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Horizontal heat exchanger installation demonstration

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Ground-Source Heat Pump Workshop

This article summarizes a workshop conducted on ground-source heat pumps to examine their status, benefits, and opportunities. The speakers and other participants were selected from Canada, Germany, the Netherlands, Sweden, Switzerland, and the United States to enable an international exchange of ideas. A key topic of the meeting was identification of research and development needs, for which the preliminary results are presented.

Meeting in Albany, New York, a group of 48 invitees reviewed the status and prospects for ground-source heat pumps (GSHPs). The participants represented six European and North American countries; they included many internationally recognized leaders in the development and application of systems using the ground as a heat source for heating, and sink for air conditioning, via heat pumps.

The workshop objectives were to examine the technologies used, benefits derived, and opportunities to broaden application of GSHPs for building heating and air conditioning. By bringing together representatives from installing firms, equipment manufacturers, utilities, engineering firms, and research organizations, an opportu-

ity was provided to scrutinize these issues from several distinct perspectives. Moreover, the international participation enabled an intensive examination using current information on successes from around the world.

The workshop was organized and sponsored by the IEA Heat Pump Center with cosponsorship by the U.S. Department of Energy (DOE) and the Electric Power Research Institute (EPRI). The host for the meeting, and also a cosponsor, was the Niagara Mohawk Power Corporation, a large investor-owned utility selling both electricity and gas in upstate New York. The meeting was held at the Desmond Americana Hotel on October 27, - November 1, 1986.

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Topics addressed

Structured in six sessions, the meeting was designed to foster frequent discussions, both formal and informal, and exchange of ideas among the participants. Toward this end, the workshop included a welcoming reception and dinner, daily luncheons, and poster displays in the evenings.

The sessions, featuring 19 prepared presentations as well as a number of informal ones and discussions, included:

- Introductions and background information on system configurations and markets,
- Review of utility benefits and soil properties,
- Vertical, horizontal, and trench heat exchanger designs for ground-coupled systems,
- Direct-expansion options,
- Simulation, design models, installation techniques, and performance tests, and
- Equipment availability and development.

These sessions resulted in considerable discussion of the relative advantages and disadvantages of various configurations, as well as the many factors affecting their successful application.

The benefits of the international information exchange were evident. For example, a ground heat exchanger design shown in Figure 1 — the *trench collector* — that is becoming popular in Austria, Germany, and Switzerland was presented. It had not, to the knowledge of the attendees, been tried in North America. Conversely, soil characterization methods, data, and measuring devices from the United States, largely derived from electric utility cable installation research, were introduced for European benefit.

Summaries of the prepared presentations and discussion highlights follow.

Introductions and background information

Attendees were welcomed and the conference introduced by George C. Horan (USA) of Niagara Mohawk. Karl O. Holzappel (FRG) of the IEA Heat Pump Center followed with a review of the objectives and agenda.

The first paper was by James M. Calm (USA/FRG), a consultant to the Heat Pump Center, on *System Configurations*. This paper surveyed the various types of GSHP systems with a view towards establishing a common nomenclature and focus for the



Figure 1. Trench Collector Installation (Courtesy of H. Gerbert)

meeting. Shallow ground heat, as extracted by vertical exchangers shorter than 15 m (50 ft), is dominated by solar and resulting climatic effects. Deeper vertical exchangers also benefit from geothermal heat, which generally dominates below 30 m (100 ft). Three GSHP categories, summarized in Figure 2, were presented: groundwater, ground coupled, and direct expansion. The first two use water, or a water solution with an antifreeze (brine or glycol), for intermediate transport of heat from (to for cooling) the ground. In groundwater systems, water is removed from underground, though often reinjected later. In ground-coupled systems, a closed loop or loops of recirculated fluid is(are) used to couple the heat pump to the ground. For direct-expansion systems, refrigerant is actually circulated in a buried heat exchanger; the name derives from direct expansion (evaporation) of the refrigerant in the ground without an intermediate heat transport fluid. The terms ground and earth are usually synonymous, depending on context. Options in heat exchanger, or collector, configurations as well as in indoor heat distribution were also addressed. As indicated in the R&D needs discussed below, the lack of common terminology for GSHP repeatedly surfaces as an impediment to communications on such systems.

The next speaker was James E. Bose (USA), of Oklahoma State University, who addressed *The North American Market for GSHPs*. Dr. Bose cited a range of estimates, 1,500-15,000, for annual installations of ground-coupled heat pumps in 1985 and 1986. He suggested a likely estimate of 5,000, and stressed the need for advertising and promotional programs. His paper also pointed out several imped-

ments to broader application of the technology, including high initial costs, lack of installer experience, similar lack of consumer awareness, and the economic effects of low-market-volume.

Palne Mogensen (Sweden), a consultant, followed with *The Market for Ground-Source Heat Pumps in Western Europe*. The paper indicated that approximately 53,000 GSHPs have been installed. Almost all were heating only with hydronic (hot water) heat distribution inside buildings. The single-mode heat pumps competed primarily with gas- and oil-fired boilers and with electric resistance base-board heaters. The sharp decline in oil prices in 1986 severely curtailed the European heat pump market.

James G. Crawford (USA), Trane Company Dealer Products Group, concluded the first session with a presentation on *Ground-Source Heat Pumps from the Perspective of a Major Air-Source Heat Pump Manufacturer*. He noted that the present world and North American markets for GSHPs was minute compared to that of ASHPs. In order to maximize use of limited capital as well as technical and marketing support resources, major manufacturers tend to concentrate on high-volume products. Such products are the ones with which they can best compete and on which their businesses depend; for the foreseeable heat pump future, this will be ASHPs. Applauding the entrepreneurial spirit of the small manufacturers nurturing the GSHP market, the presentation noted that they serve a niche, but still valuable market. That market amounts to 2% of unitary heat pump and 0.5% of the total unitary equipment (air conditioner and heat pump) sales. Also, GSHPs are unlikely to displace ASHPs in retrofit installa-



Figure 3. Installation of Vertical Spiral Heat Exchanger (Courtesy of O. Svec)

tions; one explanation for this conclusion is insufficient available time to install a ground heat exchanger when an ASHP fails and needs replacement. Structural market factors and high costs are likely to preclude rapid growth in the GSHP market for some time.

Benefits and soil properties

Richard A. O'Neil (USA), Niagara Mohawk Power Corporation, spoke on *Benefits and Utility Experience with Ground-Source Heat Pumps*. He reported on his company's residential experience noting benefits in several areas for both customers and utilities. For customers, GSHPs provided reasonable paybacks over alternative systems, offered both heating and cooling modes, and a hedge against oil and gas price fluctuations and availability. For utilities, they reduced overall peak demands and modified diurnal load profiles to morning rather than evening peaks. About 45% of weekday energy consumption was off peak, thus improving utility load factors and reducing the need for additional generating capacity.

An added paper by Terry G. Statt (USA), U.S. Department of Energy, summarized a *Regional Economic Comparison of Ground-Source Heat Pumps*. It found that northern, colder, regions are the most economic locations for GSHPs with payback periods of three years compared to ASHPs. Using two pipes in a trench typically reduces the payback period by 25-30% over single pipes for horizontal heat exchangers; reduction by 50% results from optimization of the system design. Adding domestic water heating capability to the optimized two-pipe system reduces the payback period by 60-75% compared to the single-pipe system; the resultant economics improve marketability and expand the potential market to southern locations. Direct-expansion GSHPs were projected to be the most economic of the GSHPs evaluated.

Lawrence A. Salomone (USA) of STS Consultants discussed the *Effect of Soil Properties and External Factors on the Thermal Resistivity of Soils*. Despite recent advances, heat transfer from earth-contact surfaces is not well understood. One difficulty in estimating ground heat transfer is uncertainty regarding the thermal resistivity, or the reciprocal conductivity, of the soil. In practice, moisture content is strongly variable, and thus the thermal resistivity of the soil is rarely constant. Driving factors include soil type, dry density, moisture content, water retention, temperature, and heat-flux density. An approach, consolidating information from several scientific and engineering disciplines, was presented for considering the influence of these factors. Additional work is needed to develop the necessary family of data relations (e.g., for thermal resistivity to moisture content and density and for heat-flux density to moisture and temperature) for the variety of conditions that can be encountered with GSHPs.

Heat exchanger designs

The first presentation in this session was by Otto J. Svec (Canada) on *Vertical Ground Heat Exchanger Design*. He cited the relatively high costs of GSHPs as their main disadvantage. An expensive component of the costs — representing approximately 30% and 40% of the total for horizontal and vertical systems respectively — is for installation of the ground heat exchanger. The exchanger, therefore, offers very high potential for cost reduction. By improving the ground heat exchanger design and using lower operating temperatures, heat extraction/rejection rates can be substantially increased (two to three times). The exchanger length can, therefore, be reduced correspondingly. It is possible to design a highly efficient vertical ground heat exchanger, that operates below 0 °C (32 °F), with plastic heat exchangers. Such an exchanger is shown in Figure 3.

Anders Nelson (Sweden) of Geotec: Svenska Brunnborrars Branschorganisation, an association representing 35% of Swedish well drillers, then reported on *Swedish Borehole Technology*. 6,000 and 4,000 energy wells were drilled for GSHPs in Sweden in 1984 and 1985, respectively; the decline stems from falling oil prices. 95% were drilled in rock for ground-coupled systems, referred to as bedrock heat pump systems in Sweden. Tests have also been conducted on direct-expansion systems using 45 m (150 ft) deep, 19 mm (3/4 in) diameter, copper U-tubes for exchangers. Groundwater systems are uncommon due to corrosion and scaling problems. Innovative boring techniques have been developed, including *Odex* for drilling and casing in one called rock-laden overburden. *Odex* uses a pilot drill with an eccentric reamer. The reamer slightly increases the bore for casings, but can be retracted by reversing the direction of rotation to extract the drill and reamer through the installed casing. Other developments include compact and lightweight rigs, for use close to buildings, that are powered by air compressors which remain in the street. A 150 m (500 ft) energy well in bedrock, including the installed heat exchanger, costs approximately 28,000 Kroner (U.S. \$4,000).

Burkhard Sanner (FRG) summarized *German Research of Ground-Coupled Heat Pumps with Vertical Exchangers*. Several attempts were made in the late 1970s and early 1980s to use vertical exchangers for GSHPs. Most were steel pipes driven into the ground; some used plastic pipes in drilled holes. Experiments with seasonal thermal storage were also conducted. Systematic research and development started in December 1984 with investigation of environmental effects, ground temperature recovery, and performance monitoring. Current work is centered at a full-scale research station at Schwalbach, south of Wetzlar. Heating performance measurements were started for a 22 kW (75 Mbt) hydronic GSHP in September 1985. The initial coaxial exchanger was replaced in July 1986 with a double U-tube, made of four high-density polyethylene pipes. The new design is less expensive, but has shown similar performance. Monitoring of 35 houses with GSHPs is used to qualify experimental results from Schwalbach.

Hans Ludwig von Cube (FRG), a Consulting Engineer, provided a number of practical recommendations on *Heat Exchanger Design* based on his 30-year experience with GSHPs. He cited common design deficiencies, relating to inadequate analysis or understanding, and the consequences, such as low performance and frost heaving. For heating-only systems, extraction rate and duration must be such that adequate provision is allowed for natural recharge. Horizontal exchangers should be installed as close to the ground surface as possible, subject to depth requirements to avoid frost heaving and to consideration of

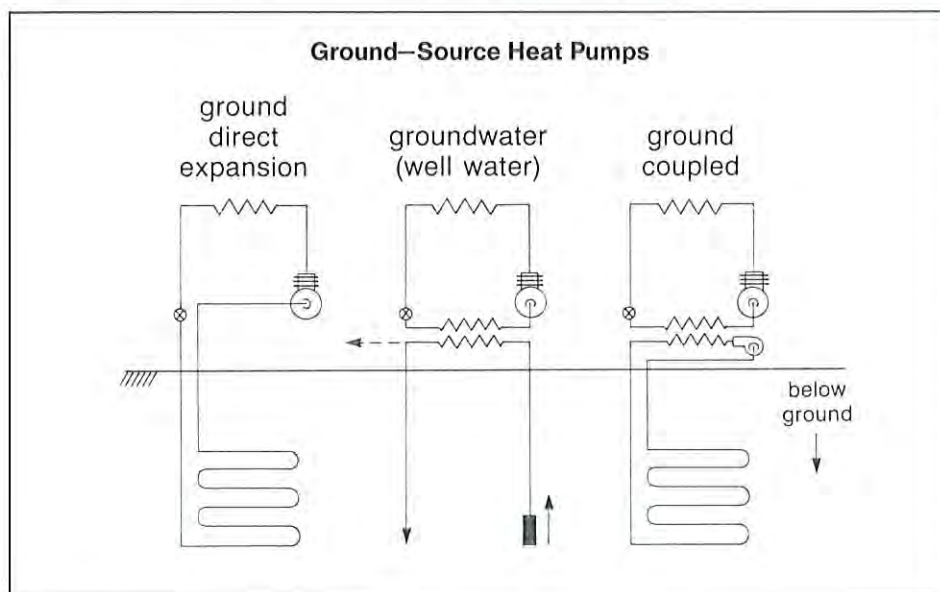


Figure 2. Ground-Source Heat Pump System Configurations

TOPIC	EUROPEAN			NORTH AMERICAN				TOTAL
	AERG	MFR	ALL	AERG	MFR	UTIL	ALL	
Develop simplified design methods	++	++	+	++	+	+++	++	++
Develop standard terminology for describing systems	+++	--	++	+	+	+++	++	+
Develop a universal rating procedure for ground heat exchangers	+	+++	+	++	+	++	+	+
Develop reliable sizing procedures to reduce oversizing		++	+	++	+	++	+	+
Develop low-cost ground-coupling heat exchangers	+	++	++	+	--	+++	+	+
Develop a GSHP information database		--		++		+++	++	+
Develop a simplified soil characterization approach	+			++		++	+	+
Develop low-cost drilling/trenching methods or equipment	++	+++	+++			++		+
Develop installation and maintenance training materials/programs	+	+++	++	+		+	+	+
Compare costs and performance of GSHP systems		+++	+	+	--	+++	+	+
Develop improved heat transfer fluids for ground-coupled systems	++	++	++	+				+
Identify regulations and laws regarding thermal-use rights	+		+		++		+	+
Monitor performance and costs of existing systems		--		+		++	+	+
Conduct seminars on accepted/recommended practices	+	++	+	+	--	++	+	+
Assess utility benefits				+	+		+	+
Assess ground-coupling fluid toxicities	--	++	--	++	+		+	+
Develop improved direct expansion systems	+	+++	++	+	--	++		+
Optimize vertical ground-coupled systems	+	++	+	+	+		+	+
Develop low-cost installation techniques	++		+			+++		+
Address the infrastructure for installation and maintenance				+	+	+	+	+
Develop direct-expansion GSHP systems that provide cooling	--	+++		+		++	+	+
AERG = University faculty and staff, engineers, researchers, government employees MFR = Manufacturer representatives UTIL = Utility and utility association representatives ALL = All of the above plus installers TOTAL = All attendees								
+++ = Very important ++ = Important + = Should be done = No consensus or recommendation -- = Majority of responses recommend against pursuing								

Table 1 Research and Development Needs Summary

how the overlying land will be used. Mathematical models have been developed and validated by field data from designs. Factors cited as retarding broader use of GSHPs were high costs due to lack of experience by installers, distrust by homeowners of unfamiliar systems, and consideration of GSHPs by architects too late in the design process, when favorable collector locations were already precluded.

Heinz Gerbert (FRG), a Consulting Engineer, described a heat exchanger for ground-coupled heat pump systems known as *The Trench Collector*. So called because of the very large — typically 3 m (10 ft) deep — trench in which it is installed, the exchanger consists of a horizontal arrangement of 20 mm (3/4 in) high-density polyethylene pipes. The pipes, typically 40 of them on each sloped-side of the trench, terminate inside a concrete manhole, at one end of the trench, where they are manifolded together. Figure 1 shows a trench collector during installation prior to backfill. The trench length varies with system capacity, but is typically 25 m (82 ft) for a single-family home; multiple trench collectors as long as 50 m (160 ft) can be used for schools or other large buildings. Approximately 600 installations have been made in Austria, Germany, and Switzerland since 1979. Typical costs for a fully installed collector including the trenching and backfill are DM 8,200 (U.S. \$4,100 at the time of the workshop, U.S. \$4,600 at current conversion rates) for a single-family house.

James R. Partin (USA), GEOSystems Incorporated, described a *Ground-Coupled Office Building in Hays, Kansas*. The system will use 73 heat pumps of various capacities and 120 vertical U-tube heat exchangers with a depth of 152 ft (46 m) each; the heating and cooling capacities are both expected to be approximately 1.5 million Btu/hr (440 kW_t). Completion of the building is expected in the spring of 1987.

Dan Ellis (USA), WaterFurnace International Incorporated, noted an installation that uses 200 heat pumps with an aggregate of 428 ton (1.5 MW) cooling capacity. More than 21 miles (34 km) of polybutylene pipe, forming and connecting 324 vertical heat exchangers each 150 ft (46 m) deep, and 7,000 gallons (26 m³) of water-glycol solution are required. The installation is a retrofit to a building being converted from a school to an office building in Terre Haute, Indiana. This installation may be the largest ground-coupled system in operation.

Direct-expansion options

Per-Eric Borin (Sweden) of AJ Tero-pump began the discussion of *Direct-expansion Systems* by comparing a direct-expansion to a ground-coupled system; both were heating-only. The latter used

365 m (1200 ft) of plastic heat exchanger contrasted to 180 m (590 ft) of copper exchanger for the former. The resulting evaporating temperatures were comparable, but the direct-expansion system reduced costs by 20% both due to the shorter collector and reduced complexity. Approximately 2500 direct-expansion systems are in operation in Sweden. The paper also reviewed schematics of heat pump systems using surface water, grey water from showers, and exhaust air as heat sources.

J. Hal Roberts (USA), ECR Technologies Incorporated, briefly described a direct-expansion GSHP system developed by his firm. Although no paper was prepared, advertising literature was circulated stating that the system achieves the efficiencies of ground-coupled systems at a cost that is just slightly above that of ASHPs. A proposed arrangement for a three-ton (10.6 kW_t cooling capacity) system uses parallel vertical heat exchangers made of coaxial copper tubes. The 90 ft (27 m) deep exchangers are spaced in a hexagonal pattern with a seventh tube in the center and with a distance of 10 ft (3 m) separating each from one-another. Claimed innovations include refrigerant flow-control devices that result in no superheat to the compressor, prevent passage of vapor from the condenser, and maintain a flooded evaporator.

An additional speaker scheduled for this session, Mr. Vahan Basmajian (USA) of the Megatech Corporation was unable to attend. His firm produced approximately 200 direct-expansion GSHPs, mostly installed in the northeastern portion of the United States, but discontinued the product in the first-quarter of 1986. Mention was also made of an unidentified German firm producing a direct-expansion GSHP.

Simulation, design, installation, and performance

Vince C. Mei (USA) of Oak Ridge National Laboratory began this topic with a description of *Heat Pump Ground Coil Models*. Five different models were developed, using rigorous physical theory, for different heat exchangers including both horizontal and vertical configurations. The models were validated with either field or laboratory experimental data, and provide more accurate analyses than simplified methods. Although computer execution time is excessive for general use, these models can be useful in qualifying designs developed with simplified methods, and — as computers become faster and more powerful — the new models will become more practical.

Palne Mogensen (Sweden) presented a second paper on *Small Computer Design Programs - European Style*. The paper notes that GSHP design differs from that for ASHP because the performance and capacity are affected not only by current

conditions, but by prior heat extraction. Designs must, therefore, address seasonal load profiles. A stepped approach, requiring approximately 13 kB memory, is given for simplified design of heating-only systems with hand-held (pocket) computers. The paper documents sample analyses. Although the program is based on verifiable physical principles, caution is expressed regarding its use outside the region for which the model was developed. Broader applicability would require additional characterizing parameters, increasing the program's size and complexity of use.

Patrick J. Hughes (USA) presented a paper coauthored with William S. Fleming and R. J. Hackner, all of W. S. Fleming and Associates, entitled *Materials, Design, and Installation Techniques*. The paper concludes that technology transfer materials to reduce risks to contractors, builders, and the design community are required and should take the form of guidelines for materials, design, and installation techniques. Ground-coupling requires considerable customizing by region due to variations in soil properties, moisture conditions, and construction practice. A major outcome of demonstrations in New York State was determination that ground heat exchanger sizes can be considerably smaller than previously indicated. Locally developed guidelines, based on physical models and monitored demonstration, are needed for other regions to support market development and to determine whether accepted sizing techniques are appropriate for the individual locations.

Van D. Baxter, Oak Ridge National Laboratory, presented a *Design Analysis and Field Evaluation Results of an Advanced Ground-Coupled Heat Pump System*; the paper was coauthored with Mark A. Catan, H. Michael Hughes, Patrick J. Hughes, and Richard A. O'Neil. It summarized research to minimize the life-cycle cost of a ground-coupled system by optimizing the heat pump and ground exchanger in concert. The resulting system was projected to offer payback periods shorter than three years relative to conventional ASHPs. Both heating and cooling performance, as well as the cost of the heat pump itself, are approximately 20% higher than for the conventional counterparts. Two prototype tests resulted in seasonal performance coefficients of 2.7 for heating and 3.3-3.5 for cooling. By improving the heat pump used, the ground exchanger size can be reduced at least 30% without overall loss of performance.

Johan Claesson (Sweden) described the *Computer Models and Analyses of Thermal Processes in the Ground* based on work at the University of Lund. Studies over the last ten years have resulted in theoretically derived computer models that have been validated by field and laboratory data. Both horizontal and vertical configurations as well as soil freezing and

interacting heat exchangers are addressed. Work to adapt the sizing and other models developed to run on micro-computers is underway. The importance of establishing the contact thermal resistance of exchangers was emphasized.

Equipment availability and development

H. Michael Hughes (USA), of Climate Master Incorporated, began the session with a paper on *Heat Pumps for Ground-Source Applications*. His presentation included slides and discussion of representative equipment and components. The paper dealt with unitary (factory-made) GSHPs for the heating-and-cooling markets of Canada and the United States. It noted that the full heating load cannot usually be met with GSHPs based on common sizing approaches. The majority of current installations are in single-family houses, but a very small market also exists in the commercial sector. Recent research to improve efficiency has resulted in equipment that is more expensive, but which uses smaller ground heat exchangers and offers lower life-cycle costs to consumers. The paper notes that optimal designs change as new components become available.

Karl Hess (Switzerland), SHF AG Entwicklung und Fabrikation von Wärmepumpen, addressed *Ground-Coupled Heat Pumps* indicating that approximately 1,000 have been installed in Switzerland. Typical designs are based on 63 W/m (66 Btu/hr ft) for vertical heat exchangers, resulting in annual extraction of 100-130 kWh/m (104-135 Mbtu/ft). Ground freezing is generally avoided, although the ethylene-glycol mixture used should withstand 17 °C (1 °F). Illustrations of typical installations and ground-temperature measurements were shown.

R&D needs identification

The need for simplified yet reliable design methods was stressed throughout the presentations and discussions. The simplified design issue, in fact, was rated as the highest priority among more than 75 research and development (R&D) topics identified. The R&D needs identification was requested by the cosponsoring organizations to steer their planning and budget preparation.

Requirements were noted whenever cited during the meeting or mentioned in the prepared papers. A roundtable discussion at the end of the second afternoon surfaced further concerns as well as responses to some previously identified needs. A list of the R&D topics, including several suggestions for marketing support, was compiled that night and introduced the next day. Each participant was asked to review it and note:

- The topics felt to be both the most and least important,
- Specific research goals and their importance,
- Potential uses and users of the results, and
- Possible approaches.

The resulting worksheets are being analyzed by the Heat Pump Center. Table 1 shows the needs identified as having the highest priorities. Perhaps less apparent from the table alone, compared to rankings of the complete set of candidate R&D topics, was general consistency between the various groupings of attendees. Split either between European and North American participants (i.e., heating-only contrasted to heating-cooling perspectives) or by professional groups as indicated in the table, agreement was high regarding topics not needing further R&D. As groups (discounting diversity within groups), the North American utility and utility association representatives and the European manufacturers tended to endorse more potential needs. By contrast, the North American manufacturers collectively found few needs warranting further attention. Note that recommendations against specific topics did not necessarily indicate disagreement with their intended result. They may also have implied perceptions that exhaustive R&D has been performed — whether or not in the public domain — and/or that the prognosis for success would be too low to justify.

An interesting contrast in opinions was for development of low-cost ground-coupling heat exchangers. Strongly endorsed by the utility group, the topic was opposed by the North American manufacturers. The same was true for comparisons of costs and performance of GSHPs, but here the European manufacturers joined the utility position. The most positive support from the North American manufacturers came for identification of regulations and laws governing thermal-use rights for the ground; other participants were less emphatic on this issue. Topics for which the European and North American participants disagreed included the development of low-cost drilling and trenching methods or equipment and — with reversed positions — development of a GSHP information database.

Further analysis and compilation of specific suggestions within these R&D topics is in process for inclusion in the proceedings of the meeting.

Proceedings

This cited proceedings will comprise papers prepared by the scheduled

speakers along with discussion summaries. All participants, whether or not scheduled as speakers, were invited to submit additional papers on relevant subjects they felt had not been adequately addressed. Copies of the proceedings will be available from the Heat Pump Center and will be announced in the Newsletter when ready. The delay is to enable editing, draft review by the participants, revisions, and publication. Completion is expected in July 1987.

Site visits

Several site visits were arranged to enable attendees, particularly those from Europe, to see typical North American systems. Preceding the workshop was a demonstration (see Figure on cover) of a horizontal heat exchanger being installed as part of a three-ton (11 kW_t cooling capacity), ground-coupled, residential system. The horizontal exchanger being installed consisted of 2,000 ft (600 m) of 3/4 in (19 mm) high-density polyethylene pipe. Four parallel pipes were installed in each of two 250 ft (75 m), 6 ft (1.8 m) deep trenches. Using equipment specifically developed for such installations, the trenching, exchanger burial, and backfill were performed in a single operation. Also shown was a popular method, derived from gas piping techniques, of fusing polyethylene pipes and fittings. The demonstration was arranged by Niagara Mohawk and the Charles Machine Works, manufacturer of Ditch Witch trenching and underground construction equipment.

Following the meeting was a two-day continuation of site visits. The first stop was in Utica, New York, to visit the plant where Climate Master Incorporated produces, among other products, water-source heat pumps for groundwater and ground-coupled systems. A leading manufacturer of such products in the United States, Climate Master has also conducted research to optimize heat pumps for use in ground-source systems. Results of this project, sponsored by DOE and Niagara Mohawk, were presented by Van D. Baxter et al and by H. Michael Hughes at the workshop (see preceding presentation summaries).

The tour continued to Syracuse, New York, headquarters of both Niagara Mohawk and the largest American manufacturer of heat pumps, the Carrier Corporation. Several Niagara Mohawk field tests of GSHPs installations were visited. Among these were a residence where performance of one of the optimized systems is being monitored. Another was an office building with multiple ground heat exchangers installed to compare their performance.

The participants then flew to Ottawa, Canada, where they spent the night before visiting the National Research Council of Canada. The GSHP research facility and

associated laboratories were shown. Included were a full-scale heat exchanger test facility, a demonstration house with a vertical ground-source system, a laboratory for testing thermal characteristics of soils, and a fully-computerized laboratory for study of soil freezing and frost heaving.

Annex VIII

The Steering Committee for Annex VIII met in conjunction with the workshop, since many of the European representatives to this Annex were attending. The Annex is one of 13 specific cooperative research activities under the IEA Implementing Agreement on Advanced Heat Pumps. Annex VIII addresses the development of novel, more-efficient ground heat exchangers. Included are:

- Development of advanced analytical models for horizontal heat exchange,
- Parametric studies and experimental validation,
- Monitoring of pilot facilities,
- Full-scale experiments on selected vertical and horizontal heat exchangers,
- Basic experimental research on heat and mass transfer characteristics of soils, and
- A demonstration project.

Organization and acknowledgements

The Organizing Committee for the workshop included Van D. Baxter, Karl O. Holz-
apfel, George C. Horan, Powell A. Joyner of the Electric Power Research Institute, Otto J. Svec, and Terry G. Statt. Sessions were chaired by George C. Horan, John D. Ryan of the Department of Energy, Phillip D. Fairchild of Oak Ridge National Laboratory, and Powell A. Joyner. Michael Cahill of Niagara Mohawk provided a welcoming address, stressing the benefits of GSHPs to both utilities and their customers, at the workshop dinner. John Borkowski, Alan F. Byrne, and George C. Horan, all of Niagara Mohawk, arranged the logistics for the meeting and site visits; they also assisted with the many details needing attention throughout the workshop. The author conducted the research needs identification and also coordinated the program and preparation of the proceedings.

Conclusions

The presentations and discussions suggested very important benefits, both for consumers and utilities, from the use of GSHPs. Still, these efficient heating and cooling devices capture only a relatively small share of either the new or retrofit construction markets in Europe, North America, and Japan. The largest share of the present market is for residential appli-

cations, but several commercial installations were noted.

The most limiting factor appears not to be the state of the technology, but the installed costs. Cost-reducing approaches were proposed, but further R&D is needed in this regard. One contribution from this workshop is a compilation of topics warranting attention.

The presentations and discussions at this meeting indicated that R&D efforts are underway for many of the topics identified. Furthermore, the individuals involved are more optimistic than pessimistic that solutions will be found. The level of expertise apparent at the workshop suggests that its subject has attracted the interest of some very creative and competent individuals and organizations.

Further attention to information dissemination and marketing is clearly also needed. The IEA Heat Pump Center can contribute in this regard.

Frequently cited as critical was proper training of the installers and inspection of their work. The importance of this issue was brought out by contrasting repeated successes by firms for which installation

had become routine to failures by crews unfamiliar with the systems.

It appears that no single approach will dominate GSHPs for the near term. The current market for groundwater systems is two-three times larger than that of ground-coupled systems, but both are being used and improved. Many different ground heat exchanger configurations are being pursued for the latter. There was strong disagreement on direct-expansion GSHPs. Some attendees felt this approach offers the highest potential for the future; others took an opposing view, even to the point of suggesting that such systems have no practical future.

Further international cooperation on GSHP advancement is clearly desirable. The sharing of methods, research and development results, and product ideas will help all involved. While the workshop itself has ended, the most valuable part will be the ongoing dialog and exchange among the leaders that met.

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O. J. Svec*

Ground Heat Exchange: Research Program in the National Research Council of Canada

The performance of the ground heat exchanger is one of the most important single considerations of a ground source heat pump system. Overall performance depends on a multitude of factors which, if fully understood and properly assessed, theoretically could be optimized to provide the "ideal" ground heat exchanger. The objective of a project at the Institute for Research in Construction (IRC) of the National Research Council of Canada is to develop a highly efficient and economical ground heat exchanger.

Introduction

Performance of a ground source heat exchanger depends upon the surrounding soil, the conceptual and physical design and the operation strategy. These factors must be considered together with initial installation costs of the system as well as the amount of site investigation required to obtain design parameters. Among important soil factors are thermal conductivity, heat capacity, moisture content, thermal diffusivity, hydraulic conductivity and ground water flow, thermal contact resistance, latent heat, and overall natural ground thermal regime. Design factors include the type of material (commonly plastic or metal), and the conceptual design such as U-pipes, concentric pipes, multichannel designs and spiral pipes as proposed here. Other factors which must be considered include size and shape

which influence the cost of installation, choice of backfill material (if any), and, finally, operating strategy.

Research program

At IRC it has been recognized that improvements in design can only be achieved by understanding each of the factors involved and by assessing the importance of each factor on the performance of ground heat exchangers. Building on many years of experience in testing the thermal characteristics of soils, in designing instrumentation for the determination of properties such as moisture content, moisture diffusivity and thermal conductivity and in developing experimental techniques, a program was initiated which progressed from determination of basic parameters through stages of experimentation to a full-scale demonstration

project. The key elements of the program were as follows:

1. Determine thermal behavior of various soils.
2. Conduct laboratory experiments using short sections of current and potential types of heat exchanger.
3. Conduct numerical analyses of new designs.
4. Undertake full scale prototype testing.
5. Carry out demonstration projects.

Stages 1 to 4 have been completed and a demonstration project is now underway. In stages 2 and 4, standard plastic U-tubes, steel finned pipes, vertical copper pipes and membrane-lined holes were compared to a spiral type of design using copper pipe. The full-scale prototype testing verified the laboratory and numerical predictions of improved performance capabilities of the spiral design. Stage 5 is now underway and the initial results are very encouraging, although the current winter (1987) has been unusually mild for the Ottawa region of Canada.

Demonstration house

An experimental house built on NRCC property has been used for testing a prototype ground heat exchange system connected to a ground source heat pump. This house was originally built for measuring heat loss from the basement to the ground and later used for research on direct passive solar gains. As a result, the overall thermal behavior of the house and its heat loss have been accurately determined. After these projects were completed, the house was modified to accommodate a water-to-air ground source heat pump using a forced-air distribution system. The house is a one-story structure with a full basement. The areas of the first floor and the basement are 85 m² each. There is a large open staircase leading to the basement, so that the heating system services both floors. The overall house heat loss coefficient is 58.5 W/°K. The temperature in the house is kept at 22°C.

There are four, 15 m deep, vertical copper spiral heat exchangers installed 6 m apart at a distance of 5 m from the house. The diameter of the tube is 2.2 cm and the diameter of each spiral is about 28 cm, with turns 15-18 cm apart. A 2.5 cm plastic tube is used as a return line. These heat exchangers were installed in 30 cm diameter predrilled holes in Leda clay. The sand backfill is saturated by the high ground water level. A two-ton water source heat pump was donated to the project by Earth Systems Ltd. of Leamington, Ontario, Canada.

Monitoring system

The performance of the heat pump system is being monitored at about 15 sec intervals during each running cycle. A computer program is initiated by the impulse

from the thermostat as it demands heating or cooling. The inlet and outlet temperature of the circulating fluid (water/methanol) is measured at the top of each heat exchanger in the ground, and at the distribution manifold in the basement of the house. There are two thermistors in each measuring location. The overall accuracy of these sensors is +0.005°C. The accuracy of the fluid flow is about 1-2%.

After each cycle, readings are summarized and the average difference of the inlet and outlet temperatures is computed. This information, together with fluid flow (accumulated volume), is used to calculate gained or dissipated energy by each ground loop as well as by the entire system. Daily, weekly, and monthly energy balances and COP are then determined.

In order to assess the overall performance of individual heat exchangers, the heat energy extraction or rejection per meter of a heat exchanger (hole) is also calculated. The heat demand from October to December has been met by using only two heat exchangers (i.e., 24 m of hole). Another important aspect of a ground source system, natural heat recovery of the ground, was observed for the prototype tests. For the demonstration house, natural recovery will be augmented by heat rejection from the air-conditioning cycles.

Results

This project commenced in the summer of 1986, so that results are only available for the first half of the first heating season (i.e., to December 1986). The October to December results may be summarized as follows:

1. Overall performance has been excellent; the temperature in the house does not fluctuate more than $\pm 1^\circ\text{C}$ in either the cooling or the heating mode.
2. The heat extraction rate per meter of the heat exchanger (per meter of the borehole) at the beginning of the heating season (October) was about 140-150 W/m and at the end of December was about 110-115 W/m. This rate is approximately three times more than the rate achieved using a conventional plastic U-tube in a similar-sized borehole.
3. The temperature of the fluid entering the ground coil was about -3°C to -3.5°C at the end of December and the return temperature was approximately 2° warmer.

Conclusion

The preliminary results are very encouraging and consistent with predictions based on the prototype tests. Copper coils larger than 25 cm in diameter can be manufactured from 2.2 cm diameter copper tube without difficulty. Up to now, relatively little effort has been made to develop a simple and rapid installation technique, but these results justify such a development. Similarly, potential corrosion problems and long-term system performance should be evaluated. Finally, design guidelines for copper or plastic spiral heat exchangers should be produced considering different soil and climatical conditions.

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B. Sanner*

Schwalbach Ground-Coupled Heat Pump (GCHP) Research Station

Since 1985, construction and operation of a full-size ground-coupled heat pump research plant (Fig. 1) has been underway at Schwalbach, just a few kilometers south of Wetzlar, Germany. The plant is equipped with one earth probe, 50 m long, a heat pump of 22 kW heating output, and blowers dumping excess heat to the air; operation is planned in heating-only mode. From September 1985 to April 1986, a heating period with a coaxial earth probe was recorded, followed by three months of thermal recovering of the ground. From fall 1986, a new type of earth probe has been used.

Schwalbach is a small village on the hills of the Rhenish Massif, 300 m above sea level. The bed rocks of the region are paleozoic sediments, mostly greywackes of carboniferous age. The ground water level varies between 12 and 14 m below ground surface. A hole 50 m deep was drilled into the rock, and a coaxial (pipe-

in-pipe) earth probe inserted. Alongside the outer wall of the earth probe a cable with temperature-sensors was fitted, one sensor every 2 m depth. Around this central hole several holes were drilled (Fig. 2), and temperature measuring probes were installed. A special recording system was developed to ensure accuracy and long-



Fig 1. Schwalbach GCHP Research Station, view to the North (Fall 1985)



Fig 3. Siemens heat pump used at Schwalbach GCHP Research Station



Fig. 4 Residential home in Schwalbach equipped with GCHP heating

term stability which works with semiconductors and microprocessors. Special regard was given to the tightness of these temperature-probes in groundwater.

The research plant can be operated in three different modes:

- Heating simulation mode (controlled by the temperature of the water supplied to the heating system)
- Constant brine temperature mode (controlled by the temperature of brine flowing to the earth probe; important for easy validation of computer simulation)
- Operator controlled (used for testing extreme limits of the system)

All data (Dec. 86: 144 ground temperatures, 4 outdoor temperatures, wind, rain, solar radiation, brine temperature, heat-pump performance) are recorded hourly by a datalogger designed exclusively for this job.

Based on the field tests in Schwalbach, the Institute of Applied Geosciences of Justus-Liebig-University, Giessen, Germany, developed computer-codes to simulate heat extraction from the ground. These simulations are done by the Finite-Difference-Method and will be validated against field data from Schwalbach and other locations. The working group of the Institute of Applied Geosciences, chaired by Prof. Dr. K. Knoblich, also carried out the hydrogeological and geological investigations at Schwalbach research site and is involved in rock heat-conductivity determination and hydrochemical analysis.

On June 23-24, 1986, a seminar on ground coupled heat pumps and seasonal heat storage took place at Giessen University. Research groups from Germany and Switzerland discussed these topics and determined fields of possible cooperation. Results of this seminar will be published in the last 1986 issue of the "Zeitschrift für Angewandte Geowissenschaften." In this issue, results of the first heating period of Schwalbach GCHP Research Station are also given.

Meanwhile, about 40 residential houses and 3 commercial buildings are equipped with heat pumps and vertical earth probes by Geotherm GmbH of Giessen-Allendorf. This company acts as a contractor in the research program and uses all results for improving the plants. At the same time, experiences from running heating systems can help to verify the experimental data of the Schwalbach research station. Two plants, one in Mülheim/Ruhr, the other in Huettenberg (south of Giessen), are equipped with monitoring devices and a new type of datalogger (48 ground temperatures, outdoor temperature, brine

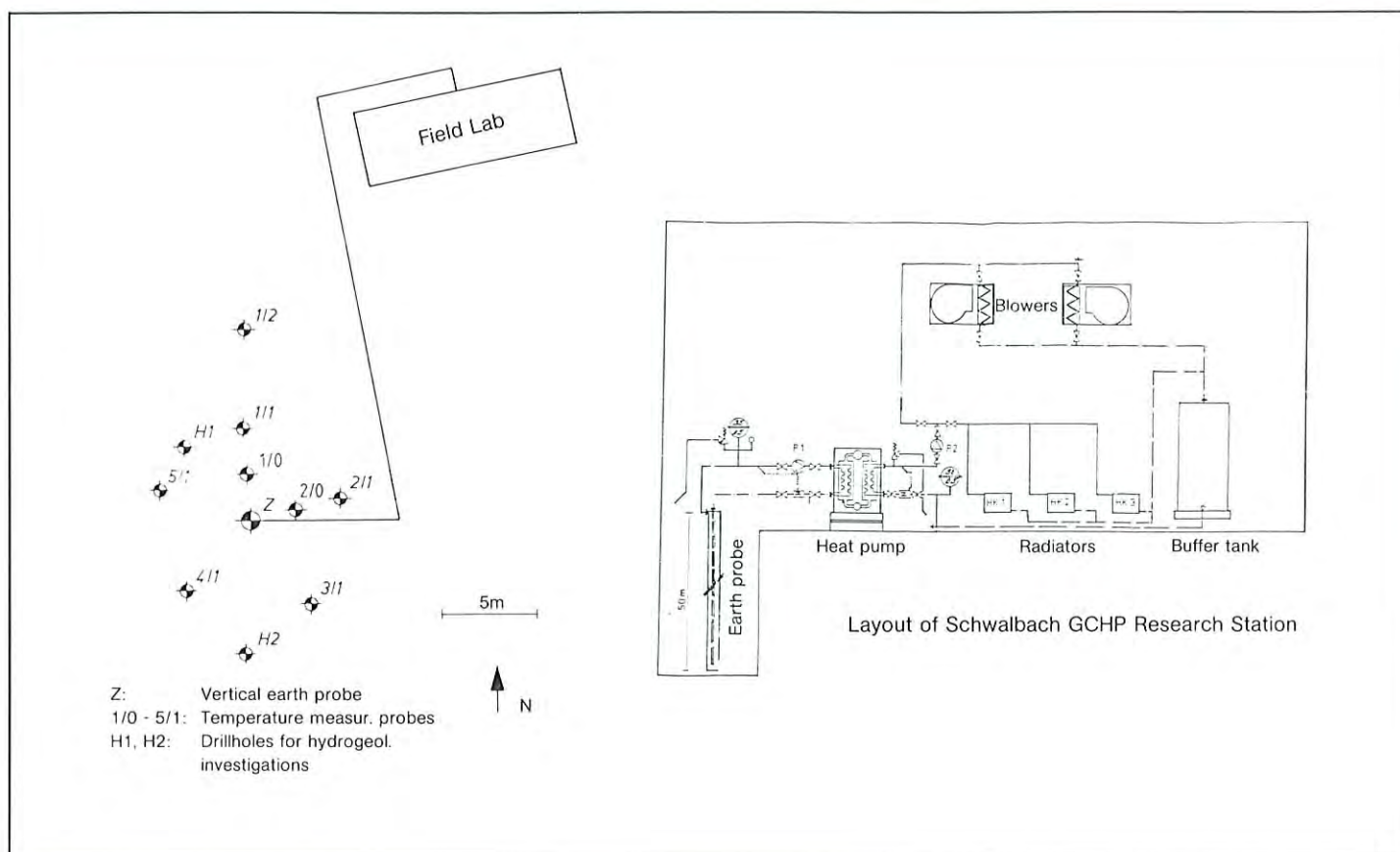


Fig 2. Schematic map and layout of Schwalbach GCHP Research Station

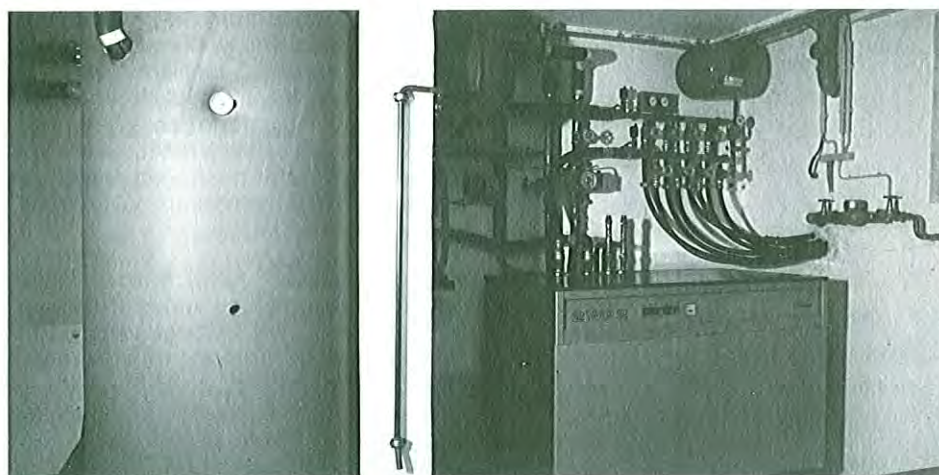


Fig 5. Heat pump and installation in Schwalbach home

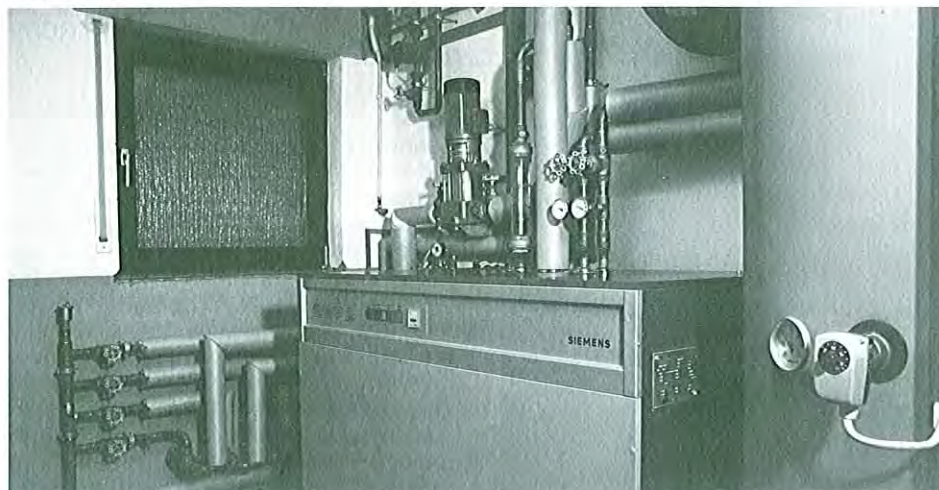


Fig 6. Heat pump and Brine system in a residential home in Wetzlar-Dutenhofen

temperature, heat pump performance). In 1987, five more plants will be monitored.

Figures 4-6 show examples of plants built by Geotherm GmbH. Each has four probes 45 m deep and a heat pump with 22 kW heating output. The plants run with an SPF of about 3; exact determination of COP and SPF will be done in the plants monitored in the research program.

In conclusion, we can report good performance and reliable operation of GCHP heating systems. Although all plants are running in heating-only mode, no problems with ground heat recovering arose since the first plant was installed in 1979. For special applications, the system can be combined with solar collectors or waste-heat storage to meet all demands. Future improvements may include reduction of drilling costs, implementation of new operational schemes, and use of nontoxic, noncorrosive brines. Currently, the degree of development is high enough to build GCHP-plants with vertical earth probes successfully and to enter the market of energy-saving heating systems in larger quantities.

We wish to acknowledge the financial contribution to this R&D-work granted by the German Federal Ministry of Research and Technology (BMFT).

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S. Goto*

National Project of Stirling Engines for Heat Pump Systems in Japan

This national project is to establish the high-efficiency Stirling engine technologies for practical use as power sources of 3 kW and 30 kW class heat pump systems for residential and commercial air conditioners, and 30 kW class electric power generator units. This project, which is a part of the energy conservation programs of the Agency of Industrial Science and Technology (AIST), is called "Moon Light Project." Total fund of this project will be ten billion yen (U.S.\$50 million) for the 6 years from 1982 to 1987.

The Stirling engine project has been developed under the cooperation between national research laboratories and the New Energy Development Organization (NEDO), as illustrated in Fig. 1. NEDO is responsible for the project management of contracting members consisting of four private companies under the programmatic direction of AIST.

Research and development in national laboratories

One research project in the Mechanical Engineering Laboratory (MEL) is to investigate the performance of the developed engines by NEDO and to obtain the fundamental data for the optimum design. The theoretical work has been completed concerning the thermodynamic analysis

of conventional type Stirling engines. In the experimental work, the performance of a displacer type engine has been examined in the two operation modes, a cryogenic engine mode and a hot engine mode. This has been followed by the performance test of "MELSE2" and "MELSE3" (Fig. 2), which are laboratory research engines newly built in MEL. In addition to these performance tests, the reciprocating seals, heat exchangers and crucial components have been studied to support the engine research.

In the National Aerospace Laboratory (NAL), low emission combustors are under experiment by using the test equipment in Fig. 3.

The National Research Institute for Pollution and Resources (NRIPR) has been re-

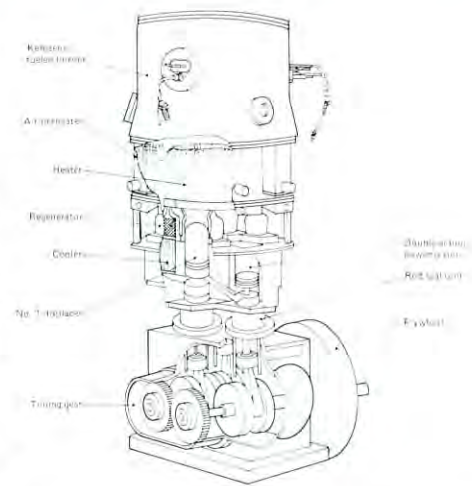


Fig. 2. Laboratory research Stirling engine Melse III (three-cylinder, double-acting displacer-type engine)

searching multi-fuel applications which make the combustion characteristics of wood fuel clear.

Research and development in NEDO

NEDO has been developing four different types of Stirling engines. The final performance and the application systems are presented in Table 1. The thermal efficiency is defined as the ratio of net shaft power to fuel thermal input with low heating value of natural gas. Figures 4-7 show the configurations of four types of Stirling engines developed.

The NSO3M engine by Mitsubishi Electric Co. (Fig. 4) is a well-known displacer type engine. A feature of the engine composition is an annular arrangement of heater head, regenerator and cooler. This arrangement contributes to the design simplicity and the engine compactness.

The NSO3T engine by Toshiba Co. (Fig. 5) is a typical two-piston type engine. The compression piston makes an angle of 60 degrees to the power piston so that the dead space is decreased to achieve the higher thermal efficiency as well as to achieve engine compactness. A heater tube comprises double tubes.

The NS30A engine by Aisin Seiki Co. (Fig. 6) is a four-cylinder double-acting type engine with a swash plate mechanism. Two types of engines, a horizontal and a vertical one, are developed. As for the vertical engine, a guide piston linked with the swash plate is directly connected to compressor pistons in heat pump systems.

The NS30S engine by Tokyo Sanyo Co. (Fig. 7) is a four cylinder double-acting type engine with a crank drive mechanism. In the final stage of the engine development, hydrogen gas is to be used as working gas. Dry lubrication for the crank drive has been applied to avoid the problem of lubricant leaking into cylinders.

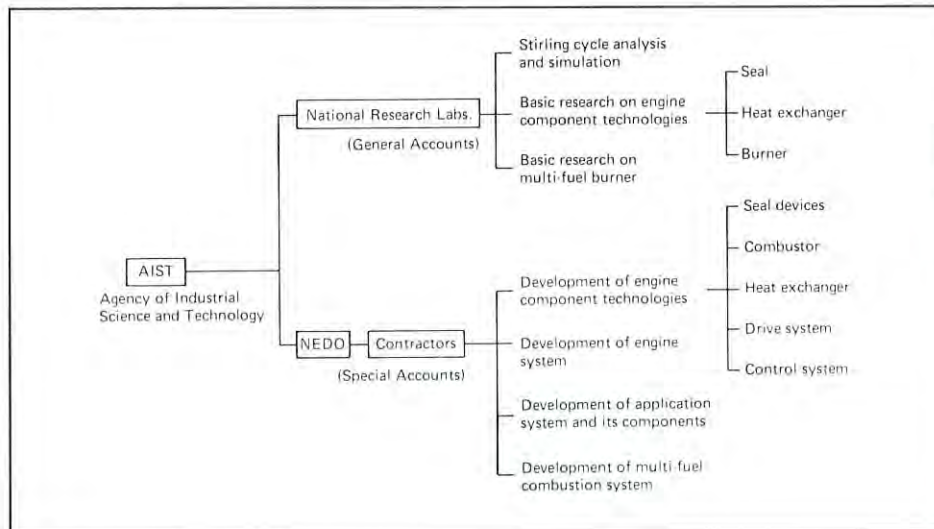


Fig. 1. Project organization and R&D approach

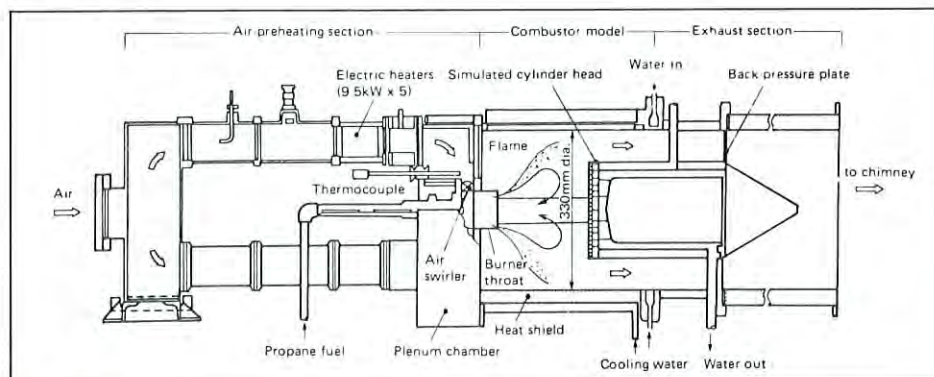


Fig. 3. High temperature combustor

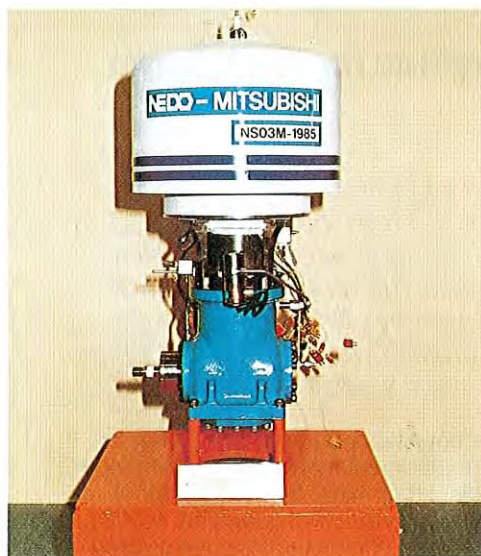
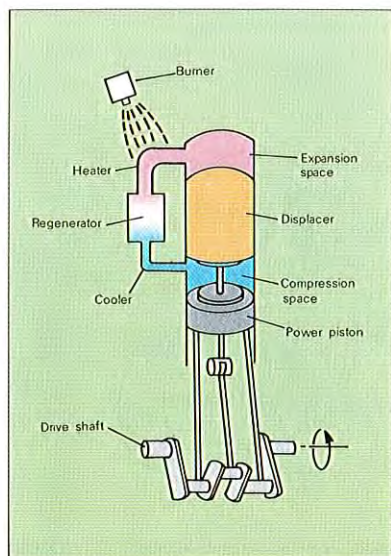


Fig. 4. NS03M 3kW Stirling engine for residential air-conditioning systems (displacer type)

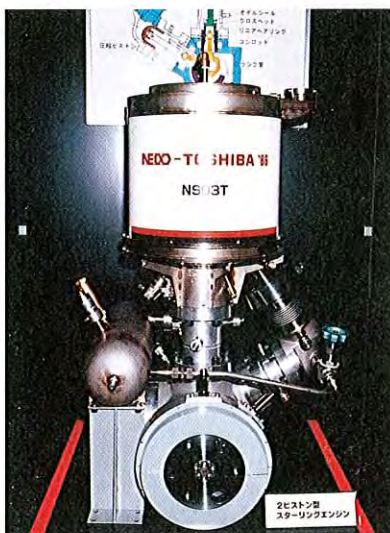
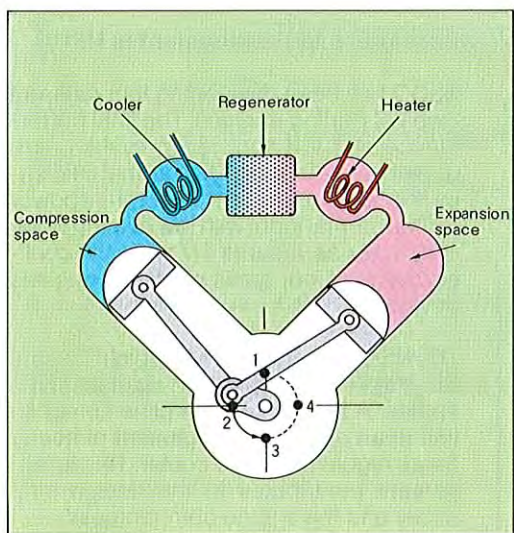


Fig. 5. NS03T 3kW Stirling engine for home air-conditioning systems (two-piston type)

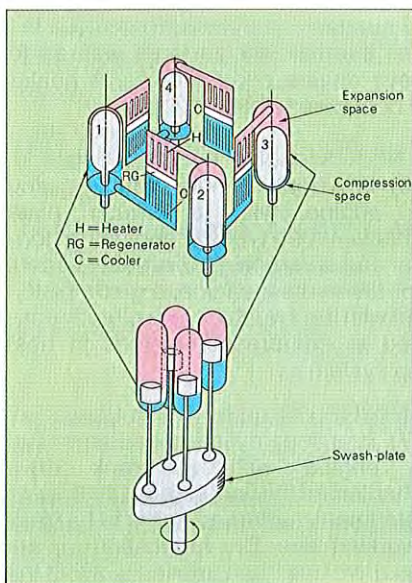


Fig. 6. NS30A 30kW Stirling engine for office air-conditioning systems (swash-plate type)

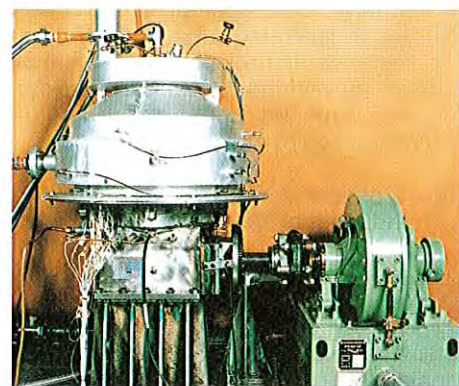
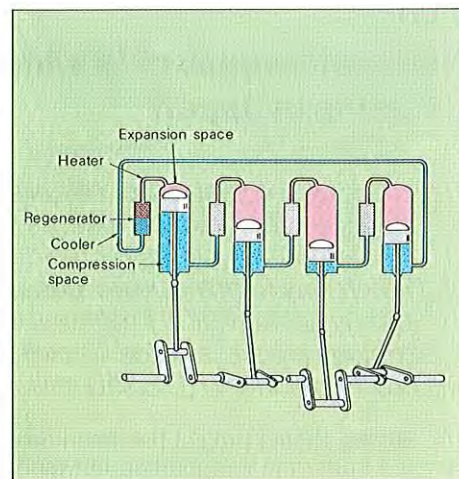


Fig. 7. NS30S 30kW Stirling engine for small-size electric power generators (crankshaft type)

Interim assessment

The interim assessment was held by AIST at the end of fiscal year 1984. Table 2 shows the results of the engine performance test of the interim assessment. All types of engines were evaluated as capable of meeting the high performance levels set as a goal and valuable enough to continue development as practical engines for the remaining development period.

Furthermore, it had to be considered that there were several technical problems to be solved. The first goal of R&D is to increase the thermal efficiency. The indicated efficiency was comparatively low, mainly because of regenerator loss. The next goal is to improve engines' suitability for practical use, with more durability, lighter weight and lower emission level of exhaust gas and noise.

Above are described outlines of the Stirling engine project in AIST. We have the intention of commercializing Stirling engines within several years. For that purpose, we wish to challenge other workers to solve difficult technical problems and consider it very important to prompt international cooperation in the research for developing Stirling engines.

**S. Goto, Ministry of International Trade and Industry, Japan*

Application			Air-conditioning system		Electric generator	Notes
Items	Brake Horse Power		3 kW class	30 kW class	30 kW class	
Engine system	Main fuel		Natural gas			Power loss caused by auxiliaries is included in each efficiency of 30 kW class engines.
	Maximum thermal efficiencies		32 %	35 %	37 %	
	Exhaust emissions		Below legislated standards for exhaust emissions from boilers.			
Application system	C.O.P.	In cooling mode	0.92 (1.43)	1.09 (1.57)	—	Values in parentheses represent the C.O.P. with hot water supply. Exhaust heat recovery : 70 % C.O.P. = Coefficient of performance.
		In heating mode	1.49	1.63		
	Life time		10 years			
	Maintenance frequency		Once a year			
	Allowable noise level		45 dB (A)	60 dB (A)		
	Fluctuations		—			

Table 1. Final goal of the Stirling engine project

	Application		Air-conditioning system			Electric generator
	Item	Model No.	NS03M	NS03T	NS30A	NS30S
Goals	Fuel		Natural gas.			
	Working gas		Helium			Helium (Finally, H ₂)
	Cooling method		Water cooling			
	Thermal efficiency		32%		35%	35% (Finally, 37%, H ₂)
	Shaft power		3 kW		30 kW	
Test results	Characteristic of max. power	Heater tube temp.	801 °C	789 °C	751 °C	763 °C
		Mean pressure	6.2 MPa	6.1 MPa	15.0 MPa	11.1 MPa
		Engine speed	1201 rpm	1250 rpm	1500 rpm	1600 rpm
		Shaft power	4.27 kW	3.46 kW	30.9 kW	23.4 kW
		Efficiency	29.5 %	27.3 %	27.8 %	26.8 %
	Characteristic of max. efficiency	Heater tube temp.	782 °C	798 °C	754 °C	812 °C
		Mean pressure	6.2 MPa	6.1 MPa	10.8 MPa	13.7 MPa
		Engine speed	800 rpm	900 rpm	750 rpm	826 rpm
		Shaft power	3.17 kW	3.00 kW	13.0 kW	18.8 kW
		Efficiency	31.6 %	31.1 %	31.6 %	29.5 %

Table 2. Goals and test results

IEA Heat Pump Center, National Team, Finland

Heat Pumps in Finland

The national team of Finland, IEA Heat Pump Center member country, presents an overview of the current and projected heat pump market in Finland.

Energy supply in Finland

Finland's energy supply is characterized by a high consumption level and a low degree of self-sufficiency. In 1985, total demand for primary energy was 27 Mtoe (electricity produced with hydro or nuclear power is multiplied by conversion factor 2.8); oil accounted for 34% of total supply and indigenous fuels for 30%. Total energy consumption has stopped increasing because of energy conservation and intensified energy use. The location of Finland in northern Europe, with annual mean temperature varying from 0°C to 5°C, keeps the demand for heating energy high.

In the beginning of the 1970's, the average specific energy consumption in residential, commercial and service buildings was typically 80-100 kWh/m³ per year. Due to energy conservation, insulation, and changed user habits, the average specific consumptions have now been brought down to 60 kWh/m³.

The total electricity supply (52 TWh) in 1985 consisted of the following:

- Nuclear power 35%
- Hydro power 24%
- Back-pressure power 23%
- Net imports 9%
- Others 9%

The real price of electricity has declined by 40% during the last 15 years. The price of electricity in the near future is expected to be relatively stable.

Heat pumps for space heating

Single family houses: The first Finnish series-produced heat pumps were installed in the 1970's. Extensive small-scale manufacturing was started, but resources for development, maintenance and training were scarce. At the same time, electricity prices were down and direct electric space heating became more and more common. Investment costs for heat pump heating were too high to attract builders, which resulted in a sharp decline in demand. The total number of installations is about 9,000, most of which are ground-coupled heat pumps.

Multi-family houses: The use of heat pumps in multi-family houses is still largely in the testing stage. Exhaust-air heat pumps are used for heating both tap water and space-heating water. Only 70 heat pumps have been installed in the 1980's.

Public or commercial buildings: The heat pumps in public or commercial buildings generally use exhaust-air or outdoor air as a heat source. Integrated systems which allow switching between heating and cooling are also used. In sparsely populated areas, competitive alternatives include heat pumps using ground and sea or lake water as a heat source. The number of heat pumps in operation is small, approximately 260 units.

District heating: The total installed heat pump capacity in the district heating system is 36 MW, which is made up of six MW-size plants. Heat sources include sea water, municipal waste water, and cooling water from cogeneration plants. Besides these plants, a few small heat pumps have been constructed, especially for experimental purposes.

Industrial heat pumps

Small heat pumps, with a capacity under 100 kW, are used for drying applications, such as timber, leather, furs, and powdered milk. The total number of units in operation is about 600, with the most common application being timber drying.

Heat from cooling processes is used as a heat source for heat pumps, especially in the food products industry. Tens of 100 kW-size heat pumps are operating in dairies and slaughterhouses, and also in the plastics industry.

All the above-mentioned applications for space heating and industrial processes use electric heat pumps. In addition, me-

chanical vapor compression is used in distillation and evaporation processes. Dairies and the wood processing industry have 25 mechanical vapor compression plants, with a total heat capacity of 230 MW.

Market potential

Heat pump sales in the field of space heating for single family houses are very modest. The economy of small exhaust-air heat pumps might improve because in the future there will be a greater emphasis on ventilation. The aim of further development is a highly standardized, inexpensive heat pump with a long utilization time.

High investment costs and operational problems will reduce the potential of multi-family heat pumps in favor of other systems. The exhaust-air heat pump might be feasible in buildings with a mechanical exhaust-air system when light fuel oil is replaced.

In public and commercial buildings, the use of heat pumps for heating and cooling will continue to increase. Heat pumps using sea and lake water are also expected to be competitive in the future.

The feasibility and competitiveness of district heating heat pumps largely depend on local conditions. To determine their impact on the market potential, a study was made covering all localities where district heat load and heat source for at least a 0.5 MW heat pump were available. Industrial waste water and municipal sewage water were considered possible heat sources. The trend of electricity and fuel prices will have a great influence on the market penetration of heat pumps. Although the survey results identified the techno-economical potential of the total heating capacity of heat pump installations in the 1990's to be several hundred megawatts, the estimated usage is less than 200.

HEAT PUMP APPLICATION	ESTIMATE FOR 1990		
	PRESENT	SITUATION	
	Number of Units	Total Capacity (MW)	Total Capacity (MW)
Single family houses	9,000	100	120
Multi-family houses	70	5	10
Public or commercial buildings	260	75	210
District heating	6	36	80
Industry	700	250	320
Total		466	740

Table 1. Current heat pump capacity and estimate for 1990

The industrial heat pump market will continue to grow. Drying and evaporation processes will be the most common industrial heat pump applications in the future.

Table 1 shows the present situation and an estimate for heat pump capacity in 1990 in each category.

Research and development

R&D on heat pumps is carried out by technical universities, energy companies, consulting engineers and heat pump manufacturers. The most significant financier is the Ministry of Trade and Industry which, during the 1980's, has granted FIM 15 million for heat pump research.

Some R&D projects currently in progress:

- Exhaust-air heat pumps in single family houses

- Space heating of public and commercial buildings by exhaust-air heat pump
- Natural gas-driven heat pumps in district heating systems and wood processing industry
- Heat pumps for heat recovery in paper machines
- Scope of absorption heat pumps in forest industry (case study)
- Ice heat pumps
- Development of evaporators for heat pump systems
- Accumulative heat pump using zeolite

The R&D work thus far has focused primarily on heat pump applications; the heat pump process itself has not been given much attention. □

22°C range had proven to be the optimum temperature for accelerated lobster growth.

A feasibility study completed in 1983 indicated that a heat pump system would be suitable for this application.

Heat pump system

The heat pump/heat exchanger system shown in Figure 1 was developed with sufficient flexibility to operate in various modes, depending on seasonal temperatures and user requirements. The two basic modes of operation are described below. The heat pump package before installation is seen in Figure 2. The conceptual design study and system monitoring were funded by the National Research Council Canada, and the heat pump was manufactured by J. H. Lock & Sons Ltd., Dartmouth, Nova Scotia, Canada.

Operating modes

The heat pump can be set for either cooling or heating mode at the control panel.

Cooling mode: The heat pump operates in the cooling mode from April to November, and runs 24 hours per day during this period. The heat pump is used to extract heat from the seawater and maintain a reduced seawater outlet temperature suitable for supply to the lobster storage system. The cold waste seawater returned from the storage reservoir passes through the cold seawater plate heat exchanger and is used to precool the incoming seawater entering the heat pump evaporator. The temperature of the seawater inlet to the lobster trays is controlled by unloading banks of compressor cylinders as required. The heat output from the heat pump is used to heat seawater for the warm seawater trays, provide building heating, or if there are no heating requirements, the heat is rejected to waste seawater via the waste heat plate heat exchanger.

J. W. Linton and D. Moscovitch*

Seawater Source Heat Pump Used for Live Lobster Storage

Clearwater Lobsters Ltd., of Halifax, Nova Scotia, is using a heat pump system to solve a commercial and culinary problem, that is, storing live lobsters throughout the entire year.

Introduction

The fishing season for lobsters in Canada occurs only in the spring and autumn of the year. However, a demand for lobster exists year round and prices rise considerably when lobsters are out of season.

Until recently, the storage of live lobsters into the off season incurred considerable product losses, especially when the seawater temperature was warmest during the summer. In order to supply lobsters through the year without these losses, Clearwater Lobsters Ltd., Halifax, Nova Scotia, Canada, approached the problem in two stages. First, Clearwater developed a dryland pound long-term storage system in which lobsters are stored in compartmented trays. The trays are stacked in a system of racks, and seawater is continuously recirculated through the tray system (see Figure 1). A percentage of the recirculated seawater must be dumped and fresh seawater introduced to remove waste products from the tray system.

After the dryland pound system proved very successful in reducing lobster mortality, it was determined that if the seawater could be maintained below 4°C during the holding period, the mortality rate would be

further reduced. The seawater supply temperature at the storage facility usually varies from 15.5°C to 1.0°C during the year, with the peak temperatures occurring during the late summer and early fall. Clearwater was also interested in the possibility of growing lobsters in captivity, and seawater temperatures in the 15°C to

Heat pump specifications

Type:	Electric motor driven-vapor compression
Compressors:	Reciprocating type, two 6 cylinders driven by a 74.6 kW motor and one 4 cylinders driven by a 56 kW motor
Working fluid:	Refrigerant 12 (Dichlorodifluoromethane)
Lubricating oil:	Synthetic oil (Chevron Zerol)
Evaporating temperature:	-6.3°C to -1.7°C
Condensing temperature:	62°C
Maximum output temperature:	60°C
Maximum heat output:	579 kW
Maximum heating COP:	2.8
Seawater inlet temperature:	1.0°C to 5.6°C

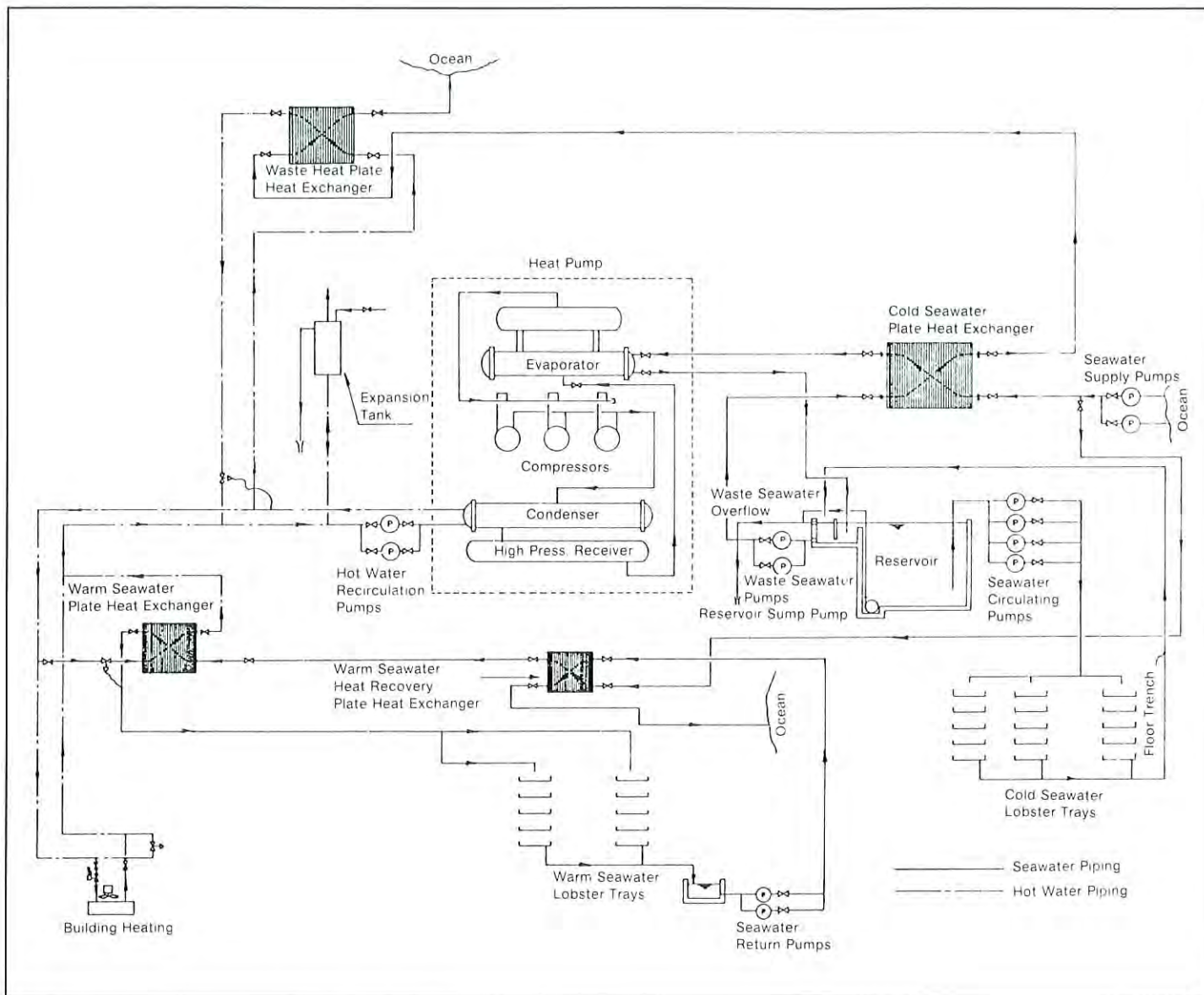


Figure 1. Schematic Flow Diagram (Cooling Mode Shown)

Heating mode: The system is operated in this mode when the ambient seawater temperature is below 4°C. No cooling of the seawater is required before inlet to the lobster trays. Sea water returned from the storage reservoir is used as the source for the heat pump before being pumped back to the ocean. Similar to the cooling mode, the heat output is used to supply the heating loads as required, but in this case the heat pump is controlled by maintaining a constant hot water outlet temperature from the condenser. The compressors unload banks of cylinders if the outlet temperature is above the set point, indicating reduced heating requirements.

Heat pump description

Compressors

The three reciprocating compressors were selected to ensure that adequate cooling would still be available if any mechanical problems developed with any one of the compressors. The compressors are equipped with refrigerant cooled oil coolers.

Heat exchangers

The evaporator and condenser are shell and tube type heat exchangers.

Evaporator: The evaporator has refrigerant in the shell and seawater in the tubes. A refrigerant separation vessel is mounted on top of the heat exchanger and connected by risers to ensure that only vapor enters the compressors.

The materials in contact with seawater had to be carefully selected for corrosion resistance and non-toxicity to the lobsters. Titanium was selected for the tubes and tube sheet cladding for these reasons and for its good thermal conductivity properties.

Condenser: The condenser contains refrigerant in the shell and fresh water in the tubes. The tubes of the evaporator are made of cupro-nickel because the fresh water available at the plant was slightly brackish.

Control system

The heat pump is controlled by a micro-processor-based programmable logic controller (PLC). The PLC controls the loading and unloading of compressors, monitors the electrical supply for phase loss, phase reversal and loss of power. The heat pump has all the safety controls required for unattended operation. In the event of a problem, the PLC uses an auto-dialler and telephone link to notify the duty operator.

Operating experience

The heat pump system has been in full operation since May 1984. The performance of the system was monitored from May 1985 to March 1986. Heat pump performance has been to design specifications with only minor problems since system commissioning.

Economics

The capital costs for this project were as follows (1984 Canadian Dollars):

Heat pump	\$200,000.00
Titanium plate heat exchangers	100,000.00
Pumps	30,000.00
Building and seawater reservoir	82,000.00
Piping, electrical, labor	40,000.00
Engineering and contingencies	30,000.00
Total installed cost	\$482,000.00

Comparing the operating costs of the heat pump system to the alternative of operating a boiler and seawater chiller, the annual cost savings are \$164,000.00. Therefore, the simple payback period for the whole system (based on energy savings only) is approximately 2.9 years.

Although an accurate figure is not available for the savings due to decreased lobster mortality, it is expected to be very significant. Thus, when energy and product savings are considered together, the heat pump system has a very favorable rate of return.

Future work

The heat pump system has proven so successful that Clearwater Lobsters Ltd. has



Figure 2. Heat Pump Used for Live Lobster Storage (Before Installation)

added an additional lobster storage facility and connected it to the heat pump system. The company also plans a new lobster holding facility at another location using a similar heat pump system.

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Development of Heat Pump Room Air Conditioners for Cold Districts

As a result of its efforts made since 1981, Tohoku Electric Power Co., Ltd. (TEPCO) has recently developed and commercialized heat pump type room air conditioners (HPRACs) used for cold districts. The new development is outlined below.

1. Necessity and history of the development of HPRACs for cold districts

The northern half of Japan's territory consists of cold districts, which include Hokkaido, the northeastern six prefectures, the northern part of Kanto District (composed of Ibaragi, Tochigi and Gunma Prefectures), as well as Niigata, Toyama, Nagano Prefectures, etc. In some of these cold districts, the lowest winter temperature drops below -20°C , complicating a normal function of usually available HPRACs due to frosting, with deficient heating capacity caused by low temperatures of ambient air as a heat source. Few HPRACs have actually been in use in these districts.

However, as space heating by HPRACs involves no combustion, it is safe, clean, easy to use, and provides a high energy

consumption efficiency. Because of these merits, HPRACs are considerably demanded in the field of residential heating and in warm districts; 12% of all room heating appliances are represented by them. In cold districts, however, the rate of their penetration is said to remain below 1/10 compared to warm districts.

In 1981 TEPCO started to develop advanced HPRACs for cold districts to improve these circumstances.

Two years later, new types of HPRAC with heaters were developed as a result of the joint development work with heat pump manufacturers. In 1985, more practical types of HPRAC without heaters were materialized which can be used satisfactorily even at temperatures -7°C .

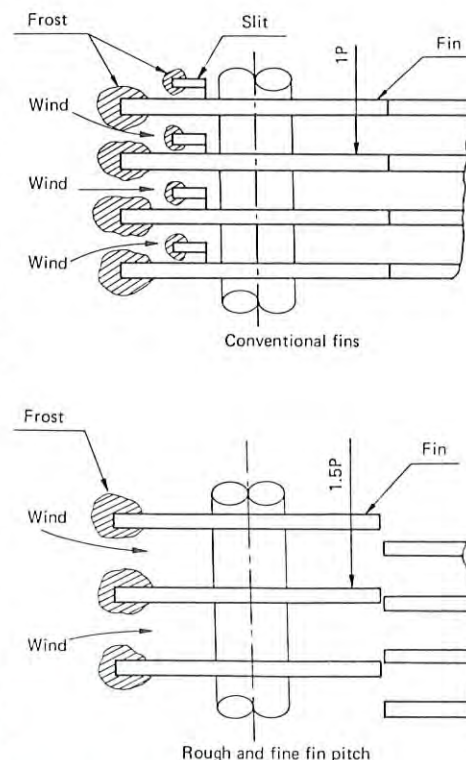


Fig. 1. Example of improved fin shape

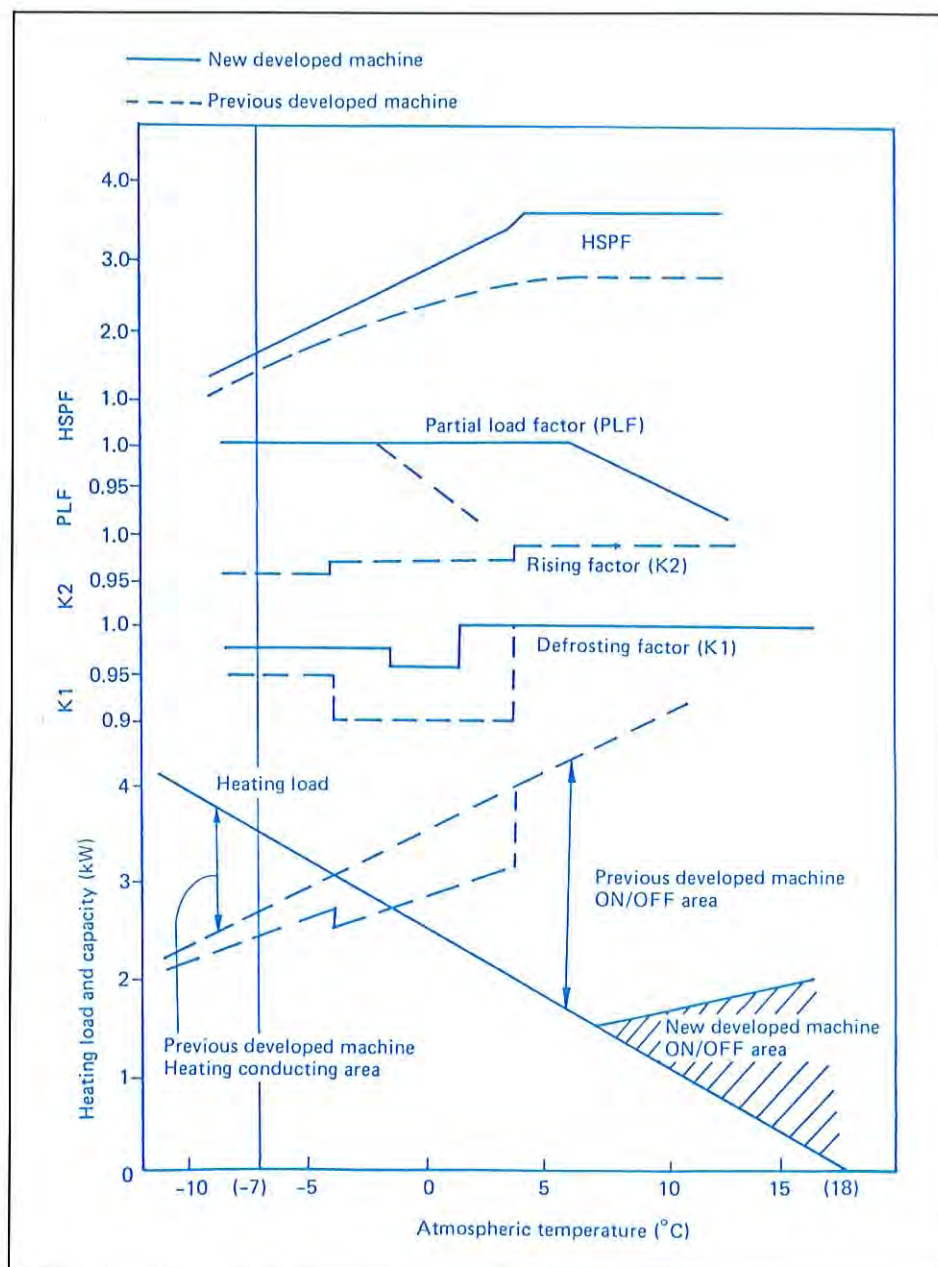


Fig. 2. Example of basic characteristics of heat pump in cold districts

2. Major element technologies developed

2.1 Outline

2.1.1 Development of inverter-controlled high-efficiency compressors for cold districts

By developing measures to reduce the pressure loss of the suction side and discharge side of the compressor during the high-speed and high-efficiency operation, and improving the mechanism for separating the working fluid and the refrigerating machine oil, high-efficiency compressors were developed.

2.1.2 Introduction of the refrigeration machine oil corresponding to the time of low outdoor temperature operation.

By selecting a refrigeration machine oil which is low in pour point and flock point,

and which provides excellent low-temperature resisting characteristics, the reliability of the compressor operation while the outdoor temperature is low was ensured.

2.1.3 Development of a pre-heating method for the compressor

During the heat-up operation, about 14.5% of the total consumption of electricity is required for heating the compressor itself. For this reason, we have developed a compressor pre-heating method in order to reduce the heat amount required for heating the compressor during the heat-up operation, or to shorten the heat-up time.

2.1.4 Introduction of a new working fluid and development of an optimum control system for the refrigeration cycle

A new working fluid (R22+R13B₁), which can operate most efficiently in a total am-

bient temperature region during room heating time and an optimum control system, has been developed. In particular, the capacity at the low ambient temperature time was increased and the starting characteristics of the compressor were improved.

2.1.5 Development of a new frost detecting method

A detection algorithm of heat source air temperatures and heat exchanger temperatures, and also a detection algorithm of heat source air temperatures/humidity and heat exchanger temperatures, which are both new frost detecting methods, were developed for prevention of ineffective defrosting operation and for reduction of defrosting time.

2.1.6 Development of a new defrosting method

In order to reduce the defrosting time, a heat accumulating method for defrosting, which accumulates heat in the refrigeration cycle, has been developed.

2.1.7 Development of a method for frost reduction

A method for frost reduction by improving the heat exchanger shape and the surface treatment of the heat exchanger was developed, in order to improve the efficiency of the defrosting operation. Fig. 1 shows an example of the improved fin shape.

2.1.8 Improved pneumatic power characteristics

By means of a new-type propeller fan and improved air inlet and air outlet shapes, it is designed to increase the amount of blast of the unit and to decrease its noise.

2.2 Results

2.2.1 Heating capacity under low outdoor temperature

Although the rated output of the recently developed compressor is 750W, its heating capacity at an ambient temperature of -7°C ranges from 3.5 to 4.4 kW without an auxiliary heater. This is approximately equivalent to the heating capacity, 3.6 to 4.5 kW, of the formerly developed model (about 1 kW in input) with an auxiliary heater.

When comparing the heating capacities of the old and new models at an ambient temperature of -7°C and a base outdoor temperature of 7°C, the percentage of the heating capacity of the formerly developed model is 64%, while that of the recently developed model is 76 to 88%. This indicates a sharply improved rate of capacity drop in the low-temperature range.

2.2.2 Heating seasonal performance factor (HSPF)

The HSPF represented an average of 2.87 to 3.21 for the three northeastern prefectures (Aomori, Akita and Iwate), where de-

monstrative experiments were carried out, showing an improvement of about 30 to 37% compared with 2.16 to 2.47 of the formerly developed model.

2.2.3 Heat-up time of the room temperature

The time required for the room temperature to rise from 0°C to 18°C when the ambient temperature is 0°C, was about 15 minutes, and could reduce the heat-up time of 40 to 45 minutes required by the formerly developed models to about 1/3.

2.2.4 Defrosting time and frosting

The time required for defrosting the outdoor heat exchanger is about 2 to 4 minutes, approximately halving the time of about 7 minutes required by the formerly developed model.

The temperature range of +4 to -4°C of the former model where the maximum frost was formed, could be reduced to +2 to -2°C. Additionally, steps were taken to prevent ineffective defrosting operation, and to achieve a strong frost reduction.

Fig. 2 illustrates the basic characteristic of the HPRAC designed for use in cold districts.

3. Demonstration results

Assessment of the HPRAC consists of such important points as the sense of heating, heat-up time, defrosting operation, operation costs, and applicability.

The demonstrations were carried out on advanced type machines without heaters for a period between December 1984 and April 1985 in 12 major cities of the north-eastern prefectures (Aomori, Iwate and Akita). The experiments are outlined below.

3.1 Sense of heating

A monitor in a living room of 13.2 to 16.5 m² in area comments that this type RAC gives a softer sense of heating than a kerosene-fired heater for producing warm wind.

3.2 Heat-up-time characteristics

Regarding room temperature increase in the morning, in addition to the reduced heat-up time, almost all monitors use a timer to perform pre-heating for a time between 30 minutes and one hour before the room occupants get up; they comment that the room is warm enough.

3.3 Defrosting operation

The duration and frequency of the defrosting operation is governed by meteorological conditions such as temperature and humidity of the ambient air used as the heat source (snowfall, etc.). Even at an ambient air temperature level of 2 to -2°C, which is the temperature region causing

the maximum frosting on the machine, the frequency of the defrosting operation can be once in 50 to 90 minutes, and its duration can be as short as 2 to 3 minutes, so that the room temperature is hardly affected by the defrosting operation. Most of the monitors comment that they took no notice of the machine operation.

3.4 Cost of machine operation

Due to the difference in living customs of monitors, the size of the room (13.2 to 23.1 m²) where the HPRAC is installed, and the length of time it is operated, the power consumption varies widely from 400 to 1,000 kWh/month depending on the monitors. However, if the machine is oper-

ated in a manner that agrees with the actual lives of common monitors, the power consumption is considered to range from 300 to 500 kWh/month.

According to a simulation carried out under the preset condition that the HPRAC is used in a living room of 13.2 m² in area, based on the data obtained from the present corroborative experiments, the average power consumption in the three prefectures of the Tohoku District is 270 kWh/month. But in view of the scattering of the values expressed by the monitors, it ranges from 180 kWh/month to 400 kWh/month. This is considered to result from the size of each monitor's living room used

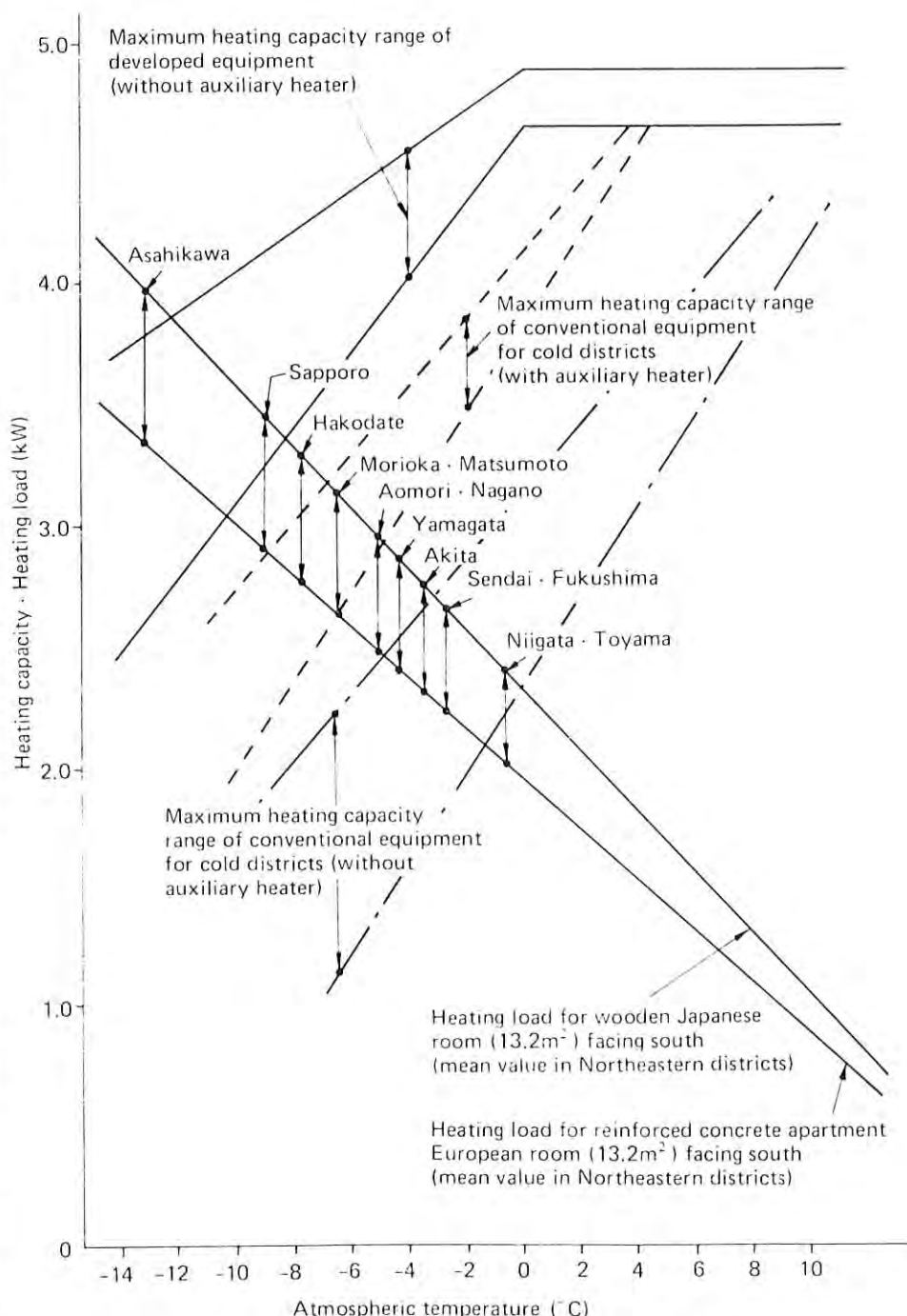


Fig. 3. Atmospheric temperature and heating capacity. Heating load characteristics

as the basis, the adiabatic capacity of buildings, etc.

The HSPF obtained from a simulation performed in the three prefectures of the Tohoku District is 2.92 to 3.27, the unit price of energy obtained is 10.39 to 11.64 yen/kWh (the unit price of electric energy of Tohoku Electric Power Co. is 33.95 yen/kWh). This is calculated for trial as ap-

proximately equivalent to the unit energy cost of 11.82 yen/kWh of the FF type kerosene stove.

4. Applicability

The hours of ambient temperature which fall below -7°C during the room heating period in cold districts are 1.9% of the pe-

riod in Morioka and 0.1% in Akita. This is a rarely seen example.

The standard extent of room heating achieved is (as shown in Fig. 3) 3.13 kWh in a wooden Japanese-style living room (13.2 m^2 in area) and 2.61 kWh in a Western-style room (13.2 m^2 in area) of a ferroconcrete apartment house building in Morioka, where the monthly average

		Entire Air Conditioning Units*	Units Including Heat Pump	Component Ratio of Heat Pump
Aomori		3.3	0.8	24%
Iwate		3.3	1.3	39%
Akita		5.2	1.6	31%
Miyagi		13.0	7.4	57%
Yamagata		7.3	2.5	34%
Fukushima		12.4	6.1	49%
Niigata		39.8	19.1	48%
Total		84.3	38.8	46%
Nationwide		2850.0	1800.0	63%
Ratio to Nationwide		3.0%	2.2%	

		1983	1984	Rate of Increase
NORTHEASTERN DISTRICTS	Entire Air Conditioning Units	66.5	84.3	26.8%
	Units Including Heat Pump	19.8	38.8	96.0%
NATIONWIDE	Entire Air Conditioning Units	2430.0	2850.0	17.3%
	Units Including Heat Pump	1160.0	1800.0	55.2%

(*Unit = 1,000)

NOTE: Source of data for 6 Northeastern prefectures and nationwide --DENPA paper; source of data for Niigata Prefecture -- DENSHOSO

Table 1. Number of Machines Shipped in 1984

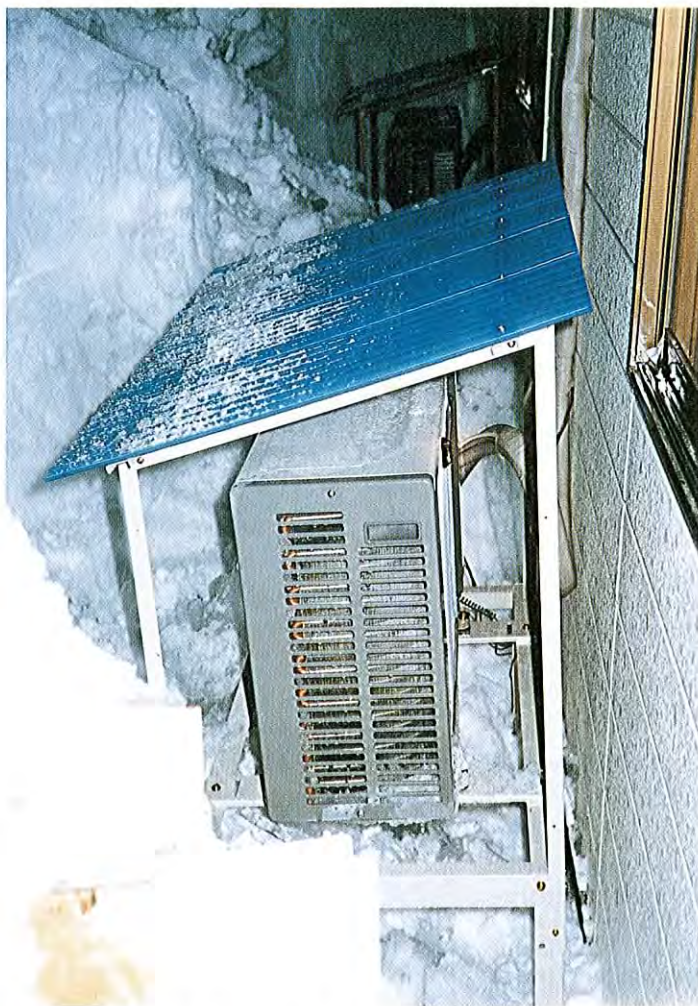


Fig. 4. The situation of outdoor unit - compressor and outdoor heat exchanger - installation

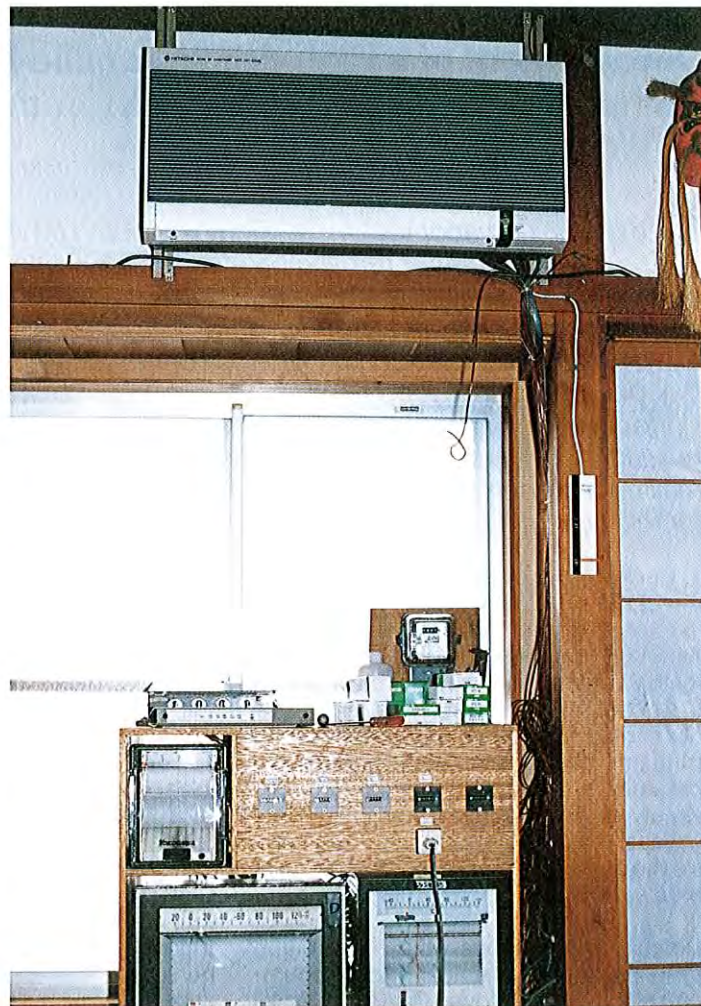


Fig. 6. The indoor unit - indoor heat exchanger - upper wall mount; downward blow



Fig. 5. The indoor unit - indoor heat exchanger - under wall mount; upward blow

value of the daily lowest temperature is lowest in the Tohoku District.

Therefore, if the room heating capacity of over 3.5 kWh at an ambient temperature lower than -7°C can be secured by the HPRAC, the prospect of this RAC is bright enough for use in cold districts, such as Tohoku District.

5. Conclusion

The number of units of the formerly developed model, with an auxiliary heater, in operation in 1984 in seven northeastern prefectures (including Niigata Prefecture) has amounted to 4,467 (11.3% of the 39,712 total RAC units in operation). It is popularized in the cold districts of Japan and its performance capabilities are highly regarded.

The new inverter-controlled model without heater (Figures 4-6), which has recently been developed and operated, has approximately solved the technical problems occurring in cold districts. It is considered to be at an almost satisfactory level, regarding performance and ratio of energy consumption. The authors hope that it will contribute more to the creation of comfortable living environments.

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Development of a High-Efficiency and High-Compression-Ratio Heat Pump System with Water as Working Fluid

The mechanical vapor recompression heat pump (MVR-HP), that recovers and reuses latent heat, is remarkably effective (COP often higher than 20) for concentration, distillation, and rectification processes. When using a conventional one-stage centrifugal compressor pressure ratios for water vapor of 1.5 to 2.0 are obtained and a maximum temperature of 135°C can be reached. This paper reports on the development of a highly efficient two-stage centrifugal compressor which is used for the compression of water vapor (Fig. 1). This advanced compressor allows a pressure ratio up to 4.5 and a maximum temperature up to 185°C. This will lead to extended applications for this type of heat pumps.

1. Background of the heat pump development

Although the heat pump's operational method of transferring heat from a colder to a hotter reservoir remains the same under various conditions, the means of realizing this method differ according to temperature level, range of temperature rise, variety of heat sources, applicable processes, etc. These differences characterize the system's design and the choice of hardware for an industrial heat pump.

Compared to various other heat pump systems, the mechanical vapor recompression heat pump type (MVR-HP) for latent heat recovery with a high thermal efficiency was thought to be better. It completely replaces conventional systems by means of newly developed component technology, improved performance, operation and service characteristics and reduced installation costs. This tendency

has been particularly important after the first oil crisis. This system, that recovers and reuses latent heat, is remarkably effective for processes such as concentration, distillation and rectification, where the COP is often higher than 20.

The compressor used for this MVR system is a specially designed steam compressor (Fig. 2).

The goal of this system is to achieve a high COP (10 to 20) by restricting the temperature rise to the minimum required for a refrigerant vapor which has a high latent heat (539 kcal/kg, 1 atmospheric pressure, 100°C in the case of water) thus also minimizing the power required. The essentials of this technology are as follows: To efficiently compress water vapor (which has a high specific volume and a comparatively low compression ratio) and raise its temperature; to efficiently exchange this latent heat with a smaller dif-



Fig. 1 Pilot plant of a high-efficiency and high-compression-ratio heat pump system with water as working fluid

ference in temperature; and, to combine both technologies. The combination of these technologies was successful (Fig. 3).

The application fields of these technologies, however, were limited with regard to the availability of process streams containing liquid water and similar materials. On the other hand, the extension of application fields of such a highly efficient system was considered to be important for various industrial processes. If the compression ratio of water vapor could be increased and result in higher temperatures, the application fields could be further extended. The recently developed high-pressure, two-stage, centrifugal compressor with a high-compression-ratio has permitted improving the technology and extending the application fields. As a basis for the development of heat pump technology, the above-mentioned technology is available not only for working with water vapor but also for working with fluids such as ammonia, Freon, hydrocarbons, etc. These technologies have been developed to provide higher temperatures and higher efficiencies. Although water is considered to be most suitable for heat pumps used for high-temperature ranges, it is desirable to produce optimum combinations for various conditions and systems.

2. System features

The main characteristics of this heat pump system are listed in Table 1.

First, a highly efficient two-stage centrifugal compressor is used for compression of water vapor. This is the first of its kind in Japan. This type provides high adiabatic efficiency, gives highly efficient flow control, has a wide range of capacity selection, and provides a large capacity. It also



Fig. 2 280 kW High-efficiency and high-compression-ratio two stage centrifugal compressor

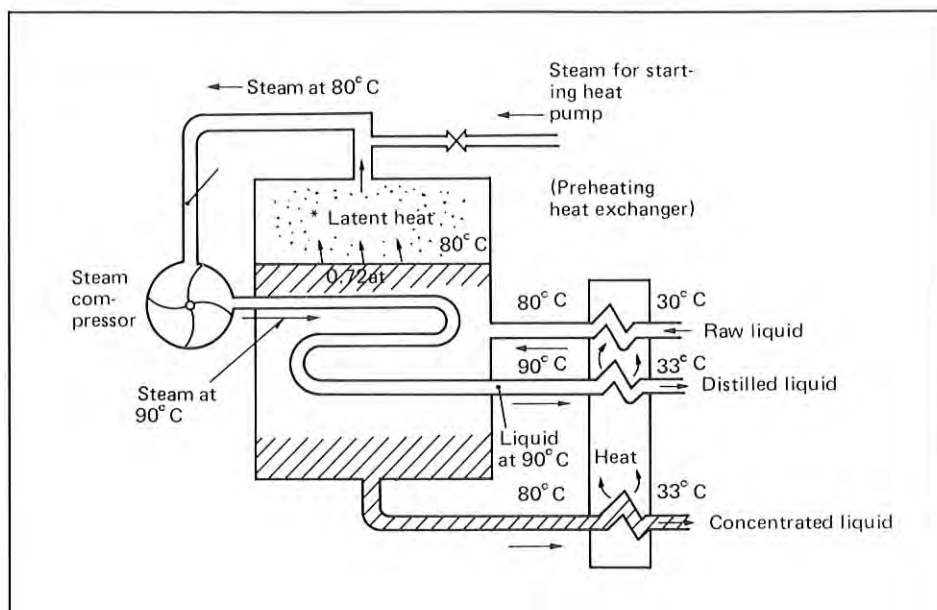


Fig. 3 Mechanical Vapor Recompression Heat Pump

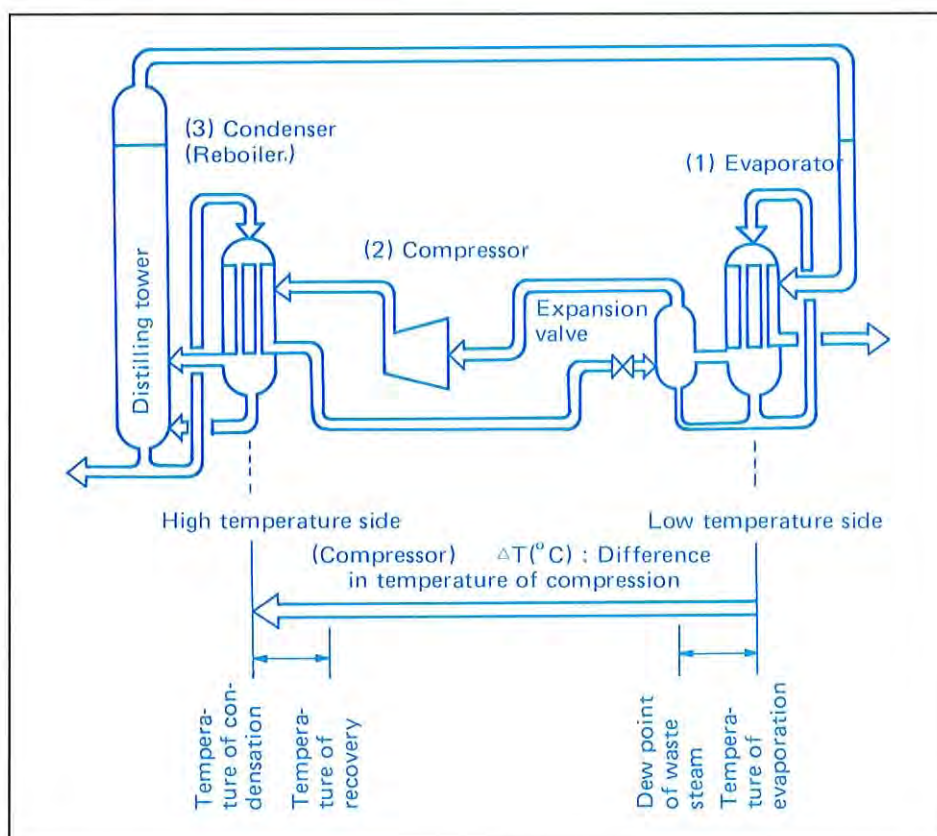


Fig. 6 System Flow

has broad applications because it can use high- or low-pressure vapor, depending on the temperature level of the heat source. In particular, the compression ratio of water vapor, which used to be 1.5 to 2.0 with the conventional single-stage centrifugal compressor, has been increased up to 4.5 when using the two-stage centrifugal compressor. This allowed the increase of maximum temperature produced by the heat pump up to 185°C.

Second, this system uses an improved heat exchanger. Instead of the conventional natural convection type or forced circulation type evaporator and condenser (which have comparatively high differences in approach temperatures when used for heating and heat absorbing sides of a heat pump) a falling film type evaporator (heat exchanger) is used in order to provide a temperature difference of 3 to 4°C, as a result of which the effective temperature lift of the heat pump has been

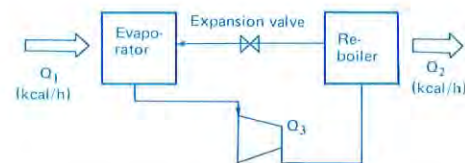


Fig. 4 Heat Flow of the Two-stage Compression Heat Pump

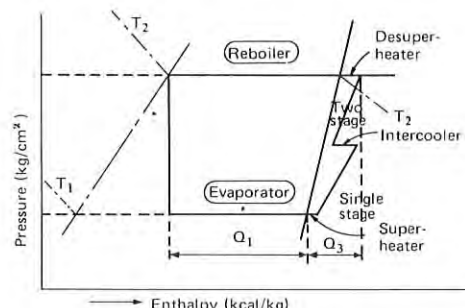


Fig. 5 Two-stage Compression Heat Pump Cycle

increased and the COP improved considerably.

In addition to the completely automatic control functions, compactly built hardware in the system design and improved economic advantage realized by equipment standardization are additional factors for the reduction in installation cost.

3. Operating principle of the two-stage compression type heat pump

Fig. 4 shows the heat flow of the two-stage compression heat pump cycle.

The water vapor discharged from the distillation tower top undergoes two-stage compression via a water working fluid.

	Newly developed system	Conventional type
(1) Working fluid	Water	Gas series working fluid, Freon, etc.
(2) Utilized temperature	Maximum 185°C	Maximum of about 130°C
(3) Compressor type	Two-stage turbo	Screw, reciprocating, etc.
(4) Heat exchanger type	Falling film evaporator	Natural convection or forced circulation
(5) Efficiency (COP)	COP: about 6	COP: about 4.5

Table 1 Comparison with Conventional Type of Industrial Heat Pump

and by increasing the temperature of the water vapor, the heat is transferred to a substance that is to be heated in the re-boiler (condenser).

Fig. 5 illustrates the above-mentioned heat pump cycle represented on a Mollier chart.

The two-stage compression begins with compression of water vapor that has left the evaporator. The suction side gas of the

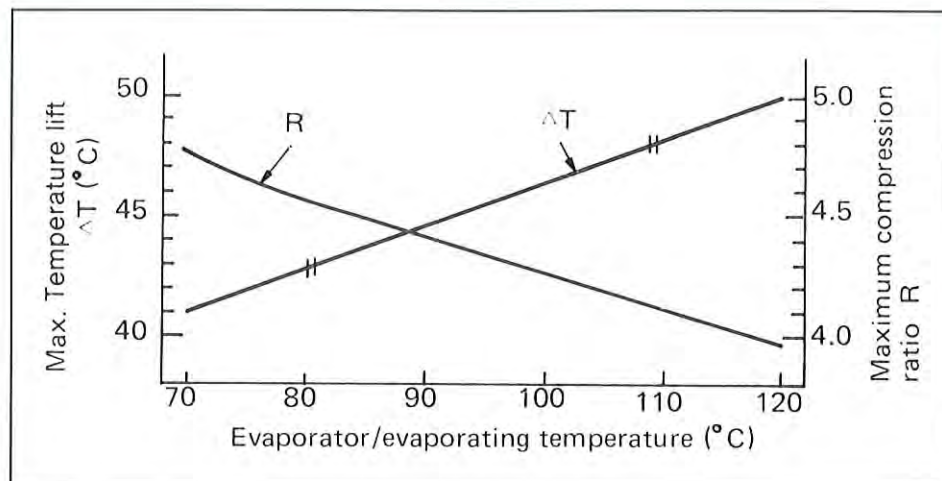


Fig. 7 Correlation between Evaporating Temperature, maximum Temperature and Compression Ratio

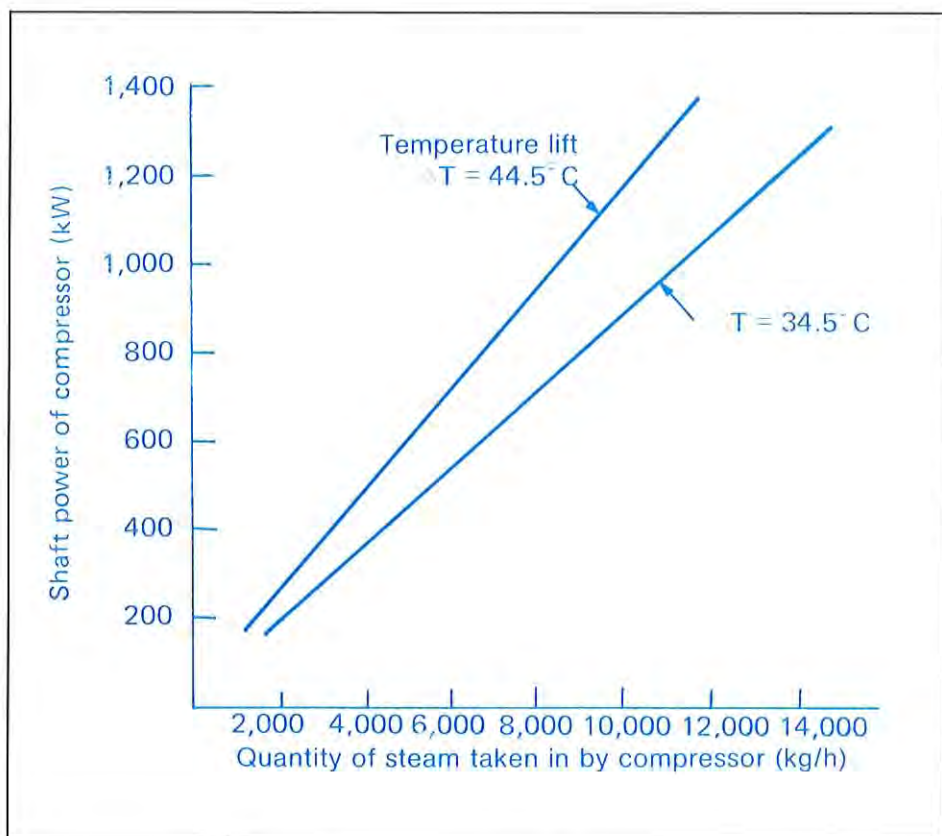


Fig. 8 Correlation between Quantity of Steam taken in by Compressor and Shaft Power of Compressor

	Quantity of steam consumption	Quantity of power consumption	Annual running Cost	Difference (merit)
Existing system	3,300 kg/h (decrease)	—	¥105,600,000	¥59,520,000/year
Heat pump system	—	320 kWh/h	¥46,080,000	

Table 2. Comparison of Running Costs (where annual operational hours: 8,000 h; unit price of steam ¥4,000/t; unit power charge ¥18/kWh)

first stage is superheated and the steam temperature from the outlet is decreased by the intercooler, and then enters the inlet of the second stage. When the water vapor at the outlet of the second stage is superheated, its temperature is decreased by the de-superheater and, additionally, the saturated steam is condensed in the re-

boiler so that it can impart heat to the substance that is to be heated. By repetition of this cycle, a high temperature product steam is maintained.

4. System flow

Fig. 6 shows the system flow. The product

vapor discharged from the distillation tower top retains 90% of the quantity of heat applied to the tower bottom. At present, all of the heat is removed via cooler, etc. From the viewpoint of energy economy it is important to recover and utilize this heat. The evaporator is a heat exchanger for extracting waste heat and causes the extracted heat to generate steam (which is the working fluid for the heat pump). This steam has its pressure and temperature increased by the two-stage compression type centrifugal compressor and enters the reboiler (which is also called a heater or condenser), where, by releasing its latent heat, it becomes condensed water and, after having its pressure reduced by the expansion valve, enters the evaporator. This operating cycle is repeated. For a period between September and October 1985 this heat pump system was installed in the pilot plant of the Chiba Factory (Ichihara City, Chiba Prefecture) of Maruzen Petrochemical Co. The experiments produced good results.

5. Major performance capabilities and selection of capacity

Fig. 7 illustrates the correlation between the evaporating temperature, the maximum temperature lift (difference between the temperature of condensation and evaporation), and the compression ratio (the ratio of the saturated vapor pressure of the suction side of the compressor to the saturated vapor pressure of the discharge side of the compressor).

Fig. 8 illustrates the correlation between the quantity of vapor taken in by the compressor and the shaft power of the compressor, which is depicted by using the temperature lift of the compressor as a parameter.

Fig. 9 illustrates a variety of two-stage centrifugal compressors corresponding to the quantity of vapor taken in by the compressor.

Temp. Industry	Below 100°C	100 to 150°C	150 to 183°C	Over 183°C
Food/tobacco	2.5	62.3	16.6	18.6
Textile	0.4	50.3	49.3	0
Wood/wooden products	1.1	9.3	6.6	83.0
Pulppaper	0	85.9	4.1	0
Chemical	4.8	26.9	50.0	18.8
Rubber products	0	26.3	53.4	20.4
Leather products	0	100.0	0	0
Ceramic, earth and stone	0	85.6	14.4	0

Table 3. Utilized Temperature Levels Classified by Industries (unit: %)

6. Economic feature (energy-saving effect)

The temperature for recovery of an organic solvent in a distilling tower in an industrial process differs from the normal boiling temperature of the solvent, which is the

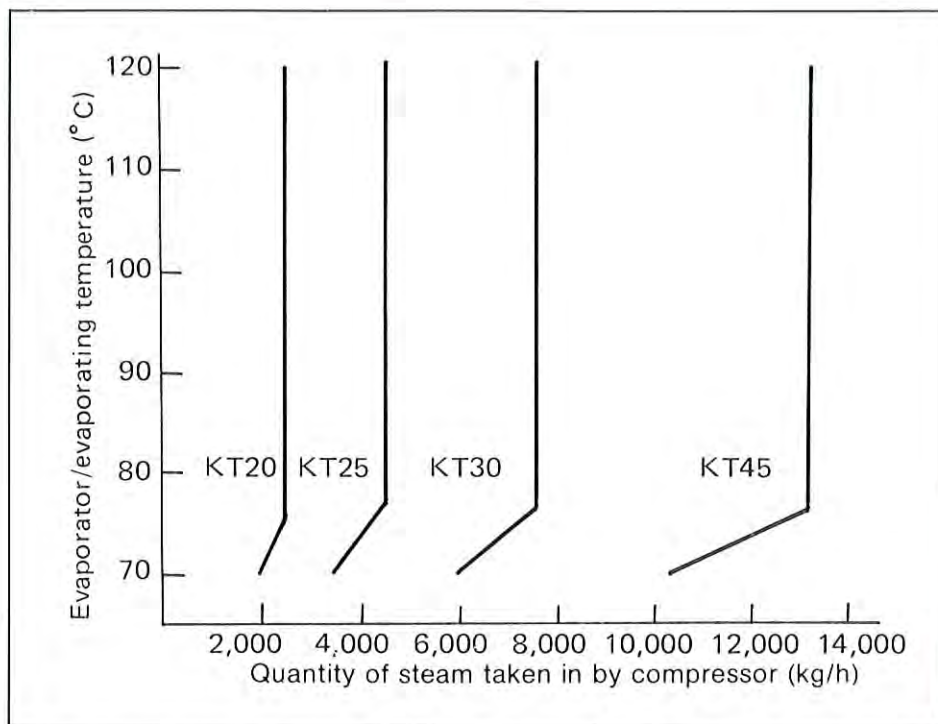


Fig. 9 Variety of Two-stage Centrifugal Compressors Corresponding to Quantity of Steam taken in by the Compressor

Petrochemical industry
High polymer chemical industry
Inorganic chemical industry
Chemical fertilizer industry
Dyeing rearranging industry
Sugar manufacturing industry
Dairy produce making industry
Stock raising and livestock produce processing industry
Marine produce processing ind.
Noodle industry
Refrigerated foods making ind.
Flavorings making industry
Alcoholics manufacturing ind.
Paper and pulp making industry
Iron and steel making industry

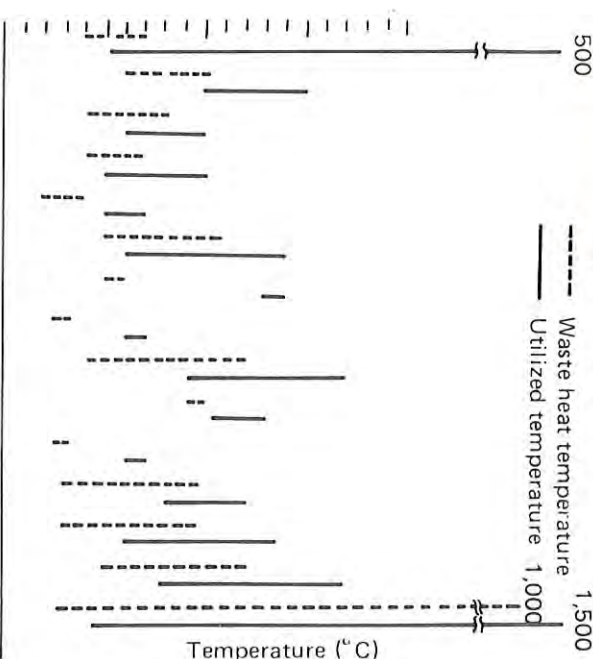


Fig. 13 Distribution of Waste Heat Utilizing Temperature Classified by Industries

product. Fig. 10 illustrates the heat balance when a heat pump is applied to the process of distilling and refining BTX (benzene-toluene-xylene). From this illustration it can be seen that the working steam obtained (from the solvent vapor through the evaporator on the heat recovering side) has a temperature of 110 °C. If this (water) steam is pressurized by a two-stage turbo type compressor with a shaft power of 230 kW, the steam temperature will increase to 145 °C. This is equivalent to steam generated in a boiler. This steam is condensed, and its latent heat of vaporization is used to heat the process fluid being distilled to 140 °C.

In case of the process which has introduced this heat pump, steam is used to heat the material only at the start-up time; the continuation of the operation is carried by the heat pump mechanism with rejected heat upgraded and reused in the refining process. The fuel-oiled boiler is replaced by the electric power of the compressor, which reduces the energy cost by 36%. The comparison of the costs is listed in Table 2.

If the waste heat from the tower top is 1,510,100 kcal/h and compared with heating by boiler steam, the annual energy-saving effect will be 800 m³ of crude oil.

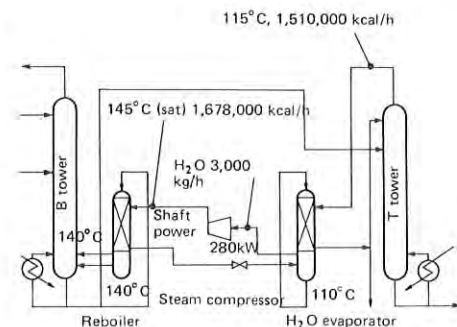


Fig. 10 BTX Distilling Process

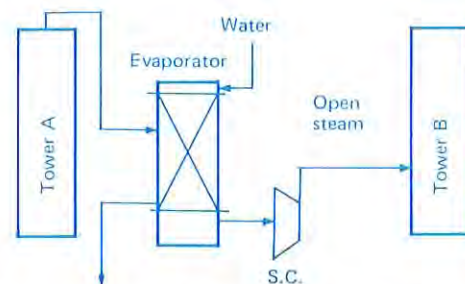


Fig. 11 Example of Application No. 1: Application to Position between Distilling Towers

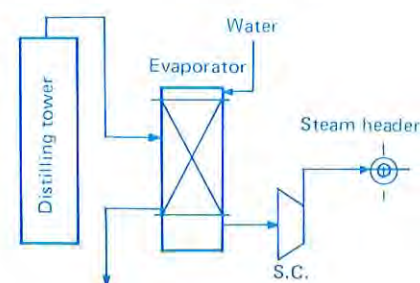


Fig. 12 Example of Application No. 2: Application to Tower Top Vapor

Although the financial advantage differs depending on the unit price of the existing industrial energy and the number of annual operational hours, it is estimated that the initial investment for plant and facilities can be amortized within about 18 months.

Example of Application No. 1 (Fig. 11) illustrates the following method: The waste heat from the tower top is recovered as low-pressure steam via the evaporator. This steam, after increase of pressure and temperature by the steam compressor, is directly supplied to the distillation or reacting tower for heating purpose.

Example of Application No. 2 (Fig. 12) illustrates the following method: The steam referred to in Example No. 2 is supplied to the header and used as process steam.

7. Field of application and feasibility

Fig. 13 shows the waste heat classified by industries and distribution of temperatures utilized. The temperature of waste heat streams ranges from 40 to 100 °C and chemical natures differ. They can be mixed and include such discharges of manufacturing processes as air, steam,

warm water, gases, etc. The temperatures utilized by manufacturing processes often range from about 50-60°C to about 160°C.

Table 3 lists the requirements of the temperature level and the ratio of total demand for some types of industry. Apart from industries such as iron and steel making which utilize high temperatures, many types of industry often utilize temperature levels from 100 to 150°C, which are considered to be suitable for widespread use of heat pumps.

For the recovery of waste heat from the tower-top vapor of organic solvents like benzene, toluene, xylene, isopropyl alcohol, ethanol, paraxylene, etc., in distilling and refining processes of chemical industries which utilize temperatures lower than 150°C, and for the recovery of waste heat from the tower-top vapor of various other organic solvent series materials, water-working fluid type heat pump systems are extremely effective.

In order to expand the use of full-scale heat pumps, it is necessary to solve the following technical problems:

- 1) Technology for equipment (hardware)
 - a) High-temperature output
The temperature level of 185°C will be further increased to a level of 300 to 350°C.
 - b) Expansion of increase in temperature lift
On the basis of economic advantage, the existing temperature lift of 40 to 50 K will be further increased.
 - c) Realization of high efficiency (improvement in COP)
Equipment that can realize high COPs, while retaining a wide range of temperature lift, will be developed.
 - d) Improvement of reliability and safety aspects
 - e) Further cost reduction
- 2) Technology for utilizing heat pumps (software)
 - a) Improvement of the adaptability to the process

- b) By considering the energy balance of the whole process, the concept of an optimum heat pump system as a total system will be formulated
- c) Improvement of system controlling features

With regard to the software aspects, the establishment of a design technology to adapt the heat pump to various process side conditions, and the development of optimizing procedures that include temperature conditions of the heat source and utilizer side, variations in heat load, material balance, heat balance, and the nature of a total system as they affect the economic advantage, will be carried out.

These researches and developments are the result of joint studies with Kimura Kakoki Co. The author expresses his thanks to those who have cooperated with him, publishing this report.

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New Design of an Ammonia-Water Absorption Cooling Process and Its Potential Applications for Heat Engineering

Three highly effective innovations utilizing internal cycle potentials are introduced to the familiar ammonia-water absorption cooling process. The result is dehydration of the refrigerant vapor in the rectification unit with no energy loss, a dramatic reduction in the driving energy required to transport the solution in the cycle and a stabilization of the refrigerant concentration in the evaporator, also with no energy losses. The mode of operation and design of the improved elements are described.

The implemented improvements make the newly designed cooling process significantly superior to current alternatives, especially with respect to heat pump operation. Hot water supply temperatures of 80°C at real thermal values of about 1.5 can be attained both in large heat pumps and low-capacity domestic heat pumps. The process may also be completely automated in an extremely simple way. With the exception of the transport equipment for the solution circulation, all steps can be implemented with currently available technology.

1. Introduction

The engineering objective is not to implement individually customized heat pumps but to realize a universally applicable and economical heat pump process. The realization of this objective is urgent and can only come from new ideas and approaches. This report deals with a future-oriented heat pump process derived from the standard ammonia-water absorption cooling process and satisfying the above objective.

A simplified block diagram of this standard process is shown in Fig. 1. It provides a useful basis for a general explanation of the principle involved and for identifying currently existing weaknesses of the process technology.

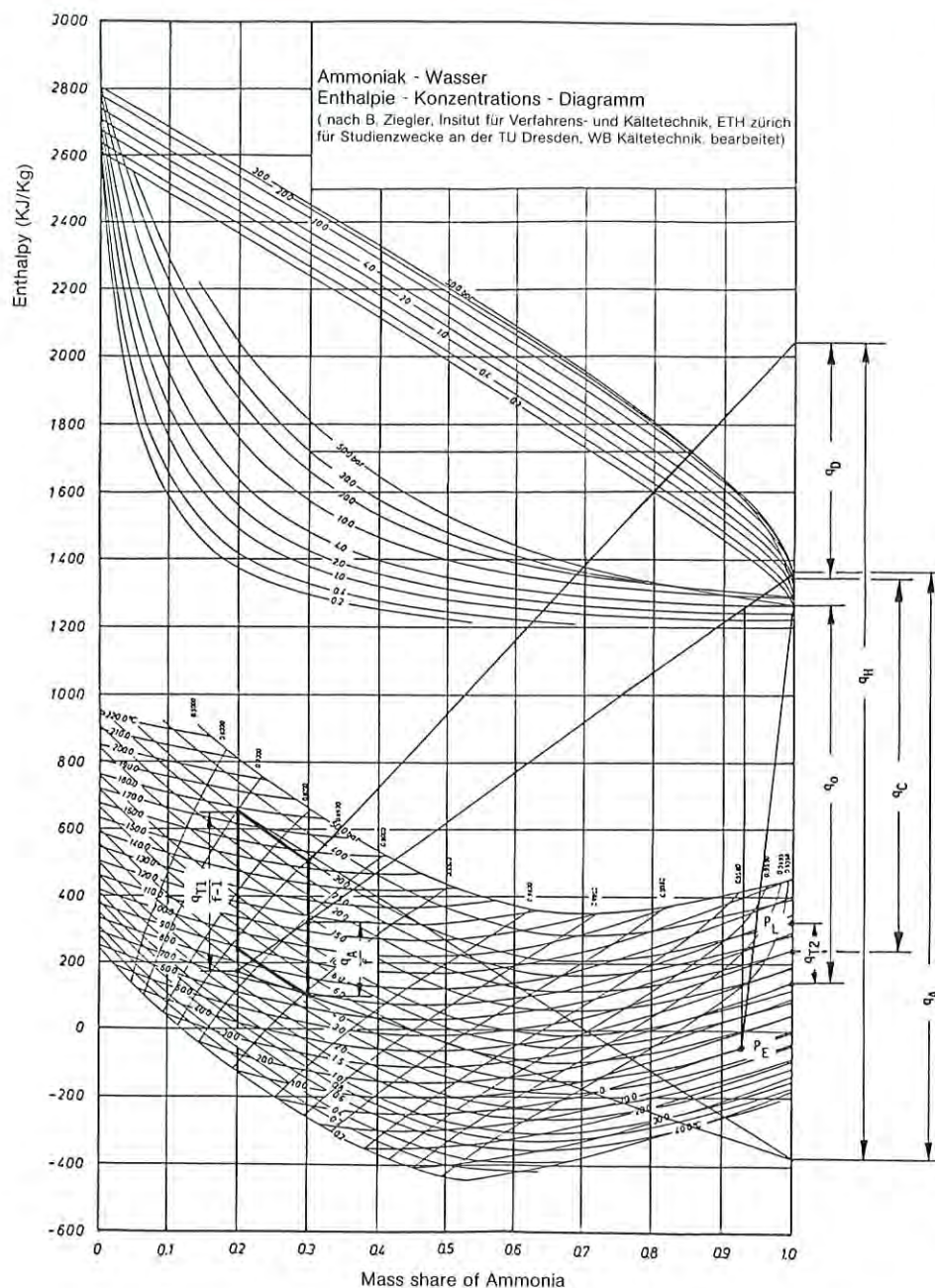
It can be seen from Fig. 1 that the refrigerant vapor generated in the evaporator is absorbed by the solution at the same pressure. The solution pump then raises the refrigerant-enriched solution to the condensation pressure and transports it

via the heat exchanger into the combined boiler-rectification unit. Here, the dissolved refrigerant is removed from the solvent in vapor form and thus dehydrated. The dehydrated refrigerant vapor leaves the head of the rectification unit and flows into the condenser, where it liquefies. The refrigerant condensate coming from the condenser then passes in countercurrent flow to the cold vapor outlet of the evaporator through the cold vapor cooler and enters the evaporator, already precooled, via the condensate expansion valve. Here it is evaporated again at reduced pressure. The stripped heated solvent is drawn off at the boiler base and passed in countercurrent flow to the cold enriched solution through the heat exchanger. The subsequently supercooled stripped solution finally reaches the absorber via the solution expansion valve and can again absorb cold vapor in a new cycle.

2. Identification of process weaknesses

Indications showing process weaknesses requiring improvement are given by the energy balance of the absorption refrigeration process and by the heat transfers in each individual unit. These may be specified clearly and with sufficient accuracy with the aid of an enthalpy concentration diagram for the ammonia-water system. Fig. 2 shows such a diagram for a closed-circuit representation of a heat pump.

The process operates in the solution cycle with the following parameters: degasification width 0.1, concentration of stripped



Legend

- q_D Specific dephlegmation heat
- q_0 Specific cooling capacity of evaporation heat
- q_C Specific condensation heat
- q_H Specific thermal heat
- q_A Specific absorption heat

- q_{T1} Specific waste heat of the stripped solution
- q_{T2} Specific subcooling of the condensate
- f Specific mass flow rate of the enriched solution
- $f-1$ Specific mass flow rate of the stripped solution
- P_E Point of evaporation
- P_L Point of liquefaction

Fig. 2 Representation of the standard absorption heat pump process in the enthalpy concentration diagram for the ammonia-water system

solution $\xi_a = 0.2$, concentration of enriched solution $\xi_r = 0.3$. The vapor pressures of the refrigerant are 30 bar in the condenser and 4 bar in the evaporator. The refrigerant vapor is concentrated in the rectification unit up to $\xi_d = 0.998$.

The process obtains the following external energy inputs: the heat from the boiler q_H , the ambient heat from the evaporator q_0 and mechanical energy from the solution pump L_p . The process outflows comprise

the liquefaction heat of the absorber, the condenser and the partial condenser cooler (q_A , q_C , q_D). Standard recovery methods include internal heat conversions in the heat exchanger q_{T1} and the cold vapor cooler q_{T2} . Although the newly designed absorption process does not dispense with this equipment, it is adapted to the new requirements.

In contrast, the high energy expenditure required for the ammonia dehydration in

the rectification unit is a source of criticism. It is characterized quantitatively by the liquefaction heat of the partial condenser q_D . This situation is unsatisfactory because the cooling potential of the excess mass flow of enriched solution in the heat exchanger suitable for rectification also remains unused or can be utilized only insufficiently in today's partial condenser configuration.

A further cause for criticism is the high expenditure of mechanical energy required to drive the solution pump, due to the efficiency chain. This situation is unsatisfactory because the reduced pressure of the refrigerant condensate and of the super-cooled stripped solution eliminates a quantitatively higher mechanical energy potential at other points.

A third weakness of the process is its imperfect mixture separation. Because the ammonia-water mixture cannot be completely separated with reasonable rectifier outlay, a small proportion of water always reaches the evaporator with the refrigerant condensate. To stabilize the refrigerant concentration in the evaporator, therefore, an output-synchronized refrigerant mass flow must be continuously removed from the evaporator and replaced by an equal mass flow of more highly concentrated condensate. At present, this mass flow can be controlled only manually, and no parameters which provide direct process indications are available to inform the operators of the correct setting values. As a consequence, installation outages and excessive refrigerant circulations are the rule. The lack of any displayable process parameters also explains why today's ammonia-water absorption refrigeration plants can only operate to a limited extent automatically, without human intervention.

3. Proposed solutions

3.1 Reduction of material separation energy

The idea of reducing the separation energy of the ammonia-water absorption refrigeration process by improved rectification equipment utilizing the cooling potential of the cold excess mass flow of the enriched solution is not new. However, well-known attempts to attain this objective failed in the last analysis because important physical relationships within the material separation process were not taken into account. Thus, one approach was to separate the excess mass flow prior to the heat exchanger and feed it separately to the rectifier column via a middle floor in the boiler section¹. Another suggestion was to introduce the separated excess mass flow into the column at the head of the upflow section and feed it downwards in a medium-separating tube within the column as far as the entry point of the mainstream solution heated by the heat exchanger. The excess mass flow then heats up to the temperature of the mainstream solution².

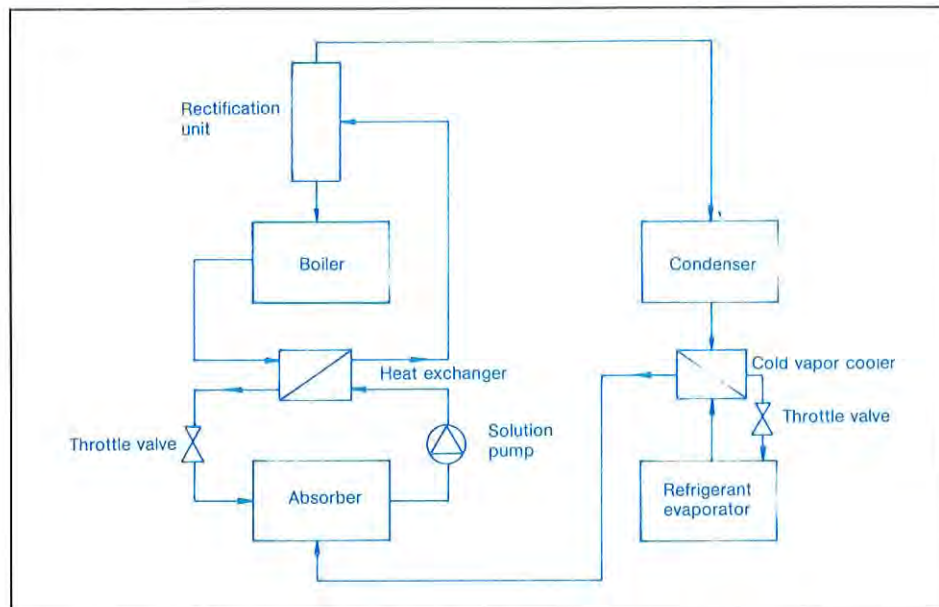


Fig. 1 Block diagram of standard absorption cooling process

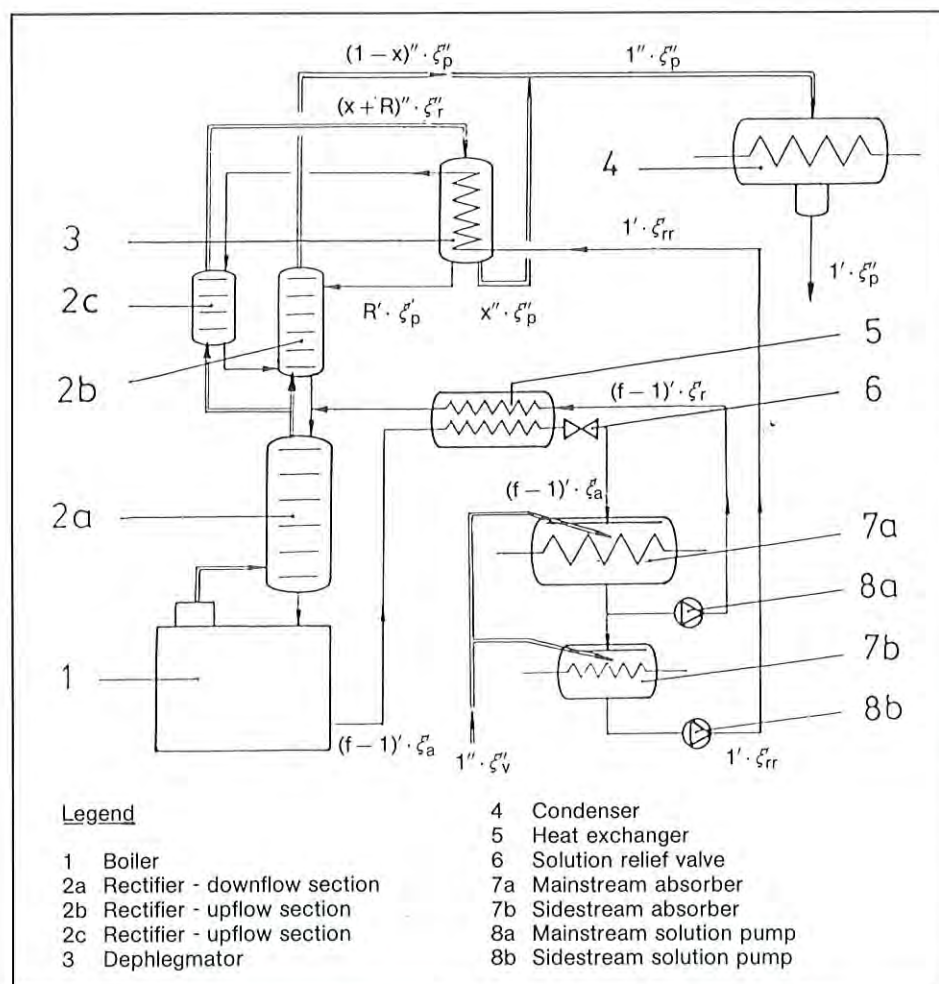


Fig. 3 Process flowchart of improved rectification unit

In the first case, two solution streams with different rectification potentials - the dephlegmator reflux and the cold excess mass flow of the enriched solution - are mixed, for example, in the upflow column. Both solution streams have different concentrations as well as different temperatures at the mixing point prior to mixing. The mixing process itself takes place exo-

thermically, the combined solution stream having a reduced concentration and an increased temperature. Ultimately, therefore, the mixing process neutralizes the rectification potential present in the separate solution streams.

In the second case, the excess mass flow heats up in the column without contribut-

ing to the rectification, so that the quantity of heat absorbed must be additionally supplied to the generator. The cold excess mass flow could contribute to the rectification in this case only if it can be ensured that it either supercools only the dephlegmator reflux flowing downstream in co-current flow or condenses out only countercurrent refrigerant vapor. But in the latter case, the purified refrigerant vapor must no longer come into contact with the solution reflux. However, a mixed operation, i.e., cooling of both the solution reflux and the refrigerant vapor, remains without effect.

Like the two attempts mentioned above, the physically effective solution to this problem makes use of equivalent heat exchange in the heat exchanger. The stripped solution is then supercooled to the extent that no spontaneous evaporation is possible in the expansion valve and the entire excess mass flow of cold enriched solution is available for the dephlegmation heat transfer.

The configuration of rectification units shown in Fig. 3 allows complete transfer of the dephlegmation heat in the absorption process by the cold excess mass flow even for very small enrichment concentrations in the solution cycle.

On the vapor side, the rectification unit (2) is connected directly after the boiler (1). Rectification downflow section 2a is followed by two parallel-operated upflow sections 2b and 2c. A dephlegmator (3) on the evaporation side is additionally connected after upflow section 2c. The evaporation streams of upflow section 2b and of the dephlegmator (3) attain product vapor quality and are fed to the condenser (4) for liquefaction.

The stripped solution leaves the base of the boiler (1) towards the heat exchanger (5), its pressure is reduced by expansion valve (6) and it is fed into the mainstream absorber (7a). There, the stripped solution is enriched with refrigerant. Subsequently, the solution pump (8a) pumps an equivalent sidestream of enriched solution through the heat exchanger (5). The enriched solution heated in the heat exchanger is fed at the head of the downflow section (2a) into the rectification unit (2).

The excess mass flow of enriched solution is fed to sidestream absorber 7b and further enriched there with refrigerant at the same pressure. Sidestream pump 8b then pumps this more concentrated refrigerant solution through the dephlegmator (3) and subsequently feeds it at the head of upflow section 2c into the rectification unit.

The refrigerant vapor coming from downflow section 2a is fed into the base of upflow section 2c and added in countercurrent flow to the more concentrated excess solution entering the column from above. The rectification potential of this solution

Transport Equipment	Centrifugal Pump	Compression Pump	Axial Piston Pressure Exchanger	
			Vapor Drive	Auxiliary Pump
Transport pressure difference bar	4 --- 31.5	4 --- 31.5	4 --- 31.5	4 --- 31.5
Transport flow rate (l/h)	183	183	393	375
Theor. energy requirement (W)	140	140	40 (14)**	11
Efficiencies				
Pump	0.11	0.4	0.85	0.11
Motor	0.75	0.6	---	0.5
Boiler*	0.93	0.93	0.93	0.93
Energy consumption (W)	1,700 el	585 el	47.2 therm	200 el
Energy costs (DM/a)	1,564	538	15.2 (6.1)**	184

Table 3.1 Comparison of alternative transport equipment for solution exchange in domestic absorption heat pumps (heat output 20 kW, 4000 h/a)

Transport Equipment	Centrifugal Pump	Compression Pump	Rotary Piston Pressure Exchanger	
			Vapor Drive	El. Drive
Transport pressure difference bar	4 --- 31.5	4 --- 31.5	4 --- 31.5	4 --- 31.5
Transport flow rate (m ³ /h)	27.5	27.5	57.0	56.4
Theor. energy requirement (W)	21,000	21,000	600	600
Efficiencies				
Pump	0.52	0.85	0.85	0.85
Motor	0.92	0.88	1.0	0.7
Boiler*	0.96	0.96	0.96	0.96
Energy consumption (W)	43,900 el	28,100 el	1,080 therm	1,010 el
Energy costs (DM/a)	67,650	43,300	580	1,557

Energy Costs:

Electricity -- 0.23 DM/kWh el
Gas -- 0.08 DM/kWh therm.

* Incl. exhaust gas heat exchanger

** With complete utilization of pressure potential

Table 3.2 Comparison of alternative transport equipment for solution exchange in industrial absorption heat pumps (heat output 3 MW, 6700 h/a)

stream is fully utilized by this means, and the refrigerant vapor reaching the dephlegmator (3) already has a higher and stabilized concentration. Here, the refrigerant vapor is concentrated to the required product purity by condensing out the components containing solvent. The resulting condensate is separated and fed as solution reflux to the head of upflow section 2b. The quantity and concentration of this solution reflux is suitable for the vapor leaving this upflow section also to attain the required product purity. Even at very low enrichment concentrations, the refrigerant vapor can be dehydrated in this way without external cooling and thus without incurring losses.

Since this rectification concept involves additive product streams, the new ab-

sorption refrigeration process may no longer be represented as a closed system in the enthalpy concentration diagram. In this case, the heat conversions of the equipment may be determined with the aid of this diagram only by means of individual balances and intermediate calculations.

The examples of results for an absorption heat pump are shown in Table 5.3, Section 5, for a refrigerant evaporation temperature of 1°C and a condensation temperature of 68°C. A hot water supply temperature of 80°C is possible under these conditions.

These results clearly bring out the advantages of the sidestream absorber. It does

		Evaporator	Condenser
Pressure	bar	4	30
Temperature	°C	1	68
Concentration	%	94	99.5
Refrigerant mass flow	kg/s	1	1
Refrigerant bleed	%	9.0	0

Table 5.1 Refrigerant Process Data

Weak solution:	
Concentration	20 %
Mass flow	6.2 kg/s
Temperatures:	
Boiler outflow	177 °C
Absorber inflow	75 °C
Enriched solution of mainstream absorber:	
Concentration	30 %
Mass flow	7.084 kg/s
Mass flow to heat exchanger	6.2 kg/s
Mass flow to sidestream absorber	0.884 kg/s
Temperatures:	
Absorber outflow	71 °C
Heat exchanger outflow	152 °C
Enriched solution of sidestream absorber:	
Concentration	38 %
Mass flow	1.0 kg/s
Temperatures:	
Absorber outflow	55 °C
Dephlegmator outflow	128 °C

Table 5.2 Solution circulation process data

Evaporator	1,105
Condenser	1,080
Mainstream absorber	1,615
Sidestream absorber	250
Boiler	1,840
Heat exchanger	2,885
Cold vapor cooler	110
Evaporation cooler	116
Dephlegmator	355
Exhaust gas heat exchanger	165

Table 5.3 Heat exchanges in the equipment (kJ/s)

5.4.1	Theoretical thermal value $Tv_{th} = 2945/1840 = 1.60$
5.4.2	Real thermal value for direct-fired boiler with exhaust gas heat exchanger in cooling water cycle $Tv_{real} = (2945 + 165)/2115 = 1.47$
5.4.3	Real thermal value for direct-fired boiler with regenerative burner air pre-heating $Tv_{real} = 2945/1957 = 1.50$

Table 5.4 Heat pump thermal values

in any case lead to no overall increase in absorption heat dissipation. The final enrichment concentration in the mainstream solution can be reduced by the increased rectification potential of the excess solution with higher refrigerant

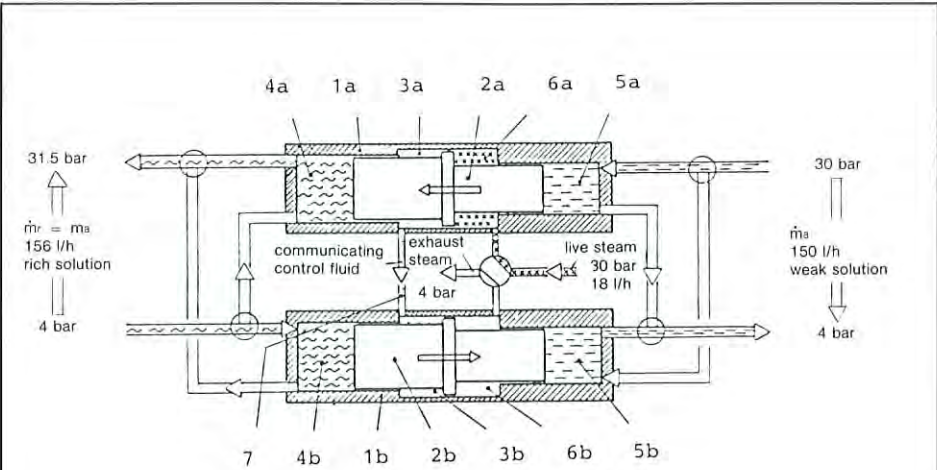


Fig. 4a Mainstream pressure exchanger with auxiliary vapor drive

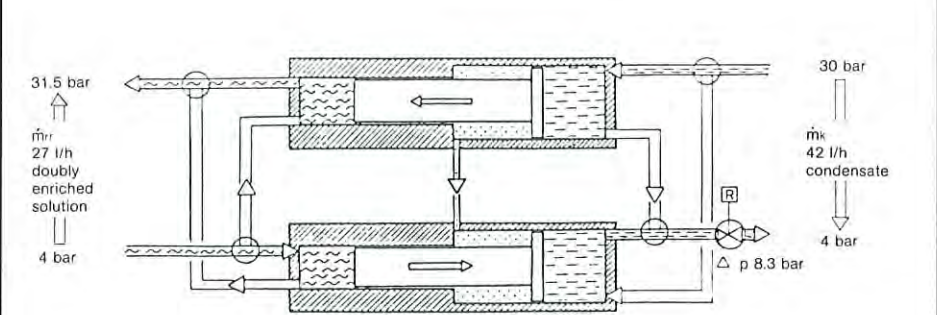


Fig. 4b Sidestream pressure exchanger without auxiliary vapor drive

Fig. 4 Principle of continuously transporting axial piston pressure exchanger for 20 kW domestic heat pump

	Standard Process	New Process
Thermal value	1.33	1.5
Energy requirement for solution circulation	43.0 kW	0.9 kW
Electricity consumption for solution circulation	301,000 kWh/a	4,320 kWh/a
Fuel consumption:		
Heat pump	9,022 MWh	8,000 MWh
Boiler	685 MWh	685 MWh
Mean annual utilization degree	1.30*	1.45

* For concentration-stabilized evaporator

Table 6.2 Operating value comparison

enrichment so that ultimately the refrigerant outflow temperature can be increased.

3.2 Utilization of existing drive potential with a new type of pressure exchanger

Up to now, the drive energy requirement of the solution pump has been neglected for

both absorber refrigeration plants and heat pumps. However, as is shown in Table 3.1, the drive energy requirement of the solution pump increases to attain considerable values particularly for low-capacity domestic heat pumps due to the efficiency chain of the transport equipment. A significant part of the primary energy-side cost saving must therefore be

Condenser inflow	50 °C
Condenser outflow	62.8 °C *
Absorber outflow	78.4 °C
Exhaust gas heat exchanger outflow	80 °C

* Condensation point: 68 °C

Table 5.5 Cooling water heating in heat exchangers

Heating power	2.5 MW
Annual heat requirement	12,600 MWh
Cooling water heating	50/80 °C
Heat pump proportion of annual energy production	12,000 MWh
Annual operating hours	7,000 h
Annual full capacity hours	4,000 h

Table 6.1 Common operating conditions

used to cover the operating costs of the solution pump.

It becomes apparent from Table 3.2 that for large heat pumps with high annual capacity utilization too, the energy costs for operating the solution pump can become considerable in spite of the relatively good efficiency chain.

In contrast, a higher mechanical energy potential is eliminated in the pressure reduction of the separated liquid streams than would be required for raising the pressure of the enriched solution stream. This potential can be exploited by a pressure exchanger operating on the compression principle so that the actual drive energy requirement for the solution circulation can be dramatically reduced. The most important precondition for this is that the liquid streams are sufficiently supercooled before their pressure is reduced, in order to avoid spontaneous evaporation.

The hydraulic principle of an axial piston pressure exchanger of the kind which may be used in domestic heat pumps is shown in Fig. 4. The modes of operation of a mainstream pressure exchanger with auxiliary vapor drive and a sidestream pressure exchanger using no external energy will be described at a later stage. This latter exchanger utilizes the drive potential of the refrigerant condensate to

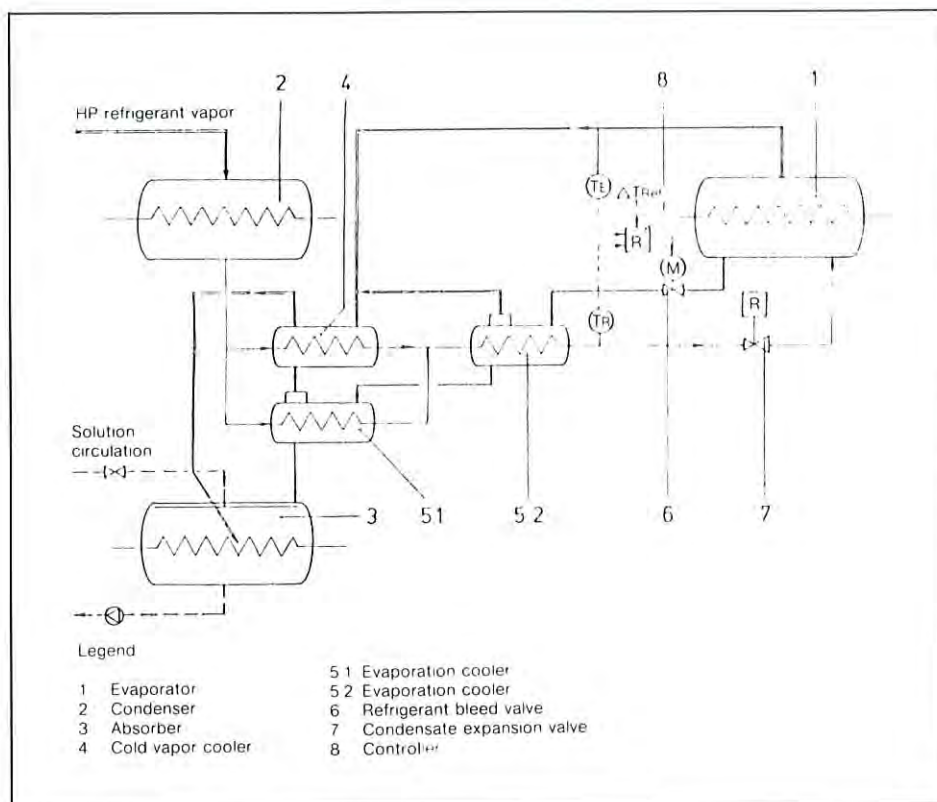


Fig. 5 Process flowchart of concentration stabilized refrigerant evaporator

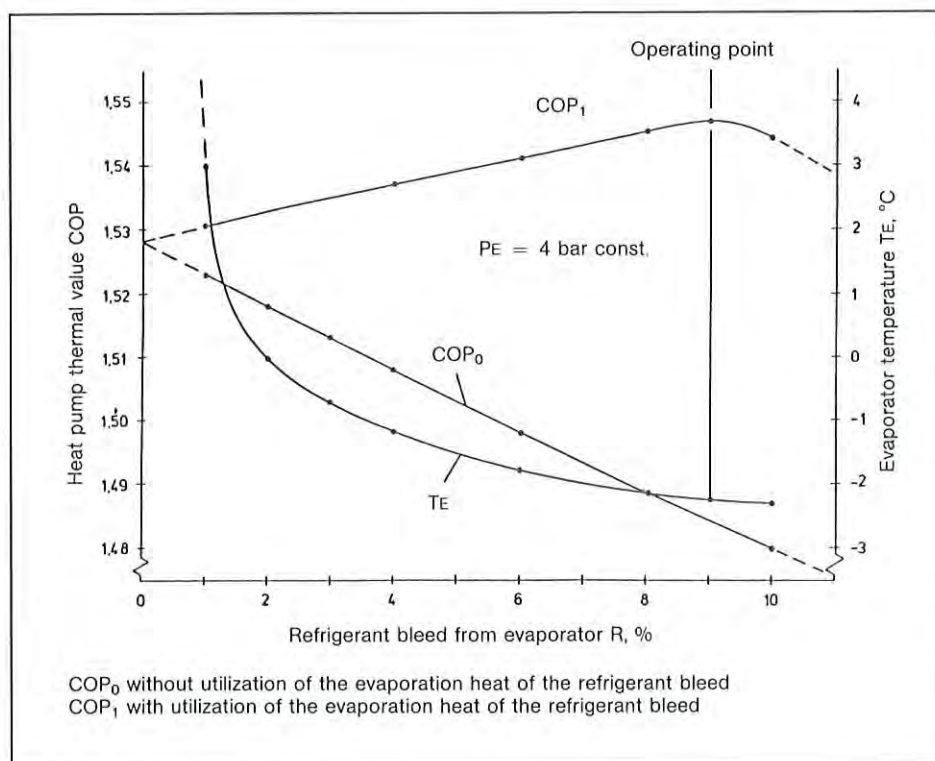


Fig. 6 Effect of condensate supercooling and regulated refrigerant bleed from the evaporator on the heat pump thermal value

transport an equivalent quantity of more enriched excess solution and to increase its pressure.

A surface-adapted piston (2a) moves within a cylinder (1a). Four operating chambers (3a - 6a) result from the configuration of the piston and the interior contour of the cylinder (1a). These chambers

expand on one side and contract on the other when the piston (2a) moves axially. Continuous transport is ensured by hydraulically connecting two equal-volume cylinders (1a and 1b) within which the pistons move back and forth in alternating rhythm. The opposing motion of the two pistons (2a and 2b) is ensured by linking the chambers (3a and 3b) via a tube (7)

and by filling both chambers and the tube (7) with an incompressible fluid.

The enriched solution flows into chambers 4a and 4b and the supercooled stripped solution into the opposite chambers (5a and 5b). If piston 2a now moves to the left starting from its right end position, then piston 2b moves to the right from its left end position. Chamber 5a is then filled with supercooled stripped solution at the prevailing condensation pressure and on the other side enriched solution is removed from chamber 4a against an equal or higher pressure. In the same operating cycle, chamber 4b in cylinder 1b is filled at the prevailing absorber pressure with enriched solution and the weak supercooled solution is removed at the same pressure from chamber 5b. When piston 2a reaches the left end position, the direction of motion of the two pistons is reversed.

To overcome the mechanical frictional losses of the pistons and the external frictional pressure losses of the solution streams, the mainstream axial piston pressure exchanger is additionally driven with refrigerant vapor. The two chambers 6a and 6b are alternately available for this purpose. Thus, when chamber 6a is filled with vapor from the condenser, chamber 6b is open to the absorber and piston 2b therefore is unilaterally relieved, so that both pistons can be moved from any position by vapor injection.

The suction and pressure flows of the stripped and enriched solutions are inhibited with respect to each other by pressure-operated backflow blocks. The speed of the continuously transporting mainstream pressure exchanger may be set via the vapor feed.

The sidestream pressure exchanger is shown in Fig. 4b. It operates without external auxiliary energy. In this case, the chambers required for vapor operation are obviated. To avoid spontaneous evaporation of the condensate in the chamber during the pressure expansion phase, the condensate must be more highly pre-cooled and a throttle valve must additionally be incorporated into the condensate outflow pipeline in which any existing excess expansion potential is eliminated. The transport speed of the sidestream pressure exchanger can thus be adjusted via the pressure difference of the throttle valve.

The use of axial piston pressure exchangers presents problems in large installations due to the volumetric flows which must be processed. Single-shaft rotary piston pressure exchangers are particularly suitable for this application. Their transport capacity is significantly greater and can be modified via the rotation. Rotary piston pressure exchangers have the additional advantage that the entire drive potential can be utilized, because the force of all the solution streams

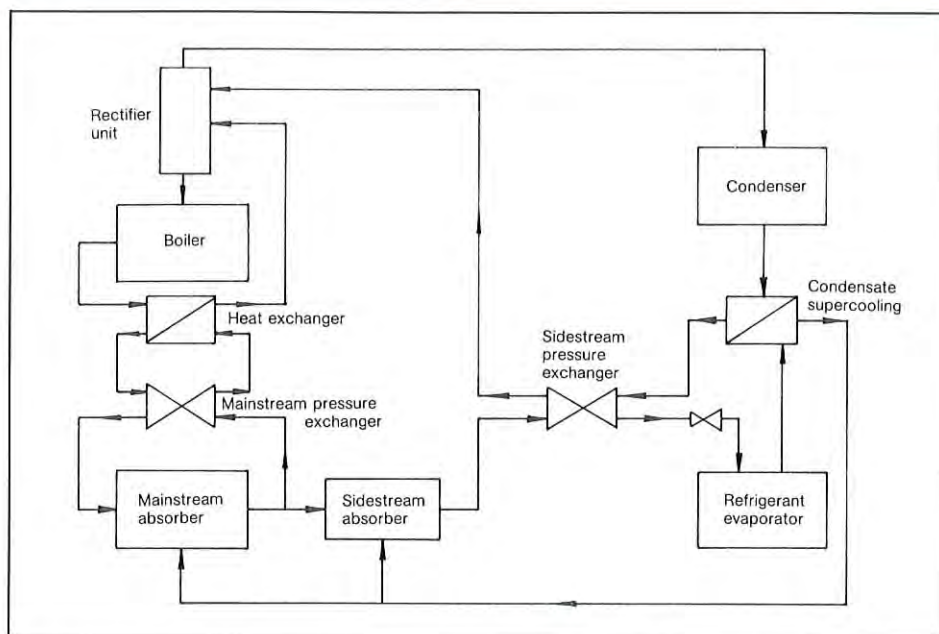


Fig. 7 Block diagram of improved absorption cooling process

Ultimately, this large heat pump requires less than DM 600 per year for maintaining the solution circulation. The electric auxiliary drive of the rotary piston pressure exchanger is also an attractive alternative for large installations. For small installations, however, its poor efficiencies tend to increase costs, so that only the axial piston pressure exchanger with auxiliary vapor drive can be considered. For medium-sized installations, the axial piston pressure exchanger may be improved by conveying a sidestream from the mainstream solution via a sidestream pressure exchanger. In this case, the throttle valve in the condensate outflow is not needed.

It is particularly advantageous to use the auxiliary vapor drive for these two types of pressure exchanger designs, because no rotating or moving parts need be run to the outside. These pressure exchangers thus have an extremely long-term seal. Operating material losses are, therefore, practi-

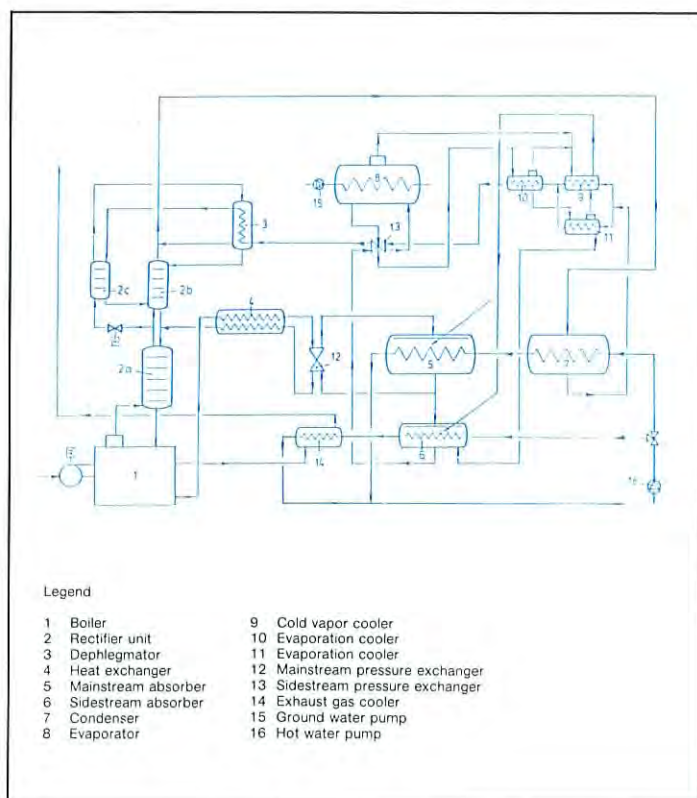


Fig. 8 Process flowchart of a domestic heat pump for thermal outputs of 20-50 kW

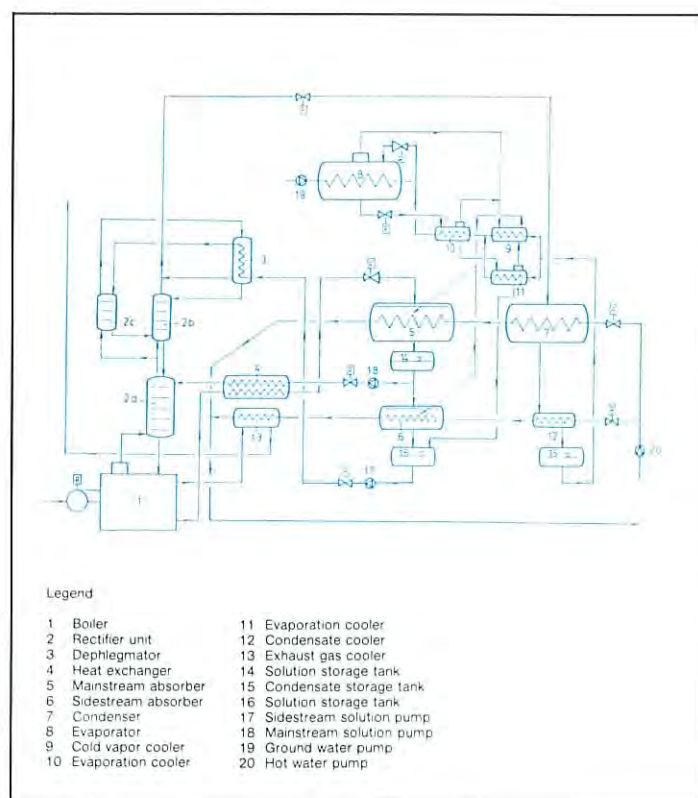


Fig. 9 Process flowchart of an industrial heat pump for thermal outputs ≥ 100 kW

affects a single shaft. The auxiliary drive powers are consequently small, so that not only auxiliary vapor drives but also auxiliary drives powered by electric motors can be used economically. Operational comparisons of the available alternative transport equipment were made for two different installation sizes. Table 3.1 shows this comparison for a domestic heat pump with a medium annual capacity load and Table 3.2 the comparative figures for a large heat pump with high annual capacity.

With regard to annual operating costs, the pressure exchanger with auxiliary vapor drive has the best showing, quite independently of its performance. Thus, the annual operating costs for a domestic heat pump for solution circulation are as low as DM 20 for 4000 hours operation under full load.

The use of a rotary piston pressure exchanger for a 3 MW heat pump with 6700 hours annual operation at full load produces a saving in operating costs in the region of DM 67,000 per year.

cally excluded. In addition to this, the volumetric distribution of the solution streams by the pressure exchangers simplifies the control effort required for automatic operation of these installations. These operating advantages, together with the cost benefits achieved, make the development of these conveying devices appear very promising. Their use is limited not only to absorption refrigeration installations and heat pumps; they can also be used advantageously in high pressure distillation plants of other types.

3.3 Introduction of concentration-regulated refrigerant evaporator

The deficiencies of today's absorber plants due to the subjective regulation of the concentration in the evaporators were clearly brought out by two operational analyses which were performed independently of each other. Thus, an absorption refrigeration plant for combined cooling and heating was shown to consume almost double the optimum heating energy, the major part of this being due to an excessive refrigerant outflow³. In the second case, a remote heat pump installation, the same deficiency gave rise to a 12% higher annual energy consumption⁴. Both results show how important an automated concentration maintenance in the evaporator is for future installations. For this reason, a highly informative and reliable process parameter must be sought and implemented for process automation.

This problem is solved by utilizing not only the cooling potential of the cold vapor outflow from the evaporator but also that of the refrigerant outflow for supercooling the condensate feed. Additionally, the temperature of the supercooled, not yet pressure-reduced condensate feed and the boiling temperature of the refrigerant in the evaporator are measured, and the mass control valve of the refrigerant outflow is automatically regulated to achieve a minimum difference between these two temperatures.

This temperature difference provides us with the sought-after informative process parameter. It is independent of the saturation vapor pressure of the boiling refrigerant and dependent only to a small extent on the evaporator capacity. As long as the supercooling of the refrigerant condensate feed and the cooling potential of the refrigerant outflow from the evaporator (bleed) are in equilibrium, this type of concentration stabilization involves no energy losses, i.e., the required refrigerant outflow depends only on the possible degree of supercooling of the refrigerant condensate. A specific stabilized refrigerant concentration in the evaporator then results from every required refrigerant bleed. This concentration can be maintained via the minimized temperature difference.

Depending upon the equipment outlay, this temperature difference can assume values between 2 and 10 K. Its measurement and process implementation is cost effective and reliable. Fig. 5 is a process flowchart showing this concentration maintenance.

The refrigerant condensate leaves the condenser (2) in two sidestreams: one of these, adapted to the cooling potential of the cold vapor, flows through the cold

vapor cooler (4) and the remaining stream flows through the evaporation cooler (5.1). Both condensate streams are subsequently combined and passed through the evaporation cooler (5.2). From here the refrigerant condensate reaches the evaporator (1) via the condensate expansion valve (7).

The refrigerant bleed from the evaporator (1) is initially passed via the control valve (6) to the evaporation cooler (5.2) and partially evaporated in this unit. The boiling refrigerant bleed then reaches the evaporation cooler (5.1) and is almost entirely evaporated there. A remainder solution stream of less than 0.5% finally reaches the absorber, almost at the concentration of the absorber solution (3).

The quantities of spent vapor from the evaporator (1) and from the evaporation cooler (5.2) are fed to the cold vapor cooler (4) and passed in the jacketed chamber in countercurrent flow to the sidestream of the refrigerant condensate feed. The spent vapor from the evaporation cooler (5.1) can either be passed directly to the absorber (3) or fed additionally into the cold vapor cooler at a point having the same temperature (4).

This heat exchange configuration can be used to cool the refrigerant condensate feed prior to its pressure reduction by the condensate expansion valve (7) up to a minimum residual temperature excess of 4 - 5 K, referred to the boiling temperature of the refrigerant in the evaporator (1).

The temperature difference between the refrigerant condensate feed before the condensate expansion valve (7) and the cold vapor leaving the evaporator (1) is measured at the two temperature measuring points T_v and T_k . This temperature difference is kept constant by controller (8) via the mass flow at the refrigerant bleed valve (6) in the entire partial load region of the heat pump.

Fig. 6 shows the thermal value profile of the heat pump for the new absorption heat pump process as a function of the refrigerant bleed from the evaporator for an evaporator pressure of 4 bar.

The upper curve applies to the newly implemented complete condensate supercooling with utilization of the common cooling potential of the refrigerant bleed and cold vapor outflow. In contrast, the lower curve applies to the unrealistic case of exactly adjusted refrigerant bleed in the hitherto usual partial condensate supercooling process in the cold vapor cooler.

But Fig. 6 also shows the effects of the refrigerant bleed on its evaporation tem-

perature at constant pressure. Whereas the evaporation temperature increases with decreasing refrigerant bleed as a result of the increasing dilution of the refrigerant, with increasing refrigerant bleed it asymptotically approaches the evaporation temperature of the pure refrigerant.

This again means that by utilizing the mass-controlled refrigerant bleed for supercooling the condensate feed, an optimum evaporator operating point is established. At this point, the overall process operates at a maximum thermal value and the refrigerant evaporation at optimum pressure. The evaporator is operated almost as a unary evaporator via the controlled temperature difference.

The control engineering concept of this concentration-stabilized evaporator is simplified as soon as the refrigerant condensate feed into the evaporator is implemented via a volumetrically conveying pressure exchanger. A mass flow of refrigerant bleed can then be transported synchronously with the output from the evaporator into the evaporation cooler in the same operating cycle. Supercooling of the refrigerant condensate feed in the reverse direction is also necessary to ensure reliable operation of the pressure exchanger.

4. The new heat pump concept in its totality

The process improvements discussed in Section 3 may be integrated step by step into the new overall concept. Thus, both existing installations and new ones can already be equipped with the improved rectification equipment and the concentration-stabilized evaporator. Similarly, the sidestream absorber may be retrofitted if required. For process automation, such retrofitting effectively represents an extension carried out in existing technology. All retrofittable process components are state of the art.

In contrast, the pressure exchangers designed to ensure energy-saving circulation of the solution are new and are therefore suitable for introduction over the longer term. Both the axial piston pressure exchangers used for domestic pumps and the rotary piston pressure exchangers used for large installations must be specifically developed for these applications. Their availability means a considerable reduction in control-engineering and operational outlay for fully automatic operation of the installations, because the mass flows can be unequivocally assigned. In this case, the burner of the direct-fired boiler and the pressure exchangers operate synchronously, controlled by the supply temperature of the heating water.

Fig. 7 shows a simplified block diagram of the newly designed absorption refrigeration processes. Its aim is to show clearly the essential modifications compared with the standard process (Fig. 1). It shows the twofold refrigerant absorption in the mainstream and sidestream absorbers with their separate solution feeds into the rectification unit and the energy-saving solution exchange via the pressure exchangers. However, the more intensive supercooling of the condensate via the controlled refrigerant bleed from the concentration-stabilized evaporator is not shown. Neither has any detailed representation been given of the improved rectification unit.

The detailed process flowcharts with all improvement steps are shown in Figs. 8 and 9. The number of heat exchangers required for these should not be assessed too highly. It means no increase of the installed heat exchange area and should be optimized in terms of process engineering on the basis of fundamental operational checks.

Fig. 8 shows the process flowchart for the newly-designed domestic heat pump. In addition to the loss-free refrigerant concentration equipment, it contains a concentration-stabilized refrigerant evaporator which also operates without loss. The domestic heat pump also contains axial piston pressure exchangers for energy-saving solution circulation as well as an exhaust gas heat exchanger for improved fuel utilization.

Fig. 9 shows the process flowchart of an extension stage for large heat pumps possible today. It also contains all process engineering improvements, with the exception of the pressure exchanger. Beyond this, large, direct-fired heat pumps can be additionally equipped with air preheaters.

The control of the mass flows within the cycle will not be treated again at this point (see Section 3). However, the cooling water flow through the various heat exchangers of the installation is of interest to allow a maximum coolant heating to be attained. A particularly advantageous solution is where the cooling water flows through the condenser and the sidestream absorber in parallel and the combined refrigerant stream finally cools the mainstream absorber and the exhaust gas heat exchanger for the direct-fired boiler.

As the check of the thermal stresses in the individual heat exchangers recorded in Table 5.5 shows, the heating water temperatures can in this way rise continuously in the supply up to 82°C and in the reflux up to 52°C.

5. Efficiency of the improved heat pump process

The efficiency of the newly designed ammonia-water absorption heat pump proc-

ess may be demonstrated on the basis of a model example. The process and power data are combined for this purpose in Tables 5.1 to 5.5.

5.6 Checking the rectification unit

The operation of the newly designed rectification unit was checked by mass, concentration and energy balances. The process data of the model example showed that the rectification potential of the solution stream which was enriched more highly with refrigerant in the sidestream absorber is greater than required. This results in higher solution reflux concentrations at the outlets of the two upflow sections. It is then assured that the refrigerant vapor is in each case concentrated without external cooling. The excess potential is then available for a further dehydration of the refrigerant vapor.

6. The energy advantages of the process

The energy advantages of the new absorption heat pump process are apparent from a comparison with the standard process. This comparison is permissible only when both processes are based on the same operating conditions. The following common operating conditions have been defined with reference to the existing remote heating absorption heat pump installation of the Waiblingen municipal utilities (Tables 6.1 and 6.2)⁴. The annual utilization efficiency and the thermal values of the standard process are unattainable ideal values, because in practical operation unstable evaporator concentrations lead to unplanned heat pump cutouts and to an increased refrigerant circulation. In the heat pump installation at Waiblingen, these effects led to a 10% reduction of the heat generation by the heat pump. These relationships are directly transferable to the standard process and cause a reduction of its annual utilization efficiency from 1.3 to 1.24.

In reality, however, the heat pump installation at Waiblingen, which is designed in standard technology, attains an annual utilization efficiency of only 1.12. This is also due to the fact that its hot water supply temperature is limited to 65°C and its heat generation proportion drops to 67%.

These disadvantages are avoided in the improved heat pump process. It attains an annual utilization efficiency of 1.45 with continuously increasing hot water supply temperature up to 80°C under the operating conditions of the heat pump installation at Waiblingen.

Apart from the energy cost savings due to the significantly reduced fuel consumption and the almost negligible electricity consumption for solution transport, com-

plete automation of the installation also allows a staff reduction. A costwise evaluation of the savings thereby attained has no general validity due to individual energy tariffs, so that no evaluation of this sort will be carried out here.

7. Conclusion

The problem solutions worked out in this paper help to provide new technological applications for the absorption principle, especially for rational heating processes. But a thorough and well thought-out market preparation is required for their implementation. It must be borne in mind that the real market is in domestic pumps. If the VDEW questionnaires are correct, there is a market for 1 million heat pumps in the Federal Republic of Germany alone. The negative experience obtained in operating current installations would therefore lead to a new generation of heat pumps.

This insight should lead to the development of domestic heat pumps on the basis of the most modern series production methods, so that market-oriented installation prices can be ensured. In contrast, the smaller market for large-scale heat pumps may already be supplied with the new installation technology. In both cases, highest priority must be given to the development of the new types of transport equipment, for these are the key components of economical domestic heat pumps and are of interest not only for the new rectification unit but also for the entire field of material separation technology.

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2. Markfort, D., patent application P 27 38 138 dated 1 Mar 79 and P 28 49 076 dated 22 May 80, "Thermal mixture separation process for decomposition of a liquid mixture of several components of different volatilities: rectification unit".
3. Vinz, P., Operating analysis of the absorption cooling plant in the Olympic ice sport stadium at Garmisch/Partenkirchen, 1984.
4. Hatje, W., "Utilization of spent heat and biogas from sewage works processes" VDI Research Report No. 591, 1986.

⁴P. Vinz, *Angewandte Thermodynamik*, München, FRG

LITERATURE REVIEW: GROUND-COUPLED HEAT PUMPS

Braud, H.J.; Klimkowski, H.; Baker, F.E., "Earth-coupled heat pump and refrigeration design and applications", CONF-8606206, 1986. 15 p. Conference: Summer meeting of the Society for Engineering in Agriculture: meeting the engineering challenges of agriculture worldwide, San Luis Obispo, CA, USA, 29 Jun 1986 (in English). The paper gives a design procedure for ground loops with a combination of heat pump and refrigeration loads. Water-cooled condensers for refrigeration, freezers, ice machines and heat pumps for hot water can be coupled along with the heat pump for space heating/cooling.

Mei, V.C., "Heat pump ground coil analysis with thermal interference", CONF-861211--17; DE87002621SO 1986, 6 p. Conference: American Society of Mechanical Engineers winter meeting, Anaheim, CA, USA, 7 Dec 1986 (in English). When two horizontal ground coils are buried in the same trench, one on top of the other, the total energy exchange coils and ground increases drastically over that of a single coil for the same trench length. However, due to the thermal interference between the two coils, the overall performance of the the two-coil trench is very difficult to determine. Traditionally this type of problem is handled by the line source mirror image approach. But this approach, due to the requirement of estimating the strength of the source, will result in extremely conservative ground coil design. This paper provides a detailed mathematical model to describe the operation of the two-coil system which can calculate more realistically the performance of the ground coils, which in turn could make the ground coil heat pump system more economically competitive with other heating and cooling systems. The effect of thermal interference is clearly shown in the calculated soil temperature profile. The effect of fluid inlet positions, from upper or lower coils, is also studied. The comparison of measured and calculated daily energy absorbed from the ground indicated a maximum error of 27%, with the average error at less than 16%, with the calculated values on the conservative side. This model can better calculate the performance of two coils in the same trench.

Soederlund, M., "Energy wells recharged by solar heat. (Lulea, Sweden.)", CONF-860641--; DE87770021, 1986. pp. 186-191. Conference: North Sun '86 - international conference, Copenhagen, Denmark, 10 Jun 1986 (in English). If solar energy is to be able to compete with other energy sources in northern regions the collector system has to be inexpensive and simple. During the summer asphalt and roof surfaces are heated by the sun to relatively high temperatures. By spread-

ing and collecting water which flows in a thin layer on such a surface a low temperature solar collector is created. Heat extraction from ground systems means a cooling of the ground. By recharging the energy wells with solar heat during the summer the temperature drop is kept within chosen limits.

Johnson, W.S.; McGraw, B.A.; Conlin, F.; Wix, S.D.; Baugh, R.N., "Annual performance of a horizontal-coil ground-coupled heat pump", CONF-860106--SO ASHRAE Trans. (1986) v. 92(1A) p. 173-185 Conference: ASHRAE semi-annual meeting, San Francisco, CA, USA, 19 Jan 1986 (in English). Performance of the ground-coupled heat pump system located in TECH House I at the University of Tennessee has been measured for the 1983-84 cooling and heating seasons. Hourly measurements of heat flows, power consumptions, and various ground, space, and system temperatures have been recorded and analyzed to give overall seasonal performance. For the heating season, the performance factor was found to be 2.6, which is higher than a typical air-air system. However, performance factor for the cooling season was determined to be only 1.1, considerably lower than a typical air-air system. Reasons for this low performance in the cooling mode include poor backfilling procedure when the ground coil was buried, soil drying, and possibly too short a ground coil. Also, the heat pump system itself had a low coefficient of performance for cooling compared to other high-efficiency systems. Future work on this project will involve reinstallation of a longer ground coil with better backfill, the installation of a ground irrigation system, a more efficient heat pump, and eventually the optimization of the ground coil and heat pump system.

Sommer, S.; Loekke, H.; Helweg, A., "Laboratory studies of chemicals leaching from ground coils connected to heat pumps. (Environmental consequences.)" Undersoegelse af miljoemaessige konsekvenser ved udslip af frostsikringsmidler fra varmeslanger til jordvarme, 1986, 72 p. (in Danish). Heat pumps win widespread use for the heating of buildings in Denmark. However, ground coils filled with anti-freeze mixtures and placed at a depth of 0.6 to 1 m below the soil surface involve a risk of ground water contamination. This project was established to investigate the dispersion and degradation in soil of chemical compounds used as anti-freeze components. The experiments were carried out in the laboratory, but physical, chemical and biological conditions were similar to those under field conditions during major parts of the year, i.e. 6-10 degrees, anaerobic or anoxic soil air, water concentrations at or above field capacity and edaphic conditions as found

in actual soil samples taken at 1 m depth from three typical sites. The results show a high mobility and a slow degradation rate for each of the compounds ethylene glycol, ethanol, methanol and sodium benzoate in deeper soil layers. It appears from the results that a risk of ground water contamination exists in cases of leaching of antifreeze mixtures from the ground coils of heat pump plants. A special risk of contamination should be expected under circumstances where the retention time of the chemical substances is low in the unsaturated zone. This might be the case if: a) a high volume of anti-freeze mixture is released; b) the hydraulic conductivity is high; c) the distance to the water-table is low; d) a high volume of precipitation is infiltrating the ground immediately after leaching. 19 refs.

Soerensen, S.; Qvale, B., "Groundwater as heat source for heat pump plant", Grundvand som varmekilde for varmepumpeanlaeg, 1986. 85 p. Ser. Title: Energiministeriets Varmepumpeforskningsprogram. Rapport, 35. ISBN: 87-7511-622-7 (in Danish). Heat pumps with groundwater as heat source have now been in operation for a number of years in Denmark and abroad. The first Danish plant was put in operation in 1978 and in May 1983 approximately 250 plants were installed in Denmark - mainly in Jutland. Since the number of controlled measurements in Denmark is very limited it is difficult to make COP-comparisons between common types of Danish heat pump installations with natural heat sources (air, soil, groundwater). Present experiences show that a groundwater heat-pump installation will meet expectations to operation and economy if: 1. The preceding hydrogeological investigations are performed conscientiously. 2. The drilling of the wells are carried out with special attention to the special demands caused by the reinjection of the water from the groundwater heat pump. 3. Installations are airtight, thus preventing contact between atmospheric oxygen and circulating groundwater, and preventing leakage of oil and refrigerant into the ground. 4. The installations in the house are dimensioned to permit the heat pump to operate at its optimal temperature conditions.

Drake, M.S., "Off peak geothermal heat pump storage system", CONF-860303--SO Energy Technol. (Wash., D.C.) (Mar 1986) v. 13 p. 1327-1330 Conference: 13. annual energy technology conference and exposition (ET '86), Washington, DC, USA, 17 Mar 1986 (in English). The basic design of a system installed in the Columbus, Ohio Zoo is described. The distribution system is described in detail. The control system is microprocessor controlled, with about 50 monitoring points. Preliminary evaluations have indicated that the system can operate with an overall coefficient of performance in excess of 3.5. 1 figure.

Bingham, G.A., "Earth-type heat exchanger for heat pump systems", US patent 4570452, 18 Feb 1986 (in English). This patent describes a heat pump heating and/or cooling system containing chlorofluorocarbon refrigerants. The system includes above-ground components and an earth-type heat exchanger unit dis-

posed during use of the system entirely below ground level. The improvement described includes a cylindrical coil assembly having cylindrical fluid-conducting coils extending in substantially concentric relationship to each other about the central axis of the assembly. Each of the coils is formed of heat conductive and

corrosion resistant tubing. The tubing of each coil is of different diameter than the tubing of each other coil; a conduit interconnects the coil assembly and the above-ground system components, during operation of the system, conducting the fluid to and from fluid-conducting coils of the assembly.



Selected Book and Report Reviews

Reviews of the most recent publications on heat pumps

Heat from the Sun and Ground (Wärme aus Sonne & Erde); Schulz, H.; Oekobuch Verlag & Versand GmbH, Freiburg, ISBN 3-922964-18-4, January 1987 (in German)

The author has developed an energy saving heating system with solar absorber, ground heat storage system and diesel engine heat pump and tested it for several years in his own home. Summer sun heat is stored in the ground via heat exchange probes that are drilled vertically into the soil. During the heating period, heat is extracted from the 1000 m² ground storage system by means of a diesel engine heat pump. With this inexpensive self-made plant, the author succeeded in reducing his fuel oil consumption from 4000-5000 liters per year to 1300-1500 liters per year without sacrificing heating and comfort. In this book are also detailed instructions on how to install single elements (ground storage system, solar absorber and diesel engine heat pump) with customary units and easy handicraft methods without assistance. Additionally, previous operating experiences are imparted and instructions are given for dimensioning similar plants that are not only useful for dwellings, but also in agriculture and industry.

Impact of Heat Pump Technologies on Ground Water and Soils; Balke, K. D. (Tübingen Univ., F.R. Germany), pp. 153-164 of Technological Evaluation of the Heat Pumps, Papers and Discussions. Marhenkel, H. and Koenig, W. (eds.). Düsseldorf, F.R. Germany, VDI-Verl. CONF-8403236 (1985) (in German)

From seminar on technological evaluation of the heat pump; Düsseldorf, F.R. Germany (13 Mar 1984). The physical, chemical, and biological effects of ground refrigeration caused by the operation of ground water heat pumps are quasi-negligible in their totality. According to tests, the negative thermal anomaly building up around the absorption well is not stationary but pulsates in conformity with the seasonable cold water seepage and the seasonal temperature of the near-surface

ground. However, as well construction and long-term operation may cause groundwater contamination, groundwater heat pump systems are not by nature ecologically beneficial. Sufficient pollution control and appropriate dislocation and construction work require the planning and site supervision of the system as a whole (i.e., the block of heating system/heat pump/well) to be engineered by only one company.

Book series "Heat Pumps", editor H. Kirn, publisher C.F. Müller, Karlsruhe

C.F. Mueller Publishers have completed the eight-volume book series "Heat Pumps" through the publication of Volume 6: "Absorption Heat Pumps" by Loewer, as well a totally revised 5th edition of Volume 2: "Application of the Electric Heat Pump" by Kirn/Hadenfeldt.

In 1976, C.F. Mueller Publishers put the book "Heat Pumps" by Kirn/Hadenfeldt on the book market. From the beginning the demand was keen, so that new editions were published within short intervals. After the increasing interest in heat pump technology, various problems that occurred with electric, diesel and gas heat pumps as well as with compression and absorption heat that arose in the 1970's, it seemed to be suitable to change the book into a series on heat pumps. With this, the publishers and authors could better correspond to the requirements of theoretically interested persons, for example students, on the one hand, and experts involved in planning and construction of plants on the other hand. The applications of heat pump technology and suitable technical solutions were so different that an entire presentation corresponded to the requirements of practice.

For this reason, the following classification was developed from the requirements of the book market:

Volume 1: "Fundamentals of heat pump technology", author: H. Kirn, edition 1983, 6th revised edition, 259 p., 202 figs., DM

62.-, ISBN 3-7880-7209-1

This volume imparts theoretical fundamental knowledge of heat pump technology for theoretically interested persons as well as planners: Thermodynamic fundamentals/heat pump process/compression heat pump/compressor types/drive engines/sorption heat pump (absorption,- adsorption,- and resorption heat pumps)/heat transformer/radiation pump/Peltier heat pump/heat exchanger/fittings/working fluids/heat transfer agents/sound control.

Volume 2: "Application of the electric heat pump", authors: Kirn/Hadenfeldt, 5th completely revised edition 1987, 320 p., 200 figs., DM 58.-, ISBN 3-7880-7209-1

Volume 2 is devoted to installers of electric heat pump plants, interested architects and experts.

Contents:

Fundamentals of heat engineering and electrical engineering/electric engine/service suitability/testing of electric heat pumps/heat sources (for ex. cellar rooms, outdoor air, ground water, ground source)/buffer storage/boiler as additional heat producer/circulating pump and network of tubes/low-temperature hot water heating system/hot water supply/regulation and control/annual energy and annual total costs of a plant/planning examples.

Volume 3: "Application of a gas- and diesel heat pump in domestic engineering", author: H. Juettemann, edition 1981, 216 p., 161 figs., DM 48.-, ISBN 3-7880-7121-4

Volume 3 is not only thought to be an instructional manual, but it will give planners and installers the necessary instructions for planning and installation of a heating system with gas or diesel heat pumps as heat producers.

Contents:

Terms and fundamental quantities for combustion engines/types of combustion engines/waste gas turbo-charged/fuel types/service life/sound emission/pollutant emission/heat output of the engine/joint utilization of spare generating sets/maintenance/profitability/absorption heat pumps/examples/realized objects.

Volume 4: "Installation, operation and maintenance of the electric heat pump", author: H. Eickenhorst, edition 1982, 131 p., 81 figs., DM 36.-, ISBN 3-7880-7151-6

This book is intended for the expert. Its content is based on the author's many years of experience with installation and operation of heat pump plants.

Contents:

System construction of heat pumps/operation modes of heat pumps/delivery of the heat pump/connections of heating systems/operating failures and their removal/electric installation/product information/standards and rules/tariffs/public funds.

Volume 5: "Heat sources and heat storage systems", author: H. Kirn, edition 1983, 196 p., 125 figs., DM 58.-, ISBN-3-7880-7194-X

Volume 5 provides information about natural heat sources that can be used by the heat pump and about the technology of their utilization.

Contents:

Terms/surface water/shore filtrate/waste water/ground source/earth probes/outdoor air/solar radiation/open air space heat source systems (absorber surfaces)/heat storage systems.

Volume 6: "Absorption heat pumps", author: H. Loewer (editor), edition 1987, 240 p., many figures and tables, ISBN 3-7880-7234-2

The book is intended for theoretically interested persons and persons who are involved in planning of heat pump plants with thermal drives. The utilization of primary energy and low-value energy mediums for driving the heat pump (waste heat) offer new possibilities of saving energy.

Contents:

Thermodynamic fundamentals/absorption heat pumps of average and high capacity/absorption heat pumps of low capacity (residential heating system)/periodically working absorption heat pumps/the heat transformer/gas absorption heat pumps/working fluids/control/standards and rules.

Volume 7: "Application of the heat pump in the industrial, commercial and agricultural sector", authors: A. Frank, H. Hohen-dorf, H. Moderegger, G. Trenkowitz, edition 1984, 214 p., 110 figs., DM 59.-, ISBN 3-7880-7200-8

The heat pump in the industrial, commercial and agricultural sector mostly serves to recover waste heat from heating or cooling processes and to supply low-temperature process heat. The book is intended for all persons who want to plan industrial, commercial and agricultural heating processes more effectively. It tries to give some advice for the planning engineer and installer of corresponding systems.

Contents:

Fundamentals/drying methods/evaporation and distillation processes/meat processing/beverage industry/agriculture/food sales/chemistry/sports arenas/large buildings/special application fields.

Volume 8: "Hot water supply with electric heat pumps", author: A. Hadenfeldt, edition 1983, 166 p., 83 figs., 28 tables, DM 63.-, ISBN 3-7880-7189-3

The book especially appeals to experts as an instruction guide and assistance for planning and installing hot water supply systems.

Contents:

Fundamentals of hot water supply/individual supply/simultaneous supply/residential supply/central supply/flow water heater/storage water heater/hot water and efficient energy demand in households, agriculture, public buildings, trade, commerce and industry/hot water supply systems for heat pumps with heating-only heat pumps, special hot water heat pumps and waste heat from refrigeration plants/precalculations of the annual costs/standards and rules.

With the technical book series "Electric Heat Pumps" a first attempt was made to offer theoretically interested persons and experts a work that gives a broad survey of heat pump technology and that corresponds to the special requirements.

The publishers, the editor and authors hope to have contributed to a rational energy application with their efforts. They are confident that there will be a keen demand for this book series in future.

Volumes 1, 2 and 3 were published in 1981, 1982 and 1983 in Italy by the publishers Tecnice Nuove Milano, Volume 1 was put on the book market in 1983 by the publishers El Instalador Madrid. In both countries the volumes were published in the national language.

As a supplement to the eight-volume series, the publishers present a work on guiding rules "Heat Pump Rules", by N. Krug/U. Pfeifferberger/T. Rinck, which has been elaborated in cooperation with "Deutscher Kälte- und Klimatechnischer Verein (DKV).

The complete series plus the work "Heat Pump Rules" are offered at a special price of DM 380.

Schedule of Conferences and Trade Fairs

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New York, New York (USA); **1987 ASHRAE Winter Meeting.** Sponsored by the American Society for Heating, Refrigerating, and Air-Conditioning Engineers. Contact: Judith Breese, ASHRAE, 1791 Tullie Circle NE, Atlanta, GA 30329 (USA), Telephone 404/636-8400.

January 19-22, 1987

New York, New York (USA); **1987 International Air-Conditioning, Heating, Refrigerating Exhibition.** Sponsored by ARI

and ASHRAE. Contact: Hank Stevens, International Exposition Company, 200 Park Avenue, Suite 1204, New York, NY 10166 (USA), Telephone 212/986-4232.

March 25-27, 1987

Oxford (United Kingdom); **3rd International Symposium on the Large Scale Applications of Heat Pumps and Exhibition.** Sponsored by the British Hydromechanics Research Association (Cranfield). Contact: Ms. J. Stanbury, British Hydromechanics Research Association (BHRA), Fluid Engineering Center, Cran-

field, Bedford MK43 OAJ (United Kingdom).

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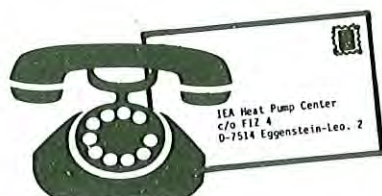
Budapest (Hungary); **Symposium on the Long-Term Impact of Energy Efficiency Improvements.** Sponsored by the United Nations (Geneva), the Economic Commission for Europe. Contact: United Nations, Economic Commission for Europe, Energy Division, Palais des Nations, CH-1211 Geneva 10 (Switzerland).



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● **Nonazeotropic Refrigerant Mixtures as Working Fluids in Compression Heat Pumps, A Bibliography**, Report No. HPC-B1, second edition, December 1985, DM 220. This report is a comprehensive bibliography of over 200 reports with complete citations and abstracts.

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- Descriptions of each session
- Biographies of speakers and chairmen
- Details of planned preconference site visits and postconference field trips

In addition, ongoing IEA activities, particularly those emphasizing heat pumps, will be discussed. Regular features (book reviews, conferences and trade fairs, news updates) will also be included. If you have further contributions relevant to ground-coupled heat pumps, please send them to the Heat Pump Center.

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