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MATHEMATICAL MODEL OF A COMPLETE VAPOR COMPRESSION REFRIGERATION SYSTEM WITH HELICAL COIL EVAPORATOR FLOODED IN THE WATER

Summary

Refrigeration cycle system modelling of the vapor compression experimental unit with the goal to predict the system performance of the cycle and geometry of the helical coil evaporator flooded in the water is presented in the paper. Design of the experimental unit is based on the commercially available scroll compressor and air cooled condenser. In order to determine the thermodynamic conditions of refrigeration cycle and heat transfer process in the evaporator the simulation model is developed. The model takes into account the specific data, dimensions and characteristics of the main components. Evaporation process, observed in three parts, and condensation process are described with appropriate heat transfer correlations. With two approximation functions, developed based on manufacturer data, the model of compression process is described. Results show relations between thermal resistance and geometrical quantities of evaporator with influence on the system performance. Analysis of thermal resistance shows that geometry of the evaporator may have important effect on the final design of these types of refrigeration applications.

Key words: *Refrigeration device; Mathematical model; Evaporation process; System design; Thermal resistance;*

1. Introduction

In many engineering applications, such as refrigeration and HVAC systems, the helically coiled tubes are used. There is very little information available in the literature related to the refrigeration system applications in which the helically coiled tube has the function of an evaporator flooded in the water. Researchers mainly gave the results obtained from the heat transfer correlations applied inside the tube and on the outer surface of the helical coil. Some interesting results can be highlighted. Jitian et al. [1] and Zhao et al. [2] proposed the heat transfer correlations for the boiling of refrigerant inside the helically coiled tubes. Ali [3] experimentally investigated and suggested the natural convection heat transfer correlation applied on the water side around the tube.

Similar methodology of refrigeration system modeling is published by Elsayed [4], where the simulation results of system performance of miniature refrigeration device with helically coiled evaporator is compared with the obtained experimental results, inside 5 % of difference for this type of evaporator. Good results with similar mathematical modeling methodology are presented by authors in reference [5], where validated mathematical model on experimental results of refrigeration device is used for “drop-in” analysis and system performance with alternative refrigerants.

Special attention should be paid to system design or redesign according to the heat transfer of helically coiled evaporator. If the water is used as cooling or heating medium in the refrigeration process, this negative effect can be especially expressed. This is pointed out by Wang [6], who asserts that the heat transfer effects of the refrigerant might be important in some circumstances, like those found in systems with water cooled condenser, where the dominant thermal resistance may be lower on refrigerant side.

In this paper the mathematical model of a complete vapor compression refrigeration system with specially designed helical coiled evaporator flooded in the water is presented. Condensation and evaporation process is developed using appropriate heat transfer correlations. Mathematical model of compression process is developed based on the manufacturer data of compressor. Using the model, simulation results of system performance, influence of the geometrical quantities and thermal resistance are observed, what was the aim of the paper from one side, and from another to present the mathematical modeling of refrigeration system where the model can be used for system design or redesign, system performance analysis, drop-in analysis, heat transfer features or some other system analysis.

2. Description of experimental device

The experimental unit with evaporator flooded in the water, schematically shown in Figure 1, is planned to be made and used for different kind of analyses with focus on the evaporation process.

A commercial hermetic rotary (scroll) compressor is used, with a displacement volume of 12 cm^3. Compressor is made for refrigerant R410A. Commercial thermal expansion valve is used for controlling the degree of superheating at the compressor inlet. The air cooled condenser is a heat exchanger consisting of 20 horizontal copper tubes, 780 mm long, inner diameter 7.92 mm and outer diameter 9.52 mm. Across the tubes, 500 aluminum fins are placed. Cooling medium (air) flows through the annular space forced by air fan. Inner diameter of the water tank is 500 mm with 100 liters volume.

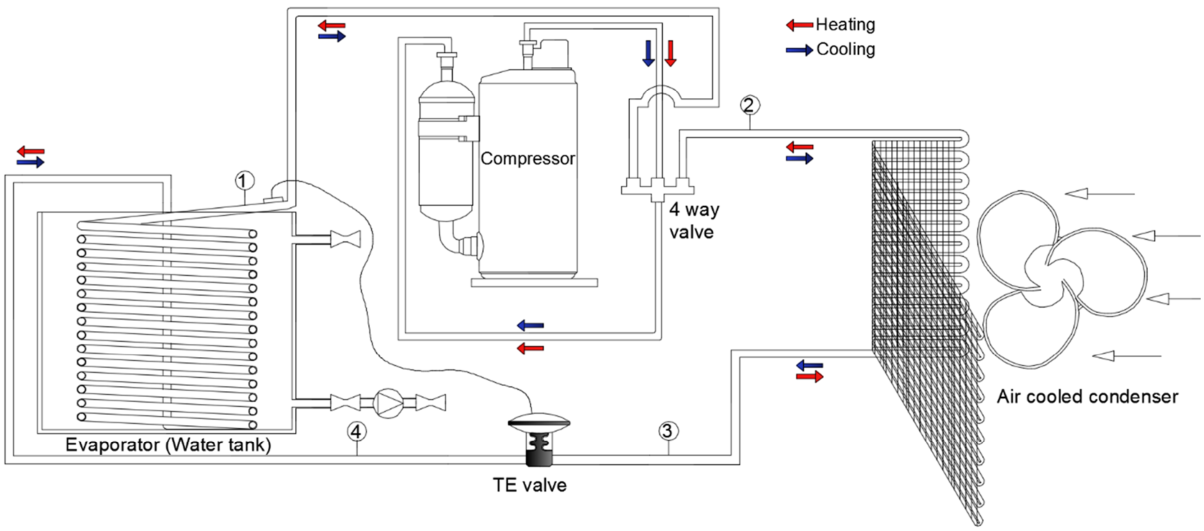

Fig. 1 Schematic representation of experimental device

In this paper, with the goal to present the results with influence on design of this heat exchanger type, the geometry of evaporator is a variable parameter. However, the following text presents adopted geometry for experimental procedure.

The evaporation process takes place in a vertically oriented helically coiled copper tube of 13 mm inner diameter (15 mm outer diameter), 15,000 mm long and consisted of 15 coils, where the angle of the coil is 2°. The coil diameter is 300 mm and coil pitch of 32 mm, how is also presented on the next Figure 2.

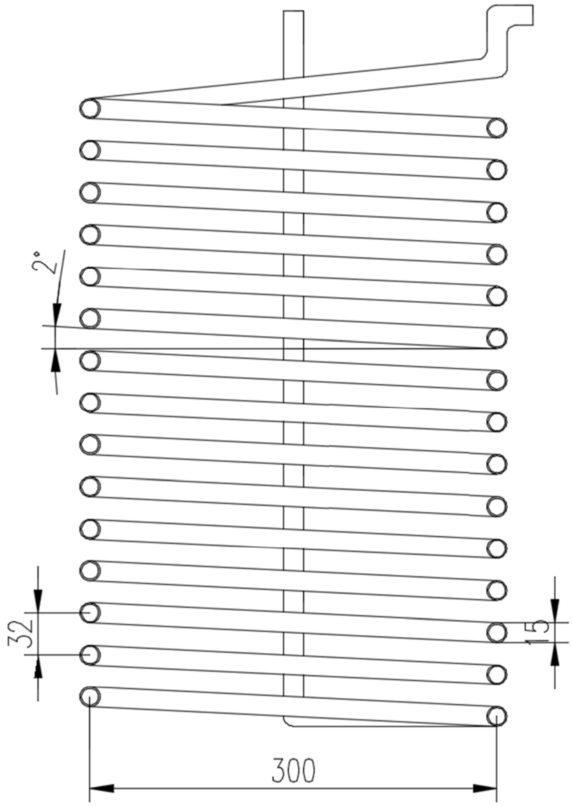

Fig. 2 Schematic representation of helical coil evaporator

The total cooling capacity of the evaporator is equal to the cooled water capacity. The refrigerant enters to the evaporator at the top of the vertical tube and flows to the bottom of the tube, then to the top across helical coil as two phase medium and leaves the coil as superheated vapor.

3. Mathematical model

Mathematical model takes into account specific data and dimensions of components to predict the operating conditions of the experimental unit. The following assumptions are set:

1. The pressure drops in the refrigerant pipes are negligible;
2. The heat losses in refrigerant pipes to surroundings are negligible;
3. Heat transfer between water tank and surroundings is negligible.

Mathematical model follows the system modelling conception from reference [5].

3.1 Compressor

Compression process is described with volumetric efficiency, to define refrigerant mass flow, and overall efficiency to obtain electric power for compressor process. Both are obtained from manufacturer report data for the mass flow rate and power consumption of the compressor for frequency value of 50 Hz, condensation temperatures of 50 °C, 55 °C, 60 °C and 65 °C and evaporating temperatures range from 0°C to 10°C, measured according to the

ASHRAE conditions. For both efficiencies higher order polynomial fit should be considered. Equation (1) is the result of the regression analysis with coefficient of determination R^2 of 95% for volumetric efficiency as a function of pressure ratio:

$$\eta_{\mathrm{vol}} = -0.0044 \cdot RC^3 + 0.0558 \cdot RC^2 + 0.2779 \cdot RC + 1.1454. \tag{1}$$

The overall efficiency also was assumed to be only a function of the pressure ratio, and the effect of the frequency was neglected. Equation (2) is the result of the regression analysis with coefficient of determination R^2 of 93 %:

$$\eta_{\mathrm{ov}} = 0.0031 \cdot RC^3 - 0.0412 \cdot RC^2 + 0.1263 \cdot RC + 0.5302\,. \tag{2}$$

Refrigerant mass flow rate is obtained from the compressor displacement $\left(V_{\mathrm{cyc}}\right)$, compressor rotation speed $\left(N\right)$, volumetric efficiency $\left(\eta_{\mathrm{vol}}\right)$, and the refrigerant density at the suction line $\left(\rho_1\right)$:

$$\dot{m}_{\mathrm{ref}} = V_{\mathrm{cyc}} \cdot \eta_{\mathrm{vol}} \cdot \rho_1 \cdot \left(\frac{N}{60}\right). \tag{3}$$

According to assumption 1, condensation and evaporation pressures define the pressure ratio:

$$RC = \frac{p_{\mathrm{C}}}{p_{\mathrm{E}}}. \tag{4}$$

The compressor rotation speed (rpm) is related to the supply frequency of the compressor:

$$N = \frac{f_{\mathrm{comp}}}{P_{\mathrm{m}}} \cdot \left(1 - s_{\mathrm{m}}\right) \cdot 60\,, \tag{5}$$

where, in this case, the number of phases and number of magnetic pole per phase is equal to 1, and slip $\left(s_{\mathrm{m}}\right)$ has been assumed to be 5 %, which is the typical value for small power motors.

Total energy supply for compression process is obtained from overall efficiency and isentropic power of compressor:

$$\dot{W}_{\mathrm{elec}} = \frac{\dot{m}_{\mathrm{ref}} \cdot \left(h_{2,\mathrm{is}} - h_1\right)}{\eta_{\mathrm{ov}}} = \frac{\dot{W}_{\mathrm{is}}}{\eta_{\mathrm{ov}}}. \tag{6}$$

3.2 Thermostatic valve

It is assumed that the thermostatic valve is perfectly insulated, and the following is valid: $h_{\mathrm{C,out}} = h_{\mathrm{E,in}}$. The thermostatic valve keeps constant the degree of superheating at the evaporator outlet:

$$T_1 = T_{\mathrm{sat}}(p_{\mathrm{E}}) + \Delta T_{\mathrm{sup}}\,. \tag{7}$$

3.3 Evaporator

3.3.1 Refrigerant side

The evaporation process unrolls inside the vertically oriented helically coiled copper tube. Evaporation total heat transfer rate of refrigerant side is given by following equation:

$$\dot{Q}_{\mathrm{E}} = \dot{m}_{\mathrm{ref}} \cdot (h_1 - h_4)\,. \tag{8}$$

Evaporation heat transfer process is considered in the three parts, presented with next three equations:

$$\dot{Q}_{\mathrm{E}} = \dot{Q}_{\mathrm{(nb+sp)}} + \dot{Q}_{\mathrm{sup}}\,, \tag{9}$$

$$\dot{Q}_{\mathrm{(nb+sp)}} = \dot{Q}_{\text{vertical pipe (nb+sp)}} + \dot{Q}_{\text{helical coil (nb+sp)}}\,, \tag{10}$$

$$L_{\mathrm{E}} = L_{\text{vertical pipe (nb+sp)}} + L_{\text{helical coil (nb+sp)}} + L_{\mathrm{sup}}\,. \tag{11}$$

<u>Vertical pipe</u>

The boiling process inside the vertical tube of the evaporator is modeled using additive formula recommended by Rohsenow and Griffith [7], equations (12), (13) and (14):

$$\dot{Q}_{\text{vertical pipe (nb+sp)}} = \dot{Q}_{\text{vertical pipe (nb)}} + \dot{Q}_{\text{vertical pipe (sp)}}\,, \tag{12}$$

$$q_{\text{vertical pipe (nb+sp)}} = q_{\text{vertical pipe (nb)}} + q_{\text{vertical pipe (sp)}} = \alpha_{\text{vertical pipe (2ph)}} \cdot (T_{\mathrm{E,t,i}} - T_{\mathrm{E}})\,, \tag{13}$$

$$\alpha_{\text{vertical pipe (2ph)}} = \alpha_{\text{vertical pipe (nb)}} + \alpha_{\text{vertical pipe (sp)}}\,. \tag{14}$$

Heat transfer coefficient during boiling ($\alpha_{\text{vertical pipe (nb)}}$) is calculated from correlation by Rohsenow [8] based on the assumption that the bulk of liquid is stationary. The heat transfer coefficient during forced flow of liquid ($\alpha_{\text{vertical pipe (sp)}}$), based on the assumption that a single-phase liquid flow at evaporation temperature flows inside the tube, is calculated from Dittus-Boelter correlation [9], equation (15):

$$q_{\text{vertical pipe (sp)}} = \alpha_{\text{vertical pipe (sp)}} \cdot (T_{\mathrm{E,t,i}} - T_{\mathrm{E}})\,. \tag{15}$$

<u>Helical coil</u>

Widely used methodology to predict heat transfer coefficient of two phase (boiling) flow in helical coil is used. The Martinelli number for turbulent flow is defined as:

$$\chi_{\mathrm{tt}} = \left(\frac{1-X}{X}\right)^{0,9} \left(\frac{\rho_{\mathrm{V}}}{\rho_{\mathrm{L}}}\right)^{0,5} \left(\frac{\lambda_{\mathrm{L}}}{\lambda_{\mathrm{V}}}\right)^{0,1}. \tag{16}$$

The two-phase heat transfer coefficient is calculated as a ratio of the liquid-only single phase heat transfer coefficient obtained by Jitian et al. [1]:

$$\frac{\alpha_{\text{helical coil (2ph)}}}{\alpha_{\text{helical coil (LO)}}} = 2,8446 \cdot \left(\frac{1}{\chi_{\mathrm{tt}}}\right)^{0,27} + \left(46162 \cdot Bo^{1,15} - 0,8762\right), \tag{17}$$

where (Bo) is Boiling number defined as:

$$Bo = \frac{q_{\text{helical coil}}}{G \cdot \Delta h_{\mathrm{E,V\text{-}L}}}\,. \tag{18}$$

Liquid only Reynolds number is calculated from the following equation:

$$\mathrm{Re}_{\mathrm{LO}} = \frac{G \cdot d_{\mathrm{E,i}}}{\mu_{\mathrm{L}}}\,, \tag{19}$$

where (G) is refrigerant mass velocity defined as:

$$G = \frac{\dot{m}_{\text{ref, V}} + \dot{m}_{\text{ref, L}}}{A_{\text{E,i}}} . \tag{20}$$

Nusselt number for liquid only phase in the helical coil is calculated from the following correlation obtained by Zhao et al. [2]:

$$Nu_{\text{LO}} = 0{,}023 \cdot \text{Re}_{\text{LO}}^{\;0{,}8} \cdot \text{Pr}_{\text{L}}^{\;0{,}4} \left[\text{Re}_{\text{LO}}^{\;0{,}05} \cdot \left(\frac{d_{\text{E,i}}}{d_{\text{coil}}} \right)^{0{,}1} \right] . \tag{21}$$

<u>Superheated phase</u>

Heat transfer coefficient for superheating process (α_{sup}) is calculated from Petukhov's [10] and Gnielinski's [11] correlation. Specific heat flux of superheated phase is:

$$q_{\text{sup}} = \alpha_{\text{sup}} \cdot (T_{\text{E,t,i}} - T_1) . \tag{22}$$

3.3.2 Water side

It is assumed that the heat transfer rate from the water to the refrigerant, according to the assumption 3, is equal to the total heat transfer rate of the evaporator. Next equation defines the time required for cooling water to achieve desired temperature:

$$\dot{Q}_{\text{w}} = \dot{m}_{\text{w}} \cdot c_{\text{p,w}} \cdot \Delta T_{\text{w}} = \dot{Q}_{\text{E}} . \tag{23}$$

For the calculation and theoretical possibility to obtain the steady state conditions the very small volume flow rate, to inside and from outside of the water tank, is appointed. From this point of view is considered that we have very small velocities of the water and natural convection around the evaporator tubes can be adopted.

Relation between water side and refrigerant is set by following equation:

$$\dot{Q}_{\text{w}} = \dot{Q}_{\text{E}} = k_{\text{E,o}} \cdot A_{\text{E,o}} \cdot \Delta T_{\text{w,ln}} , \tag{24}$$

where the logarithmic mean temperature difference between refrigerant and water is calculated as simplified model for the water tank using the next equation:

$$\Delta T_{\text{w.ln}} = \frac{\left(T_{\text{tank,start}} - \Delta T_{\text{w}}\right) - \left(T_{\text{E,ref,out}} - T_{\text{E,ref,in}}\right)}{\ln\left(\dfrac{\left(T_{\text{tank,start}} - \Delta T_{\text{w}}\right)}{\left(T_{\text{E,ref,out}} - T_{\text{E,ref,in}}\right)} \right)} , \tag{25}$$

and overall heat transfer coefficient for the outer surface is calculated with the next equation:

$$k_{\text{E,o}} = \frac{1}{\dfrac{r_{\text{E,o}}}{r_{\text{E,i}} \cdot \alpha_{\text{helical coil (2ph)}}} + \dfrac{r_{\text{E,o}}}{\lambda_{\text{Cu}}} \cdot \ln\left(\dfrac{r_{\text{E,o}}}{r_{\text{E,i}}}\right) + \dfrac{1}{\alpha_{\text{w, helical coil}}}} . \tag{26}$$

<u>Vertical pipe</u>

Heat transfer coefficient of water around the vertical pipe of evaporator is calculated from the equation (27), found in Eckert [12], which is valid for laminar natural convection of water, for Grashof numbers lower than 10^9:

$$Nu_{\text{w, vertical pipe}} = 0,55 \cdot \sqrt[4]{\frac{\rho_{\text{w},\infty} - \rho_{\text{w,t,o}}}{\rho_{\text{t,o}}} \cdot \frac{g \cdot L^3_{\text{vertical pipe}}}{\upsilon^2_{\text{t,o}}} \cdot Pr}\,, \tag{27}$$

$$\alpha_{\text{w, vertical pipe}} = \frac{Nu_{\text{w, vertical pipe}} \cdot \lambda_{\text{w,t,o}}}{L_{\text{vertical pipe}}}\,. \tag{28}$$

Helical coil

According to the predicted cooling capacity and geometry of the helical coil, the correlation for the heat transfer coefficient around the helical coil on the water side is taken from Ali [3], which is valid for specific diameter ratio of 19.957:

$$Nu_{\text{w, helical coil}} = 0.685 \cdot Ra^{0.295} \rightarrow \left(\frac{D_{\text{coil}}}{d_o} = 19.957 \right), \tag{29}$$

$$\alpha_{\text{w, helical coil}} = \frac{Nu_{\text{w, helical coil}} \cdot \lambda_{\text{w,t,o}}}{d_{\text{E,t,o}}}\,. \tag{30}$$

3.4 Condenser

The heat transfer rate from the refrigerant to the cooling medium (air) is obtained by applying energy balances for each stream, equations (31) and (32):

$$\dot{Q}_{\text{C}} = \dot{m}_{\text{ref}} \cdot (h_2 - h_3)\,, \tag{31}$$

$$\dot{Q}_{\text{C}} = \dot{m}_{\text{air}} \cdot c_{\text{p,air}} \cdot \left(T_{\text{air, out}} - T_{\text{air, in}} \right) = \dot{Q}_{\text{air}}\,. \tag{32}$$

Detailed model of condensation process is not presented here, the focus is on the evaporation process. Model details can be found in cited references. The correlation between air and refrigerant side is given as simplified LMTD thermodynamic model for performance assessment, how authors and published in reference [13] using the following equations for calculating the air temperature at the outlet and overall heat transfer coefficient of the outer surface of air cooled condenser:

$$T_{\text{air,out}} = T_{\text{air,out}} + \frac{\left(T_{\text{C}} - T_{\text{air,in}} \right) \cdot \left(\exp\left(\frac{k_{\text{C,o}} \cdot A_{\text{C,o}}}{\dot{m}_{\text{air}} \cdot c_{\text{p,air}}} \right) - 1 \right)}{\exp\left(\frac{k_{\text{C,o}} \cdot A_{\text{C,o}}}{\dot{m}_{\text{air}} \cdot c_{\text{p,air}}} \right)}\,, \tag{33}$$

$$k_{\text{C,o}} = \frac{\dot{Q}_{\text{air}}}{A_{\text{C,o}} \cdot \left(T_{\text{air, out}} - T_{\text{air, in}} \right)}\,. \tag{34}$$

3.4.1 Refrigerant side

Heat transfer coefficient of condensation process in a turbulent annular film inside horizontal smooth copper tube is obtained from correlation by Cavalini et al. [14], [15] and [16] for turbulent flow regime and predicted fully stratified flow. The degree of subcooling at the condenser outlet is calculated considering a velocity and temperature profile corresponding with laminar condensate film with negligible shear effect, as suggested by Roshenow [17].

3.4.2 Air side

The heat transfer coefficient on the cooling medium side (air) is obtained from the known heat transfer correlation Colburn *j* factor, found in Wang at al. [18].

3.5 Refrigeration system energy performance

Here, the coefficient of performance (COP) for thermodynamic cycle with isentropic compression process and for whole installation (without power of condenser air fan) is considered. Both values are calculated from cooling capacity, isentropic and electric power for compressor work:

$$COP_{\mathrm{is}} = \frac{\dot{Q}_{\mathrm{E}}}{\dot{W}_{\mathrm{is}}}, \tag{35}$$

$$COP_{\mathrm{inst}} = \frac{\dot{Q}_{\mathrm{E}}}{\dot{W}_{\mathrm{elec}}}. \tag{36}$$

3.6 Literature background of compression, evaporator and condenser processes

The mathematical model described in the previous section has been programmed using Engineering Equation Solver (EES) [19], software with built-in thermophysical properties of all working mediums of the experimental unit.

In this section the compression approximation based on manufacturer data and overview on the evaporator and compressor process modeling are given.

Compressor

Compression process is described with volumetric and overall efficiency based on manufacturer data according to the ASHRAE conditions presented in Table 1 for refrigerant R410A.

Table 1 ASHRAE test conditions of compressor

Scroll compressor (GREE) - ASHRAE conditions	
Testing power supply	220 V/ 50 Hz
Evaporation temperature range	0 °C - 10°C
Condensation temperatures	50°C, 55°C, 60°C, 65°C
Liquid temperature	46.1°C
Suction temperature	35°C

Compressor can be applied in a range of evaporation temperature from -15°C to +15°C, and condensation temperature range from 27 °C to 67 °C, for actual refrigerant.

Comparison of the refrigerant mass flow rate obtained from the expression for volumetric efficiency, equation (1), and the refrigerant mass flow rate given in manufacturer data, in the same test conditions, shows difference ± 3%, which can be seen in Figure 3.

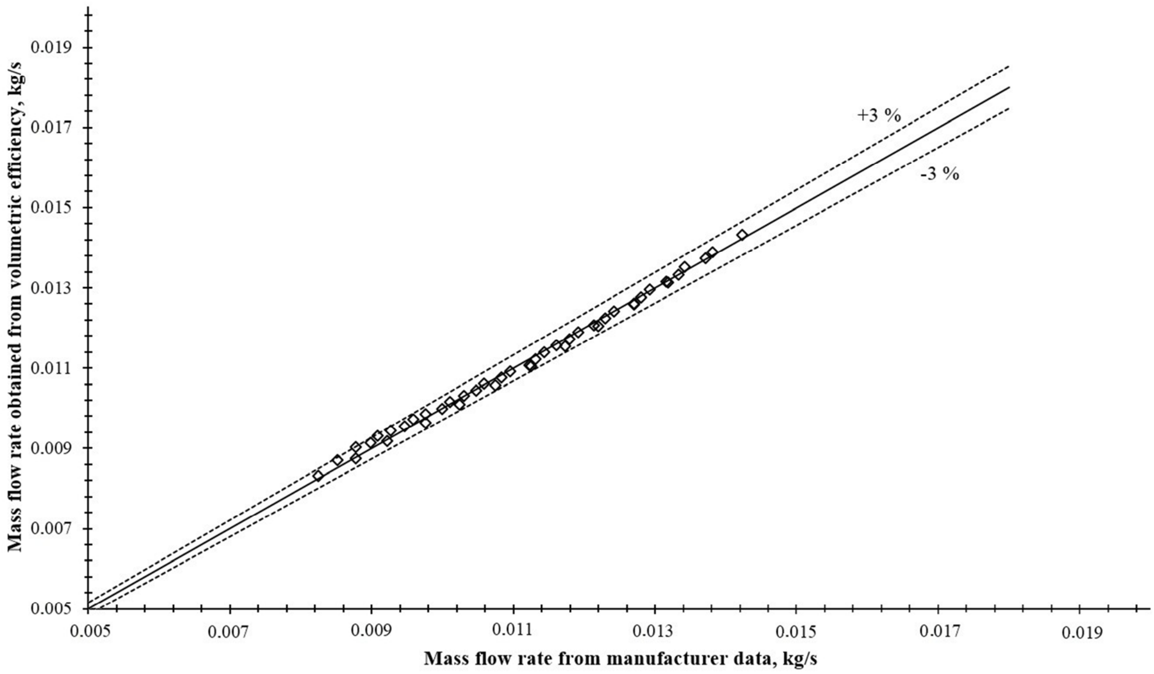

Fig. 3 Deviation between numerical results and manufacturer data of mass flow rate

Results of input power obtained from equation (2) for overall efficiency and input power from manufacturer data show difference ± 3%, for 99% of compared results, according to the presented test conditions, Figure 4.

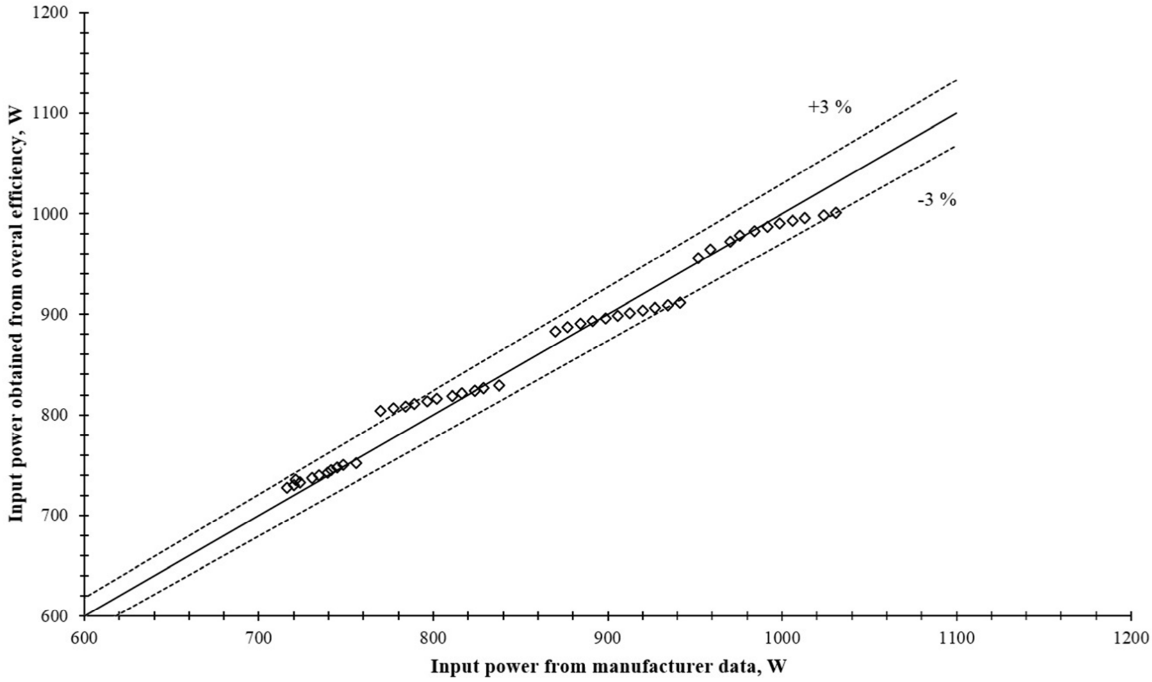

Fig. 4 Deviation between numerical results and manufacturer data of input power for compression process

Evaporator and condenser

The experimental device is in developing process, so there are no experimental results for validation of evaporation or condensation process. However, similar method applied on system modeling of refrigeration device showed that this kind of mathematical modeling is a good prediction of evaporation and condensation process, how authors and published in reference [5]. Results of prediction technique of cooling capacity in the helically coiled evaporator (inside 5 % of difference) compared with the obtained experimental results are published by Elsayed [4] using the similar methodology of complete refrigeration system modeling.

4. Results and discussion

Using the developed mathematical model, the influence on performance of vapor compression system was analyzed by varying the geometrical quantities of the helically coiled evaporator. Two different kinds of analyses were performed:

1. Analysis of geometrical solution for chosen evaporator diameter ratio of 19.957 defined according to the heat transfer correlation form equation (29);
2. Analysis of relation between geometrical quantities and thermal resistance.

The degree of superheat on 3 °C, volumetric flow rate of condenser air fan taken from manufacturer data, pressures of water and air considered as normal atmospheric pressures, inlet temperature of air at 25 °C, and starting temperature of the cooled water at 15 °C, are set as an inputs data in both analyses. Coil pitch and coil angle do not have important effect on the system performance, in that case the coil pitch is observed as constant, Figure 2 $(P_{\text{coil}} = 32\ \text{mm})$, coil angle can be changed a little without effect on the heat transfer; and at the final the number of coils are set on 8 coils $(N_{\text{coil}} = 8)$ for all analyses, because the number of coils do not changes the basic geometry of the evaporator.

All equations from (1) to (36) and thermodynamic properties of all fluids are included in the EES calculation program to obtain the results of the thermodynamic refrigeration cycle for these input data.

4.1 Analysis of geometrical solution for evaporator diameter ratio of 19.957

In this analysis only the coil diameter is additional input parameter varied from 265 to 365 m. Based on the evaporator diameter ratio used for the natural convection around the evaporator flooded in the water, published in Ali [3], it is possible to determine all other variables from the presented mathematical model.

The next two figures are in correlation.

The dependences of the water temperature, the tube diameter and evaporator length are presented at Figure 5. From the same simulation results in the next Figure 6, dependence of the cooling capacity, COP of refrigeration process and diameter coil is presented. It is clear that with increasing, both, the evaporator length and the coil diameter, COP and cooling capacity increases. However, according to the published heat transfer features this both geometrical quantities could be considered from thermodynamic process simulation, and in some similar design/redesign or thermodynamic analysis. For wanted temperature of the cooled water, the geometrical quantities of the evaporator (outer diameter of the tube and evaporator length) can be obtained from Figure 5, and follows on Figure 6 with coil diameter, cooling capacity and COP of the refrigeration device.

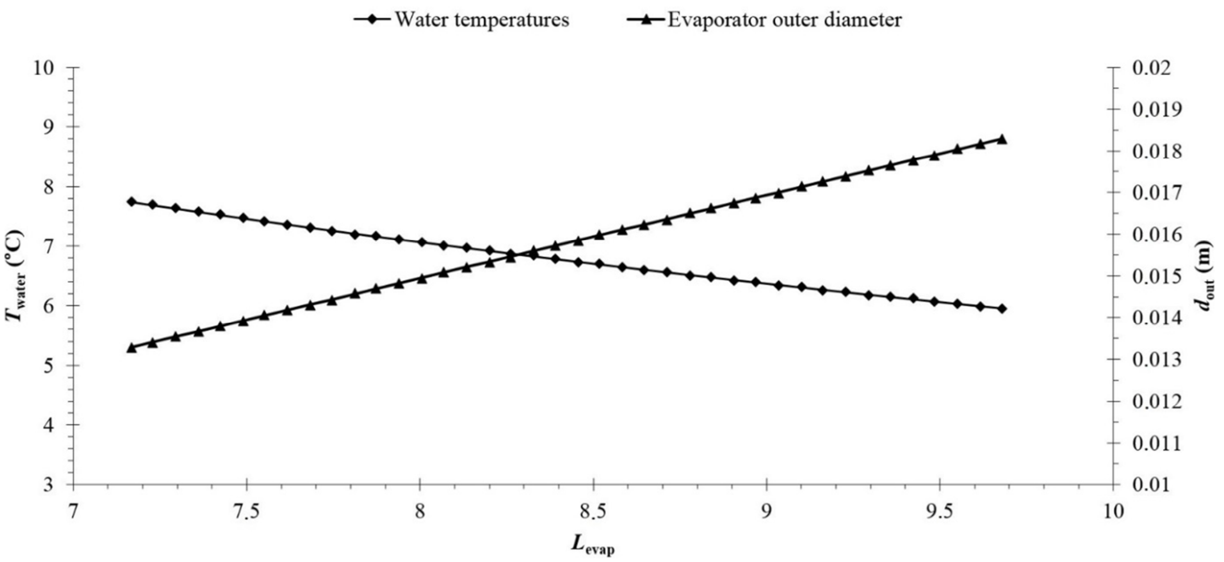

Fig. 5 Dependence of water temperature, tube diameter and evaporator length

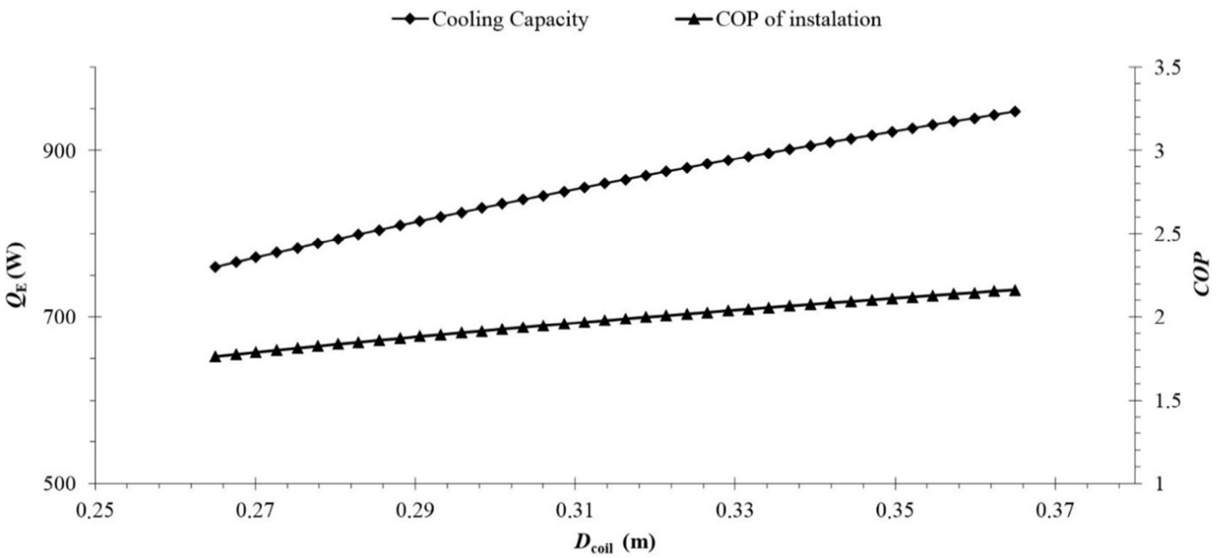

Fig. 6 Dependence of water temperature, tube diameter and coil diameter

The question is which of these two parameters (coil diameter and tube diameter) has a more significant influence on the system performance, what is and presented in the next analysis from simulation results by varying these both geometrical parameters.

4.2 Analysis of the relation between geometrical quantities and thermal resistance

In this analysis the tube diameter and coil diameter are varied where the cooled water temperature and water side thermal resistance are calculated.

Figure 7 shows results for fixed inner/outer tube diameter of 13/15 mm. It can be seen that water side thermal resistance slightly increases with increasing the coil diameter, from 265 mm to 365 mm. Vice versa, for fixed coil diameter of 300 mm, by varying the outer tube diameter of evaporator from 13.5 mm to 21.5 mm, water side thermal resistance decreases with increasing the outer tube diameter, and the cooling heat transfer increases, how is presented on Figure 8.

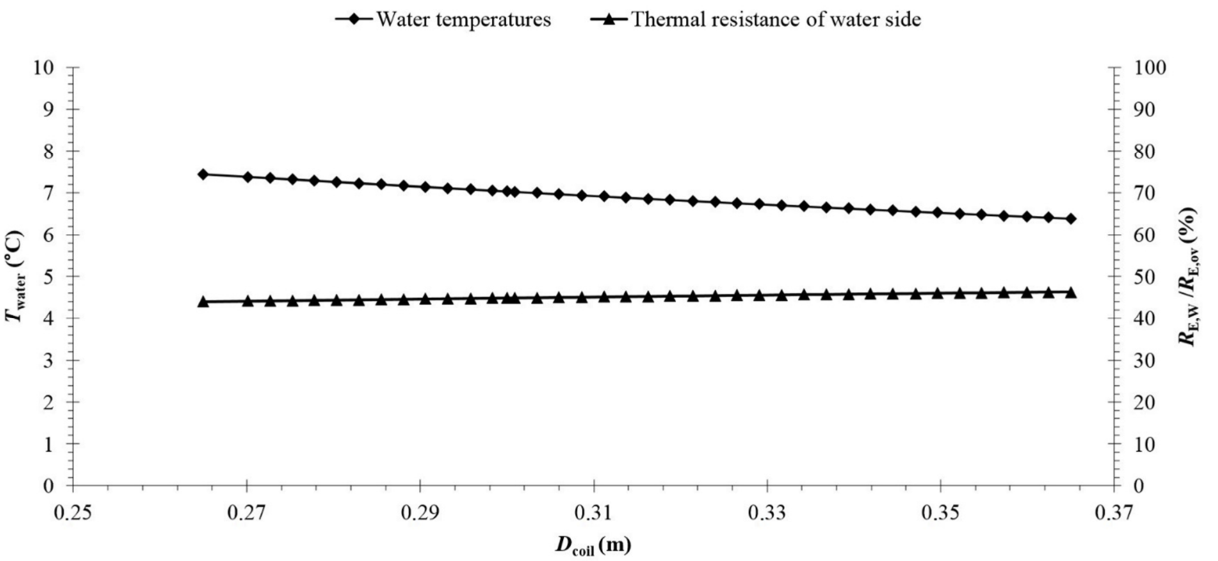

Fig. 7 Dependence of the water temperature and water side thermal resistance on the coil diameter for fixed tube diameter

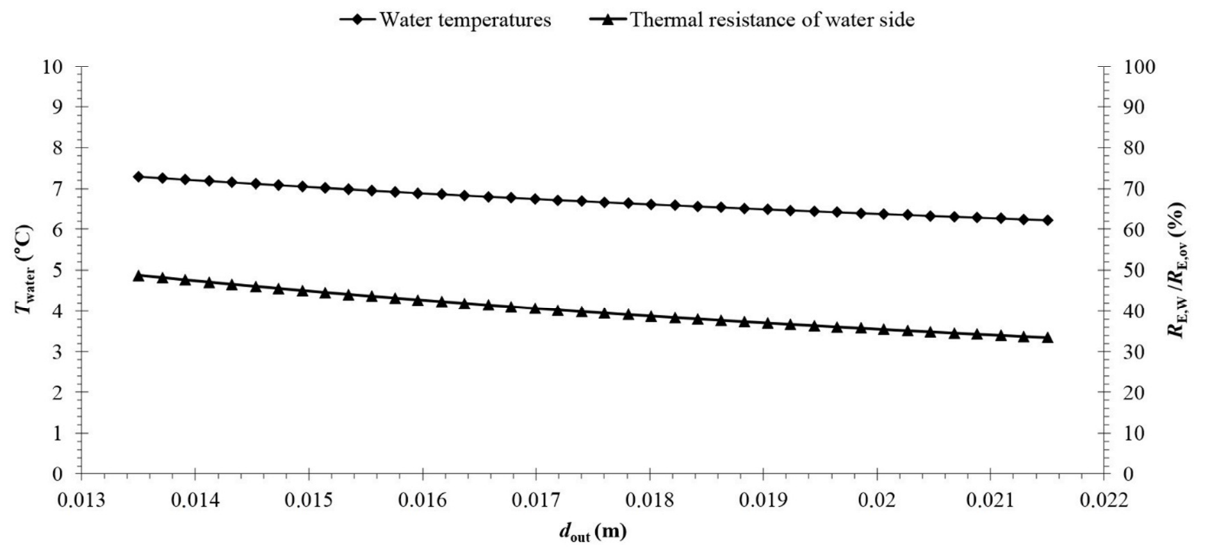

Fig. 8 Dependence of the water temperature and water side thermal resistance on the tube diameter for fixed coil diameter

Mostly, in papers from references, researchers claim that higher values of the heat transfer coefficient inside the tube are received for smaller values of the tube diameter, what should results with better system performance of cooling device. However, here, in this paper it could be seen that for higher values of the tube diameter the lower thermal resistance on the water side is present, and for higher values of the coil diameter the slightly higher thermal resistance on the water side is observed. It can be drawn that changing the tube diameter has more significant influence on the system performance then coil diameter.

Those circumstances indicate that a different conclusion can be drawn when we ask the following question: Which effect will we have on thermodynamic cycle (system performance, heat fluxes, temperature of cooling/heating medium…) choosing the adequate geometry of this type of evaporator for system design or redesign, where in the final the same water temperature around the evaporator is wanted?

5. Conclusion

This paper presents a complete mathematical model of vapor compression device used for simulations of thermodynamic refrigeration cycle, system design or redesign possibilities for the similar small power application of refrigeration devices. Based on the given mathematical model, presented results and heat transfer features of the evaporator, the following conclusions can be drawn:

- This type of mathematical modeling of the refrigeration systems can be a useful tool for determination system performance from different points of view, as thermodynamic or some geometrical and some other analyses.
- Results provide that geometrical quantities of the evaporator may have a different effect on the cooling capacity and COP. In other words, to obtain the same output conditions before and after the system improvement, using similar mathematical models, the geometry can be obtained for these types of refrigeration devices.
- Analysis of thermal resistance shows that the coil diameter of the evaporator does not have a significant effect on the system performance. On the contrary, based on the results shown, the outer tube diameter of the evaporator is the quantity with higher influence on the system performance. Finally, it can be concluded that improvement can be achieved by changing the geometry of these types of evaporators, in some circumstances with higher and in other with lower impact on the refrigeration system performance.

REFERENCES

[1] Jitian, H., Li, S., Wenwen, C., and Changnianm, C. Study on Flow Boiling Heat Transfer of R134a in Horizontal Helical Coils. *Chinese Engineering Thermophysics*. ID 093191

[2] Zhao, L., Guo, L., Bai, B., Hou, Y., and Zhang, X. (2003). Convective boiling heat transfer and two phase flow characteristics inside a small horizontal helically coiled tubing once-through steam generator. *International Journal of Heat and Mass Transfer* 46: 4779-4788

[3] Ali M.E.; Experimental investigation of natural convection from vertical helical coiled tubes. *Int. J. Heat Mass Transfer* 37(4): 665–671, 1994

[4] Ahmed Mohamed Elsayed, Heat Transfer in Helically Coiled Small Diameter Tubes for Miniature Cooling Systems, Degree of Doctor of Philosophy (2011), *School of Mechanical Engineering, University of Birmingham*

[5] Z. Janković, J. S. Atienza, J. A. M. Suárez, Thermodynamic and heat transfer analyses for R1234yf and R1234ze(E) as drop-in replacements for R134a in a small power refrigerating system, *Applied Thermal Engineering* 80 (2015) 42-52

[6] Chi-Chuan Wang, System performance of R-1234yf refrigerant in air-conditioning and heat pump system - an overview of current status, *Appl. Therm. Eng.* 73 (2014) 1412-1420

[7] Rohsenow W.M. Griffith P., Correlation of maximum heat flux data for boiling of saturated liquids, *AICHE-ASME Heat Transfer Symp*., Louisville, Ky., 1955

[8] Rohsenow W.M., A method of correlating heat transfer data for surface boiling liquids, *Trans. ASME*, Vol. 74, 1952

[9] Dittus PW. And Boelter L.M.K., Heat transfer in automobile radiators of the tubular type, *Int. Commun. Heat Mass Transfer*, Vol. 12, 1985, pp. 3-22.

[10] Petukhov; Heat transfer and friction in turbulent pipe flow with variable physical properties, *Advances in Heat Transfer*, Vol. 6, Editores J.P. Hartentt and T.F. Irvine, Academic Press, New York, 1970

[11] Gnielinski V., New equations for heat and mass transfer in turbulent pipe and channel flow, *Int. Chem. Eng.*, Vol. 16, 1976, pp. 359-368.

[12] E. R. G. Eckert, Thomas W. Jackson: Analysis of turbulent free convection boundary layer on flat plate, *National Advisory Committee for Aeronautics* (1950): 2207

[13] M. Pieve , G. Salvadori, Performance of an air-cooled steam condenser for a waste-to-energy plant over its whole operating range*, Energy Conversion and Management*, Volume 52, Issue 4, April 2011, Pages 1908–1913

[14] J. El Hajal, J.R. Thome, A. Cavallini: Condensation in horizontal tubes, part 1: two-phase flow pattern map. *Int. J. Heat Mass Transfer* 46 (1994): 3349–3363

[15] J.R. Thome, J. El Hajal, A. Cavallini: Condensation in horizontal tubes, part 2: new heat transfer model based on flow regime. *Int. J. Heat Mass Transfer* 46 (2203): 3365–3387

[16] D. Del Col, A. Cavallini, J. R. Thome: Condensation of Zeotropic Mixtures in Horizontal Tubes: New Simplified Heat Transfer Model Based on Flow Regimes, *Journal of Heat Transfer* 127 (2005): 221-230

[17] W.M. Rohsenow, Heat transfer and temperature distribution in laminar condensation, *Trans. Am. Soc. Mech. Eng. Ser. C. J. Heat. Transf.* 78 (1956) 1645-1648.

[18] Wang, C.C., Chang Y.J., Hseih, Y.C. and Lin, Y.T.: Sensible heat and friction characteristics of plate fin-and-tube heat exchangers having plane fins. *International Journal of Refrigeration* 19 (1996): 223-230

[19] S.A. Klein, Engineering Equation Solver (EES), F-Chart Software, Middleton, WI, 1992-2015.

NOMENCLATURE

A	Area	(m^2)	Pr	Prandtl number	-
COP	Coefficient of performance	-	p	Pressure	(bar)
c_p	Specific heat capacity	(J·kg^{-1}·K^{-1})	q	Specific heat transfer rate	(J·kg^{-1})
d	Diameter	(m)	$\dot{Q}$	Heat transfer rate	(W)
f	Frequency supply	(Hz)	R	Thermal resistance	(K·W^{-1})
g	Gravity	(m·s^{-2})	Ra	Rayleigh number	
k	Overall heat transfer coefficient	(W·m^{-2}·K^{-1})	RC	Compression ratio	-
L	Length	(m)	s_m	Slip	-
$\dot{m}$	Mass flow rate	(kg·s^{-1})	T	Temperature	(K)
N	Compressor speed	(rpm)	V	Volume	(m^3)
Nu	Nusselt number	-	$\dot{W}$	Power	(W)
P_m	Number of magnetic pole pairs per phase	-	X	Quality	-

Greek symbols

α	Convective heat transfer coefficient	(W·m^{-2}·K^{-1})			
Δh	Specific enthalpy difference	(J·kg^{-1})	ΔT	Temperature difference	(K)
ΔT_{lm}	Logarithmic mean temperature difference	(K)			
λ	Thermal conductivity	(W·m^{-1}·K^{-1})	ρ	Specific enthalpy difference	(J·kg^{-1})
η	Efficiency	-	μ	Viscosity	(kg·m^{-1}·s^{-1})
χ_{tt}	Martinelli number	-	v	Kinematic viscosity	(m^2·s^{-1})

Subscripts

C	Condenser	out	Outlet
Cu	Copper	ov	Overall
cyc	Cylinder of compressor (displacement)	ref	Refrigerant
E	Evaporator	sat	Saturation
elec	Electric	sp	single phase
i	Inner	sub	Subcooling
in	Inlet	sup	Superheating
inst	Installation	t	Tube
is	Isentropic	V	Vapor
L	Liquid	vol	Volumetric
LO	Liquid only	W	Water
nb	Nucleute boiling	2ph	Two-phase
o	Outer		

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