The concept of augmenting human circulation with a ventricular assist device (VAD) has been in existence for decades. Two types of VAD devices are currently under development: pulsatile and continuous flow.

The human heart produces blood flow with a pulsatile action. The first notable application of a pulsatile VAD with one long-term survival was by DeBakey (1) in the 1960s. Olsen et al. (2) discussed advantages to be gained with total artificial heart (TAH) pulsatile devices. However, long-term reliability problems with mechanical pulsatile devices have been encountered. They require a flexing polymeric diaphragm and two valves for each ventricle.

The mechanical reliability problems with pulsatile pumps has led to the concept of a continuous (non-pulsatile) flow pump. Akamatsu et al. (3), (4) have described a centrifugal impeller partially suspended magnetically. It is not fully magnetically suspended because the pump impeller is magnetically coupled to a brushless DC motor supported on rolling element bearings. O’Hara and Nose (5) describe a centrifugal pump with blood bearings, referred to as the Baylor-Gyro pump in medical literature. To date, there has not been a fully magnetically supported VAD reported in the literature.

There are several potentially viable bearing choices for blood pumps: ball bearings, fluid film bearings, and non-contacting magnetic bearings. Ball bearings cannot be lubricated directly with blood because extensive damage to the red blood cells, called hemolysis, is caused when the blood flows through the ball-race clearances. Ball bearings can only be used with a purged seal system (6). Seal failures resulted in many early pump failures. Additionally, seals accumulate protein buildup and/or thrombosis (clotting). Blood...
lubricated bearings tend to be subject to high shear due to the high rotational speeds and small clearances. The most recent design of the centrifugal pump indicated above has the rotor/impeller supported on three ball tips that ride on the housing, as reported by O'Hara and Nose (5).

Magnetic bearings offer the advantages of non-contact suspension, thus, there is no wear over time and no friction, and large clearances, as described by Allaire et al. (7). The potential for use in VAD systems was first reported by Bramm et al. (8). A prototype device was suspended magnetically. Olsen and Bramm (9) and Olsen and Wampler (10) reported on further work related to the device.

There is widespread industrial experience with low-viscosity fluid film bearings in pumps which have high bearing failure rates (11), (12). Magnetic bearings are now under development as replacements for the conventional bearings to remove these failures and should offer the same improvement for VAD bearings (13), (14). The large clearance allows for high circulation rates around the rotating impeller, thus reducing hemolysis and thrombosis.

The purpose of this paper is to describe the development of a prototype VAD consisting of a centrifugal pump supported in a five-axis active magnetic bearing system (13). The primary emphasis of the paper is on the active magnetic bearing system rather than on pump or motor performance.

VENTRICULAR ASSIST DEVICE

The expected use of the Continuous Flow Ventricular Assist Device (CFVAD) system in an adult human is illustrated in Fig. 1. The pump itself will be implanted in the abdominal cavity but the batteries and patient controller will be located external to the body in a vest/belt system. The VAD is a blood pump which will directly pump blood to assist the left ventricle. It should also be configured to assist the right ventricle. It must be designed for long-term life, over five years, at high reliability of over 90 percent. It must be compatible with the anatomical space available and have hemocompatibility.

The authors have developed a prototype continuous flow blood pump prototype. It has been constructed and preliminary performance tests have been conducted on a simulated human cardiac test loop. While the current prototype is too large for implantation, much is being learned about this type of CFVAD. The inlet to the CFVAD2 is a 14-mm ID cannula designed to simulate collecting blood from the apex of the left ventricle and returning it through the outlet near the root of the aortic arch. The cardiac output is in the range of 6-10 liters/min and the differential pressure is approximately 100 mm Hg. The CFVAD prototype has adjusted its cardiac output by adjusting the motor speed, providing a variable flow rate. When implanted, a future prototype will have a pressure sensor to provide the feedback signal to the electronic output control system. The current CFVAD prototype consists of five subsystems: pump, magnetic bearings, motor, electronic control, and bio-compatible contacting surfaces.

PROTOTYPE CFVAD

This prototype model was intended to demonstrate the concept of a small pump fully suspended in a five-axis magnetic bearing system. The CFVAD2 was 175 mm in diameter and 117 mm in axial length. The rotor had a length of 105 mm and weighed 3.65 N. The larger-diameter impeller section had an axial length of 36 mm.

A schematic diagram of the prototype is shown in Fig. 2. Fluid flowed along the axial inlet of the centrifugal impeller and then radially outward into a vaneless diffuser. The motor and magnetic bearings were integrated with the pump im-
peller and housing as shown. The motor stator and inlet radial magnetic bearing stator was located along the inlet section. The outlet radial magnetic bearing stator and thrust bearing stator were located behind the impeller. The motor rotor, inlet radial magnetic bearing rotor, outlet radial magnetic bearing rotor, and magnetic thrust bearing rotor were all located on the pump rigid rotor. A more-detailed view of the prototype is given in Fig. 3.

**THRUST BEARINGS**

Magnetic bearings, when operated essentially single-sided as in this case, are nonlinear. The force is related to the square of the coil currents (14)–(16). A typical equation for the flux density \( B \) in the magnetic circuit is given by:

\[
B = \frac{NI}{2A g R g}
\]  

where it is assumed that all of the flux is contained in the two air gaps and there is no hysteresis. Then the force \( F \) which attracts the rotor in a single-sided stator is given by:

\[
F = \frac{B^2 A g}{\mu_0} = \frac{\epsilon_0 N^2 I^2 A g}{4G^2}
\]

The term \( \epsilon \) represents the correction for leakage and fringing and it has a value of approximately 0.9 for thrust bearings. This force equation is dependent on both current \( I \) and position \( x \) of the thrust disk in the bearing where:

\[
G = G_0 + x
\]

Then \( F \) can be expressed as:

\[
F = K_{T1} I^2
\]

where the nonlinear current gain constant \( K_{T1} \) is given by:

\[
K_{T1} = \frac{\epsilon_0 A g N^2}{2G_0}
\]

for a single-sided thrust bearing. For a given bearing configuration, all of these parameters are known. It may be noted that this differs from normal practice in magnetic bearings which models the force as a linear actuator gain and a linear open loop stiffness term (7). The open loop stiffness term is assumed to be small in the following and is neglected.

For the CFVAD prototype, the thrust bearing was divided into two parts: Thrust Bearing A located at the back of the impeller and Thrust Bearing B located in front of the impeller. Both were constructed of silicon iron. The geometry for Thrust Bearing A is shown in Fig. 4. The outer pole geometry had \( OD = 33.0 \text{ mm} \), \( ID = 28.4 \text{ mm} \), and length \( L = 22.9 \text{ mm} \). The thrust collar was constructed of material larger in diameter than the outer pole geometry but the effective geometry was the same as the dimensions given above. The details of Thrust Bearing B are not given due to length restrictions. The design RMS thrust force was 9.2 N for Thrust A and 18 N for Thrust B.

The magnetic bearings supported the pump rotor and prevented it from contacting or coming close to the stationary housing. An active control system was used for the thrust bearings with an analog proportional-integral-derivative (PID) control system (7). The position feedback control signal was provided by an eddy current sensor. The space for the sensor is indicated in Fig. 4. Magnetic bearings tend to have large clearances, in the range of 0.25 to 1.0 mm which is very good for high flow rates in the clearances and low shear. However, it should be noted that the active control system keeps the actual shaft motions to 0.025 to 0.050 mm in amplitude.

Measurements were conducted on the rotor to evaluate the bearing current gain constant. For the double-acting bearing configuration, the governing equation is:

\[
F = M g [ K_{T1} I_1^2 - K_{T2} I_2^2]
\]

The bearing was placed in the configuration where the gravitational load was along the axis of the pump, as illustrated in Fig. 5. Four cases were considered as presented in Table 1. The first two cases yielded the results:

\[
K_{T1} = 0.010 \text{ N} \cdot \text{A}^2, \quad K_{T2} = 0.00721 \text{ N} \cdot \text{A}^2
\]
while the second set of cases yielded:

$$K_{T1} = 0.00979 \text{ N/A}^2, \quad K_{T2} = 0.00670 \text{ N/A}^2$$  \[8\]

These were reasonably close in value, indicating some consistency in the measurements. The average value was employed in subsequent calculations.

**RADIAL BEARINGS**

The radial bearing (A) at the outlet side was a 16-pole planar radial bearing stator shown in Fig. 6. It had dimensions of $OD = 89.5 \text{ mm}$, $ID = 43.4 \text{ mm}$, and length $= 8.2 \text{ mm}$. The rotor had dimensions of $OD = 4.24 \text{ mm}$, $ID = 3.30 \text{ mm}$ and the radial clearance was $g = 0.406 \text{ mm}$. The design RMS load capacity was $22 \text{ N}$.

The inlet side radial bearing (B) stator (not shown), was placed at the outer diameter of the inlet annulus. It had $OD = 79.4 \text{ mm}$, $ID = 26.9 \text{ mm}$, and length $8.2 \text{ mm}$. The rotor had $OD = 25.4 \text{ mm}$, $ID = 15.6 \text{ mm}$, and clearance $g = 0.406 \text{ mm}$. The design RMS load capacity was $18 \text{ N}$. The control system was analog PID and the sensors were variable reluctance sensors with locations indicated in Fig. 3.

The rotor was placed with the $z$-axis perpendicular to gravity, as shown in Fig. 7. The force balance in the $y$-direction was

$$f_A + f_B = M_x g$$  \[9\]

Here the orientation of Radial Magnetic Bearing A (at the outer diameter of impeller) was $\theta_A = -20^\circ$. The horizontal force centers were labeled as S1B and S1A, aligned with the sensor orientation, while the vertical force centers were labeled as S2A and S2B. For Radial Bearing B (at the inlet end), the orientation angle is $\theta_B = 0$. The horizontal force centers are labeled as S4A and S4B, again aligned with the sensor orientation, with the vertical force sensors labeled as S3A and S3B.
The force balance condition was applied to all of the magnetic gaps which gave the equation:

\[
K_{1A}l_A^2\sin\theta_A - K_{1B}l_B^2\sin\theta_A - K_{2A}l_A^2\cos\theta_A + K_{2B}l_B^2\cos\theta_A - K_{3A}l_A^2\cos\theta_B + K_{3B}l_B^2\cos\theta_B - K_{4A}l_A^2\sin\theta_B + K_{4B}l_B^2\sin\theta_B = M_r g
\]

assuming that the rotor is centered. This equation was employed for each of \( n \) measurements. In general, this set of \( n \) equations can be written as:

\[
[F_F] [K] = [M_R g]
\]

where \([F_F]\) is \( n \times 1 \), \([K]\) is \( 8 \times 1 \), and \([M_R g]\) is \( n \times 1 \).

The least-squares estimation of the \([K]\) matrix was given by:

\[
\hat{K} = ([F_F]^T [F_F])^{-1} [F_F]^T [M_R g]
\]

The final results were:

\[
\hat{K} = \begin{bmatrix}
K_{1A} \\
K_{1B} \\
K_{2A} \\
K_{2B} \\
K_{3A} \\
K_{3B} \\
K_{4A} \\
K_{4B}
\end{bmatrix} = \begin{bmatrix}
0.00653 \\
0.00522 \\
0.00581 \\
0.00550 \\
0.00378 \\
0.00795 \\
0.00486 \\
0.00715
\end{bmatrix} (N/A^2)
\]

These numerical values were employed in the determination of the bearing forces.

**FORCES ACTING ON THE ROTOR**

In this first fully magnetic bearing supported prototype, the rotor was centered by the bearing stiffness and damping properties (7), (17) but a number of external forces acting on the impeller were unknown (18). These included fluid forces, unbalanced forces, motor eccentricity magnetic forces, and others. Thus, the magnetic bearings were significantly overdesigned compared to the actual measured forces. One of the primary purposes for the development of CFVAD2 was the ability to measure the imposed forces on the bearings for future, smaller prototypes.

The bearing coil currents were measured and 10 forces—eight radial and two axial—acting on the rotor were calculated based upon the actuator current gains given in Eqs. (7), (8) and (18). The radial forces plotted were the forces F1B and F2B, at the impeller end of the rotor; F3B and F4A, at the inlet end of the rotor, were plotted in Fig. 8 vs. the aortic pressure (AoP) ranging from 55 mm Hg up to 160 mm Hg.

The initial static forces in air, with the rotor horizontally oriented, are shown in the first set of data points. It may be noted that the measured static vertical force \( F_2B \) = 3.5 N is relatively near the center of gravity while the weight of the rotor is 3.65 N. At the impeller end, the radial force was 1.50 N and the other forces are small. The next set of data points shown is the rotor running at 40 Hz in air and the radial forces are shown to drop somewhat due to the rotation effects.

Results with the impeller in water are also shown in Fig. 8. The measured static forces in water are similar to the static forces in air. As the pump rotor started rotating and pumping, the forces were measured. All of the forces dropped to approximately 1.50 N for all measured forces up to a pressure of 150 mm Hg. This is apparently due to centering effects of
the water which helps to take some of the impeller weight. Axial flow seals in industrial pumps are well-known to produce centering forces, positive stiffness and damping effects similar to this (19).

The measured total power delivered to the prototype magnetic bearings was approximately 50 watts (10 volts and about 5 amps). Most of this power was dissipated in the resistance losses due to bias currents of approximately 0.7 amps to each of the magnetic bearing coils.

PUMP PERFORMANCE TESTING

The pump was tested for pressure rise and efficiency, with the CFVAD2 unit connected to a simple mock circulation system (in a benchtop test) pumping water. The water was circulated with a physiological level of pump output and differential pressure. Pump inlet pressure (LAP) and pump outlet pressure (AoP) and volume flow rate were measured. Figure 9 shows the measured differential pressure head vs. flow rate at various rotor rotation rates. A typical flow of 6 liters/min was achieved against a differential pressure of 100 mm Hg at 38 Hz. A range of pressure head vs. flow rate characteristics may be obtained from this pump with small variations in pump rpm.

Figure 10 shows the efficiency curves vs. flow rate at the same rotation rates. The pump impeller was designed to have peak efficiency at approximately 16 liters/min (in the normal physiological range for humans) and the measured pump efficiency has a peak of approximately 55 percent at that flow rate. The curve shows the expected drop off in efficiency at higher flow rates. It is clear that the CFVAD2 meets the necessary flow requirements for a ventricular assist device.

CONCLUSIONS

A prototype continuous flow magnetic bearing supported ventricular assist device has been successfully levitated and rotated. The magnetic bearings operated successfully, centering the rotor in the clearance passages. A method was developed to measure the bearing gain constants for each of the 10 magnetic gaps in the pump by measuring bearing coil currents in various orientations with respect to gravitational loading. These gain constants were then employed to evaluate the external forces acting on the impeller. It was found that the radial forces on the impeller are significantly lower than expected, probably due to the load capacity, stiffness and damping effects of clearance passages.

The current prototype has several disadvantages. It is at least twice the size that is desirable for implantation, weighs too much, the power loss is too high, and too many wires are employed. However, having listed some of the shortcomings, the CFVAD2 prototype is apparently the first fully magnetically supported ventricular assist device. It has been quite successful as a test apparatus.

Work is proceeding on the next version of magnetic bearing-supported continuous flow ventricular assist device (CFVAD3) to reduce the size, weight, power loss, and wire count. The magnetic bearings are much larger than necessary in CFVAD2. With the low bearing load capacity data available from this prototype, the bearing size will be substantially reduced. The sensors will be eliminated and self sensing will be employed, reducing both size and wire count. The rotor will be substantially shortened, consisting of essentially the centrifugal impeller alone in a “pancake” type of design.

REFERENCES


