1988

Influence of Oil Injection and Pressure

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INFLUENCE OF OIL INJECTION AND PRESSURE RATIO ON SINGLE SCREW PERFORMANCES AT HIGH TEMPERATURE

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1 - INTRODUCTION

Single screw compressors are well known and their performances in refrigerating plants or in heat pump at low temperature are good enough. For industrial applications, the heat pumps must work at a higher temperature (over 60°C and up to 120°C). For these temperatures, special refrigerants have to be used, e.g. R114 or R142b [1]. In this paper we try to analyze the behaviour of this kind of screw compressor when used at high temperature (condensing temperature of 97°C) with R142b. We varied technical data of the compressor, such as oil flows and experimental conditions such as suction and discharge pressures, and we measure their influences on the isentropic and volumetric efficiencies of the compressor.

2 - TEST INSTALLATIONS

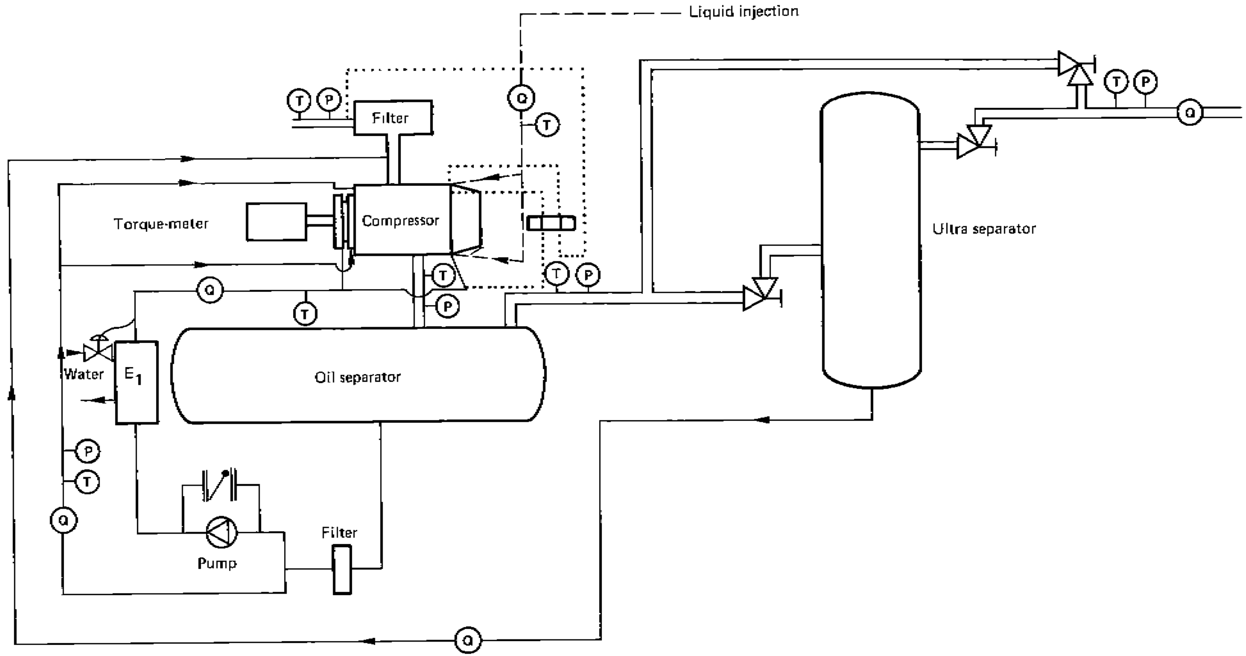
For these experiments, we used a mono screw compressor which was implanted on our experimental heat pump plant already described in [2].

The principal characteristics of the compressor are as follows :

- mono-screw HALL HS 18,
- swept volume : 311 m^3/h at 3000 r.p.m.,
- oil injection between 1.5 and 2.5 m^3/h,
- volume ratio : 2.6,
- OILON stars,
- refrigerant used : R142b.

As we can see on figure 1, the compressor unit includes a lot of gauges to measure temperatures, pressures at different locations of the plant, flow rates of oil and refrigerant. We can also connect a test-rig at the oil supply to determine the miscibility of refrigerant in oil and the viscosity of the homogeneous phase.

The monoscrew compressor, like anykind of screw compressor, needs a lot of lubrication. Most of this oil is injected inside the chamber for the sealing and the cooling, a small part is used for the lubrication of the bearings. These two main functions are very different and need separated oil lines. The first one, for lubrication, crosses an oil cooler in order that the temperature Orof the oil decreases down to 60°C. The other one for sealing and cooling is directly injected into the compressor. As we want to analyze the influence of



Oil { .... Capacity control supply
Oil { ——— Supply for lubrication and injection

Refrigerant { — — — Liquid injection
Refrigerant { ═══ Gas

Q Flow-meter

T Temperature gauge

P Pressure gauge

E_1 Water cooled oil cooler

Figure 1 — Test plant.

oil injection on the performances of this compressor, we inserted a manual valve on each oil line and so we have the possibility of modifying the flow rate of oil in each line independantly.

3 - PRESENTATION OF EXPERIMENTAL RESULTS

In order to analyze the influence of oil injection flow rates we determined the values for each experimental results for :

- The volumetric efficiency (η_v) : ratio of the flow rate of refrigerant at the suction point on the swept volume of the screw compressor.
- The total isentropic efficiency defined as the ratio of the theoritical power absorbed by the refrigerant for an isentropic efficiency and the shaft power measured for the real compression (η_{ist}).
- The coefficient of performance : ratio of the power obtained at the cold source on the power measured on the shaft of the compressor (C.O.P.).

4 - INFLUENCE OF OIL INJECTION ON PERFORMANCES

In this part of the experimentation we fixed the pressure ratio and independantly varied the flow rate of the oil for bearings between 0.1 and 0.6 m^3/h and the flow rate of the oil for sealing and cooling between 0 and 3 m^3/h, for two different values of the pressure ratios : 3.24 and 4,6 which gave respectively for evaporating and condensing temperature of the heat pump : 30°C/76°C for 3.24 and 31°C/94°C for 4.6.

4.1 - Influence of the flow rate of sealing oil

The following figure (figure 2) shows the evolution of the volumetric efficiency versus the sealing-oil flow rate for two different pressure ratios : 3.24 and 4.6 and for two values of the bearings oil flow rate : 0.125 m^3/h and 0.4 m^3/h.

We can see that this last parameter does not affect the value of η_v. But we notice that this efficiency increases continuously as does the oil sealing flow rate. We made a one-hour test without oil sealing flow rate : no incident was observed but the volumetric efficiency has a weak value.

On figure 3 we plotted the isentropic efficiency versus the oil sealing flow rate for the same experimental values. It is important to notice that over 0.7 m^3/h for P.R = 3,24 end over 1.5 m^3/h for p.r = 4.6 any increase of oil sealing flow rate has no effect on the isentropic efficiency. There are two contradictary phenomenas with the increase of oil flow rate : firstly an improvement of the volumetric efficiency ; secondly an increase of the energy spent by the compressor to convey this oil flow rate. These two phenomenas are balanced. These critical values of the oil sealing flow rate are certainly dependant on the clearances inside the compressor.

4.2 - Influence of oil bearing flow rate

The figures 4 (η_v vs oil bearings flow rate) and 5 (η_{ist} vs oil bearings flow rate) show the bad influence of the increase of oil bearings flow rate on the efficiencies. This oil, after the lubrication and the cooling of the bearings is injected for one part to the suction point of the compressor. A part of the refrigerant contained in this oil, is liberated (flash) and limits the quantity of refrigerant that the compressor could draw up. So the volumetric efficiency decreases when oil flow increases. Morever this oil, with a low temperature (60°C) cools the vapor of the refrigerant during the compression and decreases the enthalpy of discharged gas. In this way the isentropic efficiency is decreased. For an experimental point we calculated the power lost (PL) due to the oil bearings flow rate.

$$P_L = \frac{\rho_m \, C\rho_m \, D_o \, \Delta T}{3600}$$

with ρ_m : volumic mass of the oil injected (kg/m^3)

C_{pm} : specific heat of oil injected (kJ/kg/°C)

D_o : oil bearings flow rate (m^3/h)

ΔT : temperature difference between discharge point and injected oil (°C).

For Δ_T = 50°C and D_o = 0,2 m^3/h we obtained P_L = 4,9 kW approximatively 7 % of the power absorbed by the compressor. These results show that it is important to choose the smaller flow rate of the bearings oil recommended by the manufacturer to keep working conditions as good as possible.

5 - INFLUENCE OF PRESSURE RATIO ON EFFICIENCIES

In this part of the experimentation, as we wanted to analyze the influence of the pressure ratio on efficiencies, we regulated the oil flow rates with manual valves on oil tubes, in order to obtain 0,1 m^3/h for the bearings oil flow rate and successively 0,7 m^3/h (named weak flow rate) and 1,4 m^3/h (named strong flow rate) for the sealing oil flow rate. All these flow rates were regulated under a pressure ratio of 2,7 close to the optimal value corresponding to the value of the volume ratio (2,6).

For experiments at higher pressure ratio (3,24 and 4,6), we let the flow rates of oil vary according to the pressure difference between the suction and the discharge points.

5.1 - Fixed suction pressure

Figure 6 shows the variations of volumetric and isentropic efficiencies according to the value of pressure ratio for a fixed suction pressure equal to 4 bars (32°C). We notice that the volumetric efficiency is close to 78 % for a pressure ratio varying between 2.5 and 3 (according to the volume ratio of the compressor of 2.6).

Over 3 the decrease of η_v is important : 64 % for a p.r of 5. We can explain this evolution, higher the pressure ratio is the higher the temperature and the pressure are, and the greater the dilution of refrigerant in the oil is. The viscosity of the oil decreases and the internal leakages increase consequently. We can also consider that for a strong flow rate the volumetric efficiency is better than 2 %.

A quite similar evolution can be observed for the isentropic efficiency with a maximal value of 60 % at a pressure ratio of 2,8 (*), but no effect of the increase of oil sealing flow rate is noticed.

Figure 7 shows the influence of the degradation of the compressor efficiency on the COP of the heat pump. The experimental values of the COP vary from 0.5 to 0.4 time the value of the theoritical COP based on the Carnot cycle.

5.2 - Fixed discharge pressure

Compared to the previous results, we noticed a weaker decrease of the efficiencies of the compressor showed on figure 8. Two explanations can be given :

- When the pressure ratio increases for a fixed discharge pressure, the discharge temperature increases. This improves the viscosity of the oil (less refrigerant in the oil due to a better superheat). This better viscosity largely compensates the internal leakage due to the increase of pressure ratio. The decrease of η_{is} is less important than on figure 6. If the effect of non-adaptability between the internal volume ratio and the external pressure ratio is the same, that means the improvement of the viscosity of the oil has a positive influence on the internal thightness.

- The differences of pressures, during this second part of the tests are less important. In fact the pressure difference is a more significant parameter than the pressure ratio for internal leakage for this kind of compressor.

For this part of the tests, we find the same evolution efficiencies :

- An increase of 2 % for η_v when we used a strong oil sealing flow rate (not observed for η_{ist}).

- A decrease of the COP from 0.5 to 0.4 times the Carnot COP with the increase of the pressure ratio (figure 9).

6 - CONCLUSION

These tests, which lasted more than 1000 hours, showed the importance of the value of oil flows on the efficiencies of the compressor, especially for a high temperature heat pump. We also noticed the important decrease of efficiencies when the pressure ratio does not correspond to the volume ratio of the compressor.

Chemical analysis of the refrigerant and the oil after tests did not revealed any deterioration of their properties.

Further tests will be made on the same compressor with new materials for stars (small thermal dilatation) and with more viscous lubricant.

(*) A part of the decrease of η_{is} when the pressure ratio grows is due to the supplementary work induced by the non-adaptation of the internal volume ratio (2.6) and the external pressure ratio (η_p to 5).

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[2] BLAISE J.C. - DUTTO T. - Some practical results obtained with non azeotropic mixtures of refrigerants in high temperature heat pump - 1986 - IIR - Commission E2 5.8 August Purdue USA.

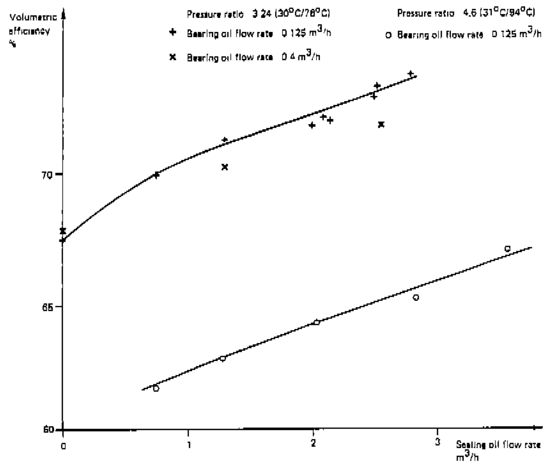

Figure 2 — Influence of sealing oil flow on volumetric efficiency.

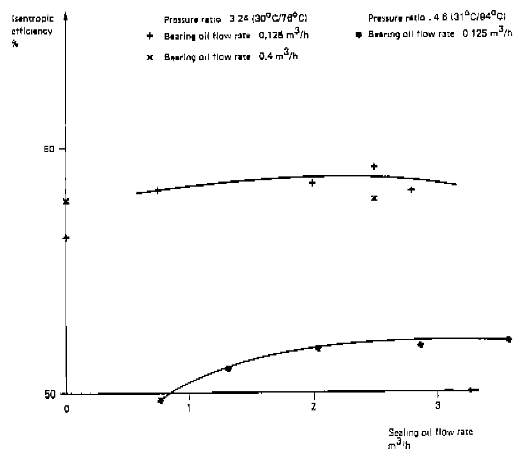

Figure 3 — Influence of sealing oil flow on isentropic efficiency.

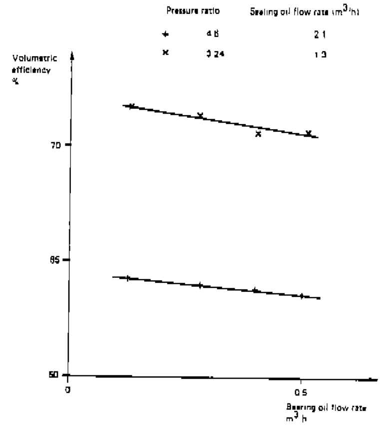

Figure 4 — Influence of bearing oil flow on volumetric efficiency.

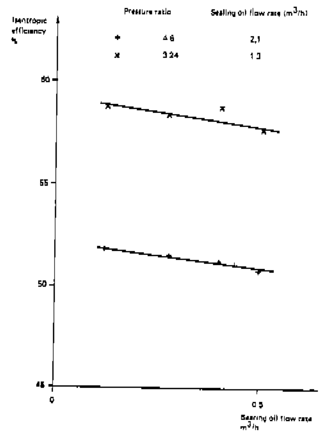

Figure 5 — Influence of bearing oil flow on isentropic efficiency.

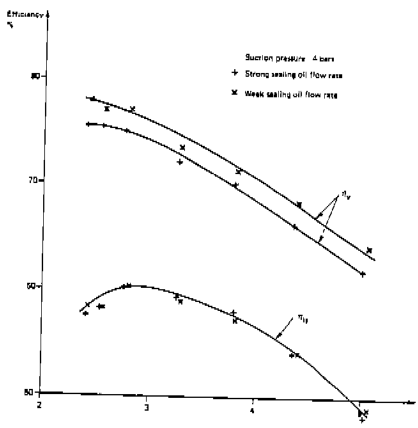

Figure 6 — Influence of pressure ratio on efficiencies at constant suction pressure.

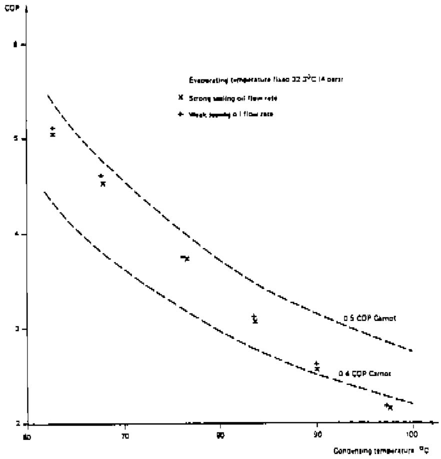

Figure 7 — Influence of the pressure ratio on COP for constant suction pressure.

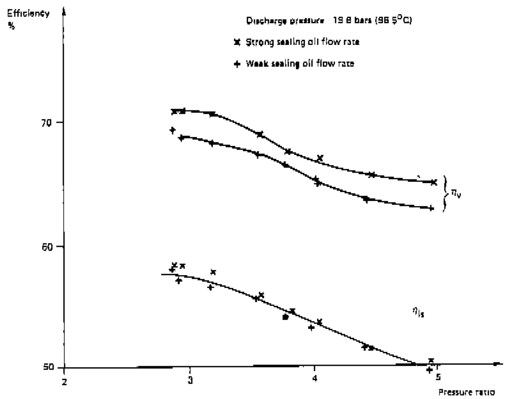

Figure 8 — Influence of the pressure ratio on efficiencies at constant discharge pressure.

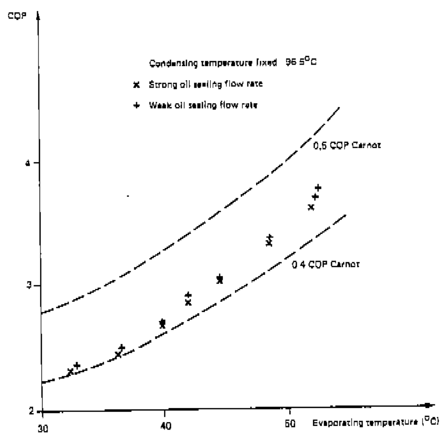

Figure 9 — Influence of the pressure ratio on COP for constant discharge pressure.