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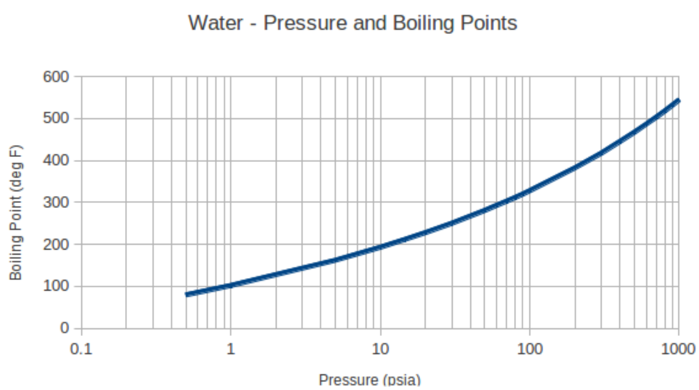
PUMPS FOR HIGH TEMPERATURE HOT WATER

Introduction

Pumping water is done at nearly every commercial and industrial facility in the world. With water pumps dating back to 3000 B.C., we have a lot of history pumping water. Pumping cold (lubricating) water is normally very straight forward with standard materials, standard seals and lower pressure components normally being utilized.

What happens to water's properties when you heat water? At 20°C (68°F) water has a specific gravity of .9982 and a viscosity of 1 cps with a vapor pressure of about .3 psia. At 80°C (176°F) water has a specific gravity of .9716 and a viscosity of .355 cps with a vapor pressure of nearly 7 psia. The hot water is lighter in weight, thinner and vaporizing to a greater degree. It has also become less lubricating. As it gets even hotter, the weight and the viscosity continue to drop and above the sea level atmospheric boiling temperature of 100°C (212°F), the pressure to maintain it as a liquid (vs. a gas) increases dramatically as shown in the graph below.



The next logical question is what effect does this higher temperature (and resulting liquid properties) have on a standard centrifugal pump and should I consider a different more reliable pump?

When pumping any liquid, especially hot liquids, a check of the available energy ($NPSH_A$ -net positive suction head available) of the incoming liquid should be done. As centrifugal pumps require the incoming stream to be in liquid form, we must not allow boiling (vaporizing) of the liquid by allowing it to reach its vapor pressure. In nearly all elevated temperature water systems above 212°F, you have a vessel (or boiler) where the pressure at the water surface is the same as the liquid vapor pressure (state of equilibrium) where external additional pressure is not being added. At the pump suction, the only energy keeping that contained water in liquid form is the pressure created by the vertical head measured from the liquid surface to the centerline of the pump impeller. We lose some of that energy due to friction loss in the suction piping. The available energy is best expressed as Net positive suction head available ($NPSH_A$) which is defined as the difference between the pressure at pump suction nozzle and the liquid vapor pressure, expressed in the same pressure terms (typically feet of liquid).

$$NPSH_A = \frac{2.31*(P_a - P_v)}{SG} + H_e - H_f$$

Where P_a is the pressure at the water surface (psia), P_v is the vapor pressure (saturation pressure) for the fluid at the given temperature (psia), H_e is the difference in height from the water surface to the pump suction centerline (feet), H_f is the suction line friction loss (feet), and SG is the fluid specific gravity. Once the $NPSH_A$ is calculated, it should be compared to the specific pump $NPSH_R$ (Net positive suction head required) to make sure ample difference between the available and required exists.

Hot water systems tend to have **low NPSH available.** The danger surrounding not having an adequate amount of $NPSH_A$, particularly in the case of water, is cavitation. Cavitation can occur at several points within a pump, but in the case of low $NPSH_A$, the cavitation will occur at the suction eye (inlet) of the impeller. As the liquid is accelerated through the suction pipe (in order to feed the pump), its velocity increases. This velocity increase causes the pressure to drop (Bernoulli's Principle). If the velocity is such that the pressure drops to the vapor pressure of the liquid being pumped then vapor bubbles will form within the liquid stream. The presence of the bubbles can affect the overall performance of the pump but the real damage occurs after the bubbles enter the impeller eye and the pressure begins to increase. This added pressure causes the bubbles to collapse. The collapsing bubbles can produce tremendous localized pressures on the surface of the impeller to the point where a small amount of metal can be removed. The tiny abrasions occur constantly as the liquid moves through the pump and over time will severely reduce the structural integrity of the metal surface. Typically pumps cavitating in water make the telltale sound of "rocks being pumped". The below two picture show a bronze pump inducer, the left being a new unused part and the right is a picture of the same part after being subjected to severe suction cavitation.


For applications where the difference in $NPSH_A$ and $NPSH_R$ is less than 2 feet, a 316 SS impeller may be used. The 316 SS impeller has greater resistance to short periods of minor cavitation. However, this is not a solution for inadequate $NPSH_A$. Consideration should be given to <u>specifying an NPSH test</u> to confirm the performance <u>when $NPSH_A$ is less than 4 feet greater than pump $NPSH_R$</u>.

Once we have determined the $NPSH_A$ we need to consider the water properties discussed above (higher temperature, lighter in weight-lower SG, thinner, less lubricating and higher vapor pressure).

Higher temperature (and resulting higher pressures)- Nearly any centrifugal pump can handle water up to about 250° F, although special consideration for the mechanical seal is required with the elevated temperature. Beyond that temperature though, much greater consideration must be given to the mechanical design, the working pressure limits of the pump, as well as the mechanical seal selection. At high temperatures, above 300° F, pumps and piping systems, due to thermal expansion, will grow and consideration must be given to this expansion. The API (American Petroleum Institute) Standard 610 recommends that pumps in this temperature range and above should be designed with centerline-supported casing which **allows the pump to thermally expand** from its center of rotation thus keeping the rotating components on axis with the pump casing. This design allows the pump casing to thermally expand freely without being fixed to the pump base like with a foot mounted pump casing. The centerline supported casing significantly **reduces the pump alignment problems** of foot mounted casing designs.

The pump casing, bolting and flanges should be designed to withstand 150% of the working pressure of the pump as defined in the following formula:

$$P_{wp}=1.50 * (P_t+P_s)$$

Where P_{wp} = pump design working pressure in psig; P_t = total developed pressure or differential head in psig; and P_s = suction pressure in psig.

Lower SG (weight per gallon)-The lower density of hot water reduces the HP required for a given flow and head using a centrifugal pump. The resulting reading on a pressure gauge will also be reduced as your column of liquid weighs less and the psi (pounds per square inch) is less.

Thinner viscosity- The lower viscosity affects the lubrication film on the mechanical seal faces with a reduced film thickness. It becomes more difficult to seal with mechanical seals and also at any pipe connections.

Less Lubricating- This is a result of the lower viscosity with higher temperatures.

Higher vapor pressure-Water must be kept as a liquid in order to be pumped. As the liquid water is heated, the vapor pressure with greater temperatures will create much higher system pressures. One of the difficulties becomes sealing the hot water as the internal (stuffing box) pressure vs. the external (atmospheric) pressure on the other side of the shaft packing or mechanical seal increases. This can quickly become a big safety concern as seal failure can allow the extremely hot water to escape at high pressures potentially resulting in injury or death.

The Pump

The standard high temperature hot water pump is the Dean Series R4000 series Pump. With Dean manufacturing pumps for high temperature service for nearly 70 years, they know how to provide the right pump for your application. The R4000 series is a heavy duty centerline-supported, end suction, top centerline discharge, back pull-out design centrifugal pump. The basic materials of construction are cast steel casings and cast iron impellers (Dean class 40). The pump is rated for 500 psig working pressure and is suitable for water at 460° F maximum temperature.

Sealing the Pump

Mechanical seals

The next consideration is the stuffing box or mechanical seal chamber of the pump. Graphoil ribbon packing is suitable for a packed box which will leak a small amount of steam and water. Leakage is necessary to lubricate the packing and reduce heat build-up in the stuffing box. Mechanical seals also depend on a very tiny amount of leakage (normally in the form of vapor) to cool and lubricate the mechanical seal faces. Leakage from packing is much greater than from mechanical seals and is generally not desired. Most units today are furnished with mechanical seals.

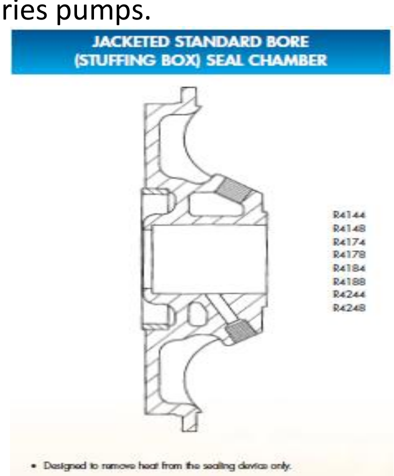
Due to the high working pressures of hot water systems, a balanced mechanical seal design is required above 220°F. (Such as John Crane Type 8 B1T). Below 220°F an unbalanced seal may be used (Such as John Crane Type 1). For either the balanced or unbalanced seal, the face materials should be a special carbon for hot water (Code P_{66} or F_{48}) vs. either tungsten carbide (Code 0_{15}) or silicon carbide (Code 0_{58}). The static seals should be elastomeric O-rings. Keep in mind elastomers do not perform up to their normal temperature ratings in hot water. The following guide lines should be followed when applying elastomers:

Viton (code X)	250°F Max.
Ethylene Propylene (code 0_{28})	280°F Max.
AFLAS (code X_{18})	350°F Max.
Kalrez (code X_5)	400°F Max.

The most common seal sold for high temperature water is the John Crane single inside balanced type 8B1T Code $X_{18}F_{48}1O_{15}1$ (316SS) with a flush gland. This seal has Aflas elastomers, Carbon vs. Tungsten Carbide seal faces and 316SS hardware. This same seal construction can be furnished in a balanced Cartridge seal arrangement which makes installation much easier.

Sealing chamber cooling

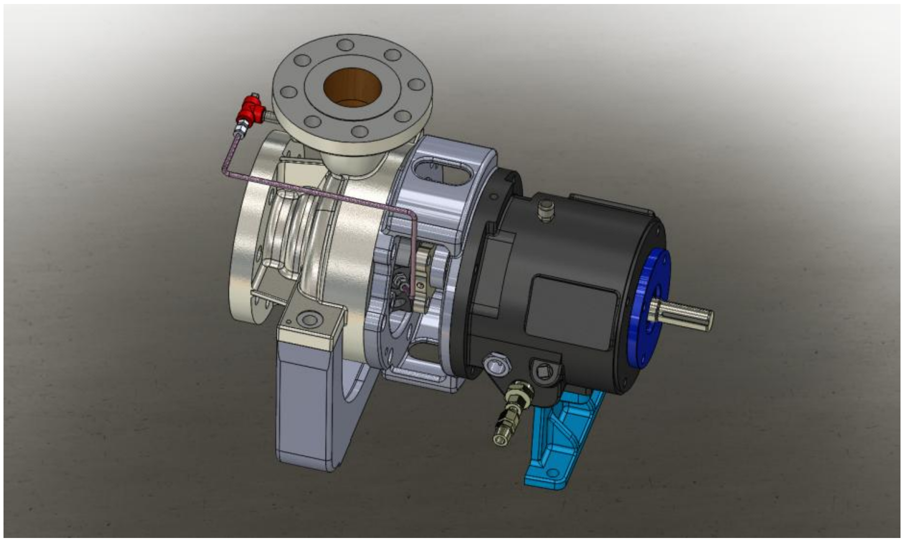
Above 250° F pumping temperature, cooling of the seal chamber is always recommended. This jacketed backhead (sealing chamber), when connected to cooling water, will help cool the seal, giving much improved seal life. It is especially useful to keep the seal parts cool when the pump is not in operation as well as when operating. The jacketed sealing chamber is standard for the Dean R4000 series pumps.



Improving Seal Life

As mentioned prior, water at elevated temperatures loses its lubricity or lubricating quality resulting in reduced seal life. Some type of seal cooling options should be considered. Even though very helpful, a hot water seal will simply not be adequately cooled solely by the seal chamber cooling jacket. A cooled flush over the seal faces should be considered.

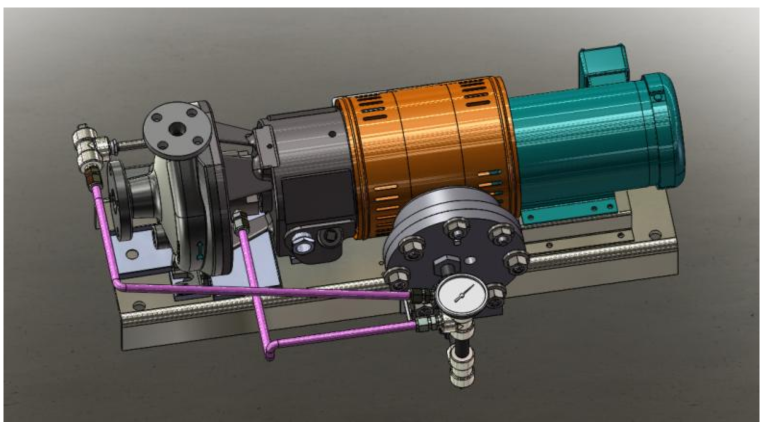
For temperatures **below 250° F** a **flush line** (as shown below) should be routed from pump discharge to the flush connection in the seal gland plate to carry away the heat generated by the seal faces.



DEAN P1200 ECONOMY FLUSH PLAN FOR PROCESS PUMPS

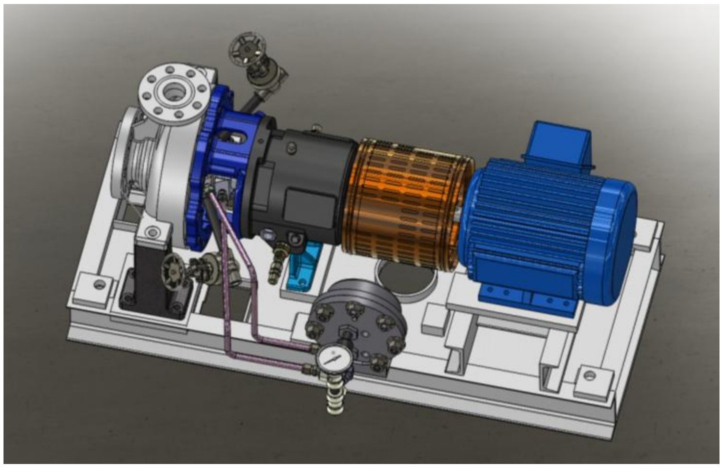
For temperatures **above 250° F,** a **heat exchanger** is highly recommended. An ANSI Plan 7321 (API Plan 21), ANSI Plan 7323 (API Plan 23), or Dean Seal Guard B System would be good options. The heat exchanger selected must be capable of bringing the temperature in the seal area down to around 160° F. If a cooled flush is not provided, the liquid film between the seal faces will begin to flash and the seal faces will "run dry" causing the seal to fail prematurely.

The ANSI Plan 7321 (API Plan 21) is a flush line from the pump discharge, through a heat exchanger and into the connection of the seal gland plate to flush the seal faces. The flow is controlled via an orifice, with the size determined by the differential pressure produced by the pump. Temperature indication is when specified (optional).



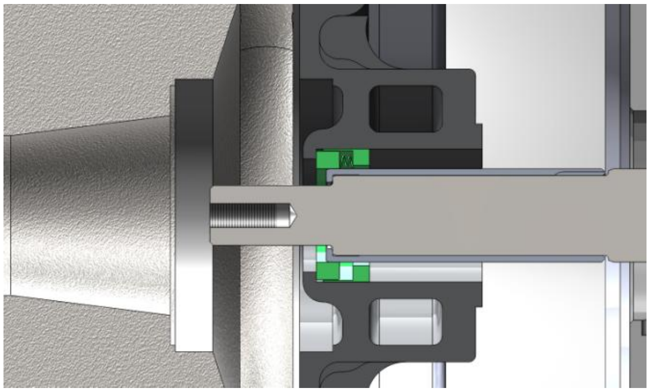
API Plan 21 (ANSI Plan 7321)

The ANSI Plan 7323 (API Plan 23) is a flush line from the seal chamber through a heat exchanger and back into the seal flush connection on the gland. This plan requires the mechanical seal to be equipped with a pumping ring to circulate the water through the heat exchanger. The advantage to this configuration is that the heat exchanger is cooling water from the seal chamber which has already been cooled by the jacket. The disadvantage is that it requires a pumping ring which does not work well at slow speeds. Temperature indication is when specified (optional).



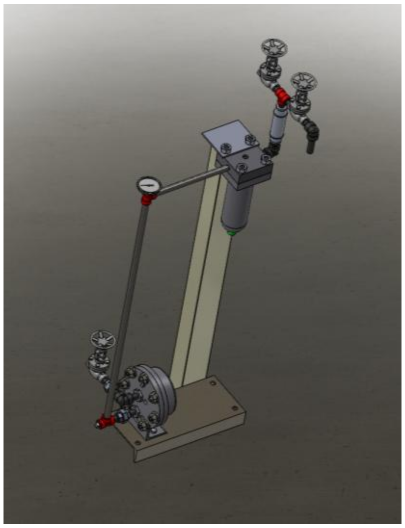
API Plan 23 (ANSI Plan 7323)

The use of a **Dean Min-Flo Type-S bushing** positioned in the bottom of the seal chamber is recommended for both plans. This bushing separates the cooler water in the seal chamber from the hotter water in the pump casing and reduces the amount of flow from the seal chamber back into the pump. The Type-S is provided with a carbon bushing and 316SS retainer, set screws and springs.



It is recommended that these seal plans be furnished with a temperature indicator (thermometer) and a needle valve located in the flush line between the cooler and the flush connection on the gland. Cooling water flow to the seal should be in the range of one-half to one gallon per minute and the temperature should be in the range of 140°F. The needle valve permits close regulation of the flow. Cooling water flow to the heat exchanger needs to be in the 7 to 15 gallon per minute range, depending on the heat exchanger.

The Dean Seal Guard B System consists of a flush line from the pump discharge, first through a heat exchanger and then through a filter before returning to the flush connection in the seal gland plate. The Seal Guard B System is similar to an ANSI Plan 7321 (API Plan 21) except it has a filter. This filter will take care of the welding slag, rust and scale common in most new hot water systems. It also "cleans up" an older system which may have scale. If the water has solids in solution, when cooled they will precipitate out and the filter in the Seal Guard B prevents them from getting into the seal chamber. Seal Guard B units are furnished with a needle valve and thermometer.



Dean Seal Guard B System

Bearing Frame Cooling

Cooling of the bearing frame is not necessary for hot water applications. Excessive cooling of the bearing frame may actually lead to early bearing failure from moisture condensation contamination of the lubricating oil. Cooling of the bearing frame is required only on services where the R4000 pumps are used in temperatures above 500°F.

Summary

Hot water pumping does not need to be a problem. Care simply needs be used in the selection of the pump, mechanical seal and seal cooling system as well as the piping system. Following this paper a general specification which may be helpful to those needing hot water pumps for their systems.

GENERAL SPECIFICATION FOR HIGH-TEMPERATURE WATER PUMPS

1. GENERAL

a. High Temperature Water (HTW) pumps shall be designed for 500 psi working pressure. They shall be horizontal, single stage, center-line supported, end suction, vertical center-line discharge, centrifugal pumps with high-efficiency enclosed impellers. The pumps will be direct coupled to their drivers and shall be mounted on fabricated steel base plates and coupled with a 3.5" (minimum) flexible spacer coupling. Suction and discharge flanges shall be faced and drilled ASTM 300 lb. Raised Face (RF) Standard.

b. Pumps shall be vertical split case and shall be back pull-out design, capable of being dismantled without disconnecting the suction and discharge piping and without disturbing the drivers.

c. Head-capacity curves shall have rising characteristics from design point to shut-off.

d. Pump specification is based on Dean Model R4000.

2. CONDITIONS OF SERVICE

		Design	Maximum
Liquid			
Capacity	GPM		
Differential Head	FT		
Pumping Temperature	°F		
Specific Gravity			
Vapor Pressure	PSIA		
Suction Pressure	PSIG		
NPSHA	FT		
Pump Speed	RPM		
Cooling water Temp	°F		

3. DETAILS OF CONSTRUCTION

a. Casing should be cast steel (ASTM #216 Grade WCB or approved equal), self-venting, centerline support. Casing gasket shall be the confined type, suitable for the specified design pressure and temperature.

b. They shall be designed for maximum interchangeability of parts between sizes.

c. A large 1" (minimum) visual sight glass will be provided to monitor the oil level.

Labyrinth type seals (bronze fitted with Viton® O-rings) on the bearing frames shall be provided in place of traditional lip seals. These seals will help ensure that the bearings are kept properly lubricated and uncontaminated throughout their projected design life.

d. The coupling guards provided will be totally enclosed, cylindrical "telescoping" design".

e. Seal chamber shall be the same material as the casing and shall have an integrally cast water cooling jacket. The seal chamber jacket shall be designed to maintain a temperature of 160° F in the seal chamber when the pump unit is not operating and when supplied with 5 GPM of cooling water at 78° maximum. The seal chamber shall be fitted with a throat restricting bushing similar to Dean Min-Flo bushing to reduce the interchange of cooled seal water in the seal chamber and hot water in the pump casing.

f. All pressure containing components shall be hydrostatically tested with ambient temperature water at 750 psig.

g. Wearing rings shall be provided on both the casing and seal chamber. The case wear ring and the seal chamber wear ring shall be made of hardened iron.

h. Impeller shall be enclosed design of cast iron. Dynamic balancing shall be provided. Impeller shall be keyed to the shaft and held tightly in place by a 18-8 SS impeller nut or by a bolt and separate washer locked to prevent bolt or nut from accidental removal.

i. The impeller shall-be-fitted with impeller wear rings of steel on both sides of the impeller. The difference in hardness between opposed wearing rings shall be at least 50 BHN.

j. Bearing housing shall be cast iron. Bearings shall be standard type anti-friction design ball, selected to give 25,000 hours minimum life at rated pump conditions but not less than 16,000 hours at maximum axial and radial loads and rated speeds. Bearings shall be oil lubricated.

k. Shaft shall be steel having a tensile strength of at least 125,000 lbs. and a yield strength of at least 100,000 lbs. It shall be of ample size and properly proportioned to limit shaft deflection to a maximum of .002 inches at the face of the seal chamber under the worst conditions of shut-off head and 1.0 sq. in the most heavily loaded casing used with the particular shaft size. The bearings shall be protected by a suitable deflector or closure at each end of the bearing housing.

l. Shaft sleeve shall be of the renewable "hook-type" clamped in place by the impeller and free to expand at the outboard end. Leakage under the sleeve shall be prevented by a lapped metal-to-metal fit against the shaft shoulder and impeller. Shaft sleeve shall be accurately machined and ground and polished through its full length. When mechanical seals are used the shaft sleeve shall be made of 316SS; when packing is used shaft sleeve shall be 420SS hardened to 500 Brinell.

m. Pumps shall be provided with casing drain. The casing discharge gauge connection shall be provided for use as a seal flush connection.

4 MECHANICAL SEAL

a. Each pump shall be fitted with a single inside balanced mechanical seal, John Crane, Inc. type 8 B1T $X_{18}F_{48}1O_{15}1$, with high temperature water carbon (F_{48}) versus tungsten carbide faces (O_{15}), 316 stainless steel fitted with Aflas (X_{18}) "O"-ring elastomers or equal. Other seal vendors to be considered are ASI and Flowserve. The gland shall be 316SS Flush type.

5 MECHANICAL SEAL COOLING SYSTEM (Option- Select a, b. or c)

a. Sealguard (Option)
Each pump shall be provided with a Dean Pump Model B500T seal guard system with a design pressure of 500 psig. The mechanical seal will be flushed through a flush gland or flush connector by the liquid being pumped using a flushing loop in which the pumped liquid is taken from the casing through a valve and into a heat exchanger. After leaving the heat exchanger, the pumped liquid will go through a suitable cartridge-type filter, then into the stuffing box or flushing gland. The filter, valve, heat exchanger and instruments shall be suitable for the operating temperatures and pressures of the system and in addition to cut-off valves to isolate the flushing loop, there shall be a thermometer fitted in the line between the heat exchanger and the filter to determine the temperature of the liquid after leaving the heat exchanger. There will be a suitable flow meter with a range for setting the flushing flow at from ¼ GPM to ½ GPM in the pipeline after the filter and before going into the stuffing box. The heat exchanger shall be of sized to cool ¼ GPM to ½ GPM from 400^0 F to 160^0 F hot water with a cooling supply to the heat exchanger of 78^0 F. Min- flow bushings shall be supplied to reduce cooling water flow back into the pump.

b. Seal Flush and Cooling - Plan 21 (Option)
Each pump shall be provided with the manufacturer's standard seal flush and cooling plan similar to ANSI Plan 7321 or API Plan 21 which shall include a flush from discharge through a heat exchanger and to the seal flush gland. The heat exchanger shall cool flush from pump discharge at high temperature to 160^0 F. A temperature indicator and valve shall allow flow adjustment of the cooled pumpage to the correct temperature. The bottom of the stuffing box shall be fitted with a close fitting restricting bushing to restrict interchange of flow between the pump case and the seal chamber.

c. Seal Flush and Cooling - Plan 23 (Option) .
Each pump shall be provided with the manufacturer's standard seal flush and cooling plan similar to ANSI Plan 7323 or API Plan 23 which shall include a flush from the seal

chamber through a heat exchanger and to the seal flush gland. The mechanical seal shall be fitted with a circulating ring to circulate the flush liquid. The heat exchanger shall cool flush from pump discharge at high temperature to 160^{0} F. A temperature indicator and valve shall allow flow adjustment of the cooled pumpage to the correct temperature. The bottom of the stuffing box shall be fitted with a close fitting restricting bushing to restrict interchange of flow between the pump case and the seal chamber.

6. ELECTRIC MOTOR (Option - engineer's specification)

7. TESTING (Optional additional charge)

a. Completed pump will be performance tested in accordance with requirements of the Hydraulic Institute at a minimum of four points on the performance curve. Results will yield capacity, head, and horsepower. An NPSH Test will be conducted only if NPSH required by the pumps is within four feet of system NPSH available at the design point.

b. A, certified curve plotted from test data including head, capacity, efficiency, brake horsepower and NPSHR shall be provided.

c. Each finished pump assembly will be hydrostatically tested to the maximum pressure of the mechanical seal and certification of this hydrostatic test shall be provided. .

7. SPARE PARTS (Optional additional charge)

With each pump, the supplier shall furnish the following spare parts, packaged individually, and properly tagged for identification. Material shall be identical with those used in the original units. .

1 -Set wear rings
1-Shaft sleeve
1-Mechanical seal assembly complete, including gaskets, less gland
1-Set bearings
2 -Casing-gaskets
1–Set labyrinth oil seals and end cover gaskets 2- Filter elements

8. START UP (optional additional charge)

The pump supplier shall provide the services of a trained individual to inspect, start up and instruct owner personnel in the operation of the equipment provided. This service shall be provided by the authorized pump distributor in the area where the pump is located.