Third in a 3-part series:

Centrifugal pump operation at off-design conditions

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The third and last part of this series of articles examines internal recirculation at the discharge, as well as the overload on the drivers of high specific speed pumps and the possible air- or gas-binding of pumps handling high concentrations of dissolved or entrained gases at low flows. It also prescribes the necessary by-pass arrangements when process demands fall below minimum flow rates recommended by the pump manufacturers. Parts 1 and 2 in this series appeared in the April and May 1987 issues of CHEMICAL PROCESSING.

Internal recirculation at the discharge

Internal recirculation at the discharge (Fig. 19), is a less frequent cause of field troubles but, on occasion, they can be quite severe. Like suction recirculation, it causes hydraulic surges and local cavitation at the impeller tips (Fig. 20). Incidentally, the fact that these effects, for both suction and discharge recirculation, are a form of cavitation is illustrated by the observation that physical damage in a high pressure multistage pump is generally limited to the first stage. Subsequent stages have a greater margin between the prevailing pressures and the liquid vapor pressure, and the effect is either minimized or eliminated.

Another effect frequently observed in connection with recirculation at the discharge of both single and double suction impellers is a marked axial instability of the rotor. This is caused by strong fluctuations in pressure outside the impeller shrouds. As the pressures acting on the two sides of the impeller vary, the axial balance is disturbed and the rotor moves now in one direction, now in the other. Kingsbury-type bearings have a very liberal axial clearance and may accommodate such motion readily. Ball thrust bearings have a much lower clearance for end-play due to tolerance build-up. The effect of the rotor axial shuttling is to cause premature failure of a ball thrust bearing by the repeated impact load. In extreme cases, this load may be sufficient to even deflect the mounting structure and cause its failure.

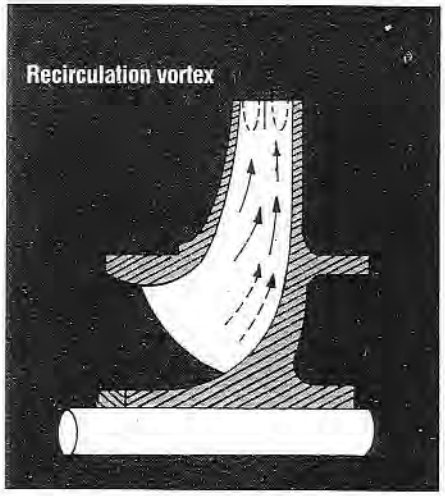


Fig. 19: Internal recirculation at impeller discharge

Just as "bulk-head" rings are sometimes used to alleviate the unfavorable effect of internal recirculation at the suction, a similar approach can be used to reduce the ill effects of discharge recirculation. It consists of providing projections from the casing walls. These projections, coming fairly close to the impeller outer shrouds as shown in Fig. 21, prevent propagation of the vortices formed by discharge recirculation down towards the center line of the impeller. This, in turn, reduces the fluctuation of pressures acting on the impeller shrouds and, hence, reduces the fluctuating axial thrust unbalance. Alternately, projecting rings can be fitted to the outer shrouds of the impeller (Fig. 22).

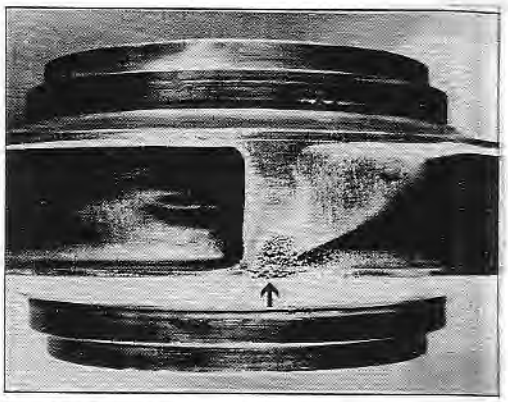


Fig. 20: Cavitation damage caused by recirculation at discharge

Damage to the impeller and pressure pulsations caused by discharge recirculation should not be confused with the effects of the so-called "vane-passing syndrome." This phenomenon is caused by the interaction between the moving impeller tips and the stationary volute tongues or diffuser vanes. A hydraulic shock occurs when the impeller vanes pass these stationary parts. The magnitude of the shock and the resulting pressure fluctuations increase with the impeller tip velocities and with the pump size, while the frequency is a multiple of the pump speed and of the number of impeller vanes.

Because the unsteady flow caused by the wake of the vortices shed from the vanes persists for some distance, the major factor in the magnitude of the shock resides in the gap between the vane tips and the volute tongues or diffuser tips. On the other hand, an excessive gap will act unfavorably on the pump efficiency. Opinions differ somewhat about

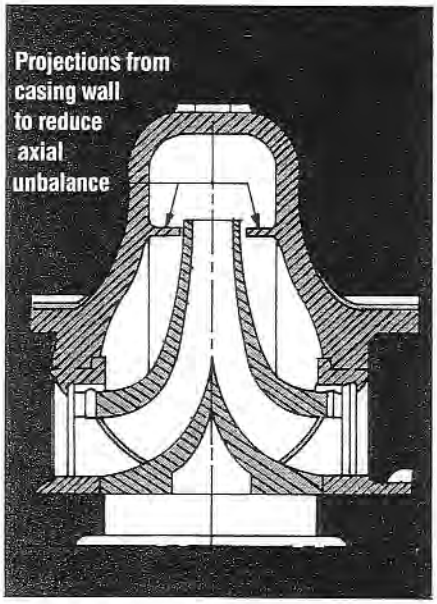


Fig. 21: Projections from casing wall provided to reduce problems created by discharge side recirculation

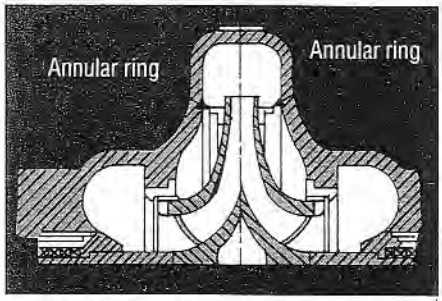


Fig. 22: Addition of two annular rings to impeller shrouds to reduce axial movement of rotor caused by internal recirculation at discharge

the minimum recommended gap, but it should not fall below approximately 4 to 6% of the impeller diameter.

Figs. 23 and 24 show the first stage and twin-volute of the first stage of a high pressure multistage pump which was subject to the vane passing syndrome because of an excessively close gap.

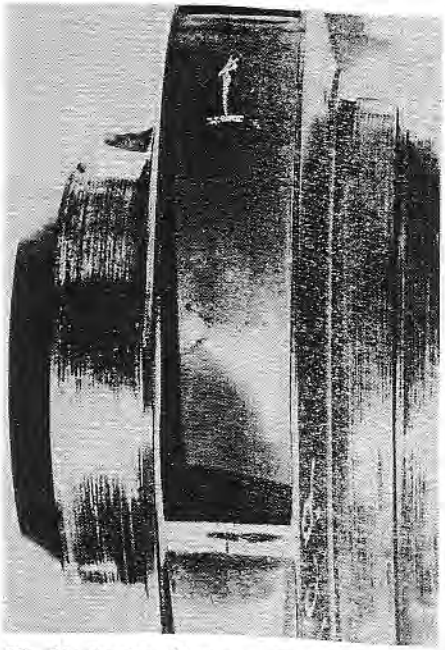


Fig. 23: Erosion of first-stage impeller tips when running with excessively close clearance

Instead of damage to the vane tips near the shrouds, it occurs near the central part of the vane. At a point where the pressure would normally be 500 to 600 psi above the vapor pressure, the marked cavitation erosion indicates that the shock pressure of the impeller vane tip passing close to the twin volute tongues was momentarily reducing the local pressure to less than the vapor pressure. This cavitation-erosion occurred only at the first stage. Pressures reached at the higher stages prevented the shock pressure from reducing the local pressures to values below the vapor pressure. This does not mean, however, that hydraulic shock and pressure pulsations were not being generated. The cure for the vane passing syndrome was to increase the gap by cutting back the volute tongues.

Tables I and II summarize the comparisons between classical cavitation and internal recirculation at suction as well as between the vane passing syndrome and internal recirculation at the discharge respectively.

Possible overload of high specific speed pumps

Pumps with low to medium specific speeds have brake-horsepower (bhp) curves that decrease with reduction in capacity. Thus, operating such pumps at lower than design flows will not overload the driver. On the other hand, if we deal with high specific speed pumps, such as one whose characteristics are illustrated in Fig. 25 (specific speed = 10,000), the bhp increases with a reduction in capacity. For instance, were such a pump to be operated at 50% of its rated capacity by throttling the discharge, the bhp would increase to almost 140% of the rated horsepower. If the motor size had not been selected with this end in view, it would be overloaded. Obviously, sizing drivers of high specific speed pumps so that they are not overloaded over the entire range of operating capacities down to 0% would be uneconomical. Thus, the minimum permissible flow is dictated by the selection of the driver horsepower.

Effect of entrained air or gases on minimum flows

When entrained air or gas enters a centrifugal pump along with the pumped liquid, the pump performance is affected unfavorably (Fig. 26). As the percentage by volume of air or gas increases, the flow at which it can be swept out of the pump increases and therefore so does the permissible minimum flow. ■

Minimum flow by-pass

If the pump is required to operate at shutoff, some means must be provided to prevent operation below the minimum permissible flows, regardless of whether the discharge valve or the check valve is closed. This is accomplished by installing a by-pass in the discharge line from the pump, located on the pump side of the

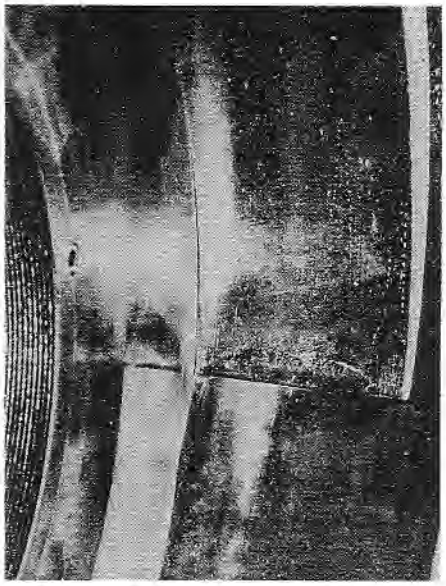


Fig. 24: Erosion of first-stage twin volute corresponding to impeller in Figure 23

TABLE I
Comparison of Classical Cavitation and Internal Recirculation at Suction

Classical Cavitation	Internal Recirculation at Suction
• Occurs at high flows	• Occurs at low flows
• Symptoms disappear a) As capacity is decreased or b) As NPSH available is increased	• Symptoms disappear as capacity is increased
• Damage to impeller on visible side of vanes	• Damage to impeller on invisible side of vanes
TO ELIMINATE:	TO ELIMINATE:
• Increase avail. NPSH or	• Operate at higher flows or
• Operate at lower flow or	• Get a redesigned impeller
• Obtain new impeller with lower req. NPSH	

TABLE II
Comparison of Vane-Passing Syndrome and Internal Recirculation at Discharge

Vane-Passing Syndrome	Internal Recirculation at Discharge
• Occurs over entire range of capacity	• Occurs at low capacities
• Frequency is multiple of pump speed and number of vanes	• Frequency is random
• Damage greatest at center of vane tips at PERIPHERY	• Damage greatest on vane tips near SHROUDS
• Can be eliminated or reduced by increasing gap between impeller and collector	TO ELIMINATE: a) Operate at higher flows or b) Get a redesigned impeller

check and gate valves and leading to some lower pressure point in the installation where the excess heat absorbed through operation at light flows may be dissipated (Fig. 27). Normally, this by-pass should not be led directly back to the pump suction, as there would be no means to dissipate the excess heat, and the entire purpose of the by-pass arrangement would be defeated.

However, if the minimum flow is established for other reasons than a permissible temperature rise (for instance because of internal recirculation conditions) and, therefore, this flow is quite significant (say, 25 to 50% of the design flow), a major portion of the by-pass may be led back to the suction piping close to the pump itself. The temperature rise at such flows is not very significant, and no harm can result if mixing occurs with the flow entering the pump. A second by-pass leading to a "heat sink" is provided to take care of the thermal effect when necessary.

In boiler feed pump installations where the pumps take their suction from a deaerating heater, the logical place to return the recirculation flow is the heater itself (Fig 28). Since the water in the by-pass line is under a pressure equal to the pump discharge pressure, an orifice must be located in this line to limit the flow through the line to the desired value. When the differential pressure broken down by the orifice is low, a single drilled orifice in a stainless steel rod 3 to 6" long can be used. Fig. 29 shows flows through orifices of different diameters and with various head differentials.

Elbows should not be located too close after an orifice to avoid damage to the piping. If the piping must bend, the orifice should be followed by a 12 to 18" length of straight pipe and T-joint (Fig. 30) leading back to the heater. The dead end of the T-joint should then be followed by another length of straight pipe, terminating in a pipe coupling. The end of the pipe coupling should be fitted with a stainless steel plug, which will take the brunt of the high velocity stream coming through the orifice. The plug will be cheaper and simpler to replace, if necessary, than an elbow in the piping.

When higher pressures than those indicated in Fig. 29 are encountered, or when the noise incidental to the breaking down of the discharge pressure through a single orifice must be reduced, special multiple pressure-reducing orifices can be obtained and installed.

The arrangement illustrated in Fig. 28 is based on a manual control of the by-pass. Two valves are used, one on each side of the orifice. One of these remains locked in the open position and is closed only to isolate the orifice for inspection or replacement. The second is the control valve and remains either open or closed. This valve is often automatically controlled and is made responsive to the flow through the pump. When the capacity to be by-passed is fairly large, modulating valves are preferred, arranged to open more or less widely, so that the flow to the process plus the flow through the by-pass valve can be made to correspond to the recommended minimum flow.

If an existing installation of several pumps that operate in parallel is provided with insufficiently sized minimum flow by-passes, one can avoid the need to replace all the by-pass valves and piping. The solution consists of manifolding the individual by-passes as in Fig. 31, which shows two boiler feed pumps in parallel for full load operation and a spare pump.

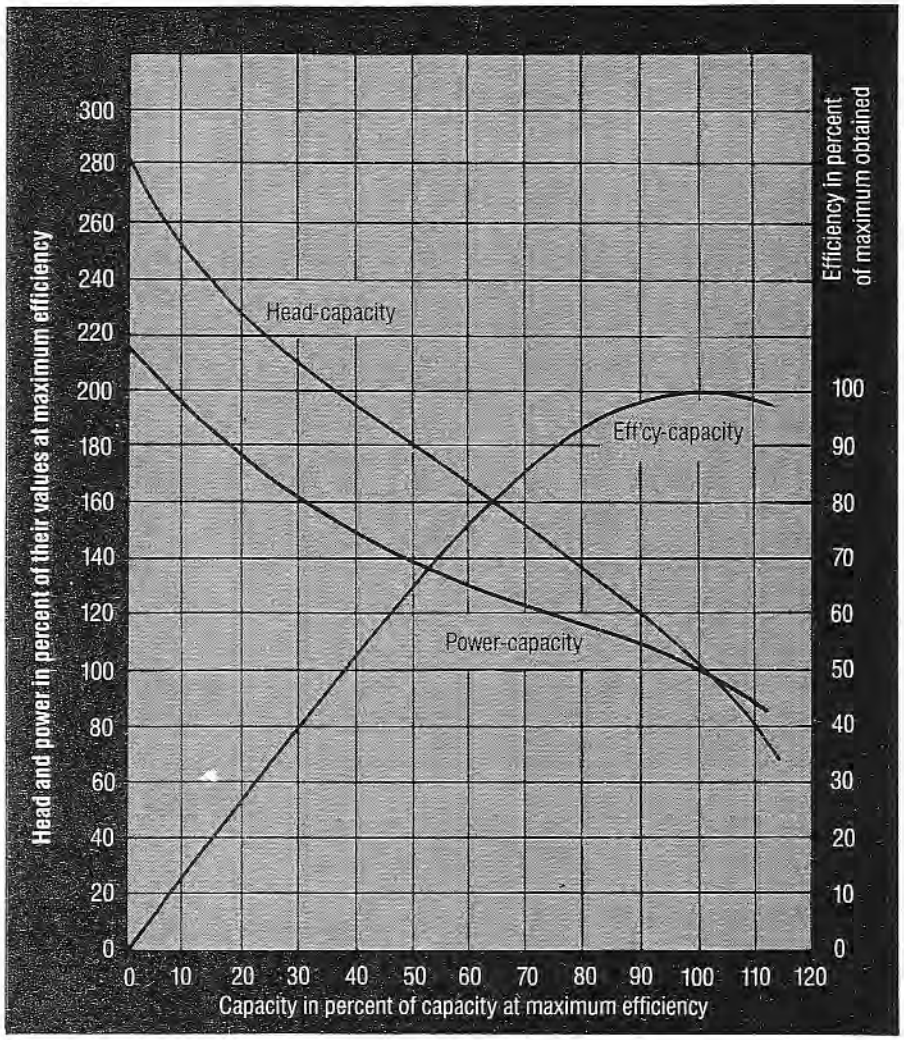
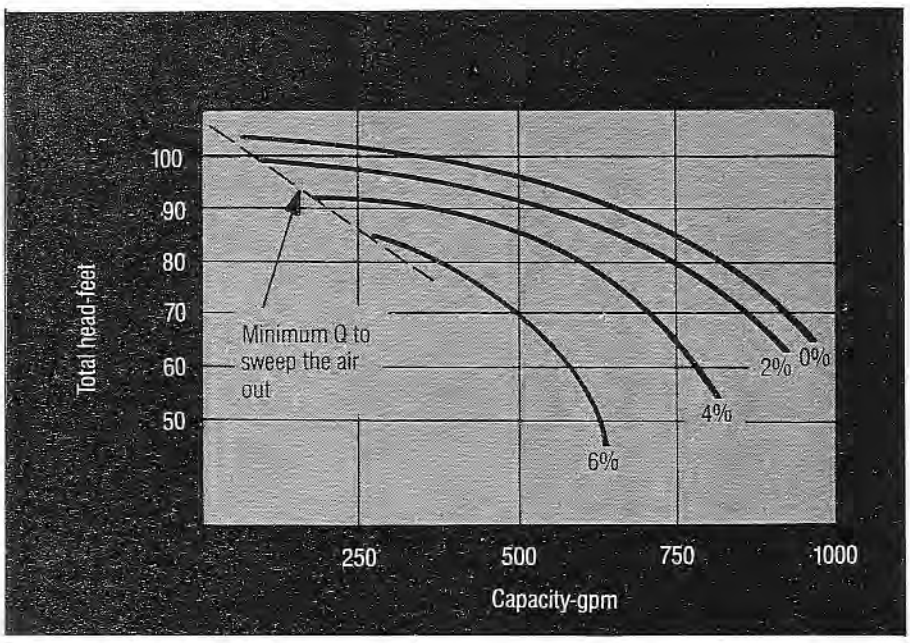


Fig. 25: Effect of high specific speed on hp/capacity curves

Fig. 26: Effect of entrained air on capacity



Assume, for instance, that the existing by-passes are sized for 20% of the design flow and that it is suspected that the proper minimum flow should be as high as 50 or 60% of design. The arrangement in Fig. 31 will permit to by-pass either 20 or 40 or 60% by manipulating the two manual control valves in the by-pass manifold.

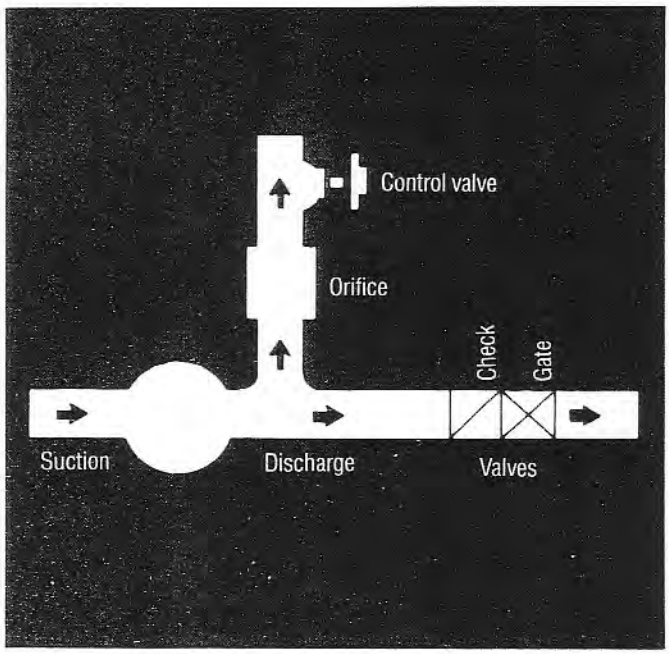
The reasoning is based on the fact that if, indeed, the pumps could operate satisfactorily with 20% minimum flow, the arrangement would permit them to do so and not waste energy. If, on the other hand, a higher minimum flow is necessary, opening one or both the manual control valves would take care of the requirement. Consider that in normal operation, the only time one may need to operate continuously with the by-pass open is when a single pump is running. The other condition requiring the by-pass to be open is during any switchover of pumps, that is when bringing a second pump on the line or when switching on the spare pump while two pumps are running, prior to shutting down one of these pumps. This is a condition of rather short duration, and no major problem should arise during the short switchover operation.

After the initial installation and the monitoring has been carried out and the decision on the proper amount of minimum flow has been reached, the operation of the three by-pass valves can be made automatic. The valves should be opened sequentially as follows, assuming that the minimum flow is established to be 60%:

1. When the flow to the boiler falls to between 40 and 60% of a single pump capacity, one valve opens.
2. When the flow falls to below 40% but above 20% of one pump's capacity, a second valve opens.
3. Finally, when the flow falls to below 20%, the third valve opens.

If the by-pass valves themselves are of the "modulating" type, they can continue to be so operated. In other words, assuming that the flow to the boiler is 50%, a single one of the by-pass control valves is used and is made to by-pass 10%, and so on.

If the minimum flow decided upon is 40%, the only time all three by-pass valves will be used is during the switching operation. Then with two pumps momentarily on the line, each pump can be run at 30% of its capacity.



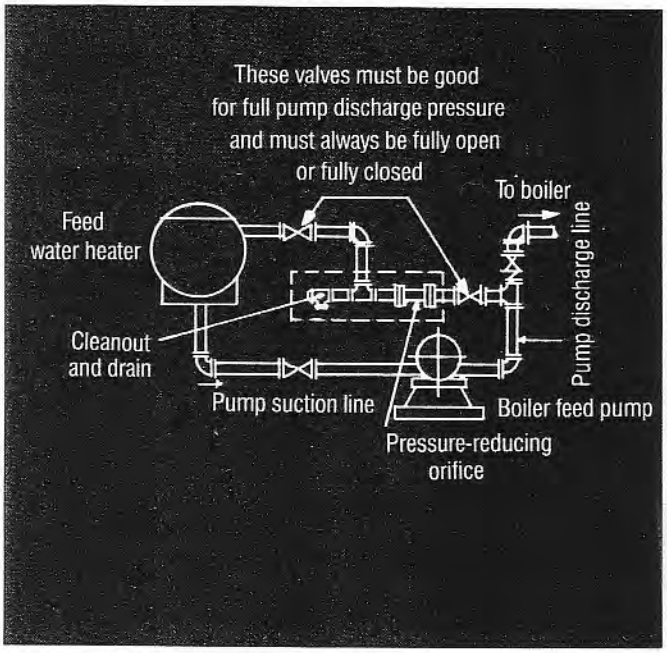


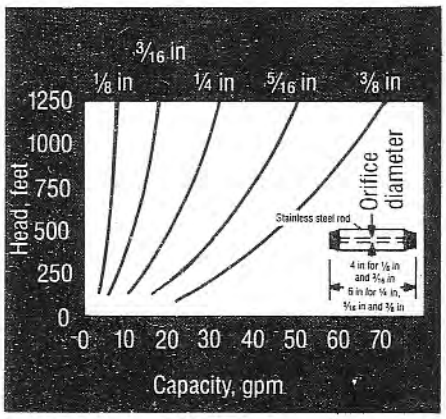
Fig. 27 (above): Arrangement for minimum flow by-pass

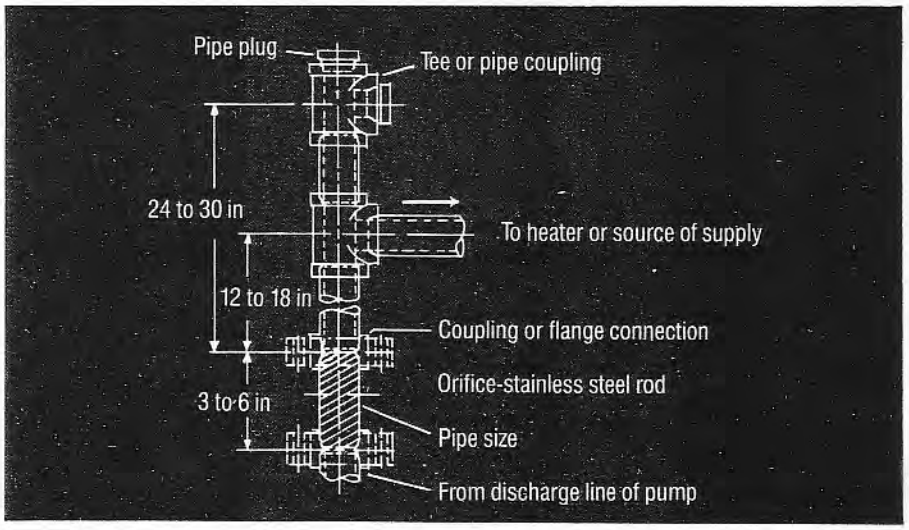
Fig. 28 (above, right): Arrangement of by-pass recirculation for boiler feed service

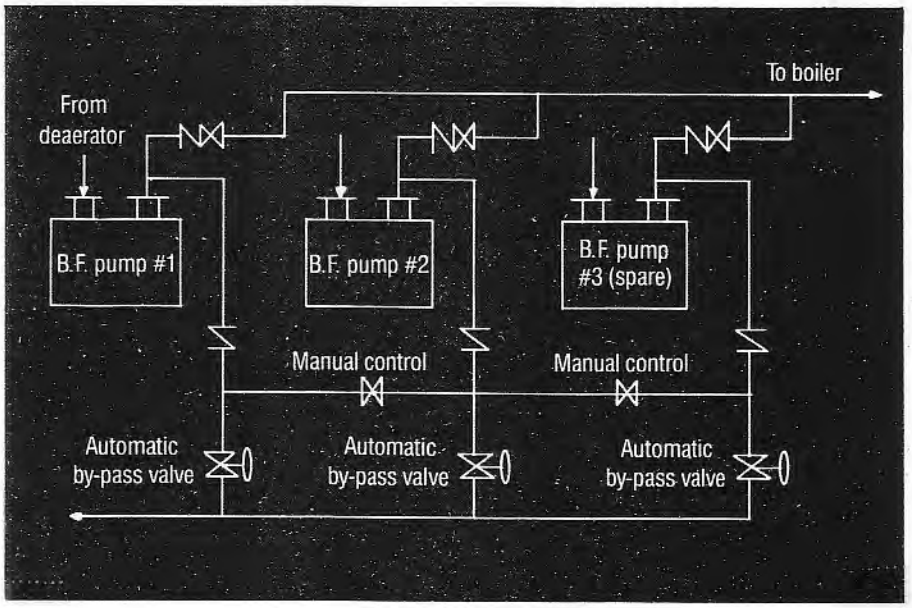
Fig. 29 (below): Flows through single orifices drilled in a stainless steel rod

Fig. 30 (right): Piping arrangement for a by-pass orifice

Fig. 31 (lower right): Suggested minimum flow by-pass arrangement







Distinction between "continuous" and "intermittent" operation

On occasion, manufacturers make a distinction between "continuous" and "intermittent" operation when recommending minimum flows for their pumps. But how do you distinguish between the two? Of course, one might select some quite arbitrary apportionment of time as a differentiation between the two. For instance, any operation of a pump at some specific rate of flow (or below it) that takes place for less than 25% of the time might be termed "intermittent," while

operation for more than 25% of the time would be "continuous." The problem created by such a definition is that if a pump were to operate for three continuous months of each year at flows below some critical value, and the remaining nine months at flows well above this value, one would find it difficult to accept the idea that the pump is operating "intermittently" at subcritical flows for the 3-month period. It would be equally difficult to assume that a pump is operating "intermittently" at subcritical flows if this were to take place regularly for six consecutive hours out of each twenty-four hour day.

We can clarify the situation somewhat if we consider that there are several separate end-results created by operation below some critical flow:

1. Excessive radial thrust, leading to shaft or bearing failure.
2. Excessive temperature rise.
3. In the case of internal recirculation, hydraulic pulsations and mechanical vibrations, leading to possible mechanical failure of the pump components, including that of the impeller, of the bearings and/or of the seals. These failures can be either progressive or instantaneous.
4. In the case of internal recirculation, cavitation type damage to the impeller. This will always be progressive.
5. Overload of drivers in the case of high specific speed pumps.
6. Air- or vapor-binding in the case of pumps handling liquids with appreciable amounts of entrained or dissolved air or gas.

The effects must be examined separately insofar as the setting-up of minimum flows is concerned. Those minimum flows dictated by effects which can lead to very rapid failure or deterioration of performance must be assumed to be discrete numbers, and operation below these flows should never be permitted. On the other hand, operation below the minimum flow dictated by cavitation type damage to the impeller need not cause an immediate destruction of the impeller, but will shorten impeller life. Therefore operation below this flow can be permitted to occur intermittently, but not continuously. ■