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Propane for Heat Pump Applications using Brazed Plate Heat Exchangers

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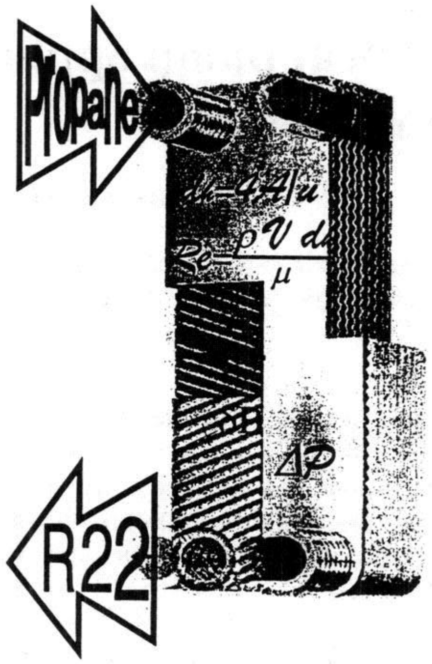

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Stockholm 1995

Project B3.2 -Propane and propane based mixtures in small heat pumps

PROJEKTET INGÅR I NUTEKPROGRAMMET: ERSÄTTNING AV MILJÖFARLIGA MEDIER I KYLANLÄGGNINGAR OCH VÄRMEPUMPAR

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Propane for heat pump applications using brazed plate heat exchangers

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Abstract

Heat transfer and pressure drop has been measured for two brazed plate heat exchangers, one used as condenser and one as evaporator, in a system simulating a small domestic heat pump. The measurements have been done with R22 and propane as refrigerants, with the aim of assessing the relative heat transfer and pressure drop characteristics of these fluids. The nominal heating capacity of the system was about 3 kW. The amount of refrigerant was 500-600g with propane and 1000-1100g with R22. The condenser tests were done at 35°C condensation temperature and the evaporator tests at +5, 0 and -5°C evaporation temperature. The flow conditions on the water/brine sides of the heat exchangers were kept constant through the tests. Any difference in the temperature difference or the overall heat transfer coefficient between the fluids could therefore be related to different heat transfer characteristics of the refrigerants.

The tests indicate that propane in the present type of plate heat exchangers gives slightly lower overall heat transfer coefficients both in condensation and evaporation than R22. However, the tests show that the pressure drop is 40-50% lower with propane both in the condenser and the evaporator. This indicates, that if the heat exchangers were redesigned for use with propane, the overall heat transfer coefficient may be as high as, or higher than with R22.

Nomenclature

k	$[W/(m^2 \cdot K)]$	overall heat transfer coefficient
$\dot{q}$	$[W/m^2]$	heat flux
$\dot{Q}$	[W]	heating or cooling capacity
A	$[m^2]$	heat exchanger surface area
$\vartheta_{\ln}$	[K]	logarithmic mean temperature difference (LMTD)
ϑ_{in}	[K]	temperature difference between the refrigerant saturation temperature and the incoming water or brine.
ϑ_{out}	[K]	temperature difference between the refrigerant saturation temperature and the outgoing water or brine.

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1. Introduction

During the last ten years, several research projects have been connected to the ”phase out” of CFC-type media, so called ”Freons”, used as refrigerants. The reason for this is the depleting effect these substances have on the stratospheric ozone layer.

Because of the concern for the stratospheric ozone, and because of a fear for other unknown environmental effects of man-made chemicals used as refrigerants, interest in Europe has shifted towards the use of naturally occurring substances as working fluids. Especially, different hydrocarbons such as propane and isobutane are being tested as refrigerants. As these substances are highly flammable, it is desirable to use heat exchangers having small internal volume.

Brazed plate heat exchangers is one type fulfilling this requirement, and they have become very popular, in Sweden and elsewhere, as evaporators and condensers in heat pumps and refrigeration systems. This type of heat exchangers allows for a very large heat transfer surface area in a small external volume, and also gives high heat transfer coefficients in boiling and condensation as well as in one-phase flow.

At the Division of Applied Thermodynamics and Refrigeration of Kungliga Tekniska Högskolan, a project is running, aimed at studying the use of propane or propane based mixtures as refrigerants in a small domestic heat pump with brazed plate heat exchangers as condensers and evaporators.

During the first year, an experimental set-up has been designed and built, sensors for flows, pressures and temperatures installed, data-acquisition system connected and a computer program for developing the data has been written. The system has then been used for testing a pair of brazed plate heat exchangers with R22 and with propane. The tests with R22 have been run to establish a base for comparison for results with other fluids.

In this report, heat transfer and pressure drop for the first set of condenser and evaporator are reported.

2. Experimental set-up

Fig. 1 shows the schema of the experimental set-up and the measuring equipment. It simulates a small domestic heat pump equipped with a hermetic compressor with a power of about 900 W and four brazed plate heat exchangers as condenser, subcooler, evaporator and superheater. The refrigerant is throttled by a thermostatic expansion valve. Two water loops are connected to the condenser and the subcooler, and a brine loop is connected to the evaporator and the superheater. The cooling and the heating capacities of the set-up and operation conditions are controlled both manually and electronically.

In the water and brine loops, volume flows, temperatures and pressure drops across the heat exchangers are measured. The same instrumentation is available for the refrigerant side with the additional possibilities to measure the mass flow and the pressures.

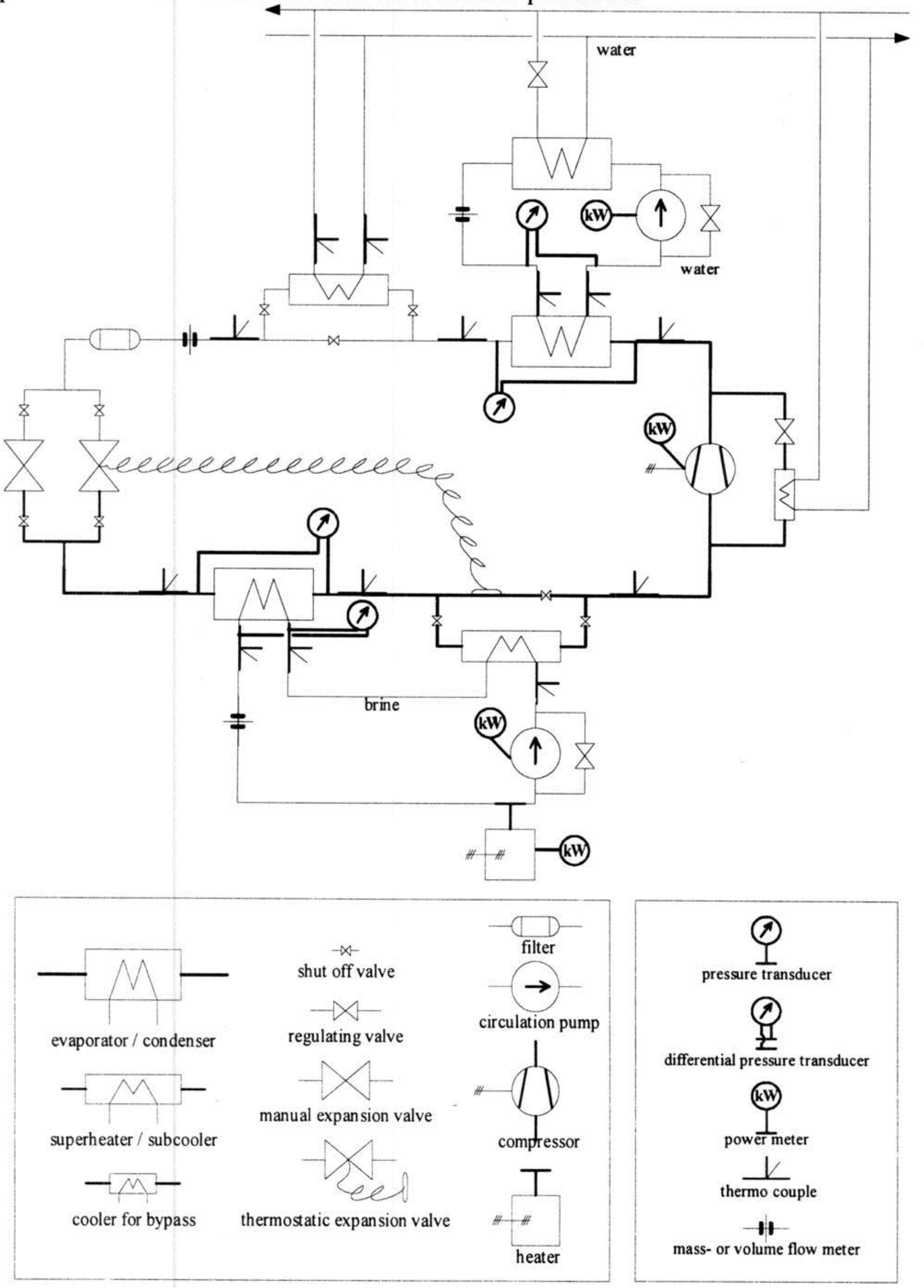

Fig. 1 : Experimental set-up and measuring equipment

3. Measurement

The condenser and the evaporator were operated as they were designed originally to be used with R22. No study has been performed on the subcooler and the superheater since it was not necessary to use them in the present tests.
Both condenser and evaporator have been studied separately to investigate the heat transfer characteristics and the pressure drops.
The first tests were carried out with R22 as refrigerant in the system. A base for comparison was gained through these measurements. The tests were re-run after replacing R22 by propane. The replacement was performed without any major changes of the system. It was sometimes necessary to change the size of the nozzle in the thermostatic expansion valve to achieve stable operation. The setting of the expansion valve was changed to receive a superheat of 5 to 6°C.
The mass flow was in these tests varied through bypassing the compressor, this in order to reduce the heating capacity and the cooling capacity for a given temperature of condensation and evaporation respectively. The bypass was cooled by water through a heat exchanger.
Table 1 and *table 2* show the test series for the condenser and the evaporator.

Table 1 : The test series for the condenser

Refrigerant	Evaporation temperature	Condenser	Condensation temperature
R22	5°C	model C	35°C
Propane	5°C	model C	35°C

Table 2 : The test series for the evaporator

Refrigerant	Condensation temperature	Evaporator	Evaporation temperature
R22	35°C	model D	5°C, 0°C, -5°C
Propane	35°C	model D	5°C, 0°C, -5°C

4. Calculations

A temperature difference between the hot and the cold side of the heat exchanger is necessary for the transfer of heat. To achieve maximum coefficient of performance of the system, it is of interest to keep the differences as small as possible. The temperature difference can be used to compare the heat transfer characteristics of different fluids. In the following, the dependencies of temperature difference on the heating and cooling capacity per square meter of heat exchanger surface are presented.

The temperature difference can be calculated in several ways. For the heat exchange between one phase media the logarithmic mean temperature difference is used, define as :

$$\vartheta_{\ln} = \frac{(\vartheta_{in} - \vartheta_{out})}{\ln(\vartheta_{in} / \vartheta_{out})} \tag{3}$$

where ϑ_{in} and ϑ_{out} are the temperature differences between the fluids at the two ends of the heat exchanger.

For condensers and evaporators, this definition is not quite appropriate since the temperature difference in these cases varies in an unsymmetrical way along the heat exchanger depending on whether the refrigerant is one phase or two phase. Fig. 2 shows that the condenser performs three operations: hot gas cooling, condensation and subcooling. The evaporator performs two operations: evaporation and superheating. Furthermore, the condensation and the evaporation temperatures drop slightly along the heat exchangers because of the pressure drop.

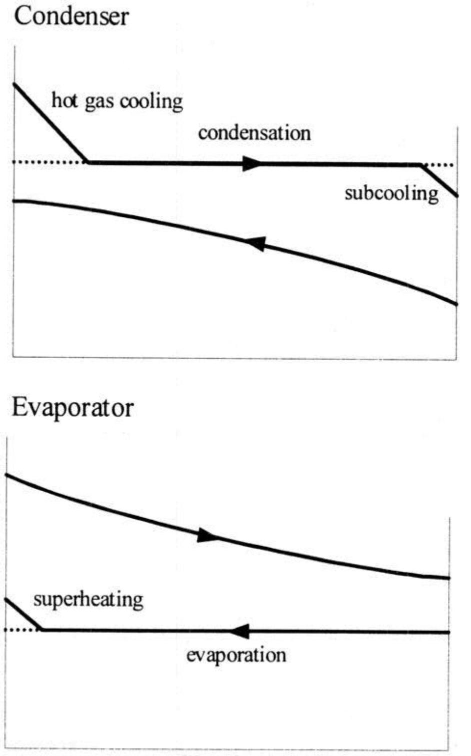

Fig. 2 : Temperature profiles for condenser and evaporator

In the next section, the heat transfer results are presented in three ways:
First, the logarithmic mean temperature difference ϑ_{ln} is given, calculated according to eq. 3, using the temperature differences between the *saturation temperature* and the water, or brine, temperature at the two ends of the heat exchangers (cf. fig. 3).
Second, the *inlet* temperature difference ϑ_{in} between the *saturation temperature* and water, or brine, is shown.
Third, the overall heat transfer coefficient k is presented. This value is calculated as

$$k = \frac{\dot{q}}{\vartheta_{ln}} \tag{1}$$

where $\dot{q}$ is the heat flux, defined as :

$$\dot{q} = \frac{\dot{Q}}{A} \tag{2}$$

and $\dot{Q}$ is the heating or cooling power and A the heat exchanger surface area.

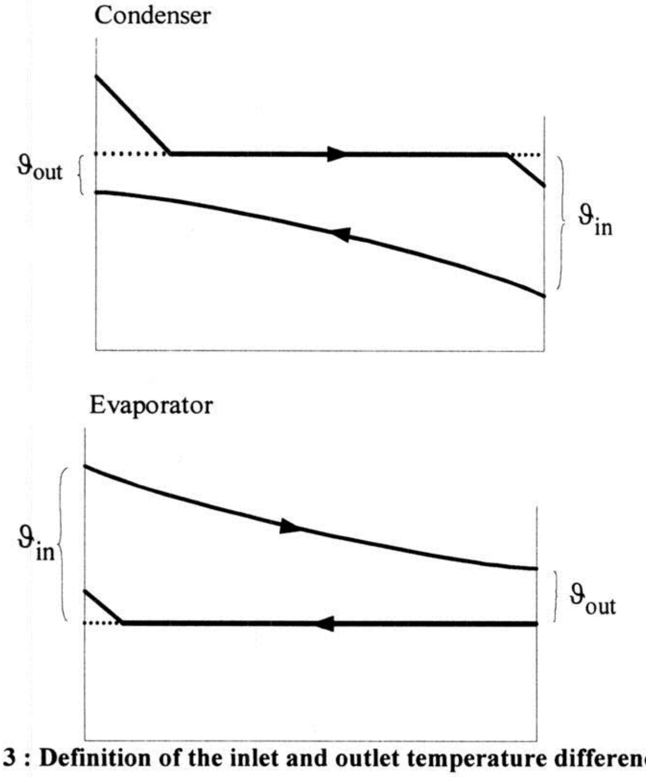

Fig. 3 : Definition of the inlet and outlet temperature differences.

By keeping the conditions on the water/brine side the same in the R22 and the propane tests, one may conclude that any difference in the mean, or inlet, temperature difference, or in the k-value, between the fluids is caused by different heat transfer characteristics on the refrigerant side of the heat exchanger.

5. Results and discussion

In the following diagrams, heat transfer and pressure drop characteristics are plotted as a function of the heating or cooling capacity.
The filled symbols and the open symbols represent propane and R22 respectively.

5.1. The condenser

Fig. 4 shows the logarithmic mean temperature difference between the two media, ϑ_{ln}, for the measurement performed at 35°C temperature of condensation. Fig. 5 shows the diagram for the inlet temperature difference, ϑ_{in}.

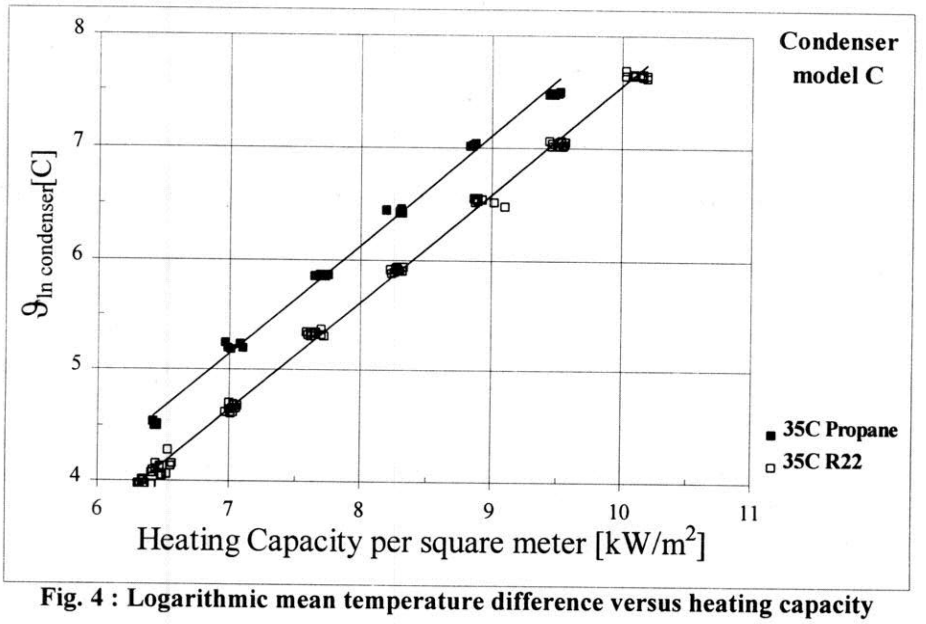

Fig. 4 : Logarithmic mean temperature difference versus heating capacity

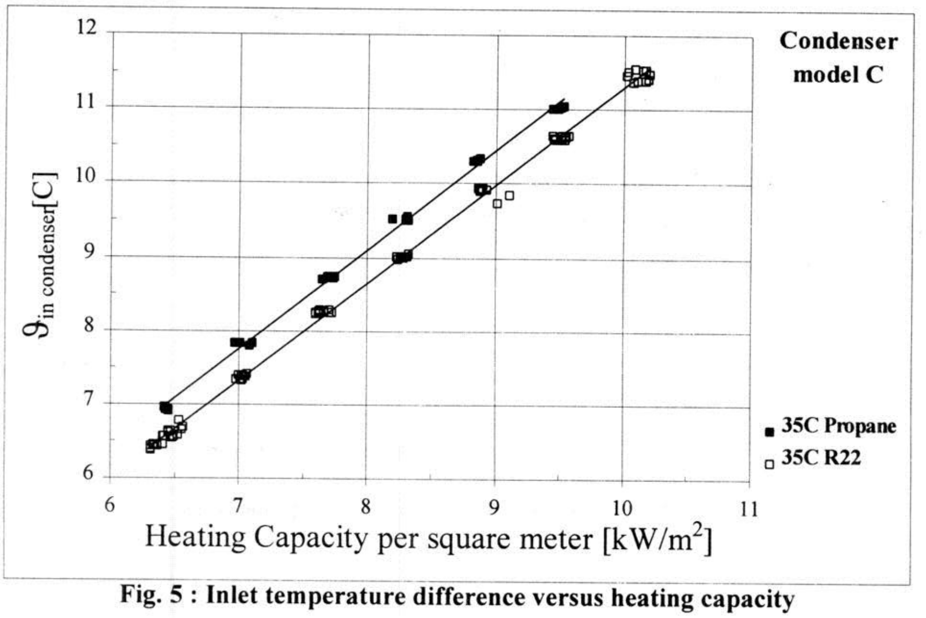

Fig. 5 : Inlet temperature difference versus heating capacity

In these tests, propane requires a slightly higher temperature difference than R22 to transfer a given capacity. The differences are small, about half a degree.

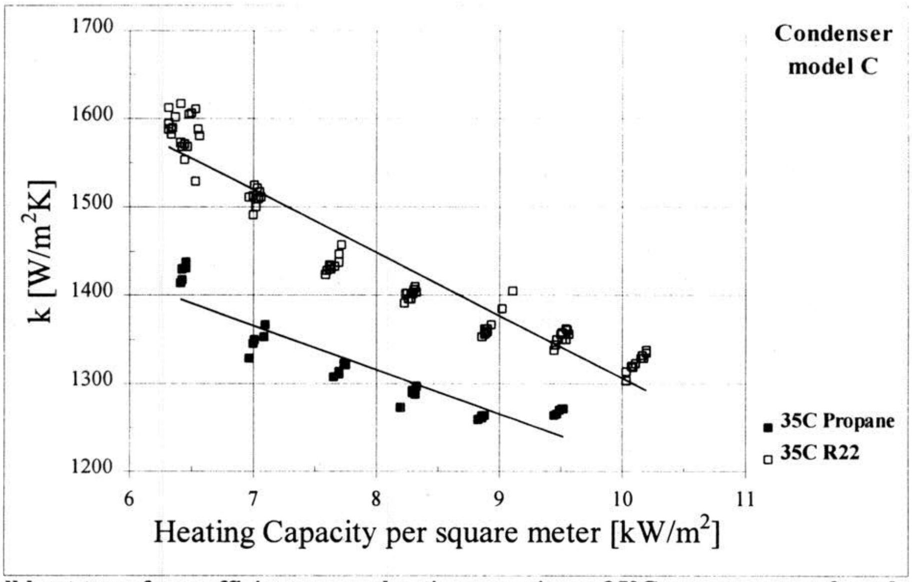

Fig. 6: Overall heat transfer coefficient versus heating capacity at 35°C temperature of condensation

From the logarithmic mean temperature difference, ϑ_{ln}, and the heating capacity the overall heat transfer coefficient k may be calculated. These values are presented in Fig. 6.
In the condenser, R22 has thus better heat transfer characteristics, with an overall heat transfer coefficient from 8% to 14% higher than propane. The difference decreases when the heating capacity increases.

In Fig. 7, the pressure drops in the condenser with the two media are compared. It can be seen that propane gives considerably lower pressure drop.
The pressure drop may also be expressed as a drop in the saturation temperature. This is shown in Fig. 8.

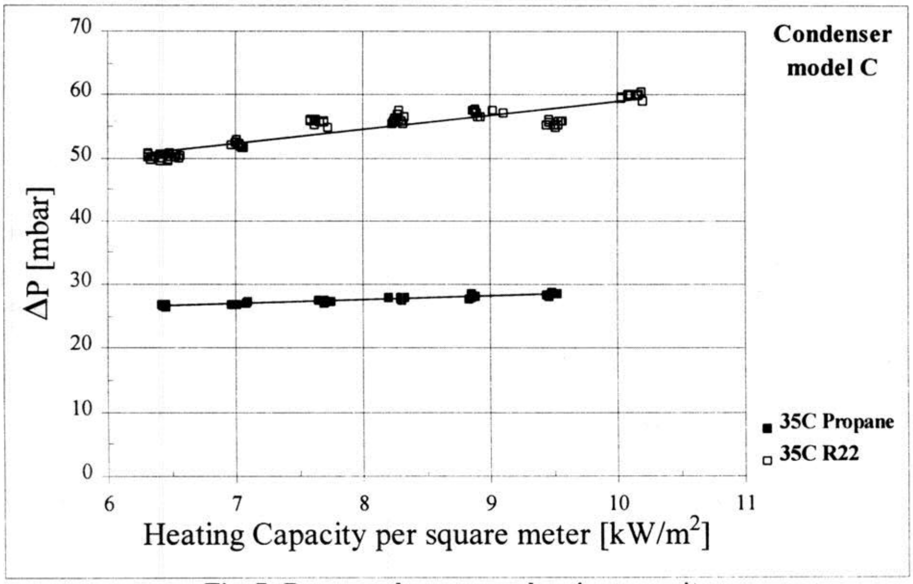

Fig. 7: Pressure drop versus heating capacity

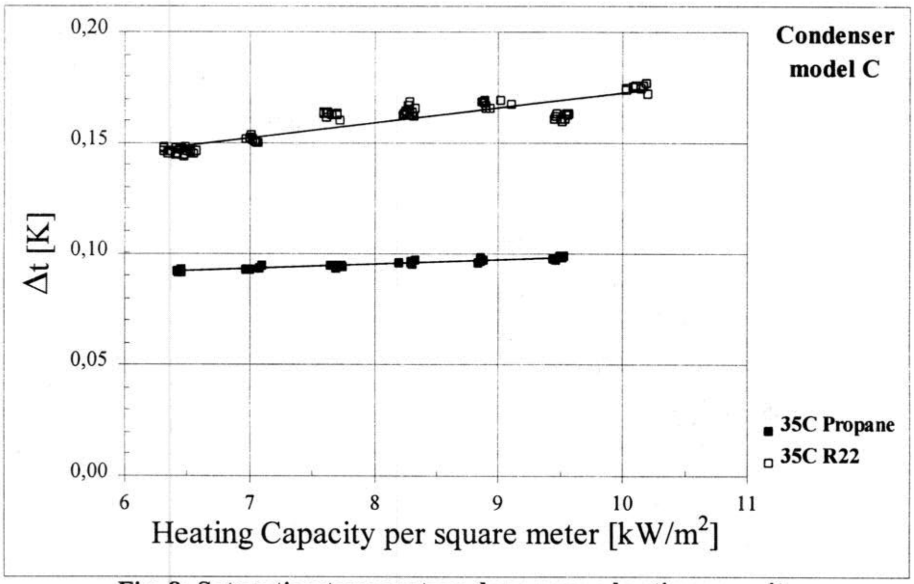

Fig. 8: Saturation temperature drop versus heating capacity

5.2. The evaporator

The same evaluation process as was used for the condenser has been used for the evaporator. Fig. 9 and 10 show the logarithmic mean temperature difference and the inlet temperature difference respectively for measurements performed at +5°C, 0°C and -5°C temperature of evaporation.

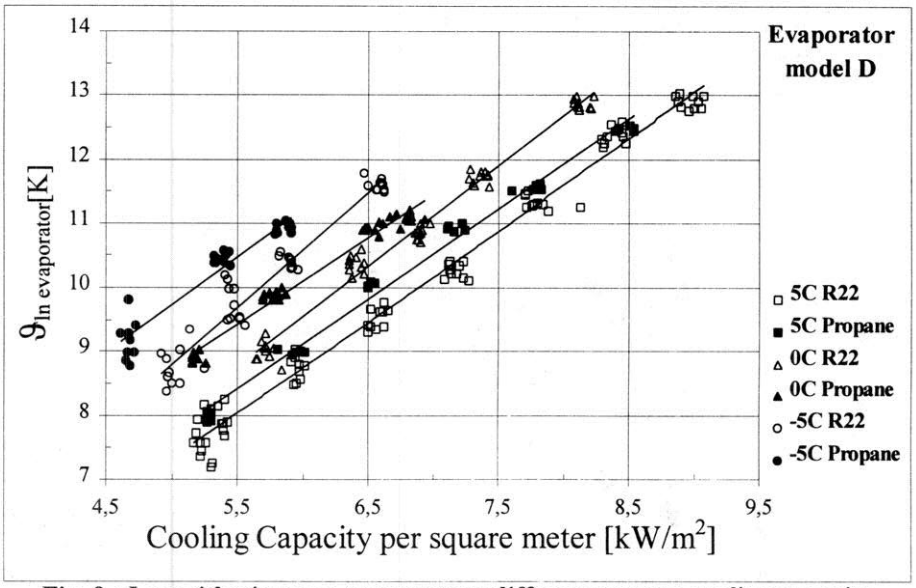

Fig. 9 : Logarithmic mean temperature difference versus cooling capacity

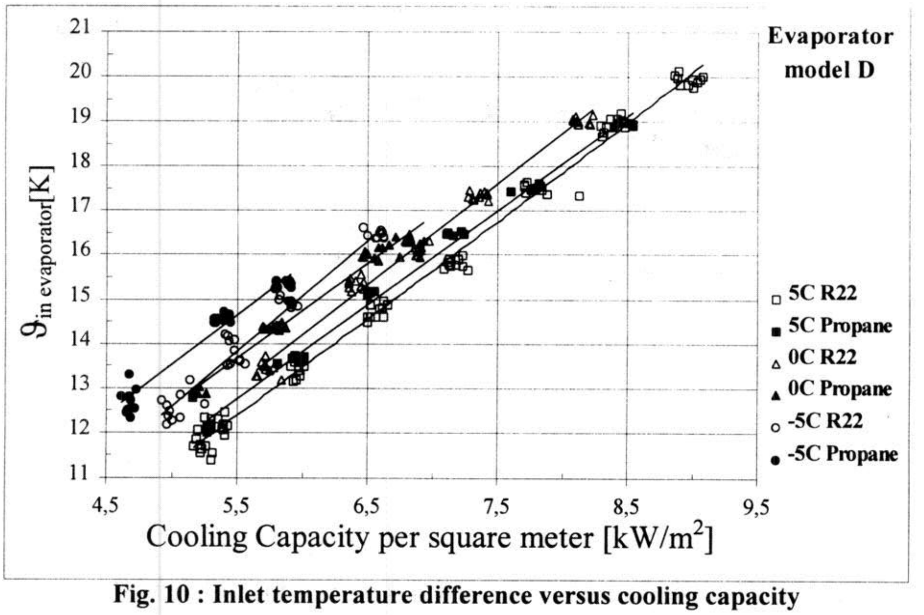

Fig. 10 : Inlet temperature difference versus cooling capacity

It can be seen that, just like in the condenser, propane needs a slightly higher temperature difference than R22 to transfer a certain capacity. The differences are small, between half a degree to one degree. The difference between the media is largest for the lower evaporation temperatures.
It should be mentioned here that preliminary tests with two other types of brazed plate evaporators has given the opposite results, with lower temperature differences for propane than for R22.
In fig. 11 the overall heat transfer coefficient in the evaporator is shown for the two refrigerants. The heat transfer coefficient is in this case about 5% higher with R22.
Note that with propane the heat transfer coefficient seem to be more dependent on the heat flux than with R22. With R22 the k-value is almost independent of the heat flux, which is surprising.

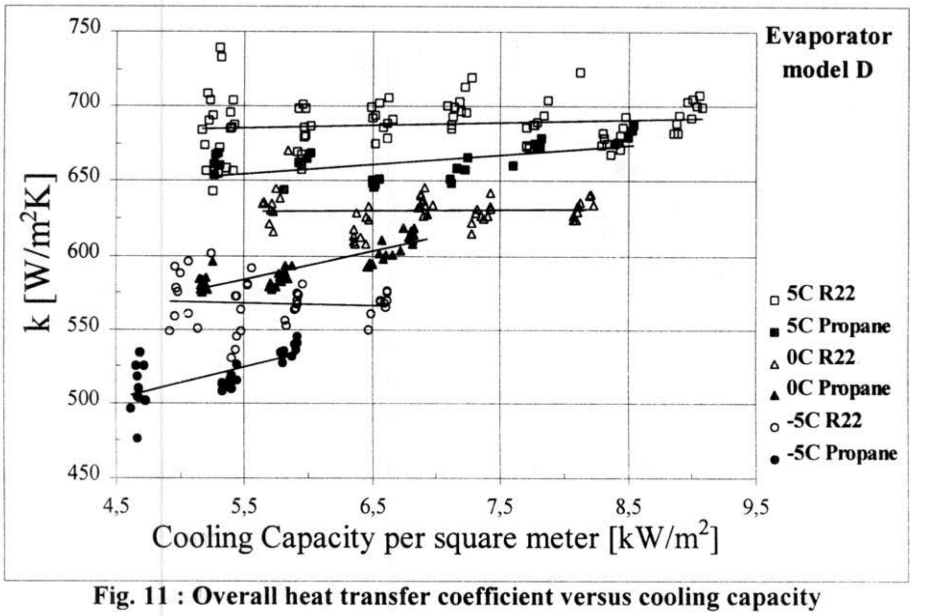

Fig. 11 : Overall heat transfer coefficient versus cooling capacity

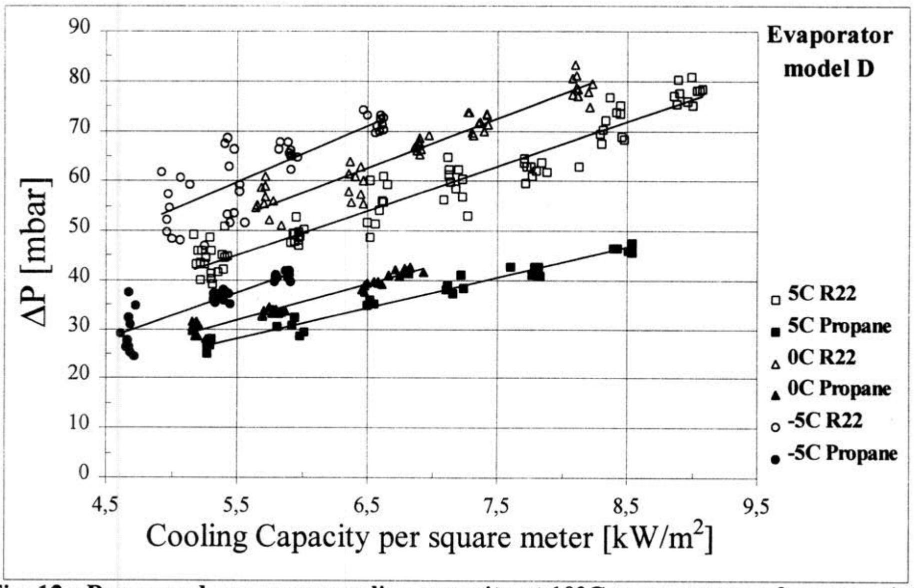

Fig. 12 : Pressure drop versus cooling capacity at 10°C temperature of evaporation

The comparison of the pressure drop characteristics, Fig. 12, shows that propane gives considerably lower pressure drop in the evaporator, just as was the case for the condenser.
The difference between the pressure drop values of propane and R22 is constant, independent of evaporation temperature. Fig. 13 shows the pressure drop expressed as the change of saturation temperature between the inlet and outlet of the evaporator.

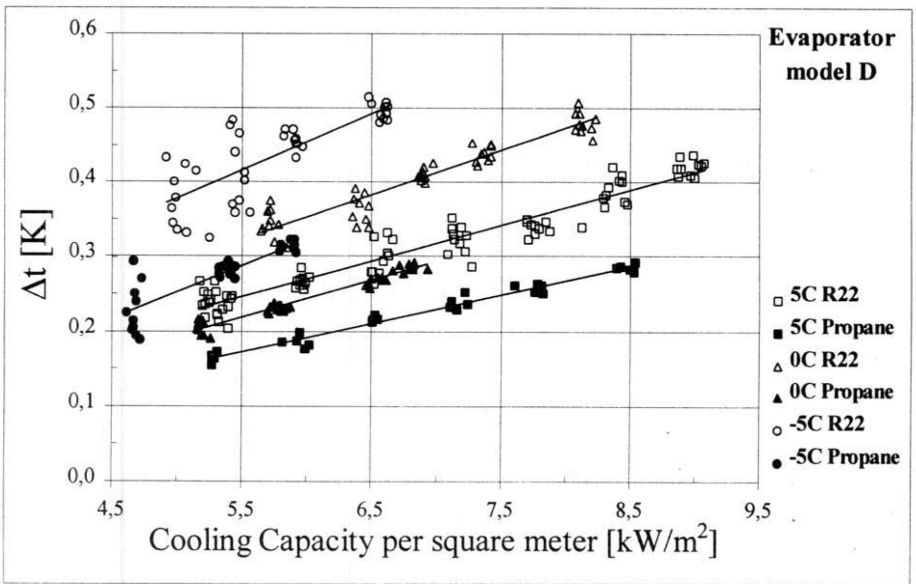

Fig. 13 : Temperature drop versus cooling capacity

6. Conclusions

To sum up, the results show that propane in the tested heat exchangers, at any given heat flux, gives overall heat transfer coefficients slightly lower than those of R22 in both condenser and evaporator. The tests also show that the pressure drop with propane is considerably lower than with R22 (40-50% lower). This fact opens a possibility for changing the design of the heat exchangers when used with propane, so as to allow a larger pressure drop and thereby achieving higher heat transfer coefficients.