Enhancing Efficiency and Reducing Carbon Footprint in Centrifugal Air Compressors with active magnetic bearings: High-Speed Operation for Specific Energy

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ABSTRACT

This study compares high-speed direct drive centrifugal compressors (DDCC) and oil-free screw compressors (OFSC), analyzing boundary conditions, component performances, and energy consumption. Results include the energy efficiency of high-speed DDCC configurations, weight reduction benefits and total cost of ownership (TCO) comparisons over 10 years. The results emphasize the potential of high-speed centrifugal compressors for sustainable industrial practices.

1. INTRODUCTION

To align with global climate change mitigation goals, industrial companies are reassessing their carbon footprint. This reevaluation is spurred by heightened industry awareness of its contribution to global warming, with government incentives and regulatory requirements further propelling this commitment. High-energy consumption sectors, particularly air compression, are receiving special attention.

In response, companies are dedicated to reducing greenhouse gas (GHG) emissions associated with their air compression operations. These efforts involve minimizing gas leaks, reducing pressure drops, and adopting more efficient energy management practices. A pivotal strategy involves embracing variable-speed technology, enabling compressor speed adjustments based on compressed air demand, leading to enhanced energy efficiency.

Companies now face the strategic decision of selecting the appropriate compressor type best suited to their unique needs. This decision depends on factors such as facility size, fluctuations in compressed air demand, operational constraints, and heat recovery capabilities.

Active magnetic bearing (AMB) technology has gained prominence across various fields, making an in-depth discussion of its operating principles and control mechanisms unnecessary (Schweitzeir *et al*, 2009). Its application in areas like chillers (Byrd *et al*, 2021) and turbo blowers (Kinoshita and Aota, 2015) has demonstrated significant energy savings, resulting in an almost 30% reduction in carbon footprint compared to traditional "oiled compressors". Furthermore, the sealed design of compressors using AMB in gas applications minimizes gas leaks, further mitigating the carbon footprint (Gelin and Lucas, 2023).

Focusing on high-speed air compression, the multi-stage configuration is favored for its specific energy gains. Typically, two or three stage compressors, incorporating intercooling, achieve the required pressure ratio with enhanced overall efficiency.

Recent comparisons of energy savings and CO_2 emissions between a three-stage compression unit equipped with variable-speed magnetic-bearing centrifugal compressors operating at nearly 36000 rpm and a two-stage unit with oil-free screw compressors operating at around 3000 rpm, both utilizing variable speed, have emphasized the significant energy and carbon emissions reductions achieved by the centrifugal solution, amounting to minimum 10% for operating pressures up to 7 bar(g) (Lopatin, 2023a).

This study aims to demonstrate how increasing speed can significantly improve specific energy efficiency. The efficiency gains of combining high-speed centrifugal compressors designed for different speeds will be explored, focusing on low-pressure air compressor units (ACU) up to 10 bars (g).

Section 2 details the boundary conditions of the study. Section 3 covers the performance of various components constituting the air-compressor. Section 4 compares the specific energy consumption of ACUs based on recorded measurements for two existing configurations, while the improved configurations (operating at 60000 rpm) are based on simulations and incorporate lessons learned from measurements conducted on centrifugal compressors spinning at 36000 rpm and certain components of the value chain (motor, VFD, cooling system) as explained in section 3. Section 5 summarizes the weight reductions achieved with higher speeds. Section 6 presents the total cost of ownership over a 10-year period.

2. BOUNDARY CONDITIONS AND HYPOTHESIS

2.1 Compressor units

This paper analyzes water-cooled (WC), oil-free (OF) air compressor units (ACUs), focusing on two distinct topologies: high-speed direct drive centrifugal compressor (DDCC) and oil-free screw compressor (OFSC), as a reminder oil-free means no oil in the compression chamber.

The high-speed centrifugal compressor (HSCC) in Figure 1 utilizes a permanent magnet motor (PMM) with active magnetic bearings (AMBs), while the other configuration in Figure 2 is driven by an induction motor (IM) with lubricated bearings. DDCC and OFSC units use variable frequency drives (VFDs) for the flow variation control.

For the three-stage (3S) HSCC unit with VFD, various speed combinations are examined, assuming one motor capable of spinning up to 36000 rpm and the other up to 60000 rpm for the same shaft power. In contrast, for the two-stage (2S) OFSC unit with VFD, the induction motor (IM) rotates at approximately 3000 rpm.

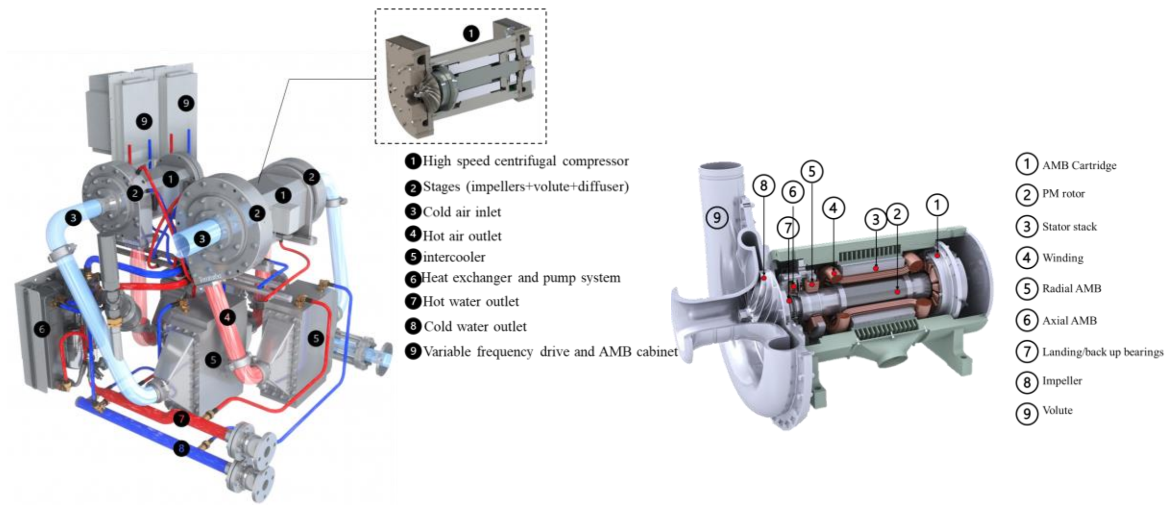

Figure 1 : Illustration of a three stage WC high-speed direct drive centrifugal compressor with VFD (ACU from Tamturbo)

Figure 2: Illustration of a two stage WC oil-free screw air compressor with VFD (ACU from Atlas Copco)

The ACUs can deliver pressures of up to 10 bars(g), but the study focuses on a service pressure of 7 bars(g), aligning with many industrial requirements.

Performance analysis follows the program established by the Compressor Air and Gas Institute (CAGI) and ISO 1217 standards. Evaluations focus on specific energy (kW/(m³/min)) based on measurements of input electrical power—including cooling devices and accessories—and output flow at specified discharge pressures and environmental conditions. The results for both the OFSC and DDCC version spinning at 36000 rpm are based on power and speed measurements in accordance with ISO standards. For the DDCC version spinning at 60000 rpm, results are extrapolated using lessons learned and the same environmental conditions.

2.2 Studied Scenarios

Figure 3 illustrates two typical scenarios that may be encountered in factories. Scenario n°1 portrays operations running at near maximum capacity throughout the week, with a noticeable reduction in workload observed during shifts and extended breaks. In scenario n°2, we see a similar pattern where the ACU operates at nearly full capacity during the daytime, with a decrease in demand during lunch breaks. Furthermore, in the late hours of the night, there is a significant reduction in flow demand, indicating that not all workstations are operational during this period.

For the OFSC, over time, there is a gradual degradation of the rotor coating due to pressure, temperature, humidity, and wear, leading to a reduction in load capacity (Nordquist *et al*, 1992). This reduction can amount to 10% within the initial thousand hours of operation and may further increase over the subsequent years prior to overhaul (IR, 2005).

The recommended maintenance interval for critical components in oil-free screw compressors is every 5 years, involving replacing parts such as male/female rotors, bearings, gears, wave springs and seals to restore performance.

Although degradation typically occurs gradually, for simulation/calculation purposes, a simplified degradation trend- as illustrated in Figure 4- is considered in evaluating the energy consumption of the 2-stage OFSC unit.

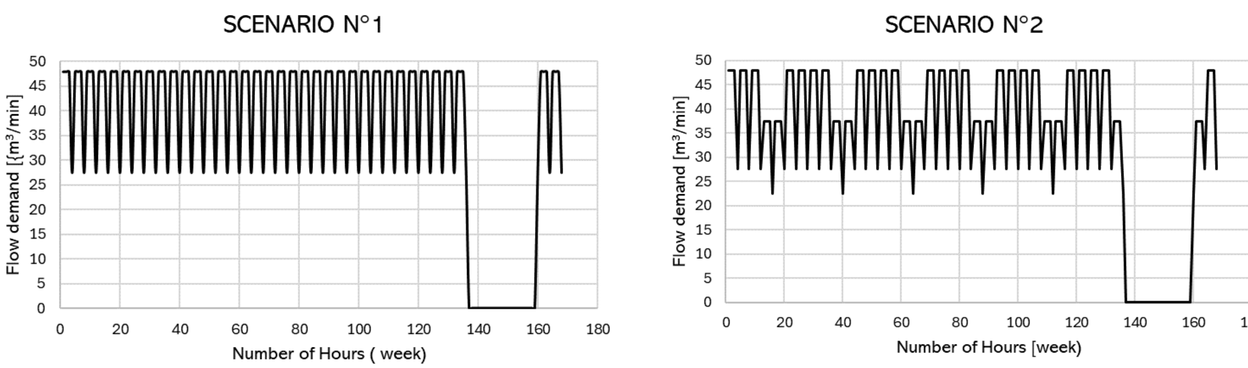

Figure 3: Example of scenarios of flow demand at 7 bars (g)

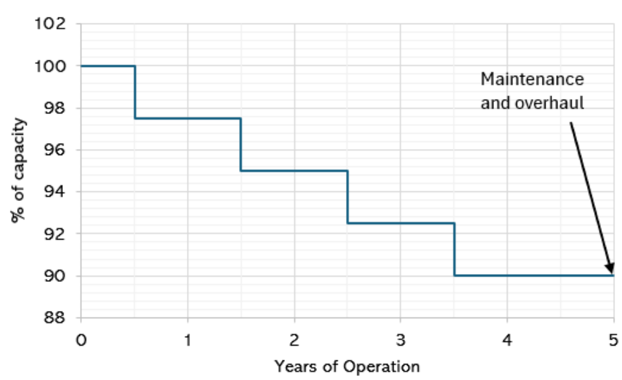

Figure 4: Scenario of the performance reduction over the years on the OFSC due to coating degradation

The energy consumption over 10 years, is based on these scenarios, assuming:

- Weekly demand repeats each week over the year.

- Each power consumption level (kW) is maintained for one hour, to facilitate the calculation of energy consumption (kWh)
- Two weeks of ACU shutdown yearly.
- ACUs do not operate during part of the weekend, as shown in Figure 3.
- Pressure demand remains constant.
- The performance degradation considered for the energy evaluation of the OFSC is based on Figure 4

Overhaul and replacement of parts are carried out during shutdown at the end of the fifth year, the OFSC recovers its original performance during the following year.

2.4 Studied DDCC configurations

For the highest speed ACU (60000 rpm), two configurations (3-stage or 2-stage) are possible. The choice depends on the customer's required pressure and volume flow. Figure 5 illustrates two possible DDCC configurations that could meet the targeted flow demand of around 49 m^3/min and pressure of 7 bars (g).

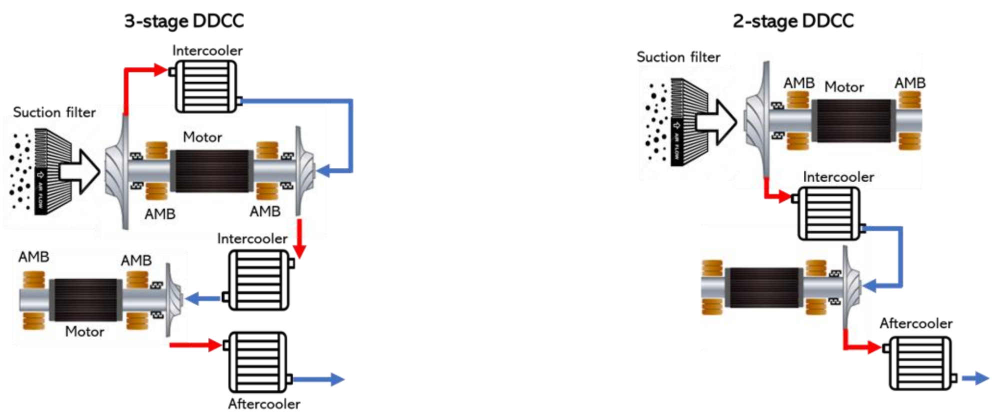

Figure 5: Artist view of a 3-stage DDCC and 2-stage DDCC configurations.

2.5 Motor performance

The performances and thermal behavior of the high-speed motors are evaluated thanks to a back-to-back test as illustrated in Figure 6.

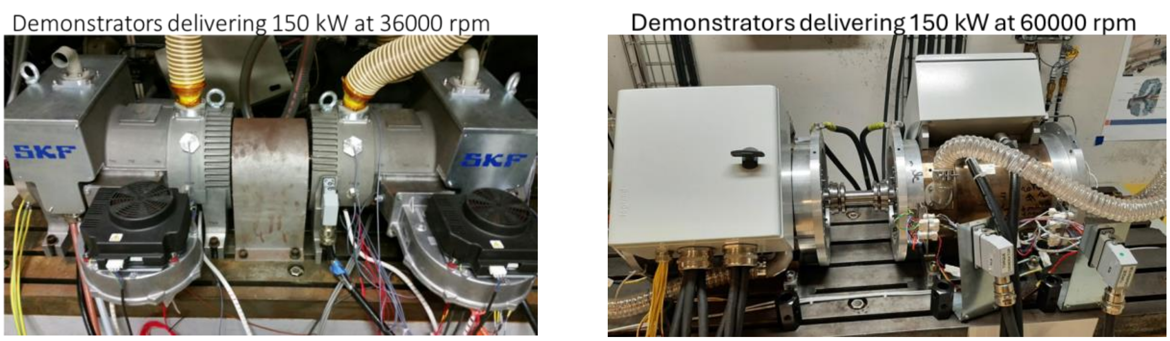

Figure 6: Back-to-back test to evaluate the performances and thermal behavior of the high-speed motors

2.6 VFD configuration

Two extensively used inverter types in high-speed applications, especially with Permanent Magnet (PM) motors, are the Two-Level Pulse Width Modulation (2L PWM) and Three-Level Neutral Point Clamped PWM (3L NPC PWM) (Holmes and Lipo, 2006a).

Figure 7 illustrates the configuration of 2L PWM and 3L NPC PWM inverters. For the 60000-rpm motor, the 3L NPC PWM switching at 16 kHz will be used without a sine filter, while for the 36000-rpm motor, the 2L PWM with sine filter is used with a switching frequency of 8 kHz.

In high-speed applications (above 30000 rpm), maintaining high-quality current and voltage waveforms is crucial to avoid additional losses and heat in PMM. The 2L PWM VFD commonly used in such applications is often paired with a sine filter to mitigate harmonic distortion. Design considerations for a sine filter include supply frequency, switching frequency, resonant frequency, and load (Nerubatskyi *et al*, 2021).

The 3L PWM inverter, gaining popularity in low-voltage high-speed applications, offers smoother current with minimal harmonics and eliminates the need for a sine filter. It provides advantages such as lower voltage steps and output current ripple (Holmes and Lipo, 2006b).

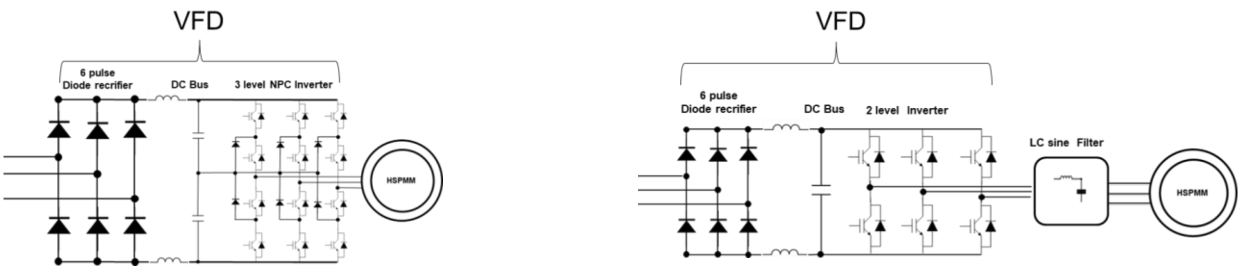

Figure 7: VFD topologies used in the HS DDCC unit

3. PERFORMANCES OF SUB-COMPONENTS USED ON DDCC

3.1 AMB and bearing technology

In high-speed centrifugal compressors, AMBs emerge as a state-of-the-art solution. They offer a sealed design, eliminating leakage, and provide maintenance-free, contamination-resistant operation. Their ability to actively control and monitor the rotor's real-time position enhances operational flexibility and responsiveness to system dynamics.

Figure 8 succinctly illustrates the superiority of AMBs compared to other bearing systems. In high-speed applications, guide elements are typically air bearings or AMBs. Unlike air bearings, AMBs boast zero friction at starts, high damping, and the capacity to handle both high loads and power ranges with remarkable reliability (McDonald, 1998) (Irving and Ibets, 2013).

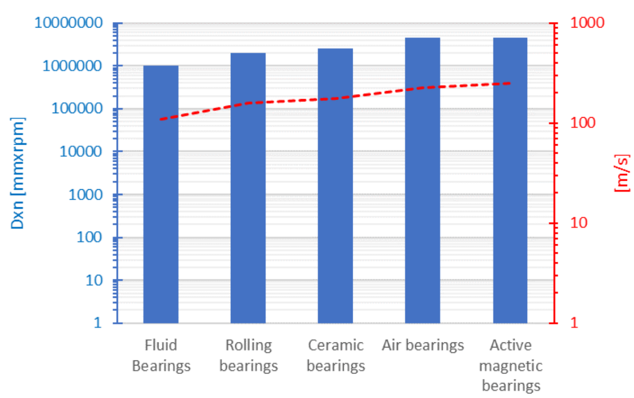

Figure 8: Speed and Diameter capabilities per bearing technology

3.2 Impeller sizing and stage compression

Designing each impeller stage follows established procedures outlined in (Lüdtke, 2004) (Gambini and Vellini, 2020). The fundamental equations concerning head rise, efficiency, and volume flow for a single-stage compressor are well-established and do not require repetition. Sizing compression wheels depends on the pressure and head targets; for instance, the same impeller speed and tip speed can offer varying results depending on impeller shape (blade, hub, inducer).

Figure 9 provides comparative results. Simulations assume all stages operate at the same rotational speed, with each motor delivering up to 150 kW on the shaft.

Transitioning from a compression wheel sized for 36000 rpm to one for 60000 rpm enhances isentropic efficiency for the same outlet pressure within flow rates of 40-55 m^3/min.

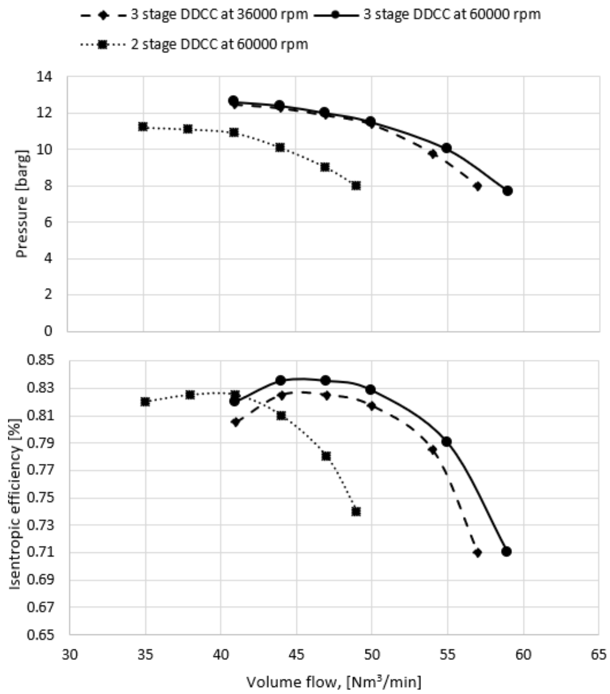

Figure 9: Predicted performance of the DDCC for different speeds and flows.

3.3 Electric motor and VFD performances

Figure 10 shows the motor efficiency map, excluding cooling system losses. The efficiency map across a wide power/speed range is extrapolated from specific measurement points obtained through the back-to-back test, combined with a calibration of the analytical model.

The results reveal higher efficiency in the motor spinning at 60000 rpm, due to modifications (material type) made on the iron sheet to mitigate iron losses in the 60000-rpm motor. While this modification comes at a higher cost per iron sheet, the overall quantity required is diminished due to the motor's more compact design.

For power ratings exceeding 100 kW, industry-standard 2-level (2L) Variable Frequency Drives (VFDs) with Silicon (Si) IGBTs operate at up to 16 kHz switching frequency. However, this increased frequency may diminish the VFD load capacity, particularly in adverse ambient or coolant temperatures. Transitioning to a 3-level Neutral Point Clamped (NPC) VFD allows for higher switching frequencies without sacrificing efficiency. This is attributed to the halved DC voltage applied to each switch (IGBT), resulting in lower dv/dt, reduced power device stress, and decreased conduction and commutation losses. Figure 11 illustrates the efficiency mapping of both VFD topologies, including the entire chain from grid to the output of the VFD. The mapping is conducted based on an analytical model calibrated based on specific measurement points. It shows that the efficiency of a 3L VFD surpasses that of a 2L VFD at the same switching frequency. It's important to note that coolant temperature impacts the load capacity limit. For illustration, coolant temperature isolines are included in Figure 11, to indicate how a given temperature could affect the power availability.

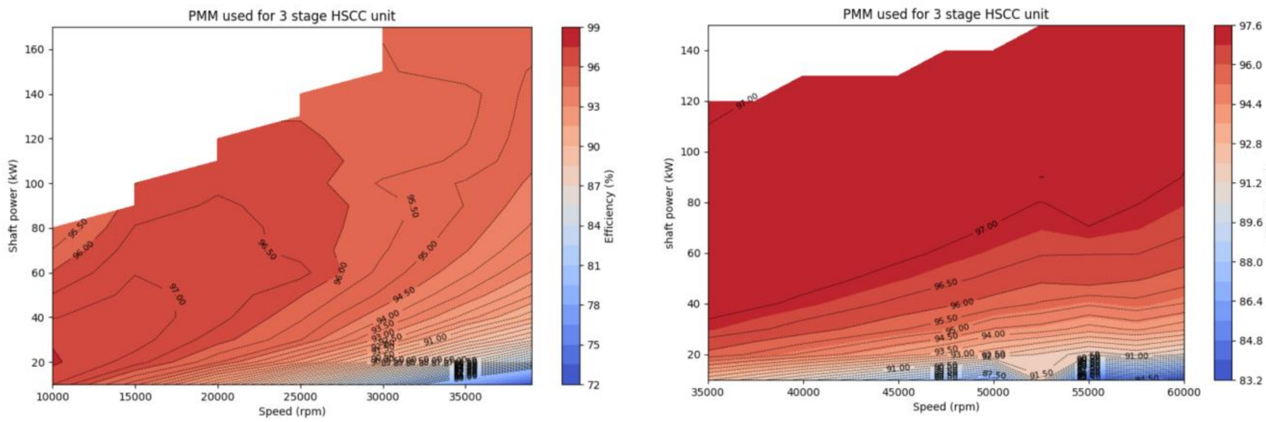

Figure 10: Motor efficiency versus speed and shaft power for the studied high speed PM motors used in HS DDCC

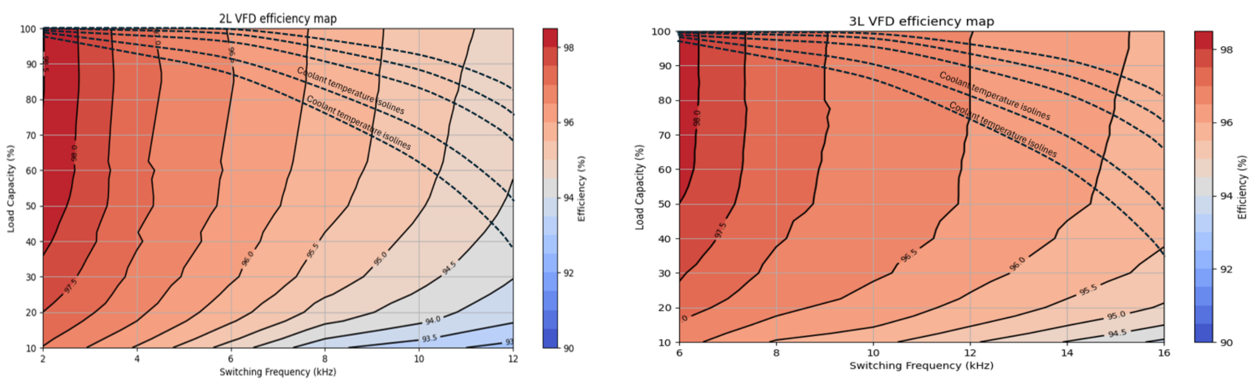

Figure 11: VFD efficiency used in HS DDCC

4. RESULTS AND DISCUSSION

4.1 Compressor unit performances

In (Lopatin, 2023b), an in-depth exploration of various annual usage scenarios for compression units was conducted. This encompassed a comparative analysis between a 3-stage HS DDCC (V1) and a 2-stage OFSC, across a spectrum of discharge pressures. Building upon this foundation, our study aims to juxtapose the performance of compression units previously examined with a distinct 2-stage HS DDCC (V2) and 3-stage HS DDCC (V3).

Figure 12 shows the specific energy versus flow at 7 bars (g) of the:

- 2-stage OFSC (1) driven by one motor that can deliver a shaft power of 300kW at 2982 rpm.
- 3-stage HS DDCC V1 (2) driven with 2 motors, each motor can deliver a shaft power up to 150kW at 36000 rpm, one moto-compressor having 2 impellers.
- 2-stage HS DDCC V2 (3) driven by two motors, each motor can deliver a shaft power up to 150kW at 60000 rpm. Each moto-compressor has one impeller. Results based on extrapolation considering the lessons learned and using the same conditions in accordance with ISO 1217.
- 3-stage HS DDCC V3 (4) driven by two motors, each motor can deliver a shaft power up to 150kW at 60000 rpm. one moto-compressor having 2 impellers.

Notice that the performances of the OFSC (1) are given, considering no wears on the coatings of both male and female rotors.

Analysis of Figure 12 highlights several points:

- In the flow range between 75% and 100%, the HS DDCC configurations exhibit better specific energy, with a significant reduction for 3-stage DDCC V3.
- In the range from 55% to 75%, the versions operating at 60000 rpm maintain better specific energy, with a preference, however, for the 2-stage HS DDCC V2.

- Between 25% and 55%, centrifugal versions can no longer regulate the minimum flow, thus imposing a potentially excessive flow that needs to be blown off. During operation between 25% and 50%, 2-stage OFSC, although with degraded specific energy, consumes nonetheless less energy than centrifugal compressors.

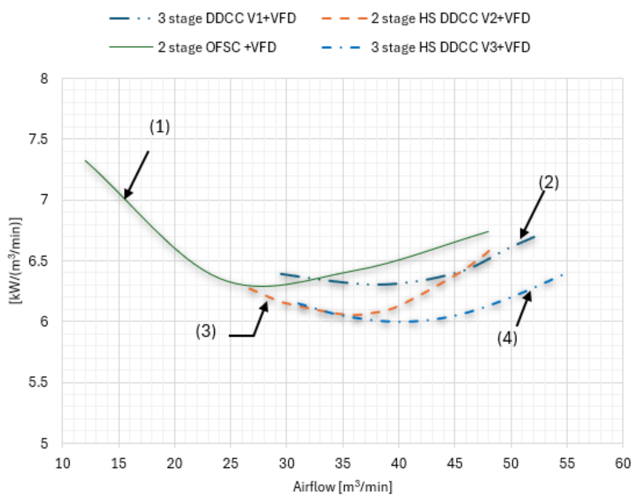

Figure 12: Specific energy (kW/(m^3/min)) of the DDSC (V1, V2, V3) and the OFSC at 7 bars (g)

4.2 Energy savings over 10 years

Figure 13 provides a summary of the energy consumption gains, translated in per unit for clarity. The calculation is based on curves from Figure 12, using scenario depicted in Figure 3 and considering for the OFSC a performance degradation trend as given in Figure 4.

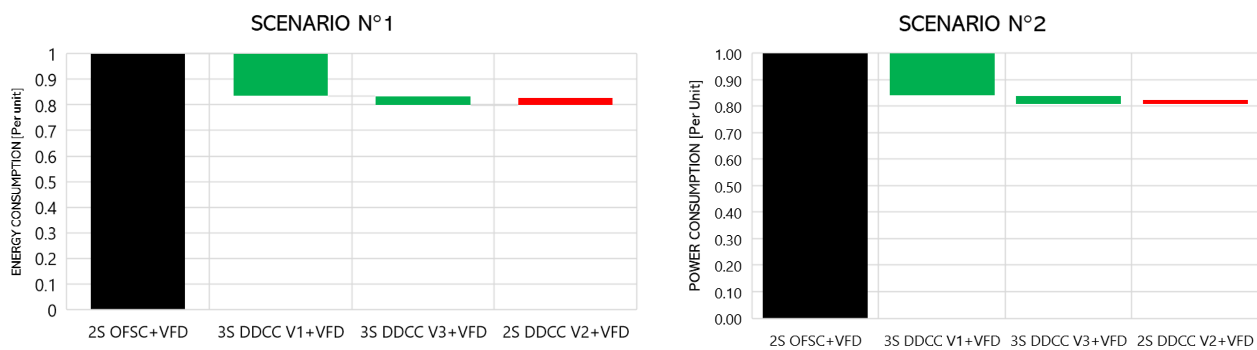

Figure 13 : Energy consumption reduction in per unit Vs APU configuration

In both scenarios, all HS DDCC configurations exhibit a reduction in energy consumption at 7 bars (g) operation. Utilizing a 3-stage DDCC (V1) spinning at around 36000 rpm enables 17% to 18% reduction in energy consumption, while employing a 3-stage DDCC (V2) spinning at around 60000 rpm allows for a reduction of around 20% in energy consumption. The 2-stage DDCC (V2) has a similar energy consumption reduction as the 3-stage DDCC (V1) spinning at 36000 rpm.
GHG emissions associated with scope 2, which pertains to energy consumption, reflect an equivalent percentage reduction. However, the actual level of CO_2 equivalent emission (TCO_2e) will vary depending on the country of analysis, whether it is France, Sweden, the USA or China.

4.3 Weight savings

Figure 14 illustrates the weight, in per unit, of the studied compression unit configurations. As already mentioned in (Lopatin, 2023c), DDCC with AMBs inherently weigh less than OFSC. However, what is highlighted in this graph is the advantage of having a higher speed. Transitioning from DDCC V1 (~36000 rpm) to DDCC V3 (~60000 rpm) further reduced the mass, primarily due to more compact motor-compressors.

Additionally, removing the sinusoidal filter with the use of a 3-level NPC VFD configuration also contributed to mass reduction. The most notable aspect is seen in DDCC V2 in weight reduction, where utilizing only 2 stages eliminated the need for an intercooler, an aerodynamic element (impeller, volute, diffuser), and ducting. It's worth noting that employing only two stages for the targeted flow and pressure was made possible by increasing the speed from 36000 rpm to 60000 rpm, naturally necessitating the resizing of all components. When considering the overhauled/replaced parts after 10 years, the weight difference becomes even more pronounced. During preventive maintenance, electronic cards with capacitors and small cooling fans in all ACUs are replaced every 5 years as well.

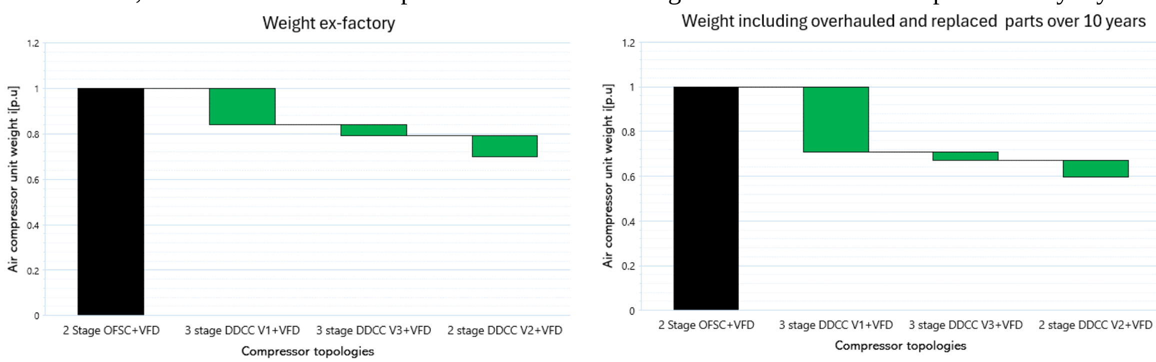

Figure 14: Weight in per unit of the different ACUs ex-factory and after 10 years operation.

4.4 Total Cost of ownership

The energy cost considered in the analysis is based on the average price per Megawatt-hour (MWh) in the USA in 2023, which stands at around \$81 per MWh (US energy information administration).

Acquisition and maintenance costs are averaged based on the prices set by compression unit suppliers for SKF production units in Europe mainly. Due to varying commercial strategies among compressor suppliers, which affect acquisition and maintenance costs differently, we strived to maintain price consistency.

Figure 15 provides an overview of the total cost of ownership (TCO) over a period of 10 years. The reference point, measured per unit, is the Capital Expenditure (CAPEX) of the two-stage OFSC, set as one.

Despite the initial cost advantage of OFSC over HS DDCC solutions, the substantial maintenance demands, and the need for periodic replacement of crucial compressor components (rotors, seals, bearings, gears) contribute to an overall higher total cost. Moreover, the energy consumption adds to this cost escalation. As depicted in Figure 15, over a decade, potential savings in TCO ranging between 14% and 18% are observable, contingent upon the chosen HS DDCC configuration and the specific scenario under analysis.

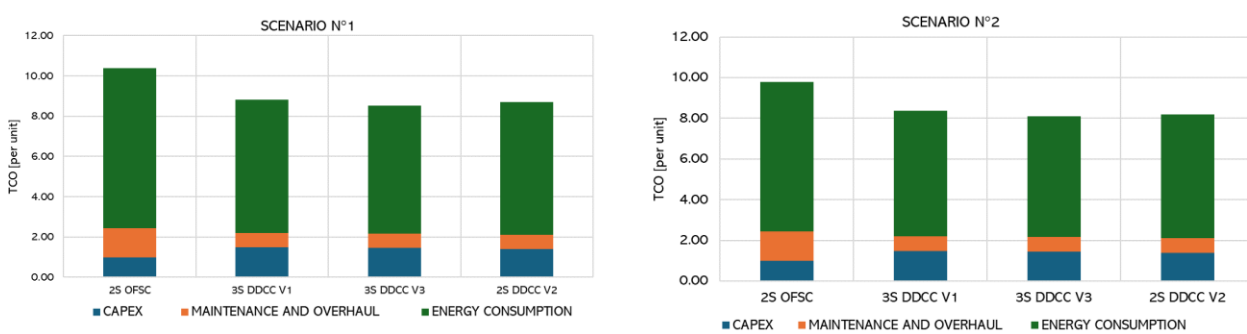

Figure 15: TCO over 10 years of exploitation

5. CONCLUSION

Based on the results presented in this paper, centrifugal compressors equipped with active magnetic bearings demonstrate excellent performances in oil-free low-pressure air compression category. The utilization of magnetic bearings to achieve high-speed, leads to notable reductions in energy consumption, reaching up to 20% in the analyzed scenarios at 7 bars (g), and substantial mass reductions compared to oil-free screw compressors operated by VFDs, excluding spare parts. It suggests even greater potential energy savings if the pressure demand falls below 7 bars (g) (Lopatin, 2023d). Despite the initial higher capital expenditure (CAPEX) associated with DDCC, a thorough evaluation of the TCO over 10 years reveals significant economic advantages, particularly evident when including maintenance and overhaul. These cost savings exceed 14% on the TCO, with potential enhancements of up to 18% attainable in scenarios that feature optimized DDCC configurations operating at 60000 rpm and nearly constant full capacity utilization.

Although the 60000-rpm version is based on simulations and interpolations of component tests (motors, VFD) and feedback from the 36000-rpm existing version, the expected gains remain realistic. Tests on the entire compression unit at 60000 rpm are planned for 2024-2025, which will validate the hypotheses used to interpolate the results.

Unlike OFSC, which frequently faces coating degradation problems leading to significant loss in load capacity surpassing the assumptions made in this paper, DDCC with AMBs emerge as the preferred choice due to their reliability, efficiency, and absence of degradation concerns, resulting in virtually maintenance-free operation. The levitated rotor design of DDCC with AMBs eliminates wear and lubrication requirements, ensuring long-term operational reliability.

The chosen scenarios, where flow demand ranging from 60% to full capacity, highlight the suitability of high-speed centrifugal compressors with AMBs. This underscores the importance of tailoring air compressor unit selection to match specific operating conditions and demand patterns. Oversizing the compressor unit for daily and annual needs, especially in scenarios with episodic peak demand, may diminish advantages of the DDCC, particularly if the compressor frequently operates under conditions requiring blow-off. Therefore, a comprehensive assessment of operating conditions, demand patterns, and compressor capabilities is essential to optimize performance and maximize cost-effectiveness in air compression systems, particularly when using multiple-air compressors.

Heat recovery is not addressed in this paper but holds significant importance. Its implementation requires additional installations to capture heat at the required temperature, which does not directly impact the intrinsic performance of the ACUs.

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