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Programme of Research and Developement on
Advanced Heat Pump Systems

Annex 5

Integration of Large Heat Pumps into District Heating and Large Housing Blocks

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Prepared by:
Benny Petersen, M.Sc., B.Com.

innosys
Consulting Engineers
Heimdalsgade 14
DK-2200 Copenhagen N

CONTENTS

LIST OF FIGURES AND TABLES.................................. 8

LIST OF SYMBOLS AND ABBREVATIONS.............................10

1 EXECUTIVE SUMMARY...11

CONCLUSIONS...11

2 GENERAL INTRODUCTION......................................15

2.1 Project Organization....................................15

2.2 Objectives of the Study.................................16

2.3 Programme of Work.......................................17

3 HEAT PUMP SYSTEMS...18

3.1 The Principle of a Heat Pump............................18

3.2 Compressor and Absorption Heat Pumps....................20

4 HEAT PUMP APPLICATIONS....................................21

4.1 Selection of Case Studies...............................23

Air as Heat Source..24

Water as Heat Source..24

Sewage Water as Heat Source.................................25

River and Sea Water as Heat Source..........................25

Other Heat Sources..26

5 COMPARISON OF DIFFERENT HEAT PUMP PROJECTS................28

5.1 Investment in Heat Pumps................................28

5.2 Expenditures for a Heat Pump Plant......................29

5.3 Energy Savings..30

5.4 Energy Cost Savings.....................................31

5.5 Economy Calculations for Changed Applications...........33

5.6 Key Figures for the Case Studies........................35

5.7 International Comparison................................38

5.7.1 Case Study Calculations for Germany...................39

5.7.2 Case Study Calculations for Denmark...................41

5.7.3 Case Study Calculations for Italy.......... 43

5.7.4 Case Study Calculations for Sweden.......... 45

5.7.5 Variations in Energy Prices and Pay Back Time.............................. 47

5.8 Conclusions from the Comparison.................... 50

5.9 Evaluation of Experience.......................... 51

5.10 Priorities for Future R, D & D..................... 55

6 INTRODUCTION TO CASE STUDIES............................ 56

6.1 Natural Gas Driven Heat Pump in Erlangen, W. Germany.............................. 57

6.1.1 Owner and Location........................... 57

6.1.2 Heating System............................... 57

6.1.3 The Heat Pump System......................... 57

6.1.4 Calculated Operation Conditions............. 59

6.1.5 Fuel Savings................................. 59

6.1.6 Investment and Operation.................... 60

6.1.7 Economy...................................... 60

6.1.8 Experience from Planning and construction... 61

6.1.9 Experience from Test and Measurement........ 61

The Result of the Plant..................... 62

6.1.10 Conclusions from the Plant.................. 63

6.1.11 Key Figures for 0.7 MW Gas Engine Driven Heat Pump.......................... 63

6.2 Absorption Heat Pump Project in Saarberg.......... 65

6.2.1 Owner and Location........................... 65

6.2.2 Heating System............................... 65

6.2.3 The Absorption Heat Pump System............. 66

6.2.4 Operation Conditions........................ 66

6.2.5 Fuel Savings................................ 69

6.2.6 Investment and Operation (1980)............. 69

6.2.7 Economy..................................... 70

6.2.8 Experience from Projecting and Construction of the Absorption Heat Pump............ 71

6.2.9 Experience from Test Running of the Plant... 72

6.2.10 Conclusions................................ 73

6.2.11 Key Figures for a 4.0 MW Absorption Heat Pump.................................. 74

6.3 Absorption Heat Pump in Waiblingen, W. Germany..... 75

6.3.1 Owner and Location.......................... 75

6.3.2 Heating System.............................. 75

6.3.3 The Heat Pump system........................ 77

6.3.4 Calculated Operation Conditions............. 77

6.3.5 Fuel Savings of Absorption Heat Pump Plant.. 79

6.3.6 Investment and Operation (1982)............. 80

6.3.7 Economy..................................... 81

6.3.8 Experience from Construction of the Plant... 82

6.3.9 Experience from Test Running................ 83

6.3.10 Comments and Conclusions on the Project..... 85

6.3.11 Key Figures for a 2.5 MW Absorption Heat Pump.................................. 86

6.4 Natural Gas Driven Heat Pump for Paderborn Sports Centre, W. Germany......................... 87

6.4.1 Owner and Location.......................... 87

6.4.2 Heating System.............................. 87

6.4.3 The Heat Pump system........................ 87

6.4.4 Calculated Operation Conditions............. 89

6.4.5 Fuel Savings................................ 90

6.4.6 Investment and Operation.................... 90

6.4.7 Economy.................................... 91

6.4.8 Experience from Projecting and Building..... 92

6.4.9 Experience from Test and Measurement........ 92

6.4.10 Conclusions on the Project................. 93

6.4.11 Key Figures for a 4.6 MW Gas Engine Driven Heat Pump........................... 94

6.5 Two-Media Solution in a Compression and Absorption Heat Pump Process in Mannheim, W. Germany.......... 95

6.5.1 Owner and Location......................... 95

6.5.2 Heating System............................. 95

6.5.3 The Heat Pump System....................... 96

6.5.4 Calculated Operation Conditions............ 97

6.5.5 Fuel Savings............................... 99

6.5.6 Investment and Operation................... 100

6.5.7 Economy.................................... 100

6.5.8 Experience from Construction of the Plant... 101

6.5.9 Experience from Test and Measurement........ 101

6.5.10 Conclusions and Comments.................. 102

6.5.11 Key Figures for a 0.4 MW Two-Media Gas Engine Driven Heat Pump................ 103

6.6 District Heating and Cooling in Reggio Emilia, Italy.................................... 104

6.6.1 Owner and Location......................... 104

6.6.2 Heating and Cooling System................. 104

6.6.3 The Heat Pump System....................... 107

6.6.4 Calculated Operation Conditions............ 109

6.6.5 Fuel Savings............................... 110

6.6.6 Investment and Operation................... 110

6.6.7 Economy.................................... 111

6.6.8 Experience during Construction of the Plant. 111

6.6.9 Experience from Test and Measurement....... 114

6.6.10 Conclusions from the Plant.................. 118

6.6.11 Key Figures for the Total Energy Scheme..... 118

6.7 Diesel Driven Heat Pump for District Heating in Frederikshavn, Denmark.......................... 119

6.7.1 Owner and Location.......................... 119

6.7.2 Heating System.............................. 119

6.7.3 The Heat Pump System........................ 119

6.7.4 Calculated Operation Conditions............. 122

6.7.5 Fuel Savings................................ 122

6.7.6 Investment and Operation.................... 123

6.7.7 Economy..................................... 123

6.7.8 Experience during Construction of the Plant. 125

6.7.9 Experience from Test and Measurement........ 125

6.7.10 Conclusion from a 10 MW Diesel Driven Heat Pump........................... 127

6.7.11 Key Figures for a 10 MW Diesel Driven Heat Pump........................... 128

6.8 Natural Gas Driven Heat Pump for District Heating in Ejby, Denmark.......................... 129

6.8.1 Owner and Location.......................... 129

6.8.2 Heating System.............................. 129

6.8.3 The Heat Pump System........................ 129

6.8.4 Calculated Operation Conditions............. 130

6.8.5 Fuel Savings................................ 132

6.8.6 Investment and Operation.................... 133

6.8.7 Economy..................................... 133

6.8.8 Experience during Construction of the Plant. 135

6.8.9 Experience from Test and Measurement........ 137

6.8.10 Conclusions from The Plant.................. 139

6.8.11 Key Figures for a 1.2 MW Gas Engine Driven Heat Pump........................... 140

6.9 Electrically driven HeatPump in Kungaelv, Sweden... 141

6.9.1 Owner and Location........................ 141

6.9.2 Heating System............................ 141

6.9.3 The Heat Pump System...................... 141

6.9.4 Calculated Operation Conditions........... 142

6.9.5 Fuel Savings.............................. 143

6.9.6 Investment and Operation.................. 143

6.9.7 Economy................................... 144

6.9.11 Key Figures for a 2.6 MW Electrically Driven Heat Pump......................... 145

6.10 District Heating and Ice Rink Cooling in Ruddalen, Sweden.............................. 146

6.10.1 Owner and Location....................... 146

6.10.2 Heating System........................... 146

6.10.3 The Heat Pump system..................... 146

6.10.4 Calculated Operation Conditions.......... 147

6.10.5 Fuel Savings............................. 148

6.10.6 Investment and Operation................. 149

6.10.7 Economy.................................. 150

6.10.11 Key Figures for a 1.4 MW Electrically Driven Heat Pump........................ 151

6.11 Electrically Driven Heat Pump in Sala-Heby, Sweden............................. 152

6.11.1 Owner and Location....................... 152

6.11.2 Heating System........................... 152

6.11.3 The Heat Pump system..................... 152

6.11.4 Calculated Operation Conditions........... 153

6.11.5 Fuel Savings.............................. 154

6.11.6 Investment and Operation.................. 154

6.11.7 Economy................................... 155

6.11.8 Experience during Construction of the Plant............................... 156

6.11.9 Experience from Test and Measurement...... 156

6.11.10 Conclusions from the Plant............... 157

6.11.11 Key Figures for a 3 MW Electric Heat Pump.................................. 158

6.12 Electrically Driven Heat Pump using Sea water as Heat Source in Visby, Sweden..................... 159

6.12.1 Owner and Location........................ 159

6.12.2 Heating System............................ 159

6.12.3 The Heat Pump System...................... 159

6.12.4 Calculated Operation Conditions........... 161

6.12.5 Fuel Savings.............................. 161

6.12.6 Investment and Operation.................. 162

6.12.7 Economy................................... 162

6.12.11 Key Figures for a 8.5 MW electric Heat Pump.................................. 163

7 REFERENCES...................................... 164

LIST OF FIGURES AND TABLES

List of figures

Fig. 1. The Principle of a Heat Pump........................ 19

Fig. 2. Heat Pump for Industrial Waste Heat................ 19

Fig. 3. Structures of Steam Turbine and Absorption Heat Pump....................................... 21

Fig. 4. Energy Flow in an Electric Heat Pump............... 22

Fig. 5. Energy Flow in a Gas or Diesel Engine Driven Heat Pump................................ 22

Fig. 6.1.1 Principal Lay Out of Heat Pump in Erlangen....................................... 58

Fig. 6.2.1 Principal Scheme of the Saar Absorption Heat Pump Project.............................. 65

Fig. 6.2.2 Absorption Heat Pump Flow Chart (Saar).......... 67

Fig. 6.3.1 The Heating System (Waiblingen)................ 76

Fig. 6.3.2 The Annual Heat Output (Waiblingen)............ 76

Fig. 6.3.3 Principal Lay-Out of the Heat Pump (Waiblingen)..................................... 78

Fig. 6.4.1 Heating System (Paderborn)...................... 88

Fig. 6.5.1 Heat Pump System (Mannheim)..................... 96

Fig. 6.5.2 Conditions for Two-Media Heat Pump............. 97

Fig. 6.5.3 Heat Pump (Mannheim)............................ 98

Fig. 6.6.1 Total Energy Plant Flow Diagram - Winter Operation (Reggio Emilia)...................105

Fig. 6.6.2 Total Energy Plant Flow Diagram- Summer Operation (Reggio Emilia)...................106

Fig. 6.7.1 Sankey Diagram for Diesel Heat Pump (Frederikshavn)..............................119

Fig. 6.7.2 Energy Balance for Diesel Heat Pump Equipped with Power Generator (Frederikshavn)............121

Fig. 6.8.1 Principal Lay Out (Ejby).......................131

Fig. 6.9.1 Brine System (Kungälv)..........................142

Fig. 6.10.1 Cooling and Heating System (Ruddalen)..........147

Fig. 6.11.4 Fundamental Hook-Up (Sala-Heby)................153

Fig. 6.12.1 Principal Lay Out (Visby)......................160

List of Tables

Table 4.1.1 Heat Pump Plant by Prime Mover and Heat Sources........................... 27

Table 5.6.1 Basic Information of Heat Pump Plants in the Case Studies.................. 36

Table 5.6.2 Key Figures for Heat Pump Plants in the Case Studies........................ 37

Table 5.7.1 Case Studies Installed under German Economic Conditions..................... 40

Table 5.7.2 Case Studies Installed under Danish Economic Conditions.................... 42

Table 5.7.3 Case Studies Installed under Italian Economic Conditions.................... 44

Table 5.7.4 Case Studies Installed under Swedish Economic Conditions................... 46

Table 5.7.5 Energy Price Index in Participating Countries.............................. 47

Table 5.7.6 Pay Back Times of Selected Projects vs. Countries.......................... 49

Table 5.9.1 Pay Back Periods with Respect to the Heat Source.......................... 52

Table 6.6.9.1 Reggio Emilia, Total energy Balance........115

Table 6.6.9.2 Reggio Emilia, Economic Balance............116

Table 6.6.9.3 Investment for the RETE Project............117

LIST OF SYMBOLS AND ABBREVATIONS

I =	Investment cost
a =	annuity (reciproc pay back time)
OC =	Operating cost per year
ED =	Energy delivered from the plant
ER =	Energy ratio
ER_O =	Energy ratio for a reference boiler
ES =	Energy saving per year
ES% =	Energy saving in per cent
P_O =	Price of reference boiler fuel
P =	Price of heat pump fuel
ESC =	Energy cost saving per year
ESC% =	Energy cost saving in per cent
EER =	Economic Energy Ratio = ER/P
EER_O =	Economic Energy Ratio of reference boiler
DM =	German Marks
D.kr. =	Danish Crowns
S.kr. =	Swedish Crowns
LIT =	Italian Lire
toe =	Tons of Oil Equivalent
IES =	Investment per yearly energy saving
COP =	Coefficient of performance
PER =	Primary Energy Ratio
=	COP divided with fuel supply efficiency

1. EXECUTIVE SUMMARY

This report contains the results and conclusions of the study of integration of large heat pumps into district heating and large housing blocks, which is the fifth study within the IEA-programme of Research and Developement on Advanced Heat Pump Systems.

The study was carried out and jointly financed by four participating countries: Denmark, Germany, Italy and Sweden.

The main objective of the study was to assess operating experience and economic performance of large heat pumps (1-10 MW) in district heating and large housing block (block centrals)applications.

The objective has been met by a study of ongoing demonstration projects and studies of large heat pumps, which were either in operation or under construction at the beginning of the study.

A documentation in English on each of the twelve chosen projects has been prepared in order to perform a uniform and schematic description of the projects. This description is included in chapter 6 of the report.

The necessary economic theory for composing key figures for the projects has been developed. These key figures has been used to estimate the economic result of all the heat pump plants according to the different economic conditions existing in the participating countries. Investment and energy prices are regulated to price level of January 1985.

CONCLUSIONS

The case studies selected for this annex have demonstrated energy savings from 17% to 77% and energy cost savings in the same range. The projects show with revised prices pay back periods from 1.7 years to an infinite number of years. Eight of twelve projects have pay back periods less than 10 years.

Under Danish economic conditions 8 projects are estimated to have a pay back time between 2 and 6.8 years. Under German conditons 8 projects will have a pay back time less than 10 years. Italian conditions give a rough estimate of 6 projects with a pay back time between 1.9 and 9.9 years. With Swedish economic conditions 8 types of projects are calculated to give pay back periodes from 1.7 to 8.6 years.

The economic evaluation shows that large electric heat pumps are competitive in Denmark and especially in Sweden, where electricity prices are extremely low.

Gas and diesel engine driven heat pumps are economically feasible in all the countries as a base load supply for district heating. The economy becomes clearly better with the size of the plant and the lower limit is expected to be about 1 MW.

Absorption heat pumps have been demonstrated as technically feasible and the projected efficiencies have been demonstrated. These plants show pay back times from 5.3 to 9.7 years, which is only slightly higher than engine driven heat pumps.With further developments the absorption heat pump could be more economical attractive.

The advantage of absorption heat pumps is, that they can use low cost fuels, but the fuel cost saving will in this case also be reduced and thereby the economy becomes less attractive. It may also be possible to find special applications, where absorption heat pumps have advantages to compression heat pumps.

If the selected projects are analysed with respect to the heat sources, the following range of pay back periods has been calculated:

Sewage water	1.7 - 8 years
Riwer and sea water	1.9 - 16 Years
Ice and misc.	3.1 - 13 years
Ambient air	3.9 - ∞ years

The most general experience in the case studies is that the heat pumps have been build too large for the actual heat demand. The projecting has been done on the basis of calculated heat losses for new buildings or an overestimated increase in the connection of new houses to the district heating net. This has often resulted in part load operation of the heat pump most of the year, which is unfavourable to the efficiency of the heat pump and the investment has been too large compared with the energy savings.

When the heat pump has been projected as a monovalent heat pump system on the basis of calculated heat demand, the economical result has in several cases been disastrous.

The conclusion is that large heat pumps should always be build as base load plants, preferably in series with peak load boilers and in winter time only preheating the district heating water.

Several heat sources have been demonstrated as well economical as technically feasible. Further developement can surely imply a reduced cost and better efficiency for some of the heat sources. Especially air as the most general heat source and heat exchangers for water near the freezing point are recommended for future R, D & D.

Also evaporators for using the heat of freezing are recommended for future research, as these in many cases would solve the problem in finding an appropriate heat source.

One should pay attention to heat pumps, where both the warm and the cold side are used. The case study of an ice rink is one example, but also cold stores, cooling of computer facilities and air conditioning of shopping centres could be mentioned.

Many factories in the plastic or chemical industry have a need for cooling water, which could be delivered from at district heating central with heat pumps.

The heat pumps in the case studies show refrigeration cycles from the simple ones, often used for refrigeration and to more complex integrated systems. The operation circumstances for a heat pump are normally more variating and complex than that of a refrigeration unit. Several cases have shown that components or the design of the system make it necessary to use regulating equipment which reduces the efficiency of the heat pump.

A large variation in heat pump cycles can be made and R, D & D is needed to investigate the posibilities of achieving higher efficiencies with only marginal increase in investment.

The diesel and gas engine driven heat pumps already show a good economy, but further demonstration projects combined with the above mentioned improved heat pump cycle are needed.

In areas without natural gas more attention should be paid to the possibilities of reducing the fuel consumption by 40-60% by diesel driven heat pumps, even though the remaining fuel consumption will be the expensive diesel fuel.

The technology in the refineries is expected to give a large fraction of diesel fuel with better cracking processes and in this way the energy saving becomes the most important argument.

2 GENERAL INTRODUCTION

The study of integration of large heat pumps into district heating and large housing blocks was put into force in February 1982 as annex of an "Implementing Agreement for a Programme of Research and Development on Advanced Heat Pump Systems". The annex should remain in force for an initial period of three years.
The four participating countries were Denmark, Germany, Italy, and Sweden.

2.1 Project Organization

INNOSYS was appointed by the government of Denmark to act as operating agent for the annex.
Each participant has nominated a representative to assist the operating agent in carrying out this annex by collecting and transferring experience from the national projects within this field.
The nominated delegates were:

Denmark:	Benny Petersen, M.Sc., B.Com., Innosys, Copenhagen.
W. Germany:	Dr. Ing. Schlagheck, Kernforschungsanlage/PLE, Jülich.
Italy:	Dr. Ing. Ettore Piantoni, CISE, Reggio Emilia.
Sweden:	Prof.Bernt Bäckström, Chalmers University of Technology, Göteborg.

The study has been performed by a working group which besides the nominated delegates has had the following members:

Germany:	Dr. Ing. Hummelsiep, Saarberg Fernwärme, Saarbrücken.
	Dipl. Ing. Mučić, Stadtwerke Mannheim, Mannheim.
	Dr. Ing. Oartloff, Stadtwerke Waiblingen, Waiblingen.
Italy:	Dr. Ing. Recchi, National Research Council, Bari.
	Prof. Ing. Ruggiero, University of Bari, Bari.
	Prof. Ing. Fortunato, University of Bari, Bari.
Sweden:	Research Secr. Bengt Lundquist, Swedish Council for Building Research, Stockholm.

2.2 Objectives of the Study

The main objective of this task was to assess operating experience and economic performance of large (1-10 MW) heat pumps in district heating and large housing block (block central) applications. An additional objective was to assess experience with heat sources appropriate for large heat pumps, if additional resources could be provided by an increase in participants.

The technical experience from installation and operation of large heat pumps is expected to lead to reduced cost of future demonstration projects as exchange of information of lessons learned

- gives a better utilization of existing ex-

perience
- reduces the risk of mistakes in new projects
- identifies subjects for new research activities.

2.3 Programme of Work

The annex was started with an investigation or study of 43 important heat pump projects, most of which were either in operation or under construction.
Among these heat pump projects twelve projects were selected for further investigations. The selection took place according to the preferences of the delegates in the working group with attention to a broad representation of different applications of large heat pumps.
Available reports of the selected projects have been distributed at the working meetings, and lectures have been arranged with several experts in large heat pumps.

The description of each project included in this report has been made by the operating agent in order to perform a uniform and schematic description of the heat pumps as the basis of a comparison of the projects.
The necessary theory for an economic comparison of the projects has been developed, and the key figures for this comparison have been calculated for each project in a uniform way.

The operating agent has made a condensed description of the experience gained in the projects, where test reports have been available before the end of the annex.

Combined with the existing know-how of large heat pumps, the experience described in the test reports has been the basis of a brief evaluation of the demonstration projects, although the evaluation of the project and the heat sources belong to an extention of the annex by increased resources.

3 HEAT PUMP SYSTEMS

Generally, heat pumps are technical devices which transform energy from one available quality to the specific quality needed for a process or for heating purposes. The purpose of the device is economic saving, normally as a result of energy saving.

The size of investment in a large heat pump often justifies a tailor-made plant, for which reason it is difficult to describe a standard heat pump.

However, heat pumps can be classified according to a few fundamental principles, from which hundreds of variants can be designed. Especially for industrial use, the number of variations is as large as the number of industries. Even heat pumps for heating purposes show large variations due to application factors such as availability and price of different fuels, and the available heat sources for the heat pump.

3.1 The Principle of a Heat Pump

A heat pump is an energy transformer which normally uses a high grade energy like electricity, fossil fuel or high temperature heat to transform a low grade energy into an intermediate temperature which is usable.

In this way the amount of usable energy is the sum of consumed high grade energy and transformed low temperature energy. The amount of usable energy can be expressed as the fuel consumption multiplied by an Energy Ratio factor (ER) which characterizes the heat pump and its operation conditions. This Energy Ratio is called the Coefficient of Performance (COP), when the fuel is electricity.

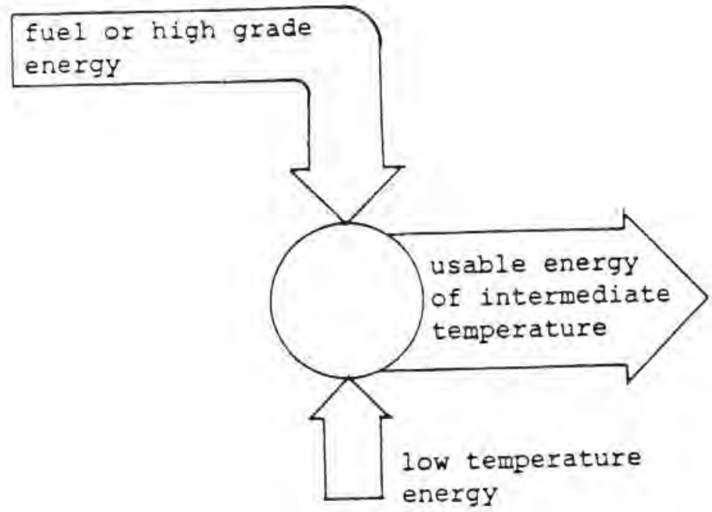


Fig.1. The Principle of a Heat Pump.

The heat production divided by the total consumption of fuel including transport, refining etc. is normally called the Primary Energy Ratio (PER). For engine driven heat pumps PER is a few percentages lower than the ER, while the PER for an electric pump is about one third of the COP on account of the conversion factor in power plants.

In a few cases the heat pump is designed for a large quantity of intermediate temperature energy (normally industrial waste heat) and thus transforms part of this energy into a higher temperature and degrades the remaining energy to a heat sink like cooling water.

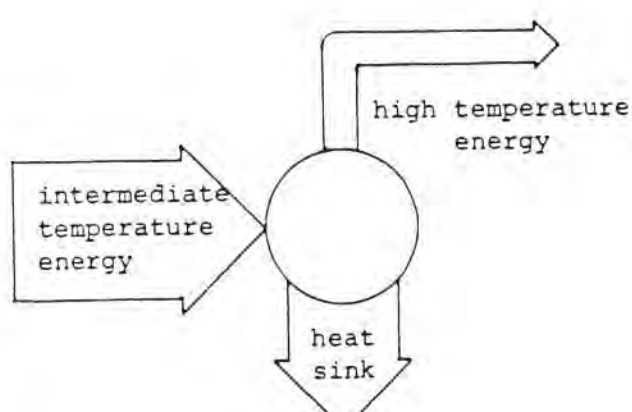


Fig.2. Heat Pump for Industrial Waste Heat

These specially designed heat pumps will not be included in this project. In Japan this principle has been demonstrated for converting industrial waste heat into useful heat at higher temperature for heating purposes.

3.2 Compressor and Absorption Heat Pumps

The heat pump principle may be divided into two main groups. The most common type is based on mechanical compression of a refrigerant or a mixture of refrigerants which circulate in a closed loop circuit. The vapour compressor consumes high grade energy as mechanical power, and this mechanical power can be produced by electricity by means of an electric motor or by fossil fuel with a diesel engine, gas engine, gas turbine, or steam turbine.
An electrically driven heat pump is in principle driven by a steam turbine, and the electricity is only used to convey the mechanical energy from the power plant to the compressor.

The other main type is absorption heat pumps, in which cold refrigerant vapour is absorbed in an absorbent, thereby releasing the heat of absorption at an intermediate temperature. After pressurizing and heating the absorbent, the refrigerant is boiled out of the absorbent by means of high temperature energy, and the energy is supplied at an intermediate temperature, when the refrigerant is condensing again. The basic thermodynamic principles of a steam turbine driven compressor heat pump and an absorption heat pump have comparable structures, cf. figure 3.

In theory, the different heat pump systems can be designed for nearly all operation conditions, but the resulting efficiency will turn out differently. The final choice between heat pump systems is not only a question of efficiency and detailed techniques, but a complex balancing of techniques and economy, where the price and availability of different fuels for the heat pump, the investment and the efficiency for the actual application are important parameters.

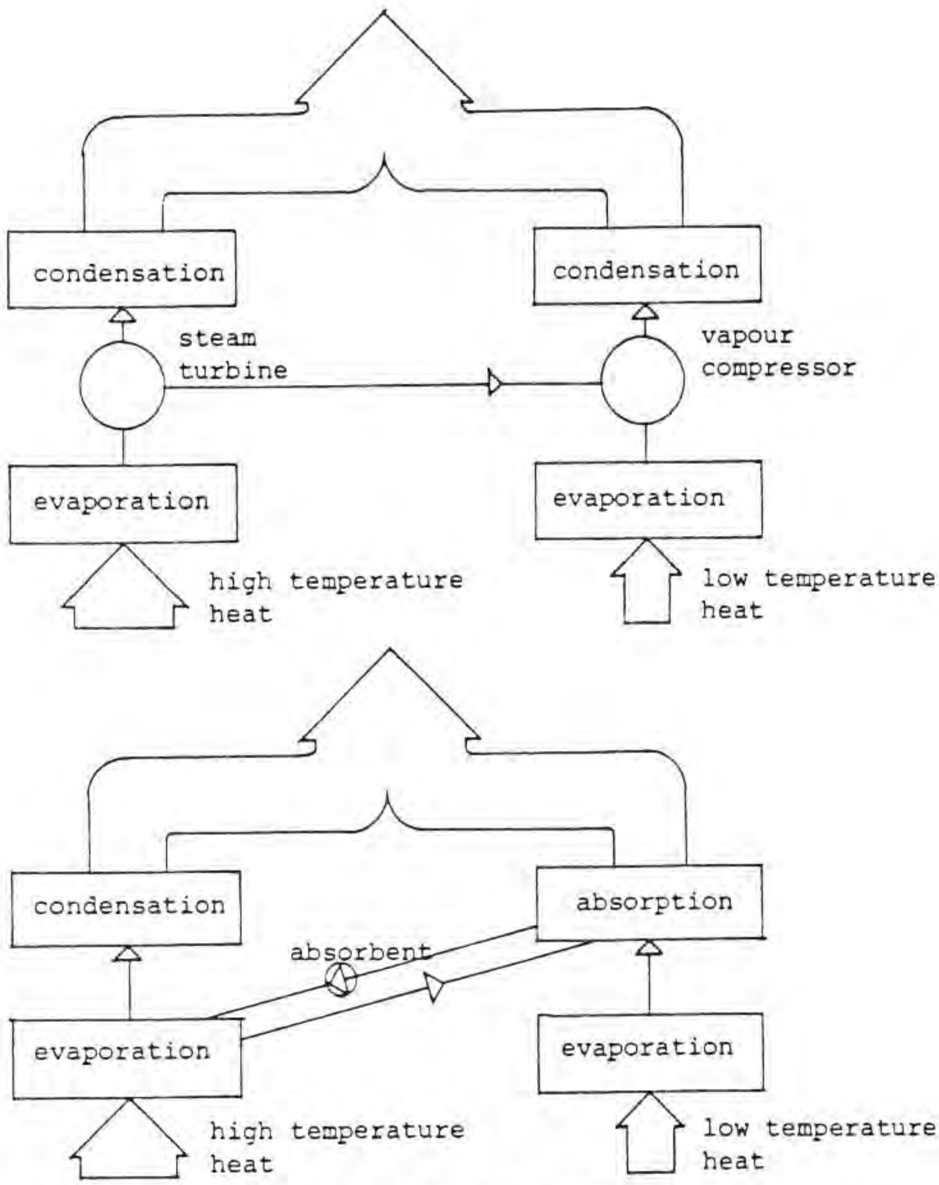


Fig.3. Structures of Steam Turbine and Absorption Heat Pump

4 HEAT PUMP APPLICATIONS

The most general heat pump application for district heating and heating of large housing blocks is a heat pump using the energy contents of the surroundings as heat source and producing hot water for the heating system.

A compressor heat pump can either be driven by an electric motor or by an engine.

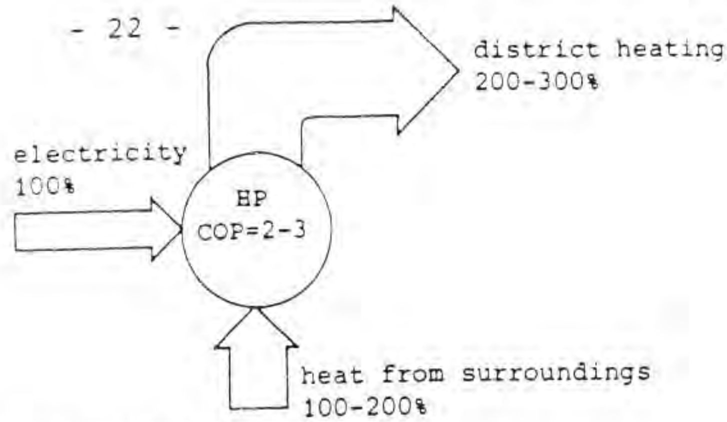


Fig.4. Energy Flow in an Electric Heat Pump

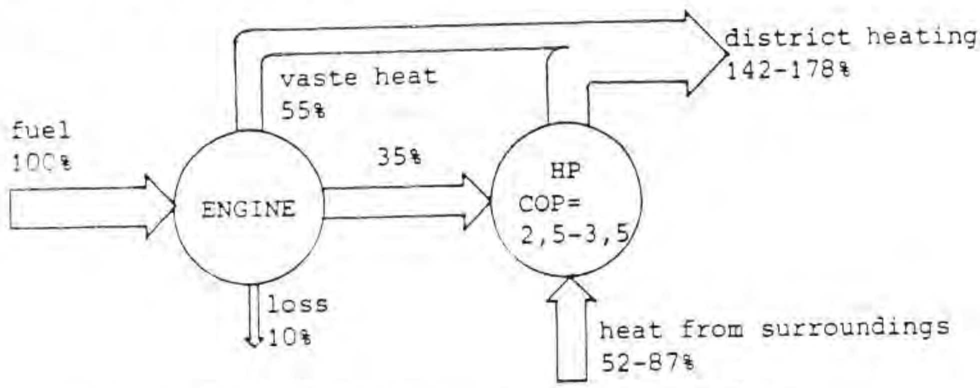


Fig 5. Energy Flow in a Gas or Diesel Engine Driven Heat Pump

Figures 4 and 5 show typical energy flows in an electric heat pump and a gas engine heat pump. Apparently, the electric heat pump is more efficient, as it has an energy ratio or coefficient of performance of 200 to 300%, whereas the gas engine driven heat pump has an energy ratio of 142 to 178%. However, these figures do not take into account that primary energy consumption for electricity production is often 2.5 to 3 times as large as the electricity output (except hydro power), while natural gas consumption is the consumption of primary fuel. Energy calculations like these show neither the energy conservation nor the best economic choice.

Unlike the energy ratio of a boiler, the energy ratio of a heat pump is very sensitive to the operation conditions in the form of the temperature of the heat source and both the supply and return temperature of the district heating water.
Relatively to the fuel consumption, the heat production or energy ratio increase is about 2% for each ^{o}C higher heat source temperature.

The sensitivity to the temperature of the heating net is dependent on the plant design, but as a rule-of-thumb the sum of the sensitivity in relation to the two temperatures is about equal to the sensitivity of the heat source temperature.

For a compressor heat pump the capacity of the plant is also sensitive to the heat source temperature, as the production decrease is about 4% for each ^{o}C decrease of the heat source temperature. For an absorption heat pump this is only the case to a minor extent.

As the operation condition has large influence on the energy ratio and thereby the energy saving and the economy, the operation condition should be borne in mind in the following chapters, when different heat pump plants are compared.
It has not been possible and it has no meaning to recalculate the operation of each demonstration plant to some sort of standard conditions, as the different plants have been chosen and designed for the actual operation conditions.

4.1 Selection of Case Studies

Chapter 6 is divided into 12 parts, 6.1 - 6.12, each containing a description of one of the 12 heat pump plants selected for this project.

A selection procedure has been carried out in co-operation with the participants. The selection has been made with regard to presenting a spectrum of typical heat sources and with respect to different possibilities of prime movers and heat pump systems.
Because of the high sensitivity to the heat source temperature, industrial waste heat has been excluded from this project. The case studies selected for the project only deal with natural heat sources which only show little variation between countries and thereby the results become more comparable.

The heat sources are represented in 4 groups; outside air, sewage water, river or sea water and various heat sources as ice rink, district heating, return water, and air conditioning.

Air as Heat Source

The most available heat source is always ambient air. Since you can nearly almost always choose air as heat source it will in practise become the least economical heat source of any interest. The econmy is disfavoured by a high investment in the evaporator compared with an evaporator for water. and the temperature of air show a larger variation during the year. The temperature is lowest when the demand for heat is highest, and the low temperature decreases the capacity of the heat pump in this period.

Besides, the air evaporator has to have defrosting equipment which increases the investment and reduces the heat output. The energy consumption of the fan in the evaporator typically accounts for 10% of the power consumption of the heat pump and thereby the operation economy of the heat pump plant is reduced.

Ambient air as heat source is demonstrated in a 0.7 MW gas engine driven heat pump in project No. 1 situated in Erlangen, W. Germany, and a 2.6 MW electrically driven heat pump in project No. 9 for Kungälv, Sweden.

A combination of air and water as heat source is used in a 1.2 MW gas engine driven heat pump in project No. 8 built in Ejby, Denmark.

Water as Heat Source

When a water based heat source is available it normally gives more favourable operation conditions for the heat pump and a better economy. The temperature variation is smaller than for ambient air which reduces the variation in capacity of the heat pump and thereby the regulation problems. The evaporator is less expensive and the better economy is often capable of justifying a large investment in pipes, if there is a distance between the heat pump and the heat source.

However, some of the natural heat sources as river water and sea water can cause problems, because the temperature is close to the freezing point in the winter time. In some cases the heat pump therefore has to be stopped for a period in the winter time.

When the heat pump plant is larger than about 5 MW, a water based heat source is believed to be necessary, as the area around the heat pump will be cooled down, when the wind is not blowing.
When selecting the case studies, distinction has been made between natural heat sources and sewage water which has a higher average temperature due to warm waste water.

Sewage Water as Heat Source

In project No. 11 a 3 MW electric heat pump in Sala, Sweden, uses sewage water as heat source. This heat source is also used in a 10 MW diesel driven heat pump in project No. 7 built in Frederikshavn, Denmark. Project No. 3 is a 2.5 MW absorption heat pump fueled partly by biogas from a sewage plant and using sewage water as heat source. This plant is situated in Waiblingen, W. Germany.

River and Sea Water as Heat Source

In project No. 12 a 10 MW electric heat pump for Visby in Sweden is projected to use sea water as heat source.
In project No. 4 river water is combined with ventilation air as heat source for a 4.6 MW gas engine driven heat pump plant in Paderborn, W. Germany.
As mentioned before, river water has been projected as a supplement to the air evaporator in the 1.2 MW gas engine driven heat pump in Ejby, Denmark. Due to environment protection rules the river water will be changed into ground water, however.
River water is also used as an alternative heat source for project No. 2 in Völklingen, W. Germany.

Other Heat Sources

The remaining types of heat sources give some examples of different heat pump applications which in no way give a total picture of the possiblities.

In project No. 6 in Regio Emilia, Italy, the fresh water supply is used as heat source and heat sink for the heat pump which is used both for heat production, for balancing the power production from a diesel driven combined heat and power plant during the winter, and for air conditioning during the summer.

In project No. 10 a 1.7 MW electrically driven heat pump in Ruddalen, Sweden, uses an ice rink as heat source which gives an exploitation of both the hot and the cold side of the heat pump.

Another general application is the use of a heat pump to cool the return water in transmission pipes for district heating. By extended cooling of the return water, the water flow is reduced with the same energy transport capacity and thereby saves power for the water pump, or the transport capacity is increased and saves the investment in a larger or an additional transmission pipe.

The motive for this application is mostly economic savings, as the energy consumption often increases. However, in some cases the application is able to increase the use of heat from combined heat and power plants and in this way it brings about a substantial energy saving.

Project No. 5 is a 0.5 MW test project for Mannheim, W. Germany, for a specially developed heat pump. This heat pump uses a combined principle of absorption and vapour compression. The compressor is driven by a gas engine.

In project No. 2 in Völklingen, W. Germany, a gas fired absorption heat pump is used for cooling the return water in a district heating transmission pipe, but in this particular case it is due to the fact that the district heating is waste heat from a coking plant, and consequently the amount of exploited waste heat is increased because of lower temperature in the heat exchangers. As an alternative operation of this heat pump, river water can be used as heat source.

Table 4.1.1 on this page shows the heat pump plants included in this project. A broad spectrum of combinations between type of heat pump and heat source is covered by the investigated plants.

Heat sources / Prime mover	air	sewage water	river or sea water	ice or misc.
electric	9	11	12	10
absorption	-	3	(2)	2
diesel/gas	1+8	7	4+(8)	5+6

Table 4.1.1.Heat pump plant by prime mover and heat sources.

5 COMPARISON OF DIFFERENT HEAT PUMP PROJECTS

The economy of an energy saving investment project often gives different results in different countries, not only because the operation conditions can vary, but mainly because the economic conditions are different. This is because capital costs are highly dependent on interest rates and subsidies to the investor, and because operation costs and savings depend on labour costs and energy prices including taxes.

In the following chapter the essential economic parameters are expounded and a method to make a rough economy calculation for comparison of the projects are developed. This economy calculation may give an indication of the value of more detailed calculations and planning of new projects in each country.

5.1 Investment in Heat Pumps

The costs of a heat pump can normally be calculated independently of the country in which it is built with an accuracy of ± 10%, as export and import between IEA countries are only subject to custom duties of a few percent.
However, the investment can show a variation in the same size within each country due to differences in operation conditions, connection costs to the district heating net and the heat source.
Within the uncertainty of international comparison it is believed that the investment cost can be translated by the exchange rate of currencies.
To account for different sizes of a heat pump plant you could calculate the investment relatively to the heat output, but this figure is very sensitive to the operation conditions and type and design of heat pump.
The investment in heat pumps is made to save energy, so the investment relative to the annual energy saving (IES) is a key figure both taking into account the size of the plant and the investment in higher efficiency.
For each project the investment per tons of oil equivalent (toe) is calculated and used as a key figure for the evaluation of the projects.

5.2 Expenditures for a Heat Pump Plant

The capital cost, which is an important part of the annual expenses for an energy saving investment, can be calculated as an annuity (a) which incorporates interest rate on the capital and depreciation during the life time of the plant.

It is generally believed that the energy prices will increase as much as the general inflation, and as the income of the investment is due to energy saving the calculated annuity for the capital cost is comparable to real interest or return on capital in fixed prices.

Normal interest rates show a large variation between countries and from time to time, but in economic theory the real interest rate is believed to be more stable and in the size of 3-9% p.a.

Another major expense for a heat pump plant is the cost of operation, repair and maintenance. These expenses are often calculated as 1-6% of the investment p.a., and it is believed to be typical of the type and complexity of the plant, while the dependency of the plant size is only a small variation of comparable sizes.

Calculated as a percentage of the investment, the operation costs (OC) are not likely to vary much between the countries. This figure is therefore calculated as the second key figure for each plant. The total expenditure, exclusive of energy costs, for the operation of a heat pump plant can be expressed as :

$$I \cdot (a + OC)$$

where international values of I and OC can be used.

5.3 Energy Savings

If a heat pump plant supplies an annual amount of energy (ED) with an average Energy Ratio (ER) of the heat pump, the project can be compared with the alternative heat production method as for example a boiler with a normal Energy Ratio (ER_o) of 85%. The Energy Saving (ES) can be calculated in the following way:

$$\text{Energy used for boiler production} = ED/ER_o$$
$$\text{Energy used for the heat pump} = ED/ER$$

$$\text{Energy Saving per year: } ES = ED/ER_o - ED/ER$$

The energy saving in percent (ES%) of the fuel consumption of the boiler is

$$ES\% = (ED/ER_o - ED/ER) \ / \ (ED/ER_o)$$

$$= 1 - ER_o/ER$$

In this way the annual energy saving can be written as the energy saving in percent times the fuel consumption of a boiler:

$$ES = ES\% \cdot (ED/ER_o)$$

The energy saving per year is strongly dependent on the application of the heat pump, especially the difference in full load operation hours between a monovalent heat pump and a base load plant.

Unlike this, the energy saving in percent is a typical figure for the heat pump and is therefore calculated as a key figure, even though it is dependent on the operation temperatures.
However, the economic result of the energy saving depends on the energy price, especially if the heat pump uses another fuel type than the alternative production method.

In case of different fuels you therefore have to calculate the energy cost savings.

5.4 Energy Cost Savings

Normally the only income for a heat pump plant is the savings in energy costs. The energy cost saving (ECS) can be derived in the following way; when P_o is the fuel price for boiler fuel and P is the fuel price for the heat pump:

$$\begin{array}{ll} \text{Energy cost for boilers:} & (ED/ER_o) \times P_o \\ \text{Energy cost for H.P.} \quad : & \underline{(ED/ER) \times P} \end{array}$$

$$\text{Energy cost saving, ESC} = ED\ (P_o/ER_o - P/ER)$$

The energy cost saving in percent of normal fuel cost is as follows:

$$\begin{aligned} ECS\% \ : \ & ED\ (P_o/ER_o - P/ER)/(ED/ER_o) \cdot P_o \\ & 1 - (ER_o/P_o)/(ER/P) \\ = \ & 1 - (ER_o/ER)\ (P/P_o) \end{aligned}$$

The energy ratio divided by the energy price can be defined as an Economic Energy Ratio (EER) which makes the expression for energy saving similar to that of energy cost saving:

$$ES\% = 1 - ER_o/ER$$

and

$$\begin{aligned} ECS\% &= 1 - EER_o/EER \\ &= 1 - (ER_o/ER)\ (P/P_o) \end{aligned}$$

From these expressions it is seen that the energy cost saving is equal to the energy saving, when the fuel or the fuel prices are the same.

The percentage of energy saving is typical for heat pump installations but the price structure of different fuels P/P_o is dependent on the country and sometimes the actual place of the plant.
However, it is seen that by knowing the fuel prices for a place and the energy saving for a heat pump plant, the energy cost saving, which is essential to the economy of the plant, can be calculated from:

$$1 - ECS\% = (1 - ES\%) \cdot P/P_o$$

The return on investment in a heat pump plant can be calculated by putting the expenditure I (OC + a) equal to the annual energy cost savings, ECS:

$$\begin{aligned} ECS &= (ED/ER_o) \cdot P_o \cdot ECS\% \\ &= \frac{ES}{ES\%} \cdot P_o \cdot ESC\% \\ &= I\ (OC + a) \end{aligned}$$

This can be transformed to:

$$\frac{I}{ES}\ (OC + a) = P_o \cdot \frac{ECS\%}{ES\%}$$

The investment I divided by the energy saving ES has been mentioned as the investment per saved ton of oil, IES:

$$OC + a = \frac{P_o}{IES} \cdot \frac{ECS\%}{ES\%}$$

From this it can be seen that by knowing the key figures OC and ES% for the heat pump plant, IES for the application and the energy prices for the place of installation (country), the energy cost savings and the annuity can be calculated and thereby give an indication of the return on investment.

When the fuel prices for a heat pump are the same as for the boiler, the energy cost savings ECS% is the same as the energy saving ES%, and the expression for the economy becomes more simple:

$$IES\ (OC + a) = P_o$$

The left side is the annual capital and operation cost for saving one ton of oil per year and P_o is the price of energy calculated as oil equivalent. From this the return on investment can be calculated from the annuity a, where

$$a = \frac{P_o}{IES} - OC \quad \text{in percent per year}$$

5.5 Economy Calculations for Changed Applications

If the application of the heat pump is changed, if for instance a monovalent heat pump is recalculated to a base load operation, the key figure IES is no more valid and it is necessary to return to more basic calculations.

The annual savings in energy expenditures compared with an existing boiler production can be calculated as:

Existing energy expenditure x
fraction delivered by the heat pump x
energy cost saving in percent

As a rule-of-thumb the fraction delivered by the heat pump can be calculated as 80 to 90 percent, when the heating effect of the heat pump is 50 to 60 percent of the maximum demand. In case of a smaller plant it can be calculated as the utilization time (e.g. 7-8000 h/year) multiplied by the average heating capacity of the heat pump.

Finally, the annual saving in energy expenditures should be related to and prove to be greater than the total expenditure on operation of the heat pump and capital cost.

As an example a base load, capital-intensive plant has a heating effect of 50 percent and delivers 80 percent of the annual heat production.

A natural gas boiler could have an energy ratio (e.g. efficiency) of 85 percent, while a natural gas driven heat pump could have an energy ratio of 165 percent.

The energy saving would be 1 - 85/165 = 48 percent, and as the natural gas price would be identical, the energy cost saving would also be 48 percent.

The annual saving can be calculated as:

(Normal fuel expenditures) x 80 percent x 48 percent
= 39 percent of usual total expenditures

The additional operation cost compared with boiler operation can be estimated to 3 percent of the investment, and the interest and the depreciation can be estimated to 10 percent of the investment in fixed prices.

Totally, the 39 percent of the fuel expenditures should be larger than the 10 + 3 = 13 percent of the investment, or the investment must not exceed 39/13 = 3 times the total annual fuel expenditures of the existing heating central.

5.6 Key Figures for the Case Studies

In chapter 6 the selected case studies are described and the key figures for the plants have been calculated from the available information. In the last section of each description, the key figures have been collected and recalculated to the price level at the end of 1984. The key figures have also been changed according to the available results from test, measurement, and operation of the plant. In a few cases a new set of key figures have been calculated for an alternative application or modified design.

Table 5.6.1 shows the basic information of the heat pump plant in the case studies. The price information is updated to the end of 1984.

On the next page table 5.6.2 shows the calculated key figures for the case studies, and from these figures the national pay-back time (1/a) has been calculated.

As the investment has been price-regulated and the energy price has changed since the decision and construction of the plant, the calculated pay-back time will normally be different from the calculation in the feasibility studies.

Table 5.6.1 Basic information of heat pump plants in the case studies

	*)	Type of Heat Pump	Heat output MW	Investment Mill.		Operation cost% p.a. OC	Operation time per year h/y	Energy delivered MWh/y	Energy saved MWh/y	Energy saving %
6.1	M	Gas engine HP/air	0.67	1.07	DM	6.4	1850	1240	248	17
6.2	T	Absorption/return water/river	4.0	3.80	DM	4.0	7000	26000	11500	35
6.3	T	Absorption/sewage water	2.5	2.70	DM	3.0	5144	12860	5600	37
6.4	M	Gas engine HP/river	4.65	2.37	DM	4.7	1153	5262	3750	60
6.4	B	"-"	4.65	2.37	DM	4.7	6000	27900	19538	60
6.5	T	Gas engine HP/return water test	0.38	0.27	DM	5	1730	657	595	77
6.6	P	Gas engine, total energy system	5.6	5880	LIT	2	2540	(22050)	13700	35
6.7	M	Diesel engine HP/sewage water	10.5	22	D.kr.	3.5	6000	63000	39300	53
6.8	T	Gas engine HP/air/ground water	1.2	7	D.kr.	4.6	8075	9700	5700	50
6.9	P	Electric HP/air	2.6	9.7	S.kr.	2.8	6385	16600	13100	67
6.10	P	Electric HP/ice rink	1.4	4.55	S.kr.	1.4	6300	8800	7300	71
6.11	T	Electric HP/sewage water	3.0	6.25	S.kr.	2	7800	23400	18170	66
6.12	P	Electric HP/sea water	8	20.2	S.kr.	3.5	7650	65000	52500	69

*) Basis of the energy figures is:

B) Calculated as base load

M) Long term measurement

P) Project values

T) Representative test results

Table 5.6.2 Key figures for heat pump plants in the case studies

	*)	Type of Heat Pump	Heat output MW	Invest-ment Mill.		Operation cost% p.a. OC	Investment per saved MWh/y IES	Energy saving ES%	Energy cost saving ECS%	Pay back time
6.1	M	Gas engine HP/air	0.67	1.07	DM	6.4	4315	17	17	oo
6.2	T	Absorption/return water/river	4.0	3.80	DM	4.0	330	35	35	11
6.3	T	Absorption/sewage water	2.5	2.70	DM	3.0	482	37	37	9.7
6.4	M	Gas engine HP/river	4.65	2.37	DM	4.7	632	60	60	51
6.4	B	"-"	4.65	2.37	DM	4.7	121	60	60	3.3
6.5	T	Gas engine HP/return water test	0.38	0.27	DM	5	454	77	77	23
6.6	P	Gas engine, total energy system	5.6	5880	LIT	2	429	35	35	13
6.7	M	Diesel engine HP/sewage water	10.5	22	D.kr.	3.5	560	53	53	2.3
6.8	T	Gas engine HP/air/ground water	1.2	7	D.kr.	4.6	1228	50	50	6.8
6.9	P	Electric HP/air	2.6	9.7	S.kr.	2.8	740	67	66	3.9 1)
6.10	P	Electric HP/ice rink	1.4	4.55	S.kr.	1.4	623	71	70	3.1 1)
6.11	T	Electric HP/sewage water	3.0	6.25	S.kr.	2	344	66	65	1.7 1)
6.12	P	Electric HP/sea water	8	20.2	S.kr.	3.5	385	69	68	1.9 1)

*) Basis of the energy figures is:

B) Calculated as base load

M) Long term measurement

P) Project values

T) Representative test results

Notes:

1) Fuel oil boiler compared with electric HP $P/P_o = 1.02$

5.7 International Comparison

In the preceding chapter 5.6 the key figures and the pay-back time have been calculated for the national conditions of the case studies.
However, the economic conditions may be slightly different in other countries, so a heat pump plant with a good economy in one country may not be economical in another country.

The key figures in the following chapter have been recalculated in relation to the currency of each participating country and the pay-back time has been calculated for the energy prices in the country.

These calculations should give an indication of the economy of each plant, if it is built in other countries, and these calculations are therefore an important part of an international comparison of the project.

5.7.1 Case Study Calculations for Germany

To investigate the economy of building the heat pump projects in the case studies in Germany, it is assumed that the figures for investment and operation can be estimated by using the exchange rate of January 1st, 1985.

The used exchange rate was:

Denmark:	27,93 DM/loo D.kr.
Italy:	0.1636 DM/100 LIT
Sweden:	35.10 DM/100 S.kr.

In the calculation of the case studies are used the following average prices for large consumers:

Diesel oil	85 DM/MWh	23.6 DM/GJ
Fuel oil	70 DM/MWh	19,4 DM/GJ
Natural gas	64 DM/MWh	17,8 DM/GJ
Electricity	160 DM/MWh	44.4 DM/GJ

From the calculations table 5.7.1 it is seen that the energy saving of the plants and the energy cost saving vary from 17% to 77%, but due to differences in taxes and prices between the primary fuels the energy cost savings for especially the electrical driven heat pumps are much smaller than the energy saving.

The economic result is that eight plants have a pay back time less than lo years under German economic conditions.

The most attractive plants are large gas and diesel engine driven heat pumps but also electrical driven and absorption heat pumps show a good economy.

Figure 5.7.1 Case studies installed under German economic conditions

	*)	Type of Heat Pump	Heat output MW	Invest-ment Mill. DM	Operation cost% p.a. OC	Investment per saved MWh/y IES DM	Energy saving ES%	ECS%		Pay back time
6.1	M	Gas engine HP/air	0.67	1.07	6.4	4315	17	17		oo
6.2	T	Absorption/return water/river	4.0	3.80	4.0	330	35	35		6.5
6.3	T	Absorption/sewage water	2.5	2.70	3.0	482	37	37		9.7
6.4	M	Gas engine HP/river	4.65	2.37	4.7	632	60	60		18.4
6.4	B	"-"	4.65	2.37	4.7	121	60	60		2.1
6.5	T	Gas engine HP/return water test	0.38	0.27	5	454	77	77		11
6.6	P	Gas engine, total energy system	5.6	9.62	2	702	35	35		14
6.7	M	Diesel engine HP/sewage water	10.5	6.14	3.5	156	53	53		2.4
6.8	T	Gas engine HP/air/ground water	1.2	1.96	4.6	343	50	50		7.1
6.9	P	Electric HP/air	2.6	3.4	2.8	260	67	24	1)	14.6
6.10	P	Electric HP/ice rink	1.4	1.6	1.4	220	71	34	1)	7.2
6.11	T	Electric HP/sewage water	3.0	2.2	2	121	66	22	1)	5.8
6.12	P	Electric HP/sea water	8	7.1	3.5	135	69	29	1)	5.5

*) Basis of the energy figures is:

B) Calculated as base load
M) Long term measurement
P) Project values
T) Representative test results

Notes:

1) Electric heat pump compared with fuel oil boiler $P/P_o = 2.29$

5.7.2 Case Study Calculations for Denmark

In the same way as for Germany, the case studies can be calculated as if they were placed under Danish economic conditions.
The investment and operation costs are calculated by using the exchange rate for January 1st, 1985.

Germany:	358.0	D.kr./100 DM
Italy:	0.5856	D.kr./100 LIT
Sweden:	125.7	D.kr./100 S.kr.

The average energy prices for large consumers in Denmark are:

Diesel oil	306 D.kr./MWh	85 D.kr./GJ
Fuel oil	259 D.kr./MWh	72 D.kr./GJ
Natural gas	238 D.kr./MWh	66 D.kr./GJ
Electricity	434 D.kr./MWh	120 D.kr./GJ

From the calculations for Denmark in table 5.7.2 it is seen that both an engine driven heat pump and an electric heat pump show a good economy. In total 9 calculations out of 13 show a pay back time less than 10 years. Two projects are not economical because they are monovalent installations with a far too large capacity.

Even the small test plant for cooling return water shows a pay back time of 11 years in spite of the fact that the plant is small and only used as peak load installation with a utilization time of 1730 hours per year. This means that the use of heat pumps in connection with long district heating transmission lines should be given special attention.

Table 5.7.2 Case studies installed under Danish economic conditions

	*)	Type of Heat Pump	Heat output MW	Invest-ment Mill. D.kr.	Operation cost% p.a. OC	Investment per saved MWh/y IES D.kr.	Energy saving ES%	ECS%	Pay back time
6.1	M	Gas engine HP/air	0.67	3.83	6.4	15440	17	17	oo
6.2	T	Absorption/return water/river	4.0	13.6	4.0	1183	35	35	6.2
6.3	T	Absorption/sewage water	2.5	9.7	3.0	1726	37	37	9.3
6.4	M	Gas engine HP/river	4.65	8.5	4.7	2267	60	60	17
6.4	B	"-"	4.65	8.5	4.7	435	60	60	2.0
6.5	T	Gas engine HP/return water test	0.38	1.0	5	1680	77	77	11
6.6	M	Gas engine, total energy system	5.6	34.4	2		35	35	13
6.7	M	Diesel engine HP/sewage water	10.5	22.0	3.5	560	53	53	2.3
6.8	T	Gas engine HP/air/ground water	1.2	7.0	4.6	1228	50	50	6.8
6.9	P	Electric HP/air	2.6	12.2	2.8	931	67	45 1)	6.3
6.10	P	Electric HP/ice rink	1.4	5.7	1.4	703	71	52 1)	3.9
6.11	T	Electric HP/sewage water	3.0	7.9	2	432	66	43 1)	2.7
6.12	P	Electric HP/sea water	8	25.2	3.5	481	69	48 1)	2.9

*) Basis of the energy figures is:

B) Calculated as base load
M) Long term measurement
P) Project values
T) Representative test results

Notes:

1) Electric heat pump compared with a fuel oil boiler $P/P_O = 1.67$

5.7.3 Case Study Calculations for Italy

As for Germany and Denmark, the case studies are recalculated to Italian economic conditions, assuming the same operation conditions could be found in Italy and that the investment and operation costs can be estimated by using the following exchange rates.

Denmark	17100 LIT/100 D.kr.
Germany	61100 LIT/100 DM
Sweden	21500 LIT/100 S.kr.

The energy prices are slightly different and for large consumers like a district heating central the following prices are used:

Diesel oil	68.040 LIT/MWh	18.900 LIT/GJ
Fuel oil	33.640 LIT/MWh	9.345 LIT/GJ
Natural gas	41.920 LIT/MWh	11.645 LIT/GJ
Electricity	109.450 LIT/MWh	30.400 LIT/GJ

The price of electricity depends on different tariffs which show variations from 10 LIT/kWh for off-peak power up to 172 LIT/kWh during peak load hours. The user's price is a tariff for a general industrial load above 100 kW.

The calculations for Italy shows that large engine driven heat pumps should be economical. The electrical heat pump on the contrary shows an operation deficit, because of the high electricity price. This high electricity price may also improve combined heat and power schemes like in the project of Reggio Emilia in a way which can not be accounted for with this calculation method. A large absorption heat pump can show a pay back time of 6 years, but this result is less than that of the corresponding engine driven heat pump.

Table 5.7.3 Table studies installed under Italian economic conditions

	*)	Type of Heat Pump	Heat output MW	Invest-ment Mill. LIT	Operation cost% p.a. OC	Investment per saved MWh/y IES 1000 LIT	Energy saving ES%	ECS%	Pay back time
6.1	M	Gas engine HP/air	0.67	654	6.4	2640	17	17	oo
6.2	T	Absorption/return water/river	4.0	2322	4.0	202	35	35	6.0
6.3	T	Absorption/sewage water	2.5	1650	3.0	295	37	37	8.9
6.4	M	Gas engine HP/river	4.65	1448	4.7	386	60	60	16
6.4	B	"-"	4.65	1448	4.7	74	60	60	1.9
6.5	T	Gas engine HP/return water test	0.38	165	5	277	77	77	9.9
6.6	M	Gas engine, total energy system	5.6	5880	2	429	35	35	13
6.7	M	Diesel engine HP/sewage water	10.5	3762	3.5	96	53	53	3.2
6.8	T	Gas engine HP/air/ground water	1.2	1197	4.6	210	50	50	6.5
6.9	P	Electric HP/air	2.6	2085	2.8	159	67	- 7 1)	oo
6.10	P	Electric HP/ice rink	1.4	978	1.4	134	71	6 1)	oo
6.11	T	Electric HP/sewage water	3.0	1344	2	73	66	- 11 1)	oo
6.12	P	Electric HP/sea water	8	4343	3.5	83	69	- 1 1)	oo

*) Basis of the energy figures is:

B) Calculated as base load

M) Long term measurement

P) Project values

T) Representative test results

Notes:

1) Electric heat pump compared with fuel oil boiler $P/P_o = 3.25$

5.7.4 Case Study Calculations for Sweden

The transformation of the case study results to Sweden is more problematic, because Sweden has no natural gas. However, the calculations have been made with the following assumptions:
Gas engines for heat pumps are more expensive than diesel engines and have a slightly lower efficiency. The price of diesel oil is used as fuel for the gas engines. The smallest heat pumps are compared with boilers for diesel fuel and the large ones with boiler for heavy fuel oil.
Absorption heat pumps are assumed fired with a boiler for heavy fuel oil and compared with a normal boiler for heavy fuel.
Generally, this assumption will give a pessimistic impression of the heat pumps.

The following exchange rates have been used:

Denmark	79.57	S.kr./100 D.kr.
Germany	284.9	S.kr./100 DM
Italy	0.4660	S.kr./100 LIT

And the following energy prices for consumption of about 1000 tons/year:

Diesel oil	293 S.kr./MWh	81.4 S.kr./GJ
Fuel oil	215 S.kr./MWh	59.7 S.kr./GJ
Electricity	220 S.kr./MWh.	61.1 S.kr./GJ

The calculations in table 5.7.4 clearly show that the best economic results are reached in the large electric heat pumps, mainly because of the low electricity prices.

There is, however, only a small difference to the economy of the large gas and diesel engine driven heat pumps, so one should pay attention to this heat pumps if any indications show that the electricity price will rise in the future or if imported natural gas should be used with a high efficiency.

Table 5.7.4 Case studies installed under Swedish economic conditions

	*)	Type of Heat Pump	Heat output MW	Investment Mill. S.kr.	Operation cost% p.a. OC	Investment per saved MWh/y IES S.kr.	Energy saving ES%	ECS%	Pay back time	
6.1	M	Gas engine HP/air	0.67	3.05	6.4	12300	17	17	oo	1)
6.2	T	Absorption/return water/river	4.0	10.8	4.0	939	35	35	5.3	2)
6.3	T	Absorption/sewage water	2.5	7.7	3.0	1374	37	37	7.9	2)
6.4	M	Gas engine HP/river	4.65	6.8	4.7	1813	60	46	23	3)
6.4	B	"-"	4.65	6.8	4.7	348	60	46	2.3	3)
6.5	T	Gas engine HP/return water test	0.38	0.77	5	1294	77	77	8.6	1)
6.6	M	Gas engine, total energy system	5.6	27.4	2	2000	35	(12)	59	3)
6.7	M	Diesel engine HP/sewage water	10.5	17.5	3.5	445	53	53	2.2	
6.8	T	Gas engine HP/air/ground water	1.2	5.6	4.6	982	50	32	10.6	3)
6.9	P	Electric HP/air	2.6	9.7	2.8	740	67	66	3.9	4)
6.10	P	Electric HP/ice rink	1.4	4.55	1.4	623	71	70	3.1	4)
6.11	T	Electric HP/sewage water	3.0	6.25	2	344	66	65	1.7	4)
6.12	P	Electric HP/sea water	8	20.2	3.5	385	69	68	1.9	4)

*) Basis of the energy figures is:

B) Calculated as base load

M) Long term measurement

P) Project values

T) Representative test results

Notes:

1) Gas oil boiler compared with diesel engine HP $P/P_o = 1$

2) Fuel oil boiler and HP instead of gas $P/P_o = 1$

3) Fuel oil boiler compared with diesel engine HP $P/P_o = 1.36$

4) Fuel oil boiler compared with electric HP $P/P_o = 1.02$

5.7.5 Variation in Energy Prices and Pay Back Periods

The preceding calculations on the economic conditions of the heat pump plants in the case studies show a considerable variation in the return on investment.

The most extreme variation between the countries is an electric heat pump (project 6.11) with pay back times from 1.7 years in Sweden, 2.7 years in Denmark, 7.2 years in Germany to infinity in Italy.

The investment and the operation conditions for these projects are identical, so the economic conditions are only determined by the energy prices in each country.

To analyse the variation in energy price a price index of diesel oil, heavy fuel oil, natural gas and electricity is calculated for each country. The index is calculated relative to the price at the end of 1984 of heavy fuel oil in Denmark without energy taxes. Table 5.7.5 shows the calculated index.

Table 5.7.5 Energy Price Index in Participating Countries

	Diesel Oil	Fuel Oil	Natural Gas	Electricity
Denmark	140	119	110	198
Germany	139	114	105	261
Italy	182	90	112	293
Sweden	168	123	-	126
max/min	131%	137%	107%	233%

The table shows that the variations between the maximum price and the minimum price is from 131% for diesel oil to 233% for electricity. In general Germany has the lowest energy prices except for electricity, which is cheapest in Sweden.
As a low energy price makes an energy saving investment in a heat pump less attractive, the lowest return on investment or highest pay back time is therefore expected in Germany. This is also the case except for the electrical heat pump which was unattractive in Italy.

To compare the return on investment in different countries the following table 5.7.6 shows the pay back time (reciprocal annuity) for all the selected projects in each country.

The table shows a large variation between the countries and of course between the projects. If you take a pay back time of 10 years as a commercial limit, it is seen that 9 projects are commercially possible in Denmark, 8 projects in Germany, 6 projects in Italy and 9 projects in Sweden - all out of 13 calculations of pay back periods.

In the column of the lowest pay back periods (min) you will find 10 out of 13 projects commercial in one of the four countries.

The result, that 3 projects are not commercial in any country, does not mean that the technology has no future. In some cases it is owing to unlucky circumstances in the demonstration project. As an example, the gas engine driven heat pump in Paderborn is not commercial in any country as demonstrated in the measuring program, but this is because the plant is 3 times too large to cover the maximum heat demand. If the project is recalculated with the measured performance, but used as a base load plant it becomes the most economical project.

The same result can be concluded from the gas engine driven heat pump with air as heat source in Erlangen, which is not economical as demonstrated but an almost identical project in Ejby, Denmark shows a reasonable economy.

Table 5.7.6 Pay Back Times of Selected Projects vs. Country

	*)	Type of Heat Pump	Heat output MW	Pay-Back Time, Country					
				Denmark	Germany	Italy	Sweden	Min.	Max.
6.1	M	Gas engine HP/air	0.67	oo	oo	oo	oo	-	oo
6.2	T	Absorption/return water/river	4.0	6.2	6.5	6.0	5.3	5.3	6.5
6.3	T	Absorption/sewage water	2.5	9.3	9.7	8.9	7.9	7.9	9.7
6.4	M	Gas engine HP/river	4.65	17	18	16	23	16	23
6.4	B	"-"	4.65	2.0	2.1	1.9	2.3	1.9	2.3
6.5	T	Gas engine HP/return water test	0.38	11	11	9.9	8.6	8.6	11
6.6	P	Gas engine, total energy system	5.6	13	14	13	59	13	59
6.7	M	Diesel engine HP/sewage water	10.5	2.3	2.4	3.2	2.2	2.2	3.2
6.8	T	Gas engine HP/air/ground water	1.2	6.8	7.1	6.5	10.6	6.5	11
6.9	P	Electric HP/air	2.6	6.3	15	oo	3.9	3.9	oo
6.10	P	Electric HP/ice rink	1.4	3.9	7.2	oo	3.1	3.1	oo
6.11	T	Electric HP/sewage water	3	2.7	5.8	oo	1.7	1.7	oo
6.12	P	Electric HP/sea water	8	2.9	5.5	oo	1.9	1.9	oo
		Min.		2.0	2.1	1.9	1.7		

*) Basis of the calculations is:

B) Calculated as base load
M) Long term measurement
P) Project values
T) Representative test results

5.8 Conclusion from Comparison between the Countries

The table shows that the most economical heat pump project in Denmark, Germany and Italy is a gas engine driven heat pump used for a low temperature heating system in a sports center. The pay back time is from 1.9 years to 2.3 years. For Sweden it is an electic heat pump used as base load plant in a normal district heating net and with sewage as heat source. The pay back time is here calculated to 1.7 years.

The operation conditions for the 4.65 MW gas engine driven heat pump is however recalculated to operating as a base load plant. Looking away from this calculation, the second best project is a 10 MW diesel driven heat pump in Frederikshavn for a normal district heating supply operated with the normal supply temperature and as second priority with respect to the load. The heat source is also sewage water and the pay back time varies from 2.2 years in Sweden to 3.2 years in Italy.

The gas engine driven heat pump in Paderborn and the diesel engine driven heat pump in Frederikshavn have in common the considerably size of the project 4.65 MW and 10.5 MW, the water based heat sources and the operating by piston engines. The heat pumps themselves are different as the first has screw compressors and the second turbo compressors, which also makes the heat pump system different.

From the 1.2 MW gas engine heat pump project in Ejby it is seen that a reasonable economy still can be reached in this size even though the heat source is the worst of all - ambient air.

5.9 Evaluation of Experience

From project no. 1 and project no. 8, both gas engine driven heat pumps with ambient air as heat source, it can be seen that even though a technology doesn't show a good economy in one project, further developement can show improvement. It is possible to get a reasonable economy with ambient air as heat source for gas engine driven heat pumps in the size of 1 MW.

From the absorption heat pump projects, project no. 2 and project no. 3, it is seen that even though a high technical performance is reached, it is difficult to get the economy as good as the economy of the engine driven heat pumps. The reasonable economy in project no. 2 could attract ones attention to the use of district heating return water as heat source as in project no. 5.

The larger gas and diesel engine driven heat pumps in project no. 4 and project no. 7 show a good economy in all countries. Together with project no. 1 these projects clearly show that heat pumps should be used as base load heating plants together with a peak load boiler and never as a monovalent supply system.

The total energy scheme in project no. 6 is difficult to evaluate to other countries as many parameters has an influence on the operation economy. However, when there is a combined need for heating, cooling and electricity there should be a basis for this kind of schemes.

The test results from project no. 4 and the total energy project show that the normal heat loss calculations for a new building and expected increase in the market potential are unsure values for projecting a heat pump. A far better economy is reached by building a small base load plant prepared for an extension when the heat demand has been measured.

Projects no. 9 to 12 are all base load electric heat pumps with different sources. They show a good economy in Denmark and Germany, but especially in Sweden the price ratio between fuel oil and electricity makes electrical heat pumps attractive.

In Denmark the electricity is produced mainly from coal and it is is a question whether these heat pump plants should be compared with a coal boiler instead of an oil boiler. Environmental considerations could, however, give preference to electric heat pumps with a centralized use of coal.

The economical experience from different heat sources can be evaluated by looking at the range of pay back times with respect to the heat source as shown in table 5.9.1.

Table 5.9.1. Pay back periods with respect to the heat source

Sewage water	1.7 - 7.9 years
River and sea water	1.9 - 16 years
Ice and misc.	3.1 - 13 years
Air	3.9 - ∞ years

As expected beforehand sewage water with the slightly elevated temperature shows the best economy, when industrial waste heat is excluded. River and sea water give almost the same result. The group with an ice rink and miscellaneous comes in between these results and finally air shows pay back times from 3.9 years to infinity.

The table also shows that a good heat source not necessarily gives a good economy. A good project or good economical conditions for an air heat pump are more important than the heat source. It is therefore recommended to compare projects with different heat sources before making the final choise of the heat source.

In many cases it can be recommended to make a case study with air as a basic alternative and in the detail projecting make a cost benefit analysis of other alternative heat sources.

The most common technical experience with waste water is that the water flow is drastically reduced in the night time. Both in Waiblingen and in Frederikshavn it has been necessary to use an existing sewage basin to accumulate water during the day time in order to keep the heat pump operating in the night time.

A general experience in using waste water is that the water can be extremely corrosive and imply scaling problems in the evaporator.

Sea water and river water are heat sources showing almost the same economy as waste water. From the demonstration projects no difficulties have been reported. It is, however, a general knowledge that these heat sources can cause problems, at least in the Nordic countries, with respect to low temperature in the winter. The evaporator and the heat source pipe system have to be designed to a very small cooling of the heat source water. Even then, it can be difficult to avoid problems with freezing of the water and in many cases it can be necessary to stop the heat pump in winter time.

In Sweden and Norway a new evaporator with stainless steel plates has been tested succesfully. The water is sprayed on to the evaporator plates and is cooled as a falling film.

Two projects have a combined use of the heating and the cooling from the heat pump. In project no. 10, the heat pump is base load for an ice rink, which means that you not only have the energy saving from the heat production but also the electricity saving in the normal cooling equipment. In this case it is however less than 10% of the electricity for the heat pump. This project shows that with a heat pump you can both save energy with a good economy and get benefits from the cold side in form of saved operation expences on an ice rink. In the summer time the ice rink functions as a solar collector. On a cloudy summer day the evaporation temperature can be $5^{o}C$ higher than the air temperature.

In project no. 6 a part of the heat source is cooled for air conditioning even though the market for district cooling seems small in the European countries.

Ambient air is used in three projects as basic heat source. If the air wasn't such an expensive and complicated heat source, a much smaller research effort on heat sources was needed.

The projects show that a large experience has been gained with utilization of air as heat source and the case studies can be regarded as the first generation demonstration plants or as prototypes. Especially defrosting and capacity regulation give problems but a good experience has been gained in these fields.

5.10 Recommendations for future R, D & D

The large variation in economic results for the different case studies shows that many lessons has to be learned and more R, D & D is needed if the technology shall be developed to a commercial level.

The developement of low cost components for large heat pumps and developement of more advanced heat pump cycles with economizer and multistage condensation is expected to increase the energy saving and to make the economy in using heat pumps for district heating even better.

The high cost of large heat pump projects calls for a continuation of an international cooperation for exchanging ideas, test and demonstration results.

There is also a large need for developement in different types of heat sources and low cost heat exchangers for the different heat sources. Especially the developement of an evaporator for air with reduced cost and efficient defrosting equipment is recommended.

The developement of evaporators for water temperatures near the freezing point and even with posibility of ice production will surely increase the use of heat pumps especially in the Northern areas.

It is believed that the use of a combination of two heat sources with different properties will make it possible to exploit the best property of both the heat sources. However, the theory and the computer models for such systems are not yet developed and tested against measuring results.

6.0 INTRODUCTION TO CASE STUDIES

In the following 12 chapters a short uniform description is given for the 12 case studies of large heat pumps in the participating countries: Germany, Italy, Sweden and Denmark. These projects are chosen out of 43 important heat pump projects. The selection took place with respect to the preferences of the participants and to a broad representation of the different applications of large heat pumps. Many problems arose in this projects and many lessons have been learned.

In some of the projects not selected for this study, the problems have even been so many that no measurement results will be available as the plants will never go into commercial operation. Some of the experience from these plants have been given to the study group through lectures and reports.

6.1 NATURAL GAS DRIVEN HEAT PUMP IN ERLANGEN, W.GERMANY

6.1.1 Owner and Location

The plant is located in the town of Erlangen, and the heat pump is owned and operated by Erlangen district heating association.

6.1.2 Heating System

The heat pump system is a monovalent heating system, which means that there is no boiler capacity for reserve and peak load. The heat is delivered to the Public Housing Association in Erlangen and used for heating 77 flats covering a total area of 6,200 m^2. The heating demand is 1.6 GWh per year with a maximum load of 0.68 MW. The plant was built in 1980 in expectation of lower operation costs compared with a normal boiler system.

6.1.3 The Heat Pump System

The system is shown in Fig. 6.1.1.

The plant has two screw compressor heat pump units using freon R12 as the refrigerant.

Each screw compressor is powered by a spark ignition natural gas engine through a gear box (ratio 1:19). The maximum power of the engine is 130 kW at 1800 r/min.

Besides the compressor, each engine drives a 25 kW generator. These generators produce electricity for the blowers at the air evaporators, and when the heating demand is low, the generators also supply electricity to the district.

The evaporators draw heat from the ambient air, and the lowest design point is an air temperature of -18°C and an evaporation temperature of -26°C.

The return water pipes from the heating system are connected to three water storage tanks of 2.5 m^3 each, and water from these tanks is heated in the oil cooler and condenser to 55°C, and then the engine cooling water and exhaust gas heat the water to an outlet temperature of 70°C. The water is then used in the heating system and for heating the tap water.

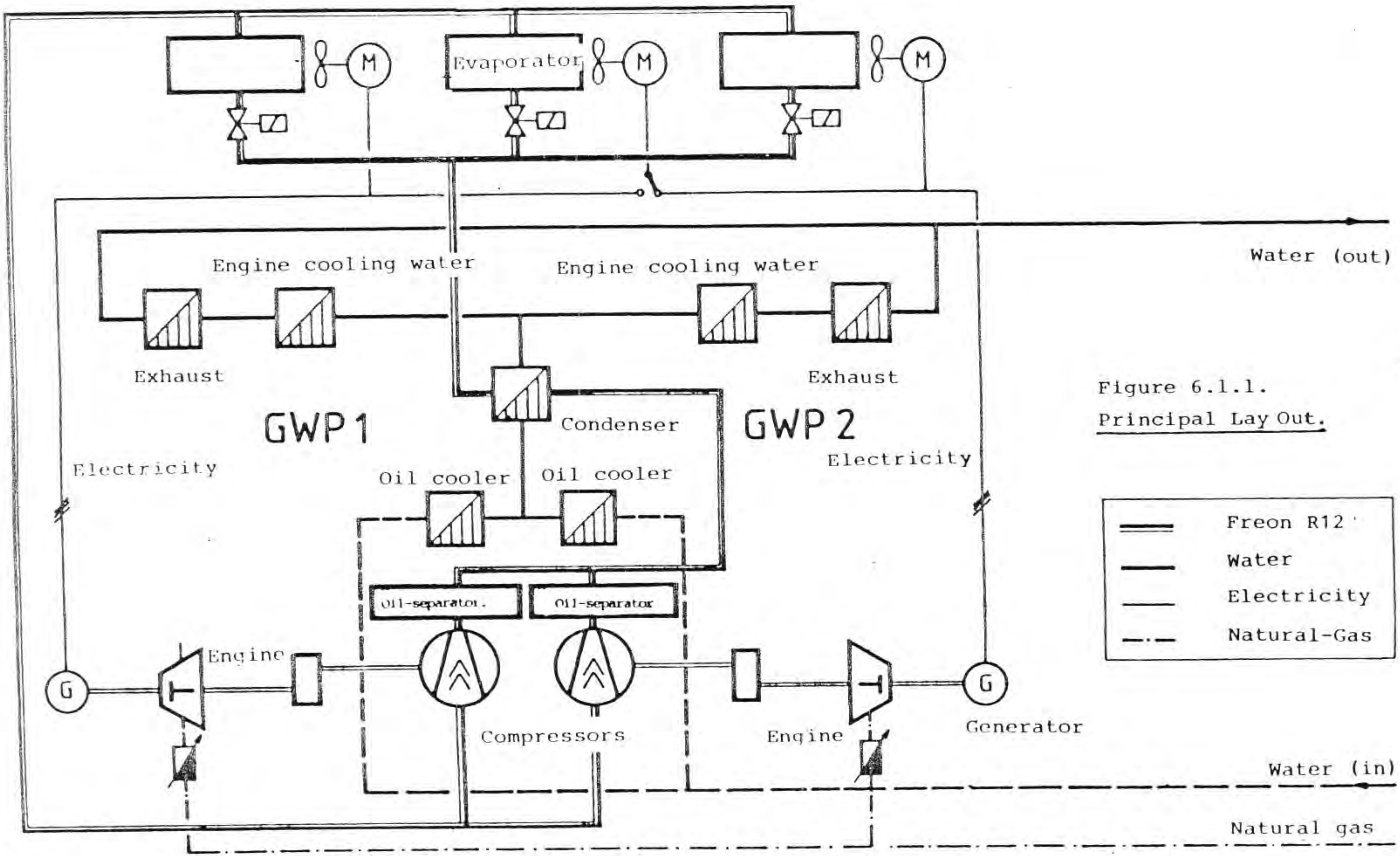


Figure 6.1.1.
Principal Lay Out.

6.1.4 Calculated Operation Conditions

The temperature of district heating to the housing blocks is regulated during the year according to the heat losses of the appartments. The hot tap water has a separate supply. The heat pump plant is expected to have the following operation conditions:

		Winter	Summer	Yearly average
Evaporator temp.	°C	-26		
Condensation temp.	°C	60		
Heat source temp.	°C	-18		9
Supply temp. from HP	°C	70		
Return temp. to HP	°C	45		
Heat from HP	MW	0.32		
Total heat from HP system (incl. waste heat)	MW	0.68		
Total load of heating system	MW	0.67		
Calculated performance				
Coefficient of performance HP		1.23		1.8
Primary energy ratio		1.1	2.5	

6.1.5 Fuel Savings

The plant is expected to supply 1.6 GWh per year at a gas consumption of 0.9 GWh per year. The calculated key figures regarding fuel saving are listed below:

Total heat production of the plant:	1.6 GWh/y
Fuel consumption:	0.9 GWh/y
Fraction of heat demand:	100 per cent
Primary fuel saving in the winter	
1 - ER/ER = 1 - 0.85/1.1 =	23 per cent
Yearly average of primary fuel savings	
1 - 0.85/1.8 =	53 per cent
Total fuel savings per year	
1.6 GWh/y (1/0.85 - 1/1.8) =	1.0 GWh/y
=	90 toe/y

6.1.6 Investment and Operation

The price of this 0.7 MW heat pump was DM 880,000 in 1980, which is DM 390,000 more than that of a conventional boiler plant. The operation costs are expected to be DM 58,000 per year.

The economic key figures are:

Total investment in the plant:	DM 880,000
Marginal investment in the plant:	DM 390,000
Operation costs:	DM 58,000
Operation costs/investment:	6.6 per cent
Investment per tons of oil saved:	DM 9.800 per toe
Marginal investment per toe:	DM 4,300 per toe
Fuel cost savings (1981)	DM 50,000 per year

6.1.7 Economy

The consumers only pay about half the total yearly costs, which corresponds to the fuel costs in a conventional boiler, for which reason the budget for heat production is as follows:

Fuel costs:	DM 44,000 per year
Operation costs:	DM 58,000 per year
Capital costs 10% (i=6.5; n=15 y):	DM 88,000 per year
Total costs:	DM 190,000 per year
Consumers pay:	DM 85,000 per year
Deficit:	DM 105,000 per year
Costs of heat production:	DM 119 per MWh
Consumer payment:	DM 56 per MWh

6.1.8 Experience from Planning and Construction

The heat pump was planned as a monovalent heat pump with a size calculated by normal maximum heat demand (DIN 4701).

This calculation gave a maximum heat load of 790 kW, which is 76 pct. more than the maximum measured heat load of 450 kW.

The plant is therefore far too large and it is concluded, that a heat pump with air as heat source should never be built as a monovalent system. The heating system shall always contain a base load heat pump and a peak load boiler.

The suction pipe from the evaporator to the compressor is equipped with a throttle valve to reduce the suction pressure to a pressure corresponding to an evaporator temperature of $0^{\circ}C$. This means that the plant would never be able to benefit from the high evaporation temperatures in the summer time and will never reach the calculated energy ratio of 2.5 in the summer time.

6.1.9 Experience from Test and Measurement

From the start of the plant in December 1980 and until February 1982 several failures occured, but from February 1982 the heat pump worked without greater problems. For the security of the heat delivery a boiler has, however, been installed.

The main technical problems with the components of the heat pump were:

- leaks of freon from the heat pump
- leaks of oil and freon from the sealings of the compressor in stand-by position
- the gasengines sucked freon from these leaks through the air intake, which caused corrosion in the exhaust system
- the expansion valve before the evaporator is regulated by overheating of the evaporatorgas, but an uneven distribu-

tion of liquid freon caused drops in the suction gas. These drops cool the sensor of the expansion valve and close the valve too much

-when the defrosting system melts the ice on the evaporator, the defrosting system stops before the water is removed.

-the heat pump system was not able to reach a sufficient supply temperature with one engine in operation, mainly because the water flow was lower than projected. The necessary increase in the water flow of the heating system caused a high return temperature.

-during stand-by of one of the heat pumps, freon is condensing in the compressor and dissolved in the oil. The high degree of part load operation of the heat pumps gives a high heat loss from the plant compared with the heat production and thereby reducing the energy ratio of the plant

- the specific work of compression from the screw compressor was only half of the work promised by the manufacturer (Linde). The Carnot efficiency was 0.28-0.31, while the expected value was 0.58.

The Result of the Plant

From the operation the following measurements are available:

Date/time	17/11	winter	summer
Heat source	2.8°C		
Supply temperature	56°C		
Return temperature	45°C		
Heat from condenser, MW	0.093		
Heat from heat pump, MW	0.277		
Primary energy ratio	0.98	0.99	1.03

This energy ratio is extremely low and it could be reached by a condensing gas boiler.

6.1.10 Conclusions from this Plant

The most serious problems with the plant are the low energy ratio and the high price for the monovalent heat pump.

The low energy ratio is due to several design and component deficiencies.

The concept with a monovalent heat pump means, that the heat pump always will run on part load, which reduces the efficiency due to regulation problems, larger heat losses from a larger plant and falling heat transfer values in heat exchangers at low water flow.

The regulation of the suction pressure by throttling makes it impossible to reach the high energy ratios in the summer time.

As things are now, a screw compressor is not able to reach the efficiency of a piston compressor under varying operation conditions. Furthermore the size of the plant clearly indicates, that a piston compressor is more suitable for this purpose.

Several components in the plant, especially the evaporator and the compressor, did not fulfil the promises, which increased the operation problems caused by the design of this heat pump system.

This plant is one of the first gas engine driven heat pumps for heating of buildings with ambient air as heat source.

The example shows that many lessons has to be learned. This plant should therefore not be taken as the state of art for heat pump plants.

6.1.11 Key Figures for 0.7 MW Gas Engine Driven Heat Pump

The test of the plant showed a considerably lower performance than calculated in the feasibility study, which is the result of many unfortunate circumstances. The following key figures for the plant should therefore not be regarded as the state of art for gas engine driven heat pumps.

Key figures for the plant according to the test results and prices regulated to January 1985 with 5 pct. per year is:

1) Heat output	W	0.67 MW
2) Investment	I	1.07 M.DM
3) Operation/Investment	OC	6.4 %
4) Energy delivered	ED	1240 MWh/y
5) Full load operation	-	1850 h/y
6) Yearly energy saving	ES	248 MWh/y
7) -"-	ES	22 toe/y
8) Investment per ES	IES	48.600 DM/toe/y
9) Relative energy saving	ES%	17 pct.

6.2 ABSORPTION HEAT PUMP PROJECT IN SAARBERG

6.2.1 Owner and Location

A 3.5 - 4.2 MW absorption heat pump owned by a joint venture between "Borsig", "Saarberg-Fernwârme" and "Saar-Ferngas" was in operation from November 1980 until 1982.

The heat pump was placed near a coking plant in Vôlklingen and supplied heat into the Saar district heating master line.

6.2.2 Heating System

The heat pump supplied heat into an existing district heating transmission line, which in beforehand received most heat from the coking plant. The heat source of the pump was either river water or return water from the district heating.

Waste heat from the coking plant was supplied to the district heating net, the cooling of the return water to the coking plant increased the exploitation of this waste heat.

Fig. 6.2.1 shows the principal lay-out of the heat pump connection to the district heating net.

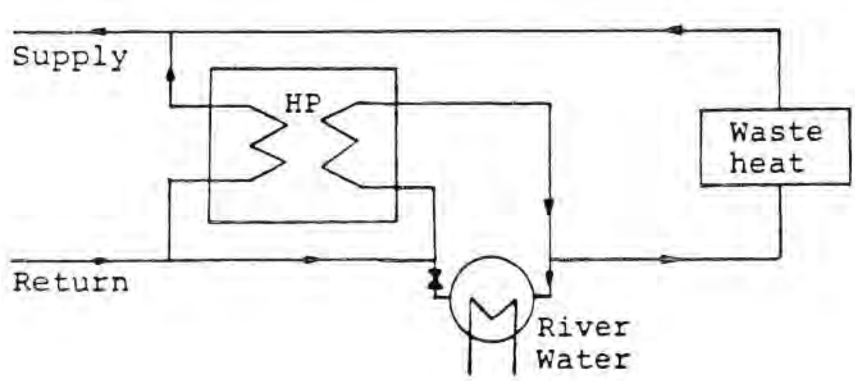


Figure 6.2.1 Principal Scheme of the Saar Absorption Heat Pump Project.

6.2.3 The Absorption Heat Pump System

A steam boiler using cokeoven gas as the fuel supplied heat to the absorption process.

The absorption heat pump is based on ammonia and water as working pair. The evaporator system is a two-stage absorption process, and the desorber system has a liquid subcooler and a separation column cooled together with the vapour cooler by the absorber solution. This ensures a high energy performance and a low water content in the ammonia from the evaporator.

Fig. 6.2.2 shows the flow chart of the absorption heat pump, followed by the key words for the drawing in English.

6.2.4 Operation Conditions

The following scheme gives the basic operation conditions of the heat pump:

		Winter	Summer	Range/average
Evaporator temp.	°C	5	25	5-45
Condensation temp.	°C	-	-	50-70
Heat source temp.	°C	30	40	10-50
Supply temp. from HP	°C	90	90	90
Return temp. to HP	°C	50	50	50
Heat from HP	MW	3.5	3.8	4
Total heat from HP system (incl. waste heat)	MW	3.5	3.8	4.0
Total load of heating system	MW	-	-	70
Calculated performance		1.42	1.50	1.53
Primary energy ratio		1.16	1.23	1.3

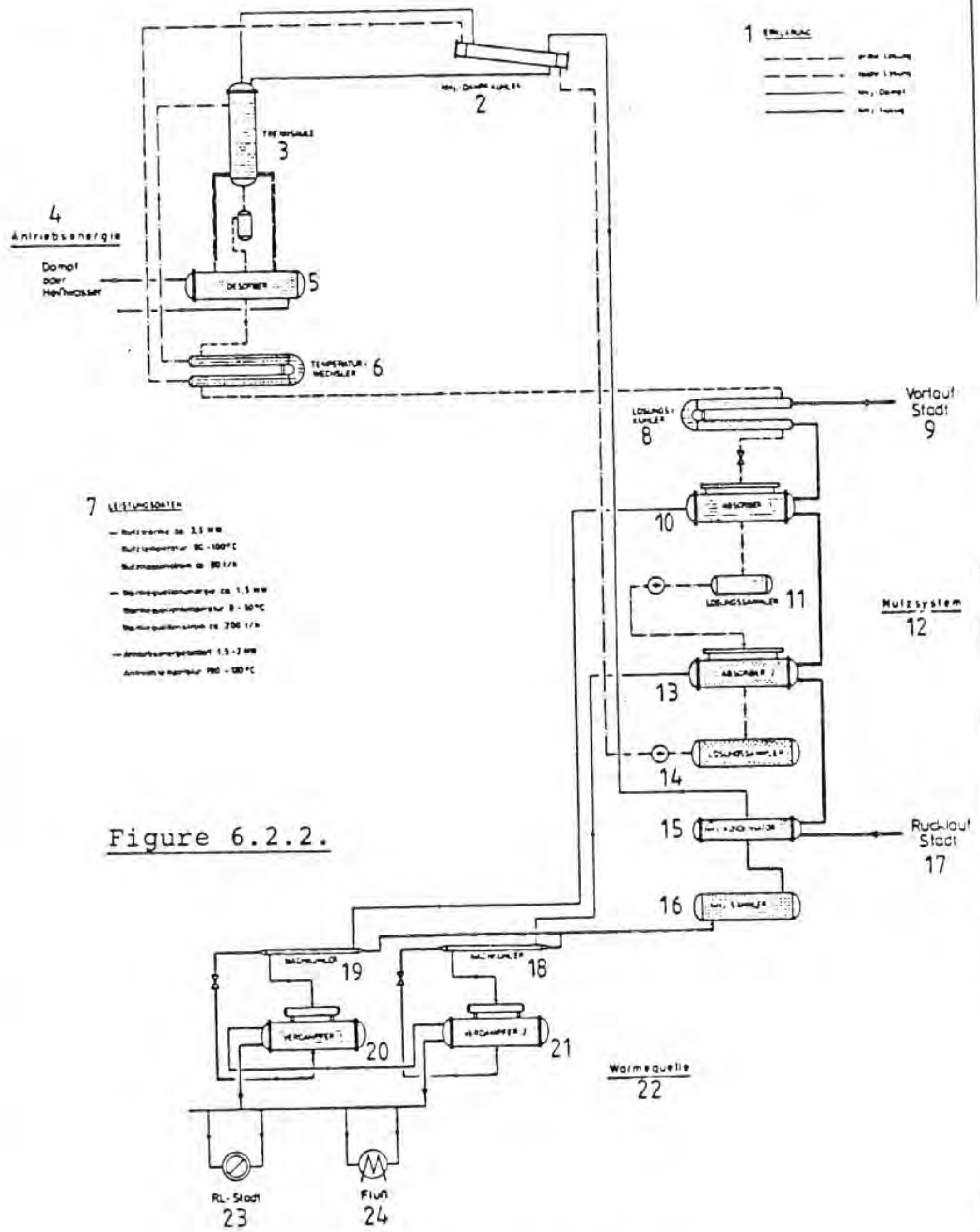


Figure 6.2.2.

Absorption heat pump flow chart

Project outline

Within the framework of a R + D project, sponsored by the "Federal Ministry for Research and Technology", "Borsig", "Saarberg-Fernwärme" and "Saar-Ferngas" have been operating a plant which is integrated into the SAAR DISTRICT HEATING MASTER LINE (location: SFW's main station in the Saarwiesen) as a joint venture, since November 1980.

Absorption heat pump flow chart

1. Key:
   low content solution
   high content solution
   NH_3 vapour
   NH_3 liquid
2. NH_3 vapour cooler
3. separation column
4. driving energy
   steam
   or
   hot water
5. desorber
6. heat exchanger
7. performance data

- effective heat	c. 3.5 MW
" temperature	90 - 100° C
" mass flow	c. 80 t/h
- heat source energy	c. 1.5 MW
" " temperature	0 - 50° C
" " flow	c. 200 t/h
- driving energy demand	1.5 - 2 MW
driving temperature	190 - 120° C

8. solution cooler
9. supply flow - town
10. absorber 1
11. solution collector
12. consumers
13. absorber 2
14. solution collector
15. NH_3 condenser
16. NH_3 collector
17. return flow - town
18. recooler
19. "
20. evaporator 1
21. " 2
22. heat source
23. RF - town
24. flow

6.2.5 Fuel Savings

The potential fuel savings of this 4 MW absorption heat pump plant using industrial waste heat as heat source can be estimated by assuming a utilization time of 7000 hours per year. This high figure is obtained because the capacity of the plant only covers a small part of the total heat demand.

Total heat production:	
4 MW x 7000 hours =	28 GWh/y
Fraction of heat demand:	26 per cent
Fuel consumption 28 GWh/y/1.3 =	21.5 GWh/y
Fuel reduction & fuel costs reduction	
1 - 0.85/ER = 1 - 0.85/1.3 =	35 per cent
Energy savings per year	
28 GWh/y x 0.35 x 1/0.85 =	11.5 GWh/y
=	1000 toe/y

6.2.6 Investment and Operation (1980)

The investment in this 4 MW demonstration plant was recorded to about DM 550 per kW, but a commercial unit can be estimated at between DM 600 and 700 per kW. The estimated total investment in a 4 MW commercial unit is therefore DM 2.6m. The heat pump itself is an open outdoor installation. A quite new district heating central with building and peak load boilers has to be build; the total cost will at least be 900 DM/kW or DM 3.6m.

The economic key figures as an energy saving investment are:

Estimated investment in 1980	
- excl. the transmission line:	DM 2.6m
Operation costs (assumed):	4 per cent
Investment per toe saved yearly	
DM 1.7m/1000 toe/y	DM 2600/toe/y
Assumed additional investment for	
connection pipe and projecting:	20 per cent
Estimated cost per toe saved:	DM 3120/toe/y

6.2.7 Economy

If the plant is regarded as an energy saving investment, the fuel savings is to pay the operation costs and capital costs of the plant.

An indication of the economy can be calculated at a general fuel cost of DM 460 per toe.

Income relative to investment DM 460/toe/DM 3120/toe/y =	15	per cent per year
Deduction of operation costs gives a capital return of (15 - 4) =	11	per cent per year
Simple pay back time	9	years
Depreciation of the plant over 15 years gives a return on the investment of	8.3	per cent per year

6.2.8 Experience from Projecting and Construction of the Absorption Heat Pump.

In the design phase it was found, that special attention should be paid to the system design.

Contrary to the general expectation, the heat utilization system should be coupled as condenser-reflux condenser-Absorber, because the system pressure otherwise would be too high.

It was also found, that the solution pump should be chosen with a pressure reserve and that care should be taken regarding the power consumption at part load. This indicates a frequency regulation of the pump. The reflux condenser should be outsized to give a proper function at high load; i.e. sufficient low water content in the ammonia. Special care should also be taken of the vertical position of the heatexchanger in the system in order to exploit the gravity force to transport solutions in the system.

The cost of the absorption heat pump system is registrated to 45o-5oo DM/kW in 1982, however, it is believed that the commercial cost is 6oo-7oo DM/kW for the plant, besides the cost of connectionto the district heating system.

The system design used in this project is recommended to be used for cooling return water in a district heating net, with the purpose to increase waste heat utilization from an industrial process or to increase the transport capacity of district heating transmission pipes from a combined heat and power plant.

6.2.9 Experience from test running of the plant.

Some problems with the regulation of the plant were experienced during test runs, but these were solved by minor changes in the regulation equipment.

In the following test periode, the plant produced 15.000 MWh with a broad spectrum of operation parameters and the extreme values of operation was tested.

The results from this test periode was, that the energy ratio of the absorption process, to a large extent, corresponded to the projected value and the total energy ratio was slightly lower due to a low efficiency of the steamboiler.

Furthermore it is recommended that the absorption heat pump is projected with the steam boiler capacity covered by at least 2 separate boilers.

6.2.10 Conclusions.

From this project it is concluded, that it is possible to use the absorption heat pump principle for capacities of more than 2 MW for an operating temperature of 70-120°C.

With a difference between the evaporator temperature and the supply temperature of 65°C a total energy ratio of 1.2-1.3 can be reached and with a temperature difference of 85°C, the energy ratio would be 1.1-1.2.

It is still an open question whether absorption heat pumps are economically competitive with conventional heat generating systems and to gas engine driven heat pumps.

The efficiency of absorption heat pumps is lower than of gas engine driven heat pumps, so the advantage of the absorption heat pump is the possibility of using fuels of lower quality as i.e. coal, straw and wood.

However the energy prices of gas and especially coal seem still too low to bear the investment in an absorption heat pump.

In Western Germany there seems to be no plans for further installations of absorption heat pumps suited for the above mentioned high temperature range.

6.2.11 Key Figures for 4.0 MW Absorption Heat Pump

The test running of the plant showed no major changes in the technical performance, but the installation cost has been underestimated in the feasibility study.

The key figures for international comparison of heat pump plants according to the test results and prices regulated to January 1985 with 5 pct. per year are:

1) Heat output	W	4.0	MW
2) Investment 1984/85	I	3.80	M.DM
3) Operation/Investment	OC	4	%
4) Energy delivered	ED	26.000	MWh/y
5) Full load operation	-	6.500	h
6) Yearly energy saving	ES	11.500	MWh/y
7) -"-	ES	1000	toe/y
8) Investment per ES	IES	3792	DM/toe/y
9) Relative energy saving	ES%	35	%

6.3 ABSORPTION HEAT PUMP IN WAIBLINGEN, W. GERMANY

6.3.1. Owner and Location

The heat pump plant is located in the town of Waiblingen in Germany and is owned by the municipality of Waiblingen.

6.3.2 Heating System

The heating system supplies heat to six groups of buildings and a district heating net links these buildings with the heat plant.

Five of the buildings, the town hall, a restaurant, a public swimming pool, a shopping centre and a hospital are designed for high temperature heating water 90/70°C (supply return). In addition to these loads there is a new city hall and a new sewage plant which is designed for this heat pump and has a low temperature heating system (60/40°C).

The district heating supply water heats the 5 high temperature loads, town hall, public swimming pool, restaurant and shopping centre and a part of the return water heats the city hall. Fig. 6.3.1 shows the principal lay-out of the heating system. The total production is 16.7 GWh per year with a maximum load of 6.5 MW.

The heat pump has a performance of 2.5 MW and supplies 77% of the heat production, which is 12.86 GWh per year. When the ambient air temperature drops below 8°C the heat pump performance becomes too low, but the heating system has two boilers for additional heating and reserve with a capacity of 7.0 MW (Fig. 6.3.2).

The supply temperature from the heat pump is 65°C. The boilers can increase the temperature to 110°C.

The total project was completed in August 1983.

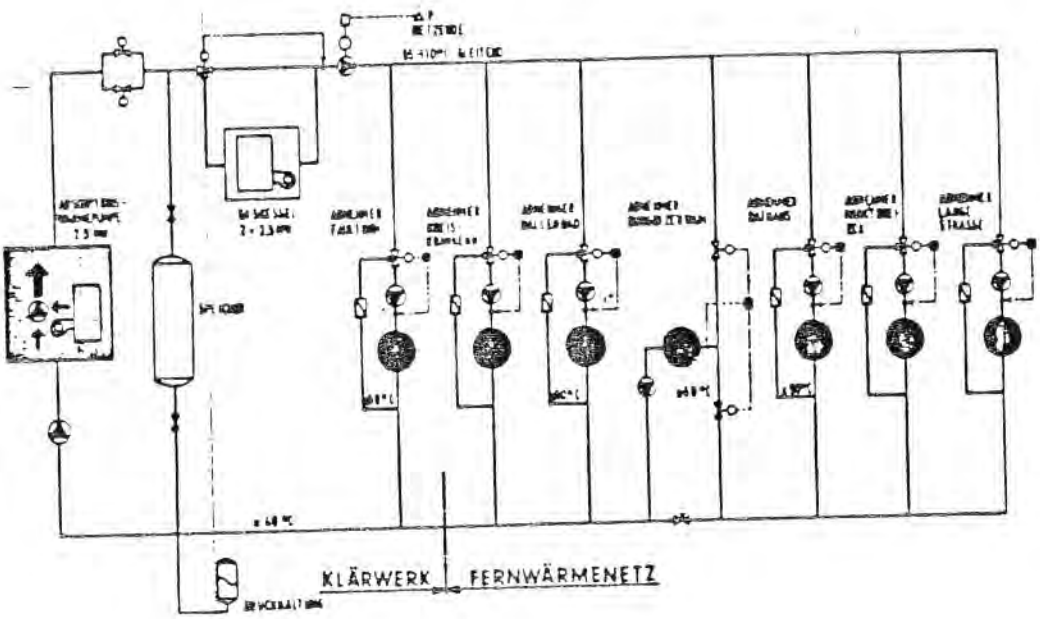


Fig. 6.3.1 The heating system

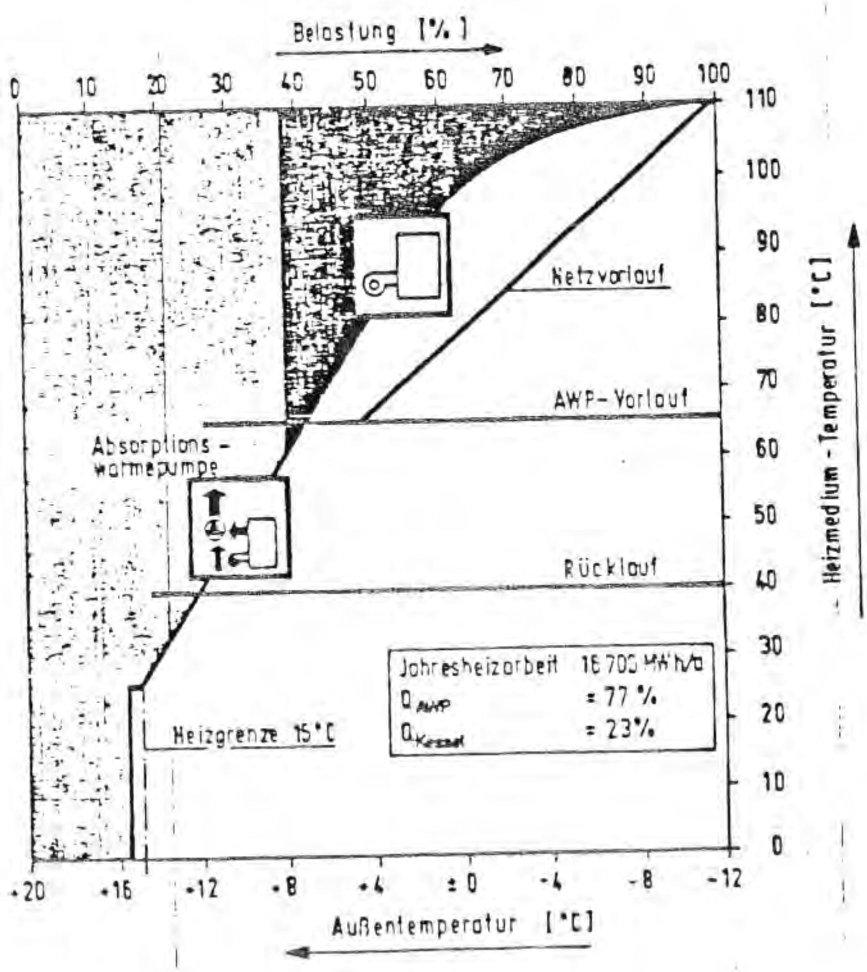


Fig. 6.3.2 Making available the annual heat output

6.3.3 The Heat Pump System

The heat pump is an absorption heat pump with water and ammonia as working pair. A principal lay-out is shown in Fig. 6.3.3. The water ammonia solution in the boiler is heated by a gas burner.

The bio gas produced in the sewage treatment plant is to be utilized effectively by an alternative bio gas/natural gas operation of the absorption heat pump.

The return water is first heated in the condenser and absorber, then the water cools the rectifier, and finally the water is heated by exhaust gas from the burner.
The ammonia from the condenser is subcooled by the ammonia gas from the evaporater.
Treated sewage is cooled by the evaporator. The water has a temperature ranging from 8 to 20 °C and is cooled about 5°C.

6.3.4 Calculated Operation Conditions

The winter operation conditions are shown for typical january conditions and the summer conditions are average for the summer months.

		Winter	Summer	Yearly average
Evaporator temp.	°C	1	1	
Condensation temp.	°C			
Heat source temp.	°C	10	18	
Supply temp. from HP	°C	65	65	
Return temp. to HP	°C	40	40	
Heat from HP	MW	(2.5)	(0.75)	
Total heat from HP system (incl. waste heat)	MW	2.5	0.75	
Total load of heating system	MW	6.5	0.75	
Calculated performance				
Primary energy ratio		1.32	1.42	1.35

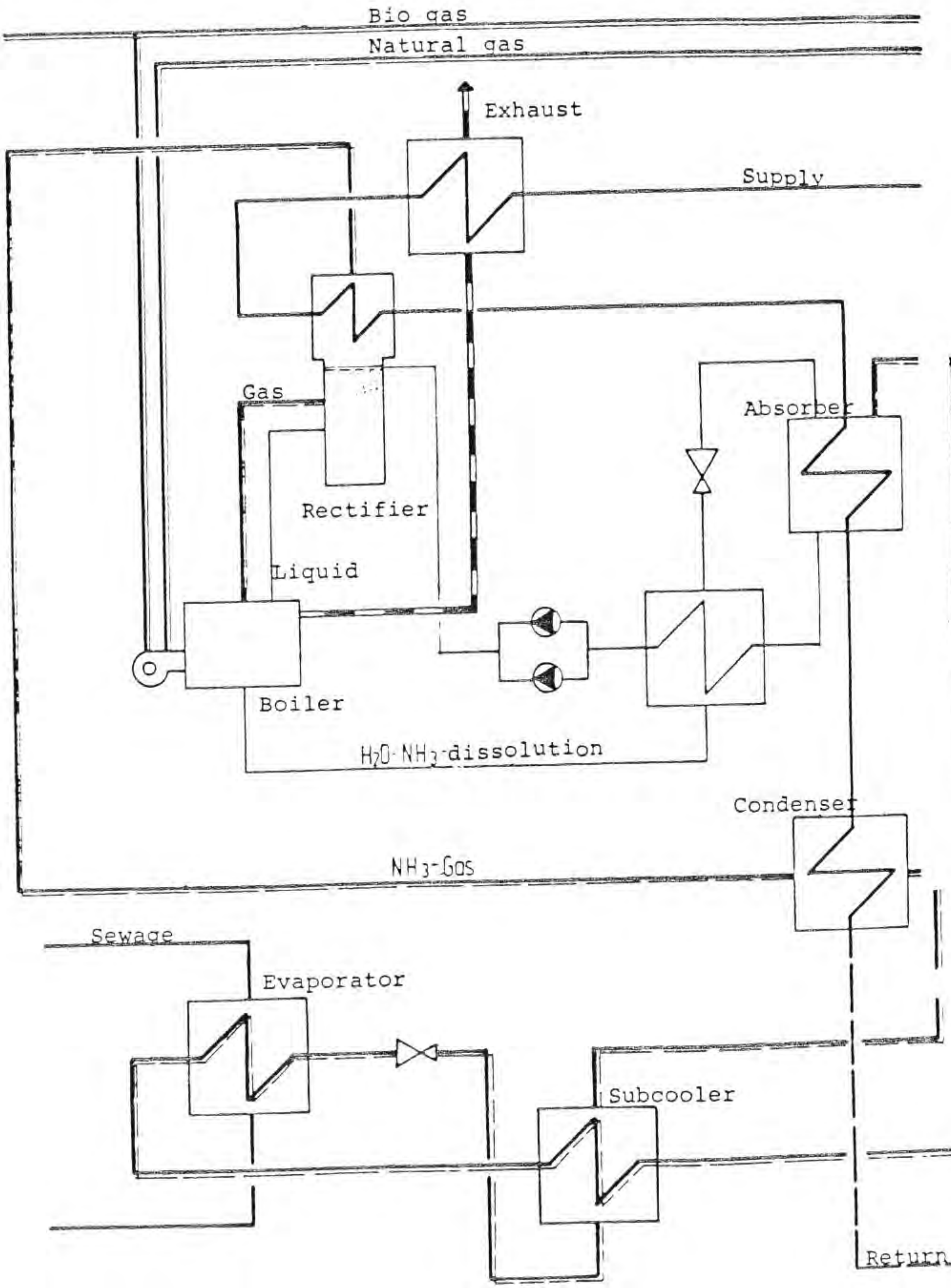


Figure 6.3.3.
Principal lay-out of the heat-pump.

Heating capacity of absorption heat pump	2,500 KW
Heating capacity boiler plant	2 X 3,500 KW
Heating capacity, total installed	9,500 KW

6.3.5 Fuel Savings of Absorption Heat Pump Plant

The primary energy ratio is 1.35 calculated as the yearly average. The primary energy saving in per cent amounts to 1 - 0.85/1.35 = 37 per cent.
The yearly heat production is expected to be 16,700 MWh per year, of which 77 per cent is supplied by the heat pump. The fuel savings correspond to 5,600 MWh per year ~ 500 toe per year.

Heat production in heat pump	12.86 GWh/year
Natural gas consumption	6.33 GWh/year
Sewage gas consumption	3.20 GWh/year
Alternative energy consumption in a boiler	15.1 GWh/year
Primary fuel saving	5.6 GWh/year

These calculations are based on a boiler efficiency of 85%, which is obtainable for a district heating central. However a new district heating net has been built in this project and the alternative heating equipment are six smaller boilers. In these boilers an average efficiency of 75% is a more realistic comparison, but then the heat loss in the district heating net also has to be included. The following calculations can be done for the total project:

Oil Saving and Substitution in Total Project.

Yearly heat production	16.7 GWh/year
Annual gas consumption in the heat pump	9.53 GWh/year
Annual gas consumption in the boilers	4.52 GWh/year

Usefull energy at consumers (loss 5 per cent)	15.8 GWh/year
Gas consumption in decentral boilers (ER=0.75)	21.1 GWh/year
Total energy saving (21.07-9.53-4.52)GWh/year=	7.0 GWh/year
Oil substitution (21.07-6.33-4.52)GWh/year=	10.2 GWh/year
	=1 mio liter oil/year

Only a small part of the biogas would have been used if the heat pump had not been build. In this way the oil substitution is reduced to approximately 900.000 liter oil/year.

6.3.6 Investment and Operation (1982)

The district heating line accounts for a large part of the total investment because it was necessary to build the plant rather far from the loads. The total cost of the project can be divided into the following parts:

Investment in the heat pump plant	DM 2.7 m
Cost of district heating system	DM 3.1 m
Reconstruction measures on consumer units	DM 0.5 m
Miscellaneous additional construction cost	DM 4.2 m
Total project	DM10.5 m

Expected operation cost incl. electricity consumption	DM 80.000 per year
Operation cost/investment in heat pump	3.0 % per year

6.3.7 Economy

The investment cost stated below does only include the heat pump unit.

The fuel price is based on natural gas, but the real fuel price is lower, because on third of the gas consumption is bio gas.

Primary fuel savings	5.6 GWh/year
Fuel price (natural gas	DM 64 pr MWh
Fuel cost saving	DM 360,0000 per year
Investment	DM 2.7 m
Operation cost	DM 80,000 per year
Total cost saving	DM 280,000 per year
Simple pay back time	10 years
Investment per saved ton of oil per year	DM 5,500 per toe

If the actual available biogas is included in the calculation, the key figures become:

Primary fuel saving	7.2 GWh/year
Fuel price (natural gas)	DM 64 per MWh
Fuel cost saving	DM 460.000 per year
Investment	DM 2.7 m
Operation cost	DM 80,000 per year
Total cost saving	DM 380,000 per year
Simple pay back time	7.1 years
Investment per saved ton of oil per year	DM 4,200 per toe

6.3.8 Experience during Construction of the Plant

The heat pump system in Waiblingen was built according to plan with the exception of the sewage availability in the night hours.

Through a minor intervention in the sewage-technical process a continuous sewage flow was realized also in the night hours. This was obtained by means of a temporary storage of the flowing non--clarified sewerage during the day time in an available storing basin with mechanical pre-clarification and extraction in the night hours between 1 a.m. and 7 a.m.

The most important recognition for the future planning of such a system is an integration as wide as possible of the heat pump into the functional process of the sewage works.

Previously this was not possible in Waiblingen due to the fact that the approval procedure had already been concluded and the rebuilding measures in the sewage works had already been started.

Consequently the costs of construction performance in Waiblingen could not be transferred to other objects.

Depending on project, sewage availability, sewage utilization, arrangement of components etc. can be realized at lower costs.

At the construction and during the initial operation it also turned out that a distribution of performance to several firms gives rise to guarantee intersections as well as co-ordination and responsibility problems.

A system like this one should be assigned to a competent company as a complete packet.

In Waiblingen the rebuilding measures at the consumers' were separated from the main order.

As a result of the diversification of responsibility the adjustment and touch-up costs to obtain the required return flow temperature of 40° C were higher than expected.

The return flow temperature of the district heating network is the most sensitive factor of the absorption heat pump system (AHP).

The construction and any rebuilding of the consumers to the heat pump conditions must be made consistently in connection with a HP system.

The higher spreading between forward and return flow temperatures connected to the rebuilding measures and the corresponding low water quantity have demonstrably shown no problems in Waiblingen; however, with the exception of a few heat circuits in the complex distribution system of the swimming pool.

6.3.9 Experience from Test and Measurement

So far there has been no unfavourable experience in connection with the system in Waiblingen.

Only the appearance of micro-organisms in the clarified sewage on 3-4 days in 1984 was not expected.

The micro-organisms caused strong contaminations of the automatic reversible flow filter, these contaminations could only be removed by means of manual cleaning.

Through expedient observations and cleaning processes such disturbances can be avoided.

Energetically seen the practical operation of the system shows both positive and negative divergences in relation to the planning data.

In the output measurements a thermal figure of ER = 1.37 was proved, according to contract a thermal figure of ER = 1.3 had been ensured.

The district heating losses had been calculated at 5.5%. In practise more favourable values are seen.

Whereas the original plan of the system provided for a discontinuous operation of the AHP in connection with lack of sewage in the night hours, an intervention in the sewage-technical field made a continuous sewage flow possible, however.

The utilization of the sewage heating is thus improved, and at the same time the switch frequency of the system is reduced, consequently there is a positive effect on the stable and economic HP-operation, and finally there is a positive effect on the operation life of the system.

These positive aspects should be seen in relation to the low heating capacity requirements and the energy consumption of the consumers.

A maximum heating capacity of 6,500 kW on a winter's day was planned, however, a maximum load of approximately 70% of this value is seen during operation.

This is to some extent also the case for the available heating consumption. Wheras 15,800 MWh had been planned, approximately 12,000 MWh are reached according to the preliminary extrapolations.

Calculations of the heating requirements according to DIN 4701 and the securities contained, unexpected low simultaneousness factors inside the building and the total system, and energy savings through rebuilding of the high-temperature consumers to low-temperature systems suitable for heat pumps have had the result that the capacity and consumption values planned have not been reached.

On the consumer side the AHP system of Waiblingen has not yet been fully built-up. Capacity for another 3,000 kW is available.

In the planning phase, constructions for the future were made. Due to economical reasons a lot of investments have been made with a view to the future (for instance the district heating transport system).

6.3.10 Conclusions from the Plant

The advantage of the source of heat of "geklärtes Abwasser" (clarified sewage) was seen first and foremost in connection with the low outside temperatures of down to - 20° C prevailing in January and February. Also at this temperatures the heat pump could be operated, unlike HP systems which use air or surface water as their sources of heat.

The advantage of fermentation gas being available in a sewage plant for the operation of the HP system will be used additionally.

6.3.11 Key Figures for a 2.5 MW Absorption Heat Pump

The efficiency of the plant is expected to fulfil the projected value, so in lack of long term operation experience, the projected values of the energy calculation are used for the key figures. On this basis the plant will have the following key figures:

1) Heat output	W	2.5	MW
2) Investment	I	2.7	M.DM
3) Operation/Investment	OC	3.0	%
4) Energy delivered	ED	12.860	MWh/y
5) Full load operation	-	5.144	h/y
6) Yearly energy saving	ES	5.600	MWh/y
7) -"-	ES	490	toe/y
8) Investment per ES	IES	5.500	DM/toe/y
9) Relative energy saving	ES%	37	%

6.4 NATURAL GAS DRIVEN HEAT PUMP FOR PADERBORN SPORTS CENTRE, W.GERMANY

6.4.1 Owner and Location

The heat pump is built in the sports centre of the town Paderborn between Hannover and the Ruhr.
It is owned by the local heating association.

6.4.2 Heating System

The plant is used instead of a boiler in a new sports centre. The heating system is specially designed for this heat pump, which is the only heating installation to serve the centre. The total heated area including halls, swimming pool and changing rooms covers 7084 m^2. In addition to these heat loads, the heat pump heats the pool water and all tap water.

The total heat load is projected to 2.8 MW, and the yearly heat consumption is 8.95 GWh.

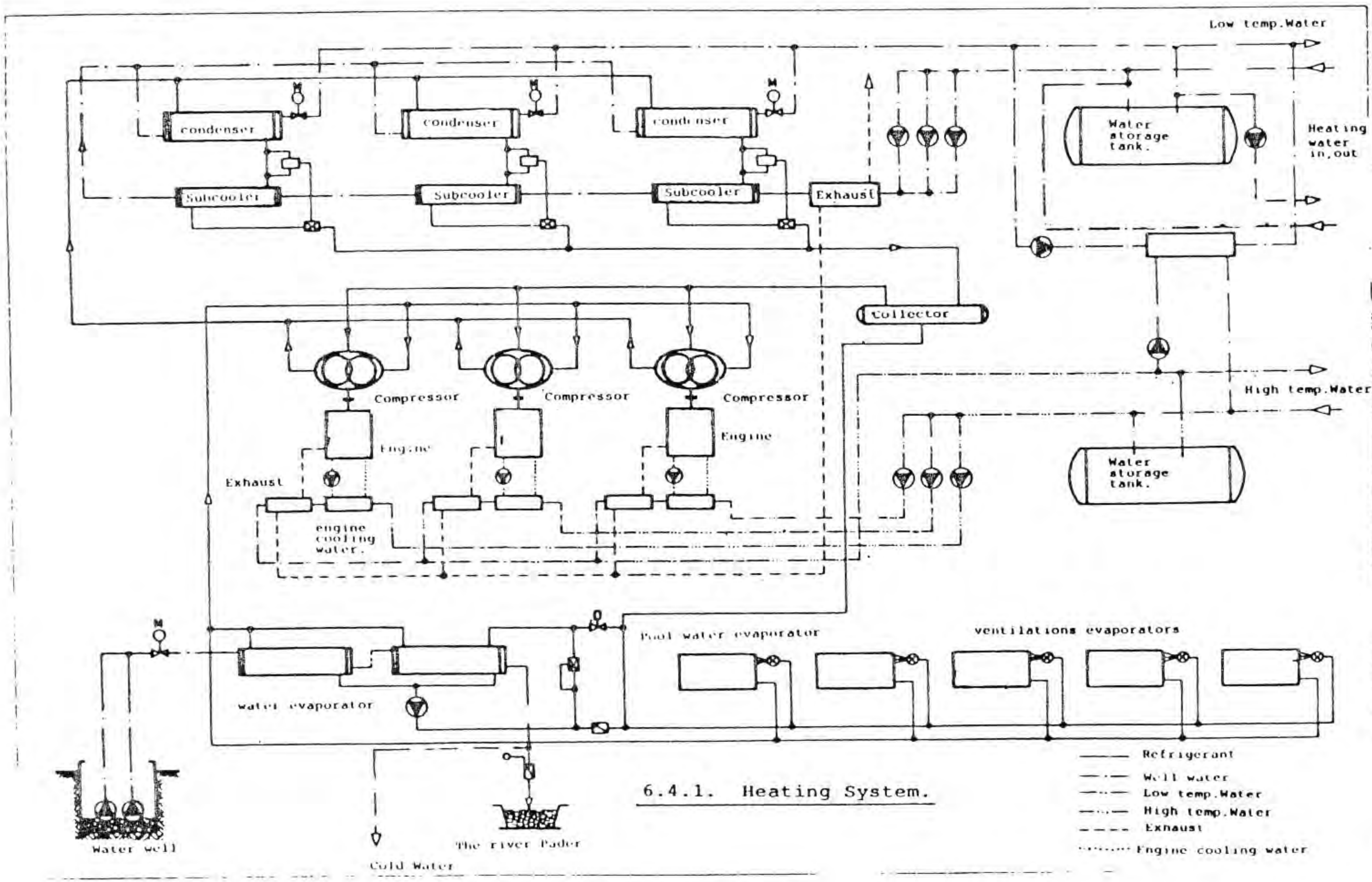
The project was finished in 1977, and the centre is visited by between 1200 and 1400 persons a day.

6.4.3 The Heat Pump System

The system is shown in Fig. 6.4.1. There are three identical heat pump units each with a screw compressor driven by a turbo-charged spark ignition gas engine. Each engine produces 253 kW at the maximum of 1650 r/min. Freon R22 is used as refrigerant.

The condensers have a condensation temperature of 56°C and they heat the return water from 36.4°C to 45.4°C in a low temperature heating system. This system is mainly used for space heating in the sports centre.

The engine cooling water and water heated by the engine exhaust gas are coupled to a separate high temperature heating system. This system is mainly used for additional heating of tap water and pool water.



6.4.1. Heating System.

The main heat source for the evaporators is ground water, and the cooled water is led into the river Pader.

The evaporation temperature is kept at 2°C to avoid ice problems. In addition to this heat source there are five evaporators heated by the swimming pool return water and by the ventilation system.

6.4.4 Calculated Operation Conditions

The heat pump is expected to have a high utilization time even in the summer, because the energy consumption for the pool water and showers is independent of the weather, and about half the heat production is used for heating this water.
The operation conditions are shown below at an ambient temperature of -15°C in the winter and 20°C in the summer.

		Winter	Summer	Yearly average
Evaporator temp.	°C	2		
Condensation temp.	°C	56		
Heat source temp.	°C	9	11	
Supply temp. from HP	°C	45.4		
Return temp. to HP	°C	36.4		
Heat from HP	MW	2.8		
Total heat from HP system (incl. waste heat)	MW	4.65		
Total load of heating system	MW	3.6	2.2	
Calculated performance				
Coefficient of performance		3.74		
Primary energy ratio		2.00	2.4	2.1

6.4.5 Fuel Savings

The high and almost constant temperature in the well and the low condensing temperature mean that the energy ratio is rather high and independent of the weather.

Yearly average primary energy ratio ER =	2.1
Average primary energy saving in per cent	
$1 - ER_o/ER = 1 - 0.85/2.1 =$	59.1 per cent

Heat production in heat pump	8.95 GWh
Alternative energy consumption in a boiler	10.52 GWh
Primary energy use in heat pump	4.26 GWh
Primary energy savings	6.26 GWh

6.4.6 Investment and Operation

The cost of the plant in 1979 was DM 1.69m. A gas boiler installation would cost about half of this amount.

Investment in the heat plant	DM 1.69m
Investment in a boiler plant	DM 0.86m
Marginal investment	DM 0.83m
Operation costs	DM 81,000
Operation costs/investment	4.7 per cent

6.4.7 Economy

The simple pay back time is 14 years as shown below, but if only the marginal investment is considered, the pay back time is 7 years.

Key figures:

Heat production of the plant		8.95 GWh per year
Fuel savings 8.95 GWh/year x (1/0.85 - 1/2.1) =		6.26 GWh per year
Fuel price (1979)	DM	30 per MWh
Fuel cost savings	DM	187,000 per year
Investment	DM	1.69m
Operation costs	DM	81,000 per year
Operation costs/investment		4.7 per cent
Total cost savings	DM	106,000 per year
Simple pay back time		14 years
Investment per saved ton of oil per year	DM	3000 per toe per year

6.4.8 Experience from projecting and building

The heat pump system was designed as a monovalent heating system for a new building complex. It was therfore designed on the basis of a calculated maximum heat loss and yearly heat consumption.

From the first year of operation it was experienced, that the maximum heat loss was only 30 pct of the calculated the dayly energy consumption was 37 pct and the yearly energy consumption was 60 pct of the calculated value.

Here as well as in other projects it can be concluded, that heat pumps are too expensive to be used as the only heat production unit.

6.4.9 Experience from test and measurements.

From the test of the plant, it was experienced, that the power of the gas engine was too small at certain speed of rotation. The power was increased by changing the turbocharger of the engines, but several problems arose with these turbochargers after 3-3500 hours of operation.

Traces of chlorine from the swimmingpool in the lubrication oil of the engines made it necessary to build an air intake with fresh air instead of suction from the engine room.

Corrosion in the silencers for the engines turned up after 6 month of operation.

It showed up to be necessary to cover the walls in the engine room with noise reducing material to reduce the noise from 110 dB to 82 dB in the engine room.

The heat pump showed good operation except that refrigerant was absorbed in the lubrication oil, due to the lack of a non-return valve in the compressor outlet.

The cost of the project was equal to the projected cost except a 10% rise due to general price increase during the project.

6.4.10 Conclusions on the project.

The project has given good experience with gas engine driven heat pumps. The good operation conditions in a building designed with a low temperature heating system give an energy ratio of about 210 pct as projected.

The plant is far too large for the purpose, but it is suggested to supply heating and cooling to a new townhall and concert building with 70.000 sq.m. in total. In this way the economy will be even better than to day.

6.4.11 Key Figures for 4.6 MW Gas Engine Driven Heat Pump

The extreme outsize of the heat pump compared with the heating demand is naturally influencing the economy of the plant. However, the key figures below are calculated on the basis of the demonstrated efficiency and this large investment regulated to January 1985 prices by 5 pct. per year:

1) Heat output	W	4.65	MW
2) Investment	I	2.37	M.DM
3) Operation/Investment	OC	4.7	%
4) Energy delivered	ED	5362	MWh/y
5) Full load operation	-	1153	h/y
6) Yearly energy saving	ES	3750	MWh/y
7) -"-	ES	326	toe/y
8) Investment per ES	IES	7278	DM/toe/y
9) Relative energy	ES%	60	%

If, however, the plant had been used as a base load plant for a much higher consumption, the economy would be much better. As the efficiency of the plant has proved to be good, the following key figures are calculated for a base load operation with a full load operation of 6000 hours per year, assuming that the operation cost will be unchanged.

1) Heat output	W	4.65	MW
2) Investment	I	2.37	M.DM
3) Operation/Investment	OC	4.7	%
4) Energy delivered	ED	27.900	MWh/y
5) Full load operation	-	6.000	h/y
6) Yearly energy saving	ES	19.538	MWh/y
7) -"-	ES	1.700	toe/y
8) Investment per ES	IES	1.394	DM/toe/y
9) Relative energy saving	ES%	60	%

6.5 TWO-MEDIA SOLUTION IN A COMPRESSION AND ABSORPTION HEAT PUMP PROCESS, MANNHEIM, W.GERMANY

6.5.1 Owner and Location

The plant is a test heat pump owned by the heating association in the town of Mannheim and sponsored by the Ministry of Research and Technology.

The plant is situated in Mannheim-Waldhof and was finished in 1983.

6.5.2 Heating System

The district heating net of the town of Heidelberg is planned to be coupled to a combined heat and power plant in Mannheim by means of a new transmission line.

This transmission line is rather long, and to save investments and power for water pumping the return water is planned to be used as a heat source in a heat pump. This heat pump will heat part of the return water, which will then be used as supply water in the district heating system of Heidelberg.

This system means that less water will be circulating between Mannheim and Heidelberg, and smaller pipes will be required.

The plant is expected to have a performance of 60 MW.

A test plant is being built in order to investigate the benefit of the recovery of the return water heat.

The test plant is designed to have a performance of 380 kW and a compressor power of 35 kW.

The heat source has a temperature of 59°C and is cooled to 36°C. The high temperature water is heated to 77°C

6.5.3 The Heat Pump System

Figure 6.5.1 outlines the system.

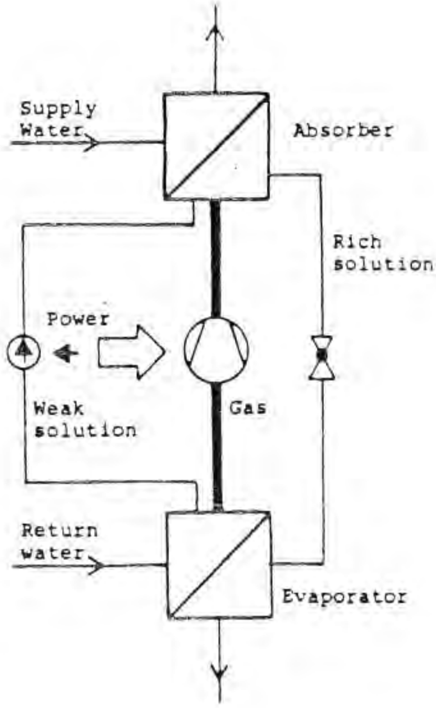


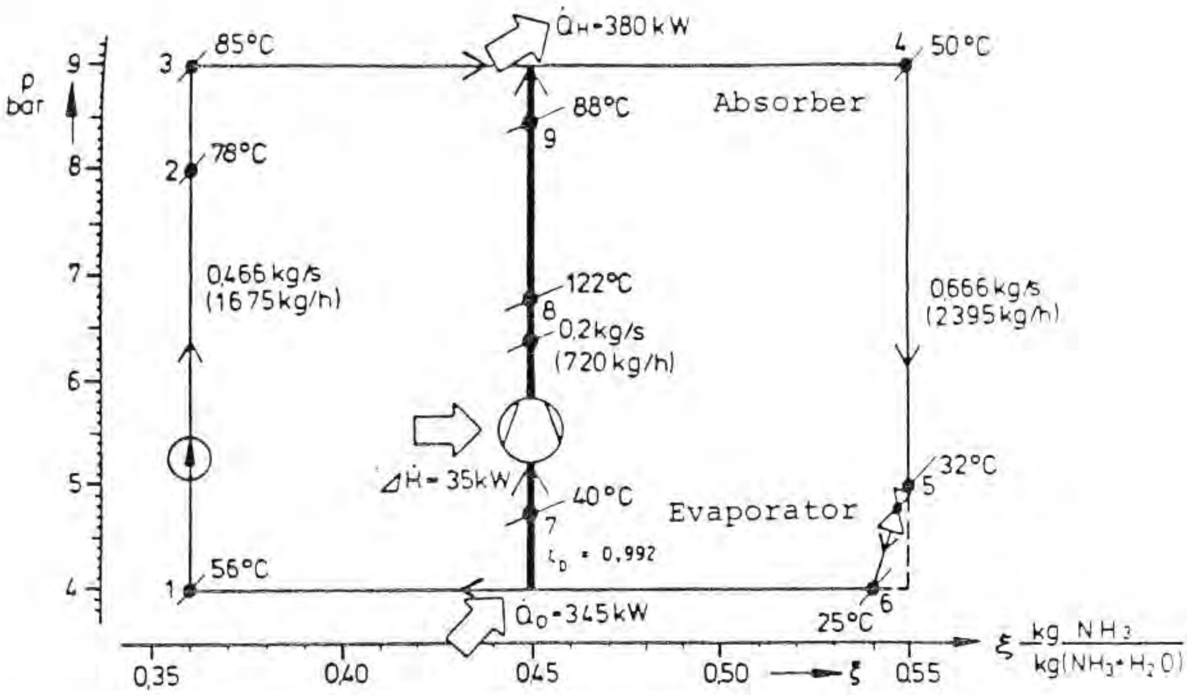
Figure 6.5.1.
Heat Pump System.

The refrigerant consists of a mixture of ammonia and water. From the absorber the rich solution is first cooled from 50 to 32°C and then supplied to the evaporator. The evaporator is heated by water of 59°C, a temperature corresponding to that of the retur water from the district heating net.

Ammonia in particular will evaporate, and the gas is sucked away by the compressor. The solution becomes weak and is then pumped back to the absorber.

Heat exchanges between the solution and the compressed ammonia ga takes place before the solution reaches the absorber. The ammonia gas is absorbed by the weak solution, and the absorption heat is released.

Fig. 6.5.2 shows the operation conditions.



Punkt		1	2	3	4	5	6	7	8'	8	9
h	kJ/kg	132	239	272	174	89	52	1712	1846	1888	1811

Figure 6.5.2. Conditions for 2-Media Heat Pump.

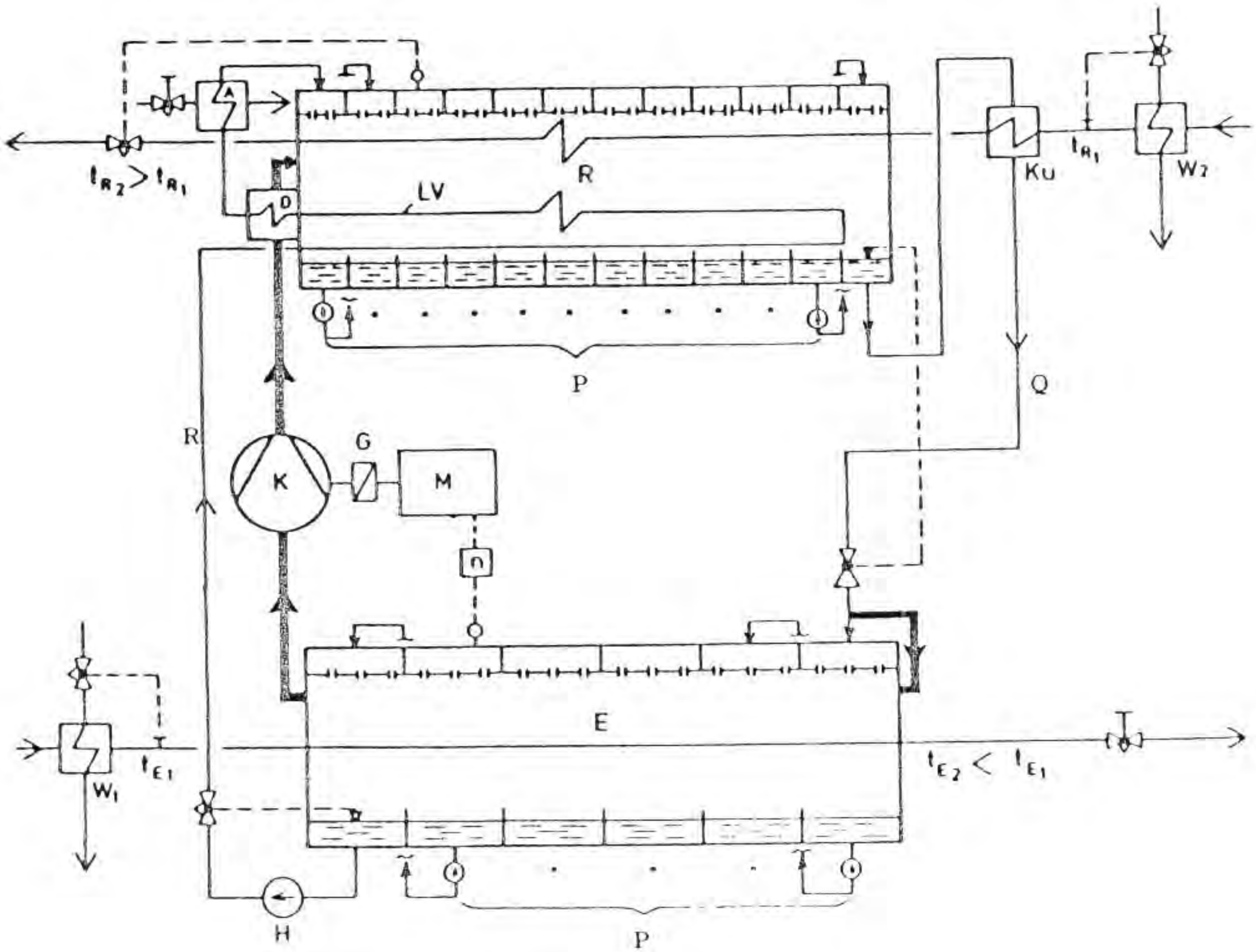
The benefit of this heat pump compared with a one-medium heat pump is a gradual cooling of the return water and a gradual heating of the supply water. This gives a small temperature difference between the solutions and the water, which causes a small entropy production.

Fig. 6.5.3 shows the plant and the graduated construction of evaporator and absorber.

Fig. 6.5.2 shows the changing concentrations in the solution from evaporator to absorber.

6.5.4 Calculated Operation Conditions

The temperatures of the incoming and outgoing solution in the evaporator and the absorber (condenser) are shown below. In this test plant the return water is cooled from 59 to 36°C, and the supply water is heated from 30 to 77°C.



D -Heat exchanger
E -Evaporator
G -Gear box
H -Solution pump
K -Compressor
Ku-Heat exchanger
LV-Solution
M -Engine
P -Sprinkling pump
Q -Rich solution
R -Weak solution
t_{e1}-Water temperature in
t_{e2}-Water temperature out
t_{r1}-Heating water in
t_{r2}-Heating water out
W_1-Heat exchanger
(only in the test plant)
W_2-Heat exchanger
(only in the test plant)

Figure 6.5.3. Heat Pump.

The COP value is calculated to Q/W = 380 kW/35 kW = 10.9. This value is excl. of power to the pumps, but this load is not expected to change the COP value significantly.

The theoretical efficiency of this two-media heat pump is about 2.1 times higher than that of a normal compression heat pump under these special operation conditions.

			Winter	Summer	Yearly average
Evaporator temp.	in/out	^oC	25/56	25/56	
Condensation temp.	in/out	^oC	85/50	85/50	
Heat source temp.	in/out	^oC	59/36	59/36	
Supply temp. from HP		^oC	77	77	
Return temp. to HP		^oC	30	30	
Heat from HP		MW	0.38	0.38	
Total heat from HP system (incl. waste heat)		MW			
Total load of heating system		MW			
Calculated performance					
Coefficient of performance			10.9	10.9	
Primary energy ratio					

6.5.5 Fuel Savings

The purpose of this plant is not so much fuel saving as savings in investments, because the plant only transforms energy from the return line to the supply line.

The saved energy is a small reduction in heat loss from the return pipe due to a lower temperature.

6.5.6 Investment and Operation

The investment in the test plant has been encounted to 243.000 DM in 1982, or about 269.000 DM in price level of 1984.

With a projected heat output of 380 kW this becomes 708 DM/kW.

6.5.7 Economy

For the proposed heat pump the following figures may be applied for considering the economy.

In the heat pump
- the return water temperature will be lowered by 23°C from 59°C and the absorber (condenser) will raise the water temperature by 18°C from 59 to 77°C.
- At a COP value of about 6 the return water will constitute about 40 per cent of the circulating water in the district heating system. The diameter of the transmission line then becomes 38 per cent smaller than if the heat pump were omitted.
This corresponds approximately to a 38 per cent reduction in transmission line costs.

6.5.8 Experience from construction of the plant

The project included a very complex design work, especially on the two counterflow heat exchangers for evaperation and absorption of ammonia. These heat exchangers contains 6 and 12 sections respectively.

6.5.9 Experience from test and measurement.

In spite of carefully cleaning of the system before starting up, several problems with the circulations pumps were experienced because of the impurities. The pumps were changed, but all the pumps in the system showed an extreme low efficiency (6-18%).
A leakage between one of the sections in the absorption heat exchanger made it necessary to by-pass this section. For this reason the performance and the operation temperatures was derated.

It was experienced during the test, that the resorber should have been designed with larger volumes of the solution sumps, with a liquid level of minimum 12o mm.

During the test a C.O.P. of 12 for the heat pump was experienced. This value was reduced to 9.1 if the power for the low efficiency pumps was included.

For a diesel or gasengine driven heat pump with an engine efficiency of 35% and 55% waste heat, this test result will give an energy ratio of 374%. Compared with a normal boiler to cover the peak load of the missing transmission capacity, this would reduce the energy consumption with (1 - 0.85/3.74)= 77%.

6.5.10 Conclusions and comments.

In the project report it is concluded, that this two-media compression heat pump is considerably better than a normal heat pump. With a new design of the resorber and with a higher efficiency of the cirkulation pumps, this special heat pump concept could lead to large energy savings.

As a comment to this conclusion, we would like to pay attention to the fact, that the test conditions for this project will not be used for a normal heat pump. As the heat source has a higher temperature than the useful heat, a simple heat exchanger would transfer 40% of the total heat and the 60% could be exploited by a normal heat pump with a COP of about 8.2, which leads to an overall COP of about 13.7 or larger than the test results.

It may, however, be possible to find operation conditions, where this new concept has better performance than a normal heat pump and further research and development may also bring improvement to the performance.

The most promising application of the two media heat pump will probably be the production of industrial process heat in the temperature range from 150-180 °C from waste heat in the range from 40-80 °C. No conventional heat pump can cover this range of temperatures.

6.5.11 Key Figures for a 0.4 MW Two Media Gas Engine Heat Pump

The figure calculations of the results from a small test plant of a much larger system can only give an indication of the possibilities. By making the standard key figure calculations as an energy saving investment, you compare a peak load boiler with this heat pump system, both installed at the end of a district heating transmission line with a reduced capacity.

By assuming that the heat pump covers the production from 50% to 80% of the maximum heat demand, the utilization time of the plant will be about 1730 hours per year and the energy delivered will be about 13% of the total consumption. The resulting pay back time (chapter 5) for the system indicates the factor of scale necessary to make the system competitive.

On this assumption, the test results and the regulated investment will give the following key figures:

1) Heat output	W	0.38	MW
2) Investment	I	0.27	M.DM
3) Operation/Investment	OC	appr. 5	%
4) Energy delivered	ED	657	MWh/y
5) Full load operation	-	1730	h/y
6) Yearly energy saving	ES	595	MWh/y
7) -"-	ES	52	toe/y
8) Investment per ES	IES	5192	DM/toe/y
9) Relative energy saving	ES%	77	%

6.6 DISTRICT HEATING AND COOLING IN REGGIO EMILIA, ITALY

6.6.1 Owner and Location

The plant is built by the Associated Gas and Water Company AGAG, and the technical and economic feasibility study was prepared by the research organization CISE. The project was supported by the EEC and the Italian National Research Council.

The plant is located in the town of Reggio Emilia in San Pellegrino. Operation began in February 1981.

6.6.2 Heating and Cooling Systems

The plant is a diesel and gas engine combined heat and power plant equipped with heat pumps for load levelling and air conditioning.

In the winter the plant produces district heat and hot tap water. In the summer it produces hot tap water and chilled water for air conditioning. Between summer and winter, when the heating or cooling demand is too low for economic operation, the heat pump is stopped and conventional boilers meet the hot tap water demand.

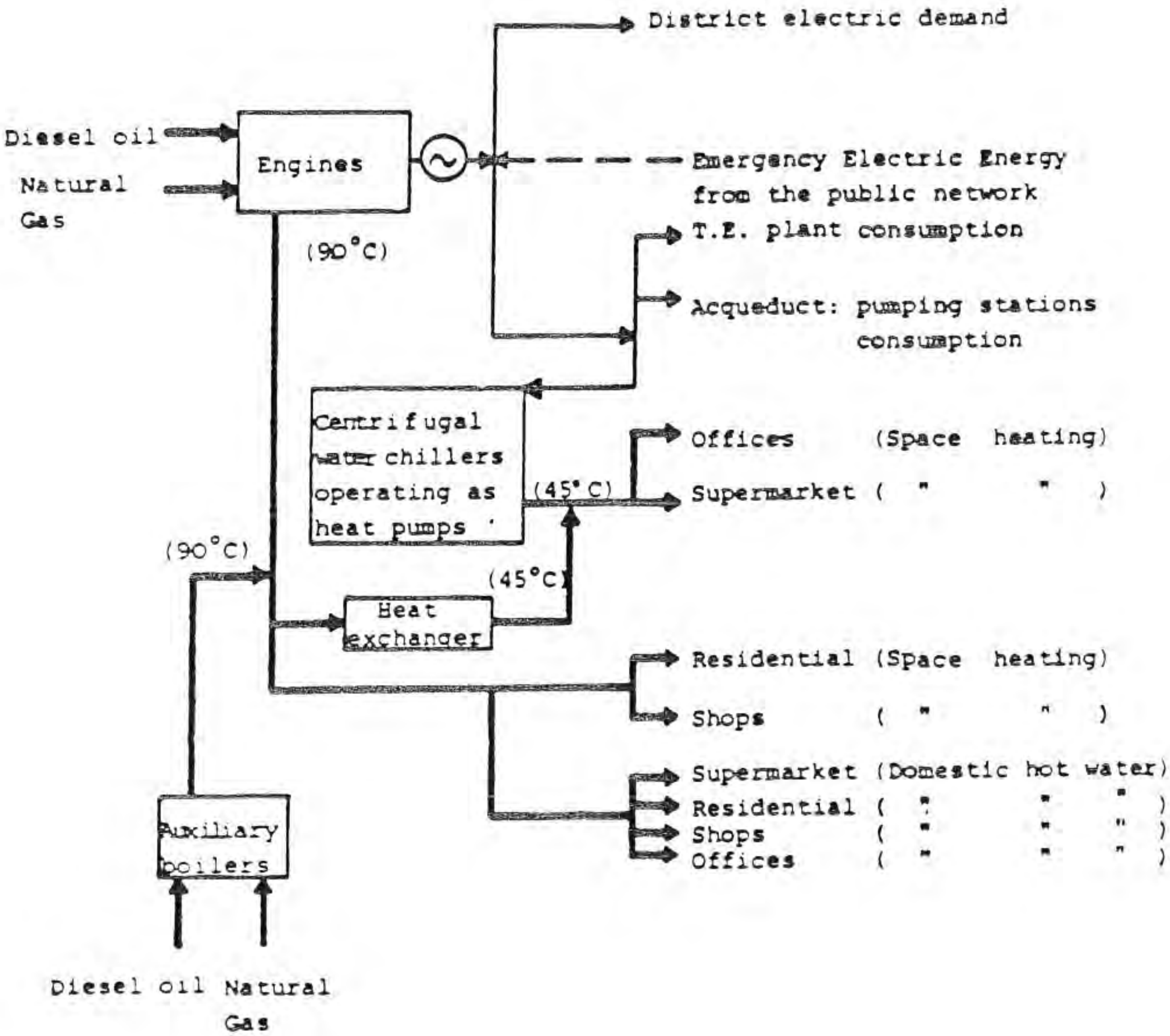
The district heating net is connected to 440 flats covering a total of 120,000 m^3 and facilities and office buildings of 125,000 ^{3}

The district heating net consists of two lines, one high temperature line of 90°C and another line for 45°C water in the winter and for chilled water of 6°C in the summer.

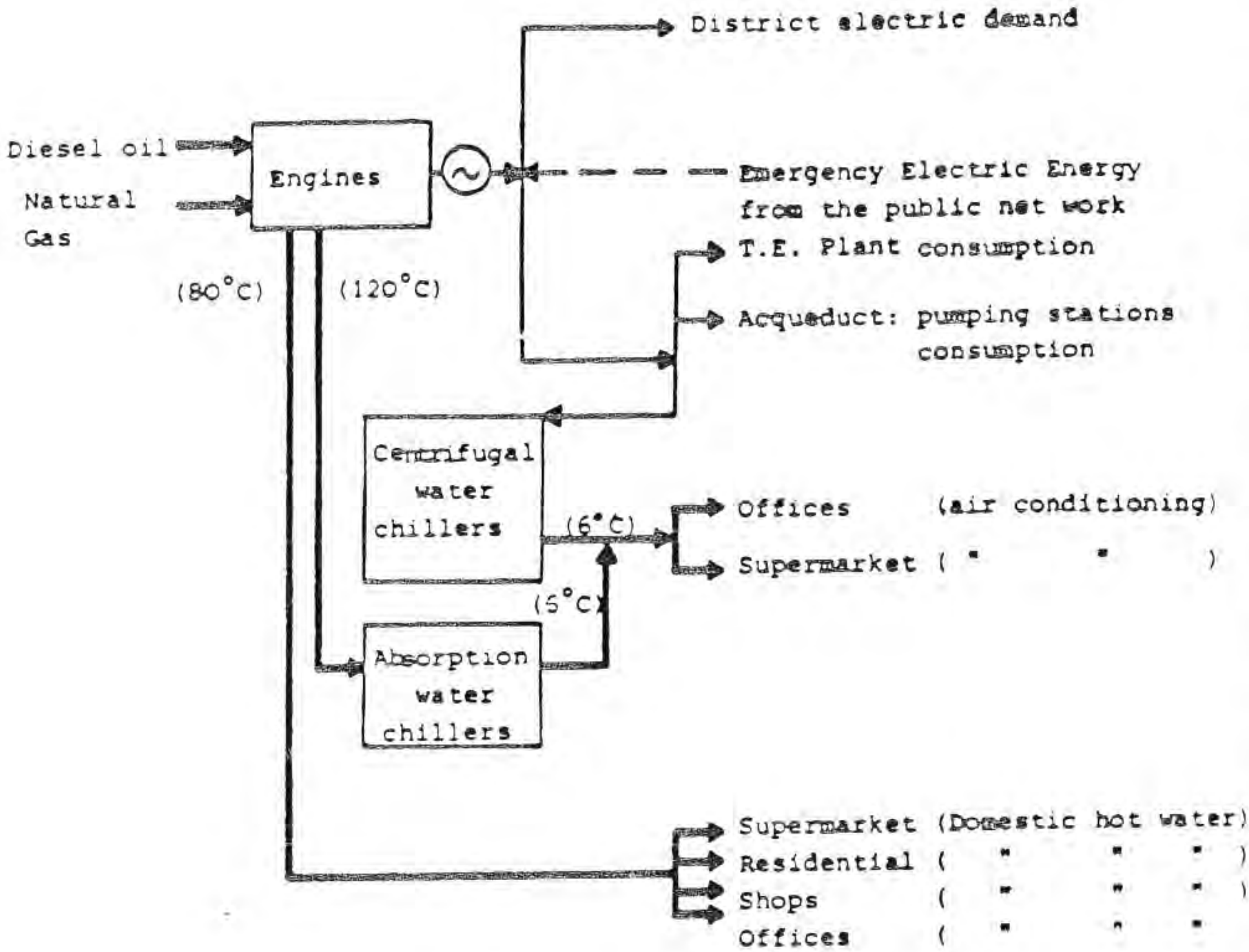
The plant produces:

- Electric power for plant consumption, the municipal aqueduct pumping station, and district buildings in the town. The yearly production is 7.8 GWh.

- 90°C hot water for heating residential buildings and producing hot tap water. The heat demand is 4.6 MW, corresponding to 8.8 GWh per year.



6.6.1 Total Energy Plant Flow Diagram - Winter Operation



6.6.2 Total Energy Plant Flow Diagram - Summer Operation

- $45^{\circ}C$ hot water for heating office buildings and a supermarket. The heating demand is 3.1 MW, corresponding to 4.7 GWh per year.

- $6^{\circ}C$ chilled water for cooling offices and a supermarket. The cooling demand is 2.1 MW, corresponding to 1.4 GWh per year.

Figs. 6.6.1 and 6.6.2 show plant flow diagrams for summer and winter conditions.

6.6.3 The Heat Pump System

The plant is equipped with two natural gas engines each of 0.886 MW and one diesel engine of 1.15 MW.

The three generators are connected to the engines with a total performance of 2.9 MW.

Winter conditions

The heat pump is equipped with two electrically driven turbo compressors each with a power consumption of 300 kW.

The condenser heats the return water in a low temperature heating line from 36 to $45^{\circ}C$; capacity 2.3 MW.

The engine cooling water and exhaust gas deliver heat to the $90^{\circ}C$ high temperature line, and part of this energy is also transferred to the $45^{\circ}C$ system by a heat exchanger. The heat recovered from the engines constitutes 4.3 MW.

The heat source is the municipal aqueduct, which is slightly preheated by the generators, transformers and coolers for the turbocharging. The water source temperature is between 9 and $15^{\circ}C$.

In addition to the heat pump there are two conventional boilers used as boosters, and also when the heat demand only amounts to the supply of hot tap water.

Summer conditions

The production in the summer period comprises hot tap water and chilled water.

The basic district refrigeration load is covered by an absorption refrigerator, which is fed with 120°C superheated water produced by exhaust gas boilers on the engines. The cooling performance is 0.93 MW.

If the cooling demand exceeds the performance of the absorption units, the heat pump is used as a refrigerator with a performance of 1.86 MW.

The water for sanitary facilities is heated by the engine cooling water.

The condenser and part of the engine cooling water is cooled in a cooling tower.

6.6.4 Calculated Operation Conditions

The scheme below gives the figures for the two district heating lines under winter and summer conditions. The COP value under summer conditions is given for the heat pump working as refrigeration plant.

The low primary energy ratio is due to a high electricity production, which causes heat losses when the heat demand is low, and during the summer this energy is lost in the cooling tower.

The figures below refer to the low temperature line/the high temperature line.

		Winter	Summer	Yearly average
Evaporator temp.	°C	4	2	
Condensation temp.	°C	48	35	
Heat source temp.	°C	14	29	
Supply temp. from HP	°C	45 / 90	6 / 80	
Return temp. to HP	°C	36 /	14.5/	
Heat from HP	MW	2.3 /	1.86/	
Total heat from HP system (incl. waste heat)	MW	5.6 / 4.1		
Total load of heating system	MW	3.1 / 4.7	2.1/ 0.8	
Calculated performance				
Coefficient of performance		4.2	3.19	
Primary energy ratio				0.865

The cooling load of 2.1 MW can primary be met by the absorption heat pump of 0.93 MW, if a surplus of high temperature waste heat is available, otherwise - by the electrically driven heat pump of 1.86 MW.

6.6.5 Fuel Savings

A conventional system might consist of boilers with a primary energy ratio (PER) of 0.85, and of a plant with engine driven generators with a PER of 0.35.

If connected, these two plants would have a PER of 0.564.

This plant has a PER of 0.865.

Energy saving in per cent = $1 - 0.564/0.865$ =	35 per cent
Electric energy production	7.85 GWh per yea
Primary energy for this production = 7.85/0.35 =	22.43 GWh per year
Heat production	14.2 GWh per year
Primary energy for this production in a boiler = 14.2/0.85 =	16.7 GWh per year
Primary energy use in a conventional system = (22.43 + 16.7) GWh/year =	39.1 GWh per year
Primary energy use in this plant = (7.85 + 14.2)/0.865 GWh/year =	25.4 GWh per year
Primary energy saving	13.7 GWh per yeai

6.6.6 Investment and Operation

Total investment (1980)	LIT 3,500m	
Operation and maintenance	LIT	70m per year
Operation/investment	LIT	2 per cent

The operation control is carried out by computers connected to measurement points at the plant and by telephone and radio links to periphery points.

6.6.7 Economy

The following figures are based on the expected production and a fuel saving of 35 per cent.

Primary energy saving	13.7 GWh per year
Fuel cost 1982	LIT 30m per GWh
Fuel cost savings	LIT 410m per year
Operation cost	LIT 70m per year
Cost savings	LIT 340m per year
Simple pay back time	10 years
Investment per saved ton of oil per year	LIT 2.85m per toe per year

Besides these cost savings there will probably be a cost saving in the cooling system, compared with conventional small separated air conditioning systems.

6.6.8 Experience during Construction of the Plant

The experimental operation of the plant has been devoted to the evaluation of the performance of the whole system as the equipment utilized in the plant is conventional.

During the three years some minor equipment failure has been encounted but the main problem was the automatic control of the plant. The impact of the innovative concepts used in the design of the plant (cogeneration, district heating and cooling) created some operating problems both in the management of the cogeneration plant and in distribution of heat to the final users. In particular the cooling need of the prime mover was in competition against the request of hot water up to 90°C for the district heating. This caused return water temperature from the users up to 70°C and some difficulties to guarantee an adequate cooling of the diesel/gas engine without intervention of the heat removal emergency system.

For these reasons some modifications of the heat exchangers of the engines have been applied with consequent reduction of the utilization time of the engine.

The automatic control of the plant was originally designed on the assumption of a constant water delivery temperature (90^{o}C) and variable flow according to the heating load. As the heating system in the buildings was realized with three way valves it was difficult to obtain the temperature difference in the district heating net: When the temperature of the apartements had reached the value of 20^{o}C the heat load became zero and through the three way valves the delivery water by-passed directly into the return flow.

This caused an increase of the return water temperature in the district heating net with consequent difficulties to cool the prime movers. To overcome this problem the control system of the plant was modified and the temperature of the delivery water chosen according to the heat load of the building.

After these changes some other problems arose due to simultaneous need for engine cooling and low supply temperature to the users.

These conditions caused the continuous presence of personnel in the control room to operate the plant in a manual mode. To solve the problem a new control system was introduced and the temperature of the delivery water chosen as function of the external temperature and the return temperature in the district heating system.The same problem arose in the heat distribution system at low temperature. Also in this case the control system has been modified from constant temperature and variable flow to constant flow and variable temperature.

Some other modifications have been carried out in order to comply with a remote automatic control of the plant to reduce the cost of personnel.

Regarding the heat pump operation no major problems were encountered during the operation of the plant. However, during the setting up phase some additional smaller problems appeared:

- there were some difficulties in the installation of the computer based alarm system for detecting loss of water in the heating net. The problems were caused by high humidity in the small valve system chambers. The problem was solved by covering the alarm cables with watertight sheating.

- one of the 3/15 KV transformers short-circuited during the starting of the plant.

- some exhaust gas leaks in the heat exchanger system of the methane engine have been repaired, since they caused condensing af water in the insulation.

- Leaks in the heat exchangers for cooling the two generators have also been repaired.

- Some of the heat exchangers in the hot water supply system have been replaced by stainless steel exchangers, as the original exchangers were corroded, due to the hardness of the water and to presence of dissolved gasses in the water.

- the fan speed of the cooling tower ventilators had to be increased, due to insufficient capacity.

- the speed of rotation on the heat pump compressor was increased in order to reach the desired temperature (46°C) easier.

6.6.9. Experience from Test and measurement

The most important problem that caused a non economical operation of the plant was the reduced developement in the amount of users compared to the figure considered as forecast. This reduction is due to the delay in both the construction and the sale of the apartments. Refering to the operational period from May 83 to April 84, the thermal energy sold has been equal to 50% of the figure calculated in the project, while the users connected to the district heating system reached 78% of the total thermal power installed. The same situation goes for cooling need.

The municipality of Reggio Emilia is considering the possibility to extend the district heating and cooling circuit to new users in order to obtain a more economical utilization of the plant.

In table 6.6.9.1 are summarized the measured performances of the plant for the three years of experimental operation while in table 6.6.9.2 are reported the corresponding economic balances forthe same period. In table 6.6.9.3 are summarized the investment costs forthe RETE plant.

Table 6.6.9.1 Reggio Emilia, Total Energy Balance

Energy consumed	1981-1982*	1982-1983*	1983-1984*
Diesel oil (kWh_t)	1.040.968	827.144	629.750
Natural gas (engine)(kWh_t)	2.039.443	5.558.702	6.519.003
Natural gas (boiler)(kWh_t)	6.206.926	3.881.897	4.071.551
Purchase of electric energy (kWh_e)	618.400	498.000	467.400
Energy produced			
Electric energy (kWh_e)	659.145	1.651.637	1.581.838
Thermal energy (sold)(kWh_t)	5.735.986	5.881.325	6.473.838
Cooling energy (sold)(kWh_t)	150.319	225.453	154.349
Total percentage of users connected by district heating	66	66	78
Percentage of electric efficiency of the plant	28.1	30.7	29.2
Total efficiency	65.3	71.2	70.2

*from May to April the following year

Table 6.6.9.2 Reggio Emilia, Economic Balance

Energy sold	1981-1982*	1982-1983*	1983-1984*
Electric energy	69.415.060	172.519.387	197.077.617
Thermal energy	265.462.153	341.951.152	383.489.672
Cooling energy	14.478.726	35.826.733	27.111.600
Energy purchased			
Electric energy	57.516.070	47.371.170	53.297.850
Diesel oil/natural gas	232.853.017	287.983.816	327.793.671
Lubricating oil	5.902.765	13.391.590	15.000.000
Other Costs			
Personnel	80.000.000	100.000.000	110.000.000
Maintenance	60.000.000	100.000.000	100.000.000
Total	-86.959.914	+28.261.876	-8.412.632
Payment of interest	-190.000.000	-190.000.000	-190.000.000
Depreciation (12%/20 years)	-122.000.000	-122.000.000	-122.000.000
Total	-398.955.914	-283.738.124	-320.412.632

* from May to April the following year

All prices are prices of the year in Lire

Table 6.6.9.3 Investment for the RETE Project

Civil structures	600.000.000
Heat plant	1.941.000.000
Heat pump	14.500.000
Auxiliary (fire protection and noise protection)	50.000.000
Control and regulation	296.000.000
Data acquisition and elaboration for R&d activities	70.000.000
Total	3.102.000.000

All prices are prices of 1980 in Lire

6.6.10 Conclusions from the Plant

In the planning stages and in the feasibility study it was foreseen that optimal operation of the plant could be reached after 3-4 years in connection with the finishing of the residential quarter. The performance of some of the fundamental components in the plant seems to be rather close to - or even better than - the expected values. Very satisfactory figures were reached for the yearly efficiency. The plant efficiency, PER, is 0.79 and satisfactory compared to the expected value of 0.81.

The loss of heat in the distribution network was less than 2.5%. The plant has only reached an operation time and heat delivery of about 11 pct. of the expected value, partly due to the low thermal heat demand.

The profit and loss account for the first working year was balancing, but there was no surplus for interests and repayments of the capital invested.

6.6.11 Key Figures for total Energy Scheme

The load on this plant has been so small that it has no meaning to calculate key figures on the basis of the test results. As the test results show a satisfactory efficiency of the plant, the key figures are based on the projected value of the loads and energy supply. The investment is regulated according to the consumers' price index.

1) Heat output	W	5.6	MW
2) Investment	I	5880	m.LIT
3) Operation/Investment	OC	2	%
4) Energy delivered	ED	(22.050)	MWh/y
5) Full load operation	-	2.540	h/y
6) Yearly energy saving	ES	13.700	MWh/y
7) -"-	ES	1.190	toe/y
8) Investment per ES	IES	4.94	m.LIT/toe/y
9) Relative energy saving	ES%	35	%

6.7 DIESEL DRIVEN HEAT PUMP FOR DISTRICT HEATING IN FREDERIKSHAVN, DENMARK

6.7.1 Owner and Location

This plant is built in the north of Jutland in the town of Frederikshavn and is owned by the municipality of Frederikshavn.

6.7.2 Heating System

The heat pump is used in the district heating system of Frederikshavn. The town has a heat demand of about 200,000 MWh per year, and the heat pump is designed to produce 80,500 MWh per year with a maximum performance of 11.5 MW.

In December 1978 the building contract was signed between the manufacturer Burmeister & Wain and the municipality of Frederikshavn. The construction was finished in 1979.

6.7.3 The Heat Pump System

The heat pump is driven by a 3 MW four-stroke turbocharged diesel engine. It is connected both to a 0.7 MW electricity generator and to the compressor through a gearbox. See Fig. 6.7.1.

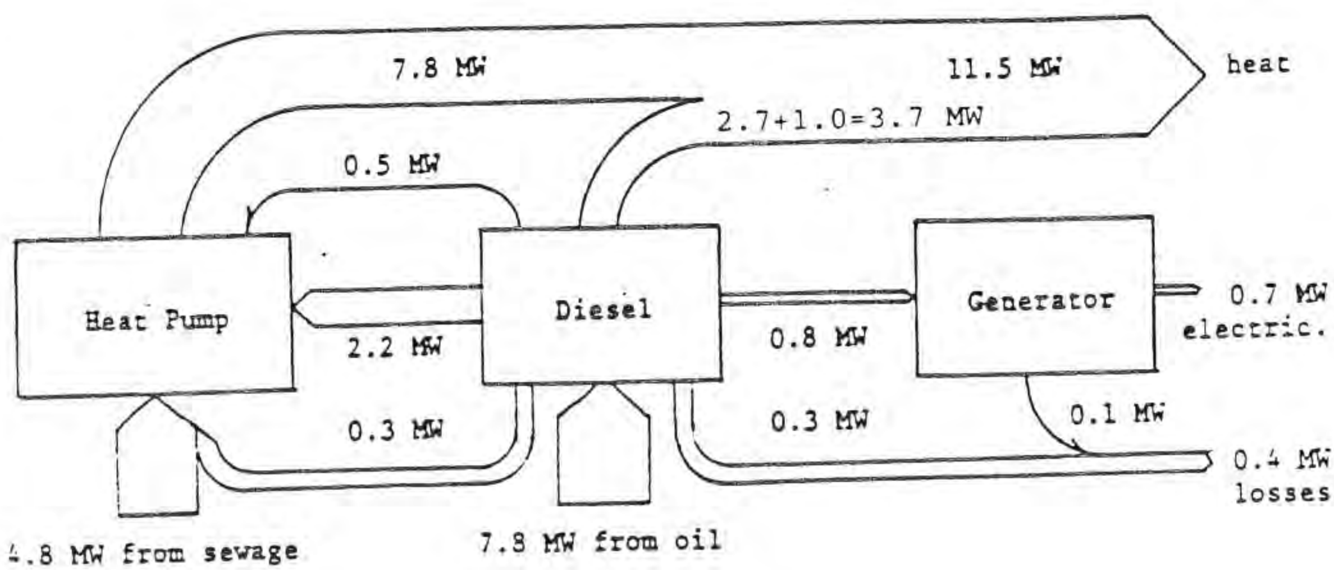


FIGURE 6.7.1 Sankey Diagram for Diesel Heat Pump

The compressor is a two-stage turbo compressor with variable diffuser angles and inlet guide vanes to obtain control of the high efficiency at different operating conditions.

It is necessary to use a two-stage compression because of the high pressure difference, but it also makes it possible to have a flash economizer in the heat pump. An additional evaporator using low grade waste heat from the diesel engine is also connected to this interstage level.

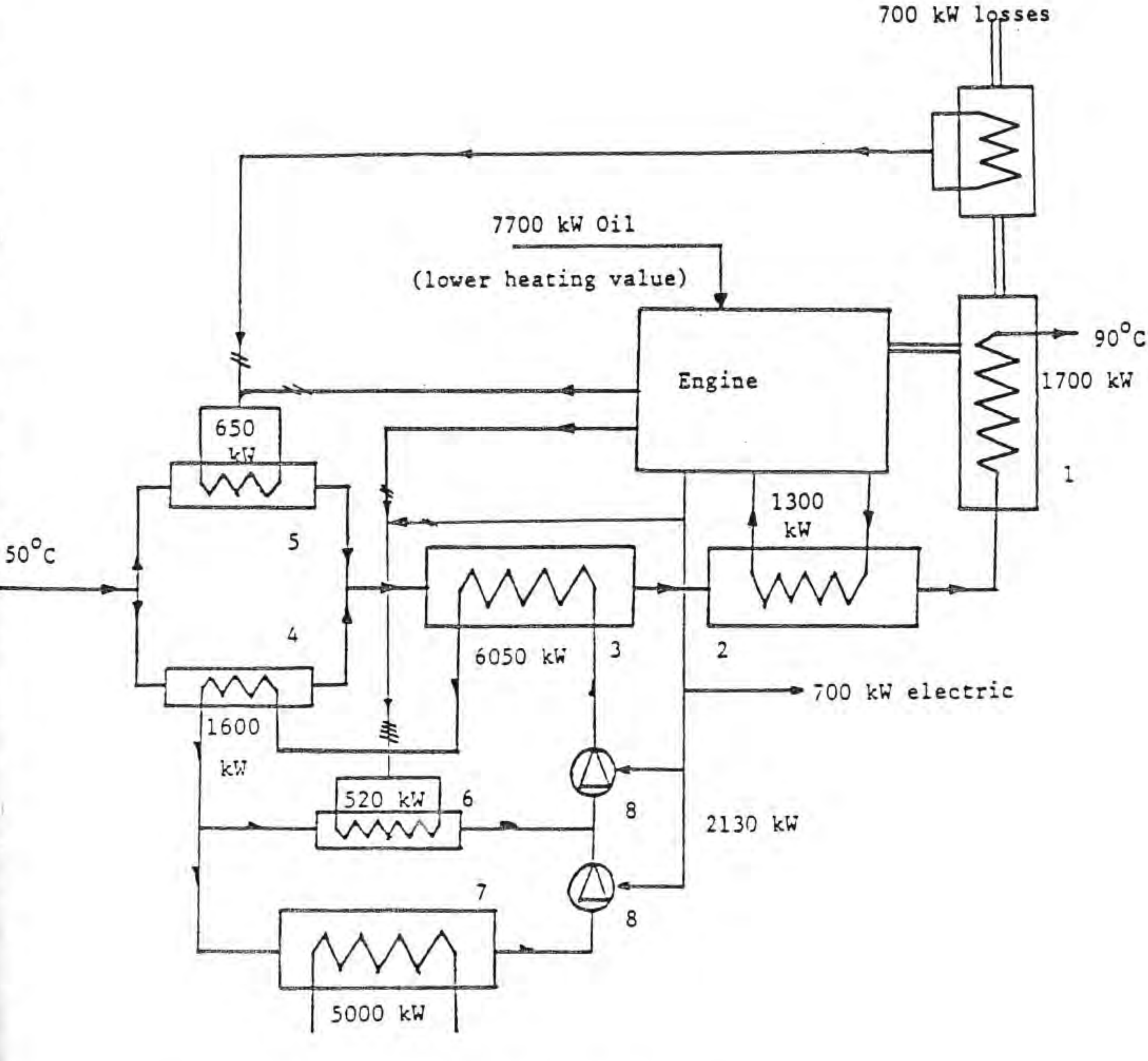
As freons with a high specific volume will provide the highest COP value in turbo compressors, freon R114 is used as refrigerant.

The heat pump is built next to the sewage plant, and cleaned wat from this plant is used as heat source for the evaporator. The sewage plant has a load of 65,000 person equivalent.
The condenser heats the district heating water. The heat production of the condenser is 7.8 MW, and the engine cooling water and the primary exhaust heat supply 3.7 MW and raise the temperature of water from the condenser to 85^{O}C.

After the primary exhaust heat exchanger, the exhaust gas heats the incoming return water from the district system, and finally the exhaust gas is washed by water and the vapour in the exhaust condenses. The washing water is cooled in the evaporator.

Fig. 6.7.2 shows an energy balance for this heat pump.

Figure 6.7.2.
Energy balance for diesel heat pump equipped with power generator



1. Waste heat recovery from exhaust gas
2. Waste heat recovery from jacket cooler.
3. Heat pump condenser
4. Under cooling of liquid refrigerent
5. Middle temperature waste heat from engine
6. Low temperature waste heat
7. Evaporator with heat from heat source
8. Compressors

6.7.4 Calculated Operation Conditions

The figures below show the conditions in winter and summer and the yearly average.

		Winter	Summer	Yearly average
Evaporator temp.	°C	0	8	2
Condensation temp.	°C	78	78	78
Heat source temp.	°C	10	18	12
Supply temp. from HP	°C	85	85	85
Return temp. to HP	°C	50	50	50
Heat from HP	MW	7.8	(8.6)	8.0
Total heat from HP system (incl. waste heat)	MW	11.5	(12.3)	11.7
Total load of heating system	MW	40	8.1	22.7
Calculated performance				
Coefficient of performance		3.54	3.9	3.64
Primary energy ratio		1.47		1.47

6.7.5 Fuel Savings

A significant part of the engine power produces electricity; this lowers the primary energy ratio and fuel savings, but not cost savings.

Average primary energy ratio of the heat pump 1.47

Primary energy saving in per cent

1 - 0.85/1.47 = 42 per cent

Average primary energy ratio of this kind of heat pump without generator 1.87

Primary energy saving in per cent

55 per cent

6.7.6 Investment and Operation

Investment in the heat pump (1981)	DKr 19m
Connections to district heating system	DKr 2.4m
Operation costs (1981)	DKr 0.6m per year
Operation costs/investment	3.15 per cent

In addition to the investment, about DKr 5m was used for research.

6.7.7 Economy

Savings and pay back time are calculated both for this heat pump and for a similar heat pump without generator.

Heat pump equipped with generator:

Investment 1981	DKr 19m
Heat production	80.5 GWh per year
Primary energy use in a heat pump	54.6 GWh per year
Primary energy use in a boiler	94.7 GWh per year
Primary energy saving	40.1 GWh per year
Fuel price 1981	DKr 200 per MWh
Electricity price 1981	DKr 580 per MWh
Fuel cost savings	DKr 8.0m per year
Savings on account of generator	DKr 2.0m per year
Operation costs 1981	DKr 0.6m per year
Total cost savings:	DKr 9.4m per year
Simple pay back time	2 years
Investment per saved ton of oil per year	DKr 5,300 per toe per year
Operation costs/investment	3.15 per cent

Heat pump without generator:

Investment 1981	DKr 17m
Heat production	75.4 GWh per year
Primary energy use in a heat pump	50.0 GWh per year
Primary energy use in a boiler	88.7 GWh per year
Primary energy saving	38.7 GWh per year
Fuel price 1981	DKr 200 per MWh
Fuel cost savings	DKr 7.7m per year
Operation costs	DKr 0.6m per year
Total cost savings	DKr 7.1m per year
Simple pay back time	2.5 years
Investment per saved ton of oil per year	DKr 4,900 per toe per year
Operation costs/investment	3.5 per cent

6.7.8 Experience during Construction of the Plant

During the construction of the plant, changes were made on the following subjects:

Flow switches were changed to pressure difference measurement because of faulty signals from the flow switches.

A contraction was made at the top of the chimney in order to get a higher exhaust gas velocity at the outlet.

The exhaust gas scrubber was originally designed so that fouling in the system called for manual adjustment of valves. This system was redesigned so that the adjustment now takes place automatically.

Problems with electrical noise in the alarm system has been removed by introducing a procedure where the computer checks the validity of all incoming signals and excludes faulty signals.

The cost of the plant is as shown in paragraph 6.7.6 but the performance is a little lower because it is necessary to control the compressor with a margin away from the surge point.

6.7.9 Experience from Test and Measurement

In the period June-September 1981 a running-in and measuring programme was carried out. The running-in of the regulation system was carried out in October and the plant was finally handed over to the town in October 1982.

During the first test of the heat pump, both the turbocompressors broke down, presumably because of a fatigue failure in an inlet guide valve. After the fabrication of two new compressors, the test and measuring continued. The compressors have operated with no failures since.

The measurement programme and the regulation system have in all 147 temperature sensors, 62 pressure sensors and 11 flow sensors, most of them recorded by the computer. This facility is important in detecting reasons for an abnormal function and for performance calculations of the plant as a whole and the component parts.

The result of operation without electricity generation gives a primary energy ratio, PER, of 1.81% on heavy fuel oil and of 1.87% on diesel oil, while the projected value was 1.84% on heavy fuel oil.

The alarm system caused problems during the test running because of many unexpected alarms and shut downs. These problems arose from the following sources:

26% electric noise
24% flow switches
18% scrubber system
16% computer hardware
10% programme errors
6% misc.

The main problem with this heat pump has been a large reduction in yearly operation time due to lower heat demand and insufficiency of the heat source.

Even though 20% more houses has been connected to the district heating net since the plant was projected, the total heat consumption has decreased with 10%.

A refuse burning plant has the highest priority in heat production, so the remaining heat load is often below the minimum capacity of the heat pump in the summertime and the heat pump is stopped.

Measurements of the amount of sewage water before the construction of the plant showed a much higher result than the reality. In addition to this, the variation of sewage water was much larger than expected and became more critical. The water flow was often below the minimum for operation in the night time. An existing sewage water reservoir is now used as storage tank to increase the sewage water flow during the nights.

From October 1982 the plant has been in operation without any major problems and the heat production in 1983 63 GWh, which should be compared with an expectation of 80 GWh.

The amount of oil saved is 35 GWh which gives a fuel cost saving of Dkr. 7.0m per year in 1981 prices.

It was planned that the heat pump plant should work without personnel but it turned out to be necessary to engage an operator to keep the running of the plant stable. This caused an increase in the first estimated operation cost from 400.000 Dkr to 600.000 Dkr.

6.7.10 Conclusion from a 10 MW Diesel Driven Heat Pump

The heat pump has a very high primary energy ratio which has been possible due to the high performance turbo compressor with economizer connection.

The high performance would be hard to reach with a screw compressor, but turbo compressors are only an advantage in plants with a high heating capacity.

After all operation problems now have been solved the plant shows a very high fuel cost saving and the pay back time is still below 3 years. This also shows that the plant has a size where the factor of scale has an important influence on the economy.

In a large heat pump it is easier to reach a high efficiency even though the temperature difference is large and the specific investment is reduced because of the large size of the plant.

6.7.11 Key Figures for a 10 MW Diesel Driven Heat Pump

The key figures has been calculated according to the test result for running on heavy fuel oil without the generator and the investment is based on the cost of the heat pump without a generator. The prices are regulated to the end of 1984.

1) Heat output	W	10.5	MW
2) Investment	I	22	mDkr
3) Operation/Investment	OC	3.5	%
4) Energy delivered	ED	63.000	MWh/y
5) Full load operation	-	6.000	h/y
6) Yearly energy saving	ES	39.300	MWh/y
7) -"-	ES	3.420	toe/y
8) Investment per ES	IES	6.433	Dkr/toe/y
9) Relative energy saving	ES%	53	%

6.8 NATURAL GAS DRIVEN HEAT PUMP FOR DISTRICT HEATING IN EJBY, DENMARK

6.8.1 Owner and Location

The heat pump is located in the town of Ejby on the island of Funen, Denmark. The Ejby district heating association has the right to take over the plant after completion of the test programme.

6.8.2 Heating System

The construction of the national natural gas system is soon to be completed in Denmark.
The Danish Oil and Natural Gas Company and the local natural gas association on Funen wished to investigate the benefits of gas driven heat pump centrals. Ejby was chosen, because the town was connected to the natural gas supply net at an early stage.

The yearly district heat production in Ejby is 16.6 GWh with a maximum load of 4.24 MW.

The heat pump is designed to yield 8.8 GWh per year, and the maximum heat supply to the district heating system is 1.3 MW. The plant is expected to be in operation by mid-1984. The heat pump central is financed by loans and subsidies by the Ministry of Energy, the Danish Oil and Natural Gas Company, and the Danish Research and Development Fund.

6.8.3 The Heat Pump System

The plant is powered by two natural gas engines of 170 kW, each connected to a piston compressor and a generator.

The generator load is 110 kW, which enables the engines to run at a high load even when the heat demand or the heat pump capacity is low.

This gives the gas engines a high fuel efficiency.
Fig. 6.8.1 shows the principal lay-out of the heat pump.

At first, the return water from the district heating system is heated in a parallel system by a low temperature exhaust gas boiler and by the oil coolers and subcoolers for the liquid refrigerant. Then the condenser supplies the main heating of the district heating water. Finally, the engine cooling water and primary exhaust gas boilers heat the water from the condenser to a supply temperature of about 70 to 80°C. This temperature is sufficient through most of the year.

In the winter, the existing district heating plant increases the supply temperature to about 90°C.

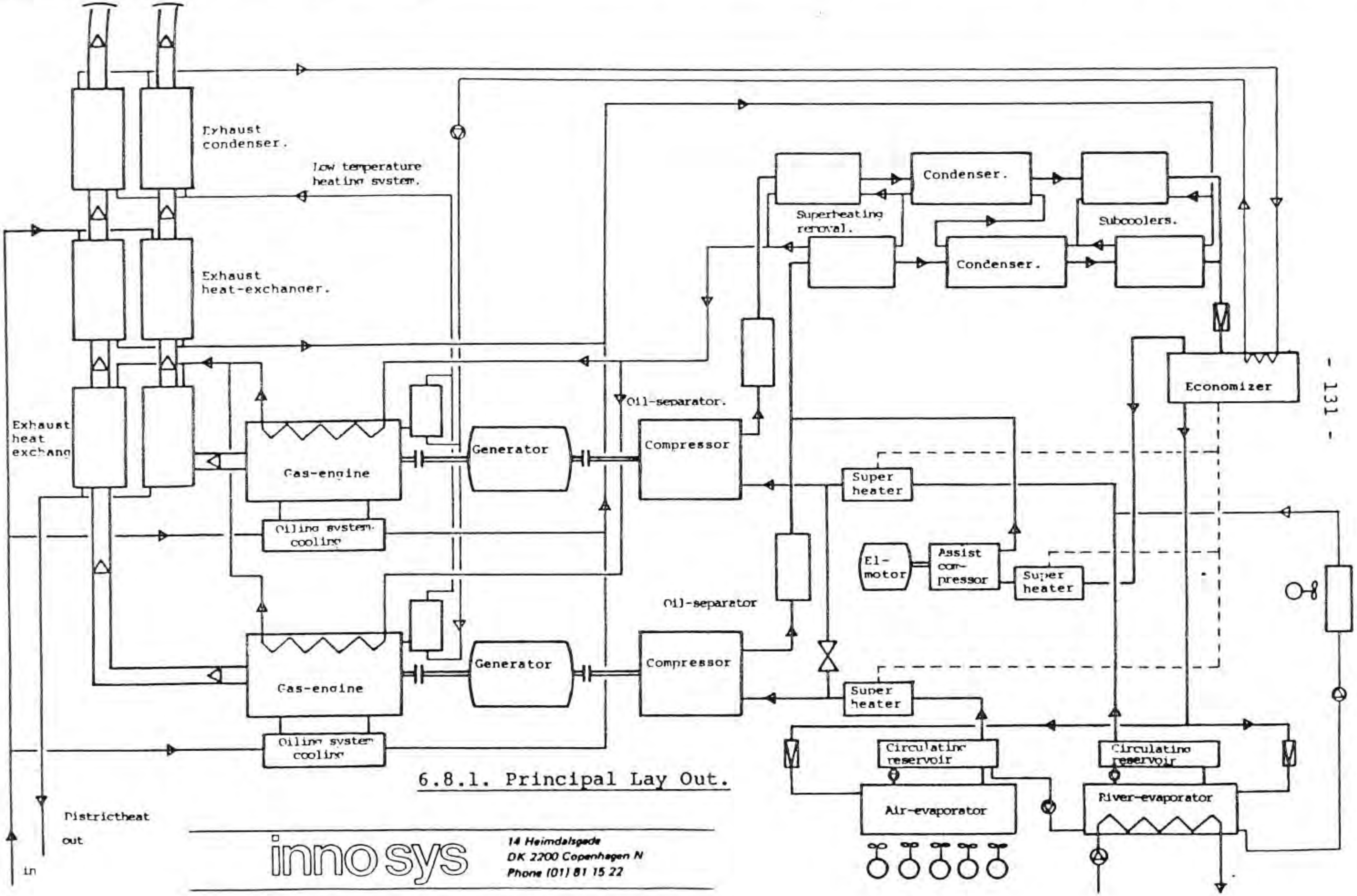
The plant is equipped with two evaporators, one for ambient air and one using river water as the heat source.

Each evaporator can be connected to one or both compressors, depending on the temperature and capacity of the heat source in question. The river is quite small and only carries enough water part of the year.

As an additional feature the heat pump is mounted with an economizer heated by an exhaust condenser.

6.8.4 Calculated Operation Conditions

In the summer, the heat pump supplies the whole load, and as a yearly average 50 per cent of the heat demand is supplied by the heat pump.



6.8.1. Principal Lay Out.

The figures given below represent the conditions when ambient air is used as the heat source.

		Winter	Summer	Yearly average
Evaporator temp.	°C	-10	0	-5
Condensation temp.	°C	60	75	65
Heat source temp.	°C	0	15	5
Supply temp. from HP	°C	65	80	
Return temp. to HP	°C	38	50	
Heat from HP	MW	0.7	0.6	
Total heat from HP system (incl. waste heat)	MW	1.2	0.9	
Total load of heating system	MW	2.7	0.9	
Calculated performance				
Coefficient of performance		3.15	3.40	3.27
Primary energy ratio		1.66	1.75	1.70

6.8.5 Fuel Savings

The design of this plant is rather complex in order to maximize the efficiency of the heat pump.

The expected results with ambient air as heat source are:

An average primary energy ratio of	1.7
Primary energy saving = 1 - 0.85/1.7	50 per cent

Heat production	8.82 GWh per year
Primary energy consumption in heat pump	5.35 GWh per year
Primary energy consumption in a boiler (ER_O = 0.85)	10.38 GWh per year
Primary energy saving	5.0 GWh per year

6.8.6 Investment and Operation

The cost of the plant is DKr 5.3m (1981), of which DKr 0.1m covers connections to the existing district heating central. In addition to this DKr 1.2m is used for research.

The river water evaporator and generator are extra equipment mounted for improving the energy saving ratio.

The river water evaporator saves both fuel and electricity. Electricity is saved by the air evaporator blowers because of a lower power demand.

The generator saves electricity costs both in the heat pump plant and at the existing heating central.

Investments 1981:

Standard heat pump unit	DKr 4.1m
Connections to heat central	DKr 0.1m
River water evaporator + piping	DKr 0.7m
Generator	DKr 0.4m
Total	DKr 5.3m
Research	DKr 1.2m
Operation costs	DKr 0.25m per year
Operation costs/investment	4.6 per cent

6.8.7 Economy

The cost savings are calculated both for this plant and for a heat pump built as a standard unit with generators and river water evaporator.

Heat pump with generator and river water evaporator:

Investment	DKr 5.3m
Primary energy saving (standard unit)	DKr 5.0 GWh per year
Energy saving of river water evaporator	DKr 0.7 GWh per year
Fuel price	DKr 200 per MWh
Fuel cost savings	DKr 1.14m per year
Saving by use of generator	DKr 0.08m per year
Operation cost	DKr 0.25m per year
Total cost savings	DKr 0.97m per year
Simple pay back time	5.5 years
Investment per saved ton of oil per year	DKr 10,400 per toe per year

Standard heat pump unit:

Investment	DKr 4.1m
Primary energy saving	DKr 5.0 GWh per year
Fuel price	DKr 200 per MWh
Fuel cost savings	DKr 1.0m per year
Operation cost	DKr 0.25m per year
Total cost savings	DKr 0.75m per year
Simple pay back time	5.5 years
Investment per saved ton of oil per year	DKr 9,150 per toe per year

6.8.8 Experience during construction of the plant

Some technical changes were made in the last stage of the design phase for the heat pump. Figure 6.8.2 shows the final design.

The main changes were the building of a receiver for liquid freon and modification of the economizer system with an assistant compressor build into the two main compressors.

The economizers require four expansion valves between the condenser and the evaporator.

To keep the freon liquid flowing, the pressure must be ajusted very carefully so that the driving pressure always is sufficient to the regulation of the expansion valves. The economizer compressors suction should also keep up a sufficient pressure in the economizer.

In the design of a heat pump with air evaporator, special care must be taken towards defrosting. The evaporator in Ejby is in 6 sections, and one section is defrosted with warm freon gas while the heat pump is operated on the remaining 5 sections. The warm gas is taken from the economizer system.

The electric generators in the plant are still not connected, because the existing transformer is too small.

The use of river water has been postponed, because a new regulation for the use of lakes and rivers has been put into force during the project . Ground water is investigated instead of river water.

A revised budget in 1983/84 prices is as follows.

Hardware cost.	5.6 mio D.kr
Projecting.	1.4 mio D.kr
Research and development	2.6 mio D.kr
	9.6 mio D.kr

The hardware and projecting cost of 7.0 mio D.kr is more than the budget in 1981 prices and slightly more than the general inflation for the periode.

Unfortunately the energi prices is about the same level as in 1981, so the pay back time will increase from 5.5 years to 7.3 years.

6.8.9 Experience from test and measurement

During the test running some of the components caused problems:

The capacity of the expansion valve to the evaporator was too small, so when both engines are in operation the evaporator dry out of freon.

The pump for liquid freon and oil return was running with heavy cavitation and has been changed.

The cooling water heat exchangers on the engines are slightly too small, which caused engine overheating in summer, when the engine heats up the district water to 8o^{o}C.

Several problems arose during the test running.

Each compressor has 16 cylindres of which up to 4 can be used for the economizers.
The pressure difference to the expansion valves has a very little margin, so during starting up with cold refrigerant, there is no flash gas in the economizers and the plant is out of balance.

A by-pass valve in the economizer compressor has now been installed to overcome the balance problem.

During the test and operation some of the installations and minor components broke down.

The heat pump has been out of operation mainly because of several leaks on the freon system:

- leaks in pipe connections both in weldings and inside the condenser between freon and water.

- two leaks in solded joints in the minor expansion valves.

- the cavitation of the freon liquid pump caused fatique fracture on a copper pipe and on 2 bolts in an expansion valve.

On the electrical system there has been problems with alarm functions, which caused engine stops without reason, specially during the starting up procedure. These problems are solved by system changings and by the removal of electricical noise.

Since the opening of the plant in August 1984 till the end of 1984 the heat pump has been in operation for 1500 hours.

During this period, the energy ratio has been about 1.70, all test running inclusive.

All operation results show a very high primary energy ratio and two examples are listed below:

Measurement of one engine in operation at maximum turns.

Date	13/12-15/12-84	20/10-24/10-84
Evaporator temp	-8 °C	0,5 °C
Condensations temp	55 °C	60 °C
Heat source	-1 °C	12 °C
Supply water temp	55 °C	60 °C
return water temp	38 °C	38 °C
Primary energy ratio	1.66	1.76

The primary energy ratio is at least as high as expected, though it is hard to give excact figures because only one engine is running (with corresponding half of the expected evaporator capacity).

6.8.10 Conclusions from the plant

The heat pump shows a very high primary energy ratio obtained by a high complexity, which necessarily caused operation problems in the first test periode.

After half a year of operation the system and component failures are found and most of them restored.

From early spring 1985 the plant is expected to run without any major problems. A ground water evaporator will be finished in autumn 1985, and from then on the plant should work as shown in section 6.8.4 or with an even better primary energy ratio.

The plant, and especially the R&D efforts, showed out to be more expensive than expected.

With the complex design it is necessary to recalculate all operation conditions for every small change in the design to avoid unpleasant surprises.

6.8.11 Key Figures for a 1.2 MW Gas Engine Driven Heat Pump

The test measurement for different periodes with operation of 24 hours per day shows that efficiency and capacity of the heat pump will be at least as high as projected.

The projected yearly efficiency and revised calculations of the yearly delivery is used for the key figures together with regulated investment cost for a plant without generators.

1) Heat output	W	1.2	MW
2) Investment	I	7.0	M.DKr
3) Operation/Investment	OC	4.6	%
4) Energy delivered	ED	9.700	MWh/y
5) Full load operation	-	8.075	h/y
6) Yearly energy saving	ES	5.700	MWh/y
7) -"-	ES	496	toe/y
8) Investment per ES	IES	14.100	Dkr/toe/y
9) Relative energy saving	ES%	50	%

6.9 ELECTRICALLY DRIVEN HEAT PUMP IN KUNGAELV, SWEDEN

6.9.1 Owner and Location

This plant is owned by the municipality of Kungaelv and is situated in the town of Kungaelv.

6.9.2 Heating System

The heating system consists of a heat pump and conventional boilers parallelly connected.

The heating central supplies a group of buildings with heat. The boilers have a performance of 14 MW with a maximum load of 8 MW, and the total heat production is 19.9 GWh per year.

The heat pump is expected to supply 16 GWh of the yearly heat production and is designed for a maximum load of 3 MW. This performance is sufficient until the ambient air temperature drops below 0°C. When the air temperature falls below -6°C the heat pump stops, and the boilers have to take the whole load.

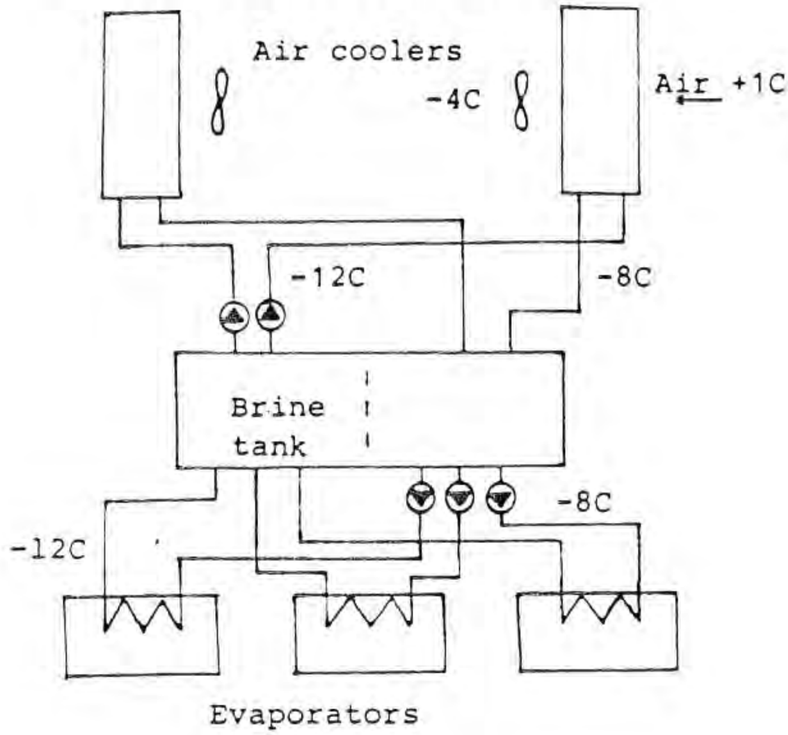
The condenser heats the supply water to between 60 and 70°C, but when the boilers are taken into use, the temperature rises steadily and reaches 100°C at an ambient air temperature of -15°C or below.

The heat pump is planned to be in operation by October 1983.

6.9.3 The Heat Pump System

Three screw compressors are driven by three electric motors, and the heat pump is fitted with economizers.
The heat source is ambient air, and a brine system is used between evaporators and air heat exchangers. See Fig. 6.9.1.

Figure 6.9.1. Brine System



6.9.4 Calculated Operation Conditions

There is a big difference between the evaporation and heat source temperatures. This causes a low evaporation temperature which again causes a drop in the COP value.

		Winter	Summer	Yearly average
Evaporator temp.	°C	-24	± 0	-10
Condensation temp.	°C	max 75	60	65
Heat source temp.	°C	min -6	+18	8.0
Supply temp. from HP	°C	max 70	65	60
Return temp. to HP	°C	55	55	55
Heat from HP	MW	2.6	1	2
Total heat from HP system (incl. waste heat)	MW	2.6	1	2
Total load of heating system	MW	5	1	2.7
Calculated performance				
Coefficient of performance		(~1.5)	(~3)	(~2.5)
Primary energy ratio		-	-	-

6.9.5 Fuel Savings

The figures below compare heat production by electricity with production by oil and are only relevant if electricity is a primary energy source; this is the case in Sweden.

Average COP value	2.5
Heat production	16.6 GWh per year
Electricity consumption	6.4 GWh per year
Primary energy consumption in a boiler ER = 0.85	19.5 GWh per year
Energy saving	13.1 GWh per year

6.9.6 Investment and Operation

Heat pump unit	SKr 5.25m
Building	SKr 2.25m
Electricity transformer	SKr 0.3m
Existing heating central	SKr 0.2m
Other	SKr 1.0m
Total	SKr 9.0m
Expected operation cost	SKr 0.25m per year
Operation costs/investment	2.8 per cent

6.9.7 Economy

An electrically driven heat pump only saves energy if the electricity is produced in a combined heat and power plant or, like in Sweden, mainly by hydro or nuclear power.

Heat production	16.6 GWh per year
Electricity consumption	6.4 GWh per year
Electricity price 1981	SKr 178.5 per MWh
Electricity cost	SKr 1.14m per year
Primary energy consumption in a boiler	19.5 GWh per year
Oil price 1981	SKr 93 per MWh
Fuel cost of a boiler	SKr 1.82m per year
Fuel cost savings	SKr 0.68m per year
Fuel cost savings in per cent	37 per cent
Operation cost	SKr 0.25m per year
Cost savings	SKr 0.43m per year
Simple pay back time	20 years

6.9.11 Key Figures for a 2.6 MW Electric Heat Pump

No results from test or measurements are available as the plant is not operating yet. The key figures are calculated on the basis of the projected values and the investment is regulated to January 1985 prices.

1) Heat output	W	2.6	MW
2) Investment	I	-	M.Skr
3) Operation/Investment	OC	2.8	%
4) Energy delivered	ED	16.600	MWh/y
5) Full load operation	-	6.385	h/y
6) Yearly energy saving	ES	13.100	MWh/y
7) -"-	ES	1.140	toe/y
8) Investment per ES	IES		SKr/toe/y
9) Relative energy saving	ES%	67	%

6.10 DISTRICT HEATING AND ICE RINK COOLING IN RUDDALEN, SWEDEN

6.10.1 Owner and Location

The plant is built in the town of Ruddalen, Sweden and is owned by the recreation administration of Gothenburg.

6.10.2 Heating System

The heat pump is built for ice rink cooling and district heating. The ice rink, which covers 12,000 m^2, was formerly constructed with a normal refrigerating plant. A large part of the cooling is now performed by the heat pump.
When the ambient air temperature drops below 0^oC the heat pump stops and the conventional refrigerating plant starts.

The total heating demand is 25.1 GWh per year, of which the heat pump produces 8.8 GWh per year and the existing heat plant produces the rest.

The heat pump cooling performance is 0.6 MW and is used for about 1000 hours per year; the heat output in this period is 1.2 MW, corresponding to 1.2 GWh per year.

When the ice rink is closed in the warm period the heat pump still delivers heat to the district heating net. This heat performance is 1.7 MW, corresponding to 7.6 GWh per year.

The plant has been in operation since May 1982.

6.10.3 The Heat Pump System

The heat pump compressor is driven by an electric motor of 600 kW. The power required is 2.4 times more than the power consumption of the original refrigerating compressors for the same cooling output but the refrigerating plant does not produce heat.

The heat pump system is shown in Fig. 6.10.1.

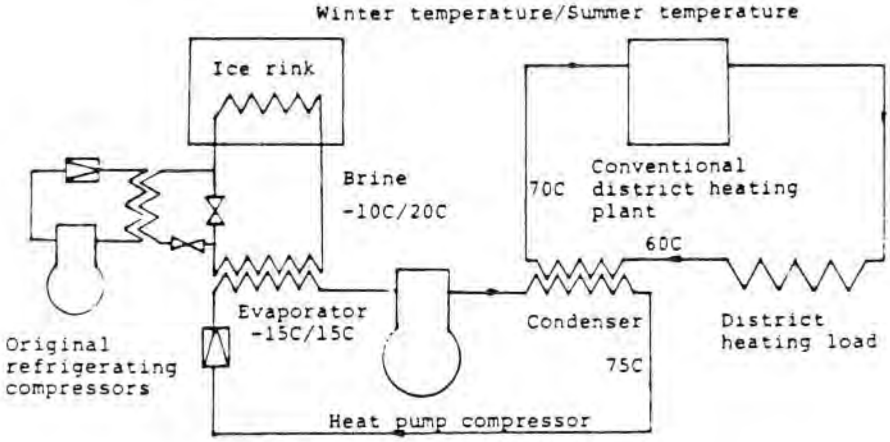


Figure 6.10.1. Cooling and Heating System.

The ice rink is cooled by a brine system with temperatures down to -10°C. During the summer the brine piping system partly works as solar collector and the temperature reaches 20°C.
The brine system is cooled by the evaporator with a difference in temperatures between brine and refrigerant of 5°C.

The condenser heats the district return water about 10°C before it is heated in the conventional district heating plant.

6.10.4 Calculated Operation Conditions

The figures below show the winter conditions at an ambient air temperature of just above 0°C and a summer temperature of about 20°C.
The maximum load of the heating system is 11 MW and the minimum load about 1 MW.

		Winter	Summer	Yearly average
Evaporator temp.	^{o}C	-15	+15	-
Condensation temp.	^{o}C	+75	+75	+75
Heat source temp. (brine)	^{o}C	-10	+20	-
Supply temp. from HP	^{o}C	+70	+70	+70
Return temp. to HP	^{o}C	+60	+60	+60
Heat from HP	MW	1.2	1.7	
Total heat from HP system (incl. waste heat)	MW	1.2	1.7	
Total load of heating system	MW	11	1.7	2.87
Calculated performance				
Coefficient of performance		2.0	2.8	2.67
Primary energy ratio		-	-	-

6.10.5 Fuel Savings

The figures below compare electricity consumption with oil consumption.
The heat pump is compared with a conventional system with separate cooling and heating.

Average COP value of the heat pump (heat production)	2.67
COP value of the original refrigerating plant (cold production)	2.4

Heat production	8.8 GWh per year
Refrigeration	0.6 GWh per year
Electricity consumption	3.3 GWh per year
Oil consumption in a boiler	10.35 GWh per year
Electricity consumption in the original refrigerating plant for the production of 0.6 GWh cooling energy	0.25 GWh per year
Total energy consumption in a conventional system	10.6 GWh per year
Energy savings	7.3 GWh per year
Oil substitution	930 tons per year

6.10.6 Investment and Operation

The fuel consumption in a boiler forms the basis of the investment per fuel saving.

Cost of the heat pump in 1981	SKr 1.5m
Piping and rebuilding the heating system in 1981	SKr 2.0m
Total investment	SKr 3.5m
Operation costs 1981	SKr 0.05m
Operation costs/investment	1.4 per cent
Investment per fuel saving	SKr 3,800 per toe per year

6.10.7 Economy

The energy consumption shown in 6.10.5 is used in this calculation:

Electricity price 1981	SKr	160 per MWh
Oil price 1981	SKr	100 per MWh
Electricity cost of the heat pump	SKr	0.53m per year
Fuel cost in a boiler	SKr	1.04m per year
Electricity cost in a normal cooling machine	SKr	0.04m per year
Energy cost in a conventional system	SKr	1.08m per year
Energy cost savings by using heat pump	SKr	0.55m per year
Energy cost savings in per cent		51 per cent
Simple pay back time		6.4 years.

6.10.11 Key Figures for a 1.4 MW electric Heat Pump

The plant has been operating since 1982, but no test report has been provided for the annex. The key figures are therefore calculated on the basis of projected values and a price regulation of the investment budget.

1) Heat output	W	1.4	MW
2) Investment	I	4.35	M.Skr
3) Operation/Investment	OC	1.4	%
4) Energy delivered	ED	8.800	MWh/y
5) Full load operation	-	6.300	h/y
6) Yearly energy saving	ES	7.300	MWh/y
7) -"-	ES	635	toe/y
8) Investment per ES	IES		SKr/toe/y
9) Relative energy saving	ES%	71	%

6.11 ELECTRICALLY DRIVEN HEAT PUMP IN SALA-HEBY, SWEDEN

6.11.1 Owner and Location

The plant is built between the conventional district heating plant and the local sewage plant in the town of Sala-Heby and owned by the local district heating association.

6.11.2 Heating System

The heat pump is used in connection with a conventional boiler plant to save energy and improve the profitability of the district heating net. A site lay-out is shown in Fig. 6.11.1.

The heating system has a yearly heat production of 120 GWh with maximum output of 40 MW. Of this load the heat pump produces 26 GWh per year with an average performance of 3.2 MW.

The plant has been in operation since June 1981.

6.11.3 The Heat Pump System

The system consists of one heat pump unit with a screw compressor powered by a 1.4 MW electric motor.

Treated water from the sewage plant is used as heat source; however, for part of the production period the flow rate is inadequate, and in such periods water is taken from a nearby river and mixed with the sewage, see Fig. 6.11.2.

The evaporator consists of tubes with inside evaporation and sprinkling water on the outside.
The water is cooled 6°C.

The condenser heats the return water from the district heating system 5 to 10°C, and in the boiler plant the water reaches its final supply temperature.

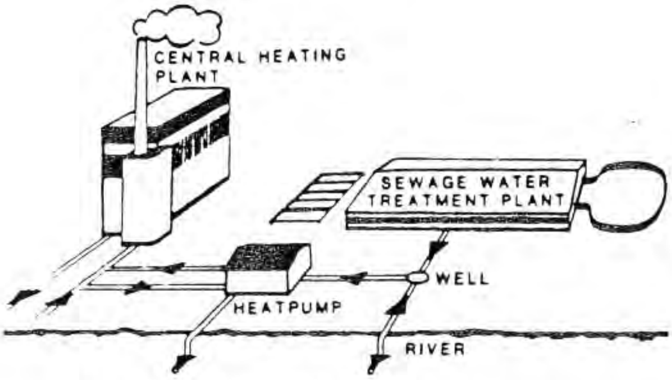


Figure 6.11.1
SITE LAY-OUT

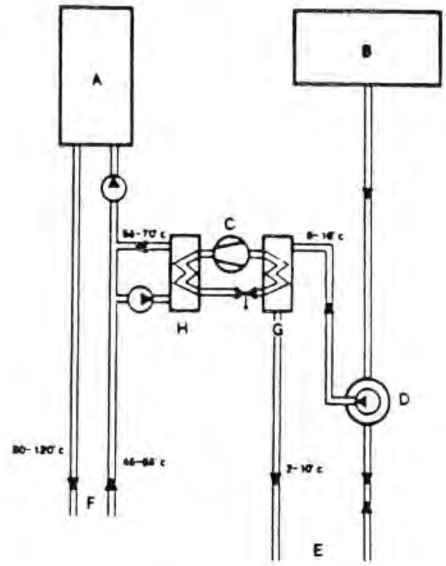


Figure 6.11.2.
FUNDAMENTAL HOOK-UP

A. Boiler plant
B. Treatment plant
C. Heat pump
D. Pump well
E. River
F. Central heating plant

6.11.4 Calculated Operation Conditions

The conditions show that even in the summer the heat pump works to full capacity.

		Winter	Summer	Yearly average
Evaporator temp.	°C	-5	5	2
Condensation temp.	°C	60	75	68
Heat source temp.	°C	7	15	10
Supply temp. from HP	°C	55	70	63
Return temp. to HP	°C	48	65	55
Heat from HP	MW	3.0	3.7	3.2
Total heat from HP system (incl. waste heat)	MW	3.0	3.7	3.2
Total load of heating system	MW	30	4	13.7
Calculated performance				
Coefficient of performance		2.1	2.7	2.6
Primary energy ratio		-	-	-

6.11.5 Fuel Savings

The figures below compare heat productions by electricity and by oil.

Average COP value	2.6	
Heat production	26	GWh per year
Electricity consumption	10	GWh per year
Primary energy consumption in a boiler	30.6	GWh per year
Energy saving	20.6	GWh per year

6.11.6 Investment and Operation

The first two years of operation a test programme is performed by the Swedish Building Research Council.

Investment in heat pump unit 1980	SKr 3 m
Ground work, pump piping etc.	SKr 2 m
Total investment	SKr 5 m
Operation costs	SKr 0.1m
Operation costs/investment	2 per cent

6.11.7 Economy

The enonomy of this plant is quite good, partly because it is working to full capacity all year.
The figures below are based on an expected electricity price.

Investment	SKr 5m
Heat production	26 GWh per year
Electricity consumption	10 GWh per year
Electricity price 1980, approx.	SKr 155 per MWh
Electricity cost	SKr 1.55m per year
Primary energy consumption in a boiler, ER = 0.85	30.6 GWh per year
Oil price 1980, approx.	SKr 88 per MWh
Fuel cost in a boiler	SKr 2.7m per year
Fuel cost savings	SKr 1.15m per year
Fuel cost savings in per cent	42.6 per cent
Operation costs	SKr 0.1m per year
Cost savings	SKr 1.05m per year
Simple pay back time	5 years

6.11.8 Experience during Construction of the Plant

During the galvanizing of the evaporator made of bare horizontal carbon steel pipes, the layer of zinc was unexpectedly thin. Examinations made by Wirsbo Works indicate that the production process in the pipe-works can influence the result of galvanizing. In two pipe-works with different processes the selective oxydation of silicone and manganese was differing. The concentration of especially silicone has a great influence on the zinc layer when galvanizing. Unfortunately all the pipes in the above mentioned evaporator came from the work having a great oxydation of silicone. After some years the evaporator is supposed to be regalvanized.

Five weeks after the starting up of the plant it was decided to improve the system by adding oil-coolers and sub-coolers to the plant. This initiative was expected to rise the heating effect by 4 pct. and the COP factor by 3.5 pct.

6.11.9 Experience from Test and Measurement

The heat production from the plant has been measured every week. In the beginning of 1982 the temperature of the return water from the town was high and the capacity of the plant was regulated. Unexpected vibrations appeared in the automatic controlled slide for capacity control on the screw-compressor and led to stop and losses in the heat production. This problem was finally solved in February 1983 by replacing the screw-compressor with a new one without slide capacity control.

Due to the above mentioned fault problems appeared with operation together with the district heating plant. Instead of capacity controling when less heat was needed, the heat pump was stopped and the district heating central took over the heat production. In February 1983 this problem was solved by installation of a motor controlled suction pressure valve and a condenser by-passing valve system. After this modification the capacity of the heat pump can be reduced from 3 MW to approx. 1.4 MW.

Compared to the slide capacity control on the screw compressor this capacity control system has a negative influence on the COP factor, but the system is more reliable and therefore less stops can be expected.

Ultimo 1982 a systematic measuring fault of approx. 7 pct. was discovered. The test results were adjusted accordingly and the fault was repaired.

By measuring the thickness of the zinc layer on the evaporator after 16 months of operation there was no unexpected decomposition of the zinc layer. Corrosion on the spray galvanized suspension frames for the evaporator has appeared and has to be repaired.

6.11.10 Conclusion from the Plant

In spite of the mentioned problems, the total heat production in 1982 reached 19.5 GWh with a working period of 6,450 hours. The average COP-factor reached 2.5. This seems satisfactory compared to the expected outcome after installation of oil- and sub-coolers and after changing the compressor capacity control system.

The heat production is expected to rise with approx. 20 pct. due to the decreasing amount of major problems and stops. The surplus in 1982 was Skr. 862,000 compared with the alternative heat production. As the total investment is 4.5 mio. Skr., this gives a pay back period of approx. 5 years, which is fairly well.

6.11.11 Key Figures for a 3 MW Electric Heat Pump

The key figures are calculated on the basis of the operation result from 1982, but modified for the expected operation time, which is 20% higher. The investment is regulated to price level 1984/85.

1) Heat output	W	3.0	MW
2) Investment	I		M.Skr
3) Operation/Investment	OC	2	%
4) Energy delivered	ED	23.400	MWh/y
5) Full load operation	-	7.800	h/y
6) yearly energy saving	ES	18.170	MWh/y
7) -"-	ES	1.580	toe/y
8) Investment per ES	IES		SKr/toe/y
9) Relative energy saving	ES%	66	%

6.12 ELECTRICALLY DRIVEN HEAT PUMP USING SEA WATER AS HEAT SOURCE, VISBY, SWEDEN

6.12.1 Owner and Location

This heat pump is built at the Baltic Sea in the town of Visby, Sweden, and is owned by the Swedish State Power Board.

6.12.2 Heating System

The heat pump is used for additional heating in the district heating system.
It is connected to the district heating system between the return and supply pipes at the end of the heating line, whereas the other heating plant is situated in the opposite part of the distirct heating net. The location was chosen in order to get the heat pump near the heat source.

The total heat production is 185 GWh per year, and the heat pump supplies 65 GWh per year of this production.

The maximum output of the conventional boiler is 80 MW and of the heat pump 12 MW. Throughout the summer the heat pump can supply the whole district.

The heat pump heats the return water from 55 to 80^{o}C in the condenser, and the water is pumped back to the boiler plant, from which it is distributed to the consumers.

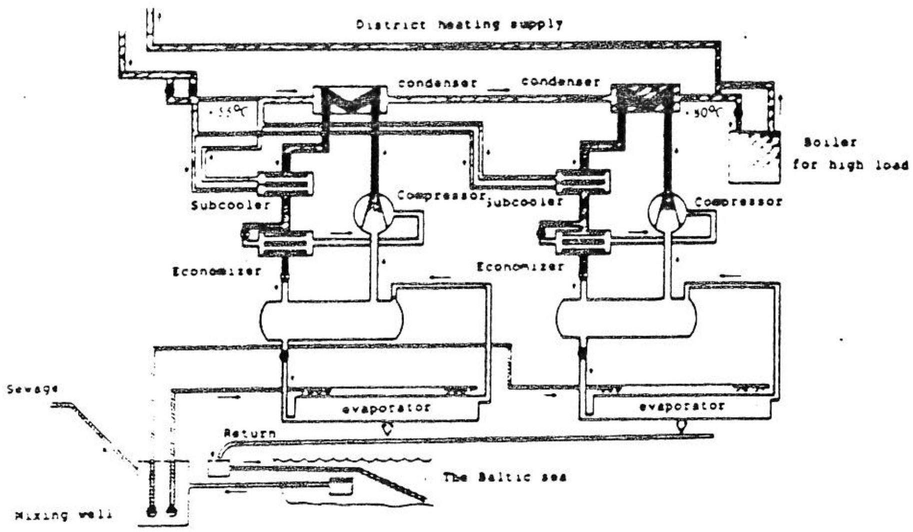
The heat pump plant should be ready for operation by March 1983.

6.12.3 The Heat Pump System

The heat pump has two separate, identical units, see Fig. 6.12.1.

Each screw compressor is driven by an electric motor of 1.8 MW. Besides condenser and evaporator each unit has a subcooler cooled by incoming district heating water and a flash economizer connected to the compressor.

Figure 6.12.1. Principal Lay Out.



The heat source is a mixture of water from the Baltic Sea and about 20 per cent treated water from the local sewage plant. The sea water is taken three metres below sea level and 110 metres from the shore. The temperature of the water mixture is between 2 and 20°C, and the temperature drop above the evaporator is abo 2°C, which gives a water flow rate of one ton per second.

6.12.4 Calculated Operation Conditions

Two figures are given for operation conditions. Which temperature applies depends on whether the year is "cold" or "warm".

		Winter	Summer	Yearly average
Evaporator temp.	°C	-3	8	
Condensation temp.	°C	86	86	
Heat source temp.	°C	+1/+4	+12/+14	
Supply temp. from HP	°C	+80	+80	
Return temp. to HP	°C	+55	+55	
Heat from HP	MW	4.3/8.5	10.3/10.7	
Total heat from HP system (incl. waste heat)	MW	4.3/8.5	10.3/10.7	
Total load of heating system	MW	80	10	
Calculated performance				
Coefficient of performance		2.0/2.5	2.8	2.7
Primary energy ratio		-	-	-

6.12.5 Fuel Savings

The figures below compare heat productions by electricity and by oil.

Average COP value	2.7
Heat production	65 GWh per year
Electricity consumption	24 GWh per year
Primary energy consumption in a boiler	76.5 GWh per year
Energy saving	52.5 GWh per year

6.12.6 Investment and Operation

Investment 1982	SKr 17m
Operation costs	SKr 0.6m
Operation costs/Investment	3.5 per cent

The plant is controlled by computers and supervised via a termina at the conventional heating central.

6.12.7 Economy

The figures are based on an expected electricity price.

Heat production	65 GWh per year
Electricity consumption	24 GWh per year
Electricity price 1982, about	SKr 200 per MWh
Electricity cost	SKr 4.8m per year
Primary energy consumption in a boiler, ER = 0.85	76.5 GWh per year
Oil price 1982	SKr 100 per MWh
Fuel costs	SKr 7.65m per year
Fuel cost savings	SKr 2.85m per year
Fuel cost savings in per cent	37 per cent
Operation costs	SKr 0.6m per year
Total cost savings	SKr 2.25m per year
Simple pay back time	7.5 years

6.12.11 Key Figures for a 8.5 MW Electric Heat Pump

The plant has been taken into operation, but no test report has been provided for the annex. The key figures are calculated on the basis of the projected values and a price regulation of the investment budget.

1) Heat output	W	8.5	MW
2) Investment	I		M.Skr
3) Operation/Investment	OC	3.5	%
4) Energy delivered	ED	65.000	MWh/y
5) Full load operation	-	7.600	h/y
6) Yearly energy saving	ES	52.500	MWH/y
7) -"-	ES	4.565	toe/y
8) Investment per ES	IES		SKr/toe/y
9) Relative energy saving	ES%	69	%

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