IEA International Energy Agency

Common Study on Advanced Heat Pump Systems

KFA Kernforschungsanlage Jülich GmbH

Verlag TÜV Rheinland

International Energy Agency
Programme of Research and Development on Advanced Heat Pump Systems

International Energy Agency
Programme of Research and Development on Advanced Heat Pump Systems

Annex I:
Common Study on Advanced Heat Pump Systems

Final Report
Volume I: Results and Conclusions

July 1980

Prepared by
D. L. Hodgett
G. Oelert (Project Leader)
R. Rudolph

M. Grapin
K. Mead
S. Pariente
A. Quinn
C. Rice

Battelle
Battelle-Institut e.V. Frankfurt

RPA
28 Avenue De Messine
75008 Paris, France

ISBN 3-88585-029-X
© Kernforschungsanlage Jülich GmbH as operating agent for the
contract parties of the IEA Implementing Agreements
„Programme of Research and Development on Advanced Heat Pump Systems"
Responsible: Project Management Energy Research (PLE)

© Kernforschungsanlage Jülich GmbH als Operating Agent für
die Vertragspartner des IEA Implementing Agreements
„Programme of Research and Development on Advanced Heat Pump Systems"
Verantwortlich: Projektleitung Energieforschung (PLE)

Gesamtherstellung: Verlag TÜV Rheinland GmbH, Köln
Printed in Germany 1981

Executive Summary

The results and conclusions of the IEA common study of Advanced Heat Pump systems are contained in this Volume I of the final report. Volumes II and III which are restricted to the participating countries, contain respectively the Technology Survey and Market Study results and conclusions. The Common Study was carried out for thirteen participating countries: Austria, Belgium, Canada, Denmark, Federal Republic of Germany, Italy, Japan, Netherlands, Spain, Sweden, Switzerland, United Kingdom and United States.

The objective of the Study was to collect and evaluate data on advanced heat pumps, i.e. heat activated systems and other systems currently under development, to assess their market potential and to derive relevant proposals for desirable developments and a future IEA R&D programme. Volume I describes the organisation of the Project and the links between the Technology Survey and Market Study, the technical assessment of individual heat pump types carried out under the Technology Survey and the market estimates for each over the period 1980 to 2000. The conclusions and R&D needs from both the technical and marketing standpoints are included in this summary volume. Volume II of the study contains a review of the state of the art of heat pumps and an assessment of the available energy supplies and heat sources in each country, a complete analysis of the technical characteristics and R&D status of each advanced heat pump type particularly in regard to the possible applications in space heating and conditioning, district heating and in industry. The technical R&D needs and recommendations for further IEA projects in this field are also included in Volume II. Volume III contains a description of the technical approach to the market study, estimations of the total market potential for each advanced heat pump system, the market potential for individual heat pump systems in the participating countries until the year 2000 and conclusions and recommendations for further R&D based on the results of the

market study.

The general conclusions of the study were as follows.

1) In the 13 participating countries there is a large and extremely diverse R&D programme covering most of the possible advanced heat pump systems. The present situation is that following the stimulus of the first energy (i.e. oil) crisis of 1973 is that heat pump technology is emerging from many years of neglect.
2) The background situation in the participating countries is a continuing high degree on oil, mainly light oil, for domestic and commercial heating, particularly in Belgium, Canada, Denmark, Germany, Italy and Switzerland.
3) Electric heat pumps (for both heating and cooling) hold a predominate position in the U.S.A. as replacement for boilers and furnaces. In W. Germany a similar situation is developing with bivalent-alternative heating only heat pumps.
4) The first priority in most countries is to substitute for oil and a second priority is to conserve all energy forms. Present electric heat pump types meet the first priority but rarely contribute anything to energy conservation, unless compared to direct electric heating. Advanced heat pump types are capable of conserving between 30 % and 50 % prime energy, and the best systems from this aspect are internal combustion engine driven. However, from the point of view of oil substition the various gas fired systems are superior i.e. the Brayton, Otto and Stirling engine driven, the Rankine-Rankine and the sorption systems. The external combustion systems have a total flexibility of fuel use and are therefore particularly attractive, especially in large unit sizes.
5) Most advanced heat pumps are not due to commercialisation until around 1985. Since they cannot be expected to make a significant contribution rapidly, the possibility of achieving worthwhile energy savings before 1990 appears low. Thus the

accelerated development of the systems with greatest potential for each application is desirable, even if in the short term their use would not be justifiable solely on economic grounds.

6) The economic potential for all the advanced heat pump systems, although significant, is limited by strong competition from conventional systems, especially natural gas boilers and conventional electric heat pumps. Strong competition among the advanced heat pumps could limit the market for each of them separately.

   The overall conclusion of the study for R&D policy in participating countries is relatively clear. To expand the market share for advanced heat pumps as a group, R&D efforts should be focused on advanced heat pumps most likely to generate market potential by 2000, such as the absorption, Rankine-Rankine, and Diesel-Rankine systems.

   The Diesel-Rankine is particularly suited to large installations, while the absorption and Rankine-Rankine show greatest potential for small heating only applications. The electric heat pump will predominate for combined heating and air conditioning applications.

7) From the Market Study it was not possible to estimate the total energy saving potential by the year 2000. However, it was estimated by the Executive Committee from the data collected in the study that the prime energy saving potential lies in the region of 5 %, depending on the successful competition of R&D and commercialisation of the systems.

8) The capital costs of the advanced systems in general are still high compared with the conventional boiler or furnace costs. The reduction of first cost, a major factor influencing choice, should be a specific R&D priority. Good design and mass-production should also greatly reduce costs.

9) The reliability, life-time, service and maintenance of advanced heat pumps will in most cases prove initially troublesome. Suitable service and maintenance facilities and guaran-

tees during the acceptance phase are essential factors in their acceptability. Pilot installations and close monitoring of earlier installations will assist in the controlled and orderly growth of the market and the necessary infrastructure.

10) There exists a diversity of financial, fiscal and other incentives to the development and use of energy conserving technologies, including heat pumps, in the participating countries. The effectiveness of these policies and possible alternatives in promoting the objectives of substitution for oil and conserving energy need investigation.

11) There are no international heat pump standards. There is a pressing need for uniform standards for performance and for definitions of terms and symbols. A further requirement is a code of practice for advertising and promotion to prevent the market for heat pumps being spoilt by misleading claims or unscrupulous promoters.

12) There is still insufficient activity to develop heat pump systems for industrial applications where large energy savings are possible through the use of this technology.

Contents

	page
Executive Summary	
1. General Introduction	1-1
1.1. Project Organisation	1-1
1.2. Objectives of the Study	1-1
1.3. Program of Work	1-3
2. Linking of Technology Survey and Market Study	2-1
2.1. Background	
2.2. Case Study Format	
3. Technology Survey	3-1
3.1. Introduction	3-1
3.2. Performance	3-2
3.2.1. Definition of Terms and Performance Criteria	3-2
3.2.2. Design Point Performance	3-4
3.2.3. Below Design Point Performance	3-8
3.2.4. Auxiliaries	3-12
3.2.5. Above Design Point Performance	3-13
3.2.6. Seasonal Performance and Energy Saving	3-16
3.2.7. Cooling Performance	3-17
3.3. Design	3-21
3.3.1. Size and Weight	3-21
3.3.2. Energy Demand	3-21
3.3.3. Safety	3-25
3.4. Operation	3-28
3.4.1. Control	3-28
3.4.2. Reliability	3-28
3.4.3. System Life	3-34
3.4.4. Service and Maintenance	3-35
3.4.5. Environmental Effects	3-37
3.5. Applications	3-40
3.5.1. Domestic	3-40
3.5.2. Commercial	3-44
3.5.3. District Heating	3-45
3.5.4. Industrial	3-51

page

4. Market Study 4-1
   4.1. Technical Approach 4-1
   4.2. Market Analyses of Advanced Heat Pump Systems 4-7
   4.3. Market Estimates by Advanced Heat Pump Systems 4-15
      4.3.1. Systems with Greater Market Potential 4-15
      4.3.2. Systems with Limited Market Potential 4-44

5. Conclusions 5-1
   5.1. General 5-1
   5.2. R+D Needs 5-4
      5.2.1. Technical R+D Needs 5-4
      5.2.2. Market Relevant R+D Needs 5-10

6. References 6-1

7. Acknowledgements and National Teams 7-1
   7.1. Acknowledgements 7-1
   7.2. National Teams 7-1

1. General Introduction

The Common Study of Advanced Heat Pump Systems was initiated in October 1978 by the Committee on Energy Research and Development (CRD) of the International Energy Agency (IEA), Paris, as Annex 1 of an "Implementing Agreement for a Programme of Research and Development on Advanced Heat Pump Systems". The participating countries (13) were Austria, Belgium, Canada, Denmark, Federal Republic of Germany, Italy, Japan, Netherlands, Spain, Sweden, Switzerland, United Kingdom and the United States of America.

1.1. Project Organisation

KFA Jülich, as representative of the Federal Republic of Germany, was appointed to act as Operating Agent with Battelle-Institut e.V. (BIeV) Frankfurt as its assistant. BIeV was thus responsible for performing the study coordination, part of the state-of-the-art survey and the general project management on behalf of KFA Jülich. The role of the national teams (see Fig. 1-1) in the study is to collect data on the state-of-the-art and on advanced heat pump systems, and later to assess the market potential for chosen systems in their respective countries.

The market study part of the programme was assigned to the USA, which nominated Resource Planning Associates Inc. (RPA) as its assistant, responsible for the definition, organisation and coordination of the market study. The allocation of responsibilities is shown in Figure 1-1.

1.2. Objectives of the Study

The objective of the study is to collect and collate data on advanced heat pumps obtained internationally, to produce an evaluation of them, to assess which developments are desirable and thereafter to propose an international R+D programme. The study would

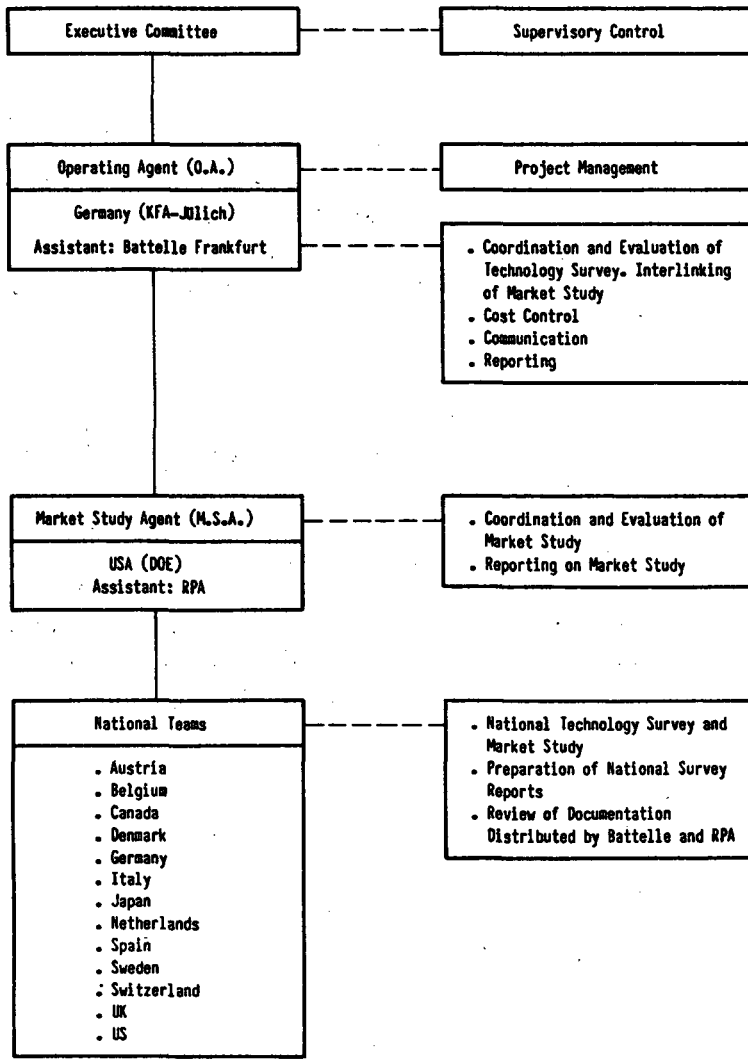


Fig. 1-1: Project organisation

encompass a critical analysis of existing heat pump systems in countries with major developments and also reveal whether experience gained with these and other technologies can be utilised for advanced heat pumps.

The international study, in which the experience and developments in many countries are collected and evaluated, is therefore intended to lead to

- better utilisation of existing experience,
- avoidance of unnecessary or undesirable developments,
- identification of trend setting developments,
- early identification of necessary research activities,
- establishment of internationally coordinated R+D programs.

1.3. Program of Work

Figure 1-2 gives the simplified project flowchart for the structure and coordination of the technology survey and market study.

The project consisted of the initial phase of the identification of the basic types of heat pumps and their classification, the market analysis and the determination of the state of heat pump development in each country. Thereafter the basic types of heat pumps of interest to the participating countries were identified and detailed technical and cost profiles prepared on each system of interest. This data was then used in the market study to estimate the potential market in each country for the chosen systems to year 2000, comparing them to the conventional competing systems prevailing in that country. Meanwhile the technical characteristics of the systems and development trends were evaluated in order to produce a ranking of their strengths and weaknesses and thereafter to propose an R+D program to augment their development. The results of the market study i. e. the estimated market for each system based on the technical and cost profiles produced earlier in the project, was then used to give weighting to the proposed R+D programme so

that development of the systems with the largest potential market would be accelerated.

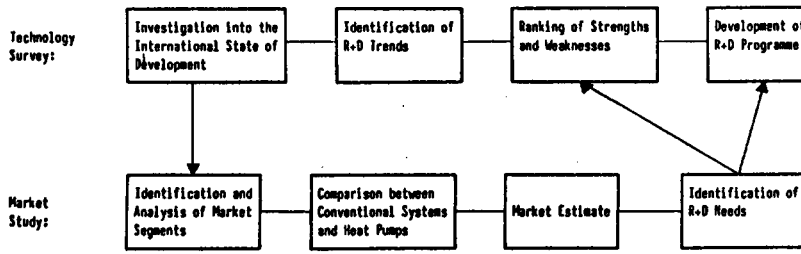


Fig. 1-2 : Overall study description

2. Linking of Technology Survey and Market Study

2.1. Background

In order to establish a link between the technology survey and the market study it was necessary to provide technical and cost data on heat pump systems which were considered by the participants to have possible applications. Initially a list of interesting profiles was made which included such aspects as operating principles (i.e. absorption, engine driven, etc.), thermal output, heat source, distribution medium and fuel. The national teams identified which systems were under development or were of interest to their country, and a questionnaire was prepared to obtain all the necessary technical and cost data. The completed, or more often semi-completed questionnaires (since most systems were in early stages of development), were then used to produce the case studies. Where data was missing the system developers were visited to obtain it, and as a last resort the data was estimated. The case studies were then distributed to the national teams for use as input data to the market study, while the questionnaires were used as material to assist the technology study.

Table 2-1 is a list of the case studies and their major characteristics. The majority of the case studies (8) were for systems being developed for single family house heating, i.e. with thermal outputs in the range 10-20 kW; these include two absorption systems, two external combustion engine driven systems, and two internal combustion engine driven systems, one electrically driven system, and one solid absorption system. Of the remainder, four were very large systems ($\geqslant$ 2 MW) for district heating, three being absorption and one Diesel engine driven. A large group (five) was for commercial building heating (and sometimes cooling) or multifamily house heating; three systems from this group were external combustion engine driven and two were internal combustion engine dri-

Heat pump profile No.	Heat pump cycle		Drive		Heat source	Thermal output kW	Case study?
	type	fluid(s)	type	fuel			
1	absorption	NH_3/H_2O	direct	oil, gas	ground	3-4	no
2	absorption	organic	direct	oil, gas	amb. air	10-12	yes
3	absorption	NH_3/H_2O	steam boiler	oil, gas	waste water	> 2000	yes
4	Rankine	R12	Diesel	oil	amb. air	15-20	yes
4a	Rankine	R22	Diesel	oil	soil	400	yes
5	Rankine	R12	Diesel	oil	amb. air	400	yes
6	Rankine	R114	Diesel	oil	sewage	> 2000	yes
7	Rankine	R12; R22	Otto	gas	amb. air	10-400	yes
8	Rankine	R12; R22	Otto	gas	air/ground/ surface water	400	yes
9	Rankine	R22	Stirling	oil, gas	air/ground surface water	10-12	yes
10	absorption	H_2O/Na_2S	solar		solar	15-20	yes
11	absorption	$H_2O/LiBr$	solar/gas		room air	3-4	no
12	Rankine	freon	Rankine	solar	room air	3-4	no
13	Rankine	R22	motor	electricity	ground	10-12	yes
14	absorption	NH_3/H_2O	direct	gas	amb. air	10-12	yes
15	Rankine	various	Rankine	oil, gas	amb. air	10-400	yes
16	Rankine	R12	Brayton	oil, gas	amb. air	50	yes
17	Rankine	steam/R11	Rankine	oil, gas	amb. air	400	yes
18	Rankine	various	Rankine	oil, gas	sea water	400	yes
19	absorption	$H_2O/LiBr$	direct	kerosene	waste water	> 2000	yes
20	abs./Ran.	$H_2O/LiBr$// R114	motor	electricity	waste water	> 2000	yes

Table 2-1: Case Studies

ven. The three systems not yet accounted for were very small (3-4 kW thermal output), and since none of the participants showed any interest in these or were able to produce case study material for them, they were dropped from the market study at an early stage, which meant that no case studies were produced.

For those systems for which insufficient data were available to produce complete case studies, but which were of interest to the participants, the necessary data were estimated by Battelle. In several cases revised data were then obtained by the national teams from the developers. When these were received in time they were used, but in some cases this was not possible and the Battelle data had to be used. In one case the developer did not agree with the Battelle data, since they considered them too pessimistic. However, since the developer in question was unwilling to provide its own estimates, the Battelle estimates had to be used.

2.2. Case Study Format

The format chosen for the case studies was one that would provide all the necessary information on each in such a way that the operating principles and performance could be easily understood, and the data on cost and performance readily utilised by each national team. The case studies therefore consisted of a data sheet giving a short description of the system's main characteristics, i.e. type, fuel, application, source and sink (i.e. distribution medium) type and temperatures, the performance at the design point (usually $0^{o}C$) and $+10^{o}C$ and $-10^{o}C$ ambient air temperature, the type and operation of the back-up, the year of introduction and the unit cost, installed cost and finally estimated maintanance cost expressed as a percentage of installed cost. Attached to each data sheet are a line diagram summarising the main characteristics of the system and an energy flow diagram for the design point and performance graphs. These graphs are of the coefficient of performance (COP_{eff} and COP'_{eff}) and the dimensionless thermal output (i.e. a given thermal

output divided by the design output = N_{ht}/N_d) against ambient air temperature. The normalisation was performed in order to compare systems of different thermal outputs.

3. Technology Survey

3.1. Introduction

The objective of the technology survey was to identify advanced heat pump systems and, by consideration of their strengths and weaknesses, to indicate which systems would be best suited to each possible application and what further research and development was necessary to lead to their commercial success. The assessment of strengths and weaknesses therefore utilised the information gathered for the case studies and for the R+D trends, in addition to discussions with experts. The analysis of this information was carried out under four critical aspects: the projected performance (under design conditions and in the proposed applications), the design (with regard to size, weight, complexity, requirements for back-up, controls, etc.), operational aspects (reliability, service and maintenance requirements, service life, noise, vibration, pollution, etc.), and finally initial cost. The latter point lay more in the area of the market study, but to ignore relative capital cost, and the possibilities of reducing it, in the technology survey would have been unrealistic.

A general assessment of strengths and weaknesses was therefore first made on the basis of these four criteria, this being followed by an assessment of each system for four major areas of application. These are single-family house heating (and to a lesser extent cooling), i.e. thermal outputs of 10 - 20 kW; multi-family house heating and cooling, i.e. thermal outputs in the range 100 to 400 kW; district heating with thermal outputs in the MW range; and finally industrial applications covering a wide temperature and size range.

3.2. Performance

3.2.1. Definitions of Terms and Performance Criteria

The major performance criteria for each system are the Coefficient of Performance (COP) and Seasonal Performance Factor (SPF). There are many possible definitions of these, depending on what the system boundaries are considered to be. Thus there is the performance of the heat pump subsystem, that including the drive subsystem, and that of the system including the electricity and fuel producing subsystems. The latter definitions are at present only significant when the energy conservation potential is assessed for electrically driven systems or those with a substantial auxiliary electric power requirement on countries with thermally generated electricity. However, future conditions with synthetic natural gas (SNG) as a possible competitor to coal or nuclear based electricity makes the definition of the system including energy production more relevant.

No internationally agreed definitions of COP and SPF exist and there are only two national standards which are relevant /25, 26/. Since these do not agree, simplified definitions of COP and SPF have been used, which are based on definitions suggested by Professor Gilli, a member of the Austrian National Team.

The following definitions of the Coefficient of Performance (COP) and Seasonal Performance Factor (SPF) were used.

For the heat pump subsystem only:

$$COP_g = N_g/N_m$$

where

N_g is the rate of heat output from the condenser (kW)

N_m is the electrical or mechanical power or rate of heat input to the heat pump, including auxiliaries (kW).

For the heat pump system, including drive subsystem:

$$COP_{eff} = N_{ht}/N_p$$

$$SPF_{eff} = Q_{ht}/Q_p$$

where

N_{ht} is the total rate of heat output (kW)

Q_{ht} is the total heat output per season (kJ)

N_p is the sum of

N_{hp}, the power or heating rate to the heat pump and

N_{aux}, the power input to the auxiliaries (kW)

Q_p is the total energy input per season to drive subsystem and auxiliaries (kJ).

For the heat pump system, including electricity generating efficiency:

$$COP'_{eff} = N_{ht}/N_{p'}$$

$$SPF'_{eff} = Q_{ht}/Q_{p'}$$

where

$$N_{p'} = N_{ht} + (N_{aux}/\eta_{el})$$

$$Q_{p'} = Q_{ht} + (Q_{aux}/\eta_{el})$$

η_{el} is the efficiency of production of electricity.

3.2.2. Design Point Performance

The only true assessment of performance is that in use i.e. the performance obtained over a working season or year, often referred to as the "seasonal performance factor" (SPF). However, for systems which have not yet been operated over a complete heating season, or worse not yet had their performance measured at the design point, it is necessary to estimate the SPF from measured or predicted design point data with estimation of the off-design point performance, making allowances for such factors such as auxiliary power consumption for pumps, fans, etc. and for defrosting if necessary. Thus, in order to assess performance it was necessary to estimate the SPF for each application by determining the design point COP_g, and heating capacity and the off-design point COP_g and N_{ht}. The latter take account of the sensitivity to source temperature variations, the power required for auxiliaries (N_{aux}), and the likely defrosting losses[+]. The most important factor determining the SPF_{eff} is the COP_{eff}. For electrically driven systems this is simply the ratio of the thermal output from the condenser to the electrical power input to the compressor, while for sorption systems it is the sum of the heat outputs from the condenser, absorber and possibly desorber divided by the sum of the thermal inputs to the generator and the electrical power to the solvent pump and auxiliaries (and sometimes also the compressor). For engine driven systems, (either internal or external combustion) the thermal output is the sum of that of the condenser and the heat recovered from the engine. The COP_{eff} is the total thermal output divided by the thermal input to the engine. This can be expressed as

$$COP_{eff} = COP_g \cdot \eta_m + f \cdot (1 - \eta_m)$$

where COP_g is for the heat pump sub-system alone, i.e. the condenser heat output divided by the shaft power, η_m is the efficiency

[+] cycling losses not included

of the drive, i.e. shaft power divided by thermal input, and f is the fraction of engine waste heat recovered. Fig. 3-1 is a graph of COP_{eff} against η_m for various values of COP_g and f with the ranges of η_m for various drives identified. From Fig. 3-1 the following conclusions can be drawn:

a) The same COP_{eff} can be achieved by different combinations of η_m, COP_g and f. For example a COP_{eff} of 1.4 can be obtained with $\eta_m = 0.2$, $COP_g = 3$ and $f = 1.0$ (a Rankine-Rankine system) or with $\eta_m = 0.33$, $COP_g = 3$ and $f = 0.6$ (Otto-Rankine system).

b) The COP_{eff} of a system with high η_m is much more sensitive to variations in the heat pump sub-systems performance, i.e. COP_g, than one with a low η_m.

c) Even under the worst possible operating conditions (i.e. with low source temperatures and COP_g 2.0) all engine driven systems have a $COP_{eff} > 0.9$, so they are always more efficient than boilers.

d) The most efficient system is the large Diesel-Rankine system with $\eta_m = 0.4$, $COP_g = 4$ (high source temperature e.g. waste heat) and $f = 0.8$. This system can achieve a COP_{eff} of about 2.1.

Table 3-1 shows the average COP_{eff} for different systems with 0^oC evaporating temperature (or heat pump sub-system minimum temperature for the resorption and Ericsson-Ericsson systems) and 60^oC condensing temperature (heat pump sub-system maximum temperature). The systems are ranked in Table 3-1 in order of COP_{eff}. Also included for the sake of completeness is the electric-Rankine system driven both by hydroelectricity ($\eta_{el} \simeq 0.9$) and thermally generated electricity ($\eta_{el} \simeq 0.3$). Three of the systems with the highest COP_{eff}, i.e. those with $COP_{eff} \geqslant 2$, are also unproven systems for which the figures are solely based on theoretical predictions (both resorption types and the Ericsson-Ericsson). Of the remaining systems those with COP_{eff} over 1.5 are the hydro-

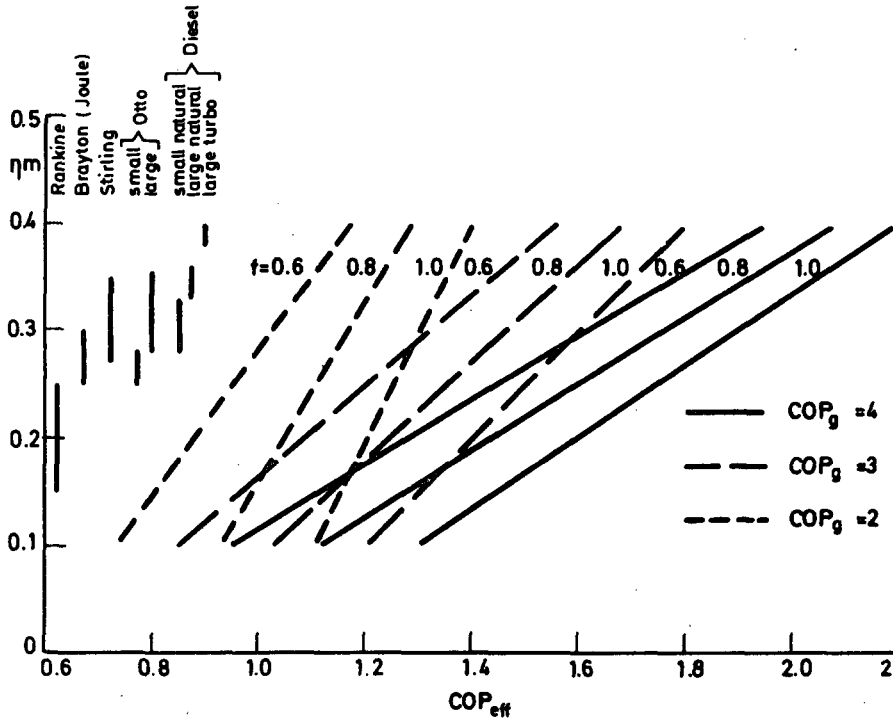


Fig. 3-1 : COP_{eff} of engine driven systems as function of drive efficiency

power driven electric heat pump (which is of limited interest) and most of the engine driven systems. Of the engine driven systems only the Rankine-Rankine (single fluid) and Brayton-Rankine lie below this figure, simply because of their low drive efficiency $\eta_m < 0.27$.

Order	System	COP_{eff}
1	resorption-electric drive (hydropower)	4
2	electric-Rankine (hydropower)	2.7
3	Ericsson-Ericsson	2
4	resorption (thermal drive)	2
5	Diesel-Rankine	1.7-2.0
6	Otto-Rankine Rankine-Rankine (dual fluids)	1.4-1.6
7	two-stage absorption	1.3-1.5
8	Rankine-Rankine (single fluid) single-stage absorption	1.2-1.4
9	Brayton-Rankine	1.2-1.3
10	resorption-electric drive (thermal power generation)	1.2
11	electric-Rankine (thermal power generation)	0.9
12	heat transformer	0.5

Table 3-1: Ranking of systems by COP_{eff} at design point

The absorption systems also have $COP_{eff} < 1.5$ while the electric-Rankine system with thermally generated electricity has $COP_{eff} \simeq 0.9$. Lowest of all is the heat transformer with a $COP_{eff} \leq 0.5$.

3.2.3. Below Design Point Performance

The variation of COP_{eff} (Fig. 3-2) and capacity (Fig. 3-3) with source temperature (usually outside temperature or loosely related to the latter if either horizontal ground source or surface water are used as source) is primarily a function of the heat extracted from the environment by the evaporator and secondly of the part-load efficiency of the drive. Table 3-2 shows the size of evaporator for each system relative to that of an air source electric heat pump - the only one for which sufficient experimental data on part-load and frosting performance exists. This figure expresses not only the relative size of the evaporator but also the proportion of the total heat output which is gained from the environment - the larger the gain from the environment the higher the COP_{eff}. It also expresses the sensitivity to changes in source temperature, since if no heat, N_{env}, is gained from the environment the COP_{eff} is the same as or worse than that of a boiler. This condition occurs when the mass flow of refrigerant ($\dot{m}_r$) though the evaporator decreases to a negligable value, which depends on the refrigerant used:

$$N_{env} \propto \dot{m}_r \propto \frac{1}{v}$$

where v, is the specific volume of the refrigerant at the compressor inlet. For reciprocating compressors, R22 or R12 is used, and between 0^oC and -10^oC the mass flow $\dot{m}_r$ decreases for both fluids by a factor of 0.72. For R114 or R11 (used in rotary compressors) $\dot{m}_r$ decreases over the same range by 0.67. Thus the systems using R114 or R11 (e.g. Rankine-Rankine systems) would be more sensitive to a decrease in source temperature than is inferred from Table 3-2. The same is true of the single-stage absorption and ejector heat pumps. In the former the vapour pressure over the absorber reaches

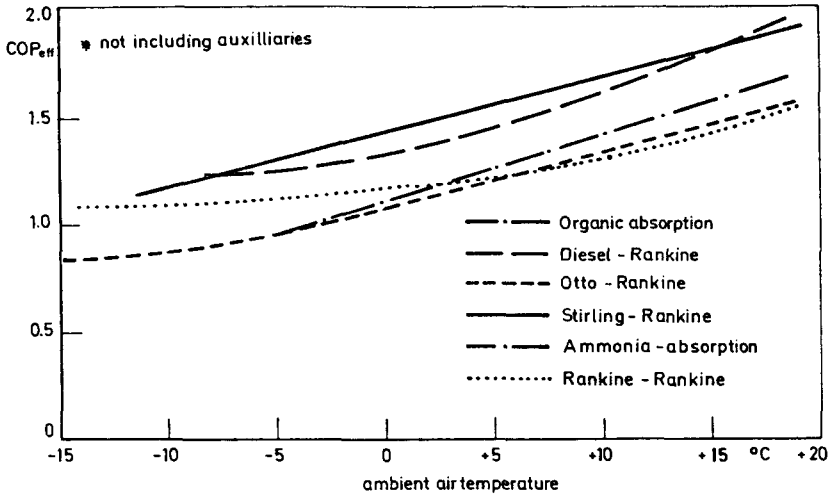


Fig. 3-2: COP^{*}_{eff} of air source heat pumps

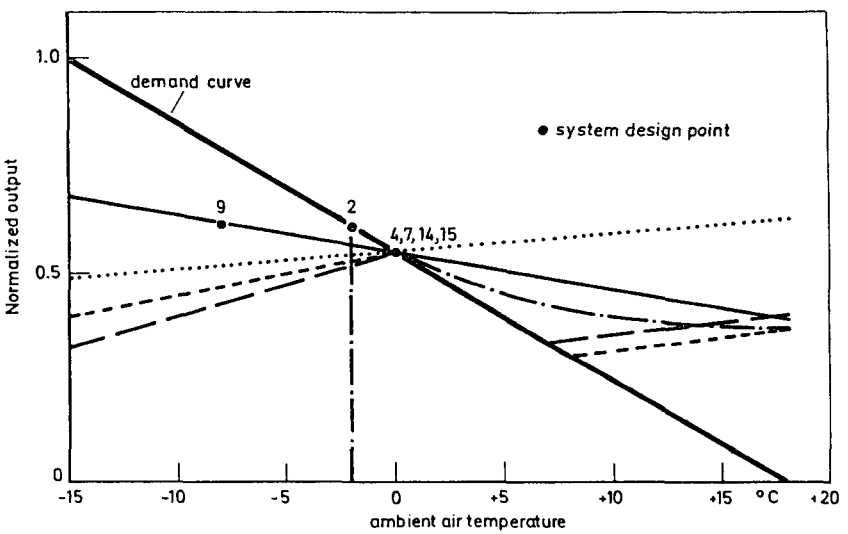


Fig. 3-3 : Normalised output of air source heat pumps

Order	System	Relative+ evaporator size	Estimated++ losses due to defrosting (%)
1	two stage absorption	0.35	1-2
2	Rankine-Rankine	0.47	1.3-2.1
3	single-stage absorption	0.45	1.4-2.3
4	ejector heat pump	0.50	1.5-2.5
5	Otto-Rankine Stirling- Rankine	0.61 0.61	1.8-3.1 1.8-3.1
6	Brayton-Rankine	0.67	2.0-3.4
7	Diesel-Rankine	0.71	2.1-3.6
8	Ericsson-Ericsson	0.67	2-3.4
9	electric-Rankine	1.0	3-5
10	heat transformer	1.14	3.5-5.7
11	resorption-electric drive	1.35	4.1-6.8

\+ Relative to electric-Rankine system

++ Estimated from defrosting losses for electric-Rankine /70/ and relative evaporator size

Table 3-2 : Ranking of systems by relative evaporator size and sensitivity to frosting

a point at which absorption becomes ineffective /49/, while for the latter the ejector pump effectiveness is a strong function of the suction vapour pressure.

Another aspect influencing below design point performance for some systems (Rankine-Rankine, absorption and ejector) is the matching of the drive subsystem and the heat pump subsystem. As already indicated the absorption system performance also depends on the absorber performance, while for the Rankine-Rankine the machinery (turbine and compressor) are likely to be mismatched off the design point. Just how mismatched cannot be quantified; although it is probably worse for single fluid systems than for two fluid systems. The capacity of the Rankine-Rankine, absorption and ejector systems below the point at which their efficiency is less than that of a boiler (or their capacity is lower than the demand) can be supplemented by operating them as a direct heating system. In the absorption system, for example, the generator/absorber loop can be used simply as a heat transfer system with the refrigerant loop isolated. Similarily the Rankine-Rankine system can operate with its feed heater being used for direct heat exchange, or a second heater or boiler can be operated in parallel with it. The most important aspects of the internal boiler operation of such systems is that this has to be designed for maximum demand and that the system is more efficient when the heat pump and internal back-up act in parallel rather than alternatively.

However Table 3-2 , to a first approximation, reflects the sensitivity of systems to variations in source temperature. Thus the systems with a middle range COP_{eff} (such as the Rankine-Rankine, and absorption systems) are less sensitive to fluctuations in source temperature than those with higher COP_{eff} at the design point, such as Diesel-Rankine. This is also true for the effect of frosting on air source systems - the smaller the contribution of the evaporator to the total heat output of the system, the lower the sensitivity to frosting losses (or fouling losses in ground water source systems). Thus the systems at the top of Table 3-2

have the advantages of

1) small evaporator, hence lower contribution to capital cost, particularly for ground source systems
2) lower sensitivity to source temperature fluctuations
3) less defrosting and fouling losses.

The effect of these three advantages is that the SPF_{eff} of these systems is better than would be expected if all systems had proportionally equal losses.

3.2.4. Auxiliaries

Another parameter that influences SPF_{eff} is the proportion of the energy input needed for the auxiliaries such as fans and pumps. For air source systems the evaporator fan power is significant, and this too, is in direct proportion to the evaporator size. Thus the absorption and Rankine-Rankine systems benefit from low fan power. However, both systems require significant pump power. The Rankine-Rankine requires a feed pump to operate over a high pressure ratio ($\simeq 5$) and the absorption systems require solvent pumps, acting over high pressure ratio. This is exacerbated by the very low efficiency ($\simeq 50$ %) of such pumps (i.e. solvent and feed pumps) for small mass flows and high pressure ratios.

However, assuming that pumps of high efficiency can be successfully developed, the below design point performance of the Rankine-Rankine and absorption systems look inherently better than in the case of the other systems.

3.2.5. Above Design Point Performance

Above design point operation of any system (see Fig. 3-3) corresponds to an excess of heat pump capacity over the demand. In simple systems, such as first generation electric heat pumps, the output is controlled by on/off switching with a small buffer store or time delay arrangement to avoid excessive starts. The range of controllability and the part-load efficiency of the systems are very important because if the control range is small or the part-load efficiency is low then the system performance above the design point will be poor. Therefore the best systems are those with a wide control range and good part-load efficiency above the design point. However, the minimum demand of a system is usually a very small fraction of that at the design point and none of the heat pump systems can be controlled (other than by switching) at very low capacities.

The parameters which effect the range of controlled output are:

1) Sensitivity of heat pump sub-system output to source temperature,
2) range of speed variation of the drive sub-system,
3) range of capacity control of the heat pump sub-system.

The sensitivity of heat pump output to source temperature has already been discussed with respect to below design point operation. Above design point the evaporator extracts increasingly more heat as the source temperature increases, which is in contrast to the decrease in demand. By reducing the capacity of the heat pump subsystem the mass flow and hence the system capacity can be reduced. For example, in compression system this can be achieved in steps, by reducing the number of cylinders in use in reciprocating compressors, or continuously, by reducing the volumetric ratio of screw compressors or by inlet vane control in centrifugal compressors. All these methods have been developed for large compressors

(>30 kW power input) and there is little reduction (≤5 %) in system efficiency by these methods. For small compressors capacity control is expensive and the part load operation is usually less efficient than full load operation. Capacity control by speed control of the drive is also possible and can be achieved in a number of ways. For electric motor systems, changing the number of poles can be used to give step changes in speed, or regulation of the supply by thyristors can give continuous speed control over a similar range (2:1). Internal combustion engines can be controlled over a range of about 2:1 to 3:1, with a reduction in drive efficiency on part load by about 5 %. The external combustion engine systems can also be speed controlled over a range of about 2:1, but as yet it is unknown what the part load efficiency is likely to be.

Table 3-3 ranks the systems by controllability and it is clear that the large engine driven systems are much more flexible than the other systems. Referring to Fig. 3-3 , which applies to small systems only, none of the systems can be controlled below about 50 % of the design capacity. At this point the output capacity begins to increase slightly due to the increasing evaporator capacity, and so below this point on/off operation becomes necessary. For most small systems it is still necessary to control by on/off switching - a method which results in reduced system life (see Sections 3.3. and 3.4.).

So far only air-source systems have been considered. Of the other sources only earth and ground water can be considered practical for small systems, while waste heat, surface water, sea water, geothermal, etc. are only applicable to larger systems. Ground water and ground sources are much less subject to temperature variations than air and do not have its frosting problems. The ground can be used as heat source with either horizontal or vertical coils. The former require a large surface area and follow the air temperature fluctuation to some extent but with heavy damping and reduced amplitude. They can also adverse environmental effects by shortening the growing season and are costly to instal.

Order	System	Control	Control range	Characteristics
1	large Diesel-Rankine large Otto-Rankine	drive speed + compressor unloading	4:1 to 9:1	continuous/ step
2	large electric-Rankine	drive speed + compressor unloading	2:1 to 4:1	step or continuous
3	small Diesel-Rankine small Otto-Rankine	drive speed	2:1 to 3:1	continuous
	absorption	solvent pump regulation of gene- rator temperature	2:1 to 3:1	continuous
	Brayton-Rankine	drive speed and on-off	2:1 to 3:1	continuous and step
4	Stirling-Rankine	speed	2:1	continuous
	small electric-Rankine	drive speed	2:1	step
	Rankine-Rankine	speed	2:1	continuous

Table 3-3: Ranking of systems by controllability

The vertical coils are much more compact, are insensitive to fluctuations in the ambient air temperature, have little environmental impact, but are also relatively expensive to instal. In both cases the cost of the heat source equipment approximately equals that of the heat pump unit. The greatest technical disadvantage of ground sources is that an intermediate heat transfer fluid has to be used, and this entails a reduction in evaporating temperature. Thus the SPF_{eff} of such systems is little better than that of air source equipment. However, the heating capacity of the heat pump does not decrease during colder spells to the same extent as air source equipment, giving a better match to the heat demand curve. A further disadvantage of vertical systems is the knowledge of local geology that is necessary before the sizing and costing of an installation can be made.

Ground water systems are similar to vertical ground source systems in their characteristics in so far as there is little fluctuation of source temperature throughout the year relative to air. Most ground water systems require two bore holes, for feed and return water, since water authorities in most countries do not permit wastage of this resource. In addition the water authorities in some countries are concerned that leaks of refrigerant could pollute the ground water and may insist on the use of an intermediate fluid. Fouling of heat exchangers can be a severe problem especially where there is a high iron content in the water.

3.2.6. Seasonal Performance and Energy Saving

The true heating SPF_{eff} can only be estimated for each application and climatic region (this was a major task of the market study). It is therefore not possible to assess the systems solely on the variation of COP_{eff} and N_{ht} with source temperature. In Section 3.5. the systems shown in Fig. 3-2 and 3-3 are compared for a particular application and climate to assess their SPF_{eff} under

these circumstances.

The energy savingcapability of the systems must in the end be assessed by comparing a complete heating season performance, i.e. SPF_{eff}, against that of a conventional system. As an example Fig. 3-4 shows the percentage potential energy saving of heat pumps as a function of the COP_{eff} and the conventional boiler efficiency η_b, for which the full load value of a new boiler should be used. For a more valid comparison, SPF_{eff} and an annual value of η_b should be used.

From Fig. 3-4 some general points can be made:

1) All systems with COP_{eff} over 1.0 save energy.
2) 30 % energy saving can be achieved with a COP_{eff} of 1.2, of which most of the advanced systems are capable.
3) 50 % energy saving requires a COP_{eff} of about 1.7. Of the systems at advanced stages of development only the large Diesel-Rankine system is capable of achieving this figure.

3.2.7. Cooling Performance

The cooling performance of advanced heat pumps is of interest to countries such as the USA, Japan, Spain and Italy, where summer cooling and air conditioning are often required. Table 3-4 compares the cooling coefficient of performance and capacity of the various systems for 35°C ambient air temperature. Also given is the US Standard, /26/, which lays down minimum requirements for cooling coefficient of performance. Neither of the coefficients of performance include auxiliaries. The cooling capacity at 35°C ambient air temperature and 20°C indoor temperature is normalised by dividing by the heating capacity, N_{ht}, at 0°C ambient and 20°C indoor temperature. In terms of cooling COP the electric system with hydro-electricity is best, but if it is ignored because of its lack of general applicability the best system is the Brayton-Rankine, followed by the Stirling-Rankine system. In fact most of the fossil

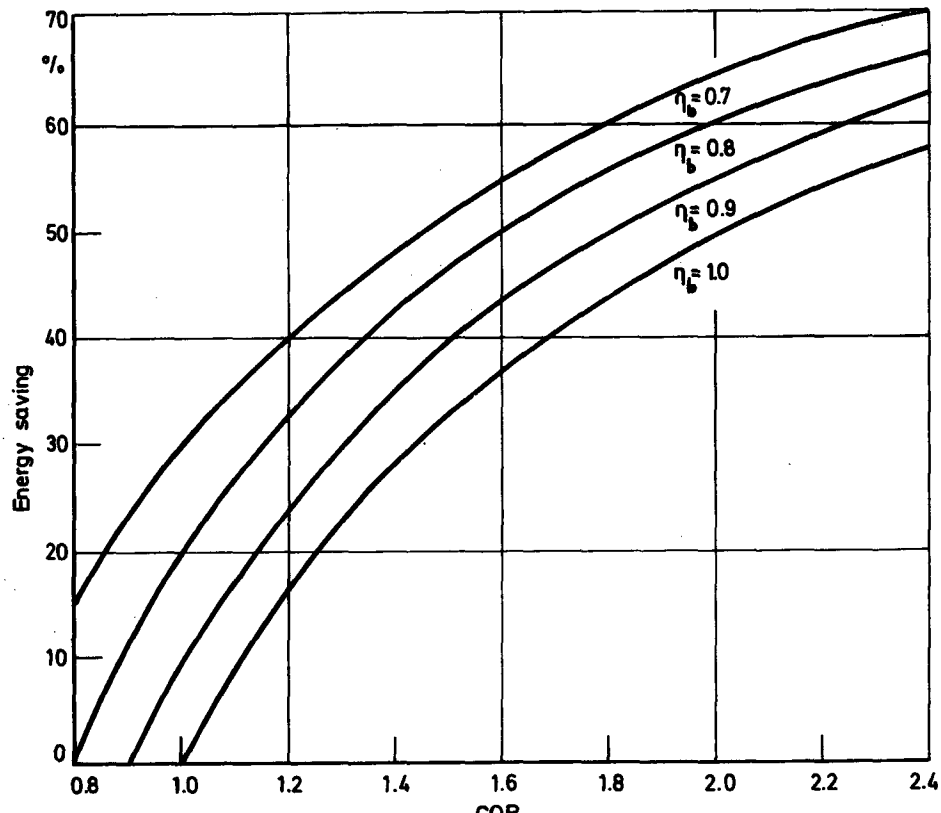


Fig. 3-4 : Energy saving potential of heat pump systems

Order		COP^{+}_{eff} 1) predicted	COP^{+}_{eff} 2) U.S.Standard	Relative 3) cooling capacity
1	electric-Rankine (hydroelectrc)	2	2.0-2.2 (1.8-2.0)	1.3
2	Brayton-Rankine	1.1	0.48	1.2
3	Stirling-Rankine	0.88	0.48	0.7
4	electric-Rankine (thermally gen. elect.)	2.5-2.7 (0.8-0.9)	2.0-2.2 (0.67-0.73)	1.3
5	large Diesel-Rankine	0.8	0.48	0.5
6	large Otto-Rankine small Diesel-Rankine	0.67	0.48	0.5
7	small Otto-Rankine	0.6	0.48	0.5
8	single-stage-absorption	0.5-0.6	0.48	0.4
9	Rankine-Rankine (single fluid)	0.4-0.5	0.48	0.4

1) COP^{+}_{eff} does not include auxiliaries

2) ASHRAE STANDARD 90-75 /26/, not including auxiliaries, primary energy-related value in parenthesis

3) Relative to design heating capacity N_d (kW)

Table 3-4 : Cooling performance of heat pumps

fuel-fired systems easily exceed the performance required by the ASHRAE Standard /26/. However, the electric-Rankine system with thermally generated power has a prime energy-related coefficient of performance in the range 0.8 to 0.9, which is superior to all but the Brayton-Rankine and Stirling-Rankine systems. With respect to cooling capacity the electric-Rankine, Brayton-Rankine and Stirling-Rankine have the highest cooling capacities relative to their heating capacity. Thus if a system is sized for a cooling load larger than the heating load, e.g. in the Southern US , then the electric-Rankine and Brayton-Rankine systems are superior. If sized for more severe winter conditions and a smaller summer cooling load, e.g. North East US, then the Stirling-Rankine is the best. Thus for combined heating and cooling applications there would appear to be no justification for the development of fossil fuelled systems other than the Brayton-Rankine and Stirling-Rankine systems, since most of them are at best equal or inferior to the well-established electric-Rankine system.

3.3. Design

The most important aspects of heat pump design are those relative to performance controllability, reliability and cost, which are discussed in Sections 3.2., 3.4. and 3.5. However, there are a number of other important aspects, which relate to the installation and use of systems and which can be significant factors in their acceptability. These include the size and weight of the system, the necessity of a back-up, the method of control, the safety and the flexibility with respect to different source and sink types and other characteristics.

3.3.1. Size and Weight

The unit size and weight are only significant for the smaller systems, especially for single room (3-4 kW) or single family house heating (10-20 kW) and cooling. The most important criterion is that the system including its back-up should not require more space than the conventional alternative, or at worst require only a little more space. For new installations the latter is more valid, but in the case of retrofit applications - which is probably the largest potential market - the available space could be severely limited. Clearly all those systems which require an additional back-up are at a disadvantage, even though in the retrofit case the existing boiler would be used as back-up. The actual size and weight of each heat pump systems depends on its application. In Section 3.5.1. the systems are discussed for single family house heating, with air as source and water as sink - the most likely combination in Western Europe.

3.3.2. Energy Demand

The major impetus towards the use of heat pumps is energy conservation, the secondary objective usually being to substitute fuel, i.e. replace imported by home produced fuel. The higher the COP_{eff} and SPF_{eff} of a system the greater the energy saving (see Fig.

3-4), but this is no guide to the degree of fuel substitution. In addition the issue is complicated by the use of mixed fuel systems such as electric heat pump with an oil-fired back-up, for which the value of the energy saving and degree of fuel substitution is contentious. A further aspect is the strain the systems would place on the energy supply system. This is a well known problem with the electric heat pumps which the electricity utilities promote, but only when bivalent (and preferably bivalent - alternative) so that the peak demand on the electricity network is not further increased. Indeed some gas utilities in Germany now discourage the use of gas-fired back-ups for electric heat pumps /69/ because the peak demand is transferred to the gas system. Therefore the possible impact of advanced heat pumps in this respect should also be considered.

Table 3-5 ranks small systems by the ratio of peak demand (fuel or electricity) to mean demand, i.e. the demand at +3°C ambient air temperature. For each system the type of back-up is specified and the mode of operation (monovalent or bivalent, parallel or alternative) and the ambient temperatures at which the heat pump is cut out and at which the back-up is cut in. Also shown is the sizing of the back-up and the heat pump as percentage of the heat demand at -15°C ambient temperature. Included in Table 3-5 is the simple gas boiler, here assumed to have 75 % efficiency at full load (-15°C) and 65 % at 45 % load (+3°C). On this basis the peak demand is approximately double the mean demand for a gas boiler.

Also included is the oil-fired Diesel-Rankine system, for which the relative peak demand is unimportant since the oil storage overcomes this problem. For two of the electric systems (those that use an oil- or gas-fired back-up) relative peak demand figures are given both uncorrected and corrected to prime energy. Comparing the four electric systems listed it is clear that the electric back-up is most unsatisfactory as it has a very high peak demand while the monovalent system requires a heat pump at least double the size of the others and as in primary energy related terms there is

Order	Operation	Heat pump		Back-up	Cut out/in temp (0°C)		% full capacity		Ratio	Advantages	Disadvantages
		drive	heat pump		heat pump	back-up	heat pump at +3°C	back-up at -15°C	peak/ mean power demand		
1	monovalent	Brayton	Rankine	-	-15	-	45	-	1.5	simplicity, compactness, low peak demand	
2	monovalent	-	-	gas boiler	-15	+18	-	100	1.5	simplicity, compactness, low peak demand	high mean demand
3	bivalent-parallel	electric	Rankine	boiler/furnace (oil)	-15	+3	45	70	4.9+ (2.1)	flexibility	large size, requires oil boiler to meet peak demand
4	bivalent-alternative	electric	Rankine	boiler/furnace (oil)	+3	+3	45	100	7.4+ (2.2)	flexibility, best electricity tariff	large size, requires oil-boiler to meet peak demand
5	monovalent	electric	Rankine	none	-15	-	100	-	2.8	simplicity	large size, high capital cost, system grossly over=sized for average conditions, poor controllability
6	bivalent-parallel	Otto	Rankine	boiler(gas)	-15	+3	45	60	3.3	flexibility	large peak gas demand, large size
7	bivalent-parallel	Stirling	Rankine	furnace (gas) internal	-15	+0	55 (at 0°C)	35	3.4	simplicity, compactness, flexibility	large peak gas demand

continued overleaf

Table 3-5: Ranking of small systems by peak demand

8	bivalent-alternative	absorption (single stage)	boiler(gas) internal	-2	-2	60 (at -2°C)	100	3.6	simplicity flexibility, compactness	large peak gas demand
	bivalent-parallel	Rankine Rankine	boiler(gas)	-5	0	55 (at 0°C)	100	3.6	simplicity, flexibility, compactness	large peak gas demand
9	bivalent-parallel	Diesel Rankine	boiler(oil)	-15	+3	45	70	3.8	flexibility	large size
10	bivalent-parallel	electric Rankine	electric	-15	+3	45	70	4.7	simplicity, compactness	high running cost, large electric peak demand, no energy saving against boiler

Table 3-5: Ranking of small systems by peak demand

little difference between the bivalent-parallel and bivalent-alternative systems. The significant difference is that for the bivalent-alternative the peak demand on the gas system is greater.

Considering only the gas-driven advanced systems, the Brayton-Rankine is clearly superior to the others (from this aspect). This results from its high degree of controllability and sizing for the maximum heating load. All the remaining systems have peak demands between 3 and 4 times the mean demand; giving a ratio which is double that of the simple boiler. This means that from the point of view of the gas utilities the same size of network has to be installed for a gas-fired heat pump as for a simple gas boiler with less gas sold per installation. Thus the widespread use of gas-fired heat pumps is likely to produce the same problems for the gas utilities as electric heat pumps produce for either them or the electricity utilities.

3.3.3. Safety

Another design aspect is safety, comprising the risks from fire, explosion and toxicity. Table 3-6 lists some of the risks and indicates which heat pumps could produce them. The major problems would appear to arise from those safety risks resulting from the introduction of these systems into domestic premises. These are mainly risks associated with the release of working fluids, their pyrolysis or combustion or interaction with air or water. Thus ammonia systems, those bringing high temperatures into proximity with organic working fluids (absorption, Rankine-Rankine), and those which could introduce toxic or possibly carcinogenic substances into domestic water show greatest risks.

Safety standards are quite strict for ammonia systems; for residential locations they require a charge of no greater than 2.5 kg /129, 130/ or prohibit their use completely (Japan). Additional restrictions prohibit their use for air conditioning, for human comfort and close location to electrical switchgear /129/.

Risk	Possible Causes	Systems likely to produce risks	Possible remedies
Fire	leak of inflammable substances	organic absorption (E181)	intermediate heat transfer fluid (expensive)
	leak of fuel	all gas- or oil-fired systems	normal risk - standard safety procedures
		Stirling engine (H_2)	use HE instead of H_2 as work fluid (expensive)
Explosion	leak of fuel	all gas- or oil-fired systems	normal risk - standard safety procedures
	leak of explosive substances	NH_3/H_2O absorption systems	system enclosure vented to atmosphere in event of leak, or double-skin system
Toxicity	leak of toxic substances	NH_3/H_2O absorption systems	system enclosure vented to atmosphere in event of leak, or double-skin system
	production of toxic substances in combustor of drive	all systems with internal or external combustion and use of R12, R22, R114 etc.	combustor air must be external-exhaust or flue must be ducted beyond risk zone
Other	physical injury by machinery	all mechanical systems with open shafts	sealed systems or interlocked enclosures
	production of corrosive substances	all combustion systems	better construction materials

Table 3-6 : Safety problems with advanced heat pumps

For Group 1 refrigerants (R12, 22, 114 etc.) the charge size is limited by the maximum permissable concentration of vapour in the smallest humanly occupied space /129/.

The remedies suggested are that -

(1) all combustion systems should breath external air only,

(2) the exhaust outlet should be remote from the inlet and from locations where it can be inhaled,

(3) domestic hot water (for baths, showers etc.) should not be directly heated by the working fluid,

(4) system working fluid charges should be as small as possible,

(5) double-skin heat exchangers or intermediate heat transfer fluids should be employed wherever possible (feed heaters, generators etc.)

3.4. Operation

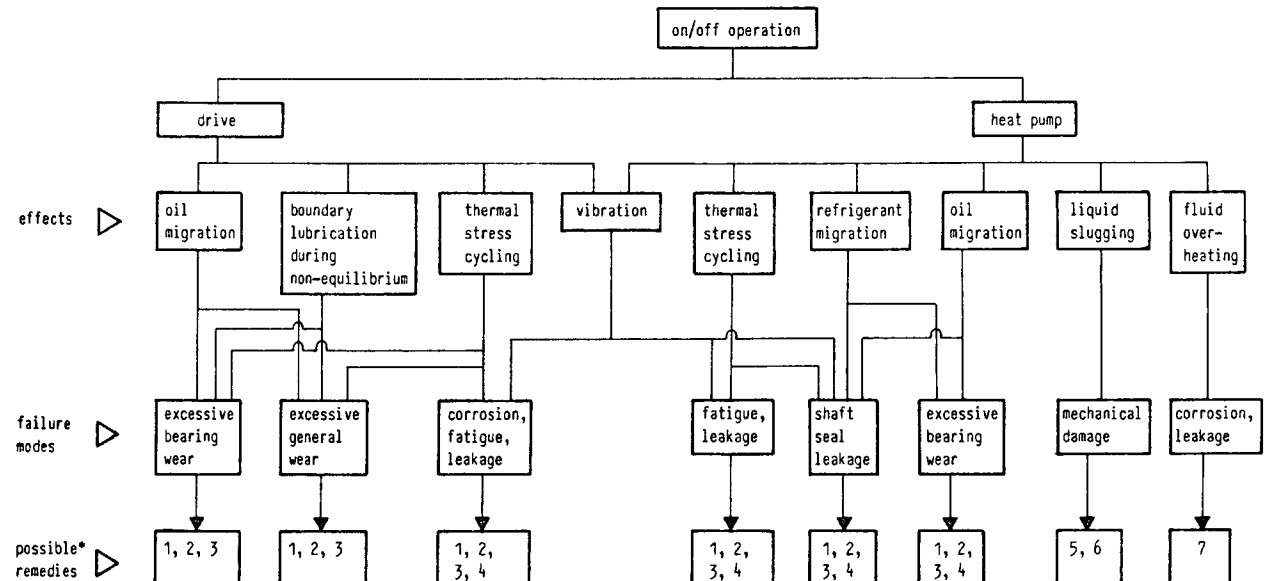
The operational aspect of advanced heat pump systems is one of the most difficult to assess as many of the systems have not yet been in operation or have operated only as prototypes. The most important factors in the operation of such systems are their controllability, reliability, servicing needs, life, and environmental effects. All these have to be considered from several different points of view, such as that of the user, the supplier, the installer and service contractor, the fuel supplier, the local authority and society in general.

3.4.1. Control

Controllability has been already discussed at length in Section 3.2. The aspects of controllability relevant to the operation of the heat pump are the complexity of auxiliary equipment required and the effect which the method and type of controls have on system life and reliability. In these respects the simple on/off control system is most unsatisfactory, inducing accelerated wear in mechanical systems and giving increased thermal losses and slow system response (e.g. in the absorption systems). More complex control methods could entail greater capital cost and possible system failures due to control system unreliability or lack of adjustment. The avoidance of on/off operation should however be a design objective for all systems, although it cannot be avoided at minimum system loadings, e.g. in spring and autumn months.

3.4.2. Reliability

Table 3-7 shows the possible failure modes for advanced heat pumps caused by on/off control. It is generally recommended that some of the techniques which have been employed for electric-Rankine systems (items 1-6) should be used for the heat pump sub-system, while the engine systems should employ additional protective measures for the motor itself (e.g. item 3).



systems ▷	Diesel-Rankine Otto-Rankine Stirling-Rankine Rankine-Rankine (free piston)	1, 2, 3, 4, 5, 6.	Rankine-Rankine (turbine) sorption	1, 2, 3, 4, 5, 6, 7. 1, 2, 4, 7.

*
1. minimum run time interlock
2. buffer heat store
3. thermostatic oil heating during off period
4. compressor unload during start up
5. use capillary expansion device and suction accumulator
6. use pump-down with thermostatic expansion valve
7. operate feed pump before start-up

Table 3-7 : Failure modes caused by on/off control

Reliability is also determined by effects other than the controls. These include fouling of the heat exchangers, loss of adjustment of settings, loss of charge due to corrosion or fatigue in normal running, air or water in the refrigerant system, etc. For the systems employing the conventional reciprocating rotary or centrifugal compressors, the normal system practices developed for electric heat pumps and other refrigerating and air conditioning systems with widely fluctuating evaporating and condensing temperatures should be employed. There is extensive experience particularly in the U.S., concerning the failure modes of such systems /64, 66/. Good system practice includes the use of suction accumulators, sump heaters, pump-downs and/or check valves, and evaporator defrost control by air-side differential air pressure transducers. System cleanliness and good control over system charging are important, as are reliable controls and intruments.

The internal combustion engine driven systems, particularly small systems (<100 kW shaft power), also have the reliability of the prime mover as a significant factor. Larger systems have been in operation as heat pump drives for about three years in West Germany and also there is considerable experience of their use for other stationary duties.

Recent reports on these Otto-Rankine heat pump installations, which are mainly for swimming pool and leisure centre heating, indicate that the initial periods were not without problems with the engines and to a certain extent the compressor. Table 3-8 lists the problems found with two such installations. The major problems appear to have been those associated with leaks of refrigerant (particularly from shaft seals) into the engine exhaust inlet with the possible production of toxic substances. The reliability problems of large Diesel engine systems can be expected to be similar to those of large Otto systems.
Small Otto and Diesel systems have not been operated long enough to be able to assess their reliability. However, development work to convert automobile Otto engines for heat pump drives has indicated a number of modifications to improve reliability and prolong

No.	System Description						Instal. date	Installation problems	Reliability problems	Oil consumption (l/h)	Service intervals		Ref.
	Output (kW)	Source temp. (°C)	Delivery temp. (°C)	Motor type and shaft power (kW)	Compressor type	Refrigerant					Oil and oil filter (hrs)	spark plugs (hrs)	
1	3x1270	+10	+45/+80	Otto 3x235	twin screw	R22	Sept. 1977	1) system initially oversized, excessive on/off operation 2) engine air inlet changed to external for safety	1) 4 months life of exhaust gas heat exchanger 2) refrigerant leaks from crankshaft seal and oil separator entered exhaust	not known	1500-4000	6000-10000	/131/
2	700	+8/+25	+24/+55	Otto 1x83	reciprocating	R22	March 1977	none stated	1) crankshaft seal replaced every 1000 hrs 2) refrigerant expansion bellows failed 3) water leak into exhaust gas heat exchanger 4) oscillations in evaporating temp. caused by wrong size expansion valve 5) starter motor changed after 3700 hours	0.09	1000	1000-1600	/132/

Table 3-8 : Reliability and service: Otto-Rankine heat pump systems

life /21/. These are summarised in Table 3-9 , along with further comments received from engine manufacturers. These modifications and system practices are considered necessary to ensure service intervals of 1000 hours, which is equivalent to about five times normal automobile industry practice and this is only approximately one half to three quarters of a heating season. A Ford engine of approximately 30 kW shaft power described in reference /21/, will incorporate the above modifications and will be available in mid-1980.

Smaller (< 30 kW) Otto engines with water cooling are not available for heat pump drives, although some heat pump manufacturers have advertised systems with them. Thus the COP_{eff} of such systems is considerable less than the theoretically possible since only exhaust waste heat is recovered.

External combustion engine driven systems should inherently be more reliable than internal combustion systems because they have stoichiometric combustion, and also no erosion of valves or oil degradation by the combustion gases. A study of past failures of such systems has been undertaken by the American Gas Association /134/ and the history of the development of free piston, turbine and absorption systems shows failure to achieve performance and first cost objectives. In addition, the reliability problems of systems such as the free-piston Otto-Rankine and rotary vane Rankine-Rankine systems led to their abandonment while the inability to develop an efficient ejector led to the abandonment of an ejector-Rankine system.

The kinematic Stirling systems are however complicated and as yet unproved in practice. The free piston Stirling system is also complex, particularly with regard to the sealing between the engine and the compressor within the hermetic casing. The free piston and turbine Rankine-Rankine systems are very much simpler in construction and should show high reliability. The latter type with a single fluid is particularly simple mechanically, with no shaft

1. Hardened valve seats and valves
2. Integrated heat recovery package for cooling water and exhaust gas and cooling water heater for off periods
3. Electronic ignition
4. Speed control by governor operation from inlet vacuum
5. Enlarged oil filter
6. Oil cooler for normal running
7. Oil heater for off periods
8. Auxilliary oil tank to increase oil charge to be sufficient for 1000 hours
9. Enlarged air filter
10. Air inlet from exterior
11. Service every 1000 hrs: oil and oil filter change, spark-plug change, valve clearance adjustment

Table 3-9: Improvements to automotive Otto engines to increase life and reliability /21/

seals, valves or lubricant to cause failure. The possible causes of unreliability will be those associated with control and regulation, since the high speed turbine assembly uses its high rotational speed to create a gas bearing, and also those associated with the possible breakdown of the work fluid if high feed heater temperatures are used to obtain high performance. The absence of oil (and iron which acts as a catalyst) from the system increases the chances of success. The feed pump could contribute to unreliability, since it has to pump liquid over a wide pressure range.

Sorption systems require a solvent pump (sometimes two) which has to operate under the same conditions as the feed pump of the Rankine-Rankine systems. Also the generator, like the Rankine-Rankine feed heater, could produce chemical breakdown, or reaction between the solvent and the refrigerant. This is particularly true of the organic systems, on which little chemical reactivity and stability data is available.

In general terms it would appear that the systems with the greatest potential reliability are the sorption and Rankine-Rankine systems, followed by the Stirling-Rankine and Brayton-Rankine systems.,

3.4.3. System Life

System life is closely related to reliability, service and maintenance. From the field of refrigeration it is clear that sorption systems have very long lives - longer than equivalent compression systems. The internal combustion engine systems, particularly small ones, are likely to have the shortest lives, while the external combustion systems would be intermediate. Over a 10-year life a heat pump system can be expected to operate approximately 10,000 to 40,000 hours, and this compares to between 1,000 and 2,000 hours for an automobile engine (with services about every 100 hours) and about 30,000 for a domestic refrigerator. It is therefore considered realistic to expect 10 to 15 years for an electric-Rankine system, similar for the Otto-Rankine and Diesel-

Rankine systems, but with periodic engine overhauls or exchanges, and 15 to 20 years for the external combustion and sorption systems.

3.4.4. Service and Maintenance

The service and maintenance requirements and costs can at present only be roughly estimated. Table 3-10 shows the service intervals, service requirements and estimated costs for similar sized systems for a single family house. It is quite clear that with the present technology the Otto and Diesel driven systems are uncompetitive in this respect. The situation is further complicated by the mix of technical skills required by such systems (the motor mechanic for the engine and the refrigeration mechanic for the heat pump). Unless an infrastructure can be established with a network of mechanics trained in both skills then service and maintenance will be a troublesome aspect in the possible use of these systems. Not included in Table 3-10 are the costs of a replacement engine or a complete overhaul, which would be required at least in the life of the system. A complete overhaul costs approximately 30 % of a new engine cost, while a replacement engine could cost 50 % or more of a new engine cost (plus installation cost).

The absorption, Rankine-Rankine and free piston Stirling-Rankine should require considerably less maintenance since they have few moving parts and these can be easily replaced. The major uncertainties concern the life, performance and reliability of the feed pump in the Rankine-Rankine and the solvent pump the the absorption systems, which pump a low mass flow over a large pressure ratio, often with liquids that cause swelling or leaching of the plastics and rubbers used in such pumps. The complete engine-compressor assembly in the free piston Stirling-Rankine and Rankine-Rankine units is likely to be their major source of unreliability, and the servicing of such units could be difficult in the

System	Service interval+	Service requirements	Annual++ service cost (DM/p.a.)
sorption	annual	clean and set burner; check charge level, fans, solvent pump and controls; clean air source evaporator	100 to 150
Rankine-Rankine	annual	clean and set burner; check charge level, feed pump, controls and fans; clean air source evaporator	100 to 150
Stirling-Rankine	annual	clean and set burner; check charge level; check controls and fans; clean air source evaporator	100 to 150
Brayton-Rankine	annual	clean and set burner; check charge level; check controls, fans and airfilter; clean air source evaporator and recuperator	100 to 150
Otto-Rankine	semi-annual	change oil and oil filter; change spark-plugs; set valves; change air filter; clean exhaust heat exchanger; check charge level; clean air source evaporator; check controls and fans	500
Diesel-Rankine	semi-annual	as Otto-Rankine, but excluding spark-plugs	500

+ 2000 hr heating year
++ costs estimated at DM 50 (travel)
DM 50/hour (labour)
includes back-up, not including recharging with refrigerant or working gas or replacing parts other than specified as necessary

Table 3-10 .: Service and maintenance requirements

field. On the other hand the complete unit could be easily exchanged.

For all systems there are two possible methods of organising service and maintenance for the domestic and commercial sectors.

1. Break-down service: this is common practice with conventional boiler systems, but depends on the manufacturer or installer having a 24 hour service network and a stock of emergency spares. This system lends itself more to the use of complete replacement units (engine, compressor units etc.) and so involves greater risks and higher service costs, but only in the event of a breakdown.

2. Contract service system: this arrangement requires the manufacturer, installer or utility (gas, electric or oil supplier) to offer the user a contract for annual or biannual servicing of the system and a 24 hour emergency call-out service. The customer pays an annual fee for the service contract, which does not include parts and which provides for the annual service visit and emergency system. This method has considerable benefits for the customer (particularly security), while for the supplier the number of emergency call-outs are minimized and he can by his approach ensure better reliability and a good reputation for the product. This type of arrangement is considered necessary for the engine driven systems, which without regular maintenance by skilled personnel could prove unreliable enough to spoil the developing market.

3.4.5. Environmental Effects

The environmental effects of the use of the advanced heat pump systems have to be considered since their noise and vibration, emissions etc. could place severe restrictions on their use. Table 3-11 shows the possible problems that may be encountered with their widespread use, from which it can be seen that the

System	Local effects			Larger scale effects	Malfunctions	Restrictions
	emission	noise	vibration			
electric Rankine	none	≃35 dBA low frequency	low	SO_2 from coal, or nuclear waste disposal; large-scale waste heat disposal; peak demand on electricity supply; little energy saving potential		bivalent alternative with oil; back-up favoured by utilities; local restrictions on noise from air source systems
Otto-Rankine	CO NO_x (high)	≃50 dBA low frequency	moderate; needs heavy base and vibration isolation	noise nuisance to neighbours from air inlet and exhaust; high peak demand on gas network	possible production of phosgene in exhaust	local noise control regulations
Diesel-Rankine	CO NO_x SO_2 (high) + soot	≃50 dBA	moderate/high; needs heavy base and vibration isolation	SO_2 in local environment; noise nuisance to neighbours from air inlet and exhaust; usually requires oil; little or no import saving	possible production of phosgene in exhaust	local noise control regulations
Stirling-Rankine (free piston)	CO NO_x (low)	≃50 dBA low frequency	*high, needs heavy* base and vibration isolation	*possible noise nuisance; possible* peak demand on gas network	possible production of phosgene in exhaust	local noise control regulations
Stirling-Rankine (kinematic)	CO NO_x (low)	≤50 dBA middle frequency	low	possible peak demand on gas network	possible production of phosgene in exhaust	
Rankine-Rankine (turbine)	CO NO_x	30/40 dBA low/ high frequency	low	possible peak demand on gas network	*possible production of* phosgene in exhause	
Rankine-Rankine (free piston)	CO NO_x (low)	≃50 dBA low frequency	high, needs heavy base and vibration isolation	possible noise nuisance; possible high peak demand on gas network	possible production of phosgene in exhaust	local noise control regualtions
Sorption	CO NO_x (low)	30/40 dBA low/ middle frequency	low	possible peak demand on gas network	NH_3 emission locally or possible phosgene production in exhaust	2.5 kg NH_3 max. permiss. charge
Brayton-Rankine	CO NO_x (low)	≃50 dBA *high frequency*	low	possible noise nuisance	possible production of phosgene in exhaust	local noise control regulations

Table 3-11.: Environmental and social effects of the use of heat pumps

external combustion systems (Brayton-Rankine, Rankine-Rankine, Stirling-Rankine and sorption) are the best, and of these the Rankine-Rankine (turbine) sorption (organic) and kinematic Stirling-Rankine have the least detrimental points.

An additional environmental problem is that associated with the possible leak of secondary fluids (such as ethylene glycol, brine or fluorocarbons if direct expansion coils are used) into the ground or ground water when either the earth or ground water are used as heat source. This could result in pollution of either the soil itself, or more importantly drinking water supplies where the ground water is used for this purpose.

3.5. Applications

3.5.1. Domestic

The potential energy saving has been the stimulus for the development of fossil fuel driven heat pumps for domestic heating, and for this application they have to compete against conventional boilers and furnaces and also against electric heat pumps (Electric-Rankine). Advanced heat pumps have the advantages over the latter of

- higher primary energy saving (see Fig. 3-4)
- lower peak energy demand (see Table 3-5)
- greater controllability (see Table 3-3)
- less sensitivity to source temperature fluctuations (see Table 3-2)
- less sensitivity to frosting (see Table 3-2).

The choice of possible systems is wide, and a selection of the most suitable systems for this application has to be based on many other factors as well as those listed above. Table 3-12 gives an analysis of the competing advanced systems (including an advanced electric driven system) for air source, water sink and 20 kW maximum heat demand against many criteria. These have been grouped under four main headings (performance, design, operation and first cost). The comparison is based on the Case Study data (see Appendix), and is a much simplified overview of the characteristics of each system. In particular the first cost data can be misleading, since for some systems the costs are target first costs indicated by the developers i.e. those which they foresee as economically attractive (Rankine-Rankine, Stirling-Rankine, Diesel-Rankine and Absorption), while others are based on present selling costs (Electric-Rankine and Otto-Rankine). In addition, German first cost estimates are often considerably higher than those from other countries.

Fuel	Electric	Gas / Oil						
		External Combustion				Internal Combustion		
Drive	Motor	Rankine (Turbine)	Stirling (Free piston)	-	-	Otto	Diesel	
Heat Pump	Rankine	Rankine	Rankine	Absorption	Resorption	Rankine	Rankine	Comments
PERFORMANCE								
COP_g	2,5 (0,75)	1,1 - 1,2	1,3 - 1,4	1,3 - 1,2		1,1	1,3 - 1,4	
Sensitivity to T_s	high	low	medium	medium		medium	medium	
SPF_g	2,7 (0,81)	1,2	1,5	1,2		1,1	1,5	
Sensitivity to frosting	high (3-5%)	low (1-2%)	medium (∿2%)	low (1-2%)	unknown	low (1-2%)	medium (∿ 2%)	COP_g at T_s = 0°C
T_s (°C) at COP_g = 1	+2 (COP_g = 3,3)	∿ - 10	≲ - 10	- 4		- 4	≲ - 10	T_H = 45°C. For
SPF_{eff}	2,5 (0,75)	1,1 - 1,2	∿ 1,3	1,1 - 1,2		∿ 1,1	1,3	electric system () refers to
Prime energy saving	0	25 - 30 %	40 - 45 %	25 - 30 %		20 - 25 %	40 - 45 %	electricity efficiency of 0,3
DESIGN								
Size (h.p. only)(m^3)	0,5 - 0,7	≲ 1	0,9	0,8 - 1	> 1	1,3	1,2	
Weight (h.p. only (kg)	150 - 200	∿ 150	∿ 200	200	∿ 200	175	185	
Back-up	50%: external	50%: internal	40%: external	100%: internal	100%: internal	60%: external	66%: external	
Control	possible speed (2:1) + on/off	speed + on/off	speed (2:1) + on/off	feed pump speed + on/off	feed pump speed + on/off	speed (2:1) + on/off	speed (2:1) + on/off	() Range of speed control
Safety: fire	low	low	low	NH_3: high; other: low	NH_3: high; other: low	low	low	
toxicity	low	low	low	NH_3: high other: low	NH_3: high other: low	low	low	
OPERATION								
Controlability	low	medium	high	medium	medium	high	high	
Reliability	medium	high	medium	high	high	low	low	
Service needs	medium	low	high	low	low	high	high	
System live (yrs)	10/15	15	15/20	15/20	15/20	10/15	10/15	
Noise and Vibration	medium	low	medium	low	low	high	high	
Pollution	high: district	medium: local	medium: local	medium: local	medium: local	medium: local	high: local	
FIRST COST (DM)								
Heat Pump	8000/-	-/3200	-/3000	8000/4000	8000+/4000+	14,9000/-	-/9,000	Relative Cost is total system cost divided by total cost of gas fired boiler system (est DM 12,000) First figure is present cost, second is target cost.
Back - up	2000	-	2000	-	-	2500	2500	
Radiators etc	9000	9000	9000	9000	9000	9000	9000	
Total	19,000	12,200	14,000	17,000/13,000	?	26,400/-	-/20,500	
Relative Cost	1,6/-	-/1,0	-/1,2	1,4/1,1	1,4/1,1 +	2,2/-	-/1,7	

1) Air Source, water distribution system, 20 kW heat load at -15°C, heat pump sized for 0°C ambient, climate North Germany

Table 3 - 12: Comparison of advanced heat pumps for domestic heating

The approximate order of merit under the six main headings are given in Table 3-13.

criteria	best	middle	worst
performance	Diesel-Rankine Stirling-Rankine	absorption Otto-Rankine Rankine-Rankine resorption	electric-Rankine
design	absorption resorption Rankine-Rankine	electric-Rankine Stirling-Rankine	Diesel-Rankine Otto-Rankine
operation	absorption resorption Rankine-Rankine	electric-Rankine Stirling-Rankine	Diesel-Rankine Otto-Rankine
first cost	absorption+ Rankine-Rankine+ Stirling-Rankine+	electric-Rankine+ Diesel-Rankine+ resorption	Otto-Rankine
energy saving++	Diesel-Rankine Stirling-Rankine	absorption Otto-Rankine Rankine-Rankine	electric-Rankine
oil substitution++	absorption Otto-Rankine Rankine-Rankine Stirling-Rankine	electric-Rankine	Diesel-Rankine

+ based on target costs

++ against oil fired boiler; all systems bivalent-parallel and gas-fired back-up except Diesel and electric-Rankine which have oil fired back-up; electricity from non-oil fired plant

Table 3-13: Order of merit of systems for domestic heating (from Table 3-12)

It is clear from Table 3-13 that the system with the best performance are not those with the best characteristics in the other respects. In particular the Diesel-Rankine system, with the best performance, scores badly in other respects. The external combustion systems offer the greatest potential, judged on all the aspects considered, for this application while the scope for their further development is considerable. These systems are the absorption, Rankine-Rankine and Stirling-Rankine systems, and the resorption and the Ericsson-Ericsson systems. The first three sys-

tems are at present at an advanced stage of development, though they are not due for commercialisation until 1983/1985. Much work is therefore still to be done before such systems become available. For their successful application to domestic heating the following areas have been identified as critical in their development

Absorption :
1. achieving design performance objectives
2. replacement of NH_3/H_2O by safer fluids with similar or better performance characteristics
3. reduction in size and first cost
4. development of high performance low cost solvent pumps
5. formulating of design methods for efficient generators and absorbers
6. development of control methods to give continuous output variation over a wide range with minimal reduction in performance
7. extension of application range to larger and smaller unit sizes.

Rankine-Rankine :
1. achieving design performance objectives
2. proving longevity and reliability of expander-compressor unit (turbine and free-piston units)
3. development of control methods to continuous output variation over a wide range with minimal reduction in performance
4. development of high performance low cost feed pumps
5. extension of application range to air distribution, summer cooling and larger unit sizes (smaller may not be possible)

Stirling-Rankine :
1. achieving design performance objectives
2. proving longevity and reliability of free piston expander-compressor unit

3. development of control methods to give continuous output variation over a wide range with minimal reduction in performance
4. extension of application range to water distribution and larger and smaller unit sizes.

The remaining systems (resorption and Ericsson-Ericsson) are at earlier stages of development and consequently must meet not only the relevant objectives listed above but their development also needs to be accelerated so that they are available in time to compete for the market against the other systems. In view of their potential for better performance than the other external combustion systems the acceleration of their development is considered desirable. In the context of air as a distribution medium and the possible cooling application of the systems (these two requirements are usually linked for domestic applications) the comparison contained in Table 3-12 needs to be conditioned by the relative cooling performance shown in Table 3-4 On this basis the Stirling-Rankine system shows the greatest potential, both in terms of capacity and COP_{eff}. For climates with a small cooling requirement the absorption and Rankine-Rankine systems may have some potential, but look uncompetitive with the already well established reversible electric heat pump.

3.5.2. Commercial

The trend since 1974 in the commercial market (offices, schools, leisure centres, public buildings, etc.) has been towards the use of heat pumps to replace air-conditioners and boilers. These have in the main been electric driven; the notable exceptions being for swimming pool and leisure centre heating and cooling for which Otto engine driven system have become well established. It is clear that on the basis of performance and cost the Otto and Diesel systems are well suited to swimming pool and leisure centres and also to the air-conditioning and heating of large commercial buildings,

particularly when there is a requirement for simultaneous power generation. These small combined heat and power plants, particularly Diesel engine driven ones, would appear logical for the providing of combined services for large public buildings and other building complexes. In the latter case the engine is used to generate power only and the engine waste heat is used for the heating of the main building. The power can then be used to drive electric heat pumps for heating remote buildings.

Smaller commercial buildings requiring heating (and often cooling) are usually too small to justify an engine driven system and for these the electric motor driven systems predominate. The earlier use of absorption systems for this use declined up until 1974 because of their higher capital costs and generally poor efficiency. In this case there would appear justification for the development of engine driven systems and also the external combustion systems (Brayton-, Rankine- and Stirling-Rankine) both to reduce energy consumption and to permit substitution of petroleum based fuels. The Brayton and Stirling engine driven systems are more suitable for this application because of their high cooling capacity and COP, though in northern climates the heating capacity and COP is more significant. The high noise level of most engine driven systems combined with localised pollution from oil driven systems makes the development of systems such as the kinematic Stirling-Rankine desirable for this application. The major problem with this systems is the present high first cost.

3.5.3. District Heating

In northern Europe district heating is a well established practice, often as part of a larger combined heat and power scheme. However, the majority of district heating systems are smaller than that needed to justify the investment associated with a combined heat and power plant. They therefore are based on a central hot water or steam boiler station providing heating to the neighbouring domestic, commercial and occasionally industrial premises. For fuel

most of these boilers use oil, sometimes combined with garbage incineration, and to a much lesser extent coal and natural gas. The substitution of oil by other fuels in such district heating schemes can be achieved by using gas, or by burning such less environmentally desirable fuels as coal, lignite, garbage, industrial waste etc. However, for a given heat demand a greater quantity of these fuels needs to be burnt than oil because of their lower calorific value and the lower efficiency with which they can be burnt. Further there are problems with pollution control, fuel handling and waste disposal. In the case of garbage and waste there is usually an unsufficient supply to meet the heat demand and so additional oil has still to be burnt in peak periods.

The problems described above in converting district heating schemes from oil can be substantially reduced by the use of heat pumps to meet at least a proportion of the heat load. Since direct fired large heat pumps are capable of having SPF's in the range 1.7 to 2 they can be used to reduce the energy demand of the district heating systems by 40 % to 50 %. When such systems are operated on wastes, such as garbage etc., then the energy requirement of the district heating scheme can be better matched by the supply and the associated problems reduced in scale.

A number of systems are possible

1. External combustion systems (such as sorption heat pumps and compression systems with Stirling, Brayton or Rankine drive cycles) can have a multifuel capability and can be combined with advanced burning techniques such as the fluid-bed boiler. The boiler can be capable of both powering the heat pump and providing the additional peak demand.

2. Internal combustion systems, such as Diesel and Otto engine driven compression heat pumps. These require natural gas, liquified petroleum gas, oil or more usefully heavy fuel oil, coal dust or combustible waste gases as fuel.

3. Electric motor driven compression systems. When the installation is near a condensing power station the condenser cooling water can be used directly (or indirectly from a warmed river or lake) as heat source. This in effect creates a small combined heat and power plant but with much greater flexibility of operation and requiring no modification to the power plant. Older coal burning power stations placed near city centers and which are already amortised and are low in the operating order can by this method have their useful life extended and their thermodynamic efficiency improved.

Table 3-14 lists the known district heating systems and shows a diversity of sources and fuels but with a concentration on internal combustion engine drives. Thermal outputs lie in the range 163 kW to 27 MW.

The most comprehensive study of heat pumps for district heating /133/ has shown that the heavy fuel oil Diesel engine driven system has the greatest fuel saving potential and best economic prospects when replacing a boiler using the same fuel. The most economic size of heat pump is with thermal outputs in the range 5 MW to 20 MW. Below 2.5 MW heavy fuel oil cannot be used while above 40 MW it is more economic to use coal fired combined heat and power plant.

The weakness of using Diesel engines for district heating heat pumps is that, although there can be a 50 % reduction in oil consumption, there is still a dependance on imported petroleum products. Dual-fuel operation (e.g. gas and oil) would tend to alleviate this weakness, but results in greater system complication. The Diesel engine driven heat pump for district heating would therefore suffer from the possible future oil price inflation (relative to other fuels) and possible shortages of fuel due to political factors. The gas engine driven systems, though less efficient, would not suffer this problem to the same extent especially when

Country	Fuel	Drive	heat Pump	Source	T_s (°C)	Thermal output (MW)	COP^*_{eff}	D.H. scheme	T_H (°C)	Back-up	Stage
Denmark	fuel oil	Diesel	compression	sewage	+10 to +16	10.2	2	new	85	oil boiler	construction
Denmark	Diesel oil	Diesel	compression	horiz.ground	+ 2 to +16	0.4	1.6	existing	45/?	garbage boiler (+oil)	construction
Germany	mine gas	Diesel or Otto	compression	coalmine cooling	+20	25	1.3	existing	90	oil boiler	planning
Germany	coke gas	steam	absorption	waste heat	+40 to +60	3.0	1.3	existing	90	oil boiler	construction
Japan	kerosene	-	absorption	waste water	+30 to +60	8.75	1.3	not known	90/150	oil boiler	proposed
Japan	electricity	electric motor	comp/abs.	waste water	+30 to +60	17.4	0.9	not known	90/150	oil boiler	proposed
Netherlands	natural gas	Otto	compression	ground water	+10	0.163	2	new	40	gas boiler	construction
Norway	electricity	electric motor	compression	sea water	+ 5 to + 6	27	3.0	new	55/75	oil boiler	proposed
Sweden	Diesel oil	Diesel	compression	air	-25 to +20	0.816	1.4	existing	80	oil boiler	planned
Switzerland	wood gas	Otto	compression	air, solar	-16 to +16	~ 0.25	not known	new	90	oil boiler	proposed
Switzerland	natural gas	Otto	compression	lake water	+ 3 to +15	0.861	1.5	existing	76	oil boiler	construction

* heat pump only, primary energy related

Table 3-14: Existing or proposed heat pump district heating schemes

locally produced gas is available. However, no fuel substitution is possible with such systems, making them vulnerable over their projected lifespan (15-20 years) to fuel price inflation and shortages. The external combustion systems, which can be designed as multifuel or can be later adapted, could possibly be alternatives. Gas turbines (Brayton cycle) cannot be operated successfully on coal and so the most hopeful systems from this aspect would appear at present to be multistage sorption systems, Stirling-Rankine and Rankine-Rankine systems. For the latter steam can be used for the power cycle, though this would be only practical for large systems for which combined heat and power would be more likely to be economically attractive. For smaller systems organic fluids are possible, but this results in lower drive cycle efficiency and larger heat exchangers than for steam. Stirling engines large enough for district heating applications are not available, though the kinematic Stirling engine has the potential advantages of high efficiency, reliability, multi-fuel capability and low noise. Thus heat pumps for district heating can be ranked (Table 3.15.) in order of merit for performance (i.e. COP_{eff} based on existing district heating distribution temperatures of +80/+90°C and source temperatures of +10/+20°C and thermal outputs in the range 400 kW to 1 MW); controllability, first cost, multifuel capability, energy saving potential and oil substitution. A more detailed comparison is not at present possible, since at present only Diesel, Otto and absorption systems have been considered for this application. In addition, a detailed comparison has also to be based on the characteristics of existing or planned district heating schemes. For example, new schemes could be based on water distribution temperatures of +40/+50°C so that a monovalent heat pump with floor warming can be used. Alternatively, in new or existing schemes, bivalent-parallel operation can be utilised to give the higher normal distribution temperatures at peak periods. In these circumstances the advanced systems, e.g. sorption, Rankine-Rankine and Stirling-Rankine, may be considered. Since these have a multifuel capability the possibility would then exist to base new district heating systems on heat pumps to both save energy and avoid use of

petroleum based fuels or natural gas.

criteria	best	middle	worst
performance+	Diesel-Rankine	Otto-Rankine Stirling-Rankine	absorption Brayton-Rankine Rankine-Rankine
controlla-bility	absorption Diesel-Rankine Otto-Rankine Stirling-Rankine	Brayton-Rankine	Rankine-Rankine
first cost	Diesel-Rankine Otto-Rankine	absorption Brayton-Rankine Rankine-Rankine	Stirling-Rankine
multifuel capability	absorption Rankine-Rankine Stirling-Rankine	Brayton-Rankine Diesel-Rankine	Otto-Rankine
energy saving++	Diesel-Rankine	Otto-Rankine Stirling-Rankine	absorption Rankine-Rankine Brayton-Rankine
oil sub-stitution++	absorption Brayton-Rankine Otto-Rankine Rankine-Rankine Stirling-Rankine	electric-Rankine	Diesel-Rankine

+ for water distribution temperature +80/+90°C, source temperature +10/+20°C.

++ see Table 3-13 for assumptions.

Table 3-15: Ranking of heat pump systems for district heating

3.5.4. Industrial

The assessment of the potential of the various advanced heat pumps in industry is made difficult by the diversity of possible applications. However, there are a number of main areas of application which have been identified and characterised, and for some of these electric heat pumps have been developed. From this characterisation the most suitable advanced heat pumps can be identified.

The largest potential markets for heat pumps have previously been identified as heat recovery from refrigeration and cooling plant for space or water heating, the coupling of heating and cooling processes within the same plant, and drying. Since refrigeration systems generally use electric motor drives, there is little potential market in this application for advanced heat pumps (which are also characterised by a much higher heating capacity than cooling capacity). Simultaneous heating and cooling is an extremely diverse market in which it is not possible to identify potential applications for standard equipment and so each application tends to require an individually designed piece of equipment. At present electric-Rankine systems are being used to couple together processes where the heat requirement is below 70°C, and sometimes using R114 as work fluid for up to 100°C. The major advantage in the possible use of direct fired systems for these applications would be the higher performance (COP_{eff}) available (over the same temperature difference as the electric-Rankine systems) or the extension of the temperature range of the heat recovered to 80/120°C.

However, the distinguishing characteristics of most industrial applications (over those in the commercial and domestic areas) are the longer operating hours (2000 to 8000 p.a.),the necessity for high reliability and low maintenance, and usually the ability to operate in a dirty environment. Under these aspects the most efficient systems (Diesel-Rankine and Otto-Rankine) have the worst characteristics, while the inherently most reliable (the absorption system) is limited with present working pairs to operate with about

90°C condensing temperature. Research and development into the other systems has not even begun.

The usual requirement is to cool or to recover heat from processes in the temperature range 30°C to 80°C and to provide hot water at 80°C to 120°C or low pressure steam at 100°C to 140°C (1 to 3.5 bar). The cooling water heat from Diesel or Otto engines cannot be recovered at the upper end of this temperature range so their efficiency is considerably reduced. The only systems that would appear feasible for these applications would be the sorption (with multistaging and new working pairs), the Rankine-Rahkine, Brayton-Rankine and Stirling-Rankine systems.

An aspect of heat recovery in industry, which applies to heat pumps as well as other possible systems, is the fouling of the heat exchangers (usually the evaporator) by the contaminated liquors or exhaust air which are used as heat source.

Convective drying in industry is carried out with temperatures in the range 40°C to 400°C, with the majority of applications in the range 80°C to 200°C. For the lower end of this range (up to 100°C) the engine driven systems would appear most attractive (i.e. Diesel and Otto), particularly for larger applications with continuous operation /107/. At higher temperatures there are three possibilities, external combustion driven- Rankine systems, the Brayton-Joule or electric-Joule system (the heat pump side using the dryer exhaust air as working fluid) or replacing the dryer airstream by steam. The latter solution is already being pursued with electric drives /91/ while the electric-Joule system has been proposed by several workers but has not yet been fully explored for this application /122, 123, 124/.

Summarising, the industrial applications for advanced heat pumps (including electric drives) would appear to lie in the area of heat recovery, the coupling of simultaneous processes and convective drying. All these applications require the development of

suitable work fluids, processes and equipment to allow operation at higher temperatures with high performance and reliability and often with contaminated heat sources.

4. Market Study

4.1. Technical Approach

National market study teams estimated the implementation potential for advanced heat pumps from 1980 to 2000 using a technical approach prepared by RPA. Although AHP systems will also have a significant market in commercial/industrial applications, only installations in single-family (SF) and multi-family residential buildings (MF) were considered in this study. Furthermore, national teams assumed that the entire housing stock in each country will require the following standard sizes of AHP installations:

SF and MFI	MFII
3-4 kW	50 kW
10-12 kW	100 kW
15-20 kW	400 kW
	2 MW (district heating)

The five major steps in the technical approach were:

1. Estimate H/C installations through 2000
2. Estimate energy requirements and operating costs of AHPs and conventional systems
3. Rank AHP and conventional systems according to decision criteria
4. Determine economic potential of AHP systems
5. Determine implementation potential.

These steps and the necessary data inputs required to complete them are outlined in Figure 4-1. Each AHP system considered in the market study is described in Chapter 3 of Volume 3, which includes detailed technical profiles in Appendix B.

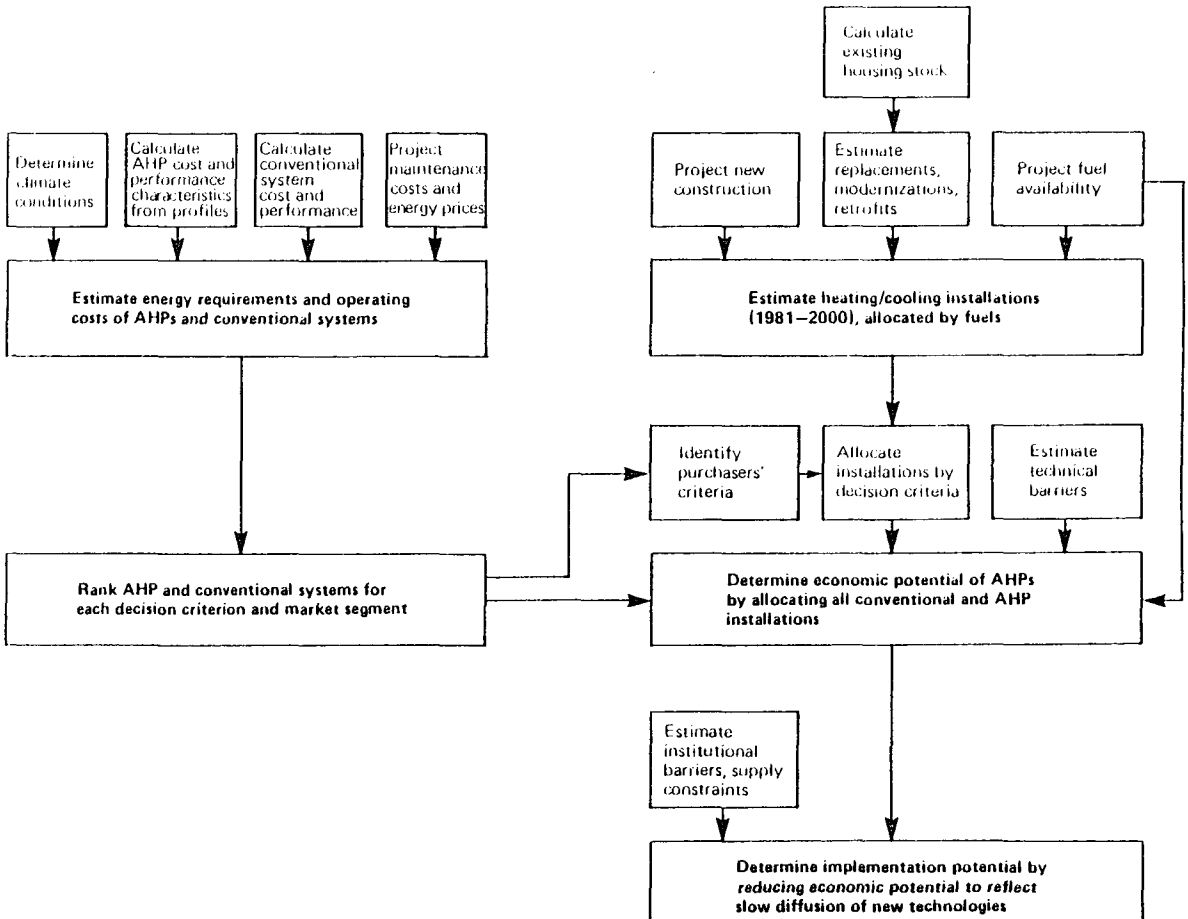
4.1.1. Estimate H/C Installations Through 2000

In the initial part of the study, national teams estimated the annual number of all heating and cooling installations between 1980 and 2000. These estimates indicate the size of the annual market in which AHP systems will compete with conventional systems (i.e., both conventional heat pump systems and conventional heating and cooling installations such as furnaces).

To complete the first part of the study, national teams first collected detailed inventories of existing housing stock and current installations of

Figure 4-1

Technical Approach for AHP Market Study



types of heating and cooling systems. Such information was necessary to estimate the potential for AHP installations in existing stock. Four categories of heating systems were defined for both single- and multifamily buildings:

Central air: Central heating plant serving either the individual housing unit or the entire building, with forced-air heat distribution; the most common system in North America.

Central water: Central heating plant with heat distribution by hot water; the most common system in Europe.

Space I: Electric baseboard (or other heating built into the building structure) and ductless furnaces.

Space II: Stoves or other room heating units (i.e., units not built into building structure), common in Japan and in older buildings in Europe.

As a result of these definitions, multifamily housing stock was further categorized by types of heating systems:

Multifamily I (MFI): Apartment buildings with individual heating systems (i.e., electric heating or stoves or other room-heating units)

Multifamily II (MFII): Apartment buildings with central heating systems (i.e., central water or air)

The inventory of existing housing stock revealed that only the United States and Japan have significant demand for residential cooling systems. All cooling systems considered in this study were categorized as either central air or Space II systems in both single- and multifamily buildings.

The existing stock was also categorized by the fuel used for heating, a key factor in determining the heating system selected for replacement, modernization, and retrofit installations. Fuel types include:

- Oil/Diesel
- Natural gas
- Electricity
- Coal
- Other (district heating, block central heating).

National teams then estimated the annual amount of new construction between 1980 and 2000, classifying single- and multifamily buildings by type of heating system as for the existing stock. Teams also made assumptions about the fuels available for heating systems in new housing, particularly the availability of natural gas.

In most countries, the largest market for heating and cooling installations is not in new construction but rather in the replacement, modernization, and retrofit of systems in the existing housing stock. Each type of installation, which is defined below, implies a different set of options for system purchasers:

- Replacement: The existing heating system is worn out and must be replaced. The distribution system (i.e., central air or water) is not changed.
- Modernization: The existing heating system is replaced to upgrade living conditions. The distribution system may also be changed. Modernizations will be most common in Europe as Space II systems are replaced with central water.
- Retrofit: The existing heating system, though not worn out, is replaced with a new system for greater energy efficiency and lower operating costs.

National market teams made varying assumptions about the percentage of the existing stock that would require installations of each type in the 1980-2000 period. Most teams assumed that installations resulting from replacement, modernization, or retrofit would continue to use the same heating fuel as existing systems.

The annual number of installations was calculated as the sum of the installations derived from new construction and those derived from replacement, modernization, and retrofit.

4.1.2. Estimate Energy Requirements and Operating Costs of AHPs and Conventional Systems

The next step in the analysis was to develop operating costs and capital costs for the heating and cooling systems that will compete in each market segment. Operating costs depend on the system's energy efficiency, which determines the amount of energy necessary to meet expected building heating and cooling loads. Unlike conventional systems whose efficiency is independent of climate conditions, a heat pump's efficiency (or coefficient of performance) varies with outside temperature, particularly if its heat source is outside air.

National teams calculated the energy requirements of heat pump systems by using aggregated climatic data for their countries (some countries, such as Canada, Japan, and the United States, performed the analysis by region). The annual hours of required heating (and/or cooling) were grouped in bins according to the prevailing outside temperatures. The coefficient of performance (COP) of the heat pump at each temperature was then taken from the specification or profile of heat pump performance and cost provided from the technical study. Analysts then calculated the energy consumption of the system for each bin of hours and added them up to derive the annual energy consumption of the heat pump system. The calculation methodology also covered estimation of the consumption of heat pump back-up systems. At the end of the calculation, analysts estimated energy consumption of the total heat pump system over the heating and cooling seasons and the seasonal performance factors. Similar calculations were made for conventional systems.

Analysts then applied estimates of expected future fuel costs to these energy consumption values to calculate energy cost; assumed maintenance and service costs were added to arrive at total annual operating costs for each system. The analysis was carried out in real money (no inflation). But, since fuel costs tend to escalate in real terms over the period, separate estimates of operating costs were made for each five year period from 1980 to 2000.

The first cost data used in the market study are in some cases uncertain and not very homogeneous, since for some systems the costs are target first costs indicated by the developers, i.e., those that they foresee as economically attractive (Rankine-Rankine, Stirling-Rankine, Diesel-Rankine, and Absorption), while others are based on present selling costs (Electric-Rankine and Otto-Rankine). In addition, German first cost estimates are often considerably higher than those from other countries.

Using capital costs in Deutschmarks taken from the heat pump profiles, national teams estimated costs for competing AHPs and conventional systems. Deutschmarks were also used to calculate the capital and maintenance costs reported in the heat pump profiles in cases when the original cost information came from a country other than Germany.

Most countries estimated heating/cooling installations by fuel type, especially for the replacement, modernization, and retrofit installations. Some countries allocated installations by fuels later in the analysis, incorporating the choice of fuel into the final selection of heating systems.

4.1.3. Rank AHP and Conventional Systems by Decision Criteria

The next step in the analysis was to introduce decision makers' criteria for selecting a heating system. National teams estimated the percentage of decision makers in each market segment that would choose among AHP profiles and competing conventional systems according to such criteria as:

- First cost
- Payback period
- Annual energy or operating cost
- Life-cycle cost.

The heating/cooling installations within a market segment were then allocated among the criteria that would apply according to these percentages.

At this point in the analysis, each national team had identified a number of market segments, characterized by:

- Number of installations in each 5-year period between 1981 and 2000
- Type of building (single-family, multifamily I, multifamily II)

- Type of installation (new construction, replacement, modernization, retrofit)
- Other distinguishing characteristics such as type of heat distribution (usually water)
- Decision criteria
- AHP and conventional systems competing in the segment.

National teams then calculated the values of the decision criteria for all the systems, both advanced heat pump and conventional, that would compete in each market segment. The systems were then ranked according to each criteria.

4.1.4. Determine Economic Potential of AHP Systems

The allocation of installations to competing systems, called the economic market potential, was performed in different ways in different countries. In most of the countries, all the installations in the market segment were allocated to that system (either advanced heat pump or conventional), which ranked first on the associated decision criterion. In many cases, however, technical limitations (such as natural gas availability or restrictions on ground heat sources) prevented this system from taking the entire market. Where this occurred, as many installations were given to the first-ranking system as technical limits would allow, and the rest given to the second-ranking system.

Some countries allocated all their installations to fuels before they allocated them to systems; most countries used this method for the replacement, modernization, retrofit segments. As a result, in these markets, competition for the market occurred only within fuels. Other countries allocated their installations to fuels and heating/cooling systems at the same time; often, countries allocated systems to new construction segments in this way. In these markets, competition among systems occurred across fuels, and more systems were able to compete. This more comprehensive analysis was done most often in those countries which analyzed a large number of systems and market segments.

4.1.5. Determine Implementation Potential

In the last step of the technical approach, the national teams estimated the implementation potential for advanced heat pumps. The implementation potential is the economic potential reduced to reflect a host of factors that slow the introduction of a new technology, including the inherent reluctance of consumers to try a new product (a reluctance not fully captured in the decision criteria), institutional barriers, supply constraints, and the like. The technical approach specified high, low, and nominal market penetration rates taken from past experience with similar products.

One important limitation in this approach can result in the underestimation of the market potential for some technologies, and the attribution of overestimated market shares to others. Except for AHP systems with an exceptionally large market potential or very little potential, no definitive conclusion should be drawn without an analysis of the sensitivity of the results to changes in costs, performances, or other factors influencing each system's economic performance.

4.2. Market Analyses of AHP Systems

Although most of the countries participating in the AHP market study followed the same technical approach, they analyzed different AHP systems in different markets and made divergent assumptions as required by specific conditions in each country. The definitions and assumptions that were used in developing the market estimates are explained in this chapter; the similarities and contrasts help illuminate the wide differences in the market potential for AHP systems in the countries covered in the study.

National teams estimated markets for heating and cooling installations in three types of buildings:

- Single-family (SF): individual houses
- Multifamily I (MFI): apartment buildings with individually heated or cooled apartments
- Multifamily II (MFII): apartment buildings with a single (building central) heating/cooling plant.

Heating distribution systems installed in existing or new buildings were limited to four types:

- Central air: central heating and cooling system serving any of the three types of buildings with forced hot and cold air distribution; the most common system in the United States and Canada.
- Central water: central heating plant with heat distribution by hot water; used in all three building types; the most common system in Europe.
- Space I: electric baseboard (or other electric heating systems built into the building structure) and ductless furnaces; used only in SF and MFI buildings
- Space II: stoves or other room heating units (i.e., not built into the building structure); common in Japan and in older buildings in Europe; used only in SF and in MFI buildings.

Most AHP systems will be installed in central heating systems. (Japan was perhaps the only country that considered room or space II installations.) Denmark, Japan, Sweden, and Switzerland also assessed the market potential for large AHPs installed in block central heating systems suppling several buildings or in district heating systems supplying urban districts.

In addition to the housing stock and types of heating and cooling installations, the potential for AHPs is conditioned by the price and availability of fuels. Each country made its own assumptions about future energy prices and their escalation rates over the period covered by the study (see Table 4-1).

Fuel prices vary considerably among countries. Canada and the United States have the lowest oil prices, Denmark, Austria, and Switzerland the highest (the latter three countries all show substantial heat pump markets). Natural gas prices are lower, in general, than oil prices for the same energy equivalent of fuel, reflecting perhaps the inertia of regulated gas prices and the desire of some governments to limit oil usage through expanded sales of gas. The Netherlands and the United Kingdom, bordering the North Sea, predicted very low gas prices, one factor that limits AHP market potential in these countries. Electricity prices are highest in the Netherlands, Austria, Japan, and Denmark and, lowest in Sweden (where AHPs have a considerable market share) and Canada.

Because gas prices are often lower than oil prices, and gas boilers are less expensive than most other heating systems, the market share for AHP systems using gas depends more on the availability of gas than on its price. Thus, the extent of the natural gas network is a key assumption in many countries. Several countries, including the United States, Canada, Spain, and Switzerland have entire regions that lack natural gas pipelines. Sweden and Switzerland have little or no gas distribution systems.

Participating countries selected the AHP systems and market segments that they would analyze in the study. Their selections are listed in Table 4-2. This choice amounted to a prejudgment of the systems most likely to succeed in the market, but limited resources forced some participants to consider less than the full number of systems that could in fact compete in their markets. AHPs 2, 13, and 15/18 were selected for analysis by the most countries, and these systems also demonstrate the most significant market potential.

All countries participating in the market study agreed on the estimated technical performance of the 16 AHP systems analyzed, but there was some divergence on the capital costs.

A number of technical constraints limit the market penetration of specific systems, including:

- Space limitations in buildings, especially in replacement/retrofit applications
- Space limitations between buildings, which limit the use of groundwater or soil-source heat pumps; particularly in urban areas
- Difficulty of sizing the unit for both heating and cooling loads

Table 4 - 1

ENERGY PRICE ASSUMPTIONS

Energy Prices (0,01 DM/kwh)	Austria	Belgium	Canada	Denmark	Italy	Germany	Japan	Netherlands	Spain	Sweden	Switzerland	United Kingdom	United States
oil price (1979)	6,5	5,5	2,2	7,0[a]	5,0	5,0	4,5		5,0	4,7[b]	6,2	3,7[c]	2,6
Projected escalation rate (%)	4,5	3,0	3,2	8,0	3,0	5,0	8,0		4,0	2,4[d]	2,5 - 3,5		2,0 - 3,0
Gas price (1979)	6,1	5,5	1,3 - 1,6		3,0	3,7	8,9	4,3	6,0		5,5	3,24	1,4 - 3,1
Projected escalation rate (%)	4,0	3,0	3,2		2,0	5,0	7,0	3,5	3,0		2,0 - 3,0	4,2	1,5 - 4,5
Electricity price, end 1979	15,0	15,0	4,0 - 5,6	14,0		9,0[e]	14,1	17,0	10 - 13,2	7,2	5,0 - 15.0	10,7	5,5 - 12,9
Projected escalation rate (%)	2,5	0	1,4	3,0		2,5	7,0	1,8	3,0	1,2[d]		< 1,0	0 - 2,0

a. Heavy oil

b. 4,5 single-family, 4 multifamily, 5 Diesel oil

c. Estimate for 1990

d. Three scenarios: 4% oil, 2% oil, 2% electricity, 2% oil, 1% electricity

e. Price for operating heat pump only, excludes taxes.

Table 4 - 2

AHP Profiles Analyzed by National Market
Study Teams: Technical and Market Characteristics

AHP Profile	Year of Introduction	Heat Pump Cycle	Drive Type	Fuel	Heat Source	Thermal Output (kW)	Market Segment*
2 a	1985 - 1986	Absorption	Direct	Oil, gas	Ambient air	3 - 2o	SF, MF I
2 b	1985 - 1986	Absorption	Direct	Oil, gas	Ambient air		SF, MF I
3	1981	Absorption	Steam boiler	Oil, gas	Wastewater	> 2,000	MF II, district
4	1982	Rankine	Diesel	Oil	Ambient air	10 - 20	SF, MF I
4 a	1980	Rankine	Diesel	Oil	Soil	> 400	MF II
5	Before 1985	Rankine	Diesel	Oil	Ambient air	100 - 6,000	MF II, district
6	1981	Rankine	Diesel	Oil	Sewage	> 2,000	MF II
7	1980	Rankine	Otto	Gas	Ambient air	10 - 400	SF, MF II
8	1981	Rankine	Otto	Gas	Air / ground / surface water	400	MF II
9	1983	Rankine	Stirling	Oil, gas	Air / ground / surface water	3 - 20	SF, MF I
13	1980	Rankine	Motor	Electricity	Ground	3 - 1,000	SF, MF I, MF II
14	1985	Absorption	Direct	Gas	Ambient air	10 - 20	SF, MF I
15/18	1985	Rankine	Rankine	Oil, gas	Ambient air	10 - 400	SF, MF I, MF II
16	1985	Rankine	Brayton	Oil, gas	Ambient air	50	MF II
19	1982	Absorption	Direct	Kerosene	Wastewater	> 2,000	District
20	1982	Absorption/Rankine	Motor	Electricity	Wastewater	> 2,000	District

* SF - Single family houses
* ML - Individually heated apartment units
ML II - Apartment units in centrally heated buildings
District - District heating

- The high temperature (90/70°C) often required by existing hot water distribution systems
- Climate conditions (cold air temperatures, icing) that render air-source heat pumps unattractive
- High noise-level problems.

In addition, several institutional barriers limit the technical applicability of AHPs, such as environmental restrictions on the use of groundwater as a heat source or safety restrictions on the use of ammonia as a working fluid. Countries considered these constraints in deciding which AHP systems would be able to compete for each market segment. In some cases, the technical constraints represented only partial limitations on the AHP, and they were applied later in the analyses when calculating implementation potentials. Because AHP systems are almost always more expensive to buy and install than conventional competitors, including electric heat pumps, their economic attractiveness depends largely on achieving low operating costs. Low operating costs in turn require a low energy consumption or high energy efficiency, in addition to reasonable maintenance costs.

The energy efficiency of a heat pump is measured by the seasonal performance factor (SPF), the ratio of energy delivered (or extracted in the case of cooling) to energy consumed over the heating or cooling season (either fossil or electricity) by the entire heat pump system, including the back-up system, if any. Because the instantaneous efficiency (the coefficient of performance or COP) varies with climate for many heat sources (e.g., outside air, soil), calculating SPF can be complex. The SPF values for heating are shown in Table 4-3. The countries with the colder, more extreme climates (e.g., Canada, Sweden, United States), show lower SPFs than those with moderate climates (e.g., United Kingdom, Belgium). The Netherlands used COP estimates from revised case studies, instead of cost profiles, used by Austria and the United Kingdom. Therefore, their calculated SPFs are low, which may partly explain why AHPs have no market in that country. The SPFs of AHP 13 (the electric heat pump) are higher than SPFs of systems burning fossil fuels, because they do not reflect electricity generating efficiency.

To calculate measures of economic attractiveness for each AHP system and for each conventional competing system, countries selected measures that reflect the criteria used to select new heating and cooling systems in each market segment. The most common criteria were:

- Initial or first cost: capital cost of the system plus installation cost; the cost of the heat or cold air distribution system was included where the system was not the same for all competing systems. Most countries adjusted costs to reflect government subsidies, if any. Rather than trying to achieve accuracy in total installation cost estimates, which was often impossible because of the very large uncertainty affecting AHP cost estimates, national teams made a special effort to impose consistency on installations cost in each market segment and from one technology to another.

Table 4 - 3

REPRESENTATIVE HEATING SEASON PERFORMANCE FACTORS (SPF) OF ADVANCED HEAT PUMPS

	Small							Medium						Large			
Country	2	4	7	9	13	14	15/18	4a	5	7	8	13	15/18	16	3	5	6
Austria		1,3			2,9	1,3			1,2			3,1					
Belgium	1,25	1,25			3,37		1,29						1,29				
Canada	1,00																
Denmark					2,1			1,59	1,2			2,52					1,9
Germany	1,38	1,39			3,74		1,10						1,18				
Italy	1,3								1,2								
Japan	1,27			1.33	2,14		1,18	1,84				2,14					
Netherlands	1,19		1,35	1,3		0,87	1,01				1,33						
Spain	1,32				3,34							3,36	1,17				
Sweden	1,1				2,76							3,36					0,85
Switzerland	1,158	1,183					1,11								1,42		
United Kingdom	1.32	1,35	0,86	1,15	3,25	1,48	1,25										
United States	1,02	1,232	0,807	1,108	2,381		1,081		1,282				1,084	0,994			

- Life-cycle cost: the capital cost plus the "present value" of the total energy and maintenance costs to be expected over the lifetime of the system.

Countries judged the relative importance of these criteria to decision makers, as shown in Table 4-4. The importance of first cost has been calculated separately from the importance of all other criteria (i.e., life-cycle costs, annual energy costs, and payback period), a key distinction for the market estimates. The technical approach assumes that the demand for heating and cooling installations in a given market segment will be met by the technically feasible conventional or AHP system ranking first on the associated decision criterion. Very often, conventional systems rank first on first cost but behind one or more AHP systems on other, longer-run, criteria. Broadly speaking, those countries with the least preoccupation with first cost (e.g., Denmark, Switzerland, Germany, Sweden) demonstrate good market penetration by heat pumps. Those most interested in first cost (e.g., Italy, the Netherlands, the United States) show smaller markets, at least in terms of market share.

The allocation of heating and cooling installations to conventional and AHP systems according to economic performance and decision criteria gives the economic potential for AHPs, the market that AHP systems would have if these factors alone determined the choice of system. In practice almost all the countries reduced these estimates to take account of other factors that are difficult to capture in the technical limitations or the economic calculations. Such factors included:

- The reluctance of consumers to purchase AHP systems until the technology has become familiar and experienced supply/installation, and service networks are established.

- The reluctance of consumers to take the extra time to install an AHP system for replacement applications. He will be strongly tempted to install another conventional system and return heat to the building as soon as possible.

- Institutional or economic constraints that cannot fully be incorporated in earlier steps of the analysis. Detailed discussion of these is contained in Volume 3.

- Supply constraints on the availability of new AHP systems.

In practice, many of the countries used diffusion or market penetration curves to simulate these effects. The result of applying these additional factors was the implementation potential, the most likely estimate of AHP potential.

Table 4 - 4

IMPORTANCE OF DECISION CRITERIA IN NEW CONSTRUCTION MARKET[a] (%)

Country	Single Family		Multifamily I		Multifamily II	
	Initial Cost	Life-Cycle Cost, Annual Energy Cost, and/or Payback Period	Initial Cost	Life-cycle Cost, Annual Energy Cost, and/or Payback Period	Initial Cost	Life-cycle Cost, Annual Energy Cost, and/or Payback Period
Austria	5o	5o	5o	5o	3o	7o
Belgium	75	25	67	33	2o	80
Canada	25	75	40	60	NA	NA
Denmark[b]	0	100	10	90	10	90
Germany	40	60	NA	NA	70	30
Italy	60	40	60	40	0	100
Japan	75	25	75	25	75	25
Netherlands	40	60	50	50	50	50
Spain[c]	25	75	45	55	35	65
Sweden[b]	45	55	45	55	30	70
Switzerland	0	100	0	100	0	100
United Kingdom	50	50	75	25	NA	NA
United States	80	20	60	40	50	50

a. Same criteria weightings apply in retrofit market, except in Austria, Belgium, Germany, Sweden, and the United States.

b. Criteriy weightings also estimated for block central and district heating.

c. 1996 - 2000

4.3. Market Estimates by AHP System

Twenty-one AHP systems were initially selected for analysis in the market study, based on the interests of the participating countries and on a general awareness of current research developments. Because not enough of the data needed for both the technical and market studies were available for all 21 systems, the number of systems analyzed was reduced to 15 (see Tables 2-1 and 4-5). The following systems were eliminated from the market study:

- Ground-source absorption (AHP 1)
- Solar-powered absorption (AHP 10)
- Gas-powered absorption with solar assist (AHP 11)
- Solar-driven organic Rankine cycle (ORC) (AHP 12)
- Steam-turbine-driven heat pump, with air-heat source (AHP 17)
- Organic Rankine cycle, with seawater heat source (AHP 18).

To ensure consistency between the market study and technical survey, RPA and Battelle agreed on detailed cost and performance data for the remaining systems. Profiles of each system were compiled and distributed to the participants.

Each country then selected system profiles for analysis, matching profiles against its energy situation, available fuels, building stock, and climate. This selection amounted to a preassessment of the system most likely to succeed in each market. However, limited study resources forced some participants to consider less than the full number of systems that could in fact compete in the markets in those countries.

Seven systems appear to have the greatest market potential, and seven have limited or insignificant market shares. In the first part of this chapter, we describe AHP systems 2, 4, 5, 9, 13, 14, and 15/18 in detail. These systems, selected for analysis by more than one country, demonstrate the most market potential. In the remainder of this chapter, we briefly explain why AHP systems 3, 4a, 6, 7, 8, 16, 19, and 20 have no significant market potential.

4.3.1. Systems with Greater Market Potential

4.3.1.1. AHP 2: Absorption System

Technical Potential

AHP 2, which will be introduced in 1985-1986, is an absorption unit. Version 2a offers heating and cooling; version 2b offers heating only. The impact of the system's technical characteristics on its technical potential market is shown in Table 4-6. The system technically applicable in all countries and was analyzed by more countries than any other system.

Table 4 - 5

AHP Profiles Analyzed by National Market
Study Teams: Technical and Market Characteristics

AHP Profile	Year of Introduction	Heat Pump Cycle	Drive Type	Fuel	Heat Source	Thermal Output (kW)	Market Segment*
1		Absorption	Direct	Oil, gas	Ground	3 - 4	SF, MF I
2a	1985 - 1986	Absorption	Direct	Oil, gas	Ambient air	3 - 20	SF, MF I
2b	1985 - 1986	Absorption	Direct	Oil, gas	Ambient air		SF, MF I
3	1981	Absorption	Steam boiler	Oil, gas	Wastewater	2,000	MF II, district
4	1982	Rankine	Diesel	Oil	Ambient air	10 - 20	SF, MF I
4a	1980	Rankine	Diesel	Oil	Soil	400	MF II
5	Before 1985	Rankine	Diesel	Oil	Ambient air	100 - 6,000	MF II, district
6	1981	Ranine	Diesel	Oil	Sewage	2,000	MF II
7	1980	Rankine	Otto	Gas	Ambient air	10 - 400	SF, MF II
8	1981	Rankine	Otto	Gas	Air / ground / surface water	400	MF II
9	1983	Rankine	Stirling	Oil, gas	Air / ground surface water	3 - 20	SF, MF I
10		Absorption	- Solar -		Solar	15 - 20	
11		Absorption	Solar	Gas	Boom air	3 - 4	
12		Rankine	Rankine	Solar	Boom air	3 - 4	
13	1980	Rankine	Motor	Electricity	Ground	3 - 1,000	SF, MF I, MF II
14	1985	Absorption	Direct	Gas	Ambient air	10 - 20	SF, MF I
15 / 18	1985	Rankine	Rankine	Oil, gas	Ambient air	10 - 400	SF, MF I, MF II
16	1985	Rankine	Brayton	Oil, gas	Ambient air	50	MF II
17		Rankine	Rankine	Oil, gas	Ambient air	50	MF II
18		Rankine	Rankine	Oil, gas	Sea water	400	
19	1982	Absorption	Direct	Kerosene	Wastewater	2,000	District
20	1982	Absorption / Rankine	Motor	Electricity	Wastewater	2,000	District

* SF = Single family houses
MF I = Individually heated apartment units
MF II = Apartment units in centrally heated buildings
District = District heating

Table 4-6

TECHNICAL CHARACTERISTICS OF AHP 2

Technical Characteristic	Impact
Working fluid: organic	No impact
Fuel: natural gas or oil	No impact
Heat source: air	No impact
Distribution system: water or air	No impact
Thermal output: 10-20 kW	Limits application to single-family and multifamily I market segments

Economic Potential

Different cost and performance data sets were prepared for both versions of AHP 2. The price of version 2a, a heating and cooling system, was estimated by the manufacturers to be rather low (e.g., 5,000 DM for 20-kW output), compared to the estimated cost of version 2b, a heating-only unit (8,000 DM for the same heat output).

The heating and cooling version (2a) was evaluated in the U.S. and Japanese markets only. The heating-only version (2b) was also evaluated by the U.S. team, which reduced the estimated cost of version 2b to approximately 4,500 DM (for 20-kW output) to make it more consistent with national conditions. Economic potential is summarized in Table 4-7.

Although AHP 2 requires no backup boiler, its high initial cost generally limits its economic potential. Initial cost is often the major criterion used by potential purchasers. Both conventional heating technologies, including electric heat pumps, and several advanced technologies (e.g., advanced boilers or furnaces, AHPs 9, 14, 15/18) have a lower initial cost.

This system has a high potential in countries such as Germany and Sweden, where the cost of conventional systems (e.g., oil boilers) is relatively high. It also penetrates the market in the United States, but only in that part of the market that uses life-cycle cost as a decision criterion.

Twelve countries evaluated the potential of AHP 2. In Belgium, the initial cost of the system is too high (by 3,000 DM) to allow a penetration of the market. Alternatively, its seasonal performance factor (SPF) should be increased to 1.5 and more.

In Canada, AHP 2 captures 42 percent of the market in the period 1986-1995, with major installations occurring in new construction. However, competition with advanced gas furances was not considered, which would reduce this share. In the 1996-2000 period, relatively low electricity tariffs will make the conventional electric heat pump very competitive, thus reducing AHP 2's share to 18 percent.

In Denmark, AHP 2 could perform well in the single-family market, as it does in Germany and Sweden, unless advanced gas boilers appear to be economically superior on energy cost or payback-period criteria.

In Germany, AHP 2 could represent half of all heating and cooling installations from 1986, on economic factors alone. With a cost reduction of 3,000 DM, the market potential could be 100 percent. In the first period after introduction, one-fifth of all AHP 2 installations would be gas units; this proportion becomes one-third in 1996-2000, as the gas network is extended. In Germany, 62 percent of all AHP 2 installations will be made in existing buildings, a large proportion of which will be replacements of conventional heat generators in centrally heated single-family houses.

In Italy, Japan, the Netherlands, and the United Kingdom, AHP 2 is not competitive with other systems and has no chance to enter the market. Cost reductions achieved through R&D efforts or subsidies would increase its market share.

Table 4-7

ECONOMIC POTENTIAL OF AHP 2
AVERAGE ANNUAL INSTALLATION (000) AND MARKET SHARE (%)

Country/ System	Market Segment	1986-1990 (Market Share)	1996-2000 (Market Share)
AHP 2a (H/C)			
Japan	SF, MFI	-	-
United States	SF, MFI	180 (3.4%)	208 (4.1%)
AHP 2b (H)			
Austria	SF	Not evaluated; potential will be lower than AHP 14.	
Belgium	SF	-	-
Canada	SF	73 (42%)	29 (18%)
Denmark	SF	Not evaluated; potential could be large.	
Germany	SF	174 (44%)	191 (48%)
Italy	SF	-	-
Netherlands	SF	-	-
Spain	SF, MFI	-	-
Sweden	SF	11 (23%)	10 (24%)
Switzerland	SF	1 (1%)	7 (8%)
United Kingdom	SF	-	-
United States	SF, MFI	84 (1.6%)	112 (2.2%)
TOTAL		523	557

In Spain, restrictions on the use of oil and the national team's assumption of rising electricity prices allow AHP 2 to enter the market in the period 1991-1995, although it is more expensive than competing systems. The Spanish market share held by AHP 2 in the period 1996-2000, however, is only 6 percent.

In Sweden, AHP 2 takes a large part of the market in single-family houses heated by individual central water systems. The economic potential stays around 10,000 units during the 1986-2000 period (25 percent of the market).

In Switzerland, it was estimated that AHP 2 has an economic potential of approximately 1,000 units in the period 1986-1990. This potential will increase to 7,000 units by the 1996-2000 period.

In the United States, heating-only and heating and cooling installations of AHP 2 will capture 5 percent of the total single-family and multifamily I markets in 1986-1990. AHP 2 systems will account for 84,000 heating-only installations and 180,000 installations of heating and cooling units. This potential is strongly limited by the minor importance of life-cycle cost in the decision process used by purchasers. From 1986-2000, AHP 2 will have an economic potential of 351,000 units (6 percent of the market).

Implementation Potential

The implementation potential of AHP 2 was analyzed in Canada, Germany, Sweden, and the United States, the four major countries with significant economic market potential for this system.

In Sweden, AHP 2 must be produced domestically to meet climatic conditions and Swedish standards. Manufacturers' capacity will therefore limit market penetration of AHP 2 for the first 5 years after its introduction (1986-1990). Other possible constraints, such as lack of consumer acceptance or limited numbers of installers and maintenance companies, will not affect its diffusion during this period. AHP 13, introduced earlier on the market, will be affected by these factors.

In Germany, Canada, and the United States, diffusion rates were derived from those observed for similar heating technologies in the past. The implementation potentials for AHP 2 in these four countries are compared in Table 4-8.

4.3.1.2. AHP 4: Diesel Rankine System

Technical Potential

AHP 4, introduced in 1982, is a Diesel Rankine unit used for heating only. The impact of the system's technical characteristics on its technical potential market is summarized in Table 4-9.

This system was analyzed by Austria, Belgium, Germany, Switzerland, the United Kingdom, and the United States. Although it is technically applicable in most other countries, it was not analyzed for various reasons. For example, low natural gas prices in Canada and the Netherlands prevent it from being economically competitive.

Table 4-8

IMPLEMENTATION POTENTIAL OF AHP 2:
AVERAGE ANNUAL INSTALLATION (000) AND MARKET SHARE (%)

Country/ System	Market Segment	1981-1985 (Market Share)	1986-1990 (Market Share)	1991-1995 (Market Share)	1996-2000 (Market Share)	Cumul-ative Sales 2000
AHP 2a (H/C)						
United States	SF, MFI	0	80.0 (1.5%)	191.0 (3.3%)	208.0 (4.1%)	479
AHP 2b (H)						
Canada	SF	0	3.5 (5%)	8.8 (12%)	15.7 (21%)	28
Germany	SF	0	5.8 (3%)	13.6 (6%)	26.2 (13%)	230
Sweden	SF	0	3.5 (8%)	10.2 (24%)	10.0 (24%)	24
United States	SF, MFI	0	37.0 (0.7%)	72.0 (1.2%)	112.0 (2.2%)	221
TOTAL			153.0	350.0	479.0	982

Table 4-9

TECHNICAL CHARACTERISTICS OF AHP 4

Technical Characteristic	Impact
Working fluid: R12	No impact
Fuel: oil	No impact
Heat source: air	No impact
Distribution system: water	Prevents retrofit application where existing system is air
Thermal output: 10-20 kW	Limits application to single family and multifamily I segments

Economic Potential

The installed cost of AHP 4 is 7,000 DM for 10- to 12-kW output and 9,000 DM for 15- to 20-kW output, which is relatively higher than competing AHP systems 2, 9, 13, 14, and 15/18. In addition, AHP 4 requires a gas- or oil-fired boiler as a backup system. Larger Diesel-driven heat pumps (e.g., AHPs 4a and 5) are relatively less expensive, indicating that Diesel-driven heat pumps generally benefit from economies of scale.

The system captures market share over the period 1982-2000 only in the United Kingdom and Switzerland. In both countries it is more expensive than competing conventional systems and does not rank first on a life-cycle cost basis. Even so, the United Kingdom assumed that AHP 4 could capture 10 percent of the portion of the single-family and multifamily I segments where gas is not available and life-cycle cost is used as a decision criterion. In Switzerland, where life-cycle cost is the only decision criterion used, experts assumed that AHP systems would gain some market share.

In Belgium, Germany, and the United States, AHP 4 penetrates the market for the first few years after its introduction, but disappears after AHPs 2 and 15/18 are introduced. It does not penetrate the market at all in Austria, where its initial cost differential over competing conventional and AHP systems is too high to be compensated by energy savings. As a result, AHP 4 is economically competitive only on the basis of life-cycle cost or payback period. The economic potential market of AHP 4 in the United Kingdom and Switzerland is summarized in Table 4-10.

Implementation Potential

The implementation potential of AHP 4 was estimated by the United Kingdom, assuming 6 years for full consumer acceptance of such a new technology (see Table 4-11).

4.3.1.3. AHP 5: Diesel-Rankine System

Technical Potential

AHP 5, introduced by 1985 or possibly earlier, is a Diesel Rankine unit. It is used for heating only, but could be designed for cooling. The impact of the system's technical characteristics on its technical potential market is summarized in Table 4-12.

The technical potential for AHP 5 was analyzed by national teams in Austria, Denmark, Italy, Sweden, and the United States. It is too large to be applicable in most other countries, although Switzerland uses large systems for district heating. However, AHP 5 was not considered by Switzerland because of its low water-delivery temperature.

Economic Potential

The unit cost of AHP 5 with an output of 800 kW is approximately 200,000 DM. An existing fossil fuel-fired boiler is required as a backup system. The economic potential market for AHP 5 is shown in Table 4-13.

Table 4-10

ECONOMIC POTENTIAL OF AHP 4:
AVERAGE ANNUAL INSTALLATIONS (000) AND MARKET SHARE (%)

Country	Market Segment/Use	1986-1990 (% Market Share)	1996-2000 (% Market Share)
United Kingdom	SF, MFI where gas is not available/heating	20.2 (4%)	17.4 (4%)
Switzerland	SF/heating	2.2 (6%)	7.0 (17%)
TOTAL		22.4	24.4

Table 4-12
TECHNICAL CHARACTERISTICS OF AHP 5

Technical Characteristic	Impact
Working fluid: R12	No impact
Fuel: Diesel oil	No impact
Heat source: air	No impact
Distribution system: water	Prevents retrofit application where existing system is air
Thermal output: 200 kW to several MW	Limits application to multifamily II segment and block centrals, district heating

Table 4-13

ECONOMIC POTENTIAL OF AHP 5:
AVERAGE ANNUAL INSTALLATIONS (000) AND MARKET SHARE (%)

Country	Market Segment	1986-1990 (Market Share)	1996-2000 (Market Share)
Austria	MFII	12.0 (18%)	14.0 (21%)
Denmark	MFII	0.11 (18%)	0.3 (51%)
	BC, DH	0.83 (55%)	0.85 (54%)
Italy	MFII	0	0
Sweden	BC	0.213 (55%)	0.287 (53%)
	DH	0.005 (87%)	0.005 (87%)
United States	MFII	0.5 (1.6%)	0.4 (1%)
TOTAL		13.68	15.83

AHP 5 has significant economic potential in Austria, Sweden, and Denmark. In the Austrian market for medium-sized systems, it ranks below AHP 13 on life-cycle cost and payback period criteria, but captures that portion of the market where AHP 13 is not technically applicable. Over the period 1981-2000, AHP 5 captures approximately 42 percent of the multifamily II new construction market and 10 percent of the retrofit market. In Sweden, the 1-MW AHP 5 captures 50-55 percent of the block central heating market, as a result of its good ranking on life-cycle cost and payback period criteria. The 10-MW AHP 5 captures approximately 87 percent of the small-size district heating system market; on the average, however, this share corresponds to only 2 percent of the capacity installed in district heating systems each year.

The system does not have significant economic potential in the United States and Italy. Because it is used for heating alone, it captures only a small portion of the market in the United States, which has a large demand for both heating and cooling. In addition, the system ranks first only on the criterion of life-cycle cost, which is not considered by most American consumers. AHP 5 captures no market share in Italy, where its initial cost is relatively high and its high maintenance costs offset energy savings over the system's life.

In Denmark, the medium-size AHP 5 captures approximately half of the multi-family II market, and the large version half of the district heating system market in the period 1996-2000. AHP 5 as a whole capture 80-90 percent of the total market for medium and large systems in Denmark. AHP 5 is not as economically competitive as other AHP systems; however, its widespread technical applicability allows it to capture some market share.

Implementation Potential

The implementation potential of AHP 5, analyzed by Denmark, Sweden, and the United States, is shown in Table 4-14. By the year 2000, about 10,700 medium-size units will be installed in Denmark and the United States, and about 10,960 large units in Denmark and Sweden.

4.3.1.4. AHP 9: Stirling-Rankine System

Technical Potential

AHP 9, introduced in 1983, is a Stirling/Rankine unit used for both heating and heating/cooling. The impact of the system's technical characteristics on its technical potential market is illustrated in Table 4-15.

AHP 9 was analyzed only by national teams in Japan, the Netherlands, the United Kingdom, and the United States; even though the system is applicable in other countries, including Sweden. AHP 9 uses an air distribution system, whereas existing distribution systems are primarily water. As a result, AHP 9 has limited technical applicability in the retrofit market. In addition, Canada felt the system was not economically competitive and that low natural gas prices would decrease the likelihood of its use.

Table 4-14

IMPLEMENTATION POTENTIAL FOR AHP 5:
AVERAGE ANNUAL INSTALLATIONS (000)

Country/System	1986-1990	1991-1995	1996-2000
Medium system:			
Denmark	0.55	0.205	0.3
United States	0.48	0.42	0.41
Large system:			
Denmark	0.42	0.84	0.85
Sweden	0.028	0.09	0.242

Table 4-15

TECHNICAL CHARACTERISTICS OF AHP 9

Technical Characteristic	Impact
Working fluid: R22	No impact
Fuel: natural gas or oil	No impact
Heat source: air	No impact
Distribution system: air	Prevents retrofit application where existing system is water
Thermal output: 10-20 kW	Limits application to single-family and multifamily I segments

Economic Potential

The installed cost of AHP 9 with an output of 15-20 kW is 5,000 DM. In addition, a natural gas or oil boiler is required as a backup system. The economic potential market of AHP 9 in Japan and the United States is shown in Table 4-16.

AHP 9 captures significant market share only in Japan, where an oil-fired configuration was analyzed. It ranks first above competing conventional systems and AHPs 2 and 15/18 on the basis of initial cost and payback period, capturing 100 percent of the single-family and multifamily market segment where heating and cooling is required (over half of the total single-family, multifamily I market segments). However, since Japan assumed an unusally low initial cost, market potential for AHP 9 may have been overestimated.

In the United States, where a gas-fired configuration was analyzed, the system captures a small portion of the market in the first few years after its introduction, but disappears from the market after AHPs 2 and 15/18 are introduced.

In both the Netherlands and the United Kingdom where gas-fired configurations were analyzed, AHP has no significant economic penetrations. In the Netherlands, it ranks below conventional systems but above other advanced systems (AHPs 2, 14, and 15/18) competing in the single-family market segment. In the United Kingdom, AHP 9 ranks below competing conventional systems as well as AHPs 14 and 15/18, which do capture market share.

Implementation Potential

Japan analyzed the implementation potential of AHP 9, assuming it would be a percentage of economic potential, as shown in Figure 4-2. Using this market penetration rate, approximately 2.5 million units will be installed in Japan (by 2000). Table 4-17 shows average annual installations over the period 1986-2000.

4.3.1.5. AHP 13: Electric/Rankine System

Technical Potential

AHP 13 is an electric motor/Rankine unit used for heating only. One version of this system, which will be introduced between 1982 and 1985, uses vertical ground coils as a heat source. The other version, introduced in 1980, uses horizontal ground coils. The impact of the system's technical characteristics on its technical potential market is summarized in Table 4-18.

AHP 13 was analyzed in the majority of countries, including Austria, Belgium, Denmark, Germany, Japan, Spain, Sweden, the United Kindgom, and the United States. Although, it is technically applicable in Italy and Switzerland, it was not analyzed in these countries. In Canada and the Netherlands, the system was not analyzed because electricity prices are relatively higher than those of natural gas.

Table 4-16

ECONOMIC POTENTIAL OF AHP 9:
AVERAGE ANNUAL INSTALLATIONS (000) AND MARKET SHARE (%)

Country	Market Segment/Use	1986-1990 (Market Share)	1996-2000 (Market Share)
Japan	SF, MFI/H & C	83.3 (46%)	250.1
United States	SF, MFI/H & C	1.3 (0.02%)	0
TOTAL		84.6	250.1

Table 4-17

IMPLEMENTATION POTENTIAL OF AHP 9:
AVERAGE ANNUAL INSTALLATIONS (000) AND MARKET SHARE (%)

Country	1986-1990	1991-1995	1996-2000
Japan	66.6 (37%)	140.6 (45%)	250.1 (53%)

Table 4-18

TECHNICAL CHARACTERISTICS OF AHP 13

Technical Characteristic	Impact
Working fluid: R22	No impact
Fuel: electricity	No impact
Heat source: ground	Prevents application where adequate ground area is not available; important barrier for version using horizontal ground coils, as an area of approximately 200 m^2 is required for a single-family system; vertical-coil system can be used only where ground is clay
Distribution system: water or air	No impact
Thermal output: 3-100 kW	No impact; application in all market segments

Economic Potential

The installed cost of AHP 13 with 10- to 12-kW output is approximately 13,000 DM, relatively higher than that of competing AHP systems (e.g., small AHPs 2, 4, 9, 14, and 15/18; medium AHPs 5 and 15/18). In addition, an electric resistance backup system is required.

The small AHP 13 (used in individually heated housing units) was analyzed in all countries, but gains significant market share only in Japan, Sweden, and Denmark. In Japan, it ranks first on payback period and captures the portion (25 percent) of the market in heating-only regions that uses payback period as a decision criterion. The competing conventional oil boiler ranks first on initial cost, and captures the remaining 75 percent of that market.

In Sweden, AHP 13 captures approximately 13-14 percent of the single-family market. It loses market share to AHP 2 on the basis of payback period, but ranks well on the criterion of life-cycle cost. Limitations on its technical applicability further reduce its economic potential market.

In Denmark, AHP 13 captures 38-50 percent of the single-family market, because heating-systems purchasers are highly sensitive to energy costs and payback period and electricity is relatively less expensive than oil. As in Sweden, AHP 13's potential in Denmark is somewhat constrained by its limited technical applicability.

In the remaining countries, the small version of AHP 13 does not capture significant market share. In Germany, it captures only 1 percent of the single-family market, primarily because of its high initial cost and consumer reluctance to install electric heating systems.

In Belgium, AHP 13 captures market share in the first few years after its introduction, but loses share when AHP 15 is introduced. In Austria, Spain, the United Kingdom, and the United States, the system is too expensive and seems to have no market potential.

The medium AHP 13 system (used in centrally heated buildings) was analyzed in Austria, Denmark, Japan, Spain, and Sweden, and gains market share in all of these countries except Spain. In Austria, it captures about 15 percent of the multifamily new construction market and 25 percent of the multifamily retrofit market over the period 1986-2000. It ranks first on life-cycle cost and payback period, but requires a large ground area, which limits its technical, and thus economic, potential markets.

In Denmark, AHP 13 captures 25 percent of the multifamily II market over the period 1981-2000. The percentage is somewhat lower in the early years (approximately 13 percent) and higher in the later years (40 percent), because it ranks well on the criterion of life-cycle cost by 1981, and well on the basis of both life-cycle cost and payback period by 1991. Limitations on its technical applicability reduce its economic potential market.

In Japan, like the small version of AHP 13, the medium-sized version ranks first on payback period and captures 25 percent of the multifamily II market segment in heating-only regions. Again, the competing conventional oil boiler captures the remaining 75 percent of the market that uses initial cost as a decision criterion.

In Sweden, AHP 13 captures approximately 15 percent of the multifamily market, but its economic potential is constrained by its limited technical applicability. In Spain, both sizes of AHP 13 are too expensive and have no economic potential.

A large version of AHP 13 (used in block central and district heating) was analyzed by Sweden and Denmark. In Sweden, this version gains only a small portion of the block-central market, because AHP 5 ranks above it on life-cycle cost and payback period, the decision criteria used in this segment. In Denmark, it gains only 10 percent of the district heating market, due to severe limitations on its technical applicability. Economically, the system ranks first on life-cycle cost, the only decision criterion used in this segment.

The economic potential market for AHP 13 is summarized in Table 4-19.

Implementation Potential

The implementation potential of AHP 13 was analyzed by Germany, Japan, and Sweden. The results are shown in Table 4-20. By 2000, approximately 537,000 small systems, 2,300 medium systems, and 286 large systems will be installed.

4.3.1.6. AHP 14: Absorption System

Technical Potential

AHP 14, introduced in 1985, is an absorption unit used for both heating and cooling. The impact of the system's technical characteristics on its technical potential market is summarized in Table 4-21.

AHP 14 was analyzed by national teams in Austria, the Netherlands, and the United Kingdom. It is also technically applicable in Canada, Italy, and Switzerland, but was not analyzed in these counties. AHP 14 is not technically applicable in the remaining market study countries, which restrict the use of ammonia as a working fluid and prefer AHP 2, a similar system using an organic working fluid.

Economic Potential

The initial cost of AHP 14 is relatively lower than that of competing AHP systems 2, 4, 9, 13, and 15/18, although still higher than competing conventional systems. The cost of a unit with 10- to 12-kW output is approximately 4,000 DM; a unit with 15- to 20-kW output costs approximately 7,000 DM. No backup system is required.

The system performs well in Austria, where the initial cost of competing conventional systems is relatively high. It ranks first on payback period and, over the period 1986-2000, captures 50 percent of the single-family new construction market and approximately 20 percent of the single-family retrofit market. It gains no market share in the Netherlands, however, because of the relatively lower initial cost of competing conventional and advanced conventional boilers.

Table 4-19

ECONOMIC POTENTIAL OF AHP 13:
AVERAGE ANNUAL INSTALLATIONS (000) AND MARKET SHARE (%)

Country/ System	Market Segment/Use	1986-1990 (Market Share)	1996-2000 (Market Share)
Small system:			
Austria	SF/heating	0	0
Belgium	SF, MFI/heating	0	0
Denmark	SF/heating	30 (41%)	25 (51%)
Germany	SF/heating	4 (1%)	4 (1%)
Japan	SF, MFI/heating	24 (13%)	56 (12%)
Spain	SF/heating	-	-
Sweden	SF/heating	5.5 (13%)	5.8 (14%)
United Kingdom	SF, MFI/heating	0	0
United States	SF, MFI/heating	0	0
TOTAL		63.5	90.8
Medium system:			
Austria	MF/heating	20 (30%)	20 (30%)
Denmark	MFII/heating	0.08 (13%)	0.23(40%)
Japan	MFII/heating	0.2 (13%)	0.5 (12%)
Spain	MFII/heating	-	-
Sweden	MF/heating	0.1 (14%)	0.02(16%)
TOTAL		20.38	20.75
Large system:			
Denmark		0.15 (10%)	0.16(10%)
Sweden	Block central/ heating	0.09 (22%)	0.1 (22%)
TOTAL		0.24	0.26

Table 4-20

IMPLEMENTATION POTENTIAL OF AHP 13
AVERAGE ANNUAL INSTALLATIONS (000)

Country/System	1986-1990	1991-1995	1996-2000
Small system:			
Germany*	0.5	0.8	1.2
Japan	16.7	31.5	50.4
Sweden	5.5	5.9	5.8
Medium system:			
Japan	0.17	0.32	0.46
Sweden	0.11	0.07	0.02
Large system:			
Sweden	0.06	0.09	0.11

* Slow penetration rate.

Table 4-21

TECHNICAL CHARACTERISTICS OF AHP 14

Technical Characteristic	Impact
Working fluid: ammonia/water	Prevents application in countries where the use of ammonia is restricted
Fuel: natural gas	Limits application to the portions of a country where natural gas connections are available
Heat source: air	No impact
Distribution system: water or air	No impact
Thermal output: 10-20 kW	Limits application to single-family and multifamily I market segments

In the United Kingdom, where competing conventional systems are also initially less expensive, the national team assumed that AHP 14 could capture approximately 30 percent of the portion of the SF and MFI market segments where natural gas is available and that use life-cycle cost as a decision criterion. The economic potential market for AHP 14 in Austria and the United Kingdom is shown in Table 4-22.

Implementation Potential

The implementation potential of AHP 14 was analyzed only by the United Kingdom, which assumed that it would take 6 years after introduction for full consumer acceptance of a new technology. During this period, its implementation potential as a percentage of economic potential would be as follows: 10 percent in year 1, 20 percent in year 2, 30 percent in year 3, 50 percent in year 4, 75 percent in year 5, and 90 percent in year 6. Starting in year 7, implementation potential would be 100 percent of economic potential. By the year 2000, approximately 1.5 million units will be installed in the United Kingdom.

4.3.1.7. AHP 15/18: ORC System

Technical Potential

AHP 15/18, introduced in 1985, is an organic Rankine/Rankine unit used for both heating and heating/cooling. The impact of the system's technical characteristics on its technical potential market are summarized in Table 4-23. The system is technically applicable in all countries. It was analyzed by national teams in Belgium, Germany, Japan, the Netherlands, Switzerland, the United Kingdom, and the United States.

Economic Potential

The target installed cost of a small AHP 15/18 with output of 10-12 kW is approximately 5,700 DM. Medium and large versions with 20 kW output or more cost approximately 500 DM/kW. In addition, an auxiliary boiler is required as a backup system.

The small system (used in individually heated housing units) was analyzed by a number of market study countries. It has significant economic potential only in Belgium, Germany, and the United Kingdom. In Belgium, the oil-fired configuration ranks above competing conventional and oil boilers and AHPs 2, 4, and 13 on payback period, allowing it to capture 30 percent of the market for small systems by 1996-2000.

In Germany, the oil-fired configuration ranks first on the basis of initial cost for new and modernization applications, allowing it to capture 12 percent of the small-systems market by 1996-2000. However, it is outranked by the competing conventional oil boiler on the basis of initial cost for replacement applications and by AHP 2 on the basis of energy cost (the only other decision criterion used in Germany) for all applications. The gas-fired configuration gains a small market share over the period 1985-1990, ranking well on the basis of energy cost. However, after 1990, AHP 15/18 has no market potential in Germany, losing to the competing AHP 2.

Table 4-22

ECONOMIC POTENTIAL OF AHP 14
Average Annual Installations (000) And Market Share (%)

Country/ System	Market Segment/Use	1986-1990 (Market Share)	1996-2000 (Market Share)
Austria	SF/heating	9 (24%)	12 (30%)
United Kingdom	SF, MFI where gas is available/heating	142 (12%)	134 (13%)
Total		151	146

Table 4-23

TECHNICAL CHARACTERISTICS OF AHP 15/18

Technical Characteristic	Impact
Working fluid: various	No impact
Fuel: natural gas or oil	No impact
Heat source: air	No impact
Distribution system: water or air	No impact
Thermal output: 10 kW and up	No impact; applicable in all market segments

In the United Kingdom, the gas-fired configuration captures 4 percent of the market where natural gas is available by 1996-2000. It ranks relatively well on life-cycle costs, and the British study team assumed it could capture 10 percent of the portion of the market using life-cycle cost as a decision criterion.

The system does not attain significant economic potential in other countries. In the United States, the oil-fired configuration gains only a small portion (less than 1 percent) of the market for small systems over the period 1985-1990. After 1990, it gains no market share, losing to AHP 2. In Japan and the Netherlands, AHP 15/18 has no economic potential market because of its poor performance and high initial cost.

The medium system (used in centrally heated buildings) was analyzed in Belgium, Germany, Switzerland, and the United States, and captures market share in each. In Belgium, the oil-fired configuration ranks first on the basis of payback period, allowing it to capture 38 percent of the multifamily II market by 1996-2000.

In Germany, AHP 15/18 wins the portion of the multifamily market that uses energy cost as a decision criterion, capturing 15 percent (oil-fired) and 11 percent (gas-fired) by 1996-2000. Competing conventional oil and gas systems rank first on initial cost.

In Switzerland, the AHP 15/18 systems rank relatively well on life-cycle cost, the only decision criterion used, and capture 16 percent of the multifamily II market by 1996-2000.

The system also ranks well on life-cycle cost in the United States, which uses primarily the gas-fired configuration, and it captures 4-5 percent of the total multifamily II market during 1986-2000. In the multifamily II market requiring heating/cooling systems, it achieves 10-percent market share over the same period. The economic potential markets for both small and medium AHP 15/18 configurations in the seven market study countries that analyzed it are listed in Table 4-24.

Implementation Potential

The implementation potential of AHP 15/18, analyzed by Belgium, Germany, the United Kingdom, and the United States, is shown in Table 4-25.

4.3.2. Systems with Limited Market Potential

4.3.2.1. AHP 3: Absorption System

AHP 3 is a large absorption system for district heating applications. It is powered by a steam boiler and uses wastewater as a heat source. Switzerland, the only country to analyze this system, predicted a 16-percent share for it in district heating applications.

Table 4-24

ECONOMIC POTENTIAL OF AHP 15/18:
AVERAGE ANNUAL INSTALLATIONS (000) AND MARKET SHARE (%)

Country/ System	Market Segment/Use		1986-1990 (Market Share)		1996-2000 (Market Share)
Small system:					
Belgium	Heating		58.6 (28%)		61 (32%)
Japan	Heating & cooling		0		0
Germany	Heating	oil	25 (6%)	oil	48 (12%)
		gas	20.4 (5%)	gas	0
Netherlands	Heating		0		0
United States	Heating,		0.8		0
	Heating & cooling		1.8 (0.05%)		0
United Kingdom	Heating		47.3 (4%)		44.7 (4%)
Total			153.9		153.7
Medium system:					
Belgium	Heating		1.8 (44%)		1.8 (38%)
Germany	Heating	oil	36 (16%)	oil	36 (15%)
		gas	25 (11%)	gas	28 (11%)
Switzerland	Heating		2.4 (5%)		7.6 (16%)
United States	Heating,		0.4 (1%)		0.3 (1%)
	Heating & cooling		1.3 (4%)		1.1 (3%)
Total			66.9		74.8

Table 4-25

IMPLEMENTATION POTENTIAL OF AHP 15/18:
AVERAGE ANNUAL INSTALLATIONS (000) AND MARKET SHARE (%)

Country/ System	1986-1990	1991-1995	1996-2000	Cumulative Total 1981-2000
Small system:				
Belgium	5.24 (3%)	14.74 (7%)	25.41 (13%)	227
Germany*	2.8 (1%)	3.4 (1.6%)	10 (5%)	81.4
United Kingdom	25.3 (2%)	44.8 (4%)	44.7 (4%)	578.4
United States	2.6 (0.05%)	1.2 (0.02%)	0	19
Medium system:				
Belgium	0.16 (3%)	0.46 (7%)	0.79 (13%)	7
Germany	0.26 (0.25%)	3.4 (3.6%)	4.6 (5.1%)	41.5
United States	0.67 (2%)	1.6 (4.5%)	1.4 (4%)	18.4

* Slow penetration rate.

AHP 3 uses ammonia as a working fluid, which constrains the system's potential market in countries that prohibit use of ammonia. A change to an organic working fluid would remove this barrier. The system could then compete with AHP 6 and 19 for district heating applications. In addition, a reduction of minimum size (now 2 MW) could make AHP 3 applicable in multifamily II buildings.

4.3.2.2. AHP 4a: Diesel System

AHP 4a is a large Diesel-driven heat pump with soil heat source for multifamily and district heating applications. For district heating, AHP 4a was analyzed only in Denmark, where it compares poorly to AHP 5 (the Diesel heat pump with air source) on economic grounds, and in Japan, where it loses to AHP 13. AHP 4a was analyzed in Sweden for multifamily II applications and loses all its potential market share to AHP 13, the electric soil-source heat pump. AHP 4a appears to have too high a first cost and too low an efficiency to compete with other AHP systems.

4.3.2.3. AHP 6: Diesel System

AHP 6 is a large Diesel-driven heat pump, using wastewater as a heat source and intended for district heating applications. This system was studied only in Denmark, where competition with other large AHPs (AHPs 5 and 13) and rather limited technical applications (the AHP 6 requires connection to a wastewater stream) reduce its potential. It is likely that the same results would occur in other countries interested in district heating applications.

4.3.2.4. AHPs 7 and 8: Otto-Cycle System

AHPs 7 and 8 are Otto-engine-driven heat pumps using ambient air or groundwater as a heat source and intended for application in single-family homes and multifamily II buildings. These systems were analyzed only in the Netherlands where, like all other AHP systems, they failed to gain any market share against conventional technologies. Generally, their costs seem too high to allow them to capture market share, even in countries with more favorable conditions.

4.3.2.5. AHP 16: Brayton-Cycle System

AHP 16 is a Brayton-cycle-driven heat pump of approximately 50-MW capacity, designed for multifamily II buildings and using an air heat source. Although applicable in many countries, this system was studied only in the United States. Due to high first cost, it generates a market only in the South, which has both heating and cooling demand. But even there, conventional systems predominate. In other regions, AHP 16 is less attractive than AHP 15/18.

4.3.2.6. AHPs 19 and 20: District Heat Absorption Systems

AHPs 19 and 20 are large absorption systems used for district heating applications. They were analyzed only by Japan, which considered AHPs 19 and 20 in competition for heating and cooling applications with other small systems (AHPs 2, 9, and 15/18) in the single-family and multifamily I market segments. In effect, the analysis by the Japanese market study team addresses the question of installing district heating or relying on small systems. The decision maker must chose between large and small systems, a choice not considered by other countries.

AHPs 19 and 20 lose market share to AHP 9 for heating and cooling applications. In a more complete analysis, it would be interesting to compare the competitive position of AHPs 19 and 20 with other district heating systems, because it seems unlikely that small AHPs will compete directly with large systems.

5. Conclusions

5.1. General

The technology survey has shown that in the 13 participating countries there is a large and extremely diverse R+D programme covering most of the possible systems. Some degree of overlap has been found, and although this may in certain circumstances be wasteful it is generally more likely to result in healthy competition to develop successful products. The present situation of heat pumps is that, following the stimulus of the first and certainly not last energy crisis of 1973, of a technology emerging from many years of neglect. Certain aspects of this crisis, recently re-emphasised by a further escalation in oil prices, the response of the participating countries and the present state of heat pump development form the background to the major R+D recommendations. These are -

(1) The continuing dependance on petroleum products, mainly light oil, for the heating of buildings in the participating countries. This is particularly significant for Belgium, Canada, Denmark, Germany, Italy, Sweden and Switzerland, for which countries oil is used for about one half or over the domestic and commercial heating requirements.

(2) The predominant position of electric heat pumps (for both heating and cooling) in the U.S. as replacement for fossil fuel fired boilers or furnaces. The situation in W. Germany is also developing in this direction because gas and oil fired systems are not fully commercialised and the market is being aggressively attacked by the electric heat pump manufacturers and utilities.

(3) The first priority is at present to substitute for oil and the second priority is the conservation of all forms of energy. The electric heat pump as presently available and promoted meets the first priority but only contributes to energy conservation when compared to electric heating. The development of gas heat pumps can be attributed to both priorities, but

dependance on gas could result in similar pressures as those in the oil market.

(4) Most advanced heat pumps, with the present development programs, are not due to commercialisation until 1985 at the earliest. Since they cannot be expected to make a significant contribution rapidly, the possibility of achieving worthwhile energy saving before 1990 appears low. By this time the energy supply situation could have changed dramatically (viz., the decade to 1980). The acceleration of the development programs of the systems with greatest potential for each application is desirable, even if in the short term their use would not be justifiable solely on economic grounds.

(5) All the advanced heat pumps appear capable of conserving between 30 % and 50 % energy. The best systems from this aspect are the internal combustion engine driven ones.Many are capable of operation on either liquid or gaseous fuels while the external combustion systems have total flexibility of fuel use, particularly in the larger unit sizes.

(6) There are systems under development for most applications. However, there appear to be no systems under development for the smallest sizes (3-4 kW - suitable for single room heating, small well insulated houses or houses in mild climates) and insufficient activity to develop systems for the industrial area.

(7) All air source heat pumps suffer from the disadvantage of declining efficiency towards peak heating demand. The peak demand imposed by the use of such systems, either monovalently or bivalently, will also restrict their use unless energy can be stored for the peak demand. This can be done by storage of energy as fuel (i.e. using an oil or LPG[+] fired boiler as back-up), as heat (hot water or other store) as chemical or latent energy. There is therefore a requirement for the development of cost effective storage systems. These could also be used to avoid low efficiency operation of systems on partload or render unnecessary sophisticated control systems.

[+] Liquified Petroleum Gas

(8) There is at present no international standard for performance for heat pumps. Two countries (U.S. and Germany) have national standards and these are completely different. There is a pressing need for uniform standards for performance and also for definitions of terms and symbols. A further requirement is for some sort of code of practice for advertising and promotion to prevent the market for heat pumps being spoilt by misleading claims or unscrupulous promoters.

(9) The reliability, life-time, service and maintenance of advanced heat pumps will in most cases prove troublesome aspects in their initial years of commercialisation. The provision of suitable service and maintenance facilities and the guarantee of installations during the acceptance phase are essential factors in their acceptability to the potential user. The use of pilot installations and close monitoring of installations during this phase will assist in the controlled and orderly growth of the market and the necessary infrastructure.

(10) The capital costs of the advanced systems in general are still high compared with the conventional boiler or furnace costs. The reduction of first cost, a major factor influencing choice, can be attained by specific R+D activities, by good design and production in large quantities.

(11) There exists a diversity of financial, fiscal and other incentives to the development and use of energy conserving technologies, including heat pumps, in the participating countries. The effectiveness of these policies and possible alternatives in promoting the objectives of substitution for oil and conserving energy needs investigation. The comparison of the various methods should enable the participating countries to choose the most cost effective methods of achieving their objectives.

5.2. R+D Needs

There have been identified a large number of R+D needs which would assist in the commercial realisation of those advanced heat pumps which appear best suited to the various applications, and to overcome disadvantages and to strengthen those systems which at present are less attractive or at an early stage of development. These R+D needs could form part of the programmes of work of those already developing these systems, of national or utility research, while some are particularly suitable for international co-operation and are more fully described in the next section.

5.2.1. Technical R+D Needs

The R+D needs have been classified by system type and are given in Table 5-1. These are not listed by priority since many have been integrated into the projects suggested as suitable for international co-operation. The remaining topics mainly apply to systems under development and could be incorporated in the existing work programmes or, where, appropriate as parts of national R+D programmes.

Sorption

Research	Development	Demonstration
• Search for new safe fluids for small fossil fuel driven units • Search for new safe fluids for high temperature industrial applications • Collection and publication of all available data on previously considered fluids • Investigation of the boosted *) absorption cycle's potential for district heating • Computer modelling of absorbers and expellers for organic and novel fluids • Computer modelling of sorption cycles for different fluids, aiming at predicting the performance of given concepts and developing new ones • Investigation of safety requirements regarding domestic use of ammonia • Investigation of alternative sorption cycles e.g. resorption and Altenkirch cycles, multistaging, to achieve better performance • Theoretical and experimental study of refrigerant mixtures in sorption systems • Fire testing of organic fluids with respect to explosivity, inflammability and formation of physiologically dangerous compounds through pyrolysis	• Design of compact, low noise and high efficiency solution pump for organic fluids • Design of high efficiency generators • Design of an organic pair, solar space cooling unit • Design of a combined heat pump/storage system to meet peak demand • Design of small reversible units to replace electric window air-conditioners • Development of sorption heat pumps with storage function (heat of reaction/dilution)	• Demonstration and comparison of prototype sorption heat pumps (in the range of 10 to 20 kW thermal output) for single family houses (see external combustion) • Demonstration of medium sized sorption units for multifamily house heating

*) Compressor between evaporator and absorber

Table 5-1: R+D Recommendations

Electric heat pumps

Research	Development	Demonstration
• Study methods of improving performance of air source heat pumps below +3°C to overcome utility restrictions to bivalent-parallel operation • R+D on short to medium term duration heat storage to supplement heat pump output at peak load conditions • Study of Lorentz cycle systems for domestic heat pumping • Investigation of methods to reduce auxiliary power requirements e.g. fans and pumps	• Development of high efficiency controllable motor-compressor units of small unit size • Development of third generation high efficiency, controllable, lower cost heat pumps for space heating only • Introduction of higher national performance standards for air-conditioners and heat pumps • Development of self defrosting evaporators for air source systems • Electronic control systems • Development of low cost plastic radiator and piping systems for low temperature water distribution systems	• Demonstration of best heat pump technology for single family house heating

Table 5-1: R+D Recommendations

Internal combustion driven

Research	Development	Demonstration
• Investigation of the possibilities of improving recip. compressor efficiency and controllability • Investigation of alternative compressor types for small engine driven systems • Investigation into control methods to achieve high efficiency output regulation, and application of microprocessor control • Investigation of free piston and rotary engine/compressor systems for small units ($<$ 30 kW output) • Study of possible methods of increasing service life and reliability of internal combustion engines for heat pump drives • Study on local pollution and noise produced by small engine systems • Recommendations for countermeasures and design practices to meet building regulations	• Design and construction of integrated engine/compressor units (10 to 20 kW output) • Development of methods to avoid soot formation or to remove soot from exhaust gas coolers of Diesel units • Development of integrated engine heat recovery packages • Development of electronic control systems for optimum performance and life of units • Development of peak heat load shaving devices (e.g. thermal stores, short term performance boosting) to avoid need for back-up boilers	• Operation of gas engine driven heat pumps on alternative gases (SNG, mine gas, town gas, biogas) • Experimental comparison of performance of internal combustion engine driven heat pumps (10 to 20 kW) • Monitoring of reliability, service and maintenance requirements of existing internal combustion engine driven systems to identify best practices and R+D requirements • Demonstration of engine driven systems for power generation and heat supply for small communities: the so-called 'Blockheizkraftwerke' (Block heating and power centres)

Table 5-1: R+D Recommendations

External combustion driven

Research	Development	Demonstration
• Investigation of suitability of external combustion systems for higher temperature applications e.g. district heating, industrial applications. • Investigation of solid fuel fired external combustion systems for medium and large applications • R+D to produce a design method (computer based) for ejectors for heat pump drives (also for sorption units) • Assessment of free piston units for small heating applications • Study of control problems and off design point performance of Rankine-Rankine systems and identify best control philosophy • Investigation of alternative heat pump cycles for external drives (Joule, Edwards, Lorentz, Resorption) • Computer modelling of solar activated systems for optimisation	• Design and development of kinematic Stirling-Rankine cycle system for medium size applications (100-400 kW thermal output) • Development of larger sized single fluid Rankine-Rankine systems • Development of Brayton and Stirling systems for combined heat and power (for heat pump drive) units for small communities • Development of high efficiency low cost feed pumps for Rankine-Rankine systems	• Demonstration and comparison of alternative systems (Rankine-Rankine, Stirling-Rankine and Ejector) for single family house heating (see sorption) • Demonstration of combined heating/cooling systems (Stirling-Rankine, Brayton-Rankine and Rankine-Rankine) for commercial building air-conditioning • Demonstration of external combustion systems for industrial applications

Table 5-1: R+D Recommendations

General

Research	Development	Demonstration
• Investigation of the fundamental problems of vertical ground sources (geology, heat transfer, heat capacity, mech. effects) • Study on the integration of large heat pump systems in district heating schemes • Evaluation of fluid mixture heat pumps to approximate the Lorenz cycle and achieve high efficiency • Assessment of the possibilities of using coal and wastes in external heat pump combustors of both sorption and vapour compression systems • Investigation of air pollution impacts generated by local exhausting of relatively cold exhaust gases • Assessment of the combined cycles' potential for district heating • Assessment of heat sources' potential for large heat pump systems • Impact study on effect of extensive use of heat pump technology on resources, environment, productive capacity, service infrastructure etc. • Study of effects of various financial, fiscal and other subsidies on heat pump development and use	• Computer modelling of vertical ground source with respect to optimisation of geometry • International standardisation of heat pump systems and testing methods • Development of concepts for large scale waste heat usage: e.g. transport of low temperature waste heat carrying water to district heating heat pumps • Development of models for the optimisation of heat pumps + improved thermal building insulation • Development of small capacity controlled high efficiency compressors • Development of low cost solar roofs as heat sources • Design of low cost and corrosion proof exhaust gas coolers for operation below the dew point • Development of small air to air heat pumps for well insulated houses using exhaust air as source • Development of high efficiency heat pumps for industrial heat recovery and drying	• Field testing of vertical ground source heat pumps (10 to 20 kW): a) electric drive b) comparison of use with on- and off-site activated heat pumps • Application of low temperature large scale solar ponds for urban heating with on-site activated heat pumps (MW-range) • Establishment of central test facility for proving of different heat pump systems (3 to 20 kW) • Field testing of solar source heat pumps (10 to 20 kW)

Table 5-1: R+D Recommendations

5.2.2. Market Relevant R+D Needs

Although 15 advanced heat pumps were analyzed in the market study, only AHPs 2, 4, 5, 9, 13, 14, and 15/18 show significant market potential in one or more countries. The conclusions of the market study for each of these systems are discussed below.

AHP 2

AHP 2, in its heating and cooling (2a) or heating (2b) versions, has the largest market potential of all the AHPs considered in the market study. This potential could be increased even more by R&D oriented toward the following objectives:

- Accelerate development to allow earlier commercialization (estimated in the analysis to occur in 1986).
- Reduce first cost. A cost reduction of 3,000 DM would open new markets for AHP 2b in Europe (e.g., Belgium, Italy, and the United Kingdom) and double its potential in Germany.

AHP 4

AHP 4, the small Diesel system, does not achieve significant market penetration in any of the countries that studied it, except in the United Kingdom and Switzerland, where the market estimate was based on qualitative judgment. In fact, AHP 2, introduced later, has better economic performance than AHP 4, and would tend to drive it out of the market. However, other, larger Diesel systems (e.g., AHPs 4a and 5) show good prospects in other market segments, indicating that Diesel-driven systems benefit from economies of scale.

AHP 4 will benefit from any delay in the commercialization of AHP 2 to establish its market position and reduce its cost through longer production runs. Time is also needed for installers and customers to adapt themselves to its more complex heating system. It is uncertain whether single-family home owners will accept a motor-driven heating system. Reducing noise and improving reliability through R&D would make the system more acceptable.

AHP 5

Despite its large size, which limits it to certain applications and countries, AHP 5 is the most successful of the Diesel systems studied. Except for Italy, the system performs well in all the countries that studied it. The Swedish analysis showed that the system performed relatively better in large installations (10 MW). No major R&D conclusions can be drawn from the results of the study.

AHP 9

AHP 9, the Stirling engine-driven heat pump, is applicable only in Japan, the United Kingdom, and the United States, due to its air distribution systems. Only in Japan does the system penetrate the market, outranking AHPs 2 and 15. This estimate is partly due to the fact that Japanese analysts assumed a lower first cost than that assumed by other countries. To achieve market share in the United States, the system will require significant first cost reductions, and R&D should be directed to this end.

AHP 13

The market share for AHP 13, the ground-source electric heat pump, is large, especially in colder countries where decision makers are sensitive to energy costs. In many of these markets, AHP 13 holds a similar position as the conventional electric air-source heat pump, which it resembles. Although economical, AHP 13 is limited by the ground area needed to install and operate the soil heat source for the horizontal system, or by the geological conditions required by the vertical system. R&D should be directed to improving heat source designs and installation methods. Such research, of course, would also benefit competitive heat pump technologies.

AHP 14

The AHP 14 absorption system should be very competitive, particularly if its first cost is as low as assumed in this study. However, its use of ammonia fluid and natural gas fuel could prevent it from entering many markets. If R&D on AHP 14 changes the working fluid to another fluid that is more widely acceptable at a reasonable cost, the system could enjoy a greatly expanded market potential.

AHP 15/18

AHP 15/18, the organic Rankine/Rankine cycle (ORC) heat pump, is one of the few AHP systems that have potential in both small and medium systems. Among AHPs, AHP 2 is its major competitor ranking ahead of AHP 15/18 on life-cycle cost and payback period. R&D efforts should be oriented toward lowering the cost and bringing the system on the market sooner. If AHP 15/18 were available significantly earlier than AHP 2, it might be able to consolidate a market position and reduce its cost fast enough to remain in market segments where AHP 2 now drives it out.

In single-family applications, AHP 15/18 might suffer from some of the same consumer reluctance that plagues AHP 4. Demonstrations and field tests may be necessary to increase consumer acceptance.

FUTURE DIRECTIONS FOR ADVANCED HEAT PUMPS

The economic potential for all AHP systems, although significant, is limited by strong competition from conventional systems, especially natural gas boilers and conventional electric heat pumps. Strong competition among AHPs limits the market for each of them separately.

The overall conclusion of the market study for R&D policy in participating countries is relatively clear. To expand the market share for advanced heat pumps as a group, R&D efforts should be focused on AHPs most likely to generate market potential by 2000: the absorption, ORC-Rankine, and Diesel/Rankine systems. R&D investments in other AHP systems may be wasted because they may never succeed in breaking into the market. If they do break in, they may merely take market share away from other AHP systems without reducing the shares enjoyed by conventional oil and gas boilers and electric heat pumps. Countries contemplating R&D investment in technologies with small or zero markets in the present study should carry out a more complete analysis and consider all the systems that will compete for the heating and cooling market. Many systems may show initial promise, but over time, each market segment is likely to be dominated by one or two technologies.

Additional market for AHPs can probably be achieved more easily by attempting to change consumers purchasing decision processes than by R&D programs to improve AHP technical performance. Government subsidies and tax write-off programs can lower the first cost of the system and, like education, encourage the consumer to take a longer run view of investment decisions. If initial production runs can be stimulated by programs such as those in Sweden or Germany, then accumulated experience will lower AHP unit cost and improve system reliability more effectively than government R&D. Such experience has already reduced the cost and improved the reliability of conventional heat pumps and should be extended to AHP technologies.

To avoid producing government-dependent markets, any subsidy program should cover only the period immediately after commercialization and be phased out over time. Such subsidies should be limited to energy-efficient technologies such as AHP systems that are desirable on broad energy-political grounds. As subsidies are phased out, normal competitive forces will determine which systems will dominate in each market.

6. References +

1. Rumpf, H.-G.:
Space heating pumps using running water as a heat source.
Paper No. 70/D2, UNIPEDE, Warsaw Congress, June 11-15, (1979)

2. Heap, R.D.:
Heat pumps for domestic use.
Proc., 2nd Int. CIB Symp. on Energy Conservation in the Built Environment, Copenhagen, May 28 - June 1, (1979)

3. Gilli, P.V. et al.:
Verringerung des Energieaufwandes mit Hilfe von Wärmepumpen
(Reduction of energy consumption using heat pumps)
dbv Verlag Graz, 1. Auflage 1978, ISBN 3-7041-0007-2

4. Soil Temperature Data, 1958 - 1972.
Environment Canada, 1973

5. Waterkotte, K.:
Erdreich-Wasser-Wärmepumpe für ein Einfamilienhaus
(Groundwater heat pump for a single family house)
Elektrowärme International, 30, Jan. (1972)

6. Research and development on extended range water source heat pumps.
Report to DBC/NRC by Hydro Quebec Research Institute, Canada, March (1978)

7. Von Cube, H.L.:
Wärmequellen für Wärmepumpen (Heat sources for heat pumps).
Proc. Conf. "Wärmepumpen", Essen, 1977, Vulkan-Verlag (1978)
ISBN 3-8027-2340-6

+This is a complete list from Volume II: The Technology Survey. Not all are referred to in this Volume, but are included as a Bibliography.

8. Lorentzen, G. and Haukas, H.T.:
The effect of climate on the economy of heat pumps.
Paper E1-56, I.I.R. XVth Congress, Venice, Sept. (1979)

9. A new and novel form of district heating using thermal effluents from electricity generating plants.
Energy, Mines and Resources, Canada, Dec. (1977)

10. Heat Pumps 1978.
Report by the Ad Hoc Committee on Heat Pumps, World Energy Conference, London, July (1979)

11. Altenkirch, E.:
Absorptionskältemaschinen, VEB Verlag Technik, Berlin (1954)

12. Plank, R., Ed:
Sorptionskältemaschinen, Handbuch der Kältetechnik, Teil 7, Springer-Verlag, Berlin (1959)

13. Lotz, H.:
Behaviour of Single- and Manifold-Medium Heat Pumps, Wärmepumpen, Vulkan-Verlag, Essen (1978),
ISBN 3-8027-2340-6

14. Ellington, R.T., et al.:
The Absorption Cooling Process, Institute of Gas Technology, Research Bulletin 14 (1957)

15. Wurm, J.:
An assessment of selected heat pump systems, Institute of Gas Technology, February (1977)

16. Hermans, M.L. and Asselman, G.A.A.:
A Stirling Engine heat pump system, Paper 789274, Vol. 3, Proc. 13th Intersociety Energy Conversion Conference, (IECEC),

San Diego, August 20-25, (1978), SAE P-75 Society of Automotive Engineers Inc., U.S.

17. Sarkes, L.A., et al.:
Gas fired heat pumps: An energy technology, ASHRAE Journal, 9, (1977), No. 3, March, p 36-41

18. Friedman, D.:
Light Commercial Brayton/Rankine space conditioning system, Energy Use Management, Ed. R.A. Fazzolare, Pergamon Press (1977), p. 243-250

19. Benson, G.M.:
Free Piston Heat Pumps, Proc. 12th IECEC, August (1977), American Nuclear Society

20. Strong, D.T.G.:
Directly fired domestic heat pump development, Antriebe für Wärmepumpen, Vulkan-Verlag, Essen, ISBN 3-8027-2341-4, (1979)

21. Pegley, A.C. and Rieke, A.:
Small gas engines as prime movers for heat pumps in domestic heating, Antriebe für Wärmepumpen, Vulkan-Verlag, Essen, ISBN 3-8027-2341-4 (1979)

22. Struck, W.:
Antrieb von Wärmepumpen mittels Dieselmotor, ibid.

23. Steimle, F.:
Absorptionswärmepumpen, Wärmepumpen Technologie, Band IV, Vulkan-Verlag, Essen (1980)

24. Mostofizadeh, Ch.:
Thermische Wärmepumpe, Elektrowärme im Technischen Ausbau, Elektrowärme International Ed. A, 35, January, (1977)

25. DIN 8900 (Draft), Part 3, Deutsches Institut für Normung e.V., (1979)

26. ASHRAE Standard 90-75, American Society of Heating, Refrigeration and Airconditioning Engineers, New York, (1975)

27. Sauer, K.-L.:
Die Absorptionswärmepumpe, Sonnenenergie und Wärmepumpe, 4, March-June, (1979)

28. Lazzarin, R. et al.:
Performance predictions of a LiBr absorption air conditioner utilising solar energy, Sun: Mankind's Future Source of Energy, Ed. de Winter, F. and Cox, M., Pergamon Press, Vol. 3, p. 1572-1580, New York, (1978), ISBN 0-08-022725-2

29. McChesney, R.M.:
Solar Heated and Cooled Financial Building, ibid.

30. Naito, S.:
Absorption Refrigerator of Natural Recirculation Type, U.S. Patent 3,978,683, Sept. 7, (1976)

31. Ward, D.S.:
Solar Absorption Cooling Feasibility, Solar Energy, 22, (1979) p. 259-268

32. Boldrun, B. and Lazzarin, R.:
How to Control a Solar Powered Absorption Chiller, Paper E1-45, XV International Congress of Refrigeration, Venezia, 23-29 Sept., (1979)

33. Grallert, K.K.H.:
Entwicklungsaspekte für solare Kühlanlagen, Sanitär- und Heizungstechnik, 3, p. 293-297, (1979)

34. Schelpuk, B.C. and Hooker, D.W.:
Development programmes in solar desiccant cooling for residential buildings, Int. J. Refrigeration, 2, No 5, p. 173-179, (1979)

35. Bolzan, M. and Lazzarin, R.:
Open cycle absorption cooling devices with spray chamber regenerator and air solar collector, Paper E1-46, XV Int. Congress of Refrigeration, Venezia, 23-29 Sept., (1979)

36. Birnbreier, H.:
Absorptionswärmepumpen,Rationelle Energieverwendung, Statusbericht (1978), BMFT, pp 528-531, (1979), ISBN 3-88135-069-1

37. Kremnyor, O.A. et al.:
Study and Development of Absorption - Type Solar-Cooling Units, Paper B2-54, XV Int. Congress of Refrigeration, Venezia, 23-29 Sept., (1979)

38. Keizer, C.:
Absorption machine driven by Solar Heat, Paper B2-60, ibid.

39. Alizadeh, S. et al.:
Design and optimisation of an absorption system operated by solar energy, Solar Energy, 22, pp 149-154, (1979)

40. Absorption Heat Pump for Waste Heat, Geothermal and Solar Energy, Sanyo Electric Airconditioning Equipment Co. Ltd., 3-10-15, Hongo, Bunkyo-Ku, Japan

41. Worsøe-Schmidt, P.:
Computer simulation of the solid absorption process, Paper B2-30, XV Int. Congress of Refrigeration, Venezia, 23-29 Sept., (1979)

42. Wolf, S.:
Hydrogen Sponge Heat Pump, Paper 759196, 10th IECEC, August, (1974), A.S.M.E., N.Y.

43. Gruen, D.M. et al.:
Lanthanum Nickel Aluminium Alloy, U.S. Patent 4,142,300, March 6, (1979)

44. Guilleminot, J.J. et al.:
Utilisation d'un cycle intermittent Zeolithe 13X-H_2O pour la refrigeration solaire, Paper E1-88, XV Int. Congress of Refrigeration, Venezia, 23-29 Sept., (1979)

45. System Tepidus, U.S. Patent Appl 865,214. Tepidus AB, Box 5607, 11486 Stockholm

46. Kinsella, E.:
Development of an absorption heat pump of improved COP, Paper B3.2 to Co-ordination Meeting of Contractors, E.C. Solar Energy R+D Programme, Brussels, 8 Dec., (1978)

47. Blakeley, R.E. et al.:
The Design and development of an absorption cycle heat pump optimised for the achievement of maximum coefficient of performance, E.C. Meeting on Heat Pumps Research, Development and Application, Brussels, 27-28 September, (1978)

48. Gelesic, A. and Kampfenkel, O.:
Sorption Heat Pumps, U.K. Patent Application GB 2,002,505A, 10 August, (1978)

49. Janssen H.A. and Oelert, G.:
Development of a Primary energy driven absorption Heat Pump for domestic heating, E.C. Meeting on Heat Pumps Research, Development and Application, Brussels, 27-28 September, (1978)

50. Janssen, H.A. et al.:
Heizverfahren mit einer Absorptionswärmepumpenanlage, German Patent DE 2803118A1,

51. Macriss, R.A.:
Selecting Refrigerant-Absorbent Fluid systems for solar energy utilisation, ASHRAE Trans, 82, (1976) Pt 1, p. 975-988

52. Renz, M.:
Suitability of Solvent/Refrigerant Agents in Absorption Heat Pumps, Wärmepumpen, ed. Paul, J., Vulkan-Verlag, Essen, (1978), ISBN 3-8027-2340-6

53. Stephan, K. and Seher, D.:
Arbeitsgemische für Sorptionswärmepumpen, Klima-Kälte-Heizung, January, (1980), p. 21-32

54. Biermann, W.J. and Offenhartz, P.O.D.:
Absorption Cooling Systems, ISES, 3rd Conf. 'Use of Solar Energy for cooling of Buildings', San Francisco, Feb. 15-17, (1978), p. 23

55. Jelinek, M. et al.:
Enthalpy-concentration diagram of the system R 22 - Dimethylformamide and performance characteristics of refrigeration cycle operating with this system, ASHRAE Trans, 84, Pt 2, (1978)

56. Colosimo, D.D.:
On-site heat activated heat pumps, Paper presented at the Conference for Technical Opportunities for Energy Conservation in Appliance, ERDA, Boston, Mass., 11 May, (1976)

57. Akerman, J.R.:
Automotive air conditioning systems with absorption refrigeration, Paper 710037, SAE Automotive Engineering Congress, Detroit, Jan. 11-15, (1971)

58. Wallner, R.:
Fuel saving future alternatives for automotive A/C plants, Paper E1-57, XV Int. Congress of Refrigeration, Venezia, 23-29 Sept., (1979)

59. Felli, M. and Muttucci, F.:
p-T-x experimental data on working mixtures for absorption refrigeration, evaluation of their performance with low generator temperatures, Paper B1-83, XV Int. Congress of Refrigeration, ibid.

60. Allen, R.A. and Murphy, K.P.:
Development of a new Gas fired Absorption Heat Pump, Paper presented at ASHRAE Semiannual Meeting, Los Angeles, 3-7 Feb., (1980)

61. Malewski, W.:
Heat pump system according to the absorption principles for performance feed into district heating systems, Wärmepumpentechnologie, Band II, Ed. Paul J., Vulkan-Verlag, Essen, (1979) p. 80-85

62. A Heat Transformer that upgrades waste heat to higher temperatures, News Item, Chemical Engineering, Sept. 24, (1979), p. 67

63. Cohen, G. et al.:
Valorisation de colones à bas niveau au moyen de cycles trithermes, Entropie, 84, Nov-Dec, (1978), p. 31-37

64. Ambrose, E.R., Ed.:
Proceedings on the Symposium 'Heat Pumps - Improved Design and Performance, San Francisco, 19-22 January, (1970), ASHRAE, N.Y.

65. Comly et al.:
Heat Pumps - Limitations and Potential, GEC Report No. 75CRD185, September, (1975), Schenectady, N.Y.

66. Kirschbaum, H.S. and Veyo, S.E.:
An investigation of methods to improve heat pump performance and reliability in a northern climate, Vol. I-III, EPRI Report EM-319, January (1977)

67. Groff, G.C. et al.:
Recent investigations of air-source heat pump performance in cold climates, Paper E1-25, XV Int. Congress of Refrigeration, Venezia, 23-29 Sept., (1979)

68. Rink, T.:
Operating experiences with electric, dual source (hybrid) heat pumps, Wärmepumpentechnologie, Band I, Ed. Paul J., Vulkan-Verlag, Essen, (1978), ISBN 3-8027-2340-6

69. Wie Energieverschwendung vertraglich gesichert wird, News Item, Frankfurter Allgemeine Zeitung, 15 January, (1980)

70. Granryd, E.:
Experiences from a domestic heat pump installation, field measurements, Winter 1979, Scandinavian Refrigeration, April, (1979)

71. Freund, P.:
Domestic Applications of small heat pump units, Paper E1-12, XV Int. Congress of Refrigeration, Venezia, 23-29 September, (1979)

72. O'Connor, T.C. and Kyne, H.F.:
The performance of air source heat pumps in Northern European climatic conditions. Presented at Meeting on E.C. Heat Pumps Research, Development and Application, Brussels, 27-28 Sept., (1978)

73. McMullin, J.T. and Morgan, R.:
The development of domestic heat pumps, ibid.

74. Burgess, A.S. et al.:
Ground Source Heat Pumps, Report No HP-5, Solar Energy Project, by Acres Consulting Services Ltd. for Division of Building Research, National Research Council of Canada, June, (1978)

75. Jacobson, L. Ed.:
Nordic Symposium on earth heat pumps, Chalmers University of Technology, Göteborg, Sweden, 15-16 October, (1979)

76. Morgenssen, H.P.:
The ground as a heat source for heat pumps-performance and reactions, Paper E1-27, XV Int. Congress of Refrigeration, Venezia, 23-29 September, (1979)

77. Granryd, E.G.:
Ground source heat pump systems in a northern climate, Paper E1-82, ibid.

78. Gouldburn, J.R. and Fearon, J.:
Deep ground evaporators for heat pumps, Applied Energy, 4, (1978), p. 293-313

79. Schlosser, K.:
A mathematical analysis of the earth system for heat pumps, Scandinavian Refrigeration, 4/79 (1979), p. 267-272

80. Van der Kooi et al.:
A solar assisted heat pump system for residential heating, Paper E1-62, XV Int. Congress of Refrigeration, 23-29 Sept., (1979)

81. Bougard, J. and Pilatte, A.:
Performance des installation solaire à pompe à chaleur, Paper E1-38, ibid.

82. Lorenz, F.:
Simulation d'une chauffage residentiel par pompe de chaleur utilisant des capteurs solaires nus comme source du froid. Paper E1-81, ibid.

83. Speidel, K.:
Dornier/RWE solar house in Essen, FRG. Proc. CCMS/ISES, Int. Solar Energy conference on Performance of Solar Heating and Cooling Systems, Düsseldorf, 19-20 April, (1978)

84. Bruno et al.:
The Philips experimental house: a system's performance study, ibid.

85. Schimmelpfennig, K.:
Energy Roof for utilisation of environmental energy for space heating, Wärmepumpentechnologie, Band III, Ed. Paul J., Vulkan-Verlag, Essen, (1979), ISBN 3-8027-2345-7

86. Kremnyov, O.A. et al.:
Geothermal Air Conditioning with heat pumps, Paper E1-22, XV Int. Congress of Refrigeration, Venezia, 23-29 September, (1979)

87. Kruse, K. and Wrede, F.:
Possibilities of energy saving in compressors for refrigeration systems, Paper B2-37, ibid.

88. Paul, J. and Steimle, F.:
Speed controlled compressors - Effects on the performance of domestic heat pumps, Paper E1-68, ibid.

89. Vester, P.:
Heat Pump Compressors, 220 V. Danfoss Energy View, KF 00.V1.02, (1979)

90. Wurm, T. and Lassota, M.J.:
New rotary refrigeration compressor - its development and potential, Paper B7-42, XV Int. Congress of Refrigeration, Venezia, 23-29 September, (1979)

91. Degueurce, B. and Pascal, M.T.:
Compression directe de vapeur d'eau par un compresseur monovis, Paper B2-15, ibid.

92. Ferns, R and Niess, R.:
High temperature heat pumps make waste heat profitable, Pulp and Paper Can., 80, No 1, January, (1979), p. 37-39

93. Polman, J. et al.:
A capacity controlled free piston electrodynamic compressors, Paper B2-12, XV Int. Congress of Refrigeration, Venezia 23-29 September, (1979)

94. Cooper, W.D. and Borchardt, H.J.:
The use of refrigerant mixtures in air-to air heat pumps, Paper E1-60, ibid.

95. Kraus, W.E. and Vollmer, D.:
Thermodynamic diagrams for nonazeotropic refrigerant mixtures, Paper B1-48, ibid.

96. Saluja, S.N. et al.:
Operating characteristics of a mixed refrigerant vapour compression system. Refrigeration and Air Conditioning, March (1978), p. 83-86, 113

97. Paul, J. Ed.:
Gaswärmepumpenpraxis, Wärmepumpentechnologie, Band IV, Vulkan-Verlag, Essen (1980)

98. Fox, U. and Broer, F.:
Zur Auslegung von Gasmotor-Wärmepumpen, HLH, 30, Nr. 8, August (1979)

99. Bussmann, W.:
Single Source Gas Heat Pump for 64 Dwellings in Dortmund, Schöffenweg, Wärmepumpentechnologie, Band III, Ed. Paul J., Vulkan-Verlag, Essen (1979), ISBN 3-8027-2345-7

100. Söllner, R.:
Fast running four-stroke gas motors, Wärmepumpentechnologie, Band II, Ed. Paul J., Vulkan-Verlag, Essen (1979), ISBN 3-8027-2341-4

101. Struck, W.:
Drive of heat pumps by means of Diesel motors, ibid.

102. Study of diesel engine heat pump for heat production in housing estates and large premises, Contract 770008-9, Swedish Council of Building Research to Volvo Flygmotor, Box S-461 01 Trollhättan, Sweden (1977)

103. Sørensen, B.E.:
Diesel heat pump for district heating, Scandinavian Refrigeration, 4/79, p. 275-281

104. Pohle, J. et al.:
Gas driven heat pumps - Development and experiences, Wärmepumpentechnologie, Band I, Ed. Paul J., Vulkan-Verlag, Essen (1978), ISBN 3-8027-2340-6

105. Bauer, G.:
Small gas and diesel motor heat pumps, Wärmepumpentechnologie, Band II, Ed. Paul J., Vulkan-Verlag, Essen (1979), ISBN 3-8027-2341-4

106. Schmitz, G.:
Diesel-powered heat pumps, Wärmepumpentechnologie, Band III, Ed. Paul J., Vulkan-Verlag (1979), ISBN 3-8027-2345-7

107. MacMichael, D.B.A. and Reay, D.A.:
Feasibility and design study of a gas engined high temperature industrial heat pump, EC. Report EUR 62 37, (1979), Brussels

108. Lior, N.:
Solar energy and the steam Rankine cycle for driving and assisting heat pumps in heating and cooling modes, Energy Conversion, 16, pp. 11-123 (1977)

109. Angelino, G. et al.:
The CNPM thermal heat pump, Papers E1-16, E1-17 and E1-18, XV Int. Congress of Refrigeration, Venezia, 23-29 Sept. (1979)

110. Rousseau, I. and Noe, J.C.:
Solar powered Rankine-cycle heat pump, Paper 769202, 11th IECEC, September (1976), A.I.Ch.E., N.Y.

111. Bauer, B.:
Eine Möglichkeit der besseren Ausnutzung von Wärmepumpen zum Antrieb von Kältemaschinen und Wärmepumpen. Brennst.-Wärme-Kraft, 30, Nr. 7, July (1978)

112. Biancardi, F.R. and Melikian, G.:
Analyses and Design of an 18-Ton Solar-Powered Heating and Cooling system. Paper 789524, 13th IECEC, San Diego, August (1978), S.A.E., Warrendale, P.A.

113. Jaumotte, A. and Bougard, J.:
Development of Rankine Machines, Paper B4.1, EC. Coordination Meeting of Contractors, E.C. Solar Energy R&D programme, 8 December (1978), Brussels

114. Berclaz, G.:
Thermocompresseur frigorifique d'une type nouveau.
La Revue Polytechnique, No 1355, p. 1093-1095, November (1976)

115. Fischer, R.D. et al.:
A solar heat pump featuring pivoting tip vanes. Proc. Conf. Solar Energy Heat Pump Systems for heating and cooling Buildings. Ed. Gilman S., ERDA Doc. COO-2560, June (1975), pp. 87-97

116. Eckard, S.E.:
Test results for a Rankine engine powered vapour compression air-conditioner for 366 K heat source, Paper 769205, 11th IECEC, September (1976), A.I.Ch.E., N.Y.

117. KB. United Stirling (Sweden) AB & Co., Fack, S 20110 Malmö 1, Sweden

118. Haines, D.J.:
Stirling Cycle Apparatus, British Patent Application GB 2012886A, 16 January (1979)

119. Zacharias, F.:
Betrachtungen zum äußeren Verbrennungssystem des Stirling-Heißgasmotors, MTZ, 32, Nr. 1, January (1971)

120. Marusak, T.J. and Chiu, W.S.:
The matching of a free piston stirling engine coupled with a free piston linear compressor for a heat pump application. Paper 789272, 13th IECEC, San Diego, (1978), S.A.E., N.Y.

121. Gerstmann, J. and Friedman, Y.:
Liquid piston heat actuated heat pump and methods of operating same, U.S. Patent 4148195, 10 April (1979)

122. Vauth, R.:
A new heat pump system operating with an air refrigeration cycle, Wärmepumpentechnologie, Band I, Ed. Paul J., Vulkan-Verlag, Essen (1978), ISBN 3-8027-2340-6

123. Hise, E.C.:
Heat pumps with air-water and other fluids, Energy, Winter (1979), p. 25-27

124. Edwards, T.C.:
A new air conditioning refrigeration and heat pump cycle, ASHRAE Trans., 84, pt II, (1978), pp. 150-172

125. Ecker, A.L. et al.:
A Tabular Boundary-state Analysis of the Edwards Cycle, ASHRAE Trans., 84, pt II, (1978), pp. 173-184

126. MüllerBörner et al.:
Thermische Wärmepumpe, German Patent DE 2739689 A 1, 2 September (1977)

127. Belic, M.:
A solid-state solar powered heat transfer device, J. App. Physics, 50, Nr. 9, September (1979), pp. 5682-5685

128. Barclay, J.A.:
The design and performance of a magnetic refrigerator and heat engine, Paper B1-75, XV Int. Congress of Refrigeration, Venezia, 23-29 September (1979)

129. British Standard BS 4434:1969, Specification for requirements for refrigeration safety, London

130. Unfallverhütungsvorschrift 10a, Kälteanlagen, (VBG 20), März 1974, Verwaltungs- und Berufsgenossenschaft, W. Germany

131. Handrock, W.:
Die Beheizung des Sportzentrums Paderborn mit einer Gaswärmepumpe, Int. Fachtagung "Wirtschaftliche Energieversorgung mit Erdgas in Bädern, Sport und Freizeitbauten", 23-24 Nov. (1978), München, Vulkan-Verlag, Essen, ISBN 0172-3154

132. Bußmann, W.:
Beheizung des Freibades Dortmund-Wellinghofen mit einer Gaswärmepumpe (Betriebserfahrungen aus zwei Jahren), ibid.

133. Petersen, B.:
Diesel heat pumps for district heating plants and for the heating of large housing blocks. Final report for the E.C. Project No. 373-78-1 EEDK, Burmeister & Wain A.S., Torvegade 2, 1449 Copenhagen K, Denmark

134. Kennedy, D.M.:
A.G.A. Air Conditioning Research - past prospects, present perspectives. Proc. 2nd Conf. on Natural Gas Research and Technology, Atlanta, Georgia, 5-7 June (1972)

7. Acknowledgements and National Teams

7.1. Acknowledgements

We wish to acknowledge the contributions of the 13 national teams for their collection of the national statistics and Case Study data, the IEA secretariat for their help and assistance; to the developers of the advanced heat pumps for their assistance in providing information on their systems; to the many experts with whom discussions were held during the project; and to the staff of the Battelle-Institut e.V. for their assistance with the compiling of the report.

7.2. National Teams

Austria

P.V. Gilli
F. Schabkar

Graz University of Technology
Kopernikusgasse 24
A-8010 Graz, Austria

Belgium

J. Berghmans

Universiteit Leuven
Instituut Mechanica
Celestynenlaan 300A
B-3030 Heverlee, Belgium

Canada

R.L.D. Cane
D. Stephenson

National Research Council
Div. Building Research
Montreal Rd.
Ottawa K1A OR6, Canada

H.F. Sullivan

University of Waterloo
Mechanical Engineering Dept.
Waterloo, Ontario, Canada
N2L 3G1

Denmark

S.R. Jacobsen	Dansk Kedelforening Gladsaxe Møllevej 15 DK-2860 Søborg, Denmark
A. Korsgaard	Teknologisk Institut Gregersensvej DK-2630 Taashup, Denmark
I. Lemming	Ministry of Energy Strandgade 29 DK-1401 Copenhagen K, Denmark
B. Petersen	Consulting Engineers Ellehegnet 10 DK-2950 Vedbak, Denmark

Germany

D. Hodgett G. Oelert R. Rudolph	Battelle-Institut e.V. Am Römerhof 35 D-6000 Frankfurt am Main Federal Republic of Germany
P. Küppers U. Plantikow	Kernforschungsanlage Jülich Projektleitung Energieforschung Postfach 1913 D-5170 Jülich Federal Republic of Germany
H. Loewer	Reinhold-Schneider-Str. 135 D-7500 Karlsruhe 51 Federal Republic of Germany

Italy

B. Buscema P. Carotenuto G. Missoni	AGIP SpA Via Giorgione 59 Roma, Italy
A. Castagnone G. La Gioia	AGIP SpA Via Due Macelli 66 I-00187 Roma, Italy
V. Recchi	O.T.B. Zona Industriale I-70123 Bari, Italy

Japan

A. Hosaka S. Kakuno T. Tsuchiya K. Ueda S. Wakabayashi	Ministry of International Trade and Industry Agency of International Science and Technology 1-3-1 Kasumigaseki Chiyodaku Tokyo, Japan
T. Mukai	Japanese Delegation O.E.C.D. 7, Avenue Hoche F-75008, France

Netherlands

H. van der Ree	TNO-MT Laan van Westenenk 501 Apeldoorn, Netherlands

Spain

J.C. Llorente	Centro de Estudios de la Energia Ministerio de Industria y Energia Agustin de Foxa 29 Madrid, Spain

Sweden

B. Backström T. Berntsson	Chalmers University of Technology Gothenburg, Sweden
T. Boström A. Boysen B. Lundquist H. Olson	Swedish Council for Building Research Sankt Göransgatan 66 S-11230 Stockholm, Sweden
B. Svedemar	National Swedish Board for Techn. Development S-10072 Stockholm, Sweden

Switzerland

J.M. Leuenberger J.C. Mayor	Swiss Federal Institute for Reactor Research CH-5303 Würenlingen Switzerland

United Kingdom

V.A. Eustace D.B.A. MacMichael D. Reay	International Research and Development Co. Ltd. Fossway Newcastle upon Tyne NE 62YD United Kingdom
M.J. Szolomicki	Glynwed Ltd. Cranmore Drive Shirley Solihull West Midlands United Kingdom

United States

P.G. Bos J. Novosielski	Resource Planning Associates 44 Brattle Street Cambridge, Mass. 02138, U.S.A.
F.A. Creswick P.D. Fairchild	Oak Ridge National Laboratory P.O.Box Y Oak Ridge, TN 37830, U.S.A.
M. Grapin K.C. Mead S. Pariente	Resource Planning Associates 28, Avenue de Messine F-75008 Paris, France
J.D. Ryan	U.S. Department of Energy 20 Massachusetts Avenue; NW Mail Stop 2221-C Washington, D.C. 20585, U.S.A.
A.C. Quinn	Resource Planning Associates 50 Church Street Cambridge, Mass. 02138, U.S.A.

IEA Secretariat

B. Kramer	International Energy Agency 2, Rue André Pascal F-75775 Paris Cedex 16, France