THE CRITICAL ROLE OF RE-RATING IN THE ELECTRIFICATION OF CENTRIFUGAL COMPRESSORS: ENSURING OPTIMAL PERFORMANCE AND RELIABILITY

Paper EUR25_18

Tomás Sanchez
Sr. Sales Engineer
Ebara Elliott Energy
901 N. Fourth Street
Jeannette, PA 15644
USA
tsanchez@elliott-turbo.com

Brian Stewart
Sr. Application Engineer
Ebara Elliott Energy
901 N. Fourth Street
Jeannette, PA 15644
USA
bstewart@elliott-turbo.com

Jeffrey Jacob
Application Engineer
Ebara Elliott Energy
901 N. Fourth Street
Jeannette, PA 15644
USA
jjacob@elliott-turbo.com

Abstract - Centrifugal compressors are considered as very energy-consuming machines used in critical operations in oil and gas, petrochemical, refining, LNG as well as ethylene industries. Transforming these machines ' drivers from mechanical to electrical type is considered as one of the effective measures toward the reduction of Scope 1 emissions. To guarantee that centrifugal compressors are suitable for both present and future operating conditions, this paper evaluates one of the key factors, which is the re-rating of centrifugal compressors. Compressor re-rating provides a new rated hardware best suited for both present and future operating conditions. It improves reliability and ensures that compressors operate at their optimal efficiency. This paper presents via case studies and executed projects how re-rating can help in substantial emission reductions and favourable return on investment.

Index Terms — Centrifugal Compressor, CO2, Efficiency, Motorization, VFD (Variable frequency drive), Re-rate

I. INTRODUCTION

Centrifugal compressors serve as vital industrial components that operate in the oil and gas along with petrochemical refineries and LNG plants and ethylene production facilities. These devices serve critical purposes because they compress gases effectively with high efficiency, which is then used for chemical reactions and/or heat transfer throughout the plant. The development of centrifugal compressors started in the early 1900s through inventions by Auguste Rateau and Aurel Stodola who created their early concepts. Progress in technology has refined centrifugal machines through improved design of aerodynamic elements together with material advancements and control system improvements which resulted in enhanced performance features. This progress has allowed wider adoption of these critical machines. Figure 1 shows a CAD 3D rendering of a horizontally split centrifugal compressor.

The industrial world now focuses on the transition from mechanical to electrical drivers because it creates substantial progress toward sustainable industrial operations. The electric conversion of centrifugal compressors drivers serves as a vital method to cut Scope 1 emission levels which result from direct source operations under owner control. Electric motors adopted as a replacement for gas or steam turbines enables industries to decrease their environmental impact substantially. Industrial electrification processes complement worldwide sustainability targets and fulfills increasing environmental regulations. Modern industries find electrification with its efficiency improvements and reliability features highly valuable for their operations.

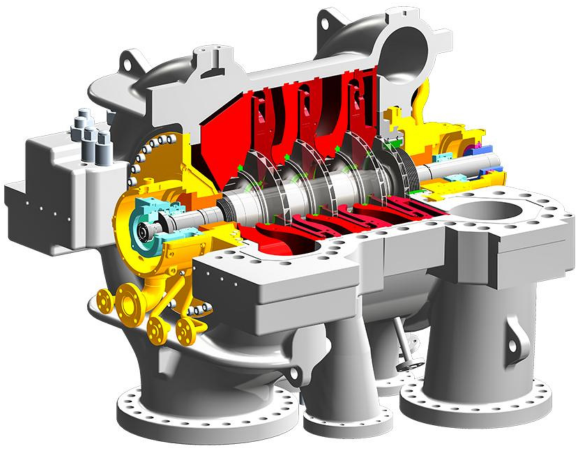

Figure 1 Horizontally split centrifugal compressor 3D cut

The present paper investigates the fundamental role of re-rating which ensures centrifugal compressors maintain operational capability for present requirements while being prepared for predicted future demands after implementation of electrification at higher efficiency rates. If predicted future demands remain the same as the present requirements, then the compressor will simply be more efficient and consume less power – furthering scope 1 emissions reduction. Increasing reliability and efficiency in compressors is an essential step for an electrified system. We aim to show how re-rating extends the life of the equipment by looking at mechanical components such as shafts, bearings and couplings but also enhances its ability to function across a wider range of operation conditions.

II. THE CONCEPT OF RE-RATING

Re-rating a centrifugal compressor requires changing the aero-path. At its core, re-rating involves recalibrating a compressor's design and operational parameters to optimize its performance under new or current working conditions. This means that diaphragms (diffusers) and rotor (impellers and shaft) will be newly supplied with modern aero design. Advances in flow path design, stage performance, aerodynamics, manufacturing technology, and materials science make it possible to achieve higher efficiency within existing casings. These component upgrades to impellers and diffusers lower energy waste in the flow path which produces significant energy savings and decreases operational costs. In addition to efficiency gains, plant process optimization and versatility can be achieved which is often overlooked in the context of transitioning from mechanical to electrical drivers. The extended operational parameters re-rating brings to the compressor supports the plant to maintain peak performance during dynamic and changing conditions.

Re-rating strengthens compressors because it improves their mechanical capacity to operate new or current operational requirements. The assessment of shafts, bearings and couplings is completed rigorously while implementing required component upgrades to enhance durability as well as robustness. Re-rating enables system reliability to increase due to optimized component selection and the resolution of potential weaknesses which reduces unexpected failures. Re-rated compressors deliver businesses strategic operational advantages that let them meet sustainability targets through enhanced performance and increased reliability. Figure 2 demonstrates the Efficiency increase opportunity for a higher flow re-rate example. Similar opportunities can be found to existing or lower flow demands. Correct analysis followed by planned implementation generates transformative results which make the initial investment for re-rating existing compressor systems more profitable in the long term.

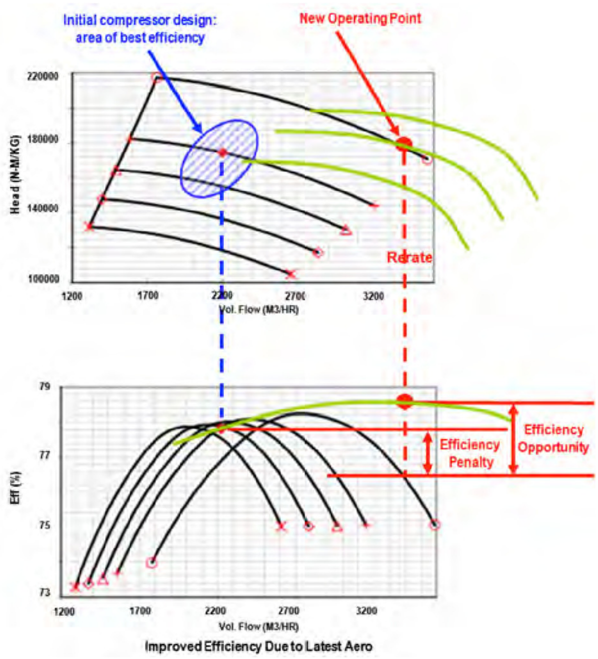

Figure 2 Re-rated Curves versus existing compressor curves

III. Mechanical Review of Centrifugal Compressor Components

Typically, in motorization projects, the existing driver is a steam turbine or gas turbine. These types of drivers adjust their operating speed and output power gradually over time. When considering an electric motor driver, however, especially a synchronous motor or one with a variable frequency drive (VFD), it needs to be considered that they will develop instantaneous pulsation torques as well as the potential for short circuit torques. These instantaneous torques can be quite large, and can introduce cyclic fatigue into the driven shaft ends. Consequently, even when considering a new motor that is rated for the same output power as an existing turbine, the driven shaft end sizes often need to increase. This can lead to additional scope on the compressor, gearbox, and couplings, over and above the new main driver. The analysis also needs to consider the revised baseplate dynamic loading evaluations, especially if the equipment is on one common baseplate. Civil engineering will need to be engaged for this evaluation.

Normally, the compressor shaft ends are tapered and hydraulically fit. Thus, for this design, we must consider not only the torque carrying capacity of the shaft end itself, but also the coupling to shaft interference fit. For the shaft torque carrying capacity, it may be possible to utilize a higher strength material for the shaft in order to accommodate the higher loading. The coupling to shaft interference fit is governed by API 671 requirements, and can many times become the limiting factor in the shaft end sizing criteria for the new driver. The interference fit is not affected by the use of higher strength materials, so in that scenario, the only option may be to increase the shaft end size and interference fit between the shaft end and coupling hub. Table 1 shows the difference in shaft end sizes between steam turbines driven from different types of Motor and drive technology.

	Shaft end size (in), Hydraulic fit		
Driver Rating 42MW	Compressor2		Compressor 1
String Configuration	Drive	Drive Thru	Drive
Steam Turbine+ Compressor2+Compressor1	6.5	5.5	5.5
Synch. Motor w/VFD (LCI) +Gear+Compressor2+Compressor1	9	8.5	8.5
Synch. Motor w/VFD (VSI) +Gear+Compressor2+Compressor1	9	8.5	8.5
Synch. Motor (DOL) Gear+Compressor2+Compressor1	10.5	10.5	10.5

Table 1 Shaft end sizing by type of driver

Thrust bearings are typically located on the non-drive end. However, when there are multiple compressor bodies driven by one driver, then all of the bodies except the last one will have drive through shaft ends. When the compressor must utilize a drive through shaft end, that shaft end size can be limited by the thrust bearing's inner diameter limit. Please see Figure 2 below. This could in turn lead to a requirement to replace the existing thrust bearing with one that has an ability to accept a larger drive through shaft end. This change normally impacts not only the thrust bearing but also the thrust bearing retainer and possibly the bearing housing.

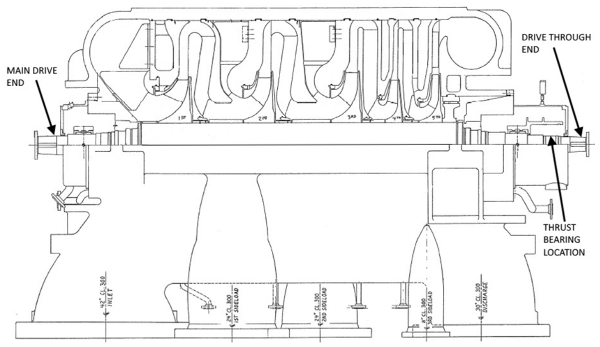

Figure 2 Location of thrust bearing on drive through end of compressor

Another issue that can present itself if the shaft end needs to increase significantly is the need to use larger main casing seals. This size increase could in turn trigger the need for new endwalls for both ends of the compressor if the larger seals cannot fit into the existing cavity.

Further, the increase in operating torques and shaft ends typically requires new couplings, or, as a minimum, new coupling hubs. When replacing the couplings, the existing guard may also need to be replaced.

It is clear that the shaft end size increase can add significant scope and complexity to a motorization project if the string rated power needs to remain the same with the new motor driver. Often it is the case that the additional scope to re-rate the compressor unit brings extra capital investment with great return on investment (ROI). On one hand the compressor scope increases, on the other hand the motor/drive system scope decreases with the lower power demand. This reduction of scope on the motor/drive system can sometimes outweigh the extra capital needed for the compressor re-rate.

IV. System Modifications and Challenges

One of the other components in the existing system that is typically impacted by a motorization project is the lubrication system. Many times, lube systems for turbine driven strings will need higher supply pressures for control oil for the governor, trip and throttle valve or other consumers. This need for high pressure oil may no longer be present with a motor driver. However, it is also very common to add a speed increasing gear to the string. Gears are typically a major consumer of both lubricating oil flow and heat load. It is normally necessary to complete an in-depth review of the existing lubrication system during the motorization project. The review should analyze all parts of the existing system, and should indicate those components which will need to be replaced, as well as to provide recommendations to optimize performance of the system, after the driver change. If the motor is directly coupled to the compressor, then the oil system could be potentially reduced. By removing the control oil, as well as the lack of need for a gear, the oil consumption could be reduced.

When changing drivers from a turbine to a motor and gear, there will likely be changes in both static and dynamic loading on the existing foundations. It is important to analyze these carefully during the design phase to ensure that there will not be any concerns with vibration issues after installation. In addition, the axial space required by the new motor and gear may also present a concern, especially if the intent is to reuse the existing main equipment baseplate or location. This could in turn drive a need to use smaller distances between the shaft ends or else a baseplate extension or a new, longer baseplate to accommodate the extra axial length. Those options assume that there is room in the plant to extend the length of the baseplate. The direct coupled motor to compressor can help alleviate this issue as well, saving sometimes very limited space at site, providing the existing baseplate can handle the new weight of the motor.

V. Benefits of Re-Rating

Even well-maintained turbomachinery in good working order can become obsolete through advances in technology, design, materials, or plant capacity. The availability and reliability of turbomachinery can have a critical effect on revenue. Each day that a piece of rotating equipment is off-line can result in millions in lost revenue and serious disruption of the downstream supply chain. Rerating equipment can increase reliability and reduce planned and unplanned outages that disrupt production and reduce revenue. Rerating a compressor increases the reliability of the unit through upgrades in not only the aero components, but potentially bearings and seals throughout the compressor. Potentially upgrading from non-abradable impeller shaft and eye seals, as well as balance piston and separation labyrinth seals with abradable seals reduces the leakage internally in the compressor thus saving power. Dry gas seals offer upgrades to the main casing seals in power savings and oil consumption savings. New bearings also provide an increase in reliability over often old liner type bearings by replacing them with new spherical seat bearings. Thrust bearings are also able to be more reliable with self-leveling equalized and Leading Edge Groove (LEG) type bearings. Changing the coupling type from gear and diaphragm oil type to dry type flexible couplings also increases the reliability of the string, as well as offering another point of tuning within the string to help avoid resonance and rotor dynamic instabilities. Finally upgrading the aerodynamic components of the compressor saves the greatest amount of power.

One of the main benefits of rerating is increasing efficiency of the compressor. Impeller + diffuser (stage) technology has greatly increased over the past 70+ years. Figure 3 shows approximate increases of stages technology historically.

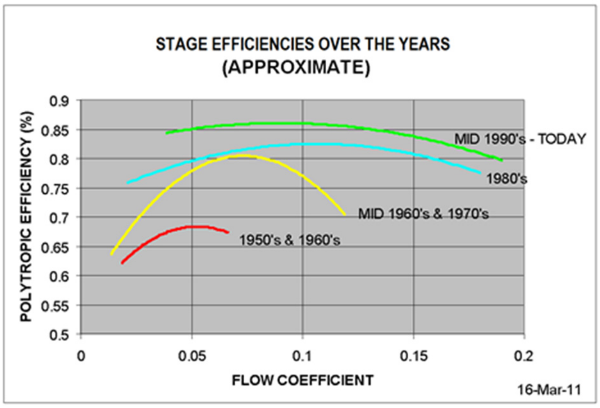

Figure 3 Stages Efficiencies over the years

By rerating the compressor, we are able to increase the efficiency of the design point while increasing the availability of the rotor for future process increases. See Figure 2 for a representation of this increase in throughput and efficiency increase. Rerates allow operational flexibility though the new technology being implemented and the ability to design for known future changes.

Summarized in table 2 below, are potential values of compressor power to power savings. Most typical instances where the rerate will be most beneficial would be older vintage compressors. As shown above in Figure 3, the operating efficiency of compressor staging has significantly increased over the past few decades. This correlates to Table 2, where the older the compressor and the higher the power the more potential savings are attainable.

Compressor Power (kW)	Potential Compressor Power Savings (kW)			
	1950-1960s	1960-1970s	1980s-1990s	1990s-Today
1000	200	70	50	30
3000	600	210	150	90
5000	1000	350	250	150
7000	1400	490	350	210
9000	1800	630	450	270
11000	2200	770	550	330
20000	4000	1400	1000	600
30000	6000	2100	1500	900
40000	8000	2800	2000	1200

Table 2 Potential compressor re-rate power saving per vintage

Table 3 shows the approximate metric tons per year for the same range of compressor powers and vintage of the compressor unit. This table was calculated with the assumption of 95% compressor availability and a factor of 0.000709 ton of CO2 for avoided kWh per EPA ref. 4.

Compressor Power (kW)	Potential CO2 emission Savings (metric Tons per year)			
	1950-1960s	1960-1970s	1980s-1990s	1990s-Today
1000	1180	413	295	177
3000	3540	1239	885	531
5000	5900	2065	1475	885
7000	8260	2891	2065	1239
9000	10621	3717	2655	1593
11000	12981	4543	3245	1947
20000	23601	8260	5900	3540
30000	35402	12391	8850	5310
40000	47202	16521	11801	7080

Table 3 Potential compressor re-rate CO2 saving per vintage

An equipment rerate is often a cost-effective, time-saving answer to changing throughput without the expense of investing in new equipment.

OPEX Analysis

In this analysis, several assumptions have been made. Firstly, an average industrial electricity cost of 68€/MWh is considered. Secondly, the CO2 price is assumed to be 60€/ton. Lastly, we assume a 95% availability of the compressor-motor train. Using these assumptions and referencing tables 2 and 3, we can estimate the potential yearly OPEX (operating expenditures) savings per compressor power targeted for motorization. For example, a compressor from the 1950s or 1960s, with an initial rotor absorbed power of 10 MW, could achieve power savings of over 15% when re-rated. Considering the assumptions of 68€/MWh, 60€/tonCO2, and 95% availability, this could lead to potential annual OPEX savings of more than 1.5M€. Figure 4 illustrates potential yearly OPEX savings for other compressor vintages and power levels, showing that greater compressor power correlates with higher potential OPEX savings.

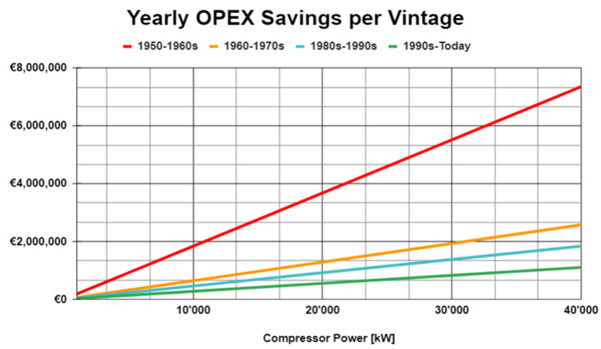

Figure 4 Yearly OPEX savings per vintage per compressor nominal power

VI. CASE STUDIES AND REAL WORLD APPLICATIONS

CUSTOMER A

A Customer in the Netherlands wanted to review and evaluate the feasibility of replacing the existing steam turbine drivers to electric motor drivers of 3 centrifugal compressors. An engineering study involving determination of the necessary changes to the compressors, specifically the shaft ends and potential bearing and seal changes, was the ideal platform to look into this electrification project. As part of the same study, a rerate evaluation was performed, as well as defining the minimum specifications of the new driver and auxiliary systems, to create RFQ's for the end user. After completion and analysis of the engineering study, the customer decided to electrify two of the three compressor drivers. These were two propylene strings from two separate OEM's (3707 rpm and 20395 kW) and (5214 rpm and 7310 kW).

Both strings of equipment were offered rerates to decrease the total power of the strings and allow the customer to select smaller motors and drive systems.

String 1	Existing Performance	Rerated Performance	Estimated Power Savings
Power (kW)	20395	18560	9.0%
Speed (rpm)	3707	3742	

Table 4 Power saving of String 1

String 2	Existing Performance	Rerated Performance	Estimated Power Savings
Power (kW)	7310	7190	1.6%
Speed (rpm)	5214	5298	

Table 5 Power saving of String 2

The rerate was purely efficiency driven. After showcasing the rerate potential, this customer requested that the existing pressure profile, flow rate, inlet temperature and inlet pressure stay the same. With these parameters frozen, an increase in rotor efficiency through new impellers, diaphragms, and seals was studied, and after some investigation and preliminary design work was developed to show about 9% in power savings of string 1. String 2 showed some improvement over the existing performance, but was not enough to provide commercial benefit to this customer.

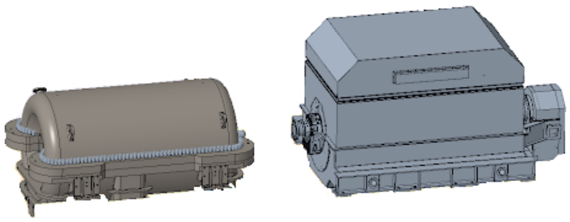

Figure 5 String layout of String 1

String 1 efficiency savings was a beneficial point for this customer to be able to take to the decision makers to provide positive FID (Final Investment Decision) for this electrification project together with the re-rate. In addition to the aerodynamic changes, other mechanical challenges need to be investigated that can affect the total CAPEX of a project. Compressor main casing seals, bearings, and shaft ends need to be evaluated based on the torsional criteria of a new electric motor over the existing turbine drivers.

The rerate of the string 1 unit saves about 9-10% in power. This results in power reduction to potentially reduce motor frame size, and compressor shaft sizing requirements. If the rerate was not chosen, the shaft end of this compressor would have had to be increased to accept the torque load of the new driver. The ability to reduce power saved a large amount of potential capital expenditure for new endwalls and casing seals.

For string 2, evaluation of the existing baseplate was necessary because the existing steam turbine and compressor were installed on a common baseplate.

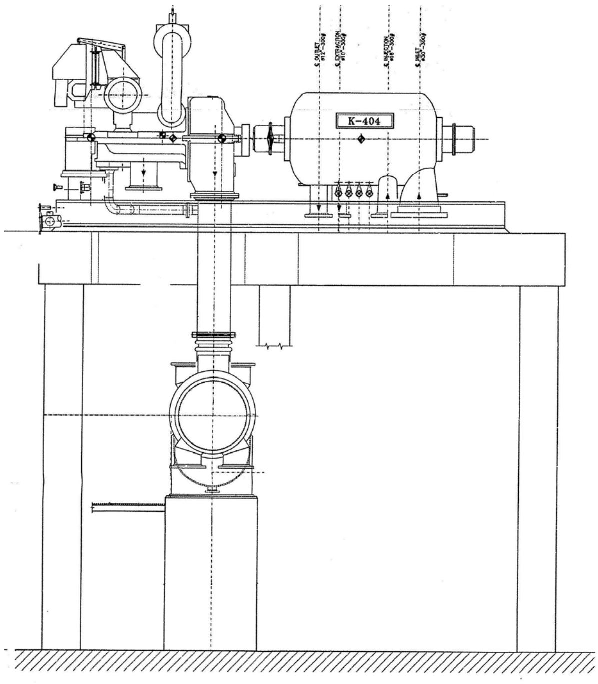

Figure 6 Existing layout for Sting 2

Different approaches were discussed. The first one was to modify the existing baseplate to receive the new motor baseplate. The second one was to cut and remove the turbine baseplate portion giving room for the new motor baseplate.

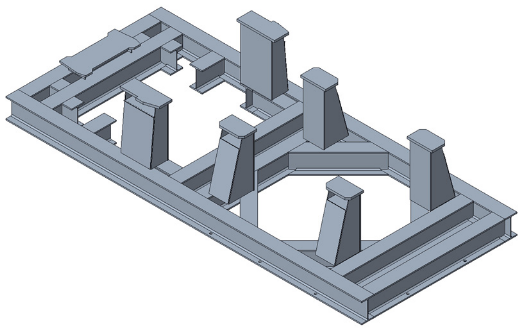

Figure 7 Existing common baseplate layout for string 2

The second method was preferred. Figure 7 illustrates the baseplate prior to the cut and figure 8 shows the baseplate after cut. However, the end user was concerned with the integrity of the existing compressor baseplate after the cut and requested that an analysis be conducted.

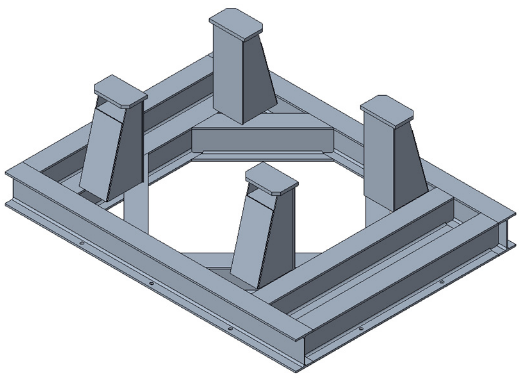

Figure 8 Cut Baseplate

The objective was to perform Static structural Analysis, Modal Analysis, and Harmonic Analysis on the Base Plate Assembly. Stress Analysis considering the point mass, Modal Analysis, Force Response Analysis was performed on the Baseplate Assembly.

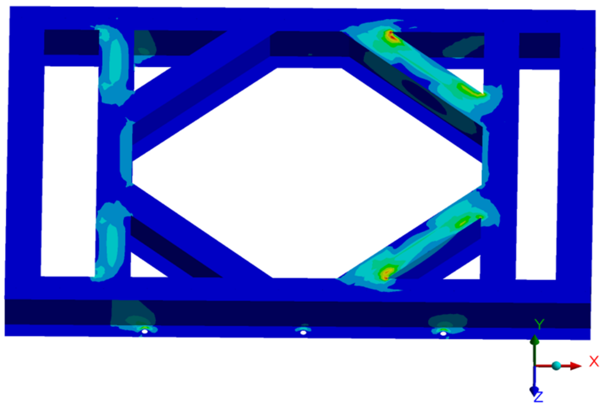

Figure 9: Modal analysis(Equivalent Stress) results of baseplate after cut

The maximum stress observed from static stress Analysis was well within the allowable limit. Based on the dynamic analysis of the base plate, the Modal Analysis showed no modal frequency is of concern with the operating speed of the Compressor. Harmonic Analysis showed that based on the maximum Alternating stress over the frequency and Mean stress the design meets the Goodman Criteria for infinite life.

CUSTOMER B

A different customer also located in the Netherlands also decided to replace their existing main driver, which in this case was a gas turbine, with a synchronous motor. The existing gas turbine was operating at approximately 5103 rpm, whereas the compressor was running at 3840 rpm, so there was a speed-reducing gear between the gas turbine and compressor. Please see the existing string layout in Figure 10. For this equipment string, there is also a steam turbine driver rated for 8.2 MW on the opposite end of the compressor which also provides some power to the string. This added an additional layer of complexity to the motorization project execution.

This project began as an engineering study in order to provide a new speed-torque curve for the new motor. The customer was initially uncertain as to whether to utilize a fixed speed driver or else add a VFD to allow for speed variation. However, during the course of the engineering study, there were checks made of the existing compressor shaft end. It was determined that, due to the reasons above in section 3, the existing compressor shaft end was undersized for the new synchronous motor driver. The situation was exacerbated with a fixed speed driver, so the decision was made to utilize the VFD for the new motor. Please see Table 6 below for the impact of the synchronous motor starting and control type upon the required shaft end size as well as the effects of using a Holset type coupling on the low speed side.

PRELIMINARY RESULTS OF SHAFT END ANALYSIS				
26 MW Motor Rating for all cases	Hyd. Fit shaft end size	Tapered & Keyed shaft end size	Hyd. Fit shaft end size - Holset coupling	Tapered & Keyed shaft end size - Holset coupling
Synchronous DOL start motor	11.5 inch	12 inch	10 inch	10 inch
Synchronous VFD motor	7.5 inch	8.5 inch	N/A	N/A
Synchronous VFD for starting only	7.5 inch	7.5 inch	N/A	N/A
The above analyses assumes no power input or inertia from the outboard steam turbine. It is assumed to be uncoupled from the string.				

Table 6 Compressor shaft end sizing based upon starting method, shaft end design, and low speed coupling type

Thus, the scope wound up becoming a new bare shaft for the compressor with a larger main drive shaft end. In addition, since the shaft end size became a larger diameter than the existing journal bearings, the journal bearings also have to be replaced with a larger size. This also required new bearing retainers for both ends. Fortunately, there was no need to replace the thrust bearing, main casing seals, or endwalls.

Figure 10, below, shows the existing string layout prior to changing out the gas turbine and speed reducing gear for the new motor and speed increasing gear.

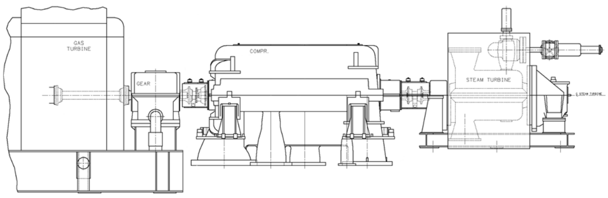

Figure 10 Existing Sting layout

Figure 11, below, illustrates the string configuration with the new motor and speed increasing gear.

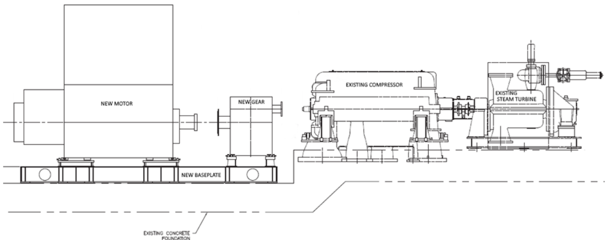

Figure 11 Revised String layout with motor

While preparing the proposal for the compressor modifications that would be needed for the new motor driver, an evaluation for the potential power savings with a full internal rerate of the compressor was investigated. The existing aerodynamic components in the compressor were from the mid-1990s, so their efficiency was moderate but not as high as current aerodynamic components. Thus, the rerate would have saved around 7% in power at the original design conditions. The gains could potentially be even greater if their current process differs from design. We explained the benefits of both CAPEX and OPEX to the customer of the rerate. However, the customer decided not to pursue the rerate at this time.

The final primary scope of this particular motorization project then became a new motor and VFD, new gearbox, new low speed and high speed couplings, a new baseplate under the new motor and gear, a new bare shaft for the compressor with a larger main drive shaft end, new journal bearings for the compressor, and new end cover and end seal for the compressor The customer also purchased a site audit of their existing oil system to verify acceptability with the new service conditions Finally, there was service center work to destack and restack the existing rotor components on the new bare shaft and then balance the rotor.

Lessons Learned and Best Practices

Both of these projects in the Netherlands explored the transition from traditional turbine drivers to electric motor drivers for centrifugal compressors and provided valuable insights and best practices regarding electrification and efficiency improvements.

To begin with, it is not possible to overstate the importance of an adequate engineering study. The studies are a baseline analysis of the feasibility and efficiency benefits of utilizing electric motor systems. By detailed observation of the infrastructure and systems in place, these studies allow the stakeholders to understand what is needed and which improvements can be made, thus maximizing the understanding of the available options and their benefits.

The engineering study evaluates the need to overcome mechanical problems that may arise in the transition to electric motors. Mechanical aspects, including the compatibility of compressor shafts, bearings, and seals with new motor systems, are major considerations that can significantly impact capital expenditure. Overlooking these aspects can lead to surprise costs and project delays. Thus, careful examination of available materials and conditions is required to ensure compatibility of existing equipment with the new drive system.

The engineering study should also contemplate compressor rerating. By optimizing the performance of existing equipment, rerating can lead to significant power savings. This alone can help to justify the initial investment in electrification, and therefore it is a worthwhile strategy for companies looking to improve their energy profile. The example of customer A is a demonstration of this valuable evaluation. However customer B did not pursue the re-rate despite great potential in power savings.

Table 7 compares the capital costs needed to be spent versus re-rating in relative units. Re-rating the compressor of customer A was 2.87 times more expensive than modifying the shaft/bearings/seals of the compressor to be capable of working with the new electrical motor. For customer B that factor was slightly higher for a lower power benefit. While this could have been a factor of decision, in reality the complexity of customer B compressor train was significantly higher. The added complexity of the steam turbine at the other end of the compressor train made the evaluation more difficult in terms of the benefits together with the steam balance of the plant - this could have been decisive for customer B in not going forward with the rerate. Table 7 also compares the rerate return on investment (ROI) in years for both customers. Customer A's investment in the rerate yielded a return on investment (ROI) in 1.5 years. Had Customer B chosen to proceed with a similar rerate project, they could have realized a comparable ROI of 1.7 years. The ROIs in Table 7 did not factor in the CAPEX savings from using smaller frame size motor/drive systems for rerated compressors. If these savings were included, the overall benefit would be even greater.

	CAPEX MODIFICATION	RE-RATE	POWER SAVINGS in kW	ROI
CUSTOMER A	1	2.87	1835	1.5
CUSTOMER B	1	2.95	1332	1.7

Table 7 CAPEX and ROI for compressor modification vs re-rate

In summary, these engineering study practices can serve as a template for future electrification projects, emphasizing the need for strict planning and execution. By prioritizing meticulous engineering studies, rerating evaluation, and mechanical analysis with caution, industries are better positioned to address the complexities of electrification, ultimately realizing enhanced efficiency and sustainability in their operations.

VII. Conclusion

Summary of Findings

This paper has explored the significant impact that compressor rerating can have on electrification/motorization projects for industrial sectors but mainly for oil and gas, petrochemicals, refining, LNG and ethylene production. It emphasized the role of centrifugal compressors as energy-intense machines, that are critical for these industries, and how electrification/motorization of these machines can play a big role in cutting Scope 1 emission levels. Furthermore it was discussed how engineering studies are important and perhaps the norm to provide visibility on the complexity of these projects. It was demonstrated that rerating provides real time power savings and can reduce scope of upgrading other components by reducing the power and torque demands. The available capital and the complexity of the system will determine the final investment decision and selection for the upgrade path.

Future Outlook

While the future of centrifugal compressor motorization and re-rating appears promising, with substantial potential for further efficiency gains and emission reductions, there are indications that the complexity of these projects and its economics aren't always as colorful. From the authors perspective and on-going other projects, it appears that motorization of centrifugal compressors projects are more favorable in regions that have carbon tax implementation, ETS (emission trading systems), public funding and in plants that are located in areas with high degree of industrialization where excess steam can be sold or utilized somewhere else. These are key factors and some of them are common to both customer A and B.

It is always advantageous to involve the compressor manufacturer or experts early in any motorization project. This allows for the identification of potential bottlenecks in the driven equipment through necessary studies and the development of the best scope of supply for existing and future operating conditions. Only with this understanding can the new driver be properly selected and sized, ensuring maximum efficiency for the compressor-motor system.

VIII. References

[1] Durantay, L., Gemert, D., Lava, J. and Spannagel, D. “A Success Story of Steam Turbine Replacement by High Speed Electric System Driven Compressor,” IEEE PCIC Europe, PCIC-EUR21_08, IEEE 2021.

[2] Malnedi, K., Harvey, S., Rauber, A. “Mechanical and electrical considerations of electric motor drivers for large centrifugal compressors in ethylene cracker applications,” Spring GCPS 2022

[3] Mcbain, M., Malnedi, K., Sanchez, T., and Das, S., “Considerations for Making the Switch from Steam Turbines to Large Motor Drives for Ethylene Plants,” IEEE Paper PCIC-2023-33, IEEE 2023

[4] Brun, K., Kurz, R.,”Compression Machinery for Oil and gas”, Elsevier 2019

[5] https://www.epa.gov/energy/greenhouse-gas-equivalencies-calculator#results

[6] https://www.oecd.org/en/topics/tax-policy.html

IX. VITA

Tomás Sanchez represents Ebara Elliott Energy as a Regional Sales Manager, covering Europe and Africa for Engineered Solutions which includes re-rates, modifications and upgrades of turbomachinery for Oil & Gas, Petrochemicals, Refining, LNG, Ethylene and power generation applications. Previously, Tomás had several roles in Elliott namely in Houston, TX as Field Service Regional Manager, in Switzerland as Technical Support Manager and in EMA (Europe, Middle-east and Africa) as Field Service Engineer. He holds bachelor’s and master’s degrees in mechanical engineering with focus on thermodynamics and ICE (internal combustion engines)
tsanchez@elliott-turbo.com

Brian Stewart represents Ebara Elliott Energy as a Senior Application Engineer, covering the Americas region for Engineered Solutions, which includes re-rates, modifications and upgrades of turbomachinery for Oil & Gas, Petrochemicals, Refining, LNG, Ethylene and power generation applications. He holds a Bachelor of Science degree in Mechanical Engineering from the University of Pittsburgh.
bstewart@elliott-turbo.com

Jeffrey Jacob represents Ebara Elliott Energy as an Application Engineer, covering Europe, Middle East and Asia Pacific regions for Engineered Solutions which includes re-rates, modifications and upgrades of turbomachinery for Oil & Gas, Petrochemicals, Refining, LNG, Ethylene and power generation applications. He holds a bachelor's degree in Mechanical Engineering Technology.
jjacob@elliott-turbo.com