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**OPTIMIZING AMINE PROCESS DESIGN USING LIQUID PHASE
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Energy Recovery
San Leandro, CA, USA**ABSTRACT**

Acid gas removal is a critical process step in natural gas processing and syngas production for ammonia and other uses. Application of a liquid phase turbocharger to the acid gas removal unit (AGRU) results in significant energy savings and improvement to reliability, availability and maintainability (RAM) of the plant. This paper describes conventional configurations with high pressure pumps and new configurations utilizing liquid phase turbochargers. Design of the equipment, process operations and controls and reliability analysis are included. The results of a RAM study comparing conventional configurations to those incorporating liquid phase turbochargers in multiple cases are also presented. From the RAM study, it can be concluded that flow sheet configurations that include a liquid phase turbocharger consistently provide lower plant downtime and maintenance costs as compared with conventional flow sheet configurations. This is in addition to the energy savings that result from energy recovery with the application of the liquid phase turbocharger to the AGRU. For the reference plant used in the study, the maintenance cost savings are as great as \$2.5M over the 20 year lifetime of the plant and average annual downtime reduction is as much as 19.8 hours.

INTRODUCTION

Natural gas is an abundant, reliable, and clean-burning source of energy that is typically processed to remove acid gases such as carbon dioxide and hydrogen sulfide before it is ready for distribution and use. A common acid gas removal (AGR) process uses an amine solvent to absorb acid gases in a high pressure contactor column. The pressure is then decreased for acid gas stripping in the regenerator. The opportunity exists to use a liquid phase turbocharger to recover the energy wasted in the pressure letdown and transfer it to the low pressure amine exiting the regenerator. This eliminates the need for a high pressure pump – providing energy savings, maintenance savings, and positively impacting plant availability. Figure 1 is a simplified process flow diagram of a typical acid gas removal unit.

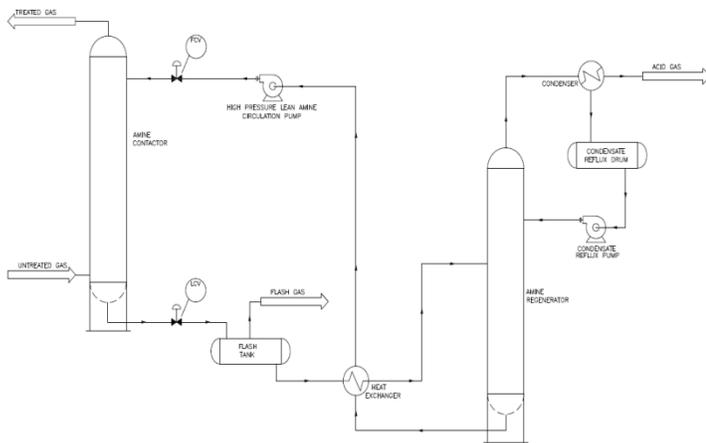


Figure 1: Simplified Acid Gas Removal Unit

In an amine based AGR process, the contactor column may typically operate at pressures up to 1100 psi (76 bar) and the amine regenerator operates at pressures closer to atmospheric. The contactor column is operated so as to maintain a liquid pool of amine at the bottom of the contactor, with the untreated gas entering the column above this liquid level. Maintaining the liquid level becomes a critically important operational requirement so that the gas is forced to move up the column and out, rather than exiting the bottom of the column and moving towards the flash tank. Various factors in the amine contactor impact the moment to moment amine holdup within the contactor column and the level control valve (LCV) is used in a feedback loop with liquid level measurement in the column to control the liquid level between the desired limits.

The high pressure lean amine circulation (HPLAC) pump pressurizes the lean amine to some level above the contactor column pressure and the flow into the column is controlled by a flow control valve (FCV) which throttles the flow from the HPLAC pump. These pumps are typically multistage centrifugal pumps. Multistage centrifugal pumps require significant regular maintenance and are subject to multiple failure modes, including those related to seals, couplings, and external oil lubrication systems. To minimize the risk of plant downtime, these pumps are commonly installed in a redundant configuration.

NOMENCLATURE

Turbocharger Nomenclature

- $P_{p,out}$ = Pressure at pump outlet
- $P_{t,out}$ = Pressure at turbine outlet
- $P_{p,in}$ = Pressure at pump inlet
- $P_{t,in}$ = Pressure at turbine inlet
- Q_p = Pump side Flow
- Q_t = Turbine side Flow

General Nomenclature

- $\sigma_{T,ult}$ = Ultimate tensile stress
- λ_p = Total pump failure rate
- λ_{SE} = Total failure rate for all pump seals
- λ_{SH} = Failure rate for the pump shaft
- λ_{BE} = Total failure rate for all pump bearings
- λ_{CA} = Failure rate for the pump casing
- λ_{FD} = Failure rate for the pump fluid driver

Conventional flow sheet and redundancy configurations

In order to assure continuous operation of the plant and to avoid any possibilities for unplanned shutdown due to equipment failures, equipment redundancy is engineered into the plant design.

LCV redundancy is typically achieved by using a standby LCV in parallel with the operating LCV. Each LCV is a “fail-close” device – meaning that if failure occurs, the valve will close. The standby LCV would then open up to maintain the level control function. LCVs tend to be quite reliable with mean time between failures (MTBFs) in excess of 30 years.

Pump redundancy is also critical for plant operations due to the much lower reliability of pumps, typically with MTBFs of less than 4 years as characterized by Offshore and Onshore Reliability Data (OREDA) organization [1]. One configuration for pump redundancy is to have one operating pump and another identical pump in parallel as a standby. Rapid switchover between pumps is engineered into the plant design and operation. Due to the relatively high capital expense associated with pumps, a second redundancy configuration is often considered. In this second configuration, three identical pumps are used. Each pump is sized to provide the full pressure boost required for contactor column operation, but only half the flow required. Two of the pumps are in operation at any given time and the third pump is on standby. In the case of pump failure of any one of the operating pumps, the standby pump comes online to maintain flow to the contactor. This configuration has the advantage of a lower capital cost due to the standby pump being sized for only half the flow. Three half size pumps are typically less expensive than two full sized pumps. The first configuration of two full sized pumps, one operating and one on standby is referred to here as the 2X100% configuration, and the second configuration of three half sized pumps, two operating and one on standby is referred to here as the 3X50% configuration.

Hydraulic Turbocharger design and operating performance

The hydraulic turbocharger is a hydraulic turbine connected via a common shaft to a single stage pump, within a single casing. The hydraulic design of the turbine side and the pump side are custom engineered using CFD programs and design heuristics to best suit the process conditions of the particular application.

Of particular interest is the hydraulic design of the turbine side. Figure 2 is a schematic of the turbine side. Key design features of the turbine side include:

1. A primary turbine nozzle directing incoming flow to the turbine volute
2. An auxiliary nozzle, also directing incoming flow to the turbine volute
3. Replaceable volute inserts that are custom designed for the particular application

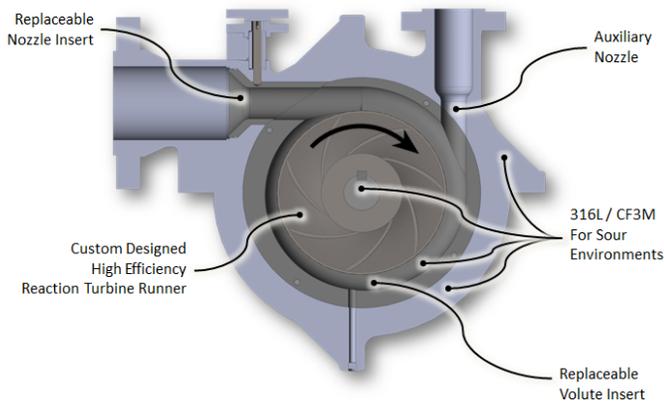


Figure 2: Hydraulic turbocharger, turbine side view

The flow directed to the auxiliary nozzle is controlled by an actuated valve, the auxiliary nozzle valve, so that the minimum flow to the turbine is with the auxiliary nozzle valve fully closed and the maximum flow to the turbine is with the auxiliary nozzle valve fully open. Actuating the valve between open and closed allows for 10-15% flow turndown capability. The most common alternate turbine design that allows for flow turndown is a variable geometry turbine utilizing guide vanes or wicket gates. The addition of multiple actuated mechanical components, the guide vanes, affects the reliability of this alternate design. The reliability of the auxiliary nozzle valve is extremely high and this design enables the functionality of a variable geometry turbine without the associated complexity and potential for failure.

Figure 3 is a schematic of the hydraulic turbocharger, showing the turbine and pump sides, with their respective inlets and outlets. Figure 4 is a schematic showing the internals of the hydraulic turbocharger.

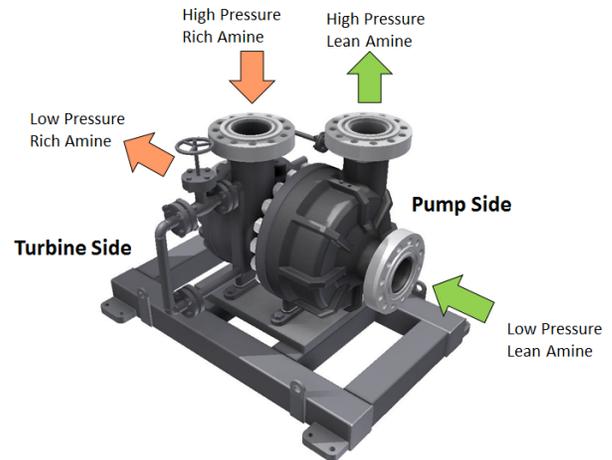


Figure 3: Hydraulic turbocharger, external view

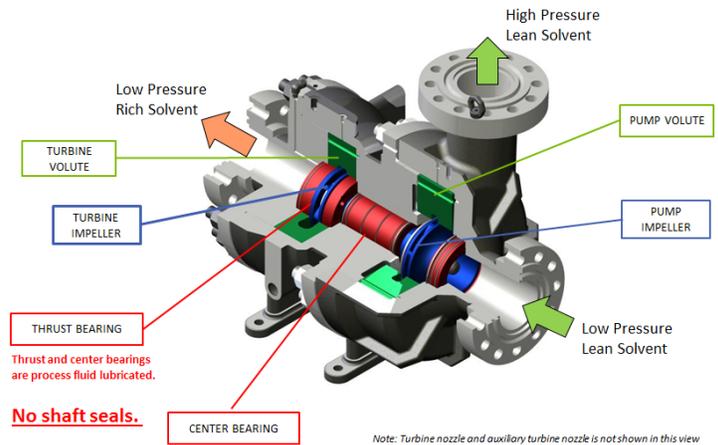


Figure 4: Hydraulic turbocharger, cutaway

The turbine runner and the pump impeller are on a common shaft, intentionally designed to be very stiff with a low L/D ratio, and supported on a center (journal) bearing. Application of the hydraulic turbocharger in AGR processes is discussed in the next section, but for the purposes of this discussion, it should be understood that the pressure of the low pressure lean amine entering the pump side of the hydraulic turbocharger is typically higher than the pressure of the low pressure rich amine leaving the turbine side of the hydraulic turbocharger. Therefore there is a net thrust force from the pump side toward the turbine side, necessitating a thrust bearing on the turbine runner. Both the center bearing as well as the thrust bearing are lubricated by the process fluid, eliminating any requirement for external oil lubrication systems. Startup procedures require flow begins to the pump side. The rotating assembly (RA) will not have torque applied until there is flow and pressure in the turbine at which point bearing lubrication has been achieved. Unlike a gas turbine there is no extended period of time where the turbine is coming up to speed or slowing down. During normal operation, the high pressure lean amine stream exiting the hydraulic turbocharger is of sufficient pressure to lubricate the bearings and is directed towards the thrust and the center

bearings. This fluid, constituting approximately 0.5% of the flow is preferably filtered prior to injection into the bearings to safeguard against the possibility of particulate material entering the bearings. The center journal bearing lubrication scheme ensures that no rich amine fluid can contaminate the lean amine so that contactor effectiveness is maintained.

The rotational speed of the hydraulic turbocharger is unconstrained and is allowed to vary according to the balance of hydraulic conditions on the turbine and the pump sides. This self-regulating mechanism ensures extremely low radial bearing loads and low vibration and minimizes risk of the related failure modes such as shaft deflection, shaft unbalance, mechanical noise, or thrust bearing failure. There is no overspeed protection needed. Overspeed in turbines most typically happens during load rejection, i.e. when an electric generator goes offline and is no longer applying a reaction torque to the turbine runner. In the case of the hydraulic turbocharger, the pump impeller is always connected and providing a load for the turbine so there is no possibility of load rejection. Additionally the pump load always balances the turbine load such that the RA spins at the ideal speed. In the circumstance that the pump load is momentarily significantly decreased (i.e. pump outlet is shutoff) the design and geometry of the system is such that the RA will operate below any critical speed and will not be damaged.

The fluid energy transfer efficiency is a key performance parameter of the hydraulic turbocharger. This parameter is defined as:

$$\text{Efficiency} = (Q_p \cdot (P_{p,out} - P_{p,in})) / (Q_t \cdot (P_{t,in} - P_{t,out}))$$

A well designed hydraulic turbocharger will typically have hydraulic efficiencies in the range of 60% to 80%, with larger hydraulic turbochargers operating at higher hydraulic efficiencies. This efficiency directly characterizes the transfer of energy from fluid stream to fluid stream, as compared to the efficiency of a hydraulic power recovery turbine (HPRT), where the efficiency is characterized as fluid stream to shaft power.

The design of the hydraulic turbocharger has certain inherent advantages from a reliability viewpoint as compared to traditional rotating equipment:

1. Since there is no shaft exiting a casing, there is no requirement for shaft seals
2. There is no requirement for an external oil lubrication system
3. There are no couplings and there is no requirement for alignment
4. The rotating assembly is not speed constrained as it does not share a common shaft with a fixed speed pump
5. Very low vibration

As a result of these factors, the reliability of the hydraulic turbocharger exceeds that of a centrifugal pump. Consequently, replacing or reducing the duty of a centrifugal pump through the application of a hydraulic turbocharger has the effect of improving plant performance.

Alternative flow sheets using hydraulic turbochargers

Hydraulic turbochargers can be used in AGR processes by utilizing the energy in the rich amine leaving the contactor to drive the turbine side of the turbocharger and thus delivering the energy to the pump side to boost the pressure of the lean amine. In this process, the LCV is essentially replaced by the turbine side of the hydraulic turbocharger. A key requirement in this replacement is that the level control functionality of the LCV be maintained.

Prior designs using hydraulic power recovery turbines (HPRT) in AGR processes required the turbines to operate at a fixed speed, and consequently at a fixed liquid flow rate. This requirement is necessitated by the fact that the turbines were typically clutch-coupled to a pump and a motor running at synchronous speed. Contactor level control was then accomplished by having a portion (typically 90%) of the rich amine contactor effluent pass through the turbine at a fixed rate and using the balance 10% of the flow to control the level in the contactor. In cases where contactor operation required flow control beyond this 10% band, the HPRT is simply taken off line, by disconnecting the clutch and allowing the motor to take the full load of pumping the lean amine to contactor pressure.

As described in a previous section, the hydraulic turbocharger has a turndown capability of 10% to 15% through the adjustment of the auxiliary nozzle valve. This turndown capability may then be used to achieve contactor level control. In the event that contactor operation required flow control beyond this 10% to 15% band, alternative measures are required to achieve contactor level control.

Figure 5 shows a simplified process flow diagram of a hydraulic turbocharger in an AGRU, where a 2x100% redundancy configuration is utilized.

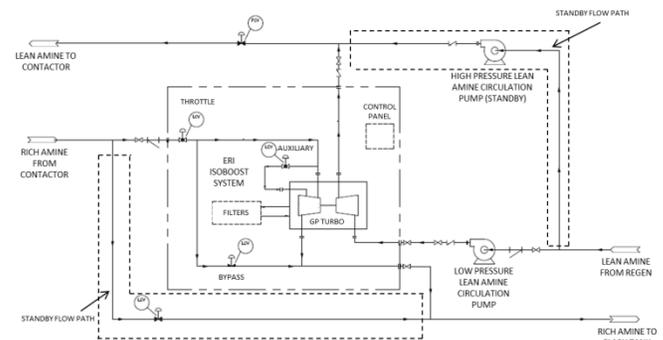


Figure 5: Simplified PFD, 2x100% redundancy

In this configuration, the full effluent from the contactor is directed towards the turbine side of the hydraulic turbocharger and the full flow of the lean amine is pressure boosted by the pump side of the hydraulic turbocharger. Since the fluid to fluid efficiency of the hydraulic turbocharger is about two-thirds, and the flows on the turbine side and the pump side are essentially similar, approximately one third of the required pressure boost is provided by a low pressure lean amine circulation pump which delivers fluid to the pump inlet of the hydraulic turbocharger.

Figure 5 also shows three control valves associated with the hydraulic turbocharger: a throttle valve upstream of the turbine inlet, an auxiliary valve controlling flow to the secondary turbine inlet, and a bypass valve enabling flow to bypass the turbine side of the hydraulic turbocharger. The throttle valve is sized to accommodate the entire flow from the contactor and to provide a partial pressure drop, the auxiliary to accommodate 10 to 15% of flow, and the bypass to accommodate ~20 % of the flow. These three valves modulate to control the contactor level in response to the plant amine contactor level controller output signal.

When the contactor level controller sends an increase level signal within the normal operating range, the auxiliary valve will close. This reduces flow to the turbine and causes the contactor level to increase. When the contactor level controller sends a decrease level signal, the throttle valve is incrementally opened. Next the auxiliary valve is incrementally opened to increase flow. If yet more flow is required, the auxiliary valve is fully opened and the bypass valve will then be incrementally opened to allow more flow to bypass the hydraulic turbocharger. In normal operations, where a 10% to 15% turndown is sufficient to achieve contactor level control, the auxiliary valve achieves control with the throttle valve fully open and the bypass valve fully closed.

Figure 6 shows a simplified process flow diagram of a hydraulic turbocharger in an AGRU, where a 3X50% redundancy configuration is utilized. Figure 6 shows a conventional multi-stage centrifugal pump as the 50% redundancy fulfilment.

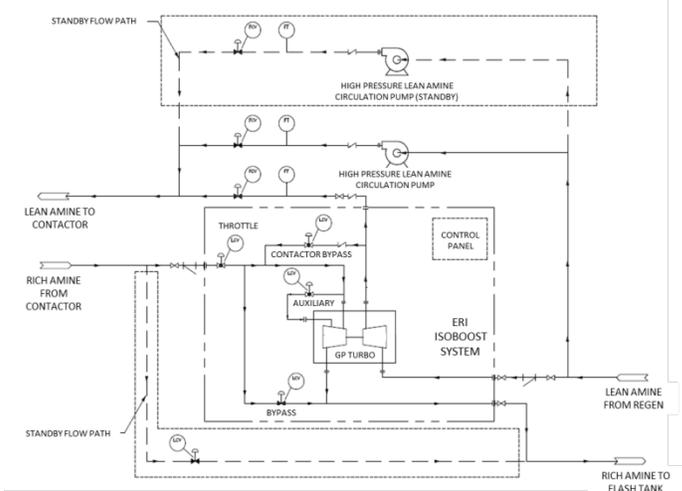


Figure 6: Simplified PFD, 3x50% redundancy

As in the previously described configuration, the full effluent from the contactor is directed towards the turbine side of the hydraulic turbocharger while approximately half of the lean amine flow is pressure boosted by the pump side of the hydraulic turbocharger. The other half of the lean amine flow is provided the full required boost by a HPLAC in parallel with the pump side of the hydraulic turbocharger. This configuration utilizes an asymmetric hydraulic turbocharger where the turbine side is sized for approximately twice the flow as the pump side. Given that the efficiency of the hydraulic turbocharger is approximately two-thirds, there is more than adequate energy in the turbine side flow to provide the full required boost to the half flow on the pump side. Thus, in this configuration, there is no low pressure lean amine circulation pump, and the only requirement is that the pump side of the hydraulic turbocharger be provided with adequate suction pressure to operate reliably without cavitation. The contactor level control is achieved in much the same manner as in the previously described 2X100% configuration.

Centrifugal pump and hydraulic turbocharger reliability

A hydraulic turbocharger can replace a conventional API 610 multi-stage centrifugal pump in an AGRU. This replacement leads to a change in overall system reliability and plant availability. The calculation of the new system reliability will vary by process and equipment arrangement. We can quantify the input to the revised system reliability equation by the difference in reliability between the replaced centrifugal pump and the replacement hydraulic turbocharger. The centrifugal pump reliability, as defined by failure rate or mean time to failure (MTTF), provides the baseline for analyzing impact of the turbocharger on the reliability of the system.

While centrifugal pumps are generally considered reliable devices, they are complex, multi-component pieces of equipment that serve critical industrial applications where they

are exposed to challenging fluids at high pressures. Regular maintenance is required and it is standard to include redundancy on pumps to ensure that interruption of plant operation is minimized.

There are multiple common failure modes for centrifugal pumps. We present descriptions of these failure modes from two primary sources – 1) OREDA operating equipment database and 2) Standard pump reliability prediction procedures from the *Handbook of Reliability Prediction Procedures for Mechanical Equipment*. From the OREDA website, “OREDA® is a project organization sponsored by eight oil and gas companies with worldwide operations. OREDA's main purpose is to collect and exchange reliability data among the participating companies and act as The Forum for co-ordination and management of reliability data collection within the oil and gas industry.” This provides an opportunity to evaluate experienced failures and understand potential root causes of these failures based on pump components that contribute to reliability. Table 1 summarizes the common failure modes as defined by both primary sources.

TABLE 1: COMMON FAILURE MODES OF MULTI STAGE CENTRIFUGAL PUMPS

Handbook of Reliability Prediction Procedures for Mechanical Equipment [2]	Operating Data, Topside Equipment, Centrifugal Pumps [1]
• Reduction in suction head	• Abnormal instrument reading
• Reduction in pump pressure	• Breakdown
• Component corrosion	• Erratic output
• Shaft deflection	• Structural deficiency
• Shaft unbalance	• Vibration
• Air leak thru gasket / stuffing box	• External leakage – utility medium
• External leakage	• External leakage – process medium
• Mechanical noise	• Fail to start on demand
• Positive suction head too low	• Noise
• Positive discharge head too high	• High output

• Suction line / impeller clogged	• Overheating
• Worn / broken impeller	• Parameter deviation
• Thrust bearing failure	• Spurious stop
	• Low output

The OREDA data used had a population of 156, aggregated calendar time in service of 3.2412 million hours and operational time of 2.1290 million hours was used. The database provides failure modes and rates for critical failure modes and degraded failure mode.

While a root cause analysis on this dataset was not performed, we can use the reliability prediction equation for a predicted failure rate. This equation is given as:

$$\lambda_P = \lambda_{SE} + \lambda_{SH} + \lambda_{BE} + \lambda_{CA} + \lambda_{FD}$$

Detailed methodology for determination of each of these component failure rates for specific equipment and operating parameters is provided. Correct determination of component failure rates is critical for an accurate result. However, it is informative to review the base failure rates to understand the relative impact of the components on failure rate. These rates are shown in Table 2.

TABLE 2: BASE FAILURE RATES FOR CENTRIFUGAL PUMP COMPONENTS

Pump Component	Base Failure Rate	Assumptions
Mechanical Seals	$\lambda_{SE,B} = 22.8$	
Shaft	$\lambda_{SH,B} = 0.01$	$\sigma_{T,ult} > 200$ kpsi
Bearings	$\lambda_{BE,B} = 0.00004$	L10=25,000 hours
Casing	$\lambda_{CA,B} = 0.01$	
Fluid Driver	$\lambda_{FD,B} = 0.2$	

We can see clearly that the component most likely to be responsible for failure is mechanical seals. This aligns well with industry experience and knowledge. As described by Marscher in his paper on avoiding failures with centrifugal pumps for the 19th Annual Pump Symposium, “Seals are considered the Achilles heel of most pumps...” [3]. With even the best materials and technology available, mechanical seals are difficult to design and manufacture to meet demanding needs of rotating equipment.

While being significantly less complex than API 610 multistage pumps, hydraulic turbochargers exhibit some of the failure modes common to rotating equipment. Bearing systems,

utilized to provide reaction forces to imparted radial and axial loads, are life limited components. The hydrodynamic journal bearings and hydrostatic axial thrust bearings typically employed in these devices have a theoretically infinite life provided the lubricating fluid is completely free of particulates. Since these bearings are typically lubricated by the process fluid itself, there is risk of bearing damage at those times when the fluid is off-spec or contains entrained particulates. Bearing damage can ultimately lead to equipment failure. Fluid filtration systems can be applied to that portion of process fluid that is being utilized for bearing lubrication which will significantly enhance the reliability of the bearing system while adding minimal complexity. Because the pump impeller typically operates at very high speed and the depressurized process fluid at the turbine exit has a tendency to flash, hydraulic turbochargers can also sustain damage from cavitation.

MTTF estimation of hydraulic turbochargers is arrived at by two separate means, one being the evaluation of hydraulic turbocharger performance in desalination where these devices have been in use for 20+ years and second being the utilization of the OREDA database and removing failures from seals, gaskets, and external couplings. These two approaches are described below:

- a) Turbochargers in desalination: A study involving a large installed base (359 units) of turbochargers in seawater reverse osmosis desalination service over a period of 17 years (1996-2013) concluded these units have a typical MTTF of greater than 10 years even though the application is relatively challenging due to the corrosive nature of the high chloride process fluid, poorly constructed plants, and relatively unskilled operators as compared to typical oil and gas installations. Failures observed were mostly associated with debris in the process fluid and chloride crevice corrosion.
- b) OREDA data [1]: On removing failures caused by seals, gaskets, and external couplings from the OREDA data, we find that MTTF extends from 2.9 years to 8.9 years. The qualification of failure types and related components is based on a root cause analysis of the OREDA pump database described above. This result correlates well with the 10 year MTTF of hydraulic turbochargers in desalination. A hydraulic turbocharger will share similar failure modes as related to impellers, changes in operating conditions, and potentially bearings in a pump. The process fluid lubricated internal bearings in a turbocharger have a lower typical failure rate than conventional bearings in centrifugal pumps as they do not require an external bearing support system, such as oil mist lubrication, but are still a potential failure mode. Since hydraulic turbocharger design does not include an external shaft, shaft seals, gaskets and

couplings, failure modes associated with these components do not apply.

A comparison of MTTF is summarized in Table 3.

TABLE 3: HYDRAULIC TURBOCHARGER AND CENTRIFUGAL PUMP MEAN TIME TO FAILURE COMPARISON

Mean Time to Failure, MTTF		
Centrifugal Pump, Typical	Centrifugal Pump, Theoretical (Without seals, gaskets or couplings)	Hydraulic Turbocharger In Sea Water Desalination
2.881	8.9	10.0

In reviewing this comparison, it is clear that the low part count and elimination of high failure rate components in the hydraulic turbocharger contributes significantly to its reliability advantage.

Hydraulic Turbocharger DFMEA

As part of this study on hydraulic turbocharger reliability, a Design Failure Modes and Effects Analysis (DFMEA) has been performed as described in the emergent API standard 691 Risk Based Machinery Management. The DFMEA identified three components of the design which had a relatively high “Risk Number” which is a metric of the overall potential hazard inherent to the component. Two of these components were bearings which was not an unexpected finding given the critical role they play in rotating equipment and the variability of the process fluid used for lubrication. The other component is the bolt used to hold the two halves of the rotating assembly together. This result was not anticipated by those performing the FMEA and thus proved the value of the exercise while providing an opportunity to upgrade the overall reliability of the system by modifying the design to reduce the likelihood of the failure mode occurring.

Field Operating Data

A hydraulic turbocharger has been operating in amine service since 2008 near Hebbbronville, TX. This is a sour gas treating facility, using a conventional amine gas treating process, with a high pressure contactor operating at about 800psi and a near atmospheric regeneration section. The hydraulic turbocharger is installed in a 2X100% configuration, and is fed with a low pressure lean amine circulation pump. A VFD is used to control the amine flow into the contactor. After initial start-up in 2008, the turbocharger has run continuously with no required maintenance or instances of failure. Table 4 gives parameters of the system, including the original design and new design with the hydraulic turbocharger.

TABLE 4: PERFORMANCE PARAMETERS OF A

HYDRAULIC TURBOCHARGER IN AN AMINE GAS TREATING PLANT IN TEXAS, USA

Parameter	Value
Lean Amine Flow (gpm)	681
Rich Amine Flow (gpm)	681
Pump Inlet Pressure (psi)	371
Pump Discharge Pressure (psi)	800
Turbine Inlet Pressure (psi)	760
Turbine Discharge Pressure (psi)	112
Bypass (gpm)	7
Hydraulic Turbocharger Efficiency	65.00%
Auxiliary Valve Position	Closed

RAM Modeling and Life Cycle Cost comparisons

To determine the impact of the change in component reliability on the system and plant, reliability, availability and maintainability modeling is performed using a Monte Carlo simulation technique. A system block diagram is created to represent the process flow diagram configuration, with each component of equipment being represented in the block diagram. Figure 7 shows an example block diagram used in the simulation.

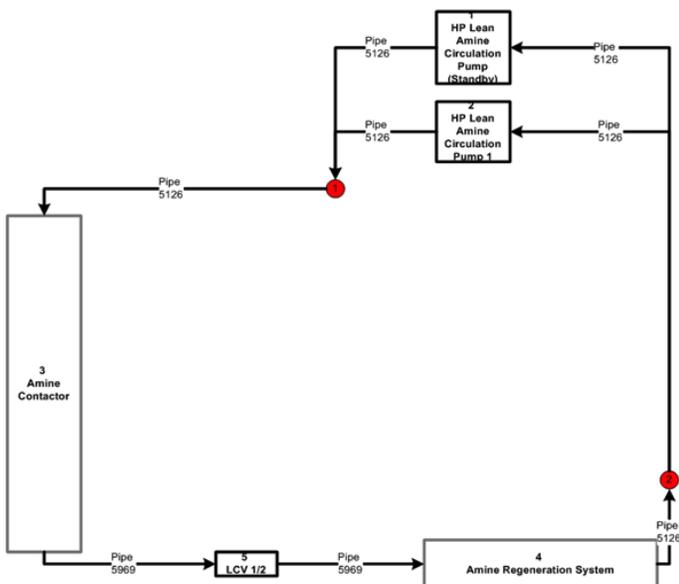


Figure 7: Process Block Diagram for RAM Simulation

Reliability data for different system components are the required inputs for the simulation. Table 5 shows the MTTF for key components in the system.

TABLE 5: COMPONENT MEAN TIME TO FAILURE

Equipment	MTTF,	Failure	Source
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Component	years	distribution	
Valves	32.8 to 42.8	Exponential	OREDA [1]
High pressure centrifugal Pump	2.9	Exponential	OREDA[1]
Low pressure centrifugal Pump	4	Exponential	OREDA[1]
Electric motor (pump driver)	9.1	Exponential	OREDA[1]
Hydraulic Turbocharger	10	Exponential	Field Data*
Variable Frequency Drive (VFD)	10	Exponential	OREDA**[1]

* Operating field data analysis of 359 large hydraulic turbochargers used in desalination plants 1996-2013

**This assumes the VFDs are current, state-of-the-art technology

Various configurations of equipment with and without hydraulic turbochargers are modeled using this technique. These are:

- I. 3x50% Base Case – 3 HPLAC pumps controlled by a common FCV; 2 pumps online and one on standby.
- II. 3x50% Configuration with hydraulic turbochargers– 2 HPLAC pumps in parallel with one hydraulic turbocharger pumps controlled by a common FCV; one pump and one hydraulic turbochargers online, one pump on standby.
- III. 2x100% Base Case with VFD – 2 HPLAC pumps in parallel, controlled by dedicated VFDs; One pump online, one pump on standby.
- IV. 2x100% Base Case with FCV - 2 HPLAC pumps in parallel, flow to the contactor controlled by a flow control valve (FCV); One pump online, one pump on standby.
- V. 2x100% configuration with hydraulic turbochargers with VFDs: One HPLAC in parallel with one hydraulic turbocharger being fed by a low pressure lean amine circulation pump, both pumps being controlled by VFDs; One hydraulic turbochargers online, one pump on standby.
- VI. 2x100% configuration with hydraulic turbochargers with FCVs: One HPLAC in parallel with one hydraulic turbocharger being fed by a low pressure lean amine circulation pump, flow to the contactor controlled by a flow control valve (FCV); One hydraulic turbochargers online, one pump on standby.

The Monte Carlo simulation of these cases is performed to convergence over a 20 year plant lifecycle and lifecycle costs (LCC) are evaluated for a reference plant. The reference plant for this analysis is a large gas processing plant with a normal lean amine circulation flow of 5,969 gpm and rated flow of

6565 gpm. The high pressure lean amine circulation pump is specified as an electric motor driven API 610 BB3 pump with pump suction pressure of 166 psi and the pump discharge pressure of 1081 psi. The radial bearing is specified as a sleeve bearing and the thrust bearing is specified as a tilt pad. A pressurized oil lubrication system is specified.

Results

This model was used to determine the overall reliability of the ERI IsoBoost System based on the proprietary liquid phase turbocharger, the GP Turbo, as well as define expected pump maintenance cost, highlight the components that contribute to non-performance, and quantify what design measures need to be taken (redundancy, upgrades, etc.) to attain system reliability and availability goals.

Results of the modeling effort show that all cases utilizing hydraulic turbochargers have lifecycle costs that are a fraction of the costs when not utilizing hydraulic turbochargers.

For the full redundancy cases, the average system availability over 20 years ranged from 99.61% (i. 3x50% Base Case) up to 99.86% for the 2x100 GP Turbo Case (v. 2x100 with VFD and IsoBoost). Average pump maintenance costs across the same fully redundant cases ranged from a high of \$4.3M (i. 3x50% Base Case) to a low of \$1.4M (v. 2x100 with VFD and IsoBoost Feed Pump). Tables 6 and 7 provide the availability and maintenance cost results for the full redundancy cases respectively.

TABLE 6: AVAILABILITY RESULTS

Case No.	Case Name	Average System Availability over 20 Years	Average Annual Downtime (hours)
i	(FR) BaseCase (3x50)	99.61%	33.8
ii	(FR) 3x50 without IsoBoost Feed Pump	99.84%	14.0
iii	(FR) BaseCase (2x100) with VFD	99.83%	15.2
iv	(FR) BaseCase (2x100) with FCV	99.83%	15.1
v	(FR) 2x100 with VFD and IsoBoost Feed Pump	99.86%	12.3
vi	(FR) 2x100 with FCV and IsoBoost Feed Pump	99.84%	14.0

TABLE 7: MAINTENANCE COST RESULTS

Case No.	Case Name	Average Pump Maintenance Costs over 20 Years
i	(FR) BaseCase (3x50)	\$4,333,842
ii	(FR) 3x50 without IsoBoost Feed Pump	\$2,211,695
iii	(FR) BaseCase (2x100) with VFD	\$3,865,888
iv	(FR) BaseCase (2x100) with FCV	\$3,782,401
v	(FR) 2x100 with VFD and IsoBoost Feed Pump	\$1,350,699
vi	(FR) 2x100 with FCV and IsoBoost Feed Pump	\$1,544,037

CONCLUSION

Based on the results of this study, it may be concluded that:

- I. Installation and maintenance of redundant pumping is critical for minimizing plant downtime and minimizing lost production
- II. Installation and maintenance of redundant pumping does not have a significant impact on maintenance costs
- III. Flow sheet configurations that include the IsoBoost system consistently provide lower plant downtime and maintenance costs as compared with conventional flow sheet configurations not including the IsoBoost system.
- IV. In the case of conventional flow sheet configurations (not involving the IsoBoost system), the 2x100% configuration provides a lower plant downtime and maintenance cost than the 3x50% configuration.
- V. In the case of flow sheet configurations that include the IsoBoost system, it is the 2x100% configuration with VFD that provides the lowest downtime of 12.3 hours, as compared with 15.2 hours for the 2x100% configuration without the IsoBoost system.
- VI. The largest difference between cases with and without the IsoBoost system is seen in the 3x50% configuration. The 3x50% configuration without the IsoBoost shows the highest downtime, 33.8 hours. The 3x50% configuration with the IsoBoost provides only 14.0 hours of downtime. Incremental lost production of ~19.8 hours/year could have a financial impact in the \$1.6M range.

Hydraulic turbochargers are a beneficial and reliable alternative for reduction of conventional pumping requirements in amine gas treating operations. Improved plant operations and availability as a result of improved component reliability and reduced maintenance are achieved.

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