

HIGH FLOW PROTECTION FOR VARIABLE SPEED PUMPS

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ABSTRACT

Low flow operation is a well-known cause of centrifugal pump failures and thus it is a custom to have low flow protection. On the other hand, protecting pumps from excessively high flow is usually not considered during project design due to natural process flow limitations or the reliance on motor overload protection.

This paper presents detailed analysis of major failures that occurred in three 27,000 HP variable speed water injection pumps in which high flow operation was a major contributor to the failures. It includes the evaluation of the impeller hydraulics and material in addition to impeller modal analysis.

INTRODUCTION

Rotating equipment normally have multiple protection systems to alert the operator or to shutdown the equipment when needed. For a pump, the typical protection system includes vibration, bearing temperatures, low suction pressure and low flow in addition to the other protection systems associated with the pump auxiliary systems such as the lube oil and sealing systems. It is unusual to have a high flow protection for a pump since there are typically many factors that prevent high flow operation such as driver power limitation, the pump inability to match the system pressure requirement at high flows, the limitation of the pump suction pressure at high flows, or the pump vibration at high flows.

For a fixed speed pump, the flow is usually controlled by a discharge control valve which easily can be used to prevent high flow operation. Variable speed pumps do not typically have control valves on their discharge line as the flow is controlled by varying the pump speed, which is a more economical way of flow control. When using a variable speed pump with no control valve, special care must be given to ensure that the operating conditions are correctly estimated for proper pump selection. Otherwise, the pump may run at high flow if the back pressure in the discharge line is found to be lower than what the pump was originally designed for even if the speed is lowered.

To illustrate the above point, this article will look at the failures of several large water injection pumps that were installed in a Saudi Aramco plant between 2000 and 2009. These are three stage horizontal pumps where the first stage has a double suction impeller and the remaining stages are of single suction design. The pumps are designed in accordance with API-610 and are driven by gas turbines which allow the pumps to vary their speed to accommodate the different flow requirements. The pumps were designed to have a normal flow of 12,000 GPM at rated speed. Each of the pumps has low flow protection and is also equipped with other protection systems including low suction pressure, vibration and temperature protection. In the following sections, the main failure modes and the different parts of the failure investigation are addressed.

FAILURE MODES

During the course of the investigation, the failure modes for the four failed rotors were evaluated (one rotor for G-1001 & G-3001 and two rotors for G-2001) and were found to be similar in nature but different in severity. The main failure modes were:

- Cavitation damage on the vanes back side (pressure side) of all the second stage impellers and some of the third stage impellers as shown in figure 1. It should be highlighted that no cavitation damage was found on the first stage which is a double suction impeller.
- Cracks on some of the second and third stage impellers vane inlets and vane outlets and breakage of some of the second stage impellers vane inlets as shown in figures 2, 3 and 4.
- Cracks on some of the second stage impeller shrouds and breakage of some of the impeller shrouds as shown in figures 5 and 6.

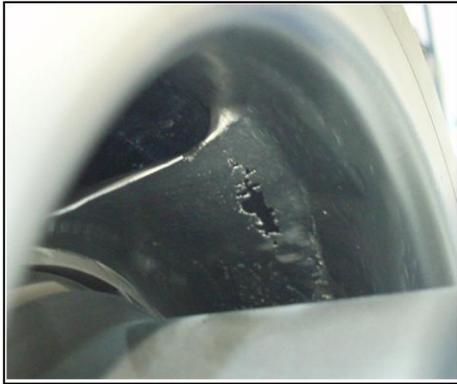


Figure 1: G-3001 2nd stage impeller cavitation



Figure 5: G-2001 new rotor 2nd stage shroud breakage



Figure 2: G-2001 old rotor 2nd stage vane inlet

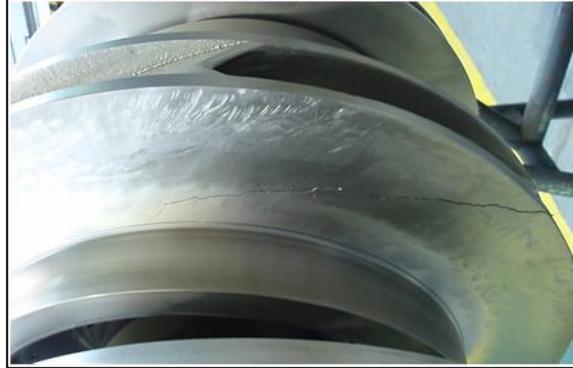


Figure 6: G-3001 2nd stage shroud crack



Figure 3: G-2001 old rotor 3rd stage vane inlet



Figure 4: G-1001 2nd stage vane outlet

FIELD ACTUAL OPERATING CONDITIONS

During the course of this failure investigation, the operating parameters of these variable speed pumps were reviewed and compared to the design parameters. Table 1 shows the summary of pump actual operating conditions in relation to the Best Efficiency Point (BEP) flowrate. Figure 7 shows the time averaged actual operating condition for the worst failure which happened on G-2001 new rotor after only 97 days of operation.

Pump	Run time (days)	Averaged Flow as % of BEP	Percentage of Operated Time as a Percentage of BEP Flow Rate				
			<120	121-140	141-150	151-160	161-180
G-1001	699	124	26	56	13	2	3
G-2001 old rotor	623	116	35	56	9	0	0
G-2001 new rotor	97	148	11	0	0	79	10
G-3001	625	122	17	79	4	0	0

Table 1: Summary of Pumps Actual Operating Conditions

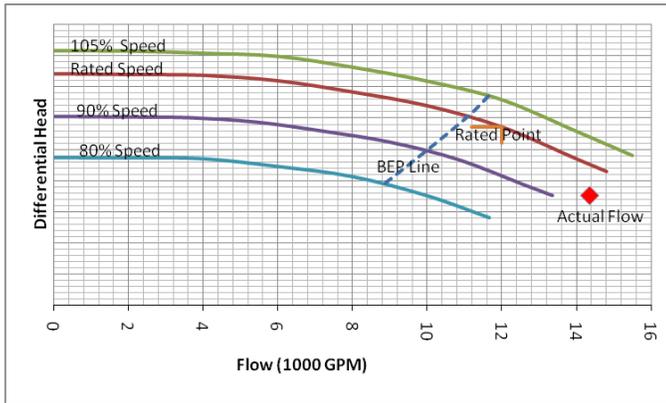


Figure 7: Operating Condition for G-2001 New Rotor

The table shows that the pumps were running for a large portion of their operating life at high flow rates ranging between 120% to 180% of BEP flow rate. This high flow operation is mainly due to a reduction in the downstream pressure requirements.

To illustrate the above point, let's take a look at the operating parameters for G-2001 new rotor shown in figure 7. The pump was designed to have a normal flow of 12,000 GPM at rated speed and to be capable of achieving 15,000 GPM by increasing its speed to 105%. The actual time averaged flow for this pump was 14350 GPM, which was achieved while operating at 87% of rated speed. Due to the low process back pressure, the pump was able to achieve high flow than rated while operating at lower than the rated speed. This actual flow is approximately 148% of the BEP flow which falls outside the allowable operating region for this pump at that reduced speed.

The pump allowable operating region is typically limited to a maximum of 120% of BEP flow. Operating at higher flow rates than this limit, such as in this case, is not recommended since it will reduce the life of the pump significantly in addition to the deterioration of the pump performance. The high flow operation for these pumps is believed to be the main contributing factor to the four failures. This conclusion is supported by the operating data, which showed that shorter life and higher damage was found on the pumps that were operated for longer portion of their life at high flow rates.

IMPELLER MATERIAL FAILURE ANALYSIS

The failed impellers received a detailed failure analysis, which showed that the material meets the chemical and mechanical properties of the impeller material which is duplex stainless steel ASTM A 890 grad 2A.

The material failure analysis was conducted on the two rotors that were used in G-2001. The first rotor lasted for 623 days of operation while the second rotor lasted for only 97 days. The analysis showed that many of the impeller cracks were caused by a combination of stress corrosion and fatigue cracking (see figure 8), which are believed to be influenced by high dynamic loading. It also showed some impeller casting and machining defects such as casting weld repairs and

reduction of vane wall thickness by grinding. These deficiencies have contributed to these failures.

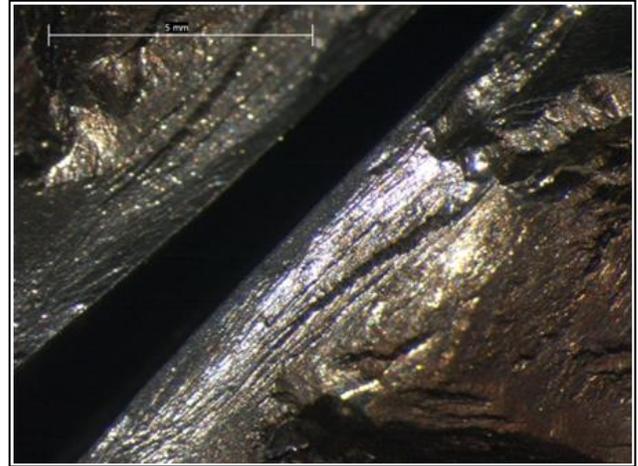


Figure 8: Beach marks on the fracture surface indicative of fatigue

IMPELLER MODAL ANALYSIS

A modal analysis was conducted to determine the impeller natural frequencies to investigate its effect on these failures especially the cracking and brakeage of the second stage shroud. The analysis was conducted using impact testing on the impeller vanes and shrouds. The natural frequencies interference diagram is shown in figure 9.

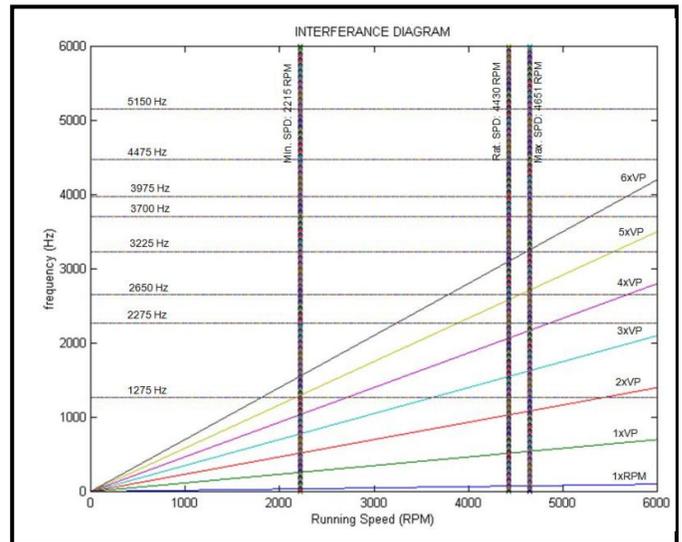


Figure 9: 2nd and 3rd Stage Impeller Interference Diagram

The interference diagram shows that there are many possible high frequency excitations that can meet the impeller natural frequencies due to the wide pump operating speeds. The diagram shows that the nearest excitation mode is much higher than the vane pass frequency. Although there are possible excitations at some of the natural frequencies, these frequencies are considered very high, which reduces the possibility for their contribution to the second stage impeller failures.

Due to the severe cavitation that was found on all the second stage impellers, a hydraulic study was conducted to determine the best cavitation point (shockless entrance capacity) for the second stage impeller. In addition, similar study was conducted on the first stage impeller which did not suffer from cavitation. Typically, cavitation in multistage pumps happens at the first stage impeller and not at the second stage as what happened in these four failures. This was the reason for including the first stage impeller in the hydraulic study.

The analysis was done by a program, which uses Vlaming empirical equations to determine the shockless entrance capacity and the 40,000 hours Net Positive Suction Head required (NPSHR). Using the measured impellers geometry, the shockless entrance capacity was calculated to be 14,550 GPM with an NPSHR of 271 ft for the first stage impeller. For the second stage impeller, the shockless entrance capacity was calculated to be 10,570 GPM with an NPSHR of 347 ft. Figure 10 and 11 shows the predicted 40,000 hours NPSHR curve for the first and second stage impellers.

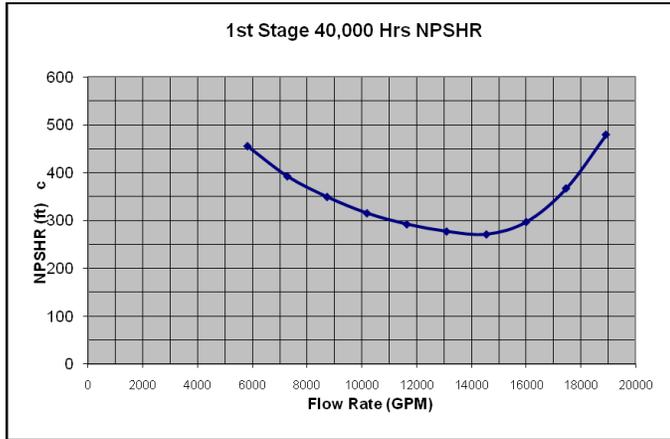


Figure 10: Predicted First Stage Impeller 40,000 Hours NPSHR Curves

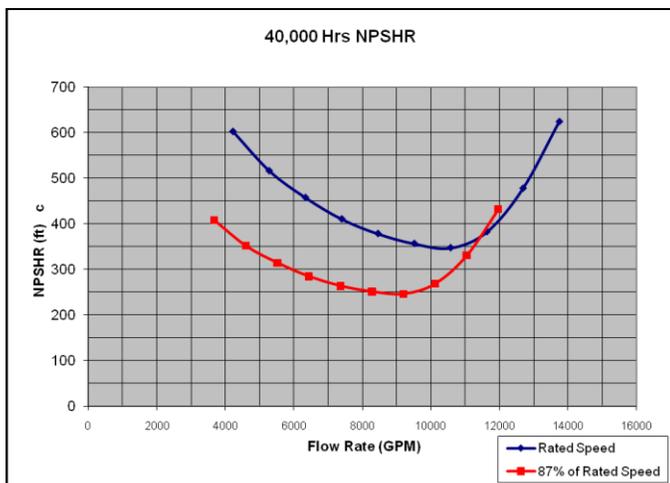


Figure 11: Predicted Second Stage Impeller 40,000 Hours

When looking at figure-10 and figure-11, it becomes clear why the high flow operation caused the cavitation damage on the second stage and not at the first stage. As was stated previously, the subject pumps were operated at high flow rates exceeding their 12,000 GPM rated flow. Figure-10 shows that the NPSHR for the first stage impeller reduces as the flow exceeds the rated flow until it reaches the shockless entrances capacity which is 14,550 GPM. For the second stage impeller, the shockless entrance capacity (10,570 GPM) is low, which causes exponential increase in the NPSHR as the flow increases above the 12,000 GPM, as shown in figure 11.

If the pumps were operated at their design condition then the first stage impeller will see a Net Positive Suction Head Available (NPSHA) of more than 900 ft while it requires an NPSHR of approximately 270 ft. Even at higher flow rates, the NPSHA is higher than the NPSHR. Having a higher NPSHA than NPSHR will help in preventing many cavitation modes. The second stage impeller has an NPSHA of approximately 2400 ft at the rated flow and speed due to the added head by the first stage impeller. During normal operation, the second stage impeller should not have cavitation even if the pump was to operate at 120% of rated design flow rate since the NPSHA significantly exceeds the NPSHR.

As the NPSHA for the second stage is a result of summing the pump NPSHA and the first stage differential pressure, the high flow operation of these pumps while running the pumps at lower speeds than rated caused a significant reduction in the NPSHA for the second stage. This reduction is mainly due to the decrease in the pump NPSHA as the flow increases, in addition to the reduction of the first stage differential head due to high flow and low speed operation which also cause an increase in the NPSHR. To illustrate the last point, the NPSHR requirement for G-2001 new rotor actual operating condition, which was running at 14,400 GPM and 87% of rated speed, was added in figure 11. As can be seen in this figure, for any high flow operation above 11,500 GPM, the NPSHR requirement for the 87% rated speed curve is higher than that for rated speed. For G-2001 actual operating condition, the NPSHR requirement is much higher than 700 FT. This reduction in NPSHA and increase in NPSHR is believed to be the reason for the observed cavitation damage on the second stage impellers for all the four failed rotors.

CORRECTIVE MEASURES

Based on the failure investigation results for the subject pumps, the two main causes for the failures are the high flow operation of the pumps and impellers manufacturing deficiencies. To resolve these problems, the following corrective actions were implemented:

- A monitoring system was added to the Distributed Control System (DCS) to show the pump operator whether the pump is operating within or outside its allowable operating range at the running speed.

- The impeller material was upgraded from duplex to super duplex stainless steel (ASTM A 890 Grade 5A), which has higher fatigue and corrosion resistance than the original duplex stainless steel material.
- Implemented better control on the new impellers casting and casting repair quality.

CONCLUSIONS

The reduction of the process back pressure and the reliance only on varying the pump speed for controlling the flow rate allowed the pumps to operate at high flow rates while running at low speeds. This high flow operation (up to 180% of BEP) was concluded to be the main cause of the pump failures in addition to some impellers manufacturing deficiencies. As pumps with variable speed drive do not typically have control valves on their discharge line, it becomes critical to know the process pressure requirements and any possible changes in them. In cases where the process pressure and flow requirements may change significantly, a discharge control valve and a high flow protection should be considered during the project development.

NOMENCLATURE

NPSHA = Net Positive Suction Head Available

NPSHR = Net Positive Suction Head Required

BEP = Best Efficiency Point

DCS = Distributed Control System

(L²)

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