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ABSTRACT

The Compressed Air Sector (CAS) is responsible for a relevant part of energy consumption, accounting for a mean 10% of the world-wide electricity needs. This ensures about the importance of the CAS issue when sustainability, in terms of energy saving and CO_2 emissions reduction, is in question. Since the compressors alone account for a mean 15% of the industry overall electricity consumption, it appears vital to pay attention towards machine performances.

The paper deals with compressor technology and it discusses the energy consumptions, on the basis of a comprehensive analysis of data for existing machines, mainly provided by the Compressed Air and Gas Institute (CAGI) for the US scenario, and PNEUROP, on the European compressors market. Data referring to different machine technologies, were processed to obtain consistency with fixed reference pressure levels and organized as a function of main operating parameters. Saving directions for different compressor types, screws & rotary vanes, have been analyzed. Main factors affecting overall efficiency have been split and all different efficiency terms (adiabatic, volumetric, mechanical, electrical, organic) considered separately. This has allowed a term-by-term evaluation of both the margin for improvement and the impact of each term on the "step change" in energy saving, leading to the evaluation of how efforts in the CAS contribute to the 20-20-20 policy emissions reduction targets. If a negligible growth in efficiency is achievable by further increase of volumetric and mechanical terms (few tenths percent), wide margins for improvement come from an upgrade of the transformation, through the adoption of a dual-stage intercooled compression. Its potential has been compared to that of an internal cooling strategy, with a fine oil spray injected within the flow: if the former solution requires the use of dedicated cooling devices between stages, the latter has its main drawback in the presence of cooling medium vapor phase within the flow, leading to a growth in compression work. Since the heat provided by oil cooling is available at a temperature (70-90°C range) that allows the conversion into mechanical energy by means of an Organic Rankine Cycle (efficiency range 8-10%) and considering that the thermal power from the oil and the mechanical power absorbed by the compressor are of the same order of magnitude, energy recovery is interesting as well. This measure, coupled with that of a multi-stage compression, has the potential to overcome the globally shared goals on energy and carbon saving.

1. INTRODUCTION

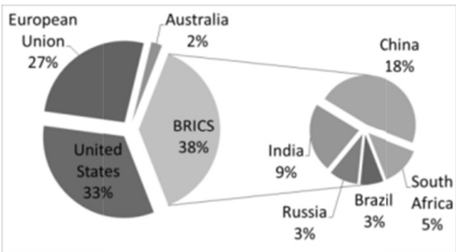
The global instability experienced in the recent past and still affecting every Country in the World suggests the complexity of the energy issue. Furthermore, the difficulty in depicting the present energy scenario represents the reason for great uncertainty in the near future. Those criticalities adversely influence the delineation of a shared energy policy and contribute to vanish all efforts oriented at the definition of a globally accepted energy planning.

Since the paper focuses on CAS, whose energy demand mainly concentrates on electric energy, the overall electricity consumption for different Countries, rather than the one including the thermal contribution, is worth to be considered. Data of interest, provided by the International Energy Outlook 2011 (IEO 2011) are reported in Table 1 and Figure 1 and the electricity need relative share among sectors is specified. The electric energy demand for the industrial sector covers a great part of the cumulative electricity demand, both in developed and developing Countries. Among them, great attention has been paid to Brazil, Russia, India, China and South Africa, commonly designated as BRICS Countries, expected to be the most rapidly growing economies in the near future.

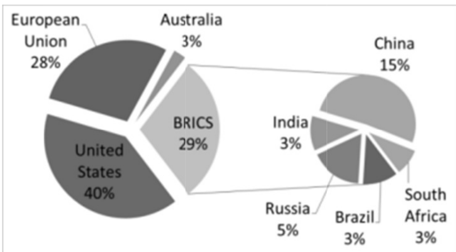
Table 1: Electricity demand by sector and industry share (%)

	Electric Energy Demand by Sector (TWh)					Share (%)
Country	**Residential**	**Commercial**	**Industrial**	**Transportation**	**Total**	**Industrial**
United States	1348	1407	1026	11	3792	27.1
European Union	1084	997	1319	103	3503	37.5
Australia	88	88	117	3	296	40.0
Brazil	117	117	205	3	442	46.7
Russia	147	176	528	85	936	56.3
India	352	117	440	29	938	46.9
China	733	528	3400	56	4717	72.0
South Africa	205	88	322	3	618	52.4
World	4074	3517	7357	293	15241	48.3

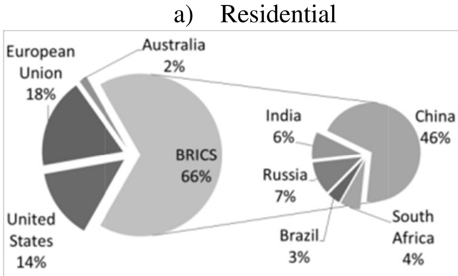
Percentages over 40% can be observed, with China showing the greatest unbalance among sectors: more than 70% electricity demand comes from industry, as a consequence of the growth in the last decades. This datum, along with future growth perspectives, suggests where energy efficiency measures can be more effective in meeting restrictions in carbon emissions. In these contexts, energy saving should become a crucial technological action and a driver for the overall industrial sector development. Fossil fuels (oil, gas and coal) dominate on the other energy sources (85%), with a negligible contribution of hydro, nuclear and other renewables (15%) (Cipollone *et al.*, 2012). The result of this fossil dependence is the CO_2 amount into the atmosphere growing at a rate that, considered the present concentration of 396.16 ppm (Dlugokencky and Tans, 2013), will soon frustrate the attempt to meet the limit of 450 ppm by 2035, considered by the Intergovernmental Panel on Climate Change (IPCC), a safeguard limit to avoid irreversible environmental and socio economic problems. This unbalance between fossil and sustainable development suggests that, at least in the near future, energy efficiency and energy recovery will be, most likely, the only measures left to take.



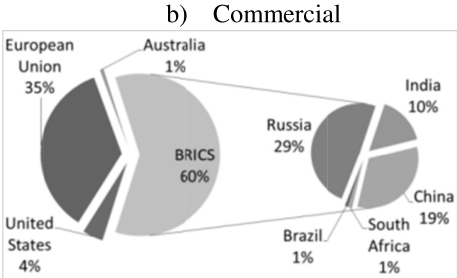
a) Residential



b) Commercial



c) Industrial



d) Transportation

Figure 1: Electricity by sectors

The percentages in Table 1, yet critical for the present scenario, depict an even worse picture, when matched to projections for future energy consumption, assuming 2020 and 2035 time horizon as a reference: they fix a proportion between electricity demand by sector, that, given the growing trend in electric energy consumption, will result in the use of a greater amount of energy and consequently a greater amount of CO_2 into the atmosphere. The risk to exceed limitations is then higher when those percentages are expected to raise from the present values, because of a higher presence of developing Countries industry-based economies, expected in the near future. Energy sector (47%) industry (18%) and transports (22%) are the main responsible for the CO_2 increase in the atmosphere today, resulting in more than 85% of global emissions, according to data presented in the World Energy Outlook (WEO) Special Report, by the International Energy Agency. A relevant part of this share comes from electricity, since the conversion efficiency has to be considered to evaluate the related emissions. Absolute values show that electricity use in industry is greater in EU (37.5%) with respect to US (27.1%), while for BRICS Countries it ranks at 32.2% of the overall electricity consumption, with China alone representing 69.5% of this share.

2. ENERGY CONSUMPTION IN INDUSTRIAL COMPRESSED AIR SYSTEMS

From many independent studies, compressed air production in developed Countries accounts for a mean value around 15% of the overall electricity consumption in the industrial sector, with ceramics and glass industry, automotive and aerospace sectors and petroleum refining processes being the biggest contributors to this percentage. By adding all the other "compressed air needs", i.e. the commercial and residential markets, the consumption grows to 20% of the industrial electricity needs (Cipollone *et al.*, 2012), making the CAS an interesting sector of application of technological improvement, when CO_2 reduction is considered as a major future concern. Energy saving in the CAS is possible by adopting a huge variety of interventions. According to a commonly accepted approach, the distinction between two families is possible, depending on where, with respect to the compression, the measures apply: upstream (preceding the compression, e.g. compressor choice, use of high efficiency motors, definition of adequate control systems) and downstream (following the compression, e.g. energy recovery/saving, heat re-use, leakages). Compressor technology and operation accounts for a mean consumption share close to 10-20%, while pressure losses in pipes, leakages and an inappropriate use of compressed air account for a greater share. So, machines more oriented to reduce energy consumption and most preferably, oriented to reduce consumption in flow rate modulation (matching between demand and offer of compressed air) should be preferred (Radgen and Blaustein, 2001). Even if downstream offer greater room for improvement than upstream measures, the interest towards the compression transformation is a key factor and in the following it is investigated on different machines.

Being the energy aspect crucial, an analysis of the energy consumption of existing machines, mainly screw type, has been done. Interesting source of data has been the CAGI, where data sheets of different manufacturers' compressors are available. A clear overview of the energetic performances of present compressor technology came out from the processing of thousands of data sheets. Even if available in a standardized form and with an unified structure, data required a deep and time consuming treatment, in order to avoid inconsistencies in comparing performances. Main reason for this, is the difference between measured pressure levels and those considered as references: to allow a comparison between machines at the same delivering pressure, the following procedure has been applied:

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% of the reference value assumed, data are processed and modified in order to refer 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. Energy consumption can then be referred to common pressure reference values, by observing that the overall compressor efficiency can be seen as:

$$\eta_{glob.} = \eta_{ad.is.} \cdot \eta_{vol.} \cdot \eta_{mech.} \cdot \eta_{el.} \cdot \eta_{org.} \tag{1}$$

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:

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

$$\eta_{ad.is.} = \frac{\Delta h_{ad.is.}}{\Delta h_{real}} = \frac{\tilde{R} \cdot T_{inl} \cdot \left[\beta^{\frac{k-1}{k}} - 1\right]}{c_p \cdot T_{inl} \cdot \left[\beta^{\frac{p-1}{p}} - 1\right]} \quad (3)$$

$$\frac{d\eta_{ad.is.}}{d\beta} = const(k, \tilde{R}, T_{inl}, c_p, p) \cdot \beta^{-\frac{1}{k}} \quad (4)$$

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

$$\frac{\Delta q_s}{q_s^{msr}} = \frac{q_s^* - q_s^{msr}}{q_s^{msr}} = -\frac{\Delta\eta_{glob.}}{\eta_{glob.}} = -\frac{\beta^* - \beta^{msr}}{(\beta^{msr})^{\frac{1}{k}}} \quad (5)$$

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

$$q_s{}^* = \left(1 + \frac{\Delta q_s}{q_s^{msr}}\right) \cdot q_s{}^{msr} \quad (6)$$

while, for the mass flow rate:

$$m^* = m^{msr} \quad (7)$$

It has to be noted that such an approach applies in case of small difference between measured and reference pressures, so that the assumption on the nature of the transformation will not affect the calculations[1]. Applying the correction procedure, almost 25% of the original data has been excluded, since the distance between measured and reference pressure was outside the 5% range. Figure 2 shows the performances of the existing compressors on the market, when all the data are recalculated at reference pressures: for sake of privacy, the data are presented anonymously, eliminating the manufacturer's name. Data have been organized as a function of pressure delivered at rated (usual design operation) and zero (frequent in case of load/unload control) flow and as a function of oil cooling technique (air or water). The following considerations apply:

a) specific power decreases in general with respect to flow rate, so bigger machines are more efficient: for air cooling, at 5 m^3/min, 20 m^3/min and 50 m^3/min flow rates, the mean specific power is 7.7 kW/(m^3/min), 7.0 kW/(m^3/min) and 6.5 kW/(m^3/min), respectively; for the same flow rates, water cooled machines show specific power consumption of about 7.0 kW/(m^3/min), 6.7 kW/(m^3/min) and 6.5 kW/(m^3/min), respectively. This trend is particularly evident for air cooled machines and for higher pressures delivered;

b) machines, in which the oil is cooled by water, show higher efficiencies than those in which oil is cooled by air, since a pump (circulating an incompressible fluid) always has a negligible impact on package power demand, with respect to a fan;

c) for a delivered pressure equal to 8 bar, air cooled machines have a significant technological scatter (among manufacturers): for a rated flow rate of 10 m^3/min, the mean value is 6.8 kW/(m^3/min), with a scatter of

[1] In reality, rotary volumetric machines show an isochoric phase immediately after the valve opening, as pressure inside the vane is different from line pressure, quite far from the assumption of a polytropic.

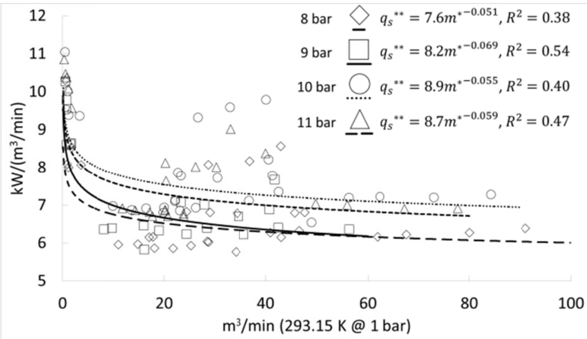
1.2 kW/(m^3/min); for these conditions, the compressors must be chosen properly and huge saving in energy absorption can be achieved. When flow rate increases specific consumption shows an asymptotic trend, with lower values for both mean specific consumption and scatter (for a flow rate of 50 m^3/min, the mean value is about 6.2 kW/(m^3/min) and the scatter is 0.5 kW/(m^3/min)). Higher flow rates present a scatter close to zero. For water cooled machines, the scatter is an order of magnitude lower than the corresponding in air cooled family, resulting in a negligible effect on performances;

d) for higher delivered pressures, the differences among air and water cooled increase, inviting to water cooled version. For a delivered pressure of 9 bar, the scatter further reduces, leading to a compressors population that is tightly distributed around the performances mean value: for a flow rate of 30 m^3/min, the mean value is around 6.6 kW/(m^3/min), with a scatter of about 0.4 kW/(m^3/min). The previous considerations keep their validity when referred to water cooled compressors, as demonstrated by the negligible scatter that characterizes performances yet for a 20 m^3/min flow rate;

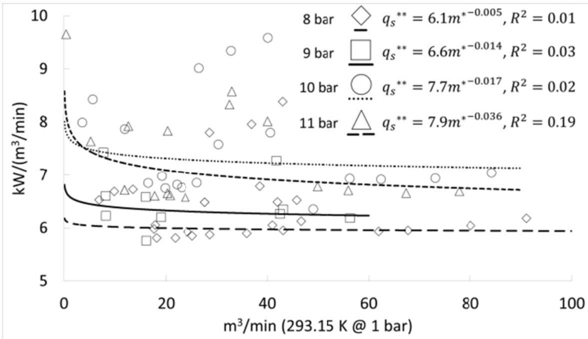
e) Figure 2 shows that the technology is not aligned to best standards: a reason for this is the massive market presence of machines, designed years ago, with no evidence of any upgrading effort on them.

As known, the coefficient of determination R^2 in Equation (8) is a statistical measure of how well the regression model (in this case: $q_s^{**} = A \cdot m^{*-a}$) approximates energetic performances of existing machines (real data points): R^2 low values demonstrate what previously discussed about how technology meets the best standards expectations.

$$R^2 = \left(\frac{\sum(m^* - m^{**})(q_s^* - q_s^{**})}{\sqrt{\sum(m^* - m^{**})^2 \sum(q_s^* - q_s^{**})^2}} \right)^2 \tag{8}$$



a) Fixed speed, Air cooled



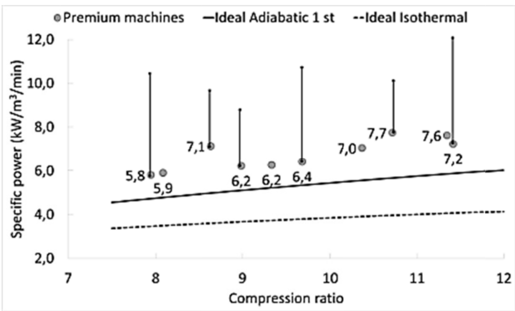
b) Fixed speed, Water cooled

Figure 2: Market compressors performances (volumetric flow rate evaluated at 1 bar, 293.15 K)

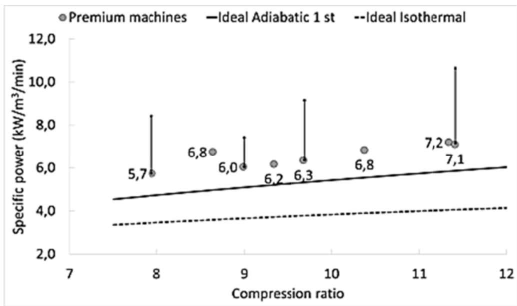
3. POTENTIAL ENERGY SAVING

The potential energy saving during compression is given by the distance from the electric specific consumption of real machines and that referred to the ideal datum, calculated by considering ideal compressions inside ideal machines. To set up a minimum energy saving, only premium machines, i.e. those with minimum specific energy requirement for any compression ratio, are considered. The higher spread (i.e. the lower R^2) among all the others, assures that in mean situations, higher energy savings are quite probable. Usually, premium values are referred to big size compressors, considering what already observed in Figure 2, about the asymptotic trend assumed by performances when the flow rate increases. Figure 3 reports the specific work, estimated for two ideal transformations: an adiabatic isentropic and an isothermal. While the former represents the best approximation for real compression, the latter represents the reference transformation with the minimum work required. In order to evaluate the potential savings, each term contributing to global efficiency in Equation (1), deserves some attention:

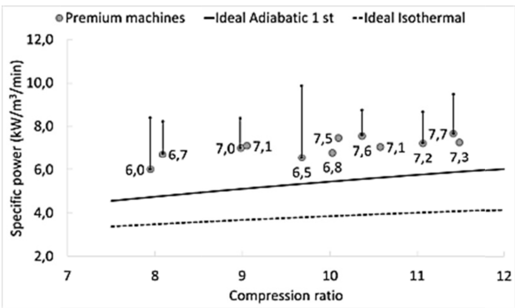
- $\eta_{ad.is.}$: it accounts for the thermodynamic transformation during compression and it is expressed by defining a reference transformation. In present machines, both screw and rotary vane, one stage of compression is considered and due to the compression technology, the adiabatic isentropic is the reference transformation. The efficiency is reported in Equation (3), in which Δh_{real} can be evaluated according to the indicated cycle, that differs from the ideal one in intake and exhaust discharge phases (Cipollone *et al.*, 2005). This is mainly due to unsteady phenomena during the discharge, since the pressure inside the cells does not meet the pressure line at the vane opening. A further loss is due to vane filling, when pressure differs from the ambient one because of the residual expansion after exhaust port closing. Experimental data show the residual importance of these phenomena and preliminarily allow to fix this efficiency to one. From a thermodynamic point of view, an isothermal transformation would introduce a relevant reduction in specific power requirement (Figure 3), but it would require a different compression technology. Recent advancements in rotary vane compressors aim to decrease compression work, by cooling the air during compression, through the injection of a fine oil spray in the vanes: in such case, the growth in compression work, due to the presence of the oil in its vapor phase within the flow is considered as a major concern (Valenti *et al.*, 2013). An external cooling stage, still appears to be the most effective way to reduce energy consumption and to produce the highest gain in performances;



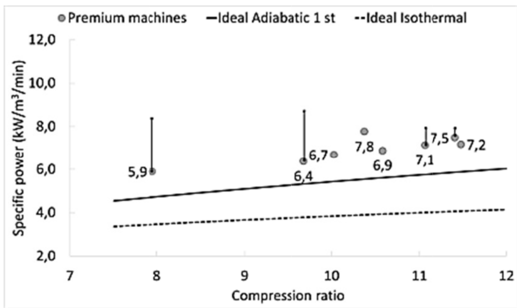
a) Fixed speed, Air cooled



b) Fixed speed, Water cooled



c) Variable speed, Air cooled



d) Variable speed, Water cooled

Figure 3: Premium machines vs ideal adiabatic and isothermal (vertical sticks = best vs worst in class spread)

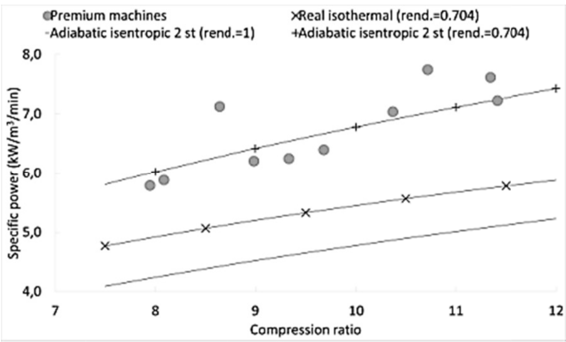
- $\eta_{vol.}$: it takes into account the fact that air compressed is greater than air delivered. This difference is due to the amount of air standing within the compressor, once the discharge phase is completed, also because of the unsteadinesses following the port opening. In premium machines, the comparison between ideal and real flow rate demonstrates that this efficiency is high and preliminarily it can be approximated to one. It is known that, according to compressor type, margins for improvement are still appreciated in rotary vane - mainly reducing air leaks on the vertical planes, where sealing effect is not guaranteed – and in screw machines - on the so called "blow hole line";

- $\eta_{mech.}$: it gives the greatest contribution to efficiency shift from the unit, accounting for friction between parts in relative motion. Room for improvement is still appreciated in both rotary vane (i.e. reduction in friction between blade tip and slot, rotor and stator and even if negligible with respect to the previous contributions, friction in bushings) (Cipollone et al., 2005) and screw machines (i.e. reduction in friction between rotor and stator, driving and driven rotors, motion transmission and speed reduction gears and friction in bearings) (Cipollone et al., 2012);

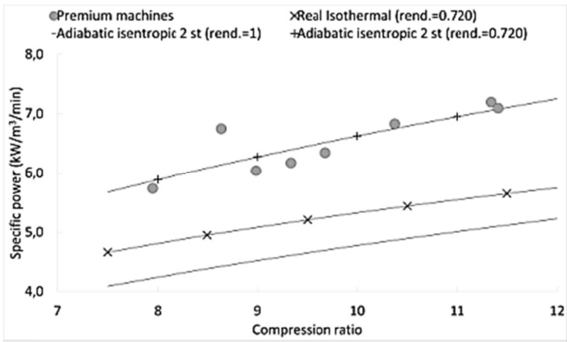
- $\eta_{el.}$: prime mover is usually an electric motor. Its inefficiencies affect the package power requirements;

- $\eta_{org.}$: it takes into account the power absorption by auxiliaries, like the fan that serves the radiator, when oil is cooled by air, or the power absorbed by the pump. This efficiency is very close to one when water provides the oil cooling, while for air cooled mainly depends onto the radiator aeraulic 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 mechanical and electrical term. Premium machines, operating at fixed speed, are very well fitted by an adiabatic transformation, with an overall global efficiency close to 0.70 if oil is cooled by air and 0.72 if oil is cooled by water (Figure 4a and 4b). The shift between the adiabatic which interpolates the best machines and the ideal data stands within a 1.3-1.6 kW/(m^3/min) for the former, and between 1-1.1 kW/(m^3/min), for the latter, with a compression ratio varying from 8 to 12. The difference between an ideal isothermal transformation and the adiabatic one is between 1.3 at 8 bar and 1.9 at 12 bar.



a) Fixed speed, Air cooled



b) Fixed speed, Water cooled

Figure 4: Dual stage (pinch point temperature difference=30°C) vs real isothermal transformation

Therefore, the improvement of both the mechanical and electrical efficiency appears an effective direction, even if the mechanical one represents a much more 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 the ones where technological effects in terms of mechanical and electrical efficiency already reached an asymptotic value. A significant margin for improvement, therefore, should be searched towards the thermodynamic direction. In Figure 5, the improvement concerning the thermodynamic transformation during compression is presented. In the bottom part of the figure, three lines are shown. They represent the specific power related to ideal transformations (i.e. $\eta_{glob.} = 1$): an isothermal (lower), an intercooled two stage adiabatic (middle) and a one stage adiabatic (upper). When a two stage ideal adiabatic transformation is considered, a 30°C pinch point temperature at the heat exchanger is fixed. It is evident how an ideal intercooled two stage compressor recovers half a distance from the ideal isothermal which is the best reference transformation. In the top part of the figure, the curves refer to the best achievable $\eta_{mech.} \cdot \eta_{el.}$: they represent an intercooled two stage adiabatic compression and an adiabatic one stage. The latter provides the best fitting for the best in class machines performances.

The performance that seems to be achievable adopting an intercooled two stage compression is very promising. A quick comparison with best in class machines allows to quantify the gain in performances in percentages between 10 and 15%, with delivered pressure swinging from 6 and 15 bar. The European Community 20% reduction of energy consumption is quite far from being disclosed. It is important to remark that these improvements underestimate the maximum achievable saving, if one considers that splitting in two separate stages the compression, the mechanical efficiency tends to increase. An interesting additional option is shown in the top part of Figure 5: the dotted line represents a sliding vane rotary compressor performances, when the oil is injected through a fine spray (Stosic *et al.*, 2003). A mean 5% of improvement can be estimated.

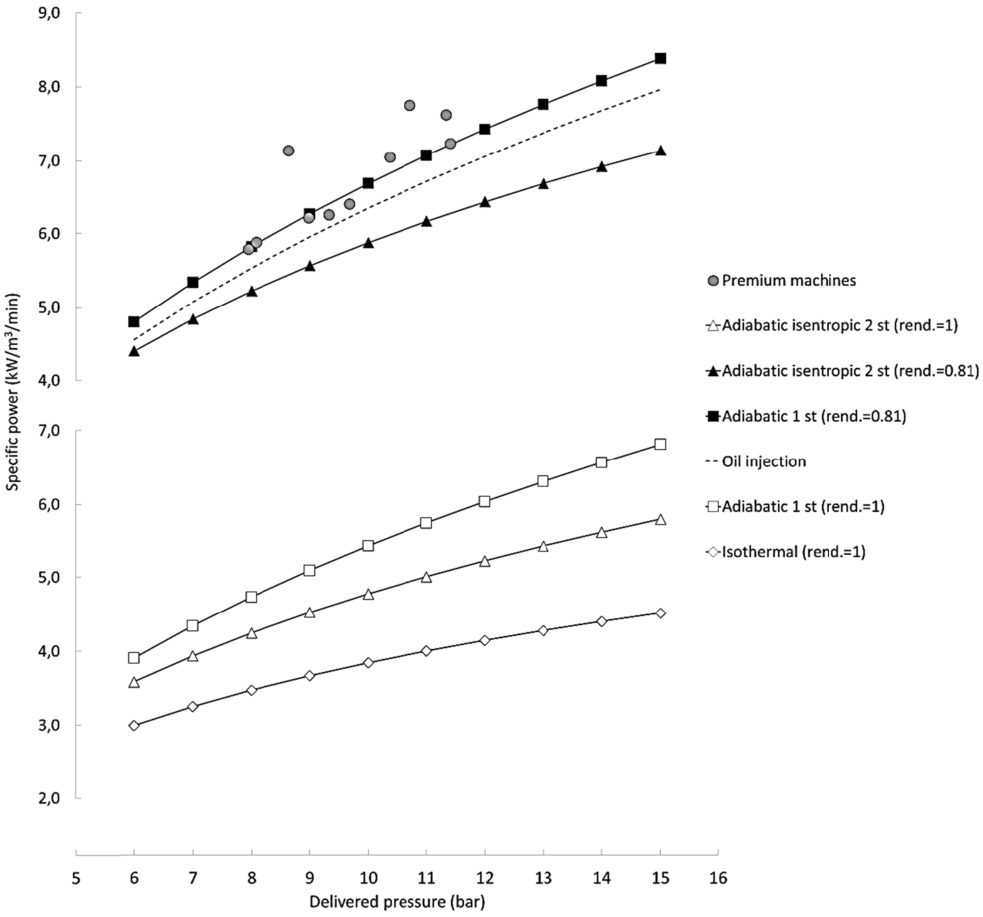


Figure 5: Fixed speed, Air cooled (approx. efficiency=0.81)

Figure 6 shows the most recent data on compressed air energy consumption, as a function of flow rate and fixed and variable speed technology as parameter: they are equal to 40 TWh and 20 TWh, respectively (PNEUROP).

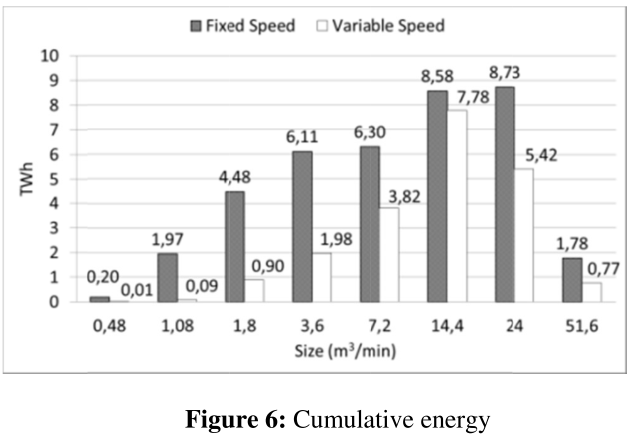


Figure 6: Cumulative energy

Considering a mean penetration market factor equal to 35%, an overall yearly saving of 2 TWh could be reached. This is an interesting issue, when compared with all the other measures.

4. CONCLUSIONS

Energy and carbon saving, along with energy generation from renewable sources, are expected to be the most effective ways to deal with sustainability commitments of all the Countries in the World, with the efforts mostly expected in sectors responsible for high energy consumption, such as industry. Inside it, it appears vital to pay attention towards electricity needs. This is the main reason why the compressed air sector is being re-thought as an area offering great opportunities for improvements, being responsible for an electric energy consumption equal up to 10% of the world-wide overall electricity needs. Given the electrical energy share with respect to overall energy needs, compressed air sector energy requirements can be directly related to the overall energy consumption and the saving potential it offers can be compared with other energy efficiency measures (e.g. renewable energy in electricity generation).

In the article, an overview of present compressor technology is given, according to the data sheets from CAGI. These data have been processed in order to obtain consistency with fixed reference pressure levels. With reference to the premium machines, considered at different compression ratios, an overall efficiency value, able to closely fit real performances, has been derived equal to 0.81. An analysis of the various terms of the overall compressor efficiency shows that such a number represents a consistent estimation of mechanical and electrical terms contribution. Considering that both electric and mechanical technology have reached a good level of development (even if improvements still remain possible), a great potential saving is possible on the thermodynamic side. Present technology allows to consider a mean saving of 15% for an intercooled compressor. Correlating this issue with the present compressor market distribution, the most suitable power range in which the introduction of a multi stage compression would give the greatest results, in terms of energy saving, is 15-20 m^3/min.

The heat provided by the oil cooling is available at a temperature within the range 70-90°C, which allows a recovery into mechanical energy, by means of an Organic Rankine Cycle (ORC) based plant. A recent study demonstrates that the overall conversion efficiency reaches 8-10%, which is a very interesting recovery, considering that the thermal power of the oil is of the same order of magnitude of the mechanical power absorbed by the compressor. This contribution, along with that of a multistage intercooled strategy, leads to conclude that energy saving and energy recovery have the potential to overcome the goals expressed by EC and World main Countries, in terms of energy and carbon saving.

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NOMENCLATURE

η	efficiency	(-)
Δ	variation	(-)
h	specific enthalpy	(J/kg)
$\tilde{R}$	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	(-)
β	compression ratio	(-)
q_s	specific power consumption	(kW/(m^3/min))
m	volumetric flow rate	(m^3/min)
R	Pearson's product moment correlation coefficient	(-)
A, a	regression model parameters	(-)

Subscripts and Superscripts

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

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