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IEA Heat Pump Centre

# NEWSLETTER


IEA heat pump centre

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***Front Cover***
*The World Trade Center Building in Amsterdam.*

***International Energy Agency***
*The International Energy Agency (IEA) was established in 1974 within the framework of the Organisation for Economic Cooperation and Development (OECD) to implement an International Energy Programme.*

*A basic aim of the IEA is to foster cooperation among the 21 IEA participating countries to increase energy security through energy conservation, development of alternative energy sources and energy research, development, and demonstration (RD&D). This is achieved in part through a programme of collaborative RD&D consisting of 42 Implementing Agreements, containing a total of over 80 separate energy RD&D projects. This publication forms one element of this programme.*



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# Editorial

From an energy efficiency point of view, engine-driven heat pumps clearly have potentially good prospects. These systems have obtained a market share in countries with low natural gas-to-electricity price ratios like Germany and The Netherlands. Gas engine-driven heat pumps are also used in France, England, Austria and Japan, and a growing interest is visible in the USA.

The dominating markets for engine-driven heat pumps were commercial and apartment buildings and glasshouses. Until now, these heat pumps were rarely installed in single-family homes, with the exception of Jin apan, where small gas engine air-conditioner heat pumps began selling remarkably well in 1986.

Evaluation of existing installations shows that engine-driven heat pumps are successful in many cases. One such case is described in the article on the application of a heat pump in the Amsterdam World Trade Center. However, experience also shows that some of the so-called first 'generation installations' suffer from disappointing performance and operating costs. These problems often originate from poor design (capacity, control, integration into heating systems etc.) and from an over-estimation of their technological knowledge. In many cases oversizing is one of the causes of the problems incurred.

Performance of engine-driven heat pumps have recently been analysed in Germany and The Netherlands. It is very important that designers and engineers make use of this information in order to learn from experience gained (Vol. 8, No. 1, Bibliography and HPC Report HPC-R-6).

New systems are currently being developed and commercialised. These systems are better tuned to the requirements which evolve from the combination of heat pump and gas engine. System integration and automation aspects also receive the required attention. This is exemplified by two Japanese articles, one dealing with small heat pump/air-conditioners, and the other one dealing with a medium-size gas engine-driven heat pump used for heating/cooling and hot water supply. The growing interest in the USA is illustrated by an article by the Gas Research Institute (GRI) on field tests of heat pumps in the 10 to 18 kW range in 19 locations, chosen to cover a wide spectrum of climatic conditions.

The Japanese and American articles show that engine-driven heat pumps are making progress, and that under favourable conditions and if well designed, can be a justified solution in many cases, especially for the environment.

The non-topical article in this Newsletter deals with a preliminary screening of 26 industrial processes for heat pump applications. The article clearly demonstrates the use of pinch analyses, a concept discussed in earlier Newsletters.

**Jos W.J. Bouma**
**General Manager HPC**

# Analysis of Internal Combustion Engine-Driven Heat Pumps

* G. Wolkerstorfer and P. De Jaegher

**SUMMARY**

A critical factor in evaluating internal combustion engine driven heat pumps (ICE-HP) is the amount of primary energy saved and the ratio of usable heat output to primary energy input (seasonal performance factor). ICE-HPs from different fields of application were thus analysed and evaluated on the basis of energy utilisation.

When evaluating the efficiency of heat pumps, financial savings are of particular interest. Various users were surveyed to find out what their investment costs had been and what their current annual service and maintenance costs were. The economic feasibility of ICE-HPs is described on the basis of the heat generating costs.

## Introduction

Compared with conventional heating systems, ICE-HPs make optimal use of primary energy. The question is, however, to what extent is the limited amount of available primary energy actually exploited by ICE-HPs; in addition, what is the seasonal performance factor for a heating period given non-steady state operating conditions.

In order to make a comparison with other heat generating systems the costs and heat output must be determined.

For the purposes of this investigation, ICE-HP users were interviewed and asked about their experiences with this technology. In addition, a compilation and description of actual heat pump units was made in order to have an overview. Relevant data of over 600 gas and diesel engine-driven heat pumps has been compiled and stored in a data bank. The ICE-HPs range from small heat pumps (heating capacity from approx. 20 kW) to large heat pump units of several MW.

## SPFs for ICE-HPs

The seasonal performance factor (SPF) is the ratio of usable heat output to primary energy input required in one heating period. SPFs dependence upon annual operating hours for various ICE-HPs are given in Figure 1. It is interesting to note that the SPFs rise with the operating hours. The heat pumps in the marked range are used for space heating. Some are also used for heating swimming pools. Outside of this range in Position 1a and 1b (different operating years), there is a heat pump used for local heat supply. Industrial waste heat of approx. 35°C to 40°C is used as the heat source for this heat pump. For this reason, very high SPFs are attained. Another positive contributing factor is the high number of annual operating hours.

The heat pump in Position 2 is used to heat an outdoor swimming pool. Water from the pool flows directly through the condenser. In this way, low condensation temperatures and high SPFs are attained. The ICE-HP in Position 3a is used in the industrial sector and attained over 7,000 hours of operation a year. This heat pump is used for both heating and cooling. Looking at only the heating side of the unit, the lower SPF (Position 3b) results. If the cold water produced is seen as an additional benefit, an SPF of over 2 is obtained.

It should be mentioned, that ICE-HPs with a high degree of utilisation in industry also attain good SPFs despite a high temperature level. In addition, efficient use of the heat pump results in high energy savings.

The investigations show that ICE-HPs used for space heating have SPFs from 1.2 to 1.4. SPFs are in the upper range for water/water heat pumps and in the middle and

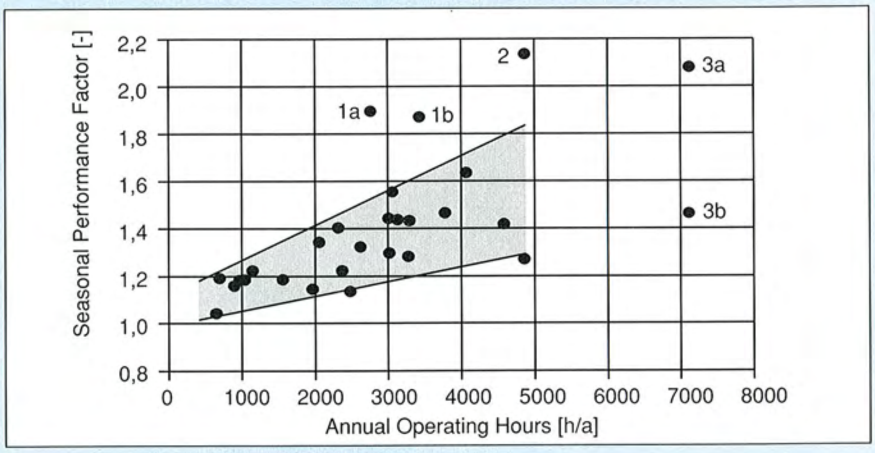



*Figure 1: Seasonal Performance Factor for ICE-HPs.*

lower range for air/water heat pumps. Heat pumps used in this sector are in operation for approx. 2,000 to 2,500 hours a year (max. 3,000 hours); the bottom limit is approx. 1,000 to 1,500 hours per year. Heat pumps which have to supply a large amount of heat during the year (outdoor and indoor swimming pools), or which have to meet a base load, naturally have a higher number of annual operating hours.

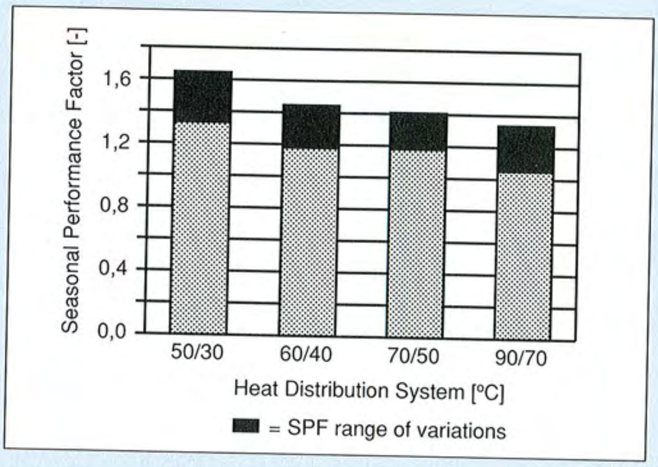



*Figure 2: SPFs Dependence upon Heat Distribution System.*

Another factor which strongly influences the SPF is the temperature level of the heat distribution systems used as shown in Figure 2. The SPF rises as the temperature drops. Distribution systems with low and middle supply/return temperatures are the most advantageous. Yet considerable energy savings are also possible with high temperature distribution systems (90/70°C) given that other conditions are favourable.

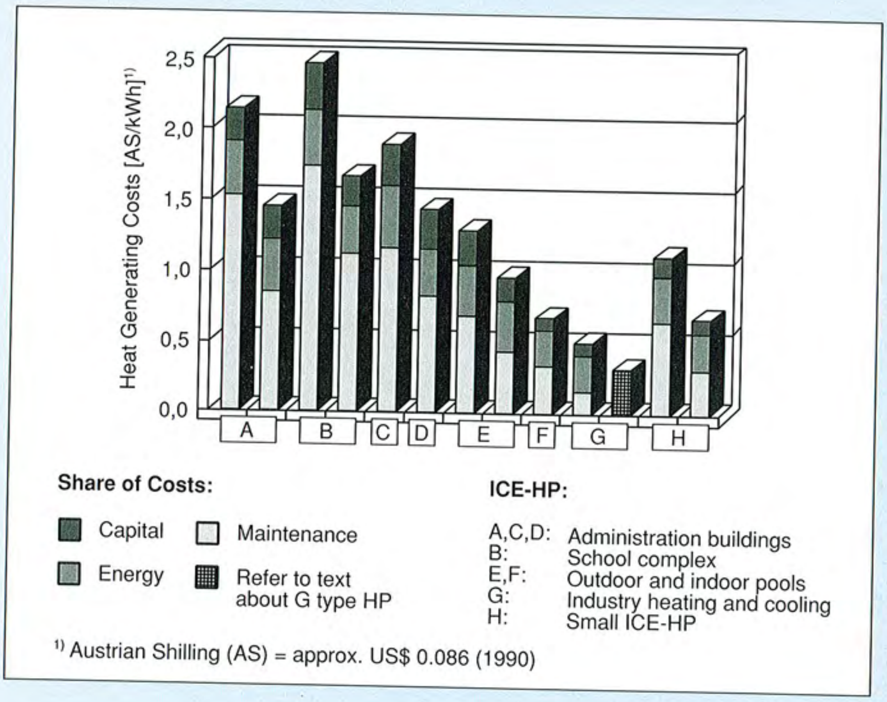
## Heat Generating Costs

Figure 3 shows heat generating costs for ICE-HPs installed in various sectors of application. Costs for energy, maintenance and, above all, capital costs have varying effects on the heat generating costs. Capital costs are obtained from investment costs and an annuity of 10.3% (general service lifespan of 15 years, 6% interest).

Maintenance costs are based on figures given by individual users for maintenance, repair and service costs necessary to keep the system running properly. Energy costs are based on the energy consumption (gas, oil, electricity) and the price.

The individual ICE-HPs, with the exception of the small heat pump, were listed according to their number of operating hours per year. The small ICE-HP (H) supplies one and two family homes with heat. The heating capacity of the large units (A - G) is from approx. 300 kW up to the MW range.

*Figure 3: Heat Generating Costs with ICE-HPs.*




Air is the heat source for ICE-HPs (A, E, and H). Ground water is used for units (D and F). Surface water is utilised for heat pump (B) and ICE-HP (C) uses waste heat. ICE-HP (G) produces both heat and cold water.

ICE-HP (A) received a government subsidy. The bar on the left indicates what heat generating costs would be without a government grant while the bar on the right depicts costs with the subsidy included. The high costs are due primarily to the fact that the unit is over-dimensioned (actual heating demand is only half of what was calculated). Thus the unit does not operate at the optimal degree of utilisation; furthermore, operating hours are low and investment costs are considerably higher than needed.

In the case of the school complex heat pump (B), the bar on the right shows heat generating costs in a favourable year. The average over three years of operation is shown by the bar on the left. The high costs (share of capital is over 1.50 AS/kWh) resulted from various breakdowns (mainly peripheral) and thus less heat supply.

Waste air (approx. 20°C to 25°C) is used as the heat source for unit C, supplied to the heat pump via a brine cycle. Due to the high supply temperatures of the distribution system, it is only possible to use the waste heat in inter-seasonal periods and not during the entire heating period. Waste heat during the cold season is emitted into the air via the brine cycle mentioned above and an electrically-driven motor chiller.

Units (A, B and C) are all in operation for relatively few hours per year (approx. 900 to 1,050). ICE-HP (D), by contrast, operates for more than twice the above number of hours. As the unit is generating more heat, heat generating costs are lower although initial investment costs, when compared to the above units, were considerable higher (approx. 80 to 100 % higher).

Units E and F, used for indoor and outdoor swimming pools, show better results but also operate under more favourable conditions. Firstly, the extent of the heat supply is higher (annual operating hours are three to four times higher) than for units (A, B and C). Secondly, in addition to heating the pools, buildings are heated as well, depending upon how much heat is left over for use. On the one hand this makes better use of the unit in inter-seasonal periods, while, on the other hand this necessitates higher temperature levels of the heat pump than would be necessary if only the indoor pool were being heated.

The two bars for unit E show the range of heat generating costs for two very different years. The bar on the left is for a year during which there was a longer non-operational period (breakdown of the unit); the bar on the right is for a year during which there were no significant operational disturbances. The other ICE-HP used in this same field of application (F) supplies a large amount of heat (approx. 4,000 hours per year). The large amount of heat generated leads to a low share of capital cost in the heating price.

The use of ICE-HPs in factories is particularly advantageous because of the high degree of utilisation and the favourable heat demand profile (base load) as shown in the case of heat pump (G). Here, the heat pump is being exploited to particular economic advantage, being used not only for producing heat, but also for cold water. As can be seen in the left bar for heat pump (G), capital costs are only a very minor factor in the heating price. Maintenance costs are also low; this is due not only to the fact that the heat pump has a high degree of utilisation, but also to the fact that it is maintained efficiently and conscientiously. The above evaluation concerns the heating aspect only, thus assuming that the cold water is produced at no cost. With a conventional system, the cold water would have to be produced in a separate system.This would involve additional energy, investment and service costs. The bar on the right reflects only the energy savings for the heating/ cooling system; if the investment and maintenance costs saved by not having to install a separate cooling system were also taken into account, an even better cost profile would emerge.

The heat pump units described above are large scale systems with heating capacities of over 300 kW. In contrast, the small heat pump (ICE-HP H) is meant for heating one and two family homes. This heat pump was tested by the manufacturer in several field studies and the results were analysed. Both cost relationships are based on the data given by the manufacturer which in turn were made on the basis of his field studies.

The bar on the left is for a heat pump in a one family home, whereas the one on the right is for a heat pump which also supplies heat for an additional consumer (e.g. a swimming pool). Mass production of such units leads to a lower capital cost share when compared to the larger units (A to D).

Specific capital costs depend primarily on operating hours and on the heat delivered. Low operating hours and low heat delivery lead to the specified shares of the total heat generating costs. It is interesting to note that the small heat pump (H)' lies in the middle range in terms of capital costs. In addition, maintenance costs are lower for units which have a high degree of utilisation. This is not the direct result of higher heat delivery but rather of proper maintenance.

## Conclusion

Investigations have shown that expected SPFs rise with the number of annual operating hours. This is due to the longer running periods per start and thus the higher share of steady-state operating phases. The reasons for shorter running periods and lower SPFs can often be found in the oversizing and design of the system.

There was quite a disparity in heat generating costs among the various heat pumps examined. A closer analysis showed that the heat generating costs were acceptable in only a few cases. In fact, using a conventional boiler for heating in the case of A would result in a heating price of approx. 0.62 to 0.65 AS/kWh including investment, maintenance and energy costs.

It is, no doubt, true that in the present energy price situation an energy-saving heat pump is economically less feasible due to its higher purchase price. If the heat pump is dimensioned, planned, constructed and maintained properly, however, it can be equal to conventional systems despite the present prices of primary energy. In special cases, ICE-HPs can even be economically more advantageous today than other systems.

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# Compression Heat Pumps in the Dutch Glasshouse Sector

* V.P. Fonville, M.G. Telle and N.J.A. van der Velden

**SUMMARY**

The application of compression heat pumps in the Dutch glasshouse sector as a primary heat source has been evaluated. A heat pump could save a lot of energy on a glasshouse holding because of its high efficiency. However, for economic reasons the application of gas engine-driven heat pumps in glasshouses is not currently interesting.

## Introduction

The size of the Dutch glasshouse sector is approximately 9,500 hectares, where mainly natural gas boilers are used for heating. The annual fuel consumption is about 3.5 billion $m^3$ natural gas. There is a large variety in holding circumstances, particularly regarding size and fuel consumption ($m^3$ gas per $m^2$ glasshouse).

During the early 1980's growers were interested in energy saving because of the high energy prices. A gas-fired boiler can, when a double gas condenser is used, reach an efficiency of over 100% (based upon the lower heating value of 31.65 MJ per $m^3$ natural gas). Application of a heat pump with high efficiency could lead to energy saving in comparison with a conventional gas boiler. About 40 heat pumps were installed in Dutch glasshouses, most of which were gas engine driven. To learn more about heat pump performances, 8 compression heat pumps were monitored for at least one year. This concerned 4 water-to-water heat pumps, 3 air-to-water heat pumps with "silent evaporator" i.e. without cooling fans, and 1 with fan-operated evaporators. The Agricultural Economics Research Institute (LEI) and the Institute of Agricultural Engineering (IMAG) in the Netherlands have gathered and analysed the available performance data and experience gained in order to come to an overall conclusion of heat pumps in the glasshouse sector. This information was used to make an economic evaluation.

## Glasshouse Heating System

The heating system in a Dutch glasshouse is usually a natural gas-fired boiler with a (double or single) exhaust gas condenser. The high grade heat (maximum water temperature 90°C), is brought into the glasshouse by steel pipes. This is the primary heating system. The gas condenser produces a small quantity of low grade heat (temperature approximately 40°C), and this heat is brought into the glasshouse by a secondary heating system with a separate pipe network.

A heat pump differs from a gas-fired boiler in that a heat pump produces low grade heat, which requires large water flows and reduces the capacity of the heating system. As a result, the secondary heating system must be extended and a new strategy for heating has to be developed.

For high efficiency of the heat pump, the return water temperature from the glasshouse should be low. Therefore, the capacity of the secondary heating system must correspond with the heating capacity of the heat pump. However, a large secondary heating system in a glasshouse is problematic for the plants and the grower. Therefore, the maximum heating capacity of a heat pump is limited by the extension of the secondary heating system which is still feasible. For example, in the case of 8 poly-ethylene tubes per 3.2 m width of the glasshouse for low grade heat from the condenser, the maximum heating capacity for the heat pump is 40 W per $m^2$. This is 20% of the total heating capacity of a glasshouse.

## Heat Pump Performance

The monitoring results of the 8 heat pumps showed that there are large differences in performance of heat pumps in glasshouse holdings. Heat pump performance in the glasshouse sector is hardly comparable with other heat pump applications for a number of reasons: firstly, glasshouses are poorly insulated and have no heat storage capacity; secondly, the influence of direct solar radiation on the heat demand is largely due to it being subject to wide fluctuations during the day (e.g. cloudy weather).
Heat pump performance depends, among other things, on the temperature of the heat source. The temperature of ground water in the Netherlands is 10-12°C throughout the year. Therefore, a water-to-water heat pump performs better than an air-to-water heat pump. In many cases the heat pump is driven by a gas motor and extended with a generator to produce electricity. Extension with a generator leads to a decrease in efficiency of the total installation (heat pump and generator) and the investment of capital will be larger because of the costs of the generator and the larger capacity of the engine.

A major problem for heat pumps in the glasshouse sector is $CO_2$ enrichment. $CO_2$ has a positive effect on the growth of plants. Dutch growers use the boiler exhaust gases in their glasshouses to raise the $CO_2$ concentration of the ambient air. The exhaust gases of the engine are not pure enough to be used for $CO_2$ enrichment. If $CO_2$ demand during the day exists, the $CO_2$ needs to be produced by the gas boiler. The boiler heat produced as a by-product does not have to be produced by the heat pump. This means that the heat pump will remain unused during a great part of the day.

With the monitored performance data it was possible to estimate expected heat pump performances. Table 1 presents the performance figures for a compression heat pump driven by a gas engine (in the case of 40 W heating capacity per m² with $CO_2$ enrichment by a separate boiler and no generator) which assess the seasonal primary energy ratio (PER), the coverage of the annual heat demand and the additional electricity consumption (the latter of which is not included in the PER).

*Table 1: Expected Performance of Gas Engine Heat Pump*

| Type of Gas Engine Heat Pump | PER (%) | Coverage (%) | Electricity (kWh/GJ) |
|---|---|---|---|
| Water-to-water | 210 | 38 | 2 |
| Air-to-water, silent evap. | 180 | 35 | - |
| Air-to-water | 175 | 30 | 5 |

## Economic Evaluation

In the economic evaluation the price of natural gas has been calculated so that no difference in costs occurs between heating only with the gas boiler, and using a heat pump in base load (40 W heating capacity per m²), assisted by a gas boiler for peak load. This price is called the breakeven point. The reference situation has a heat demand of 50 m³ natural gas equivalent per m² of glass per year, a gas boiler with double gas condenser and no electricity generator. The breakeven point for a water-to-water heat pump in a holding of 12,500 m² (capacity of the heat pump 500 kW) is Dfl. 0.67 per m³ natural gas (1 Dfl. is approx. 0.55 US$). This means that, in this situation, a heat pump is more profitable when the gas price is higher than Dfl. 0.67. For an air-to-water heat pump with a silent evaporator the breakeven point is Dfl. 0.84 per m³ gas and for an air-to-water heat pump with fan-cooled evaporator Dfl. 1.25 per m³. The water-to-water heat pump produces the best results because of the higher PER and heat coverage. A larger holding and a higher heat demand per m² will result in a lower breakeven point, e.g. a holding size of 25,000 m² reduces the above breakeven prices by approximately 10%, and an increase of the heat demand to 70 m³ natural gas equivalent per m² glass brings the breakeven price for a water-to-water heat pump in the large holding (25,000 $m_2$) down to Dfl. 0.42. The use of an electric generator has no economic advantages. The present (subsidised) gas price for Dutch growers is Dfl. 0.23 per m³ natural gas. This means that gas engine-driven heat pumps are currently not of interest to the glasshouse sector.

Sensitivity analyses show that for the larger holding size a decrease of 25% in investment costs reduces the breakeven price by approximately Dfl. 0.09, a 25% higher heat demand coverage by approximately Dfl. 0.13 and a 10 percentage point increase in PER by about Dfl. 0.03. Therefore, future developments, which lead to a lower investment of capital and maintenance costs, improvement of the performance and a higher gas price, should improve the economic perspectives.

## Conclusion

Heat pump performance in the glasshouse sector depends on the technical design of the heat pump and its integration in the heating system, and on the specific conditions of the holding. The water-to-water heat pump has the best future. Using a heat pump as a heat source in a glasshouse implies a drastic change in the heating system and heating strategy. Among other things, the heating capacity is limited by the feasible extension of the secondary heating system. Currently, it is more profitable for a grower to use the conventional gas boiler instead of an engine-driven heat pump.

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# Small Gas Engines for Gas Engine Heat Pumps

* H. Kazuta


### SUMMARY

Since placing a 1.3 RT (4.6kW) model small gas engine heat pump (GEHP) on the market in 1987, Yamaha Motor Co., Ltd. has developed and marketed 2.4 RT (8.4 kW), 4.0 RT (14.1 kW) and 5.3 RT (18.7 kW) models. While the engines used for the former two models were new designs for GEHPs, those for the latter two were made available by improving small automobile engines. The number of these engines produced and installed in GEHPs so far has exceeded 10,000.


## Engine Profiles

The 1.3 RT engine is of the 4-stroke, horizontal, single-cylinder ø 64 x 76 mm type, while the 2.4 RT engine is of the ø 70 x 76 mm type, i.e. with a larger cylinder bore than that of the 1.3 RT engine. The 4.0 RT and 5.3 RT engines are of the 4-stroke, vertical, three-cylinder ø 62 x 60.5 mm type. Various technological features of these GEHPs are described below.

### • Low Vibration

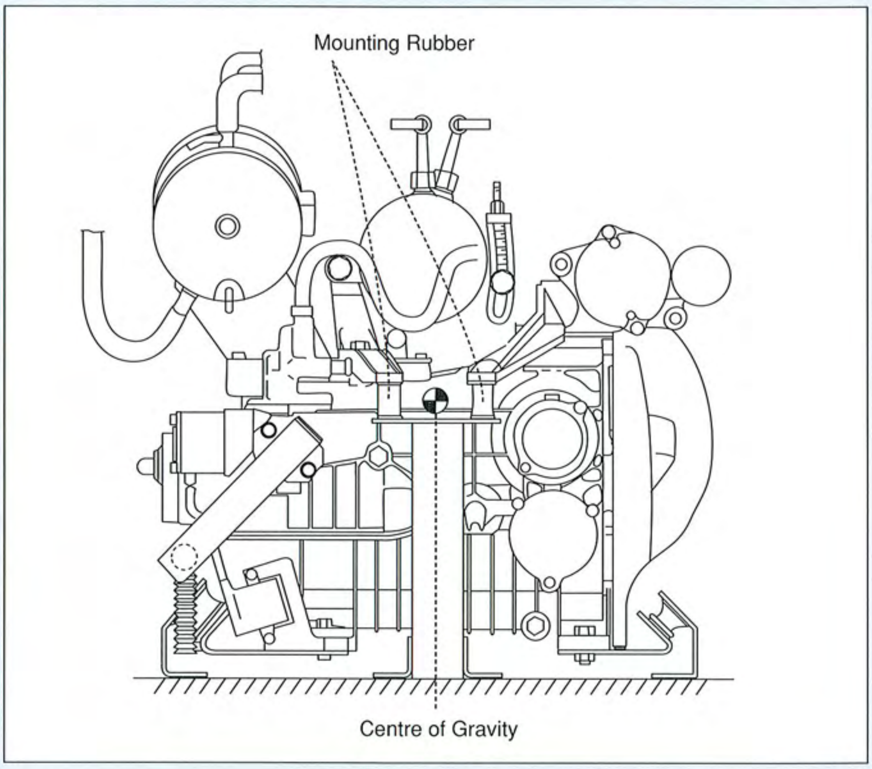
In order to improve heat pump operation efficiency, the electric heat pump employs an inverter to control the quantity of circulating refrigerant. GEHPs achieve this by varying the engine speed, but the upper limit of the speed is restricted because of the noise and the durability, while the lower limit is restricted mainly because of engine vibrations. The main cause of engine vibrations of GEHPs lies in torque fluctuations. In the case of single-cylinder engines which cause large torque fluctuations, the frequency of engine vibrations is close to the characteristic frequency of the engine mount system. This is a major problem in low-speed high load operation. In order to solve the problem, a centre of gravity mount system (Figure 1) was employed for the single-cylinder engines to achieve an engine mount system with a lowered characteristic frequency. This resulted in reduced vibrations. As for the compressor driving system, the compressor is rigidly fitted to the engine so that the engine mount system is not adversely affected by the reaction force via the compressor driving belt. Since three-cylinder engines incur less torque fluctuations, an ordinary elastic mount system is employed.

The speed range of the engines is as follows: 1100 rpm to 2000 rpm for the 1.3 RT, 1200 rpm to 2500 rpm for the 2.4 RT, 900 rpm to 2500 rpm for 4.0 RT, and 900 rpm to 3000 rpm for the 5.3 RT heat pumps.

### • Low Noise Levels

In order to reduce noise levels of the engine system, counter-measures have been taken against the noise sources, being the air intake noise, the exhaust noise and the noise emitted from the engine body. The air intake system uses a large capacity intake silencer, while the exhaust system uses an interference muffler which also serves as an exhaust gas heat exchanger. In the case of single-cylinder engines, a plastic exhaust silencer is added to reduce the intermittent exhaust noise. The noise emitted from the engine body is reduced by the enclosure. The air is supplied and exhausted through the silencer duct to ventilate the enclosure. The noise level of the outdoor unit is as follows:
- 50 db(A)/1 m for the 1.3 RT,
- 56 db(A)/1 m for the 2.4 RT,

*Figure 1: Centre of Gravity Suspension System.*



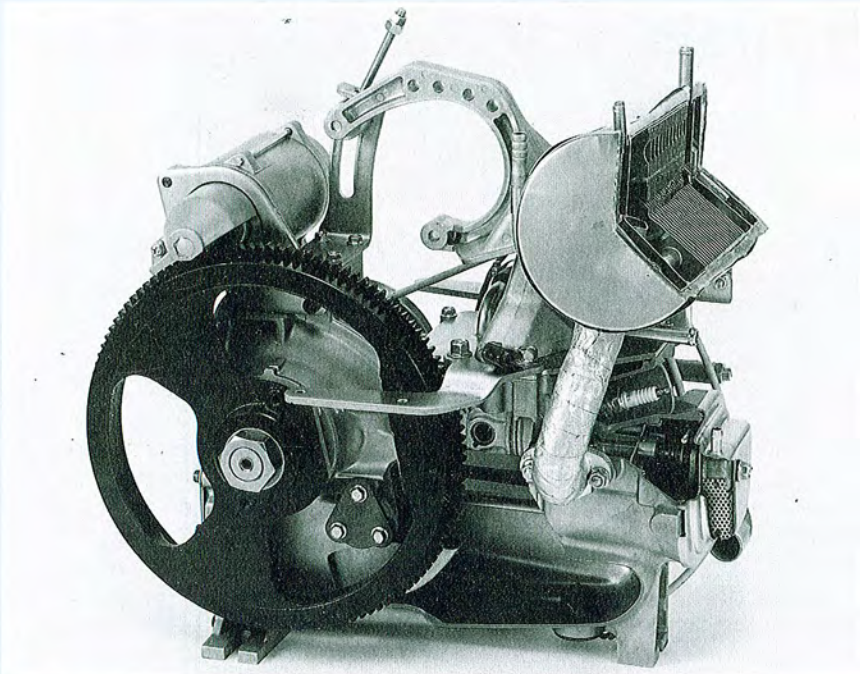


*Figure 2: The Engine for 1.3 RT (Sectioned Model).*

- 59 db(A)/1 for the 4.0 RT, and 60 db(A)/1 m for the 5.3 RT unit.

### • *Savings on Maintenance*

Main maintenance items are engine oil, ignition plug, belts, oil filter, air filter, and tappet clearance adjustment. The engine requires a special oil that ensures basicity for long-term operation and reduced sludge, and contains extreme pressure additives for low speed operation. The ignition plug uses a platinum electrode. The belts feature a reinforced wire and long life rubber. The tappet clearance of the 1.3 RT and 2.4 RT engines is designed so that it requires almost no adjustment. As for the electric source, the engine requires a 100V AC supply and no batteries. Although the maintenance interval differs from item to item, it is set at 2,000 hours according to the item with the shortest maintenance interval. In future, the interval will be increased.

### • *Automatic Operation*

Automatic operation commences upon designating the operation mode and pushing the start switch. The gas solenoid valve, starter motor, water pump, water solenoid valve, throttle actuator, etc. in the engine system are controlled automatically.

### • *Reliability*

For safe use of gas fuel, double gas solenoid valves are provided and the engine room is ventilated before starting the engine. When a problem occurs, the system is automatically stopped, and the problem mode is displayed. In an overload situation, the system is operated automatically to minimise the possibility of stopping its operation.

### • *Durability*

The valve train has been improved in wear resistance by selecting an excellent valve seat material and analysing the behaviour of the valve train. In addition, all sliding parts of the engine are subject to forced lubrication in order to reduce wear. Gas fuel leaves less deposits in the combustion chamber, which results in fewer piston ring and cylinder troubles.

### • *Auxiliary Device*

Auxiliary devices, such as a gas mixer, zero governor, throttle actuator, AC power 100V DC starter motor, AC power igniter unit and exhaust gas heat exchanger, have also been developed exclusively for GEHPs. The zero governor is designed so that it can withstand even the pulsations of the single-cylinder engine intake system, and the throttle actuator uses a stepping motor to improve its vibration resistance and response. The starter motor dispenses with batteries for simplified maintenance. The exhaust gas heat exchanger is made up of a welded stainless steel construction and excels in waste heat recovery and noise reduction performance. The heat exchanger resists the attack of drain water. Figure 2 shows the engine for the 1.3 RT heat pump.

## Future Advances

For future advances, it is necessary to reduce the $NO_x$ content of the exhaust gas, realise maintenance-free operation, and also continue to improve the technologies outlined above. Since GEHPs are characterised by low energy consumption, these small gas engines are expected to make progress as an air-conditioner driving source.

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# Development of a Medium-Size Cooling/ Heating and Hot Water Supply Gas Engine Heat Pump System

* *Shinya Kawasaki, Shigeaki Morimoto, Tadashi Fukuda, Yoshiharu Ito*

**SUMMARY**

The development goals for a 40 RT (141 kW) Gas Engine-driven Heat Pump are presented in this paper. The heat pump uses the waste heat from the gas engine as a heat source for hot sanitary water supply and for air-conditioning. The development goals, a cost-effective, automated energy conservation design, were achieved. The heat pump is intended for buildings such as hotels and sports centres.

## Introduction

The development of the gas engine heat pump (GEHP) began in 1980 in Japan, due to its remarkable energy conservation and cost-effective qualitiess. A GEHP system is a total energy system which uses a gas engine as the compressor driving source and utilises its waste heat for the hot water supply or air-conditioning heat source. During the following years there was a boom in heated swimming pools/health centres, running parallel with a sudden increase in the building of hotels. This resulted in many systems being installed around 1984, with a variety of unique systems being proposed for different uses. However, in the latter part of the 1980s the installation of new systems remained at a low level. From 1986 onwards, small-size gas engine heat pump air-conditioners (GHPs) began to sell remarkably well, resulting in the GHPs becoming synonymous as Japan's main gas engine heat pump (see Figure 1).

The increasing preference for GHPs may be explained by the ongoing pursuit of energy conservation, coupled with the growing demand for system simplification, simplified control, reduced installation costs and easier maintenance - needs which cannot necessarily be satisfied by the currently available GEHP systems. This being the case, in order to prevent GEHP systems from being restricted to acceptance only as a special system which is suitable for limited uses, it was imperative to develop more versatile products which could satisfy the market needs. This report therefore introduces a 40 RT GEHP which has recently been commercialised.

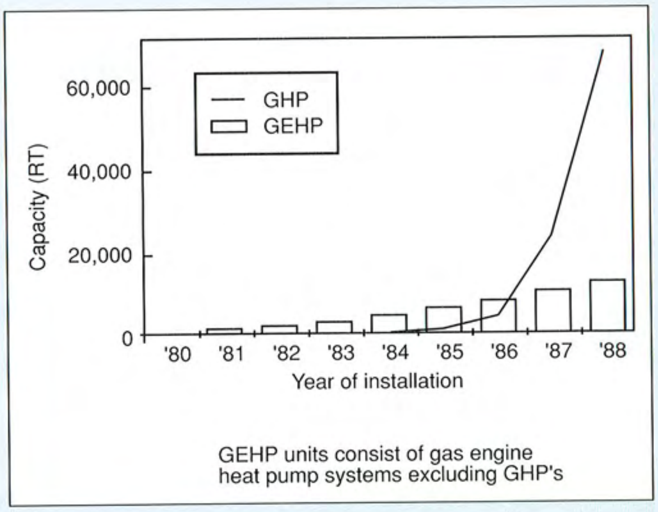



*Figure 1: Installed Capacity (Cooling Capacity).*

## System Problems

The difficulties encountered with conventional system designs have existed in realising an 'energy conservation design' most suitable for the piping/control equipment on the secondary side. Inevitably, the more energy conservation is pursued, the more complicated the equipment and operation becomes, resulting in a significant increase in equipment cost.

The main problems encountered have been as follows:

- High cost of equipment installation.
- Complicated equipment and mode selection operation.
- Too much time and labour for engineering.
- High overall system cost as GEHPs still need to be standardised.

## Development Goals of the New GEHP System

The new type GEHP system developed, can maintain a high efficiency comparable to conventional systems, and can also be as easily installed as a mono-function unit, leading to a reduction in equipment cost. The development goals for this system were as follows:

- ***Energy Conservation Design***

  Realisation of a high efficiency system which effectively utilises the condensation heat of the heat pump and the engine waste heat for heating the hot water supply.

- ***Reduced Cost of GEHP***
  Achievement of cost reductions by using R22 as a refrigerant and standardising the system, and by adopting a recently developed compressor specially suited for this type of GEHP.

- ***Hot Water Supply***
  Development of a control system which enables delivery of hot (60°C) water, despite the use of R22.

- ***Simplification and Reduction of Costs of Equipment on the Secondary Side***
  Elimination of complicated piping and control by arranging the hot water supply line as a one-way passage (see Figure 2).

- ***Simplified Operation***
  Since cooling/heating and hot water supply load control and mode selection is made automatically inside the GEHP according to the operating conditions, troublesome manual operation is unnecessary.

# System Outline

In April 1990, development of a system with a cooling capacity of 40 RT was completed. Table 1 shows the main specifications.

Figure 3 is a conceptual drawing of the system installation. Since the hot water supply line has been arranged as a one-way passage, the system installation has been simplified significantly. Either an open type or stratified type hot water storage tank can be used.

# System Flow & Capacity Characteristics

The system flow scheme is shown in Figure 4. The refrigerant system is composed of a compressor and three heat exchangers ((A), (B), (C)). The control scheme is based on the requirement for cooling/

*Table 1: Main Specifications*

| | | | |
|---|---|---|---|
| External dimensions (mm) | Height<br>Width<br>Depth | | 3,000<br>3,825<br>1,900 |
| Weight | | (kg) | 6,400 |
| Capacity | Cooling | (kW) | 140 (12°C to 7°C) |
| | Heating | (kW) | 155 (44°C to 45°C) |
| | Hot water supply | (kW) | 236 (60°C) |
| | Cooling + hot water supply | (kW) | (Cooling) 140<br>(Hot water supply) 81~236 |
| | Heating + hot water supply | (kW) | (Heating) 155~0<br>(Hot water supply) 81~236 |
| Electric power consumption | | (kW) | 5.7 |
| Gas consumption | | ($Nm^3/h$) | 14.6 |

*Figure 2: Hot Water Supply Flow Diagram.*

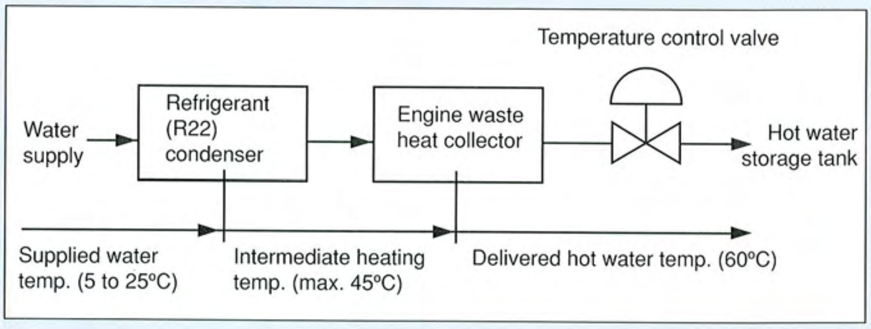
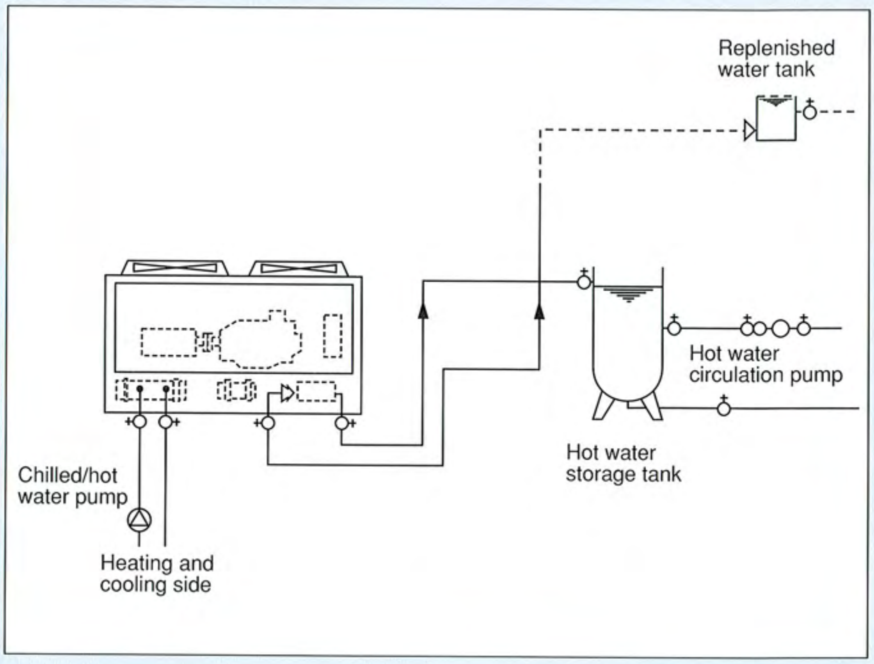



*Figure 3: Installation Layout.*



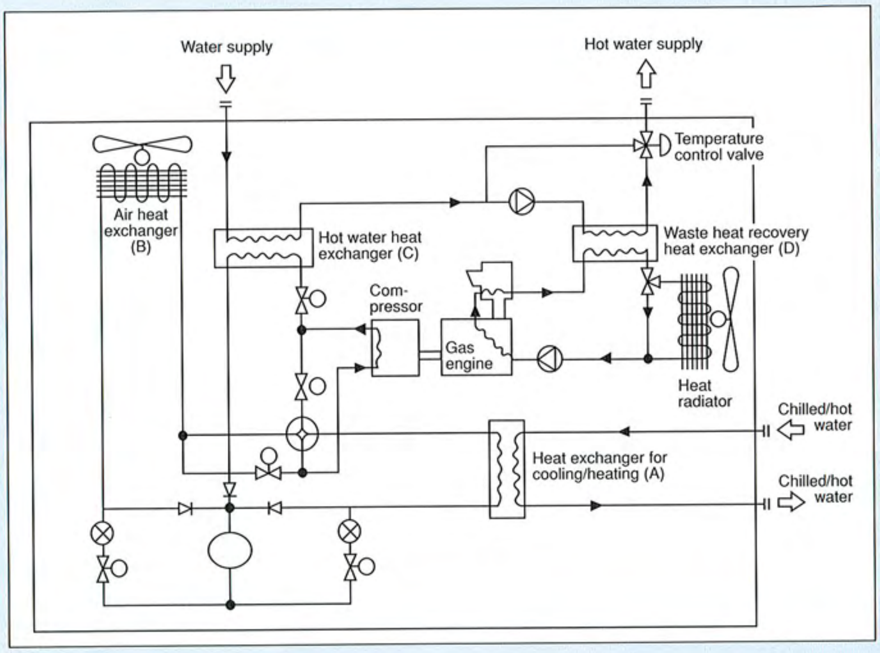



*Figure 4: Diagram of the System Flow .*

*Table 2: Operation Modes and Uses of Heat Exchangers (HEs)*

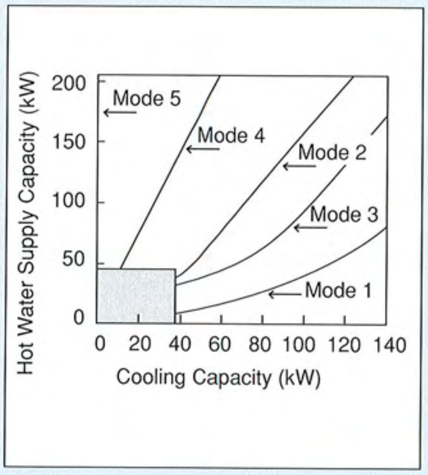
| Mode (load condition) | | HE for cooling (A) | Air HE (B) | HE for hot water (C) | Waste heat recovery supply HE (D) |
|---|---|---|---|---|---|
| | 1. Cooling>> hot water supply | EVA | CON | ___ | O |
| Cooling + hot water supply | 2. Cooling= hot water supply | EVA | ___ | CON | O |
| | 3. Cooling> hot water supply | EVA | CON | CON | O |
| | 4. Cooling< hot water supply | EVA | EVA | CON | O |
| Hot water supply | 5. Hot water supply alone | ___ | EVA | CON | O |
| Heating + hot water supply | 6. Heating>> hot water supply | CON | EVA | ___ | O |
| | 7. Heating+ hot water supply | CON | EVA | CON | O |

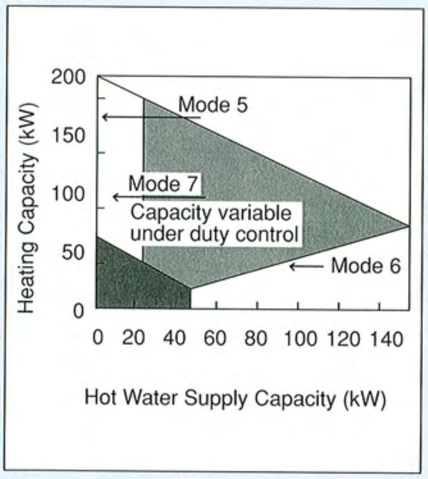
**O** = used ___ = not used **CON** =condenser **EVA** = evaporator

heating and hot water supply. It has 7 modes of operation which are related to the load ratio shown in Table 2. The microcomputer built in the system automatically selects an optimum mode. In the case of a requirement for "Cooling + hot water supply", for instance, the "Cooling = hot water supply" mode is the most efficient operation mode in which heat collected from the cooling load and the engine waste heat are totally used for heating the hot water supply. Usually, however, the cooling load does not coincide with the hot water supply load. So, continuous operation is possible at any heat load balance by shifting to mode 1 or mode 3 when the cooling load is high, or to mode 4 when the hot water supply load is high. Figure 5 shows relations between the operation modes and capacity characteristics.

In the case of "heating + hot water supply", as shown in Figure 5, the ratio of operation time per unit time in mode 7 is controlled by varying the amount of refrigerant to the heat exchanger for hot water supply according to the required heating and hot water supply load, and thereby subjecting the heating capacity and hot water supply capacity to distributive control. The overall system capacity is controlled within the range of 24% and 100%

*Figure 5: Operation Modes and an Example of Capacity Characteristics.*




**Cooling + Hot Water Supply**

(Measuring conditions)

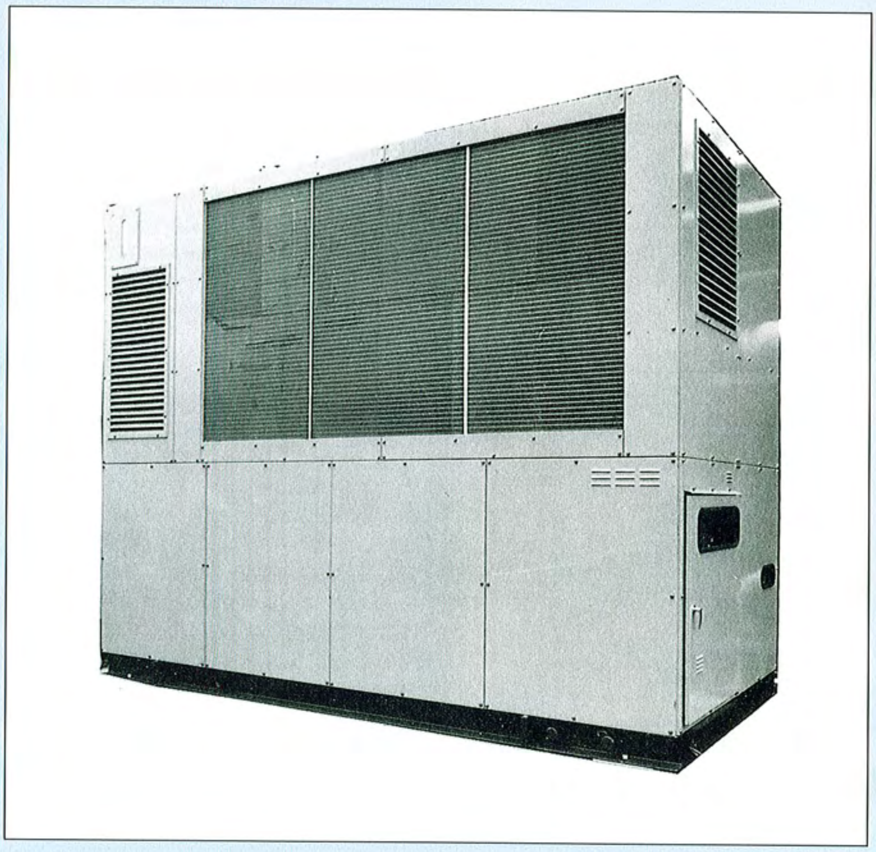
| | |
|---|---|
| Outdoor temperature: | 30°C |
| Chilled water temperature: | 12°C - 7°C |
| Water supply temperature: | 23°C |
| Hot water supply temperature: | 60°C |

**Heating + Hot Water Supply**

(Measuring conditions)

| | |
|---|---|
| Outdoor temperature: | 12°C |
| Warm water temperature: | 40°C - 45°C |
| Water supply temperature: | 15°C |
| Hot water supply temperature: | 60°C |

*Figure 6: 40 RT GEHP.*



through the engine speed control and the unloaded compressor control.

## Conclusion

The 40 RT multi-function gas engine heat pump is capable of heating/cooling and also supplying hot water (60°C). Compared to conventional gas engine heat pumps, it approx. yields a 40% reduction of the costs of the main unit and equipment. Our initial development goals - simplified operation, reduced maintenance cost, etc. - were also achieved. This system will be widely installed, mainly in buildings, such as hotels and sports centres where there is a substantial demand for hot water supply. Figure 6 shows the unit.

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# Combustion Engine-Driven Gas Heat Pumps

* S.I. Freedman, C.E. French and G.H. Myers


### SUMMARY

The Gas Research Institute (GRI) is conducting a large development programme on gas-fuelled heat pumps and chillers for residential and commercial market applications. Gas engine-driven systems play a prominent role in this programme. Field tests are underway on both a 3 RT (10.6 kW) and a 5 RT (17.6 kW) engine-driven vapour compression gas heat pump (GHP). The performance, reliability, and maintenance of these GHPs point to favourable prospects for commercialisation, thereby increasing the efficiency of gas heating from 95% to over 130% and increasing the efficiency of residential gas cooling from a gas-based COP of 0.6 to over 0.9. Substantial gas ratepayer benefits will result from such product commercialisation. Table 1 shows a comparison of steady state COP measured at standard ARI (Air Conditioning and Refrigeration Institute) rating points of 8.3°C (47°F) - heating - and 35°C (95°F) - cooling.


*Table 1: Comparison of Seasonal and Steady-State COP*

| | Seasonal COP | Steady-State COP |
|---|---|---|
| Heating COP (8.3°C, 47°F) | 1.3 | 1.7 |
| Cooling COP (35°C, 95°F) | 1.3 | 0.9 |

## Introduction

GRI is a private, not-for-profit organisation whose membership includes 37 natural gas pipeline companies, 156 natural gas distribution companies, 50 municipal utilities, 52 natural gas producers, and 21 associate members from gas companies outside the United States. GRI plans, manages, and develops financing for a co-operative research and development programme on natural gas which includes its supply, transportation, storage, and end use. One of the largest programmes that GRI is conducting is that of the development of gas-fuelled residential and commercial heat pumps and chillers.

Gas engine-driven vapour compression chillers and air-conditioners have been and are being field tested by GRI in sizes of 15 RT, 25RT, 150 RT, 250 RT, and 500 RT. The 150 RT unit has now entered commercial service.

Approximately 80 such units have already been installed and are operational, more are on order. Additional manufacturers are entering the gas engine-driven chiller field as customer acceptance grows. These gas-engine systems are mentioned here because, while they are not heat pumps in terms of the delivery of space conditioning heat, they do operate thermodynamically in cooling as heat pumps at condenser and evaporator temperatures very close to those of heat pumps and, consequently, could well be of interest to other nations whose commercial buildings are more conducive to the use of heat pumps than those in the U.S.

## Programme Overview

In a previous paper[1] the overall heat pump programme, including sorption systems and both external combustion engine and internal combustion engine-driven vapour

*Table 2: Status of Engine-Driven Gas Heat Pumps Under Development by GRI*

| Size | Technology | Status |
|---|---|---|
| 3 RT | Free Piston Stirling/Rankine | Stirling engine technology demonstrated. DOE continuing development with different contractor (Sunpower). |
| 3 RT | Heavy Duty Internal Combustion Engine/Rankine | Field test underway by Battelle. Briggs & Stratton/York planning commercial venture. |
| 3 RT | Rotary (Wankel) Engine/Rankine | Terminated. |
| 5 RT | Automotive Engine/Rankine | Market entry in Japan. Field test of version configured for U.S. market. |
| 7 RT | Rotary (Wankel) Engine/Rankine | Engine technology appraisal. |
| 10 RT | Kinematic Stirling/Rankine | Engine performance and reliability testing. |

compression systems, was described. In the past four years, significant progress has been made on the reciprocating engine-driven system. Table 2 shows the status of the engine-driven gas heat pumps reported in the 1986 paper and those currently under development or being field tested now.

The free-piston Stirling engine 3 RT heat pump was built as a research model and tested. It was found to produce COPs of just over 0.9 (cooling) and 1.6 (heating) under ARI conditions. While these performance values are attractive, the engine required close control of manufacturing tolerances and did not present adequate promise for low cost production of a high performance system. Consequently, work on the free-piston Stirling engine has been ended. The 3 RT rotary engine project at Wedtech terminated with their bankruptcy. The 7 RT commercial rotary engine-driven heat pump concept is being re-evaluated since the air charge cooled engine design does not provide adequate rotor bearing life. The other 10 RT commercial heat pump, driven by a kinematic Stirling engine, has been involved in engine performance and reliability testing and awaits favourable data on these characteristics as well as the participation of a company to bring a heat pump into the commercial market.

## The 3 RT and 5 RT GHPs

The two engine-driven gas heat pump projects with most GRI activity at this time are the field test programmes on the Battelle/Briggs & Stratton/York 3 RT heat pump and the Aisin Seiki 5 RT heat pump. Figure 1 shows the 3 RT unit and Figure 2 shows the 5 RT unit. They both use gas-engines to drive mechanical compressors but differ appreciably in component selection.

The 3 RT heat pump is based on a long-standing 5 hp (3.7 kW) engine designed by Battelle Columbus Laboratories with participation by Ricardo Consulting Engineers (U.K.) and subsequently developed by Briggs and Stratton with the inclusion of manufacturing considerations. The engine contains a number of features to ensure the highest reliability and longest practical engine lifespan without requiring an overhaul. It includes many of the engineering techniques and materials used on large, high power engines used in pipeline compressor station engines, locomotive and ship propulsion diesel engines, and similar engines noted for their long lifespan and high reliability. Engine efficiency of 27% based on the higher heating value and $NO_x$ emissions of 1.5 g/hphr [= 2 g/kWh output or approximately 170 g/GJ fuel input, ed. HPC] have been demonstrated. Endurance testing predicts an engine lifespan in excess of 40,000 hours as long as there is an annual maintenance.

The 5 RT heat pump is based on the use of an automotive engine and automotive air-conditioning compressor. These components were originally selected for use in the Japanese market, where required product lifespan is 20,000 hours.

The 3 RT heat pump uses a direct-driven open shaft version of a hermetic reciprocating compressor, also having an expected lifespan of 40,000 hours, and the 5 RT heat pump uses two belt-driven automotive scroll compressors having an expected lifespan of 10,000 hours.

In the 5 RT unit, the performance and reliability of the automotive engine and compressors in heat pump service in Japan is recognised as being of considerable value in achieving market entry in the U.S. without facing the difficult hurdle of manufacturer investment in a totally new engine. It is to be pointed out here that GRI developed the special 5 hp (3.7 kW) residential heat pump engine for the 3 RT unit because no engine was available in that size range with the requisite long lifespan features. GRI will be following the experience of the Japanese with their residential and commercial engine-driven gas heat pumps to

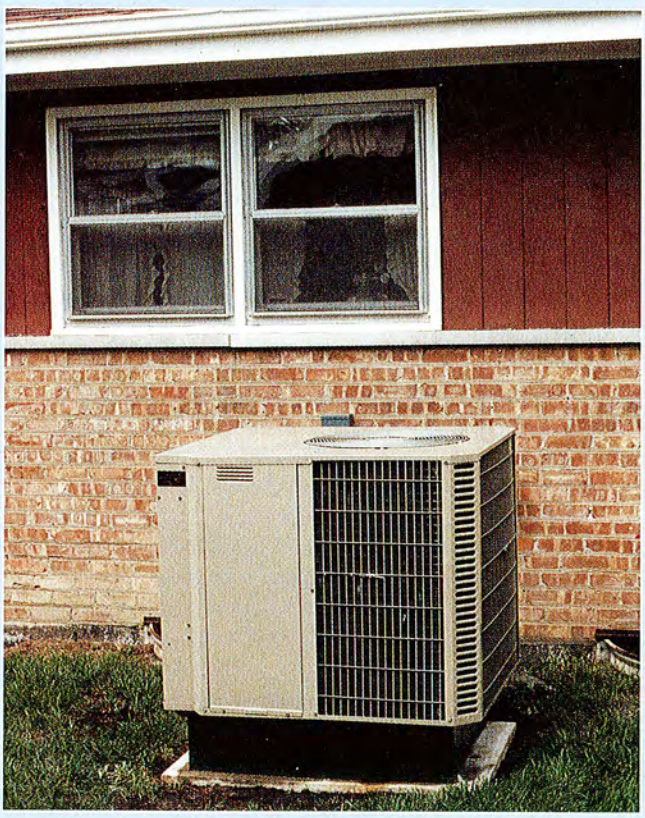

*Figure 1: Battelle 3 RT Gas Engine-Driven Heat Pump.*

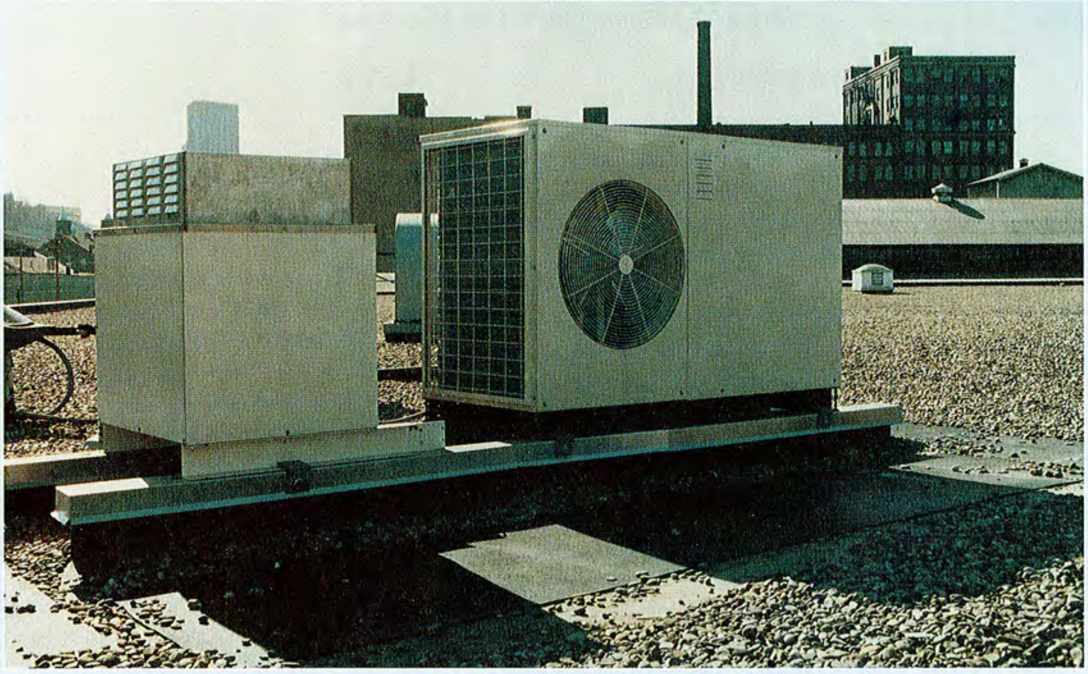

*Figure 2: Aisin Seiki 5 RT Gas Engine-Driven Heat Pump.*

see whether their lifespan is long enough for consumer acceptance in the U.S. with its (regionally) severe and long winter and summer space conditioning system operation requirements. The moderate Japanese climate eases the lifespan requirement on an engine when used to provide space conditioning in Japan. One of the purposes of the GRI field test is to determine the annual load profile of an engine-driven gas heat pump in actual service in representative climates in the United States to form a basis for engine lifespan testing as well as for correlation and normalisation of computer programs that simulate residential space conditioning system performance.

In order to ensure that the heat pumps will provide adequate heat at low outdoor temperatures, they have gas-fired supplemental heaters. These are of an efficiency slightly above the minimum requirements of the National Appliance Energy Conservation Act (NAECA) of 1987. As the supplemental heaters will operate only for short periods, such minimum efficiency is economically justified since the high COP engine-driven system with engine exhaust and engine coolant heat recovery will provide the majority of the heat on a seasonal basis.

Because of the use of engine heat and the supplemental gas-fired heaters, engine-driven gas heat pumps are expected to provide excellent comfort. Under the same operating conditions, a GHP will always provide a higher delivered air temperature than an electric heat pump. Capacity modulation provides better load matching and better humidity control. This variable speed feature has recently been incorporated into the latest models of electric heat pumps at a considerable increase in cost.

## Field Tests

The field test sites are shown in Table 3. They were selected in order to obtain adequate variability in climate to test heat pump features which are difficult to simulate in the laboratory, such as defrosting and cycling; to determine the adequacy of the variable speed engine loading control system in providing full indoor comfort with the least shortening of engine lifespan; and to expose the systems to extremes of cold winters and hot summers. It is to be noted that several of the host utilities are combination gas and electric utilities. Several of these utilities have recognised that the use of gas heat pumps can contribute to the resolution of their expected problem of meeting demand during peak summer periods in the mid-1990s.

Each of the 3 RT and 5 RT heat pumps is equipped with an extensive data acquisition system (DAS). These systems have been standardised by GRI to provide uniform instrumentation, reporting, and performance analysis. The DAS measures key heat pump operating parameters, as well as indoor and outdoor temperatures and gas and electricity used to operate the heat pump. The data area analysed and provide invaluable performance feedback to the equipment designer, as well as operating economic information for use in comparing the performance of the engine-driven gas heat pumps with other space conditioning systems.

## Conclusion

With the completion of the field test of these units in 1991, it is expected that GHPs will then be introduced on a limited basis by the manufacturers in the 1992 time period, and on a general commercial basis shortly thereafter. The availability of such space conditioning equipment is expected to offer ratepayers lower energy bills to heat and cool their homes and shops and will alleviate the electric utilities problem of

| *Table 3: Field Test Sites:* | *Briggs & Stratton / York 3 RT Heat Pump* | |
|---|---|---|
| **Location** | **Climate/Remarks** | **Utility** |
| York, PA | York Test House/ Component Improvement | |
| Chicago, IL | GRI Instrumented Test House | Peoples Gas Co. |
| Girard, OH | Cold winter, moderate summer DHW auxiliary heat | The East Ohio Gas Co./ Consolidated Natural Gas Co. |
| Wheaton, IL | Cold winter, moderate summer | Northern Illinois Gas Co. |
| Baltimore, MD | Mild winter, humid summer | Baltimore Gas & Electric Co. |
| Maplewood, NJ | Average winter, average summer | Public Service Electric & Gas Co. |
| Atlanta, GA | Hot, humid summer | Atlanta Gas Light Co. |
| Phoenix, AZ | Hot, dry summer | Southwest Gas Company |
| Salt Lake City, UT | High altitude | Mountain Fuel Supply Co. |
| New York City, NY | Average climate with modified control strategy | Brooklyn Union Gas Co. & Consolidated Edison |
| | *Aisin Seiki 5 RT Heat Pump* | |
| **Location** | **Climate/Remarks** | **Utility** |
| Detroit, MI | Cold winter, moderate summer | Consumers Power Co. |
| Pittsburg, PA | Moderate winter, moderate summer | Equitable Resources, Inc. |
| Tulsa, OK | Moderate winter, hot summer | Oklahoma Natural Gas Co. |
| Salt Lake City, UT | High altitude | Mountain Fuel Supply Co. |
| Atlanta, GA | Hot, humid summer | Atlanta Gas Light Co. |
| Mesa, AZ | Hot, dry summer | City of Mesa |
| Los Angeles, CA | Mild | Southern California Gas Co. |
| Brooklyn, NY | Average climate | Brooklyn Union Gas Co./ Consolidated Edison Co. of New York, Inc. |
| Toronto, Canada | Cold winter, mild summer | The Consumer's Gas Co., Ltd. (Canada) |

building new power plants to meet the growth of summer peak demands, which may shortly be severe in selected regions. Figure 3 shows the percentage of new single family residences built in the United States between 1983 and 1987 that have central air-conditioning. It is apparent from the figures that since over 70% of new single family residences have central air-conditioning, there must be a gas heating and cooling option in order to compete effectively in the space conditioning market.

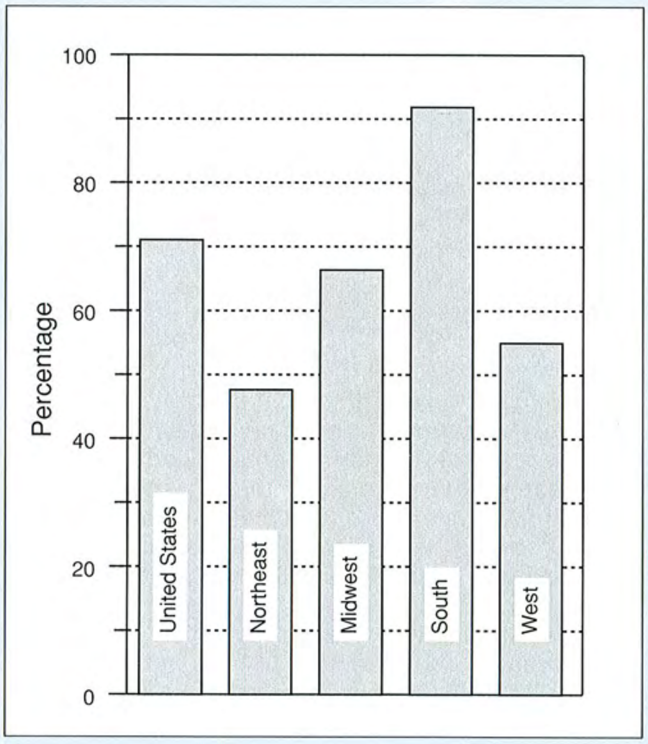


*Figure 3: Percentage of New Single Family Homes with Central Air-Conditioning (1983 -1987).*

*Source: Characteristics of New Housing.*

*References:*
*[1] Freedman, S.I., and Maret, A.R., "Heat Pump Research at the Gas Research Institute, "IEA-HPC Newsletter, Vol. 4, No. 3, October, 1986, p. 26.*

*Authors:*
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# The Amsterdam World Trade Center's (W.T.C.) Heat Pump

*P.H.H. Leijendeckers

**SUMMARY**

This article describes the integration of a gas engine-driven heat pump system in the air-conditioning and heating system of the World Trade Center premises in Amsterdam. The heat pump provides for both cooling and heating during summer and winter periods and also provides emergency power in the case of interrupted public power supply.

*Table 1: Technical Details*

| Conditions | Summer (kW) | Winter (kW) |
|---|---|---|
| Cooling capacity: | 1952 | 1645 |
| Power consumption compressor: | 531 | 446 |
| Shaft power gas engine: | 551 | 462 |
| Condenser cooling capacity: | 2363 | 1989 |
| Oil cooling capacity: | 120 | 102 |
| Gas consumption (gross heating value): | 1799 | 1508 |
| Jacket cooling system: | 574 | 481 |
| Exhaust gas cooling system: | - | 350 |
| Miscellaneous (effective): | 45 | 39 |
| Heat pump output: | - | 2961 |
| PER factor: | - | 1.96 |
| Cooling capacity per kW natural gas supply: | 1.09 | |

## Introduction

The design of the Amsterdam World Trade Center's (W.T.C.) heat pump system dates from the beginning of the 1980s and is based on the technology which was frequently used at that time. The system covers three functions, namely that of a heat pump in the winter, a refrigeration system in the summer, and an emergency power supply system whenever necessary. The system comprises a gas engine which drives the screw compressor of a chiller. This chiller is part of the refrigeration system and also functions as a heat pump whenever the heat can be used efficiently to heat the building. Heat is recovered from the ventilation air before it leaves the building. The waste heat from the gas engine's cooling system and the exhaust gases are also used. In addition, the gas engine drives a 420 kW electric generator to provide emergency power. While emergency power is supplied, the screw compressor is disconnected from the gas engine via an electromagnetic coupling. The heat pump has a capacity of 3 MW, or 37.5% of the estimated heat required to heat the building. The cooling capacity is 1.95 MW, or 40% of the total cooling load. Technical data are given in Table 1.

## Amsterdam World Trade Center

The W.T.C. building has a two storey car park with 1200 parking spaces. Four office blocks have been built above the underground car park. Two of the office blocks have seventeen floors and the others have twelve. The total floor area is 110,000 m$^2$; the offices and service areas cover 75,000 m$^2$ and the car park covers 35,000 m$^2$. The total volume of the building is 350,000 m$^3$. Each office block has its own rooftop air-conditioning unit. One of the two tallest office blocks also provides space in the rooftop for the central boiler house and the other for the refrigeration plant for the whole of the complex (see photograph on the cover of the Newsletter).

## The Heat Pump

The heat pump (Figure 1) is built around a turbo charged G-398-TA Caterpillar gas engine with a shaft power of approx. 550 kW. This engine continuously drives a 420 kW electric emergency power generator and a GHH CR-200 type screw compressor via an electromagnetic coupling to cover the cooling load. While emergency power is supplied, the screw compressor is disconnected. When the system functions as a heat pump, the condenser heat and the heat from the jacket cooling, as well as the lube oil and exhaust gas cooling of the gas engine is supplied to the central heating system. Heat is then recovered via the chiller's evaporator, from the exhaust air and the building's chilled water circuit, which is installed throughout the building to provide local cooling.
In the summer, the above system and a second electrically-driven cooling compressor system both cool the building. All excess heat is then discharged into the atmosphere via cooling towers. The heat pump functions throughout the year, both during the summer and the winter.

Figure 2 shows the complete design of the heating and cooling systems. It illustrates the relationship between the boiler system and the refrigeration system, on the one hand, and the heat pump as a linking component, on the other. The heat pump removes heat from the cooling circuit which is designed for a supply temperature of 0°C and a return temperature of 5°C. The central heating system is designed for a return temperature of 35°C or lower depending on the load, which is possible because the building is heated by an air heating system. The return water first flows through the condenser and then through the gas engine's lube oil cooling and the jacket water cooling systems. Finally, the heat from the exhaust gases is recuperated. The central heating water is heated in this way to 45°C if the outside air temperature is 0°C and two additional boilers are used to heat the water to 60°C if the outside air temperature is at -10°C. The system's total heating capacity is 7.8 MW, of which 3 MW is supplied by the heat pump.

During the summer period, when the building does not have to be heated, the heat pump and two other monoscrew compressors operate together in one cooling system. Both electrically-driven machines cool the return water in the cooling system from 15°C to 5°C before it flows to the heat pump where it is further cooled to 0°C. The cooling system's temperature range is between 0-15°C so that the diameter of the piping system can be kept to a minimum. The system's total cooling capacity is 5.2 MW, of which 1.95 MW is supplied by the heat pump.

The building has a so-called secondary chilled water circuit with a temperature range between 12-16°C so that extra local cooling can be provided for local cooling facilities near computers and other equipment. This system is connected to the main cooling system via a plate heat exchanger and has a capacity of 500 kW. During the past few years, it has proved to be effective, which is why this type of system is being used increasingly in new designs.

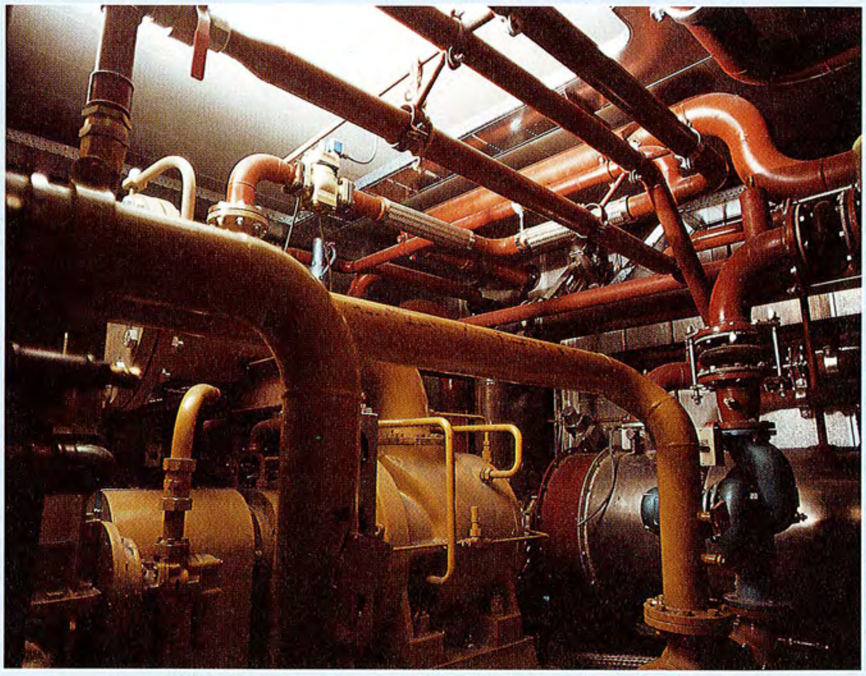


*Figure 1: Gas Engine-Driven Heat Pump.*

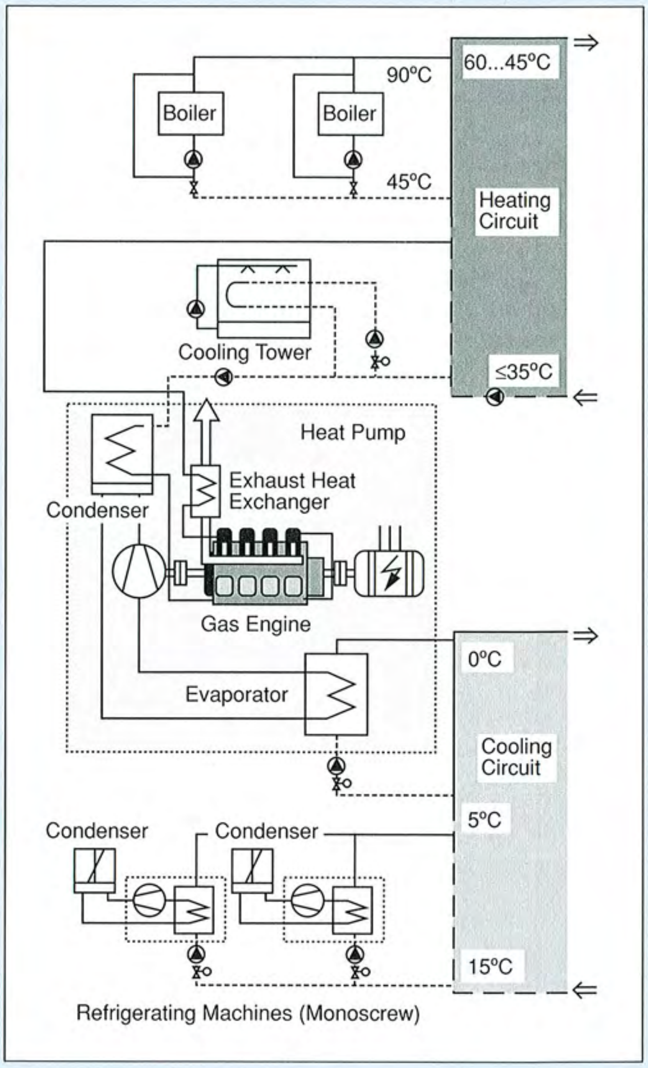
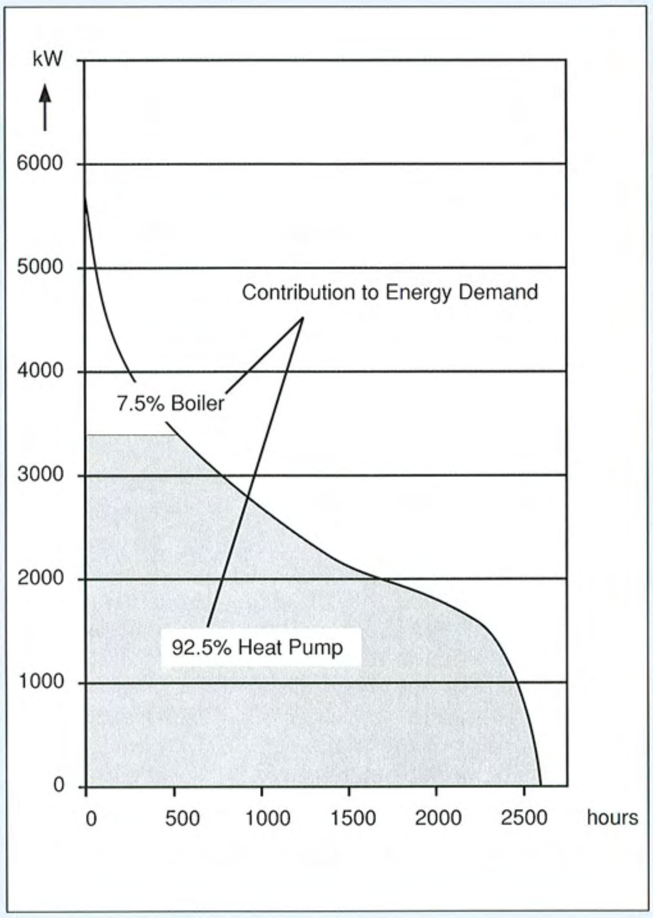



*Figure 2: Heating and Cooling System.*

*Figure 3: Annual Heating Load Diagram for Office Hours.*



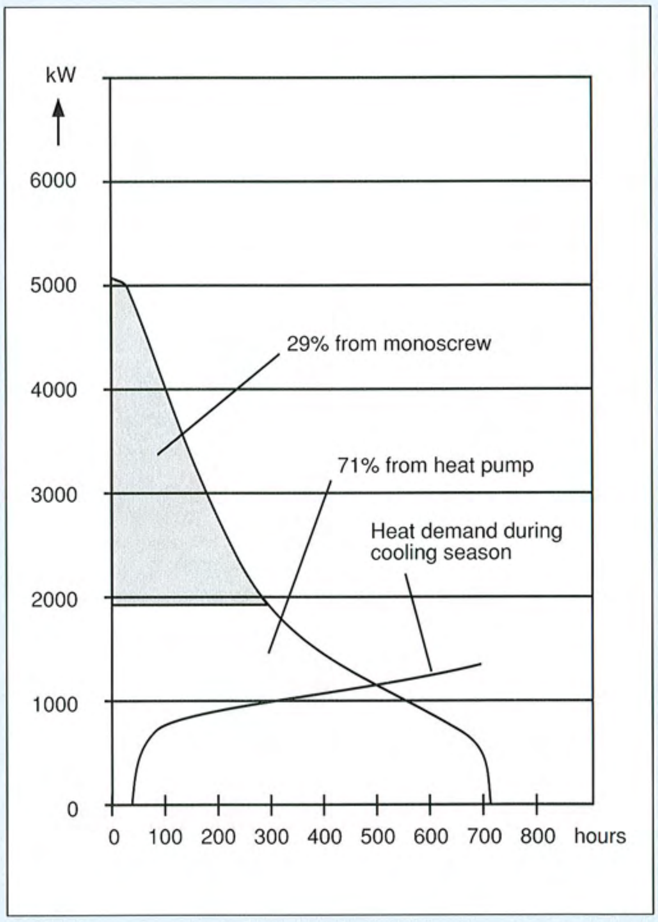




*Figure 4: Annual Cooling Load Diagram for Office Hours.*

# Energy Consumption

Figures 3 and 4 show the annual load diagrams for the heating and cooling systems. It can be deduced from these diagrams that during office hours the heat pump can provide approximately 92% of the heat requirement (21.3 TJ) and 71% of the cooling requirement. The annual amount of gas consumed by the heat pump is 286,000 $m^3$ and 480,000 $m^3$ by the boilers. It is estimated that about a 34% saving can be avhieved on primary energy when compared to a conventional design with boilers and an electrically-driven cooling system. When the system was installed in 1984, a payback period of less than five years was estimated, based on the energy prices at that time.

# Experiences So Far

The installation is functioning satisfactorily. Apart from a failure of the screw compressor during the guarantee period the installation has been fully available till to date under normal maintenance conditions. The gas consumption of the building complex has developed quite favourably to a level of 525,000 $m^3$ during 1989. This is about 30% less than the original estimate. A reason for this is the high internal heat load from the office equipment, which during the heating season reduces the heat demand of the building considerably.

*References:*

*A. Jansen; Technical Installations World Trade Center, Energie-consulent, No. 3/1987 (in Dutch).*

*Ir. H.J.M. Knipscheer; Practical Application of Large Capacity Heat Pumps, Energieconsulent, No. 3/1987 (in Dutch).*

*Author :*

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*Technical Consulting Office van Heugten, The Netherlands.*

# Preliminary Screening of Processes for Industrial Heat Pump Applications

* P. Tripathi, Y.T. Chao

**SUMMARY**

A procedure for quick preliminary screening of industrial processes to determine their suitability for cost-effective heat pump placement is presented. This procedure can be used to identify heat pump markets as well as development needs for advanced heat pumps. As Phase I of a United States Department of Energy (USDOE) funded project, this procedure was applied to 26 diverse industrial processes and many good opportunities have been identified.

## Introduction

A preliminary screening procedure for heat pump applications was developed as part of Phase I of the USDOE funded work in response to the following questions:

1. Which industrial processes have the highest likelihood of cost-effective heat pump placement?
2. What are the characteristics of an industrial process for which advanced heat pumps should be developed?
3. What development are required for advanced heat pumps?

## The Fundamentals

The procedure developed was based on (a) the concepts of process pinch and Grand Composite Curve, (b) the theory of correct thermodynamic placement and appropriate sizing, and (c) the definition of COP and the fact that it varies inversely with the temperature lift. These topics have been discussed in length in several other sources[1-5].

Visualisation of the process Grand Composite Curve (GCC) gives the first indication of whether a process is suitable for heat pump placement or not. The GCC gives a "profile" of the energy requirement of a process. It also provides important information on the heat pump potential present in a process.

To be effective, a heat pump must be "appropriately" placed. In the concept of pinch technology, this means that the heat pump must move thermal energy from below the "pinch" temperature where there is a surplus of thermal energy, to above the pinch where there is a need of thermal energy. A pinch must be present for a process to be suitable for heat pumping opportunities. Whereas most processes have both net heating and net cooling requirements, there are some so called "threshold processes" where either the heating requirement or the cooling requirement is absent. Another requirement for heat pump placement is that the temperature lift dictated by the structure of the GCC must be attainable by available compressors.

## Procedure

The procedure consists of the following steps:

1. Calculate or approximate temperature enthalpy data for process streams which undergo a change in heat content during processing.
2. Construct a Grand Composite Curve. An important parameter defining the shape of the grand composite is the specified minimum approach temperature ($DT_{min}$) between the hot and cold composite curve.
3. Determine the pinch region (when present). This tells us at which level a heat pump may operate beneficially in the particular process.
4. Visually inspect the shape of the grand composite at and near the pinch to determine the suitability for heat pump placement.
5. Compute the COP corresponding to the lowest feasible temperature lift.
6. Use the COP and the grand composite as tools for preliminary screening.

## General Results

The basic function of a heat pump is to absorb heat at a low temperature and deliver heat at a higher temperature. In order for a heat pump to fulfill this function it is necessary that there should be both a suitable heat source and a suitable heat sink. If the heat pump is to be used in an industrial application, the source and the sink should be within an industrial process.

The conventional processes for the production of ammonia, sulphuric acid, and nitric acid are all examples of a "threshold process". It is clear that such processes do not satisfy the basic requirement for heat pump application.

Certain unit operations are intrinsically amenable to heat pump applications. Evaporation and distillation, both of which have large, constant temperature heating and cooling duties, are two such operations. Some drying processes are also good candidates. Heating of process water is also often a good candidate for heat pumping. However, the presence of one or more of these unit operations within a process is not a guarantee that a suitable application exists. Moreover, it is essential that no unit operation should be viewed in isolation when the possibility of heat pumping is considered. Rather, the entire process must be analysed as an integrated entity to ensure that the most appropriate energy saving strategies are adopted.

## Specific Results

This screening procedure was applied to 26 processes[6]. The processes ranged from petroleum refining and organic chemicals to food processing and inorganic chemicals. The result of the screening shows the following process to have good potential for heat pumping:

- Ethylene (above ambient)
- Aromatics-BTX Unit*
- Crude Unit
- Chlorine-Caustic Soda
- Urea
- Food Processing
- Sugar Refining
- Liquor Distilling
- Pulp and Paper Evaporator
- Pulp and Paper Digester

*BTX = Benzene- Toluene- Xylene.

The following processes were found to have less than attractive heat pump potential:

- Ethylene (below ambient)
- Sodium Chloride (salt)
- Milk Processing
- Beer Brewing
- Polypropylene
- Aromatics-Prefractionator
- Aromatics-Reformer
- Terephthalic Acid
- Ammonia
- Nitric Acid
- Sulphuric Acid
- Soap Manufacture
- Coffee
- Fluid Catalytic Cracking
- Textile Milling
- Margarine Manufacture.

The data used in this screening does not reflect the operating conditions of an actual site but is typical of the process being investigated. The results of this screening depend on the definition of the process boundary under consideration. Numerical results are summarised in Tables 1 and 2 for those good candidates.

Some typical Grand Composite Curves are shown in Figures 1, 2 and 3. Figure 1 illustrates the grand composite curve for the manufacture of ammonia via hydrocarbon reforming. All sections of the plant, including desulphurisation, reforming, $CO_2$ shift, $CO_2$ removal, methanation and ammonia synthesis, have been used to construct the Grand Composite Curve. For a practical $DT_{min}$, it is a threshold problem and is not suitable for heat pump placement.

Figure 2 illustrates a grand composite for the manufacture of soap. It is an example of a process with an ill-defined pinch region.

The grand composite curve for the Aromatics-BTX is presented in Figure 3. This is an ideal process for heat pump placement. The pinch region is sharply defined. The pinch also occurs at a reasonable low temperature making conventional working fluids a possibility.

## Limitations

The heat pumps specified in Tables 1 and 2 are the ones with the lowest feasible lift. Identification of the "optimum" heat pump cycle to match against any given process Grand Composite Curve requires a matching of heat pump lifts with process requirements. An evaluation of the sizes and costs of

| Process | Lift (°C) | Delivery Temp. (°C) | Hot Utility* Saving (%) | Type+ |
|---|---|---|---|---|
| Sugar | 72 | 143 | 15 | a |
| Liquor | | -- | | -- |
| Chlorine/Caustic | 83 | 149 | 45 | a |
| BTX | 78 | 204 | 31 | a |
| Ethylene | 39 | 88 | 10 | a |
| | 56 | 121 | 30 | b |
| Food (Gel) | | -- | | -- |
| Crude | 139 | 316 | 50 | b |
| Urea | 67 | 177 | 50 | a |
| Pulp & Paper Digester | | -- | | -- |
| Pulp & Paper Evaporator | 56 | 99 | 64 | a |

* = Hot utility saving as approximately % of residual load after conventional heat pump has been installed.
+ = Type: a - Implies constant temperature load (latent heat). b - Implies variable temperature load (sensible heat).

*Table 1 : Scope for Advanced Heat Pumps.*

*Table 2: National Replication Potential for Identified Heat Pumps.*

| Process | Temp. Lift (°C) Closed | Temp. Lift (°C) Semi-open | Delivery Temp(°C) | Current Heat Pump Load* MW | No. of Plants | National Savings GJ/Yr x $10^{-6}$ |
|---|---|---|---|---|---|---|
| Sugar | 28 | 19 | 139 | 41.0 | 36 | 7.1 |
| Liquor | 29 | 21 | 113 | 7.0 | 126 | 5.2 |
| Chlorine/Caustic | | -- | | | 33 | |
| BTX | 31 | 24 | 158 | 5.0 | 26 | 3.3 |
| Ethylene | 25 | 38 | 94 | 6.5 | 33 | 3.5 |
| Food (Gel) | 17 | 8 | 110 | 0.5 | 6 | 0.3 |
| Crude | 36 | 34 | 186 | 2.0 | 230 | 9.4 |
| Urea | 31 | -- | 139 | 3.0 | 35 | 2.8 |
| Pulp & Paper Digester | 27 | 20 | 143 | 2.0 | 45 | 2.5 |
| Pulp & Paper Evaporator | 21 | 14 | 63 | 1.0 | 45 | 0.6 |
| Total | | | | | 582 | 34.7 |

Overall national percentage reduction by heat pumping = 10.5% allowing for relative weighings of processes.

* Load based on heat pump evaporator (or equivalent load for semi-open cycles) in the specific plant studied. Heat pump condenser load is typically 10% to 20% greater.

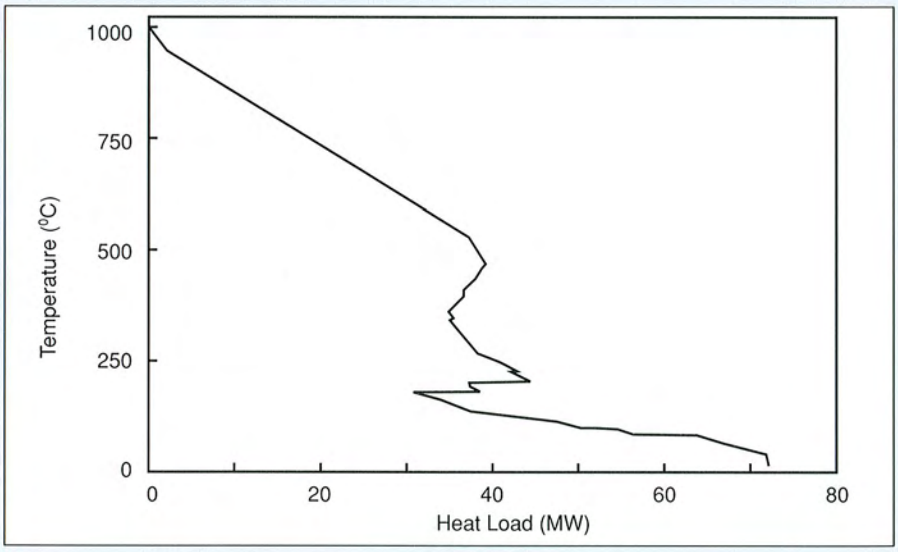



*Figure 1: Grand Composite Curve for Ammonia Process ($DT_{min}$ = 28°C).*

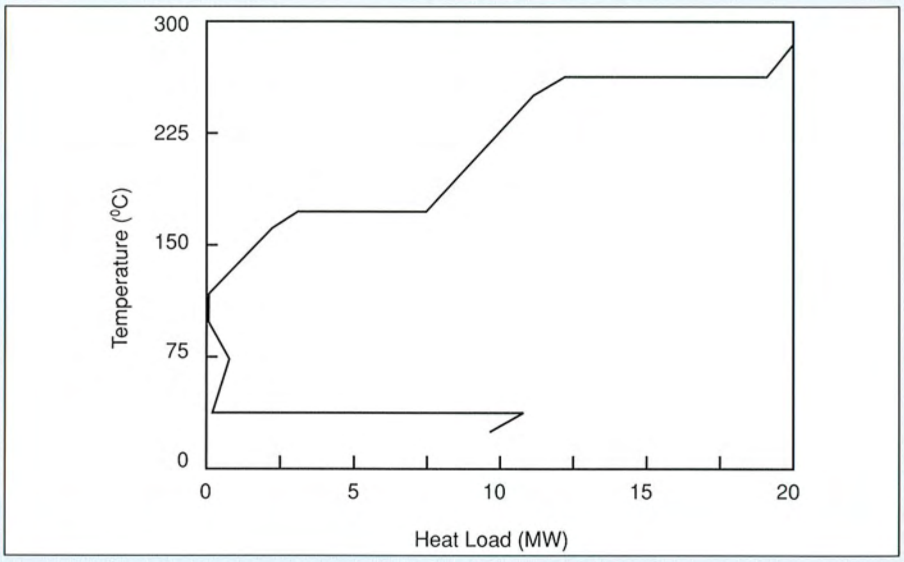
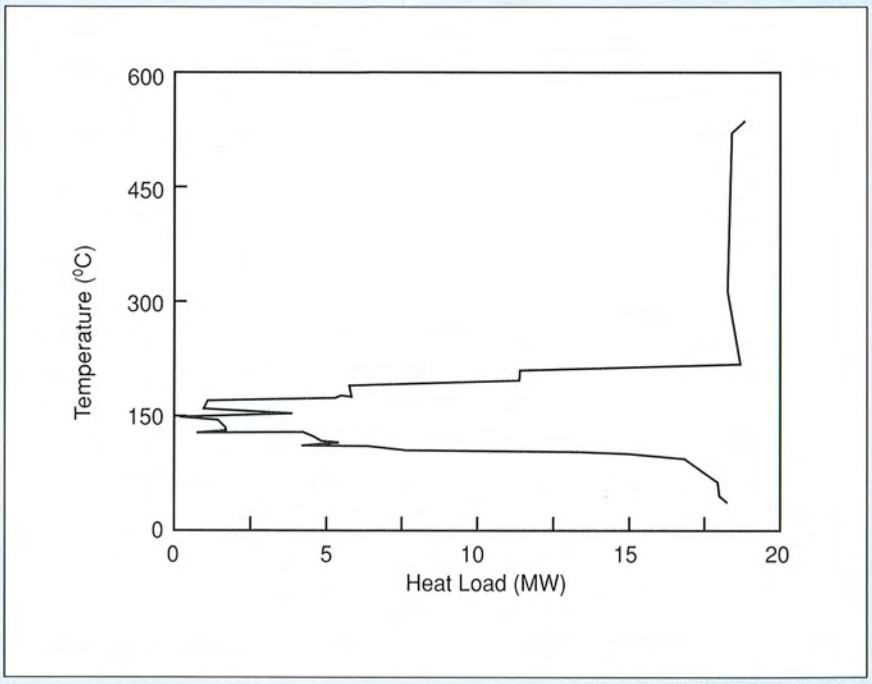



*Figure 2: Grand Composite Curve for Soap Manufacture ($DT_{min}$ = 11°C).*

*Figure 3: Grand Composite Curve for BTX Unit ($DT_{min} = 11°C$).*




the associated evaporators, condenser, and compressors is required.

The results obtained in this study should be viewed as a general indication concerning potential for heat pump placement rather than a definite verdict. Other factors such as the cost of fuel and power, the availability of fuel and power, and payout requirement etc., have to be considered before a conclusive decision can be reached.

## Conclusions

Application of the pinch concept to appropriate placement and sizing of industrial heat pump systems has been demonstrated through a step by step procedure. As a result of the knowledge and insight obtained in this study, further works have been conducted by TENSA Services through fundings from USDOE and EPRI (Electric Power Research Institute). A computer program HPSCAN (Heat Pump SCreening ANalysis)[7] funded by EPRI was developed to systemise the pinch analysis and screening procedure. Several selected site studies have further confirmed the result of this project. The continuation of this study is currently underway toward the second phase which aims at design, fabrication, and implementation of several of these heat pump applications.

*Acknowledgement:*

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*References:*

[1] *Hindmarsh, E., D. Boland, and D.W. Townsend, 1985, "Maximizing Energy Savings for Heat Engines in Process Plants", Chem. Engineering, February 4, p. 38.*

[2] *Ranade, S.M., E. Hindmarsh, and D. Boland, 1987, "Industrial Heat Pumps: New Insights on Their Integration in Total Sites", 3rd Int. Syp. on Large Scale Applications of Heat Pumps, Oxford, England.*

[3] *Ranade, S.M. And Y.T. Chao, "Industrial Heat Pump: Where and When". Proceedings of the 11th IETC Meeting, 1989.*

[4] *TENSA Services Inc., "Heat Pumps in Complex Heat and Power Systems", Report Prepared for EPRI, Palo Alto, California: EPRI EM-4694, April, 1989.*

[5] *Chappell, R.N. And S.J. Priebe, "Process Integration of Industrial Heat Pumps", Proceedings of the 8th IETC Meeting, 1986.*

[6] *TENSA Services Inc. "Optimum Heat Pump Placement in Industrial Processes", Report Prepared for the U.S. DOE, DOE/ID/12583-1, March, 1987.*

[7] *TENSA Services Inc., "Heat Pump Screening Analysis - HPSCAN User's Manual", Report and Software Prepared for EPRI, Palo Alto, California: EPRI CU-6446-CCML, Sep. 1989.*

*Authors:*
** Paul Tripathi is the president of TENSA Services Inc. and Y.T. (Peter) Chao is a Senior Engineer in TENSA, USA.*

# News and Views

## Refrigerants/CFCs

### Use of CFCs Prohibited by the Year 2000

At the end of June the "Second Meeting of the Contracting Parties to the Montreal Protocol on Substances that Deplete the Ozone Layer" was held in London. At this international meeting the ministers of the environment decided to ban the production and consumption of CFCs and halons by the year 2000, instead of halving it as was agreed in the 1987 Montreal Protocol. In order to aid developing countries to attain this goal, the industrialised countries will establish a US$ 240 million fund for the forthcoming three years.

The existing obligation to report on production and import of CFCs was extended to include R22. In a separate declaration regarding R22 the countries committed themselves to handling R22 with care and to strive towards replacing R22 by environmentally, fully acceptable, refrigerants between 2020 to 2040.

*(Source: private communication TNO.)*

### Germany Bans CFCs at Higher Pace

A few weeks before the above international conference in London, Germany announced that it is to ban the domestic use and production of CFCs by 1995.

In 1991 the use of CFCs in aerosol propellants and in the manufacture of foam products will be banned, and by 1995, this will extend to banning the use of CFCs in, amongst other things, air-conditioning systems. The use of R22 as a refrigerant will be permitted until 2000.

Germany's two CFC producers, Hoechst and Kali Chemie, presented the German Minister of Environment with a written statement in which they committed themselves to ceasing production by 1995. The German Chemical Society issued a similar statement. The German minister said that "including R22 in the ban sends a worldwide signal that, for the German Government, the substitution of partially halogenated CFCs is not a long-term solution to the ozone problem".

*(Source: New Scientist, June 9, 1990.)*

### China Speeds up Action on CFC Problem

Sponsored by the United Engineering Societies of the Chinese Science and Technology Association and co-sponsored by other Chinese societies, a symposium on, Technical Countermeasures regarding Restriction and Phase Out of CFCs, was held in Qingdao, in the province of Shandong, from April 24-29, 1990. In addition to 166 Chinese specialists and 2 government officers, 17 delegates from 7 foreign institutes (US, UK, Federal Republic of Germany and Japan) were invited. The most important results were:

1. Further promotion of the urgency to control and prohibit CFC substances;
2. Improved recognition of the need for protection of the environment for human existance;
3. The suggestion of a policy for implementation of countermeasures;
4. Formation of an information network in China.

*(Source: Guangzhou Institute of Energy Conversion.)*

### Dupont Starts Pilot Production of Refrigerant R124

Dupont recently announced that it is now operating a pilot plant to produce R124, being a constituent in the ternary mixture R124/R22/R152a, to replace R12 and R500. The plant is located at its Chambers Worksite in New Jersey, U.S.A. R124 is now undergoing toxicity testing. Limited quantities of R124 and the R124/R22/R152a blend will be available for performance tests and development purposes.

*(Source: Dupont Information Sheets, 28 February 1990.)*

### Danfoss Announces Pilot Production of R134a Units

Replacement of R12 by the new "ozone-friendly" refrigerant R134a will only be possible if, not only the refrigerant, but also the necessary equipment, is available.

Danfoss recently announced that it will be commencing production of R134a units: pilot production starts this autumn (1990), accelerating to full-scale production of six sizes of domestic refrigerators in March 1991. By the end of 1993, Danfoss aims to produce a full range of R134a hermetic units. Danfoss has succeeded in optimising R134a systems to such an extent that their units are expected to provide unchanged performance, noise and reliability levels, compared to R12 systems.
*(Source: Refrigeration and air-conditioning, May 1990.)*

## Environment

### Dutch Utilities' Environmental Action Plan

The Dutch electricity distribution, gas distribution and district heating utilities have jointly presented their first environmental action plan to reduce $CO_2$ emissions. The aim is an annual investment of Dfl 1350 million (approx. US$ 700 million) in energy saving techniques in households, industry and the energy distribution sectors. Measures, such as insulation of houses, improvement in lighting and heating efficiency, combined heat and power and wind turbines should lead to energy savings of 13% by the year 2000. An increase in government grants for energy saving is advocated or alternatively an increase of 1.25-1.5% of the electricity and gas tariffs to finance the action plan.
*(Source: Energiespectrum, May, 1990.)*

# IEA

## Technical Annexes of Implementing Agreement on Advanced Heat Pumps

- **Annex IX on High Temperature Industrial Heat Pumps has been completed.**
  The annex started in 1987 and had as participating countries: Belgium, Finland, Germany, United States, Netherlands, Switzerland, Japan and Sweden.

  In the final report the state-of-the-art for high temperature industrial heat pumps is described. Overviews are given of the different types of heat pumps on the market including, where possible, listings of manufacturers. A detailed description is also given of the R&D efforts going on in the field.

  The overviews show that open cycle compression heat pumps are able to reach temperature levels of up to 200°C. The closed cycle compression systems in actual use are limited to temperatures of about 130°C. For sorption machines in actual use only the heat transformer is able to reach significantly higher temperatures than 100°C. However, the temperature limit is presently around 135°C. As for the economics, under the present conditions of fuel oil costs high temperature heat pumps can be applied in only a few countries where the cost of electricity to drive the heat pumps is extremely low (e.g. Norway). The run-time of heat pumps must be as high as possible and must bein the order of at least 6,000 hours per annum in order to ensure acceptable payback periods.

  The report concludes with detailed case studies in which the energetic and economic performance of existing heat pump systems are analysed. In general, it is found that the systems perform according to expectations except for the economic aspects. Several systems built a few years ago are deemed uneconomical under the present cost conditions.

- **Annex XVIII concerning Thermophysical properties of the environmentally acceptable refrigerants has received official status.**
  The operating agent is the USA. The other participating countries are presently Sweden and Canada. Other countries which have shown interest have not as yet definitely decided to participate. For more information, please contact:
  National Institute for Standards and Technology,
  Mr Mark McLinden, Tel: +1-303-497-3580.

- **Belgium has proposed a new Annex concerning the different aspects of the safe use of working fluids (including $NH_3$) in heat pumps and refrigeration equipment, both of the compression and of the absorption type.**
  The aspects to be investigated are mainly:

  - flammability and effect of fire;
  - explosiveness and effect of explosions;
  - toxicity and asphyxiation.

  The work programme proposed consists of:

  I. Evaluation of international experience and national regulations;
  II. Developments of techniques to evaluate risks involved.

The total project time will be 2 years and a detailed proposal is under development.

## IEA/OECD Project on Global Climate Change: The Energy Dimension

The IEA and OECD are carrying out a study on the energy-related aspects of greenhouse gas emissions. The study will provide an overview of the greenhouse gas emissions arising from human activities, a global energy outlook up the year 2005, an analysis of greenhouse gas emission factors and emission projections up to the year 2005, energy technology options for limiting emissions of greenhouse gases, and finally, short term policy instruments to implement greenhouse gas limitation strategies in the energy sector. This report will form the basis for future IEA work in this area, and will be provided as a contribution to the work of the UN Inter-governmental Panel on Climate Change (IPCC). The chapter on energy technology options will contain an assessment of energy technologies that are currently available or that might become available in the long term. An extended report based on this chapter (amongst others containing R&D priorities) is due for completion towards the end of 1990. For further information contact: Dr. Denis Kearney,, IEA Secretariat 2 rue Andre Pascal, 75775 Paris Cedex 16, France
Tel: +33-1-4524-9958
*(Source: IEA Energy Technology Newsletter.)*

# Market News

## Japanese Market Figures for Room and Packaged Air-Conditioners

The Japan Refrigeration & Air-Conditioning Industry Association (JRAIA), has recently estimated the world demand and supply of room and packaged A/Cs for residences, buildings etc. that existed in 1988. Heat pump types are included, but the figures are exclusive of chiller-heater systems. Within the category room A/Cs, the window type, and the Japanese small household split type were included. All A/Cs other than room ACs were contained in the category of packaged A/Cs. The world demand for A/Cs in 1988 is estimated at 14,696,000 units for room A/Cs and 5,618,000 units for packaged A/Cs. Japan's share of world demand was 32% for room A/Cs and 13% for packaged A/Cs. Japan's share of world production was 40% for room and 16% for packaged A/Cs.

An independent survey of exports indicated that Japan satisfies 13% of overseas demand for room A/Cs and 3.3% of that for packaged A/Cs. According to the figures presented by JRAIA the share of the room A/C world demand and world production was 4.9% and 3.5% respectively for Europe and 30.3% and 23.9% respectively for North America. For packaged A/Cs world demand and world production figures were 2.9% and 2.1% respectively for Europe and 76.2% and 77.9% respectively for North America.

JRAIA also disclosed the Japanese shipment figures for the 1st quarter of 1990. Figures within brackets show the growth (+) or decline (-) with respect to 1989. For cooling-only room A/Cs and for heat pump room A/Cs the domestic shipment figures were 308,561 units (+ 44.1%) and 909,048 units (+ 19.5%) respectively and for cooling-only packaged A/Cs and heat pump packaged A/Cs 26,651 units (+ 5.3%) and 145,850 units (+ 18.1%) respectively. The numbers of exported units were for room A/Cs 445,565 (- 4.1%) and for commercial A/Cs 49,491 (+ 23.8%).
*(Source: Japan Air Conditioning, Heating & Refrigeration News, May 25, 1990.)*

### New Technologies at Brown Boveri-York

Excerpt from a report on BBC York Press Conference on 23 February 1990.

According to BBC York, $NH_3$ installations are being increasingly used. A $NH_3$ water chiller for the application in A/C installations in buildings will have completed the test room phase by summer 1990 to be tested afterwards together with the German Refrigeration Society (DKV). It contains only 28 kg $NH_3$ for a cooling capacity of 350 kW. BBC York is now able to replace R11 by R123 - which at the moment can only be procured from the US - for turbo-compressor water chillers of large capacity. In the range between turbo-compressor and reciprocating-compressor water chillers, new screw-compressor installations have been developed using R22 as the refrigerant. BBC York contemplates using Propane as a refrigerant for industrial chilling applications in particular, in the chemical industry.

In spite, of the said high investment costs and high energy consumption, the demand for absorption cooling installations has risen steeply due to an environmental consciousness, a trend set by the chemical industry. In the commercial sector R134a will be used shortly as a replacement for R12 and instead of R502, two-stage R22 installations will be utilised.
*(Source: Klima-Kälte-Heizung, Vol. 5, 1990.)*

## News - Member Countries

### Emperor Akihito Visits DHC Plant in Tokyo

On 20 June, 1990, the Japanese Emperor Akihito visited the Hakozaki district heating and cooling plant, run by the Tokyo Electric Power Company (TEPCO).

It is a heat pump based DHC (District Heating and Cooling) plant which uses the thermal energy of the Sumidagawa River and is the first of its kind in Japan. The plant began heat service on 1 April 1989. The heat is utilised for heating and cooling of the peripheral office buildings and high rise apartments. Its service capacity of the heat generator is 27.7 MW for heating and 39.3 MW for cooling.
*(Source: Heat Pump Technology Centre of Japan.)*

# Conferences

## * Call for papers

**IKK 90 International Trade Fair for Refrigerating and Air-Conditioning**
*October 4-6, 1990/Nurnberg (Fed. Rep. of Germany)*
Contact: Nurnberg Messe GmbH, Messezentrum, D-8500, Nurnberg 50. Germany.
Tel.: +49-911-86060, Fax : +49-911-8606-228.

**HVAC-days, Trade Fair for Heating, Ventilation and Air-Conditioning**
*October 24-27, 1990/Oslo (Norway).*
Tel.: +47-2 601-390.

**1990 International Conference on CFC and Halon Alternatives**
*November 27-29, 1990/Baltimore, Maryland (USA)*
Contact: Galany Productions Inc., P.O. Box 868, Frederick, MD 21701, USA.
Tel.: +1-301-662-9400.

**ASHRAE 1991 Winter Meeting**
*January 20-23, 1991/New York (USA)*
Contact: ASHRAE Meetings Section, 1791 Tullie Circle NE, Atlanta GA 30329, USA.
Tel.: +1-404-636-8400, Fax : +1-404-321-5478.

**International Conference on Conventional & Nuclear District Heating**
*March 18-22, 1991/Lausanne (Switzerland)*
The conference includes a session on integration of heat pumps in district heating.
Contact: Secretariat International Conference on Conventional & Nuclear District Heating, EPFL-LASEN, CH 1015 Lausanne, Switzerland.
Tel.: +41-216-93-2495 or 2516,
Fax : +41-216-93-2863.

**1991 Australian Refrigeration, Air-Conditioning and Heating (AIRAH) Conference and Exhibition.**
*April 21-14, 1991/Melbourne (Australia)*
Contact: AIRAH 1991 Annual Conference, C/manager, AIRAH Federal Office, 191 Royal Parade, Parkville, Vic 3052, Australia.

*** 1991 Energy and Environment International Symposium.**
*August 25-28, 1991/Espoo (Finland)*
Abstracts to be sent to: Prof. Ilmari Kurki-Suonio, Centre of Energy Technology, Helsinki University of Technology, Otakaari 4, 02150 Espoo, Finland.
Tel.: +358 0-4513-580.

# Bibliography

## Refrigerants/CFCs

### Refrigeration & CFCs, IIR Int. Colloquium of Brussels, March 19-20, 1990

*(English, French).*

11 papers were presented concerning: education and training; recovery and recycling of halogenated refrigerants; substitutes for CFCs (compression, absorption, solid sorption cycle); the role of R22 in refrigeration and A/C equipment; development of toxicology profile of a chemical compound; environmental impact of CFCs; socio-economic impact of the use of substitutes in developing and industrialised countries.

### New Data on Ozone-Safe Refrig. R134a

1. **Comparison of Refrigerants HFC 134a & CFC 12.** *B. Petersson, H. Thorsell, Int. Journal of Refrigeration, Vol. 13, May 1990, pp. 176-180 (English).*
2. **Pipe-sizing & Pressure Drop Calculations for HFC 134a.** *T. Atwood, ASHRAE Journal, April 1990, pp. 62-66 (English).*
3. **Influence of the FCFC Issue on the Capabilities, Use & Construction of Liquid Refrigerant Heat Exchangers.** *M. Hage, Klima-Kälte-Heizung, 1990, No. 4, pp. 156-160 (German).*

Petersson and Thorsell report on a theoretical and experimental comparison of R134a and R12 for open piston compressors. The results indicate that the tested compressor will give greater refrigerating capacity and a lower COP with R134a than with R12, the latter possibly due to the use of polyalkene-glycol lubricant.

Thermodynamic data (Wilson and Basu) and viscosity data (Shankland et al.) are used by Atwood to analyse pipe-sizing and pressure drop for R134a piping. Data are presented on mass flow rates, vapour pressure drop, vapour velocities, liquid pressure drop and liquid velocities. For suction lines it can be concluded that R134a pipe-sizing will be similar to that of R12 pipes. For discharge piping and liquid lines, the pressure drop for R134a will be substantially reduced: 25 to 30% in equivalent pipe sizes. This gives rise to opportunities for size reduction.

The paper by Hage relates to the consequences of the change from current CFC refrigerants to R22 and R134a for refrigerant liquid heat exchangers. Attention is given to shell-and-tube condensers, flooded and dry evaporators.

### Role of Refrigerant Mixtures as Alternatives to CFC's

*D.A. Didion, Nat. Institute of Standards, D.B. Bivens, Du Pont De Nemours, Int. Journal of Refrigeration, Vol. 13, May 1990, pp. 163-175 (English).* The near azeotropic blend R22/R152a/R124 - a candidate replacement of R12 in existing installation - is discussed. In particular, preferential leakage of one of the components is dealt with.

As to zeotropic (or non-azeotropic) mixtures - which require new system designs -the paper gives a general discussion on such topics as: efficiency increase by minimizing log mean temperature differences ("Lorenz cycle"), heat exchanger design, non linearities in saturation temperature profiles, inter-cycle heat exchange and desorber-absorber cycles.

## R&D

### Experimental Investigation of Performance Control with Non-Azeotropic Mixtures

*K.-D. Gerdsmeyer, H. Kruse Klima-Kälte-Heizung, Vol. 6, 1990 (German).*

Frequent consideration has been given in the past to the possibility for a low-loss and continuous control of the performance of cooling plants with non-azeotropic refrigerant mixtures. However, it has been virtually impossible to find evidence of rigorous investigation into the actual suitability of this technique. In this article the focuss is on experimental tests of two different methods for altering concentrations, particularly in heat pump applications: partial condensation and evaporator overflow. The attainable control ranges are particularly interesting together with the energetic behaviour.

### An Intensified Absorption Heat Pump

*C. Ramshaw, Imperials Chemical Industries, T.L. Winnington, Caradon Mira Ltd., Proceedings of the Institute of Refrigeration, 1988-89, pp. 2-1 to 2-6 (English).*

This article describes the R&D of an intensified absorption heat pump. This innovative hermetic closed rotating heat pump applies the centrifugal field to intensify the heat and mass transfer processes in the unit. The heat pump, uses water as a low pressure refrigerant in combination with new absorbents related to this non-conventional design. The heat pump is gas-fired and the fields of application are heating of homes and air-conditioning.

## IEA Publications

### Proceedings of the Workshop on High Performance Heat Pumps, Wider Application and Markets

*Held in Susono City Japan, March 1990, IEA-HPC - WR6 (English).* In this HPC workshop, 15 papers were presented, which were divided in 4 main categories (English):

- **High Performance Compression Heat Pumps**:
  4 papers concerning the Japanese Super Heat Pump Programme, i.e 2 super high performance heat pumps, 1 for heating only and 1 for heating and cooling, and 2 high temperature heat pumps, 1 for generating 150°C, the other for 300°C output heat.
- **Absorption Heat Pumps**:
  3 papers dealing with: a Dutch absorption heat pump for commercial space heating; a Japanese absorption compression heat pump for temperature use up to 200°C and a US triple-effect absorption chiller.
- **Components**:
  2 papers concerning new heat exchanger development in Japan, 1 for non-azeotropic mixtures, the other for condenser, evaporator and chemical storage applications; and 1 paper comparing a centralised to a distributed DHC system in Norway.
- **Applications**:
  5 papers on the following topics: French work on exergy and economic optimisation of sorption heat pumps and heat transformers; experience with Swedish district heating systems; possible applications of Japanese super heat pump and chemical storage systems in industry; residential heat pump requirements in Austria; and the activities of the Norwegian Government to promote the use of heat pumps.

Furthermore, 3 working groups discussed questions related to high performance heat pumps. HPC member countries can obtain the proceedings for Dfl 60.-.

### *New Technologies for Heating & Cooling Supply in Office Buildings, CADDET Analyses Series No.3.*

*E. Abel et al CASU/CADDET, Sweden, June 1990. (English).*

The purpose of the report is to support those who have to make initial decisions about heating and cooling supply systems in buildings and those who have to prepare the background material needed for the decisions. Information is presented about 26 commercial buildings with advanced supply systems in 7 CADDET member countries. Comprehensive data on energy use was available for 7 of the buildings. The demonstration systems can be divided into 3 main groups:
- Advanced heating systems, including various types of heat pumps and solar heating systems;
- Interactive heating and cooling systems, e.g. systems including seasonal storage and using a water reservoir or aquifer;
- Advanced cooling systems such as absorption cooling.

A comparative analysis of different energy systems with respect to electricity and fossil fuel requirements, operation costs, additional investment, and $CO_2$ emissions has been made.

The report includes cost nomograms for various systems. An analysis method that facilitates initial planning is introduced. The report can be obtained at CADDET, P.O. Box 17, 6130 Sittard, the Netherlands at a price of US$ 25.- for IEA-CADDET member countries, US$ 100.- for other IEA countries, and US$ 200.- for non IEA countries.

## *New ASHRAE & IIR Publications*

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   Price: US$ 46.00; ASHRAE-members US$ 31.00.

3. **1990 Winter Meeting Atlanta, Georgia, ASHRAE Transactions 1990, Vol. 96, Part 1, code 82901.**
   Contains 194 papers.
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### *International Institute of Refrigeration*

**Sample, compiled from the 1990 publications catalogue.**

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2. **Status of CFCs - Refrigeration Systems and Refrigerant Properties (ISBN 2903633428), Purdue Meeting, 1988, 50 papers (English), 437 p., IIR code 88/2.**
   Example of papers: Thermodynamic Properties of Refrigerants and their Non-Azeotropic Mixtures; Refrigeration Systems (Compressors, Expansion Valves, Heat Exchangers); CFCs and the Environment; Impact of the Limitation of CFC Emissions on the Field of Refrigeration and on Equipment and Refrigeration Systems; new Refrigerants; Properties of 134a; Ways to reduce the Emission of R12 into the Atmosphere.
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