# FLOW RECIRCULATION IN CENTRIFUGAL PUMPS.

by Warren H. Fraser

*Recirculation, a potentially damaging flow reversal at the inlet or discharge tips of the impeller vanes of a centrifugal pump, has long been a source of consternation to centrifugal pump designers and users. Until recently, the pump industry had little recourse other than to cope with the effects of recirculation after they had already begun wreaking havoc on a pumping system.*

*Now, after years of analysis and testing, Worthington has presented its findings on the mathematical relationship which makes it possible to determine the flow patterns which exist when recirculation occurs. This research makes it possible to design a pump for specified flow ranges to avoid or minimize the impact of recirculation.*

*Analysis and testing has also led to the determination that the symptoms associated with recirculation are very specific. This article, extracted from a paper presented at a 1981 ASME meeting, and another presented at Texas A&M in 1981, discusses some of these symptoms, diagnoses their causes, and, based on Worthington test and field experience, explains how many of these problems can be avoided.*

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The pressure field produced in a centrifugal pump impeller at a flow corresponding to peak efficiency is more uniform and more symmetrical than at any other flow. At flows less than that at peak efficiency, the pressure field becomes increasingly distorted until at some point the pressure gradient reverses and a localized reversal of the flow takes place. This is the point of recirculation, which can occur at the discharge or the suction of the impeller, or at both the suction and the discharge.

Recirculation characteristics are dependent on the design of the impeller. It is inherent in the dynamics of the pressure field that every impeller design must recirculate at some point–it cannot be avoided. It is important for both the designer and the operator of centrifugal pumps to realize that the capacity at which discharge recirculation occurs can be reduced through design procedures, but only at a reduction in the rated efficiency of the pump.

Similarly, the capacity at which suction recirculation occurs can be reduced, but only with an accompanying increase in the rated net positive suction head (npsh) required. Optimization of efficiency requires a reduction in the margin of safety between the rated capacity and the discharge recirculation capacity. Also, the higher the design suction specific speed, defined as:

$$S_s = \frac{\text{rpm}\sqrt{\text{gpm}}}{(\text{npsh/ft.})^{3/4}},$$

the narrower is the margin of safety between the rated capacity and the suction recirculation capacity.

### Causes of recirculation.

Why does a reversal of flow occur at reduced flows? The answer to this question seems to be related to the fact that the pressure field not only increases from suction to discharge, but also that the total head produced is the sum of the centrifugal head and the dynamic head.

The centrifugal head for any given impeller diameter and speed is independent of the rate of flow. The dynamic head, however, is a function of the absolute velocity that is related to the rate of flow. At some point on the head capacity curve, the dynamic head will exceed the centrifugal head. At this point, the pressure gradient reverses, and the flow is from the discharge to the suction of the impeller. Because the pressure field is now not symmetrical, and because the vanes themselves distort the pressure field, the reverse or back flow takes place in the vicinity of the vane itself. The condition now exists where a small portion of the total flow has reversed its direction, and the shear face between the two flows produces vortices that are "locked" into the vane system and rotate with it.

This is the point of recirculation, and this is the capacity at which noise, vibration, and cavitation damage to the pressure surface of the vanes are most likely to occur. The severest damage at this capacity most often occurs locally.

### Effects of recirculation.

It is now well established from field observation and from tests in the laboratory that surging and cavitation at the inlet vanes of the impeller can be caused by suction recirculation, despite the fact that a wide margin exists between the npsh required and the npsh available.

Similarly, extensive damage to the pressure side of the impeller vane at the discharge has been observed in many pumps operating at reduced flow rates. These effects are the more obvious results of recirculation. In addition, there are less obvious and more subtle symptoms and operational difficulties that are associated with operation of the pump in recirculation zones. The most common symptoms and a diagnosis of the causes of each are shown in Table I on page 10.

### Suction recirculation.

The reversal of flow in the eye of the impeller at the point of suction recirculation has been observed in laboratory tests. Figure 1 shows a Worthington laboratory test arrangement of a 6-inch end-suction pump installed in a test loop with a transparent suction pipe that permits visual observation of flow patterns in the impeller eye. Figure 2 shows a more detailed view of the pump with streamers attached to the inside of the transparent pipe to show the flow patterns of the vortex produced by suction recirculation. As the suction recirculation progresses down the pipe, a high velocity annulus of fluid is produced at the wall, while at the same time, fluid is approaching and entering the eye of the impeller through the core of the annulus. The steep gradient between the flow through the core and the rotating annulus produces vortex streets that cavitate and produce random sharp crackling noise.

The suction vortex is the external effect of a flow reversal that is occurring at the inlet of the impeller between the vanes themselves. Between the shear face of the flow entering the impeller vanes near the hub, and that ejected at the impeller eye diameter, a fixed vortex is produced that travels around with the rotation of the vane system. This vortex will cavitate at its core and attack the metal surface of the pressure side of

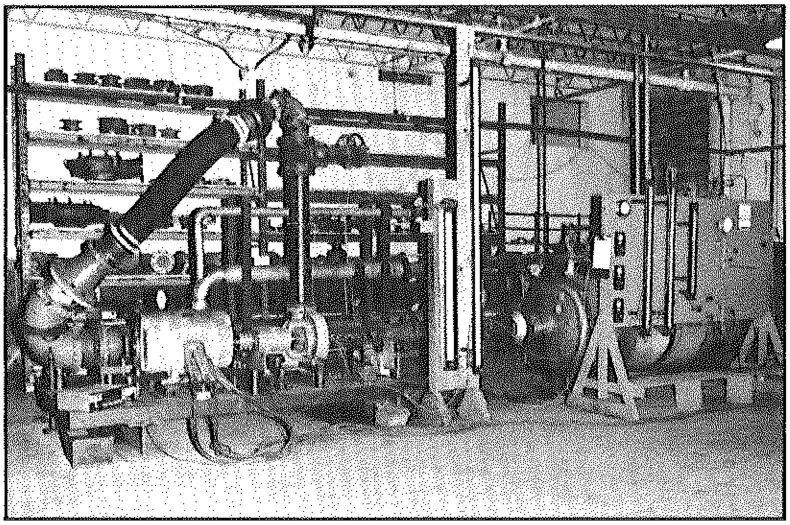


**Figure 1–Laboratory test pump.**

the vane in the area approximately midway between the hub and the shroud. Figure 3 shows schematically the flow of the impeller eye during recirculation.

**Discharge recirculation.**

The reversal of flow at the discharge of the impeller is more difficult to examine directly than is the suction recirculation. One technique is to record a trace of the pressure pulsations in the discharge casing as the output flow of the pump is reduced. At some point, that magnitude of the peak-to-peak pressure pulsations will increase at a very steep rate. This is the point of discharge recirculation.

The mechanics of the attack from discharge recirculation are very similar to that in the suction. At some point on the head capacity curve, the flow reverses on the pressure side of the vane and produces a vortex that rotates with the vane system.

Figure 4 shows schematically the flow at the impeller discharge during recirculation. If the velocities of the reverse flow are of sufficient magnitude, the vortex will cavitate and attack the metal surface of the vane. Figure 5 shows the damage to the pressure side of the discharge vane from operation in the discharge recirculation zone.

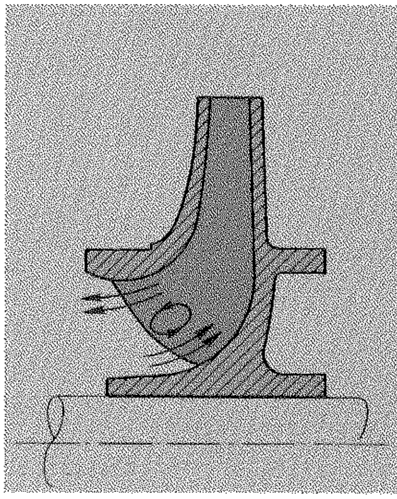


**Figure 3–Suction recirculation at the impeller eye.**

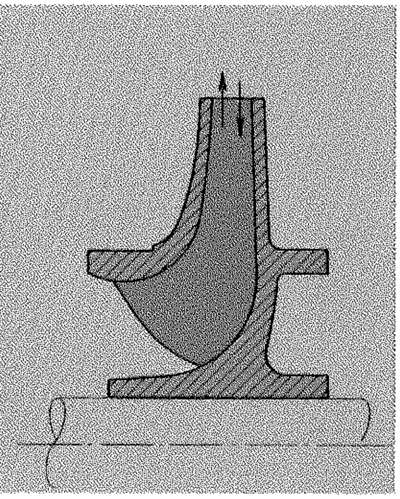


**Figure 4–Discharge recirculation at the impeller discharge.**

**Predicting recirculation.**

Figures 6 and 7 show the approximate values of suction recirculation as a percentage of the best efficiency flow for various suction specific speed designs. The suction recirculation values also increase as the specific speed ($N_s$) increases.

$$N_s = \frac{\text{rpm}\sqrt{\text{gpm}}}{(\text{th/ft.})^{3/4}}$$

This effect is shown in the form of two prediction charts.

Figure 6 should be used for specific speeds in the 500 to 2,500$N_s$ range, and Figure 7 for specific speeds from 2,500 to 10,000$N_s$. While these predictions are based on the analysis of hundreds of pumps in actual operation, as well as pumps tested in the laboratory, it must be recognized that these are guidelines for a wide range of pumps and services. While some designs may be better or worse than the charts indicate, the predictions do represent the average or the norm for good commercial designs.

A more precise prediction of recirculation values can be determined if the design parameters of the impeller are available. The appendix provided on pages 15-17

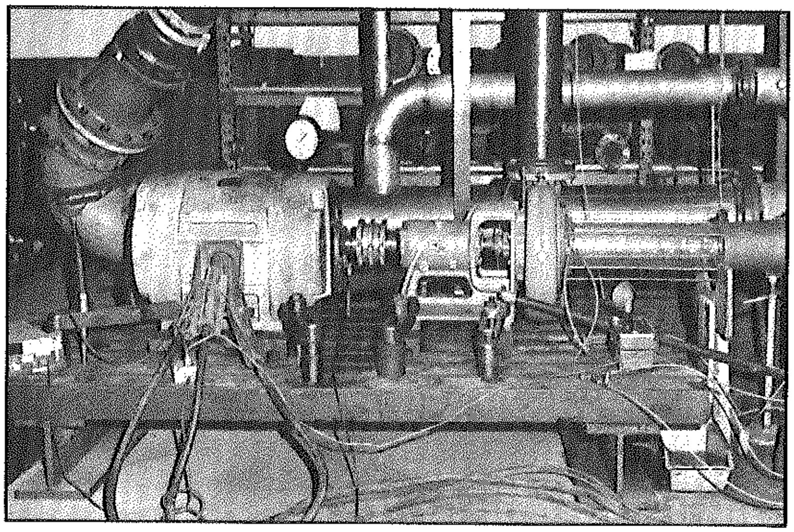


**Figure 2–Laboratory test pump showing transparent suction pipe.**

presents a more detailed analysis of this procedure.

While Figures 6 and 7 can be used to predict the suction recirculation values, the recommended minimum flow in actual operation will depend upon the size of the pump, as well as the fluid pumped. For example, in pumps of 2,500 gpm or less and heads per stage of up to 150 feet (approximately 100 hp), the energy levels may not be sufficient to cause damage or operational problems even though the pumps are operated in the recirculation zone.

As a general rule for pumps of 2,500 gpm or less and heads per stage up to 150 feet, the minimum flow values can be set at 50 percent of the recirculation values for continuous operation and 25 percent for intermittent operation. For higher flows and heads, the minimum flow should correspond to the full suction recirculation flow.

On the other hand, for pumps in that larger size range but handling hydrocarbons, the minimum flow value can be set at 60 percent of the recirculation flow for continuous operation and 25 percent for intermittent operation.

**Problems result from misuse of guidelines.**

There are a number of cases where appreciable deviation from the guidelines has resulted in very serious operational problems.

For example, a double-suction pump on cooling tower service was selected with a specific speed of 2,750$N_s$ and a suction specific speed of 10,000$S_s$. Most of the operation

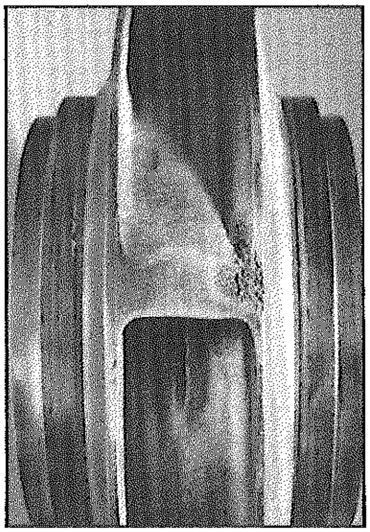


**Figure 5–Damage to the pressure side of the discharge vane from recirculation.**

of the pump occurred at 75 percent of the best efficiency flow, and after only a few weeks of operation in suction recirculation the inlet vanes of the cast iron impeller were cavitated to the point where they broke away in pieces. A steel impeller was substituted for the cast iron, and the life was extended to approximately six months before repair welding of the impeller was required.

In another case, a high-pressure boiler feed pump of 1,500$N_s$ specific speed was operated for extended periods at low loads during the start-up of a plant. During this period of operation in the discharge recirculation zone of the pump, the diffuser vanes were subjected to severe cyclic loading, and several of the vanes failed in fatigue and broke away. Once the plant was operating at or near full load, however, the pump operated above the discharge recirculation capacity, and no further damage was experienced.

**Mechanical damage.**

Experience has shown that many centrifugal pumps operate either continuously or intermittently in suction or discharge recirculation, or both. Under these conditions, mechanical damage may or may not develop, depending upon the size, horsepower rating, the head developed, the charac-

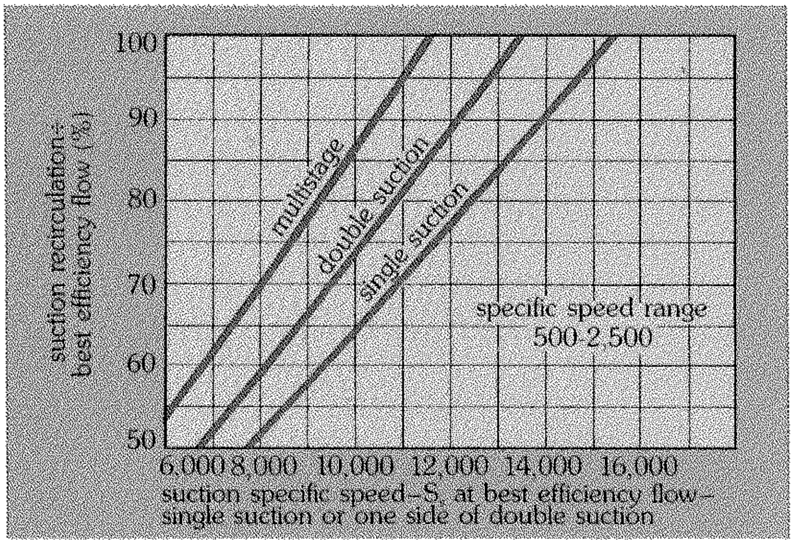
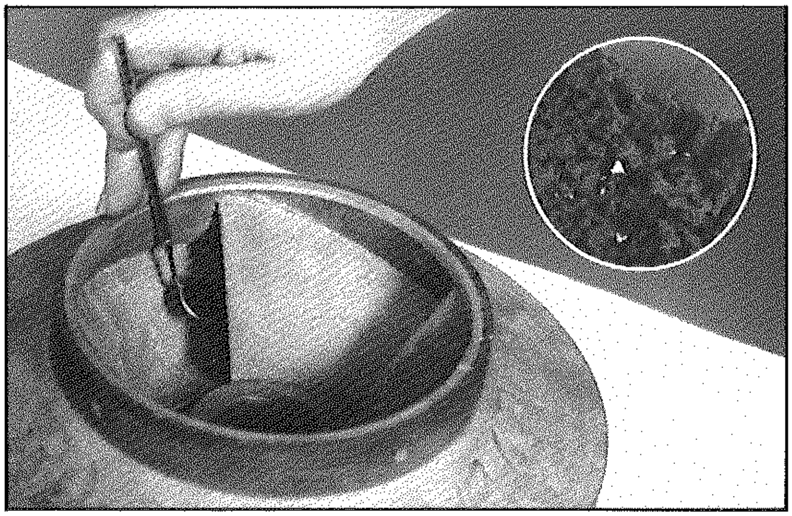



**Figure 6–Suction recirculation–500 to 2,500N .**

teristics of the fluid pumped, and the materials of construction.

For example, a 2,500-gpm pump operating at 100 feet of total head may not exhibit any damage, but a 25,000-gpm pump of the same design and specific speed will produce high noise and vibration levels in the recirculation zone with progressive cavitation damage to the impeller and to the casing.

Similarly, while the 25,000-gpm pump handling water will have the symptoms associated with recirculation, the same equipment pumping a hydrocarbon will exhibit lower levels of noise, vibration, and cavitation damage.



**Inspection of pressure side of inlet vane for recirculation damage. Inset: Damage to the pressure side of the inlet vane from suction recirculation.**

**Evaluate specific speeds; know your flow rates.**

The potential effect of suction specific speed on the point of suction recirculation, and the effect of discharge recirculation on efficiency, should be evaluated for pumping installations of 2,500 gpm or larger. As a general rule, suction specific speeds in excess of 9,000$S_s$ for pumps designed for specific speeds of 2,550$N_s$ or higher should be evaluated very carefully to avoid recirculation within the operating range of the system.

The analysis of suction recirculation has revealed that the higher the design suction specific speed, the closer will be the point of suction recirculation to the rated capacity. Similarly, the closer the discharge recirculation capacity is to the rated capacity, the higher will be the efficiency.

There is a great temptation to design for the highest possible efficiencies and suction specific speeds, but this may result in designs that are very limited in their range of operation. Both the designer and user of centrifugal pumps must know the flow rates at which suction and discharge recirculation occur. These values are as much a part of the specified performance of any given design as are the head, capacity, efficiency, or npsh.

With complete performance characteristics in hand, a more realistic evaluation can be made as to the risks associated with operation at or near recirculation as against the anticipated savings in power costs and lower npsh requirements.

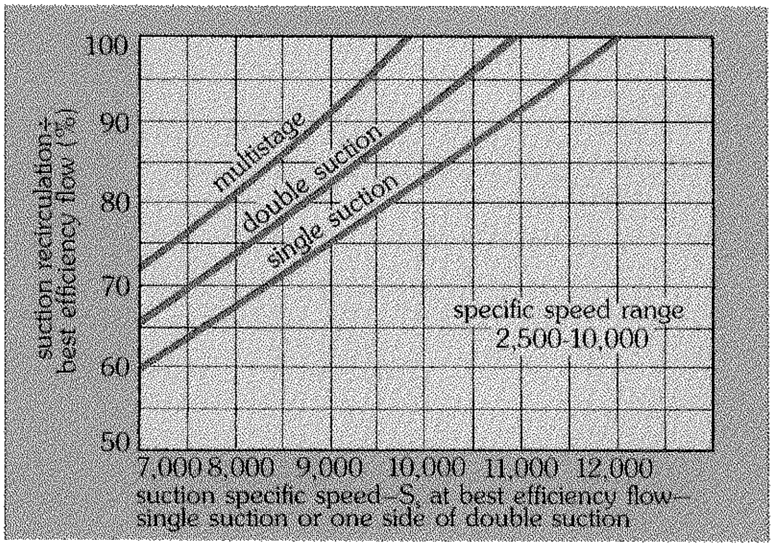



**Figure 7–Suction recirculation–2,500 to 10,000$N_s$.**

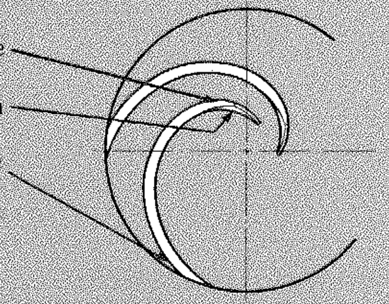



**TABLE I**
**Common Recirculation Symptoms, Diagnoses, and Causes**

| Symptom | Diagnosis | Cause |
|---|---|---|
| **1. Cavitation damage on the pressure surface of the inlet of the impeller vane (see Table I inset).** | Pump is operating in suction recirculation with the flow entering the impeller through the central core of the vortex produced at the eye diameter by the flow reversal. | Cavitation damage is caused by the cavitating core of a local vortex locked into the space between the vanes. |
| **2. Periodic crackling noise in and around the suction of the pump.** | Same as #1. | Periodic crackling noise comes from the formation and decay of string-type vortices produced by the shear surfaces of the incoming flow at the center of the reverse flowing annulus in the pump suction. |
| **3. Cavitation damage to the stop pieces or cross vanes in the pump suction.** | Same as #1. | Once suction recirculation starts, any reduction in the pump output will force the swirl pattern further upstream from the impeller eye. Any obstruction in the form of vanes or stop pieces will produce severe cavitation where the swirl pattern impinges on the vane. |
| **4. Surging in the suction.** | Same as #1, except that two-phase flow is involved with a mixture of a gas or vapor and the liquid. | Instability is produced by the dynamics of the compressibility of the gas or vapor bubble produced in the pump suction by the turbulence of recirculation. The hazards are particularly high on systems designed for npsh values of 10 feet or less. |
| **5. Cavitation damage on the pressure surface of the discharge portion of the impeller vane (see Table I inset).** | Pump is operating in discharge recirculation with a portion of the flow jetting back into the impeller channels. | Cavitation is caused by a local vortex locked into the space between the vanes. |
| **6. Mechanical failure of portions of the impeller shroud with pieces broken out between the vanes.** | Same as #5. | Pressure pulsations produced by the cavitating vortices on the vane surface between the shrouds are of sufficient magnitude and frequency that the metal shrouds fail in fatigue. |
| **7. Cavitation damage to the tongue of the volute or to the diffuser vanes in a multivane diffuser design.** | Same as #5. | As the flow reverses during the discharge recirculation, the tongue or diffuser vanes become an obstruction to the flow pattern. Cavitation will occur on the underside or the surface facing the impeller as this becomes the side of separation and reduced pressure. |
| **8. Axial oscillation of the shaft with possible damage to the thrust bearing and mechanical seals.** | Same as #5. | As the flow reverses during discharge recirculation, a portion of the flow is directed down along the outside surface of the shrouds of the impeller. This flow has a high rotational component and increases the velocity of the vortex between the impeller and the walls of the casing. The reverse flow is not stable, and the pressure on the two shrouds of the impeller will not be equal. The result will be axial unbalance. |
| **9. Shaft failures of multistage and double-suction pumps.** | Same as #5. | Thrust reversals as high as 10,000 cycles per second are imposed on the portion of the shaft between the impeller or impellers and the outboard thrust bearing causing fatigue failures of the shaft in tension. |

# FLOW RECIRCULATION IN CENTRIFUGAL PUMPS: FROM THEORY TO PRACTICE.

**by Igor J. Karassik**

***The lack of a mathematical model has for years left centrifugal pump users and designers to their own devices when attempting to specify acceptable values of minimum flows for various "S," or suction specific speed, values. This, fortunately, is no longer the situation. There does now exist a mathematical model which will permit users and designers to establish sensible limits for centrifugal pump minimum operating flows, as already discussed in this issue of Power & Fluids, in "Flow recirculation in centrifugal pumps."***

***But before this theory can be put into practice effectively, it is essential that users and designers first gain a thorough understanding of the critical relationship between minimum flows and recirculation flows. This article discusses the nature of this relationship, its effect on pump operation, and a practical approach to applying the mathematical method of predicting recirculation.***

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It should be understood that when utilizing the data presented in "Flow recirculation in centrifugal pumps" (page 5), S must always be calculated for the conditions corresponding to the capacity at best efficiency. Values calculated for any other flow do not have any relevancy and do not describe the characteristics of that pump. The guaranteed conditions of service may or may not correspond to this best efficiency flow; in fact, they seldom do.

What may be less obvious is that the S value for any pump must be calculated on the basis of pump performance with the maximum impeller diameter for which it was designed. This stricture becomes apparent when we consider that the internal recirculation at the suction occurs because of certain conditions that arise in and around the inlet of the impeller, conditions that are not necessarily affected by decreasing the impeller diameter.

Decreasing impeller diameter moves the best efficiency point to a lower flow value, but does not reduce the capacity at which suction recirculation will occur. As a matter of fact, in certain cases where the onset of suction recirculation may be triggered by the discharge side recirculation, decreasing the impeller diameter may actually increase the actual flow at which suction recirculation takes place.

### Pump and pumping system distress.

Operation of a pump at flows below the recirculation flow creates an array of events which leads to unfavorable effects, or distress, on the performance and integrity of the pump, and sometimes on the system in which it is operating. The degree of distress depends on a variety of factors such as: size of the pump (capacity, total head, and horsepower); liquid characteristics; materials of construction; length of time the pump operates below certain critical flows; and the degree of tolerance exhibited by the pump user to the signs of distress exhibited by his equipment.

### Size of the pump.

It was stated in "Flow recirculation in centrifugal pumps" that, "For pumps of 2,500 gpm or less and heads per stage up to 150 feet, the minimum flow values can be set at 50 percent of the recirculation flows for continuous operation, and 25 percent for intermittent operation." This indicates that pump size and energy level directly affect the degree of damage that can be experienced when a pump is operated within the zone of internal recirculation.

A 25-, 50-, or 100-hp pump will not severely damage itself when operating in full recirculation. A 500-hp pump may suffer considerable damage to its impeller, and a 5,000- or 10,000-hp pump has enough energy under certain conditions to shake its piping loose. A sense of proportion is important when sizing a pump.

### Liquid characteristics.

It is clear that liquid characteristics do not affect the flow at which recirculation takes place. It is only the symptoms and the accompanying damage which are affected by these characteristics. In the case of cavitation caused by recirculation, liquid properties which mitigate cavitation damage in "classical" cavitation are equally effective in the case of recirculation cavitation. In the case of flow force effects, a lower fluid density may reduce the force level of pressure pulsations. Consequently, minimum flows for pumps handling hydrocarbons need not be selected as conservatively as for water pumps.

**Materials of construction.**

Materials of construction are important factors to consider when establishing minimum flow limitations. The erosion effects of internal recirculation cavitation on pump parts are increased or decreased depending upon the relative degree of resistance to this kind of damage of the materials of construction. Type 316 stainless is a superior material which is better than bronze, which in turn is better than cast iron. However, the use of better materials will not help very much if the effects of internal recirculation manifest themselves not only in material degradation, but also in the form of pulsations and vibrations.

**The time element.**

There is a certain degree of ambiguity when discussing the time element contribution to pump distress. We differentiate between "continuous" and "intermittent" operation, but these two terms have never really been defined. It would be impractical to specify exactly the number of hours that demarcate the boundaries between these two conditions, but it is possible to be more specific.

There are two separate periods in the operating life of a pump that should be reviewed in the application of any centrifugal pump. The first is the period where the system is in full operation, and the range of capacities and heads encountered can be predicted with reasonable accuracy. We can define this continuous operation as that range of capacities which will be encountered at least 75 percent of the time. Intermittent operation during this period would cover the remaining range of operating conditions 25 percent or less of the time if they differ substantially.

The second period refers to the start-up conditions of the system. It is difficult to predict what the duration of pump operation at any capacity may be in this period. Unfortunately, this segment of pump life is seldom given any consideration in the design stage of the pumping system, while the pump may operate over very wide swings in capacities for extended periods of time. The result is that frequently there is significant damage to the pump caused by recirculation, even before the system goes into full operation.

**Tolerance of user.**

Another important factor to be considered is the degree of pump user tolerance to the signs of distress exhibited by his equipment. This is what makes the choice of minimum flow guidelines such a subjective one. For instance, while some users will accept the fact that an impeller may have to be replaced every year, others will be dissatisfied if an impeller on exactly the same service lasts only three or four years. Noise and pulsation levels which are perfectly acceptable to one user are totally unacceptable to another.

What confuses the issue even further is that recommendations of minimum flow to the user by pump manufacturers frequently seem to bear no relation to the suction specific speed of a particular pump offering. Indicative of this is the manufacturer with a pump offering with a higher S value who quotes a lower minimum flow than another manufacturer with a more conservative design; hence, a lower recirculation flow. These differences merely reflect more or less conservative estimates on the part of a designer as to what constitutes tolerable distress.

We can expand on this. Starting with the proposition that the required npsh, and, therefore, the suction specific speed of a pump, is determined by the conditions of service and by the geometric configuration of the impeller, it follows that two pumps with the same S value will have the same suction recirculation flow. It also follows that if an equal amount of distress is to be considered acceptable, the minimum flow to be recommended should be the same for equal values of S. Therefore, the higher the S value, the higher should be the value of acceptable minimum flow.

**Probability of low-flow operation.**

There are applications where a pump will never be expected to operate at flows below its design flow. This is the case, for instance, with constant-speed condenser circulating pumps operating in parallel, since it is not normal practice to throttle their discharge.

The situation can best be understood by reference to Figure 1, which describes the operation of three cooling tower pumps operating in parallel,

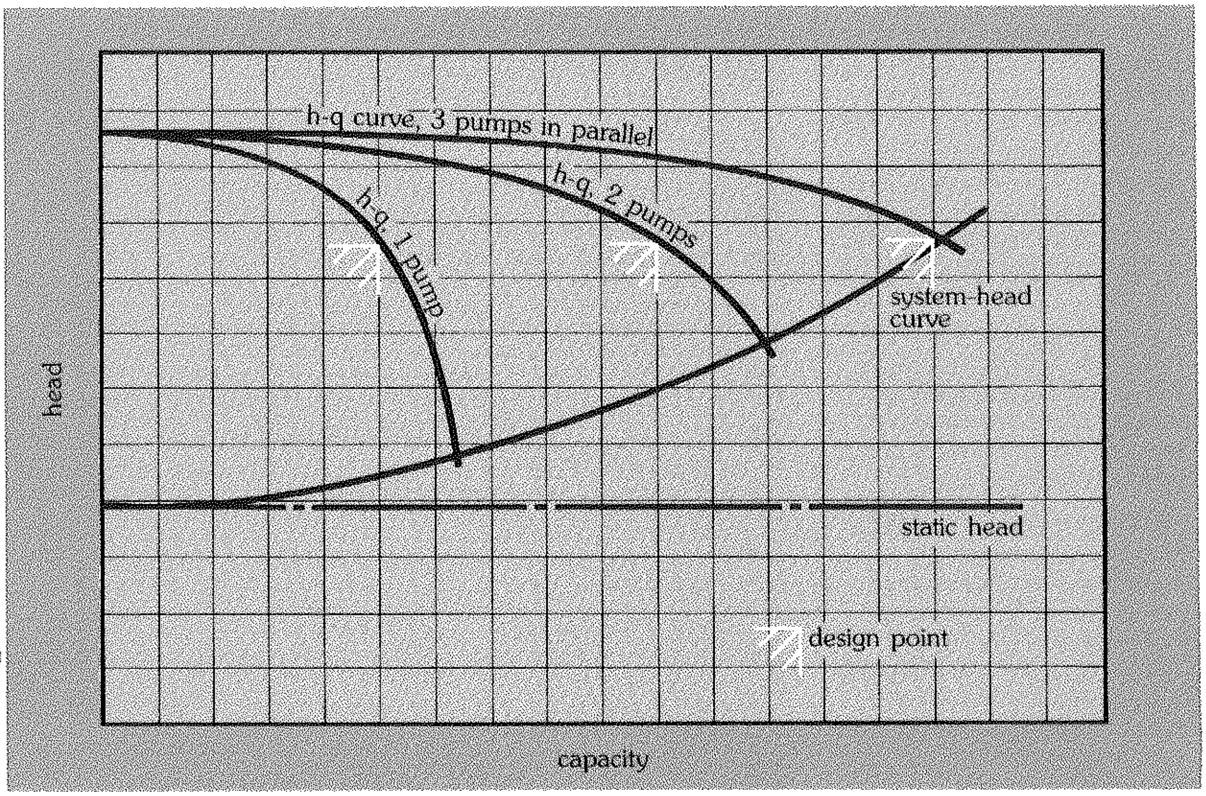



**Figure 1—Operation of three cooling tower pumps in parallel.**

superimposed on the system-head curve. Note that where 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 where the system-head curve has been drawn up pessimistically high. In a case such as this, there would be few objections to using S values considerably higher than in a case where a pump may be called upon to operate over a wide range of capacities to the left of its design condition.

However, this approach should not be applied if the pumps discharge through several heat exchangers in parallel, some of which may, on occasion, be taken out of service. The system-head curve under these circumstances would steepen, and the operating capacity reduced significantly below the design conditions.

A typical case of a pump in such a cooling system is illustrated in Figure 2. Here, the pump may discharge through either 10 or 5 heat exchangers. In the latter case, the intersection of the system-head curve with the head-capacity curve occurs at a flow sufficiently reduced to cause the pump to operate in the recirculation zone. To avoid this, a bypass must be provided, increasing the capacity just beyond the recirculation flow whenever only 5 heat exchangers are used.

## Application guidelines.

Based on the previous discussion, the following centrifugal pump application guidelines are suggested as ways to help avoid problems resulting from internal recirculation.

- Unless there is a compelling reason to do so, do not specify npsh values which result in S values much above 9,000.
- When dealing with relatively small pumps, the effect of recirculation is not apt to be as significant as for larger pumps; hence, you can afford to be less conservative.
- Pumps handling hydrocarbons can be operated at lower flows than equivalent pumps handling cold water.
- The risks of operating at flows much below the recirculation flow, and the user's reaction to

the resultant problems, can be better determined after the pump is in operation. To avoid expensive "after-the-fact" changes, provide for the possibility of increasing the minimum flow bypass if there is a suspicion that too optimistic a decision has been made at the time the pump was selected.

- If a 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 recirculation.

With the publication of "Flow recirculation in centrifugal pumps," another brick has been added to the edifice of knowledge on centrifugal pumps. By no means, however, does it mean that our knowledge is complete. With continuing research efforts, we can hope that we may manage a better coexistence with internal recirculation, even to the point of regulating its occurrence to lower flows, or mitigating its effects after onset without sacrificing the advantages of higher suction specific speed values, i.e. lower required npsh.

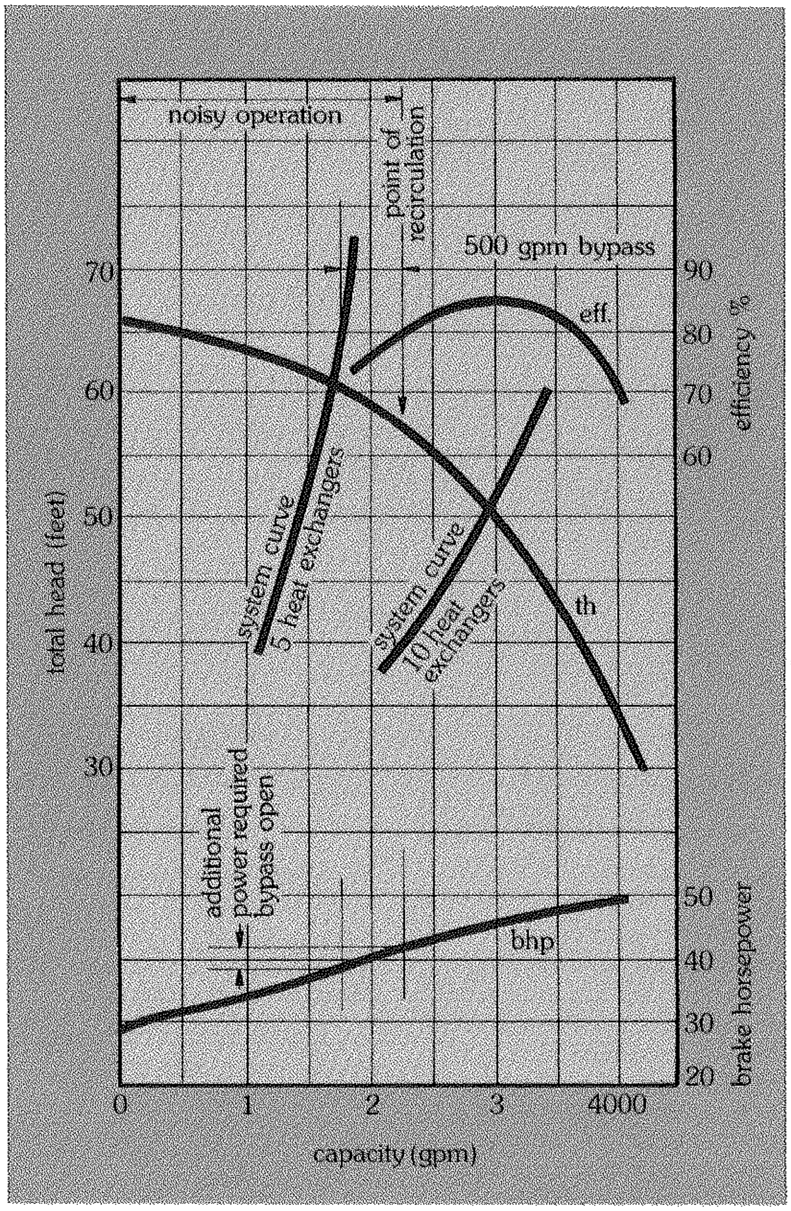



**Figure 2—Pump installation discharging through either 10 or 5 heat exchangers in parallel.**

**Recirculation questions solicited for future issue:**

Power & Fluids will address questions related to the material presented here in an upcoming issue. Please submit your questions to:

Editor, Power & Fluids
Worthington Group
McGraw-Edison Company
270 Sheffield Street
Mountainside, NJ 07092

# Appendix

## Discharge Recirculation:

1. Calculate discharge vector diagram angle $\beta_2$:

$$\text{Sin } \beta_2 = \frac{F_2}{\pi x D_2 x B_2} \text{ (refer to Figures A \& B)}$$

2. Determine $\frac{C_{m2}}{U_2}$ from Figure C (page 16)

3. The flow in gpm at discharge recirculation is equal to:

$$\frac{D_2^2 x B_2 \text{ x rpm}}{23.5} \text{ x } \frac{C_{m2}}{U_2} \quad \text{......} \quad (1)$$

## Suction Recirculation:

**Case 1 where:** $\mathbf{\frac{D_1}{D_2} < 0.5}$

1. Calculate inlet vector diagram angle $\beta_1$:

$$\text{Sin } \beta_1 = \frac{1.273F_1}{D_1^2 - h_1^2} \text{ (refer to Figures A \& B)}$$

2. Determine $\frac{V_e}{U_1}$ from Figure D (page 16)

3. The flow in gpm at suction recirculation is equal to:

$$\frac{D_1(D_1^2 - h_1^2) \text{ x rpm}}{93.45} \text{ x } \frac{V_e}{U_1} \quad \text{......} \quad (2)$$

**Case 2 where** $\mathbf{\frac{D_1}{D_2} > 0.5}$

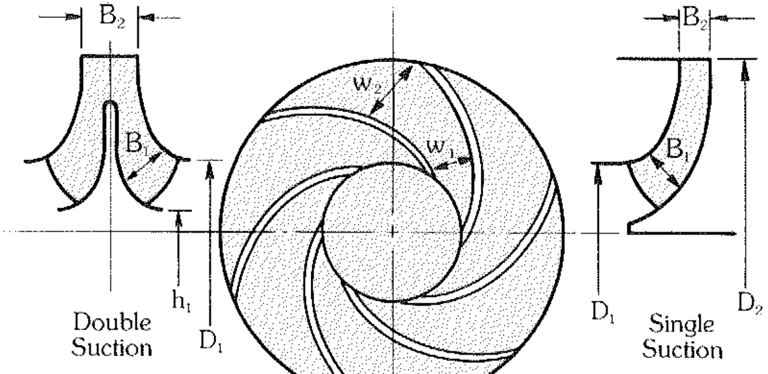
Whenever $D_1/D_2$ is equal to or greater than 0.5, the suction recirculation is either equation 1 or 2, whichever is greater.

### Nomenclature

**$B_2$** –width of the impeller waterway at the discharge diameter
**$C_{m2}$** –meridional velocity at the impeller discharge
**$D_1$** –eye diameter of the impeller
**$D_2$** –discharge diameter of the impeller
**$F_1$** –area between the vanes at the impeller inlet normal to the average meridional velocity
**$F_2$** –area between the vanes at the impeller discharge normal to the average meridional velocity
**$h_1$** –shaft diameter through the impeller eye
**$U_1$** –peripheral velocity of the impeller eye
**$U_2$** –peripheral velocity of the impeller discharge diameter
**$V_e$** –axial fluid velocity in the impeller eye
**$\beta_2$** –discharge vector diagram angle–degrees (see Figure B)
**$\beta_1$** –inlet vector diagram angle–degrees (see Figure B)
**npsh** –net positive suction head

**Appendix is extracted from "Recirculation In Centrifugal Pumps," presented as a paper at the 1981 annual ASME meeting.*

**Figure A**



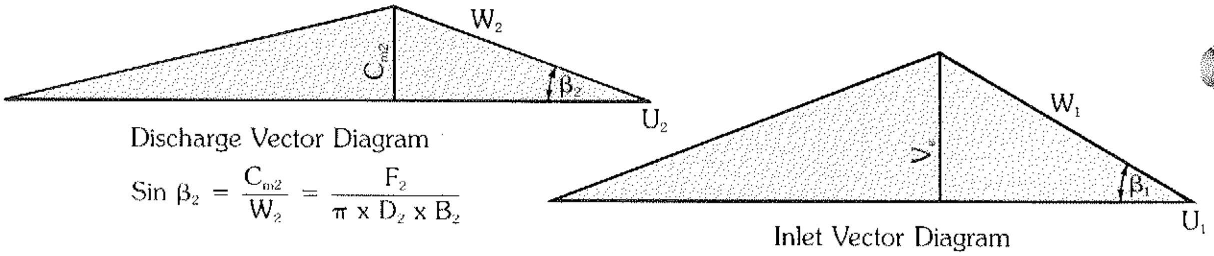

Double Suction:

$F_2 = B_2 \times w_2 \times \text{No. Vanes}$
$F_1 = B_1 \times w_1 \times \text{No. Vanes} \times 2$

Single Suction:

$F_2 = B_2 \times w_2 \times \text{No. Vanes}$
$F_1 = B_1 \times w_1 \times \text{No. Vanes}$

**Figure B**




Discharge Vector Diagram

$$\text{Sin } \beta_2 = \frac{C_{m2}}{W_2} = \frac{F_2}{\pi \times D_2 \times B_2}$$

Inlet Vector Diagram

$$\text{Sin } \beta_1 = \frac{V_e}{W_1} = \frac{1.273\ F_1}{D_1^2 - h_1^2}$$

**Figure C**

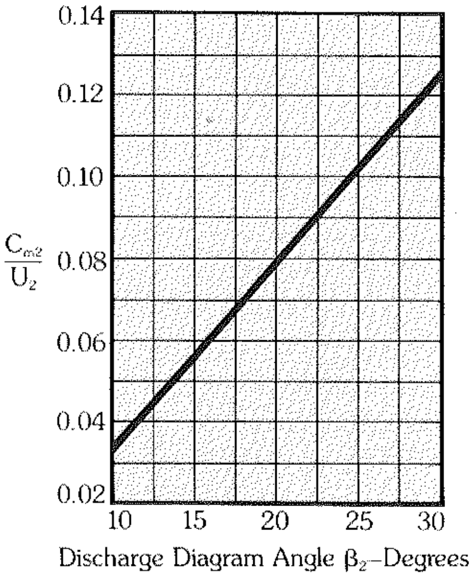
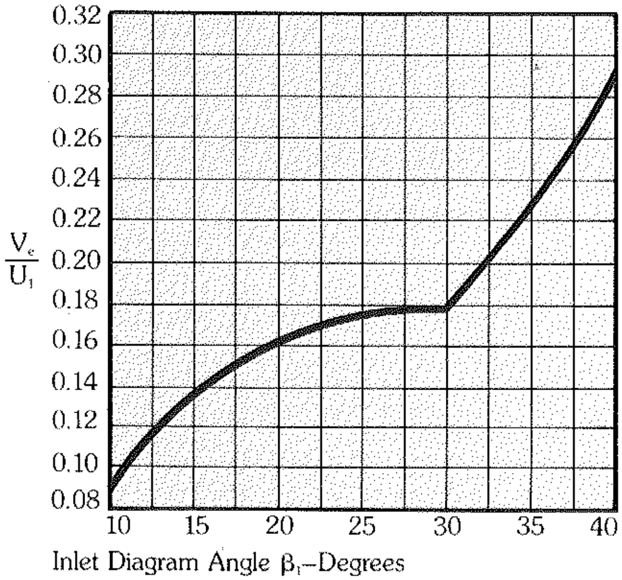



**Figure D**



## Examples

**1. Double suction pump where:**

$D_2$ = 15.5 in. $h_1$ = 2.875 in.
$B_2$ = 1.75 in. $F_1$ = 30 in.$^2$
$F_2$ = 30 in.$^2$ $\frac{D_1}{D_2} = 0.47$
$D_1$ = 7.185 in.

**Pump rating at peak efficiency is:**

3500 gpm
220 ft. thd
1750 rpm
17 ft. npsh

**Discharge recirculation:**

$$\text{Sin } \beta_2 = \frac{F_2}{\pi \times D_2 \times B_2} = \frac{30}{\pi \times 15.5 \times 1.75} = 0.352\ (20.6°)$$

$$\frac{C_{m2}}{U_2} = 0.084 \text{ from Figure C}$$

$$\text{Flow} = \frac{D_2^2 \times B_2 \times \text{rpm}}{23.5} \times \frac{C_{m2}}{U_2} = \frac{15.5^2 \times 1.75 \times 1750}{23.5} \times 0.084 = 2630 \text{ gpm}$$

**Suction recirculation:**

$$\text{Sin } \beta_1 = \frac{1.273\ F_1}{2(D_1^2 - h_1^2)} = \frac{1.273 \times 30}{2(7.185^2 - 2.875^2)} = 0.44\ (26.0°)$$

$$\frac{V_e}{U_1} = 0.172 \text{ from Figure D}$$

$$\text{Flow} = \frac{D_1\ (D_1^2 - h_1^2) \times \text{rpm}}{93.45} \times \frac{V_e}{U_1} = \frac{7.185\ (7.185^2 - 2.875^2) \times 1750}{93.45} \times 0.172 = 1000 \text{ gpm} \times 2 = 2000 \text{ gpm}$$

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**2. Single suction pump where:**

$D_2$ = 11.875 in. $h_1$ = 0
$B_2$ = 1.875 in. $F_1$ = 17 in.$^2$
$F_2$ = 19 in.$^2$ $\frac{D_1}{D_2} = 0.547$
$D_1$ = 6.5 in.

**Pump rating at peak efficiency is:**

1750 gpm
100 ft. thd
1750 rpm
17 ft. npsh

**Discharge recirculation:**

$$\text{Sin } \beta_2 = \frac{F_2}{\pi \times D_2 \times B_2} = \frac{19}{\pi \times 11.875 \times 1.875} = 0.271\ (15.7°)$$

$$\frac{C_{m2}}{U_2} = 0.062 \text{ from Figure C}$$

$$\text{Flow} = \frac{D_2^2 \times B_2 \times \text{rpm}}{23.5} \times \frac{C_{m2}}{U_2} = \frac{11.875^2 \times 1.875 \times 1750}{23.5} \times 0.062 = 1220 \text{ gpm} \qquad (1)$$

**Suction recirculation:**

$$\text{Sin } \beta_1 = \frac{1.273\ F_1}{D_1^2 - h_1^2} = \frac{1.273 \times 17}{6.5^2 - 0} = 0.512\ (30.8°)$$

$$\frac{V_e}{U_1} = 0.18 \text{ from Figure D}$$

$$\text{Flow} = \frac{D_1\ (D_1^2 - h_1^2) \times \text{rpm}}{93.45} \times \frac{V_e}{U_1} = \frac{6.5\ (6.5^2 - 0) \times 1750}{93.45} \times 0.18 = 925 \text{ gpm} \qquad (2)$$

In this case, $\frac{D_1}{D_2}$ exceeds 0.5; therefore, the flow at suction recirculation is the greater of equations 1 and 2, or 1220 gpm.