Project Management Institute. Kingdom of Saudi Arabia

Digital Insights:

HPRT Paper

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ABSTRACT

Pumps are critical components in industrial operations, ensuring efficient fluid transportation across refineries, manufacturing plants, and power facilities. However, their high energy consumption presents a significant challenge, driving the need for innovative solutions to improve efficiency and sustainability. One solution is the integration of Hydraulic Power Recovery Turbines (HPRTs), which harness excess hydraulic energy and convert it into useable power, reducing overall energy consumption and operational costs.

This paper explores the implementation of HPRTs in one of Saudi Aramco Mega Project, detailing the design, manufacturing and testing phases. It discusses the challenges encountered during development, including technical complexities, material selection and system integration, as well as the strategies used to overcome them. Additionally, the expected energy savings and economic benefits derived from HPRT implementation are analyzed, highlighting its long-term impact on sustainability and cost reduction. Furthermore, the different types of HPRTs available in the market and the rationale behind our selection are presented

Furthermore, this paper examines the different types of available energy equipment technologies and explains the rationale behind selecting the most suitable technology for our specific application. By comparing HPRTs with alternative energy recovery technologies, basis on why this solution was chosen and how it enhances overall system performance are demonstrated. The case study aims to provide valuable insights into the benefits and practical considerations of incorporating energy recovery systems in industrial operations, contributing to a more energyefficient and environmentally responsible future.

1.0 INTRODUCTION:

In industrial applications, energy consumption is a critical factor influencing operational efficiency, cost-effectiveness and sustainability. Pumps, as fundamental components of fluid transportation systems, account for a significant portion of industrial energy use. During the engineering phase of our project, one of the primary challenges was identifying a technology or solution that could effectively reduce energy consumption while maintaining the same level of productivity. With the advancements in energy-saving technologies available today, numerous options were considered. However, our objective was to select the most reliable, efficient, high quality, and cost-effective solution.
Rotating equipment plays a crucial role in optimizing energy usage, and pumps, in particular, offer significant potential for energy savings. Among the various solutions evaluated, Hydraulic Power Recovery Turbines (HPRTs) emerged as the most suitable choice due to their ability to recover and reuse excess hydraulic energy within the system. HPRTs convert otherwise wasted energy into usable power, reducing overall energy demand and improving system efficiency. This technology aligns with our project's goals of enhancing sustainability while ensuring operational reliability.

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2.0 TURBOCHARGER INSTEAD OF HPRT:

The primary objective of implementing a Hydraulic Power Recovery Turbine (HPRT) is to utilize the lost hydraulic energy to assist in driving the pumps, thereby reducing overall energy consumption. However, replacing the HPRT with a turbocharger presents certain limitations that impact system efficiency and energy savings.

In the current configuration, the system consists of three pumps (A, B, and C). Normally, two in operation while one is on standby mode.

- Pumps A and B are driven by motors.
- Pump C is driven by a motor in combination with an HPRT.

If a turbocharger was applied instead of an HPRT, it would fundamentally change the system dynamics and hydraulics. In this scenario, the turbocharger would act as a pump driver, meaning that instead of reducing energy consumption, it would introduce an additional power requirement to operate effectively. As a result, no energy savings would be achieved, and in fact, the system could experience increased power demand, contradicting the original goal of energy efficiency.

This analysis highlights why the HPRT remains the optimal choice, as it enables energy recovery without the need for additional power input, ensuring a more efficient and sustainable solution for the project.

2.1 HYDRAULIC ENERGY RECOVERY TURBINE TYPES & APPLICATIONS:

Hydraulic turbines are widely used in various applications, including hydroelectric power generation, industrial processes, pumping stations, and renewable energy projects. Their primary function is to convert kinetic and potential energy from fluid flow into mechanical energy in the form of rotational motion, which can then be used to drive other equipment or generate electricity.

There are several types of hydraulic turbines, categorized based on their design and flow characteristics. The main types include:

- **Pelton Wheel Turbine** – Best suited for high-head, low-flow applications where water is directed through nozzles to strike the turbine's buckets.
- **Francis Turbine** – A reaction turbine designed for medium-head applications, offering high efficiency and adaptability to varying flow conditions.
- **Kaplan Turbine** – A propeller-type turbine ideal for low-head, high-flow applications, commonly used in large-scale hydroelectric plants.
- **Cross-Flow Turbine** – A versatile turbine suitable for small-scale applications, where water passes through the blades twice for improved efficiency.

With our design basis established, selecting the most suitable Hydraulic Energy Recovery Turbine (HPRT) requires careful consideration of system requirements, including flow rate, pressure drop, efficiency, and operational stability. The classification of hydraulic turbines is essential in ensuring optimal performance and maximum energy recovery for our specific application.

3.1 LIQUID CHARACTERISTICS:

The fluid entering the Hydraulic Energy Recovery Turbine (HPRT) is rich amine from the absorber, carrying specific physical and chemical properties that influence the turbine's design and material selection. The key characteristics of the fluid are as follows:

- Temperature: 144°F.
- Relative Density (at normal flow): 1.05.
- Viscosity: 3.41 cP.
- Vapor Pressure: 973.4 PSIA.
- Corrosive Components: H_2S, CO_2, acid degradation products.
- Positive Hydrogen (pH level): 8.1.
- Specific Heat Capacity: 0.783 Btu/(lb•°F).

Due to the presence of H_2S, CO_2, and acid degradation products, corrosion resistance is a critical factor in material selection for the HPRT components to ensure durability and long-term operational efficiency. The thermal and physical properties of the rich amine also play a crucial role in determining the energy recovery potential of the turbine.

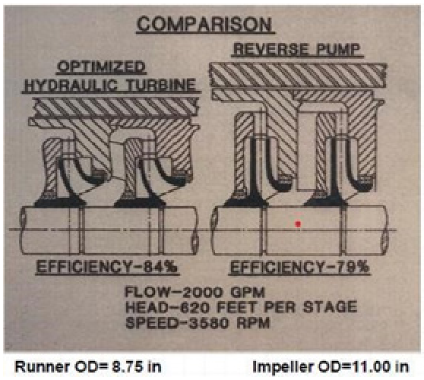
3.2 MATERIAL SELECTION:

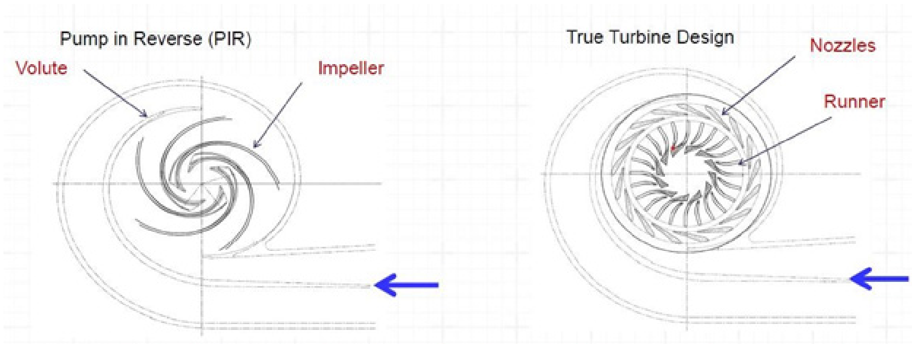
The material selection was carried out in accordance with Code HIS and American Society of Testing Materials (ASTM) standards. The selected materials include:

Material	Standard
Casing	ASTM A351 CF3M
Impellers	ASTM A743 CF3M
Case Wear Rings	ASTM A276 XM19
Shaft	ASTM A276 XM19
Shaft Sleeves	ASTM 316L SS
Throat Bushings	ASTM A276-S21800
Gland / Mechanical Seal Flange	ASTM 316L SS
Bearing Brackets	ASTM A216 WCB
Non-Wetted Fasteners	ASTM A193 Grade B-7 / ASTM A194 Grade 2H
Piping / Fittings	ASTM A312 TP316 / ASTM A182 F316L
Base Plate	ASTM S27JR
Mechanical Seal Springs	ASTM Alloy316 / 718L
Mechanical Seal Other Metal Parts	ASTM 316 SS
Throttling Sleeves	ASTM A276 XM19

3.3 POWER RECOVERY:

Power recovery involves harnessing the pressure energy from the reject stream of a process to generate useful work. In industrial systems, this is typically achieved by extracting energy from high-pressure fluid streams that would otherwise be wasted. Initially, turbines were introduced to utilize the energy of the reject stream to assist in driving the shaft of high-pressure pumps. This direct mechanical coupling helps avoid efficiency losses associated with multiple energy transformations.
The current design incorporates the Pump-in-reverse configurations behaving like a true turbine. They are specifically designed and developed for a given application to optimize energy recovery and overall system performance. This pump configuration features impeller and diffuser arrangement, enabling efficient energy transfer. To further improve predictive accuracy, multiphase CFD simulations are continually being developed to refine empirical performance models, particularly for applications involving gas evolution





Pump's impeller and volute or diffusor are called runner and nozzle when used as a turbine.

3.4 GAS EVOLUTION EFFECTS ON HPRT PERFORMANCE

Fluid streams with dissolved gases are common in HPRT applications. Furthermore, high amounts of gas at the turbine exhaust affect performance. HPRTs typically recover more power with evolving gas but can recover less power if not sized correctly.

3.5 GAS EVOLUTION

Because the turbine encounters gases that may come out of solution when pressure is reduced through the runner, special design considerations are required:

- A double suction impeller should be used at the first stage to reduce the velocity of the fluid exiting the runner.
- Non-metallic rings may be required to reduce galling of the rings and bushings.
- The impact on the rotor dynamics of the pump must be considered.
- The impact on thrust must also be evaluated.
- A true turbine can be designed to take advantage of the evolving gas and recover more power in the field.

4.0 MANUFACTURING

The HPRT system is designed as a horizontal, between-bearings API BB5 barrel pump with an opposed impeller design and volute inner casing, in accordance with the latest API 610 standards. Key design features include:

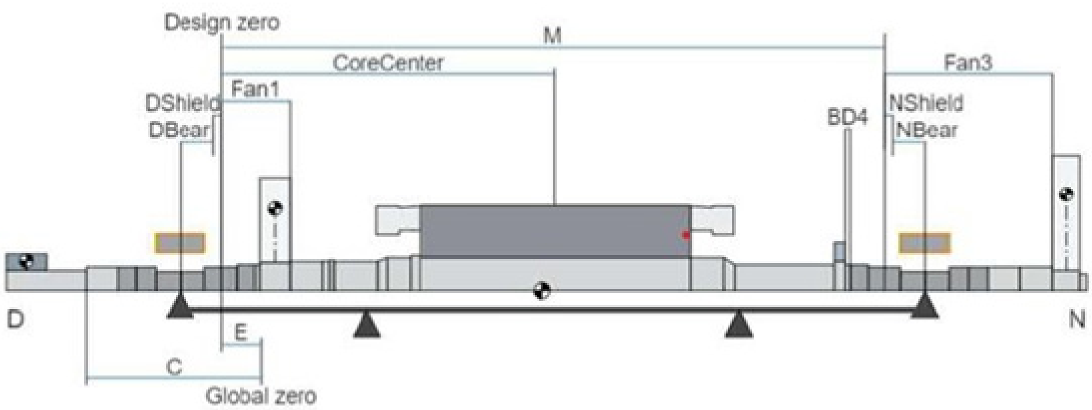
- Opposed impeller design.
- Ball/Ball, Sleeve/Ball, or Sleeve KTB bearing configuration.
- Rotor balancing.
- Sag bore option for seven stages or more.
- Double suction first stage.

5.0 CHALLENGES:

During manufacturing and testing phase, several issues have been encountered on such a complex train. Most notably, the double shaft ended motor connecting a pump from one side and the HPRT from another, faced many challenges. This section highlights some of them alongside resolutions.

5.1.1 CRITICAL SPEED MARGIN API REQUIREMENTS

Critical Speed Margin Less Than API Requirements During the Factory Acceptance Test for the main motor, it was observed that four single-shaft motors exhibited critical speeds that fell below the API-required margin. The API standard calls for a %15 separation margin, but all four motors demonstrated a critical speed margin between %8 and %8.5. One of the motors exceeded the shaft vibration limit of 38.1 µm, as specified by API 5 541th edition.
Initial critical speed calculations, using standard values, estimated the critical speed around 2900 rpm (~%20 separation margin).



Critical speed [r/min]	Amplification factor	Separation margin [%]	Criteria [%]	Ok
2365	4.2	34	10	Yes
2899	21	19	10	Yes

However, in field tests, rotors proved stiffer than anticipated, pushing the actual critical speed to ~3260 rpm, which is uncomfortably close to the motors' normal operating speed. To resolve this, it was confirmed that the rotor core stiffness was greater than calculated. Modifications to the shaft geometry were required to reduce the critical speed and increase the separation margin, thereby ensuring compliance with API standards and safe motor operation.

5.1.2 TOTAL INDICATED RUNOUT ROTOR:

Run out the rotors of motors and were balanced after shaft modification. However, the Total Indicated Runout (TIR) values exceeded the API runout limit of 9.53 µm:
DE = 8.3 µm, NDE = 19.0 µm
DE = 17.9 µm, NDE = 11.1 µm

TIR was measured using proximity probes and a dial gauge.
Shaft diameter was machined next to the rotor core to compensate for increased stiffness.

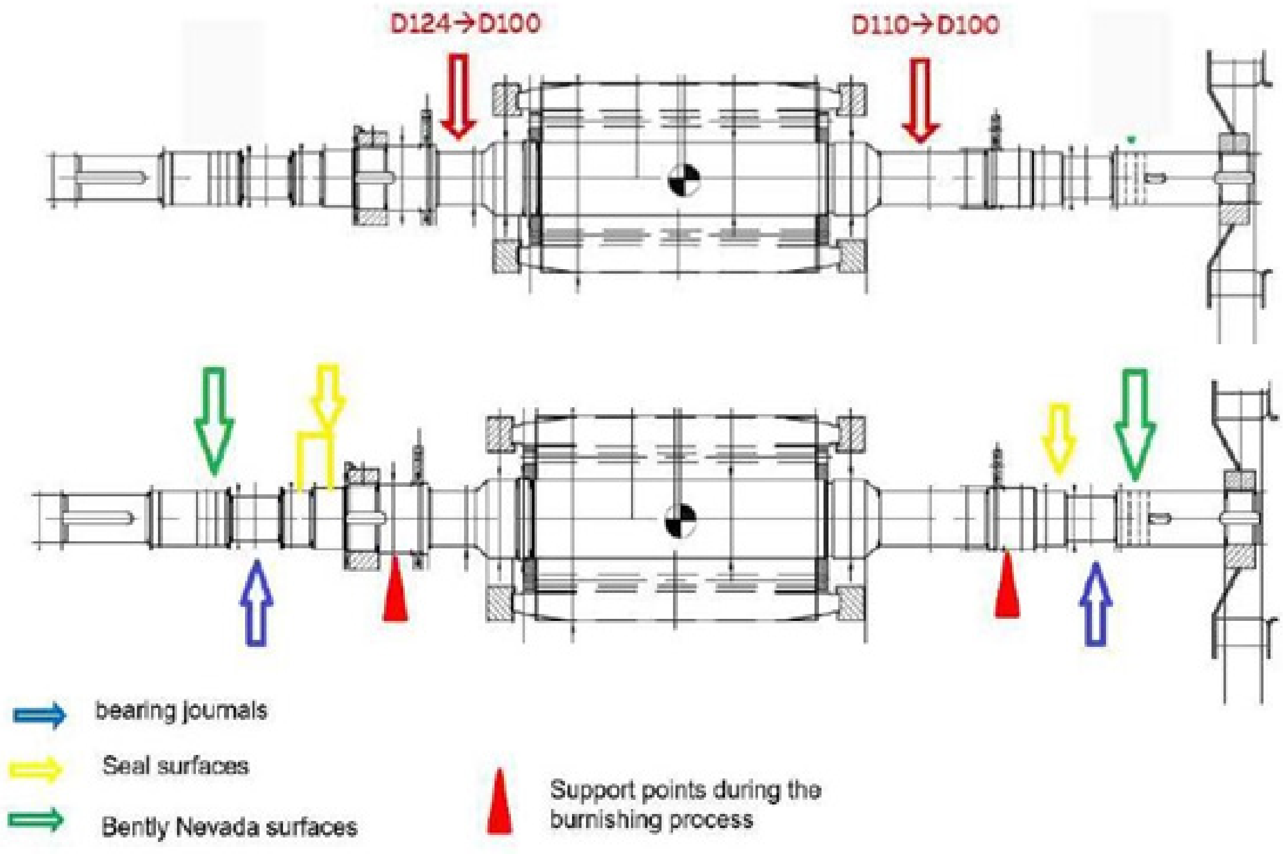


Figure 2. Burnished areas of the shaft

followed by burnishing the measurement surfaces—a standard procedure ensuring concentricity. After machining and burnishing, shafts were rebalanced and retested. Post-burnishing measurements confirmed acceptable performance, and the motors were successfully reassembled.

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5.1.3 4.1.4 SMALL BEARING JOURNAL DIAMETER:

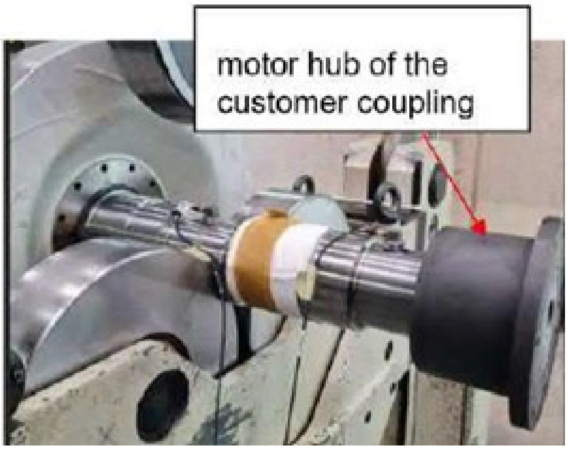
Unfortunately, working clearances of these surfaces were limited after shaft modification and burnishing. The DE bearing journal diameter became too small after re-burnishing. This issue arose because the shaft had already undergone final machining before the modifications, leaving minimal allowance. The insufficient allowance made it impossible to correct the mechanical runout at the DE

Measuring surface
Bearing journal after final machining
Bearing journal after first burnishing
Bearing journal after second re-burnishing

As a contingency, it was decided to purchase new shafts for all motors in the project in case future rotor modifications failed, minimizing the risk of additional project delays.

5.1.4 4.1.5 ROTOR BALANCING AND COUPLING ASSEMBLY ISSUES:

A dummy balancing hub—matching the weight of the motor-side hub of the customer coupling— was installed on the rotor D-end for the purpose of rotor balancing. After balancing, this dummy hub was removed and replaced with the actual customer coupling to verify the combined balancing state.



However, the residual unbalance with the coupling installed exceeded the maximum allowable residual unbalance.

Maximum allowable residual unbalance	DE (MAX)	NDE (MAX)	
	572	598	gmm
	DE	NDE	
Initial unbalance	8030	9060	gmm
	329	220	(deg) O
	DE	NDE	
Residual unbalance	68	163	gmm
	9	137	(deg) O
	DE	NDE	
Residual unbalance	2900	4280	gmm
with coupling	244	35	(deg) O

An investigation into the balancing machine and tools was conducted. It was discovered that the balancing state of the rotor changed after detaching and reattaching the rotor tools, revealing that the repeatability of the balancing machine was not up to standard. A spare cardan was then installed to recalibrate and rebalance the rotor, significantly improving the repeatability and overall results.

Maximum allowable residual unbalance	DE (MAX)	NDE (MAX)
	572	598
	DE	NDE
Initial unbalance before detaching the cardan	129	170
	1.2	1.6
	DE	NDE
Residual unbalance after reattaching the cardan	369	56
	56	187

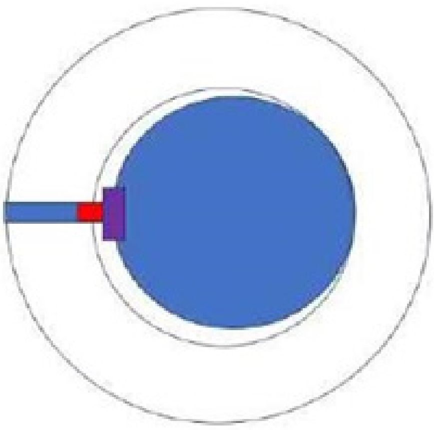
gmm

gmm
(deg) O

gmm
(deg) O

Several factors were identified that might affect residual unbalance after reattaching the cardan:

- Possible shift in cardan shaft centerline due to fit tolerances between shaft and coupling.
- Thermal bow in the rotor core caused by frictional heating during rotation and uneven cooling upon stopping.
- Creep in the rotor's laminated core when warm and stationary, which can distort the balancing state.



Additionally, the dummy hub was designed with a clearance fit; tightening the key bolt caused an offset that introduced further unbalance.

The selected balancing grades were:

- G1.0 for reduced vibration design.
- G0.7 for low vibration design

(per ISO 11:2016-21940).

Balanced for 3582 rpm in accordance with	
Balanced to ISO 21940 G2.5	√
Balance to API 610 8th edition (ISO 21940 G1.0)	
Assembly Balance only to API 671 = (ISO 21940 G0.66)	
Component balance and Check balance to commercial limits (ISO 21940 G2.5)	
Balance to API 671 Method 1 (Components only)	
Balance to API 671 Method 2 (Component + Assembly Check)	
Balance to API 671 Method 3 (Component + Assembly Balance)	
Component Balance only to AGMA Class 10 (G6.3)	
Other balance in accordance with customer specification OR (ISO 21940 G6,3)	

Calculated Unbalance (IEC)	
Coupling mass	9 kg
Operating speed	3600 rpm
Balancing grade	0,7 G
Allowed unbalance	16,7 gmm

Calculated Unbalance (IEC)	
Coupling mass	9 kg
Operating speed	3600 rpm
Balancing grade	2,5 G
Allowed unbalance	59,7 gmm

The customer coupling, balanced to ISO 21940 G2.5, consists of three parts: motor side hub, pump side hub, and transmission unit. Despite being of a different balancing grade, the effect on overall rotor unbalance was minimal—approximately %10 of the rotor's allowable unbalance.

A new balancing procedure was followed using the customer dummy hub (s/n 93038), and final verification was performed using the actual customer coupling hub (s/n 93052).

Maximum allowable residual unbalance	DE (MAX)	NDE (MAX)
	572	598
	DE	NDE
Initial unbalance	13200	9190
	321	230
	DE	NDE
Residual unbalance with customer coupling dummy hub	35	92
	209	91
	DE	NDE
Residual unbalance run 1 with customer coupling hub	519	363
	290	102
	DE	NDE
Residual unbalance run 2 with customer coupling hub	469	353
	305	106
	DE	NDE
Residual unbalance run 3 with customer coupling hub	513	412
	307	102
	DE	NDE
Result of the unbalance with customer coupling dummy hub	PASS	PASS
	DE	NDE
Result of the balancing check with customer coupling	PASS	PASS

gmm

gmm
(deg) 0

gmm
(deg) 0

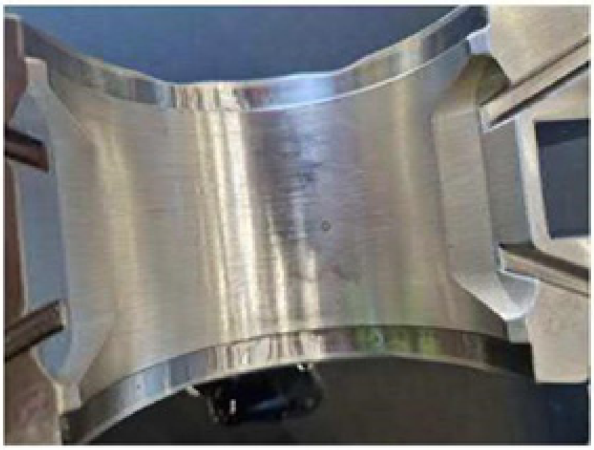
gmm
(deg) 0

gmm
(deg) 0

gmm
(deg) 0

5.1.5 BEARING SHELL DENT:

During inspection, a small dent was observed on the bottom half of the NDE (non-drive end) bearing shell. This dent was likely caused by a loose particle trapped between the bearing journal and the shell during motor assembly. Given that the bearing shell is made from softer white-metal compared to the shaft, it sustained damage while the shaft remained unaffected. It was concluded that this particle had not been properly removed during cleaning prior to assembly.



5.1.6 DENTS ON D-END BEARING SHELL SURFACE:

Dents were observed on the surface of the lower bearing shell at the drive end (Dend). While it is a typical feature for sleeve bearings to exhibit slight scratches or marks during regular motor use, in this instance, the D-end shell showed abnormal dents on the bearing's sliding surface. These irregularities exceeded the expected wear patterns and required further inspection to ensure continued reliability and operational safety.

5.1.7 TOTAL INDICATED RUNOUT (POSITION 200)

Following the shaft modification and burnishing process performed, measurement results were obtained and are summarized. These results indicated that the available working allowances on the shaft bearing journal were limited after the shaft modification.

200			
Measuring surface	NDE [mm]	DE [mm]	Tolerance min [mm]
Bearing journal after burnishing at sub-supplier after shaft modification	79,88	79,87	79,769

After the shaft modification, the rotor of motor at position 200 was balanced. However, the Total Indicated Runout (TIR) values measured during balancing exceeded the API allowable limit of 9.53 µm. These measurements were taken from as part of the balancing procedure. TIR reflects a combination of mechanical and electrical runout. The mechanical component of runout was also measured independently using a dial indicator on the balancing machine.

Motor pos.	Acceptance criteria API	D-end	N-end
200	9.53 um (peak-to- peak unfiltered 38.1 (1,5 mil)*0.25)	12.9 um	13.9 um

Motor pos.	D-end	N-end
200	3 um	4 um

It is important to note that runout values can vary when measured on the bearings. Despite this, the runout readings exceeded API acceptance limits during field testing. According to API 5 ,541th Edition, the allowable combined electrical and mechanical runout in an assembled machine must not exceed %30 of the peak-to-peak unfiltered vibration limit (38.1 µm). This results in a maximum permitted value of 11.43 µm (38.1 µm × 0.30).

Motor pos.	Acceptance criteria API	D-end	N-end
200	11.43 um	13.60 um	23.92 um

Furthermore, excessive vibration levels were detected at the NDE Y-axis location of the motor, primarily due to high runout values. This contributed to an overall vibration amplitude that surpassed the 38.1 µm threshold.

Vibration	Acceptance criteria API	Shaft vibration (um p-p) DE x	Shaft vibration (um p-p) DE Y	Shaft vibration (um p-p) NDE X	Shaft vibration (um p-p) NDE Y
vibrations, cold conditions	38.1um	33.08 µm	22.14 um	26.82 µm	41.93 um

Mechanical runout represents the deviation of the shaft's surface from a perfect circle during rotation. Irregularities such as an elliptical shape or eccentricity increase mechanical runout. Electrical runout, by contrast, arises from inconsistencies within the shaft material itself, such as residual magnetism, microscopic metallurgical variations, or localized stress concentrations. These lead to fluctuating electrical properties that manifest as elevated electrical runout.

Following a comprehensive case review, the likely root causes of the elevated runout were attributed to several factors within the burnishing process. Mechanical runout measurements suggest that the issue was primarily mechanical in nature and introduced during the burnishing phase. Post-testing, the quality department inspected the bearing shells and confirmed that no anomalies were present. The rotor required modification due to critical speed concerns. Burnishing of seal, and bearing journal surfaces is standard procedure following rotor machining. A risk assessment was conducted prior to initiating the shaft modification. One known risk was potential machining. subsequently reviewed and updated process instructions in collaboration to mitigate such risks.

6.0 ENERGY EFFICIENCY, ENVIRONMENTAL IMPACT, AND COST SAVINGS:

The implementation of the HPRT system has led to significant energy and environmental benefits, as well as notable cost savings. The project achieved an energy saving of approximately 1.2 MW, directly contributing to reduced operational demands. From an environmental perspective, this energy efficiency translates to a reduction of nearly 7,800 tons of CO_2 emissions annually, aligning with sustainability and climate action goals. Additionally, the energy savings have resulted in substantial financial benefits, with estimated annual cost savings of around 400,000 USD of yearly cost. These outcomes highlight the dual advantage of deploying advanced recovery technologies—enhancing performance while promoting economic and environmental sustainability.

7.0 CONCLUSION:

The implementation of **Hydraulic Power Recovery Turbines** (**HPRTs**) presents a significant opportunity to enhance energy efficiency in industrial applications. By recovering excess hydraulic energy, HPRTs contribute to lower operational costs, reduced power consumption, and improved system performance. This paper has explored the design, liquid characteristics, turbine types, and energy-saving benefits of HPRTs, while also comparing them with alternative solutions such as turbochargers. Our findings confirm that HPRTs are the most effective choice for optimizing energy recovery in our project. Future developments in turbine technology and material selection will continue to improve the efficiency and reliability of these systems, further supporting global efforts toward industrial sustainability. Furthermore, this paper highlights the design and factory testing challenges associated with the HPRT train—particularly the double-ended shaft electric motors—and the engineering solutions applied to overcome them.

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