THE INVESTIGATION OF SUITABILITY OF ABRADABLE SEAL MATERIALS FOR APPLICATION IN CENTRIFUGAL COMPRESSORS AND STEAM TURBINES

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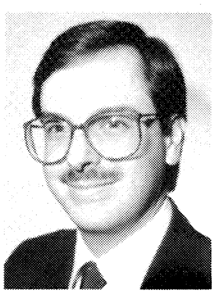

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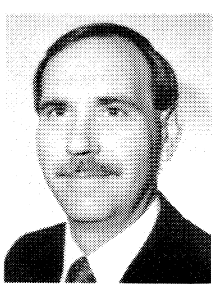

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ABSTRACT

Clearances between sealing devices of rotating and stationary components of turbomachinery have always been important to engineers in controlling gas leakages. One way of accomplishing this is by the use of abradable seals which reduces clearances and limits the risk of damage to the rotating/stationary member if a rub occurs. The program presented evaluates various candidate abradable materials through performance in a test rig and the development of the final process parameters. The results of the tests indicated that for centrifugal compressor applications, the mica filled tetrafluoroethylene (TFE), silicone rubber, aluminum polyester and nickel graphite are all acceptable materials for abradable seals. Temperature limits were specified for each of these materials. For steam turbines, nickel chromium and nickel graphite are suitable and, again, temperature limits were specified.

Application of abradable seals to centrifugal compressors has shown efficiency improvements of up to 0.5 percent per stage for high flow machines, and up to 2.5 percent per stage for low flow machines. Performance improvements in excess of two percent have been calculated for noncondensing steam turbines utilizing abradable materials in shaft end packing, blade tip and stage diaphragm seals.

INTRODUCTION

Energy savings has become one of the important factors in today's marketplace. This is especially true for manufacturers of turbomachinery equipment who are striving for even greater improvement in efficiency of their equipment. Significant energy savings can be made with even modest improvements in efficiency. One way of accomplishing an efficiency improvement is by reducing or eliminating gas leakages. Clearances between sealing devices of rotating and stationary components of rotating equipment have always been important in controlling leakage, and if the clearance is too small, a rub may occur which may damage one or both members. Conventional seals tend to be designed on the conservative side with regards to clearances and, therefore, result in higher leakages and lower efficiencies.

The use of abradable seals (rub tolerant materials) is one way of accomplishing reduced clearances and also limits the risk of damage to the rotating/stationary member if a rub occurs. The development of abradable seal materials has been actively pursued by many researchers for at least the past 20 years. The concept of abradable seals was first used in aircraft engines where materials that are low in resistance to abrasion are selected.

The use of abradable seals has given the design engineer the opportunity to design for low clearances on the premise that if an unintentional rub occurs clearances wear in without any damage to the rotating component. This concept is applicable to both shrouded and unshrouded blade tips and knife-edge contact (as in labyrinth seals).

In more recent years, the use of abradable seals has become more common in centrifugal compressors and in steam turbines. The material that is most commonly used in compressors at this time is a mica-filled tetrafluoroethylene (TFE) material. The abradable mica filled TFE material has been used for at least 10 years with a great deal of success. However, this material has certain limitations regarding operating temperature. In 1989, an abradable seal product development program was undertaken at Elliott Company to evaluate the mica filled TFE and other candidate materials which have been used in aircraft engines for application in compressors and steam turbines [1]. The evaluation process included screening and selection of candidate materials, the determination of physical and performance characteristics of the candidate material through performance testing and the development of the final process parameters. In considering possible abradable seal candidates, the following data was generated: abradability of the seal material, tip wear of rotating component, and operating temperature of the seal material.

DESCRIPTION OF SEALS IN CENTRIFUGAL COMPRESSORS

The most common design of seal for centrifugal compressors is a labyrinth or knife edge seal (Figure 1). This simple seal works by contracting the gas as it flows through the close clearance points underneath the teeth and then expanding it between the teeth. This alternating contraction and expansion reduces the energy of the gas and lowers its flowrate. Leakage through the seal is proportional to the clearance.

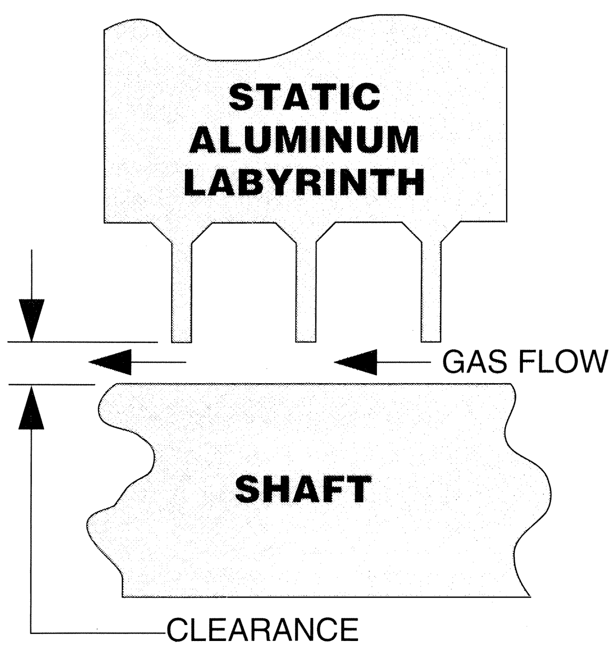


Figure 1. Nonabradable Seal.

In a nonabradabale seal, like that shown in Figure 1, the seal remains stationary while the impeller or shaft rotate inside it. The seal is normally made of a material, such as aluminum, that is much softer than the rotating parts which are typically steel. This allows the seal teeth to deform, should slight contact occur between the rotor and the seal. If the seal were as hard as the rotor, contact would cause severe rotor vibration and possible rotor damage. Unfortunately, when seal teeth are deformed the clearance beneath them is increased and so is the leakage through the seal (Figure 2).

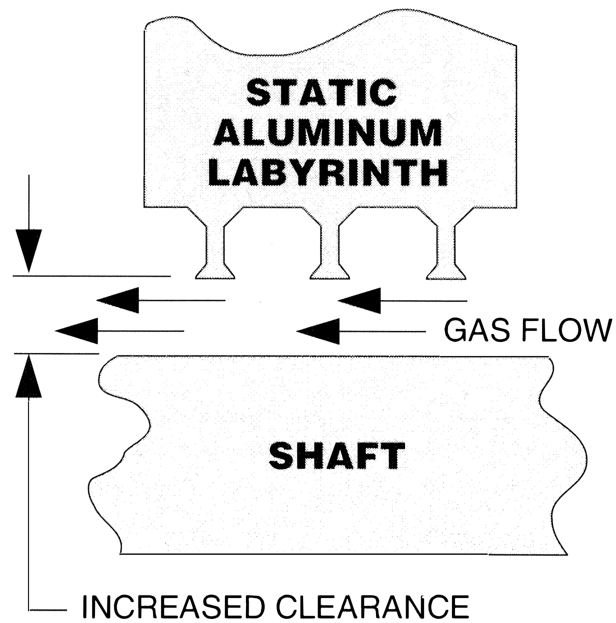


Figure 2. Rubbed Nonabradable Seal.

An abradable seal design is shown in Figure 3. The labyrinth teeth are now on the rotating component and the seal has a smooth bore. The seal is made of a relatively soft and easily abradable material such as mica filled TFE or a base material covered with

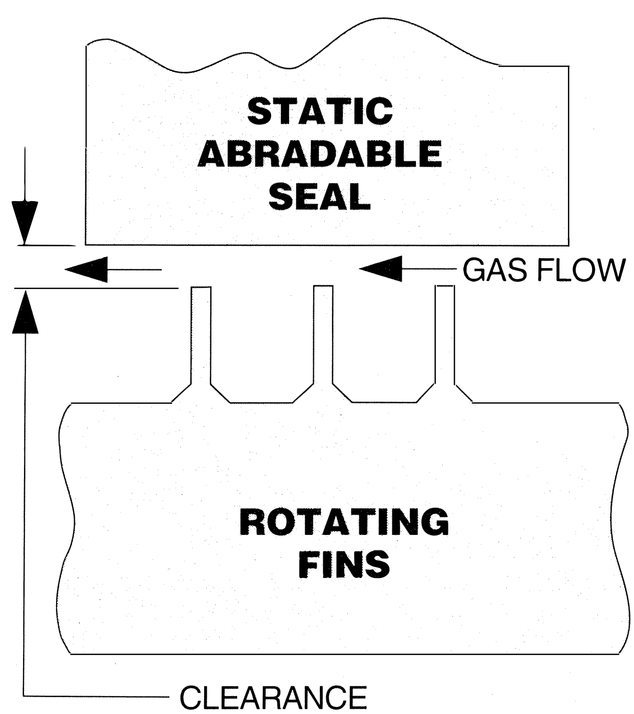


Figure 3. Abradable Seal.

an abradable coating. Should contact occur between rotor and seal, the rotating teeth will cut into the seal and remove some of its material. The teeth, being made of steel, will not deform. With a good choice of seal material, the amount that is worn away will be limited to the dimensions of the teeth plus any axial or radial movement of the rotor that occurred during contact. Axial movement is restricted by the rotor's thrust bearing and normally is quite small. A seal that has been rubbed will look like the one in Figure 4. Note that the effective clearance between the teeth and seal bore is unchanged and leakage through the seal will not be noticeably increased [2].

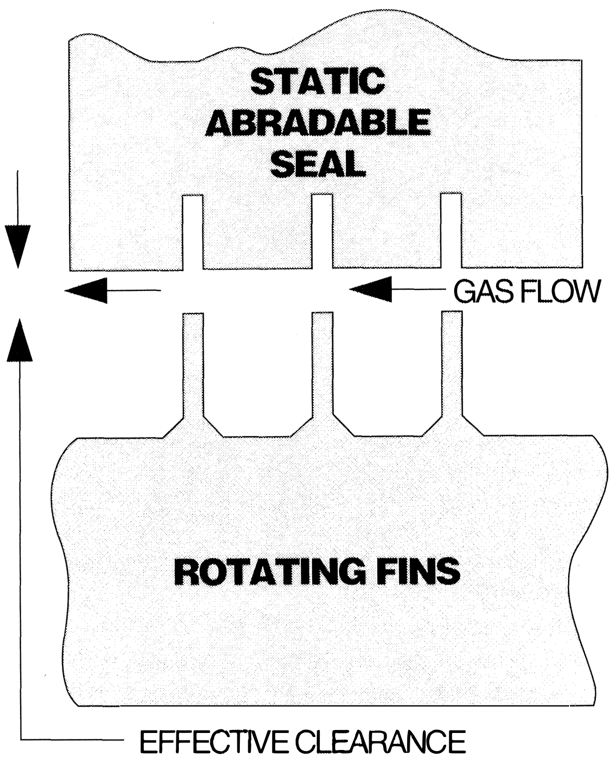


Figure 4. Rubbed Abradable Seal.

Typical centrifugal compressor stage and seal locations are shown in Figure 5. Gas entering the impeller eye at a pressure P_1 will be increased to a pressure P_2 by the work input of the impeller. The eye seal minimizes the flow of gas back into the impeller eye as shown by the arrow labelled W_e. The increasing area of the diffuser reduces the velocity of the gas and further increases its static pressure to a value P_3. Traveling down the return channel has a small effect on the static pressure of the gas so the shaft seal is required to minimize leakage back to the area behind the impeller where the pressure is lower. This is shown by the arrow W_s.

Another area to be sealed in a compressor is the balance piston. This component is used to balance the axial thrust of the rotor so that the size of the thrust bearing can be reduced. Refer again to Figure 5. The entire area of the impeller larger than the diameter of the eye seal is exposed on both sides to pressures less than or equal to P_2 (static). The cover side pressure is lower so there will be a net axial force acting on the area in the direction of the impeller eye. The area between the eye seal diameter and shaft diameter is exposed to pressure P_2 static on the back side and P_1 on the eye side. This results in a net axial force equal to $(P_2 - P_1)$ times this area and acting in the direction towards the impeller eye. Summing these forces for each stage gives the total rotor axial thrust.

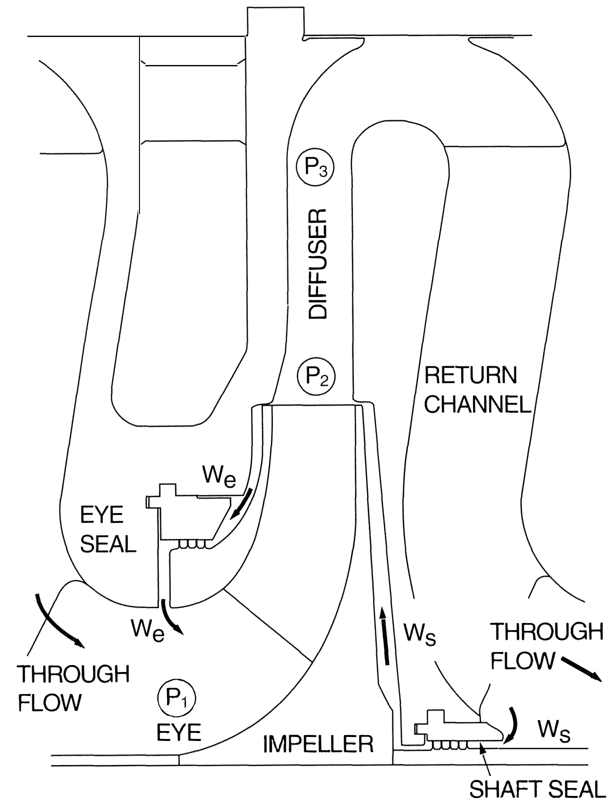


Figure 5. Typical Centrifugal Compressor Stage.

The balance piston, shown in Figure 6, rotates with the shaft. One side is exposed to the static pressure from the discharge of the last impeller, P_f. The other side is exposed to a lower pressure (usually the compressor inlet), P_i. This is done by means of a pipe, or balance line, connecting the chamber behind the balance piston to the inlet. The large pressure difference $(P_f - P_i)$ times the area of the balance piston from its outside diameter to shaft diameter provides an axial force of similar magnitude and opposite direction to that of the impellers. The important seal on the outside diameter of the balance piston reduces the leakage, W_{bp}, of gas from the compressor discharge back to the inlet.

If a nonabradable balance piston seal was rubbed as shown in Figure 2, there could be serious consequences for the compressor. Leakage back to the inlet, W_{bp}, would increase. This would in turn raise the pressure drop through the balance line so that the pressure in the chamber behind the balance piston increased and the pressure difference across the balance piston decreased. With less pressure difference the axial force would also be less and the load on the thrust bearing increased, possibly to the point of bearing failure. The increased leakage also hurts machine performance, as explained in the next section.

Given that an abradable seal can tolerate a rub with negligible affect, there is less risk should rotor contact occur. With this in mind, seal to rotor clearances can be reduced to a point where occasional contact is almost certain (passing through the rotor first critical, etc.). During normal operation, the smaller clearance reduces leakage which reduces horsepower requirements.

There are limits to which the clearance can be reduced. Aircraft engines operate with line to line or almost zero clearance. This would be ideal for centrifugal compressors but is not practical. The high operating temperatures of the gas turbine causes thermal expansion of the rotating components. This means that while clearance is near zero at operating temperatures, it is much larger

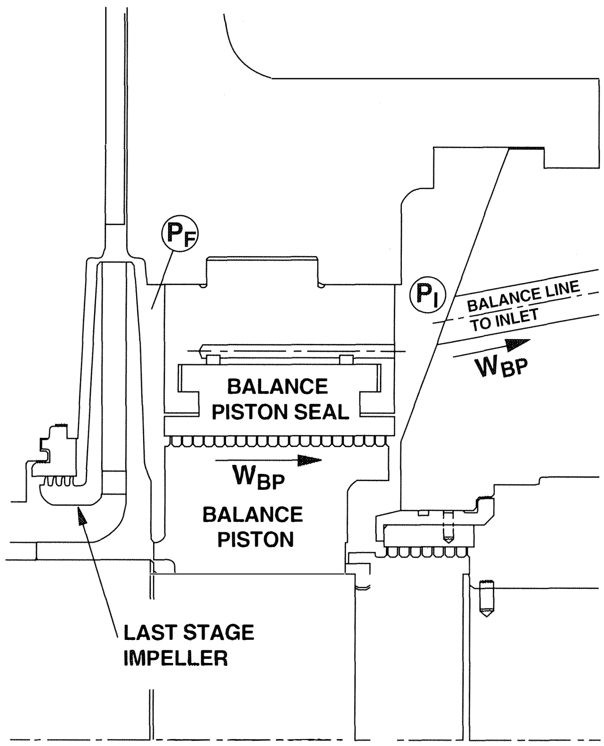


Figure 6. Balance Piston.

at startup. Most centrifugal compressors in petrochemical service have a maximum operating temperature of 350F or less, so thermal expansion is not as significant. Some clearance is necessary during operation to allow for adequate clearance at startup. Otherwise, starting torque would be high and the rotor first critical speed possibly elevated. This is particularly true for smaller, lighter rotors.

EFFICIENCY IMPROVEMENTS FOR A CENTRIFUGAL COMPRESSOR

Reduced leakages improve horsepower consumption in a compressor stage mainly by the reduction of recirculated mass flow. Recirculated flow must be added to the desired through flow of the stage when calculating gas horsepower using the equation:

$$GHP = (W\ H_p)/(33000\ Eff_p)$$

where:

W = Mass Flow (lb/min)
H_p = Polytropic head (ft-lb_f/lb_m)
Eff_p = Polytropic efficiency

It can be seen that reducing mass flow directly reduces gas horsepower. When looking at compressor stages in terms of polytropic efficiency, gains of 0.5 percent to 2.5 percent are possible with the use of abradable seals. The biggest gains occur in low volume flow stages where leakage is a larger percentage of the through flow.

Balance piston seal leakage is recirculated through the entire machine. Horsepower savings are calculated using the same equation with head and efficiency values for the machine rather than the stage. Since balance piston leakage comes from the discharge of the machine it is at a higher temperature than the inlet. As it mixes with the through flow it raises the temperature to the first stage. The higher temperature requires more work be done to compress the gas. Abradable seals reduce the amount of leakage and so reduce the inlet temperature rise, again saving horsepower.

For an example, refer to Table 1. The machine is a three stage compressor designed for 88.1 mmscfd of a hydrocarbon gas mixture. The base design has all aluminum eye, shaft, and balance piston seals. It can be seen that the lowest volume flow (last) stage shows the largest horsepower reduction of the three when abradable seals are used. The increase in polytropic efficiency for these stages is 0.7, 0.8, and 1.0 from first to last. The effects of mass flow and inlet temperature reduction were separated for the balance piston to show the contribution of each. Temperature rise from inlet to discharge is only a little over 100°F for this machine. Had it been higher, the inlet temperature reduction would have had a more significant effect on horsepower.

If the savings for abradable seals on all stages and for the balance piston seal are added together, the sum does not agree with the total at the bottom of the table. This is because the reduction of recirculated flow with the abradable balance piston seal shifts the stages to lower flows and other points on their characteristic head and efficiency curves.

Table 1. Predicted Horsepower for a Three Stage Compressor.

	Machine GHP	Horsepower Reduction	Percentage Reduction
Base - Aluminum Seals	5330	-	-
Stages			
Abradable seal stage 1 only	5311	19	0.36
Abradable seal stage 2 only	5310	20	0.38
Abradable seal stage 3 only	5308	22	0.41
Abradable seal all stages	5269	61	1.14
Abradable Balance Piston Seal			
Mass flow reduction only*	5300	30	0.56
Temperature reduction only*	5324	6	0.11
Total for balance piston seal	5294	36	0.68
Total for all abradable seals	5240	90	1.69

*These would be combined in actual operation and are listed separately to show the relative effects of each.

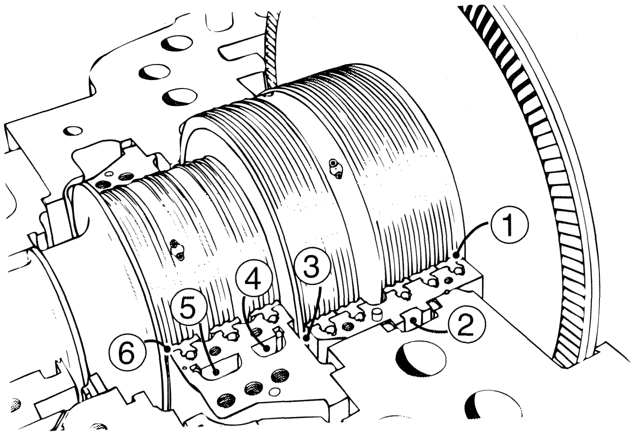
The authors have had good experience in meeting these predicted horsepowers using abradable seals, the majority being made of mica filled TFE. Other materials discussed herein will use the same clearances and are expected to have the same results.

STEAM TURBINE APPLICATION

Abradable seals can also be applied to steam turbines. Reducing seal leakage results in additional steam available to produce power. Papers published to date have not addressed these potential performance gains.

Steam turbine efficiency is influenced by aerodynamic and mechanical losses. The application of abradable materials to steam turbines will affect only mechanical losses attributed to labyrinth seals. Within the steam turbine casing, labyrinth seals are used to control steam flow at three locations. A typical leakage flow and sealing arrangement in Figure 7 shows where the rotating shaft extends through the casing end wall. The design of this shaft end seal incorporates intermediate leak offs (locations 2 and 3) to recover steam to do work at downstream stages. Not all of the steam, however, is recoverable. Typically, 10 percent to 13 percent is drawn with air to a gland condenser at the last leakoff (location 5).

Each turbine stage consists of stationary nozzles and rotating blades (Figure 8). The stationary nozzles are an integral part of a



1. First-stage pressure equals 300 PSIA at full load.
2. Leakoff to turbine stage at approximately 125 PSIA.
3. Leakoff to turbine stage at approximately 40 PSIA.
4. Sealing steam at startup approximately 18 PSIA.
5. Steam and air drawn to gland condenser at approximately 13.5 PSIA.
6. Small amount of atmospheric air at 14.7 PSIA drawn in.

Figure 7. Leakage Flow and Sealing Steam Arrangement for Steam End of Condensing Turbine. Conditions above are typical values.

diaphragm. A labyrinth seal is used to control leakage through the diaphragm bore. This leakage eventually passes through the rotating blades after mixing with the flow between the diaphragm and rotating disk. This leakage does not contribute to producing mechanical work because it bypasses the stationary nozzles.

Simply stated, a steam turbine operates by utilizing heat energy which it extracts from the steam and which it converts into mechanical work (power). Heat and mechanical work are both forms of energy, and, therefore, can be converted from one to the other.

First, heat energy is converted into velocity (kinetic) energy. This is accomplished when steam expands in a nozzle discharging at high velocity. The total heat (enthalpy) of the steam is converted to kinetic energy (velocity) by the general equation:

$$V_j = 224\,(h_2-h_1)^{1/2}$$

where:

h_1 = Inlet steam enthalpy, BTU/lb.

h_2 = Isentropic exhaust steam enthalpy, BTU/lb.

Second, a steam turbine does mechanical work by virtue of the steam velocity (V_j) which strikes moving blades. Power is the product of this mechanical work and stage mass flow. For a simple system, shown in Figure 9, power is defined by the general equation:

$$\text{Power} = K\,W\,(V_jV_b-V_b^{\,2)}$$

where:

K = Conversion factor

W = Stage mass flow, lb/hr

V_j = Nozzle exit velocity, ft/sec

V_b = Blade velocity, ft/sec

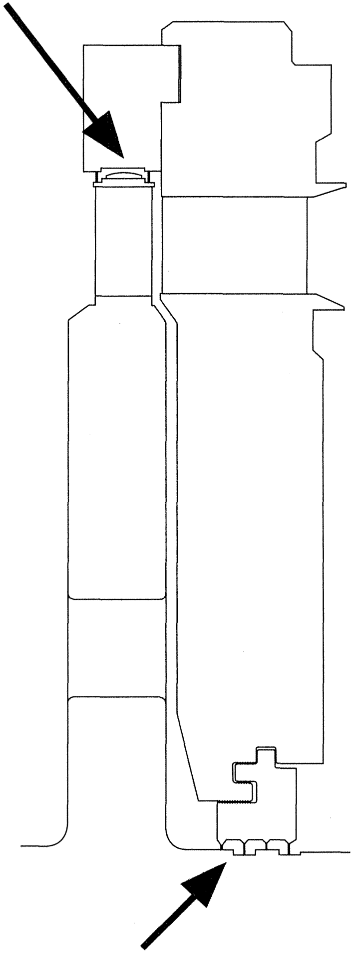


Figure 8. Typical Steam Turbine Stage Labyrinth Seals at Diaphragm Bore and Blade Tip Locations.

W is the mass flow leaving the stationary nozzles and is fixed. A portion of this mass flow becomes entrained in the flow being pumped between the diaphragm and rotating disk by disk friction. The diaphragm bore seal leakage discharges into this region. It mixes with the disk pumping flow and eventually joins with the nozzle exit mass flow to satisfy the blade flow requirements.

Lastly, labyrinth seals are used to control blade tip leakage (Figure 9). This seal reduces mass flow around the rotating blades, thereby, increasing the mass flow through the blade passages. This, in turn, increases the stage power output.

It is normal industry practice to design these labyrinth seals with relatively large radial clearances. This is done to prevent seal damage caused by rubs between stationary and rotating components during operation. The application of abradable material to the identified seal locations would permit tighter clearances resulting in lower seal losses plus provide rub tolerant seals. Lower seal losses can lower steam turbine throttle flow requirements for new applications. This can reduce operating costs for the end user.

ABRADABLE SEAL BENEFITS

Steam turbines can be classified as either condensing or noncondensing. Overall performance of each turbine class is affected differently by the reduction of seal losses due to abradable seals.

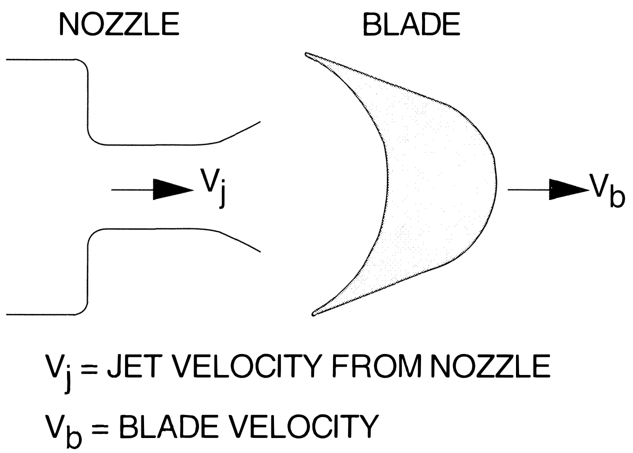


Figure 9. Simple System of a Steam Turbine Nozzle and Rotating Blade.

A theoretical review was made for applications in both steam turbine classes. Stepped labyrinth seals were chosen for both shaft end and diaphragm bore seal locations. A straight through labyrinth was used for all blade tip seals. Radial clearance was the only seal geometry parameter varied.

Seal clearance reduction and resulting performance gain for condensing and noncondensing turbines are listed in Table 2. Noncondensing turbines will benefit more from abradable seals than condensing turbines. This is because condensing turbines, by design, are more efficient than noncondensing turbines. Typically, condensing turbines are also higher flow machines. As a result their shaft end and diaphragm seal leakages are a smaller percentage of throttle flow. Therefore, reducing seal clearances and corresponding flows with abradable seals produces smaller mass flow gains for condensing turbines.

Table 2. Abradable Seal Benefits.

Turbine	Clearance Reduction (%) Shaft End	Diaphram Bore	Blade Tip	Performance Gain (%)
Condensing	43-76	31-39	50	0.3-0.5
Non-Condensing				
Single Valve	81	68	50	1.0-2.1
Multi-Valve	81	68	50	2.2-2.4

ABRADABLE SEAL MATERIALS

The screening and selection process of possible candidate materials was done by researching previous work [3, 4, 5, 6], discussion with aircraft engine manufacturers, and discussions with vendors of coatings and apparatus used for their application. In the selection process, serious consideration was given to ease of refurbishment of the abradable seal components. The mica filled TFE material does not lend itself to this attribute and, therefore, becomes a redundant part when damaged. Therefore, materials selected were in a form that could be either sprayed or injection/adhesive bonded to a metallic substrate. From the information collected, seven materials were selected for further evaluation for application in both centrifugal compressors and steam turbines. The materials that were selected and a brief description of the applied process used for attachment to the metallic substrate are reflected in Table 3.

When selecting an abradable seal beside the obvious abradable segment, there are other important considerations. The seal material must be able to withstand the environment which the machine operates under. This will include any erroneous mechanical forces and/or temperature excursion that may occur during the machines operation. Consideration must be given to the corrosion resistance of the seal since in centrifugal compressor application the gaseous mixtures can range from a hydrocarbon, sour natural gas to chlorine gaseous condition. The following is a discussion on the seal material selected and the associated physical and chemical properties.

Table 3. Brief Description of the Selected Seven Abradable Materials.

Material	Description	Process Application
A	Mica-Filled Tetrafluorethylene	Purchased in Bulk Form
B	Silicone Elastomer Containing Hollow Glass Microspheres	Vacuum Injection
C	Blend of Powders, Aluminum Alloy Containing Silicon and Polyester Resin	Plasma Sprayed
D	Proprietary Material Similar to Material "C" except Finer Particle Size	High Velocity Oxygen Fuel Gun Sprayed
E	Nickel-Graphite blended Powder	Oxygen Fuel Gun Sprayed
F	Aluminum Silicon Graphite	Oxygen Fuel Gun Sprayed
G	Nickel Chromium Powder with Meltable Polymer	Plasma Sprayed

Mica Filled Tetrafluorethylene

The mica filled TFE is a proprietary material and is available in bulk form in sizes only up to 37 in diameter. The coefficient of thermal expansion ofthe material is $1.25\text{-}1.5 \times 10\text{–}5$ in/in/per °F, which is well above that for steel and must be taken into account in design. The material is generally impervious to corrosion attack from most gaseous mixtures. The tensile strength of the material is low (1000 to 1200 psi). Consequently, this can impose certain restrictions in its application especially when subjected to high ΔP values across the seal.

Silicone Rubber

This is a proprietary material consisting of silicone elastomer containing hollow glass microspheres. The material cannot be used in sprayed coatings because the hollow spheres fracture too easily. Consequently, the material is injection molded under vacuum onto the metallic substrate.The coefficient of thermal expansion of the material is $4.4 \times 10\text{–}5$ in/in/°F. The material is generally impervious to corrosion attack from most gaseous mixtures.

Aluminum Silicon Polyester

This material is a blend of aluminum and silicon powders with polyester resin plasma sprayed to a metallic substrate. The corrosion resistance of the coating is not as good as the nickel graphite coatings but has a higher erosion resistance.

Aluminum Silicon Polyester (finer particle size)

Same as "C" except that the powders are finer to give increase in erosion resistance. This coating was applied by the high velocity oxygen fuel gun with a corrosion resistance similar to that of "C" coating.

Nickel-Graphite

This material is a blend of nickel graphite powder which is oxygen fuel gun sprayed onto the metallic substrate. The coefficient of thermal expansion of the material is compatible for that of steel which is desirable for the design engineer. The material has better corrosion resistance than both of the aluminum silicon polyester sprayed coatings.

Nickel/Chromium

This material is a blend of nickel and chromium powder and polymer applied with conventional plasma spray equipment to a metallic substrate. The sprayed material is then exposed to an oven bake in air at 600°F to 650°F to drive off the polymer leaving a dense porous metallic structure. The corrosion resistance of the coating is better than all the sprayed coatings that were tested and has a high erosion resistance, as indicated by its higher hardness.

Aluminum Silicon Graphite

This material is a blend of aluminum silicon graphite powders and organic solids which are oxygen fuel gun sprayed. It has similar corrosion resistance to the aluminum polyester coatings. The erosion resistance would be lower than the other sprayed coatings.

TEST APPARATUS

The test apparatus was a vertical grinder that was modified to obtain the required rubbing velocities. Various pulleys of appropriate diameters were fitted to the electric motor to drive the spindle at 16500, 8000, and 2950 rpm. This resulted in rubbing velocities of 450, 220, and 82 ft/sec, respectively. The two higher rubbing velocities were selected to represent compressor shaft and impeller eye seals. The lowest velocity (82 fps) represented shaft end seals on small steam turbines. The test apparatus and set up are shown in Figures 10 and 11. An electrically driven linear actuator was used to control the radial feed of the spindle and rotor. This was done to give a consistent rate and depth of penetration.

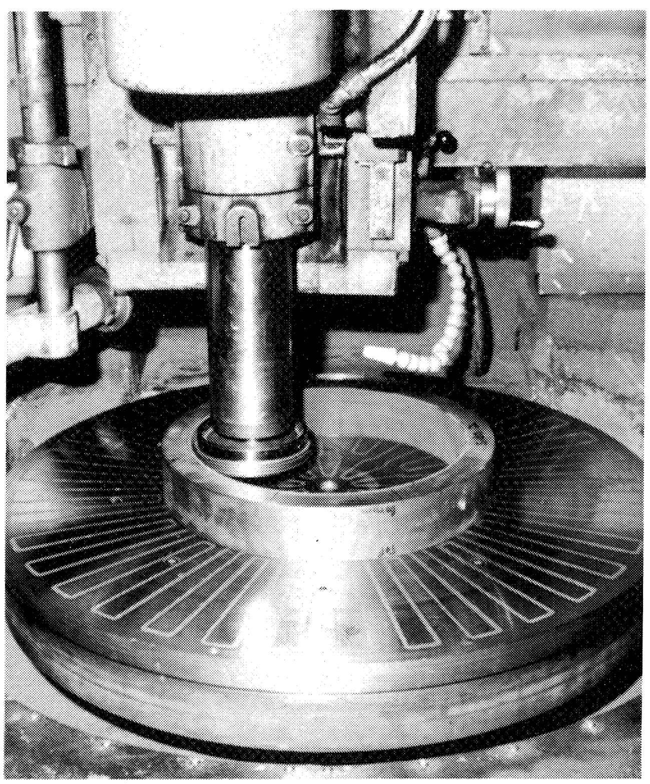


Figure 10. Overall View of Test Ring Coated with Silicone on Magnetic Table of Grinder. Test rotor on lower end of spindle.

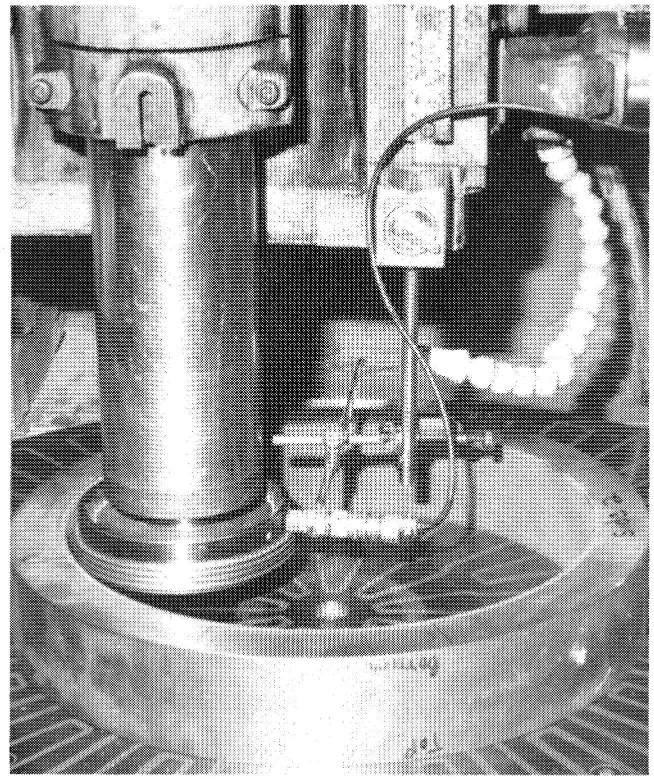


Figure 11. Closer View of Test Setup with Speed Counting Sensing Device in Place.

The test samples were rings of carbon steel which served as the substrate for the coating materials. The coatings were applied to the inner cylindrical surface and after machining obtained a coating thickness of 0.1 in (Figure 12). In the case of mica filled T F E, a 1.0 in thick ring was inserted inside the steel ring.

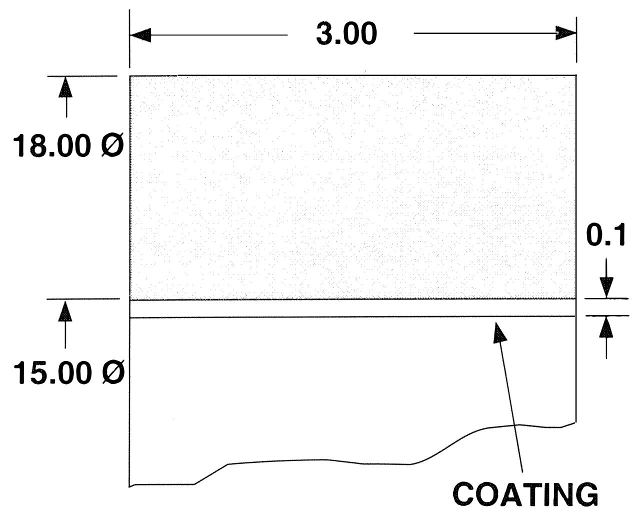


Figure 12. Cross Section of Test Ring.

The test rings were magnetically held to the grinder table and rotated in the direction opposite to that of the rotor. This resulted in a negligible effect on speed, but spread the wear around the circumference of the ring. This was considered to be preferable to having wear in one place. The rotors were made in a labyrinth tooth form, which is used in the majority of centrifugal compressors and steam turbines (Figure 13).

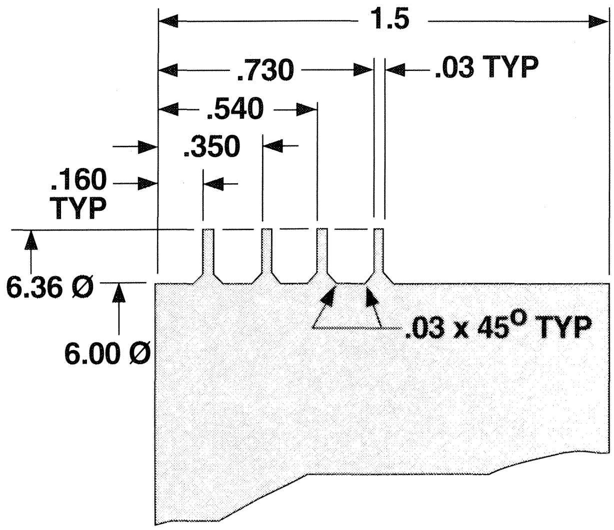


Figure 13. Section at Outside Diameter of Rotor.

TEST PROCEDURE

To optimize good abradability and erosion resistance, a certain amount of preliminary testing was done to obtain the process parameters of the oxygen gun fuel spray and plasma spray equipment for the materials under consideration. For thermal sprayed abradable coatings, a hardness range of HR15Y30 and HR15Y80 was used as a coating parameter. Generally coatings below this range have poor erosion resistance, while coatings above this range usually have poor abradability, and consequently, excessive blade tip wear. Extreme care needs to be taken in the measurement of hardness of these coatings and the procedure followed. It is important that the coating be measured for hardness be at least 0.075 in in thickness to avoid influence of the base material.

Each of the test rings were subjected to velocities of 450, 220, and 82 ft/sec, respectively. As stated in the test apparatus, the electrically driven linear actuator controls the rate and depth of penetration from which interaction rates used for the test were 0.1 and 1.0 mil per second. The interaction rate is the rate at which the labyrinth teeth wear (abrade) into the test ring. The low rate of interaction was thought to be representative of conditions occurring due to thermal expansion. The high rate was more typical of high vibration due to passing through a critical speed or a rapid change in load. The depth of penetration in the abradable material was 0.021 in.

During the test, the rotor was brought up to speed with a clearance of 0.003 in before contact with the test ring. The radial movement was then activated and the rotor made contact with the test ring. The rotor automatically retracted when it reached the 0.021 in depth. Before and after each test the labyrinth tooth form of the rotor was measured for wear. Also, visual examination of the tooth form appearance was recorded after each test. Following each test, the groove depth of the seal material was measured and recorded. In most of the published literature on abradable seals, an objective is to obtain an abradability ratio of at least eight and preferably 15. This is the ratio of groove depth in the seal to metal loss on the rotor. The appearance of the grooved test samples for the various materials were recorded. Perpendicular sections were taken through the grooved test samples and scanning electron microscopy was done to observe the interaction at the bottom of the grooves and the general contour appearance of the groove.

TEST RESULTS

The conditions for each test run performed are summarized in Table 4. All the tests were under four minutes with most of the tests at interaction of 0.1 were 3.0 and 3.5 minutes. At an interaction of 1.0, the time was less around 30 seconds. In contrast to some of the tests reported in the literature which ran less than a minute, the objective was to spread the test over a longer interval thereby making it easier to distinguish between materials. However, the test results indicated that the distinction between materials was more marked at a higher rate of interaction and low rubbing velocity. The test results are summarized in Table 5 for each test run on the various abradable materials.

Table 4. Test Conditions for Each Test Run.

Test Run	Abradable Material	Rotor rpm	Velocity fps	Duration Secs	Depth of Groove	Interaction mils/sec
1	B	16500	450	234	0.025"	0.1
2	B	16500	450	231	0.024	0.1
3	C	16500	450	201	0.022	0.1
4	E	16500	450	212	0.023	0.1
5	E	16500	450	204	0.023	0.1
6	G	16500	450	202	0.025	0.1
7	G	16500	450	202	0.023	0.1
8	A	16500	450	201	0.022	0.1
9	D	16500	450	178	0.022	0.1
10	D	16500	450	222	0.024	0.1
11	F	16500	450	223	0.030	0.1
12	F	16500	450	220	0.029	0.1
13	B	16500	450	31	0.024	1.0
14	B	8000	220	31	0.024	1.0
15	A	16500	450	31	0.023	1.0
16	A	8000	220	32	0.020	1.0
17	A	8000	220	30	0.016	1.0
18	E	8000	220	30	0.023	1.0
19	E	16500	450	29	0.025	1.0
20	F	16500	450	29	0.025	1.0
21	F	8000	220	29	0.030	1.0
22	C	8000	220	33	0.027	1.0
23	C	16500	450	30	0.024	1.0
24	D	16500	450	30	0.024	1.0
25	D	8000	220	30	0.025	1.0
26	G	8000	220	17	0.009	1.0
27	G	8000	220	230	0.025	0.1
28	E	2950	82	29	0.024	1.0
29	D	2950	82	29	0.023	1.0
30	C	2950	82	29	0.021	1.0
31	A	2950	82	28	0.010	1.0
32	B	2950	82	28	0.014	1.0
33	G	2950	82	265	0.022	0.1
34	G	2950	82	198	0.016	0.13

All of the materials except the nickel chromium (G) were tested at an interaction rate of 0.1 and 1.0 mil/sec and rotational speed of 450,220 and 82 ft/sec. The nickel chromium was tested at all three rotational velocities but only at an interaction rate of 0.1 and 0.13 mil/sec. An attempt to test the nickel chromium material at an interaction rate of 1.0 mil/sec caused severe vibration which resulted in the test not being completed. A summary of all these tests (shown in Table 5) classifies the abradability of each material for the various test conditions. Hardness tests were taken for all the materials tested and are shown in Table 6. Measurements taken of the rotor before and after each test run indicated that there was no measurable reduction of the rotor diameter (less than 0.0005 in). Even though there was evidence of heating of the rotor teeth in

Table 5. Classification of Abradability for Each Material.

Velocity (fps) Int. Rate (mps) Abradable Material	450 0.1	450 1.0	220 1.0	82 1.0	
B	Grooves well defined-VG	Grooves well defined-VG	Grooves well defined-VG	Grooves well defined-VG	Insensitive to variations in-VG
E	Grooves well defined-VG	No change-VG	No change-VG	No change-VG	Insenitive to changes in test-VG
A	Grooves sharp; few small threads on corners-VG	Slightly more threads on corners-VG	Increased threads-VG	Further increase in threads-G	Tendency for corner threads; increases with interaction rate and with decreasing velocity-G-VG
D	Groove corners sharper than Material C-G	More breakage of corners-F	Increased corner breakage-F	Little further increase in corner break-F	Performance somewhat better than Material C-F-G
C	Groove corners good, but less sharp than Materials A & B-G	More breakage of corners-F	Increased corner breakage-F	Little further increase in corner breakage-F	Corner breakage increase with interaction rate and decreased velocity-F-G
G	Some thermal cracking & chatter-G		Could not run at 1.0 mps. more thermal cracking that at higher velocity-P	Little change from 220 fps-P-	Highest hardness of coatings tested. Not suitable for use at room and moderately elevated temperatures-P-G
F	Corner breakage Layer separation-F	Increased damage-P	Still more damage-P	Not tested	Unsatisfactory due to extent of corner breakage and layer separation. Difficult to apply. Effort to refine application not justified-P-F

VG-Very Good F-Fair
G-Good P-Poor

some of the tests, especially nickel chromium, there was no measurable reduction of the rotor diameter (Figure 14). As stated in the TEST PROCEDURE section, the objective is to obtain an abradablility ratio of at least eight and preferable 15. Since the metal loss on the rotor was less than 0.0005 in, the abradability ratio greater than 15 was achieved on all completed tests.

Materials A, B, and E were found to have *good* to *very good* abradability depending upon the test condition of the test run. There was no evidence of any thermal cracking or layer separation in any of these materials. The materials "B" and "E" gave well defined grooves for all test runs (Figures 15, 16, 17 and 18). Even with the most severe abrasion test (the high rate of interaction and low rubbing velocity), materials "B" and "E" performed very good. This tended to indicate that these materials were insensitive to all the various test conditions performed. However, material "A" at the slower velocity and the higher rate of interaction gave noticeable less sharp grooves and a tendency for corner threads to form (Figures 19 and 20). The scanning electron microscope work done confirmed there was no evidence of any thermal cracking or layer separation in any of these materials (Figure 21, 22, and 23).

Materials "D" and "C" (aluminum silicon materials) performed well at the lower rate of interaction and higher rubbing velocity. But, with increase in interaction rate and/or decrease in velocity, corner breakage of the seal grooves became more evident (Figures 24 and 25). Confirmation of the corner breakage was done using the scanning electron microscope. Although there was light spark-

Table 6. Showing Hardness Range and Actual Hardness Values for the Various Abradable Materials.

Material	Hardness Range	Hardness of Material
E	*HR15Y 30-50	46HR15Y
F	*HR15Y 40-50	45HR15Y
C	*HR15Y 60-80	74HR15Y
D	*HR15Y 70-90	77HR15Y
G	*HR15Y 70-90	80HR15Y
B	–	**60 Shore Durometer Hardness A-2
A	***HRR 45-65	52HRR (52HR15Y)

NOTES

* 1. HR15Y is a Rockwell superficial hardness test using a 15 kg major load and a 1/2" diameter ball indenter.

** 2. Shore A-2 Durometer Test is an indentation hardness measurement specifically for rubber products and is not convertible to any other hardness scale.

***3. HRR is a Rockwell hardness test using a 60 kg major load and a 1/2" diameter ball indenter.

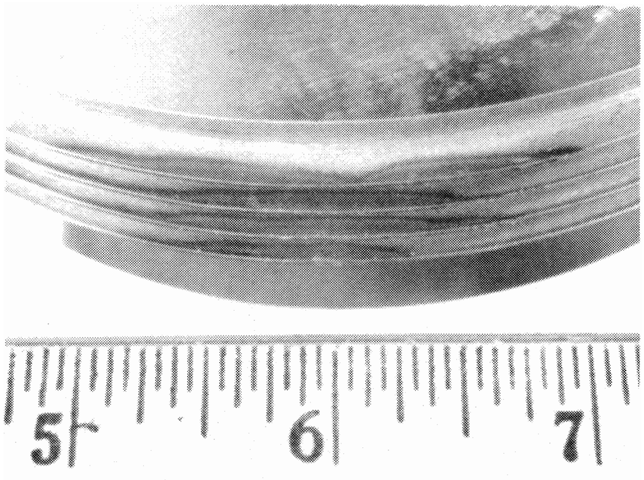


Figure 14. Showing Evidence of Heating of the Rotor Teeth on Test Rotor Number 1.

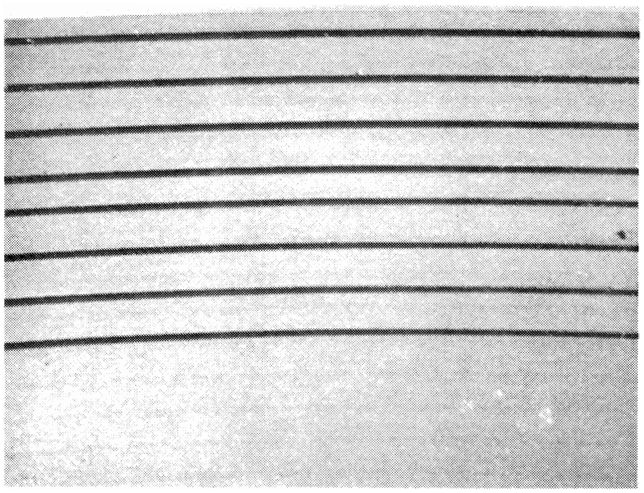


Figure 15. Showing Evidence of Good Abradability for Material "B" in Test Runs 1 and 2.

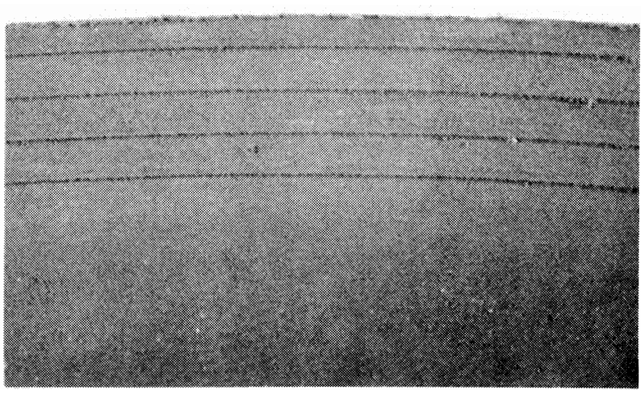


Figure 16. Showing Evidence of Good Abradability for Material "B" in Test Run 32.

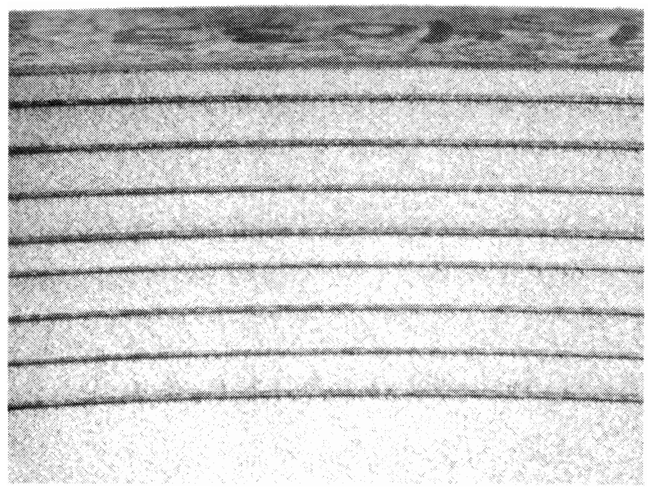


Figure 17. Showing Evidence of Good Abradability for Material "E" in Test Runs 6 and 7.

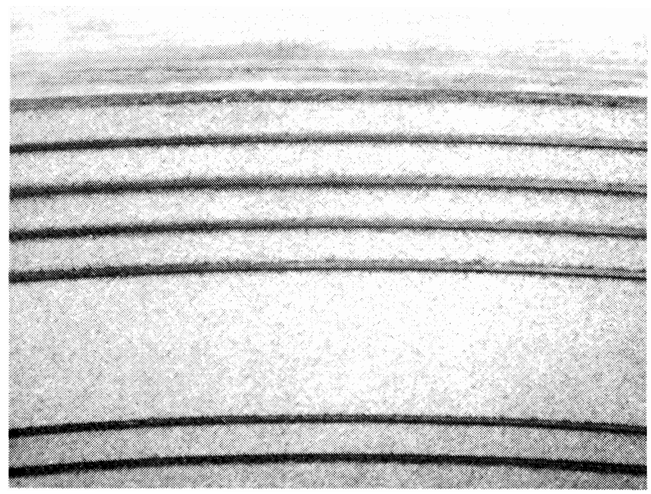


Figure 18. Showing Evidence of Good Abradability for Material "E" in Test Run 28.

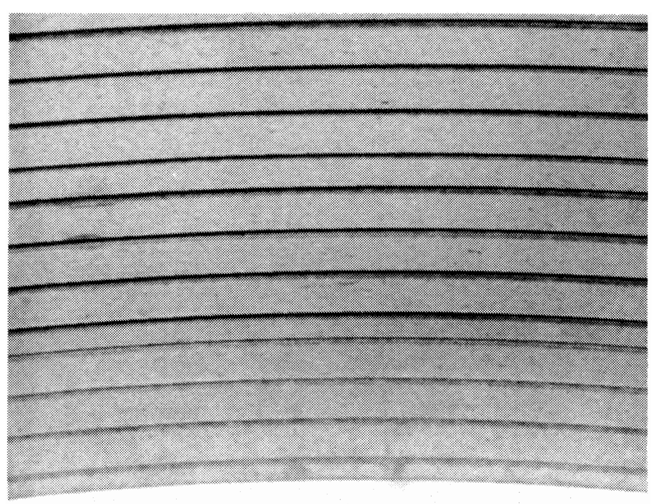


Figure 19. Showing Evidence of Good Abradability for Material "A" in Test Run 8.

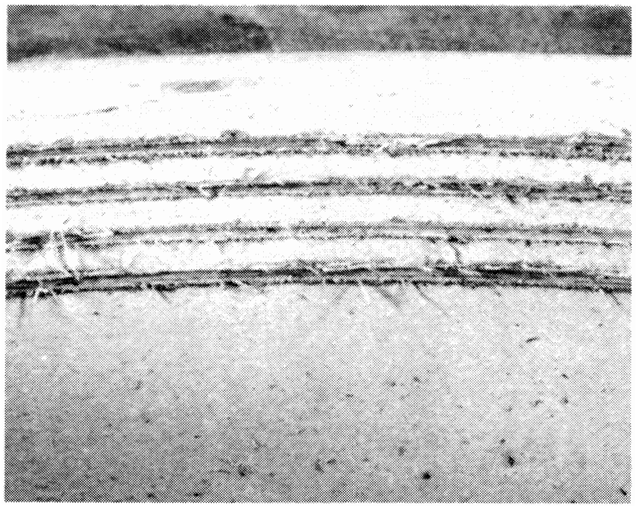


Figure 20. Showing Evidence of Loose Threads Attached to Groove Corners for Material "A" in Test Run 31.

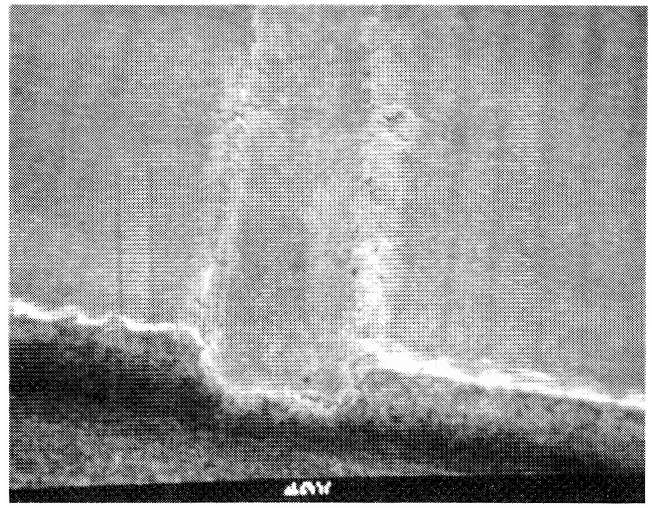


Figure 23. Showing Slight Corner Damage with No Thermal Cracking in Material "A".

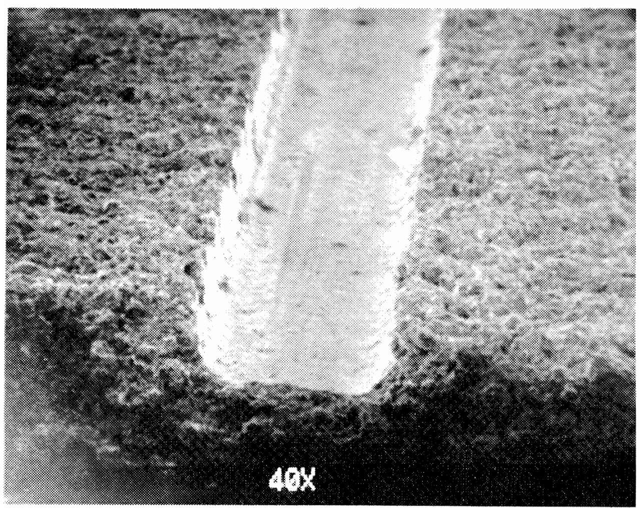


Figure 21. Showing Sharp Groove Edges with No Corner Damage or Thermal Cracking in Material "E."

ing evident, there was no evidence of any thermal cracking or layer separation in any of these materials (Figures 26 and 27). The material "D" was slightly better than "C" in obtaining sharper grooves which may be attributed to a lower porosity level in the material. The high velocity oxygen fuel gas sprayed coatings tend to give a more dense matrix as compared to the oxygen fuel sprayed material. Material "G" (nickel chromium) had good abradability at the higher velocity and lower interaction rate (Figure 28). Although some slight evidence of thermal cracking and chatter was observed, the rotor teeth did not show any appreciable evidence of material removed. However, at the lower velocity and higher interaction rates, excessive sparking was more prevalent which resulted in more thermal cracking. The scanning electron microscopy (SEM) work done shows the effect of thermal cracking and chatter (Figure 29). The test runs at the higher interaction rates and lower velocities were not completed due to violent reaction between the rotor and nickel chromium material. Although there was excessive interaction between the rotor and sprayed materials, none of the tests showed any evidence of failure of the kind between the coating and substrate.

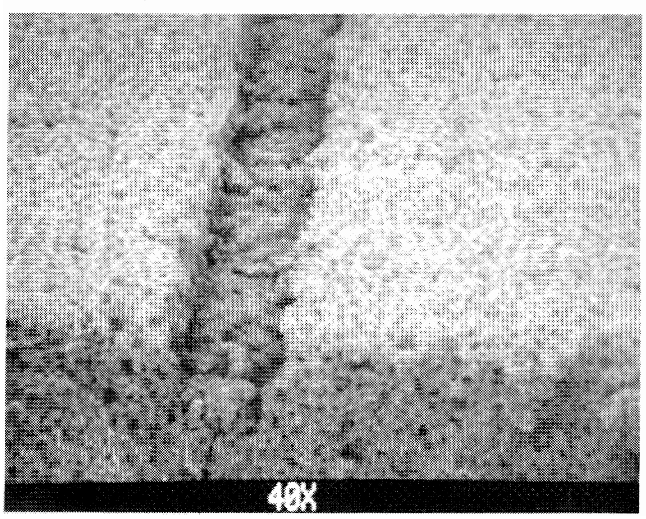


Figure 22. Showing No Corner Damage or Thermal Cracking in Material "B".

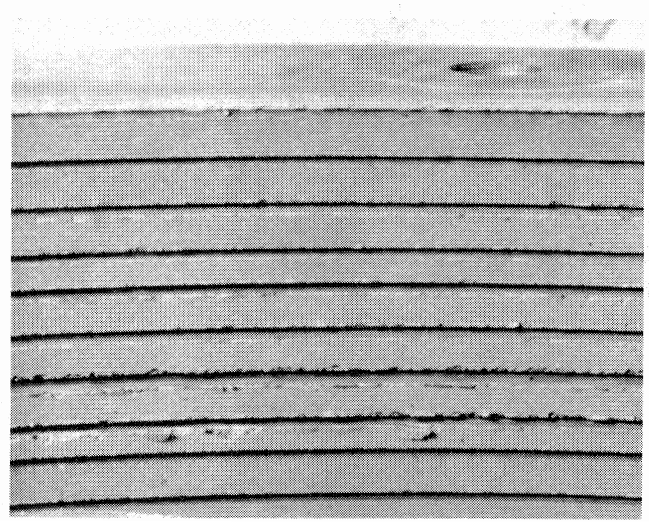


Figure 24. Showing Evidence of Corner Breakage for Material "C" in Test Runs 22 and 23.

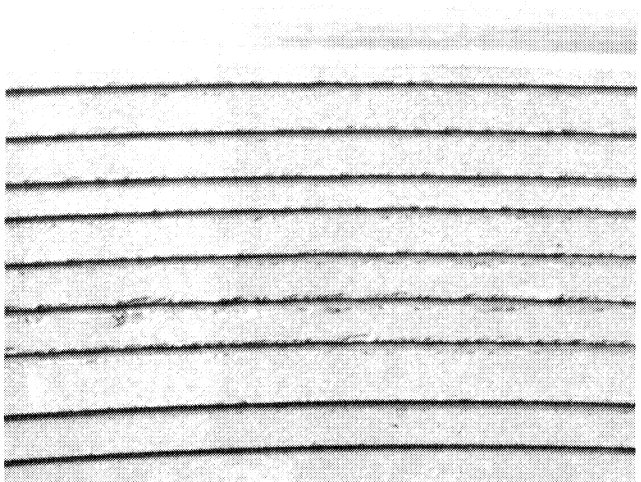


Figure 25. Showing Evidence of Corner Breakage for Material "D" in Test Runs 24 and 25.

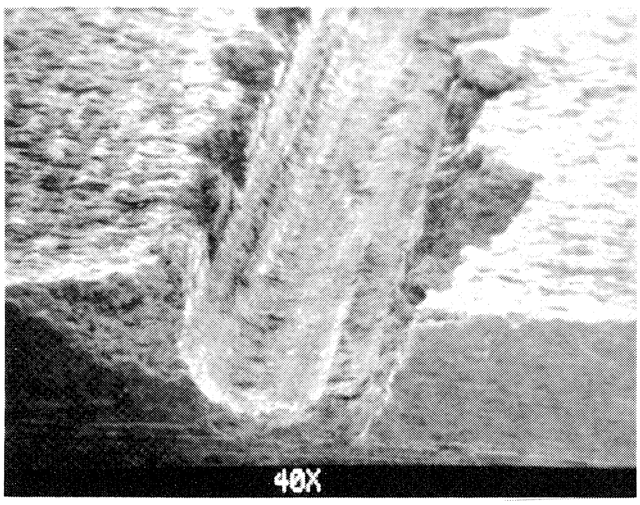


Figure 26. Showing Evidence of Corner Breakage but No Thermal Cracking or Layer Separation for Material "D."

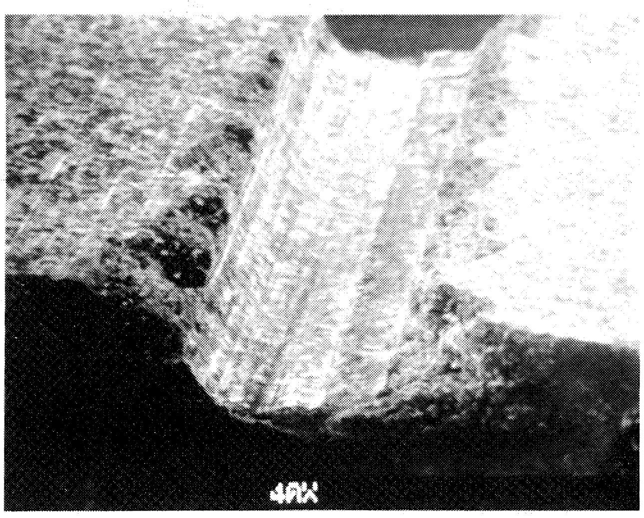


Figure 27. Showing Evidence of Corner Breakage but No Thermal Cracking or Layer Separation for Material "C."

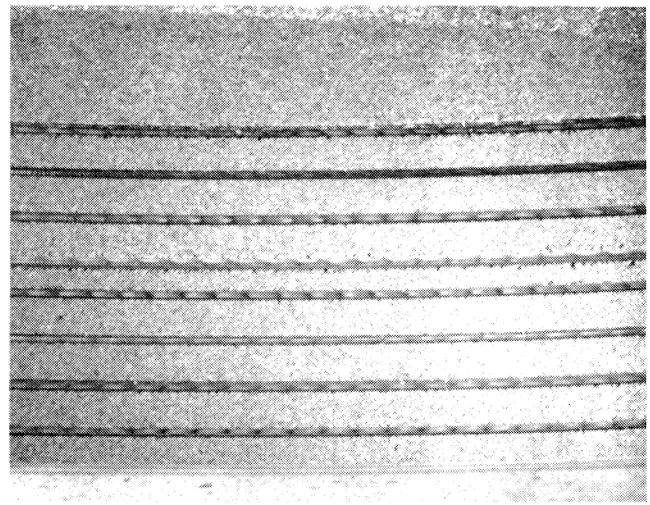


Figure 28. Showing Evidence of Good Abradability for Material "G" in Test Runs 6 and 7.

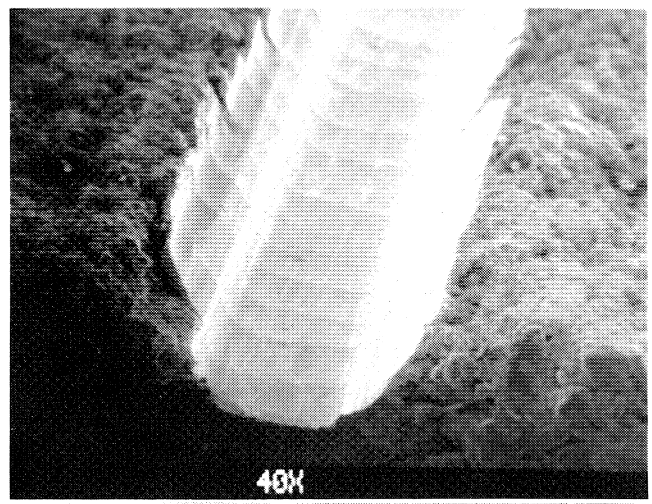


Figure 29. Showing Evidence of Thermal Cracking and Chatter for Material "G."

The material "F" (silicon graphite) produced the poorest results of all the materials tested. Not only was there evidence of corner breakage, but extensive failures between layers of coatings were observed (Figure 30). Since the tests at higher velocity and low interaction rate were unsuccessful, this material was not tested at the lower velocity. The results obtained were not surprising since this material was the most difficult to apply in the spraying process. Obtaining the optimum process parameters for the sprayed coatings is vital in order to achieve the desired abradability needed for the intended operating conditions.

The SEM work done confirmed evidence of corner breakage (Figure 31). In some cases the grooves formed in the tested material were slightly wider than the teeth in the rotor and the depth of the grooves were slightly greater than the fixed 0.021 in radial intrusion of the rotor. This may have been attributed to bits of debris from the abraded seal being lodged in the groove thereby causing what has sometimes been called *self erosion* of the seal. The wear debris of the silicone rubber stuck to the seal surface to a greater degree than that of any of the other abradable materials. It, however, is so soft that there was no detectable effect attribut-

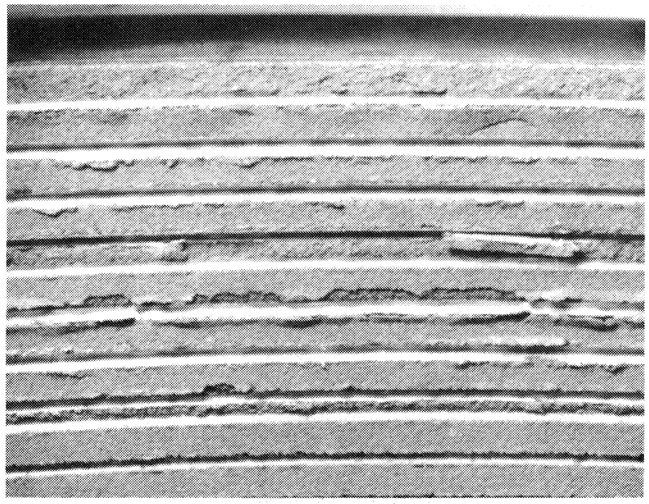


Figure 30. Showing Evidence of Corner Breakage and Layer Separation for Material "F" in Test Runs 20 and 21.

able to self erosion. It was observed that greater depths occurred with the material "F" (aluminum silicon graphite) which may have been attributed to the poor quality of the spray coating.

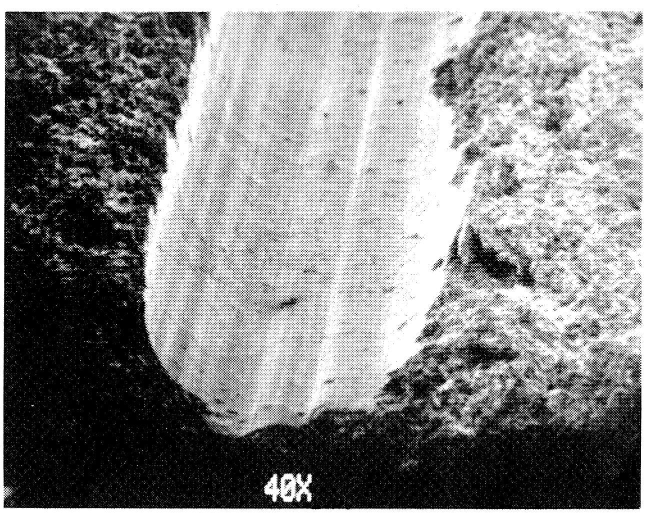


Figure 31. Showing Evidence of Corner Breakage for Material "F."

The hardness results done on the material indicated that the harder materials tended to have the lower abradability, while the softer materials gave good abradability. In all cases, the actual hardness fell within the optimum range selected R15Y30 to R15Y80 (Table 6).

The test results indicated that with the exception of materials "G" and "F," all of the materials are suitable for application in compressors. However, there are other material factors that must be considered when selecting the most suitable abradable material. One of these factors is temperature limits which varies for different materials as indicated in Table 7. This temperature restriction must be taken into account when selecting the most appropriate material for a known process application. Although the mica filled TFE has a restrictive minimum temperature of –40°F, there is a question whether this is justified. If one considers that this material has negligible toughness at room temperature any further reduction in temperature may not adversely effect the already low toughness. The silicon rubber material at lower temperatures (–100°F) becomes glass-like and, consequently, would deteriorate the abradability property of the material. There is limited information on sprayed materials, but based upon general information, it appears that these coatings could be used to –320°F. Since they are deposited in a form that is both porous and abradable, the strength of these materials will be low even at any temperatures.

Corrosion resistance is another property factor that must be considered when selecting the material. The mica filled TFE is generally impervious to corrosion attack from most gaseous mixtures and consequently would be acceptable under all hydrocarbons, gases, and sour natural gas to chlorine gaseous conditions. The silicone rubber material is recommended for most hydrocarbon gases. However, exposure to oil, water, and steam would have an adverse effect on the materials performance. The aluminum silicone coatings "C" and "D" would operate under the same gaseous conditions that are generally applied to conventional aluminum seals. Exposed to extreme sour natural gases and chlorine gaseous conditions, these materials would be attacked severely. The nickel graphite coating "E" could be used for all hydrocarbon gases and most sour natural gases and chlorine gaseous conditions.

Table 7. Temperature Limits for Various Materials.

Abradable	Temperature (°F) Minimum	Temperature (°F) Maximum	Recommended Product Line Application
A	–40	350	Compressor
B	–80 to –100	500	Compressor
C	(1)	650	Compressor
D	(1)	650	Compressor
E	(1)	900	Compressor-Steam Turbine
F	(1)	900	Not Recommended
G	(1)	1200-1400	Steam Turbine

(1) Believed to be applicable to –320°F

The coefficient of expansion of the material must be taken into account when designing the seals. The mica filled TFE material has a coefficient well above that of steel and can cause some difficulties in the overall design of the compressor. Since the sprayed coatings are only 0.1 in thick bonded to the metallic substrate, then the coefficient of the substrate would apply for design purposes. For silicone rubber, which is 100 percent dense, the material is soft and elastic. Consequently, since the strength of the material is well below that of the substrate, the silicone rubber will deform elastically to accommodate any thermal strains caused by the substrate.

For steam turbine application, the materials recommended are nickel graphite, and nickel chromium materials. The aluminum silicon polyester coatings would not be suitable for steam environment, due to corrosive attack. Both the nickel graphite and the nickel chrome coatings could accommodate the temperature range for all steam turbine conditions. It is the authors' opinion that the nickel chromium coatings would be used more appropriately at the higher temperature and where steam erosion may be a concern.

Erosion, whether by a process gas or wet steam, is a factor that must be considered when selecting the abradable material. Since the coatings need to be porous to obtain these good abradability properties, this could have an adverse effect on the erosion resistance of the material. At present, the authors' company is running a series of tests to evaluate the abradable seals in a steam turbine environment. Although the nickel chromium did not rank high in abradability compared to other materials, the material may per-form better for erosion resistance and especially at the higher temperature range.

In all cases, whether for compressor or steam turbine application, availability and ease of refurbishment becomes important factors when selecting the abradable material. The sprayed abradable seal materials, whether by high or low pressure oxygen fuel process, lend themselves more easily for refurbishment during a machine overhaul, compared to fiber metal and metal honeycomb which require an extensive brazing operation to attach these materials to the substrate seal. Refurbishment of a sprayed seal would require removal of damaged sprayed material by machining followed by recoating with either existing or new spray coatings. All the sprayed materials except the nickel chromium (a proprietary coating) are readily available from a number of suppliers. The mica filled TFE not only has a size limitation of 37 in diameter, but when damaged, becomes a redundant part and must be replaced as a complete new seal. The silicone rubber can be refurbished by vacuum injection molding, but this involves special tooling and, therefore, a quick field repair is not possible. Other possible techniques for applying the silicone rubber are pour molding and adhesive bonding of injection molded silicone rubber to the substrate. A lot of difficulties have been encountered in pour molding in preventing air getting into the mixture. This can form unacceptable air bubbles, especially if the size and clusters are large. An aircraft engine manufacturer claims to have proprietary procedures that produce acceptable pour molded seal components. The other method is to adhesively bond vacuum injection molded strips to a metallic substrate. Preliminary tests done by the authors' company have indicated that this is a viable method. The tests indicated that the adhesive bond shear strength was greater than that of the silicone rubber material.

CONCLUSIONS

- For centrifugal compressor application the following materials are recommended:
  - Mica filled TFE (Material A)
  - Silicone Rubber (Material B)
  - Nickel Graphite (Material E)
  - Aluminum Polyester (Material C & D)

Temperature limits vary among these materials.

- For steam turbine application the following materials are recommended:
  - Nickel Graphite (Material E)
  - Nickel Chromium Alloy (Material G)

Again, temperature limits must be taken into account.

- Higher interaction rates and/or lower velocities have a greater effect on the abradability of the tested material, i.e., corner breakage, thermal cracking and self erosion were more evident.
- Performance improvements in excess of two percent have been calculated for noncondensing steam turbines utilizing abradable materials in shaft end packing, blade tip and stage diaphragm seals.
- Application of abradable seals to centrifugal compressors have shown efficiency improvements of up to 0.5 percent per stage for high flow machines and up to 2.5 percent per stage for low flow machines.

FUTURE WORK

- Further studies should be concentrated on high interaction rates and low rubbing velocities.
- The test trials highlighted in the paper were of a configuration that allowed wear debris to be easily disposed from the grooves. Further test trials need to be done to assess the effect of self erosion from wear debris in a simulated operating condition.
- Further testing needs to be done to assess the erosion aspect of the abradable materials in a steam environment. This work is already underway.

REFERENCES

1. Cameron, J., "Abradable Seals for Compressors and Steam Turbines," Elliott Company Technical Report 375 (1989).
2. Gresh, M. T., "Compressor Performance: Selection, Operation, and Testing of Axial and Centrifugal Compressors," Butterworth-Heinemann, Stoneham, Massachusetts (1990).
3. Lucas, G. M., Streeter, R. T.; "Steam Turbine Abradable Seal Erosion and Wear Test," ASLE Preprint 84-AM-5.A-1, (1984).
4. Hoppock, W. G., Silvaggio, J. A., and Van Bramer, K. G., "Steam Turbine Abradable Seals," *Proceedings of the 11th Turbomachinery Symposium,* Turbomachinery Laboratory, Department of Mechanical Engineering, Texas A&M University, College Station, Texas (1982).
5. Novinsky, E. R., "The Selection and Performance of Thermal Sprayed Abradable Seal Castings for Gas Turbine Engines," SAE Aerospace/Turbine Plating and Metal Finishing Forum, (March 1989).
6. Sundberg, D. V., Dennis, R. E., and Hurst, L. G., "Abradable Compressor and Turbine Seals," 1, NTIS NoCR-159600, Project Completion Report, NASA Contract NaS3-20073 (May 1979).