FLOW RECIRCULATION IN CENTRIFUGAL PUMPS

by

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ABSTRACT

Recirculation in centrifugal pumps is a flow reversal at the inlet or at the discharge tips of the impeller vanes. Through analysis and tests it is now possible to determine the flow patterns that must exist to produce the observed results. All impellers exhibit a point of suction recirculation and a point of discharge recirculation at some specific capacity, and depending on the size and speed of the pump, the effects of recirculation can be very damaging, not only to the operation, but also to the life of the impeller and casing.

The symptoms associated with recirculation are very specific and are listed along with the diagnoses of the cause. Many of these problems can be avoided by specifying and designing pumps for lower suction specific speeds and limiting the range of operation to capacities above the point of recirculation.

INTRODUCTION

The pressure field produced in a centrifugal pump impeller at a flow corresponding to the peak efficiency is more uniform and more symmetrical than at any other flow. At flows less than that at the 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. Recirculation can occur at the discharge of the impeller, at the suction of the impeller, or at both the suction and at 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. Similarly, the higher the design suction specific speed, the narrower is the margin of safety between the rated capacity and the suction recirculation capacity.

CAUSE AND EFFECT

In the analysis of the cause and effect of recirculation one might begin by asking the question why a reversal of flow occurs at all at reduced flows. The answer seems to be related to the fact that the pressure field not only increases from suction to discharge, but also to the fact 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, the direction of flow reverses, and the flow is from the discharge to the suction of the impeller.

Because the pressure field is 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 is most likely to occur.

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 are listed below:

Symptoms Associated with Suction Recirculation

1. Cavitation damage to the pressure side of the vane at the inlet to the impeller.
2. Cavitation damage to the stationary vanes in the suction.
3. Random crackling noise in the suction as contrasted to

the steady crackling noise associated with cavitation from inadequate NPSH.

4. Surging in the suction of the pump.

Symptoms Associated with Discharge Recirculation

1. Cavitation damage to the pressure side of the vane at the discharge of the impeller.
2. Axial movement of the shaft with or without damage to the thrust bearing.
3. Cracking or failure of the impeller shrouds at the discharge of the impeller.
4. Shaft failures on the outboard end of double suction and multistage pumps.
5. Cavitation damage to the tongue or to the inlet of the diffuser vanes of the casing.

DESCRIPTION

Suction Recirculation

The reversal of the flow in the eye of the impeller at the point of suction recirculation has been observed in laboratory tests. Figure 1 shows a laboratory test arrangement of a six inch end suction pump installed in a test loop with a transparent suction pipe that permits the visual observation of the 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 vortex in the suction 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 the vane in the area approximately midway

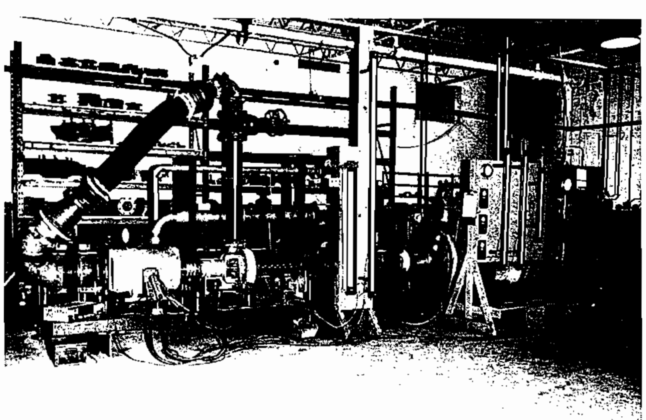

Figure 1. Laboratory Test Pump.

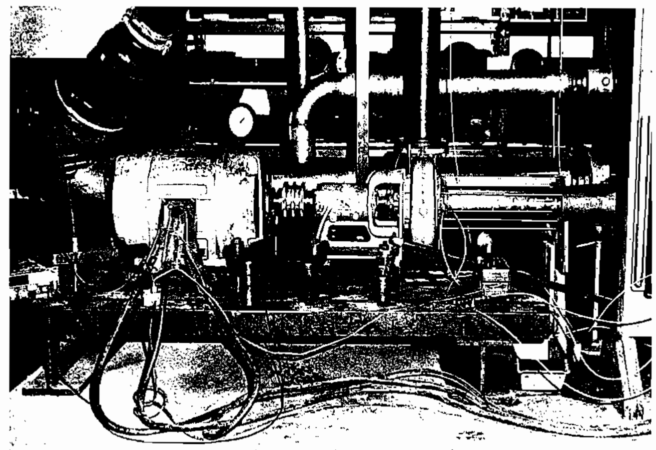

Figure 2. Laboratory Test Pump Showing Transparent Suction Pipe.

between the hub and the shroud. Figure 3 shows schematically the flow at the impeller eye during recirculation. Figure 4 shows the use of a mirror to examine the pressure or underside of the vane for suction recirculation damage. Figure 5 shows a section of a vane removed from the inlet of a large impeller heavily damaged from suction recirculation.

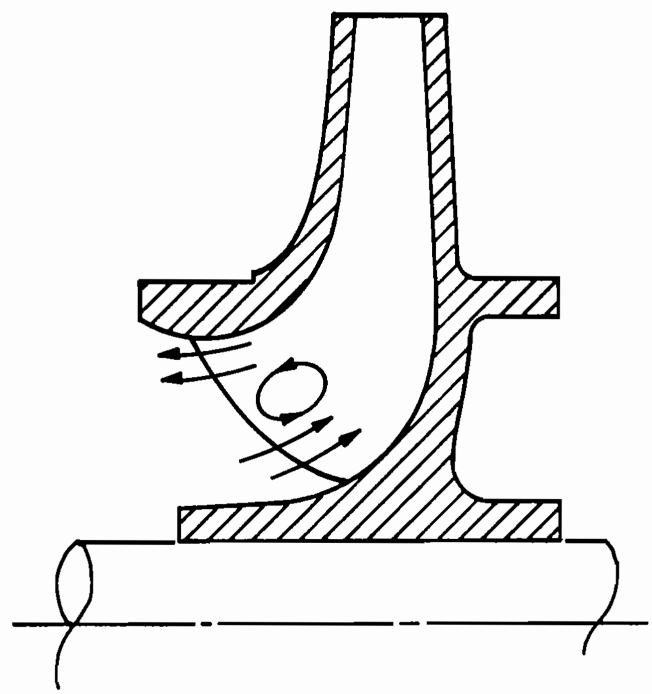

Figure 3. Suction Recirculation.

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-

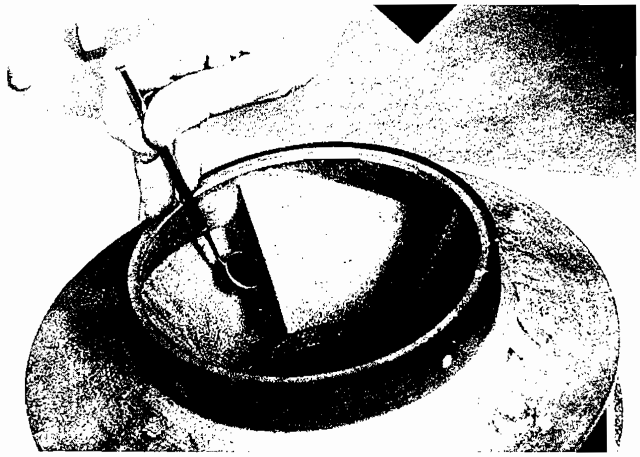

Figure 4. Inspection of Pressure Side of Inlet Vane for Recirculation Damage.

Figure 5. Damage to the Pressure Side of the Inlet Vane from Suction Recirculation.

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 is 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 6 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 7 shows the damage to the pressure side of the discharge vane from operation in the discharge recirculation zone.

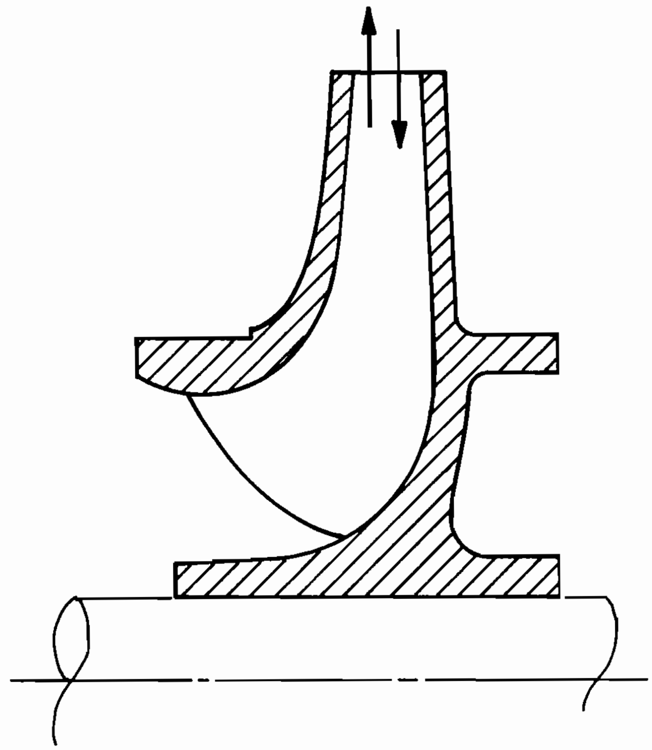

Figure 6. Discharge Recirculation.

Figure 7. Damage to the Pressure Side of the Outlet Vane from Discharge Recirculation.

SYMPTOMS AND DIAGNOSES

The following are some of the many symptoms of poor pump operation with possible causes of this performance.

Symptoms	Diagnoses
1. Cavitation damage on the pressure surface of the inlet of the impeller vane (Figure 8).	The 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. The 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 case 1. The periodic crackling noise comes from the formation and the 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 case 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 case 1 except that two-phase flow is involved with a mixture of a gas or vapor and the liquid. The 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 ten feet or less.
5. Cavitation damage on the pressure surface of the discharge portion of the impeller vane (Figure 8).	The pump is operating in discharge recirculation with a portion of the flow jetting back into the impeller channels. The 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 case 5. The pressure pulsations produced by the cavitating vortices on the vane surface between the shrouds is 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 case 5. As the flow reverses during the discharge recirculation the tongue or diffuser vanes now 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 case 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 case 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.

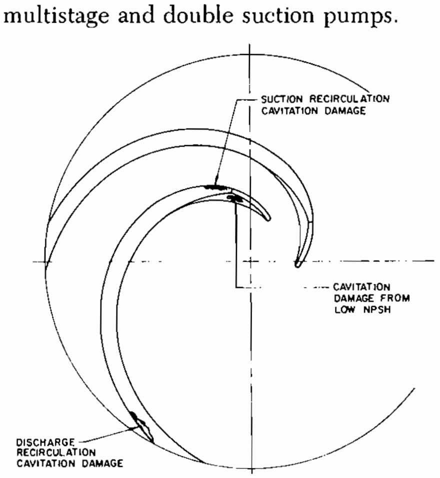

Figure 8. Damage Types and Locations.

SELECTION CHARTS

Figures 9 and 10 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 increases. This difference is shown in the form of two selection charts. Figure 9 should be used for specific speeds in the 500 to 2500N_s range and Figure 10 for specific speeds from 2500 to 10000N_s. While these recommendations 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 recommendations do represent the average or the norm for good commercial designs.

While Figures 9 and 10 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 2500 GPM or less and heads up to 150 feet 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 2500 GPM or less and heads of 150 feet or less, the minimum flow values can be set at fifty percent of the recirculation values for continuous operation and twenty-five percent for intermittent operation. Similarly for pumps of all sizes handling hydrocarbons, the minimum flow values can be set at sixty percent of the recirculation flow for continuous operation and twenty-five percent for intermittent operation.

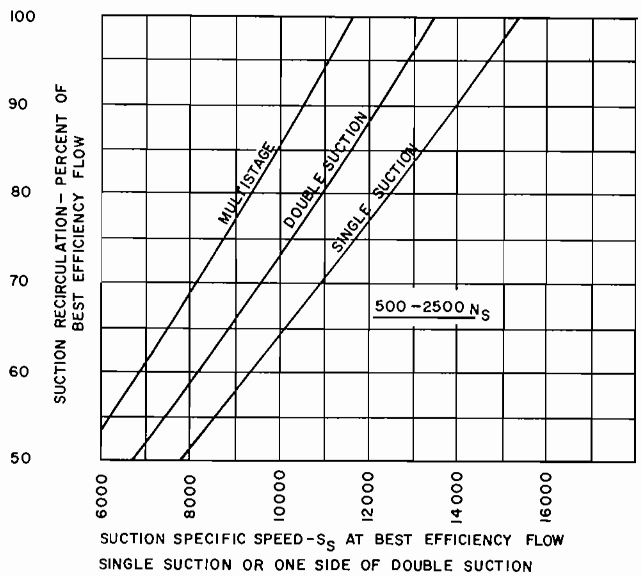

Figure 9. Suction Recirculation — 500 to 2500N_s.

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 2750N_s and a suction specific speed of 10000N_s. Most of the operation of the pump occured at seventy-five 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

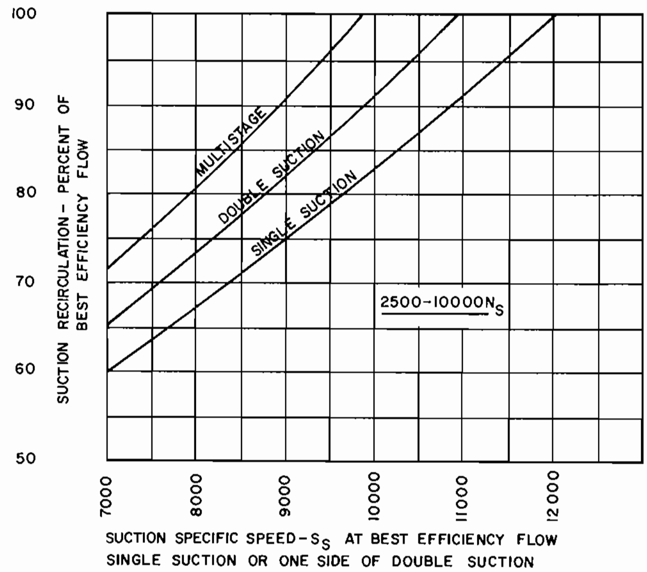

Figure 10. Suction Recirculation — 2500 to 10000N_s.

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 1500N_s specific speed was operated for extended periods at low loads during the start-up of the 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.

CONCLUSIONS

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 characteristics of the fluid pumped, and upon the materials of construction. For example, a 2500 GPM pump operating at 100 feet of total head may not exhibit any damage, but a 25000 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 25000 GPM pump handling water will have the symptoms associated with recirculation, the same pump pumping a hydrocarbon will exhibit lower levels of noise, vibration and cavitation damage.

The question of the 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 2500 GPM or larger. As a general rule, suction specific speeds in excess of 9000S_s for pumps designed for specific speeds of 2550N_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 the complete performance characteristics 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.