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Design Considerations for Bearingless Rotary Pumps

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Abstract: The designs of rotary blood pumps have shown substantial technical progress over recent years, especially contact bearing designs. However, the concern for potential thromboembolism remains and can only be eliminated by the use of bearingless pumps. Bearingless designs can be achieved through the application of magnetic, hydrodynamic, and hydrostatic forces or a proper combination of these forces. Although a purely magnetically suspended, actively controlled system can be designed, judicious use of hydraulic forces can allow simplification of device configuration and control. In this study, bearingless designs were evaluated for both axial and centrifugal pump configurations. Trade-offs between shear rates and bearing leak rates were considered based upon constraints imposed by hemolysis and residence time. These principles were used for determining the design feasibility of a rotary pump using combined magnetic and hydraulic stabilizing forces. **Key Words:** Rotary pump—Blood bearing—Magnetic bearing—Hydrostatic bearing—Hydrodynamic bearing.

Rotary blood pumps generally have a size advantage over standard pulsatile blood pumps. In many designs, the small sizes of the rotary pumps make these pumps less intrusive for implantable cardiac support applications. However, this size advantage is obtained at the expense of creating a need for contact bearings to support the rotors that impart momentum to the blood. Such bearings remain a potential source for thromboembolic complications. For this reason, many investigators (1–3), including our group, have explored the use of magnetic suspension of the rotor to eliminate contact between the moving and stationary parts of a pump. Alternatively, a hydrodynamic bearing is being evaluated as a potential blood bearing (4). We have further demonstrated the use of a combination of hydrostatic and magnetic forces for achieving contact-free rotor stability (1). This paper considers the characteristics of these types of noncontact bearings, hydrodynamic, hydrostatic, and magnetic suspension and the adequacy of their use in rotor suspension for the 2 most common types of rotary pumps, the axial and the centrifugal pump.

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PUMP PARAMETERS

There are 2 primary types of loads in a rotary pump, radial and axial. The radial load is perpendicular to the rotational axis of the rotor for the axial flow pump and is ideally merely the weight of the rotor, approximately 10–20 g for a typical blood pump. The radial load is somewhat higher in the centrifugal pump because of the asymmetry introduced by the tangential outflow. However, additional radial loads may be present because of imbalances in the rotor drive magnets, and these loads can be as much as 100–200 g. The axial, or thrust load, can range up to as much as 1,000 g as a result of the pressure increase across the pump. Axial flow pumps generally have a higher thrust load than centrifugal pumps. Thus, the range of loads encountered by the rotors which the bearings have to support is 100–1,000 g.

The fluid shear in the bearing gaps should not be excessive so that cellular damage is not induced, and the leakage flow through the bearing should be high enough to insure good wash out to avoid thrombosis. It is well established that the shear rate acceptable for cellular preservation is strongly dependent upon the duration to which cells are exposed to shear. Examples of acceptable shear rate (SR) and exposure time (ET) combinations are respectively, $SR = 4 \times 10^4\ s^{-1}$ at $ET = 10^2$ s, $SR = 10^5\ s^{-1}$ at $ET = 10^{-2}$

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s, and $SR = 10^6\ s^{-1}$ at $ET = 10^{-4}$ s (5). Most notably, there is a strong dependence of the acceptable shear rate upon the exposure time.

In the design and choice of noncontact bearings, the bearing dimensions, rotational speeds, and loads to be supported all affect the desirable operating gap width of the bearings. This, in turn, influences leakage flow and the shear rate experienced by blood cells. The physics of the hydrodynamic, hydrostatic, and magnetic bearings will be described and characteristics of each type of bearing compared. The design considerations indicate that a combination of permanent magnets and hydrostatic forces can provide a stable bearing system that has acceptable shear rates and good wash out of the bearing surfaces. Other combinations, such as hydrodynamic and permanent magnets, may also be feasible.

HYDRODYNAMIC BEARING

The basic principle of operation of a hydrodynamic bearing is illustrated in Fig. 1 where a moving surface A is shown with first a decreasing gap relative to a stationary surface B along the x-axis, followed by an increasing gap as A moves or rotates. The fluid trapped in the gap experiences an increasing shear stress perpendicular to the fluid motion (u). This leads to a positive pressure gradient in the direction of motion. This situation is reversed as the gap size increases. The pressure distribution is illustrated in the accompanying diagram with the highest pressure developing in the narrowest gap region. This principle of operation for the hydrodynamic bearing is illustrated in both the journal (radial) bearing and the thrust (axial) bearing shown in Fig. 2. Using standard equations for hydrodynamic bearings (6), forces supportable by the bearings can be estimated. Figures 3 and 4 show the load (force) versus the mean bearing gap (c) for a journal and a thrust bearing, respectively. These curves are calculated for the specific bearing dimensions shown in the figure where r is the radius of the bearing. A typical rotational pump speed of 6,000 rpm is used. For the journal bearing, the length (ℓ) of the bearing is also shown. The parameters are representative of a rotary pump with bearing sizes that are not excessive. A family of curves exists for the journal bearing based on the eccentricity (ϵ) of the rotor, defined as the ratio of the offset of the rotor center referenced

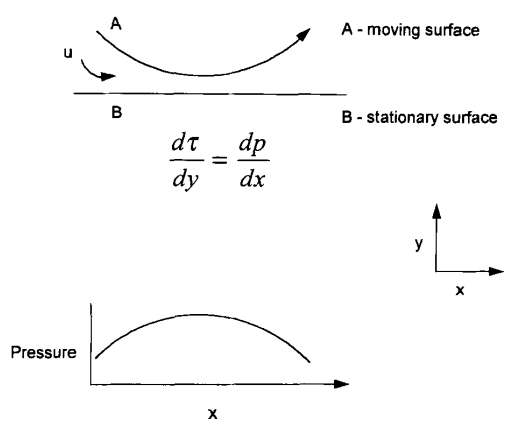

FIG. 1. The shear generated pressure rise as a result of the varying gap between bearing surfaces is shown.

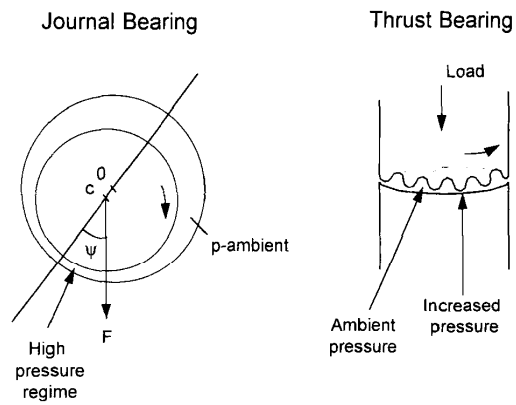

FIG. 2. The illustrations are of journal and thrust bearings. The off-center rotating shaft generates a lift for the journal bearing as a result of the changing shear gradient. The undulating thrust bearing surface results in high pressure in the narrow gap region of the bearing.

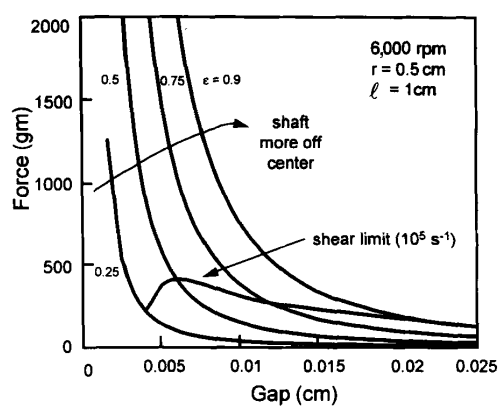

FIG. 3. The graph shows load capacity (force) for a journal bearing as a function of the mean gap between the rotating shaft and the stationary surface. The family of curves is for the degree of off-center operation of the rotating shaft denoted by the parameter ϵ. A shear rate limited curve is shown.

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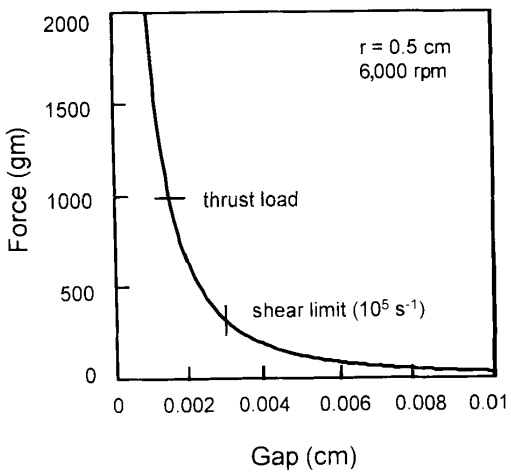

FIG. 4. The graph shows load capacity versus gap for a thrust bearing with a gap ratio (wide to narrow) of 2 and a gap (narrow) to radius ratio of 0.006.

to the stationary center divided by the mean gap between the two parts. The range for ϵ is from 0 (when the centers are aligned) to 1 (when the rotor touches the stationary wall). At $\epsilon = 0$, the bearing is not supporting a load. At $\epsilon = 0.9$, the minimal operating gap is only 10% of the mean gap. For example, with a mean gap of 0.01 cm and an offset of 0.005 cm, the supportable weight is ~150 g. At an offset of 0.009 cm, this weight increases to nearly 750 g, a five-fold increase. The sustainable force (F) scales as

$$F \propto \frac{\mu \Omega r^3 l}{c^2} \tag{1}$$

where μ and Ω are respectively the viscosity of blood and the angular frequency of the bearing. The shear rate SR at the minimum gap is

$$SR = \frac{\Omega r}{(1 - \varepsilon)c} \tag{2}$$

The locus of the constant shear rate of 10^5 s^{-1} is shown in Fig. 3. For a journal bearing, a maximum supportable weight occurs near an offset of 0.5 as illustrated in the figure. Under the constraint of a maximum shear rate, r/c and Ω are kept constant, and the weight scales linearly with the radius r. For a shear rate limit of 10^5 s^{-1}, 350 g can be supported by a bearing of 0.5 cm radius, 1 cm length, and 0.006 cm gap rotating at 6,000 rpm. If the bearing radius were increased to 2 cm and the gap increased to 0.024 cm, the supportable weight would increase by a factor of 4 to ~1400 g. Whether such large bearing surfaces can be practical depends upon the residence time of blood in the gap region. The residence time can be readily estimated from the ratio of the volume of the bearing in the annular region and the leakage flow rate through the bearing (7). For the radial bearing, this time is ~50 ms for a pressure rise across the pump of 150 mm Hg and the parameters chosen.

Similar considerations, illustrated in Fig. 4, show that a load of 350 g can be supported by a thrust bearing of 0.5 cm radius, a 0.003 cm gap at the bearing perimeter rotating at 6,000 rpm. To maintain constant shear, the bearing gap at any radius must be made proportional to the radius. The calculation was performed for a wide to narrow gap ratio of 2 (Fig. 2). For a thrust bearing, it is not likely that the bearing dimension could be increased without impeding flow. Table 1 summarizes 2 sets of design and operating parameters for the radial hydrodynamic bearing and a set for the thrust bearing. For the operating parameters given in Table 1, the estimated heat loss resulting from shear (power loss = $\mu \cdot SR \cdot \text{area} \cdot u$) is between 0.1 and 0.2 W for the smaller bearing, a negligible quantity for small bearings. For larger radial bearings, the power dissipation can be substantial. The residence time for blood in the thrust bearing region cannot be easily estimated because it is design specific. The high shear rate for the thrust bearing would require a short residence time of less than 10 ms. In addition, the operating bearing gap is not much larger than cellular dimensions, ~10^{-3} cm for red cells. Such small gaps invite blood activation.

HYDROSTATIC BEARING

A hydrostatic bearing utilizes the inherent pressure distribution around a rotor to provide the restoring force necessary to stabilize the rotor. Figure 5 shows the principle of operation of such a bearing. A configuration similar to that of a centrifugal pump is used to illustrate this principle. Commonly, the inflow near the center of the rotor is at low pressure, and the outflow near the perimeter is at high pres-

TABLE 1. *Hydrodynamic bearing parameters*

Parameters	Radial	Radial	Thrust
Radius (cm)	0.5	2	0.5
Length (ℓ) (cm)	1	2	–
Speed (rpm)	6000	3000	6000
Gap (cm)	0.006	0.024	0.002
Shear rate (s^{-1})	8×10^4	4×10^4	1.5×10^5
Bearing area (cm^2)	3	60	0.8
Load capacity (g)	350	1400	1000
Heat generated (W)	0.2	1.6	0.1
Residence time (ms)	50	12.0	–

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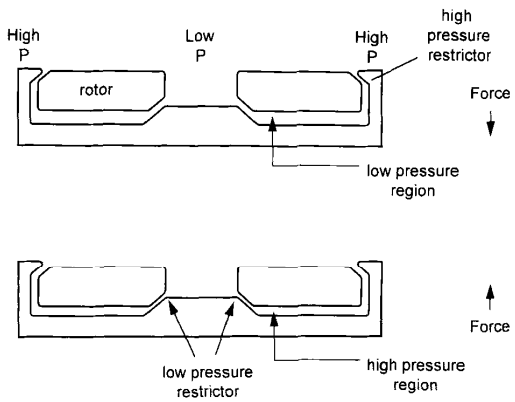

FIG. 5. The illustrations of a hydrostatic bearing show the 2 flow restrictors that passively regulate the pressure in the bypass flow region to generate a restoring force for the rotor.

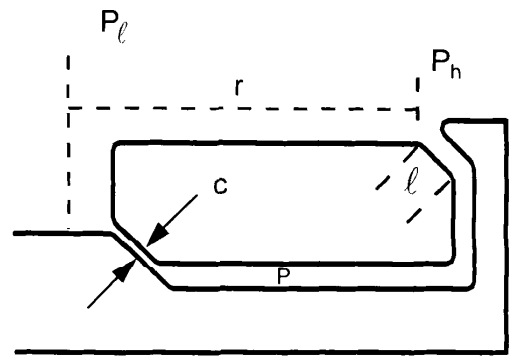

FIG. 6. The illustration is of the flow restrictor where c is the restrictor gap, ℓ is the restrictor length, r is the radius of the rotor, and P_h, P_ℓ, P are respectively the pressures of the high, low, and bypass flow regions.

sure. An annular flow return path around the rotor is used to provide washout of the rotor surfaces, diverting a fraction of the main blood flow. Two annular restrictors are designed into this return flow path, one near the high pressure side and the other near the low pressure side, respectively. Depending upon the direction of the forces exerted on the rotor, either the high or the low pressure restrictor provides the primary flow resistance, resulting in respectively either a low or a high pressure in the return path region with restoring forces to stabilize the rotor axially.

The restoring force can be calculated by recognizing that to order first the return leakage flow, Q, governs the pressure in the return flow region and is calculated as

$$Q = \frac{P_h - P}{R_h} = \frac{P - P_l}{R_l}, \tag{3}$$

where the subscripts h and ℓ represent the high and low pressure regions, R is the resistance of the restrictor, and P is the pressure in the return flow region. Pertinent parameters are defined and shown in Fig. 6. The resistance is calculated as (7)

$$R = \frac{6\mu l}{\pi r c^3} \tag{4}$$

The size of the restrictor gap is the governing parameter varying inversely to the cube power in determining the flow resistance. The resulting pressure difference across the rotor surfaces can be estimated by

$$\Delta P = P - \frac{1}{2}(P_h + P_l) \tag{5}$$

Figure 7 shows the axial restoring force on the rotor as a function of the deviation of the gap from the equilibrium position. The ratio ℓ/r is chosen to be the same for the high and low pressure restrictors. With an equilibrium gap of 0.014 cm, restoring forces of up to 1,000 g can be readily achievable. Table 2 summarizes the important parameters relevant to a hydrostatic bearing. For the hydrostatic bearing, shear rates, blood residence time in the bearing region, and the leakage flow required to support the bearing can be readily calculated and are listed in the table. Residence times are sufficiently short (~4 ms) in the gap region so that cell damage is not an issue. Heat generation due to viscous loss is manageable (~1 W) for the operating parameters shown in Table 2. The hydrostatic bearing approach is ideally suited as an axial balancing mechanism for centrifugal pumps.

PERMANENT MAGNET BEARING

In addition to hydrodynamic and hydrostatic bearings that use fluid forces to achieve the stability of

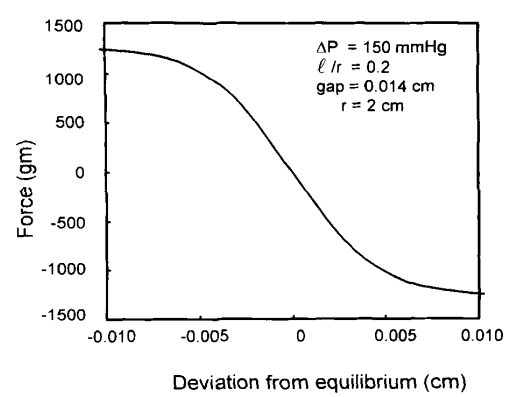

FIG. 7. The graph shows load capacity versus the deviation from equilibrium position for the hydrostatic bearing.

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TABLE 2. *Hydrostatic bearing parameters*

Radius	2 cm
Speed (rpm)	6,000
Load	1,000 g
Gap	1×10^{-2} cm
Shear rate	6×10^4 s^{-1}
Residence time	4 ms
Bearing area	0.6 cm^2
Leakage rate	1 L/min
Heat generated	1.1 W

rotors, magnets in the appropriate configuration can be used to maintain balance. Although there are no fundamental limits on the use of fluid forces to maintain balance in all degrees of freedom, there is a limit on the use of permanent magnets to achieve full stability. This fundamental limit is known as the Earnshaw theorem (8). With the use of permanent magnets, there will always be one unstable degree of freedom that must be stabilized either by actively controlling electromagnets to achieve balanced operation or by using other restoring forces, such as fluid forces, to maintain rotor equilibrium.

Within a limited range of excursion around an equilibrium position, the restoring forces can be expressed as

$$F_r = K_r r \tag{6}$$

and

$$F_z = K_z z \tag{7}$$

where r and z are respectively the radial and axial coordinates of a rotor, and K_r and K_z are the respective force constants. For an axially stable permanent magnet bearing, K_z is less than zero, such that the force vector in the z direction opposes the displacement of z. However, by the Earnshaw theorem, the radial force constant would relate to the axial force constant such that

$$K_r > -\frac{1}{2} K_z \tag{8}$$

resulting in an instability in the radial direction. For a radial bearing, K_r would be negative,

$$K_z > -2K_r \tag{9}$$

leading to an instability in the axial direction. Angular stability should be considered in any permanent magnet design; however, it is not discussed in this paper because it will not further clarify the design choices for the suspension methods. Refer to Delamare et al. (9) for details.

Figure 8 shows 2 of many magnet configurations for radially stable and axially stable permanent magnet arrangements (10). Each stability mode is illustrated by an example derived from a repulsive and an attractive layout. The arrows shown in the figure indicate the magnet alignments. For example, the repulsive radially stable but axially unstable case is shown with the magnets of the inner and outer concentric rings aligned in the same direction. Because of the radial repulsion, the inner ring seeks to be coaxial with the outer ring. However, the repulsive force also tends to force the 2 rings from lying in the same plane, resulting in the axial instability. The magnitudes of such instability can be readily estimated from (10)

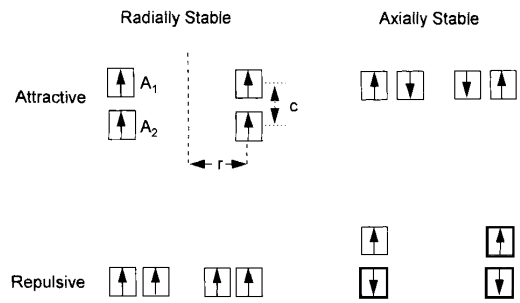

FIG. 8. Shown are examples of a radially stable and axially unstable permanent magnet configuration and an axially stable and radially unstable configuration in both the attractive and repulsive modes.

$$F_z = -2r \frac{M_1 M_2}{\mu_0 c^3} A_1 A_2, \tag{10}$$

where c is the size of the gap between magnets, r is the radius of the magnets from the axis, μ_0 is the permeability of free space, A_1 and A_2 are the cross sections of the ring magnets, and M_1 and M_2 are the magnetizations of the permanent magnet. The equation is valid as long as the magnet dimension $(A)^{1/2}$ is not larger than c, the gap size. Figure 9 shows the calculated forces versus the gap size between the ring magnets using samarium-cobalt magnets (M = 0.85 tesla) and magnet dimensions of 0.2 cm × 0.2 cm and a 2 cm radius. The distinguishing characteristic of a magnetic bearing is the large force generated at the magnet separations (of the order of mm), which are substantially larger than those (of the order of 1/10 mm) of fluid bearings. There are no shear rate or residence time limitations associated with magnetic bearings.

BEARING RESTORING FORCES

The previous analyses provide a number of useful parameters that can aid in the choice of noncontact

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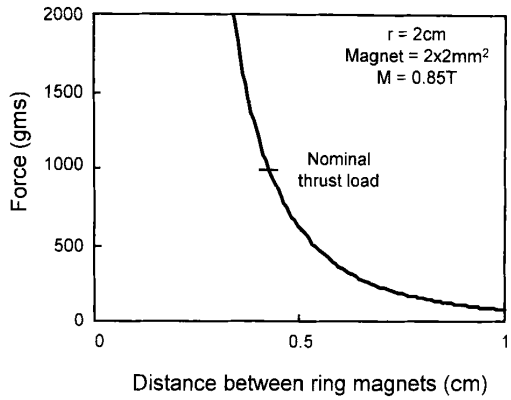

FIG. 9. The graph shows load capacity versus distance between ring magnets for the magnet parameters shown.

bearings for rotary blood pump application. Key among those parameters are the force constants associated with these bearings listed in Table 3. It is immediately obvious that fluid bearings have stiffer force constants that are approximately one order of magnitude larger than force constants derivable from permanent magnet bearings. It is, therefore, practical to select fluid bearings as a means to constrain the instability associated with the one degree of freedom in permanent magnet bearing designs.

BEARING CHOICES

Although active electromagnets have been used for noncontact blood pump bearings (1,2,3), the extra complexity associated with position sensing of the rotor and the feedback control necessary to balance a rotor inevitably result in a pump system that is large for a rotary pump. To reap the benefits of the potential size advantage of a rotary pump and yet avoid contact in the bearing system, a judicious choice of bearings can lead to optimal pump configurations. Based on the analyses discussed in this paper, the ideal combination of bearings for a centrifugal pump configuration can be achieved by the use of permanent magnets for radial and angular stability. The axial instability introduced by the permanent magnets can be offset by the hydrostatic bearing described previously. Shear rates and blood residence times in the bearing region are compatible with minimal blood damage.

TABLE 3. *Bearing force constants in g/cm (N/m)*

Bearing	Axial	Radial
Hydrodynamic	1×10^5	1×10^5
Hydrostatic	2×10^5	–
Permanent magnet	2×10^4	1×10^4

For an axial flow configuration, a potential combination would be a permanent magnet set for axial stability while the resultant radial and angular instability can be offset by the use of hydrodynamic bearings. Such a hydrodynamic bearing would have a large radius (~2 cm) and long length (2 cm), each spanning portions of the inflow and the outflow to insure that the pressure generated by the pump, as well as the gap dimensions (~1/4 mm), are large enough to insure adequate leakage flow (0.5 L/min) through the bearing. The estimated residence time of the blood leakage flow through the bearing is ~25 ms. Heat dissipation of a few watts can be a problem (Table 1). Whether the bearing is nondamaging to blood depends upon the required operating rpm. At 6,000 rpm, at which the shear rate approaches 10^5 s^{-1}, the bearing could be hemolyzing blood during the 12 ms traverse time of the blood. However, at 3,000 rpm or lower, the bearing should be innocuous to blood because of the rapid transition from an unacceptable to an acceptable shear rate below 10^5 s^{-1}, permitting substantially longer residence time in the bearing region with minimal cellular alteration.

The design trade-offs presented in this paper suggest that the judicious choice of bearing types, especially combinations of fluid and permanent magnets, can result in compact rotary pumps with noncontact bearings as the lubrication mechanism. A more comprehensive design trade-off analysis is in progress to define specifically the bearing parameters necessary for the physiologic pressure and flow regime.

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