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VANE FRICTION AND WEAR INFLUENCE ON ROTARY VANE COMPRESSOR EFFICIENCY AND OPERATION:RESEARCH AND ANALYSIS REVIEW

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In developing rotary vane compressors for compressed air or refrigerating systems the biggest challenges are predicting of friction forces and the overcoming wear between vane and roller (stator) surfaces. In this review compressor energy balance in different operating conditions and effect of vane mass and compressor geometry are observed. Also rotary vane compressor friction and wear investigations in refrigeration systems are observed, influence of several vane surface coatings and lubrication is investigated. Different outlet pressures and revolution speeds on test compressor indicated power increase with rising discharge pressure and revolution speed. The reduction compressor speed from 1500 RPM to 1000 RPM would decrease the effects of friction to almost twice. It’s important to use coatings like TiN or WC/C, which have good and very good, wear resistance properties in order to reduce friction and wear on rotary vane friction pair.

Keywords: Rotary-vane compressor, vane, friction, wear, energy saving, coating.

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Introduction

In the IEO2016 (International Energy Outlook 2016) Reference case, worldwide industrial sector energy consumption is projected to increase by an average of 1.2%/ year. Most of the long-term industrial sector energy growth occurs in non-OECD (organization of economic cooperation and development) countries. From 2012 to 2040, industrial energy consumption in non-OECD countries grows by an average of 1.5%/year, compared with 0.5%/year in OECD countries. Despite the expected growth in non-OECD industrial sector energy use, the industrial share of total delivered energy in the non-OECD countries declines during the projection period, from 64% in 2012 to 59% in 2040, as a result of the move away from energy- intensive manufacturing in many emerging, non-OECD economies and as a result of more rapid growth of energy consumption in all other end-use sectors [1].

Most of the life cycle costs of an industrial Compressed Air System (CAS) are the energy costs. For this reason, CAS are responsible of 10% of electricity consumptions in EU-15 European countries, 9.4% in China and about 10% of the global industrial energy use in the United States. Recent studies state that the saving potential in industrial CAS ranges from 20 % to 50 %. Hence, around 400 TWh/year of primary energy could be saved. [2]. The reason why the CAS (compressed air sector) is addressed as an interesting sector of application of technological improvement, when CO_2 reduction is considered as a major future concern, is twofold. Firstly, it accounts for a 10% worldwide overall electricity consumption. Secondly, energy saving/recovery appears very promising, since compressed air systems show a high level of complexity in spite of very low characteristic efficiencies. [1].

Particularly, rotary volumetric machines, and among them, Sliding Vanes (SVRC) and Screw types are analyzed, as the most common and widespread technologies. Also there are more examples where rotary vane technology can be used, for instants for small pneumatic motors, which can be a good alternative to electromagnetic motors used in micromachining processes. Such kind of motors mainly has polytetrafluoroethylene (PTFE) loaded vanes with molybdenum disulfide, which is a self-lubricating material. [3]. Another good example could be small rotary vane compressor use in automobile air conditioning systems as this type of compressors are about 40 % lower volume and 30 % higer efficiency in comparison with wobble type compressors. [4].

In Screw and SVRC the global efficiency mainly depends on: a) the mechanical, organic and electrical efficiency, whose improvement appears an effective direction, with the mechanical representing a much more interesting matter of development, b) the thermodynamics of the compression during closed volumes. [1].

Structure and operation of the sliding vane rotary compressor

Conventional sliding vane type rotary compressors have free motion vanes built in rotor channels. Due to the eccentric assembly of the rotor into the stator cylinder and to the centrifugal forces action, vanes are moving in their channels and they are sticking to the stator cylinder sharing in cells the left space between the rotor, the stator cylinder and the vanes. The volume of one cell is changing continuously during one rotation of the rotor, which makes possible to reach stages of the compression process (aspiration, compression, discharge and expansion), (Fig. 1, a) compression process for vane type rotary compressor [5]. The volume of a cell as function of the rotation angle is defined by (Fig 1, b):

$$V_{\omega} = R \times e \times L \times (\beta + 2sin\frac{\beta}{2} \times cos\varphi + \frac{1}{2} e_R \beta cos2\varphi - \frac{1}{2} e_R \beta) \quad (1)$$

where: R – radius of the stator cylinder; e – eccentricity; $e = R\text{-}r$; r – radius or the rotor; L – length of the rotor, $\beta = \frac{2\pi}{Z}$ – the angle between two consecutive cells; Z – the number of the vanes; $e_R = \frac{e}{R}$ – relative eccentricity.

The maximum and minimum values of the cells volume is achieved for $\varphi = 0$ and $\varphi = \pi$, which means:

$$V_{max} = R \times e \times L \times (\beta + 2sin\frac{\beta}{2} + \frac{1}{2} e_R\, sin\beta - \frac{1}{2} e_R \beta) \quad (2)$$

$$V_{min} = R \times e \times L \times (\beta - 2sin\frac{\beta}{2} + \frac{1}{2} e_R\, sin\beta - \frac{1}{2} e_R \beta) \quad (3)$$

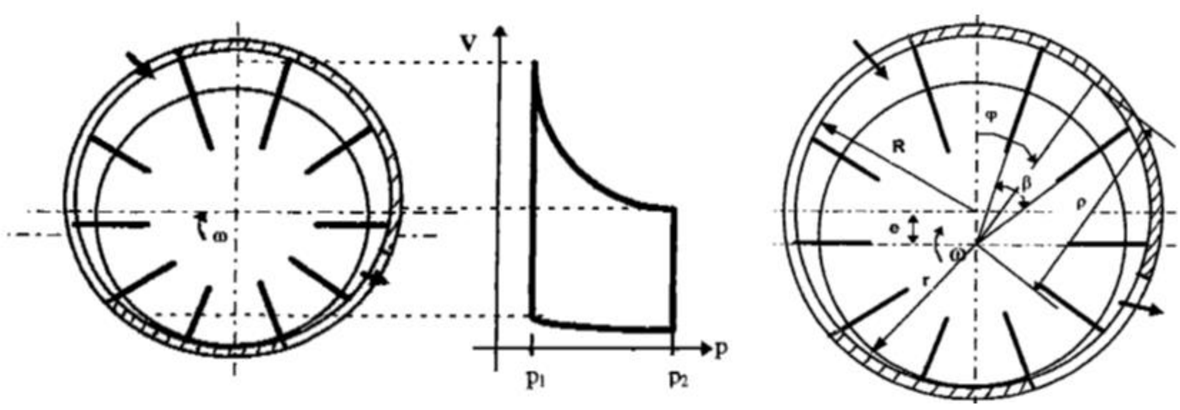
Loss power due to vanes friction

The vane has a complex motion, which is resulted from the composition of the transport motion (motion of rotation) round about rotor axis and the relative motion (motion of translation) reported to the rotor. To acquire friction forces F_1. F_2 and F_3 (see Figure 2) it is necessary to establish which forces are operating on the vane as a function of the rotor rotation angle [5]. The forces are presented as below:

The pressure force P is estimated using the pressure difference Δp between two consecutive cells:

$$P = L \times \mathrm{e} \times \left(1 + cos\varphi - \frac{e\,R \times \sin 2\varphi}{2}\right) \times \Delta p \quad (4)$$

where L – free vane surface;



a) b)

Figure 1. The construction scheme and the compression process for vane type rotary compressor a) the main construction dimensions b) [5].

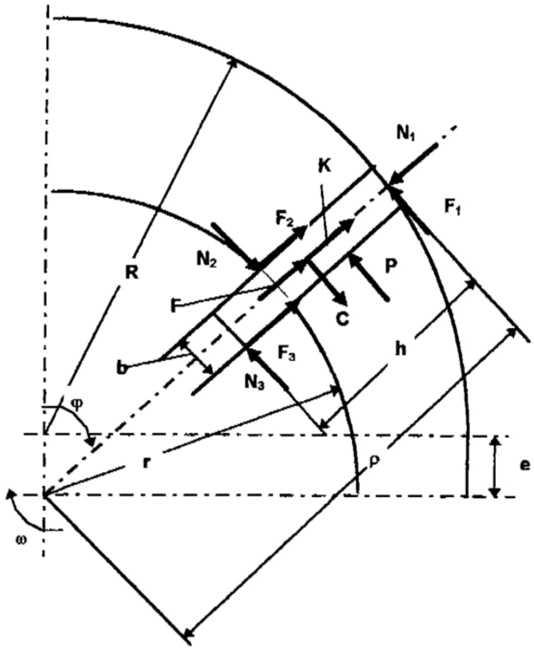


Figure 2. The forces that are acting on the single vane [5]

The centrifugal force F results from the rotation motion with angular speed ω = const. Considered as a concentrated force and with application point in the mass center of the vane:

$$F = m \times \omega^2 \times (\varphi - \frac{h}{2}) = m \times \omega^2 \times R \times (1- \frac{hR}{2} + e_R \times cos\varphi - e^2_R/2\ sin^2\varphi) \quad (5)$$

where: m – mass of the vane, kg; h_R – h/R relative height of the vane, m; h – height of the vane, m

The relative force K results from the relative accelaration:

$$K = m \times e \times \omega^2 \times (cos\varphi + e_R \times \frac{2\ cos2\ \varphi - 1}{T} + e^3_R \frac{sin2\ \varphi \times cos2\varphi}{T3}),$$
$$T = \sqrt{1 - e2R\ \times\ sin2\ \varphi} \quad (6)$$

The Coriolis force C results from the Coriolis acceleration:

$$C = 2 \times m \times e \times \omega^2 \times (sin\varphi + \frac{eR}{T} \times\ sin\varphi\ \times\ cos\varphi) \quad (7)$$

Mechanical work $L_{1,2,3}$ due to the vane friction at a complete rotation is:

$$L_{F1} = \mu \int_0^{2\pi} N_1 \rho d\varphi,\ L_{F2,3} = \mu\,(- \int_0^{2\pi} N_{2,3}\, d\rho + \int_{\pi}^{2\pi} N_{2,3}\, d\rho) \quad (8)$$

where: ρ – variable radius at the top of the vane and its equation is: $\rho = R\ (1+e_R\ cos\varphi - e^2_R/2\ sin^2 \varphi)$, and
$d\rho = -e\ sin\varphi\ (1 + e_R\ cos\varphi)\ d\varphi$ – the infinitesimal variation of radius; μ – friction coefficient of the vanes.

Total lost mechanical work due to vane friction, for all Z vanes is:

$$P_F = \frac{Z \times n}{60 \times 10^3} \sum_{j=1}^{3} L_{Fj} \quad [kW] \quad (9)$$

where: n – rotational speed (min^{-1}); Z – number of the vanes. [5].

We can see how compressor mechanical losses depend on friction power and it rises with rise of friction coefficient of the vanes [5].

Rotary vane compressor energy balance at different operating conditions

Nowadays, the most widespread industrial compression technology is the rotary volumetric one since it allows covering a wide range of flow rates and delivery pressures. The energy conducted saving potential achievable through friction power reduction in sliding vane rotary compressors was investigated using experimental and modeling approaches. Tests on a new mid-size industrial compressor operating at different steady conditions (outlet pressure 9, 12.5, 14.5 bar at 1000 and 1500 RPM) assessed the machine performance through measurement of mechanical power and the reconstruction of the pressure-volume diagram. An experimental methodology was also developed to quantify the power lost by friction and its measurement uncertainty using the concept of indicated mean effective pressure. [2]. Fig. 3 shows the layout and sensors types: low frequency pressure transducers and thermocouples were installed in relevant points of the machine while a gear flow meter and an ISA 1392 nozzle provided the oil and air mass flow rates measurements respectively. A flange torque meter was eventually used to measure the mechanical power as product of torque and revolution speed.

Four piezoelectric transducers were circumferentially mounted on an end wall plate in order to give a continuous pressure monitoring (Fig. 3). Since each of them provided a differential pressure with cyclic dispersion from one vane passage to the other, data were preliminary ensemble averaged over 10 acquisitions and further offset according to the pressure measurements at the intake and exhaust ports. The industrial vane compressor was tested at different outlet pressures and revolution speeds. Fig. 4 shows a bar chart with the energy balance. [2].

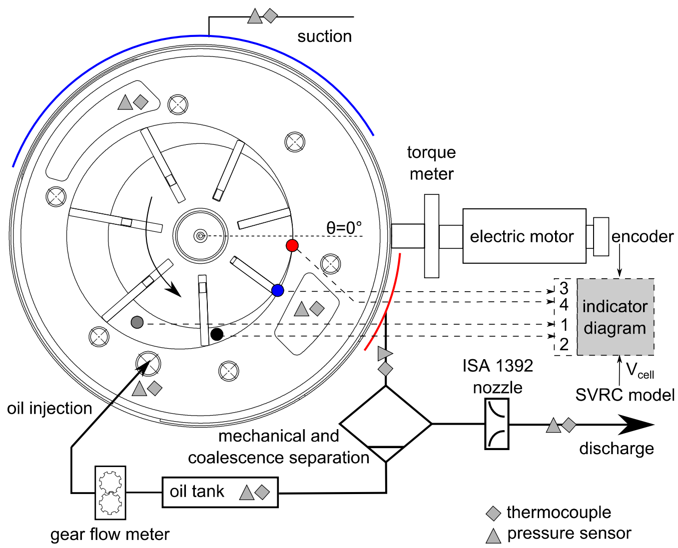


Figure 3. Experimental setup. Compressor Mattei ERC 22L [2].

The indicated power increased with discharge pressure and revolution speed. Indeed, the first parameter acts on the area of the indicator diagram while the second one on the overall mass flow rate compressed. At 12.5 bar, the compressor efficiency ranged from 87% at 1000 RPM to 86% at 1500 RPM while the specific energy consumptions at ISO 1217 conditions (20 ^{0}C, 1 bar) were 5.6 kW/(m^3/min) and 6 kW/(m^3/min), respectively. An unexpected result was the magnitude of power requested by the oil circulation that varied with discharge pressure but not with revolution speed. Even though the oil density with respect to the air one is almost three order of magnitudes higher, the power requested to fulfill the oil injection accounted for up to 7 % of the shaft power. This is mainly due to the amount of oil flowing inside the machine.

Therefore, a design suggestion would be to limit as much as possible the oil circulation without going below the minimum flow rate required for vane sealing and stable lubrication between components in relative motion. The amount of oil that exceeds this minimum threshold represents an energy waste. Compressor speed influenced friction power since it acted both on centrifugal and Coriolis forces as well as all the slip velocities. Since the centrifugal force is proportional to the square of revolution speed and the peripheral tip speed is linearly dependent with ω, an overall cubic dependency could be stated between compressor speed and friction power. On the other hand, the discharge pressure did not have a relevant influence on friction power. [2].

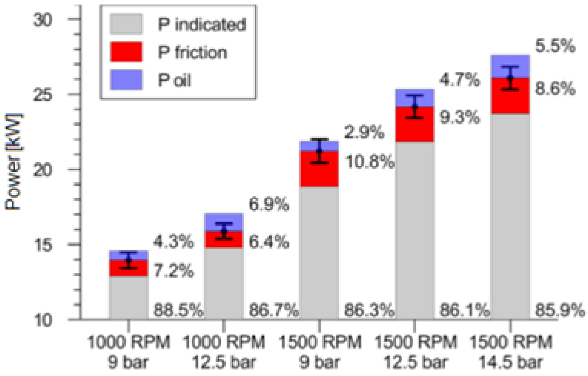


Figure 4. Compressor energy balance at different operating points (percentage refers to the mechanical power) [2]

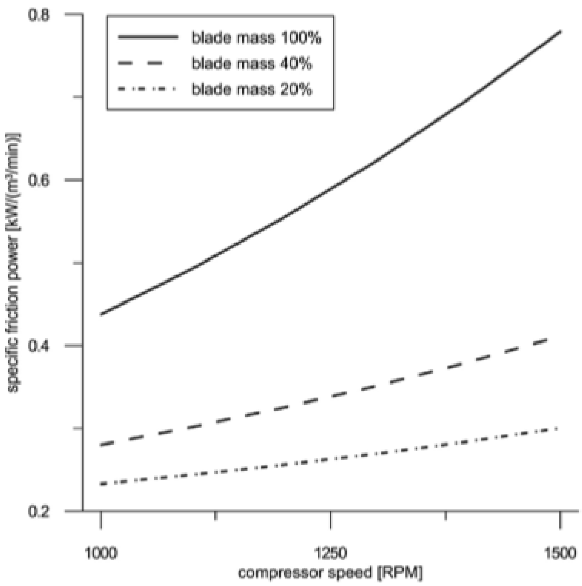


Figure 5. Effects of blade mass and revolution speed on the specific friction losses compressor operating at 12.5 bar [2]

It's important to mention that compressor rotation speed and vane mass also has an influence on the specific friction power at different compressor speeds. Considering that nominal speed of current industrial sliding vane compressors is usually 1500 RPM, the reduction to 1000 RPM would decrease the effects of friction from 0.78 kW/(m^3/min) to 0.44 kW/(m^3/min). Mass reduction is also effective on friction power reduction: at 1500 RPM, a 60% decrease of the blade mass would produce a 52% saving while a blade 80% lighter than the conventional one would lead to a saving of 61%. However, potential drawbacks of lowering the blade inertia could be oil film instability (dry contacts) as well as a significant change in the dynamic effects that could alter the vane sealing, thus the volumetric efficiency of the machine. [3].

Stator or rotor length and diameter influence on compressor aspect ratio and friction

Friction increases the overall energy expenditure for the compressed air production and it is still an open issue for the energy saving in SVRC (sliding vane rotary compressor). Part of the mechanical power supplied is dissipated by friction at the shaft bushes, between the rotor and the side covers of the compressor and because of the blade dynamics. However, the first two phenomena do not produce a noticeable contribution to the overall friction power thanks to the bush technology and to the absence of axial loads from the rotor to the covers. [6].

Using an experimental setup shown in a fig. 6 several investigations was made: compressor performance at ISO 1217 suction conditions (1 bar, 20 ^{0}C), model validation, influence at the aspect ratio on the compressor geometry and friction power dissipated, variation of the friction power per unit air delivered with revolution speed and blade tilt. The last two investigations are most actual for this review. Investigations were made by the same Italian scientist Giuseppe Bianchi and Roberto Cipollone from L'Aquila University (Italy). [6].

Fig. 7 shows the effects of varying either the stator or the rotor diameter on the compressor aspect ratio and friction power. To keep the reference mass flow rate, axial length of the compressor was varied accordingly. The resulting machine layouts are of two categories: an elongated one, in which the axial length is predominant with respect to the radial extent and a flat one which exhibits bigger diameters and shorter lengths than the actual geometry on tested compressor. A reduction of the stator diameter limits the peripheral tip speed (U) and leads to a decrease of the most significant contribution to the friction power.

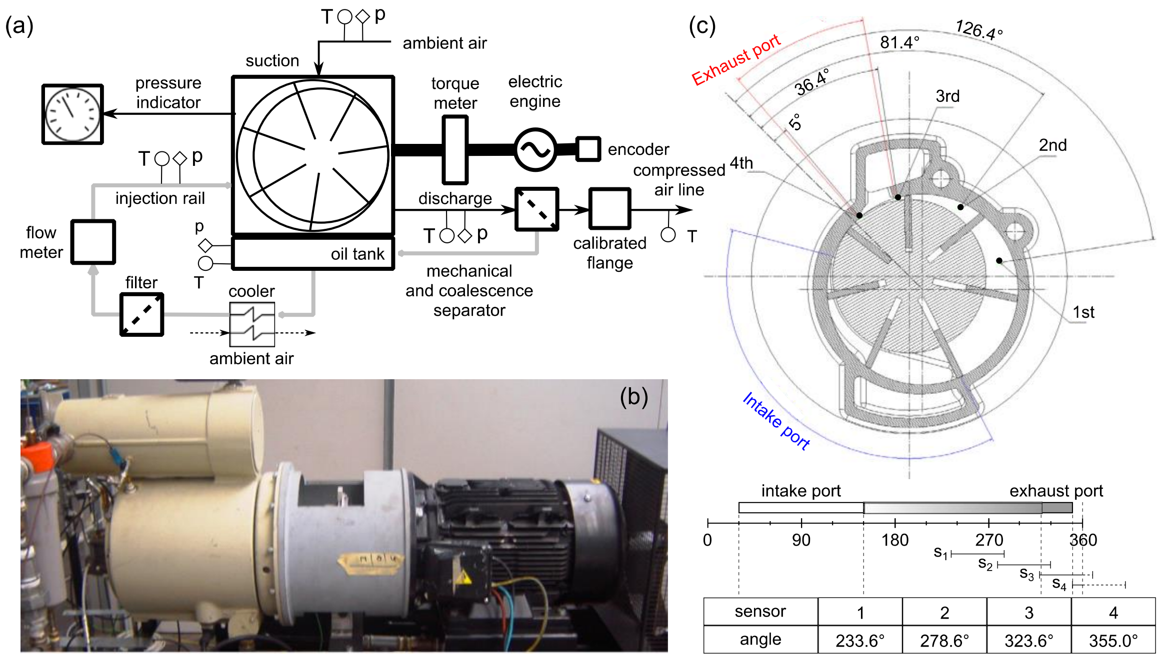


Figure 6. Experimental setup – transducers layout (a), test bench (b), piezoelectric sensors positioning (c). [6]

On the other hand, being the revolution speed kept constant, a decrease in the rotor diameter results in an increase of the sliding velocity v_{bl}, since the displacement that the blades have to accomplish in the same amount of time (i.e. the revolution period) increases. Even if this contribution has a minor importance compared to the one at the tip, a worsening of the mechanical efficiency would result from this aspect ratio.

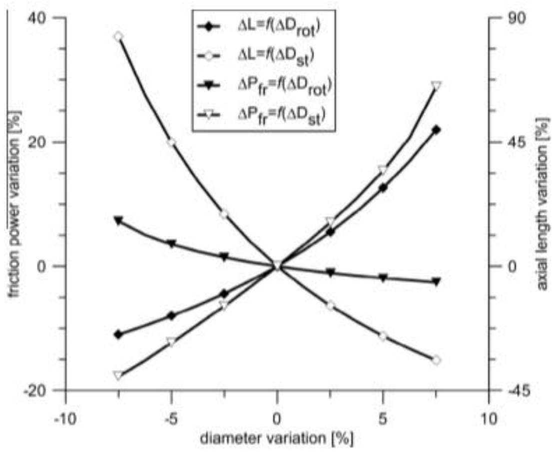


Figure 7. Influence of the aspect ratio on the compressor geometry and friction power dissipated [6]

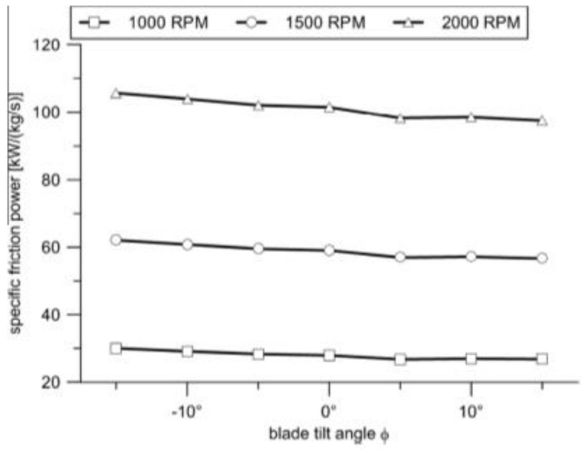


Figure 8. Variation of the friction power per unit air delivered with revolution speed and blade tilt [6]

A combined analysis on the effects of revolution speed and blade tilt on the mechanical efficiency of the compressor is reported in Fig. 8 in terms of specific power dissipated by friction with forward and backward blade tilt varying the angular velocity from 1000 RPM to 2000 RPM. The reduction of revolution speed has a direct and significant effect on friction power since it reduces both the inertial and fictitious forces as well as all the slip velocities.

Friction and wear investigations for rotary vane compressor used for refrigerating systems

One of the serious challenges in developing a rotary compressor with HFC (hydrofluorocarbon) refrigerant is the prediction of friction forces and wear amounts between vane and roller surfaces. [7]. Among all sliding pairs, the wear between vane and roller is the most critical, since these surfaces operate under boundary and mixed lubrication condition. Reviewed article compares the wear rate of TiN coated vane and uncoated original vane and investigates the optimum initial surface roughness to break in and to prolong the wear life of sliding surfaces using step load tests. TiN coating has typically been considered more effective than other coatings in resisting abrasive and adhesive wear. Also, TiN is expected to be compatible with the vane material, which is made of high-speed tool steel. Therefore, TiN coating for improving the tribological characteristics was selected and applied on the vane surface. [7].

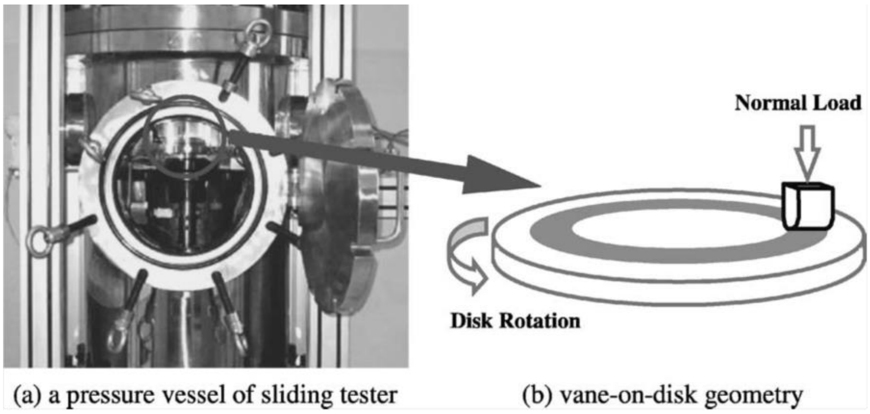


Figure 9. High pressure wear tester [7]

Vane-on-disk geometry was used in the sliding tests. The disks were cut from the cylindrical bar of the roller material. The vane sample was also machined from the same type of vane material as is used in the real compressor in order to du- plicate the mechanical properties. The TiN coating was applied only on the vane surface. The vane material was made of the high-speed tool steel and had hardness in the range of 850–900 HV. The roller material was made of Ni–Co–Mo gray cast iron and had hardness in the range of 550–600 HV. The surface roughness R_a of the disk ground was 0.14 m, which is close to the original roughness of the roller. The sliding tests were carried out in repeated-pass sliding using a vane-on-disk type tribometer under the various normal loads and sliding speeds in R410A/POE mixed environment as shown in Fig. 10b. This tribometer was capable of measuring the frictional forces and normal forces. And this tester has the pressure vessel, as shown in Fig. 10a, which is able to apply pressure of 20 bar. Test pressure of 3 bar was applied and test temperature was set at 50 ^{0}C initially. The vane was located in a holder that was clamped to a fixed arm with a transducer for friction force. The lower flat disk (roller material) was mounted on a rotating shaft in the oil bath. The contact was achieved by pressing the vane against the flat surface under a normal load applied by a spring force, which reduces the variation of normal force during sliding.

Under each test condition, three sliding tests were carried out. The wear scar widths for various normal loads and sliding speeds are shown in Fig. 10. The wear scar width is averaged. The error bars of the figure present the deviation of the wear scar widths. At the sliding speed of 100rpm, the wear amount of TiN coated vane was a little higher in the range of 30–70 kg, as shown in Fig. 10a. But at 90 kg, the wear of the uncoated vane increases accompanied by rapid increase in friction coefficient. As the sliding speed increases to 500 rpm, the wear of both vanes increases, and the wear of the uncoated vane was larger than TiN coated vane at all loads. However, at 1000 rpm, the wear amount decreased. It is suspected that the lubrication regime shifted from the boundary lubrication to the mixed lubrication. Also, as the sliding speed increased, the differences in wear amount between the coated and uncoated vane samples increased. In addition, the wear amount of the vane increases with increasing normal load. [7].

Wear coefficient *K*, were calculated from specimen wear. Wear coefficient is defined as wear volume loss per unit normal force per unit normal sliding distance. Fig. 11 shows the wear coefficient of the TiN coated vane and original vane for the various sliding speeds. At all speeds, the wear rate of TiN coated vane was lower than that of original vane.

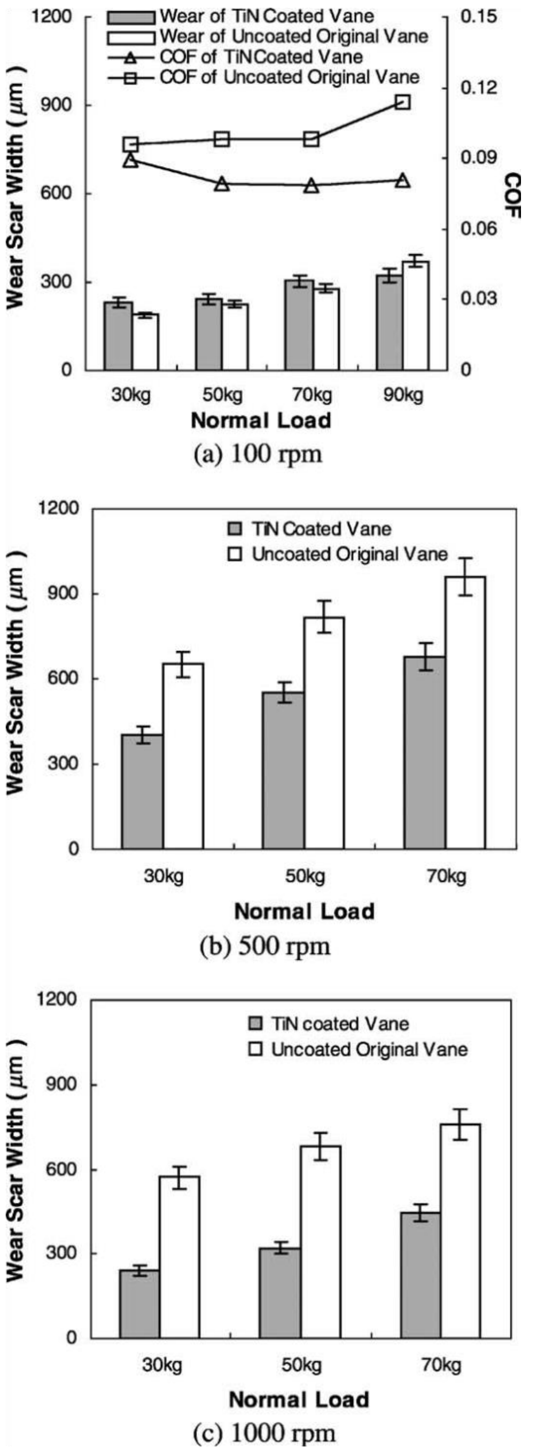


Figure 10. Wear scar width of the vane tip under various normal loads and sliding speeds. [7]

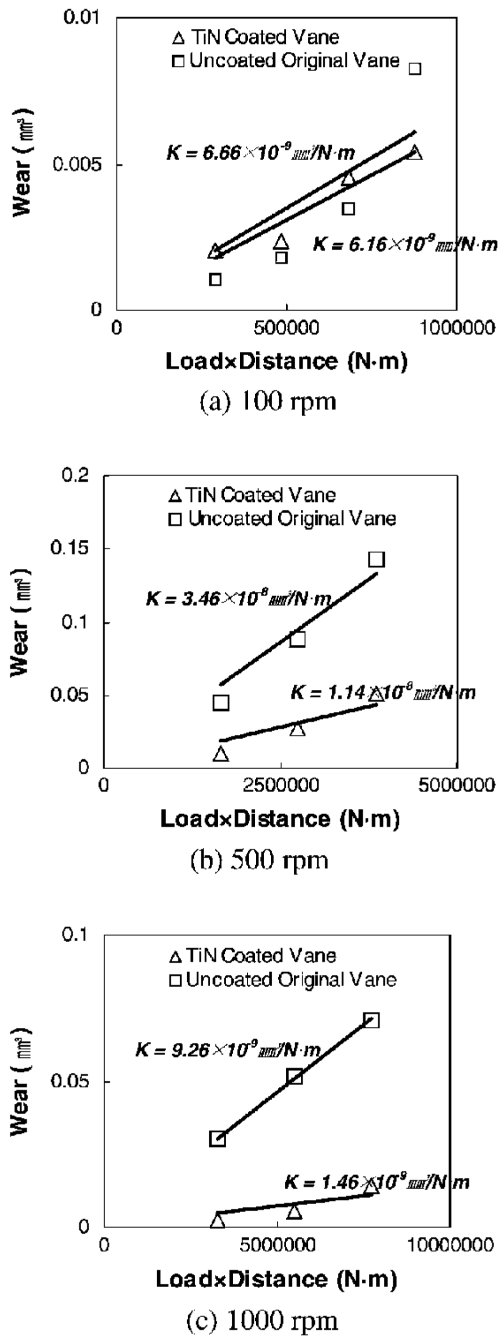


Figure 11. Wear coefficient *K* of the vane at each sliding speeds. [7]

The wear resistance of the coated vane is better than that of the original vane. Therefore, the wear resistance of the vane was improved by TiN coating. And at the sliding speed of 500 rpm, wear coefficient was the highest. It seems that, 500 rpm for the operating speeds was more severe than the other speeds in vane–roller contact. [7].

Tribological characteristics of different contact surface coatings for rotary compressor vane

In developing rotary compressors that will use HFC refrigerants, the biggest challenge is overcoming wear of the various components. In particular, severe wear between the vane and roller sliding pair is a persistent problem. In the next reviewed article, several hard coatings were applied on vane surfaces in order to improve the tribological characteristics, and their performances were evaluated experimentally.

In rotary compressors, wear takes place mainly between vane and roller, shaft and bearing, roller and flange as shown in Fig. 12. Among these sliding pairs, the wear between vane and roller is the most critical, since these surfaces operate under boundary and mixed lubrication conditions. [8].

Hard coating has typically been considered more effective than other coatings in resisting abrasive and adhesive wear. Further, hard coatings are expected to be compatible with the vane material, which is made of SKH51 (high speed tool steel). Therefore several hard coatings with potential for improving the tribological characteristics were selected and evaluated in this experiment. The coatings tested were: TiN, TiAlN, WCrC (tungsten carbide carbon), DLC (diamond-like carbon) and ion nitriding treatment. Also, a carbon vane material was tested for reference purposes. The Carbon vane, which was mainly composed of Al_4C_3 has been observed to prevent wear and scuffing phenomena during frequent stop-and-start operation of the compressor. However, its application as a vane material is limited due to its high brittleness and low hardness. All tests in described study were conducted in a Falex multi-specimen wear tester as a bench wear tester. In order to simulate the operating condition of the real compressor, a high pressure chamber which can sustain pressures up to 0.9 MPa was assembled into the wear tester. A schematic diagram of the tester is given in Fig. 12. The temperature of the chamber was raised using a cartridge heater located in the bottom plate and controlled by a microprocessor. To prevent a sudden increase of the temperature inside the chamber due to frictional heating during test, water was circulated through the chamber wall.

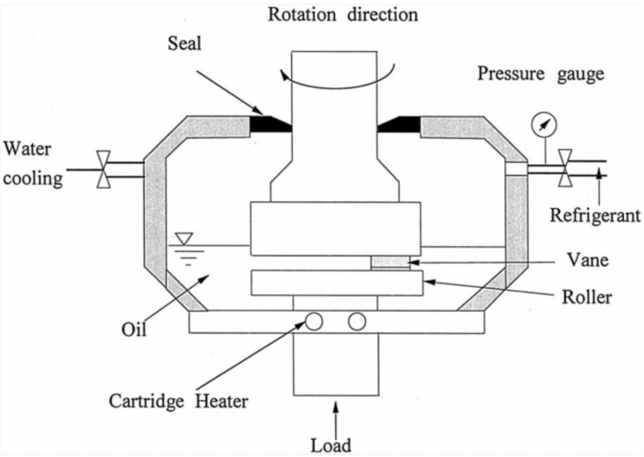


Figure 12. Schematic diagram of Falex wear test machine with high pressure chamber [8]

The coatings were applied only on the vane surface. The vane (upper specimen) attached to the rotating spindle was pressed against a stationary disk (bottom specimen). The vane material was made of SKH51 and had hardness in the range of 850–950 Hv. The roller (disk) material was made of Ni – Cr – Mo gray cast iron and had hardness in the range of 440–600 Hv. The worn surfaces of the sample vane tips were examined with optical microscope, SEM and EDX. SEM pictures of wear scar for each coated vane are shown in Fig. 13. Worn surfaces typically were smoother than the initial surfaces. For the TiN(I) coating, as shown in Fig. 13a, the worn surface is very smooth. The TiN(II) coating had a worn surface quite different from that of TiN(I) coating, as shown in Fig. 13b. The worn surface showed extensive plastic deformation, along with formation of the deep and wide grooves. It can be seen that the TiN(II) coating, which was deposited by a sputtering method did not seem to have proper wear resistance probably due to the poorer mechanical properties likely related to hardness and adhesion strength. The worn surface of the TiAlN coating was characterized by irregular and sharp edges of contact along with small grooves Fig. 13c. On the other hand, smoothening of the asperities of the worn surface was observed for the WCrC coating Fig. 13d, which showed good wear and friction performance in this test. [8].

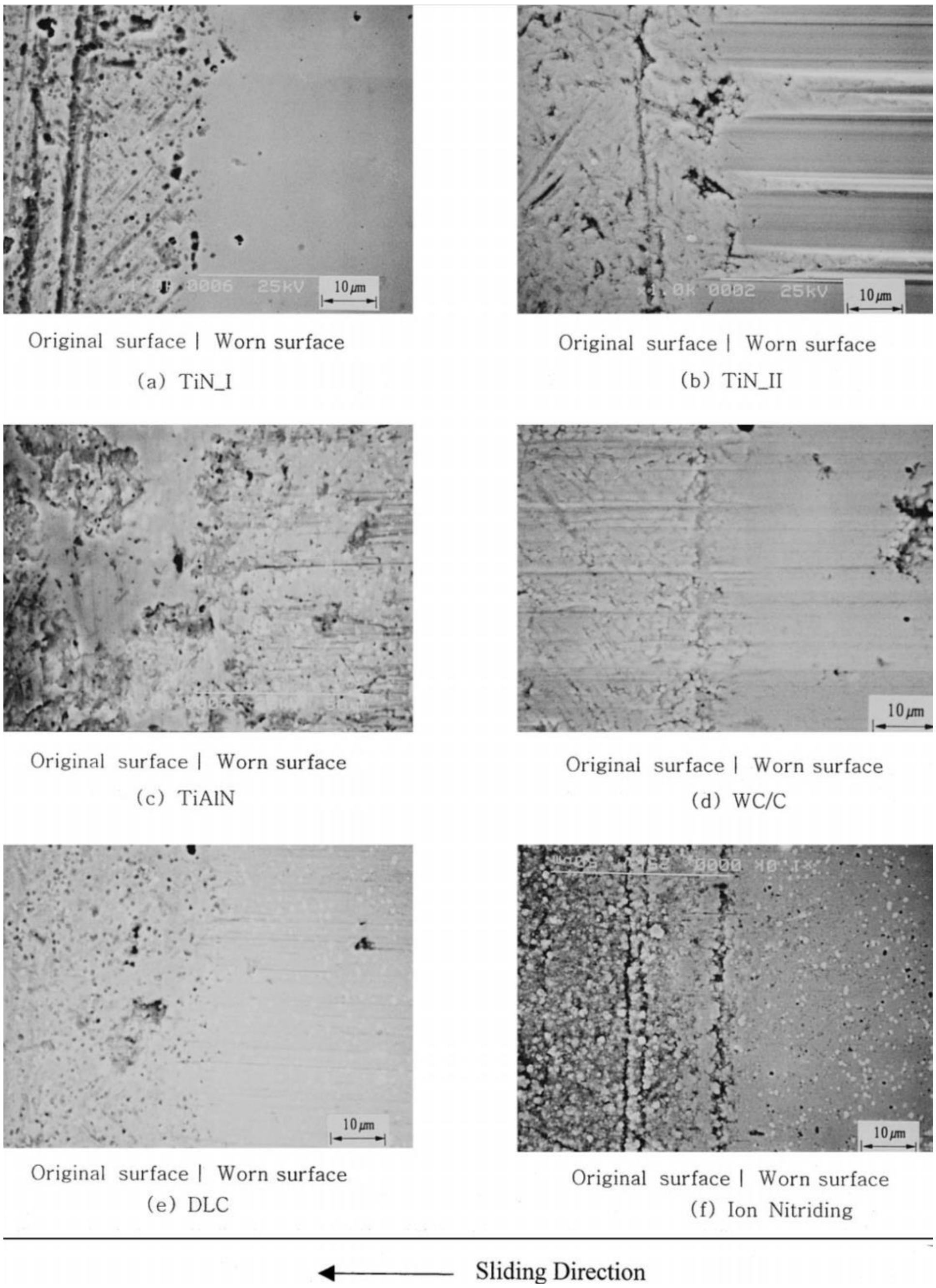
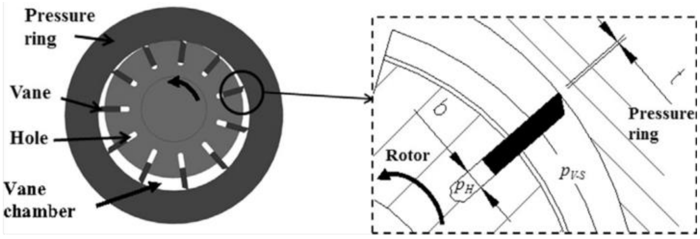


Figure 13. SEM pictures of the wear on the vane tip for various coatings. (a) TiN(I)·; (b) Tin(II)·; (c) TiAIN; (d) WC/C; (e) DLC; (f) ion nitriding in R407C and POE oil mixture tested in normal load of 440 N, speed of 350 rpm, 708C, and tested duration of 10 h. [8]

Hydrodynamic lubrication regime influences on wear optimization using rotary vane compressors

Nowadays the target of the designer consists in the development and implementation of the best ways to make the manufacturing sector, specifically of pumping systems, as energy efficient as possible. In order to help designers to improve the performance of high-pressure vane pumps by reducing the wear, increasing the volumetric efficiency and decreasing the maintenance cost, it is necessary to know what kind of lubrication regime exists in the vane-pressure ring contact, Fig. 14. Thus, the selection of the adequate materials and suitable design methodologies is desired, in order to achieve optimized wear. For example, wear reduction can be achieved by improving surface material performance if boundary lubrication regime. [9].



Figuer 14. Sketch of the pressure ring (stator), rotor, vanes and holes [9]

Described article goes deeper regarding high pressure variable displacement vane pumps and presents a work carried out by the Dept. of Engineering of the University of Ferrara in co-operation with BERARMA s.r.l (Casalecchio di Reno, Bologna, Italy). This kind of volumetric pumps is widely used in machine tools and in hydraulic systems, thanks to their control strategy. High-pressure vane pumps of the PHV series can work in a pressure range from 20 to 250 bar.

In this article, the Archard's wear equation is used to estimate the generalized wear coefficient in order to define the lubrication regime that is established in the sliding contact between vanes and pressure ring:

$$V = \frac{Kx\ FN\ x\ s}{H} \tag{11}$$

Where: V – wear volume [m^3] concerning one of the surfaces, s – is the sliding distance [m], K – is dimensionless generalized coefficient, F_N – represents the normal contact force [N] and H – is the material penetration hardness [N/m^2] of the surface which is worn away.

An analytical model based on elasto-hydrodynamic lubrication (EHD) has been used in order to estimate the minimum lubricant film thickness (h_0) established between the vane and the pressure ring in operational conditions. The minimum film thickness together with the roughness of the surfaces in contact (R_{a1} and R_{a2}) were used for the estimation of λ parameter:

$$\lambda = \frac{h_0}{\sqrt{{R_{a1}}^2 + {R_{a2}}^2}} \tag{12}$$

In the literature, λ parameter is correlated to the type of EHD lubrication. In particular, λ values lower than 1 indicate direct contact between surface asperities, so surface smearing or deformation accompanied by wear may occur. This is the typical situation of boundary lubrication. Values of λ between 1 and 3 indicate that a lubricating film separates the surfaces, but some contacts between the surface asperities are allowed. This type of lubrication is considered in the literature as mixed or partial EHD lubrication. Finally when λ values become greater than 3, full separation of the surfaces by EHD film can be expected. The results of the investigation are:

The best performance in terms of wear and friction is given by the nitriding steel pressure ring, which allows the establishment of the partial elasto-hydrodynamic lubrication in the high load regions (Inlet–Outlet and Outlet–Inlet regions). Coated pressure rings (TiN, DLC) have been selected and used since they guarantee a low friction in dry condition. In the vane-pressure ring contact, they have shown higher wear volume than the nitriding steel, due to the different lubrication regime. The coated pressure rings suffer a greater wear due to the establishment of boundary lubrication. Moreover, the higher hardness of the coated pressure rings (TiN, DLC) caused a wide wear volume on the vanes not acceptable for costumers. The results experienced for the TiN coating pressure ring may be due to the high hardness of the coating and the high roughness of the surface. The results experienced for the DLC coating pressure ring may be due to the high percentage of graphite and the low cohesion between coating and base material. The EHD model used in this research states that a lubrication condition between elasto-hydrodynamic and boundary is established in the sliding contact between vanes and pressure ring in three of the four regions the pressure ring is divided. This is in agreement with the results of the proposed experimental method. In the Inlet region the EHD model cannot be applied since the non-dimensional load parameter lies outside its typical range, due to the low level of normal contact force. [9].

Discussion

Nowadays problem to create maximum efficient rotary vane compressors is very connected to tribological studies of vane and stator (roller) pair. Many researches investigating different coatings for vane surfaces under lubrication conditions in compressed air or refrigeration appliances. In order to reduce an ambient pollution and use rotary vane machines in clean environments It is very important to study also so called oil free rotary-vane compressors and vacuum pumps were carbon vanes are used and investigate non lubricated compressor systems looking for more efficient, low friction and energy efficient vane and stator surface materials. In review article carbon vane, which was mainly composed of Al_4C_3 has been observed to prevent wear and scuffing phenomena during frequent stop-and-start operation of the compressor. [8]. However, its application as a vane material is limited due to its high brittleness and low hardness. It shows that further investigations in oil free compressors are much needed.

Conclusions

1. Developing new rotary vane compressors important attention has to be given to reduce rotation speed because lowering rotation speed friction effect decreases significantly as well as specific power consumption of the compressor, friction losses can be reduced decreasing the blade mass.
2. To restore the volumetric capacity of the machine lowered by the revolution speed reduction, the geometry of the compressor can be modified increasing the axial length of the compressor rather than stator diameter.
3. The TiN coating for the vane surface improves the wear resistance when compared with uncoated original vane in vane-stator geometry almost at all test loads and at different 100 rpm, 500 rpm and 1000 rpm.
4. On a Falex multi-specimen wear tester the TiN coated vane showed good wear resistance properties WC/C coating showed the best tribological performances among all tested coatings while very hard coatings TiAlN and DLC are not suitable since they wear very quickly and produce high friction.
5. The best performance in terms of wear and friction is given by the nitriding steel pressure ring, which allows the establishment of the partial elasto-hydrodynamic lubrication in the high load regions (Inlet–Outlet and Outlet–Inlet regions).

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