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Sealing Performance of a Mobile AC Compressor Shaft Seal

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ABSTRACT

In the last years, environmental concerns such as the destruction of the ozone layer and global warming led to the demand of tighter air conditioning (AC) systems. The compressor has been identified as the most emissive component of the AC system. Due to its complex structure and sealing mechanism, the shaft seal is the leak-prone part. This paper presents an experimental study of several lip seal designs to characterize the gas leakage among contact surfaces and through the polymeric lips. An original experimental setup has been developed in order to measure the gas leakage throughout the seal and to visualize the polymeric lips deformation in standstill mode. Leak rate measurements were carried out by measuring the variation of HFC-134a concentration inside an accumulation volume using an infrared gas spectrophotometer. The gas pressure and the shaft surface roughness impact on those results, as well as the presence of lubricating oil in the sealing zone are discussed.

1. INTRODUCTION

Air conditioning compressors in mobile applications are known to be the most leak prone components (Clodic and Yu, 2006). The Center for Energy and Processes (CEP) has been measuring, since many years, the refrigerant leak flow rate of mobile air conditioning (MAC) systems and the contribution of each component to the system high emissive rate. It has been concluded that the compressor represents 50 to 70% of refrigerant losses for new MAC systems, which corresponds to 5 to 7 g/yr. This result clearly shows that improvements in the compressor tightness will lead to significant reduction of the MAC system gas emissions and its impact on the environment.

The MAC compressor is externally driven by the vehicle engine through a belt pulley mechanism. The different compressor blocs are joined together using different seal types: o-rings, gaskets, and a shaft seal. The shaft seal, which allows the transmission of the rotary motion to the compression mechanism, is known to be a complex structure and the leak-prone part. Its design has been changing and improving gas and lubricating oil tightness. Until 1986, the mechanical seal has been commonly used as a sealing device for compressors in mobile applications. High leakages occur under excessive wear and high refrigerant pressure resulting in a high gas leakage through the seal gap. A new shaft seal design has been introduced by Ohtaki *et al*. (1986): the lip type seal. This new seal design incorporates two polymeric lips that permanently contact the shaft surface. This compact seal represented a benefit in terms of lubricating oil and gas leakage and, therefore, preventing compressor from breaking down. (Shimomura *et al*., 1989)

This shaft seal design can only be found in MAC compressors and has been studied by other authors, more specifically in terms of oil and refrigerant leakage in running mode, although, the running mode represents a small part of a vehicle life, 2 to 5% in Europe. Furthermore, the compressor is not always running with the vehicle engine, except for a clutchless type, thus the compressor running mode time is lower than the vehicle engine time. In addition, the European Regulation 706/2007 requires the leak test of new MAC systems in standstill mode. This paper presents the experimental analysis of the refrigerant emissions through several lip type seal designs in standstill mode. Five new shaft seals have been tested under dry and lubricated conditions. A transparent shaft has also been used to understand the shaft seal behavior under pressure.

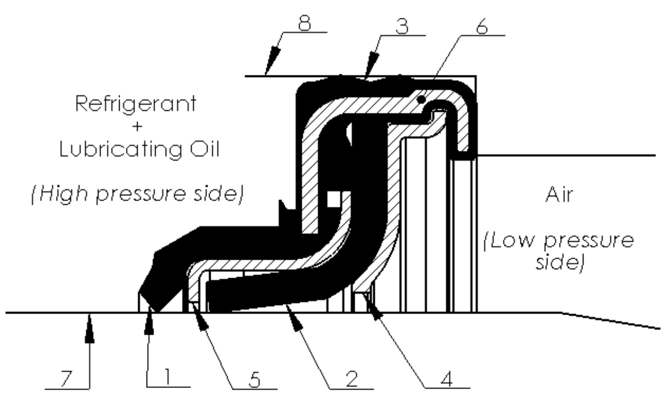
2. SHAFT SEAL DESCRIPTION

The compressor can be considered as a pressure container that is permanently under pressure, thus resulting in a continuous gas emission through seals. When the compressor is running, the pressure is nearly equal to the evaporating pressure (varying from 300 to 400 kPa). On the contrary, when the engine is off, the pressure in the compressor is rapidly equal to the saturating pressure corresponding to the ambient temperature, for example 1020 kPa at 313 K. Therefore, the pressure difference responsible for the compressor refrigerant emissions is larger in standstill mode than in running mode.

Shaft seals are designed in order to limit emissions when rotating, thanks to the lubrication mechanisms and the lip seal shapes. However, the shaft seal of a MAC compressor performs more than 90% of the year as a static seal.

The first automotive compressors were equipped with a mechanical face seal, which was also used as a seal for pumps. Nevertheless, this seal type presented an important deterioration of the matting surfaces and, consequently, leaking troubles occurred with time (Thomas, 1982). A polymeric shaft seal was introduced by Ohtaki *et al*. (1986) and was exclusively employed in the MAC system compressor. This new shaft seal was composed of one rubber lip ring that faces the fluid (refrigerant + lubricating oil) followed by one polytetrafluoroethylene (PTFE) lip ring in the atmospheric side and assembled in a metallic structure. As reported later by Shimomura *et al*. (1989), this lip type seal represents an important reduction of fluid leakage in running mode.

A lip type seal is generally composed of four main parts (Figure 1). A rubber lip ring *1* at the high-pressure side, a plastic lip ring *2* at low-pressure side, both contacting the shaft surface *7,* and peripheral rides *3*, in some cases an O-ring, that contact the seal housing surface *8* are all put together in a metallic structure 6 and 4 to rigidify the assemblage. The rubber lip has a sharp edge to increase the contact pressure on the shaft surface reducing and preventing leakage when the shaft is not rotating. When the shaft rotates, the fluid leaks to the plastic lip and the dynamic sealing is achieved by the elastohydrodynamic effects in the interface.



1- Rubber lip ring	**3**- Peripheral rides	**5**- Rubber support ring	**7**- Shaft surface
2- PTFE lip ring	**4**- Backup ring	**6**- Metallic structure	**8**- Seal housing surface

Figure 1: Sectional view of a MAC compressor lip type seal.

Different shapes of the rubber lip have been designed to control the contact pressure with the shaft and, so, the leakage in standstill mode, as well as to allow the lubrication of the plastic lip in running mode. Yamada *et al*. (2003) introduced irregularities on the rubber lip edge to guide the fluid through the contact with the shaft surface. Hosokawa *et al*. (2002) and Osako *et al*. (2003) redesigned the rubber lip to reduce contact pressure. The plastic lip can also be found with spiral grooves to improve fluid tightness in running mode (Yamada *et al*., 2003) and Hiromi *et al*. (1999) introduced a rubber support ring to prevent the rubber and plastic lip rings to be pressed with excessive force against the shaft. These are some examples of lip seal design improvements. Using the testing system developed at the CEP laboratory different designs have been tested and compared.

The shaft sealing system mainly depends on maintaining an adequate interference among the elastic contacting surfaces throughout the life of the seal. The sealing force is obtained by the fluid pressure and the shaft seal leakage results from two different phenomena:

1. fluid flow through the contact lip rings/shaft surface and peripheral rides/seal housing surface, and
2. permeation of materials.

In the first case, the fluid will diffuse through the gaps in the sealing interface that result from the surfaces manufacturing process. Manufacturing clearances are compensated by the polymers elastic deformation. However, the surface roughness is only partially eliminated as contact pressure increases. By defining the leak path dimension, the surface roughness has a remarkable influence in the sealing performance.

Permeation of materials should also be considered as a sealing parameter, since it includes the transport of the refrigerant across the seal polymeric materials. For the polymeric surface under pressure the permeation contribution is small, as it will be shown in the following tests.

3. RUNNING MODE VS. STANDSTILL MODE

Two long time run-in and two brand new compressors have been leak tested in running mode. It has been observed that run-in compressors perform better than new ones indicating that the oil film formation in the shaft seal contact zone is improved with shaft surface wear tracks (Sousa and Clodic, 2007). It has also been noticed that the presence of lubricating oil in the shaft sealing zone, when the shaft is not rotating, can reduce emissions by about 50%, but only in the first 12 hours of the test. This phenomenon will be studied later on in this paper.

In order to evaluate the impact of the shaft seal in the compressor overall emissions, 24 new compressors have been leak tested according to the European regulation 706/2007. Emission levels, without the Correlation Factor, ranges from 5 to 32 g/yr with an average value of 14 g/yr. Aged compressors coming from two end-of-life vehicles have been tested in the same conditions and gas emissions were higher than 180 g/yr, which demonstrates that the mating faces wear significantly increases emissions in standstill mode.

The seals and the shafts used in this study are brand new and brought from an Original Equipment Manufacturer (OEM) as spared parts.

4. EXPERIMENTAL SETUP

An original experimental setup has been developed to leak test the lip type seal in standstill mode and is based on the gas concentration rise method (Figure 2). It is composed of two stainless steel chambers: the refrigerant chamber and the accumulation volume, which are separated by the shaft seal. First, the refrigerant chamber is evacuated to 10 Pa, the accumulation volume is filled up with N_2 / O_2 (80 / 20%), and then the shaft seal is put under pressure by an external refrigerant boiler that is heated to control the gas pressure. All the refrigerant that passes through the seal will be stocked in the accumulation volume and measured by an infrared gas analyzer. The seal housing is heated just above the saturating temperature to avoid refrigerant condensation in the shaft seal. One of the advantages of this method relies on the continuous recording of gas concentration allowing to easily point out any change in seal tightness.

The gas transfer from the high-pressure side to the low-pressure side, at atmospheric pressure, will be recorded by the gas analyzer and plotted in a gas concentration vs. time curve (Figure 3) to determine the shaft seal emission level. The seal mass flow rate is defined as the quantity of gas flowing per unit time into the accumulation volume, as indicated in Equation (1):

$$\dot{m} = \frac{V_{av}}{V_{mol}} \frac{\partial C}{\partial t} M \qquad (1)$$

where $\dot{m}$ is the seal mass flow rate, $\frac{\partial C}{\partial t}$ is the variation of gas concentration along the time, V_{av} is the free accumulation volume, V_{mol} is the molar volume and M is the molar mass .

Before leak tests, the infrared gas analyzer is calibrated using three different refrigerant concentrations: 0, 100, and 450 ppm, and the accumulation volume is calculated by means of a calibrated leak (Figure 3). The number and accuracy of measurement devices are presented in Table 1. The mass flow rate measurement relative uncertainty is about 6%.

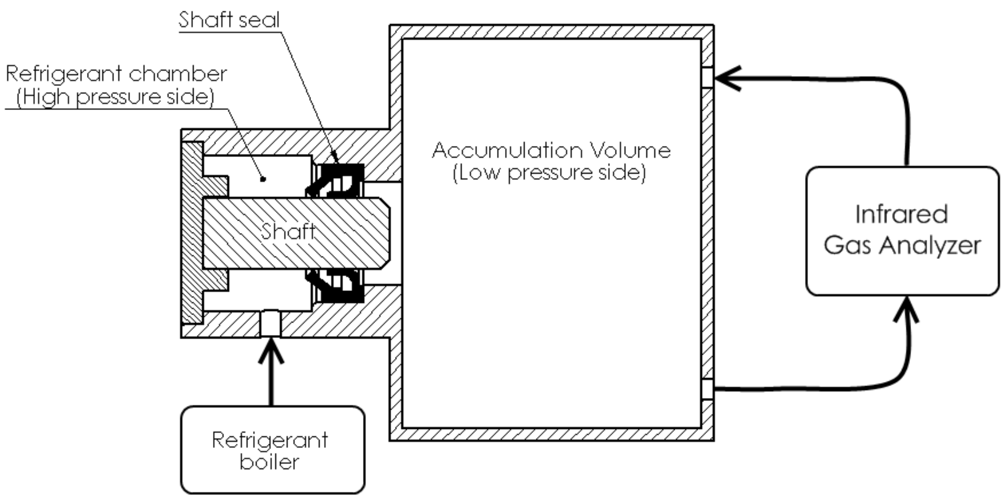


Figure 2: Schematic arrangement of the shaft seal experimental setup.

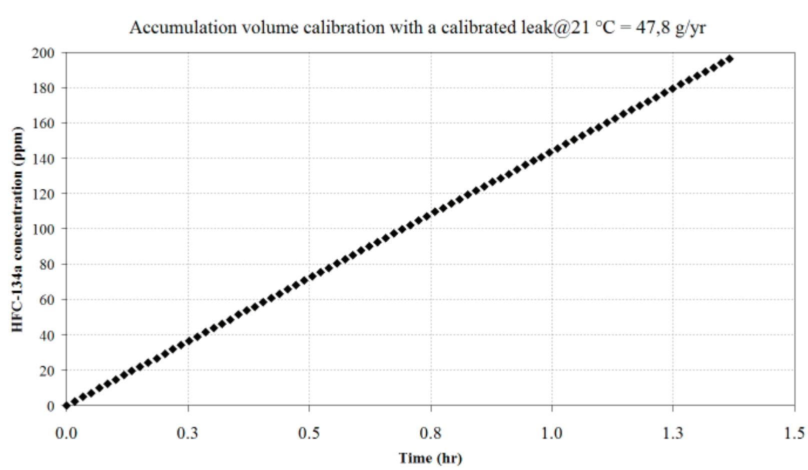


Figure 3: The accumulation volume calibration by the gas concentration rising method.

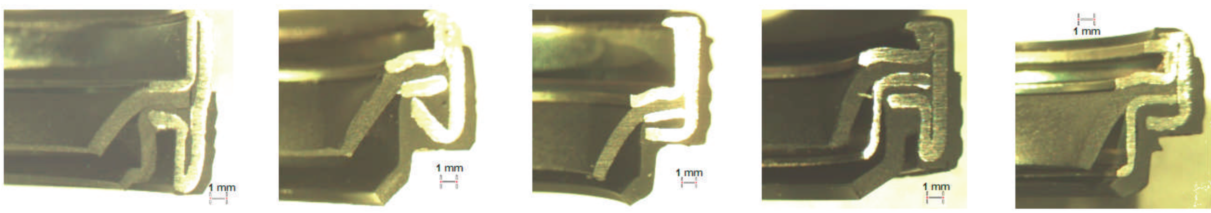
Table 1: Measurement devices.

Device	Number	Accuracy
Gas analyzer	1	± 0.5 ppm
Thermal resistance	3	± 1 K
Pressure sensor	2	0.1%

5. LEAK TEST RESULTS

Five different seal designs have been brought from an OEM as spared parts, samples A to E (Figure 4). They all have a rubber and a plastic lip and for samples D and E the lip rings are separated by a metal ring that limit rubber lip deformation. The shafts used for these tests have been taken from new compressors also brought from OEM suppliers. According to the manufacturer instructions, samples A, B, C, and E have been tested with a 0.0142 m shaft and sample D with a 0.0135 m shaft.

The five new shaft seals have been leak tested in standstill mode and with a dry contact at three different refrigerant pressures (Figure 5). The test results show sample emissions ranging from 1.3 g/yr at 764 kPa to 16.7 g/yr at 1320 kPa. The sample E is clearly the tightest shaft seal with only 3.4 g/yr at 1323 kPa. This test clearly shows that by reducing the rubber lip ring deformation and, thus, maintaining a high rubber lip contact pressure, it is possible to substantially reduce the mass flow rate through the shaft surface roughness.



Sample A Sample B Sample C Sample D Sample E

Figure 4: New compressor shaft seals tested in standstill mode.

Then Sample A has been leak tested with a lubricated contact. The shaft surface has been coated for each test with a constant Polyalkylene Glycol (PAG) oil film and inserted in the seal. The rising of refrigerant concentration is plotted in the gas concentration vs. time curve (Figure 6). It can be seen that concentration rises in two steps: the first step corresponds to the presence of the oil in the contact that limit gas leakage and the second one to the rapidly increase of emissions due to the absence of the lubricating oil in the contact. The time of the first step increases with the decrease of refrigerant pressure. The emissions for these two steps are plotted in Figure 7 proving that the gas barrier effect of oil is very significant. The second step corresponds to the emissions previously found with a dry contact (Figure 5), meaning that all lubricating oil present in the contact zone is forced by the refrigerant to exit the contact (Figure 8).

A third test has been performed on Sample A with a transparent shaft made of polymethyl methacrylate (PMMA) with a dry and a lubricated contact (Figure 9). The test results show that, for a polish surface, the gas emissions are significantly lower than for the original metallic shaft (Figure 5) and the lubricating oil in the seal contact has no effect on emissions level. There is also no transition phenomenon as reported for real shafts, which means that the oil present in the shaft surface does not traverse through the contact and then the shaft surface roughness can be neglected.

The PMMA shaft can be considered having a perfect smooth surface and then the gas emissions measured come from the permeation of polymers and the contact between the seal and the housing.

The transparent shaft has a central hole that allows the installation of a visualizing system. An endoscope is inserted in the shaft hole, coupled with a light source and a camera in order to visualize the contact between the lip rings and the shaft. The images show an important rubber lip deformation as refrigerant pressure increases and a lip movement that significantly depends on the friction factor. The smallest rubber lip contact surface is obtained for seal Sample E, which is the tightest one. The lip rings contact observation will be developed in future work.

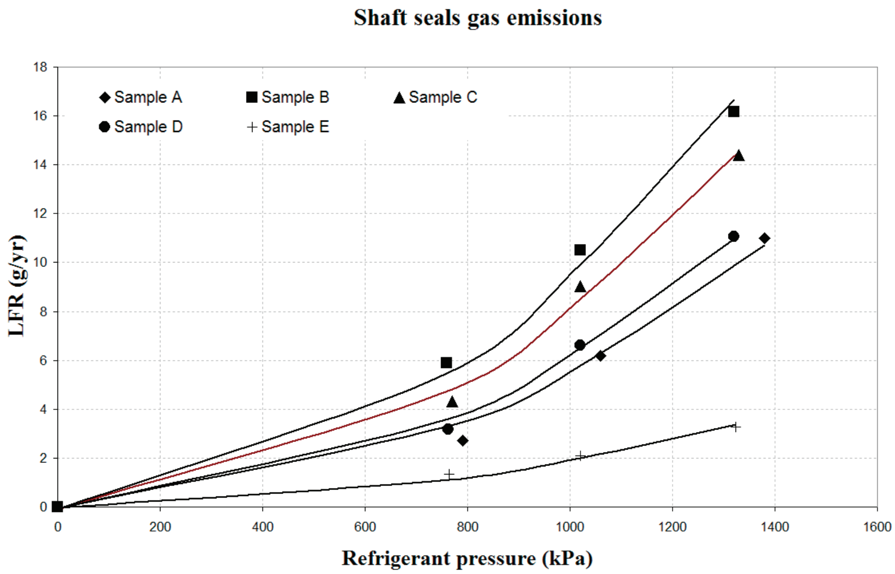


Figure 5: The shaft seals gas emissions in standstill mode with a dry contact.

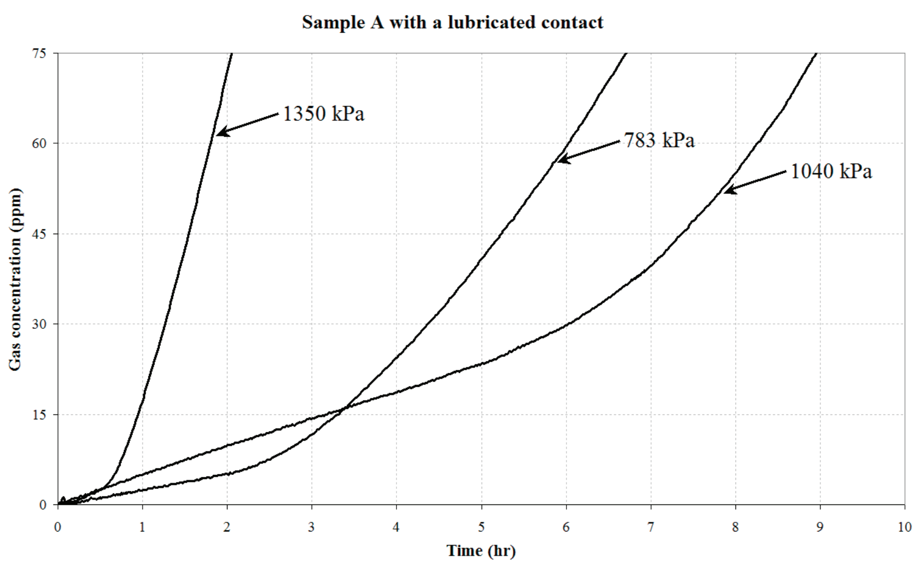


Figure 6: Sample A gas concentration in the accumulation volume with a lubricated contact.

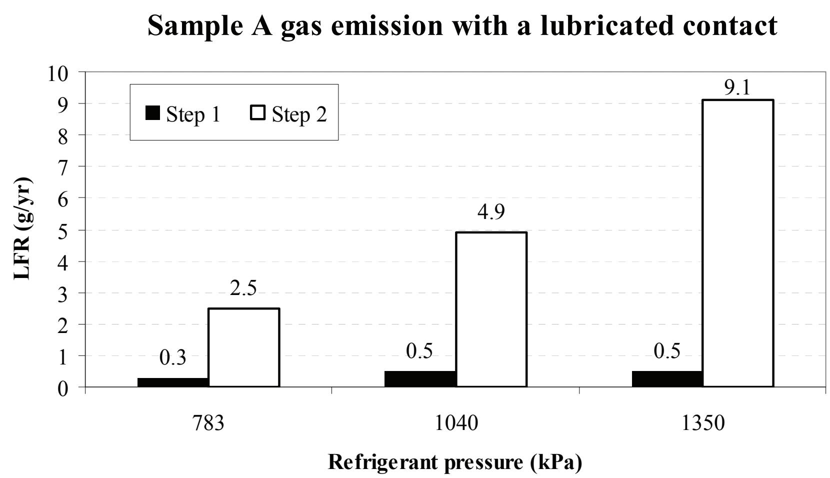


Figure 7: Sample A gas emissions in standstill mode with a lubricated contact.

Figure 8: Sample A lubricating oil deposit at the low pressure side after a leak test in standstill mode.

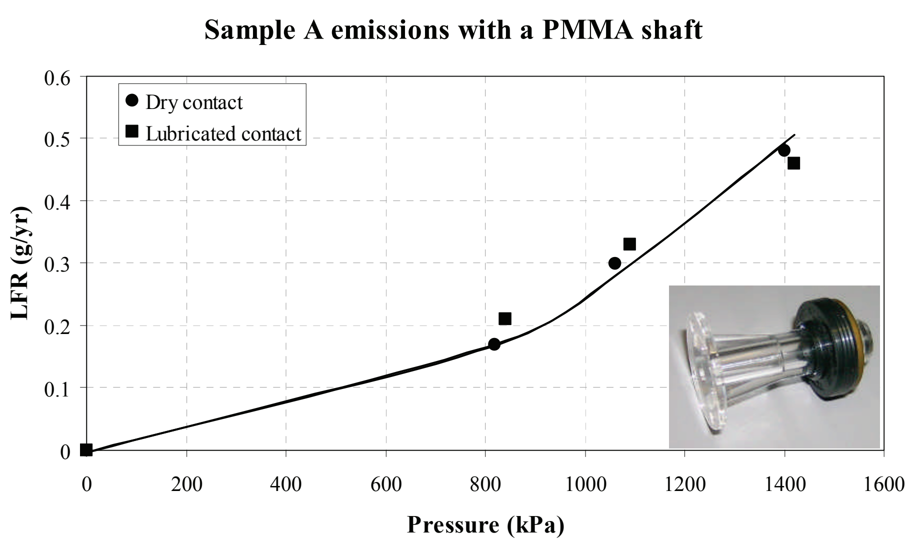


Figure 9: Sample A gas emissions with a transparent PMMA shaft.

6. CONCLUSIONS

Five new MAC compressors shaft seals have been leak tested in standstill mode. By testing these seals with the original shafts, it was possible to determine their impact on compressor gas emissions. Therefore, an emission average value of 7 g/yr for 1020 kPa has been found, which represents 50% of compressor emissions for new compressors in standstill mode proving the great interest of the shaft seal leak study. The seal Sample E has proved to be the tightest one, due to a design that improves rubber lip contact pressure

When the shaft is not rotating, the presence of lubricating oil in the shaft seal contact zone drastically reduces emissions but this is a temporally effect, since the oil is forced out by the refrigerant pressure and the emissions level rises and becomes identical to the one measured for dry contact.

The use of a PMMA polish shaft allowed to leak test the shaft seal with a negligible shaft surface roughness effect. Accordingly, the emissions due to polymer permeation and to the contact between the seal housing surface and the seal peripheral rides are about 0.3 g/yr at 1020 kPa. Therefore, emissions coming from the contact between the lip rings and the shaft surface represent 95% of total shaft seal mass flow rate. These test results show that the surface roughness has a remarkable influence in the sealing performance and explain the compressor high emission levels as seal mating surfaces wear goes on.

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