THERMOPLASTIC LABYRINTH SEALS FOR CENTRIFUGAL COMPRESSORS

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ABSTRACT

For over 15 years now, the efficiency and reliability of centrifugal compressors have been enhanced by the application of thermoplastic materials to the eye, shaft, and balance piston labyrinth seals. Traditionally these seals have been manufactured from metallic materials and have required relatively large clearances for reliability reasons. By upgrading to carefully selected engineered thermoplastics, the clearances can be reduced without sacrificing reliability. This results in increased compressor efficiency and the added benefit of easier installation. This tutorial will review the design and application of thermoplastic seals as used in centrifugal compressors. The tutorial will not cover in any detail abradable seals (such as babbitt or lead lined), reduced cross-coupling seals, or "other" seals (such as honeycomb, brush, carbon ring, dry gas, etc.).

INTRODUCTION

Labyrinth Seals

A labyrinth seal is a series of annular orifices utilized to seal a region of high pressure from a region of low pressure. These are "clearance seals" and as such, have a certain amount of leakage. In a centrifugal compressor the impeller eye seals, the shaft seals between impellers, and the balance piston seal (Figure 1) all are sealing a high pressure area from a low pressure area. These seals can consist of rotating "teeth" sealing against a smooth abradable surface, referred to as tooth on rotor (TOR) designs (Figure 2); stationary teeth sealing against a smooth rotating surface (tooth on stator or TOS) (Figure 3); or a combination of these (interlocking) (Figure 4). TOS and TOR seals are sometimes called "see-through" seals since there is no interlocking of teeth, which would obstruct the view when looking between the sealing surface and the teeth.

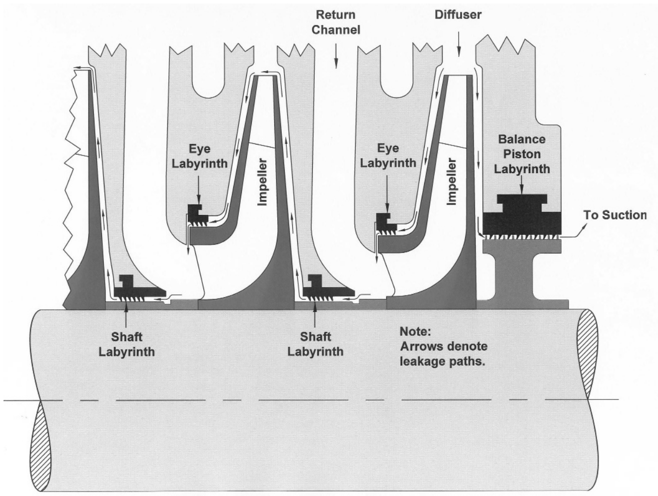

Figure 1. Partial Section Through a Typical Centrifugal Compressor Illustrating Eye, Shaft, and Balance Piston Seals as Well as Typical Leakage Paths.

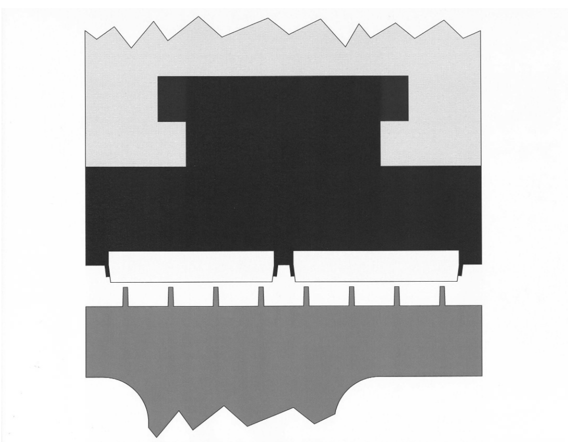

Figure 2. Tooth on Rotor (TOR) Configuration.

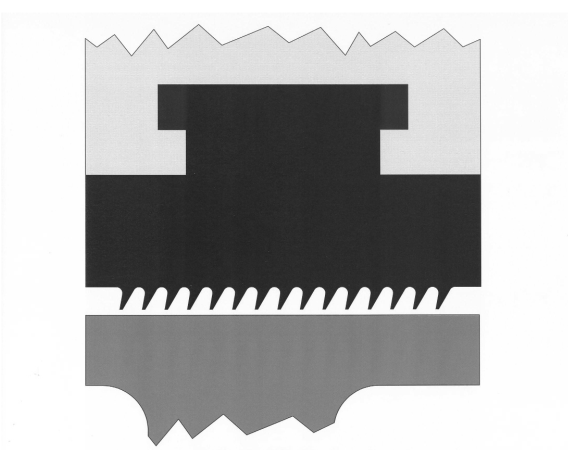

Figure 3. Tooth on Stator (TOS) Configuration.

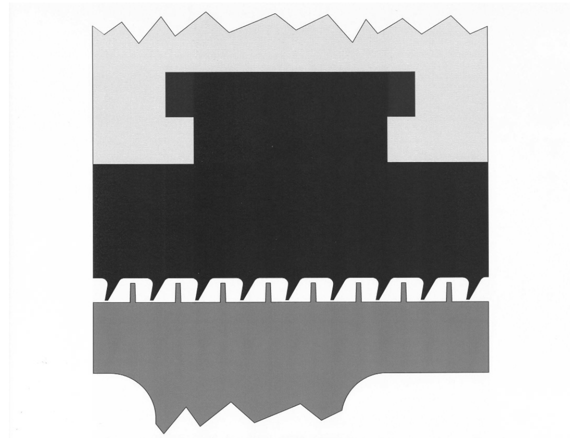

Figure 4. Interlocking, or Hi-Low, Configuration.

The flow of a fluid can be categorized in one of two areas: turbulent flow and laminar flow. By definition, turbulent flow occurs when the local fluid velocities and pressures fluctuate irregularly, in a random manner. The drawing in Figure 5 is an airfoil with turbulent flow behind it. Also by definition, laminar flow is flow that is not turbulent. The drawing in Figure 6 is of an airfoil with laminar flow passing around it. Turbulent flow involves energy transfer in the fluid and, for the case of a labyrinth seal, this turbulence helps to reduce leakage. Figure 7 is a drawing that illustrates the turbulence between teeth in a TOS seal.

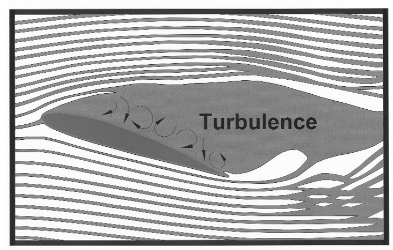

Figure 5. Turbulent Flow as Demonstrated by Airflow over an Airfoil.

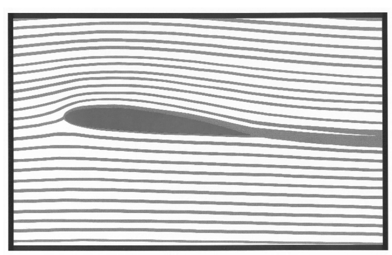

Figure 6. Laminar Flow as Demonstrated by Airflow over an Airfoil.

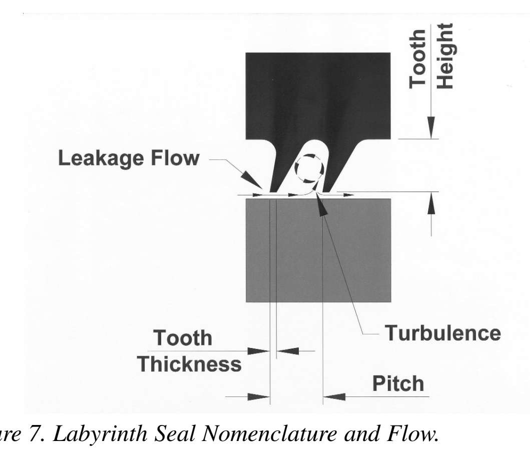

Figure 7. Labyrinth Seal Nomenclature and Flow.

Labyrinth seals (labys) in centrifugal compressors can have a significant impact on compressor efficiency. In a centrifugal compressor, work is done to increase the gas pressure and this pressure is contained by labyrinth seals throughout the compressor. Gas that leaks past one of these seals needs to be recompressed. Since the gas has already been compressed once, it is hot and therefore needs even more energy to get it back up to pressure. Compressor efficiency can be improved by reducing labyrinth seal leakage.

Labyrinth Seal Leakage

The first usable publication on the calculation of the leakage of labyrinth seals was by H. M. Martin (1908). His publication "Labyrinth Packings" presented equations that could be used to estimate leakage through labyrinth seals. Martin's work was expanded upon by Egli (1935) with his widely referenced paper, "Leakage of Steam Through Labyrinth Seals," which is still used today to estimate labyrinth seal leakage. Twenty-five years later Geza Vermes (1961) further expanded upon Martin's work by presenting leakage equations for straight, stepped, and combination seals.

At the time of the above referenced work, the impact of seal leakage on overall machine efficiency was considered trivial. It was not until the mid to late 1960s that the impact of labyrinth seal leakage on turbine and compressor efficiency started to become a concern. Also around this time, it became clear that labyrinth seals could influence the rotordynamic behavior of a turbomachine. Further work on labyrinth seals then started to concentrate on their impact on rotordynamics; leakage flow concern became secondary once again. Oddly enough, most of the modern day computer programs that are used to calculate rotordynamic coefficients of labyrinth seals use a version of Martin's equation to estimate the axial flow through the seal. This is because axial flow impact on the coefficients is trivial; it is the circumferential flow that creates the destabilizing forces. For an excellent discussion on labyrinth seal impacts on rotordynamics, refer to Childs' book, *Turbomachinery Rotordynamics, Phenomena, Modeling, and Analysis* (1993).

Additionally, some work has been done on further understanding the leakage of labyrinth seals. Present day labyrinth seal leakage research uses laboratory testing and computational fluid dynamics (CFD) tools to predict seal leakage. This work has resulted in understanding the effect of clearance on leakage (Rhode and Hibbs, 1993), the effect of tooth thickness on leakage (Rhode and Hibbs, 1992), the impact of rounded teeth (Zimmerman, et al., 1994), and rub grooves (Denecke, et al., 2002) on seal performance.

Labyrinth Seals in Centrifugal Compressors

How much of an impact can a labyrinth seal upgrade have on compressor performance? For the purposes of this discussion, we will assume leakage is directly proportional to clearance. We will also assume that the labyrinth eye, shaft, and balance piston seals account for 4 percent of the compressor's efficiency loss. In this case, if the leakage could be reduced by 50 percent by reducing the seal clearance by 50 percent, we could appreciate a 2 percentage point increase in efficiency. If all the major compressors in an ethylene plant (cracked gas, propylene, and ethylene compressors) realize a 2 percent efficiency increase this could equate to a \$700,000 annual energy savings in a two billion lb/year facility. This example assumes the efficiency gains would be used to reduce power consumption. Quite often however, the gains can be used to allow the plant to produce more product, and this could result in an even greater positive economic impact.

However, the seal clearance cannot just be arbitrarily reduced, because reduced clearance seals can rub and open up—resulting in a negation of the efficiency gain, possible rub related vibration problems, and possible damage to the rotating element. Labyrinth seals made from high performance thermoplastics, when properly designed and installed, can be used because when they rub, during normal transients such as traversing a critical speed, they will "give" and then regain their original "prerub" geometry. This is the main driving force behind the use of reduced clearance thermoplastic labyrinth seals in centrifugal compressors.

A typical metallic labyrinth seal in the "as-installed," "during rub," and "after rub" conditions is illustrated by the drawings in Figure 8. Note the seal is installed with clearance to the shaft in the "as-installed" case. The "during rub" drawing depicts the rotor contacting the seal and causing permanent deformation of the seal tooth tip. Therefore, in the "after rub" case the tooth remains deformed and excessive clearance results, leading to increased leakage. As shown, it is possible that galling can take place on the rotating surface as contact between the metallic seal and the rotor occurs. Also during the rub, enough energy may be imparted to the rotor to cause vibration problems and the associated reliability concerns. The drawings in Figure 9 depict a similar chain of events with a thermoplastic seal installed. In this case, the "as-installed" clearance is typically tighter than with the metallic seal. During the rub, the tooth deflects, moving with the rotor during this transient. After the rub the tooth regains its original "as-installed" configuration, no damage occurs to the rotor, and the initially reduced clearances are regained.

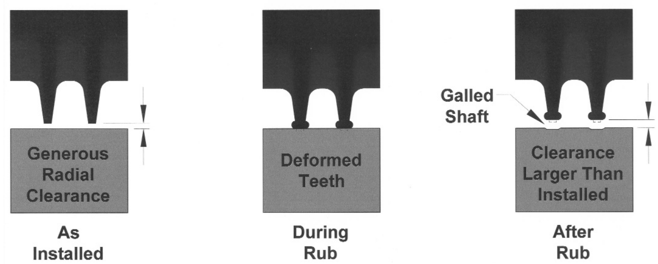

Figure 8. Typical Metallic Seal Prerub, During Rub, and Post Rub. (Note the permanent deformation of the metallic tooth and the possibility of galling damage to the rotating element.)

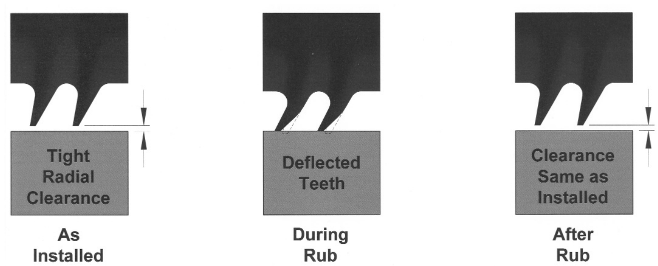

Figure 9. Typical Thermoplastic Seal Prerub, During Rub, and Post Rub. (Note the flexing of the tooth during contact and the damaged free rotating element after the rub.)

Over the years, the authors have seen many seals that have been examined after a typical multiyear run. In almost all of the cases, the seal bore has remained unchanged with no appreciable damage noted to the tooth tips. When clearance or bores have been checked, they have been very close to the as-installed values from years before. This counters the users' previous experiences with aluminum labyrinth seals where the seals usually show signs of contact and may have even scored the rotating sealing surface. From time to time machines have problems and run for extended periods in a high vibration situation. For these cases, there has been damage to the thermoplastic seals noted. However, based upon most users' experiences, this damage is significantly less than what would be expected if aluminum seals were installed. Just about every one of these cases also confirms that damage to the rotor does not occur with thermoplastic seals installed.

THERMOPLASTICS

The most common thermoplastics used to manufacture labyrinth seals for centrifugal compressors are TORLON® and PEEK™. Other materials used include Fluorosint® and Vespel® products. All of these products are supplied in various grades where the blending of the final product can influence mechanical properties and chemical compatibility.

Most plastics fall into one of two categories. They are either thermosetting plastics or thermoplastics. Thermosets chemically change (crosslink) during processing so that they will never melt again. Thermoplastics are melted and frozen into the desired shape during processing. Thermoplastics can be melted again, are thermoformable, and can be processed by a variety of methods including injection molding, compression molding, and extrusion.

Most of the products that are used for labyrinth seals are produced by the compression molding process. In this process, the material is packed into a mold and compressed in a press. The material is compressed to about 1/4 the original pack. Most common molds cannot be more than 24 inches tall to fit in the press. Therefore the maximum final ingot lengths are about six inches. Ingots longer than six inches could also exhibit pressure decay in the center of the ingot wall, resulting in lower mechanical properties. The product must be heated during the molding process; either the entire mold itself is heated with heater bands in the press, or the compressed material is secured in the mold under pressure and the entire pressurized mold is transferred to the oven for heating. PEEK™ materials require a post mold cure to finalize curing and to reduce molding stresses.

Thermal Properties

Thermal properties to consider include the glass transition temperature (Tg), the melting temperature (Tm), the continuous use temperature (CUT), the heat distortion temperature (HDT), the stiffness at temperature (DMA), creep resistance, and the coefficient of linear thermal expansion (CLTE).

The glass transition temperature is the temperature at which the polymer chains of a thermoplastic become active and the polymer begins to soften. Below the Tg, thermoplastics are rigid; above the Tg, they become rubbery. The melt temperature is the temperature at which the polymer flows freely.

The heat distortion temperature is based upon an American Society for Testing and Materials (ASTM) test (ASTM D-648) in which a standard test specimen (typically 1/2 inch wide by 1/2 inch thick by 5 inches long) under a load of 264 psi will deflect .010 inch (5 percent). Essentially the HDT is the temperature at which the flexural modulus of the polymer is reduced to 100,000 psi.

The stiffness at temperature can be quantified by dynamic mechanical analysis (DMA). DMA curves plot modulus as a function of temperature and can be used to evaluate the mechanical properties (specifically stiffness) of a material at an elevated temperature. Figure 10 is a plot of DMA for various engineered thermoplastics including Fluorosint® and various grades of TORLON® and PEEK™. TORLON® is polyamide-imide, abbreviated PAI. PEEK™ is a member of the polyaryletherketone polymer family. Note that the knees in the curves roughly correspond to the glass transition temperatures of the various materials.

Coefficient of linear thermal expansion is a very important property to consider when evaluating the suitability of a material for a high temperature, close tolerance application. The CLTE describes how the size of a part will change with changes in temperature. Most materials expand when heated and shrink when cooled. The CLTE is used to calculate how much expansion or contraction will be observed when a part is heated or cooled. The smaller the CLTE, the more dimensionally stable a part made from that material will be as temperatures are varied up and down. Figure 11 is a plot of CLTE versus temperature for two grades of PEEK™, a friction and wear grade of TORLON® and aluminum (a typical nonpolymer labyrinth material). Note the dramatic change in CLTE when the PEEK™ materials traverse the Tg; also note that the fillers have no significant impact on the Tg.

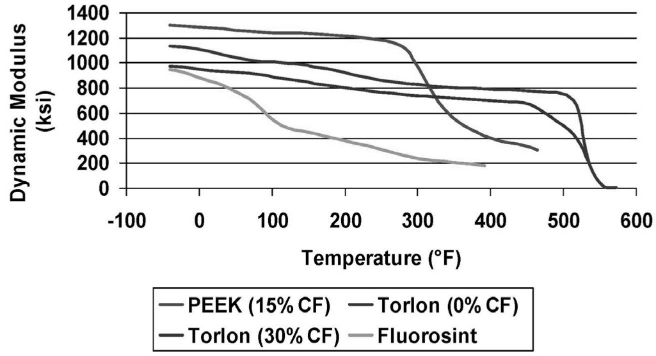

Figure 10. DMA Plot for Various Thermoplastic Materials.

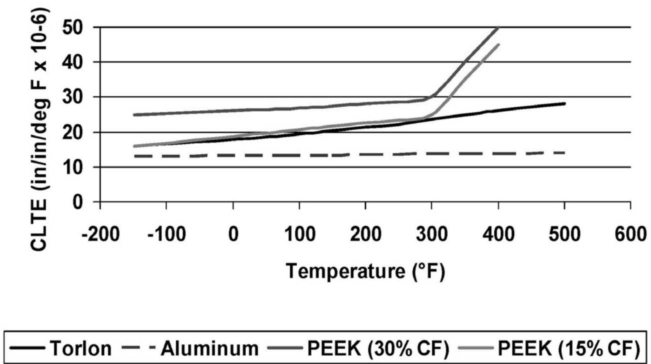

Figure 11. Coefficient of Linear Thermal Expansion (CLTE) Versus Temperature for TORLON®, PEEK™, and Aluminum. (Note the sharp knee in the PEEK™ curves, the start of this knee corresponds to the Tg of the material. Also note how the thermoplastic CLTEs increase with increasing temperatures.)

Continuous use temperature is the temperature a polymer can be exposed to for 100,000 hours (11.4 years) and still maintain 50 percent of its original (initial) properties. It represents the loss of mechanical properties due to thermal aging of the material. It has been found that this material degradation manifests itself as a loss of ductility when TORLON® seals have run at high temperatures (over 350°F) for an extended period (two to three years or more). This phenomenon is due to partial oxidative crosslinking of the polymers. As it turns out this increased brittleness does not seem to affect the performance of the seal but it does result in seal breakage upon removal from the compressor. This is because in service the stresses on the part are such that the brittleness does not come into play. However, upon removing the parts, there are other stresses put on the part, and tooth breakage has occurred. This phenomenon has only been found in the discharge end of air compressors (due to the high temperature found there), and the only real problem is the need to plan to replace the seals whenever the case is split.

Amorphous and Semicrystalline

Most thermoplastics can be categorized as either amorphous or semicrystalline. By definition, the term amorphous pertains to a solid that is noncrystalline, having no definite structure. Amorphous thermoplastics have their Tg close to their melt temperature and, as such, are not usually used above their Tg. Unlike semicrystalline thermoplastics, amorphous thermoplastics do not normally require reinforcements in their blend. Semicrystalline pertains to the crystal-like ordered structure of the material. These plastics do become amorphous above their Tg and their Tg is usually well below their Tm. Semicrystalline thermoplastics can be used above their Tg with the proper reinforcement (carbon or glass fibers). PEEK™ is a semicrystalline plastic while TORLON® is completely amorphous; there are no crystals in TORLON®.

TORLON® and PEEK™ have significant advantages over general engineered plastics in that they can be used at higher temperatures; they have better dimensional stability and better chemical resistance. Figure 12 is a plot of the various thermoplastics' thermal properties for comparison purposes.

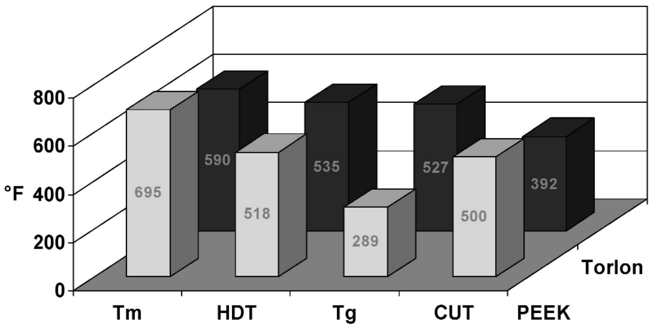

Figure 12. Relative Thermal Properties of TORLON® and PEEK™.

Chemical Resistance

Chemical attack depends on temperature, concentration, pressure, and time.

The chemical resistance of PEEK™ is very good, having resistance against halogenated hydrocarbons and is partially resistant against strong oxidizing acids. As a semicrystalline polymer, PEEK™ is highly resistant to chemical attack but will be attacked by concentrated (over 30 percent concentration) strong acids at high temperature. PEEK™ is sensitive to chromic acid, hydrofluoric acid, nitric acid, and chlorine (wet and dry). PEEK™ is unaffected by acetic acid (at 10 percent concentration), amines, and hydrocarbons.

The chemical resistance of TORLON® is very good against oils, solvents (chlorinated, fluorinated), and hydrocarbons. TORLON® is partially resistant to dilute acids and alkalis. It has poor resistance against strong acids, alkalis, and oxidizing agents. TORLON® also has poor resistance against steam and boiler feedwater (since most feedwater additive packages contain amines).

As a PTFE polymer based compound (polytetrafluoroethylene, which is better known by the trade name Teflon®), Fluorosint® is highly chemically resistant to aliphatic hydrocarbons (butane), aromatic hydrocarbons, strong oxidizing acids, dry and wet chlorines, ammonia, and amines. It is sensitive to fluorinated hydrocarbons.

Summary of Thermoplastic Use in Compressors

The following discussion contains information the user of these materials may find useful. It is meant to supply information directed to the use of the materials for centrifugal compressor labyrinths. When comparing data sheets it is important to realize that the molding method can influence some properties. For compressor labyrinths, the thermoplastics typically must be compression molded, while the datasheets may be for compression, extruded, or injection molded shapes. Fillers can also significantly influence selected properties, making it important to understand what fillers are used and why (for instance, glass fillers should never be used because of their potential to damage the rotating element).

TORLON®

The two grades of TORLON® commonly used are 4540 and 7530; these are both compression-molded grades. The 4540 grade is composed of PAI blended with 12 percent graphite powder and 5 percent PTFE, this is the most commonly used grade for labyrinths. The graphite powder and PTFE supply low friction properties and classify the material as a bearing grade, or a friction and wear grade. The 7530 grade is blended with 1 percent PTFE and 30 percent short, high modulus, carbon fibers; the carbon fibers adding strength and lowering the CLTE. Both of these properties act to make the material suitable for higher temperature applications and/or applications requiring added strength.

TORLON® is challenging to machine because it has very low conductivity and most of the heat of machining is absorbed by the machine tool, thereby breaking the tool down quickly. TORLON® also has a higher CLTE than most metals (about twice that of aluminum) making close tolerance machining require a "stabilizing" period prior to making final cuts. This ensures the part has cooled enough to ensure it will not change dimensionally once thoroughly cooled to room temperature. This can be even more of a concern with parts of significant cross sections; the outer portions of the part may be cool but the inner core still warm enough to influence the dimensions. This problem is compounded with the 7530 grade, as the shearing of the carbon fibers adds to the heat generated, breaking down tooling faster and putting more heat into the part, requiring more stabilizing time.

Both grades of TORLON® are hygroscopic; they absorb moisture. This needs to be considered when storing parts for an extended period (several months or longer), as the dimensions of the parts can change as the material swells. Should this happen the parts can be dried out by placing them in an oven at 200°F for 24 to 72 hours. Care must be taken to not use a drying temperature that is too hot as the parts can blister as the entrained moisture turns to steam, creating thermal shock.

The key property of TORLON® is its Tg; at around 527°F it is the highest of all thermoplastics and therefore has the highest use temperature of the reviewed materials. Another key property is the HDT; at 534°F, the material still maintains a flexural modulus of 100,000 psi. TORLON® is also creep resistant, has good chemical resistance, is self-lubricating, has a low coefficient of friction, and a relatively low CLTE, as compared to some other plastics.

TORLON® is only available from Quadrant Engineering Plastic Products. The formulation was originally an Amoco product with the base resin produced at an Amoco plant in Greenville, South Carolina, and molded at an Amoco facility in Georgia. Now Solvay owns the resin plant and the product is molded by Quadrant, at its plant in Reading, Pennsylvania, utilizing technology and equipment acquired from Amoco.

PEEK™

PEEK™ (polyetheretherketone), as used for centrifugal compressor labyrinths, typically comes in two compression-molded grades. The exact grade designation and formulation depend on the vendor but, as with TORLON®, there is a standard friction and wear grade, and a stronger friction and wear grade. The standard friction and wear grade typically has about 15 percent carbon fibers, 10 percent graphite powder, and 2 percent PTFE. The higher strength grade has 30 percent carbon fibers and a small amount of PTFE (as a mold release agent), and is used where the added strength is needed.

Like TORLON®, PEEK™ is challenging to machine to close tolerances due to its low heat conductivity and high thermal expansion coefficient. However, even the regular friction and wear grade of PEEK™ has 15 percent carbon fibers making the machining even more difficult due to the shearing of these fibers. The stabilization time is also important when finish machining PEEK™ seals.

Unlike TORLON®, PEEK™ requires a post mold heat treatment to reduce molding stresses and to increase crystallinity. This involves a careful heating to the glass transition temperature, holding for a period of time determined by the part cross section, then slow cooling to room temperature. Most responsible vendors have tight specifications they follow on this process. It is very important to ensure the material has the desired properties to be able to machine parts to close tolerances. From time to time, another heat treatment is required during the manufacturing phase if material movement can result in an unacceptable part. This is because mechanical and thermal stresses imparted into the component during the machining process may result in a part that goes significantly out of round such that it cannot be properly installed in the compressor.

The friction and wear grade of PEEK™ has an HDT of about 518°F, which is close to TORLON®, but its Tg of 289°F is much lower than TORLON®. PEEK™ has very good chemical resistance (better than TORLON®) and good dynamic fatigue strength (making it a great material for wear components in reciprocating compressors).

The base PEEK™ resin is produced by a few companies, but Victrex® Industries produces most of the PEEK™ that is compression molded for compressor labyrinths. Several molders take the base PEEK™ resin (called "neat" or "virgin" PEEK™), blend it with their own proprietary formulas, and mold into ingots.

Fluorosint®

Fluorosint® 500 is synthetic granular mica reinforced PTFE that is very chemically inert but has low strength and a low HDT of 210°F. It is very abradable and easy to machine, compared to TORLON® and PEEK™. Fluorosint® is very weak with a tensile strength of 1100 psi at room temperature. This needs to be taken into consideration when designing seals from this material. It is really best used when inserted in a metallic holder and used as an abradable stationary seal, running against rotating teeth, allowing the teeth to cut into the soft material.

Other Labyrinth Seal Materials and Designs

As mentioned earlier, there is a myriad of seal designs and materials that are used in centrifugal compressors. As with Fluorosint®, some are abradable; that is they are designed with smooth bores and the rotating teeth "cut" into the material during a rub situation. These include babbitt lined, lead lined, feltmetal, and nickel graphite lined seals.

Other seals used in centrifugal compressors are applied for their positive impact on the rotordynamics of the machine. These include honeycomb seals and damper seals.

Babbitt and lead lined seals employ a metallic base ring, usually a low-carbon steel, with a babbitted or lead lined bore. These components are usually used as balance piston seals, and from time to time, as end case seals in some air compressors. The soft metal lining (babbitt or lead) seals against teeth that are machined onto the sealing surface outer diameter. These rotating teeth can then abrade easily into the soft metal allowing close running clearances. These seals are sometimes difficult to fit since the long smooth surface needs to be scrapped to ensure they are installed with the proper clearance. The babbitt and lead can also flow when high temperatures and/or high-pressure drops are encountered. There are several instances where the babbitt or lead flowed during process upset conditions and failed the seal.

Feltmetal seals are used from time to time as another smooth bore option. These seals do not have the same "flow" problems as lead or babbitt, but they are difficult to scrape to fit. The feltmetal is an abradable material that is brazed, or otherwise fused, to a metallic holder. There are few vendors of this technology and often the lead times are significant.

Nickel graphite seals are similar to feltmetal, but the abradable surface is sprayed into the bore of the metal holder. Several companies can perform this metal spraying. Nickel graphite is very abradable, so much so that it is easy to damage in handling and installation. Sharp edges (such as at the split line of the seal) are susceptible to damage. It is not uncommon to observe an apparent gap at the splitline of the seal as the sharp edge breaks down. Nickel graphite is one of the better seal materials for smooth bore applications where rotating teeth ride against the seal bore. This type seal can be used to 900°F, and, when properly applied, the bond strength is 2000 psi. Since there is a bond between the nickel graphite and the base metal there is an area of potential failure. The authors, however, are not aware of any failures attributed to this.

Honeycomb seals use a honeycomb material brazed to a metallic holder. Since the honeycomb surface retards gas flow in all directions it minimizes the cross-coupling present in labyrinth seals. This destabilizing cross-coupling is generated by the circumferential gas flow in a seal, much as the circumferential oil flow in a sleeve bearing can introduce cross-coupled terms. By retarding this circumferential flow, the destabilizing forces can be minimized. Honeycomb seals can also introduce direct damping, a positive rotordynamic force, by utilizing the column of gas in the individual honeycomb cells to achieve damping. Hole pattern seals have recently been developed as another option to this concept. The hole pattern concept utilizes several radial holes drilled into the bore of the seal, resulting in a design similar to the honeycomb (Childs and Wade, 2004). The benefits of the hole pattern seal include:

- No need to braze a material to a backing ring.
- The seal is relatively easy to manufacture.
- There can be many sources for the seal (currently there are few sources for honeycomb seals).
- Lower cost and shorter delivery (due to the above).
- The holes sizes can be any design desired, whereas the honeycomb material configuration is dictated by available honeycomb materials.

Pocket damper seals are a special labyrinth seal with geometry that reduces cross-coupled stiffness terms and introduces relatively large amounts of positive damping. This seal design is attractive for rotordynamics reasons, much like the honeycomb and hole pattern seals.

ENGINEERING

When performing an upgrade from a set of metallic labyrinth seals to a set of thermoplastic seals there are several factors that need to be considered. For instance the thermoplastic materials have relatively high thermal expansion coefficients (two to three times that of aluminum), and these coefficients are not linear with temperature (the higher the temperature the larger the coefficient). Therefore, thermal expansion calculations need to be performed. Thermoplastic seals also have less strength than the metallic being replaced, so pressure-area forces and compressive stresses (from differential thermal expansion) need to be considered. In addition, during the upgrade process there is an opportunity to revisit the labyrinth design and evaluate the possibility of optimizing the number of teeth, the tooth pitch, the tooth height, and raking the teeth to encourage turbulence. In order to properly address all these issues certain engineering data are required.

Data Required to Engineer a Compressor Upgrade

In order to properly address the issues introduced above the following information is required:

- Process makeup, including any injections, for chemical compatibility confirmation. This does not need to be overly detailed if there is a concern with disclosing proprietary information, just enough information to evaluate the material compatibility with the gas.
- Suction and discharge temperatures and pressures, and any other available temperatures and pressures (including side streams, extractions, interstage, etc.), for thermal expansion calculations and material mechanical property determination.
- Speed, for centrifugal growth estimations.
- Cross sectional drawing of the compressor, for determination of seal arrangement for tooth rake and other considerations (such as

whether the compressor is horizontally or vertically split and locations of additional suction and/or discharge nozzles).

- Sample seals for reverse engineering, or appropriate drawings, as a starting point for thermoplastic seal drawings.

- Actual sealing diameters for rotor being installed. For optimal efficiency, it is advisable to size the seal bores to the actual impeller eye, shaft sleeve, and balance piston diameters. Quite often these areas are undersized due to skim cuts taken to reclaim the surfaces after a rub with the aluminum seals (rubs with thermoplastic seals do not damage the rotating element). If available, it is also advisable to supply runout values for consideration during the engineering.

- Bearing clearances; as will be discussed later will be needed to size the operating clearance of the upgraded seals.

- For completeness, it is convenient to record the original equipment manufacturer (OEM) of the compressor, the compressor frame, OEM design clearances, unit designation, other designations for the compressor, the driver type, and even the rotor serial number.

Thermal Expansion

As mentioned earlier, the thermal expansion of the seal is an important consideration in the engineering phase of an upgrade project. As the plot presented earlier (Figure 11) shows, the coefficient of linear thermal expansion of the thermoplastics is considerably higher than aluminum and even higher yet than the surrounding materials (impellers, shaft sleeves, diaphragms, etc.). Generally the CLTE of the compressor internals will be around 6.5E-6 in/in/°F while aluminum is about twice that and the thermoplastics are about two times that of aluminum (making the CLTE of the thermoplastic seals about four times that of the surrounding compressor components). Since the desire is to design to close clearances, it is important to consider the thermal expansion to understand how the clearances will change with temperature. This discussion will assume the seal is above its manufacturing temperature; the section will conclude with a discussion on low temperature applications.

The first step in the process is to estimate the temperature at each seal location. The OEM usually has this information readily available, but for an aftermarket upgrade, it is sufficient to estimate these values by assuming a linear change in temperature from suction to discharge. For configurations that are more complicated it may be necessary to assume stage efficiencies and calculate stage temperatures accordingly; and consider mass flow rates and mixing when side streams are involved. Usually this level of detail is not required since it takes gross errors in stage temperatures to have significant impacts on the final seal design.

Once the seal temperature is determined, the mechanical properties of the thermoplastic are determined. This can be done by going to available published data and interpolating or, if these calculations are performed on a regular basis, it may make sense to curve fit the data and use the resulting equations to calculate these properties. The properties calculated are the CLTE, the strength, and the modulus of the material (since both strength and modulus drop off at elevated temperatures).

Now that the mechanical properties of the seal are known and the temperature has been estimated the expansion of the parts can be calculated. One way to do this is to calculate the free thermal expansion of the following:

- The seal bore and fit diameter.

- The sealing surface outside diameter (OD).

- The diaphragm bore at the seal fit.

With these data, it is now possible to determine the amount of "crush" or interference between the seal and the diaphragm. Since the modulus of the thermoplastic is so low and the section is relatively thin compared to the diaphragm, it is safe to assume that all of this crush will act to close up the bore of the seal. At this time, the compressive stress in the material can be calculated and compared to the compressive strength. Should the stress be too high a redesign or material change can be considered.

Low Temperature Considerations

The above discussion assumed the seal operating temperature was higher than the manufacturing temperature, causing the seal to grow due to thermal expansion. For low temperature applications (such as the suction end of refrigeration machines), it is important to consider the contraction of the seal relative to the diaphragm and the rotating sealing surface. The most important consideration is to ensure the hook of the seal (assuming the seal has a "hook") has sufficient clearance after the parts have thermally stabilized. With low temperature applications, the seal contracts relative to the diaphragm, and it is important to ensure the hook still has clearance to the diaphragm after this contraction. This ensures the seal does not "hang up" on the hook, forcing the split line open. Thermoplastic seals have successfully been applied to applications as low as 150°F below zero. Since there is limited low temperature information on these materials, it is often necessary to extrapolate the thermal and mechanical properties from the high temperature data.

Impeller Eye and Balance Piston Centrifugal Growth

The growth of the sealing surfaces due to rotation needs to be considered to ensure the seals do not end up at zero or negative clearance at speed. This can be calculated utilizing finite element techniques, but this is quite time consuming and costly to perform just for a thermoplastic seal upgrade. Usually the impellers in a compressor change substantially from stage to stage requiring the analysis be performed for each stage. Another technique used by one company was to spin the rotor up to speed in a high-speed balance facility and use noncontact displacement probes at the eye locations to observe the centrifugal growth, again this is very costly and time consuming.

It is sufficient to use an annular disk centrifugal growth calculation that uses modifying factors to model impeller geometries. This is the least accurate calculation performed during an upgrade, but it is certainly accurate enough as proven by thousands of seals that are running that were designed this way. Because of the uncertainty in the growth calculation, it is advisable to design the seals to have clearance at operating speed. One method to determine this clearance is to set it equal to the bearing clearance. This is convenient since the clearance is tied to journal size, which is tied to machine size. Therefore, larger compressors will have larger designed operating clearances.

As the drawings in Figure 13 illustrate, the radial clearance of a thermoplastic seal in a horizontally split compressor can be designed to match the radial bearing clearance. In vertically split machines (barrel compressors), it is advisable to increase this design clearance somewhat since usually as-installed clearances cannot be checked. Using 1 1/2 times bearing clearance has worked well in the past, but it may be advisable to consider the machine design and age to estimate how well the seals will be aligned to the rotor after the bundle is installed in the compressor, and the rotor is up on bearings. One other factor to consider with thermoplastic seal clearance is the interlocking seal case where rotating teeth seal against the bore between stationary teeth. For these cases, it has worked well when the clearance between the rotating teeth and the smooth bore between thermoplastic stationary teeth is set to twice bearing clearance. PEEK™ and TORLON® materials do not work well as abradable seals so it is important that the seal be designed to avoid interaction in this area. The drawing in Figure 13 illustrates this clearance rule of thumb concept.

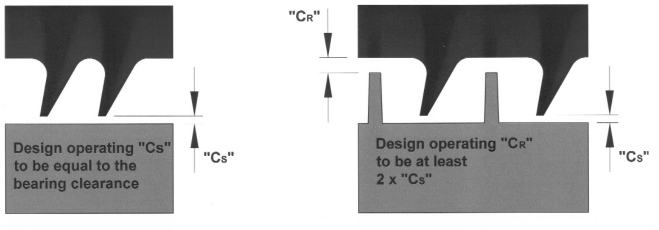

Figure 13. Drawing Illustrating Thermoplastic Seal Clearance Guidelines.

Butt Gaps

A butt gap is a designed-in gap between the two halves of a split seal. The theory is that by calculating the linear thermal expansion of the part a butt gap can be designed such that this gap will close at operating temperature. For true optimization, this gap would need to be a different value at each stage of the compressor since the temperatures (and therefore the circumferential thermal growth) are different for each stage.

However, since the compressive modulus of the materials used is so low, it is often more convenient to design for zero butt gaps and let the part absorb this growth as a compressive stress. For most applications, this stress is a couple orders of magnitude below the compressive strength of the material.

Impact on Rotordynamics

Labyrinth seals can influence the stability of the compressor. It is known that conventional labyrinth seals do impart cross-coupled forces to the rotor acting to lower the logarithmic decrement. Considerable work has been done over the years to calculate these forces. Most of the modern techniques use computational fluid dynamics programs to evaluate these forces.

For the most part, an upgrade to thermoplastic seals results in:

- Reduced operating clearances.
- Optimized tooth profiles.
- Optimized tooth height and pitch.
- Possibly more teeth.

These factors have a trivial impact on the seals rotordynamic coefficients. If there is an issue where reduced seal cross-coupling is desired, then the seal redesign can accommodate this by incorporating swirl breaks, shunts, or other stabilizing geometry.

Stress Analysis

As stated earlier, the modulus and strength of these materials decreases with increasing temperature, and this needs to be considered when performing stress calculations. The plot in Figure 14 demonstrates how the strength of selected thermoplastic materials drops off with increasing temperature. Also plotted is strength versus temperature for aluminum 6061-T6 material; it is important to keep in mind that the aluminum is ductile while the thermoplastics are not. Pressure area forces that may overstress axial hooks must also be evaluated. With an early balance piston seal, there was a failure when the pressure area forces overcame the strength of the part, and it broke in service. Since these materials are considerably more brittle than aluminum, stress concentration effects must be considered. Because of this, extra care should be used in the design phase to reduce these effects by using generous radii and minimizing other stress riser areas.

The photographs in Figures 15 and 16 are of the balance piston seal that failed at the hook in service. Figure 17 is a picture of another seal that was redesigned to minimize the impact of pressure area forces on the weaker thermoplastic material. In this case, a TORLON® insert was manufactured to roll into an aluminum holder. This allows the stronger aluminum to absorb these forces while taking advantage of the thermoplastic properties at the labyrinth teeth. This Figure also illustrates the interlocking seal design described earlier.

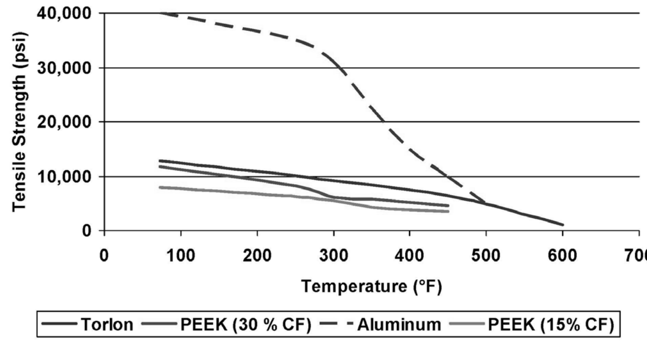

Figure 14. Material Tensile Strength Versus Temperature.

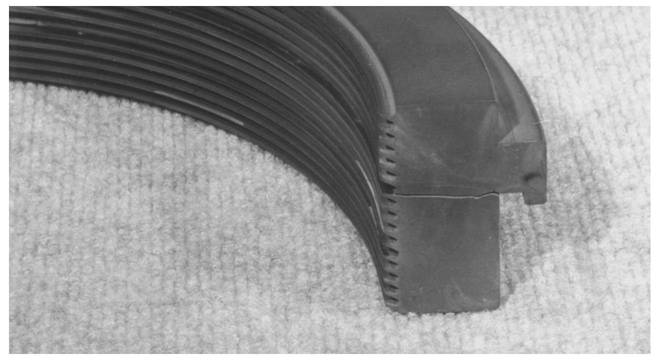

Figure 15. Picture of Broken Balance Piston Seal.

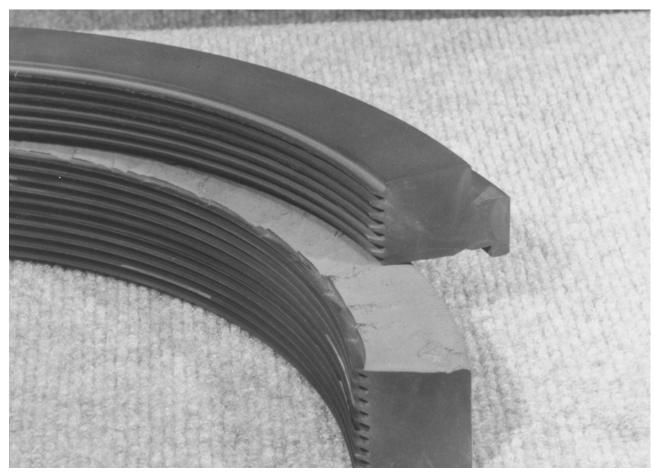

Figure 16. Another Picture of Broken Balance Piston Seal.

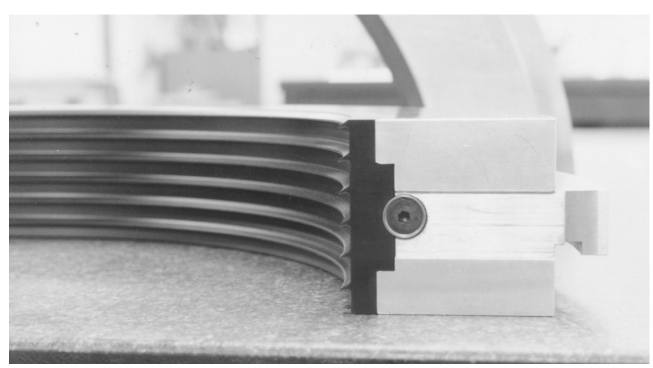

Figure 17. Picture of Thermoplastic Insert in Metallic Holder. (Note that this seal is of an interlocking balance piston design, having rotating teeth running between the stationary teeth.)

Material Selection

Due to its high Tg, good thermal and mechanical properties, and ease of machining, TORLON® 4540 is the material of choice for centrifugal compressor labyrinth seal upgrades. Even though a PEEK™ blend may be acceptable, TORLON® is a better choice because of its high Tg, since it can handle higher heat from process upsets and the heat generated by rubs. In addition, since it is the most stable to machine, it is the preferred material choice for manufacturing considerations. Of course, there are drawbacks to TORLON® including the fact that it is not as chemically inert as PEEK™; it is hygroscopic; and due to its molding process, it is available in limited ingot sizes. PEEK™ materials have a much larger selection of mold sizes to choose from, allowing material to be procured closer to the net shape, thereby reducing material cost and machining time.

The main reason to use PEEK™ is when chemical compatibility issues arise. When evaluating the materials against the process conditions, it is occasionally discovered that PEEK™ must be used instead of TORLON®; an example is the cracked gas compressors in an ethylene plant; with some plant designs there is the possibility of a caustic carryover into one or more compressors, requiring the use of PEEK™ at these locations.

THERMOPLASTIC SEAL PACKAGING, STORAGE, AND INSTALLATION

Thermoplastic parts are different in many ways from the metallic parts being replaced. Care should be taken with the packaging, storage, and installation of these seals.

Packaging and Storage

Of the thermoplastic materials covered in this tutorial, only TORLON® is hygroscopic, that is it will absorb and retain moisture. Fully soaked TORLON® parts (soaked in 180°F water for six months) can swell as much as 2 percent dimensionally (5 percent by weight), making them unacceptable for installation. The parts can be dried out by placing them in an oven for a few days at about 200°F. Care should be taken not to get the parts too hot as the entrained moisture can flash to steam and blister the seal, causing significant damage.

An acceptable way to address the moisture absorption issue is to:

- Coat the part with mineral oil to act as a barrier to moisture.
- Package the part in airtight packaging.
- Add desiccant bags to the packaging to absorb captured moisture.
- Do not open the packaging until just prior to installation.

Moisture absorption in service is not an issue because the seal is trapped by the diaphragm, the gas flow sets up a boundary layer, and the higher operating temperatures keep the part dry. As an example, an air compressor in an ammonia plant in Louisiana ran for over two years, and when the case was split, the seals rolled right out—they had not swelled at all in service, even through the very humid summers in Louisiana. Oddly enough, the spare seals, which had been removed from their packaging before being stored in the nonhumidity controlled warehouse, had absorbed moisture from the atmosphere and swelled significantly. These seals were carefully dried out and returned to their original design dimensions, but this took several days and could have affected the turnaround timing.

A last concern on packaging is the fact that these parts have less strength than the metallic parts being replaced and are relatively brittle. An impact that would normally dent an aluminum tooth may break a thermoplastic tooth. As such, it is important to carefully package these parts, preferably with form fitting packaging materials, to ensure they are not damaged while the packages are being handled.

Handling and Installation

As stated above thermoplastic seals are more fragile than aluminum seals, and they can be damaged while being handled. Areas to watch out for are the teeth, especially at the split line where they are most vulnerable, and any thin sections such as flanges. This is another reason why it is a good idea to keep the parts packaged until they are needed.

Again, careful handling during installation will help ensure a successful upgrade project. Thermoplastic labyrinths have some very good characteristics that make them very easy to work with, and generally speed up the installation into the compressor. Care must be used to ensure that the proper seal is installed in its proper location. In many compressors, there are seals that will have approximately the same bore but are designed to be installed in different locations within the compressor. There may be instances, for example, where the same OEM part number was installed in two or three different diaphragms, but now, due to eye diameter changes and/or temperature variations, there are individual thermoplastic seals for these locations. The location is important to ensure the integrity of the upgrade.

Upon installation, it is important not to force or drive a seal into its fit or diaphragm. If a seal is tight in the hook or is rubbing hard on the shaft, a careful evaluation must be made to determine the extent of the problem. Minor problems can be corrected with 100 to 300 grit emery paper. Light sanding on the tight spot will remove material at a rapid rate. Care must be taken to ensure that too much clearance is not put into the seal with the use of the emery paper.

The following installation procedures should be followed after first checking all labyrinths to assure that they have been stored properly and are damage free, and to verify the shaft and impeller dimensions are correct. Then:

1. Install both top and bottom halves of the labyrinths in the compressor case with the rotor removed to be sure the hook fit is correct.

    a. Stand laby next to its fit to assure that OD is same size as fit in case. Large differences should be investigated before an attempt is made to install laby.

    b. If the laby is tight rolling into hook, check the seal for rubs or scratches that may indicate a burr in the hook. Do not force the laby into the hook. Remove the burr or sand the laby to obtain a good fit. The laby should roll in without force. In addition, use a lubricant such as WD-40® as needed.

2. After the labyrinths have been installed in the top and bottom case and the fit is okay, remove the seals and install the rotor in the machine. Reinstall the seals in the bottom case.

3. Once the rotor has been installed in the lower half of the case, a careful check must be made to ensure that the seals have clearance and ensure that the rotor is going through the center of the diaphragm. Start at one end of the machine and use long feeler gauges to check the split line clearances. Once this has been done and recorded, minor sanding of the seals may be needed to achieve clearance. A diaphragm that is sitting to the left or right can cause this problem.

4. In the top half, it is recommended that masking tape be layered on the rotor (masking tape is approximately.005 thick), then lower the upper casing. It is recommended that tape be applied just above the split lines (refer to Figure 18) and top dead center. Raise the upper casing and look where the masking tape has been touched by the laby; this will be the clearance. Sand seals as needed to achieve proper clearance. *Note:* Be especially careful when lowering the upper casing as any impact on the thermoplastic seals could damage or break them.

The above instructions cannot cover every scenario or problem that might arise, but are a general guide. It is recommended that

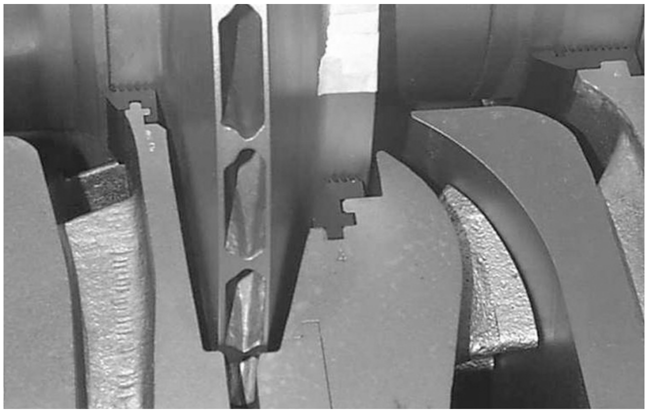

Figure 18. Photograph Illustrating the Use of Tape to Determine Radial Seal Clearance in the Upper Half of the Compressor.

first time users of thermoplastic labyrinths contract with the seal manufacturer to supply experienced personnel for the initial installation, to get a feel for the proper procedures.

Startup procedures vary from machine to machine, but care must be taken to ensure that the machine is not left in a high vibration mode that may affect the seals and cause damage.

UPGRADE PAYBACK CALCULATIONS

For the purposes of evaluating a project for justification, it is common to use the "rule of thumb" that the upgrade may yield efficiency improvements of 1/4 to 1/2 percent per impeller. It is important to realize that this is a "rule of thumb" defined (Merriam-Webster, 2000) as follows:

1: a method of procedure based on experience and common sense

2: a general principle regarded as roughly correct but not intended to be scientifically accurate.

By definition, a rule of thumb does not need to be scientifically accurate, but it does need to provide useful information, and the above stated estimate has proven to be accurate enough for upgrade evaluations.

Some compressors will experience more of a performance improvement than others will. The factors to consider include:

- High-pressure compressors will benefit more than lower pressure machines since the seals play a more important role in the overall efficiency.
- Low flow compressors will benefit more than high flow compressors because the labyrinth leakage is a larger percentage of the overall flow.
- Compressors with high pressure drop seals (such as balance piston or center seals) that are upgraded will realize more of an improvement than double flow machines (which typically do not have pressure drop seals) or compressors where these seals are not included in the upgrade.
- Older compressors with large clearance seals will benefit greatly from the reduced clearance thermoplastic seals.

When preparing to estimate the value of the efficiency gains, the above factors should be considered and an appropriate factor should be used. Usually even using 1/4 percent per impeller will more than justify the upgrade.

Most projects are evaluated based upon energy savings since these calculations are easiest to perform, do not need to consider feedstock, or end product supply and demand economics, and are usually sufficient for the approval process.

As presented earlier: "If all the major compressors in an ethylene plant (cracked gas, propylene, and ethylene compressors) realize a 2 percent efficiency increase this could equate to a $700,000 annual energy savings in a two billion lb/year facility." This statement, assuming an upgrade efficiency gain of 2 percent, is actually based upon a somewhat conservative efficiency gain estimate. Even though it can be considered "conservative," it should be more than enough to justify the upgrade project costs, which do not take into account the possibility of reduced labyrinth seal installation times.

One of the statements often made is: "After a major turnaround I expect to see compressor efficiency improvements." This is understood and accounted for in the analysis. As the case histories that follow demonstrate, the efficiency gains presented are based upon the historical performance of the compressor after a major turnaround.

Along these lines is the concern that other efficiency improvements made during the turnaround will make it impossible to break out the improvements that can be attributed to the labyrinth seal upgrade. This also is a valid statement. It has been found that if the other efficiency upgrades are sound, then a proportioning of the overall gains can be made, and all projects can be credited enough for justification purposes. It seems obvious that none of the projects would be undertaken if there were not a high level of confidence in their success, and this holds true for the thermoplastic seal upgrades.

Another item brought up is: "I can't measure my compressor efficiency accurately enough to quantify the performance improvement." What is needed is the confidence, due to the application of sound engineering principles and documented case histories, that the upgrade will result in an efficiency gain. There are several cases in the literature (Whalen, 1994) where the upgrade was undertaken independent of other factors that would affect the compressor efficiency, and the gains presented were substantial and reinforce the 1/4 to 1/2 percent per wheel gain predictions. Some of these cases even report total plant output gains attributed to the seal upgrade. The two case histories that follow also document how thermoplastic seal upgrades resulted in cost savings and plant output increases.

CASE HISTORIES

There are several case histories briefly documented in Whalen (1994) that the authors will not present here. Two detailed case histories that are more recent will be reviewed here.

Canadian Ethylene Plant

The majority of the data presented here is taken from Chow and Miller (1998) and from Whalen and Miller (1998). The Chow and Miller paper is "Optimizing Performance of an Ethylene Plant Cracked Gas Compressor Train." At the time of the writing, Chow and Miller were rotating equipment specialists working at a world-class ethylene plant in Alberta, Canada.

The Chow and Miller (1998) paper presents their experiences on coatings and thermoplastic seals as well as a discussion on compressor performance monitoring. The Whalen and Miller (1998) paper mostly covers the thermoplastic seal upgrade from this same outage, which took place in 1996. During this turnaround, thermoplastic seals were installed in two cracked gas compressors and the propylene compressor. The propylene compressor is very large and has some of the largest TORLON® seals running. The first stage eye seal has a 45 inch bore and the balance piston seal has a bore of 36 1/2 inches.

One point addressed was the upgrade of the original balance piston sealing arrangement, which used rotating teeth on the balance piston sealing against a smooth babbitt surface in the balance piston seal. The upgrade involved machining the teeth off the balance piston and utilizing a toothed TORLON® labyrinth seal. This new seal arrangement allowed for an easier seal installation. As discussed earlier, the fitting of thermoplastic seals can be easier, and therefore faster, than the fitting of aluminum seals.

Chow and Miller (1998) reported: "Installation and fitting thermoplastic seals vs. aluminum labyrinths and babbitt balance drum seals has proven to be a real time saver. Memories of fighting, fitting, hammering and occasional breakage of the aluminum seals and memories of scraping, checking and more scraping of balance drum babbitt seals are a thing of the past." They also wrote, "Evaluation of process data has shown a 2-3% efficiency increase per compression stage with the installation of thermoplastic seals."

Chow and Miller's company was pleased with the upgrade and went on to upgrade the compressors in their other unit two years later; "The Rotating Equipment Specialists at the Joffre site believe that the use of thermoplastics is important in optimizing performance, increasing run lengths, and reducing turnaround costs."

Experiences with Thermoplastics at a Texas Ethylene Plant

This case history is documented in a paper by Whalen and Dugas (2000). It discusses the upgrade of seven compressors at a world-class ethylene plant in Orange, Texas.

This case history is very useful, as it follows the upgrade process and includes the upgrade of a single compressor as a benchmark for performance improvement expectations. At this facility, there is a booster compressor ahead of the traditional cracked gas train that can be removed from service without forcing the rest of the unit down. It was decided to upgrade this compressor to evaluate thermoplastic seals. When the compressor was brought down to install the TORLON® seals, the only work performed that would affect efficiency was the installation of the upgraded seals. This allowed the efficiency change to be attributed solely to the thermoplastic seals. After the upgrade, the plant personnel determined the compressor flow increased 3.1 percent and the steam turbine driver steam consumption was reduced 2.7 percent.

Based upon this, the plant easily decided to upgrade the other six candidate compressors that would be opened during the major turnaround in 1998. The compressors upgraded include: the first two cases of the cracked gas train, the two propylene compressors, the ethylene compressor, and the purge propylene compressor. Three compressors not upgraded at that time where the last case of the charge gas, which had abradable seals running against rotating teeth, and the two methane compressors that were being considered for replacement or major rerates. The plant has decided to go ahead and remove the teeth off the last cracked gas compressor and upgrade with thermoplastic seals. As of this writing, the seals are in the process of being manufactured.

Another interesting item with this project was with the eyes of the propylene compressors. These compressors had stepped eyes such that the seal had two small bore teeth with a large bore tooth between them. It was known that these three teeth would seal better than three teeth of the same bore, but it was not known if machining the step off the eye would yield a more efficient design by allowing the use of more teeth. Figure 19 contains a drawing of the stepped and straight bore options. To ascertain which design is more efficient, a computational fluid dynamics analysis was performed. The results of the CFD analysis are summarized by the plots in Figures 20 and 21. The analysis concluded that the stepped bore design was more efficient due to the added turbulence generation caused by the step. Based upon this, it was decided to leave the stepped configuration in place.

A problem discovered during this propylene upgrade project was the inability to install all of the TORLON® seals in the exact same manner as the aluminum seals. Some of the seals are eight segment seals that attach to the diaphragm with five axial bolts per segment, which are 40 axial bolts per seal (Figure 22). With the aluminum seals, the bolts can be secured by staking the aluminum material to trap the bolts. Figure 23 is a photograph of the TORLON® seals installed in the compressor. A redesign was undertaken that involved making a seal that utilized an aluminum holder and a TORLON® insert. This design is presented by the drawing in Figure 24. Note that the resulting components have thin sections so

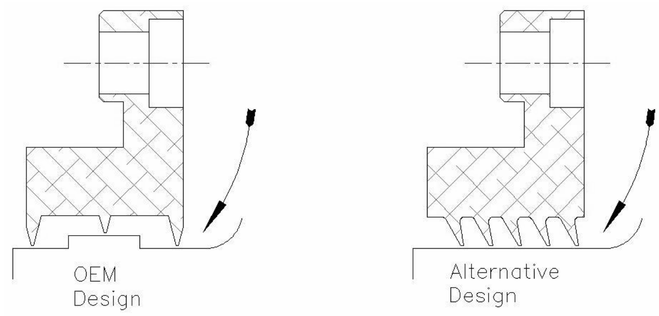

Figure 19. Two Labyrinth Options for Propylene Compressor Eye Seals, Original Stepped OEM Design on the Left and an Alternative "See-Through" Design on the Right.

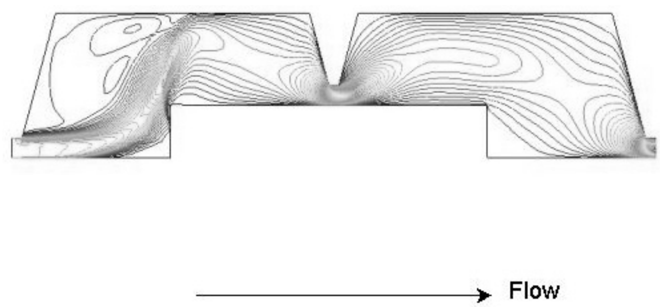

Figure 20. Flow Field Results of CFD Analysis for OEM Design.

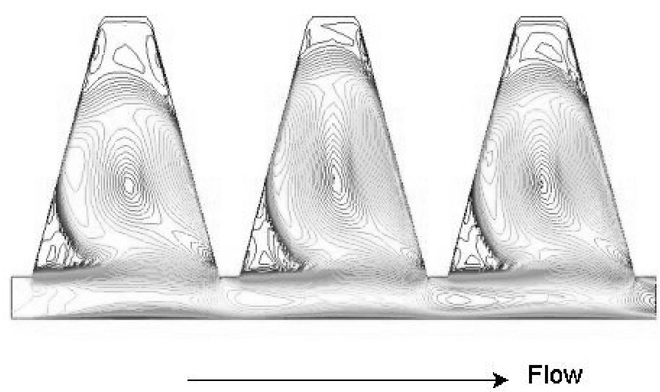

Figure 21. Flow Field Results of CFD Analysis for "See-Through" Design.

a finite element analysis (FEA) was performed to ensure the components were not overstressed. The FEA, as presented by the plot in Figure 25, predicted that the stress levels were well below the strength of the parts. Due to the timing of this redesign, these propylene seals were not installed during the 1998 outage.

As stated earlier, it is difficult to accurately determine the true impact of a seal upgrade project. However, by carefully analyzing all factors the user was able to estimate the following (attributed to the thermoplastic seal upgrade project):

- A 7 percent reduction in steam flow to the ethylene driver.
- A 17 percent increase in head coupled with a 5 percent gas flow increase accompanied by a 5 percent increase in motor power consumption with the purge propylene train.
- A 9 to 16 percent increase in flow with the propylene compressor coupled with an 8 percent speed increase and a 4 percent steam flow increase. (Note: due to the seal problem mentioned earlier only a fraction of the seals were upgraded.)
- A 14 percent flow increase in the charge gas train with a 5 percent steam flow increase.

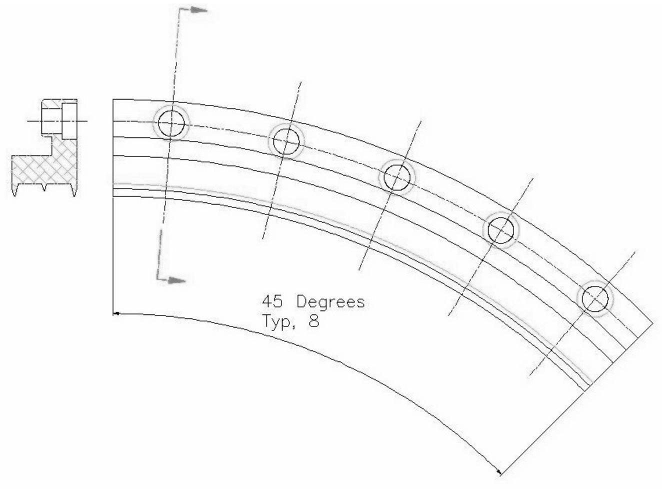

Figure 22. Drawing of OEM Designed Axially Bolted Eye Labyrinth.

Figure 23. Photograph of TORLON® Seals Installed in Propylene Compressor. (Note the axially bolted construction.)

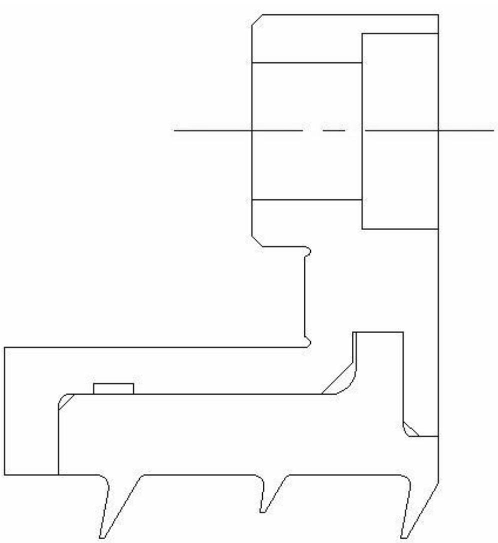

Figure 24. Drawing of Redesigned Eye Seal Consisting of Metallic Holder with TORLON® Labyrinth Insert.

It was estimated that due to the thermoplastic seal upgrade project, the total plant output increased 5 percent. During reduced production process states, the full output increase may not be realized but energy savings more than justify the project costs. As stated in the paper, "…the user is extremely pleased with the performance gains attributed to the polymer seals."

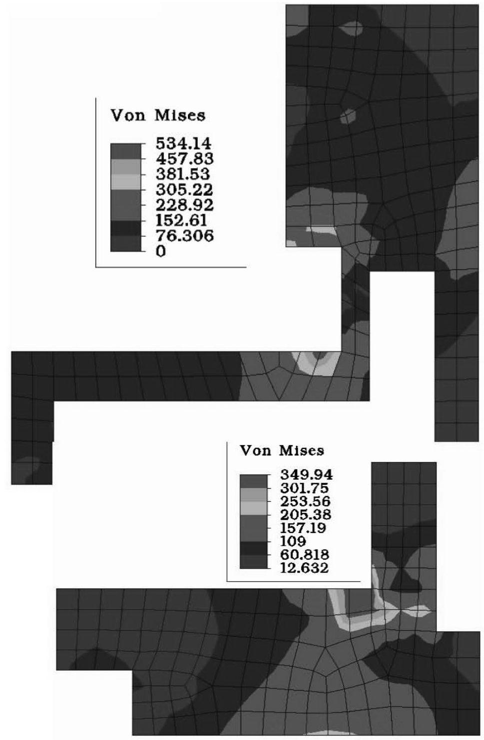

Figure 25. FEA Plot Showing Low Stress Levels in Both the Holder and the Insert for the Redesigned Seal.

CONCLUSIONS

A lot of material has been presented that represents almost 50 years of the three authors' cumulative experience with thermoplastic use as a labyrinth seal in centrifugal compressors. This tutorial attempted to present all relevant material, but it is recognized that there is a lot of ground not covered, and there are also changes occurring regularly that may make some of this information obsolete by the time it is presented. There are new materials being developed, new blends being formulated, new labyrinth seal designs being introduced, and ongoing testing to help understand labyrinth seal flows and dynamic effects.

New Materials

Some of the new materials being considered for labyrinth seals are not necessarily new formulations, but are new in this type application.

Victrex® is marketing PEEK-HT™ a PEEK™ blend formulated for high temperature applications. PEEK-HT™ is a member of the polyaryletherketone family of polymers, generically known as polyetherketone (PEK), and can be molded with carbon fibers and PTFE for high temperature applications. The compression molding technology required for molding large ingots is, as of this writing, currently under development for this new product. Even though the Tg of the material is at 315°F, it does maintain its properties closer to the Tg, has better wear resistance, has long-term creep resistance and higher tensile strength at higher temperatures, and improved compressive strength. It may prove to be a better material alternative for high temperature applications, possibly pushing the maximum PEEK™ use temperature up another 50°F.

Solvay is also marketing a new friction and wear grade of TORLON®, TORLON® 4435. This material was engineered for high pressure-velocity (PV) applications, but may prove to be a logical choice for select labyrinth seal applications. The key to high PV materials is their ability to maintain mechanical properties at elevated temperatures.

Several companies offer a composite PEEK™—continuous carbon fiber material that has good circumferential tensile strength coupled with very low circumferential thermal expansion. The material is weak in the other principle directions making its use as a labyrinth material limited. There are, however, similar materials being developed that allow orientation of the fiber in multiple directions, making the material potentially better suited for labyrinth seals.

DuPont™ offers Vespel® CR 6100 and 6200 that are Teflon® matrix carbon fiber reinforced materials. They have had very good success as wear ring materials in pumps (Bloch, 2004) and will probably one day be used as a labyrinth seal in centrifugal compressors.

There are also literally hundreds of other high performance materials that can be considered for this type of application. For now, the TORLON® and PEEK™ based materials have proven themselves and should be around for a long time.

New Labyrinth Seal Designs

As the CFD analysis with the second case history demonstrated, there are some labyrinth designs that are more effective than others. As CFD analysis becomes more prevalent, the application to labyrinth seal design will increase. One researcher has indeed taken the step of utilizing a custom written CFD code to optimize labyrinth geometry by optimizing the turbulence generation. The results of this work have been verified with laboratory and field-testing. For a given configuration, the optimized seal will leak 1/3 to 1/2 that of a standard labyrinth.

This technology has been licensed and is in the process of being tested for performance. The technology has already been used in the field as impeller eye wear rings in a large double suction pump with extraordinary success. Coupling this seal design with thermoplastics should make for very efficient installations, taking laby seal leakage to new lows. This seal is now marketed as the L^3 (L-cubed) seal to acknowledge that it is a low-leakage-labyrinth.

Dynamic Testing and Analysis

As rotordynamic behavior becomes more understood, the focus can shift to specific components. This means the study of labyrinth seals can be extended to better understand their dynamic role in the turbomachine. Most seal codes utilize control volume approaches with the newer codes breaking the seal flow area into three control volumes for added accuracy. The developers are able to run more accurate CFD analyses to validate their codes.

NOTE

Trade and vendor names were used throughout this tutorial since, as a tutorial, the authors felt it was necessary to refer to the materials as they are normally referred to in industry. Endorsement of any product is not intended.

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