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Energy saving potential analysis of a short cycling industrial air compressor in a marine equipment manufacturing plant in Türkiye

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ABSTRACT

Compressed air systems are recognised as significant energy users and are characterised by their notably inefficient energy consumption. This ensures their significance and potential for decarbonisation through cleaner and more responsible energy consumption in manufacturing facilities, such as marine equipment manufacturing plants in the shipbuilding industry, in order to address the various economic and regulatory challenges related to energy use and climate change. In the relevant literature, there are numerous studies on various energy-saving measures for compressed air systems; however, none concentrates on the problem of short-cycling phenomena and associated energy-saving potential. In this study, using a novel and systematic energy audit methodology, a detailed energy audit of a rotary-type screw air compressor was conducted at a marine equipment manufacturing plant in Türkiye. The systematic energy audit methodology was based on the measurement of power consumption and the evaluation of various operation parameters to assess the existing performance of the compressor, including compressed air demand, compressed air production, cycle speed, air tank volume, specific capacity, and duty cycle. The audit results revealed that the air compressor was short cycling, resulting in excessive energy consumption. Comprehensive technical and economic assessments were conducted to determine the root cause of the compressor's short cycling and to identify energy-saving potentials. It was determined that the compressor was oversized relative to the plant's compressed air demand patterns, while the air tank was inadequately sized, causing the compressor to engage in short cycling. To replace the existing short-cycling compressor, a scenario analysis revealed that the deployment of an optimised system consisting of a fixed-speed baseload compressor and a variable-speed trim compressor can reduce the plant's energy consumption for the compressed air system by a significant 73%. This results in annual energy savings of 74,160 kWh, annual cost savings of €9.373,8, and an annual reduction of approximately 49,9 tonnes of carbon emissions. This application requires an initial investment of €24.280 and is anticipated to redeem itself in 2,2 years. Moreover, it is anticipated to generate a net present value of €147.602 over its 20-year lifespan.

1. Introduction

Compressed air (CA) is one of the most prevalent forms of energy utilised in industrial settings due to the advantages it provides, such as cleanliness, practicality, and ease of use (Nehler, 2018). Contrary to the common misconception that it is a free resource (Cabello Eras et al., 2020; Dindorf, 2012), CA production is one of the most energy-intensive processes, with the energy use of a compressed air system (CAS) accounting for nearly 80 percent of the total cost from a life-cycle cost perspective (Nehler, 2018). Also, it is reported that a well-designed CAS system has very poor efficiency, and CA is one of the most expensive and inefficient utilities, accounting for as much as 30% of the manufacturing electric cost (Tempiam et al., 2020). Nevertheless, it is widely acknowledged that the energy efficiency of CASs can be enhanced, as there appears to be untapped potential (Nehler, 2018). For example, McKane and Hasanbeigi (2011) reported a 56 percent energy savings potential for existing CASs with the implementation of low-cost energy efficiency measures. Moreover, it is emphasized that an optimised CAS is 66% more energy efficient than a conventional system (Marshall, 2012; Nehler, 2018).

Considering the above facts, CASs can play a key role in global endeavours aimed at net zero emissions targets through reducing energy consumption and curbing greenhouse gas emissions (Benedetti et al., 2016, 2018; Introna et al., 2014). As political and global concerns about

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climate change and industrial energy use grow, so does the need for energy management tools, such as energy audit methodologies, to assist industries in adapting to climate change by measuring, analysing, and improving the energy efficiency of their energy-consuming systems (Uyan et al., 2023) through cleaner and more responsible consumption. Given their substantial energy consumption and classification as typical Significant Energy Users (Wu et al., 2021), CASs should be considered in the development and implementation of these tools and energy efficiency actions in industry (Benedetti et al., 2018). Therefore, appropriate methods and techniques for assessing the existing performance of CASs and identifying energy-saving potentials must be developed.

As a matter of fact, the author of this paper conducted a detailed energy audit of all energy-consuming systems in an energy-intensive marine equipment manufacturing plant (MEMP) in Türkiye. In this paper, an analysis of the CAS of the audited plant was presented; other energy-using systems are the subject of other papers; for example, lighting systems were studied in Uyan et al. (2023). The rationale behind the emphasis on a MEMP originates from the aim to promote sustainable and low-carbon manufacturing practices in the shipbuilding industry's MEMPs by increasing energy efficiency and the use of renewable energy resources. Current energy use and climate change concerns in the marine industry are focused on the design and operation phases of a ship's life cycle through regulations such as the Energy Efficiency Design Index (EEDI) and the Carbon Intensity Indicator (CII), while the manufacturing phase is neglected (Vakili et al., 2021). Parallel to this, current research and development in the MEMPs has centred on the development and production of more environmentally friendly and sustainable marine equipment and machinery (EC and ECOFYS Netherlands B.V., 2015). However, manufacturing activities in MEMPs involve energy-intensive and environmentally polluting processes and, as such, deserve equal consideration with regard to energy consumption and climate change (Uyan et al., 2023; Vakili et al., 2021). There are also various economic and regulatory reasons for the MEMPs to pay attention to their energy use and environmental impacts in their manufacturing operations. For instance, while the Turkish MEMPs are already affected by the recent economic volatility in the country, rising and volatile energy prices worsen the uncertainty and diminish their competitiveness in the global shipbuilding market, which is one of the most competitive markets (Stopford, 2009). In addition, based on their energy-intensive nature, the MEMPs can be targeted by the Nationally Determined Contribution (NDC) of the Turkish government under the Paris Agreement. The recent EU Carbon Border Adjustment Mechanism (CBAM), which establishes a carbon price for carbon-intensive products imported into the EU, should also be considered by the Turkish MEMPs. During its transitional period, the CBAM will encompass energy- and carbon-intensive sectors such as cement, iron, and steel, but it is anticipated that other sectors will be included in the near future (Simões, 2023). Considering that European firms hold the largest share of the export market of the Turkish marine equipment industry (OECD, 2023), the Turkish marine equipment manufacturing companies may face challenges resulting from the EU's CBAM in the near future, and if they do not take the necessary actions, they may lose their leadership position to marine equipment suppliers that perform low-carbon manufacturing. Keeping these facts in mind, the MEMPs should adopt cleaner, more responsible, and low-carbon manufacturing practises and review their energy consumption and carbon footprint reduction strategies.

During the energy audit at the case study MEMP, although no compressed air-related production interruptions had previously been reported in the plant, it was discovered that the compressor was operating very inefficiently by conducting very frequent short cycles. The compressor was a rotary screw compressor, which is a commonly used compressor type in industrial facilities (Rane et al., 2013). It was fixed speed (FS) and equipped with load/unload control. Extensive investigations were conducted to determine the root causes of inefficiency and identify energy-saving potential within the system. Short-cycling (also referred to as over-cycling) in industrial air compressors is a significant source of inefficiency, leading to a substantial increase in energy consumption ranging from 20% to 50%, as well as mechanical wear and system failures (Abels and Kissock, 2011; McIntyre, 2017). It is difficult to determine if an in-service compressor is short-cycling without in-depth analysis, as the compressor in a short-cycling condition can continue to supply CA to the system without any interruption. Plant management and technicians consider their compressors to be in good working condition unless a breakdown halts production (Çağman et al., 2022; Kaya et al., 2002). Short-cycling can go undetected for an extended period of time, resulting in significant energy waste and emissions, potential mechanical damage to the compressor (McIntyre, 2017), and associated financial losses. In addition, determining the underlying cause of short-cycling is more difficult than implementing other measures such as replacing components, performing maintenance, or repairing them (McIntyre, 2017). Therefore, air compressors should be an essential part of any energy auditing activities in industrial plants, and their energy efficiency should be evaluated and checked to determine if they are short-cycling or not, even if they work without issue and supply CA to end users without a problem.

Within the existing body of literature, there have been many efforts to improve the efficiency of CAS through the examination of various aspects and the implementation of energy-saving measures. Suppliers, supply associations, and experts also propose energy efficiency measures for CASs through handbooks, manuals, and guidelines (Nehler, 2018). These measures include installing high-efficiency motors, reducing the compressor air inlet temperature, installing variable-speed drives, implementing automatic control, reducing compressed air pressure, repairing air leaks, etc. (Abdelaziz et al., 2011; Kaya et al., 2002; Mousavi et al., 2014; Neale and Kamp, 2009; Nehler, 2018; Saidur et al., 2010). While some studies provided a review of these well-known measures, some researchers developed novel measures to save energy within CASs. For instance, Ignjatovic et al. (2012) developed a wireless filter monitoring system for CAS filters, which cause energy losses via pressure drop if they are not regularly replaced or cleaned. Sambandam et al. (2017) proposed using a 46° branch instead of a conventional T branch within the CAS distribution lines as an energy-saving measure based on their CFD analysis, which demonstrated energy is saved through pressure reduction. In another study, Goodarzia et al. (2017) developed a technique to remove moisture from compressor inlet air based on the use of a desiccant wheel that employs compression heat from the compressor's first stage. Eret et al. (2012) proposed a practical approach referred to as "end-use catalogues" to provide detailed information on CA consumption across an industrial site, which involves the evaluation of typical end-use application profiles.

Fixing air leaks within the CASs is often addressed as an energy-saving measure. Abela et al. (2020) conducted an experimental analysis on a CAS test bed to assess the impact of air leakages on energy consumption and costs. Silva et al. (2017) conducted a leak analysis of the air compressor system of a steelmaking plant's blast furnace in order to increase the plant's energy efficiency by identifying and eliminating air leaks in the distribution line. Çağman et al. (2022) proposed a simple set of equations to calculate the mass flow rate of air leakage. Czopek et al. (2022) proposed an acoustic monitoring method to identify leaks with holes larger than an over diameter within a CASs range, based on the relationship between leak diameter and sound emitted by the leak. Zahlan and Asfour (2015) proposed a simulation-optimisation model to determine the optimal location of a single compressor in a facility to minimize the distance CA must travel to reach high demand and high pressure zones, thereby reducing pressure drop, air leaks, and associated energy losses.

Some research has proposed benchmarking methods to identify the energy efficiency potential within CASs. For example, Benedetti et al. (2018) designed a benchmarking system for CASs based on an assessment of the current state of the art of CASs' energy efficiency in industrial sectors in Italy. In another study, Cabello Eras et al. (2020) proposed a six-step local energy benchmarking methodology to assess the energy performance of CASs based on monitoring and controlling the production and use of CA at a plant through the real-time monitoring of relevant variables to calculate energy performance indicators, energy baselines, and CUSUM charts. Benchmarking methods are useful tools to determine how much energy efficiency potential may exist within a system based on a comparison with other systems' performance, which is assumed to be efficient. However, short-cycling and its root causes cannot be identified from benchmarking. Energy audits carried out in existing CAS of industrial facilities have been reported. For instance, Doner and Ciddi (2022) investigated the energy-saving potential of a CA system in an industrial facility through the elimination of air leaks, loaded-unloaded operation, and waste heat recovery. Jovanovic et al. (2014) conducted an energy-saving analysis of a water bottle manufacturing system.

It is observed from the literature that no study has specifically addressed the issue of short cycling in existing air compressors in industrial facilities, despite the fact that it is of critical importance in terms of energy efficiency. To fulfil this critical research gap and contribute to the efforts to improve the energy efficiency of existing industrial CASs, this paper presents the methodology and analyses adopted during the energy auditing of the case study MEMP's air compressor, which was found to be short cycling. It is also aimed at raising awareness of sustainable, cleaner, and more responsible consumption of energy in the MEMPs of Türkiye's shipbuilding industry.

More specifically, a novel methodology approach for energy auditing was adopted based on power consumption measurement and assessments of operation parameters such as CAD, CA production, compressor cycle speed (CS), air storage tank volume, specific capacity (SC), and duty cycle (DC). Following the investigation of the existing performance and exploration of the root causes of short cycling, a scenario analysis was formulated in which multiple design alternatives were proposed. Based on comprehensive technical and economic assessments, an optimised compressor system that operates efficiently and provides energy and energy cost savings was recommended for the plant. Additionally, the reduction in the plant's carbon footprint was addressed as an environmental benefit of energy savings.

The novelty of this study based on the author's best knowledge stems from this being the first energy audit study focusing on a short cycling air compressor in a MEMP in Türkiye. Herein, a structured and tailored methodology is presented, providing significant information regarding how a short-cycling rotary screw-type air compressor is analysed, the investigation of the root cause of the compressor's short-cycling, and the energy-saving potential together with economic and environmental benefits. In addition, the current study is conducted within the context of MEMP in the Turkish shipbuilding industry, a sector that, to the author's knowledge, has been studied scantly. This is a crucial consideration given the urgent need for MEMPs to increase their energy efficiency and decarbonise their processes in order to tackle a variety of regulatory and economic challenges. Therefore, it is anticipated that this research will provide a valuable contribution to the existing body of literature regarding energy efficiency in industrial air compressors, as well as from the standpoint of the marine equipment manufacturing industry.

Section 2 describes the materials and methods utilised in this study, including a description of the case study MEMP and the methodology approach. Step 1 of the energy audit is described in Section 3, including its methodology, application, results, and recommendations for Step 2. Step 2 of the energy audit is described in Section 4, along with its methodology, application, and results, which include scenario analyses, energy-saving potentials, and economic and environmental evaluations. The discussion is presented in Section 5. The conclusions are given in Section 6.

2. Materials and methods

2.1. Description of the case study plant

The manufacturing plant under consideration is an energy intensive MEMP producing various marine equipment such as marine propellers, stern tubes, and rudders, deck machinery, and so on for shipyards and shipowners. The MEMP is located in the Marmara Region of Türkiye, which is the most industrialised region and a major shipbuilding hub of the country. The plant has two air compressors; both are rotary screw compressors, the most common kind of air compressor in industrial facilities (Rane et al., 2013). The technical specifications of the air compressors are presented in Table 1. Compressor 1 has a bigger capacity while Compressor 2 has smaller. The plant uses Compressor 2 in night shift because the plant management assumes that CAD is lower in night shifts so that the smaller compressor can cover the CAD. The type of both compressors is Fixed Speed (FS) rotary screw controlled with load/unload controlling. The CAS system has an air storage tank of 2 m^3. The operating CA pressure range is 6.5–7.5 bar. A simplified diagram of the CAS of the case study plant is given in Fig. 1.

2.2. Overview of the methodology approach

A structured and stepwise methodology was adopted in the energy audit. Power consumption measurements of the air compressor served as the basis for subsequent analyses and evaluations throughout the energy audit. The power demands of compressors were logged for three consecutive production days. The author questioned the very frequent cycling power demand of the air compressor and set out to investigate the reason behind this.

The methodology steps followed in this study are illustrated in Fig. 2. The energy audit methodology is comprised of two major steps. The first step (Step 1) involves the investigation of the compressor's existing performance and the determination of the root cause of short cycling. In this step, the power demand per second data obtained through power measurement is compiled and utilised to conduct analyses and assessments on the operation parameters, including compressor CS, DC, cycle time (CT), CAD, and volume of the air storage tank. In Step 1, it is checked whether the compressor's CS is within the allowed limits for an efficient and safe operation. Thereafter, the compressor's DC is calculated to determine its capacity utilisation to find out whether it is efficiently used or not. Also, the CAD of the plant, which is critical in terms of determining the performance of the existing compressor system as well as making informed decisions regarding the selection of a new compressor system, is established. Considering the existing compressor's capacity and the plant's CAD patterns, the required minimum volume for the air storage tank that would provide an efficient compressor operation is determined. The determined minimum volume is subsequently compared to the existing volume of the air storage tank. Thereafter, based on the results of this step, recommendations are provided to save energy by implementing optimised compressor systems within the audited facility.

Considering the findings and recommendations outlined in Step 1, a

Table 1
Specifications for the compressors in case study plant.

Specifications	Compressor 1	Compressor 2
Type	Fixed Speed Screw	Fixed Speed Screw
Control type	Load/unload + auto shut off	Load/unload + auto shut off
Rated power	55 kW	18 kW
Specific Capacity (SC)	10.153 m^3/min	3.25 m^3/min
Specific power consumption (SPC)	0.1595 m^3kW/min	0.1796 m^3kW/min
Employment	Day shift	Night shift
Air Storage Tank	2 m^3	

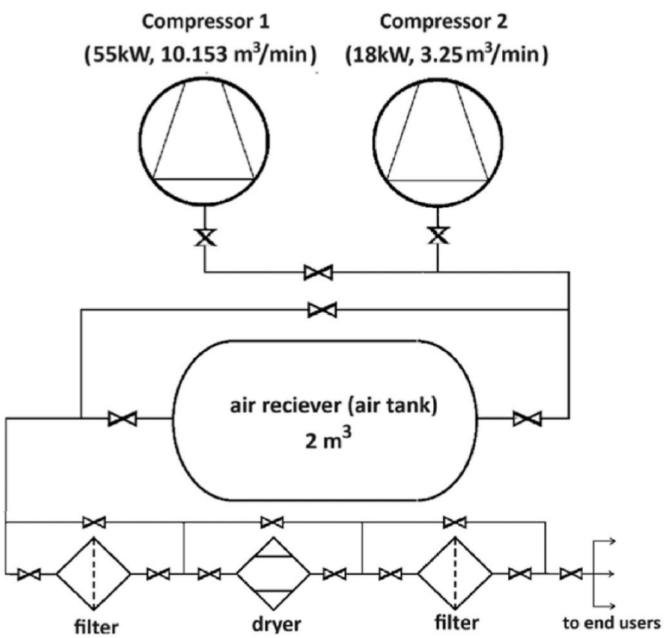

Fig. 1. Illustration of the CAS in the case study plant.

scenario analysis is formulated in Step 2. The scenario analysis entails the proposal of various design alternatives with the objective of replacing the existing short-cycling system and attaining energy savings. In Scenario 1, the design of a single compressor system using an FS compressor is considered. Scenario 2 involves the design of a multiple compressor system consisting of an FS baseload compressor and a VS (variable speed) trim compressor. In each scenario, several alternatives are proposed. A thorough analysis is conducted to evaluate the technical and economic aspects, resulting in the identification of a compressor system as the most suitable choice among various design alternatives. This recommended system demonstrates high operational efficiency through reduced cycle speed and higher capacity utilisation and offers significant energy savings and economic performance for the plant. The environmental benefits thanks to the energy savings are addressed as a decrease in the carbon footprint of the plant.

The methodologies and results for Step 1 and Step 2 are presented in separate sections, namely Section 3 and Section 4, respectively. It is worth noting the performance of the plant's Compressor 1 was determined to be extremely poor and inefficient, characterised by frequent short cycling. In comparison to Compressor 1, the performance of Compressor 2 appeared to be relatively adequate. Consequently, the energy audit methodology detailed in this study was implemented on Compressor 1 and the results were presented accordingly.

3. Investigation of the existing performance, determining the root cause of short cycling, and recommendations (Step 1)

3.1. Methodology

3.1.1. Power consumption measurement

The power demand of Compressor 1 was logged at 1 s intervals by using a power and energy data logger for three consecutive daytime production shifts (from 08:35 to 16:45). One day shift was chosen as a representative to study. The power demand measurement is illustrated in Fig. 3. The secondly power demand throughout the production shift was extracted to the excel spreadsheet to use in the analyses.

3.1.2. Compressor cycle speed (CS)

Cycle speed (CS) is an important parameter that shows the number of cycles of a compressor's motor in an hour. It is calculated as follows (Bierbaum and Hütter, 2004):

$$CS = \frac{60}{average\ CT\ within\ operation\ period} \quad (1/h) \qquad \text{Eq 1}$$

Where CT is the cycle time which is the sum of the length of time (mi-

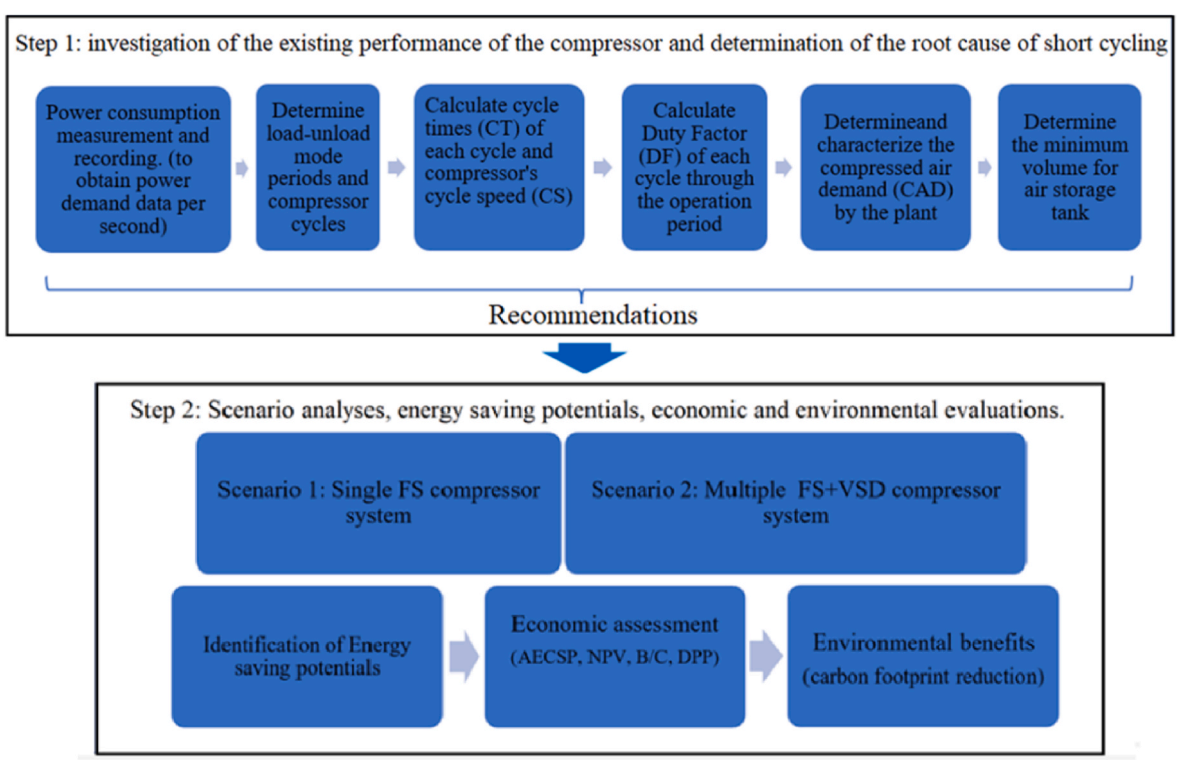

Fig. 2. Overall methodology flow chart.

Fig. 3. Power demand measurement from the case study compressor.

nutes) it takes for the compressor to load and unload (Heney, 2018) whereas operation period is the measurement period (i.e. a production shift in the case study). In other words, CT is the sum of the lengths of load (t_{load}) and unload-mode (t_{unload}) durations in a cycle as expressed as follows:

$$CT = t_{load} + t_{unload} \quad (min) \qquad \text{Eq 2}$$

For a safe and efficient compressor operation, the CS of a compressor should be below certain safety limits specified in Table 2 with regards to the motor power ratings.

The operation of a load/unload-controlled compressor is comprised of a number of consecutive cycles. The average CT within the compressor's operation period can be estimated based on the durations of t_{load} and t_{unload} for each compressor cycle, whereby the CS can be calculated using Equation (1). To identify the compressor cycles and compute the CT for each cycle, the durations for load and unload modes within each cycle were established based on the fact that a compressor with load-unload control draws about 105–115% of its power rating while working in load mode (Schmidt and Kissock, 2005). As illustrated for the 132nd cycle of compressor operation in Fig. 4, the power demand of Compressor 1 in load mode was greater than 60 kW and less than 59 kW in unload mode. Based on this, the time steps (i.e., seconds) during which the compressor's power demand was greater than 60 kW (i.e., load mode power demand) were counted and found to be 12 s and established as the load mode duration, while those that were less than 60 kW were counted to determine the unload duration and found to be 24 s. Thus, the CT was found to be 36 s for the 132nd cycle. This has been performed for each cycle of the compressor's operation in a production shift to find the average CT.

Table 2
Allowed CS for an electric motor depending on the power rating of the motor (Bierbaum and Hütter, 2004).

Motor power rating (kW)	Allowed CS (1/h)
4–7.5	30
11–22	25
30–55	20
65–90	15
110–160	10
200–250	5

3.1.3. Compressed air demand (CAD)

Understanding the CAD of the plant is a crucial factor in determining the existing performance of an air compressor as well as choosing and sizing a new compressor for the facility. The CAD of an existing facility can be estimated by measuring the compressor's CA output using an inline or non-intrusive flow meter. There was no flow measurement device installed on the compressor, and it was not possible during the auditing period as it would require additional cost and time and cause a disruption to production that cannot be tolerated by production managers (Eret et al., 2012). Instead, the plant's CAD can be estimated based on compressed air production (CAP) by the compressor. The CAP of a load/unload-controlled compressor can be calculated based on the lengths of load-mode periods and specific power consumption (SPC) since the compressor produces CA in load-modes. Therefore, the compressed air production (CAP) in the load-mode of a cycle (CAP_{cycle}) by the compressor can be calculated as follows:

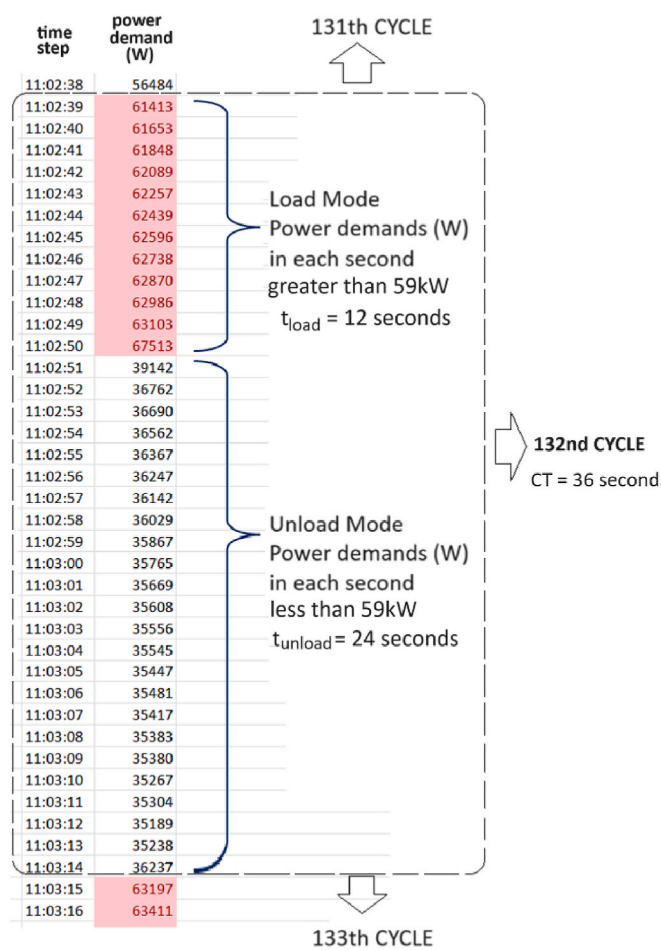

Fig. 4. Identification of compressor cycles and the durations for load and unload modes of each cycles based on the power demands.

$$CAP_{cycle}\ (m^3) = SPC\ *\ P_{average_{load}}\ *\ t_{load} \quad \text{Eq 3}$$

where CAP_{cycle} is the compressed air production in a load-mode cycle, SPC is the specific power consumption of the compressor m^3/min/kW, $P_{average\_load}$ is average power demand (kW) in load-mode of the cycle, and t_{load} is load-mode length (min) in the cycle. The power demands in the load-modes of each cycle were determined through the power measurements. The SPC of a compressor can be obtained from the compressor manual or nameplate.

The total CAP in the entire compressor operation period (i.e., in a production shift in the audit) can be expressed as the sum of each CAP_{cycle} in each load mode of the consecutive compressor cycles:

$$CAP_{total}\ (m^3) = SPC * \sum_{1}^{i} P_{average_{load}i} * t_{load\ i} \quad \text{Eq 4}$$

If the CAP in a load mode period of a cycle is divided by the overall cycle length, the instantaneous CAD by the plant throughout that cycling (i.e., CAD_{cycle}) can be estimated as follows:

$$CAD_{cycle}\ (m^3 / min) = \frac{CAP_{cycle}}{cycle\ length} \quad \text{Eq.5}$$

Similarly, the CAD (m^3/min) throughout the entire compressor operation can be produced by summing the consecutive CAD_{cycle} values.

3.1.4. Duty cycle (DC)

To determine the capacity utilisation of an existing air compressor, it is necessary to calculate the DC. This parameter serves as an indicator of the compressor's capacity utilisation by quantifying the frequency at which the compressor operates in loaded or unloaded (Melissa, 2009). The DC for a cycle can be calculated as follows (Uyan, 2019):

$$DC(\%) = \frac{t_{load}}{t_{load} + t_{unload}}\ (\%) \quad \text{Eq 6}$$

t_{load} and t_{unload} were identified in Section 3.1.2.

The DC for Compressor 1's all cycles throughout the operation period was calculated along with the descriptive statistics (i.e., mean, maximum, minimum, standard deviation, etc.) to determine its capacity utilisation as its operation is comprised of a number of consecutive cycles.

3.1.5. Minimum volume for air storage tank (Vs)

For an effective air compressor operation with a minimum number of compressor cycles, it is crucial that the air tank has the optimal capacity (Boehm and Franke, 2017). In fact, using an adequate storage tank in a CAS is necessary to prevent short-cycling (Beals, 2009). The minimum air tank volume for a CAS can be determined as follows (Agricola et al., 2003):

$$V_S = \frac{SC * 60 * [x - x^2]}{CS_{max} * \Delta P} \quad (m^3) \quad \text{Eq 7}$$

where ΔP is pressure difference (bar), SC is specific capacity of the compressor (m^3/min), and x is utilisation factor, which is calculated as follows (Agricola et al., 2003):

$$x = \frac{CAD_{max} - CAD_{avg}}{SC} \quad \text{Eq 8}$$

where CAD_{max} and CAD_{avg} are maximum CAD (m^3/min) and average CAD (m^3/min), respectively, which were determined in Section 3.1.3.

ΔP is calculated as follows (Agricola et al., 2003):

$$\Delta P = P_U - P_L \quad \text{Eq 9}$$

Where P_U is upper activation pressure (bar) and P_L lower activation pressure (bar). ΔP for the case study plant is 1 bar since Pu and PL are 7.5 bar and 6.5 bar, respectively.

3.2. Application and results

3.2.1. Power consumption measurement

The power consumption of Compressor 1 was recorded for a typical production shift. As examples, Fig. 5 shows the power demand profiles between 09:00–10:00 (a), 12:00–13:00 (b), and 15:00–16:00 (c). As evidenced by Fig. 5, the power demand profile of Compressor 1 is extremely cyclical. Normally, cycling operation is a characteristic of FS screw air compressors with load/unload control systems because they cycle between two modes: load mode, in which compressed air is generated, and unload mode, in which the compressor idles (Abels and Kissock, 2011). However, regarding Compressor 1, its power demand goes up and down extremely frequently between load-mode power demand (i.e., 60–65 kW) and unload-mode power demand (i.e., around 35 kW). Also, unload-modes are too short; as a result, the auto-shutoff mechanism cannot function. Normally, an auto-shutoff mechanism is anticipated to turn off a compressor to cool it down and avoid unnecessary energy consumption if the compressor works in unload-mode for longer than a predetermined amount of time. The highly cyclical operation can also be observed from the compressor's pressure profile logged at 1 s intervals for 20 min, as shown in Fig. 6. The power demand profile and the pressure profile suggested that Compressor 1 operates in a manner known as "short cycling" (Bierbaum and Hütter, 2004), which is undesirable because it strains the compressor's electric motor and leads to excessive cycling in order to meet CAD.

3.2.2. Cycle speed

In total, 455 cycles throughout the operation period (i.e., a typical production day) were identified for Compressor 1. The average, maximum, and minimum CS for Compressor 1 were found to be 57, 92, and 38, respectively, while the allowed CS for the compressor's 55 kW electric motor is 20, as indicated in Table 2. Evidently, Compressor 1's CS also demonstrated that the compressor exhibits extremely rapid short-cycling and therefore operates in an inefficient manner.

3.2.3. Duty cycle

The DCs for Compressor 1's 455 cycles throughout the operation period were calculated. The descriptive statistics for the DC (i.e., mean, maximum, minimum, standard deviation, etc.) were calculated so as to identify the capacity utilisation of the compressor. Table 3 shows the descriptive statistics for the DC. Fig. 7 shows the DC of each cycle of Compressor 1 throughout the entire operation between from 08:35 a.m. to 4:45 p.m. As seen, the DC varies between around 10% and 60%, while the spots on Fig. 7 are very intense between 10% and 20%. The average DC for Compressor 1 in a typical production shift was 18%, which implies that Compressor 1 only uses its 18% capacity to meet the CAD of the plant.

Fig. 8 shows the relationship between the average DC and CS within the 30-min reference intervals. It is evident that an increase in DC results in an increase in CS, implying that the utilisation of the compressor increased in parallel to the increasing CAD. However, Compressor 1 had to work partly loaded with lower DC and perform a series of short cycles to meet the increasing CAD. Normally, the compressor could show the same CAP by performing lower CS and having higher DC.

In addition, Compressor 1 operates in unloaded mode for 82% of the operation time and consumes electricity, although it does not produce useful output. While the auto-shut-off system is anticipated to turn off the compressor during the unload-mode periods to save energy, it cannot function because the lengths of the unload-mode periods are too short due to the short-cycling. The length of load-mode and unload-mode periods in an average cycle were found to be 12 s and 55.3 s, respectively.

3.2.4. Compressed air demand (CAD)

The CAD profile of the case study plant for a daytime production shift is shown in Fig. 9. As seen, the CAD fluctuates significantly over a

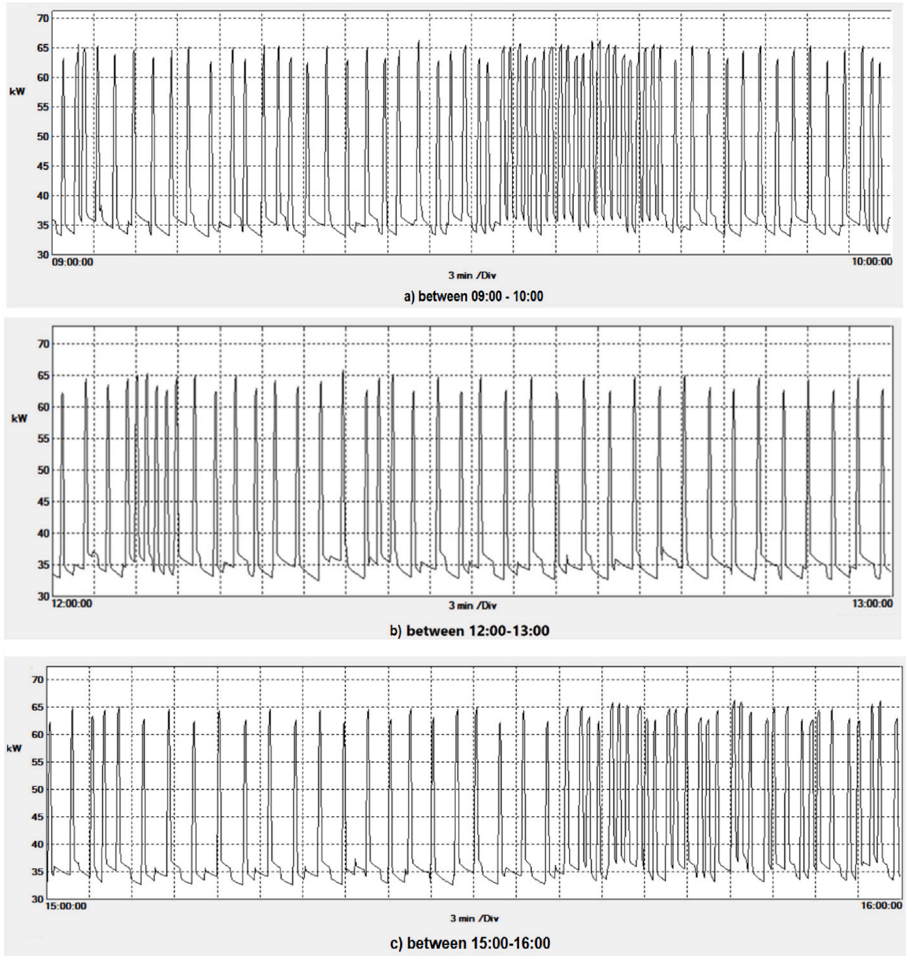

Fig. 5. Compressor 1 power demand profiles: between 09:00–10:00 (a); 12:00–13:00 (b); and 15:00–16:00 (c).

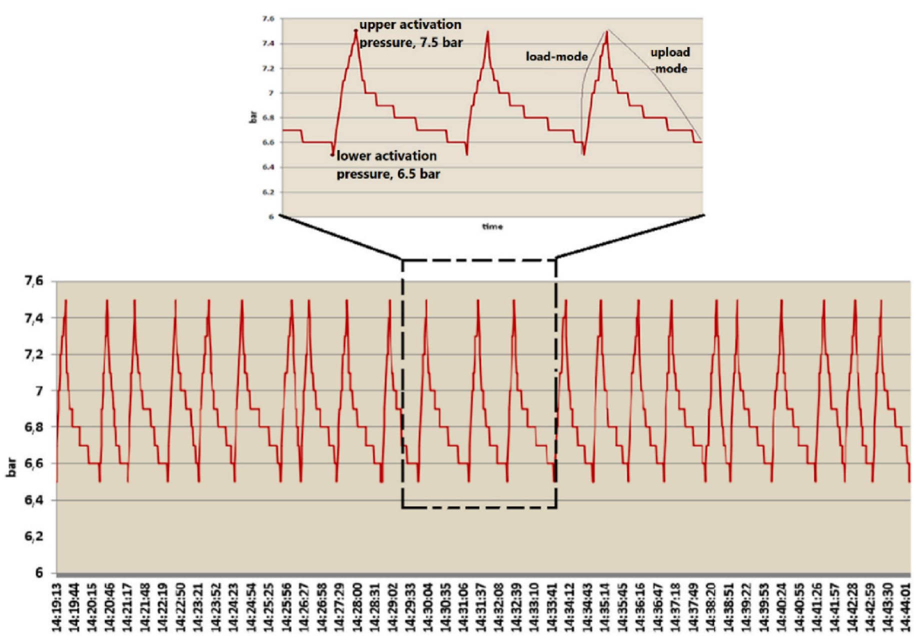

Fig. 6. Pressure profile of the compressor for 20 minutes.

Table 3
Descriptive statistics for DC% of cycles for Compressor 1.

Mean	0.18
Standard Deviation	0.14
Range	0.52
Minimum	0.08
Maximum	0.61
Count (number of cycles)	455

relatively wide range of air flow rates. Table 4 gives the descriptive statistics for the CAD of the plant during the daytime production shift. The maximum, minimum, average, and standard deviation of CAD values are equal to 6.056 m^3/min, 0.735 m^3/min, 2.03 m^3/min, and 1.15 m^3/min, respectively, whereas Compressor 1's SC is 10.153 m^3/min.

3.2.5. Volume for air storage tank (Vs)

The minimum air storage tank volume for the case study plant's

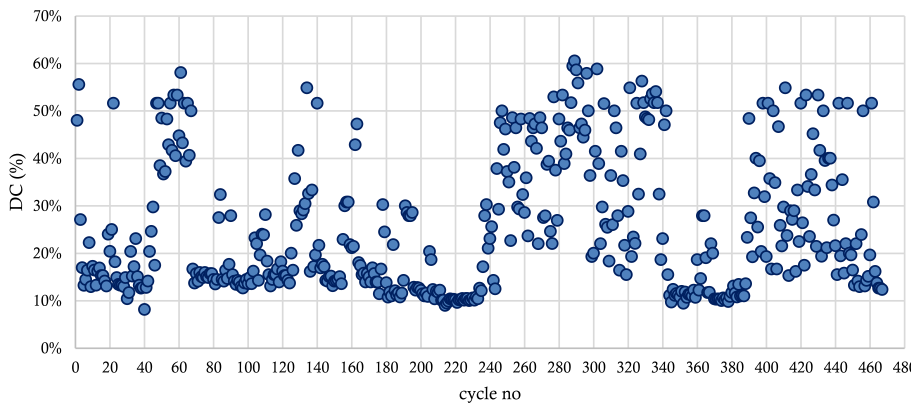

Fig. 7. DC% for Compressor 1's cycles throughout the entire operation period.

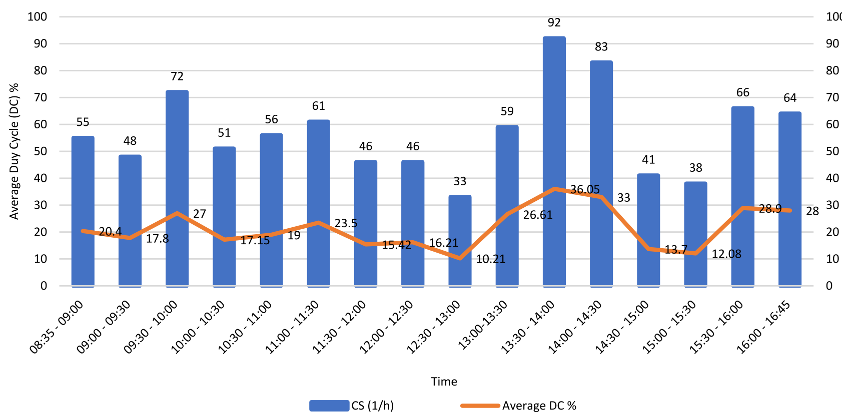

Fig. 8. Change in CS and average DC for compressor 1.

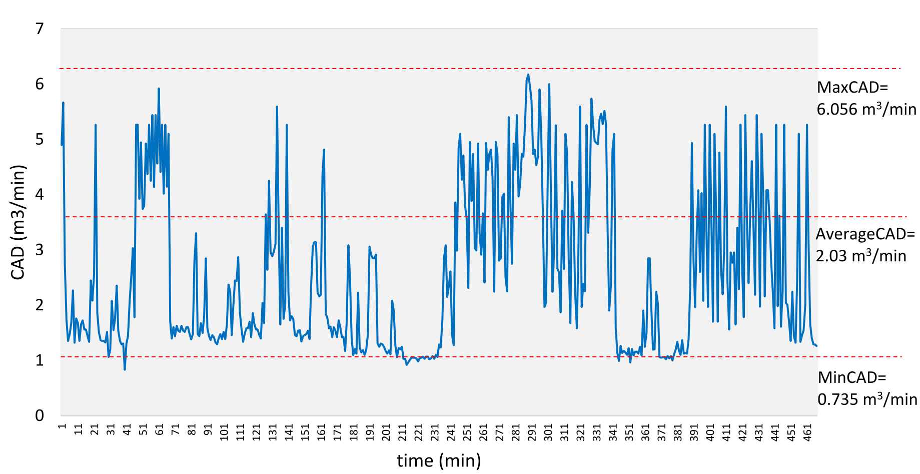

Fig. 9. CAD of the case study plant for a production shift.

Table 4
Descriptive statistics for CAD (m^3/min) of the subject plant in the day-time production shift.

Mean	2.03
Standard Deviation	1.15
Mode	1.06
Range	5.32
Minimum	0.735
Maximum	6.056
Sum	922.7
Count (total duration in minutes)	455

existing Compressor 1 and CAD was calculated to be 7.25 m^3. Thus, it is evident that the existing air storage tank of 2 m^3 is undersized. Similarly, the Vs for other CS values of 16, 12, 8, and 4 for Compressor 1 were found to be 9.06 m^3, 12.08 m^3, 14.49 m^3, 18.12 m^3, and 36.23 m^3, respectively. As seen, there is an inverse proportion between CS and Vs. This implies that very large air storage tanks must be employed to reduce the compressor's CS; however, it is impractical to locate such enormous tanks in the compressor room of the plant in the case study.

3.3. Concluding remarks and recommendations in Step 1

The power consumption measurement and subsequent assessments suggested that Compressor 1 is short cycling. The compressor's CA generation capacity, also known as its SC, was found to be greater compared to the plant's CAD characteristic, which is highly variable. Also, the volume of the existing air storage tank was found to be considerably less than the required minimum volume. Because of these, the DC of Compressor 1 is around 18%, which indicates a low utilisation of its capacity. Further, the amount of CA produced by the compressor rapidly fills the air tank and distribution lines, causing the system pressure to rapidly reach the upper activation pressure, thereby terminating the load-mode period and initiating the unload-mode period very quickly. This results in rapid filling and emptying, which short cycles the compressor, makes it operate inefficiently, and wastes energy.

Using a smaller compressor is an efficient option when the CAD imposed on an existing compressor is far less than the compressor's rated capacity, as is the case for Compressor 1 (Boge, 2016). The DC of Compressor 1 of the case study plant was discovered to be around 18%, indicating lower capacity utilisation. Therefore, a smaller FS compressor with an air storage tank of adequate volume can be a suitable option for the case study plant's CA loads. Moreover, considering the highly variable CAD of the plant with long periods of low CADs that are relatively constant, two compressors can be employed in the case study plant. One can be sized to supply the baseload CA, while the other can be used as a trim compressor to supply the variable CAD greater than the baseload CAD.

A FS compressor is recommended for the baseload CA of the case study plant because it is the most suitable option for CA baseloads with low variability, allowing the compressor to operate at close to full capacity efficiently (Boge, 2016). As for the variable CA load, the VS compressor type is chosen as a trim compressor because VS compressors require less energy for part-load operation conditions compared to FS compressors (Mousavi et al., 2014).

Bearing the above mind, the following options are recommended to save energy in the CAS of the audited plant.

- Using a single FS compressor (Scenario 1)
- Using a FS baseload compressor + a VS trim compressor (Scenario 2)

A scenario analysis was conducted to find the most suitable option for the audited plant and presented in the following section.

4. Scenario analyses, energy saving potentials, economic and environmental evaluations (Step 2)

4.1. Methodology

4.1.1. Scenario 1: using a single FS compressor

In this scenario, the objective was to examine the feasibility of using a single FS compressor with load/unload control to meet the entire CA demand of the audited plant during a daytime production shift. It was investigated if this approach could successfully meet the plant's CAD through a minimum allowed cycle speed for the compressor's electric motor, an adequate volume of air storage tank, higher capacity utilisation of the compressor, and reduced energy consumption and cost compared to Compressor 1's case.

In order to fulfil this objective, eight FS compressors from a vendor (Copco, 2023), each with a different SC that is smaller than that of Compressor 1, were chosen. Also, Compressor 2 of the plant was considered in the analysis to determine if it could supplant Compressor 1 for use during the daytime shifts. The allowed CS for the chosen compressors' electric motors was determined based on Table 2. The minimum Vs for each compressor was calculated using Equation (7) based on SC, CS, CAD_{max}, CAD_{avg}, and ΔP.

The FS compressor's DC in Scenario 1 and Scenario 2 is required to be calculated to determine its capacity utilisation under the case study plant's CAD conditions. Throughout the duration of operation, the DC for a FS compressor might vary depending on the varying CA loads and SC of the compressor. To determine the overall DC of the compressor during the operation duration (a typical production shift), the CAD values of the case study plant, which were determined in Section 3.2.3, were binned into six different groups (i.e., CAD intervals). In each CAD interval, the compressor's DC would vary. Therefore, the overall average DC for the compressor for the entire operation period can be expressed as the sum of the DC multiplied by the CAD frequency in each CAD interval as follows:

$$overall\ average\ DC = \sum_{i=1}^{n} \frac{DC_i\ x\ f_i}{n} \quad \text{Eq 10}$$

Where f is the frequency of CAD and n is the number of cycles. DC in each cycle is calculated using Equation (6).

4.1.2. Scenario 2: FS baseload compressor + VS trim compressor

In this scenario, the objective was to examine the viability of utilising a combination of an FS compressor and a VS compressor. The utilisation of the FS compressor with load/unload control is implemented to provide the baseload CA load for the plant. This enables the compressor to operate at optimal efficiency, operating at or near full load and achieving higher capacity utilisation. On the other hand, a VS compressor is employed to supply variable CA loads that exceed the capacity of the base-load FS compressor's SC. The compressor configuration in this scenario will also require a master control which will be coordinating the operation of baseload and trim compressors (Mehltretter, 2012). Its cost is also included in the economic analysis.

The FS compressors that demonstrated higher DC in Scenario 1 were chosen to be used as a baseload compressor in Scenario 2. Also, the existing Compressor 2 of the case study plant was also considered a baseload FS compressor in this scenario.

The VS trim compressor in Scenario 2 should be sized to handle the maximum CAD to ensure an uninterrupted air supply. Bearing this in mind, a safety factor of 1.2 was applied to the maximum CAD value of the plant to determine the SC for the VS compressor. Thus, the required minimum SC of the VS compressor, $SC_{VS\text{-}Min}$, can be determined as follows:

$$SC_{VS-Min} = 1.2\ x\ \mathrm{CAD}_{MAX} - \mathrm{SC}_{FS_{baseload}} \quad (\mathrm{m}^3/\mathrm{min}) \quad \text{Eq 11}$$

where $SC_{FSbaseload}$ is the specific capacity of the baseload FS compressor. Based on the required SC and the plant's processes' pressure requirements, VS compressors of the rotary screw type were chosen from a vendor (Copco, 2023).

4.1.3. Calculation of energy consumption

The power consumption of a load- or unload-controlled FS screw compressor (E_{FS}) is comprised of two components: load-mode power consumption and unload-mode power consumption. E_{FS} during the operation period (i.e., a production day) can be estimated by summing the energy consumption in each cycle, which consists of energy consumption in load mode and energy consumption in unload mode. This can be expressed as follows:

$$E_{FS} = \sum_{i=1}^{n} (P_{load_i} * t_{load_i} + P_{unload_i} * t_{unload_i}) \quad (kWh/day) \qquad \text{Eq 12}$$

where P_{load} and P_{unload} are the compressor's load-mode power demand (kW) and unload-mode power demand (kW) at each cycle, respectively, and n is the number of cycles in the operation period.

The energy consumption of a VS compressor varies depending on the CAD, as it has multiple part-load-specific SPCs for different CA generation capacities. The energy consumption of a VS compressor (E_{VS}) during the course of its operation can be calculated by multiplying its SPC by the CAD imposed on the compressor at each time step:

$$E_{VS} = \sum_{i=1}^{t} SPC_i * CAD_i \quad (kWh/day) \qquad \text{Eq 13}$$

where *t* is the total operation period (minutes). A top-up CAD profile to be supplied by the VS trim compressor was generated by subtracting the SC of the base-load compressor from the plant's CAD at each time step.

The total energy consumption, E, in Scenario 2 can be calculated as follows:

$$E = E_{FS} + E_{VS} \quad (kWh/day) \qquad \text{Eq 14}$$

The annual energy consumption (AEC) can be calculated as follows:

$$AEC = E \ x \ d \quad (kWh \,/\, year) \qquad \text{Eq 15}$$

Where d is the number of working days in a year. D is 300 days for the case study plant.

In order to consider the impact of machine ageing on AEC throughout the years, a 0.5% factor was applied to account for machine ageing (Vittorini and Cipollone, 2016), as follows:

$$AEC_i = AEC \ x \ (1+a)^t \quad (kWh \,/\, year) \qquad \text{Eq 16}$$

Where a is the machine ageing factor and AECi is the AEC for t = 0 when there is no aging yet.

4.1.4. Energy saving potentials

Annual energy saving potential (AESP) through implementing the scenarios can be estimated as follows:

$$AESP = AEC_{basecase} - AEC_{new} \quad (kWh \,/\, year) \qquad \text{Eq 17}$$

Where $AEC_{basecase}$ is the energy consumption of the existing air compressor system, and AEC_{new} is the energy consumption when a scenario is implemented.

4.1.5. Economic evaluations

4.1.5.1. Economic evaluation criteria. While an energy-saving intervention would result in cost savings due to a reduction in energy consumption, it would also require additional investment and/or installation costs (Vittorini and Cipollone, 2016). Therefore, the effectiveness of additional investment and/or installation costs to save energy should be studied to see if it is financially viable. In this study, net present value (NPV), Benefit-to-Cost (B/C) methods, and Discounted Payback Period (DPP) were used to conduct economic assessments. NPV is calculated as follows (Uyan et al., 2023):

$$NPV = \sum_{t=0}^{T} \frac{(AECSP, ARCs, SVs, other\ savings) - (ICC, RCs, MC)}{(1+i)^t} \quad (€) \qquad \text{Eq 18}$$

Where AECSP is the annual energy cost-saving potential, ARC is the avoided replacement cost, SV is the salvage value, ICC is the initial capital cost, RC is the replacement cost, MC is the maintenance cost, and i is the real interest rate (%). AECSP is calculated as follows:

$$AECSP = AESP \ x \ eucr \quad \left(\frac{kWh}{year}\right) \quad (€) \qquad \text{Eq 19}$$

Where eucr is the electricity unit cost rate, which is 0.1264 euro for the audited plant. An interest rate of 2%, f, was applied to the electricity cost, as follows (Vittorini and Cipollone, 2016):

$$AECSP_i = AECSP_o \ x \ (1+f)^t \quad (€) \qquad \text{Eq 20}$$

The project life was assumed to be 20 years. The existing Compressor 1 in the case study plant is 8 years old, and assuming a 20-year lifespan (van Elburg and van den Boorn, 2014), it will need to be replaced in the 12th year of the duration of the project. If a new compressor is purchased to save energy, it will not be necessary to replace the existing compressor at the end of its useful life. Thus, the associated ARC for the existing compressor throughout the duration of the project is subtracted from the ICC. In this study, the purchasing costs for compressors, storage tanks, and master controller were obtained through a survey in the Turkish market. Installation cost (IC) is integrated into the purchasing cost by calculating as follows (van Elburg and van den Boorn, 2017):

$$IC = 10 \ x \ \text{input power (kW)} + 800 \ (€) \qquad \text{Eq 21}$$

MC was assumed to be 5% of the purchasing value of the compressors (van Elburg and van den Boorn, 2017). As for SV, it was assumed to be 5% of the ICC through market survey.

The total present value of the benefits (PVB) divided by the total present value of the expenses (PVC) yields the B/C ratio, which is represented as follows (Uyan et al., 2023):

$$\frac{B}{C} = \frac{PVB}{PVC} \qquad \text{Eq 22}$$

DPP indicates the number of years required for the sum of the present values of benefits and costs to equal the initial investment (Eltamaly and Mohamed, 2018). Estimating DPP involves determining the year, y, for which the current NPV becomes equal to zero after the initial investment in the year y = 0, ICC, as follows: (Puertas-Frías et al., 2022):

$$NPV \ (y) = -ICCo + \frac{C \ x \ (1-(1+i)^{-y})}{i} \qquad \text{Eq 23}$$

Where C is annual cash flow.

4.1.6. Environmental benefits

The environmental benefits of the energy-saving potential are quantified as a reduction in the plant's carbon footprint annually. In this study, the reduction in the plant's carbon footprint was expressed as annual CO_2-equivalent emissions reduction potential (CO_2e-AERP) and calculated as follows (Uyan et al., 2023):

$$CO_2\text{e-AERP} = \text{AESP x } CO_2\text{eEF}\left(\text{kg-}CO_{2eq}/\text{year}\right) \qquad \text{Eq 24}$$

Where CO_2eEF is the CO_2e emission factor (EF) of the electricity used in the plant. The CO_2eEF for the audited plant is 0.499 kg-CO_2eq/year (Scarlat et al., 2022).

4.2. Results

4.2.1. Scenario 1: single FS compressor

The new compressors chosen to use as a single FS compressor in this scenario have power ratings of 55 kW, 45 kW, 37 kW, 30 kW, 22 kW, 18 kW, 15 kW, and 11 kW. Their respective SC values are 8.9 m^3/min, 6.9 m^3/min, 5.8 m^3/min, 4.7 m^3/min, 3.6 m^3/min, 2.9 m^3/min, 2.3 m^3/min, and 1.77 m^3/min. The allowed CS for the chosen compressors' electric motors was determined based on Table 2.

The SC, the maximum allowed CS, and the minimum Vs are presented in Table 5. Each FS compressor and the associated Vs in this scenario are considered a FS compressor configuration as presented in Table 5. For example, the minimum Vs in configuration S1_C2 (Configuration 2 in Scenario 1) for a compressor with SC = 8.9 m^3/min and CSmax = 20/h was found to be 6.35 m^3.

To determine the overall DC of the compressors during the operation duration, the CAD values of the case study plant were binned into 6 different groups (i.e., CAD intervals), and the average value and frequency of each bin were identified as illustrated in Fig. 10. As seen in Fig. 10, the majority of the CAD is cumulated at around 1.23 m^3/min. In each CAD interval, the compressors' DC will vary. The overall average DC for each compressor was determined and presented in Fig. 11.

If S1_C2 is employed for the case study plant, the 8.9 m^3/min compressor's DC is 22.78%. For S1_C3, the 6.9 m^3/min-compressor's DC is 29.34%. Although their DCs are greater than those of the existing Compressor 1, they are still insufficient for efficient operation because the compressors will work partially loaded for most of their operation. The reason behind this is the fact that the plant has a very varying CAD profile, with a distribution cumulated around 1.4–1.7 m^3/min, whereas the compressors' SC are 6.9 m^3/min and 8.9 m^3/min, respectively.

The configurations S1_C1, S1_C4, S1_C5, S1_C6, S1_C7, S1_C8, and S1_C9 were found to be infeasible for the CADs greater than the compressors' SCs. In those periods where the plant's CAD is greater than the compressors' SC, these compressors are undersized, and the plant's CAD cannot be met. The system pressure would drop below the lower activation point in these scenarios. To meet the CAD, either the compressors must over-cycle, which is undesirable and restricted by integrating CS_{max} in Equation (7), or very large air tanks must be employed, which the plant's compressor room cannot accommodate.

Bearing the above in mind, using a single FS compressor was infeasible for the case study plant. However, as demonstrated in Fig. 11, these configurations' overall average DC for the feasible CAD intervals are 49.34%, 34.66%, 40.3%, 46.5%, 52.5%, 60.94%, and 74.06%, respectively. Particularly, the compressors in S1_C7, S1_C8, and S1_C9 have higher DC than the others as they are exposed to CADs very close to their SC. Consequently, they will run closer to full load, resulting in higher capacity utilisation and substantial energy savings compared to the configurations with lower DC. These results suggest that an additional compressor can be used as a trim compressor for varying CA loads

Table 5
FS Compressor configurations, compressor power ratings, SC, CSmax, and minimum Vs values.

Configuration No	Compressor Power Rating (kW)	SC (m^3/min)	CSmax (1/h)	Vs (m^3)
Base case[a]	55	10.135	20	2
S1_C1[b]	18	3.25	25	2.3
S1_C2	55	8.9	20	6.35
S1_C3	45	6.9	20	4.92
S1_C4	37	5.8	20	4.14
S1_C5	30	4.7	20	3.35
S1_C6	22	3.6	25	2.57
S1_C7	18	2.9	25	1.66
S1_C8	15	2.3	25	1.31
S1_C9	11	1.77	25	1.01

[a] Compressor 1.
[b] Compressor 2.

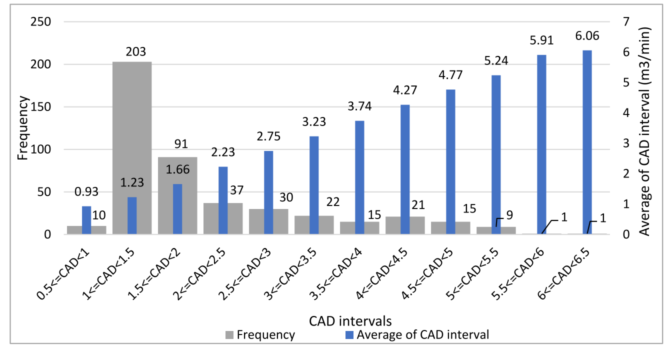

Fig. 10. Distribution of CADs binned into different CAD interval groups and averages.

greater than these compressors' SCs, which is considered in Scenario 2 in the following subsection.

4.2.2. Scenario 2: FS baseload compressor + VS trim compressor

The FS compressors in the configurations S1_C7, S1_C8, and S1_C9 demonstrated higher DCs in the feasible CAD intervals in Scenario 1, as seen in Fig. 11. Their SCs are 1.77 m^3/min, 2.3 m^3/min, and 2.9 m^3/min. One of these FS compressors can be used as a base-load compressor. Therefore, they have been considered an FS baseload compressor in Scenario 2.

Using Equation (11), the minimum SCs necessary for the VS compressors to operate with the baseload FS compressors of 1.77 m^3/min, 2.3 m^3/min, and 2.9 m^3/min were found to be 5.49 m^3/min, 4.96 m^3/min, and 4.36 m^3/min, respectively, as also demonstrated in Fig. 12. Based on these values and the plant's processes' pressure requirements, three VS compressors of the rotary screw type were chosen from a vendor (Copco, 2023). Their nominal power ratings are 30 kW, 25 kW, and 22 kW, whereas their nominal SCs are 5.84 m^3/min, 5.11 m^3/min, and 4.46 m^3/min, respectively. Their part-load SPCs for various part loads are provided in Table 6 based on the vendor's data. Each FS baseload compressor and VS trim compressor were considered a compressor configuration, as presented in Table 5. For example, S2_C1 refers to the configuration in Scenario 2, which includes 11 kW FS and 30 kW VS compressors (see Table 7).

In this scenario, the baseload FS compressor will be running and supply CA as long as the plant's CAD is less than its SC. When the baseload compressor cannot meet the CAD, the system pressure will drop to a setting point, immediately activating the VS compressor to supply CA, and the CAD of the case study plant will be supplied with no shortage.

All configurations in Scenario 2 are considered technically feasible and capable of meeting the CAD of the plant, as each configuration has been sized accordingly. In order to determine the configuration that offers the greatest potential for energy savings and is the most economically appealing, additional analyses were carried out in the subsequent section.

4.2.3. Energy saving potentials, economic assessments, and environmental benefits

The energy consumption values for FS baseload compressors were calculated using Equation (12). The energy consumptions of the VS trim compressors were determined using Equation (13), summing their part-load power consumption calculated for each time step (i.e., minute) based on the top-up CAD values at each time step and the corresponding part-load SPC of the VS compressors. The part-load SPC values were obtained directly or through linear interpolation from Table 6. The overall energy consumption of each configuration in Scenario 2 was calculated and presented in Table 8 together with the associated AESP, AECSP, and CO_2e-AERP. The results for economic assessments are

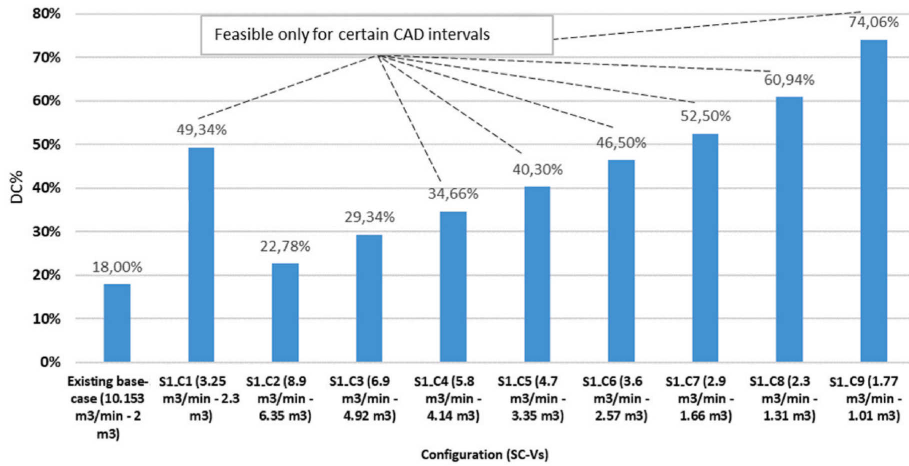

Fig. 11. The average DC for the SC-Vs configurations throughout the entire operation period.

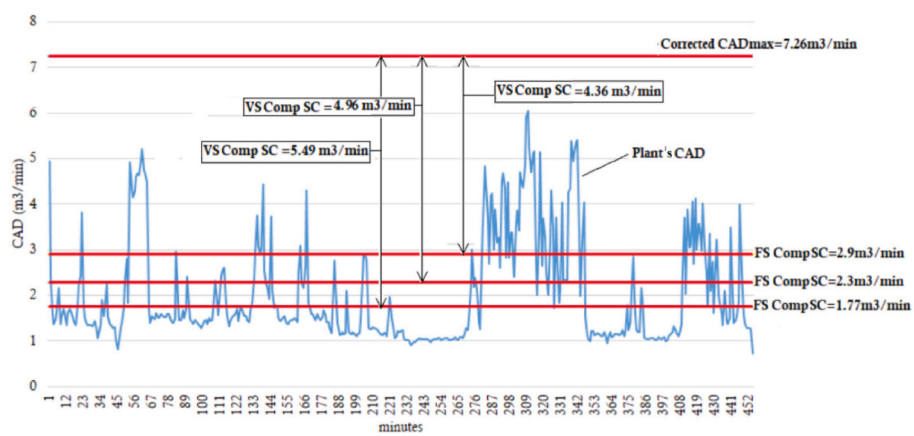

Fig. 12. Demonstration of case study plant's CAD, SCs of the FS base-load compressor and required minimum SCs for VS trim compressors.

Table 6
SPC of chosen VS compressors at part-loads.

30kW-5.84 m^3/min		26kW-5.11 m^3/min		22kW-4.46 m^3/min	
Part-load (m^3/min)	SPC (kW. min/m3)	Part-load (m^3/min)	SPC (kW. min/m3)	Part-load (m^3/min)	SPC (kW. min/m3)
5.84	6.7	5.11	6.75	4.46	6.39
5.16	6.56	4.73	6.64	3.92	6.22
4.36	6.26	3.90	6.59	3.62	5.50
2.71	6.19	2.68	6.64	1.87	6.36
1.06	7.35	1.02	8.43	1.05	7.05
0.90	7.77	0.87	9.08	0.89	7.42

Table 7
Configurations in Scenario 2.

	S2_C1	S2_C2	S2_C3
FS base-load compressor (Power rating – SC)	11 kW - 1.77 m^3/min	22 kW - 2.3 m^3/min	26 kW - 2.9 m^3/min
VS trim compressor (Power rating – SC)	30 kW- 5.84 m^3/min	26 kW - 5.11 m^3/min	22 kW - 4.46 m^3/min

presented in Table 9.

In the base case scenario where Compressor 1 is utilised, the annual energy consumption amounts to 99,946 kWh, resulting in an annual energy consumption cost of €12.633. Additionally, this leads to the release of 49,973 kg-CO_2eq. in indirect emissions per year.

The implementation of S2_C1 in Scenario 2, comprised of an 11 kW FS compressor and a 30 kW VS compressor, results in an AESP of 74,160 kWh, representing 73% of the energy consumption realised in the base case. The calculated value for the equivalent annual energy consumption of AECSP is 9.373,8, whereas the CO_2e-AERP amounts to 37,006 kg-CO_2eq.

S2_C2 in Scenario 2, which consists of a 22 kW FS compressor and a 26 kW VS compressor, in place of the existing Compressor 1, yields an AESP of 61,134 kWh, resulting in a 60% reduction in annual energy consumption, along with an AECSP of €7.727,3 and a CO_2e-AERP of 35,506 kg-CO_2eq.

S2_C3 in Scenario 2 necessitates the use of a 26 kW FS compressor and a 22 kW VS compressor and yields 66,681 kWh of AESP along with AECSP and CO_2e-AERP values of €8.571.4 and 33,838 kg-CO_2-eq, respectively.

Based on the economic assessments conducted, it has been determined that all design options within Scenario 2 exhibit strong economic viability. These options are projected to yield positive NPVs throughout the entire 20-year duration of the project.

The implementation of S2_C1 requires an ICC of €27.210. The economic analysis shows that the investment to apply this configuration returns an NPV of €177.217 over the course of a 20-year project life. The B/C ratio is 4.5, and the DPP is 2.3 years. The configuration S2_C2 necessitates an ICC of € 24.280 and returns an NPV of €147.602. The DPP for the investment in this configuration is 2.6 years, whereas the B/C is 4.3. As for S2_C3, the ICC to implement this configuration is €24.280, and the NPV it returns during the project period is €166.366. The DPP is

Table 8
Energy consumption in each configuration in Scenario 2 and ESP, AESP, AESCP, and CO_2-AERP.

		E_{FS} (kWh)	E_{VS} (kWh)	E_{TOTAL} (kWh)	ESP (kWh)	ESP %	AESP (kWh)	AECSP (€)	CO_2-AERP (kgCO_{2eq}
Basecase		338,8	–	338,8	–				
Scenario 2	S2_C1	65,27	26,33	91,6	247,2	73%	74,160	9.373,8	37,006
	S2_C2	114,02	21	135,02	203,78	60%	61,134	7.727,3	30,506
	S2_C3	101,46	11,3	112,76	226,04	66.7%	67,812	8.571,4	33,838

Table 9
Economic performance results for Scenario 2.

	ICC (€)	AESP (kWh)	AECSP (€)	NPV (€)	B/C	DPP	Economic Feasibility
S2_C1	27.210	74,160	9.373,8	177.217	4.3	2.3 years	Yes
S2_C2	24.280	61,134	7.727,3	147.602	4.2	2.6 years	Yes
S2_C3	24.280	67,812	8.571.4	166.366	4.6	2.25 years	Yes

2.2 years, while the B/C is 4.8.

5. Discussion

As the results of the systematic energy audit conducted on Compressor 1 demonstrate, the compressor was short cycling, and there is substantial potential to save energy through the replacement of the compressor with an optimised system.

In the first step of the audit, Compressor 1's capacity utilisation was determined to be approximately 18%, indicating a significantly low level. During approximately 82% of its operational duration, the compressor functioned in an unloaded state, resulting in the absence of any productive output while consuming electrical energy. Typically, the compressor's automatic shutdown mechanism should activate in order to save energy when the compressor remains in unload mode for an extended duration. However, the average duration of load-mode and unload-mode periods throughout the operation period was determined to be 12 s and 55 s, respectively. These durations are notably short, indicating that the controller lacks sufficient time to deactivate the compressor and achieve energy conservation.

The CAD of the audited plant is highly variable, exhibiting a range of 6.056 m^3/min (maximum) to 0.735 m^3/min (minimum), with an average of 2.03 m^3/min and a standard deviation of 1.15 m^3/min, while Compressor 1's SC is 10.153 m^3/min. It is evident that Compressor 1 was oversized. Furthermore, the volume of the existing air storage tank (2 m^3) was found to be very small in terms of the characteristics of the plant's CAD and the compressor's SC. Because of these, the air tank and distribution lines are filled and emptied very rapidly, thereby causing the system pressure to continuously fluctuate between the upper and lower activation pressure points (as depicted in Fig. 6). Consequently, the compressor experiences overcycling, leading to a highly inefficient operational state characterised by increased energy consumption, costs, and emissions.

It is evident from the results that the inappropriate sizing of the system, characterised by an oversized compressor and an undersized air tank, is the underlying cause for the compressor's short-cycling operation. The selection of Compressor 1 and the required volume of the air storage tank appear to have been made without adequate consideration of the compressed air consumption characteristics of the plant. This resulted in a very inefficient compressor operation, resulting in excessive energy consumption.

Based on the recommendations obtained from Step 1, two primary scenarios were examined in Step 2 to explore the possibility of replacing the current compressor system with an optimised alternative to achieve energy savings. In Scenario 1, the feasibility of utilising a single FS compressor system was examined, whereas in Scenario 2, the feasibility of employing a multiple compressor system consisting of an FS compressor and a VS trim compressor was assessed. It was found that using a single FS compressor with an air tank of appropriate volume was not a feasible option for the case study plant's CAD. This is because the FS compressors either operate with a very low DC or they cannot meet the plant's CAD greater than their capacity. Alternatively, the proposed FS + VS compressor system configurations in Scenario 2 (i.e., S2_C1, S2_C2, and S2_C3) were found to be technically feasible as they can meet the CAD of the plant and operate with a higher DC, indicating a higher capacity utilisation. What is more, it was found that the annual energy saving potential through the implementation of the configurations in Scenario 2 to substantiate the existing short cycling compressor system varies from 60 to 73%, which is significant.

As the results of comprehensive economic assessments in Step 2 demonstrate, it is evident that all the proposed configurations in Scenario 2 have a highly favourable return on investment. Their DPPs are relatively short, further highlighting the efficiency and effectiveness of the investments in these configurations. In terms of energy consumption and costs, it is worth noting that the S2_C1 configuration offers the most promising potential. The investment in this configuration returns the highest NPV (i.e., €177.217) for a 20-year project life, while the cost of electricity consumption and compressor replacement for the existing Compressor 1 within this period would amount to €370.730. Through implementing the S2_C1 configuration, comprised of an 11 kW FS compressor and a 30 kW VS compressor, the case study plant can significantly reduce its annual energy consumption for its CAS by an impressive 73%. A reduction in energy usage will not only lead to substantial cost savings but also contribute to a substantial decrease in indirect carbon emissions, amounting to approximately 37 tonnes per year. The 11 kW FS compressor in this configuration will operate at 74.06% DC, demonstrating not only high capacity utilisation but also highlighting the efficient compressor operation.

It is important to note that the initial investment cost for the S2_C1 configuration is slightly higher compared to the S2_C2 and S2_C3 configurations. Both the S2_C2 and S2_C3 options come with equal investment costs. The investment in S2_C3 is highly appealing thanks to its ability to offer greater energy efficiency and cost savings compared to S2_C2. Additionally, S2_C3 boasts superior economic advantages, making it a more attractive option for investment. In a scenario in which the case study plant gives priority to lower investment costs with less consideration to greater savings in energy use and carbon emissions, it would be advisable to consider implementing S2_C3 as a replacement for the existing short-cycling Compressor 1.

While the case study MEMP can obtain considerable financial benefits thanks to the energy savings through the materialisation of the proposed retrofittings in Scenario 2, additional monetary benefits can be achieved through carbon credit gains. Despite the fact that there is no carbon market in Türkiye at the moment, the Turkish government has planned to establish a national ETS (emission trading system) in line with the country's 2053 net-zero target (UNFCCC, 2023). In a scenario where the case study MEMP can obtain carbon credit gains from the mitigation of 37 tonnes of CO_2eq. in S2_C1 thanks to the electricity consumption reduction, additional gains of 3145 euros annually can be achieved based on a carbon price of €80 per ton (World Bank, 2023), further increasing the financial feasibility.

The analysis in this study was conducted based on the results of power consumption measurements conducted on a representative production day. For future studies, it is recommended to perform longer measurements in order to obtain a more realistic and accurate characterisation of the plant's CAD. The initial purchasing costs associated with the compressor systems were gathered from the Turkish market, and it is important to note that these costs can differ depending on the country and the specific technology provider. It is highly advisable for other researchers to consider utilising the cost figures specific to the geographical region where the energy audit is being conducted.

The methodology approach employed in this study has the potential for further enhancement by incorporating considerations of inefficiencies on the demand side. By thoroughly examining the inefficiencies stemming from the improper utilisation of CA through various measures, such as fixing air leaks, the plant's CAD can be reduced. This, in turn, will lead to a reduction in the necessary capacity of new compressors, ultimately reducing the initial investment costs and enhancing the profitability of the investments. Also, the impact of the operating pressure band of the compressor over the short cycling operation could be investigated. The CAD of the plant was determined by calculating the CAP based on power demands in load modes and the SPC of the compressor. The methodology can be further improved by determining the plant's CAD by using a flow meter with a data logger.

6. Conclusions

A detailed energy audit of a rotary screw air compressor with a load/unload control system in a MEMP in Türkiye was carried out. A unique and systematic energy audit methodology comprised of two major steps was adopted. In the first step, the existing performance of the compressor was investigated through power consumption measurements, assessments, and evaluation of various operation parameters such as CS, DC, CAD, and volume of air receiver. The air compressor was found to be performing short cycling and operating very inefficiently, although it supplies CA without any interruption. The root cause of the short cycling was investigated, and recommendations were made to save energy through an optimised compressor system. In the second step, a scenario analysis was performed in order to replace the existing compressor with an efficient compressor system that provides financial and environmental benefits.

It was found that the existing compressor system of the case study MEMP was sized inappropriately with no regard to the plant's CAD characteristics. The current FS compressor system exhibits an annual electricity consumption of approximately 99,946 kWh, resulting in associated costs of €12.633 and CO2eq emissions of approximately 49.9 tonnes annually. By implementing a synergistic combination of a FS baseload compressor and a VS trim compressor, a substantial decrease of 73% in these metrics can be achieved. The audited plant would experience an annual energy savings of 74,160 kWh, resulting in a cost savings of €9.373,8. Additionally, this would lead to a considerable reduction of 37 tonnes of CO_2eq. emissions annually, thereby enhancing the plant's performance in both monetary and environmental aspects. To apply this configuration, the plant would need to allocate an initial capital of €24.280, resulting in an NPV of €147.602 over a 20-year project life. It is projected to redeem its costs within just around 2.2 years.

The systematic methodology presented in this study offers valuable insights into the analysis of a short cycling rotary screw-type air compressor with load/unload control. This methodology not only focuses on identifying the root cause of the compressor's short-cycling issue but also explores the potential for energy savings, along with the associated economic feasibility and environmental benefits. Also, the methodology presented in this study can be employed to determine the CA consumption patterns of a plant, a crucial aspect in evaluating the performance of air compressors currently in operation as well as making informed decisions in selecting and sizing a new compressor system.

This study has demonstrated that an air compressor exhibits very poor operational performance and consumes an excessive amount of energy, despite its uninterrupted supply of CA to end users. Based on the results, it is strongly advised that MEMPs and manufacturing plants from other industrial sectors incorporate their CA systems in any energy planning or auditing activities to assess the performance of their compressors, even if the CA end users operate without any failure. Herein, the methodology presented in this study can be beneficial to both industrial and academic practitioners to evaluate the performance of air compressor systems. This study is expected to increase the awareness of sustainable manufacturing principles, such as energy efficiency and cleaner and more responsible consumption, among marine equipment manufacturers of the shipbuilding industry in Türkiye and around the world.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

The data that has been used is confidential.

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Glossary

AEC	annual energy consumption
AECSP	annual energy cost saving potential
AESP	annual energy saving potential
ARC	avoided replacement cost
CA	compressed air
CAD	compressed air demand
CAS	compressed air system
CAP	compressed air production
CFD	computional fluid dynamics
CS	cycle speed
DC	duty cycle
DPP	discountaed payback period
eurc	energy unit cost rate
FS	fixed speed
IC	installation cost
ICC	initial capital cost
NPV	net present value
PV	present value
PVB	present value of benefits
PVC	present vale of costs
RC	replacement cost
MC	maintenance cost
SC	specific capacity
SPC	specific power consumption
SV	salvage value
VS	variable speed

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