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ABSTRACT

Compression of pure refrigerants in the two-phase flow region has a number of advantages for the performance of vapor compression systems. Compressors may be lubricated by liquid refrigerant so that the process side of the cycle is not contaminated by lubricant. The compression process can be realized in the two-phase region preventing superheating with its associated irreversibility losses. The compression work is removed by the latent refrigerant heat so that discharge temperatures remain low even at large compression ratios.
The effect and requirements for wet compression of pure refrigerants are investigated. The perspectives of oil-free compressors in ammonia vapor compression are investigated with a simulation model. The performance of wet compression systems is compared to the performance of conventional dry compression systems. An overview is given of the requirements imposed to wet compressors and vapor compression systems that make use of wet compression.

1. INTRODUCTION

When the medium of a vapor compression cycle undergoes a compression process as a two-phase liquid-vapor mixture, the process it undergoes is said to be a wet compression process. Moran and Shapiro (2000) state that wet compression is generally avoided because the presence of liquid droplets can damage the compressor and that in actual systems the compressor handles vapor only (dry compression). Recent studies by Zaytsev (2003) and Infante Ferreira et al. (2006) identified screw compressors to be most suitable for operation under wet compression conditions: they are tolerant to liquid carry over and have rather high efficiencies. A screw compressor has been modified to make it suitable for wet compression operating conditions and it has been implemented as the compressor of a wet compression compression-absorption heat pump set-up. The experimental results indicate that, depending on the operating conditions, reasonable isentropic efficiencies may be attained under wet compression conditions.

The expected advantages of wet compression with pure refrigerants are:

i. The refrigerant remains oil free: lubricant contamination is prevented;
ii. Irreversibilities caused by vapor superheating during compression are prevented;
iii. Single stage compression for low temperature applications becomes feasible without too high discharge temperatures.

The performance of wet compressors when operating with pure refrigerants will be investigated in this paper. The simulation program developed for predicting the performance of wet twin-screw compressors working with ammonia-water mixture (Infante Ferreira et al., 2006) could, in principle, be applied for pure ammonia. Pure ammonia represents in fact one extreme case for simulation, i.e. when the overall concentration is set to unity. In this case however, an error occurs while the properties of the mixture in the two-phase region are evaluated based on vapor quality corresponding to thermodynamic equilibrium

$$x = \frac{X_o - X}{Y - X} \tag{1}$$

For pure ammonia, $X=Y=X_o=1$ induce an error of the kind "zero over zero" when any thermodynamic property of the mixture is evaluated.
Two possibilities to overcome this difficulty may be considered:

i. Elaborating a new wet compressor model specifically for pure refrigerants;
ii. Using a high overall concentration, close to unity, and simulating the process with the "ammonia-water" program. Refrigeration quality ammonia has mostly a water residual of about 0.25%.

In Section 2 the ammonia water program is used for high ammonia water concentrations (option ii) to investigate the compressor performance under wet conditions when pure ammonia is the refrigerant used. Option i leads to a simplification of the ammonia-water simulation model as reported by Zamfirescu et al. (2004). The remaining Sections of this paper are dedicated to a quantification of the advantages and requirements imposed by wet compression.

2. PERSPECTIVES OF OIL-FREE COMPRESSORS IN AMMONIA VAPOR COMPRESSION SYSTEMS

The homogeneous model developed for ammonia-water has been used to simulate the compressor behavior when pure ammonia is applied. The model results have been compared with experimental results in Infante Ferreira et al. (2006). This validated program for ammonia water wet compression has been used to predict the performance of a wet compressor working with pure ammonia for low temperature applications. A high overall concentration of about 0.995 should be set in the program input file. However, numerical instability occurred very often in the code at such a high concentration. Consequently, a lower concentration – namely 0.95 – was used for simulation, and this situation proved to be stable.

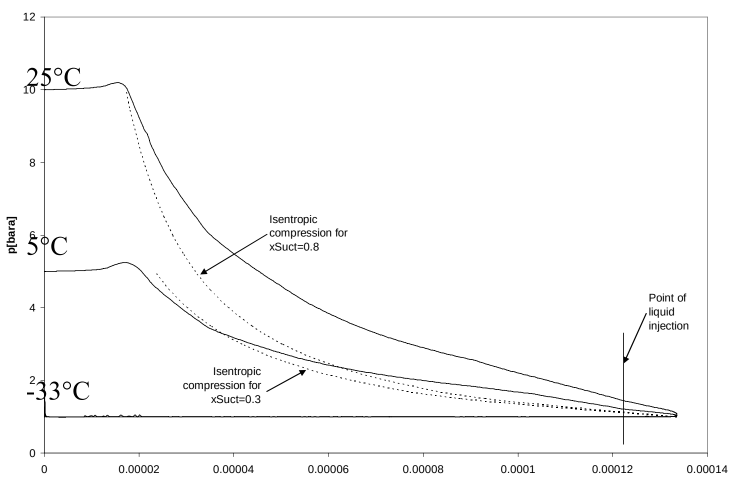

Fig. 1. Predicted indicated diagrams for (almost) pure ammonia.

The results are plotted in Fig. 1 on an indicated diagram. Fig. 1 shows also the isentropic compression lines so that the order of magnitude of the irreversibilities can be visualized. It appears that the compression process from 1 to 5 bar does not differ too much from the isentropic compression indicating a high isentropic efficiency (about 0.89). The compression process from 1 to 10 bar shows a large discrepancy with the isentropic process. Here the leakage flow becomes quite significant reducing the isentropic efficiency significantly (about 0.67).

For pure ammonia, if the discharge is maintained at 15.55 bar (condensation temperature of 40°C) then suction is at 3.1 bar (-8.4°C) for the pressure ratio case of 5 and at 1.55 bar (-24.5°C) for the pressure ratio case of 10. Operation in refrigerated systems with application temperatures around 0°C seems promising. Low temperature applications will require two-stage operation or will present a relatively low isentropic performance. The advantages remain that the discharge temperature is low and that the flow is lubricant free.

Liquid injection flow rates are a point of concern. For the pressure ratio of 5 the quantity of vapor taken from the suction is approximately 20 g/s. To attain suction conditions with a vapor quality of 80% (the inlet quality expected to lead to COP's close to a maximum), only 4 g/s liquid refrigerant should be injected from suction. As it has been discussed in Infante Ferreira et al. (2006), this flow is by far not enough to guarantee high isentropic efficiencies for the compressor. The results shown in Fig. 1 for the smaller pressure ratio have been obtained with liquid injection flows between 40 and 80 g/s. For the larger pressure ratio an injection flow of 100 g/s has been assumed. This indicates that for pure ammonia wet compression, re-injection from the discharge side will be needed to guarantee the availability of enough liquid refrigerant.

3. WET VERSUS DRY COMPRESSION OPERATION

Assume that, similarly to the compression-absorption application, also efficiencies as predicted by the model can be realized in practice. The advantages of operating a vapor compression refrigeration cycle in the wet compression region can then be quantified. Assuming a constant isentropic efficiency of 0.65, the results presented in Figure 2 have been obtained for pure ammonia as refrigerant. Lubrication of the compressor with the liquid refrigerant requires that the outlet conditions remain in the two-phase region limiting to certain extent the range of operation. For higher efficiencies the operating range will be larger than indicated in Figure 2. Each line in Figure 2 indicates the COP gain of wet compression in relation to dry compression for a certain inlet quality of the two-phase liquid-vapor mixture. The COP gain is modest: 5 to 7 %.

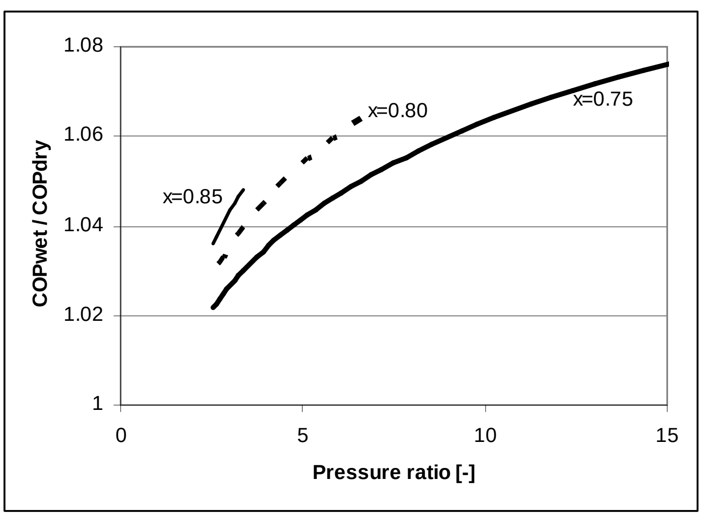

Fig. 2 – COP gain when wet compression is applied instead of dry compression.

Figure 2 has been obtained by maintaining the discharge pressure at 15.55 bar (40°C) while the suction pressure was varied. Table 1 indicates for three pressure ratios the corresponding evaporating temperature and the predicted discharge temperature for the dry vapor compression cycle assuming the same isentropic efficiency and adiabatic compression.

Table 1. – Comparison of discharge temperature level for dry and wet vapor compression operation.

Pressure ratio [-]	Evaporating temperature [°C]	Discharge temp. dry [°C]	Discharge temp. wet [°C]
5	-8.4	159.5	40.0
10	-24.5	222.7	40.0
15	-33.0	262.0	40.0

The discharge temperature reduction attained with wet compression is quite significant and allows single stage ammonia operation up to large pressure ratios. As demonstrated in Section 2, the larger pressure ratios will lead to reduced isentropic efficiencies.

4. COMPRESSOR REQUIREMENTS

Zaytsev (2003) reviewed the different compressor types available in the market for their applicability as compressor for wet compression resorption cycles. Taking the specific requirements of these cycles, the single and twin screw compressors appeared to be the most suitable. They are tolerant to liquid carry over and have rather high efficiency. The disadvantage is their built-in volume ratio.
Zaytsev and Infante Ferreira (2005) proposed a method for twin screw profile generation that allows for rotor design with increased efficiency when operating under wet compression conditions. The potential of the method is illustrated for an application example for which the contact line of the rotors with the proposed design is 5% shorter. Also the blow hole area for the proposed profile is almost two times smaller than for the original profile. The torque on the female rotor resulting from both the gas pressure and friction forces does not change its sign. If this would be the case, the periodic change of the power transfer direction would cause unstable operation of the compressor: increased wear of the rotors, vibration and noise. The average torque ratio between female and male rotor is 0.06, much smaller than other profiles conventionally used. For example, according to You et al. (1995) the average torque ratio is 0.15 for the SRM D profile. The higher torque induced on the female rotor by the gas pressure may result in higher wear of the rotors, see Zaytsev and Infante Ferreira (2005). When the seals and bearings friction torque is added, the absolute value of the female rotor torque will increase, but the sign will not change.

Rotor contact tribologic conditions in wet compressors are more severe than in serial production machines. Unlike most of the refrigeration screw compressors, a low-viscous refrigerant is used for lubrication instead of oil. Besides employing profiles with low torque transfer, additional surface treatment will be needed. Aim of this treatment is the reduction of rotor contact friction and wear.
Decreasing of contact friction losses will directly and indirectly reduce the total compressor power. Direct reduction is the reduction of the mechanical power spent on friction. Indirect reduction owes to the fact that less friction power would be converted into heat, hence the compressed gas temperature and indicated power would be reduced. Reduction of wear increases the compressor durability. A possible way to decrease friction and wear losses is to apply a coating on the rotor surface. The appropriate coating should meet a number of requirements imposed by the compressor application.

- The coating should insure low friction coefficient
- The coating should have low wear rate
- Thickness of the coating should not exceed several micrometers
- The coating should be chemically inert and compatible with refrigerant environment

- The coating should sustain operation temperatures of at least 150^0C

Based on the results on efficiency loss caused by rotor friction, Zaytsev (2003) suggests that the coefficient of friction must be limited by a value of 0.2. He proposes Eq. (2) to calculate the maximal allowed specific wear rate of the coated surface

$$\zeta = \frac{h}{w_c} \cdot \frac{v_{ci}}{v} \cdot \frac{2\pi}{\omega_i t}, \tag{2}$$

where h is the allowed wear depth; w_c is the linear contact load; v_{ci} is the contact point velocity relative to the surface of the rotor i; v is the relative rotor velocity (slip velocity) at the contact point; ω is the angular speed of rotor i; t is the compressor operation time. Results of Eq. (2), obtained for an allowed wear depth of 1 µm per compressor operation time of 1000 hours with shaft speed of 3000 rpm and with a contact load profile that applies for the proposed rotor geometry, indicate that the rotor coating specific wear rate should have an order of magnitude of 10^{-18} $m^3/(N \cdot m)$ or lower.

The tribology of coated surfaces is extremely complicated. Holmberg et al (1998) presented the fundamentals of coating tribology among with an extended list of references. The tribological contact processes are classified into macro-, micro-, nano-mechanical and tribo-chemical contact mechanisms and material transfer. Numerous properties play a role in the contact, such as the substrate and coating hardness, thickness, roughness, shear strength, elasticity, adhesion, chemical reactivity, thermal stability and many others. Various types of surface coatings have been developed during the last few decades; however, up to now there are only a few of them that meet all the requirements listed. Niebuhr et al (1997) studied wear mechanism of chromium nitride and titanium nitride coatings using test equipment, which simulates the sliding-rolling friction of screw compressor rotors. The test results have shown a wear rate of 0.5 µm/h (specific wear rate of $6 \cdot 10^{-15}$ $m^3/(N \cdot m)$) for the CrN and 5 µm/h (specific wear rate of $6 \cdot 10^{-14}$ $m^3/(N \cdot m)$) for the TiN coatings. Such wear rates are three orders of magnitude higher than required and can therefore not be accepted.

Another coating type, which offers a wear resistance more than an order of magnitude better than nitride coatings is the diamond-like carbon (DLC) coating type. The name DLC covers a wide class of coatings with different kinds of deposition techniques, structure, non-carbon additives, single of multiple layers, etc. Among deposition techniques the chemical (CVD) or physical vapour deposition (PVD) from a carbon-containing gas can be named as well as sputtering from solid carbon. The coating structure is an amorphous solid with graphite and diamond boundings in various combinations or a polycrystalline diamond in an amorphous carbon matrix. The additives may by hydrogen and different metals such as chromium, titan, tungsten, their oxides and carbides. Additions may form one single layer or may be deposed as multi-layers. Typical thickness of a DLC coating is several micrometers. They have high hardness (from 20 to 40 GPa) and low friction coefficient (order of 0.1).

PVD and CVD deposed DLC coatings have proved to be successful in improving the performance of cutting and forming tools, however their applications in heavy loaded machinery such as gears, bearings, engines is limited. The coatings are very hard, which makes them brittle and affects adhesion at high loads.

Development of the magnetron sputtering deposition technique leads to the origin of new series of amorphous carbon coatings. Although these coatings are still called DLC, they are graphite-like rather than diamond-like, Yang et al (2000). With metallic additives, they have shown improved tribological properties when compared with "conventional" DLC coatings, Jones et al (1998), Fox et al (2000). The wear rate is in the range of 10^{-17} – 10^{-18} $m^3/(N \cdot m)$, even values as low as 10^{-19} $m^3/(N \cdot m)$ have been reported, Teer (2001). The friction coefficient is still low, 0.05 – 0.15. The hardness is several times lower than of the "conventional" DLC coatings. Combination of such properties makes new coatings suitable for applications with heavier loads, where previous coating types have found little or no success, Yang (2000), Teer (2001).

Table 2 is a summary of properties of four different commercially available coatings considered for application in the compressor.

Table 2. Comparison of four different carbon-based coatings

Number	1	2	3	4
Coating type	CrC-C (Amorphous carbon with 20% Cr) with a chromium interlayer which improves the adhesion to the substrate	WC-C (tungsten carbide-carbon) with a chromium interlayer which improves the adhesion to the substrate	(t-a:C) tetrahedrally bounded amorphous carbon	(a-C:H), (a-C:H/Ti/O) amorphous hydrogenated carbon with titanium oxide interlayer which improves the adhesion to the substrate
Hydrogen containing	No	Yes (however, applications in fresh water- and seawater pumps are reported)	No	Yes
Substrate must be hardened	Recommended	No	Recommended	
Hardness of coating, HV	Around 2500	1000 – 1600	5000	5000 - 7000
Thickness, µm	Up to 5, typically 2.5	1-4, standard 3	0.1-5, normally 1	2-4
Friction coefficient	Dry 0.07-0.1, de-ionised water and brine 0.05	Dry, against steel 0.2	0.1-0.2 in comparison to steel against steel	Dry against steel 0.1-0.2
Wear rate, $m^3/(N \cdot m)$	Dry $3 \cdot 10^{-17}$, in water $5 \cdot 10^{-18}$	No data	Against steel$<5 \cdot 10^{-17}$	Dry $5 \cdot 10^{-17} – 10^{-16}$
Operation temperature, ^0C	up to 450	up to 300	up to 500-700	up to 300
Price for one rotor pair, €	410	880	800	1500 including pre-polishing
Number	1	2	3	4

Initially the compressor rotors have been coated with an amorphous hydrogenated carbon (a-C:H) diamond-like coating (4 in Table 2). After about 30 hours of running, the coating on the driving flanks was worn out. Studies of tribological performance of different types of DLC coatings by Ronkainen et al (1998), (2001) and Drees et al (1996) have shown that the hydrogenated films show good performance in dry conditions, but in aqueous environments they suffer from severe wear. Hydrogen-free amorphous carbon (a-C) films show good performance in water. This indicates a possible reason of the failure: the water in the working mixture caused severe wear of the (a-C:H) coating. Another reason of the coating failure could be the soft substrate (stainless steel) that could not provide enough load support for the thin and hard coating layer, which might become brittle under such conditions.

A slightly modified set of stainless steel rotors has been polished and then hardened by carbon diffusion. A special hardening process has been applied, which does not affect the corrosion resistance. The carbon diffusion treatment has increased the hardness of the stainless steel rotor surface roughly from 200 to 1000 HV offering a better load support. Afterwards, the rotors were coated with a hydrogen-free amorphous carbon coating (1 in Table 2), which contains chromium additives to improve substrate adhesion and to make the coating less brittle and therefore, suitable for heavier loads. According to Table 2, this coating has the lowest friction coefficient and the lowest wear rate. With this compressor, Infante

Ferreira et al. (2006) investigated experimentally the effect of the liquid injection angle and total injected liquid flow on the performance of a compressor operating under wet conditions. The ideal location for liquid injection is during the start phase of compression so that flashing into the suction cavity is prevented. Also the labyrinth seal leakage flow must be re-injected at the same location. The labyrinth seals allow separation of the process side from the oil lubricated bearing housing by PTFE shaft seals. Under the experimental conditions reported in Infante Ferreira et al. (2006) a minimum of 40 to 60 g/s injection flow of liquid refrigerant was needed to guarantee reasonable isentropic efficiencies.
The liquid in the suction line was separated and injected into the compressor cavities after they are disconnected from the suction port.

5. CONSEQUENCES FOR SYSTEM DESIGN AND REFRIGERANT CONTENTS

Liquid vapor separator / receiver designs with small refrigerant contents are needed at compressor suction and condenser outlet, respectively. These separators should be designed to guarantee that the total system contents remain low.

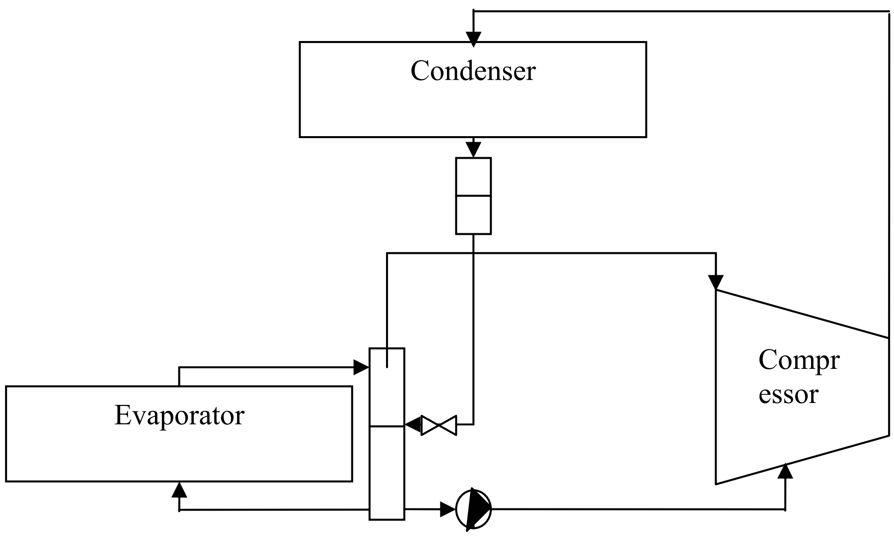

Fig. 3. – Possible arrangement for vapor compression cycle with wet compression.

Figure 3 illustrates a possible arrangement. Part of the liquid is pumped from the low pressure separator into the compression cavity after start of compression which has a pressure slightly higher than the suction pressure. The liquid injection to the low pressure separator is liquid level controlled and the evaporator is natural gravity fed.
To allow for low refrigerant mass contents, the condenser would preferably be a falling film condenser with no liquid accumulation in the heat transfer area. Also evaporator and low pressure separator should be designed for minimum refrigerant contents. Alternatively the evaporator should also be a falling film heat exchanger with a liquid receiver at the bottom. In this case the liquid pump would both deliver liquid for injection and to the evaporator.

6. CONCLUSIONS

Model predictions indicate that wet compression of pure refrigerants is feasible while reasonable isentropic efficiencies can be attained. These efficiencies are in the range 0.89 to 0.67 when the compression ratio varies from 5 to 10.

The operating conditions required imply that only a relatively small liquid mass flow rate can be taken from the suction side. To attain reasonable efficiencies, additional liquid should be injected. To prevent significant losses associated with the expansion of this liquid flow, heat recovery may be required in the liquid re-injection line. In the compression resorption heat pump cycle (see Infante Ferreira et al., 2006) this implies the application of an extra heat exchanger.

The COP of ammonia refrigeration systems operating with wet compression is 5 to 7% higher than the COP of dry compression systems. The discharge temperature reduction attained with wet compression is quite large.

The tribologic conditions of the contact between rotors in wet compressors impose surface treatment of the (screw compressor) rotors.

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