# DEVELOPING SEALLESS INTEGRAL MOTOR PUMPS USING AXIAL FIELD, PERMANENT MAGNET, DISK MOTORS

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## ABSTRACT

Problems with mechanical shaft seals account for the majority of pump maintenance costs. When pumping critical or hazardous fluids, users can resort to sophisticated mechanical shaft sealing systems, complicated buffered double seals, or sealless pumps in either a magnetically coupled or canned motor configuration. User preference based on past experience usually determines which concept is selected. While seal manufacturers have developed and marketed new seal configurations to meet the challenges of environmental regulations and seal reliability, sealless pump manufacturers have focused on component developments aimed at improving the reliability of existing designs.

Now, an innovative sealless pump configuration is presented as an alternative to the traditional approaches used to seal pumped fluids from the environment. This configuration utilizes an axial field, permanent magnet motor integrated directly into the impeller structure and pump casing. This integration produces a configuration that has many mechanical advantages over conventionally designed sealless pumps and pumps using sophisticated mechanical seal packages. The sealless centrifugal axial-field motor pump (SCAMP) requires a polyphase inverter (similar to a variable frequency drive), which offers additional operational benefits. A range of prototypes has been built and tested. Development of commercial applications is now in progress.

## INTRODUCTION

All centrifugal pump applications require that the pumped fluid be isolated from the environment by some reliable means. Mainly, this isolation is achieved through the use of a sealing barrier around the shaft that drives the pump impeller. This barrier takes the form of mechanical seal(s) or, on older pumps, packing material. Studies show that these sealing elements are a major source of reliability problems for pump users. The nuisance value of leaks plus economic, safety, and environmental concerns have forced an increased focus on leaking pumps (if the seal is failed or operating in an abnormal manner) and emissions (present even if the seal is operating properly).

Seal manufacturers have responded with the introduction of improved designs (using compliance with API 682 (1994) as a basis), utilizing ever more sophisticated sealing technology, to achieve reliable containment of the pumped fluid and the hazardous vapors sometimes associated with those fluids. The results are seals that are easier to install, more reliable, and will meet environmental regulations but that are still considered critical components. Pump manufacturers have responded over the years with the introduction of sealless pumps (either of the magnetically coupled or canned-induction motor variety). Their recent development efforts have focused on improving components (such as bearing materials and designs, and pressure barrier construction) and making the internal flowpaths more robust when handling difficult fluids. With the implementation of double containment features (which for a magnetically coupled pump could include a

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backup mechanical shaft seal on the magnetic coupling drive shaft or a double containment shell), these types of pumps can reasonably isolate the fluid from the environment. However, they are still perceived as complicated, difficult to maintain, expensive, and only marginally reliable in difficult services.

The pumps described in this paper constitute an alternative sealless pump design approach. This new approach is a radical departure from traditional sealless pump designs. It is based upon several technologies that have been successfully used in other product areas. These technologies are:

• High strength, rare earth, permanent magnet materials applied to disk motor configurations

• High current, high frequency switching semiconductors

• Product lubricated bearing materials and designs

Most pumps today (whether sealless or equipped with shaft seals) use a radial field, induction type motor. The sealless centrifugal axial-field motor pump (SCAMP) uses a permanent magnet disk motor, with the lines of electromagnetic and magnetic flux oriented across an axial gap between the stator and rotor, which is integrated within the pump's pressure boundary. The motor is operated by a combined rectifier/polyphase inverter drive. The power density of the motor design (inherent with a permanent magnet design) is large enough so that motor components can fit into the impeller disk(s) and casing end cover structure(s), and can generate higher torque levels over a wide range of rpm. The integration of this type of motor inside the pump, along with the use of product lubricated bearings, provides a mechanical package that eliminates shaft seals (and in fact eliminates any rotating shaft), is very compact, and eliminates large bearing spans and high bearing loads resulting from overhung rotating masses. The disk motor can be integrated into either a single-sided (or single-suction) or double-sided (double-suction) pump configuration. Either way, the advantages inherent in this approach result in improved reliability and reduced life-cycle cost over traditional systems.

Employment of this approach is suited for applications where the pump configuration includes a substantial radial orientation of the impeller shrouds. Proper matching of the motor torque characteristics with the pump impeller design (to achieve the desired head and flow) is important. Since motor commutation is supplied by the inverter, the maximum pump speed is not dictated by the line frequency like an induction motor. With variable speed, matching of a pump to the system requirements does not require impeller diameter cuts (the pump is always operating at the most efficient impeller diameter). The inverter and control electronics provide a basis for process control and also create a platform for continuous diagnostic monitoring of speed, input power, and other electro-mechanical information.

The sealed integral motor is cooled by the pumped fluid. Since the permanent magnet equipped motor is subject to the same temperature limitations as magnetically coupled pumps, fluid temperatures determine limits for some application. However, external routing of the pumped liquid through a cooler and returned around the rotor is possible, which would extend the useful temperature range of the SCAMP.

The Ingersoll-Dresser Pump Company and Kaman Aerospace, Electromagnetic Development Center, have jointly participated in the development of this pumping concept using both internal and external funded research and development (R&D) programs. Numerous prototypes have been and are currently undergoing tests for a variety of applications.

## CONVENTIONAL VERSUS SEALLESS PUMPS

Centrifugal pumps are essential to the operation of chemical and hydrocarbon processing plants. Clark and Littlefield (1994) have studied the economic relevance of pumps to the process industry. They estimate that:

• 20 percent of the cost of new plant equipment purchases involves pumps

• 80 percent of all electrical power consumed in the plant is used for pumps

• 40 percent of all maintenance expense is related to pumps

An added complexity for centrifugal pumps in a process environment is the degree of environmental regulation regarding leakage and fugitive emissions. Users can meet these requirements by using either sophisticated mechanical seal systems or completely sealless pumps (typically with double containment design features). The type of process plant (hydrocarbon or chemical processing) determines the kinds of fluids to be pumped, the imposed duty cycle, and the approach taken regarding selection of pumping equipment.

Hydrocarbon processing plants or refineries typically buy equipment that complies with API standards (standards that have a large influence over the mechanical design of equipment like pumps and seals). Historically, these plants have avoided sealless pumps in favor of pumps equipped with API compliant mechanical seals and seal systems.

Chemical processing plants use pumps that typically conform to ANSI specification. These specifications are less design focused and more configuration related. As such, purchasers can more easily apply sealless pumps and not violate the intent of the governing specification. These plants handle fluids that are often more dangerous than refineries and have historically been more inclined to apply sealless pumps, where double containment design makes them nearly fail-safe.

Both types of plants must still comply with environmental regulation on leakage and emissions. Also, they must satisfy the inexorable drive to reduce maintenance costs while achieving higher levels of process availability and improved efficiency of the process. This translates to a reduction in the total life-cycle cost of the pumping equipment. Any new pump design must successfully address these issues.

Flach, et al. (1998), studied the maintenance repair records from five chemical plants and three refineries and identified that 24 percent of the analyzable repairs throughout the plant were due to failures attributable to shaft seal design and selection. This study and others show that shaft seal reliability is a critical factor in the successful operation of a modern processing plant. Seal manufacturers have had to address seal reliability, but also had to deal with stricter environmental regulations governing leakage and emissions.

Pump manufacturers responded to the tightening emissions regulations with a "surge" in sealless pump development. These pumps were derivatives of existing sealless designs and used either a canned motor (with a sealed induction motor assembly submerged in the pumpage) or a magnetic coupling (using a set of rotating magnet assemblies separated by a pressure boundary that isolated the pumped fluid from the environment). These pumps required careful specification, manufacture, and installation to achieve acceptable life. To many users, these pumps are perceived as being prone to problems (mostly associated with the product lubricated bearings), and so are not considered as a universal solution to the goals of meeting stricter environmental regulations and reducing maintenance and operating costs.

A review of the *Cross-Referenced Indexes to the Proceedings of the International Pump Users Symposia* (1996) and from the *Proceedings* of the subsequent two years (1997, 1998) reveals trends regarding technical papers that focus on either mechanical shaft sealing systems or sealless pumps. The chart shown in Figure 1 compares the number of technical papers published in these two categories. (With seal related papers dominating the *Proceedings* by an almost two to one margin over sealless pumps. The only other two topics that rival sealless pumps are "cavitation" and "rotordynamics.")
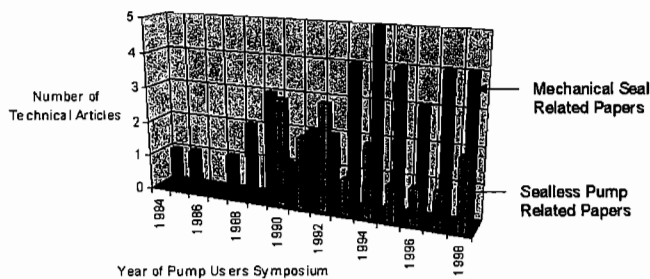
*Figure 1. Texas A&M International Pump Users Symposia Paper Topic Distribution.*

As seen in Figure 1, increased interest of pump manufacturers and users led to an increase in technical papers relating to sealless pumps for the years 1990 to 1992. These years correspond to increased concern over government regulations regarding fugitive emissions from process pumps. However, the year 1993 shows a dramatic increase in technical papers by seal manufacturers, most relating to seal reliability and emission control through application of new seal component and system designs. During the period from 1993 to the present, seal papers outnumbered papers on sealless pumps by an almost three to one margin.

What happened to the sealless pump momentum generated in the early 1990s? By studying the papers written following this period, technology development efforts were aimed at improving components (essentially the bearing materials and designs), advocating monitoring of the sealless pump to predict failure, or adding features to provide dual containment in the event of a primary containment failure. This collective work was defensive in nature and failed to yield any new approaches to sealless pump design. Today, sealless pumps and advanced mechanical sealing systems continue to coexist in the commercial marketplace. Both must contend with reliability and cost issues. Sealing systems that comply with API 682 (1994) are more likely to be found in refinery operations than sealless pumps. Chemical processing plants are also now using the improved seal technology in many applications. Sealless pumps appear to be viewed as specialty equipment, which are to be used only in certain critical applications.

However, over the last decade, a radically different sealless pump configuration has been undergoing development. With R&D support from a variety of sources, the sealed centrifugal axial-field motor pump (SCAMP) has been maturing. A pump representative of this type of machine is shown in Figure 2. (Also shown in Figure 2 are the key force vectors (other than the mass of the rotor itself) that influence the design of the single bearing assembly.) This configuration simplifies the mechanical layout of the conventional sealless pump. It uses a permanent magnet, brushless DC disk motor design and relies on high efficiency, current switching semiconductors built into the power section of a polyphase inverter to drive it at variable speed. What follows is a description of the development of these unique machines and how their application will benefit plant reliability and operability and lower the pump's total life-cycle cost.

## DISK MOTORS AND CENTRIFUGAL PUMPS

While disk motors are not new, and their integration into centrifugal pumping machines has been considered long ago, only recently have enabling technologies been developed that provide a suitable level of performance at a reasonable level of cost.

### Background—Disk Motors and Centrifugal Pumps

Dolman (1952) shows an axial air gap motor (with an axially oriented electromagnetic field, presumed to be an induction machine) integrated into a centrifugal pump. This is the earliest documented representation of this concept that the authors have found. Undoubtedly, it never evolved due to the minimal torque delivery available from motors oriented in such a manner.
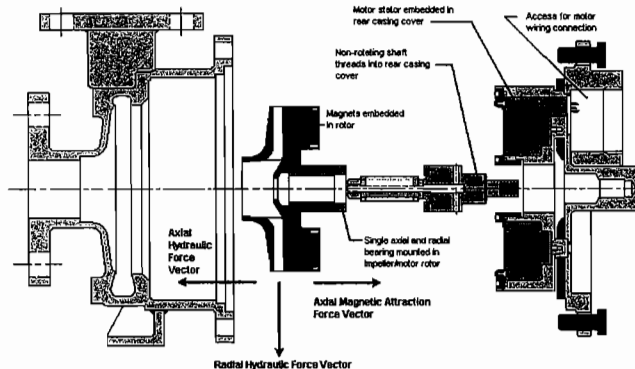


*Figure 2. SCAMP Cross-Section/Exploded. (This shows the basic configuration of a SCAMP.)*

Others have had notional concepts of such a machine since then (most notable being Igor Karassik, although no reference is available), but the lack of a motor that could develop sufficient power and torque within a diameter that provided practical pump performance prevented further work.

Igor Karassik (1990) did identify the desirability of operating pumps without seals, where the motor components were hermetically sealed within the pump casing. Although similar to the familiar canned motor pump, Karassik was envisioning a more intimate integration of motor and pump components. Karassik and McGuire (1988) also identified the benefits of variable speed operation apart from the well-known efficiency benefits achieved from matching pump output to process demands. These benefits came from slow start of the pump, reducing excessive internal forces arising from fixed speed operation at off-design flowrates, and minimizing stress on the pumping system due to unsteady pressures and flow delivery from pumps operating in suction and/or discharge recirculation mode.

In order to pursue the direct integration of high-power disk motors into the pump, new motor technology was required. Developments by Slemon (1994), Mongeau (1996), and Hawsey and Bailey (1992) are typical of the activity focused on advancing permanent magnet, axial field disk motor technology (including performance, reliability, and practical manufacture). As will be seen later, torque and power delivery several times greater than a conventional induction machine are possible with these machines. Once a practical means of cooling the motor stator (and, to a much lesser degree, the rotor) is established, power densities several times greater than induction motors can be realized. In turn, this increased power density makes it practical to integrate the disk motor directly into the pump impeller and casing.

Many patents have been issued relating to the integration of a disk motor(s) within the structure of the pump. A patent issued to White (1956) defined the basic elements of today's SCAMP concept. A copy of one of the patent figures is shown in Figure 3. White's concept actually predates Dolman, since the White patent was applied for in 1950. White included the nonrotating shaft that located the impeller (although he incorporated an antifriction bearing set sealed within the impeller hub). He also addressed the cooling of the motor stator. However, he undoubtedly was limited to the use of ordinary induction motor technology and so could not provide enough torque, for a given diameter, to make the concept worthwhile.

Mizobuchi, et al. (1989), patented a configuration similar to White (1956). The significant claim from this patent was the use of a single spiral groove axial thrust bearing on a single, nonrotating shaft. No specific claims were made relative to the motor.

A patent issued to Cooper and Bulson (1992) claimed the integration of a pair of disk motors within the structure of a double-suction impeller (for the rotor components) and the pump casing (for the stator components). The Cooper and Bulson (1992) patent was specific in its claiming the use of a permanent magnet, disk type motor arrangement.

July 3, 1956       H. T. WHITE       2,752,857

MOTOR PUMP UNIT WITH AXIAL GAP MOTOR

Filed June 8, 1950            2 Sheets—Sheet 2
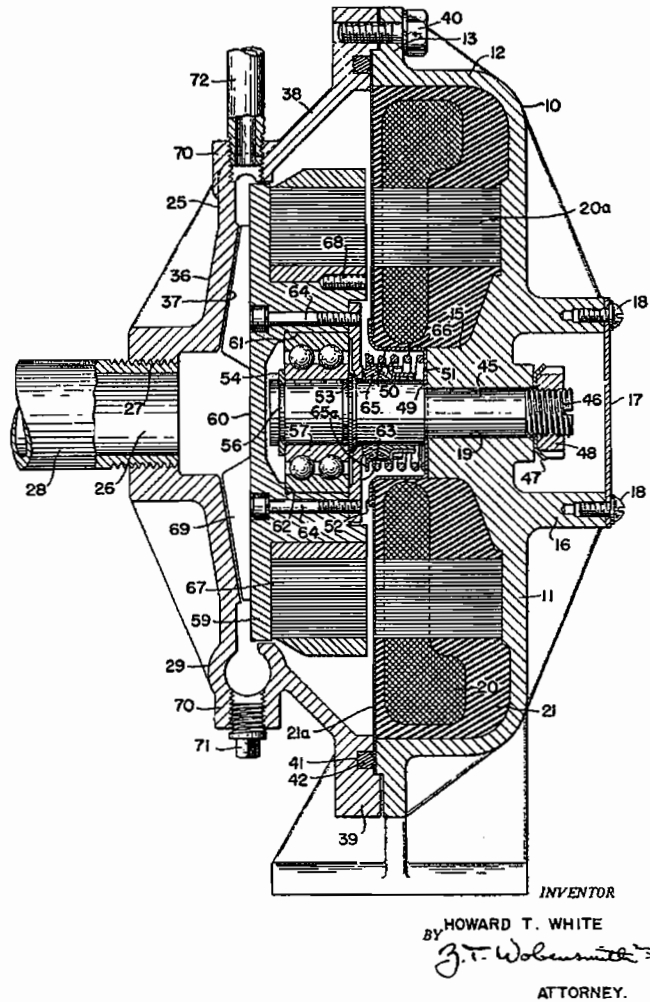


INVENTOR

BY HOWARD T. WHITE

ATTORNEY.

*Figure 3. Axial Field, Integral Motor Pump of White (1956). (This is the earliest detailed drawing of an axial field motor integral with a centrifugal pump.) (Courtesy US Patent and Trademark Office)*

At the same time, Kricker and Childs (1994) were issued a patent on a single-suction machine similar to that of Mizobuchi, et al. (1989), and White (1956), but also claiming a permanent magnet configuration. Kricker and Childs (1994) major claim covered the rotor position sensor arrangement necessary to provide feedback to the motor controller. This machine, like that of Mizobuchi, appears to be limited in performance due to lack of sufficient provision for cooling the stator coils.

In a patent pending by the authors of this paper, Sloteman, et al. (that was conceived in 1993, and applied for in 1997), a unique cooling flowpath for motor cooling was claimed, as were several unique constructional features that facilitated assembly and disassembly of the unit. The claims in this patent application form the groundwork for successfully integrating high powered, permanent magnet motors into the structure of a centrifugal pump and generating sufficient torque to provide a commercially useful level of head, flow, and efficiency.

*Disk Motors—Design and Performance*

The availability of high strength permanent magnets (neodymium iron boron or samarium cobalt) has enabled motor designers to develop a new class of high torque and very power

dense electric motors. These permanent magnet (PM) machines can be configured in either a radial or axial gap configuration. The conventional induction motor applied to pumping applications is a radial gap machine. For integration into centrifugal pumps, the disk motor is ideal. The PM motor (whether in disk or cylindrical configuration) offers a number of features that enhance their value in a number of applications, while at the same time presenting the designer with a number of challenges. For the integral motor pump applications discussed in this paper, we will see how those features and challenges are applied and met.

Mongeau (1996) provides a fundamental explanation of the nature of the electromagnetic behavior of the PM motor, along with a practical framework for analyzing the performance characteristics of these machines. The development of the axial gap PM motors has resulted in a number of commercial motor packages being available. The motor shown in Figure 4 is such a machine (not shown in this picture is the rectifier/inverter drive package needed to operate the motor). This particular motor (liquid cooled), which uses an 18.3 inch rotor diameter, delivers 500 hp at 6000 rpm and can provide rated torque at zero rpm (a feature not required to drive pump loads where the torque required varies inversely with the square of the speed). PM motors are synchronous in nature in that they require precise rotor positional information to be supplied to the inverter so that proper commutation can be achieved.
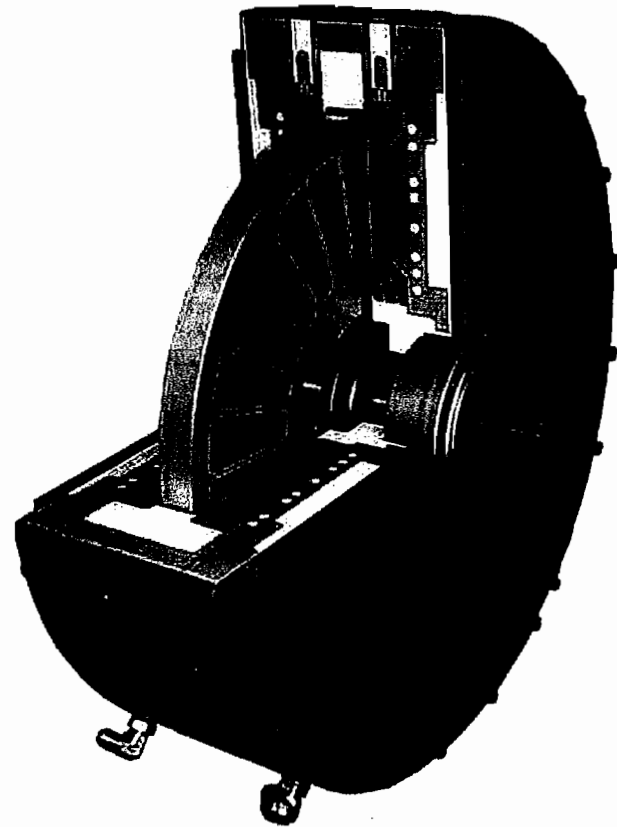


*Figure 4. Cutaway of Modern Permanent Magnet Disk Motor.*

The detailed explanation of the electromagnetic operation of this motor is beyond the scope of this paper. However, four important concepts must be considered in order to understand how the PM disk motor concept can be applied to sealless pump applications:

- Magnet/rotor configuration (surface or interior)
- Electromagnetic shear stress

- Motor efficiency

- Thermal management

A basic disk motor (single-sided) configuration is shown in Figure 5. In this notional motor design, the magnets are mounted in "surface" configuration on a magnetically conductive back iron. The magnets are positioned so that the N-S poles are oriented axially, with the polarity reversed with each adjacent magnet. Mongeau (1996) describes several detailed reasons why the surface configuration is preferred; put simply, it brings more magnetic flux to bear on the stator, improves cooling of the magnets, and allows the motor to operate at practical axial rotor-to-stator gaps.
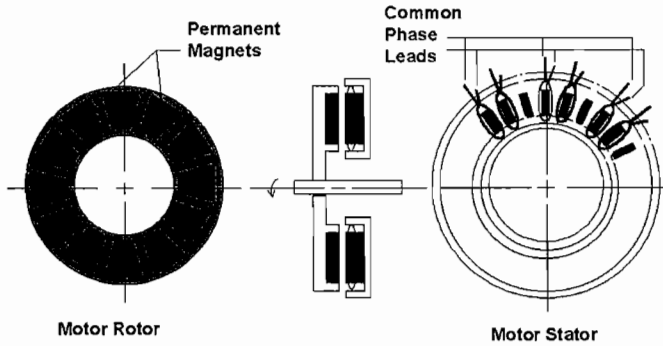


*Figure 5. Disk Motor Construction (Notional). (The basic elements of a disk motor, the rotor with surface mounted permanent magnets, and the stator are shown.)*

The electromagnetic shear stress is a useful metric for evaluating not only PM disk type motors but all motors, no matter what the topology. This shear stress, $\sigma$ (lbf/in$^2$), is defined as the torque producing force developed per unit gap area of the machine. In electromagnetic terms it is defined as:

$$\sigma = 2 \ K_o \ B_{avg} \ / \ \pi \qquad (1)$$

where $K_o$ is the amplitude of the stator current loading and $B_{avg}$ is the average gap magnetic flux density. The $B_{avg}$ is a function of the magnet material and its volume on the rotor, and the $K_o$ is a function of the ability to get current through the copper windings that are a part of the stator. Mongeau (1996) shows that shear stress values of 10 lbf/in$^2$ are possible with specially designed PM motors (the motor shown in Figure 4 develops a $\sigma$ of 5.5 lbf/in$^2$). Various design factors influence the level of electromagnetic shear stress developed by any one machine. By contrast, today's fan cooled induction motors develop a shear stress of between .5 to .75. Liquid cooling of the induction motor (as in a canned motor pump) will increase this value to 1.0 to 1.5. We find that the higher the shear stress value, the more critical the design and performance of the motor components. Therefore, it is necessary to evaluate the requirements of a subject application to avoid over-design of the motor.

The shear stress can also be related to developed torque in non-electromagnetic terms. By making the reasonable assumption that the electromagnetic shear stress does not vary significantly with radius, the torque developed can be defined as an integral function. The following equations relate the physical geometry of the disk motor (Figure 6) to shear stress:

$$\text{Torque} = \times \ \sigma \ r \ (2\pi r) \ dr \qquad (2)$$

Solving this integral between the limits of the outer and inner magnetic radii, the relationship between torque and shear stress is:

$$\sigma = \text{Torque} \ / \ (2/3) \ \pi \ (r_{m,2}{}^3 - r_{m,1}{}^3) \qquad (3)$$
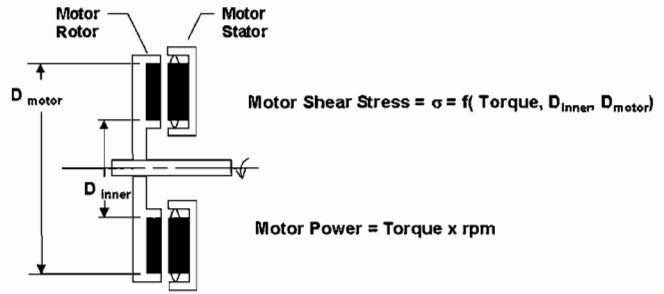


*Figure 6. Disk Motor Key Dimensions.*

This equation connects the electromagnetic design (expressed as the shear stress value) with the physical size of the motor and defines the available torque that is used with the motor speed (rpm, $N$) to calculate the effective shaft horsepower:

$$P_{shaft} = \text{Torque} \times N \qquad (4)$$

The efficiency level of the PM motor depends on stator and rotor design as well as the matching of the polyphase inverter to the specific motor characteristics. Typically, the efficiency is better than found in ordinary, commercial induction motors. Mongeau (1996) reports motor efficiencies in excess of 95 percent. The efficiency of the PM motors drops off slowly as torque is reduced from the design point. Also, large reductions in motor rpm can lead to drop-off of efficiency unless the DC bus voltage is also reduced to account for the lower induced back electromotive force (EMF) voltage. However, proper design and application of the PM motor and its polyphase inverter will usually provide superior motor efficiency.

A key aspect of "proper design and application" lies in the thermal management of the losses generated in both rotor and stator. Rotor losses, although small, are dominated by the eddy current heating in the magnets. Some rotor cooling is required to keep the magnet temperature below the level at which demagnetization occurs, typically between 150°C and 180°C. If the pumped fluid temperature is significantly higher than these temperatures, the useful operating temperature range of the machine can be extended by selection of magnet grade (trading off temperature capability with field strength) and the use of secondary cooling loops. The majority of loss in the motor occurs in the stator. For machines discussed here, resistance losses dominate and are caused by the current flow in the copper and iron. The ability to remove the resulting heat and keep the temperature of the copper windings at levels commensurate with the insulation system used is key to maximizing the absolute level of stator current loading, $K_o$ used in Equation (1). Several design features of the PM motor described by Mongeau (1996) lend themselves to minimizing the stator losses, thus creating higher motor efficiency and reducing the total amount of heat to be removed.

Significant design and engineering effort have been put into the thermal management of disk motors. In their application to pumps, rotor cooling is accomplished by the submergence of the rotor disk in the pumped fluid. The stator is also cooled by the pumped liquid (although external cooling has also been implemented for special applications). This type of cooling can be referred to as "indirect conduction" (Mongeau, 1996). In most pumping applications, the stator components are sealed from the pumpage, which forces the heat path from the copper to the liquid through several elements in series (copper, iron, and sealing surface). Minimizing the conduction resistance in each element and ensuring good thermal interfaces between them is essential.

The design features incorporated into the PM disk motor to ensure good heat transfer offer another benefit. These features also ensure a solid stator/copper assembly. This means that the stator is immune to large shock loads and the thermal potting material used to impregnate the stator and provide good heat conduction also

serves to solidly encapsulate the copper coils. This encapsulation minimizes flexing and movement of the copper coils and prolongs the life of the windings insulation system.

### The Inverter/Rectifier

The operation of a PM motor is synchronous in nature. Rotation is accomplished by switching the flow of current through sets of stator teeth (the phases) in a sequential manner. The permanent magnet field emanating from the rotor augments and follows this rotating electromagnetic field. The notional motor configuration shown in Figure 5 includes a view of the stator. Here, coils of every third stator tooth are connected together and form one of the phases of the motor. High performance PM motors are configured as either three-phase or six-phase machines. The inverter is the device that supplies the current to the motor phases. The inverter has the ability to switch the current between the phases at precisely the correct time. This is essential for the commutation of the motor and is responsible for achieving high levels of efficiency.

The block diagram shown in Figure 7 presents a simplified view of the motor drive. The inverter switches current between the motor phases to maintain the desired rotational speed of the motor. The ramp up and ramp down of this switching function is ideally sinusoidal in nature. However, an alternative, the six-step waveform, can be substituted to simulate the sine wave. The six-step approach, while requiring less expensive electronics, generates unwanted harmonics both upstream and downstream of the inverter. The rotational speed of the motor (rpm) is determined by the frequency of the drive ($f_d$ in cycles/sec) and is dependent on the number of pole pairs ($p_p$) built into the motor (the number of stator teeth divided by the number of motor phases):

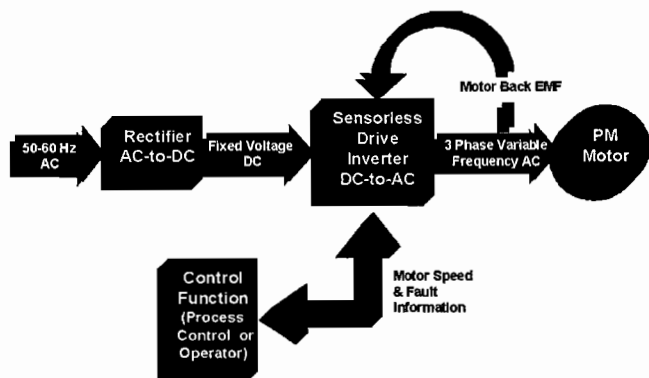$$RPM = 120\ f_d\ /\ p_p \qquad (5)$$



*Figure 7. Inverter/Rectifier Drive for Integral Motor Pump. (This identifies the basic elements that comprise a sensorless drive used for brushless DC permanent magnet motors like those used in SCAMPs.)*

The speed of the PM motor is not limited to synchronous line frequency as is the induction motor, but is instead a function of the controller selected switching frequency of the semiconductor based power transistors. The controller is commanded to operate the motor at the desired rpm. This command can either originate from a process control set point or be manually instructed by an operator. Variable speed operation is inherent in the drive.

The timing of the current switching between motor phases determines the rpm, but another method is used to vary the amount of current supplied to each phase. Superimposed on the phase currents is a pulse width modulated signal (PWM) of very high frequency, typically between 8000 to 20,000 Hz, depending on the design of the inverter. This PWM signal operates between the ground state voltage and the maximum DC voltage delivered by the rectifier. The duration that each high frequency pulse is left on determines the amount of current delivered to the stator phase. If the controller senses that the motor rpm is dropping below the motor set point speed (indicating an increase in load), the width of each of the high frequency pulses is increased. Conversely, if the speed is sensed to be rising above the set point (indicating a reduction in load), the pulse width is reduced.

Current is supplied to the inverter via an AC-to-DC rectifier. The rectifier converts the AC voltage to DC. The rectifier must be properly matched with the inverter and motor so that a suitable DC voltage level is available on the "DC bus." The matching makes sure that the DC voltage available is greater than the back EMF generated by the rotating magnetic field in the motor.

The back EMF serves a valuable role in terms of motor operation. As mentioned earlier, the PM motor requires precise rotor position information in order to properly command the switching of current between the motor phases. Traditionally, this has required some form of position sensor be integrated into the motor assembly. The sensors then fed the information to the controller, which in turn commanded the switching of the power transistors and also the width of each PWM pulse. Now, sophisticated circuits with signal processing capability can be used to analyze the voltage signal on each phase line and determine the rotor position based on the fluctuations of voltage levels due to individual magnets passing the stator teeth. This sophistication eliminates costly and potentially unreliable sensors, which in the case of a sealless integral motor pump would have had to be built into the pump casing and reside within the pressure boundary.

The presence of the inverter and controller electronics serves as a platform for a variety of diagnostic tools that can be used to protect the machine as well as identify potential failures. For the example of the sealless integral motor pump, run dry can be detected as a sudden drop in motor input power, without a reduced rpm command. In response, the motor can be shut off and wait for a reset command. Another example would be detecting rotor position changes (indicating wear of the thrust bearing face) by an increase in the motor back EMF voltage, indicating a reduction in rotor to stator clearance. More features can be integrated to provide additional protective and diagnostic capabilities.

With the cost of the current switching semiconductors and attendant control electronics dropping, the use of variable speed motor drives (both for induction type motors as well as the PM motor) is becoming more economically viable. By using these drives, users can realize the following benefits:

- Higher process efficiencies

- Reduced "wear and tear" on bearings (slow start and reduced forces at off-design flow condition)

- Built-in pump condition monitoring

A photo of a high performance rectifier/inverter/control system is shown in Figure 8. This drive is capable of powering the motor, shown in cutaway fashion, of Figure 4.

## MATCHING DISK MOTOR PERFORMANCE WITH CENTRIFUGAL PUMP PERFORMANCE

The geometry of the disk motor lends itself to integration within the structure of a centrifugal pump (Figure 9). In order to define the integral motor pump design, there are several pump performance relationships that must be correlated with the motor Equations (2) and (3).

If it is assumed that the motor shaft power equals the pump input power, then:

$$P_{shaft} = Q\ H\ /\ \eta_{pump} \qquad (6)$$

The pump head, $H$, can be calculated from:
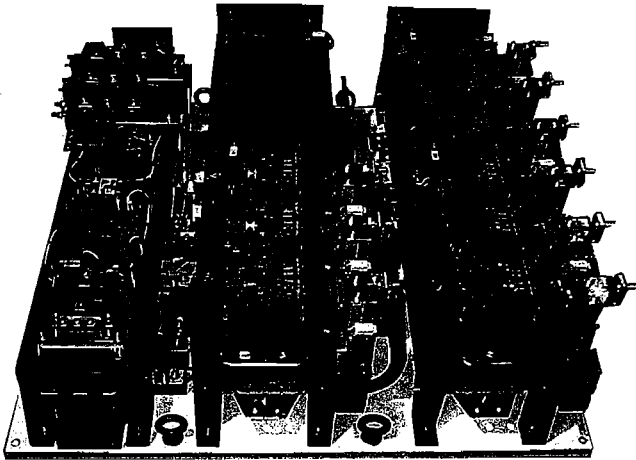
$$H = \Psi\ (\omega\ R_{impeller})^2\ /\ g \qquad (7)$$

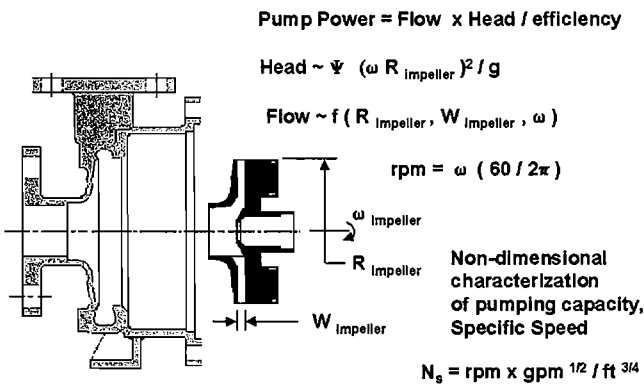*Figure 8. Photograph of a Drive for a High Power, Permanent Magnet Disk Motor.*

**Pump Power = Flow x Head / efficiency**

**Head ~ Ψ (ω R $_{impeller}$ )$^2$ / g**

**Flow ~ f ( R $_{impeller}$ , W $_{impeller}$ , ω )**

**rpm = ω ( 60 / 2π )**

ω Impeller

R Impeller

W Impeller

**Non-dimensional characterization of pumping capacity, Specific Speed**

**N$_s$ = rpm x gpm $^{1/2}$ / ft $^{3/4}$**

*Figure 9. Pump Dimensions.*

The Ψ represents the impeller head coefficient, which is basically determined by the impeller blade geometry. The flowrate, $Q$, is more complicated to calculate but can be expressed as a function of the following:

$$Q = f(R_{impeller}, W_{impeller}, \omega) \qquad (8)$$

The $\eta_{pump}$ term encompasses all the pump inefficiencies (due to leakage, disk drag, and hydraulic losses). The motor efficiency is not included in this relationship. However, the disk drag power and the leakage flows (including motor cooling) that are related to the presence of an integral motor will be included in this efficiency. Sufficient flow must be bled off from the discharge to remove the losses manifested as heat generated by the motor stator and rotor ($\eta_{motor}$):

$$\text{Heat Rejection} = (1 - \eta_{motor}) \times P_{shaft} \qquad (9)$$

The extra disk drag comes from the incremental width to the impeller due to inclusion of the motor rotor magnets and backiron (refer to Figure 2).

The hydraulic efficiency is often dependent on the specific speed of the pump:

$$N_s = N \, Q^{1/2} / H^{3/4} \qquad (10)$$

The Worthington efficiency curves found in Figure 10 show that the pump efficiency increases with increasing specific speed (for $N_s < 2,000$). The use of the PM motor allows operation of the pump at speeds greater than synchronous. The result is a pump that

operates at a higher rpm ($N$) (within the limitations of the available suction pressure) and will enjoy a hydraulic efficiency advantage over a pump forced to operate at a line frequency derived motor rpm (and hence a lower $N_s$).
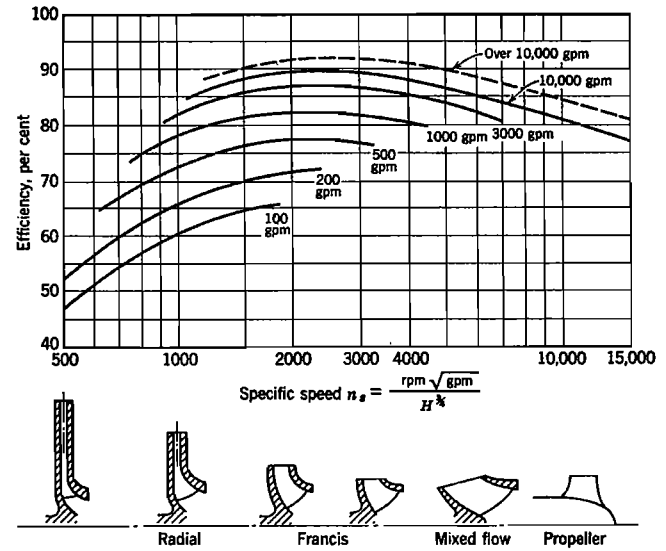


*Figure 10. Pump Specific Speed Versus Efficiency and Impeller Geometry.*

Considering the factors of increased leakage required to cool the motor, larger disk drag power losses and improved hydraulic efficiency, it is usually found that the $\eta_{pump}$ for the integral motor pump is similar to that found on conventional pumps and is probably superior to today's sealless pumps.

Figure 10 also provides insight into the optimum impeller geometry for a given specific speed. The impeller cross sections at the bottom of the figure show the relative differences in outer diameter to inlet diameter ratio for a range of specific speeds optimized for efficiency. At specific speed less than about 1500, the optimum geometry has a substantial radial shroud orientation. As the impeller becomes more "mixed-flow" in configuration, the radial extent of the impeller shrouds disappears. It is obvious that the disk type motor, integral with the impeller shrouds, is more adaptable to lower specific speed applications.

Using the simple equations (Equation (1) through (9)) that describe the motor and pump geometry required to meet a desired performance requirement, it can be said that lower pump specific speeds require disk motors with lower electromagnetic shear stress. Conversely, the higher the specific speed, the higher the shear stress must be (or else the motor disk would have to be larger in diameter than the impeller that would incur higher disk drag losses). Disk motors with higher shear stress values tend to more aggressive designs (both thermally and electromagnetically) and lead to higher manufacturing costs. It is therefore practical to limit potential applications to specific speeds of less than 1500.

## ELEMENTS OF INTEGRAL MOTOR PUMP DESIGN

There are several elements of the pump design that require special consideration in the context of SCAMP type machines. In all cases, these elements require no careful application of existing, proven technologies.

*Pump Configuration—*
*Single-Suction Versus Double-Suction*

Thus far, we have considered single-sided disk motors, integrated into single-suction pumps, with the motor integrated within the hub of the impeller and the rear casing wall (Figure 2).

Another concept places disk motors into both shrouds of a double-suction pump (Cooper and Bulson, 1992). Here (Figure 11), the pair of disk motors supply power to the impeller. Support of the rotor is via bearings that are located over the inlet sections at each end of the impeller. The double-suction configuration doubles the motor torque available for a given impeller diameter. It handles twice the flow of the single-suction configuration. The double-suction configuration also offers balance to the magnetic attraction force inherent in the PM motor design.

## Conventional, Double-Suction Rotor Configuration
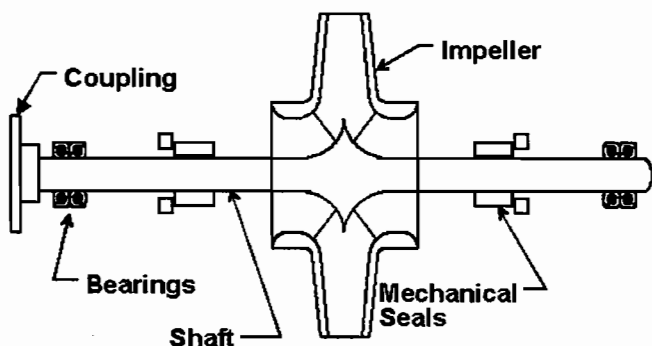


## Integral Motor Pump, Double-Suction
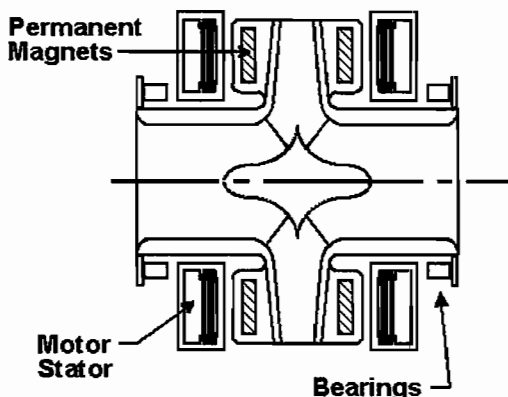


*Figure 11. Double-Suction Configuration.*

However, the double-suction configuration adds to the complexity of construction, assembly, and wiring of the two motors (wired in parallel for operation by a single inverter, or wired independently and driven by two inverters to provide a measure of redundancy). It is used when the application demands its special features and increased pumping capacity.

### Internal Forces

The forces present in a centrifugal pump using an integral PM include:

- Pressure forces acting on stationary casing parts
- Pressure forces acting radially on rotor/bearing components
- Pressure forces acting axially on rotor/bearing components
- Magnetic attraction forces acting on rotor/bearing components
- Rotor unbalance force

- Weight of rotor components
- Hydrodynamic forces generated in close radial clearances

The first item is dealt with in all centrifugal pumps by careful design of the external pressure boundary. In fact, the compact dimensions of SCAMP type pumps would simplify the casing design for high pressure applications. Additional pressure loads related to explosion-proof capability must also be accounted for in the casing structure design. Unbalance force and rotor mass is easily determined. Hydrodynamic forces are of major concern on high-energy, high-speed pumps, but are not of significant interest on the single-stage devices discussed here. The other forces require additional discussion in the context of the axial field motor topology. Refer to Figure 2 to identify the orientation of each of the following force vectors acting upon the pump rotor.

The typical radial and axial hydraulic forces described in Agostinelli, et al. (1960), and Guinzburg and Buse (1995), act on SCAMP impellers in the same manner as conventional pumping machinery. The compact mechanical configuration of SCAMPs places the radial bearing(s) in very close proximity to the radial load, thus reducing overhung moment found in conventional pumps. The axial thrust that is developed is actually used to offset magnetic attraction forces inherent in the SCAMP design.

With SCAMP, integral permanent magnets generate a large attraction force between the rotor and stator. On double-suction designs, this force is essentially balanced. Only the variations in axial clearance between the sides will produce a net attraction force. However, on single-suction pumps, the attraction force is considerable. In general, the magnetic attraction force is a function of the type of magnets selected, the swept area, and the effectiveness of the flux path defined by the magnetic circuit.

For single-suction machines, if the designer can maintain:

$$F_{axial, \, magnetic} > F_{axial, \, hydraulic} \qquad (11)$$

for all flow and speed conditions, then only one thrust bearing face is required.

Testing performed on a prototype machine similar to the one shown in Figure 2, but equipped with an axial load cell, confirmed the applicability of the single axial thrust face. The data shown in Figure 12 were measured for typical rotor-to-stator gap and commercial values of internal pump clearances. The load map in Figure 12 was measured over a wide speed range and with the pump operating at best efficiency flow. At zero rpm, it is observed that the load cell only measures the force caused by the magnets. As rpm increases and the pump begins to generate head (and pressure), the force measured by the load cell decreases. The $F_{axial, \, hydraulic}$ is increasing by the square of the rpm increase. However, the $F_{axial, \, magnetic}$ remains constant, unaffected by the pump rpm. This characteristic was observed at flowrates near shutoff as well.
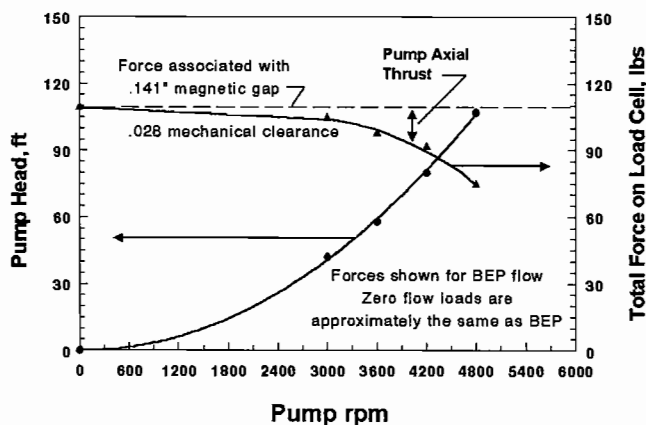


*Figure 12. Axial Thrust Characteristic of a Four Inch SCAMP.*

Besides allowing the use of a single thrust bearing in the machine, the reduction in net force with increasing rpm actually lowers the PV value (the product of the unit pressure loading on the thrust face multiplied by the linear face velocity). This characteristic is opposite to the behavior of many conventional sealless pump thrust bearings (which rely on double-acting systems). In those machines, the higher the head (or pressure), the larger any residual net axial thrust that must be reacted by the bearing face. This leads to PV values that increase with the cube of the rpm.

Double-suction machines would require two thrust bearing faces to account for the uncertainty of the direction of the ultimate net axial thrust vector, unless unusual design features are incorporated to orient the net thrust in one direction at all times.

*Bearings*

Radial and axial bearings are required to support the pump/motor rotor relative to the casing. With the sealless design (and no rotating shaft present), antifriction bearings are not practical due to the variety of fluids being pumped, temperatures, contaminants, and the churning power losses. Instead, hydrodynamic bearings of suitable materials, operating in the pumped liquid and operating on a fluid film are the most practical bearing for this application. A magnetic bearing system applied to the sealless pump, as envisioned by Allaire, et al. (1989), and Cooper and Bulson (1992), would provide the ultimate in sealless pump reliability by eliminating the potential for internal contacting or particle damage in the fluid film type bearing. However, with today's technology, magnetic bearings are being applied only in special applications, where their cost can be justified by the extra value they provided.

The bearing size is determined from the physical loads and moments of the system. The loads are calculated from information discussed in the previous section. In addition, the static weight of the rotor, loads due to misalignment of the bearing journals, and also the contribution of residual unbalance in the rotor assembly must be considered. Proper operation of the bearing is based on the ability of the fluid film to support the imposed load. A bearing exposed to a heavy load will break through the film and operate in a boundary lubrication mode resulting in increased wear. In this regime, the PV value is important in determining the ultimate life of the bearing. Conversely, if the bearing is too lightly loaded, instability will occur in the bearing journal and excessive vibration will result. If the pump's operating range covers a wide range of loads, more sophisticated bearing designs are available to provide the load carrying capability and maintain stability.

Advances in product lubricated bearings is one of the enabling SCAMP technologies. Advances in design and material of construction result in bearings that are highly resistant to entrained contaminants in the fluid. The most notable is silicon carbide and porous variants impregnated with material like graphite makes it reliable when called upon to run dry for brief periods of time. The range of bearing design and materials available for sealless pump operation is described by Swann, et al. (1997). Through design and material selection, the correct bearing design for the load, speed, temperature, viscosity, and other operational requirements is available.

*Internal Flowpath Considerations*

The motor components that are sealed within the SCAMP require a source of cooling. Also, the bearing system requires a flow of liquid to carry away heat due to friction and to flush contaminants. Usually the pumped liquid serves both of these purposes (although in some cases the motor stator can be externally cooled). Two distinct cooling flowpaths are used in a SCAMP. These paths originate at the outer diameter (OD) of the rotor and are driven by the pressure difference at the rotor OD and suction (Figure 13).
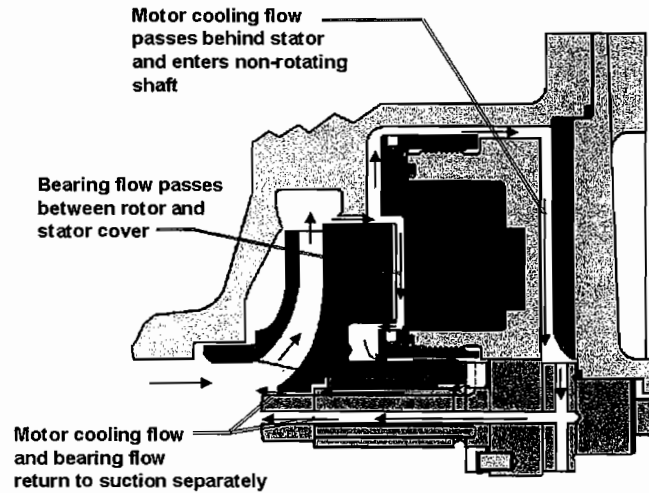


*Figure 13. SCAMP Cooling and Bearing Flowpaths.*

The fluid used for stator cooling is directed along an annulus and enters a passage behind the stator pocket. This radial passage channels the flow inward. As the flow passes over the metal surface isolating the stator core from the liquid, the temperature difference between the fluid and the stator core drives heat from the core into the fluid using both conduction and convective heat transfer. The heated fluid enters passages installed in the nonrotating shaft. These passages in turn direct the stator cooling flow back to the suction. Once leaving the region of the rotor OD, the cooling flow does not encounter any rotating surface or channels in its path back to the suction. This greatly simplifies the fluid path and prevents any sudden reductions in static pressure that could lead to flashing and resultant interruption of the cooling flow.

The second of the two paths is used to lubricate (cool) the radial and axial bearing assembly (Figure 13). Liquid for the second path leaves the stator cooling flow at the rotor OD and enters the gap between the impeller and stator. What little heating has occurred in the rotor magnets is removed, through the metallic can, by this flow stream. This stream does not suffer any temperature rise from the stator heating since a nonmetallic cover plate is used to isolate the stator from the pumped liquid. This cover must be nonmetallic to prevent the formation of eddy currents in the plate. The cover plate serves as an effective heat transfer barrier and prevents this flow stream from incurring any significant temperature rise before it gets to the bearing surfaces. This fluid then passes through the bearing (initially the axial face and then the radial journal) and returns to suction.

The segregation of these two flow streams reduces the risk of cavitation occurring in the bearing. This cavitation can take place when pumping fluids very near their vapor pressure. The resulting cavitation leads to loss of bearing support, additional bearing temperature rise, and ultimate failure of the bearing itself.

Careful attention must be paid to this cooling flowpath design. Models similar to those developed by Guinzburg and Buse (1998) are necessary to properly configure the internal flowpaths to achieve the desired pressure distribution and flowrates needed to cool both the motor and the bearing. These models are also important for determining the levels of axial thrust present in the machine. If a single axial bearing design is used, the direction of the net axial hydraulic force must be opposite to the magnetic attraction and smaller in magnitude.

For double-suction configurations (Figure 11), it is impractical to achieve the segregated cooling flows. Here, in order to simplify the internal design, the motor cooling flow will be forced through the bearing on its way back to suction. Careful attention must be given to prevent bearing cavitation (as is true with conventional magnetic drive or canned motor pumps) due to the heated return flow.

Another area of concern is the presence of high concentration of particles in the flow streams. Some degree of tolerance to particles is inherent in the design. Hard surface bearings (like silicon carbide (SiC)) are used in the presence of particles. Also, the passages should be configured to prevent accumulation of solids. If significant solids are expected, a wash-flow filter can be integrated at the rotor OD. This type of filter is commonly used in sealless pumps to minimize the solids concentration in the fluid used to feed the rotor bearings.

*Materials of Construction*

SCAMP type machines impose only a few constraints to the selection of materials. Since the motor components (copper, iron, magnets, connectors, etc.) are completely sealed from the pumpage, the criteria for selecting the casing, impeller, and bearing components are identical to conventional sealless pump practice. Materials are selected for their compatibility and corrosion resistance with the pumped fluid. If explosion-proof requirements exist due to the fluid and location of service, higher strength materials should be evaluated to reduce casing section thickness requirements.

Three areas require special material selection. The cover plate that separates the magnet surfaces from the pumped liquid needs to be nonmagnetic. Using metals that are magnetic increases eddy current activity and results in power losses, and will require unwanted cooling of the rotor surface. The plate should be suitable for welded connection to the rotor body. Special welding processes are used to accommodate the strong magnetic fields present when trying to attach the cover plate to the impeller structure. Nonmetallic materials can be considered for some components in the pump. Completely molded impeller/motor rotor assemblies can be envisioned. Here, careful attention is required to minimize the rotor heating potential, since the nonmetallic encapsulation acts as a thermal barrier as well as a sealing interface.

Similarly, the cover over the stator must be nonmetallic. Here, the large variations in the rotating permanent magnet field interact with the stator teeth to form the potential for strong eddy currents. Nonmetallic covers eliminate this problem and also form an ideal barrier to heating the bearing cooling flow that passes between the rotor and stator.

Finally, the material used between the stator core and the motor cooling fluid must be chosen to minimize the fouling due to corrosion or biological growth. A suitable fouling factor must be used in designing a SCAMP type machine in order to achieve a suitable life of useful operation.

## DEVELOPMENT EXPERIENCE

To date, four different SCAMP prototypes have been built and tested. Each represents a slightly different adaptation of the integration of high-power disk motors within the pressure boundary of a centrifugal pump. Each pump also addresses different requirements and design problems that can confront pump designers when facing challenging applications.

*Electrolyte Battery Pump—QPM*

The quiet pump motor (QPM) was developed to circulate a NaOH electrolyte solution through a high capacity battery system. This battery was intended to provide energy for an electric drive underwater vehicle. The compact and sealless nature of the SCAMP concept made it ideal for this application. Special requirements included design for very short total life and achieving low noise signature levels. The performance and size metrics for QPM are:

- 80 gpm

- 80 psid, pressure rise (in NaOH)

- 5000 rpm

- 8.8 hp, in NaOH (effective shaft power)

- 5.00 inch impeller/rotor diameter

A cross section of this single-suction design is shown in Figure 14. Stator cooling was to be accomplished using seawater ducted from outside the vehicle. The sealing of the stator was accomplished by clamping a Teflon disk between the nonrotating shaft and the outer casing. Stainless steel was used for the rotor assembly with a welded cover plate sealing the magnets within a pocket. The impeller blades were machined to assure uniformity and precise replication of the design shape and a cover plate bolted to those blades. The pump casing and rear casing cover (incorporating the motor stator and cooling passages) were made from aluminum, with the casing being plated using electroless nickel. To reduce the front leakage flow, a floating ring design was used.
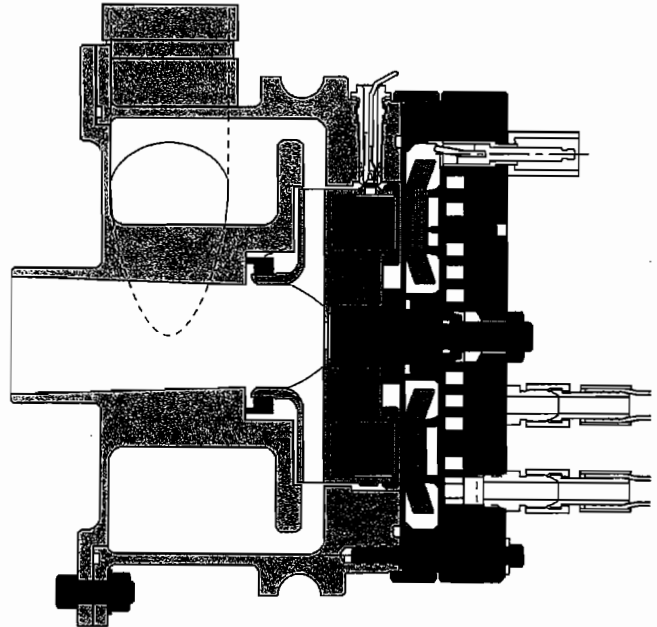


*Figure 14. QPM Cross-Section. (This pump is typical of the application of the SCAMP concept to a special purpose application.)*

The motor rotor and stator were specially designed to reduce their influence on fluidborne noise and structureborne vibration signatures. The inverter electronics were also specially selected and modified to comply with stringent noise specifications.

A photo of the pump, installed in a bulkhead housing of the underwater vehicle, is seen in Figure 15. The pump passed all mechanical and motor performance tests using water as a test medium. Due to cancellation of the program, the pump was never installed in a battery section for operation with NaOH. However, the system (pump, motor, and inverter) is in the hands of the US Navy for assessment of its noise signature and evaluation of the disk motor technology for other critical applications.

*SCAMP Prototype*

A commercially oriented single-suction pump prototype was developed to evaluate the use of the disk motors in pumps intended for use in an industrial environment. This SCAMP unit, while being similar in configuration to the earlier QPM, differed greatly in terms of disk motor design and construction features. While the QPM was considered a "cost-no-object" design to meet a critical specification, the SCAMP shown in Figure 16 was configured to be readily producible, maintainable, and to be operated using relatively common power electronics. The performance of this unit is as follows:
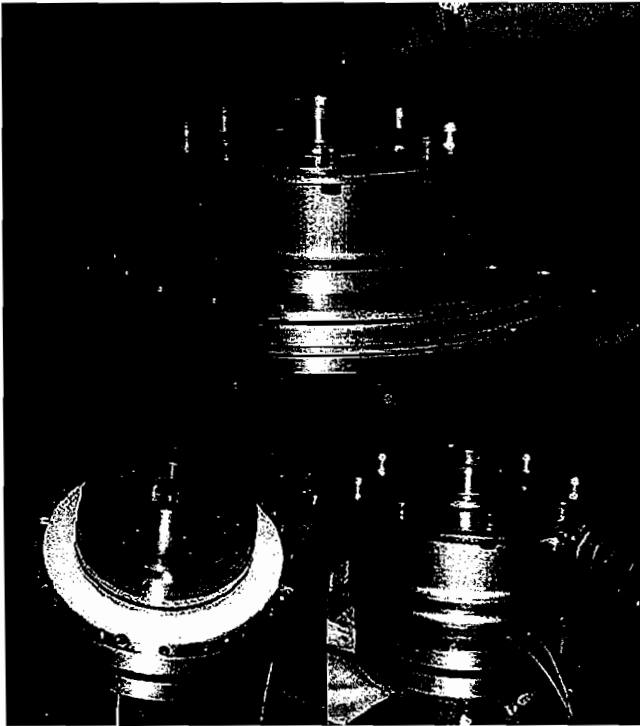
*Figure 15. Photograph of QPM Mounted in Vehicle Bulkhead. (The compact size and special mounting design used to position the pump are visible here.)*

- 60 gpm
- 120 ft TDH
- 5600 rpm
- 3.6 hp (effective shaft power)
- 4.10 inch impeller/rotor diameter

These prototypes were used to develop the cooling path design described earlier in this paper (for which a patent application has been filed). One prototype was specially outfitted with an axial load cell and used to characterize the axial thrust performance and verify satisfactory operation with the single thrust bearing approach. Other prototypes were used to test different bearing materials, although selection of bearing materials (such as SiC, polymers, or carbon graphite-based materials) follows standard sealless pump practice. One significant development goal was the design of a low cost disk motor stator. The resultant design employs simpler stator construction and coil winding, with only a small reduction in torque production as measured by the electro-magnetic shear stress (2.5 to 3.5, depending on the ultimate diameter of the disk motor). Finally, these prototypes were initially designed with hall sensors embedded in the stator core to provide rotor position information to the inverter controller. Successful use of the sensorless feedback control was implemented on these prototypes. This eliminates the need for costly and potentially unreliable sensors being located in the motor components, and ultimately inside the pump.

These prototypes have been demonstrated to the public and are serving as a tool for developing applications where SCAMPs unique features can be applied to meet user needs for sealless pumping machinery (Figure 17).

### Double-Suction Prototype

We have seen that utilization of disk motors is not restricted to the single-suction pump configuration. The double-suction concept of Cooper and Bulson (1992) offers axial balance of not only
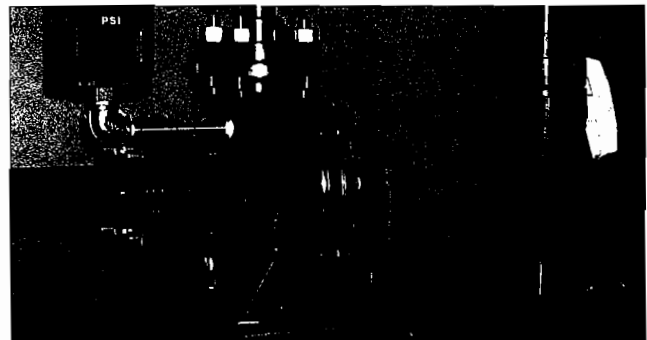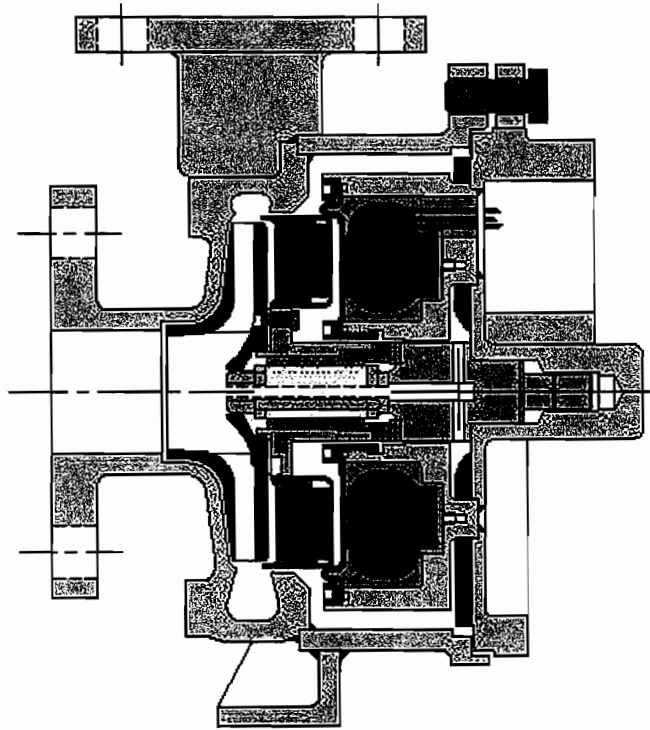




*Figure 16. SCAMP Prototype Cross-Section. (This prototype contains many of the significant design features that characterize the SCAMP concept.)*
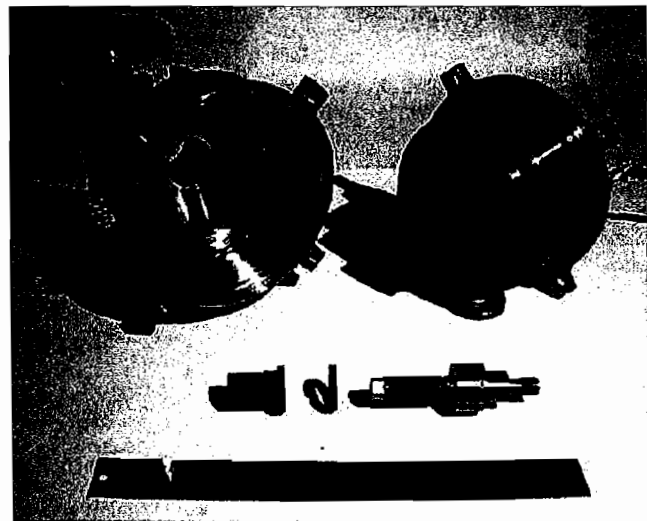


*Figure 17. SCAMP Prototype and Components. (This shows a disassembled SCAMP prototype with internal hardware (shaft, radial bearing, and axial bearing) laid out in the foreground.)*

hydraulic forces, but also the axial forces generated by the incorporation of permanent magnets in the rotor. Both impeller shrouds incorporate an identical complement of magnets. The stators are installed within a pair of stainless steel structures located within the pressure boundary of the pump. This concept was put into practice with a large 400 hp prototype. A Teflon cover plate was used to seal the front of each stator from the pumped fluid. Cooling was accomplished using the pumped fluid, passing behind the stator assemblies. In this case, the cooling fluid also served to provide lubrication to the product lubricated bearings. The twin disk motors double the available torque over the single-suction version, so that, for a given head generated by the impeller, about twice the flow can be delivered. The pump described below was constructed using a production impeller and pump casing (heavily modified to accommodate the motor stators). These elements were selected from different pumps to facilitate construction of this prototype, so the hydraulic performance should not be considered optimized.

- 3000 gpm
- 350 ft TDH
- 2450 rpm
- 400 hp (effective shaft power)
- 18.3 inch impeller/rotor diameter

A cross-section of this configuration is shown in Figure 18. A photograph of this pump, showing the modified production casing is shown in Figure 19. The mechanical design is unusual since there is ostensibly no driven shaft in the pump. The impeller/rotor is supported by a set of product lubricated bearings, which are positioned around what appears to be an elongated wear ring land. The inner diameter (ID) of the tubular extension directs the fluid from the suction bay to the impeller eye. The OD of this tube locates the thrust collar, radial bearing sleeve, lock nut, and the leakage control surface. The bearing design utilized tilting-pads for both radial and axial bearings. Pads were made of silicon carbide and ran against silicon carbide rotating surfaces. Tilting-pad bearings were selected over simpler journal bearings in order to prevent potential bearing instability arising from the expected light unit loads (due to the large bearing surface area dictated by the location of the bearing around the impeller eye). This unique configuration positioned the radial load directly between the short bearing span. The orientation of the axial bearings, being so close to the impeller and integrated with the radial bearings, also proved effective at reacting the residual axial force.

No evidence of bearing wear was observed throughout the development testing. No special filtering was used to protect the bearings; carbon steel piping and a carbon steel pump casing were used. Testing of the pump to minimum flow (about 10 percent of design flow) produced little increase in vibration or rotor motion, attesting to the robustness of the closely mounted bearing assemblies.

*Single-Suction, Explosion-Proof Prototype*

The fourth and most recent prototype pump uses a single 18.3 inch disk motor, identical in design to the motors used in the double-suction pump just described. A cross-section of this pump is shown in Figure 20. The performance of the pump is listed below:

- 1600 gpm
- 400 ft TDH
- 2800 rpm
- 240 hp (effective shaft power)
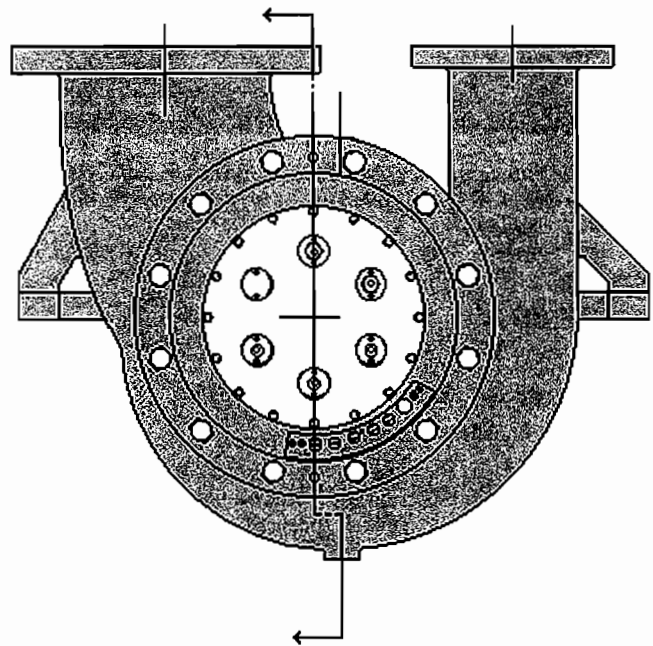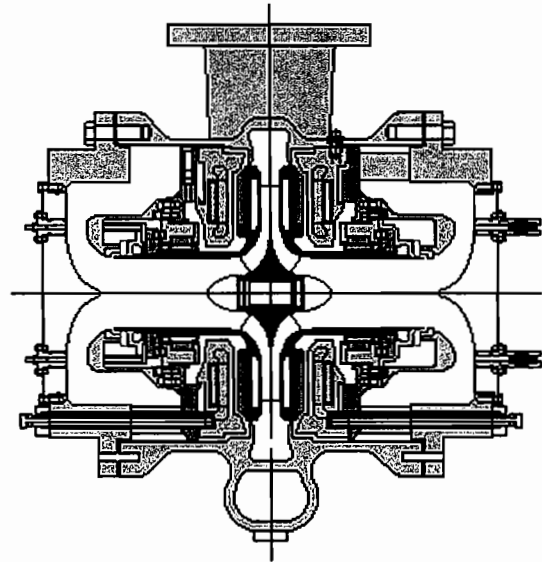- 18.3 inch motor rotor diameter
- 14.5 inch impeller diameter



*Figure 18. Double-Suction SCAMP Prototype Cross-Section. (This displays the design features incorporated into a radially split, single-stage, double-suction pump.)*

The pump is configured similarly to the small SCAMP prototype. However, there are two important differences. The impeller disk is considerably larger than the impeller. In order to extend the flow and power range of single-sided SCAMP machines, this practice is necessary. The extended disk causes additional disk friction power losses with a result of a loss of efficiency of five to 10 points. Also, the extended disk creates additional axial hydraulic thrust (over and above what one would expect from just an impeller without the motor disk). In order to compensate for this thrust (so that the magnetic attraction force would still control the rotor position on the shaft) the internal cooling flow and bearing flow arrangement were altered. In this case, the motor cooling does not return to suction through a hollow, stationary shaft, but instead flows from behind the stator, through the annulus formed by the ID of the stator housing and the OD of the shaft piloting arrangement. The flow then splits with one portion passing through the bearings and returning to suction, and
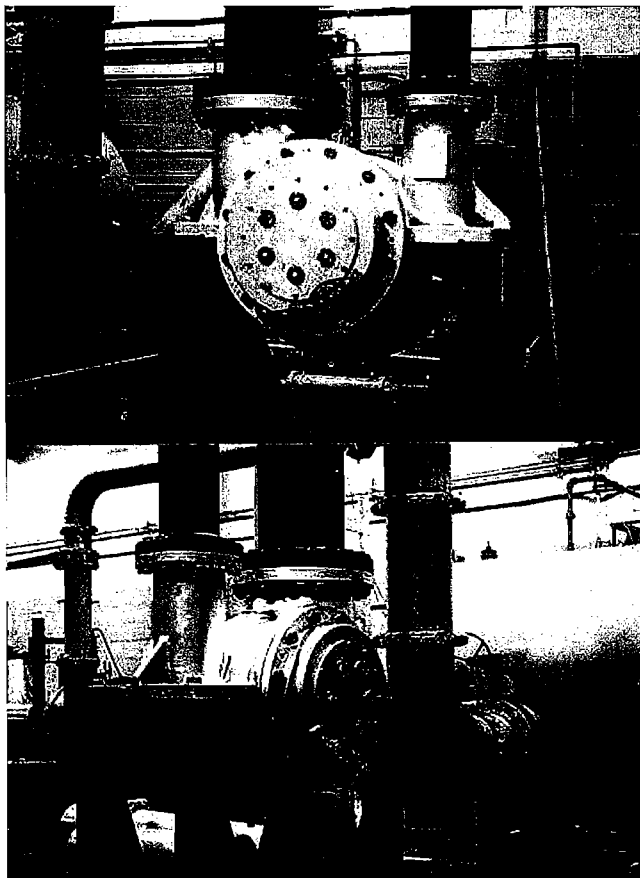
*Figure 19. Double-Suction, 400 HP SCAMP Prototype on Test. (This shows the prototype installed into a test loop. The absence of bearing housings, mechanical seals, shaft coupling, and external motor drive is apparent.)*
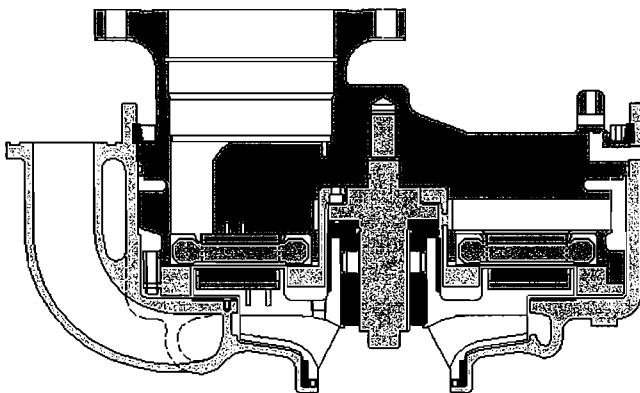


*Figure 20. 240 HP Super-SCAMP Cross-Section. (This pump is designed for submerged operation in hazardous environments.)*

the other portion returning to suction through balance holes. This redirection of the cooling flow is necessary to reduce the pressure behind the impeller and hence reduce the hydraulic thrust toward suction. This enables the single axial bearing and magnetic forces to work together to control the axial force on the rotor.

The other unique feature of this prototype is its explosion-proof design. A review of the governing regulations from various sources (National Electric Code, Factory Mutual, International Electrotechnical Commission) identified the governing classifications for integral motor pumps operating in a submerged tank. It is assumed that Class1/Group C and D gas groupings (typically ethylene and propane) are used for the evaluation. The motor/pump

design was analyzed as two separate but related areas; the stator and rotor cavities.

The stator cavity is classified as a division 2/zone 2 type area, meaning that flammable material may be present abnormally. The method of protection chosen for the stator cavity is explosion-proof. This means that in the event of an explosion, the structure is designed to prevent leakage of hot combustion gas or flame to the surrounding areas. The rotor cavity is classified as a division 1/zone 1 type area, meaning that flammable material is present intermittently. This area uses an increased safety approach that implies that no arcs, sparks, or hot surfaces will exist to ignite the vapor.

The rotor safety is achieved via design evaluation. The stator safety must be evaluated through actual testing, where an explosive gas mixture is placed in the cavity and ignited. The resulting pressure is measured and recorded. Next, a hydro-test, using a test liquid, is conducted on the cavity at a pressure four times that of the recorded explosion pressure. No leaks of the hydro-test fluid are allowed. The design of the structure to contain such an explosion and prevent deformation accounts for the somewhat massive size of the pump casing and rear cover (incorporating the stator cavity) seen in Figure 20. The requirement to contain the significant forces that could be generated by the explosion testing led to the use of a shear ring type enclosure to connect the rear cover to the pump casing. This type of enclosure is economical of space and material, and has a very large load carrying capability.

Photographs of this pump are shown in Figure 21. It is in a vertical orientation and is supported by a stand. The dual discharge volute requires a piping manifold to connect to the discharge piping.
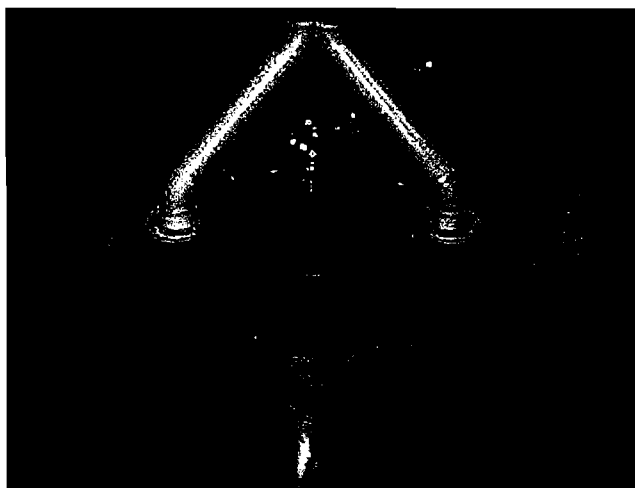


*Figure 21. Super-SCAMP on the Test Stand. (This prototype pump is adapted for operation in a closed test loop.)*

## CONCLUSIONS

Pumped fluids can be isolated from the environment in two ways: rely on mechanical shaft seals or resort to completely sealless pumps. Both methods have advantages and disadvantages with the final selection being based on issues of severity of application, cost, history of equipment reliability, and availability of suitable equipment. The sealless centrifugal axial-field motor pump (SCAMP) approach to sealless pump design takes a step toward improved sealless pump reliability and offers the user a new means to adapt the installed pumps to a specific or variable application requirement.

The close integration of the motor components within the traditional structure of the pump itself offers significant mechanical advantages that translate into better reliability, reduced parts count, and smaller, more compact pumping systems. The characteristics of the permanent magnet disk motors used in SCAMP machines allow pump engineers to design pumps for operation above the conventional 60 (or 50) cycle, two-pole motor speed. This implies that suction conditions become the limiting factor to operating at higher speed (and the higher hydraulic efficiencies come from this high-speed operation). Existing product lubricated bearing technology can be used for SCAMP with the operation of these bearings enhanced, due to the advantages of mechanical configuration and the operating characteristics of the variable speed, permanent magnet disk motor.

The commutation of this high power, compact, liquid-cooled motor is made possible through the use of an efficient polyphase inverter. This inverter (along with the attendant controller and AC/DC rectifier) provides infinitely variable operating speeds well over 3600 rpm (or 3000 rpm for 50 Hz duty). All the benefits of variable speed operation can be realized, such as the elimination of expensive control valves, optimum matching of hydraulic performance to process demands, avoiding the need to trim impellers to less efficient diameters, and soft start capability to reduce wear and tear on internal parts. The presence of the inverter and controller also provides a framework for inclusion of specific diagnostic and self-preservation features that help the operators prevent premature failure.

Presently, four prototypes have been built and tested. Each unit addresses some different design issue or application. Equivalent shaft power range from 3 hp to 400 hp configurations include both double-suction (using twin-disk motors) and single-suction pumps. Intended applications include hydrocarbon and chemical processing, noise-quiet underwater service, and explosion-proof service in a submerged tank. With the technology demonstrations accomplished, the SCAMP concept is being considered for industrial and military use, where its features can be applied to pumping applications to provide reliable and efficient service.

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