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Abstract. Lube-oil injection is used in positive-displacement compressors and, among them, in sliding-vane machines to guarantee the correct lubrication of the moving parts and as sealing to prevent air leakage. Furthermore, lube-oil injection allows to exploit lubricant also as thermal ballast with a great thermal capacity to minimize the temperature increase during the compression. This study presents the design of a two-stage sliding-vane rotary compressor in which the air cooling is operated by high-pressure cold oil injection into a connection duct between the two stages. The heat exchange between the atomized oil jet and the air results in a decrease of the air temperature before the second stage, improving the overall system efficiency. This cooling system is named here *intracooling*, as opposed to intercooling. The oil injection is realized via pressure-swirl nozzles, both within the compressors and inside the intracooling duct. The design of the two-stage sliding-vane compressor is accomplished by way of a lumped parameter model. The model predicts an input power reduction as large as 10% for intercooled and intracooled two-stage compressors, the latter being slightly better, with respect to a conventional single-stage compressor for compressed air applications. An experimental campaign is conducted on a first prototype that comprises the low-pressure compressor and the intracooling duct, indicating that a significant temperature reduction is achieved in the duct.

Nomenclature

Symbols

$\boldsymbol{\beta}$	pressure ratio	(-)
$\boldsymbol{\gamma}$	cone spray angle	(rad)
$\boldsymbol{\mu}$	viscosity	(Pa s)
$\boldsymbol{\rho}$	density	(kg m^{-3})
$\boldsymbol{\sigma}$	surface tension	(N m^{-1})
$\boldsymbol{P}$	power	(kW)
$\boldsymbol{SMD}$	Sauter Mean Diameter	(m)
$\boldsymbol{T}$	temperature	(K)
$\boldsymbol{V}$	volume	(m^3)
$\boldsymbol{d}$	diameter	(m)
$\boldsymbol{p}$	pressure	(Pa)
$\boldsymbol{q}$	size distribution parameter	(-)

Subscript

$\boldsymbol{el}$	electrical
$\boldsymbol{g}$	gas
$\boldsymbol{id}$	ideal
$\boldsymbol{in}$	inlet
$\boldsymbol{l}$	liquid
$\boldsymbol{or}$	orifice
$\boldsymbol{out}$	outlet
$\boldsymbol{tot}$	total

1. Introduction

Intercooling is an established practice in *compression technology* for reducing discharge temperature and power requirement [1,2]. As known, intercooling is the cooling of the compressed gas between two compression stages by way of a *heat exchanger* employing typically water or air as coolant. Overall, thermal power is transferred from the gas to the *environment*.

The present work proposes a similar yet not identical concept: *intracooling*, which 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 entering the next compression stage. The liquid shall be separated at the end of the last compression stage. Overall, thermal power is transferred from the gas to the liquid, but not to the environment. When applied to an *oil-flooded* compressor, such as a sliding-vane air compressor considered here, the lubricant itself can be used as sprayed liquid. In this case, intracooling with respect to intercooling allows for fewer pieces of equipment, as shown by Figure 1, and it allows for oil to be available at the highest pressure, a condition that may be exploited for a better injection. Figure 1 depicts the intracooled layout and compares it against an intercooled one: the low-pressure oil separator as well as the intercooling heat exchanger are substituted by the *intracooling duct*. For sake of clarity, the authors will refer to the low-pressure stage as LP and to the high-pressure stage as HP.

The first objective of this work is to predict the thermodynamic performance of an intracooled compressor, taking a single-stage compressor and an intercooled compressor as references. This analysis is executed with an in-house software developed for sliding-vane machines [3,4], but the outcomes are valid for the generic oil-flooded compressor. Moreover, the second objective is to prove the concept on a preliminary prototype that comprises only the LP compressor and the intracooling duct (while the HP compressor will be coupled in the next phase of the project).

To the knowledge of the authors, the concept of intracooling has never been formulated before and, hence, this work proposes a novel terminology as well as novel numerical and experimental analyses.

The following sections describe: the rationale of intracooling, its numerical modelling, the test case adopted for the investigation, the experimental setup for the first prototype, the numerical as well as experimental results, and lastly the conclusions.

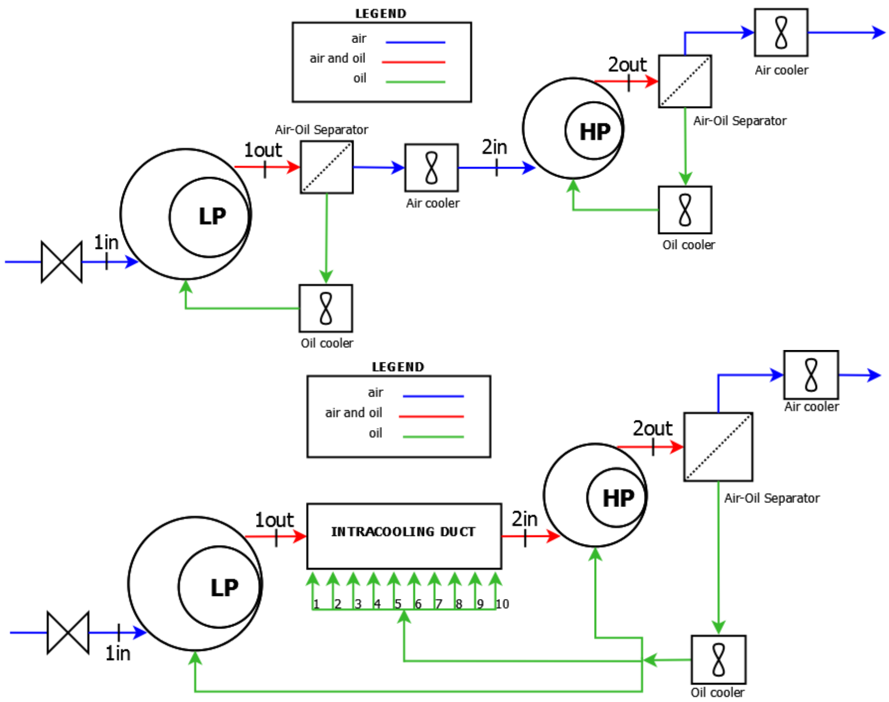

Figure 1. Application of intercooling (top) and intracooling (bottom) to oil-flooded (positive-displacement) compressors, such as sliding-vane air compressors as considered in this work (LP indicates low pressure, HP high pressure).

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2. Rationale of intracooling

The well-known fundamental notion of both inter- and intracooling states that the lower the initial temperature of a gas compression, the lower the compression work; hence, an overall compression shall be divided into two (or even more) stages between which the gas is cooled by some mean in order to reduce the overall work requirement. In intercooling, the gas cooling is achieved by a heat exchanger; in intracooling, instead, the gas cooling is achieved by spraying cooled lubricant oil as small droplets into the gas flow within the intracooling duct. The sprayed oil is not separated from the gas, but it is let in the flow for lubricating the next compressor. Oil is separated from the gas only at the discharge of the second (last) compression stage.

The oil spraying in the compressors and in the intracooling duct can be accomplished by *pressure-swirl nozzles*. Hence, the oil sprays in intracooling may not cool the gas with the same effectiveness of the heat exchanger in intercooling. However, the availability of oil at the highest pressure leads to a fine and beneficial atomization in the LP compressor and in the intracooling duct.

3. Numerical modelling

Considering a two-stage compression, the *pressure ratio of the first stage*, β_{stage}^{id}, minimizing the overall compression work is:

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

where β_{tot} is the total compression ratio, T_{1in} [K] and T_{2in} [K] are the inlet temperatures of the LP and HP 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 a single stage is equal to the square root of the total compression ratio.

Once established the compression ratio of each stage, it is possible to proceed to the compressors sizing. The heat transfer between gas and liquid within the process affects directly the compressor performance and dimensioning. About this topic, a detailed simulation code has been developed by the authors, as reported in the previous works [3,4]. It is based on the following assumptions:

- the gas behaves as an ideal gas and the liquid as an incompressible fluid;
- the gas and the liquid are characterized by a constant specific heat;
- the gas is non-soluble into the liquid and the liquid is non-volatile into the gas;
- the compression is adiabatic toward the surroundings;
- the droplets are non-deformable spheres and the coalescence of droplets is neglected;
- the temperature variation inside the droplets is neglected.

The process is discretized so that gas and liquid temperatures and the gas pressure within the compression chamber can be calculated in every time step of the process. The chamber volume is calculated by geometrical means as a function of the angular position.

The injection system for the lubricating oil is based on both *plain orifices* and *pressure-swirl nozzles* to exploit the thermal effect of sprayed oil inside the compression chamber. The injection system is characterized in terms of number and position of the injectors as well as their jet characteristics.

The oil sprayed through pressure-swirl nozzles splits into a number of droplets of different dimensions affecting strongly the heat transfer from the air to the oil. The probability of a droplet size within a liquid jet, $f(d)$ [-], is described by the Rosin-Rammler relation as detailed by Mugele and Evans (1951):

$$f(d) = q\frac{d^{q-1}}{SMD^q}e^{-\left(\frac{d}{SMD}\right)^q} \tag{2}$$

where d [m] is the droplet diameter, SMD [m] is the Sauter Mean Diameter, defined as the diameter of a sphere that has the same volume-to-surface area ratio of the entire droplet sample, and q [-] is the shape parameter providing a measure of the spread of droplet sizes. In its turn, q is taken equal to 3.5 (Lefebvre, 1989) and SMD computed via the relation by Wang and Lefebvre (1987):

$$SMD = 4.52\left(\frac{\sigma_l\,\mu_l}{\rho_g\,\Delta p}\right)^{0.25}\left[2.7\left(\frac{d_{or}\,\dot{m}_l\,\mu_l}{\rho_l\,\Delta p}\right)\cos\frac{\gamma}{2}\right]^{0.25} + 0.39\left(\frac{\rho_l\,\mu_l}{\rho_g\,\Delta p}\right)^{0.25}\left[2.7\left(\frac{d_{or}\dot{m}_l\,\mu_l}{\rho_l\,\Delta p}\right)\cos\frac{\gamma}{2}\right]^{0.75} \quad (3)$$

where σ_l is the liquid surface tension [N m^{-1}], Δp the differential pressure [Pa], ρ the density [kg m^{-3}], d_{or} the nozzle/orifice diameter [m], γ the cone spray angle [rad], $\dot{m}_l$ and μ_l are the liquid mas flow rate [kg s^{-1}] and viscosity [Pa s]. The droplet diameter depends strictly on the nozzle differential pressure, which is calculated as the difference between the injection pressure and the pressure inside the compression chamber at the moment of injection.

The same model, properly adapted, is used to determine the heat transfer inside the intracooling duct. The main differences between the two systems are: the compression chamber is a closed system to mass transfer where the geometrical volume decrease causes an increase of the gas pressure; the intracooling duct is a system open to mass transfer where pressure is, as a first approach, assumed constant and where the gas volume decrease thanks to the cooling operated by the sprayed oil.

In case of inter-cooling, the pressure at the end of the first compression stage has to be opportunely increased because of the pressure drop on the cooling fan. In case of intra-cooling no pressure drop occurs into the connection duct, leading to a perfect matching between the two stages.

The second stage sizing is based on the compatibility between the volume of air-oil mix coming from the duct and the volume of the first chamber available in the second stage. Once the compressors geometries are fixed, the simulation allows to calculate the indicated power, as the area enclosed by the pressure-volume curve and the mechanical power of each compression stage. Finally, the overall input power, $P_{el,tot}$ [kW] is calculated as:

$$P_{el,tot} = P_{el,LP} + P_{el,HP} + P_{el,FAN} \quad (4)$$

where $P_{el,LP}$ [kW] and $P_{el,HP}$ [kW] are the low and high pressure input power respectively and $P_{el,FAN}$ [kW] is the electric power adsorbed by the fans, calculated specifically for each fan showed in Figure 1.

4. Case study

The case study is a *large-size sliding-vane air compressor* operating as LP stage. It is characterized by a hybrid injection system comprising plain orifices along the compressor on the stator and three couples of pressure-swirl nozzles on the end plates. The intracooling duct is directly connected to the compressor discharge and is equipped with ten pressure-swirl nozzles, equally spaced on the tube length (Figure 2). Downstream the intracooling duct a cyclonic air-oil separator is installed. An auxiliary pump is used to simulate the effect of the HP compressor for lubricant injection. Before being injected, the lubricant is water-cooled in a plate heat exchanger.

Figure 2. The first prototype comprising the intracooling duct (black) and the LP compressor (red). This prototype is taken as case study for the analysis.

5. Experimental testing

An experimental campaign is conducted to verify the intracooling effectiveness. The *prototype* is equipped with the instruments necessary to measure and control the process (Table 1): temperatures and pressures of air and oil in all key points of the circuit, the injected oil flow rate, and the electrical power.

The experimental data are acquired via the National Instruments cDAQ-9178 and processed via the NI Signal Express 2016. The experimental procedure comprises the simultaneous startup of the compressor and the auxiliary pump. The air delivery pressure is set to the chosen value acting on a valve. Once the system is in steady state conditions, the measurements are recorded.

The *reference configuration* for the experimental campaign is that with the intracooling deactivated. Consequently, the pressure swirl nozzles are activated progressively. Thirteen different configurations are tested, characterized by different number and positions of active nozzles, but at the same pressure and temperature injection conditions. The parameter used to evaluate the intracooling effectiveness is the temperature difference between the intracooling duct inlet and outlet.

Lastly, the thermal distribution on the intracooling duct is visualized by an infrared camera.

Table 1. Test rig instrumentation.

Instrument	Measured quantity	Absolute uncertainty	Unit
T-type thermocouple	Temperature	0.5	°C
Pressure transducer	Relative pressure	4000	Pa (gauge)
Oil flow meter	Volume flow rate	5	$l\ min^{-1}$
Power analyzer	Active power	0.2% reading	kW

6. Results and discussions

Three compression systems are investigated numerically: a conventional single-stage, an intercooled and an intracooled compressor. The overall pressure ratio as well as the air and liquid inlet temperatures are the same for all cases. Simulations of intercooled and intracooled compressors are executed, using the detailed simulation code based on the numerical model presented in Section 3. , where the compressors geometries are established in order to guarantee the same air mass flow rate and total amounts of injected oil in both the systems. This quantity is higher in two-stage compressors with respect to the single-stage because all compressors must be correctly lubricated. The main inputs and outputs are showed in Table 2.

As shown by the numerical results, the potential energy saving of the two-stage compression systems, compared to the single-stage, is up to 10%. The performance of the intercooled and the intracooled compressors are comparable, with a better result for the intracooled compressor (-2.3% of total input power). The compression power of the LP stage is slightly higher in the intercooled system. Despite the amount of oil injected into the compressor is the same, the lower injection pressure causes a coarser atomization that generates larger droplets dimension, as explained in Section 3. In this condition, the heat exchange between air and oil is penalized leading to an increase of air temperature and pressure as well as a greater consumed power.

The two systems differ also in terms of intermediate pressure between the two stages. On one hand, β_{stage}^{id} of the intercooled LP stage is lower than the intracooled one, because of the air temperature differences at the inlet of both the HP stages. On the other hand, the intercooled system is affected by the air pressure drop across the intercooler that influences the intermediate pressure between the two stages. This affects directly the intercooled HP stage that has a lower inlet pressure compared with the intracooled one, and consequently, a higher adsorbed power. Furthermore, despite the higher initial compression temperature in the HP intracooled compressor, the amount of liquid deriving from the intracooling duct allows to stabilize the thermal process since the intake port closing. In this system, the role of heat carrier fluid imputable to the lubricant is amplified and optimized to cool the air during the overall compression process. Lastly, the fan powers are similar between the two compression systems.

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These numerical results indicate that oil atomization via pressure-swirl nozzle is an interesting technology from the input power perspective not only for the process occurring within a compressor, but also between two compressors.

Table 2. Numerical simulation results (NP stands for "not present").

	Unit	Conventional single-stage	Intercooled two-stage	Intracooled two-stage
		Inputs		
Air mass flow rate $\dot{m}_g$	kg s^{-1}	0.3	0.3	0.3
Motor speed	rpm	1000	1000	1000
LP Inlet gas temperature, T_{1in}	°C	40	40	40
HP Inlet gas temperature, T_{2in}	°C	NP	40	45
Liquid injection temperature	°C	40	40	40
Total compression ratio β_{tot}	-	8	8	8
		Outputs		
Total oil volume flow rate $\dot{m}_l$	l min^{-1}	112.8	188.6	187.2
LP stage compression power $P_{el,LP}$	kW	96.8	46.5	46.2
LP stage fan power $P_{el,LP\ FAN}$	kW	1.3	0.5	NP
HP stage compression power $P_{el,HP}$	kW	NP	40.9	39.0
HP stage fan power $P_{el,HP\ FAN}$	kW	NP	0.6	1.2
Total compression power $P_{el,LP+HP}$	kW	96.8	87.4	85.2
Total fan power $P_{el,LP+HP\ FAN}$	kW	1.3	1.1	1.2
Total input power $P_{el,tot}$	**kW**	**98.1**	**88.5**	**86.4**

Table 3 and Figure 3 report the results of the experimental campaign conducted on the first prototype of the intracooled compressor by varying the number and the position of the activated nozzles.

Table 3. Configurations of the intracooled duct differing by activated nozzles (RC stands for reference configuration, C1 to C12 to configuration from 1 to 12).

		Active Nozzles										Number of total active nozzles
		1	2	3	4	5	6	7	8	9	10	
Cases	RC											0
	C1											1
	C2											1
	C3											1
	C4											3
	C5											3
	C6											3
	C7											3
	C8											5
	C9											5
	C10											5
	C11											7
	C12											10

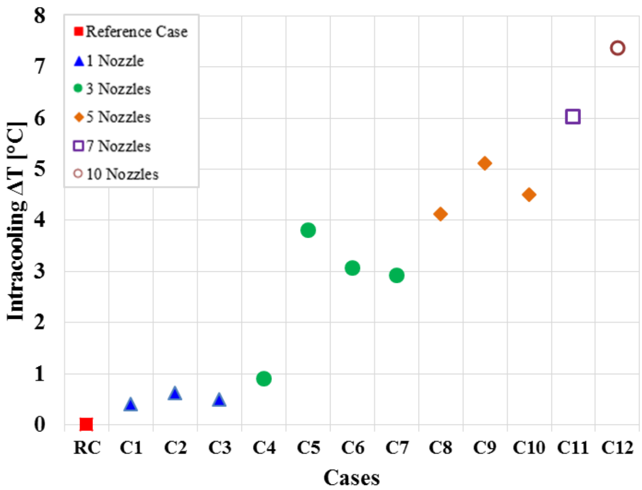

Figure 3. Measured temperature difference between inlet and outlet of the intracooled duct depending upon the investigated configuration.

The temperature difference between the intracooling duct inlet and outlet is strongly influenced by the injected oil flow rate: the higher is the number of active nozzles, and thus the flow rate, the higher is the temperature decrease along the duct. In particular:

- Cases C1 thru C3 comprise only one activated nozzle: the intracooling effectiveness is low and the position of the injector has a minor influence;
- Cases C4 thru C7 comprise three activated nozzles: the major amount of liquid injected into the duct leads to a higher cooling effect;
- Cases C8 thru C12 comprise five, seven and ten activated nozzles: the intracooling effect is higher and higher as the number of nozzles increase.

The cases with three and five nozzles show a better effectiveness when the last nozzles are activated (C5 and C9). The magnitude of the phenomenon is greater for C5 and will be investigated in the future.

In brief, the best temperature reduction is 7.4°C for case C12. The infrared analysis confirms it: the temperature distribution for the reference case, RC, and the best case, C12, are shown in Figure 4.

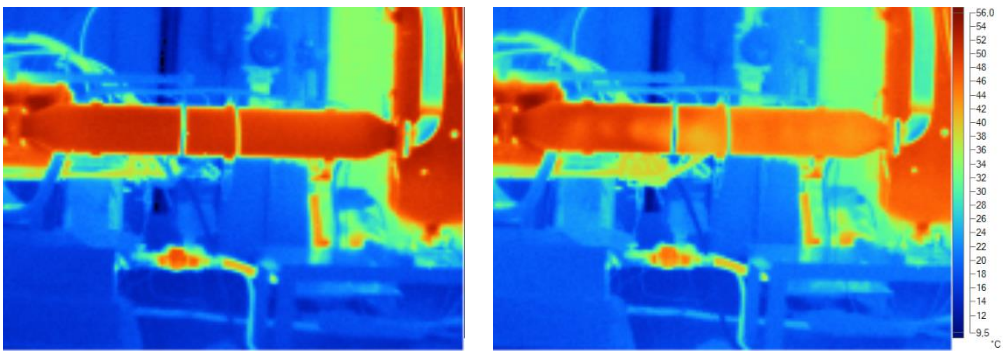

Figure 4. Infrared images of the intracooling duct for the reference case, RC, on the left and the best case, C12, on the right.

7. Conclusions

This work presents the novel concept of intracooling for reducing the overall compression requirement in a similar, yet not identical, fashion to intercooling. The concept of intracooling consists in exploiting part of the amount of oil needed to the high-pressure compressor, not only to guarantee the lubrication effect and the internal sealing, but also for enhancing the air cooling between the two stages.

This work draws the following conclusions.

- The intracooled two-stage compressor is a novel system promising an energy saving up to 10% with respect to a conventional single-stage compressor for typical air compression applications.
- The performance of the intercooled and the intracooled compressors are numerically comparable, with a better result for the intracooled system (-2.3% of total input power).
- The low-pressure stage of the intracooled compressor shows a better compression process compared to the intercooled one, due to the higher injection pressure that results in a finer injected oil atomization; in terms of absorbed power, the first stage of both inter- and intracooled compressors are slightly similar.
- The high-pressure stage has a lower adsorbed power (39.0 kW) compared with the intercooled compressor (40.9 kW); it presents a higher inlet temperature and pressure compared with the intercooled one; the liquid coming from the intracooling duct allows to stabilize the thermal process since the intake port closing.
- The measured intracooling effectiveness depends strongly on the number of active nozzles and, consequently, on the oil flow rate as well as on their position.
- The highest intracooling effect, resulting in a measured temperature reduction of 7.4 °C, can be achieved by activating all ten nozzles.

The future work will focus on the design of the high pressure stage and a control system that governs the nozzles activation in order to maintain constant the temperature at the inlet of the second stage.

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