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Propane for Heat Pump Applications Using Brazed Plate Heat Exchangers (part 3)

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Propane for Heat Pump Applications Using Brazed Plate Heat Exchangers

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Olivier Pelletier, Björn Palm

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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

Propane and R22 have been tested and compared as refrigerants in a small laboratory set-up simulating a domestic heat pump with brazed plate heat exchangers as condenser and evaporator. Two types of plate heat exchangers were tested, and each type was tested both as evaporator and as condenser. The two types differed only in the angle of the V-shaped pattern of the plates.

For the two refrigerants, the logarithmic mean temperature differences in the heat exchangers were measured and the overall heat transfer coefficient calculated. The conditions on the water/secondary refrigerant side of the heat exchangers were kept constant in all tests and the overall heat transfer coefficient therefore gave an indication of the relative film heat transfer coefficients on the refrigerant sides. The tests showed that the heat transfer coefficients in the tested evaporators were slightly higher with propane than with R22, while in the condenser the heat transfer coefficients were slightly lower with propane.

The pressure drop in both the condensers and the evaporators were about half as large with propane as with R22. As heat transfer coefficients increase with increasing pressure drop it is believed that with redesigned heat exchangers, propane could give higher heat transfer coefficients than R22 in both condenser and evaporator at equal pressure drop.

The tests were done with a small rolling piston type rotary compressor. It was found that both the total efficiency and the volumetric efficiency of the compressor were the same with the two refrigerants.

The heating capacity Q_1 of the heat pump was about 10% lower with propane than with R22. This difference is considerably smaller than anticipated from theoretical calculations based solely on the thermodynamic properties of the refrigerants. The heating coefficient of performance COP_1 was 4-5% higher for propane. Both results are in agreement with results reported elsewhere.

In the first part of the report thermodynamic and transport properties of propane and R22 are presented. Also, theoretical refrigeration process data and figures of merit for pressure drop and different types of heat transfer are compared. It is found that the results of the theoretical predictions are in good agreement with the experimental results.

NOMENCLATURE

A	heat exchanger surface area	[m^2]
Cp	specific heat capacity	[kJ/(kg.K)]
d	diameter	[m]
$\dot{E}$	power	[W]
h	specific enthalpy	[kJ/kg]
k	overall heat transfer coefficient	[W/(m^2.K)]
$\dot{m}$	mass flow	[kg/s]
p	pressure	[bar]
$\dot{q}$	heat flux	[W/m^2]
q_{vol}	volumetric capacity	[J/m^3]
$\dot{Q}_1, \dot{Q}_2$	heating and cooling capacity, resp.	[W]
t	temperature	[°C]
v	specific volume	[m^3/kg].
$\dot{V}$	volume flow	[m^3/s]

Greek letters

α	heat transfer coefficient	[W/(m^2.K)]
δ	wall thickness	[m]
ε_{vol}	volumetric energy demand	[J/m^3]
Δ	change of temperature, pressure or enthalpy	
ϑ	temperature difference	[K]
λ	thermal conductivity	[W/(m.K)]
μ	dynamic viscosity	[N.s/m^2]
υ	kinematic viscosity	[m^2/s]
ρ	density	[kg/m^3]

Subscripts

1	condenser
1c	condenser inlet
2	evaporator
2c	evaporator outlet
b	brine (evaporator side)
dis	displacement (compressor)
e	expansion valve
el	electrical power
h	hydraulic
is	isentropic
ln	logarithmic
r	refrigerant
tot	total
vol	volumetric
w	water (condenser side) / wall

Abbreviation

COP	Coefficient Of Performance
GWP	Global Warming Potential
ODP	Ozone Depleting Potential

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1. HISTORICAL BACKGROUND

Since the first refrigerating machine using ether was invented in the middle of the 19th century a large number of chemical substances which could be used as working fluids in refrigeration and heat pump compression systems have been tried. Up to the 1920´s only fluids naturally occurring in the environment were used. In the beginning of the 1930's the synthetic chloro-fluoro-carbons (CFCs) were introduced and reduced the interest for natural working fluids, except for Ammonia, which continued to be used in large refrigeration plants. The CFCs were at this time called "safety refrigerants" as they were non flammable and non toxic and had no known detrimental effect on the environment. Half a century later, reports on the CFCs´ and HCFCs´ effects on the Ozone layer and on their contribution to the green house effect lead to an international agreement to phase out of these substances, the "Montreal Protocol". This agreement started a search for new, environmentally benign, refrigerants. The findings about the CFCs´ influence on stratospheric Ozone also pointed to the danger of utilising synthetic substances on a large scale, and environmental groups and organisations therefore started to advocate the use of natural fluids as refrigerants. An important event in the re-introduction of natural working fluids was the presentation in 1992 by Greenpeace of a small refrigerator with hydrocarbon refrigerant and (H)CFC-free insulation.

At the Division of Applied Thermodynamic and Refrigeration of Kungliga Tekniska Högskolan, KTH, studies of alternative working fluids started in the mid-1980s. These studies concern heat transfer characteristics of new synthetic substances and new mixtures as well as of natural refrigerants. Among the natural refrigerants a special interest is directed towards the alkanes and in particular propane which is the focus of this report.

2. PROPANE AS A REFRIGERANT

2.1 Suits today's technology

From a technical point of view, propane and propane-based mixtures can easily replace halocarbons in small refrigerating and heat pump systems. It is a main advantage of halocarbons compared to ammonia and other natural fluids that no change in technology is required as these fluids are compatible with all materials commonly used in (H)CFC refrigeration plants. For example, hermetic compressors, standard R-22 expansion valves, copper tubing etc., work well with propane, normal lubricating oils return better with propane than with any halocarbon, and the discharge temperature from the compressor is lower with propane than with R-22. This means that equipment produced in large scale for (H)CFC refrigerants can be used also with propane. The compatibility is demonstrated by the fact that several R22 plants have been retrofitted with propane or other alkanes.

The charge (mass) of refrigerant needed when using hydrocarbons is considerably smaller than with the halocarbons. Since the densities of CFC liquids are more than two times that of propane, the charges may often be reduced to less than half in a retrofit situation. Also, when designing new equipment, smaller pipes can be used with this refrigerant without increasing the pressure drop. The present development towards compact heat exchanger equipment, especially plate heat exchangers, has opened the possibility of reducing the required liquid charge dramatically in new systems, down to 50 g per kW in certain cases.

The main problem when using hydrocarbons as refrigerants is the flammability, and technically, to design safe refrigerating systems. Without neglecting or underestimating the hazards, it is the opinion of the present authors that these problems could be overcome. Propane or city gas is already used in many homes world-wide in gas stoves and water

heaters. These uses must be considerably more hazardous as the gas in these cases is let out into the air through manually operated valves, and then voluntarily ignited. A refrigerating system, on the other hand, is normally hermetically sealed and could be designed without sources of ignition. Also, the amount of gas which could escape from stoves and water heaters etc. in case of a leak is always much larger than in the case of a refrigerator, a freezer or a domestic heat pump.

2.2 Environmental impact of propane

As this report will show, the thermodynamic qualities of propane make it an excellent refrigerant. In addition to those qualities propane has the benefit of being environmentally safe. Contrary to the CFCs and HCFCs propane has no ozone depletion potential, a negligible global warming potential (used as refrigerant), and no toxic or irritating decomposition products. Also, there are less problems with toxic waste produced during production. The following table summarises important environmental properties of some refrigerants /1/. As can be seen the only negative aspect of propane is its flammability, which means that adequate safety measures must be taken when retrofitting or designing new refrigeration and heat pump systems using propane.

Table A: Environmental- and safety characteristics of some refrigerants

Refrigerant	CFC R-12	HFC R-134a	HCFC R-22	C_3H_8 R-290 Propane	NH_3 R-717 Ammonia	CO_2 R-744
Natural	No	No	No	Yes	Yes	Yes
ODP	1.0	0	0.05	0	0	0
GWP[1] 100 years	7100	1200	1500	0	0	1 (0)[2]
20 years	7100	3100	4100			1 (0)
Maximum safe amount per room vol.[3] (vol. % / kg/m^3)	4.0 / 0.2	-	4.2 / 0.15	0.44 / 0.008	-	5.5 / 0.1
Flammable/explosive[4]	No	No	No	Yes	Yes	No
Flammability limits in air (vol. %)	-	-	-	2.2 / 9.5	15.5 / 27	-
Toxic/irritating decomposition products	Yes	Yes	Yes	No	No	No
Toxicity class[5]	6	6	5a	5b	2	5
Molar mass	120.92	102.03	86.48	44.1	17.03	44.01

[1] Global Warming Potential in relation to CO_2, with 20 and 100 years integration time (IPCC 1990, 1992).
[2] Abundant amounts of CO_2 is recovered from waste gas. Thus, the effective GWP of commercial CO_2 used as refrigerant is 0.
[3] Maximum refrigerant charge in relation to refrigerated room volume, as suggested in ANSI/ASHRAE 15-1989 : Safety Code for Mechanical Refrigeration.
[4] Although considered to be non-flammable, both R-134a and R-22 are combustible in certain mixtures with air at elevated pressures, but ignition may be difficult.
[5] PAFT-Test. 6 = non toxic

3. SOME SAFETY CONSIDERATIONS

3.1 Regulations on flammability

As there has been no interest for using propane and other flammable hydrocarbons as refrigerants during a long time, the national regulations governing the use of refrigerants do not include any guidelines for the design of refrigeration systems with these fluids. Instead, the use of flammable fluids as refrigerants have been prohibited in many countries. Presently, the regulations are being changed in many European countries to allow the use of hydrocarbons as refrigerants under certain conditions. As the lack of regulations has been a main obstacle for the introduction of these fluids it can be expected that the new regulations will lead to a major increase in their use.

In spite of the lack of regulations, hydrocarbons have been allowed in domestic refrigerators and freezers in almost all European countries, in Australia and in at least some countries in Asia and South America /2/. In some countries hydrocarbons are also allowed in small domestic heat pumps. In Germany, about 30% of new domestic heat pumps are utilizing hydrocarbon refrigerants, and propane is also being used in indirect systems in supermarkets /3/.

In the UK, the use of hyrocarbon refrigerants has been promoted by the company Calor Gas which sells hydrocarbons and hydrocarbon blends of high quality for use as refrigerants. Several British manufacturers have already introduced, or have announced their intentions to introduce, products such as display cases for food and beverages, air conditioning units, heat pumps, cabinets for food industry and water chillers using hydrocarbon refrigerants /4/. The introduction of these products has been made possible by the new British Standard BS4434 which allows, in any type of building, up to 1.5 kg of hydrocarbons in sealed systems and up to 5 kg if the system is indirect and the refrigerant containing parts are placed in a special machinery room. In offices, small shops and small restaurants, 2.5 kg is allowed in sealed systems and up to 10 kg in indirect systems with machinery room. Considering the low density of these fluids this means that fairly large heating or refrigerating capacities may be achieved.

In Sweden, hydrocarbons are now being used in domestic refrigerators and freezers as well as in some brands of heat pumps for domestic use. Apart from these small appliances there are approximately 20 larger plants installed or under construction. This in spite of the lack of regulations, which complicates the design and necessitates the contact with several authorities like the local fire department, the national inspectorate of explosives and flammables (Sprängämnesinpektionen), the swedish electrical commission (Sveriges elektriska kommission), for special permits. The interest from industry for hydrocarbon technology is large, and when the swedish refrigeration norm (Svensk Kylnorm) has been revised to include flammable refrigerants it can be expected that the number of plants with these refrigerants will increase.

3.2 Retrofit of hydrocarbon refrigerants

The general opinion in Sweden has been that (H)CFC plants should not be retrofitted with hydrocarbon refrigerants, even though this is technically possible. The reason for this seem to be the concern that these older plants are more liable to start leaking than new ones. Outside Sweden there seem to be no such general agreement although the number of retrofitted plants is limited. For the moment no good estimate of the number of retrofitted plants is available. According to Stene /3/, "a large number" of small air conditioners, split-type heat pumps and water chillers in the UK have been converted to hydrocarbon blends during the last few years,

while in Germany the number of retrofitted plants is "...relatively small". According to Paul Blacklock, general manager of Calor Gas, retrofits have been done on CFC, HCFC and HFC systems, "though not in any large quantities".

As already noted, hydrocarbons are compatible with the components and materials usually found in (H)CFC systems as well as with mineral/ester oils. The only changes necessary at a retrofit are thus those to ensure the safety of the system. This means that the electrical system has to be looked over. Parts of the electrical system probably must be moved or put inside an air tight enclosure, while other parts should be exchanged. Also, the tubing system should be changed so as to minimize the risk of refrigerant leakage.

In general, when dealing with hydrocarbons as refrigerants, the machinery rooms should be vented to the outside, a refrigerant vapour detector should be installed, and it is wise to have a switch isolating all electrical circuits other than emergency lightning, ventilation and gas detectors in case of a leak.

The risks connected with flammability depend, however, on the risks of the occurrence of a leak and the amounts of gas coming out of the system in case it occurs.

4. PROPERTIES OF PROPANE VS. R22

In this chapter, the properties of propane will be presented and compared to those of R22. Thermodynamic and transport properties are presented, as well as refrigeration properties, e.g. theoretical volumetric cooling capacity and COP, and figures of merit for heat transfer and pressure drop, calculated from well known correlations. When nothing else is stated, the information is taken from a computer code /5/.

4.1 Thermodynamic and transport properties

4.1.1 Saturation pressure

Propane is a possible substitute for refrigerant R22 as the saturation pressures of the two fluids are similar in the temperature range in which R22 has by tradition been used (-20°C to +60°C).

As shown in Figure 1, the saturation pressures are almost equal at low temperatures, while for higher temperatures the pressure of propane is lower.

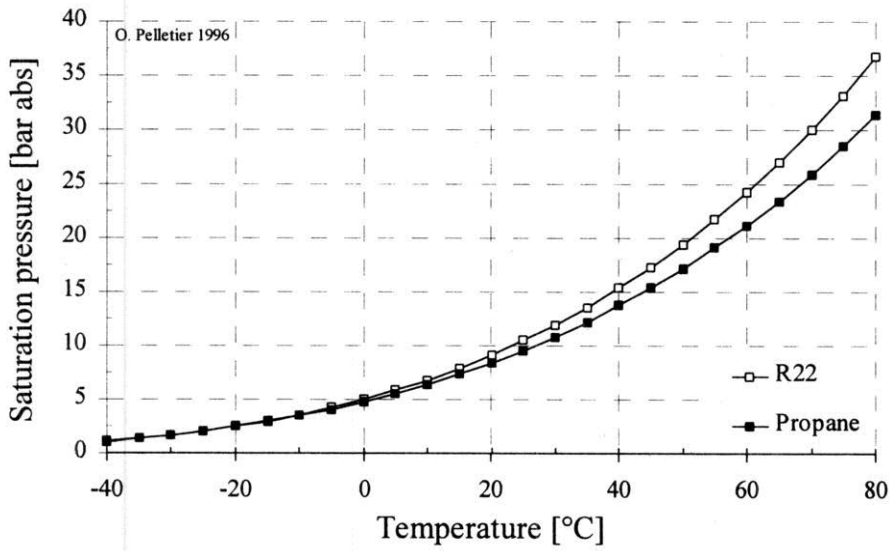

Figure 1 - Saturation pressures for R22 and propane

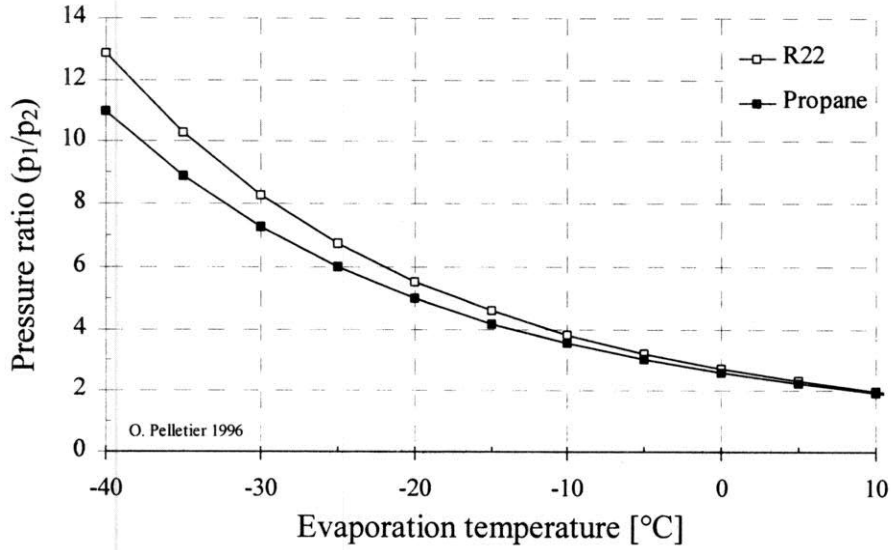

Figure 2 - Pressure ratio as a function of evaporation temperature at 35°C condensation temperature.

This difference results in two benefits for propane as compared to R22. First, using equipment designed for a certain maximum pressure, propane allows condensation at a higher temperature than R22, second, for any given temperature lift, the pressure ratio is lower for propane than for R22, Figure 2, which is beneficial for the volumetric efficiency of the compressor.

4.1.2 Density

Figure 3 and Figure 4 show the densities of saturated R22 and propane in liquid and vapour phase. As shown, the density of propane is about half of that of R22, in both phases. As a result the mass of propane required in a certain refrigerating system is about half of that required if R22 is used. As will be shown later, the difference in density will also influence the pressure drop and the heat transfer.

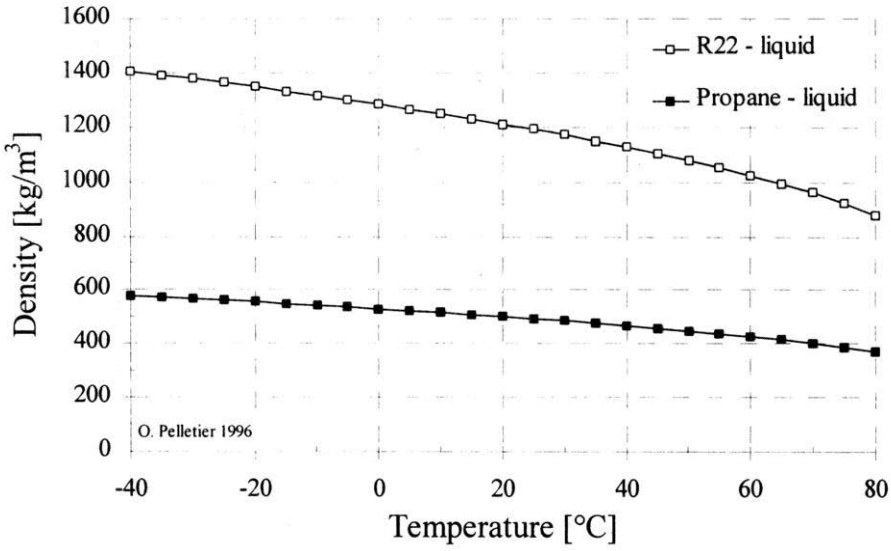

Figure 3 - Specific volume of saturated liquid

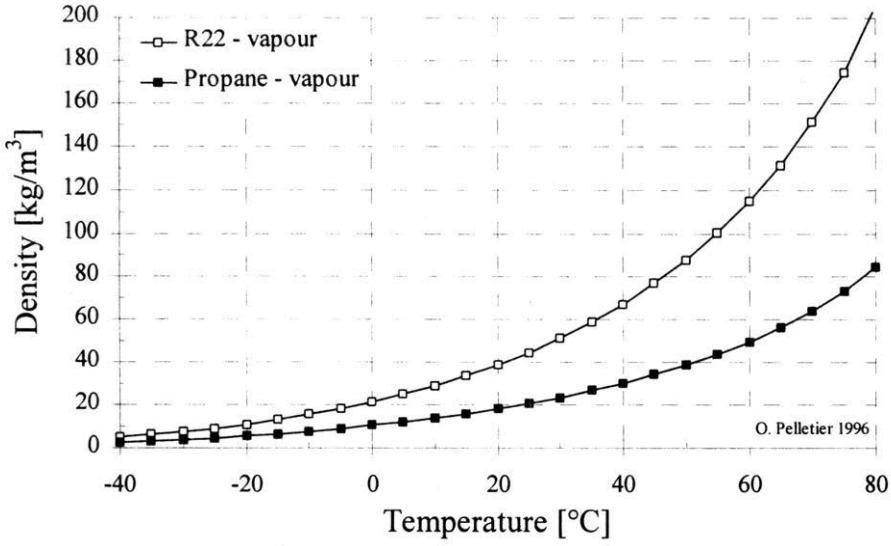

Figure 4 - Specific volume of saturated vapour

4.1.3 Molecular weight

As may be understood from the vapour specific volume, the molecular weight of propane is about half of that of R22. (44,10 for propane and 86,47 for R22). The higher molecular weight of R22 may be expected to increase the losses in the compressor, thus reducing the compressor efficiency as compared to propane.

4.1.4 Heat of vaporisation

As shown in Figure 5 propane has a considerably larger heat of vaporisation than R22. This means that, at equal mass flows, the capacity of a propane system is considerably larger than of an R22 system. As the density of propane is considerably lower, at equal *volume* flow, the capacities will be quite similar (cf. Figure 10).

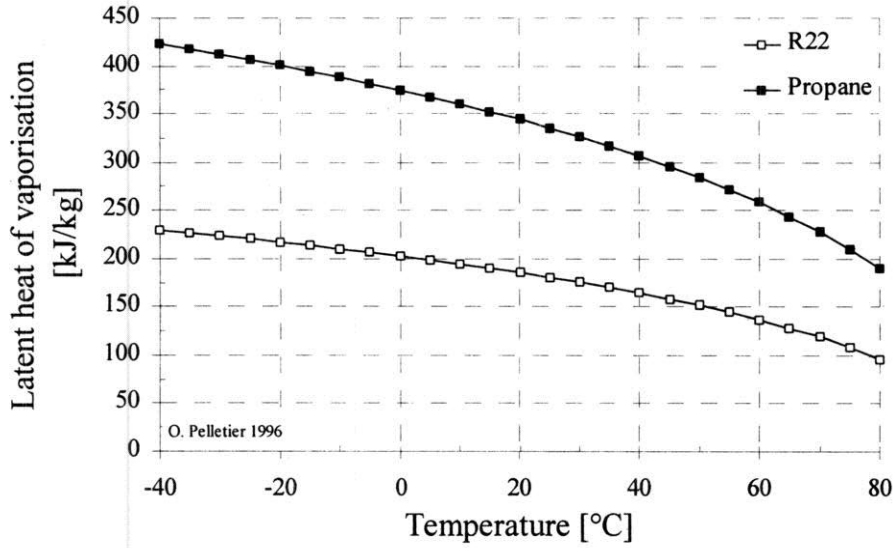

Figure 5 - Heat of vaporisation vs temperature for R22 and propane

4.1.5 Thermal conductivity

The thermal conductivity is of importance to heat transfer both in single phase and two phase flows. Figure 6 shows that propane has higher thermal conductivity in both liquid and vapour phase. As will be shown later, this will influence the figures of merit for heat transfer.

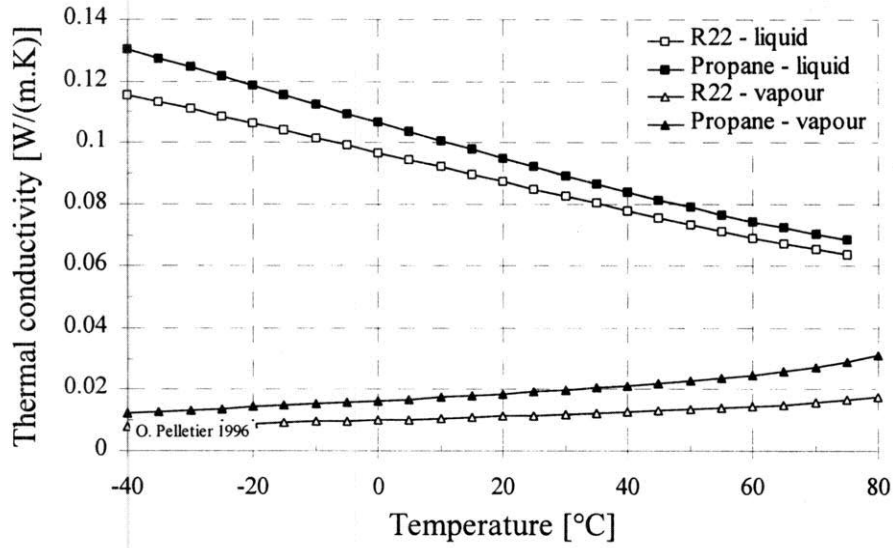

Figure 6 - Thermal conductivity of propane and R22 in saturated liquid and vapour phase.

4.1.6 Viscosity

The viscosity of the fluid is important both to heat transfer and to pressure drop. As shown in Figure 7, the *dynamic* viscosity is considerably higher for R22 than for propane. The *kinematic* viscosity is defined as the dynamic viscosity divided by the density of the fluid:

$$\upsilon = \frac{\mu}{\rho} \qquad \text{Eq. 1}$$

The *kinematic* viscosities are rather similar for R22 and propane, Figure 8.

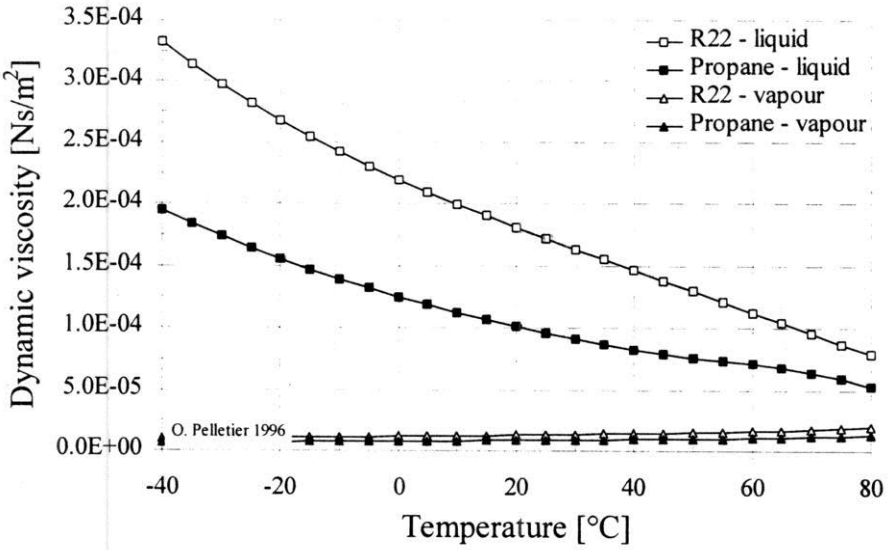

Figure 7 - Dynamic viscosity of R22 and propane in saturated liquid and vapour phase

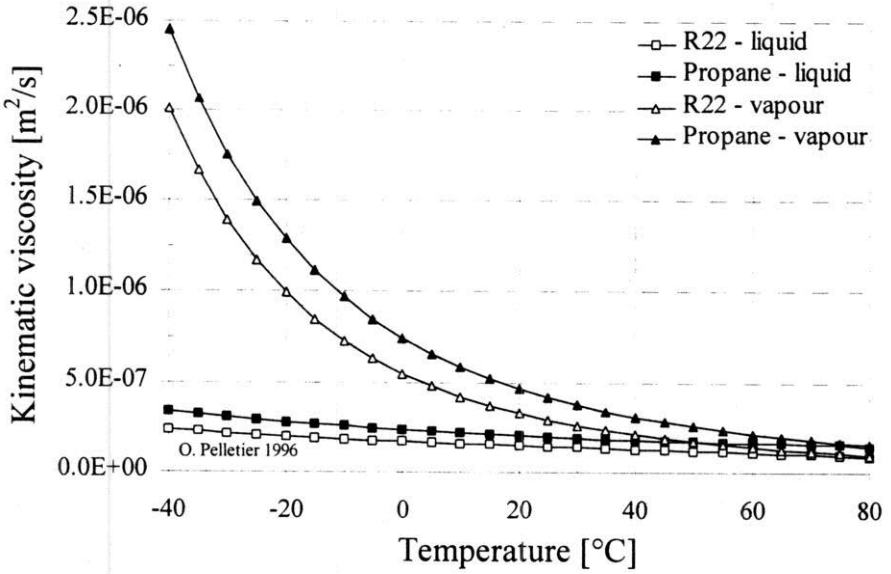

Figure 8 - Kinematic viscosity of R22 and propane in saturated liquid and vapour phase

4.1.7 Surface tension

The surface tension of propane is slightly lower than that of R22, as shown in Figure 9, according to /6/. This property is important to the heat transfer in the evaporator in case of nucleate boiling, as it influences the nucleation of vapour bubbles. The boiling heat transfer coefficient is expected to decrease with increasing surface tension.

Surface tension may also influence the condensation heat transfer on finned surfaces or other geometries with non-smooth surfaces where the vapour liquid interface is curved. In this case, high surface tension has a positive influence on heat transfer as it acts to reduce the liquid film thickness on the condenser surfaces.

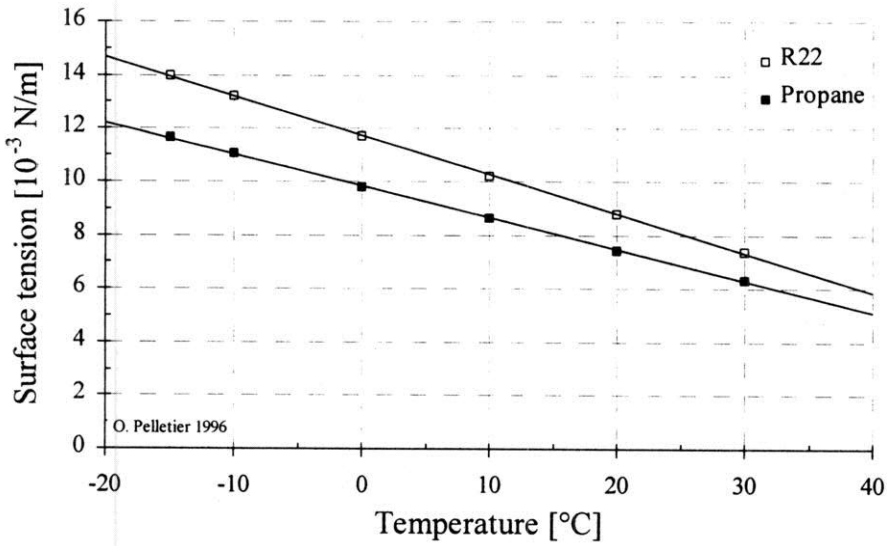

Figure 9 - Surface tension of R22 and propane

4.2 Refrigeration properties

4.2.1 Volumetric capacity

The refrigerating capacity of a certain system using the two refrigerants can be compared by calculating the volumetric cooling capacity q_{vol} defined as:

$$q_{vol} = \frac{h_{2c} - h_e}{v_{2c}} \qquad \text{Eq. 2}$$

where
h_{2c} = enthalpy at compressor inlet
h_e = enthalpy at evaporator inlet
v_{2c} = specific volume at compressor inlet.

For heat pumps it may be of interest to define similarly a volumetric heating capacity as

$$q_{vol,h} = \frac{h_{1c} - h_s}{v_{2c}} \qquad \text{Eq. 3}$$

where
h_{1c} = enthalpy at compressor *outlet*

Figure 10 and Figure 11 show the *theoretical* volumetric cooling and heating capacities as a function of the evaporation temperature assuming constant condensation temperature of 35°C and 55°C, no subcooling, 5°C superheat and *isentropic compression*. Both figures show that the capacity of a refrigerating or heat pump system may be expected to be slightly higher with R22 than with propane. The difference is 12-20% depending on the conditions, lower at the lower condensation temperatures. This theoretical comparison is qualitatively in agreement with the experimental results reported here and with results in numerous reports in the literature. The experimentally measured differences are usually much smaller than the theoretical differences. According to some estimates /7/ and experimental results /8/, the use of pure propane reduces the heating capacity by 4 to 15% in an R22 system, depending on the operating conditions.

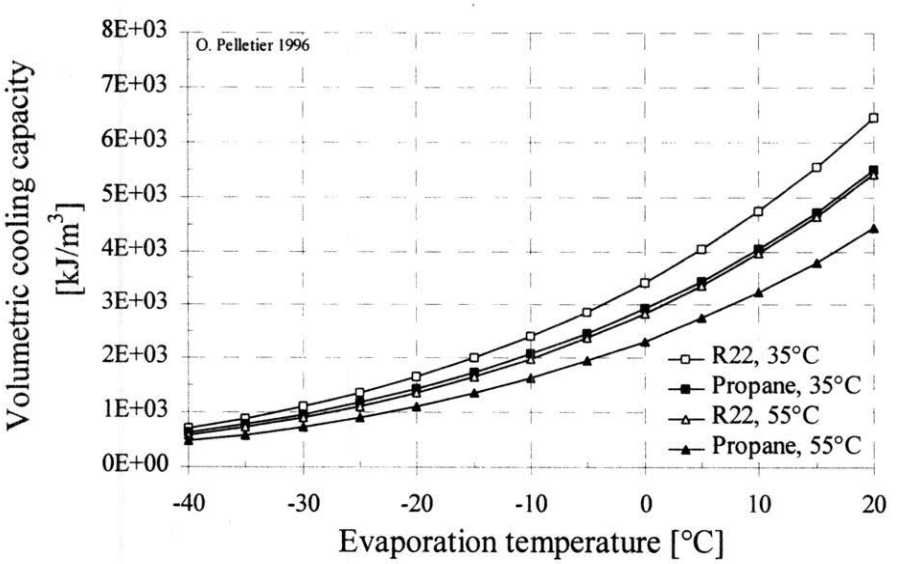

Figure 10 - Volumetric cooling capacity of R22 and propane at 35°C and 55°C condensation temperature.

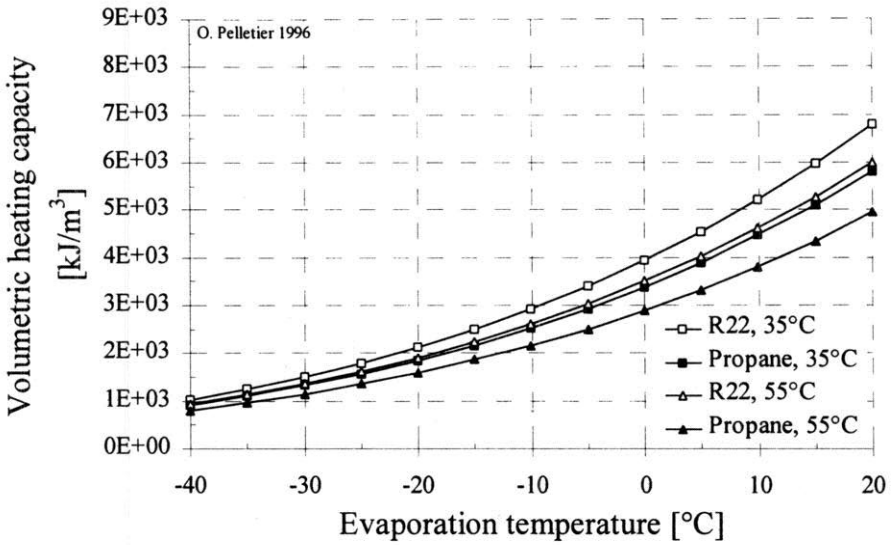

Figure 11 - Volumetric heating capacity for R22 and propane at 35°C and 55°C condensation temperature.

4.2.2 Volumetric energy demand

It is also of interest to compare the necessary driving power to the compressor with the two refrigerants. This may be done by comparing the volumetric energy demand, ε_{vol} defined as:

$$\varepsilon_{vol} = \frac{h_{1c} - h_{2c}}{v_{2c}} \qquad \text{Eq. 4}$$

ε_{vol} is the energy needed for the compression of one unit (by volume) of refrigerant. In Figure 12, ε_{vol} is plotted vs. the evaporation temperature for the condensation temperatures 35°C and 55°C, assuming isentropic compression, no subcooling and 5°C of superheat.

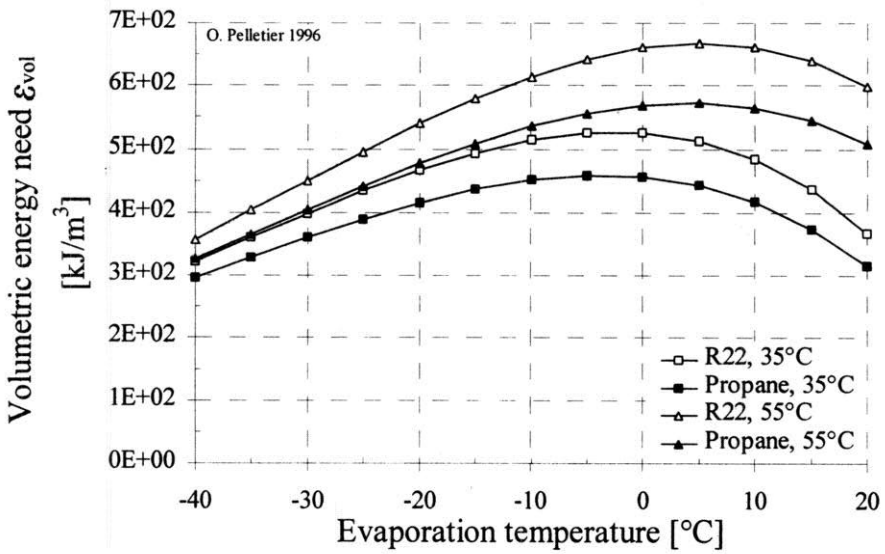

Figure 12 - Volumetric energy demand of compressor with R22 and propane at 35 and 55°C condensation temperature.

It is noted that the volumetric driving energy is 8-16% lower for propane than for R22, the difference being smallest at low evaporation temperatures. This is in agreement with test results reported here and with results reported in the literature. The lower volumetric energy demand means that a hermetic compressor designed for R22 has a slightly oversized motor when used with propane.

For the condensation temperatures of interest, the volumetric energy demand has a common maximum for the two refrigerants at evaporation temperatures around 0°C.

4.2.3 Coefficient of performance, COP

The coefficients of performance may be calculated from the volumetric capacities and the volumetric driving energies as:

$$COP_1 = \frac{q_{vol,h}}{\varepsilon_{vol}}$$

$$COP_2 = \frac{q_{vol}}{\varepsilon_{vol}} \qquad \text{Eq. 5}$$

where COP_1 and COP_2 mean heating and cooling coefficient of performance, respectively.

As noted above, both the numerators and the denominators in these equations are slightly larger for R22 than for propane. In a theoretical comparison, the differences in COP between the two refrigerants will therefore be very small, and depend on the conditions at which the comparison is done. Under the conditions used above (5°C superheating, no subcooling, isentropic compression) the COP_1 is 0-7% higher for R22. From experiments it is usually reported that the COPs of propane is higher than of R22, about 4 to 12%, depending on the operating conditions /7/ /8/. This increase cannot be explained only by the fact that the heat exchangers are operating at lower load with propane than with R22.

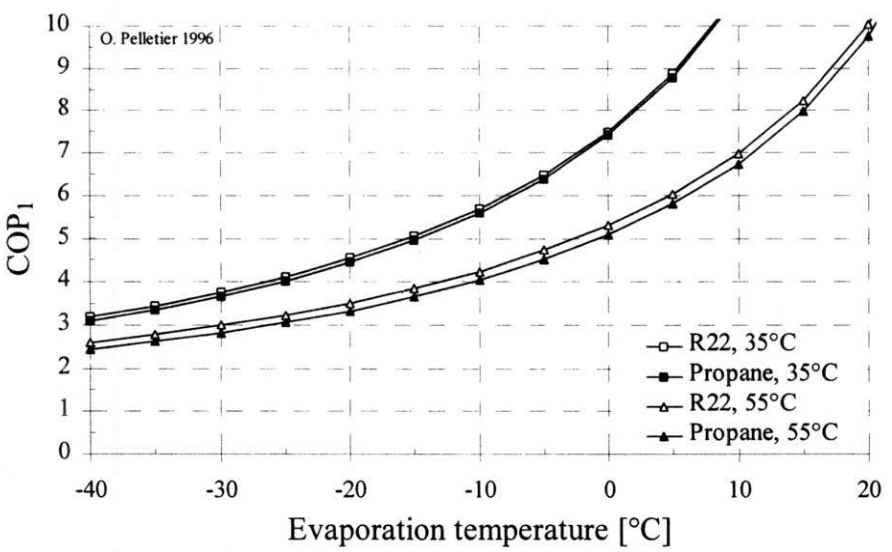

Figure 13 - COP_1 for R22 and propane at 35°C and 55°C condensation temperature.

4.2.4 Influence of superheat and subcooling on COP1

In a theoretical comparison, the degree of superheat before the compressor inlet has opposite effects on the COP_1s of the two refrigerants. With R22, the COP_1 decreases with increasing superheat, while the opposite is true for propane. For low condensation temperatures, the COP_1 of propane is higher than for R22 at high superheats, Figure 14. For high condensation temperatures, the COP_1 of R22 is always higher, but the difference decreases with increasing superheat. This means that with propane it is advantageous to use an internal heat exchanger between the suction line and the liquid line.

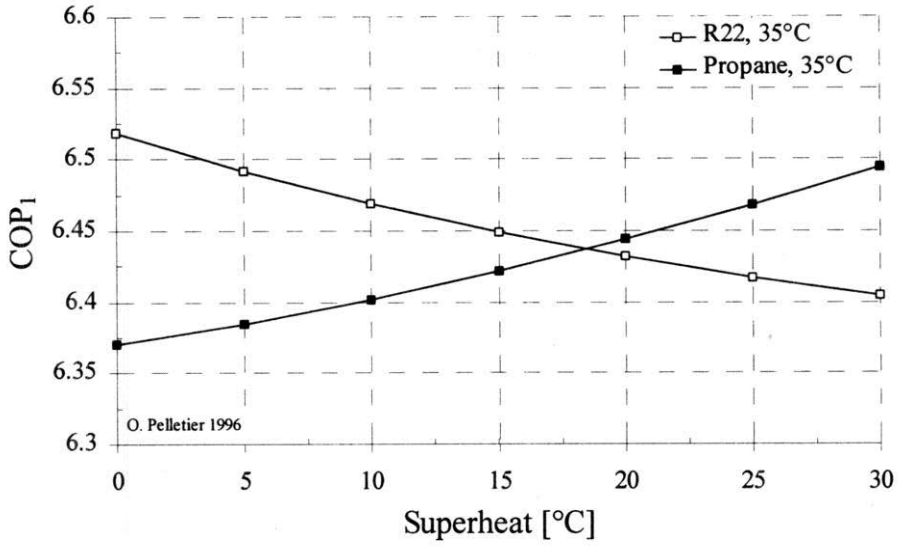

Figure 14 - Influence of superheat on COP_1, assuming 35°C condensation temperature, -5°C evaporation temperature and no subcooling.

Subcooling will of course always have a positive influence on COP_1 as long as the heat can be used. As shown in Figure 15 subcooling is more advantageous with propane than with R22.

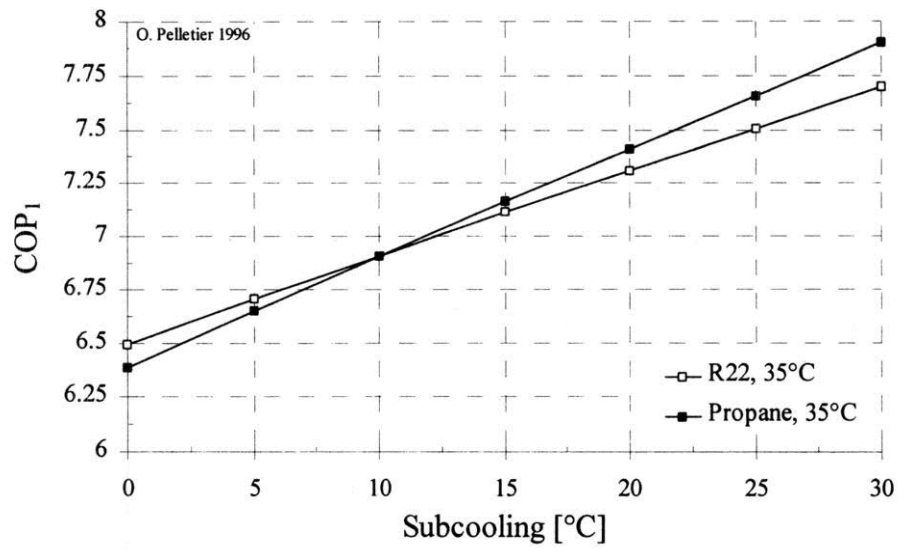

Figure 15 - Influence of subcooling on COP_1, assuming 35°C condensation temperature, -5°C evaporation temperature and 5°C superheat.

4.2.5 Discharge temperature

The discharge temperature is often a limiting factor for how large pressure ratio can be achieved in one compression stage. Figure 16 shows theoretical discharge temperatures calculated assuming isentropic compression and 5°C superheat for the condensation temperatures 35°C and 55°C. As shown, propane gives considerably lower discharge temperatures than R22, thus allowing higher temperature lifts in one-stage systems. This is probably one of the most important differences between the two refrigerants from a refrigerating engineer's point of view.

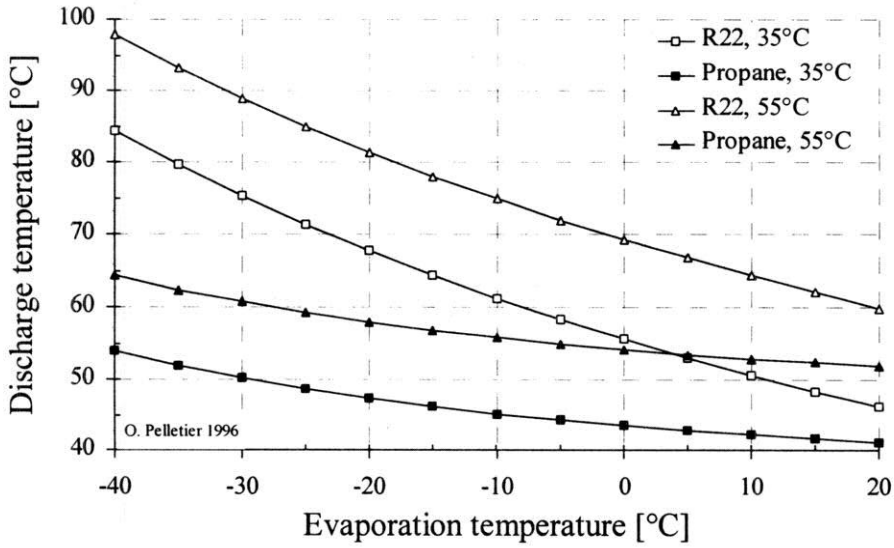

Figure 16 - Discharge temperatures for R22 and propane at 35°C and 55°C condensation temperatures.

4.3 Figures of merit for heat transfer and pressure drop

4.3.1 Friction pressure drop

In one-phase flow, the pressure drop may be calculated as:

$$\Delta p = f_1 \cdot \rho \cdot w^2 \cdot \frac{L}{d} \qquad \text{Eq. 6}$$

where f_1 is the friction factor, which for turbulent flow in smooth circular tubes may be expressed by Blasius' correlation:

$$f_1 = 0{,}158 \cdot \mathrm{Re}^{-1/4} \qquad \text{Eq. 7}$$

where *Re* is the Reynolds number:

$$\mathrm{Re} = \frac{w \cdot d}{\upsilon} \qquad \text{Eq. 8}$$

Considering a heat pump system having a certain heating capacity, the velocity of the refrigerant can be approximately calculated as:

$$w = \frac{\dot{V}}{A} = \frac{\dot{m}}{\rho \cdot \frac{\pi \cdot d^2}{4}} = \frac{\frac{\dot{Q}}{r}}{\rho \cdot \frac{\pi \cdot d^2}{4}} = \frac{4 \cdot \dot{Q}}{r \cdot \rho \cdot \pi \cdot d^2} \qquad \text{Eq. 9}$$

Combining the above equations we get an expression for the pressure drop for turbulent flow in circular tubes:

$$\Delta p = 0{,}241 \cdot \frac{L \cdot \dot{Q}^{7/4}}{d^{19/4}} \cdot \frac{\mu^{1/4}}{\rho \cdot r^{7/4}} \qquad \text{Eq. 10}$$

When comparing the two refrigerants, the ratio of thermodynamic properties in the last factor of this equation can be used as a figure of merit (a low value indicates a low pressure drop). In Figure 17 the ratio of these figures of merit are presented for liquid and vapour flow.

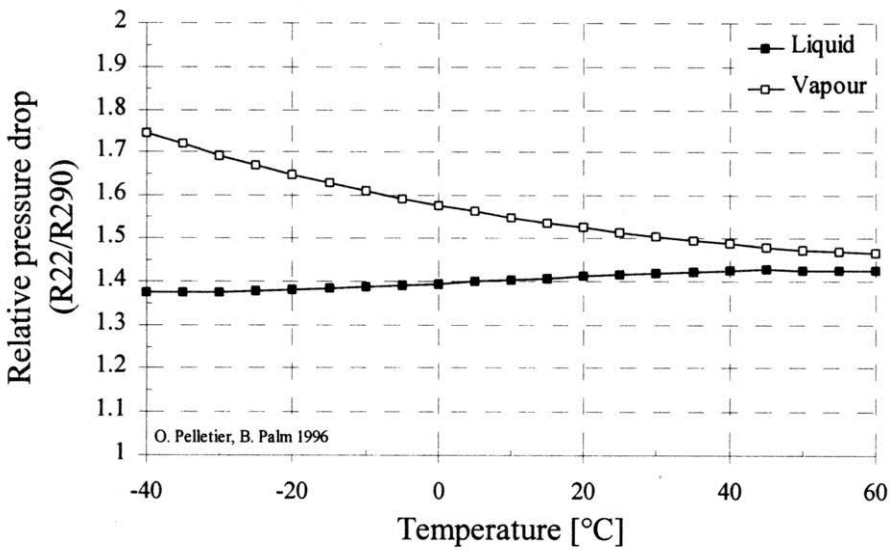

Figure 17 - Relative pressure drops of R22 and propane calculated from pressure drop equation for turbulent flow in circular tubes.

As shown, we may expect 30-35% higher pressure drop with R22 in the liquid phase and up to 70% higher pressure drop in the vapour phase at low temperatures. In the two phase flow of an evaporator or condenser the ratio may be expected to be in between that of liquid and vapour flow. These estimated differences are found to agree quite well with measurements (see following chapters).

4.3.2 Heat transfer in condenser

Heat transfer in condensation on smooth vertical surfaces is well described by the equations of Nusselt. According to these equations, the heat transfer coefficient can be calculated as:

$$\alpha = 0{,}924 \cdot k \cdot \left[\frac{r \cdot g \cdot \rho}{\upsilon \cdot L \cdot \frac{\dot{q}}{A}} \right]^{1/3} \qquad \text{Eq. 11}$$

As a figure of merit for condensation heat transfer we may thus use:

$$FOM_C = \left[\frac{k^3 \cdot r \cdot \rho}{\upsilon} \right]^{1/3} \qquad \text{Eq. 12}$$

In Figure 18, the ratio for the FOM_C for R22 and R290 is presented. The diagram shows that in the same condenser, at equal heat flux, R22 may be expected to give slightly higher (2-5%) heat transfer coefficients than propane.

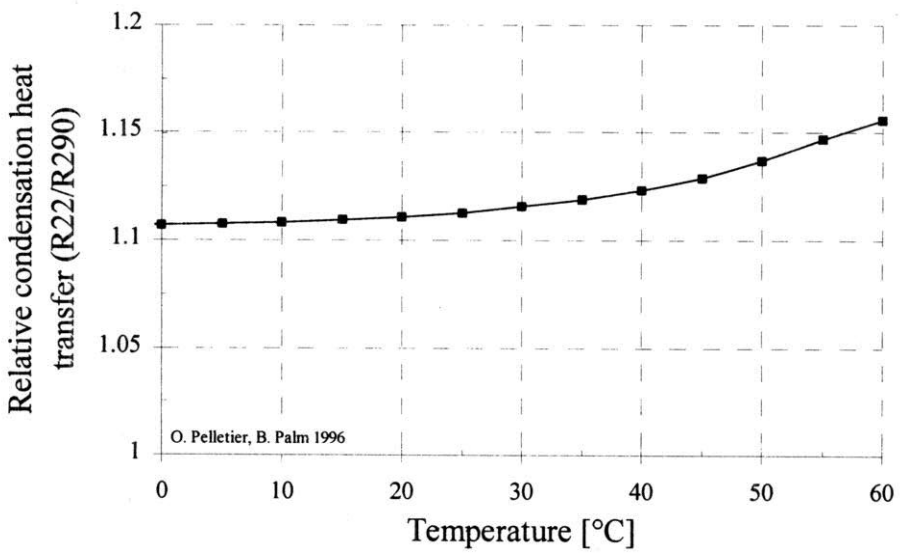

Figure 18 - Heat transfer in condensation for R22 relative to propane, calculated from Nusselt's equation.

4.3.3 *Heat transfer in turbulent one-phase flow*

In turbulent one-phase flow heat transfer is well represented by the Dittus-Boelter equation:

$$\alpha = 0{,}023 \cdot \mathrm{Re}^{0,8} \cdot \mathrm{Pr}^{0,4} \cdot \frac{\lambda}{d} \qquad \text{Eq. 13}$$

Using the definitions of Re and Pr, and approximating the mass flow as the ratio of the heating capacity of the system and the heat of vaporisation, this equation may be written as:

$$\alpha = 0{,}023 \cdot \left(\frac{4}{\pi}\right)^{0,8} \cdot \frac{\dot{Q}^{0,8}}{d^{1,8}} \cdot \frac{c_p^{0,4} \cdot k^{0,6}}{\mu^{0,4} \cdot r^{0,8}} \qquad \text{Eq. 14}$$

The last factor in this equation may be used as a figure of merit when comparing the two refrigerants. The ratio of these factors for R22 and propane are presented in Figure 19. As shown, the heat transfer of R22 is expected to be lower than of propane in both liquid and vapour phase. In a refrigeration system or a heat pump single phase heat transfer takes place only at the evaporator outlet and condenser inlet (and outlet, if liquid is subcooled). The single phase heat transfer may therefore seem to be of minor interest. However, in convective boiling the heat transfer can be expected to be related to that of single phase liquid flow, and the heat transfer ratio presented for liquid phase may thus indicate that convective boiling heat transfer coefficients are higher in propane than in R22.

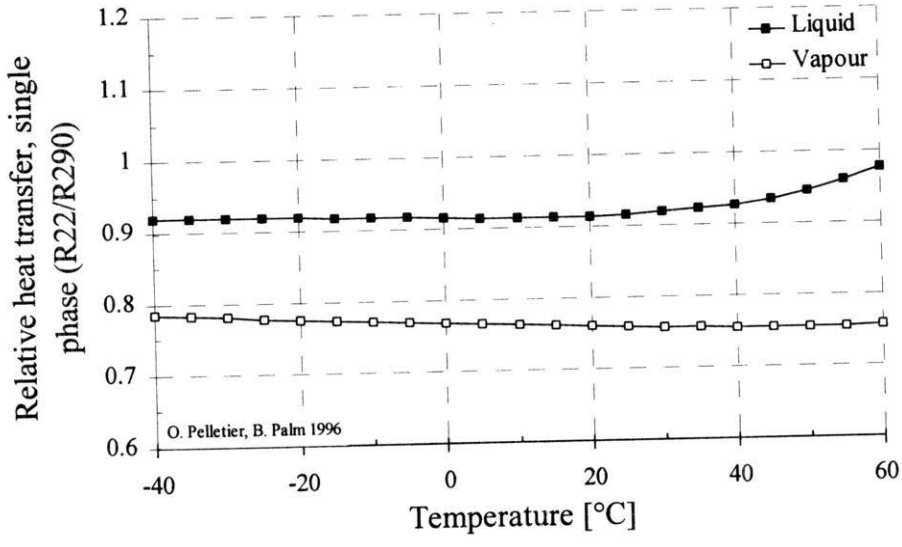

Figure 19 - Heat transfer for R22 relative to propane according to the Dittus-Boelter equation.

4.3.4 *Convective boiling*

No one correlation is generally accepted for flow boiling heat transfer. A simple correlation which has proved to give reasonable results in boiling of (H)CFC refrigerants in horizontal tubes has been proposed by Pierre. For complete evaporation, this equation can be expressed as:

$$\alpha = \frac{\lambda}{d} \cdot \mathrm{Re}^{0,8} \cdot K_f^{0,4} \qquad \text{Eq. 15}$$

where K_f is the Pierre boiling number defined as:

$$K_f = \frac{\Delta h}{g \cdot L} = \frac{\Delta x \cdot r}{g \cdot L} \qquad \text{Eq. 16}$$

and Δx is the change in vapour quality in the evaporator. Assuming $\Delta x = 1$ and the mass flux to be equal to the cooling capacity divided by the heat of vaporisation, the Pierre equation may be written as:

$$\alpha = 0{,}01 \cdot \left(\frac{4}{\pi}\right)^{0,8} \cdot \frac{1}{g^{0,4}} \cdot \frac{\dot{Q}^{0,8}}{d^{1,8} \cdot L^{0,4}} \cdot \frac{k}{r^{0,4} \cdot \mu^{0,8}} \qquad \text{Eq. 17}$$

The last factor of this equation may be used as a figure of merit for convective heat transfer. In Figure 20 the ratio of these factors for propane and R22 is plotted. It is clear that the Pierre equation, like the Dittus-Boelter equation, indicates that boiling heat transfer coefficients should be higher for propane than for R22.

The above analysis should be kept in mind when viewing the experimental results in the following chapter. It will be found that there is generally good agreement, at least qualitatively, between the experimental results and the simple analysis above.

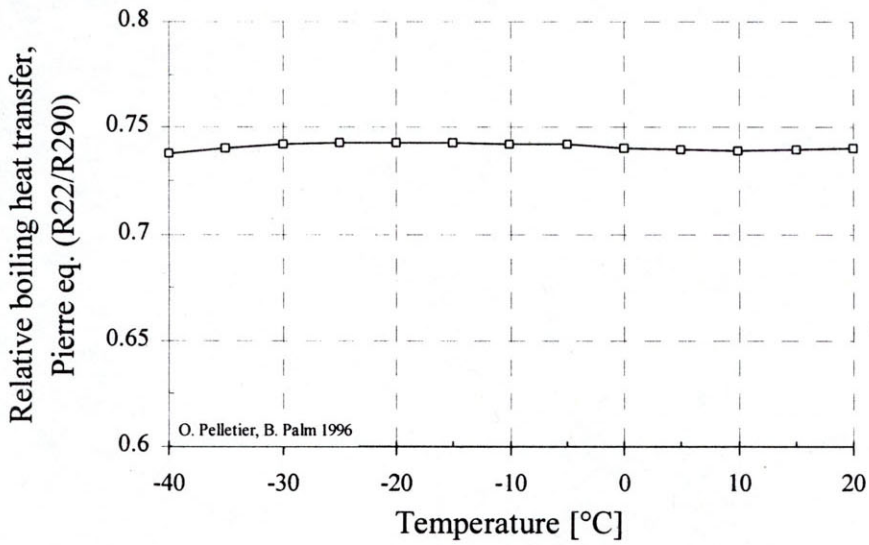

Figure 20 - Heat transfer for R22 relative to propane in convective boiling according to the Pierre equation.

5. EXPERIMENTAL APPARATUS

In order to verify some of these theories and the qualities of propane as a substitute of R-22 new experiments have been carried out at the Department of Applied Thermodynamic and Refrigeration, KTH, since a year.

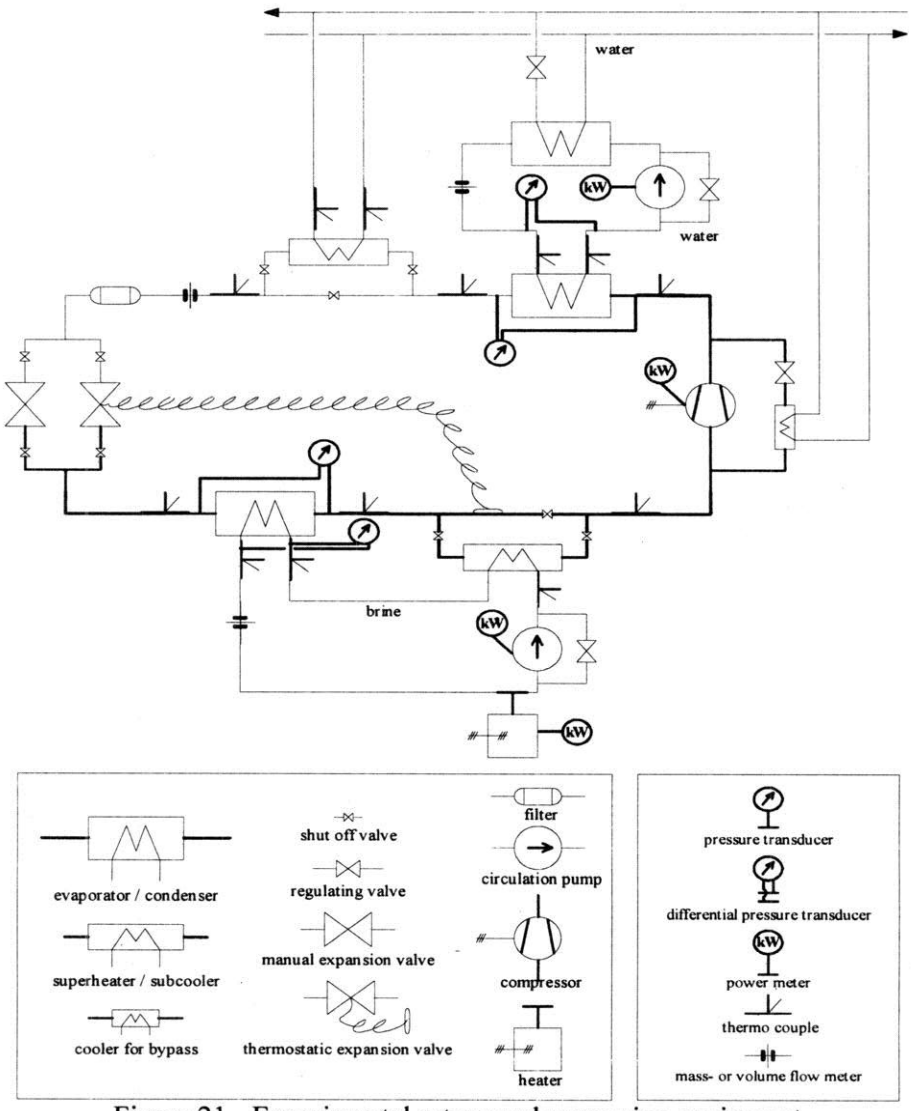

Figure 21 - Experimental set-up and measuring equipment

Figure 21 shows the schema of the experimental set-up and the measuring equipment. It simulates a small domestic heat pump with a total heat capacity of 3 kW. It is equipped with a hermetic compressor with a nominal power of 900 W and four brazed plate heat exchangers as condenser, subcooler, evaporator and superheater. The refrigerant is throttled by a thermostatic expansion valve. Most of the tubes and component connections are brazed. The refrigerant circuit is filled with 0,5 kg propane or 1 kg R22, since the specific volume is the double for propane.

Two water loops are connected to the condenser and the subcooler, and a brine loop is connected to the evaporator and the superheater. 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.

The cooling and the heating capacities of the set-up and operating conditions are controlled

both manually and electronically. All electric components are installed at least one meter from the heat pump. The heat pump is normally vented and equipped with two gas detectors. A switch isolating all electrical circuits other than computer, ventilation and gas detectors is installed.

6. MEASUREMENT RESULTS WITH COMPRESSOR

6.1 Measurement method

The compressor which was used in the experimental set-up was the subject of special tests. The results of the tests are summarised in order to compare propane with R22.

The compressor is a hermetic rolling piston from Mitsubishi. It was operated as it was designed originally, i.e. with R22 in the evaporating temperature range -10°C/+15°C. It has a displacement equal to 18,9 cm^3/rev. A partly-synthetic oil, Diamond MS 56, was used.

The tests with propane were performed without any major changes of the system. The mass flow was varied by changing the evaporation temperature. The setting of the expansion valve was changed to maintain a superheat of 5°C. The temperature of condensation was held constant at 35°C. The compressor was not insulated.

6.2 Total isentropic compressor efficiency

The total efficiency η_{tot} of the compressor can be calculated from the ratio:

$$\eta_{tot} = \frac{\dot{E}_{is}}{\dot{E}_{el}} = \frac{\dot{m}_r.(h_{is} - h_{2c})}{\dot{E}_{el}} \qquad \text{Eq. 18}$$

This efficiency includes all partial efficiencies, such as the electrical efficiency of the motor and the mechanical efficiency of the compressor. The electrical power consumption, $\dot{E}_{el}$, of the compressor is related to the power, $\dot{E}_{is}$, which would be needed in case of an ideal isentropic compression (i.e. no heat exchange between the refrigerant and the surroundings, and no working losses from mechanical or molecular friction).

Figure 22 shows the compressor's total isentropic efficiency as a function of the pressure ratio. Propane seems to have slightly higher efficiency, even though no significant differences can be observed.

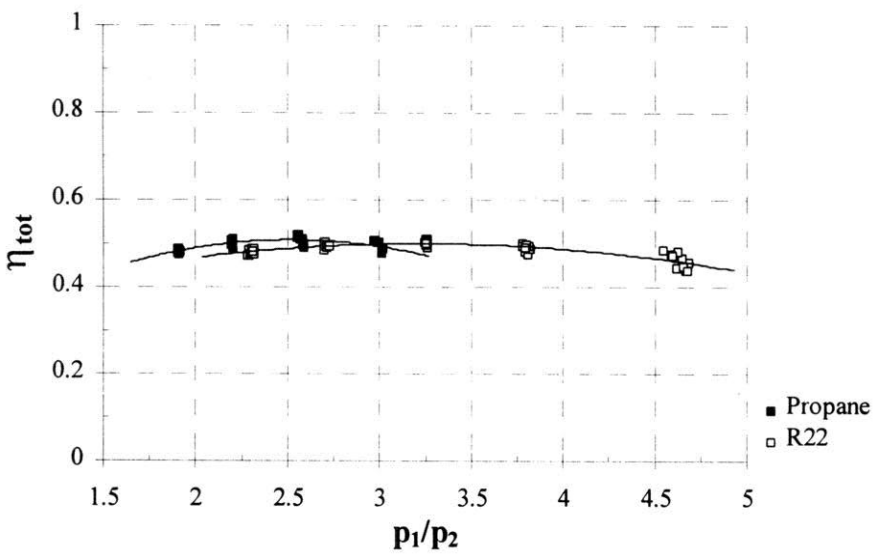

Figure 22 - Total isentropic efficiency versus pressure ratio

6.3 Volumetric compressor efficiency

The volumetric efficiency of the compressor has been calculated with the following equation:

$$\eta_{vol} = \frac{\dot{V}}{\dot{V}_{dis}} = \frac{\dot{m}_r . v_{2c}}{\dot{V}_{dis}} \qquad \text{Eq. 19}$$

This efficiency takes into account the displacement characteristic of the compressor.

As shown in Figure 23, propane seems to have slightly higher volumetric efficiency, even though the differences are small.

Propane is not only compatible with the compressor (originally designed for R22) but also offers as good characteristics as R22 with the advantage of producing a considerably lower discharge temperature.

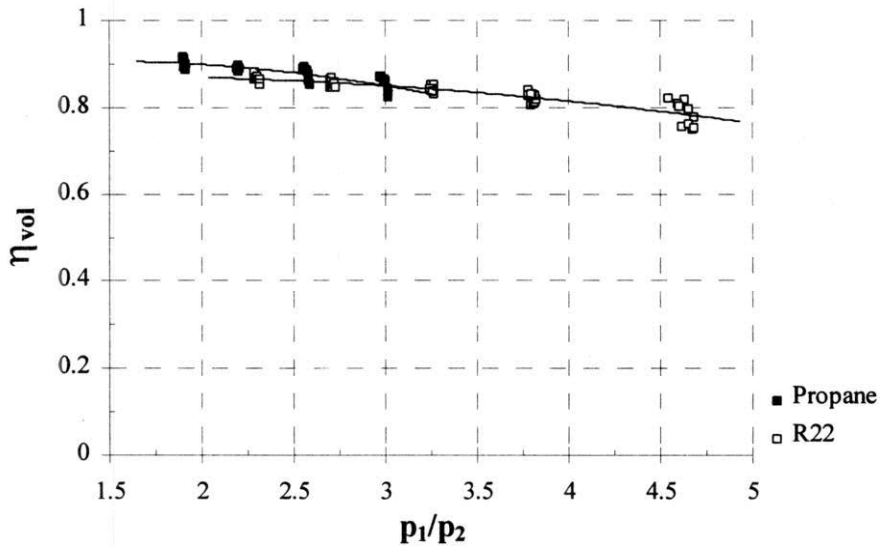

Figure 23 - Volumetric efficiency versus pressure ratio

6.4 Coefficient Of Performance

The heating capacity $\dot{Q}_1$ produced by the heat pump can be calculated as:

$$\dot{Q}_1 = \dot{v}_w . \rho_w . Cp_w . \Delta t_w = \dot{m}_r . \Delta h_r \qquad \text{Eq. 20}$$

In Figure 24 the heating capacity of the heat pump is plotted as a function of the evaporation temperature. R22 gives higher heating capacity than propane, about 200-400 W, which correspond to 7-10% higher value. This difference is much lower than expected from a theoretical calculation with an ideal cycle, which gives a heating capacity 15% higher for R22.

A first explanation for this difference can be that the pressure ratio for R22 is higher. This gives a compressor volumetric efficiency slightly lower, about 1-2% according to /9/, which decreases the heating capacity. This result also corresponds with that observed in Figure 23.

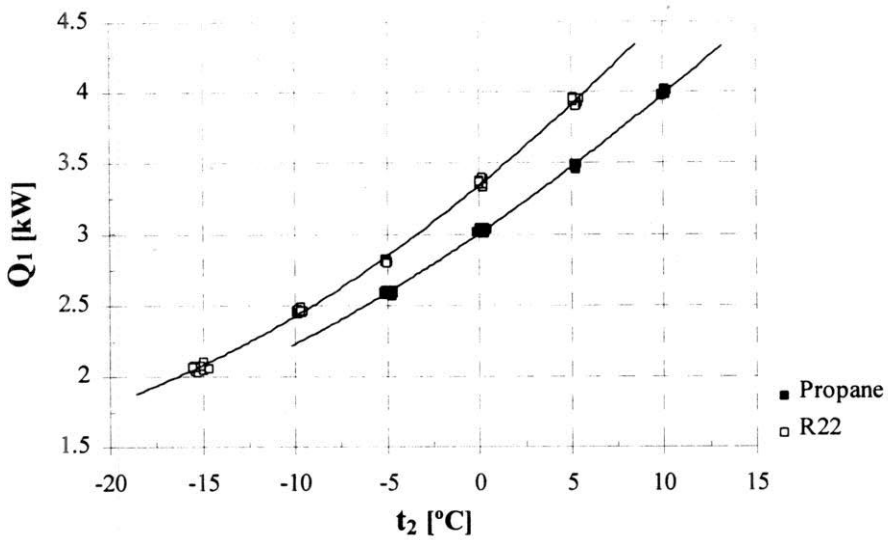

Figure 24 - Heating capacity versus evaporation temperature

From the heating capacity $\dot{Q}_1$ and the measured electrical power consumption of the compressor $\dot{E}_{el}$, the heating coefficient of performance has been calculated as follows:

$$COP_1 = \frac{\dot{Q}_1}{\dot{E}_{el}} \qquad \text{Eq. 21}$$

Figure 25 shows the COP_1 as a function of the evaporation temperature.

It is interesting to note that propane clearly gives a higher COP, about 4-5% higher value, although the theoretical calculation for an ideal cycle gives the same value, according to /8/ and to the previous chapters. This may indicate that the compressor losses are smaller with propane than with R22.

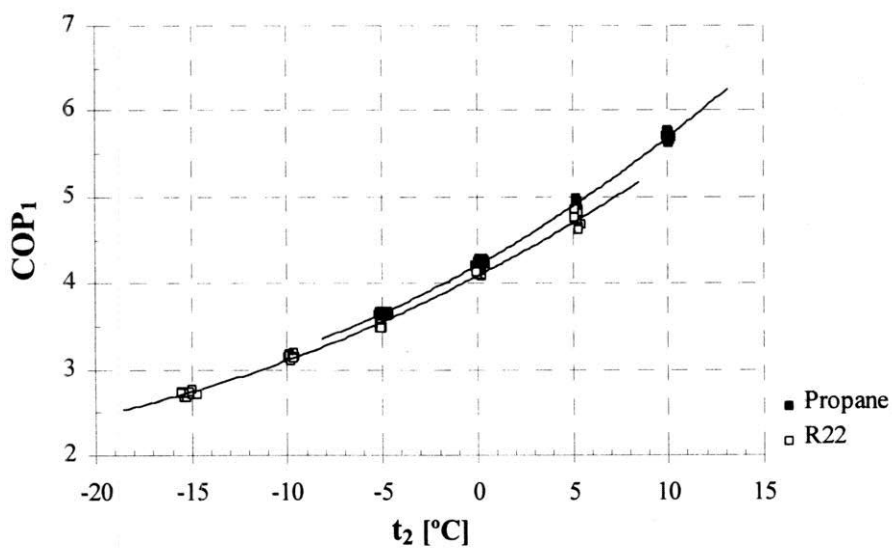

Figure 25 - Heating coefficient of performance versus evaporation temperature

7. MEASUREMENT RESULTS WITH PLATE HEAT EXCHANGERS

7.1 Measurement method

The condenser and the evaporator were operated as they were originally designed to be used with R22. No study has been performed on the subcooler and the superheater.

Both condenser and evaporator have been studied separately to investigate the heat transfer characteristics and the pressure drops. Two plate heat exchangers with different geometry have been tested. Here they will be referred to as model A and B.

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. As in the compressor tests, the replacement was performed without any major changes of the system.

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.

The setting of the expansion valve was adjusted to maintain a superheat of 4-6°C in all tests. The temperatures of condensation and evaporation were held constant at 35°C and 5°C, respectively. The flow rates of the secondary refrigerants were also held constant.

7.2 Calculations

A temperature difference between the hot and the cold side of the heat exchanger is necessary for the transfer of heat. To achieve the 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 fluids the logarithmic mean temperature difference is used, defined as:

$$\vartheta_{ln} = \frac{(\vartheta_{in} - \vartheta_{out})}{\ln(\vartheta_{in} / \vartheta_{out})} \qquad \text{Eq. 22}$$

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 phases.

The logarithmic mean temperature difference ϑ_{ln} was calculated according to Eq. 5, but using the temperature differences between the saturation temperature of the refrigerant and the temperature of the secondary refrigerant at the two ends of the heat exchangers, Figure 23.

Then, the overall heat transfer coefficient k was calculated as:

$$k = \frac{\dot{q}}{\vartheta_{ln}} \qquad \text{Eq. 23}$$

where $\dot{q}$ is the heat flux.

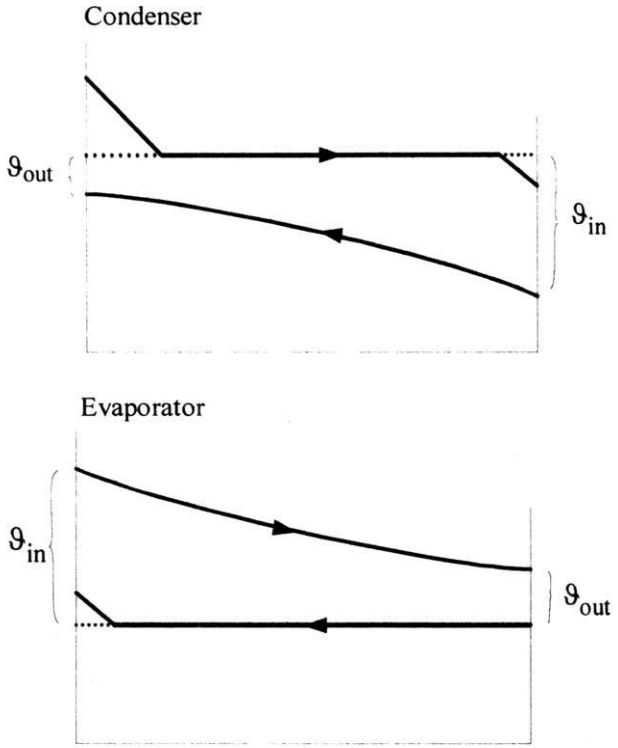

Figure 26 - Definition of the inlet and outlet temperature differences.

By keeping the conditions on the secondary refrigerant / water side the same in the R22 and the propane tests, one may conclude that any difference in the mean 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.

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

7.3 The condenser

Figure 27 shows the logarithmic mean temperature difference between the two fluids, ϑ_{ln}.

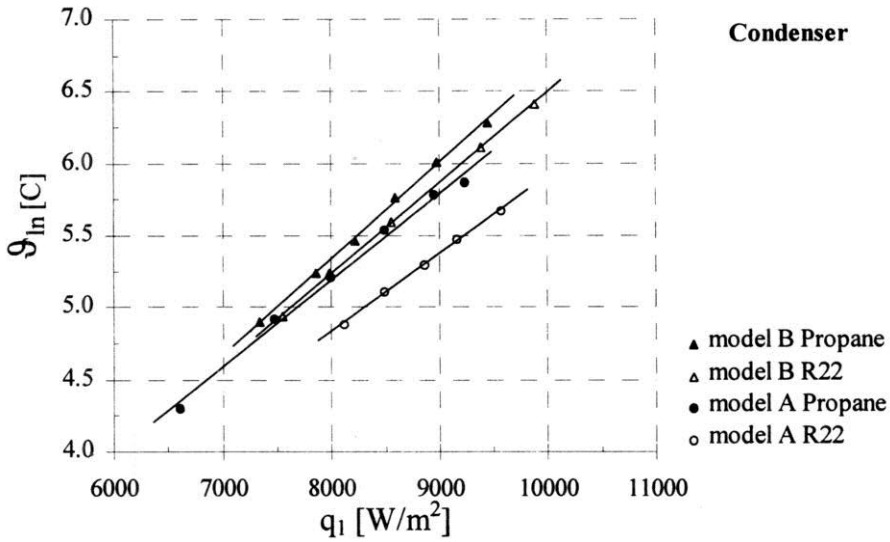

Figure 27 - Logarithmic mean temperature difference versus heat flux $\dot{q}_1$

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 with model A. With model B, the difference is insignificant and the temperature can be assumed as equal.

From the logarithmic mean temperature difference, ϑ_{ln}, and the heat flux, $\dot{q}_1$, the overall heat transfer coefficient, k, may be calculated. These values are plotted in Figure 28.

R22 has better heat transfer characteristics in the condenser, with an overall heat transfer coefficient on the average 9% higher than propane for model A and 4% higher for model B. The difference can be assumed constant and independent of the heat flux.

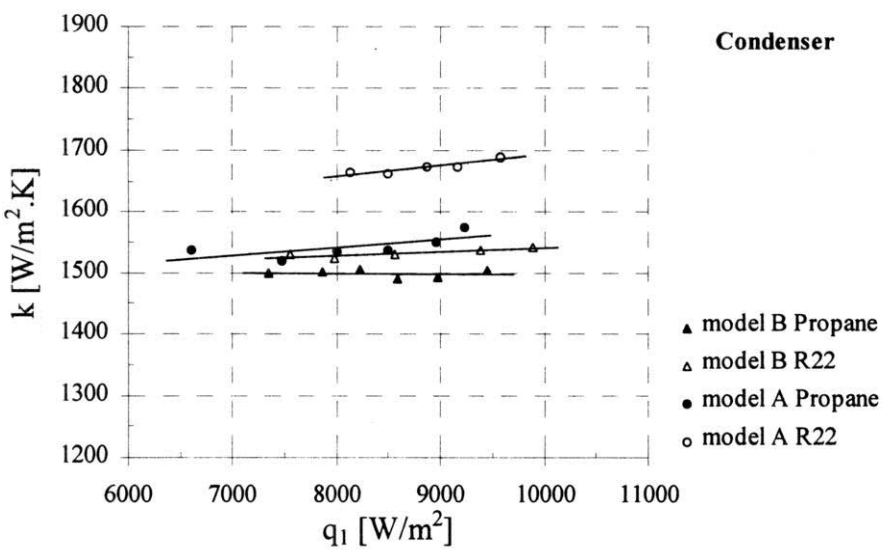

Figure 28 - Overall heat transfer coefficient versus heat flux $\dot{q}_1$

In Figure 29, the pressure drops in the condenser with the two fluids are compared. It can be seen that propane gives a considerably lower pressure drop. Model A gives higher pressure drop for both refrigerants. The difference between R22 and propane is the same for both models.

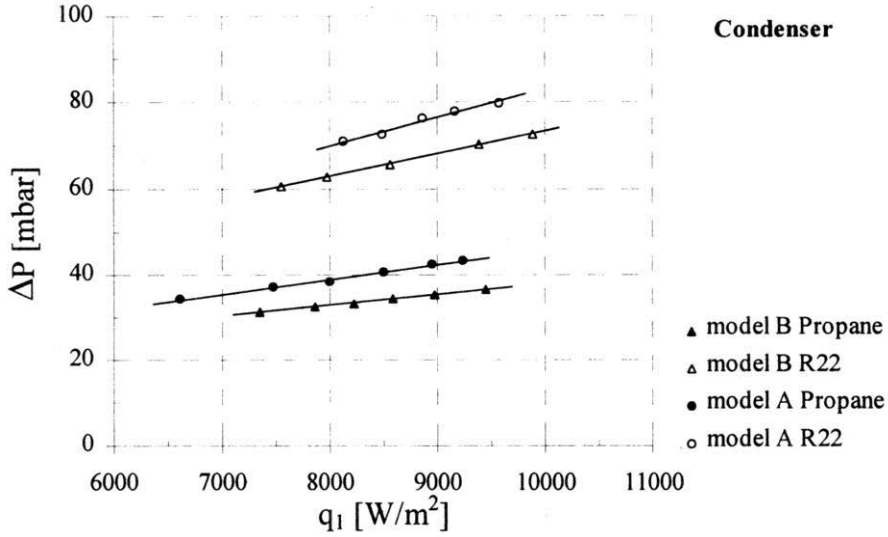

Figure 29 - Pressure drop versus heat flux $\dot{q}_1$

7.4 The evaporator

The evaluation process used for the condenser has also been used for the evaporator.

Figure 30 shows the logarithmic mean temperature difference. It can be seen that, contrary to the case with the condenser, R22 needs a higher temperature difference than propane to transfer a certain capacity. The differences between the fluids are small, half a degree to one degree. The difference increases with $\dot{q}_2$.

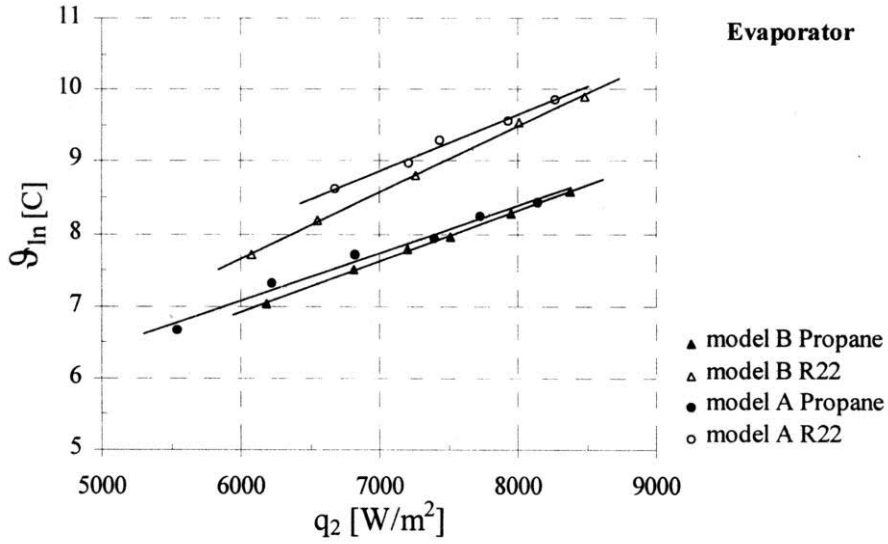

Figure 30 - Logarithmic mean temperature difference versus heat flux $\dot{q}_2$

In Figure 31 the overall heat transfer coefficient in the evaporator is shown for the two refrigerants. The heat transfer coefficient can be considered equal for model A and B. It is in this case about 10% to 16% higher with propane. This difference increases rapidly with $\dot{q}_2$.

Note that with propane the heat transfer coefficient seem to be more dependent on the heat flux than with R22.

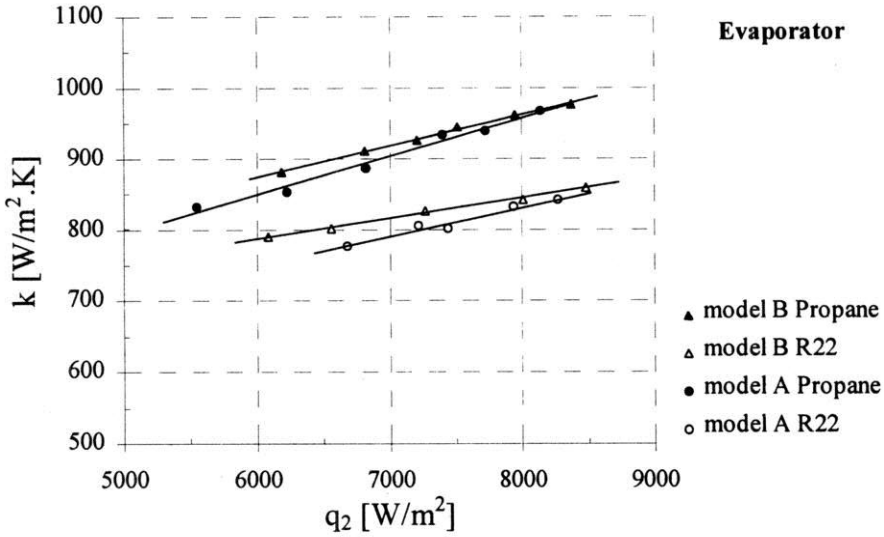

Figure 31 - Overall heat transfer coefficient versus heat flux $\dot{q}_2$

The comparison of the pressure drop characteristics, Figure 32, shows that propane gives considerably lower pressure drop in the evaporator, just as was the case with the condenser.

The difference between the pressure drop values of propane and R22 is constant, independent of the model of plate heat exchanger or heat flux.

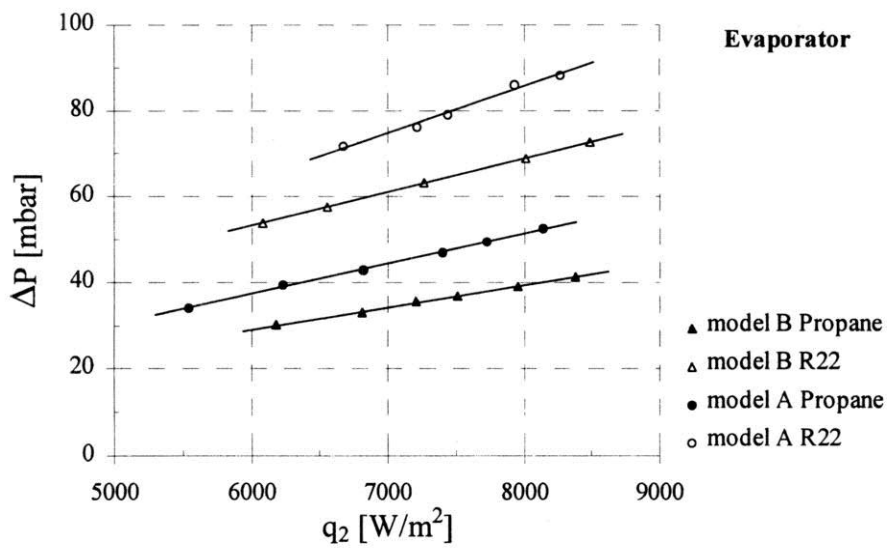

Figure 32 - Pressure drop versus heat flux $\dot{q}_2$

7.5 Heat Transfer Coefficient

Figure 33 shows the plot of the refrigerant heat transfer coefficient α as a function of $\dot{q}_2$, for the evaporator model A.

This α-value was obtained by determining the heat transfer coefficient on the secondary refrigerant side by applying the so called "Wilson plot method". By keeping the conditions on the refrigerant side the same and varying the flow rate on the secondary refrigerant side, one may obtain an expression for the heat transfer coefficient on the secondary refrigerant side as a function of the flow rate only.

The refrigerant heat transfer coefficient α is then calculated by the equation of the heat transfer resistance through an exchanger, expressed as:

$$\frac{1}{k} = \frac{1}{\alpha_1} + \frac{\delta}{\lambda} + \frac{1}{\alpha_2} \qquad \text{Eq. 24}$$

together with the overall heat transfer coefficient, k, found in the diagram above. As the heat transfer resistance in the wall is small, this term may be omitted.

The heat transfer coefficient then obtained is plotted in Figure 33 as a function of the heat flux $\dot{q}_2$. The α-value is about 30% to 33% higher for propane. For both fluids, the heat transfer coefficients increase rapidly with $\dot{q}_2$. This strong dependency could indicate a major influence of nucleate boiling. This hypotheses will thus be a subject for further work.

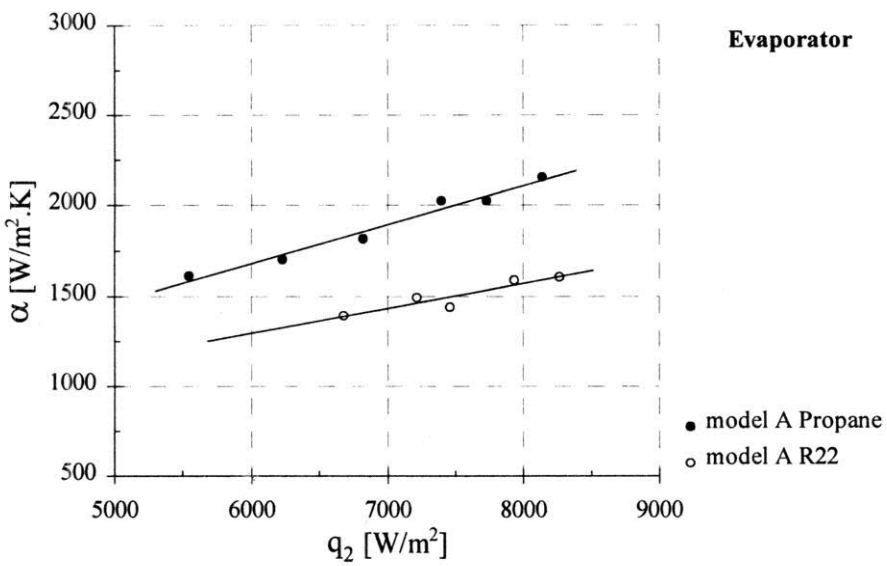

Figure 33 - Heat transfer coefficient versus heat flux $\dot{q}_2$

8. CONCLUSION

To sum up, the results show that propane in the tested plate heat exchangers, at any given heat flux, gives overall heat transfer coefficients slightly lower than those of R22 in the condenser, but slightly higher in the 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.

The compressor tests show that the compressor, designed for R22, could be used with propane without problems, and that the volumetric and total efficiencies were about the same. With propane, the heating capacity was 7-10% lower, while the heating coefficient of performance was 4-5% higher than with R22.

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