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**TEST ON A TWIN-SCREW COMPRESSOR :  
COMPARISON BETWEEN TWO COOLING MODES**

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**ABSTRACT**

An open twin-screw compressor supporting two distinct cooling modes (by external oil cooling or liquid injection into the compressor) has been tested with HCFC 22 and an alkylbenzene lubricant. The test procedure was as specified in the ISO 917 standard (testing of refrigerant compressors). Using both cooling methods, compressor efficiencies for evaporation from - 35 °C to - 5 °C, condensation from 30 °C to 47 °C, and discharge superheat of 35 K are determined. To conclude the study, compressor cooling needs for different values of discharge superheat and of compressor capacities are analysed.

**INTRODUCTION**

When designing or implementing compressors using HCFC 22, full allowance must be made for the phenomenon of refrigerant heating during compression, especially under harsh operating conditions. In many cases, the compressor will require cooling. Rather than cooling the injected lubricant, which is the conventional method for cooling screw compressors, it might be economically advantageous to inject liquid HCFC 22.

To compare the impact of cooling methods on compressor efficiencies and cooling needs, both cooling modes were tested on the same twin-screw compressor, having a swept volume of 294 m<sup>3</sup>.h<sup>-1</sup>.

**TEST PROCEDURE AND OPERATING CONDITIONS**

The compressor was implemented on a test installation consisting of a dry-expansion refrigerating system. The refrigerant HCFC 22 was used with an ISO 100 grade alkylbenzene lubricant.

To accurately determine compressor efficiencies and ensure results reliability, ISO 917 procedure (testing of refrigerant compressors) was applied to the existing installation and the compressor under test. The preliminary study of ISO 917 standard revealed the following points :

- reference test conditions are compressor suction pressure, discharge pressure, suction temperature and shaft speed, with tolerances for each values specified. For coolable compressors, the ISO 917 procedure is not sufficient to define a unique set of test conditions : for a full definition, discharge temperature or superheating is also to specify.

In the present work, we determined compressor efficiencies under refrigerating system conditions (see table 1), at nominal compressor capacity, with discharge superheating of 35 K. We assessed compressor

cooling needs at 50 %, 75 % and 100 % compressor capacity, for discharge superheating values between 20 and 40 K (discrete points only).

		Suction pressure			
		1.3 bar (- 35 °C)	2 bar (- 25 °C)	3 bar (- 14.6 °C)	4 bar (- 6.5 °C)
Discharge pressure	11.9 bar (30 °C)	⊗		⊗	
	13.5 bar (35 °C)		x	x	
	16.5 bar (43 °C)	⊗c	x	⊗c	x
	18 bar (46.7 °C)		x	x	⊗cv

○ cooled lubricant injection  
 x HCFC 22 liquid injection  
 c variable capacity  
 v variable discharge superheating  
 suction superheat : 10 K

Table 1 : Test conditions

- Basically, the ISO 917 procedure involves determining the refrigerant mass flow rate through the compressor, which conditions calculation of the other values (volumetric and isentropic efficiencies, refrigerating power and COP). In order to determine this mass flow rate, two distinct methods are necessary ; in our case, flow rates of liquid at expansion valve inlet and of vapour at the discharge of the compressor should not differ by more than 4 %. Both of these values are taken to determine test mass flow rate (average of the two methods).
- This values must be multiplied by a corrective factor to bring the test point into line with the reference conditions. Here, we have rectified certain inconsistencies in the ISO 917 standard.
- Efficiencies and accuracy errors were calculated in accordance with ISO 917 definitions. Isentropic efficiency is defined as the ratio between isentropic compression power and compressor mechanical power demand. This definition was adopted in all cases, noting that there is no specific reference to compressor cooling in ISO 917.
- ISO 917 makes no allowance for the fact that, in liquid injection mode, part of the discharge refrigerant flow rate is drawn off for cooling the compressor. To allow for this, we compare the volumic flow rate of liquid HCFC 22 directed to the expansion valve with the flow delivered by the compressor, subtracting the flow of liquid HCFC 22 injected during compression, measured using a volumic flow meter.

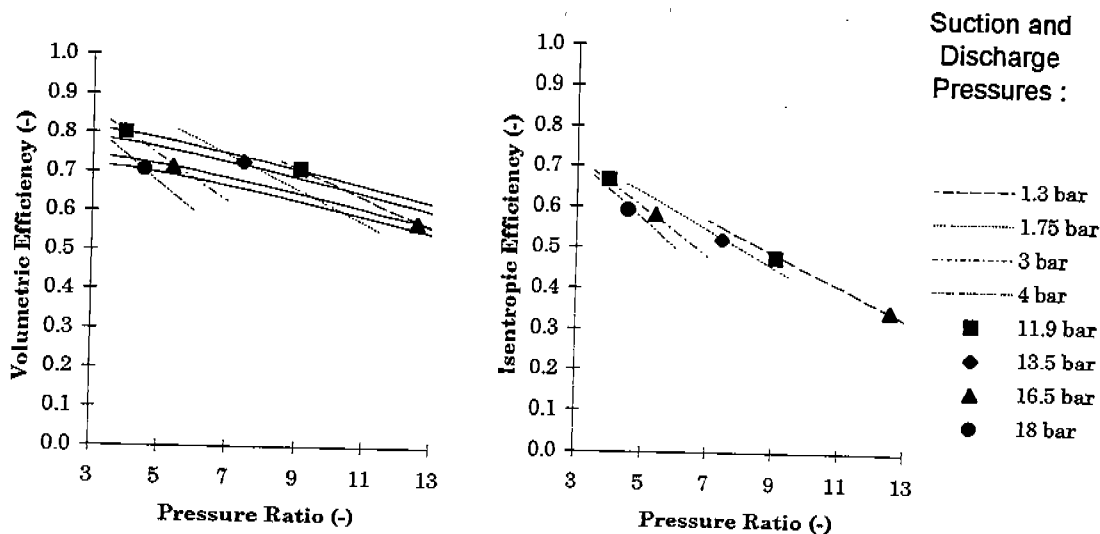
### COMPRESSOR EFFICIENCIES IN LUBRICANT COOLING MODE

For compressor standard configuration (external lubricant cooling and nominal capacity), tests covered the five points shown in table 1, plus the point given by suction/discharge pressures of 1.75/13.5 bar. Figures 1 and 2 summarise results for volumetric efficiency (preferred to mass flowrate) and isentropic efficiency.

The volumetric efficiency curve is typical of that found with fixed  $V_i$  screw compressors. For pressure ratios from 3 to 13, volumetric efficiency varies widely, from 0.8 to 0.57. Volumetric efficiency is highest at pressure ratios approaching manufacturers specified  $V_i$ , though there is no maxima value within the studied range. The rate of decrease in volumetric efficiency with pressure ratio is constant at all discharge pressures (2.5 % decrease in efficiency per point increase in pressure ratio). At any given suction pressure, we observe a steeper decline in efficiency (5 % to 8 %) which is as more important as the suction pressure increases. At fixed pressure ratio, an increase of 1 bar in suction pressure produces an 11.7 % drop in volumetric efficiency, whereas an increase of 1 bar in discharge pressure produces a 1.5 % drop in volumetric efficiency.

A wide variation in isentropic efficiency with pressure ratio is also observed : from 0.67 to 0.35. However the curves for constant discharge pressures are very close (less than 0.05 deviation between curves for 11.9 bar and 18 bar). The variation in isentropic efficiency with pressure ratio is more

pronounced at constant suction pressure (slope from 0.04 to 0.06) than at constant discharge pressure (slope 0.03).



Figures 1 and 2: Volumetric and Isentropic Efficiencies versus pressure ratio, in lubricant cooling mode

### IMPACT OF HCFC 22 LIQUID INJECTION ON COMPRESSOR BEHAVIOUR

On this compressor, cooling by liquid injection is only usually applied for ammonia. We therefore extended tests using HCFC 22 and this cooling mode to ten points.

As compressor cooling is for this method performed by drawing off part of the liquid refrigerant (i.e. part of the discharge flow rate) from the HP receiver, for reinjection into the compressor, it is interesting to analyse the HCFC 22 flow rates throughout the system : the refrigerating power only depends on HCFC 22 mass flow rate, whereas the mechanical power depends on the sum of the suction and injected flow rates (since the injected fluid must be compressed from the intermediary pressure to the discharge pressure).

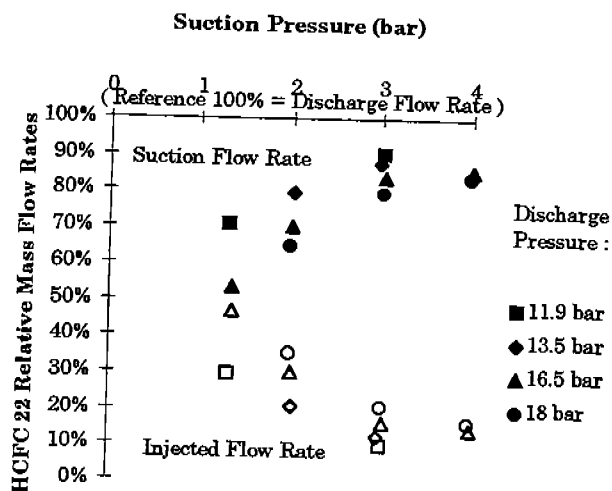
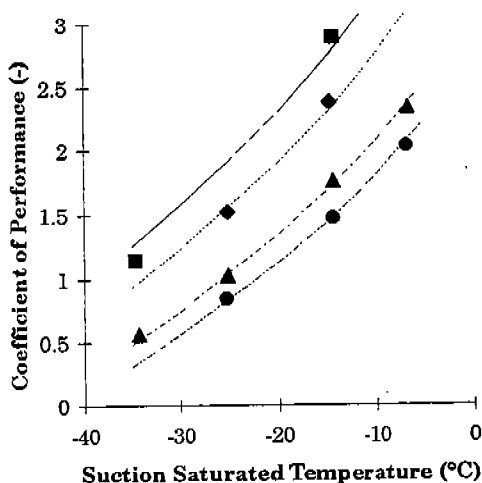
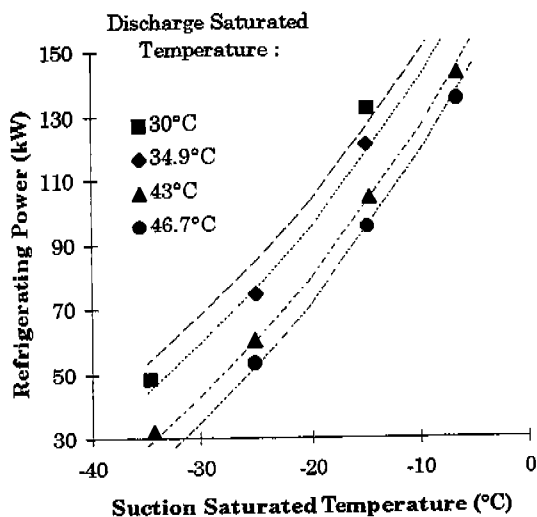


Figure 3 shows the part of HCFC 22 discharge flow rate used for refrigeration and the part used for compressor cooling. At low suction pressures, the part of discharge flow rate used for compressor cooling (for a constant discharge superheating of 35 K) rises with high discharge pressure up to 47 %. At high suction pressures (i.e. lower pressure ratios) 80 % to 90 % of the discharge flow rate is used for refrigeration.

Figure 3: HCFC 22 Mass Flow Rates in the test loop

We then examine how compressor efficiencies are affected by the different mass flow rates associated with this cooling mode. Figures 4 and 5 show refrigerating COP and refrigerating power curves for liquid injection mode. Tables 2 and 3 compare refrigerating power and COP between both cooling modes, at five identical points.



Figures 4 and 5 : Refrigerating Power and COP versus suction saturated temperature, for liquid injection cooling mode

		Suction Pressure		
		1.3 bar	3 bar	4 bar
Discharge pressure	11.9 bar	92 %	97 %	
	16.5 bar	84 %	92 %	
	18 bar			96 %

Table 2 : Refrigerating Power compared to results obtained for lubricant cooling mode (reference 100 %)

		Suction Pressure		
		1.3 bar	3 bar	4 bar
Discharge pressure	11.9 bar	88 %	93 %	
	16.5 bar	79 %	89 %	
	18 bar			92 %

Table 3 : Refrigerating COP compared to results obtained for lubricant cooling mode (reference 100 %)

Under identical compressor operating conditions, as gas suction density and enthalpies are the same for both cooling modes, the refrigerating power is only affected by the difference in suction flow rates, i.e. a decrease from 8 % to 16 % (3 % to 8 %) for an evaporation temperature of - 35 °C (- 15 °C) and condensation temperatures from 30 to 43 °C. The drop in refrigerating power only becomes significant when suction and discharge saturated temperatures start to diverge.

As for coefficient of performance, it depends directly on the ratio between suction mass flow rate and mechanical power. Under worst conditions (high pressure ratio or large difference between saturating temperatures), we observe a 21 % drop in compressor performance with respect to the COP obtained using the usual cooling mode, but only a 6 % drop in the most favourable conditions. The drop in COP is higher than drop in refrigerating power because of the increase in mechanical power (3 to 5 %) when liquid injection cooling is considered.

### COMPRESSOR COOLING NEEDS

The ISO 917 test procedures fails to specify whether or not the compressor is cooled and the test report specified by ISO 917 makes no mention of parameters relative to compressor cooling needs. However, the complete study of the compressor is also based on the analyse of these terms.

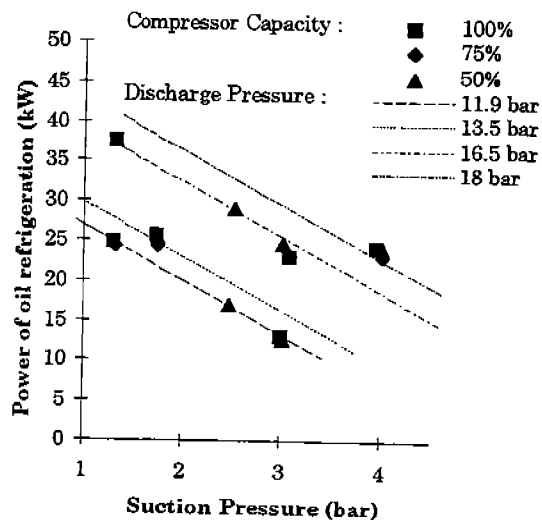
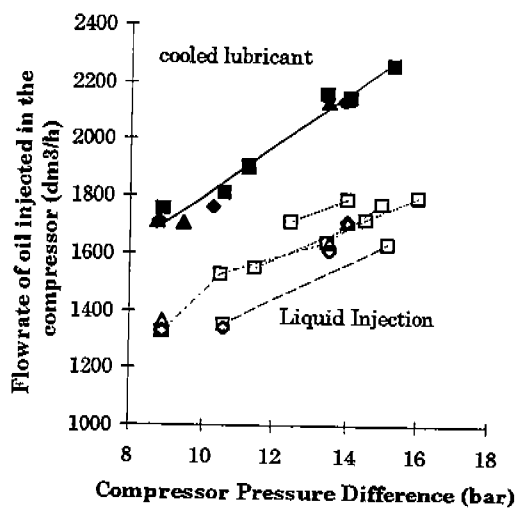
So the following parameters are examined : temperature and flow rate of lubricant injected into the compressor (all functions included) ; power required for cooling the compressed gases, calculated at the water side of the external water/lubricant exchanger (for lubricant cooling method only) ; flow rate of refrigerant injected during compression and injection pressure (for HCFC 22 liquid injection method only).

- Figure 6 shows the oil injection flow rate versus pressure difference at compressor inlet and outlet, for both cooling modes.

In lubricant cooling mode, the injected oil flow rate varies linearly with the difference between suction and discharge pressures.

In liquid injection mode, the points on the graph show  $\pm 100 \text{ dm}^3/\text{h}$  dispersion around an average value. Lubricant flow rates are 15 % to 25 % below the values found previously and, within the compressor range under study, depend primarily on the discharge pressure. This difference in injected oil flow rate behaviour between both cooling modes may be attributable to the pressure difference between discharge and lubricant injection orifices, which is responsible for circulating the lubricant.

In both cooling modes, the flow rate of lubricant injected into the compressor remains constant, when discharge superheating or compressor capacity varies. This confirms the previous conclusions, given the lubricant flow rate as only dependant on pressure difference.



Figures 6 and 7 : Characterisation of compressor cooling needs for cooled lubricant mode

- The power required for compressor cooling is examined, for constant discharge superheat of 35 K, by reference to figure 7, which shows the usual behaviour cooling power with suction and discharge pressures : compression cooling need (and thus cooling power) drops as the temperatures of the heat sink and cold source get closer together (i.e. best-case conditions for compressor). The power increase required for compressor cooling is estimated at +6.9 kW per bar decrease in suction pressure, and +2.7 kW per bar increase in discharge pressure. The ratio between cooling power and mechanical power at nominal capacity varies from 30 % (for lowest  $\Delta T$  between sources : 45 K) to 70 % (for highest  $\Delta T$  between sources : 79 K).

Thanks to tests at 50 % and 75 % capacity, it can be concluded that cooling compressor power level is independent of the compressor capacity stage : under the same suction and discharge conditions, compressor cooling needs remain unchanged regardless of compressor power.

Tests also confirmed that the compressor cooling power varies with discharge superheating. In lubricant cooling mode at the 4/18 bar point, the compressor cooling power drops from 30 kW to 24 kW as discharge superheating goes from 25 K to 35 K, though the mechanical power required for compressing the HCFC 22 remains the same. The magnitude of heat exchanges between lubricant and refrigerant during compression thus appears to have no effect on the mechanical power, at least over the parameter range covered by our tests. However, this observation would need checking for the case of zero lubricant cooling, i.e. uncontrolled discharge superheating.

- For a discharge pressure of 11.9 bar, a suction pressure from 1.3 to 3 bar and a discharge superheat of 35 K, oil temperatures vary from 35 °C to 46 °C at the injection orifices. For a discharge pressure of 16.5 bar, temperatures ranged from 42.2 °C to 52.4 °C. The highest lubricant injection temperature was observed at the 4 bar/18 point. The injection temperature is important as it determines the viscosity of the lubricant/refrigerant mixture injected into the compressor (to ensure lubrication, leaktightness between screws, etc...). For a comparison, in liquid injection mode, the injection temperature varies between 58 °C and 77 °C and only depends on the discharge temperature (i.e. it is only affected by losses in the main lubrication circuit). These temperatures are on average 6 K lower than the discharge temperature. In this last case, owing to the levels of viscosity ranged at the high end of the temperature range, viscosity may become a limiting factor for use of liquid injection cooling mode at discharge pressure greater than 16 bar.
- In liquid injection mode, the flow rate of the refrigerant injected into the compressor (parameter analysed above) and the intermediary injection pressure, used for sizing the valve of the liquid injection circuit (controlled by means of a bulb at the compressor discharge), were analysed. At nominal capacity, HCFC 22 injected flow rate ranged from 300 to 730 kg/h, for pressure difference (discharge minus intermediary pressures) from 7 to 12.5 bar. Though the proportion of mechanical power responsible for compressing the HCFC 22 coming from compressor suction is lower than in external cooling mode, the additional flow of fluid to be conveyed from intermediary pressure to discharge pressure is responsible for an observed increase of 3 % to 5 % in mechanical power.

The intermediary injection pressure varies from 3 to 5.8 bar (corresponding to saturating temperatures of - 16 °C to 5 °C), depending on the compressor cooling needs and thus on the pressure difference at the compressor inlet and outlet.

## CONCLUSION

The efficiencies difference obtained by implementing liquid injection cooling on a compressor that is usually cooled by the lubricant have been analysed. Significant deviations between these two cooling methods, and divergence increased as operating conditions worsened are observed.

The results of our investigation should prove useful when selecting compressors cooled by liquid injection. Liquid-injection-cooled compressors have lower performance than lubricant-cooled compressors of the same swept volume, which means they will need a higher swept volume rating to operate under the same evaporation temperature and refrigerating power conditions. However, this performance penalty may be compensated by lower investment and running costs for certain items (simplified auxiliary circuits, no external exchanger, no cooling water). The long-term profitability of adopting this lower-performance method should be examined case by case.

## REFERENCES

ISO 917 standard, « Testing of refrigerant compressors », 1989, second edition.