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The Current Sealing Performance and Future Developments of Edge Welded Metal Bellows Mechanical Seals in Industrial Refrigeration Compressors

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THE CURRENT SEALING PERFORMANCE AND FUTURE DEVELOPMENTS OF EDGE WELDED METAL BELLOWS MECHANICAL SEALS IN INDUSTRIAL REFRIGERATION COMPRESSORS

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ABSTRACT

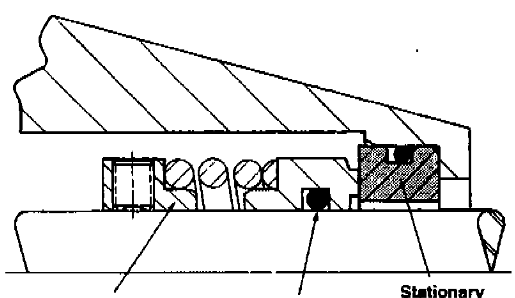
This paper covers the general performance capability and limitations of the various types of mechanical shaft seals in use today in refrigeration compressors and describes the benefits gained by the change to edge welded metal bellows seals. Additionally, new developments and enhancements to these seals including techniques for future environmental control are outlined.

INTRODUCTION

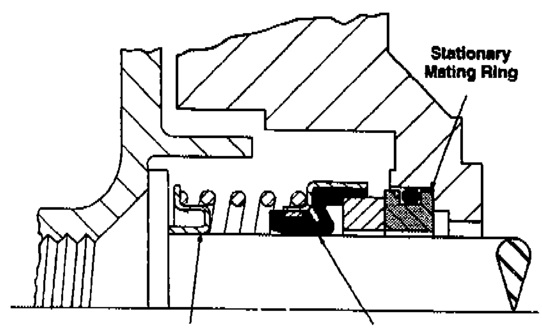
In the past most types of refrigerant compressors were sealed by simple "pusher" (fig.1) or rubber bellows/diaphragm mechanical seals (fig.2). These were produced by the compressor manufacturer or commercially sourced. However, beginning in the mid 1980's with the market demand for a higher performance seal, the convention in seal selection began to change to the point today where edge welded metal bellows seals have become the preferred sealing device.

Edge welded metal bellows seals have been qualified for use on a number of piston, screw and vane compressors throughout the European and North American refrigerant industry. The history of the usage of welded metal bellows seals has moved on from initial qualification during the mid to late 1980's to full production programs over the last 6 or 7 years.

The conditions under which the seals work vary with compressor type and size but are typically shaft diameters from 25–100 mm (1–4"), running at up to 3600 r.p.m. and handling up to 25 bar (375 psi) at up to 120° C (250 °F). In most instances a single seal is installed and the fluid to be sealed is a mineral or synthetic oil containing up to 25% refrigerant gas such as C.F.C.s, H.C.F.C.'s and Ammonia.



(fig. 1) Simple "Pusher" Seal



(fig. 2) Rubber Bellows Seal

GENERAL PRINCIPLES OF SEALING

Mechanical face seals are the most versatile type of seals for rotating shafts. Their performance depends upon mechanical stability, axial load and lubricating efficiency. Standardized products are required to perform under a wide variety of operating conditions and to run a number of years without maintenance while being able to withstand occasional system upsets. In its simplest terms a mechanical seal is a compromise between face wear and leakage rate. The sealing faces ride on a thin film of lubricant which seals as well as providing lubrication. Short term leakage will occur when a closing force imbalance produces a condition where the lubricating film is too thick. Conversely, short term wear will result from an imbalance which will not support the film, leading to heavy contact between the faces. Therefore seal performance and reliability depend on seal design, choice of friction face materials, choice of elastomeric secondary seal materials, maintaining a suitable seal environment and quality of fitting.

To optimize these factors required a detailed and comprehensive development testing program to establish long term seal performance and to improve upon the deteriorating leakage performance with time so commonly experienced with other seal types. Leakage rates and wear values of seals are difficult to measure and predict accurately and these figures are normally based on statistical history and O.E.M. qualification test programs. However, as a reference for establishing the factors which influence leakage, the equation shown below is normally cited:

$$Q = \frac{C \times d_m \times h_o^3 \Delta P}{\zeta \times b}$$

(Ref. 1)

<u>Where</u>

Q is leakage flow rate

C is a constant dependent on units

d_m is the mean diameter of the interface

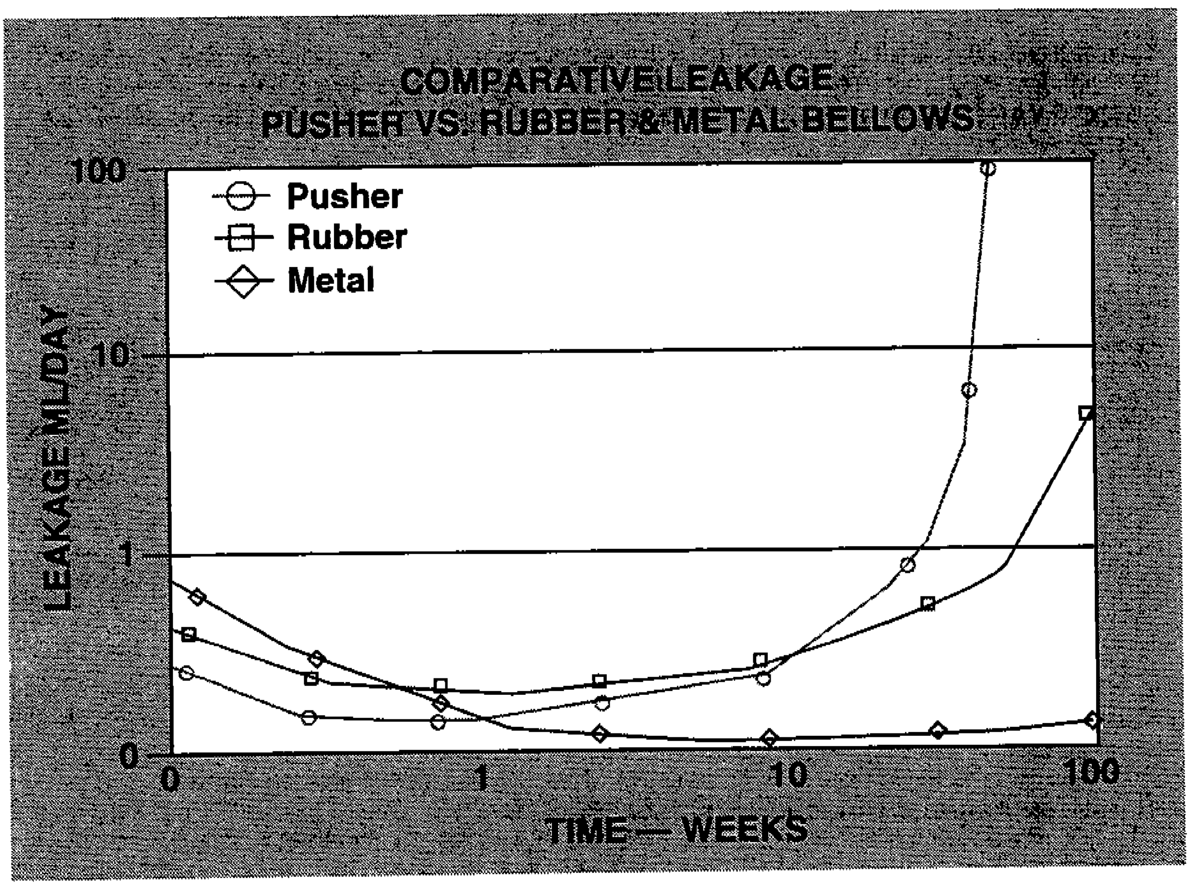
ΔP is the differential between sealed and atmospheric pressure

h_o is the separation of the interface

ζ is absolute viscosity

b is width of interface

From the formula it will be seen that the critical factor is the separation of the interface, h_o; the reason for the difficulty in analytical prediction of leakage is the uncertainty of this dominant factor. The interface gap is dependent upon actual load, fluid properties and the inevitable microscopic distortions of the faces due to thermal and mechanical effects. Additionally, face material properties will affect the thermal conditions, as will the structural design of the seal. However the principle uncertainty is seal deterioration over time. While many seal designs will provide good initial seal performance, the long term reliability is often unpredictable (fig.3).



(fig.3)

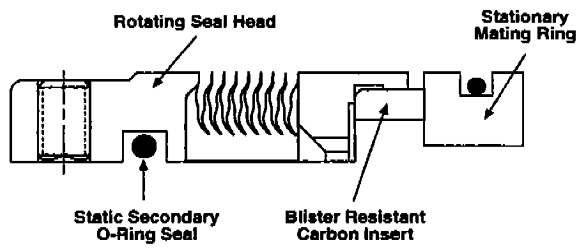
BENEFITS OF EDGE WELDED METAL BELLOWS SEALS

A typical metal bellows seal for use in a refrigerant compressor is shown in fig. 4. Materials of construction are typically high alloy stainless steel for all the rotating metal components with a blister resistant carbon face running against a premium grade cast iron seat. The static secondary seals in the seal and seat are usually of a proprietary supply. The selection of the face, seat and elastomeric materials combined with the flatness and surface finish of the faces are critical for long term performance and reliability. The principle features of edge welded metal bellows seals can be summarized as follows:

a) Low axial operating face load.
b) Inherent hydraulic balance.
c) Evenly distributed face load.
d) All secondary seals are well shrouded static applications.
e) Low profile envelope.

These inherent design characteristics result in the following benefits:

1) Elimination of secondary seal hysteresis.
2) No fretting/ micro wear damage to the shaft.
3) Lower face wear even under marginal lubrication conditions.
4) Lower long term leakage.
5) Easily adapted to existing compressor designs.



(fig. 4) Metal Bellows Seal

HISTORY OF EDGE WELDED METAL BELLOWS SEALS

Welded metal bellows have been used as sealing elements in mechanical seals, valve stems and other equipment since 1950. They were originally developed for the aerospace industry, in particular for accessories and aero–engine main shaft seals. In these industries they have been used for their integrity, reliability, toughness and high temperature resistance. Operating conditions ranged from 150,000 r.p.m., 50 bar (750 psi) and –250 °C to +600 °C (–420 °F to 1110 °F) in separate duties.

In the 1960's these aerospace developed products were adapted for general industrial and process applications. The driving force was to produce highly standardized, higher volume off–the–shelf components which would combine high performance with cost effectiveness. Thus a small range of products were able to meet a very wide spectrum of operating conditions, mainly for use in pumps. The main focus at this time was on high temperature applications higher speed equipment along with a wide variety of materials to handle a range of corrosive conditions. (Ref. 2) Currently, edge welded metal bellows are available in stainless steels, super stainless steels, precipitation hardened stainless steels, Inconel for high temperatures and Hastelloy or Monel for more aggressive products.

In 1984 the first metal bellows seal was applied into a refrigeration compressor with successful results. Continued acceptance of the technology in the industrial refrigeration compressor market has led to high volume production, allowing for standardized core materials and designs, which in turn has produced cost reductions. Slight modifications to standard machined seal components allow for the seal to be applied into many different machines and configurations.

CURRENT SEAL PRACTICE AND DEVELOPMENT

An earlier paper (Ref. 3) described how the original driving force for better seal performance was initiated by compressor operators, resulting in a number of qualification test programs instigated jointly by the compressor and seal manufacturers. The seal manufacture's in–house test programs concentrated on seal design and material optimization for best performance consistent with costs acceptable in the market place. The type approval tests by the compressor manufacturers were carried out as a fitness–for–purpose evaluation at the extreme operating conditions of their machines, including the most difficult gas/oil combinations. It was found in almost all cases that one basic combination of materials gave the best results.

Due to the requirement for long term reliability, product acceptance by the compressor manufacturers was based upon a long and rigorous test regime. This included design evaluation, manufacturers test bed evaluation and, after audit review, any necessary fine tuning of the seal and seal environment, field trials and final acceptance for production.

Initially this procedure took 2–3 years but growing confidence, as a result of experience gained from the many thousands of production seals in field service, has shortened this development time. This experience has led to the current market position where metal bellows seals are rapidly becoming the industry standard in Europe and are quickly gaining acceptance in North America. Each new application is the result of close co–operation and exchange of engineering skills between compressor and seal manufacturer. Some typical operating conditions where metal bellows seals are operating are listed in fig.5.

Figure 5

REFRIGERANT	OIL	TEMP °C/°F	PRESSURE BAR/PSI
R12	FUCHS KES 100	130/266	20/300
	VG 46 – VG 100	80/176	25/375
R22	FUCHS KES 100	130/266	15/225
	SUNAISO 46	50/122	12/180
	VG 46 – VG 100	80/176	25/375
AMMONIA	M68 MOBIL ARCTIC 300	70/158	18/270
	P68 SHELL CLAVIUS G68	70/158	18/270

OIL LEAKAGE RATES ACCEPTABLE TYPICALLY:	0.05–0.25 ml/hr
OIL LEAKAGE RATES ACHIEVED TYPICALLY:	0.01 – 0.05 ml/hr
GAS LEAKAGE RATES ACHIEVED TYPICALLY:	5–25 gram/yr (0.2–0.9 oz/yr) (depending upon the compressor type and operating conditions)
TYPICAL SEAL LIFE ACCEPTABLE:	3 YEARS
TYPICAL SEAL LIFE ACHIEVED:	5+ YEARS

FUTURE DEVELOPMENTS OF METAL BELLOWS SEALS

As a direct result of continuing close liaison between customer and supplier, further development and higher performance of shaft seals is being market driven and this will undoubtedly lead to improvements and enhancements in seal design. The most obvious development in the refrigeration industries is the requirement to comply with the 4th meeting of the Montreal Protocol in November, 1992. Thus far this has not given great problems to the seal manufacturer. For example, where existing machines have been adapted to handle ammonia and an appropriate oil, this mixture tends to shrink current secondary seal elastomers. The solution is the proper selection of secondary elastomers and a modified mechanical design. All recently designed seals include these simple revisions.

Many manufacturers are considering Ammonia as the most readily available environmentally friendly gas from the stand point of zero global warming and ozone depletion, but there are considerable operational limitations and safety requirements. The development of substitute refrigerants such as R22 and R134A are being widely considered and so far have not given problems at the seal. Any requirement for higher temperatures at the seal cavity can be met without modification to the current seal design, with the exception of the proper selection of secondary elastomeric seals. Currently used elastomers are successful up to 90–95° C (195–205 °F) and Fluorocarbon elastomers have been used at temperatures to 120° C (250 °F). Any effects to mechanical properties and swell is not usually a difficult problem because of the static **only** sealing mode of secondary elastomers in bellows seals.

An elastomer, resulting from the hydrogenation of nitrile rubber, commonly known as HNBR is available for temperatures of –40° C to +150° C (–100 to +300 °F) . While HNBR has not been applied as yet to seals for industrial refrigeration compressors, it is a potential candidate. It has been adopted by three major European car manufacturers for use as "O" rings in air conditioning units containing R12 with mineral oil and R134A with polyalkylene glycols (PAGs) and polyolester oils.

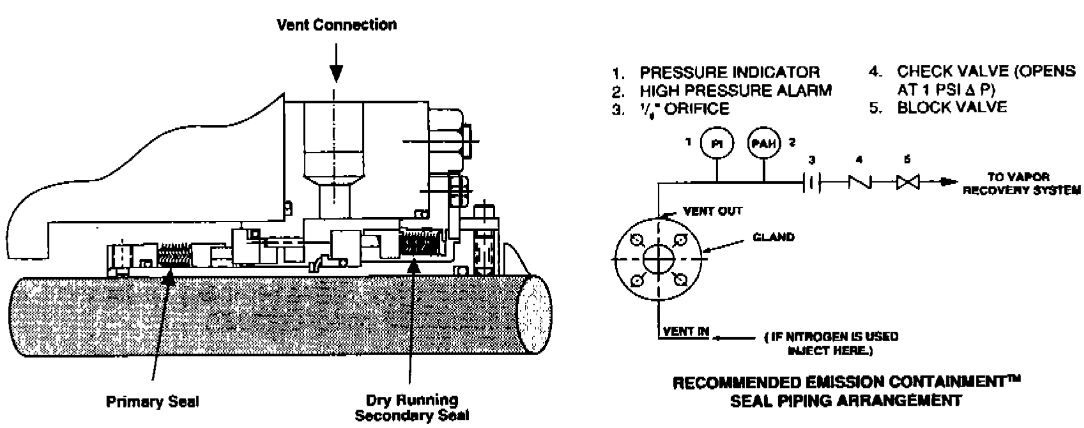
There has been a small but significant requirement for edge welded metal bellows seals to handle dynamic pressures beyond the capabilities of a single ply bellows configuration (20 bar, 300 psi). A two ply bellows design roughly doubles the pressure capability and such seals have been supplied successfully for a number of years as standard items in the petrochemical and process industries. The higher face loads associated with the higher pressures may require hard face materials such as silicon carbide but this again is common practice. Test seals are being actively considered at the present time by one or two compressor manufacturers.

Similarly, requests for higher shaft speeds up to 50 m/s (10,000 rpm) can be catered for by using a well proven solution. In order to reduce the stresses on the metal bellows and rotational dynamics, the seat is designed to rotate with the shaft while the bellows seal is fixed to the casing of the compressor. (Ref. 4) Once again hard faces may be required, and as with higher pressures continuous and adequate lubrication of the seal faces will require careful design consultation between supplier and customer. Here again some actual testing is underway.

At least one compressor manufacturer is considering operating without a separate lubricating oil pump and a test rig was set up using a metal bellows seal with the usual materials of construction, running without oil for 5 minutes before being fully lubricated. A further requirement was for the seal to run 30 minutes lubricated then drained and then run "dry" for 30 seconds before oil was re–introduced. This cycle was repeated 50 times and throughout the trials the dynamic pressure on the seal was 10–12 bar (150–180 psi). The seal performed well with minimal evidence of increased wear of the carbon and the results were encouraging enough to consider full scale compressor trials.

The necessity to meet ever more stringent emissions control legislation gives rise to the latent demand to reduce leakage rates of both oil and gas. To meet this environmental need and to reduce the cost of replacing expensive refrigerant gases lost to the atmosphere a dual seal employing a dry–sliding secondary containment seal (see fig. 6) is already available. The design concept was developed in Europe as the Secondary Containment Seal in the mid eighties. The driving force at this time was the safety of plant personnel from the leakage of dangerous and flammable products rather than global environmental concerns.The design featured a secure secondary seal capable of taking full process pressure in the event of a primary seal failure. Up until that time many refineries and chemical plants fitted a throttle bushing or tandem seal with a liquid barrier system to reduce the effects of disastrous leakage. Field testing began in 1986 and from this a reliable safety seal was made available to industry.

As environmental regulations in the USA and throughout the world became more stringent, the design was adapted to produce an off the shelf dry–sliding secondary containment seal, the Emission Containment Seal (ECS), which provided emergency back–up for the primary seal and contained the normal seepage of either liquid or gas which passed across the primary seal faces. Field prototype testing of this improved design was accelerated in Europe and in 1990 extended into the USA where a propane test rig was installed to allow rigorous in–house testing. (Ref. 5)



(fig. 6) Emission Containment Seal & Typical ECS Piping Plan

This new technology incorporates two main design features:

1) A special edge welded metal bellows core which allows the faces to run dry for a minimum of 3 years continuous service. Unlike other designs currently available, the Emission Containment Seal (ECS) uses proven bellows seal technology with static elastomers which provide the more consistent control of face loading, essential for obtaining low face wear rates in dry running seals.
2) Face materials which enable the seal to run dry without liquid lubrication. The proprietary grade of carbon running against a special grade of silicon carbide proved after a comprehensive and demanding test program to be the optimum combination to reliably give a minimum of 3 years continuous service.

The seal is a cartridge arrangement capable of either wet or dry running and will handle a mixture of liquid and gas which can be vented to a recovery system or a storage container. Alarm systems can be fitted to indicate that leakage across the primary seal has reached an unacceptable level and action should be taken without the environment being under immediate threat. This allows preventative maintenance to be carried out in an orderly manner.

Since its introduction as a standard product in late 1991, nearly one thousand of these seals have been delivered and are operating successfully. With so much experience it is a short step to adapting this technology to a refrigeration compressor application. This is currently being carried out by a major compressor manufacturer in North America.

CONCLUSIONS

The edge welded bellows seal has successfully proven over the last decade to be a highly effective and economic solution for shaft sealing of a variety of industrial refrigeration compressors and is widely accepted under current performance requirements. Its superiority to previous seal types is well established and any foreseeable further performance developments or conditions imposed by current or future legislation can be readily accommodated by the adaptation of existing proven bellows seal technology and designs.

The close co–operation between the compressor and seal manufacturers already established is likely to develop further as seal requirements change. It should be also noted that the international nature of the refrigeration industry means that these type of partnerships will be increasingly required to cross national boundaries in order to achieve optimum seal performance for this growing and changing worldwide marketplace.

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