ABRADABLE SEALS IN TURBOMACHINERY

by

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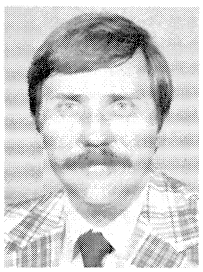

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INTRODUCTION

All turbomachines share at least one common trait: they all handle some type of fluid. Since fluids have a rather insidious inclination to find flow paths other than those intended, the subject of fluid leakage and its control through the use of sealing devices has always been of importance to the rotating equipment designer. The use of seals in turbomachinery dates back to the first pumps, turbines and compressors of nearly a century ago.

Since seals have such a long history, it might be assumed that the subject was a static one, and readers may question why there is still so much interest in seals. The answer lies in two words — energy and efficiency. Rising energy costs of the last few years have put greater emphasis on the improvement of machinery efficiency, including reduction of leakage rates through improved seal designs. The desire to improve efficiency by reducing leakage has led to the development and use of abradable seals. In an abradable seal the rotating element is actually allowed to wear into the stationary sealing element, which is made of a material that can tolerate abrasion. This technique permits reduced clearance between the rotating and stationary parts of the seal, and the reduced clearance results in lower leakage rates across the seal.

In a seal which is used to contain the fluid within the machine, this lower leakage rate obtained with abradable seals can be viewed as a reduction of throughput losses. While in a seal used to separate internal parts ,the reduced leakage cuts down on internal recirculation. In either case, the end result is improved overall machine efficiency.

The intent of this paper is to describe various aspects of the application of abradable seals to industrial turbomachinery. Topics covered include sealing locations, materials, comparative studies and examples of the use of abradable seals. The paper will illustrate the use of the abradable seal as an energy saving device.

Although the basic concepts discussed in this paper are generally applicable to several types of turbomachines, the scope of this paper will be directed primarily toward centrifugal compressors, since the compressor offers several different seal applications for study and because most of the work done on abradable seals has been relative to compressors.

LOCATIONS WHERE ABRADABLE SEALS CAN BE USED

Many sealing locations exist in turbomachinery where abradable seals can be utilized to their fullest extent. Locations that require a close clearance type seal, where the rotating element runs in close proximity to the stationary element, make good locations for the application of abradable seals. An ideal situation for the application of an abradable seal is where the rotating element grows radially or axially toward the stationary element. The greater the amount of radial or axial excursion, the more suitable is the application of an abradable seal design. Wherever an abradable seal design is applied, a seal with minimum operating clearance results.

Examples of locations in turbomachinery, where abradable seals are used, can generally be classified into two groups. These two groups are vane tip clearance seals and labyrinth clearance seals.

Vane tip clearance seals are located where vane tips come in close proximity to the stationary housing. Figure 1 shows two examples of vane tip abradable seals; i.e. an axial compressor blade tip seal and an open centrifugal compressor impeller vane tip seal. In both cases the vane tip of the compressor blade must be in close, optimum proximity to the stationary housing to give efficient operation of the compressor stage. The less leakage that occurs across the vane tip, between the vane tip and housing, the more efficiently the compressor vane will function; and, therefore, result in a more efficient operating compressor stage. In a vane tip application, the abradable seal material is applied to the stationary element (shroud surface) in the vicinity of the vane tip. As the compressor rotor speed is increased to its operating speed, the blade tips will grow radially toward the stationary shroud. When an abradable seal material is used as the stationary shroud element, the clearance should be kept to the minimum necessary for start-up of the rotor. As the rotor increases in speed and the blades grow radially, the vane tips are allowed to freely abrade away the seal material and find their own minimum running clearance. The running clearance using an abradable seal design will be less than the running clearance of a conventional design. Hence, the abradable seal configuration will have less leakage and higher stage efficiency than a configuration without an abradable seal.

If an abradable material is not used on the stationary shroud, ample clearance must be provided for the radial elongation of the blade, as well as any rotor dynamic deflections, to ensure that no contact is made between the rotating and stationary elements, since contact could possibly result in a catastrophic failure of the rotor assembly.

The second group of abradable seal locations are those utilizing labyrinth clearance seals, where pressure reduction is accomplished by a series of annular orifices. Figure 2 shows an example of an abradable labyrinth seal that is used to seal along an axial surface. Any location in a turbomachine that presently utilizes a labyrinth seal is a potential location for the use of an abradable labyrinth seal. As in vane tip abradable seals, the abradable material is applied to the stationary element. The labyrinth teeth are applied to the rotating element. The clearance between the rotating labyrinth teeth and stationary abradable material is kept to the minimum required for the assembly and start-up of the rotor. As the labyrinth teeth grow radially from rotation and move radially due to rotor dynamics, the labyrinth teeth are allowed to freely cut into the abradable seal material.

Since an abradable seal material allows the rotating labyrinth teeth to seek their own minimum running clearance, the leakage is kept to a minimum. If the axial excursions are kept to a minimum, the abradable labyrinth seal configuration will take on the characteristics of a staggered labyrinth seal after the abrading has taken place. Figure 3 shows an abraded labyrinth seal in an operating configuration corresponding to a staggered labyrinth seal.

Abradable labyrinth seals can also be used on radial faces, provided that the rotating labyrinth teeth will have an axial

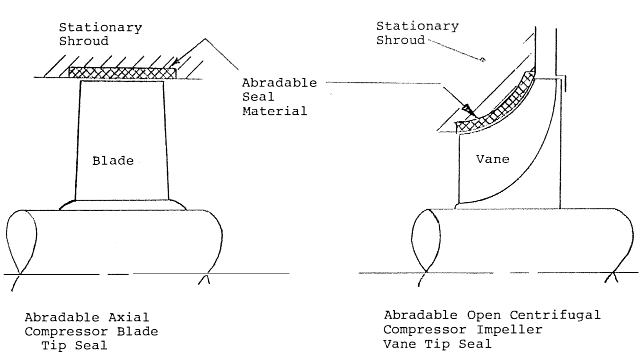

Figure 1. Two Examples of Vane Tip Abradable Seals.

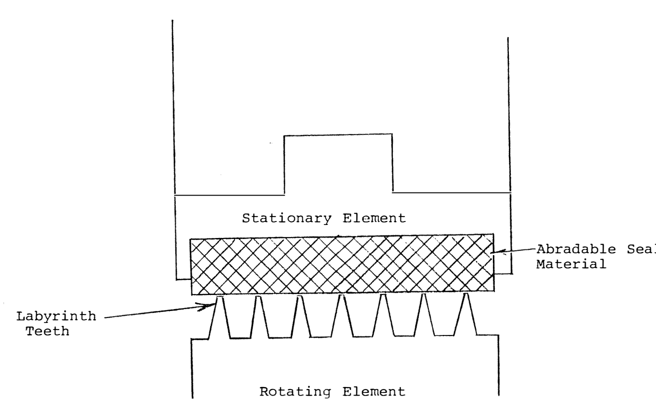

Figure 2. An Example of an Abradable Labyrinth Seal That is Used to Seal Along an Axial Surface.

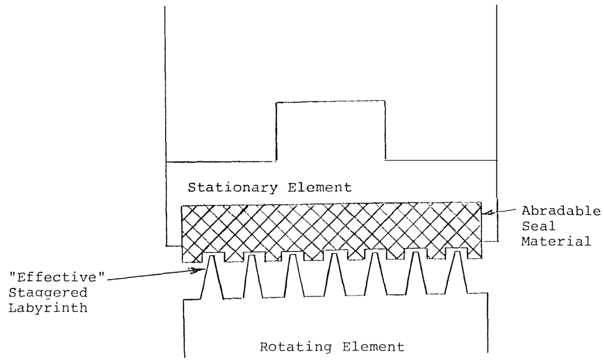

Figure 3. Operating Configuration of an Abraded Labyrinth Abradable Seal Showing the Formation of a Staggered Labyrinth Seal.

excursion into the stationary abradable seal material. The potential benefit of less leakage is similar to an abradable labyrinth seal along the axial surface.

Examples of locations of abradable labyrinth seals for a typical centrifugal compressor are shown in Figure 4. The closed centrifugal impeller shroud seal (intrastage seals) and interstage centrifugal impeller seals act to minimize recirculation losses. The balance piston or drum seal is intended to minimize the leakage from discharge pressure to suction pressure. The purpose of the balance drum is to balance the resultant aerodynamic thrust of all the impellers. The shaft-end seals act to contain the gas being compressed with the case. Depending on the nature of the gas, and because labyrinth seals leak, labyrinth shaft end seals are not always appropriate.

New locations for the application of abradable seals are continuously being explored. The designer must realize how abradable seals function to ensure that the location is favorable to an abradable seal design.

ABRADABLE SEAL MATERIALS

In addition to understanding how an abradable seal functions, the designer must be familiar with the properties of

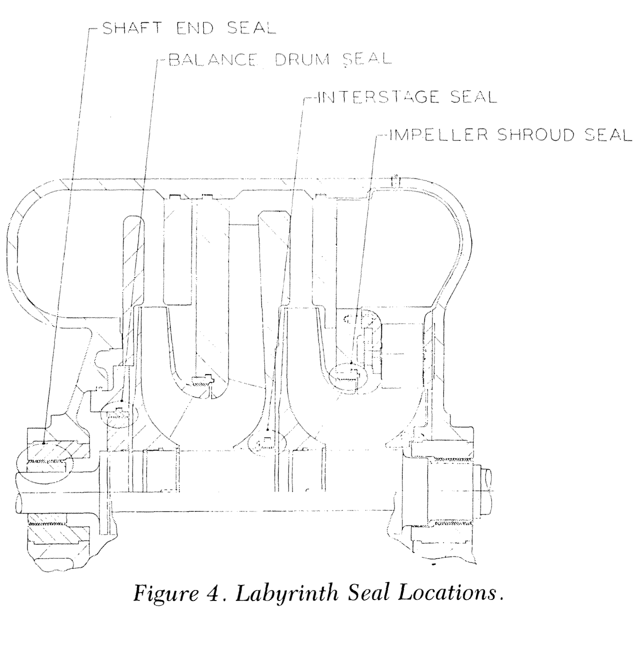

Figure 4. Labyrinth Seal Locations.

material used in abradable seals. In a typical application the abradable material is mounted on the stationary part of the assembly. Besides the obvious abradability requirement, there are other important considerations. The material selected must be compatible with the environment at the installation site. It must be mechanically capable of withstanding the forces resulting from a rub or from differential pressure across the seal or from temperature excursions. It must be resistant to erosion by high velocity leakage flow especially in labyrinth seals. It must be compatible with a wide variety of gases and mixtures of gases ranging from air in a gas turbine compressor to isobutane in a process compressor application to sour natural gas in an oil field application. As part of the abradability requirement, seal materials must be able to accept damage from foreign objects without failing. In addition, the material must not cause significant wear on the blade or knife edge.

There are many materials available which show promise for use in abradable seals. We will not cover all of them here. We will discuss several which are currently being used. One of the better known abradable seal materials is Feltmetal®. Feltmetal seals are made from Brunswick Technetics proprietary metal fibers. The material is produced as a uniform mat of metal fibers, sintered and compressed to a specified thickness and density. Two popular materials used to make the Feltmetal® mat are Haynes 188 and Hastelloy X. When specified in the 21% density product, Feltmetal® has good abradability while causing minimum wear to the blade tips and knife edges.

As with all good things there are limitations. The Feltmetal® noted above is susceptible to severe erosion when exposed to wet steam in a labyrinth seal. Another limit which is of little concern in most compressor applications is the upper temperature at which the material may be applied. Depending on the backing and mounting, this limit is approximately 1200°F.

Feltmetal® is by nature a porous material. The designer must take this into account when using it in an abradable seal. At low differential pressures across the seal the leakage through the material — as opposed to through the clearance gap — is minor. At higher differential pressures, provision must be made to "seal" the porosity. The differential pressure where this change in philosophy is required depends on the specific application and estimates of acceptable leakage.

Fluorosint® is another material which has been successfully used in abradable seals. It is a proprietary development of the Polymer Corporation consisting of a synthetic mica-filled TFE Fluorocarbon. The addition of the mica improves its performance characteristics over pure TFE. The reduced coefficient of thermal expansion over pure TFE virtually eliminates fit and clearance problems in seal parts made from Fluorisint®. Fluorosint® has good form stability to approximately 650°F and exhibits good heat distortion characteristics. In addition it is practically impervious to chemical attack. Only gaseous flourine at high temperature and pressure, and molten alkali metal show any significant attack. Some fluorinated compounds cause swelling.

Unlike Feltmetal® which can be brazed to a suitable mounting device, Flourisant® must be mechanically held in place. Since it is a plastic material it is usually mounted in some kind of metallic holder. The holder may be part of the compressor structure or a separate assembly may be provided.

Honeycomb is one of the oldest materials used for abradable seals. It has been applied as turbine blade-tip seals and as labyrinth seals in centrifugal compressors. It is generally made of stainless steel strip. When used as a labyrinth seal, the basic honeycomb shape when cut by the rotating knives, may develop various and widespread leak paths. To overcome this objection, the cells of the honeycomb may be filled with a suitable abradable material. In this configuration, the honeycomb provides the mechanical strength and resistance to erosion, and the filler prevents spurious leak paths. Because of the metallic construction of the honeycomb, this material causes somewhat more wear to the rotating mating parts than other seal materials.

PMG® is an abradable seal material developed by Gould, Inc. for turbine and compressor tip, labyrinth and interstage seals. It is a powder metal grid available in several composite systems. A typical composite consists of a modified nickel-graphite substrate gridded with L-605 cobalt base alloy. It can be applied from room temperature to 1250°F. A PMG® seal can use any sinterable material available in powder form. The grid serves to preserve the structure under gas stream erosion conditions while the substrate acts as the seal and low-wear rubbing component. PMG® exhibits the durability of filled honeycomb with less component wear. It is available as a finished component ready to install or as rough material for user finishing.

Several grades of glass filled TFE and Nylon have been used in abradable seal applications. In general they have been unsuccessful. This is attributed to the large coefficient of linear expansion of the glass-filled materials, the lower melting point (compared to Fluorosint®) and the highly abrasive characteristics of the glass filler.

When considering materials for abradable seals, the designer should also consider the material of the rotating part. In most cases, standard materials that are compatible with the gas being compressed can be used. However, some applications may have special considerations where particular care must be taken. For example, pockets of gas between labyrinth teeth may result in localized "hot spots," which could cause problems in some applications, such as oxygen or chlorine. In these instances, materials with the proper heat-resistant properties should be employed. Other instances involving particularly corrosive environments should be considered accordingly when selecting the proper material for the rotating seal part.

EXAMPLES OF ABRADABLE SEALS

Locations where abradable seals can be used were discussed in general in a previous section. Figures 5 through 9 illustrate specific applications of abradable seals in various configurations designed and applied by Transamerica Delaval Inc., including some patented designs.

Figure 5 illustrates the shroud and interstage seals which act to minimize recirculation, both around the impeller and from other stages. This sealing arrangement can be used on most centrifugal compressors.

Figures 6 and 7 show two balance piston seal designs for different temperature limits.

Figure 8 shows abradable labyrinth seals being used in conjunction with an oil-type shaft-end seal. The number of labyrinth seals used can vary depending on the type of buffer gas or eductor system. This configuration is commonly used on compressors handling hydrocarbon gases.

Figure 9 shows a multi-labyrinth shaft-end seal for a combination buffer/eductor seal system. This type of seal is most often used on compressors handling non-hydrocarbon gases, such as chlorine, which cannot be allowed to leak into the atmosphere.

These illustrations demonstrate the versatility of abradable seals as applied to a wide variety of compressor applications.

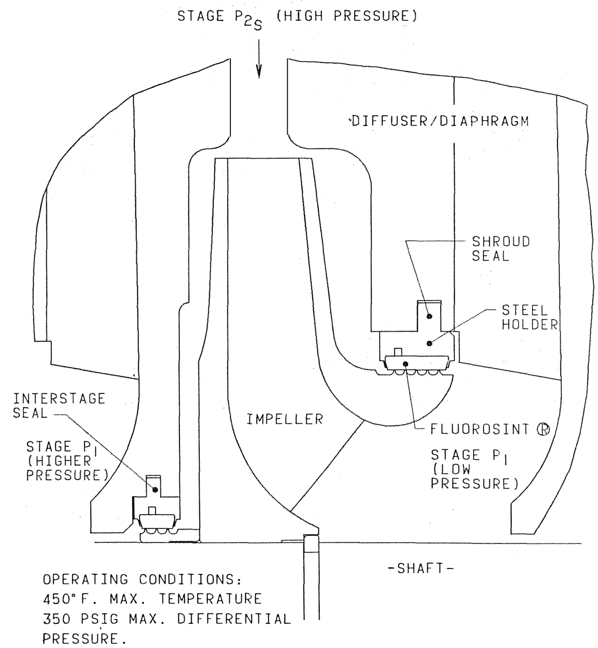

Figure 5. Typical Centrifugal Compressor Stage Seals.

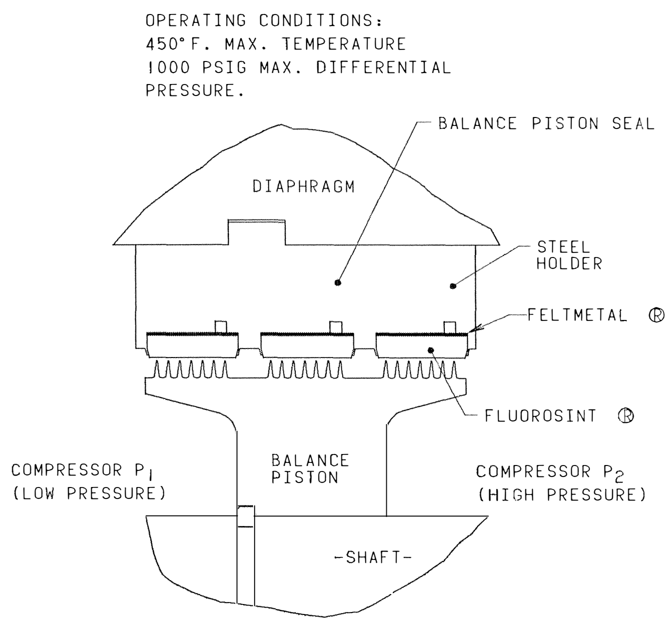

Figure 6. Typical Centrifugal Compressor Balance Piston Seal for Normal Temperatures and Intermediate Pressures.

COMPARATIVE STUDIES USING ABRADABLE SEALS

Various studies have been conducted to compare the use of abradable labyrinth seals to the use of conventional labyrinth seals. Primarily, the studies were conducted to verify the theoretically calculated efficiency improvements, due to the reduction of leakage that is available when abradable labyrinth seals are used.

One of the these studies was undertaken to improve the stage efficiency of wedge-type low specific speed compressor

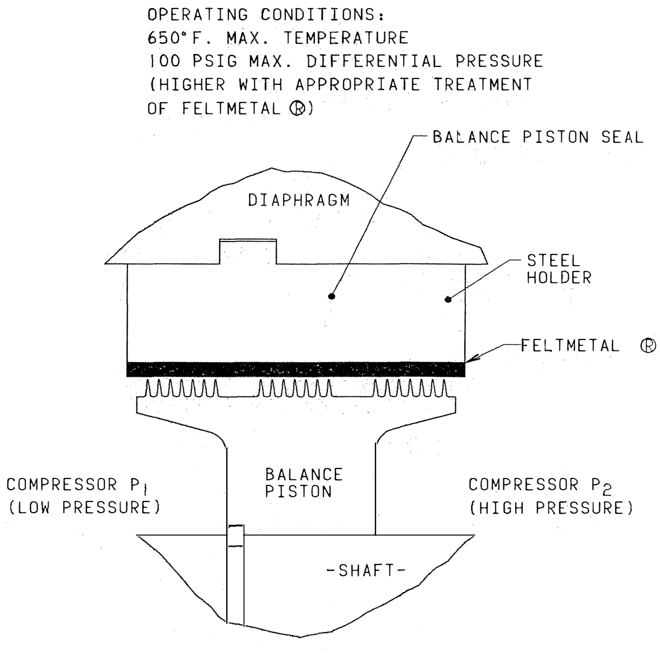

Figure 7. Typical Centrifugal Compressor Balance Piston Seal for High Temperatures and Low Pressures.

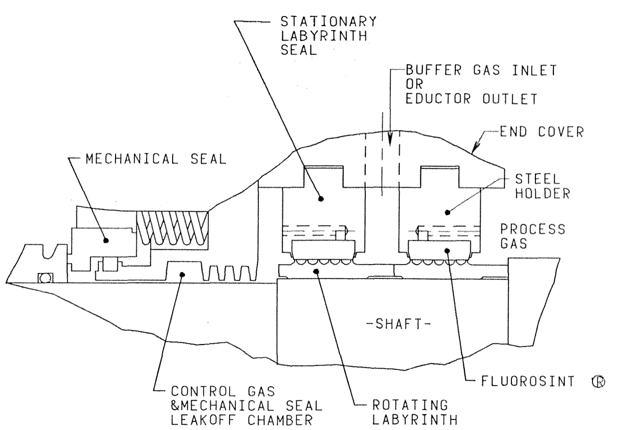

Figure 8. Typical Centrifugal Compressor Shaft End Labyrinth Seal.

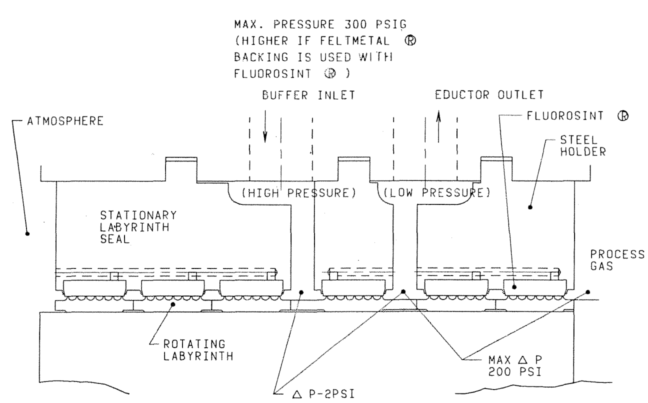

Figure 9. Special Purpose Centrifugal Compressor Shaft-End Seal.

impellers by incorporating an abradable seal for the shroud seal. The internal deficiency, η_i, of a compressor stage can be expressed as,

$$\eta_i = \eta_{ad}\eta_{DF}\eta_{vol} \tag{1}$$

where

η_{ad} = adiabatic efficiency
η_{DF} = disc friction efficiency
η_{vol} = volumetric efficiency

The volumetric efficiency can be expressed as,

$$\eta_{vol} = \frac{m}{m + \Delta m} \tag{2}$$

where

m = impeller mass flow
Δm = leakage mass flow (across shroud seal)

The above equations show that as the leakage flow is reduced, the volumetric efficiency increases and stage efficiency increases.

Egli [2] presented an equation for the leakage through a straight labyrinth seal as,

$$G = A\alpha\phi\gamma\sqrt{g\frac{P_o}{V_o}} \tag{3}$$

where

G = leakage flow (lb_m/sec)
A = area of the throttling opening, $\pi D\delta/144$ (ft^2)
δ = radial clearance (in)
D = seal diameter (in)
α = contraction factor (a function of δ and Δ)
Δ = labyrinth strip width (in)
ϕ = pressure ratio factor (a function of P_o/P_f and n)
γ = carry over factor (a function of δ/s and n)
g = acceleration of gravity (ft/sec^2)
n = number of labyrinth teeth
P_o = absolute pressure before labyrinth (lbf/ft^2)
P_f = absolute pressure after labyrinth (lbf/ft^2)
V_o = specific volume before labyrinth (ft^3/lb_m)
s = pitch between labyrinth teeth (in).

See Figure 10 for the terminology used.

If a series of calculations are made for a straight labyrinth design application with all geometric parameters remaining

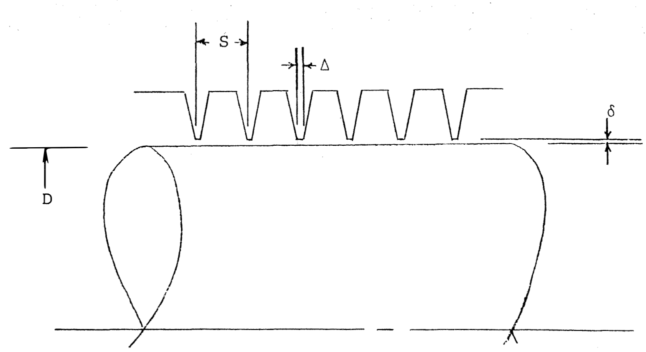

Figure 10. Labyrinth Seal Terminology.

constant except for the radial clearance, a curve of leakage versus radial clearance can be generated.

First, consider the case of the standard straight labyrinth shroud seal of a low specific speed wedge-type impeller operating under the following conditions (See Figure 11):

D = 7.625 in.
δ = .007 in.
Δ = .031 in.
s = .125 in.
n = 4
P_o = 18.0 psia
P_f = 14.7 psia
T = 600°R

Using the above parameters, the leakage was calculated to be .029 lb_m/sec. Since low specific speed impeller efficiencies can be substantially improved by reducing the shroud seal leakage [1], abradable seals were designed for this application with the following parameters. (See Figure 11)

D = 8.248 in.
δ = .001 in. to .010 in.
Δ = .013 in.
s = .140 in.
n = 6
P_o = 18 psia
P_f = 14.7 psia
T = 600°R

The resulting curve of leakage versus radial clearance calculated for the above parameters is shown in Figure 12. To verify the substantial reduction in leakage calculated with abradable seals and the resulting stage efficiency improvement, a test program was initiated. Various abradable seal designs were implemented at the shroud seal to determine the effect on stage performance which includes both efficiency and head [1].

Figure 13 shows the normalized performance improvement when abradable seals were installed as the shroud seal. It can be seen from Figure 13 that stage performance improvements of up to 4% were realized when abradable seals were implemented at the shroud seal.

Similar comparative studies were also made for the successful implementation of abradable seals at other locations in a centrifugal compressor. These locations include interstage centrifugal compressor seals, multistage centrifugal compressor balance drums, and multistage centrifugal compressor shaft-end seals.

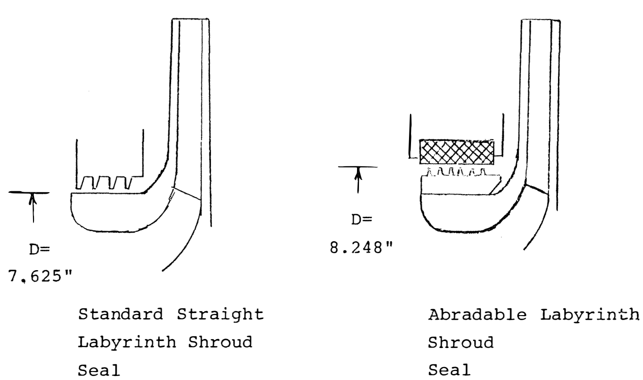

Figure 11. Standard and Abradable Shroud Seal Configurations for the Low Specific Speed Impeller.

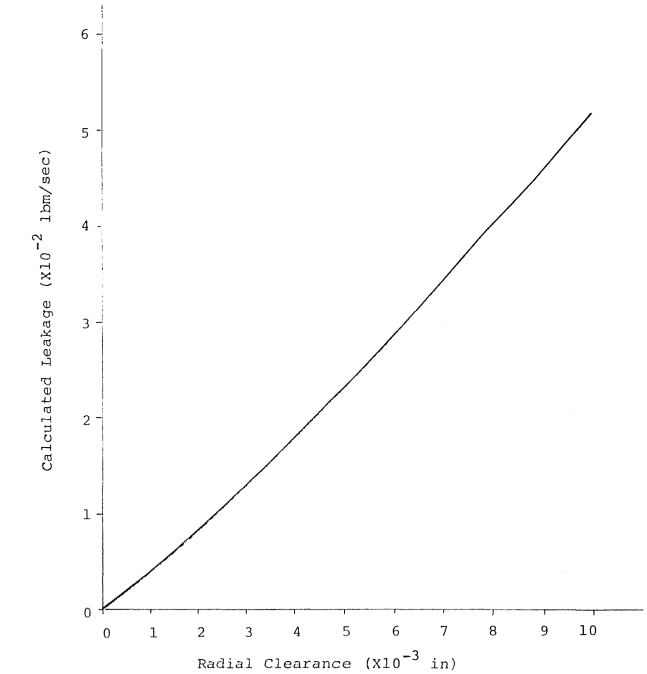

Figure 12. Calculated Leakage Versus Radial Clearance for an Abradable Labyrinth Shroud Seal.

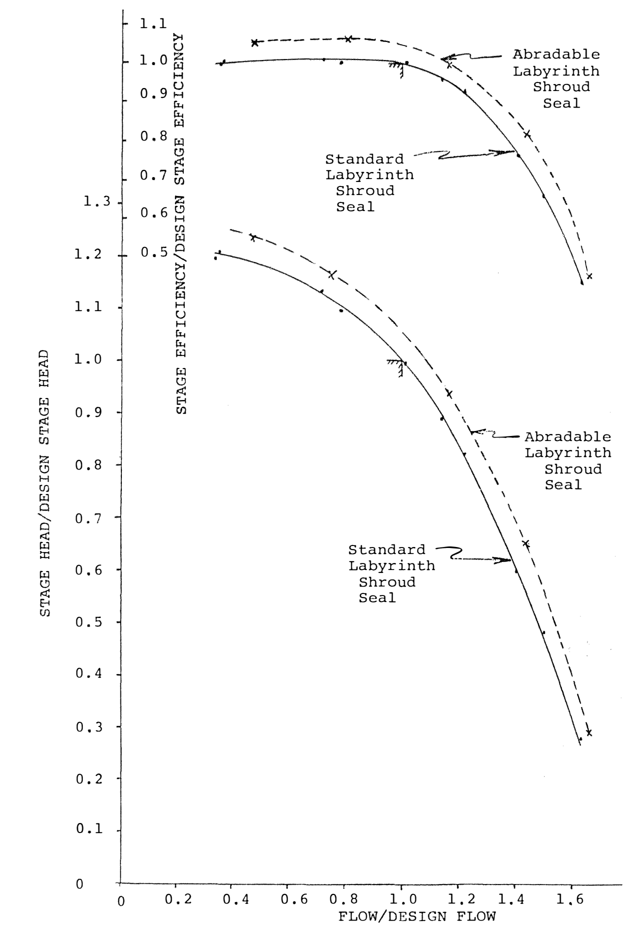

Figure 13. Performance Improvement When Abradable Seals are Used.

CONCLUSION

We have now shown that abradable seals can be designed in a variety of configurations using several materials, and can be applied to many sealing locations in rotating machinery, particularly in centrifugal compressors. Let us now return to the introductory remarks claiming the abradable seal as an energy-saving device.

Figure 13 clearly indicates the ability of abradable seals to improve compressor stage efficiency by improving volumetric efficiency equation (2). The application of abradable seals to balance piston seals and shaft end seals can also result in compressor efficiency improvement by reducing throughput losses. This has the effect of increasing the compressor's overall volumetric efficiency.

The overall efficiency improvement attainable by using abradable seals in a compressor varies with several factors, most notably the size of the compressor. Flow capacity increases as the square of the impeller diameter, while seal clearance increases more linearly with impeller size and is also dependent on other factors such as bearing clearances and manufacturing tolerances. Therefore as the compressor size increases, the leakages involved become a smaller portion of the total flow. As this happens, the improvements gained by reducing these leakages have a diminishing impact on the machine's overall efficiency. This relationship is illustrated in Figure 14.

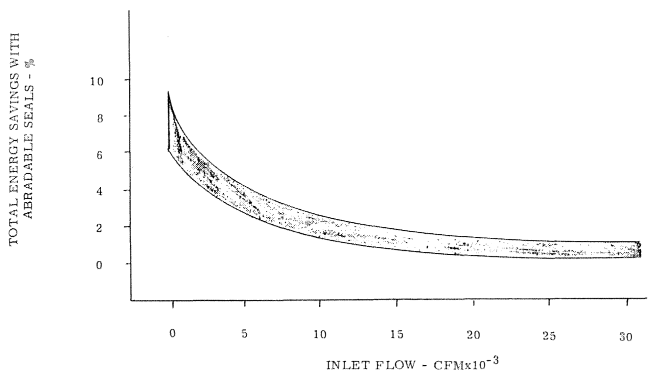

Figure 14. Energy Savings With Abradable Seals.

Another factor to consider when weighing the efficiency gains obtained with abradable seals is the prospect of seal wear over the operating life of the compressor. In a conventional labyrinth seal where the teeth are in the stationary part, the seal's clearance may increase over time as the rotating part touches the stationary part and wears down the teeth. The extent of this wear depends on the initial clearance and the amount of movement of the rotating part. The rate of wear can increase if the machine is subjected to frequent upsets or if rotor dynamics is not properly controlled. Over a typical 3 year continuous operation, it is possible for the clearance of conventional labyrinth seals to increase to twice the initial value, which would result in a doubling of the leakage rate.

With an abradable seal, if radial and axial excursions of the rotating teeth are not excessive, the effective clearance can remain nearly as installed for long periods of operation. Thus with an abradable seal the leakage rates need not increase substantially with time. In the case of a compressor running continuously for 3 years, the cumulative savings with abradable seals could be as much as 150% of that shown in Figure 14.

We can conclude, therefore, that abradable seals can be successfully applied in various sealing locations of turbomachinery, particularly centrifugal compressors, and can be used to improve overall efficiency of these machines.

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2. Adolf Egli, "The Leakage of Steam Through Labyrinth Seals" Transactions of the American Society of Mechanical Engineers, Fuels and Steam Power, FSP-57-5, Vol. 57, 1935, pp. 115-122.