Section TECH-B
Pump Application Data

TECH-B-1 Corrosion & Materials of Construction

Selecting the right pump type and sizing it correctly are critical to the success of any pump application. Equally important is the selection of *materials of construction.* Choices must be made between metals and/or non-metals for pump components that come into contact with the pumpage. In addition, gaskets and O-ring material selections must be made to assure long leak-free operation of the pump's dynamic and static sealing joints. To assist in proper selection, included in this section is a brief discussion of specific types of corrosion and a general material selection guide.

Corrosion

Corrosion is the destructive attack of a metal by chemical or electra-chemical reaction with its environment. It is important to understand the various types of corrosion and factors affecting corrosion rate to properly select materials.

TYPES OF CORROSION

(1) Galvanic corrosion is the electro-chemical action produced when one metal is in electrical contact with another more noble metal, with both being immersed in the same corroding medium called the electrolyte. A galvanic cell is formed and current flows between the two materials. The least noble material called the anode will corrode while the more noble cathode will be protected. It is important that the smaller wearing parts in a pump be of a more noble material than the larger more massive parts, as in an iron pump with bronze or stainless steel trim.

Following is a galvanic series listing the more common metals and alloys.

Corroded End
(Anodic, or least noble)
Magnesium
Magnesium Alloys
Zinc
Aluminum 2S
Cadmium
Aluminum 175T
Steel or Iron
Cast Iron
Stainless Steel, 400 Series (Active)
Stainless Steel, Type 304 (Active)
Stainless Steel, Type 316 (Active)
Lead-tin Solders
Lead
Tin
Nickel (Active)
Nickel base alloy (active)
Brasses
Copper
Bronzes
Copper-Nickel Alloy
Monel
Silver Solder
Nickel (Passive)
Nickel Base Alloy (Passive)
Stainless Steel, 400 Series (Passive)
Stainless Steel, Type 304 (Passive)
Stainless Steel, Type 316 (Passive)
Silver
Graphite
Gold
Platinum Protected End
(Cathodic, or most noble)

(2) Uniform Corrosion is the overall attack on a metal by a corroding liquid resulting in a relatively uniform metal loss over the exposed surface. This is the most common type of corrosion and it can be minimized by the selection of a material which offers resistance to the corroding liquid.

(3) Intergranular corrosion is the precipitation of chromium carbides at the grain boundaries of stainless steels. It results in the complete destruction of the mechanical properties of the steel for the depth of the attack. Solution annealing or the use of extra low carbon stainless steels will eliminate intergranular corrosion.

(4) Pitting Corrosion is a localized rather than uniform type of attack. It is caused by a breakdown of the protective film and results in rapid pit formation at random locations on the surface.

(5) Crevice or Concentration Cell Corrosion occurs in joints or small surface imperfections. Portions of the liquid become trapped and a difference in potential is established due to the oxygen concentration difference in these cells. The resulting corrosion may progress rapidly leaving the surrounding area unaffected.

(6) Stress Corrosion is the failure of a material due to a combination of stress and corrosive environment, whereas the material would not be affected by the environment alone.

(7) Erosion-Corrosion is the corrosion resulting when a metal's protective film is destroyed by high velocity fluids. It is distinguished from abrasion which is destruction by fluids containing abrasive solid particles.

pH VALUES

The pH of a liquid is an indication of its corrosive qualities, either acidic or alkaline. It is a measure of the hydrogen or hydroxide ion concentration in gram equivalents per liter. pH value is expressed as the logarithm to the base 10 of the reciprocal of the hydrogen ion concentration. The scale of pH values is from zero to 14, with 7 as a neutral point. From 6 to zero denotes increasing hydrogen ion concentration and thus increasing acidity, and from 8 to 14 denotes increasing hydroxide ion concentration and thus increasing alkalinity.

The table below outlines materials of construction usually recommended for pumps handling liquids of known pH value

pH Value	Material of Construction
10 to 14	Corrosion Resistant Alloys
8 to 10 6 to 8 4 to 6	Iron, Stainless Steel, Bronze, Carbon Steel
0 to 4	Corrosion Resistant Alloys

The pH value should only be used as a guide with weak aqueous solutions. For more corrosive solutions, temperature and chemical composition should be carefully evaluated in the selection of materials of construction.

TECH-B-2 Material Selection Chart

The material selection chart (Table 1, Page 15) is intended to be a guide in the preliminary selection of economic materials. Corrosion rates may vary widely with temperature, concentration and the presence of trace elements or abrasive solids. Blank spaces in the chart indicate a lack of accurate corrosion data for those specific conditions. Maximum temperature limits are shown where data are available.

Compatibility data for fluoropolymers EPDM, FKM, FFKM, PVDF and ECTFE (see code chart) were supplied by manufacturers.

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CODE FOR TABLE 1.	
A	Recommended
B	Useful resistance
X	Unsuitable
Steel	Carbon steel, cast iron and ductile iron
Brz	Bronze
316	Stainless steel
A-20	Carpenter stainless
CD4MCu	CD4MCu stainless steel
Alloy 2205	Alloy 2205 stainless steel
C-276	Wrought Hastelloy® C-276 alloy
Ti	Titanium unalloyed
Zi	Zirconium
ETFE	Ethylenetetrafluoroethylene (Tefzel®)
FP	Fluoropolymers (e.g., Teflon®) including perfluoroalkoxy (PFA), Hyflon (MFA),polytetrafluoroethylene (PTFE) and fluorinated ethylene propylene (FEP)
FRP	Fiber-reinforced plastic (vinylester resin)
EPDM	Ethylenepropylene rubber (Nordel®)
FKM1	Standard grades; dipolymers of hexafluoropropylene (HFP) and vinylidene fluoride (VF_2; Viton®, Tecnoflon®)
FKM2	Specialty grades; terpolymers comprising at least three of the following: HFP, VF_2, tetrafluorethylene (TFE), perfluoromethylvinyl ether (PMVE) or ethylene (E). Specialty grades may have significantly improved chemical compatibility compared to standard grades in many harsh chemical environments (Viton®, Tecnoflon®).
FFKM	Copolymer of TFE and PMVE (Kalrez®, Tecnoflon®)
PVDF	Polyvinylidene fluoride (Kynar®, Hylar, Solef®)
ECTFE	Ethylene chlorotrifluoroethylene (Halar®)

NOTE: Compatibility is dependent on specific form and/or grade. Contact elastomer manufacturer.

Guidelines for information purposes and not design guidelines*

TABLE 1. MATERIAL SELECTION CHART.

Corrosive	Steel	Brz	316	A-20	CD4MCuN	Alloy 2205	C-276	Ti	Zi	ETFE	FP	FRP	EPDM	FKM1	FKM2	FFKM	PVDF	ECTFE
Acetaldehyde, 70°F	B	A	A	A	A	A	A	A	A	A	A	X	A	X	X	A	X	B
Acetic acid, 70°F	X	A	A	A	A	A	A	A	A	A	A	X	A	X	B	A	A	A
Acetic acid, <50%, to boiling	X	B	A	A	B	A	A	A	A				A	X	B	A	B	A
Acetic acid, >50%, to boiling	X	X	B	A	X	A	A	A	A	104°C	A	X	B	X	B	A	X	B
Acetone, to boiling	A	A	A	A	A	A	A	A	A	104°C	A	X	A	X	X	A	X	X
Aluminum chloride, <10%, 70°F	X	B	X	B	X	B	A	B	A	A	A	A	A	A	A	A		A
Aluminum chloride, >10%, 70°F	X	X	X	B	X	B	A	B	A	A	A	A	A	A	A	A	A (to 40°C)	A
Aluminum chloride, <10%, to boiling	X	X	X	X	X	X	B	X	A	104°C	A	X	A	A	A	A	A	A
Aluminum chloride, >10%, to boiling	X	X	X	X	X	X	B	X	A	104°C	A	X	A	A	A	A	A (to 40°C)	A
Aluminum sulphate, 70°F	X	B	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Aluminum sulphate, <10%, to boiling	X	B	B	A	B	A	A	A	A	104°C	A		A	A	A	A	A	A
Aluminum sulphate, >10%, to boiling	X	X	X	B	X	B	A	X	B	104°C	A		A	A	A	A	A	A
Ammonium chloride, 70°F	X	X	B	B	B	B	A	A	A	A	A	A	A	A	A	A	A	A
Ammonium chloride, <10%, to boiling	X	X	B	B	X	B	A	A	A	104°C	A		A	A	A	A	A	A
Ammonium chloride, >10%, to boiling	X	X	X	X	X	X	B	X	X	104°C	A		A	A	A	A	A	A
Ammonium fluosilicate, 70°F	X	X	X	B	X	B		X	X							A		
Ammonium sulphate, <40%, to boiling	X	X	B	B	X	B	B	A	A	104°C	A		A	X	B	A	A	A
Arsenic acid, to 225°F	X	X	X	B	X	B				A	A		A	A	A	A	A	A
Barium chloride, 70°F <30%	X	B	X	B	X	B	A	B	B	A	A	A	A	A	A	A	A	A
Barium chloride, <5%, to boiling	X	B	X	B	X	B	B	A	A	104°C	A		A	A	A	A	A	A
Barium chloride, >5%, to boiling	X	X	X	X	X	X	B	X	X	104°C	A		A	A	A	A	A	A
Barium hydroxide, 70°F	B	X	A	A	A	A	B	A	A	A	A	A	A	A	A	A	A	A
Barium nitrate, to boiling	X	X	B	B	B	B		B	B	104°C	A					A	A	A
Barium sulphide, 70°F	X	X	B	B	B	B		A	A	A	A	A	A	B	A	A	A	A
Benzoic acid	X	X	B	B	B	B	A	A	A	A	A		X	A	A	A	110°C	A
Boric acid, to boiling	X	X	B	B	B	B	A	B	B	104°C	A		A	A	A	A	A	A
Boron trichloride, 70°F dry	B	B	B	B	B	B	B									A	A	A
Boron trifluoride, 70°F 10%, dry	B	B	B	A	B	A	A						X	X	X	B	A	A
Brine (acid), 70°F	X	X	X	X	X	X	B	B		A	A	A	A	A	A	A	A	A
Bromine (dry), 70°F	X	X	X	X	X	X	B	X	X	A	A	X	X	A	A	A	A	A
Bromine (wet), 70°F	X	X	X	X	X	X	B	X	X	A	A	X	X	B	A	A	A	A
Calcium bisulphite, 70°F	X	X	B	B	B	B	B	A	A	A	A		X	A	A	A	A	A
Calcium bisulphite	X	X	X	B	X	B	B	A	A	A	A		X	A	A	A	95°C	A
Calcium chloride, 70°F	B	X	B	B	B	B	A	A	A	A	A	A	A	A	A	A	A	A
Calcium chloride <5%, to boiling	X	X	B	B	B	B	A	A	A	104°C	A		A	A	A	A	A	A
Calcium chloride >5%, to boiling	X	X	X	B	X	B	A	B	B	104°C	A		A	B	A	A	A	A
Calcium hydroxide, 70°F	B	B	B	B	B	B	A	A		A	A	A	A	A	A	A	A	A
Calcium hydroxide, <30%, to boiling	X	B	B	B	B	B	A	A		104°C	A		A	B	A	A	A	A
Calcium hydroxide, >30%, to boiling	X	X	X	X	X	X	B	A		104°C	A		A	B	A	A	A	A
Calcium hypochlorite, <2%, 70°F	X	X	X	X	X	X	A	A	A	A	A	X	B	A	A	A	A	A
Calcium hypochlorite, >2%, 70°F	X	X	X	X	X	X	B	A	B	A	A	X	B	A	A	A	A	A
Carbolic acid, 70°F (phenol)	X	B	A	A	A	A	A	A	A	A	A		B	B	A	A	50°C	A
Carbon bisulphide, 70°F	B	B	A	A	A	A		A		A	A		X	A	A	A		A
Carbonic acid, 70°F	B	X	A	A	A	A	A	A	A	A	A		A	A	A	A	A	A
Carbon tetrachloride, dry to boiling	B	B	A	A	A	A	B	A	A	104°C	149°C		X	B	A	A	A	
Chloric acid, 70°F	X	X	X	B	X	B	B			A	A					A	A	A
Chlorinated water, 70°F	X	X	B	B	B	B	A	A	A	A	A		A	A	A	A		A
Chloroacetic acid, 70°F	X		X	X		X	A	A	B	A	A	A	A	X	B	A	X	A
Chlorosulphonic acid, 70°F	X	X	X	X	X	X	A	B	X	A	A	A	X	X	X	A	X	A
Chromic acid, <30%	X	X	X	B	X	B	B	A	A	65°C	A	X	X	A	A	A	80°C	A

Corrosive	Steel	Brz	316	A-20	CD4MCuN	2205	C-276	Ti	Zi	ETFE	FP	FRP	EPDM	FKM1	FKM2	FFKM	PVDF	ECTFE
Citric acid	X	X	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Copper nitrate, to 175°F	X	X	B	B	B	B	B	B		A	A	A				A	A	A
Copper sulphate, to boiling	X	X	X	B	X	B	A	A	A	104°C	A		A	A	A	A	A	A
Cresylic acid	X	X	B	B	B	B	B			A	A		X	A	A	A	65°C	A
Cupric chloride	X	X	X	X	X	X	B	B	X	A	A			A	A	A	A	A
Cyanohydrin, 70°F	X		B	B	B	B				A	A	A		X	X	A		
Dichloroethane	X	B	B	B	B	B	B	A	B	65°C	A			B	A	A	A	23°C
Diethylene glycol, 70°F	A	B	A	A	A	A	B	A	A	A	A		A	B	A	A	A	A
Dinitrochlorobenzene, 70°F (dry)	X	B	A	A	A	A	A	A	A	A	A			X	B	A	A	
Ethanolamine, 70°F	B	X	B	B	B	B		A	A	A	A		B	X	X	A	X	X
Ethers, 70°F	B	B	B	A	A	A	A	A	A	A	A		X	X	X	A	B	B
Ethyl alcohol, to boiling	A	A	A	A	A	A	A	A	A	104°C	A		A	B	A	A	A	A
Ethyl cellulose, 70°F	A	B	B	B	B	B	B	A	A	A	A		B	X	X	A		A
Ethyl chloride, 70°F	X	B	B	A	B	A	B	A	A	A	A	X	X	B	A	A	A	A
Ethyl mercaptan, 70°F	X	X	B	A	B	A				A	A	X	X	B	B	A	A	A
Ethyl sulphate, 70°F	X	B	B	A	B	A	A			A	A	X		X	X	A		A
Ethylene chlorohydrin, 70°F	X	B	B	B	B	B	B	A	A	A	A	X	B	A	A	A	A	A
Ethylene dichloride, 70°F	X	B	B	B	B	B	A	A	A	A	A	X	X	A	A	A	A	A
Ethylene glycol, 70°F	B	B	B	B	B	B	A	A	A	A	A	A	A	A	A	A	A	A
Ethylene oxide, 70°F	X	X	B	B	B	B	A	A	A	A	A		X	X	X	A	A	A
Ferric chloride, <5%, 70°F	X	X	X	X	X	X	A	A	B	A	A	A	A	A	A	A	A	A
Ferric chloride, >5%, 70°F	X	X	X	X	X	X	B	B	X	A	A	X	A	A	A	A	A	A
Ferric nitrate, 70°F	X	X	B	A	B	A	B			A	A	A	A	A	A	A	A	A
Ferric sulphate, 70°F	X	X	X	B	X	B	B	B	B	A	A	A	A	A	A	A	A	A
Ferrous sulphate, 70°F	X	X	X	B	X	B	A	A	A	A	A	A		A	A	A	A	A
Formaldehyde, to boiling	B	B	A	A	A	A	B	A	A	104°C	A		A	X	B	A	X	B
Formic acid, to 212°F	X	X	X	A	B	A	A	X	A	A	A		A	X	X	A	A	A
Freon, 70°F	A	A	A	A	A	A	A	A	A	A	A		A/X[1]	A/X[1]	A/X[1]	A/B[1]	A	A
Hydrochloride acid, <1%, 70°F	X	X	X	B	X	B	A	B	A	A	A	A	A	A	A	A	A	A
Hydrochloric acid, 1% to 20%, 70°F	X	X	X	X	X	X	A	X	A	A	A	A	A	A	A	A	A	A
Hydrochloric acid, >20%, 70°F	X	X	X	X		X	A	X	B	A	A	X	A	B	A	A	A	A
Hydrochloric acid, <$^1/_2$%, 175°F	X	X	X	X	X	X	A	X	A	A	A	X	X	B	A	A	A	A
Hydrochloric acid, $^1/_2$% to 2%, 175°F	X	X	X		X		A	X	A	A	A	X	X	B	A	A	A	A
Hydrocyanic acid, 70°F	X	X	X	B	X	B				A	A	A	A	A	A	A	A	A
Hydrogen peroxide, <30%, <150°F	X	X	B	B	B	B	B	A	A	A	A		B	B	A	A	B	A
Hydrofluoric acid, <20%, 70°F	X	B	X	B	X	B	A	X	X	A	A		X	B	A	A	A	A
Hydrofluoric acid, >20%, 50°F	X	X	X	X	X	X	B	X	X	A	A		X	B	A	A	A	A
Hydrofluoric acid, to boiling	X	X	X	X	X	X	B	X	X				X	X	B	A	B	A
Hydrofluorsilicic acid, 70°F	X		X	B	X	B	B			A	A		B	A	A	A	A	A
Lactic acid, <50%, 70°F	X	B	A	A	A	A	A	A	A	A	A		A	A	A	A	A	A
Lactic acid, >50%, 70°F	X	B	B	B	B	B	A	A	A	A	A		A	A	A	A	A	A
Lactic acid, <5%, to boiling	X	X	X	B	X	B	B	A	A	104°C	A		X	B	A	A	50°C	A
Lime slurries, 70°F	B	B	B	B	A	B	A	B	B				A	A	A	A		A
Magnesium chloride, 70°F	X	X	B	A	B	A	A	A	A	A	A	A	A	A	A	A	A	A
Magnesium chloride, <5%, to boiling	X	X	X	B	X	B	A	A	A	104°C	A		A	A	A	A	140°C	A
Magnesium chloride, >5%, to boiling	X	X	X	X	X	X	B	B	B	104°C	A		A	A	A	A	140°C	A
Magnesium hydroxide, 70°F	B	A	B	B	A	B	B	A		A	A	A	A	A	A	A	A	A
Magnesium sulphate	X	X	B	A	B	A	A	B	B	A	A		A	A	A	A	135°C	A
Maleic acid	X	X	B	B	B	B	A	A		A	A		B	A	A	A	120°C	A
Mercaptans	A	X	A	A	A	A				A	A			X	B	A	A	A
Mercuric chloride, <2%, 70°F	X	X	X	X	X	X	B	A	A	A	A		A	A	A	A	A	A
Mercurous nitrate, 70°F	X	X	B	B	B	B	X			A	A			A	A	A	A	A
Methyl alcohol, 70°F	A	A	A	A	A	A	A	A	A	A	A		A	X	A	A	A	A

Corrosive	Steel	Brz	316	A-20	CD4MCuN	2205	C-276	Ti	Zi	ETFE	FP	FRP	EPDM	FKM1	FKM2	FFKM	PVDF	ECTFE
Naphthalene sulphonic acid, 70°F	X	X	B	B	B	B	B			A	A					A	A	A
Napthalenic acid	X	X	B	B	B	B	B			A	A		X	A	A	A	A	A
Nickel chloride, 70°F	X	X	X	B	X	B	A	B	B	A	A	A	A	A	A	A	A	A
Nickel sulphate	X	X	B	B	B	B	B		A	A	A		A	A	A	A	A	A
Nitric acid	X	X	B	B	B	B	X	B	B				X	X	B	A	70%, 50°C	90%, 70°C
Nitrobenzene, 70°F	A	X	A	A	A	A	A	A		A	A	X	A	B	A	A	A	A
Nitroethane, 70°F	A	A	A	A	A	A	A	A	A	A	A	X	B	X	X	A	A	A
Nitropropane, 70°F	A	A	A	A	A	A	A	A	A	A	A	X	X	X	X	A	B	A
Nitrous acid, 70°F	X	X	X	X	X	X	A			A	A	A		X	X	A		A
Nitrous oxide, 70°F	X	X	X	X	X	X				A	A			B	B	A	X	A
Oleic acid	X	X	B	B	B	B	B	X	X	A	A	X	B	B	B	A	120°C	A
Oleum acid, 70°F	B	X	B	B	B	B	B	B		A	A	X	X	B	A	A	X	B
Oxalic acid	X	X	X	B	X	B	B	X	A	A	A	X	A	A	A	A	50°C	A
Palmitic acid	B	B	B	A	B	A	B			A	A		B	A	A	A	120°C	A
Phenol (see carbolic acid)										A	A		B	B	A	A	50°C	A
Phosgene, 70°F	X	X	B	B	B	B	B			A	A			X	X	A	A	X
Phosphoric acid, <10%, 70°F	X	X	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A	A
Phosphoric acid, >10% to 70%, 70°F	X	X	A	A	A	A	A	B	B	A	A	X	A	A	A	A	A	A
Phosphoric acid, <20%, 175°F	X	X	B	B	B	B	A	X	B	A	A	X	A	A	A	A	A	A
Phosphoric acid, >20%, 175°F, <85%	X	X	X	B	X	B	A	X	X	A	A	X	A	A	A	A	A	A
Phosphoric acid, >10%, boil, <85%	X	X	X	X	X	X	B	X	X				A	A	A	A	A	A
Phthalic acid, 70°F	X	B	B	A	B	A	A	A	A	A	A			B	B	A	A	A
Phthalic anhydride, 70°F	B	X	A	A	A	A	A			A	A			B	B	A		A
Picric acid, 70°F	X	X	X	B	X	B	B			A	A		B	A	A	A	A	A
Potassium carbonate	B	B	A	A	A	A	B	A	A	A	A	A		A	A	A	140°C	A
Potassium chlorate	B	X	A	A	A	A	B	A	A	A	A	A		A	A	A	95°C	A
Potassium chloride, 70°F	X	X	B	A	B	A	A	A	A	A	A	A	A	A	A	A	A	A
Potassium cyanide, 70°F	B	X	B	B	B	B	A			A	A	A	A	A	A	A	A	A
Potassium dichromate	B	B	A	A	A	A	B	A	A	A	A		A	A	A	A	140°C	A
Potassium ferricyanide	X	B	B	B	B	B	B	A	A	A	A			B	B	A	140°C	A
Potassium ferrocyanide, 70°F	X	B	B	B	B	B	B		B	A	A	A		B	B	A	A	A
Potassium hydroxide, 70°F	X	X	B	A	B	A	A	B	A	A	A	A	A	X	B	A	X	A
Potassium hypochlorite	X	X	X	B	X	B	B	A		A	A			X	X	A	95°C	A
Potassium iodide, 70°F	X	B	B	B	B	B	A	A	A	A	A			A	A	A	A	A
Potassium permanganate	B	B	B	B	B	B	B			A	A	A		B	B	A	120°C	A
Potassium phosphate	X	X	B	B	B	B		B	B	A	A	A		A	A	A		A
Seawater, 70°F	X	B	B	A	B	A	A	A	A	A	A	A	A	A	A	A	A	A
Sodium bisulphate, 70°F	X	X	X	B	X	B	B	B	A	A	A	A		A	A	A	A	A
Sodium bromide, 70°F	B	X	B	B	B	B	B			A	A	A		A	A	A	A	A
Sodium carbonate	B	B	B	A	B	A	A	A	A	A	A	A		A	A	A	140°C	A
Sodium chloride, 70°F	X	B	B	B	B	B	A	A	A	A	A	A	A	A	A	A	A	A
Sodium cyanide	B	X	B	B	B	B	A	B		A	A		A	A	A	A	135°C	A
Sodium dichromate	B	X	B	B	B	B	B	B		100°C	A			A	A	A	95°C	A
Sodium ethylate	B	A	A	A	A	A	A			A	A					A	A	A
Sodium fluoride	X	X	B	B	B	B	A	B	B	A	A			A	A	A	140°C	A
Sodium hydroxide, 70°F	B	B	B	A	B	A	A	A	A	A	A	A	A	B	A	A	X	A
Sodium hypochlorite	X	X	X	X	X	X	B	A	B	A	A	X	B	B	A	A	40%, 95°C	A
Sodium lactate, 70°F	B	X	X	X	X	X	A			A	A	A				A	A	A
Stannic chloride, <5%, 70°F	X	X	X	X	X	X	B	A	A	A	A	A	A	A	A	A	A	A
Stannic chloride, >5%, 70°F	X	X	X	X	X	X	B	B	B	A	A		A	A	A	A	A	A
Sulphite liquors, to 175°F	X	X	B	B	B	B	B	A					B	A	A	A	A	A
Sulphur (molten)	B	X	A	A	A	A	A	A					A	A	A	A	120°C	
Sulphur dioxide (spray), 70°F	X	X	B	B	B	B	B	X		A	A		A	A	A	A	A	A

Corrosive	Steel	Brz	316	A-20	CD4MCuN	2205	C-276	Ti	Zi	ETFE	FP	FRP	EPDM	FKM1	FKM2	FFKM	PVDF	ECTFE
Sulphuric acid, <2%, 70°F	X	X	B	A	B	A	A	B	A	A	A	A	A	A	A	A	A	A
Sulphuric acid, 2%t o 40%, 70°F	X	X	X	B	X	B	A	X	A	A	A	A	B	A	A	A	A	A
Sulphuric acid, 40%, <90%, 70°F	X	X	X	B	X	B	A	X	X	A	A	X	B	B	A	A	A	A
Sulphuric acid, 93% to 98%, 70°F	B	X	B	B	B	B	A	X	X	A	A	X	X	B	A	A	A	A
Sulphuric acid, <10%, 175°F	X	X	X	B	X	B	A	X	B	A	A	A	X	A	A	A	A	A
Sulphuric acid, 10% to 60% & >80%, 175°F	X	X	X	B	X	B	B	X	X	A	A	X	B	A	A	A	A	
Sulphuric acid, 60% to 80%, 175°F	X	X	X	X	X	X	B	X	X	A	A	X	X	B	A	A	A	A
Sulphuric acid, <$^3/_4$%, boiling	X	X	X	B	X	B	A	X	B				X	B	A	A	120°C	120°C
Sulphuric acid, $^3/_4$% to 40%, boiling	X	X	X	X	X	X	X	X	B				X	B	A	A	120°C	120°C
Sulphuric acid, 40% to 65% & >85%, boiling		X	X	X	X	X	X	X	X	X			X	X	B	A	B	120°C
Sulphuric acid, 65% to 85%, boiling	X	X	X	X	X	X	X	X	X				X	X	B	A	95°C	120°C
Sulphurous acid, 70°F	X	X	X	B	X	B	B	A	B	A	A	A	X	X	B	A	A	A
Titanium tetrachloride, 70°F	X		X	B	X	B	A			A	A		X	B	B	B	A	A
Tirchlorethylene, to boiling	B	X	B	B	B	B	B	A	A				X	B	A	A	A	20°C
Urea, 70°F	X	X	B	B	B	B	A	B	B	A	A			B	A	A	A	A
Vinyl acetate	B	B	B	B	B	B	A			A	A			B	B	A	120°C	20°C
Vinyl chloride	B	X	B	B	B	B	A	A		A	A		X	A	A	A	95°C	B
Water, to boiling	B	A	A	A	A	A	A	A	A				A	A	A	A	A	A
Zinc chloride	X	X	B	A	B	A	A	A	A	A	A	A	A	A	A	A	140°C	A
Zinc cyanide, 70°F	X	B	B	B	B	B	B	B	B	A	A					A	A	A
Zinc sulphate	X	X	A	A	A	A	A	A		A	A	A	A	A	A	A	140°C	A

*** NOTE:**
The use of tables, graphs and charts, and text suggestions contained in these guidelines is provided for information purposes only. The performance of materials in services can be affected by minor variations in the operating environment and pumping operating conditions that may affect corrosion performance.

It is the responsibility of the user to determine the operating conditions and suitability of selected materials. It is the user's responsibility to ensure that a material will be satisfactory in the intended service and environment. Before using any material, the end user should satisfy himself as to the suitability of any material for the proposed end use.

Elastomer Selection Guide

Please use the following chart as a general guide only. Refer to detailed selection tables or the factory for specific elastomer recommendations.

Elastomer	Shore (A) Hardness	Max Temp Limit	pH Range	Abrasion	Resistance to Moderate Chemicals	Oils Hydrocarbons
Natural Rubber	40	154 F	5 - 12	E	G (1)	P
Polyurethane	81	149 F	3 - 11	E (2)	G (1)	E
Neoprene	60	212 F	3 - 12	G	G (1)	G
Nitrile	60	220 F	4 - 12	G	G (1)	E
Hypalon	55	230 F	1 - 14	G	E	G
Chlorobutyl	50	300 F	3 - 12	G	E	P

(1) Poor for oxidizing chemicals and strong acids.
(2) Fine particles only (200 mesh or less).

E = Excellent
G = Good
P = Poor

TECH-B-3 Piping Design

The design of a piping system can have an important effect on the successful operation of a centrifugal pump. Such items as sump design, suction piping design, suction and discharge pipe size, and pipe supports must all be carefully considered.

Selection of the discharge pipe size is primarily a matter of economics. The cost of the various pipe sizes must be compared to the pump size and power cost required to overcome the resulting friction head.

The suction piping size and design is far more important. Many centrifugal pump troubles are caused by poor suction conditions.

The suction pipe should never be smaller than the suction connection of the pump and, in most cases, should be at least one size larger. Suction pipes should be as short and as straight as possible. Suction pipe velocities should be in the 5 to 8 feet per second range unless suction conditions are unusually good.

Higher velocities will increase the friction loss and can result in troublesome air or vapor separation. This is further complicated when elbows or tees are located adjacent to the pump suction nozzle, in that uneven flow patterns or vapor separation keeps the liquid from evenly filling the impeller. This upsets hydraulic balance leading to noise vibration, possible cavitation, and excessive shaft deflection. Cavitation erosion damage, shaft breakage or premature bearing failure may result.

On pump installations involving suction lift, air pockets in the suction line can be a source of trouble. The suction pipe should be exactly horizontal, or with a uniform slope upward from the sump to the pump as shown in Fig. 1. There should be no high spots where air can collect and cause the pump to lose its prime. Eccentric rather than concentric reducers should always be used.

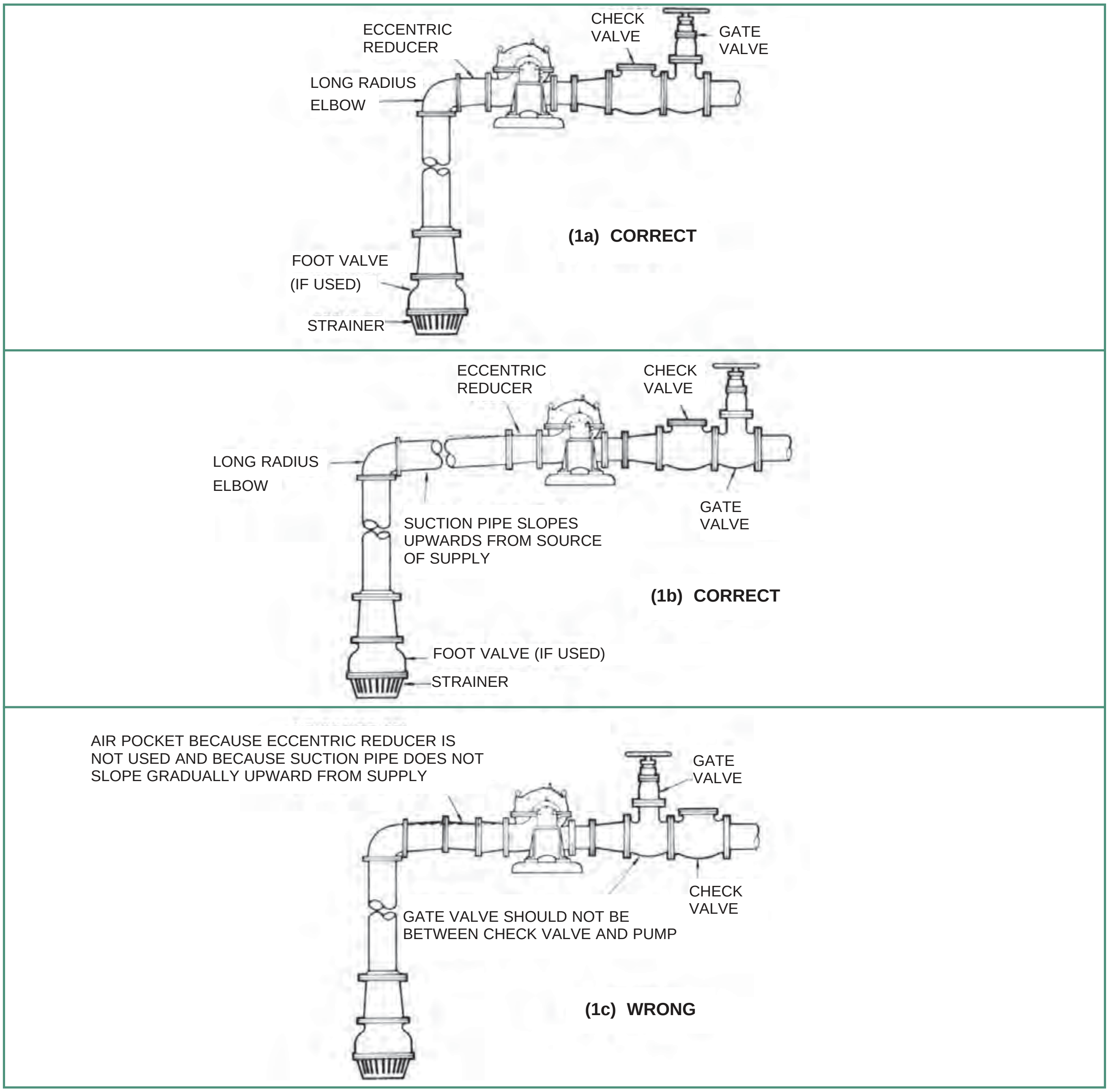


Fig. 1 Air Pockets in Suction Piping

If an elbow is required at the suction of a double suction pump, it should be in a vertical position if at all possible. Where it is necessary for some reason to use a horizontal elbow, it should be a long radius elbow and there should be a minimum of three diameters of straight pipe between the elbow and the pump as shown in Fig. 2, for low suction energy pumps, and five pipe diameters for high suction energy pumps. Fig. 3 shows the effect of an elbow directly on the suction. The liquid will flow toward the outside of the elbow and result in an uneven flow distribution into the two inlets of the double suction impeller. Noise and excessive axial thrust will result.

There are several important considerations in the design of a suction supply tank or sump. It is imperative that the amount of turbulence and entrained air be kept to a minimum. Entrained air may cause reduced capacity and efficiency as well as vibration, noise, shaft breakage, loss of prime, and/or accelerated corrosion.

The free discharge of liquid above the surface of the supply tank at or near the pump suction can cause entrained air to enter the pump. All lines should be submerged in the tank, and baffles should be used in extreme cases as shown in Fig. 4.

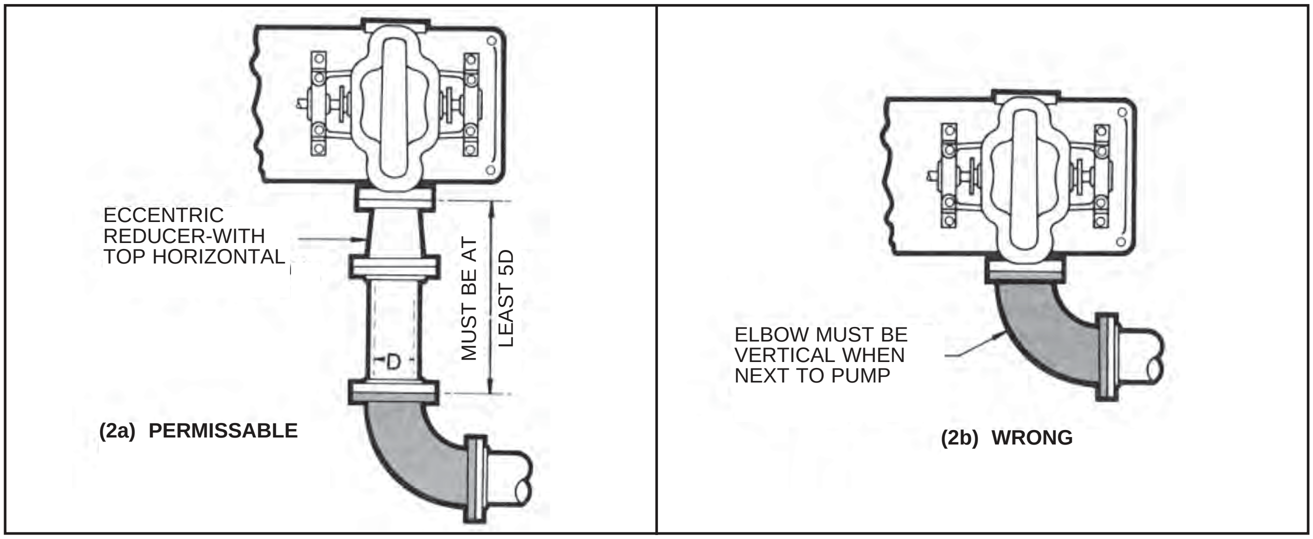


Fig. 2 Elbows At Pump Suction

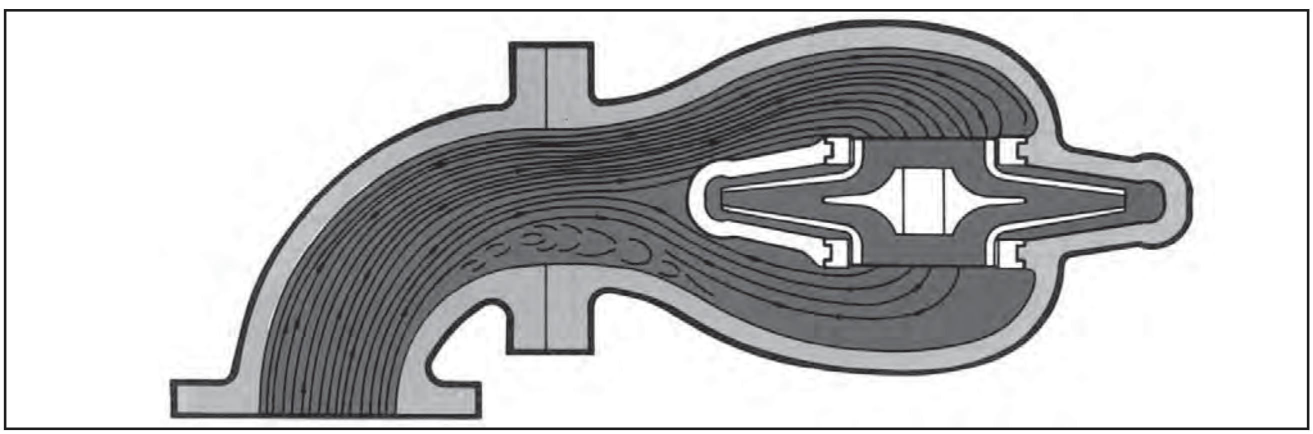


Fig. 3 Effect of Elbow Directly on Suction

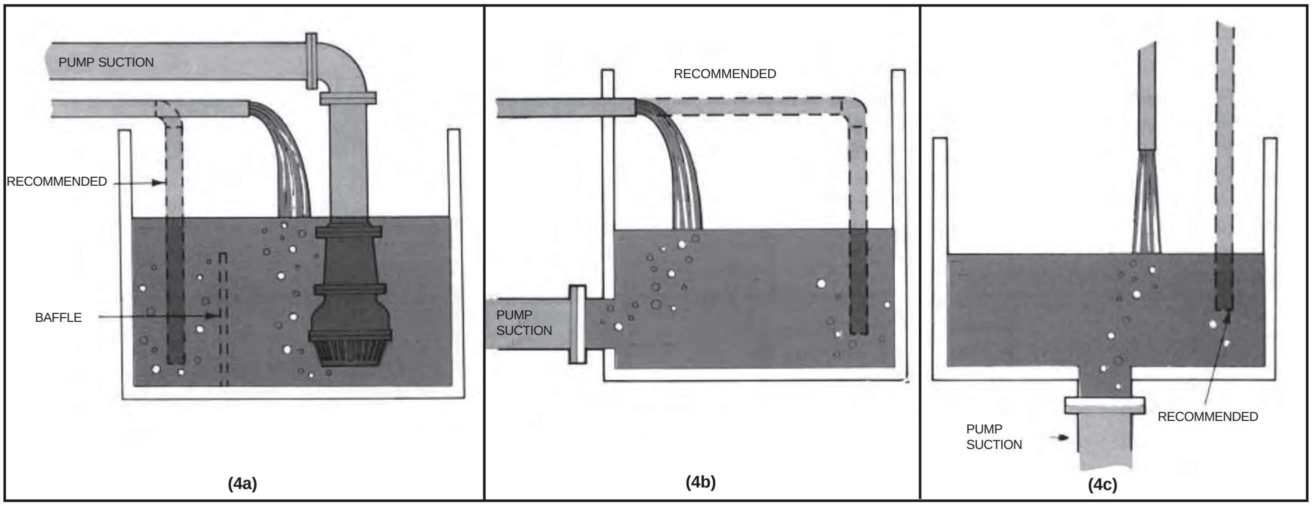


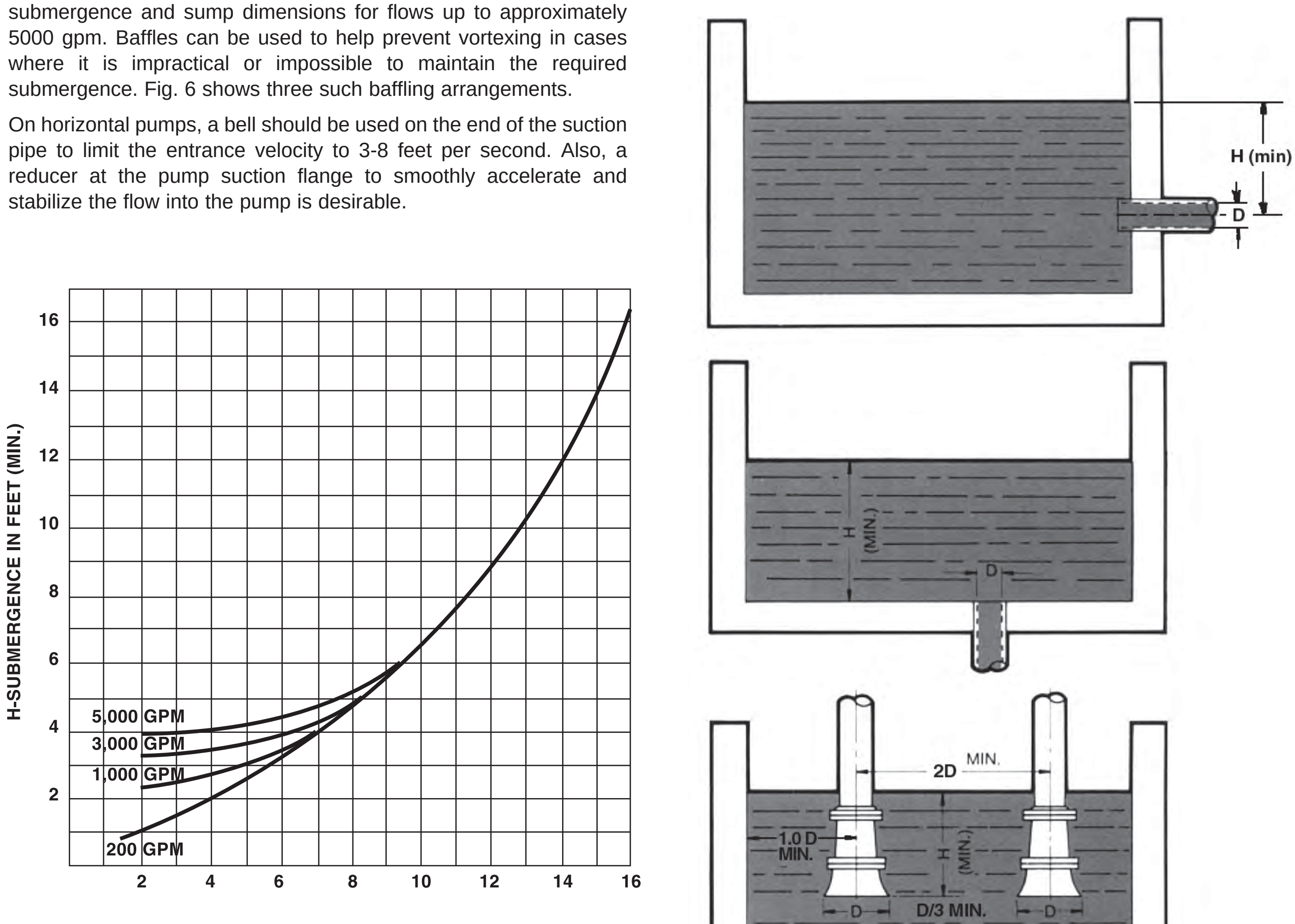
Fig. 4 Keeping Air Out of Pump

Improper submergence of the pump suction line can cause a vortex, which is a swirling funnel of air from the surface directly into the pump suction pipe. In addition to submergence, the location of the pipe in the sump and the actual dimensions of the sump are also important in preventing vortexing and/or excess turbulence.

For horizontal pumps, Fig. 5 can be used as a guide for minimum submergence and sump dimensions for flows up to approximately 5000 gpm. Baffles can be used to help prevent vortexing in cases where it is impractical or impossible to maintain the required submergence. Fig. 6 shows three such baffling arrangements.

On horizontal pumps, a bell should be used on the end of the suction pipe to limit the entrance velocity to 3-8 feet per second. Also, a reducer at the pump suction flange to smoothly accelerate and stabilize the flow into the pump is desirable.

The submergence of the suction pipe must also be carefully considered. The amount of submergence required depends upon the size and capacity of the individual pumps as well as on the sump design. Past experience is the best guide for determining the submergence. The pump manufacturer should be consulted for recommendations in the absence of other reliable data.



$$\text{VELOCITY IN FEET PER SEC.} = \frac{\text{QUAN. (G.P.M.) x .321}}{\text{AREA (inches)}^2} \text{ OR } \frac{\text{G.P.M. x .4085}}{D^2}$$

Fig. 5 Minimum Suction Pipe Submergence and Sump Dimensions

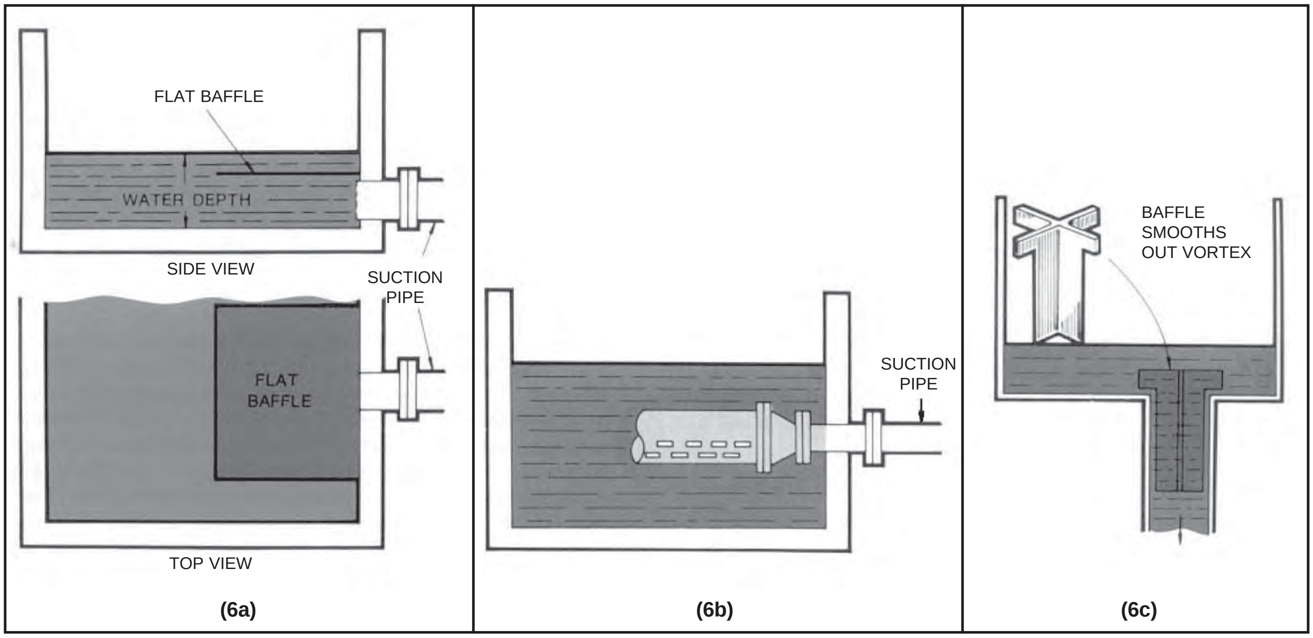


Fig. 6 Baffle Arrangements for Vortex Prevention

For larger units (over 5000 GPM) taking their suction supply for an intake sump (especially vertically submerged pumps), requires special attention.

The following section (Intake System Design) addresses these larger pumps.

INTAKE SYSTEM DESIGN

The function of the intake structure (whether it be an open channel, a fully wetted tunnel, a sump, or a tank) is to supply an evenly distributed flow to the pump suction. An uneven distribution of flow, characterized by strong local currents, can result in formation of surface or submerged vortices and with certain low values of submergence, may introduce air into the pump, causing a reduction of capacity, an increase in vibration and additional noise. Uneven flow distribution can also increase or decrease the power consumption with a change in total developed head.

The ideal approach is a straight channel coming directly to the pump or suction pipe. Turns and obstructions are detrimental, since they may cause eddy currents and tend to initiate deep-cored vortices.

The amount of submergence available is only one factor affecting vortex-free operation. It is possible to have adequate submergence and still have submerged vortices that may have an adverse effect on pump operation. Successful, vortex-free operation will depend greatly on the approach upstream of the sump.

Complete analysis of intake structures can only be accurately accomplished by scale model tests. Model testing is especially recommended for larger pumping units.

GENERAL DATA INFORMATION

Subject to the qualifications of the foregoing statements, Figures 7 through 10 have been constructed for single and multiple intake arrangements to provide guidelines for basic sump dimensions.

Since these values are composite averages for many pump types and cover the entire range of specific speeds, they are not absolute values but typical values subject to variations.

All of the dimensions In Figures 7 through 10 are based on the rated capacity of the pump. If operation at an increased capacity is to be undertaken for extended periods of time, the maximum capacity should be used for obtaining sump dimensions.

If the position of the back wall is determined structurally, dimension B in Figures 7 to 10 may become excessive and a false back wall should be installed.

Dimension S in Figures 7 and 9 is a minimum value based on the normal low water level at the pump or suction pipe bell, taking into consideration friction losses through the inlet screen and approach channel. Note that this dimension represents submergence at the intake, or the physical height of the water level above the intake relating to the prevention of eddy formations and vortexing.

The channel floor should be level for at least a distance Y (see Figures 7 through 10) upstream before any slope begins. The screen or gate widths should not be substantially less than W, and heights should not be less than the maximum anticipated water level to avoid overflow. Depending on the approach conditions before the sump, it may be necessary to construct straightening vanes in the approach channel, increase dimension A and/or conduct an intake model test to work out some other combination of these factors.

Dimension W is the width of an individual pump cell or the center-to-center distance of two pumps if no dividing wall is used.

On multiple intake installations, the recommended dimensions in Figures 7 and 8 apply as noted above, and the following additional factors should be considered.

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As shown in Fig. 10 (A), low velocity and straight in-line flow to all units simultaneously is a primary recommendation. Velocities in the sump should be approximately one foot per second, but velocities of two feet per second may prove satisfactory. This is particularly true when the design is based on a model study. Not recommended would be an abrupt change in the size of the inlet pipe to the sump or the inlet from one side introducing eddying.

In many cases, as shown in Fig. 10 (B), pumps operate satisfactorily without separating walls below 5,000 GPM. If walls must be used for structural purposes or some pumps operate intermittently, then the walls should extend from the rear wall approximately five times the D dimension given in Fig. 7.

If walls are used, increase dimension W by the thickness of the wall for correct centerline spacing and use round or ogive ends of walls. Not recommended is the placement of a number of pumps or suction pipes around the sides of a sump with or without dividing walls.

Abrupt changes in size, as shown in Fig. 10 (C), from inlet pipe or channel to the sump are not desirable. Connection of a pipe to a sump is best accomplished using a gradually increasing taper section. The angle should be as small as possible, preferably not more than 10 degrees. With this arrangement, sump velocities less than one foot per second are desirable.

Specifically not recommended is a pipe directly connected to a sump with suction intakes close to the sump inlet, since this results in an abrupt change in the flow direction. Centering pumps or suction pipes in the sump leaves large vortex areas behind the intake which will cause operational trouble.

If the sump velocity, as shown in Fig. 10 (D), can be kept low (approximately one foot per second), an abrupt change from inlet pipe to sump can be accommodated if the sump length equals or exceeds the values shown. As ratio Z/P increases, the inlet velocity at P may be increased up to an allowed maximum of eight feet per second at Z/P 10. Intakes “in line” are not recommended unless a trench-type of intake is provided (per ANSI/HI 9.8), or the ratio of sump to intake size is quite large and intakes are separated by a substantial margin longitudinally. A sump can generally be constructed at less cost by using a recommended design.

As shown in Fig. 10 (E), it is sometimes desirable to install pumps in tunnels or pipe lines. A drop pipe or false well to house the unit with a vaned inlet elbow facing upstream is satisfactory in flows up to eight feet per second. Without inlet elbow, the suction bell should be positioned at least two pipe (vertical) diameters above the top of the tunnel. The unit should not be suspended in the tunnel flow, unless the tunnel velocity Is less than two feet per second. There must be no air along the top of the tunnel, and the minimum submergence must be provided.

In general: Keep inlet velocity to the sump below two feet per second. Keep velocity in sump below 1.5 foot per second. Avoid changing direction of flow from inlet to pump or suction pipe, or change direction gradually and smoothly, guiding flow.

D	**=**	**$(.0744Q)^{0.5}$ Recommended**			
W	**=**	**2D**	**S**	**=**	**$D + 0.574Q/D^{1.5}$**
Y	**≥**	**4D**	**Where:**		
A	**≥**	**5D**	**S**	**-**	**inches**
C	**=**	**.3D to .5D**	**Q**	**-**	**Flow (GPM)**
B	**=**	**.75D**	**D**	**-**	**inches**

Fig. 7 Sump Dimensions

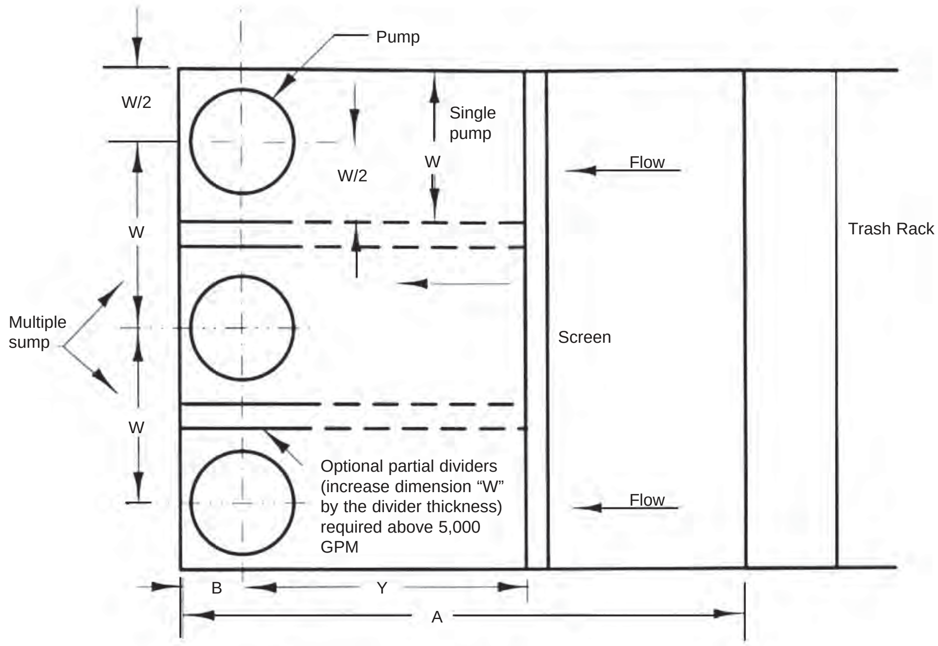


Fig. 8 Sump dimensions, plan view, wet pit type pumps

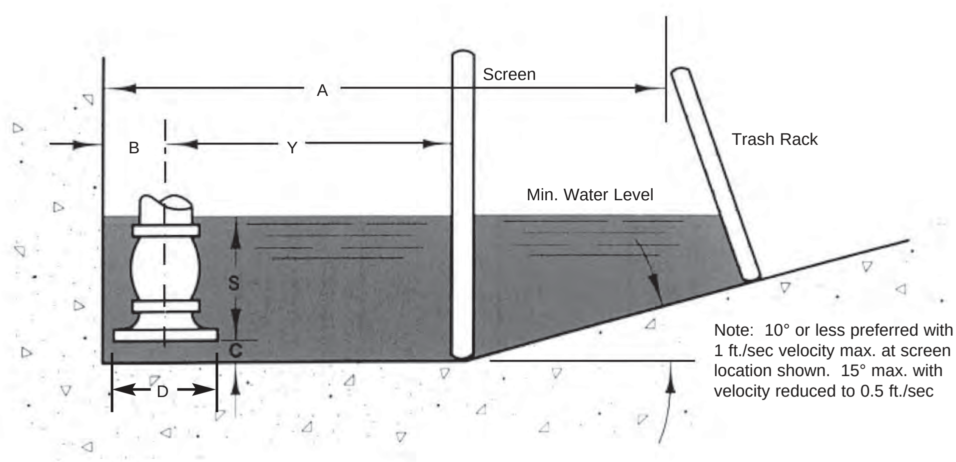
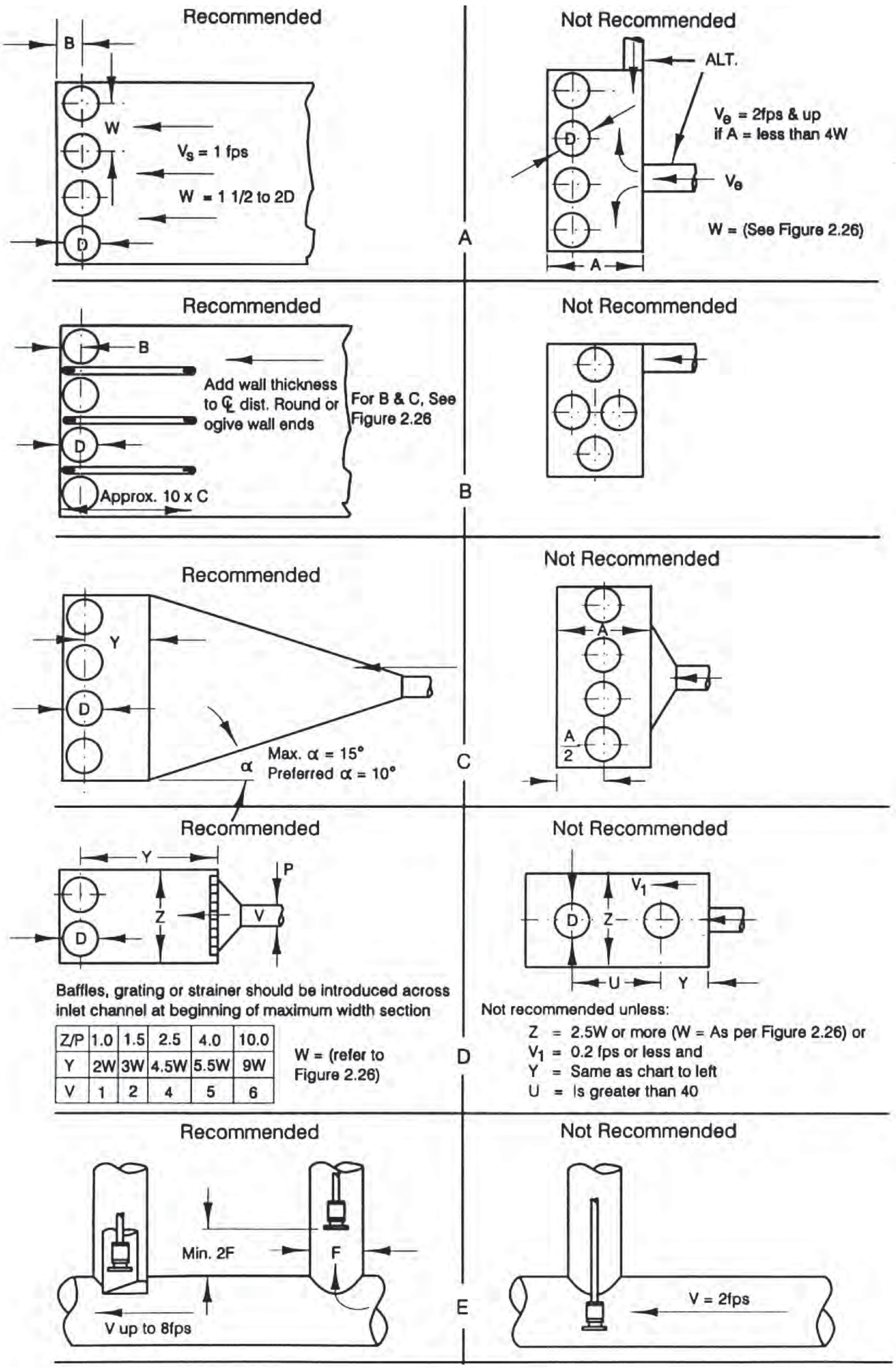


Fig. 9 Sump dimensions, elevation view, wet pit type pumps



Z/P	1.0	1.5	2.5	4.0	10.0
Y	2W	3W	4.5W	5.5W	9W
V	1	2	4	5	6

Fig. 10 Multiple pump installations

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TECH-B-4A Sealing

The proper selection of a seal is critical to the success of every pump application. For maximum pump reliability, choices must be made between the type of seal and the seal environment. In addition, a sealless pump is an alternative which would eliminate the need for a dynamic type seal entirely.

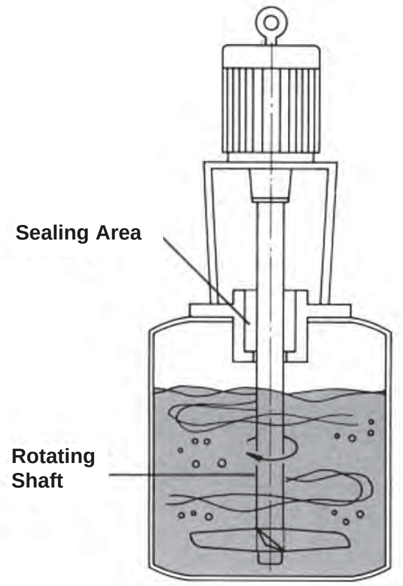


Fig. 1 Cross Section of Tank and Mixer

Sealing Basics

There are two basic kinds of seals: *static and dynamic*. Static seals are employed where no movement occurs at the juncture to be sealed. Gaskets and O-rings are typical static seals.

Dynamic seals are used where surfaces move relative to one another. Dynamic seals are used, for example, where a rotating shaft transmits power through the wall of a tank (Fig. 1), through the casing of a pump (Fig. 2), or through the housing of other rotating equipment such as a filter or screen.

A common application of sealing devices is to seal the rotating shaft of a centrifugal pump. To best understand how such a seal functions, a quick review of pump fundamentals is in order.

In a centrifugal pump, the liquid enters the suction of the pump at the center (eye) of the rotating impeller (Figures 3 and 4).

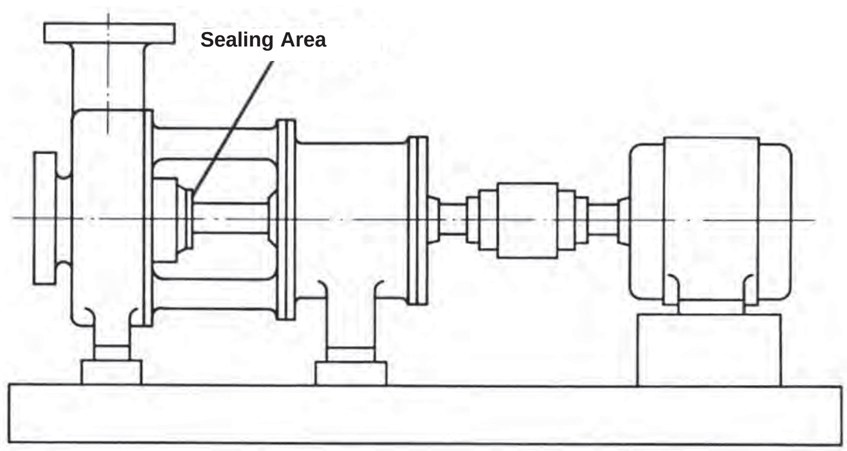


Fig. 2
Typical Centrifugal Pump

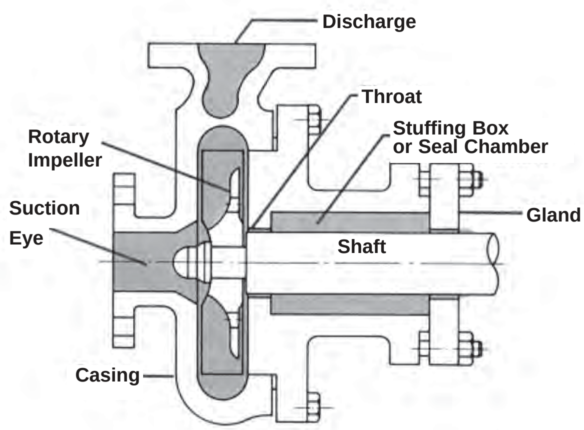


Fig. 3
Centrifugal Pump,
Liquid End

As the impeller vanes rotate, they transmit motion to the incoming product, which then leaves the impeller, collects in the pump casing, and leaves the pump under pressure through the pump discharge.

Discharge pressure will force some product down behind the impeller to the drive shaft, where it attempts to escape along the rotating drive shaft. Pump manufacturers use various design techniques to reduce the pressure of the product trying to escape. Such techniques include: 1) the addition of balance holes through the impeller to permit most of the pressure to escape into the suction side of the impeller, or 2) the addition of back pump-out vanes on the back side of the impeller.

However, as there is no way to eliminate this pressure completely, sealing devices are necessary to limit the escape of the product to the atmosphere. Such sealing devices are typically either compression packing or end-face mechanical seals.

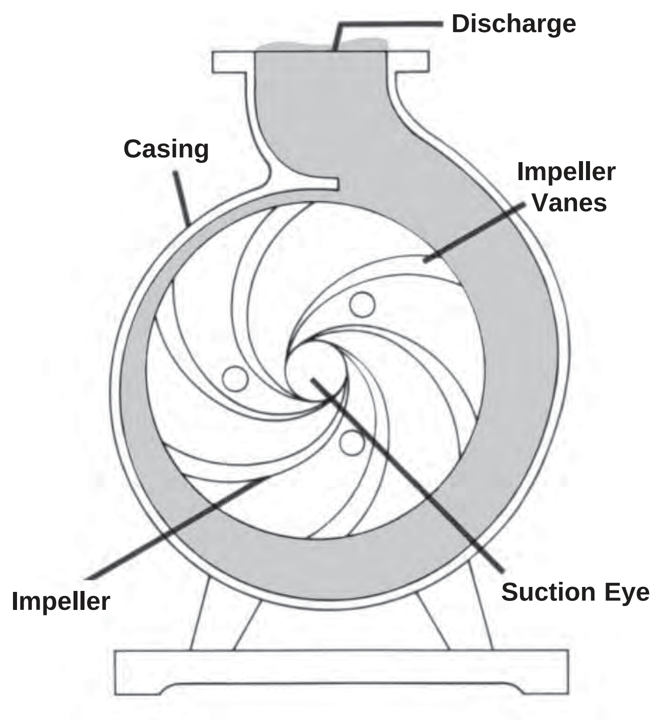


Fig. 4
Fluid Flow in a Centrifugal Pump

Stuffing Box Packing

A typical packed stuffing box arrangement is shown in Fig. 5. It consists of: A) Five rings of packing, B) A lantern ring used for the injection of a lubricating and/or flushing liquid, and C) A gland to hold the packing and maintain the desired compression for a proper seal.

The function of packing is to control leakage and not to eliminate it completely. The packing must be lubricated, and a flow from 40 to 60 drops per minute out of the stuffing box must be maintained for proper lubrication.

The method of lubricating the packing depends on the nature of the liquid being pumped as well as on the pressure in the stuffing box. When the pump stuffing box pressure is above atmospheric pressure and the liquid is clean and nonabrasive, the pumped liquid itself will lubricate the packing (Fig. 6). When the stuffing box pressure is below atmospheric pressure, a lantern ring is employed and lubrication is injected into the stuffing box (Fig. 7). A bypass line from the pump discharge to the lantern ring connection is normally used providing the pumped liquid is clean.

When pumping slurries or abrasive liquids, it is necessary to inject a clean lubricating liquid from an external source into the lantern ring (Fig. 8). A flow of from .2 to .5 gpm is desirable and a valve and flowmeter should be used for accurate control. The seal water pressure should be from 10 to 15 psi above the stuffing box pressure, and anything above this will only add to packing wear. The lantern ring is normally located in the center of the stuffing box. However, for extremely thick slurries like paper stock, it is recommended that the lantern ring be located at the stuffing box throat to prevent stock from contaminating the packing.

The gland shown in Figures 5 through 8 is a quench type gland. Water, oil, or other fluids can be injected into the gland to remove heat from the shaft, thus limiting heat transfer to the bearing frame. This permits the operating temperature of the pump to be higher than the limits of the bearing and lubricant design. The same quench gland can be used to prevent the escape of a toxic or volatile liquid into the air around the pump. This is called a smothering gland, with an external liquid simply flushing away the undesirable leakage to a sewer or waste receiver.

Today, however, stringent emission standards limit use of packing to non-hazardous water based liquids. This, plus a desire to reduce maintenance costs, has increased preference for mechanical seals.

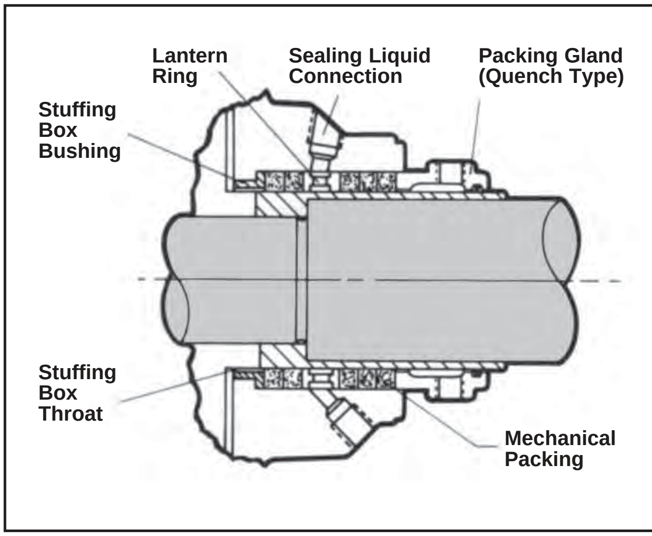


Fig. 5 Typical Stuffing Box Arrangement (Description of Parts)

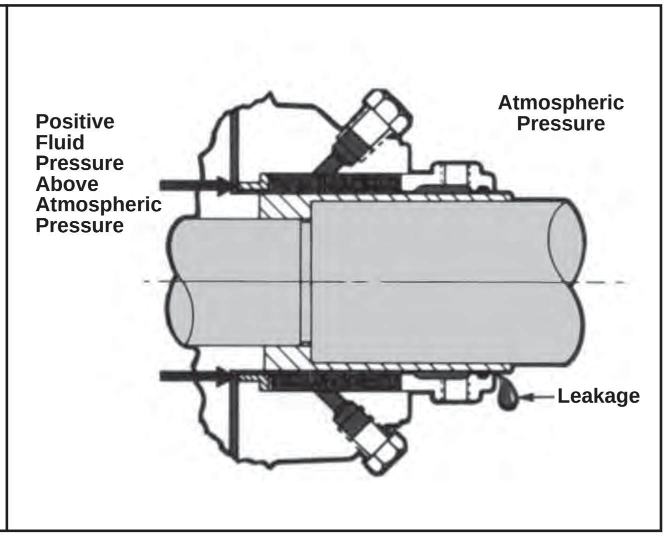


Fig. 6 Typical Stuffing Box Arrangement When Stuffing Box Pressure is Above Atmospheric Pressure

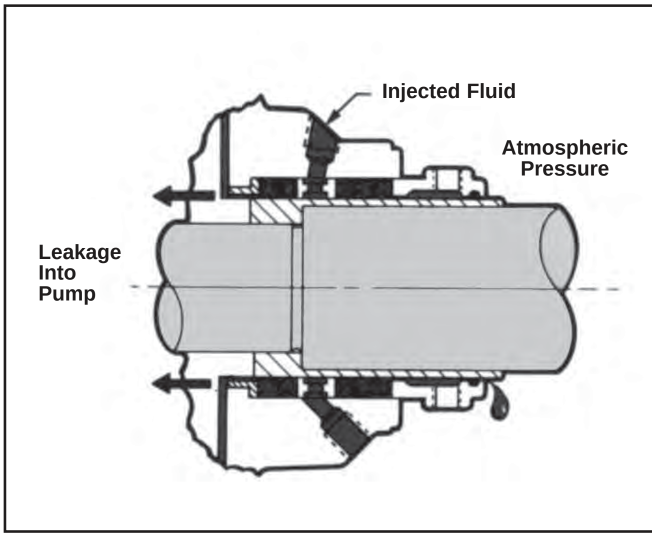


Fig. 7 Typical Stuffing Box Arrangement When Stuffing Box Pressure is Below Atmospheric Pressure

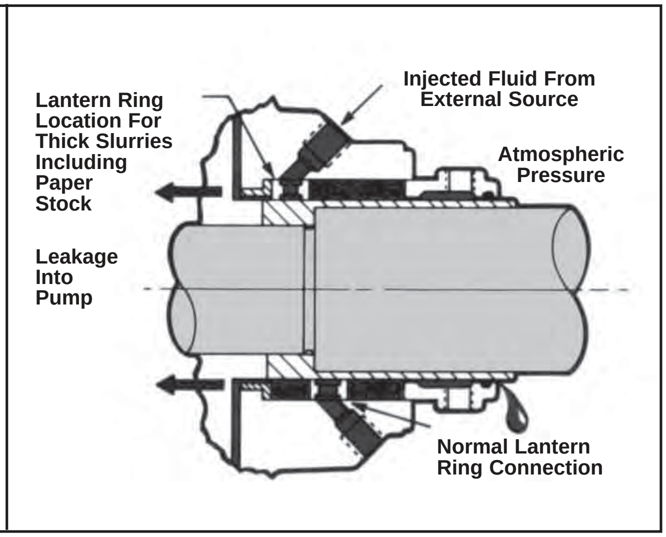


Fig. 8 Typical Stuffing Box Arrangement When Pumping Slurries

Mechanical Seals

A mechanical seal is a sealing device which forms a running seal between rotating and stationary parts. They were developed to overcome the disadvantages of compression packing. Leakage can be reduced to a level meeting environmental standards of government regulating agencies and maintenance costs can be lower. Advantages of mechanical seals over conventional packing are as follows:

1. Zero or limited leakage of product (meet emission regulations.)
2. Reduced friction and power loss.
3. Elimination of shaft or sleeve wear.
4. Reduced maintenance costs.
5. Ability to seal higher pressures and more corrosive environments.
6. The wide variety of designs allows use of mechanical seals in almost all pump applications.

The Basic Mechanical Seal

All mechanical seals are constructed of three basic sets of parts as shown in Fig. 9:

1. A set of primary seal faces: one rotary and one stationary...shown in Fig. 9 as seal ring and insert.
2. A set of secondary seals known as shaft packings and insert mountings such as O-rings, wedges and V-rings.
3. Mechanical seal hardware including gland rings, collars, compression rings, pins, springs and bellows.

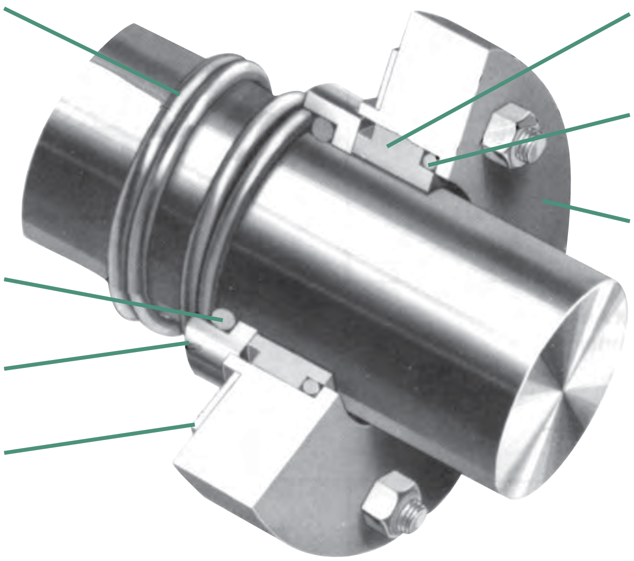


Fig. 9 A Simple Mechanical Seal

How A Mechanical Seal Works

The *primary seal* is achieved by two very flat, lapped faces which create a difficult leakage path perpendicular to the shaft. Rubbing contact between these two flat mating surfaces minimizes leakage. As in all seals, one face is held *stationary* in a housing and the other face is fixed to, and *rotates* with, the shaft. One of the faces is usually a non-galling material such as *carbon-graphite*. The other is usually a relatively hard material like *silicon-carbide*. Dissimilar materials are usually used for the stationary Insert and the rotating seal ring face in order to prevent adhesion of the two faces. The softer face usually has the smaller mating surface and is commonly called the *wear nose.*

There are four main sealing points within an end face mechanical seal (Fig. 10). The primary seal is at the seal face, Point A. The leakage path at Point B is blocked by either an O-ring, a V-ring or a wedge. Leakage paths at Points C and D are blocked by gaskets or O-rings.

The faces in a typical mechanical seal are lubricated with a boundary layer of gas or liquid between the faces. In designing seals for the desired leakage, seal life, and energy consumption, the designer must consider how the faces are to be lubricated and select from a number of modes of seal face lubrication.

To select the best seal design, it's necessary to know as much as possible about the operating conditions and the product to be sealed. Complete information about the product and environment will allow selection of the best seal for the application.

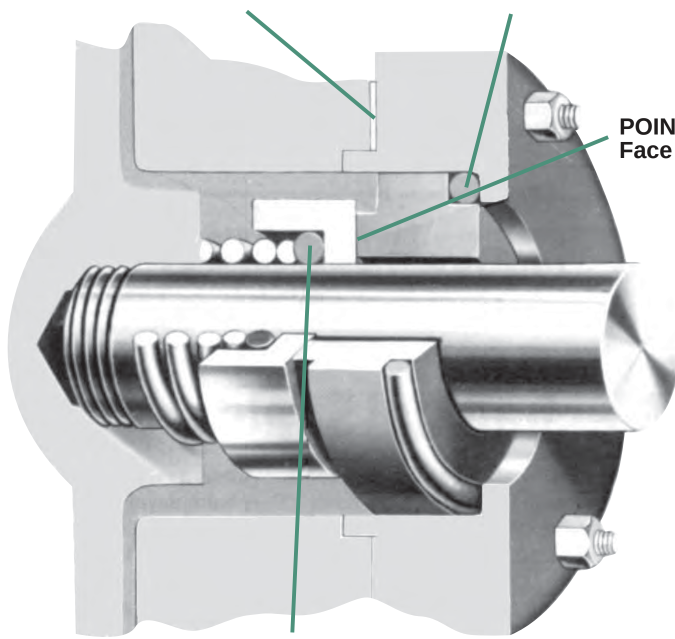


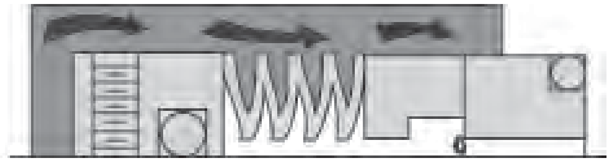
Fig. 10 Sealing Points for Mechanical Seal

Mechanical Seal Types

Mechanical seals can be classified into several types and arrangements:

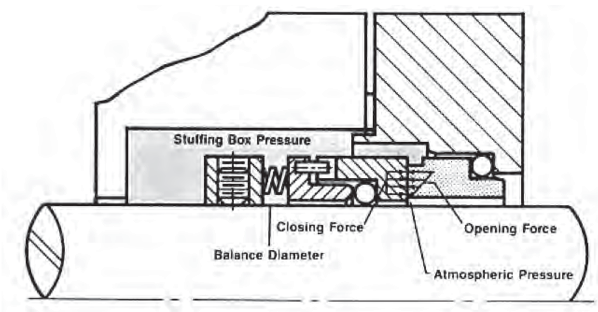
PUSHER:

Incorporate secondary seals that move axially along a shaft or sleeve to maintain contact at the seal faces. This feature compensates for seal face wear and wobble due to misalignment. The pusher seals advantage is that it's inexpensive and commercially available in a wide range of sizes and configurations. Its disadvantage is that it's prone to secondary seal hang-up and fretting of the shaft or sleeve. Examples are Dura RO and Crane Type 9T.



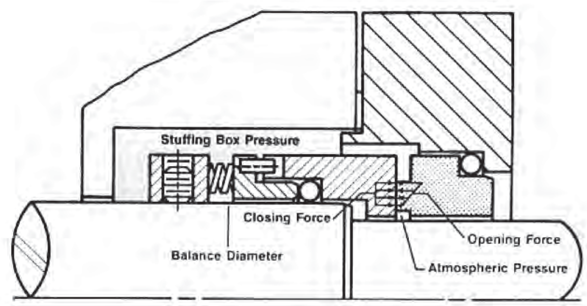
NON-PUSHER:

The non-pusher or bellows seal does not have to move along the shaft or sleeve to maintain seal face contact. The main advantages are its ability to handle high and low temperature applications, and does not require a secondary seal (not prone to secondary seal hang-up). A disadvantage of this style seal is that its thin bellows cross sections must be upgraded for use in corrosive environments. Examples are Dura CBR and Crane 215, and Sealol 680.



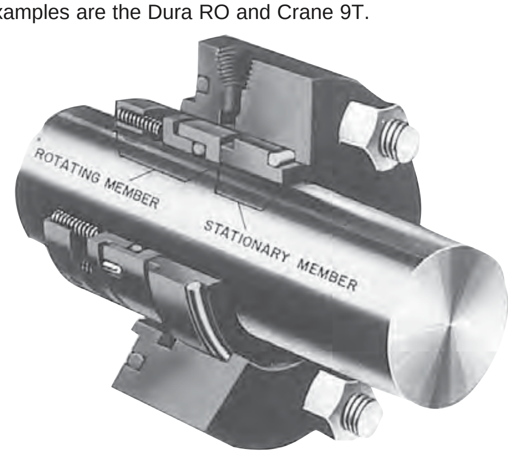
UNBALANCED:

They are inexpensive, leak less, and are more stable when subjected to vibration, misalignment, and cavitation. The disadvantage is their relative low pressure limit. If the closing force exerted on the seal faces exceeds the pressure limit, the lubricating film between the faces is squeezed out and the highly loaded dry running seal fails. Examples are the Dura RO and Crane 9T.



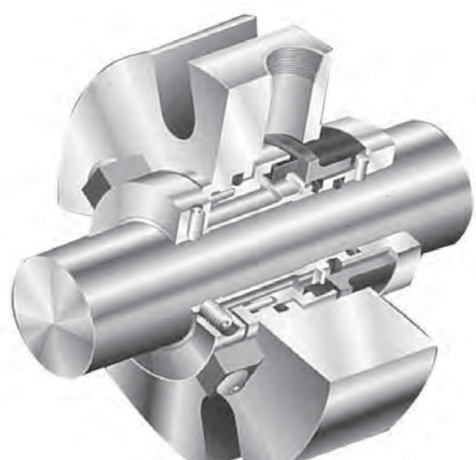
BALANCED:

Balancing a mechanical seal involves a simple design change which reduces the hydraulic forces acting to close the seal faces. Balanced seals have higher pressure limits, lower seal face loading, and generate less heat. This makes them well suited to handle liquids with poor lubricity and high vapor pressures such as light hydrocarbons. Examples are Dura CBR and PBR and Crane 98T and 215.



CONVENTIONAL:

Examples are the Dura RO and Crane Type 1 which require setting and alignment of the seal (single, double, tandem) on the shaft or sleeve of the pump. Although setting a mechanical seal is relatively simple, today's emphasis on reducing maintenance costs has increased preference for cartridge seals.



CARTRIDGE:

Examples are Dura P-50 and Crane 1100 which have the mechanical seal premounted on a sleeve including the gland and fit directly over the Model 3196 shaft or shaft sleeve (available single, double, tandem). The major benefit, of course, is no requirement for the usual seal setting measurements for their installation. Cartridge seals lower maintenance costs and reduce seal setting errors.

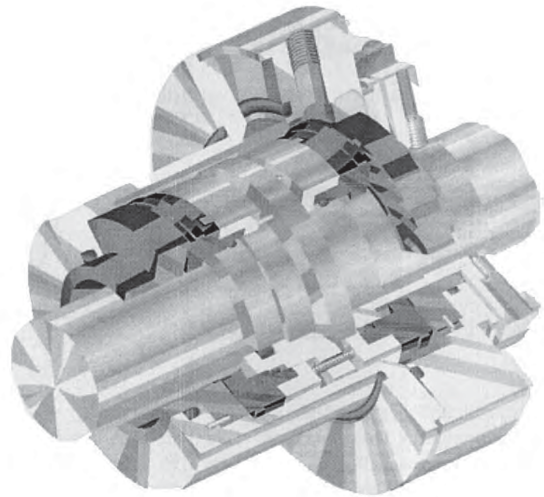
Mechanical Seal Arrangements

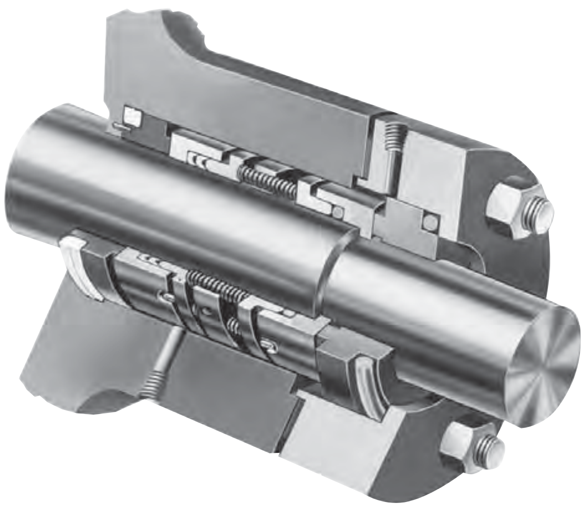
SINGLE INSIDE:

This is the most common type of mechanical seal. These seals are easily modified to accommodate seal flush plans and can be balanced to withstand high seal environment pressures. Recommended for relatively clear non-corrosive and corrosive liquids with satisfactory lubricating properties where cost of operation does not exceed that of a double seal. Examples are Dura RO and CBR and Crane 9T and 215. Reference Conventional Seal.

SINGLE OUTSIDE:

If an extremely corrosive liquid has good lubricating properties, an outside seal offers an economical alternative to the expensive metal required for an inside seal to resist corrosion. The disadvantage is that it is exposed outside of the pump which makes it vulnerable to damage from impact and hydraulic pressure works to open the seal faces so they have low pressure limits (balanced or unbalanced).





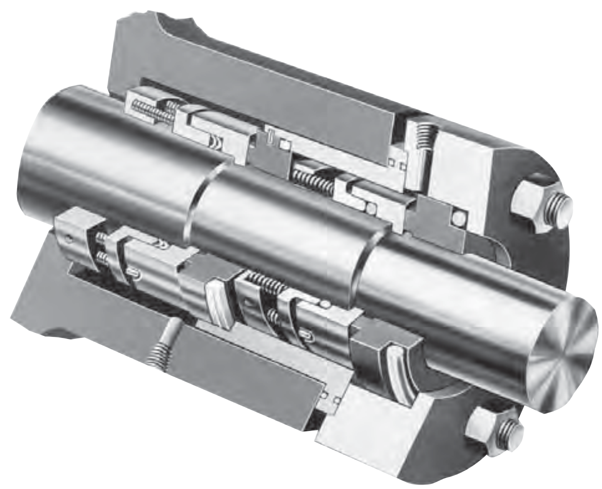
DOUBLE (DUAL PRESSURIZED):

This arrangement is recommended for liquids that are not compatible with a single mechanical seal (i.e. liquids that are toxic, hazardous [regulated by the EPA], have suspended abrasives, or corrosives which require costly materials). The advantages of the double seal are that it can have five times the life of a single seal in severe environments. Also, the metal inner seal parts are never exposed to the liquid product being pumped, so viscous, abrasive, or thermosetting liquids are easily sealed without a need for expensive metallurgy. In addition, recent testing has shown that double seal life is virtually unaffected by process upset conditions during pump operation. A significant advantage of using a double seal over a single seal.

The final decision between choosing a double or single seal comes down to the initial cost to purchase the seal, cost of operation of the seal, and environmental and user plant emission standards for leakage from seals. Examples are Dura double RO and X-200 and Crane double 811T.

DOUBLE GAS BARRIER (PRESSURIZED DUAL GAS):

Very similar to cartridge double seals...sealing involves an inert gas, like nitrogen, to act as a surface lubricant and coolant in place of a liquid barrier system or external flush required with conventional or cartridge double seals. This concept was developed because many barrier fluids commonly used with double seals can no longer be used due to new emission regulations. The gas barrier seal uses nitrogen or air as a harmless and inexpensive barrier fluid that helps prevent product emissions to the atmosphere and fully complies with emission regulations. The double gas barrier seal should be considered for use on toxic or hazardous liquids that are regulated or in situations where increased reliability is the required on an application. Examples are Dura GB200, GF200, and Crane 2800.



TANDEM (DUAL UNPRESSURIZED):

Due to health, safety, and environmental considerations, tandem seals have been used for products such as vinyl chloride, carbon monoxide, light hydrocarbons, and a wide range of other volatile, toxic, carcinogenic, or hazardous liquids.

Tandem seals eliminate icing and freezing of light hydrocarbons and other liquids which could fall below the atmospheric freezing point of water in air (32°F or 0°C). (Typical buffer liquids in these applications are ethylene glycol, methanol, and propanol.) A tandem also increases online reliability. If the primary seal fails, the outboard seal can take over and function until maintenance of the equipment can be scheduled. Examples are Dura TMB-73 and tandem PTO.

Mechanical Seal Selection

The proper selection of a mechanical seal can be made only if the full operating conditions are known:

1. **Liquid**
2. **Pressure**
3. **Temperature**
4. **Characteristics of Liquid**
5. **Reliability and Emission Concerns**

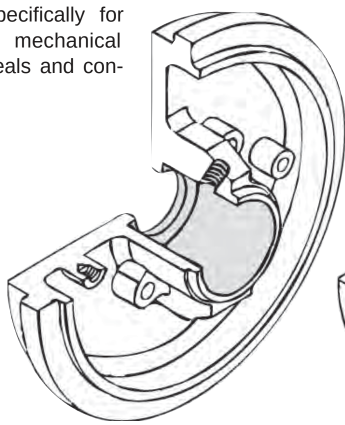
1. **Liquid.** Identification of the exact liquid to be handled is the first step in seal selection. The metal parts must be corrosion resistant, usually steel, bronze, stainless steel, or Hastelloy. The mating faces must also resist corrosion and wear. Carbon, ceramic, silicon carbide or tungsten carbide may be considered. Stationary sealing members of Buna, EPR, Viton and Teflon are common.
2. **Pressure.** The proper type of seal, balanced or unbalanced, is based on the pressure on the seal and on the seal size.
3. **Temperature.** In part, determines the use of the sealing members. Materials must be selected to handle liquid temperature.
4. **Characteristics of Liquid.** Abrasive liquids create excessive wear and short seal life. Double seals or clear liquid flushing from an external source allow the use of mechanical seals on these difficult liquids. On light hydrocarbons balanced seals are often used for longer seal life even though pressures are low.
5. **Reliability and Emission Concerns.** The seal type and arrangement selected must meet the desired reliability and emission standards for the pump application. Double seals and double gas barrier seals are becoming the seals of choice.

Seal Environment

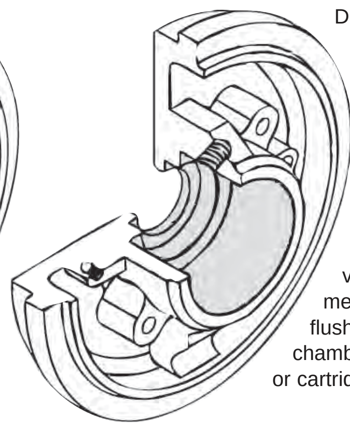
The number one cause of pump downtime is failure of the shaft seal. These failures are normally the result of an unfavorable seal environment such as improper heat dissipation (cooling), poor lubrication of seal faces, or seals operating in liquids containing solids, air or vapors. To achieve maximum reliability of a seal application, proper choices of seal housings (standard bore stuffing box, large bore, or large tapered bore seal chamber) and seal environmental controls (CPI and API seal flush plans) must be made.

STANDARD BORE STUFFING BOX COVER

Designed thirty years ago specifically for packing. Also accommodates mechanical seals (clamped seat outside seals and conventional double seals.)



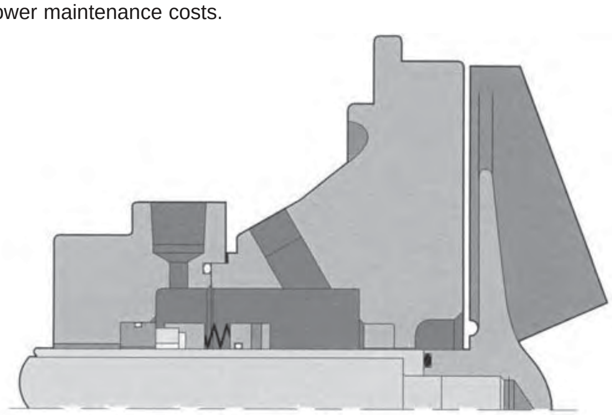
CONVENTIONAL LARGE BORE SEAL CHAMBER



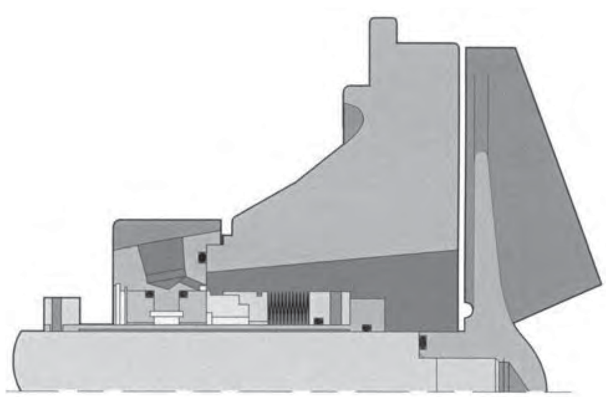
Designed specifically for mechanical seals. Large bore provides increased life of seals through improved lubrication and cooling of faces. Seal environment should be controlled through use of CPI or API flush plans. Often available with internal bypass to provide circulation of liquid to faces without using external flush. Ideal for conventional or cartridge single mechanical seals in conjunction with a flush and throat bushing in bottom of chamber. Also excellent for conventional or cartridge double or tandem seals.

LARGE BORE SEAL CHAMBERS

Introduced in the mid-80's, enlarged bore seal chambers with increased radial clearance between the mechanical seal and seal chamber wall, provide better circulation of liquid to and from seal faces. Improved lubrication and heat removal (cooling) of seal faces extend seal life and lower maintenance costs.



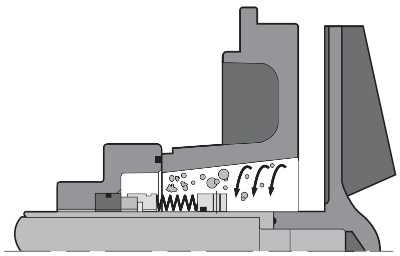
BigBore™ Seal Chamber



TaperBore™ Seal Chamber

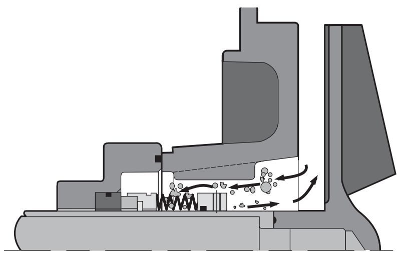
Large Tapered Bore Seal Chambers

Provide increased circulation of liquid at seal faces without use of external flush. Offers advantages of lower maintenance costs, elimination of tubing/piping, lower utility costs (associated with seal flushing) and extended seal reliability. The tapered bore seal chamber is commonly available with ANSI chemical pumps. API process pumps use conventional large bore seal chambers. Paper stock pumps use both conventional large bore and large tapered bore seal chambers. Only tapered bore seal chambers with flow modifiers provide expected reliability on services with or without solids, air or vapors.



Conventional Tapered Bore Seal Chamber:
Mechanical Seals Fail When Solids or Vapors Are Present in Liquid

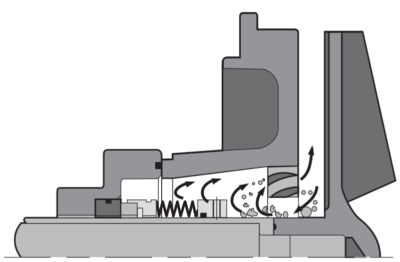
Many users have applied the conventional tapered bore seal chamber to improve seal life on services containing solids or vapors. Seals in this environment failed prematurely due to entrapped solids and vapors. Severe erosion of seal and pump parts, damaged seal faces and dry running were the result.



Modified Tapered Bore Seal Chamber with Axial Ribs:
Good for Services Containing Air, Minimum Solids

This type of seal chamber will provide better seal life when air or vapors are present in the liquid. The axial ribs prevent entrapment of vapors through improved flow in the chamber. Dry running failures are eliminated. In addition, solids less than 1% are not a problem.

The new flow pattern, however, still places the seal in the path of solids/liquid flow. The consequence on services with significant solids (greater than 1%) is solids packing the seal spring or bellows, solids impingement on seal faces and ultimate seal failure.

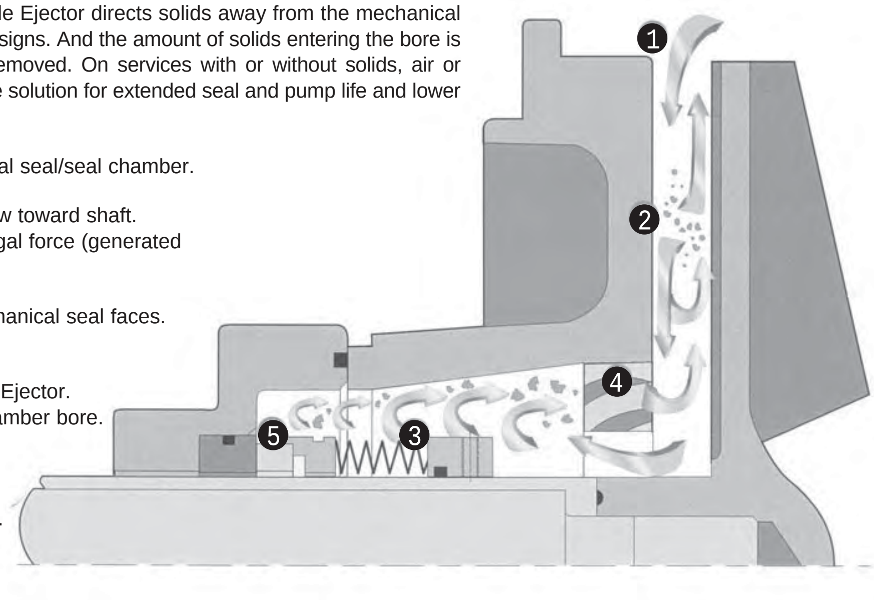


Goulds Standard TaperBore™ *PLUS* Seal Chamber:
The Best Solution for Services Containing Solids and Air or Vapors

To eliminate seal failures on services containing vapors as well as solids, the flow pattern must direct solids away from the mechanical seal, and purge air and vapors. Goulds Standard TaperBore™ *PLUS* completely reconfigures the flow in the seal chamber with the result that seal failures due to solids are eliminated. Air and vapors are efficiently removed eliminating dry run failures. Extended seal and pump life with lower maintenance costs are the results.

Goulds TaperBore™ *Plus*: How It Works

The unique flow path created by the Vane Particle Ejector directs solids away from the mechanical seal, not at the seal as with other tapered bore designs. And the amount of solids entering the bore is minimized. Air and vapors are also efficiently removed. On services with or without solids, air or vapors, Goulds TaperBore™ *PLUS* is the effective solution for extended seal and pump life and lower maintenance costs.



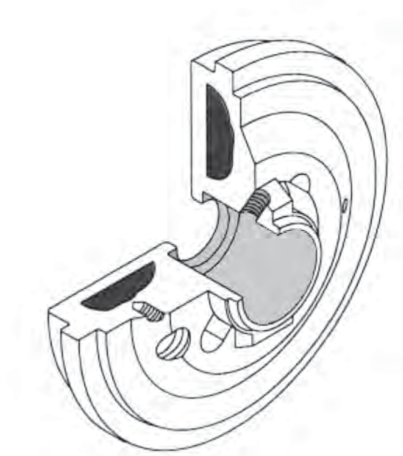
❶ Solids/liquid mixture flows toward mechanical seal/seal chamber.

❷ Turbulent zone. Some solids continue to flow toward shaft. Other solids are forced back out by centrifugal force (generated by back pump-out vanes).

❸ Clean liquid continues to move toward mechanical seal faces. Solids, air, vapors flow away from seal.

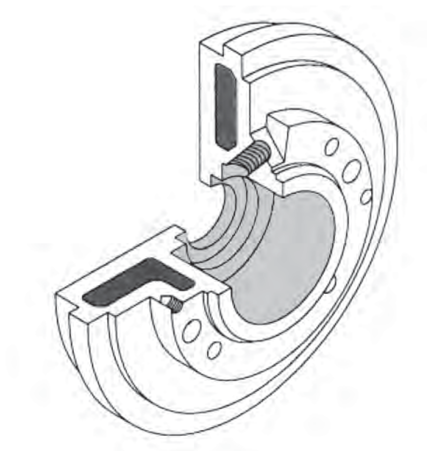
❹ Low pressure zone create by Vane Particle Ejector. Solids, air, vapor liquid mixture exit seal chamber bore.

❺ Flow in TaperBore™ *PLUS* seal chamber assures efficient heat removal (cooling) and lubrication. Seal face heat is dissipated. Seal faces are continuously flushed with clean liquid.



JACKETED STUFFING BOX COVER

Designed to maintain proper temperature control (heating or cooling) of seal environment. (Jacketed covers do not help lower seal face temperatures to any significant degree). Good for high temperature services that require use of a conventional double seal or single seal with a flush and API or CPI plan 21.



JACKETED LARGE BORE SEAL CHAMBER

Maintains proper temperature control (heating or cooling) of seal environment with improved lubrication of seal faces. Ideal for controlling temperature for services such as molten sulfur and polymerizing liquids. Excellent for high temperature services that require use of conventional or cartridge single mechanical seals with flush and throat bushing in bottom of seal chamber. Also, great for conventional or cartridge double or tandem seals.

Stuffing Box Cover and Seal Chamber Guides

The following two selection guides are designed to assist selection of the proper seal housing for a pump application.

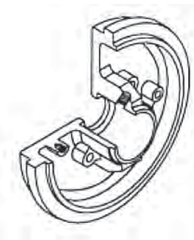
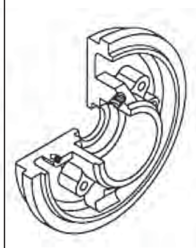
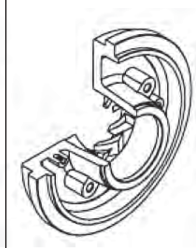
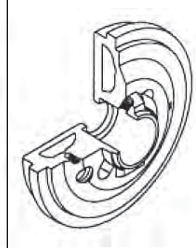
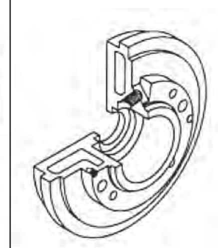
Stuffing Box and Seal Chamber Application Guide

Stuffing Box Cover Seal Chamber	Application
Standard Bore Stuffing Box Cover	Use for soft packing. Outside mechanical seals. Double seals. Also, accommodates other mechanical seals.
Jacketed Stuffing Box Cover	Same as above, but used in high temperature applications when the temperature of the seal area needs to be controlled.
Conventional Large Bore	Use for all mechanical seal applications where the seal environment requires use of CPI or API seal flush pans. Cannot be used with outside type mechanical seals
Jacketed Large Bore	Same as Large Bore but also need to control temperature of liquid in seal area.
Tapered Large Bore with Axial Ribs	Clean services that require use of single mechanical seals. Can also be used with cartridge double seals. Also, effective on services with light solids up to 1% by weight. Paper stock to 1% by weight.
Tapered Large Bore with Patented Vane Particle Ejector (Alloy Construction)	Services with light to moderate solids up to 10% by weight. Paper stock to 5% by weight. Ideal for single mechanical seals. No flush required. Also, accommodates cartridge double seals. Cannot be used with outside mechanical seals.

Selection Guide

Goulds engineered seal chambers provide best seal environment for selected sealing arrangements/services.

A	Ideally Suited
B	Acceptable
C	Not Recommended

Service Acceptable Ideally Suited	**TYPE 1** **Standard Bore Stuffing Box Cover** *Designed for packing. Also accommodates mechanical seals.*	**TYPE 2** **Conventional Large Bore** *Enlarged chamber for increased seal life through improved lubrication and cooling. Seal environment should be controlled through use of CPI flush plans.*	**TYPE 3** **Tapered Bore** *Lower seal face temperatures, self-venting and draining. Solids and vapors circulated away from seal faces. Often no flush required. Superior patented design maximizes seal life with or without solids and vapor in liquid.*	**TYPE 4** **Jacketed Stuffing Box** *Maintains proper temperature control (heating or cooling) of seal environment.*	**TYPE 5** **Jacketed Large Bore** *Maintains proper temperature control (heating or cooling) of seal environment with improved lubrication of seal faces. Ideal for controlling temperatures on services such as molten sulfur and polymerizing liquids.*
					
Ambient Water With Flush	A	A	A	-	-
Entrained Air or Vapor	C	B	A	C	B
Solids 0-10%, No Flush	C	C	A	C	C
Solids up to and greater than 10% With Flush	B	A	A	B	A
Paper Stock 0-5%, With No Flush	C	C	A	-	-
Paper Stock 0-5%, With Flush	B	A	A	-	-
Slurries 0-5%, No Flush	C	C	A	C	C
High Boiling Point Liquids, no flush	C	C	A	C	C
Temperature Control	C	C	C	B	A
Self-Venting and Draining	C	C	A	C	C
Seal Face Heat Removal	C	A	A	C	A
Molten or Polymerizing Liquid, No Flush	C	C	B	C	C
Molten or Polymerizing Liquid With Flush	C	B	B	C	A

Environmental Controls

Environmental controls are necessary for reliable performance of a mechanical seal on many applications. Goulds Pumps and the seal vendors offer a variety of arrangements to combat these problems:

1. Corrosion
2. Temperature Control
3. Dirty or Incompatible Environments

CORROSION

Corrosion can be controlled by selecting seal materials that are not attacked by the pumpage. When this is difficult, external fluid injection of a non-corrosive chemical to lubricate the seal is possible. Single or double seals could be used, depending on if the customer can stand delusion of his product.

TEMPERATURE CONTROL

As the seal rotates, the faces are in contact. This generates heat and if this heat is not removed, the temperature in the stuffing box or seal chamber can increase and cause sealing problems. A simple by-pass of product over the seal faces will remove the heat generated by the seal (Fig. 25). For higher temperature services, by-pass of product through a cooler may be required to cool the seal sufficiently (Fig. 26). External cooling fluid injection can also be used.

DIRTY or INCOMPATIBLE ENVIRONMENTS

Mechanical seals do not normally function well on liquids which contain solids or can solidify on contact with the atmosphere. Here, by-pass flush through a filter, a cyclone separator or a strainer are methods of providing a clean fluid to lubricate seal faces.

Strainers are effective for particles larger than the openings on a 40 mesh screen.

Cyclone separators are effective on solids 10 micron or more in diameter, if they have a specific gravity of 2.7 and the pump develops a differential pressure of 30-40 psi. Filters are available to remove solids 2 microns and larger.

If external flush with clean liquid is available, this is the most fail proof system. Lip seal or restricting bushings are available to control flow of injected fluid to flows as low as $^{1}/_{8}$ GPM.

Quench type glands are used on fluids which tend to crystallize on exposure to air. Water or steam is put through this gland to wash away any build up. Other systems are available as required by the service.

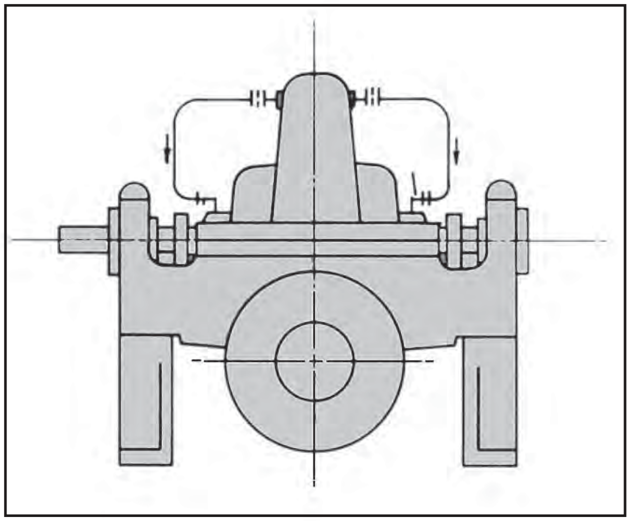


Fig. 25

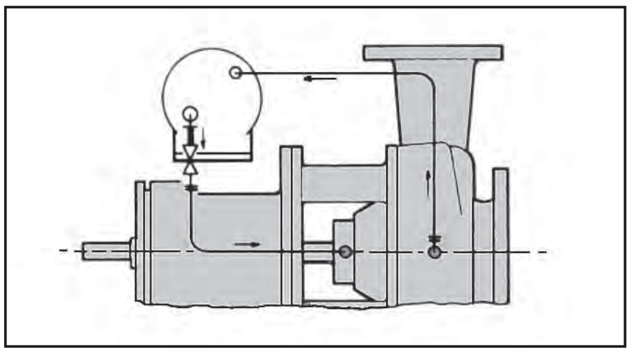


Fig. 26

API and CPI Plans

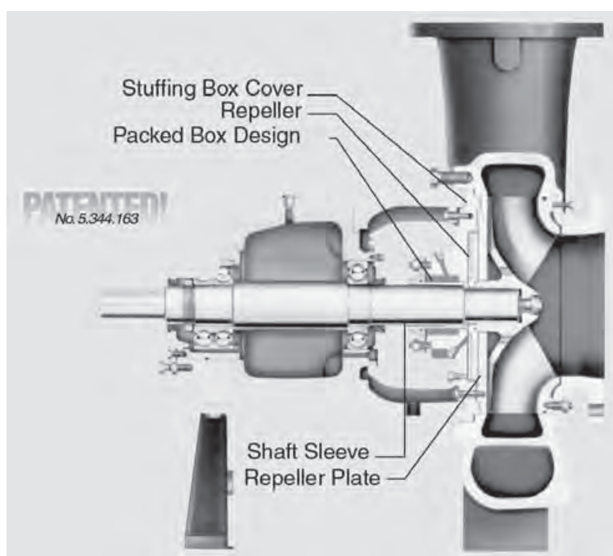
API and CPI mechanical seal flush plans are commonly used with API and CPI process pumps. The general arrangement of the plans are similar regardless of the designation whether API or CPI. The difference between the flush plans is the construction which provides applicable pressure-temperature capability for each type of pump. API plans have higher pressure and temperature capability than CPI plans. Each plan helps provide critical lubrication and cooling of seal faces to maximize seal reliability.

Plan No.	Recommended Applications
01	Single mechanical seals and TDH less then 125 feet.
02	Used with some outside seals. In most cases not recommended.
11	Single and tandem seals. Always consider a plan 11 with balanced seals. Apply when TDH is greater than 125 ft.
12	Same application as 11. Additionally, a 12 will strain particles from the flush liquid. This helps prevent solid impingement on seal faces.
13	Single and tandem seals. Use when difference in pressure between the seal chamber or stuffing box and pump suction exceed 35 psi.
21	Single and tandem seals. Required when the flush needs to be cooled before flushing at the seal faces. (ex. water above 200°F, light hydrocarbons or any other liquids with poor lubricating qualities and high vapor pressures.)
22	Same application as 21. Additionally, a plan 22 will strain particles from the flush liquid. This helps prevent solid impingement on seal faces.
23	Single and tandem seals. Use when difference in pressure between the seal chamber or stuffing box and pump suction exceed 35 psi. 3600 RPM only.
31	Single and tandem seals. Apply when strainers are inadequate to clean flushing liquid.
32	Single and tandem seals. Required when pumpage is not suitable to lubricate seal faces. Use of bushing or lip seal is also recommended.
33	Used with double seals when external system is available from user.
41	Apply with liquids that require simultaneous cyclone separation and cooling. (Single and tandem seals).
51	Single seals. Required when sealed liquid will crystallize, coke, solidify, etc. at seal faces if contact with air. Common blankets are isopropyl alcohol, glycol, and water. Normally used with FVD gland and bushing or packed auxiliary box.
52	Tandem seals. Plan provides buffer liquid for outside seal. A plan 01 or plan 11 is also recommended with tandem seals to properly flush inboard seal. Pumping rings recommended.
53	Double seals. Plan provides flushing and cooling to both sets of seal faces. Pumping ring recommended.
54	Double seals or packed auxiliary stuffing box.

Maximum Sealing Flexibility - Dynamic Seal

For Elimination of Mechanical Seal Problems and Reduced Maintenance

Goulds' Dynamic Seal pumps are designed to handle the tough applications where conventional mechanical seals or packing require outside flush and constant, costly attention. The major advantage is that through Goulds' patented design (No. 5,344,163) external seal water is not required, thus eliminating leakage, pumpage contamination, product dilution and problems associated with piping from a remote source.



TECH-B-4B Magnetic Drive Pumps

INTRODUCTION

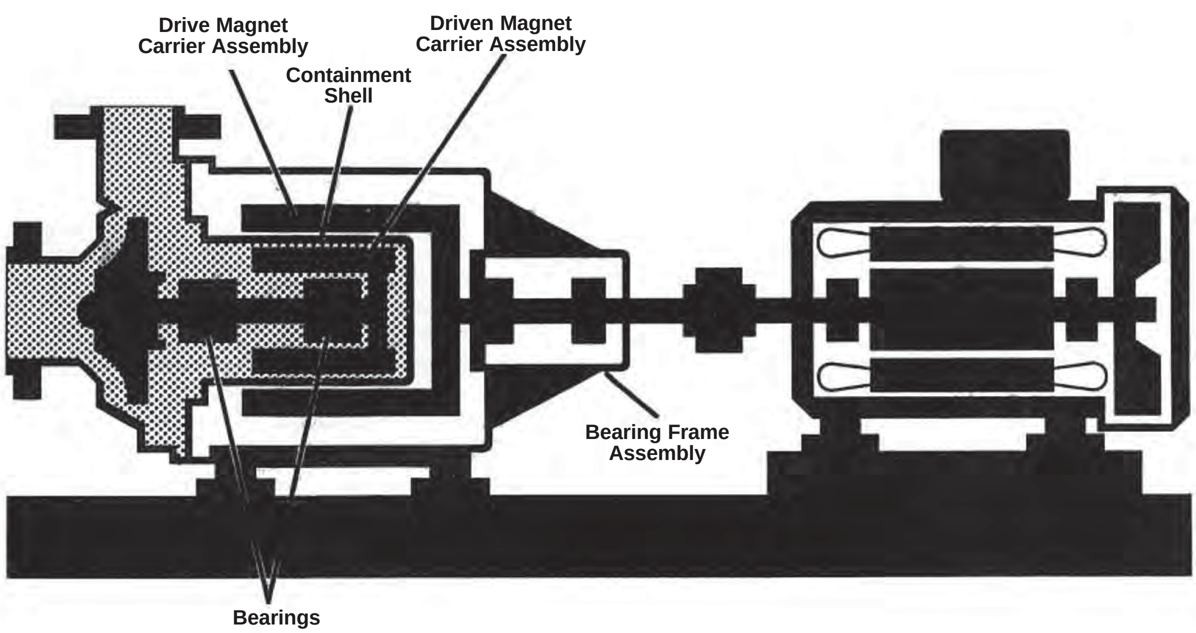
Environmental concerns and recurring mechanical seal problems have created a need for sealless pumps in the chemical and petrochemical industries. In some cases, more stringent regulations by the EPA, OSHA and local agencies are mandating the use of sealless pumps. One type of sealless pump is the magnetic drive pump which uses a permanent magnetic coupling to transmit torque to the impeller without the need for a mechanical seal for packing.

PRINCIPLES OF OPERATION

Magnetic drive pumps use a standard electric motor to drive a set of permanent magnets that are mounted on a carrier or drive assembly located outside of the containment shell. The drive magnet assembly is mounted on a second shaft which is driven by a standard motor. The external rotating magnetic field drives the inner rotor.

The coaxial synchronous torque coupling consists of two rings of permanent magnets as shown in Fig. 1. A magnetic force field is established between the north and south pole magnets in the drive and driven assemblies. This provides the no slip or synchronous capability of the torque coupling. The magnetic field is shown as dashed lines and shaded areas in Fig. 3.

Fig. 1 Typical Magnetic Drive Pump



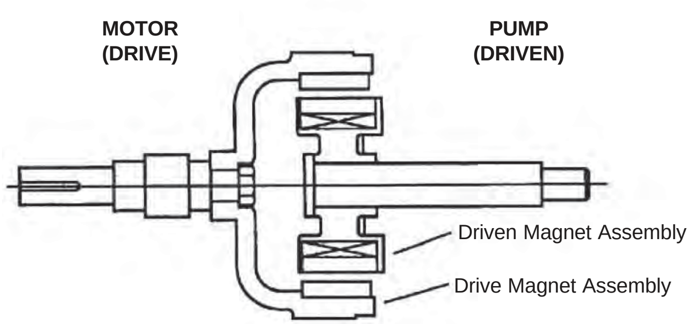


Fig. 2. Coaxial Synchronous Magnetic Torque Coupling

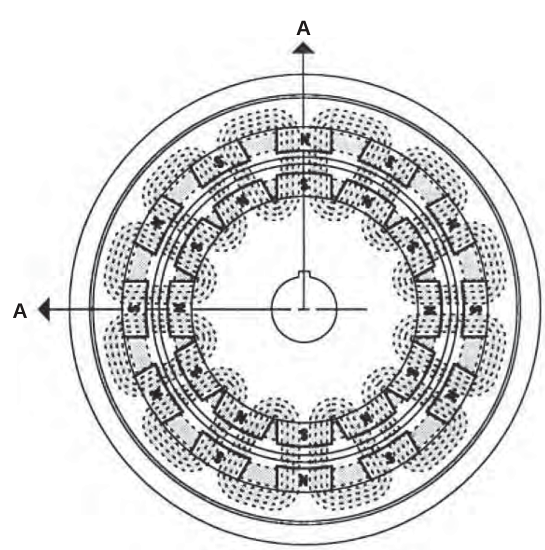


Fig. 3

Containment Shell Designs

The containment shell is the pressure containing barrier which is fitted between the drive and the driven magnet assembly. It must contain full working pressure of the pump, since it isolates the pumped liquid from the atmosphere. One-piece formed shells offer the best reliability, eliminating welds used for two-piece shells.

Since the torque coupling magnetic force field must pass through the shell, it must be made of a non-magnetic material. Non-magnetic metals such as Hastelloy and 316SS are typical choices for the containment shell. The motion of the magnets past an electrically conductive containment shell produces eddy currents, which generate heat and must be removed by a process fluid recirculation circuit.

The eddy currents also create a horsepower loss, which reduces the efficiency of the pump. Metals with low electrical conductivity have lower eddy current losses, providing superior pump efficiency. Hastelloy has a relatively low electrical conductivity and good corrosion resistance, thus is an excellent choice for metal containment shells. Electrically non-conductive materials such as plastic and ceramics are also good choices for containment shells, since the eddy current losses are totally eliminated. This results in pump efficiencies equal to conventionally sealed pumps. Plastic containment shells are generally limited to lower pressures and temperatures due to the limited strength of plastics.

Sleeve and Thrust Bearings

Magnetic drive pumps utilize process lubricated bearings to support the inner drive rotor. These bearings are subject to the corrosive nature of the liquids being pumped, thus need to be made from corrosion resistant materials. Two commonly used materials are hard carbon and silicon carbide (SIC). Pure sintered SIC is superior to reaction bonded SIC, since reaction bonded SIC has free silicon left in the matrix, resulting in lower chemical resistance and lower strength.

Hard carbon against silicon carbide offers excellent service life for many chemical applications and also offers the advantage of short term operation in marginal lubrication conditions.

Silicon carbide against silicon carbide offers excellent service life for nearly all chemical applications. Its hardness, high thermal conductivity, and strength make it an excellent bearing material. Silicon carbide must be handled carefully to prevent chipping. Silicon carbide against silicon carbide has very limited capability in marginal lubrication conditions.

Recirculation Circuit

All magnetic drive pumps circulate some of the process fluid to lubricate and cool the bearings supporting the inner rotor.

Magnetic drive pumps with metal containment shells, also require a circulation of some process fluid through the containment shell to remove heat generated by eddy currents. For pumps with metal containment shells, the fluid recirculation path must be carefully engineered to prevent vaporization of the process liquid necessary to lubricate the bearings. A pressurized circuit as shown in Fig. 4 offers excellent reliability for pumps with metal containment shells.

Magnetic drive pumps with electrically non-conductive containment shells, such as plastic or ceramic have no heat generated by eddy currents. Since no heat is required to be removed from the containment shell, a much simpler recirculation circuit can be used.

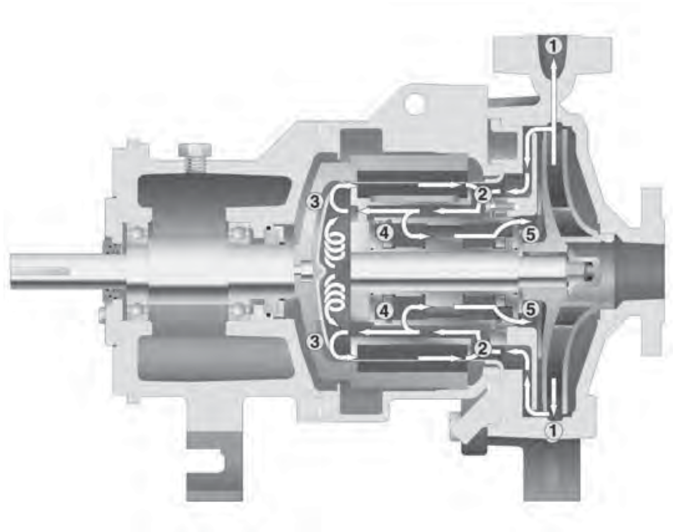


Fig. 4 Recirculation Circuit

Fail Safe Devices

DESCRIPTION

Condition monitoring of the pump is a "key objective" and provides the user with an assurance of safety and reliability.

System and pump malfunctions can result from the following:

- No-flow condition through the pump
- Dry running as a result of plugged liquid circulation paths in the pump bearing and magnets assembly section
- Cavitation due to insufficient $NPSH_A$
- Uncoupling of the magnetic drive due to overload
- Temperature and pressure transients in the system
- "Flashing" in the pump liquid circulation paths due to pressure and temperature transients

These malfunctions can contribute to:

- Overheating of the drive and driven magnet assemblies
- Overload of drive motor and drive magnetic assembly
- Extreme pump bearing load conditions
- Damage to pump due to extremes in temperatures and pressures due to transients that exceed normal design

Various fail safe devices are available with the pump to control malfunctions and provide safety and reliability including:

- Thermocouple / controller
- Low amp relay
- Liquid leak detector
- Power monitor

TECH-B-5 Field Testing Methods

A. Determination of total head

The total head of a pump can be determined by gauge readings as illustrated in Fig. 1.

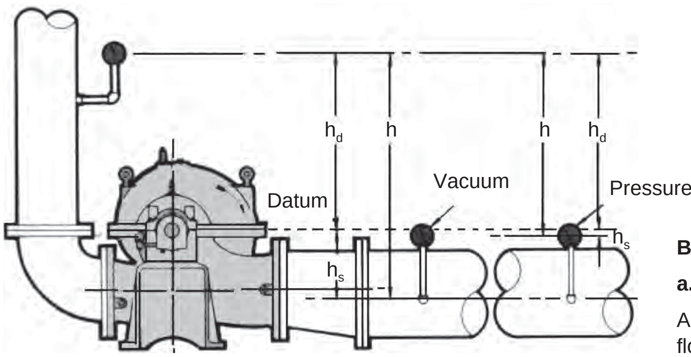


Fig. 1 Determination of Total Head From Gauge Readings

Negative Suction Pressure:

TDH = Discharge gauge reading converted to feet of liquid + vacuum gauge reading converted to feet of liquid + distance between point of attachment of vacuum gauge and the centerline of the discharge gauge, h, in feet + $\left(\frac{Vd^2}{2g} - \frac{Vs^2}{2g}\right)$

Positive Suction Pressure:

or TDH = Discharge gauge reading converted to feet of liquid-pressure gauge reading in suction line converted to ft. of liquid + distance between center of discharge and suction gauges, h, in feet + $\left(\frac{Vd^2}{2g} - \frac{Vs^2}{2g}\right)$

In using gauges when the pressure is positive or above atmospheric pressure, any air in the gauge line should be vented off by loosening the gauge until liquid appears. This assures that the entire gauge line is filled with liquid and thus the gauge will read the pressure at the elevation of the centerline of the gauge. However, the gauge line will be empty of liquid when measuring vacuum and the gauge will read the vacuum at the elevation of the point of attachment of the gauge line to the pipe line. These assumptions are reflected in the above definitions.

The final term in the above definitions accounts for a difference in size between the suction and discharge lines. The discharge line is normally smaller than the suction line and thus the discharge velocity is higher. A higher velocity results in a lower pressure since the sum of the pressure head and velocity head in any flowing liquid remains constant. Thus, when the suction and discharge line sizes at the gauge attachment points are different, the resulting difference in velocity head must be included in the total head calculation.

Manometers can also be used to measure pressure. The liquid used in a manometer is normally water or mercury, but any liquid of known specific gravity can be used. Manometers are extremely accurate for determining low pressures or vacuums and no calibration is needed. They are also easily fabricated in the field to suit any particular application. Figs. 2 & 3 illustrate typical manometer set ups.

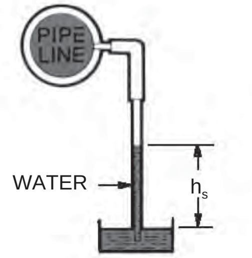


Fig. 2 Manometer Indicating Vacuum

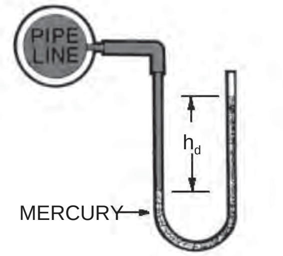


Fig. 3 Manometer Indicating Pressure

B. Measurement of capacity

a.) Magnetic Flow Meter

A calibrated magnetic flow meter is an accurate means of measuring flow in a pumping system. However, due to the expense involved, magnetic flow meters are only practical in small factory test loops and in certain process pumping systems where flow is critical.

b.) Volumetric Measurement

Pump capacity can be determined by weighing the liquid pumped or measuring its volume in a calibrated vessel. This is often practical when pumping into an accurately measured reservoir or tank, or when it is possible to use small containers which can be accurately weighed. These methods, however, are normally suited only to relatively small capacity systems.

c.) Venturi Meter

A venturi meter consists of a converging section, a short constricting throat section and then a diverging section. The object is to accelerate the fluid and temporarily lower its static pressure. The flow is then a function of the pressure differential between the full diameter line and the throat. Fig. 4 shows the general shape and flow equation. The meter coefficient is determined by actual calibration by the manufacturer and, when properly installed, the Venturi Meter is accurate to within plus or minus 1%.

$$Q(GPM) = 5.67\, C D_2^2 \sqrt{\frac{H}{1 - R^4}}$$

C = Instrument Coefficient

D_1 = Entrance Diameter in Inches

D_2 = Throat Diameter in Inches

$R = D_2/D_1$

H = Differential Head in Inches = $h_1 - h_2$

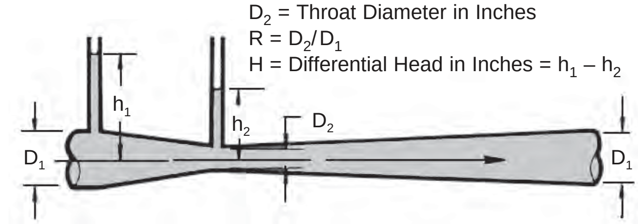


Fig. 4 Venturi Meter

d.) Nozzle

A nozzle is simply the converging portion of a venturi tube with the liquid exiting to the atmosphere. Therefore, the same formula can be used with the differential head equal to the gauge reading ahead of the nozzle. Fig. 5 lists theoretical nozzle discharge flows.

Theoretical Discharge of Nozzles in U.S. GPM

Head		Veloc'y of Disch. Feet per Sec.	Diameter of Nozzle in Inches												
Lbs.	Feet		1/16	1/8	3/16	1/4	3/8	1/2	5/8	3/4	7/8	1	1 1/8	1 1/4	1 3/8
10	23.1	38.6	0.37	1.48	3.32	5.91	13.3	23.6	36.9	53.1	72.4	94.5	120	148	179
15	34.6	47.25	0.45	1.81	4.06	7.24	16.3	28.9	45.2	65.0	88.5	116.	147	181	219
20	46.2	54.55	0.52	2.09	4.69	8.35	18.8	33.4	52.2	75.1	102.	134.	169	209	253
25	57.7	61.0	0.58	2.34	5.25	9.34	21.0	37.3	58.3	84.0	114.	149	189	234	283
30	69.3	66.85	0.64	2.56	5.75	10.2	23.0	40.9	63.9	92.0	125.	164.	207	256	309
35	80.8	72.2	0.69	2.77	6.21	11.1	24.8	44.2	69.0	99.5	135.	177.	224	277	334
40	92.4	77.2	0.74	2.96	6.64	11.8	26.6	47.3	73.8	106.	145.	189.	239	296	357
45	103.9	81.8	0.78	3.13	7.03	12.5	28.2	50.1	78.2	113.	153.	200.	253	313	379
50	115.5	86.25	0.83	3.30	7.41	13.2	29.7	52.8	82.5	119.	162.	211.	267	330	339
55	127.0	90.4	0.87	3.46	7.77	13.8	31.1	55.3	86.4	125.	169.	221.	280	346	418
60	138.6	94.5	0.90	3.62	8.12	14.5	32.5	57.8	90.4	130.	177.	231.	293	362	438
65	150.1	98.3	0.94	3.77	8.45	15.1	33.8	60.2	94.0	136.	184.	241.	305	376	455
70	161.7	102.1	0.98	3.91	8.78	15.7	35.2	62.5	97.7	141.	191.	250.	317	391	473
75	173.2	105.7	1.01	4.05	9.08	16.2	36.4	64.7	101.	146.	198.	259.	327	404	489
80	184.8	109.1	1.05	4.18	9.39	16.7	37.6	66.8	104.	150.	205.	267.	338	418	505
85	196.3	112.5	1.08	4.31	9.67	17.3	38.8	68.9	108.	155.	211.	276.	349	431	521
90	207.9	115.8	1.11	4.43	9.95	17.7	39.9	70.8	111.	160.	217.	284.	359	443	536
95	219.4	119.0	1.14	4.56	10.2	18.2	41.0	72.8	114.	164.	223.	292.	369	456	551
100	230.9	122.0	1.17	4.67	10.5	18.7	42.1	74.7	117.	168.	229.	299.	378	467	565
105	242.4	125.0	1.20	4.79	10.8	19.2	43.1	76.5	120.	172.	234	306.	388	479	579
110	254.0	128.0	1.23	4.90	11.0	19.6	44.1	78.4	122.	176.	240	314.	397	490	593
115	265.5	130.9	1.25	5.01	11.2	20.0	45.1	80.1	125.	180.	245.	320.	406	501	606
120	277.1	133.7	1.28	5.12	11.5	20.5	46.0	81.8	128.	184.	251.	327.	414	512	619
125	288.6	136.4	1.31	5.22	11.7	20.9	47.0	83.5	130.	188.	256.	334.	423	522	632
130	300.2	139.1	1.33	5.33	12.0	21.3	48.0	85.2	133.	192.	261.	341.	432	533	645
135	311.7	141.8	1.36	5.43	12.2	21.7	48.9	86.7	136.	195.	266.	347.	439	543	656
140	323.3	144.3	1.38	5.53	12.4	22.1	49.8	88.4	138.	199.	271.	354	448	553	668
145	334.8	146.9	1.41	5.62	12.6	22.5	50.6	89.9	140.	202.	275.	360	455	562	680
150	346.4	149.5	1.43	5.72	12.9	22.9	51.5	91.5	143.	206.	280.	366.	463	572	692
175	404.1	161.4	1.55	6.18	13.9	24.7	55.6	98.8	154.	222.	302.	395.	500	618	747
200	461.9	172.6	1.65	6.61	14.8	26.4	59.5	106.	165.	238.	323.	423	535	660	799
250	577.4	193.0	1.85	7.39	16.6	29.6	66.5	118.	185.	266.	362.	473.	598	739	894
300	692.8	211.2	2.02	8.08	18.2	32.4	72.8	129.	202.	291.	396.	517.	655	808	977
			1 1/2	1 3/4	2	2 1/4	2 1/2	2 3/4	3	3 1/2	4	4 1/2	5	5 1/2	6
10	23.1	38.6	213	289	378	479	591	714	851	1158	1510	1915	2365	2855	3405
15	34.6	47.25	260	354	463	585	723	874	1041	1418	1850	2345	2890	3490	4165
20	46.2	54.55	301	409	535	676	835	1009	1203	1638	2135	2710	3340	4040	4810
25	57.7	61.0	336	458	598	756	934	1128	1345	1830	2385	3025	3730	4510	5380
30	69.3	66.85	368	501	655	828	1023	1236	1473	2005	2615	3315	4090	4940	5895
35	80.8	72.2	398	541	708	895	1106	1335	1591	2168	2825	3580	4415	5340	6370
40	92.4	77.2	425	578	756	957	1182	1428	1701	2315	3020	3830	4725	5280	6380
45	103.9	81.8	451	613	801	1015	1252	1512	1802	2455	3200	4055	5000	6050	7210
50	115.5	86.25	475	647	845	1070	1320	1595	1900	2590	3375	4275	5280	6380	7600
55	127.0	90.4	498	678	886	1121	1385	1671	1991	2710	3540	4480	5530	6690	7970
60	138.6	94.5	521	708	926	1172	1447	1748	2085	2835	3700	4685	5790	6980	8330
65	150.1	98.3	542	737	964	1220	1506	1819	2165	2950	3850	4875	6020	7270	8670
70	161.7	102.1	563	765	1001	1267	1565	1888	2250	3065	4000	5060	6250	7560	9000
75	173.2	105.7	582	792	1037	1310	1619	1955	2330	3170	4135	5240	6475	7820	9320
80	184.8	109.1	602	818	1070	1354	1672	2020	2405	3280	4270	5410	6690	8080	9630
85	196.3	112.5	620	844	1103	1395	1723	2080	2480	3375	4440	5575	6890	8320	9920
90	207.9	115.8	638	868	1136	1436	1773	2140	2550	3475	4530	5740	7090	8560	10210
95	219.4	119.0	656	892	1168	1476	1824	2200	2625	3570	4655	5900	7290	8800	10500
100	230.9	122.0	672	915	1196	1512	1870	2255	2690	3660	4775	6050	7470	9030	10770
105	242.4	125.0	689	937	1226	1550	1916	2312	2755	3750	4655	5900	7290	8800	10500
110	254.0	128.0	705	960	1255	1588	1961	2366	2820	3840	5010	6350	7840	9470	11300
115	265.5	130.9	720	980	1282	1621	2005	2420	2885	3930	5120	6490	8010	9680	11500
120	277.1	133.7	736	1002	1310	1659	2050	2470	2945	4015	5225	6630	8180	9900	11800
125	288.6	136.4	751	1022	1338	1690	2090	2520	3005	4090	5340	6760	8350	10100	12030
130	300.2	139.1	767	1043	1365	1726	2132	2575	3070	4175	5450	6900	8530	10300	12290
135	311.7	141.8	780	1063	1390	1759	2173	2620	3125	4250	5550	7030	8680	10490	12510
140	323.3	144.3	795	1082	1415	1790	2212	2670	3180	4330	5650	7160	8850	10690	12730
145	334.8	146.9	809	1100	1440	1820	2250	2715	3235	4410	5740	7280	8990	10880	12960
150	346.4	149.5	824	1120	1466	1853	2290	2760	3295	4485	5850	7410	9150	11070	13200
175	404.1	161.4	890	1210	1582	2000	2473	2985	3560	4840	6310	8000	9890	11940	14250
200	461.9	172.6	950	1294	1691	2140	2645	3190	3800	5175	6750	8550	10580	12770	15220
250	577.4	193.0	1063	1447	1891	2392	2955	3570	4250	5795	7550	9570	11820	14290	17020
300	692.8	211.2	1163	1582	2070	2615	3235	3900	4650	6330	8260	10480	12940	15620	18610

NOTE: – The actual quantities will vary from these figures, the amount of variation depending upon the shape of nozzle and size of pipe at the point where the pressure is determined. With smooth taper nozzles the actual discharge is about 94% of the figures given in the tables.

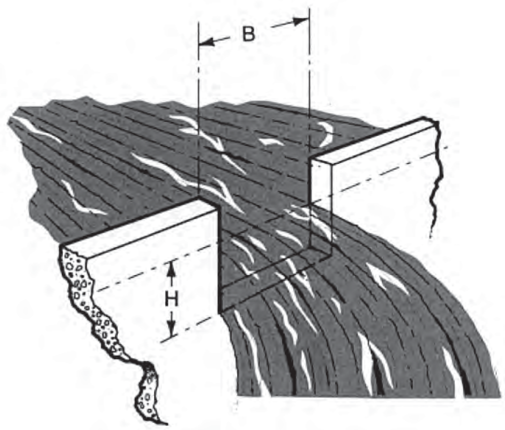
Fig. 5

e.) Orifice

An orifice is a thin plate containing an opening of specific shape and dimensions. The plate is installed in a pipe and the flow is a function of the pressure upstream of the orifice. There are numerous types of orifices available and their descriptions and applications are covered in the Hydraulic Institute Standards and the ASME Fluid Meters Report. Orifices are not recommended for permanent installations due to the inherent high head loss across the plate.

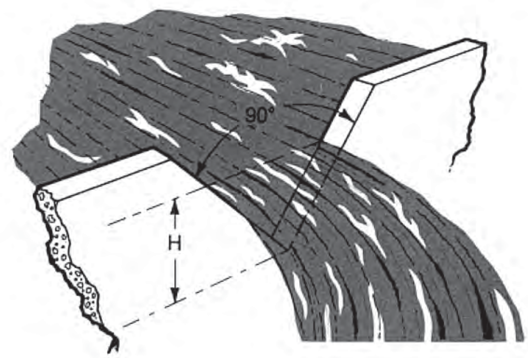
f.) Weir

A weir is particularly well suited to measuring flows in open conduits and can be adapted to extremely large capacity systems. For best accuracy, a weir should be calibrated in place. However, when this is impractical, there are formulas which can be used for the various weir configurations. The most common types are the rectangular contracted weir and the 90 V-notch weir. These are shown in Fig. 6 with the applicable flow formulas.



(6a) - Rectangular Weir With Complete End Contractions

Q(G.P.M.) = 1495 $H^{3/2}$ (B-O.2H)
H = Head in Feet Above Weir
B = Crest Width in Feet



(6b) - 90° V-Notch Weir

Q(G.P.M.) = 1140 $H^{5/2}$
H = Head in Feet Above Weir

Fig. 6 Weirs

g.) Pitot tube

A pitot tube measures fluid velocity. A small tube placed in the flow stream gives two pressure readings: one receiving the full impact of the flowing stream reads static head + velocity head, and the other reads the static head only (Fig. 7). The difference between the two readings is the velocity head. The velocity and the flow are then determined from the following well known formulas.

$\mathbf{V = C\sqrt{2gh_v}}$ where C is a coefficient for the meter determined by calibration, and h_v = velocity head,

Capacity = Area x Average Velocity

Since the velocity varies across the pipe, it is necessary to obtain a velocity profile to determine the average velocity. This involves some error but, when properly applied, a calibrated pitot tube is within plus or minus 2% accuracy.

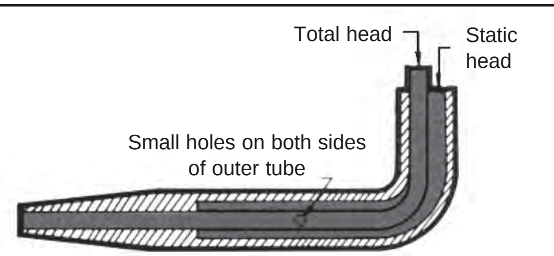


Fig. 7 Pitot Tube

TECH-B-6 Vibration Analysis

Vibration analysis equipment enables you to tell when "normal" vibration becomes "problem" vibration or exceeds acceptable levels. It may also allow you to determine the source and cause of the vibration, thus becoming an effective preventive maintenance and troubleshooting aid.

A vibration analyzer measures the amplitude, frequency and phase of vibration. Also when vibration occurs at several frequencies, it separates one frequency from another so that each individual vibration characteristic can be measured.

The vibration pickup senses the velocity of the vibration and converts it into an electrical signal. The analyzer receives this signal, converting it to the corresponding amplitude and frequency.

The amplitude is measured in terms of peak-to-peak displacement in mils (1 mil = .001") and is indicated on the amplitude meter.

Some instruments are equipped with a frequency meter which gives a direct readout of the predominant frequency of the vibration. Other instruments have tunable filters which allow scanning the frequency scale and reading amplitude at any particular frequency, all others being filtered out.

A strobe light is used to determine the phase of vibration. It can be made to flash at the frequency of the vibration present or at any arbitrary frequency set on an internal oscillator.

A reference mark on a rotating part viewed under the strob light flashing at the vibration frequency may appear as a single frozen (or rotating) mark, or as several frozen (or rotating) marks. The number of marks viewed is useful in determining the source of the vibration. The location of the mark or marks is used in balancing rotating parts.

The first step in vibration analysis is to determine the severity of the vibration, then, if the vibration is serious, a complete set of vibration readings should be taken before attempting to analyze the cause. Fig. 1 is the general guide for horizontal centrifugal pumps as published by the Hydraulic Institute. The amplitudes shown are the overall maximum obtained without filtering to specific frequencies. Amplitudes at specific frequencies, such as vane pass frequency with multi-vane impellers, should be less than 75% of the unfiltered amplitudes allowed in Fig. 1 at the operating RPM. For horizontal non-clog and vertical submerged pumps, refer to Hydraulic Institute standards or pump manufacturer.

Severity of vibration is a function of amplitude and pump speed; however, it should be noted that a change in severity over a period of time is usually a warning of impending failure. This change is often more important than vibration in the "slightly rough" or "rough" ranges which does not change with time.

Complete pump vibration analysis requires taking vibration readings at each bearing in three planes (horizontal, vertical and axial). Readings at the pump suction and discharge flanges may also be useful in some cases.

After all data has been tabulated, it can be analyzed to determine the most likely cause or causes of vibration and the identifying characteristics of each.

By analyzing the tabulated vibration data one or several causes may be found. Each must be checked, starting with the most likely cause or easiest to check.

For example, assume the axial vibration is 50% or more of the radial vibration and the predominant frequency is the same as the RPM of the pump. The chart indicates probable misalignment or bent shaft. Coupling misalignment is probably the most common single cause of pump vibration and is one of the easiest to check. If after checking, the alignment proves to be good, then inspect for flange loading. Finally, check for a bent shaft. Cavitation in a pump can cause serious vibration. Vibration at random frequencies can also be caused by hydraulic disturbances in poorly designed suction or discharge systems.

The use of vibration equipment in preventive maintenance involves keeping a vibration history on individual pieces of equipment in a plant. A form similar to that shown in Fig. 4 can be used to record the vibration data on a periodic routine basis. Abrupt changes are a sign of impending failure. A gradual increase in vibration can also be detected and corrective measures can be taken before it reaches a dangerous level.

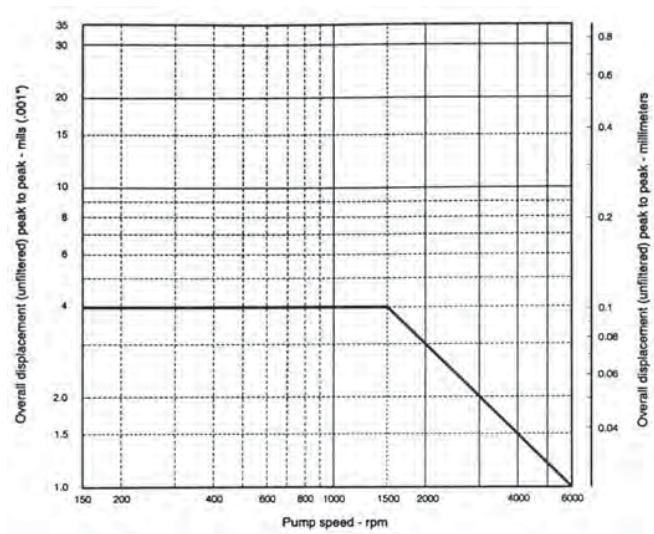


Fig. 1 Acceptable field vibration limits for horizontal or vertical in-line pumps (Figures 1.107 to 1.109) - clear liquids

Vibration Analysis – Continued

Cause	Amplitude	Frequency	Phase	Remarks
Unbalance	Largest in radial direction. Proportional to unbalance	1 x RPM	Single reference mark	Unbalance
Misalignment of coupling or bear-ings and bent shaft	Axial direction vibration 50% or more of radial	1 x RPM normally	single, double, or triple	Easily recognized by large axial vibration. Excessive flange loading can contribute to misalignment
Bad Anti-friction bearings	Unsteady	Very high. Several times RPM	Erratic	Largest high-fre-quency vibration near the bad bearing.
Mechanical looseness		2 x RPM	Two reference marks. Slightly erratic.	Check grouting and bed plate bolting.
Bad drive belts	Erratic or pulsing	1, 2, 3 & 4 x RPM of belts	Unsteady	Use strobe light to freeze faulty belt.
Electrical	Disappears when power is turned off.	1 or 2 x synchro-nous frequency	Single or rotating double mark	3600 or 7200 cps for 60 cycle current.
Hydraulic forces		No. of impeller vanes x RPM		Rarely a cause of serious vibration

Fig. 3 Vibration Identification Chart

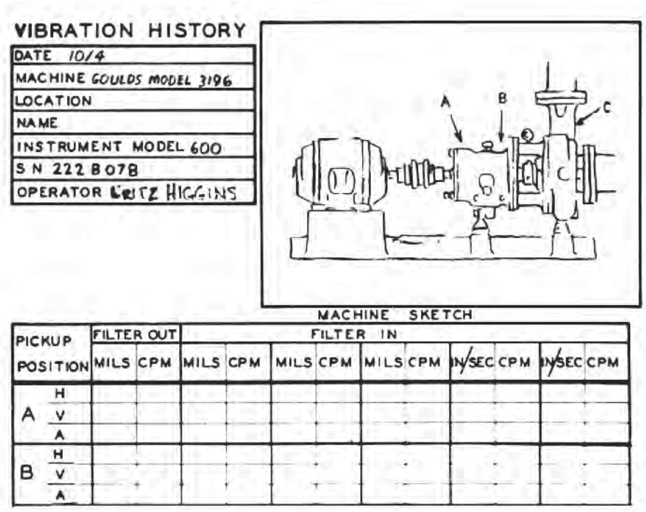


Fig. 4 Vibration Data Sheet

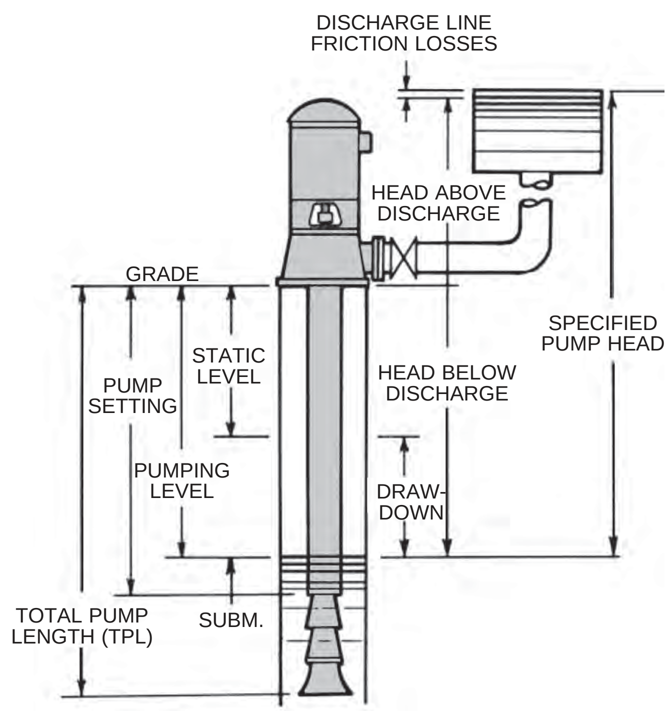
TECH-B-7 Vertical Turbine Pumps

Turbine Nomenclature

1. **DATUM OR GRADE -** The elevation of the surface from which the pump is supported.
2. **STATIC LIQUID LEVEL -** The vertical distance from grade to the liquid level when no liquid is being drawn from the well or source.
3. **DRAWDOWN -** The distance between the static liquid level and the liquid level when pumping at required capacity.
4. **PUMPING LIQUID LEVEL -** The vertical distance from grade to liquid level when pumping at rated capacity. Pumping liquid level equals static water level plus drawdown.
5. **SETTING -** The distance from grade to the top of the pump bowl assembly.
6. **TPL (TOTAL PUMP LENGTH) -** The distance from grade to lowest point of pump.
7. **RATED PUMP HEAD -** Lift below discharge plus head above discharge plus friction losses in discharge line. This is the head for which the customer is responsible and does not include any losses within the pump.
8. **COLUMN AND DISCHARGE HEAD FRICTION LOSS -** Head loss in the pump due to friction in the column assembly and discharge head. Friction loss is measured in feet and is dependent upon column size, shaft size, setting, and discharge head size. Values given in appropriate charts in Data Section.
9. **BOWL HEAD -** Total head which the pump bowl assembly will deliver at the rated capacity. This is curve performance.
10. **BOWL EFFICIENCY-** The efficiency of the bowl unit only. This value is read directly from the performance curve.
11. **BOWL HORSEPOWER-** The horsepower - required by the bowls only to deliver a specified capacity against bowl head.

$$\text{BOWL HP} = \frac{\text{Bowl Head x Capacity}}{\text{3960 x Bowl Efficiency}}$$

12. **TOTAL PUMP HEAD -** Rated pump head plus column and discharge head loss. **NOTE**: This is new or final bowl head.
13. **SHAFT FRICTION LOSS -** The horsepower required to turn the lineshaft in the bearings. These values are given in appropriate table in Data Section.



14. **PUMP BRAKE HORSEPOWER -** Sum of bowl horsepower plus shaft loss (and the driver thrust bearing loss under certain conditions).
15. **TOTAL PUMP EFFICIENCY (WATER TO WATER)** -The efficiency of the complete pump, less the driver, with all pump losses taken into account.

$$\text{Efficiency} = \frac{\text{Specified Pump Head x Capacity}}{\text{3960 x Brake Horsepower}}$$

16. **OVERALL EFFICIENCY (WIRE TO WATER) -** The efficiency of the pump and motor complete. Overall efficiency = total pump efficiency x motor efficiency.
17. **SUBMERGENCE -** Distance from liquid level to suction bell.

Vertical Turbine Pumps - Calculating Axial Thrust

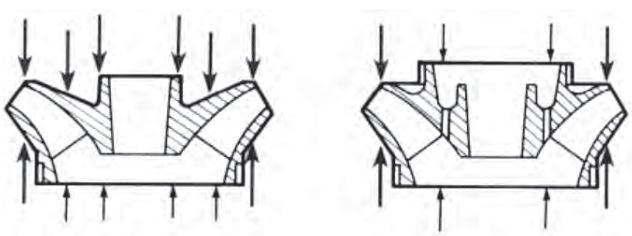
Under normal circumstances Vertical Turbine Pumps have a thrust load acting parallel to the pump shaft. This load is due to unbalanced pressure, dead weight and liquid direction change. Optimum selection of the motor bearing and correct determination of required bowl lateral for deep setting pumps require accurate knowledge of both the magnitude and direction (usually down) of the resultant of these forces. In addition, but with a less significant role, thrust influences shaft H.P. rating and shaft critical speeds.

IMPELLER THRUST

Impeller Thrust in the downward direction is due to the unbalanced discharge pressure across the eye area of the impeller. See diagram A.

Counteracting this load is an upward force primarily due to the change in direction of the liquid passing through the impeller. The resultant of these two forces constitutes impeller thrust. Calculating this thrust using a thrust constant (K) will often produce only an approximate thrust value because a single constant cannot express the upthrust component which varies with capacity.

To accurately determine impeller thrust, thrust-capacity curves based on actual tests are required. Such curves now exist for the "A" Line. To determine thrust, the thrust factor "K" is read from the thrust-capacity curve at the required capacity and given RPM. "K" is then multiplied by the Total Pump Head (Final Lab Head) times Specific Gravity of the pumped liquid. If impeller thrust is excessively high, the impeller can usually be hydraulically balanced. This reduces the value of "K". Balancing is achieved by reducing the discharge pressure above the impeller eye by use of balancing holes and rings. See diagram B.



(A) (B)

Suspection Pressure

Discharge Pressure

NOTE:

Although hydraulic balancing reduces impeller thrust, it also decreases efficiency by one to five points by providing an additional path for liquid recirculation. Of even greater concern is that should the hydraulic balancing holes become clogged, (unclean fluids, fluids with solid content, intermittent services, etc.), the impeller thrust will increase and possibly cause the driver to fail. Hydraulically balanced impellers cannot be used in applications requiring rubber bowl bearings because the flutes on the inside diameter of the bearings provide an additional path to the top side of the impeller, thus creating an additional down thrust.

Hydraulically balanced impellers should be used as a "last resort" for those situations where the pump thrust exceeds the motor thrust bearing capabilities.

DEAD WEIGHT

In addition to the impeller force, dead weight (shaft plus impeller weight less the weight of the liquid displaced) acts downward. On pumps with settings less than 50 feet, dead weight may be neglected on all but the most critical applications as it represents only a small part of the total force. On deeper setting pumps, dead weight becomes significant and must be taken into account.

NOTE:

We normally only take shaft weight into consideration as dead weight, the reason being that impeller weight less its liquid displacement weight is usually a small part of the total.

SHAFT SLEEVES

Finally, there can be an upward force across a head shaft sleeve or mechanical seal sleeve. In the case of can pumps with suction pressure, there can be an additional upward force across the impeller shaft area. Again, for most applications, these forces are small and can be neglected; however, when there is a danger of upthrusts or when there is high discharge pressure (above 600 psi) or high suction pressure (above 400 psi) these forces should be considered.

MOTOR BEARING SIZING

Generally speaking a motor for a normal thrust application has, as standard, a bearing adequate for shutoff thrust. When practical, motor bearings rated for shutoff conditions are preferred. For high thrust applications (when shutoff thrust exceeds the standard motor bearing rating) the motor bearing may be sized for the maximum anticipated operating range of the pump.

Should the pump operate below minimum flow for a short period of time, anti-friction bearings such as angular contact or spherical roller can handle the overload. It should be remembered, however, that bearing life is approximately inversely proportional to the cube of the load. Should the load double, motor bearing life will be cut to $^1/_8$ of its original value. Although down thrust overloading is possible, the pump must never be allowed to operate in a continuous upthrust condition even for a short interval without a special motor bearing equipped to handle it. Such upthrust will fail the motor bearing.

CALCULATING MOTOR BEARING LOAD

As previously stated, for short setting non-hydraulic balanced pumps below 50 feet with discharge pressures below 600 psi and can pumps with suction pressures below 100 psi, only impeller thrust need be considered.

Under these conditions:	Where:
Motor Bearing Load (lbs.)	Impeller Thrust (lbs.)
$T_{imp} = KH_L \times SG$	K=Thrust factors (lbs./ft.)
	H_L, = Lab Head (ft.)
	SG = Specific Gravity

For more demanding applications, the forces which should be considered are impeller thrust plus dead weight minus any sleeve or shaft area force.

In equation form:

Motor Bearing Load = T_{imp} + Wt[1] – sleeve force[2] – shaft area force[3] = T_t

CALCULATING AXIAL THRUST - CONTINUED

Shaft Dia (in)	Shaft Dead Wt. (lbs/ft.) Open Lineshaft	Shaft Dead Wt. (lbs/ft.) Closed Lineshaft	Shaft Area (in^2)	Sleeve Area (in)
1	2.3	2.6	.78	1.0
$1\frac{3}{16}$	3.3	3.8	1.1	1.1
$1\frac{1}{2}$	5.3	6.0	1.8	1.1
$1\frac{11}{16}$	6.7	7.6	2.2	1.5
$1\frac{15}{16}$	8.8	10.0	2.9	1.8
$2\frac{3}{16}$	11.2	12.8	3.7	2.0

(1) Wt.= Shaft Dead Wt. x Setting In Ft.
(2) Sleeve Force=Sleeve area x Discharge pressure
(3) Shaft Area Force = Shaft area x Suction pressure
*Oil Lube shaft does not displace liquid above the pumping water level and therefore has a greater net weight.

THRUST BEARING LOSS

Thrust bearing loss is the loss of horsepower delivered to the pump at the thrust bearings due to thrust. In equation form:

$$L_{TB} = .0075 \left(\frac{BHP}{100}\right)\left(\frac{T_t}{1000}\right)$$

where:

L_{TB} = Thrust bearing loss (HP)
BHP = Brake horsepower
T_t = Motor Bearing Load (Lbs.)
= $T_{imp} + Wt^{(1)}$ – sleeve force$^{(2)}$ – shaft area force$^{(3)}$

Vertical Turbine Bearing Material Data

(For specific applications where a given bearing material is specified and these limitations are exceeded, refer to factory.)

	Material Description	Temperature and Specific Gravity Limits	Remarks
1.**	Standard Bronze (Federalloy III) 7% Tin/2-4% Zinc /85-89% Cu	-50 to 180°F Min. S. G. of 0.6	General purpose material for fresh/salt water light abrasive services up to 50 ppm. This is a non-leaded bronze material that will **not** dezincify in seawater because of low zinc content. Not suitable in ammonia, hydrogen sulfide and acetylene services.
2.**	Resin Impregnated Carbon	-50 to 300°F All gravities	Good corrosion resistant material suitable for light abrasive services up to 10 ppm. Special materials available for temperatures beyond 300°F. Good for low specific gravity fluids (e.g. ethane, propane, butane, ethylene) because the carbon is self-lubricating.
3.	Metal (e.g. Nickel) impreg-nated carbon bearings	-380° to 700°F All gravities	Excellent corrosion resistant except for strong oxidizing solutions**(1)**. Suitable for abrasive services up to 50 ppm. Special materials available for severe acid services. Good for low specific gravity fluids because the carbon is self-lubricating.
4.	Teflon (metal backed) 25% Graphite with 75% Teflon	-50° to 250°F All gravities	Excellent corrosion resistant except for strong oxidizing solutions**(1)**. Suitable for abrasive services up to 5 ppm. (Glass filled Teflon also available.) Limited applications - Call V.P.O.
5.	Cast Iron ASTM-A-48 CL30 (I. D. Electroless Nickel Coated)	32° to 180°F Min. S.G. of 0.6	Limited to mildly caustic **(2)** and light abrasive services up to 10 ppm & some petroleum products (e.g. tar, heavy crude) with good lubricity.
6.**	Rubber (Nitrile Butadiene or Neoprene) with phenolic **(3)** or metal backing	32° to 150°F	First choice in abrasive fresh/salt water services up to 5000 ppm. Shafting should also be hardfaced for abrasive content above 100 ppm. Bearings must be wet prior to start-up if non-submerged ("dry column") length is greater than 50 ft. **Do not** use in oil, hydrocarbon services, and strong oxidizing agent(1). Contact the factory If the pumpage is other than fresh/salt water. **Do not** use for stuffing box or mechanical seal housing bushings; instead, use standard bronze for light abrasive service up to 50 ppm or hard faced bearing and shaft over 50 ppm. **Do not** use with hydraulically balanced impellers.
7.**	Stainless shell with hardfacing	-100° to 300°F Min. S.G. of 0.6	Alternate for corrosive/abrasive services up to 5000 ppm. Coating or hardfacing material is typically chromium oxide. Contact factory if the bearing shell and/or coating need to be upgraded for better corrosion/abrasion resistance. **Always** use in combination with hard-faced shaft journals.
8.	Boron diffusion coated	up to 400°F	Recommended only for geothermal-brine services **without** the present of oxygen. Hard-faced coated surfaces typically in the range of Rc75. High temp. chemical vapor deposition (CVD). Hardness penetrates into various parent material (substrates), and will not flake, chip, or separate under severe applications (with no O_2 presence).
9.	30% carbon fiber reinforced and compression molded PEEK	-80° to 300°F	Good for services that are corrosive + high temp + abrasives up to 250 ppm. Good chemical resistance (not as good as carbon and Teflon.) Low coefficient of friction and impact/ thermal shock resistance. Not suitable for strong acids, halogens or hot solvents services. Difficult to machine.
10.	Continuous carbon fiber reinforced PEEK	-80° to 600°F	Good for services that are corrosive + high temp + abrasives up to 1,000 ppm. Good chemical resistance (not as good as carbon and Teflon.) Low coefficient of friction and impact/ thermal shock resistance. Not suitable for strong acids, halogens or hot solvents services. Difficult to machine. This material is very difficult to install for high temperature (over 300°F) service.
11.	Nitronic 60	-50° to 300°F Min. S.G. of 0.6	For seawater service. Good for abrasives up to 50 ppm. For use **ONLY** with nitronic 50 shaft.
12.	Thordon	32° to 150°F Min. S.G. of 0.6	Snap rings must be used. Absorbs water, tends to swell. Used at customer request only.

** Denotes bearing materials which are included in Prism cost database.
(1) Example of strong oxidizing agents: hydrochloric acid HCl, nitric acid HNO_3, and sulfuric acid H2SO4(hot).
(2) Common name for sodium hydroxide NaOH is lye or caustic soda.
(3) Rubber with phenolic backing will be furnished unless other backing is specified by customer.

TECH-B-8 Self Priming Pump System Guidelines

Self-priming pumps are inherently designed to allow the pump to re-prime itself typically under lift conditions. These pumps are very effective to the end user in that they will eliminate the need for foot valves, vacuum and ejector pumps which can become clogged or be impractical to use for prolonged or remote operation. Although the pump itself is designed to accomplish this task, it is important to understand the principle of how self-priming is achieved so that the piping system can be designed so as not to conflict with this function.

A self-priming pump, by definition, is a pump which will clear its passages of air if it becomes air bound and resume delivery of the pumpage without outside attention. To accomplish this, a charge of liquid sufficient to prime the pump must be retained in the casing (See Fig. A) or in an accessory priming chamber. When the pump starts, the rotating impeller creates a partial vacuum; air from the suction piping is then drawn into this vacuum and is entrained in the liquid drawn from the priming chamber. This air-liquid mixture is then pumped into the air separation chamber (within the casing) where the air is separated from the liquid with the air being expelled out the discharge piping (Fig. B) and the liquid returning to the priming chamber. This cycle is repeated until all of the air from the suction piping has been expelled and replaced by pumpage and the prime has been established (Fig. C).

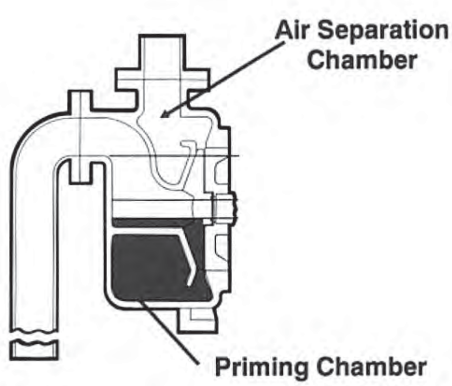


Fig. A

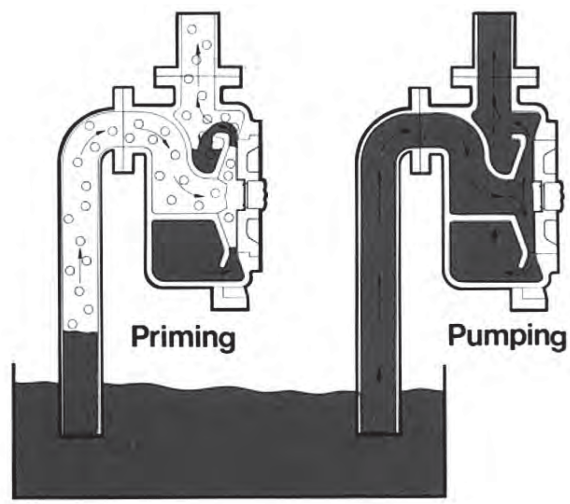


Fig. B

Fig. C

The following considerations should be made when designing a piping system for which a self-priming pump is to be used:

- Care should be exercised to insure that adequate liquid is retained in the priming chamber. For outdoor/remote installations a heating element may be required to prevent freezing. For dirty services a strainer may be required to keep solids from accumulating in the priming chamber, thus displacing priming liquid.
- The static lift and suction piping should be minimized to keep priming time to a minimum. Excessive priming time can cause liquid in the priming chamber to vaporize before prime is achieved.
- All connections in the suction piping should be leak-free as air could be sucked in, thus extending/compromising priming of the pump. (Pumps sealed with packing should be flushed to prevent air from being introduced.)
- A priming bypass line (See Fig. D) should be installed so that back pressure is not created in the discharge piping during priming which would prevent the pump from priming itself. (Self-priming pumps are not good air compressors!)
- The suction piping should be designed such that no high points are created where air can be trapped/accumulated which can prevent priming. Historically this has been problematic on top unloading of rail cars. (See Fig. E)

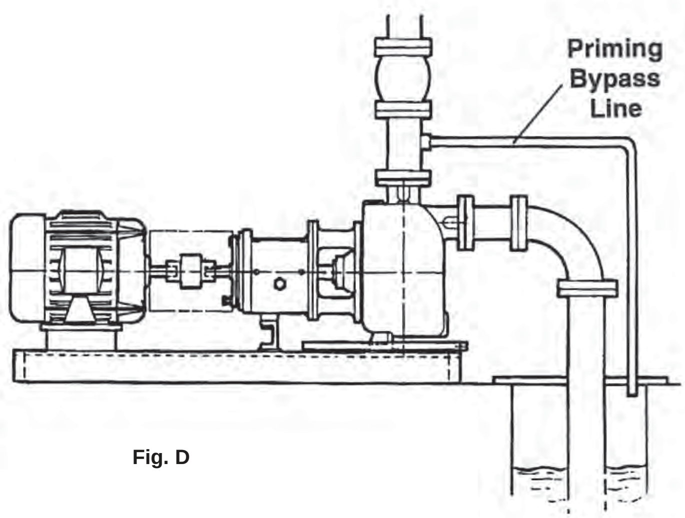


Fig. D

NOTE: Goulds Model 3796 self-priming process pump is outlined in Section 1F.

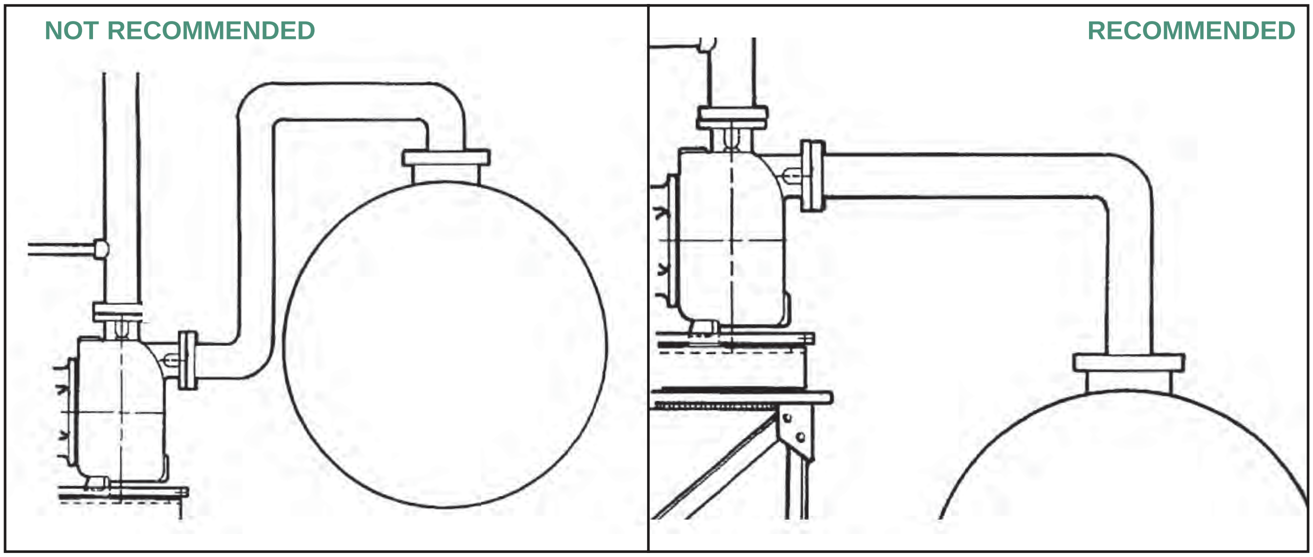


Fig. E Tank Car Unloading

TECH-B-9 Priming Time Calculations

Priming time data for each Model 3796 pump size and speed is displayed on the individual performance curves where priming time is plotted versus effective static lift for maximum, minimum and intermediate impeller diameters. This data is for suction piping of the same nominal diameter as the pump suction, i.e. 3" piping and 3" pump suction, and must be corrected for suction pipe diameters different from the pump suction and for suction pipe lengths greater than the effective static lift.

To calculate the total priming time for a given system:

1. Select the correct size and speed pump from the performance curve for the given rating.

2. Calculate the **NPSH Available** for the system. The available NPSH must be **equal to or greater than the NPSH Required** by the selected pump at the rating point.

$$NPSH_A = P - (L_s + V_p + h_f)$$

where: P = Pressure on surface of liquid in feet absolute

L_s = Maximum static lift in feet from free surface of the liquid to the centerline of the impeller.

V_p = Vapor pressure of the liquid at maximum pumping temperature in feet absolute.

h_f = Suction pipe friction loss in feet at the required capacity.

3. Determine the effective static lift.

$$L_{es} = Ls \times Sp. Gr.$$

where: L_{es} = Effective static lift in feet.

L_s = Maximum static lift in feet from free surface of the liquid to the centerline of the pump suction, or the highest point in the suction piping, whichever is greater.

Sp. Gr. = Specific gravity of the liquid.

4. Enter the priming time curve at the effective static lift calculated in Step 3. Proceed across to the impeller diameter selected for the specified rating and then downward to the bottom coordinate to determine the priming time (PT_{Les}) to achieve the given lift.

5. Insert the priming time from Step 4 into the following formula to calculate the total system priming time:

Priming Time - Seconds

$$PT_T = PT_{Les} \; x \; \frac{SPL}{L_{es}} \; x \left(\frac{D_p}{D_s}\right)^2$$

where: PT_T = Total system priming time.

PT_{Les} = Priming time in seconds for the effective static lift (Step 4.)

SPL = Total suction pipe length above the free surface of the liquid in feet.

L_{es} = Effective static lift.

D_p = Nominal pipe diameter.

D_s = Nominal pump suction diameter.

Sample Priming Time Curve