EXPERIMENTAL STUDY OF THE INFLUENCE OF BACKFLOW CONTROL ON PUMP HYDRAULIC-MECHANICAL INTERACTION

by

Paul J. Kasztejna

Supervising Design Engineer

Charles C. Heald

Manager, Engineering

Engineered Pump Division

Phillipsburg, New Jersey

and

Paul Cooper

Senior Staff Researcher

Research Center

Ingersoll-Rand Company

Princeton, New Jersey

Paul J. Kasztejna is Supervising Design Engineer for Process and Industrial Pumps, Engineered Pump Division, Ingersoll-Rand Company. He received a B.S. degree in Mechanical Engineering from Pennsylvania State University in 1972. Since joining Ingersoll-Rand in 1973, Mr. Kasztejna has held various positions involved with pump design, hydraulic design and application. He has been responsible for slurry pump project engineering as well as special development projects relative to process pumps and slurry pumps in conjunction with Ingersoll-Rand Research, Incorporated. His most recent assignments have involved hydraulic and mechanical design integration with CAD/CAM.

Charles C. Heald is Manager, Engineering, for the Engineered Pump Division of Ingersoll-Rand Company in Phillipsburg, New Jersey. He is responsible for design, development, and product engineering for the company's process and industrial pump lines. He is also responsible for Computer Aided Design and Drafting and Technical Services for the Engineered Pump Division.

Mr. Heald received a B.S. degree in Mechanical Engineering from the University of Maine in 1960. Since that time, he has held various positions with Ingersoll-Rand, including Development Engineer, Senior Hydraulic Design Engineer and Manager, Pump Design. He is a member of ASME, AWS and the API Centrifugal Pump Manufacturer's Subcommittee, and is currently serving on the API-610 7th Edition Task Force.

Paul Cooper is a Senior Staff Researcher at the Ingersoll-Rand Research Center in Princeton, New Jersey. He is responsible for hydraulic research on turbines and pumps, and the application of this work to the design and performance analysis of these machines. Included is research on cavitation damage in pumps and hydraulic design approaches needed to improve performance at both design and off design flow conditions.

Dr. Cooper was involved previously with pump and inducer hydraulic research and development at TRW, Incorporated. He holds several patents in the area of pump design.

Dr. Cooper received a B.S. degree in Mechanical Engineering in 1957 from Drexel, a M.S. degree in Mechanical Engineering in 1959 from Massachusetts Institute of Technology and a Ph.D. in Engineering in 1972 from Case Western Reserve. He is a member of the Executive Committee of the Fluids Engineering Division of ASME and serves on the Advisory Committee for the International Pump Symposium at Texas A&M University.

ABSTRACT

The hydraulic/mechanical response of a high suction specific speed end suction process pump is investigated experimentally at reduced flowrates, where backflow or suction recirculation was observed and cavitation surge experienced. Introduction of a backflow recirculator completely eliminated the attendant strong low frequency fluctuations of suction pressure and axial thrust. It is concluded that by this means, high suction specific speed pumps can be designed to operate reliably and durably at flowrates much lower than the BEP value.

INTRODUCTION

The trend of centrifugal process pump requirements until the late 1970s led to smaller, cost-effective pumps running at higher speeds, delivering more flow and head per stage at lower net positive suction head required ($NPSH_R$) values. Thus, end suction process pumps became high suction specific speed (Nss) machines with Nss over 11,000 ($Nss = [N \text{ (rpm)}\ Q^{1/2} \text{ (gpm)}]/[(NPSH_R)^{3/4} \text{ (ft)}]$) and with a characteristic low inlet flow coefficient ($C_{m.1}/U_e$), where $C_{m.1}$ is the absolute inlet eye velocity. This corresponds to large impeller eye diameters and high blade tip speeds (U_e) at the inlet. This impeller inlet geometry readily produced backflow when the flowrate was reduced below the best efficiency point (BEP). The resulting hydraulic instabilities meant that these pumps were restricted to a narrow range of flowrates in order to ensure smooth operation.

Quoted minimum flowrates of 50 to 80 percent of that at BEP are not uncommon for such machines. In fact, most processes are not controlled closely enough for pumps involved to operate within such limits. A statistical survey by Hallam [1] indicated that pumps capable of $Nss \geq 11{,}000$ have been failing at twice the rate of those with $Nss \leq 10{,}000$. While these failures can be safely categorized as mechanical, i.e., the failure of a bearing or seal, or the breakage of the shaft or the impeller, hydraulic instability associated with backflow at the inlet eye is cited as the underlying cause of failure.

Thus, researchers have been seeking to understand the nature and conditions for occurrence of backflow fields both at the inlet and discharge of pump impellers [2,3,4]. In fact, design remedies for resulting mechanical failures have recently been identified. In one instance, cavitation surge, which is the interaction of backflow and two-phase flow in the impeller eye region of a centrifugal pump, was shown to exist at typical NPSH values. The introduction of an annular stationary passage around the inlet pipe, known as a backflow recirculator (BFR), was demonstrated to eliminate this hydraulic instability [5].

In another case, Makay and Barrett removed adverse mechanical response to unsteady discharge recirculation by altering the sizes of the gaps between the outer circumference of the impeller and the surrounding casing elements [6]. However, end suction process pumps of high suction specific speed have sufficiently low energy levels so as not to be mechanically sensitive to these discharge recirculation-related disturbances. In fact, they would not readily be expected to suffer from such problems, since pumps of the same specific speed and lower Nss (and, therefore, of the same impeller exit design) give satisfactory service. The flow problems of these pumps are obviously associated with the suction instabilities of high Nss machines, as postulated by Hallam [1].

These instabilities and their characteristic low frequency (1 to 6 Hz) pressure pulsations [7] can be experimentally connected with the kind of mechanical response that can reasonably be expected to cause the failures experienced in Hallam's survey. A remedy for this problem, namely, the backflow recirculator, was included in these experiments. The reduction that this device produced in these pressure pulsations [5] is confirmed, and a resulting reduction in mechanical response is obtained, namely in axial-thrust fluctuations.

The 16,000-Nss end suction double-volute pump reported in [5] was selected for this experimental program. To minimize the effect of discharge-related hydraulic instabilities, a smaller-diameter, cut-down impeller was used that extended radially only slightly beyond the sealing-ring diameter and which had a radial clearance from the two cutwaters that was 33 percent of the impeller radius. The end suction configuration is the most susceptible to cavitation surge, because the associated vapor core that forms in the suction pipe by reason of the vorticity of the backflow is not supplanted by a shaft extending through the eye, as is the case in double suction pumps [5]. Observation of this core and its behavior during the surge cycle was afforded by the use of a transparent suction pipe. This pipe also allowed observation of the improvements resulting from the introduction of an insertable backflow recirculator into the test pump. This device, shown in Figure 1, consists of (a) an annular slot for the reception of swirling backflow emerging from the impeller, (b) a set of straightening vanes in an annulus surrounding the inlet pipe to remove the swirl and (c) a second annular slot for returning the deswirled backflow to the main flow stream.

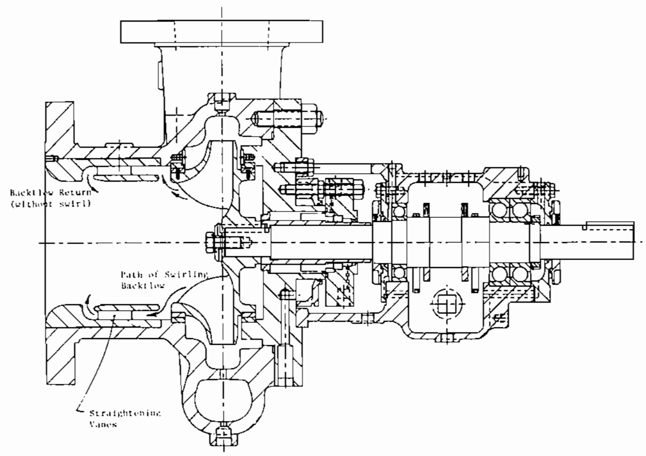


Figure 1. Sectional Drawing of 6 × 10 × 12 Test Pump Showing Backflow Recirculator Inserted into Suction Nozzle.

Testing was conducted at flowrates at and below 60 percent of BEP, corresponding to the zone in which extensive instability can be expected for high Nss [6]. Because the frequency of cavitation surge is somewhat random with a fairly wide bandwidth, it is difficult to get meaningful vibration data on this motion, which does not have a constant periodicity. Therefore, though vibration measurements were attempted, they did not reveal significant data. Yet the visual and auditory observation of the pump—especially the adjacent piping at all flowrates (constant speed)—were dramatic and were greatly affected by the backflow recirculator. For this reason, such observations can reasonably be included with the pressure and thrust data that were taken, the sum total of this information providing an indication of the mechanical response, reliability and durability of the pump.

TEST SET-UP

All experiments and tests were conducted on a commercial, end suction, single stage process pump. Some of the significant design parameters of the 6 × 10 × 12 pump used in all of the testing are given in Table 1. A cross-section of the pump under study is shown in Figure 1. The pump is typical of current process pump designs except that it accommodates a backflow recirculator within the suction nozzle. The backflow recirculator is of the standard design used in conjunction with this pump. No design modifications were made to any area of the pump used for this test program. All parts, except for the axial load cell adapter/insert, were production components.

The test loop was specifically designed for flow visualization studies at the pump suction, with and without the use of the backflow recirculator. The test facility and the suction line fitted with a 54 in long, 10 in diameter clear section is shown in Figure 2. A schematic diagram of the complete test loop is shown in Figure 3. The suction pressure transducer was located 6 in upstream of the suction flange. The discharge pressure transducer was located 36 in downstream of the discharge flange.

In constructing the loop, as much stainless steel was used as possible, to eliminate the need for rust inhibitors and still

Table 1. Design Parameters of a 6 × 10 × 12 Pump.

Pump Data	
Speed	3570 RPM
Flow Rate (BEP)*	2500 GPM
Head*	560 Ft.
NPSHR (3% Head Loss – BEP)	25 Ft.
Specific Speed	1550
Suction Specific Speed	16,000
Max. Impeller Diameter	12.75 In.
Test Impeller Diameter	10.25 In.
Impeller Data:	
Eye Diameter	7.50 In.
Number of Vanes	5
Inlet Vane Angle	12 Deg.
Casing Cutwater Diameter	13.62 In.

*These data apply to the maximum impeller diameter. Figure 5 shows the performance data for both maximum diameter and test diameter.

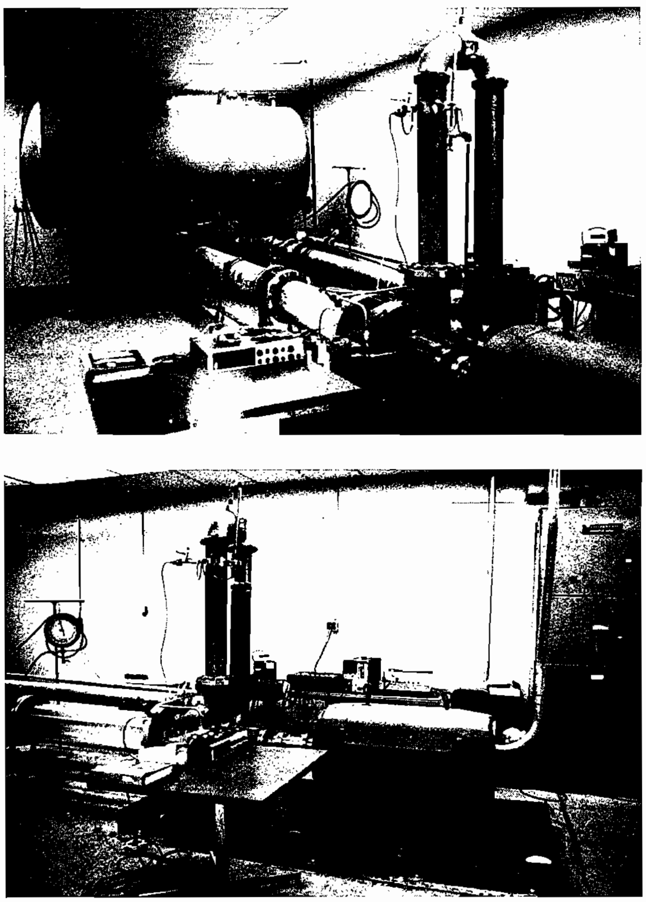
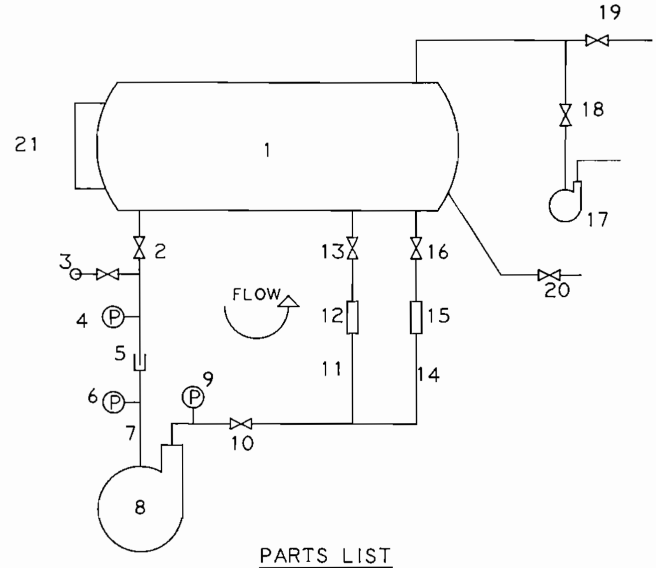


Figure 2. Test Facility. a) View of Loop Piping and Tank. b) View of Test Pump and Transparent Suction Line.



PARTS LIST

1) 1750 gal. TANK	11) 6" DISCHARGE LINE
2) 10" SUCTION ISOLATION VALVE	12) 6" TURBINE FLOW METER
3) 2" DRAIN VALVE	13) 4" ISOLATION VALVE
4) SUCTION VACUUM GAGE	14) 2" DISCHARGE LINE
5) 10" EXPANSION JOINT	15) 2" TUBINE FLOWMETER
6) SUCTION PRESS. TRANSDUCER	16) 2" ISOLATION VALVE
7) 10" DIA.–48" LONG CLEAR PIPE	17) VACUUM PUMP
8) 6X10X12 TEST PUMP	18) 2" VACUUM CONTROL VALVE
9) DISCHARGE PRESS. TRANSDUCER	19) 1" VACUUM BLEED VALVE
10) 6" DISCHARGE CONTROL VALVE	20) 1" FILL LINE
	21) TANK LEVEL INDICATOR

Figure 3. Test Loop Schematic Diagram.

maintain clarity of the water. Each time the backflow recirculator was installed or removed, the suction line and the pump itself had to be drained, making the use of rust inhibitors impractical. The test loop was arranged such that the backflow recirculator could be installed or removed in a matter of minutes by removing the clear section of suction pipe. This allowed test observers to witness quickly the stabilizing effect of the backflow recirculator under identical operating conditions.

TEST PROGRAM

The test program was initially established to visually demonstrate the nature of backflow at off-design flows within the impeller approach and adjacent suction pipe, by means of the 54 in long clear pipe section bolted to the pump suction. To quantify the magnitude of these off-design related disturbances, pressure transducers were mounted to both suction and discharge pipes. Proximity probes were also mounted to the pump to aid in the measurement and recording of shaft vibrations and shaft movement at various flows and NPSH levels. The only location available to install the proximity probes on the commercial pump (without modifying the pump) was immediately adjacent to the radial ball bearing. Readings at this location showed an insignificant difference in amplitude with varying flows and NPSH levels, even when the pump was obviously in distress. An accelerometer fixed to the top of the bearing housing also failed to reflect significant variations, even though suction disturbances were quite significant (both visually and audibly). It was concluded that because of the location of the proximity probes and the sensitivity of the accelerometer, the random low backflow frequency pulsations associated with recirculation and hydraulic surge were not accurately determined by these instruments.

To evaluate the effects of suction pressure pulsations on thrust bearing axial loading, a load cell adapter/insert was designed to fit within the existing standard pump bearing housing—without the need for machining modifications to standard pump parts (Figure 4). By using smaller than standard duplex thrust bearings, a bearing inset/retainer, and modified bearing housing end cover and studs, load cells were installed and preloaded to measure dynamic axial thrust loads in either direction. Load cells were installed on all four end cover studs, and readings were electronically averaged. By using this device, it was possible to measure and record the instantaneous thrust bearing load at various flowrates and NPSH levels. The accuracy of this method of thrust measurement was verified by testing at preloads of 1000 lbs and 2000 lbs (and achieving identical results) and by loading the pump shaft by inserting an independent axial load cell between the pump and motor shafts.

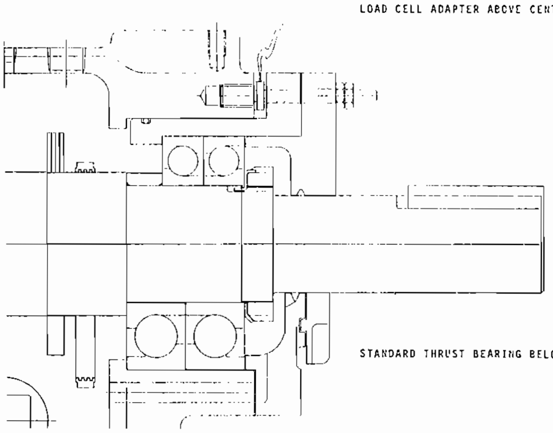


Figure 4. Detail of Axial Thrust Load Cell Installation in Test Pump.

A tabulation of the various tests that were conducted is presented in Table 2. Test Group I did not include provisions for recording the axial load. Therefore, Group II was undertaken in order to explore axial load pulsations. It was during this testing that shaft deflection/vibrations and bearing housing accelerations were abandoned because of the lack of response. Flowrates studied were primarily those at 60 percent of BEP and lower. Higher flowrates were not studied because of limitations of the loop. It was anticipated that the effects of backflow or recirculation would be of no significance at higher flowrates [5], and this was verified by test results at 1500 gpm.

Tests were conducted at constant capacity, varying NPSH from above atmospheric pressure down to an assured breakdown in performance (as listed). Typically, no more than two or three flowrates were run in succession. Water temperature variations of +10°F were conducted within a range of 70°F to 110°F. Tests within the water temperature range at exactly the same NPSH levels are not always shown, because of the temperature variation within this range.

A maximum diameter performance curve is shown in Figure 5, along with a curve of performance at 10.25 in diameter. The variations of head and efficiency as functions of the flowrate are identical for the pump fitted with or without the backflow recirculator. No significant changes in the head-capacity performance were noted during the testing. Test results given in this discussion are those obtained using the impeller at the 10.25 in reduced diameter. The smaller diameter was beneficial in energy consumption and system heat dissipation. The reduced impeller exit diameter of these tests had negligible effect on the pump suction performance.

The instrumentation used during the test program is listed in Table 3. The last column lists the accuracy as specified by the manufacturer or obtained via the calibration system used prior to testing. Pressure measuring transducers and readouts were calibrated as a unit to give accuracies of ±0.01 percent of reading or better. The complete system accuracy of the transducer, readout, recorder, analyzer and plotter was calculated at ±2 percent for the resulting pressure pulsation data. The thrust load measurement system of load cells, junction box, readout, recorder analyzer and plotter yielded an accuracy of ±5 percent for the data presented. Flowrate measurements were recorded manually and, although the equipment yielded an accuracy of ±0.2 percent, overall flow measurement accuracy was ±4 percent at the low flowrates (300, 500, 750 gpm) and better than ±2 percent at the higher flowrates (1000, 1500 gpm).

Table 2. List of Tests.

	Group I Data Recorded: Suction & Discharge Pressure Pulsations Shaft Vibration & Deflection Bearing Housing Accelerations		Group II Data Recorded: Suction & Discharge Pulsations Axial Thrust Bearing Loads	
	NPSH Levels		*NPSH Levels*	
Flow (GPM)	*A) With BFR*	*B) Without BFR*	*A) With BFR*	*B) Without BFR*
300	32.3	32.5	33.2	35.8
"	27.3	29.7	24.2	28.3
"	19.3	23.7	17.7	18.8
"	13.3	19.2	10.2	16.8
"	10.3	16.7	7.25	14.8
"	8.5	--	5.6	12.1
"	7.1	--	--	--
500	32.0	50.4	27.8	25.9
"	26.6	39.9	19.1	18.4
"	18.4	22.2	12.3	13.9
"	12.4	18.2	11.0	11.9
"	10.1	14.2	--	15.9
"	9.2	9.7	--	--
"	8.2	8.7	--	--
"	--	7.9	--	--
750	31.2	31.3	32.1	26.0
"	27.4	27.2	26.5	18.3
"	17.9	20.2	17.0	14.5
"	12.9	14.9	11.5	12.5
"	10.4	11.2	10.0	10.5
"	15.4	9.9	8.6	--
1000	26.6	39.4	31.5	26.2
"	18.6	23.3	27.0	16.7
"	15.6	17.5	18.5	14.5
"	13.6	12.4	14.0	12.3
"	--	9.1	11.9	11.1
"	--	7.3	--	--
1500	25.8	30.3	33.5	25.2
"	19.8	26.3	26.7	19.0
"	15.3	21.3	17.0	14.8
"	13.8	16.3	14.1	16.0
"	14.8	14.3	12.3	--

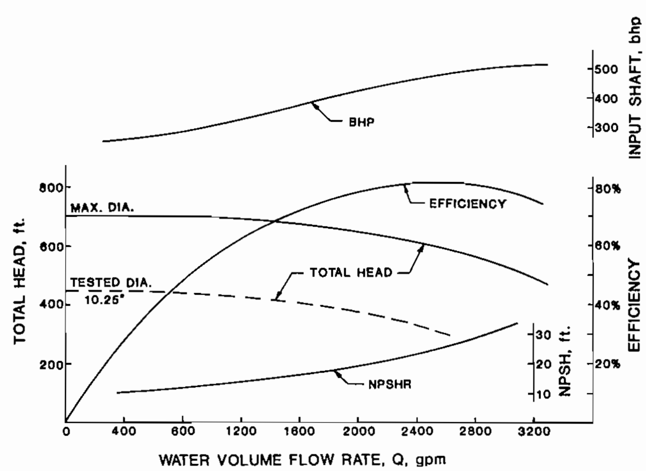


Figure 5. Performance Curves of 6×10×12 Test Pump at 3570 RPM. Results obtained with and without the backflow recirculator differed negligibly. $NPSH_R$ is on the basis of three percent head deviation.

Table 3. Instrumentation

Manufacturer	Model	Instrument	Accuracy
Nicolet	446	Mini-ubiquitous FFT Computing Spectrum Analyzer	±.25%
Hewlett-Packard	7044A	X-Y Recorder	±.5%
Racal Recorder	14DS	Tape Recorder 14 Channel	±.75%
Ampex	797	Wideband Recording Tape (Tape Speed 3¾"/Sec.)	----
Tektronix	5110	Oscilloscope - 2 Channel	±.5%
Trig-tek	401A	Portable Calibrator (Frequency/Amplitude Generator)	±.5%
Daytronic	3270	Strain Gage Conditioner/Indicators (Suction and Discharge)	±.15%
Viatran	104	Pressure Transducers (Suction and Discharge)	±.15%
Red Lion	DT3D	Digital Indicator/Readout (2) (6" Meter and 2" Meter)	±.15%
Flow Technology	FT32 -	2" Flow Turbine Meter/Mag. Pickup	±.03%
	FT96 -	6" Flow Turbine Meter/Mag. Pickup	±.03%
Lebow	3711-250	(4) Load Sensors	±.05%
Lebow	7535	Digital Strain Gage Indicator	±.02%
Lebow	C-20	Load Cell Summing Box	±.05%
Heist	35314	Vacuum Gage	±1.5%
Bently Nevada	5MM	Proximity Probes	±2%
Bently Nevada	7200	Proximeter	±1%
Ithaco	454	Amplifier/Accelerometer	±1%
Endevco	2236	Accelerometer	±5%

TEST RESULTS

The effects of the 16,000 suction specific speed impeller with and without backflow control are presented with respect to parameters of suction and discharge pulsations, axial thrust bearing load pulsations, and visual suction disturbances.

The suction and discharge pulsations for all flows rated below 50 percent of BEP when not using a backflow recirculator typically follow that shown in Figure 6. Without the use of the backflow recirculator, NPSH levels in excess of three times that required at three percent head deviation were necessary to begin to suppress major pressure pulsation (Figure 6-1). These high amplitude pressure pulsations are the result of the interaction between unsteady, separated swirling reversed flow and the attendant fluctuations of vapor volume in the impeller inlet region known as cavitation surge [5]. Note also that at significant head deviation (beyond 10 percent), the impeller cavitation begins to create discharge pulsations, and the suction pulsations are no longer significant, because the inlet region of the impeller is full of two-phase fluid and is incapable of driving significant backflow (Figure 6-7).

By using a backflow recirculator, the suction pressure pulsation amplitudes are reduced by an order of magnitude and the discharge pulsation amplitudes are similarly decreased. This can be seen by comparing Figures 6 and 7, both at the same flowrate of 500 gpm (approximately 25 percent BEP). Note that the suction pressure pulsations on Figure 7 remain steady and do not induce ~~fur~~ther pulsations at the discharge until the impeller is in full cavitation and performance is broken down well beyond three percent head deviation. The data obtained at the other flowrates of 300 gpm, 700 gpm, 1000 gpm, and 1500 gpm were similar to these results. The 500 gpm flowrate data typically illustrate the nature of the phenomenon at all flows.

Suction pressure pulsations are shown again in Figures 8 and 9, but here the upper track shows the axial thrust pulsations obtained from the load cells. It can be seen that the suction pressure pulsations without the backflow recirculator are accompanied by a dramatic pulsation or pounding of the thrust bearing within the pump. The thrust wave is characterized by a steady load punctuated by spikes of as much as 800 lbs. These

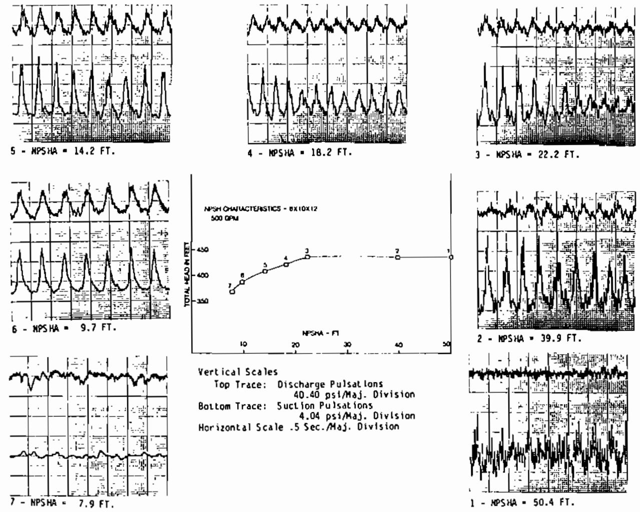


Figure 6. Suction and Discharge Pressure Pulsation Data at 500 GPM and 3570 RPM without the Backflow Recirculator.

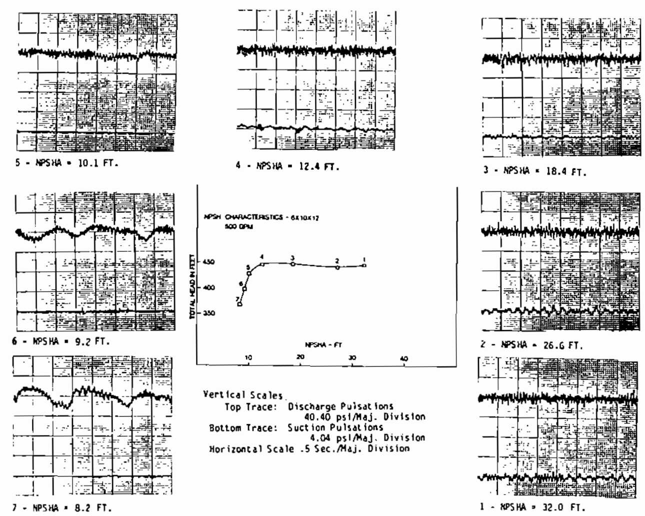


Figure 7. Suction and Discharge Pressure Pulsation Data at 500 GPM and 3570 RPM with the Backflow Recirculator.

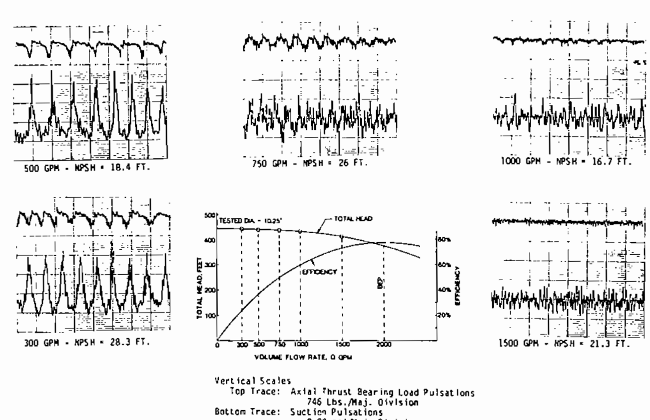


Figure 8. Effect of Flowrate on Axial Thrust without the Backflow Recirculator. Corresponding suction pressure pulsations are also shown to illustrate the relationship between these two quantities.

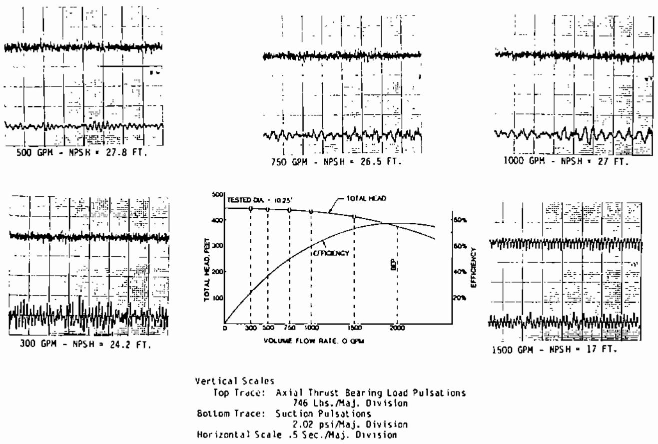


Figure 9. Effect of Flowrate on Axial Thrust with the Backflow Recirculator.

spikes represent a reduction in the axial thrust of the impeller toward the inlet, and they correspond to the suction pressure peaks (maxima). It should be noted that all scales are identical when comparing similar conditions with and without backflow control. These comparisons were made at various NPSH levels, as noted in Table 2. The NPSH level shown at each flowrate is that which produced maximum pulsation activity without the backflow recirculator.

Visual evidence of the improvement in operation when controlling backflow in the pump suction is perceived most dramatically if one stands beside the test loop. It is very difficult to convey the dynamics of what is happening in photographs. Attempts to illustrate this fact are demonstrated in Figures 10, 11 and 12, but they hardly do justice to the disturbances observed by the authors and others during the testing, without the backflow recirculator. This is evidenced by the fact that no photographs are shown for flowrates less than 1000 gpm, because the attendant physical movement of the piping could have resulted in failures of the transparent suction piping. No such concern existed when running at 300 gpm with the backflow control device in place (Figure 12).

DISCUSSION

The foregoing results illustrate the mechanical consequences of suction instabilities at off-design flows. The photographs herein illustrate the connection between the hydraulically induced backflow phenomenon and simultaneous measurements of axial thrust loads and pressure pulsations on the tests which had an impeller designed for 16,000 Nss.

The effects of fluctuating axial loading on a pump rotor can be severe, depending on the frequency. These tests have shown this loading to be a direct function of flow and $NPSH_A$. At high flowrates (closer to BEP), the frequency of the axial thrust pulsations was relatively high and the amplitude was relatively low, even as $NPSH_A$ was reduced to near breakoff. As the flowrate was reduced from that of BEP, the amplitude of axial thrust pulsations increased and the frequency went down. Under cavitation surge conditions, the high amplitude "spikes" occurred at intervals as great as 0.5 second or more. When this occurs, *if* the magnitude of the thrust pulses was greater than the nominal axial thrust load, actual fluctuating thrust reversal could take place; whereas, if the frequency was very high, the inertia of the rotor could eliminate the possibility of thrust reversals. However, during these tests on cold water, with low $NPSH_A$, the impeller unbalanced axial thrust was toward the suction and was of a magnitude such that thrust reversal did not occur. Obvious side effects of such reverse loadings would

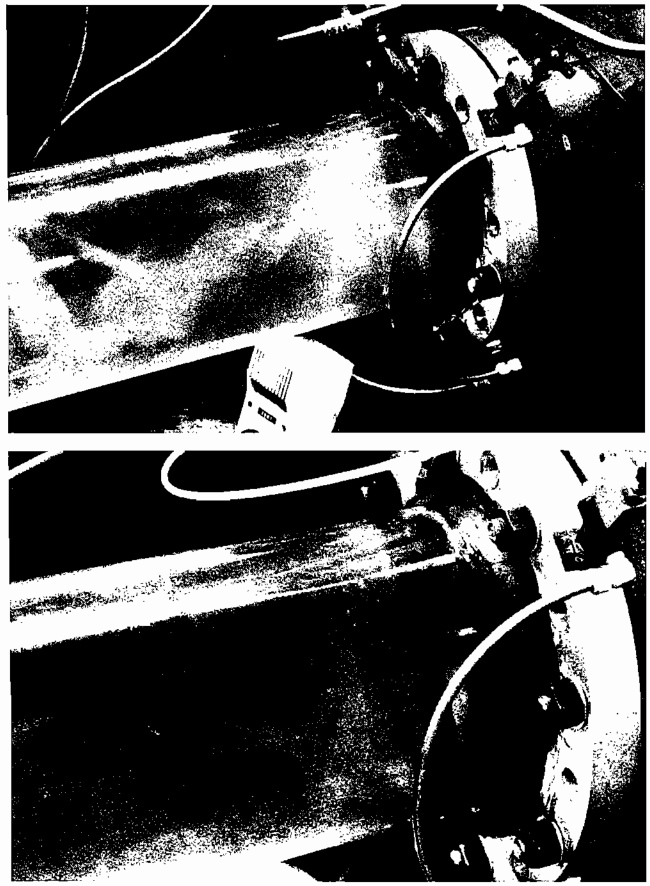


Figure 10. Transparent Suction Pipe at 1500 GPM. a) Without Backflow Recirculator. b) With Backflow Recirculator.

include axial "shuttling" of the rotor, bearing and mechanical seal face fatigue, and possible bearing "brinnelling." All these effects—and certainly others—can contribute to premature mechanical failures.

It is most important to understand that *low* frequency effects—especially low frequency, high amplitude pulses—should be avoided. This is especially true relative to system response, wherein pump-initiated pulsations can excite other components in the pumping system. It is therefore important to note that test results with the backflow recirculator show that the pressure pulsations at low flows and low NPSH levels occur at significantly higher frequencies, as well as lower amplitudes.

By controlling the backflow, the high suction specific speed test pump was shown to be capable of running to extremely low flowrates without generating potentially damaging low frequency, high-amplitude pulsations of suction pressure, discharge pressure and axial rotor load.

CONCLUSIONS

The mechanical response of a high suction specific speed end suction process pump was measured at flowrates from 12 to 60 percent of the best efficiency point with and without a backflow recirculator. The pump had a head of about 400 ft and operated at 3570 rpm. Cavitation surge was observed visually and was recorded in terms of pressure pulsations and axial thrust fluctuations. Without the backflow recirculator, axial thrust spikes of 800 lbs at a frequency of approximately 2 Hz accompanied suction pressure pulsation peak-to-peak amplitudes of three times the average inlet pressure. While such

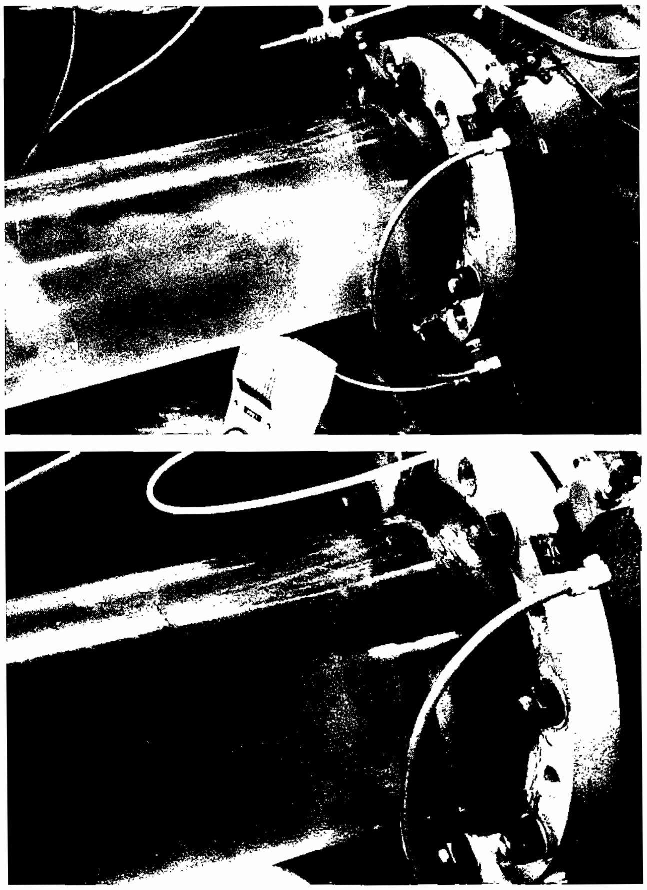


Figure 11. Transparent Suction Pipe at 1000 GPM. a) Without Backflow Recirculator. b) With Backflow Recirculator.

thrust fluctuations did not reverse the impeller thrust, the potential for such a reversal continuously occurring exists if the inlet pressure (at the same NPSH) is high enough (as is the case in many high vapor pressure processes). This would obviously cause both premature bearing and seal failure and, even without a thrust reversal, such load fluctuation would still have an adverse mechanical effect.

However, with the backflow recirculator inserted into the pump, the amplitude of the thrust fluctuation was more than halved and the frequency increased to the blade passing value. This was accompanied by in-phase suction pressure pulsations that were reduced by an order of magnitude. A similar reduction occurred in the discharge pressure fluctuations.

Visual and auditory observations accompanying this behavior were dramatic. Without the backflow recirculator in place, a pulsing core of bubbles extended upstream of the impeller in the suction pipe, the axial extent of this flow pattern becoming greater at the lower flowrates [7]. The piping attached to the pump vibrated laterally by as much as 0.75 in, and while different piping arrangements can be expected to respond differently, these results strongly suggest such behavior in field installations. All of this motion ceased, and the vapor disappeared—as did all evidence of backflow—with the backflow recirculator inserted into the pump suction branch. Smooth operation resulted at all flowrates, and it is concluded that such a device, properly designed, redeems the high suction specific speed end suction process pump, making it mechanically reliable and durable at all flowrates. Thus, the original goal of a lighter, faster, lower-$NPSH_R$ pump can now be safely realized and the trend in this direction re-established without penalty to users.

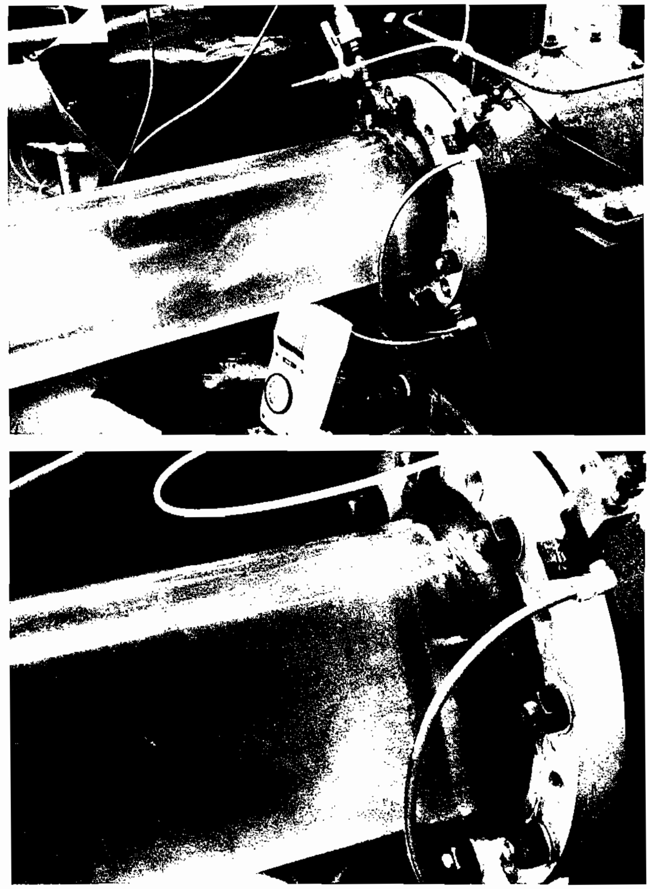


Figure 12. Transparent Suction Pipe. a) Without Backflow Recirculator at 1000 GPM. b) With Backflow Recirculator at 300 GPM.

This improvement in mechanical response resulted entirely from treatment of the backflow problem at the impeller inlet. Extension of such improvement to higher energy applications and to neutrally balanced single-stage double suction pumps may require attention to the design remedies [6] offered to eliminate the adverse effects of discharge recirculation as well.

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