A USER'S ENGINEERING REVIEW OF SEALLESS PUMP DESIGN LIMITATIONS AND FEATURES

by

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Mr. Hernandez earned a B. S. degree in Mechanical Engineering from McNeese State University (1974). He is an ASME member and is active in the ASME South Texas Section Professional Development Seminar Programs. He is currently Professional Development Chairman of the ASME South Texas Section. In 1989, he received the J. George H. Thompson award for most outstanding, meritorious service to the ASME South Texas Section.

ABSTRACT

Exxon Chemical, like many other petrochemical companies and refineries, is announcing emission plans with a long term goal to achieve continuous emission reduction from the plant sites. Exxon Chemical's emission plan has reduced air emission by 90 percent since 1975, with work continuing to further reduce emissions over the next five years. These goals are being driven by company policy with concern for the environment, business competition, government regulations, and public concern over the past several years. Since many petrochemical and refinery employees, friends, and relatives live in areas where plants are located, they are just as concerned about safety and hygiene impact of emissions and waste disposal from the plants as the general public, environmental activist groups, and government agencies.

Due to the steadily increasing public concern over the environment, the government regulations are becoming more restrictive in leakage criteria. A good example of this is the current regulatory debate to the Clean Air Act, which would require industry to utilize maximum achievable control technology.

To reduce emissions, great improvements have been made over the past several years in pump mechanical seal design, evaluation, and auxiliary support systems. However, the mechanical seal design inherently requires a minute leakage across the faces for lubrication, along with costly auxiliary systems and environmental monitoring programs. Even with the improvements, mechanical seals remain the most fragile and weakest link of the pump; thus, they are the primary cause of pump failures [1] that can also contribute to emissions.

Mechanical seals typically fail as a result of many factors, such as process upsets, vibration problems, bearing failures, auxiliary support system failures, and problems caused from maintenance and operating personnel. Consequently, one solution to reducing pump failures due to mechanical seals and resulting emissions is to apply sealless pumps to more hazardous, toxic, or hydrocarbon services. For this reason, an engineering survey was launched to evaluate magnetic drive and canned motor pumps.

The engineering survey was limited to metallic, single stage, horizontal pump designs, similar to the American National Standards Institute, (ANSI B73.1 "Specification for Horizontal End Suction Centrifugal Pumps for Chemical Process"), and American Petroleum Institute (API 610 "Centrifugal Pumps for General Refinery Service") designs that are typically used in the petrochemical and refinery industries. The magnetic drive pump evaluated was of the couple design, because the author's experience with close couple design has been poor, and because the close couple design is rarely used in large petrochemical and refinery applications. The engineering survey addressed mechanical design features, performance ranges, horsepower limitations, materials of construction, manufacturing location, assembly and test location, delivery, available options, and most of all, questions about potential problem areas.

Like any piece of machinery, there are design limitations, as well as potential design deficiencies, that can lead to reliability problems that show up as increased maintenance cost and possible production losses. The advantages and deficiences in current magnetic drive and canned motor pump designs will be shown from a user's viewpoint. The views expressed are the author's opinions and not necessarily the company's views.

INTRODUCTION

The sealless pump has been around for many years, with the primary design being of the canned motor pump rather than the magnetic coupling design. The canned motor pumps have been used in many applications around Europe and the United States. Prior to the late 1970s [2], magnetic drive pumps were limited to the magnet material availability and power transmission capability with the magnetic coupling design. [3] With the rapid development of rare earth magnets after 1978 [2], the magnetic drive pump operating capabilities have greatly increased. Within the last year, magnetic drive pumps have become a competitive product for the canned motor pumps and will steadily increase in the near future, because more manufacturers are starting to design and produce the pumps. Although the magnetic drive pumps have been primarily used in Europe for many years, the pumps have recently been applied and used in pump applications in the United States beginning in mid 1980s, according to available manufacturer's experience lists.

Since the magnetic drive pump is a relative new product in the US market, the first step of the project was to locate manufacturers of magnetic drive pumps. The most logical starting point was to contact all the standard ANSI and API pump manufacturers in the US and to use a source, such as the Thomas Register For American Manufacturers, that lists manufacturers for various type pumps and other equipment. The pump types found were vane, gear, metering, screw, and centrifugal. Since the survey was limited to the metallic, single stage, horizontal pump design, several manufacturers were eliminated by this criteria. After the elimination, there were 14 pump manufacturers selected, of which 93 percent responded to the survey.

Locating the canned motor pump manufacturers was much easier, because of the author's previous experience with the canned motor pump design. According to the Thomas Register for American Manufacturers, there are many canned motor pump vendors, but only four of the major manufacturers were selected because of past experience with this type pump design. Only 75 percent of the manufacturers selected responded to the survey.

After locating the manufacturers, contact was made to obtain their standard brochures that led to a sales call to further discuss the best features of their pump designs. Some manufacturer's sales personnel were so enthusiastic and eager to sell the new magnetic drive pump that several stated their pumps were the answer to most all mechanical seal problems. As a result of the sales discussions, it was not long before several potential problem areas were uncovered. At this point, a detailed engineering survey was developed to determine the pump design, hydraulic and temperature limitations, mechanical design limitations, and questions to address the potential problem areas, refer to APPENDIX 1. The same survey and questions were also sent, at a later date, to three selected canned motor pump manufacturers, after a decision was made to include canned motor pumps in the engineering review.

During the early stages of the engineering review, the first item found to be confusing was the various nomenclature the vendors used in their brochures. To clarify the nomenclature

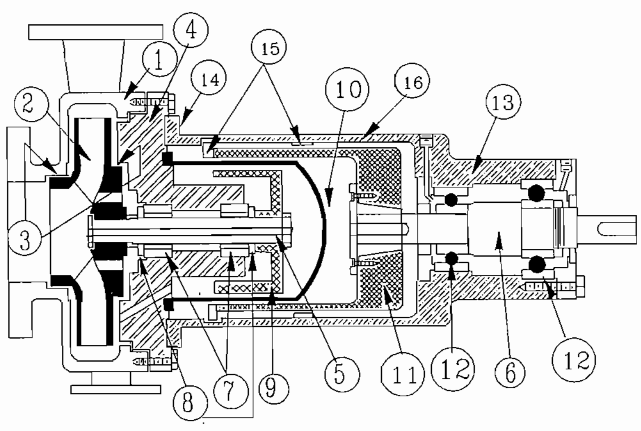

Figure 1. Magnetic Drive Pump. 1) Casing, 2) impeller, 3) wear ring, 4) back cover, 5) internal shaft, 6) external shaft, 7) radial sleeve bearing, 8) thrust bearing, 9) inner magnet rotor, 10) containment shell, 11) external magnet rotor, 12) antifriction bearing, 13) bearing housing, 14) transition flange, 15) safety rub ring, and 16) outer containment shell.

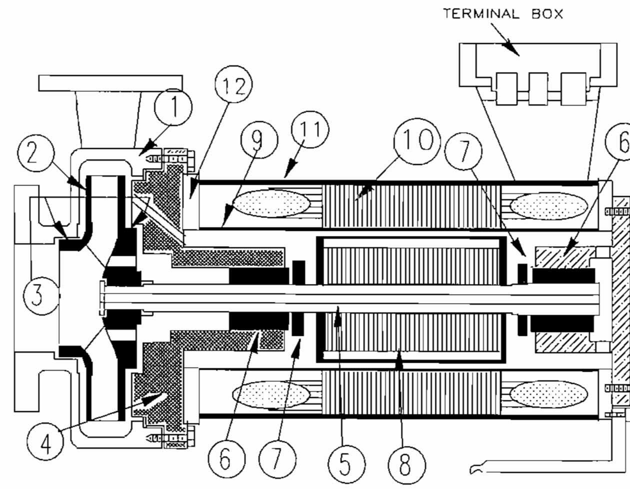

Figure 2. Canned Motor Pump. 1) Casing, 2) impeller, 3) wear ring, 4) back cover, 5) internal shaft, 6) radial sleeve bearing, 7) thrust bearing, 8) inner rotor, 9) inner can, 10) stator windings, 11) outer can, and 12) transition flange.

used in this paper, the pump components are identified as shown in Figures 1 and 2.

SYSTEM DESIGN CONSIDERATIONS

Anyone that has had any dealings with applying pumps knows the system design is vital to the pump operation and can have severe effects on the pump safety and reliability. The pump can either work properly or can possibly destroy itself. Since the system design is critical, the machinery engineer's predominant concern should be to understand the pump system design characteristics, fluid properties, and potential problems caused from low or high production unit rates. The process engineer should convey the critical process information to the machinery engineer by using a process data sheet, as shown in Figure 3. Once the system is understandable, the machinery engineer can then select the best pump for the application, along with conveying the information to the pump manufacturers by way of pump specification and data sheet (Figure 4). With the case of the sealless pump design, not only does the owner's machinery engineer need to understand the system design, but the pump manufacturer also needs to know how the pump is being applied. The author's experience with sealless pumps has been good in one petrochemical plant and poor in another plant. The primary reason for the different experience was because of the products being pumped along with the system design.

Since there are many books and papers published on pump system design, a few of the leading problem areas that will affect the sealless pump safety and reliability are discussed in this paper. The fluid properties such as specific gravity, vapor pressure, viscosity, and specific heat are some of the most important elements of the system design, because they affect the NPSH (net positive suction head) available, mechanical design, process emergency conditions, and bearing design, as in the case of the sealless pump. The fluid specific gravity and viscosity will affect the pump horsepower and head requirements. Furthermore, the lower the specific gravity comes the increase chance of the fluid flashing. For this reason, the vapor pressure rate of change, curve slope, is extremely critical in the operation of a sealless pump. The higher the vapor pressure rate of change, the more risk of the fluid flashing with a small increase

PUMPS

PROCESS DATA SHEET

UNIT:____ BY:____ DATE:____
APPROPRIATION:____ LAST REV. BY:____ DATE:____

GENERAL DATA			
Item			
Service			
Number Req'd			
PUMPING CONDITIONS			
Fluid			
Temp.,°F			
Sp. Gr. @ Temp.,°F			
Sp. Heat @ Temp.,°F			
Vis. @ Temp.,°F			
Max. Vis. @ Min. Temp.,°F			
Vapor Press., psia			
Solids			
Corrosive Materials			
Erosive Materials			
Chloride Concentration, PPM			
Particle Concentration, % by wt.			
Particle Size, Micron			
Start-up Fluid			
Min. Flow Rate, GPM @ °F			
Normal Flow Rate, GPM @ °F			
Design Flow Rate, GPM @ °F			
Design Suction Press., psig			
Min. NPSH Available, Ft.			
Design Discharge Press., psig			
Design Differential head, Ft.			
BHP @ Design Conditions			
CONSTRUCTION			
Pump Type			
Usage Factor			
Seal Type			
Materials - Case			
Impeller			
Trim			
Cooling Water Req'd			
EMERGENCY CONOITIONS			
Max. Suct. Press.,psig @ Temp.,°F			
Max. Sut. Temp., °F @ Press., psig			
Max. Disch. Press.,psig @ Temp,°F			
DRIVER			
Type			
Steam Cond. for Steam Turbine			

PROCESS PUMP OATA SHEET -2-

GENERAL DATA			
Item			
SYSTEM DESCRIPTION			
SUCTION VESSEL:			
OPEN TO ATMOSPHERE or CLOSED SYSTEM			
SUCTION VESSEL PRESSURE, PSIG			
PUMP LOCATION: BELOW OR ABOVE VESSEL			
SUCTION VESSEL HAVE LEVEL CONTROL:	YES NO	YES NO	YES NO
SUCTION VESSEL HAVE PRESSURE SENSOR:	YES NO	YES NO	YES NO
HOW IS SUCTION VESSEL PRESS. MAINTAINED? Press.			
IF FLUID LEVEL OR TANK PRESSURE OROPS TOO LOW, WILL SYSTEM AUTOMATICALLY STOP THE PUMP?	YES NO	YES NO	YES NO
COULD PUMP POSSIBLY RUN DRY?	YES NO	YES NO	YES NO
REMARKS:			
SITE AND UTILITY DATA			
LOCATION: INDOOR or OUTDOOR	IN. OUT.	IN. OUT.	IN. OUT.
HEATED or UNHEATED	YES NO	YES NO	YES NO
PARTIAL SIDES			
OTHER			
WINTERIZATION / TROPICALIZATION REQ'D			
ELECTRICAL AREA CLASS. CL / GR / DIV	/ /	/ /	/ /
UNUSUAL CONDITIONS: DUST or FUMES			

SITE DATA:
ELEVATION____FT. BAROMETER____PSIA
RANGE OF AMBIENT TEMP. MIN./MAX.____°F
RELATIVE HUMIDITY: MIN./MAX.____%
UTILITY CONDITIONS:
STEAM: DRIVERS HEATING
MIN.____PSIG____°F____PSIG____°F
MAX.____PSIG____°F____PSIG____°F
ELECTRICITY: DRIVERS HEATING CONTROL SHUTDOWN
VOLTAGE____ ____ ____ ____
HERTZ ____ ____ ____ ____
PHASE ____ ____ ____ ____
INSTRUMENT AIR, PSIG: MAX.____MIN.____
COOLING WATER: WATER SOURCE____
TEMP. INLET ____°F MAX. RETURN____°F
PRESS. NORM.____PSIG DESIGN____PSIG
MIN. RETURN ____PSIG MAX. ALLOW DELTA P____PSI

REMARKS

Figure 3. Sealless Pump Process Data Requirements.

in temperature; thus, greater risk of the fluid flashing in the bearings or around the internal rotor causing cooling and wear problems. Pumps where the fluid is operating near the vapor pressure are typically found on a distillation column, reflux service, or reboiler service. The fluid viscosity is also important because of the hydrodynamic or frictional losses from the internal flow passages. In the case with the sealless pump, high viscosity fluids affect the internal bearing lubrication, starting torque requirements, and drag on internal rotor. As a result of the increased drag on internal rotor and through the small passages, the increase in fluid viscosity greatly increases the hydrodynamic losses.

The fluid specific heat is another important parameter, because it is the ratio of the amount of heat transferred to raise a unit mass of fluid one degree, divided by the heat required to raise a unit mass water one degree at some specified temperature. Since mosts tests and calculations are based on water, the ratio is used as a correction for other type fluids. In another words, the specific heat is used to determine the fluid flow required to remove the heat generated by the bearings and heat produced by the magnetic coupling or motor windings. As pointed out, the fluid not only affects the hydraulic and mechanical component design, but also affects the materials of construction. When the pump manufacturer selects the material of construction, the material selection depends upon the operating stress and effects of corrosion, erosion, abrasion, temperature, entrained vapors, and solids content. Therefore, the fluid properties play a vital role in the system design.

As with most petrochemical and refinery industries, the piping system layout is ordinarily arranged by pipe designers that are directed by either a project engineer or process design engineer. In either case, the pipe designer's principal concerns are pipe routing, elevations, anchors, valve locations, and etc., for towers, drums, exchangers, and in some cases, machinery. As in most cases, the pipe designer has standards for piping layouts around control valves and fixed equipment, but very few pipe designers are aware of the piping requirements around machinery; at least, this has been the author's experience. It seems like the pipe designer always wants to install an elbow, a reducer, or a strainer at or near the pump suction flange, along with a reducer, check valve, and/or a block valve at or near the pump discharge flange. The predominant reason for this type piping layout is that the pipe designer is trying to conserve real estate and doesn't fully understand piping effects on machinery performance and reliability. Furthermore, by the time the pump manufacturer sends the installation manuals with the piping requirements, the pipe designer usually has the entire system designed, spool sheets completed, and pipe fabrication is either well underway or completed. Even if the machinery engineer and pump manufacturer properly apply the pump, a faulty system layout and installation can destroy the pump performance and reliability. In addition, more attention should be paid to suction piping layout because of the destructive effects on the pump performance as compared to the discharge piping [7]. The only effect the discharge piping has is to increase friction losses because of high velocities that transform into head and horsepower losses when the piping is too small, or to increase piping cost if the discharge pipe is oversized. To help resolve the problem, the manufacturer should place pipe installation requirements on the pump outline drawing as well as send the installation manual with the preliminary outline drawing. Based on the author's experience the following guidelines have been proven to be effective.

- Suction, discharge, and minimum flow piping should be symmetrical to avoid pump problems during startup and parallel operation.

SEALLESS -- MAGNETIC DRIVE & CANNED MOTOR
CENTRIFUGAL PUMP DATA SHEET - CUSTOMARY UNITS

JOB No.____ ITEM No.____
PURCHASE ORDER NO.____ DATE____
INQUIRY No.____ BY____
REV.____ DATE____

APPLICABLE TO: [] PROPOSAL [] PURCHASE ORDER [] AS BUILT
FOR____ UNIT____
SITE____ NO. REQ'D____
SERVICE____ PUMP SIZE,TYPE & STAGES____
MFG.____ MODEL____ S/N____

NOTE: O INDICATES INFORMATION COMPLETED BY PURCHASER [] BY MANUFACTURER [O] BY MANUFACTURER OR PURCHASER

GENERAL

PUMPS TO OPERATE IN PARALLEL/SERIES WITH____

	MOTOR	GEAR	TURBINE
NO. DRIVEN			
PUMP I/N			
DRIVER I/N			
PROVIDED BY			
MOUNTED BY			
DATA SHEET NO'S			

O OPERATING CONDITION
CAPACITY, GPM: NORMAL____ RATED____
OTHER____
SUCTION PRESS., PSIG: MAX____ RATED____
DISCHARGE PRESS., PSIG:____
DIFFERENTIAL PRESS., PSI:____
DIFFERENTIAL HEAD, FT.:____ NPSH AVAIL., FT.____
HYDRAULIC POWER, HP:____
SERVICE: O CONTINUOUS O INTERMITTENT (START/DAY____)

O SITE AND UTILITY DATA
LOCATION:
O INDOOR O HEATED O UNDER ROOF
O OUTDOOR O UNHEATED O PARTIAL SIDES
O GRADE O MEZZANINE O ____
O WINTERIZATION O TROPICALIZATION REQ'D.
O ELECTRICAL AREA CLASS. CL____ GR____ DIV____
SITE DATA:
O ELEVATION____ FT. BAROMETER____ PSIA
O RANGE OF AMBIENT TEMP. MIN./MAX.____ °F
O RELATIVE HUMIDITY: MIN./MAX.____ %
UNUSUAL CONDITIONS: O DUST O FUMES
O OTHER____
O UTILITY CONDITIONS:
STEAM: DRIVERS HEATING
MIN.____ PSIG____ °F____ PSIG____ °F
MAX.____ PSIG____ °F____ PSIG____ °F
ELECTRICITY: DRIVERS HEATING CONTROL SHUTDOWN
VOLTAGE____
HERTZ____
PHASE____
INSTRUMENT AIR, PSIG: MAX.____ MIN.____
REMARKS:____

O SITE AND UTILITY DATA (CONT'D)
COOLING WATER: WATER SOURCE____
TEMP. INLET____ °F MAX. RETURN____ °F
PRESS. NORM.____ PSIG DESIGN____ PSIG
MIN. RETURN____ PSIG MAX. ALLOW DELTA P____ PSI

O LIQUID
O TYPE OR NAME OF FLUID____
O PUMPING TEMP., °F NORM____ MAX____ MIN____
O SPECIFIC GRAVITY____ @ MAX. TEMP.
O VAPOR PRESSURE,PSIA____
O SPECIFIC HEAT____ CP (BTU/LB°F)
O VISCOSITY____ cP @____ °F
O MAX. VISCOSITY____ @ MIN. TEMP. °F____
O CORROSIVE/ EROSIVE AGENT____
O CHLORIDE CONCENTRATION (PPM)____
O H_2S CONCENTRATION (PPM)____
LIQUID: O TOXIC O FLAMMABLE O OTHER
PARTICLE CONCENTRATION, % BY WEIGHT____
PARTICLE SIZE:____ HARDNESS____

O SYSTEM DESCRIPTION
O SUCTION VESSEL: O OPEN TO ATMOSPHERE O CLOSED SYSTEM
O SUCTION VESSEL PRESSURE, PSIG____
O PUMP LOCATION: O BELOW VESSEL O ABOVE VESSEL
O SUCTION VESSEL HAVE LEVEL CONTROL: O YES O NO
O SUCTION VESSEL HAVE PRESSURE SENSOR: O YES O NO
O HOW IS SUCTION VESSEL PRESS. MAINTAINED____
O IF FLUID LEVEL OR TANK PRESSURE DROPS TOO LOW, WILL SYSTEM AUTOMATICALLY STOP THE PUMP? O YES O NO
O IS IT POSSIBLE THE PUMP COULD RUN DRY? O YES O NO
REMARKS:____

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SEALLESS -- MAGNETIC DRIVE & CANNED MOTOR
CENTRIFUGAL PUMP DATA SHEET - CUSTOMARY UNITS

JOB No.____ ITEM No.____
PURCHASE ORDER NO.____ DATE____
INQUIRY No.____ BY____
REV.____ DATE____

[] PERFORMANCE
[] RPM____ PROPOSAL CURVE No.____
[] IMPELLER DIA., IN.____ RATED____ MAX____ MIN
[] RATED POWER, BHP____ EFFICIENCY,%____
[] MIN. CONTINUOUS FLOW, GPM____ THERMAL____ STABLE
[] MAX. HEAD RATED IMPELLER____ FT.
[] MAX. POWER RATED IMPELLER____ BHP
[] NPSH REQUIRED AT RATE CAP.____ FT. H_2O
[] SUCTION SPECIFIC SPEED____
[] MAX. SOUND PRESS. LEVEL____
REMARKS:____

[] CONSTRUCTION
[] MAIN CONNECTIONS:

	SIZE	RATING	FACING	POSITION
SUCTION				
DISCHARGE				
BAL. DRUM				
OTHER				

[] OTHER CONNECTIONS

SERVICE	NO.	SIZE	TYPE/RATING	POSITION
VENT				
DRAIN				
PRESS. GAGE				
TEMP. GAGE				
WARM-UP				
OTHER				

[] CASING MOUNTING:
[] CENTERLINE [] NEAR CENTERLINE
[] FOOT [] SEPARATE MOUNTING PLATE
[] VERTICAL [] SUMP
[] IN-LINE
[] CASING SPLIT: [] AXIAL [] RADIAL
[] CASING TYPE:
[] SINGLE VOLUTE [] DOUBLE VOLUTE
[] BARREL [] DIFFUSER
[] STAGGERED VOLUTES [] VERTICAL DOUBLE CASING
[] IN-LINE
[] IMPELLER:____ THRUST BAL. MAX. SOLID SIZE____
[] CLOSE [] OPEN [] SEMI-CLOSE
[] BETWEEN BEARINGS [] OVERHUNG
O IMPELLERS INDIVIDUALLY SECURED
[] CASE AND CONTAINMENT SHELL PRESSURE RATING, PSIG:
O CASE AND CONTAINMENT SHELL DESIGNED FOR SAME
MAWP____ THICKNESS, IN.____
MAWP @ 60°F____ @ NORM. PUMP TEMP.____
HYDRO. TEST, PSIG____
REMARKS____

[] CONSTRUCTION (CONT'D)
[] ROTATION (VIEWED FROM COUPLING END)
[] CW [] CCW
REMARKS____
[] SHAFT DIAMETERS, IN.'S, AT
SLEEVE____ COUPLING____
RADIAL BRG.____ THRUST BRG.____
BETWEEN BRG'S____ IMPELLER____
[] SHAFT SPAN, IN.'S
BETWEEN BRG.'S____
RD. BRG. TO IMPELLER____
REMARKS____
[O] COUPLING: MAKE____ MODEL____
CPLG. RATING, HP/100 RPM____ SERVICE FACTOR____
SPACER LENGTH____ LIMITED END FLOAT REQ'D____
LUBRICATION____ DYNAMIC BALANCE AGMA CLASS____
DRIVER HALF MOUNTED BY:
O PUMP MFG. O DRIVER MFG. O PURCHASER
O COUPLING PER API 671
REMARKS____
[] MAGNETIC COUPLING: [] SYNCHRONOUS [] EDDY CURRENT
[] SOFT START REQ'D [] HARD START CAPABILITY
EDDY CURRENT LOSSES THRU SHELL____
HEAT GENERATION IN SHELL, BTU'S____
DECOUPLE TORQUE, LB-IN____ SAFETY FACTOR____
LOCK ROTOR TORQUE, LB-IN____

	OUTER	INNER
MOUNTING METHOD		
TEMP. LIMITATION, °F		
HERMETICALLY SEALED		
NO. OF MAGNETS		
TOTAL TORQUE, LB-IN.		

BEARINGS AND LUBRICATION
BEARING (PUMP DRIVER SHAFT): MFG/TYPE/ NUMBER/LIFE/LOAD, LBS.
RADIAL (CPLG. END)____
THRUST (IMPELLER END)____
LUBRICATION: OIL VISCOSITY, ISO GRADE____
[] GREASE [] FLOOD [] RING [] FLINGER [] PURGE OIL MIST
[] PURE OIL MIST [] CONSTANT LEVEL OILER [] PRESSURE
[] API 610 [] API 614 [O] OIL HEATER REQ'D(O ELECTRIC O STEAM)
[] OIL PRESS. GREATER THAN COOLANT PRESS.
BEARING (PUMP DRIVEN SHAFT): (INCHES)
RADIAL: DIA.____ LENGTH____ CLEARANCE____
NUMBER____ SPAN____ LOAD____ PSI OR LBS.
THRUST: TYPE____ END PLAY____ DIA.____
NUMBER____ LOAD____ PSI OR LBS.

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CENTRIFUGAL PUMP DATA SHEET - CUSTOMARY UNITS

JOB No.____ ITEM No.____
PURCHASE ORDER NO.____ DATE____
INQUIRY No.____ BY____
REV.____ DATE____

[] MATERIAL
[] BARREL/CASE____ IMPELLER____
[] WEAR RINGS: CASE____ IMPELLER____
[] SHAFT:DRIVE____ DRIVEN____
[] CONTAINMENT SHELL____
[] DRIVEN SHAFT RADIAL BRG.____
[] DRIVEN SHAFT BEARING JOURNAL____
[] DRIVEN SHAFT THRUST BRG.____
[] DRIVE SHAFT THRUST FACE____
[] DRIVER BEARING HOUSING____
[] ADAPTER OR TRANSITION PLATE____
[] WETTED FASTENERS____
[] COUPLING HUBS____ SPACER____
[] BASEPLATE____
[] VERTICAL LEVELING SCREWS____
[] HORIZONTAL POSITION SCREWS____
[] MAGNET MATERIAL:
OUTER____ INNER____
REMARKS:____

[] MOTOR DRIVE
[] MANUFACTURER____
[] HORSEPOWER____ [] RPM____
[] HORIZONTAL [] VERTICAL
[] FRAME____ ENCLOSURE____
O TYPE____ SERVICE FACTOR____
O VOLTS/PHASE/HERTZ____
O EXPLOSIONPROOF "T" RATING____
O MINIMUM STARTING VOLTAGE,%____ TEMP. RISE____
[] LOCK ROTOR AMPS____ FULL LOAD AMPS____
[] INSULATION____
[] STARTING METHOD____
[] BEARINGS: TYPE____ MFG:____ NUMBER____
[] LUBE____
O 2 X THRUST RATING____
O SHAFT: SOLID____ HOLLOW____
[] VERTICAL THRUST CAPACITY, LBS: UP____ DOWN____
REMARKS____

[] INSTRUMENTATION
[] VIBRATION:
O ACCELEROMETER O SEISMIC O NONCONTACTING
O PROVISION FOR MOUNTING ONLY O API 671 DATA SHT.
REMARKS:____
[] TEMPERATURE AND PRESSURE
O TEMP. GAUGES O THERMOWELLS O PRESS. GAGES
O PROVISIONS FOR INSTRUMENTS ONLY
O CONTAINMENT SHELL TEMP. PROBES ONLY
O CONTAINMENT SHELL TEMP. PROBES & INST. READOUT
O OTHER____
REMARKS:(MFG. & MODEL)____
O ADAPTER LIQUID OR LEAK DETECTOR (MFG. & MODEL)____
O MOTOR LOAD SENSORS (MFG. & MODEL)____
O DIFFERENTIAL PRESS. SWITCH (MFG. & MODEL)____
O FLOW DETECTORS (MFG. & MODEL)____
O SECONDARY CONTAINMENT SHELL (THICKNESS, DESIGN, CAPACITY, ETC.)____
REMARKS____

COOLING WATER & JACKET PIPING
O COOLING WATER REQ'D O STEAM JACKET REQ'D
O OTHER____
O SIGHT FLOW INDICATORS O MANIFOLD OUTLET VALVE
O GALVANIZED PIPE SCH'D____ O COPPER TUBING
O STAINLESS STEEL TUBING SIZE & WALL____
[] COOLING WATER REQUIREMENTS: FLOW, GPM____ @PSIG____
[] STEAM REQUIREMENTS: FLOW, SCFM____ @PSIG & TEMP. °F____
REMARKS:____

SURFACE PREPARATION AND PAINT
O MANUFACTURER'S STD.
O OTHER____
PUMP:
O SURFACE PREPARATION SSPC-SP____
O PRIMER____ FINISH COAT____
BASEPLATE:
O SURFACE PREPARATION SSPC-SP____
O PRIMER____ FINISH COAT____
GROUTING REQ'D O YES O NO
O SURFACE PREPARATION SSPC-SP____
O PRIMER____
REMARKS:____
SHIPMENT:
O DOMESTIC O EXPORT O EXPORT BOXING REQ'D
O OUTDOOR STORAGE MORE THAN 6 MONTHS
SPARE ROTOR ASSEMBLY AND SPARE PARTS
O HORIZONTAL STORAGE O VERTICAL STORAGE LBS.
O TYPE OF SHIPPING PREPARATION____
REMARKS:____

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SEALESS -- MAGNETIC DRIVE & CANNED MOTOR
CENTRIFUGAL PUMP DATA SHEET - CUSTOMARY UNITS

JOB No.____ ITEM No.____
PURCHASE ORDER NO.____ DATE____
INQUIRY No.____ BY____
REV.____ DATE____

[] WEIGHTS
MOTOR DRIVEN PUMP:
[] WEIGHT OF PUMP, LBS.____
[] WEIGHT OF BASEPLATE, LBS.____
[] WEIGHT OF MOTOR, LBS.____
[] WEIGHT OF GEAR, LBS.____
[] TOTAL WEIGHT, LBS.____
TURBINE DRIVEN PUMP:
[] WEIGHT OF PUMP, LBS.____
[] WEIGHT OF BASEPLATE, LBS.____
[] WEIGHT OF TURBINE, LBS.____
[] WEIGHT OF GEAR, LBS.____
[] TOTAL WEIGHT, LBS.____
REMARKS____

O OTHER PURCHASER REQUIREMENTS
O COORDINATION MEETING REQ'D.____
O REVIEW FOUNDATION DRAWINGS
O REVIEW PIPING DRAWINGS
O OBSERVE PIPING CHECKS
O OBSERVE INITIAL ALIGNMENT CHECK
O CHECK ALIGNMENT AT OPERATING TEMPERATURE
O ROTOR BALANCED DURING ASSEMBLY OF EACH ELEMENT
O VENDOR DEMONSTRATION OF MAX. ALLOWABLE VIBRATION AT MINIMUM FLOW TO MAX. DESIGN FLOW
O LATERAL RESPONSE ANALYSIS REQUIRED
O PUMP ONLY O ALL EQUIPMENT
O CRITICAL SPEED ANALYSIS
O STIFFNESS MAP OF UNDAMPED ROTOR
O TORSIONAL ANALYSIS
O PROGRESS REPORTS REQUIRED
O OTHER____
REMARKS:____

QA INSPECTION & TEST
O REVIEW OF VENDORS QA PROGRAM____
O PERFORMANCE CURVE APPROVAL
O MATERIAL CERTIFICATION REQUIRED
O CASING O IMPELLER O SHAFT
O CASTING REPAIR PROCEDURE APPROVAL REQ'D
O INSPECTION REQUIRED FOR NOZZLE WELDS
O MAGNETIC PARTICLE O LIQUID PENETRANT
O RADIOGRAPHIC O ULTRASONIC
REMARKS:____

[] QA INSPECTION & TEST (CONT.)
O CHARPY IMPACT TEST REQUIRED FOR____
O INSPECTION REQ'D FOR____
O MAGNETIC PARTICLE O LIQUID PENETRANT
O RADIOGRAPHIC O ULTRASONIC
O HARDNESS TEST REQ'D FOR____
O SURFACTANT HYDROTEST
O VENDOR SUBMIT TEST PROCEDURES
O RECORD FINAL ASSEMBLY RUNNING CLEARANCES
O VENDOR MAINTAIN RECORDS FOR 5 YEARS
O SHOP INSPECTIONS

TEST	NON-WIT	WIT	OBSERVE
HYDROSTATIC	O	O	O
PERFORMANCE	O	O	O
NPSH	O	O	O
COMPLETE UNIT TEST	O	O	O
SOUND LEVEL TEST	O	O	O
O DISMANTLE & INSP. AFTER TEST	O	O	O
O CLEANLINESS PRIOR TO FINAL ASSEMBLY	O	O	O
O PIPE LOAD TEST	O	O	O
O ____	O	O	O
O ____	O	O	O
O ____	O	O	O
O ____	O	O	O
O ____	O	O	O
O ____	O	O	O
O ____	O	O	O

APPLICABLE STANDARDS
O API 610 O ANSI B 73.1 & B73.2
O OTHER____
O VENDOR UNIT RESPONSIBILITY
O GOVERNING SPECIFICATION (IF DIFFERENT)____
REMARKS:____

SPARE PARTS
O START-UP O RECONDITIONING O CRITICAL SERVICE
O SPECIFY____
REMARKS:____

Figure 4. Sealless Pump Specification Data Sheets.

• Suction line should have a minimum of five straight pipe diameters to the first obstruction (elbow, reducer, valve, etc.), while the discharge should have a minimum of three straight pipe diameters.

• Piping and transitions should be designed to prevent vapor pockets or entrain vapor. An eccentric reducer should be used for making transition when pipe size changes, one pipe size smaller, near pump inlets.

• All process piping flanges should be parallel within 1/16 in.

• Maximum horizontal offset should not exceed 1/8 in on all process piping flanges.

• Local gauges should be installed upstream and downstream of suction strainer (pressure only along with one temperature), as well as the discharge (pressure only).

• Flow measurement devices should be located in the discharge line. In addition, the minimum flow recirculation lines should have local flow measurement capabilities.

• In a rare case, the pipe inside diameter to pump nozzle alignment should be checked.

Not only is the piping layout critical, but one item overlooked many times is the suction piping entrance nozzle design and location. If the entrance nozzle is not designed correctly, the pump will have poor performance, particularly when the pumping fluid is in equilibrium, such as columns and towers found in most cases in the petrochemical and refinery industries. When suction is taken from the bottom or side of a vessel, tank, or tower tray, sufficient liquid height or static head above the entrance nozzle must be provided to account for entrance and velocity head losses at the nozzle, plus any submergence that is required to prevent vortices from entering the suction line. Vortices can be controlled by decreasing nozzle velocities or increasing submergence, as shown in Figure 5 [7]. Submergence required to control vortices can be reduced by using anti swirl devices, commonly know as vortex breakers, or using baffles to eliminate the swirl, as shown in Figure 6 [6].

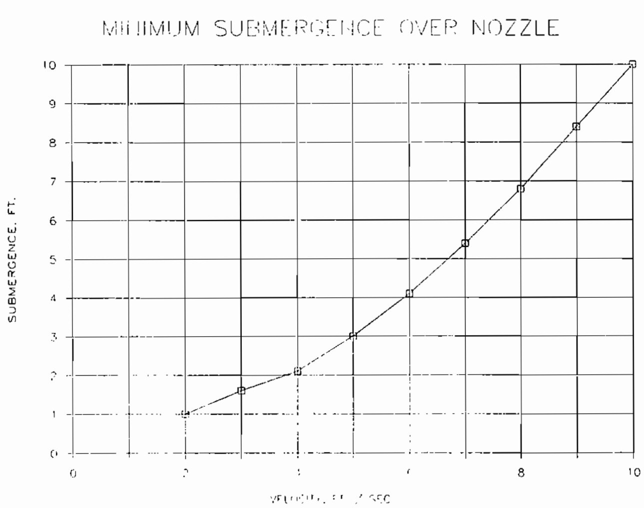


Figure 5. Submergence Vs Nozzle Velocity.

Even though numerous papers and pump books have been published on net positive suction head (NPSH), a great percentage of pump problems can be traced to NPSH related problems. NPSH is critical in any pump application, but much more vital to the sealless pump because of the internal bearing design, fluid lubrication, and cooling requirements. Furthermore, the process design engineer who normally sets the NPSH available

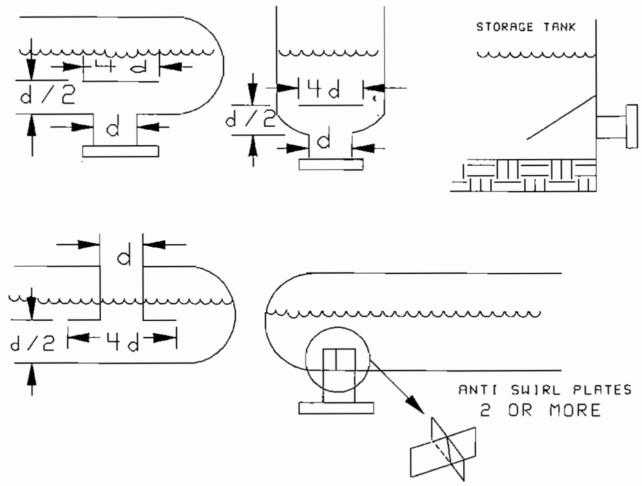


Figure 6. Antiswirl Devices.

for a pump application may not understand the implications or penalties for NPSH shortage. The penalties show up as cavitation, bearing damage, impeller damage, or other pump problems. NPSH available is frequently defined as the total suction head in feet of liquid above the liquid vapor pressure, measured at the centerline of the pump impeller eye, less friction losses in suction pipe and entrance nozzle. NPSH available can be calculated by:

$$\text{NPSH (A)} = H_a \pm H_s - H_v - H_f$$

NPSH (A) = Net Positive Suction Head in feed of liquid

H_a = Absolute pressure in feet at the liquid service

H_s = Static height in feet the liquid level is above or below the pump centerline or impeller eye

H_v = Liquid vapor pressure in feet at pumping temperature

H_f = All suction line and entrance losses in feet

The NPSH available should be calculated at the maximum pump capacity, minimum vessel level, suction piping fouling, and flow ranges along with any fluctuating suction pressures [5]. The limits should be specified on the pump data sheet, provided information is available from the process design engineer. The NPSH available should then be compared to the pump NPSH requirements. NPSH margin is consider minimal when the difference between NPSH available and required is three feet or less. When the NPSH is marginal, a test should be run to verify the pump actual NPSH requirement, because manufacturer's curves seem to show lower NPSH than the actual test results. On a new installation, the NPSH available should exceed the pump NPSH requirement by a minimum of five feet or more in order to prevent the fluid from flashing in the suction eye of the impeller.

Since the sealless pump's weakest design link is the internal bearings, the pump may experience damage ranging from minor to a possible catastrophic failure if the NPSH available is not adequate or if the suction vessel runs dry. On occasions, process problems occur, and the suction vessel liquid level gets dangerously low or becomes empty; hence, the NPSH available decreases until pump damage occurs. Although standard ANSI or API pumps are not designed to run dry, the pumps may experience bearing damage, wear ring galling, or other mechanical damage. Should a sealless pump run dry, the pump would experience similar problems, but most likely, major damage to the containment shell and internal bearings would occur. Major damage to a sealless pump is very expensive, and in some cases, the pump cannot be repaired and has to be replaced. As a minimum, the suction vessel should have level control protection in order to protect the pump from running dry.

MECHANICAL DESIGN

Centrifugal pumps used in petrochemical and refinery industries in the US generally comply with one of the two industry standards, ANSI B73.1 or API 610. The ANSI pump specifications are basically for light and medium duty pumps with dimensional interchangeability for handling corrosive, hazardous, or toxic chemicals distinctive in the chemical industry. The API 610 is a high quality, heavy duty pump designed for pressures, flows, safety, and reliability for typical rugged duty service. Mainly, API 610 pumps are specified when the fluid is flammable or toxic with a specific gravity of less than 0.7 at pumping temperature, pumping temperature exceeds 300°F, or discharge pressure exceeds 300 psig.

Since magnetic drive pumps are primarily produced in Europe or Japan, many of the pumps are designed and fabricated to other country standards, such as DIN 24256 (Germany), NFE 44121 (France), and ISO 2858 (International Standards Organization), but most are similar to the ANSI B73.1 (United States).

Due to the design of the canned motor pumps, this type pump is designed and fabricated to manufacturer's standards rather than the ANSI or API specifications. The canned motor pump should be considered a specialty engineered type product, because the motor and pump are combined into a single piece of equipment.

According to the engineering survey (Figure 7), many of the pumps available in the US conform to the ANSI flange size, but not necessarily to the ANSI envelope or dimensions. Only 33 percent of the manufacturers responding to the survey have pumps conforming to the ANSI envelope. Furthermore, 83 percent do employ the back pull out feature commonly found with ANSI and API pumps, while most (67 percent) pumps utilized the self venting design with the top, center mounted discharge flange.

PUMP PARTS	PUMP MANUFACTURER % MATERIAL AVAILABILITY										
MAGNETIC ORIVE	CS	CI	OI	304SS	304L	316SS	316L	AL-20	HAST.	MONEL	TITANIUM
CASING	58	0	25	8	8	67	8	50	42	8	8
IMPELLER	33	25	0	8	8	67	8	50	42	8	8
WETTED FASTENERS	0	0	0	15	8	67	8	23	8	8	8
CONTAINMENT SHELL	0	0	0	0	0	42	23	0	54	0	0
INTERNAL SHAFT	8	0	0	8	8	42	8	33	17	8	8
EXTERNAL SHAFT	67	0	0	0	0	0	D	0	0	0	0
BEARING HOUSING	17	42	33	0	0	0	0	0	0	0	0
TRANSITION FLANGE	33	17	17	0	0	8	0	0	0	0	0
CANNEO MOTOR	CS		DI	304SS	304L	316SS	316L	AL-20	HAST.	MONEL	TITANIUM
CASING	25		0	0	25	75	50	50	25	50	0
IMPELLER	0		0	50	25	75	0	50	25	50	0
WETTEO FASTENERS	0		0	50	0	75	0	50	25	50	0
INNER CAN	0		0	0	0	0	25	0	75	0	0
INTERNAL SHAFT	0		0	50	25	75	0	50	25	50	0
OUTER CAN	75		0	0	0	0	0	0	0	0	0

Figure 7. Mechanical Design Features.

Although the canned motor pumps do not comply with ANSI or API specifications, the pump design commonly uses ANSI size flanges and utilizes the self venting design feature, as found with the magnetic drive pumps.

Since the magnetic drive and canned motor pumps are designed to handle corrosive, hazardous, or toxic chemicals, the pumps are available in many types of materials, depending upon the manufacturer. The casing, impeller, and shaft for magnetic drive pumps are available in carbon steel, ductile iron, 316 SS, 316 L, 304 SS, alloy 20, Hastelloy B, and Hastelloy C, along with some exotic materials such as monel and titanium, depending upon the manufacturer (Figure 8). Should the pump mechanical design require wear rings (Figure 7), the material available is nearly always the same, or at least compatible to, the casing and impeller material. Likewise, the wetted end fasteners are normally 316 SS or the same alloy as the casing and impeller.

FEATURE	PUMP MANUFACTURER % MAGNETIC DRIVE	CANNEO MOTOR
IMPELLER		
SEMI-OPEN IMPELLER	8	25
SEMI-CLOSE IMPELLER.	8	0
CLOSE IMPELLER	75	75
OVERHUNG	100	75
THRUST BALANCE		
WEAR RING TYPE OESIGN	50	75
BALANCE HOLES	58	75
BACK IMPELLER RIB	17	0
CASING		
SELF VENTING	67	75
BACK PULL OUT	83	75
ANSI ENVELOPE	33	0
ANSI FLANGE TO FLANGE	50	50
ANSI FOOTPRINT	33	0
ANSI FLANGES	67	75
HYOROSTATIC TEST @ 1.5 X MAWP	83	75

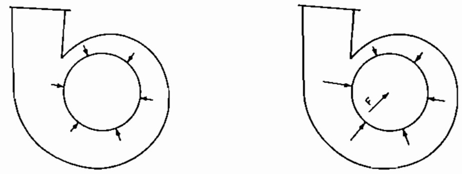
Figure 8. Pump Material Availability and Mechanical Design Features.

Canned motor pump material availability is the same as the magnetic drive, with the exception of ductile iron and titanium (Figure 8).

The bearing housing material of construction for a magnetic drive pump is available in cast iron, ductile iron, and carbon steel, depending upon the manufacturer. However, bearing housing transition flange to casing is available in carbon steel only from 17 percent of the manufacturers responding to the survey. The outer magnetic rotor shaft is available in carbon steels such as AISI (American Iron and Steel Institute) 1040 or 4140 grades.

Magnetic drive and canned motor pump casings serve two purposes like the standard ANSI and API: 1) to convert fluid velocity into pressure, and 2) to contain fluid pressure. The pump casing uses a volute to convert the fluid velocity imparted by the impeller into pressure. A volute is the spiral like shape that collects the discharge fluid from the impeller. As the volute areas increase at a rate proportional to the rate of discharge liquid for the impeller, a constant velocity around the impeller periphery is produced. The fluid velocity is then diffused in the casing nozzle to produce pressure (Figure 9) [8]. Due to the volute spiral shape, a radial load is produced perpendicular to the shaft, 90 degrees from the cut water, at 50 percent and 120 percent of best efficiency point (BEP) flow [6], on the impeller because of the uneven pressure distribution around the impeller. The greatest radial load is at shutoff, highest pump pressure, and decreases as the flow increase and near the minimum load at BEP flow [6]. The radial load increases again in the opposite direction from BEP flow to pump full flow [6]. When the pump is operating near the end of the curve, the radial load then changes directions and begins to increase because of the uneven pressure distribution. Since radial load can vary as the process changes, the internal bearings have to absorb or transfer the radial loads to the bearing housing. The radial load for a single volute can be calculated as shown in Figure 9 [8]. Most magnetic drive pumps use a single volute because of the relative small size. At this time, no magnetic drive pumps use a double volute. The advantage of the double volute is to significantly reduce impeller radial loads; therefore, bearing life is improved. The double volute does not completely hydraulic balance the radial loads because of casting imperfections and uneven casting surfaces [6].

On the canned motor pumps, a concentric casing (Figure 10) is used to reduce radial impeller loads, even though the concentric casing is less efficient than a single volute casing.

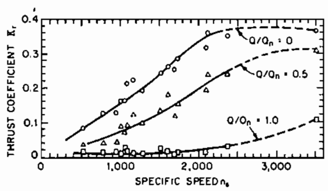


Uniform casing pressures exist at design capacity, resulting in zero radial reaction.

At reduced capacities uniform pressures do not exist in a single-volute casing, resulting in a radial reaction *F*.

$F_r = k\,K_r\,(Sp.\ Gr.)\,H\,D_2\,b2$

where
F_r = Radial load, pounds
k = 0.433
K_r = Experimentally determined coefficient
Sp. Gr. = Specific Gravity of fluid
H = Pump head, Ft.
D_2 = Impeller outside diameter, in.
b_2 = Impeller discharge width, including shrouds, in.



K_r as a function of specific speed and capacity for single-volute pumps (to obtain n_s in SI units, multiply by 0.6123).

Figure 9. Volute Casing" Radial Impeller Load.

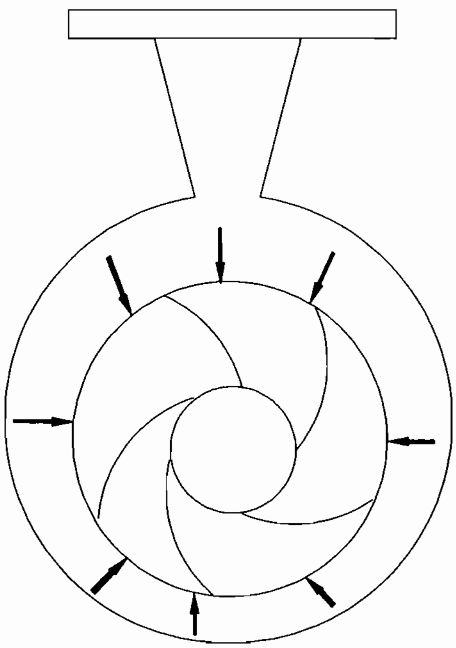


Figure 10. Concentric Casing.

Since the pump casing has to contain the process fluid pressure plus the increased fluid pressure from the impeller, the ANSI and API pump casings are designed to the ASME Section VIII Pressure Vessel code stress values. To ensure the casing integrity, the manufacturer's hydrostatic test the casing routinely at 1.5 times the maximum allowable working pressure. The maximum allowable working pressure most commonly is defined in API 610 as the maximum discharge pressure plus allowance for head and speed increases caused by impeller or driver replacements. Also, the maximum allowable working pressure could be the process emergency conditions. Should the latter be the case, the emergency conditions, pressure and temperature, need to be conveyed to the pump manufacturer in order for the casing to meet the higher design requirements than the standard manufacturer's design requirements. As a result of the engineering survey, it was found that only 33 percent of the manufacturers responding to the survey design the casing to the ASME Section VIII stress values.

From a user's view, the preferred casing design is with the back cover bolting separate from the bearing housing transition flange, shown in Figure 11, and designed in accordance with ASME Section VIII code stress values. Furthermore, the preferred material for the bearing housing is carbon steel, due to ease of repairs and reduced chance of cracking as compared to the other material, regardless of case bolting design and nozzle loading.

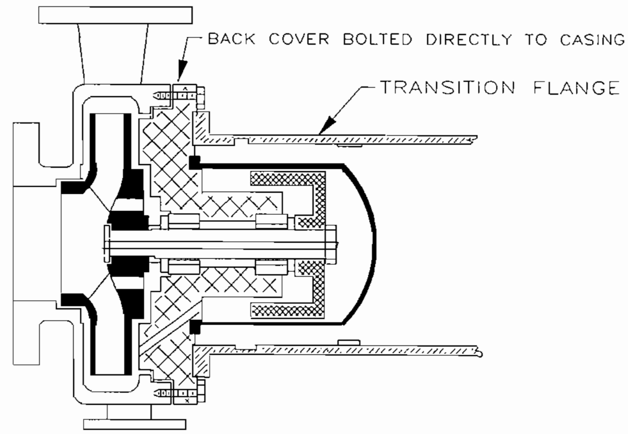


Figure 11. Casing" Preferred Design.

HYDRAULIC DESIGN

Currently, the hydraulic coverage of the sealless pump is somewhat limited; in the magnetic drive pump design to the magnetic coupling size and torque capability, and in the canned motor pump design to the motor design, along with the stator and rotor size. At this time, the performance range is shown in Figure 12. The performance range, head, flow, and horsepower are functions of the specific gravity, viscosity, temperature, and impeller design. Limitation in specific gravity is due to the hydraulic performance of the impeller, head, and horsepower requirements. According to the engineering survey, the maximum specific gravity is 3.0 and the lowest is 0.3, depending upon the manufacturer. The average range was from 2.0 to 0.5. When the sealless pump is applied to specific gravity ranges below 0.7, special consideration should be given to case design and internal fluid flow, pressures, and temperatures, for reasons described in other sections of this paper.

On the other hand, the viscosity limitation is due to the hydrodynamic losses that affect the torque or horsepower requirement, bearing lubrication and cooling to remove heat from electric motor, or magnetic coupling losses. According to the survey, the maximum viscosity limits vary with pump manufacturer and are shown in Figure 12.

The temperature limitation is dependent upon material stress levels in pressure containing parts, bearing material and lubrication, magnet material, and gasket material. As an outcome of the survey, the temperature limit varied from 750°F to -80°F. On the 750°F temperature, the manufacturer uses an eddy current type magnetic coupling. Also, one manufacturer with a synchronous type magnetic coupling with samarium cobalt magnet material, a special grade, had a temperature limit of

MAGNETIC DRIVE	SMALL SIZE RANGE				
MAX. HORSEPOWER, BHP	7.5	15	20 TO 25	30	36
MAX HEAD, FT.	20 TO 160	20 TO 300	13 TO 320	35 TO 300	15 TO 600
MAX FLOW, GPM	3 TO 100	2 TO 280	2 TO 300	6 TO 350	3 TO 600
SPECIFIC GRAVITY	0.5 TO 2.0	0.5 TO 2.0	-20 TO 482	-70 TO 700	-100 TO 750
TEMPERATURE, °F	-100 TO 750	-100 TO 750	0.5 TO 2.0	0.1 TO 3.0	0.1 TO 300
VISCOSITY, cp	0.1 TO 100	0.1 TO 100	0.4 TO 600	0.3 TO 300	0.3 TO 2.0
SPEED, RPM	1750 TO 3500	1750 TO 3500	900 TO 3500	1200 TO 3500	1750 TO 3500
NOZZLE SIZE					
SUCTION, INCHES	1 TO 2	3 TO 2	1-1/2 TO 3	1-1/2 TO 8	3 TO 6
DISCHARGE, INCHES	1 TO 1-1/2	1-1/2 TO 3	1-1/2 TO 2	1 TO 6	1-1/2 TO 3

CANNED MOTOR					
MAX. HORSEPOWER, BHP	7.5	15	20	30	36
MAX HEAD, FT.	90		110	450	425
MAX FLOW, GPM	55		400	300	400
SPECIFIC GRAVITY	0.3 TO 2.0			0.3 TO 2.0	
TEMPERATURE, °F	-150 TO 850			-150 TO 850	
VISCOSITY, cp	0.1 TO 150			0.1 TO 150	

MAGNETIC DRIVE	MIDDLE SIZE RANGE				
MAX. HORSEPOWER, 8HP	40	50	60	70	75
MAX. HEAD, FT.	15 TO 295	15 TO 425	64 TO 440	120 TO 175	100 TO 604
MAX. FLOW, GPM	5 TO 500	20 TO 1200	20 TO 625	225 TO 290	95 TO 1600
SPECIFIC GRAVITY	0.5 TO 2.0	0.5 TO 2.0	-76 TO 700	-40 TO 400	-20 TO 700
TEMPERATURE, °F	-80 TO 500	-20 TO 482	0.1 TO 300	0.1 TO 1.1	0.1 TO 300
VISCOSITY, cp	0.1 TO 100	0.1 TO 100	0.3 TO 300	0.3 TO 300	0.3 TO 2.0
SPEED, RPM	1750 TO 3500	1750 TO 3500	1750 TO 3500	1750 TO 3500	1750 TO 3500
NOZZLE SIZE					
SUCTION, INCHES	1 TO 1-1/2	3 TO 6	1-1/2 TO 5	3	3 TO 8
DISCHARGE, INCHES	2 TO 3	2 TO 4	1 TO 4	2	2 TO 6

CANNED MOTOR				
MAX. HORSEPOWER, BHP	40	50		70
MAX. HEAD, FT.	120	600		440
MAX. FLOW, GPM	1300	1200		450
SPECIFIC GRAVITY		0.3 TO 2.0		
TEMPERATURE, °F		-150 TO 850		
VISCOSITY, cp		0.1 TO 150		

MAGNETIC DRIVE	LARGER SIZE RANGE		
MAX. HORSEPOWER, BHP	100	140	150
MAX. HEAD, FT.	14 TO 680	10 TO 475	65 TO 700
MAX. FLOW, GPM	30 TO 1600	20 TO 4000	90 TO 4500
SPECIFIC GRAVITY	0.3 TO 3.0	0.5 TO 2.0	-80 TO 400
TEMPERATURE, °F	-80 TO 700	-185 TO 840	0.1 TO 200
VISCOSITY, cp	0.1 TO 300	0.1 TO 100	0.3 TO 300
SPEED, RPM	1200 TO 1800	900 TO 3500	1750 TO 3500
NOZZLE SIZE			
SUCTION, INCHES	4 TO 8	1-1/2 TO 12	4 TO 8
DISCHARGE, INCHES	3 TO 6	1 TO 10	3 TO 8

CANNED MOTOR	
MAX. HORSEPOWER, 8HP	120
MAX. HEAD, FT.	280
MAX. FLOW, GPM	1100
SPECIFIC GRAVITY	
TEMPERATURE, °F	
VISCOSITY, cp	

Figure 12. Sealless Pump Performance Range.

700°F. Most other manufacturers with same type coupling and samarium cobalt magnet material had temperature limits ranging from 300°F to 500°F. The maximum useful temperature for the samarium cobalt material is 500°F.

Canned motor pump temperature limits range from -150°F to 850°F, but require special cooling systems at temperatures above 650°F, depending upon pump manufacturer.

As illustrated by the survey, the owner's machinery engineer needs to understand and review the temperature limitations when applying sealless pumps. When fluid temperature approaches or exceeds 500°F, special design considerations are required to keep the magnet temperature below the useful operating temperature levels. Some manufacturers are capable of operating the magnetic drive pump with a fluid temperature above 500°F by using special techniques to control the temperature in the magnet area.

The present sealless pump designs, both magnetic drive and canned motor, use three types of impellers, such as the semiopen, semiclosed, and closed. The semiopen and semiclosed impeller designs are commonly found in the standard ANSI pump design, but only eight percent of the manufacturers use the semiopen design, while eight percent offer the semiclosed design in sealless pumps. Even though some semiopen design impellers are found in standard API pumps, the closed impeller is more commonly found (75 percent) in the API pumps, as well as most sealless pumps, both magnetic drive and canned motor. The advantage of the closed impeller design over the semiopen design is reduced axial loads and generally higher performance and efficiency.

In the case of the sealless pump, the impeller affects the performance, head, flow, horsepower, and efficiency, and also influences the thrust bearing design and internal cooling, which is covered in another section of this paper. The thrust bearing capacity is dependent upon the impeller design because of the pressure gradient on the front side, suction eye side, and the back cover of the impeller. A semiopen impeller has a different pressure gradient on the front side and back cover as compared to a closed impeller. Besides the impeller types, other features, such as ribs on back cover or back wear rings on closed impeller, also changes the pressure gradient. Impeller design and pressure gradient and thrust direction are shown in Figure 13 [6 and 8]. From the survey findings, the pressure near the hub on a closed impeller is approximately 60 percent of discharge pressure as compared to a closed impeller with a back wear ring and balance holes is approximately 5.0 to 10 psi above suction pressure [5]. The back rib feature would reduce the pressure at the hub from about 60 percent to 25 percent or less of the discharge pressure. When the pressure is reduced on the back cover of the impeller, the reduction in pressure reduces the thrust load and decreases the thrust bearing size.

On the magnetic drive pump, the magnetic coupling is sometimes used to offset the impeller thrust load. One manufacturer has vanes on the inner magnetic coupling rotor that increase the pressure in the containment shell. With the increased pressure and the thrust bearing acting as an orifice, the internal rotor is thrust balanced, and the rotor is free to float [3]. Most other manufacturers do not use this balancing feature and rely strictly on impeller balance holes (58 percent), and/or case wear ring design style (50 percent). Only one manufacturer has a special patented thrust bearing capable of handling the thrust load from a semiopen impeller.

Since canned motor pumps also use a closed impeller, the thrust balance is very similar to the magnetic drive pump, but some canned motor pump manufacturers also use special features to balance thrust loads. One canned motor pump manufacturer uses a set of fixed orifices on the front and back side of the impeller, where wear rings are typically located on a closed impeller. The impeller also has balance holes. As the impeller thrusts, in either direction, one orifice is increased while the other is decreased; thus, increasing pressure on one side of the impeller that centers the impeller. Another canned motor pump manufacturer allows the rotor assembly to rotate on a stationary hollow mandrel where orifices are drilled in the rotor shaft and mandrel. As the impeller thrusts, in either direction, the pressure behind the motor rotor increases or decreases because of the alignment of the orifice holes in rotor and the hollow stationary mandrel; hence, the rotor is self centering.

As a result of the survey, the NPSH requirements and efficiencies are very similar to the standard ANSI and API pump impeller designs currently being used. A caution one needs to be aware of is, when the NPSH requirement during actual testing is higher than quoted, some manufacturers will polish the impeller suction eye area, between vanes in entrance area, and casing in order to improve the NPSH requirement. If the NPSH margin between available and required is five feet or less, the manufacturer should not modify the impeller or casing, because the user could experience problems later in the field when an impeller is replaced. Some manufacturers claim to assign a new part number to the modified impeller, but the problem arises as to how much material was removed and at what locations. The manufacturer should not modify a standard impeller or casing, under any circumstances, without the owner's machinery engineer approval, because minimum process changes could be made to accept the slightly higher NPSH required impeller. If process changes are not acceptable to alleviate the NPSH available problem, an inducer should be installed rather than modify a standard impeller and casing. It is up to the owner's machin-

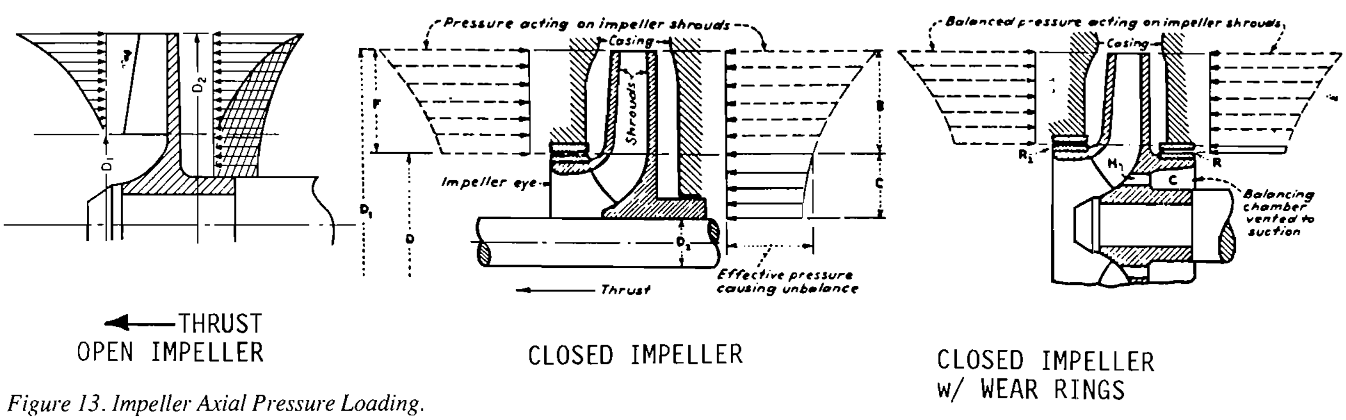


Figure 13. Impeller Axial Pressure Loading.

ery engineer to make sure the impeller modifications are properly documented in all necessary manuals, part lists, and other equipment records to prevent future problems.

From a user's viewpoint, the information on impeller axial load balance, impeller radial load, and thrust and journal bearing design should be requested as part of the pump specification. During the bid evaluation, the impeller load capacities should be compared to other manufacturers.

CONTAINMENT SHELL OR CAN DESIGN

Containment shell, or can design parameters, are influenced by the pump working pressure and temperature, internal rotor bearing support system, magnetic coupling eddy current losses, and motor electrical losses. Depending upon the containment shell design requirements on magnetic drive pumps, the shell is available in several materials (Figure 7), such as 316 stainless steel, Hastelloy, ceramic material, and polyetheretherketone, commonly known as PEEK, depending upon manufacturer. Canned motor pump materials (Figure 7) are available in 316 stainless steel and Hastelloy, depending upon manufacturer.

A word of caution on using 304 SS or 316 SS in a service where chlorides are present because of the possibility of stress corrosion cracking. Since the can or containment shells are relatively thin with residual stresses due to forming, the containment shell could experience stress corrosion cracking, depending upon stress level, operating temperatures, and the chloride concentration.

As a result of the engineering survey, there was a variety of different features offered by the pump manufacturers. As shown in Figure 14, some of the features, along with the percentage of manufacturers incorporating the features into the canned motor and magnetic drive pump design, are displayed.

PARAMETERS	PUMP MANUFACTURER % MAGNETIC DRIVE	CANNED MOTOR
ASME SECTION VIII DESIGN (STRESS LEVEL)	33	50
HYDROSTATIC TEST @ 1.5 X MAWP	67	75
SELF VENTING	67	50
SELF DRAINING	58	50
BEARING SUPPORT MOUNTED ON CONTAINMENT SHELL	23	N/A
SECONDARY CONTAINMENT SHELL OPTION	0	75
INTERNAL FLOW -- DISCHARGE TO SUCTION	67	75
DISCHARGE TO AREA NEAR DISCHARGE	17	0

MAGNETIC DRIVE PUMPS		CANNED MOTOR PUMPS	
CONTAINMENT SHELL THICKNESS, INCHES	MAXIMUM ALLOWABLE WORKING PRESSURE, PSI	CAN THICKNESS, INCHES	MAXIMUM ALLOWABLE WORKING PRESSURE, PSI
0.029	----	0.015	150 (300 w/sleeves)
0.040	150 TO 232	0.025	300
0.047	275		
0.050	350		
0.052	300		
0.057	275		
0.060	250-300		
0.067	300		
0.078	754		
0.093	500		
0.198	500		

Figure 14. Containment and Can Design Features.

From a user's view, the preferred containment shell or can design should incorporate the ASME Section VIII code stress values with a minimum 0.015 in corrosion allowance, self draining, self venting, no longitudinal welds, and no bearing support attached to containment shell because of added stress induced from vibration. These features are selected to improve safety, improve pump reliability, and reduce the possibility of emissions and personnel exposure to the fluid. Furthermore, in some toxic or hazardous services, the secondary containment shell is also recommended on the magnetic drive pump design. Since the canned motor pump has an outer shell, the outer shell should be capable of acting as a secondary containment shell and should also have the motor leads sealed for reasons covered in a later section.

CONTAINMENT SHELL OR CAN INTERNAL FLOW PATH

As an outcome of the engineering survey, most magnetic drive pump manufacturers (67 percent) have internal flow route from discharge back to suction, while other magnetic drive pump manufacturers (17 percent) use the internal flow routing from discharge back to another area at or near the pump discharge pressure. Besides these two commonly found flow paths, there are many special design configurations using an external fluid source for cooling and lubrication. Since there are several techniques that can be used to apply an external fluid source, this section will deal with the two commonly found flow paths rather than the special configurations. Any special requirements should be reviewed with the pump manufacturer to determine which is the best solution for the application.

The first and most common flow path being used on magnetic drive pumps is taking a slip stream from the pump discharge and returning the fluid to the suction after the fluid has removed the heat from the bearings and magnetic coupling in the containment shell, as shown in Figure 15. Each pump manufacturer has a slightly different flow pattern through the containment shell on magnetic drive pumps. In brief, the flow path originates from the impeller exit area through an orifice or hole in the casing back cover. On pumps where the magnetic coupling overhangs the internal bearings, the flow passes through the face of the rear bearing, through the bore of the rear bearing, through the bore of the front bearing, through the balance holes in the impeller, and thus back into the suction eye of the pump. On some pumps, the fluid enters between the bearings, and the stream splits where one goes through the rear bearing, and the remaining fluid goes down the center of the shaft to the rear area of the containment shell. Another fluid stream passes between the inner rotor and the containment shell and through a

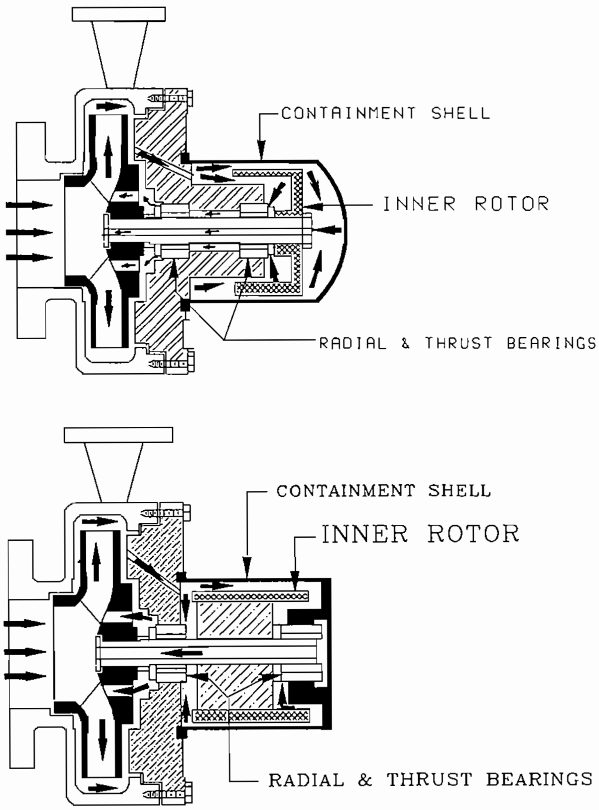


Figure 15. Magnetic Drive Pump Containment Shell Internal Flow Path Discharge to Suction.

hole down the center of the pump shaft, and thus, back to the suction eye of the pump.

On pumps where the magnetic coupling is between the bearings, the flow originates in the same way as described above, but the flow passes through the face of the front bearing, through the front bearing bore, through the impeller balance holes, and thus back to the impeller suction eye. Another stream passes between the inner magnetic coupling and the containment shell, at which time the stream may split, depending upon the pump manufacturer, where all or part of the flow passes through the face of the rear bearing, through the rear bearing bore, and enters a hole in the end of the pump shaft where the fluid flows back to the suction eye of the impeller. If all of the fluid does not pass through the rear bearing, a stream flows through holes in the outside diameter of the shaft to a hole in the center of the shaft where the fluid is returned to the impeller suction eye.

The second type flow path, shown in Figure 16, being used on magnetic drive pumps, is taking a slip stream from the pump discharge and returning the fluid to the rear of the impeller hub that is a percentage of discharge pressure, or returning it to an area near full discharge pressure after the fluid has removed the heat from the bearings and magnetic coupling in the containment shell. The flow path originates in the same manner as described above. Since the magnetic coupling overhangs the internal bearings, the flow passes between the inner magnetic coupling rotor and containment shell, through a hole in the inner rotor shaft, through the face of the rear bearing, through the bore of the rear bearing, through the bore of the front bearing, to an area near the impeller hub. On another design pump, some fluid passes though the front bearing bore and mixes with the slip stream fluid. Then the stream enters holes in the outside diameter of the shaft to a hole in the center of the shaft and flows to the rear area of the containment shell, where vanes on the magnetic coupling are used to increase the fluid pressure and return the fluid back to discharge. Once the fluid pressure is increased, some fluid passes through the face of the rear bearing, through the rear bearing bore, and meets the incoming fluid.

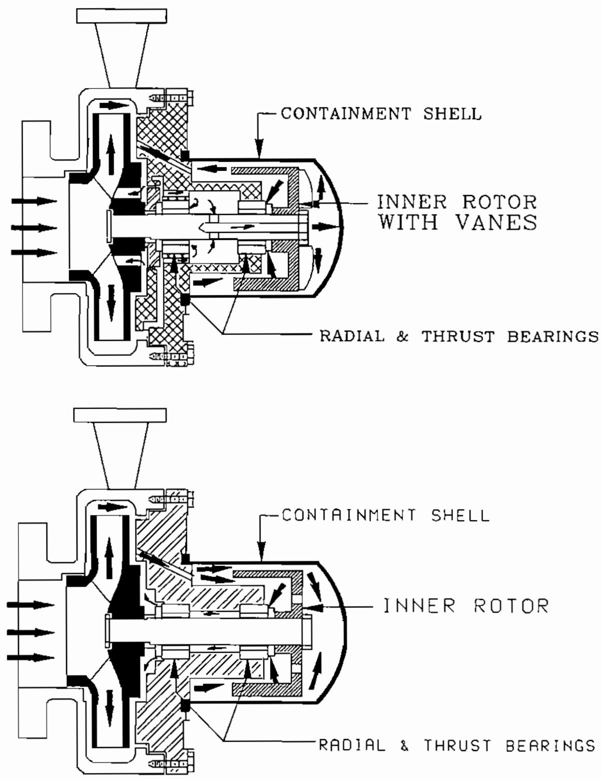


Figure 16. Magnetic Drive Pump Containment Shell and Can Internal Flow Path Discharge to Discharge.

On canned motor pumps (Figure 17), the flow path being used is taking a slip stream from the pump discharge and returning the fluid to the suction, after the fluid has removed the heat from the bearings and motor rotor in the can. In brief, the flow path originates from the impeller exit area through an orifice or front bearing area behind the impeller. The flow splits, and some fluid passes through the face of the front bearing, through the front bearing bore, and through balance holes in the impeller, thus back to the suction eye of the impeller. The remaining fluid passes between the motor rotor and stator can, where the fluid stream splits again. The fluid passes through the rear bearing face, through the rear bearing bore to a hole in the end of the shaft, while the remaining fluid flows through holes or orifices, depending upon the pump manufacturer, to the hole in the end of the shaft. All fluid that enters through the end of the shaft is returned to the suction eye of the impeller.

As illustrated above, each manufacturer has a slightly different internal flow path through the canned motor and magnetic drive pump designs. From a user's view, the primary concerns with the fluid flow path are whether or not dead spot areas for solids to settle out in the flow path are present, and the pressure and temperature margin from the fluid vapor pressure at various points along the flow route. Either flow path could be acceptable, provided the fluid is not encroaching on the fluid vapor pressure curve, solids cannot settle out, and ferric oxide (rust) or other magnetic particles do not build up on the inner magnetic

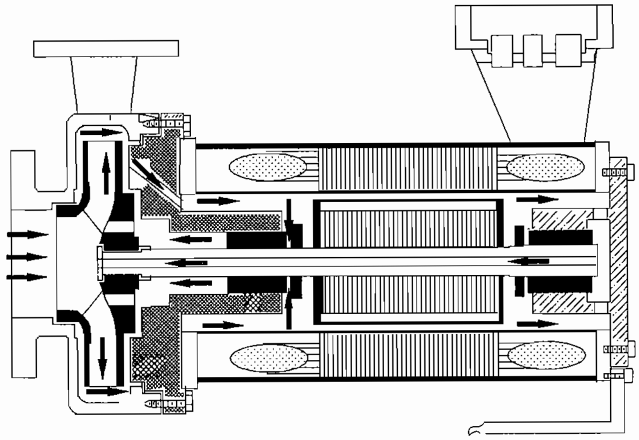
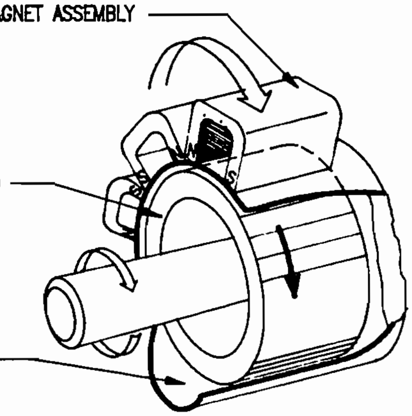
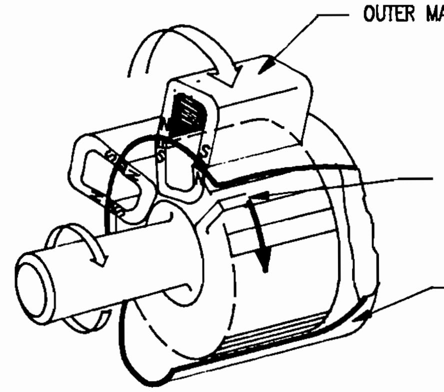


Figure 17. Canned Motor Pump Internal Flow Path.



EDDY CURRENT TYPE



SYNCHRONOUS TYPE

Figure 18. Magnetic Coupling Types.

coupling rotor. Should the fluid approach the vapor pressure along the flow path, the fluid could vaporize and cause problems with removing the heat generated by the magnetic coupling losses, affect the bearing cooling and lubrication, and affect the NPSH requirements of the impeller. Any or all of these problems could be disastrous for the pump. Since the internal flow is vital to a reliable sealless pump application, the user should require the pump manufacturer to supply the flowrate, pressure, and temperature at the critical areas along the flow route, such as bearing faces and bores, magnetic coupling rotor, fluid entrance and exit points, and between rotor and containment shell or can. The internal flow should be reviewed as part of the bid evaluation and compared to other manufacturers to determine which pump has the best operating fluid vapor pressure margin.

MAGNETIC COUPLING DESIGN

According to the engineering survey, all manufacturers offer the synchronous drive magnetic coupling, magnet to magnet, as compared to the eddy current (eight percent) magnetic coupling. In each coupling, the outer magnets are mounted in a housing that is attached to a drive shaft. The outer magnets on the drive shaft are then direct coupled to a standard induction type electric motor. The outer rotor with the magnets rotate in the ambient environment, while the inner rotor with the magnets rotate in the process fluid.

An eddy current magnetic coupling, shown in Figure 18 [7], works on a similar principle as the standard induction motor, but uses rare earth magnets to produce a magnetic field rather than an electric current. The magnets are mounted with a high temperature adhesive in a rotor that is directly coupled to a standard electric motor, while copper bars parallel to the shaft are mounted and encapsulated with a nonmagnetic sheath on the inner rotor. As the outer magnet assembly rotates, a rotating magnetic field is established and passes through the air gap, through the containment shell, through the nonmagnetic sheath, through the copper bars and rotor material beneath the torque ring, and returns to the magnet to complete the circuit. When the rotating magnetic field produces eddy currents in the copper bars, the eddy currents develop electromagnets which follow the rotating magnetic field; therefore, producing torque. The larger the slip between the outer magnet and the inner electromagnet and copper bars, the larger the eddy currents flowing; thus, developing greater torque [7].

A synchronous drive magnetic coupling (Figure 18) [7], uses rare earth magnets on the outer and inner rotor. The magnets in the outer are mounted in the same manner as with the eddy current type coupling. Magnets on the inner rotor are mounted with a high temperature adhesive and are encapsulated in a nonmagnetic sheath, depending upon manufacturer. Since both the inner and outer rotor contain the same number of magnets that determine the torque transmissibility, the inner rotor rotates in synchronism with the outer rotor that is directly coupled to a standard electric motor.

Eddy current magnetic coupling advantages are there are no magnets exposed to process temperature, no magnets to attract ferric oxide particles in process fluid, cannot decouple, and higher slip capability for high starting torque applications. On the other hand, synchronous magnetic coupling advantages are constant torque for low input speed applications, maximum power required in minimum dimensional constraint, maximum efficiency at design flowrate, and slightly more efficiency than the eddy current coupling design [2].

In either magnetic coupling design, there are two commonly found rare earth magnets that the pump manufacturers have selected. The rare earth magnets are neodymium iron boron and samarium cobalt. Neodymium iron boron magnets have a stronger magnetic field strength than the samarium cobalt, but the maximum temperature limitation is 150°C (300°F). Because increased temperature affects the demagnetization, the neodymium iron boron torque rating needs to be reduced at a rate of 0.09-0.120 percent per °C. Samarium cobalt has a temperature limitation of 350°C (500°F), and the torque derating needs to be reduced by 0.03-0.045 percent per °C [9]. Depending upon the magnet material, grade, and magnet manufacturer, the temperature limitation and derating factor may vary between pump manufacturers, and these parameters should be addressed during the pump selection and evaluation. In addition to temperature affecting the torque rating, the rare earth magnet loses magnetic field strength with age at a rate of one to one and one-half percent per year, according to one pump manufacturer.

Regardless of which magnetic coupling is applied, the chief concern is the magnetic field problem with a metallic barrier that creates eddy currents in the conductive material [9]. When the induced eddy currents establish a magnetic field, the induced field is subtractive from the initial field. As the magnetic field is reduced, heat is produced and torque is reduced; consequently, efficiency is reduced. As with the synchronous magnetic coupling design, the induced eddy currents produce heat very rapidly when the inner and outer magnets decouple. The heat can rise to high levels in a matter of one to three minutes, depending upon magnetic coupling size. Since there are many parameters that affect the magnetic coupling design, some are listed below [9].

- Coupling capability decreases with the inverse square of the magnet to magnet gap.
- Losses increase with the square of speed.
- Losses increase with square of increase in drive radius.
- Losses are inverse by proportional to containment shell resistivity.
- Additional losses from induced eddy current magnetic field.

The user's leading concern is not necessarily which type coupling to use, but rather the magnet material limitation, torque derating due to temperature and age, and decoupling safety factor with the synchronous magnetic coupling design. Furthermore, the inner magnet material must be compatible with process fluid, or preferably hermetically sealed magnets to protect the magnet material from being chemically attacked by the fluid. The preferred decoupling safety factor should be a minimum of 1.5 over maximum driver horsepower with maximum impeller for operation of entire pump curve. Typical safety factors shown from the survey ranged from 1.1 to 1.5 with some as high as 1.8 to 2.0. In some cases, the decoupling safety factor may have to be based on maximum horsepower for rated impeller, with a percentage allowance for future uprate capabilities. The latter case may be necessary to avoid oversizing the magnetic coupling that could be very inefficient at rated impeller; consequently, expending excessive heat into the cooling fluid passing through the containment shell that could affect the NPSH requirements on some magnetic drive pump designs.

From a user's viewpoint, there is a safety hazard with separating or assembling the magnetic couplings over 5.0 hp. When the maintenance personnel are separating or assembling the magnetic coupling, a person's finger(s) could get caught between the transition flange and the casing; hence, causing a minor hand injury, such as lacerations, or even a major hand injury, such as a crushed or missing limb, finger(s), with the larger horsepower magnetic couplings. To alleviate this safety hazard, jack screws, a minimum of three at 120 degrees apart, should be installed on the transition flange. The jack screws will help separate the magnetic coupling, or during assembly, the jack screws can be used to ease the magnetic coupling together very gently. By having jack screws, the magnetic coupling will not cause the transition flange to impact the rear casing and possibly causing a hand injury.

CANNED MOTOR PUMP DESIGN

The canned motor pump differs from the magnetic drive pump in the motor design. The magnetic drive pump utilizes a standard induction type electric motor to drive a magnetic coupling, where as the canned motor uses an unconventional, special design electric motor. The motor stator and rotor are isolated from the process fluid by thin metal cans of corrosion resistant nonmagnetic material. Design and construction of the cans are the core of the canned motor pump and require extremely high quality fabrication and fitting.

Characteristic of the canned motor pump (Figure 2), the rotor have sleeves, 0.015 to 0.125 in thick, shrunk onto the outside diameter of the rotor, with end covers welded to the sleeve and the shaft, depending upon pump manufacturer. Prior to welding, the rotor is preheated to more than 300°F for several hours to remove any moisture. If the moisture is not removed, the sleeve could bulge and rub the can during dry running operation. This design provides for an air tight nonmagnetic container as well as minimizing the air gap. Likewise, the stator also requires a high quality fabrication because of the thin can, 0.015 to 0.025 in, and relative high pump design pressure. Because of the thin cans, a potential problem can occur when the pump is shut down. Since the motor windings hold heat, the heat could flash fluid inside the can; thus, swelling the thin can. The can is normally press fitted inside the stator windings to reduce the air gap which improves motor efficiency. Furthermore, a nonmagnetic sleeve or can, such as 316 SS or Hastelloy, are used to reduce induction heat from the magnetic field between the stator and rotor.

The motor design uses special lamination and winding combinations to achieve high electrical efficiency and low winding temperatures. Special stator winding laminations are used because of the larger air gap, rotor to stator clearance, required by canned motor design, as compared to the standard induction type motor. The power losses in the form of heat transferred into the fluid is a function of the air gap, rotor to stator alignment, and drag as the rotor rotates in the fluid. Typical clearance between rotor and stator cans are 0.026 to 0.040 in, depending upon pump manufacturer.

The stator is commonly installed in sealed, heavy outer steel casing, and sometimes filled with a special fluid. Motor windings have a class H or N insulation rating, depending upon pump manufacturer, along with special insulation lead materials such as teflon or equal material. One pump manufacturer offers a ceramic type insulation for high temperature services. As illustrated, the motor is a special design.

With the special motor design, the canned motor efficiencies approach the standard induction motor efficiencies. As a result of the efficiency, the stator and winding temperatures are lower, which improves motor life expectancy.

One major problem with the canned motor pump design is what happens when the stator liner ruptures as a result of the internal rotor rubbing the can. When the stator liner ruptures, the process fluid being pumped enters the winding area and flow through the electrical junction box and in to the conduit. For this reason, some manufacturers offer an option to seal the motor leads. Following is a brief description to show what potential problems could occur.

- If process fluid enters the switch gear via the conduit, the fluid could cause a fire or explosion in the switch gear or motor control center, corrode starters or other electrical components, or degrade wiring insulation.
- If process fluid enters control room via the conduit, the fluid could overcome operating personnel, damage instrumentation and computers, or cause a fire or explosion.
- If process fluid enters conduit, the fluid could degrade the wiring insulation, which could cause ground faults that may lead to similar problems described above.

EXTERNAL ROTOR SHAFT BEARING DESIGN

On the magnetic drive pump design, two antifriction bearings are used to support the drive shaft with outer magnet assembly, to properly position the outer to inner magnet assemblies, and to absorb the forces that are transmitted from the magnetic coupling. Since the bearing load is basically radial, no duplex angular contact or double row bearing is required to absorb thrust. The only thrust load the bearing would have is from the slight misalignment of the inner and outer magnets, caused from machining and assembly tolerances or incorrect installation of the rotor, depending upon manufacturer's design. According to the engineering survey, all manufacturers use two deep groove, Conrad type radial bearings, such as the 6000 series (medium duty). Furthermore, the survey revealed several different methods used to mount the bearings in the housing. One manufacturer mounts the bearing similar to the standard ANSI type pumps. This method uses a snap ring fitted in a groove on the bearing outer race that is a simple and space saving means to axially locate the bearing in the housing. The bearing cap then clamps the snap ring to prevent any axial movement of the bearing. Other manufacturers use shoulders in the bearing housing to axially locate the bearing as well as clamping the bearing outer race between the shoulder and the bearing cap. One manufacturer uses two snap rings to locate the bearings, along with securing the bearing in place. There is a spacer between the two radial bearing to help clamp the outer bearing races.

The clamped outer races and shoulders should be the preferred design (Figure 19). In addition to the mounting method, the bearing housing alignment is also critical because of the flexing of the bearing cages that causes the balls to rub the cage and initiate fatigue cracks at the stress points from the thermal load cycling. Since lubrication is critical at the rubbing contact area because of heat generation, the desirable bearing separator should be phenolic, machined bronze, pressed brass strips, press steel strips, and riveted steel strips, in the preferred order as listed [6].

FEATURES	PUMP MANUFACTURER % MAGNETIC DRIVE
LUBRICATION	
GREASE	58
OIL	33
OIL MIST	17
SNAP RING AXIAL POSITION	17
CLAMP BEARING AXIAL POSITION	42

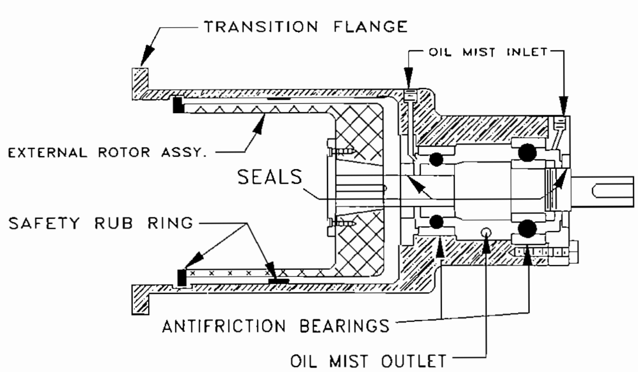


Figure 19. External Rotor and Bearing Design.

According to the survey, most manufacturers standards are to use grease lubrication for the bearings. Wet sump oil lube and oil mist are offered as an option by some manufacturers. The manufacturer's lubrication standards and options are summarized in Figure 19.

Based on the author's experience, the dry sump oil mist is the most reliable bearing lubrication method. To ensure the bearings obtain proper lubrication, the oil mist reclassifier should be properly sized, and the oil mist should enter from one side of the bearing and vent on the opposite side of the bearing. The installation shown in Figure 19 is preferred, because the oil mist enters from the outer side of the bearings and vents in the middle, where a collection bottle with a vent is used to collect the residual oil for a sample if necessary, or to visually check the oil for metal contamination from the bearings. The oil mist aids in cooling the bearings, and most of all, protects the bearings from the high humidity, such as the Gulf Coast atmosphere, whether the pump is operating or idled. Furthermore, the oil mist reduces operating and maintenance personnel problems with adding too much oil or using an oil can containing a mixture of oil and water.

Most manufacturers use standard lip seals to protect the bearings from the surrounding environment. Some manufacturers offer magnetic seals as an option for protecting the bearing. The magnetic seal is an excellent method but is usually very expensive as compared to other alternative type seals, along with long delivery on magnetic seal replacement parts, if the user does not stock the parts. A good alternate seal design is the close clearance rotating labyrinth seal, currently offered by several seal manufacturers. The alternate design costs much less than the magnetic seal design, 50 percent or more on some sizes, and requires fewer modifications to the pump manufacturer's standard bearing housing than with the lip seal design.

Should the antifriction bearings fail, the drive rotor with the outer magnets could come in contact with the containment shell; therefore, the containment shell could rupture. Several of the pump manufacturers (42 percent), have some type of safety protection built into the pump design to prevent the outer rotor from rubbing the containment shell. Two of the manufacturers have outer rotor rub rings (Figure 1), while others rely on the smaller clearance between the outer rotor and the bearing housing, as compared to the clearance between the outer rotor and the containment shell. The preferred design would be the replaceable rub ring with a nonsparking material, currently offered by one or two pump manufacturers depending upon the pump frame size. All manufacturers should incorporate a nonsparking safety feature into their pump in order to prevent a source of ignition that could lead to a major fire or explosion, similar to that experienced with a mechanical seal failure.

INTERNAL ROTOR BEARING DESIGN

The most fragile or weakest design in the sealless pump, magnetic drive or canned motor, is the internal bearings and shaft with or without sleeves. Since the internal bearings are the core of the pump, the service life depends upon its environment and loading. As discussed earlier, the hydraulic design has a great impact on the life expectancy of the bearing, along with the bearing design and material. Any defects in design can result in reliability problems, along with costly repairs. It has been the author's experience that when severe bearing wear or damage occurs on canned motor pumps, the metal can could rupture and cause environmental problems, plus the repair cost could equal the cost of a new pump. A similar sequence of events can be seen with the magnetic drive pump design.

Internal rotor bearings are the cylindrical radial sleeve bearing type design which is immersed in the pumping fluid. The fluid is utilized to provide a lubricating and cooling film between the journal, shaft or sleeve, and the radial bearing. Many factors can influence the wear and life expectancy, such as radial and axial loads from hydraulic design, bearing misalignment, fluid properties (lubrication and vapor pressure), and solids content in the fluid. The better surface finish quality on shaft journal and bearing sleeve greatly improves the lubrication film development from minimal to full hydrodynamic film conditions during startup and minimizes bearing wear during periods of marginal film conditions.

The survey shows a variety of shaft journal and bearing sleeve materials are available for the sealless pump manufacturer to apply in the many different pump applications. Some of the materials available for shaft journals and bearing sleeves are silicon carbide, carbon, filled teflon, ceramic, solid graphite impregnated with molten metal, or the shaft material with or without hard surface coatings as shown in Figure 20. In selecting the appropriate material, the shaft journal and bearing sleeve must be compatible with the fluid, capable of supporting radial and axial loads, resistance to erosion from inevitable foreign particles in the process stream, and have good wear properties with marginal lubrication.

FEATURES	PUMP DESIGN MAGNETIC DRIVE	CANNED MOTOR
BEARING SLEEVE & JOURNAL MATERIAL		
SILICON CARBIDE	75	75
CARBON	42	75
FILLED TEFLON	33	25
SHAFT MATERIAL	17	0
SHAFT COATING	25	75
LUBRICATION GROOVES		
RADIAL BEARING	67	75
THRUST BEARING	67	75
BEARING MATERIAL MOUNTED IN METALLIC HOLDER		
RADIAL BEARING	67	25
THRUST COLLAR	67	25
ONE PIECE BEARING CARRIER	33	0

Figure 20. Internal Rotor and Bearing.

The more frequently used bearing materials along with some of the properties are listed below.

• Silicon carbide is a dense, fine grained, ceramic material with a high thermal conductivity that was primarily developed for the nuclear industry. The types of silicon carbide are the reaction bonded, alpha sintered, and beta sintered. The reaction bonded has free silicon metal, which is advantageous for possible dry run, but has a main disadvantage in that the material should not be used in fluids with a pH of less than four or greater than 11, because the fluid will attack the free silicon metal. The alpha sintered is more corrosion resistant, which is more advantageous, but the disadvantage is the material requires a fluid film to have good life. The beta sintered is very similar to the alpha sintered and has no major advantage over the alpha grade. The properties for the different grades of silicon carbide are shown in Figure 21 [10 & 11].

MATERIAL	HARDNESS KNOOP	COMPRESSIVE STRENGTH 10^3 PSI	THERMAL EXP. COEFF. 10^6 IN/IN/°F
REACTION BONDED	2700	NOT AVAILABLE	2.61
ALPHA SINTERED	2800	560	2.20
BETA SINTERED	565	2.20	

Figure 21. Silicon Carbide Properties.

• Carbon is a material consisting of fillers, binders, and additives. The filler could be graphite, petroleum coke, lamp black or charcoal, and is typically 20 to 80 percent of the material. The binder could be synthetic resin, tar pitch, petroleum pitch, and other type substances, and is 15 to 50 percent of the material. The additive could be film formers, antioxidant, graphitizing aids, or other substances, and is 0 to 10 percent of the material. As shown by the ingredients mixture, there are many grades and property variations with carbon. The properties for the most common grade of carbon used in bearings for sealless pumps are shown in Figure 22 [11].

MATERIAL	HARDNESS SHORE C	COMPRESSIVE STRENGTH 10^3 PSI	THERMAL EXP. COEFF. 10^6 IN/IN/°F
GRADE 6038-C	80	21	2.0
GRADE 658RCH	95	31	2.7

(NOTE: THE 6038-C is very corrosion resistance as compared to 658RCH)

Figure 22. Carbon Properties.

• Filled teflon is a fluorocarbon material with various fillers for improving the virgin teflon properties, such as creep, hardness, etc. Many substances can be blended into teflon, but the more common fillers are glass and carbon. The properties for filled teflon are shown in Figure 23 [12].

MATERIAL	HARDNESS Durometer D Scale	COMPRESSIVE STRENGTH 10^3 PSI	THERMAL EXP. COEFF. 10^5 N/IN/°F	PV LIMIT AT@ 1000 FT./MIN.
TFE	52	600	5.5	2,500
TFE w/15% Glass Fiber	54	1,000	3-8	15,000
TFE w/25% Glass Fiber	56	NO LISTING	NO LISTING	16,000
TFE w/15% Graphite	55	1,080	4-7	28,000

Figure 23. Filled Teflon Properties.

Besides the variety of bearing materials available, the survey showed most bearing sleeves are mounted into a metallic carrier with a press fit or slip fit, depending upon material and pump manufacturer. Bearing with a slip fit is pinned to prevent rotation. Although some manufacturers use the shaft material or place a hard coating surface on the shaft, most manufacturers provide shaft journals with replaceable slip on sleeves. Most manufacturers rely on clamping the shaft journal sleeve between the impeller and the magnetic coupling rotor, but one goes a step further to protect the sleeve material from rotating on the shaft or from cracking due to thermal expansion. This manufacturer uses expansion rings between the shaft and sleeve along with Graphoil® packing to seal between the sleeve face and shaft. In addition to the mounting arrangement, the most common diametrical clearances range from 0.002 to 0.004 in/in of bearing diameter for bearings ranging from 1.5 to 3.5 in in diameter. The survey showed some manufacturers bearing clearances were as low as 0.001 in/in of bearing diameter. Bearing length to diameter ratio (L/D) varied from 0.27 to 1.5, depending upon manufacturer, but the majority of the bearings were designed with an L/D ratio of 0.8 to 1.1. A larger L/D ratio is desirable because the load capacity and bearing life increases as L/D increases.

From a user's viewpoint, the preferred design is the silicon carbide bearing sleeve running against a silicon carbide shaft journal with thermal expansion protection to prevent the silicon carbide from cracking or rotating on the shaft. In addition, the silicon carbide should be mounted in a metallic holder to prevent the pieces from destroying the pump internals, should the silicon carbide fracture. Futhermore, regardless of bearing material used, all bearings should have lubrication grooves to aid in removing heat and to help flush foreign particles out of the bearing. Also, the user should review the number and type of positive alignment fits, rabbet or dowel type, that are used to maintain bearing alignment. The fewer number of alignment fits, the less chance of assembly stack up errors, which can cause bearing wear problems.

INSTRUMENTATION AND PROTECTIVE SYSTEMS

Since the internal flow is critical to the reliability of the sealless pump, the pump manufacturers offer several instrumentation options for the magnetic drive and canned motor pumps. The instrumentation applicable to each pump design, as well as the percentage of manufacturers offering the option, is shown in Figure 24.

FLUID	FLAMMABLE OR TOXIC	NON-FLAMMABLE
NPSH MARGIN	5 Ft. OR LESS	ABOVE 5 Ft.
VAPOR PRESSURE MARGIN	50 °F OR LESS	ABOVE 50 °F
TYPE INSTRUMENT		
MOTOR CURRENT SENSORS	YES	OPTIONAL
CONTAINMENT SHELL OR CAN TEMP. PROBE	YES	YES
INTERNAL FLOW RETURN TEMP. PROBE	YES	OPTIONAL
VIBRATION PROBES	OPTIONAL	OPTIONAL
VIBRATION PROBE MOUNTING	YES	OPTIONAL
LEAK DETECTOR	OPTIONAL	NO
DIFFERENTIAL PRESSURE SWITCH	OPTIONAL	NO
PRESSURE GAGES SUCTION & DISCHARGE	YES	YES
FLOW MEASUREMENT	YES	OPTIONAL
SUCTION VESSEL LEVEL ALARM & SHUTDOWN	YES	ALARM ONLY

Figure 24. Instrumentation Minimum Requirements.

If the user is requesting the pump manufacturer to supply protective instrumentation other than the temperature probe only, the pump manufacturer needs to ask questions as to how the protective instrumentation, such as leak detectors, motor current sensors, etc., are going to be used, and how the instrument is going to interface with the user's control room instrumentation. The primary reason for this word of caution is the author has experienced interface problems with vibration monitoring equipment and a process computer. There are many different type process computers being used with various type input and output interfaces that have to match the instrument output signal. Also, some petrochemical and refinery industries have certain instruments that are preferred over others, and there is no way the pump manufacturer will know which type to supply unless the user specifies the supplier, type, and model. It is the author's opinion that the user should supply all protective instrumentation, other than temperature probes, in order to prevent instrument interface problems. This allows the plant instrument personnel to work the instrument problems, and not the machinery engineer.

From a user's perspective, the preferred minimum instrumentation for a sealless pump installation is the temperature probe on the containment shell for magnetic drive pumps, shaft position on canned motor pumps, motor current sensors on both designs, and suction vessel low level alarm and shutdown, along with local pressure gauges across the suction strainer and on discharge line and flow measurements on the discharge. Since the most critical section of the sealless pump design is the return fluid stream, all pump manufacturers should have a temperature probe to measure the return stream temperature. If the manufacturer's current pump design does not incorporate the return fluid temperature probe feature, the manufacturer should consider improving the pump design to where a probe could be installed to monitor and protect the sealless pump; therefore, improve the sealless pump reliability.

COST COMPARISON BETWEEN SEALLESS AND STANDARD TYPE PUMPS

To determine the cost difference between the standard ANSI or API pumps and the sealless pump, canned motor and magnetic drive, six services containing hydrocarbon or a mixture of hydrocarbons with toxic chemicals were selected. The services ranged from flows of 2.5 to 650 gpm and heads of 59 to 332 ft. The operating temperatures ranged from 130°F to 165°F, vapor pressure ranged from 2.2 to 165 pisa, and specific gravity ranged from a minimum of 0.55 to a maximum of 0.98. The results of the comparison between a sealless and a standard design pump are shown in Figure 25.

Since the best sealless pump may not be the lowest cost pump, a decision making chart or an equivalent evaluation

	FLOW GPM	HEAD TDH, FEET	NPSHA	SUCTION PRESS., PSIG RATED	MAX.	TEMP., °F	VAPOR PRESS., PSIA	SPECIFIC GRAVITY
SERVICE 1	8.0	47	20	17	64		2.2	0.98
SERVICE 2	2.5 & 4.5	59 & 106	20	15 & 11	288		2.2 & 7.4	0.63 & 0.98
SERVICE 3	550	315	8	B3	126		B1	0.55
SERVICE 4	71	326	9	100	140		95	0.89
SERVICE 5	277	134	6	142	180		95	0.86
SERVICE 6	130	165	9	169	1B5		165	0.63
SERVICE 7	650	95	B	7	50		4	0.85

% ABOVE BASE COST -- PUMP, BASE, & DRIVER

	COST COMPARISON, %				APPLICATION COMPARISON, %				
PUMP TYPE	ANSI[1] STD.	API[2] STD.	MAGNETIC[3] DRIVE	CANNED[4] MOTOR	MFG. QUTOTED	CURVE FIT POOR	GOOD	ACCEPT.	N/A[5]
SERVICE 1	BASE	206	156-34B	69-301	60	89	0	0	11
SERVICE 2	BASE	221	115-374	79-167	53	88	0	0	12
SERVICE 3	BASE	211	213-350	17B-360	33	20	20	0	60
SERVICE 4	BASE	174	168-408	158-325	53	25	25	3B	12
SERVICE 5	BASE	209	22B-445	143-302	40	0	17	17	66
SERVICE 6	BASE	158	174-372	185-325	47	0	14	57	29
SERVICE 7	BASE	180	218-450	201-390	47	0	43	2B	27

Note:
1) Material is 316SS which is typically ANSI standard.
2) Material is API A-8 (316 SS).
3) Material is 316 SS (75%) and Hast. C (25%)
4) Material is 316 SS.
5) NOT ACCEPTABLE -NPSHR > NPSHA

PUMP INSTALLED COST IS ESTIMATED AT 3 TIMES BASE EQUIPMENT COST

Figure 25. Sealless Vs Standard ANSI and API Pump Cost.

should be used to select the best pump for the application (Figure 26) [13]. A similar chart could be used to decide whether to use a standard pump with mechanical seals, magnetic drive pump, or a canned motor pump for a specific application.

CONCLUSION

When properly applied and installed in compatible fluids, the sealless pump should operate troublefree for many years. Canned motor and magnetic drive pumps are a good solution to reduce emissions from mechanical seals but are not a cure for all mechanical seal problems. Since sealless pumps have certain design and construction limitations, exceeding the limits will affect mechanical reliability and/or leakproof capability; consequently, the pumps can become the most troublesome pumps in the plant.

The pump design limitation will most likely cause the pumps to fail or wreck due to radial or thrust bearing problems. A sealless pump should not be applied in dirty service, fluid with a high vapor pressure rate of change, or fluid with high viscosity, above 100 centerpoise. Because of the design limitations and severity of potential problems, the sealless pump should be considered a special engineered type product. Sealless pump failure can cause potential consequences to personnel safety, environmental impact, and possible economic impact if the pump is misapplied, not operated properly, or installed with inadequate instrumentation protection. As a result, additional up front engineering, application review, vendor selection, equipment procurement, installation procedures, and operator training need to be allocated to have a successful and reliable sealless pump installation.

For the future, the pump manufacturers should consider a vertical inline magnetic drive pump similar to current ANSI and API vertical inline pumps with bearing brackets. Over the past three years, the author has purchased more vertical inline pumps with bearing brackets rather than horizontal pumps because of space limitations and lower installed cost.

APPENDIX 1

SURVEY QUESTION LIST

- What is the internal fluid flow path through the containment shell and bearings?
- What are the anticipated pressures at various points along the internal flow path, i.e., thrust bearing face, pressure differential across journal bearing, and etc.?

DECISION MAKING CHART EVALUATION ITEMS

SAFETY & ENVIROMENTAL
CONTAINMENT SHELL DESIGN:
ASME DESIGN
HYDROSTATIC TEST, 1.5 X MAWP
SELF DRAINING
SAFETY RUB PROTECTION RING ON ROTOR OR HOUSING
NON SPARKING TYPE RING MATERIAL
INNER ROTOR TO CONTAINMENT SHELL =< 0.1 INCHES
OUTER ROTOR TO CONTAINMENT SHELL =< 0.15 INCHES
INSTRUMENTATION OPTIONS:
DRIVEN SHAFT BRG.S TEMP. PROBES
CONTAINMENT SHELL SKIN TEMP. PROBE
CONTAINMENT SHELL FLUID RETURN TEMP. PROBE
OUTER HOUSING (SAFETY RING AREA) TEMP. PROBE

MECHANICAL DESIGN & RELIABILITY
BACK PULL OUT PUMP
SELF VENTING PUMP
IMPELLER: OPEN, CLOSE,
SEMI-CLOSE THURST BALANCED
FLANGES: ANSI OR EUROPEAN
CONTAINMENT SHELL DESIGN:
MATERIAL AVAILABITIES
WELDED CONSTRUCTION
INTERNAL BEARING DESIGN (DRIVEN SHAFT):
MATERIAL AVAILABITIES
BEARING MOUNTED ON COMMON CARRIER
BEARING MOUNTED ON CONTAINMENT SHELL
LUBRICATION GROOVES IN RADIAL BEARING
LUBRICATION GROOVES IN THRUST BEARING
THRUST COLLAR MOUNTED IN METAL RETAINER
SHAFT SLEEVES DESIGNED FOR THERMAL EXPANSION
EXTERNAL BEARING DESIGN (DRIVE SHAFT):
DEEP GOOVE ANTIFRICTION BEARINGS
SINGLE ROW RADIAL BEARINGS
40,000 Hr. BEARING LIFE
LUBE: OIL MIST, GREASE, PURGE MIST, WET SUMP, WET SUMP w/PURGE MIST
MATERIAL AVAILABILITY:
SHAFT, CASING, IMPELLER
WETTED FASTENERS

OPERATIONAL DESIGN & RELIABILITY
CONTAINMENT SHELL DESIGN:
SELF VENTING
CONTAINMENT SHELL FLUID FLOW:
DISCHARGE BACK TO SUCTION PRESS. AREA
DISCHARGE BACK TO DISCHARGE PRESS. AREA
DEAD SPOT AREAS
EXTERNAL FLUSH AVAIALBILITY
INTERNAL STRAINER
MAGNETIC COUPLING:
HERMATICALY SEALED INNER MAGNETS
TEMPEATURE LIMITATION >= 500 F
HARD START CAPABILITY
INSTRUMENTATION OPTIONS:
VIBRATION PROBE PROVISIONS
FLOW DETECTORS
MAGNETIC STRAINERS AVAIABILITY
MOTOR SENSORS AVAILABILITY
DIFFERENTIAL PRESSURE SWITCH AVAILABILITY

SOLIDS HANDLING ABILITY
CONTAINMENT SHELL DESIGN:
INNER ROTOR TO CONTAINMENT SHELL =< 0.1 INCHES
IMPELLER: OPEN, CLOSE, SEMI-CLOSE, THURST BALANCED
INTERNAL BEARING DESIGN (DRIVEN SHAFT):
BEARING MOUNTED ON CONTAINMENT SHELL
LUBRICATION GROOVES IN RADIAL BEARING
LUBRICATION GROOVES IN THRUST BEARING
CONTAINMENT SHELL FLUID FLOW:
DISCHARGE BACK TO SUCTION PRESS. AREA
DISCHARGE BACK TO DISCHARGE PRESS. AREA
DEAD SPOT AREAS
EXTERNAL FLUSH AVAIALBILITY
INTERNAL STRAINER

DESIGN SIMPLICITY
UNIQUE DESIGN DIFFICIENY FEATURE
UNIQUE DESIGN IMPROVEMENT FEATURES
CONTAINMENT SHELL DESIGN:
ASME DESIGN
HYDROSTATIC TEST, 1.5 X MAWP
INTERNAL BEARING DESIGN
(DRIVEN SHAFT):
BEARING MOUNTED ON COMMON CARRIER
LUBRICATION GROOVES IN RADIAL BEARING
LUBRICATION GROOVES IN THRUST BEARING
EXTERNAL BEARING DESIGN (DRIVE SHAFT):
DEEP GOOVE ANTIFRICTION BEARINGS
SINGLE ROW RADIAL BEARINGS
LUBE: OIL MIST, GREASE, PURGE MIST, WET SUMP, WET SUMP w/PURGE MIST

PROVEN DESIGN
PUMP USERS' LIST
PUMP POWER FRAME SIZES -- AVAILABLE
PUMP POWER FRAME SIZES --PLANNED
STANDARDS:
EUROPEAN
AMERICAN NATIONAL (ANSI)
AMERICAN PETROLEUM (API)
INTERNATIONAL (ISO)

US AVAILABILITY
ENGINEERING CAPABILITIES
APPLICATION GROUP
MANUFACTURING
ASSEMBLY
PERFORMANCE TESTING
PARTS STOCKING

COST
STD. ANSI vs SEALLESS ANSI
EQUIPMENT ONLY
INSTALLED
DELIVERY
REPAIR
STD. API vs SEALLESS API
EQUIPMENT ONLY
INSTALLED
DELIVERY
REPAIR

DECISION MAKING CHART
SEALLESS PUMPS

FACTOR BEING RATED	WEIGHTED I TO IO	PUMP MANUFACTURER'S RATING 1 TO IO															
		MFG: X	MFG.: U	MFG: T	MFG: V	MFG: R	MFG: Q	MFG: O	MFG: N	MFG: P	MFG: H	MFG: J	MFG: Y	MFG: Z	MFG: E	MFG: F	MFG: M
SAFETY & ENVIRONMENTAL	10	8	8	7	7	6	0	8	8	6	7	7	2	5	8	7	7
MECHANICAL DESIGN & RELIABILITY	8	7	7	8	6	5	0	7	7	5	6	5	2	4	6	6	6
OPERATIONAL DESIGN & RELIABILITY	9	8	7	7	6	5	0	6	7	6	6	5	2	4	6	6	6
SOLIDS HANDLING ABILITY	5	6	5	5	4	4	0	5	5	4	4	4	2	3	4	4	4
DESIGN SIMPLICITY	6	8	8	8	8	6	0	7	6	6	5	6	4	5	7	7	7
PROVEN DESIGN	7	7	6	8	5	2	0	6	4	5	7	4	1	2	8	6	7
US AVAILABILITY	7	4	8	1	7	8	0	4	8	7	5	2	2	2	7	7	2
COST	6	5	4	0	5	0	0	5	0	0	4	2	0	0	7	6	2
TOTALS	580	393	394	333	354	271	0	357	344	294	330	265	109	191	391	361	309
DECISION PERCENTAGES		67.8	67.9	57.4	61.0	46.7	0.0	61.6	59.3	50.7	56.9	45.7	18.8	32.9	67.4	62.2	53.3

Note: "0" indicates no data supplied for evaluation

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Figure 26. Sealless Pump Decision Making Chart.

- What are the advantages and disadvantages of your specific pump design as compared to other magnetic drive pumps on the market?
- How many seconds or minutes does it take for the heat to build up in the containment shell?
- Should the driver shaft decouple with driven shaft?
- How do you calculate the heat generation from the magnets during normal operation and during decouple operation?
- What is the containment shell fluid typical design margin, pressure, and temperature, as compared to the fluid vapor pressure?

What options are available on the magnetic drive pumps to detect problems with reduced internal fluid flow, decoupling of the magnetic drive coupling, containment shell leakage, suction loss, and etc.?

- Where are the pumps manufactured, assembled, and tested?
- What is the typical delivery on the magnetic drive pumps?
- Do you have engineering capabilities in the US?
- Does the containment shell design meet the ASME Section VIII, Division I stress levels?
- Does the casing design meet the ASME Section VIII, Division I stress levels?
- What welding code is used in fabricating the containment shell?
- Should the antifriction bearings fail, does the pump have a safety feature to prevent the outer magnet rotor from rupturing the containment shell?

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