Mathematical Model of Heat Pumps' Coaxial Evaporator with Distributed Parameters

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Abstract: The aim of this article is to create a very precise mathematical model with distributed parameters of the heat pumps' coaxial evaporator in order to examine the behavior of evaporator in the steady-state operation mode. The mathematical model consists of the balance equations, the equations of viscous friction coefficient, the convective heat transfer, and the state equation of refrigerant R134a. The boundary conditions, which are the thermal-expansion valve and the compressor, in this work, have not yet been taken into account. It is the task and the goal of the next article. For the solution of mathematical model an iterative numerical procedure was applied. Simulation results were obtained in numerical form and are presented graphically. The 'new' in this paper: implanted the Kandlikar's evaporative heat transfer coefficient as a function of vapor quality, applied the viscous friction coefficient of the refrigerant according to Friedel and state equations of R134a according to Martin-Hou. In addition, using the new mathematical model simulated the behavior of coaxial evaporator.

Keywords: coaxial evaporator; heat pump; performance; mathematical model; distributed parameters; efficiency

Nomenclature

$\dot{m}$	Mass flow rate $[kg/s]$
α	Convective heat transfer coefficient $[W/m^2/°C]$
λ	Conductive heat transfer coefficient $[W/m/°C]$
C_p	Specific heat, p = const $[J/kg/°C]$
t, T	Temperature [°C], $[K]$

dt	Temperature difference [°C]
f	Function of viscous friction $[-]$
A	Cross section area of flow $[m^2]$
di	Latent heat $[J/kg]$
q	Heat flux, performance $[W]$
Re	Reynolds number $[-]$
Pr	Brandt number $[-]$
Fr	Fraud number $[-]$
C	Coefficient $[s/kg^2/m]$
ρ	Density $[kg/m^3]$
δ	Thickness $[m]$
d	Diameter $[m]$
w	Velocity $[m/s]$
i	Specific enthalpy $[J/kg]$
R	Gas constant $[J/kg\ K]$
V	Volume $[m^3]$
x	Vapor quality [-]
P	Pressure $[Pa]$

Subscripts and superscripts

w	water
f	refrigerant (Freon)
I	input
o	output
m	middle, normal
c	critical
p	parallel
v	volumetric

1 Introduction

The evaporator is the central component of the heat pump. The coefficient of performance (COP) of the heat pump depends largely on the construction of the evaporator. The construction of the evaporator significantly influences the refrigerant flow pattern within the pipes and on the shell side in water. The flow pattern of streaming fluids greatly affects the value of heat transfer coefficient and the value of viscous friction coefficient.

The quality of the mathematical model of heat pump's evaporator depends largely on the accuracy of the mathematical formulation of the heat transfer coefficient and of the viscous friction coefficient in both streaming fluids. The most significant effect on the accuracy of the simulation results is the heat transfer coefficient of evaporation.

In the mathematical model applied governing, equilibrium equations are derived and long since proved. Uncertainty in the model regarding to accuracy represents only the heat transfer coefficients and the viscous friction coefficient of refrigerant.

In this investigation, we paid special attention to choose the heat transfer model of refrigerant evaporation. Among the many models we chose Kandlikar's [1] improved model. Kandlikar's model for the heat transfer of refrigerant evaporation gave the best results in the simulation, according to the tendency and according to the intensity. Róbert Sánta [4] in his work has confirmed the justification of applying the Kandlikar's model.

Besides the evaporator construction, the system parameters significantly affect the coefficient of performance (COP) of the heating-cooling system when using heat pumps.

We carried out numerical simulation to verify the accuracy of the mathematical model of the evaporator, and simultaneously investigated the behavior of the evaporator in steady-state operation for various values of water mass flow rate as well.

In this paper the boundary conditions, which are the thermal expansion valve and the compressor, have not yet been taken into account. It is the task and the goal of the next article.

In this paper is a new methodology: implant the evaporative heat transfer coefficient as the function of vapor quality according to Kandlikar [1], apply the viscous friction coefficient of the refrigerant according to Friedel [2] and use the state equations of R134a, according to Martin-Hou [3]. In addition, using the new model the behavior of coaxial evaporator was simulated.

2 Physical Model of the Coaxial Evaporator

Structurally, the considered evaporator as a coaxial tube heat exchanger. From thermal aspect, in the evaporator the heat transfer is performed between the refrigerant and the well water. In the observed case, refrigerant Freon R134a flows inside the evaporator tubes, while the cooled mass - the well water - flows inside the shell of evaporator.

In the evaporator, the parallel pipes are connected with the baffles. The baffles are placed perpendicular to the pipe bundle at a distance of about 150 mm in the direction of the tube axis. Water flows between the baffles like a sinusoid.

The evaporator works most efficiently when the full length of parallel pipes is filled with refrigerant which still contains the liquid phase. The quantity of evaporated refrigerant depends on the compressor capacity; the temperature and the mass flow rate of well water respectively.

The thermo-expansion (TEX) valve doses an adequate quantity of refrigerant into the evaporator. This valve uses the sensors monitors the temperature and the pressure of the outlet refrigerant from the evaporator. Based on the measured temperature and pressure, the valve carries out optimal dosing of the refrigerant.

In case of a refrigerant quantity deficit, the shorter length of the pipes evaporates the liquid phase of refrigerant, and in the remaining part of the evaporator flows only vapor which superheats. The heat transfer of the Freon's' vapor phase is multiple times lower than of the liquid phase. The vapor of the refrigerant in the dry section superheats.

Over-dosing of the refrigerant means that the liquid phase cannot evaporate and some quantity of the liquid phase leaves the evaporator. This phenomenon reduces the refrigeration effect, for example the performance of the heat pumps' evaporator. Interpretation: non-evaporated liquid phase evaporates in the compressor. Consequently, hydro hummer can happen in the compressor. In the correct operation mode from the evaporator dry vapor flows out, superheating up to 4 °C.

The evaporator is connected to the compressor. From the aspect of the evaporator, the compressor only sucks out the superheated vapor from the evaporator. From the aspect of compressor, the compressor using mechanical work compresses the vapor which primarily increases the temperature on the adequate level. The compression is unfortunately accompanied with intensive increase of the pressure as well.

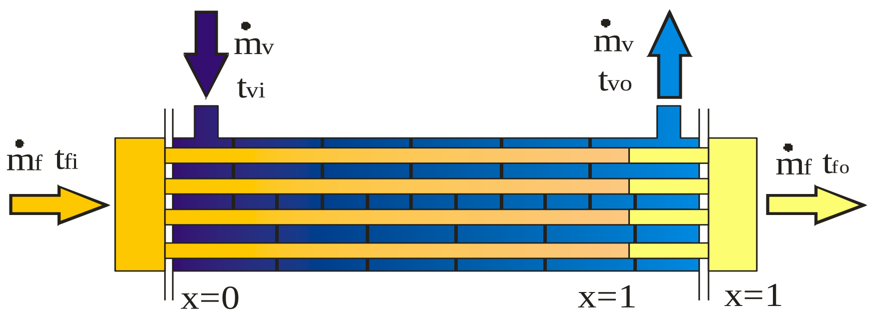

Figure 1

The physical model of the heat pumps' evaporator with the system parameters

3 Stationary Mathematical Model of the Evaporator

3.1 Conditions and Approximations

During the modeling the next conditions and approximations have been applied.

- Distributed parameters,
- Operation mode is steady-state,
- Pressure drop in the refrigerant is taken account,
- Evaporator walls' thermal resistance is not neglected,
- Specific heat of the water is taken as a constant because it slightly depends on the temperature,
- Specific heat of the refrigerant vapor is taken as constant because it slightly depends on the temperature,
- Evaporator's heat gain from the environment is neglected.

3.2 Governing Equations

3.2.1 Balance Equations of the Refrigerant

Mass conservation

$$\frac{\partial \left(\rho \cdot w\right)}{\partial z} = 0 \tag{1}$$

If the operation mode is steady-state

$$\rho \cdot w \equiv \dot{m} = const. \tag{2}$$

Impulse conservation

$$\frac{\partial \left(\rho \cdot w \cdot w + p\right)}{\partial z} + f(x) = 0 \tag{3}$$

Energy conservation

$$\frac{\partial \left(\rho \cdot w \left(w \cdot w / 2 + i\right)\right)}{\partial z} - q(x) = 0 \tag{4}$$

3.2.2 Energy Balance Equation of the Pipe Wall Using the Energy Conservation Law

$$C_{cv} \cdot (T_w - T_c) - C_{cf} \cdot (T_c - T_f) = 0 \tag{5}$$

3.2.3. Energy Balance Equation of the Well Water which Flows through the Evaporator, Based on the Energy Conservation

$$\pm w_v \cdot \frac{\partial T_w}{\partial z} + C_w \cdot (T_w - T_c) = 0 \tag{6}$$

3.3 Auxiliary Equations

3.3.1 State Equations of the Refrigerant R 134a

The thermodynamic equations of refrigerant R 134a are taken from ISO / DIN 17584 standards.

The equations are applicable at the following conditions:

$0.60 < T_r < 0.96$

$0.5\ [bar] < p < 25\ [bar]$

$T_{max} = 500\ [K]$

3.3.2 The Balance Equation according to Wagner

$$lnp_r = \frac{1}{T_r} \cdot \left[\begin{array}{l} A_1 \cdot (1 - T_r) + A_2 \cdot (1 - T_r)^{B1} + A_3 \cdot (1 - T_r)^{B2} \\ + A_4 \cdot (1 - T_r)^{B3} + A_5 \cdot (1 - T_r)^{B4} + A_6 \end{array} \right] \tag{7}$$

Where:

The relative temperature

$$T_r = \frac{T}{T_C} \tag{8}$$

The relative pressure

$$p_r = \frac{p}{p_C} \tag{9}$$

The critical temperature of R134a

$$T_C = 374.21 [K] \tag{10}$$

The critical pressure of R134a

$$p_C = 40.5\ [bar] \tag{11}$$

3.3.3 The Parameters of the Refrigerant of Saturated Area

Specific enthalpy of the mixture

$$i = i' + x \cdot (i'' - i') \tag{12}$$

Specific internal energy of the mixture

$$u = u' + x \cdot (u'' - u') \tag{13}$$

Specific volume of the mixture:

$$v = v' + x \cdot (v'' - v') \tag{14}$$

3.3.3 State Equation of R134a according to Martin - Hou [3] for the Superheated Area

Martin-Hou used Van der Waals equation of real gases and adapted the equation with the correctional coefficients for refrigerants (i.e. for Freon). The equation is applicable for the Freon vapor superheated area.

$$p = \frac{R \cdot T}{V - b} + \sum_{i=2}^{5} \frac{A_i + B_i \cdot T + C_i exp(-{K \cdot T}/{T_{crit}})}{(V - b)^i} \tag{15}$$

Where:

The coefficients from the equation

$$b = 2.99628 \cdot 10^4 \left[\frac{m^3}{kg} \right]$$

$$K = 5.475$$

3.3.4 Specific Enthalpy of the Freon Vapor

$$\begin{aligned} i(T_r, v) = \mathrm{i}_o + (p \cdot v - R \cdot T) + D_1 \cdot T + {D_2 \cdot T^2}/{2} + \\ + D_4 \cdot lnT + {E_1}/{z} + {E_2}/{2 \cdot z} + {E_3}/{3 \cdot z} + {E_4}/{4 \cdot z} + \\ \mathrm{e}^{-\mathrm{kT_R}} \cdot (1 + \mathrm{k} \cdot \mathrm{T_R}) \cdot \left({G_1}/{z} + {G_2}/{2 \cdot z^2} + {\mathrm{G}_4}/{4 \cdot z^4} \right) \end{aligned} \tag{16}$$

3.3.5 Pressure Drop in the Single-Phase and the Bi-Phase of Refrigerant f(x)

For the determination of pressure drop in the refrigerant, Friedel's mathematical model was applied.

Friedel's [2] mathematical model is used in the area of single-phase and bi-phase evaporation as well.

$$\frac{\Delta p}{dz} = \Phi^2 \cdot \left(\frac{dp}{dz} \right)_f \tag{17}$$

Pressure drop factor of the liquid phases

$$\left(\frac{dp}{dz}\right)_f = \frac{4 \cdot \lambda_f \cdot G^2}{2 \cdot D \cdot \rho_f} \tag{18}$$

Friction factor of the flowing fluids was determined according to Blasius

Where:

For the liquid phase

$$\lambda_f = \frac{0.079}{Re_f^{0.25}} \quad Re_f = \frac{G \cdot D}{\mu_f} \tag{19}$$

For the vapor phase

$$\lambda_g = \frac{0.079}{Re_g^{0.25}} \quad Re_g = \frac{G \cdot D}{\mu_g} \tag{20}$$

The characteristic friction factor in bi-phase flow

$$\Phi^2 = E + \frac{3.24 \cdot F \cdot H}{Fr^{0.045} \cdot We_f^{0.035}} \tag{21}$$

3.3.6 Latent Heat as a Function of the Evaporation Temperature

$$di = a_o + a_1 \cdot t_{fo}^1 + a_2 \cdot t_{fo}^2 \tag{22}$$

Where the constants of the refrigerant R 134a are:

$a_o = 200.5965715$

$a_1 = -0.709168$

$a_2 = -0.00596796$

3.3.7 Evaporative Heat Transfer Coefficient of the Refrigerant

For the determination of the evaporative heat transfer coefficient of refrigerant inside the pipe Kandlikar's improved mathematical model was used. The same model was used for single-phase and bi-phase refrigerant as well. Improved Kandlikar's model separately estimates the heat transfer coefficient of the liquid and the vapor phase. The result is taken as the maximum value of the calculated two.

$$\alpha_{tp} = max \ [\alpha_n, \alpha_c] \tag{23}$$

3.3.7 Heat Transfer Coefficient of the Convective Area

$$\alpha_n = 0.6683 \cdot Co^{-0.2} \cdot (1-x)^{0.8} \cdot \alpha_{lo} \cdot f\left[Fr_{lo}\right] + 1058 \cdot Bo^{0.7} \cdot (1-x)^{0.8} \cdot F_{FL} \cdot \alpha_{lo} \tag{24}$$

3.3.8 Heat Transfer Coefficient of the Bubble Area

$$\alpha_c = 1.136 \cdot Co^{-0.9} \cdot (1-x)^{0.8} \cdot \alpha_{lo} \cdot f\left[Fr_{lo}\right] + 667.2 \cdot Bo^{0.7} \cdot (1-x)^{0.8} \cdot F_{FL} \cdot \alpha_{lo} \tag{25}$$

3.3.9 Heat Transfer Coefficient in the Single Phase Region according to Petukhov and Popov

Interval of the applicability:

$$0.5 < Pr_{lo} < 2000 \tag{26}$$

$$10^4 < Re_{lo} < 5 \cdot 10^6 \tag{27}$$

$$\alpha_{lo} = \frac{Re_{lo} \cdot Pr_{lo} \cdot \left(\frac{\xi}{2}\right)}{1.07 + 12.7 \cdot \left(Pr_{lo}^{2/3} - 1\right) \cdot \left(\frac{\xi}{2}\right)^{0.5}} \cdot \frac{\lambda}{d} \tag{28}$$

3.4 Equations of the Heat Source Fluid - Well Water

The complex flow of the water has been taken into account when the heat transfer coefficient of the cooled mass, i.e. of well water, which was determined by applying the circular baffles.

The baffles are placed perpendicularly on the evaporators' tubes. The baffles force the water to flow in the form of a sinusoid. The water stream is separated approximately in two sections, parallel with the pipes and perpendicular to the pipes' direction.

3.4.1 Mathematical Model of the Heat Transfer Coefficient of Well Water according to Hausen

$$\alpha_w = 0.0222 \cdot R_{ew}{}^{0.6} \cdot Pr_w{}^{0.33} \cdot \frac{\lambda_w}{d_k} \tag{29}$$

Reynolds number of cooled mass - well water

$$Re_w = \frac{\dot{G}_{eq} \cdot d_k}{\eta_w} \tag{30}$$

Mass velocity in the case of an equivalent cross section:

$$\dot{G}eq = \left(\dot{G}p \cdot \dot{G}m \right)^{0.5} \tag{31}$$

$$A_e = \left(A_p \cdot A_m \right)^{0.5} \tag{32}$$

The mass velocity in case of parallel flow

$$\dot{G}p = \frac{\dot{m}_w}{A_p} \tag{33}$$

$$A_p = \left(D^2 - z \cdot d_k{}^2 \right) \cdot \frac{\pi}{4} \cdot n \tag{34}$$

The mass velocity in case of normal flow

$$\dot{G}m = \frac{\dot{m}_w}{A_m} \tag{35}$$

$$A_m = \frac{\left(D \cdot b \cdot \left(s - d_k \right) \right)}{s} \tag{36}$$

3.5 Conditions and Initial Numerical Values of the Simulation

The initial conditions and values of the simulation in stationary operating modes of the evaporator:

Refrigerant is R134a

Mass flow of the Freon m = 0.07 kg / s

Temperature of the well water t = 13°C

Pressure in the condenser p_k = 15 bar

Inlet pressure of the evaporator p_i = 3 bar

Shell tube diameter D = 62 mm

Number of the pipes in the evaporator and the pipe diameter 14 x ø8 x 1 mm

3.6 Condition of Linking Two Parts of the Model

The mathematical model consists of two parts: the evaporating sector and the superheating sector. Governing equations of both sectors are the same, but the physical parameters are different. The thermal and the hydraulic parameters are different for the refrigerant mixture and the refrigerant vapor respectively.

Namely, the evaporating sector the refrigerant is in liquid and vapor aggregate state, while in the superheating sector the vapor is superheated. The condition for the change of physical coefficients in the mathematical model during the simulation is the vapor quality. The value of the vapor quality in the critical position is $x = 1$.

4 Numerical Method

Mathematical model consists of system coupled partial differential equations with boundary conditions. The boundary conditions are algebraic equations to describe the behavior of thermal expansion valve and the compressor. In this case, the boundary conditions are only numbers, not functions. For example, the effect of the thermal expansion valve is represented with the value of the refrigerants' input temperature in the evaporator.

For solving the mathematical model the numerical recursive procedure Runge-Kutta and iterative Mac Adams were applied. The procedure of Runge-Kutta is used only for getting initial solutions at discrete points, while the procedure Mac Adams converges the approximate initial solutions by iterative manner to the accurate values. Namely, the recursive procedure Runge-Kutta solves only one-point boundary problems; in our case, the problem is two-points. The system of the differential equations with two-point boundary conditions is solved by an iterative procedure, one of these is Mac. Adams. The obtained results are numerical and presented visually in graphical form. Graphics are visible in Figures 2, 3, 4 and 5.

5 Simulations

A few independent variables were investigated to explain the behavior of the heat pump’s evaporator. The investigations were carried out using the mathematical model and the numerical method.

The independent variables:

• Refrigerant superheated vapors' outlet temperature.

• Pipe wall temperature.

• Well water outlet temperature.

The simulation includes the following investigations:

a) Refrigerant convective heat transfer coefficient as a function of the pipe length

$$\alpha_f = f(z) \tag{37}$$

b) Refrigerant temperature in the evaporators' pipes as a function of the pipe length

$$t_{fo} = f(z) \tag{38}$$

c) Well water temperature in the evaporator as a function of the pipe length

$$t_{wo} = f(z) \tag{39}$$

d) Evaporators' thermal performance as a function of the pipe length

$$q = f(z) \tag{40}$$

6 Simulation Results

The obtained results are based on the created mathematical model and numerical methods. These results are presented in the Figures 2, 3, 4 and 5.

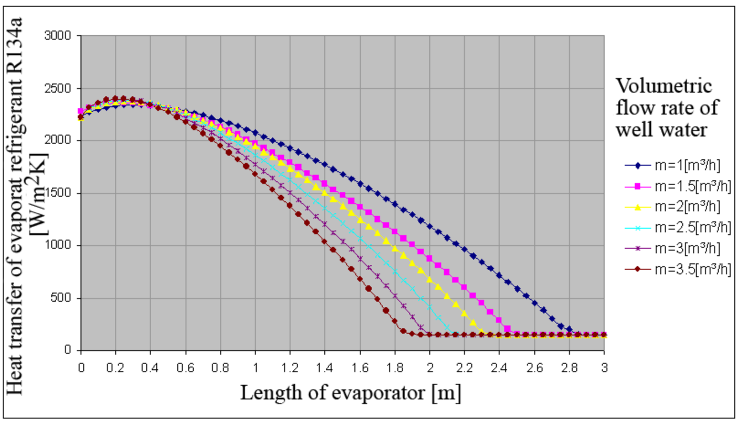

Figure 2

Refrigerant convective heat transfer coefficient as a function of the pipe length

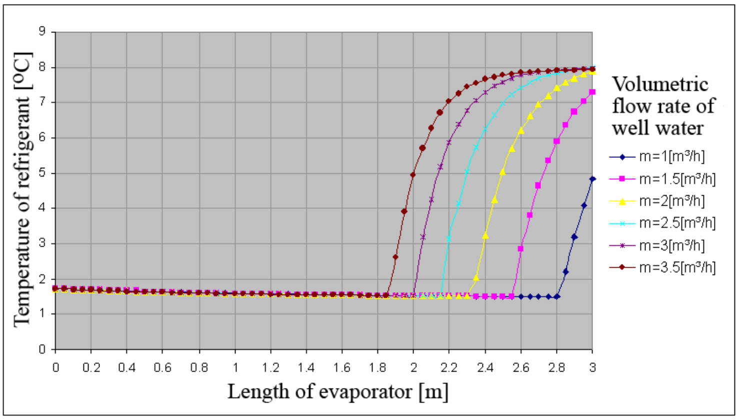

Figure 3

Refrigerant temperature in the evaporators' pipes as a function of the pipe length

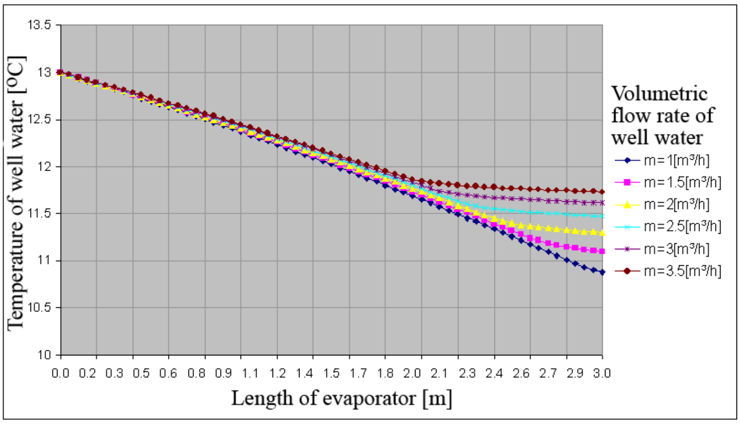

Figure 4

Well water temperature in the evaporator as a function of the pipe length

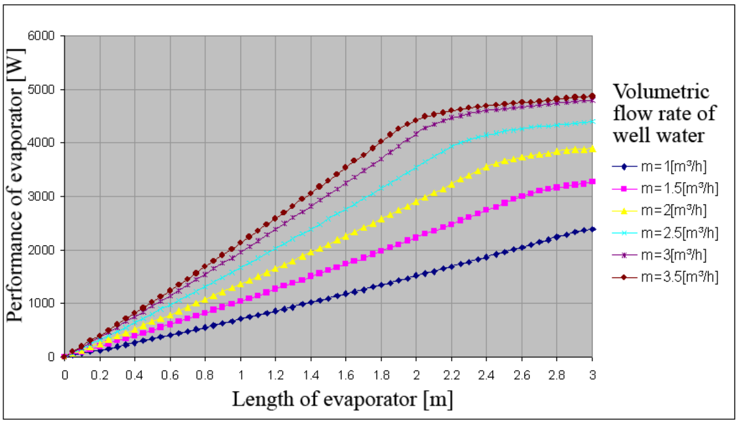

Figure 5

Evaporators' thermal performance as a function of the pipe length

Conclusions

a) Comments on the Mathematical Model

- In the mathematical model applied governing, equilibrium equations are derived and already proven. Uncertainty in the model regarding to accuracy represents only the heat transfer coefficients and the viscous friction coefficient of refrigerant,
- The stationary mathematical model of the evaporator consists of the system coupled partial differential equations and the explicit and implicit non-linear algebraic equations,
- The model is one-dimensional with distributed parameters, for steady-state operation mode,
- The pressure drop in the coaxial exchanger is taken into account,
- The evaporator wall's thermal resistance is not neglected,
- Specific heat of the water is taken as a constant because it slightly depends on the temperature,
- Specific heat of the refrigerant vapor is taken as a constant because it slightly depends on the temperature,
- The evaporator's heat gain from the environment is neglected,
- Solving the mathematical model is possible only numerically,
- The applied numerical procedures:
  - To obtain the initial solutions, a recursive one point numerical procedure, Runge-Kutta was used,

    - To obtain the final solution, the iterative two-point numerical procedure, Mac. Adams was used.
- The obtained solutions are in numerical form,
- The solutions are presented visually in graphic form.

b) Comments on the heat pumps' evaporator

- The simulations were performed at**:**
    - Constant temperature of the well water inlet,
    - Constant mass flow rate of the well water.

c) Conclusions based on the simulation results

- The accuracy of the mathematical model depends largely on the applied heat transfer coefficient of both streaming fluids,
- With increasing the mass flow rates in both fluids increases the whirling as well this phenomenon causes the increasing of the heat transfer coefficient,
- To a smaller extent, the viscous friction coefficient of both fluids also affects the accuracy,
- Based on the tested five models, Kandlikar's heat transfer coefficient model of the refrigerant proved the most appropriate. At the beginning and at the end of evaporation, the values of the heat transfer coefficient correspond to the expectations. Character i.e. the tendency of the coefficient changes along the evaporator is real. It looks like a parabola [4],
- The evaporation heat transfer coefficient of the Freon R134a shows a logical tendency. While decreasing the quantity of the liquid phase, the heat transfer coefficient exponentially decreases and achieves the value of heat transfer coefficient of superheated vapor (Figure 2),
- With increasing the well water's mass flow rate the consequences:
    - Decreases the length of evaporation, the result is the length of the evaporator decreases as well,
    - The refrigeration capacity of the evaporator increases,
    - The quantity of superheated vapor increases and tends asymptotically towards a constant value.
    - The temperature of the evaporating refrigerant slightly decreases in the evaporating sector. The temperature decrease is caused by the pressure drop because the viscous friction is present.
    - By increasing the mass flow rate of well water, the water temperature decreases.

New Scientific Results

a) Evaporative heat transfer coefficient in the model was implanted as a function of vapor quality according to Kandlikar [1],

b) Viscous friction coefficient of refrigerant was applied according to Friedel [2],

c) State equations of refrigerant R134a were used according to Martin-Hou [3],

d) In addition, using the new model behavior of heat pumps' coaxial evaporator was simulated (Figures 2, 3, 4 and 5).

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