

DESIGN AND OPERATION OF PUMPS FOR HOT STANDBY SERVICE

by

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ABSTRACT

Pump thermal distortion problems routinely arise in refinery applications where hot fluids must be handled. This is particularly the case when warming up standby pumps. The single stage, double suction configuration, which is best suited for such services, is analyzed in terms of the mechanical interferences that arise from thermal distortion. This is done with the aid of finite element modelling, which not only led to the solution of the interference problems via design improvements, but also led to recommendations for application and operation of pumps under hot-fluid conditions.

APPLICABILITY OF SINGLE STAGE DOUBLE SUCTION PUMPS

In a variety of refinery pumping services, high flowrate and developed head requirements dictate the type of pump required. In addition, these services can also have high pumping temperatures (above 400°F/200°C) and liquid specific gravities below 0.70.

Pump design alternatives for these applications include the following [1]:

- single stage, double suction, vertically split
- single stage, overhung, vertically split
- multistage vertically split (barrel)
- two stage vertically or horizontally split (single case).

Advantages of the single stage, double suction, vertically split design (Figure 1) over the other alternatives listed makes it the obvious selection:

- A double suction design requires less suction pressure than a single suction design (typically at least 30 percent less) at the same speed. This also means that a smaller, higher speed pump can be selected, which may also cost less and be more efficient than a single suction unit.

- A single stage unit can more easily be arranged with a radial (circular) main casing joint with a confined, controlled compression gasket to more positively contain hazardous (flammable, toxic) liquids than a multistage unit.

- A double suction symmetrical design can be constructed with a heavy shaft extending through the impeller eyes (inlets) and supported with bearings at both ends of the shaft (an API-610 requirement for double suction impellers). This reduces shaft deflection at mechanical seal faces and wear rings and re-

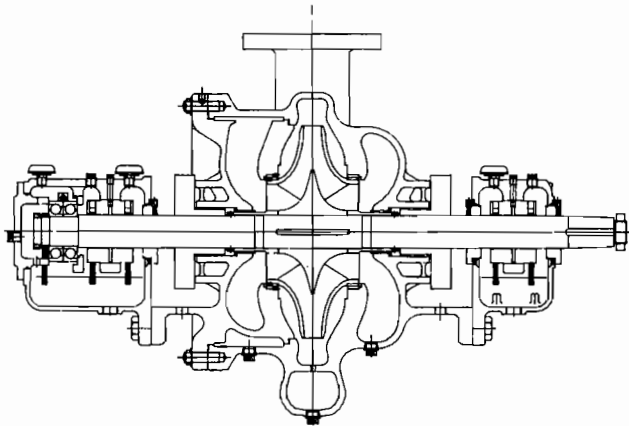


Figure 1. Typical single Stage, Double Suction, Vertically Split Process Pump Design.

sults in a stable rotor design that operates well below first lateral critical speed levels.

- The nominal axial balance of the single stage double suction design minimizes thrust bearing loads when suction pressures are high. Overhung pump designs must accommodate suction pressure thrust because of the single stuffing box/seal chamber design. Multistage pumps are usually arranged to balance the pressure on each seal chamber, but this is usually accomplished in conjunction with close fitting throttling devices, such that axial balance can change (significantly) with normal pump wear.

- Suction and discharge nozzles can usually be arranged for either top or side locations, depending on which is more convenient for the specific application. Usually top suction and discharge arrangements are preferred for hot services with overhead piping to improve piping arrangements and provide more resistance to nozzle forces and moments created by thermal growth.

- A double suction impeller can have a lower inlet tip speed than a single suction impeller. Since refinery fluids may contain suspended solids, this means impellers with lower tip speed will tend to suffer less abrasive erosion.

There are, of course, some disadvantages to the single stage double suction design as well:

- A two bearing design requires two mechanical seals, whereas a singlestage overhung design requires only one seal; this means that for low flows, and where suction pressures are adequate for NPSH requirements, the overhung design may be preferred.

- The higher energy-per-stage in a single stage double suction pump may dictate higher minimum flows than a multistage unit. Also, for higher head applications, the specific speed of a multistage pump may be more nearly optimum, and result in higher operating efficiency. Units of two or more stages, however, are more susceptible to internal alignment problems when high operating temperatures impose severe thermal forces and temperature gradients throughout the pump.

- Double suction pumps—whether single stage or the first stage of multistage pumps—require symmetrical flow to the suction nozzle. If piping leading to the pump creates unequal flow to each impeller eye (inlet), an axial unbalance can occur, leading to high thrust bearing loads. This also results in unequal flows within the impeller, which may cause one side of the impeller to cavitate. Unequal flows can also result in rotor oscillations, which cause unsteady bearing and seal loads—and premature failures. Suction piping arrangements are less critical on single suction pumps.

This presentation will outline experiences with large single stage double suction refinery pumps applied to severe hot bottoms service. It will discuss the application and operational problems encountered and how they are being addressed and resolved.

The pumps described in this application are single stage, double suction, vertically split with a 10 in discharge, 16 in suction and 31 in nominal impeller diameter. The pumps are rated for 5,000 US gpm, 295 ft, operating at 1780 rpm. The service is main fractionator bottoms, 640°F (338°C) with a specific gravity of 0.552 at pumping temperature. The pump was designed for 720 psi working pressure at the pumping temperature. Material of construction is carbon steel (ASMT-A216 WCC). The pump is equipped with sleeve radial bearings and ball thrust bearings (Duplex, back-to-back) with oil ring lubrication and purge mist. The pump was built in accordance with API-610, 5th Edition. The mechanical seals were specified to be of single, balanced cartridge design, with external cool oil flush.

This pump represents one of the largest single stage machines applied to this type of service anywhere in the world. Not only is the pumping temperature high, but the ambient temperature (Middle East) is also very high.

INSTALLATION AND OPERATING PROBLEMS

The operational problems discussed herein will concentrate on those associated with high temperatures.

Each pumping unit was arranged (as is typical) with two pumps—one operational and one spare (hot standby). Suction and discharge piping was oriented at the top of the pump (top nozzles) with hangers and supports to reduce nozzle loads. The suction pipe, at about 10 pipe diameters from the pump nozzle (flange) turned 90 degrees and ran parallel to axis of the pump shaft. In this particular instance, the distance of the 90 degree turn from the suction nozzle was probably sufficient to at least minimize distortion of the flow into the suction nozzle—and thus to the impeller inlets. However, normally, it is critical that attention be given to suction piping (and flow distribution) on double suction pumps (impellers) to ensure that flow is equal to both sides of the impeller to prevent poor suction performance, axial rotor unbalance and axial rotor shuttling—especially at flow less than design flow.

The mechanical performance of the first unit, which was started cold and brought up to design temperature slowly with the process, was satisfactory. Major problems arose, however, when the “hot” standby pump rotor would not turn by hand without experiencing internal rubbing (hard contact) after approximately 180 degrees of turn. After an elapsed period of about four minutes, the rotor could undertake a further rotation of 180 degrees before contact was experienced again. Thus, hot standby startup could not be achieved.

Investigation revealed several possible factors contributing to this inability to rotate the pump rotor freely on hot standby:

- The warmup flow to the pump case (Figure 2) consisted of a singular pipe to the discharge piping of the standby unit such that uniform distribution of the warmup liquid through the pump casing and around the rotor was not being achieved. In view of the large physical size of the pump and the thermal stratification effects, the location of the warmup line, and probably the quantity of warm-up liquid as well, was inadequate [2].

- Coupling alignment was changed significantly from the “cold” to the “hot standby” condition (as much as 0.020 in- 0.030 in). It was suspected that nozzle loads were moving the pump relative to the drive. It was also suspected that the pump was trying to expand axially and transversely and was being restricted, and that the shaft was moving independently of the pump casing.

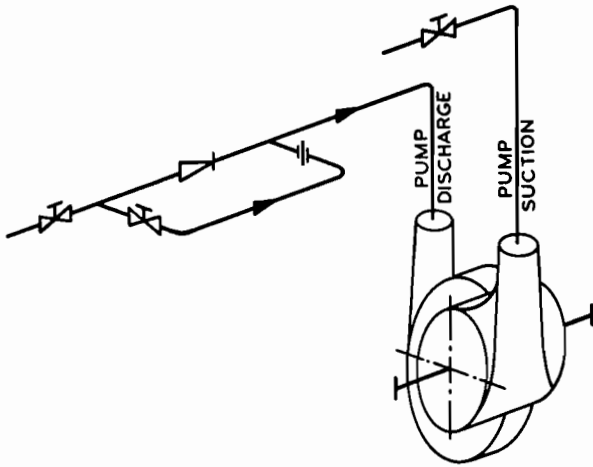


Figure 2. Warmup Piping Showing Top Warmup Connection.

INVESTIGATIONS

To evaluate the influence of the original warmup flow to the pump casing, a finite element model (Figure 3) of a portion of the nearly symmetrical pressure containment vessel was created such that thermal gradients could be studied and distortions (movements) could be evaluated relative to known measurements (Figure 4). When we imposed known deflections on the model, the total displaced geometry could be viewed and studied.

It was found that the original warmup flow arrangement was doing very little good. It appeared to be short-circuiting (Figure 5) and simply going in and out of the top of the pump case. It was, in fact, adding to the distortion (Figure 6).

The configuration of the casing was also reviewed to determine if changes in construction (reinforcing ribs, etc.) could add significantly to structural rigidity and add resistance to thermal distortion. After reviewing the finite element model, this was rejected in favor of maintaining a smooth casing profile. Thermal

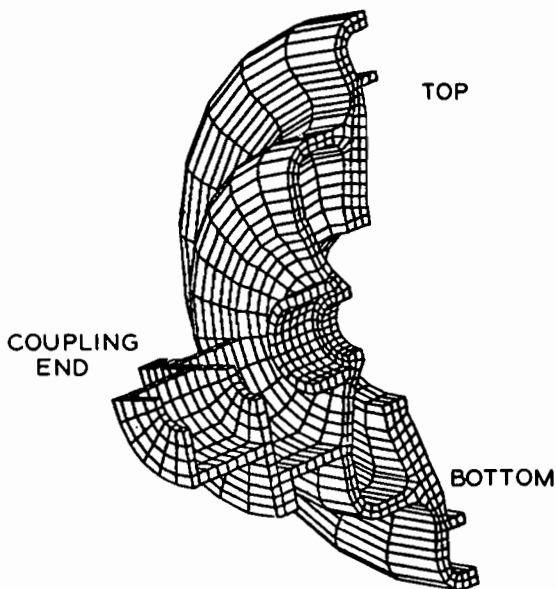


Figure 3. A Finite Element Model of a Portion of the Nearly Symmetrical Pressure Containment Vessel.

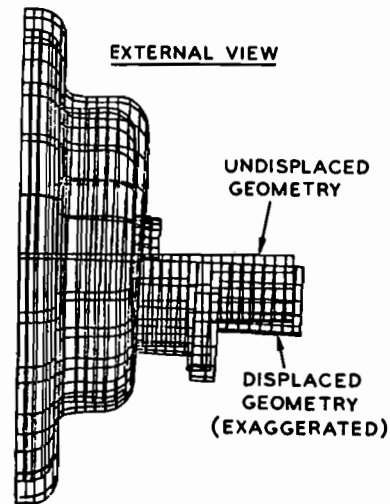


Figure 4. Model Showing deflection resulting from Original Warmup Flow into Top Connection..

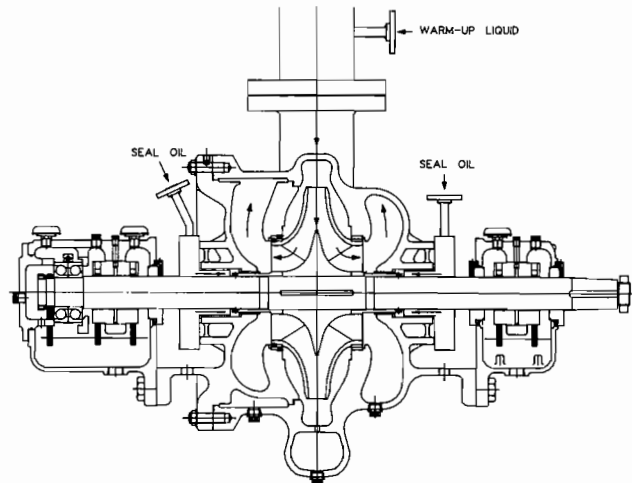


Figure 5. Top Warmup Flow Connection Results in Short Circuit. Warmup Flow Simple Goes in and out of the Top of the Pump Case, Resulting in Severe Thermal Gradients.

gradients in major reinforcing ribs could actually create higher levels of distortion, and *stress*, if not precisely located and sized.

Originally the pump was dowelled securely to its mounting pedestals without provision for expansion in any direction. This is routinely done on smaller units of this type to maintain pump position on its bedplate and maintain alignment with the driver. With such a large pump, however, it appeared that “planned” expansion, particularly in the radial direction, could help eliminate distortion and shaft movement at the coupling.

Because of the volatile nature of the pumpage, the single balanced mechanical seals were arranged with cold gas oil flush to each seal chamber from an external source (API-610 Seal Piping Plan 32/62) (Figure 5). This flush was injected into the seal at a single point. Since the flush has to be on the idle standby pump as all times (for safety purposes), it has the effect of cooling the shaft and seal chamber area while the rest of the pump was being heated. Actually, the casing was being heated nonuniformly with the single point warmup flow in the discharge pipe and the shaft and seal chamber area were being cooled nonuniformly

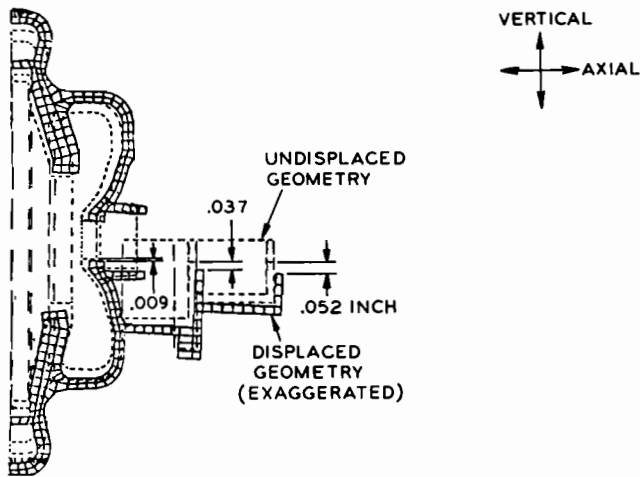


Figure 6. Model Shows Casing Displacements Resulting from Top Warmup Flow Which Actually Added to Distortion.

with the single point injection of the cold gas oil flush. Further nonuniform cooling took place as the spent cold gas oil stratified in the bottom of the pump casing adjacent to the seal chambers. The net result was severe "hogging" of the shaft, bowing of the casing, and binding of the rotor.

CORRECTIVE ACTIONS

The finite element model was constructed such that field measurements of pump casing surface temperatures (top to bottom) could be used as input. When this was done, resulting distortions were in close approximation to those values actually experienced on the hot standby pump. The distorted model with actual measurements of deflections imposed is shown in Figure 6.

The finite element analysis showed that multiple point warm-up injection would be required to uniformly heat the casing. With single point injection, the bearing housing brackets were actually moving and twisting relative to the close running clearance diameters in the casing, thus causing rotor contact as well as bearing misalignment. By replacing a single 3/4 inch injection

point at the top of the pump with three one-inch connections on the bottom of the casing (one in the volute and one in each of the suction passages, adjacent to the seal chambers and bearing brackets—Figure 7), it was predicted that the flow of warm-up liquid would be substantially increased and would flow more uniformly throughout the pump casing, resulting in uniform casing growth (Figure 8). In fact, by this means, the original approximately 250°C differential across the casing was reduced to an acceptable 25°C. The ways casing temperatures were distributed and inputted to finite element program are shown in Figure 9. Another displacement output to evaluate effect of 360 degree bearing bracket is shown in Figure 10. In arranging for injection into the suction passages, it was determined that convective upward flow should be controlled so as not to impinge on the pump shaft. The concern was that such impingement contributed to additional bowing of the shaft.

Again studying casing thermal growth, it was decided that the pump, because of its physical size and near cube-like overall dimensions, should not be dowelled, in order to avoid any possible internal distortion. Instead, keys were fitted at the centerline of both pump feet to allow free radial expansion and axial expansion toward the drive (Figure 11).

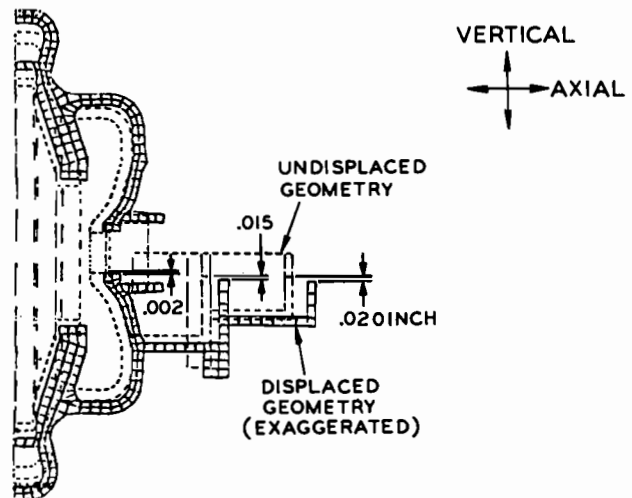


Figure 8. Model Shows Improvement in Casing Displacements with Improved Bottom Warmup Connections.

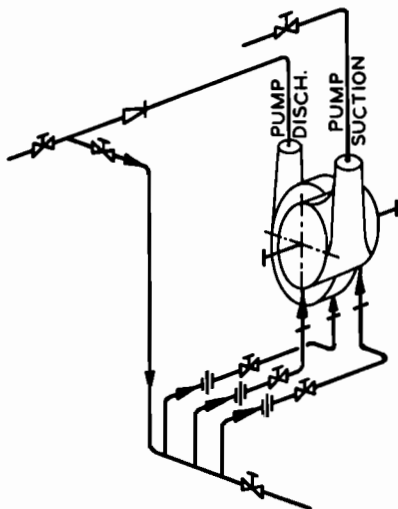


Figure 7. Improved Warmup Flow Piping into the Bottom of Each Casing Passage.

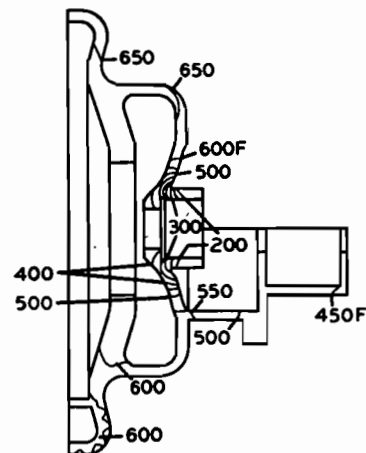


Figure 9. Distribution of Casing Temperatures.

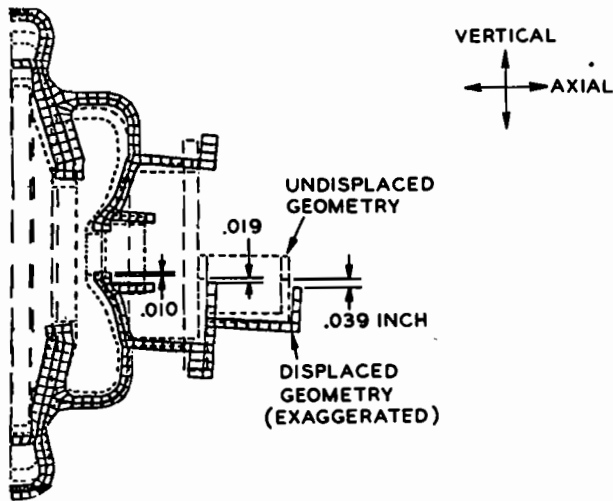


Figure 10. Effect of 360 Degree Bearing Support on Distortion. Compare to Figure 8.

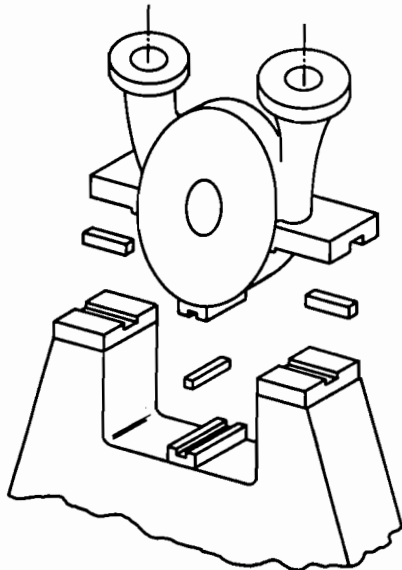


Figure 11. Keys Fitted to Allow Free Radial and Axial Expansion while Minimizing Distortions Caused by Restricted Movement.

In addition, a key block was added to the lower centerline of the pump casing to mate with a block on the baseplate. By providing these sliding features to allow the pump to expand both axially and transversely, alignment with the driver could be assured while avoiding any restrictions which could add to the distortion problem.

The coupling alignment is maintained by the transverse key fitted at the bottom of the pump. The keys were fitted with 0.005 in side clearance (maximum) to limit angular movement. To evaluate the effectiveness of corrective actions, the pump was warmed up with the coupling spacer removed and sufficient time allowed for the pump to reach stabilization. Pump to motor alignment was determined with the pump in the hot condition using a "three clock" method. Three dial indicators were attached to the motor coupling hub with one indicator registering pump coupling hub rim runout and two indicators registering pump coupling hub face runout. Two indicators were necessary

for the face measurement to compensate for pump shaft axial movement during rotation. Indicator support arms were sufficiently rigid to avoid sag over the distance between the two hubs. Alignment (to less than 0.002 in) was achieved by adjusting the motor position and shimming under the motor feet. A "soft foot" check was performed on the motor after alignment.

Pump casing skin temperatures were obtained by the use of ten thermocouples attached strategically to the casing by spot welding. Thermocouples were also attached to the warmup line to monitor the warmup liquid temperature and to the seal oil feed line to monitor the seal oil temperature. All thermocouples were connected to a chart recorder which registered each temperature every 60 seconds.

The rate of flow to each warmup injection point was controlled by a valve such that a warmup rate not exceeding 3°C/min and a maximum differential temperature at any point in time between all pump thermocouples of 30°C could be achieved. It was found that the maximum differential objective was attainable, but it was not possible to maintain the 3°C/min rise at all points in the very initial stages of warmup. A steady state temperature was reached after approximately two hours of introducing hot liquid. Ultimately, the warmup liquid control valves were to be replaced by sized orifices.

Shaft to casing relative movement could be monitored only where the shaft extended from the coupling end bearing housing, because of insufficient space between the seal glands and the bearing housings. A single dial indicator was attached to the coupling end bearing housing to register the relative vertical movement of the shaft during warmup. It was shown that this indicator was a dependable guide to the extent of bowing of the shaft at the coupling, the deflection at that point being approximately half that at the coupling. This deflection could be verified during shaft rotation after warmup. A limit of 0.005 in was placed on the acceptable shaft runout for startup, knowing that the shaft would fully straighten very shortly after startup.

To minimize any possible error arising from the growth of structures in close proximity to the hot pump casing, all dial indicator supports were carefully insulated. Care was taken to ensure that support structures themselves were sufficiently rigid to withstand reasonable accidental knocks. The axial movement of the pump suction line relative to a fixed point was also monitored to establish its possible effect on the pump casing movement during warmup. In addition to the continuous monitoring of temperatures, all indicator readings were recorded at 15 minute intervals.

These modifications resulted in a significant improvement. The shaft was now capable of being rotated continuously through full 360 degrees. However, an unacceptable upward shaft bowing at the coupling prevented automatic startup from the hot standby condition. Coupling misalignment was more than was desired prior to startup. Further testing clearly indicated a relationship to the cold flush oil injection. Shaft movement at the coupling could be modified significantly by altering the rate of flow of the cold flush oil injection, obviously not an acceptable situation.

It was determined that the shaft bowing resulted from the cold oil flush impinging on the shaft over a restricted circumference due to the single point seal injection (Figure 12). In conjunction with the seal manufacturer, the seals were modified to a simplified multipoint injection. At the same time, an attempt was made to avoid direct impingement of the hot "warmup" liquid on the underside of the shaft at the coupling end of the pump. However, these modifications resulted in only *limited success*.

To obtain further improvement and minimize the effects of the cold flush oil impinging on the shaft, arrangements were made to *preheat* the gas oil to approximately 100°C (212°F) in a

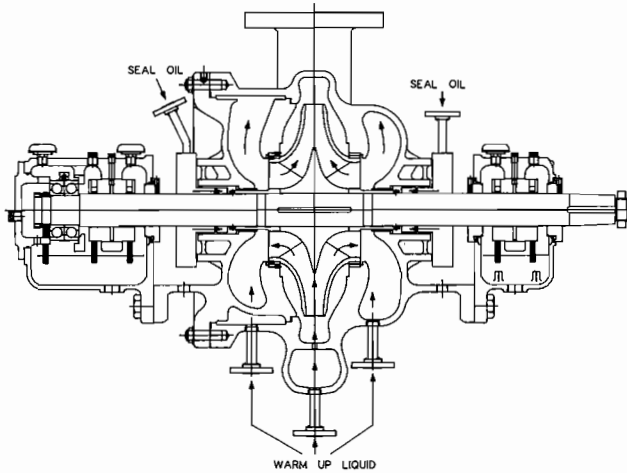


Figure 12. Bottom Warmup Connections in Each Pump Passage Improves Distribution of Warmup Liquid and Results in More Uniform Casing Growth.

steam heater. By this means, bowing of the shaft was reduced to an acceptable level such that the rotor did not bind and movement at the coupling was virtually eliminated. A successful start-up from the hot standby condition was made.

However, preheating of the gas oil is not considered an acceptable long term solution for many reasons, among which is the temperature of the shaft at entry to the bearings (high temperature close to the babbitt material).

Other limitations have necessitated further changes. Experience has shown that different rates of warm up flow are desirable to avoid excessive rates of temperature rise from cold to that required to hold the pump at a sufficiently high temperature to avoid thermal shock on start up. The use of different orifice plates in the warm up harness to control the rate and direction of flow would be desirable, but has been shown to be impractical in this particular installation because of the danger of plugging.

A long term solution has been being pursued based on the experiences gained to date (Figure 13). This will include the use of double mechanical seals, thus limiting the quantity of cold flush, and containing it so that it cannot enter the pump suction passages and cause thermal stratification and subsequent distortion. Tangential introduction of the hot warmup liquid (Figure 14) will minimize the impingement of hot liquid on the lower

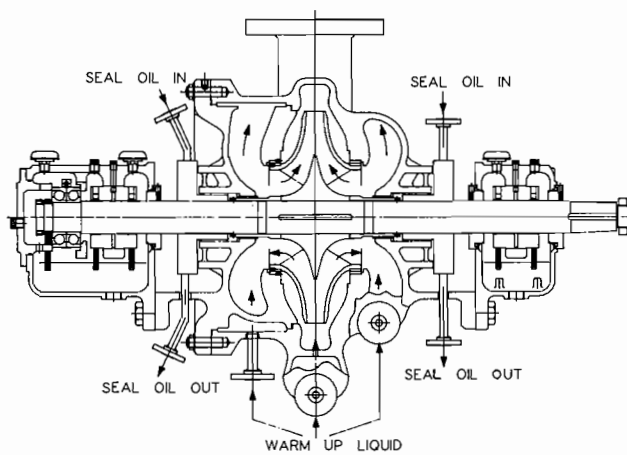


Figure 13. Tangential Injection of Warmup Flows Prevented Impingement on Shaft.

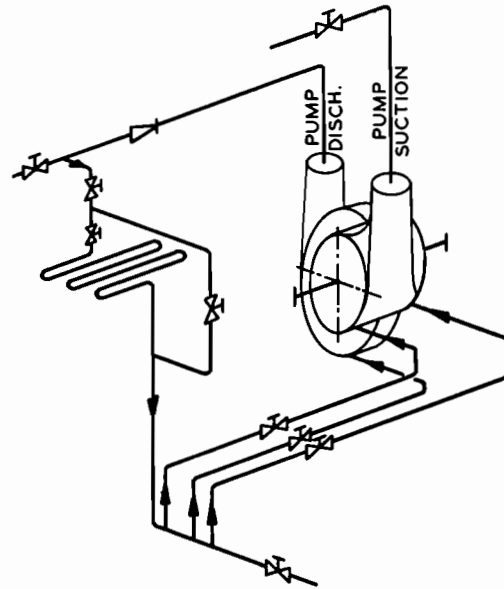


Figure 14. Warmup Piping Showing Tangential Injection and Temperature Control of Hot Flush Liquid.

portion of the shaft. A natural convection cooler may also be introduced into the warmup line to provide control of the pump warmup rate from ambient conditions.

CONCLUSION

It has been common practice to warmup pumps by introducing hot liquid into the discharge pipework and permitting the flow to permeate through the pump body to the pump suction. In general, this has been perfectly satisfactory in achieving two vital conditions for immediate hot startup:

- Controlling the warmup rate of the pump casing and holding the pump at a sufficient high temperature to avoid thermal shock on startup.
- Holding an acceptable differential temperature throughout the pump body to avoid thermal distortion and rotor binding.

As pumps become larger, a simple warmup flow to the discharge piping is not sufficient.

Thermal gradients within the body of a large pump can overcome the natural liquid flow resulting in stagnant areas within the pump and extensive temperature differentials across sections of the pump casing. Such thermal differentials can easily result in enough distortion to cause rotor binding (contact) and seizure.

The method of pump warmup must be given careful consideration for each pump installation. There is no universal "standard" solution. At present, it would appear that pump manufacturers are rarely consulted. It would be advantageous to manufacturers and users alike if more communication in this important area took place at the initial stages of a contract.

Many factors come into play in the decision to use a single or double mechanical seal, but *seldom* is one of those factors the consideration of thermal distortion of the pump rotor or casing. Experience has shown, however, that even mechanical seal selection can be important to satisfactory pump startup from a hot standby condition.

Both pumping temperature and pump size must be considered when evaluating warmup flow requirements for hot standby pumps. Larger pumps at high pumping temperatures

demand special attention to many of the procedures routinely applied to smaller, lower temperature machines. Each installation deserves individual consideration, and it is in the best interest of both the user and the pump manufacturer to communicate on warmup arrangements early enough to avoid unnecessary start-up problems.

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