Failure of the HP Boiler Feed Water Pump on an Ammonia Plant

A catastrophic failure occurred on the main High-Pressure Boiler Feed Water Pump Train on an Ammonia Plant. The direct cause of the failure was flow reversal from the Boiler Feed Water system and/or the Steam Drum, to the turbine driven High Pressure Boiler Feed Water Pump. While the failure of the pump is the focus of the paper, the identified causal factors involve failures of several auxiliary operating systems, which are also discussed. The paper presents the sequence of events and associated root causes that resulted in the failure of the pump as well as recommendations to prevent a re-occurrence.

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Introduction

The high-pressure boiler feed water pump train of an ammonia plant experienced a failure on 25th September 2015. The failure occurred in a 1,850 MTPD (2,040 STPD) ammonia plant. This paper describes the failure that occurred, the investigations and the lessons learnt.

System Configuration

As shown in Figure 1, boiler feed water (BFW), leaving the deaerator, is directed to the steam drum for the purpose of high pressure (HP) steam generation using two (2) identical boiler feed water pumps. The main HP BFW pump is driven by an API 611 condensing turbine which utilizes medium pressure (MP) steam, and the standby HP BFW pump is driven by an electric motor, which operates at the 4,160V level.

The HP BFW pumps are API 610 centrifugal multi-stage, opposed impeller, horizontally split volute pumps. Automatic recirculation (ARC) valves in the discharge piping of the pumps are designed to prevent liquid vaporization in the casing and pump damage, by ensuring positive flow. The ARC valves on each pump also act as check valves to prevent reverse flow. The pumps are also furnished with minimum flow lines from the ARC valves that converge to a single line back to the deaerator.

The turbine utilizes a mechanical hydraulic speed governor for speed control. The governor controls turbine speed by adjusting the quantity of MP steam supplied to the turbine. The governor has an output between 0 – 100% representing minimum (3,026 rpm) to maximum (3,728 rpm) governor speed settings. The DCS boiler feed water flow controller, acts in two ways. It controls the speed set point of the turbine and also the output to the control valve on the discharge of the standby pump.

Deaerated boiler feed water from the BFW pumps is fed to the steam drum via the three paths each with a control valve and a tilting disc check valve. Two of the paths include BFW pre-heating and steam generating shell and tube heat exchangers.
**Incident Description**

On 25th September 2015, the plant experienced a spurious activation of the process air compressor shutdown signal resulting in the synthesis gas compressor and downstream steam generating equipment to shut down as per the plant’s design interlock logic. Despite the shutdown signal that initiated these events, the process air compressor remained online which resulted in a deficit MP steam mass balance since the process air compressor is one of the largest consumers of MP steam.

The ensuing steam system upset resulted in the following sequence of events over a 40 minute period: a high steam drum level, low BFW flow to the steam drum, the automatic start of the standby HP BFW pump on low flow, no BFW flow to the steam drum and eventually low steam drum level. These events culminated in the activation of the steam drum low level interlock which initiated a shutdown of the primary reformer and downstream systems (total plant trip) as per design.

With the main process systems offline a plant operator was dispatched to initiate shutdown of the main and standby BFW pumps which were still online. The operator utilized the mechanical trip lever on the turbine driver for the main HP BFW pump and initially observed a speed reduction. Within seconds, the operator heard an accelerating sound and observed the turbine speed to be increasing past 5200 rpm (well over the maximum of 3728 rpm). An instruction was given by the shift supervisor to evacuate the area immediately and within seconds the catastrophic failure of the main BFW pump and turbine unit occurred. Thankfully, only minor injury was sustained by an operator which was treated by on site first aid. Figure 2 shows the resulting damage.
Actions Before the BFW Pump Failure

During the steam upset with the trip of the synthesis gas compressor train, the steam drum level was rising. The plant operators attempted to control the steam drum level by reducing the BFW flow from the main pump. To reduce the BFW flow, the speed of the BFW turbine was reduced by using the governor control to bring the turbine speed to minimum governor. With the level still rising, the flow through the steam inlet valve for the turbine driver of the main BFW pump was reduced by a plant operator partially closing the valve. This action only momentarily dropped the main pump’s speed from 3500rpm to 1900rpm. The pump’s speed returned to 3500rpm while the closing of the valve was still in progress. It is believed that the turbine’s mechanical hydraulic governor compensated for the reduction in steam flow by opening to allow more flow as evidenced by a return to 3500rpm. The corresponding drop in discharge flow also had the unintended side effect of auto-starting the standby pump against its closed discharge control valve. The steam inlet valve was then reopened from the partially closed position.

As the steam system was now being managed, the steam drum level began to fall. The plant operators at this time did not observe any flow being indicated on the outlet of the pumps. Without isolating the main pump, the discharge control
valve on the standby pump was opened fully, but still no flow was observed. The steam drum level continued to fall until the eventual plant trip due to low-low steam drum level.

Findings

Inspection Findings

The steam turbine was a total wreck with the turbine casing disintegrated and the rotor components dispersed across the site. The damage noted on the pump skid and foundation indicated clockwise rotation from the drive end of the train, indicating that the machine was experiencing reverse rotation as it usually operates in a counterclockwise direction from this frame of reference. This damage, combined with the dispersion of parts shown in Figure 2, indicated reverse rotation of the unit. Additionally, more evidence of reverse rotation was observed upon inspection of the pump’s internal components. The shaft was bent toward the non-drive end as shown in Figure 2D with significant deterioration to the casing wear rings, inter stage rings, impeller rings, impeller and bushings at the discharge end of the pump, some of which is shown in Figure 3A.

Inspection of the ARC valve on the main pump revealed the presence of debris lodged between and around the main check valve and lower body seating surfaces, likely preventing the check valve from closing fully, shown in Figures 3B & 3C. The vortex plug on the recirculating/bypass end of the ARC was also found to be stuck in the valve body, possibly restricting bypass flow. The debris was confirmed to be a section of the pump’s casing inter-stage ring. The pump suction strainer, shown in Figure 3D, was found to be distorted in a collapsed inverted condition which suggests flow reversal and was found to be fouled on both the upstream and downstream sides with the pump rotor’s wear ring material also found in the ARC valve.

Of the three check valves on the lines to the Steam Drum, two (CH-V1 & CH-V3) were found stuck in the open position with debris lodged in the seating area – shown in Figures 3E & 3F. The debris that was found in one of the valves was identified to be SMAW welding electrodes. On the other valve, twisted tape turbulators from within the tubes of a BFW preheater/steam generator were noted between the disc and seating area.

![Figure 3. A – Main HP BFW pump wear rings found damaged; B & C – Wear ring debris lodged in main BFW pump ARC check valve seat; D – Main HP BFW pump strainer found fouled on both sides; E & F Check valves found open](image-url)
Discussion

Analysis of the Sequence of Events

The trip signal generated by the air compressor and resulting steam upset led to the shutdown of the synthesis gas compressor. The output of the HP BFW pump began falling automatically to match the steam generation by the plant’s logic, causing the steam drum level to begin falling. In an effort to manage the steam drum level, the combined HP BFW flow controller was placed in manual, and the control valves to the steam drum were operated in an effort to maintain a suitable level in the drum. This had the desired effect of causing the steam drum level to begin rising.

The steam drum eventually reached 98%, at which point the combined HP BFW flow controller was reduced to 0%, causing the turbine driven pump to reduce to minimum governor speed. In an effort to further reduce the output of the pump, an attempt was made to partially close the HP BFW pump turbine’s steam inlet valve, causing only a momentary drop in speed and flow. As a result, the standby HP BFW pump automatically started. It is suspected at this instant, damage was done to the main pump’s wear rings, causing fragments to lodge in the seat of the ARC valve.

The standby pump’s flow controller was in manual at 0%. This gives the operator finite control of the BFW output such that when this pump starts automatically, there are no unwieldy swings in the BFW output flow. Based on this, there were three pathways available to the fluid being pumped – the common recirculation line, the common discharge line, and most importantly, because of its reduced output, back to the main pump. With both pumps running on the common recirculation line, which was designed for the flow of one pump (via RO-1), both pumps began to gradually deteriorate as the minimum flow requirements were not being met. Both pumps were also flowing to the common discharge, and with a compromised ARC valve on the main pump, the output of the standby pump is expected to have caused continuing damage to the main pump.

The HP BFW flow dropped slowly to 0 klb/h, and in an attempt to restore flow, the standby pump’s discharge controller was set to 100%. This caused a momentary increase in flow, then a similar gradual decrease as this pump was also damaged. The steam drum level fell to the point of low-low alarm, tripping the plant. The turbine was then manually tripped, resulting in the reverse flow of a high energy BFW and steam two phase medium that caused the pump and turbine to rapidly increase in rotational velocity in the reverse direction. This reverse rotation likely caused the turbine discs to loosen, vibrate, rub and separate from the shaft and ultimately causing the disintegration of the turbine.

Figure 4 shows the eventual flow path from the steam drum through the pump and to the deaerator.
Failure of the ARC

Inspection of the ARC valve revealed debris trapped between the main check valve and lower body seating faces. Results from positive material identification checks of this debris were similar to that of the pump wear rings. The gap created was 7mm of the 48mm gap for the full valve travel. The equivalent cross sectional flow area of this 7 mm gap is estimated to be that of a 3” orifice which with the expected pressure drop in the reverse condition would allow for more than 400 klb/h flow.

Loss of Forward Flow

The throttling of the main BFW pump turbine’s steam inlet valve resulted in reduced capacity of the pump, to the point where it could not deliver a forward flow into the header. From this point, the pump’s output was directed to the deaerator via the recirculation line. In parallel, when the standby BFW pump automatically started with the discharge control valve (FCV) manually closed, the flow from this pump would also have been directed to the deaerator through the same recirculation line. The recirculation lines for both the main and standby pumps join and pass the flow through a restriction orifice, which was found to be sized for only a single pump’s recirculation flow as confirmed by the plant designers after the incident. As a direct result of this, it is suspected that for a period of time, both pumps operated below the minimum required flow. In the post incident analysis, it was found that the current practice of the plant designers dictates separate recirculation lines for each pump.
It is known that subjecting a pump to low flow conditions will result in increased temperatures in the pump due to recirculation, opening of the clearances due to rubbing of the wear rings and therefore, degraded performance. With the installed thermoplastic casing wear rings of this pump, rubbing and damage is expected to occur more easily than for pumps with metal wear rings due to the reduced clearance.

Severe damage to the thermoplastic wear rings was noted in the inspection findings of the rotor. This type of wear ring allows for smaller pump clearances and greater efficiency in operation. However, notably, API 610 refers primarily to metal wear rings but does not explicitly prohibit the use of non-metallic wear components.

It is also known that, prior to the temporary partial closing of the turbine’s steam inlet valve, the pump was online with no observed performance issues based on the boiler feed water flow rate, but, subsequent to this, the pump was unable to deliver any forward flow. Therefore, it is certain that a change in pump performance occurred over that period and it is likely due to the low flow conditions created by the temporary partial closing of the steam inlet valve and the restrictions created by only one recirculation line, when the bypass pump started.

It is notable that after this time, the discharge temperature indication for the BFW pumps did in fact reach its maximum saturation point. However, plant design does not call for an alarm for this condition, and it remained unrecognized until post incident analysis.

**Reverse Rotation and Flow**

Inspection findings and the observation of the turbine deceleration followed by reacceleration after being tripped suggest that reverse rotation of the BFW pump and turbine occurred.

The maximum speed that can be attained by the BFW pump with reverse flow of service fluid was calculated. The suction specific speed for the BFW pump was calculated from the normal operating point on the pump datasheet and was applied to the Hydraulic Institute’s curve\(^1\) for reverse runaway speed vs specific speed – shown in Figure 5.

![Figure 5. Hydraulic Institute’s curve](image)

A reverse runaway speed of 1.18 times normal operating speed or 4,095 rpm was estimated. A literature review for reverse runaway speeds in centrifugal pumps suggest that 1.25 times normal operating speed is a typical finding\(^2,3,4\). For the HP BFW pump, the maximum reverse runaway speed with service fluid (BFW) was therefore taken as 4,450 rpm.

The observation of the plant operator was that a speed of at least 5,200 rpm was achieved prior to the failure. This observation when compared to the maximum expected runaway speed of 4,450 rpm indicates that the medium involved in the reverse flow must have been not only BFW but a combination of steam and BFW which are available in the downstream steam generating heat exchangers and steam drum.
Additional evidence supporting reverse flow of BFW and steam from the downstream equipment include:

a. A mass balance of the deaerator in which a 107 klb/h additional input was noted. A rise in deaerator pressure was also observed to the point of transmitter saturation.

b. It was found that -4.2 klb/h BFW flow was recorded for the no flow conditions observed. Post incident field checks of the flow indicator reveal that the actual value transmitted was reproduced by simulating reverse flow and this value is in fact the lower saturation limit of the instrument.

c. Even though it is known that low flow conditions will cause increased pump temperatures, reverse flow was the more likely cause of the temperature excursion observed on the BFW transmitter. This is supported by the rate of the temperature increase as well as the location of the damage on the pump shaft.

It should be noted that the control valves on the lines with the two compromised check valves (CV1 & CV3) were closed at the time. However, the control valves have shut off classifications of ANSI Class II and III, which allow some leakage across the valves. As a result of this, the quantity of BFW/steam in reverse flow from the steam drum is not certain. Notwithstanding this fact, the volume of BFW/steam available in the steam generating heat exchangers is significant enough to reverse rotate the pump train to failure.\(^5\)

### Pump and Turbine Failure

The tripping of the turbine removed the last of the resistance to reverse rotation and allowed full reverse flow of the steam and BFW media from the downstream equipment and piping, back through the pump. This directly resulted in the rapid acceleration and unstoppable gross overspeed of the turbine in the reverse direction and consequent catastrophic failure.

Based on observations, both the pump and turbine appear to have failed completely at around the same time. However, the damage to the turbine was of greater magnitude. After considering several possible scenarios, the likely explanation is that the machine speed in reverse was high enough for the turbine discs to become loose on the shaft.

Calculations show that this is possible at as low as 130% of the normal forward operating speed, which was certainly exceeded, as the failure occurred at approximately 146% of the operating speed. Operating with loose discs would lead to significant and unpredictable vibration due to rubbing between the rotor and the stators, completely freeing the discs, leading to the turbine failure. Once the turbine failed, the pump shaft would be left running in reverse at this speed with an overhung coupling that would likely quickly bend the shaft and cause the pump to seize.

Examination of the pump design documentation shows that design for reverse rotation was considered. However, the recommended factor did not consider the specific pump configuration of a double impeller suction as well as actual suction and discharge pressures. Despite this, reverse flow design does not consider the conditions or the two phase medium that this pump was exposed to. It is also notable that the option on the turbine datasheet to indicate that the machine was designed for reverse rotation was not selected. It is however not a normal practice to select a turbine designed for reverse rotation. Additionally, the coupling was not designed to fail in case of reverse rotation.
Analysis and Cause

Loss of forward flow and failure of the ARC valve allowed a reversal of a high energy steam/BFW flow and the subsequent failure of the pump and turbine. The loss of forward flow was caused by the temporary closure of the turbine’s steam inlet valve and it was exacerbated by the failed steam drum inlet check valves. Testing of the ARC was not historically done unless issues were identified. The root causes of this incident were determined to be:

1. Inadequate understanding of the potential consequences of reverse flow in the Process Hazard Analysis.
2. Inadequate safeguards to prevent reverse flow in the HP BFW system.
3. Inadequate management of the Steam Drum level.

1. Inadequate understanding of the potential consequences of reverse flow in the Process Hazard Analysis.

The design of the HP boiler feed water system relies on a single engineering control, the ARC valve, to safeguard against reverse flow through the pump. The check valves at the inlets to the steam drum cannot be considered as a second dissimilar check valve in series due to the inventory of BFW and steam available between the ARC and these valves.

The PHA performed at the detailed engineering phase for this plant identified check valve failure as a cause of reverse flow but failed to recognize the potential consequence and its severity; as the magnitude of the resulting reverse flow was not evaluated. As a result, additional safeguards were not identified.

The 2015 revalidation PHA addressed the scenario of a failed check valve and reverse pump rotation, but only for the case where the pump is started while rotating in the reverse direction. It did not recognize that reverse rotation on its own could have resulted in the failure of the turbine and was not identified as a consequence. Hence, additional safeguards were again not identified.

2. Inadequate safeguards to prevent reverse flow in the HP BFW system.

Adjusting of the steam inlet block valve should not have been performed as it is against operating procedure to safeguard the boiler feed water pump. Review of the SOP shows that the procedure was also reliant on the functionality of the single safeguard, the ARC valve as a non-return valve.

The minimum governor speed is set at the lower safe operating limit for the pump based on a minimum flow. As such, the only method for reducing the speed lower than this is by partially closing the turbine’s steam inlet valve. However, closing the inlet steam valve decreases the pump’s ability to achieve minimum flow and can result in pump failure. The use of the turbine’s steam inlet valve to control the output of the main BFW pump below the minimum governor speed was not warned against in the SOP. This oversight was also subsequently clarified.

3. Inadequate management of the Steam Drum level.

The SOP guiding actions following a trip of the air compressor identifies specific actions for control of steam drum level. Further procedure review identified additional actions that would have allowed effective management of the steam drum level. However, none of these were documented in the SOP. It was concluded that the SOP could have provided enhanced guidance for this emergency situation. The SOP was subsequently amended to include the additional steps.
Recommendations

To prevent a reoccurrence, the following recommendations were made:

1) Review and amend the PHA of the HP BFW System:
   Although the PHA did identify the possibility of reverse flow, the consequence and severity were underestimated. Because of the inventory between the discharge of the pumps and the steam drum inlet check valves, the severity is elevated, and the consequence of turbine failure was not considered.

   The PHA was reviewed, and it was recommended that separate recirculation lines as well as automatic shut-off valves be installed.

2) Amend SOP’s:
   Closing the pump discharge prior to shutting down the turbine or motor is now included in the SOP. This will prevent flow from one pump to the other through the common discharge line. The closing of these valves will be done manually until the automatic shut-off control valves are installed.

   Clarification on the actions the operator is allowed to take with the steam turbine inlet valve was included in the SOP. This information will prevent attempts to use the turbine’s steam inlet valve to lower the machine’s speed below the speed required for the pump’s minimum flow specification.

   Emergency procedures have been amended to include all identified methods of managing the steam drum level. This information will allow informed, safe actions that can be executed based on current plant conditions.

3) Implementation of automatic shut-off valves:
   Automatic shut-off valves are to be installed on the pump discharge. These valves, when activated, will immediately close on low pump discharge pressure. After pump start-up, these valves will positively isolate the discharge of the pump should the discharge pressure fall below a predetermined value.

4) Installation of separate recirculation lines for each pump:
   A separate recirculation line is to be installed for each pump. This will allow both pumps to establish proper minimum flows when both are online at the same time. This can occur under normal circumstances such as plant startup (when the motor driven standby pump is used before steam can run the turbine), and when switching pumps to conduct routine PM’s and other maintenance activities.

5) The ARC's and several other critical check valves have now been identified for periodic testing.

Conclusion

Consequence of failure of a high-pressure boiler feed water pump train can be significant in certain situations. Safeguards for existing systems and procedures governing pump operations should be evaluated in light of this incident.

References