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A NTU-Based Model to Estimate Suction Superheating in Scroll Compressors

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ABSTRACT

Suction superheating plays a major role in determining the efficiency degradation of hermetic scroll compressors. Current models to predict superheating are usually experimentally calibrated and therefore can only be applied to existing compressors. This paper presents a thermal model to estimate suction superheating in scroll compressors, based on the NTU method for heat exchangers design. The model considers an isothermal surface exchanging heat with the gas in the suction path and in the discharge plenum. Compared to other models, the new approach described herein has the advantage of not requiring any experimental input data. The thermal model is coupled to a thermodynamic model and applied to evaluate the performance of a scroll compressor. The model was capable to predict the suction gas temperature in good agreement with experimental data, making it particularly useful for compressor design.

1. INTRODUCTION

The scroll compressor is formed by two identical spiral-shaped scroll elements, which are mounted inverted and rotated 180° in relation to each other. The contacts between the scroll elements at sealing points form multiple compression volumes (pockets). As the orbiting scroll performs its motion in relation to the stationary scroll, the sealing points migrate inwards, pushing the pockets to the center of the scrolls, reducing their volumes and compressing the gas inside.

Heat transfer in hermetic compressors usually results in efficiency loss. One of such inefficiencies is the so-called suction superheating, due to the temperature increase of the gas as it flows from the compressor inlet to the suction chamber. This increase of temperature directly reduces the gas density and, hence, the mass flow rate.

Caillat et al. (1988) proposed one of the first thermal models for suction superheating in scroll compressors. Other models based on lumped formulations have been developed to estimate the compressor temperature distribution (Chen *et al.*, 2002; Diniz *et al.* 2012). In addition to the aforementioned models, semi-empirical models have been proposed to represent both the suction superheating and the compression process, such as those reported by Winandy *et al.* (2002) and Cuevas and Lebrun (2009). Although very useful, such models require experimental calibration and, therefore, can only be applied to existing compressors.

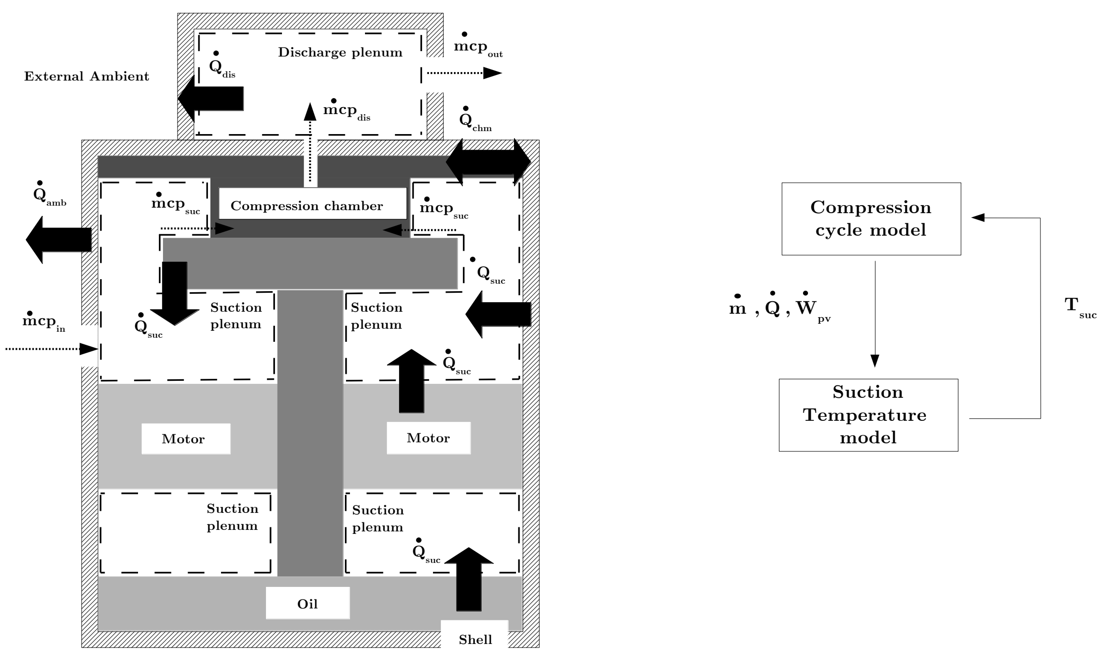
The present paper reports a thermal model to predict suction superheating in scroll compressors. The approach is based on the ε-NTU method for heat exchangers design (Kakac *et al.*, 2002). Coupled to a thermodynamic model for the compression process, it is capable of predicting suction temperature without any experimental calibration. The model was tested in two different compressors and a sensitivity analysis was performed to evaluate its applicability.

2. COMPRESSION PROCESS MODEL

In compressor modelling, the suction temperature is required in the simulation of the compression cycle. In a typical low-pressure shell scroll compressor, illustrated in Figure 1(a), the inlet gas thermally interacts with hot components, such as the shell and the lubricant oil, before reaching the compression chamber. Electromechanical inefficiencies associated with the electrical motor and bearing system also represent a significant source of heat in scroll compressors. Other aspects, such as heat transfer in the compression and discharge chambers also contribute to increase the gas temperature in the suction path.

As illustrated in Figure 1(b), the thermal model for the suction temperature requires input data from the compression cycle model, such as mass flow rate, heat transfer in the compression chamber and indicated power. On the other hand, the suction temperature is an input data in the compression process model. Therefore, the compression process model and thermal model have to be simulated in a coupled manner to correctly characterize the compressor performance.

In the model proposed herein, the compression process was modeled employing the approach described in Pereira and Deschamps (2010), by adopting a lumped formulation for the conservation Equations. Chamber volumes and available flow areas during compression cycle were described through accurate mathematical expressions. The heat transfer coefficients inside the compressor pockets were obtained from numerical simulations based on the finite volume method, as detailed in Pereira and Deschamps (2012). Leakages through the top and flank gaps were estimated from correlations developed by Pereira (2012) via numerical simulations, which were validated with reference to experimental data made available by Suefuji et al. (1992) and Xiuling et al. (1992). The approach followed by Pereira and Deschamps (2010) requires the simulation of a certain number of compression cycles to achieve convergence.



(a) (b)

Figure 1 – (a) Heat interactions in a typical scroll compressor and (b) Compression cycle and suction temperature model interaction.

3. THERMAL MODEL

The thermal model developed to estimate the suction temperature is based on a proposal of Winandy et al. (2002), which represents superheating as a temperature increase experienced by the gas as it flows in contact with an isothermal plate. This isothermal plate is also responsible for the temperature reduction of the gas temperature (discharge cooling-down) in the discharge plenum. Winandy et al. (2002) obtained the necessary thermal conductances through experimental calibration.

In order to represent suction superheating and discharge cooling-down, the isothermal surface should represent an average temperature of different regions inside the compressor so as to allow heat transfer with the gas in such regions. In this way, all thermal interactions inside the compressor can be related to this isothermal surface, including heat generation due to mechanical and electrical inefficiencies, as well as heat transfer within the compression chamber.

In the model proposed herein, the gas path inside the compressor, before and after the compression chamber, is modelled as turbulent flow in a circular duct with an isothermal wall of negligible thickness. This choice seems appropriate since the gas flows through channels in a typical scroll compressor, especially in the suction path. Therefore, the suction superheating and the discharge cooling-down are represented through thermal interactions with the duct wall of temperature T_w. A schematic view of the gas path from the inlet to the outlet and the associated heat transfer considered in the model is depicted in Figure 2. The model can be seen as much simplified representation of the thermal interactions illustrated in Figure 1(a).

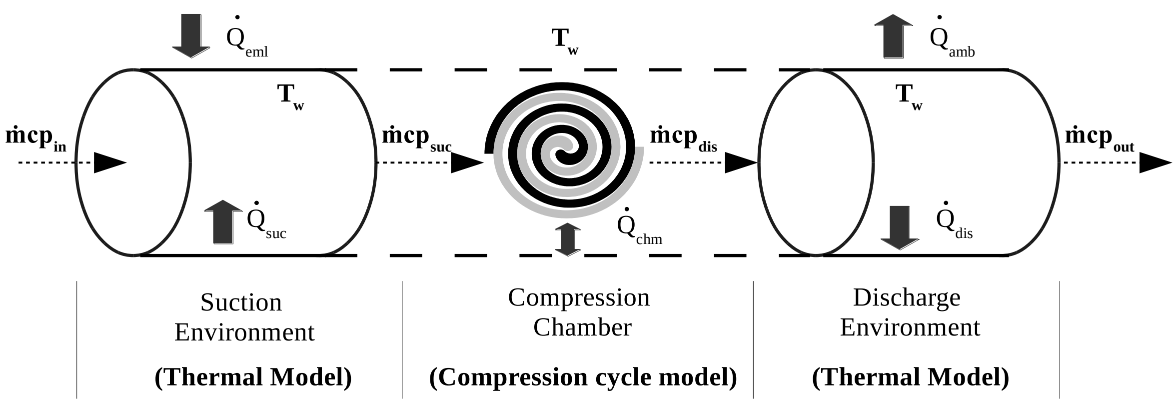


Figure 2 – Simplified model of gas flow and heat interactions in a typical scroll compressor.

A steady state thermal balance in the duct surface shows that:

$$\dot{Q}_{suc} + \dot{Q}_{amb} = \dot{Q}_{dis} + \dot{Q}_{chm} + \dot{Q}_{eml} \,. \tag{1}$$

Heat transfer between the gas being compressed and the isothermal surface, $\dot{Q}_{chm}$, is obtained from the compression process model. The heat generation due to electromechanical losses, $\dot{Q}_{eml}$, is calculated from estimates for the indicated power, $\dot{W}_{pv}$, and the electromechanical efficiency (η_{eml}):

$$\dot{Q}_{eml} = \frac{\dot{W}_{pv}}{\eta_{eml}} - \dot{W}_{pv} \,. \tag{2}$$

The heat transfer term between the isothermal surface and the external ambient, $\dot{Q}_{\sup-amb}$, is given by:

$$\dot{Q}_{amb} = h_{amb} A_{cmp} (T_w - T_{amb}) \,, \tag{3}$$

where the heat transfer coefficient, h_{amb}, is estimated by using the following correlation:

$$Nu_{amb} = \frac{h_{amb} D_{cmp}}{k_{amb}} = 0.3 + \frac{0.62\,\mathrm{Re}_{D_{cmp}}^{1/2}\,\mathrm{Pr}^{1/3}}{[1+(0.4/\mathrm{Pr})^{2/3}]^{1/4}} \left[1+\left(\frac{\mathrm{Re}_{D_{cmp}}}{282000}\right)^{1/2}\right] \tag{4}$$

and A_{cmp} is the compressor external area.

The suction superheating, $\dot{Q}_{suc}$, due to heat transfer between the isothermal surface and the suction environment is modelled via the $\varepsilon - NTU$ method (Kakac *et al.*, 2002):

$$\dot{Q}_{suc} = \varepsilon_{suc} \dot{m} c_P (T_w - T_{in}) \,, \tag{5}$$

where the effectiveness, ε_{suc}, is given by:

$$\varepsilon_{suc} = 1 - \exp(-h_{suc} A_{suc} / \dot{m} c_P) \,. \tag{6}$$

The heat transfer coefficient h_{suc} is obtained from:

$$Nu_{suc} = \frac{h_{suc} D_{suc}}{k} = 0.023\,\mathrm{Re}_{suc}^{\;0.8}\,\mathrm{Pr}^{0.4} \,, \tag{7}$$

where the Reynolds number is calculated using the following expression:

$$\mathrm{Re}_{suc} = \frac{4\dot{m}}{\pi D_{suc} \mu_{suc}} \,. \tag{8}$$

D_{suc} is a hydraulic diameter given by:

$$D_{suc} = \frac{4 V_{suc}}{A_{suc}} \,, \tag{9}$$

where V_{suc} is the volume occupied by the gas in the suction environment and A_{suc} is the area available for heat transfer. Chen *et al.* (2002) employed this method to evaluate the hydraulic diameter of the compression chambers in scroll compressors.

With the suction superheating, $\dot{Q}_{suc}$, obtained from Equation (5), it is possible to obtain the suction temperature from a simple energy balance:

$$\dot{Q}_{suc} = \dot{m} c_P (T_{suc} - T_{ent}) \,. \tag{10}$$

The discharge cooling-down, $\dot{Q}_{dis}$, is calculated in a similar way:

$$\dot{Q}_{dis} = \varepsilon_{dis} \dot{m} c_P (T_{dis} - T_w) = \dot{m} c_P (T_{dis} - T_{out}) \,. \tag{11}$$

$$\varepsilon_{dis} = 1 - \exp(-h_{dis} A_{dis} / \dot{m} c_P) \,, \tag{12}$$

$$Nu_{dis} = \frac{h_{dis} D_{dis}}{k} = 0.023 \mathrm{Re}_{dis}^{\ 0.8} \mathrm{Pr}^{0.3} . \tag{13}$$

At each time-step during the solution procedure of the thermal model, the temperatures of the compressor components are calculated iteratively until the residual of the energy balance shown in Equation (1) is less than 0.01 % of the electrical energy consumption. This solution procedure is illustrated in Figure 3(a).

As discussed previously, the compression process model requires the simulation of a certain number of cycles to achieve convergence. In the present solution procedure, the compression process model and the thermal model exchange input data at the end of each cycle. This procedure is depicted in Figure 3(b) and starts after the simulation of two compression cycles.

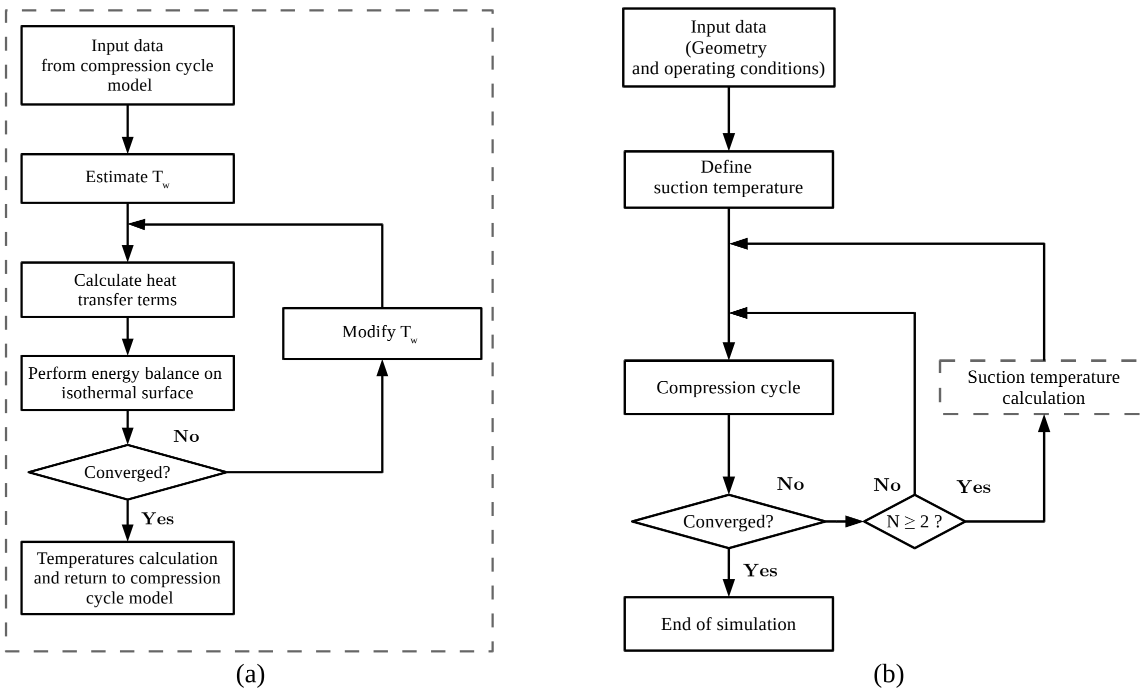


Figure 3 – Solution procedure - (a) Thermal model and (b) Coupled model.

4. RESULTS AND DISCUSSION

4.1 Comparisons with experimental data

The coupled model was applied to simulate two compressors (A and B) and the predictions were compared with experimental data. Both compressors are vertically oriented and have a low-pressure shell, where the suction gas thermally interacts with the compressor hot components before entering the compression chamber. Other important characteristics of both compressors are given in Table 1, whereas the geometrical input data required in the thermal model are summarized in Table 2.

The operating conditions and predictions of temperature at the suction and outlet regions of compressors A and B are presented in Tables 3 and 4, respectively. In the same tables, experimental are also provided for an assessment of the model accuracy. Only two operating conditions were considered in the measurements of compressor A, whereas six operating conditions were investigated for compressor B. Electromechanical efficiencies of 0.67 and 0.63 were used in all simulations of compressors A and B, respectively. Such values were estimated from predictions of indicated power and measurements of electrical consumption.

The results in Tables 3 and 4 show good agreement between predictions and experimental data for the suction temperature. Moreover, the thermal model was able to represent the suction temperature in two different compressors with different fluids and operating conditions. However, the model failed to characterize the compressor outlet temperature, especially for compressor A. This shortcoming is probably associated with the correlation used to estimate the heat transfer coefficient in the discharge plenum, Equation (13).

Table 1: Main characteristics of compressors A and B.

Compressor	Fluid	Displacement (cm^3/rev)	Built in pressure ratio	Top gap (µm)	Flank gap (µm)
A	R404A	7.17	8.31	12	12
B	R410A	6.10	3.40	6	6

Table 2: Geometrical input data required in the thermal model.

Compressor	A_{cmp} (mm^2)	V_{suc} (mm^3)	A_{suc} (mm^2)	D_{suc} (mm)	V_{dis} (mm^3)	A_{dis} (mm^2)	D_{dis} (mm)
A	103399	463961	159382	11.65	115133	13832.6	33.29
B	104139	518767	123873	16.75	114505	14587.4	31.42

Table 3: Simulation results for compressor A.

Speed	T_e	T_c	T_{in}	T_{suc} [˚C]			T_{out} [˚C]		
				Exp.	Num.	Error (%)	Exp.	Num.	Error (%)
8000	-27	42	35.6	77.9	76.0	-2.4	143.6	174.6	21.6
10000	-27	42	34.4	75.8	71.9	5.2	143.0	169.7	18.7

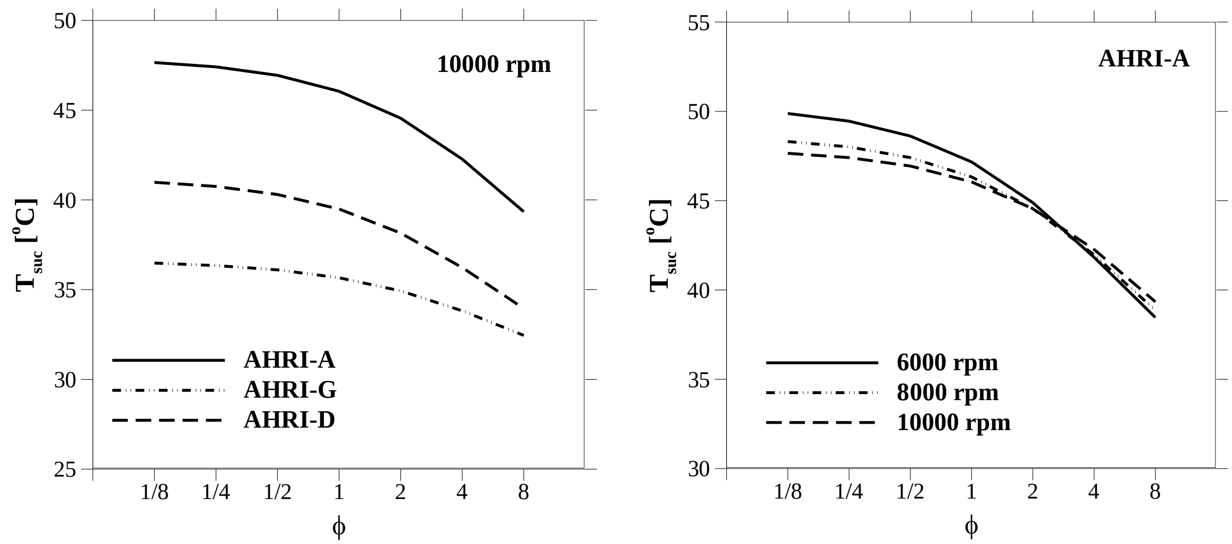
Table 4: Simulation results for compressor B.

Speed	Condition	T_e	T_c	T_{in}	T_{suc} [˚C]			T_{out} [˚C]		
					Exp.	Num.	Error (%)	Exp.	Num.	Error (%)
8000	AHRI-A	7.2	54.4	21.4	44.4	45.9	3.4	107.3	116.3	8.4
10000		7.2	54.4	20.7	44.0	47.3	7.5	109.2	118.5	8.5
8000	AHRI-D	-1.1	43.3	12.7	41.2	43.6	5.8	99.2	111.6	12.5
10000		-1.1	43.3	12.6	41.00	44.6	8.8	101.5	112.8	11.1
8000	AHRI-G	1.7	32.2	16.13	40.82	39.0	-4.5	83.5	89.6	7.3
10000		1.7	32.2	15.32	40.46	41.5	2.6	86.3	93.4	8.2

4.2 Model sensitivity analysis

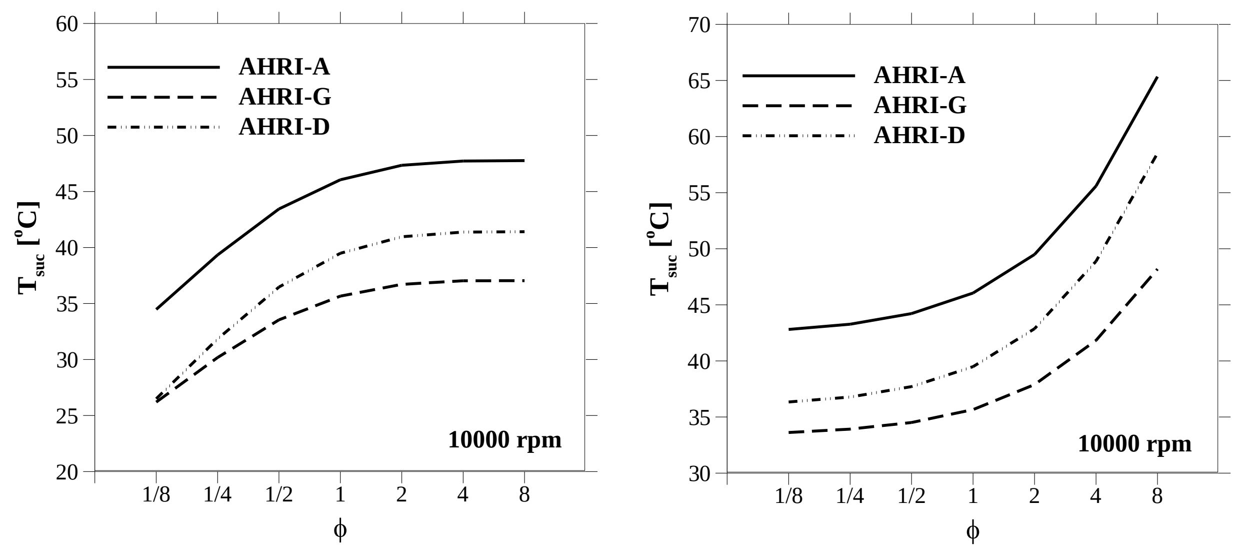
The thermal model proposed herein was adopted to analyze the influence of operating conditions on the suction temperature of compressor B. For instance, Figure 4 shows the effect of the convective interaction between the isothermal surface and the external environment, which could be an outcome of different cooling alternatives for the compressor shell. The baseline value estimated via Equation (4) for h_{amb} (around 20 W/m^{2}K), is multiplied by different factors ϕ (1/8, 1/4, 1/2, etc.). Figure 4(a) shows that the suction temperature is decreased as h_{amb} is increased, as one would expect. On the other hand, Figure 4(b) shows that the suction temperature is not significantly affected by the compressor speed.

Figure 5 shows predictions obtained when the values of h_{suc} and h_{dis} obtained from Equations (7) and (13) are multiplied by different factors ϕ. The values of such coefficients for the compressor operating under the AHRI-A condition at 10000 rpm are equal to 290 W/m^{2}K and 125 W/m^{2}K, respectively. One can observe from Figure 5(a) that the suction temperature is more influenced by variations in h_{suc} and h_{dis} at lower evaporating temperatures, such as in the AHRI-D condition. In addition, it is also noted that h_{dis} affects the suction temperature more strongly than h_{suc}.



(a) (b)

Figure 4: Sensitivity analysis - h_{amb} influence over suction temperature.



(a) (b)

Figure 5: Sensitivity analysis – h_{suc} and h_{dis} influence over suction temperature

4.3 Sensitivity to operating conditions

Figure 6 shows the variation of the suction temperature in relation to the condensing temperature at two different evaporating temperatures (T_e = -1.1 ^oC and 7.2 ^oC). As can be seen, the variation of the suction temperature is greater in the case of lower evaporating temperature. It is evident from Figure 6(a) that the suction temperature decreases with the compressor speed in the case of T_e = -1.1 ^oC, probably because leakage becomes less important. However, for the higher evaporating temperature, represented by T_e = 7.2 ^oC, leakage is not dominant and other effects are influential on the suction temperature. The increase of the suction temperature with pressure ratio indicates that absolute suction superheating ($\dot{Q}_{suc}$) also rises with pressure ratio, but in all conditions represented in Figure 6 the ratio $\dot{Q}_{suc}/W_{ele}$ was seen to remain almost constant in the range 35-40%.

Despite the continuous increase of the suction temperature with pressure ratio, Figure 7 shows that the isentropic efficiency, η_s, reaches its maximum value when the compressor compression ratio matches the pressure ratio established by the operating condition. As expected, at low-pressure ratios, over compression losses have significant impact on the isentropic efficiency, especially for the higher evaporating temperature. Similar negative effect on the compressor performance occurs in the case of under compression at high-pressure ratios.

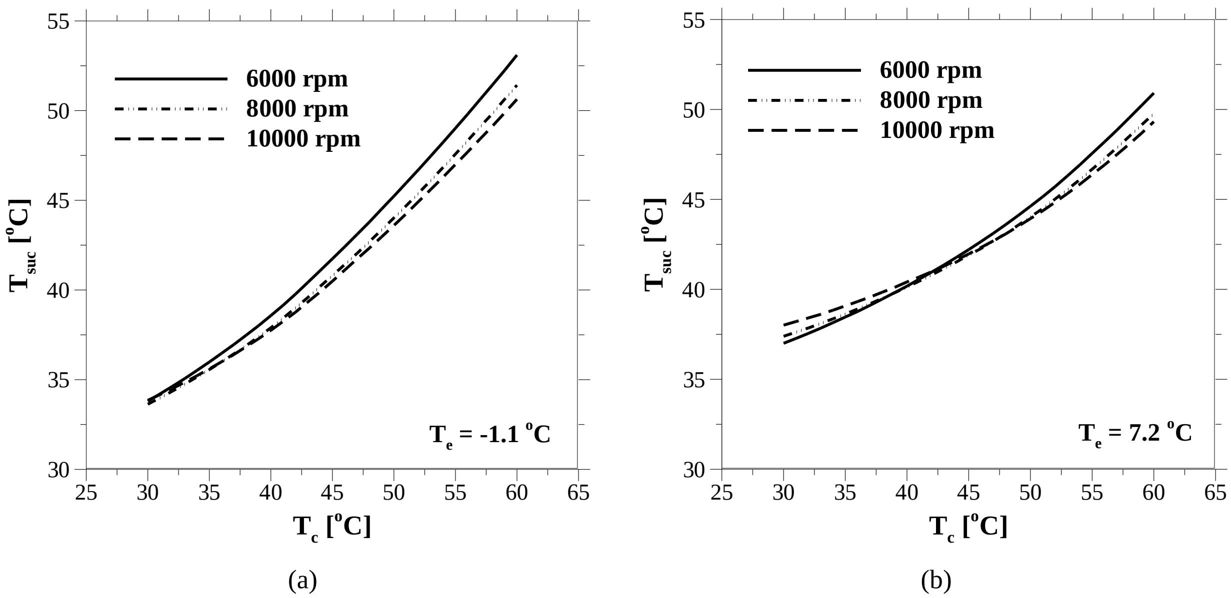


Figure 6: Suction temperature variation with condensing temperatures.

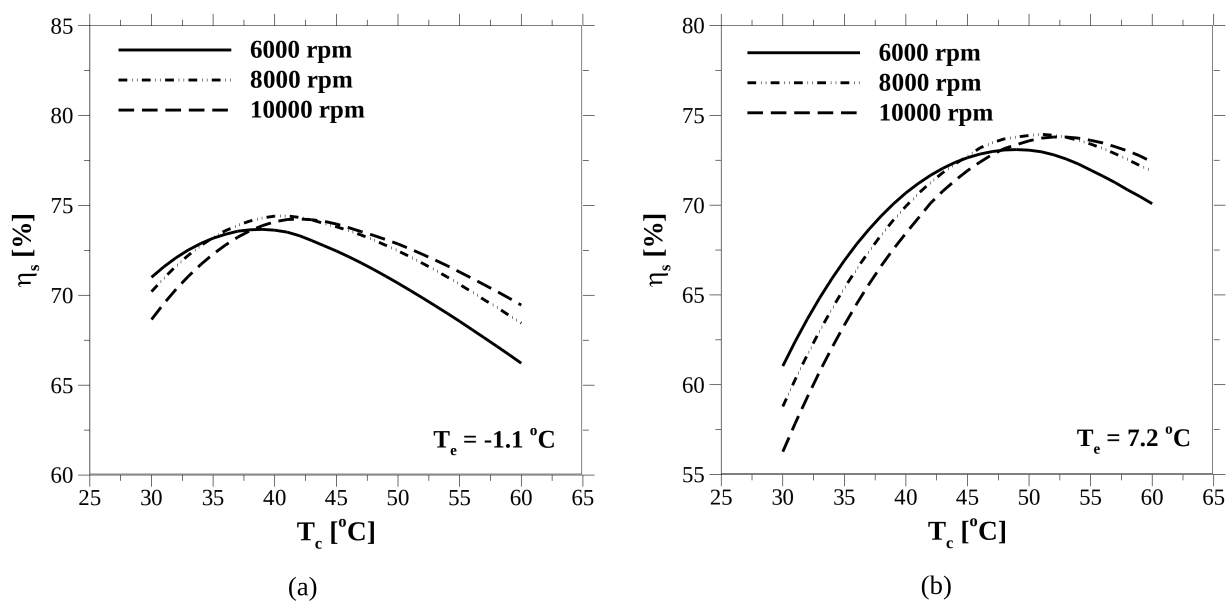


Figure 7: Isentropic efficiency variation with condensing temperatures.

5. CONCLUSIONS

This paper presented a simple approach to characterize suction superheating in scroll compressors. The model, based on the $\varepsilon - NTU$ method developed for heat exchangers, was coupled to a lumped model for the compression process. In comparison with the model of Winandy *et al.* (2002), the present model includes a lumped transient formulation for the compression process and correlations to estimate the required heat transfer coefficients. Simulations of two different compressors with different refrigerants showed that the model is capable to predict the suction temperature in good agreement with measurements without requiring any experimental calibration. However, the same level of agreement was not seen for the outlet temperature and different heat transfer correlations should be tested to improve the model. Finally, the model was used in sensitivity analysis regarding superheating as a function of geometrical parameters and operating conditions. In all situations, the model provided physically consistent predictions.

NOMENCLATURE

General and Greek symbols

A	Heat transfer area	(m^2)	Re	Reynolds number	( )
c_p	Specific heat	(J/kgK)	T	Temperature	(K)
D	Diameter	(m)	T_e	Evaporating temperature	(°C)
h	Heat transfer coefficient	(W/m^2K)	T_c	Condensating temperature	(°C)
k	Thermal conductivity	(W/mK)	V	Volume occupied by the gas	(m3)
$\dot{m}$	Mass flow rate	(kg/s)	$\dot{W}_{pv}$	Indicated Power	(W)
Nu	Nusselt number	( )			
Pr	Prandtl number	( )	ε	Effectiveness	( )
$\dot{Q}$	Heat transfer rate	(W)	μ	Dynamic viscosity	(Pa.s)
$\dot{Q}_{eml}$	Heat generation due to electromechanical losses	(W)	η_s	Isentropic efficiency	( )
$\dot{Q}_{chm}$	Heat transfer between tube surface and compression chamber	(W)	η_{eml}	Electromechanical efficiency	( )

Subscripts

amb	Related to external ambient	out	Related to compressor outlet
cmp	Related to the compressor	suc	Related to compressor suction
dis	Related to compressor discharge	w	Related to isothermal surface
in	Related to compressor inlet		

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