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Experimental Investigation On
A Novel Two-Stage Sliding-Vane Air Compressor Based On The Intracooling Concept

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ABSTRACT

Intercooling is a well-known practice in compression technology for reducing the discharge temperature and the power consumption of the process. *Intracooling*, a similar yet not identical concept, is the cooling of the compressed gas between two compression stages by way of spraying a liquid coolant in the gas flow without separating that liquid prior to the second compression stage. This liquid coolant can be the cooled lube oil. The present work reports the experimental experience on a first prototype of a small-scale two-stage sliding-vane compressor based on the concept. The prototype is designed for a relatively low delivery pressure, 0.7-1.0 MPa. Moreover, it is characterized by an oil injection system comprising pressure-swirl nozzles placed on the end-plates of the compression stages and along the intracooling duct. This duct is equipped with eight nozzles: six of them perform a radial inward injection and are equally spaced on the tube length, while the other two are located at its ends for an axial injection, one cocurrent and the other countercurrent to the air flow direction. The experimental tests differ by the number and the position of the active nozzles along the duct. The outcomes indicate that intracooling does not yield operability issues and that the intracooling effectiveness increases with the number of active pressure-swirl nozzles, reaching a decrease in temperature along the duct of about 5°C. However, the configuration with the lowest mechanical specific power, by 4.4% with respect to a single-stage compressor, has only one nozzle active and spraying along the axial flow direction. The results suggest that the compromise among oil flow rate, number of active nozzles and their position, is the best solution to obtain the maximum efficiency for the overall system. In the future, an improved intracooling duct and a mid-size intracooled compressor for higher pressures will be manufactured and tested.

1. INTRODUCTION

Intercooling is a well-known practice in compression technology for reducing both the discharge temperature and the power consumption of the process. A similar yet not identical concept is presented by Murgia *et al*. (2017): *intracooling* is the cooling of the compressed gas between two compression stages by way of spraying a liquid coolant in the gas flow without separating that liquid prior to the second compression stage. When applied to oil-flooded air compressors, the liquid coolant is the cooled lube oil itself. Conceptually, intracooling is an extending application of the benefits of proper oil atomization inside positive-displacement compressors, which are documented extensively by a number of investigations (Singh and Bowman, 1986; Stosic *et al*., 1988; Fujiwara and Osada, 1995; Valenti *et al*., 2013; Valenti *et al*. 2014, Valenti et al. 2016).

The present work reports the experimental experience on a first prototype of a small-scale two-stage sliding-vane compressor based on the intracooling concept. An experimental campaign is conducted to assess the operability and the potential of this concept. As a first step in a longer project, the prototype is designed and manufactured considering a relatively low delivery pressure, whereas higher pressures will be addressed in the future. The next sections describe the methodology for the rapid design of the prototype; moreover, the experimental campaign on the manufactured prototype, with results and discussion, conclusions and, ultimately, the future development of the project.

2. PROTOTYPE DESIGN

This section outlines the steps for the rapid design of the prototype: the determination of the pressures as well as of the volume relations of air and the lube oil, and, lastly, the calculation of the compressor sizes and performances.

2.1 Pressures

As well known, the first step to design correctly a two-stage compressor is to define the *pressure ratio of each stage*, β_{stage}^{id} [-]. It is obtained by minimizing the overall compression work, yielding:

$$\beta_{stage}^{id} = \sqrt{\beta_{tot}\left(\frac{T_{2in}}{T_{1in}}\right)^{\frac{\gamma}{\gamma-1}}} \tag{1}$$

where β_{tot} [-] is the total compression ratio, T_{1in} [K] and T_{2in} [K] are the inlet temperatures of the low- and high-pressure stages, respectively, and γ [-] is the gas heat capacity ratio. If the air temperatures at the inlet of each stage are the same, the ideal compression ratio of the single stage is equal to the square root of the total compression ratio. As a first approach, the same pressure ratio can be assumed for each stage, here defined as β_{stage} [-]. Once the compression ratio of each stage is established, the pressures of the system are consequently as follows:

$$p_{1out} = p_{1in}\,\beta_{stage} \tag{2}$$

where p_{1in} [Pa] is the pressure at the inlet of the low-pressure stage, typically equal to atmospheric pressure. In a two-stage intracooled compressor, this pressure characterizes also the intracooling duct. It does not vary appreciably prior to the high-pressure stage because no sensible pressure losses occur during the intracooling process:

$$p_{intracooling} = p_{1out} = p_{2in} \tag{3}$$

The delivery pressure of the system is the pressure at the outlet of the high-pressure stage p_{2out} [Pa], equal to:

$$p_{2out} = p_{intracooling}\,\beta_{stage} = p_{1in}\,\beta_{tot} \tag{4}$$

2.2 Air volume relations

Once all pressures are defined, the sizing of each stage can be performed. In positive displacement compressors, the increase of the gas pressure is achieved by reducing the volume in which the gas is trapped. Assuming for simplicity an adiabatic compression, the relation between the pressure and the volume is defined as follows:

$$p\,V^{\gamma} = constant \tag{5}$$

and, hence, for each stage:

$$p_{in}\,{V_{in}}^{\gamma} = p_{out}\,{V_{out}}^{\gamma} \tag{6}$$

Consequently, the volume relation for air is:

$$\beta_V = \frac{V_{in,stage}}{V_{out,stage}} = \left(\frac{p_{out,stage}}{p_{in,stage}}\right)^{1/\gamma} = {\beta_{stage}}^{1/\gamma} \tag{7}$$

2.3 Lube oil volume flow rates

An injection system comprising exclusively pressure swirl nozzles is applied to each compression stage and the intracooling duct of the prototype. Lube-oil injection is used in positive-displacement compressors to guarantee the correct lubrication of the moving parts and to minimize air leakage. Furthermore, lube-oil atomization allows to exploit lubricant as thermal ballast with a great thermal capacity to minimize the temperature increase during the compression. The amount of the liquid injected in the first and in the second stage are $\dot{V}_{oil,LP}$ [m^3/min] and $\dot{V}_{oil,HP}$ [m^3/min], respectively. Moreover, $\dot{V}_{oil,intracooling}$ [m^3/min] is the liquid injected into the intracooling duct through the series of nozzles along the duct. These volume flow rates are calculated using the correlation:

$$\dot{V}_l = F\,\dot{V}_2 = F\,\dot{V}_1\,\left(\frac{\Delta p_2}{\Delta p_1}\right)^{0.4} \tag{8}$$

where $\dot{V}_l$ is the liquid volume flow rate [m^3/min] and Δp is the pressure difference across the nozzle [Pa]. Subscripts 1 and 2 identify two different test conditions: the first is associated to the manufacturer test condition with water while the second to the actual one with lubricant oil; F is a density conversion factor, in this case equal to 1.06.

2.4 Compressor final sizes

The final size of each compression stage is performed considering the operating conditions and the oil volume injected during the process. For the low-pressure stage, the inlet volume is set by the target flow rate $\dot{V}$ [m^3/min]:

$$\dot{V} = V_{1in}\, n\, N\, \eta_V \tag{9}$$

where V_{1in} [m^3/chamber] is the volume of the first closed compression chamber, n [chamber/revolution] is the number of closed compression chambers in one revolution of the rotor, N [revolution/min] is the number of revolutions in one minute, and η_V [-] is the volume efficiency. Rearranging, V_{1in} [m^3/chamber] is calculated from Equation (9) and, consequently, V_{1out} can be calculated from Equation (7). Starting from them, all the other geometrical parameters of the low-pressure stage can be derived. The first closed compression chamber of the high-pressure stage is calculated considering both the gas and the oil coming from the low-pressure stage and the intracooling duct.

$$\dot{V}_{2in} = \dot{V}_{1out} + \dot{V}_{oil,LP} + \dot{V}_{oil,intracooling} \tag{10}$$

Solving Equation (9) and Equation (10) for the high-pressure stage, the volume and rotational speed necessary for the correct processing of the air and the oil flow rates can be determined. Ultimately, V_{2out} [m^3/chamber] can be calculated from Equation (7), determining the chamber volume necessary to reach the target delivery pressure p_{out} [Pa].

2.5 System performances

The parameter used to evaluate the intracooling effectiveness is the temperature difference between the intracooling duct inlet and outlet:

$$\Delta T_{intracooling} = T_{1out} - T_{2in} \tag{11}$$

The two-stage compressor efficiency is evaluated by the specific mechanical power SP [kW/(m^3/min)], which is the ratio between the total mechanical power P_{mech} [kW] and the air volume flow rate at the inlet conditions $\dot{V}_{1in}$ [m^3/min]:

$$SP = \frac{P_{mech}}{\dot{V}_{1in}} \tag{12}$$

The total mechanical power P_{mech} [kW] for the two-stage compressor is equal to:

$$P_{mech} = P_{LP} + P_{HP} \tag{13}$$

where P_{LP} [kW] and P_{HP} [kW] are the first and second stage mechanical power consumptions, respectively.

3. EXPERIMENTAL TESTS

This section illustrates the first experimental campaign on the *prototype of a small-scale sliding-vane intracooled two-stage air compressor*, manufactured after completing the rapid design detailed above (Figure 1). The delivery pressure and the compressor size here considered in the design of the prototype are relatively low, in the range of conventional small-scale industrial air compressors (0.7-1.0 MPa and 8-10 kW). Higher pressures and powers, for instance in the range 1.5-2.0 MPa and 75-100 kW, will be addressed in the future.

3.1 Test rig

Two sliding-vane compressors, designed for the purpose, are used as low- and high-pressure stages. As mentioned, the *prototype* is characterized by an injection system comprising exclusively pressure-swirl nozzles of different sizes, placed on the end-plates of the compression stages and located along the intracooling duct. In particular, the duct is equipped with eight nozzles: six of them perform a radial inward injection and are equally spaced on the tube length, while the other two are placed at its ends and perform an axial injection, one cocurrent and the other countercurrent to the air flow direction. Downstream the high-pressure stage a cyclonic air-oil separator is installed. Before reinjection into the compressor, the oil is cooled by cooling water in a conventional plate heat exchanger.

The *prototype* is equipped with all the instrumentation necessary to measure and control the process (Table 1):

- temperatures and pressures sensors are used to control the air and oil status in all key points of the circuit;
- two flowmeters are used to measure the cooling water and the injected oil flow rates;
- a standard orifice and a differential pressure flowmeter to measure the air flowrate according to ISO 5167;
- two power analyzers to measure the electrical consumption by the stages. This measure allows calculating the mechanical power thanks to the characteristics of the electrical components (electric motor and inverter).

The data are acquired via National Instruments cDAQ-9178 and processed via NI Signal Express 2016. The procedure comprises the simultaneous switch on of the two stages, while the delivery pressure is set acting on a valve. Once the system is in steady state conditions, the measures are recorded and performances evaluated according to ISO 1217.

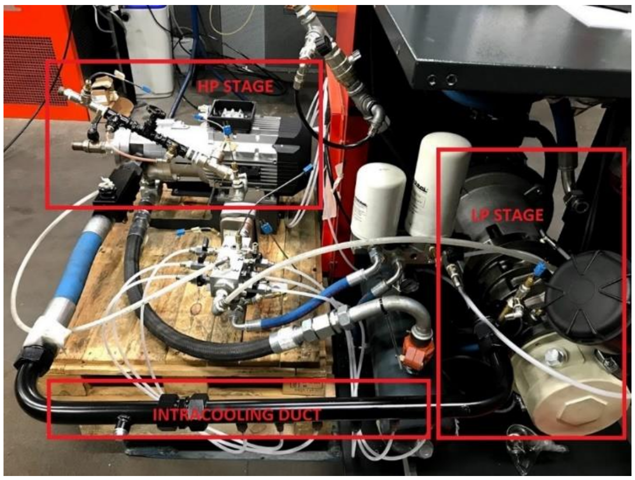

Figure 1: Picture of the small-scale sliding-vane prototype based on the intracooling concept.

Table 1: Instrument list for the test.

Instrument	Quantity	Uncertainty
T-type thermocouple	Temperature	0.5°C
Pressure transducer	Relative pressure	4000 Pa
Flow meter	Volume flow rate	1% reading
Power analyzer	Active power	0.2% reading

3.2 Test cases

The *reference configuration* has all the intracooling nozzles deactivated. Starting from this configuration, the pressure-swirl nozzles are activated progressively. Nine different configurations are analyzed, characterized by different number and positions of the active nozzles, but at the same pressure and temperature injection conditions. Table 2 reports a schematic of the configurations (0, 1, A, B, 2, 3, 4, 5, 6), while Figure 2 depicts a simplified overall scheme. In all the configurations, the same delivery pressure is set equal to 0.8 MPa.

	Active Nozzles								Number of total active nozzles
Configurations	A	1	2	3	4	5	6	B	
C0									0
C1		■							1
CA	■								1
CB								■	1
C2		■	■						2
C3		■	■	■					3
C4		■	■	■	■				4
C5		■	■	■	■	■			5
C6		■	■	■	■	■	■		6

Table 2: Investigated configurations of the intracooled duct differing by the activated nozzles.

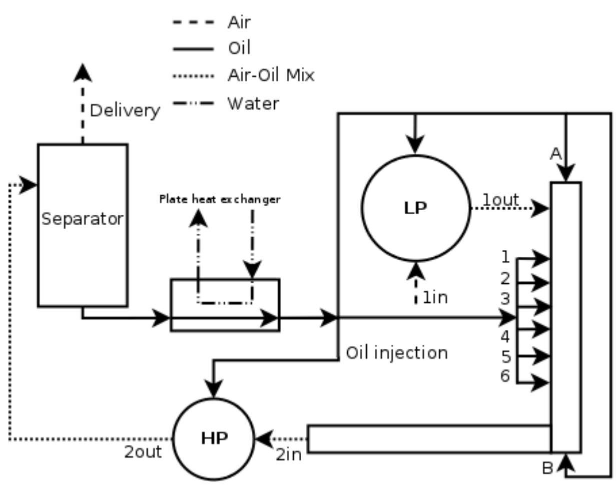

Figure 2: Simplified scheme of the small-scale sliding-vane prototype based on the intracooling concept.

4. RESULTS AND DISCUSSION

This section provides and discusses the experimental outcomes of the nine test cases on the rig previously described. The focus is on the measured temperatures, pressures, volume flow rates, electric power consumptions and, ultimately, on the computed mechanical specific powers. In particular, no operability issues are observed during the tests.

The effects of intracooling on the air temperature and pressure in the intracooling duct are shown in Figure 3. By increasing the number of active nozzles, the intracooling effect is more evident. The increase in oil flowrate in the configurations C2, C3, C4, C5, and C6 determines a progressive reduction of the temperature along the intracooling duct. The configurations C1, CA and CB are characterized by only one active nozzles and determine practically the same intracooling effect. The intracooling pressure, measured inside the intracooling duct, also increases with the number of active nozzles. The intracooling effectiveness reaches its maximum value (about 5°C) in C6 configuration.

Figure 4 indicates that the higher intracooling effect, with a higher number of activated nozzles, the lower the inlet temperatures in the high-pressure stage and, consequently, the lower the outlet temperatures.

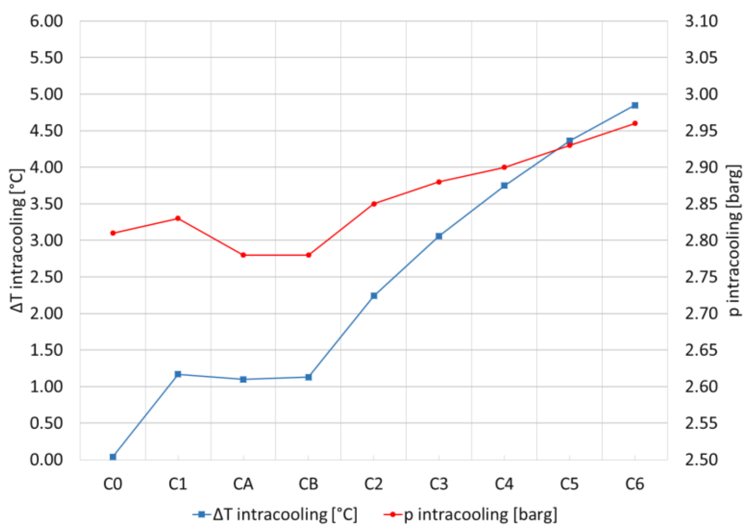

Figure 3: Intracooling effectiveness (blue on left axis) and intracooling pressure (red right).

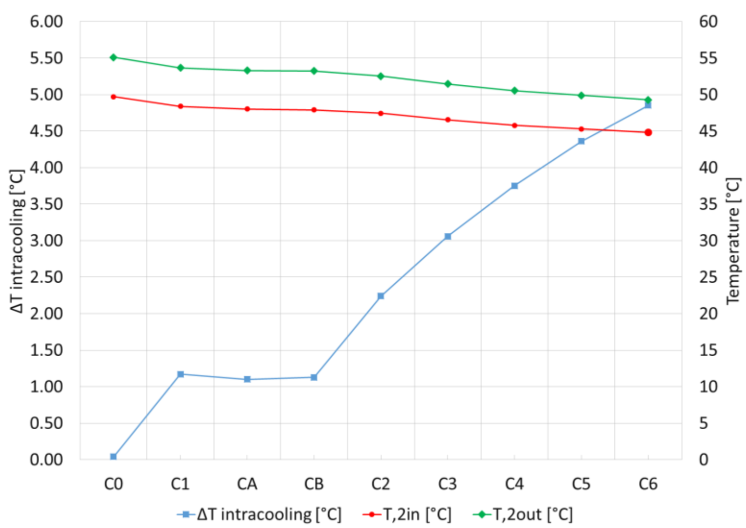

Figure 4: Intracooling effectiveness (blue on left axis) as well as inlet and outlet temperatures of the high-pressure stage (red and green on right).

With the nozzles activation, the mechanical power slightly increases for both compression stages (Figure 5). Although the C6 configuration with all nozzles active allows for reaching the maximum intracooling effect, this configuration yields an appreciable rising of the mechanical power consumptions for both stages:

- the power required by the first stage increases due to the increase of the pressure in the intracooling duct;
- the power required by the second stage increases due to the increase of the oil flow rate.

Figure 6 illustrates the mechanical specific power, *SP* [kW/(m3/min)], of the intracooled two-stage compressor (indicated by 2S) compared against the reference cases with no activated nozzles along the intracooling duct (C0) and against a single-stage compressor (1S), which is constant because independent from the number of nozzles activated. Configuration C0 up to C4 show a better performance of the two-stage system. The best specific power is achieved by CA configuration, which is 4.4% lower than the single-stage compressor (1S). In CA configuration, only one nozzle is activated and it is directed with a spray along the axial direction. This configuration allows for a better atomization process compared with the radial injection, with a lower amount of oil impacting onto the duct surface, and, consequently, with a better heat exchange process through the high-pressure stage. Ultimately, the better heat exchange allows reaching a reduction of the mechanical specific power leading to the best performance of the overall system. The experimental outcomes suggest that the compromise among oil flow rate, number of active nozzles and their position, is the best solution to obtain the maximum efficiency for the overall system.

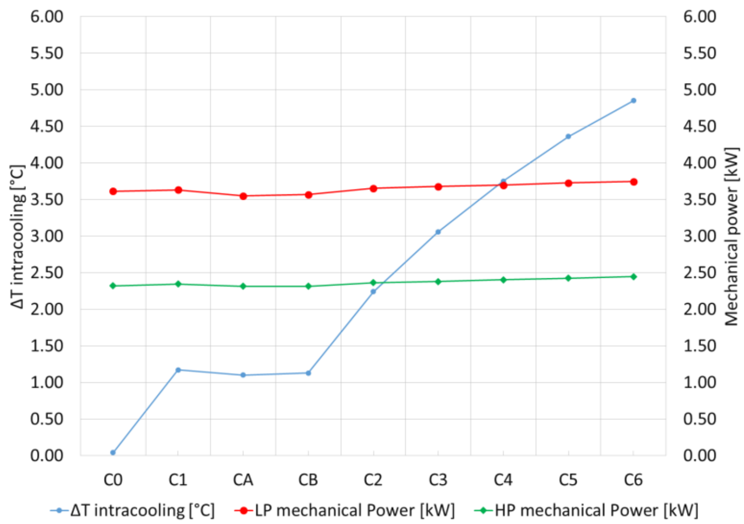

Figure 5: Intracooling effectiveness (blue on left axis) as well as low- and high-pressure stage mechanical power (red and green right).

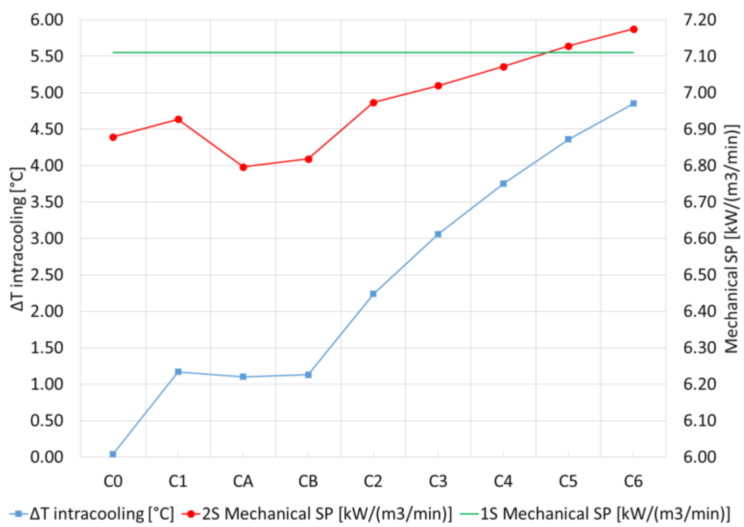

Figure 6: Intracooling effectiveness (blue on left axis) as well as two-stage (2S) and single-stage (1S) mechanical specific power (red and green right).

5. CONCLUSIONS

This work presents the experimental outcomes from the tests of a small-scale sliding-vane prototype based on the intracooling concept. Oil is injected within the compressors and the intracooling duct by way of pressure-swirl nozzles. The work draws conclusions as follows.

- Intracooling does not yield operability issues, even in case of low or high volumes of lube oil.
- The intracooling effectiveness increases with the number of active pressure-swirl nozzles, reaching its maximum value in C6 configuration, of about 5°C, which comprises six nozzles active.
- C6 configuration is not the best in terms of the overall compressor performance because the higher oil flow rate and the greater operating pressure inside the intracooling duct lead to a higher mechanical specific power with respect to the reference configuration (C0) with no nozzles active along the duct itself.
- Configuration C0 up to C4, with 0 up to 4 nozzles active, show better performances compared against the single stage. In particular, CA configuration, with one nozzle active spraying along the axial flow direction, is the best configuration among all with a mechanical specific power lower than reference by 4.4%.
- The experimental outcomes suggests that the compromise among oil flow rate, number of active nozzles and their position, is the best solution to obtain the maximum efficiency for the overall system.

The future activities will cover an improved design of the intracooling duct, based on axial injection and better atomization by an optimal number and position of active nozzles as well as diameter of the duct itself. In addition, a new prototype based on a mid-size two-stage intracooled compressor for high pressures will be assembled and tested.

NOMENCLATURE

Symbols

β	pressure ratio	(-)
γ	specific heat ratio	(-)
η	efficiency	(-)
Δ	variation	(-)
n	number of chambers	(chamber/revolution)
p	pressure	(Pa)
F	density factor	(-)
P	power	(kW)
SP	specific power	(kW/ (m^3/min))
T	temperature	(K)
V	volume	(m^3)
$\dot{V}$	volume flow rate	(m^3/min)

Subscript

in	inlet
out	outlet
LP	Low Pressure
HP	High Pressure

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