VESPEL®

USING VESPEL BEARINGS

DESIGN AND TECHNICAL DATA

DU PONT®

Putting VESPEL* to work in your system

DuPont VESPEL SP polyimide bearings have been hard at work for more than twenty years, keeping equipment running longer and with less maintenance than conventional bearing materials. VESPEL bearings are the cost-effective choice in thousands of applications, because they're tough and lightweight, and resist wear and creep—even at extremes of temperature. They can outperform metals and other engineering plastics under a wide range of conditions.

The design guide is provided to help you choose the VESPEL bearing that is best suited to your application. Inside you will find:

■ general information about bearing design;

■ a method for determining pressure-velocity (PV) loading in your application;

■ guidelines for selecting the correct SP polyimide for PV loadings found in practice;

■ considerations for use in the design of VESPEL bearings; and

■ a sample bearing design problem.

If you have any questions on bearings that are not answered in this brochure, contact your local VESPEL sales engineer or the sales office nearest you.

* DuPont registered trademark for polyimide parts and machining stock.

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VESPEL Bearings vs. Other Materials

The ability of a bearing to perform in a given application depends, in general, on:

■ the operating environment, including temperature and lubrication

■ load or pressure on the bearing surface

■ sliding velocity of the mating surface relative to the bearing

■ hardness and finish of the mating surface

■ frictional behavior of the bearing material

■ thickness of the bearing material combined with the material's ability to dissipate heat of friction.

VESPEL parts, made from DuPont's SP polyimide resins, perform well with or without lubrication under conditions that destroy most other plastics and cause severe wear in most metals. VESPEL bearings reduce or eliminate problems with abrasion, corrosion, adhesion, fatigue and wear that plague conventional bearing materials, especially when used without lubricants.

VESPEL bearings can accommodate higher pressure-velocity (PV) loading than most high-performance engineering plastics. In addition, VESPEL bearings excel over a wide range of temperatures and stresses because they retain their outstanding creep resistance, abrasion resistance and strength. They have performed successfully in the following adverse environments:

■ air and inert gases at 700°F (371°C)

■ gamma and electron beam radiation

■ high vacuum (10^{-10} torr)

■ hydraulic fluids and jet fuels

■ liquid hydrogen

Unlike ordinary ball, needle and roller bearings, VESPEL* bearings:

- need no external lubrication
- perform at temperatures where lubricants break down
- perform well in dirty environments
- can reduce noise, weight and costs

Compared with bronze, brass and porous metal bearings, VESPEL bearings:

- extend the life of other components by eliminating metal-to-metal wear
- withstand combinations of temperature, prssure and surface velocity beyond the reach of unlubricated metals
- resist creep and poundout
- eliminate problems of lubricant loss in the presence of paper dust or lint

Compared with other polymer bearings, VESPEL bearings:

- perform at temperatures, pressures and surface velocities that other plastics cannot survive
- increase creep and poundout resistance
- machine like brass and hold tighter tolerances

How to Choose a Bearing Material

PV Loading—A Prime Factor in Material Selection

PV is the product of load or pressure (P) and sliding velocity (V). A plastic bearing subjected to increasing PV loading will eventually reach a point of failure known as the PV limit. The failure point is usually manifested by an abrupt increase in the wear rate of the bearing material.

As long as the mechanical strength of the bearing material is not exceeded, the temperature of the bearing surface is generally the most important factor in determining PV limit. Therefore, anything that affects surface temperature—coefficient of friction, thermal conductivity, lubrication, ambient temperature, running clearance, hardness and surface finish of mating materials—will also affect the PV limit of the bearing.
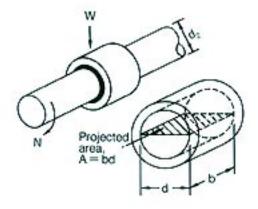
The first step in evaluating a bearing material consists of determining whether the PV limit of that material will be exceeded in your application. It is usually prudent to allow a generous safety margin in determining PV limits, because real operating conditions often are more rigorous than experimental conditions.

Determining Your PV Requirements

1. First determine the static loading per unit area (P) that the bearing must withstand in operation.
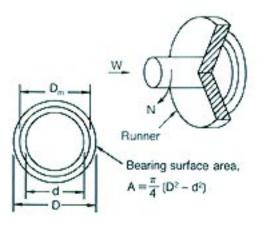
For journal bearing configurations:

$$P = W/(d \times b)$$

P = pressure, psi (kg/cm²)
W = static load, lb (kg)
d = bearing surface ID, in. (cm)
b = bearing length, in. (cm)
N = rotation speed, rpm

For thrust bearing configurations:

$$P = 4W/\pi(D^2 - d^2)$$

P = pressure, psi (kg/cm²)
W = static load, lb (kg)
d = bearing surface ID, in. (cm)
D = bearing surface OD, in. (cm)
N = rotation speed, rpm

For either bearing configuration, pressure (P) should not exceed the values shown here at room temperature:

Allowable Static Bearing Pressure								
Composition	**SP-1**		**SP-21**		**SP-22**		**SP-211**	
Fabrication Process	**Mach.**	**Direct Formed**	**Mach.**	**Direct Formed**	**Mach.**	**Direct Formed**	**Mach.**	**Direct Formed**
P, psi	7,400	4,800	6,600	4,900	6,000	3,700	5,400	4,000
P, kg/cm^2	520	337	464	345	422	260	380	281

2. Next, calculate the velocity (V) of the bearing relative to the mating surface:

	Journal Bearing	**Thrust Bearing**
Continuous Rotation	$V = \pi(DN)$	$V = \pi(D_M N)$
Oscillatory Motion	$V = \pi(dN)(\theta/180)$	$V = \pi(D_M N)(\theta/180)$

where:
N = speed of rotation, rpm or cycles/min
$\mathbf{D_M}$ = $(D + d)/2$, in. (cm)
θ = angle between limits of oscillation, degrees
V = surface velocity, in./min (cm/min)

3. Finally, calculate PV:

PV (psi-ft/min) = **P** (psi) $\times$ **V** (in/min) / 12
or, in metric units:

PV (kg/cm^2-m/sec) = **P** (kg/cm^2) $\times$ **V** (cm/min) / 6000

PV Limits of Unlubricated Bearing Materials

Table 1 shows the maximum PV limits for unlubricated VESPEL parts and several other unlubricated bearing materials under conditions of continuous motion. Properly lubricated VESPEL parts can withstand approximately 1 million psi-ft/min.

TABLE I—PV LIMIT GUIDELINES**					
		lb-ft	**kg-m**	**Maximum Contact Temperature**	
Material	**Filler**	**in^2-min**	**cm^2-sec**	**°F**	**°C**
SP-21	15% Graphite	300,000	107	740	393
SP-22	40% Graphite	300,000	107	740	393
SP-211	15% Graphite 10% PTFE	100,000	36	500	260
PTFE*	Unfilled	1,800	0.64	500	260
PTFE*	15–25% Glass	12,500	4.5	500	260
PTFE*	25% Carbon	20,000	7.1	500	260
PTFE*	60% Bronze	18,500	6.6	500	260
Nylon	Unfilled	4,000	1.4	300	217
Acetal	PTFE	7,500	2.7	250	201
	Unfilled	3,500	1.2		

*At 100 fpm.
** These guideline values are supplied for reference only. PV limits for any material vary with different combinations of pressure and velocity as well as with other test conditions. Consult manufacturer's literature for detailed information.

Designing VESPEL Bearings

Effects of Surface Temperature on Wear Characteristics

PV is a very useful measure in determining the suitability of a material for a bearing application. However, contact pressure and sliding velocity alone do not adequately characterize bearing materials. Temperature, system geometry and mating surface material also play significant roles in wear of bearings.

Of the factors just named, temperature is generally the most important, because it not only affects the coefficient of friction but also determines the usable combinations of pressure and sliding velocity, or PV. Wear characteristics of VESPEL bearings will be moderate even at high PVs if sufficient cooling is provided. Wear can be severe at any PV if the ambient temperature is too high. The wear resistance of a VESPEL bearing operating at a temperature below its limit can be predicted from an experimentally determined *Wear Factor.* The wear factor is derived from an equation relating the volume of material removed by wear in a given time per unit of load and surface velocity.

$$v = KFVT$$

where:

V = wear volume, in^3 (cm^3)

K = wear factor, in^3-min/ft-lb-h (cm^3-min/m-kg-h)

F = supported load, lb (kg)

T = time, h

V = velocity, ft/min (m/sec)

For flat surfaces the equation is modified so that:

$$X = KPVT$$

where:

X = wear depth, in. (cm)

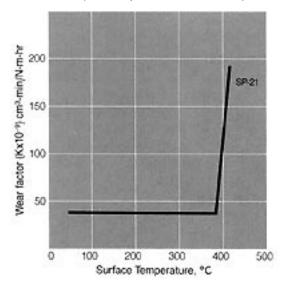**P** = pressure, psi (kg/cm^2)

Wear Transition Temperature

The wear rate of a plastic material operating in air is proportional to the product of pressure and velocity (PV) if the surface temperature does not exceed a critical value called *Wear Transition Temperature.* Above the wear transition temperature, wear increases dramatically. For SP resins, the wear transition temperature is in the range 900° to 1000°F (482° to 538°C) in vacuum or inert gases, and 700° to 750°F (371° to 399°C) in air.

As Figure 1 shows, the wear factor of VESPEL bearings made with SP-21 resin is essentially constant over a wide range of operating conditions, as long as surface temperature does not exceed the wear transition temperature.

FIGURE 1
Wear Factor vs. Surface Temp for SP-21 vs. Carbon Steel Thrust Bearing Tester— No Lubrication PV of 1,000 to 500,000 lb/ $in^2 \times$ ft/min (310-155,000 N/cm$^2 \times$ m/min)
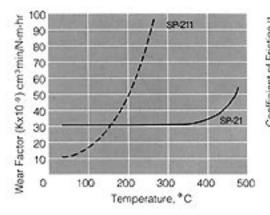
Frictional Behavior

Temperature, pressure and velocity all affect the dynamic coefficient of friction. Typical coefficients of friction for various SP polyimide compositions are shown in Table 2 below.
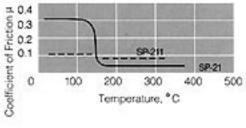
The coefficients of friction for filled SP compositions undergo a transition at about 300°F (149°C), as shown in Figure 2. Below this temperature the frictional behavior is similar to that of 66 nylon, but above 300°F (149°C) the frictional forces drop sharply, and in the range of 400° to 1000°F (204° to 538°C), the friction characteristics of SP compositions remain independent of temperature. The friction transition is not associated with wear transition. The magnitude of the transition, and the wear rate below 300°F (149°C), are greatly reduced in SP-211.

The designer must allow for the higher frictional forces, resulting from two separate phenomena, which may be present during start-up. One is the transfer of a layer of SP polyimide resin/filler composition to the mating surface and the second is the temperature transition for SP polyimide resins. During restart, it may not be necessary under service conditions to break in a new layer, but the temperature effect is reversible and will continue to operate at each restart.

TABLE 2— Typical Coefficients of Friction—Unlubricated Thrust Bearing Test			
MEASUREMENT	**COMPOSITION**		
Conditions British (SI) Units	SP-21 polyimide	SP-22 polyimide	SP-211 polyimide
Static	0.30	0.27	0.20
P = 50 psi (0.34 MPa) V = 500 fpm (2.54 m/s)	0.24	0.20	0.21
P = 100 psi (0.69 MPa) V = 100 fpm (0.51 m/s)	0.30	0.24	0.24
P = 100 psi (0.69 MPa) V = 300 fpm (1.52 m/s)	0.28	0.21	0.20
P = 100 psi (0.69 MPa) V = 1000 fpm (5.08 m/s)	0.12	0.09	0.08

FIGURE 2
Wear Factor and Friction Coefficient for Unlubricated Operation Against Mild Carbon Steel
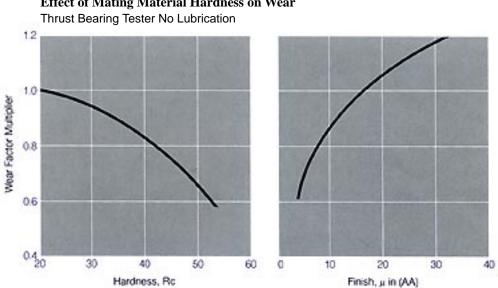

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Mating Material and Surface Finish

The wear performance of VESPEL® parts can be substantially affected by the hardness of the mating material and its surface finish. Unlubricated bearing wear rates can be reduced by increasing the hardness and decreasing the roughness of the mating surface. In general, a ground surface finish on the mating material is preferable to a turned surface. A fine polishing operation is often beneficial. The finishing operation should be in the same direction as the bearing motion relative to the mating surface.

Aluminum and zinc are not good mating surfaces for plastic bearings because the softness of these materials can lead to rapid wear. If used, aluminum should be hardened or, preferably, anodized. Die-cast aluminum with high silica content is very abrasive to VESPEL. Chrome plating is not necessary and may cause greater wear than a polished steel surface. The porosity of the chrome tends to micro-machine the VESPEL surface. In general, the metal surface should be as hard and smooth as is practical.

Plastic is not a good mating material for VESPEL bearings and, if used, should be limited to low PV conditions. The softness of a plastic mating surface can lead to high wear. In addition, since plastics are relatively poor thermal conductors, plastic-to-plastic bearing interfaces run hotter than plastic-to-metal interfaces, so metal plastic bearing systems have higher PV limits than plastic-plastic bearing systems. Figure 3 illustrates the effects of mating material hardness and finish on wear performance.

FIGURE 3
Effect of Mating Material Hardness on Wear
Thrust Bearing Tester No Lubrication

Lubrication and Other Bearing Design Considerations

When determining whether bearings need to be lubricated, the following points should be considered:

■ A one-time lubrication, consisting of an initial greasing or use of dry lubricant, generally reduces break-in wear and improves overall wear resistance.

■ Lubrication of bearings can increase the PV limit by reducing coefficient of friction and helping to remove wear debris. Circulation of the lubricant can further increase the PV limit by cooling the bearing.

■ Lubrication with a chemically compatible fluid to wet VESPEL bearings will reduce both friction and wear rates. The amount of reduction increases with increasing fluid film thickness, which in turn increases with fluid viscosity and surface velocity, and decreases with increasing bearing pressure. Application geometry will also affect the reduction of friction. Even thin film lubricants can reduce dry wear rates by a factor of 10 or more. Thick films, which cause complete separation of the solid mating surfaces, can theoretically reduce wear to negligible proportions.

■ The frictional behavior of a bearing system using thin film lubrication is determined by the properties of the bearing material as well as by the properties of the lubricant. Frictional behavior is determined exclusively by the lubricant properties with thick film lubrication.

■ Unlubricated bearings should have surface grooves to carry wear debris out of the interface. In lubricated systems the grooves can help increase the supply of lubricant. The effect of grooving on bearing pressure should be considered.

■ Because it does not wet SP resin, water is not an effective thin film or boundary lubricant for VESPEL bearings. In fact, water can adversely affect the wear rate of dry VESPEL bearings. However, periodic contamination by casual water should not cause any problems.

■ Purging an unlubricated VESPEL bearing with nitrogen gas can reduce wear rates to less than 20% of the corresponding rate in air. In addition, operation in nitrogen can increase the wear transition temperature by at least 100°F (56°C) above the value in air.

■ For applications in dirty environments, sealing or purging should be considered for prevention of bearing surface contamination.
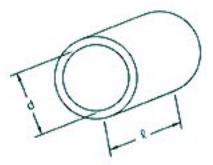
Proportions

Journal Bearings: For optimum performance of VESPEL journal bearing, ℓ/d (length/diameter) ratios in the order of 1/2 to 3/2 are suggested. If a long bearing is required, consider using two bearings with a gap between them. Smaller values of ℓ/d will result in:

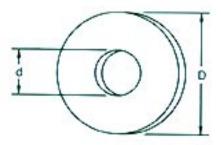■ more efficient debris removal

■ less sensitivity to shaft deflection and misalignment

■ better heat dissipation

■ cost advantages due to lower fabrication costs

Thrust Bearings: For optimum performance of VESPEL thrust bearings, it is best not to exceed a ratio of outside to inside diameter (D/d) of 2. Ratios greater than 2 can cause overheating at the outside edge, and problems may arise from lack of flatness and from trapped wear debris.

Running Clearances for Journal Bearings

Although VESPEL bearings have much lower coefficients of thermal expansion than most plastics, minimal running clearances are required. Normal operating clearances for VESPEL journal bearings are from 0.3% to 0.5% of shaft diameter, depending on the application. In general, heavier loads require larger clearances. Closer running clearances can be engineered by slotting the bearing to allow for circumferential thermal expansion.

Use the following formula to determine VESPEL bearing design inside diameter adjusted for thermal expansion of the bearing system:

Bearing design ID = shaft diameter at room temperature + change in shaft diameter due to temperature change + operating clearance of shaft + change in the bearing wall thickness due to temperature – change in housing diameter due to temperature.

$$ID = D(1 + \alpha S \Delta T_1 + C) + 2t\,\alpha Sp\Delta T_2 - d\alpha B\Delta T_3$$

where:

d	=	housing diameter
D	=	shaft diameter at ambient temperature
C	=	shaft operating clearance, percent of shaft diameter
α**S**	=	coefficient of expansion of shaft material
α**Sp**	=	coefficient of thermal expansion of VESPEL bearing
t	=	VESPEL bearing wall thickness
ΔT_1	=	temperature rise for the shaft
ΔT_2	=	temperature rise for the bearing
ΔT_3	=	temperature rise for the housing
α**B**	=	coefficient of expansion of bore material

Running clearances for VESPEL bearings usually do not have to be adjusted for moisture, because SP polyimides absorb very little moisture. See the brochure "Properties of VESPEL Parts and Shapes", for moisture absorption curves.

TABLE 3—Coefficient of Thermal Expansion (α)								
Composition	**SP-1**		**SP-21**		**SP-22**		**SP-211**	
Fabrication Process	**Mach.**	**Direct Formed**	**Mach.**	**Direct Formed**	**Mach.**	**Direct Formed**	**Mach.**	**Direct Formed**
α, 10^{-6} in/in/°F	30	28	27	23	21	15	30	23
α, 10^{-6} cm/cm/°C	54	50	49	41	38	27	54	41

Wall Thickness for Journal Bearings

VESPEL journal bearing walls should be as thin as possible, because thin walls:

■ improve dissipation of frictional heat

■ reduce running clearance variations resulting from thermal- and moisture-related dimensional changes

■ reduce distortion under high loading

For most applications, the typical wall thickness for VESPEL bearings ranges from 0.06 in. to 0.125 in. (1.5 to 3.2 mm).

Installation of Journal Bearings

VESPEL journal bearings can be installed either mechanically or with adhesive.

To press fit VESPEL bearings into metal, the suggested practice is to use a low-interference fit. After it is pressed into place, the bore of the bearing will be reduced by 90% of the calculated diametral interference, which will result in a small compressive load in the bearing wall. A typical interference fit is 0.5%, but press-fit interference should be adjusted to the needs of the application.

VESPEL parts can be used with most commercial adhesives. The "VESPEL Adhesives Bulletin" discusses selection of adhesives, surface preparation and other considerations. With any adhesive, it is important to follow the manufacturer's recommendations for best results.

Sample Design Problem

VESPEL bearings are being considered for a blender dryer with the following requirements:

■ *Two bearings on a 1.5 in. (38 mm) shaft must support 4000 lb (1814 kg), at temperatures ranging from 70°F to 525°F (21°C to 274°C).*

■ *Maximum bearing length is 1.5 in. (38 mm) and maximum allowable running clearance hot or cold is 0.015 in. (0.38 mm).*

■ *To prevent product contamination, the bearings cannot be lubricated and they must operate without service 40 hours per week for 3 years.*

■ *The blender-dryer shaft rotates intermittently, 5% on and 95% off, at 20 rpm.*

Will VESPEL bearings meet these requirements? Refer to the table "PV LIMIT GUIDELINES", under "PV Limits of Unlubricated Materials" for temperature and PV limits.

Solution (British Units)

1. Check temperature limit.

Limiting surface temperature of SP-21 polyimide in air is 740°F, so unless PV is very high, the surface temperature should rise less than the 215°F difference between 525°F and 740°F.

2. Check PV.

Calculate bearing pressure:

$$P = \frac{F}{LD} = \frac{2000 \text{ lb (per bearing)}}{1.5 \text{ in} \times 1.5 \text{ in}}$$

$$= 890 \text{ psi}$$

Calculate shaft Speed:

$$V = \pi DN = \frac{\pi \times 1.5 \text{ in} \times 20 \text{ rpm}}{12 \text{ in/ft}}$$

$$= 7.9 \text{ fpm}$$

Calculate PV:

$$PV = 800 \text{ psi} \times 7.9 \text{ fpm}$$
$$= 7040 \text{ psi-fpm}$$

At this low PV, SP-21 polyimide will operate in its mild wear regime, so PV will not limit, especially considering the intermittent operation.

3. Check wear resistance.

Calculate running time:

$$T = .05 \times \frac{40 \text{ hrs}}{\text{wk}} \times \frac{52 \text{ wks}}{\text{yr}} \times 3 \text{ yrs}$$

$$= 312 \text{ hours}$$

Radial wear = **wear factor** × **PV** × **running time** =

$$33 \times 10^{-10} \frac{\text{in min}}{\text{ft lb hr}} \times$$

$$7040 \text{ psi-fpm} \times 312 \text{ hrs} = 0.0073 \text{ in}$$

This wear is less than the maximum allowable operating clearance. If the difference is enough to accommodate thermal expansion, then VESPEL Parts will meet the requirements.

4. Design-Determine Room Temperature Clearance.

At this point, experience and judgment play a big role, and one can only approximate a solution. Experience dictates that the shaft, bearing surface, bearing OD and housing will all reach different operating temperatures.

Assume that:

— the contact surface reaches 100°F higher than the dryer temperature, but:

— the bearing body average temperature is only 50°F higher than the dryer, while:

— the housing remains at room temperature and restrains the bearing securely, so:

— the bearing will expand inward when the temperature rises, and

— the shaft will expand outward.

With these assumptions, initial room temperature clearance, C_4, can be determined with the following equation:

$$C_4 = D(\alpha s \Delta T_1 + C) + 2t\alpha s\rho \Delta T_2$$

D = shaft diameter

αs = coefficient of thermal expansion for shaft material

= 6×10^{-6} in/in/°F

C = operating clearance, usually 0.001 in/in

t = bearing wall thickness

$\alpha s\rho$ = coefficient of thermal expansion for bearing = 24×10^{-6} in/in/°F

ΔT_1 = temperature rise for shaft

ΔT_2 = temperature rise for bearing

If one picks a wall thickness of $1/16$ in

$C4 = 1.5[(6 \times 10^{-6})(625 - 70) + .001] + (2 \times .0625 \times 24 \times 10^{-4})(575 - 70)$

$$= 1.5(.0043) + .0015$$

$$= .00645 + .0015$$

$$= .008 \text{ in}$$

5. Check maximum clearance.

Initial clearance plus wear after 3 years will then be 0.008 + 0.007 = 0.015 in which just meets the stated requirements.

Thus, VESPEL bearings do meet the requirements to operate without lubrication in this elevated temperature situation.

If you encounter expansion problems, slot the bearing.

Solution (SI Units)

1. Check temperature limit.

Limiting surface temperature of SP-21 polyimide in air is 666°K, so unless PV is very high, the surface temperature should rise less than the 119°K difference between 547°K and 666°K.

2. Check PV.

Calculate bearing pressure:

$$P = \frac{F}{LD} = \frac{8900N \text{ (per bearing)}}{.038 \text{ m} \times .038 \text{ m}}$$

$$= 6160 \text{ kPa}$$

Calculate shaft Speed:

$$V = \pi DN = \pi \times .038 \text{ m} \times 20 \text{ rpm}$$

$$= 2.39 \text{ m/min}$$

Calculate PV:

$$PV = 6136 \text{ kPa} \times 2.41 \text{ m/min}$$

$$= 14{,}720 \text{ kPa-m/min}$$

At this low PV, SP-21 polyimide will operate in its mild wear regime, so PV will not limit, especially considering the intermittent operation.

3. Check wear resistance.

Calculate running time:

$$T = .05 \times \frac{40 \text{ hrs}}{\text{wk}} \times \frac{52 \text{ wks}}{\text{yr}} \times 3 \text{ yrs}$$

$$= 312 \text{ hours}$$

Radial wear = **wear factor** $\times$ **PV** $\times$ **running time** =

$$40 \times 10^{-9} \frac{\text{cm}^3\text{min}}{\text{m N hr}} \times$$

$$14{,}720 \frac{\text{kPa m}}{\text{min}} \times$$

$$312 \text{ hrs} \times 10^{-4} \frac{\text{m}^2}{\text{cm}^2}$$

$$= .0184 \text{ cm}$$

This wear is less than the maximum allowable operating clearance. If the difference is enough to accommodate thermal expansion, then VESPEL Parts will meet the requirements.

4. Design-Determine Room Temperature Clearance.

At this point, experience and judgment play a big role, and one can only approximate a solution. Experience dictates that the shaft, bearing surface, bearing OD and housing will all reach different operating temperatures.

Assume that:

— the contact surface reaches 56°K higher than the dryer temperature, but:

— the bearing body average temperature is only 28°K higher than the dryer, while:

— the housing remains at room temperature and restrains the bearing securely, so:

— the bearing will expand inward when the temperature rises, and

— the shaft will expand outward.

With these assumptions, initial room temperature clearance, C_d, can be determined with the following equation:

$$C_d = D\,(\alpha s\Delta T_1 + C) + 2t\alpha s\rho\Delta T_2$$

D = shaft diameter

αs = coefficient of thermal expansion for shaft material

= 10.8×10^{-6} m/m/°K

C = operating clearance, usually .001 cm/cm

t = bearing wall thickness

$\alpha s\rho$ = coefficient of thermal expansion for bearing = 43×10^{-6} m/m/°K

ΔT_1 = temperature rise for shaft

ΔT_2 = temperature rise for bearing

If one picks a wall thickness of .159 cm:

$$C_d = 3.8[(10.8 \times 10^{-6})(602-294) + 0.01] + (2 \times .159 \times 43 \times 10^{-4}) \times (575 - 294)$$

$$= 3.8\,(.0043) + .0038$$

$$= .0163 + .0038$$

$$= .020 \text{ cm}$$

5. Check maximum clearance.

Initial clearance plus wear after 3 years will then be 0.20 + 0.18 = 0.38 mm which just meets the stated requirements.

Thus, VESPEL bearings do meet the requirements to operate without lubrication in this elevated temperature situation.

If you encounter expansion problems, slot the bearing.

More information on the benefits and properties
of VESPEL® parts is available in these brochures:

"Introduction to VESPEL Parts"
(E-61486)

"Summary of Typical Properties"
(E-61477)

VESPEL Shapes: Machining Stock of SP Polyimide"
(E-61482)

"Using VESPEL Seal Rings"
Design Considerations and Technical Data
(E-73911)

"VESPEL and Radiation. A Guide for Users"
(E-73910)

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