
Energy Saving in Sliding Vane Rotary Compressors

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Abstract

In industrial contexts, electrical energy for compressed air represents an important share of the global electricity consumption: this figure accounts for 4-5 % of the total. Among the existing compressor technologies, rotary volumetric machines proved to be more suitable than other types (dynamic, reciprocating, etc.) in terms of pressure and flow rate delivered. Even though not as widespread as screw machines, but thanks to the technological improvements made in the last two decades, Sliding Vane Rotary Compressors are characterized by premium specific energy consumptions and demonstrate an unforeseen potential in terms of energy saving due to some intrinsic features specifically related to these machines. The current research focuses on an innovative oil injection technology that is not only able to fulfill the sealing and lubrication requirements but also to cool the air during the compression phase. A comparison between the mathematical model of the new oil injection technology and the experimental p-V diagrams measured through a set of piezoelectric transducers is shown. The compression work reduction, predicted in the model and further measured at the shaft and observed in the indicator diagrams, gives a strong consistency to the injection technology.

Keywords

sliding vane rotary compressor; positive displacement compressor; pressure swirl nozzle; oil spray injection; indicator diagram

1. Introduction

Energy saving will be one of the most important drivers in the next years for many energy consuming users. Energy saving is recognized as equivalent to an energy source but with a much lower cost per unit energy (saved rather than produced).

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Compressed air is mainly produced by electrical energy. It is a non-replaceable utility and, with respect to the global electric energy consumptions, it is responsible for a 4-5 % share (1; 2) of the global industrial electricity consumption. Typical industrial flow rate and pressure requirements make rotary volumetric machines the most widespread of compressor technology. Among them, screw compressors are the most diffused and characterized by a proven and reliable technology. Sliding Vane Rotary Compressors (SVRC) come second. However, in the Authors' opinion, these machines are not very well known in terms of energy saving potential. In the past two decades, they have been subjected to massive technological improvements making them today's most efficient single-stage compressors in the industry.

In recent years, some of the Authors deeply focused their attention on the physics behind the behavior of such types of machines: main processes such as air intake, compression and exhaust, oil circulation and injection, blade motion inside the rotor slots and friction phenomena have been investigated (3; 4; 5). This led to the development of a virtual simulation platform that, once validated with experimental data, became a tool to improve performances and conceive new design solutions (6). At the same time, an innovative experimental methodology has been developed to measure the pressure inside the compressor cells, when intake, compression and exhaust occur. The further processing of p-V data allowed the identification of the friction coefficient (6; 7), a better understanding of the cooling action done by the oil during injection (8), and suggested geometrical re-shaping of the machine for optimization purposes (7). Furthermore, thanks to this experimental activity, the specific energy consumption for some models of SVRC reached the lowest values in the industrial compressor global market: values around $5.4\text{-}5.8 \text{ kW}/\text{m}^3/\text{min}$ - ISO 1217 conditions (p intake = 1 bar; T intake = 293.15 K) - at 7 bar as pressure delivered.

The reconstruction of the indicator diagram allowed to state that the oil injected with current technology (simple calibrated holes) is unable to effectively cool the air during the compression phase (8). This is due to some thermophysical properties of the oil itself (such as the molecular diffusion coefficient, saturation curve, etc.) but also from the mean drop dimensions produced by the jet: indeed, big drops are hardly heated or evaporate while they travel inside the cell. Moreover, the bigger the droplets the higher their inertia is downstream of the injector orifice. Hence, all the oil injected suddenly impinges on the metallic walls of the cell (rotor and blade surfaces) and any heat exchange with the air being compressed is prevented (8). If any oil evaporation took place, the effects on the compression work would be twofold but opposing: on the one hand a strong air cooling would lead to an isothermal compression so decreasing the input work; on the other hand, the compression of the oil vapours would require more energy than the pressurization in the liquid phase so increasing the specific work. Therefore, enough care must be taken in order to have the benefits of the air cooling and to avoid the work increase due to the compression of the oil in the vapour phase. In order to identify the optimal set of operating parameters for the spray injection system (droplet dimensions, injection pressure, nozzle type, etc.) some of the Authors recently developed a complex comprehensive oil injection model that describes the phenomena occurring during the jet breakup, particle formation, droplets dynamics, heat transfer between oil and air, oil heating and vaporization, re-condensation due to pressure increase during compression, oil puddle formation due to the interaction between the oil spray and the metallic surfaces of the compressor cells. These results were simulated using an existing mid-size industrial SVRC (22 kW at 1500 RPM and 8.5 bar as absolute delivery pressure).

In this paper the Authors present an experimental activity on the same compressor in which proper pressure swirled injectors replaced the conventional injection rail. The injectors were specifically designed for this application and optionally fed by an external pump in order to test higher pressures with respect to those produced by the machine. Pressure increase was required to vary continuously the dimension of the droplets. p-V data was recorded as well as the mechanical shaft power.

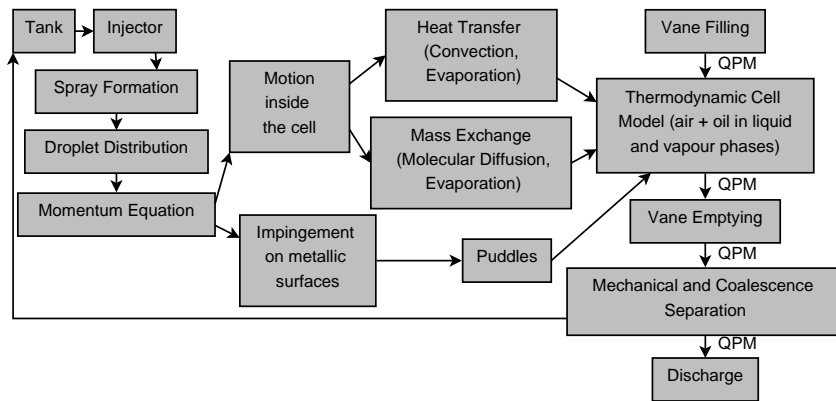


Fig. 1. Interactions between the SVRC model and the one related to the spray injection system

2. Oil injection modeling inside the vane

Some of the Authors have already developed a comprehensive, physically consistent, model which describes the effects of an oil spray produced by a pressure swirl nozzle on a SVRC cell during the compression phase (8). The spray injection module has been later embedded into a previous model which predicts pressure and temperature inside the cell and evaluates the performances of a SVRC subdividing the physical behavior in a set of processes that interact each other (3; 8). Briefly, it considers:

1. Vane filling and emptying through intake and exhaust ports suitably described in order to closely match their real shape. The mass transfers rely on a 1D unsteady formulation named QPM (Quasi Propagatory Model): it considers the transients occurring during the suction and, mainly, discharge processes taking into account the inertial, capacitive and resistive features of the flow (7);
2. Pressure and temperature inside the cell during the compression phase by means of the energy conservation equation in a lumped parameter form. Energy exchanges between air and oil are considered as well as the heat transfer through the stator surfaces.

The oil injection model is different according to the technology adopted. The conventional solution is represented by a series of calibrated holes fed by a common rail in which pressurized oil is supplied (after separation from the compressed air at the SVRC discharge and subsequent cooling). The resulting jets do not produce any cooling effect on the air during compression (8). A technological breakthrough happened using pressure swirled injectors which demonstrated the capability to cool the air and reduce compression work. The spray injection model predicts:

1. The break up distance from the nozzle orifice, from which the oil jet starts to atomize. The jet is subdivided in several portions and for each of them, using a Rosin-Rammler drop size density distribution, the number of particles for each class of dimensions is calculated. The initial speed of all the particles is defined through semi-empirical correlations related to the oil flow rate injected while the direction of propagation for each class is randomly chosen inside the experimentally observed spray cone;
2. The droplet trajectories as a result of the solution of momentum equation in which aerodynamic (drag and shear lift), inertial (virtual mass and Basset history), volume (gravity and buoyancy), fictitious (Centrifugal and Coriolis) and pressure forces are considered in order to evaluate the droplet motion;
3. Oil puddles that occur because of the impingement of the jets onto the metallic surfaces of the cell;
4. Heat transfer between oil droplets and air due to forced convection: oil heating and air cooling results until the saturation temperature is reached. From this point on, oil vaporizes and the latent heat of vaporization is exchanged proportionally to the oil mass evaporated;



Fig. 2. Images taken with a high speed camera of two different types of nozzles (left: full-cone large angle; right: full-cone narrow angle) spraying oil at $80\text{ }^{\circ}\text{C}$ and 6 bar_g into an ambient reservoir

5. Oil mass diffusion during the droplets motion because of the molecular diffusivity of the oil in air: this phenomenon modifies the droplets mass and their momentum;

The interactions between cell model and oil injection is represented in the block diagram of Figure 1: the basic concept is that the heat exchange between oil droplets and air realizes an internal inter-refrigeration for the compression process leading to a lower pressure at the discharge so reducing the amount of energy required.

3. Experimental Activity

In order to verify experimentally the theory explained above, the injection system of a 22 kW SVRC was modified from the conventional setup in which oil is injected through straight calibrated orifices to an enhanced architecture in which oil is sprayed via pressure-swirled solid-cone nozzles. A few types of nozzles were specifically designed to be fitted on the compressor either radially or axially. Figure 2 shows the images, taken with a high speed camera, of two different types of nozzles spraying oil at $80\text{ }^{\circ}\text{C}$ and at 6 bar_g into an ambient reservoir. These pictures show that oil break-up takes place within a short distance in conditions typical for an air compressor, yet it does not lead exclusively to spherical droplets but also to ligaments, ramifications and undefined structures (9).

A conventional 22 kW SVRC and the modified model featuring the new injection system were tested on a rig (see Figure 3). This experimental test bench employs the necessary instrumentation to measure:

- temperature, pressure and humidity of air at ambient conditions;
- temperature and pressure of air and oil along the process;
- pressure drop across an ISA 1932 nozzle;
- volume flow rate of oil through the external pump;
- rotational speed of the compressor;
- electrical power input to the package.

From the data, the actual delivered flow rate of air is computed accordingly to standard ISO 5167. Subsequently, the standard volume flow rate of air and the specific energy requirement of the packaged compressors are calculated accordingly to standard ISO 1217. The combined measurement uncertainty on the standard volume flow rate is in the 4.5-5.5% range (computed from the practical working formula of ISO 5167), the measurement uncertainty on the electrical power input is at most 1% of the read value (the wattmeter belongs to the accuracy class 1%). Finally, all the compressor stators have four miniature piezoelectric pressure transducers placed circumferentially along one end plate. These transducers are used to measure pressure versus time. From the data acquired along 20-30 revolutions, the indicator diagram is reconstructed

with the methodology developed by Cipollone et al.(7). The measurement uncertainty of the transducers is 0.5% of the full scale (25 bar).

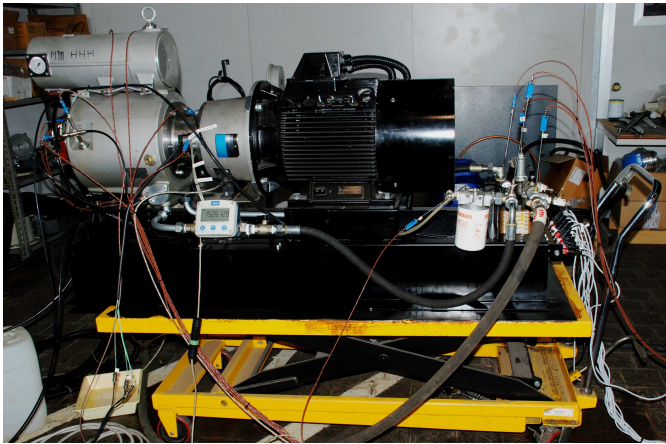


Fig. 3: Compressor rig

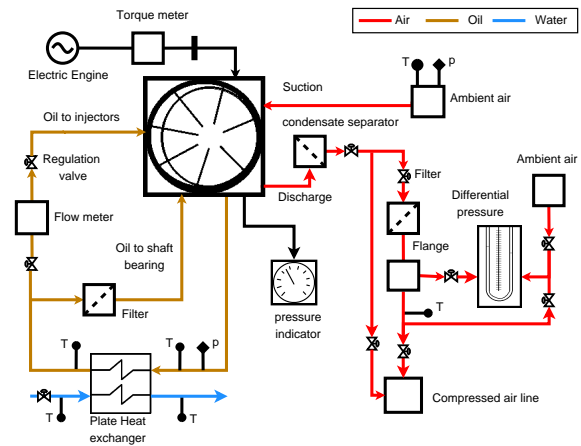


Fig. 4: Process flow diagram of the experimental rig

A process flow diagram of the test rig is depicted in Figure 4 illustrating: the air stream (red) with highlighted the rated flange for measuring the volume flow rate on the discharge line; the oil stream (brown) which is split into the oil to shaft bearing and the oil the injection system, both conventional and enhanced; the water stream (cyan) for oil cooling; and the power stream (black) that drives the shaft. The test rig allows to test the conventional and the enhanced compressor while varying: the discharge pressure, the rotational speed, the injected oil temperature and pressure; for the enhanced compressor it allows to modify which nozzles are activated.

4. Results

A number of tests have been conducted to create an experimental database that could be used for the validation of the simulation code as well as for understanding the margins of further improving the compressor specific energy.

Because in general the flow rate through a nozzle is much smaller than that through an orifice at same pressure and temperature, the total injected oil was relatively low in the enhanced compressor despite the number of nozzles installed. In order to improve flow rate and make a finer spray, a pump was utilized to boost upstream injector pressure. The analysis was done in order to:

1. Further validate the belief that current oil injection technology doesn't produce any significant air cooling effect;
2. Compare the compressor performances, with respect to the current ones, when the upstream injector pressure is equal to the pressure of the compressed air;
3. Compare the compressor performances when the upstream injector pressure is increased using an additional pump.

Most of the comparisons were made with reference to the p-V data which represent the most intimate information concerning real behaviour. Among the wide testing done, the cases reported in Table 1 have been chosen as most representative for the goal of the analysis.

Case #1 represents a typical condition in which the oil is injected according to the conventional technology: the pressure inside the rail is 7.9 bar_a . In reality, the compressor works with a conventional oil flow rate. Case #2 refers to a pressure swirled injection fed at 20.2 bar_a , assisted by an external pump. Only a similar oil flow rate - slightly lower - has been reached, due to the fact that nozzles require higher pressures to achieve the same flow rate as orifices. Case #3 refers to a pressure swirled injection fed at 8.1 bar_a , without any external pump: this is the main reasons of a reduced oil flow rate with respect to previous values.

Parameter	Unit	Experimental conditions		
		1	2	3
Rotation speed	RPM	1500	1498	1504
Injection pressure	bar_a	7.9	20.2	8.1
Free Air Delivery	l/min	3984	4001	3848
Air flow rate, dry	kg/s	0.069	0.070	0.068
Air temperature	$^{\circ}C$	80.1	73.9	90.8
Oil temperature	$^{\circ}C$	67.4	60.0	60.0
Oil flow rate	l/min	37	31	15
End pressure	bar_g	7.5	7.5	7.5
Indicated power	kW	20.90	19.41	19.68
Shaft power	kW	23.08	21.4	21.89
Mechanical efficiency		0.90	0.91	0.90

Table 1: Experimental cases

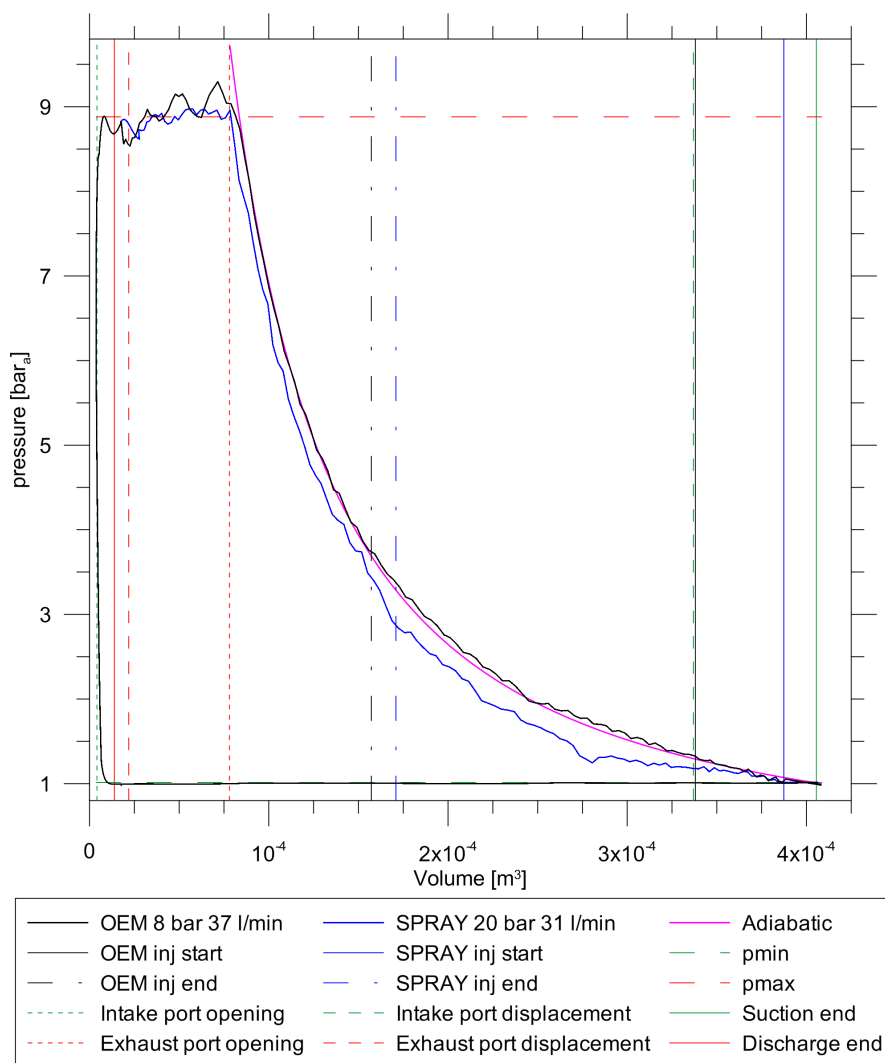


Fig. 5. p-V data, measured and calculated, Case #1 and Case #2

Figure 5 shows the p-V diagram measured referred to Case #1 and Case #2. Most evident aspects are:

1. the compression during Case #1 closely stays on an adiabatic transformation, so the conventionally injected oil doesn't produce any air cooling;
2. oil injection at high pressure (Case #2) realises a visible cooling of the air due to the spray which reduces mean drop diameters. The cooling remains effective during the all injection duration. When this is ended, the pressure trace continues more or less parallel to the adiabatic transformation, slightly lower than the conventional case;
3. The lower oil flow rate and the less effective oil spray, produce a higher air temperature even though a slightly lower indicated power with respect to the conventional case is measured (Case #3). It doesn't behave too differently from the conventional case, so no additional references are given.

The difference between the pressure trace between case #1 and case #2 integrated in the V direction is equal to the actual heat removed by the spray. From the integration the result is equal to 1.50 kW which is very close to the differences found in the shaft power, Table 1. A theoretical verification of this important datum has been done following the mathematical representation of the interactions between compressed air and spray, (8). With reference to the conditions of test Case #2, spray *SMD* (Sauter Mean Diameter) has been calculated, for an orifice diameter D_{or} , by :

$$SMD = 4.52 \left(\frac{\sigma_o \mu_o^2}{\rho_a \Delta p} \right)^{0.25} \left[2.7 \left(\frac{D_{or} \dot{m} \mu_o}{\rho_o \Delta p} \right) \cos \theta \right]^{0.25} + 0.39 \left(\frac{\rho_o \mu_o}{\rho_a \Delta p} \right)^{0.25} \left[2.7 \left(\frac{D_{or} \dot{m} \mu_o}{\rho_o \Delta p} \right) \cos \theta \right]^{0.75} \quad (1)$$

being ρ , μ , σ the density, viscosity and surface tension; θ the half spray cone angle, Δp the pressure difference across the injector; "o" and "a" refer to oil and air. \dot{m} is the oil mass flow rate. From the SMD knowledge, the assumption of a Rosin-Rammler drop size distribution gives the number of particles for each class of dimension chosen. For each droplet, the thermal power exchanged with air \dot{Q}_{a-d} results from:

$$\dot{Q}_{a-d} = \pi d_d k_{mix} N_u^* (T_a - T_d) \quad (2)$$

being d the droplet diameter, k_{mix} the thermal conductivity of the air and oil mixture, T the temperature. N_u^* is the corrected Nusselt number, (10). From previous studies (8), the thermal power exchanged between jet and air is almost exclusively due to forced convection, having observed that oil vaporization doesn't occur. The overall thermal power exchanged with the air is the sum of all the contributions given by the all particles, whose properties change during motion before impingement.

Equation 1 is reported in Figure 6: for an upstream injection pressure equal to 20.2 bar_a (the mean pressure difference is close to 16-17 bar), and for an orifice diameter close to 0.5-0.7 mm, a SMD in the range of 70-75 μm is calculated.

Figure 7 shows the overall thermal power exchanged vs. spray SMD. The predicted theoretical value corresponds to 2.2 kW. This estimation is almost 1.45 times greater than the experimental measured data. This difference can be retained satisfactory, considering all the approximations introduced in the spray modelling: the most important one is related to the fact that each sub injection considered (to reproduce the real process) do not interact. In reality, droplets coalescence occurs, the smaller drops being caught and enveloped by the bigger ones. This reduces heat transfer removed by the air. In this sense, the model appears to be a good compromise between the need of a physical representation (with its high complexity) and the need of having a model which can be run in an engineering software platform.

5. Conclusions

Oil injection technology in SVRC can further improve specific energy with respect to current values. This is due to the additional effect that oil can introduce, with respect to friction reduction and sealing, related to the air cooling process during compression.

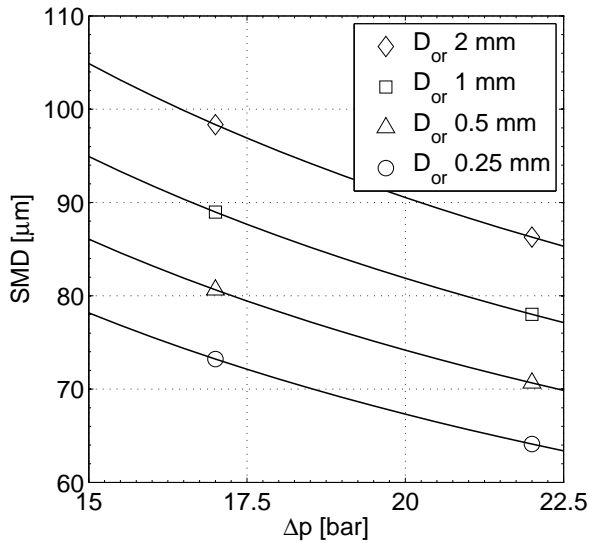


Fig. 6: SMD versus orifice dimension and feeding pressure, Case #2

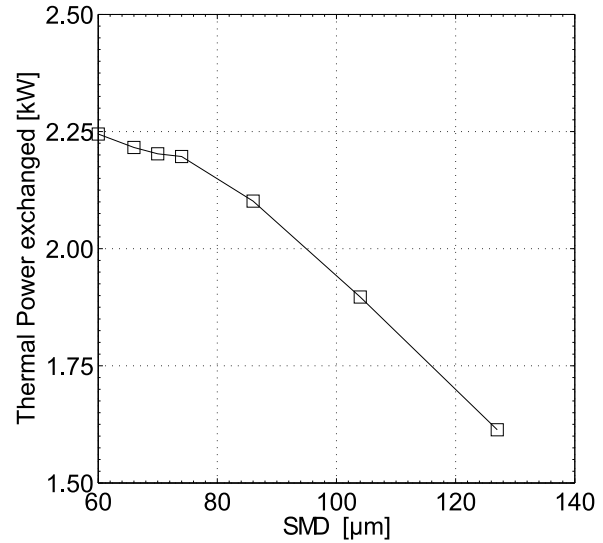


Fig. 7: Thermal power exchanged as a function of SMD, Case #2

In this paper the Authors presented the results obtained making reference to pressure-swirled nozzles substituting the current technology which employs a series of calibrated orifices fed by an oil common rail. The main goal was the investigation of the cooling effect induced by the sprayed oil on the air which can result in a lower compression work, thus in a more efficient compressor. In order to perform and deeply analyze this effect, a theoretical model reproducing the performance of a SVRC was presented in literature and recently updated with a mathematical model of a pressure swirled oil injection system. The main result was the calculation of the heat removed from the air during compression which resulted from oil drops heating till saturation and subsequent evaporation.

Furthermore, an experimental SVRC test rig was built in order to: (a) validate spray modelling in particular concerning the heat removed from the air (cooling); (b) verify compressor performances when the new injection technology is used. A 22 kW existing industrial compressor was tested at diverse working conditions. The experimental results demonstrate that with pressure swirled injectors fed at 20 bar_a thanks to an additional pump, the shaft power decreases by about 1.7 kW; this datum corresponds to the the area between the experimental indicator diagrams referred to the conventional and innovative oil injection technologies. The spray modelling applied to the condition tested gives a thermal cooling equal to 2.2 kW while the treatment of p-V data gives 1.5 kW. The difference can still be considered as satisfactory, in spite of the model simplicity.

Currently, the experimentation on the enhanced compressor is continuing. This initial experimentation has shown that the current common rail position may not be ideal for the new pressure swirl injectors. New injector positions on the compressor are being tested and initial results show interesting improvements on specific energy values even without the necessity of an external oil pump.

6. Acknowledgement

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