IEA Heat Pump Programme

Annex 33

Compact Heat Exchangers in Heat Pumping Equipment

Final Report

Operating Agent: United Kingdom

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Preface
This project was carried out within the Heat Pump Programme, HPP which is an Implementing agreement within the International Energy Agency, IEA.

The IEA
The IEA was established in 1974 within the framework of the Organization for Economic Cooperation and Development (OECD) to implement an International Energy Programme. A basic aim of the IEA is to foster cooperation among the IEA participating countries to increase energy security through energy conservation, development of alternative energy sources, new energy technology and research and development (R&D). This is achieved, in part, through a programme of energy technology and R&D collaboration, currently within the framework of over 40 Implementing Agreements.

The IEA Heat Pump Programme
The Implementing Agreement for a Programme of Research, Development, Demonstration and Promotion of Heat Pumping Technologies (IA) forms the legal basis for the IEA Heat Pump Programme. Signatories of the IA are either governments or organizations designated by their respective governments to conduct programmes in the field of energy conservation.

Under the IA collaborative tasks or “Annexes” in the field of heat pumps are undertaken. These tasks are conducted on a cost-sharing and/or task-sharing basis by the participating countries. An Annex is in general coordinated by one country which acts as the Operating Agent (manager). Annexes have specific topics and work plans and operate for a specified period, usually several years. The objectives vary from information exchange to the development and implementation of technology. This report presents the results of one Annex. The Programme is governed by an Executive Committee, which monitors existing projects and identifies new areas where collaborative effort may be beneficial.

The IEA Heat Pump Centre
A central role within the IEA Heat Pump Programme is played by the IEA Heat Pump Centre (HPC). Consistent with the overall objective of the IA the HPC seeks to advance and disseminate knowledge about heat pumps, and promote their use wherever appropriate. Activities of the HPC include the production of a quarterly newsletter and the webpage, the organization of workshops, an inquiry service and a promotion programme. The HPC also publishes selected results from other Annexes, and this publication is one result of this activity.

For further information about the IEA Heat Pump Programme and for inquiries on heat pump issues in general contact the IEA Heat Pump Centre at the following address:

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[1] Note that the scientific contributions of Sweden and the USA are in the main text, with the latter providing full data on the HPC web site, see references.

1. Summary

This Report presents the outcomes of Annex 33, which was concerned with compact heat exchangers (CHEs) in heat pumping equipment.

The objective of this Annex was to present a compilation of possible options for compact heat exchangers, used as evaporators, condensers and in other roles in heat pumping equipment. The aim is to minimise the direct and indirect effect on the local and global environment due to operation of, and ultimate disposal of, the equipment.

The Annex involved five countries – Austria, Japan, Sweden, the United States and the United Kingdom, the latter acting additionally as Operating Agent. The Annex ran for three years, the final Annex meeting being held in the UK in September 2009.

The outcomes of the Annex consist of a wide variety of data ranging from fundamental research on boiling in narrow channels to guidelines for selecting and using CHEs in heat pumping systems. There are considerable market data available within the Report and the cited references, and a number of novel heat exchanger concepts, including the use of new materials and the application of process intensification methods, should allow equipment manufacturers in the future to achieve the Annex aim, as summarised above.

The scope has not been limited to heat pumps in buildings – the input from the UK has been particularly focussed on industrial heat pumps, where the potential for energy savings, assuming capital costs can be made attractive, is high.

2. Introduction

The objective of this Annex was to present a compilation of possible options for compact heat exchangers, used as evaporators, condensers and in other roles in heat pumping equipment. The aim is to minimise the direct and indirect effect on the local and global environment due to operation of, and ultimate disposal of, the equipment. Further data on the overall Annex activities are given later in the introductory Section, but this publication relates specifically to design data for compact heat exchangers used in heat pumping systems at the process/large commercial level.

2.1 Background

During the last two decades there have been substantial developments affecting equipment used within the refrigeration and heat pump industries, brought about largely due to environmental concerns, principally related to vapour compression cycle systems: Firstly, the realisation of the detrimental effect of chlorinated hydrocarbons on the ozone layer led to a quick phase out of these fluids, a process which is now complete in many parts of the world. Secondly, as more focus and concern has been directed towards the issue of global warming, the global warming potential (GWP) of many of the commonly used working fluids, in particular hydrofluorocarbons (HFCs), has been topping the agenda. Thirdly, and discussed later, the materials content of heat exchangers in fan-coil units (and in other heating/cooling units) are regarded rather negatively from the points of view of environmental impact and life cycle assessment[2].

Due to these perceived negative effects on the global environment, parts of the industry as well as parts of the research community and some governmental institutions, in particular in Europe, have suggested the use of natural fluids, meaning primarily ammonia, hydrocarbons and carbon dioxide, as working fluids. All these fluids are suitable from a technical point of view, although each is not necessarily ideal for all applications. However, they all have drawbacks, relating either to flammability or toxicity, or operation at high pressure. If used in public areas, the quantities of working fluid in the systems should therefore be kept as low as possible.

Having the global and local environment in mind, it is clear that future refrigeration and heat pump equipment should have as small as possible internal volume. This conclusion is equally true independently of the refrigerant chosen in the system: With HFCs and other high GWP-fluids a low charge will reduce the total leakage, thus reducing the influence on the global warming. With ammonia, hydrocarbons and carbon dioxide, a low charge will reduce the risks of accidents in case of leaks.

An additional parameter which must be taken into account is the indirect influence on the global warming from any type of refrigeration or heat pump unit. As long as we manage to keep our systems tight these secondary effects caused by the CO_2 emissions connected to electricity production are far larger than the direct effects caused by the leakage of refrigerant. A reduction of charge must thus not be accomplished at the cost of reduced energy efficiency of the system.

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[2] See research of Matajaz Prek at the University of Ljubljana, for example, reproduced in *Energy and Buildings*, 36 (2004) 1021-1027.

Looking at the distribution of the charge of working fluid within a heat pumping system it is quite clear that the dominating part is located either in the evaporator or in the condenser for vapour compression cycle and mechanical vapour recompression cycle units and, where absorption cycle systems may be considered, additionally in the generator and absorber. Reducing the charge is therefore mainly a matter of redesigning the heat exchangers. (A possible exception to this rule is the case of multisplit, direct expansion systems where a large amount of liquid working fluid is circulated through long tubing systems. For this type of system, a first step, it is suggested, should be to redesign to an indirect system, using a secondary fluid to distribute the heating/cooling capacity). To reduce the volume on the working fluid side without decreasing the energy efficiency of the system may seem a difficult task. The heat transfer areas on the two sides of the heat exchanger should preferably not be decreased, as this may increase the temperature difference between the fluids and thus reduce the efficiency.

The obvious solution to this equation is to decrease the channel area cross section, i.e. the hydraulic diameter. Fortunately, a decreased diameter may offer possibilities of increasing the heat transfer coefficients. In single phase flow it can easily be shown that in most instances, for a fixed temperature difference, a fixed pressure drop and given heat- and mass flows, the heat transfer coefficient will increase and the necessary heat transfer surface area will decrease with decreasing tube diameter. For two-phase flow, condensation and evaporation, the analysis is not as simple partly because there are no reliable correlations available for predicting heat transfer and pressure drop for small-diameter channels. However, there are increasing indications that in two-phase flow decreased channel diameter will lead to increased heat transfer and thus the possibilities of increasing the system performance.

For certain applications small diameter channel heat exchangers have already been implemented, for other reasons than the decrease of working fluid inventory. One area is for cooling of electronics, where active cooling of individual components by fluid channels incorporated into the structure has been discussed for long. As an alternative, cold-plates with mini-channels or mini-channel heat pipes are used for cooling certain types of components. A second area is automotive air conditioning, where extruded multichannel aluminium tubes have been used for several years, primarily as condensers[3]. There are also some manufacturers who have specific methods of manufacturing heat exchangers with small hydraulic diameters. These producers may target specific applications, such as off-shore gas processing or chemical process intensification (where reduced inventories are a selling point), or they may rely on having customers in different areas, all having specific requirements concerning the heat transfer, which may be met by the compact designs.

Developments in the area of compact heat exchangers (CHEs) are partly driven by demands from industry, e.g. the electronic industry and the chemicals sector. A second reason for the development is the progress in the area of materials science: It is now possible to manufacture small size objects with high precision in large quantities at low cost. We believe that these possibilities have not yet been totally explored for the benefit of the heat pump and refrigeration industry.

[3] The development of micro-channel finned heat exchangers for residential split air conditioners was described recently by Salvatori Macri of Carrier S.p.A. at the AiCARR Meeting in Milan in March 2006. Charge reduction was highlighted as one major benefit.

Highly relevant to all refrigeration and heat pump systems is the cost of disposal. It is believed that disposal/recycling will be facilitated by employing CHEs in the whole range of heat pump equipment, regardless of cycle type, as the quantities of metals to be disposed of, as well as the fluid(s) within the systems, should be reduced. The energy use in manufacture should also be reduced, reducing the overall environmental impact.

2.2 Goals of Annex 33

The principal goal of the Annex was to identify compact heat exchangers[4], either existing or under development, that may be applied in heat pumping equipment – including those using vapour compression, mechanical vapour recompression and absorption cycles. This has the aims of decreasing the working fluid inventory, minimising the environmental impact of system manufacture and disposal, and/or increasing the system performance during the equipment life, thereby reducing the possible direct and indirect effects of the systems on the global and local environments.

A second goal was to identify, where necessary propose, and document reasonably accurate methods of predicting heat transfer, pressure drop and void fractions in these types of heat exchangers, thereby promoting or simplifying their commercial use by heat pump manufacturers. Integral with these activities was an examination of manifolding/flow distribution in compact/micro-heat exchangers, in particular in evaporators.

A third goal was to present listings of operating limits etc. for the different types of compact heat exchangers, e.g. maximum pressures, maximum temperatures, material compatibility, minimum diameters, etc. and of estimated manufacturing costs or possible market prices in large scale production. It is intended within this context that opportunities for technology transfer from sectors where mass-produced CHEs are used (e.g. automotive) will be examined and recommendations made. Much of these data are presented within this document.

[4] A selection of CHEs and their applications may be found in an IEA-CADDET Publication: Learning from Experiences with Compact Heat Exchangers. CADDET Analyses Series No. 25, June, 1999.

3. Country Reports

3.1 Austria

3.1.1 Participants Details

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3.1.2 Projects Offered

3.1.2.1 Heat Pump Testing at the AIT

Principal Investigator: Austrian Institute of Technology (AIT)

Summary: The Austrian Institute of Technology (AIT) is performing heat pumps tests for more than ten years now. Within this period, over 130 heat pumps have been successfully tested. The tested samples include different heat pump types like brine/water, water/water and direct expansion heat pumps as well as different refrigerant types. The analysis presented in this report is primarily based on results obtained from this heat pump test activity, which thus represents the main contribution to this project. As all of the devices tested include plate heat exchangers, which represent the state-of-the-art technology of compact heat exchangers, the obtained analysis of this data represents the current technological status of heat exchangers used in heat pumps and may be used as a benchmark for new compact

3.1.3: Market research

3.1.3.1: Introduction

In the following, an overview of the Austrian heat pump market is given. The information includes the most important heat source systems, the development of the heat pump market together with the systems in operation, production and export ratios, the state of the art as well as future technological trends and the specifics of the Austrian heat pump market. This information illustrates the general environment in which the Austrian heat pump industry operates. Future research and development in the field of compact heat exchangers in Austria will be related to this industry. The products of heat exchanger manufacturers based in Austria are analysed regarding their types produced and industries supplied.

3.1.3.2: Heat pump technologies in Austria

3.1.3.2.1: Brine/water systems (indirect systems)

Brine/water heat pumps are mainly used in combination with horizontal collectors, which are efficient, reliable and long lasting. The collector pipes are buried under the frost layer of the ground. The brine, a mixture of water and an anti–freeze, is pumped through the plastic pipes of the collector in order to capture the energy stored in the underground. There are many different configurations of the pipes and the collector. Figure 1 shows the functional scheme of a brine/water heat pump.

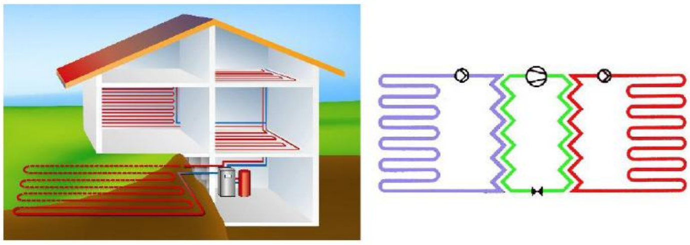


Figure 1: Functional scheme of a brine/water system with a single loop horizontal heat exchanger; (Böswarth et al., 2006)

Vertical collectors are becoming more and more important, as excavations of the gardens and frost heaves can be avoided. (Böswarth et al., 2006).

3.1.3.2.2: Direct expansion systems

In direct expansion heat pumps the system's refrigerant circulates as heat transfer medium within the collector, where it evaporates. Therefore the heat exchanger between the refrigerant circle and a brine circle, as well as the circulation pump, can be eliminated. These systems use horizontal (ground) collectors, which are also called evaporation loops. Figure 2 shows an example of a direct expansion heat pump and the principle of a direct exchange heat pump.

a) b)

Figure 2: a) Example of a direct expansion heat pump; b) Principle of a direct exchange heat pump; (Böswarth et al., 2006)

The collector pipes used are copper pipes with a polyethylene casing (Böswarth et al., 2006). Direct expansion heat pump systems are a particular feature of the Austrian heat pump market.

3.1.3.2.3: Water/water systems

The principle of water/water systems, which is shown in Figure 3, is to pump ground water through the evaporator or an intermediate heat exchanger in the brine circuit to the evaporator. The ground water is extracted from a borehole, pumped through the heat pump, and then re-injected into the ground via a second borehole. The advantage is that due to the higher temperature of the ground water, a higher coefficient of performance (COP) can be reached (Böswarth et al., 2006).

Figure 3: Principle of the water/water heat pump system; (Böswarth et al., 2006)

In Central Europe the average groundwater-temperature, which varies just slightly in a depth of 10m or more, is about 10°C. Instead of ground water basically also surface water of the sea can be used as heat source. (Böswarth et al., 2006)

3.1.3.2.4: Air/water and air/air systems

Air/water and air/air heat pumps use indoor and outdoor air as heat sources. When the outdoor temperature falls, the heat load for the building simultaneously increases, while the heating capacity and efficiency of the heat pump decreases. Indoor compact devices include all the heat pump components in one casing. The necessary airflow is produced by centrifugal fans.

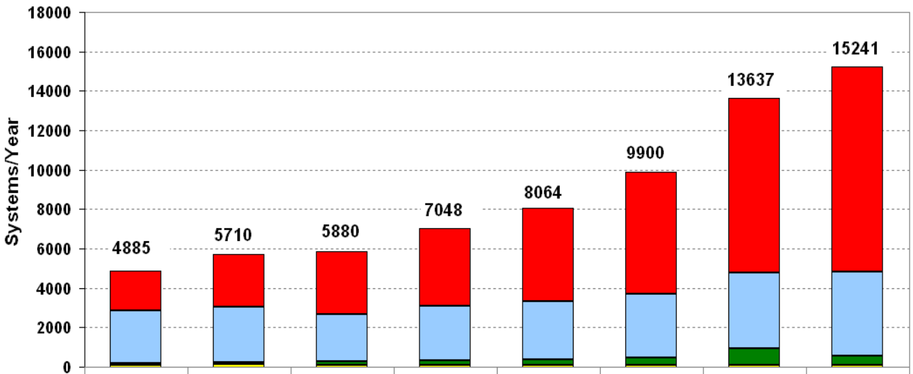
3.1.3.3: Market Development

The Austrian heat pump market has a long lasting track record with the first heat pump systems introduced in 1976, at the beginning of the first energy crisis. The technology experienced high yearly growth rates up to 1987, in both, provision of domestic hot water and space heating. From 1987 till 2000, the sales of heat pumps developed anti-cyclically to the development of energy prices.
Due to the introduction of energy policy guidelines concerning environmental protection and therefore promotion of energy efficiency and regenerative energy[5], and substantial subsidies for the housing segment, Austria experienced steady growth rates in the segment of domestic heat pumps from the year 2000 onwards, when energy

[5]Energy-efficient driven heat pumps are currently categorised as sustainable heating systems and are part of the climate initiative of the Austrian government.

prices rose again. This expansion was mainly triggered by the growing demand of heat pumps for space heating, as shown in Figure 4.

The Austrian market can be described as quite volatile concerning the type of heat pumps preferred. While direct expansion systems were the most popular type in the 90s, brine/water heat pumps have been favoured since then, reaching a market share of 45% in 2008. Second in place in terms of sales figures are air/water heat pumps experiencing an increase in sales of 92.3% from 2007 to 2008, thus gaining a market share of 30,9%. As shown in Figure 5, air/air and direct expansion systems are currently the least favoured heat pump systems in Austria.



	2000	2001	2002	2003	2004	2005	2006	2007
Space heating	2025	2660	3200	3953	4748	6193	8853	10398
Domestic hot water	2690	2810	2420	2761	2962	3253	3863	4264
Ventilation	80	120	160	221	258	349	842	486
Dehumidification	90	120	100	113	96	105	79	93

Figure 4: Development of the domestic heat pump market in Austria from 2000 to 2007; Year 2006 (Faninger, 2007), Year 2000 – 2007 excluding 2006 (Biermayr et al., 2008)

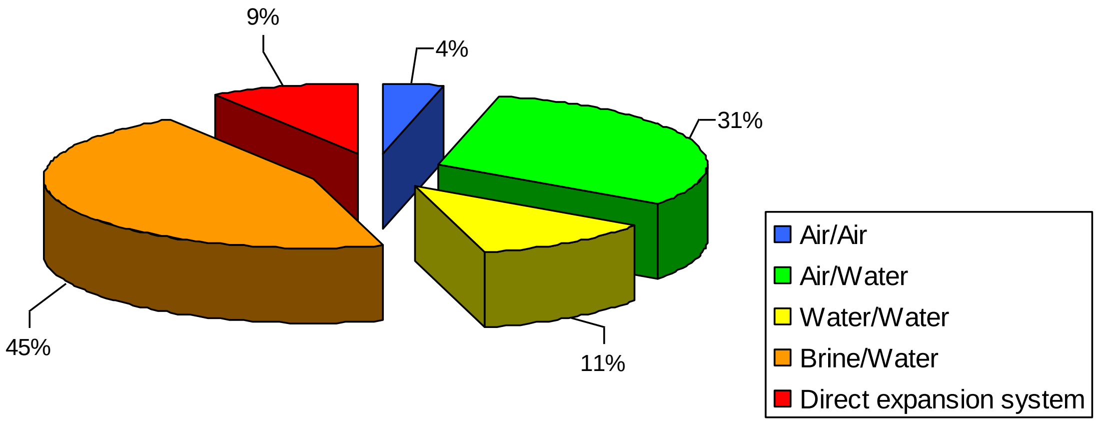


Figure 5: Types of heat pumps sold in 2008 in % of total sales, (Biermayr et al., 2009)

3.1.3.3.1: Heat pumps in operation

In 2008, about 156,500 heat pumps were in operation in Austria. They are used for the following applications (Biermayr et al., 2009):

- 82,500 for domestic hot water production
- 68,800 for space heating
- 2,900 for air ventilation and
- 2,300 for dehumidification of swimming pools.

Although the figures of new installations are still high, the Austrian market for heat pump technology experiences currently a temporary decrease in total number of installations. This development can be best explained using sales figures from the past. Most heat pumps sold in the 1980s and early 1990s were used for hot water production. Assuming a lifespan of 20 years, these systems gradually have been put out of operation from 2000 onwards, leading to a decrease of heat pumps in operation for domestic hot water production.

3.1.3.3.2: Production, import and export

In 2007 about one third out of 12.778 heat pumps units produced in Austria were exported; in 2008 the export quota reached 37%. The increase of about 5% was triggered mainly by a rising foreign demand for Austrian heat pumps for space heating.
The favourite export markets are (Biermayr et al., 2009):

- Germany
- Switzerland
- Italy
- Others: Ireland, Netherlands, Hungary, Slovenia, Czech Republic, Poland, Croatia, Russia, Greece, Romania, Liechtenstein (Biermayr et al., 2008)

Austria is importing heat pumps or heat pump components from the following countries, ranked according to their importance for the Austrian heat pump industry (Biermayr et al., 2009):

- Germany
- Sweden
- Italy
- Others: Netherlands, France, Belgium, Switzerland, Denmark, USA

3.1.3.4: State of the art

Heat pumps for heating purposes can be classified as follows:

- Air/water-split systems
- Air/water-compact systems
- External air-heat pumps with air preheating in the soil
- Brine/water–borehole heat exchangers
- Brine/water and direct expansion with horizontal ground heat exchangers
- Water/water–systems

Data of the heat sources for heat pumps for space heating have been collected since 1989. Since then, brine/water heat pumps have increased their market share continuously, while air/water heat pumps have gained importance only during the last years. The rise in sales is partly caused by the growing number of refurbished buildings, where air is often the only useable heat transfer medium (Biermayr et al., 2008).

Figure 6 depicts the development of heat source systems of heat pumps for space heating from 1993 onwards.

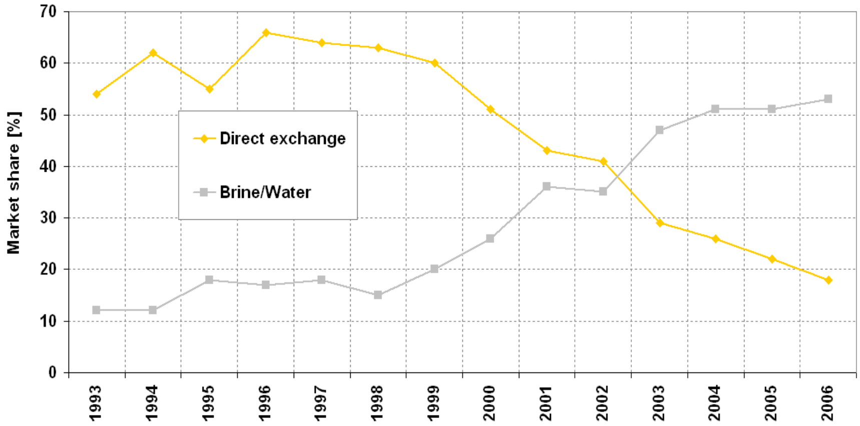


Figure 6 : Market share of direct expansion heat pumps and brine/water heat pumps regarding to heat pumps for space heating; (Faninger and Biermayr, 2007)

Evidently, direct expansion heat pumps lost importance from 1996 on, whereas brine water systems were getting more popular. This development was triggered by the spreading of vertical collectors and the spectrum of systems offered. (Biermayr et al, 2008)

Another important factor influencing the type of heat pump to use is the seasonal performance factor, in short SPF. Figure 7 shows the correlation of the SPF of various heat source systems at different levels of design temperature.

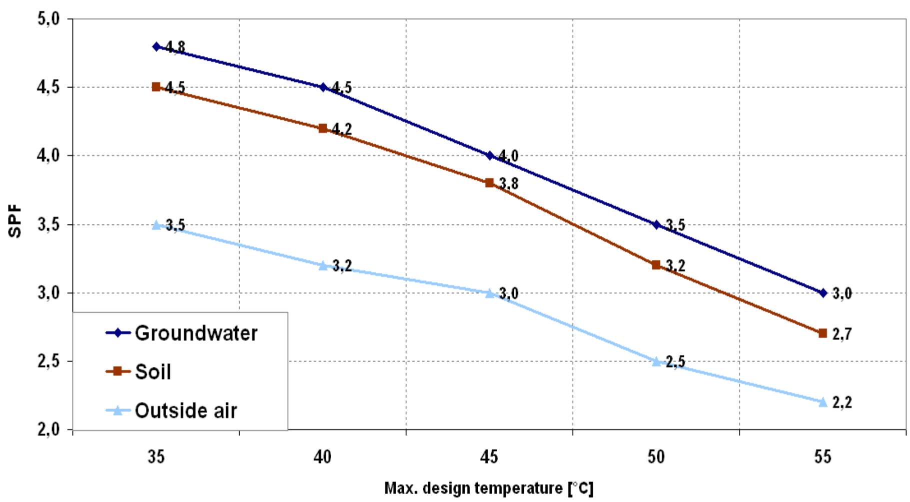


Figure 7: Heat pumps for space heating in Austria; SPF dependence on heat source and max. design temperature of the heating system; (Faninger, 2007)

Obviously, heat pumps using groundwater as heat source, achieve the highest SPF followed by heat pumps deploying soil or outside air. All systems reach their highest SPF at low design temperatures of about 35°C. An increase in design temperature leads to constant decreases of SPFs within all systems.

As shown in Figure 8, direct expansion heat pumps achieved the best SPFs over the years. The second best performing system were water/water heat pumps, which seem to close the gap towards direct exchange ground coupled water heat pumps. The lowest SPFs were reached by air/water heat pumps.

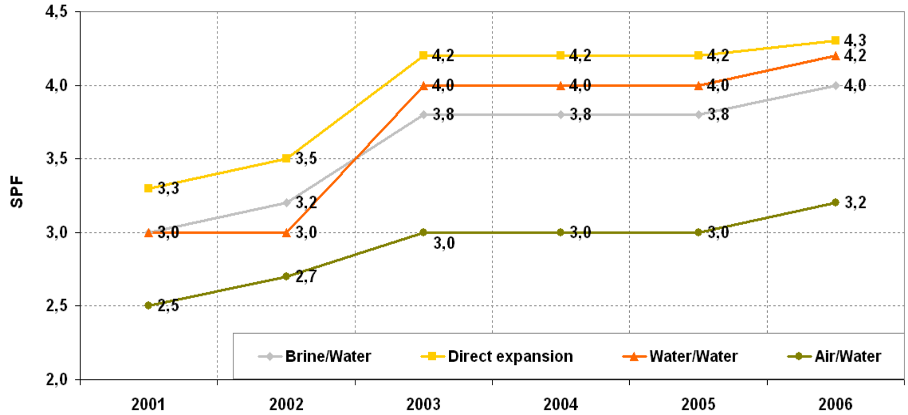


Figure 8: Development of SPFs of heat source systems from 2001 to 2006; (Faninger, 2007)

3.1.3.5: Future trends

3.1.3.5.1: Technology

As mentioned above, the Austrian domestic market is currently focusing on the use of heat pumps for space heating, domestic hot water production and combined systems. The most common system in place is the compression heat pump using ground heat exchangers, borehole heat exchanger, ground water wells or air heat exchangers as heat sources. The systems are mainly driven by electrical power.
It is expected, that the sales of hybrid systems providing for heating and cooling will increase, as the demand for air conditioning is rising steadily. Another development showing already, is the continuous increase of sales of heat pumps using air as heat source. These systems meet the requirements of the retrofitting building industry, which is expected to be the next market segment to penetrate, best.
Furthermore it is most likely that natural gas and other energetically useable gases will be more important as driving energy for both, systems with large capacity but also for capacities around 10 kW (Biermayr et al., 2008). This will stimulate the market penetration of sorption head pumps, which could also be used in thermal cooling application.

3.1.3.6: Austrian heat pump market compared to other European heat pump markets

As shown in Fig. 9, Sweden is the country with most ground sourced heat pumps (GSHP) per capita, followed by Finland, Switzerland and Austria. If the existing

GSHP are calculated per area, the ranking looks slightly different: Switzerland is taking the lead, in front of Sweden, Austria and Germany.

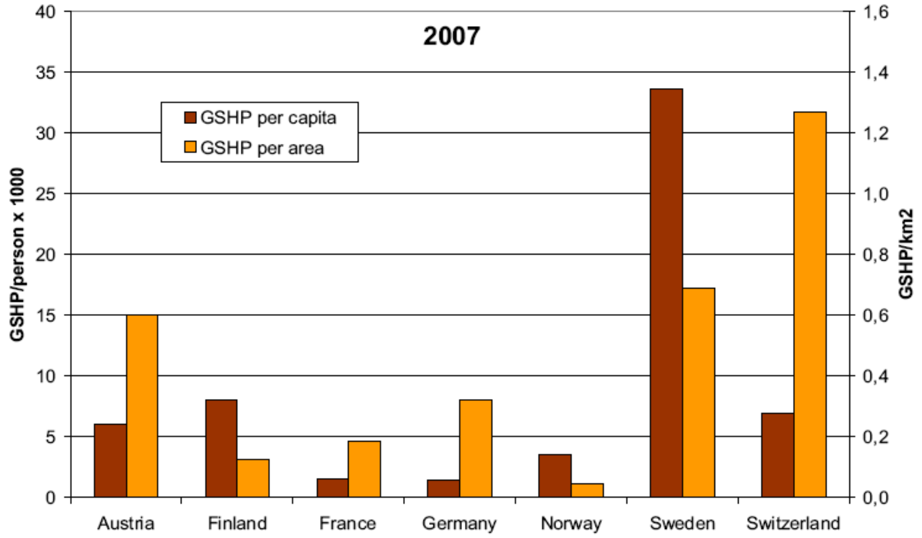


Figure 9: Total number of existing ground sourced heat pumps (GSHP) for the year 2007, per capita and per area; (Sanner, 2008)

EHPA heat pump statistics reveal that direct expansion or direct condensation heat pumps exist in three European countries: Austria, France and Germany. The importance, expressed in terms of market penetration, differs in the three nations. Whereas Austria (17.8%) and France (14.5%) show quite high ratios of market penetration, the importance in Germany (2.7%) may be qualified as rather low. Figure 10 displays the European Heat Pump Association (EHPA) statistics of 2006.

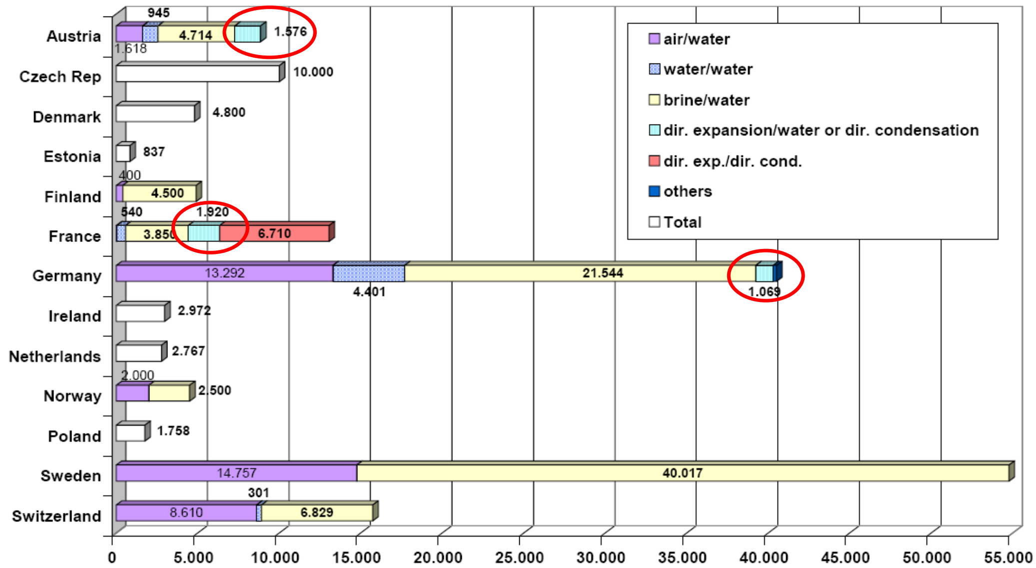


Figure 10: EHPA heat pump statistics 2006: sales figures heating only (excluding exhausted air heat pumps); (EHPA, 2007)

3.1.3.7: Market overview of Austrians heat exchanger manufacturers

3.1.3.7.1: Survey of Austrian heat exchanger manufacturers

The following data, which represents most[6] Austrian manufacturers of heat exchangers, was gathered through telephone interviews and internet research from November to December 2008 by AIT. The products of multinational companies operating in Austria are included only so far, as they are relevant for the domestic market.
The heat exchanger producing companies in Austria manufacture mainly tube heat exchangers focusing on shell & tube heat exchangers, which cover about one third of the market, as displayed in the following Figures 11 and 12. The figures visualise, how many of the twenty companies that were interviewed produce certain heat exchanger types.

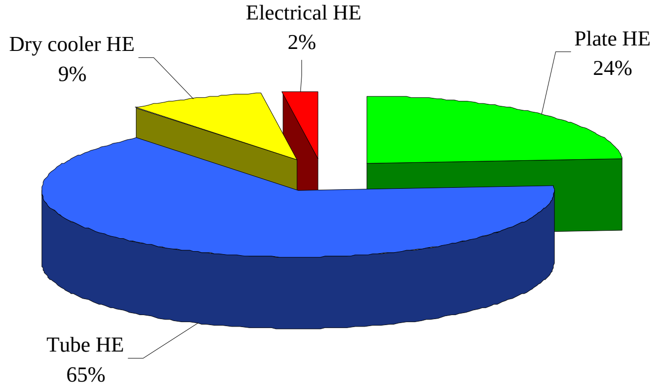


Figure 11: Overview of types of heat exchangers (HE) produced by Austrian companies (AIT, 2008)

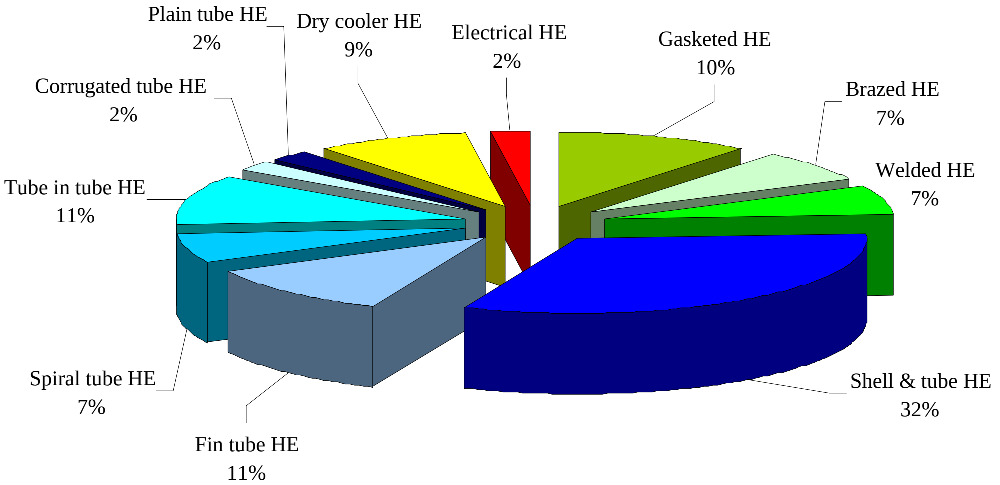


Figure 12: Types of heat exchangers (HE) produced by Austrian companies in detail (AIT, 2008)

Table 1 displays a matrix of the Austrian heat exchanger manufacturers and their types of heat exchangers produced, including the categories: plate heat exchangers, tube heat exchangers, dry coolers and electrical heat exchangers.

[6] Some companies were not willing to participate in the survey.

Table 1: Overview of heat exchangers supplied by Austrian companies

Manufacturer	Heat exchanger types: Plate heat exchanger			Tube heat exchangers							
	Gasketed plate heat exchangers	Brazed plate heat exchangers	Welded plate heat exchangers	Shell-and-tube heat exchanger	Fin tube heat exchangers	Spiral tube heat exchangers	Tube-in-tube heat exchanger	Corrugated tube heat-exchanger (Drallrohr - WT)	Plain tube heat exchanger (Glattrohrwärmetauscher)	Dry cooler (fin tube heat exchanger)	Electrical heat exchanger (directly heated gas preheater)
UNEX Heatexchanger Engineering GmbH,	X	X	X			X	X	X		X	
APL Apparatebau GmbH			X	X	X						
Enco Energie Componenten Ges.m.b.H.				X	X		X				
INNOWELD Metallverarbeitung GmbH				X	X	X	X				
Friedrich Pölzl				X							
Energie- und Umwelttechnik Ges.m.b.H. - EUT	X	X									
Fischer Maschinen- und Apparatebau AG	X										
IAF Industrieanlagentechnik Frauental GmbH				X							
ECOTHERM Austria (former Bremstaller GmbH & Co KG)				X	X	X					
GIG Karasek GmbH			X	X							
Holger Andreasen & Partner Ges.m.b.H.				X							
Apparatebau-Schweißtechnik Ges.m.b.H.				X							
SIROCCO Luft- und Umwelttechnik GmbH	X	X		X			X			X	
GEA Klimatechnik Produktion GmbH					X				X	X	
Josef Bertsch Ges.m.b.H. & Co				X							
Heat wärmetechnische Anlagen Gesellschaft m.b.H.				X			X				X
Karrer Steel Components GmbH				X							
Martin Konvektoren Gesellschaft mbH & Co										X	
Schiff & Stern KG	X			X							
Olaer Austria GmbH				X						X	

As shown by Figure 13, 95% of the Austrian manufacturers interviewed are producing heat exchangers for the plant engineering and construction industry. The food and chemical industry are addressed by 80% of suppliers each; followed by the power plant and biotechnology industry, which are of importance for 75% and 70% of the producers of heat exchangers. All these sectors require rather customized heat exchangers with large capacities.

Between 40 and 55% of the manufacturers are into the production of heat exchangers for the HVAC sector, like heating, refrigeration and air conditioning. Only 3 out of 20 Austrian firms - Unex, Ecotherm Austria and Fischer Maschinenbau Apparatebau AG - are manufacturing heat exchangers for the heat pump industry. All three provide a rather diversified portfolio of heat exchangers covering at least 10 different industries including, in two cases, all other relevant HVAC sectors.

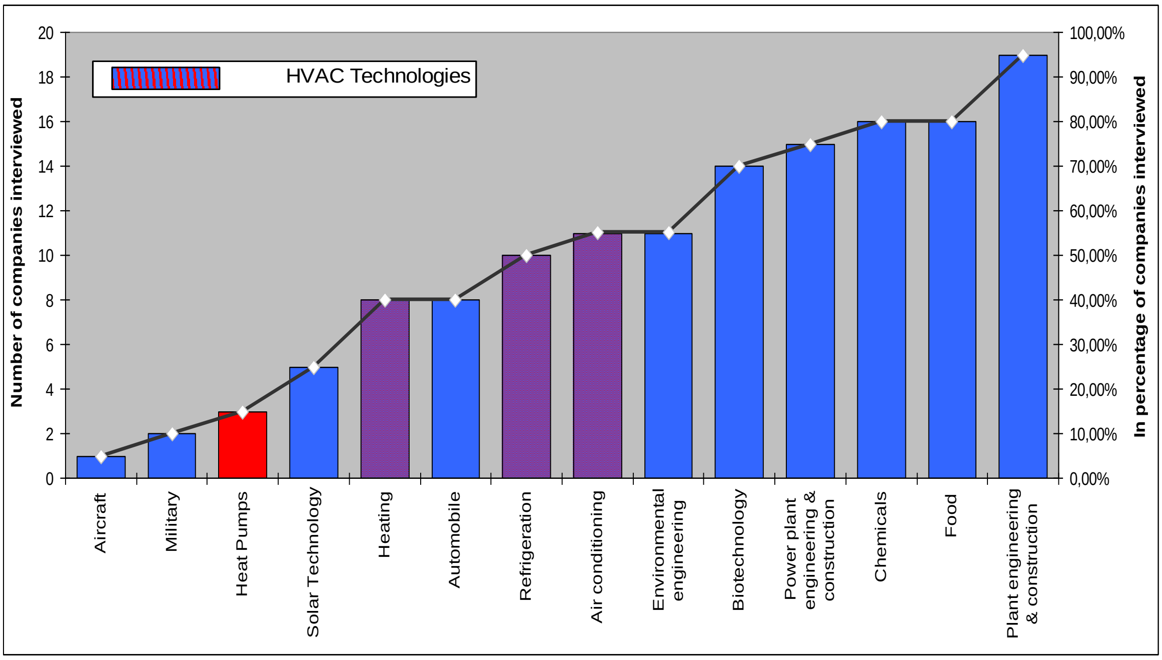


Figure 13: Overview of industries supplied by the interviewed companies with heat exchangers, AIT, 2008

Table 2 shows the Austrian heat exchanger manufacturers and their supplied sectors of industry as a matrix.

Table 2: Overview of industries supplied by Austrian heat exchange manufacturers

Manufacturer	Supplied sectors of industry													
	Heat pumps	Solar technilogy	Heating	Air conditioning	Refrigeration	Plant engineering and construction	Power plant engineering and construction	Biotechnology	Environmental engineering	Chemicals industry	Food industry	Automobile industry	Aircraft industry	Military
UNEX Heatexchanger Engineering GmbH,	X	X	X	X	X	X	X	X	X	X	X	X		
APL Apparatebau GmbH			X		X	X		X	X	X	X			X
Enco Energie Componenten Ges.m.b.H.						X	X	X	X	X	X			
INNOWELD Metallverarbeitung GmbH		X	X	X	X	X	X	X	X	X	X	X		
Friedrich Pölzl						X					X			
Energie- und Umwelttechnik Ges.m.b.H. - EUT		X	X	X	X	X		X	X	X	X	X		
Fischer Maschinen- und Apparatebau AG	X		X	X	X	X	X	X		X	X	X	X	
IAF Industrieanlagentechnik Frauental GmbH				X		X	X	X		X	X			
ECOTHERM Austria (former Bremstaller GmbH & Co KG)	X	X	X	X	X	X	X	X	X	X	X	X		
GIG Karasek GmbH						X	X	X	X	X	X			
Holger Andreasen & Partner Ges.m.b.H.		X			X	X	X	X		X	X			
Apparatebau-Schweißtechnik Ges.m.b.H.						X				X				
SIROCCO Luft- und Umwelttechnik GmbH			X	X		X	X		X	X	X			X
GEA Klimatechnik Produktion GmbH				X	X	X	X	X	X	X	X	X		
Josef Bertsch Ges.m.b.H. & Co							X							
Heat wärmetechnische Anlagen Gesellschaft m.b.H.			X		X	X			X					
Karrer Steel Components GmbH						X	X	X		X				
Martin Konvektoren Gesellschaft mbH & Co				X		X	X				X			
Schiff & Stern KG			X	X	X	X	X	X		X	X	X		
Olaer Austria GmbH				X		X	X	X	X	X	X	X		

3.1.3.7.2: Investigation of heat exchangers in tested heat pumps

For this investigation, certification test data gathered from 134 heat pump tests is analysed. As shown in Figure 14, a total number of 19 heat pump manufacturers from 9 European countries had their heat pumps tested at AIT between the years 2000 and 2008.

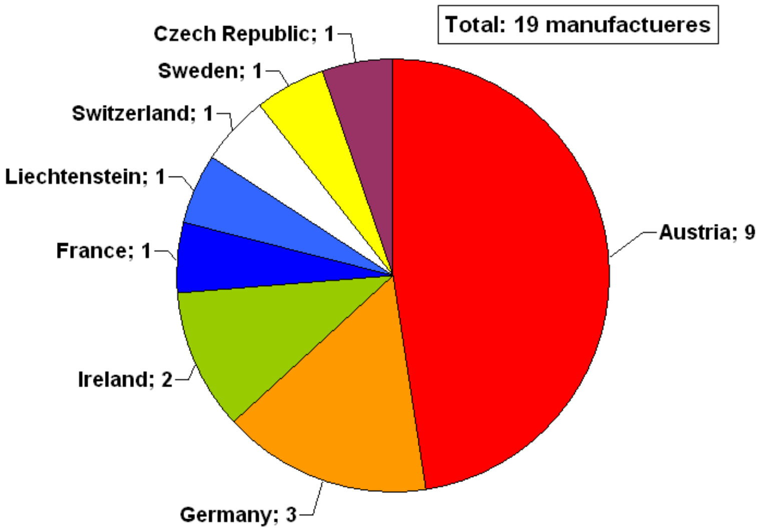


Figure 14: Number and nationality of heat pump manufacturers which tested heat pumps at AIT; (AIT, 2008 b)

The objects of analysis are heat exchangers used in water-, direct expansion- and water/water heat pumps. Table 3 shows the variety of materials used for evaporators and condensers for the different types of heat exchangers tested.

Table 3: Heat exchanger types and materials for evaporators and condensers of tested heat pumps (AIT, 2008 b)

Heat exchanger	Evaporator		Condenser
	Plate heat exchangers	Collector line	Plate heat exchangers
Material	Stainless steel	Stainless steel	Stainless steel
		Copper	Copper

As far as direct expansion heat pumps are concerned, collector lines of steel or copper are used as evaporators, whereas the brine/water and water/water pumps tested are equipped with plate heat exchangers made of stainless steel. All heat pumps use plate heat exchangers as condensers either made of stainless steel or copper.
The heat exchangers of the heat pumps manufactured in Austrian originate either in Sweden or Germany. This information refers to the heat pump tests where the heat exchanger manufacturer is documented (AIT, 2008 b).

3.1.4: The evaluation of the performance of compact heat exchangers relevant to heat pumps

3.1.4.1: Heat pump data base and analysis of data

Since 2000, AIT has tested 134 different heat pumps on in-house test facilities. So far, the obtained data was not stored in a single database and thus not accessible for further investigation. Within the scope of this project, the data of these tests were collected and transferred to a single database which can now be utilized for further investigations. A first basic thermodynamic analysis of the data is presented in this report.

All of the heat pumps tested include plate heat exchangers which represent the state-of-the-art in compact heat exchangers. Results obtained from analysis of this database are thus a useful basis (or benchmark) for comparison with new technologies. Table 4 is a shortened list of the heat pumps tested including types, model and customer names; COPs at a standard test point and dates and so on. Pump types and company names have been removed from the table.

Table 4: Shortened heat pump list

Nr.	System	Pumpentyp	Firma	?T	COP	Ordner	Pr.-Nr.	Datum
1	B0/W35			5	4.20			25/08/2000
2	W10/W35			5	5.33			25/08/2000
3	E4/W35			5	4.43			25/08/2000
4	B0/W35			10	4.06			25/08/2000
5	W10/W35			10	4.88			25/08/2000
6	E4/W35			5	4.26			25/08/2000
7	E4/W35			10	5.00			19/02/2001
8	E4/W35			10	4.40			19/02/2001
9	E4/W35			5	4.25			08/03/2001
10	E4/W35			5	4.54			08/03/2001
11	E4/W35			10	4.40			03/08/2001
127	B0/W35			10	4.30			03/06/2008
128	B0/W35			5	4.20			03/06/2008
129	B0/W35			10	4.40			03/06/2008
130	W10/W35			5	5.20			03/06/2008
131	W10/W35			10	5.50			03/06/2008
132	W10/W35			5	5.30			03/06/2008
133	W10/W35			10	5.60			03/06/2008
134	E4/W35			5	3.89			24/07/2008

Each heat pump was tested with different combination of source and sink temperatures. For a full test, a heat pump is tested in total eight different temperature conditions as shown in Table 5 for example.

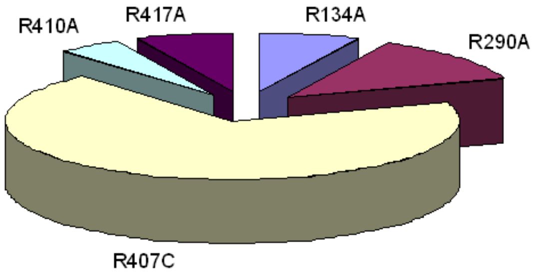
Table 5: Example of a full test

	mittlere Heizleistung [kW]	mittlere Leistungsaufnahme [kW]	**Leistungszahl**	Unsicherheit Heizleistung [± kW]
B0/W35 *	**14.456**	**3.622**	**4.0**	**0.219**
B5/W35	16.447	3.789	**4.3**	0.222
B-5/W35	12.218	3.490	**3.5**	0.215
B0/W45	12.896	4.142	**3.1**	0.216
B5/W45	14.994	4.378	**3.4**	0.220
B-5/W45	10.749	3.798	**2.8**	0.213
B0/W55	11.293	4.422	**2.6**	0.213
B5/W55	13.387	4.818	**2.8**	0.217

* WNA ΔT = 5 K

The heat pump in Table 5 was a brine-to-water (BW hereunder) heat pump. Test conditions are noted in the first column on the left designating source temperature by the number behind B and sink temperature by the one behind W. For example, B-5/W45 means source is brine at -5°C and sink is water at 45°C. From the second column to the fifth, average heating power, electric power consumption, COP and the uncertainty in heating power are listed.

In total, five refrigerants were identified in the database, namely R134A, R290A, R407C, R410A, and R417A. As shown in Figure 15, absolute majority is R407C.

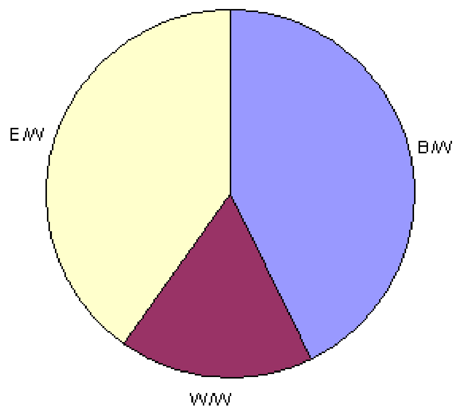


Refrigerant	No of cases
R134A	34
R290	61
R407C	324
R410A	23
R417A	33
Total No of cases	475

Figure 15: Type of refrigerant considered

Note that the figures in Figure 15 are number of test cases, not of heat pumps. One set of tests or a project contains different number of test cases since customers can choose from minimum (two-point tests) to full (eight-point) test.

The distribution of heat pump types is shown in Figure 16. BW (brine-to-water) and EW (direct expansion or ground-to-water) are almost equal and WW (water-to-water) is the smallest in number of unit. Regarding the refrigerant in different heat pump types, it is notable that no EW heat pump was based on R134a or R417A.



Refrigerant	Source type
R134A	B/W: 26 cases
	W/W: 8
R290A	B/W: 18
	E/W: 43
R407C	B/W: 140
	W/W: 59
	E/W: 125
R410A	E/W: 23
R417A	B/W: 19
	W/W: 14

Figure 16: Type of heat source considered

Figure 17 is a diagram showing all measured parameters from one case of test, which is included in the final test report. In each case, only the steady and continuous part of the measurement that satisfies the certification criteria is time-averaged and automatically recorded in such a diagram.

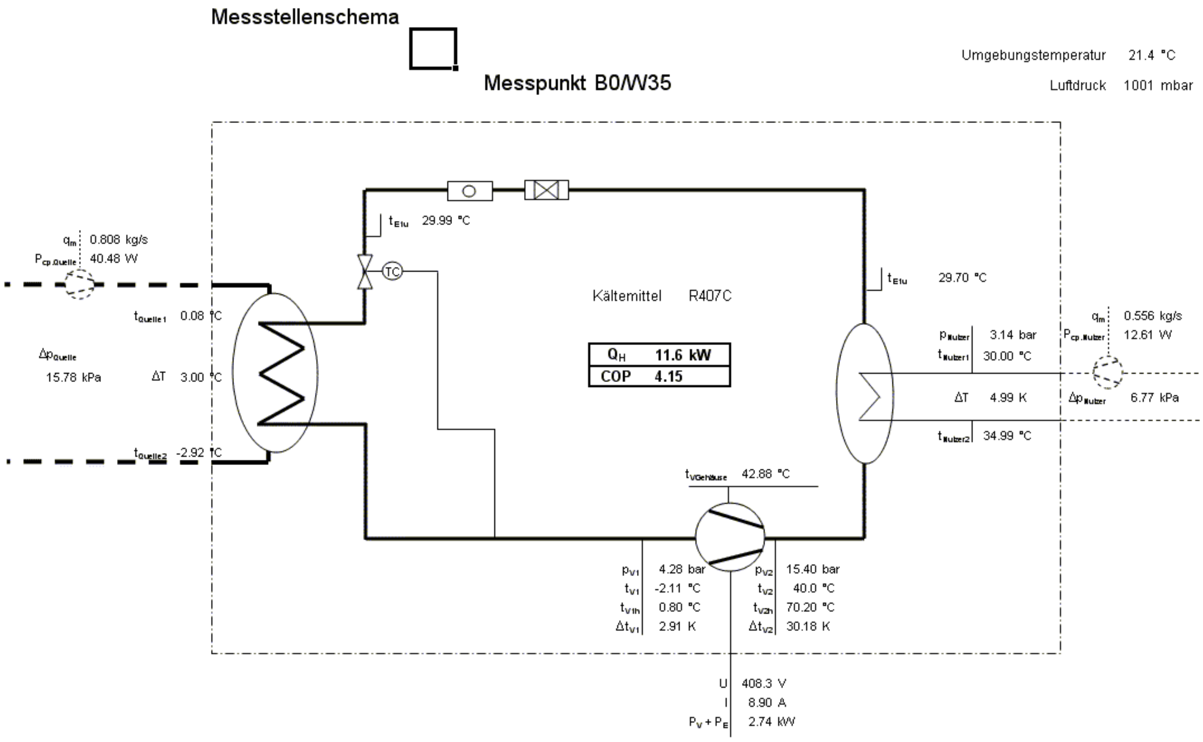


Figure 17: Typical configuration and measurement of a heat pump

The particular heat pump in Figure 17 is based on R290 (see the middle of the diagram) and equipped with an internal heat exchanger. Source is brine at 0°C and sink is water at 35°C. Heating power is measured 14.5kW with COP of 4.12. Various other parameters are also measured including primary and secondary temperatures, secondary flow rates, suction- and discharge pressures and so on.
For the analysis in this report, not all the parameters are needed. Figure 18 shows the measurement points of interest.

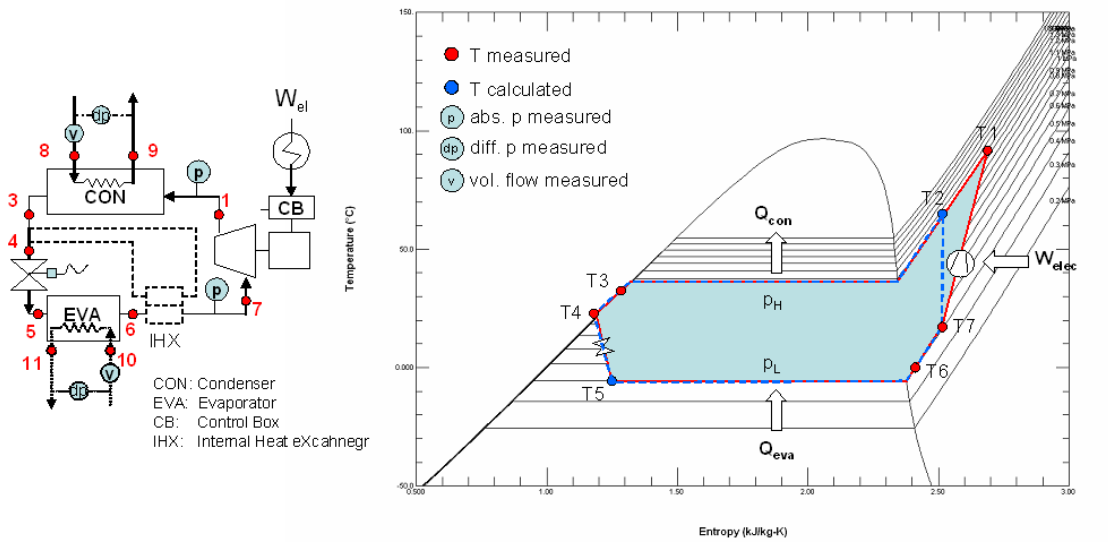


Figure 18: Schematics of a heat pump and a heat pump cycle

The analysis is largely divided into two parts, heat pump cycle and heat exchanger. In the cycle part, the thermodynamic behaviour of the heat pump is investigated and in the heat exchanger part, the influence of the heat exchanger on the cycle performance is investigated. For this purpose, focus is given to the following processes in a heat pump (refer to the figure for indices).

7→1:	Actual compression
7→2:	Isentropic compression
1→3:	Heat removal from the primary side in a condenser
3→4:	Heat removal from liquid in internal heat exchanger
4→5:	Isenthalpic expansion
5→6:	Heat addition to the primary side in evaporator
6→7:	Heat addition to the gas in internal heat exchanger
8→9:	Heat addition to secondary side in a condenser
10→11:	Heat removal from secondary side in evaporator

Thermodynamic analysis of these processes requires temperature and pressure at each point on primary side and temperature and flow rate on secondary side. However, not all the parameters are available from the database. Primary pressure is normally measured only at two points across compressor, i.e. at point 7 and 1. T5 (temperature at point 5) is not always measured and T2 is only obtained from calculation. Point 3 and 6 do not exist when there is no internal heat exchanger. When unavailable, some assumptions will be applied to provide alternatives.

Table 6 shows a list and description of the data collected from the database for the purpose of the analysis.

Table 6: Collected data from the database

	Parameters	Description
1	date	Test date
2	model name	Manufacturer + heat pump type
3	test name	Sink/Source + temperature difference on hot water side
4	comp	Model type + Model name
5	Hxcon	Manufacturer + Model type
6	HX_{eva}	Manufacturer + Model type
7	$type_{ref}$	Refrigerant name
8	cnfg	With Internal heat exchanger → 1 Without internal heat exchanger → 0
9	ref_{chrg} [Kg]	Filled quantity of the refrigerant
10	W_{el} [kW]	Total active power consumption (excl. circulation pumps)
11	Q_{con} [kW]	Heating power (User Side)
12	Q_{eva} [kW]	Cooling power (Source Side)
13	Δp_{con} [kPa]	Pressure difference heat exchanger (User Side)
14	Δp_{eva} [kPa]	Pressure difference heat exchanger (Source Side)
15	Wp_{con} [kW]	Circulation pump work user side
16	Wp_{eva} [kW]	Circulation pump work source side
17	p_1 [bar]	Pressure after compressor
18	p_3 [bar]	Pressure after condenser, if available
19	p_6 [bar]	Pressure after evaporator, if available
20	p_7 [bar]	Pressure before compressor
21	m_8 [kg/s]	Secondary mass flow rate into condenser
22	m_{10} [kg/s]	Secondary mass flow rate into evaporator, if available
23	T_1 [°C]	Temperature after compressor
24	T_3 [°C]	Temperature after condenser
25	T_4 [°C]	Temperature before expansion valve
26	T_6 [°C]	Temperature after evaporator
27	T_7 [°C]	Temperature before compressor
28	T_8 [°C]	Secondary inlet temperature condenser
29	T_9 [°C]	Secondary outlet temperature condenser
30	T_{10} [°C]	Secondary inlet temperature evaporator
31	T_{11} [°C]	Secondary outlet temperature evaporator

3.1.4.2: Method of analysis

The method of analysis is detailed in Appendix 1, Section A1.1, together with results.

3.1.4.3: Conclusions

Since 2000, 134 heat pump tests with three different types of heat pumps based on five different refrigerants have been performed at AIT. Within this project, this data was collected and organized in a database in order to become available for further analysis. The data was subsequently analyzed to characterize the thermodynamic behaviour of heat pumps and the heat exchangers therein. The analysis clearly showed dependence of the 1st and second law efficiencies on temperature lift and heat exchangers' LMTDs. From the results, the following conclusions were made:

- COP_{el} is found linearly proportional to the reciprocal of temperature lift and the linear relationship has been identified for all types of heat pump.
- COP_{el} is found higher in the order from water-to-water, brine-to-water and direct expansion heat pumps. Low COP_{el} of direct expansion heat pumps is suspected attributable to the relatively large pressure drops in collect lines.
- Among five refrigerants, R407C is found to give the highest level of COP_{el}.
- All plate heat exchangers in R407C heat pumps could be characterized in terms of LMTD within ca. 30% of uncertainty.
- LMTDs of heat exchangers are found to be directly related to the second law efficiency of heat pumps showing linear relationships.
- The influence of the evaporator could not be separated from that of condenser on the second law efficiency of the heat pumps. This task requires further study and remains as a future task.

3.1.5 Dissemination and Education

In order to inform the Austrian stakeholders of the research activities and results gained in this project, AIT held a workshop on compact heat exchangers in heat pumping on October 28th, 2009 in Vienna. This workshop was part of a public customer event at the AIT. Additionally, the database will be further developed and updated on a regular basis. It is intended to extend this analysis and publish the results in relevant scientific papers.

3.2 Japan

3.2.1 Participant Details of Japan National Team

The organizational structure of Japan National Term (JNT) is shown in Figure 3.2.1-1. JNT is operated and managed as one of technical sub-committees of Heat Pump and Thermal Storage Technology Center of Japan (HPTCJ). The activities of JNT are supported by METI and NEDO.

Members of JNT are experts from universities, institutions and industries. Members of the steering committee of JNT are listed in Table 3.2.1-1 and member companies are also listed in Table 3.2.1-2.

The directions of the activity of JNT which we set up at the start of the work are as follows:

(1) to investigate recent trend of R & D on heat exchangers for refrigeration & air-conditioning systems
(2) to analyze and report recent research progress on heat exchanger technology in Japan, such as heat exchange characteristics of alternative refrigerants, heat transfer enhancement, flow distribution, etc.
(3) to organize JNT meeting in order to collect the information about R & D on heat exchangers for refrigeration and air-conditioning applications, 4 times a year.

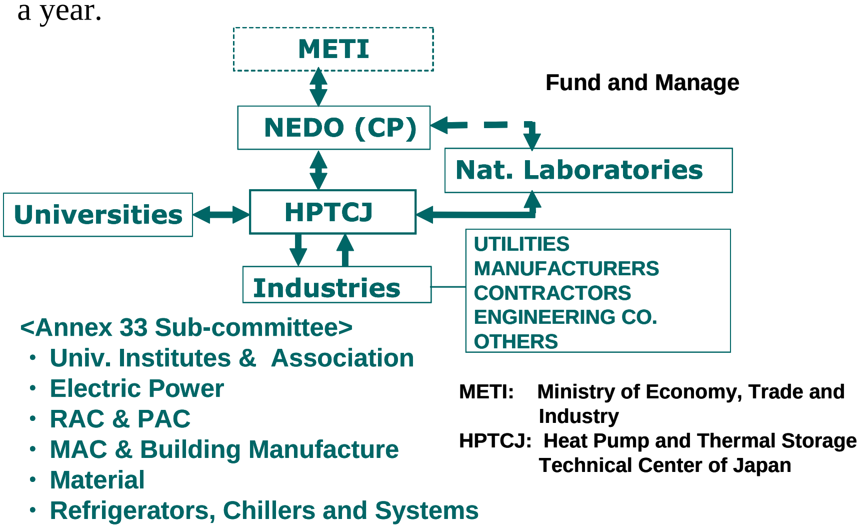


Fig. 3.2.1-1 Organizational structure of Japan national term

Table 3.2.1-1 Steering committee of JNT

Member	Institution	
Prof. Shigeru KOYAMA	Kyushu University	Chair
Prof. Akio MIYARA	Saga University	Vice-chair
Prof. Hitoshi ASANO	Kobe University	Vice-chair
Dr. Katsumi HASHIMOTO	CRIEPI	Vice-chair
Prof. Eiji HIHARA	The University of Tokyo	
Prof. Isao KATAOKA	Osaka University	
Prof. Masafumi KATSUTA	Waseda University	
Dr. Fumio TAKEMURA	AIST	
Dr. Yu SESHIMO	JRAIA	
Mr. Makoto TONO	HPTCJ	Secretary-general

CRIEPI : Central Research Institute of Electric Power Industry
AIST : National Institute of Advanced Industrial Science and Technology
JRAIA : The Japan Refrigeration and Air Conditioning Industry Association

Table 3.2.1-2 Member companies

Electric Power (3)
Chubu Electric Power Co., Inc.
The Kansai Electric Power Co., Inc.
The Tokyo Electric Power Co., Inc.
MAC &Vending Machine (5)
Calsonic Kansei Corp.
Fuji Electric Advanced Technology Co., Ltd.
Fuji Electric Retail Systems Co., Ltd.
Sanden Co.
T. RAD Co., Ltd.
Refrigerators, Chillers and Systems (5)
Ebara Refrigeration Equipment & System Co., Ltd.
Mayekawa MFG. Co, Ltd.
Mitsubishi Heavy Industries, Ltd.
Takasago Thermal Engineering Co, Ltd.
Toyo Engineering Works, Ltd.

RAC & PAC (7)
Daikin Industries Ltd.
Fujitsu General Institute of Air-Conditioning Technology Ltd.
Hitachi Appliances, Inc.
Mitsubishi Electric Co.
Panasonic Co., Ltd.
Panasonic Home Appliances, Inc.
Toshiba Carrier Co.
Material (6)
Hitachi Cable, Ltd.
Kobelco & Material Copper Tube, Ltd.
Showa Denko K. K.
Showa Tansan Co., Ltd.
Sumitomo Light Metal Industries, Ltd.
The Furukawa Electric Co., Ltd.

3.2.2 Market Research in Japan

Principal Investigator & Co investigator

Makoto Tono

Institution

Heat Pump & Thermal Storage technology Center of Japan

Heat pump systems are mainly employed in domestic and commercial air-conditioners in Japan. In addition to them, water heaters which have been replaced from combustion systems now become one of the main demands of heat pump systems. The promotion of Energy Saving Laws for reduction of greenhouse gases and its supportive measures for further progression accelerate development of component technology and system efficiency.

Residential air-conditioner:

Fig. 3.2.2-1 shows that the shipment numbers of residential air-conditioners have been between 7 million and 7.5 million units since 2005. Most of them are direct expansion heating and cooling with air-source and are categorized under 7.1 kW. 90 percent of all residential air-conditioners are under 4.0 kW category. The APF standard has been employed to show annual efficiency since the Energy-Conservation Standards was set in 2004. And also "The Top Runner Standards" have been adopted after being reviewed the target level of the APF. To meet the top runner standard, the system performance is getting dramatically improved. The improvements are seen in heat pump systems, heat exchangers and compressors etc.

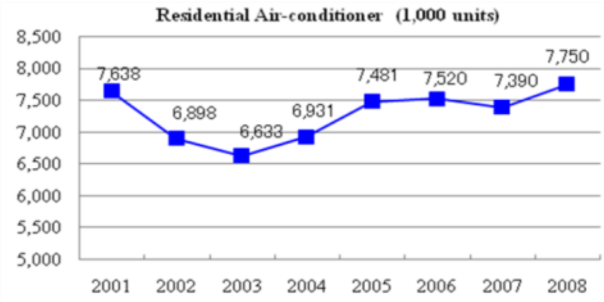


Fig. 3.2.2-1

Commercial air-conditioner:

The shipment numbers of commercial air conditioners have been between 750,000 and 800,000 since 2004. 93 percent of them adopt heat pump systems and they equip both cooling and heating. The number has remained at the same level. The annual energy efficiency has become obliged to show when we had JIS revised in March 2006. The APF has been required to be displayed along with the COP since October 2006. Manufactures developed and commercialized products targeted for cold regions then started selling them so that they can increase their demands in the market. They have developed products by making full use of the inverter technology, adopting a double-stage compressor and adding an injection cycle to a refrigerant circuit to have high efficiency and quick heating technology when outside temperature is low.

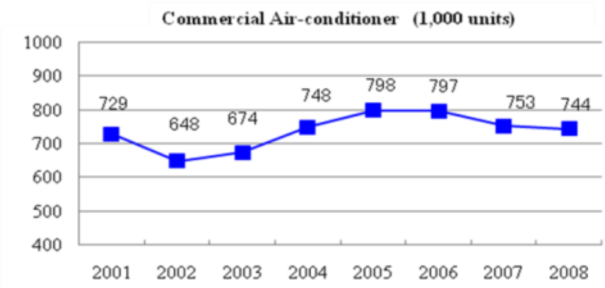


Fig. 3.2.2-2

Echo cute:

As Fig. 3.2.2-3 shows, the Echo Cute, the air-source heat pump water heaters that are used natural refrigerant, CO_2 have rapidly increased their domestic market share. The Japanese government has had financial supporting system for those who purchase an Echo-Cute system so that we can expect large reduction in domestic energy consumption. The accumulated amount of the subsidy is 58 billion yen since 2001 when the supporting system started. The APF should be 3.1 or above in 2009 to apply to the subsidy. It is on first-apply-first-served basis. The sales achieved 1.74 million units in March 2009 and it hit half a million units in single year of 2008.

The efficiency of Eco Cute system has grown year after year thanks to developments in compressors and water heat exchangers and so on. Japan has adopted the APF to the Eco Cute systems as a way to show their efficiency like air conditioners since 2008 and it achieved 3.6 in 2009.

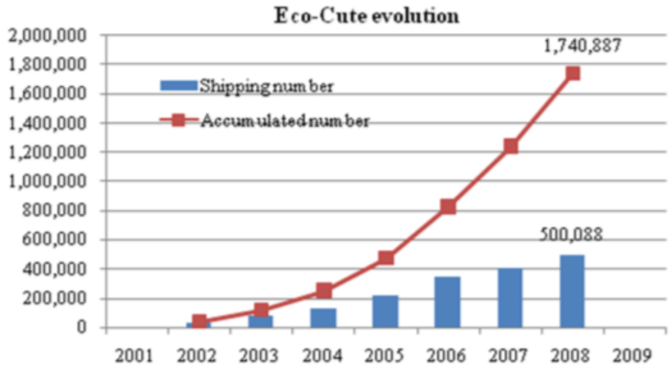


Fig. 3.2.2-3

Chiller:

The shipment numbers of chiller is around 10,000 in Japan and the number has been stable these years.

70 percent of all chillers in Japan are air cooling, and the rest of 30 percent are water cooling. Research and development for high efficiency brought products with COP 5 which have inverter technology for compressors, highly-modularized devises and using R410A refrigerant. The inverter technology and modularizing have contributed to energy saving in accordance with part load performance.

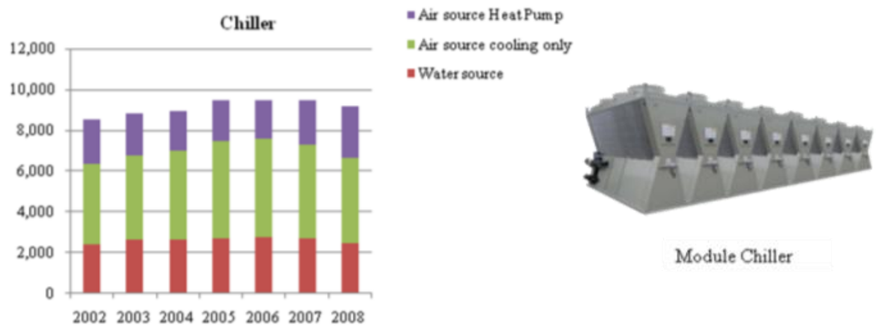


Fig. 3.2.2-4

The above data are based upon the JRAIA heat pump statistics.

3.2.3 Survey of R & D Needs on Heat Exchangers for Refrigeration & Air-conditioning Applications

Principal Investigator & Co investigator

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Institution

Faculty of Engineering Sciences, Kyushu University

Summary

The subcommittee of heat exchanger technology, JSRAE has been established for the purpose of technological development survey and future direction planning of heat exchangers for refrigeration and air conditioning applications. This subcommittee has also been searching suitable research topics to develop a consortium in the field of heat exchangers. In this report, R & D trend survey results on the heat exchangers for refrigeration and air conditioning applications in Japan are summarized, (see also Koyama et al, 2007 – In Japanese)

a. Methodology of R & D trend survey

The subcommittee of heat exchanger technology, JSRAE carried out the R & D trend survey on the heat exchangers for refrigeration and air conditioning applications by using the following methodology:

(1) People at whom the survey was directed were nominated senior engineers from Japanese refrigeration and air-conditioning industries (total 8 persons).

(2) Nominated persons answered the questionnaire of the survey. Specific subjects which should be tackled by JSRAE were described in unrestrictive manner. Questionnaire of the survey is shown in Table 3.2.3-1. Technology fields for the survey were classified to eighth items as, 1) refrigerant type, 2) refrigerant side heat transfer processes, 3) heat transfer surface, 4) secondary refrigerant side heat transfer for air and brine, 5) heat exchangers, 6) new research field related to heat exchanger technology, 7) applied technologies such as nano technology, 8) other problems not mentioned in items 1) to 7).

Table 3.2.3-1 Items of 1st questionnaire

Item No.	Contents
1	Refrigerant types (Next generation refrigerant, Future refrigerant, Refrigerant mixture and other perspectives)
2	Refrigerant side heat transfer processes (Condensation, Evaporation, Absorption, Adsorption and other perspectives)
3	Heat transfer surface (Surface type and geometry, Copper tube, Aluminum tube, Micro channel and other perspectives)
4	Secondary refrigerant side heat transfer for air and brine (Heat transfer of fins, Frosting, Defrosting, Mist and other perspectives)
5	Heat exchanger (Heat exchanger type, Distribution of refrigerant, Performance improvement, Compactness and other perspectives)
6	New research field related to heat exchanger technology
7	Applied technologies such as nano technology
8	Other problems not mentioned in Items 1 to 7

(3) Collected survey results should only be handled by the sub-committee chairperson as to protect information leakage of technical strategy of individual companies. Then, collected survey results were rearranged and summarized by the sub-committee chairperson.
(4) Summarized survey results are returned to the nominated persons for checking and modification if necessary.
(5) The sub-committee chairperson finalizes the survey results.
(6) The finalized survey results were evaluated by nominated persons according to the grading procedure shown in Table 3.2.3-2. In this table, importance has five steps of evaluation, 1 point represents the subject is O.K or may be important, 3 point means the subject is important, and 5 points means the subject must do and is the most important. Time schedule is categorized in three steps as short, medium and long terms. If the nominated person thinks that the subject is suitable for consortium, he will put circle symbol in the subject.

Table 3.2.3-2 Grading procedure

1. Importance	5 Steps of evaluation (2 or 4 also O.K.) 1 : Having O.K. · May be important 3 : Important
2. Time Schedule	Following 3 Steps evaluation Short term : Start from 2006 between 2008 (1Year) Medium term : Start from 2009 between 2011 (4 Years) Long term : Start from 2011 onward (7 Years)
3. Consortium	If agree, put ○ symbol

b. Results of R & D trend survey

Table 3.2.3-3 shows the R & D trend survey results of Item 1) refrigerant types. Important research subjects are natural refrigerants and their mixtures, evaluation of influence of refrigerants on global warming impact, and selection of optimum refrigerants for individual operating conditions. Time schedule for these important subjects is about two to four years.

Table 3.2.3-4 shows the R & D trend survey results of Item 2) refrigerant side heat transfer processes. In this item, important research subjects are heat transfer of CO_2, effect of oil on CO_2 heat transfer, visualization and numerical simulation of two phase flow, and heat transfer in non-circular tubes. In these subjects, CO_2 heat transfer research is the most urgent and important topic and should be solved within a short term.

Table 3.2.3-5 shows the R & D trend survey results of Item 3) heat transfer surface. Important research subjects are channel size design limitation of micro-channel and grooved shape for CO_2 applications, in- and out-tubes heat transfer of natural refrigerant, and heat transfer enhancement on air side. Time schedule for these important subjects is about three years and highly reliable academic results are essential to design heat exchangers.

Table 3.2.3-6 shows the R & D trend survey results of Item 4) secondary refrigerant side heat transfer. Relatively important research subjects are heat transfer enhancement by using new type fins such as delta fin, surface treatment, and frosting and defrosting phenomena. Time schedule for these subjects is about four to five years.

Table 3.2.3-3 R & D survey results of Item 1)

Refrigerant types (Next generation refrigerant, Future refrigerant, Refrigerant mixture and other perspectives)			Average	Average (Year)	Itemwise Count
No.	Theme	Remarks	Importance	Time Schedule	Consortium
1(1)	Pure refrigerants	CO_2, air, water, propane, butane, ammonia, etc.	4.1	2.3	2
1(2)	Natural refrigerant mixtures	CO_2/HC type refrigerant mixtures and R723 (ammonia/DME refrigerant mixtures) and others	3.0	4.4	3
1(3)	Search of next generation refrigerants and their application possibilities	Search of next generation refrigerants accepting from the viewpoint of environmental load, system performance, safety (flammability)	3.7	5.7	1
1(4)	Intended purpose of refrigerants (purpose, application condition)	Intended purpose of refrigerants, country · area (purpose, application condition)	2.1	4.9	0
1(5)	Total evaluation regarding global warming of refrigerants	Total evaluation from the viewpoint of effectiveness of global warming prevention, energy efficiency and others	4.0	3.6	3
1(6)	Selection of refrigerants from academic viewpoint	We should select suitable refrigerants from academic viewpoint (should not be selected by national policy)	3.9	3.6	4
1(7)	Preferred technology considering refrigerant characteristics	Refrigerant characteristics (merit and demerit) matching: high pressure, flammability, corrosion and others	3.6	3.1	0
1(8)	Refrigeration system using secondary refrigerants such as HC cycle + CO_2	Refrigeration system using secondary refrigerants such as HC cycle + CO_2	3.6	3.6	0
1(9)	New fluorocarbon based alternative refrigerants having low global warming impact potential	Search of new fluorocarbon based refrigerants having low GWP (in the case of automobile air conditioning)	2.9	4.0	2

Table 3.2.3-4 R & D survey results of Item 2)

Refrigerant side heat transfer processes (Condensation, Evaporation, Absorption, Adsorption and other perspectives)			Average	Average (Year)	Itemwise Count
No.	Theme	Remarks	Importance	Time Schedule	Consortium
2(1)	Heat transfer characteristics of CO_2/HC refrigerant mixtures	Condensation and evaporation heat transfer characteristics of CO_2/HC refrigerant mixtures	3.1	3.1	2
2(2)	Heat transfer characteristics of oil mixed CO_2	Heat transfer reduction due to oil mixed in CO_2	4.9	1.0	3
2(3)	CO_2 solid-vapor two-phase heat transfer	CO_2 solid-vapor two-phase heat transfer, chilled energy below triple point temperature (–56.6 ºC), sublimation latent heat utilization, heat transfer enhancement	2.4	4.0	1
2(4)	Development of empirical equation of heat transfer characteristics of super critical CO_2 based on experimental results	Experiment on heat transfer characteristics of super critical CO_2 and development of empirical equation (specially in the case of pseudo-critical temperature)	4.4	1.0	4
2(5)	In-tube two-phase flow visualization and numerical analysis	Flow visualization in actual heat exchanger and numerical analysis	3.9	4.0	5
2(6)	Heat transfer characteristics in non-circular tubes	Experimentally determined heat transfer characteristics in non-circular tubes (plate type, fin and plate, corrugated fin and flat tube, etc.)	3.4	3.6	2
2(7)	Heat transfer performance in a single small diameter circular tube	Heat transfer performance in a single small diameter circular tube which has the diameter of around 5 mm to lower than 1 mm	3.1	3.1	3
2(8)	Heat transfer tubes having low pressure drop during evaporation and high heat transfer coefficient during condensation	Heat transfer tubes having low pressure drop during evaporation and high heat transfer coefficient during condensation	2.4	4.0	0
2(9)	Improvement of heat transfer characteristics of liquid phase during low void fraction	Improvement of heat transfer characteristics of liquid phase during low void fraction	2.7	4.4	0
2(10)	Approach from the viewpoint of reliability and quality of product	Academic approach from the viewpoint of reliability and quality of product as well as refrigerant side heat transfer performance	2.0	6.1	1

Table 3.2.3-5 R & D survey results of Item 3)

Heat transfer surface (Surface type and geometry, Copper tube, Aluminum tube, Micro channel and other perspectives)			Average	Average (Year)	Itemwise Count
No.	Theme	Remarks	Importance	Time Schedule	Consortium
3(1)	Effect of u-bend on heat transfer	Effect of u-bend and other bends on heat transfer	2.3	3.6	1
3(2)	Performance evaluation of oval tube	Utilization of oval tube used in evaporation cooling, (in-tube condensation, evaporation at the outer surface of tube): quantitative evaluation is necessary	2.0	4.9	1
3(3)	Discussion and evaluation on heat transfer surface of CO_2 solid-vapor two-phase flow	Discussion and evaluation on heat transfer surface of CO_2 solid-vapor two-phase flow considering heat transfer surface type, surface geometry and materials	2.6	4.2	1
3(4)	Search of optimum heat transfer surface	Search of optimum heat transfer surface (need not consider the development procedure)	3.0	4.6	0
3(5)	Channel size design limitation of micro-channel and grooved shape for CO_2 applications	Channel size design limitation of micro-channel and grooved shape for CO_2 applications	3.4	3.6	4
3(6)	Materials for heat exchangers (all Al, all Cu, Al-Cu)	Discussion on heat exchanger: (1) Material (Al, Cu, etc.), (2) All Al, all Cu, cross fin heat exchanger using Al and Cu	2.0	5.5	0
3(7)	Outer surface heat transfer for thermal recycling from waste water	Self cleaning of outer surface (secondary refrigerant side), heat recovery from waste water	1.6	5.7	1
3(8)	Discussion on micro-channel: reduction of refrigerant charging amount	Discussion on cu tube, micro-channel, reduction of refrigerant charging amount, environment load reduction	2.3	3.4	0
3(9)	In-tube and outer tube heat transfer of natural refrigerant	Natural refrigerant + inner grooved tube, natural refrigerant + outer tube heat transfer (condensation, boiling)	3.7	2.7	3
3(10)	Enhancement of heat transfer area (utilizing nano particles)	Enhancement of heat transfer area (porous material made of nano particles)	3.0	4.4	2
3(11)	Heat transfer enhancement of al tube (grooved tubes having large helix angle)	Heat transfer enhancement of al tube (grooved tubes having large helix angle)	2.3	4.4	1
3(12)	Heat transfer enhancement on air side (swirl, pulsated motion, pin fin)	In between fins of air side: (1) Heat transfer enhancement utilizing swirl, (2) Heat transfer enhancement using air intermittent flow, (3) pin fin	3.1	2.7	2
3(13)	Optimization of ice making machine: heat transfer surface which enhanced ice removal	Optimization of ice making machine: ice removal and production enhancement by surface treatment	2.0	4.8	0

Table 3.2.3-6 R & D survey results of Item 4)

Secondary refrigerant side heat transfer for air and brine (Heat transfer of fins, Frosting, Defrosting, Mist and other perspectives)			Average	Average (Year)	Itemwise Count
No.	Theme	Remarks	Importance	Time Schedule	Consortium
4(1)	Heat transfer enhancement by using new type fins such as delta fin	Heat transfer enhancement by treating surface of new type fins such as delta fin	3.3	4.0	1
4(2)	Surface treatment and geometry of fins・heat transfer performance of fins during frosting	Effect of surface treatment of fins on frosting, effect of geometry of fins on frosting and defrosting	3.0	4.0	2
4(3)	Optimization of heat transfer in fins at wet condition and water drainage from fin surface	Optimization of heat transfer in fins at wet condition and water drainage from fin surface	2.6	4.4	1
4(4)	Applicability of CO_2 as brine : advanced thermophysical properties, latent heat utilization	Energy efficiency achievement employing advanced thermophysical properties, latent heat utilization	2.7	3.6	1
4(5)	Development of high performance fin for application in frosting condition	Development of high performance fin for application in low ambient temperature condition	2.6	4.0	1
4(6)	Relationship between wettability of surface and frosting	Relationship between wettability of fin surface and frosting	2.7	4.0	3
4(7)	Mist formation mechanism and its countermeasure	Mist formation mechanism and its countermeasure	2.3	4.0	1
4(8)	Secondary refrigerant (brine, CO_2, etc.)	Secondary refrigerant (brine, CO_2, etc.)	2.4	4.9	2
4(9)	Waste water (pollution・high viscosity fluid)	Waste water (pollution・high viscosity fluid)	1.3	5.7	0
4(10)	Suppression of frosting and shortening defrosting time	Suppression of frosting and shortening defrosting time during heating mode	2.1	4.0	1
4(11)	Optimization of mist cooling of heat exchanger applied in evaporation cooling	Optimization of mist cooling by minimizing sensible heating in evaporation cooling process	1.7	4.4	0
4(12)	Research overview on frosting and defrosting	Recent research overview on frosting and defrosting (removing moisture from air using desiccant dehumidifier)	3.1	4.0	2

Table 3.2.3-7 shows the R & D trend survey results of Item 5) heat exchangers. Important research subjects are compact heat exchangers using mini-and micro channels, application of internal heat exchangers, all-aluminum and all-copper heat exchangers. Time schedule for these subjects is about four years.

Table 3.2.3-7 R & D survey results of Item 5)

Heat exchanger (Heat exchanger type, Distribution of refrigerant, Performance improvement, Compactness and other perspectives)			Average	Average (Year)	Itemwise Count
No.	Theme	Remarks	Importance	Time Schedule	Consortium
5(1)	Ultra thin tubes heat exchanger	Ultra thin tubes heat exchanger : simultaneous development of fins and tubes, is it possible to use multi-port extruded tubes?	3.4	4.4	2
5(2)	Development of general design method for refrigerant distributor	Development of general design method for refrigerant distributor	3.6	4.0	3
5(3)	Direct contact heat exchange between air and refrigerant	Direct contact heat exchange between air and refrigerant	2.1	6.6	1
5(4)	Direct contact heat exchange between liquid (secondary refrigerant) and refrigerant	Direct contact heat exchanger : (1) ice production by injecting refrigerant directly into water, (2) dry ice production by injecting CO_2 directly into low-temperature brine, and optimization of heat transfer in the above mentioned and similar cases.	2.1	6.1	0
5(5)	Heat transfer enhancement during simultaneous utilization of sensible and latent heats	Heat transfer enhancement during simultaneous utilization of sensible and latent heats: cascaded, evaporation-condensation, etc.	2.3	4.4	1
5(6)	Performance improvement of fin-tube heat exchanger	Performance improvement and compactness of fin-tube heat exchanger by understanding local characteristics of heat transfer and pressure drop	3.6	3.1	3
5(7)	Experiment on refrigerant distribution and development of correlation equation	Experimental investigation of refrigerant flow behavior in branching conduits, header and distributor; and development of correlation equation	3.9	3.6	4
5(8)	Refrigerant-refrigerant heat exchanger	Study on internal heat exchanger will be important in near future due to wide application of natural refrigerants and so on.	3.6	4.0	2
5(9)	Compactness of heat exchanger and reduction of materials	Compactness of heat exchanger and reduction of materials and space reduction	2.6	4.6	1
5(10)	Same as 5-(1)	Reduce diameter	3.3	4.4	3
5(11)	Multi-port extrude tube heat exchanger	Multi-flow type heat exchanger by using Multi-port extrude tube	3.4	4.4	2
5(12)	All aluminum heat exchanger for the application of heat pump	Parallel flow type all aluminum heat exchanger for the application of heat pump	3.3	4.4	1
5(13)	Same as 5-(8)	Refrigerant-refrigerant heat exchanger	2.3	5.7	0
5(14)	Heat exchanger for air conditioner using radiation heat transfer	Heat exchanger for air conditioner using radiation heat transfer or other hybrid heat transfer	2.7	4.9	1

Table 3.2.3-8 shows the R & D trend survey results of Item 6) new research field related to heat exchanger technology. Subjects of this item are not so important for engineers working in refrigeration and air-conditioning fields. Time schedule for these subjects is about four to six years.

Table 3.2.3-9 shows the R & D trend survey results of Item 7) applied technologies such as nano technology. Important subject of this item is fundamental study on nano heat transfer science. Time schedule for this subject is about five years.

Table 3.2.3-10 shows the R & D trend survey results of Item 8) Other Problems not mentioned in Items 1) to 7). Subjects in this item are evaluation of effectiveness on systems and cycles, and thermal contact resistance. Time schedule for these subjects is about three to four years.

Table 3.2.3-8 R & D survey results of Item 6)

New research field related to heat exchanger technology

No.	Theme	Remarks	Average Importance	Average (Year) Time Schedule	Itemwise Count Consortium
6(1)	Application of heat exchanger technology in electronic device	Application of heat exchanger technology in electronic device	2.4	4.9	0
6(2)	Heat exchanger for vapor re-compression (VRC) system	Heat exchanger for vapor re-compression (VRC) system (in-tube : condensation of compressed vapor/outer tube: solidification of solution or slurry)	2.1	4.9	1
6(3)	Same as 5-(5)	Enhancement of heat transfer	2.4	4.4	1
6(4)	Application of heat transfer enhancement technology in other fields	Search on application of heat transfer enhancement technology in new field	2.7	6.6	4
6(5)	Optimization of combined heat exchangers in air conditioning and hot water supply systems	Selection criteria of combined heat exchangers in air conditioning and hot water supply systems	3.1	4,4	0
6(6)	Refrigerant charge amount adjustment in heat exchanger and capacity control of the system	Refrigerant charge amount adjustment in heat exchanger employing evaporation and condensation phenomena and capacity control of the system	2.7	3.6	1
6(7)	Active heat transfer enhancement technology	Active heat transfer enhancement technology (water, refrigerant air, etc.)	2.9	5.7	3

Table 3.2.3-9 R & D survey results of Item 7)

Applied technologies such as nano technology

No.	Theme	Remarks	Average Importance	Average (Year) Time Schedule	Itemwise Count Consortium
7(1)	Precise treatment of heat transfer surface	Heat transfer enhancement by precise treatment of heat transfer surface	3.1	4.4	2
7(2)	Nano order evaporation and condensation characteristics	Nano order evaporation and condensation characteristics in the case of two-phase flow	3.7	4.9	3
7(3)	Heat transfer characteristics of nano order tube	Clarification of heat transfer characteristics of nano order tube	3.6	6.1	4
7(4)	Adsorption-desorption characteristics of adsorption heat pump system	Adsorption-desorption characteristics of adsorption heat pump system: (1) Low temperature waste heat utilization, (2) Life span enhancement and reduction of initial cost	2.4	4.9	1
7(5)	New material development for PSA equipment that collects emitted CO_2	New material development for PSA equipment that collects emitted CO_2 : new adsorbent for CO_2 adsorption-desorption	2.4	4.3	0
7(6)	Advanced surface treatment	Advanced surface treatment : (1) Anti pollution, anti fungal, adsorb smell, hydrophobic, (2) Method to develop light entrance procedure	2.3	4.9	1
7(7)	Survey on trend of treatment technology and its application	Trend of top runner treatment technology, development and application of heat transfer enhancement technology and precise treatment technology	2.9	4.4	2
7(8)	Fundamental study on nano heat transfer science	Fundamental study on nano heat transfer science for the development of ultra high heat transfer processes	4.1	4.9	4
7(9)	Frosting control・pollution control coating	Avoiding the efficiency drop by frosting control and pollution control coating	2.9	4.4	0
7(10)	Heat transfer enhancement of refrigerant mixtures using nano-fluids	Is it possible to enhance heat transfer using nano-fluids?	2.7	6.1	1

Table 3.2.3-10 R & D survey results of Item 8)

Other problems not mentioned in Items 1 to 7

No.	Theme	Remarks	Average Importance	Average (Year) Time Schedule	Itemwise Count Consortium
8(1)	Prediction of thermal contact resistance between heat transfer tube and fin	Prediction of thermal contact resistance between heat transfer tube and fin	2.7	3.1	1
8(2)	Analysis of frozing phenomena during pressurized condition and development of freezer	(1) Analysis of frozing phenomena during pressurized condition, (2) development of heat exchanger and freezer, (3) formation of ozone content ice	1.6	4.9	1
8(3)	Evaluation of effectiveness on systems and cycles	Numerical investigation on heat exchanger technology along with the evaluation of effectiveness on systems and cycles	3.0	4.0	2
8(4)	Development of new material for heat exchanger	Development of new material for heat exchanger (development of functional material, application)	2.4	6.6	0
8(5)	Thermal management by considering atmospheric condition such that humidity control	Thermal management and thermal flow direction control by considering atmospheric condition such that humidity control	2.4	5.3	3

The selected important subjects through the R & D trend survey in Japan are summarized as follows.

- Subjects to be investigated urgently -
  1) Effects of Oil on CO_2 Heat Transfer (2(2))
  2) Establishment of Correlation of CO_2 Heat Transfer at Super-critical Condition(2(4))
- Subjects to be investigated within three years -
  1) Heat Transfer in Nano-ordered Tube (7(3))
  2) In- and Out-tube Heat Transfer of Natural Refrigerants (3(9))
  3) Properties of Pure Component Natural Refrigerants (1(1))
  4) Refrigerant Distribution in Heat Exchangers (5(7))
  5) Selection Methodology of Suitable Refrigerants Based on Academic Viewpoints (1(6))
  6) New Fluorinated Alternative Refrigerants Based on Ethers, Olefin etc. (1(9))
  7) Performance Improvement of Fin & Tube Heat Exchangers (5(6))
  8) Miniaturization and Optimization of Grooved Tubes for CO_2 (3(5))
  9) Heat Transfer in Non-circular Tubes (2(6))
- Subjects to be investigated within around five years -
  1) Visualization and Numerical Simulation of Two Phase Flow (2(5))
  2) Refrigerant Distribution in Heat Exchangers (5(2))
  3) Nano-scale Heat Transfer and Its Applications for Heat Transfer Enhancement (7(8))
  4) Nano-scale Phase Change Phenomena (7(2))
- Universal Subject -
  1) Exploring New Alternative Refrigerants and Evaluation of Their Applicability

Relation between ‘Importance’ and ‘Time for Research Subjects’ can be characterized as: In case of important subject, the higher the rank is, the shorter the time schedule is. Lower ranked subjects have longer time schedule. It may be noted that lower ranked subjects do not mean their un-necessities in a long run.

In the following Sections, data on Japanese research projects are given in the form, of Project Summaries.

Full data are given in Appendix 2 - see after each summary for the specific Appendix reference.

3.2.4 R & D on Heat Exchangers relevant to CO_2 Heat Pump Water Heaters

3.2.4.1 Study on heat transfer characteristics of carbon dioxide at both supercritical and subcritical pressures

Principal Investigator

Eiji Hihara and Chaobin Dang

Institution

Dept. of Human and Environmental Engineering, University of Tokyo

Summary

Since the CO_2 heat pump works at a relatively high operation pressure, the heat transfer characteristics of CO_2 differ from conventional CFC or HFC refrigerants, and a systematic investigation on both the cooling heat transfer of CO_2 at supercritical pressure and the flow boiling heat transfer at subcritical pressure is critical for the design and optimization of the heat exchangers for CO_2 heat pump.

To clarify the characteristics of flow and heat transfer of supercritical CO_2 cooled in smooth tube, experimental measurement, theoretical analysis and numerical simulation were conducted. From the comparison between the experimental results and numerical simulation results, the proper selection of turbulence models for supercritical CO_2 was discussed. Based on the understanding of temperature and velocity distributions at cross-sectional direction provided by the numerical simulation, a new prediction correlation was proposed which agreed well with the experimental results. In addition, the effect of lubricating oil on the heat transfer characteristics of supercritical CO_2 was studied through heat transfer measurement and flow pattern visualization. The heat transfer characteristics of supercritical CO_2 inside a small-sized micro-fin tube were also investigated.

Flow boiling heat transfer of CO_2 was experimentally investigated for smooth tube with tube diameter ranging from 1 ~ 6 mm and for micro-fin tube with mean inner diameter of 2 mm. heat transfer characteristics of pure CO_2 and the effect of lubricating oil were studied under different mass flux and heat flux conditions. Nucleate boiling was found the dominant heat transfer mechanism for both the smooth and micro-fin tubes due to its low surface tension and high density ratio of vapor to liquid of CO_2. Although the CO2 has a high heat transfer coefficient at pre-dryout region, the dryout quality is usually low compared to conventional refrigerants. Contaminating lubricating oil into CO_2 led to a sharp decrease in heat transfer coefficient. By using micro-fin tube, the dryout shifted to a high vapor quality side and the heat transfer performance enhanced significantly.

Full data are given in Appendix A2.2 and A2.3.

3.2.4.2 Heat transfer characteristics of carbon dioxide inside spirally grooved tubes

Principal Investigator

Shigeru Koyama

Institution

Faculty of Engineering Sciences, Kyushu University

Summary

Recently, from the viewpoint of grovel environmental protection, the development of high-performance heat pump and refrigeration systems using naturally occurring substances, such as CO_2, NH_3, i-C_4H_{10}, as working fluids has become the most important challenge. Therefore, we have investigated the heat transfer characteristics of CO_2 in spirally grooved tubes experimentally. This report deals with our group's experimental research concerning the forced convection cooling heat transfer of CO_2 in the supercritical pressure range and the flow boiling heat transfer of CO_2 in the subcritical pressure range inside several spirally grooved tubes.

For reference, with the grooved tube G1 as a representative example, the relations between the quality of the predicted values and experimental values calculated by the correlation equation of heat transfer are shown in Fig. 3.2.4.2-6. The symbol open circle in the figure represents an experimental value, and the solid lines represent predicted values of heat transfer coefficients, the broken lines represent the contributions of nucleate boiling heat transfer in the predicted values of heat transfer coefficients. As for the predicted values of heat transfer coefficients, the contribution of nucleate boiling heat transfer accounts for the most part of the region and the contribution of forced convection heat transfer accounts for little in the low quality region. The contribution of nucleate boiling decreases in proportion to the increase in quality. Under the pressure of 3 MPa, if the quality is about 0.8, the contribution of nucleate boiling is very small.

Further data are given in Appendix A2.4.

3.2.4.3 Experimental Study of Carbon Dioxide Flow Boiling Heat Transfer Coefficient in Horizontal Grooved Tubes.

Principal Investigator & Co investigator

Katsumi Hashimoto[a)] and Naoe Sasaki[b)]

Institution

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Summary

Protection of the environment is becoming more important globally, and considerable effort is being spent on developing heat pumps with natural working fluids. After

surveying the status of R&D on heat pumps with natural working fluids, at the Central Research Institute of Electric Power Industry (CRIEPI) focused on a CO_2 heat pump, upon which a basic study was commenced in 1995. In Japan, the world's first CO_2 heat pump for residential hot water heaters was developed in a joint research project involving DENSO, the Tokyo Electric Power Company and CRIEPI, and was released as a commercial product in May 2001 (Saikawa et al. 2000). With other manufacturers joining the market, this added impetus to the residential hot water heater market. Total shipments in the 2008 fiscal year numbered about 500,000 units, with the figure since 2001 exceeding 1,500,000 in 2008. The Japanese government has a strong interest in this technology because of its potential to save energy and abate greenhouse gases. To develop a more efficient and compact heat pump, the evaporator needs to perform better and be smaller than a conventional evaporator. It is important to accurately measure the flow boiling heat transfer coefficient and to understand the heat transfer mechanism. In our previous report, it is understood that mean heat flux, like mass velocity, considerably affects the mean evaporating heat transfer coefficient in a horizontal smooth tube (99.999% purity, without oil, outer diameter 6 mm, thickness 0.4 mm). In this report, the heat transfer coefficient and pressure drop of a smooth tube and 3 kinds of grooved tubes are measured in a collaborative study between Sumitomo Light Metal Ltd. and CRIEPI in order to find suitable inner groove form for carbon dioxide.

The heat transfer coefficient and pressure drop of three kinds of groove tubes and a kind of smooth tube were measured via collaboration between Sumitomo Light Metal, Ltd. and CRIEPI and the following conclusions were obtained:

1) Pressure drop increases according to the increment ζ of tubes,
2) The heat transfer coefficient of the grooved tubes increases monotonously, namely, does not decrease in a region with a mass velocity exceeding 400 kg/(m^2s) like that of the smooth tube,
3) The heat transfer coefficient of grooved tubes increases according to ζ, and
4) Suitable inner groove form is GT2 as far as we examined in this report.

Full data are given in Appendix A2.5.

3.2.4.4 Development of World's first CO_2 heat pump water heater (ECO CUTE) and compact heat exchanger for water heating.

Principal Investigator & Co investigator

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Institution

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Summary

In Japan, energy demand for hot tap water accounts for about 30 % of the total residential final energy consumption, but most of this demand (over 90 %) is met by

the direct combustion of fossil fuel. The development of high-performance heat pump water heaters using natural working fluids is thus eagerly anticipated for energy conservation and greenhouse gas reduction. If the COP of a heat pump water heater is 3 or more, the heat pump water heater uses about 30 % less primary energy than a combustion water heater.

The Central Research Institute of Electric Power Industry (CRIEPI) has been studying heat pumps using CO_2 as a refrigerant since 1995, and has found by theoretical analysis that the CO_2 cycle has unique characteristics and can achieve a higher COP than conventional refrigerants for domestic hot water production, and by experiments that the CO_2 cycle can be effectively controlled by the combination of an automatic expansion valve and a variable-speed compressor.

The annual average COP was evaluated using performance test results for the final prototype. The evaluated system COP, which includes the power input to the air fan and water pump, was 3.4. This value was better than the targeted value of 3.0. In addition, it was confirmed that the final prototype could produce hot water at 90°C at an ambient air temperature of -15°C. In May 2001, CO_2 heat pump water heater named ECO CUTE was commercialized. "ECO CUTE" is a name used by electric power companies and water heater manufacturers, and refers only to heat pump water heaters using CO_2 as a refrigerant. Because energy saving and greenhouse abating potential of ECO CUTE were highly evaluated, ECO CUTE won various prizes.

Full data are given in Appendix A2.6.

3.2.4.5 CO_2 Heat Pump Water Heater Gas-Cooler Enhancement Technology for CO_2 Heat Pump Water Heater Gas-Coolers

Principal Investigator & Co investigator

Kazuhiko Machida, Takumi Kida and Shoichi Yokoyama

Institution

Home Appliances Company, Panasonic Corporation

Summary

The curtailment of CO_2 emission volumes has been called for in recent years from the perspective of protection of the global environment and the diffusion of CO_2 heat pump water heaters that offer greater energy saving and have a smaller impact on the global environment than conventional heating appliances is progressing. Given such circumstances, the increasingly higher efficiency of products is advancing at a rapid rate in association with the increasingly expanding market for CO_2 heat pump water heaters.

This report introduces examples of efforts made in recent years at Panasonic in regard to technology to increase the performance of gas coolers, which play a crucial role in the greater efficiency of CO_2 heat pump water heaters.
We developed [1] twist, [2] dimple and twist, and [3] high density dimple and twist specifications for heat transfer tubes as an undertaking to enhance the performance of gas coolers. In particular, with the high density dimple and twist specifications, it is

possible to improve on the straight tube water side heat transfer ratio by a factor of about 2.0 while keeping pressure drops at low levels. This has contributed significantly to the enhanced performance of Panasonic gas coolers.

Further data are given in Appendix A2.7.

3.2.5 R & D on Compact Heat Exchangers relevant to Heat Pumps

3.2.5.1 Heat transfer issues in fin and tube heat exchanger

Principal Investigator

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Summary

Fin and tube heat exchangers are widely used in air-conditioners and refrigeration systems. For higher COP, heat exchangers which have high heat transfer performance and low pressure loss are desired. Heat transfer resistances of the fin-tube heat exchanger are (1) the convective heat transfer resistance of refrigerant flowing inside the tube, (2) the thermal contact resistance between tubes and fins, and (3) the convective heat transfer resistance of air flowing through fins. It is said that degrees of the resistances are 15~20% for the refrigerant side, 10~15% for thermal contact of tube and fin, and 70~75% for air-side. In the last decades, the efforts have been poured into development of convection heat transfer enhancement for refrigerant and air sides. The air side heat transfer has been considerably enhanced by wavy, louver, and offset fins. The heat transfer enhancements of condensing or evaporating refrigerant have been exclusively achieved by microfin tubes, because they give considerable improvement of heat transfer coefficient accompanying small increase of pressure loss. Although the contact resistance is relatively small, it becomes not to be ignored because other resistances are reducing.

Further data are given in Appendix A2.8.

3.2.5.2 Development of New Printed Circuit Heat Exchanger

Principal Investigator

Yasuyoshi Kato

Institution

Research Laboratory for Nuclear Reactors, Tokyo Institute of Technology

Summary

A new Printed Circuit Heat Exchanger (PCHE) has been proposed as a Micro Channel Heat Exchanger (Fig.3.2.5.2-1) through three dimensional thermal-hydraulic simulations for recuperators of a carbon dioxide gas turbine cycle, which has discontinuous “S”-shape fins and provides flow channels with near sine curves. Its

pressure drop is one-sixth reference to the conventional PCHE with zigzag flow channel configuration while the same high heat transfer performance inherits. Thermal-hydraulic performance of the new PCHE has been studied parametrically changing the "S" shape fin design parameters: fin angle, fin width, size, and edge sharpness. The pressure drop reduction is ascribed to elimination of recirculation flows and eddies that appears around bend corners of zigzag flow channels in conventional PCHE [1]. Moreover, Empirical correlations of heat transfer and pressure drop performance have been proposed for the micro channel heat exchanger for a hot water supplier with S-shaped fins, using supercritical carbon dioxide as the heating medium [2]. The test section was constructed from flat copper plates that had fluid flow passages chemically etched on their surfaces. The plates were then diffusion-bounded together into a block to from a heat exchanger core. The new PCHE has core dimensions of 78 x 1000 x 14 mm and the flow channels take sine-wave forms by S-shaped fins. Total numbers of flow channels are 54 and 4, respectively, for the hot and cold side. This PCHE has one H2O and two CO2 plates which are arranged in a sandwich structure in what we call a double banking model. The fluid flow of H2O and CO2 is arranged as counter-current flow.

See Appendix A2.9 for Figures & Tables.

3.2.5.3 Numerical analysis on laminar film condensation of pure refrigerant inside a vertical rectangular channel

Principal Investigator & Co investigator

Shigeru Koyama and Tatsuya Matsumoto

Institution

Faculty of Engineering Sciences, Kyushu University

Summary

In the present study, a theoretical analysis for the laminar film condensation in a finned vertical rectangular channel is carried out to clarify the heat transfer characteristics of plate-fin condensers. In the analysis the following assumptions are employed. The bulk vapour is pure and saturated, and the effect of viscous shear of vapour on the liquid film is negligible. The heat conduction in the fin is one-dimensional, and the base surface temperature is a constant. The local characteristics of liquid film shape and fin temperature are examined, and a heat transfer correlation including the effects of fin shape parameters is proposed.

See Appendix A2.10 for full data.

3.2.5.4 Experimental Study on Gas-Liquid Flow Distribution to Multiple Upward Channels of Compact Evaporator

Principal Investigator

Masafumi Hirota

Institution

Department of Mechanical Engineering, Mie University

Summary

The multi-pass channels have been used in evaporators for the automobile air-conditioning system to improve its heat transfer performance. In those channels the maldistribution of gas and liquid from the dividing header to the branches has been a serious problem. Many studies have been conducted on this subject, but no systematic results have been obtained to date because the distribution characteristics change depending on many parameters. Among those parameters, the pressure distribution in the combining header, i.e., outlet pressure of branches, would be one of the most important factors. In most studies, however, the condition at the outlets of branches has been quite obscure. Therefore, in this study, air-water flow distribution into multiple upward channels that simulates the evaporator, was examined with attention to the influence of the outlet pressure of branches on the flow distributions.

See Appendix A2.11 for further data.

3.2.5.5 Characteristics of Air-Water Flows in Micro-grooved Evaporator Flat Channels with a Sharp 180-Degree Turn

Principal Investigator

Masafumi Hirota

Institution

Department of Mechanical Engineering, Mie University

Summary

In the air-conditioning systems of automobiles, evaporator should be equipped in quite a small space of the passenger’s compartment, in which the temperature reaches more than 60 degrees Celsius. In response to such demands, a drawn cup type heat exchanger (see Appendix A2.12) had been developed. It consists of sharp 18-degree turned refrigerant channels with a very flat rectangular cross-section of 40 mm x 2 mm.

In general, a stagnation and dry-out of refrigerant liquid films lead to reduced heat transfer performance. Although protruding cross-ribs had been applied and cooling performance could be improved, pressure loss of refrigerant flow and the resulting refrigerant compressor load also increased. In order to develop a new type of channel wall with high ability of liquid film formation and low pressure loss, microgrooves had been engraved on the channel wall.

In this report, experimental results of local void fractions and pressure loss for air-water two-phase flow in microgrooved channels are shown.

See Appendix A2.12 for full data.

3.2.5.6 Visualization and Measurement of Refrigerant in a Commercial Plate Heat Exchanger by Neutron Radiography

Principal Investigator

Hitoshi Asano

Institution

Department of Mechanical Engineering, Kobe University

Summary

Plate heat exchanger (PHE) is widely used in industrial process as a liquid-liquid heat exchanger. Recently, applications of plate heat exchanger to gas-liquid two-phase flows have become of interest for the compactness and heat transfer performance improvement. A PHE is made by brazing 20 to 280 sheets of stainless steel wavy plates. Working fluids flow through the gaps between these plates. The channel of each fluid is arrayed alternately, that is, a plate heat exchanger has many parallel channels. In the case where working fluids flow as a gas-liquid two-phase mixture, the dynamic behaviour of gas-liquid two-phase flow greatly affect on the heat transfer performance. In order to design or improve such heat exchanger for gas-liquid two-phase flows, such as evaporator and condenser, it is important to understand liquid distribution not only in each channel but also into parallel channels. In particular, in the case where the inlet flow is a two-phase flow, the effects of inlet flow conditions and inlet configuration on liquid distribution characteristics should be clarified.

To understand gas-liquid two-phase flow behaviour in a compact heat exchanger with a complicated shape, flow visualization is much efficient. However, since heat exchangers are usually made by metallic material, it is difficult to visualize by optical ray. Radiography is a visualization technique by using the difference in attenuation rate of a radio ray to irradiated materials. X-ray, γ-ray, and thermal and fast neutron rays can be used as a radio ray. Mass attenuation coefficients, μm, of thermal neutron are quite different from those of X-ray and γ-ray as shown in Fig.3.2.5.6-1. While the mass attenuation coefficients of X-ray increase with an increase in atomic number, the attenuation coefficients of thermal neutron ray is high for hydrogen, and is low for metallic elements. Therefore, neutron radiography is efficient for the visualization of gas-liquid two-phase flow in a metallic vessel.

Flows in a single- and multi-channel commercial brazing PHE were visualized by neutron radiography. In the single channel experiments, the effect of flow direction, vertically upward or downward, on boiling heat transfer performance and liquid distribution was examined. On the other hand, in the multi channel experiment, adiabatic air-water two-phase flows were visualized and average liquid hold-up in each channel was measured from neutron radiographs. The effect of inlet flow condition on phase distribution into each channel was considered.

Further data are given in Appendix A2.13.

3.2.5.7 The enhanced heat transfer in the plate-type evaporator by using an ejector for recirculation

Principal Investigator & Co investigator

Tatasunori Man'o[a)], Masayuki Tanino[a)], Takashi Okazaki[b)], and Shegru Koyama[c)]

Institution

a) Research and Development Center, Takasago Thermal Engineering Co. Ltd.
b) Living Environment Systems Laboratory, Mitsubishi Electric Corporation
c) Faculty of Engineering Sciences, Kyushu University

Summary

We have proposed a new method to enhance the average evaporative heat transfer ($\alpha_{r,m}$) in the refrigeration cycle. This method uses an ejector to recirculate vapor refrigerant from the outlet of evaporator to the inlet. It improves the $\alpha_{r,m}$ by increasing vapor quality(x) and mass flow rate of refrigerant at the evaporator inlet. In this study, experiments were performed to evaluate the $\alpha_{r,m}$ and refrigerant flow distribution characteristics of the plate-type evaporator in the refrigeration cycle with the proposed method. The $\alpha_{r,m}$ in an actual evaporator with 50 plates were compared with $\alpha_{r,m}$ calculated by an empirical formula on an evaporator with 6 plates. Furthermore, the refrigerant distribution was experimentally investigated by direct measurement of refrigerant and cooled fluid temperature at each refrigerant path outlet. As a result, the $\alpha_{r,m}$ in the evaporator with 50 plates were drastically improved as consequence of improvement of mal-distribution on liquid refrigerant flow at the inlet header of the evaporator.

Full data are given in Appendix A2.14.

3.2.5.8 Development of heat exchangers for natural refrigerants

Principal Investigator

Akito Machida

Institution

Research and Development Center, Mayekawa Mfg. Co. Ltd

Summary

Mayekawa has developed refrigeration and air conditioning systems using natural refrigerants, ammonia, carbon dioxide, air, water and hydro carbons. Carbon dioxide is used as a secondary refrigerant in refrigeration and storage applications as well as a primary refrigerant in trans-critical heat pumps. Air is the refrigerant in the air refrigeration cycle with an operation range of -50℃ to -100℃ while water is used as the refrigerant in adsorption cycles. Ammonia and hydro carbons are used in a wide range of refrigeration and air conditioning applications.

In all these systems, heat exchangers play important roles in terms of performance, refrigerant charge, safety and reliability. Use of compact heat exchangers with high

heat transfer performance characteristics not only improves the COP of the system but also reduces refrigerant charge and improves safety and reliability of the system.

Further data are given in Appendix A2.15.

3.2.5.9 Trends in next-generation heat exchangers for heat pump systems

Principal Investigator

Shigeharu Taira

Institution

Daikin Industries., Ltd

Summary

Development of air conditioning products to address environmental and air pollution concerns is now an imperative for the industry. Heat exchangers, which are widely used in room air conditioners, packaged air conditioners, heat pump water heaters and other applications, are one of the most important components and affect the performance, reliability (quality) and cost of the product. In this paper, therefore, trends in the development of heat exchangers by air conditioning manufacturers, focusing on actual cases have been investigated.

Full data are given in Appendix A2.16.

3.2.5.10 Development of a High-Performance Fin-and-Tube Heat Exchanger with Vortex Generator for a Vending Machine

Institution of Principal Investigator & Co investigator

Fuji Electric Advanced Technology, Co. Ltd.
Fuji Electric Retail Systems, Co. Ltd.

Summary

Beverage vending machines are widely used on street corners and offices in Japan. Since such vending machines have a refrigerator and heater to keep beverage cold or hot, energy saving is strongly required as well as refrigerators and air-conditioners. A fin and tube heat exchanger is used for heat exchange between refrigerant and air. A heat exchanger used in beverage vending machines has a problem of condensate and frost on fin in airflow. Delta-wing-vortex generators were developed for heat transfer enhancement with small pressure loss.

Flow and wall temperature field around the vortex generators were visualized using double size model. Heat transfer enhancement mechanism was considered based on the visualized results, and the configuration of turbulence promoter was optimized. Moreover, heat transfer performance and pressure loss for a prototype heat exchanger was measured.

Full data are given in Appendix A2.17.

3.2.5.11 Numerical Analysis of Gas-Liquid Two-Phase Flow

Principal Investigator

Isao Kataoka

Institution

Department of Mechanical Engineering, Osaka University

Summary

Accurate prediction of thermal hydrodynamic behaviors of gas-liquid two-phase flow is quite important in designing heat pump where various gas-liquid two-phase flows appear in many components such as evaporator, condenser, ejector etc. Gas-liquid two-phase shows quite complicated characteristics of flow and heat transfer compared with single-phase flow. Therefore, numerical analysis of two-phase flow is much more difficult than that of single-phase flow. However, recently, knowledge of gas-liquid two-phase flow and technology of numerical analysis has been considerably improved and accurate numerical predictions become possible even using personal computer. In view of these, developments and present status of numerical analyses of gas-liquid two-phase flow were reviewed.

See Appendix A2.18 for further data.

3.2.6 Manufacturing and Materials

3.2.6.1 Development trends in inner-grooved tubes in Japan

Principal Investigator & Co investigator

Mamoru Houfuku

Institution

Hitachi-Cable, Ltd.

Summary

Tubes with internal grooves are used in air conditioners to conserve energy. The demand for high-performance grooved tubes continues to increase in reaction to the new Conservation of Energy law regarding alternative refrigerants in Japan.

In this paper, changes in the production methods of these inner-grooved tubes and the shapes of the inner surface grooves of these tubes in Japan are presented with regards to the tubes used in fin-tube-type heat exchangers.

The full paper is given in Appendix A2.19.1.

3.2.6.2 Development of High Strength Copper Tube

Principal Investigator

Takashi Shirai

Institution

Research & Development Section, Technical Department,
Kobelco & Materials Copper Tube, Ltd.

Summary

Phosphorus deoxidized copper tube (JIS H3300 C1220T) has been widely used in heat exchanger and tubing etc. for air-conditioning and refrigerating units from the past, because of its balanced performance among strength, workability, thermal conductivity, soldering and brazing, and corrosion resistance.

However recent Japanese refrigerating and air-conditioning market is demanding superior performance tube in strength and heat resistance to current C1220 tube. Because of the market strongly desire to prevent going to thicker wall thickness of tube due to alternative refrigerant and to do cost saving using thinner wall thickness of tube. Three kinds of high strength copper tubes are introduced those are MA5J, KHRT and HRS35LT.

Further data are given in Appendix A2.19.2.

3.2.6.3 Corrosion Control Design for Automotive Aluminium Heat Exchangers

Principal Investigator

Kazuhiko Minami

Institution

Showa Denko K.K.

Summary

Aluminium compact heat exchangers are used in automotive air-conditioning systems and include condensers, evaporators, and heater cores. In this report, we explain the outline of corrosion resistant methods for the condenser which is exposed to the severest corrosive environment in the automobile, and then, explain the formation of sacrificial corrosion layer caused by zinc and the effect of zinc coating from the standpoint of zinc diffusion behaviour in aluminium.

See Appendix A2.19.3 for further data.

3.3 Sweden

3.3.1 Participants Details

Sweden has been represented in the Annex by the Royal Institute of Technology, KTH, Department of Energy Technology, Division of Applied Thermodynamics and Refrigeration. The principal investigator has been Professor Björn Palm.

The Division of Applied Thermodynamics and Refrigeration has a long history of research on refrigeration and heat pump systems, heat transfer and heat exchangers. A special topic of interest has been boiling heat transfer. During the last ten years several projects have focused on compact heat exchangers and mini-channel heat transfer. The following is a brief résumé of projects and publications from the Swedish participant organization particularly relevant to the present project:

- More than 50 years ago, Professor Pierre (1956) conducted experiments on flow boiling in horizontal tubes. These measurements resulted in correlations for average heat transfer in complete and incomplete evaporation which have been widely cited and used in the refrigeration and heat pump industry.
- A minichannel water to refrigerant heat exchanger made from rectangular shaped copper tubes stacked into piles with narrow distance between the tubes was described by Granryd (1986). Heat exchangers of this design was manufactured and used as evaporators and condensers by the heat pump manufacturer Thermia during several years.
- Miniature minichannel heat exchangers were investigated by Khodabandeh for the application of cooling of high power electronic components. The heat transfer in these devices was described e.g. in Khodabandeh and Palm (2002, a, b)
- Visualization of single phase flow in plate heat exchangers was reported by Dovic et al. (2000, 2009). The latter paper also includes a method of estimating single phase heat transfer in plate heat exchangers.
- Flow boiling in brazed plate heat exchangers has been investigated and reported by Claesson and co-workers in several papers. A review of this work is found in Palm and Claesson, 2006.
- Heat transfer and pressure drop in single phase and two phase flow in minichannels with diameters from 0,8 to 1,7 mm was investigated and reported by Owhaib and co-workers. The results were among the first confirming that classical correlations for single phase flow still apply for channels in this range (Owhaib and Palm, 2004). Heat transfer in two-phase evaporative flow was reported by Owhaib et al. (2004).

Sweden is a small country, but in spite of this, it has several companies active on the international market in the area of heat pumps and heat exchanger technology. Among the heat pump manufacturers we would like to mention Nibe, Thermia and IVT which are all well known on the European market.

Among the heat exchanger manufacturers the largest of the Swedish companies is Alfa Laval, which is the company introducing plate heat exchangers, originally for single phase applications. Another important company is SWEP, which introduced *brazed* plate heat exchangers, now very popular as evaporators and condensers in all liquid to liquid heat pumping systems. Other companies, Swedish or international with production in Sweden, are Asarums Industri AB, GEA, Aircoil … some of the products from these companies will be presented below.

3.3.2 Projects Offered

As is the case for the activities in many IEA-annexes, the financing of the Swedish activities within the annex has to be found from different national sources, and the initiation of these activities have not been immediately triggered by the initiation of the Annex. On the contrary, the fact that several activities were already ongoing in the area was important for the decision for Sweden to enter the Annex.

The projects described below are thus such that have been running at the laboratory of the laboratory of the Department of Energy Technology at KTH during the time of the Annex.

3.3.2.1 Microchannel heat transfer

Principal Investigator & Co investigator

Professor Björn Palm, Dr. Wahib Owhaib and Ph.D students Claudi Martín-Callizo, Rashid Ali and Hamayun Maqbool.

Institution

Royal Institute of Technology, Department of Energy Technology, Division of Applied Thermodynamics and Refrigeration

Summary

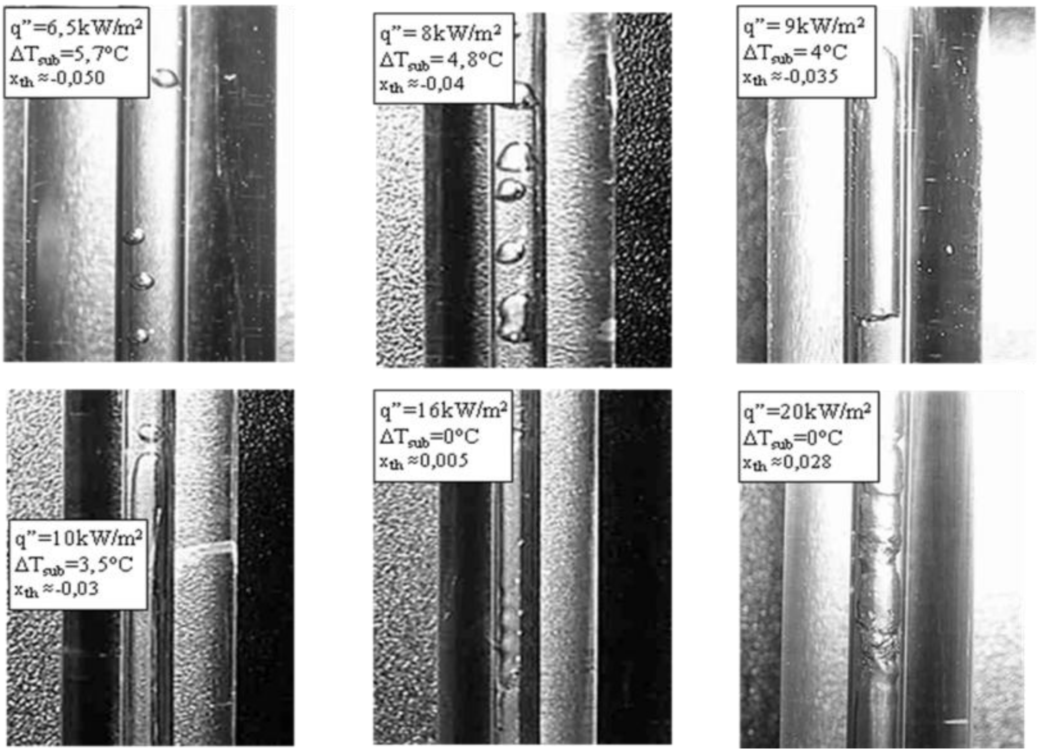

Fig. 3.3.1 Pictures of bubble development during evaporation of R134a in a vertical narrow tube.

Owhaib et al. (2006) reports on an experimental study of saturated flow boiling of R134a inside a circular vertical quartz tube coated with a transparent electrically conducting layer used as a heater – data are shown in Fig. 3.3.1. The inner diameter of the tube was 1.33 mm and the heated length 235.5 mm. The flow pattern at high vapor qualities and the dryout of the liquid film were studied using a high speed CCD camera at the mass fluxes 47.4 and 124.4 kg/m^2 s in up-flow at 6.425 bar. The heat fluxes ranged from 5 to 13.6 kW/m^2 for the lower mass flux and from 20 to 32.4 kW/m^2 for the higher mass flux. The behavior of the flow close to dryout was found to be different at low and high mass flux. At low mass flux the location of the liquid front fluctuated with waves passing high up in the tube. In between the waves, a thin film was formed, slowly evaporating without breaking up. At high mass flux the location of the liquid front was more stable. In this case the liquid film was seen to break up into liquid streams and dry zones on the tube wall.

The article by Martín-Callizo et al. (2007a) also reports on flow boiling visualization of refrigerant R-134a in the same vertical circular channel as above. Flow patterns have been observed along the heated length with the aid of high-speed CMOS digital camera. From the flow boiling visualization, seven distinct two-phase flow patterns have been observed: *Isolated bubbly flow*, *confined bubbly flow*, *slug flow*, *churn flow*, *slug-annular flow*, *annular flow*, and *mist flow*. Two-phase flow pattern observations are presented in the form of flow pattern maps. The effects of the saturation temperature and the inlet subcooling degree on the two-phase flow pattern transitions are elucidated. Finally, the experimental flow pattern map is compared to models developed for conventional sizes as well as to a microscale map for air-water mixtures available in the literature, showing a large discrepancy.

Two-phase flow pattern observations are presented in the form of flow pattern maps. *Annular*-type flow patterns are dominant for x_{th}>0.2.

Increasing the saturation temperature shifts all transition boundaries towards higher vapor qualities. No significant influence of the inlet subcooling degree has been observed on the *slug-annular*/*churn* to *annular flow* (*intermittent* to *non-intermittent*) transition. However, larger inlet subcooling seems to move all other transitions to earlier vapor qualities.

The experimental flow pattern maps are compared to models developed for conventional sizes available in the literature and a microscale map developed for air-water, showing a large discrepancy. On the other hand, the results are consistent with other studies using refrigerants in minichannels.

Subcooled flow boiling heat transfer for refrigerant R-134a in vertical cylindrical tubes with 0.83, 1.22 and 1.70 mm internal diameter was experimentally investigated and reported by Martín-Callizo et al. (2007b). The effects of the heat flux, q” = 1–26 kW/m^2, mass flux, G = 300 – 700 kg/m^2 s, inlet subcooling, $\Delta T_{sub,i}$ = 5–15 °C, system pressure, p = 7.70–10.17 bar, and channel diameter, D, on the subcooled boiling heat transfer were explored in detail. The results are presented in the form of boiling curves and heat transfer coefficients. The boiling curves evidenced the existence of hysteresis when increasing the heat flux until the onset of nucleate boiling, ONB. The wall superheat at ONB was found to be essentially higher than that predicted with correlations for larger tubes. An increase of the mass flux leads, for early subcooled boiling, to an increase in the heat transfer coefficient. However, for fully developed subcooled boiling, increases of the mass flux only result in a slight improvement of the heat transfer. Higher inlet subcooling, higher system pressure and smaller channel diameter lead to better boiling heat transfer. Experimental heat transfer coefficients are compared to predictions from classical correlations available in the literature. None of them predicts the experimental data for all tested conditions.

The prediction of dryout occurrence is crucial in the design and safety of compact heat exchangers. For this purpose, a series of experimental tests have been performed to investigate dryout incipience and critical heat flux (CHF) in a circular vertical microchannel with internal diameter of 640 µm and a uniformly heated length of 213 mm using refrigerants R-134a, R-22 and R-245fa as working fluids. The results were reported by Martín-Callizo et al. (2008). The effects of mass flux, system pressure, and refrigerant on the dryout heat flux and vapour quality are explored in detail. The experimental results show that the heat flux at dryout incipience and the CHF increase with increasing mass flux, but are essentially independent of the system pressure for the range of experimental conditions. The dryout quality is believed to be controlled by the force balance between gas shear and liquid surface tension.Finally, the experimental CHF results are compared to existing correlations both specially developed for microchannels and for macroscale geometries. The comparison shows good agreement with the classical Katto and Ohno (1984) correlation, developed for conventional large tubes. On the other hand, none of the microchannel correlations predicts in agreement the experimental data.

A series of experiments, reported by Ali et al. (2009) were conducted to measure the heat transfer coefficients in a minichannel made of stainless steel (AISI 316) having an internal diameter of 1.7mm and a uniformly heated length of 220mm. R134a was used as working fluid and experiments were performed at two different system pressures corresponding to saturation temperatures of 27 ºC and 32 ºC. Mass flux was varied from 50 kg/m^2 s to 600 kg/m^2 s and heat flux ranged from $2kW/m^2$ to $156kW/m^2$. The test section was heated directly using a DC power supply. The experimental results show that the heat transfer coefficient increases with imposed wall heat flux while mass flux and vapour quality have no considerable effect. Increasing the system pressure slightly enhances the heat transfer coefficient. The heat transfer coefficient is reduced as dryout is reached. It is observed that dry out phenomenon is accompanied with fluctuations in outer wall temperature and a larger standard deviation in wall temperatures.

Pressure drop in two phase flow in minichannels was investigated and reported by Ali et al. (2009). Test tube is made of fused silica having an internal diameter of 781µm with a total length of 261mm and a heated length of 191mm.The external surface of the test tube is coated with an electrically conductive thin layer of ITO (Indium Tin Oxide) for direct heating of test section. Refrigerant R134a was used as the working fluid and mass flux during the experiments from 100 to 650 kg/m^2sec. Experiments were performed at a system pressure of 7.70 bar (corresponding to saturation temperature of 30^oC). Two-phase frictional pressure drop characteristics with different mass flux, vapour fraction and heat flux were explored in detail. Finally, the prediction capability of well known available correlations in the literature, developed for macrochannels and others especially developed for microchannels, was also checked. The homogeneous model predicts the data fairly well with a mean absolute deviation (MAD) of 19% and 69% of data within ±20% error band. The Müller-Steinhagen and Heck (1986) correlation developed for macrochannels predicts the data with a MAD of 19% and 61% of data within ±20% error band. The Mishima and Hibiki (1996) correlation, developed for microchannels, also shows fairly good approximation of the data with MAD of 19% and 57% of data within ±20% error band.

3.3.2.2 A compact water to water heat exchanger

Principal Investigator and Co investigators

Professor Björn Palm, Ph.D student Primal Fernando, Professor Per Lundqvist, Professor Eric Granryd, Professor Tim Ameel

Institution

Royal Institute of Technology, Department of Energy Technology, Division of Applied Thermodynamics and Refrigeration

Brief Summary

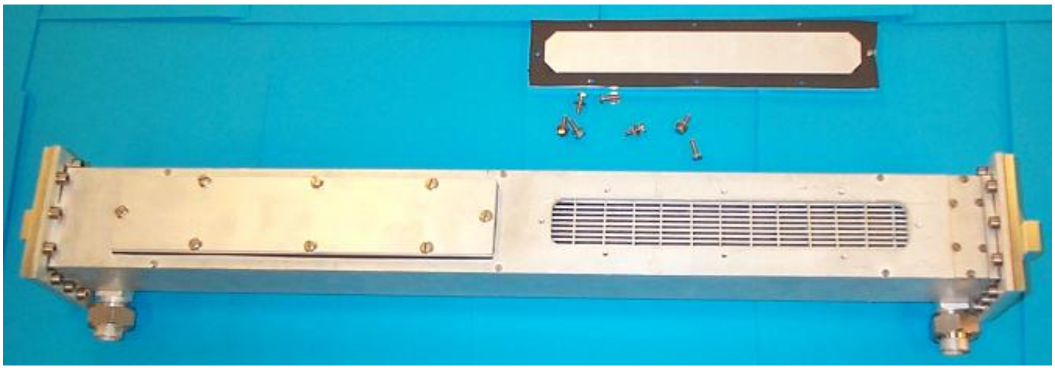

Fig. 3.3.2. Prototype compact water to refrigerant heat exchanger designed from flat, multichannel aluminium tubes. Used as evaporator and condenser.

A project to develop a low-charge heat pump system with propane as refrigerant has been finalized within the last couple of years. The project as a whole is reported by Fernando (2007). As a part of this project, a prototype liquid-to-refrigerant heat exchanger was developed with the aim of minimizing the refrigerant charge in small systems. To allow correct calculation of the refrigerant side heat transfer, the heat exchanger was first tested for liquid-to-liquid (water-to-water) operation in order to determine the single-phase heat transfer performance. These single-phase tests are reported in part one of the triple- paper by Fernando et al. (2008). The heat exchanger was made from extruded multiport aluminium tubes and was designed similar to a shell-and-tube heat exchanger. The heat transfer areas of the shell-side and tube-side were approximately 0.82m^2 and 0.78 m^2, respectively. There were six rectangular-shaped parallel channels in a tube. The hydraulic diameter of the tube-side was 1.42mm and of the shell-side 3.62 mm. Tests were conducted with varying water flow rates, temperature levels and heat fluxes on both the tube and shell sides at Reynolds numbers of approximately 170–6000 on the tube-side and 1000–5000 on the shell-side, respectively. The Wilson plot method was employed to investigate the heat transfer on both the shell and tube sides. In the Reynolds number range of 2300–6000, it was found that the Nusselt numbers agreed with those predicted by the Gnielinski correlation within 5% accuracy. In the Reynolds number range of 170–1200 the Nusselt numbers gradually increased from 2.1 to 3.7. None of the previously reported correlations for laminar flow predicted the Nusselt numbers well in this range. The shell-side Nusselt numbers were found to be considerably higher than those predicted by correlations from the literature.

The second part of the paper presents heat transfer data for the same multiport minichannel heat exchanger, vertically mounted as an evaporator in a test-rig simulating a small water-to-water heat pump. Refrigerant propane with a selected inlet vapour quality flowed upward through the tubes and exited with a superheat of 1–4 K. A temperature-controlled glycol solution that flowed downward on the shell-side supplied the heat for the evaporation of the propane. The heat transfer rate between the glycol solution and propane was controlled by varying the evaporation temperature and propane mass flow rate while the glycol flow rate was fixed (18.50 l/min). Tests were conducted for a range of evaporation temperatures from -15 to +10 °C, heat flux from 2000 to 9000W/m^2 and mass flux from 13 to 66 kg/m^2 s. The heat transfer coefficients were compared with 14 correlations found in the literature. The experimental heat transfer coefficients were higher than those predicted by many of the correlations. A correlation which was previously developed for a very large and long tube (21 mm diameter and 10 m long) was in good agreement with the experimental data (97% of the data within ±30%). Several other correlations were able to predict the data within a reasonable deviation (within ±30%) after some adjustments to the correlations.

The third part of the paper reports on heat transfer results obtained during condensation of propane inside the same minichannel aluminium heat exchanger vertically mounted in an experimental setup simulating a water-to-water heat pump. Propane vapour entered the condenser tubes at the top end and sub-cooled liquid exited from the bottom. Coolant water flowed upward on the shell-side. The heat transfer areas of the tube-side and the shell-side of the condenser were 0.941 m^2 and 0.985 m^2, respectively.

The heat transfer rate between the two fluids was controlled by varying the evaporation temperature while the condensation temperature was fixed. The applied heat transfer rate was within 3900–9500W for all tests. Experiments were performed at constant condensing temperatures of 30°C, 40°C and 50°C, respectively. The cooling water flow rate was maintained at 11.90 l/min for all tests. De-superheating length, two-phase length, sub-cooling length, local heat transfer coefficients and average heat transfer coefficients of the condenser were calculated. The experimental heat transfer coefficients were compared with predictions from correlations found in the literature. The experimental heat transfer coefficients in the different regions were higher than those predicted by the available correlations.

3.3.2.3 Compact plate heat exchanger evaporator with porous surface

Principal Investigator and Co investigator

Professor Björn Palm, Ph.D student Richard Furberg

Institution

Royal Institute of Technology, Department of Energy Technology, Division of Applied Thermodynamics and Refrigeration

Brief Summary

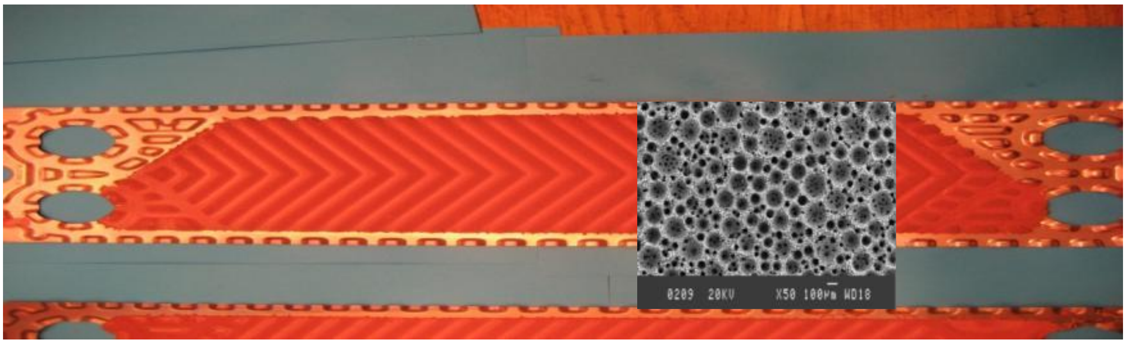

Fig. 3.3.3. Stainless steel plate of plate heat exchanger, covered by porous copper structure. Insert shows microscope photo of surface

A porous surface for enhancing boiling heat transfer has been developed at the Department of Energy Technology together with the Division of Functional Materials, both at KTH. Furberg et al. (2009) describes an experimental study of the performance of a standard plate heat exchanger evaporator, both with and without the novel nano- and microporous copper structure, used to enhance the boiling heat transfer in the refrigerant channel. Various distance frames in the refrigerant channel were also employed to study the influence of the refrigerant mass flux on two-phase flow heat transfer. The tests were conducted at heat fluxes ranging between 4.5 kW/m^2 and 17 kW/m^2 with 134a as refrigerant. Pool boiling tests of the enhancement structure, under similar conditions and at various surface inclination angles, were also performed for reasons of comparison. The plate heat exchanger with the enhancement structure displayed up to ten times enhanced heat transfer coefficient in the refrigerant channel, resulting in an improvement in the overall heat transfer coefficient with over 100%. This significant boiling enhancement is in agreement with previous pool boiling experiments and confirms that the enhancement structure may be used to enhance the performance of plate heat exchangers. A simple superposition model was used to evaluate the results, and it was found that, primarily, the convective boiling mechanism was affected by the distance frames in the standard heat exchanger. On the other hand, with the enhanced boiling structure, variations in hydraulic diameter in the refrigerant channel caused a significant change in the nucleate boiling mechanism, which accounted for the largest effect on the heat transfer performance.

3.3.3 Market Research

One of the aims of this Annex has been to identify compact heat exchangers suitable for use in heat pumping equipment as evaporators or condensers. Especially, new types, or designs used in other industries, which may be suitable for this application have been sought. In this section we will report on some designs fitting to this description. We will also try to highlight some of the latest development trends concerning compact heat exchangers in general. The present section will focus on products from Swedish manufacturers. The products will be grouped based on the manufacturer.

3.3.3.1 Alfa Laval

Alfa Laval is one of the world’s largest manufacturers of heat exchangers, having a large range of products suitable for different applications and for all sizes, from the smallest to the largest. It is therefore only possible here to give a short glimpse of some of the recent developments and products.

Plate heat exchangers of different types have been one of the most important type of products from Alfa Laval. For the application of plate heat exchangers as evaporators, the distribution of refrigerant between the different parallel channels is a well known design problem. Alfa Laval have introduced a special design of the distributor in many of their plate heat exchanger evaporators called Equalancer (see Fig. 3.3.4). This design is said to give a “double mixing” of the refrigerant, and thereby a more even distribution between the parallel channels.

Fig. 3.3.4. "Equalancer" system for distribution of refrigerant into the parallel channels of plate heat exchangers.

Fig. 3.3.5. Left: Brazed plate heat exchanger Alfa Laval CBXP52 designed for pressures up to 70 bar. Right: Plate heat exchanger for transcritical CO_2 systems, pressures up to 167 bar.

Several of Alfa Laval's plate heat exchangers are also available for extra high pressures, for applications with R410A or carbon dioxide. An example is the brazed plate heat exchanger CBXP52 designed for pressures up to 70 bar. A new group of products for transcritical CO2 systems is released September 2009, called AXP10 and AXP14. These are designed for pressures up to 167 and 152 bars respectively.

3.3.3.2 Swep

Swep is a company specialized in brazed plate heat exchangers. Just like Alfa Laval they have introduced new heat exchangers sustaining higher pressures. The model DP is available in different sizes, all available for pressures up to 45 bar, enough for use with R410A. The DP300 series has assymmetrical plates, with narrower refrigeration channels, thereby reducing the hold-up volume by about 30%.

A recent design is the B16DW-U, which, through an external support, is able to withstand pressures up to 140 bar, enough for use in transcritical carbon dioxide systems.

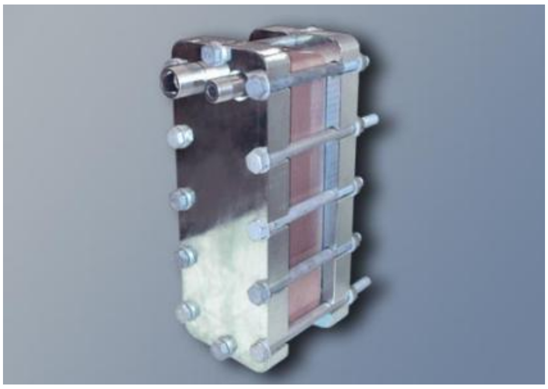

Fig. 3.3.6. Plate heat exchanger B16DW-U from SWEP for pressures up to 140 bar.

3.3.3.3 Tranter

Tranter is a large heat exchanger manufacturer with production mainly in Sweden, USA and India. Most of the products are based on plates as the heat transfer surface, including plate and frame, fully welded and shell and plate heat exchangers. Worth mentioning here is the SuperMax shell and plate heat exchanger, designed for operating pressures up to 70 bar, and thus useful for low temperature carbon dioxide systems.

Fig. 3.3.7. SuperMax shell and plate heat exchanger from Tranter.

3.3.3.4 Multichannel

Multichannel AB is a small company with an interesting plate heat exchanger as one of their main products. This heat exchanger is a combined evaporator-condenser-internal heat exchanger made of one single package of plates. The design is described by

Fig. As shown, the unit is divided vertically so that the hot gas is condensed at one side of the heat exchanger, the liquid is then transferred to the top of the unit through a narrow section in the middle of the unit, also acting as a separator between the hot and the cold side. The high pressure subcooled liquid refrigerant is taken from the top of the unit, through the expansion valve and down to the bottom inlet of the evaporator section. Finally, the superheated refrigerant gas is drawn from the top of the unit to the compressor.

To avoid oil accumulating in the bottom of the unit, a re-circulation line can be connected from the bottom of the unit to the inlet line just after the expansion valve.

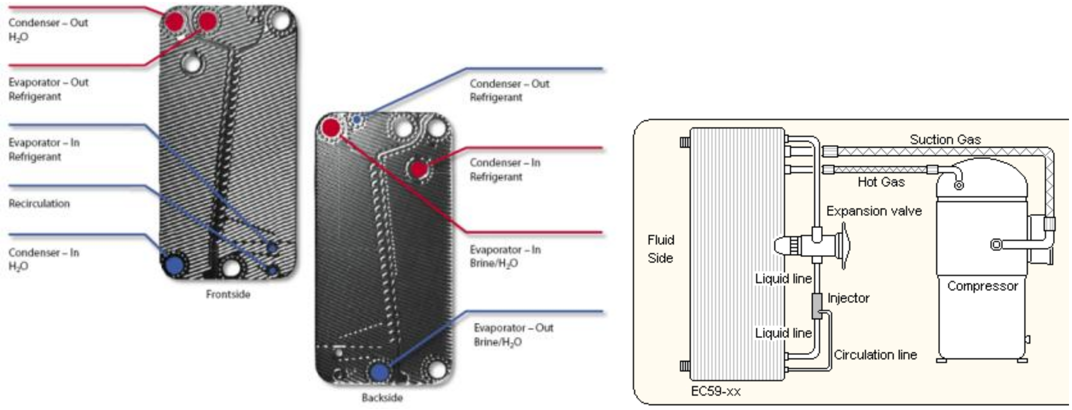

Fig. 3.3.8. Combined evaporator/condenser from Multichannel AB.

3.3.3.5 Laminova

Laminova manufactures a very special heat exchanger mainly used as an oil cooler or intercooler in e.g. high performance cars. The heat exchanger is cylindrical, with the core consisting of an extruded aluminium tube which has been turned so as to get very thin fins with extremely close spacing on the outside. A typical design is shown in Fig. 3.3.9. Efforts have been made to develop the design further to make it possible to use as an evaporator or condenser of heat pumping systems. However, it has not as yet resulted in any products for this application.

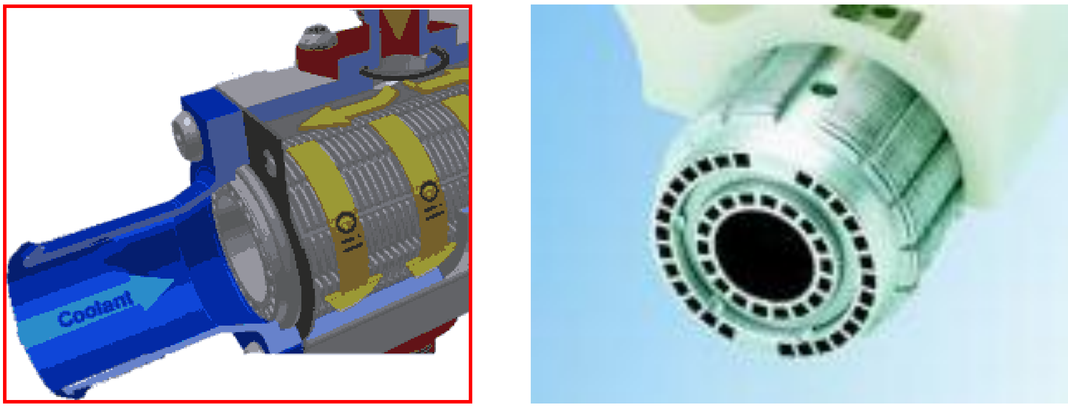

Fig. 3.3.9. Laminova heat exchanger, usually used as oil cooler or intercooler in engines.

3.3.4 Manufacturing techniques

The development of compact heat exchangers is highly dependent on new manufacturing techniques. We would therefore here also like to include some new techniques which have, or may be, used in the development of new compact heat exchangers.

SAPA is a large manufacturer of aluminium and aluminium profiles used for many different applications. The company has a special heat transfer division helping customers with design and production of heat exchangers. Within the company, there are manufacturing units producing extruded multichannel tubes similar to the one shown to the left in Fig. 3.3.10. An alternative method of producing miniature channels from the company's extruded profiles is shown in the right part of Fig. 3.3.10. Two, or several, profiles are designed so that they, when assembled together, form multiple parallel channels with complex geometries. In this case, the profiles are "brazed" with the help of a thin surface layer of the aluminium being doped to lower its melting point.

Fig. 3.3.10. Left: Extruded multichannel aluminium tube. Right: Multichannel tubes from SAPA produced by brazing of extruded profiles.

Minchannels can also be produced from copper strips, as show in Fig. 3.3.11. This particular tube has cross section areas of about 3 mm^2, wall thickness of 0,2 mm and can withstand pressures above 100 bar.

Fig. 3.3.11. Multi-channel tube of copper from Luvata.

3.4 Country Report – United Kingdom

3.4.1 Participants Details

The UK, which was also Operating Agent, was represented by Brunel University, Uxbridge, Middlesex. Professors Tassos Karayiannis and John Lewis represented the University at some meetings, and the co-ordination of activities was subsumed to Professor David Reay, Heriot-Watt University and Dr. Peter Kew, also of Heriot-Watt University.

Dr. Kew represented the UK at all overseas meetings of the Annex.

3.4.2 Projects Offered

The projects offered by the UK spanned three main activities – scientific research, market studies and dissemination/education. Most data are given in Appendix 3, but the activities are summarised below.

3.4.2.1 Boiling and condensation in Microchannels.

Principal Investigator and Co-investigators

Professor Tassos Karayiannis, Brunel University. Co-investigators included, but were not limited to, Prof. David Kenning, Brunel University, Dr. Khellil Sefiane, Edinburgh University, Prof. Anthony Walton, Edinburgh University, Prof. David Reay, David Reay & Associates; Dr. Peter Kew, Heriot-Watt University, Dr. David McNeil, Heriot-Watt University, Assoc. Prof. Yuying Yan, Nottingham University, Prof. John Rose, QMUL, Dr. Huasheng Wang, QMUL, Dr. Ryan McGlen, Thermacore Ltd., Eric Ferguson, Selex Galileo, and representatives of ANSYS, Chart Energy & Chemicals and Wieland-Werke.

Institutions

Brunel University, Heriot-Watt University, Edinburgh University, Nottingham University, QMUL (London University), and 9 companies.

Summary

Evaporation and condensation are very efficient and potentially near-isothermal heat transfer processes. In closed systems, they must go together. They are employed in power and refrigeration systems and for the cooling of devices with high thermal power density. As power ratings are increased and sizes are decreased, multiphase heat transfer in channels of very small cross-section becomes increasingly attractive, compared to single-phase cooling. Multiphase pressure drop imposes a constraint on the lower limit of cross-sectional dimension of ~50 μm and channels in the range of practical interest of 50 –1000 microns are conventionally referred to as **microchannels** (which does not match the conventions of terminology in some other applications of microfluidics). As an example, cooling by boiling in microchannels embedded in the silicon chip is a strong contender for cooling new electronic devices at heat fluxes exceeding $1MW/m^2$. The cooling requirements of microelectronics are becoming a bottleneck for more powerful, sophisticated microchips with increased component densities. The move from bipolar to CMOS and the change in voltage from 5V to a voltage lower than 1.5V have extended the lifetimes of standard cooling systems, at least for low power electronics (PC, phone chip, etc.), but the NEMI and ITRS Roadmaps indicate no slow down in the rate of increase in cooling demands for the next generations of chips. Predictions for 2006 were of peak heat fluxes around 1.6 MW/m^2.

There are savings in cost, size, fluid inventory and safety from employing microchannels in multiphase fluid-fluid heat exchangers for process applications, such as heat pumps and refrigerators. Empirically designed condensers with microchannels have been used for some years in automobile air conditioners, resulting in condensers 4 times smaller and unit efficiencies 10-20% higher than earlier technologies. Rational design will bring further improvements and savings in energy if microchannels can be employed in units of larger capacity.

Because of the complicated nature of multiphase flow, the design of industrial systems employing large channels currently depends on empirical correlations. As the channel cross-section is reduced, surface tension forces increase in importance and in some respects the heat transfer processes become more amenable to modelling. In boiling, the number of bubbles to be considered at any time decreases and flows are more likely to be laminar. In condensation, a significant portion of a microchannel wall is covered by a thin laminar condensate film. Even so, there is still controversy about the flow regimes and mechanisms, particularly in regions of ultra-thin liquid films, and worldwide effort is now being directed to boiling and condensation in microchannels (see for instance the 1st and 2nd ASME International Conferences on Microchannels and Minichannels in 2003 and 2004). Much of the experimental data are for heat transfer and pressure drop averaged over a channel or group of channels, whereas the validation of mechanistic models requires synchronised local measurements of several parameters. The development of reliable models is even more important because of the opportunities for improving heat transfer in microchannels by methods that are not feasible in large channels. Small non-circular cross-sections give rise to strong surface tension forces transverse to the flow direction that create thin liquid films; in some systems, effects associated with the surface condition of the solid wall may be made more reproducible and may be enhanced by microfabricated features. The possibilities are so large that they can only be optimised by using reliable theoretically-based, experimentally validated computer models. Microfabricated instrumentation may be used to obtain localised measurements to validate the models and to open the way to feedback control of the flow conditions in individual channels.

The objectives of this project were to develop models and simulations for flow boiling and condensation, validated against a combination of global and localised experimental data and to investigate several methods of enhancement. The results of the investigation were presented as design methods, together with an appraisal of applications for which particular methods of enhancement are appropriate. Although much of the current interest was on cooling of microelectronics, the application of micro-heat exchangers to heat pumping systems was not overlooked.

The consortium brought to the project experience at an internationally leading level of modelling, global and localised experimental measurements in microchannel boiling and condensation, expertise in the fabrication of micro-instrumentation with outstanding facilities for microfabrication in silicon and expertise in the numerical simulation of multiphase flow with deformable interfaces. A particular strength is the combination of boiling and condensation in one project, leading to cross-fertilisation of ideas relating to processes with shared characteristics.

The studies of boiling and condensation will employ channels with a variety of cross-sectional shapes ranging from rectangles to much more complicated finned arrangements, with dimensions in the range 50 μm – 1 mm. Boiling and condensation both involve flow regimes in which a thin liquid film is present on the wall of the channel and vapour flows in the middle, creating an axial shear stress on the film. Conduction through the liquid film is a significant thermal resistance, which is reduced by making the film thinner. In micro-channels of non-circular cross-section, the thickness is modified by lateral pressure gradients caused by surface tension acting on regions of different interfacial curvature, which draw liquid towards corners. The controlled creation of regions of thin film is of vital importance for the enhancement of condensation. In evaporation, the improvement in heat transfer by film thinning is limited by the breakdown of the film into dry patches. The very high heat fluxes through ultra-thin films create significant non-equilibrium between the liquid interface and the vapour, requiring molecular-level modelling of the Knudsen layer. Continuum modelling also breaks down in the liquid film in the region of the triple contact zone due to Van der Waals forces dependent on the properties of the solid wall. Gradients of surface tension may cause Marangoni flows additional to the flows due to gradients of interfacial curvature. There is worldwide activity on the modelling of the triple contact region but the particular implications for multiphase heat transfer in microchannels are not understood. Improved modelling of the thin-film regime is vital for boiling and condensation and comparison of their experimental heat transfer coefficients may indicate ways of improving the modelling.

Other issues such as the development of flow regimes are more specific to boiling or condensation. In condensation, the thin-film regime develops smoothly from the onset of condensation and is stable on well-wetted surfaces. The issue is to maintain the thinning by surface tension over a significant length through understanding of its dependence on the channel shape, fluid properties and condensation rate. In boiling, there must first be nucleation and growth of confined bubbles, surrounded by thin films and separated by liquid slugs. At high heat fluxes, nucleation of small bubbles may continue in the thin films, further improving heat transfer but affecting the film stability. Film thickness and stability may be influenced by controlling nucleation, either passively by modification of surface microgeometry and wettability or actively by triggering the formation of confined bubbles. The pressure pulse caused by bubble growth interacts with the inlet flow, which may be temporarily reversed. The pressure pulse is modified by interconnections between parallel channels. All the boiling studies will involve localised measurements of wall temperature as a function of time, based on micro-instrumentation and circuitry previously developed at Edinburgh University (Royal Society Grant R36706). Local pressures along channels were measured by a range of methods. Local measurements were synchronised with high-speed imaging.

The strategies for controlling liquid film thickness and stability depend on the mass flow rate through the channel. As virtually all applications involve many parallel channels, uniformity of distribution is a common concern for boiling and condensation. Achieving this is difficult even in single-phase flow; when inlet and outlet connections are formed on the same highly-conductive substrates as the channels, there may also be phase change within the connections that affects distribution and compressibility. Events within parallel channels may also interact by thermal conduction.

All the experimental groups pursued their work on development of analytical models that usually require numerical solution. The numerical modelling group at Nottingham University developed fully-3 D numerical simulations to validate and extend these models. The experimental groups provided Nottingham University with data for comparison with the simulations.

The planned studies are summarised in Table 3.4.1.

Table 3.4.1. Summary of activities.

Univ.	**conditions**	**fluids**	**pressure**
	boiling, electrical heat source		
LSBU	Single channel, variable cross-section, silicon substrate. Passive and active control of inlet flow, bubble nucleation.	R134a, water	1 – 5 bar
EDU	Parallel channels, non-uniform heat flux, silicon substrate. Flow distribution and interactions between channels. Interactions by conduction in silicon substrate.	R134a, water	1-2 bar
HWU	Multi-channels with and without interconnections, metal substrates. Flow regimes, pressure drop, dryout.	R134a, 141b, PFCs	1 – 10 bar
	condensation, water heat sink		
QMUL	Parallel channels (metal extrusions) with optimised microchannel geometry and microchannels with microfins. Superheated and saturated incoming vapour; model fluids with widely different properties.	FC72, H_2O (expt. and theory) R134a, R22, R410a, R152a, CO_2 (theory)	any (theory) near atm (expt.)
	numerical simulation		
NU	Bubble growth in single and cross-connected channels. Thin film heat transfer and flow due to axial stresses and lateral pressure gradients caused by variations in interfacial curvature. Heat transfer across ultra- thin films and liquid-vapour-solid contact zones, including non-equilibrium in Knudsen layer. Conduction between the main heat transfer zone and end connections.	any	any

3.4.3: Market research

The Summary Data presented in Appendix 3 are extracted from documents prepared in support of an analysis of Process Heat Recovery (PHR) in the United Kingdom which was directed at assisting the Carbon Trust to (a) quantify the amount of heat that could be fully recovered in the process industries, (b) identify the technologies that might be use to recover this heat and (c) recommend ways in which industry can be encouraged to increase its take-up of PHR, including, but not limited to, the Enhanced Capital Allowance scheme – a scheme that allows companies to write off the investment in energy-saving equipment, provided that the equipment is 'approved' by government agencies. Industrial heat pumps are not yet in the scheme, but some heat pumps for buildings are – see www.eca.gov.uk/

The work was carried out by David Reay & Associates, in conjunction with AEA Energy & Environment. The aim is of course to substantially reduce carbon emissions.

In this consultation exercise the contractors concentrated on the sectors that are believed to have the greatest potential for PHR – the seven sectors highlighted in bold in Appendix 3.1.

The information given includes the sector total energy use; the estimated sector potential for PHR; the generic heat recovery types that can be used in each sector, including heat pumps which can be related to the nature of the unit operations that are identified in broad terms. The detailed breakdown of PHR opportunities in the seven chosen sectors – now divided into subsectors – are also. The nature of each subsector is given in the Appendix, as well as discussions of the PHR technologies considered.

It is interesting to consider a 'broad brush' approach to PHR using a technology that could be applied widely to a group of unit operations. For example, if heat pumps and MVR systems were applied to all applicable drying/separation processes, which consume 2821 ktoe/annum, energy savings of about 35% might result, saving 987 ktoe/a, or about equal to the TOTAL savings predicted in the more conservative approach, the results of which are detailed here. At present, of course, the manufacturing capability to produce such equipment within a reasonable timescale does not exist.

The data were initially presented at a meeting of HEXAG – the Heat Exchanger Action Group – www.hexag.org held at Newcastle University.

3.4.4: The evaluation of the performance of compact heat exchangers relevant to heat pumps

The activities within this area of the Annex were associated with the EPSRC Project: Boiling and Condensation in Microchannels – see Section 3.4.2.1 above. Papers are available – see Appendix 3 and the Bibliography.

3.4.5 The evaluation of properties and operating limits of such equipment

See Appendix 3 for the Guidance Notes on Compact Heat Exchanger Selection for comprehensive data on compact heat exchangers that might be used in heat pumping systems.

3.4.6 Dissemination and Education

Dissemination activities included publication of meeting proceedings via the project web site – www.compactheatpumps.org (now closed) – talks to the Heat Pump Association and the Heat Exchanger Action Group, and presentations of data at conferences, including the Micro Nano Conference at Brunel University, 2009 and the UK Heat Transfer Conference, QMUL, also in 2009.

A paper is being prepared for the journal *Applied Thermal Engineering*, reporting upon the outcomes of the projects in the Annex.

3.5 Country Report – United States

3.5.1 Participants

The United States contribution to this Annex consisted of six research reports exploring the fundamental phenomena affecting performance of compact heat exchangers. Five of the projects were sponsored by the Air Conditioning Heating and Refrigeration Institute (AHRI) through its research arm, the Air-Conditioning & Refrigeration Technology Institute (ARTI), with financial assistance from the U.S. Department of Energy, the Copper Development Association, the New York State Energy Research and Development Authority, the California Energy Commission, the Refrigeration Service Engineers Society, and the Heating Refrigeration Air-Conditioning Institute of Canada. The sixth was sponsored directly by the US Department of Energy through its Oak Ridge National Laboratory.

3.5.2 Projects Offered

3.5.2.1 High Performance Heat Exchangers for Air Conditioning and Refrigeration Applications (Non-circular Tubes)

Principal Investigator and Co-investigators: A. M. Jacobi, Y. Park, Y. Zhong, G. Michna, and Y. Xia

Institution: Air Conditioning and Refrigeration Center (ACRC), University of Illinois at Urbana-Champaign, Urbana, IL 610801

Summary

A study of flat-tube heat exchangers and their air-side thermal-hydraulic performance under dry-, wet-, and frosted-surface conditions is reported. This multi-pronged research project includes fundamental and applied work. The six chapters and eight appendices present findings that can be generally classified into the following areas, with the following contributions:

• Analysis, modeling and interpretation of air-side thermal-hydraulic data for flat-tube heat exchangers under wet- and frosted-surface conditions.

−We provide a new solution, and mathematical proofs, for conduction in a two-dimensional slab on a one-dimensional fin. The solution is simplified to model-frosted fin performance for flat-tube heat exchangers.

−We provide an analytically rigorous development of a new data-interpretation scheme, based on an intuitive extension of a *UA*-LMTD approach. The development includes a critical assessment and comparison to extant methods.

• Studies of the effects of fin design parameters on the performance of plain, wavy, strip, and louvered fins for flat-tube heat exchangers.

−Analysis and experiments from the literature are used and extended to assess fin design parameters and make design recommendations for plain, wavy, strip and louvered fins.

−New flow visualization is reported to elucidate the role of boundary layer effects on flow through louvered fins, providing guidance toward optimal fin design.

−New convective data are obtained using naphthalene sublimation to explore the high- Reynolds-number behavior of offset-strip fin, flat-tube heat exchangers.

• Studies of the retention and drainage of water from the air-side surface of flat-tube heat exchangers with plain, wavy, strip, and louvered fins.

−A new dynamic dip test method is validated and used to obtain drainage data from plain, wavy, strip, and louvered-fin heat exchangers. An analytical model of drainage successfully predicts the magnitude and trends in these transient data.

−Wind-tunnel experiments are used to quantify the *in situ* retention of condensate on these heat exchangers over a range of operating condition. Ancillary experiments are conducted to examine off-cycle retention.

• Characterization of the thermal-hydraulic performance of plain, wavy, strip, and louvered-fin, flat-tube heat exchangers under dry-, wet-, and frosted-surface conditions.

−Extensive experiments are conducted to quantify the frictional and heat-transfer behavior of plain-, wavy-, strip-, and louvered-fin heat exchangers. Data for each heat exchanger are correlated, and the physical mechanisms affecting performance are discussed.

−Wet-surface performance is characterized in terms of *multipliers*, developed for the Darcy friction factor, *f*, and the Colburn *j*-factor for each heat exchanger.

−Frosted-surface behavior is characterized in terms of frost distribution, accumulation, pressure drop, and thermal resistance.

−The performance of flat-tube heat exchangers is compared to that of round-tube heat exchangers using a Second-Law-based analysis.

−We provide a new correlation that covers a broader parameter space more accurately than any existing louvered-fin performance prediction.

3.5.2.2 High Condensing Temperature Heat Transfer Performance of Low Critical Temperature Refrigerants.

Principal Investigator: Dr. Srinivas Garimella

Institute: George W. Woodruff School of Mechanical Engineering, Georgia Institute of Technology, Atlanta, GA 30332-0405

Summary

A comprehensive study of heat transfer and pressure drop of refrigerant blends R410A and R404A during condensation and supercritical cooling at near-critical pressures was conducted. Local heat transfer coefficients and pressure drops were obtained at five nominal pressures: 0.8, 0.9, 1.0, 1.1 and 1.2 ×Pcrit. Refrigerant R410A was tested in commercially available horizontal smooth tubes of 9.4 and 6.2 mm I.D., whereas R404A was tested in a 9.4 mm I.D. tube. Heat transfer coefficients were measured using a thermal amplification technique that measures heat duty accurately while also providing refrigerant heat transfer coefficients with low uncertainties.

For condensation tests, local heat transfer coefficients and pressure drops were measured for the mass flux range 200 < G < 800 kg/m2-s in small quality increments over the entire vapor-liquid region. For supercritical tests, local heat transfer coefficients and pressure drops were measured for the same mass flux range as in the condensation tests for temperatures ranging from 30 – 110oC. The average heat transfer coefficients uncertainties in this study were 10%, while average pressure drop uncertainties were 2% except at extremely low mass fluxes
(G = 200 kg/m2-s). For both phase-change condensation and supercritical cooling, frictional pressure gradients were calculated by removing the contraction and expansion component at the ends of the test section and the deceleration component due to momentum change from the measured pressure gradients.

The condensation tests showed increasing heat transfer coefficients and pressure drops with quality and mass flux. The effect of reduced pressure on heat transfer coefficients is not very significant, while this effect is more pronounced in the pressure gradient, with pressure drops being lower at higher reduced pressures. The data from this study were compared with the available heat transfer and pressure drop models in the literature for similar situations, and explanations for agreements/disagreements were provided. In general, no correlation from the literature was successful in predicting heat transfer or pressure drop under the present conditions with an acceptable degree of accuracy, primarily because the available correlations were developed for different geometries, fluids, or operating conditions.

In the absence of other applicable flow regime transition criteria, the criteria developed by Coleman and Garimella (2003) for condensation of R134a were used to designate the prevailing flow regimes for a given combination of mass flux and quality. The condensation data collected in the present study were primarily in the wavy and annular flow regimes. Thus, wavy flow and annular flow heat transfer models were developed. A transition region was then defined to provide a smooth transition between the wavy and annular flow models and thereby remove discontinuities in the overall predictions. To predict the condensation pressure drop data, a modified form of the Friedel (1979) two phase multiplier model was developed in this study. The average absolute deviation of the entire data set from the heat transfer model predictions was 6%, with 95% of the data predicted within ±15%. The pressure drop model resulted in an average deviation of 7%, with 90% of the data predicted within ±15%.

In supercritical cooling, the sharp variations in the thermophysical properties in the vicinity of the critical point were found to have substantial effect on heat transfer coefficients and pressure drops. Due to these abrupt property variations, the heat transfer coefficients show sharp peaks while pressure drops show sudden jumps near the critical temperature. Based on the variations of the specific work of thermal expansion (contraction) for each fluid, the data from the supercritical tests were grouped into three regimes: liquid-like, pseudo-critical transition and gas-like.

Supercritical heat transfer coefficient and pressure drop correlations for each regime were developed. The average absolute deviation of the entire data set from the heat transfer model predictions was 14%, with 83% of the data predicted within ±25%. The pressure drop model resulted in an average deviation of 15%, with 87% of the data predicted within ±25%.

3.5.2.3 Near-critical/Supercritical Heat Transfer Measurements of R-410A in a Small Diameter Tube.

Principal Investigator: Dr. Srinivas Garimella

Institution: George W. Woodruff School of Mechanical Engineering, Georgia Institute of Technology, Atlanta, GA 30332-0405

Summary

A study of heat transfer and pressure drop of zero ozone-depletion-potential (ODP) refrigerant blends in small diameter tubes was conducted. The refrigerant blend R410A (equal mass fractions of R32 and R125) has zero ODP and can replace R22 in many applications. Its use requires new equipment designs and it is not a direct substitute, but is of interest in high-temperature-lift space-conditioning and water heating applications.

Smaller tubes lead to higher heat transfer coefficients and are better suited for high operating pressures. Heat transfer coefficients and pressure drops for R410A were determined experimentally during condensation across the entire vapor-liquid dome at 0.8, 0.9×$P_{critical}$ and gas cooling at 1.0, 1.1, 1.2×$P_{critical}$ in three different round tubes (D = 3.05, 1.52, 0.76 mm) over a mass flux range of 200 < G < 800 kg/m^2-s. A thermal amplification technique was used to accurately determine the heat duty for condensation in small quality increments or supercritical cooling across small temperature changes while ensuring low uncertainties in the refrigerant heat transfer coefficients.

The data from this study were used in conjunction with data obtained under similar operating conditions for refrigerants R404A and R410A in tubes of diameter 6.22 and 9.40 mm to develop models to predict heat transfer and pressure drop in tubes with diameters ranging from 0.76 to 9.40 mm during condensation. Similarly, in the supercritical states, heat transfer and pressure drop models were developed to account for the sharp variations in the thermophysical properties near the critical point.
The physical understanding and models resulting from this investigation provide the information necessary for designing and optimizing new components that utilize R410A for air-conditioning and heat pumping applications.

3.5.2.4 Novel materials for Heat Exchangers.

Principal Investigator and Co-investigators: Anthony M. Jacobi (PI), Xiaohong Han, Young-Gil Park, Andrew Sommers, Christophe T'Joen, Qin Wang

Institution: Air Conditioning and Refrigeration Center, University of Illinois at Urbana-Champaign, Urbana, IL 61801

Summary

Heat exchangers are important to the overall efficiency, cost, and compactness of air-conditioning, refrigeration, and energy-recovery systems. Current heat exchanger designs rely heavily upon copper and aluminum constructions of fin-and-tube or plate

heat exchangers. However, recent advances in polymers, metal and carbon foams, lattice structures and other materials open opportunities for novel heat exchanger designs, exploiting the properties of these new materials. Although some research has been reported on using these materials for heat exchangers in other applications, there has not been a comprehensive study of the use of these emerging materials in conventional HVAC&R systems.

The overarching objective of this study is to identify new materials that hold promise for use in heat exchangers and to assess their potential benefits and feasibility for application in HVAC&R systems. Supporting this objective are six specific research tasks: (A) a literature review to identify promising new materials for heat exchangers in HVAC&R systems; (B) a critical evaluation of the potential benefits of using these material; (C) a study of the best ways to exploit the properties of these new materials; (D) an assessment of the feasibility of implementing these materials in heat exchangers; (E) a study of cost and performance benefits of using new materials; and (F) recommendations for further research on this topic.

In order to identify novel materials that are most promising for heat exchangers in HVAC&R applications, a comprehensive literature review is conducted. Ideas are collected from a wide span of industry and applications in the technical literature including journal papers, conference proceedings, reports, patents, and online documents. Over 500 technical articles are collected, organized, categorized, and the germane work is reviewed in detail. We explore the use of polymers, metals, carbonaceous materials, and ceramics, and all of these materials in composite forms.

The thermal and mechanical properties of individual materials are collected in tabular form; however, thermal-hydraulic performance data are found to be limited. Heat exchanger designs are explored, considering the replacement of materials in existing designs and use of new material with dramatic changes in heat exchanger configuration. Practical issues related to implementing the designs, including manufacturing issues, are considered.

Component simulations are used to compare to the performance of conventional metallic heat exchangers to new designs. The simulations show that in some applications polymeric heat exchangers can surpass metallic counterparts in weight, with the potential for attendant cost savings. Likewise, for some applications high porosity metal foams are also shown to hold excellent promise for use as air-side surfaces.

The recommendations identify directions unlikely to be useful as well as research directions with promise. The focus of the recommendations is on the development and testing of prototype heat exchangers similar to the designs analyzed in this study.

3.5.2.5 Modeling and Analysis of a Heat Exchanger with Carbon-fibre Fin Structures.

Principal Investigator and Co-investigators: D. G. Walker[a], E. A. Vineyard[b] R. Linkous[b] J. Hemrick[b]

Institutions: [a] Department of Mechanical Engineering, Vanderbilt University, Nashville, TN 37235 and [b] Oak Ridge National Laboratory, Oak Ridge TN, 37831

Summary.

A novel heat exchanger design utilizing a high-conductivity carbon fibre weave as fins was analyzed for heat transfer capacity. The design is a 12-pass copper tube with carbon fibre material perpendicular to and between the tubes. The model is an equivalent resistance model that incorporates lateral conduction to describe the heat transfer between tubes via the fibre. Results of the analysis suggest that high conductivity fibres can indeed improve the heat transfer. However, for extremely large conductivities, the performance will degrade due to parasitic heat losses. Further- more, the contact resistance between the fibre and the tube govern the performance more than the fibre conductivity.

A prototype was constructed, installed in a frame and tested in a wind tunnel over a range of air flow rates and water and air inlet conditions. Experimental results were analyzed using an NTU model, and showed that contact resistance was the limiting factor. Since optimal circuiting depends on both contact resistance and fiber conductivity, the results indicated that the next prototype should await a more detailed investigation of contact mechanisms and employ an alternative weave configuration to reduce pressure drop.

3.5.2.6 Void Fraction Measurement and Modelling for Condensing Refrigerant Flows in Small Diameter Tubes.

Principal Investigator: Timothy A. Shedd

Institution: Department of Mechanical Engineering, University of Wisconsin, Madison, WI 53706-1572

Summary

The understanding and knowledge of two-phase pressure drop and void fraction of refrigerants inside minichannels is important due to the fact that a better understanding would lead to improved accuracy in the design of HVAC and refrigeration systems. A novel flow facility has been fabricated that allows for the visualization and measurement of two-phase flow of R410A in a minichannel. This facility allows for very precise control of the operating conditions such that the saturation temperature in the test section is maintained to within ±0.5°C for flow conditions from 0 to 100% quality and mass fluxes ranging from 200 to 800 kg m^{-2} s^{-1} in a 3 mm and 1 mm nominal I.D. tube at 30 °C and 50°C.

Two-phase frictional pressure drop was measured for mass fluxes ranging from 200 to 800 kg m^{-2} s^{-1} at 30 and 50 °C for 3 mm and 1 mm inner diameter stainless steel tubes for vapor qualities from $0 \le x \le 1$. Empirical data obtained were compared to a large number of widely known two-phase frictional pressure drop models, but those proposed by Zhang and Webb (2001), Schubring and Shedd (2008), and the widely

known correlation by Müller-Steinhagen and Heck (1986) performed significantly better than the others in the 3 mm I.D. channel with an averaged (for all conditions) mean absolute error of 18.93 %, 40.06 %, and 22.70 % respectively.

In the case of the 1 mm I.D. tube, the correlations by Zhang and Webb (2001), Müller-Steinhagen and Heck (1986), and Mishima and Hibiki (1996) performed better than the others with an averaged mean absolute error of 27.6 %, 22.79 %, and 21.141 % respectively. Therefore, it can be said that the above models for pressure drop can be used with some confidence to predict two-phase frictional pressure drop for R410A in round tubes of nominal inner diameters of 3 mm and 1 mm for mass fluxes ranging from 200 to 800 kg m^{-2} s^{-1} at 30 °C and 50°C.

A new calibration procedure to measure void fraction by means of capacitance using the ring-type sensor by forcing homogeneous flow to occur in a rigid flow loop was proposed. In homogeneous flow, the slip between the vapor and the liquid goes to zero, and the void fraction can be found from knowledge of the volume occupied by each phase in the loop.

A single relationship between a normalized capacitance of the sensor and void fraction was found using this technique. Void fraction data for R410A were obtained from this calibration in round tubes with diameters of 3 mm and 1 mm for mass fluxes ranging from 200 to 800 kg m^{-2} s^{-1} at 30 °C and 50 °C.

The Rouhani (1969) model remains as one of the best models for the prediction of void fraction. In addition, as a good estimate, particularly at lower qualities (below x = 0.20, for instance) the homogeneous model provides a reasonable estimate. The mean absolute error for the homogeneous model for the 1 mm void fraction data was found to be 18.18%, while the MAE for the Rouhani (1969) model was 12.09%. The overall mean absolute errors for the homogenous model and Rouhani (1969) model for the 3 mm tube were 13.43 and 8.13 % respectively. This is shown in Figure 1. This also provides further support for the use of the new calibration for the ring type capacitance sensors proposed in this work.

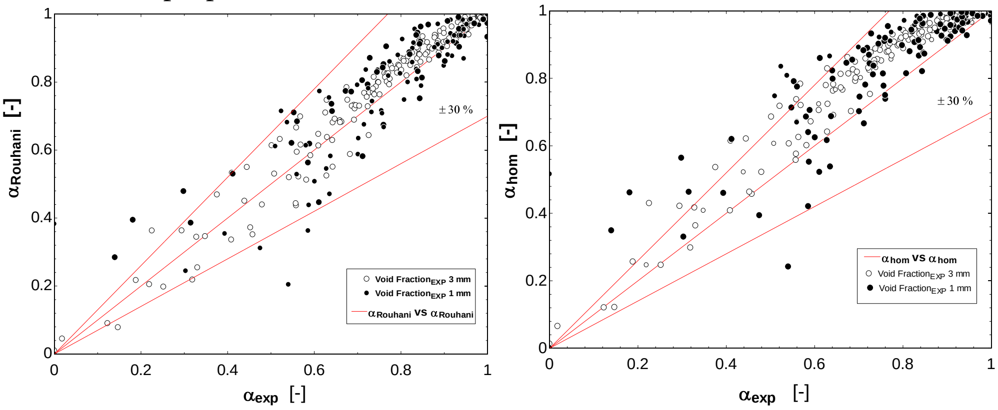

Figure 1- Experimental Void Fraction vs. Rouhani Model (left) and the Homogeneous model (Right)

Key findings of this study to date are:

- Existing pressure drop models are adequate, though not excellent, in predicting the two-phase flow of R410A through 1 mm and 3 mm tubes
- Void fraction data could be accurately obtained using a ring capacitance sensor and time averaging
- The void fraction could be adequately modeled using the Rouhani-Axelsson correlation
- Pressure drop data were no better correlated with void fraction than with quality.
- Pressure drop data were not strongly correlated to observed flow regimes, nor with the Thome, et al. flow regime map
- Slip ratio data were somewhat inconclusive as to their correlation with pressure drop

Work is ongoing to extend this study to 0.5 mm tubes.

3.5.3 Market research

Since these projects were designed to explore basic concepts in a research laboratory, no market research was conducted. However the projects were motivated by market research conducted by the industrial project sponsors, which indicated a need for an expanded and accessible body of detailed knowledge about factors affecting manufacturing and performance of compact heat exchangers.

3.5.4 The evaluation of the performance of compact heat exchangers relevant to heat pumps

One of the projects addressed overall heat exchanger performance, while the other five focused on the underlying factors affecting compact heat exchanger performance. Performance data of three types were acquired in laboratory experiments: heat transfer and pressure drop on the heat exchangers' air- and refrigerant sides; and on the refrigerant side the void fraction was measured because of its substantial influence on charge requirements, heat transfer and pressure drop. Data were then analyzed to construct broadly applicable correlations for use by designers of compact heat exchangers.

3.5.5 The evaluation of properties and operating limits of such equipment

All of the experiments spanned the full range of operating conditions generally encountered in heat pump applications. Several were focused particularly on the region near the critical temperature and pressure of commonly used refrigerants, since key properties that affect performance can vary abruptly at operating conditions near the critical point.

3.5.6 Dissemination and education

All five projects yielded project summaries and final reports that the sponsors are making available. These are posted in Report form on the HPC web site.

4. Conclusions

The outcomes of Annex 33, which was concerned with compact heat exchangers (CHEs) in heat pumping equipment have been many, quite diverse in their nature and comprehensive.

The objective of this Annex was to present a compilation of possible options for compact heat exchangers, used as evaporators, condensers and in other roles in heat pumping equipment. The aim of the work was to highlight technologies and techniques to minimise the direct and indirect effect on the local and global environment due to operation of, and ultimate disposal of, the equipment.

The Annex involved five countries – Austria, Japan, Sweden, the United States and the United Kingdom, the latter acting additionally as Operating Agent. The Annex ran for three years, the final Annex meeting being held in the UK in September 2009.

The outcomes of the Annex consist of a wide variety of data ranging from fundamental research on boiling in narrow channels to guidelines for selecting and using CHEs in heat pumping systems. There are considerable market data available within the Report and the cited references, and a number of novel heat exchanger concepts, including the use of new materials and the application of process intensification methods, should allow equipment manufacturers in the future to achieve the Annex aim.

Particular aspects that it is considered worth highlighting in the Conclusions are:

1. The increasing interest in and use of, CO_2 as a working fluid. This has interesting implications in terms of the equipment used and the concepts for heat pumping that might be applied – see particularly the inputs from Austria and Japan.
2. The growing market for domestic heat pumps, where efficiency, arising in part out of the increased use of CHEs, is critical to further sustained market growth, particularly in countries where heat pump use has been slow to materialize.
3. The vast portfolio of research on heat transfer and fluid dynamics in narrow channels in CHEs. The research highlighted in Sweden, Japan and the USA are of particular note.
4. The role heat pumps could play in industry, where reduced payback times could be aided by CHEs. The UK study highlights the market possibilities.
5. There is a need to educate the heat pump industry in the use of CHEs, their merits and limitations, and the types that are available. The use of new materials, as indicated in some of the research in the USA, could reveal new opportunities.

The project has brought together many experts in the heat pump/CHE field and the Annex Report will, it is believed, be a major and constructive source of data for those interested in using CHEs in heat pumping equipment.

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(NOTE – The references and Bibliography include all papers cited by the Annex members and/or listed as Relevant Publications, except for those cited in the US Section, which will appear in the Reports on the HPC web site).

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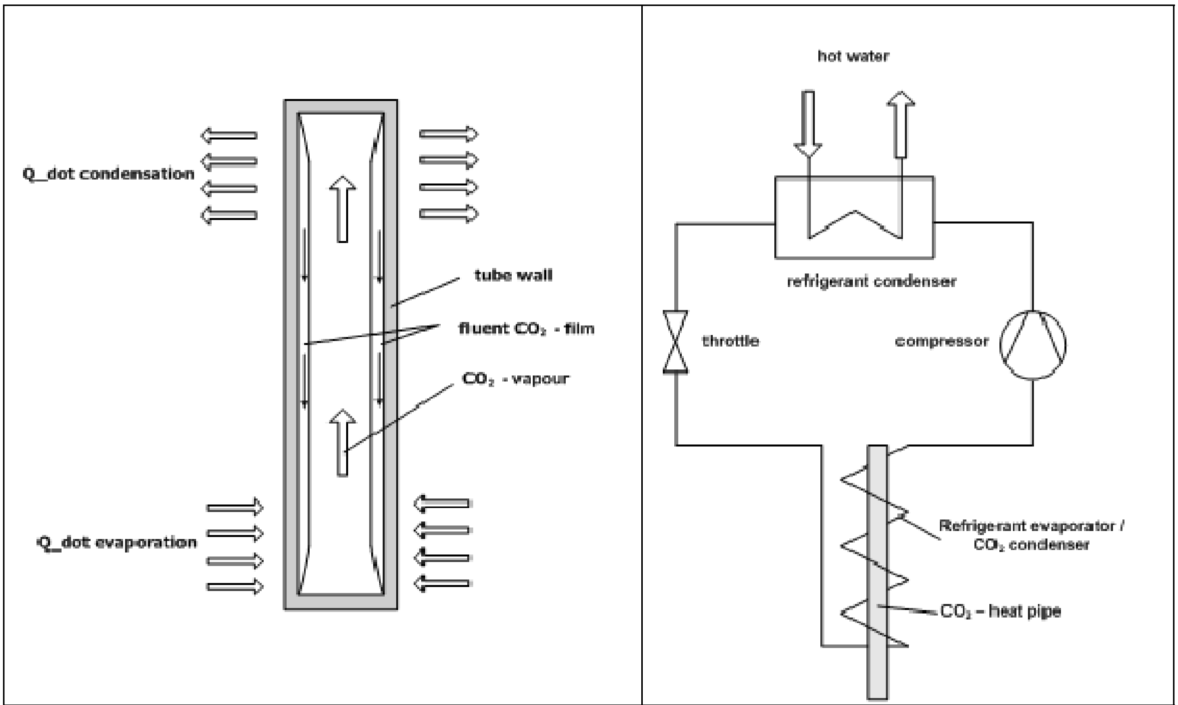
APPENDIX 1 Austria – Scientific Contributions

Technological development of heat pumps using CO_2 heat pipes in Austria

The CO_2 heat pipe can be best described as a vertical collector, made of copper, with a diameter of 15 cm and a depth of about 70 – 100m. This vertical collector is filled with CO_2 at a pressure of 45 bar. The gas is circulating and flows from the top to the bottom and evaporates. The vapour rises in the middle of the pipe and condenses at the evaporator, thus transferring the thermal energy from the soil to the heat pump. The heat pump transfers the temperature level of the evaporator to the desired temperature level. The condensed CO_2 flows back to the bottom thus closing the circle. The soil temperature regenerates through absorbed solar radiation, rain- and groundwater and, in deeper layers, through geothermal heat. As the CO_2 circulates on its own, no energy for pumps is needed. CO_2 heat pipes can't be used for cooling applications. Fig. A1.1a shows the functionality of a CO_2 heat pipe, while Fig. A1.1b depicts the flow chart of the refrigerant circle.



a) b)

Fig. A1.1 a) Functionality of the CO_2 heat pipe; b) Flow chart of the refrigerant circle; (M-TEC, 2008)

More than 500 projects with CO_2 heat pipes have been realised in Austria so far with SPFs up to 5.5. Due to the rise in price of copper pipes, corrugated stainless steel pipes are being used as heat pipes (Faninger, 2007).

A1.1 Method of analysis

Thermodynamic characterization of a heat pump

Fig. A1.2 shows T-s diagram of a heat pump cycle with parameters of interest. All parameters are obtained from Table 6, except T2 and T5.

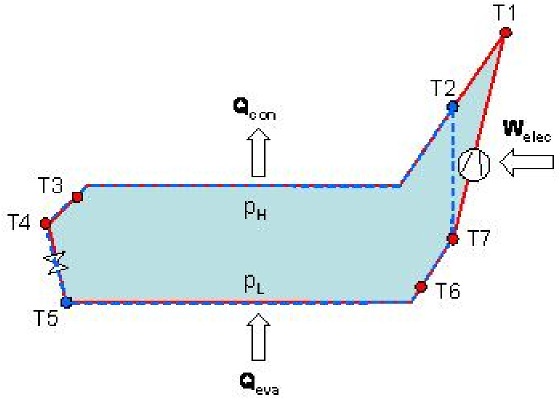


Fig. A1.2. T-s diagram of a heat pump cycle

Thermodynamic state and properties at each point in the figure are determined with the measured temperature and pressure at the point except point 2 and 5. Point 2 and 5 are determined assuming isentropic and isenthalpic processes from point 7 and 4, respectively. For example, T_2 and T_5 are determined by

$$T_2 = T(p_H, s_2) \quad (1)$$

where pH is measured discharge pressure (p1 in Table 6 – see main Report) and entropy is assumed the same for point 2 and 7, i.e. s2=s7, and

$$T_5 = T(p_L, h_5) \quad (2)$$

where p_L is the measured suction pressure (p_7 in Table 6, if p_6 is not available) and enthalpy is assumed the same for point 4 and 5, i.e. h_5=h_4.

With all the thermodynamic states determined, a heat pump is characterized with the following parameters.

Electrical COP of a heat pump is calculated using

$$COP_{el} = \frac{Q_{con}}{W_{el}} \quad (3)$$

where heating power Q_{con} and power consumption W_{el} are readily available from Table 6.

Isentropic efficiency of a compressor is calculated with

$$\eta_{is} = \frac{h_2 - h_7}{h_1 - h_7} \quad (4)$$

Second law efficiency of the heat pump is calculated with

$$\eta_{2nd} = \frac{COP_{el}}{COP_{Carnot}} \tag{5}$$

where Carnot cycle efficiency COP_{Carnot} is defined by

$$COP_{Carnot} = \frac{T_{\text{sink}}}{T_{\text{sink}} - T_{\text{source}}} \tag{6}$$

where sink temperature T_{sink} is an arithmetic average of the secondary temperatures in a condenser, i.e. $T_{sink}=(T_8+T_9)/2$ and source temperature T_{source} is that of an evaporator, i.e. $T_{source}=(T_{10}+T_{11})/2$.

Temperature lift is a parameter frequently used in heat pump analysis. Temperature lift ΔT_{Lift} in this study is defined by

$$\Delta T_{lift} = T_{bub,con} - T_{bub,eva} \tag{7}$$

where $T_{bub,con}$ is the bubble temperature of the refrigerant in the condenser which is calculated with measured pressure as

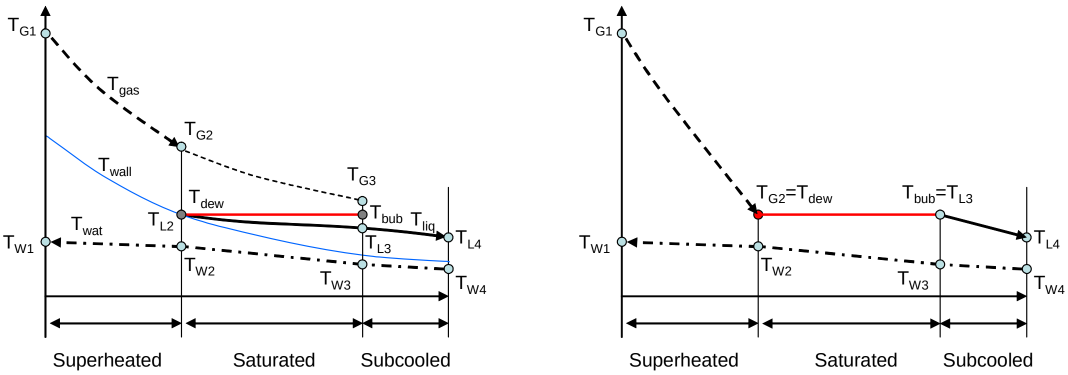
$$T_{bub,con} = T_{bub}(p_{\text{H}}) \tag{8}$$

and similarly $T_{bub,eva}$ is that of an evaporator calculated as

$$T_{bub,eva} = T_{bub}(p_{\text{L}}) \tag{9}$$

Characterization of heat exchangers

The condenser and the evaporator in a heat pump are characterized as follows. Fig. A1.3 shows two temperature profiles in a condenser, actual and simplified. The actual profiles in Fig. A1.3a are complicated from the fact that a condenser has three distinctive heat transfer regions namely, the 'superheated' region where superheated gas undergoes sensible cooling, the 'saturated' region where condensation takes place and the 'subcooled' region where the condensate is further cooled down in the abovementioned order in flow direction. Analysis of this heat exchanger is even more complicated considering that bulk gas is not really in equilibrium with condensate in the saturated region. For this reason, the temperature profiles are simplified to those shown in Fig. A1.3b.



(a) Actual profiles (b) Simplified

Fig. A1.3. Temperature profiles in a condenser

In Fig. A1.3b, it is assumed that bulk gas is in equilibrium with condensate at the onset of condensation. Resultantly, the bulk gas temperature T_{G2} at the onset of condensation is assumed equal to dew temperature for the measured pressure, i.e. $T_{G2}=T_{dew}$. It is also assumed that the bulk condensate temperature at the inlet of sub-cooled section T_{L3} is equal to the bubble temperature, i.e. $T_{L3}=T_{bub}$.

Corresponding secondary temperatures T_{W2} and T_{W3} are calculated as follows.

From Table 6, primary and secondary temperatures at both ends of the condenser, i.e. T_{G1} (T_1 in Table 6), T_{L4} (T_3), T_{W1} (T_9) and T_{W4} (T_8) are given. Since refrigerant mass flow rate m_{ref} can be estimated by

$$m_{ref} = \frac{Q_{con}}{h_1 - h_3} \qquad (10)$$

and therefore T_{W2} can be estimated by

$$T_{W2} = T_{W1} - \frac{Q_{\sup}}{m_8 Cp_8} \qquad (11)$$

where sensible heat of superheat Q_{sup} is

$$Q_{\sup} = m_{ref} Cp_{gas}\left(T_{G1} - T_{dew}\right) \qquad (12)$$

and similarly T_{W3} is estimated by

$$T_{W3} = T_{W4} + \frac{Q_{\text{sub}}}{m_8 Cp_8} \tag{13}$$

where sensible heat of sub-cooling Q_{sub} is

$$Q_{sub} = m_{ref} Cp_{liq} \left(T_{bub} - T_{L4}\right) \tag{14}$$

Then, the UA value can be estimated for each region as follows. For the superheated region, UA is given by

$$UA_{\text{sup}} = Q_{\text{sup}} / LMTD_{\text{sup}} \tag{15}$$

where

$$LMTD_{\text{sup}} = \frac{\left(T_{G1} - T_{W1}\right) - \left(T_{G2} - T_{W2}\right)}{\ln\left[\left(T_{G1} - T_{W1}\right) / \left(T_{G2} - T_{W2}\right)\right]}. \tag{16}$$

For the sub-cooled region, UA is given by

$$UA_{\text{sub}} = Q_{\text{sub}} / LMTD_{\text{sub}} \tag{17}$$

where

$$LMTD_{\text{sub}} = \frac{\left(T_{L3} - T_{W3}\right) - \left(T_{L4} - T_{W4}\right)}{\ln\left[\left(T_{L3} - T_{W3}\right) / \left(T_{L4} - T_{W4}\right)\right]}. \tag{18}$$

For the saturated region, UA is given by

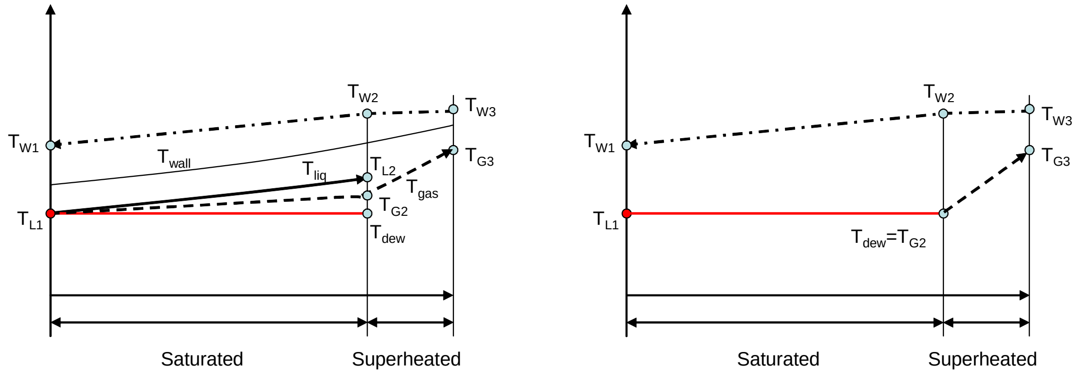
$$UA_{\text{sat}} = Q_{\text{sat}} / LMTD_{\text{sat}} \tag{19}$$

where Q_{sat}=Q_{con}-Q_{sup}-Q_{sub} and

$$LMTD_{\text{sat}} = \frac{\left(T_{G2} - T_{W2}\right) - \left(T_{L3} - T_{W3}\right)}{\ln\left[\left(T_{G2} - T_{W2}\right) / \left(T_{L3} - T_{W3}\right)\right]}. \tag{20}$$

Note that pressure drop is neglected in the analysis. It is assumed that pressure inside the condenser is constant at the measured discharge pressure.

The evaporator is simplified similarly. Fig. A1.4 shows temperature profiles in an evaporator. Two-phase refrigerant enters from left and liquid evaporates in saturated region and gas is heated in superheated region. It is assumed that bulk gas temperature at the inlet of superheated region is equal to the dew temperature for the measured pressure, i.e. $T_{G2}=T_{dew}$.



(a) Real profiles (b) Simplified

Fig. A1.4. Temperature profiles in an evaporator

Secondary temperatures at both ends of the evaporator, i.e. T_{W1} (T_{11} in Table 6) and T_{W3} (T_{10}) are given iin Table 6. The bulk gas temperature at the outlet, i.e. T_{G3} (T_6) is also available from Table 6, but T_{L1} (T_5) is calculated from Eq. (2) because it is either inaccurate or not available.

Similar to the condenser, T_{W2} is calculated by

$$T_{W2} = T_{W3} - \frac{Q_{\sup}}{m_{10} Cp_{10}} \tag{21}$$

where sensible heat of superheat Q_{sup} is

$$Q_{\sup} = m_{ref} Cp_{gas} \left(T_{G3} - T_{dew} \right). \tag{22}$$

The UA value of each region is estimated as follows.

$$UA_{\sup} = Q_{\sup} / LMTD_{\sup} \tag{23}$$

where

$$LMTD_{\sup} = \frac{(T_{W2} - T_{G2}) - (T_{W3} - T_{G3})}{\ln\left[(T_{W2} - T_{G2}) / (T_{W3} - T_{G3}) \right]}. \tag{24}$$

For saturated region, UA is given by

$$UA_{sat} = Q_{sat} / LMTD_{sat} \tag{25}$$

where $Q_{sat}=Q_{eva}-Q_{sup}$ and

$$LMTD_{sat} = \frac{(T_{W1} - T_{L1}) - (T_{W2} - T_{G2})}{\ln\left[(T_{W1} - T_{L1})/(T_{W2} - T_{G2})\right]}. \tag{26}$$

Note that pressure drop is neglected in the analysis. It is assumed that pressure inside the evaporator is constant at the measured suction pressure. For this reason and because they are not plate type, the evaporators in direct expansion systems are not considered.

A1.2 Results and discussion

Before presenting the results, it would be helpful to discuss some fundamental aspects of a heat pump.

Noting $Q_{con}=Q_{eva}+W_{el}$, the COP_{el} defined in Eq. (3) may be rewritten as

$$COP_{el} = 1 + \frac{Q_{eva}}{W_{el}} \tag{27}$$

One may expect that cooling power Q_{eva} would be proportional to m_{ref} but inversely proportional to compression ratio or temperature lift ΔT_{lift} due to the flashing loss in an expansion device. That is

$$Q_{eva} = c_1 \times m_{ref} h^{fg} \times \left(\Delta T_{lift}\right)^{m_1} \tag{28}$$

On the other hand, one may expect that the power consumption W_{el} would be proportional to both m_{ref} and ΔT_{lift}. That is

$$W_{el} = c_2 \times m_{ref} \times \left(\Delta T_{lift}\right)^{m_2} \tag{29}$$

Then, Eq. (27) may be rewritten as

$$COP_{el} = c \times \left(\Delta T_{lift}\right)^{m} + d \tag{30}$$

where c, m and d are functions of some other parameters which may be reasonably assumed constant in a narrow operating range.

Fig. A1.5 shows COP_{el} of R407C BW heat pumps against ΔT_{lift}.

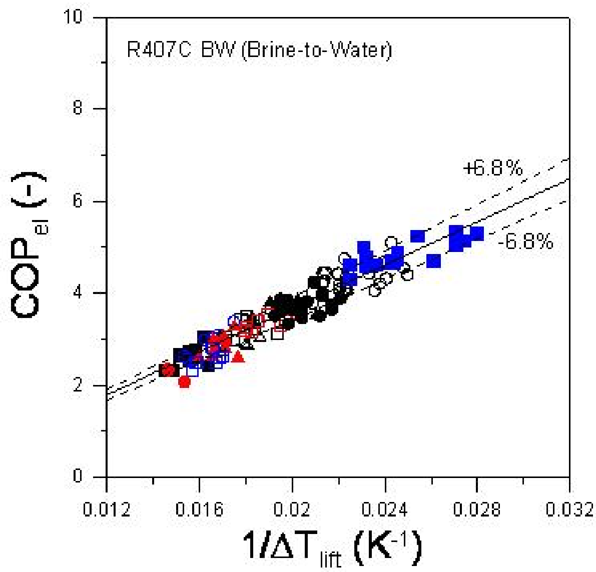


Fig. A1.5. COP_{el} vs. ΔT_{lift} for R407C Brine-to-Water heat pumps

The solid line is a linear fit and dashed lines denoted the boundaries of ±6.8% standard deviation from the fit. It seems that the exponent m=-1 in Eq. (30) is a reasonable value.

Note that ΔT_{lift} defined in Eq. (7) includes no information on the secondary side. Therefore Fig gives no explicit information on heat exchangers.

The following Figs. (A1.6 and A1.7) show corresponding results for EW (direct expansion) and WW (water-to-water) heat pumps.

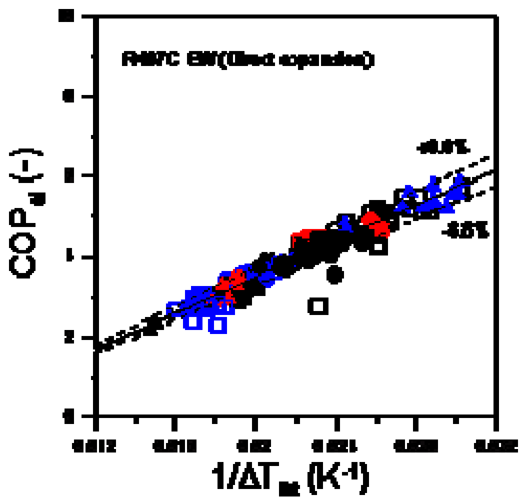


Fig. A1.6. COP_{el} vs. ΔT_{lift} for R407C EW.

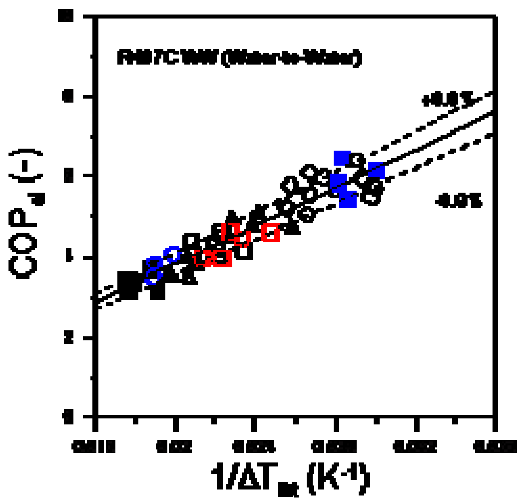


Fig. A1.7. COP_{el} vs. ΔT_{lift} for R407C WW

Table A1.1 summarizes information on the linear fits in Figs A1.5-A1.7.

Table A1.1. Constants in Eq. (30) for R407C heat pumps

	c	m	d	Standard deviation (±%)
BW	235.1	-1	- 1.036	6.8
EW	224.2	-1	- 1.004	6.6
WW	237.6	-1	- 0.929	6.9

Comparison of those of the linear fits in Table A1.1 may give some interesting information on the relation between COP_{el} and heat pump type. Fig. A1.8 shows the three linear fits for comparison.

As can be expected from the similar c values in Table A1.1, three linear lines look almost parallel to each other. For the same ΔT_{lift}, COP_{el} is the highest with water-to-water (WW in the figure) heat pumps and the lowest with direct expansion (EW) heat pumps. For two adjacent linear lines, one is near to but outside the standard deviation boundary of the other implying that the COP_{el} difference is meaningful.

Although not completely clear, high COP_{el} of WW heat pumps may be attributable to better performance of water as secondary fluid. And low COP_{el} of EW heat pumps may be attributable to the relatively large pressure drop in collector lines.

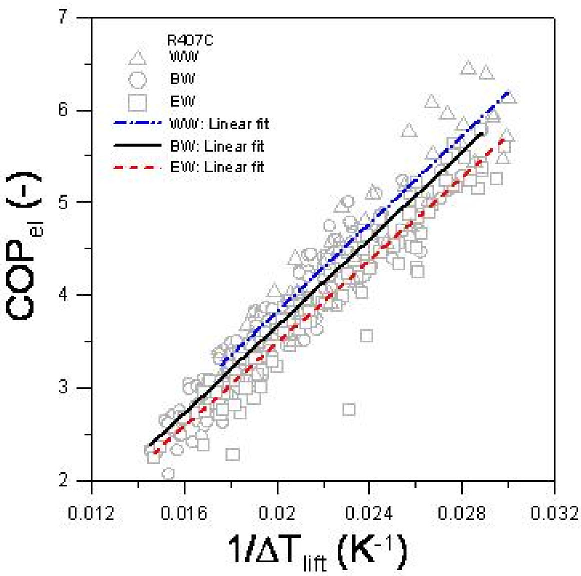


Fig. A1.8. COP_{el} vs. ΔT_{lift} relations for different types of R407C heat pumps

Note that the low COP_{el} of EW heat pumps do not mean poorer performance in the field. A direct expansion heat pump is intended to avoid the secondary loop between heat pump and ground in a ground-coupled system. Absence of the secondary loop outbalances the efficiency drop due to the pressure drop in the collect line. It is also interesting to compare different refrigerants in terms of COP_{el}. For this purpose, neglecting heat pumps types, all R407C data were fitted with linear line as shown in Fig. A1.9. The solid linear line in Fig. A1.9 fits the data with standard deviation of ±7.5%. All refrigerants and fitting results are summarized in **Fel! Hittar inte referenskälla.**.

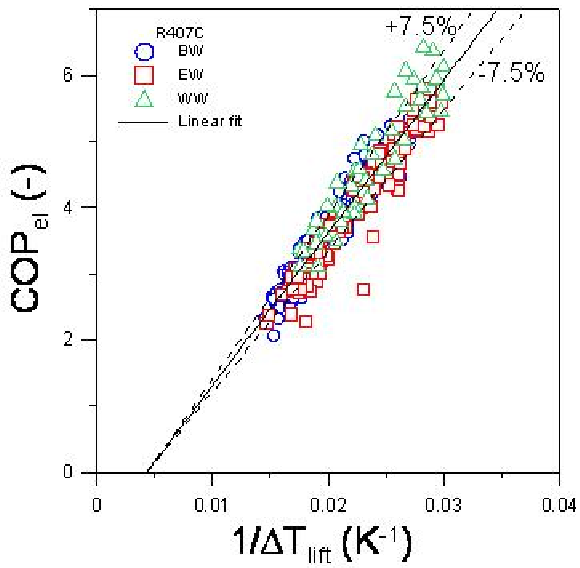


Fig. A1.9. COP_{el} vs. ΔT_{lift} for all R407C data

Table A1.2. Constants in Eq. (30) for all refrigerants

	c	m	d	Standard deviation (±%)	No. of data
R407C	231.2	-1	- 1.008	7.5	324
R290	177.2	-1	+ 0.016	5.2	61
R134A	214.5	-1	- 0.998	9.3	34
R417A	220.3	-1	- 0.714	2.5	33
R410A	204.8	-1	- 0.646	3.1	23

Fig. A1.10 is a graphical representation of Table A1.2. It is readily seen that R407C and R417A form the highest COP_{el} group. R134A and R410A form the lowest group with R410A slightly better than R134A. It is peculiar that R290 is the lowest in low lift region (in large $1/\Delta T_{lift}$) but one of the highest in high lift region (in small $1/\Delta T_{lift}$).

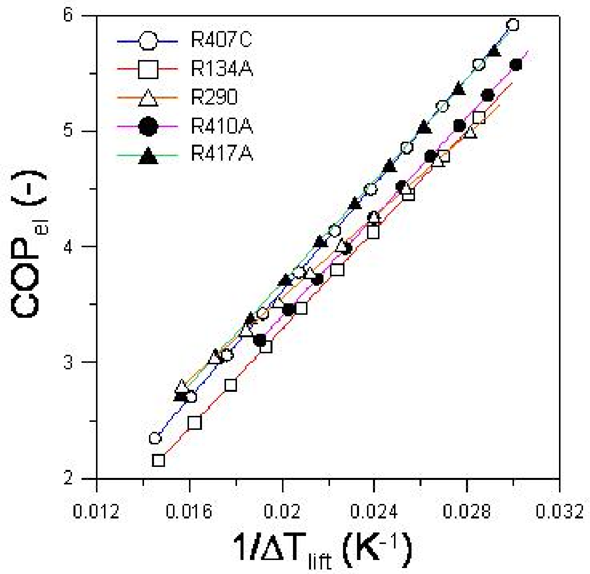


Fig. A1.11. COP_{el} vs. ΔT_{lift} for all refrigerants

As previously described, condensers were divided into superheat, saturated and subcooled regions and three UA values are available for each of the condensers. Evaporators were divided into two regions and UA values are available for saturated and superheated region for each of them except for the collect lines in direct expansion heat pumps.

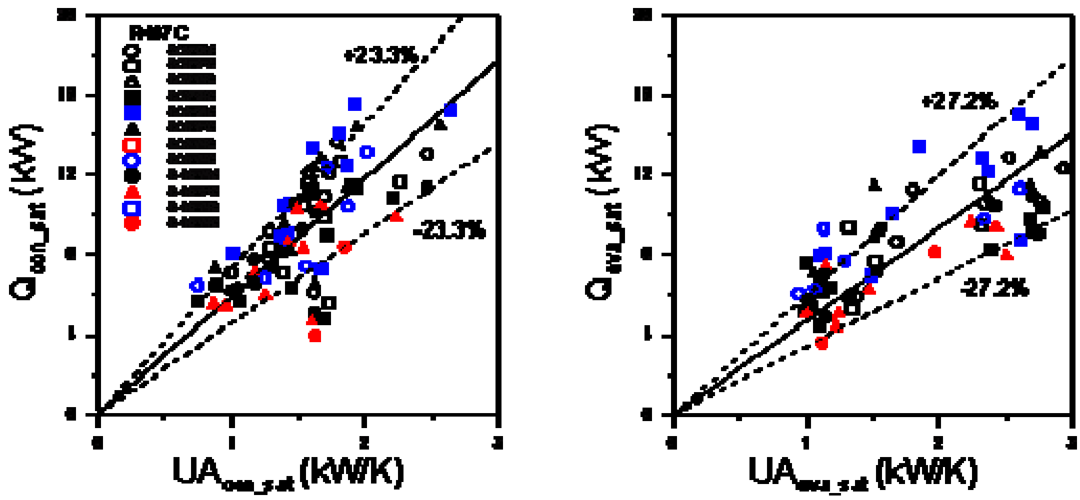
However, unlike the COP_{el} vs. ΔT_{lift} relations presented above, UA values of the heat pumps based on other refrigerants than R407C show large scatter when drawn against main performance figures like heating power Q_{con} and COP_{el} probably because of the comparably small number of available samples. Therefore it was decided to use only R407C data in the following analysis.

Different heat pumps are equipped with different combinations of heat exchangers. And because a heat pump operates under various operating conditions according to the changes in environment and the corresponding control actions. Therefore it is difficult or may be even impossible to characterize such a heat pump in terms of a few simple parameters regarding heat exchangers, which is tough desirable in the present analysis.

Efficiencies of heat pumps, if compressor efficiency is the same, would be dependent mainly on the sizes, i.e. UA values of the condensers and evaporators because they are the main parameters that determine system pressures and therefore compressor work. UA value of a heat exchanger, however, is not constant but subject to change according to operating conditions, for example, with primary and secondary flow rates. Therefore UA value cannot represent a heat exchanger under variable operating conditions.

Starting from the idea that the heat transfer rate is proportional to the refrigerant flow rate and so is the heat transfer coefficient; it was decided to check the correlation between heat transfer rates and UA values hoping to find out a meaningful relationship.

Fig. A1.12 shows the heat transfer rate against UA for the saturated regions in the condensers and evaporators of R407C BW heat pumps. Secondary flow rates were adjusted during the tests to maintain 5K difference between the inlet and outlet temperatures in the condenser, i.e. ΔT_{sink}=5K.



(a) Q_{con_sat} vs. UA_{con_sat} (b) Q_{eva_sat} vs. UA_{eva_sat}

Fig. A1.12. Q vs. UA for R407C BW heat pumps with ΔT_{sink}=5K

The solid line in the figure is a linear fit defined by

$$Q = c \times UA \tag{31}$$

where c is a constant. c=5.91 in Eq. (31) reproduces the data in
a with ±23.3% of standard deviation and c=4.7 in
b with ±27.2% of standard deviation. According to the definition in Eq. (31), c is equivalent to LMTD (logarithmic mean temperature difference) which is frequently used in the heat transfer equation for a heat exchanger and can be considered as an average temperature difference between primary and secondary flows. Apparently, Q and UA in the saturated regions of the condensers have a relationship that may be represented with a linear line. It turned out that this linear relationship holds for the rest of heat exchangers in other types of heat pumps too.

Tables A1.3 and A1.4 summarize fitting results for all the condensers and evaporators in R407C heat pumps, respectively.

Table A1.3. Fitting results of Q against UA with Eq. (31) for the saturated regions in the condensers in R407C heat pumps

Type	* ΔT_{sink} (K)	c	Standard deviation (±%)
BW	5	5.91	23.3
EW	5	5.15	24.0
WW	5	6.79	17.9
BW	10	6.17	20.0
EW	10	6.33	24.8
WW	10	7.35	23.0

* Flow rate is adjusted to maintain the temperature difference in secondary medium across the condenser during the test.

Table A1.4. Fitting results of Q against UA with Eq. (31) for the saturated regions in the evaporators in R407C heat pumps

Type	* ΔT_{sink} (K)	c	Standard deviation (±%)
BW	5	4.70	27.2
WW	5	5.44	23.5
BW	10	5.07	24.3
WW	10	4.96	32.9

* Flow rate is adjusted to maintain the temperature difference in secondary medium across the condenser during the test.

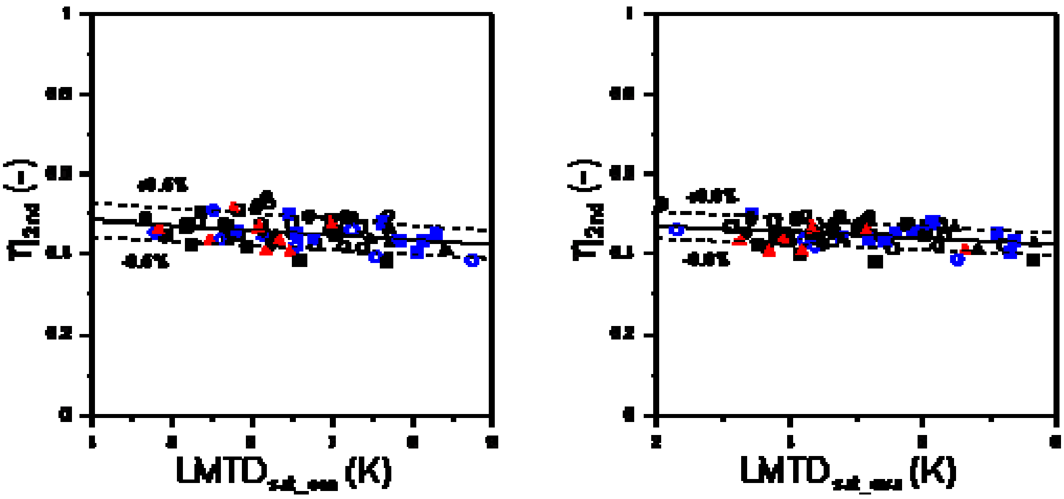
In these two Tables, it is notable, that c values for condenser are distributed in a relatively wide range from 5.2 to 7.4K while those for evaporator are closely distributed between 4.7 and 5.4K. It is also notable that the condensers in WW heat pumps have in general, larger c values than those in the other types. It may be understood that c values in these Tables are the average LMTDs of the heat exchangers in the heat pumps tested. Uncertainties are represented by the standard deviation in the rightmost column which ranges from 18 to 33%.

Now let us see how a heat exchanger influences the performance of a heat pump. It is important to choose right parameters to see the influence. It was seen that COP_{el} had clear relationship with ΔT_{lift} in Fig – Fig. A1.6 and Table A1.2. ΔT_{lift} itself, however, contains no explicit information on heat exchangers since it is based on measured pressures. Therefore it is hard to see heat exchangers' effects on COP_{el} in COP_{el} vs. ΔT_{lift} relations. It is necessary to choose other parameters that could explicitly shown an efficiency-heat exchanger relationship. For this purpose, the second law efficiency η_{2nd} defined in Eq. (5) is chosen to represent heat pump efficiency and LMTD is chosen for representation of a heat exchanger. By definition, η_{2nd} includes information on both primary and secondary side temperatures and therefore it was thought that it might show a certain trend against LMTD which represents a heat exchanger's capacity in terms of the temperature difference between primary and secondary flows. Fig. A1.13 shows η_{2nd} drawn against $LMTD_{sat\_con}$ (LMTD of the saturated region in condenser) for the R407C BW heat pumps tested with ΔT_{sink}=5K. The data shows that η_{2nd} is inversely proportional to $LMTD_{sat\_con}$. This is a certainly expected result considering that large UA (thus small LMTD) decreases ΔT_{lift} for given set of sink and source temperatures and consequently increases COP_{el} as in Figs. A1.1 – A1.6 resulting in higher η_{2nd}.

In Fig. A1.13b corresponding result is shown for the evaporators and a similar trend can be seen. Again, similar to Eq. (30), η_{2nd} may be expressed as

$$\eta_{2nd} = c \times LMTD^{m} + d \qquad (32)$$

The solid lines in Fig. A1.13 are linear fits (m=1) and standard deviations are ± 8.5% and ± 6.8%, respectively.



(a) η_{2nd} vs. $LMTD_{sat\_con}$ (b) η_{2nd} vs. $LMTD_{sat\_eva}$

Fig. A1.13. η_{2nd} vs. LMTDs for R407C BW heat pumps with ΔT_{sink}=5K

Linear fit is deemed enough for the present analysis and no other attempt has been made. Table A1.5 summarizes fitting results for η_{2nd} against $LMTD_{sat\_con}$ for all types of heat pumps.

Table A1.5. Fitting results of η_{2nd} against LMTD with Eq. (31) for the saturated regions in the condensers in R407C heat pumps (m=1)

Type	*ΔT_{sink} (K)	c	d	Standard deviation (±%)
BW	5	-0.0123	0.534	8.5
EW	5	-0.0099	0.490	7.3
WW	5	-0.0122	0.521	6.9
BW	10	-0.0049	0.469	6.4
EW	10	-0.0162	0.502	8.7
WW	10	-0.0204	0.554	9.5

* Flow rate is adjusted to maintain the temperature difference in secondary medium across the condenser during the test.

The gradient c ranges from -0.0204 to -0.0049 and d ranges from 0.469 to 0.554. Standard deviation of the fits ranges from ±6.4 to ±9.5% being comparable to those in Table A1.6. Physically, the c value in Table A1.5 means that every 1K drop in $LMTD_{sat\_con}$ increases η_{2nd} by from ca. 0.005 to 0.02.

Table A1.6 summarizes fitting results for η_{2nd} against $LMTD_{sat\_eva}$ for all types of heat pumps.

Table A1.6. Fitting results of η_{2nd} against LMTD with Eq. (31) for the saturated regions in the evaporators in R407C heat pumps (m=1)

Type	$^{*}\Delta T_{sink}$ (K)	c	d	Standard deviation (±%)
BW	5	-0.0080	0.488	6.8
WW	5	-0.0140	0.515	6.6
BW	10	-0.0098	0.486	5.8
WW	10	-0.0228	0.528	8.1

* Flow rate is adjusted to maintain the temperature difference in secondary medium across the condenser during the test.

The c values in Table A1.6 are in the same range as the condensers in Table 5 and show that every 1K drop in $LMTD_{sat\_eva}$ increases η_{2nd} by from ca. 0.008 to 0.023.

From the results above, it seems that influences of the evaporators and the condensers on the system performance became clear. However, note that $LMTD_{sat\_eva}$ is not constant when η_{2nd} is fitted with $LMTD_{sat\_con}$ in Fig. A1.13(a) and vice versa in Fig. A1.13(b). That is, neither in Table A1.4 or Table A1.6 are the influences of the evaporators and condensers separated. Considering that $LMTD_{sat\_con}$ and $LMTD_{sat\_eva}$ are, basically, independent variables expected to have the same magnitude of influences on η_{2nd}, it is necessary to separate the influence of one from the other. A few attempts have been made to separate those influences but failed. It would require some time to devise a proper model for the purpose. However, observing the very similar trends in Fig. A1.13, it may be expected that $LMTD_{sat\_con}$ and $LMTD_{sat\_eva}$ can be somehow combined to form such a single variable that it clearly shows the influence of each parameter on η_{2nd}. This task is reserved for the future.

APPENDIX 2 Japan – Scientific Contributions

Study on heat transfer characteristics of carbon dioxide at both supercritical and subcritical pressures

Eiji Hihara and Chaobin Dang（The University of Tokyo)

A2.1 Introduction

Since the CO_2 heat pump works at a relatively high operation pressure, the heat transfer characteristics of CO_2 differ from conventional CFC or HFC refrigerants, and a systematic investigation on both the cooling heat transfer of CO_2 at supercritical pressure and the flow boiling heat transfer at subcritical pressure is critical for the design and optimization of the heat exchangers for CO_2 heat pump.

To clarify the characteristics of flow and heat transfer of supercritical CO_2 cooled in smooth tube, experimental measurement, theoretical analysis and numerical simulation were conducted. From the comparison between the experimental results and numerical simulation results, the proper selection of turbulence models for supercritical CO_2 was discussed. Based on the understanding of temperature and velocity distributions at cross-sectional direction provided by the numerical simulation, a new prediction correlation was proposed which agreed well with the experimental results. In addition, the effect of lubricating oil on the heat transfer characteristics of supercritical CO_2 was studied through heat transfer measurement and flow pattern visualization. The heat transfer characteristics of supercritical CO_2 inside a small-sized micro-fin tube were also investigated.

Flow boiling heat transfer of CO_2 was experimentally investigated for smooth tube with tube diameter ranging from 1 ~ 6 mm and for micro-fin tube with mean inner diameter of 2 mm. heat transfer characteristics of pure CO_2 and the effect of lubricating oil were studied under different mass flux and heat flux conditions. Nucleate boiling was found the dominant heat transfer mechanism for both the smooth and micro-fin tubes due to its low surface tension and high density ratio of vapor to liquid of CO_2. Although the CO2 has a high heat transfer coefficient at pre-dryout region, the dryout quality is usually low compared to conventional refrigerants. Contaminating lubricating oil into CO_2 led to a sharp decrease in heat transfer coefficient. By using micro-fin tube, the dryout shifted to a high vapor quality side and the heat transfer performance enhanced significantly.

A2.2 Cooling heat transfer of supercritical CO_2

A2.2.1 Heat transfer of pure CO_2 [1][2]

Experimental measurement, theoretical analysis and numerical simulation were conducted to clarify the characteristics of flow and heat transfer of supercritical CO_2 cooled in smooth tube, with tube diameter varied between 1 ~ 6 mm. Effects of parameters such as mass flux, pressure, heat flux, and cooling-tube diameter on the heat transfer coefficient and pressure drop were analyzed. Experimental results were compared with several existing correlations, and a modification of the Gnielinski correlation was proposed for the heat transfer coefficient. The heat transfer coefficient increases with increasing mass flux, the effects of heat flux and tube diameter depend on the property variation in the radial direction. The pressure drop increased with increasing mass flux. The pressure drop seems independent of inlet pressure at temperature lower than the pseudocritical temperature. When the temperature is higher than the pseudocritical temperature, the pressure drop decreases significantly with increasing pressure. In addition, a modified Gnielinski equation by selecting reference temperature properly was developed as follows. The modified correlation predicted experimental data with an accuracy of 20%.

$$\alpha = Nu k_f / d, \tag{1}$$

$$Nu = \frac{(f_f/8)(\mathrm{Re}_b - 1000)\,\mathrm{Pr}}{1.07 + 12.7\sqrt{f_f/8}(\mathrm{Pr}^{2/3} - 1)}, \tag{2}$$

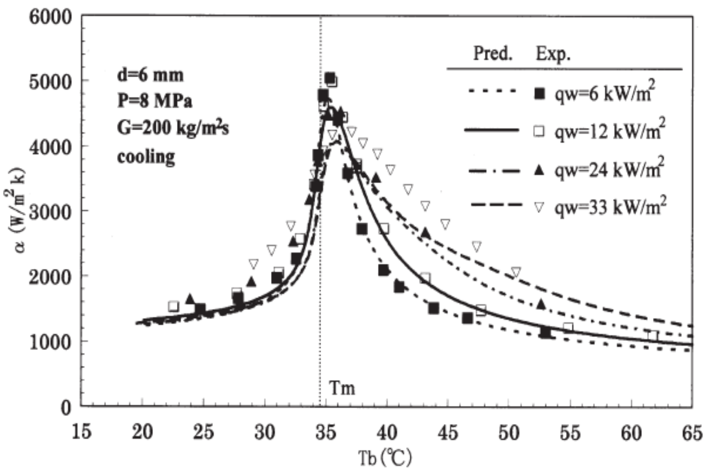
$$\mathrm{Pr} = \begin{cases} c_{p_b}\mu_b / k_b, & \text{for } c_{p_b} \ge \bar{c}_p \\ \bar{c}_p \mu_b / k_b, & \text{for } c_{p_b} < \bar{c}_p \text{ and } \mu_b/k_b \ge \mu_f/k_f \\ \bar{c}_p \mu_f / k_f, & \text{for } c_{p_b} < \bar{c}_p \text{ and } \mu_b/k_b < \mu_f/k_f \end{cases}, \tag{3}$$

$$\bar{c}_p = (h_b - h_w)/(T_b - T_w), \tag{4}$$

$$\mathrm{Re}_b = Gd/\mu_b, \tag{5}$$

$$f_f = \left[1.82\log_{10}(\mathrm{Re}_f) - 1.64\right]^{-2}, \tag{6}$$

$$\mathrm{Re}_f = Gd/\mu_f, \tag{7}$$



(a) JL model

Fig. A2.1 Comparison between the simulation results and experimental results.

Numerical simulation was conducted to examine the applicability of different turbulence models to the prediction of heat transfer coefficient of supercritical CO_2. The JL (Jones and Launder, 1972) model works well under most conditions, although at large heat flux, it slightly underestimates the measured heat transfer coefficient near the pseudocritical temperature. The LS (Launder and Sharma, 1974) model significantly underestimates the measured heat transfer coefficient, mainly due to the small damping function in the near-wall region. In the BR model and MK model, different definition of y^+ in the damping function leads to significantly different results. The integrated effect of properties should be considered to describe the effect of heat flux on heat transfer coefficient correctly. In addition, according to a comparative computation using different models of turbulent Prandtl number, the turbulent Prandtl number does not affect the calculation results of the heat transfer coefficient for the experimental conditions considered here. The comparison between numerical simulation results and experimental results demonstrated that the numerical simulation can be a feasible tool in analyzing the heat transfer characteristics of supercritical fluids, if only the turbulence model being properly selected.

A2.2.2 Effect of lubricating oil [3][4][5][6]

The heat transfer characteristics of supercritical CO_2 cooled in horizontal circular tubes with a small amount of entrained PAG-type lubricating oil were investigated experimentally. The oil concentrations were varied from 0 to 5 wt% along with variations in the mass flux, pressure, heat flux, and tube diameter, and their effect on the heat transfer coefficient and pressure drop were analyzed. The main conclusions of this study are summarized as follows:

(1) From flow observations, it was found that the flow pattern varies with the temperature, mass flux, and tube diameter. For a 2 mm I.D. tube at a low temperature, the oil flows in the bulk region with a diameter from 50 to 100 mm. With an increase in the temperature, an oil-rich layer is observed in the near-wall region. For a 6 mm I.D. tube, the flow pattern was a separated wavy flow at a low mass flux, and it transformed into an annular-dispersed flow at a high mass flux. See Fig. A2.2.

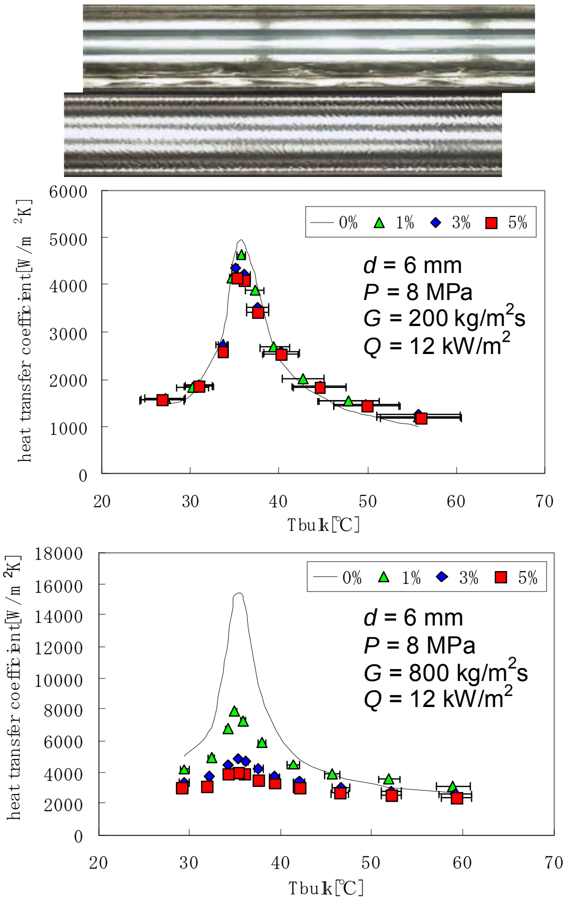


Fig. A2.2 Flow pattern of CO_2-oil and heat transfer coefficient for 6 mm I.D. tube

(2) For a large-sized tube with a mass flux lower than 400 kg/m^{2}s, no significant difference in the heat transfer coefficient was observed until the oil concentration reached 1%. A sharp decrease occurred at an oil concentration approximately between 1 and 3%. A further increase in the oil concentration above 3% did not decrease the heat transfer coefficient significantly.

(3) For small-sized tubes, the heat transfer coefficient dropped at a rather low oil concentration. At an oil concentration of 1%, the heat transfer coefficient was60 and 50% of that at the oil-free condition for the 2 and 1 mm I.D. tubes, respectively.

(4) The pressure drops for the 2 and 1 mm I.D. tubes did not increase significantly at an oil concentration of 1%. For the 2 mm I.D. tube, there was no difference between the pressure drops for the oil concentrations of 5 and 3%. However, for the 1 mm I.D. tube, the pressure drop increased monotonously with an increase in the oil concentration.

(5) A neural network method is proposed for the prediction of the heat transfer coefficient of supercritical CO_2 with entrained PAG-type lubricating oil. The results from the proposed model agreed well with the experimental results, with the deviation of 87.3% data (1146/1313) being within ±20%; 96.1% data (1262/1313) were found to be within a deviation of ±30%.

In addition, the effects of lubricating oil on cooling heat transfer coefficient of supercritical carbon dioxide were discussed applying three different types of lubricants, Polyalkylene Glycol(PAG), Polyvinyl Ether(PVE) and PAG-PVE Copolymer (ECP). The varied flow pattern of two-phase fluid caused by the different type of oil leads to the big difference on the heat transfer. For oils with good solubility with CO_2 at the inlet temperature of bulk CO_2 lower than the pseudocritical temperature, the sharp deterioration of heat transfer will be prevented due to the difficulty on the oil film formation. In case of the inlet temperature of bulk CO_2 over the critical temperature, the thinner oil film is formed comparing to other oil with poor solubility, which slows down the decrement of heat transfer. By means of the visible experiments, it is found that a thick oil film of PAG is easy to form in comparison to PVE and ECP due to the poor miscibility of PAG and CO_2. The ECP oil mixing in CO_2 has a better heat transfer performance in comparison to that of PAG oil – Fig. A2.3.

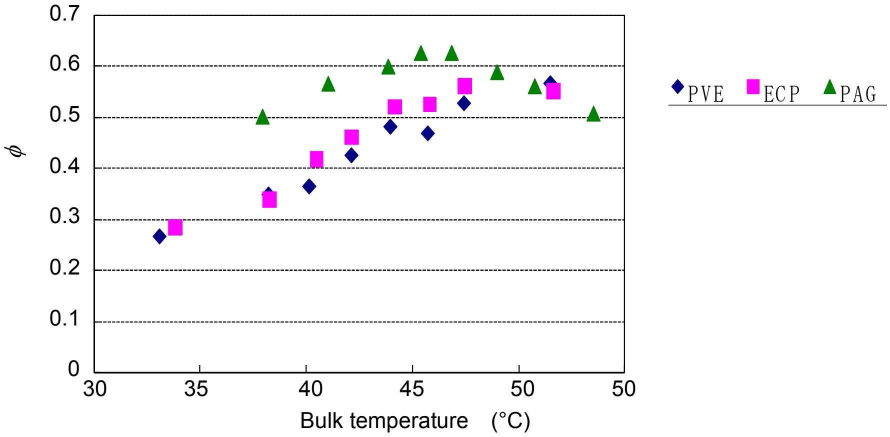


Fig. A2.3 Comparison of the decrease ratio of heat transfer coefficient of CO_2 with three types of oils (G = 800 kg/(m^2·s), P = 10 MPa, x = 2 wt%)

A2.2.3 Heat transfer characteristics inside micro-fin tube [7][8]

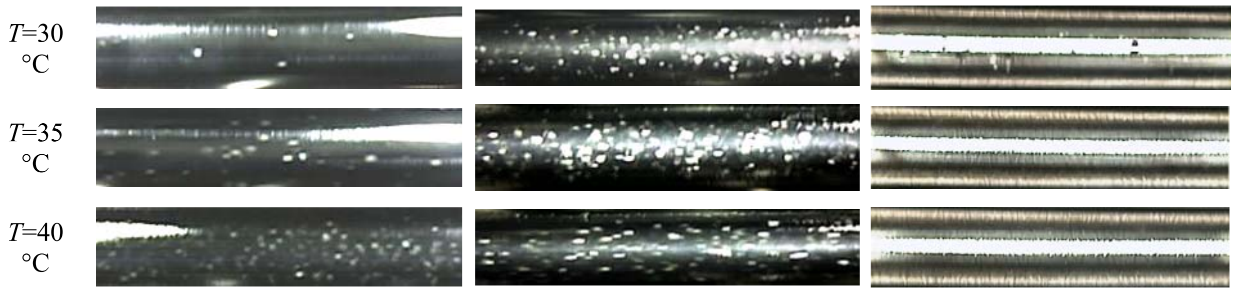
In this study, the heat transfer characteristics of supercritical carbon dioxide cooled inside a small-sized internally grooved tube was experimentally investigated, Fig. A2.4. Main conclusions of the study are summarized as follows:

(1) Under oil-free conditions, the heat transfer coefficient and pressure drop for the grooved tube are found to be proportional to those for a smooth tube. The prediction model proposed for smooth tubes can also be used for grooved tubes when the area enlargement ratio is taken in to consideration.

(2) An increase in oil concentration leads to a drastic decrease in the heat transfer coefficient. The most drastic drop occurs in the pseudo critical region, Fig. A2.5.

(3) The drop in heat transfer coefficient increases to 30%~50% when the oil concentration increases to 1%, and the drop in the heat transfer coefficient reaches 50%~70% when the oil concentration increases to 3%. However, for an increase in the oil concentration from 3% to 5%, there is no distinct decrease in the heat transfer coefficient, showing saturation phenomena in the heat transfer deterioration due to the breakup of oil film.

(4) The heat transfer coefficient of CO_2 entrained with a PAG-type lubricating oil and cooled in the grooved tube is found to be higher than that in the smooth tube, even when the area enlargement ratio is taken into consideration, especially at the low-temperature side. These results indicate that the grooved tube has a large heat transfer area and also aids in the breaking up of the oil film and formation of oil droplets; this further enhances the heat transfer performance.



(a) PVE (b) ECP (c) PAG

Fig. A2.4 Flow visualization of CO_2 with entrained oil
(P = 8MPa, G = 800kg/(m^2·s), x= 1 wt%)

A2.3 Flow boiling heat transfer of CO_2

A2.3.1 Flow boiling heat transfer of pure CO_2[9][10]

Boiling heat transfer coefficients of carbon dioxide in horizontally located smooth tubes were experimentally investigated. The inner diameter of heat transfer tubes was 1, 2, 4, and 6 mm. Experiments were conducted at evaporating temperature of 5 and 15 °C, heat fluxes from 4.5 to 36 kW/m^2, and mass fluxes from 360 to 1440 kg/m^{2}s. The following conclusions can be drawn.

(1) The effects of heat flux on the heat transfer coefficient are much significant in the pre-dryout region, which is related with the activation of nucleate boiling. On the contrary, the effects of mass flux are relatively low due to the small two-phase density ratio near the critical point.

(2) As the inner tube diameter increases, the effect of heat flux on the heat transfer coefficient decreases, in particular, the effect of heat flux is not observed in the 6 mm I.D. tube.

(3) Pre-dryout heat transfer coefficient increases with the increase in evaporating temperature, while the dryout inception quality and post-dryout heat transfer coefficient are not affected greatly by the evaporating temperature.

(4) Mass flux is the dominating factor that affects the dryout quality. At a high mass flux of 1440 kg/m^{2}s, dryout may occur at a small quality of 0.4. Increasing tube diameter or heat flux can decrease the dryout quality slightly.

(5) Mass flux is the dominating factor that affects the heat transfer coefficient in post-dryout region. At small mass flux, the heat transfer coefficient decreases with the increase of quality, while at large mass flux such as 1440kg/m^{2}s, the heat transfer coefficient turns to increasing with the quality.

(6) Due to the complicity of the thermodynamic non-equilibrium phenomenon, none of the previously proposed models can reasonably correlate the experimental data obtained for CO_2 at post-dryout region. A modified prediction model based on the Shah model was proposed and the comparison showed that the 80% of the experimental data are well predicted with a divergence of less than ±30%. See Fig. A2.5.

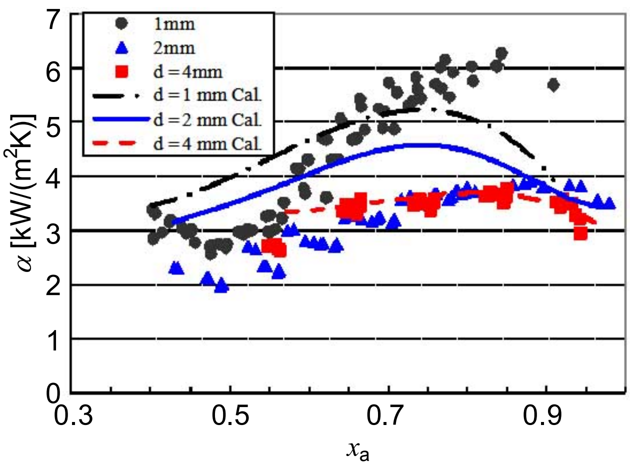


Fig. A2.5 Proposed correlation for the post-dryout heat transfer prediction (d = 1 ~ 4 mm, G = 1440 kg/(m^2·s), Q = 36 kW/m^2)

A2.3.2 Effect of lubricating oil[11]

In this study, the flow boiling heat transfer of CO_2 with a small amount of PAG-type lubricating oil was studied experimentally. The main conclusions are summarized as follows:

(1) The heat transfer coefficient decreases sharply at a critical oil concentration. The critical oil concentration for a tube with an inner diameter of 2 mm is approximately 0.5%, while that for tubes with inner diameters of 4–6 mm is approximately 1%. However, the heat transfer coefficient does not decrease further at higher oil concentrations.

(2) The heat flux positively influences the pre-dryout heat transfer coefficient at a small mass flux in the low quality region. This is due to the effect of nucleate boiling. At a large mass flux or in the high quality region, no obvious difference is observed when the heat flux is varied. Moreover, the heat flux does not influence the dryout quality and the post-dryout heat transfer coefficient.

(3) At a low heat flux, the heat transfer coefficient in the pre-dryout region increases with the mass flux. However, at a large heat flux, the influence of the mass flux on the pre-dryout heat transfer coefficient is not significant. The dryout quality decreases at a large mass flux.

(4) As the tube diameter decreases, the increase in the heat transfer coefficient due to the heat flux in the pre-dryout region becomes significant.

(5) The pressure drop monotonously increases with the oil concentration. This is attributed to an increase in the viscosity of the CO_2-oil mixture and the formation of an oil layer along the inner wall of the tube.

Data are given in Fig. A2.6, with and without oil.

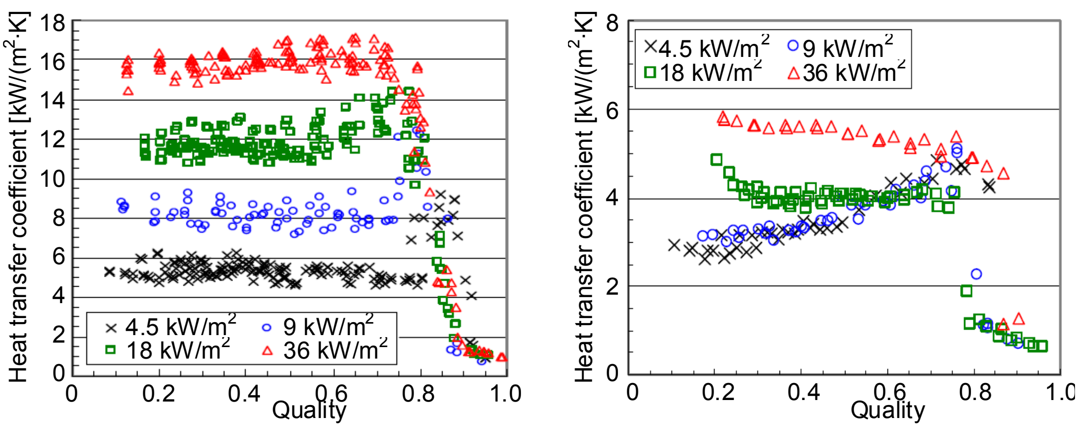


Fig. A2.6 Comparison of flow boiling heat transfer characteristics of CO_2 with and without oil (d = 2 mm, G = 360kg/m^{2}s, Q = 4.5 ~36 kW/m^2)

A2.3.3 Flow boiling heat transfer inside micro-fin tube[12]

In this study, the flow boiling of CO_2 inside a small micro-fin tube was studied by comparising the measured results with that of smooth tube. The conclusions derived from this study can be summarized as follows:

(1) Nucleate boiling is the dominant heat transfer mechanism for flow boiling of CO_2 in both the micro-fin and smooth tubes. However, suppression of nucleate boiling was observed in the case of the micro-fin tube, especially under a high mass flux of 720 kg/m^{2}s and low heat flux condition of 9kW/m^2.

(2) The use of the micro-fin tube increases both the pre-dryout heat transfer coefficient and dryout quality. The enhancement ratio of heat transfer coefficient was 1.9 to 2.3, and the dryout occurred at a higher value ranging from 0.9 to 1.0.

(3) The enhancement ratio of pressure drop was 1.5 to 2.1, which is smaller than the enhancement ratio of heat transfer coefficient. Data are given in Figs. A2.7 and A2.8.

(4) Since the heat transfer coefficient increases with the decrease in mass flux and heat flux, while the pressure drop changed reversely, it is suggested that a relatively low mass flux and high heat flux should be applied for the evaporator in order to obtain a high heat transfer coefficient at a low penalty pressure drop.

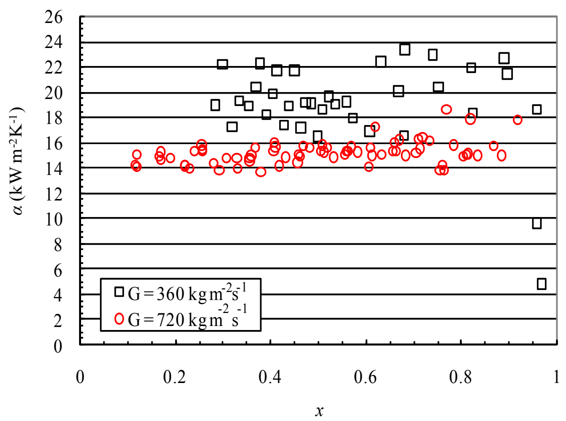


Fig. A2.7 Effect of mass flux on heat transfer coefficient for micro-fin tube

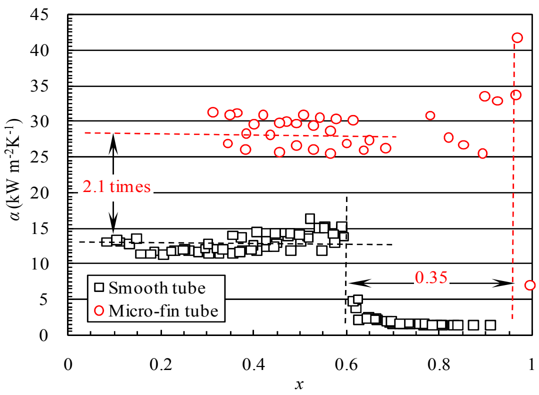


Fig. A2.8 Comparison of heat transfer coefficient between micro-fin tube and smooth tube

A2.4 Heat transfer characteristics of carbon dioxide inside spirally grooved tubes

Shigeru Koyama, Ken Kuwahara, and Shinya Higashiiue (Kyushu University)

A2.4.1 Summary

Recently, from the viewpoint of greater environmental protection, the development of high-performance heat pump and refrigeration systems using naturally occurring substances, such as CO_2, NH_3, i-C_4H_{10}, as working fluids has become the most important challenge. Therefore, we have investigated the heat transfer characteristics of CO_2 in spirally grooved tubes experimentally. This report deals with our group's experimental research concerning the forced convection cooling heat transfer of CO_2 in the supercritical pressure range and the flow boiling heat transfer of CO_2 in the subcritical pressure range inside several spirally grooved tubes.

A2.4.2 Experimental apparatus

Fig. A2.9 shows a schematic diagram of our experimental apparatus used in the experiment of cooling heat transfer of CO_2 under supercritical pressure. The experimental apparatus is a vapor compression heat pump system that consists of a refrigerant loop, PAG oil loop and heat source water loop. Major elements of refrigerant loop include a compressor, two oil separators, a pre gas cooler, a heat transfer test section, a CO_2-PAG oil mixture sampling section, an after gas cooler, an accumulator, a mass flowmeter and two electrically heating evaporators. The heat source loop is used as the cooling heat source of CO_2 in the test section. In the experiments, oil contained in CO_2 is fully removed by the oil separator installed at the outlet of the compressor.

Fig. A2.10 shows a schematic diagram of the test section used in the experiment of cooling heat transfer of CO_2 under supercritical pressure. The test section is a double-tube counter-flow heat exchanger, in which CO_2 flows in the inner tube and cooling water flows in the annulus part surrounding the inner tube. The overall length of test section is 2,064 mm, and the test section consists of three subsections, each of which is 688 mm. The heat transfer effective length of each subsection is 500 mm.

Experiments of boiling and evaporating heat transfer of CO_2 are also done by inserting the test section, which is identical as the one shown in Fig. A2.10, between the two electric heating evaporators in the refrigerant loop in Fig. A2.9.

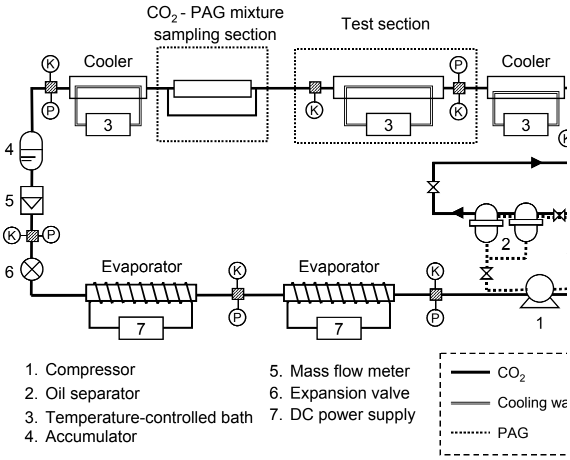


Fig. A2.9 Experimental apparatus

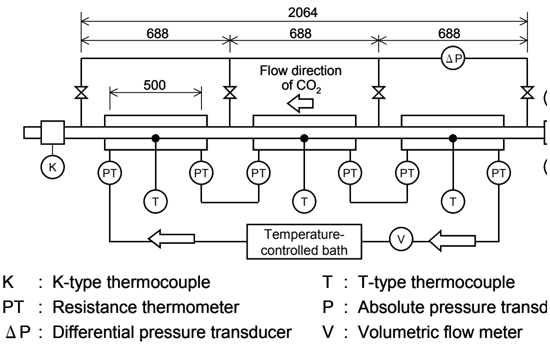


Fig. A2.10 Test section

A2.4.3 Cooling heat transfer characteristics of CO_2 at supercritical pressure

1. Test tubes and experimental range

Table A2.1 shows specifications of tested heat transfer tubes. For the experiments, three types of copper spirally grooved tubes of different fin shapes and one type of copper smooth tube were used. As for the outer diameters of test spirally grooved tubes, the outer diameters of both spirally grooved tube and smooth tube are 6.02 mm. As for the shapes of spirally grooved tubes, the height of fins is 0.15-0.24 mm, the lead angle is 5-25 degrees, the number of fins is 46-52, and the area expansion ratio is 1.4-2.3. For reference, the average inner diameter "d_i" of spirally grooved tubes is defined as the inner diameter of the smooth tube that has the same inner cross-sectional area as the actual inner cross-sectional area of spirally grooved tube.

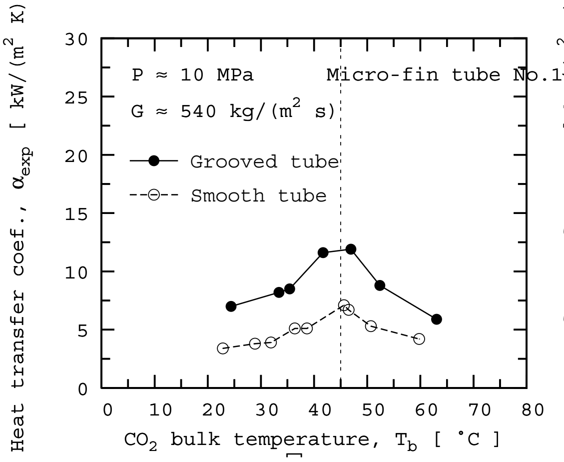
Table A2.1 Specifications of spirally grooved tubes and smooth tube

Type		Smooth tube	Grooved tube		
			No.1	No.2	No.5
Outer diameter, d_o	[mm]	6.00	6.02	6.02	6.00
Maximum inner diameter, d_r	[mm]	4.42	4.9	4.91	5.28
Average inner diameter, d_i	[mm]	–	4.76	4.76	5.11
Depth of groove, h	[mm]	–	0.15	0.18	0.24
Helix angle, β	[deg]	–	24	5	25
Number of Grooves, N	[-]	–	52	46	50
Area expansion ratio, η	[-]	1	1.4	1.4	2.3

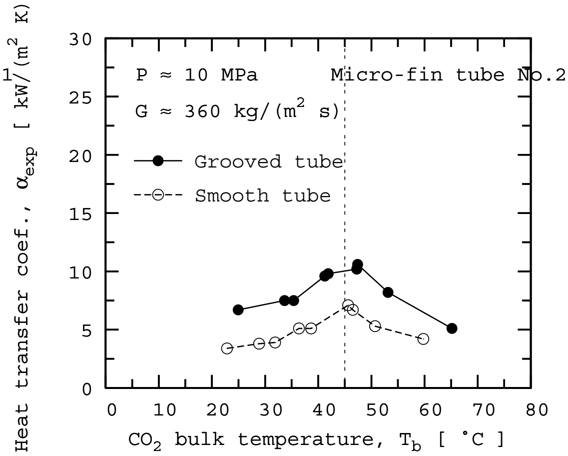
In the experiments, the heat transfer rate of almost pure CO_2 (mass fraction of 99.93% or higher) was measured within the range of pressure between 8 and 10 MPa, mass velocity of 360-690 kg/(m^2·s), and CO_2 bulk temperature of 20-75 ^oC. The temperature and flow rate of cooling water was regulated to allow the heat exchange rate of the test section as a whole to reach about 1,000W.

2. Experimental results and heat transfer correlation

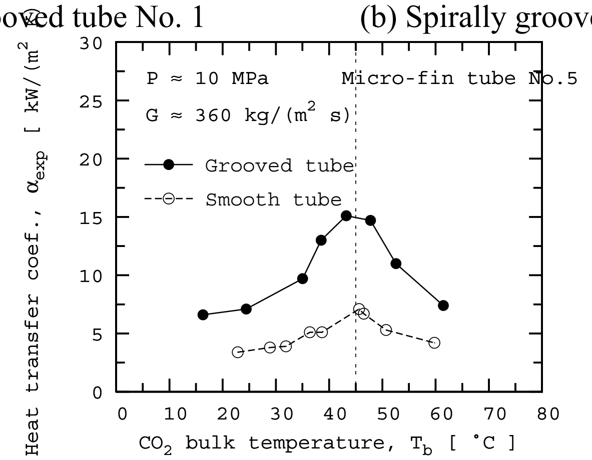
In order to confirm the heat transfer enhancement effect of spirally grooved tubes used in the experiments, the measured values of heat transfer coefficients obtained are compared with the case of smooth tube. As the representative examples of experiment results, Fig.A2.11 shows the measured values of heat transfer coefficients of grooved tubes and smooth tube in the case where the pressure is 10 MPa and the mass velocity is 540 kg/(m^2·s). Figs. A2.11 (a), (b) and (c) show the respective results of No. 1, No. 2 and No.5 spiral grooved tubes, which is furnished in Table A2.1. The heat transfer coefficient of any of the grooved tubes showed higher value than that of the heat transfer coefficient of smooth tube in the wide range of bulk temperatures of CO_2. As compared to the smooth tube, the heat transfer enhancement ratios of No. 1 and No. 2 tubes were 1.4 times, while that of No. 5 tube was about twice. These results show that the heat transfer enhancement by spirally grooved tubes is effective in the cooling process of CO_2 in the supercritical pressure range.



(a) Spirally grooved tube No. 1



(b) Spirally grooved tube No. 2



(c) Spirally grooved tube No. 5

Fig. A2.11 Comparison of heat transfer characteristics of spirally grooved tubes and smooth tube

Based on the results of measurement of heat transfer of CO_2 in spirally grooved tubes in the supercritical pressure range, the following correlation equation of heat transfer was developed:

$$Nu_f = 0.036\eta_a^{0.58}(\cos\beta)^{-0.31} \cdot \mathrm{Re}_f^{0.8}\,\mathrm{Pr}_b^{0.6} \tag{1}$$

where, η_a is the area expansion ratio and β is the lead angle of spiral groove. Nu_f, Re_f and Pr_b are Nusselt number, Reynolds number and Prandtl number, respectively, and defined as follows:

$$Nu_f = \frac{\alpha\, d_i}{\lambda_f},\ \mathrm{Re}_f = \frac{G\, d_i}{\mu_f},\ \mathrm{Pr}_b = \frac{\mu_b\, c_{pb}}{\lambda_b} \qquad (2, 3, 4)$$

Subscripts "b" and "f" show that physical properties are calculated by using bulk temperatures T_b and film temperatures $(T_{wi} + T_b)/2$, respectively.

A2.4.4 Boiling heat transfer characteristics of CO_2 at subcritical pressure

1. Test tubes and experimental range

Table A2.2 shows the specifications of five types of copper spirally grooved tubes and one type of copper smooth tube, which were used in the boiling heat transfer experiments of CO_2. The scope of groove shape parameters of experimental spirally grooved tubes is the same as that of the tubes used in the cooling heat transfer experiment in the supercritical pressure range.

2. Experimental results and heat transfer correlation

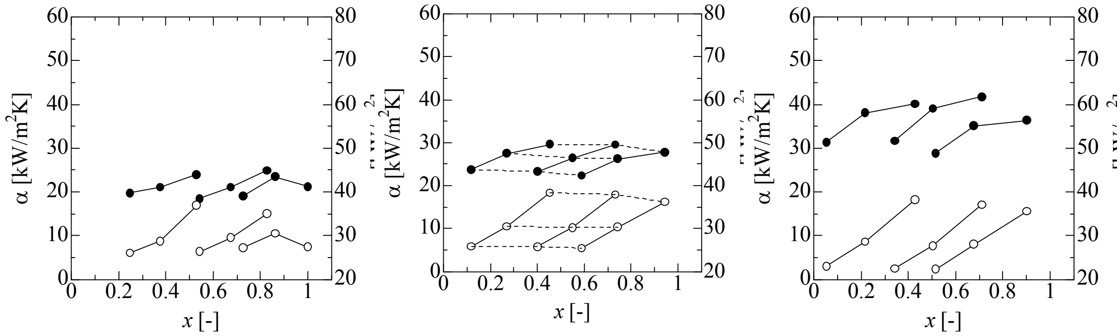
Fig. A2.12 shows the representative examples of measurement results of the heat transfer coefficient"α," heat flux "q," quality "x" in the case where the pressure in the grooved tube G1 is changed. In this figure, the horizontal axis represents quality, the vertical one on the left represents heat transfer coefficients (represented by closed circle), and the vertical axis on the right represents heat fluxes (represented by open circle). Here, the heat transfer coefficients of grooved tubes are defined based on the inner surface area of smooth tube of inner diameter d_i which is the same as the average inner diameter of grooved tube. With this figure, it can be known that the higher the pressure increases and the more the heat fluxes increase, the higher the value of heat transfer coefficient increases in the grooved tubes. This tendency is as same as in the case of the smooth tube; it shows the significant effect of nucleate boiling on heat transfer characteristics. On the other hand, the decrease in heat transfer coefficients and heat fluxes due to dryout does not clearly manifest itself in the grooved tubes. This is because it is considered that the value of quality that causes dryout due to the effect of spiral grooves becomes higher than that of the smooth tube.

Table A2.2 Specifications of experimental spirally grooved tubes and smooth tube

Type		Smooth tube	Grooved tube				
			G1	G2	G3	G4	G5
Outer diameter, d_o	[mm]	6.00	6.02	6.02	6.02	6.02	6.00
Maximum inner diameter, d_r	[mm]	4.42	4.9	4.91	5	5.3	5.28
Average inner diameter, d_i	[mm]	–	4.76	4.76	4.87	5.12	5.11
Depth of groove, h	[mm]	–	0.15	0.18	0.18	0.24	0.24
Helix angle, β	[deg]	–	24	5	24	10	25
Number of Grooves, N	[-]	–	52	46	35	55	50
Area expansion ratio, η_a	[-]	1	1.4	1.4	1.4	2.3	2.3

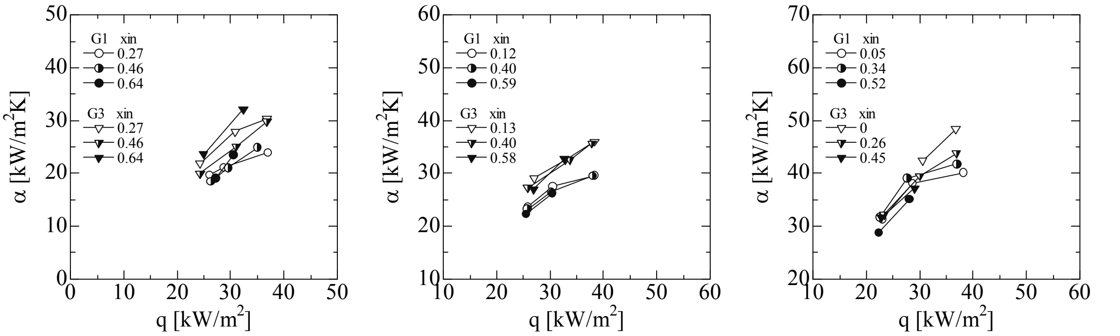
Fig. A2.13 shows the comparison of heat transfer coefficients between the grooved tube G1 (the depth of groove is 0.15 mm and the number of grooves is 52) and the grooved tube G3 (the depth of groove is 0.18 mm and the number of grooves is 35), of which the area expansion ratios and helix angles are almost the same. At any pressure, the heat transfer coefficient of the grooved tube G1 is smaller than that of the grooved tube G3. Moreover, the lower the pressure decreases, the larger the difference in heat transfer coefficients between the grooved tube G1 and the grooved tube G3 becomes. This is because it is considered that if the pressure decreases, the vapor velocity increases and the liquid films formed in grooves become thinner, and that if the groove becomes deeper, the liquid films formed in grooves become thinner. Moreover, as the number of fins of the grooved tube G1 is larger than that of the grooved tube G3, it can be seen that the effect of the groove depth on heat transfer coefficients is larger than that of the number of grooves, if the area expansion ratios and helix angles are the same.

An attempt was made to develop the correlation equation of the boiling and evaporative heat transfer coefficients in spirally grooved tubes with reference to the correlation equation of Takamatsu *et al.*(2) of the boiling heat transfer of CFC refrigerants in tubes and the correlation equation of Goto *et al.*(3) of the single-phase turbulent flow heat transfer coefficients in grooved tubes.



(a) P = 3 MPa, G=364 kg/(m^2·s)　(b) P = 4 MPa, G=363 kg/(m^2·s)　(a) P = 5 MPa, G=366 kg/(m^2·s)

Fig. A2.12 Results of measurement of heat transfer coefficients of grooved tube G1



(a) P = 3 MPa, G=360 kg/(m^2·s)　(b) P = 4 MPa, G=360 kg/(m^2·s)　(a) P = 5 MPa, G=360 kg/(m^2·s)

Fig. A2.13 Relations between heat transfer coefficients and heat fluxes (Comparison between G1 and G3) G = 360 kg/(m^{2}s)

The heat transfer coefficient α_{real} based on the actual area can be expressed by the sum of the forced convection heat transfer coefficient $(\alpha_{cv})_{real}$ and the nucleate boiling heat transfer coefficient $(\alpha_{nb})_{real}$. That is:

$$\alpha_{real} = (\alpha_{cv})_{real} + (\alpha_{nb})_{real} \tag{5}$$

where $(\alpha_{cv})_{real}$ and $(\alpha_{nb})_{real}$ that we developed here are shown in Table A2.3. In this Table, as the correlation equation of Goto et al. for single-phase turbulent flow heat transfer coefficients, which was used to calculate $(\alpha_{cv})_{real}$, is correlated by the heat transfer coefficient calculated based on the maximum inner diameter, it must be based on the actual heat transfer area, and it should be noted that the value of numerical subscript "real" is calculated based on the actual heat transfer area and that of "max" based on the maximum inner diameter. As for the constants C_1 and C_2 in the equation related to $(\alpha_{nb})_{real}$, the heat transfer characteristics of five types of grooved tubes from G1 to G5 could be best correlated with each other, if C_1=26×10^{-5} and C_2=2.7.

For reference, with the grooved tube G1 as a representative example, the relations between the quality of the predicted values and experimental values calculated by the correlation equation of heat transfer are shown in Fig. A2.14. The symbol open circle in the figure represents an experimental value, and the solid lines represent predicted values of heat transfer coefficients, the broken lines represent the contributions of nucleate boiling heat transfer in the predicted values of heat transfer coefficients. As for the predicted values of heat transfer coefficients, the contribution of nucleate boiling heat transfer accounts for the most part of the region and the contribution of forced convection heat transfer accounts for little in the low quality region. The contribution of nucleate boiling decreases in proportion to the increase in quality. Under the pressure of 3 MPa, if the quality is about 0.8, the contribution of nucleate boiling is very small.

Table A2.3 Correlation of boiling and evaporative heat transfer of CO_2 in spiral grooved tubes

(a) Contribution of forced convection heat transfer: $(\alpha_{cv})_{real}$

$$(\alpha_{cv})_{real} = F(\alpha_1)_{real}$$

where,

$$F = 1 + 2.0 X_{tt}^{-0.88}, (\alpha_1)_{real} = \frac{(\alpha_1)_{eq}}{\eta_{eq}} = \frac{(\alpha_1)_{max}}{\eta_{max}} = \frac{(\alpha_1)_{max}}{\eta_{eq} \cdot (d_i / d_{max})},$$

$$(\alpha_1)_{max} = 0.031 \mathrm{Re}_1 \mathrm{Pr}_1^{1/3} \eta_{max}^{0.8} (\cos\beta)^{-1.3} \left\{1 + (10p/d_{max})^{2.3}\right\} \left(\frac{\lambda_1}{d_{max}}\right), \mathrm{Re}_1 = \frac{G(1-x)d_{max}}{\mu_1}$$

(b) Contribution of nucleate boiling heat transfer: $(\alpha_{nb})_{real}$

$$(\alpha_{nb})_{real} = K^{0.745} S(\alpha_{pb})_{real}$$

where,

$$K^{0.745} = \frac{1}{1 + 0.768\eta + 0.351\eta^2 + 0.347\eta^3 + 0.131\eta^4}, \eta = \frac{(\alpha_{cv})_{real}}{S \cdot (\alpha_{pb})_{real}}, S = \frac{1 - e^{-\zeta}}{\zeta}, \xi = \frac{D_b \cdot (\alpha_{cv})}{\lambda_1}$$

$$D_b = C_1 \left(\frac{\rho_1}{\rho_v} \frac{Cp_1}{\Delta h} T_{sat}\right)^{1.25} \left(\frac{2\sigma}{g(\rho_1 - \rho_v)}\right)^{0.5}, (\alpha_{pb})_{real} = C_2 \times 207 \left(\frac{\lambda_1}{D_{be}}\right) \left(\frac{q_{real} D_{be}}{\lambda_1 T_{sat}}\right)^{0.745} \left(\frac{\rho_v}{\rho_1}\right)^{0.581} \mathrm{P}$$

$$d_{be} = 0.51 \left\{\frac{2\sigma}{g(\rho_1 - \rho_v)}\right\}^{0.5}$$

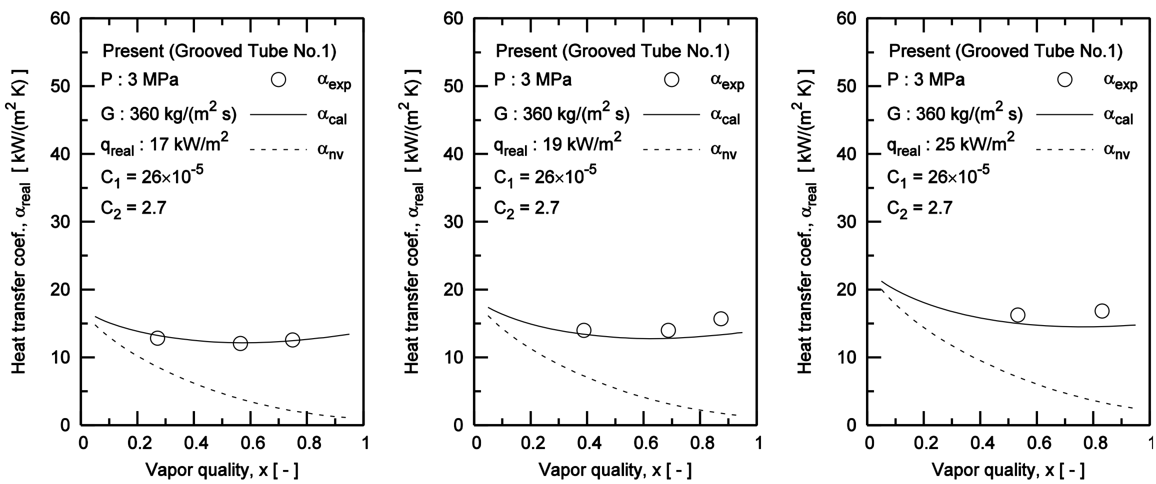


Fig. A2.14 Relations between heat transfer coefficients and quality (comparison between experimental values and correlation equation of heat transfer)

Nomenclature

C_1, C_2	: constant	[-]
c_p	: isobaric heat	[kJ/(kg K)]
D_b	: bubble departure diameter	[m]
D_{be}	: equilibrium break-off diameter	[m]
d_i	: average inner diameter	[m]
$d_{\max}$	: maximum inner diameter	[m]
d_o	: outer diameter	[m]
F	: two-phase multiplier factor	[-]
G	: mass velocity	[kg/(m^2 s)]
g	: gravitational acceleration	[m^2/s]
h	: specific enthalpy	[kJ/kg]
K	: function of η	[-]
Nu	: Nusselt number	[-]
P	: pressure	[MPa]
p	: fin pitch	[m]
Pr	: Prandtl number	[-]
q	: heat flux	[kW/m^2]
Re	: Reynolds number	[-]
S	: suppression factor	[-]
T	: temperature	[K or $^{\circ}$C]
x	: quality	[-]
α	: heat transfer coefficient	[kW/(m^2 K)]
β	: lead angle	[deg]
η	: parameter of nucleate boiling	[-]
η_a	: area expansion ratio	[-]
λ	: thermal conductivity	[W/(m K)]
μ	: viscosity	[Pa s]
ξ	: parameter of nucleate boiling	[-]
ρ	: density	[kg/m^3]
σ	: surface tension	[N/m]
X_{tt}	: Martinelli parameter	[-]

Subscripts

b	: value at the bulk fluid
cv	: convection heat transfer contribution term
exp	: experimental value
f	: value at the film temperature
l	:liquid
nb	: nucleate boiling heat transfer contribution term
pb	: pool boiling
real	: based on real surface area
sat	: saturation state
v	:vapor

A2.5 Experimental Study of Carbon Dioxide Flow Boiling Heat Transfer Coefficient in Horizontal Grooved Tubes.

Katsumi Hashimoto (Central Research Institute of Electric Power Industry)
Naoe Ssasaki (Sumitomo Light Metal Industries, Ltd.)

A2.5.1 Introduction

Protection of the environment is becoming more important globally, and considerable effort is being spent on developing heat pumps with natural working fluids. After surveying the status of R&D on heat pumps with natural working fluids, at the Central Research Institute of Electric Power Industry (CRIEPI) focused on a CO_2 heat pump, upon which a basic study was commenced in 1995. In Japan, the world's first CO_2 heat pump for residential hot water heaters was developed in a joint research project involving DENSO, the Tokyo Electric Power Company and CRIEPI, and was released as a commercial product in May 2001 (Saikawa et al. 2000). With other manufacturers joining the market, this added impetus to the residential hot water heater market. Total shipments in the 2008 fiscal year numbered about 500,000 units, with the figure since 2001 exceeding 1,500,000 in 2008. The Japanese government has a strong interest in this technology because of its potential to save energy and abate greenhouse gases. To develop a more efficient and compact heat pump, the evaporator needs to perform better and be smaller than a conventional evaporator. It is important to accurately measure the flow boiling heat transfer coefficient and to understand the heat transfer mechanism. In our previous report, it is understood that mean heat flux, like mass velocity, considerably affects the mean evaporating heat transfer coefficient in a horizontal smooth tube (99.999% purity, without oil, outer diameter 6 mm, thickness 0.4 mm). In this report, the heat transfer coefficient and pressure drop of a smooth tube and 3 kinds of grooved tubes are measured in a collaborative study between Sumitomo Light Metal Ltd. and CRIEPI in order to find suitable inner groove form for carbon dioxide.

A2.5.2 Experimental Apparatus

1. CO_2 Heat Pump Loop

Fig. A2.15 shows a schematic diagram of the experimental apparatus in CRIEPI. The apparatus consists of a compressor, three gas coolers, two electro motion expansion valves and two evaporators. The compressor is an oil-free reciprocating model, driven by a variable speed motor (inverter driven). This apparatus has heat source and heat sink equipment. Brine (antifreeze liquid) is used as the heat source medium and water as the heat sink medium, while the temperatures and flow rates of these fluids are controlled automatically.

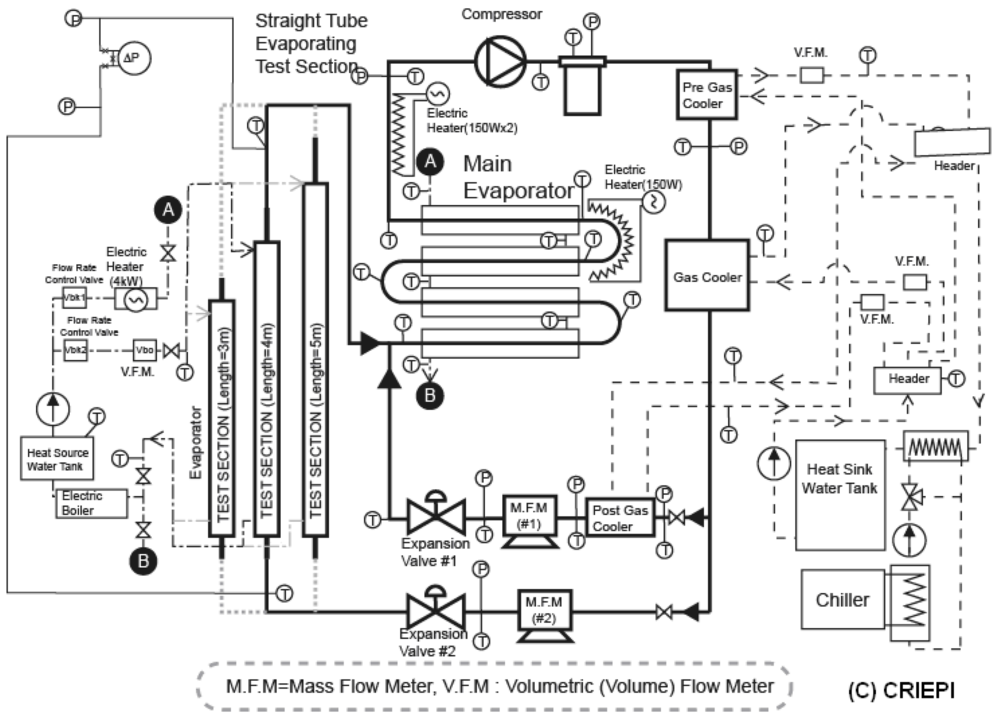


Fig. A2.15 Schematic Diagram of the CO_2 Heat Pump Loop

2. Details of the Test Section

Main evaporator and straight tube evaporating test section (test section) were connected serially. Two expansion valves made it possible to control mass flow rate into the test section and pressure of the test section. Three different length of test section are prepared and connected in parallel (temperature and pressure measuring devices are common). Fig. A2.16 shows details of the test section, which consists of double tubes. CO_2 was blown into the inner tube and brine into the annular part in a counter-current flow. Inner tube is testing tube made of copper. Tubes' outer diameter (O.D., d_I) is 6mm and length is 4m (results of other length are described in relevant publication). Tube length is corresponded to the distances between the two electrodes for voltage measurement. Testing tube and piping are fully isolated electrically at the electrical insulating joints on both ends of the tube. Outer tube is made from transparent polyvinyl chloride and had an O.D. of 22 mm and an inner diameter of 12 mm.

The CO_2 temperature and pressure on the inlet and outlet of the test section are measured. In addition, pressure difference between the inlet and outlet is measured too. The mass flow rate of CO_2 into the test section can be measured independently of the main mass flow rate, as shown in Fig. A2.15. Brine temperatures are measured at the inlet and outlet of the test section and the volumetric flow rate is also measured at the inlet. The thermometers were high-accuracy electric-resistance thermometers, calibrated using a standard mercury-in-glass thermometer.

The wall temperature of the inner tube, T_w, is measured using the electric-resistance method as shown in Fig. A2.16. A DC current supplier passed an almost constant DC current through the copper tube. The value of the DC current (I) was checked by measuring the voltage of standard resistance (1mΩ±0.002mΩ) using a high-accuracy digital multi-meter (DMM). The value of the voltage (E) of the copper tube was measured by high-accuracy DMM, too. The value of the resistance of the tube (R) was calculated by E and I. Meanwhile, the correlation of the copper tube resistance and temperature was estimated in advance, in the vacuum conditions in the copper tube, each time the experiment was set up. Wall temperatures involved in this method are flow directional and the circumferential average temperature, although this is believed to be more accurate than a surface thermometer.

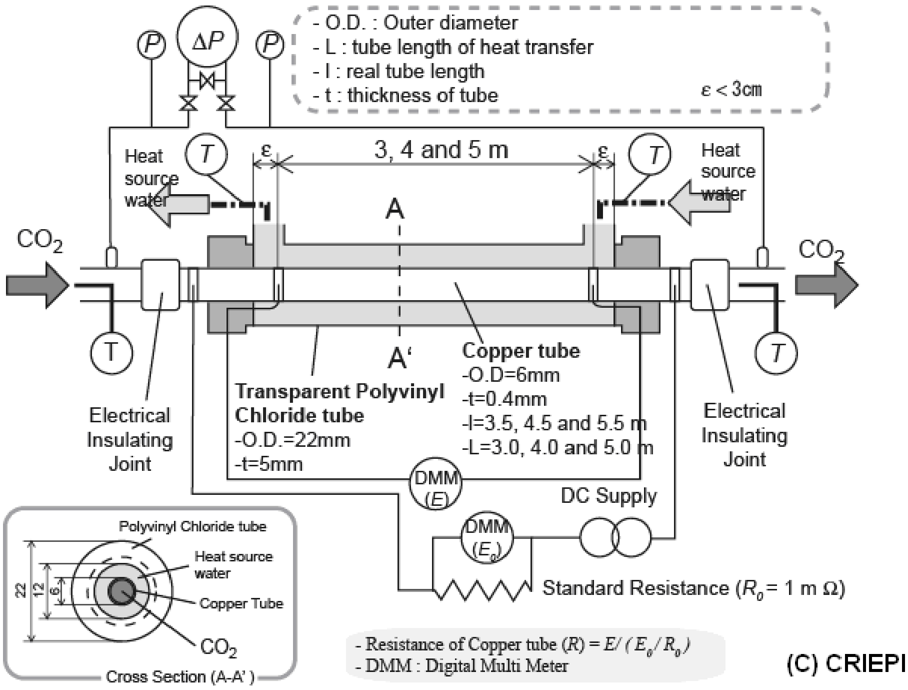


Fig. A2.16 Schematic Diagram of the Evaporating Heat Transfer Test Section

3. Derivation of the Flow Boiling Heat Transfer Coefficient

All experimental data is taken from a steady state for over 20 minutes (data accumulated every two seconds), and averaged for analysis. The value of the flow boiling heat transfer coefficient is calculated via the following process:

Amount of heat transfer from brine, Q_b, is calculated by eq. (1),

$$Q_b = \rho_b \, Cp_b \, V_b \, (T_{b,i} - T_{b,o}) \tag{1}$$

Reference temperature of CO_2, T_r, is the mean value between the inlet and outlet saturation temperatures; calculated by the pressure. The reference temperature of the brine, T_b, is the mean value between the inlet and outlet temperatures.

The mean heat transfer coefficient of CO_2 is defined as eq. (2).

$$h_{reqvEX} = \frac{Q_b}{S_{eqv}\left(T_w - T_r\right)} \tag{2}$$

Where, heat transfer area is determined based on an equivalent diameter, d_{eqv}, defined as eq. (3).

$$S_{eqv} = \pi L d_{eqv} = \pi L\left(\sqrt{d_1^{\ 2} - 4U/\pi\rho_{Cu}}\right) \tag{3}$$

Where U is the unit weight of the tube.

Heat flux q_{eqv}, is defined based on equivalent heat transfer area. Mass velocity, G, is defined based on the equivalent diameter.

4. Experimental Conditions

The experimental conditions are: CO_2 mass velocity from about 160 to about 660 kg/(m^{2}s), pressures of CO_2 is 4.0MPa (saturation temperatures is 5^{o}C), as well as tube length, results of other experiment conditions are described in relevant publications). Quality at the inlet of the test section is of 0.17 and CO_2 super heat at the outlet of test section is of 5 K (two runs with different CO_2 super heat temperatures at outlet were performed keeping other parameters fixed, and interpolation done in order to estimate the heat transfer coefficient set super heat temperature to 5K).

5. Tube Characteristics

Three kinds of grooved copper tubes are made via the component rolling method (Iijima 1994). The groove is characterized by groove parameters as shown in Table A2.4 and Fig. A2.17, the former of which shows the groove parameters. "ST" means the smooth tube and "GT" means the grooved tube. d_l is the outer diameter of the tube, while the maximum diameter, d_m, is the diameter between the bottoms of the groove, d_e is the equivalent diameter based on the unit weight and defined as eq. (3). The depth of groove, H, the top angle of groove, γ, the number of grooves, N, and the lead angle, α, are shown in Fig. A2.17. ζ is defined as the ratio of the grooved tubes' wetting heat transfer area relative to the smooth tube. In Fig. A2.18, shadow photographs of the cross-section of each of the grooved tubes are shown

Table A2.4 Groove parameters

ID	d_l	d_m	d_e	H	α	γ	N	ζ
	(mm)	(mm)	(mm)	(mm)	(°)	(°)		
ST	6	5.2	5.20	0	0	0	0	1
GT2	6	5.2	5.10	0.155	14	15	70	2.01
GT3	6	5.2	5.06	0.210	14	15	60	2.17
GT10	6	5.2	5.01	0.220	20	15	80	2.83

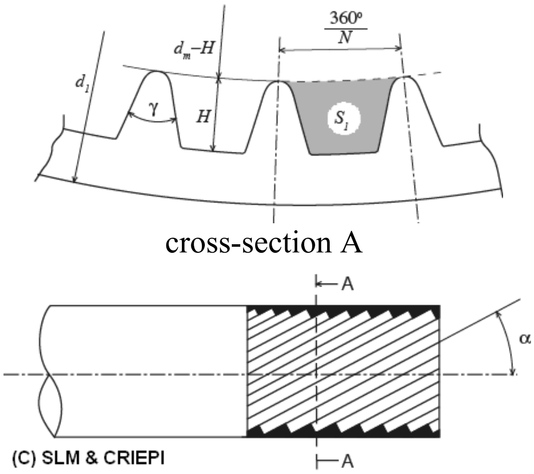


Fig. A2.17 Groove parameters

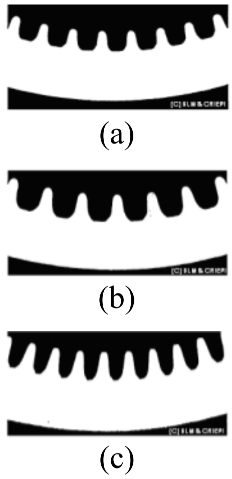


Fig. A2.18 Shadow photographs of groove tubes

A2.5.3 Results and Discussion

1. Pressure drop with mass velocity

In Fig. A2.19, the pressure drop of tubes, *dp/dz*, and mass velocity are represented as both logarithmic axes. The pressure drop, which is represented, is divided by the tube length. In this figure, only the pressure drop in 4.0MPa is displayed.

The pressure drop increases monotonously and the descending order of pressure drops is GT10, GT3, GT2 and the smooth tube, which is the same as that of ζ.

2. Mean heat transfer coefficient with mass velocity

In Fig. A2.20, the mean heat transfer and mass velocity are represented as both logarithmic axes, while the heat transfer coefficient is based on the equivalent heat transfer area as defined in eq. (3). It emerges that the heat transfer coefficient of a smooth tube has a peak value of around 400 kg/(m^{2}s), but that the heat transfer coefficient of grooved tubes increases monotonously. It has been reported that in the case of the R22, the heat transfer coefficient of grooved tubes is higher than that of smooth tubes throughout all mass velocities (Iijima 1994). In the case of CO_2, at a lower mass velocity region than 400kg/(m^{2}s), the heat transfer coefficients of grooved tubes and smooth tube are almost identical. In the higher mass velocity region, otherwise, the heat transfer coefficient of grooved tubes is higher than that of smooth tubes. It is assumed that the reason come from difference of flow pattern between fluorocarbon and CO_2. According to modified Baker diagram, in case of CO_2, until quality becomes 0.5, flow pattern is the slug flow and dose not become easily annular flow. On the other hand, in case of fluorocarbon, it is predictable that flow pattern becomes annular flow when quality is 0.2 or more. It is thought that heat transfer coefficient of CO_2 in grooved tube at lower mass velocity region is not higher than that of smooth tube by these flow pattern difference.

The descending order of the heat transfer coefficients of the grooved tubes is GT10, GT3 and GT2, which is the same as that of ζ.

In the previous report, we suggested that the reason why the mean heat transfer coefficient of the smooth tube has a peak value is due to dryout at lower quality. The mean heat transfer coefficient of grooved tubes increases monotonously, which is presumably because dryout does not occur until high quality is present. The reason why no dryout occurs with lower quality seems to be helical flow and liquid entering the gap. A helical groove induces the helical flow of refrigerant liquid, while the centrifugal force of the helical flow pushes the liquid onto the wall. In addition, small gaps between fins retain the refrigerant liquid. For these two reasons, refrigerant liquid can remain on the wall of grooved tubes until a higher level of quality than that of smooth tubes.

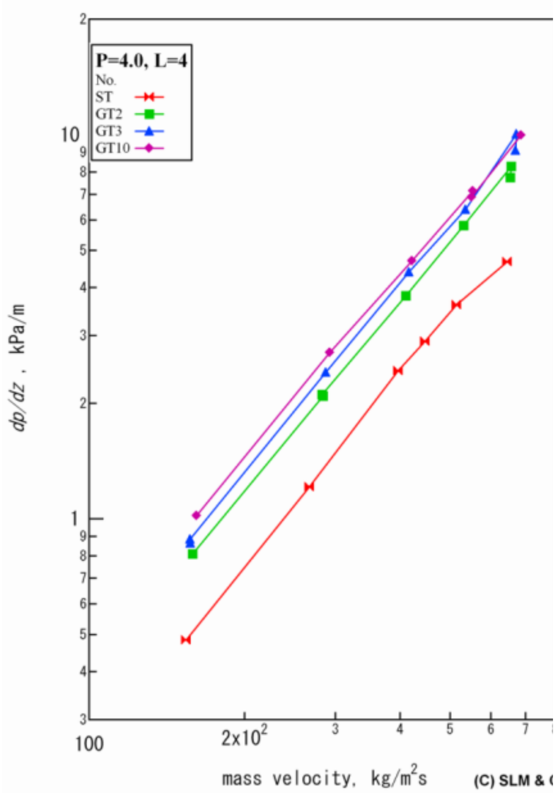


Fig. A2.19 Pressure drop with mass velocity

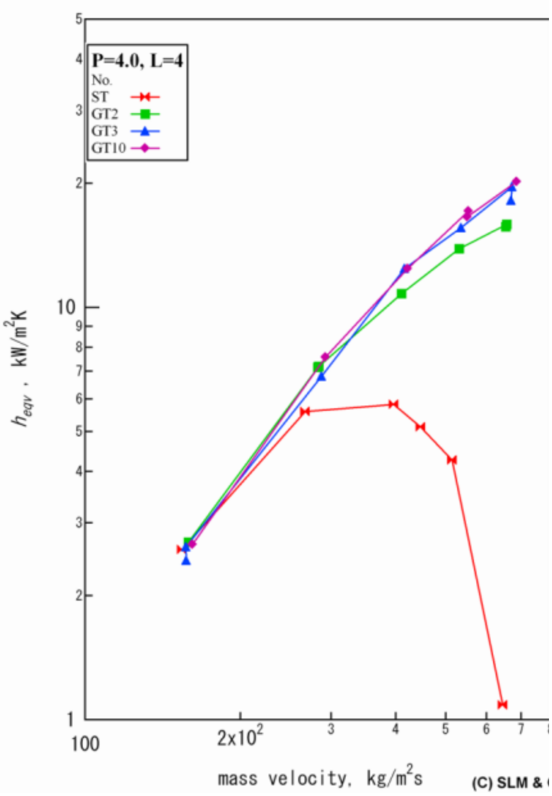


Fig. A2.20 Indicates the heat transfer coefficient with mass velocity

3. Suitable inner groove form for CO_2

We will discuss that suitable form for CO_2 flow boiling heat transfer enhancement. Unlike mean heat transfer coefficient of smooth tube, mean heat transfer coefficient of grooved tubes increases monotonously with the increment of mass velocity and dose not decrease at higher mass velocity region. From this fact, grooved tube is effective for heat transfer enhancement. Moreover, pressure drop of CO_2 in grooved tube increases monotonously with the increment of mass velocity. Here, we discuss pressure drop and heat transfer coefficient at certain value of mass velocity. Mass velocity in common use of supercritical heat pump water heater like ECO CUTE is about 400 kg/(m^{2}s). In Fig. A2.19 and Fig. A2.20, the difference of pressure drop is clear depending on groove form, but the difference of heat transfer coefficient is little. Therefore, GT2 is most suitable for CO_2 flow boiling heat transfer enhancement as far as we examined, because heat transfer coefficient is almost equal to others and pressure drop is obviously small. Feature of GT2 is depth of groove is comparatively small according to Table A2.4. It suggests that decrement of heat transfer coefficient will not occur if liquid of CO_2 exists on heat transfer surface. In the future, we would like to study the lowest depth of groove that CO_2 liquid film can maintain.

A2.5.4 Conclusions

The heat transfer coefficient and pressure drop of three kinds of groove tubes and a kind of smooth tube were measured via collaboration between Sumitomo Light Metal, Ltd. and CRIEPI and the following conclusions were obtained:

1) Pressure drop increases according to the increment ζ of tubes,
2) The heat transfer coefficient of the grooved tubes increases monotonously, namely, does not decrease in a region with a mass velocity exceeding 400 kg/(m^{2}s) like that of the smooth tube,
3) The heat transfer coefficient of grooved tubes increases according to ζ, and
4) Suitable inner groove form is GT2 as far as we examined in this report.

Nomenclature

A	heat transfer area, m^2	N	number of grooves
c_p	specific heat capacity, kJ/(kgK)	P	pressure, MPa
d	diameter, m	Pr	Prandtl Number
E	electrical voltage, V	Q	amount of heat transfer, kW
G	mass velocity of refrigerant (CO_2), kg/(m^{2}s)	q	heat flux, kW/m^2
		R	resistance, Ω
h	heat transfer coefficient, kJ/kg	Re	Reynolds Number
H	depth of groove, mm	S	cross section area of CO_2 path, m^2
I	electrical current, A	t	inner tube thickness, m
K	overall heat transfer coefficient, W/m^{2}K	T	temperature, ^{o}C
		V	volumetric flow rate, m^3/s
L	length of tube, m	x	quality
L	latent heat, kJ/(kgK)		
[Greek]			
α	lead angle of groove, degree	μ	dynamic viscosity, Pa-s
Δt_m	log. mean temperature diff., K	λ	thermal conductivity, kW/(mK)
η	heat balance	σ	surface tension, N/m
ρ	density, kg/m^3	ζ	area expansion ratio
[Subscripts]			
1	outer of inner tube	l	liquid
2	inner of outer tube	m	maximum inner diameter
b	heat source medium (brine)	o	outlet

CO	based on Colburn correlation	*r*	refrigerant (CO_2)
EX	based on wall temperature	*v*	vapor
eqv	equivalent inner diameter	*w*	wall
i	inlet		

A2.6 Development of World's first CO_2 heat pump water heater (ECO CUTE) and compact heat exchanger for water heating.

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T. Kobayakawa and K. Kusakari (Tokyo Electric Power Company)
M. Ito, H. Sakakibara, T. Okinotani, N. Kawachi, and K. Yamamoto (DENSO Corp.)

A2.6.1 Introduction

In Japan, energy demand for hot tap water accounts for about 30 % of the total residential final energy consumption, but most of this demand (over 90 %) is met by the direct combustion of fossil fuel. The development of high-performance heat pump water heaters using natural working fluids is thus eagerly anticipated for energy conservation and greenhouse gas reduction. If the COP of a heat pump water heater is 3 or more, the heat pump water heater uses about 30 % less primary energy than a combustion water heater.

The Central Research Institute of Electric Power Industry (CRIEPI) has been studying heat pumps using CO_2 as a refrigerant since 1995, and has found by theoretical analysis that the CO_2 cycle has unique characteristics and can achieve a higher COP than conventional refrigerants for domestic hot water production, and by experiments that the CO_2 cycle can be effectively controlled by the combination of an automatic expansion valve and a variable-speed compressor.

A2.6.2 Joint development of a CO_2 heat pump water heater

Using CO_2 heat pump technologies developed independently by CRIEPI and Denso (Denso possessing compressor technologies for car air-conditioning systems), a CO_2 heat pump water heater for residential use was jointly developed by the Tokyo Electric Power Company (TEPCO), Denso Corporation and CRIEPI, starting in 1998. In this collaboration, TEPCO mainly provided the concept for the product, while the main Denso contributions were the research and development of components and methods of improving performance of heat pump water heater. CRIEPI concentrated on evaluation of the prototype and finding ways to improve performance.

1. Modification of the prototype and improving its performance

Components such as the compressor and heat exchanger were developed and modified, and a prototype that was assembled from these components was installed in the test chamber and tested. As a result, COP improved from 2.1 (first prototype) to 3.4 (final prototype) under winter conditions (ambient temperature: 8°C, tap water temperature: 8°C and heated water temperature: 65°C). Fig. A2.21 shows test results for both prototypes in a *T-s* (temperature - specific entropy) diagram. The final prototype *T-s* gradient of the compression stage was steeper than that of the first prototype (in Fig. A2.21 (a)) because the compression efficiency increased.

Outlet CO_2 temperature of the heat exchanger between CO_2 and water (CO_2 gas cooler) of the first prototype was relatively high as shown in Fig. A2.21 (b). It is effective to lower the outlet temperature of CO_2 for the improvement of COP.

Because the heat exchanger was double tube type, the temperature was not able to be lowered easily. That's why we launched to create new and favourable style heat exchanger for supercritical CO_2.

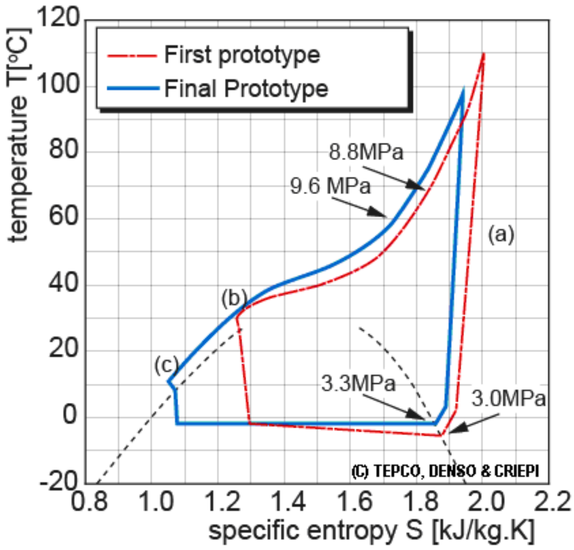


Fig. A2.21 *T-s* diagram of the final prototype in comparison with the first one.

2. Creation new and favourable heat exchanger for water heating

There were many requirements for the heat exchanger of supercritical CO_2 and water such as high temperature efficiency, a small volume, a high pressure withstanding performance, a leakage detection performance, durability etc. at the same time. The supercritical CO_2 temperature gradually decreases by heat release. Therefore, the temperatures difference between CO_2 and water becomes smaller than that of condensation heat transfer. The heat transfer area was not able to be expanded in order to achieve a small volume heat exchanger. Using capillary tubes (inner diameter is 0.5mm) for CO_2 flow path achieved high withstand pressure of the heat exchanger and high heat transfer coefficient of CO_2. However, the pressure loss increases. In order to correspond to this issue, 150 capillary tubes were connected in parallel. The flow rate of water is too small to become turbulent flow because of high temperature difference heating. Therefore, heat transfer coefficient of water side is very low. Using plate with off set inner fin for water flow path achieved high heat transfer coefficient of water because of forward edge effect. In order to increase the durability of the heat exchanger, the height of the water path was adjusted to avoid blockage by the scale adhesion. To ensure high temperature efficiency, CO_2 and water flow was semi-counter current. With these technologies, small and high temperature efficiency heat exchanger, capillary type heat exchanger, as shown in Fig. A2.22 was created. This heat exchanger became great contribution to increment of COP of the heat pump as shown in the Fig. A2.21 (b) and (c) and achieved the one fifth volume of the conventional heat exchanger.

Moreover, the heat exchanger won the Technical Award of Heat Transfer Society of Japan in 2002.

A2.6.3. Evaluation of the final prototype and commercialisation as ECO CUTE

The annual average COP was evaluated using performance test results for the final prototype. The evaluated system COP, which includes the power input to the air fan and water pump, was 3.4. This value was better than the targeted value of 3.0. In addition, it was confirmed that the final prototype could produce hot water at 90°C at an ambient air temperature of -15°C. In May 2001, CO_2 heat pump water heater named ECO CUTE was commercialised as shown in Fig. A2.23. "ECO CUTE" is a name used by electric power companies and water heater manufacturers, and refers only to heat pump water heaters using CO_2 as a refrigerant. Because the energy saving and greenhouse abating potential of ECO CUTE were highly evaluated, ECO CUTE won various prizes.

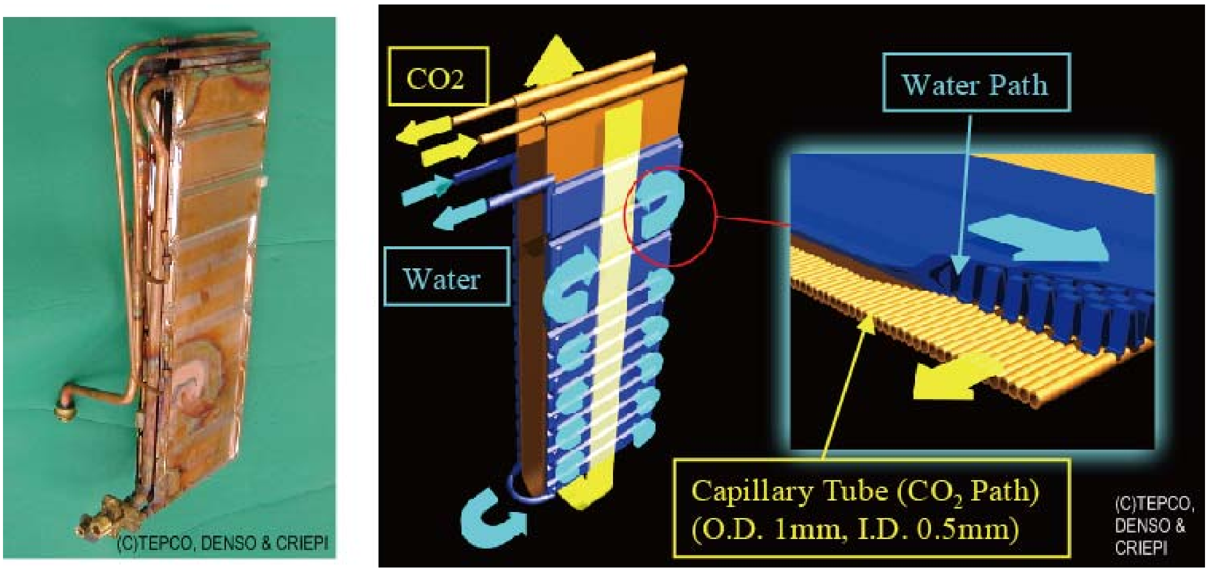


Fig. A2.22 Photograph and schematic diagram of capillary tube heat exchanger.

Left: Heat pump unit (81×32×65 cm), Heating capacity: 4.5 kW
Right: Hot water tank (109×45×152 cm), Capacity: 300 litters
Fig. A2.23 Photograph of ECO CUTE commercialized in May 2001.

A2.7 CO_2 Heat Pump Water Heater Gas-Cooler Enhancement Technology Enhancement Technology for CO_2 Heat Pump Water Heater Gas-Coolers

Kazuhiko Machida, Takumi Kida, and Shoichi Yokoyama (Panasonic Corp.)

A2.7.1 Introduction

The curtailment of CO_2 emission volumes has been called for in recent years from the perspective of protection of the global environment and the diffusion of CO_2 heat pump water heaters that offer greater energy saving and have a smaller impact on the global environment than conventional heating appliances is progressing. Given such circumstances, the increasingly higher efficiency of products is advancing at a rapid rate in association with the increasingly expanding market for CO_2 heat pump water heaters.

This report introduces examples of efforts made in recent years at Panasonic in regard to technology to increase the performance of gas coolers, which play a crucial role in the greater efficiency of CO_2 heat pump water heaters.

A2.7.2 The Basic Configuration and Heat Transfer Characteristics of Panasonic Gas Coolers

As shown in Fig. A2.24, Panasonic gas coolers employ a double tube heat exchanger involving two CO_2 refrigerant tubes that make up the CO_2 flow channels inside the water tube, and are configured so that the water and CO_2 flow in opposite directions to each other.

The two CO_2 refrigerant tubes are configured in a straight formation, referred to as TYPE-A below.

We used a block segmentation method [1] to understand heat transfer performance and forecast the heat transfer rates on the water and refrigerant sides. As shown in Fig. A2.25, we determined that the heat transfer rate on the water side is very small in comparison with the refrigerant side. Consequently, the main point in increasing the performance of our gas coolers was increasing the rate of heat transfer on the water side.

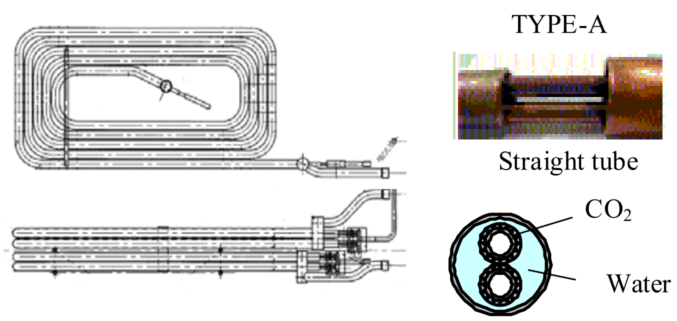


Fig. A2.24 Schematic view of the present gas-cooler

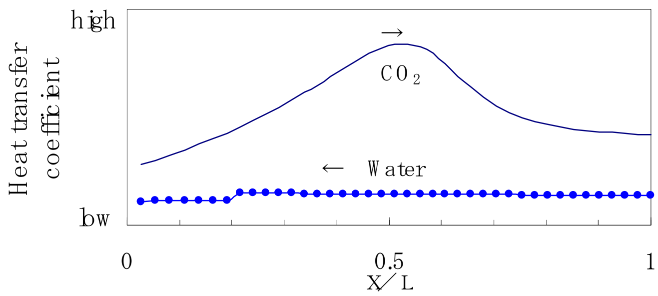


Fig. A2.25 Calculation data of heat transfer coefficient

A.2.7.3 Development Specifications

Aiming at targets such as the promotion of turbulence on the water side and the disturbance of the thermal boundary layer on the surface of the CO_2 refrigerant tubes, we developed heat transfer tubes in the 3 configurations (TYPE-B - D) shown in Fig. A2.26.

- Twist specifications (TYPE-B)

Configuration with two CO_2 refrigerant tubes twisted together

- Dimple and twist specifications (TYPE-C)

The two CO_2 refrigerant tubes of TYPE-B are processed to create dimples on the surface of the tubes.

- High density dimple and twist specifications (TYPE-D)

Configuration with twice as many dimples as TYPE-C

A2.7.4 Test Method

The hot water experimental apparatus used in the evaluation of the heat transfer tubes is shown in Fig. A2.27. The parts tested were tubes made using the specifications shown in Fig. A2.26 (heat transfer length 1m). Hot water was run through the tubes and cold water was run outside the tubes and the flow volume within the tubes varied. The ratio of heat transfer outside of the tube was calculated using the Wilson plot method.

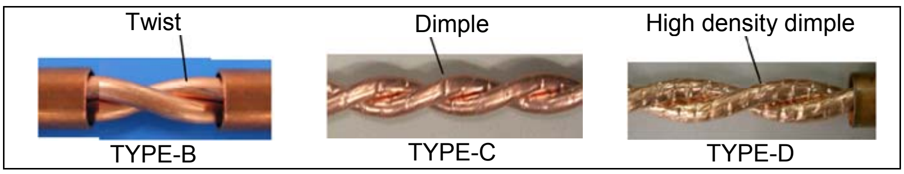


Fig. A2.26 New enhanced tubes.

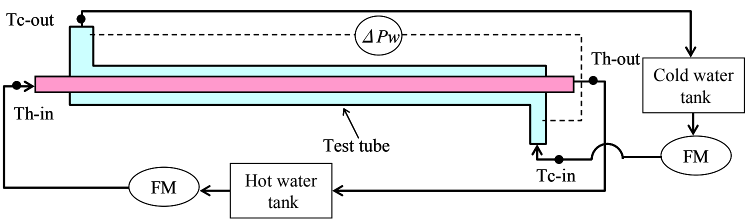


Fig. A2.27 Experimental apparatus.

A2.7.5 Development Results

1. Twist specifications

Figs A2.28 and A2.29 show the heat transfer ratio α_w and pressure drop ΔP_w characteristics of the TYPE-B (twist) tube against the TYPE-A tube (straight). The heat transfer ratio α_w of the TYPE-B tube in the Reynolds number range Re_w: 1000 - 3000, which is the gas cooler usage range, has increased in comparison to the heat transfer ratio α_{w0} of the TYPE-A tube. In addition, although the pressure drop ΔP_w of the TYPE-B tube has also increased in comparison to the pressure drop ΔP_{w0} of the TYPE-A tube, because α_w has increased more than this, the TYPE-B tube can be described as a useful means for the promotion of heat transfer. Fig. A2.30 shows water temperature distribution using a cross-section of the water flow channel outlet analyzed with general purpose thermal fluid software. The TYPE-B tube has less heat irregularity than the TYPE-A tube so it is understood that the water has been disturbed well.

2. Dimple and twist specifications

Figs A2.31 and A2.32 show heat transfer ratio α_w and pressure drop ΔP_w characteristics of the TYPE-C and TYPE-D tubes against the TYPE-A tube. The α_w of the TYPE-C and TYPE-D tubes are higher, in sequence, than the TYPE-B tube. In spite of this, the pressure drop ΔP_w of the TYPE-C and TYPE-D tubes show roughly the same values as for the TYPE-B tube so the TYPE-C and TYPE-D tubes are very effective as a means of promoting heat transfer on the water side. Furthermore, the distribution forecast values for the heat transfer ratios of gas coolers fitted with tubes configured to these specifications are shown in Fig. A2.33. The TYPE-D specifications are able to improve on the water side heat transfer ratio by a factor of about 2.0 in comparison to the TYPE-A specifications and have contributed significantly to the enhanced performance of Panasonic gas coolers.

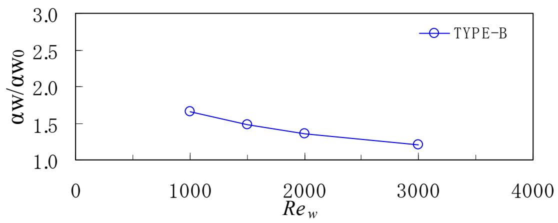


Fig. A2.28 Heat transfer characteristic of TYPE-B

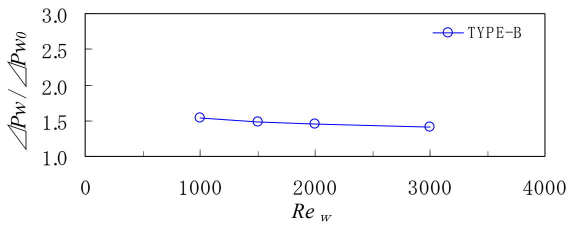


Fig. A2.29 Pressure drop characteristic of TYPE-B

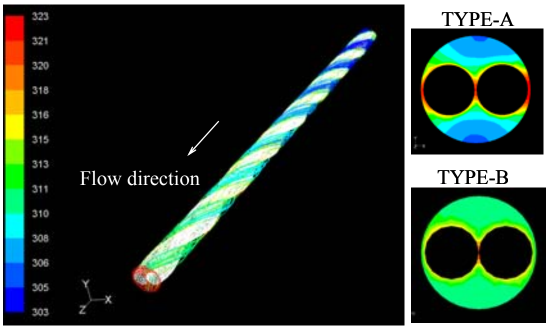


Fig. A2.30 Water temperature distribution

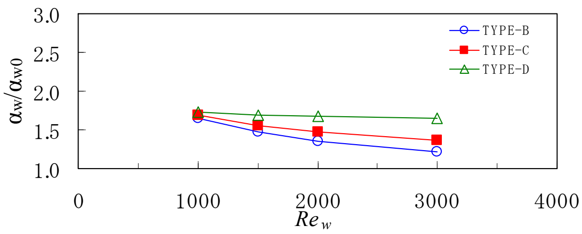


Fig. A2.31 Heat transfer characteristic

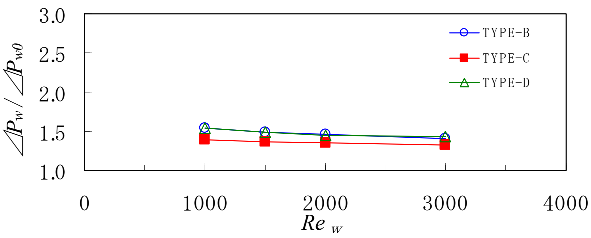


Fig. A2.32 Pressure drop characteristic

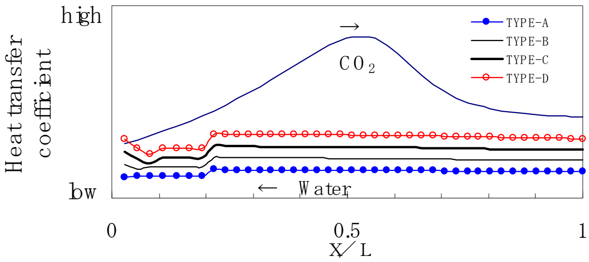


Fig. A2.33 Calculation data of heat transfer coefficient

A2.7.6 Conclusion

We developed [1] twist, [2] dimple and twist, and [3] high density dimple and twist specifications for heat transfer tubes as an undertaking to enhance the performance of gas coolers. In particular, with the high density dimple and twist specifications, it is possible to improve on the straight tube water side heat transfer ratio by a factor of about 2.0 while keeping pressure drops at low levels. This has contributed significantly to the enhanced performance of Panasonic gas coolers.

Acknowledgements

This research describes results obtained based on a joint research contract with the New Energy and Industrial Technology Development Organization (NEDO).

A2.8 Heat Transfer Issues in Fin and Tube Heat Exchangers

Akio Miyara (Saga University)

A2.8.1 Introduction

Fin and tube heat exchangers are widely used in air-conditioners and refrigeration systems. For higher COP, heat exchangers which have high heat transfer performance and low pressure loss are desired. Heat transfer resistances of the fin-tube heat exchanger are (1) the convective heat transfer resistance of refrigerant flowing inside the tube, (2) the thermal contact resistance between tubes and fins, and (3) the convective heat transfer resistance of air flowing through fins. It is said that degrees of the resistances are 15~20% for the refrigerant side, 10~15% for thermal contact of tube and fin, and 70~75% for air-side. In the last decades, the efforts have been poured into development of convection heat transfer enhancement for refrigerant and air sides. The air side heat transfer has been considerably enhanced by wavy, louver, and offset fins. The heat transfer enhancements of condensing or evaporating refrigerant have been exclusively achieved by microfin tubes, because they give considerable improvement of heat transfer coefficient accompanying small increase of pressure loss. Although the contact resistance is relatively small, it becomes not to be ignored because other resistances are reducing.

A2.8.2 Heat transfer enhancement by microfin tubes

Microfins have four kinds of effect on heat transfer enhancement, such as (1) augmentation of heat transfer area, (2) generation of turbulence, (3) thinning of liquid film, and (4) feed of liquid. The heat transfer enhancement mechanism differs with flow conditions. The film thinning effect plays the important role on the enhancement of condensation and evaporation. In the case of condensation, the condensate covers the entire fin surface and the film becomes thinner around corner of microfins by the surface tension effect. Larger heat transfer is achieved at the thin film area. For the evaporation, meniscus film is formed in the groove and very thin films exist at the tips of the meniscus film where the evaporation is very active and liquid is supplemented to the tips by the surface tension effect. In the case of single phase flow, the area augmentation and turbulence generation enhance the heat transfer. Relation between fin height and thicknesses of viscous sublayer and thermal boundary layer are important for heat transfer enhancement and pressure drop.

In the case of herringbone microfin tube, microfins work to remove liquid at fin-diverging parts and collect liquid at fin converging parts. For condensation, microfins remove condensed liquid and keep very thin film then the heat transfer coefficients are higher than those of smooth and helical microfin tubes. For evaporation, the liquid removal would cause lack of liquid and start earlier dryout. However, the evaporation heat transfer coefficient of the herringbone tube shows higher than that of the helical microfin tube when the refrigerant mass velocity is sufficiently large and tube has suitable fin shape. The reason is that considerable droplets are entrained into vapor flow and four circulation flow generated by herringbone fins convey the droplets to tube side and the dryout is avoided.

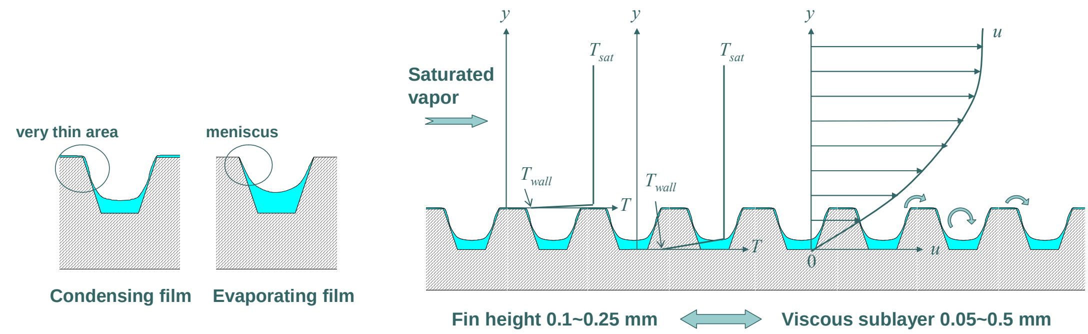


Fig. A2.34 Liquid film formed around fins and boundary layers

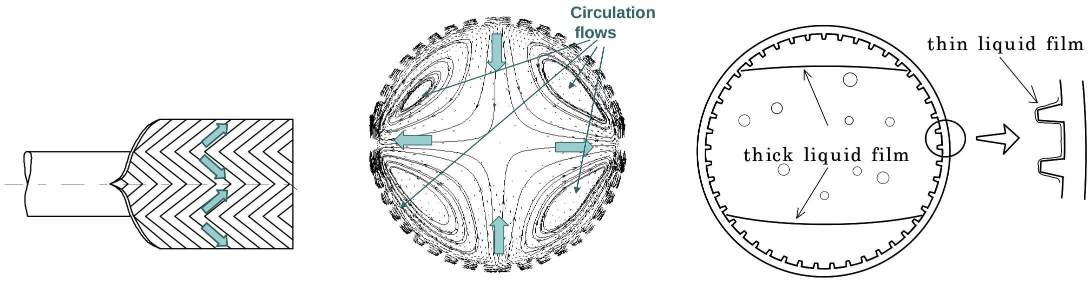


Fig. A2.35 Liquid separation and circulation flows by the herringbone fins

A2.8.3 Thermal contact resistance between tubes and fins

Thermal contact resistance consists of contact surface and triangular space formed by tube and fins, where the tube and fin are not contacted. The triangular space is more dominant in the contact resistance. A protrusion of fin collar affects on the air-side heat transfer. A numerical simulation has been conducted by using a commercial code FLUENT. Increasing the form ratio, which indicates the rate of non-contact area to total area, the total thermal resistance, consisted of the contact resistance and air-side convection resistance, increases according to the increase of the contact resistance. The contact resistance of 20% form ratio is 17 % of total resistance and that of 70 % form ratio is 33 % of total resistance.

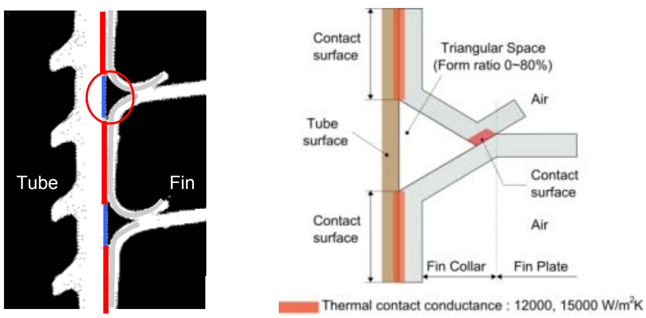


Fig. A2.36 Increase of thermal resistance with increase of form ratio

A2.9 Development of a New Printed Circuit Heat Exchanger

Yasuyoshi Kato (Tokyo Institute of Technology)

A2.9.1 Introduction

A new Printed Circuit Heat Exchanger (PCHE) has been proposed as a Micro Channel Heat Exchanger (Fig. A2.37) through three dimensional thermal-hydraulic simulations for recuperators of a carbon dioxide gas turbine cycle, which has discontinuous “S”-shape fins and provides flow channels with near sine curves (Fig. A2.38). Its pressure drop is one-sixth reference to the conventional PCHE with zigzag flow channel configuration while the same high heat transfer performance inherits (Fig. A2.39). Thermal-hydraulic performance of the new PCHE has been studied parametrically changing the “S” shape fin design parameters: fin angle, fin width, size, and edge sharpness. The pressure drop reduction is ascribed to elimination of recirculation flows and eddies that appears around bend corners of zigzag flow channels in conventional PCHE [1]. Moreover, Empirical correlations of heat transfer and pressure drop performance have been proposed for the micro channel heat exchanger for a hot water supplier with S-shaped fins, using supercritical carbon dioxide as the heating medium [2]. The test section was constructed from flat copper plates that had fluid flow passages chemically etched on their surfaces. The plates were then diffusion-bounded together into a block to from a heat exchanger core. The new PCHE has core dimensions of 78 x 1000 x 14 mm as shown in Fig. A2.40, and Fig. A2.41 shows that the flow channels take sine-wave forms by S-shaped fins. Total numbers of flow channels are 54 and 4, respectively, for the hot and cold side. This PCHE has one H_2O and two CO_2 plates which are arranged in a sandwich structure in what we call a double banking model, as illustrated in Fig. A2.42. The fluid flow of H_2O and CO_2 is arranged as counter-current flow. The fin design parameters are summarized in Table A2.5.

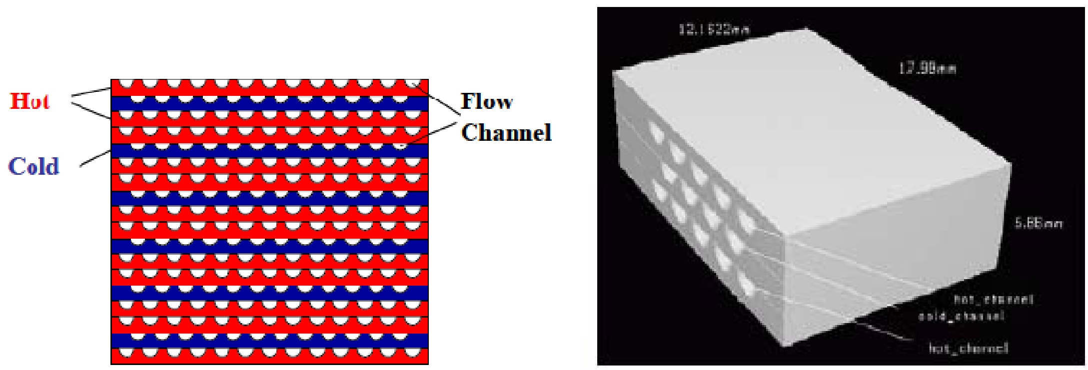


Fig. A2.37 Microchannel heat exchangers.

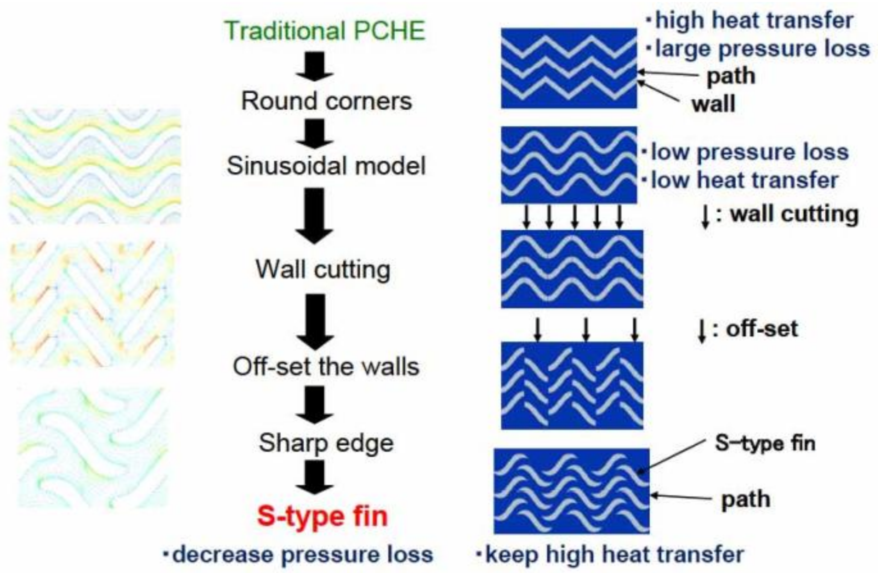


Fig. A2.38 Types of microchannel heat exchangers.

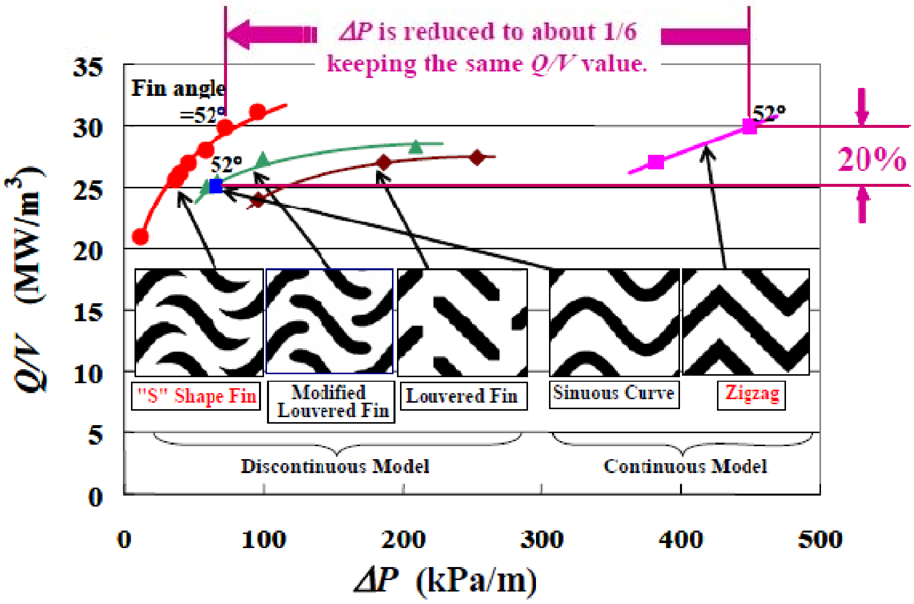


Fig. A2.39 Heat transfer rate vs. pressure drop.

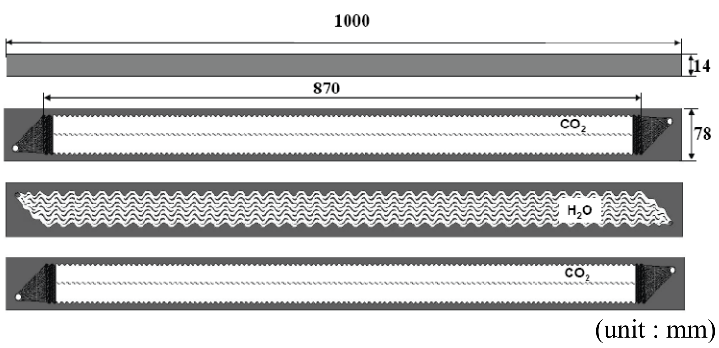


Fig. A2.40 Configuration of new PCHE

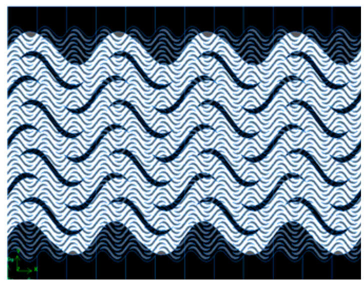


Fig. A2.41 Superimposition of channels

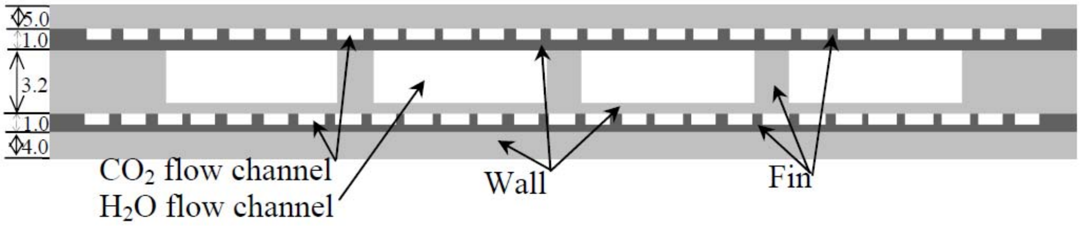


Fig. A2.42 Cross-sectional View of PCHE

Table A2.5 Specification parameters of PCHE

Items	CO_2 side	H_2O side
Dimensions (mm)	W78×H1000×D14	
Fin angle	52°	52°
Fin length, (mm)	4.8	14.4
Fin width, (mm)	0.4	1.2
Fin depth, (mm)	0.47	2.5
Fin gap, (mm)	0.87	6.17
D_h, (mm)	0.59	3.40
Number of channels	54	4
Flow length, (mm)	1455	1455
Heat transfer area, A (m^2)	0.225	0.109
Free flow area, A_c (m^2)	22.1×10^{-6}	61.7×10^{-6}

A2.10 Numerical Analysis on Laminar Film Condensation of Pure Refrigerant Inside a Vertical Rectangular Channel

Shigeru Koyama and Tatsuya Matsumoto (Kyushu University)

A2.10.1 Introduction

Plate-fin heat exchangers are widely used in chemical plants for their high heat transfer performance, and are taken a great interest in as heat exchangers for heat pump and refrigeration systems. Many researchers have studied this kind of heat exchanger in single phase region in detail, but the almost did not extent their research to two-phase flow region.

This study aims to clarify the basic characteristics of heat transfer in a vertical plate-fin condenser, where refrigerant flows downward and cooling water flows counter-currently. The numerical analysis for the free convective laminar film condensation of pure refrigerant on a vertical finned surface is carried out using a simplified model for shapes of fin and liquid film to clarify the relation between heat transfer characteristics and dimensions of the fin.

A2.10.2 Numerical model and basic equations

Fig. A2.43 shows the simplified model of refrigerant path in a vertical plate-fin condenser, where g is the gravitational acceleration, p the fin pitch, h the fin height, t the fin thickness, and r_0 the radius of concave joint region of plate and fin; For the symmetry of plate and fin in the refrigerant path, a quarter of one path, in which fin and plate surfaces are shaded, is only treated as a calculation domain in the present numerical analysis.

Fig. A2.44 shows the model of liquid film on the vertical finned surface. In the figure L and δ denote the fin length and the liquid film thickness, respectively. The coordinates for liquid film are z in the gravitational direction, s in the horizontal direction along the fin surface and y in the direction perpendicular to the fin surface. For the heat conduction in the fin, the coordinates from the symmetric point are X in the direction of fin height and Y in the direction perpendicular to X. The main assumptions in the numerical analysis are as follows:

(1) The condensed liquid film is laminar filmwise.
(2) The vapor shear stress at the vapor-liquid interface is negligible.
(3) The pressure in the liquid film is determined by the relation between the surface tension σ and the radius of the liquid film surface in the horizontal direction r.
(4) The liquid film flow is controlled by not only the gravitational force but also the surface tension.

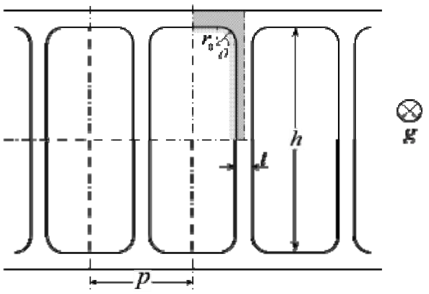


Fig. A2.43 Simplified model of a plate-fin

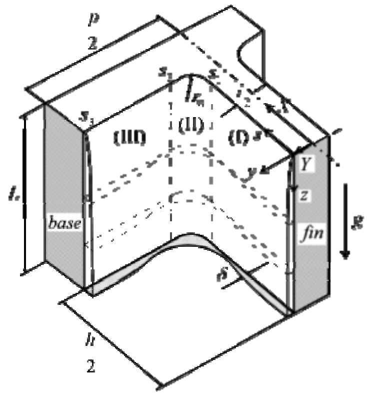


Fig. A2.44 Liquid film model on a finned surface

The governing equations for the liquid film are as follows:

$$\frac{\partial u}{\partial s}+\frac{\partial v}{\partial y}+\frac{\partial w}{\partial z}=0 \qquad \text{(mass conservation)} \qquad (1)$$

$$\frac{\partial^2 u}{\partial y^2}=\frac{1}{\mu_l}\frac{\partial P}{\partial s} \qquad \text{(s-direction momentum conservation)} \qquad (2)$$

$$\frac{\partial^2 w}{\partial y^2}=-\frac{1}{\mu_l}\rho_l g \qquad \text{(z-direction momentum conservation)} \qquad (3)$$

$$\frac{\partial^2 T}{\partial y^2}=0 \qquad \text{(energy conservation)} \qquad (4)$$

where u, v and w denote velocity components in s, y and z coordinates, respectively, P is the pressure in the liquid film, T is the temperature, and μ_l and ρ_l are the viscosity and the density of liquid, respectively. The boundary conditions to solve equations (1) to (4) are as,

$$y=0:\ u=0,\ v=0,\ w=0 \qquad (5),(6),(7)$$

$$T=T_w \qquad (8)$$

$$y=\delta\ :\ \frac{\partial u}{\partial y}=0,\ \frac{\partial w}{\partial y}=0 \qquad (9),(10)$$

$$\dot{m}=\rho_l\left(u\frac{\partial\delta}{\partial s}+w\frac{\partial\delta}{\partial z}-v\right) \qquad (11)$$

$$\lambda_l\frac{\partial T}{\partial y}=\dot{m}\Delta h_{vl}\ ,\ T=T_{sat} \qquad (12),(13)$$

where T_w is the wall temperature, $\dot{m}$ is condensation mass flux, λ_l is the liquid thermal conductivity, Δh_{vl} is the latent heat of condensation, and T_{sat} is the saturated vapor temperature. The pressure in the liquid film is calculated as,

$$P=-\frac{\sigma}{r}+P_{sat} \qquad (14)$$

where σ is the surface tension, P is the pressure in the liquid film, P_{sat} is the vapor pressure at the saturated temperature T_{sat}, and the horizontal directional radius of the liquid film r is given by

$$\frac{1}{r}=\frac{\left[\frac{1}{r_w}+\left(-\frac{2}{r_w^2}+\frac{\delta}{r_w^3}\right)\delta+\left\{\frac{2}{r_w}\frac{\partial\delta}{\partial s}+\delta\frac{\partial}{\partial s}\left(\frac{1}{r_w}\right)\right\}\frac{\partial\delta}{\partial s}+\left(1-\frac{\delta}{r_w}\right)\frac{\partial^2\delta}{\partial s^2}\right]}{\left\{\left(1-\frac{\delta}{r_w}\right)^2+\left(\frac{\partial\delta}{\partial s}\right)^2\right\}^{1.5}} \qquad (15)$$

In equation (15) the fin surface radius r_w is given by

$s=0\sim s_1$ (Region I) and $s=s_2\sim s_3$ (Region III) : $r_w=\infty$ (16)

$s=s_1\sim s_2$ (Region II) : $r_w=r_0$ (17)

The heat conduction equation in the fin is as follows,

$$\lambda_w\frac{\partial}{\partial s}\left[\frac{t/2+r_0(1-\cos\theta)}{\cos\theta}\frac{\partial T_w}{\partial s}\right]+\frac{\lambda_l(T_{sat}-T_w)}{\delta}=0 \qquad (18)$$

where λ_w is the thermal conductivity of fin material, and θ is given by

$s=0\rightarrow s_1$ (Region I) : $\theta=0$ (19)

$s=s_1\rightarrow s_2$ (Region II) : $\theta=(s-s_1)/r_0$ (20)

The boundary conditions to solve equation (18) are as,

$s=0$: $\partial T_w/\partial s=0$ (21)

$s=s_2(\theta=\pi/2)$: $T_w=T_{wb}$ (22)

where T_{wb} is the temperature of the base wall surface. In region III, the wall temperature is constant as

$$T_w=T_{wb} \qquad (23)$$

A2.10.3 Numerical results

To evaluate the effects of fin pitch p (range: 1.4 mm to 14 mm), fin height h (range: 1.26 mm to 13.8 mm), radius of concave joint region of plate and fin r_0 (range: 0.2 mm to 0.4 mm), fin thickness t (range: 0.1 mm to 0.4 mm) and fin length L (range: 1mm to 25 mm) on the condensation heat transfer characteristics of pure refrigerant on a vertical finned surface, the calculation is carried out on the following conditions: (1) the refrigerant vapors are HCFC123, HFC134a and HFC32 with their saturated temperature at T_{sat}=298.15 K, (2) materials of finned surface are aluminum and SUS304, (3) the fin-base plate temperature ranges from $T_{wb}=278.15$ K to 293.15 K.

Fig. A2.45 shows the examples of numerical results at fin parameters: p=3.0 mm, h =6.5mm, r_0=0.2 mm and t=0.2 mm, where figures (a) and (b) show the results of HCHC123/Alimumum and HCFC123/SUS304, respectively. In Fig. A2.45 (a), the liquid film thickness has increase with increase of s and has the maximum value. Then, it has the minimum value around s=s_1 and has the maximum value again in the region II, and then it has the minimum value around s=s_2 and has the maximum value at s=s_3 again. In Fig. A2.45 (b), the liquid film thickness shows the similar tendency to that in Fig. A2.45 (a). However, the liquid film thickness is almost zero around s =0 due to low thermal conductivity of fin material.

Fig. A2.46 shows s-directional distribution of dimensionless fin temperature $(T_w - T_{wb})/(T_{sat} - T_{wb})$ at the same calculation conditions as Fig. A2.45, where Figs. A2.46 (a) and (b) show the results of HCHC123/Aluminium and HCFC123/SUS304, respectively. In both figures the fin temperature decreases with increase of s.

Fig. A2.47 shows the effects of fin pitch p on the relation between average Nusselt number Nu_m and the group of dimensionless numbers $Ga_L Pr/Ph$ at the fixed fin parameters as h = 6.5 mm, r_0 = 0.2 mm, t = 0.2 mm and L = 25 mm, where figures (a), (b) and (c) show results of HCFC123/Aluminium, HCFC123/SUS304and HFC134a/Almimum, respectively. The average Nusselt number and other dimensionless numbers are defined as:

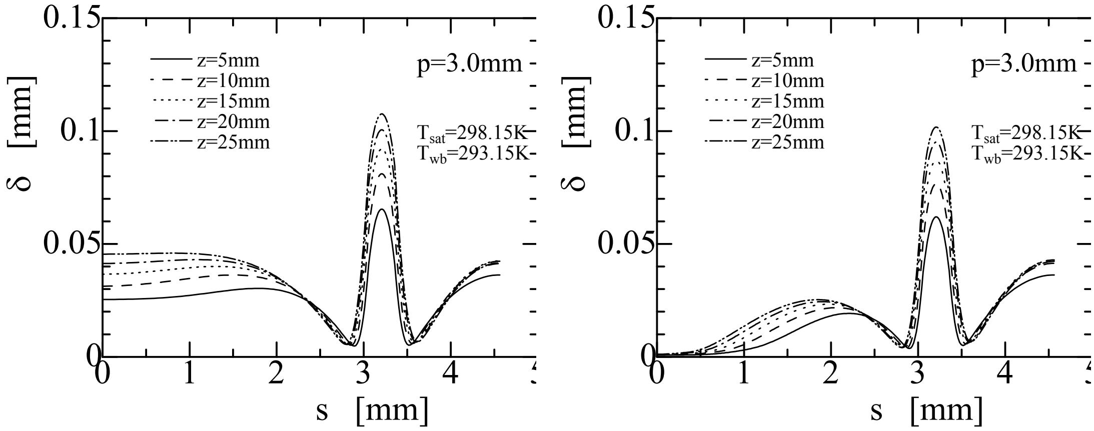
$$Nu_m = \frac{2\lambda_l \int_0^L \int_0^{s_3} q_{ws}\, ds\, dz\, L}{p\, L\, (T_{sat} - T_{wb})}, \qquad Ga_L = g\, L^3 / \nu_l^2$$

(24), (25)

$$Pr = \mu_l\, c_{pl} / \lambda_l\,, \qquad Ph = c_{pl}\,(T_{sat} - T_{wb}) / \Delta h_{vl}$$

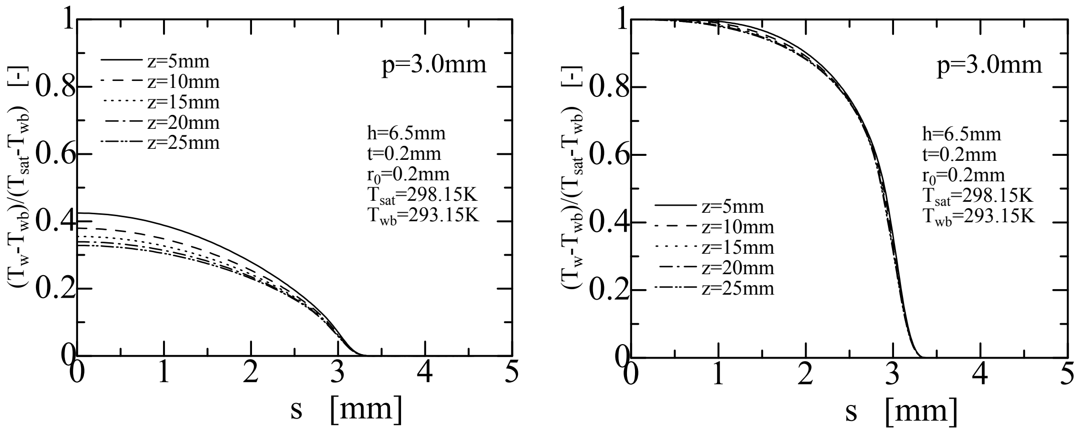
(26), (27)

where Ga_L, Pr, and Ph denote the Galileo number, the Prandtl number and the phase change number, respectively, and ν_l and c_{pl} denote the dynamic viscosity and isobaric specific heat of liquid, respectively. In any cases of pitch the Nusselt number of finned surface is higher than that of the Nusselt theory denoted by dotted chain line. At the same value of $Ga_L Pr/Ph$, the Nu_m value increases with a decrease in the value of pitch. It is also seen that the Nu_m value in the case of HFC123/Aluminum fins are higher than that of HFC123/SUS fins at the same fin dimensions.



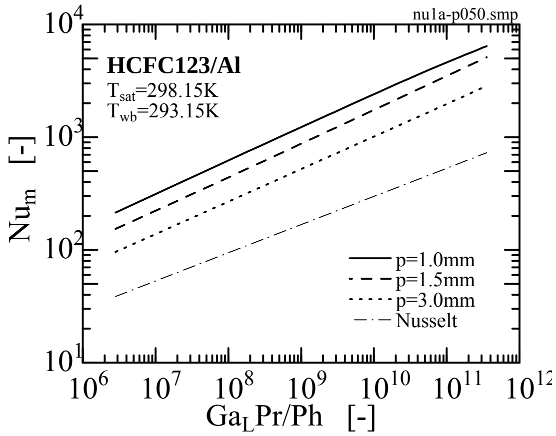
(a) In case of HCFC123/Al (b) In case of HCFC123/SUS304

Fig. A2.45 s-directional distribution of liquid film thickness

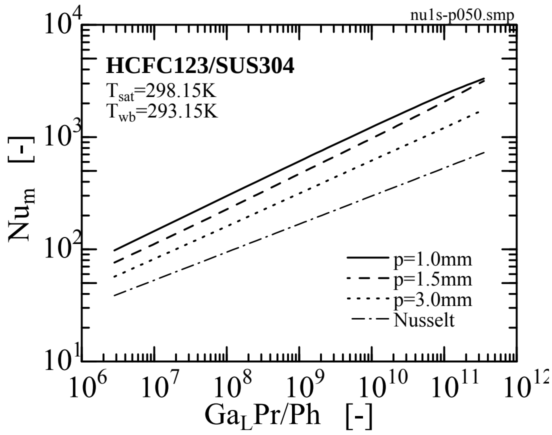


(a) In case of HCFC123/Al (b) In case of HCFC123/SUS304

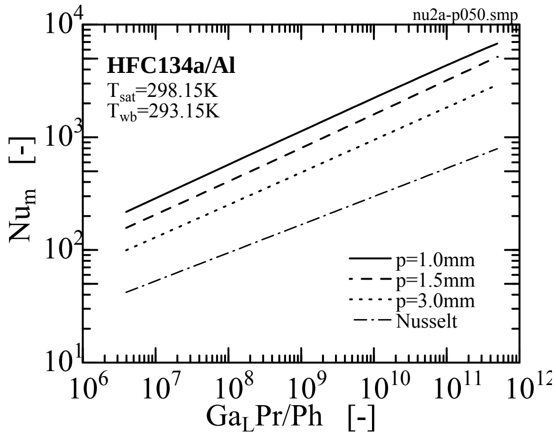
Fig. A2.46 s-directional distribution of fin temperature



(a) In case of HCFC123/Al



(b) In case of HCFC123/SUS304



(c) In case of HFC134a/Al

Fig. A2.47 Relation between Nu_m and $Ga_L Pr/Ph$ (Effects of fin pitch)

A2.10.4. Proposal of heat transfer correlation

Based on the present numerical calculation results, the following heat transfer correlation is obtained.

$$\frac{(Nu_m)_{cor}}{Nu_{m0}} = 1 + 0.43\ Bo^{-0.22}\ Pr^{0.07}\left(\frac{\lambda_w}{\lambda_l}\right)^{0.19}\left(\frac{p}{r_0}\right)^{-1.03}\left(\frac{h}{r_0}\right)^{036}\left(\frac{t}{r_0}\right)^{0.06}\left(\frac{L}{r_0}\right)^{0.28} \quad (28)$$

where Nu_{m0} denote the Nusselt equation given by,

$$Nu_{\mathrm{m0}} = 0.943\left(\frac{Ga_{\mathrm{L}}Pr}{Ph}\right)^{0.25} \quad (29)$$

and Bo is the Bond number defined by,

$$Bo = \frac{g\,\rho_l\,r_0{}^2}{\sigma} \quad (30)$$

The applicability range of equation (28) is as follows,

$2.2\times10^{-2} < Bo < 0.34$, $2.9\times10^{-2} < Ph < 0.16$, $6.3\times10^{5} < Ga_L Pr < 1.6\times10^{8}$
$2.5 < p/r_0 < 250$, $5 < h/r_0 < 100$

A2.11 Experimental Study on Gas-Liquid Flow Distribution to Multiple Upward Channels of Compact Evaporator

Masafumi Hirota (Mie University)

A2.11.1 Introduction

The multi-pass channels have been used in evaporators for the automobile air-conditioning system to improve its heat transfer performance. In those channels the maldistribution of gas and liquid from the dividing header to the branches has been a serious problem. Many studies have been conducted on this subject, but no systematic results have been obtained to date because the distribution characteristics change depending on many parameters. Among those parameters, the pressure distribution in the combining header, i.e., outlet pressure of branches, would be one of the most important factors. In most studies, however, the condition at the outlets of branches has been quite obscure. Therefore, in this study, air-water flow distribution into multiple upward channels that simulates the evaporator was examined with attention to the influence of the outlet pressure of branches on the flow distributions.

A2.11.2 Experimental apparatus and method

Fig. A2.48 is a schematic diagram of the experimental apparatus. The experiments were conducted with an isothermal air-water flow system. The test channel has a horizontal dividing header with a square cross section of 20 mm × 20 mm, and ten upward branches of 20 mm × 2 mm are connected to it at intervals of 20 mm. In order to examine the influence of the outlet pressure of branches on the flow distributions, we tested two outlet conditions as shown in Fig. A2.49; Case A is the non-uniform pressure condition, and Case B is the uniform pressure condition. In Case A, a gas-liquid separator and an air-flow meter are connected to each branch and the pressure in the separator, i.e., outlet pressure of each branch, is measured by a pressure gauge. In this case, the pressure in the separator varies depending on the flow distribution. In Case B, one branch to measure the flow distribution is connected to the separator, and other branches are connected to a combining tube. The valves at the exits of the separator and the tube are adjusted so that the pressure difference between them becomes zero. Since the combining tube is quite large, the pressure in it is nearly constant. Therefore, in Case B, one can measure the flow distribution under the constant outlet pressure condition for all the branches.

The superficial gas velocity j_G and liquid velocity j_L at the entrance of the header are set at 1.0 ~ 5.0 m/s and 0.015 ~ 0.045 m/s, respectively, referring to the operating condition of an evaporator. A developing channel is connected to the entrance of the header, and the flow pattern in it is the stratified wavy flow.

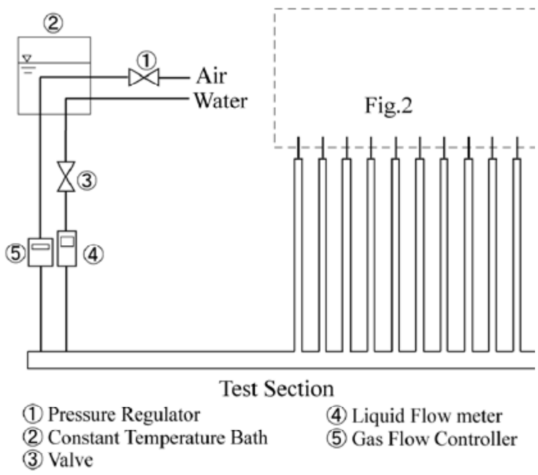


Fig. A2.48 Schematic diagram of experimental apparatus.

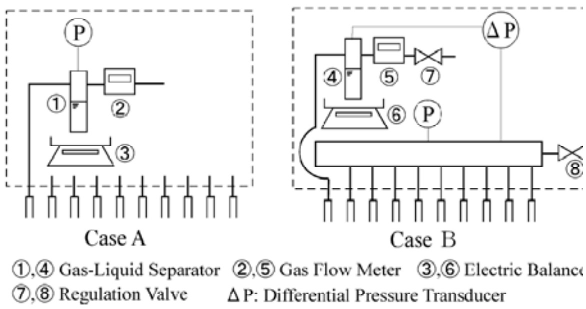


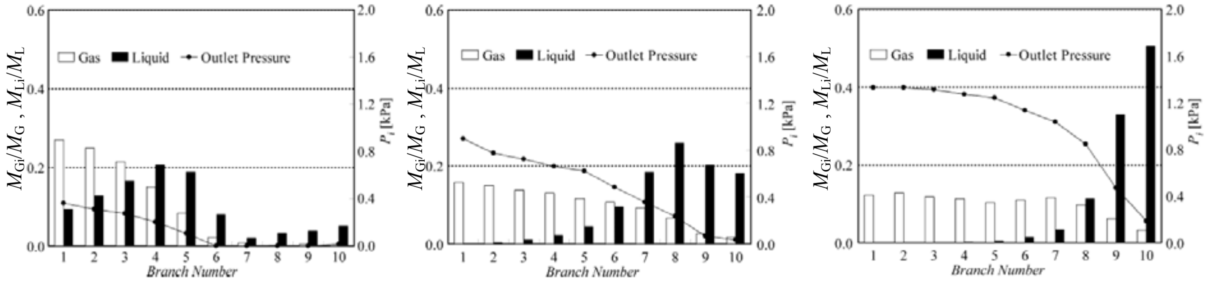
Fig. A2.49 Details of outlet conditions of branches.

A2.11.3 Experimental results and discussion

Fig. A2.50 shows the typical results of gas and liquid distributions in Case A. The abscissa shows the branch number, and the ordinate shows the gas distribution ratio M_{Gi}/M_G, liquid distribution ratio M_{Li}/M_L, and the outlet pressure P_i in each branch (i = 1~10). The value of j_L was fixed at 0.03 m/s and j_G was varied from 1 to 5 m/s. At the low j_G of 1 m/s, larger amount of gas was distributed to branches located nearer the header entrance and M_{Gi}/M_G was almost zero after the branch 6. M_{Li}/M_L also showed large values in upstream branches but it attained the maximum in branch 4. This location corresponded to the occurrence of large waves on the liquid surface. In the downstream region, the header was nearly blocked by stagnant liquid and no gas could be distributed to branches. As j_G was increased, both gas and liquid were distributed to further downstream branches. At j_G = 5 m/s, gas was distributed almost uniformly to all the branches but liquid was preferentially distributed to branches 9 to 10. A close correlation was observed between the gas distribution and the outlet pressure of branches.

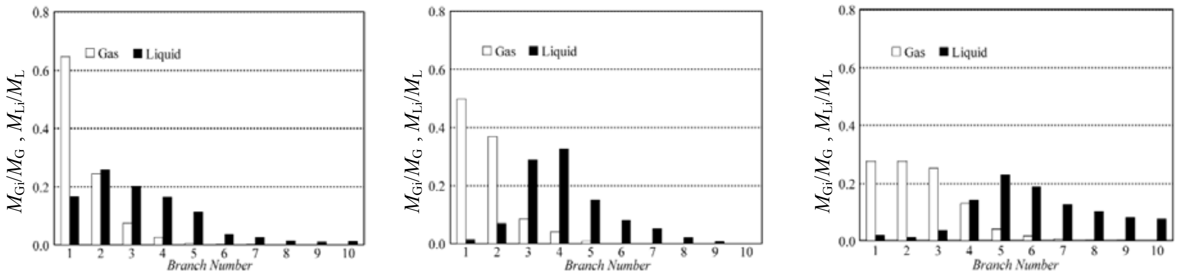
The results of Case B are shown in Fig. A2.51. Similar to Case A, gas and liquid tended to be distributed to downstream branches as j_G was increased. The maldistribution of gas to upstream branches was, however, more serious than in Case A. As a result, the preferential distribution of liquid to downstream branches observed in high j_G in Case A was remarkably relieved in Case B.

Next, the influence of liquid velocity on the gas-liquid distributions was examined in Figs. A2.52 and A2.53. Fig. A2.52 shows results for Case A; broken and solid lines show the gas and liquid distributions, respectively. In both distributions, the results fall into one line, suggesting that the distributions of gas and liquid are determined independently of j_L in Case A. In Case B, the gas and liquid distributions for different j_L show qualitatively similar characteristics. However, j_L exerted greater influence on the flow distributions than in Case A, and both M_{Gi}/M_G and M_{Li}/M_L in the upstream branches increase as j_L was increased.



(a) j_G =1.0 m/s (b) j_G =3.0 m/s (c) j_G =5.0 m/s

Fig. A2.50 Gas and liquid distributions under non-uniform outlet pressure condition (Case A, j_L = 0.03 m/s)



(a) j_G =1.0 m/s (b) j_G =3.0 m/s (c) j_G =5.0 m/s

Fig. A2.51 Gas and liquid distributions under uniform outlet pressure condition (Case B, j_L = 0.03 m/s)

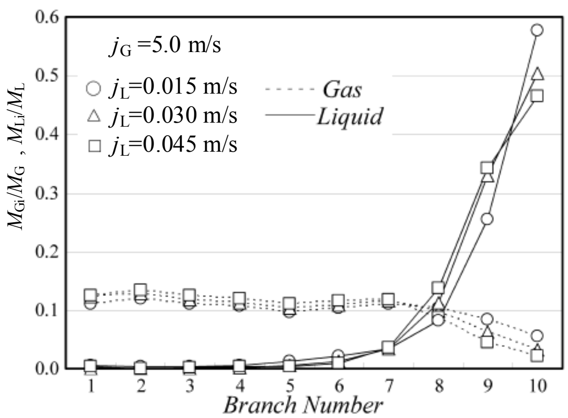


Fig. A2.52 Flow distribution in Case A (j_G=5.0 m/s)

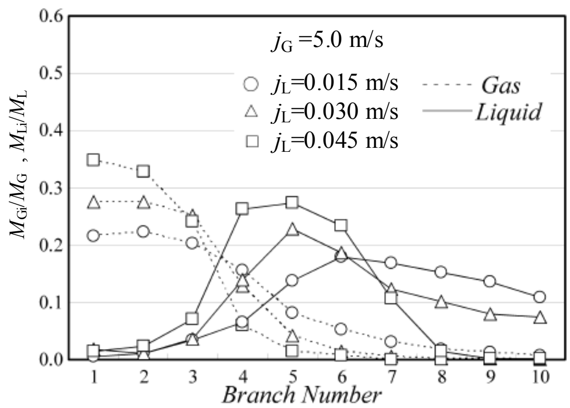


Fig. A2.53 Flow distribution in Case B (j_G=5.0 m/s)

A2.12 Characteristics of Air-Water Flows in Micro-grooved Evaporator Flat Channels with a Sharp 180-Degree Turn

Masafumi Hirota (Mie University)

A2.12.1 Introduction

In the air-conditioning systems of automobiles, evaporator should be equipped in quite a small space of the passenger's compartment, in which the temperature reaches more than 60 degrees Celsius. In response to such demands, a drawn cup type heat exchanger as shown in Fig. A2.54 had been developed. It consists of sharp 18-degree turned refrigerant channels with a very flat rectangular cross-section of 40 mm x 2 mm.

In general, a stagnation and dry-out of refrigerant liquid films lead to reduced heat transfer performance. Although protruding cross-ribs had been applied and cooling performance could be improved, pressure loss of refrigerant flow and the resulting refrigerant compressor load also increased. In order to develop a new type of channel wall with high ability of liquid film formation and low pressure loss, microgrooves had been engraved on the channel wall.

In this report, experimental results of local void fractions and pressure loss for air-water two-phase flow in microgrooved channels are shown.

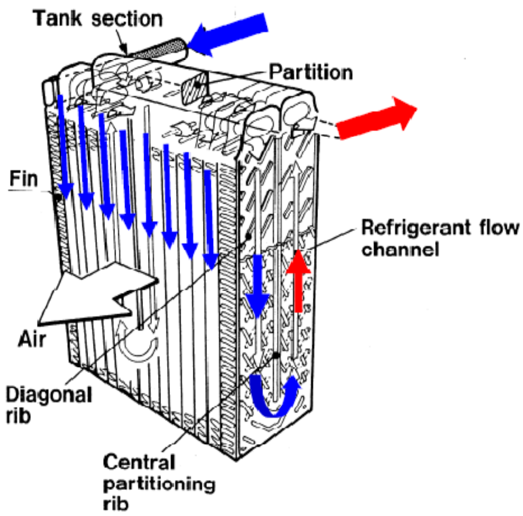


Fig. A2.54 Drawn cup type heat exchanger for car air-conditioner.

A2.12.2 Experiments

Detail of the test channel is shown in Fig. A2.55. The entrance section and test channel are made of transparent resin plates. The cross-section of the channel is of a flat rectangular shape with 40 mm x 2 mm (mean hydraulic diameter D_h = 3.81 mm). The test channel consists of two straight sections and one sharp 180-degree turn with a clearance of 40 mm, and the width of the rubber partition sheet in the straight section is 5 mm. These dimensions of the test channel are almost the same as those of practical evaporator. Pressure loss was evaluated from measured differential pressure between two pressure taps shown as P1 and P2 in Fig. A2.55.

Fig. A2.56 shows the details of microgrooves engraved over the entire surfaces of the long-side walls. The microgrooves have a triangular cross-section with the depth of 0.5 mm, pitch of 0.98 mm, and vertical angle of 60°. Two types of grooved surface with different groove-inclination angles of 45 and 90-degree to the longitudinal channel axis were tested. In MG45 with 45-degree groove-inclination angle, the microgrooves on one channel wall are inversely inclined to those on the other channel wall, and thus the front view looks like a cross-ribbed channel. The test channel was placed in several positions to evaluate the gravity effect.

The local void fractions were measured by a capacitance probe. Fig. A2.57 shows the schematic of void fraction measurement. Spatial average void fractions α were measured from the change in capacitance C by the following equation, where d, S and ε is the thickness, measuring area, and dielectric constant of each material, respectively. The capacitance of resin wall could be compensated using the measured value without water.

$$\alpha = 1 - \frac{1}{d}\left(\frac{1}{\varepsilon_{gas}} - \frac{1}{\varepsilon_{water}}\right)^{-1}\left(\frac{S}{C} - \frac{d_{resin}}{\varepsilon_{resin}} - \frac{d}{\varepsilon_{gas}}\right)$$

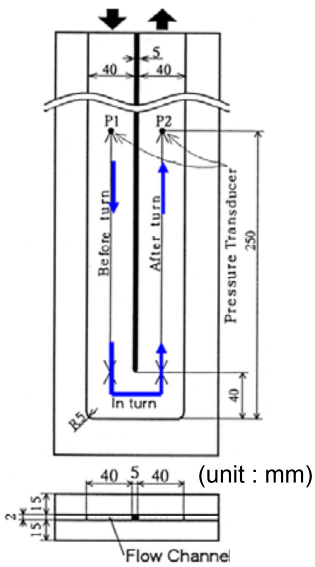
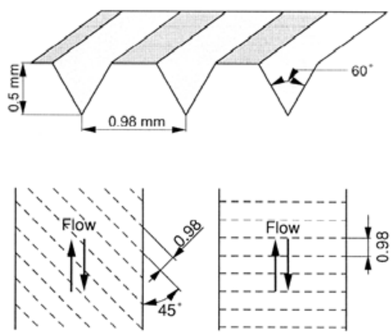


Fig. A2.55 Configuration simulated sharp turn narrow channel.



(a) MG45 (b) MG90

Fig. A2.56 Configuration of microgroove.

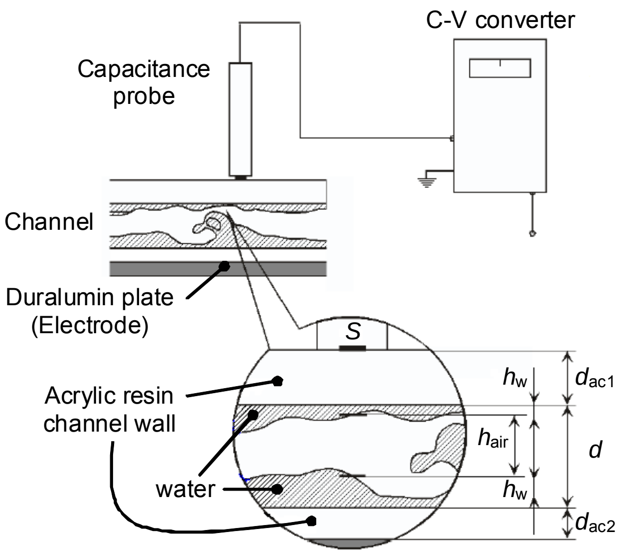


Fig. A2.57 Schematic diagram of measuring section for local void fraction by capacitance probe.

A2.12.3 Experimental results and discussion

Figs A2.58 (a-d) show void fraction distributions for varied quality with a constant liquid volumetric flux j_L. In each figures of (a) to (d), the effect of microgrooves can be evaluated. The blackened area corresponds to the region of α<0.1, and it is called "liquid stagnation region". The hatched area in the figures presents the region of α>0.9, in which the liquid film is nearly deficient (called "liquid film deficit region").

For the smooth U-shaped channel in a vertical plane with lower quality of 0.04, quite a large liquid stagnation region appeared around the bottom of the turn. The stagnant liquid was carried upward by air flow and was swept off at higher quality. In the straight section after the turn, the air flow deviated toward the outer-side of the channel, forming high void fraction region there. This air flow deviation was caused by the centrifugal force exerted on the air flowing around the 180-degree turn, and therefore it is more pronounced in the α distribution of x = 0.2 than that of x = 0.04.

For MG45 with inclined microgrooves, the liquid stagnation region also appeared near the exit of the turn section at low x condition. At higher quality, the stagnant liquid almost vanished. It was remarkable that the deviation of the high α region toward the outer-side of the straight section after the turn, which appeared obviously in the smooth channel. It can be said that the microgrooves with 45-degree inclination are quite effective to improve the span wise uniformity of the liquid film distribution after the turn. Although the liquid stagnation at low quality also appeared in MG90, the area became smaller than that in the smooth channel. This result suggests that the 90-degree inclined microgrooves are also effective the deviation of the α distribution caused by centrifugal force, but its performance to form the uniform liquid film distribution is inferior to the 45-degree inclined microgrooves.

For inverted U-shaped channel (Fig. A2.58 (b)), no liquid stagnation was observed in each channel. In lower quality condition, the α distribution was quite uniform in the smooth channel. Such a distribution will be favourable to the heat transfer performance of the evaporator. However, at higher quality, high α region over 0.9

appeared near the outside wall. In the microgrooved channels also, the uniformity of the α distributions was quite high in the low quality condition. At higher quality, α became rather large in the turn section but the liquid film deficit regions did not appear at all. This means that the microgrooves have high ability to form and keep the liquid films over the channel walls. As observed in the U-shaped channel, the spanwise uniformity of the α distribution after the turn in MG45 was much higher than that in the smooth channel and MG90.

For the horizontal sharp turn channel (Fig. A2.58 (c)), the α distributions were similar with those for the inverted U-shaped channel shown in Fig. A2.58 (b). The uniformity of the α distribution after the turn was lower than that in the inverted U-shaped channel. For either surface, the region with high void fraction was observed along the outside of straight channel after the turn. It can be seen that the inclined microgrooves in MG45 was effective to avoid liquid film deficit region.

For the upward sharp turn channel in a vertical plane (Fig. A2.58 (d)), gravity effect on phase distribution was observed in the straight channel before the turn. After the turn, the α distribution was seemed to be uniform.

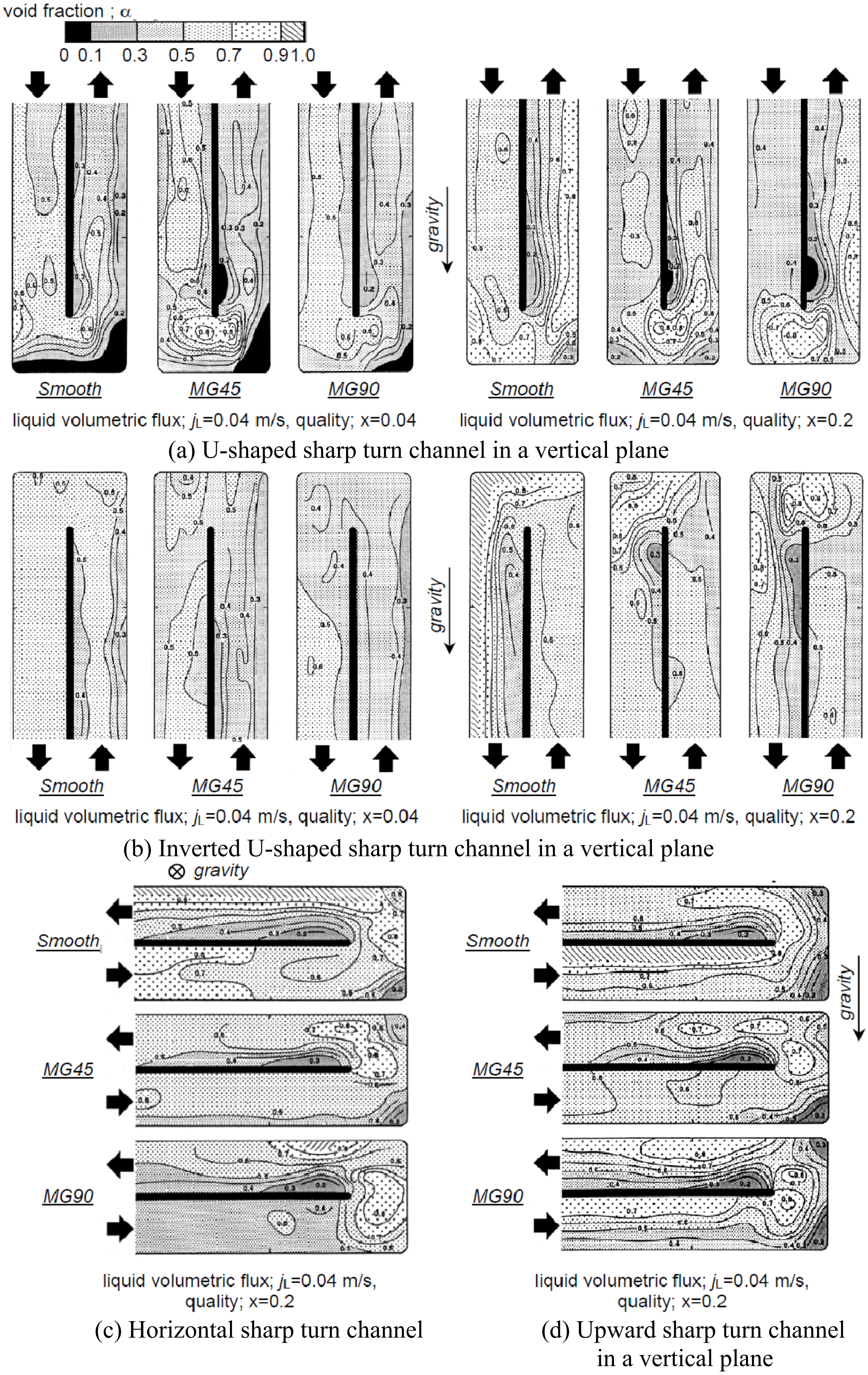


Fig. A2.58 Void fraction distribution of air-water two-phase flow in a narrow sharp turn channel measured by capacitance method.

A2.13 Visualization and Measurement of Refrigerant in a Commercial Plate Heat Exchanger by Neutron Radiography

Hitoshi Asano (Kobe University)

A2.13.1 Introduction

The plate heat exchanger (PHE) is widely used in industrial processes as a liquid-liquid heat exchanger. Recently, applications of the plate heat exchanger to gas-liquid two-phase flows have become of interest for the compactness and heat transfer performance improvement. A PHE is made by brazing 20 to 280 sheets of stainless steel wavy plates. Working fluids flow through the gaps between these plates. The channel of each fluid is arrayed alternately, that is, a plate heat exchanger has many parallel channels. In the case where working fluids flow as a gas-liquid two-phase mixture, the dynamic behaviours of gas-liquid two-phase flow greatly affect on the heat transfer performance. In order to design or improve such heat exchanger for gas-liquid two-phase flows, such as evaporator and condenser, it is important to understand liquid distribution not only in each channel but also into parallel channels. In particular, in the case where the inlet flow is a two-phase flow, the effects of inlet flow conditions and inlet configuration on liquid distribution characteristics should be clarified.

To understand gas-liquid two-phase flow behaviors in a compact heat exchanger with a complicated shape, flow visualization is much efficient. However, since heat exchangers are usually made by metallic material, it is difficult to visualize by optical ray. Radiography is a visualization technique by using the difference in attenuation rate of a radio ray to irradiated materials. X-ray, γ-ray, and thermal and fast neutron rays can be used as a radio ray. Mass attenuation coefficients, μm, of thermal neutron are quite different from those of X-ray and γ-ray as shown in Fig. A2.59. While the mass attenuation coefficients of X-ray increase with an increase in atomic number, the attenuation coefficients of thermal neutron ray is high for hydrogen, and is low for metallic elements. Therefore, neutron radiography is efficient for the visualization of gas-liquid two-phase flow in a metallic vessel.

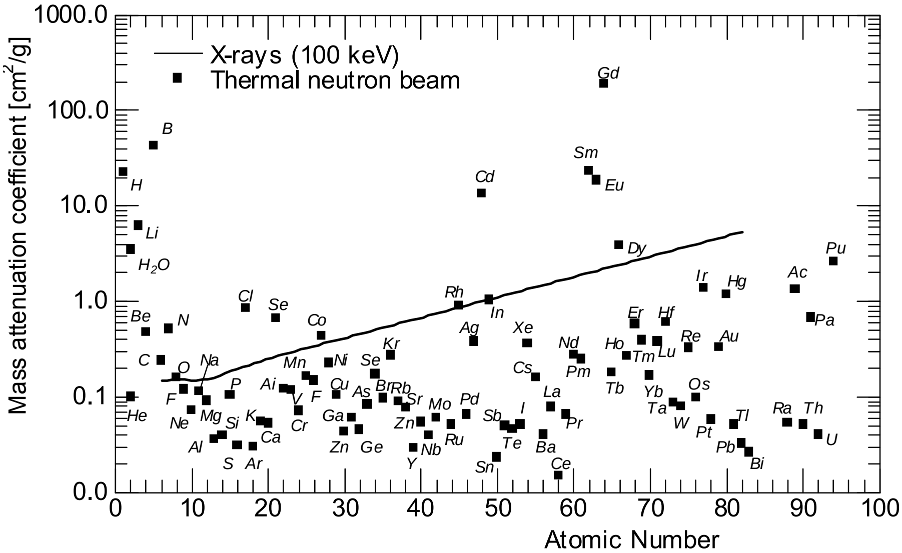


Fig. A2.59 Mass attenuation coefficients of thermal neutron and X rays.

Flows in a single- and multi-channel commercial brazing PHE were visualized by neutron radiography. In the single channel experiments, the effect of flow direction, vertically upward or downward, on boiling heat transfer performance and liquid distribution was examined. On the other hand, in the multi channel experiment, adiabatic air-water two-phase flows were visualized and average liquid hold-up in each channel was measured from neutron radiographs. The effect of inlet flow condition on phase distribution into each channel was considered.

2.13.2 Neutron radiography

Assuming the brightness of a visualized image is proportional to the beam intensity on a scintillation converter and neglecting the attenuation term due to gas phase, the brightness of two-phase flow image S_{TP}(x,y), the image without liquid $S_{\alpha=1}$(x,y), i.e., α(x, y) = 1, and the image full of liquid $S_{\alpha=0}$(x,y), i.e., α(x, y) = 0, is expressed as the following equations.

$$S_{\mathrm{TP}}(x,y) = G(x,y)\exp[-\rho_{\mathrm{w}}\mu_{\mathrm{mw}}t_{\mathrm{w}}(x,y) - \{1-\alpha(x,y)\}\rho_{\mathrm{L}}\mu_{\mathrm{mL}}t_{\mathrm{c}}(x,y)] + O_{\mathrm{TP}}(x,y) \quad (1)$$

$$S_{\alpha=1}(x,y) = G(x,y)\exp[-\rho_{\mathrm{w}}\mu_{\mathrm{mw}}t_{\mathrm{w}}(x,y)] + O_{\alpha=1}(x,y) \quad (2)$$

$$S_{\alpha=0}(x,y) = G(x,y)\exp[-\rho_{\mathrm{w}}\mu_{\mathrm{mw}}t_{\mathrm{w}}(x,y) - \rho_{\mathrm{L}}\mu_{\mathrm{mL}}t_{\mathrm{c}}(x,y)] + O_{\alpha=0}(x,y) \quad (3)$$

where ρ and μ_m are density [kg/m^3] and mass attenuation coefficient [m^2/kg], respectively, and those are physical properties of objects. The function of t is thickness along a neutron beam. The subscripts of w, L, and c mean wall, liquid, and conduit, respectively. α(x, y) mean the average void fraction along the neutron beam at coordinate (x, y). G(x, y) is gain and depends on the position due to non-flatness of initial beam intensity and of the sensitivity in the imaging system. O_{TP}(x,y), $O_{\alpha=1}$(x,y), and $O_{\alpha=0}$(x,y) are offset value in brightness.

The offset value is caused by neutron scatterings in an object, optical scatterings in a neutron camera, and the dark current of an imaging system, that is, depends on flow conditions. Therefore, the offset value under each condition is necessary for quantitative measurements of void fraction.

Using equations (1) to (3), a two-dimensional void fraction distribution can be expressed as

$$\alpha(x,y) = \ln\left[\frac{S_{TP}(x,y) - O_{TP}(x,y)}{S_{\alpha=0}(x,y) - O_{\alpha=0}(x,y)}\right] \Big/ \ln\left[\frac{S_{\alpha=1}(x,y) - O_{\alpha=1}(x,y)}{S_{\alpha=0}(x,y) - O_{\alpha=0}(x,y)}\right] \quad (4)$$

Equation (4) means that two-dimensional void fraction distributions can be calculated without knowing the thickness and the physical properties of wall and liquid, so far as the three images for without liquid, filled with liquid, and the offset for the each condition were recorded in the same configuration as two-phase visualization. Detail information on principle of neutron radiography and quantitative measurement method are shown by Takenaka and Asano, (2005).

A2.13.3 Boiling flow in a single channel

The detail of plate configuration is shown in Fig. A2.60. The heat transfer area of a plate is 0.0123 m^2, and the volume of single channel is 0.02 L. The average hydraulic diameter of the single channel was calculated to 3.36 mm. Tested PHE has three channels by 4 thin SUS wavy sheets. The working fluid of HCFC-142b in the center channel was heated by the heating medium of fluoro-carbon in the both sides channel. Neutron beam was irradiated vertically to plates, and boiling behavior was visualized.

Original radiographs of the test section with liquid single phase flow and without working fluid are shown in Fig. A2.61(a), (i) and (ii), respectively. From these two images, only liquid can be visualized by image processing. Processed image is shown in Fig. A2.61 (b). Brightness becomes lower with thicker liquid thickness. Since the image (i) is for liquid single-phase flow, the black net shows the channel of working fluid.

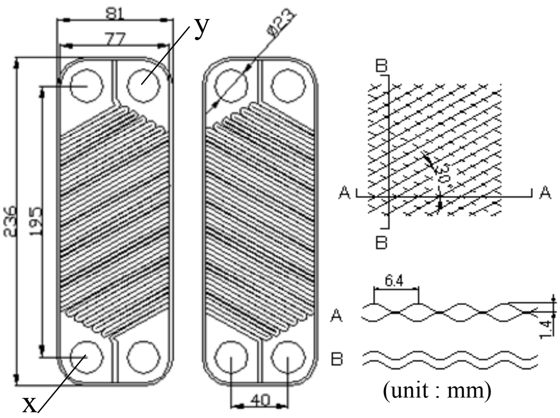
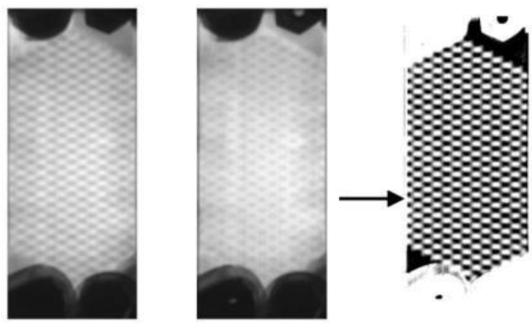


Fig. A2.60 Plate configuration.



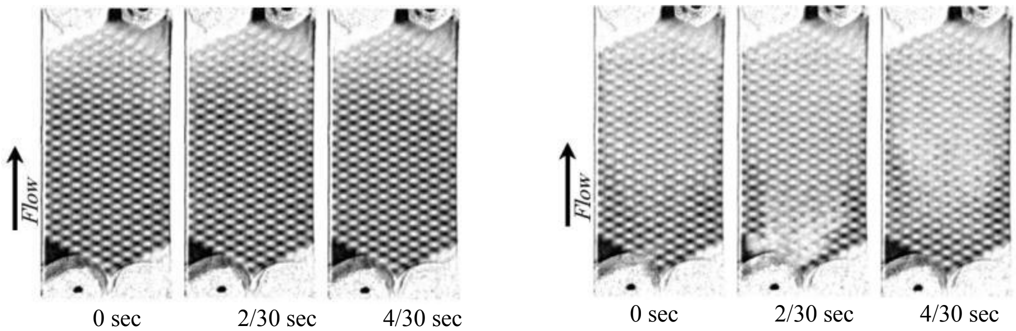
(i) Liquid image for single-phase visualization (ii) Without liquid (b) Processed liquid

(a) Original image

Fig. A2.61 Image processing to visualize liquid behavior.

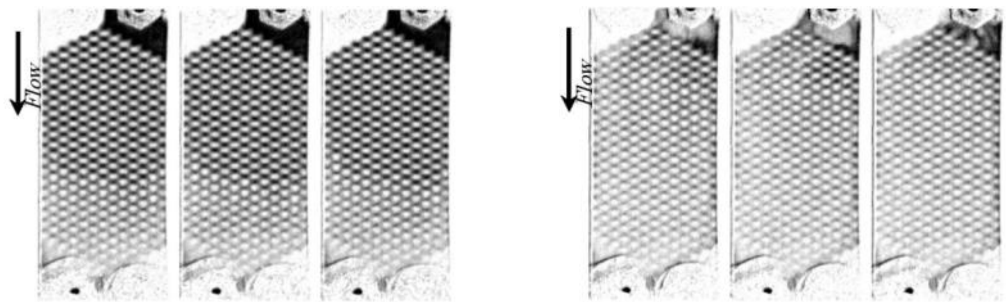
Continuous frames of liquid thickness distribution are shown in Fig. A2.62. The brightness is proportional to liquid thickness, that is, darker part shows thicker liquid thickness. Fig. A2.62 (i) shows boiling behaviour for inlet subcooling condition, and Fig. A2.62 (ii) shows those for inlet wet vapor condition for upward flows in Fig. A2.62 (a) and downward flows in Fig. A2.62 (b), respectively. For inlet subcooling condition, the area where boiling occurred can be clearly observed. Single-phase flow area was larger at the centre of channel for each direction. The reason might be on liquid flow rate distribution, that is, liquid flow rate around the centre might be higher. Boiling two-phase flow area for downward flow was larger than that for upward flow. In this case, there is little difference in overall heat transfer coefficient. For upward flow with inlet wet vapor condition, vapor flowed into the test section intermittently. Liquid stagnation was observed at the both side around the inlet. Heat transfer might be lower in the area with liquid stagnation. On the other hand, for downward flow with inlet wet vapor condition, though vapor flowed into the test section intermittently, liquid stagnation was not observed. For inlet wet vapor condition, overall heat transfer coefficients of the downward flow were higher than those of upward flow. The difference in flow pattern might lead to the difference in overall heat transfer coefficient.

Generally designers of heat exchanger think for boiling flow that heat transfer performance of downward flow in a tube might be lower than upward flow, because buoyancy operates in the opposite direction to main flow and slip velocity between liquid and vapor becomes lower. However, heat transfer in PHE was strongly influenced by flow pattern, and downward flow was better for heat transfer performance.



(i) Inlet subcooling (ΔT_{subin}=10K) (ii) Inlet wet vapor (x_{in}=0.01)

(a) Upward flow



(i) Inlet subcooling (ΔT_{subin}=10K) (ii) Inlet wet vapor (x_{in}=0.01)

(b) Downward flow

Fig. A2.62 Liquid flow behaviors in boiling two-phase flow
Effect of inlet condition and flow direction (Mass flow rate□0.026 kg/s)

A2.13.4 Flow distribution into parallel channels

Air-water two-phase mixture was supplied to a multi-channel PHE, and flows were visualized under adiabatic condition. From a projection image in the direction parallel to plates, the sum total of liquid thickness in each channel can be calculated. Fig. A2.63 shows an appearance of multi-channel PHE with 18 parallel channels. Holes in each plate form the cylindrical distributing header of 23 mm in diameter and 69.7 mm in length. Fig. A2.64 shows two-dimensional distribution of liquid thickness with arbitrary scale. Higher brightness shows thicker liquid thickness. Since 18 white vertical lines are clearly seen, it can be said liquid volume in each channel can be measured individually. Assuming that the volume of conduit at arbitrary cross-section is the same, the ratio of liquid volume is equivalent to the ratio on cross-sectional average liquid fraction.

Calculated results at 60 mm upward from the inlet (y=60 mm) are plotted in Fig. A2.65 (a) and (b). Vertical axis shows liquid volumetric fraction normalized by the average value. In Fig. A2.65 (a), at the lower gas volumetric flux j_G, liquid fraction became higher with deeper channel due to the higher liquid momentum. However, at the higher j_G, the liquid distribution has the opposite tendency with the case of lower j_G. Liquid fraction became lower with the deeper channel. On the other hand, in the Fig. A2.65 (b), the same tendency was observed. That is to say, liquid distribution into each channel strongly depended on balance between gas and liquid flow rate. It is estimated that there is a condition in which liquid is distributed homogenously. Under the every condition, the liquid fraction at the first channel from the inlet was high. The reason might be that liquid phase easily stagnated at the enlarged section.

For the downward flow, since the water flew into each channel from the lower part of the header, it can be estimated that water flows easily into a nearer channel from the inlet. At the lower liquid volumetric flux j_L of 0.01 m/s in Fig. A2.66 (a), the results showed such tendency. However, when the j_L increased to 0.03 m/s, the opposite tendency was observed. That is to say, the liquid fraction became higher with a deeper channel. In all cases, the effect of j_G on the liquid distribution was small.

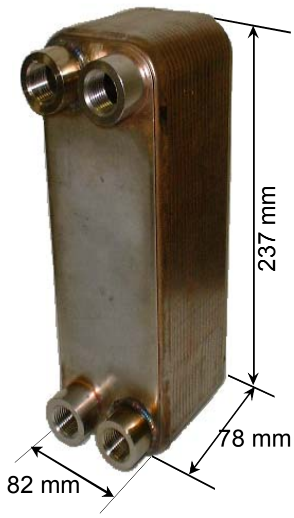


Fig. A2.63 Appearance of tested PHE.

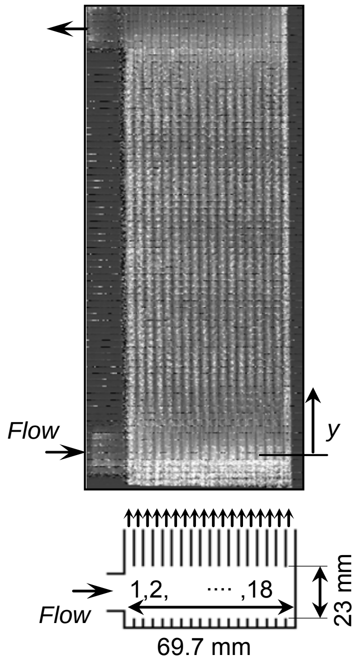
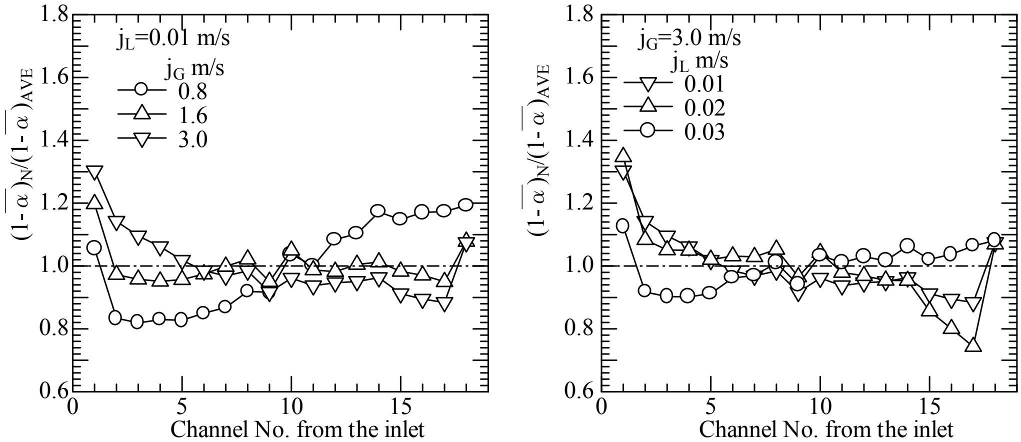
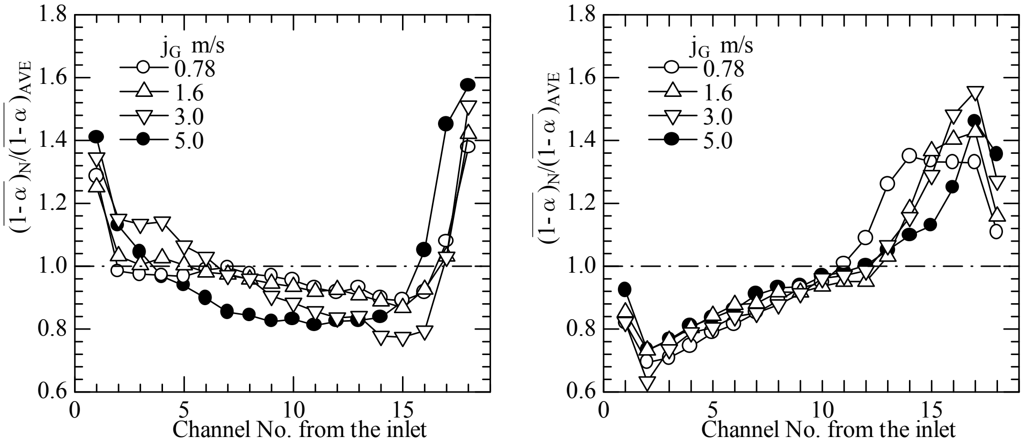


Fig. A2.64 Processed image for liquid hold-up in 18 channels PHE with arbitrary scale.



(a) Effect of gas velocity (b) Effect of liquid velocity

Fig. A2.65 Liquid distribution of the upward flow in the multi channel plate heat exchanger.



(a) j_L=0.01 m/s b) j_L=0.03 m/s

Fig. A2.66 Liquid distribution of the downward flow in the multi channel plate heat exchanger.

A2.14 The enhanced heat transfer in the plate-type evaporator by using an ejector for recirculation

Tatasunori Man'o and Masayuki Tanino (Takasago Thermal Engineering Co. Ltd.)
Takashi Okazaki (Mitsubishi Electric Corp.)
Shegru Koyama (Kyushu University)

2.14.1 Introduction

It is known that the evaporative heat transfer coefficient (α_r) in annular flow regime in horizontal pipes increases while increases of the vapor quality of liquid-vapor two phase refrigerant flow. In this point of view, we proposed to enhance the α_r in the evaporator by installing an ejector in the pressure reducing section of the vapor compression refrigeration cycle. In this method, the ejector is used to recirculate a portion of the vapor refrigerant at the evaporator outlet to the inlet. It leads to improve the α_r. In plate-type evaporator utilized in chiller, the liquid-vapor two phase refrigerant flow into the evaporator changes to an upward perpendicular direction at the horizontal inlet header. It causes the problem of a deterioration in the $\alpha_{r,m}$ due to mal-distribution of refrigerant flow in the evaporator with a large number of plates (number of refrigerant branches). This paper describes results from experimentally evaluating effects brought about on liquid-vapor two phase refrigerant distribution at the plate-type evaporator inlet, when utilizing an ejector to recirculate vapor refrigerant in refrigeration cycles with an actual plate-type evaporator.

2.14.2 Evaporative heat transfer enhancement method using an ejector

Fig. A2.67 shows compression refrigeration cycle system diagram with a proposed evaporative heat transfer enhancement method. An ejector is installed in a pressure reducing section. The evaporator outlet pipe is connected to the suction port of the ejector. The ejector suctions the vapor refrigerant G_{ej} from the evaporator outlet and recirculates it, simultaneous with pressure reducing of the high-pressure liquid refrigerant G_n from the condenser. Therefore, the evaporative heat transfer coefficient can be improved by increasing the vapor quality and mass flow rate of refrigerant in the evaporator. However, increasing the vapor quality and mass flow rate of refrigerant flow into the evaporator has the effect of increasing dry-out region in the evaporator refrigerant flow path. Fig. A2.68 shows calculation results of local evaporative heat transfer coefficient $\alpha_{r(cal.)}$ and tube length in dry-out state L_d in horizontal smooth evaporative tube for each of the recirculation ratios G_{ej}/G_n, where ordinate and abscissa represents the $\alpha_{r(cal.)}$ and dimension-less tube length L respectively. As shown in Fig. A2.68, the L_d is increased as the G_{ej}/G_n is increased. However, the $\alpha_{r(cal.)}$ from the evaporator inlet to L at initiation point of dry-out is increased as the G_{ej}/G_n is increased, and there is not a large difference between the $\alpha_{r(cal.)}$ after the initiation point of dry-out regardless of the G_{ej}/G_n. Consequently, the average evaporative heat transfer coefficient for the entire evaporator is improved through an integrated value of $\alpha_{r(cal.)}$ is increased as the G_{ej}/G_n is increased by using the proposed method.

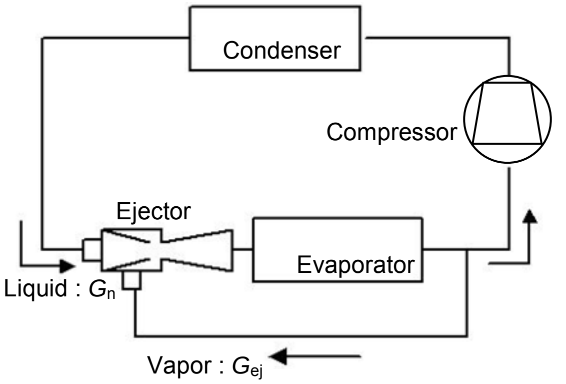


Fig. A2.67 Refrigeration cycle with a vapor recirculation evaporator by an ejector

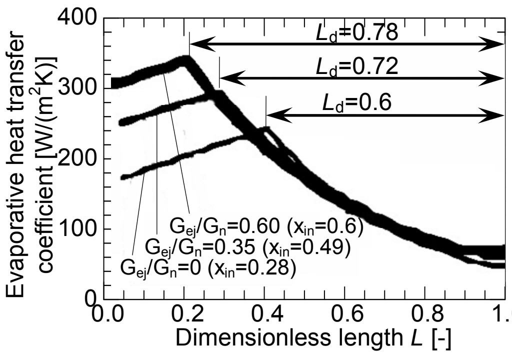


Fig. A2.68 Correlation of L with $\alpha_{r(cal)}$ and L_d

2.14.3 Experimental apparatus

Fig. A2.69 and Table A2.6 shows, respectively, a schematic overview of the experimental apparatus and specifications of a plate-type heat exchanger with 50 plates (refrigerant channels: 24) used as an experimental evaporator to evaluate the $\alpha_{r,m}$ and refrigerant flow distribution in the inlet header of the plate-type evaporator. The detail about experimental apparatus can be seen in a relevant publication as stated above.

2.14.4 Experimental conditions and procedures

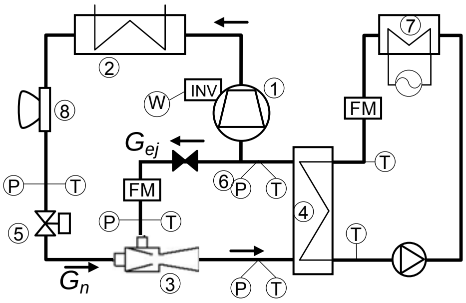
Table A2.7 shows the experimental conditions, HFC-404A was utilized as the refrigerant, the condensing temperature T_c was approximately kept of 50 °C, and experiments were performed for the cases where the average mass velocity of refrigerant in an evaporator channel $m_{r(0)}$ was 40kg/(m^{2}s), and the evaporative temperature without recirculation $T_{e(0)}$ was 0 °C. The experimental procedures were as follows:

1) The recirculation flow rate G_{ej} was changed while adjusting the valve ("6" in Fig. A2.69) to an optional degree of opening from fully closed to a fully open.

2) The compressor rotation speed and opening degree of the electronic expansion valve ("5" in Fig. A2.69) were adjusted so that heat exchange rate in the evaporator and the degree of super heat of the refrigerant at the evaporator outlet were always a fixed quantity.

3) The cooled fluid temperature of the experimental evaporator inlet was regulated to a fixed temperature at this time by using a dummy load ("7" in Fig. A2.69).

4) The evaporative temperature and pressure were measured after the refrigeration cycle reached a steady state. The $\alpha_{r,m}$ of refrigerant in the experimental evaporator was obtained by finding the over-all heat transfer coefficient from the heat exchange rate, heat transfer area and logarithmic mean temperature difference, and then utilizing the cooled fluid side heat transfer coefficient found from the experimental correlation formula for the single phase flow heat transfer coefficient. Here, brine was utilized as the cooled fluid.



1. Compressor 5. G_n control valve Ⓣ Thermometer
2. Condenser 6. G_{ej} control valve Ⓟ Pressure gauge
3. Ejector 7. Dummy load FM Turbine flow meter
4. Evaporator 8. Mass flow meter

Fig. A2.69 Schematic view of experimental apparatus

Table A2.6 Specification of experimental evaporator

Number of plates	: N_p		50
Heat transfer area	: A	[m^2]	0.672
"A" per a plate	: A_1	[m^2]	0.014
Plate height	: H_p	[m]	0.207
Plate width	: w_p	[m]	0.077
Plates pitch	: p_p	[mm]	2.35
Plate thickness	: t_p	[mm]	0.35
Channel volume	: v_{ch}	[l]	0.020
Hydraulic diameter	: d_h	[mm]	2.86
Average channel cross section area	: A_{ch}	[m^2]	9.7×10^{-5}
Inlet pipe diameter	: d_{in}	[mm]	16.9
Header section diameter	: d_h	[mm]	20.5

Table A2.7 Experimental conditions

Refrigerant			HFC-404A
Condensing temperature	: T_c	[°C]	50
Evaporating temperature (no-recirculation)	: $T_{e(0)}$	[°C]	0
Inlet vapor quality	: x_{in}	[-]	0.44~0.72
Degree of refrigerant superheat	: SH	[K]	1.0~20
Average velocity in a channel(no-recirculation)	: $m_{r(0)}$	[kg/(m^{2}s)]	40
Refrigerant flow rate	: G_n	[kg/h]	337
Heat exchange rate	: Q_e	[kW]	10.3
Evaporator inlet brine temp.	: $T_{b,in}$	[°C]	16.6
Evaporator outlet brine temp.	: $T_{b,out}$	[°C]	11.6
Brine flow rate	: V_b	[l/min]	31.0

2.14.5 Method to evaluate the refrigerant distribution in the plate-type evaporator

Fig. A2.70 shows a measuring equipment to evaluate the distribution characteristics of refrigerant at the inlet header of the plate-type evaporator. As shown in Fig. A2.70, the measuring equipment consists of a T-type sheathed thermocouple (Class 1) with an outer diameter of 1.6 mm and with its tip bent 90 degrees and a shaft seal mechanism to move the thermocouple axially. This measuring equipment were connected to refrigerant pipe and brine pipe at the evaporator outlet respectively, then refrigerant temperature($T_{r,ch}$) and brine temperature($T_{b,ch}$) at the plate channel outlets were directly measured by a thermocouple moved in the horizontal direction and rotated in the evaporator outlet header. The $T_{r,ch}$ were measured at the 1st, 7th, 13th, 19th and 24th channel outlets from the upstream end of the evaporator inlet header (downstream end of the outlet header), and the $T_{b,ch}$ was measured at the 2nd, 7th, 13th, 19th, and 24th channel outlets. The distribution of refrigerant was evaluated by comparing the degrees of refrigerant superheat (SH_{ch}) profiles and heat exchange rate(Q_{ch}) profiles for the individual channels from the case of no recirculation with the corresponding data from the case of recirculation. The SH_{ch} were calculated from measured $T_{r,ch}$ and the saturation temperature that equivalent to the evaporator outlet pressure. The Q_{ch} were calculated from measured $T_{b,ch}$ and bribe temperature at evaporator inlet. In the calculation of the Q_{ch}, it was assumed that the brine flow as single-phase would flow evenly into the individual plate channels.

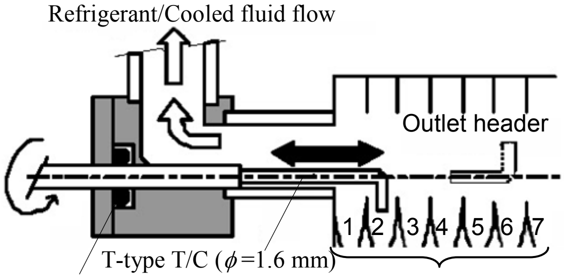


Fig. A2.70 Channel outlet temperature measurement equipment.

2.14.6 Average evaporative heat transfer coefficient

Fig. A2.71 shows correlation of inlet vapor quality x_{in} with experimental and calculation results of average evaporative heat transfer coefficient($\alpha_{r,m(exp.)}$ and $\alpha_{r,m(cal.)}$) and average mass velocity in an channel m_r, where left and right ordinate represent the $\alpha_{r,m}$ and the m_r respectively, and the abscissa represents the x_{in}. The $\alpha_{r,m(cal.)}$ was calculated using a experimental correlation formula about plate-type evaporator with 6 plates (refrigerant channels: 2) which has same specification plate to evaporator as shown in Table A2.6. As shown in Fig. A2.71, in the case of no recirculation with x_{in}=0.44, the $\alpha_{r,m(exp)}$ (1650W/(m^{2}K)) is much lower than the $\alpha_{r,m(cal.)}$ for the same conditions (4400 W/(m^{2}K)). Even after taking into account the accuracy of the $\alpha_{r,m(cal.)}$, the $\alpha_{r(exp.)}$ value is extremely lower than the $\alpha_{r,m(cal.)}$. The experimental correlation formula to calculate the $\alpha_{r,m(cal.)}$ were obtained in the case where the uniform distribution of refrigerant in the inlet header of evaporator was assumed. In addition, the $\alpha_{r,m(exp.)}$ were calculated based on the actual total heat transfer area. From these facts, it can be said that the reason for the low $\alpha_{r(exp.)}$ relative to the $\alpha_{r,m(cal.)}$ is a reduction in the effective evaporative heat transfer area due to mal-distribution of refrigerant in the inlet header. In the case of recirculation with x_{in}=0.63, The $\alpha_{r,m\ (exp.)}$ increased 1.8 times as compared to $\alpha_{r,m\ (exp.)}$ in the case of no recirculation. In addition, from result of the $\alpha_{r,m(cal.)}$, it is found that the $\alpha_{r,m(cal.)}$ in a plate-type evaporator with 6 plates in which the effect of mal-distribution of the refrigerant is very small can be increased about 10% by increasing x_{in} to 0.63 from 0.44 and m_r to 65 kg/(m^{2}s) from 40 kg/(m^{2}s). From these results, in the case of a plate-type evaporator with 50 plates, it can be said that the $\alpha_{r,m(exp.)}$ was significantly improved by increasing the effective evaporative heat transfer area through improvement in mal-distribution of liquid phase refrigerant in the inlet header by recirculating vapour refrigerant.

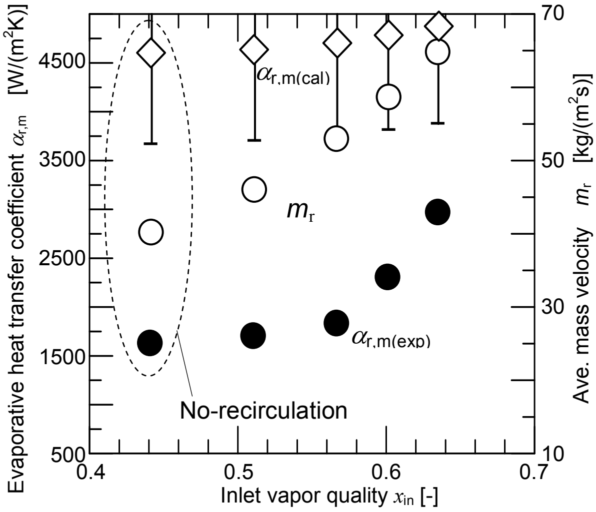


Fig. A2.71 Correlation of x_{in} with $\alpha_{r,m}$ and m_r

2.14.7 Refrigerant distribution in the inlet header of the evaporator

Fig. A2.72 shows the SH_{ch} profiles in cases of x_{in}=0.44 (no recirculation) and x_{in}=0.63 (recirculation). Circle and triangle symbols correspond the result of no recirculation and recirculation respectively. As shown in Fig. A2.72, in the case of no-recirculation, the SH_{ch} at the 1st channel outlet was 1K, which was close to the saturation state, while the SH_{ch} at the 7th to 19th channel outlets were 16K or higher and the SH_{ch} at the 24th channel outlet was 11K. In the case of recirculation (x_{in}=0.62), the SH_{ch} at the 1st channel outlets increased to 6 K and the SH_{ch} at the 2nd to 24th channel outlet decreased to 2 to 8K.

Fig. A2.73 shows the Q_{ch} profiles in cases of no recirculation and recirculation. Circle and triangle symbols correspond to the results of the no recirculation and recirculation respectively. The dash-and-dot line in this figure represents the heat exchange rates in the case where the refrigerant is distributed evenly to the individual plate channels ($Q_{ch(eq.)}$). In the case of no recirculation, the Q_{ch} at the 2nd and 24th channel were more than 440W, and the Q_{ch} at the 7th channel was the lowest at 276W. The difference in the Q_{ch} with the highest and lowest was 179W, so it is confirmed that the dispersion in the Q_{ch} with respect to $Q_{ch(eq.)}$ was as large as 40%. In the case of recirculation, the Q_{ch} at the 2nd channel was the lowest at 335W and that at the 24th channel was the highest at 400W. The difference in the Q_{ch} with the highest and lowest was 65W, which was much smaller than that in the case of no recirculation. So, the dispersion in Q_{ch} with respect to $Q_{ch(eq.)}$ was +8% and -2%, which means that roughly uniform heat exchange took place.

From these results of the SH_{ch} and the Q_{ch} profiles as described above, it can be guessed that following. In the case of no recirculation, the liquid phase refrigerant flowed into channels located near the upstream end of the inlet header were excessively high compared to the liquid phase refrigerant flow rate required to maintain the required heat exchange rate while liquid phase refrigerant shortages occurred at other channels. Therefore, gross mal-distribution of liquid phase refrigerant occurred in the inlet header. In the case of recirculation, the liquid phase refrigerant flowed into downstream side channels as well. Therefore, mal-distribution in the case of no recirculation is dissolved.

To expand on above evaluation, Fig. A2.74 shows the correlation of the vapor phase refrigerant velocity at the evaporator inlet pipe u_v with the $\alpha_{r,m}$ increased ratio $\alpha_{r,m(exp.)}/\alpha_{r,m(exp,0)}$ for the results from Fig. A2.73. The $\alpha_{r,m}/\alpha_{r,m(0)}$ relative $\alpha_{r,m}$ to $\alpha_{r,m(0)}$ with no recirculation. The u_v was calculated using the inner diameter of the refrigerant pipe connected to the inlet of the evaporator. As shown in Fig. A2.74, the $\alpha_{r,m(exp.)}$ was improved in 10% or more at u_v=9.5m/s or more by vapor refrigerant recirculation. Fig. A2.75 shows the flow pattern map [Mandhane et al.(1974)] for the results from Fig. A2.74. Open symbols represent the case of the $\alpha_{r,m(exp.)}$ was improved in 10% or more. From the observation of refrigerant flow in evaporator inlet pipe from sight glass, it is found that the $\alpha_{r,m}$ is improved when flow pattern of refrigerant at the inlet of evaporator come to annular-mist flow regime by increasing vapor phase velocity to 9.5m/s or more.

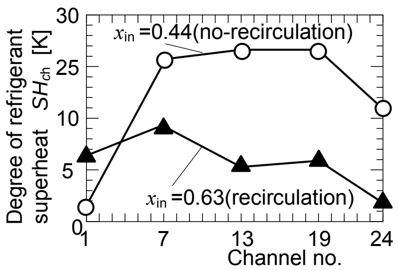


Fig. A2.72 SH_{ch} profiles at channel outlet

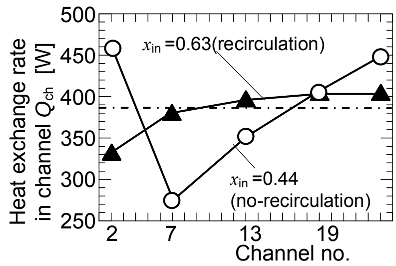


Fig. A2.73 Q_{ch} profiles at channel outlet

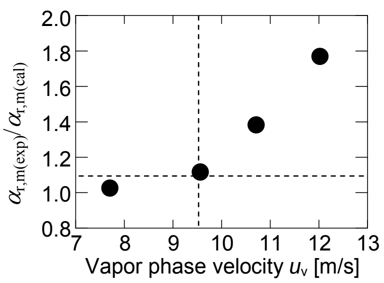


Fig. A2.74 Correlation of u_v with $\alpha_{r,m(exp)}/\alpha_{r,m(cal)}$

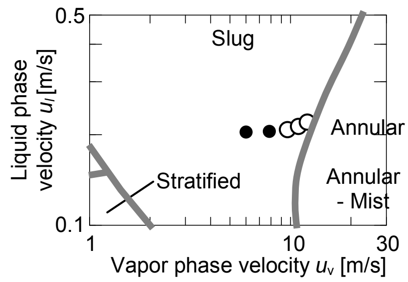


Fig. A2.75 Flow pattern map

2.14.8 Conclusion

A new method to enhance the heat transfer in the evaporator and the effect of the new method on heat transfer enhancement were described. And then, effects of the new method on average evaporative heat transfer coefficient and refrigerant flow distribution at inlet header of the plate-type evaporator was evaluated, experimentally, using the refrigeration cycle with an actual plate-type evaporator. As the result, it was confirmed that the mal-distribution of refrigerant flow at the inlet header was improved and a better average evaporative heat transfer coefficient obtained by increasing the vapor phase refrigerant velocity to 9.5 m/s or more by vapor recirculation.

A2.15 Development of Heat Exchangers for Natural Refrigerants

Akito Machida (Mayekawa Mfg. Co. Ltd.)

A2.15.1 Introduction

Mayekawa has developed refrigeration and air conditioning systems using natural refrigerants, ammonia, carbon dioxide, air, water and hydro carbons. Carbon dioxide is used as a secondary refrigerant in refrigeration and storage applications as well as a primary refrigerant in trans critical heat pumps. Air is the refrigerant in the air refrigeration cycle with an operation range of -50℃ to -100℃ while water is used as the refrigerant in adsorption cycles. Ammonia and hydro carbons are used in a wide range of refrigeration and air conditioning applications.

In all these systems, heat exchangers play important roles in terms of performance, refrigerant charge, safety and reliability. Use of compact heat exchangers with high heat transfer performance characteristics not only improves the COP of the system but also reduces refrigerant charge and improves safety and reliability of the system.

A2.15.2 System

The critical pressure and temperature of carbon dioxide are 7.37MPa and 30.98C respectively. Compared to the conventional secondary refrigerants, therefore, carbon dioxide operates at high pressures and heat exchangers must be designed to withstand the high pressures (table 1 -30C, 1.43MPa).

The system mainly consists of an ammonia refrigeration unit, CO_2 receiver, CO_2 pump, and an air cooler. CO_2 is condensed in the evaporator of the ammonia refrigeration unit and flows into the CO_2 receiver. From the receiver, CO_2 is pumped to the air cooler in the freezer where it exchanges heat with food products and evaporates. The evaporated CO_2 flows back into the evaporator of the ammonia refrigeration unit where it is condensed again. A plate and shell heat exchanger was used as the evaporator and a copper tube air cooler in the freezer.

A2.15.3 Plate and shell heat exchanger

Compact heat exchangers are of great interest in industrial refrigeration and air conditioning because of their potential to reduce refrigerant charge. Plate and shell heat exchangers have been applied in ammonia and ammonia/CO2 refrigeration systems.

Fig.A2.76 shows a picture of the plate and shell heat exchanger. The heat exchanger consists of circular plates welded in a pack and housed in a shell. The Plate and shell heat exchanger combines the best features of the plate heat exchanger which are compactness and high heat transfer performances as well as those of the shell and tube heat exchangers which are high pressure and temperature capabilities. It is fully welded with no gaskets therefore refrigerant leakage is greatly minimized.

It has been applied as an evaporator and a condenser. And in the case of evaporators it has been applied in both flooded and direct expansion heat exchangers.

As shown in Fig. A2.77, the flooded heat exchanger offers higher heat transfer performances compared to direct expansion heat exchangers. This is because the flooded heat exchanger's heat transfer area is fully wetted where as the direct expansion heat exchanger has a portion for super heating. This portion results in single phase gas heat transfer characteristics which lower the over all heat transfer coefficients.

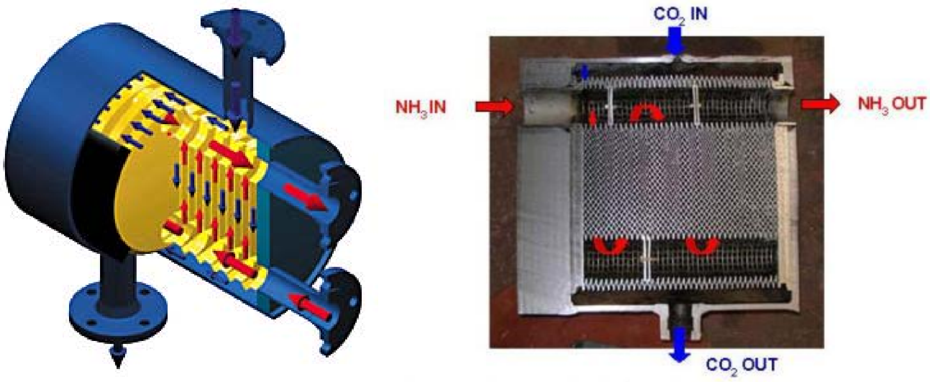


Fig. A2.76 Picture of the plate and shell heat exchanger.

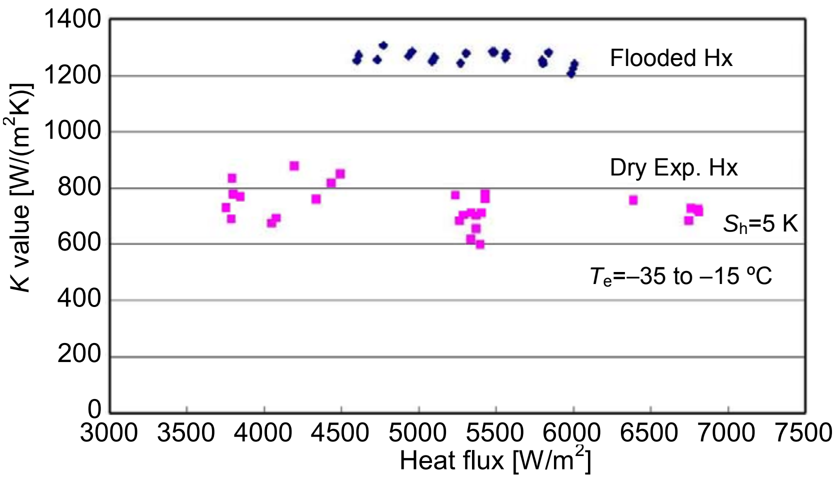


Fig. A2.77 Comparison of K values for flooded and direct expansion heat exchangers.

A2.15.4 CO_2 air cooler

Applications using carbon dioxide as a secondary refrigerant have been introduced in the market. Because of the good transport properties of carbon dioxide, turbulent flow is easily attained using less pump power. And utilization of the latent heat improves the heat transfer characteristics. The over all heat transfer coefficients are improved by more than two times compared to the conventional secondary refrigerants and as a result the heat exchanger size is reduced.

A2.15.5 Conclusions

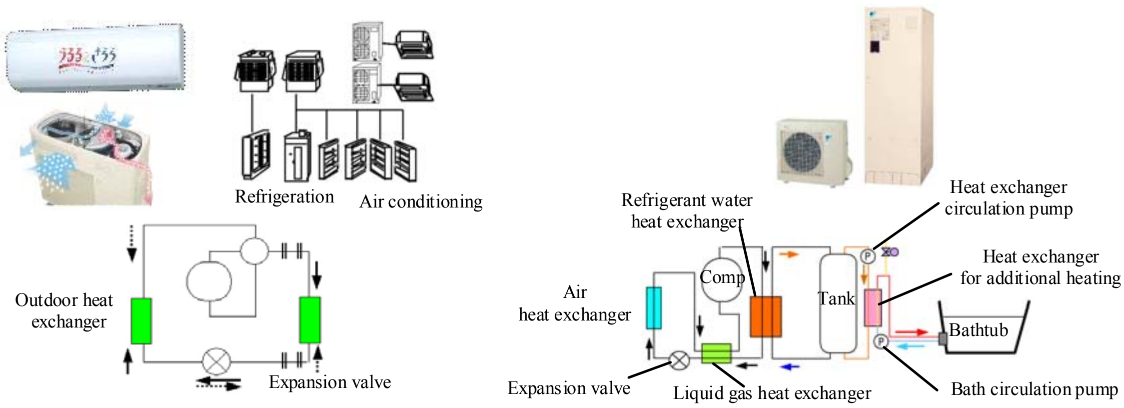
Plate and shell heat exchangers are gasket free and fully welded. Their application in industrial refrigeration offers reduced refrigerant charge, compactness as well minimized refrigerant leakage.

A2.16 Trends in Next-generation Heat Exchangers for Heat Pump Systems

Shigeharu TAIRA (Daikin Industries., Ltd.)

A2.16.1 Introduction

In the air conditioning industry, heat pump systems are mainly used in air conditioners and heat pump water heaters. Fig. A2.78 (a) shows the outline of air conditioning products and a system circuit diagram. Fig. A2.78 (b) shows a heat pump water heater. “Heat pump water heater” refers to models, such as the Eco-cute, jointly developed by electric power companies and water heater manufacturers. The heat exchanger is the key system component in both air conditioners and water heaters.



(a) Heat pump air conditioner (b) Heat pump hot water supply unit (Eco-cute)

Fig. A2.78 A heat pump system from an air conditioning manufacturer
(air conditioner and hot water supply system)

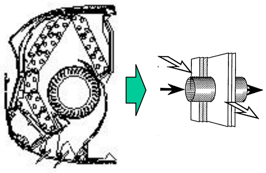
The revised Law Concerning the Rational Use of Energy in Japan requires manufactures of air conditioners to improve the energy efficiency of their products. Development of products incorporating heat exchangers of improved performance is therefore essential, and competition in this field has become fierce in recent years.

This paper examines trends in the development of heat exchangers by air conditioning manufacturers, focusing on actual cases.

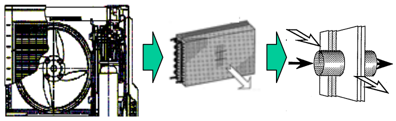
A2.16.2 Trends in various types of heat exchangers

1. Heat pump air conditioner

Heat exchangers for air conditioners are roughly categorized into those for indoor units and those for outdoor units. Cross-fin heat exchangers consisted of copper tube and aluminium fin are under development separately for both indoor and outdoor units. Those for indoor units are categorized into main heat exchangers and auxiliary heat exchangers. Many recent products are equipped with auxiliary heat exchangers to enhance the performance. Cross-flow fin heat exchangers, shown in Fig. A2.79, are used for air conditioners. Fig. A2.79 (a) shows an indoor heat exchanger, and Fig. A2.79 (b) shows an outdoor heat exchanger. Heat exchangers are largely divided into two types: those for indoor use and those for outdoor use. They consist of aluminium fins about 0.1 mm thick and copper pipes 6 to 9.5 mm in diameter. Recent heat exchangers product development is driven by structural design technology to optimise air flow and refrigerant flow, while also considering noise level, water spray at the heat exchanger and heat exchanging performance.



(a) Indoor heat exchanger



(b) Outdoor heat exchanger

Fig. A2.79 Air conditioner heat exchanger

2. Heat pump water heater (Eco-cute)

The heat source of the Eco-cute works by a principle similar to that of the outdoor unit of an air conditioner, and is categorized as both an air heat exchanger and heat exchanger to heat water. Air heat exchangers are cross-fin heat exchangers like those of the outdoor unit of air conditioners, and exchange heat between a refrigerant and the air as an evaporator. To improve their performance, the shape of the heat exchanger tube and that of the fin must be optimized. A heat exchanger to heat water is an element device to heat water through exchange of heat between water and the refrigerant made hot by adiabatic compression by a compressor. Various manufacturers have developed their own structures and forms, and are striving to improve their efficiency by expansion of the heat transfer area, optimization of the groove shape, etc. However, problem of clogging due to the structure and water flow speed is a concern.

A2.16.3 Technological Trends in next-generation heat exchangers for heat pump systems

This section examines possible future trends of various heat exchangers for heat pump air conditioners and the Eco-cute, focusing on heat exchangers for room air conditioners and the Eco-cute.

1. Heat exchangers for air conditioners

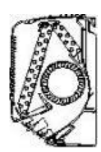
Table A2.8 shows the historical change in the layout of indoor units of representative models of room air conditioners manufactured by Daikin's [1]. As the table shows, heat exchangers of indoor units have become larger year after year, and have evolved to the present form, whereby they surround the fan. This has made providing sufficient air passage more difficult, and has complicated the layout of components. Reliability in terms of air flow, noise level and water spray must be taken into consideration. As Table A2.8 shows, heat exchanger tubes have become thinner from φ9.52 to φ7, and heat transfer performance has been improved by the adoption of grooved tubes and slit fins. Since 1995, energy efficiency and a compact exterior to enable more flexible installation have become more important, and development efforts have focused on improving efficiency through increased volume of heat exchangers. However, as Table A2.8 shows, the introduction of multi-stage heat exchangers starting in 1990 had the effect of changing the layout—with a larger heat exchanger—without having much effect on the volume of the product. Heat exchangers are expected to become more integrated in the future.

Technological development of heat exchanger tubes, fins and their materials, and improvements in performance, reliability and cost are likely to become increasingly important in heat exchangers for air conditioners.

Fig. A2.80 shows the historic change in heat exchanger tubes of air heat exchangers. They evolved from the smooth tubes used before 1980 to tubes with processed (grooved) inner surfaces with improved heat transfer area and characteristics by 1995. Improvement in processing and manufacturing technologies has helped to enhance further the heat exchanging performance. Since 1995, and in response to the adoption of substitute refrigerants R410A and R407C, research and development of heat transfer technology and improvements in processing to improve heat transfer and energy efficiency have led to better performance. New shapes of welded pipes, such as those with a W groove or N groove, have been developed to suit the characteristics of specific refrigerants. In the future, further advancement of processing technology to incise the groove is expected to enhance further heat transfer efficiency, but cost may be a concern.

Table A2.9 also shows the historical change in the fins of air heat exchangers in indoor and outdoor units. Table A2.9 shows that fins of air heat exchangers of indoor units have become slimmer as the tubes have become thinner, and slit shapes have become diverse. This has been necessitated by factors other than performance, such as air flow noise and other practicality factors, including cost. The shapes of fins of outdoor units have also become diverse due to the need to improve defrost performance during heating operation and to reduce operating noise. In the future, further advancement of overall system optimisation is needed, including fin processing technology, cost reduction, optimisation of fin pitch, and improvement of tube expanding ratio.

Table A2.8 Change of layout of indoor units

Year	1970's	1980's	Early 1990's	1993 onwards	1995 onwards	1999 onwards
Structure						
Heat transfer pipe	φ 9.52 Flat	φ 9.52 Flat	φ 7spiral (Trapezoid ditch)	φ 7 spiral (Trapezoid ditch)	φ 7 spiral (Trapezoid ditch)	φ 7 spiral (Trapezoid + slim ditch)
Fin	Plate fin	Slit fin	Looper fin	Looper fin	Looper fin	Looper fin

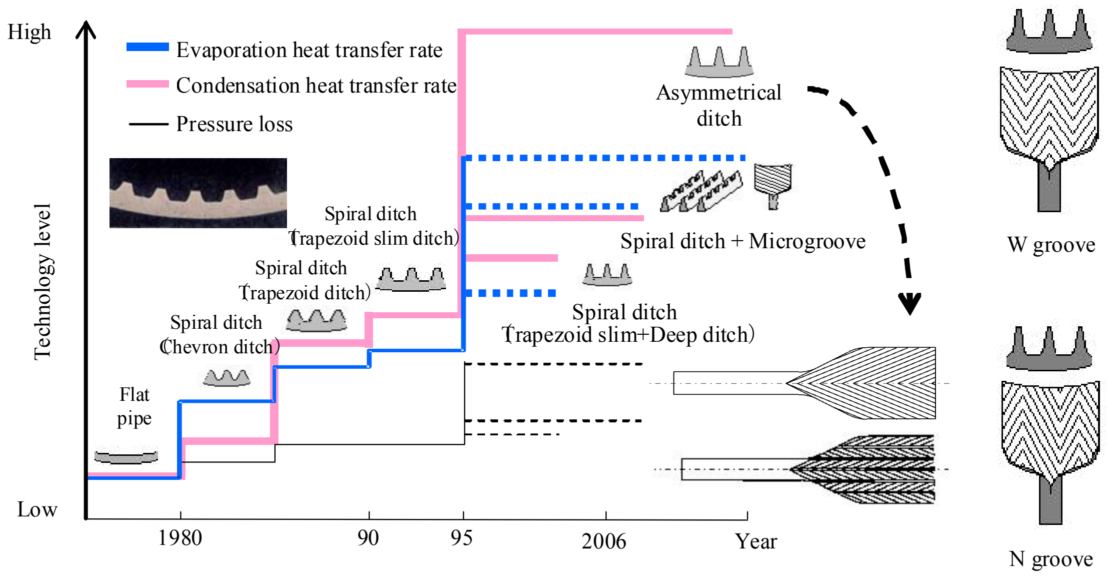
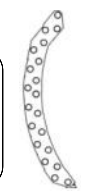
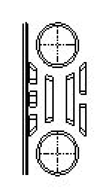
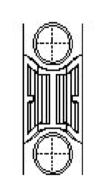
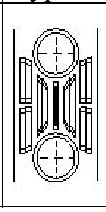
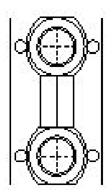
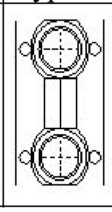
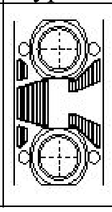
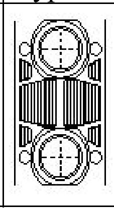
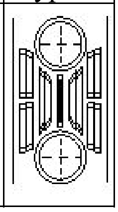


Fig. A2.80 Trend in heat transfer pipe of heat exchangers

Table A2.9 Trend in heat exchanger fin

Use	Indoor				Outdoor				
Type	Type A	Type B	Type C	Type E	Type A	Type B	Type C	Type D	Type E
Fin shape Line pace Step pace Fin pace									

2. Heat exchangers of heat pump water heater (Eco-cute)

Heat exchangers to heat water are especially important as the next generation of heat exchanger for the Eco-cute. Generally, a double tube construction is used in water heat exchangers. When this is applied to heat pump water heaters, a leakage detection ditch is added to prevent mixture of refrigerants or oil and water in the event of refrigerant leakage (see double tube shown in Table A2.10). However, this type of double tube has the drawbacks of being heavy, expensive, and difficult to reduce in size because the bending R cannot be reduced. To address this, Daikin developed a new heat exchanger with a capillary tube acting as a flow channel for CO_2 (see smooth tube shown in Table A2.10), wound around the core pipe as water flow and both brazed. The shape of the water heat exchanger is presented in Fig. A2.81. The heat transfer coefficient on the water is much smaller, also the weight and volume of the water heat exchanger is about 10 to 30% less than those of a double-tube exchanger.

For further energy conservation, a water heat exchanger using an innovative dimple water tube was developed. As Fig. A2.82 shows, water in the dimpled tube is stirred, which results in efficient mixing of high temperature and low temperature water compared with conventional heat exchangers to heat water, in which the water flow is straight.

The improvement in performance was proven by the time-series images of the dimple tube. Fig. A2.83 shows time-series images of the dimple tube. The images of the protrusion captured from the side of the acrylic tube show the detailed flow around the protrusion. Fig. A2.84 shows the downstream whirlpool structure around the protrusion. The flow near the wall continuously forms a small vertical vortex after passing the protrusion; the vortex has with its central axis along the tube axis, and is about the size of the radius of the protrusion.

Table A2.10 Comparison of water heat exchanger

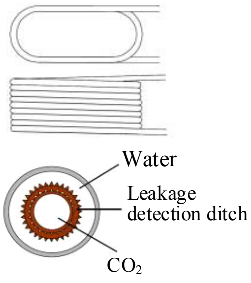
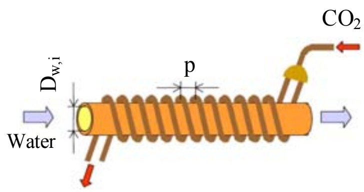
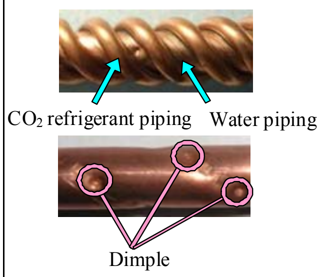
	Former type	New type	
Spec.	Double tube	Smooth tube（initial time)	Dimple tube (latest)
Outline of shape			
Capacity ratio	1.00	0.89	0.89
Weight ratio	1.00	0.74	0.64
Performance	1.00	1.00	1.00

Fig. A2.81 Appearance of water heat exchanger

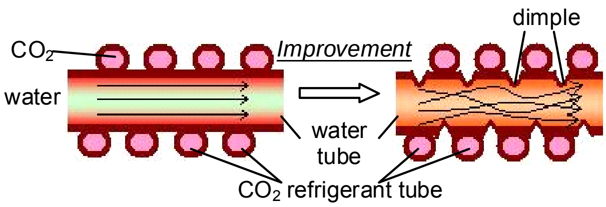


Fig. A2.82 Improvement of a heat transfer rate of a water heat exchanger

Along with this vortex, the flow of the main flow side forms a similar vortex as if it is induced, and both are then mixed as shown in Fig. A2.83. As shown by the dashed arrow in Fig. A2.84, a small vertical vortex is formed after the protrusion, which reaches to the tube centre on both sides.

By using the dimple tube, the heat transfer performance is doubled compared to a smooth tube. The accelerated heat transmission can be seen especially in the low Reynolds number field as shown in Fig. A2.85.

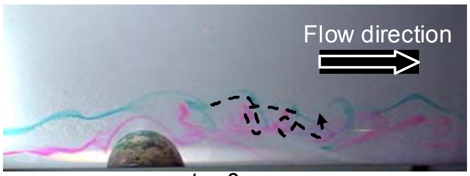


Fig. A2.83 Visualization of stream-lines around the dimple.

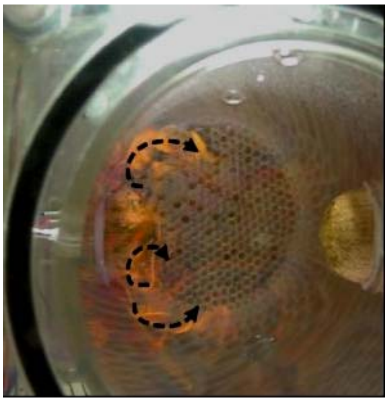


Fig. A2.84 Vortex structure around the dimple.

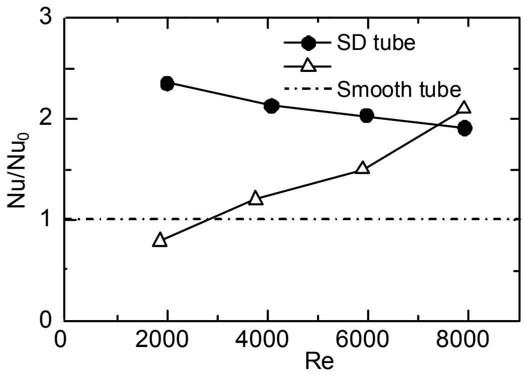


Fig. A2.85 Heat transfer performance of the water heat exchanger

A2.16.4 Conclusions

The above investigation of development trends in heat exchangers in air conditioning based on actual cases reveals the following key points for future development:

1. Air heat exchangers: more compact, higher performance, lower cost, measures against water spray.
2. Heat exchangers to heat water: higher performance, higher defrost performance, better reliability, more compact, lower cost.
3.Thanks to development of water heat exchangers, expander-compressors, and other key components and improvement of system technologies, heat pumps may contribute significantly in future to addressing the challenge of global warming.

In the future, development of breakthrough technology in performance, practicality, cost, materials and other aspects is needed. And DAIKIN wishes to contribute to the global environment protection as being in the slogan, "The Heat pumps saves the earth".

A2.17 Development of a High-Performance Fin-and-Tube Heat Exchanger with Vortex Generator for a Vending Machine

Fuji Electric Advanced Technology, Co. Ltd.
Fuji Electric Retail Systems, Co. Ltd.

A2.17.1 Introduction

Beverage vending machines are widely used on street corners and offices in Japan. Since such vending machines have a refrigerator and heater to keep beverage cold or hot, energy saving is strongly required as well as refrigerators and air-conditioners. A fin and tube heat exchanger is used for heat exchange between refrigerant and air. A heat exchanger used in beverage vending machines has a problem of condensate and frost on fin in airflow. Delta-wing-vortex generators as shown in Fig. A2.86 were developed for heat transfer enhancement with small pressure loss.

Flow and wall temperature field around the vortex generators were visualized using double size model. Heat transfer enhancement mechanism was considered based on the visualized results, and the configuration of turbulence promoter was optimised. Moreover, heat transfer performance and pressure loss for a prototype heat exchanger was measured.

A2.17.2 Visualization and measurement methods

Flow and heat transfer characteristics of flows between fins were investigated using a double scale model. In the flow visualization experiment, water was used as the working fluid. Streamlines around vortex generators and tubes were visualized by particle tracing method, and wake area behind tubes was visualized by dye injection method. On the other hand, in the heat transfer measurements, air was used as the working fluid. Fin surface was heated by a thin conductive plastic film heater. The vortex generators and tube wall was not heated. Wall temperature field was measured by an infrared thermometer. Emissivity of the surface was calibrated using measured temperature by thermocouples at 6 points. Local Nusselt number, *Nu*, and overall Nusselt number, Nu_m, were defined by the following equations:

$$Nu = \left[q/(T_w - T_b)\right](2H)/\lambda$$
$$Nu = \left[q/\Delta T_m\right](2H)/\lambda$$

where q is heat flux, H is fin pitch, l is thermal conductivity, T_w is local wall temperature, T_b is air bulk mean temperature, and ΔT_m is integrated temperature difference over the heat transfer area.

The configuration of vortex generators is shown in Fig. A2.87. Delta wing was set in the front of tube and two delta winglets were set behind tube.

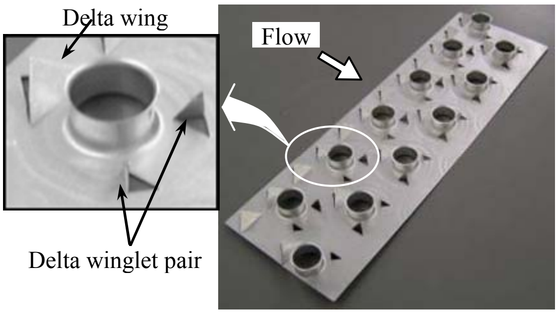


Fig. A2.86 View of the fin with vortex generator.

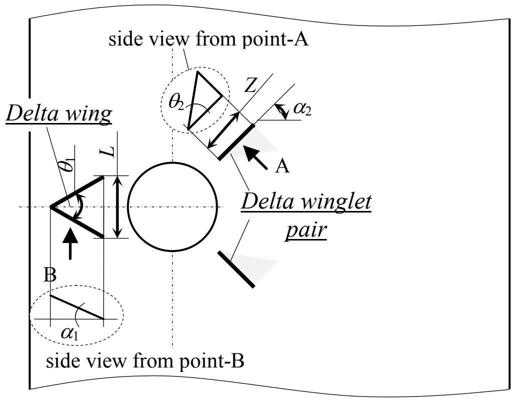


Fig. A2.87 Configuration of delta wing and delta winglet pair.

A2.17.3 Experimental results

Experimental results on streamlines around vortex generators and *Nu* distribution on the heat transfer surface are shown in Figs. A2.88 (a), (b). For the fin with only a delta wing (Fig. A2.88 (a)(i)), a pair of longitudinal vortex was visualized behind the delta wing. The area with high heat transfer was observed along the longitudinal vortex. It was shown in Fig. A2.88 (b) that the longitudinal vortex was disturbed by the delta winglet, and flow into the wake area behind the tube was observed. It can be said that delta winglet is effective for a decrease in wake area behind tube as a turbulence promoter. The effect led to an improvement in heat transfer in the area behind of the tube.

Overall average Nusselt numbers over the fin surface, Nu_m, are plotted in Fig. A2.89 against Reynolds number, *Re*. Heat transfer could be improved 50 to 70 % by delta wing and delta winglets. The enhancement ratio increased with increasing *Re*. The reason might be that turbulent induced by vortex generator became stronger for flow with higher *Re*.

The effect of vertex angle of delta wing, θ_1, on heat transfer enhancement ratio is shown in Fig. A2.90. The heat transfer enhancement ratio is the ratio to that for smooth fin without delta wing. The attack angle, α_1, was set to 35-degree. Heat transfer enhancement ratio became higher with an increase of θ_1. The effect of delta winglet configuration on wake area behind the tube is shown in Fig. A2.91. The inclined angle, θ_2, was varied from 20-degree to 90-degree. The attack angle, α_1, was set to 35-degree. The area is shown as the ratio to that for smooth surface without delta winglet. Flow visualization results by dye injection method are shown in the upper. The lower part in the left picture was for the smooth surface. It was shown that the area became the minimum at θ_2=45-degree.

In order to evaluate the effect of delta wing and delta winglets on the performance of heat exchangers, a test model of heat exchanger was manufactured. Tested results are shown in Fig. A2.92. It was shown that high heat transfer enhancement could be obtained by delta wing and delta winglets even in low *Re* flow condition.

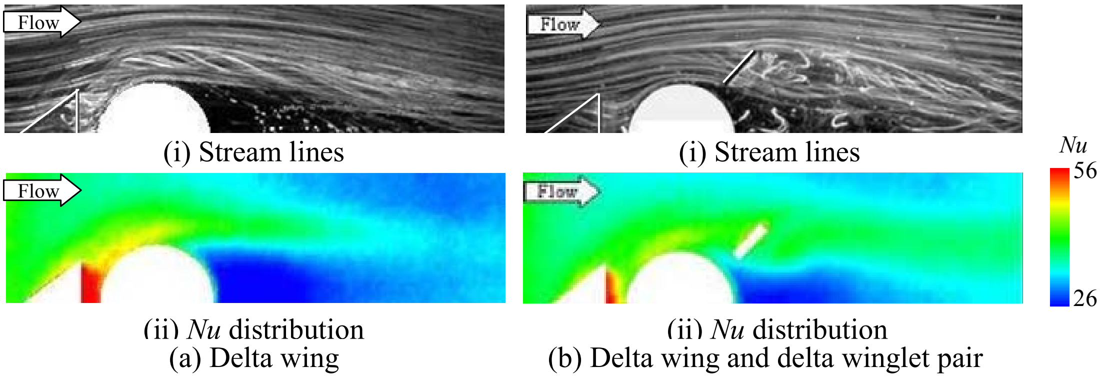


Fig. A2.88 Comparison between streamlines and *Nu* distribution around vortex generator
(*Re* = 1000).

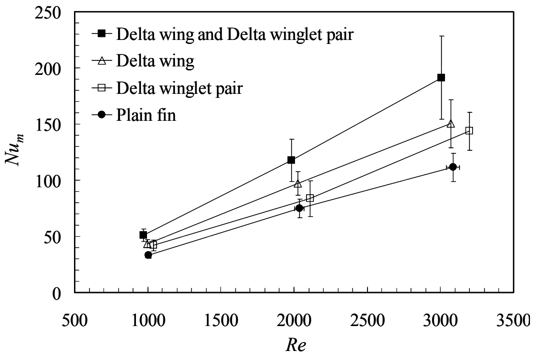


Fig. A2.89 Measured mean Nusselt number Nu_m.

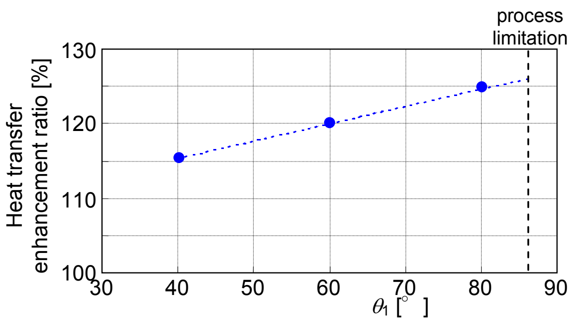


Fig. A2.90 Effect of delta wing angle θ_1 on average heat transfer

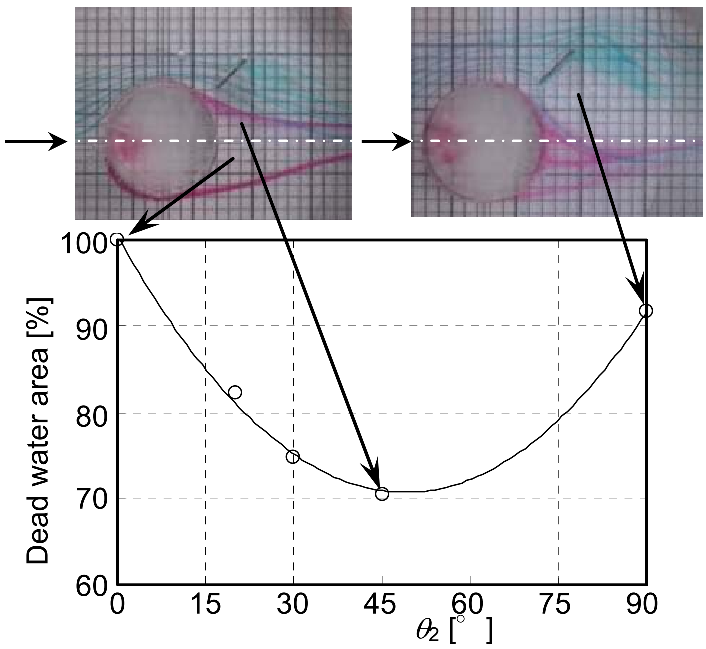


Fig. A2.91 Effect of delta winglet angle θ_2 on dead water area

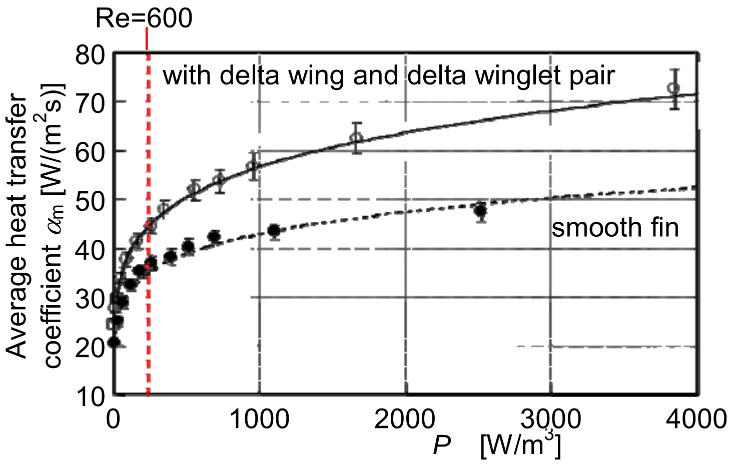


Fig. A2.92 Correlation between heat transfer performance and fan power per unit volume P

A2.18 Numerical Analysis of Gas-Liquid Two-Phase Flow

Isao Kataoka (Osaka University)

A2.18.1 Overview

Numerical analysis of two-phase flow is carried out by numerically solving the basic conservation equations of mass, momentum and energy conservations of two-phase flow. Since gas and liquid phases coexist in gas-liquid two-phase, conservation equations are derived using time and/or space averaging based on appropriate two-phase flow models. Homogeneous flow model, slip model and drift flux model are established as mixture model where two-phase flow is treated as one fluid of gas and liquid mixture. Two-fluid model is also developed where gas and liquid phases are treated separately and mass, momentum and energy conservation equations are formulated for each phase. Two-fluid model is the most accurate model among various models and widely used in numerical analyses of gas-liquid two-phase flow. However, in practical numerical analysis, basic equations based on appropriate model are used considering flow regime, treated phenomena, calculation time and accuracy. Since basic equations of gas-liquid two-phase flow are given in averaged forms, in solving the basic equations, constitutive correlations for pressure drop, heat transfer coefficient, relative velocity between phases, interfacial transfer terms etc. are needed. Basic conservations equations and constitutive correlations in typical two-phase models were introduced and explained.

Numerical methods for gas-liquid two-phase flow were reviewed and explained. Basic methods of numerical analysis of gas-liquid two-phase flow are same as those of single phase flow. Explicit method, semi-implicit method and implicit method are used depending upon the purpose of analysis. However, in practical analyses, semi-implicit method (SMAC method) is commonly used also in gas-liquid two-phase flow. Since gas and liquid phases coexist in gas-liquid two-phase flow, in numerical analysis, gas-liquid two-phase flow should be treated as compressible fluid. Therefore, continuity equation (mass conservation equation) should be treated implicitly. Furthermore, interfacial momentum transfer term and interfacial heat transfer term in two-fluid model should be treated implicitly. Some examples of finite difference equations based on semi-implicit method were shown.

Some examples of numerical analyses of gas-liquid two-phase flow related to heat pump were shown. Transient air-water two-phase flow in vertical pipe (flow and void excursion), blow down phenomena where vapor-liquid two-phase flow is ejected from high pressure pipe due to the pipe break, two-phase natural circulation in vessel (bubble induced tow-phase flow in vessel), boiling two-phase flow in complicated geometry (rod bundle) were numerically analysed and compared with experimental data with successful agreements.

More advanced methods of numerical analyses of gas-liquid two-phase flow were also reviewed and explained. Lagrange simulations of bubbles or droplets are recently carried out as detailed numerical analyses of bubbly flow and droplet flow. In this simulation, trajectories of each bubble and droplet are numerically calculated by solving equation of motion in continuous phase. Calculations of numerous number of bubbles and droplets become possible due to the remarkable improvement of computer capacity. Using Lagrange simulation, detailed behaviour of bubble flow and droplet flow can be analysed such as diffusion of bubbles or droplets and deposition of droplet to pipe wall. Numerical processes and necessary correlations for bubble and droplet behaviours were explained and some examples of simulations were shown.

Interface tracking method was also reviewed and explained. In this method, direct numerical analysis of Navier-Stokes equations in gas and liquid phase were carried out without using averaged equations. The behaviours of gas-liquid interfaces of bubbles or droplets are directly calculated in details. As a typical interface tracking method, VOF method (volume of fluid method) was explained and some examples of simulation results were shown. Since the load of calculation in this method is very high, the application of this method is limited to a few bubbles and droplets. However, this method is useful in obtaining the detailed behaviours of bubbles and droplets in gas-liquid two-phase flow.

A2.19 Manufacturing and Materials

A2.19.1 Development trends in inner-grooved tubes in Japan

Mamoru Houfuku (Hitachi-Cable, Ltd.)

1. Introduction

Inner-grooved copper tubes are widely used as the refrigerant side heat transfer tubes employed in fin-tube-type heat exchangers and in shell and tube type heat exchangers.

In our company, the mass production of the above tubes used in fin-tube-type heat exchangers, shown in Fig. A2.93, started in the latter half of 1970's.

In this paper, changes in the production methods of the above tubes and in the shapes of the inner surface grooves are presented for the inner-grooved tubes produced by our company since the beginning.

Fig. A2.93. Fin-tube-type heat exchanger

In Japan, where resources are poor, the above tubes have been used since the 1970's because of the strong consciousness regarding saving energy and the highly functional nature of air conditioning equipment. Recently, the performance of these tubes has been further improved largely due to two environmental issues.

One of these issues is CFC (e.g. Freon) regulation. In order to protect the ozone layer, alternative refrigerants such as R410A, R407C etc. were adopted in the 1990's in Japan. R407C was the strongest candidate as alternative refrigerants at that time because of its similarity with the conventional refrigerants R22. However, the heat transfer coefficient of this refrigerant is lower, 30% to 40%, compared with R22. As a result the tubes needed to be improved in terms of function.

The second issue is global warming. In order to prevent global warming from occurring due to the emissions of carbon dioxide, energy saving laws were enacted to improve energy efficiency. The above tubes also had been to be improved in function to comply with the energy saving requirements of air conditioning systems.

In our company, improvements in the function of the above tubes have been made through changing the inner surface groove shapes in response to these environmental issues.

2. Changes in the shapes of the inner-grooved tubes

Fig. A2.94 shows the changes in the shapes of the inner-grooved tubes. The heat transfer characteristics are expressed by the ratio of the heat transfer coefficient at the inner surface to that of the smoothed tube, which is assumed to be 1.

The tubes shown in Fig. A2.94 are representative mass-produced inner-grooved tubes from our company. The ratio of the heat transfer coefficient with condensation and the ratio of the heat transfer coefficient with evaporation are shown by the solid line and broken line, respectively.

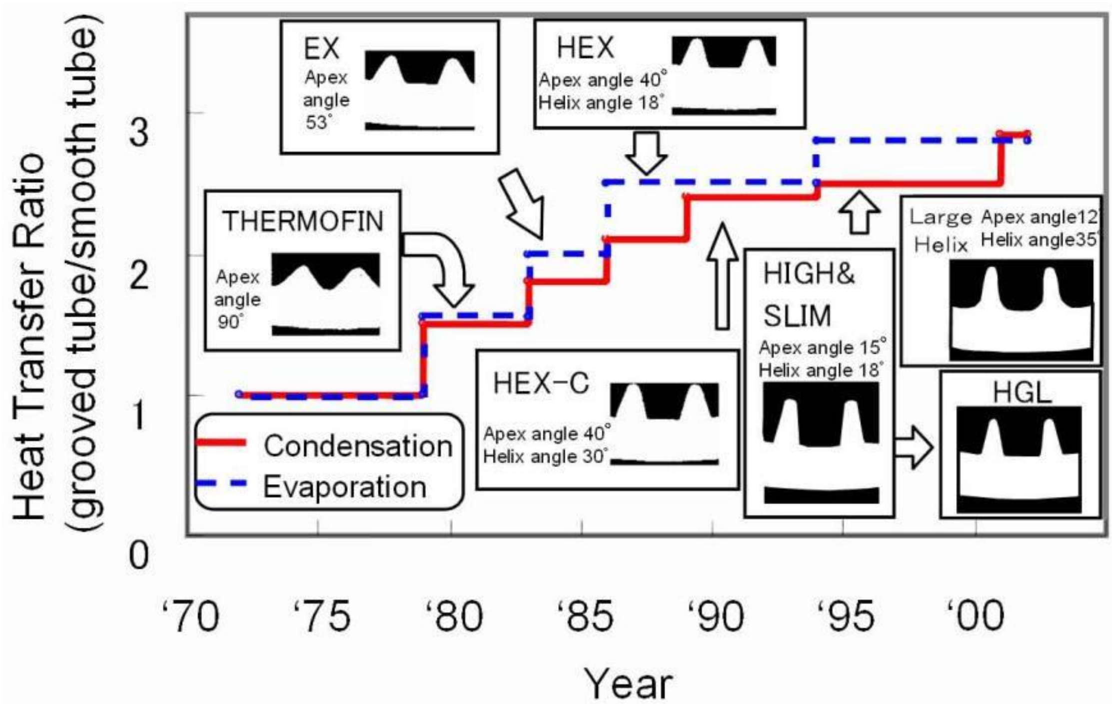


Fig. A2.94 Changes in the shapes of the inner-grooved tubes

When the mass production of the above tubes began, the fin apex angle was 90 degrees of a large triangle. After that, EX with a fin apex angle of 53 degrees and HEX with a fin apex angle of 40 degrees were developed to improve the heat transfer performance through adopting slim fins. Additionally, in the latter half of the 1980's, HEX-C, which improved condensation performance, was developed by changing the helix angle of HEX to 30 degrees.

In the 1990's, high and slim inner-grooved tube was developed and adopted for alternative refrigerants. At this time, highly functional tubes using crossed grooves and fine structural secondary grooves, etc. were developed.

In the early 2000's, HGL, for which the weight was reduced by the optimization of groove shapes, and large helix tube, for which the condensing function was improved by the large helix angle of the grooves, were developed and mass-produced.

A2.19.2 Development of High Strength Copper Tube

Takashi Shirai (Kobelco & Materials Copper Tube, Ltd.)

1. Overview

Phosphorus deoxidized copper tube (JIS H3300 C1220T) has been widely used in heat exchanger and tubing etc. for air-conditioning and refrigerating units from the past, because of its balanced performance among strength, workability, thermal conductivity, soldering and brazing, and corrosion resistance.

However recent Japanese refrigerating and air-conditioning market is demanding superior performance tube in strength and heat resistance to current C1220 tube. Because of the market strongly desire to prevent going to thicker wall thickness of tube due to alternative refrigerant and to do cost saving using thinner wall thickness of tube. Three kinds of high strength copper tubes are introduced those are MA5J, KHRT and HRS35LT. And they have already marketed. The status of market result and the reduction rate of unit weight of these tubes compared with C1220 tube are shown in Table A2.11.

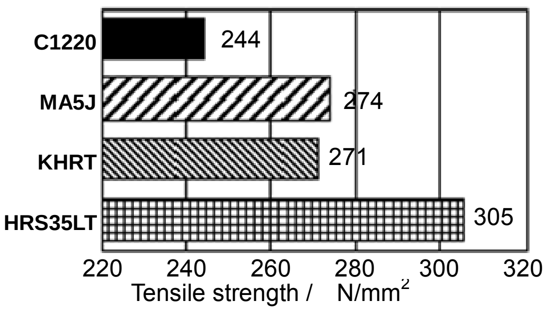


Fig. A2.95 Tensile strength of high strength copper tube (φ 9.52 × t0.80 mm, OL)

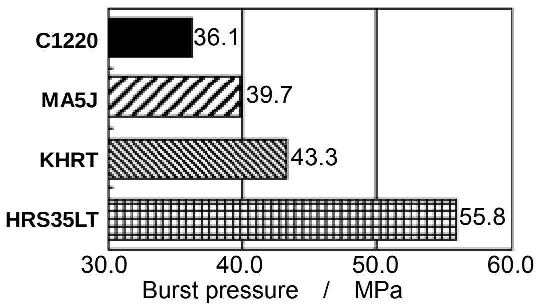


Fig. A2.96 Burst pressure of high strength copper tube (φ 9.52 × t0.80 mm, OL)

Furthermore these high strength copper tubes were registered in Japanese Industrial Standard (JIS) on 20th July 2009. Now, they are proceeding with the application to High Pressure Gas Safety Law as well.

It is expected that the low-alloyed copper tube will penetrate into the next generation of products.

Table A2.11 Weight reduction in model case (Compared with C1220)

Application	MA5J (C1565)	KHRT (C5010)	HRS35LT(C1862)
Heat exchanger for ACR	10 – 20 %		
Connection tubing		15 – 30 %	
Muffler, Accumulator			20 – 40 %
Furnace brazing heat exchanger		20 – 30 %	

A2.19.3 Corrosion Control Design for Automotive Aluminium Heat Exchangers

Kazuhiko Minami (Showa Denko K.K.)

1. Introduction

Aluminium compact heat exchangers are used in automotive air-conditioning systems and include condensers, evaporators, and heater cores. In this report, we explain the outline of corrosion resistant methods for the condenser which is exposed to the severest corrosive environment in the automobile, and then, explain the formation of sacrificial corrosion layer caused by zinc and the effect of zinc coating from the standpoint of zinc diffusion behaviour in aluminium.

2. Corrosion Prevention Technology for Condenser

Condenser is composed of a header, tubes and fins as illustrated in Fig. A2.97. Inside the tube, the refrigerant is flowing to work as air-conditioning system. If the tube however, is pitted due to corrosion and the refrigerant leaks, the condenser does not work any more as an air-conditioning system. Moreover, the cross-sectional shape of the tube is extremely fine. Therefore, a sacrificial corrosion layer is formed on the surface of the tubing for the long-term corrosion resistance.

In order to protect condenser tubes from corrosion, a conventional method has been used, in which zinc is sprayed on the tube surface (Fig. A2.98), and then diffused by a brazing heat treatment to form a sacrificial corrosion layer of Al-Zn alloy on the tube surface.

Zinc-added aluminium alloy changes in electrical potential. Zinc diffusing layer has lower potential compared with the core layer and works as a sacrificial corrosion layer. Fig. A2.99 shows zinc diffusion from the tube surface analysed by EPMA. The brazing heat treatment process enhances the diffusion of the zinc spraying layer attached to the tube surface towards the tube core layer and the core layer is protected from corrosion. Since the zinc diffusion distance means the thickness of the sacrificing corrosion layer, the zinc diffusion layer needs to be controlled in order to realize lighter and more efficient tubes.

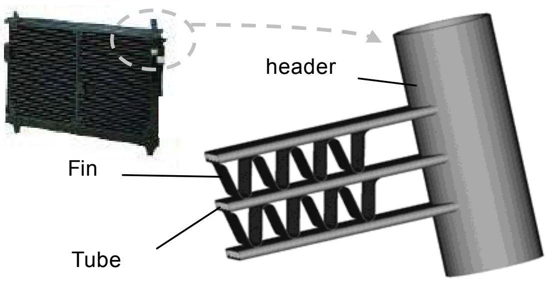


Fig. A2.97 Configuration of condenser

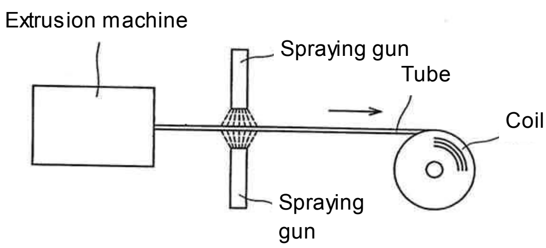


Fig. A2.98 Zinc spray to tube surface

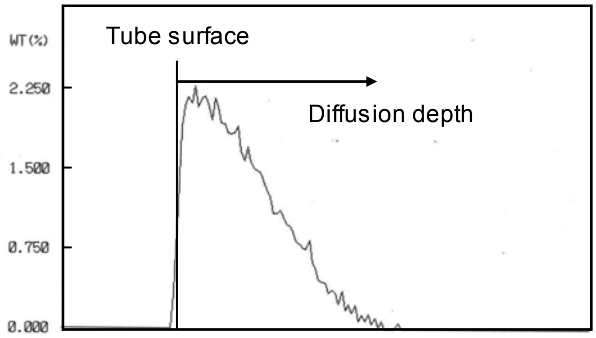


Fig. A2.99 Zinc diffusion layer in aluminum

3. Zinc Diffusion

As for the zinc diffusion in aluminium, if its diffusion coefficient (D) keeps constant despite the position, the second equation of Fick's diffusion is indicated as in Eq. (1).

$$\frac{\partial c}{\partial t} = D\frac{\partial^2 c}{\partial x^2} \tag{1}$$

When the boundary conditions for the thin-film diffusion source are satisfied, the solution will be Eq. (2).

$$C = \frac{A}{\sqrt{\pi D t}}\exp\left(-\frac{x^2}{4Dt}\right) \tag{2}$$

where A is the zinc film thickness and D is the diffusion coefficient indicated in Eq. (3) with activation energy (Q) of the zinc diffusion in aluminium and frequency term (D_0).

$$D = D_0 \exp\left(-\frac{Q}{RT}\right) \tag{3}$$

From the above, reducing the zinc coating amount in order to decrease corrosion depth for realizing lighter weight and higher performance is considered to be a possible method because zinc diffusion distance varies depending on brazing temperature, time and zinc coating amount. When computed from Eq. (2) using the brazing condition 873K × 10 min, the zinc-coating thickness 1.4 μm (10 g/m^2), frequency term (D_0) and Q, results are obtained as shown in Fig. A2.100. Similarly, results obtained when zinc-coating thickness is 0.7 μm (5 g/m^2) are also shown. From this, it is expected that reducing zinc coating amount can lead to the decrease in zinc diffusion layer and therefore to reduction of sacrificial corrosion layer.

However, in the area where the zinc coating amount is smaller, zinc spray particles are coated unevenly on the tube surface and therefore, there is a possibility that the sacrificial corrosion layer may not work effectively. So we have observed the corrosion resistance of the less zinc coated area during the combined cycle corrosion testing in the neutral environment.

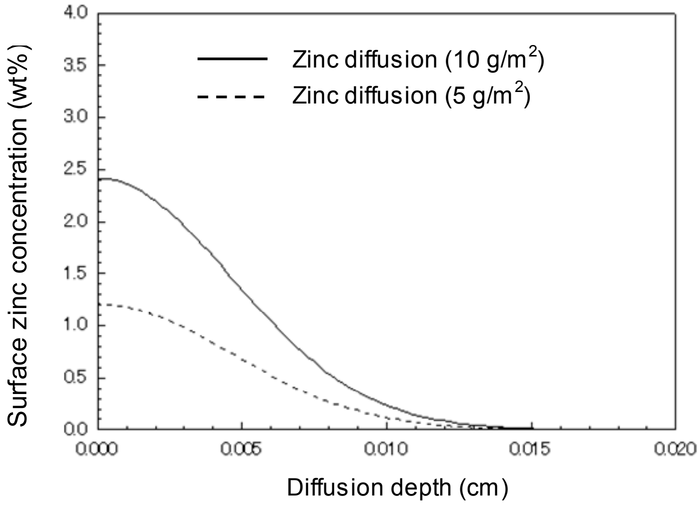


Fig. A2.100 Calculated results of zinc diffusion in aluminum

4. Experimental Methodology

The material for a heat exchanger core used in this study is as below.
Fin: A3203 + 1.2% Zn core material brazing sheet
Tube: Al-0.4% Cu-0.2% Mn

Zinc is sprayed to the tube immediately after extrusion to obtain the zinc coating amount. To investigate zinc diffusion on the tube surface, zinc plated tubes are also made. The condition for brazing a heat exchanger core is 873K × 10 min. As for the zinc plated samples, the tube was tested separately by the infrared gold image furnace in the same heat conditions as the brazing conditions.

Zinc spraying amount (coating amount) of each sample was measured at five points by fluorescent X-ray analysis and the average was taken. The method to investigate the degree that the tube is coated with zinc was as below. For clear differentiation, we used COMPO image to binarize zinc and aluminium coated on the tube surface and measured how much the tube was coated with zinc as shown in Fig. A2.101. As for the measurement of the zinc diffusion layer on the tube surface, surface zinc concentration and diffusion depth were measured by conducting line analysis in the perpendicular direction to the tube cross section using EPMA. The corrosion depth was measured by a focus depth method using a measuring microscope and where the greatest corrosion was seen, the cross-sectional microscopic observation was conducted.

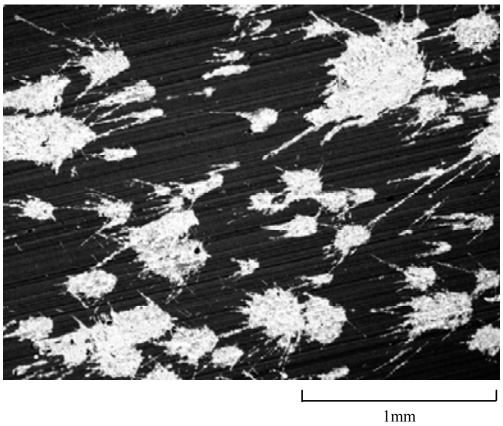


Fig. A2.101 COMPO figure of zinc sprayed tube (zinc amount 3 g/m^2, zinc coating ratio 30%)

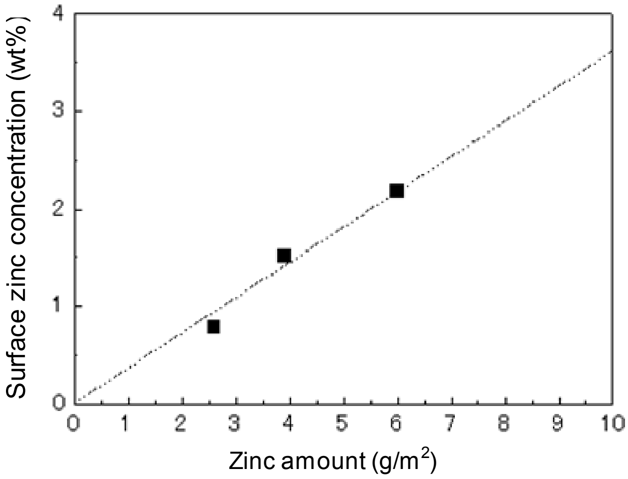


Fig. A2.102 Zinc amount and surface concentration

5. Zinc Concentration Results

Distribution of zinc concentration on the tube surface is shown in Fig. A2.102. Results of EPMA mapping analysis conducted on tube thickness cross-section are also shown in Fig. A2.103. These results confirmed the presence of zinc even in the area of the tube surface where zinc spraying particles were not coated.

In order to examine the reasons why zinc exists in the area where no zinc spraying particles are coated, we performed a heat treatment testing on the zinc plated tube and investigated the zinc surface diffusion behaviour on the tube surface. We measured the surface diffusion distance by using the surface zinc concentration obtained by EPMA line analysis. Fig. A2.104 shows a graph of surface zinc concentration and surface diffusion distance.

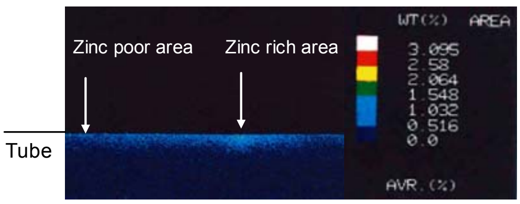


Fig. A2.103 EPMA mapping analysis

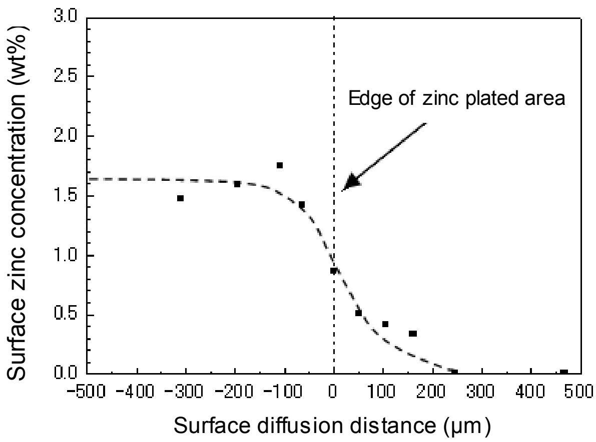


Fig. A2.104 Surface zinc concentration and diffusion distance

This result shows that zinc diffusion distance on the tube surface was approx. 250 μm. Here, when the spraying particles are dispersed evenly and the degree of the surface covered with zinc is α, the average radius of a spray particle is r, the distance between particles is d, and number of particles is n, the total area is

$$n\pi\left(\frac{d}{2}+r\right)^2 \tag{4}$$

and the total area is also obtained from α and the area of a zinc particle can be written as

$$\frac{n\pi r^2}{\alpha} \tag{5}$$

and from the Eqs. (4) and (5),

$$n\pi\left(\frac{d}{2}+r\right)^2 = \frac{n\pi r^2}{\alpha} \tag{6}$$

and when Eq. (6) is solved to obtain d,

$$d = \frac{2r}{\sqrt{\alpha}} - 2r \tag{7}$$

Using Eq. (7), if the spray particle radius is 100 μm and the percentage of zinc covered tube surface is 50%, the distance between particles is calculated to be 83 μm. Since zinc surface diffusion is larger than the distance between particles, it was confirmed that there was no zinc diffusion area existing. It was also confirmed that zinc required for anti-corrosion resistance exists based on the results of surface zinc concentration which can be estimated from the data of the surface zinc diffusing onto the areas where no zinc attached.

6. Corrosion Test Results

A cross-sectional microscopic picture taken after corrosion testing is shown in Fig. A2.105 and the results of zinc coating amount and measured maximum depth of corrosion in Fig. A2.106. These results clearly indicate that on the heat exchanger core which has less zinc coating, the type of corrosion is not pitting corrosion and there is enough zinc exists to protect the tube from corrosion, and therefore good corrosion resistance is obtained when tested by the combined cycle corrosion test in a neutral environment. It suggests a possibility to reduce the corrosion depth, meaning a thinner sacrificial corrosion layer, and therefore a lighter heat exchanger for the automotive application.

Fig. A2.105 Cross section of max. corrosion

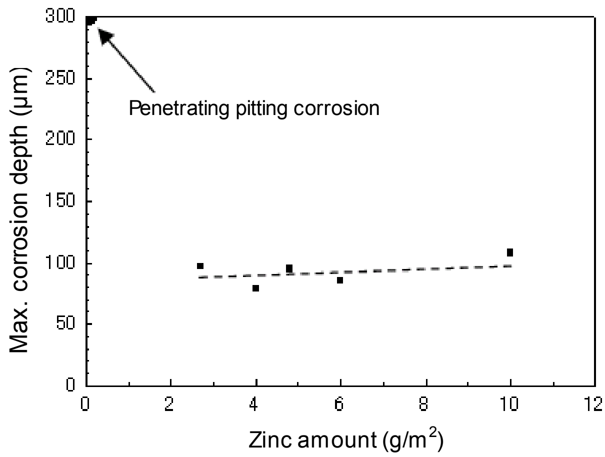


Fig. A2.106 Zinc amount and max. corrosion depth

7. Conclusions

When approx. 3 g/m^2 of zinc was sprayed, although there was some area where no zinc was attached, the entire area was covered with zinc after being given a brazing heat treatment.

We think zinc existed there because the zinc diffusion layer was formed around the spray particles due to the relatively fast surface diffusion.

The combined cycle corrosion test in a neutral environment showed a good corrosion resistance on the heat exchanger which had less zinc coating amount.

By reducing the zinc coating amount, it will become possible to thin the sacrificial corrosion layer, and therefore to realize a lighter heat exchanger for automotive application.

APPENDIX 3 United Kingdom – Scientific Contributions

A3.1 UK Process Heat Recovery Market Study – Summary Data Highlighting UK Industrial Heat Pump Market

Contents

A3.1.1 Background

The Summary Data presented below are extracted from documents prepared in support of an analysis of Process Heat Recovery (PHR) in the United Kingdom which was directed at assisting the Carbon Trust to (a) quantify the amount of heat that could be fully recovered in the process industries, (b) identify the technologies that might be use to recover this heat and (c) recommend ways in which industry can be encouraged to increase its take-up of PHR, including, but not limited to, the Enhanced Capital Allowance scheme – a scheme that allows companies to write off the investment in energy-saving equipment, provided that the equipment is 'approved' by government agencies. Industrial heat pumps are not yet in the scheme, but some heat pumps for buildings are – see www.eca.gov.uk/

The aim is of course to substantially reduce carbon emissions.

In this consultation exercise we concentrated on the sectors that are believed to have the greatest potential for PHR – the seven sectors highlighted in bold in Table A3.1 below.

The information given here includes the sector total energy use; the estimated sector potential for PHR (all in Table A3.1); the generic heat recovery types that can be used in each sector (Table A3.2- including heat pumps – highlighted in yellow) which can be related to the nature of the unit operations that are identified in broad terms in Table A3.3. The detailed breakdown of PHR opportunities in the seven chosen sectors – now divided into subsectors – are given in Tables A3.4-A3.9 (heat pumps being highlighted in yellow). The nature of each subsector is listed in Annex A3.1.1, while Annex A3.1.2 discusses the PHR technologies considered.

It is interesting to consider a 'broad brush' approach to PHR using a technology that could be applied widely to a group of unit operations – such as one of the categories in Table A3.3. For example, if heat pumps and MVR systems were applied to all applicable drying/separation processes, which consume 2821 ktoe/annum, energy savings of about 35% might result, saving 987 ktoe/a, or about equal to the TOTAL savings predicted in the more conservative approach, the results of which are detailed here. At present, of course, the manufacturing capability to produce such equipment within a reasonable timescale does not exist.

The data herein were initially presented at a meeting of HEXAG – the Heat Exchanger Action Group – www.hexag.org held at Newcastle University.

A3.1 2 Bases of Data Analysis

The figures are based upon 2004 data on UK industrial subsector energy use, by fuel, and 2004 subsector data by sector unit operation breakdown (these unit operations include space heating, low temperature process heat, distillation, drying, refrigeration etc.). Knowing the energy uses in each subsector for 2004, the subsector unit operation energy use can be estimated from overall sector figures, selecting appropriate fuels. *Note that electricity use has not been included in most instances as generating a source of waste heat, and conservative estimates of heat recovery equipment efficiency, in terms of recoverable percentages, have been assumed.*

The heat recovery potential has been assessed upon the basis of the consultant's knowledge of and research into the expected savings arising out of the use of appropriate heat recovery systems, data from a number of Government strategy documents and recent publications on PHR, and discussion with specialists in sectors and relevant technologies. Heat pumps have been included in a number of perceived opportunities, but as yet no account has been taken of a switch from, for example, gas to electricity to drive the heat pump. (Of course, gas engine-driven heat pumps may be the best option in some cases).

TABLE A3.1. SECTOR PHR POTENTIAL[1]
(Those in bold type are selected for detailed attention in Tables 4-10)

Sector	See Table	Sector Use (ktoe)	PHR Potential (ktoe)
Food & drink	**4**	**3841**	**239.3**
Tobacco products		22	1.5
Textiles	**5**	**944**	**21**
Wood & wood products		884	20
Pulp, paper & paper products	**6**	**1370**	**29**
Publishing, printing & reproduction		1090	8
Coke, refined petroleum etc.	**7**	**8900[2]**	**290**
Chemicals & chemical products	**8**	**5801**	**136.6**
Rubber & plastics products		2454	10
Other non-metallic mineral prod.	**9**	**2309**	**76**
Basic metals	**10**	**2879[3]**	**62**
Fabricated metal products		976	29
Machinery & equipment		623	-
Electrical machinery & equipment		437	8.5
Radio, TV & communication equip.		239	3
Motor vehicles, trailers etc.		998	49.5
Other transport equipment		535	1
Totals		**34,302**	**984.4**

[1] Note that space heating and compressed air heat recovery have been largely excluded, as has heat recovery from refrigeration compressors and boilers. These are generic unit operations.
[2] The data for refining of petroleum products and separate data for the chemical industry may involve some overlap. This is not evident from the statistical data, however.
[3] Note that a relatively small proportion of this energy originates from natural gas – i.e. where regenerative or recuperative burners could be used – typically 35%.

TABLE A3.2. GENERIC HEAT RECOVERY EQUIPMENT TYPES IDENTIFIED ON A SECTOR BASIS

Sector	HX Gas-gas	HX Gas-liquid	HX Liquid-liquid	Heat pumps Open cycle	Heat pumps Closed cycle	Power recovery	Direct reuse
Food & drink	**x**	**x**	**x**	**x**	**x**		
Tobacco products	x				x		
Textiles	**x**	**z**	**x**		**x**		**x**
Wood & wood products	x				x		x
Pulp, paper & paper products	**x**	**x**	**x**	**x**	**x**		
Publishing, printing & reproduction	x						
Coke, refined petroleum etc.	**x**	**x**	**x**	**x**			
Chemicals & chemical products	**x**	**x**	**x**	**x**	**x**	**x**	**x**
Rubber & plastics products	x				x		x
Other non-metallic mineral prod.	**x**				**x**		**x**
Basic metals	**x**						**x**
Fabricated metal products	x	x					x
Electrical machinery & equipment	x		x		x		
Radio, TV & communication equip.	x	x	x		x		
Motor vehicles, trailers etc.	x		x		x		x
Other transport equipment	x						

TABLE A3.3. INDUSTRIAL ENERGY CONSUMPTION BY END USE – 2004. Data in thousands of tonnes of oil equivalent (ktoe)

SIC (92) code		High temperature process	Low Temperature Process	Drying / Separation	Motors	Compressed Air	Lighting	Refrigeration	Space Heating	Other	Total
15	**Food products and beverages**	**-**	**2,466**	**282**	**262**	**-**	**-**	**272**	**-**	**559**	**3,841**
16	Tobacco products	-	14	1	1	-	-	2	-	3	21
17	**Textiles**	**-**	**374**	**117**	**110**	**-**	**-**	**-**	**343**	**-**	**944**
18	Wearing apparel	-	10	-	10	-	-	-	40	-	60
19	Leather	-	14	-	6	-	-	-	13	-	33
20	Wood and wood products	-	44	530	124	-	-	-	186	-	884
21	**Pulp, paper and paper products**	**-**	**466**	**740**	**34**	**7**	**-**	**-**	**123**	**-**	**1,370**
22	Publishing, printing and reproduction	-	-	-	228	399	-	-	-	463	1,090
23	**Coke, refined petroleum products and nuclear fuel**	**-**	**-**	**-**	**-**	**-**	**-**	**-**	**-**	**-**	**-**
24	**Chemicals and chemical products**	**681**	**1,880**	**1,033**	**856**	**221**	**-**	**242**	**150**	**738**	**5,801**
25	Rubber and plastic products	-	1,668	-	408	127	-	-	-	251	2,454
26	**Other non-metallic mineral products**	**1,670**	**125**	**118**	**185**	**-**	**-**	**-**	**33**	**178**	**2,309**
27	**Basic metals**	**2,446**	**-**	**-**	**134**	**-**	**-**	**-**	**-**	**299**	**2,879**
28	Fabricated metal products	56	521	-	12	37	41	-	287	22	976
29	Machinery and equipment	36	317	-	7	22	31	-	188	22	623
30	Office machinery and computers	-	12	-	1	3	11	-	33	7	67
31	Electrical machinery and equipment	22	153	-	7	17	61	-	157	20	437
32	Radio, television and communication equipment and apparatus	-	43	-	2	12	38	-	120	24	239
33	Medical, precision and optical instruments	12	83	-	4	10	33	-	86	11	239
34	Motor vehicles, trailers and semi-trailers	45	369	-	10	30	60	-	439	45	998
35	Other transport equipment	21	166	-	5	16	32	-	246	49	535
36	Furniture	-	168	-	-	-	-	-	404	270	842
37	Recycling	-	-	-	-	-	-	-	-	-	-
	All industry	**4,989**	**8,893**	**2,821**	**2,406**	**901**	**307**	**516**	**2,848**	**2,961**	**26,642**

Source: Department of Trade and Industry estimates derived from Office for National Statistics, Future Energy Solutions and Department of Trade and Industry data

TABLE A3.4. PHR POTENTIAL IN FOOD PRODUCTS & BEVERAGES SECTOR

	Gas-gas HX[4,5]	Gas-liquid HX	Liquid-liquid HX	Closed cycle heat pump	Open cycle heat pump	Combined heat & power[6]	Power cycles	Heat/cold storage	Heat transfer enhancement	Process integration	New process technology/ intensification
1511		10[7]	3	9							
1512	2.4[8]		3.2								
1513	5		8	45							
1520	1.8[9]					1.5					
1531		1.5	0.3								
1532			0.1		0.5						
1533	3.5		3.5	0.4							
1541	2		2		2					x[10]	
1542	4									x[11]	
1551	2	1	6	4							
1552		0.2									
1561	8[12]										
1572			0.5	0.5						x	
1581	12										
1583					20[13]					x	
1588	0.5										
1589	2.5		2.5		1[14]						
	Gas-	Gas-	Liquid-	Closed	Open	Combined	Power	Heat/cold	Heat transfer	Process	New process

[4] HX = heat exchanger
[5] All figures are in ktoe/a
[6] Included where absorption cooling may be used from an existing or future CHP unit
[7] Principally heat recovery from refrigeration condensers for washing etc.
[8] Excludes opportunities for heat recovery for space heating in poultry houses.
[9] A proportion of this could be used for space heating rather than process re-use.
[10] The energy savings due to process integration are not quantified – this needs highly specialised analyses.
[11] 35% savings are suggested on new plant using process integration
[12] Includes a small proportion for space heating
[13] Most conventional PHR opportunities are exploited already
[14] An ill-defined subsector in terms

	gas HX[15]	liquid HX	liquid HX	cycle heat pump	cycle heat pump	heat & power[16]	cycles	storage	enhancement	integration	technology/ intensification
1591	2	3	3	5	5					x	
1592	1										
1594			0.5	0.5[17]							
1596			1	5	2					x	
1597	10			10[18]		x[19]					x[20]
1598			2								
TOTALS	**67.4**	**15.7**	**38.8**	**79.9**	**35**	**1.5**	Note[21]	Note[22]	Note[23]	Note[24]	**1**[25]
SECTOR TOTAL											**239.3 ktoe/a**

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[15] HX = heat exchanger
[16] Included where absorption cooling may be used from an existing or future CHP unit
[17] Equally divided between PHR using heat exchangers and CCHPs (Closed Cycle Heat Pumps).
[18] Equally divided between gas-gas heat recovery and CCHPs on malt kilns.
[19] Exhaust gases cannot be used directly in malt kilns, but could via a heat exchanger.
[20] Microwaves might be an opportunity.
[21] The number of process opportunities in the food sector for using waste heat to generate power is believed negligible.
[22] Heat/cold storage is perhaps of greatest interest when off-peak electricity can be used to generate ice for storage for daytime use. Where absorption chillers could use waste heat, there may also be 'coolth' storage opportunities.
[23] Heat transfer enhancement is an integral part of many hat recovery installations. In the food sector enhancement features can introduce cleaning problems, so their use here is currently believed to be limited.
[24] Process integration can be used in several sectors – it has been applied in the edible oils sub-sector, for example. Quantification of the benefits necessitates a full site analysis on a case-by-case basis, and is beyond the scope of this study.
[25] Process intensification, as with process integration, requires a specific knowledge of each process/unit operation in order to quantify benefits, and many PI technologies are as yet difficult to quantify in terms of energy savings. It is believed that many opportunities exist in this sector and in chemicals.

TABLE A3.5. PHR POTENTIAL IN THE TEXTILES SECTOR

	Gas-gas HX[26],[27]	Gas-liquid HX[28]	Liquid-liquid HX	Closed cycle heat pump	Open cycle heat pump	Space heating	Compressed air	Heat/cold storage	Heat transfer enhancement	Process integration	New process technology/ intensification
1730	6	2	8	(8)[29]			1	x[30]	x[31]	x[32]	
1740						x[33]					
1754	0.5		0.5								
1910	1.0			3[34]							
TOTALS	**7.5**	**2**	**8.5**	**2**			**1**				
SUBSECTOR TOTAL											**21.0 ktoe/a**

[26] HX = heat exchanger
[27] All figures are in ktoe/a
[28] Opportunities for spray recuperators – but corrosion needs to be taken into account.
[29] These would be used instead of liquid-liquid PHR, so are not included in the totals.
[30] Storage of heat between batches in, for example, dyeing may be feasible.
[31] Where shell and tube heat exchangers are used for effluent heat recovery, tube inserts may be beneficial.
[32] PI may be applied in larger plant, but no examples are known to date.
[33] Modest amount of PHR for space heating may be feasible. Too small to sensibly quantify.
[34] Some of this would replace gas-gas heat recovery equipment n the longer term, so only 2 ktoe counted.

TABLE A3.6. PHR POTENTIAL IN PULP, PAPER & PAPER PRODUCTS

	Gas-gas HX[35],[36]	Gas-liquid HX	Liquid-liquid HX	Closed cycle heat pump	Open cycle heat pump	Combined heat & power[37]	Power cycles	Heat/cold storage	Heat transfer enhancement	Process integration	New process technology/ intensification
2110						x				x	x[38]
2121	4		2								x
2122	6		(2)[39]	10	5[40]						
2123											
2124	2[41]										
2125											
TOTALS	12		2	10	5						
SUBSECTOR TOTAL											**29 ktoe/a**

[35] HX = heat exchanger
[36] All figures are in ktoe/a
[37] Included as CHP opportunities may still exist in paper plant
[38] See text for brief discussion
[39] Figure in brackets indicates that it would be an integral part of an overall dryer PHR scheme.
[40] A heat pump might be a future option for improved PHR.
[41] Possibly integrated with a solvent recovery process.

TABLE A3.7. PHR POTENTIAL IN COKE, REFINED PETROLEUM ETC.

	Gas-gas HX[42],[43]	Gas-liquid HX	Liquid-liquid HX	Closed cycle heat pump	Open cycle heat pump	Combined heat & power[44]	Power cycles	Heat/cold storage	Heat transfer enhancement	Process integration	New process technology/ intensification
2310										x	x
2320	45	30	15		200[45]		x[46]			x	x
TOTALS	**45**	**30**	**15**		**200**						
SUBSECTOR TOTAL											**290 ktoe/a**

[42] HX = heat exchanger
[43] All figures are in ktoe/a
[44] Included for absorption cooling, where relevant
[45] Distillation is a major opportunity for heat pumps, but barriers exist, such as high capital cost.
[46] It is worth flagging up at this point (the first opportunity)

TABLE A3.8. PHR POTENTIAL IN CHEMICALS & CHEMICAL PRODUCTS

	Gas-gas HX[47], [48]	Gas-liquid HX	Liquid-liquid HX	Closed cycle heat pump	Open cycle heat pump	Combined heat & power[49]	Power cycles	Heat/cold storage & refrigeration	Heat transfer enhancement	Process integration	New process technology/ intensification
2411		5[50]						10		x[51]	
2412	4	3	3							x	x[52]
2413	3.6		1		0.35[53]						
2414	2.75[54]	0.8[55]				3[56]		0.4			x
2415	4	7	8				(1)[57]			x	x[58]
2416	14	5	2		7.5[59]					x	x
2417	1			0.5	0.8						
2420	2	1	1							x	x
2430	2		2							x	x
2441	2	2	1	0.5	2						
2442	7.5	3	2.3	2	5			1.2[60]			
2451	2		1.2	1	1.2						
2452[61]											

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47 HX = heat exchanger
48 All figures are in ktoe/a
49 Included for absorption cooling, where relevant
50 Heat recovery from gas compressor after-coolers for boiler water heating
51 Process integration can be widely applied, as it is, across the chemicals sector.
52 Process intensification and new process technology could make major inroads into energy se in the whole sector.
53 On evaporation plant
54 Several of the gas-gas opportunities are at high temperatures.
55 Includes compressed air PHR
56 Distillation heat pumps are long-term aim.
57 Sulphuric acid plant
58 Polymerisation is a strong candidate for process intensification.
59 Includes both evaporators and distillation plant.
60 Refrigeration plant heat recovery
61 No data, but small subsector

	Gas-gas HX[62], [63]	Gas-liquid HX	Liquid-liquid HX	Closed cycle heat pump	Open cycle heat pump	Combined heat & power[64]	Power cycles	Heat/cold storage & refrigeration	Heat transfer enhancement	Process integration	New process technology/ intensification
2462[65]											
2463[66]											
2464[67]											x
2466	3			1	2						
2470	1										
TOTALS	**48.85**	**26.8**	**21.5**	**5.0**	**18.85**	**3**	**(1)**	**11.6**			
SUBSECTOR TOTAL											**136.6 ktoe/a**

[62] HX = heat exchanger
[63] All figures are in ktoe/a
[64] Included for absorption cooling, where relevant
[65] Very low energy use
[66] Very low energy use
[67] No data, but modest energy use – opportunity for process intensification (now used).

TABLE A3.9. PHR POTENTIAL IN OTHER NON-METALLIC MINERAL PRODUCTS

	Gas-gas HX[68], [69]	Gas-liquid HX	Liquid-liquid HX	Closed cycle heat pump	Open cycle heat pump	Combined heat & power[70]	Power cycles	Heat/cold storage & refrigeration	Direct heat reuse	Process integration	New process technology/ intensification
2611							x[71]				
2613	26[72]							2	2[73]		
2614											
2615											
2621	0.5			0.5							
2622	0.5										
2624											
2626	0.5										
2630	2			1							
2640	22 + 7[74]			8							
2651[75]										x	x
2661											
2662	2			2							
2663											
2666											
TOTALS	**60.5**			**11.5**				**2**	**2**		
SUBSECTOR TOTAL											**76 ktoe/a**

[68] HX = heat exchanger
[69] All figures are in ktoe/a
[70] Included for absorption cooling, where relevant
[71] The potential for heat recovery to produce power in glass making processes could be of long-term interest, but corrosion might inhibit take-up
[72] This figure is the potential for gas-gas high temperature heat recovery across the sector – regenerative and recuperative burners and recuperators.
[73] Cullet preheating using waste gases.
[74] Includes high temperature (22 ktoe) and low temperature heat – the latter being split between gas-gas and closed cycle heat pumps.
[75] Cement plants are fully integrated and special cases. Heat recovery is not considered for these. There may be options for revisiting the overall manufacturing process, however, using process integration/process intensification, as was done at Blue Circle some 20 years ago by the author (D.A. Reay).

ANNEX A3.1.1 DEFINITION OF UK SECTOR CODES

SIC(92) codes	Description
	FOOD PRODUCTS & BEVERAGES
1511	Production and preserving of meat
1512	Production and preserving of poultrymeat
1513	Production of meat and poultrymeat products
1520	Processing and preserving of fish and fish products
1531	Processing and preserving of potatoes
1532	Manufacture of fruit and vegetable juice
1533	Processing and preserving of fruit and vegetables n.e.c.
1541	Manufacture of crude oils and fats
1542	Manufacture of refined oils and fats
1551	Operation of dairies and cheese making
1552	Manufacture of ice cream
1561	Manufacture of grain mill products
1571	Manufacture of prepared feeds for farm animals
1572	Manufacture of prepared pet foods
1581	Manufacture of bread; manufacture of fresh pastry goods and cakes
1582	Manufacture of rusks and biscuits; manufacture of preserved pastry goods and cakes
1583	Manufacture of sugar
1584	Manufacture of cocoa; chocolate and sugar confectionery
1585	Manufacture of macaroni, noodles, couscous and similar farinaceous products
1586	Processing of tea and coffee
1587	Manufacture of condiments and seasonings
1588	Manufacture of homogenized food preparations and dietetic food
1589	Manufacture of other food products n.e.c.
1591	Manufacture of distilled potable alcoholic beverages
1592	Production of ethyl alcohol from fermented materials
1594	Manufacture of cider and other fruit wines
1596	Manufacture of beer
1597	Manufacture of malt
1598	Production of mineral waters and soft drinks
	TEXTILES
1730	Finishing of textiles
1740	Manufacture of made-up textile articles, except apparel
1754	Manufacture of other textiles n.e.c.
1910	Tanning and dressing of leather
	PULP, PAPER & PAPER PRODUCTS
2110	Manufacture of pulp, paper and paperboard
2121	Manufacture of corrugated paper and paperboard and of containers of paper and paperboard
2122	Manufacture of household and sanitary goods and of toilet requisites

2123	Manufacture of paper stationery
2124	Manufacture of wallpaper
2125	Manufacture of other articles of paper and paperboard n.e.c.
	COKE, REFINED PETROLEUM PROD. & NUCLEAR FUEL
2310	Manufacture of coke oven products
2320	Manufacture of refined petroleum products
	CHEMICALS & CHEMICAL PRODUCTS
2411	Manufacture of industrial gases
2412	Manufacture of dyes and pigments
2413	Manufacture of other inorganic basic chemicals
2414	Manufacture of other organic basic chemicals
2415	Manufacture of fertilizers and nitrogen compounds
2416	Manufacture of plastics in primary forms
2417	Manufacture of synthetic rubber in primary forms
2420	Manufacture of pesticides and other agro-chemical products
2430	Manufacture of paints, varnishes and similar coatings, printing ink and mastics
2441	Manufacture of basic pharmaceutical products
2442	Manufacture of pharmaceutical preparations
2451	Manufacture of soap and detergents, cleaning and polishing preparations
2452	Manufacture of perfumes and toilet preparations
2462	Manufacture of glues and gelatines
2463	Manufacture of essential oils
2464	Manufacture of photographic chemical material
2466	Manufacture of other chemical products n.e.c.
2470	Manufacture of man-made fibres
	OTHER NON METALLIC MINERAL PRODUCTS
2611	Manufacture of flat glass
2612	Shaping and processing of flat glass
2613	Manufacture of hollow glass
2614	Manufacture of glass fibres
2615	Manufacture and processing of other glass, including technical glassware
2621	Manufacture of ceramic household and ornamental articles
2622	Manufacture of ceramic sanitary fixtures
2624	Manufacture of other technical ceramic products
2626	Manufacture of refractory ceramic products
2630	Manufacture of ceramic tiles and flags
2640	Manufacture of bricks, tiles and construction products, in baked clay
2651	Manufacture of cement
2661	Manufacture of concrete products for construction purposes
2662	Manufacture of plaster products for construction purposes
2663	Manufacture of ready-mixed concrete
2666	Manufacture of other articles of concrete, plaster and cement

	BASIC METALS
2710	Manufacture of basic iron and steel and of ferro-alloys (ECSC
2721	Manufacture of cast iron tubes
2722	Manufacture of steel tubes
2734	Wire drawing
2735	Other first processing of iron and steel n.e.c.; production of non-ECSC ferro-alloys
2742	Aluminium production
2743	Lead, zinc and tin production
2744	Copper production
2745	Other non-ferrous metal production
2751	Casting of iron
2752	Casting of steel
2753	Casting of light metals
2754	Casting of other non-ferrous metals

Source: Department of Trade and Industry

ANNEX A3.1.2

PROCESS HEAT RECOVERY EQUIPMENT CONSIDERED

A3.1.2.1. Introduction

There are many types of equipment that can be used for process waste heat recovery. Most may be described as heat exchangers, but there are others, such a heat pumps and systems that can use waste heat for the production of refrigeration (e.g. the absorption refrigerator) or electricity (e.g. organic Rankine cycle machines) that may involve several heat exchangers and/or other unit operations, such as compressors or turbines.

The selection of the most appropriate unit for recovering the heat depends upon many factors – see for example Good Practice Guide 141 on Waste Heat Recovery in the Process Industries – but those that primarily determine the selection are the nature of the source of waste heat and the best opportunity present for using the heat when it is recovered. This is reflected in the basic categories of heat exchangers, as listed below:

Gas-gas heat exchangers
Gas-liquid heat exchangers
Liquid-liquid heat exchangers[76]

Within each of these categories there are several types of heat exchanger, briefly described below.

Another category of equipment that is effective in process heat recovery encompasses the recuperative and regenerative burner. These burners are marketed with integral heat exchangers and are marketed as combustion units rather than heat exchangers. However their main attributes are due to the presence of the heat exchangers, which are gas-gas types included within our basic categories listed above.

When we look at heat pumps (of which there are several variants), the use of waste heat for providing refrigeration, or for the generation of electrical energy, the complexity of the equipment tends to be greater, and the word 'system' is used more accurately to describe the equipment.

Another category of PHR is heat/cold storage. This may be of interest when the demand does not coincide with the supply, or there is excess heat/cold available at any one time.

The direct reuse of waste heat without the need for a heat exchanger is an ideal situation, but one that does not occur often. It may be feasible, for example, where a gas turbine exhaust could be used for a drying duty, or where the exhaust from a heat treatment oven may be recycled for preheating the oven charge.

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[76] Wherever a category involves a liquid stream, there may be two-phase heat transfer (boiling or condensing). For example a waste heat boiler is one sub-category within 'gas-liquid heat exchangers' in which the liquid steam undergoes a change of phase to a vapour.

Not strictly a heat exchanger, but a piece of equipment that can improve the performance of heat exchangers, is a tube insert. This falls within the area of 'heat transfer enhancement' and is included below.

There are methodologies that can be used to assist the optimisation of PHR. The most important of these is Process Integration. Pioneered by ICI and the University of Manchester, process integration can be used to minimise the heating and cooling utilities in a process by proper heat exchanger selection and placement. Used extensively in the chemical and related sectors, it has been also applied in food processing plant.

Finally, a radical redesign of the process may minimise or completely remove the need for heat recovery. Process intensification is one technique that can help towards this aim.

A3.1.2.2 Equipment Types

Heat Exchangers

Gas-gas Heat Exchangers

The gas-gas heat exchangers encompass a wide range of equipment types, spanning most temperature ranges met within the process industries. The range of efficiencies is large, the regenerator typically being regarded as the most effective – up to around 90%, but it is important to note that a heat exchanger having an efficiency of only 55% may be the most appropriate for a specific application, because the heat transfer surface my be designed to handle fouling in the exhaust gas stream.[77]

Gas-gas heat exchangers (the gas may of course be air or air with other gases in it) are used in four main areas:

- Heat recovery for preheating the gas stream within the originating process – which may of course be a prime mover
- Heat recovery to preheat another process gas stream
- Heat recovery to preheat air for space heating
- Heat recovery from ventilation extract air (e.g. in a factory) to preheat incoming air[78]

In terms of economic benefits the first of these areas is generally the most attractive.

Some of the gas-gas heat exchangers may be used across a range of the 'low temperature' duties, but others are unique to high temperature tasks, such as in the metals and ceramics/glass sectors.

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[77] Such factors are not unique to gas-gas heat exchangers – a liquid effluent in the textile industry from which heat is being recovered may be highly fouled and therefore need a heat exchanger with large flow passages – implying lower efficiency.

[78] In a cooled space, such a heat exchanger may be employed to recover 'coolth' *to precool* the incoming make-up air – thus saving on air-conditioning load. The market may be in the pharmaceutical industry, but most extensively in the buildings sector.

Rotating regenerator

The rotating regenerator is potentially the most efficient of the gas-gas heat exchangers. Heat is transferred between adjacent gas streams by alternate heating and cooling of the regenerator matrix (which may be a metal 'pan-scrubber' type structure or a honeycomb matrix) as the matrix passes between them. Rotational speed of the 'heat wheel' is low – typically 10 rpm. High temperature ceramic regenerators have been used in gas turbines, and studied for the steel sector, but life/reliability were poor.

Plate heat exchanger

Not to be confused with the liquid-liquid variant of the same name, the gas-gas plate heat exchanger is available for a wide range of process conditions and can be engineered to handle dirty gas steams. Classed as a recuperative heat exchanger (heat transfer taking place between streams, across a wall separating the two), it can be manufactured using plates in aluminium, polymer, steel or other metals. Where corrosion is a severe problem, glass has been used.

The 'run-around coil' or liquid-coupled heat exchanger

The run-around coil is a versatile unit comprising two or more gas-liquid heat exchangers that are normally finned coils, connected by a pumped liquid loop. The fluid used may be water or a thermal oil. The principal advantage of this type of heat recovery unit is that it can readily accommodate heat sources and sinks (the users) that are some distance apart. It also offers a safety feature where cross-contamination may occur – there are two walls between the streams, rather than one, as in most recuperators.[79]

The heat pipe or thermosyphon heat exchanger

Heat pipes/thermosyphons use tubes containing a liquid that undergoes an evaporation/condensation process to transfer heat from one end – immersed in the waste heat stream – to the other. Fins are put on the outside of the pipes to aid heat transfer into and out of them. They tend to be more expensive than other types, largely due to small production runs. They do have advantages in the separation of streams that might otherwise not be considered as heat sources because of cross-contamination fears.

[79] Interestingly, one of the first EEO Demonstration Projects on PHR, at Berk Spencer Acids, used the run-around coil principle but the two heat exchangers were *liquid-liquid* plate heat exchangers (CHEs). These allowed separation of a waste hot acid stream from a boiler feedwater supply (the 'sink') in a manner that put two walls between the acid and the feedwater – thus satisfying the concerns of the boiler house supervisor! As the liquid-liquid plate heat exchanger can be 95% efficient, the cost in terms of a higher temperature drop between the source and sink than available using only one heat exchanger was minor. This type of use of CHEs could be highly relevant to heat pumps in the process sectors – these being expensive pieces of equipment where pollution of the working fluid would be disastrous.

Tubular recuperators

There are two types of tubular recuperator – the convection recuperator and the radiation recuperator. The former tends to be used at the lower end of the process temperature ranges, where radiation does not dominate heat transfer.

Convection recuperators comprise a bundle of tubes normal to the flow of one gas stream, the other passing through the tubes. There can be a mismatch between the outside surface (which will in most cases be finned, unless the tubes are made of polymer or glass) and the inside, which may be plain. Thus efficiencies may be rather low.

This need not deter from their attractiveness in terms of easy cleaning and the ready availability of tubes in corrosion-resistant materials such as the aforementioned glass and polymers. At higher temperatures ceramics may be used.

Radiation recuperators may comprise concentric cylinders (see recuperative burner later), or resemble convection recuperators – tubes mounted between two headers.

Static regenerators

Static regenerators comprise, in the majority of process applications, one or two beds of heat retention material, such a bricks (in the glass industry) or ceramic balls (in the regenerative burner – see later) which remove and store heat from the hot gas stream passing through it, later giving up that heat to the stream to be preheated. They tend to be associated with high temperature processes, often where severe acid fouling may occur.

Recuperative and regenerative burners

The recuperative (or self-recuperative) burner uses a built-in tubular recuperator to recover exhaust gas heat to preheat combustion air. Although of interest for smaller furnaces (5 tonne/h or less), it has been superseded by the regenerative ceramic burner in many metal processes, which operates to preheat incoming combustion air..

Gas-liquid Heat Exchangers

Unlike many of the heat exchangers described earlier, gas-liquid heat exchangers are more likely to be integral parts of another unit operation, such as a boiler. Some cannot be strictly regarded as PHR units as they are standard items of plant in boilers, such as economisers and superheaters. As such, they may appear as a component of a condensing boiler, included in the ETL.

Nevertheless, there remains considerable scope for gas-liquid heat recovery in the process industries, both for using hot exhaust gases for heating liquids or raising process steam. Additionally, there is growing interest and application in recovering heat from prime movers or processes for generating a vapour that can be expanded through a turbine for power generation.

The gas-liquid heat exchanger is also beneficial in applications where one wishes to condense out moisture from a gas steam – either to boost the heat recovery capability by recovering latent heat – as in a condensing economiser, or to remove impurities.

Economisers

The economiser is a gas-liquid tubular heat exchanger, generally associated with boiler plant, where it is used to recover heat from exhaust gases to preheat boiler feed-water. The gas flows over the tubes, which are normally finned, and the water through the tubes, in one or more passes. Although common on boilers, there is scope for using these heat exchangers on process plant for, as an example, heating heat transfer fluids or domestic hot water.

Condensing economiser

Most commonly associated with modern domestic boilers, the condensing economiser (or condensing heat exchanger) is used where it is desirable to cool exhaust gases below their dew point – promoting condensation and recovering a proportion of the latent heat. This may of course result in a dilute acid solution, so materials selection is important. It can be a separate heat exchanger, or form the back few tube rows of a conventional economiser – a 'condensing section'.

By definition, these heat exchangers operate at the low temperature end of the gas stream, so the recovered heat may not be useful in processes. However, the unit can be a beneficial part of an exhaust gas clean up system.

Waste heat boiler

The waste heat boiler in one of its forms resembles the economiser, except that evaporation takes place within the tubes, allowing steam, or another vapour, to be generated. A second variant – the 'fire-tube waste heat boiler' – takes the gases through the tubes, and the boiling takes place outside the tubes.

Unlike the economiser, the waste heat boiler may need water treatment facilities to minimise scaling and corrosion. Downstream components such as steam traps – as with conventional boilers – may be 'extras'. In instances where the source of waste heat may vary in intensity over time, supplementary firing may be incorporated in order to sustain steam output. In some cases it may then become purely a boiler.

The association with power generation, where steam or an organic working fluid vapour may be produced by the waste heat boiler, is one which could grow, as the demand for electricity rises.

Spray condensers

The spray condenser, or spray recuperator, is a direct contact heat exchanger – at least in the first stage of heat recovery no wall separates the heat source from the heat sink. Typically, water is sprayed into a humid air-stream, the water is heated and supplemented by the warm moisture as it condenses. The heated water can be used directly in another process, or cooled by another heat exchanger before being returned to the spray inlet. The system has seen valuable application in the pulp and paper sector.

Liquid-liquid Heat Exchangers

It is perhaps slightly misleading to use the term liquid-liquid because many of the heat exchangers within this category function as evaporators and/or condensers. Within the compact heat exchanger field, the plate heat exchanger and derivatives also are classed as liquid-liquid units, and are in some cases already eligible for inclusion in the ETL (Energy Technology List for Enhanced Capital Allowances – a UK support scheme for energy-saving equipment, including some heat pumps and compact heat exchangers).

Liquid-liquid heat exchangers are prolific within the process industries, particular the 'low temperature' sectors. Liquid effluent streams are common heat sources, and these can be fouled – so care needs to be taken in selecting the appropriate type of exchanger[80]. The two most common types are the shell-and-tube unit and the plate that exchanger mentioned above.

An interesting feature of many liquid-liquid heat exchangers is that they are frequently perceived as necessary in order to operate processes with acceptable levels of thermal efficiency. The plate heat exchangers used on pasteurisers are a good example of this, and one may categorise such applications as 'dead weight' using the terminology adopted in analyses by the Carbon Trust.

Shell-and-tube heat exchanger (S&T HX)

The S&T HX, described by some as the 'workhorse' of the process industries dominates the heat exchanger market. It is also used in process chiller and refrigeration plant, as an evaporator or condenser. Because it can be large, it is seen in some process sectors as a target for the more compact and more efficient compact heat exchangers, such as the plate unit. However, its versatility and ease of cleaning are strong marketing points.

There are many attempts to improve these heat exchangers using spiral/twisted tubes and modifications to the baffles – the aim being to reduce size and/or improve efficiency. This can complicate attempts to categorise the units. (See also Section 4 on heat transfer enhancement).

The use of a fluidised bed on the shell or the tube-side of variants of shell-and-tube heat exchangers can allow them to be used in duties where fouling would normally preclude heat recovery. The paper sector has seen such an example.

[80] It is important to recognise that a heat exchanger specifically engineered to handle a fouled stream (liquid or gas) may have a lower efficiency than one designed for clean streams. Thus the setting of high effectiveness values for some types of exchanger as a means of categorising them for inclusion in an 'approved list' may rule out a number of beneficial PHR opportunities. A low efficiency does not necessarily imply a poor PHR unit, but perhaps one 'designed for purpose'.

Plate heat exchangers

The plate heat exchanger (PHE) is the most common of the 'compact' heat exchangers that have been included in the ECA Scheme. With efficiencies of up to 95% their benefits to processes where they may contribute to energy efficiency are potentially substantial – and this is particularly true in the food and drink sector, where they are well-established.

Generally available in a gasketed form, which can be opened for cleaning, the PHE may also be welded or brazed. This allows operation at higher process temperatures and pressures.

Plate and shell heat exchangers

The plate and shell heat exchanger combines the merits of shell-and-tube and plate units, as the name implies. The compactness and high efficiency of a plate unit, especially configured to fit inside a shell, allows advantage to be taken of the higher operating pressures and temperatures afforded by the shell.

Compact Heat Exchangers

Compact heat exchangers (CHEs) are already included in the ETL, but there are difficulties in meeting the criteria in terms of cataloguing, particularly for the more advanced types such as metal foam heat exchangers and the units manufactured by Chart Energy & Chemicals, Inc. and Heatric Ltd.

Heat Pumps[81]

Closed Cycle Heat Pumps

There has been a reluctance in the UK, not seen in other countries, for process industries to invest in heat pump technology, particularly the closed cycle type. These heat pumps, operating on the vapour compression cycle or, more rarely, the absorption cycle, should be more readily retrofitted to plant as an add-on component than open cycle heat pumps, that need close integration with the process, ideally during the design and specification stage of the latter, (although there are some exceptions). The heat pump is one of the few heat recovery units that can produce effective cooling, as well as upgrading waste heat to a higher temperature.

Vapour compression cycle heat pumps

Using either a warm gas stream (which may be humid) or liquid stream, the heat pump can recover heat from the source and deliver it to the heat sink (the user) at temperatures typically up to 40°C higher than the source. With such a modest

[81] The term heat pump is used to include machines used for heating and/or cooling, using as the heat source a waste heat stream of any type.

‘temperature lift’ the heat pump would use typically 20-30% of the energy recovered to deliver the higher-grade energy.[82]

A highly effective role of heat pumps that has been exploited by one or two equipment suppliers in the UK is dehumidification. In this case, the exhaust stream from, for example, a batch dryer may be dehumidified and reheated by passing it in turn over the heat pump evaporator (which cools it) and condenser (which reheats it).

Absorption cycle heat pumps

The absorption unit operates on a different cycle to that of the vapour compression system. The principal difference in terms of energy use is that it is heat-driven and the heat input may be a waste heat steam or a burner. Currently the units are more complex than their vapour compression counterparts, with cost implications.

The demand for absorption units is currently driven by two applications, both involving heat recovery to provide cooling/refrigeration. The first is for improving the efficiency of gas turbine cycles (using exhaust heat) by cooling the inlet air – a popular use for absorption cycle units in the Middle East. The second is to use the exhaust heat (from a prime mover or a process) for refrigeration/cooling.

Open Cycle Heat Pumps

Two forms of open cycle units are of interest. The vapour recompression system is the most common, but recently the Brayton-cycle unit has been proposed (as a gas compressor) in adsorbent regeneration.

Mechanical vapour recompression

The mechanical vapour recompression (MVR) unit, is becoming established as a heat recovery unit on evaporators in the food and drink sector, and has the potential to reduce energy use in effluent concentration and in distillation (including chemical distillation). Unlike the closed cycle heat pump, the MVR unit upgrades the pressure and temperature of the process stream (normally in this case steam or a process vapour such as an organic chemical). The potential energy savings are significant, but capital investment in the larger units can be high.

Interest in the marketplace is growing, and a number of units are currently being installed in the UK.

Brayton-cycle heat pump

The Brayton cycle uses air or a gas as the working fluid (it is used in some train air conditioning systems and aircraft). The heat pump based upon this cycle is proposed

[82] The energy input needed to deliver this typically 40 deg.C rise in temperature is via a compressor, that is driven by an electric motor (most common) or a gas engine. Thus in terms of carbon emissions, the nature of the energy input and, if electricity, the efficiency of primary energy generation and the fuel used (e.g. nuclear/oil/gas/coal) become important. A coal-fired power station with carbon capture, a combined cycle plant, or a nuclear station will be the ‘best’ energy sources for an electric drive heat pump. A combined heat and power plant on site is an attractive alternative.

for adsorbent regeneration in adsorption systems that are used for solvent recovery in the process industries. Energy savings of up to 25% compared to conventional adsorbent regeneration.

Power Cycles

It is feasible to convert waste heat in industrial processes into electrical power by employing a Rankine cycle machine. Turbine expanders, with steam or an organic fluid vapour raised in a boiler fed by the waste heat, can effectively provide electricity with conversion efficiencies between 30 and 5%, depending upon the temperature of the waste heat source.

Although not directly associated with waste heat, energy recovery can be implemented using a turbine through which a high-pressure gas stream is expanded. This can replace a valve, for example. This is not considered further in this study.

Rankine Cycle Machines

The Rankine cycle PHR unit is manufactured by a number of companies, and has been used for process heat recovery as well as in renewable energy power generation (using solar concentrators). The selection of the fluid used in the cycle depends upon the temperature of the waste heat available – water/steam being appropriate for high temperature processes while an organic fluid (less efficient) as the available temperature drops. (Hence the acronym ORC – Organic Rankine Cycle) used at the low temperature end of the scale).

Low temperature heat sources imply low cycle efficiencies, and the cost-effectiveness based upon normal industrial criteria may be difficult to justify, unless a steam turbine can be used. In this case the process options are few.

Stirling engines

The Stirling engine can operate to produce power using waste heat. Proposed for domestic micro-generation systems, Stirling engines could conceivably be used to aid PHR in niche applications.

Heat/Cold Storage (or Thermal Energy Storage – TES)

The storage of heat has been practiced for many decades, steam accumulators being a common feature of process plant in the last Century. The static regenerator is a short-term form of sensible heat storage, but the principal difficulty with longer-term storage of sensible heat is the volume of store needed – orders of magnitude greater than the domestic storage 'radiator' containing bricks or oil.

In order to overcome this size constraint, latent heat storage media have become of interest. Phase change media (PCMs) which change from solid to liquid and vice-versa as they absorb and reject heat (or 'coolth') are being used in buildings, in chilled ceilings to reduce air conditioning loads, for example.

While the storage of heat or 'coolth' in most processes is not high on the list of priorities in most companies, there are precedents for cold storage in industry. For example Rohm & Haas has minimised electricity costs by storing ice generated using off-peak electricity, lowering chiller operating costs. TES has been used in the Middle East for gas turbine inlet air cooling. At Nottingham University a Carbon Trust project is recovering 'coolth' overnight for space cooling during the day – a system that might have potential in some process cooling situations. At a Dorset dairy TES was use to aid yoghurt cooling.

Direct Reuse of Waste Heat

In a number of sectors, where cross-contamination is not a problem in terms of product quality or purity, the direct use of waste heat in, for example, a downstream process may be feasible. The manufacture of titanium dioxide has employed exhaust gases from a gas turbine for a drying process, and there are other examples. At the 'heavy' end of the chemicals sector, the use of double-effect distillation (and in other sectors multiple-effect evaporation) are forms of direct reuse of waste heat. In these cases, the heat may be used in a modified downstream processes rather than being re-introduced at the start of the original process. (See GPG 269 for examples)

Care has to be exercised – for example the exhaust gases from gas-fired engines were used in malt kilns, but nitrozamine concerns are believed to have led to this practice being discontinued.

Any sensible assessment of PHR will first of all examine direct reuse, but opportunities can be overlooked.

Heat Transfer Enhancement

Heat transfer enhancement takes many forms – fins on the tubes of heat exchangers and microwaves to heat a sheet of plywood to speed up drying are examples of passive and active enhancement, respectively. Heat transfer enhancement has been targeted by the Government in the past. This was based upon similar arguments to those used to promote compact heat exchanger and process intensification strategies.

In the context of the current study, the most appropriate enhancement method is believed to be the tube insert. This can take several forms, but is in all cases an object inserted into a heat exchanger tube to enhance fluid mixing and to improve single- or two-phase heat transfer coefficients. Commonly used as a retrofit method for boosting the capability of shell-and-tube heat exchangers, tube inserts are of relatively low cost and can be used to 'upgrade' an exchanger, obviating replacement.

Process Integration

Process integration (PI) can be described as (see GPG 242): '*the selection of processing steps and their interconnection to provide an optimum manufacturing solution*'. This definition gives little clue to the main feature of much PI – the optimisation of the performance of groups of heat exchangers and/or other unit operations so that the heat recovery between the hot and cold streams is maximised.

PI has been the subject of a number of R&D projects supported by the Energy Efficiency Office about 20 years ago, but the initiative since then has been taken by individual companies and consultancies, mainly serving the chemicals sector, and the consortium based at Manchester University.

Process integration is an 'enabling technology' that can assist companies maximise the efficiency (and minimise emissions) in complex processes. As such it has an important role to play in carbon reduction, but needs support in a way not currently possible under ECA criteria.

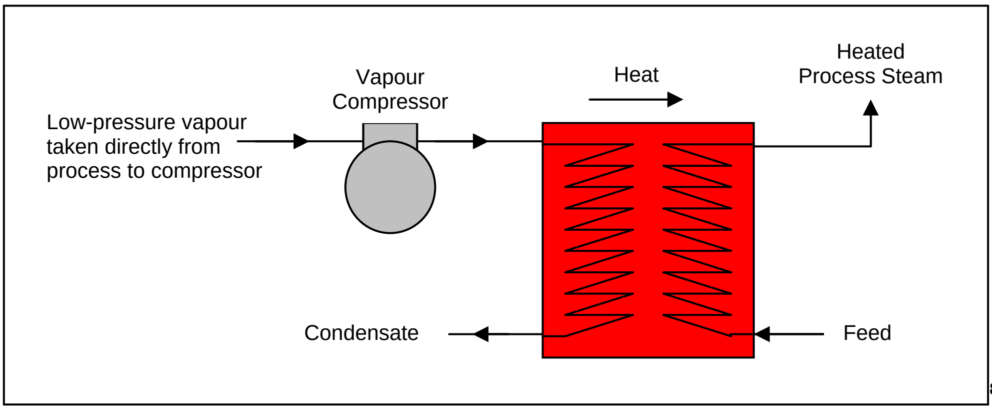
Process Intensification

Process intensification may be defined as 'any engineering development that leads to a substantially smaller, cleaner, safer and more energy-efficient technology'.

The potential of process intensification to influence process heat recovery is typified by the comments of Professor Ramshaw at the HEXAG meeting in Newcastle in November 2006, where he suggested that catalytic plate reactors could replace much larger less efficient process reactors that use excessive quantities of heat. In some of these alternative processes the amount of recoverable waste heat would be much reduced, or even eliminated.

Thus process intensification is seen as an option which could eliminate or much reduce the need for PHR, particularly in some processes in chemicals, food & drink and possibly textiles.

A3.2 Guidance Notes for Compact Heat Exchanger Selection



[83]

Contents

[83] The illustration shows an open-cycle mechanical vapour recompression unit, that may use a compact plate heat exchanger, employed for effluent concentration in the process industries.

A3.2.1 Introduction to the Annex and Background to the Document

The objective of this Annex was to present a compilation of possible options for compact heat exchangers, used as evaporators, condensers and in other roles in heat pumping equipment. The aim is to minimise the direct and indirect effect on the local and global environment due to operation of, and ultimate disposal of, the equipment. Further data on the overall Annex activities are given later in the introductory Section, but this publication relates specifically to design data for compact heat exchangers used in heat pumping systems at the process/large commercial level.

Introduction and Background

During the last two decades there have been substantial developments affecting equipment used within the refrigeration and heat pump industries, brought about largely due to environmental concerns, principally related to vapour compression cycle systems: Firstly, the realisation of the detrimental effect of chlorinated hydrocarbons on the ozone layer led to a quick phase out of these fluids, a process which is now complete in many parts of the world. Secondly, as more focus and concern has been directed towards the issue of global warming, the global warming potential (GWP) of many of the commonly used working fluids, in particular hydrofluorocarbons (HFCs), has been topping the agenda. Thirdly, and discussed later, the materials content of heat exchangers in fan-coil units (and in other heating/cooling units) are regarded rather negatively from the points of view of environmental impact and life cycle assessment[84].

Due to these perceived negative effects on the global environment, parts of the industry as well as parts of the research community and some governmental institutions, in particular in Europe, have suggested the use of natural fluids, meaning primarily ammonia, hydrocarbons and carbon dioxide, as working fluids. All these fluids are suitable from a technical point of view, although each is not necessarily ideal for all applications. However, they all have drawbacks, relating either to flammability or toxicity, or operation at high pressure. If used in public areas, the quantities of working fluid in the systems should therefore be kept as low as possible.

Having the global and local environment in mind, it is clear that future refrigeration and heat pump equipment should have as small as possible internal volume. This conclusion is equally true independently of the refrigerant chosen in the system: With HFCs and other high GWP-fluids a low charge will reduce the total leakage, thus reducing the influence on the global warming. With ammonia, hydrocarbons and carbon dioxide, a low charge will reduce the risks of accidents in case of leaks.

An additional parameter which must be taken into account is the indirect influence on the global warming from any type of refrigeration or heat pump unit. As long as we manage to keep our systems tight these secondary effects caused by the CO_2 emissions connected to electricity production are far larger than the direct effects

[84] See research of Matajaz Prek at the University of Ljubljana, for example, reproduced in *Energy and Buildings*, 36 (2004) 1021-1027.

caused by the leakage of refrigerant. A reduction of charge must thus not be accomplished at the cost of reduced energy efficiency of the system.
Looking at the distribution of the charge of working fluid within a heat pumping system it is quite clear that the dominating part is located either in the evaporator or in the condenser for vapour compression cycle and mechanical vapour recompression cycle units and, where absorption cycle systems may be considered, additionally in the generator and absorber. Reducing the charge is therefore mainly a matter of redesigning the heat exchangers. (A possible exception to this rule is the case of multisplit, direct expansion systems where a large amount of liquid working fluid is circulated through long tubing systems. For this type of system, a first step, it is suggested, should be to redesign to an indirect system, using a secondary fluid to distribute the heating/cooling capacity). To reduce the volume on the working fluid side without decreasing the energy efficiency of the system may seem a difficult task. The heat transfer areas on the two sides of the heat exchanger should preferably not be decreased, as this may increase the temperature difference between the fluids and thus reduce the efficiency.

The obvious solution to this equation is to decrease the channel area cross section, i.e. the hydraulic diameter. Fortunately, a decreased diameter may offer possibilities of increasing the heat transfer coefficients. In single phase flow it can easily be shown that in most instances, for a fixed temperature difference, a fixed pressure drop and given heat- and mass flows, the heat transfer coefficient will increase and the necessary heat transfer surface area will decrease with decreasing tube diameter. For two-phase flow, condensation and evaporation, the analysis is not as simple partly because there are no reliable correlations available for predicting heat transfer and pressure drop for small-diameter channels. However, there are increasing indications that in two-phase flow decreased channel diameter will lead to increased heat transfer and thus the possibilities of increasing the system performance.

For certain applications small diameter channel heat exchangers have already been implemented, for other reasons than the decrease of working fluid inventory. One area is for cooling of electronics, where active cooling of individual components by fluid channels incorporated into the structure has been discussed for long. As an alternative, cold-plates with mini-channels or mini-channel heat pipes are used for cooling certain types of components. A second area is automotive air conditioning, where extruded multichannel aluminium tubes have been used for several years, primarily as condensers[85]. There are also some manufacturers who have specific methods of manufacturing heat exchangers with small hydraulic diameters. These producers may target specific applications, such as off-shore gas processing or chemical process intensification (where reduced inventories are a selling point), or they may rely on having customers in different areas, all having specific requirements concerning the heat transfer, which may be met by the compact designs.

Developments in the area of compact heat exchangers (CHEs) are partly driven by demands from industry, e.g. the electronic industry and the chemicals sector. A second reason for the development is the progress in the area of materials science: It is

[85] The development of micro-channel finned heat exchangers for residential split air conditioners was described recently by Salvatori Macri of Carrier S.p.A. at the AiCARR Meeting in Milan in March 2006. Charge reduction was highlighted as one major benefit.

now possible to manufacture small size objects with high precision in large quantities at low cost. We believe that these possibilities have not yet been totally explored for the benefit of the heat pump and refrigeration industry.
Highly relevant to all refrigeration and heat pump systems is the cost of disposal. It is believed that disposal/recycling will be facilitated by employing CHEs in the whole range of heat pump equipment, regardless of cycle type, as the quantities of metals to be disposed of, as well as the fluid(s) within the systems, should be reduced. The energy use in manufacture should also be reduced, reducing the overall environmental impact.

Goals of Annex 33

The principal goal of the Annex was to identify compact heat exchangers[86], either existing or under development, that may be applied in heat pumping equipment – including those using vapour compression, mechanical vapour recompression and absorption cycles. This has the aims of decreasing the working fluid inventory, minimising the environmental impact of system manufacture and disposal, and/or increasing the system performance during the equipment life, thereby reducing the possible direct and indirect effects of the systems on the global and local environments.

A second goal was to identify, where necessary propose, and document reasonably accurate methods of predicting heat transfer, pressure drop and void fractions in these types of heat exchangers, thereby promoting or simplifying their commercial use by heat pump manufacturers. Integral with these activities was an examination of manifolding/flow distribution in compact/micro-heat exchangers, in particular in evaporators.

A third goal was to present listings of operating limits etc. for the different types of compact heat exchangers, e.g. maximum pressures, maximum temperatures, material compatibility, minimum diameters, etc. and of estimated manufacturing costs or possible market prices in large scale production. It is intended within this context that opportunities for technology transfer from sectors where mass-produced CHEs are used (e.g. automotive) will be examined and recommendations made. Much of these data are presented within this document.

This Document

Compact heat exchangers were one of a range of energy-efficient technologies that the UK Energy Efficiency Best Practice Programme (EEBPP) promoted from the 1970s onwards, and remain of significance today. The EEBPP was a Government-funded initiative aimed at producing, for industry, the commercial and domestic sectors, a range of useful data on energy efficient technologies, practices etc., based additionally on case studies, research and development projects, and demonstration projects. It ran in parallel with similar initiatives from the European Commission –

[86] A selection of CHEs and their applications may be found in an IEA-CADDET Publication: Learning from Experiences with Compact Heat Exchangers. CADDET Analyses Series No. 25, June, 1999.

the Non-Nuclear Energy R&D Programme, more recently being known as the Framework Programmes for R&D.

For the process industries and the large commercial heat pump/refrigeration sector, for which the heat exchangers described within this document are most relevant, the activities were managed by the then Energy Technology Support Unit (ETSU) – now AEA Energy & Environment – at Harwell, Oxfordshire.

The original Training Package upon which this current one, for compact heat exchangers that might be used in process (and some other) heat pumps, is partially based, was produced by ETSU and W.S.Atkins, a private company in the UK. Unfortunately it is no longer available, according to the Carbon Trust web site, but it had been distributed as a CD-ROM and the basis of much of the data this document is derived from the CD and updated information from the EEBPP Good Practice Guides 89 and 198 already available through the Best Practice Programme[87]. All three documents provided information on compact heat exchanger technology, applications, selection, operation and Best Practice.

Other significant data sources on compact heat exchangers are the CADDET Guide listed in footnote 3 and the book on compact heat exchangers by Dr. John Hesselgreaves, a leading expert on heat exchanger design.[88] This covers, in addition to process CHEs, a number of types more relevant to small commercial/domestic heat pumps.

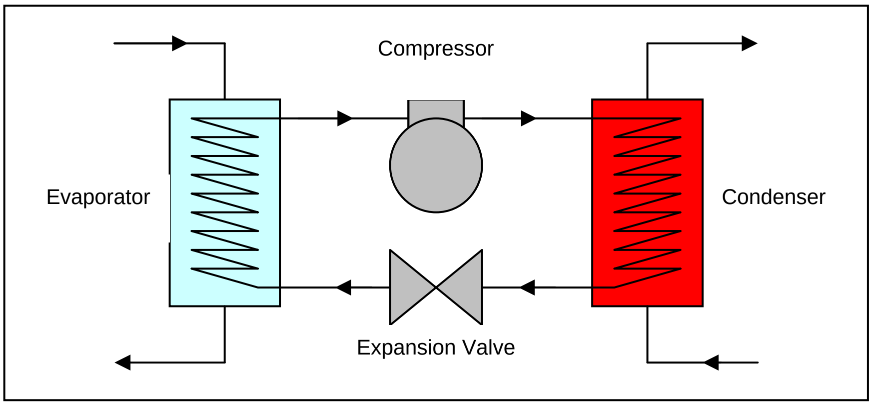


Fig. A3.2.1. A closed cycle heat pump

In its basic form, we are concerned with the evaporator and condenser of closed cycle heat pumps. Each of these heat exchangers might be 'compact' and within these guidance notes a number of CHEs are described which might be used for heat pump

[87] EEBPP Good Practice Guide 89 – Guide to Compact Heat Exchangers, EEBPP Good Practice Guide 198 – Experience in the Operation of Compact Heat Exchangers and EEBPP Good Practice Guide 141 – Waste Heat recovery in the Process Industries (that includes data on heat pumps/MVR units) were compiled by experts, including Professor Reay of Annex 33. The Good Practice Guide 198 is available to download as a .pdf file from the Carbon Trust – www.carbontrust.co.uk – while Good Practice Guide 141 may be ordered as a paper copy. There may be a modest charge for copes sent outside the UK.

[88] Hesselgreaves, J.E. Compact Heat Exchangers – Selection, Design and Operation. Pergamon. Elsevier Science, Oxford, 2001. ISBN 0-08-042839-8.

evaporator, condenser and other duties. In absorption cycle machines, the opportunities for CHE use are believed to be even greater.

A3.2.1.1 Introduction to Compact Heat Exchangers

A3.2.1.1.1 Definition of Compact Heat Exchangers

Compact heat exchangers (CHEs) are characterised by their high 'area density': this means that they have a high ratio of heat transfer surface to heat exchanger volume. Because of this, the initial development and use of compact heat exchangers was in the aerospace, road transport and marine sectors.

In appropriate applications, compact heat exchangers offer a number of advantages over more conventional designs including:

Improved heat exchanger effectiveness, closer approach temperatures and high thermal effectiveness.
Smaller volume and weight for a given duty.
Lower installed cost (in most cases).
Multi-stream and multi-pass configurations.
Tighter temperature control.
Energy savings.
Reduced inventory volume giving short residence times.

One somewhat arbitrary definition of a compact heat exchanger is having an area density (area/volume ratio) greater than 700 m^2/m^3. However, this strictly refers to the gas side of a gas-liquid heat exchanger. Table A3.2.1 lists the threshold area/volume densities of a range of generic heat exchanger types that could all be classed as 'compact'.

Table A3.2.1. Area Densities of Compact Heat exchangers

Compact Heat Exchanger Type	**Area Density (m^2/m^3)[(1)]**
Liquid-liquid	$\geq$200
Gas-liquid	$\geq$700
Gas-Gas	$\geq$700 (generally not relevant to heat pumps)

Notes:
(1) The area density includes secondary surfaces, such as fins.

A conventional shell and tube heat exchanger with 19 mm diameter tubes has an area density of about 100 m^2/m^3 on one fluid side although there are some 'compact' shell and tube heat exchangers with higher heat transfer areas, such as those used for fuel-oil heat transfer in aircraft engines. The use of shell and tube exchangers in absorption plant can lead to units as shown in Fig. A3.2.3.

Fig. A3.2.2. The typical absorption cycle chiller – a collection of large shell and tube heat exchangers.

Other types of heat exchanger also may be regarded as 'compact', for example the rotating regenerator or heat wheel type frequently has an area ratio in excess of 5,000 m^2/m^3.

Compact heat exchangers are commonly used for both single phase and two-phase applications as shown in Table A3.2.2.

Table A3.2.2. Fluid Streams in CHEs, According to Type.

Type	**Nature of Streams**
Gasketed plate and frame	Liquids; two-phase (1)
Brazed plate	Liquids; two-phase
Welded plate and frame	Liquids; two-phase (2)
Plate-fin	Gases; liquids; two-phase
Printed circuit	Gases; liquids; two-phase
Compact shell and tube	Liquids; two-phase

Notes:
(1) Two-phase includes boiling and condensation.
(2) One flow path may have welded plates, the other retaining gaskets.

A3.2.1.1.2 Compact Heat Exchanger Types

There are various types of compact heat exchanger available on the market.

Those particularly relevant to heat pumping systems include:

Plate Heat Exchanger Types, which include:
- Plate and Frame Heat Exchangers
- Partially Welded Plate Heat Exchangers
- Brazed Plate Heat Exchangers
- The Alfa-Rex fully-welded Plate Exchanger

Plate-fin Heat Exchangers
- Brazed Plate-fin Heat Exchangers
- Welded Plate-fin Heat Exchangers
- Diffusion-Bonded Plate-fin Heat Exchangers

Printed Circuit Heat Exchangers

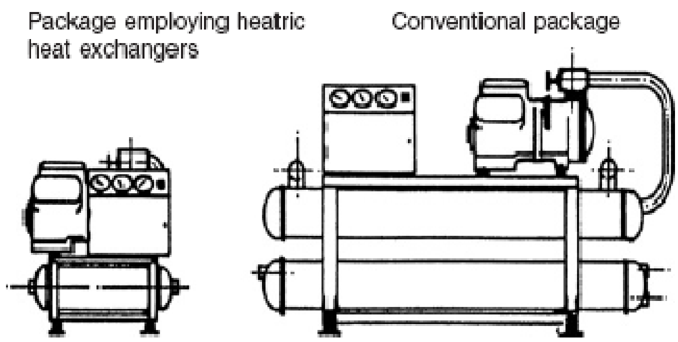


Fig. A3.2.3. The use of a PCHE allows one to re-engineer a chiller plant – the PCHE is used on the unit on the LHS.

Data in Table A3.2.3 give comparisons of the relevant properties/configurations and materials used in a range of CHEs that might be used in heat pumping systems.

A shell and tube heat exchanger is included for comparative purposes.

Table A3.2.3. Data on CHEs

Type of Heat Exchanger	Area Density (m^2/m^3) (0)	Stream Types (1)	Materials (3)	Temperature Range (^oC)	Maximum Pressure (bar) (2)	Fluid Limitations	Cleaning Methods	Corrosion Resistance (18)	Multi-stream Capability	Multi-pass Capability
Plate and frame (Gasketed)	→200	liquid-liquid gas-liquid two-phase	s/s, Ti, Incoloy Hastelloy, graphite, polymer	-25 to +175 Special –35 to +200	Normal 25 Special 40	Limited by gasket type. Not normal for gases	Mechanical (14) Chemical	Good (7)	Yes (9)	Yes
Partially welded plate	→200	gas-liquid liquid-liquid two-phase	s/s, Ti, Incoloy Hastelloy	-35 to +200	25	Few - Some types need clean fluids	Mechanical (4, 14) Chemical (6)	Good (7)	No	Yes
Fully welded plate (AlfaRex)	→200	gas-gas gas-liquid liquid-liquid two-phase	s/s, Ti, Ni alloys	-50 to +650	40	Few - Some types need clean fluid	Chemical	Excellent	No	Yes
Brazed plate	→200	liquid-liquid two-phase	s/s	Cu braze –195 to +220 Ni braze →400	Cu braze 30 Ni braze 16	Must be compatible with braze	Chemical (5)	Good (8)	No	No (10)
Brazed plate-fin	800 – 1,500	gas-gas gas-liquid liquid-liquid two-phase	Al, s/s, Ti Ni alloy	Al –270 to +200 s/s cryogenic to +650	120	Low fouling Many limitations with Al	Chemical	Good	Yes	Yes

Table A3.2.3. Data on CHEs

Type of Heat Exchanger	**Area Density (m^2/m^3) (0)**	**Stream Types (1)**	**Materials (3)**	**Temperature Range (^oC)**	**Maximum Pressure (bar) (2)**	**Fluid Limitations**	**Cleaning Methods**	**Corrosion Resistance (18)**	**Multi-stream Capability**	**Multi-pass Capability**
Diffusion-bonded plate-fin	700 - 800	gas-gas gas-liquid liquid-liquid two-phase	Ti, s/s, Ni	→400	200	Low fouling	Chemical	Excellent	Yes	Yes
Printed circuit	200 - 5,000	gas-gas gas-liquid liquid-liquid two-phase	s/s, Ni, Ni alloys Ti	-200 to +900	500	Low fouling	Chemical	Excellent	Yes	Yes
Shell and tube (19)	→100	gas-gas gas-liquid liquid-liquid two-phase	s/s, Ti, (shell also in c/s), many different materials	-100 to +600	Shell 300 Tubes 1400	Few	Mechanical (16, 14) Chemical (17)	Good	No	Yes

Notes for Table A3.2.3 → = up to, s/s = stainless steel, Ti = titanium, Ni = nickel, Al = aluminium, Cu = copper, c/s= carbon steel

(0) Area includes the secondary surface (such as fins)
(1) Two-phase includes boiling and condensing duties
(2) The maximum pressure capability is unlikely to occur at the higher operating temperatures, and assumes no pressure/stress-related corrosion
(3) Other special alloys are frequently available
(4) On gasket side
(5) Ensure compatibility with copper braze
(6) On welded side
(7) Function of gasket as well as plate material
(8) Function of braze as well as plate material
(9) Not common
(10) Not in a single unit
(11) On tube side
(12) Only when flanged access provided, otherwise chemical cleaning
(13) Five fluids maximum
(14) Can be dismantled
(15) Shell may be composed of polymeric material
(16) On shell side
(17) On plate or tube side
(18) Primarily a function of construction materials rather than the exchanger type
(19) Not a compact exchanger technology (given for comparison)

A3.2.1.2 Plate Heat Exchangers

There are several types of plate heat exchanger, which are seeing increasing duties within heat pumping equipment, particularly in small systems where brazed plate heat exchangers can be highly cost effective. The most common PHE is the plate and frame, or gasketed, type.

A3.2.1.2.1 Plate and Frame Heat Exchangers

The plate and frame heat exchanger was one of the first compact exchangers to be used in the UK process industries, being originally introduced in 1923; the first plates were made of gunmetal. It is currently second to the shell and tube heat exchanger in terms of market share.

The most common variant of the plate and frame heat exchanger consists of a number of pressed, corrugated metal plates compressed together into a frame. These plates are provided with gaskets, partly to seal the spaces between adjacent plates and partly to distribute the media between the flow channels. The most common plate material is stainless steel.

Plate and frame heat exchangers were first used in the food and dairy industries, where the ability to access plate surfaces for cleaning is imperative.

There are numerous suppliers of plate and frame heat exchangers. While all manufacturers follow the same basic construction method, the differences in performance claimed tend to be associated with the patterns on the plates that form the flow channels, and the choice of gasket materials. Newer designs can accommodate features such as grossly unequal flow rates on each side of the plate.

Construction

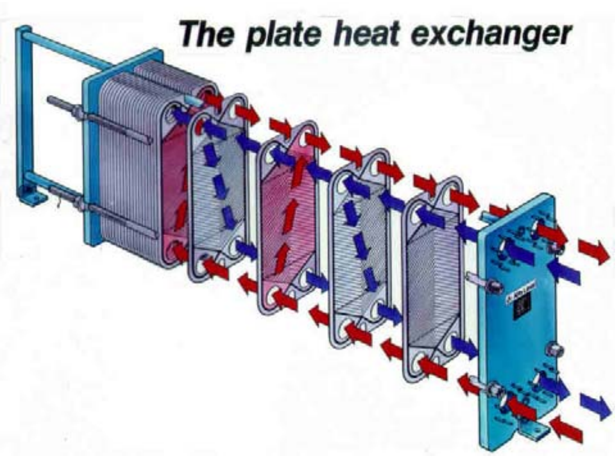


Fig.A3.2.4 (LHS). An exploded view of a typical plate and frame heat exchanger design, and the assembled unit (RHS).

The heat transfer surface consists of a number of thin corrugated plates pressed out of a high grade metal. The pressed pattern on each plate surface induces turbulence and minimises stagnant areas and fouling. Unlike shell and tube heat exchangers, which can be custom-built to meet almost any capacity and operating conditions, the plates for plate and frame heat exchangers are mass-produced using expensive dies and presses. Therefore, all plate and frame heat exchangers are made with what may appear to be a limited range of plate designs.

Although the plate heat exchangers are made from standard parts, each one is custom designed. Variation in the trough angle, flow path or flow gap can alter the performance of the heat exchanger.

The plate pack is clamped together in a frame suspended from a carrying bar. Gaskets are fitted to seal the plate channels and interfaces. The frame consists of a fixed frame plate at one end and a moveable pressure plate at the other. The moveable plate facilitates access for cleaning or exchanging the heat transfer surfaces. A feature of this type of heat exchanger is the ability to add or remove surface area as necessary.

The plates are grouped into passes with each fluid being directed evenly between the paralleled passages in each pass. Whenever the thermal duty permits, it is desirable to use single pass, counter flow for an extremely efficient performance. Although plate and frame exchangers can accept more than two streams, two-pass arrangements are the most common.

Plates can be produced from all pressable materials. The most common construction materials are:

- Stainless steel (AISI304, 316).
- Titanium.
- Incoloy.
- Hastelloy.

Where corrosion is a problem, some manufacturers offer plate and frame heat exchangers in non-metallic materials, such as a graphite/fluoroplastic composite or a polymer. These may be of use in some instances in absorption or even adsorption cycle machines.

Gasket properties have a critical bearing on the capabilities of a plate and frame heat exchanger, in terms of its tolerance to temperature and pressure. The gasket material, if it comes into contact with the heat pump working fluid, should of course show excellent compatibility.

Gaskets are commonly made of:

- Nitrile rubber.
- Hypalon.
- Viton.
- Neoprene.
- EPDM.

Originally, most manufacturers used glue to fix the gaskets to the plates. Several proprietary fixing techniques are available that eliminate the need to use glue, and most manufacturers have adopted these methods. These so-called 'glueless' gaskets are suitable for some heavy-duty industrial applications. The simplified removal and location of such gaskets can be beneficial, as it reduces downtime when on-site changing is necessary.

Care should be taken in locating the gaskets during reassembly, as imperfect sealing is the main disadvantage of the plate and frame heat exchanger. Many OEMs (original equipment manufacturers) suggest that the units be sent back to them for gasket replacement.

Double-wall units are another variant catering for specific process situations. Here two special non-welded plates, fitted with a non-glued gasket to seal and hold the plates together, replace the single plate normally separating the two media. Consequently, two walls separate the product and service medium giving additional protection against cross contamination and the occurrence of a hostile reaction. The partially welded plate unit (see Section A3.2.1.2.2) is designed for handling aggressive media.

Operating Limits

The operating limits of gasketed plate and frame heat exchangers vary slightly from manufacturer to manufacturer. Typically, the operating temperature range of the metal plates is from -35°C to +200°C. Design pressures up to 25 bar can be tolerated, with test pressures to 40 bar.

Heat transfer areas range from 0.02 m^2 to 4.45 m^2 (per plate). Flow rates of up to 3,500 m^3/hour can be accommodated in standard units, rising to 5,000 m^3/hour with a double port entry. Approach temperatures as low as 1°C are feasible with plate and frame heat exchangers.

The surface pattern on the plates tends to induce good mixing and turbulence, and in general this type of heat exchanger has a low propensity for fouling. Fouling resistances of typically 25% of those for shell and tube heat exchangers have been measured by the Heat Transfer Research Incorporated (HTRI) in the USA.

Where fouling is a concern, the gap between the plates can be widened. For example, one manufacturer offers plates with a 13 mm gap and coarse contours for viscous liquids and fluids containing fibres, solids, crystals, pulp, etc.

Applications

Gasketed plate and frame heat exchangers have a large range of applications typically classified in terms of the nature of the streams to be heated/cooled as follows:

- Liquid-liquid.
- Condensing duties.
- Evaporating duties.

Gasketed units may be used in refrigeration and heat pump plants and are extensively used in the processing of food and drinks, where the ease of plate cleaning and re-gasketing are important. In the chemicals sector, a substantial list of heating and cooling applications includes cooling isoparaffin, sulphuric acid, salt solutions, hexane and kerosene. Heating glycerine and condensing ethanol are other routine uses. In the accompanying document on the market for process heat pumps, the vast potential for heat pumps in the chemical process industries is evident.

A3.2.1.2.2 Partially Welded Plate Heat Exchangers

Externally, partially welded plate heat exchangers or twin plate heat exchangers resemble a fully-gasketed plate and frame unit. However, the difference is the plate pack has alternating welded channels and gasketed channels.

The advantage of welding the plate pairs is that, except for a small gasket around the ports, other materials are eliminated and corrosion is slightly reduced.

Construction

The overall construction is similar to that of the gasketed plate and frame heat exchanger (described in Section A3.2.1.2.1), with one important exception: each plate pair is welded together, normally using laser welding. Porthole gaskets fabricated from highly resistant elastomer or non-elastomer materials, are attached using a glueless method.

Plate construction materials are as for the gasketed plate and frame heat exchanger. The plate material is normally selected for its resistance to corrosion.

Operating Limits

As for the gasketed plate and frame type, but with the added protection from leaks afforded by the partially welded construction. The welded section should tolerate higher pressures.

Applications

As for gasketed plate and frame heat exchanger, but extended to include more aggressive media.

Partially welded plate heat exchangers are used for the evaporation and condensation of refrigerants such as ammonia and hydrofluorocarbons (HFCs), and for chemical and general process duties involving aggressive liquids. Industrial heat pump applications could benefit from the use of these exchangers.

A3.2.1.2.3 Brazed Plate Heat Exchangers

The brazed plate heat exchanger consists of a pack of pressed plates brazed together, thus completely eliminating the use of gaskets. The frame can also be omitted.

Brazed plate heat exchangers tend to be offered by the principal suppliers of the plate and frame type and tend to be directed at niche markets such as refrigeration. These exchangers have heat transfer capabilities up to 600 kW, depending on the supplier.

The corrugated plates induce a highly turbulent flow such that the scouring action of the turbulence reduces surface deposits in the heat exchanger.

Construction

Brazed plate heat exchangers consist of a number of pressed stainless steel plates joined together by brazing. Typically a very high content copper braze is used, and the brazing process is carried out under vacuum. Capillary forces collect the brazing material at the contact points between the plates.

As well as sealing around the periphery of the plates, the internal herringbone contact points are also brazed, permitting higher pressures to be tolerated than in gasketed units.

Stainless steel is usually used as the plate material.

Operating Limitations

Copper brazed units are available for temperatures up to 225^{o}C and a maximum operating pressure of 30 bar, but copper braze may produce an incompatibility with some working media. Nickel brazed units are available for temperatures up to 400^{o}C and maximum operating pressures of 16 bar.

Applications

The brazed plate unit is in part directed at the refrigeration/heat pump market for evaporators and condensers (water-cooled), but it is also suitable for process water heating, heat recovery and district heating systems. Brazed plate heat exchangers can also be used as desuperheaters, subcoolers, economisers and oil coolers.

The role of brazed plate heat exchangers in 'intensified' heat pumps is typified by their use in the Rotex/Rotartica unit as a solution heat exchanger (absorption cycle), illustrated in Fig. A3.2.5. The solution heat exchanger s visible towards the bottom of the sketch.

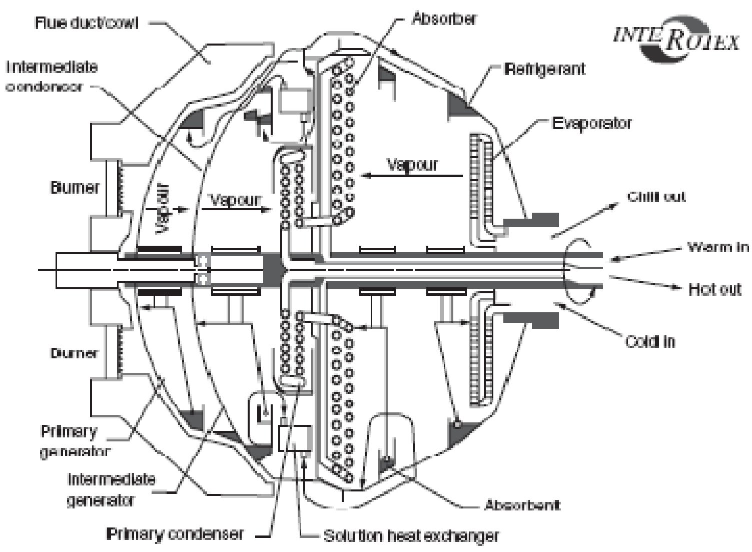


Fig. A3.2.5. The Rotex absorption cycle heat pump.

A3.2.1.2.4 The AlfaRex Welded Plate Heat Exchanger

The AlfaRex heat exchanger was the first full-size, gasket-free heat exchanger. The herringbone plate design creates channels with high fluid turbulence that increases thermal efficiency and minimises the risk of fouling.

Media flow is counter-current which is optimal for heat transfer, particularly in heat recovery duties. Per unit of surface area, counter current flow achieves 20% higher heat transfer values than cross flow.

Construction

Plates with the traditional herringbone pattern are laser-welded together to form a plate pack in which both media are in full counter-current flow.

Plate materials include:

- AISI 316.
- SMO.
- Titanium.
- Palladium-stabilised Titanium.
- Hastelloy C276.
- Nickel.

Operating Limits

The AlfaRex design operating temperature range is -50°C to +350°C at pressures up to 40 bar. The exchanger is capable of handling flowrates up to 800 m^3/hour.

Applications

Typical heat pumping duties for the AlfaRex heat exchanger include: Evaporation and condensing of ammonia or carbon dioxide in heat pump and adsorption systems.

A3.2.1.3 Plate-fin Heat Exchangers

Plate-fin heat exchangers (PFHEs) are a matrix of flat plates and corrugated fins in a sandwich construction.

Brazed aluminium plate-fin heat exchangers exhibit certain features and characteristics that distinguish them from other types of heat exchanger.

These include:

- A very large heat transfer area per unit volume of heat exchanger. This surface area is composed of primary and secondary (finned) surfaces. Typically, the effective surface area is over five times greater than that of a conventional shell and tube heat exchanger. Area densities range from 850 to 1,500 m^2/m^3.
- A single heat exchanger can incorporate several different process streams and the unique plate-fin construction allows these to enter/exit the exchanger at intermediate points along the exchanger length rather than just at the ends.
- Very close temperature approaches between streams (typically 1 to 3^oC) can be accommodated leading to operational cost savings.
- High thermal efficiency, use of aluminium and multi-stream capability combine to form a compact, low-weight structure.

The versatility of plate-fin heat exchangers, coupled with the ability to manufacture them in a variety of other materials, makes them ideal for a range of duties in heat pumping systems, although they have yet to be seriously exploited in these areas. Work on using PFHEs has concentrated to date on absorption cycle units, but they could be used in vapour compression cycle units, including those using CO_2 working fluid.

A3.2.1.3.1 Brazed Plate-Fin Heat Exchangers

This section describes brazed plate-fin heat exchangers, an example of which is pictured in Fig. A3.2.6.

Fig. A3.2.6. Large Chart Inc. Aluminium Plate-Fin Heat Exchanger
(not all CHEs are small!)

Construction

The heat exchanger is assembled from a series of flat sheets and corrugated fins in a sandwich construction. Tube plates (i.e. parting sheets) provide the primary heat transfer surface. Tube plates are positioned alternatively with the layers of fins in the stack to form the containment between individual layers. These elements are built into a complete core and then vacuum brazed to form an integral unit.

The heat transfer fins (see Fig. A3.2.7 and Table A3.2.3) provide the secondary heating surface for heat transfer. Fin types, densities and heights can be varied to ensure that exchangers are tailor-made to meet individual customer requirements in terms of heat transfer performance versus pressure drop.

Distributor fins collect and distribute the heat transfer fluid from the header tank to the heat transfer fins at the inlet and reverse the process at the outlet. Distribution fins are taken from the same range as the heat transfer fins, but tend to be less dense.

The heat exchanger core is then encased in a welded structure that incorporates headers, support plates and feed/discharge pipes.

Most plate-fin heat exchangers are made of aluminium, with a vacuum-brazed core. Corrosion-resistant and heat-resistant brazing alloys can be used; for example plate-fin heat exchangers can also be assembled in stainless steel, a variety of nickel-based alloys, and some other specialist alloys.

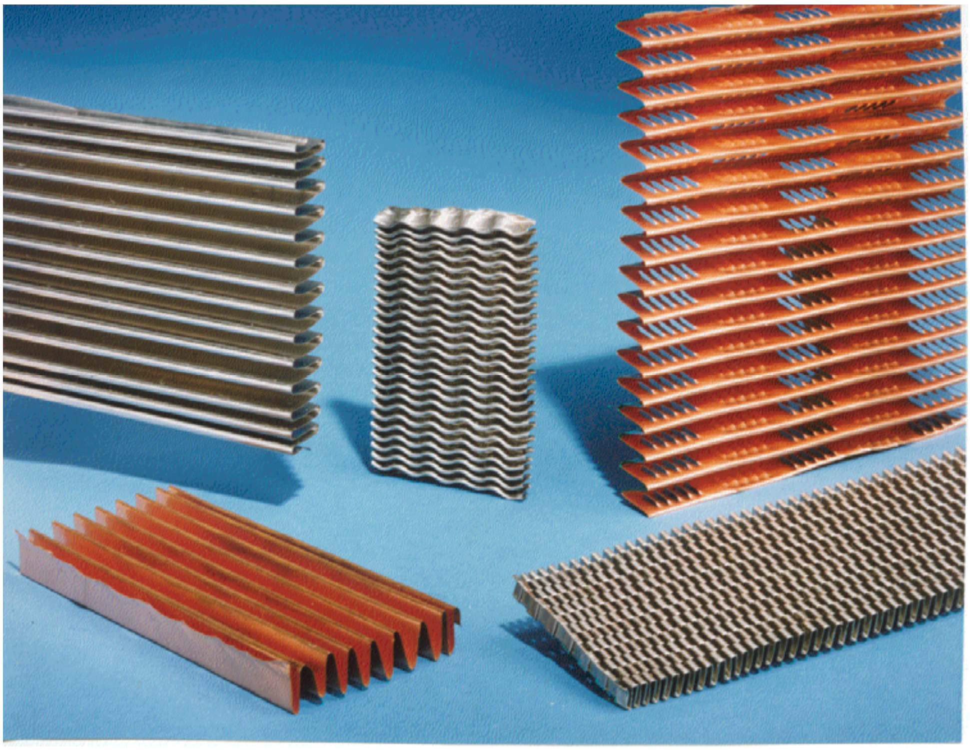


Fig. A3.2.7. Fin structures such as these are used between the plates in PFHEs.

3.1.2 Operating Limits

The maximum operating temperature of a plate-fin heat exchanger is a function of its construction materials. Aluminium brazed plate-fin heat exchangers can be used from cryogenic temperatures (-270^{o}C) up to 200^{o}C, depending on the pipe and header alloys. Stainless steel plate-fin heat exchangers are able to operate at up to 650^{o}C, while titanium units can tolerate temperatures approaching 550^{o}C.

Aluminium brazed units can operate at up to 120 bar, depending on the physical size and the maximum operating temperature. Stainless steel plate-fin heat exchangers are currently limited to 50 bar, with developments expected that will extend the capability to 90 bar. Higher pressures can be tolerated by using a diffusion-bonded structure.

The size of a plate-fin heat exchanger is a function of the procedure used to assemble the core. In the case of aluminium vacuum-brazed units, modules of 6.25 m x 2.4 m x 1.2 m are available.

When selecting brazed aluminium plate-fin exchangers, the engineer should ensure that:

- All fluids must be clean. Filtration must be used to remove particulate matter over 0.3mm.
- Fluids must be non-corrosive to aluminium (if this metal is used). Water is suitable if it is a closed loop and contains corrosion inhibitors.
- Fluids must be in the temperature range –270 to +200^{o}C.
- The maximum design pressure is less than 120 bar.

Table A3.2.3. Brazed Plate-Fin Types (see also Fig. A3.2.7 above)

Fin Type	**Application**	**Features**	
		Relative Δp	**Relative Heat Transfer**
Plain	General	Lowest	Lowest
Perforated	Boiling streams	Low	Low
Herringbone	Gas streams with low allowable ΔP High pressure streams Gas streams for hydrocarbon and natural gas applications	High	High
Serrated	Gas streams in air separation applications General	Highest	Highest

Applications

The plate-fin heat exchanger is suitable for use over a wide range of temperatures and pressures for gas-gas, gas-liquid and multi-phase duties. Many of those listed in Table A3.2.4 involve heat pumping systems – mainly vapour compression cycle.

Table A3.2.4 Typical Applications of Brazed Plate-Fin Heat Exchangers

Plant Types	Products and Fluids	Typical Temperature Range (°c)	Typical Pressure Range (bar.g)
Industrial Gas Production - Air Separation - Liquefaction	Oxygen Nitrogen Argon Rare Gases Carbon Dioxide	-200 to +65	1 to 60
Natural Gas Processing (NGP) - Expander Type - Nitrogen Rejection Unit (NRU) - Liquefied Petroleum Gas (LPG) - Helium Recovery	Methane Ethane Propane Butane Pentane Nitrogen Helium Hydrogen Hexane Carbon Dioxide	-130 to +100	15 to 100
Liquefied Natural Gas (LNG) - Base Load - Peakshaver	Liquefied Natural Gas Multi-component refrigerants	-200 to +65	5 to 75
Petrochemical Production - Ethylene - MTBE - Ammonia - Refinery Off-Gas Purification	Ethylene Propylene Ethane Propane MTBE Ammonia Carbon Monoxide Hydrogen	-200 to +120	1 to 100
Refrigeration Systems - Cascade Cooling - Liquefaction	Helium Freon Propane Ethylene Propylene Nitrogen Hydrogen Multi-component Refrigerants	-270 to +100	15 to 45

It should be noted, as highlighted in the introduction to the PFHE, that the units, and their welded or diffusion bonded equivalents, can be used in absorption cycle heat pumping systems to considerable effect, reducing the volume significantly.

A3.2.1.4 Printed Circuit Heat Exchangers

Printed circuit heat exchangers are highly compact, corrosion resistant heat exchangers capable of operating at pressures of several hundred atmospheres and temperatures ranging from cryogenic to several hundred degrees Celsius.

The printed circuit heat exchanger (PCHE) design offers a unique combination of innovative manufacturing technology and potential breadth of application. In common with some other compact heat exchangers, it is potentially more than just a compact plate heat exchanger; the structure has applications in a variety of other unit operations, including reactors, mass transfer and mixers and as a structural member if required.

PCHEs are constructed from flat alloy plates with fluid flow passages photo-chemically machined (etched) into them. This process is similar to manufacturing electronic printed circuit boards, and gives rise to the name of the exchangers. An example of a plate showing a 'herringbone' pattern of flow paths is shown in Fig. A.3.2.8. Other illustrations are given on the Company web site – www.heatric.com

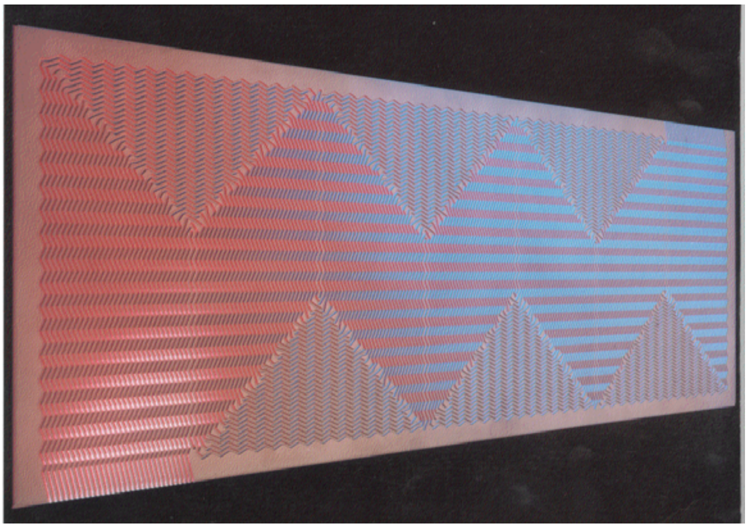


Fig. A.3.2.8. Fluid Flow Paths on a Typical Printed Circuit Heat Exchanger Etched Plate

Heatric originally developed printed circuit heat exchangers in Australia, where this type of heat exchanger first became commercially available for refrigeration and process applications in 1985. In 1990, Heatric moved to the UK and has supplied printed circuit exchangers into the offshore and process sectors, both in the UK and overseas.

Construction

The standard manufacturing process involves chemically milling (etching) the fluid flow passages into the plates. This allows enormous flexibility in thermal/hydraulic design, as complex new plate patterns require only minimal re-tooling costs.

This plate/channel forming technique can produce a wide range of flow path sizes, the channels varying typically from 0.5 to 2.0 mm in depth.

Stacks of etched plates, carrying flow passage designs tailored for each fluid, are diffusion bonded together to form a compact, strong, all-metal heat exchanger core. No gaskets or brazing materials are required for the assembly. Diffusion bonding allows the plates to be joined so that the bond acquires the same strength as the parent metal. The thermal capacity of the exchanger is built to the required level by welding together diffusion bonded blocks to form the complete heat exchanger core. Finally, fluid headers and nozzles are welded to the cores, in order to direct the fluids to the appropriate sets of passages. Fig. A3.2.9 shows a completed heat exchanger unit.

Materials of construction include stainless steel (SS 300 series) and titanium as standard, with nickel and nickel alloys also being commonly used. A copper variant is being developed.

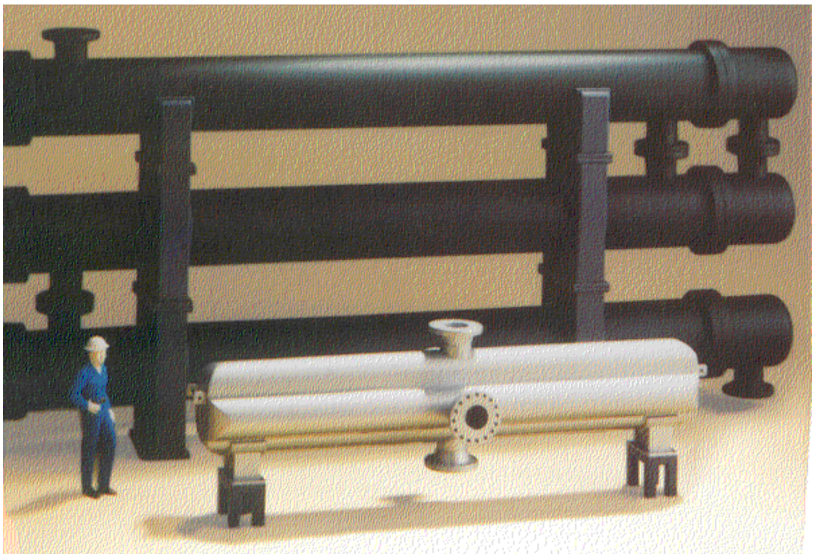


Fig. A3.2.9. A PCHE shown in front of the three shell and tube heat exchangers it replaced.

Operating Limits

Mechanical design is flexible: etching patterns can be adjusted to provide high pressure containment where required. Due to its construction, the printed circuit heat exchanger is able to withstand substantial pressures. Pressures as high as 200 bar are routine, with values in the range 300 - 500 bar being possible.

The all welded construction is compatible with very high temperature operation, and the use of austenitic steel allows cryogenic operation. Operating temperature ranges from -200°C to +900°C, the upper limits depending on the metal selected and the pressure duty.

Passages are typically of the order of 2 mm semi-circular cross-section (i.e. 2 mm across and 1 mm deep) for reasonably clean applications, although there is no absolute limit on passage size.

Prime heat transfer surface densities, expressed in terms of effective heat transfer area per unit volume, can be up to 2500 m^2/m^3. This is higher than prime surface densities in gasketed plate exchangers, and an order of magnitude higher than normal prime surface densities in shell and tube exchangers.

Operation

Printed circuit heat exchangers are all welded so there is no braze material employed in construction, and no gaskets are required. Hence the potential for leakage and fluid compatibility difficulties are reduced and the high level of constructional integrity renders the designs exceptionally well suited to critical high pressure applications, such as gas compression cooling exchangers on offshore platforms.

The thermal design of printed circuit heat exchangers is subject to very few constraints. Fluids may be liquid, gas or two-phase, multi-stream and multi-pass configurations can be assembled and flow arrangements can be truly counter-current, co-current or cross-flow, or a combination of these, at any required pressure drop.

Where required high heat exchange effectiveness (over 98%) can be achieved through very close temperature approaches in counter-flow. To simplify control, or to further maximise energy efficiency, more than two fluids can exchange heat in a single core. Heat loads can vary from a few watts to many megawatts, in exchangers weighing from a few kilograms to thousands of kilograms.

Flow induced vibration, an important source of failure in shell and tube exchangers, is absent from printed circuit heat exchangers.

A simple strainer upstream of the unit will remove outsize particles, while the corrosion resistant materials of construction for printed circuit heat exchangers, the high wall shear stresses, and the absence of dead spots assist in resisting fouling deposition.

Design

Detailed thermal design of printed circuit heat exchangers is supported by proprietary design software developed by the manufacturer that allows infinite geometric variation to passage arrangements during design optimisation. Variations to passage geometry have negligible production cost impact since the only tooling required for each variation is a photographic transparency for the photo-chemical machining process.

Although the scope of printed circuit heat exchanger capabilities is much wider, as a sizing guide it is safe to assume that channel patterns can be developed to mimic any j- and f- factor characteristics (found in publications such as “Compact Heat Exchangers” by J.E. Hesselgreaves) for aluminium surfaces, or data presented by gasketed plate manufacturers.

It is rarely necessary to apply a correction factor substantially less than 1 to the LMTD calculated for an heat exchanger, no matter how high the effectiveness required, because of the printed circuit heat exchanger counter-flow capabilities. Pressure drops can be specified at will, however as with all heat exchangers, lower allowable pressure drops will result in lower heat transfer coefficients and hence larger exchangers.

Applications

Printed circuit heat exchangers extend the benefits of compact heat exchangers into applications where pressure, temperature or corrosion prevents the use of conventional plate exchangers. As mentioned above, the printed circuit heat exchanger can handle gases, liquids and two-phase flows. Application areas where heat pumping duties might be involved are:

- Power and energy:
  - Feedwater heating.
  - Geothermal generation.
  - Chemical heat pumps.
- Refrigeration:
  - Chillers and condensers.
  - Cascade condensers.
  - Absorption cycles.

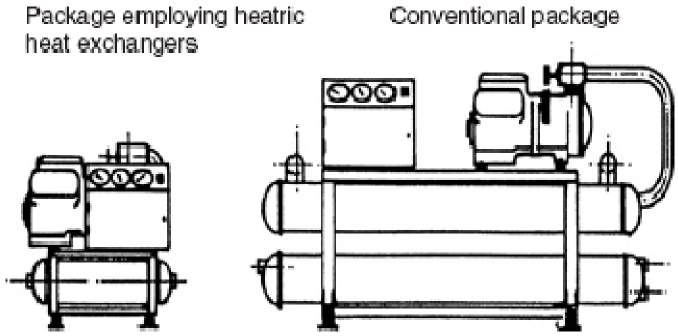


Fig. A3.2.10. The PCHE-based package on the left has been ‘re-engineered’ to have a much smaller footprint than the chiller using conventional heat exchangers.

The heat exchanger can be fabricated in any material which can be diffusion bonded - stainless steel, titanium, higher alloy steels and nickel. As an alternative, brazing can be used, which allows metals such as aluminium to be considered. This potentially could lead to capital cost reductions compared to those fabricated using more bespoke procedures.

A3.2.2 Advanced CHE Types that could Influence Heat Pump Design

The Section below, taken from: Reay. D.A., Ramshaw, C. and Harvey, A.P. 'Process Intensification – Engineering for Efficiency, Sustainability and Flexibility' (Butterworth-Heinemann, 2008) identifies some concepts that may come to play important roles in compact heat pumping equipment in the future.

A3.2.2.1 The 'Chart-flo' Heat exchanger

The Chart-flo heat exchanger and its stable-mates in the heat exchanger-reactor field are the latest truly innovative designs to enter the CHE marketplace. Produced by Chart Energy & Chemicals, the Chart-flo unit extends the option for those who are looking for high integrity, highly compact units able to operate over a range of pressures and temperatures not met with more conventional gasketed or welded CHEs. The manufacturing procedures are similar to those of the PCHE (chemical etching and diffusion bonding – although brazing is possible – see below), but the construction allows the use of small passageways, which significantly increases the porosity (free volume fraction) of the heat exchanger core. This can result, in appropriate applications, in a substantially higher area density than the PCHE. For example, a doubling of porosity, other factors being equal, results in a halving of the volume for a given surface area.

An expanded view of a Chart-flo unit is shown in Fig. A3.2.11, while some plates – the shims – are illustrated in Fig. A3.2.12.

Fig. A3.2.11. The Chart-flo compact heat exchanger – in this variant larger passages are provide to take into account possible fouling.

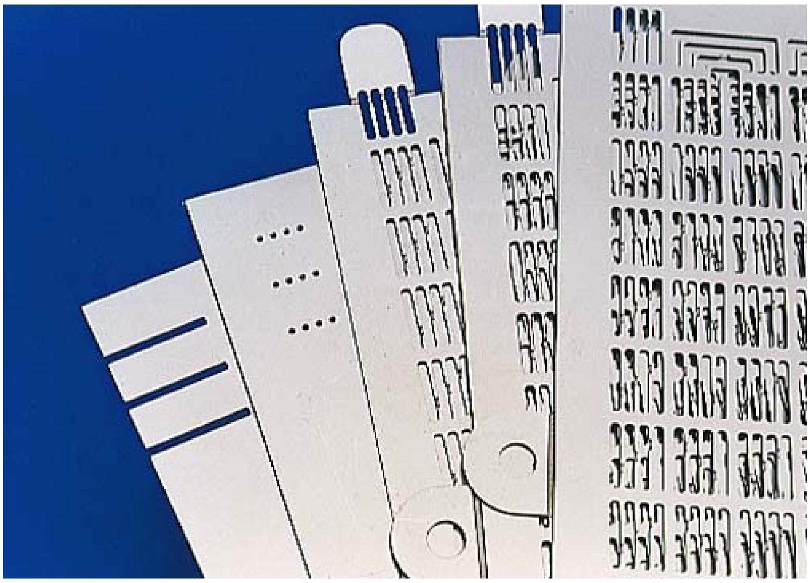


Fig. 3.2.12. Plates used in the Chart-flo Heat Exchanger

A particular feature of the Chart-flo is that it was developed at the outset for applications which could encompass reactions as well as pure heat transfer duties - in its Chart-pak and other forms. The compactness (see below) implies that it could be an integral part of many 'intensified' processes, and a study of its implications for reducing the size of absorption cycle refrigeration plant has illustrated the benefits of multi-stream, multi-pass and multi-functional use within a single module.

As mentioned earlier, the high porosity of the Chart-flo unit, together with the ability to use low hydraulic diameters, gives it advantages over several other CHEs in terms of compactness. For clean stream duties, the volume of a Chart-flo heat exchanger could be as low as 5% of that of the equivalent shell and tube heat exchanger.

An interesting aspect of compact heat exchanger structures such as the Chart and some Heatric variants and materials like metal foams (see Section A3.2.2.2 below) is that they may have a dual role – a heat transfer and a structural one. The nature of some compact heat exchanger surfaces when formed into thin multiple plate assemblies is that they are light in weight but strong and/or rigid. This allows heat transfer functions to be directly integral with a structural member, such as the skin of an aircraft.

A3.2.2.2 Foam Heat Exchangers

Rigid foam has benefits for heat transfer enhancement and it is also used as the basis of heat exchangers. One example of its use is in a gas-liquid heat exchanger, the tubes penetrating the bulk of the foam, which replaced conventional extended surfaces. The tubes are sintered into the foam, giving good contact. Foam pore size was 0.1-5 mm. Foam could be put inside tubes or between plates, configuring a gas-gas heat exchanger. The foam can be described as a *three-dimensional extended surface*. Work has been under way at Brunel University in the UK (Zhao et al, 2006) to characterise the heat transfer and pressure drop behaviour of the foam, involving some mathematical modelling. This could lead to designs for tube-in-foam heat exchangers of co/counter or cross-flow configuration. Other research at ETH, Zurich has investigated the use of metal foams as compact heat exchangers, (Boosma et al, 2003).

Advantages of the foam include a choice of metals to 1000^{0}C, low weight, compact and the ability to be formed in complex shapes. Some potential applications include gas turbine tail gas heat exchangers and hydrogen liquefaction plant.

Within the area of heat exchangers, the work at ETH was based on aluminium foam of the type shown in Fig. A3.2.13.

The Brunel work on tubes containing foams is illustrated in Figs. A3.2.14 and A3.2.15 (Zhao et al, 2006). It can be seen that the overall heat transfer coefficient of the metal-foam filled heat exchanger is significantly higher than that of conventional finned tube heat exchangers. For example, the overall heat transfer coefficient of a 10 ppi (pores per inch) copper-foam filled heat exchanger is shown to be more than double that of a finned tube heat exchanger (spiral fins 5000 fins/m, height 1 mm, thickness 0.075 mm and inner tube grooves width 0.1 mm, height 1 mm). Therefore, it is clear that the use of metal foams can greatly enhance the heat transfer, and metal foams have significant potential in the manufacture of compact heat exchangers.

Other applications include aircraft wing structures for the aerospace industry, core structures for high strength panels, and containment matrices and combustors. ETH has stated that due to the high surface-area density and strong mixing capability for the fluid, open cell metal foams are now regarded as one of the most promising materials for the manufacture of efficient compact heat exchangers. Again, it is worth highlighting the dual role of heat transfer and structural member that materials such as foam can satisfy.

The application within heat pumps and refrigerators was not considered at Brunel, but the use of foams should be feasible.

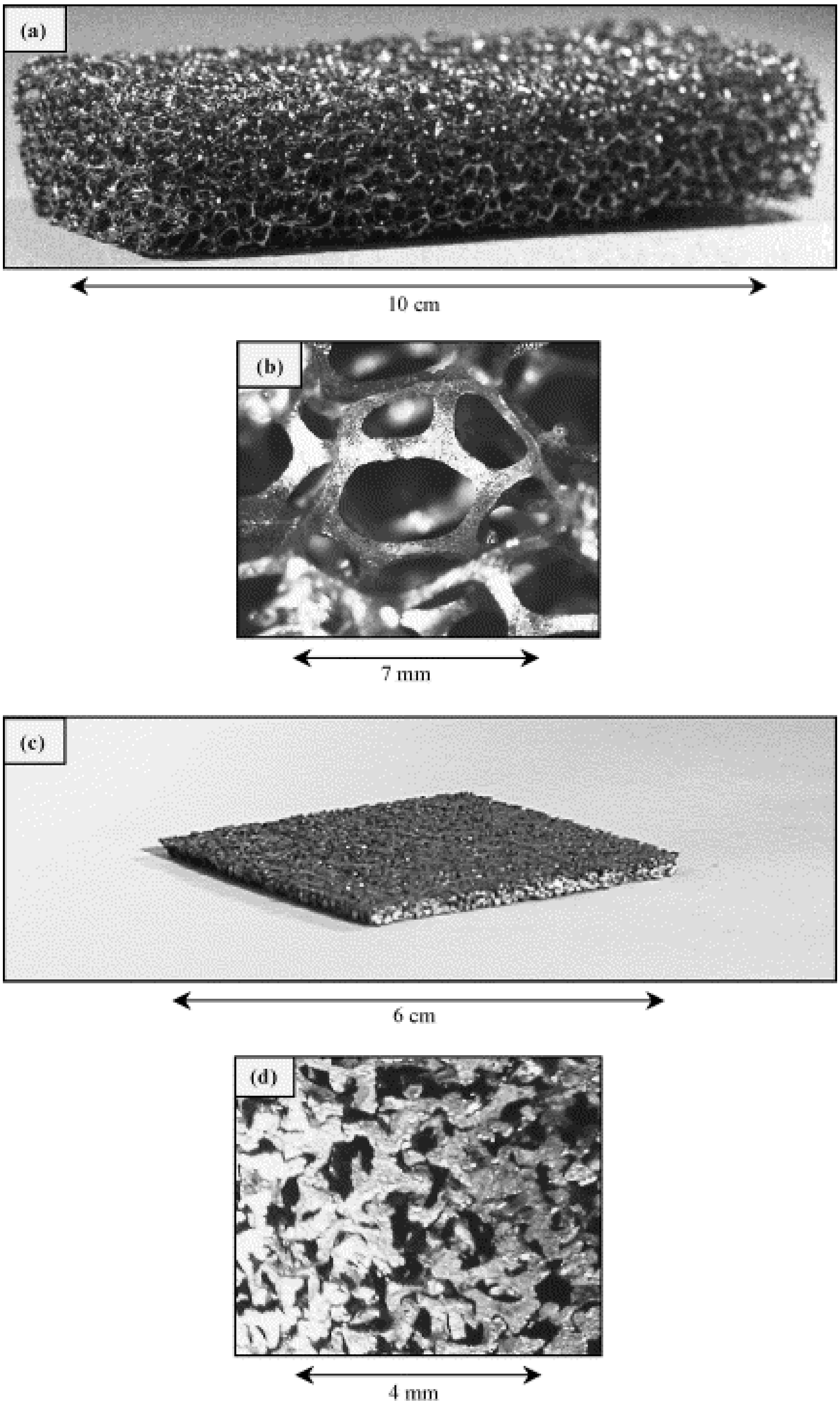


Fig. A3.2.13. An open-cell aluminium foam. This can be compressed to form smaller pore structures for CHEs, as was done at ETH (c and d).

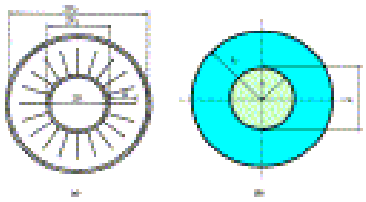


Fig. 3.2.14. The two geometries tested at Brunel, a tube-in-tube exchanger with internal fins, and one of identical size with the fins replaced by foam.

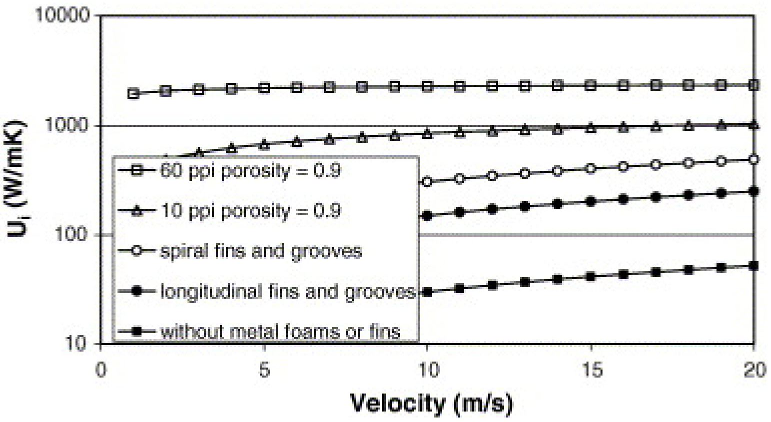


Fig. 3.2.15. Performance data comparing the Brunel University foam and finned units. The superiority of the foam is evident.

A3.2.2.3 Mesh Heat Exchangers

Structure is mentioned specifically in research reported recently at Cambridge University (Tian et al, 2007). Here metal meshes (woven as textiles) have been tested as heat exchangers. Although not a new concept, the mesh heat exchanger is an alternative to metal foams, and as readers who are familiar with heat pipe technology will recognize, woven meshes and twills are available in a wide range of materials and pore sizes, and can be stacked and brazed, where necessary.

The unit illustrated in Fig. A3.2.16 may be configured as a cross flow heat exchanger. In the case in the Figure, the cooling flow is flowing horizontally through the mesh, removing heat from devices convecting to the lower surface face sheet. Pressure may be applied (as it is a load bearing structure). The measurements carried out by the research team are shown in Fig. A3.2.17, where data are compared with other porous heat exchanger types. Note that an LFM is a lattice-frame material. The mesh is comparable in terms of heat transfer to the foam.

Pressure drop and heat transfer are strong functions of the path length through the mesh (or foam). Probably the mesh is less reproducible than foam structures, once several mesh layers are stacked, but there may be advantages in accessibility of material and flexibility. Interestingly a mesh reactor, albeit with the mesh fulfilling a slightly different purpose, has been constructed at University College London – see www.pinetwork.org

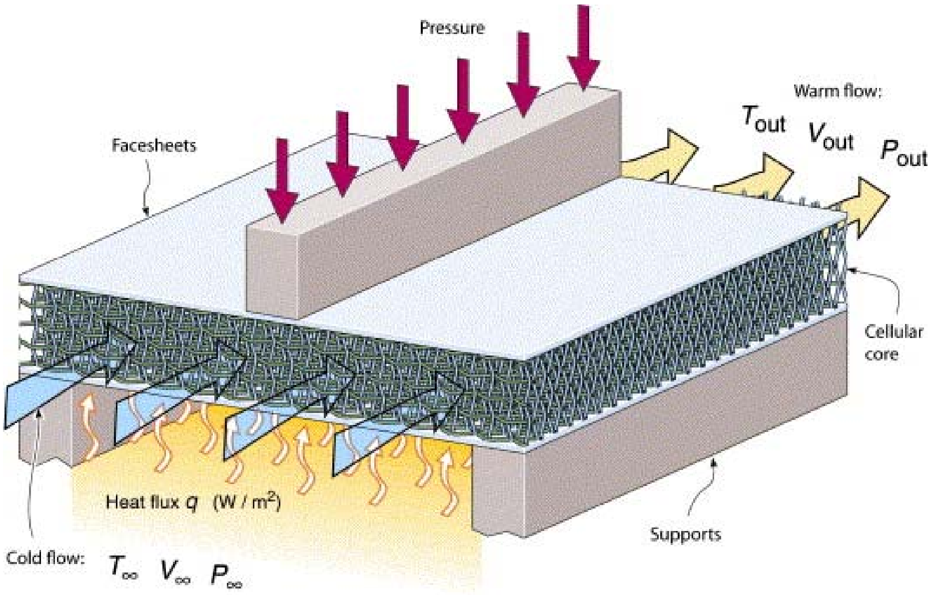


Fig. A3.2.16. The mesh unit, where sandwich construction with textile technology: (a) a transient liquid phase joins the wire mesh screen laminated at all points of contact; (b) face-sheets are added to the textile core.

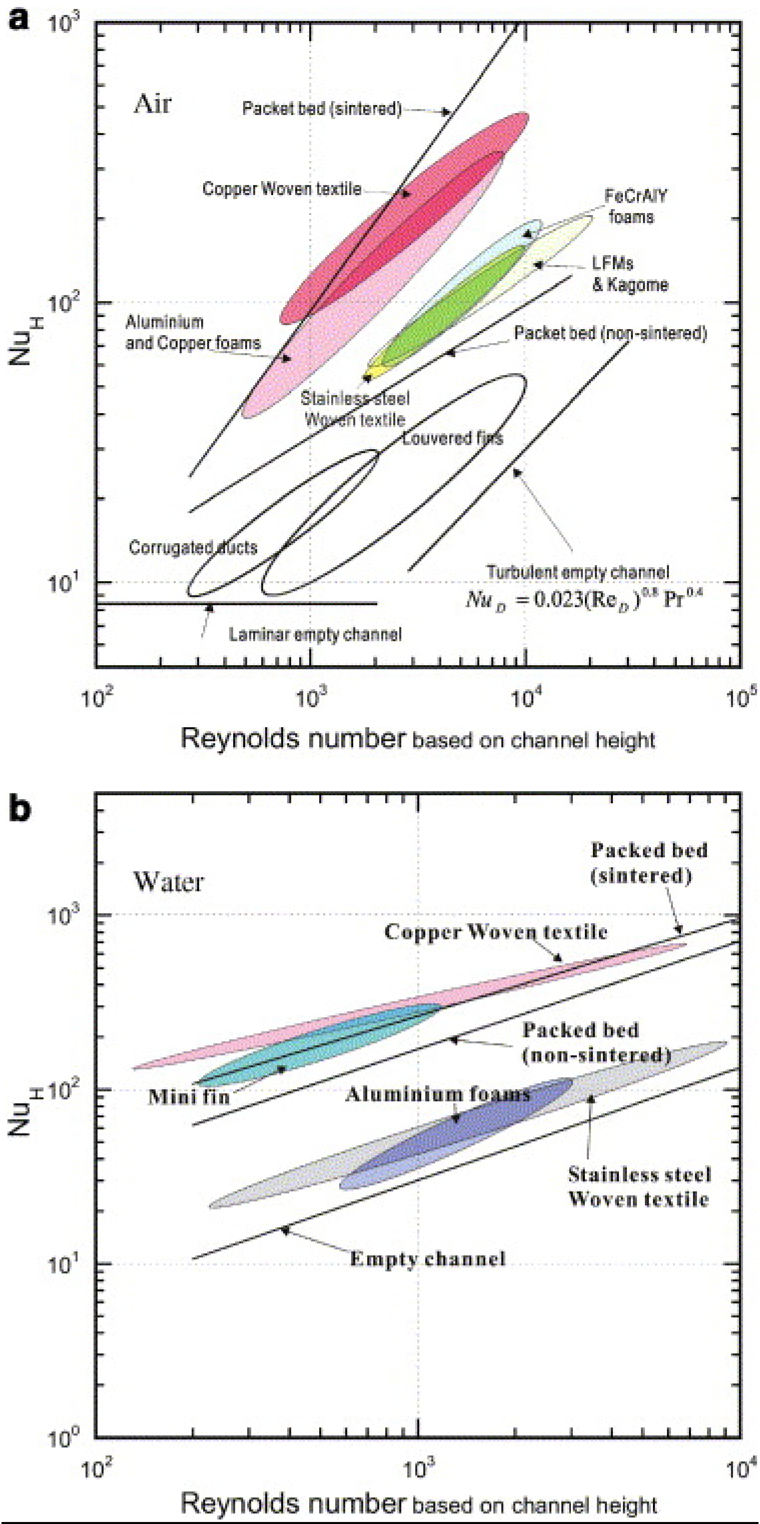


Fig. 3.2.17. The comparison of the mesh textile unit with other porous heat exchangers, such as sinters and metal foams, is illustrated here. The Nusselt number is based upon channel height (as for Re).

A3.2.3 Advantages of Compact Heat Exchangers

Compact designs can, in many applications including heat pumps, offer substantial advantages over more conventional exchanger designs, usually shell and tube exchangers.

The requirement in most heat transfer applications is to maximise the amount of heat transferred, subject to capital cost and pressure drop constraints. Both cost and heat transfer performance generally increase in proportion to exchange surface area, and the CHE permits large amounts of surface to be contained within a smaller volume than, for example, a 19mm tube diameter shell and tube exchanger.

A3.2.3.1 Generic Advantages of Compact Designs

The main benefits of using compact heat exchangers in the context of heat pumping equipment are:

- Improved heat exchanger thermal effectiveness.
- Closer approach temperatures.
- High heat transfer coefficients and transfer areas per exchanger volume.
- Smaller size.
- Energy savings.
- Reduced fluid inventory.
- Process intensification.

These technological advantages can be converted into reduced operational and capital costs, and conserve energy, compared to shell and tube units, and are discussed below.

A3.2.3.2 Improved Heat Exchanger Thermal Effectiveness

A major advantage of most compact designs is their greater efficiency or thermal effectiveness (E). The effectiveness of a heat exchanger can simply be expressed as the ratio of the actual heat transfer to the maximum possible heat transfer. The effectiveness is a function of the heat capacity of the fluid streams, the overall heat transfer coefficient and the area of heat transfer surface.

The benefits of improved heat exchanger efficiencies are described below. One advantage of a higher effectiveness - a closer approach temperature difference - is a beneficial effect on the relative position of the hot and cold composite curves utilised in a process integration analysis. This can lead to significant energy cost savings in process heating and cooling duties.

Heat transfer performance can be measured by assessing the efficiency or, more strictly, the 'effectiveness' of heat exchangers. Consider the case of two counter-current streams as depicted in Fig. A3.2.18.

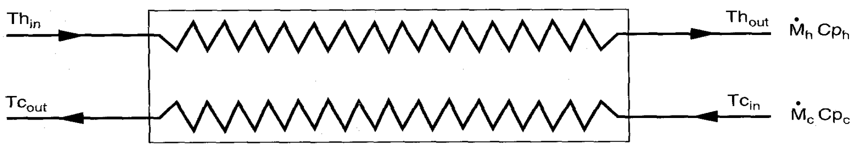


Fig. A3.2.18. Idealised Conditions for Two Counter-current Streams in a Heat Exchanger

Nomenclature for the theory below;

A	heat transfer surface area of heat exchanger	m^2
Cp	specific heat capacity	J/kg.K
E	heat exchanger effectiveness	-
LMTD	Log Mean Temperature Difference	K
$\dot{M}$	mass flow-rate	kg/s
$\dot{Q}$	heat load of heat exchanger	W
$\dot{Q}_{max}$	maximum possible heat load	W
R_f	fouling resistance	m^2.K/kW
R_K	$(\dot{M}Cp)_h/(\dot{M}Cp)_c$	-
T	stream temperature	K
ΔT	temperature difference	K
U	overall heat transfer coefficient	W/m^2K

Subscripts

av	average
c	cold
h	hot
in	in
m	mean
out	out

The theoretical maximum amount of heat that can be transferred in this simple configuration is given by:

$$\dot{Q}_{max} = (\dot{M}Cp)_{min}(T_{h,in} - T_{c,out}) \qquad (1)$$

where $(\dot{M}Cp)_{min} = \dot{M}_h Cp_h$ if $\dot{M}_h Cp_h < \dot{M}_c Cp_c$

$(\dot{M}Cp)_{min} = \dot{M}_c Cp_c$ if $\dot{M}_h Cp_h > \dot{M}_c Cp_c$

The effectiveness is a measure of heat transfer performance. The thermal effectiveness (E) may be defined as:

$$E = \dot{Q}_{act}/\dot{Q}_{max} \qquad (2)$$

where $\dot{Q}_{act}$ = actual amount of heat transferred

$\dot{Q}_{max}$ = theoretical maximum heat transfer

The actual amount of heat transferred ($\dot{Q}_{act}$), is given by:

$$\dot{Q}_{act} = \dot{M}_h Cp_h (T_{h,in} - T_{h,out}) \qquad (3)$$

$$\dot{Q}_{act} = \dot{M}_c Cp_c (T_{c,out} - T_{c,in})$$

Therefore E is given by the equation:

$$E = \frac{\dot{M}_h Cp_h (T_{h,in} - T_{h,out})}{(\dot{M}Cp)_{min}(T_{h,in} - T_{c,in})} \qquad (4)$$

$$E = \frac{\dot{M}_c Cp_c (T_{c,out} - T_{c,in})}{(\dot{M}Cp)_{min}(T_{h,in} - T_{c,in})}$$

If $\dot{M}_h Cp_h = \dot{M}_c Cp_c$, then

$$E = \frac{(T_{h,in} - T_{h,out})}{(T_{h,in} - T_{c,in})} \qquad (5)$$

$$E = \frac{(T_{c,out} - T_{c,in})}{(T_{h,in} - T_{c,in})}$$

For shell and tube heat exchangers, the thermal effectiveness (E) is typically 0.75. Values in excess of 0.95 are economically possible with compact heat exchangers - up to 0.98 for PCHEs and the Chart units.

A3.2.3.3 Closer Approach Temperatures

Approach temperature is an alternative measure of heat exchanger performance. Equation (6) shows that as the outlet temperature of the cold stream ($T_{c,out}$) approaches the inlet temperature of the hot stream $(T_{h,in})$, the thermal effectiveness (E) increases.

A shell and tube heat exchanger with an effectiveness of 0.75, heating a single phase fluid from 10°C with a hot source stream at 100°C, will give a cold stream outlet temperature of 77.5°C: that is, an approach temperature of 22.5°C.

A compact heat exchanger with an effectiveness of 0.95, used for the same application, would give a cold stream outlet temperature of 95.5°C: that is, an approach temperature of only 4.5°C.

$$\Delta T_{min} = (1 - E)(T_{h,in} - T_{c,in}) \qquad (6)$$

A3.2.3.4 Heat Transfer Coefficient and Area

Heat transfer can be predicted by the equation:

$$\dot{Q} = UA\Delta T \qquad (7)$$

where $\dot{Q}$ = heat transfer rate, W/m^2
U = overall heat transfer coefficient, W/m^2/K
A = heat transfer surface area, m^2
ΔT = temperature difference between the streams, K or °C

Increasing U and/or A will increase the heat transfer rate for given temperature conditions and will, therefore, improve effectiveness.

Compact heat exchangers have:

- High heat transfer coefficients due to the small hydraulic diameter of the flow passages.
- High heat transfer surface areas for a given volume of heat exchanger. Typically, compact exchanger area/volume ratios may be up to an order of magnitude greater than those of shell and tube exchangers depending on the exchanger type.

In many instances the high heat transfer coefficient is frequently achieved without excessive pressure drop.

An arrangement of shell and tube heat exchangers in series could provide high heat transfer areas and the effectiveness of such an assembly could thus be made as high as required. However, the greater area/volume ratio and high heat transfer coefficients of compact heat exchangers give a high effectiveness with a much smaller overall volume. A cost-effective heat exchanger with a high effectiveness can therefore be achieved with a compact design.

A3.2.3.5 Smaller Size

Obvious from the above discussion, another advantage of compact heat exchangers is their smaller physical characteristics for a given heat transfer duty compared to most shell and tube heat exchangers.

This has benefits that extend well beyond the heat exchanger itself, including, for example, reduced support structure and more convenient location (particularly when installed on a 'greenfield' site). Obviously in restricted space applications, reduced size and weight are important selection criteria. Also, the additional space needed to remove the tube bundle from the shell of a shell and tube unit should not be overlooked; the equivalent space required on a compact heat exchanger with a removable core is proportionally less.

Compact heat exchangers, particularly when their total installed cost is considered, tend to be significantly cheaper than their conventional counterparts. While this may not always be the case when low-cost metals can be used, the benefits of compact heat exchangers are much more apparent when the heat exchanger has to be made from an expensive material such as nickel or titanium. Here the cost per kilogram of raw material dominates the cost of the exchanger, and often adjacent ancillaries.

A3.2.3.6 Energy Savings

The ability of compact heat exchangers to operate with smaller driving temperature differences between streams means that it is possible to reduce the power requirements of plant items such as refrigeration compressors that were previously sized for the greater temperature differences required with shell and tube heat exchangers. The principal driver of IEA HP Annex 33 has been energy savings, coupled with size and cost reductions in heat pumping systems.

A3.2.3.7 Reduced Fluid Inventory

Compact heat exchangers operate with much lower fluid inventories (the 'hold-up') than many conventional heat exchangers.

The implications of lower fluid inventories include:

- Safer operating conditions when handling hazardous fluids – important for ammonia, for example.
- Reduced volume when handling highly valuable products – as the costs of refrigerants/working fluids rises, the reduced inventory in CHE-based systems becomes increasingly attractive.

It should be noted that the low hold-up of compact heat exchangers, compared to conventional designs, allows them to react quickly to changes in conditions. This should be taken into account when designing associated control plant and other upstream equipment, but in some process heat pumping applications a lower response time can have a positive effect on, for example, product quality.

A3.2.3.8 Process Intensification

Process intensification (15) has various definitions, but is generally associated with active (and in the case of many CHEs, passive) heat and mass transfer phenomena that allow one or two orders of magnitude reduction in the size of equipment.

Illustrated in Fig. A3.2.19, the motivation behind the design of this vapour compression unit was principally compactness. The heat pipe forms the central core of the unit, but rotation is employed in several other ways, with the intention of enhancing performance. As shown, the air conditioning unit spans the wall of a building, requiring a relatively small hole to connect the condenser section to the inside of the room. The reject heat from the cycle is dissipated by convection induced in the outside air by a rotating conductive fin, or, not shown, by a fan.[89] In the space to be conditioned, the liquid refrigerant flows into the hollow fan blades where it expands through orifices near the blade tips to fill them with cold vapour, which extracts heat from the room air. The warmed vapour enters the compressor, and then flows to the condenser – data are given in Gray, V.H. (1969).

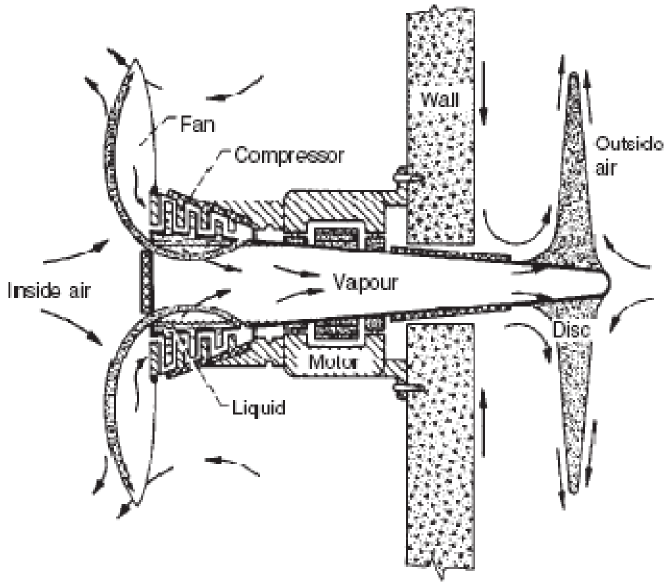


Fig. A3.2.19. The rotating air conditioning unit, based upon heat pipes.

The most radical (to date) use of process intensification in heat pumping equipment is the Rotex – more recently known as Rotartica – rotating absorption cycle heat pump/chiller/air conditioning unit. This is illustrated elsewhere in this document.

[89] One could envisage the rotating fin as being hollow, but not connected to the main vapour space. This could then act as another rotting heat pipe, in series with the main unit, to aid dissipation.

A3.2.4 Compact Heat Exchangers in Practice

In this Section a number of factors affecting, ultimately, the long-term satisfactory operation of compact heat exchangers are discussed. Having decided upon the type of heat exchanger to be used in a particular situation, (an exercise which will have already considered some aspects of fouling), the installation of the unit must be carried out, and guidance given to those involved in the operation and maintenance of the unit. Fouling, and to a lesser extent corrosion, and their minimisation remain key priorities during the life of many heat exchangers, particularly those in arduous process industry duties, and these aspects should never be neglected during the life of the unit.

It should of course be noted that the majority of the fouling in heat exchangers in heat pumps, particularly vapour compression types, will take place n the process side(s) of the heat exchangers and the working fluid itself should remain relatively clean. Bear in mind that wear can lead to metallic particles from bearings, etc., and the oil used as a lubricant, if in the working fluid, can affect the heat transfer characteristics.

The discussion is targeted towards process applications, although some of the principles are relevant to all applications of compact heat exchangers. Domestic heat pumps will suffer from air-side fouling, possibly fouling due to dirty water, and of course ground coils (not really 'compact') do have a 'fouling resistance' associated with their contact with the soil/earth.

The factors to be considered are set out in the following order:

- Specification/purchase
- Installation
- Commissioning
- Operation
- Maintenance:
  - General factors
  - Fouling & corrosion

Note that the CHE is discussed in isolation in most of this Section, rather than as a heat pump component. It is often compared with a shell and tube heat exchanger – in many instances of their use, the CHE may well be replacing an earlier plant/process design that incorporated the larger heat exchanger variant.

A3.2.4.1 Specification/Purchase

The steps the process engineer, business manager, technical director etc. should take in order to get full advantage from the information given above on the types of CHE available are important to the ultimate success of any heat pumping equipment installation. It is recommended that a check list, an example of which is given below, be followed to ensure that important considerations are not neglected. There are also a lot of organisations which can assist in the many stages from project conception to commissioning

This check list is arranged as a logical series of questions which should assist the process engineer or other personnel involved in addressing CHEs (as a potential user) to make a preliminary assessment of their feasibility in a process heat pump, specifying the unit and finding suppliers able to quote..

This applies principally to someone looking at replacing an existing plant/process with one using CHEs. For 'green field' sites and new plant, the questions concerning the existing plant/process may be less relevant, but other questions below will need addressing.

QUESTION	YES/NO/EXP-LANATION
Is my plant/process currently running under optimum conditions?	
If not, why not?	
Is it cost-effective to reach optimum running conditions with my current plant/process?	
If my plant/process is optimised, is there still a case for looking at incorporating CHEs?	
Do I know all the current operating parameters on my plant/process?	
Can I obtain these data?	
Do I have information on the operating parameters of any replacement plant?	
Will any replacement plant/process have significantly different requirements to the existing plant (throughput, be continuous or intermittent (batch), product type, operating temperatures, pressures, quality control, fouling etc.)?	
Are these requirements realistic for my existing plant, or a plant based upon ‘standard’ heat exchanger technology?	
If they are, why do I want to study a system that might incorporate CHEs?	
If they are not realistic for a conventional plant, how will a more compact unit help?	
Will a radical change of design/heat exchanger type affect my product?	
Is my knowledge of heat & mass transfer sufficient to answer the above?	
If not, are there others in my organisation who could assist me?	
Should I consider employing outside consultants?	
Do I have sufficient time and resources within my own team?	
Do my timescales and potential budget allow for any R&D work?	
If not, am I limited to equipment that is already fully developed?	
If this is the case, is such equipment available?	
Do I then have the information to carry out:	
A risk assessment (health & safety)?	
A technical feasibility study?	
A rigorous financial appraisal?	
Have I considered the availability of grants?	
Can I now make a comprehensive case for examining a ‘compact’ solution to my plant/process?	

A3.2.4.2 Installation

Many of the features of compact heat exchangers which affect installation will no doubt have been considered during the specification stage. In some cases there may be a need for instrumentation which monitors the performance of the compact heat exchanger continuously, because of its higher sensitivity to some changes in conditions than, say, standard shell and tube units. This will have been considered before installation, but correct location is of course important. In the case of compact heat exchangers, the provision of on-line pressure drop monitoring will provide a useful guide to the build up of fouling, which will of course be reflected in an increase in pressure drop across the unit.

Where fouling is of particular concern, insofar as it could affect the time the heat exchanger spends on line, the use of two identical heat exchangers in parallel should be considered. Then, if one becomes partially blocked, the other unit can be brought into service during cleaning of the fouled unit, without significant process interruption. It must be remembered that the cost of production lost due to the need to shut a stream while a heat exchanger is cleaned, or worse, can greatly outweigh the extra cost of a second heat exchanger in a parallel bypass stream. Such a precaution may be appropriate for some important process heat pumping applications, but of course in most instances the working fluid side would need to be replicated too.

There are substantial differences between installations where an old heat exchanger is being replaced by a new one, or a new one added to and existing plant – retrofitting – than where a completely new plant is being constructed with compact heat exchangers as part of this new installation. An aspect of retrofitting sometimes overlooked is the need to either replace or at least fully clean pipework upstream of the new heat exchanger. (Of course, owing to size differences, the pipes immediately local to the new heat exchanger will need replacing). Compact units, being susceptible to particulates in the fluid stream, require much better control of upstream conditions than a conventional shell and tube heat exchanger, or other types with large passages. Even in an all-new plant, debris may get into pipework during plant construction, which could affect subsequent compact heat exchanger performance. ALPEMA[90] makes specific recommendations for the start up of brazed plate-fin heat exchangers, concerning purging and cool-down (where used in cryogenic duties – the first real large scale application of CHEs in heat pumping equipment), to clean the unit, prevent freezing, and preserve its integrity.

The installer who is used to making substantial provision of space for bundle removal in conventional shell and tube heat exchanger installations will find that on an all-new installation, the amount of space required for removal of the core during maintenance will be substantially less[91]. Lifting gear needs will also be different, and it will have become evident during the design and specification of the installation that the options

[90] ALPEMA is The Brazed Aluminium Plate-Fin Heat Exchanger Manufacturers' Association (see ref. 14).

[91] Note that not all compact heat exchangers have cores which can be removed for mechanical cleaning, for example. Where brazing or diffusion bonding is used, alternatives should be considered – see section on fouling.

for location of the heat exchanger will have increased owing in part to its lower weight for a given duty.
Installation is frequently carried out while other activities are under way in the locality. It may seem obvious, but compact heat exchangers need to be protected from adverse conditions during installation and commissioning. Ingress of dust could be disastrous, and the introduction of any fluid which is not that anticipated within process operation could also create problems, such as corrosion. Obviously if the plant is located on-shore, but near the sea, a salt-laden atmosphere would be damaging to an aluminium plate-fin heat exchanger, for example.

Bott, in his *Fouling Notebook*, (ref. 1) highlights problems which might occur if hydraulic testing (carried out on site prior to commissioning) is not properly implemented. Some corrosion control by inhibitors, for example, is recommended immediately following the test, and good quality water should of course be used for the hydraulic test itself. It may be that the inhibitor will be left in the heat exchanger for some time, prior to final installation and commissioning.

Of course, as pointed out by Bott, the heat exchanger should be thoroughly checked on its arrival at the site, to make sure that all components and the whole assembly meet the specification requested.

Heat Pumps: There is no particular reason why a large heat pump installation that currently uses shell and tube (or other non-compact) heat exchangers should not have these replaced by compact units, although the benefit that might result will relate solely to the COP – this should be raised due to closer approach temperatures being possible.

A3.3.4.3 Commissioning

It is particularly important to be careful when commissioning a compact heat exchanger installation. As mentioned earlier, if the installation is a retrofitted one, all pipework accessing the heat exchanger(s) should be fully cleaned. Heat exchangers with small passages, particularly those where access for cleaning is difficult, are particularly sensitive to operation outside their design envelope, in terms of temperature and velocity of the fluid(s) passing through the heat exchanger. High temperature can change the nature of the fluid at the heat exchanger wall/fluid interface, leading to fouling which would not, under normal circumstances, be anticipated, (see later in this Chapter). If the velocity of the fluid passing thought the unit is reduced during commissioning, well below that to prevent deposition of particulates on surfaces, for example, accelerated fouling can occur and some deposits which were not anticipated when the heat exchanger was specified may be found difficult to remove. Beyond the commissioning stage, of course, similar difficulties might arise during start up of the whole process plant, where conditions through the heat exchanger may be substantially different to those normally encountered during steady state operation. (An analogy may be made with a car engine clean-up system, where the cold engine can affect catalyst performance due to increased throughput of unburned hydrocarbons, for example).

Many compact heat exchanger types are still relatively new in some market sectors. It is therefore essential for even large user companies to make full use of the experience of the heat exchanger vendor gained during installation and commissioning at other sites. Even if the vendor was not directly involved, the contractor will no doubt have provided some feedback, particularly if any problems had been encountered. ‘Learning from the experience of other’ (ref. 2) is a key recommendation for anyone contemplating using heat exchanger equipment.

A3.2.4.4 Operation

The operation of compact heat exchangers requires a degree of control over conditions which tends to be greater than that exercised with conventional shell and tube heat exchangers. This arises from the faster transient conditions consequent on the lower fluid inventory of compacts. Some may find that this limits the flexibility of the overall installation, but as long as the user is aware of the operating parameters of the process streams when he is making the selection of the heat exchanger and associated plant, this should not be a problem. Indeed, some types of compact heat exchanger are specifically designed to be expanded or reduced in size to cope with differing process conditions over the life of a plant. The gasketed plate heat exchanger is one such example. Tight design can affect the performance of the heat exchanger where a reduction in stream velocity might occur, for example. The effect this might have has already been highlighted in the section on commissioning above.

A perhaps obvious aspect of compact heat exchanger operation is the need to ensure that operators have the necessary training to address any problems which might arise, and are conversant with cleaning procedures, the handling needs of particular types of surface and/or materials, and the degree of tightness when installing gaskets, for example.

If one is concerned about possible failure of any heat exchanger, due to pressure rupture, excessive temperatures affecting gasket material, or fouling leading to complete blockage, the main way to ensure that damage limitation is fully implemented is to *monitor*, preferably continuously, critical factors such as stream temperatures and pressure drop across the sides (there may be more than two streams) of the heat exchanger.

An aspect of operation which cannot readily be anticipated is a change in the nature of the product which is being processed in the plant of which the compact heat exchangers are essential components. While it may be acceptable, albeit with some change in overall heat transfer coefficients, to change the material going through a shell and tube heat exchanger having tubes of 19 mm internal diameter, to do the same with a printed circuit unit with an hydraulic diameter of channels of 2-3 mm is to increase the degree of risk substantially. As with all of the advice given, it is assumed that reputable plant operators will be aware of such hazards and avoid them. There will, however, be those who remain ignorant, and one can in such cases only point out potential sources of erroneous judgement!

A3.2.4.5 Maintenance

In this Section, the maintenance of compact heat exchangers is discussed. Because fouling (and corrosion) are factors of compact units (apart from their small size) most regularly mentioned whenever these heat exchanger types are considered, it is important to discuss in some detail the types of fouling mechanisms, some of which are directly associated with corrosion, which can occur. Even more significant is a description of the techniques used to overcome the several forms of fouling which can occur in ‘conventional’ or compact heat exchangers.

Thus, following a general discussion of maintenance aspects of compact units, a sub-section is devoted to a description of fouling mechanisms, and solutions for overcoming fouling, in particular in compacts.

Maintenance – General Factors

In many cases, maintenance of compact heat exchangers can be easier than that of larger conventional shell and tube units. As an example, consider the volume differences between a shell and tube unit with provision for removing the whole tube bundle, and a welded plate heat exchanger of the same duty. In the former case, a distance equivalent to the whole heat exchanger length should be provided downstream of one of the end covers for bundle removal, while for the plate unit a distance, either vertically or horizontally, of about 30% of that needed for the shell and tube unit is necessary for an identical operation. While on-shore such space considerations may not be critical, on an off-shore platform the cost of platform space is such that reduced space needs are dominant in equipment selection.

The reduced volume of the compact core also has other advantages – it can be readily shipped off site for maintenance and/or cleaning, for example. An important aspect of maintenance sometimes neglected by users it associated with gaskets. Where these are used, for example between plates, uniform gasket compression should be ensured. In recent years, the original equipment manufacturers have also become concerned about the use of ‘cheaper’ gaskets which are not approved by the heat exchanger supplier. As in many other sectors of industry, ‘bogus’ spare parts are a serious problem affecting service reliability, and companies should never contemplate sacrificing the perhaps higher cost of spares from a reputable company in order to seemingly save capital expenditure. The pitfalls of doing this are obvious.

Maintenance of the heat exchanger is useless if carried out in isolation. A heat exchanger is but one component of a process stream, and associated with the heat exchanger will be controls, valves, filters and sensors, such as pressure transducers and thermocouples. Checking of the calibration of equipment which is used to monitor the performance of the heat exchanger is essential – this goes without saying – but it is also vitally important, as mentioned in the section on ‘operation’, to ensure that such equipment is used during heat exchanger running. If monitoring equipment is in place, but is neglected, no-one other than the site person responsible can be blamed for the consequences of failure. The cleaning of filters is a regular maintenance operation, not to be forgotten. If the pressure drop across the heat exchanger is being monitored (as it should be), the location of the pressure tappings may be such that the filters are included in the pressure drop being measured; if they are outside the region between tappings, a second pressure drop monitoring system covering the upstream filters, could be usefully installed.

Preventative maintenance is designed, of course, to reduce the chances of breakdown in any of the components making up a process stream. However, contingency plans should always be present in case failure of the whole, or part, of a heat exchanger does occur while on line. The remedial action may involve directing the process stream(s) through an adjacent second heat exchanger, (installed for such an eventuality), or blocking off the offending layers in the compact heat exchanger.

Maintenance - Fouling and Corrosion

Fouling, while perceived as one of the limitations affecting increased use of CHEs, can be addressed in a number of ways to minimise its effect on the system, and on the CHE in particular, (ref. 3). In this sub-section the mechanisms of fouling which may occur in CHEs (and of course in other types of heat exchanger) are described, and solutions put forward. There are also a number of heat exchanger types, not all of them CHEs, which are designed specifically to handle fouled process streams. These are briefly discussed at the end of this sub-section. For those wanting a more detailed treatment of fouling in heat exchangers, a recent text by Bott (ref. 4) is recommended.

The principal types of fouling are:

- Crystallisation or precipitation fouling
- Particulate fouling or silting
- Biological fouling
- Corrosion fouling
- Chemical reaction fouling
- Freezing or solidification fouling

Crystallisation or precipitation fouling.

This occurs when a solute in the fluid stream is precipitated and crystals are formed, either directly on the heat transfer surface, or in the fluid, to be subsequently deposited on the surface. When the fluid concerned is water, and calcium or magnesium salts are deposited, the mechanism is frequently referred to as scaling. Waxes can also precipitate out as temperatures of fluids are reduced through the heat exchanger.

Scaling is a function of wall temperature and possibly also of kinetics. The wall temperature profile through most compact heat exchangers is well-defined relative to that in shell and tube units, and it is suggested (ref. 5) that a less conservative view of scaling should be taken with CHEs, and any permissible increase in the outlet temperature of sea water, when used as a coolant, would be welcomed because of the associated reduction it brings in pump sizes, etc.

It is important to note that there are a number of compounds, known as inverse solubility salts, which exhibit a reduction in solubility with increasing temperature. These include calcium carbonate and magnesium silicate. In these cases, identification of the *highest* cooling water temperature likely to occur in a heat exchanger with narrow channels is therefore important for determining the water treatment strategy.

In general, crystallisation or precipitation fouling is avoided either by pre-treatment of the fluid stream (e.g. by acid addition to cooling water to remove carbonate) or by the continuous addition of chemicals that reduce or eliminate deposit formation. There are positive indications that the addition of appropriate particles into the fluid stream - seeding or germination - can be effective in minimising deposition on heat transfer surfaces, (ref. 6). Additives can be used to minimise waxing, but environmental pressures are increasingly mitigating against excessive use of chemical treatments. Mechanical cleaning methods such as high pressure lances are unlikely to be usable in compact heat exchangers because of the small passage sizes and their sometimes tortuous flow paths. These features make it difficult to clean any passages which are completely blocked. Note that care should be taken to ensure that cleaning chemicals are compatible with materials of construction, including brazes, gaskets etc.

Recently, electromagnetic descaling technology – sometimes called physical water conditioning by the equipment suppliers, which is extensively promoted for scale prevention in water-carrying pipes to inhibit calcium carbonate deposition on the surfaces, has been examined as a protective measure to inhibit heat exchanger fouling. The work indicates that the method can be effective for both inverse solubility salts and those exhibiting normal solubility characteristics, such as silica. Most of the scale inhibitors which are marketed extensively for pipe scale control use a controlled electromagnetic field through which the water stream passes, affecting the size and structure of mineral crystals within the stream. This reduces their ability to adhere to each other, and to the pipe or heat exchanger surfaces, thus reducing the propensity to scale.

Although patents were taken out on the concept as early as 1895, companies such as Hydromag (UK) state that a considerable amount of time has been expended on achieving the correct field strength in combination with positioning of the lines of the magnetic field with regard to the direction of the water stream. Although more strictly categorised as a device for the prevention of scaling, rather than one for removing it once it builds up, the system has to date been of particular value in calorifiers and humidifiers. It has also been used effectively in cooling/'clean in place'/washdown systems in, for example, breweries. The units are largely maintenance-free, and can be cleaned by polarity reversal. Throughputs, in terms of water flow rate, are typically 0.1 – 30 l/s for a single unit.

Data specific to the used of such scale reduction techniques on compact heat exchangers are sparse, but tests have been reported arising out of a research project in the USA (ref. 7). Here it was proposed that the electromagnetic descaling unit, located upstream of the plate heat exchanger, led to agitation of the charged mineral (calcium and bicarbonate) ions, resulting in them colliding with one-another and thereafter precipitating. Once the dissolved ions are converted to insoluble crystals, the level of super-saturation of the water greatly decreases, preventing new scale deposits forming on the surfaces of the heat exchanger. The cited paper reports on field trial on an Alfa Laval heat exchanger with titanium plates. A 7% brine solution was cooled by river water, and scaling, which occurred on both sides of the heat exchanger, had resulted in a 10% reduction in the 'U' value every week. This necessitated regular acid cleaning. However, the electromagnetic descaling system eliminated degradation in performance during the trial period of 16 weeks.

Of potential use in compact heat exchangers is a new method of minimising scaling. This is based on surface treatment, similar to that used to create surfaces for drop-wise condensation promotion. Research at the University of Surrey in the UK has shown that surface modifications of a metallurgical nature, such as ion implantation and magnetron sputtering can lead to a greater than 70% reduction in the growth of crystalline deposits and an even greater reduction in bacteria adhesion. The treatment should be readily applicable to the plates of compact heat exchangers before they are assembled, and the surface treatment is also believed to offer resistance to corrosion and erosion, (ref. 13).

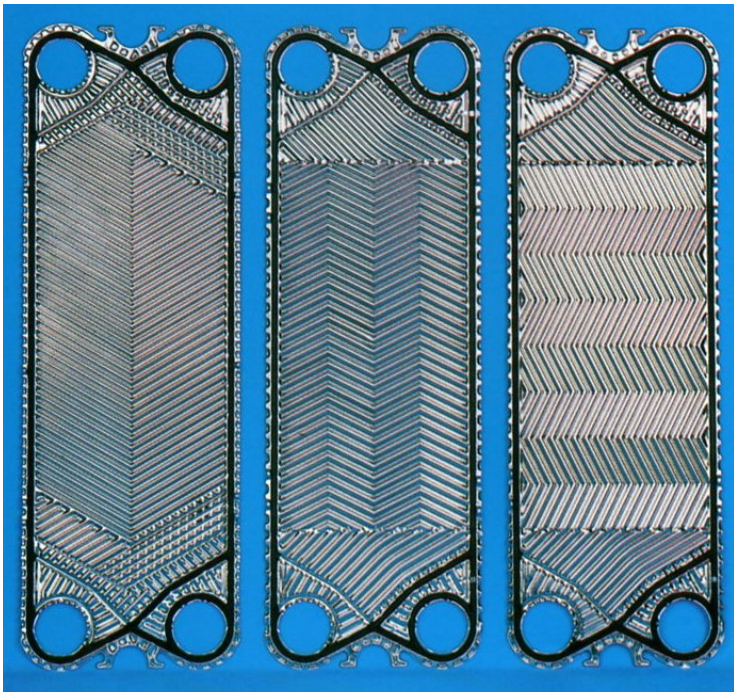


Fig. A3.2.20. The ability to open up gasketed plate heat exchangers allows excellent cleaning. The use of CIP – cleaning in place – is also regularly used in the food processing sectors.

Particulate fouling (silting)

Silting is the deposition of solid particles on a surface – the phrase ‘silted up’ is commonly used to describe a pipe or channel which has a thick covering of particles at one or more locations. Small particles can be harder to remove than larger ones, as the forces holding them together and to the surface can be greater. Much has been written about electrophoretic and thermophoretic effects in attracting small particles to surfaces, and these forces do play a role in many particulate fouling events. However, these, and other temperature-driven effects are less in compact heat exchangers because of the (generally) lower temperature differences.

Particulates by themselves are not too difficult to remove. Problems arise when the particulates are combined with other fouling mechanisms, in particular tar formation. Then removal needs more drastic measures, such as chemical/solvent treatment.

Pure particulate fouling can be reduced by using high fluid velocities, except in cases where an adhesive component may be mixed with the particles. Filtration of particles can be applied and for CHEs a suitable strainer can be installed upstream. Severe pressure pulses, such as obtained by rupture of a bursting disc, can also remove fouling. For gasketed PHEs, it is suggested (ref. 8) that steel/stainless steel brushes be avoided when cleaning opened exchanger plates, nylon brushes being recommended, with water. For greasy deposits, kerosene can be used, together with brushing. In welded plate heat exchangers, vacuum cleaning of the inlet headers is recommended.

For those with an interest in the detail of fouling mechanisms, Karabelas *et al* (ref. 9) have studied this aspect with a view to improving design standards for novel and conventional heat exchangers. Fouling data are given for plate heat exchangers with various corrugation angles and particles of mean diameter 5 mm. The results showed that fouling is adhesion-controlled and that the maximum fouling resistance is almost one order of magnitude less than the TEMA[92] recommendations. The corrugated surface geometry common to such compact heat exchanger plates assisted particle detachment from the surfaces, the geometry leading to tangential hydrodynamic forces. Data obtained for specific particle sizes revealed a distribution of adhesive strengths which are strongly influenced by the acidity (pH) level.

Biological fouling

Biological fouling is caused by the deposition and growth of organisms on surfaces. Bacteria are the organisms most likely to cause problems in CHEs, and their presence can also assist corrosion by, for example, reducing sulphate to hydrogen sulphide, which is corrosive to common stainless steels and many other materials.

The best control method, especially for closed systems, is treatment with biocides. Non-oxidising biocides are normally alternated to prevent the development of resistance, and may kill the bacteria but not remove the biofilm. Some biocides have detergent properties which can disrupt the film. Chlorine and ozone are oxidising biocides, which kill the bacteria and oxidise the film. High concentrations may be necessary for them to be fully effective, but the smaller volume and fluid inventory in a CHE, compared to shell and tube heat exchangers, helps to minimise the quantities needed. The chemical diffuses to the biofilm, and in narrow channels this should be relatively rapid.

Applicable not to just biological fouling is an innovative method for fouling inhibition patented by workers in the UK. The company Applied Coolant Technologies called this the 'micro-mechanical approach to fouling'. The concept uses micro-fibres, of an insoluble polymer, which are added to a circulating cooling water stream. In the first experiments on a closed system – a car engine cooling water circuit – it was shown that fouling inside the engine and radiator was prevented and cavitation was also reduced. Plans are in hand at the time of writing to use the method in a number of heat exchanger types. Other reports (ref. 10) suggest that the addition of wood pulp fibres can also have a dramatic effect on reducing scale deposits. As an example, the

[92] TEMA is the Tubular Heat Exchanger Manufacturers' Association

addition of 0.05% wood pulp fibres to synthetic hard water almost completely eliminated the formation of calcium sulphate deposits.

At this time, more rigorous analyses are required of the effect of fibre addition, and the type of fibre, on the fouling mechanisms, both for scaling and biological fouling, but the data to data shows that a promising technique may be evolving.

Another recent approach to biological fouling minimisation is to use tube inserts, of the type made by Cal Gavin Ltd., the 'HiTRAN' element. While data on the trials are not yet published, there seems to be no reason to doubt that benefits will accrue to the use of these tube inserts, although of course in many applications the tubes will be somewhat larger than those associated with compact heat exchangers. They have been used in tubes of 3 mm bore, however.

Corrosion fouling

Chemical reactions involving the heat transfer surface, or the carrying of corrosion products from other parts of the system, to be left on the heat transfer surface, lead to corrosion fouling. The formation of deposits can by itself lead to corrosion underneath them, for example as the result of formation of electrolytic oxygen concentration cells.

To avoid this type of fouling, construction materials which are resistant to corrosion should be selected. This is routinely done, of course, by the heat exchanger vendors, but there always remains the possibility that the user may change process streams passing through the heat exchanger. Where polymers may be appropriate to counter corrosion, a recent US Gas Research Institute report (ref. 11) gives much background on types of polymer, operating limits etc. Alternatively, inhibitors can be used. Cathodic protection can lead to cathodic scales being formed if hard water/brines are the flow streams. Several types of compact heat exchanger have no dissimilar metals within them, thus making the likelihood of corrosion attack more predictable. Nevertheless, if there is a possibility of a stainless steel unit being stored for a period in a salt air environment, the surfaces should be protected to avoid stress corrosion cracking, (ref. 12).

It should be noted that corrosion products could be significant in fouling whilst not endangering the exchanger from the point of view of leaks or pressure containment.

Chemical reaction fouling

Chemical reaction fouling arises from reactions between constituents in the process fluid, leaving viscous or solid layers on the heat transfer surface. The latter is not involved in the reaction. Polymerisation reactions are commonly associated with this form of fouling, and if the deposit turns from a tar to a hard coke, removal is very difficult. Recently, a new type of chemical reaction fouling will need to be examined - in this case in heat exchanger-reactors. Comparatively little quantitative data exist to date, but it is evident that if by-products of a reaction are produced within such units, or if the reaction is not closely controlled, the implications for fouling would be important. The converse is also potentially possible, whereby a reaction (primary or secondary) may take place which would involve and remove a product or by-product which would otherwise foul the exchanger.

For heat exchangers alone, careful control of fluid and heat transfer surface temperatures, and reductions in residence time at the higher temperatures which enhance this type of fouling, should minimise it. A number of the benefits of compact heat exchangers described in this book will help to control chemical reaction fouling in compact heat exchangers, where it may not be so easy in larger channel systems, because of the inherently good temperature control and low residence times.

Freezing or solidification fouling

If the temperature of a fluid passing through a heat exchanger becomes too low, the fluid can solidify at the heat transfer surface.

Control of this form of fouling is relatively easy, and only requires a rise in temperature to melt any deposits. In compact heat exchangers, the generally lower metal mass and low quantities of fluid make this quite a rapid process.

The type of fouling depends in certain cases on whether the stream being studied is the 'hot' or 'cold' one, and also upon the direction of heat flow. Particulate fouling of course is not necessarily influenced by temperature, but obviously freezing or solidification fouling tends to occur on the side from which excessive heat is being removed. This may be the hot or cold stream, depending upon the solidification temperature of the process fluid.

Scaling is a difficult area – because, as mentioned above, of the presence of some inverse solubility salts, which precipitate out as the water becomes warmer. Other crystals of course precipitate on cooling. Biological growth tends to accelerate, to a certain limit, with temperature.

Corrosion fouling is normally associated with increasing temperature of a process stream, the reaction proceeding more rapidly as the fluid becomes warmer. However, those who maintain condensing economisers (normally not classed as compact heat exchangers) will advise caution when shutting down the plant. If an acid condensate – as occurs in such economisers on the gas side) - is left on the surfaces of the heat exchanger during a shut down period, although the temperature may be atmospheric ambient, some nice pockets of corrosion will be found on inspection some days or weeks later!

Chemical reaction fouling tends to be associated with increasing temperatures in the stream receiving the heat.

Heat Exchangers Designed to Handle Fouling

There are many heat exchangers which are designed specifically to handle fouled streams. The spiral heat exchanger (not common in heat pumps) is perhaps the main type of compact unit which is marketed specifically for streams such as those carrying sludge, followed by the plate and frame unit, which has a versatility in fouling duties difficult to rival by other ‘mainsteam’ compacts.

The ability of a heat exchanger to handle particulates is often a function of the gap available for passage through the unit of such particulates. Manufacturers of welded (and other) plate heat exchangers offer ‘wide gap’ variants which are specifically designed to handle fouled steams on one or more sides. Some have removable headers, as well as wide plate gaps, to allow ready access to the core for cleaning. Typical applications would be in the sugar industry (evaporators and/or MVR) and as a top condenser on a distillation column – both areas where open or closed cycle heat pumps might be contemplated.

Applications of Compact Heat Exchangers and Fouling Possibilities

The types of compact heat exchangers used in various sectors of industry do vary, depending upon the sector experience and degree of confidence. To a considerable degree the confidence is based upon a knowledge of fouling. In the Table below, the compact heat exchangers most commonly used in a variety of sectors, and two generic unit operations (refrigeration and prime movers) are identified, and comments made on the types of fouling likely to be encountered.

Table A3.2.5. Comments on Fouling in a Number of Sectors Commonly Using Compact Heat Exchangers. (Note that the 'process' side of the heat exchanger is under discussion here – not the side through which the refrigerant/working fluid flows).

Sector/Application	Type of Heat Exchanger	Comments on Fouling
Chemicals & petrochemicals	Plate & frame heat exchanger Brazed plate heat exchanger Welded plate heat exchanger Spiral heat exchanger Plate-fin heat exchanger Printed circuit heat exchanger Compact shell & tube heat exchanger Compact types retaining a shell	Because of the great variety in terms of process streams within this sector, each application and heat exchanger type should be examined independently. The plate & frame, spiral and compacts retaining a shell, such as the plate & shell hx, are more likely to be used where fouling is anticipated within the exchanger. Not all of the heat exchangers listed are likely to be used in open or closed cycle heat pumps in this sector, but the environment can be highly fouling, except where MVR may be used in distillation/evaporation processes.
Cryogenics	Plate-fin heat exchanger Printed circuit heat exchanger	One characteristic, apart from the low temperature operation, which sets cryogenic applications apart from most others is that the streams are likely to be clean. Impurities in gases can occur, for example mercury, but this impinges more on materials selection than fouling.

Food & drink	Plate & frame heat exchanger Welded plate heat exchanger Compact types retaining a shell	The processing of food and drink is a vast application area for some types of compact heat exchangers. Cleanliness is of course critical, and the ability to use heat exchangers which can be cleaned, and which are made using stainless steel, tends to limit the choice. The plate & frame unit is the most commonly used, because it is easy to dismantle, is flexible, and is well known in the sector. Crystalline, biological fouling and silting are common types of fouling associated with the sector. This is seen as an important growth area for heat pumps – both open and closed cycle. Cleanliness and the 'clean in place' philosophy need to be studied when looking at heat exchanger selection.
Paper & board	Plate & frame heat exchanger Spiral heat exchanger	Particulates, specifically fibres, are the principal contaminant in this sector, which is not a major user of compacts. However there may be uses in paper drying and in heat recovery/upgrading from black liquor and waste water.

Textiles & fabric care	Plate & frame heat exchanger Spiral heat exchanger	The textile industry produces large quantities of warm effluent, contaminated with dyes and fibres. It is therefore important to use compacts which can be opened up for cleaning. The sector also uses specially-designed heat exchangers with rotating surfaces for such effluents, mainly for recovering heat in liquid-liquid duties. This can make the selection of CHEs for heat pumps difficult, as the working fluid side of the heat exchanger may be totally different to the effluent side.
Refrigeration	Plate & frame heat exchanger Brazed plate heat exchanger Plate-fin heat exchanger Printed circuit heat exchanger	On the refrigerant side, fouling is unlikely. However, oil can be carried around the circuit from a lubricated compressor, and the presence of this could affect compact heat exchanger behaviour. On the secondary side, the fluid is determined by the cooling requirement, and any form of fouling might occur. Thus brazed plate heat exchangers would be selected with care if properties of the secondary fluid were unknown. Now we are much more interested in using ‘natural’ refrigerants, such as ammonia, hydrocarbons, CO_2 and air. Ammonia is toxic, hydrocarbons are flammable and CO_2 may be operating at or around its critical point, with relatively high pressures. All of these features make compact heat exchangers, in conjunction with other ‘compact’ units to reduce fluid inventories, desirable. However, while the refrigerant will be clean, the fluid on the secondary circuit may not be.

A3.2.5 What Next for Compacts in Heat Pumps?

We have briefly mentioned Rotex above, where CHEs are used as the solution heat exchangers (originally purpose-designed and built by SWEP.

One concept (Fig. A3.2.21) proposed by Dr. Peter Kew at Heriot-Watt University, Edinburgh (Reay et al. 1999) was a water-water heat pump employing a rotary vane compressor embedded in a printed circuit-type heat exchanger. The unit shown was designed to operate using R134a with evaporating and condensing temperatures of –4°C and 60°C respectively, an output of 3 kW and a COP of 3. The unit (excluding water-side headers, water pumps and compressor motor) would occupy a block 25 mm x 700 mm x 200 mm, or 0.0035 m^3. Interestingly, the unit here has a similar duty per unit volume to that of an absorption cycle unit (ammonia/water) that was investigated by Heriot-Watt University and Absotech in Germany based upon the Hesselgreaves compact heat exchanger surface, (Hesselgreaves, 1997). This design had a 100 kW cooling duty using ammonia and water working fluid pair, and a core size of 1000 mm x 550 mm x 200 mm, or 0.11 m^3, (Reay et al. 1998).

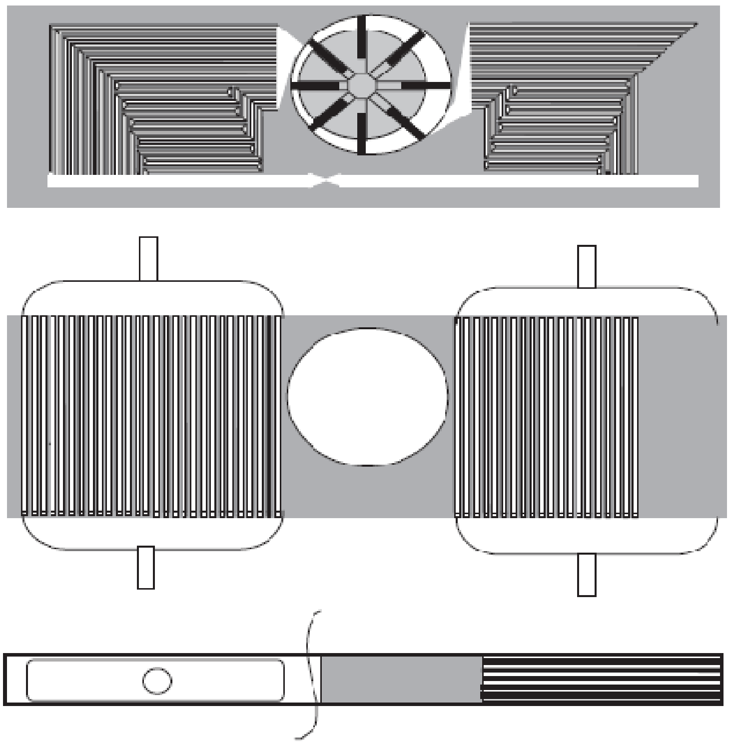


Fig. A3.2.21. A heat pump, based on PCHE-type structures but incorporating the compressor within the 'heat exchangers'.

Since the proposals reported in 1999, others have adopted similar strategies in order to 'compress' heat pumping/refrigeration cycles. The ARCTIC project at the University of California is directed at chip cooling in microelectronics, and necessarily adopts a rotary compressor (as selected by Kew at Heriot-Watt in 1998 – see above), in this case a Wankel type that is projected to give a compression ratio of 4.7:1, (Heppner et al. 2007).

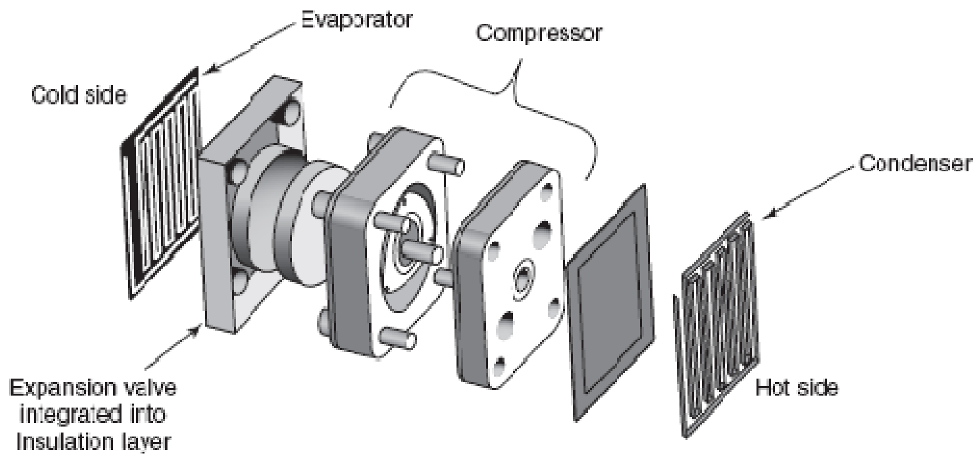


Fig. A3.2.22. The ARCTIC micro-heat pump/refrigerator

The schematic of ARCTIC is shown in Fig. A3.2.22. By using MEMS components within the unit, and a compressor with a 'footprint' about the size of an Intel Pentium 4 chip (25mm x 30mm) the researchers indicate that a theoretical COP of 4.6 is achievable, with a cooling capacity of 45W, a temperature difference of 40K and a compressor speed of 1000 rpm.

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