DESIGN, TESTING, COMMISSIONING AND OPERATING EXPERIENCE OF A 2000HP HYDRAULIC TURBOCHARGER FOR ACID GAS RECOVERY PROCESS

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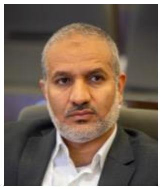

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ABSTRACT

Four sets of hydraulic turbocharger based energy recovery systems (2000 hp each) were successfully designed, manufactured, tested and commissioned for Acid Gas Recovery plants (AGR) in Saudi Arabia. The hydraulic turbocharger consists of a liquid phase turbine runner and a pump impeller mounted in a back-to-back configuration on a common rigid shaft, in a single seal-less casing. The shaft is supported by process lubricated hydrodynamic radial and axial (dual-acting) bearings of non-metallic construction. The turbine side extracts wasted pressure energy from the rich amine and along with a series of throttle, auxiliary and bypass valves, controls the level in the absorber. The pump side boosts the lean amine solvent to the absorber. Such a design provides major advantages compared to a conventional configuration of a reverse running pump-as-turbine coupled to a pump. These advantages include a compact single stage hydraulic design (as opposed to a multi-stage design for a reverse running pump), smaller footprint, higher efficiency, absence of mechanical seals & associated support system, absence of lubrication oil system as well as simpler control.

Several complex design and testing challenges were overcome through development of novel design features and incorporation of provisions in a highly specialized test loop. Some of the unique experiences are shared in the paper. The seal-less design of the turbocharger with no exposed rotating or bearing parts presented testing challenges such as determination of shaft power, individual pump side and turbine side performance and efficiencies; measurement and validation of axial thrust & axial position; as well as monitoring & control of bearing performance parameters. The test loop was engineered to operate the hydraulic turbocharger through its testable operating range with provisions to measure, control and monitor all the needed parameters. Testing was performed using water with viscosity 2.5 times lower than the operating fluid. This implied suitable modification to the design, monitoring and controlling of process-lubricated bearings along with the internal system during testing to account for the change in viscosity. Data was evaluated to conduct root cause analysis of apparent bearing failures such as rub marks and to eliminate root causes by improving the manufacturing method to reduce misalignment & also using non-metallic bearings.

Four identical units were built and completed successful Factory Acceptance Tests (FAT) in early 2018. The turbochargers achieved & surpassed the pump side rated requirements of 2148ft of head at 2820 gpm, at a rated speed of 8000 rpm, as part of the FAT. All the units were installed & commissioned in early 2020 by the End User. Each unit is part of 4x50 configuration – which includes the turbocharger and three circulation pumps running in parallel. The paper will provide details about the commissioning effort of the turbocharger units in the field, and their operating experience over a period of one year.

INTRODUCTION

Hydraulic turbochargers are widely used as energy recovery in Sea Water Reverse Osmosis (SWRO) industry. As noted by Kadaj (2018) turbochargers were first introduced in the 1980's and became widely adopted in the 1990's. These turbochargers can have efficiencies as high as 80%, and are used to recover wasted pressure energy from the brine concentrate water rejected by the reverse osmosis membranes using a turbine. The turbine drives a pump impeller mounted on a common shaft to raise the pressure of the fresh water to that required by the RO membranes. The turbine flowrate to pump flowrate is typically 0.6:1 for these applications. A booster pump in series raises 40-60% of the pressure while the turbocharger pump raises the rest. Due to this high pump suction pressure, the rotating assembly experiences uni-directional thrust, and these turbochargers therefore have a single hydrostatic bearing. The shaft speed can range from about 8000 rpm for large turbochargers (8000 gpm) to about 25,000 rpm for the small turbochargers (250 gpm). Since the process fluid is non-toxic, the turbochargers typically run un-spared, without any condition monitoring or control system. They are also not designed to be compliant with API 610.

As compared to this, the AGR application has to handle toxic substances such as H_2S and CO_2, in a refinery environment and therefore the design requirements for this turbocharger are more stringent. The End User based on their experience with some catastrophic failures of hydraulic turbochargers made by the OEM's competitor in SWRO application came up with a list of design recommendations for the

OEM to consider.

The OEM had experience with a successful installation of a 500 hp (370 kW) hydraulic turbocharger in a West Texas AGR plant in 2008, which ran without any maintenance for more than 10 years. The design was used as a starting point for the 2000 hp (1490 kW) Fadhili Gas Plan (FGP) application, however based on End User's technical requirement, the OEM decided to completely redesign the turbocharger energy recovery system as part of a technology project. This involved kicking-off the development of a proof-of-concept 500 hp (370 kW), 2:1 flow ratio turbocharger with dual-acting hydrodynamic bearings in 4Q2015. Lab testing of the turbocharger helped develop the high energy, low specific speed hydraulic design, bearing and system-level design and analysis tools required to design a turbocharger for FGP. It also provided a test vehicle to try out different lubrication flow paths, bearing materials, condition monitoring techniques and turbocharger test loop design & methods that were finally implemented in the development of the FGP turbochargers. The design for the FGP turbocharger commenced in 2Q2016 and the first Hasbah (HSBH) unit turbocharger passed factory acceptance test in 1Q2018. The turbocharger based recovery system SKID's were commissioned at the Middle East AGR plant in 1Q2020 and have been running for more than one year without any issues.

The paper will provide a background of the different energy recovery solutions that have been implemented in the past, their drawbacks and the reasons for why a hydraulic turbocharger based energy recovery system was selected and implemented at Fadhili Gas Plant (FGP). The FGP has five AGR gas processing trains, four of them processing gas from offshore HSBH field, while one plant processing has from on-shore Khursaniyah (KRSN) field. Each train has one turbocharger based energy recovery system – the four HSBH trains have identical 2000 hp (1490 kW) turbochargers and the fifth KRSN train has a 1000 hp (746 kW) turbocharger. This paper will provide some of the unique design details, the commissioning & operating experience of the HSBH Unit One turbocharger only.

BACKGROUND

Natural gas from the wells is processed to remove acid gases such as carbon dioxide and hydrogen sulfide. Typically, the acid gas removal (AGR) process uses high pressure amine solvent to absorb acid gases in the contactor column. The acid gases absorbed in amine are then released in the regenerator column after the pressure of amine is decreased. In all pre-Fadhili Gas Plants of End User, the means of pressure reduction of high pressure amine is pressure-let down valves aka throttle valves. After the acid gases are collected in the regenerator, the amine is re-pressurized through high pressure amine pumps and routed back to the absorber to repeat the cycle. Almost 90% of the hydraulic energy input by High Pressure Amine Pumps (HPAP) is lost in the throttle valves. Figure 1 is a simplified process flow diagram of a typical contactor side of acid gas removal unit. Since 1970's efforts have been made to develop systems to capture this wasted energy by running a Hydraulic Power Recovery Turbine (HPRT) hence reducing net energy input to the system.

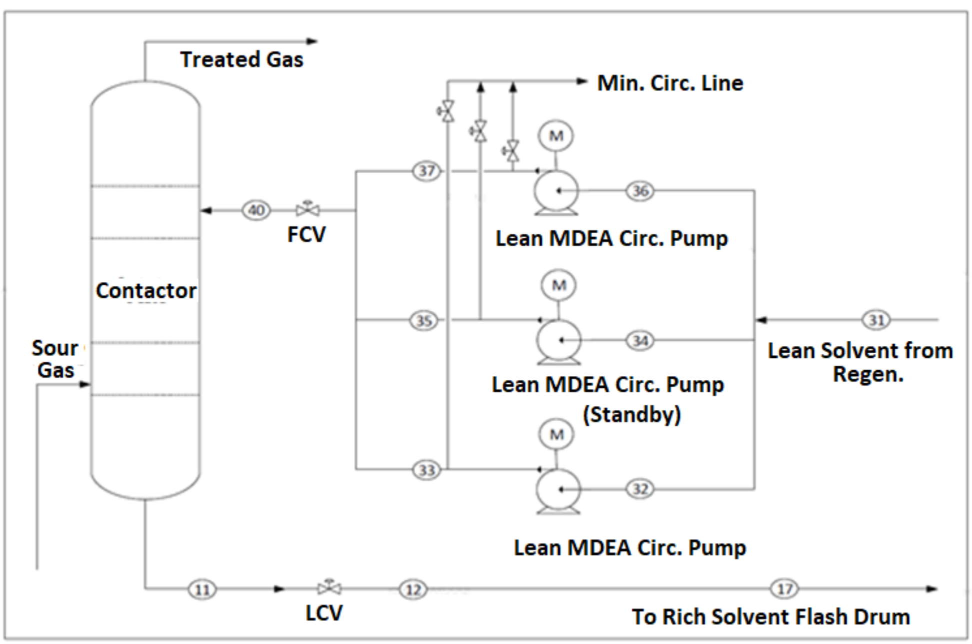


Figure 1: Contactor (Absorber) side of Acid Gas Recovery process

Below is a summary of the energy recovery solutions now deployed in industry to replace wasted pressure energy across rich amine throttle valve.

Hydraulic Power Recovery Turbines (HPRT)
The most common energy recovery method utilizes conventional API 610 reverse running pumps as HPRT. They are coupled in tandem

with a high-pressure amine pump using an over-running clutch as described in API 610 Annex C. This arrangement reduces the power input to the system by the electric motor and saves electrical energy. Gopalkrishnan (1986) in his pump symposium paper details some of the complexities of such an arrangement leading to several major challenges due to the complex multistage system. Two additional mechanical seal systems isolate very high concentrations of acid gas, which, if sour, may contain up to 40,000 parts per million (ppm) of toxic dissolved hydrogen sulfide. Multistage turbines provide sufficient residence time for gas bubbles to be released out of solution and interact with reverse running pump hydraulic components resulting in inconsistent hydraulic performance. Off-design operation will result in high vibrations on the turbine side. The overall reliability of the system is lower due to the very high number of system components prone to different failure modes. Figure 2 shows the simplified arrangement of the Turbine-Motor-Pump system. No such systems are in operation with this End User as of today.

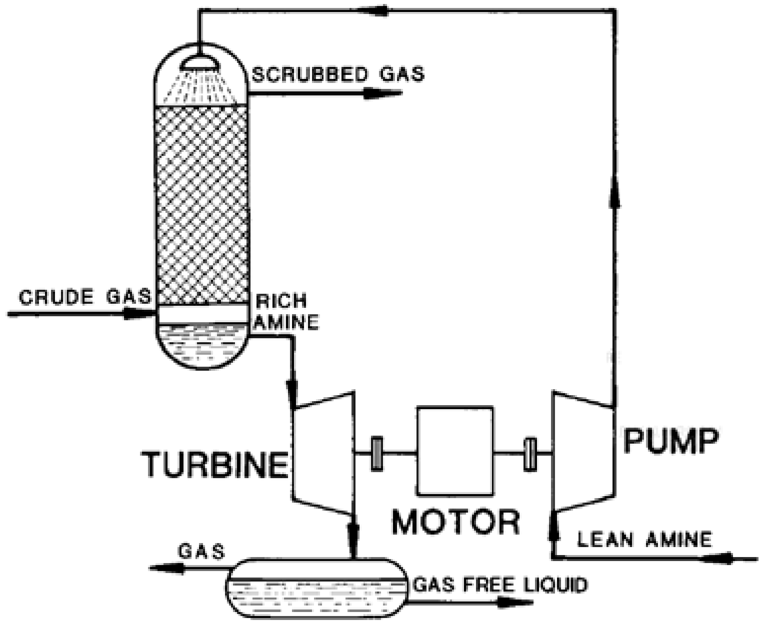


Figure 2: Conventional Pump-Motor-Turbine system (from Gopalkrishnan (1986))

Hydraulic turbo-generator

In 2015 OEM in concert with the End User developed a hydraulic turbo-generator which was a single stage hydraulic turbine coupled to an electric generator (turbo-generator configuration). A variable frequency drive generator was connected to the electrical grid. This system was piloted by the end user in Hawaiyyah Gas Plant (HGP) as shown in Figure 3.

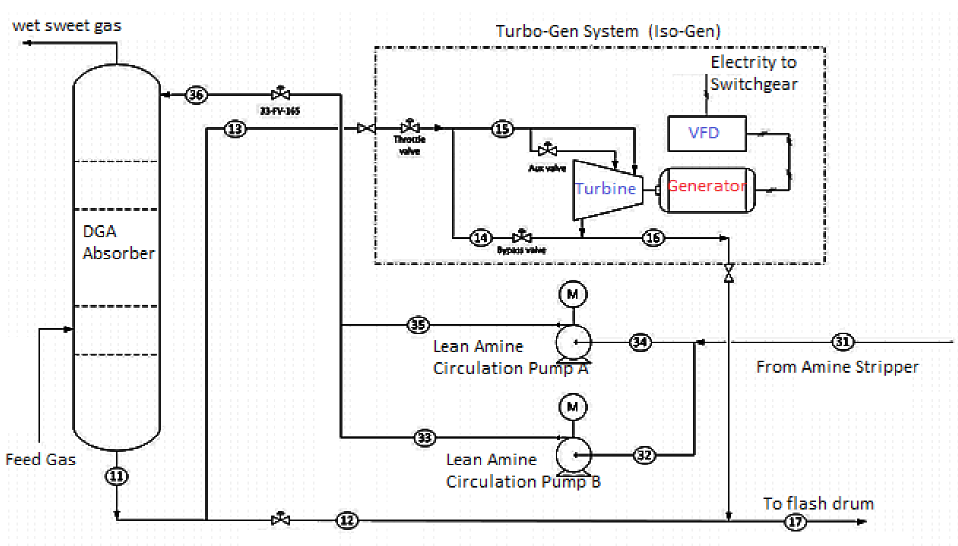


Figure 3: Turbo-Generator generates electricity and feeds it to the grid. Piloted at End User's Hawaiyah Gas Plant.

As noted by Haji et al (2015) the piloting of the Turbo-Gen system at HGP demonstrated recovery of 400 hp (300 kW) of electricity, the system faced challenges due to mechanical seal support system failures with significant equipment downtime. It is pertinent to indicate as noted by Marscher (2002) mechanical seals have been rightly termed as Achilles heel of pump systems as being the largest contributors to pump down time.

Hydraulic Turbocharger

The hydraulic turbocharger is an energy recovery device that uses the hydraulic energy in the rich solvent to boost the lean amine solvent up to the contactor pressure. Figure 4 shows the cross-section of a typical hydraulic turbocharger. The liquid phase turbocharger is the heart of this energy recovery system. The working principle is similar to a gas phase turbocharger normally used in turbocharged diesel engines. Inside the turbocharger casing is a single rotating assembly (turbine impeller, shaft and pump impeller), replaceable volute inserts, process lubricated thrust and journal bearing. Since the driver and pump are contained within the casing, there are no mechanical shaft seals. No mechanical seals is a significant advantage compared to other competing technologies mentioned previously, since mechanical seals are typically the major cause of equipment downtime. The high-pressure rich solvent enters on the turbine side and drives the turbine impeller. The rotating assembly transfers energy to the incoming lean amine inside the pump and provides with sufficient head to reach contactor pressure. The shaft is free to rotate at a speed determined by the operating conditions. Based on customer operating conditions, the turbine & pump hydraulics designs are power-matched so that the shaft operates at a speed, which maximizes turbocharger efficiency. Since the shaft is not limited by synchronous speed, hydraulic turbochargers can be designed for much higher speeds, which results in reduction in size

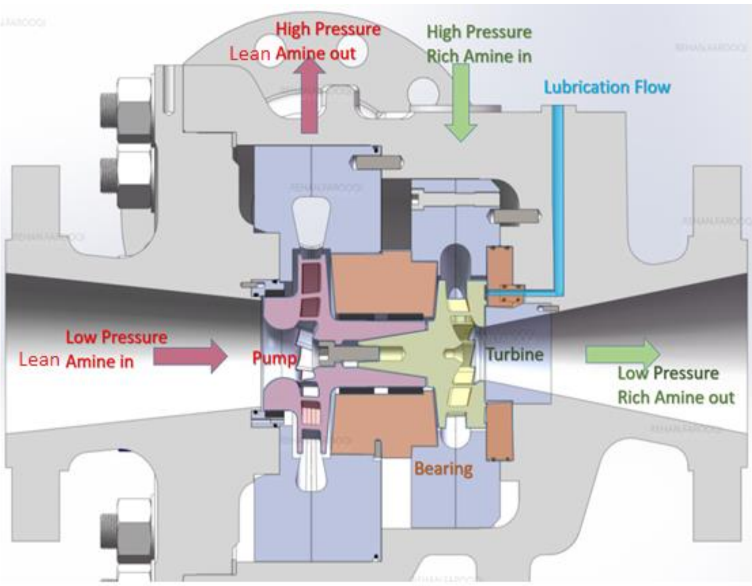


Figure 4: Cross section of hydraulic turbocharger

Table 1: Operating conditions at Fadhili Gas Plant

			Summer		
			Rated	Normal	Turndown
Pump	Q	gpm	2820	2563	1025
	dP	psid	931	953	1040
	Pin	psig	161.5	207	247
	Pout	psig	1092.5	1160	1287
	T	°F	144		
	Sp. Gr.	dimensionless	1.001		
	Viscosity	cP	2.784		
	Speed**	rpm	8000		
	NPS/DHA	ft	394.4		
Turbine	Q	gpm	6565.9	5969	2387.6
	dP	psid	787.9	790.5	800.6
	Pin	psig	936.3	937.7	943
	Pout	psig	148.4	147.2	142.4
	T	°F	177		
	Sp. Gr.	dimensionless	0.947		
	Viscosity	cP	1.730		
	Speed**	rpm	8000		
	NPS/DHA	ft			

Fadhili Gas Plant handles four gas streams from HSBH field in four AGR plants. For sake of brevity, Table 1 shows the worst case of "summer" rated conditions. A quick evaluation based on Fadhili operating data was carried out indicating maximum Power recovery potential at Fadhili Gas Plant for one stream only at greater than 1.7MW. For all four streams, the total power recovery potential was 6.8MW

Figure 5 shows the base case & the different energy recovery equipment solutions considered for Fadhili Gas Plant. They are as follows:

- Option 1 (Base Case): 2x100% and 3x50% High Pressure Lean Amine pumps without power recovery. 3x50% case is shown in the figure.
- Option 2: 2x100% and 3x50% high pressure lean amine pumps with conventional Pump-Motor-Clutch-Turbine type Power recovery (HPRT). 2x100% configuration is shown in the figure.
- Option 3: 2x100% configuration with Turbocharger. Note the figure 5 below show an option of additional pump set to compensate for 20% to 30% system losses including pump and turbine efficiency losses. However, it is to be noted that the system includes Low Pressure lean amine pumps (not shown), these pumps in a variable speed configuration can possibly be used as turbocharger booster pump.
- Option 4: 3x50% configuration with Turbocharger.

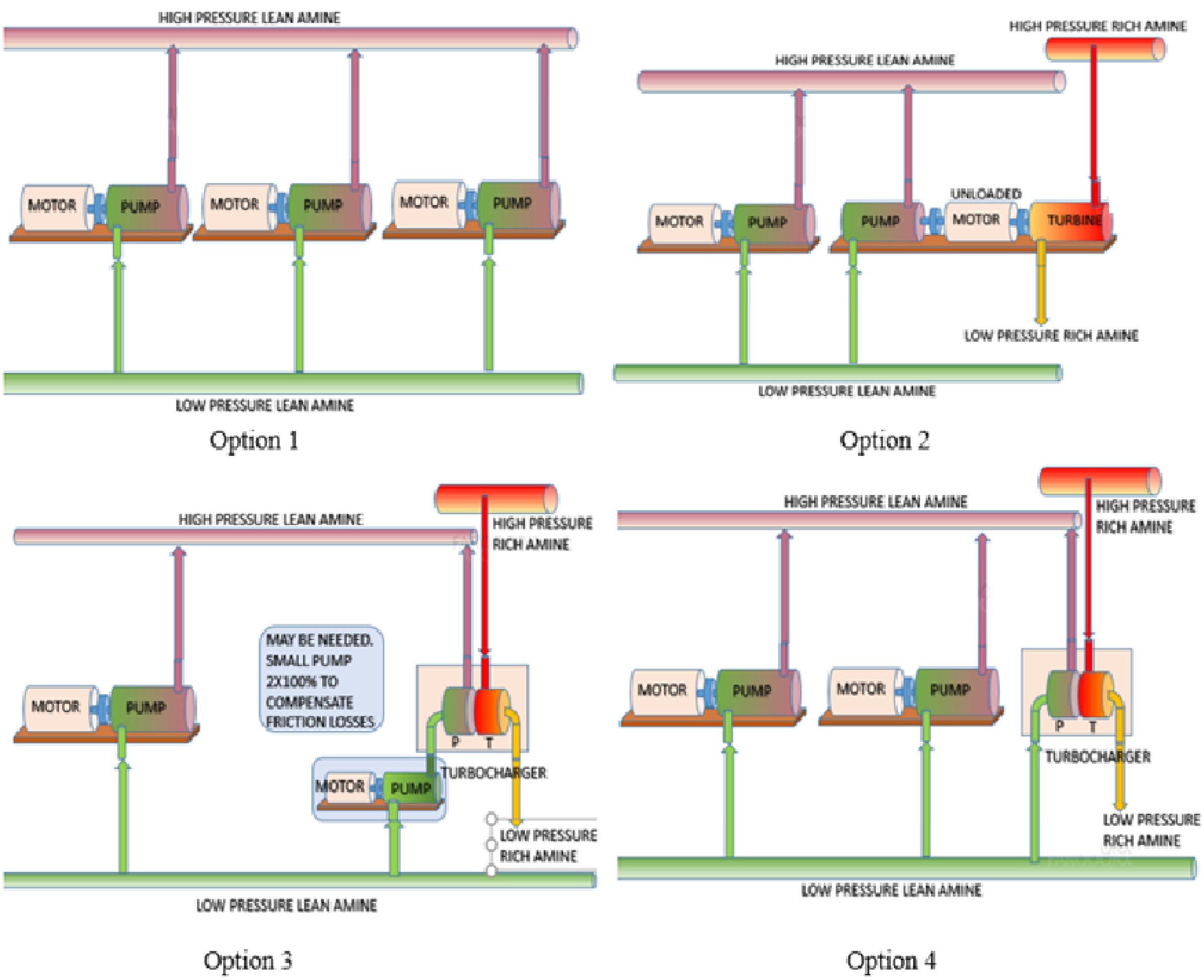


Figure 5: Base case & different energy recovery solutions considered for FGP

Table 2 shows the summary of the short-listed major energy recovery solutions considered. The End User decided to proceed with a 3x50% turbocharger arrangement based on expected reliability and energy recovery potential. To mitigate any risk the turbochargers were selected to be additional equipment to a conventional 3x50% lean amine pump arrangement.

Table 2: Evaluation of major energy recovery solutions

Option	2x100% HPRT	3x50% Turbo charger	2x100% Turbo charger
# of Pumps	2	2	3
# Motors	2	2	3
#Turbines	1	-	-
#Turbo Charger (TC)	-	1	1
Speed Control	Synchronous with Grid	No control	No control
Speed of Turbine / TC	3600 RPM	8000-9000 RPM	6000-7000 RPM
Speed of Pump	3600 RPM Max	3600 RPM Max	3600 RPM max.
Lubrication System	API 610 and 614 compliant lubrication system	process lubricated bearings for turbocharger	process lubricated bearings for turbocharger
Head Per Stage	Less than 1000 feet	Up to 1870 feet in turbine side, Up to 2200 feet on pump side of TC	Up to 1870 feet in turbine side, 1477 feet in pump side of TC
Power Per Stage	1000 hp for turbine. 1000 to 1500 hp for pump side.	Up to 2250 hp* *equivalent motor rating	Up to 3000 hp* * equivalent motor rating
Turbine / pump arrangement	API between bearing	Single stage turbocharger,	Single stage turbocharger,
Power recovery potential % of power input to system	66% Depends on turbine efficiency.	50% Eliminates one pump	71%
Reliability issues	API compliant system 2x100% not supported by RAM study	Very high speed, Very high energy per stage, high head per stage.	High speed, high energy per stage, High head per stage. 2x100% not supported by RAM study

Technical Requirements for Hydraulic Turbochargers on Amine Gas Recovery Systems

Liquid phase turbochargers are widely used in Seawater Reverse Osmosis Plants with power ratings typically less than 670 hp (500 kW). Failures are infrequent; however, when failures occur they can be catastrophic, resulting in major equipment damage if not detected early.

End User studied their own operating experience as well as from major users such as Saline Water Conversion Corporation Saudi Arabia. Significant history of catastrophic failures was collected which included issues like radial bearing damage, thrust bearing damage leading to rotor and shaft damage, welding cracks and welding damage. Figure 6 shows photographs of some as found damaged conditions for turbos designs by other OEM's for desalination service. Despite many potential benefits of turbocharger, the reliable operation of installed equipment at Fadhili Gas Plant was critical for the End User because of the possibility of major commercial losses as well as safety consequences of released toxic H_2S fluid due to equipment failure.

Figure 6: Experience of catastrophic damage history of hydraulic turbocharger on water service

End User's, engineering team concluded that the proposed turbocharger system at Fadhili Gas Plant must undergo a redesign exercise. The End User and OEM agreed on following the framework of design requirements summarized in Table 3.

Table 3: Design challenges and recommended approach

Design Challenge	Recommended Design Approach
1. Risk of catastrophic failures of pump and turbine impellers due to high energy and high head per stage.	Re-design for high-energy application. Conduct additional analysis to verify pressure pulsation and fatigue life. Pressure pulsation magnitude measured during tests.
2. Risk of over speed and associated damages in case of process upset or mechanical damage.	Evaluate design, calculate and verify runaway speed. Mechanical re-design for worst condition
3. Single volute design leading to off design radial loading.	Develop twin volute or diffuser design and determine off design radial thrust coefficients and radial load values.
4. No design feature to accommodate thrust reversal during off design operation and process upset.	Design, manufacture and test features to handle worst-case thrust reversal.
5. Risk of bearing damage as a result of high radial loading	Evaluate bearing design to handle worst combination of radial load. Radial and thrust bearings must be lubricated by filtered fluid. Design a duplex filtration system.
6. Potential of off-design operation of turbine or pump side and associated complications due to mismatching of hydraulics and as result of either pump or turbine side control	Design hydraulics from scratch. 1. Matching range of hydraulic characteristics for pump and turbine through iterative process 2. Calculate the turbocharger operating envelope by CFD simulation and non-dimensionalzed analytical methods
7. Potential of NPSH inception and NPSH related damage issues due to high tip speeds	1. Determine NPSH inception by CFD of final design. 2. If needed do a 2-Phase cavitation analysis to estimate life of impeller against cavitation damage.
8. Compliance to API 610 although API 610 does not cover turbochargers.	1. Design to meet or exceed API-610 requirement. 2. Documents the list of API requirements that do not apply and list of API requirement to which an exception will 3. Fully comply with all applicable End User standards.
9. Brazed impeller construction and its limitations in a high energy application.	Change manufacturing process. Investment cast impellers only with design features to reduce stress concentration areas are required.
10. Rotor dynamic analysis to verify critical speeds.	Confirm by analysis that the rotor is 'classically stiff' even at runaway speed if possible.
11. Operating scenarios and configurations leading to complex controls especially in a 3x50% configuration	Extensive process simulation and redesign process controls to ensure turbocharger can operate in parallel in 3x50% configuration.
12. Insufficient design procedures for high-energy equipment design. Need to carry out fatigue life evaluation.	CFD, FEA will be carried out to design against rotor-stator interaction, resonance, fatigue failure and dynamic response.
13. Material selection for major components, bearings, wears rings and thrust disk.	Revisit material selection. Comply with SAES-G-005. Radial and thrust bearing material selection.
14. Product lubricated static bearing design and filtration system adequacy for the process fluid.	1. Filtration system to be designed for worst-case solid particle. 2. Non-metallic component in static bearing to be avoided (detailed testing by OEM showed non-metallic bearings to have better start-up performance than WC bearings) 3. Prove thrust bearing design through testing in a bearing test rig.
16. Inadequate test procedure for turbocharger performance and mechanical testing.	1. Complete range performance test for turbine and pump. 2. Define operating envelope, complete range test for turbocharger. 3. NPSH test if required. 4. Full train endurance test
17. Need to model coupled turbine and pump for the turbocharger design to simulate behavior in process plant application and for design performance test.	1. Process simulation model to be developed which will be validated first for the test loop. 2. All process operating conditions and worst caseworst-case scenarios to be modeled and evaluated.

18. Insufficient Hydraulic design process.	Develop hydraulic design process to include 1-D and 3-D design with CFD steady state as well as transient analysis-for both pump and turbine side. Develop a non-dim model to determine turbocharger iso-torque operating range.
19. Need to improve design review process.	Add End User review at critical design location.
20. Carry out component level FMEA in collaboration with CSD.	FMEA to evaluate consequence turbocharge component failure and potential impacts.
21. No means to determine Rotational speed measurement and recording in system.	Rotating speed determination either directly or indirectly by a reliable method e.g. 1. Seismic vibration sensors on bearing housing 2. Dynamic pressure transducers measuring pressure pulsation frequency. 3. Flow rate and head across pump and turbine side and running speed.
22. Insufficient instrumentation and no emergency Shut down system.	1. Novel arrangement of axial position probe to be added. 2. Seismic vibration probe to be added. 3. Determine operating conditions requiring emergency shutdown and program in DCS. 4. Design a reliable mechanism for emergency shutdown.

DESIGN

Almost designed from scratch, re-engineering of the new generation turbocharger took 52 weeks. For sake of brevity, this section will present only the major and novel design approaches. Designing a turbocharger from scratch involves three major area – hydraulic design, mechanical design, and turbocharger SKID design. Mechanical design further involves casing design and bearing design. Figure 7 shows the high level design workflow for designing a turbocharger from scratch. It starts with analyzing the customer operating conditions and down-selecting one design point for the hydraulics, and determining the pump and turbine design point. Generally, the hydraulics is designed to meet the summer/winter worst rated conditions with the auxiliary valve closed. Next, the most important step is selecting the shaft speed that will maximize the turbocharger efficiency – which is a product of the pump efficiency and turbine efficiency. Other than maximizing turbocharger efficiency, the shaft speed may need to be limited by considerations about pump or turbine cavitation and rotordynamics.

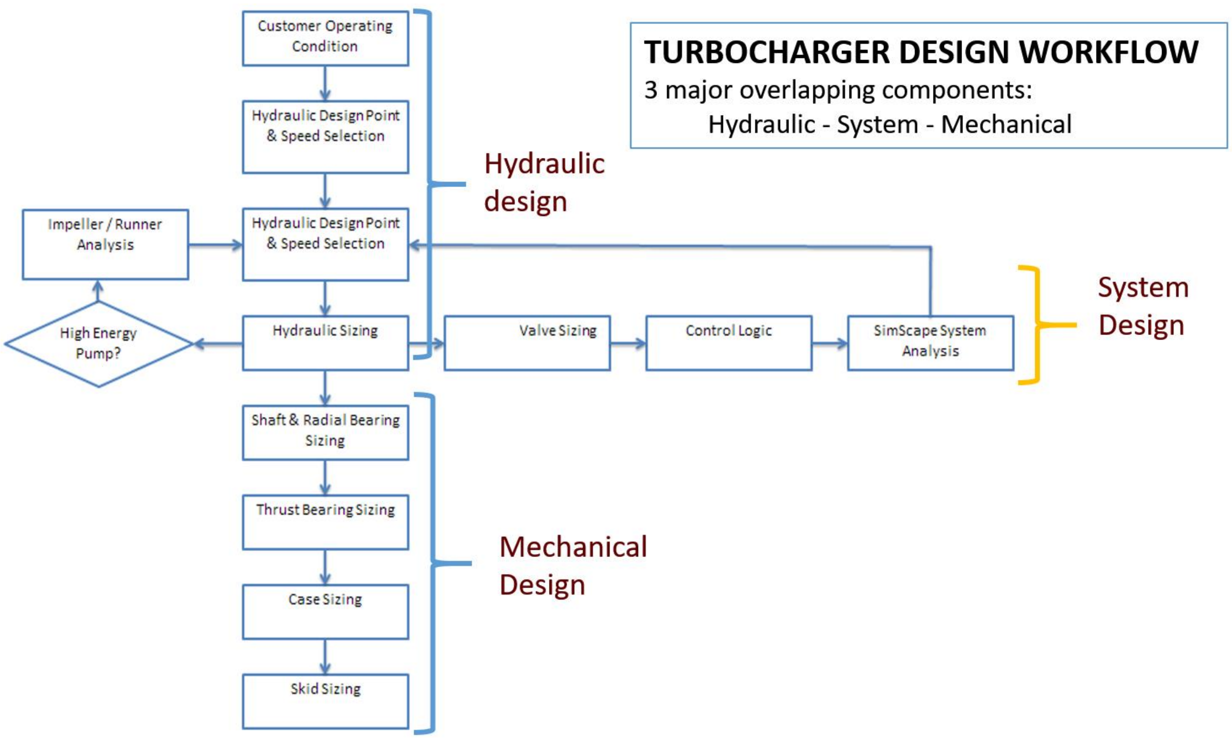


Figure 7: Turbocharger design workflow

When designing from scratch, hydraulic sizing takes precedence over bearing and casing sizing. Finally, different valves and lines on the SKID are sized. The process simulation of the entire amine plant helps to determine whether the turbocharger in concert with the lean amine circulation pumps can achieve the dual function of contactor level control and supply lean amine into the contactor. Typically a 50-50 flow ratio – i.e. turbocharger pump supply 50% of total lean amine flow into the contactor.

In an AGR plant, the turbocharger recovers energy from the rich amine side and uses it to boost the pressure of lean amine. Therefore, the design of the turbocharger is integral with the design of the entire AGR plant. Also since the turbine and pump side are on a common shaft, and will run at same shaft speed and shaft power, their designs as inter-twined with each other. The OEM makes standard hydraulic turbocharger, which are used in the SWRO industry for over 30 years, where a waste pressure energy from membrane reject, brine water is recovered using a turbine and used to boost the pressure of fresh water going into the membranes. In this application, a booster pump is in series with the pump side of the turbocharger. The high suction pressure on the pump side means that the axial thrust is uni-directional hydrostatic thrust bearing. However, for this project, the 3x50% configuration was selected i.e. a turbocharger and circulation pump in parallel boost the lean amine flow by the same head but half the total required flow. In 3x50% configuration, the pump suction pressure is much lower compare to a 2x100% configuration, and so there is possibility of thrust reversal during startup/shutdown. This necessitated a bi-directional thrust bearing. Some of the other important design considerations and challenges that were unique to HSBH turbocharger design are listed below:

- **Maximizing efficiency**: The End User had requested that maximize efficiency was primary goal, and the turbocharger meet efficiency target within a -0/+2% target per API 610. Since each turbocharger has custom hydraulics, estimation of the efficiency at the design stage had to be very precise
- **Design for cavitation control**: Turbocharger is not limited by motor synchronous speed, and hence are designed for high speed to minimize wheel sizes which makes them compact – however this also means that the hydraulics have to be designed to ensure cavitation is not a concern at the high inlet tip speeds

  Bearing design: Design for bi-directional thrust. Design bearings for both water and amine application. Prevent mixing of rich amine into lean amine
- **System design and control:** Ensure seamless switch-over of turbocharger & circulation pump to two circulation pumps

Turbocharger based HSBH AGR plant design

Table 4 shows total six operating conditions for the turbocharger system on the rich and lean amine side. Summer and winter includes conditions for rated, normal and turndown. The summer and winter conditions are close together, and summer rated represents the maximum flow condition. During preliminary design stage, summer rated condition was chosen as the design point and based on initial hydraulic sizing it was determined that the turbocharger would be able to meet normal and rated conditions but it will be difficult to meet turndown conditions due to significant increase in turbocharger efficiency. A process simulation tool – Turbocharger System Analysis Software (TSAS) developed specifically for this project showed the extent of the turbocharger turndown at the detailed design stage. The system analysis tool was also used to develop and fine tune the control narrative to seamlessly transition the AGR plant from having the turbocharger and one circulation pump in operation to having the rich amine led down valve and two circulation pumps online, while the turbocharger is idled or shutdown based on the duration of expected turndown.

Table 4: Summer and winter operation conditions for HSBH turbocharger

			Summer			Winter		
			Rated	Normal	Turndown	Rated	Normal	Turndown
Pump	*Q*	gpm	2820	2563	1025	2778	2525	1010
	dP	psid	931	953	1040	932.3	959	1067.5
	Pin	psig	161.5	207	247	158.7	209	253.5
	Pout	psig	1092.5	1160	1287	1091	1168	1321
	T	°F	144			144		
	Sp. Gr.	dimensionless	1.001			1.002		
	Viscosity	cP	2.784			2.841		
	*Speed***	rpm	8000					
	NPS/DHA	ft	394.4					
Turbine	*Q*	gpm	6565.9	5969	2387.6	6457	5870	2348
	dP	psid	787.9	790.5	800.6	788.2	790.7	800.6
	Pin	psig	936.3	937.7	943	936.5	937.8	943
	Pout	psig	148.4	147.2	142.4	148.3	147.1	142.4
	T	°F	177			173		
	Sp. Gr.	dimensionless	0.947			0.950		
	Viscosity	cP	1.730			1.871		
	*Speed***	rpm	8000					
	NPS/DHA	ft						

Figure 8 shows the P&ID of the HSBH AGR plant with turbocharger system and three amine circulation pumps in parallel. Double check valves on pump discharge safeguard against reverse flow. During start-up or shutdown, minimum recirculation line prevent sudden dead heading of the turbocharger pump. Turbocharger bearings are process fluid lubricated. The lean amine from turbocharger pump discharge as well an external line from circulation pumps is filtered and injected into turbocharger for lubrication. To prevent mixing of rich into lean amine and to ensure turbocharger bearings are always flooded with clean filtered lean amine, an external line from circulation pumps is used. The lubrication line also measures the temperature and the flow rate, and turbocharger system is shutdown if the bearing lubrication flow temperature exceeds 180F or the flow rate drops below 5 gpm.

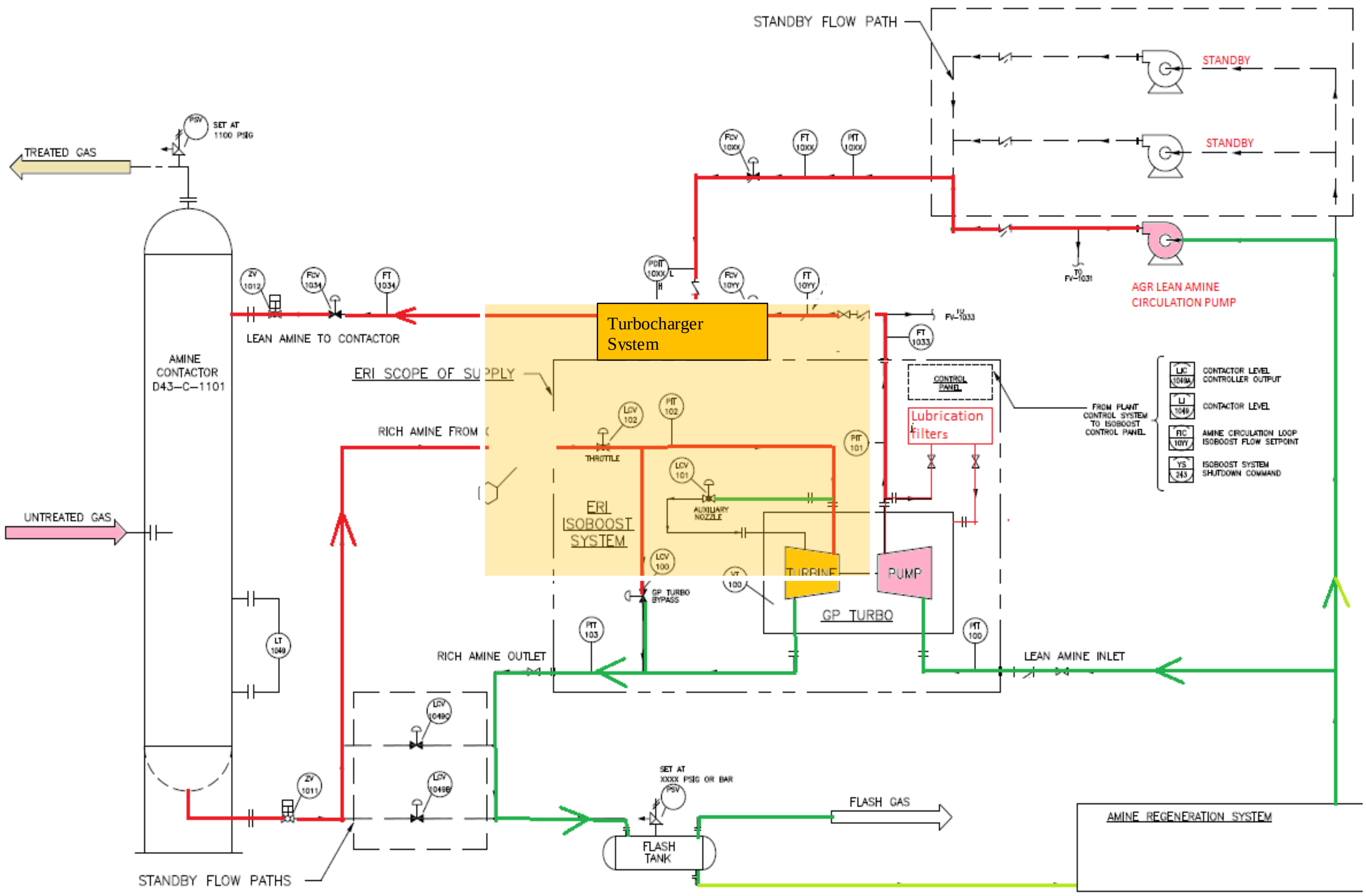


Figure 8: Turbocharger in lieu of contactor level control valve (LCV) and an electrically driven circulation pump

Figure 9 shows the turbocharger skid CAD model. The key components of the skid are the turbocharger, the lubrication line and the three turbine side control valves - throttle, auxiliary and bypass valves. These valves along with the turbine help control the amine level in the contactor.

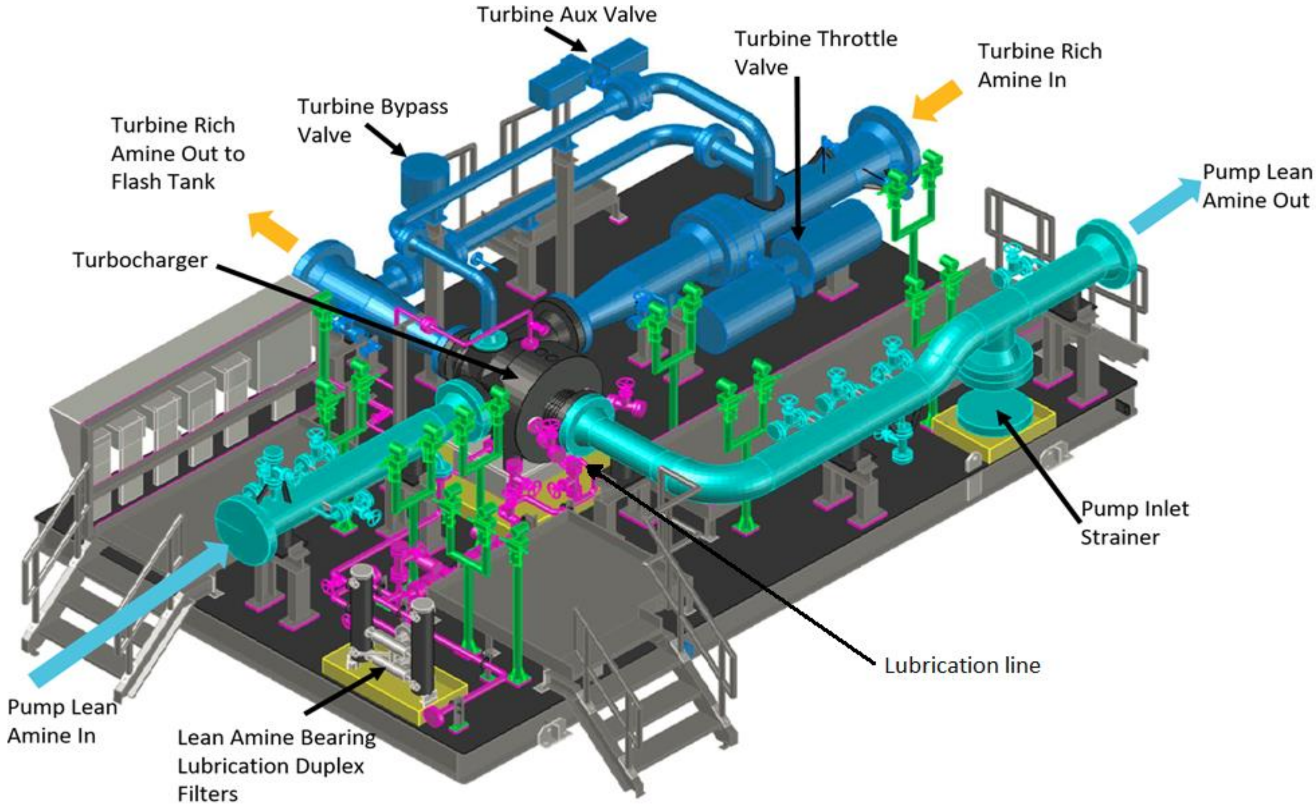


Figure 9: CAD model of turbocharger skid

Figure 10 shows the cross-section of the redesigned 3x50% turbocharger for HSBH AGR plant. The left side is the hydraulic turbine runner coupled to the pump impeller on the right. The turbine volute converts the inlet hydraulic energy to the runner through the main nozzle insert. As shown in Figure 11, it also has a provision for an auxiliary nozzle which feeds the runner as well. With the auxiliary valve close, the runner gets all its flow from the main nozzle, and when the auxiliary valve is opened the turbine is fed by both the main and auxiliary nozzle, thus moving the resistance curve for the turbine to the right.

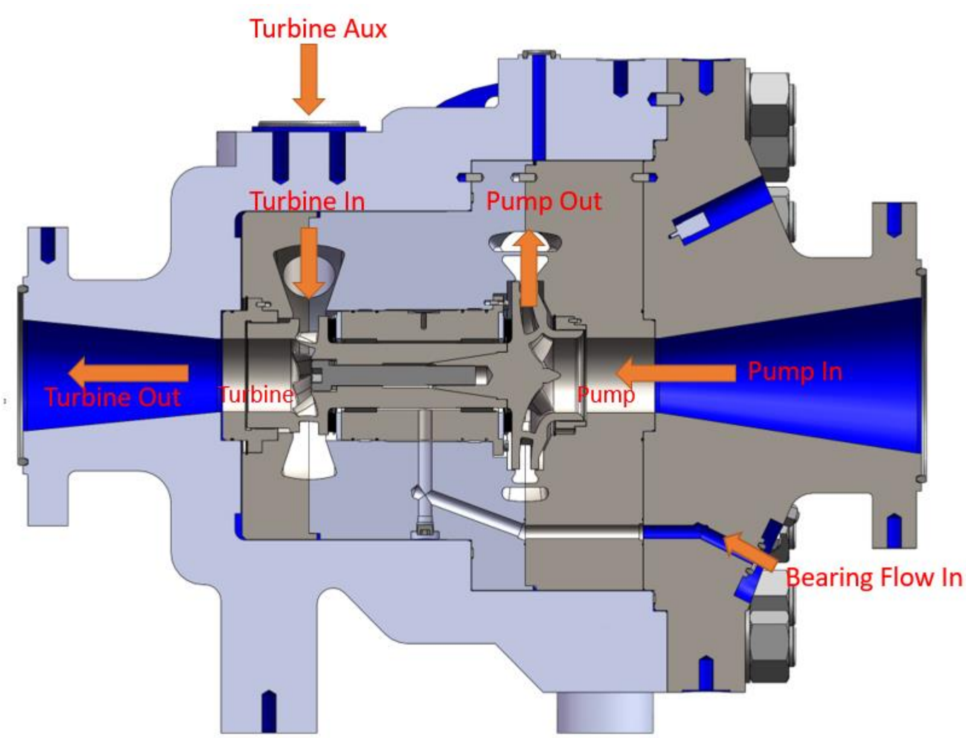


Figure 10: Cross-section of re-designed 3x50% HSBH turbocharger

On the bottom right there is a flanged connection for bearing lubrication flow which is injected in the between the two journal bearings which support the shaft in between the turbine and the pump. On the top right there is a provision for a velometer to measure casing vibration as close to the impeller as possible.

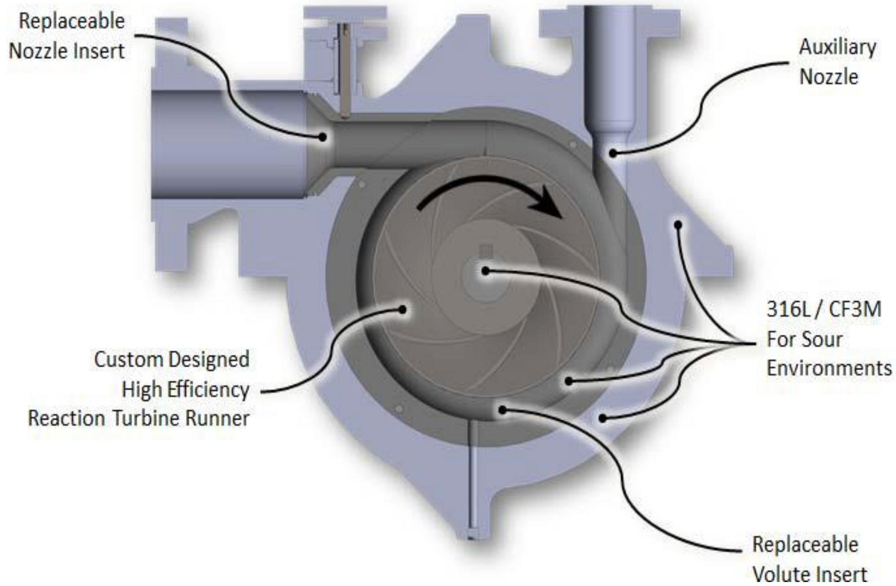


Figure 11: Hydraulic turbocharger, typical turbine cross section showing auxiliary nozzle

The turbocharger shaft design speed is 8000 rpm and the pump side rated point per datasheet is 2820 gpm at 2195 ft of head, thus making it a high-energy pump. Figure 12 shows the HSBH pump impeller hydraulics plotted against historical high energy pumps as taken from Karassik et al (2000). It was therefore decided to change the pump volute design to a double volute to reduce hydraulic radial loads.

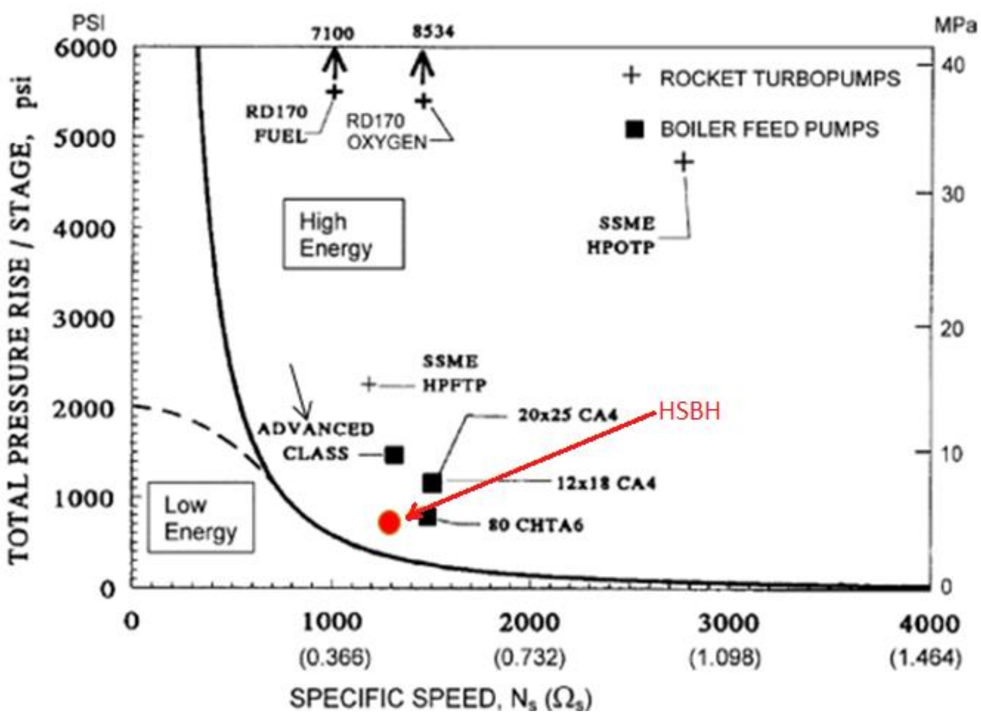


Figure 12: Pump energy level defined in terms of stage pressure rise (from Karassik et al (2000))

Table 5 shows the comparison of AGR process without turbochargers (Case 1) and with turbochargers in 3x50% configuration (Case 2). Case 1 shows a total power requirement of 3961 hp (2954 kW) at rated condition, whereas for Case 2 the power requirement at rated conditions with the turbocharger is only 1981 hp (1477 kW) i.e. 50% power savings. This assumes a even 50-50% flow split between turbocharger pump and lean amine pump i.e. each providing 2820 gpm of lean amine flow. However, during lab testing (FAT) the

HSBH turbochargers were more efficient (60%) than estimated at bid stage (51%). Therefore, based on the as-tested characteristic curves of the pump and turbine, TSAS was used to simulate the AGR process under field process fluid conditions. And it was showed that the plant could run with a flow split of 55.5-45.5% i.e. turbocharger pump could provide up to 3130 gpm of lean amine flow rate and the electrically driven circulation pumps will make up the rest at 2510 gpm of lean amine flow rate. This would translate to energy savings of about $1.15 million dollars per year and CO2 reduction of about 10,100 tons per year. Therefore, for the four combined HSBH AGR plants, the total energy savings could be up to $4.59 million dollars per year and CO2 reduction of about 40,400 tons per year. This assumes cost of energy of $0.08 per kWh.

Table 5: Annual energy savings and CO2 emission reduction per HSBH turbocharger – bid stage estimate and measured based on lab testing

	Case 1, existing Process	Case 2, with turbocharger 3x50% configuration	
	LA Circ. Pumps G-1105 A/B	Turbocharger System + LA Circ. Pump G-1105 A	
	Rated [1]	Rated [1]	Measured @ FAT [2]
LA flow from Regen., gpm	5,640	5,640	5,640
RA flow from contactor, gpm	6,566	6,566	6,566
RA pressure from contactor, psi	936	936	936
Flash drum pressure, psi	148	148	148
LA pump suction pressure	165	165	165
Amine Sp. Gr., Lean/ Rich	1.00/ 0.95	1.00 / 0.95	
IsoBoost			
IsoBoost pump flow, gpm		2,820	3,130
Pump suction pressure, pri		165	165
Pump discharge pressure, psi		1,080	1,080
IsoBoost turbine flow, gpm		5,283	6,214
turbine suction pressure, psi		936	936
turbine discharge pressure, psi		148	148
LA Circ. Pump	**LA Circ. Pumps G-1105 A/B**	**LA Circ. Pump G-1105 A**	
flow	5,640	2,820	2,510
suction pressure	165	165	165
discharge pressure	1,080	1,080	1,080
pump efficiency	80%	80%	
motor efficiency	95%	95%	
TOTAL POWER REQUIREMENT, kW	2,954	1,477	1,314
Power Saving, kW		**1,477**	**1,639**
Power savings, %		50%	56%
Cost of energy, kWh		0.080	
ANNUAL ENERGY SAVING, MWh		**12,938**	**14,362**
ANNUAL ENERGY SAVING, $/Yr		**1,035,000**	**1,149,000**
CO2 EMISSION REDUCTION, Tons CO2/Yr		**9,100**	**10,100**
Potential extra saving for four units, $/Yr			**456,000**
Assumptions		Notes:	
Contactor pressure, psi	936	LA: lean Amine	
Amine circ. pumps suction pressure, psi	165	RA: Rich Amine	
Amine circ. pumps dischare pressure, psi	1,080		
Flash drum pressure, psi	148	[1]	110% of Normal Flow
LA flow rate, gpm	5,640	[2]	Avg of four HSBH units
RA flow rate, gpm	6,566	*: Summer condition for both cases	

As part of the redesign effort, the turbocharger layout was also changed to a cartridge-style design with one endcover instead of two to allow for ease of vertical assembly and to reduce radial tolerance stack-ups. The cartridge consists of the pump volute half, center bearing assembly and the turbine volute halves bolted on to the end cover. Figure 13 shows the installation of the cartridge into the casing in vertical position during lab testing.

Figure 13: Installation of cartridge in tubocharger casing for FAT

Hydraulic design

For the selected design point preliminary hydraulic sizing was performed. The speed selection was based on the following considerations:

- Maximize turbocharger efficiency which is a product of pump efficiency and turbine efficiency
- Maximize speed to reduce wheel size and fit into a smaller casing
- Pump cavitation – i.e. impeller inlet tip speeds and NPSH margin as noted in Schiavello (2009)

The head and flow for pump and turbine are known and the shaft speed common to both pump and turbine is selected which will determine the specific speed for both pump and turbine such that efficiency is maximized. This requires empirical design charts of efficiency vs specific speed for pump and turbine separately. For 3x50 configuration selected for HSBH, the turbine to pump flow ratio is 2:1 **-** and pressure ratio is typically close to 1:1. This means that for AGR 3x50% application pump specific speed is close to around 1000 in US units and turbine specific speed (using same definition as pump) will be around 2000 and above in US units. These numbers could vary based on size of plant and the lean amine flowrate.

Based on above considerations the selected design speed for HSBH was 8000 rpm. Once the speed is selected for the given operating conditions, the pump & turbine specific speed is fixed. Figure 14 shows the hydraulic design workflow. A preliminary hydraulic sizing & performance estimation of the pump & turbine hydraulic components is performed using empirical relations & past designs based on these specific speeds.

During detailed design, 3D geometry of the pump impeller and turbine runner with respective volute are created that are analyzed using CFD. The design is iteratively modified until the turbine and pump shaft power is matched and the design requirements such as turbocharger efficiency are met. CFD & other proprietary 1D sizing codes are used to estimate the performance characteristics (head vs flow, efficiency vs flow) of the pump & turbine separately over the operating flow range. Based on estimated efficiencies of pump and turbine the shaft diameter is selected to safely transmit the required shaft power and torque. Bearings and wear rings are then sized based on empirically estimated axial & radial loads.

After this pump & turbine performance characteristic curves are adjusted for volumetric & mechanical losses using 1D loss models. The CFD+1D characteristic performance curves are fed to TSAS which analyzes the relevant amine circuit including the turbocharger energy recovery system (turbine, pump, throttle, bypass & auxiliary valves), Level Control Valve (LCV), Contactor Flow Control Valve (CFCV), booster pumps, lean amine circulation pumps and their discharger & minimum recirculation flow control valves and pressure safety valves. AGR Process simulation code is used to ensure adequate sizing of the control valves and test the turbocharger energy recovery system control logic for startup, shutdown, normal and upset conditions. During this analysis if the range of operating points requested by the customer is not met, hydraulic design is modified until the requested range of operation is met.

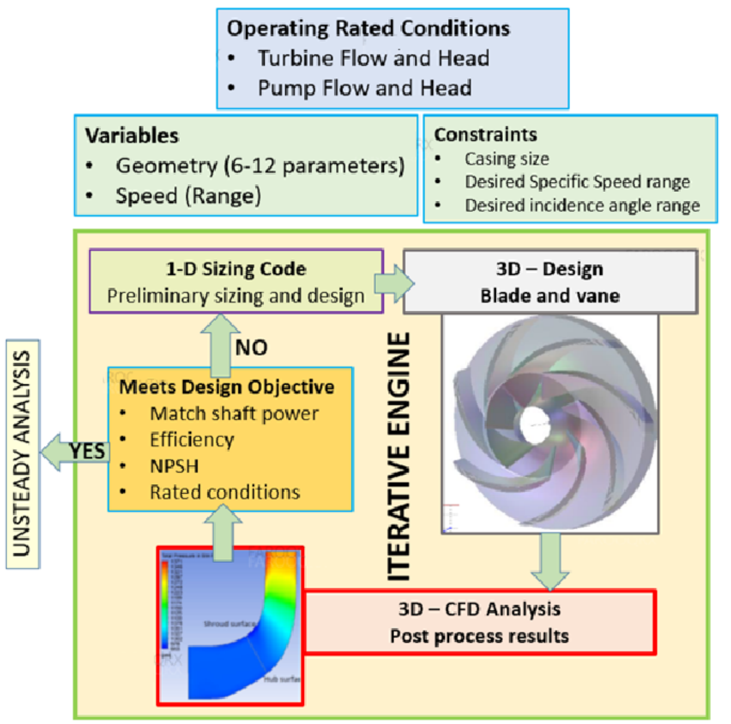


Figure 14: Hydraulic design workflow

Bearing design

The OEM designs fully customized optimum hydraulics for each order, and uses standard uni-directional axial hydrostatic bearing and center journal bearings for SWRO application. For this order, due to the above mentioned challenges with bi-directional axial thrust, and API 610 requirements, the bearing configurations was designed from scratch. To limit the design space certain sizes of pre-designed hydrodynamic tilt-pad bearings are utilized. The hydrodynamic thrust bearing supports the rotating assembly on back shroud of the pump impeller or turbine runner, there is an interdependence of hydraulic sizing and bearing selection. Figure 15 shows the various interdependent relations between hydraulic sizing, thrust and radial bearing sizing/design and power loss calculations. Bearing design is driven by hydraulic design, and vice versa. Thrust load depends on hydraulic sizing, bearing configuration and the design of the lubrication flow path. In addition, final hydraulic design (i.e. pump and turbine matching) is dependent on secondary losses such leakage, disk friction etc inside the pump and turbine, which may not be trivial depending on the size of turbocharger.

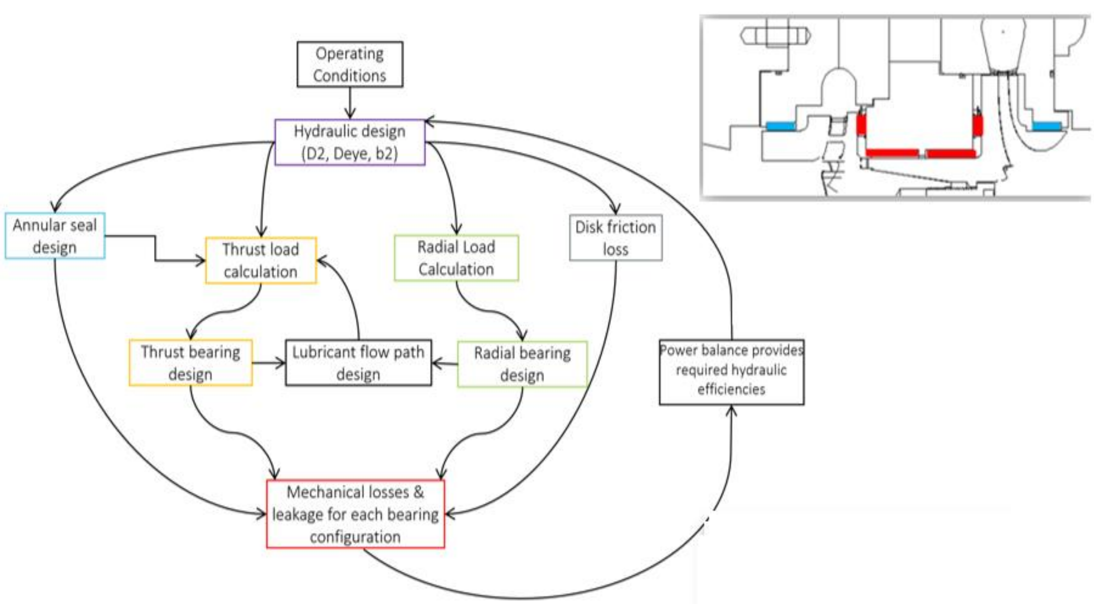


Figure 15: Inter-dependency of Hydraulic and Bearing design

Table 6 shows the altenative bearing configurations that were considered taking into account operating conditions and pressure gradients to ensure only filtered lean amine is fed to the bearings. However, the chosen configuration ('A' from Table 6) was the only one that satisfied the following criteria for the application: prevent mixing of rich amine into lean amine, ensure filtered lean amine lubricates the bearings at a controlled temperature.

Table 6: Bi-directional bearing configurations for turbocharger

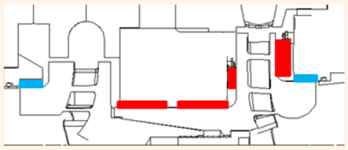
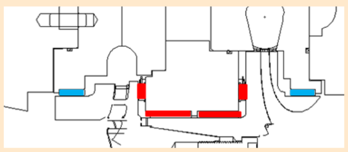
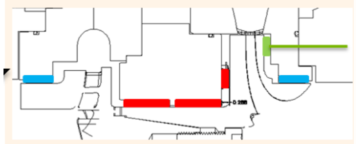
	BrgConfig 'C' – 2x3x5 TPTB on impeller back shroud & 3x5x7 TPTB on impeller front shroud	- Excessive power loss (4% brg loss. Desal total loss is 3%.)	- Greater load capacity - Proven design for Aquabold pumps - Both Thrust bearings can be lubricated by lean amine flow
	BrgConfig 'A' – 2x3x5 TPTB on impeller back shroud & 2x3x5 TPTB on impeller front shroud	- Possibility of inadequate thrust load capacity in water	- Easy to implement in accelerated schedule - Proven design for Aquabold pumps - Can handle misalignment - Power loss not excessive in water
	BrgConfig 'B' – 2x3x5 TPTB on impeller back shroud & HS bearing on impeller front shroud	- Cannot handle mis-alignment - Excessive leakage if thrust direction reverses (or use const. flow pumps)	- HS have greater thrust load capacity - Smaller power loss - HS Load capacity is same for water for amine

Figure 16 shows the final bearing configuration and its implementation in the HSBH turbocharger. The new designed HSBH turbocharger utilizes product lubrication radial & dual acting thrust bearing to handle any thrust reversals. This is the first time hydrodynamic dual acting thrust bearings have been implemented in a hydraulic turbocharger. The bearings are made of a PEEK composite. Different grades of non-metallic and carbide bearings were tested as part of the 500 hp proof-of-concept turbocharger, before finalizing the bearing material to the current grade of PEEK.

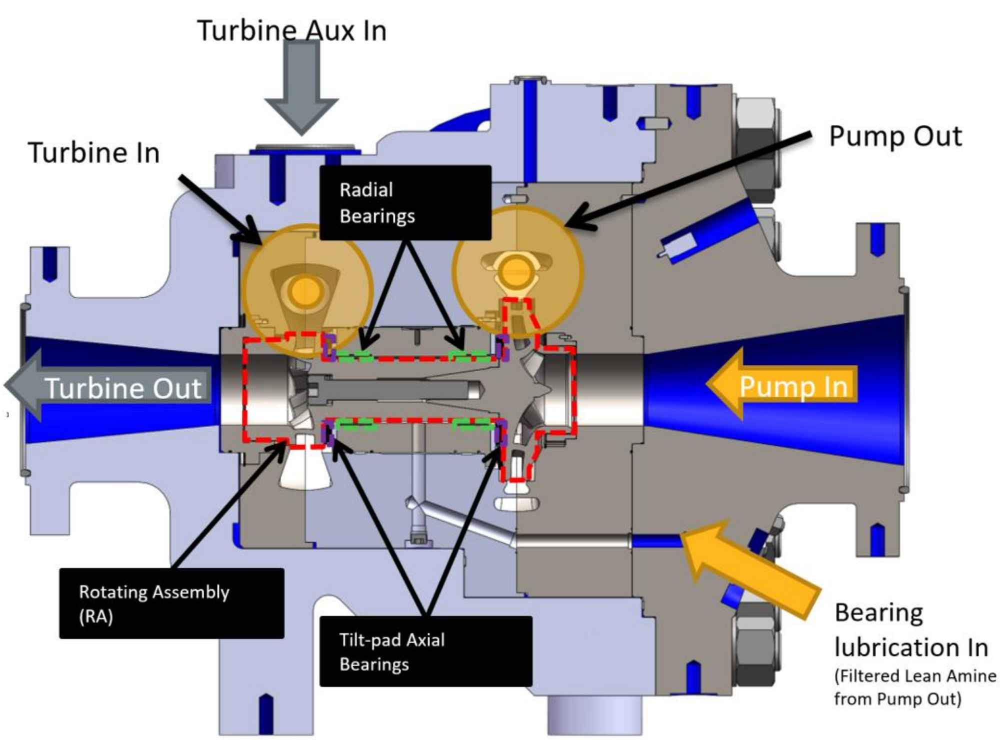


Figure 16: Turbocharger bearing configuration

Figure 17 outlines the bearing lubrication flow path that was the final choice. Filtered lean amine is taken from the turbocharger pump-side discharge line. It is then injected at the center of the two radial bearings. From there it flows to either side past the radial bearings to both the pump and turbine thrust bearings. As mentioned previously, to ensure that the bearings are always flooded with filtered lean amine even when turbocharger is shutdown, a lean amine from the circulation pumps is fed via an external lubrication line upstream of the turbocharger skid filters.

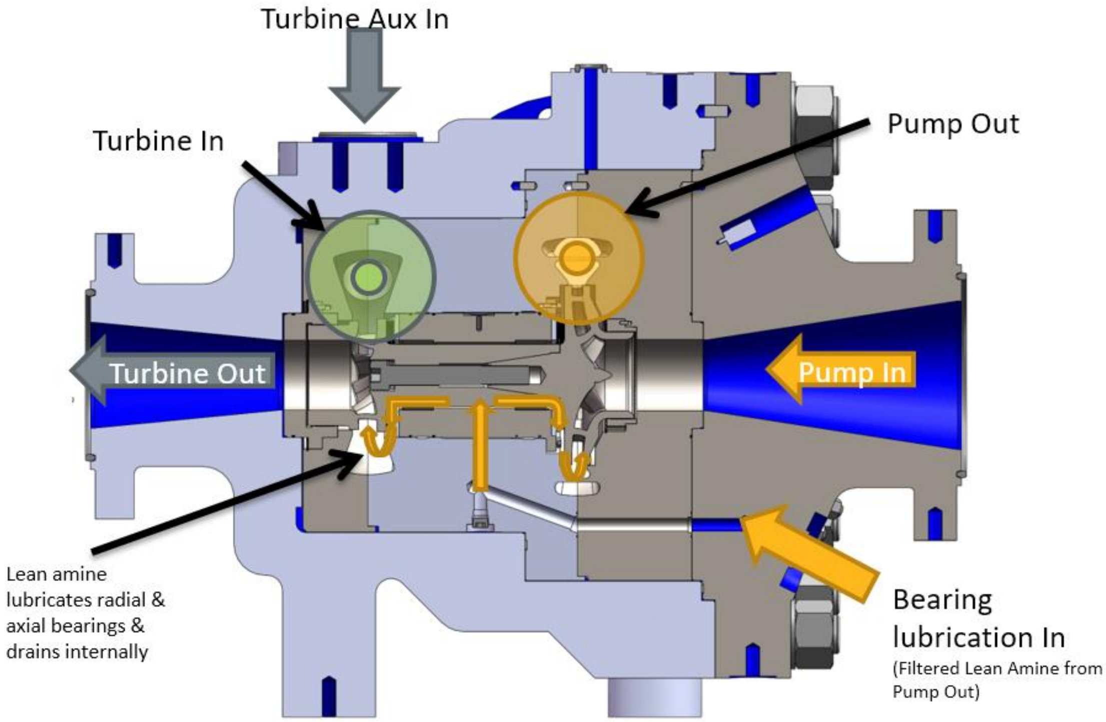


Figure 17: Turbocharger lubrication flow path

The main challenges with bearing design were:

- Hydraulic loads and bearing capacity:
  - Accurately estimate & validating hydraulic axial and radial loads, since there was no room to oversize the bearings (constrained by impeller and runner rear shroud sizing
  - Determining the axial and radial bearing load capacity under various scenarios
- Bearing lubrication flow:
  - Determine the minimum requirements of bearing flow to prevent
    - Starvation of bearings
    - Excessive temperature in the bearings (amine breaks down at 250F)
  - Ensure that the pressure driven lubrication flow under various scenarios is always greater than the minimum required flow rate

Figure 18 shows the various forces and pressure distribution acting on the rotating assembly of the turbocharger. It also shows the typical pressure distribution in the side wall gaps of the pump impeller and turbine runner. The axial thrust calculation for a pump is notoriously difficult to estimate accurately as it is subtraction of two large thrust forces opposing each other. For a turbocharger the uncertainity of the axial thrust calculation is doubled because the net axial force is subtraction of the thrust forces acting on the pump impeller and the turbine runner. One of the reason for the uncertainty in determining the axial thrust is knowing the radial pressure distribution in the side wall gaps as explained in detail by Gulich (2010). Therefore the radial pressure distribution, and hence the swirl ratio was determined experimentally for 500 hp proof-of-concept turbocharger test rig using pressure taps on the front shroud and the rear shroud. As the specific speeds of the 500 hp and 2000 hp hydraulics were similar, swirl ratio data was used to calculate the axial thrust on the 2000 hp HSBH units as well.

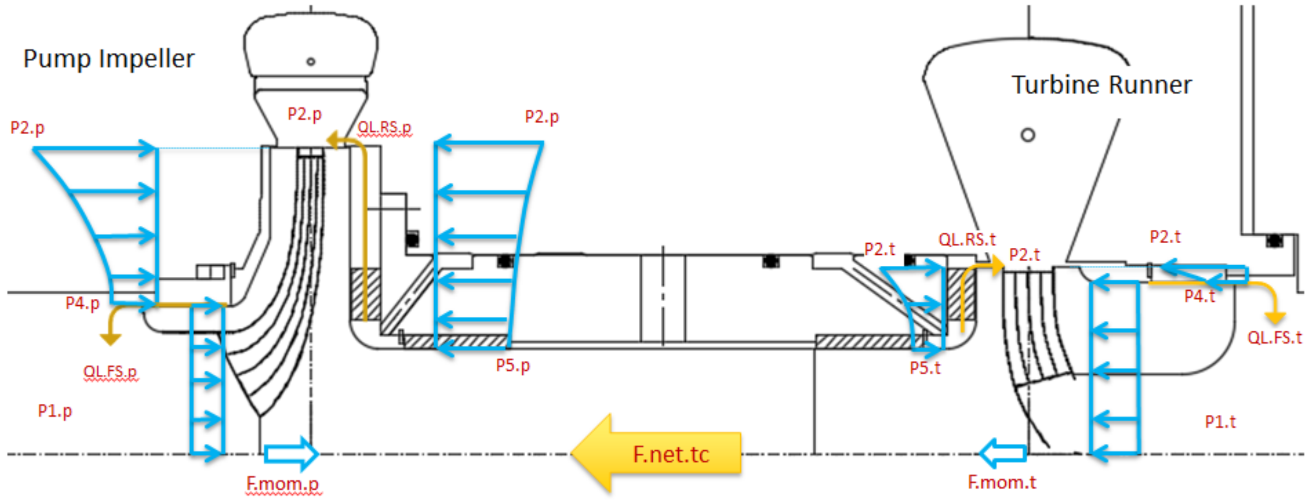


Figure 18: Axial thrust calculation for turbocharger

Figure 19 shows a pressure distribution of the thin film over the thrust bearings using commercial thrust bearing analysis software. By running the analysis at various bearing tips speeds and bearing loads, the bearing thrust capacity was calculated for a minimum film thickness based on the surface finish of the bearing and the running counter surface for various scenarios such as:

- Rated operating condition with water as the process fluid (FAT)
- Rated operating condition with amine at the maximum temperature as the process fluid (summer in the middle-east)
- Partial runaway condition

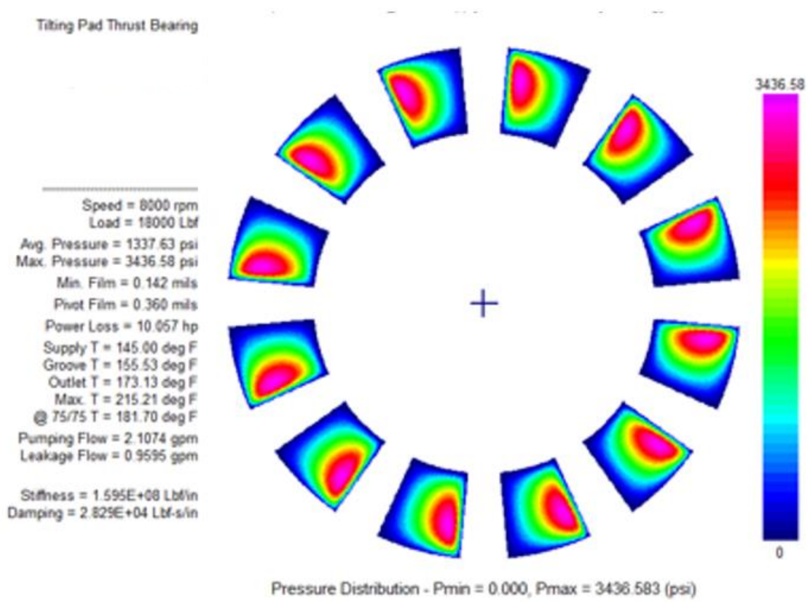


Figure 19: Pressure distribution on the thrust bearing calculated using commercial thrust bearing analysis software

Figure 20 shows the minimum film thickness curves for the tilt-pad bearing under water and amine as the bearing fluid calculated using commercial thrust bearing analysis software – the solid blue and orange curves are the recommended design capacity for the bearings in water and amine respectively, whereas, the dotted blue and orange curves are the film breakdown capacity in water and amine respectively. The bearing design capacity for water at the same tip speed of say 250 ft/s is about 40% lower compared to amine (compare 1000 psi vs 700 psi). The horizontal purple line indicates the mechanical limit at which the PEEK bearings will deform,based on lab tests.

Also superimposed on this is graph is the hydraulic thrust load for rotating assembly calculated for the two extreme scenarios of rated condition and partial runaway. As noted previously due to test loop limitations, the turbochargers were tested with water which has about 1.5-2 times lower viscosity compared to amine which the process fluid in the field. Another important data point is the red diamond which corresponds to the 500 hp proof-of-concept turbocharger test data point, where the thrust unit load of about 800 psi was indirectly calculated based on the pressure taps as noted previously. This is much worse than about 700 psi of bearing unit loading calculated for

HSBH rated condition. The successful testing of the 500 hp turbocharger provided the confidence that the much larger 2000 hp turbocharger bearings will be able to survive the FAT with the lower viscosity water as the lubricating fluid.

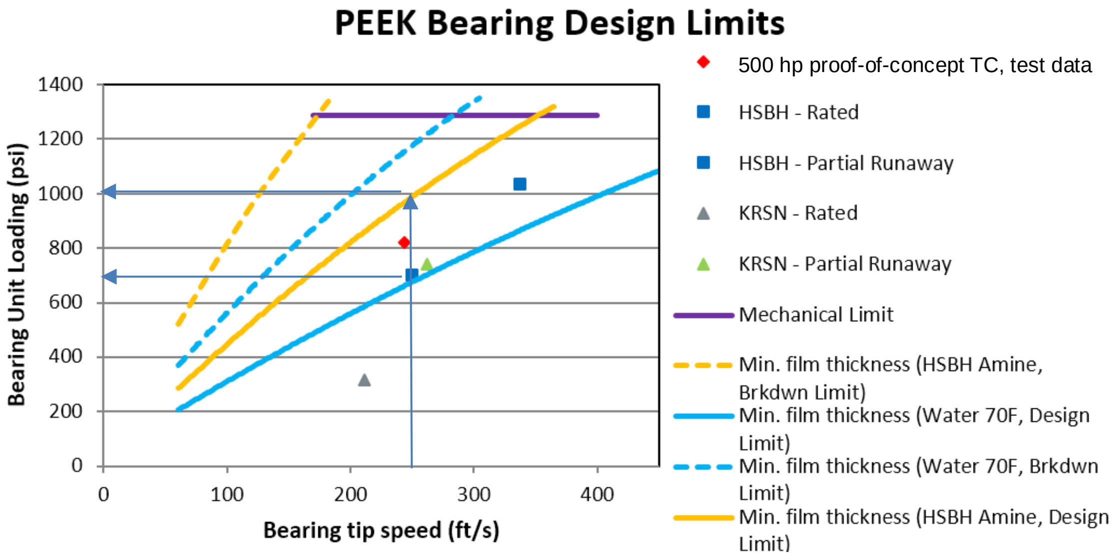


Figure 20: Non-metallic thrust pad bearing design limits

A detailed 1D-circuit analysis of the lubrication flow path was also performed using TSAS to ensure the pressure driven flow is greater than the minimum required for the radial & axial bearings based on rated loads & thermal consideration as well (about 5-6 gpm).

Rotating assembly fatigue analysis

Since the energy level for the turbocharger per stage is quite high, fatigue analysis of the pump impeller and turbine runner were performed. For the sake of brevity, only the fatigue life investigation of only the pump impeller element is presented here for the worst case which is the summer rated point. The loading on the element has a static component and a dynamic or cyclic component. For the cyclic component, the S-N curve, as seen in Figure 21, serves as a starting point to analyze allowable alternating stress for a given number of loading cycles. The diagram is typically created using a rotating specimen where-in the mean stress is zero. The endurance limit, S_e, is the maximum alternating stress below which the material can be subjected to an infinite number cycles without failure. The ASME BPVC (2010) Section VIII has been used for the results presented here.

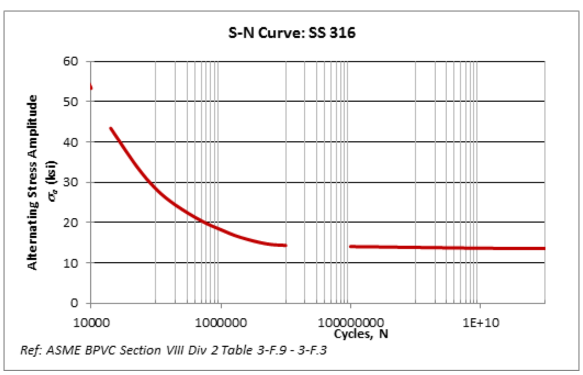


Figure 21: 316SS SN curve for rotating assembly (pump and impeller) fatigue analysis

Figure 22 shows the mean stresses that arise from the static component of the loading in the part are accounted for using a Goodman diagram drawn using the ultimate tensile strength, S_{ut}, and the endurance limit, S_e. An equivalent alternating stress is, S_{eqvlt_alt}, calculated by extending a line drawn between the points (S_{ut}, 0) and (σ_{mean}, σ_{alt}) to the y-axis. S_{eqvlt_alt} can be used with the S-N curve to calculate the factor of safety if it is lesser than S_e, or the allowable load cycles if greater.

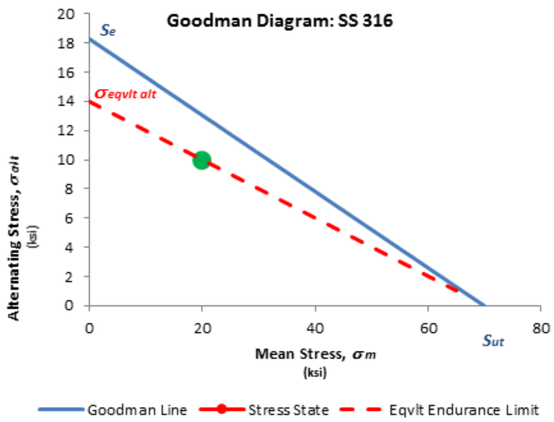


Figure 22: Impeller & runner Goodman diagram

For the FEA, the loading condition takes into account both static and cyclic loading to do a fatigue analysis. The static loading comprises of the centrifugal loading as a result of rotational velocity (8000 rpm), pressure loads at the pump suction, wear ring, external faces of the front and back shrouds. For the internal pressure distribution, a point cloud from CFD results was mapped into the structural model, as seen in Figure 23. For the cyclic loading as a result of the pressure pulsations with the impeller blade and volute cutwater interaction, a conservative assumption of 10% of the pressure rise per stage was considered for this amplitude with a linear relationship with radius. The load has been applied to the blade pressure side and inner sides of the front and back shrouds.

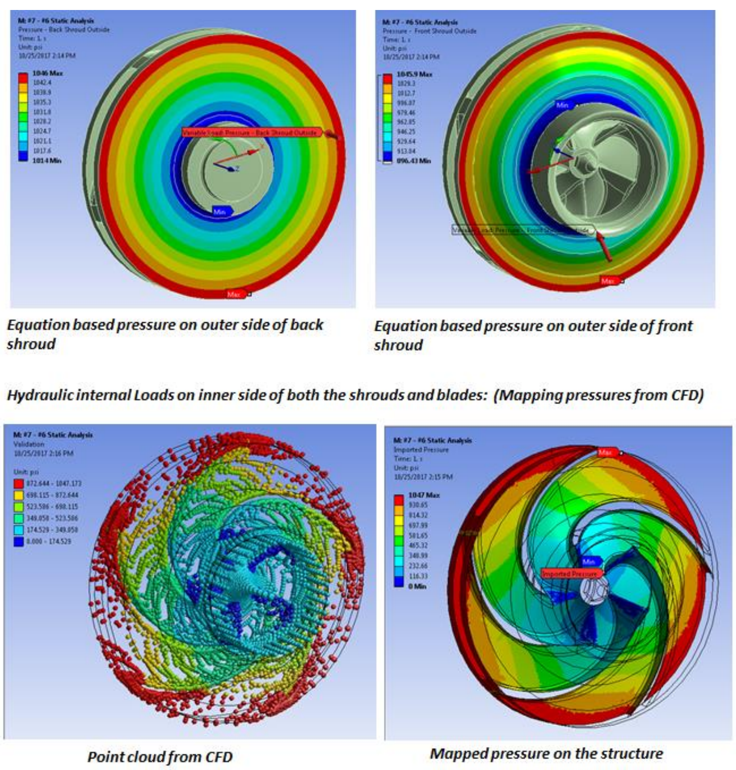


Figure 23: Mapping of hydraulic pressure on impeller walls

Figure 24, shows the results from the impeller fatigue analysis, the maximum equivalent alternating stress is calculated to be **6.1 ksi**, which occurs locally at the trailing edge of the impeller. The stresses are fairly low compared to the endurance limit S_e of **13.6 ksi,** the factor of safety being **2.23.** This is based off uniaxial S-N curve data from ASME.

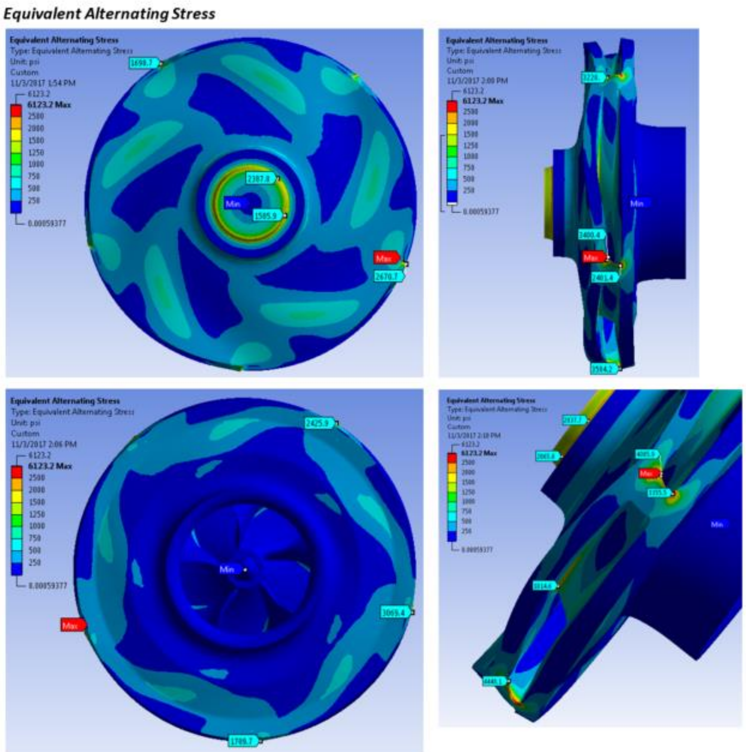


Figure 24: Pump Impeller fatigue FEA analysis

Bearing condition monitoring
In typical pumps and turbines, a mechanical seal or equivalent is used to prevent exposure of the bearings to the process fluid. Axial and radial bearings are located in bearing housings under atmospheric pressure lubricated with clean oil. The bearing monitoring system consists of two radial proximity probes (to obtain the shaft orbit), two axial probes (to measure the axial movement of the shaft) and a key phasor (to measure rotational speed). As these do not see the pressurized process fluid, the implementation is relatively straightforward, all being mounted on the bearing housing bolted on or integral to the casing and/or end cover of the machine. A turbocharger uses product-lubricated bearings with no mechanical seals present. The shaft is not easily accessible as a result. The bearing monitoring probes would need to be exposed to the process fluid to accurately capture the axial and radial movement of the shaft.

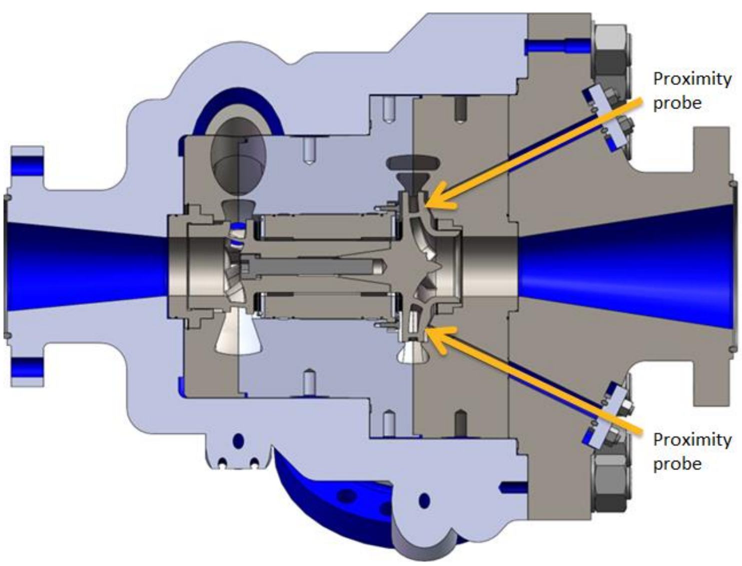


Figure 25: Angled axial proximity probes

An angled probe system was invented as described by Pattom (2020) to overcome this by coming in at an angle with two proximity probes as shown in Figure 25 and then using trigonometry to derive the axial shaft position. These probes can be used to measure thrust bearing condition i.e. wear which would cause position to change, and thrust direction. In addition, the casing allows the accommodation of a radial proximity probe and a key phasor to measure the radial bearing location and speed. Figure 26, shows implementation of radial proximity probes.

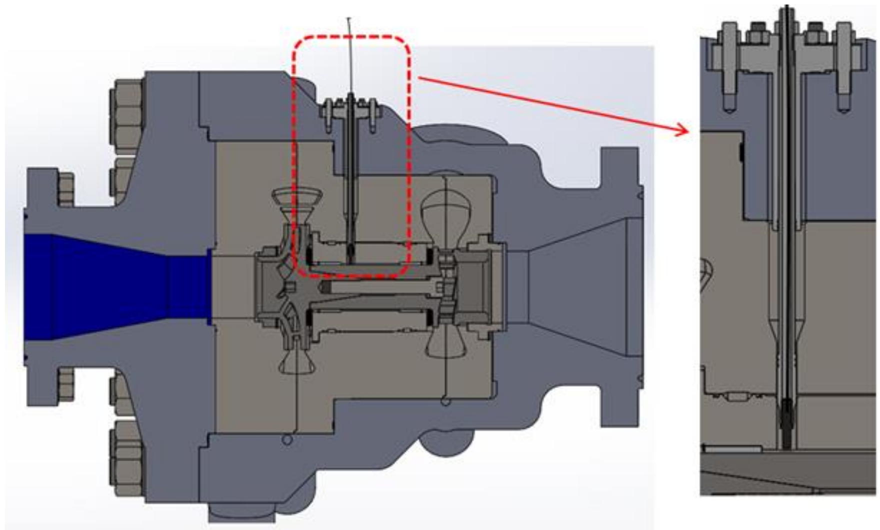


Figure 26: Radial proximity probes

Process Simulation

As mentioned previously, the TSAS was developed specifically for the project to analyze various what-if turbocharger based AGR control methods. Figure 27 shows the interface for the TSAS tool. The following elements are modeled in TSAS apart from the turbocharger - contactor, lean amine booster & circulation pumps, lean amine turbocharger and circulation pump balancing valves, flash tank, lean amine check valves, contactor flow control valve, pressure safety valve, lean amine minimum recirculation line, rich amine turbocharger skid throttle, auxiliary and bypass valves.

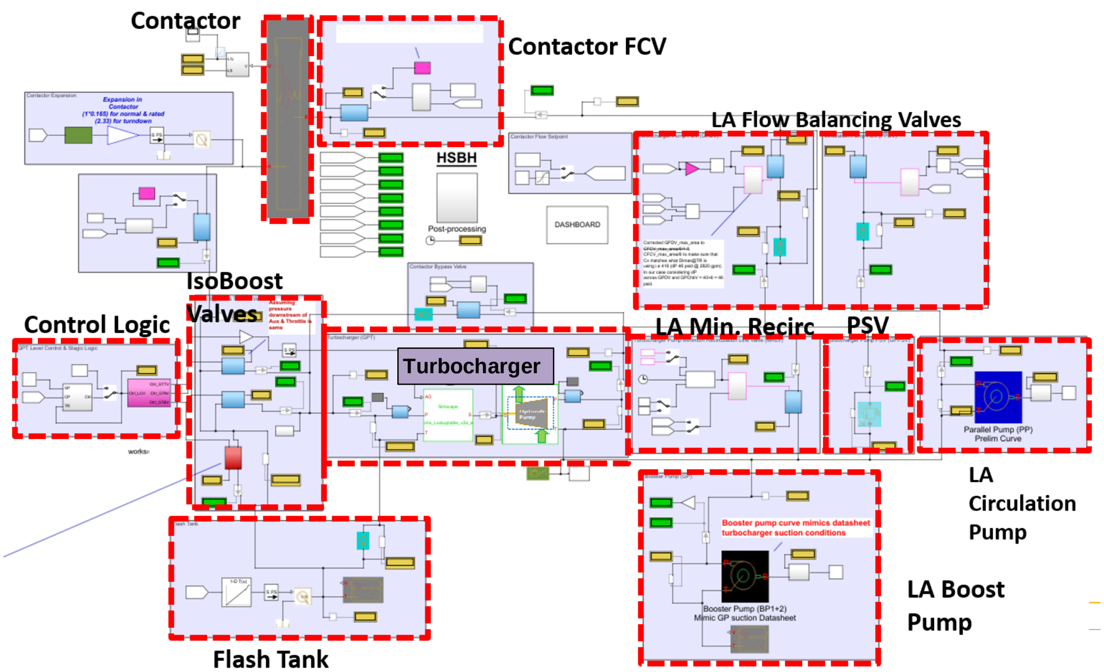
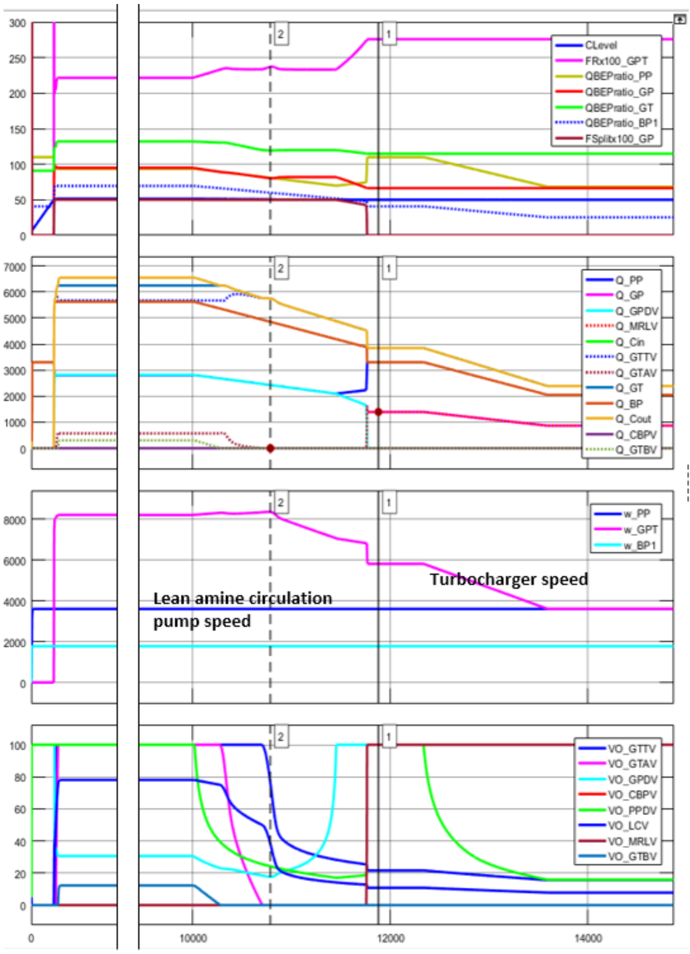


Figure 27: AGR plant with turbocharger, circulation pump, process simulation using in-house developed software - TSAS

Figure 28 shows a typical turndown simulation for HSBH AGR using TSAS. The input to TSAS is pump and valve curves, contactor dimensions, and normal, rated and turndown operating conditions for the plant. The graphs show the plant coming online with the turbocharger and lean amine circulation pump running in 3x50% configuration. Then as the flow setpoint for the total flow into the contactor is decreased, the controls automatically throttles the lean amine balancing valves to reduce the flow through the turbocharger and circulation pumps. This also reduced flow coming out from the contactor, and thus at one point the turbine cannot absorb enough energy from the rich amine stream to power the pump to pressurize the lean amine to open the check valve. The exact flow rate at which this happens is not a straight forward calculation and can be easily calculated using TSAS.



Key non-dimensional parameters (%)
CLevel : Contactor level
FRx100_GPT : Turbine to pump flow ratio (x 100)
QBEPratio_PP : Ratio of circulation pump flow rate to its BEP flow rate
QBEPratio_GP : Ratio of circulation pump flow rate to its BEP flow rate
QBEPratio_GT : Ratio of circulation pump flow rate to its BEP flow rate
QBEPratio_BP1 : Ratio of circulation pump flow rate to its BEP flow rate
FSplitx100_GP : Ratio of turbocharger pump to total lean amine flow into contactor (typical 50%)

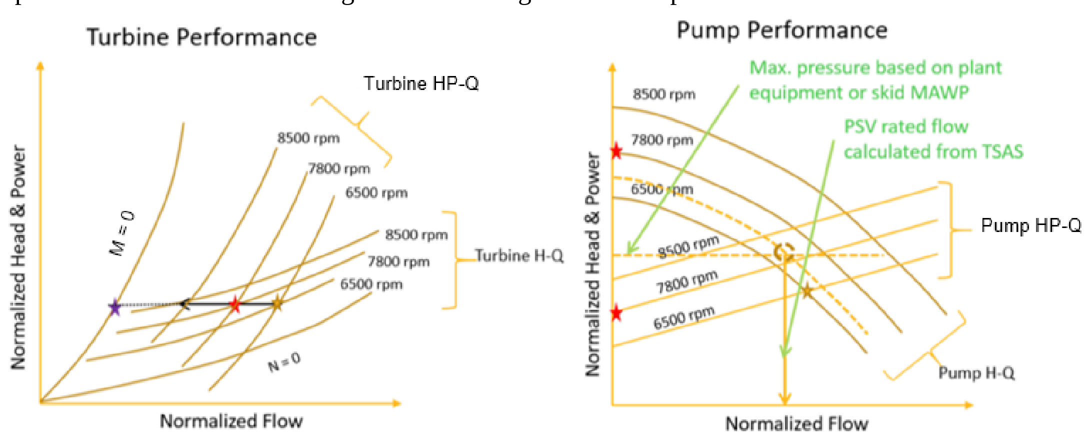
Flow rates (gpm)
Q_PP : Electrically driven lean amine circulation pump flow rate
Q_GP : Turbocharger pump lean amine flow rate
Q_GPDV : Turbocharger pump flow through discharger flow control valve
Q_MRLV : Turbocharger pump flow through minimum recirculation line valve
Q_Cin : Total lean amine flow into the contactor
Q_GTTV : Rich amine flow through turbocharger turbine throttle valve
Q_GTAV : Rich amine flow through turbocharger turbine auxiliary valve
Q_GTBV : Rich amine flow through turbocharger turbine bypass valve
Q_GT : Rich amine flow through turbocharger turbine = Q_GTTV + Q_GTAV
Q_BP : Lean amine flow through booster pumps
Q_Cout : Rich amine flow out of contactor

Shaft speed (rpm)
w_PP : Electrically driven lean amine circulation pump shaft speed
w_GPT : Turbocharger shaft speed
w_BP1: Booster pump shaft speed

Valve Openings (% open)
VO_GTTV : Valve opening of turbocharge throttle valve
VO_GTAV : Valve opening of turbocharge throttle valve
VO_GTBV : Valve opening of turbocharge throttle valve
VO_GPDV : Valve opening of turbocharge throttle valve
VO_CPBV : Valve opening of contactor bypass valve
VO_PPDV : Valve opening of circulation pump discharge valve
VO_LCV : Valve opening of level flow control valve (in parallel with turbine)
VO_MRLV : Valve opening of turbocharger minimum recirculation line valve

Figure 28: HSBH plant turbocharger start-up, turndown & shutdown process simulation using TSAS

Another important use of TSAS is to determine the partial run-away shaft speed of the turbocharger. Partial-runaway happens when the turbocharger pump is dead headed due to blockage or insufficient energy generated by the turbine to open the lean amine pump discharger check valve. Figure 29 shows turbine performance at different speeds. The left-most curve is the zero-torque curve and the right-most curve is the zero shaft-speed curve while there is flow through the turbine side. If the pump side is suddenly blocked, then shaft power requirement decreases. If the turbine hydraulic power input does not decrease in a similar proportion then the turbine can over speed until the shaft power is balanced again. Since the pump differential pressure varies as a squared of the shaft speed – it is important to ensure that the casing MAWP is designed for such pressure event.



Operating Condition	Speed (rpm)	Pump Shut-off (psid)	Pump suction pressure (psig)	Pump shut-off discharge pressure (psig)
Normal	6500	1138	147	1285
Partial Runaway (shut-off)	7800	1639	147	1786

Figure 29: Partial runaway condition for turbocharger (Torque (T) = 0 – runaway condition; Shaft speed (N) = 0 – stalled rotor)

TESTING

A state-of-the-art bespoke test loop was built to be able to test the turbocharger units at full rated conditions as required by the End User. Figure 30 shows the simplified P&ID of the test loop. After considering various options, the final test loop design used two separate loops for the turbine and the pump side to allow for independent control of the turbine and pump side, with water as the working fluid. The test loop serves for three tests on the turbocharger, namely, performance test, NPSH test, and mechanical run test.

Test Loop Description

Five 1000 hp multistage pumps in parallel, each with a rated capacity of 1200 gpm at 2300 ft of head, provide required flow to test turbine. A discharge flow control valve controlled the test pump flowrate. The turbine discharge pressure and the pump suction pressure was regulated independently using two bladder-type expansion tanks rated to 250 psig (seen in the background of Figure 31). Two heat exchangers maintain the temperature on the turbine and pump side loops.

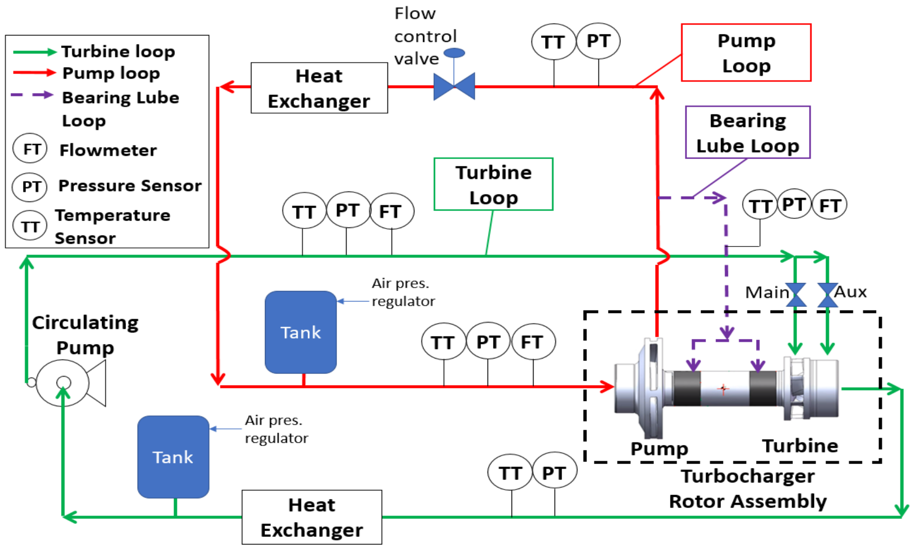


Figure 30: Simplified FAT loop P&ID

Figure 31: Turbocharger FAT setup showing instrumentation at inlets and outlets of the turbocharger

Figure 32: Turbocharger FAT setup with lubrication skid in the foreground

The bearings are lubricated by filtered water from pump discharge, using a lubrication skid as seen in Figure 32. Since viscosity of water is about 2.5 times lower than MDEA amine used at SITE, the temperature of water was maintained below 80F which is a viscosity of 0.86 cP. This necessitated the use of chillers to reject heat from the cold side of loop heat exchangers and maintain the required approach temperature.

The OEM also built a state-of-the-art Labview based data acquisition and control system to test the turbocharger which included live monitoring of turbine and pump performance and bearing condition monitoring. Table 7 shows all the key parameters measured for the FAT.

Table 7: FAT Key Instrumentation

#	Measurements	Function
1	Flow meter	Pump, turbine, and bearing lubrication flow rate
2	Pressure transmitter	Pump inlet & outlet pressure, turbine inlet & outlet pressure, differential pressure across pump & turbine journal bearings, tank pressure
3	Temperature	Bearing lubrication injection temperature downstream of heat exchanger, RTD
4	Vibration	Casing vibration measured using velometer mounted in the endcover
5	Shaft Position & Thrust Pad Wear	Shaft axial position and thrust pad wear was measured using two proximity probes
6	Speed	Shaft speed cannot be measured directly due to inaccessibility of shaft and hence will be calculated based on vane pass frequencies detected using casing accelerometer
7	Shaft power	Turbocharger shaft power is calculated based on measured differential head, flow and pump/turbine temperatures.

Turbocharger Test Procedure

The turbocharger test procedure was developed to meet API 610 and End User standards. The turbocharger underwent the following tests as part of factory acceptance tests - a) constant speed performance test b) pump NPSH test c) mechanical run test d) disassembly and inspection of bearing and critical rotating surfaces. All the turbocharger test procedures and the instrumentation & controls required to run the tests was developed by the OEM as part of a technology project before implementing for the FAT. The test procedures and acceptance criteria are explained below for reference:

Performance test
The objective is to verify the turbocharger performance meets customer operating conditions given in the turbocharger datasheet. In the field, the turbine side sees rich amine and the pump side sees lean amine. Specific gravity of rich and lean amine can differ by about 5-10%, which makes it difficult to predict the performance of the turbocharger in the field using just hand calculations. Therefore the performance test was developed to calculate the characteristic curves of the pump and turbine in water and the performance in the field with amine can be calculated using process simulation software. The key steps of the performance test are presented below.

- Turbocharger shaft speed is maintained constant by regulating flow through the turbine for different pump discharger valve openings. The shaft speed selected for the constant speed test was within +/- 20% of speed specified in the 'Performance Data' section of the datasheet (HI 14.6, 14.6.5.7.2). Witness test includes seven test pump flow rates from 10-15% of BEP flow to 120% of BEP flow.
- The test points mentioned above were taken when the turbine auxiliary valve is fully closed and fully open.
- Each test point was recorded only after the temperature in the turbine & pump loops were stabilized over 10 minutes.
- The constant speed turbine and pump water characteristics was input to OEM's Turbocharger System Analysis Software (TSAS) to evaluate 'turbocharger package' performance in process fluid at – Summer Normal & Rated operating condition, and Winter Normal & Rated operating condition. Specific gravity, speed & viscosity corrections were applied by the software based on HI 14.6.
- Turbocharger partial runaway speed was calculated using TSAS for the following scenario: turbine differential pressure at rated conditions and pump discharge at shutoff.

The turbocharger performance is considered to be acceptable if the pump differential head calculated by TSAS, as mentioned in above section is greater than -2% of the pump datasheet head. Dismantling of turbocharger is only required if turbocharger fails performance test.

Figure 33 shows the performance test results during the FAT and predictions using CFD and loss model. The prediction is in red. The performance of the turbocharger includes developed head, shaft power, and efficiency. The pump and turbine's performance are documented separately. For the turbine, the test results are shown for auxiliary valve closed fully and auxiliary valve opened fully. In general, the model predicts well the performance of both pump and turbine, except the turbine's efficiency. The data were collected for water service at 8000 rpm. The performance of the turbocharger is then scaled accordingly for amine service as noted previously using TSAS.

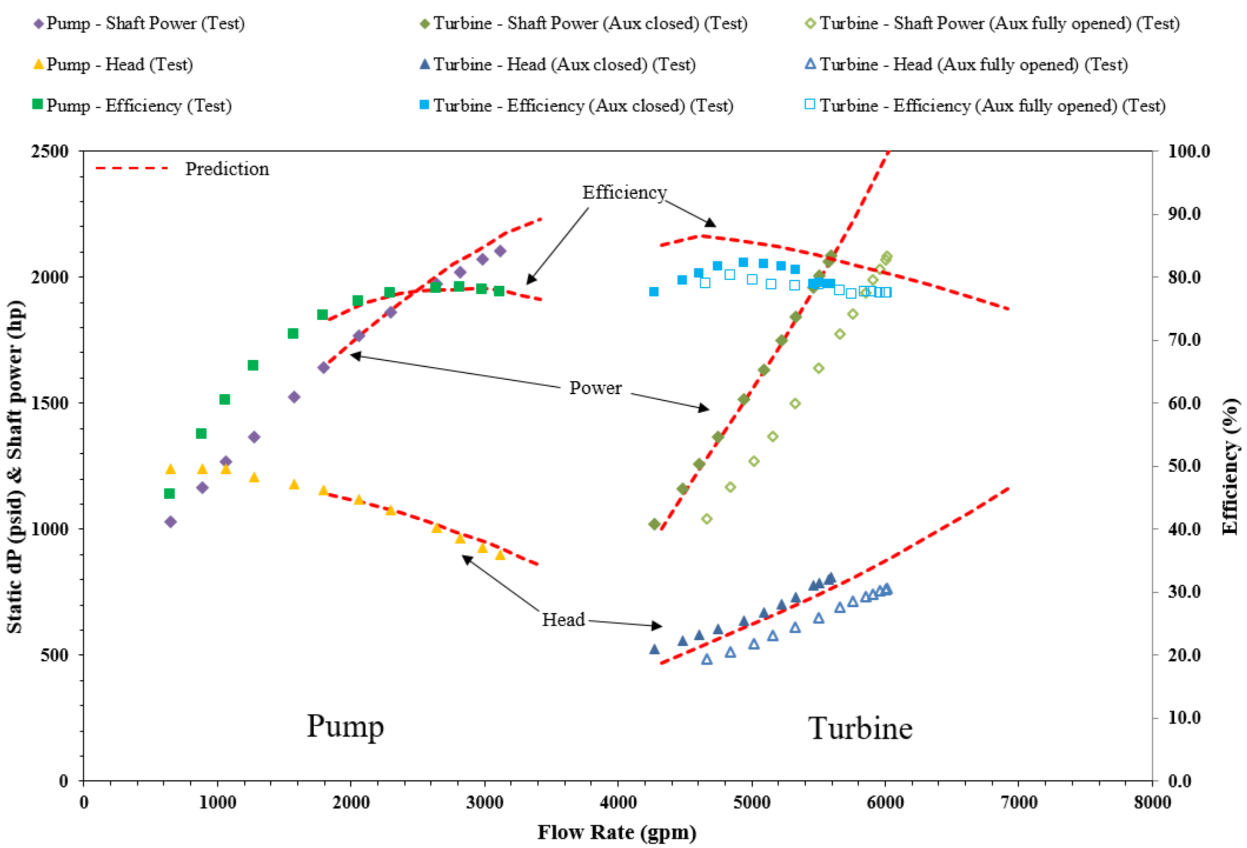


Figure 33: Turbocharger lab vs CFD+1D model predicted performance result with water

NPSH test
This test verifies the pump NPSHR at the summer rated operating conditions (which is also the maximum turbocharger shaft speed for given datasheet conditions) is sufficiently less than NPSHA specified in turbocharger package datasheets (see Table 4).

During NPSH test, the pump suction pressure is lowered (while maintaining turbine discharge pressure) until pump head drops by 3%. This results in significantly increased axial thrust on the bearings. The axial thrust bearings bearing capacity is directly proportional with the viscosity of the lubricating medium. Since the test liquid is water with a viscosity about 3 times less than the process fluid (lean amine) used for lubrication, the NPSH test will be done at 80% speed. The turbine differential pressure and pump flow rate at the reduced test speed will be calculated from affinity laws. Similarly the measured NPSHR3% will be projected back to 100% speed using conservative square law per HI 14.6.

Turbocharger was tested at the summer rated operating condition, as listed in the datasheet, and the pump NPSH procedure was as follows:

- Turbine discharge pressure will be no more than 110% of values given in turbocharger datasheet (per API 610 8.3.3.6).
- Turbocharger shaft speed will be 80% of datasheet speed
- Differential pressure across turbine & shop turbine flow control valves combined will be 64% (per affinity law) of that prescribed as specified in the datasheet, without correcting for difference of specific gravity between process fluid and test fluid.
- Turbine flow rate will be controlled using shop turbine throttle & auxiliary valves. It will be ensured that it does not exceed the value specified in the datasheet. The maximum turbine throttle & auxiliary valve openings will correspond to fully open, valve manufacturer certified Cv's skid throttle & auxiliary valves, respectively
- The pump flow rate will be set to a value 80% (per affinity law) of that specified in the datasheet using shop pump throttle valve.
- Wait until test loop as well bearing lubrication injection temperature has stabilized.
- Once the above conditions are satisfied, pump suction pressure will be decreased, while maintaining the pump flow rate & the turbocharger shaft speed. The pump suction pressure, at which the pump differential head drops by 3%, will be recorded using appropriate formulae as net position suction head required (NPSHR3) at rated conditions. Temperature of water at pump inlet will also be recorded and used to calculate vapor pressure of water.

As discussed in the test procedure, during NPSH test, the pump suction pressure is lowered (while maintaining turbine discharge pressure) until pump head drops by 3%. The turbocharger was operated stably at 6440 rpm and 2212 gpm of pump flow during NPSH test. As shown in the Figure 34, the TDH dropped 3% when NPSH reached 121ft.

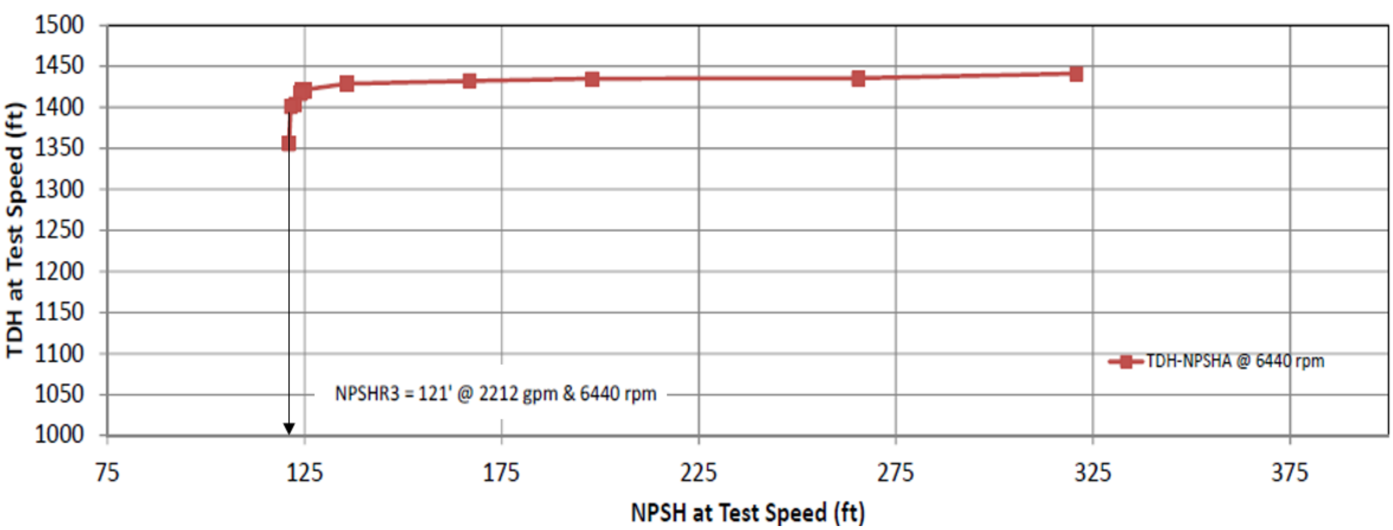


Figure 34: Pump NPSH lab test result

Mechanical run test
The objective is to demonstrate the satisfactory mechanical operation of the turbocharger at the rated conditions, including vibration levels, lack of leakage from gaskets, adequate lubrication flow and free-running operation of rotating assembly. Turbocharger will be tested at the summer rated operating condition, as listed in the datasheet, and the Mechanical run test will be as follows:

- Pump suction & turbine discharge pressure will be no more than 110% of values given in datasheet (per API 610 8.3.3.6).
- Differential pressure across turbine & shop turbine flow control valves combined will be prescribed as specified in the datasheet, without correct for any difference of specific gravity.

- Turbine flow rate will be controlled using shop turbine throttle & auxiliary valves. It will be ensured that it does not exceed the number specified in the datasheet. The maximum turbine throttle & auxiliary valve openings will correspond to fully open, valve manufacturer certified Cv's skid throttle & auxiliary valves, respectively
- The pump flow rate will be set to the value specified in the datasheet using shop pump throttle valve.
- Turbocharger speed, and shop turbine throttle & auxiliary valve command positions will be recorded for informative purposes only.
- Once the above conditions are satisfied, the turbocharger will be run continuously for 1 hour. Record data per sample test report every 15 mins.

The mechanical performance is considered to be acceptable when each of the following is achieved:

- Vibration does not exceed limits specified by API 610, (Overall RMS: 0.18 in/s, Discrete RMS: 0.12 in/s).
- Sound pressure level (SPL) not to exceed 85 dBA per datasheet.
- No visible leakage through turbocharger pressure boundary i.e. case, end cover and mating flanges.
- Dismantling of turbocharger is only required if turbocharger fails mechanical run test

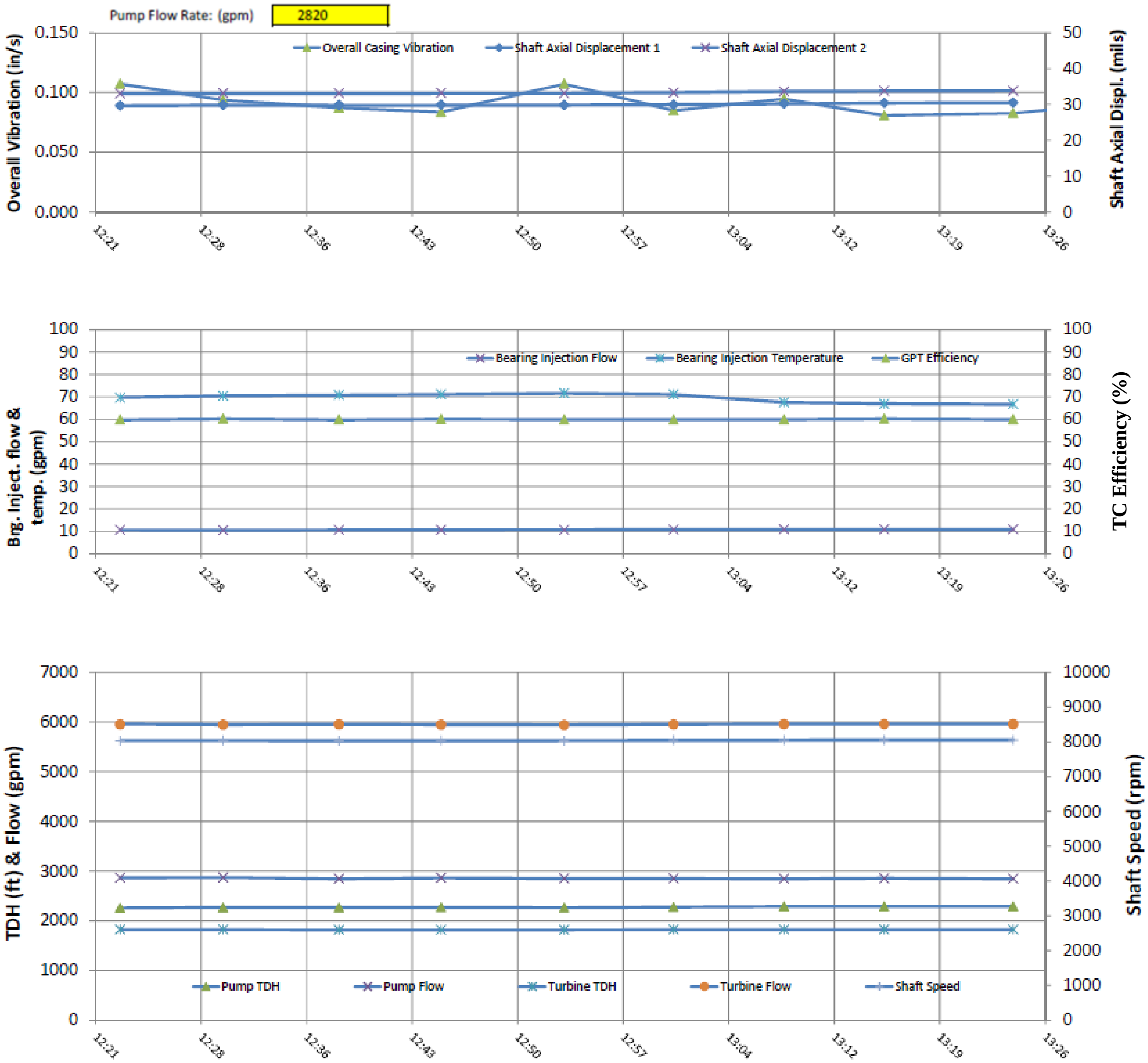


Figure 35: Mechanical run – trend plots

Trend plots for the mechanical run are shown in Figure 35. In general, the mechanical run monitors the following parameters: turbine & pump TDH & flow, shaft axial displacement, casing vibration, bearing flow and temperature and shaft speed of the turbocharger. As shown in the figure, all trends are stable and within the acceptance criteria.

The data from the factory acceptance test was analyzed, and projected at field amine conditions using TSAS and compared against the acceptance criteria set in the test procedure, as seen in Table 8. The HSBH Unit 1 passed all the test acceptance criteria, and the project partial runaway speed of the turbocharger, as defined previously, was calculated as 11,890 rpm which is less than that the bearings are capable of.

Table 8: HSBH Unit 1 turbocharger factory acceptance test criteria and results

Turbocharger Performance Test Acceptance Criteria [1]					
	Units	Summer		Winter	
Turbine Side		**Rated**	**Normal**	**Rated**	**Normal**
Flow	gpm	6566	5969	6457	5870
Sp. gravity	-	0.947		0.95	
Viscosity	cP	1.73		1.871	
Diff. head	ft	-1922	-1928	-1917	-1923
Pump Side		**Rated**	**Normal**	**Rated**	**Normal**
Flow	gpm	2820	2563	2778	2525
Sp. gravity	-	1.001		1.002	
Viscosity	cP	2.784		2.841	
TDH (Total differential head)	ft	2148	2200	2151	2210
Throttle Valve Opening	%	100%	100%	100%	100%
Auxilary Nozzle Opening	%	100%	57%	100%	43%
Bypass Nozzle Opening	%	13%	0%	10%	0%
Pump TDH calculated by TSAS*	ft	2414	2573	2451	2575
Allowable tolerance on Pump TDH	%	-2% (Per 31-SAMSS-004, 8.3.3.4). There will be no positive tolerance to allow for maximum turbocharger recovery per Note 12 in datasheet.			
Pump TDH deviation	%	12.4%	17.0%	13.9%	16.5%
Pass / Fail?	-	PASS	PASS	PASS	PASS

Projected partial runaway speed of turbocharger	11,835	rpm

Note: Runaway speed is calculated using TSAS when turbine is at rated conditions and pump is at shut-off .

Note: 1. Turbocharger performance corrected for SG, viscosity and shaft speed per datasheet conditions based on Performance Test Log Data input to TSAS (Turbocharger System Analysis Software) - (see test log).

Disassembly and inspection

The HSBH Unit 1 turbocharger was initially tested with tungsten carbide (WC) journal bearings and PEEK thrust pads as requested by End User to be able to handle debris in process fluid. However even though the turbocharger passed the performance test failed the disassembly and inspection test as rub marks were observed post mechanical run test at the journal bearing & wear ring locations, as seen in Figure 36 and Figure 37.

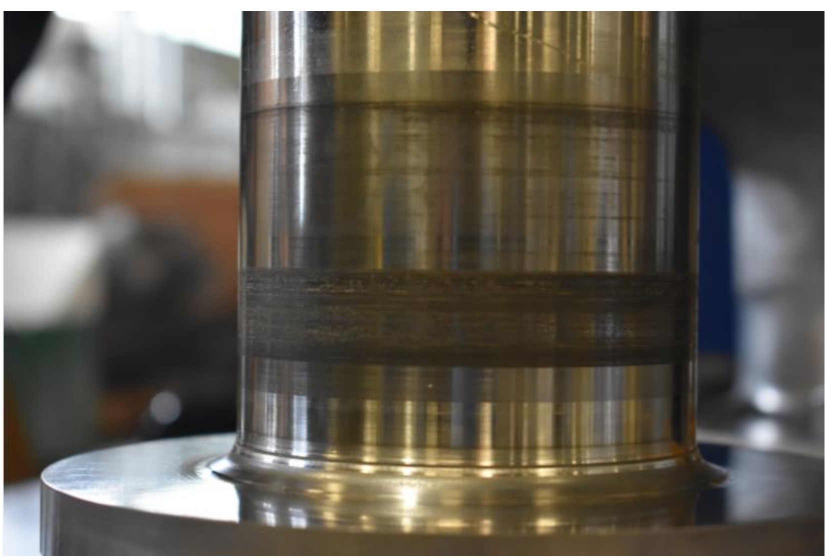


Figure 36: Shaft showing rub marks at journal bearing location when tested with WC bearings

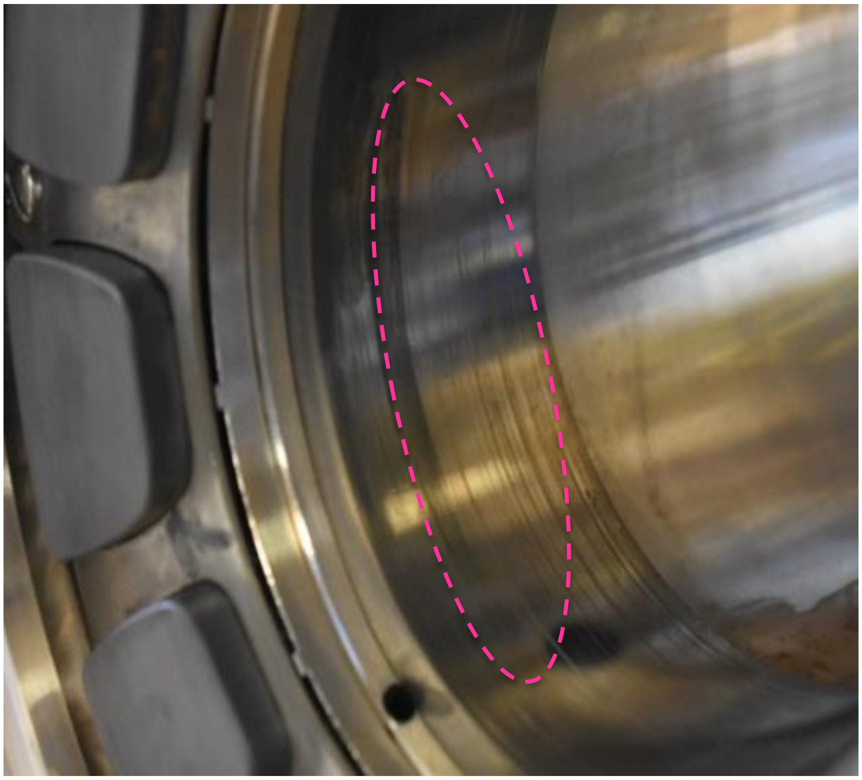


Figure 37: Rub marks post-test on WC journal bearings on turbine side

A thorough and detailed root cause analysis was performed, and the key findings where:

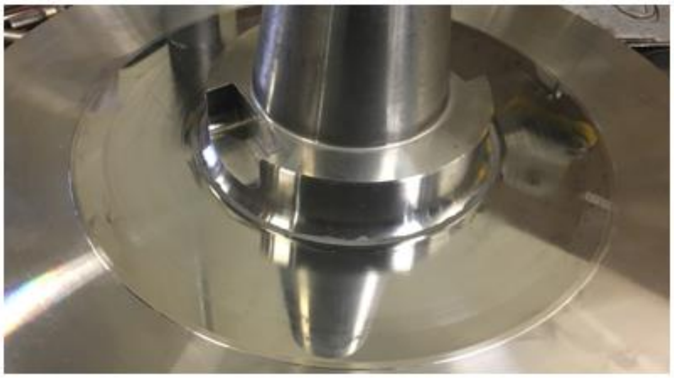
- The turbocharger passed the performance test and the absence of any high vibrations indicated that the bearing rubs were most likely a transient phenomena
- A thorough CMM inspection of the journal bearing and wear rings indicated that they were misaligned. This meant the eccentricity ratio of the impeller in the wear rings was greater than 0.5 during start-up, which could result in negative stiffness at the wear rings based on research by Arghir et al (2014), especially at transient conditions when the speed is still low
- Higher than initially anticipated radial loads could cause tilting of the rotating assembly, further increasing the eccentricity ratio of the impeller in the wear rings
- Hard/hard bearing material pair of tungsten carbide on tungsten carbide meant during transient events such as start-up and shutdown, there is possibility of contact before hydrodynamic bearing film lifts off the rotor which could result in a thermal runaway event.

The following fixes were implemented to resolve the bearing rubs:

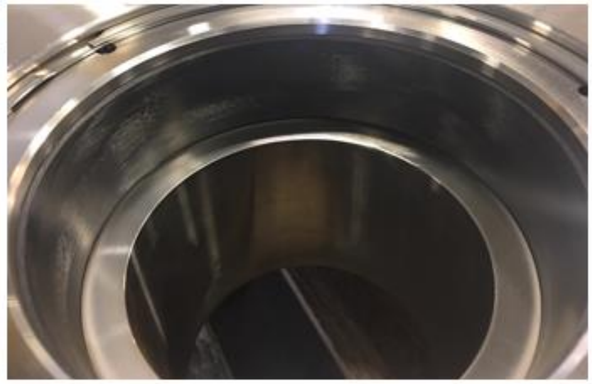
- The wear rings were machined concentric with respect to the journal bearing within 0.001” with side volutes assembled with the center volute, to reduce eccentric at the wear rings
- The rotating assembly imbalance was improved
- The bearing material was changed from WC/WC to PEEK composite with chamfered edges to prevent edge loading, and to allow for partial dry-running conditions during transient events such as start-up and shutdown.

The HSBH Unit 1 turbocharger was retested and then disassembled. Figure 38 and Figure 39 show all the bearings and their counter running surfaces in good condition – thus validating the fixes. Subsequently all the remaining turbochargers were modified and all passed the factory acceptance test by May 2018.

a) Turbine back shroud (running face for active thrust bearing) & shaft surface in good condition after lab test

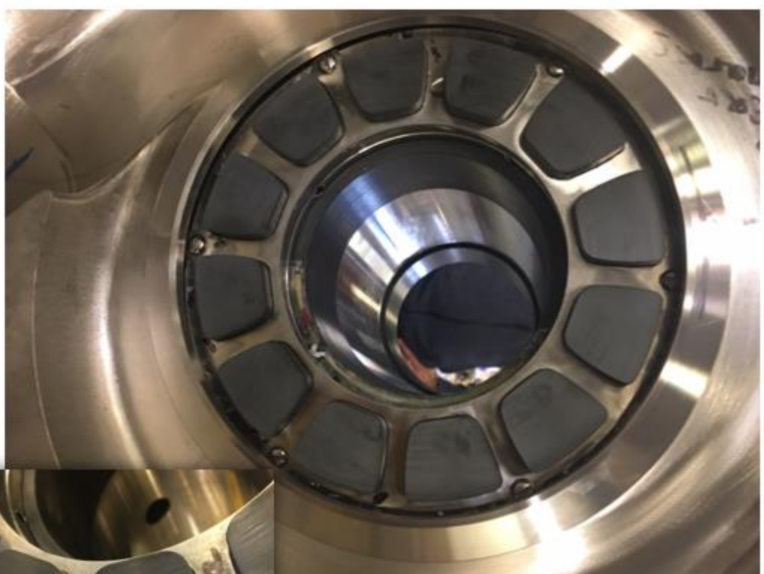


b) Pump back shroud (running face for inactive thrust bearing) in good condition after lab test

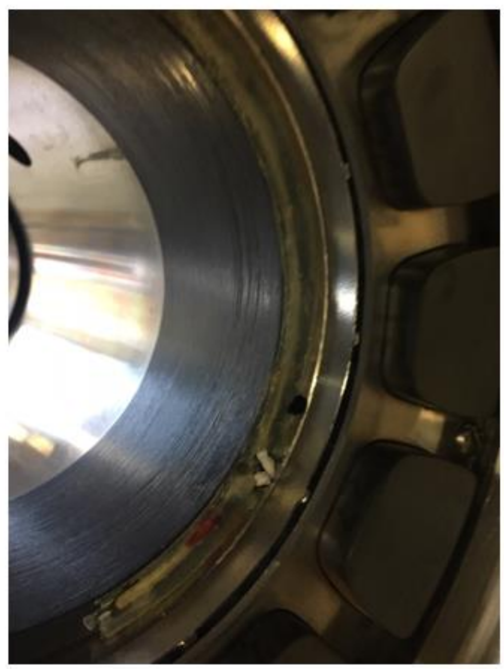


c) Pump wear ring in good condition after lab test

Figure 38: Rotating assembly running surfaces and wear ring are in good condition post mechanical run test after bearing rub issue was fixed



a) Active thrust bearing pads after lab mechanical run test, inset shows close-up of thrust pad in good condition after lab test



b) Journal bearing in good condition after lab mechanical run test

Figure 39: Thrust and journal bearings are in good condition post mechanical run test after bearing rub issue was fixed

COMMISSIONING

The turbochargers were then assembled on to the SKID and then shipped to the FGP in the middle-east for commissioning. The turbocharger units were installed and fully commissioned by the End User and EPC in early 2020.

In the past two hydraulic turbocharger were implemented in AGR plant – both used uni-directional process lubricated hydrostatic bearings. As noted by Shirazi et al (2016), a 750 gpm, 400 hp lean amine turbocharger was installed in a Texas AGR plant in 2008 and ran for more than 12 years without any maintenance for the turbocharger. In 2012, two turbochargers were designed for pumping 1300 gpm, 440 hp of lean amine were installed at a plant in Asia, however "within a few minutes of starting the units, the bearing lubrication filters were clogged, forcing a shutdown of the turbocharger. Krish P., Shriazi M. et al (2015) explain that a significant amount of metal and inorganic debris was found in the turbocharger as well as in the filtration media. It was hypothesized that incomplete flushing of the process piping was the root cause of this debris. Due to loss of lubrication flow, debris from the process fluid had made its way to the bearing surfaces and had caused damage to the bearings and the rotating assembly".

Based on this experience, it was recommended by the End User that the AGR system is clean and stable before starting the turbocharger. Starting a new facility is a challenging activity. It comes at the end of multiple parallel construction, testing, cleaning, and commissioning activities. Hydrocarbons (and H2S) at high pressure are introduced to the facility for the first time. The "Soft Startup" concept calls for the proactive testing of a facility using safer mediums well ahead of mechanical completion. This has been proven to eliminate operational and safety risks pertaining to equipment, piping and personnel.

Soft startup process calls for the proactive and full testing of a facility using safer mediums well ahead of Mechanical Completion Certificate (MCC). FGP successfully implemented it in all Gas Treatment (GT) trains, where amine circuits were filled with demineralized water (instead of amine), then pressurized by Nitrogen (instead of flare gas) up to the operating pressure. This was done for multiple safety and reliability reasons: to test system for leaks with a safer medium, debug Process Control and ESD Systems, test the functionality and reliability of rotating equipment, clean the system from remaining debris, and to limit the potential for foaming incidents during future operations.

The adoption of the Soft Startup Concept at FGP is one of the major contributors to the safe and reliable startup of all FGP facilities. As a result, all FGP facilities where the "soft startup" process was applied were subsequently put in service safely and reliably, without any process, DCS, instrumentation interruptions or rotating equipment failures.

After completing the soft start, the amine loading to the train was started along with the cold and hot amine circulation. Then the train was kept running without turbocharger for at least 4 months to assure that all piping is cleaned especially during hot circulation. Also, it was made sure that the control system is stable and tuned.

The following steps were followed during turbocharger commissioning preparation:

1. Conduct borescope inspection for internal part of turbocharger through turbocharger casing plugs before introduce any liquid to Turbocharger.
2. Conduct borescope inspection for the strainer and commissioning mesh availability at both sides (Pump and Turbine sides).
3. Remove lubrication filter elements and conduct water flushing for the line upstream the filter till upstream the turbocharger bearings.
4. Conduct another flushing by amine through opening the external lubrication line without filter elements and should be isolate it before turbocharger bearings by venting the amine to safe location.
5. Before introducing rich amine to turbine, isolate the turbocharger rich side, and conduct several flushing for turbine upstream line through opening and closing the suction isolation valve of the rich side. High pressure hose should be utilized to drain the line to safe location.
6. Conduct step (4) for the lean amine side (Pump Side).
7. Open Lean Amine line valve gradually and drain the turbocharger casing.
8. Install lubrication filter elements and open the auxiliary lubrication line to bearings. Then, open the discharge line drain of turbine side to safe location.
9. Open Lean Amine line valve gradually and drain the turbocharger casing.
10. Vent all instrumentation transmitters and ensure its running healthy.
11. Make sure that Lean Amine Booster Pumps A/B/C are in service (2 out of 3)
12. Open Lean Amine line valve gradually and drain the turbocharger casing.
13. Prime Turbocharger pump side.
14. Ensure the lubrication flow is within design readings.
15. Open Turbine suction isolation valve gradually and monitor pump and turbine pressure.

OPERATING EXPERIENCE

After the commissioning of the turbochargers was complete, and the train was running with the turbochargers for about 6 months, the end user shared one month of operating data with the OEM.

Figure 40 shows some of the key operating parameters of the AGR plant. The data shows that the control system smoothly transitioned the plant operation from one electric circulation pump and turbocharger-based energy recovery system to two electric circulation pumps and rich amine level control valve on 15 October 2020, and back the same day. On 8 Oct, lean amine circulation pumps was swapped without affecting the turbocharger or plant operation in any way. From 22-27 Oct, the total lean amine flow was met by two circulation pump (50% of flow) and the rest by the turbocharger pump. This shows the flexibility offered while using the turbocharger-based energy recovery system with electric-driven circulation pumps.

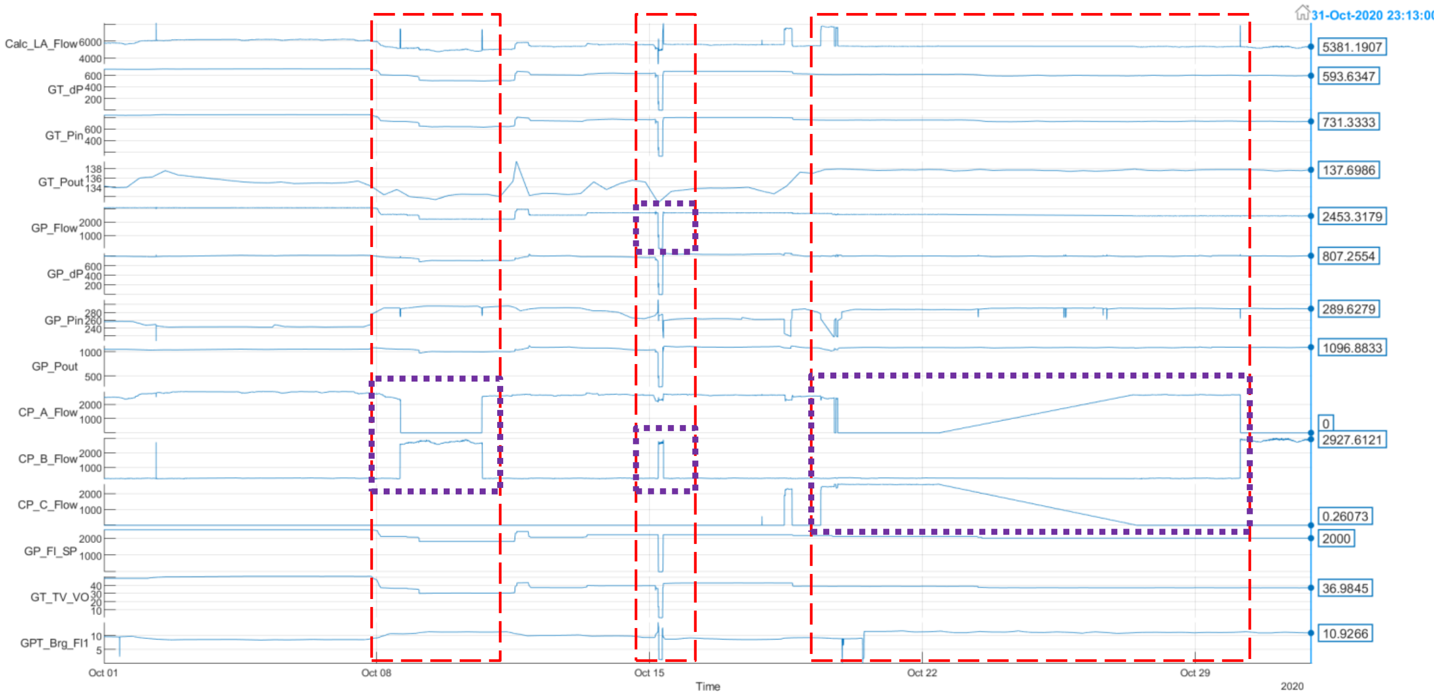


Figure 40: HSBH Unit 1 AGR plant – key operational parameters for the month of October 2020, showing switch-overs between circulation pump and turbocharger

The variables plotted are as follows:

- Calc_LA_Flow is calculated lean amine flowrate (gpm),
- GT_dP is turbine differential pressure (psid), GT_Pin is turbine inlet pressure (psig),
- GT_Pin is turbine inlet pressure (psig)
- GT_Pout is turbine outlet pressure (psig),
- GP_Flow is turbocharger pump flowrate (gpm),
- GP_dP is turbocharger pump differential pressure (psid),
- GP_Pin is turbocharger pump inlet pressure (psig),
- GP_Pout is turbocharger pump outlet pressure (psig),
- CP_A_Flow, CP_B_Flow, CP_C_Flow is lean amine circulation pump A, B, C flow rate (gpm) respectively,
- GP_Fl_SP is turbocharger pump flow setpoint (gpm),
- GT_TV_VO is turbine throttle valve opening (%),
- GPT_Brg_Fl1 is turbocharger bearing flow rate (gpm).

Figure 41, shows zoomed in view of the data showing how the turbocharger was shutdown by closing the throttle valve and circulation pump B was brought online from 5 am to 6am, after which the turbocharger is isolated and bearing lubrication flow also drops, then after some time the turbocharger was brought online 8am. Note in this case the End User chose to initiate the bearing lubrication flow about 10 minutes before starting the turbocharger with the help of the external lubrication line from the circulation pumps – this ensures the internal bearing flow passages are flooded with filtered temperature-controlled lean amine and prevents any mixing of rich into lean amine.

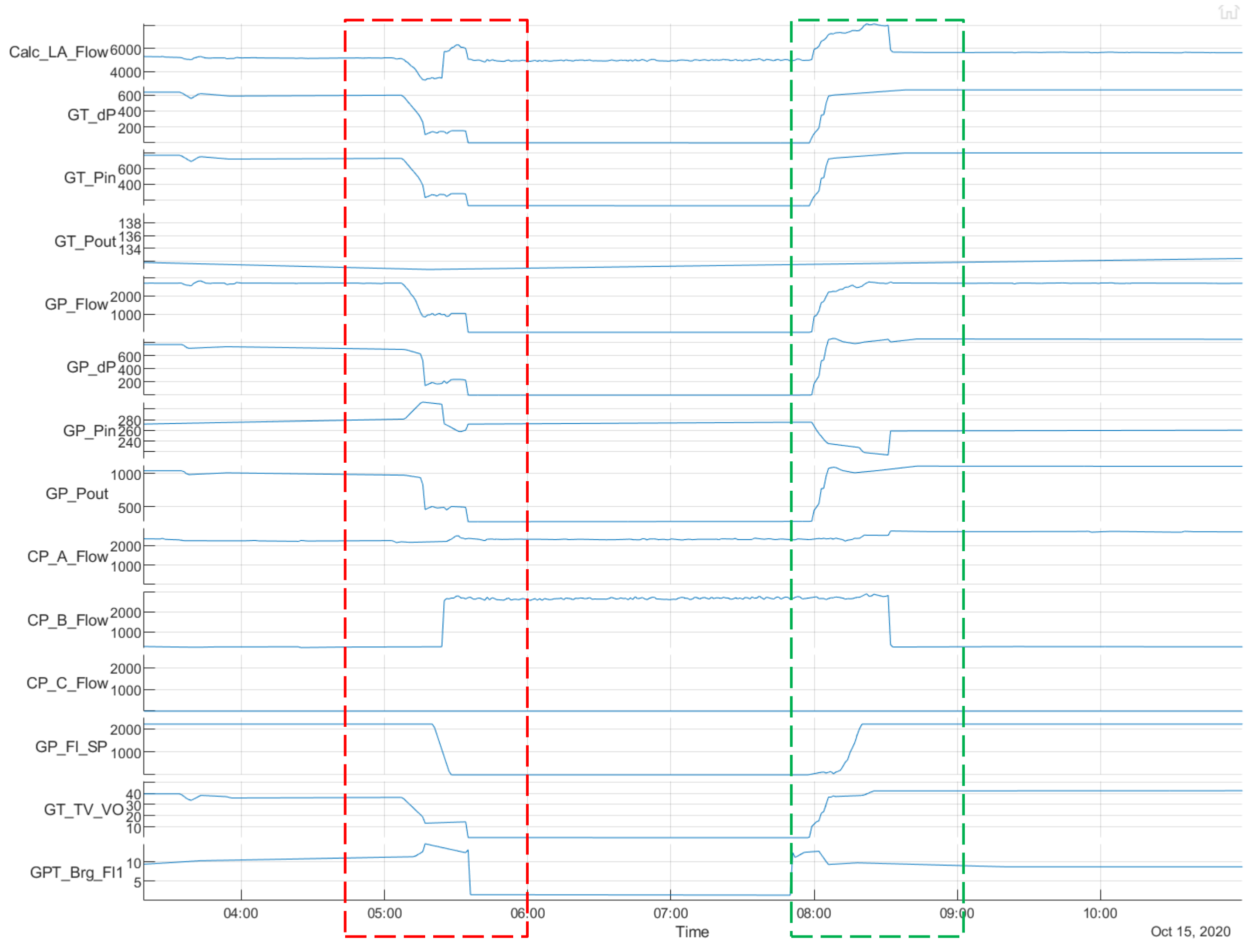


Figure 41: HSBH Unit 1 AGR Plant - October 15, 2020 field data which shows shutdown (5-6 am) and start-up (7:45-8:45a m) of the turbocharger

In the field turbine side rich amine flowrate and shaft speed was not measured, hence turbocharger efficiency and where is the pump and turbine running with respect to its best efficiency point could not be calculated. However based on the operating data it is estimated that HSBH unit 1 is running at around 7000 rpm which is 90% of rated speed. Over this period, Figure 42 and Figure 43 shows the turbocharger pump lean amine flow varied between 2000-3000 gpm and provided 50% of the total lean amine flow into the contactor and the rest being supplied by the electrically driven circulation pump. This confirms the turbocharger based energy recovery system as designed is capable of saving at least 50% of power consumption for pump lean amine into the contactor, while maintaining contactor level using the turbine at the same time. In the absence of measured rich amine flowrate, the turbine flowrate is assumed same as the total lean amine flowrate going into the contactor - to understand the hydraulic energy input to the turbine. Note this could result in under-estimating the turbine flow rate by few percentage points depending on the volumetric expansion of lean to rich amine due to chemical absorption of the impurities by the amine solvent.

Figure 42: Operational data of turbocharger and circulation pump lean amine flow rates of HSBH Unit 1 AGR plant for Oct 2020

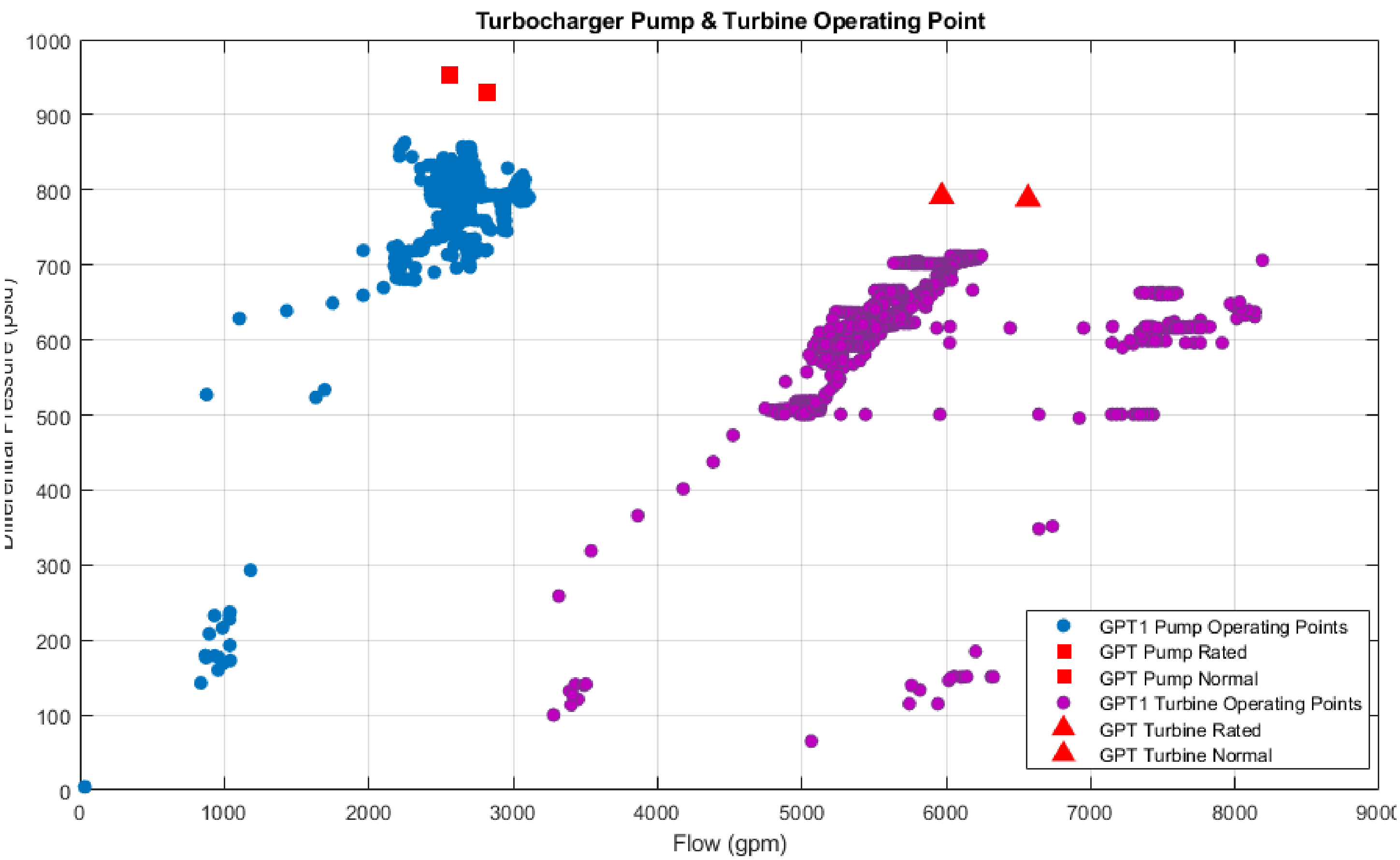


Figure 43: HSBH Unit 1 AGR Plant turbocharger operational data for October 2020

For the turbocharger to function as-designed the most important thing to confirm is that the bearing lubrication flow is greater than the minimum required for proper functioning of the bearings. Figure 44 shows the turbocharger casing vibration and lean amine bearing lubrication flow – the lubrication flow is always greater than the minimum required flow of 5-6 gpm, except for transients such as start-up and shutdown. The vibration observed in the field is lower than during FAT. This could be explained by the fact water was used as the lubrication fluid, whereas in the field lean amine is the lubricating medium and it has higher viscosity than water.

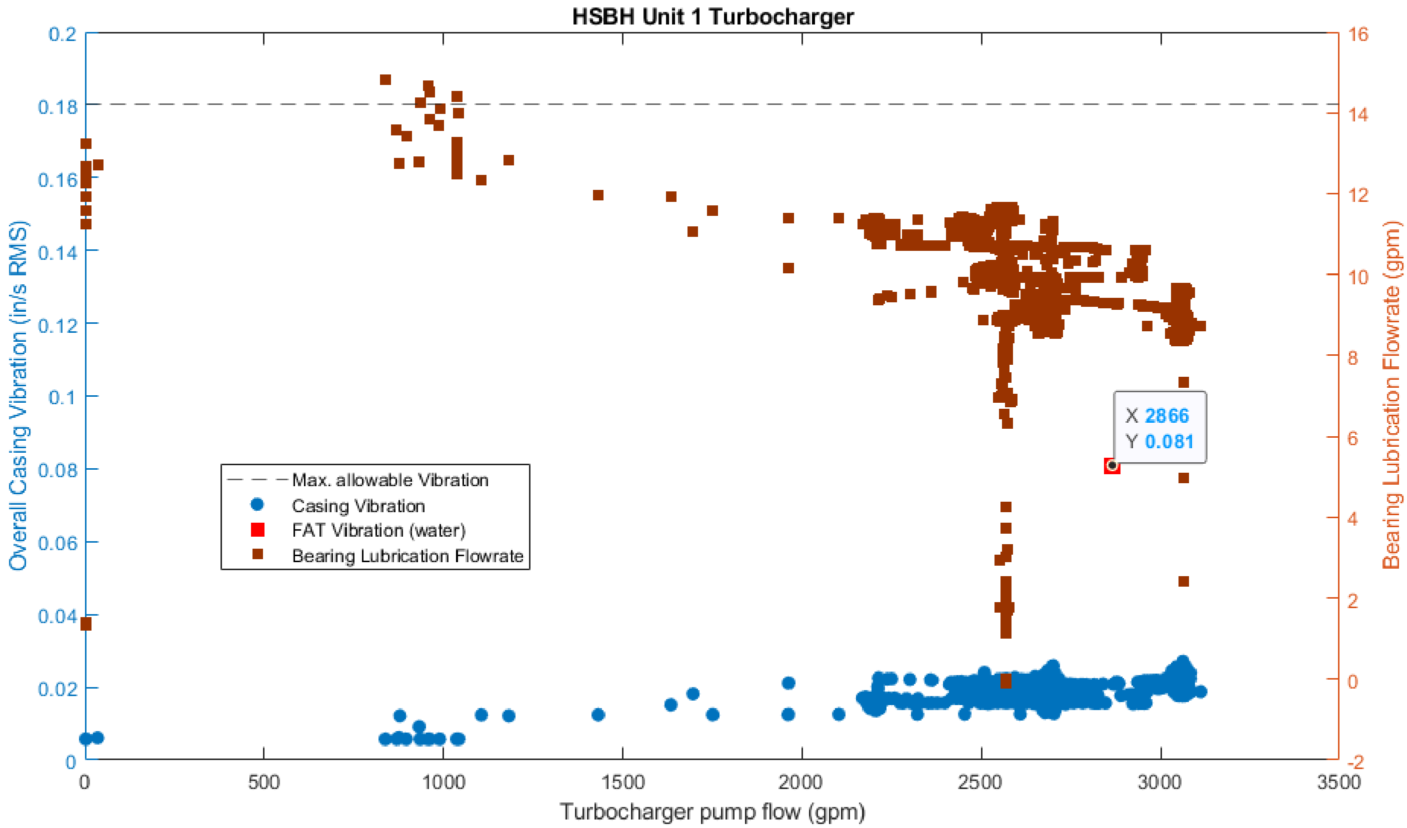


Figure 44: HSBH Unit 1 AGR Plant - turbocharger, casing vibration & bearing lubrication flow rate for October 2020

The End User observed some additional operational benefits of using a turbocharger at FGP, which are as follows:

- During plant start up very high vibrations were observed at the let down valves, the operations teams elected to fix the issue by adding many additional pipe supports but the issue was not fully eliminated. In addition, after five months of operation there was a leakage in the downstream pipe of the let down valve, it was decided to start the HSBH-1 turbocharger ahead of plan to avoid the plant shut down and work on parallel on the let down station.
- SITE had power dip issue which cause the electric motor pump to shutdown, while the turbocharger kept running for more than 2 minute before rich amine in the contactor could reach the low-low level. During this upset condition, 2 minutes gave us sufficient time to restart the circulation pumps, and prevented a complete shutdown of the gas treatment plant.
- Additional advantage of using the turbocharger is the significant noise reduction in the gas treatment unit compared to the let down valves with two electric motor lean amine circulation pumps.

CONCLUSION

This hydraulic turbocharger is the first known application which uses bi-directional tilt-pad hydrodynamic bearings – which can be concluded to be operating very well after over one year of successful operation with no down time. Thus a new 2000 hp hydraulic turbocharger based energy recovery system was developed for AGR process. This new energy recovery solution is extremely compact, requires no alignment, is seal-less, requires no support systems, has very few moving parts compared to existing energy recovery solutions, which results in a highly reliable device. The successful operation of the four HSBH units from March 2020, for over one year validates the design of hydraulic turbocharger developed, and reinforces the need of a close working partnership between the OEM and the End User.

During COVID pandemic, due to travel restrictions in place for travel into Kingdom of Saudi Arabia, engineers from the OEM could not travel to support the commissioning effort. The successful commissioning of the turbochargers indicates the use of proper startup procedures implemented by the End User is important, since it ensured clean filtered process fluid is supplied to the process lubricated turbocharger bearings. In addition, commissioning of the turbocharger in the absence of OEM personnel shows the ease of installation (no alignment necessary), operation & control of hydraulic turbocharger energy recovery system. This new energy recovery system installed at four HSBH AGR plans helped replace an API 610 multi-stage lean amine circulation pump and a level control valve, recover more than 50% of power consumption of the lean amine circulation pumps, reduce CO2 emissions, while also improve working conditions by reducing noise level at the plant.

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NOMENCLATURE

AGR	=	Acid Gas Recovery
ASME	=	American Society of Mechanical Engineering
API	=	American Petroleum Industry
BPVC	=	Boiler Pressure Vessel Code
CAD	=	Computer-Aided Design
CFD	=	Computational Fluid Dynamics
CFCV	=	Contactor Flow Control Valve
CMM	=	Coordinate Measuring Machine
DCS	=	Distributed Control System
ESD	=	Emergency Shutdown
EPC	=	Engineering, Procurement & Construction Firm
FAT	=	Factory Acceptance Tests
FEA	=	Finite Element Analysis
FMEA	=	Failure Mode Effect Analysis
FGP	=	Fadhili Gas Plant
GT	=	Gas Treatment
HI	=	Hydraulic Institute
HPAP	=	High Pressure Amine Pump
HPRT	=	Hydraulic Power Recovery Turbine
HGP	=	Hawaiyyah Gas Plant
HS	=	Hydrostatic
HSBH	=	Hasbah
KRSN	=	Khursaniyah
LCV	=	Level Control Valve
MCC	=	Mechanical Completion Certificate
MAWP	=	Maximum Allowable Working Pressure
NPS/DHA	=	Net Positive Suction or Discharge Head
OEM	=	Original Equipment Manufacturer
RAM	=	Reliability, Availability & Maintainability
RO	=	Reverse Osmosis
SWRO	=	Sea-water Reverse Osmosis
TC	=	Turbocharger
TPTB	=	Tilt-pad Thrust Bearing
TSAS	=	Turbocharger System Analysis Software

H	=	Head	[ft]
HP	=	Power	[hp]
Q	=	Flow	[gpm]
M	=	Torque	[ft-lbf]
N	=	Shaft speed	[rpm]
dP	=	Differential Pressure	[psid]
P	=	Pressure	[psig]
T	=	Temperature	[F]
S_{ut}	=	Ultimate Tensile Strength	[ksi]
$S_{eqvlt_{alt}}$	=	Equivalent Alternating Stress	[ksi]
σ	=	Stress	[ksi]

REFERENCES

API Standard 610, September 2010, “Centrifugal Pumps for Petroleum, Petrochemical and Natural Gas Industries”, Eleventh Edition, *American Petroleum Institute*

American Society of Mechanical Engineers. (2010). “ASME boiler and pressure vessel code, Section VIII – Rules for construction of Pressure Vessels”. New York: American Society of Mechanical Engineers, Boiler and Pressure Vessel Committee

Arghir, M., Mariot A., 2014, “About the negative direct static stiffness of highly eccentric straight annular seals,” *Proceedings of ASME Turbo Expo*, Paper No. GT2014-27070.

Karassik, I. J., Messina J. P., Cooper P., Heald C. C., 2000, "Pump Handbook," *Third Edition, McGraw-Hill*, New York, New York, USA.

Gopalakrishnan, S., 1986, "Power Recovery Turbines for the process industry," *Proceedings of the Third International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 3-11.

Gulich, J. F., 2010, "Centrifugal pumps," Second Edition, Springer Publishing, New York, New York, USA.

Haji M., 2015 "Hydraulic Energy Recovery Technology (Iso-Gen) Pilot Test Report," Internal End User Report

HI Standard 14.6, 2011, "*Rotordynamic Pump for Hydraulic Performance Acceptance Tests*", Hydraulic Institute

Kadaj, E., Bosleman E., 2018, "Renewable Energy Powered Desalination Handbook: Chapter 11 - Energy Recovery Devices in Membrane Desalination Processes," First Edition, Elsevier Inc., Amsterdam, Netherlands.

Krish, P., Shirazi, M., A., Gains-Germain, A. 2015, "Optimizing Amine Process Design Using Liquid Phase Turbochargers," *Proceedings of ASME Turbo Expo*, Paper No. GT2015-42774.

Krish, P., Gains-Germain, A., Thorp, J., Martin, J., and Shirazi, M., 2015, "Optimizing Amine Process Design Using Liquid Phase Turbochargers," *Proceedings of ASME Turbo Expo*, Paper No. GT2015-42774.

Marscher, William D., 2002, "Avoiding Failures in Centrifugal Pumps," *Proceedings of the Nineteenth International Pump Users Symposium*, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 157-175.

Pattom M.; Deshpande C.; Martin J., 2020, "System and method for monitoring operating condition in a hydraulic turbocharger", Patent US10712235B2

Schiavello, Bruno; Visser, Frank C. (2009). Pump Cavitation: Various NPSHR Criteria, NPSHA Margins, Impeller Life Expectancy. Texas A&M University. Turbomachinery Laboratories.

Shirazi, M., 2016, "Hydraulic Turbochargers in the Acid Gas Removal Units at Natural Gas Processing Plants," *Proceedings of the Laurance Reid Gas Conditioning Conference*, Norman, Oklahoma, USA.

Winkler, F., 2014, "Reducing Electricity Consumption of Amine Gas Sweetening Using a Variable Speed Hydraulic Turbogenerator", *Presented at the 93rd GPA Annual Convention, April 13-16, 2014 – Dallas, TX*

