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EXPERIMENTAL ANALYSIS OF AN INVERTER-DRIVEN SCROLL COMPRESSOR WITH LIQUID INJECTION

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ABSTRACT

An air-to-water heat pump with R22 as the working fluid was equipped with an inverter and a hermetic scroll compressor. Liquid refrigerant was injected into the suction pipe to decrease the high discharge temperatures and thereby extend the operating range.

This paper emphasizes the characteristic of the Inverter Compressor Combination at different suction vapor qualities and speeds. The leakage across radial and axial clearances is investigated theoretically considering the motion, the pressure differential and the mutual solubility of refrigerant and oil.

It was observed in an experimental analysis that the isentropic compressor efficiency and the mass flow efficiency decrease substantially with decreasing suction vapor quality. Only the energetic efficiency, accounting for the compressor shell losses, improves due to the lower shell temperatures.

NOMENCLATURE

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$g$ gap width

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INTRODUCTION

Continuous capacity control by varying the compressor speed with an inverter, a technique that has been successfully applied to heat pump air conditioners in the U.S. and in Japan, is applied to an air-to-water heat pump with R22 as the working fluid. A problem many compressor manufacturers are faced with in changing from R12 to R22 is the high discharge temperature occurring at some operating conditions. Defects of the electric motor and decomposition of the refrigerant can result.

More efficient compressor cooling yielding lower discharge temperatures can be achieved by staging with intercooler, injection of cooling oil or liquid refrigerant. The latter method, i.e. injection of liquid refrigerant into the suction pipe, can be applied since most scroll compressors are in some sense compliant allowing a two-phase suction state with a vapor quality of down to 80 % mass.

THEORETICAL BACKGROUND

There are few theoretical investigations about the wet compression process. One group of authors [Ozaki 1990] studied the optimum liquid injection condition depending on working fluid, suction wetness and injection pressure ratio from the theoretical point of cycle analysis. They showed that for an idealized compression process the COP improvement at lower vapor qualities increases for working fluids with high latent heat. Although no COP increase was attainable for wet compression with R22, the superheat of the discharge vapor could be reduced very effectively. Other authors concluded for wet compression in a reciprocating compressor that flashing of liquid refrigerant droplets during the suction process lowers the effective suction flow rate [Reischelt 1974] and that the cyclic dissolution of refrigerant in the lubricating oil is of importance [Weinberg 1951].

There remains the question of how a real compressor behaves during a wet compression process. To achieve a better understanding in this field the leakage to the suction chamber is investigated theoretically by describing the various leakage mechanisms and their influence on the effective suction volume, in comparison to overall measurements. Furthermore some characteristic key values of inverter and scroll compressor are compared for different speeds and suction vapor qualities in an experimental analysis.

LOSES IN INVERTER AND SCROLL COMPRESSOR

The mass flow efficiency

\[ \eta_{\text{me}} = \frac{\dot{m}_{\text{in}}}{\dot{m}_{\text{th}}} = \frac{m_{\text{in}}}{\rho_{\text{g}} v d N_{\text{rot}}} \]  

accounts for thermal and volumetric effects reducing the ideal mass flow rate.

The total power \( P_{\text{el}} \) fed into the inverter is only partly transferred to the refrigerant, because of the inverter losses (expressed through the inverter efficiency \( \eta_{\text{inv}} \)) and of the shell heat loss to the surroundings (expressed through the energetic efficiency \( \eta_{\text{en}} \)):

\[ \eta_{\text{inv}} = \frac{P_{\text{el}}-P_{\text{el}}}{P_{\text{el}}} \]

\[ \eta_{\text{en}} = \frac{P_{\text{en}}}{P_{\text{el}}-P_{\text{el}}} = \frac{m_{\text{en}} (h_d-h_s)}{P_{\text{el}}-P_{\text{el}}} \]

Thus

\[ M_{\text{re}} (h_d-h_s) = \eta_{\text{en}} \eta_{\text{inv}} P_{\text{el}} \]

The energy transferred to the refrigerant can be compared to an ideal one, which may be conveniently defined as the isentropic work input necessary for the same pressure ratio supposing that all the mass flow is at suction vapor state \( \nu \). In this case the isentropic power is

\[ M_{\text{re}} (h_{d-is}-h_s) = M_{\text{re}} \frac{k \rho_s}{k-1} \left( \frac{P_{\text{el}}}{P_{\text{el}}} \right)^{\frac{k-1}{k}} \]

where a mean value for \( k \) is taken (it depends in reality on the temperature).

The comparison of (4) and (5) allows the definition of an isentropic work factor \( (1 + \zeta_{\text{wa}}) \), which corresponds to the inverse of an isentropic efficiency \( \eta_{\text{ws}} \)

\[ 1 + \zeta_{\text{wa}} = \frac{1}{\eta_{\text{ws}}} = \frac{h_d-h_s}{h_{d-is}-h_s} \]

\( \zeta_{\text{wa}} \) deviates from the value 0 for different reasons:

(a) The normal dissipation occurring during compression and passage through suction and discharge openings, including also motor and bearing losses and taking into account the shell losses. These effects are expressed by \( \zeta_{\text{ws}} \).

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(b) The influence of leakages, expressed by \( \zeta_w \).

(c) The influence of eventual wet compression, expressed by \( \zeta_e \).

As the losses according to (a) are common in every compressor, we shall not discuss it here anymore.

**Influence of Leakages**

Leakages have a double effect, on one side on the mass flow efficiency, because the leakage flowing back into the suction chamber reduces the net mass flow rate \( M_{ne} \) (it will be discussed in the part LEAKAGE MODEL), on the other hand on the work required for the compression, because the initial temperature \( T_s \) is higher by \( \Delta T_s \) (giving rise to an increase of the compression work) and because of the part of the compression effort invested in the leakage mass flow:

\[
P_{\text{comp}} = [M_{ne} \Delta h_i (1+\zeta_e) + M_{leak} \Delta h_i (1+\zeta_e) C_{leak}] \left[ 1 + \frac{\Delta T_s}{T_s} \right]
\]

(7)

where \( C_{leak} \) takes into account the average relative compression effected before the leakage occurs; \( C_{leak} \) is between 0 and 1.

The increase of the suction temperature is

\[
\Delta T_s = \frac{C_{leak} \Delta h_i (1+\zeta_e) M_{leak}}{(M_{ne} + M_{leak}) C_p}
\]

(8)

Therefore equation (7) becomes

\[
P_{\text{comp}} = [M_{ne} \Delta h_i (1+\zeta_e) + M_{leak} \Delta h_i (1+\zeta_e) C_{leak}] \left[ 1 + \frac{C_{leak} \Delta h_i (1+\zeta_e) M_{leak}}{M_{ne} + M_{leak}} \right]
\]

(9)

as the leakage is a relatively small mass flow, it is possible to simplify (9) as follows

\[
P_{\text{comp}} = M_{ne} \Delta h_i (1+\zeta_e) \left[ 1 + (C_{leak} \sigma) \right]
\]

(10)

with \( \sigma \) being the relative leakage flow \( \sigma = M_{leak} / M_{ne} \)

and \( C_{leak} \) being approximately

\[
C_{leak} = C_{leak} + C_{leak} \frac{1}{1+\sigma} \frac{\Delta h_i (1+\zeta_e)}{c_p T_s} = C_{leak} \left[ 1 + \frac{\Delta h_i (1+\zeta_e)}{c_p T_s} \right] = C_{leak} \left[ 1 + \frac{P_s k - 1}{P_s k - 1} (1+\zeta_e) \right]
\]

(12)

Denoting

\[
C_{leak} \sigma = \zeta_b
\]

(13)

we have therefore

\[
P_{\text{comp}} = M_{ne} \Delta h_i (1+\zeta_e) (1+\zeta_b)
\]

(14)

**LEAKAGE MODEL**

**Where Does Leakage Occur?**

Several authors [Tojo 1986; Caillat 1988] have described the main leakage paths in a scroll compressor, i.e. leakage due to the radial clearance between the flanks of the wraps and due to the axial clearance between the tip of the wraps and the end plate.

A problem is to define the appropriate clearances. For the following calculations values from a scroll deformation analysis [Suefuji 1988] of 30 \( \mu \)m for the radial clearance and 10 \( \mu \)m for the axial clearance have been taken. Since the radial clearance is three times the axial clearance flank leakage becomes dominant. An average pressure difference of 1.5 \( 10^5 \) Pa between first compression and suction chamber was used for the leakage flow calculations resulting from the pressure differential.

The leakage fluid is assumed to be fully liquid (centrifugal effects cause a liquid layer accumulating on the walls of the scroll wraps) and is treated as a homogeneous isotropic refrigerant / oil solution that is released by rolling, transported by dragging and by the pressure differential to the suction chamber, where flashing of the dissolved refrigerant takes place. The solution properties are taken for equilibrium conditions.
Mechanisms for Leakage

- **Rolling (only flank leakage)**
  The leakage volume flow due to rolling is proportional to \( \delta \), \( W \), \( N_{rot} \) and \( F_r \), with the gap clearance and the gap width \( \delta \) and \( W \), the rotor speed \( N_{rot} \) and the rolling factor \( F_r \):
  \[
  F_r = \frac{\text{rolling distance}}{\text{sliding distance}} = \frac{R_s}{\delta} \phi \cdot 2
  \]
  with \( \phi \) as involute wrap angle of the outer contact point.

- **Dragging**
  The leakage volume flow due to dragging is proportional to \( \delta \), \( W \) and \( \frac{1}{N_{rot}} \), assuming a laminar flow of a Newton Fluid.

- **Pressure Differential**
  The leakage volume flow due to the pressure differential is proportional to \( \delta \), \( W \), \( \Delta p \) and \( 1/\mu \), where \( \Delta p \) is the pressure difference to the following compression chamber and \( 1/\mu \) is the reciprocal of the dynamic viscosity (Hagen-Poiseuille Flow).
  The average pressure difference between suction chamber and isentropically compressed first compression chamber is used for calculating the leakage resulting from the pressure differential between adjacent chambers, as viscous flow in a wide rectangular channel.
  The gap geometry of a wide rectangular channel with Hagen-Poiseuille flow can be expressed by a factor \( F_{b-p} \):
  \[
  F_{b-p} = \frac{B^2}{12 L} \cdot \frac{1}{\delta} \cdot n(\phi)
  \]
  for flank leakage with \( n(\phi) \) as \( L/\delta \), where \( \delta \) is the minimum clearance and the length \( L \) is taken to the point where the clearance doubles.

- **Flashing**
  The leakage volume increase due to flashing is proportional to \( v''/v' \), \( \Delta \gamma_e \) and \( V_{leak} \), where \( v''/v' \) accounts for the ratio of vapor to liquid volume, \( \Delta \gamma_e \) for the relative amount of flashed refrigerant and \( V_{leak} \) for the liquid leakage volume flowing back into the suction chamber. The flashing factor \( F_f \) gives the volume increase due to the flashed refrigerant.
  \[
  F_f = \frac{\Delta \gamma_e \cdot v''}{v}
  \]

Leakage Volume Flow

The fluid flow to the suction chamber can be calculated by superposing several linear terms accounting for the total leakage flow rate:
\[
V_{leak} = A_C \cdot F_T \left[ 2 \pi \left( \frac{1}{2} + F_r \right) N_{rot} + F_{b-p} \Delta p/\mu \right]
\]
where \( F_T \) is the total leakage factor.
yielding a reduction in the suction volume

$$\Delta V_s = \frac{V_{s,s}}{V_{d,s}} N_{rot}$$

From (19) the reduction in mass flow efficiency is calculated as

$$\Delta \eta_{m} = \frac{V_s - \Delta V_s}{V_s} \frac{T_s}{T_s + \Delta T_s}$$

**Solubility Characteristic of the Refrigerant / Oil Solution**

The solubility of the refrigerant / oil solution influences the viscosity and the flashing. Fig. 1 shows the *kinematic viscosity* of the refrigerant / oil solution as a function of temperature and refrigerant concentration. Isobars of 3 bar and 5 bar and, at the lower left side, the area of *partial miscibility* between R22 and oil are depicted.

**Fig. 1: Viscosity of a 55 eSt naphthenic mineral oil / R22 solution**

It has to be noted that the *kinematic viscosity* of the solution can be 3 - 4 orders of magnitude lower than for pure oil if the temperature approaches the saturation condition of the liquid refrigerant, as occurs during wet compression.

Integrating the rolling factor of (15) over the involute wrap angle during the suction process gives an average rolling factor of 6.5 for the type of scroll compressor measured. Fig. 2 shows the reduction of the suction volume for different refrigerant concentrations resulting from liquid leakage. The volume reduction due to *dragging* and *rolling* remains constant for all speeds since the liquid leakage and the theoretical refrigerant flow rate are proportional. Leakage due to the *pressure differential* appears only at high refrigerant concentrations where the viscosity dependent Hagen-Poiseuille term becomes relevant. Its influence on the effective suction volume increases with decreasing speed. It is interesting that the leakage due to *rolling* becomes more important at lower refrigerant concentrations for the suction process than the other leakage sources.

**Fig. 2: Calculated reduction of the effective suction volume due to liquid leakage**
Fig. 3 shows the large reduction in suction volume due to the backflashing leakage. It is assumed that the refrigerant flashes at suction pressure. The vapor quality after flashing is calculated by applying an energy balance to the liquid leakage assuming that the liquid is throttled adiabatically. On the lower half of fig. 3 it is shown that the vapor quality increases with decreasing refrigerant concentrations. This effect results from the decreasing amount of refrigerant and the high heating capacity of the remaining oil.

**THE INFLUENCE OF LIQUID INJECTION**

Liquid injection influences the net mass flow rate:

- the suction temperature approaches the saturation temperature, i.e., superheating disappears, giving rise to an increase in mass flow
- the leakage flow, which in oil injected scroll compressors consists mainly of oil, has, due to liquid injection, an increased refrigerant content in the oil. The leakage flowing back into the suction chamber shows therefore a pronounced flashing of the dissolved refrigerant due to throttling from the high pressure in the first compression pocket to the low pressure in the suction chamber. This effect contributes to a partial filling of the suction chamber and therefore leads to a reduction in mass flow.

The experiments (see also fig. 6-7) showed that the measured range of reduction in mass flow corresponds well with the calculations. The circumstance that the measured reduction becomes smaller at high speed seems to indicate that the amount of flashing is limited at high speed by the heat flux to the flashing refrigerant.

**Liquid Injection Influences also the Power**

Equation (14) extended by the factor \(1 + \zeta_b\) gives

\[
P_{\text{comp}} = M_{\text{te}} \Delta h_{\text{in}} (1+\zeta_b) (1+\zeta_b) (1+\zeta_b)
\]

(21)

- due to the presence of liquid droplets, the temperature rise (8) will not take place.

Therefore for the leakage influence we have

\[
C_{\text{leak}} = C_{\text{leak}} \quad \text{instead of (12)}
\]

or

\[
\zeta_b = \sigma C_{\text{leak}}
\]

(22)

(23)
corresponding to a decrease in the work required

\[
\delta P_{\text{comp}} = \delta \zeta_b \, M_{\text{re}} \, \Delta h_{\text{is}} \left( 1 + \zeta_s \right)
\]  

(24)

where

\[
\delta \zeta_b = -\sigma \, (C_{\text{is}} - C_{\text{ls}}) = -\sigma \, C_{\text{ls}} \, \frac{\Delta h_{\text{is}} \left( 1 + \zeta_s \right)}{c_p \, T_s}
\]

(25)

For \( \kappa = 1.1 \) and \( \zeta_s = 0.3 \), the relative reduction of the leakage effect \( \delta \zeta_b / \zeta_b \) is roughly 10-20%:

\[
\begin{array}{ccc}
\text{pressure ratio} & 2 & 3 & 4 \\
\hline
\delta \zeta_b / \zeta_b & -0.08 & -0.14 & -0.17
\end{array}
\]

• another reduction of the power is due to the fact, that the refrigerant is initially partly in the liquid state, being vaporized only gradually during compression as shown in fig. 4. Thus only a part of the compression is effected in the vapor phase; the other part undergone in liquid state requires very little power. Denoting an average value of the end pressure of liquid state by \( \Pi^* p_s \) and of the corresponding density by \( p_s^* \) we find for the isentropic compression power

\[
P_{\text{comp-is}} = \frac{M_{\text{exp}} \, K}{K-1} \, P_a \left[ \frac{K-1}{K} - 1 \right] \text{ initial vapor}
\]

\[+ M_{\text{liq}} \, \frac{K}{K-1} \, \frac{\Pi^*}{p_s} \left[ \frac{K-1}{K} - 1 \right] \text{ initial liquid}
\]

\[
= \left( M_{\text{exp}} + M_{\text{liq}} \right) \frac{P_a}{p_s} \left[ \frac{K-1}{K} - 1 \right] \left[ 1 - \frac{M_{\text{liq}}}{M_{\text{re}}} \left[ 1 - \frac{\Pi^*}{p_s^*} / p_s \left[ \frac{K-1}{K} - 1 \right] \right] \right]
\]

(26)

\( \Pi^* \) is not known; by approximating it by \( \sqrt{\Pi} \) and \( p_s^* / p_s \) by \( \Pi^* \), we obtain for the second term in brackets 0.5 \( M_{\text{liq}} / M_{\text{re}} \) (the constant 0.5 varies hardly with the pressure ratio \( \Pi \)), which corresponds to the relative reduction in power \( \delta \zeta^* \) due to this effect.

\[
\delta \zeta^* = \frac{\Delta P_{\text{re}}}{P_{\text{re}}} = -0.5 \, M_{\text{liq}} / M_{\text{re}}
\]

(27)

Therefore

\[
\zeta = \delta \zeta_b + \delta \zeta^*
\]

(28)

\[\text{Fig. 4: Temperature-entropy diagram of wet compression}\]
There are other influences, because the efficiency of the motor or the inverter may change due to a change in power required (different mass flow with liquid injection).

\[ \Delta \eta_{el} = \frac{\Delta P_{el}}{P_{el}} = \frac{\partial (\eta_{inv} \eta_{inv})}{\partial \Delta \eta_{el}} \frac{\Delta M}{M} = C_{el} \frac{\Delta M}{M} \]

(29)

where \( C_{el} \) depends on the motor and inverter characteristics and

\[ \frac{\Delta M}{M} = \frac{\Delta \eta_{inv}}{\eta_{inv}} = \Delta V_{s-eff} \frac{\eta_{inv}}{V_{s-eff}} + \frac{\Delta T_s}{T_s} \]

(30)

The first part corresponds to the change in effective suction volume, the second part to the disappearance of an eventual superheat \( \Delta T_s \) due to liquid injection. \( \Delta V_{s-eff} \) is negative, as shown by the experiments, probably due to the flashing effect.

Therefore the total effect of liquid injection on the electric power consumption is

\[ \Delta P_{el-sot} = \left[ \frac{\Delta \eta_{b} + \Delta \eta_{c}}{\eta_{inv}} + \Delta \eta_{el} \right] P_{el} \]

(31)

Still another effect of liquid injection concerns the shell losses, which are normally reduced due to a lower temperature difference with the surroundings; the energetic efficiency \( \eta_{en} \) according to (3) may thus increase by \( \delta \eta_{en} \). Neglecting cross effects on the compression process, we obtain finally

\[ \Delta P_{rot-sot} = \eta_{inv} \left[ \frac{\Delta \eta_{b} + \Delta \eta_{c}}{\eta_{inv}} + \Delta \eta_{el} + \delta \eta_{en} \right] P_{el} \]

(32)

**Experimental Setup**

The part of the test rig concerned with the inverter and scroll compressor measurements is depicted in fig. 5. The inverter is of the commonly used pulse width modulating type\(^1\). The scroll compressor is hermetically sealed in a high-pressure shell, where the induction motor is cooled by the discharged refrigerant gas.

Pressure and temperature have been measured with piezo-resistive pressure transducers and thermocouples respectively. The rotor speed of the hermetic compressor was determined by analyzing the pressure pulsations by means of a Fast Fourier Transformation (FFT). The refrigerant mass flow rates were measured with Coriolis type mass flow meters and the electric power input with a precision wattmeter, tolerating harmonics of up to 10 kHz.

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**Fig. 5**: Instrumentation of inverter and scroll compressor

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1 pulse width modulation is a method of creating voltage at variable amplitude and frequency by means of rectangular pulses with differing time widths to operate induction motors at variable speed.
The experiments have been carried out at low and high speed, i.e. 44 Hz and 110 Hz, with a suction pressure of 3 bar, an operating pressure ratio of 4 and three different injection ratios. The results are shown in figures 6 and 7.

- Low Speed

The inverter efficiency increases slightly from 93% to 95% with decreasing suction vapor quality. The reason is the higher compression work due to the higher suction density and the fact that pulse width modulating inverters work more efficiently at higher load.

The mass flow efficiency decreases considerably from 98% to 80% at a suction vapor quality of 86% (see also fig. 3). One of the main reasons could be the backflashing of liquid refrigerant accumulated on the outer wall of the compression chambers; the partial flashing of this liquid into the suction chamber reduces the net suction flow rate. Other possibilities are changes in the geometry of the gaps induced by the differential thermal expansion of the material and interaction with the lubricating oil. The isentropic efficiency strongly decreases with liquid injection from 68% to 41% whereas the energetic efficiency increases from 76% to 84% due to lower temperature differences with the ambient. The decrease of the isentropic efficiency can be explained partly by heat transfer to the refrigerant instead of loss to the surroundings.

The specific compression work that includes inverter and compressor losses remains almost constant for all injection ratios at low speed, except for one point where the operating pressure ratio was slightly higher.

- High Speed

The results at high speed show the same trend as at low speed differing only in the amount of efficiency change due to the higher mass flow rate as shown in fig. 7. Since the inverter works more efficiently at high speed, its efficiency is 96%; approximately 2% higher than at low speed. The mass flow efficiency decreases from 95% to 87%, the isentropic efficiency from 58% to 49% whereas the energetic efficiency increases slightly from 88% to 91%. The trend of the specific compression work decreasing by approximately 10% with lower vapor quality favors liquid injection at high speed, where high discharge temperatures usually occur.
CONCLUSIONS

The leakage fluid in a scroll compressor with liquid injection can be described as a backflowing refrigerant / oil solution that partly flashes in the suction chamber. Assuming that the radial clearances are completely filled with liquid, the characteristic type of motion of the contact point of the scroll wraps (a combination of rolling and sliding) and the flashing of the leakage fluid are probably the main causes for the reduction of the theoretical suction volume. The leakage resulting from the pressure differential has a smaller impact and depends strongly on the viscosity due to the solubility characteristic of the refrigerant / oil.

The experimental analysis showed that a wet compression process lowers the expected gain in mass flow, presumably due to flashing effects. Thus, most of the efficiency indicators, such as mass flow and isentropic efficiency, tend to decrease with liquid injection. Only the energetic efficiency increases due to lower shell temperatures and thereby reduced shell losses. The consequences of wet compression on the life time of the compressor have not been considered in this investigation.

REFERENCES


