Second in a 3-part series:

Centrifugal pump operation at off-design conditions

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Part 1 in this 3-part series of articles, published in the April 1987 issue of CHEMICAL PROCESSING Magazine, examined the effects of radial thrust and temperature rise on pumped fluids when pumps operate above and below the best efficiency point on the performance curve. Part 2 deals with internal recirculation in the impeller. Part 3, scheduled for the June issue, will discuss 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 will also prescribe the necessary by-pass arrangements when process demands fall below minimum flow rates recommended by the pump manufacturers. (Please note that consecutive figure numbers are used throughout the 3-part series of articles.)

Internal recirculation

The subject of internal recirculation, until very recently, had been understood by only a small number of pump designers. At certain flows, generally below that of best efficiency, all centrifugal pumps are subject to internal recirculation in the suction and discharge areas of the impeller. The result of these two phenomena is a great increase in pressure pulsations (Fig. 9). It should be noted that the capacities at which the recirculation at the suction and at the discharge arises are not necessarily coincidental. Internal recirculation at the suction is most frequently the cause of field problems.

Internal recirculation at the suction

One method used to reduce the required NPSH is to increase the eye diameter of the impeller, thereby reducing the entrance velocities (Fig. 10). This, in turn, increases the peripheral velocity of the impeller at the eye and, at some capacity, the distortion of the velocity triangles causes the flow at the outer eye diameter to reverse itself and flow back out of the impeller.

The exact flow at which suction recirculation takes place is dependent on the design of the impeller. The important fact to remember is that the larger the impeller eye diameter, and the larger the areas at the impeller suction relative to its overall geometry (and, therefore, the lower the required NPSH at a given capacity and speed), the higher will be the capacity at which recirculation takes place in percentage of the capacity at best efficiency (Fig. 11).

Internal recirculation causes the formation of very intense vortices (Fig. 12) with high velocities at their core and, consequently, a significant lowering of the static pressure at that location. This, in turn, leads to intense cavitation accompanied by severe pressure pulsations and noise and can be damaging to the operation of the pump and to the integrity of the impeller material. It should be noted, incidentally, that the location of the material damage is an excellent diagnostic tool in identifying whether the cause is classic cavitation or internal suction recirculation. If the damage is on the visible side of the inlet impeller vanes, the cause is classic cavitation. If the damage is to the hidden pressure side of the vanes and must be seen with the help of a small mirror, the cause is suction recirculation (Fig. 13).

Field problems caused by the above phenomenon obviously did occur from time to time, but it was in the early 1960's that they became intensified. Two factors led to a greater incidence of problems:

1. More pressure was exerted by the users to have the pump designers reduce required NPSH values. This could only be achieved by increasing the impeller suction eye areas, bringing the onset of internal recirculation closer and closer to the best efficiency capacity.
2. Higher heads per stage and larger pump capacities were increasing the energy levels of individual impellers, intensifying the unfavorable effects of internal recirculation.

Information on the explanation of the phenomenon was first released in 1972 in a limited circulation paper and later, more widely, in an article (Reference 1). For certain obvious reasons, the mathematical solution was considered proprietary and was not published until 1981. In the interim, however, guidelines were suggested indicating that Suction Specific Speed values ("S") not be allowed to exceed a range of 8500 to 9500 to avoid field problems if operation at significantly reduced flows was contemplated. The following will explain the thinking that led to establishing these guidelines:

At this time it is not practical to determine the amount of the static pressure converted into kinetic energy at the center of the vortex; in other words, it is not possible to establish a mathematical relation for the NPSH required at this vortex to suppress cavitation. We can, however, establish some qualitative descriptive curves which are useful to understand the phenomenon. If, for instance, we are dealing with an impeller with very low NPSH (high Suction Specific Speed "S"), the onset of recirculation occurs very close to the best efficiency point, as shown in Fig. 11, and the NPSH at the vortex exceeds the NPSH available. Damage is certain to occur in such a case. If, on the other hand, we select a more conventional impeller, with a moderate value of NPSH and, hence, a moderate "S" value, recirculation occurs

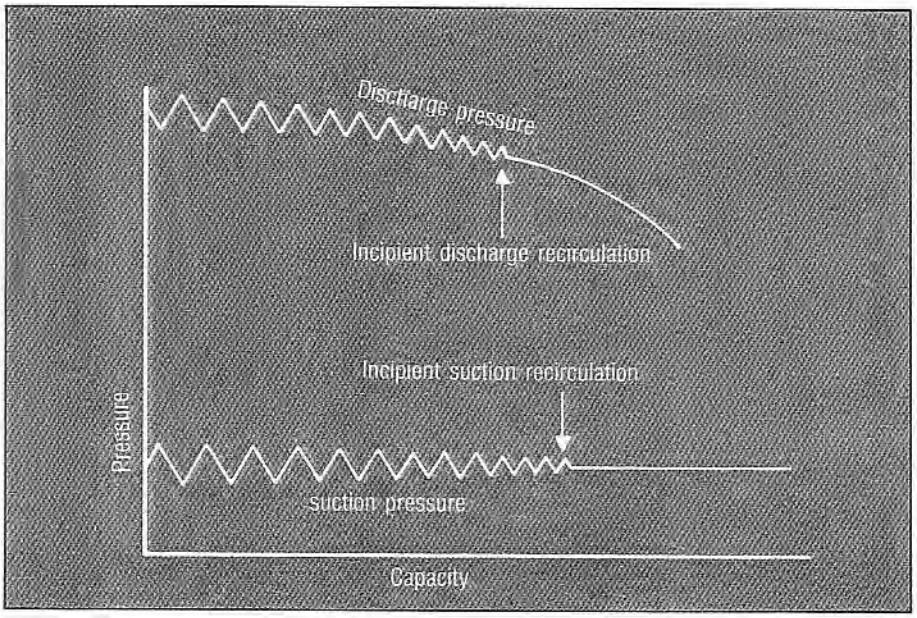

Figure 9: Characteristics of internal recirculation

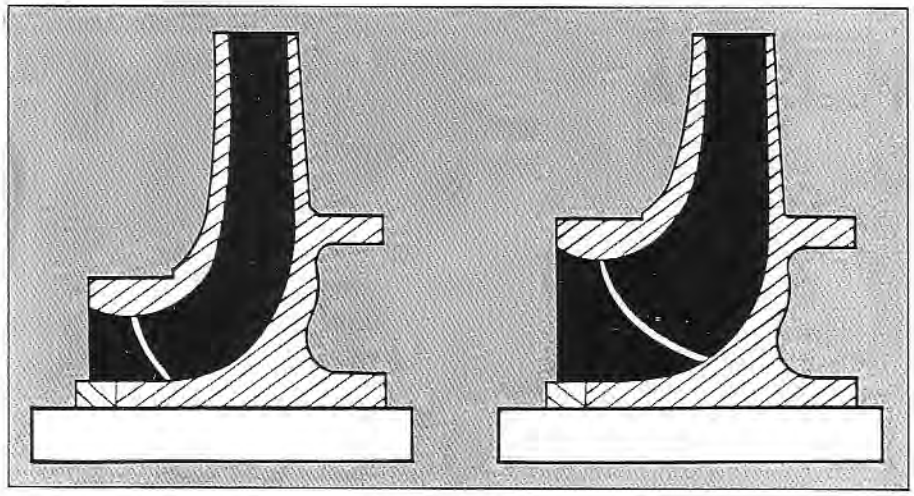

Figure 10: Enlarging impeller eye area to reduce NPSH

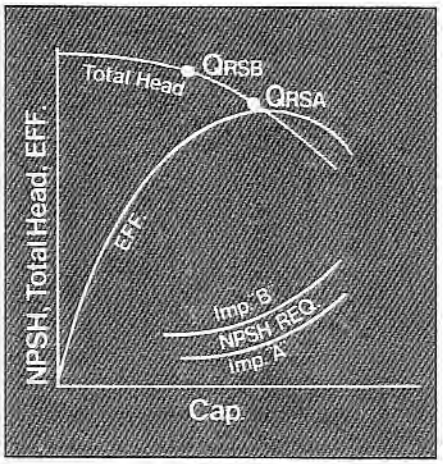

Figure 11: Effect of NPSH on recirculation flow

at a much lower percentage of b.e.p. If the NPSH required at the vortex is more modest and, possibly, does not exceed the available NPSH, frequently no damage occurs even if the pump operates in the recirculation zone. Another way of expressing this thought is illustrated in Fig. 14 which describes the safe operating zones for normal and high "S" value impellers.

W. H. Fraser's paper (Reference 2) gave exact formulas to calculate the flow at which internal recirculation would start at the suction once certain geometric data were known about an impeller. Another paper (Reference 3) provided close approximation curves in the event that these data are not readily available. If data are available to permit the more exact calculations given in Reference 2, these formulas should be used instead of the curves.

In all of this discussion it must be noted that "S," the Suction Specific Speed of the pump, must always be calculated for the conditions corresponding to the capacity at best efficiency for that pump. The guaranteed conditions of service may not correspond to this best efficiency flow; in fact, they rarely do. It must also be calculated on the basis of the pump performance with the maximum impeller diameter for which it was designed. This constraint becomes obvious when one considers that the internal recirculation at the suction occurs because of conditions that arise in and around the impeller inlet, conditions which are not necessarily affected by cutting down the impeller diameter. This cutting down will move the best efficiency point to a lower flow value, but

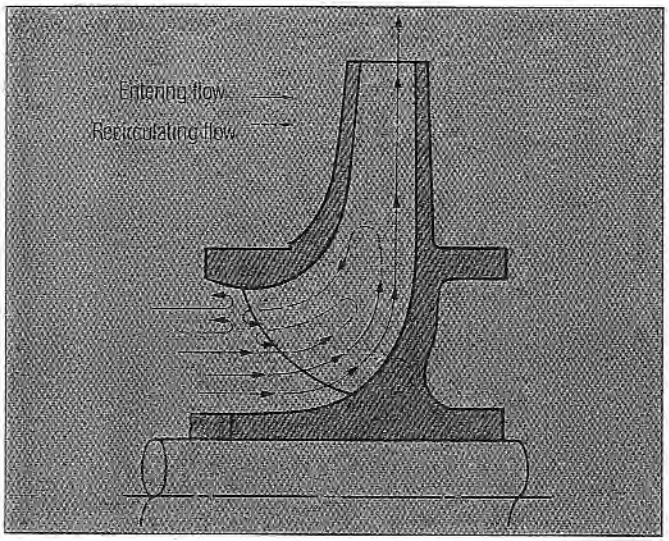

Figure 12: Section through a single suction impeller indicating schematically the recirculation of liquid at the inlet during operation at low capacities

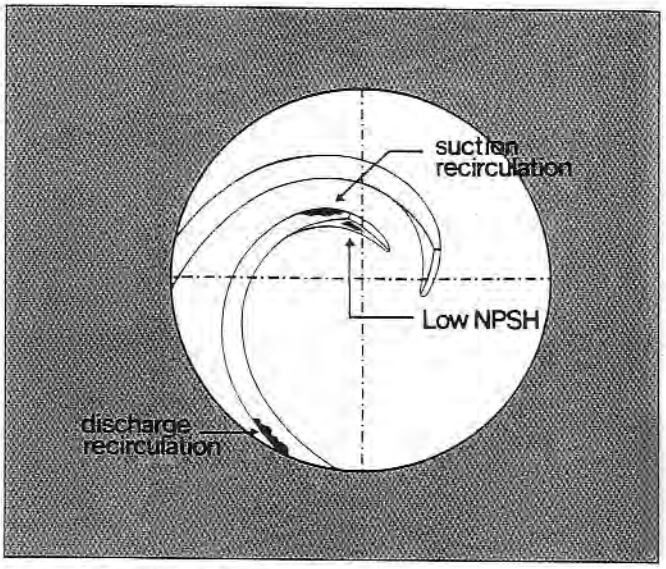

Figure 13: Diagnosis of cavitation damage

will not reduce the flow at which suction recirculation will occur.

While liquid characteristics cannot and do not affect the flow at which internal recirculation takes place, both the symptoms and the accompanying damage are affected by these characteristics. In the case of cavitation caused by recirculation, liquid properties which mitigate cavitation damage caused by "classical" cavitation (that is cavitation resulting from insufficient NPSH available) are equally effective. In general, there are a number of factors which operate to do so when handling liquids such as hydrocarbons. The most important of these factors is the relatively low ratio of the specific volumes of the vapor and of the liquid phases when compared to this ratio for cold water. Consequently, minimum flows for pumps handling hydrocarbons need not be selected as conservatively as for cold water pumps.

Field experience has amply demonstrated the validity of the above explanation. I know of no major case of impeller damage sustained by pumps handling hydrocarbons caused by operation within the recirculation zone, while very serious damage has occurred with exactly the same pumps operated in the same manner on cold water.

It would seem that now that means are available to calculate the onset of internal recirculation, users and designers of centrifugal pumps should be in the position to establish sensible limits for minimum operating flows. To do so, it would be necessary to establish some sort of guidelines between minimum flows and recirculation flows. But there's the rub! Setting up such guidelines is a fairly complex task for a number of reasons.

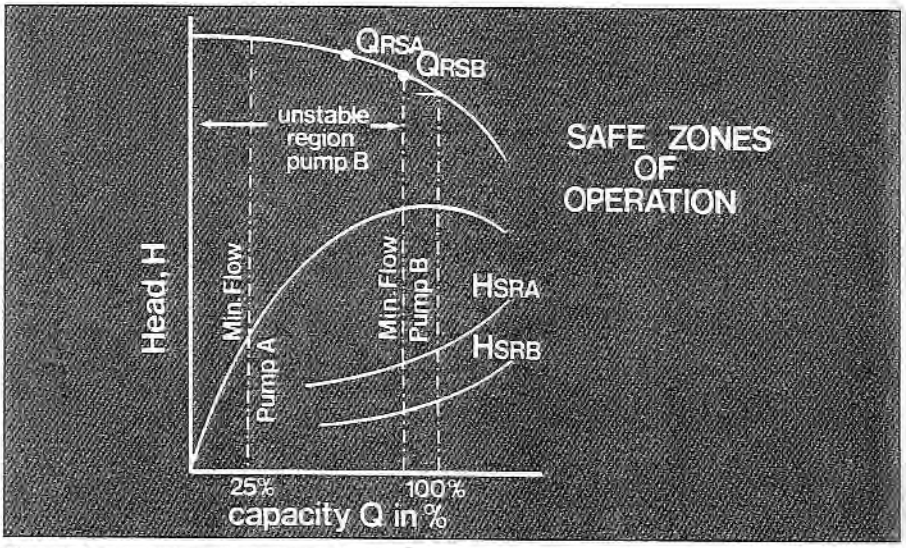

Figure 14: Comparison of safe zones of operation for normal and for high "S" value impellers

Operation of a pump at flows below the recirculation flow leads to a variety of events, all of which contribute to unfavorable effects on the pump performance and on the ultimate life of the impeller. The events can be lumped together under the term "distress." In turn, the degree of "distress" will depend on a variety of factors, such as:

- size of the pump, i.e. capacity, total head and horsepower;
- the value itself of the Suction Specific Speed;
- fluid characteristics;
- materials of construction;
- length of time the pump operates below certain critical flows;
- most important, the degree of tolerance of the pump user to the signs of distress exhibited by his equipment.

The last factor makes the choice of minimum flow guidelines a very subjective one. Some users will accept unquestionably the fact that an impeller may have to be replaced every year, while others will complain if an impeller on exactly the same service lasts only three or four years. Similarly, noise and pulsation levels which are perfectly acceptable to one user are cause for bitter recrimina-

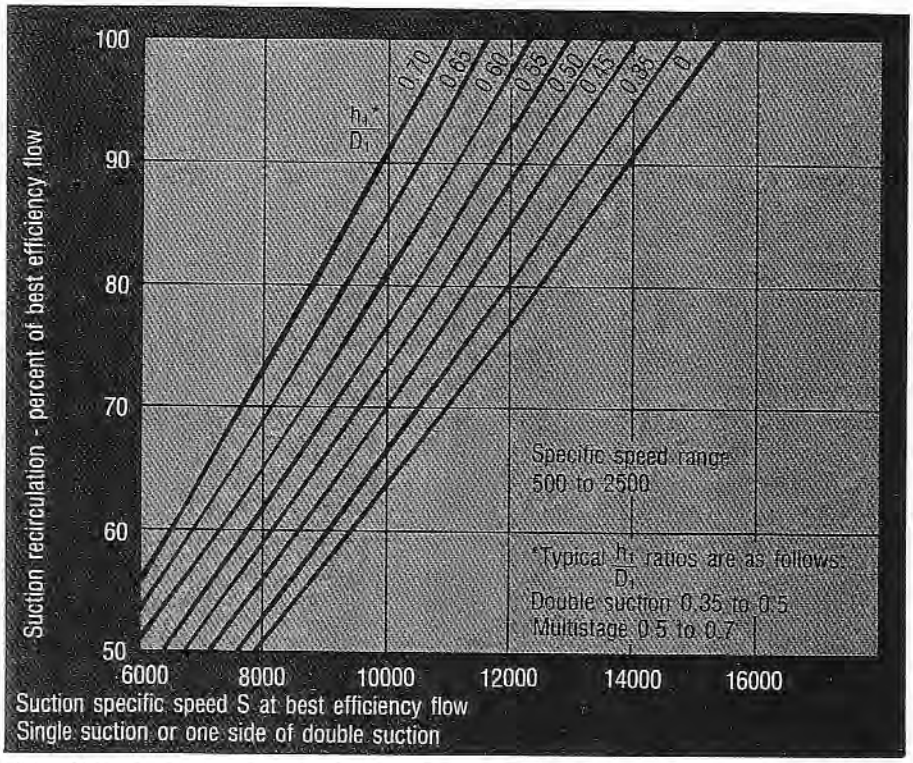

Figure 15: Suction recirculation as a function of suction specific speed and hub diameter (h_1) to eye diameter (D_1) ratio

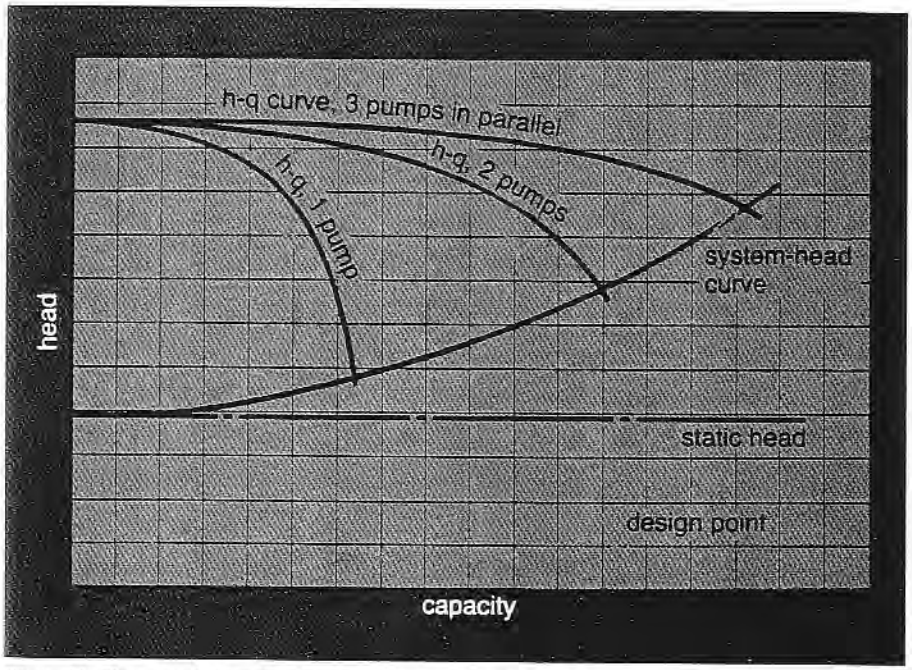

Figure 16: Three condenser circulating pumps operating in parallel

tions on the part of other users.

It can easily be demonstrated that there is a definite relationship between acceptable minimum flows in percentage of recirculation flow and the Suction Specific Speed itself, by both intuitive reasoning and actual field experience. It should be noted that for the same speed and capacity, a higher suction speed design requires a larger impeller eye diameter than does a lower suction specific speed design. This means that when the backflow starts at the eye diameter of the impeller, the peripheral velocity of this backflow will be higher with the higher suction specific speed design. It follows that the higher backflow velocity represents a higher energy level, and that all the symptoms of operation in the recirculation zone will be intensified with the higher suction specific speed design.

There is one observation, however, which may give some guidance to users, and that is with respect to the effect of the "S" value on what constitutes an acceptable minimum flow. Referring to Fig. 15, and assuming a hub to eye diameter (h/D_1) ratio of 0.45, an impeller with an "S" value of 14,000 will have its suction recirculation occur at about 100% of its best efficiency flow. If we are dealing with a fairly substantial pump handling cold water, we should be most reluctant to consider running such a pump below this 100%. On the other hand, a pump with the same hub-to-eye-diameter ratio and an "S" value of 8000 will have a recirculation flow of only 56% of best efficiency flow, and we need have no hesitation running it when necessary at as little as 25% of best efficiency flow.

But one can set some guidelines, vague as they may appear to be:

1. Unless there is a compelling reason to do so, do not specify NPSH values which result in "S" values much above 9000 for water or above 11,000 for hydrocarbons.
2. When dealing with relatively small pumps, say under 100 hp, the effect of suction recirculation is not apt to be as significant as for larger pumps.
3. Pumps handling hydrocarbons can be operated at lower flows than equivalent pumps handling cold water.
4. The risks of operating at flows much below the recirculation flow can best be determined after the pump is in operation. Provision should therefore be made to increase the minimum flow by-pass if there is a suspicion that too optimistic a decision as to this minimum flow has been made at the time the pump was selected.
5. When the pump is never expected to operate at flows below its design condition, higher "S" values can be used without concern for unfavorable effects from internal suction recirculation.

This will be the case, for instance, with constant speed condenser circulating pumps operating in parallel since it is not the practice to throttle their discharge. This situation can best be understood by reference to a set of curves

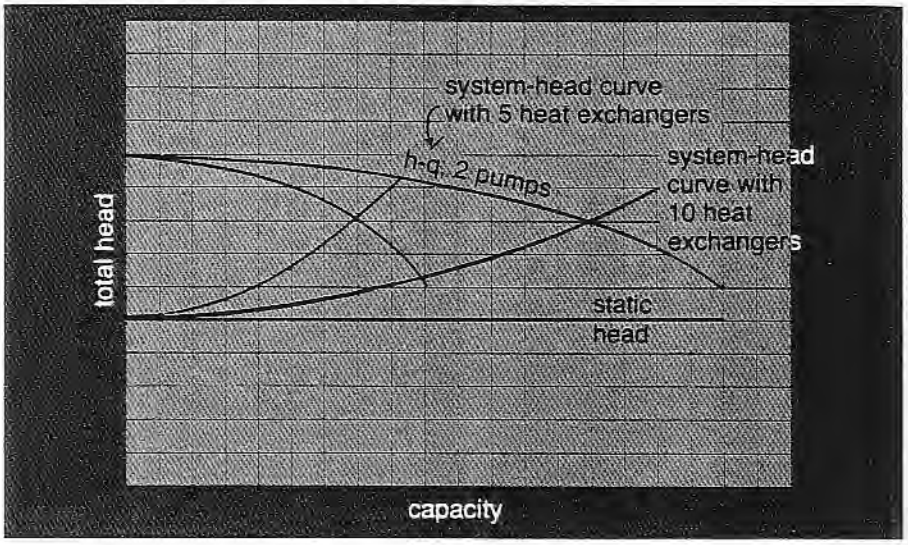

Figure 17: Pumps operating in parallel with ten and with five heat exchangers

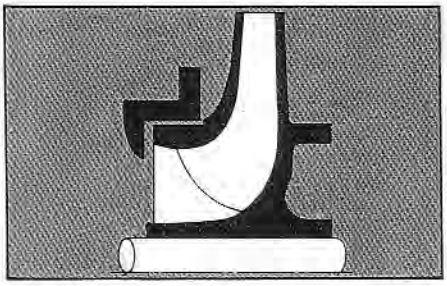

Figure 18: "Bulk-head ring" construction eliminates unfavorable effects of excessively large impeller eye diameter

which describes the operation of three condenser circulating pumps operating in parallel superimposed on the system-head curve (Fig. 16). You will note that whenever only one or two pumps are operating, the head-capacity curves intersect the system-head curve at flows in excess of the design capacity. As a matter of fact, this situation is even further accentuated whenever the system-head curve has been drawn up pessimistically high. In a case such as this, one can certainly use "S" values considerably higher than when a pump may be called upon to operate over a wide range of capacities to the left of its design condition.

The only precaution one must take is not to apply the approach if the pumps discharge through several heat exchangers in parallel, some of which may, on occasion, be by-passed or taken out of service. This is described in a curve showing two cooling tower pumps discharging through 10 heat exchangers (Fig. 17). Whenever, let us say, five of these heat exchangers are by-passed, the system-head curve steepens, and the operating capacity is reduced to much below the design conditions.

I should mention here a "post-factum" modification which, in a number of cases, has been used quite successfully to reduce and even sometimes to eliminate the unfavorable effects of suction recirculation. It consists of retrofitting pumps with stationary casing rings the apron of which extends inwardly of the impeller eye diameter (Fig. 18). If preferred, such rings can instead be rotating and mounted on the impeller. Such rings are commonly referred to as "bulk-head rings." This prevents the recirculation vortex from extending axially beyond the plane formed by the apron. Of course, since this does increase the required NPSH, the use of these bulk-head rings can only be resorted to if there is sufficient margin in the available NPSH.

References:

1) Bush, Fraser and Karassik, "Coping with Pump Progress: The Sources and Solutions of Centrifugal Pump Pulsations, Surges and Vibrations," *Pump World*, Summer 1975 and March 1976.

2) Fraser, W. H., "Recirculation in Centrifugal Pumps," Winter Annual Meeting of ASME, Nov. 16, 1981.

3) Fraser, W. H., "Flow Recirculation in Centrifugal Pumps," Texas A. & M. Turbomachinery Symposium, Houston, Texas, Dec. 1981.