Review

Energy saving potential in existing industrial compressors

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A R T I C L E   I N F O

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A B S T R A C T

The Compressed Air Sector accounts for a mean 10% worldwide electricity consumption, which ensures about its importance, when energy saving and CO2 emissions reduction are in question. Since the compressors alone account for 15% overall industry electricity consumption, it appears vital to pay attention to machine performances.

The paper presents an overview of present compressor technology and focuses on saving directions for screw and sliding vanes machines, according to data provided by the Compressed Air and Gas Institute and PNEUROP. Data were processed to obtain consistency with fixed reference pressures and organized as a function of main operating parameters. Each sub-term, contributing to the overall efficiency (adiabatic, volumetric, mechanical, electric, organic), was considered separately: the analysis showed that the thermodynamic improvement during compression achievable by splitting the compression in two stages, with a lower compression ratio, opens the way to significantly reduce the energy specific consumption.

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1. Introduction

At the present day, many different circumstances concur to increase the difficulty in defining a comprehensive and concerted energy policy. Factors as energy sources availability, the too weak presence of renewables in the global market, the effects of the economic downturn on energy issues and the difficult ongoing of international climate negotiations, all these aspects represent different facets of the same problem, i.e. the attempt to define a form of sustainable energy development consistent with growth rate of each economy in the World. Although true in general, these considerations apply to the electric energy scenario more than anyone else, since more than 50% electricity demand comes from the industrial sector, driver in the economy of both developed and developing Countries (Fig. 1) [1].

A greater detail for data in Fig. 1 can be found in Table 1, where the electricity need relative share among sectors is reported, suggesting the great role played by BRICS Countries (acronym for Brazil, Russia, India, China and South Africa) in the present electricity demand. The industry is the most present sector among all the others: percentages, always close to 50%, rank at 78% and 68% for China and India, respectively, giving a clear insight of where the greatest outcome in term of emissions reduction should be.
expected, when energy saving and energy recovery measures were adopted.

The relevance of emissions-related climate changes and the fact that the above mentioned measures can be no longer postponed are both stated in many independent studies [2,3]. Among them, the one performed by the IPCC (Intergovernmental Panel on Climate Change) confirms that at the present GHG (greenhouse gases) emission rate, a warming exceeding 4 °C14 on the average global temperature by the end of the century is going to be unavoidable [4]. Such a rise in temperature almost doubles the 2 °C increase in the 21st century, commonly accepted as a limit to avoid irreversible damage to climate systems and to prevent the socio-economic models from collapsing. The datum on temperature rise fixes a cap on global GHG emissions by 2020, at 44 GtCO2eq. Projections, based on the current development scenario and on the current energy sources market composition (up to 87% energy demand covered with fossil fuels), bring global GHG emissions from 50 GtCO2eq in 2010 to 59 GtCO2eq in 2020, i.e. 15 GtCO2eq beyond the 44 GtCO2eq limit.

Even less optimistic perspectives shall be considered when future growth scenarios are taken into account. Fig. 2 shows the trend in electricity demand on a 45 years basis (1990–2035), for the main World Countries: a positive trend can be always detected, even though differences in increase rates can be appreciated for each context [5,6].

The growth in electricity demand by 2035, as a percentage of its value in 2012, is expected to rank at 187.7% for Russia and 115.6% for China, while at only 11.1% for Asia-Oceania. This depicts a complex and dishomogeneous situation, in which it appears hard to take shared energy initiatives. Main concern of the electricity growth in the future is CO2 concentration into the atmosphere; even though greater attention is being paid to market related initiatives for carbon emissions reduction, strong differences still occur, among different geographical areas and socio-economic contexts, which make very difficult the adoption of a shared political action.

In light of all the above mentioned factors, when the goal is the reaching of the 2 °C target, the adoption of mitigation policies is no longer an option. Nevertheless, a compromise between the call for emissions reduction and the current development scenario, still strongly dependent on fossil fuels, is needed as well. The most effective way to achieve this goal, is offered by the adoption of energy efficiency/recovery measures in present energy-intensive (and particularly, electro-intensive) applications. Another aspect worth considering is the importance of defining common actions on each and any specific sector and to privilege those characterized by the greatest probability of success. Among them, the CAS (compressed air systems), whose energy demand is almost entirely electricity-oriented, exhibit a great potential for improvement, under both the energy and emissions point of view. It has to be

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**Table 1**

<table>
<thead>
<tr>
<th>Country</th>
<th>Residential</th>
<th>Commercial</th>
<th>Industrial</th>
<th>Transports</th>
<th>Total</th>
<th>Industrial</th>
</tr>
</thead>
<tbody>
<tr>
<td>United States</td>
<td>1374.6</td>
<td>1323.8</td>
<td>1021.6</td>
<td>6.8</td>
<td>3726.8</td>
<td>27.4</td>
</tr>
<tr>
<td>European Union</td>
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<td>844.9</td>
<td>1059.2</td>
<td>64.1</td>
<td>2796.6</td>
<td>37.9</td>
</tr>
<tr>
<td>Australia</td>
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<td>60.9</td>
<td>82.3</td>
<td>4.1</td>
<td>209.3</td>
<td>39.3</td>
</tr>
<tr>
<td>Brazil</td>
<td>117.6</td>
<td>119.6</td>
<td>232.9</td>
<td>2.8</td>
<td>472.9</td>
<td>49.2</td>
</tr>
<tr>
<td>Russia</td>
<td>132.2</td>
<td>161.9</td>
<td>354.2</td>
<td>92.1</td>
<td>740.3</td>
<td>47.8</td>
</tr>
<tr>
<td>India</td>
<td>190.9</td>
<td>76.1</td>
<td>586.4</td>
<td>15.4</td>
<td>868.8</td>
<td>67.5</td>
</tr>
<tr>
<td>China</td>
<td>621.9</td>
<td>244.3</td>
<td>3209.9</td>
<td>51.9</td>
<td>4128.1</td>
<td>77.8</td>
</tr>
<tr>
<td>South Africa</td>
<td>38.8</td>
<td>28.2</td>
<td>126.3</td>
<td>3.8</td>
<td>197.1</td>
<td>64.1</td>
</tr>
<tr>
<td>Total</td>
<td>3366</td>
<td>2860</td>
<td>6673</td>
<td>241</td>
<td>13,140</td>
<td>50.8</td>
</tr>
</tbody>
</table>

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**Fig. 1.** Electricity by sectors (year: 2013).

**Fig. 2.** Electricity demand (1990–2035).
noted, indeed, that a big share of electricity consumption in present industrial scenario is CAS related: up to 20% industry electricity consumption (i.e. 1335 TWh/y) is due to air compression and delivery to final uses, which corresponds to an even higher figure in terms of primary energy consumption, considered the low characteristic efficiencies of electricity generation worldwide. Bearing in mind that, despite their complexity, compressed air systems have low characteristic efficiencies and that up to 40% compressed air of any installation is used in a wasteful way, it appears clear why energy saving/recovery is considered the only viable option to reduce the energy impact and Carbon footprint of CAS.

2. Energy consumption in industrial compressed air systems

According to many independent statistics, industry alone is responsible for up to 50% electricity consumption in the CAS, with peaks in glass and ceramic production, refining processes and automotive/aerospace manufacturing [7]. Smaller contributions, still not negligible, come from residential, commercial and transports sectors, whose sum covers the remaining 50% CAS-related electricity consumption. These data clearly state the importance of compressed air in present economy and development models and give evidence that CAS-related consumption mainly concentrates on industry. Thus, in order to stress the relevance of the CAS in the present energy and emissions scenario and to have a better understanding of the potential improvements it offers, the compressed air electricity consumption can be conveniently seen as a 20% of the industry electricity need, rather than a 10% of the overall one (i.e. the sum of electricity consumption in residential, commercial, industrial and transports sectors) and the analysis will focus mainly on CAS for industrial purposes.

As known, both upstream and downstream interventions can be performed on a compressed air system, dependently on whether the measures take place before or after the compression. A typical upstream measure consists in a proper selection of components (e.g. compressor, electric motor, control system), while downstream measures mainly consist in the adoption of energy saving/recovery strategies (e.g. heat re-use, leakages detection and fixing).

Many independent studies aim at ranking the saving potential of different measures, to address main criticalities of CAS. In present compressed air chain, end use devices optimization to avoid a wasteful use of the air (35–40%), air leaks reduction (20%) and use of sophisticated control systems, to better match the air demand profile through flow modulation instead of throttling (15–20%), appear to be the most effective actions. Furthermore, since the energy consumption for compressed air production can be seen as the product of the flow rate delivered and the specific energy consumption of compressors, it appears clear that a decrease in the compressor and electric motor technology proportionally reflects on overall energy consumption. As a consequence, the compressor upgrading, whose contribution ranks at 10–20% energy saving potential, along with the use of high efficiency motors (5%) are considered interesting areas of intervention. Lower values characterize frictional pressure losses reduction and the CAS thermal management (i.e. the recovery and re-use of waste heat from air and lubricant), both within the 2–5% range [7–10].

Since the compression section alone accounts for a 10–20% electricity consumption, great attention is being currently drawn by the quest for increased compressor performances and a reduced consumption in flow rate modulation [7]. Given the relevance of the compressor efficiency issue, a comprehensive analysis of present market compressors performances is of the essence and is performed in the following, on a wide variety of machine types, with a particular focus on screw and sliding vanes compressors, according to the data provided by the CAGI (Compressed Air and Gas Institute) for the US market [11] and Pneurop for the EU scenario [12,13].

As a preliminary remark, it has to be noted that data provided by Pneurop mainly refer to the market dimension of compressors in Europe, accounting for aspects as sales volume and operating hours per year, in order to investigate the market acceptance level for compressors, different in types and sizes (fixed and variable speed drive, flow rate), with very little or no investigation of performances and technological standards. As a result, the presence in the Pneurop data population of obsolete design machines, still sold and present on the market, with lower efficiencies than the most advanced compressors, has to be expected. CAGI data, on the contrary, concentrate on the technological side, providing detailed information on current standard in compressors performances and allow a comprehensive assessment of the present technological scenario. It should not surprise, then, if slight unevenesses were detected on the two data distributions, given their different nature and goals.

The analysis aimed at the evaluation of market compressors efficiency as starting point to assess the room for improvement provided by different actions: whereas data on Pneurop outlook are available in a standardized form and refer all to the same delivery pressure (9 bar absolute), the declared values for specific consumption on CAGI website do not refer to a unified set of delivery pressures. This represent a big issue, since small variations around specified levels (those usually assumed as reference) make the direct comparison among different machines useless, when the goal is to investigate the state-of-the-art in compressors efficiency. Even though CAGI provides the most important set of data among those currently available, such a lack in the respect of reference values is frequent (e.g., on CAGI datasheets, two different machines operating with a 8.6 and 8.9 bar delivery pressure, are both filed as machines providing air at 9 bar; same inconsistency in data shows up when two actual delivery pressures of 8.4 and 8.1 bar are equally approximated to an 8 bar one; when a comparison was made, a severe distortion in the results should be expected). Thus, in order to compare performances of compressors from different manufacturers as provided by CAGI, the specific consumption had to be recalculated to match fixed reference pressures, by means of the trimming procedure consisting in the following steps:

a) reference pressures are fixed equal to 8, 9, 10 and 11 bar;
b) if the pressure delivered by a specific machine remains in the 5% reference value assumed, data are processed and modified in order to be referred to reference pressure levels; otherwise, they are excluded;
c) for the processed data, reference and measured pressures are known and the corresponding compression ratios are known as well. Hence, energy consumption can be referred to common reference pressures, by observing that the overall compressor efficiency can be seen as:

$$\eta_{glob} = \eta_{ad.is} \cdot \eta_{vol} \cdot \eta_{mech} \cdot \eta_{rel} \cdot \eta_{org}. \quad (1)$$

The volumetric, mechanical, electric and organic terms are considered as constants, when the energy consumption from the measured values is referred to reference values. For this reason:

$$\Delta \eta_{glob} = \Delta \eta_{ad.is}. \quad (2)$$

$$\eta_{ad.is} = \frac{\Delta \eta_{ad.is}}{\Delta \eta_{real}} = \frac{R \cdot T_{ini} \cdot \beta_{1}^{1/2} - 1}{c_{p} \cdot T_{ini} \cdot \beta_{1}^{1/2} - 1}. \quad (3)$$
\[
\frac{d\eta_{ad.is}}{d\beta} = \text{const} \left( k, \bar{R}, T_{\text{int}}, c_p, p \right) \cdot \beta^{-1}
\]  

(4)

The specific consumption variation related to the global efficiency and the pressure ratio is given by:

\[
\frac{\Delta q_s}{q^{msr}_s} = \frac{q^*_s - q^{msr}_s}{q^{msr}_s} = \frac{\Delta \eta_{glob}}{\eta_{glob}} = \beta - q^{msr}_s
\]

(5)

The specific consumption that corresponds to the compression ratio assumed as reference is then:

\[
q^*_s = \left( 1 + \frac{\Delta q_s}{q^{msr}_s} \right) q^{msr}_s
\]

while, for the mass flow rate:

\[
m^* = m^{msr}
\]

(7)

It is worth observing that such an approach applies only when measured and reference pressures are close, so that no differences can be detected in the nature of the transformation (assumed to be a polytropic) at the vane opening in real and reference case. Thus, in order not to lose validity on the assumption of a polytropic, a tight tolerance needs to be fixed on delivery pressure (±2.5% reference value) and the consistency of actual pressures to this condition becomes the main criterion to establish whether the machine can undergo the analysis. The criticality of this issue can be particularly appreciated in rotary volumetric compressors, where the pressure inside the vane always matches the line one with a constant-volume transformation (isochoric phase); hence, the bigger the shift between measured and reference pressures, the wider the gap between real transformation and reference polytropic, the less fitting the assumption of a polytropic in the calculations.

The exclusion procedure reduced the number of machines taken into account to almost 75% CAGI original data (1000 machines against 1200 original data). The normalized ones are reported in Fig. 3: for sake of privacy, no mention is made to the manufacturer and the data are organized in four groups (fixed and variable speed, oil cooling by air and water) and sub-divided according to the delivery pressure. The y-axis reports efficiency values, calculated having the adiabatic isentropic as reference, according to:

\[
\eta^*_{ad.is} = \rho \cdot \bar{R} \cdot T_{\text{int}} \cdot \left( \frac{k}{k-1} \right) \cdot \left[ \beta - q^{msr}_s \right] - 1
\]

(8)

\[
\eta = \frac{q^*_s}{q_s}
\]

(9)

A \( \eta^* = A \cdot m^{*\alpha} \) regression model is chosen to interpolate real data points, in order to highlight the technological trend in actual compressors. The coefficient of determination \( R^2 \) is a statistical parameter that allows to evaluate the goodness of the fitting: the closer \( R^2 \) to one, the less disperse real data points around their mean value, the smaller the technological scatter among different manufacturers.

\[
R^2 = \left( \frac{\sum (m^* - m^{**}) (\eta^* - \eta^{**})}{\sqrt{\sum (m^* - m^{**})^2 \sum (\eta^* - \eta^{**})^2}} \right)^2
\]

(10)

The following considerations apply:

a) bigger machines (i.e. machines operating at higher flow rates) are characterized by higher efficiencies: for air cooling, at 5 m³/min, 20 m³/min and 50 m³/min, the mean value for efficiency is 65%, 73% and 82% respectively; at the same flow rates, characteristic efficiencies for water cooled are 68%, 78% and 86% respectively. This trend is particularly evident at higher delivery pressures;

b) lower specific consumption characterizes water cooled machines; main reason for this is the fact that in air cooled machines, a fan is needed, while in water cooled, a pump is employed. Since a fan circulates a compressible fluid, it

![Fig. 3. CAGI – Market compressors performances (volumetric flow rate evaluated at 1 bar, 293.15 K).](image-url)
shows a higher consumption which adversely affect the package global efficiency;
c) an asymptotic trend can be detected at all pressure levels, with tendency lines flattening on an average efficiency value with increasing mass flow rate; 8 bar delivery pressure machines show the greatest technological scatter among manufacturers: for a 30 m³/min flow rate, the average value of efficiency in fixed speed air cooled machines is 0.80, with a scatter of 0.20, while yet at 50 m³/min the average value is 0.81, with a 0.10 scatter. This datum provides a clear indication on where the compressor choice allows the greatest reduction in energy absorption, with the scatter in efficiency (i.e. the performance variability) becoming a more impacting factor than the efficiency itself (e.g. in the above mentioned situation, a 20 m³/min increase in flow rate corresponds to an increase in efficiency an order of magnitude lower than the shift in efficiency scatter). Greater accuracy should then be put into the machine choice at lower flow rates, where the actual efficiency can significantly differ from the average one.

Similar considerations apply to water cooled machines;
d) when variable speed machines are considered, the comparison between water and air cooled shows higher efficiencies for the former, especially at higher delivery pressures. It is worth observing that 9 bar machines are those characterized by the smallest scatter (i.e. the highest R²) in both water and air cooled family, at any flow rate. Water cooled machines have higher efficiencies than air cooled (0.8 vs 0.7);
e) isolated points on the diagrams show that the machines are not aligned to the best standards; this is mainly due to the presence of obsolete designs on the market.

For sake of clarity and in order to have, at the same time, a ready-to-use information and a datum that accounts exclusively for machine performances, when the goal is to set up an average value for compressors efficiency a simplified approach is possible: since no difference in compressor technology can be appreciated between air and water cooled families, exception made for the need for a radiator to provide water cooling in the latter, the only distinction worth considering is the one between fixed and variable speed drive. The results of such an analysis are in Fig. 4, in which no distinction between air and water cooled compressors is made: at each flow rate, an average efficiency is derived for machines delivering air at 8, 9, 10, 11 bar, along with the one referring to the whole class (i.e. the whole spectrum of delivery pressures), in both fixed and variable speed families.

Fig. 4a and b give evidence that:

a) the CAE (class average efficiency) is higher in fixed speed than in variable speed, which is particularly evident at flow rates lower than 40 m³/min. A 1% difference is detected at 10 m³/min (72% vs. 71% in fixed and variable speed, respectively), while at flow rates higher than 40 m³/min, the distance tends to decrease and never overstays the 0.6% limit (81.6% vs. 81% at 100 m³/min, in fixed and variable speed, respectively);
b) differences can be appreciated in the distribution of efficiency values for each delivery pressure, with respect to the CAE: in fixed speed class, the efficiencies at 8 and 9 bar delivery pressures are close one other and always lower than the CAE, with the difference increasing with increasing flow rate (1.6% difference at 10 m³/min, 2.4% at 100 m³/min). Efficiencies at 10 and 11 bar delivery pressure are higher than the corresponding CAE at any flow rate, with the minimum and maximum shift detected at 10 m³/min (1.7%) and 100 m³/min (5.2%) respectively. Variable speed is characterized by a less predictable distribution in efficiency values: efficiencies for 8 and 9 bar delivery pressure always stand below the corresponding CAE, whereas 10 bar delivery pressure is characterized by efficiency values, whose distance from the CAE ones never exceeds the 0.7%, reached at 10 m³/min (70.3% vs. 71% CAE);
c) an average value of efficiency for both fixed and variable speed families can be calculated as the mean value of CAEs in the flow rate range of interest for market compressors, leading to a 78.2% efficiency for fixed speed and to a 77.5% efficiency for variable speed; even though, when it comes to establish whether to choose a fixed or a variable speed machine, many aspects, strictly related to the actual operating conditions of the compressor (e.g. need for load/unload control, speed modulation applicability), have to be considered, the values provided here still represent a reliable tool (since they come out of standardized and structured data) in evaluating what technology should be preferred, when the goal is to maximize the energy performance of the system.

Since CAGI machines operating with 9 bar delivery pressure have the lowest performance variability among manufacturers, a comparison with 9 bar delivery pressure machines in the European scenario can be performed, based on data provided by Pneurop (Fig. 5). Since Pneurop only differentiate between fixed and variable speed machines, air and water cooled are gathered together: as a consequence CAGI air and water cooled machines collapse into one single population, to be compared to the Pneurop corresponding one. Best and worst in class machines at any flow rate, for both CAGI and Pneurop, are reported; the trend lines fit the whole population of 9 bar delivery pressure, air and water cooled machines: the efficiency values for CAGI and Pneurop are close, even though a higher performance variability characterizes Pneurop machines in fixed speed family, particularly at 60 m³/min, where actual values can be a 10% lower than the fitting ones. Main reason for this is the fact that Pneurop data offer an overview of all the machines on the market and make no distinction between up-to-date and obsolete design compressors. A much more balanced situation appears in the variable speed class, where a 15% difference between the present efficiency and the fitting values can be appreciated, on both best and worst in class sides on the whole flow rate range, with no meaningful fall in efficiency for bigger machine sizes.

Further evidence of the similarities between CAGI and Pneurop data is in Fig. 6, where the isentropic efficiency of market compressors is reported as a function of air flow rate (log scale); solid and dashed lines in Fig. 6 report the technological standards for the European scenario for best in class machines (BIC, i.e. those machines characterized by an efficiency higher than the average one), WIC (worst in class machines, i.e. those machines characterized by the lowest values of efficiency) and AIC (average in class machines, i.e. machines with an average efficiency level), according to Pneurop data. CAGI data, referring to the main compressors manufacturers whose compressors are compliant with the above mentioned criteria in data selection (i.e. the constraint in the value of delivery pressure), are reported as well. Data are presented anonymously and thus the manufacturer is referred to as M1, M2, M3, M4: the characteristic efficiency values are consistent with Pneurop expectations, with data points that mainly concentrate in the area between Pneurop average and best in class baselines. It appears then clear, that compressors performances are aligned to the best standards in both the EU and US contexts. A particular focus is on SVRC (sliding vanes rotary compressors), that, especially in variable speed family, represent a reference for present technology, since their characteristic efficiencies are always between the baselines for average and best in class.
3. Compressors performances assessment

When dealing with potential energy saving in compression, two transformations can be assumed as reference: an isothermal (as the one characterized by the minimum energy requirement) and an adiabatic isentropic (as the closest one to present compression). Once reference transformations are fixed, the distance between actual and ideal specific consumption expresses the maximum achievable reduction in real compression. The lower limit to this figure can be fixed by taking into account, among present compressors, premium machines, since they are characterized by the lowest specific consumption (i.e. the highest efficiency) at any compression ratio. As a consequence, the analysis performed on premium machines provides an underestimation of the potential energy saving achievable in compression: higher energy savings should be expected when the same measures apply to medium efficiency-level compressors. As previously observed, the highest efficiencies can be appreciated at the highest flow rates, to whom high $R^2$ values correspond. In Fig. 7, the efficiency values for real compressions are calculated having the adiabatic isentropic as reference, according to Equation (9).

This definition is not in conflict with the one provided in Equation (1), where other terms, apart from the thermodynamic one, are taken into account: when, at a preliminary stage of analysis, as the one considered here, the mechanical, volumetric, electric and organic terms are fixed at their maximum values, the efficiency improvement depends solely on the thermodynamic term. At different compression ratios, the efficiency is reported for real compression, with particular emphasis on best and worst in class machines.
Characteristic values of efficiencies for these machines are labeled at whiskers top and bottom. Other three lines (rectangles horizontal bases) are reported and represent, from the bottom to the top, the first quartile, the median and the third quartile: these lines, along with the upper and lower cap, subdivide the data population into four groups, each one containing a 25% of the overall population. Hence, a clear indication of how the efficiency values distribute around the average one is provided: the higher the distance between the quartiles, the higher the degree of dispersion of data (i.e. the greater efficiency spread). This datum, along with the one referring to the average efficiency, provides different elements for a comprehensive analysis of the efficiency scenario in market compressors:

a) the bigger the distance between best and worst in class machines, the huger the margin for improvement in market compressors, with respect to the one calculated on premium machines;
b) data distribution around the median varies with compression ratio and machine type (air or water cooled), suggesting that great variability in efficiency values must be expected (e.g. in Fig. 7a, characteristic efficiencies for WIC and BIC machines operating at a compression ratio of 7.95 vary from 46% to 81%, with an AIC efficiency of 71%; in Fig. 7d, the same data for a compression ratio of 11.48 are 74%, 82% and 78%);
c) the variable speed family always shows the smallest scatter in efficiency, between BIC and WIC compressors, giving evidence of a greater performance control in this family: the maximum scatter in variable speed ranks at 26% in air cooled, at 9.68 compression ratio (Fig. 7b) and at 23% in water cooled, at 7.95 compression ratio (Fig. 7d); a wider scatter in efficiency characterizes fixed speed air cooled compressors (in
**Fig. 7a**, the lowest one ranks at 22% at a compression ratio of 10.72;

d) when operated at rated flow rates, variable speed air cooled machines show a slightly lower efficiency: this is due to the presence of the inverter, which introduces an additional loss.

When flow rates are matched through speed modulation (instead of throttling or on/off load managing), the higher efficiency of variable speed machines is confirmed;

**Fig. 7.** Premium machines vs ideal adiabatic — Efficiency.

e) for greater pressure ratios, higher efficiencies correspond to bigger machines (i.e. higher flow rates); this means that it is difficult to realign all the machines to the best in class ones. Subclasses should be considered as a function of the flow rates, as reported in Figs. 5 and 6.

A greater detail in the comparison between CAGI and Pneurop data can be appreciated in **Fig. 8**, where the focus is on those machines operating at the same compression ratios.

**Fig. 8.** CAGI (left box) vs Pneurop (right box) — Comparison at common compression ratios.
More specifically, attention should be paid to the following points:

a) for AIC (average in class machines), the efficiencies are always close one other, except for fixed speed machines at a compression ratio of 8.99; as a result, a common standard for market compressors in the EU and US can be fixed, with efficiencies always in the range 60–70%;

b) whether premium machines show no (as in fixed speed at 8.99 compression ratio) or low scatter in efficiency (4% in fixed speed at 8.64, 14% in fixed speed at 8.99 and 16% in variable speed at 8.99), most likely because machines with low efficiency are still chosen in the market;

c) a higher performance variability can be appreciated on Pneurop data, suggesting that the compressor choice should receive greater attention in the EU context than in the US: the higher the shift in efficiency, the higher the room for reduction in specific consumption, achievable by orienting the choice towards up-to-date high efficiency compressors.

An adiabatic isentropic transformation provides the best fit of premium machines, in all four families; as shown in Fig. 7, the value of approximating efficiency (i.e. the one that minimizes the distance between premium machines and the interpolating adiabatic) changes among machine types, with fixed speed water cooled showing the greatest efficiency value (0.829), followed by variable speed water cooled (0.817) and fixed speed air cooled (0.813); the lowest efficiency is in variable speed air cooled compressors (0.786). CAGI data on compressors efficiency can be conveniently read in terms of specific consumption, as in Fig. 9: a direct evaluation is possible, of how far the specific consumption in actual compressors (BIC, AIC and WIC compressors at each compression ratio) is from the one that characterizes an ideal adiabatic compression (dashed line).

It has to be noted that:

a) the premium machines that best approximate the specific consumption of adiabatic isentropic compression can be found in the variable speed family: a 16% difference is detected at 9.68 compression ratio for air cooled (Fig. 9b) and 16.1% at 11.07 compression ratio for water cooled (Fig. 9d); the same trend characterizes the specific consumption in AIC machines, with values ranging from 24.8% to 29.8% in air cooled compressors and from 27.3% to 35.8% in water cooled;

b) 36.9% and 44.4% are the minimum and maximum difference in specific consumption between AIC machines and the adiabatic isentropic compression in fixed speed water cooled (Fig. 9c); the same data for fixed speed air cooled are 35.7% and 60.5% (Fig. 9a); the higher differences on AIC machines suggest that, when the improvement is intended as a best approximation of an adiabatic isentropic compression, the greatest room for performance improvement should be sought in fixed speed air cooled compressors;

![Fig. 9. Premium machines vs ideal adiabatic — Specific consumption.](image-url)
c) the shift between the adiabatic which interpolates the best machines and the ideal datum stands, for air cooled compressors, within a 0.9—1.4 kW/(m³/min) for fixed speed and between 0.9 and 1.2 kW/(m³/min) for variable speed, for water cooled, the specific consumption ranges between 1.2 and 1.6 kW/(m³/min) and 0.9—1.3 kW/(m³/min) for fixed and variable speed respectively.

Such a variability in data clearly states the need for an in-depth investigation of all terms in Equation (1), in order to evaluate where, according to machine type and operating conditions, the greatest contribution to efficiency penalty comes from.

4. Potential achievable energy savings

In order to evaluate the overall potential savings, each term contributing to the global efficiency in Equation (1), deserves some attention:

- \( \eta_{\text{ad,ale}} \): it refers to the thermodynamic of the compression and it expresses the distance between the actual compression and the one assumed as reference. In both screw and rotary vanes, the one stage compression closely approximates an adiabatic isentropic. Once the indicated cycle is available, the \( d\h_{\text{ad,ale}} \) in Equation (3) can be evaluated and experimental activity proves that the main differences between ideal and actual compression show up at intake and discharge, because of the pressure regime at the vane—line interface: at discharge, the vane pressure doesn’t meet the line one, while at intake, the residual expansion after exhaust port closing is responsible for the mismatch between vane and ambient pressure. Experimental data suggest that these phenomena have a negligible effect, thus efficiency can be fixed to one. Given the technological difficulties in realizing a constant temperature compression, the isothermal can be only ideally considered a reference. Nonetheless, recent advancements in rotary volumetric compressors aim to decrease the compression work by cooling the air during compression, both internally and externally. The internal cooling is performed through the injection of a fine oil spray in the air stream during compression: once in the vane, the heat transmission to the oil takes place mainly through direct contact between air and oil droplets and through the adhesion of oil droplets to vane walls at higher temperature. As a consequence, a partial vaporization of the lubricant can take place inside the vanes, resulting into the unwanted presence of oil vapor phase in the amount of air being compressed and to an increase in compression power. As presented in different research works [14,16], a mean 5% reduction in specific consumption with respect to the actual compression is achievable. Nonetheless, even if such a strategy is currently under investigation, with the main matter of concern represented by the need to control heat exchange and the oil-air interaction dynamics (e.g. injection pressure, direction, optimum droplets dimension), the external cooling of the air, by means of dedicated devices, still represents the most effective technique to reduce the compression power [14,15]. As discussed in the following, mean values for the energy saving related to the adoption of a dual stage compression, with intermediate air cooling and a 30 °C pinch-point at the heat exchanger, range between 10 and 20% specific consumption of single stage premium machines, dependently on the compression ratio adopted (i.e. the higher the compression ratio, the higher the potential saving);

- \( \eta_{\text{mech}} \): it almost entirely accounts for the loss represented by the energy expense to compress air that stands within the compressor and is not discharged at the port opening, because of unsteadiness at the vane—line interface. For best in class machines, real and ideal flow rates are close and the volumetric efficiency can be approximated to one. However, it is known that, according to compressor type, margins for improvement associated to the leaks reduction can still be appreciated: in rotary vanes, on the vertical planes, where sealing effect is not guaranteed [15]; in screw machines, on the so called “blow hole line” [17]. As said, the greatest detrimental effect on volumetric efficiency comes from losses that characterize the filling and emptying processes: while intake can be considered steady, when air is exhausted an intrinsic unsteady behavior is present and losses occur. Other aspects having a negative impact on the volumetric efficiency are those related to internal leaks: the vane is squeezed during rotation, but a residual quantity of air inside it cannot be expelled, resulting in a mass transfer between the vanes. The effect on the volumetric efficiency of rotational speed is worth being discussed. In SVRC, there are no operational constraints on its value: the presence of a stable oil film between surfaces in relative motion prevents internal pressure-controlled leaks through clearances, allowing to fix this efficiency to one. Actual values, as presented in Ref. [18] are always comprised between 88% and 97%. In screw compressors, low rotational speeds lead to an increase in the air recirculated through the blow-hole line, while they have residual effects on the leaks through the screws/compressor housing clearances and between meshing rotors. The resulting continuous fluid transfer inside the machines represents a fixed flow loss. Furthermore, the continuous recompression of an already worked gas represents a fixed power loss;

- \( \eta_{\text{lm}} \): the greatest penalty to compressor efficiency comes from friction between parts in relative motion, accounting for a 20—25% package power in sliding vane rotary compressors. In both rotary vanes and screw compressors, the reduction of friction between rotor and stator is addressed as the most effective way to improve mechanical efficiency, whereas, dependently on the machine type, further specific measures can be adopted. In rotary vanes, a further loss reduction can be achieved through an appropriate design of contact surfaces (i.e. at the blade tip—stator interface the contact occurs because of centrifugal force on the blade; the differential pressure inside adjacent vanes induces contact at the slot—blade interface) and friction reduction in bushings. When the stator-rotor coupling is concerned, the choice of appropriate geometry and rotational speed is of the essence in friction reduction: the variation of the geometrical diameter-length ratio appears the most powerful tool, since it modifies reaction forces on the blades (i.e. higher stator diameters lead to higher centrifugal forces; an increase in rotor diameter leads to a lower eccentricity, with lower relative speed between blade lateral surfaces and the slot). Even though the adoption of lower rotational speeds has opposite effects on the centrifugal forces responsible for friction, down-speeding provides interesting margins for improvement: the mass flow rate entering the machine suffers from the reduction in rotational speed and consequently an increased compressor length and an increased blades mass is needed to keep the mass flow rate constant. Such a mass increase gives a positive contribution to centrifugal force (linear increase), still lower than the negative (quadratic) one that down-speeding allows. In screw compressors, the reduction of friction in bearings and between driving and driven rotors is the key for mechanical efficiency increase. Friction depends on bearings accuracy and rotors accurate shaping; because of the principle of operation of screw compressors, contact during rotation cannot be avoided and contributes to bearings aging and surfaces damage at the rotor—rotor contact areas, thus increasing loss in mechanical
power. Rotational speed reduction certainly decreases the frictional losses but it is detrimental on the volumetric efficiency: its potential in screw machines is then well below the one in SVRC [16,17];

- $\eta_{fr}$: the prime mover is usually an electric motor, whose losses affect the package power requirements, because of the power absorption by the fan (1% package power) and internal losses to the motor (8%);

- $\eta_{org}$: both fan (in those compressors in which the oil is cooled by air) and pump (when oil is cooled by water) are responsible for energy absorption that contributes to the package power consumption. The power needed to pressurize the oil inside the machine should be still considered as power absorbed by an auxiliary and since in both SVRC and screw machines abundant quantity of oil is employed, its impact on machines performance deserves some attention. Even though the high oil density with respect to the air should make the power requested for oil pressurization negligible, this power can be still significantly decreased by reducing the oil circulation. In present machines, the oil contribution reaches 5–7%; the oil flow rate reduction would produce a proportional power absorption reduction, but sealing and friction should be taken under control. Mechanical efficiency is very close to one when water provides the oil cooling, while for air cooled mainly depends onto the radiator aerodynamic permeability.

So, while the fluid-dynamic losses strongly affect the overall performance in turbo-compressors, in screw and rotary vane machines the global efficiency mainly depends on the mechanical and electrical terms, with the mechanical one, strictly dependent on the compressor technology and currently representing one of the most interesting matter of development.

Spread reduction produces always a positive effect, even though it is in conflict, mainly in screw compressors, with flow rate delivery and leakages among vanes. Nevertheless, even though the improvements could be reached, premium machines have to be considered as those in which technological advancements in terms of mechanical and electrical efficiency have already reached an asymptotic value. Room for further improvement is still available and is currently being explored by means of a massive involvement of CFD approaches and deep experimental techniques, even though the thermodynamic side appears as the one offering the greatest margin for a growth in performances of present market compressors [19].

In Fig. 10, the improvement concerning the thermodynamic transformation during compression is presented. In the figure, the circles represent premium machines for each class. The lines stand for ideal and actual compressions: the former refer to the ideal thermodynamic transformation inside ideal compressors, i.e. the compression having all efficiency terms equal to one, whereas the second is the compression attainable in actual machines, thus the one affected by losses and characterized by an efficiency value lower than one. At any delivery pressure, the dual stage ideal adiabatic (30°C pinch point at the heat exchanger) (solid line, blank triangle) has the lowest specific consumption and the distance from the one stage ideal adiabatic (solid line, blank square) ranges from 0.5 to 1.0 kW/(m³/min), when the delivery pressure varies between 7.5 and 11.5 bar. As previously observed, the compression in premium machines is closely approximated by a one stage adiabatic (solid line, black square) with the efficiency lower than one (0.813 for fixed speed air cooled, 0.786 for variable speed air cooled, 0.829 for fixed speed water cooled, 0.817 for variable speed water cooled).

Fig. 11 reports a dual stage compression too (solid line, black triangle), with the same efficiency than the single stage one, i.e. under the assumption that the dual stage results from splitting the compression in two moments, with no technological upgrade of the machines involved. The differences between present best in class machines and this curve represent the improvement that can be

![Fig. 10. Premium machines — Improvement potential.](image-url)
reached today, simply splitting the overall compression in two stages (30°C pinch point at the exchanger). Something better could be expected if one considers that when split into two stages, thanks to the lower pressure ratio, all the efficiencies would have a benefit: as a rule of thumb, every 0.5 bar decrease in the delivery pressure leads to a 5% saving on the electric power consumption by the motor, not accounting for leaks and open blowing. It is worth observing that, dependently on the delivery pressure, such a dual stage is able to allow a 50–70% reduction of the distance between premium machines and the ideal single stage adiabatic: typical values for the saving in specific consumption are 0.60–1.0 kW/(m³/min) (fixed speed air cooled), 0.59–0.76 kW/(m³/min) (variable speed air cooled) and 0.70–0.93 kW/(m³/min) (variable speed water cooled) when the delivery pressure ranges from 8 to 11.5 bar. A further additional reduction in specific consumption can be achieved by injecting a fine oil spray within the air flow, as experienced in SVRC, thus reducing the air temperature during compression: 5% of the one stage adiabatic consumption is a good estimation of the achievable saving by adopting the oil injection technique (dashed line) [20].

A comparison between real compression, performed according to a one stage adiabatic, a two stage adiabatic and an isothermal, leads to Fig. 11, where specific consumption is reported for each compression technique, along with the reduction allowed by the adoption of a two stage adiabatic, expressed, for each compression ratio, as a percentage of the distance between the single stage adiabatic and the isothermal. Apart from the technical issues to deal with when approximating an isothermal compression, it has to be noted that yet with only one intermediate stage of cooling, the gain in specific power ranges between 37.9% and 42.5% dependently on the machine type and the compression ratio. It reaches its maximum in fixed speed water cooled machines at 11.5 bar, where about 42.5% distance between one stage adiabatic and isothermal is recovered. Comparable values characterize variable speed air cooled (42.4%), fixed speed air cooled and variable speed water cooled (42.2%).

Fig. 12 shows the most recent data on compressed air energy consumption in Europe, in terms of cumulative electricity consumption and specific consumption for PTL (present technological level) compressors, as a function of flow rate: most recent data on cumulative electricity consumption rank at around 45 TWh for fixed speed technology and 30 TWh for variable speed compressors [13].

When it comes to evaluate the actual reduction in the energy consumption related to the CAS, achievable through the adoption of the above mentioned measures, it is of utmost importance to consider that those measures can't be directly and effectively applied to all compressed air stations and that the outcome of such interventions is a function of a great variety of factors, e.g. the energy saving allowed by a specified measure can significantly differ, dependently on the machine size and/or the machine type, as well as the applicability and economic feasibility of a specified measure can vary according to the amount of hours per year of operation and the nature of the application.

For instance, it is well known that a reduction in the air temperature during compression leads to a reduced specific consumption and as a common understanding, this reduction is fixed in 1% compressor energy use each 3°C temperature reduction, with paybacks up to 5 years.

The level of acceptance of those interventions by the market, along with the availability of capitals to be allocated for energy saving in the present industrial scenario, play a fundamental role
an energy saving of 2 TWh has been estimated, by simply replacing low efficient compressors. In order to allow the evaluation of the saving potential associated to the dual stage option, Fig. 12 reports, along with the aggregated datum on the cumulative electricity consumption, the average specific power of present compressors in the flow range reported on the y-axis. When the compression is split into two stages, the potential reduction in specific power ranks at about 20%, by far more relevant than the 2% allowed by the simple machine replacement, with higher efficiency compressors. Nonetheless, in order to evaluate actual financial feasibility, it has to be remarked that, for both interventions, higher investment and installation costs have to be expected and such an increase has to be compared to the benefit associated to the electricity saving, which, in turn, calls for an in depth analysis of both the specific industrial environment the measures apply to (i.e. constraints on the payback period, hours of operation, plant configuration) and the specific socio-economic context under investigation (i.e. cost of electricity, inflation rate). For all the reasons above, any attempt to provide an isolated datum with a general validity is not only out of the scope of the present work, but also devoid of scientific meaning. Anyway, when the focus is on the air cooling, an average value for the attainable benefit is 1400 US$/year saved and a 5 months payback time [10].

It is worth observing that the improvements discussed here make reference to an average technological level for present compressors, with no distinction between best and worst in class machines. As a consequence, when energy saving/recovery measures were implemented in all market compressors, including those with efficiencies lower than premium machines, a huger saving would be possible. This perspective is the main reason why research is being carried out on a great variety of possible interventions and even the slightest margin for improvement is addressed as extremely appealing in a technology that is already at its technological asymptote.

5. Conclusions

Industry, among all the other sectors, is currently responsible for the biggest electricity consumption, with up to 15–20% of it coming from the compressed air sector: this percentage means, today, an electricity consumption for compressed air production between 75 and 80 TWh/y. Since the compression section alone accounts for 10–20% of the overall CAS consumption, the compressor is being addressed as the component offering the greatest room for improvement. This calls for an in-depth investigation of both compressors technology and market. Such an analysis has been performed in the present study: starting from the present technological scenario, different energy saving/recovery options have been analyzed and their energy dimension evaluated.

The paper provides an overview of present technological scenario, based on data available on compressors performances from CAGI and Pneurop, for the US and EU contexts respectively. A great dispersion in terms of specific energy consumption characterizes present compressors technology: consequently, it is hard to make reference to an average technological level when the potential improvement allowed by different energy saving/recovery options is under investigation: small and medium size machines appear to have the greatest variability between actual and average value, with up to 50% shift between actual and average specific consumption. A mean value for such a difference in big size compressors is 3%: this means that in big size compressors a closer control on performances, along with higher characteristic efficiencies (80–82% in big size, 76–78% in medium size, 70–72% in small size), shall be expected. The baseline consumption can be identified for different machines type. Average in class machines always show a specific consumption between 7.3 and 8.4 kW/(m³/min) in fixed speed air cooled compressors, 6–7.6 kW/(m³/min) in variable speed air cooled compressors, 7.3–7.9 kW/(m³/min) in

Fig. 12. Cumulative electricity consumption – Present technological level vs saving scenario.
fixed speed water cooled compressors and 6.4–7.5 kW/(m³/min) in variable speed air cooled compressors, when the compression ratio varies between the most common values in the industrial practice (8 and 12).

A baseline efficiency is derived, in the range of flow rates of interest for industrial applications, for best and worst in class machines (i.e. those machines characterized by the lowest and highest specific consumption), as well as for average in class machines: this means that a lower and an upper limit for compressors efficiency can be found, along with a baseline efficiency that refers to the mean compressor technology for each machine type. It is worth observing that, when the flow rate shifts from 0.1 to 100 m³/min, in both fixed speed (best in class baseline efficiency: 65–85%, worst in class baseline efficiency: 45–70%) and variable speed (best in class baseline efficiency: 55–85%, worst in class baseline efficiency: 40–65%) families, sliding vane rotary compressors have characteristic efficiencies between the average (50–75% in fixed speed, 45–75% in variable speed) and best in class baseline. This suggests that sliding vane rotary compressors can be considered the reference for present technology.

Data on compressors, available from different companies and data derived from European sales, as presented by Pneurop, demonstrate that the awareness in energy saving still doesn’t receive the attention it deserves. This may appear in contrast with the potential energy benefit associated to energy saving/recovery options (e.g. 10–15% reduction in specific consumption achieved in premium machines by splitting the compression in two stages), but the feasibility of an intervention can’t be evaluated only on the energy gain it allows. Other factors, such as the impact on the financial management of the plant must be taken into account.

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Nomenclature

$\eta$ efficiency, (–)
$\Delta$ variation, (–)
$h$ specific enthalpy, (J/kg)
$R$ individual gas constant, (J/(kg·K))
$T$ absolute temperature, (K)
$c_p$ specific heat at constant pressure, (J/(kg·K))
$k$ adiabatic isentropic exponent, (–)
$p$ polytropic exponent, (–)
$\beta$ compression ratio, (–)
$\rho$ air density, (kg/m³)
$q_s$ specific power consumption, (KW/(m³/min))
$m$ volumetric flow rate, (m³/min)
$R$ Pearson’s product moment correlation coefficient, (–)
$A$, $a$ regression model parameters, (–)

Subscripts and superscripts

glob. global
gls. adiabatic isentropic
vol. volumetric
mech. mechanical
org. organic
el. electrical
real real
inl inlet
msr measured
* reported to reference pressure levels
** interpolating