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RECIPROCATING AND SCREW AIR COMPRESSORS: A CASE STUDY

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Abstract.
This study focuses on economic results related to the conversion of electric energy to compressed air. It was developed with the purpose of presenting a case study of electrical efficiency and effective flow as the final product of the conversion, without observing the generation losses in the airend, in a system with reciprocating piston air compressors. Subsequent to this analysis, a comparison was made with the same demands and needs of the system being serviced by a screw-type positive displacement air compressor. This is an evaluation of the collection of quantities such as electric current, manometric pressure and volumetric flow of air compressors, in order to present a proposal that presents greater efficiency in the generation of compressed air, that is, generate a certain volume with a certain pressure with the least possible consumption of electricity. It is proposed an mathematical model that allows precise analysis to delineate the limits in the use of reciprocating piston compressors in the generation of compressed air of low pressure (up to 15 Bar) in relation to compressors of screw type air. It has been found that there is a superiority of the screw type compressors from a number of daily working cycles, presenting greater efficiency in the conversion of electric energy to compressed air.

Keywords: *Compressed air systems, efficiency in compressed air, case study screw and piston compressors.*

1. INTRODUCTION

Compressed air is a type of mechanical energy, generated from the work of compression. From the beginning of its use from the nineteenth century until the present time, such equipment has been perfected. There are several types of compressors (Hanlon, 2001) (Brown, 1997). The most commonly used and commercially available ones are screw-type and reciprocating piston positive displacements (Greenough, 2000). In the search for better results and greater efficiency, compressed air has the potential to be against these ideals because it deals with the product of a thermodynamic transformation of a thermal machine that uses a large amount of energy to transform atmospheric air into energy mechanics (Rocha, 2005). In most cases, the electric power is the power supply of the motors that drive the airend.

Currently the use of compressed air is very widespread and accepted by the industries, being considered "fourth utility" at many facilities. In some of these cases, air compressors are the largest consumers of electricity, and can exceed up to 40% of the total costs of an industrial plant. To convert 1 kW to compressed air power, the process can consume between 7 kW to 8 kW of electricity in generation (CAC, 2005). In this way, it is one of the energy areas with the greatest saving potential from controls, care and evaluations of your system as a whole.

In the present case study, an auditing was performed in a compressed air generation matrix with four reciprocating piston air compressors, which operated in alternating cycles and provided a daily consumption of up to 45.2 kW (60 HP) operating conditions and equipment conditions and configurations, the opportunity for improvement with cost reduction and increased efficiency in the generation of compressed air through a screw compressor of 30 kW (40 HP) was found, whose efficiencies and yields present in a theoretical model, superiority mainly due to the drive and control mode and the engine and airend efficiencies. This study aims to indicate the limits between the use of piston and screw compressors, through a mathematical modeling that allows the user to enter the data of the system with the desired conditions and parameters, and to obtain in response the type of equipment more efficient in relation to the cost of acquisition and generation of compressed air.

2. GENERAL OVERVIEW OF THE STUDY

2.1 Applicability of types of compressors

In the action of air compression in the airend, there is a loss of the electric energy transmitted from the motor to the airend through friction, the generation of the heat of compression, leakage of the volume sucked by gaps and passages as well as the initial and final volume ratio from the atmospheric air composition since there is water vapor and suspended solid particles being drawn in. The association of these in the discharge of the airend is given as an effective flow, which in the absence of condensed moisture is called the FAD flow (free air delivery). In a brief analysis of a 11.3 kW (15 HP) engine piston air compressor, we find a theoretical displacement or flow of 0.02831682 m³/s at 1 MPa pressure, according to the manufacturer (Schulz, 2018). Its constructive format and the type of composite elements in the compression mechanism is indicated by the manufacturer himself the effective flow rate in 30% unless the theoretical flow, that is, treating the same equipment, the effective flow rate is approximately 0.019349827 m³/s. On the other hand, analyzing a screw compressor of 11.3 kW (15 HP), its flow reaches 0.02359735 m³/s in 1 MPa of pressure, being this flow, according to manufacturer (Ingersoll Rand, 2016), the effective flow FAD.

Although there are such differences, the cost of acquisition can be up to 60% higher than the screw compressor in relation to the reciprocating piston, being the viability of the acquisition associated with the use profile or duty cycle as well as the amortization time. Such a study can be carried out from a life cycle analysis, with an average life span of 10 years (Orsi, 2011).

2.2 Methodology

The central point of the study is the performance results of each equipment, under the same conditions, being carried out analyzes of the quantities collected from the equipment and the distribution network of compressed air to define the profile of consumption and its fluctuations during the day. To determine the magnitudes of the compressed air generation matrix, data were collected from a compressed air generation plant with four reciprocating piston compressors connected in parallel in the network, with their characteristics as shown in Tab. 1 and Fig. 1:

Table 1. Compressor data analyzed in the study.

	C1	C2	C3	C4
Pressure ON	0.661 MPa	0.744 MPa	0.771 MPa	0.771 MPa
Pressure OFF	0.944 MPa	0.982 MPa	0.985 MPa	0.999 MPa
Drive and Control Method	ON/OFF	ON/OFF	ON/OFF	ON/OFF
Three Phase Voltage	380V	380V	380V	380V
Frequency	60Hz	60Hz	60Hz	60Hz
Theoretical Flow	0.0284 m³/s	0.0284 m³/s	0.0284 m³/s	0.0284 m³/s
Full Load Power	13,42 kW	13,42 kW	13,42 kW	13,42 kW
Full Load Current (In)	18,81 A	18,15 A	18,43 A	17,26 A
Motor Performance (*Ƞ*)	0,86	0,86	0,86	0,86

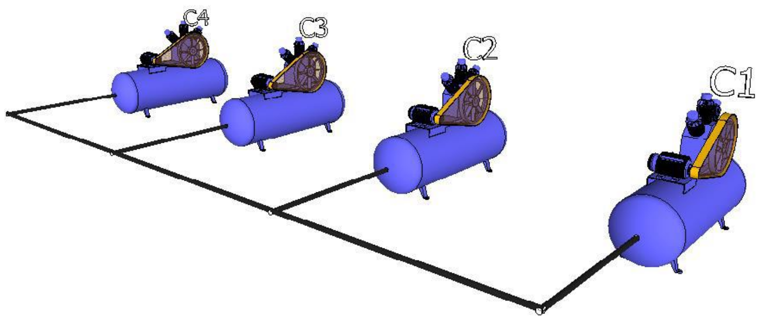

Figure 1. Installation representation compressors.

The equipment was configured for operation in cycles similar to a standard cascade configuration as Fig. 2. However, the starter system did not allow the equipment to be actuated at precisely the load pressure or target pressure. The operating data, electrical current of one of the three phases and the common pressure of the equipment's network, were collected during three days.

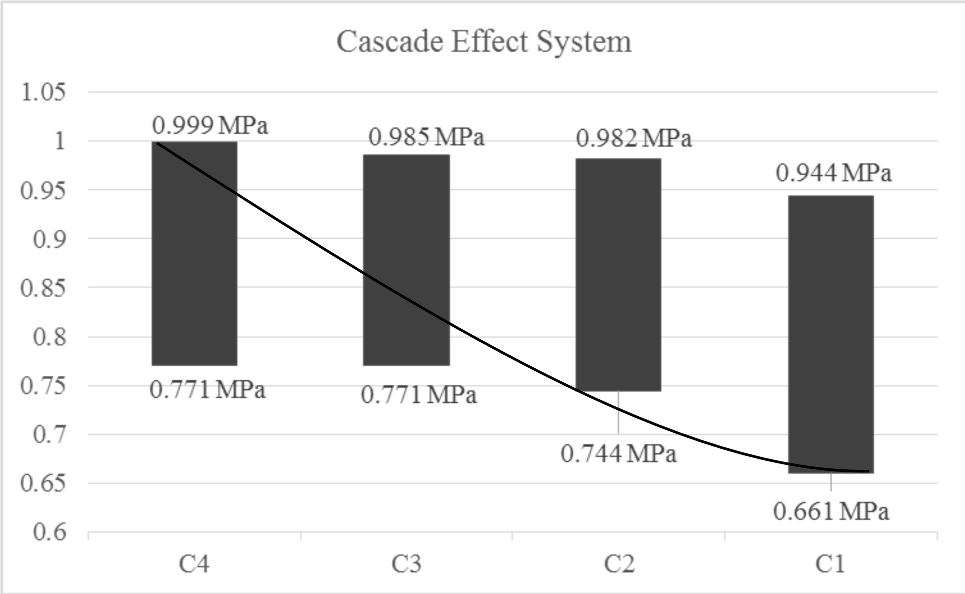

Figure 2. Graphic representative of the pressure drop in cascade effect in the adjustment of compressor pressures.

The evaluation of the system is divided into daily periods for a more precise and accurate point-to-point analysis, after this fractional analysis is made the joining of all cycles and an evaluation of the overall effect of the system and equipment.

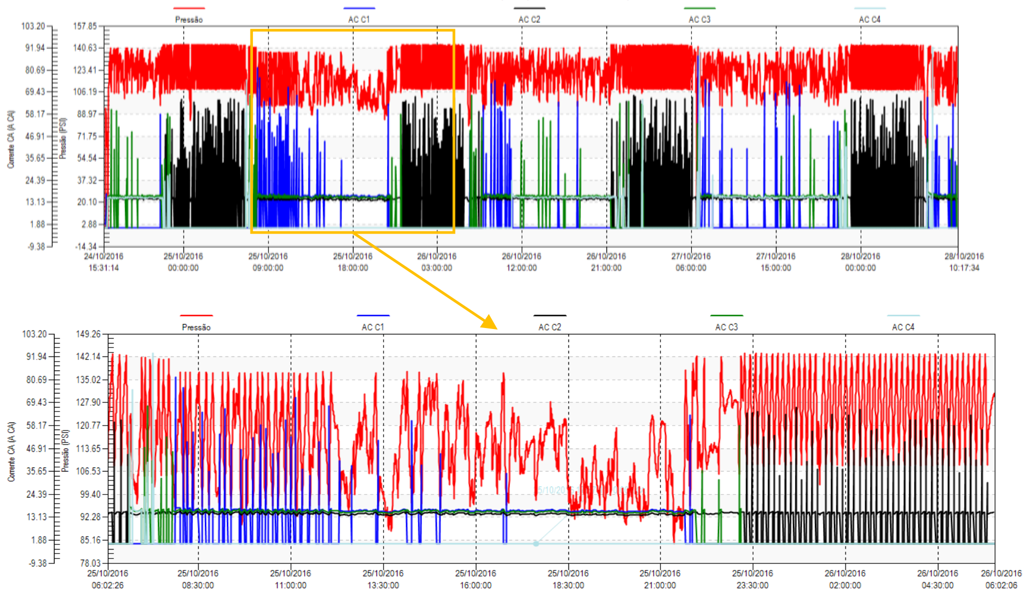

Figure 3. Graph of the data compiled in the total and partial period, presented variation of current and pressure of the network.

The data collection, analysis and verification can allow the visualization of the options and ways to obtain the highest efficiency in relation to the compressed air generation in that analyzed system. Some of the most significant benchmarks for analysis of a system as indicated (Rocha and Monteiro, 2005):

- Maximum Consumption Flow (m³/min)
- Specific Power or Compression Efficiency (kW/m³)
- Specific cost (R$/m³)
- Minimum Working Pressure

3. MATHEMATICAL MODELING USED AND DATA ANALYSIS

The starting method and operating control is one of the first aspects to achieve higher efficiency in air compressor matrices (Mousavi and Bernard, 2014). Three main types are most commonly used: on/off, load/relief (unload) and VSD. Each has its systems and subsystems that allow the execution of the control of the operation of the equipment, however the choice and the form that applies one or more equipment with such methods of control and departure, is that it allows efficiency in the generation of compressed air.

3.1 Start and control ON/OFF

In the ON/OFF method, there are two condition states for the equipment, the control is normally carried out by transducers that start and stop the equipment from the pressure reading made.

Air compressors are equipments that have greater efficiency in the electrical aspect when they are operating in maximum nominal load. Although this is the most efficient and economical method of theoretical control (Dindorf, 2012), in practice there are many aspects that do not allow such a condition to occur: characteristics limitations of the induction motors in the overhead temperature rise (Abou et al. , 1996); low compressed air storage capacity in the distribution network due to an undersized system or the high presence of leaks. Although these devices almost always have motors with high insulation ratings, their use in ON/OFF cycles in short service regimes with constant stops and starts that does not allow the electric motor to achieve thermal equilibrium can cause the engine to burn in the absence of the installation of suitable thermistors, but also results in low efficiency since high temperature reduces efficiency.

This control method is applied in systems and equipment of small air compressors, from 0.746 kW up to 22 kW, of which there is a small variety of screw type compressors that operate in such a cycle, whereas in the reciprocal piston the situation is the majority for this operating profile.

This cycle can be observed in Fig. 4:

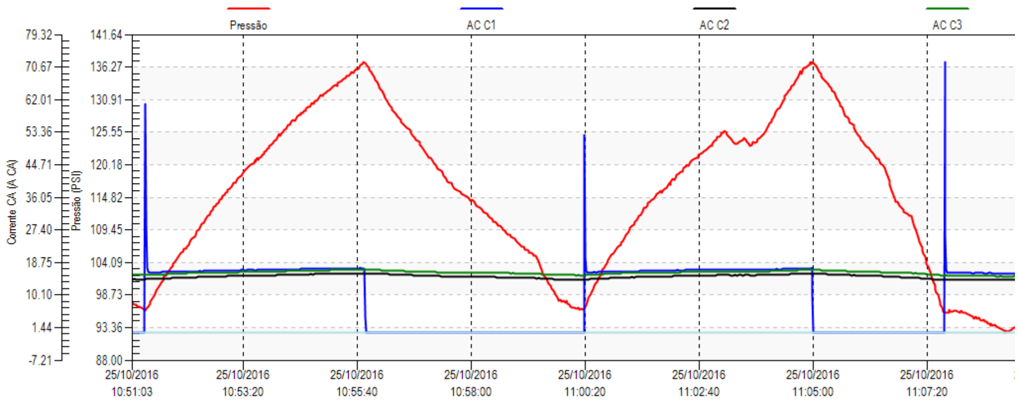

Figure 4. Graph of two ON/OFF cycles from the mains pressure.

To define the operation of these equipment's, the following deduction is adopted, since it is necessary to set the target pressure or OFF pressure and the ON pressure or the starting pressure.

$$Ton = \forall 1 / Qg \tag{1}$$

$$Toff = \forall 2 / Qc \tag{2}$$

Where: T_{on} the operating time (min), product of the ratio between: ∀1 system volume or estimated system capacity (m³) at OFF pressure, and Qg, which is the volumetric flow rate of the compressor equipment given in (Sm³/min), this flow being Qg at the same pressure as ∀1.

T_{off} is time (min) that the equipment will remain off product of ∀2, which is system volume at ON pressure with respect to Qc which is the system consumption volumetric flow in the source condition (Sm³/min).

To obtain an estimate of the energy cost during a normal life cycle of an air compressor, it is necessary to define the amounts of daily cycles, using Eq. (1) and Eq. (2):

$$C = Td / (Ton + Toff)(3) \tag{3}$$

Where: C is the dimensionless daily operating cycle; Td is the operating time in which there is air consumption during a day (min).

From Eq. (3), we can find the operating time:

$$Tl = \left[\left(C \right) Ton \right] / 60 \tag{4}$$

The power used for this type of cycle can be calculated by Eq. (5):

$$Pm = \left[\left(In \right) \left(U \right) \left(\sqrt{3} \right) \left(\cos \varphi \right) \right] / 1000 \tag{5}$$

Pm is the motor power (kW/h) calculated from the motor nominal current In (A), operating voltage (V) and motor power factor (cosφ). The operating time Tl (min) is also required. With such data, by quantifying the operable days in a normal life cycle of an air compressor equipment of 10 years, it is possible to introduce data for an analysis and comparison between systems and generation methods:

$$Dcv = \left(Qd \right) \left(10 \right) \tag{6}$$

$$Cg = \left\{ \left[\left(Pm \right) \left(Tl \right) \left(Dcv \right) \left(C \right) \right] / \eta_{motor} \right\} + CC \tag{7}$$

Being: Cost of generation in cycle of life of the compressor Cg in monetary unit (R$, US$ etc.); Qd number of operating days in one year; C the cost of kW/h in monetary unit (R$, US$ etc.); CC the initial acquisition cost of the system, which involves the purchase of the air compressor equipment and the entire system integrated therein.

3.2 Start and control LOAD/RELIEF

In screw-type compressors from a considered power, it is possible to find operation control mechanisms from the LOAD/RELIEF method, which is a partial solution to avoid problems as mentioned in the ON/OFF starting method. When the equipment reaches the target pressure of the system which would be OFF pressure, it remains operating, but with low admission of mass of air in the entrance door of the airend, after the pressure of the network reaches what would be the ON pressure, the equipment becomes to operate at full load. However even in relief, the equipment consumes electrical energy to maintain the minimum load of operation and therefore cooling of the electric induction motor.

To find the Generation Cost (Cg) of this method, it is necessary to obtain the operating powers of the equipments in the two conditions, applying Eq. (1), (2), (3) and (4) we reach:

$$Tr = \left[\left(C \right) \left(Toff \right) \right] / 60 \tag{8}$$

$$Pml = \left[\left(In_{load} \right) \left(U \right) \left(\sqrt{3} \right) \left(\cos \varphi \right) \right] / 1000 \tag{9}$$

$$Pmr = \left[\left(In_{relief} \right) \left(U \right) \left(\sqrt{3} \right) \left(\cos \varphi \right) \right] / 1000 \tag{10}$$

Being: Tr relief time (min); Pml is motor power (kW/h) under load condition and Pmr, which is motor power (kW/h) when operating in relief.

So finding the Cg for air compressors with LOAD/RELIEF operation:

$$Cg = \left\{ \left\{ \left[\left(Pml \right) \left(Tl \right) \left(Dcv \right) \left(C \right) \right] + \left[\left(Pmr \right) \left(Tr \right) \left(Dcv \right) \left(C \right) \right] \right\} / \eta_{motor} \right\} + CC \tag{11}$$

3.3 Start and control VSD

In the analyzed system a solution was found from the replacement of the four reciprocating air compressors piston, by a screw type with starting system and control through the variable speed drive (VSD). As shown in Fig. 5, the control of the equipment operation is carried out by reading the system pressure, the electric current is always the inverse of the pressure in the system. This method has greater efficiency in floating loads and in systems with great alternation in the consumption during the day of operation, however it is less efficient than an equipment ON/OFF or LOAD/RELIEF in full load by a long period of operation, (Kara et al., 2014).

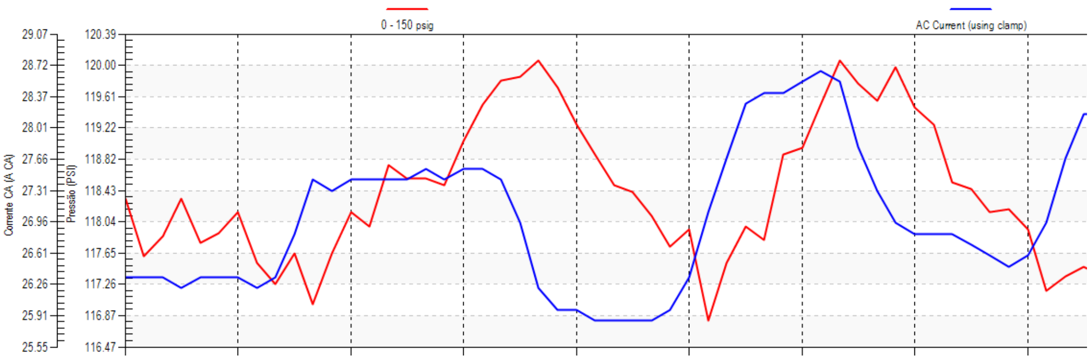

Figure 5. Compressor operating behavior from electric current and mains pressure.

For the Generation Cost of a matrix that has such equipment, the modeling follows from the value obtained in Eq. (6), thus:

$$Pma = \left[\left(In_{\mathrm{var}} \right) (U) \left(\sqrt{3} \right) (\cos \varphi) \right] / 1000 \tag{12}$$

$$Cg = \left\{ \left[(Pma)(Td)(Dcv) \right] / \left[\left(\eta_{motor} \right) \left(\eta_{driver} \right) \right] \right\} + CC \tag{13}$$

Where: Pma is the average power in hours (kW/h), Ina is the average current of the analyzed float cycle. η motor and η driver are their respective engine and VSD yields.

3.4 Control of operation in effect cascade and pressure adjustment

As shown in Fig. 2 for the analyzed system, in compressed air generation matrices that have two or more equipment, it is possible to perform the input and output adjustment by cascade effect when connected to the same network (Kara et al., 2014). Such a configuration based on the consumption demand allows a smaller number of possible equipment to operate at the same time, providing less consumption of electrical energy. It is also possible to carry out this control method from remote electronic controllers or through the configurations of each equipment (CAC, 2005).

For the study under analysis, it is applied as shown in Eqs. (5), (9), (10) and (11) whose power is directly proportional to the current increase and that it is proportional to the workload in the generation of compressed air. Thus, adjusting and setting an appropriate pressure, which meets the minimum requirements, promotes an economy that can reach up to 15% for every 0.1 MPa reduced (Saidur, 2010).

4. CASE STUDY FINDINGS AND CONSIDERATIONS

As shown, based on the principles demonstrated, the construction of the consumption profile of the system and, consequently, the analysis of the compressed air generation matrix, focused on the fundamental Benchmarks, was carried out in the case analysis. For the case study system at the start point, the following data were obtained after the analysis:

Table 2. Results of the analysis of data compiled from the second day of the case study matrix.

Analysis Data				
	COMPRESSOR 1	COMPRESSOR 2	COMPRESSOR 3	COMPRESSOR 4
Analysis Time (min)	24.00	24.00	24.00	24.00
ON Time (min)	1.82	20.09	15.20	13.58
Number of Matches	39	74	20	34
Percent of Working Time	8%	84%	63%	57%
Generated Flow (m³/min)	0.088	0.971	0.735	0.657
Average Power (kW)	9.64	8.42	9.03	8.74
Cost (R$)	R$ 11.68	R$ 112.72	R$ 93.67	R$ 79.15

Table 3. Benchmarks of the analyzed system.

Global System Data	
Average Power (kW)	35.82
Specific Power (kW/m³/min)	14.62
Specific Cost (R$/m³/min)	R$ 7.48
Maximum Consumption (m³/min)	2.45
Target Pressure	135 PSI

Evaluating the demand presented, in interaction with the compressed air storage capacity of the system, it was determined that the best conditions would be between the use of a 22 kW (30HP) screw type compressor in LOAD/RELIEF starting system or, looking for an expansion and therefore future consumption, a 30 kW (40HP) bolt-type air compressor in VSD starting and control system. This second option was selected and acquired. Finally, after the installation of the equipment, the analysis and determination of the new fundamental Benchmarks was carried out:

Table 4. Results of the analyzed data of the system with the new compressor of 30 kW.

New System - Analysis Data	
	COMPRESSOR 30 kW
Analysis Time (min)	24.00
ON Time (min)	24.00
Number of Matches	1
Percent of Working Time	55%
Generated Flow (m³/min)	2.750
Average Power (KW)	17.49
Cost (R$)	R$ 223.40

Table 5. Selected new system benchmarks.

New Global System Data	
Average Power (kW)	17.49
Specific Power (kW/m³/min)	6.36
Specific Cost (R$/m³/min)	R$ 3.38
Maximum Consumption (m³/min)	2.75

The evaluation of the system with the initial and final conditions is presented in Fig. 6, which transcribes the functionality of the application of mathematical modeling proposed for the evaluation of the applicability as well as the viability of the given system in using the use of one of the two types of compressors. To illustrate how this happened, the CC values were used according to given equations, for CG PISTON COMPRESSOR of R $ 44,000.00 and CG SCREW COMPRESSOR of R $ 96,000.00, such being the formation of the price of acquisition of the system.

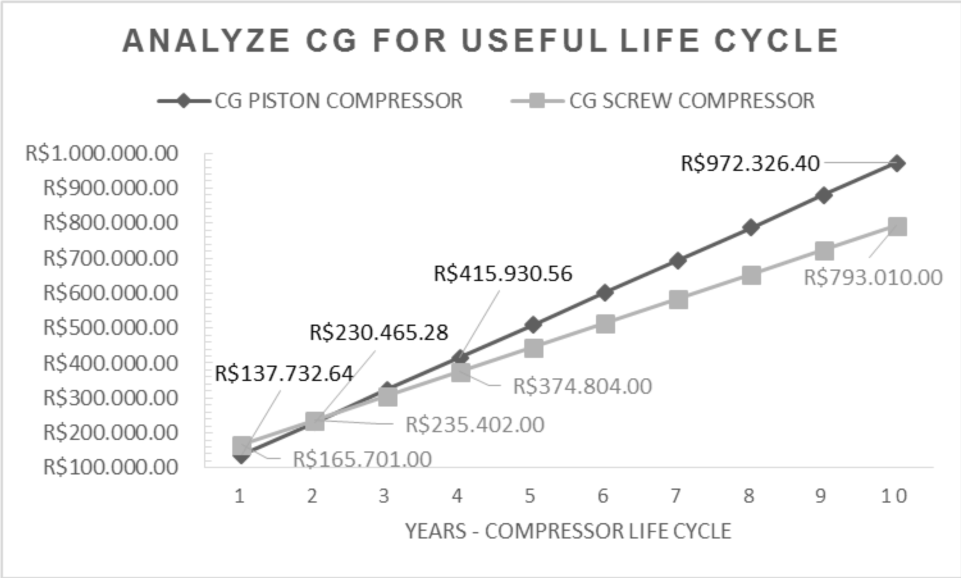

Figure 6. Comparison of the accumulated costs of each equipment within the life cycle of compressors.

5. CONCLUSIONS

In the case study presented, comparing the generation matrix with four air compressors of the piston type in relation to one air compressor of the screw type, it was identified that in less than 3 years the cost trend between screw and piston compressors are, as shown in Fig. 6. For the following years, there is a progressive increase in economy of the screw type compressor, reaching 2.09 GW of economy in 10 years. The factors for this result are associated to the starting and control mechanisms, the equalization of the target system pressure, and higher volumetric flow rates for the case of the bolt-type air compressor with greater electrical efficiency of the package.

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