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ABSTRACT

Being part of a model which simulates the whole consecutive overall compression process in a scroll compressor by solving equations of mass, momentum and energy balance for fluid refrigerant (Rovira *et al.*,2006), an updated version is presented. In this new model, an energy balance over the scroll wraps is implemented; where temperatures and heat fluxes are obtained dividing the wall into 36 parts (slices) each turn.

The scroll wrap is divided into different solid slices; energy balance is carried out, taking into account: i) conduction along the scroll wrap; ii) convection heat transfer between each slice and each fluid chamber, with special attention on solid slice - fluid chamber contact at each time step.

The numerical model shows the one dimensional and transient temperature, pressure and mass flow rate, at each fluid chamber along the scroll compressor, among detailed solid wrap temperature distribution.

The whole numerical model has been experimentally validated against experimental data from technical literature (Halm,1997)(Chen *et al*.,2004a)(Chen *et al*.,2004b), comparing mass flow rate, discharge temperature, compression work and power consumption.

Finally, the influence of wall temperatures and wall heat fluxes on the compressor performance and other output variables is analyzed.

1. INTRODUCTION

Following the previous work by (Rovira *et al*,2006), an improvement has been introduced focused in wall thermal behavior in wrap scroll. The main objective is: The very usual hypothesis to simulate the wall temperature as a lineal with angle between suction at entry (T_s) and discharge at center (T_d) is it correct?

This lineal simulation is simple to program and calculate, but it is a bit inexact because when we force the wall slices to have a temperature values, they are not in balance of heat fluxes, but there are net fluxes to or from each slice (which affect the fluid). At the end of a turn, the temperatures are again forced linearly from the new T_s and T_d. However the simulation converges for the simple reason that it does not change form iteration to next. But this does not mean they are in thermal equilibrium.

(Sunder,1997) did a complete treatment of wall heat exchanges and presented results for one compressor. (Ishii *et al*., 2002a) and (Ishii *et al*. 2002b) used the wall lineal temperature approach from measurements. (Jang&Jeong,1999) published accurate experimental measures with lineal distribution of wall temperatures and heat fluxes but they do not compare with their numerical model. (Sun *et al*., 2010) did a wide experimental and numerical study isothermalizing the fixed scroll wall and the fixed basis. They used the lineal model (T_s,T_d) for the orbiting scroll wall. Thus we think the difference between the two numerical models is not fully answered.

In our work we compare the lineal model of scroll wall slices temperatures between T_s and T_d, with the same cases but letting each slice be in thermal equilibrium of heat fluxes, and analyzing the temperatures profile, comparing the difference and presenting its effect on Compression Work, Input Power, Mass Flow rate, fluid Discharge Temperature (T_d) and fluid exit compressor Temperature (T_{gas}).

2. MODEL DESCRIPTION

The previous model (Rovira *et al*.,2006) included mass, momentum and energy balance for fluid, at entry pipe, at scroll inside, and outside at the compressor solid lumps fluid contact, (including leakages and heat interchanges), (see Figure 1). Compressor solids were considered one block each, with heat accumulation to steady state. Heat exchanges between compressor lumps were calculated supposing thermal resistances as (Chen,2004,ab) in all cases. So, all solids get thermal equilibrium from heat fluxes, changing its temperatures, except scroll walls which temperatures are supposed lineal with angle from T_s to T_d. That is unsatisfactory because we were forcing temperature values at the end of each turn, but they were far from thermal equilibrium. However, it converges only because the simulation values do not change from iteration to next one, despite heat fluxes balance was not zero in steady state. To evaluate if the lineal approach is good enough, a wall wrap model is made and presented here.

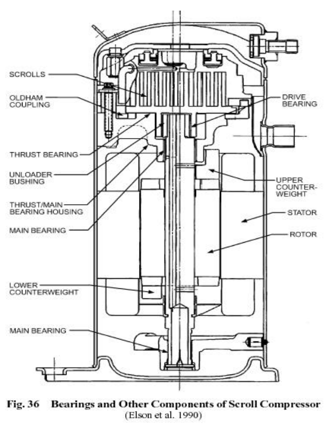


Figure 1: ASHRAE Handbook 2008 HVAC Systems and Equipments

In this updated version, scroll walls are divided into slices, each one being in thermal equilibrium, when the simulation converges. There are a geometrical model and a thermal model. Geometrically scroll walls are divided in trapezoidal volumes with six flat borders, two side neighboring slices, two fluids (an internal and external chamber), and top and bottom borders which are considered isolated from the basis and thermally separated from the basis (adiabatic); and friction affects only the bases. Thus thermally wall interactions are considered only between scroll slices and fluid, as well as slices with their neighbors. In this model version the compressor is symmetrical and walls are both with the same metal (all aluminum or all steel). Wall temperatures are recalculated at the end of each turn, being constant during whole time steps of turn. This is a good approach because slices temperatures change slowly in comparison to fluid temperature. The two instant heat fluxes for every slice (see Figure 2b) are calculated every time step:

$$Q_N^{K\theta}=\alpha_{j+1}^{\theta}A_N^K\,(T_{ll}^K-T_{j+1}^{\theta}) \tag{1}$$

$$Q_S^{K\theta}=\alpha_j^{\theta}A_S^K\,(T_{ll}^K-T_j^{\theta}) \tag{2}$$

At the end of the whole turn, heat fluxes across slices are added:

$$Q_N^K = \Sigma_\theta Q_N^{K\theta} \tag{3}$$
$$Q_S^K = \Sigma_\theta Q_S^{K\theta} \tag{4}$$

As well as heat fluxes along wrap scroll calculation at end of turn:

$$Q_W^K = -\lambda A_{ll} (T_{ll}^{K-1} - T_{ll}^K) / ((\delta^{K-1} + \delta^K)/2) \tag{5}$$

$$Q_E^K = -\lambda A_{ll} (T_{ll}^{K+1} - T_{ll}^K) / ((\delta^{K+1} + \delta^K)/2) \tag{6}$$

With four fluxes it is possible to calculate next slice temperature from:

$$\rho^K C_p^K ((T_{ll}^K - T_{ll}^{Ko})/\Delta t) V^K = Q_N^K + Q_S^K + Q_W^K + Q_E^K \tag{7}$$

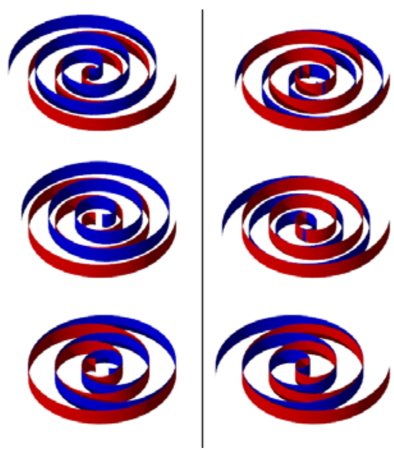


Figure 2.a: Scroll wall pairs.

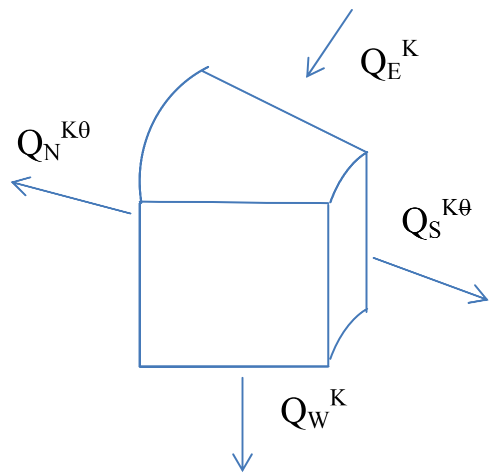


Figure 2.b: Wall Slice instant heat fluxes

With this information and previous slices temperatures, T_{ll}^K can be recalculated simultaneously, using a TDMA algorithm. Details are in (Patankar,1980). Geometry is solved once at the beginning and the solution put in tables. Observe that the slices per chamber change at each time step (See Figure 2a). Two new slices are added to chamber J with angle increment and two others are left behind. The knots (border between chambers) coincide with slices borders, for simplification. And slice "K" is in front of slice "K+180º" also for simplification. This last simplification is acceptable despite of scroll motion eccentricity because the main lateral displacement happens when the chamber is wide, whilst when chamber is narrow coincides with slice in front its counterpart. Only convection is considered between fluid-wall and conduction wall-wall.

$Q_{in}^{J,\theta}$ and $Q_{out}^{J,\theta}$(where in and out corresponds to internal and external borders of chambers, respectively), are obtained adding instant heat fluxes for every time step (angle), every chamber, adding each slice heat flux belonging to chamber J.

$$Q_{in}^{J,\theta} = \Sigma_K Q_N^{K\theta} \tag{8}$$

$$Q_{out}^{J,\theta} = \Sigma_K Q_S^{K\theta} \tag{9}$$

The moment to fill $Q_{in}^{J,\theta}$ and $Q_{out}^{J,\theta}$ is in the function which calculates fluid temperature every chamber. Fluid temperature and instant wall-fluid heat fluxes interact with both each other, which implicates an interactive process: $Q_{in}^{J,\theta}$, $Q_{out}^{J,\theta}$ vs $T^{J,\theta}$. The whole fluid chamber is considered to have the same temperature $T^{J,\theta}$.

3. VALIDATION AND NUMERICAL RESULTS

OP1 Tgas=80ºC Ts,Tdmodel Td=70.45ºC Tgas=73.36ºC Tll,beqmodel Td=77.67ºC Tgas=77.11ºC	OP2 Tgas=92ºC Ts,Tdmodel Td=92.52ºC Tgas=90.80ºC Tll,beqmodel Td=98.60ºC Tgas=93.89ºC	OP3 Tgas=105ºC Ts,Tdmodel Td=115.87ºC Tgas=108.09ºC Tll.beqmodel Td=120.33ºC Tgas=110.24ºC
OP4 Tgas=69ºC Ts,Tdmodel Td=54.11ºC Tgas=62.01ºC Tll.beqmodel Td=61.17ºC Tgas=66.30ºC	OP5 Tgas=81ºC Ts,Td model Td=72.21ºC Tgas=75.69ªC Tll,beq model Td=78.82ºC Tgas=79.66ºC	OP6 Tgas=91ºC Ts,Td model Td=91.25ºC Tgas=88.98ºC Tll,beq model Td=96.79ºC Tgas=92.25ºC
OP7 Tgas=63ºC Ts,Tdmodel Td=45.04ºC Tgas=60.22ºC Tll,beqmodel Td=53.47ºC Tgas=66.42ºC	OP8 Tgas=73ºC Ts,Td model Td=59.93ºC Tgas=68.52ºC Tll,beq model Td=66.65ºC Tgas=73.19ºC	OP9 Tgas=83ºC Ts,Td model Td=76.24ºC Tgas=78.91ºC Tll,beq model Td=82.40ºC Tgas=83.09ºC

Table 1: Numerical comparison with experimental temperatures from (Halm,1997).

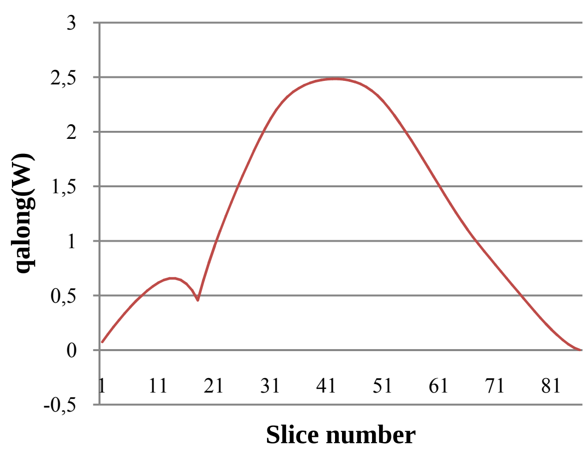
Pexp = 840W P(Ts,Td)= 831W (341W) P(beq) = 837W (344W)	Pexp = 955W P(Ts,Td) = 965W (431W) P(beq) = 968W (433W)	Pexp = 1074W P(Ts,Td) =1087W (523W) P(beq) =1086W (522W)
Pexp = 817W P(Ts,Td) = 785W (325W) P(beq) = 793W(328W)	Pexp = 936W P(Ts,Td) = 902W (440W) P(beq) = 908W (443W)	Pexp = 1055W P(Ts,Td) = 998W(555W) P(beq) =1001W(557W)
Pexp = 804W P(Ts,Td) = 887W(295W) P(beq) = 914W(304W)	Pexp = 923W P(Ts,Td) = 908W(412W) P(beq) = 919W(417W)	Pexp = 1029W P(Ts,Td) = 979W(555W) P(beq) = 985W(559W)

Table 2: Numerical comparison with exp. vs. num. input Power and Compression Work

Mexp = 44Kg/h M(Ts,Td)= 36Kg/h M(beq) = 36Kg/h	Mexp = 43Kg/h M(Ts,Td) = 35Kg/h M(beq) = 35Kg/h	Mexp = 42Kg/h M(Ts,Td) = 34Kg/h M(beq) = 34Kg/h
Mexp = 62Kg/h M(Ts,Td) = 53Kg/h P(beq) = 53 Kg/h	Mexp = 60 Kg/h M(Ts,Td) = 53Kg/h M(beq) = 52Kg/h	Mexp = 59Kg/h M(Ts,Td) = 50Kg/h M(beq) = 50Kg/h
Mexp = 83 Kg/h M(Ts,Td) = 74Kg/h M(beq) = 74Kg/h	Mexp = 81Kg/h M(Ts,Td) = 73Kg/h M(beq) = 73Kg/h	Mexp = 79Kg/h M(Ts,Td) = 72Kg/h M(beq) = 71Kg/h

Table 3: Numerical and experimental Mass Flow Rate comparison

Heat flux slice to slice qalong (Q_W^K and Q_E^K) can be put in a graphic and slice temperatures too. All slices are numbered from suction to discharge, in (degrees/10). Slices are numbered from 1 the most external (at φ_e) to 86 the most inner (at φ_s). For OP5 and OP7 as illustration:



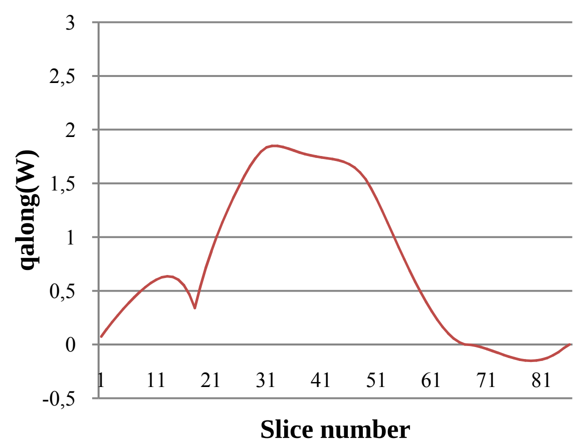
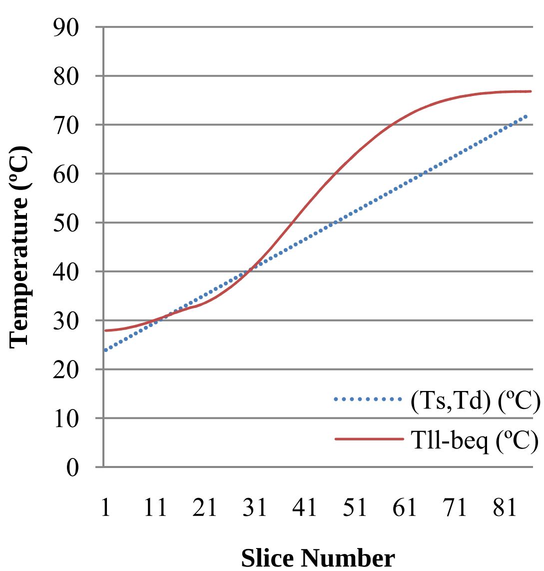


Figure 3.a: qalong between slices in Watt for case OP5. **Figure 3.b**: qalong between slices in Watt for case OP7

In OP7 It can be seen a bit of negative heat flux (from outer to center) as (Jang&Jeong,1999). All simulations have been done with R22, and with the compressor of (Halm,1997), thus we compared his results with ours, with both models. Here for cases OP5 and OP7, there are wall temperatures in function of angle.



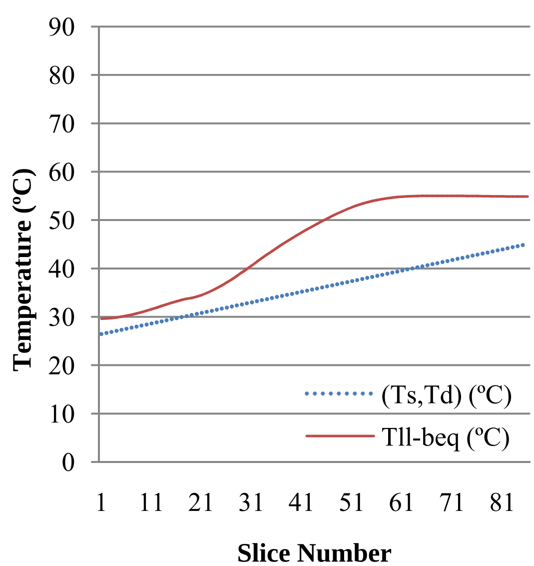
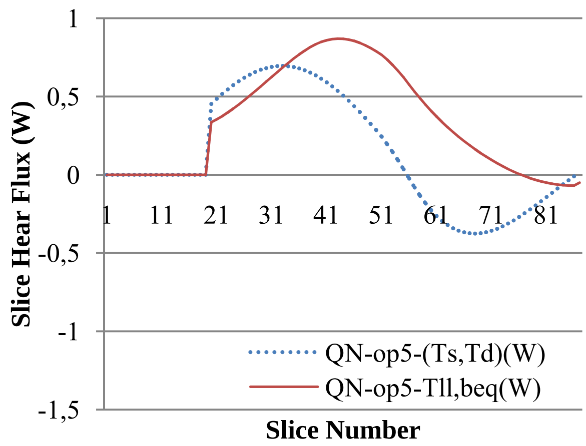


Figure 4.a: Central case OP5 Wall Temperatures in both models. **Figure 4.b**: Case OP7 Wall Temperatures in both models.

Maximum difference is situated just finishing compression.



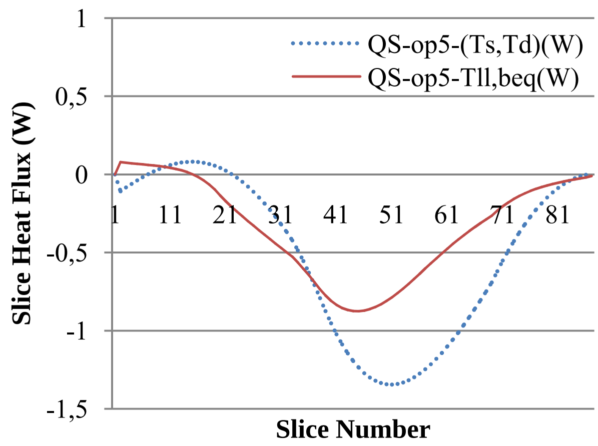


Figure 5.a: and **Figure 5.b**: Slices heat fluxes to exterior and to interior in Watt, both models, accumulated in one turn. Entry channel is considered adiabatic.

The precedent graphics are from a point of view of wall 86 slices. Following two graphics are from chambers fluid heat flow point of view, 3 turns, 108 (degrees/10).

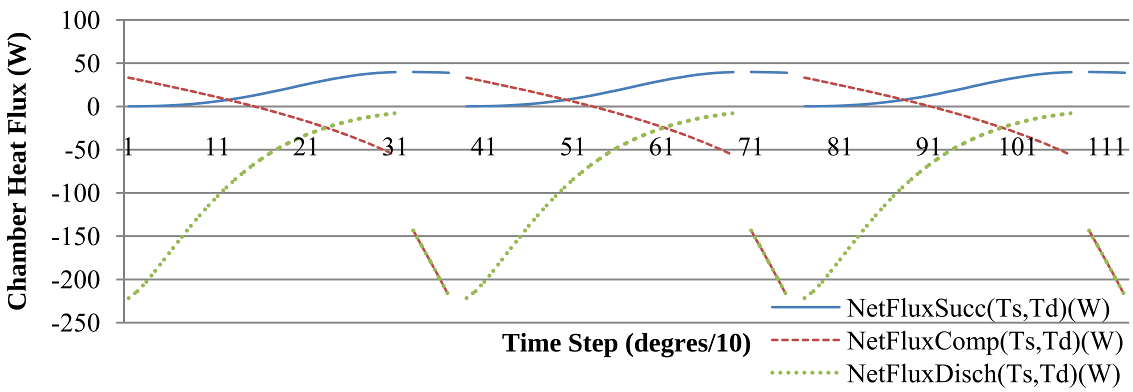


Figure 6.a: Net heat fluxes in lineal model. Discontinuities are deliberated and correspond to moment of suction creation and discharge-compression fusion. Graphics of 108 (degrees/10), 3 turns.

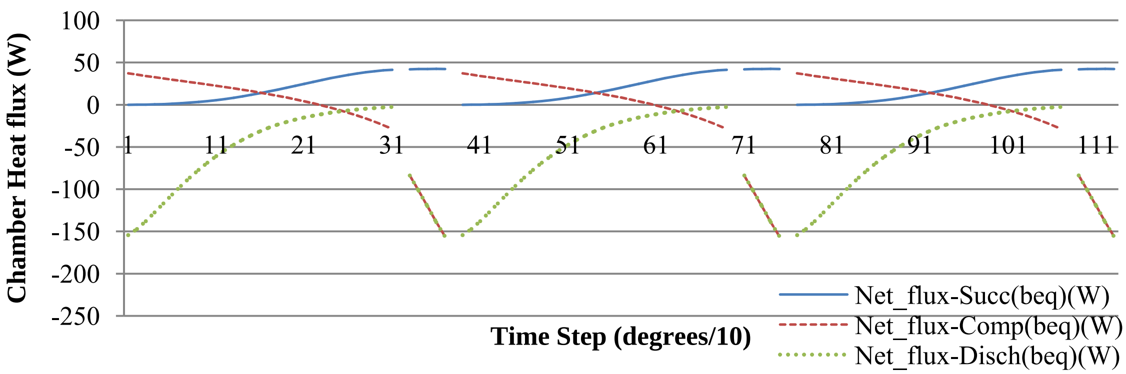


Figure 6.b: Net heat fluxes in beq model. Discontinuities are deliberated and correspond to moment of suction creation and discharge-compression fusion. Graphics of 108 (degrees/10), 3 turns.

In second figure Heat is a more balanced one. In the first model there is a net flux from fluid to wall, because walls are colder in this model. If we let the equilibrium, walls are hotter and heat fluxes balanced.

In lineal model (T_s,T_d) there is a net heat flux from fluid to walls of -65W for OP5 case, with a maximum of -115W for OP7 case; thus calculated fluid T_d and T_{gas} temperatures are inferior compared to heat balance of equilibrium (beq) calculus which net heat flux from fluid to walls is nearly zero in steady state. Discharge Temperature at exit scrolls(T_d) is +6.6ºC higher and exit compressor temperature(T_{gas}) is +4.0ºC higher in thermal equilibrium than in lineal model; and finally Mass flow rate is 0.20Kg/h a bit lower in thermal equilibrium than in lineal model.

To test the influence of Wall Temperature on input Power and Discharge Temperature, a simulation has been carried out, with the following results: ΔCompressionWork=+8W and ΔPower=+16W; ΔT_d=+8ºC and ΔT_{gas}=+5ºC; all for ΔT_{ll}=+10ºC.

The influence of T_{entry} on input Power is scarcely -1W for +10ºC increase; and the influence of T_{entry} on T_d is +10ºC for +10ºC increase.

Mass flow rate decrement (-1%) due T_{ll} increment (+10ºC) and mass flow rate decrement (-4%) due T_{entry} increment (+10ºC) are the other effect.

4. CONCLUSIONS

An optional improvement of wall thermal model has been added to (Rovira *et al*.,2006), similar to (Sunder,1997), but integrated with the thermal fluid model including leakages and compressor lumps heat exchanges. It has been validated against experimental results of Halm'1997. It also clarifies an important doubt, under the usual approximation of wall temperature being forced to be lineal with angle(θ) between fluid suction (T_s) and discharge (T_d) temperatures, the wall slices were not in thermal equilibrium, and the discordance gets a maximum heat flux of 115W for OP7 case and 65W for the central OP5 case from fluid to walls.

The new calculated equilibrium wall temperature distribution is considerably over the lineal approach. The calculated indicated-compression-work is similar but not equal, around 3W higher with a maximum difference of 9W in case OP7. Thus the input Power in both calculus have a difference of 6W for the central case OP5 and 27W for max case OP7, based on the (Halm,1997) efficiency formula.

The major part of wall heat flux difference comparing both calculi is manifest to discharge temperature difference and the minor part to compressor work difference.

NOMENCLATURE

α	Heat Transfer Coefficient	[W/(m^2 K)]
A	Area	[m^2]
T_{LL}	Slice Temperature	[K]
T	Fluid Temperature	[K]
Q	Heat Flux	[W]
λ	Heat Solid Conductivity	[W/(K m)]
δ	Slice width at gravity center	[m]
C_p	Heat Coefficient	[W/(Kg K)]
V	Volume	[m^3]
Δt	Time increment	[s]

Subscripts

J:	(Ncc+1..0)	Chamber index
K:	1..nslice	Slice index
θ:	1..360º	Angle index
N:	(outer)	North slice face
S:	(inner)	South slice face
W:	(near suction)	West slice face
E:	(near discharge)	East slice face
o:		Previous period

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