Development of a multi-purpose heat pump for domestic applications
(Paper nr. 257)

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1. Introduction

The development of the heat pump aims at a compact heat pump providing heating and cooling in an environmentally acceptable and energetically less consuming way.
The heat pump fulfils the following functions : heating in winter time and cooling in summer time and producing sanitary hot water during all seasons. Besides the energy savings inherent to the principle of the heat pump, combining these functions reduces also the energy demand. During the cooling season, when the heat pump provides space cooling, the heat pump can simultaneously provide water heating without consuming extra power. When the heat pump is in heating mode, it will provide both space and water heating, with a total power demand equal to that of the heat pump, but it will run longer to meet the combined requirements of space and water heating.
The work has been done for a joint European Project in the framework of the Joule program. The project is called : 'Development of a multipurpose heat pump for domestic applications'. There are three participants involved in this project :
The university KTH (Royal Institute of Technology) in Stockholm, Sweden at the division of Applied Thermodynamics & Refrigeration.
ATECNIC, located in Sintra Portugal, is a manufacturer of heat exchangers for airconditioning equipment. They assemble heat pumps.
The KULeuven in Belgium at the division of Applied Mechanics and Energyconversion. Prof. J. Berghmans of the KULeuven is the co-ordinator of the project.

The objective of the project is to improve the position of European manufacturers of heat pumps on the European air-conditioning market. The market for space heating heat pumps is still small in Europe at this moment. Hot water heat pumps are sold in limited numbers and are usually not combined with space cooling. A disadvantage of domestic heat pumps often stated is that they consume electric energy. Electricity is in most countries a rather expensive form of energy. In some countries however, electricity is distributed at very low prices. This is due to hydro and nuclear electricity production as is for example the case in Sweden and other scandinavian countries. This makes the heat pump a more interesting alternative in those countries.
The market for single room air-conditioners has grown rapidly the last few years and this mainly in southern climates. International experience shows that the heat pump is successful mainly in areas in which a space cooling demand exists. Combined space heating and cooling indeed has a large effect upon heat pump economics. Installing a reversible heat pump (heat pump/air-conditioning) justifies the high investment costs of the heat pump.
Furthermore, the heating load connected with sanitary hot water production increases, relatively speaking, when compared with the space heating load. This is due on the one hand to the increase in hot water consumption and the improved home insulation on the other hand. For these reasons heat pumps of the type described should have a large and increasing potential for application.

It is known that heat pumps through their potential for energy conservation can give rise to considerable reduction in CO_2 production. The heat pump proposed will obviously result in considerable energy conservation as most hot water is produced by direct electric resistance heating in southern Europe.
The assignments in the project were : The study and evaluation of the heat pump. A simulation program is developed to determine the optimal heating and cooling capacities, taking into account the meteorological conditions, the hot water needs, etc... . With this program the most optimal type of heat pump is chosen and the dimensioning parameters are derived from it. An economic evaluation is made taking into account the cost of electricity and other fuels for the different climates.
The heat exchangers are studied thoroughly in relation with improved heat transfer.
After identification of the different components, the heat pump is assembled by ATECNIC. This prototype is tested and evaluated. The aim of the tests was to formulate some improvements for the second prototype.
In this paper, especially the test results and their evaluation will be discussed.

2. Heat Pump Configuration

A scheme of the heat pump configuration is shown in figure nr 1.
R134a is used for as the refrigerant. The compressor is of the scroll and is filled with esteroil to enable to work with R134a.
The heat exchangers are all chosen plate heat exchangers. At the high pressure side there are three heat exchangers. As freon cools down the freon changes from superheated gas to two-phase and subcooled freon. Hence there are three heat exchangers : a desuperheater , a condenser and a subcooler.
The desuperheater gives off his heat energy to the sanitary hot water which is delivered from a vessel by a circulation pump. The temperature and pressure of the freon is measured at the in and outlet of the desuperheater (cf. scheme fig nr. 1). The sanitary hot water temperatures are measured also at the entrance and outlet of the desuperheater. A thermostat is placed inside the vessel for control purposes. A magnetic volumetric flowmeter is placed in the sanitary hot water duct to measure the flowrate.
After the desuperheater, the condenser transmits the condensation energy to either the room space (in heat pump regime) or the outside air in air-conditioning regime. To ensure a swift and easy switch between heat pump regime and air-conditioning regime, a water - water distribution system is used. This secondary coolant is water with 30% ethylene glycol forming a brine with a freezing temperature of -15°C. All the temperatures and pressures of the freon and brine water are measured.
In the middle season (i.e. spring and autumn) there is only need for sanitary hot water and there is practically no space heating demand. In this case the freon in the desuperheater takes up the complete condensation. Hence the desuperheater and the condenser itself are dimensioned equally. The desuperheater and condenser have both a design capacity of 6.6 kW. A plate heat exchanger with 14 plates is chosen : this means 6 channels for the refrigerant and 7 channels for the brine water. This gives a total heat transfer area of 0.76 m².
After the condenser a vessel is placed to buffer the amount of liquid refrigerant. An additional subcooler with a design capacity of 0.6 kW, ensures a sufficient degree of subcooling and gives a better performance of the heat pump. A small plate heat

exchanger is used with 10 plates and a total heat transfer area of 0.12 m². A degree of 10°C of subcooling is attained.
The expansion valve takes care of a superheat of 8 to 9°C.
The evaporator receives its heat from the outside environment in the winter season. In summer however, the evaporator cools down the inside room. The evaporator has a design cooling capacity of 5 kW and has 28 plates with a total heat transfer area of 1.64 m².
As stated earlier, the energy is transferred to the outside environment and the indoor room with an secondary coolant : brine water (30% ethylene glycol). There are two separate circuits : a hot brine circuit and a cold brine circuit. The brine circuits are switched between the indoor fan coil unit and the outside fan by two 4-way valves. (cf. scheme fig nr. 1)
Flowmeters, placed after the 4-way valves give the flowrate of the brine through the inside fancoil unit and the outside fan.
In both the brine circuits, circulation pumps are installed. They are set up at the highest speed, to insure a turbulent flow and hence good heat transfer.
The motor of the inside fancoil unit is adjustable at three speeds but is operated at the lowest speed to suppress noise disturbance inside the room. The room temperature and the temperature of the blown air are measured. A temperature sensor at the fan is used for control purposes.
The outside fan is located in a climate room to be able to simulate all outdoor temperatures. Evidently, room temperature and the temperature of the blown air are also measured.

3. Evaluation of the measurements

On the scheme the different measurement points of temperature and pressure and flow rate are shown. With these measurements the heat capacities and efficiencies are determined. An overview of the different formulae used is given :

The condensation temperature is determined as the saturation temperature of the pressure at the entrance of the condenser. The evaporation temperature is taken as the temperature at the entrance of the evaporator.

$$T_{condensation} = T_{saturation}(P(2))$$

$$T_{evaporation} = T(7)$$

The enthalpies at each point are determined with the use of correlations for vapour , liquid and two phase R134a.
At the entrance of the evaporator the refrigerant is in two-phase and the enthalpy is determined as the enthalpy before the expansion valve :

$$H(7) = H(6)$$

The state condition at the entrance of the condenser can either be vapour, two-phase or liquid. In case of the two-phase state condition the enthalpy is solely determined by the energy balance over the desuperheater.

$$H(2) = H(1) - \frac{Q_{desuperheater}}{M_{freon}}$$

$$Q_{desuperheater} = M_{hot\ water} \cdot C_{p_{hot\ water}} \cdot \Delta T$$

Determination of the refrigerant mass flow : normally the refrigerant mass flow could be deduced by an energy balance over the different heat exchangers. However, during the tests a serious (flow) leakage and hence also heat leakage is detected over the two 4-way valves. Consequently, the energy balances do not give a real insight of the mass flow in the different heat exchangers. The refrigerant mass flow is then determined from the electric power to the compressor. An estimation is made of the heat losses of the compressor to the environment by measuring the temperature at the surface of the compressor and the surrounding temperature. The heat losses are estimated at 3 to 4% of the compressor power.

$$M_{freon} = \frac{W_{compressor} - W_{losses}}{H(10) - H(9)}$$

This results in the heat capacities of the different heat exchangers:

$$Q_{desuperheater} = M_{freon} \cdot (H(1) - H(2))$$

$$Q_{condenser} = M_{freon} \cdot (H(2) - H(3))$$

$$Q_{subcooler} = M_{freon} \cdot (H(5) - H(4))$$

$$Q_{evaporator} = M_{freon} \cdot (H(8) - H(7))$$

The compressor efficiency : two compressor efficiencies can be determined. The isentropic efficiency is the relation between the isentropic enthalpy change and the measured enthalpy change. The isentropic enthalpy change is determined by a correlation for R134a and related to the suction and outlet pressure and the superheat of refrigerant vapour at the suction of the compressor.

$$\eta_{isentropic} = \frac{\Delta H_{isentropic}}{H(10) - H(9)}$$

The volumetric efficiency of the compressor is the relation of the real displaced volume to theoretical swept volume of the compressor.

$$\eta_{volumetric} = \frac{M_{freon} \cdot v(9)}{V_{swept}}$$

Determination of COP : The different COP values are calculated depending on the regime the heat pump works

1) Heat Pump Mode

$$COP_{freon} = \frac{H(1) - H(5)}{H(10) - H(9)}$$

$$COP_{water} = \frac{Q_{heating}}{W_{compressor}}$$

2) Air-conditioning Mode

$$COP_{freon} = \frac{H(8) - H(7)}{H(10) - H(9)}$$

$$COP_{water} = \frac{Q_{cooling}}{W_{compressor}}$$

3) Hot Water Production

$$COP_{freon} = \frac{H(1) - H(2)}{H(10) - H(9)}$$

$$COP_{water} = \frac{Q_{desuperheater}}{W_{compressor}}$$

Evaluation of the heat exchangers : for the different heat exchangers values for the LMTD, UA are calculated and the performance is evaluated.

The LMTD is defined as

$$LMTD = \frac{\Delta T_1 - \Delta T_2}{\ln\left(\frac{\Delta T_1}{\Delta T_2}\right)}$$

The temperature differences for the counterflow plate heat exchangers are :

$$\Delta T_1 = T_{hot,inlet} - T_{cold,outlet}$$

$$\Delta T_2 = T_{hot,outlet} - T_{cold,inlet}$$

Hence, the UA-value of the heat exchangers can be determined from :

$$Q = UA \cdot LMTD$$

The efficiency of the heat exchanger is set as the relation between the exchanged heat capacity and the maximum possible heat exchange. C_{min} is the heat capacity of the fluid with the smallest heat capacity.

$$\eta = \frac{Q}{C_{min} \cdot (T_{hot,inlet} - T_{cold,inlet})}$$

4. Test Conditions

As stated above, the heat pump is tested at different regimes. The following table demonstrates these different regimes :

EXPERIMENTAL CONDITIONS	
Heat Pump, no hot water production	– varying outdoor temperatures 3 °C - 14 °C (winter season)
Heat Pump, only hot water	– varying $T_{HOT\ WATER}$: 20 °C - 65 °C – constant outdoor temperature (10 °C)
Heat Pump, only hot water	– varying $T_{HOT\ WATER}$: 20 °C - 65 °C – constant outdoor temperature (10 °C)
Air-conditioning, no hot water production	– varying outdoor temperatures 22 °C - 36 °C (summer season)
Air-conditioning, only hot water	– $T_{HOT\ WATER}$ constant (20 °C, 35 °C, 45°C) – varying outdoor temperatures 20 °C - 28 °C

Table 1

4.1 Heat Pump, no hot water production

The condensation temperature is almost constant since the condensation temperature in heat pump regime is determined by the room temperature which is fixed at 20° to 22°C. The outside temperature is varied from 3° to 14°C, this gives a variation in the evaporation temperature. In the following figures 2 and 3 the heat capacities and the COP are set against the evaporation temperature.
The different heat capacities have a small ascending trend with increasing evaporation temperature and so does the compressor power (figure2). Hence, there is practically no influence on the COP (figure 3).

4.2 Heat Pump, only hot water production

4.2.1 $T_{OUTSIDE}$ Constant

The outdoor temperature was kept constant at 10°C. The sanitary water was warmed up gradually from 20°C to 65°C. This is the outer limit for producing hot water with the heat pump. The heat capacity of the desuperheater is decreasing with increasing hot water temperature, while the power to the compressor goes sharply up (figure 4). The COP is decreasing sharply from 6 to 2.5 (figure 5).

4.2.2 $T_{HOT\ WATER}$ Constant

The outdoor temperature is varied while the hot water temperature is kept constant. Three different set of tests are carried out with different hot water temperatures. The condensation temperature is nearly constant, depending on the hot water temperature.

$T_{HOT\ WATER}$ [°C]	$T_{CONDENSATATION}$ [°C]
45	48
35	41
20	30

Table 2

In these tests there is no space heating demand because the outdoor temperatures are set as middle season temperatures. However no complete condensation is observed in the desuperheater.
In all tests the heat capacity of the desuperheater and evaporator increases slightly with increasing evaporation temperature (i.e. outdoor temperature). The compressor power however is nearly constant. Consequently the COP increases slightly with increasing evaporation temperature.
Comparing the three sets of tests the COP is higher for lower hot water temperature.

4.3 Air-conditioning, no hot water production

The evaporator temperature is determined by the room temperature while the varying outdoor temperature (22°C - 36°C) results in a varying condensation temperature. The figures 6 and 7 show that the heat capacities remain rather constant with increasing condensation temperature.

4.4 Air-conditioning, only hot water production

The outside temperature and hence the condensation temperature are varied while the hot water temperature is kept constant. Again three sets of tests were carried out varying the hot water temperature.

The hot water temperature has also an influence on the condensation temperature (increasing condensation temperature with increasing hot water temperature). The COP decreases then with increasing condensation temperature and increasing hot water temperature.

5. Component Analysis

5.1 The valves

As stated earlier, the mass of refrigerant is not determined by an energy balance over the different heat exchangers. The reason for this is that the experiments show substantial losses within the two 4-way valves. As the water flow meters are placed after the valves, it is not possible to have a clear idea about the water flow through the heat exchangers. The following plots show how large the heat losses are (figures 8, 9 and 10). These heat losses can not only be explained by heat transfer through insufficient isolated tubes, but there is also an important mixing of the fluids. The valves should satisfy to the following requirements : they should be guaranteed leak proof and the inner core of the valves should be made of thermally isolating material. For obtaining a better user friendliness one 8-way valve will be considered instead of the two 4-way valves.

5.2 The compressor

An analysis is made of the efficiency of the compressor. The tests in heat pump regime, without hot water production are used for this end. The calculation formulae are stated earlier in this paper. Figure 11 shows a volumetric efficiency roundabout 80 to 90%. The isentropic efficiency however is remarkably low : it hardly reaches 50%. This low efficiency is due to the fact that the scroll compressor used in this heat pump is initially not designed for R134a. There are only scroll compressors for R134a available for much larger power capacities. The design for this compressor is based on R22 and is optimised for a fixed pressure rate. As the pressure rates for R22 are much lower than the pressure rates for R134a, the compressor is not working in an optimal working range with R134a. There are two possible solutions : For the next prototype a new compressor will be placed suited for R134a. Current work is done searching for a convenient compressor. Measurements will be carried out using propane as the refrigerant. A preliminary study shows that propane has pressure ratios closer to those of R22. The table shows a comparison of the behaviour of R134a, R290 (propane) and R22. Figure 12 shows an estimation of the isentropic efficiency in relation to the pressure ratio. It shows an optimum for pressure ratios of 2.5 to 3.3. The pressure ratio for R134a ranges from 4.9 to 6.3 for the same condensation and evaporation temperatures. The pressure ratio for R290 is smaller than those for R22 for equal range of condensation and evaporation temperatures.

Pressure	R134a <R290<R22
Pressure ratio	R290<R22<R134a
Mass flow	R290<R134a<R22
Cooling Capacity	
ΔH	R134a<R22<R290
Q	R134a<<R290<R22
$\eta_{isentropic}$	R134a<R290<R22
Compressor Power	R134a<R290<R22

Table 3

5.3 Condenser

For evaluating the performance of the condenser the tests in heat pump regime without hot water production are chosen. This means that the condenser takes up the complete desuperheating of the freon. The condenser can be divided in three parts : a part where the desuperheating takes place, a condensation part and a small area of the condenser is used for subcooling the refrigerant. In figure 13 the heat capacities are plotted as function of the LMTD value and secondly, on figure 14 the UA-value is set as function of the heat capacity. Figure 15 shows the efficiency, defined earlier, as function of the outdoor temperature. As these plots show, the condenser has a very good performance.

5.4 Desuperheater

An identical analysis is made for the desuperheater. It should be mentioned that there was no complete condensation and hence no subcooling part. For the evaluation the tests are taken in heat pump regime with only heat water production ($T_{HOT\ WATER}$ = 45°C). Figure 16 shows the capacity of the desuperheater as function of the LMTD. Figure 17 shows the efficiency related to the condensation temperature. As there is no complete condensation the condensation part show a smaller efficiency than the desuperheating part.

6. Conclusive Remarks

These measurement campaign has delivered some remarks for improvement of the first prototype.

As the distribution system is concerned, the following remarks are made : A 8-way valve is placed in the brine circuits. The tubes are sealed with O-rings to avoid leakage. Furthermore the inner core of the valve is made of plastic to better isolate the circuits.

As second remark, it is observed that the outdoor fan is rather noisy and takes a lot of power. In a next prototype, the outdoor unit can be dimensioned much smaller.

For the heat pump system itself, the current prototype will be tested with propane as to improve the compressor efficiency. A second prototype will be build using a more convenient compressor.
Although the condenser has a very good performance, there was interest from a heat exchanger manufacturer to test a new type of condenser. This type of condenser uses spiral wiring on the tube to enhance the heat transfer. Its performance will also be tested.

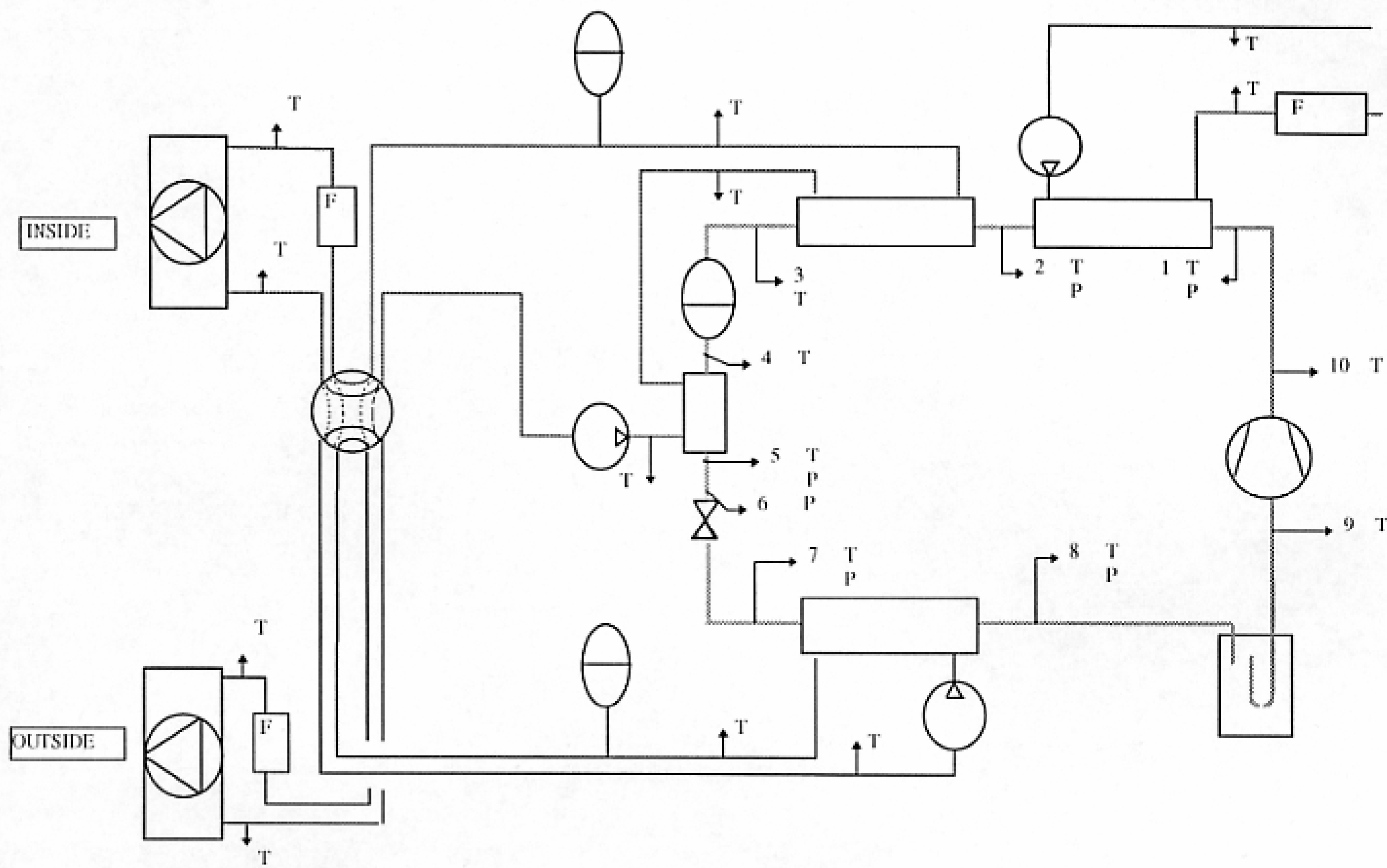


Figure 1

HEAT PUMP, NO HO WATER PRODUCTION

$T_{Condensation} = 57 \pm 3°C$

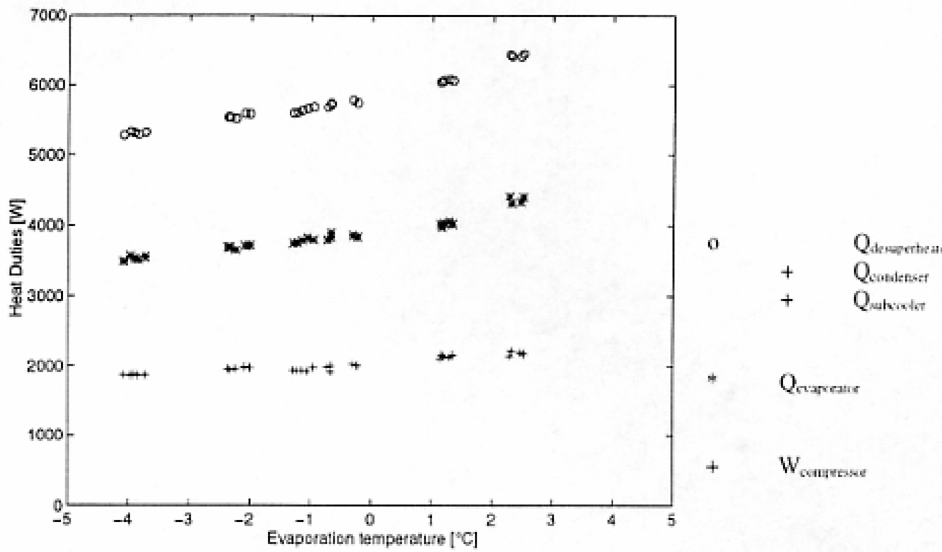


figure 2

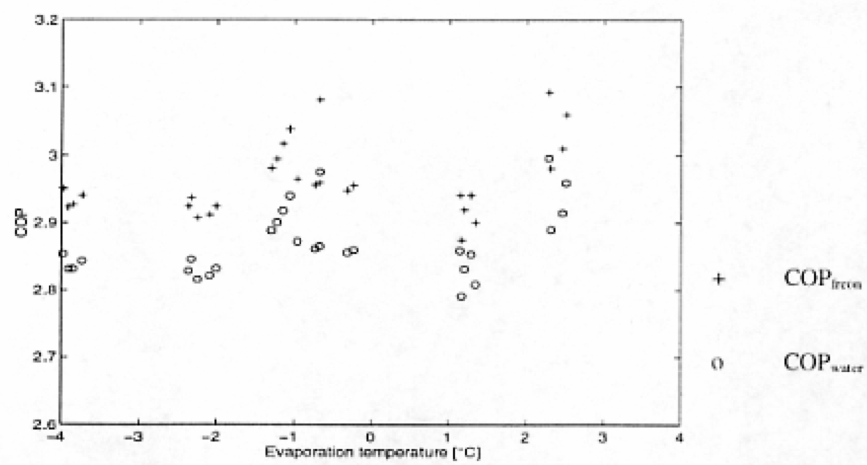


figure 3

HEAT PUMP, ONLY HOT WATER

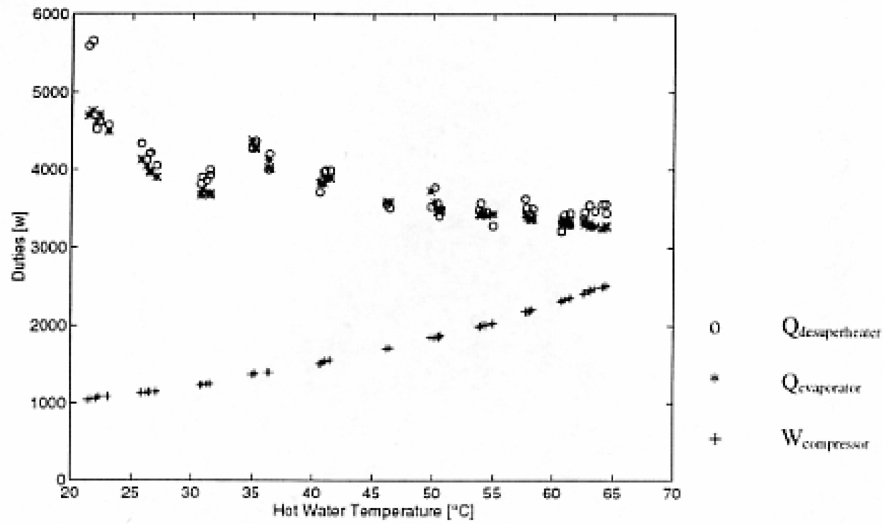


figure 4

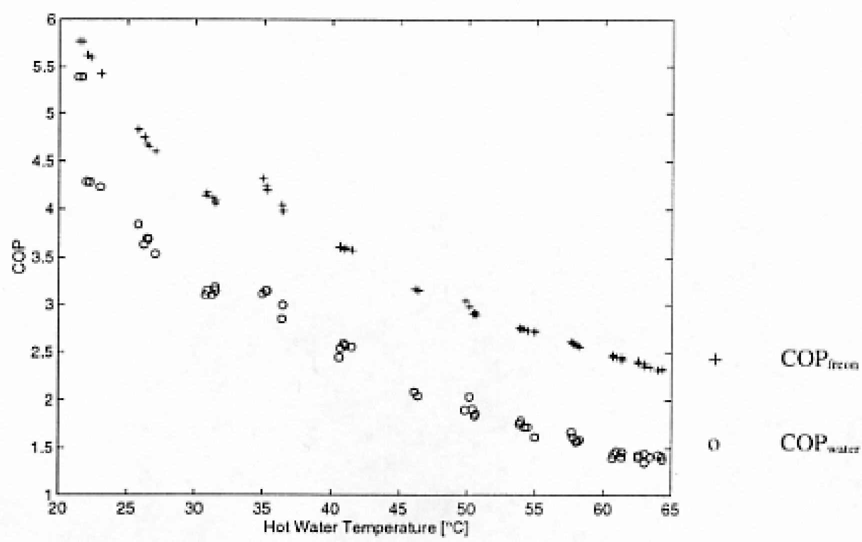


figure 5

AIR-CONDITIONING, NO HOT WATER PRODUCTION

$T_{Evaporation} = 0 \pm 3°C$

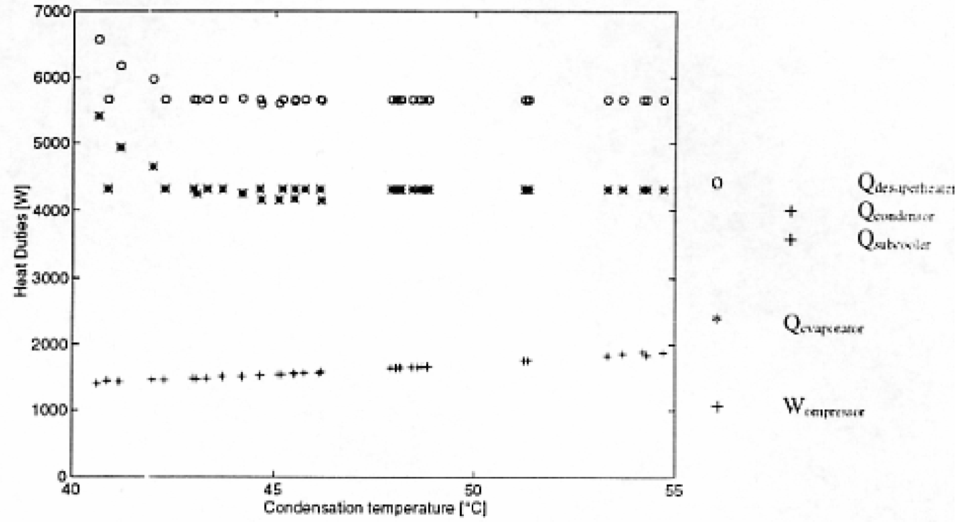


figure 6

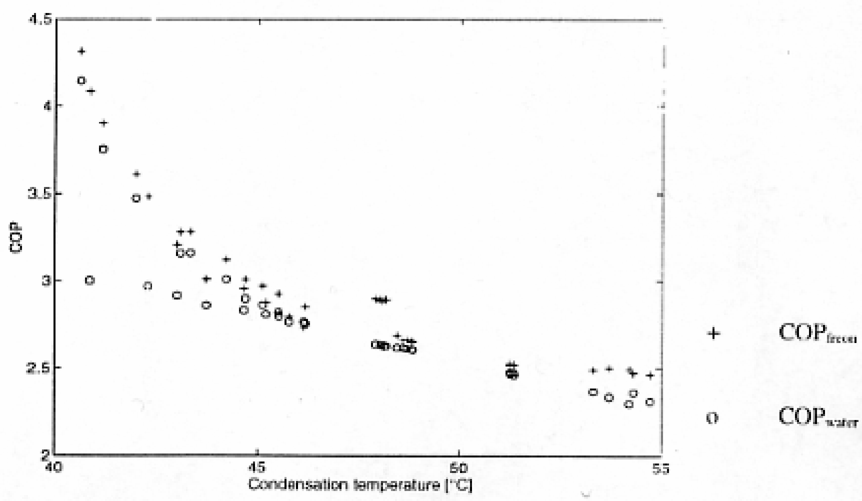


figure 7

HEAT PUMP, NO HOT WATER

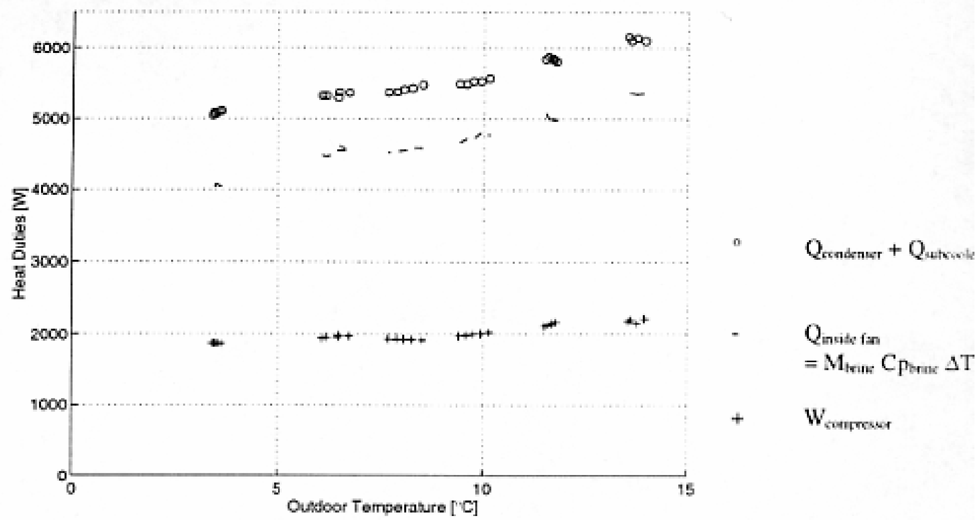


figure 8

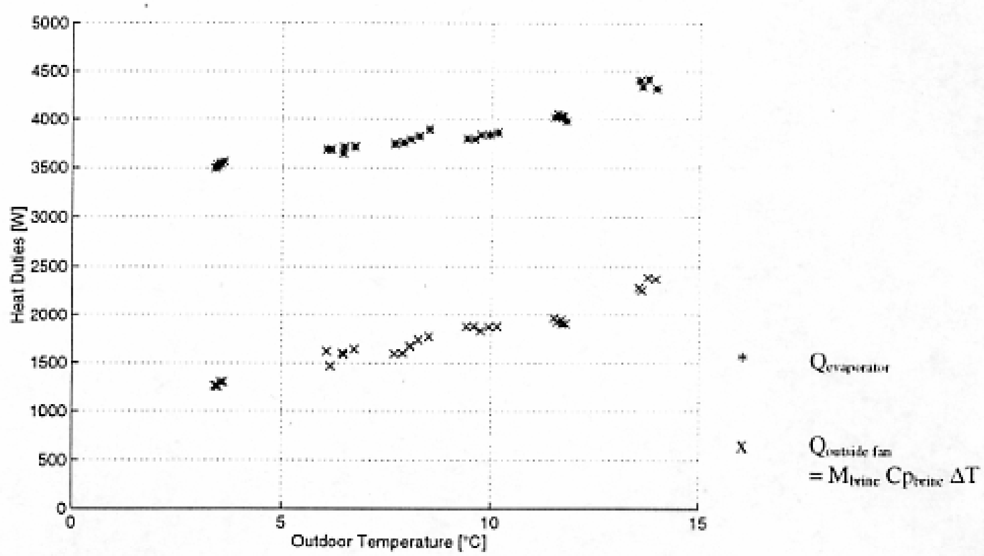
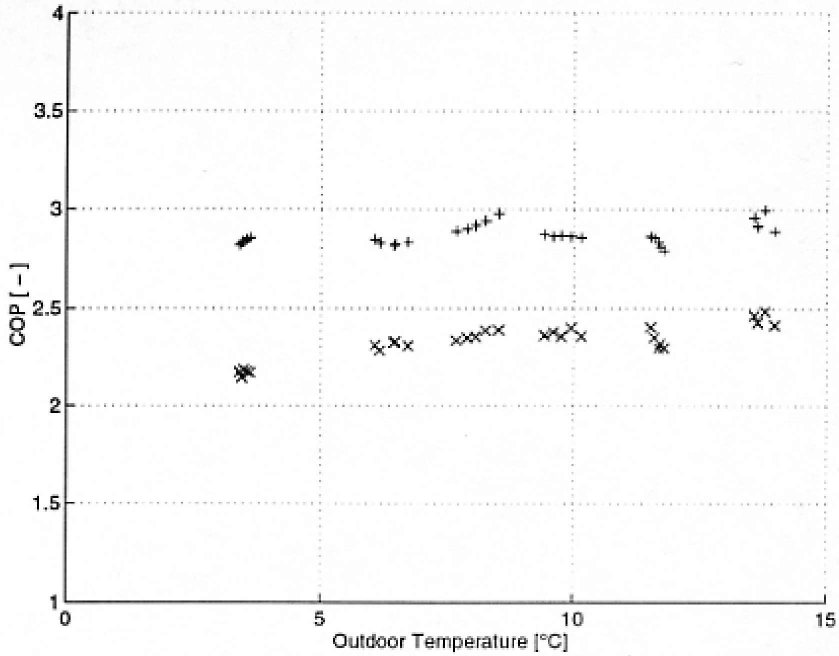


figure 9

HEAT PUMP, NO HOT WATER



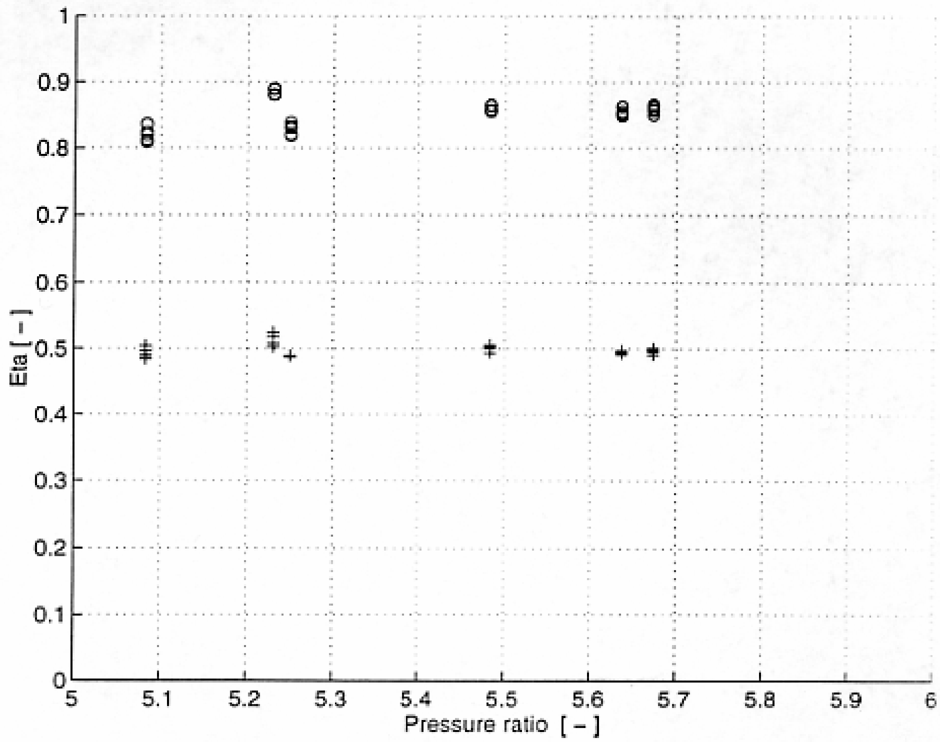
+ $COP = (Q_{condenser} + Q_{subcooler}) / W_{compressor}$

x $COP = Q_{inside\ fan} / W_{compressor}$

figure 10

COMPRESSOR EFFICIENCY

Heat pump, no hot water production



o $\eta_{volumetric}$

+ $\eta_{isentropic}$

figure 11

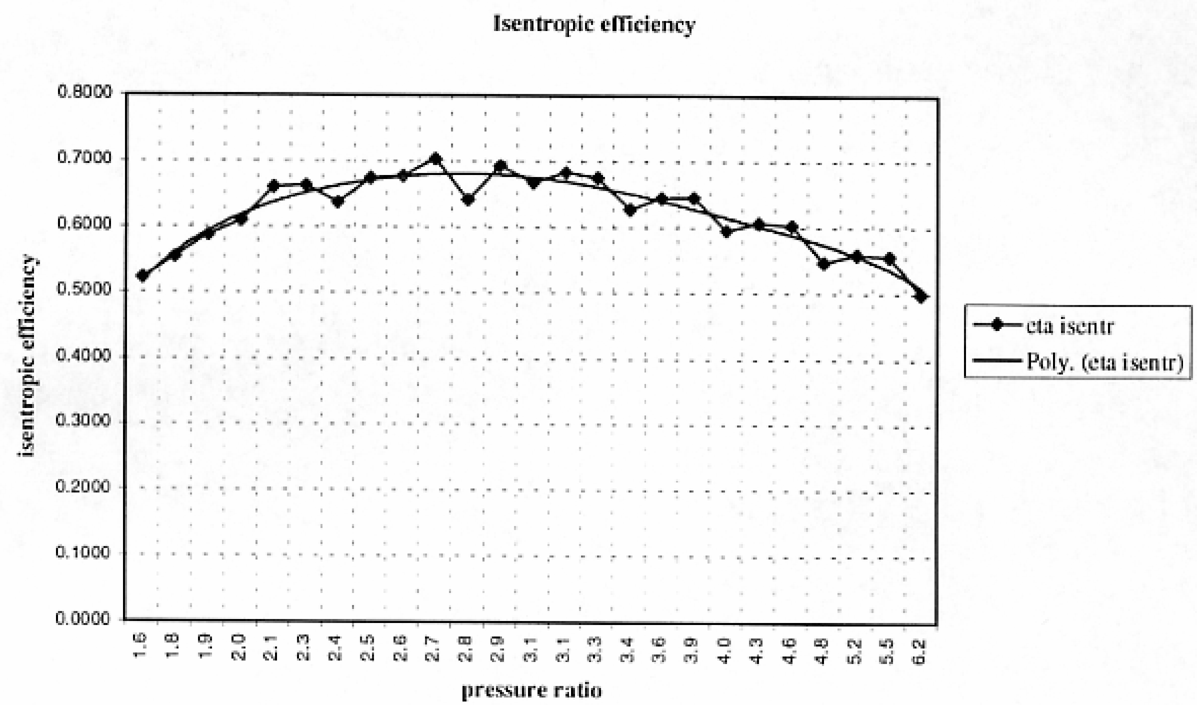
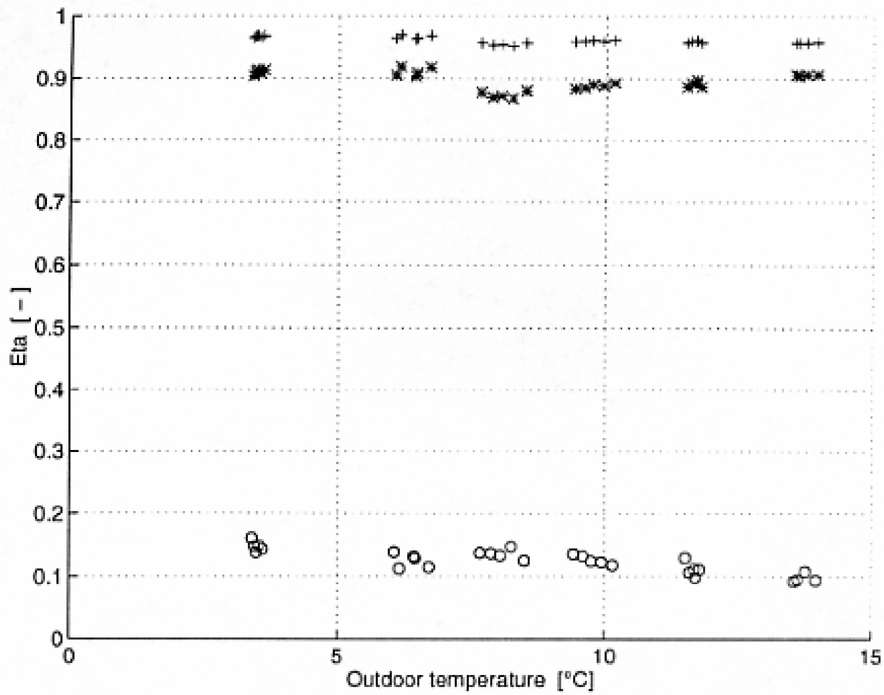

Isentropic efficiency
0.8000
0.7000
0.6000
0.5000
0.4000
0.3000
0.2000
0.1000
0.0000
isentropic efficiency
1.6
1.8
1.9
2.0
2.1
2.3
2.4
2.5
2.6
2.7
2.8
2.9
3.1
3.1
3.3
3.4
3.6
3.9
4.0
4.3
4.6
4.8
5.2
5.5
6.2
pressure ratio
eta isentr
Poly. (eta isentr)

figure 12

CONDENSER



\+ Desuperheating

* Condensatie

° Subcooling

figure 15

CONDENSER

Heat Pump, no hot water production

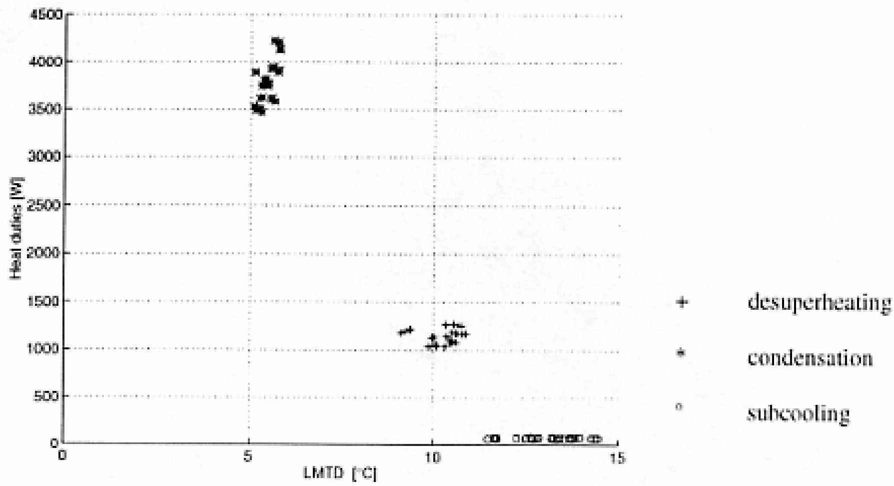


figure 13

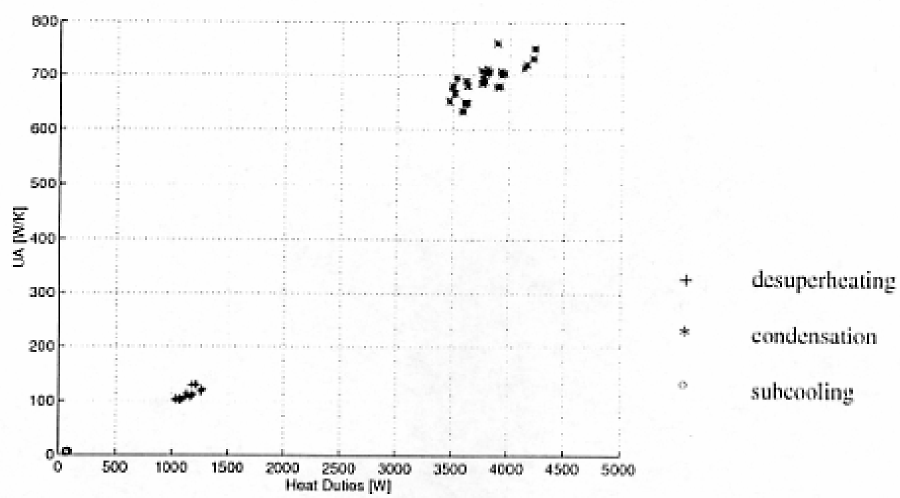


figure 14

DESUPERHEATER

Heat pump, $T_{Hot\ water} = 45°C$

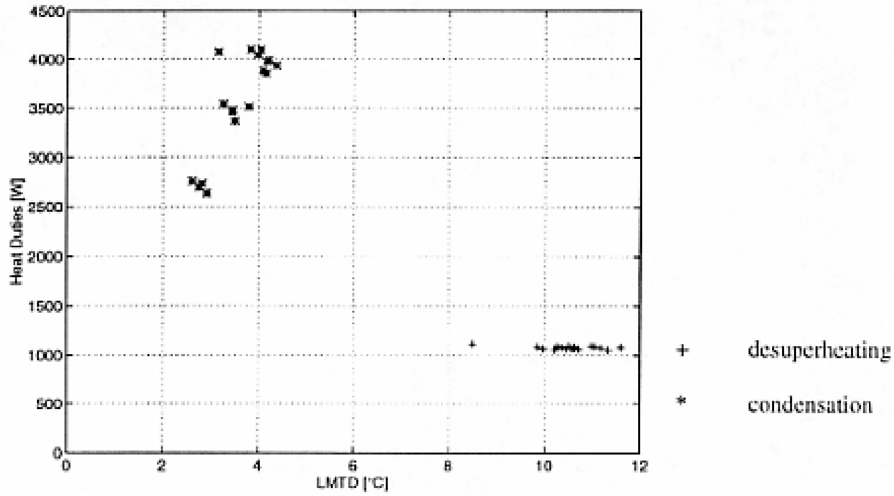


figure 16

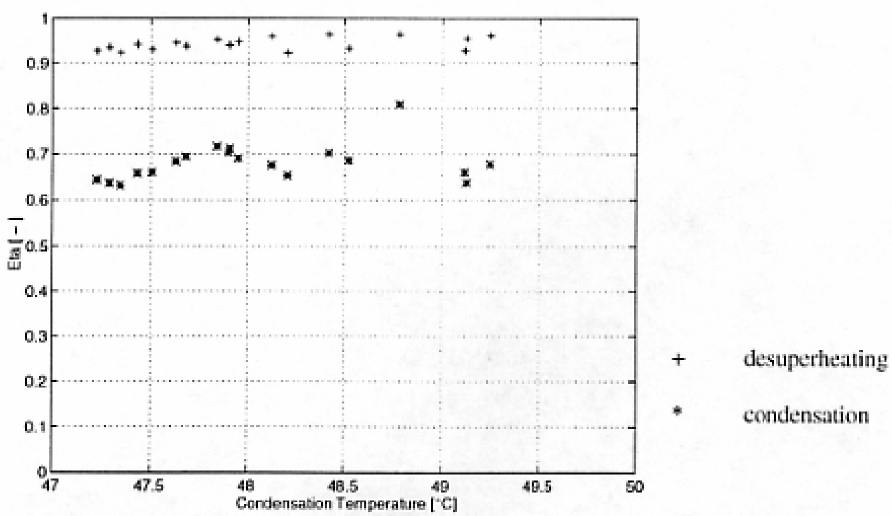


figure 17