Proceedings of the 3rd International Symposium on the

LARGE SCALE APPLICATIONS OF HEAT PUMPS

Oxford, England: 25-27 March, 1987

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Editor: Lorraine Grove

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When citing papers from the volume, the following reference should be used:- Title, Author(s), Paper No., Pages, Proceedings of the 3rd International Symposium on the Large Scale Applications of Heat Pumps, Oxford, England. Organised and Sponsored by BHRA, The Fluid Engineering Centre, Cranfield, Bedford, England. (25-27 March, 1987).

BL British Library Cataloguing in Publication Data

International Symposium on the Large-Scale Applications of Heat Pumps (3rd)

Proceedings of the 3rd International Symposium on the Large-Scale Applications of Heat Pumps.
1. Heat Pumps
I. Title II. Grove, Lorraine III BHRA
621.402'5 TJ62

ISBN 0-947711-24-4 BHRA, The Fluid Engineering Centre, Cranfield, Bedford, MK43 0AJ
ISBN 3-540-17750-7 Springer-Verlag Berlin Heidelberg New York
ISBN 0-387-17750-7 Springer-Verlag New York Heidelberg Berlin

Large Scale Applications of Heat Pumps

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ACKNOWLEDGEMENTS

The valuable assistance of the Organising Committee and Panel of Referees is gratefully acknowledged.

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PAPER A1

EXPERIENCES FROM PERFORMANCE TESTING OF LARGE HEAT PUMPS

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SYNOPSIS

Experience from performance testing of 20 large heat pump plants has demonstrated the importance of such tests and the importance of clear guarantee conditions. The heat pump plants which were evaluated in this investigation consisted of 26 heat pump units, each with a thermal output in the range of 10-30 MW. All plants are connected to municipal district heating systems using treated sewage water, lake water, industrial waste water or geothermal ground water as the heat source.

The tests show that stability criteria for temperatures and flowrates, as required by the test method developed by the National Testing Institute, could be fullfilled in all cases. The typical error of measurement is in the range of 2-4 % and the average deviation of the measured COP from guaranteed values is -4 % for full load operation.

NOMENCLATURE

Roman letters

P_1 = heating power
P_2 = cooling power
P_e = total electric power consumed excluding pumps for heat source and heat sink flows
COP = coefficient of performance = P_1/P_e
COP_c = Carnot efficiency of heat pump cycle
EB = energy balance = $\frac{P_1-P_2-P_e}{P_1}$

t_1 = heating water temperature out from heat pump
t_2 = heating water temperature in to heat pump
t_3 = heat source temperature in to heat pump
t_4 = heat source temperature out from heat pump
q_1 = heating water flowrate
q_2 = heat source flowrate
ρ_1 = heating water density
c_{p1} = heating water specific heat

Additional subscripts

m = measured value
g = guaranteed value
o = operating conditions

INTRODUCTION

Good operating economy and short pay back times have resulted in a large number of heat pumps being installed for base load operation in district heating systems in Sweden. In May 1986 approximately 55 large units were installed or contracted with a total output exceeding 1100 MW. Unitary turn-key installations with outputs up to 30 MW per unit exist and the units can be combined to cover any desired load. The most common heat sources are purified sewage water and lake or sea water but industrial waste water, geothermal ground water, untreated sewage water and outdoor air are also used.

In order to demonstrate the profitability of a heat pump plant accurate calculations have to be performed when the plant is contracted. How profitable the plant will be is decided by the size of the investment, the availability of the installation, the cost of operation and maintenance and the relative energy prices.

The main part of the variable costs consists of the electricity bills for the driving power of the compressor. This is considered in contracts by stating guaranteed values for heat output and coefficient of performance for one or several sets of operating conditions. In most cases bonuses or penalties are stipulated depending on how well the supplier is able to conform to contracted conditions. Thus there are strong economic incentives for both vendor and purchaser to verify the performance of the plant in the most accurate possible way. How strong these incentives can be is illustrated by the following example.

Using the performance data given in table 1 the annual extra energy consumption to produce the yearly guaranteed amount of thermal energy can be estimated.

Annual extra power consumption
Part load: 1554 MWh
Full load: 2069 MWh

In this example it has been assumed that the heat pump has operated under part load conditions during four summer months and under full load conditions during the rest of the year.

Using current Swedish high voltage rates

Held at St. Catherine's College Oxford, England. Organised and sponsored by BHRA, The Fluid Engineering Centre, Cranfield, Bedfordshire, MK43 0AJ England.

for electricity (0,183 SEK/kWh in May-September, 0.221 SEK/kWh the rest of the year and 305 SEK/installed kW including all taxes and other charges) the following annual extra cost can be determined.

Annual extra cost
Part load: 313 kSEK
Full load: 503 kSEK
Power rates: 244 kSEK
Total: 1060 kSEK
(Approx. £ 103000)

This example is just intended to give an idea of the changes in the operating costs that can be caused by seemingly minor deviations from guaranteed values. Of course these deviations may work both ways as will be shown later in this paper.

Despite the moderate discrepancy between measured and guaranteed coefficients of performance in this example operating costs are increased by more than 1 million SEK per annum. A fairly large part of this increase is due to the power rates for the extra power consumption (305 SEK/kW). Using a real interest rate of 5 % and a pay back time of 5 years the present value of this extra cost will amount to 4.7 million SEK. Thus a performance test can carry a fairly sizeable cost.

For this type of commercially contracted heat pump installation the National Testing Institute of Sweden has designed a test method and used it on more than 20 large heat pump plants. The test method describes stability criteria and accuracy of measurement.

DESCRIPTION OF THE TESTED HEAT PUMPS

The units considered for this particular evaluation all operate on municipal district heating systems. Of the 20 plants 13 utilize purified sewage water as heat source, 3 operate on lake water, 2 use industrial waste water and 2 are supplied with geothermal ground water.

All units are furnished with turbo compressors powered by high voltage (11 kV) electric motors. Due to the high thermal outputs these plants carry fairly sizeable refrigerant charges. Using a rough estimate of 1 ton of refrigerant per MW thermal output a plant of 3 x 13 MW would have at total refrigerant charge of 39 tons.

Sewage water

The principle layout of a heat pump plant using purified sewage water as heat source is shown in figure 1. Sewage water is pumped from the treatment plant to a well and then pumped through the evaporator (2), normally a shell and tube design.

The typical design uses a two stage turbo compressor (3) and a two step expansion process by means of a high pressure (11) and a low pressure (13) control valve. Flash gas is separated in an economizer vessel (12). To achieve a maximum amount of sub cooling the district heating water is separated into two flows through the condensor (6) and sub cooler (7) respectively. These flows are joined again before returning to the district heating system through the delivery flow meter (q1).

In table 2 a list is given of the various sewage water heat pumps giving their noninal output, type of refrigerant and the year they were taken into operation.

Lake water

In figure 2 the layout of a lake water heat pump is illustrated. The principle is very similar to the sewage water design apart from the evaporator. Due to the very low water temperatures during winter when lakes are frozen the evaporators are of a special falling film design. Water is sprayed on the outside of the evaporator making it easy to clean the heat transfer surfaces from algae and ice.

During winter water is taken at a substantial depth through the winter inlet (2). In summertime the warm surface water is utilized by means of another shallow summer inlet (1).

In table 3 a list of lake water heat pumps is given.

Waste water

Two installations using industrial waste water have been tested (see table 4). The principle design is the same as for sewage water, as described in figure 1.

Ground water

Geothermal ground water is special by being highly aggressive, therefore requiring the use of titanium tubes in the evaporator. Otherwise the plant configuration is similar to the sewage water design. Two installations are given in table 5.

TESTED PERFORMANCE DATA

In most cases contracts prescribe guaranteed performance data for several sets of operating conditions. This is to enable tests to be carried out with varying temperatures and flowrates for the heat source and heat sink since these parameters normally can not be controlled. Furthermore many installations are designed to operate with reduced output and therefore part load performance is often guaranteed.

In order to give an idea of the performance to be expected from these types of large heat pump installations, measured values for heat output and coefficient of performance are given in tables 6-8. Since guaranteed operating conditions are different for every individual contract the figures given in the tables can not be compared with each other. The incoming heat source temperature (t_3) and out going heating water temperature (t_1) are therefore given to designate the operating conditions as (t_3/t_1). Only one performance point is given for each heat pump unit. Note that COP values are fairly high in spite of very high forward feed temperatures. Typical COP values range between 60-65 % of the Carnot efficiency.

COMPARISON WITH GUARANTEED PERFORMANCE DATA

Guaranteed performance data can be given in the form of diagrams but is generally stated for one or several sets of fixed operating conditions. This requires a correction to be made either to the test results or to the guaranteed data since tests can not generally be carried out at the exact operating conditions of the guarantee. This will be discussed in the section on guarantee problems.

In the present investigation the measured thermal output (P_1), electric power con- sumption (P_e) and coefficient of perfor- mance (COP) have been compared with cor- rected guaranteed values. On the average measured values of P_1 for full load operation exceeded guaranteed values by 4.0 % with a standard deviation of 6,3 %. The maximum differences from guaranteed values of P_1 were +12.9 % and -24.0 % respectively. For part load operation the average difference is +3,8 % with a maximum deviation of +6.4 %. If P_1 on the average has exceeded expectations so has P_e also. For full load operation the average difference is +6.4 % with maximum differences of +16.8 % and -17.6 % respectively. Corresponding figures for part load operation are +13.3 % mean difference and a highest value of +25.0 %.

These discrepancies for P_1 and P_e result in COP values being 4 % lower than expected on the average for full load operation. The standard deviation is 5.5 % with high and low values of +26.7 % and -10.5 % respectively. The distribution of deviations for COP values is illustrated in figure 3 for full load operation. It can be seen in the diagram that the bulk of the tests lie between +2 % and -6 % deviation between actually measured and guaranteed COP calues.

Sometimes only the cooling power (the "free" energy) is guaranteed and then it might be argued that COP values are unimportant. This however is only true when the heat pump operates as base load with electric heating to cover the peak. Otherwise the extra electric energy consumption and particularly the extra power tariff will prove costly.

In figure 4 the deviations from guaranteed values of P_1 and COP are compared with calculated values of the error of measurement. This diagram will show the occurence of penalties or bonuses being paid. Deviations smaller than the error of measurement are normally not considered which further emphasizes the importance of accuracy of measurement.

A common figure for the size of a bonus or penalty is 0.25 % of the total contract sum for each % deviation of P_1 and P_e. For a typical installation this would amount to something like 40000 SEK (approximately £ 4000) for each % deviation resulting in 160000 SEK (£16000) for a typical deviation of 4 %. This is excluding the error of measurement, making each % of this error equally costly. Therefore an improvement of the error of measurement by 1 % is worth 40000 SEK. This also makes the estimation of measuring errors extremely important.

DISCUSSION OF GUARANTEES

In many cases contracts guarantee that the heat output (P_1) should not be lower than a certain value and that the electric power consumption (P_e) should not exceed a certain value. Sometimes this will prove disadvantageous for the heat pump manufacturer. If the heat pump has a higher output than guaranteed the power consumption may exceed guaranteed levels and even though COP values may be better than promised a fine will have to be paid. This example shows the importance of how a contract is laid out.

The recommended practice would be to guarantee heat output and COP. However particular problems arise in the case of part load operation. Two separate situations occur.

In the first case part load operation is induced by a reduction of the heat demand. This being the case part load performance should be defined in terms of achieving a guaranteed COP value for a defined percentage of the full load output for the same set of operating conditions.

The second case however is more complicated. This occurs when the heat pump output has to be reduced due to design limitations in terms of excessive motor currents, extreme heat source or heat sink temperatures etc. Then the guaranteed operating range of the heat pump in terms of design limitations may be far more important than any possible deviations in efficiency. For instance in the case of a lake water heat pump the economic losses from not being able to extract the full amount of energy from the water due to a low lake water temperature is of much greater importance than COP values. In this case premature operation of protection devices should be checked carefully. Even fractions of degrees are important. Selection of the full capacity rating points should be chosen carefully to avoid close proximity to any of the heat pump operating limits. It is generally very difficult to get meaningful results when different types of protection devices start to affect heat pump performance.

Another important aspect concerns the recalculation of measured data to make them comparable to guaranteed values in spite of deviations in the operating conditions. Normally the manufacturer uses computer programs to calculate data for any set of operating conditions. It has to be stated in advance how large deviations from the design point in temperatures and flowrates that can be tolerated without invalidating the computer program. Prior to the performance test either curves showing how performance data vary with temperature and flowrate or several calculated values for different sets of operating conditions (with only one parameter changed at a time) should be supplied by the manufacturer.

In figure 5 there is a diagram showing the deviations from the design heat source temperature for the present number of tests. The diagram indicates that most of the tests have been within ± 2 K and a large portion within ± 1 K of the heat source temperature of the guarantee conditions. Figure 6 demonstrates the same thing for the heat sink temperature. It is evident that in this case the scatter is much larger. However most temperatures lie within ± 3 K of guaranteed conditions. The mean deviation from guarantee conditions of the heat source temperature is -0.4 K and for the heat sink temperature -1.5 K. Extreme deviations are +3.2 and -3.8 K and +3,4 K and -9.8 K respectively.

Flowrates have differed considerably both for the heat source and the heat sink. These discrepancies normally stem from too little knowledge of the heat source and the district heating network by the purchaser. Heat source flowrates have varied between a maximum of +98 % and a minimum of -18 % from design values with a mean value of +12.5 %. Similarly heat sink flowrates have varied between +108 % and -19 % with a mean deviation of +9.4 % from design conditions.

As discussed in the previous section accuracy of measurement is of great importance. The principle for calculating errors and the errors ascribed to installation deficiencies and calibration should preferably be decided and agreed upon in advance.

MEASURING PROBLEMS

Stability criteria

Heat pump performance is greatly affected by temperature and flowrate. This can be designated by

$$P_1 = P_1\ (t_1,\ t_3,\ q_1,\ q_2)$$
$$P_e = P_e\ (t_1,\ t_3,\ q_1,\ q_2)$$

Uncertainty in determining the exact operating conditions during a test will result in an additional error, ΔP_{1o}.

$$\Delta P_{1,o} \quad \frac{\partial P_1}{\partial t_1} \bullet \Delta t_1 + \frac{\partial P_1}{\partial t_3} \bullet \Delta t_3 + \frac{\partial P_1}{\partial q_1} \Delta q_1 + \frac{\partial P_1}{\partial q_2} \bullet \Delta q_2$$

It is therefore important not only to have operating conditions close to design values but also to keep them constant. Variations will unavoidably increase Δt and Δq in the expressions above thereby increasing $\Delta P_{1,o}$.

The test method used in these tests require heat source temperatures to be stable within ± 0.5 K, heat sink temperatures within ± 1 K and flowrates to be stable within ± 5 % for a stabilizing period of 1 hour and a measuring period of 0.5 hours. It has been possible to fullfill these requirements on all occasions. The mean variation for t_1 is 0.4 K with a maximum of 1 K. For t_3 the mean variation is 0.1 K with a maximum of 0.5 K. Flowrates are normally very stable with mean variations of 1.2 % and 1.8 % for q_1 and q_2 respectively.

The error due to instability of the operating conditions is typically 0.5 %. In some installations it has reached a high value of 1.5 % although variations have still been within the limits prescribed by the test method.

Accuracy of measurement

A prerequisite for obtaining accurate results is the use of high quality equipment with up to date calibration status. It is important that calibration has been performed for the actual operating conditions of the application. Flowmeters for instance are mostly calibrated using cold water where as the heat output flowmeter operates in the range 50-100 °C. For instruments with several output signals (e.g. frequency, current etc) only the output used in the calibration process should be used.

The above conditions are necessary but unfortunately not sufficient. In most of the tests in this report estimated errors because of installation of measuring equipment deviating from ideal conditions make a major contribution to the total error. This applies to flowmeter installations in particular.

The error due to measuring uncertainties is calculated from

$$P_1 = q_1 \bullet \rho_1 \bullet c_{p1}(t_1 - t_2)$$

Logarithmic differentiation yields

$$\frac{\Delta P_{1,m}}{P_1} = \frac{\Delta q_1}{q_1} + \frac{\Delta \rho_1}{p} + \frac{\Delta c_{p1}}{c_{p1}} + \frac{\Delta (t_1 - t_2)}{(t_1 - t_2)}$$

Accuracy of measurement of the individual parameters will determine the total error of measurement.

Some particular problems are listed below.

- Temperature.
  To stand the high temperature and pressure of the district heating system, thermometer wells have to be used giving fairly slow response. Normally 2-4 PT100 transducers are used in a measuring plane. Thermal stratification has not been experienced as a problem in pipe measurements although this has been difficult to check in practice.

  In some installations flowrates have been very high compared to contracted values giving low temperture differences, thus increasing the relative error. For lake water it may sometimes be difficult to get a representative value due to large gradients around the inlet. The total uncertainty for the temperature difference is normally less than 0.2 K (calibration of instrument better than 0.005 K).

- Flowrate.
  In most installations electromagnetic flowmeters have been used with good results. Unfortunately the heat delivery meter has often been installed with insufficient straight lengths of pipe before and after the meter. A typical installation is shown in figure 7. For this type of installation the uncertainty of the meter performance is the single most important contribution to the total error.

  In one installation electrode leakage in the meter housing led to readings being consistently 10 % low. This was detected through the energy balance check.

  In a number of installations ultrasonic flowmeters have been used for the heat source flowrate with poor results. Deviations have ranged from +7 % to -17 %.

  In one installation problems with interference were experienced when connecting the frequency output of the meter to the test instrument. This caused the heat pump to shut down due to faulty flow wignal.

- Electric power.
  For measurement of electric power 2 or 3-phase measuring instruments are employed. The most difficult problem is to get the power meter connected to the high voltage measuring transformers with short notice. On one occasion the power consumption had to be measured in a substation 1,5 km away from the heat pump plant and therefore transmission losses had to be considered.

  Sometimes commercial kWh-meters have been connected to the same set of measuring transformers loading these enough to derate the transformer accuracy class one step.

Total accuracy

Total accuracy is normally derived by adding the errors quadratically and stated at a confidence level of 95 %

$$\Delta P_1 = \sqrt{\Delta P_{1,m}^2 + \Delta P_{1,o}^2}$$

In figure 8 results for the total error of measurement are shown. The average value is ± 2.8 % with a maximum of ± 4.7 % and a minimum of ± 1.4 %. This is the range of accuracy values to be expected in this type of testing using first class instrumentation.

Whenever possible an energy balance calculation should be performed. The heat output (P_1) should equal the heat input (P_2) plus the electric power input (P_e) minus any losses to the surroundings.

$$EB = \frac{(P_1 - P_2 - P_e)}{P_1} \, 100 \ \%$$

Results from these tests are illustrated in figure 9. It can be seen that in most cases the balance agrees within ± 2 %. All the extreme values are due to inferior quality flowmeters being used on the heat source flow. For tests using the same quality of equipment for both the heat source and heat sink the average balance is -0.3 % with the largest deviation being 3.2 %. Thus if a heat balance shall be of use high quality instrumentation must be used all around.

OPERATING PROBLEMS

Since performance testing for the purpose of guarantee evaluation is carried out during the course of 1 or 2 days there are normally few problems with the operation of the heat pump. Long term experience has also returned annual availability figures in excess of 90 %, sometimes approaching 100 % (VAST, 1986). Some examples of problems which have affected performance testing are given below.

On one lake water installation there was a problem of too low water temperatures due to an unsually mild autumn. This delayed ice coverage causing larger than usual heat losses from the lake. Due to this climatic problem it was impossible to operate the heat pump on full load.

On some sewage water installations there have been problems with leaves blocking screens and with unstable temperatures and flowrates due to flushing of filters. Furthermore tube vibration in the evaporator caused a few tubes to crack requiring these tubes to be plugged in one installation.

Insufficient or unappropriate instrumentation has delayed testing in a few installations. There have also been problems in using existing flowmeters directly on a few occasions due to the control system of the heat pump. If the flowsignal is affected the flow control unit may stop the compressor.

Factors directly affecting performance have been for instance too little refrigerant charge, refrigerant level control in condensors and evaporators, hot gas distribution in condensors and control of hot gas by pass in connection with potential pumping of the compressor. Estimation and control of the actual refrigerant level in the system seems to be a generic problem. Level indicators have degraded due to heat and vibration and since there is no accumulator capacity for refrigerant in the system the charge has to be changed and optimized according to the operating conditions of each individual installation. This makes the function of the monitoring system all the more important.

Factors affecting heat pump performance on a long term basis have been tube vibration in heat exchangers, refrigerant leakage and poor control of cooling of oil and electric motors. In particular problems with refrigerant leakage through seals constitutes a general problem. Also the oil has sometimes been degraded through pollution by refrigerant which has decayed due to high temperatures when it has leakaed through bearings. The degradation of the oil has caused one or two bearing failures.

Finally one compressor failed on starting up due to too tight clearances between impeller and housing and one system lost a sizeable amount of refrigerant when a safety valve opened. This was due to accidentally passing too hot water through the condensor of a unit not in operation while trying to control the condensor temperature of another unit for testing purposes.

It must be emphasized that most problems are very specific and have happened once or twice. In general these types of heat pump plants have proved very reliable. The only generic problem of importance concerns refrigerant leakage.

CONCLUSIONS

This survey of a number of large heat pump installations has demonstrated that on average performance compares well with guaranteed values. It has also been indicated that due to the economic consequences accuracy of measurement is vital in this type of performance testing. In some cases money has been saved in the first cost of measuring equipment and its installation leading to unnecessarily high inaccuracies in the evaluation of the heat pump.

In order to provide unambiguous conditions for evaluation of guaranteed performance data the method of test, the method of error calculation, the data to be supplied prior to testing, choice and calibration status of equipment and last but not least clear definitions of the guaranteed performance should all be stated or referenced in the contract. The relevant guarantees should normally be comprised of heat output and COP for full load operation, COP at a defined percentage output for normal partload operation and the operating range for part load operation due to design limitations.

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TABLE 1. Example of performance test data

	Guaranteed	Measured
Full load		
- P_1, MW	16.6	18.3
P_e, MW	5.0	5.8
COP	3.3	3.1
Part load		
- P_1, MW	8.4	9.4
P_e, MW	2.7	3.6
COP	3.1	2.6

TABLE 2. List of sewage water heat pumps

Nominal output (MW)	Year taken into operation	Type of refrigerant
13.5	1982	R12
3x13	1982	R12
13	1982	R12
13	1983	R12
27	1983	R12
12	1983	R12
14	1984	R12
12	1984	R12
15	1984	R12
7	1984	R12
29	1984	R12
9	1986	R500
2x15	1986	R12

TABLE 3. List of lake water heat pumps

Nominal output (MW)	Year taken into operation	Type of refrigerant
10	1982	R12
3x26	1986	R22
2x20	1986	R500/R22

TABLE 4. List of waste water heat pumps

Nominal output (MW)	Year taken into operation	Type of refrigerant
12	1984	R12
2x14	1984	R22

TABLE 5. List of ground water heat pumps

Nominal output (MW)	Year taken into operation	Type of refrigerant
19	1985	R500
19	1986	R500

TABLE 6. Full load performance of sewage water heat pumps

P_1 (MW)	COP	(t_3°C/t_1°C)
12.9	3.7	(14.7/61.2)
13.3	3.5	(17.8/68.5)
13.5	3.4	(12.9/60.4)
13.5	3.4	(14.9/67.0)
13.3	3.3	(12.1/67.3)
12.0	3.1	(4.1/68.4)
29.5	3.1	(13.6/77.2)
11.6	3.3	(11.1/67.1)
14.0	3.4	(11.2/58.8)
11.3	2.8	(3.6/73.1)
12.4	3.4	(9.2/65.1)
4.3	3.0	(10.7/79.1)
28.2	2.7	(12.1/87.8)
9.3	2.8	(7.5/74.7)
31.4	3.4	(8.3/61.2)

TABLE 7. Full load performance of lake water heat pumps

P_1 (MW)	COP	(t_3°C/t_1°C)
11.0	2.8	(2.4/61.6)
78.6	3.4	(4.1/57.0)

TABLE 8. Full load performance of waste water and ground water heat pumps

P_1 (MW)	COP	(t_3°C/t_1°C)
11.0	3.5	(20.4/64.7)
14.3	3.4	(11.2/62.4)
45.1	3.2	(20.8/70.1)

1. Waste water pump
2. Evaporator
3. Turbo compressor
4. Electric motor
5. Gear box
6. Condensor
7. Sub cooler
8. Shut off valve
9. District heating water pump
10. Shut off valve
11. High pressure control valve
12. Economizer
13. Low pressure control valve

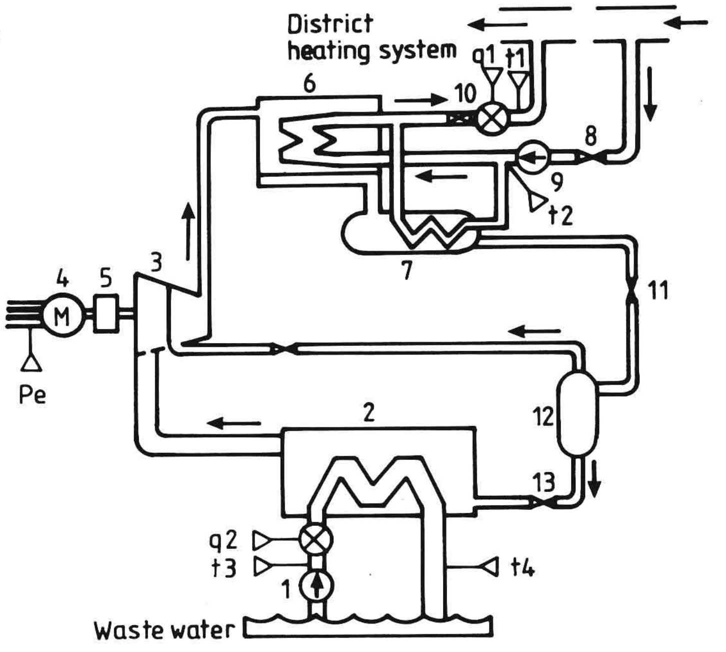

Figure 1. Principle design of a waste water heat pump (ASEA STAL) with the necessary measuring points included.

1. Summer inlet
2. Winter inlet
3. Lake water pump
4. Panel evaporator
5. Turbo compressor
6. Electric motor
7. Gear box
8. Condensor
9. Sub cooler
10. Shut off valve
11. District heating water pump
12. Shut off valve
13. High pressure control valve
14. Economizer
15. Low pressure control valve

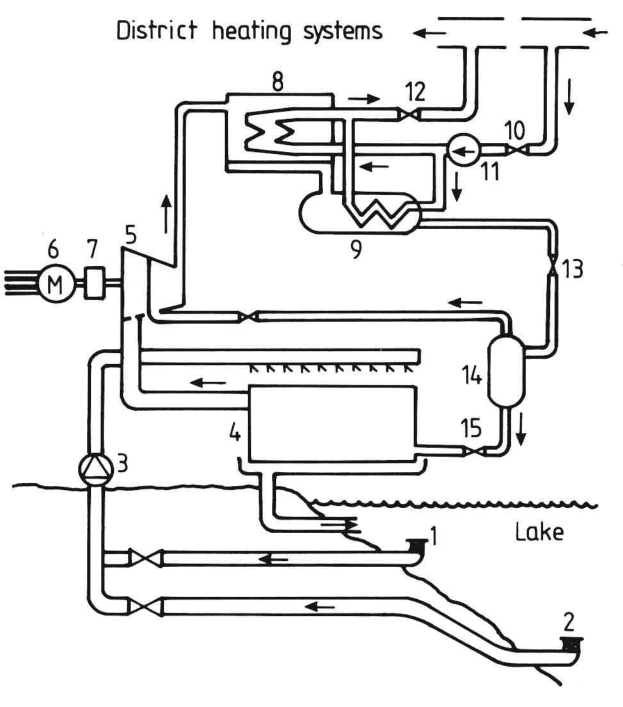

Figure 2. Principle design of a lake water heat pump (ASEA STAL).

DEVIATION BETWEEN MEASURED AND GUARANTEED COP

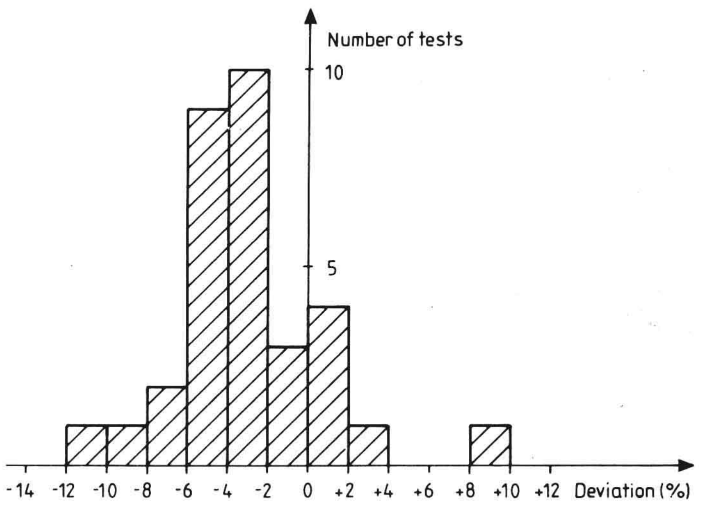

Figure 3. Number of tests with a certain deviation between measured and guaranteed COP in 2 % intervals.

DEVIATION BETWEEN MEASURED AND GUARANTEED HEAT SOURCE TEMPERATURE

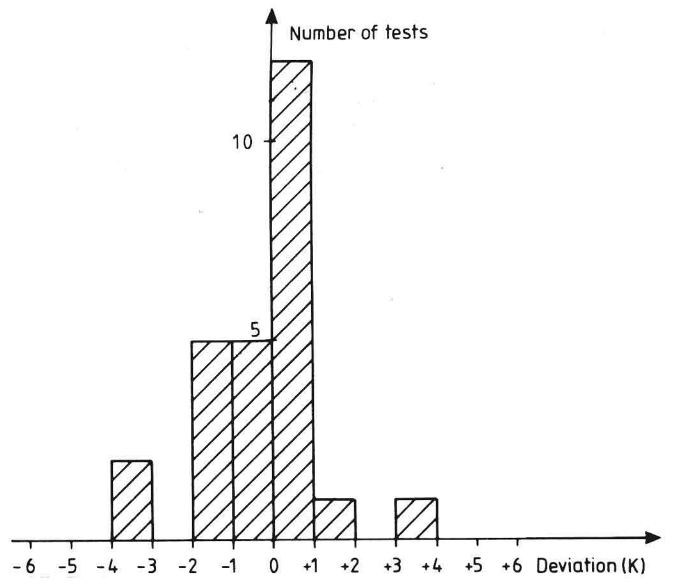

Figure 5. Number of tests with a certain deviation between the measured heat source temperature and the temperature for guaranteed conditions in 1 K intervals.

DEVIATION BETWEEN MEASURED AND GUARANTEED HEAT SINK TEMPERATURE

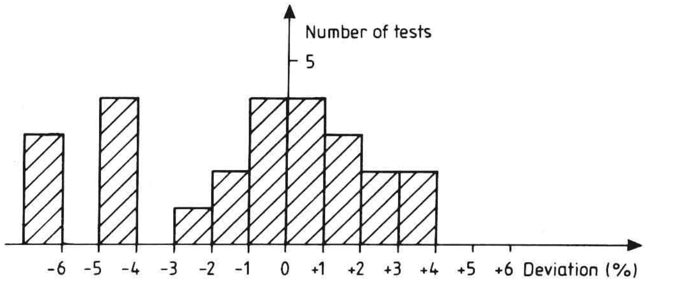

Figure 6. Number of tests with a certain deviation between the measured heat sink temperature and the temperature for guaranteed conditions in 1 K intervals.

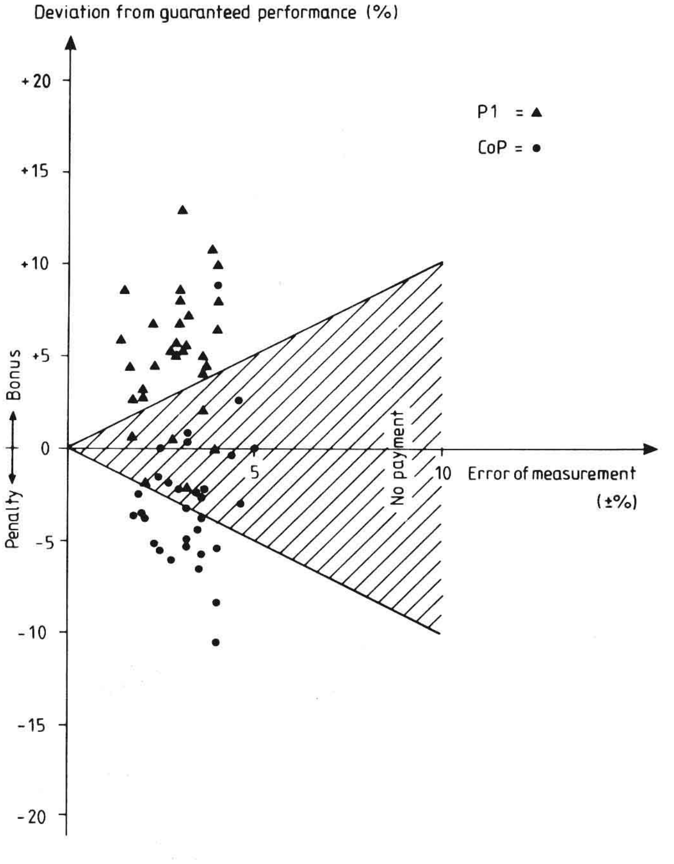

Figure 4. Diagram indicating the influence of error of measurement on payment of penalty or bonus.

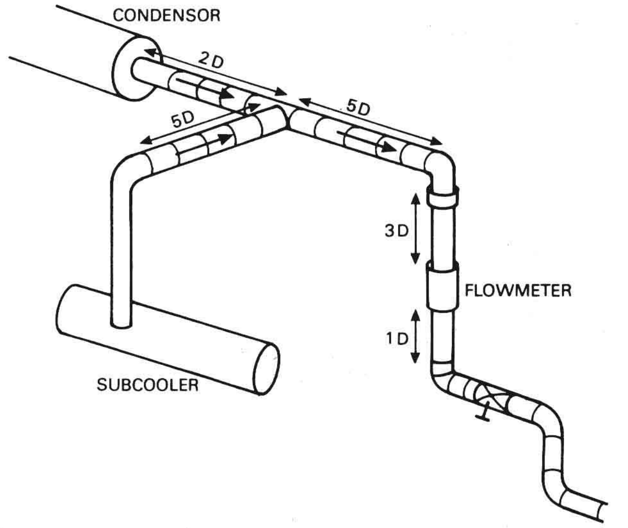

Figure 7. Typical flowmeter installation in a large heat pump unit.