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# SIMULINK MODEL FOR A HEAT-EXCHANGER

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***Abstract:*** *The main purpose of the dynamic analysis is to study the behavior of the phase-change heat-exchanger as component of the whole refrigeration system. The matter of modification of the operating conditions has been treated very often in the last few years because the refrigeration systems "on the field" are stressed by changes in temperature regime, pressure mass flow, heat transfer conditions and, especially, by changes of the thermal load. These regards have imposed the approach of the refrigeration systems from the dynamic point of view of the functioning regime. The major advantage brought by this dynamic approach is that it makes possible the designing and the analysis of a refrigeration system before its physical existence and a constructive and functional optimization which can be made with lower costs*
***Keywords: dynamic system, heat exchanger,*** *Simulink modeling*

## 1. INTRODUCTION

The purpose of the dynamic analysis is to study the behavior of the thermal systems, but also the behavior of each part of this system, at different constraints. This dynamic analysis can lead to results with direct action on manufacturing costs, exploitation and maintenance costs, environmental protection and also, on the operating safety, on the reliability, the functioning and the maintenance of the system.

The major advantage brought by this dynamic approach is that it makes possible the designing and the analysis of a thermal system before its physical existence and a constructive and functional optimization can be made with lower costs. More than, the dynamic regime functioning model gives the opportunity to make a better choice of the automatization elements for the system, to diagnose possible malfunctions of the system under certain conditions, and to develop new constructive models of the system with improved thermodynamic and economic features.

## 2. BACKGROUND

Simulink model for heat-exchanger with phase-change, in this case the shell-tube condenser, it is using the mathematical model for this type of heat-exchanger, based on functional model presented in Figure-1, and it is containing a differential equation system is presented. This differential equation system, the theoretical approach to obtain a numerically functional model is described considering a model with lumped parameters. The condenser numeric model (see Figure-2) is constructed using four sections, corresponding to the condenser main constructive parts, respectively: the vapor condensation section (see eq.-1), the tube section (see eq.-3), the liquid section (see eq.-4) and the shell section. Each section is represented by a differential equation system resulted from differential form of mass and energy conservation equations.

Heat transfer coefficients used are modeled as was possible like a general model with small number of coefficients depending on the refrigerant type and depending on the temperature and pressure. For presented case of study R134a was chosen as refrigerant, and water as secondary cooling medium. The thermodynamic properties for R134a are included like refrigerant library, likewise for water properties. These general form libraries can be very easily replaced to analyze performance of the other pair refrigerant-cooling media for condenser.

Input process variables considered in the model were: the refrigerant inlet mass flow, the refrigerant inlet enthalpy (this input variable was taken same as the enthalpy of end of compression), the water inlet mass flow, and, the water inlet temperature. Simulations performed for numerical model were considered for step variations of the input process variables, and for any of these changes the output variables shows the new dynamic regime state. The variations for input variables considered in simulation are: refrigerant: R134a; inlet refrigerant temperature is $t_{in}$=+69°C, corresponding to a superheated state with specific enthalpy $h_{in}$=438.5kJ/kg, inlet refrigerant vapor mass flow, step variable in range of [0.3…0.35]kg/s; secondary cooling medium: water, inlet water temperature, step variable in range of [+15…+25]°C, inlet water mass flow, step variable in range of [2.4…3.0]kg/s. The constructive variables considered in the model simulation were: length of flow section is 2.33m, tube steel dimensions are Ø21/19mm; shell tube condenser is built with 3 flow sections on secondary medium side, with 30 tubes on each flow section, heat transfer surface is 12.5m$^2$; shell dimensions: Di/De=Ø324/330mm.

To cover all common cases of dynamic regimes the step variations for input variables were considered as decreasing values for inlet water temperature at 500sec. the moment of time from the beginning of simulation, increasing values for water mass flow at 1000sec. the moment of time, and finally increasing values for refrigerant mass flow at 1500 sec. the moment of time. The range from 0 to 500sec. the moment of time is left without input variable changes to find out the initial steady-state regime.

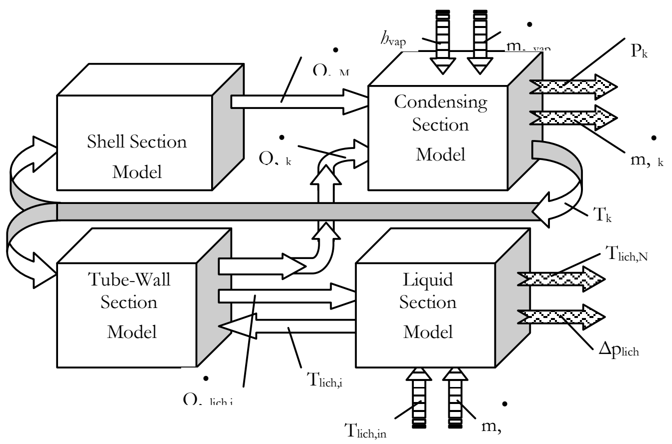


**Figure 1:** Functional model of shell-tube heat-exchanger

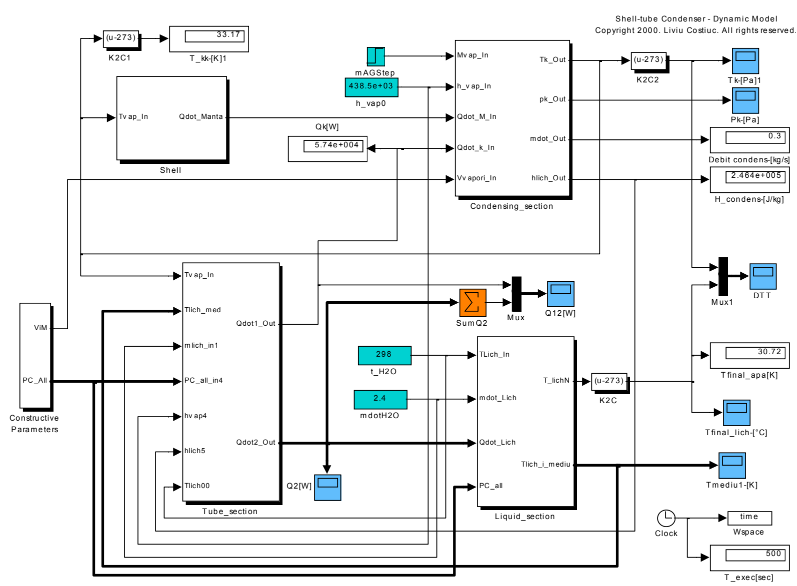


**Figure 2:** Simulink model for heat-exchanger

## 3. MATHEMATICAL MODEL

The mathematical model is developed using the differential form of energy, mass balance equations corresponding to each model section as follows:

- the differential equation for *Shell Section*:

$$\frac{dT_{Shell}}{dt} = \frac{\dot{Q}_{shell}}{m_{shell} \cdot c_{p\,shell}} \ \text{[K/s]} \tag{1}$$

where: $\dot{Q}_{Shell}$ [W]- absorbed heat flow by the shell material, calculated with:

$$\dot{Q}_{shell} = \alpha_{MV} \cdot \mathrm{A}_{\mathrm{shell}} \cdot \left(T_{vap} - T_{shell}\right) \ \text{[W]} \tag{2}$$

and: $\alpha_{MV}$ [W/m$^2$.K] - is the heat transfer coefficient between refrigerant vapor and shell material; $A_{Shell}$ [m$^2$]- is the shell heat transfer area; $m_{Shell}$ [kg]- shell mass; $c_{p\,Shell}$ [J/kg.K]- specific heat flow coefficient of the shell material; $T_{Shell}$ [K]- shell temperature; $T_{vap}$ [K]- refrigerant vapor temperature;

- differential equation for *Phase-Change Section*:

$$\frac{dT_{vap}}{dt} = \left( \frac{\dot{m}_{\mathrm{vap}}\,(\mathrm{h}_{\mathrm{vap\_in}} - \mathrm{h}_{\mathrm{lich}}) - \dot{Q}_{\mathrm{Shell}} - \dot{Q}_{\mathrm{K}}}{\mathrm{CT}_{\mathrm{vap}}} \right) \tag{3}$$

where: $\dot{m}_{vap}$ [kg/s] – inlet refrigerant mass flow; $h_{vap\_in}$ [J/kg]- masic enthalpy of the inlet vapor refrigerant; $h_{lich}$ [J/kg]- masic enthalpy of the outlet liquid refrigerant; $\dot{Q}_K$ [W]- condensing heat flow, it is computed into the *Tube-Wall Section*. The relations for condensed liquid mass flow, $\dot{m}_{cond}$ and $CT_{vap}$ – thermal capacity of the refrigerant vapors, are computed with:

$$\dot{m}_{cond} = \dot{m}_{vap} - \frac{dT_{vap}}{dt}\left(\frac{\partial \rho}{\partial T}\right) \cdot V, \ \text{[kg/s]} \tag{4}$$

$$CT_{vap} = V\left[\rho_{vap}\left(\frac{\partial h}{\partial T}\right)_{sat} + \left(h_{vap,in} - h_{lich}\right)\left(\frac{\partial \rho}{\partial T}\right)_{sat} - \left(\frac{\partial P}{\partial T}\right)_{sat}\right], \text{ in [J/K]} \tag{5}$$

where: $V$ [m$^3$] -the vapor volume inside the shell condenser, $\rho_{vap}$ [kg/m$^3$] –the vapor density, and the partial derivatives for pressure, density and enthalpy in relation (5) are considered for saturated refrigerant vapor state, and the expressions are obtained using thermodynamic properties relations.

- differential equation for *Tube Wall Section*:

$$\frac{dT_{wall}}{dt} = \left( \frac{\dot{Q}_{\mathrm{k}} - \dot{Q}_{\mathrm{TL}}}{\mathrm{N}_{\mathrm{tv}}\ \mathrm{m}_{\mathrm{tv}}\ \mathrm{c}_{\mathrm{p\,tv}}} \right), \ \text{[K/s]} \tag{6}$$

where: $T_{wall}$ [K]- wall tube temperature; $\dot{Q}_{TL}$ [W]- heat flow transferred to the cooling liquid medium; $N_{tv}$- number of tubes on a flow section; $m_{tv}$ [kg]- mass for one pipe; $c_{p\,tevi}$ [J/kg.K]- specific heat flow coefficient of the pipe material;

The transferred heat flow to the cooling liquid was computed for each flow section, and resulting total heat flow $\dot{Q}_{TL}$ it is the sum of the $\dot{Q}_{TL,i}$ section heat flow. These heat flows, $\dot{Q}_{TL,i}$, where calculated using mean liquid temperature, heat transfer coefficient and corresponding thermal properties at the mean liquid temperature. The used expression it is:

$$\dot{Q}_{TL} = \Sigma \dot{Q}_{TL,i}\text{, } i = 1...N\text{, and where: } \dot{Q}_{TL,i} = \frac{1}{Rt_{tv-lich,i}}(\mathrm{T_{tevi}} - \overline{T}_{\mathrm{lich,i}}) \text{ [W]} \quad (7)$$

$N$ –number of sections, $Rt_{tv\text{-}lich,i}$ –thermal resistance for section-$i$ , at water side.

For condensing heat flow $\dot{Q}_{\mathrm{k}}$ the relation used it is:

$$\dot{Q}_k = \frac{1}{Rt_{tv-vap}}(\mathrm{T_{vap}} - T_{\mathrm{tv}}) \text{ [W]} \quad (9)$$

where $Rt_{tv\text{-}vap}$ is the thermal resistance at refrigerant side, $\alpha_{tv\text{-}vap}$ [W/m$^2$.K] -heat transfer coefficient between refrigerant vapor and pipe wall, $\alpha_{tv\text{-}lich}$ [W/m$^2$.K] -heat transfer coefficient between cooling liquid and pipe wall, $\lambda_{tv}$ [W/m.K] - thermal conductivity of the pipe wall; $d_e$ , $d_m$ , $d_i$ [m] - external, medium and internal pipe diameter, $A_{i,tv}$ , $A_{e,tv}$ [m$^2$] - internal and external, for one pipe, heat transfer area, $L_{tv}$ [m]- pipe length, $\overline{T}_{lich,i}$ - mean liquid temperature for section $i$, computed like mean arithmetic temperature between pipe inlet and outlet temperatures.

- differential equation for *Liquid Section,* section-$i$:

$$\frac{dT_{lich,i}}{dt} = \left( \frac{\dot{Q}_{TL,i} + \dot{m}_{lich} \cdot c_{p\,lich,i} \cdot (T_{lich,i-1} - T_{lich,i})}{\rho_{lich,i} \cdot c_{p\,lich,i} \cdot V_{lich,i}} \right), \quad (10)$$

where: $i = 1...N$, $N$ number of tubes, $T_{lich,i}$ – outlet liquid temperature from section-$i$ [K], $\dot{Q}_{TL,i}$ – heat flow transferred to liquid in section-$i$ [W], $\dot{m}_{lich}$ – liquid mass flow [kg], $c_{p\ lich,i}$ - specific heat flow coefficient of the liquid for section-$i$ [J/kg.K], $T_{lich,\ i-1}$ – inlet temperature for section-$i$ [K], $\rho_{lich,i}$ - liquid density for section-$i$ [kg/m$^3$], $V_{lich,i}$- liquid volume from section-$i$ [m$^3$].

## 4. SIMULATION RESULTS

The simulation results are presented in Figure-3 to Figure-8. The step variations for input variables were considered as decreasing from +25°C to +15°C for inlet water temperature at 500 sec. the moment of time from the beginning of simulation, increasing from 2.4kg/s to 2.8kg/s for water mass flow at 1000 sec. the moment of time, and increasing from 0.3kg/s to 0.35kg/s for refrigerant mass flow at 1500 sec. the moment of time.

In all cases the dynamic regime reaches the new functioning state for time ranges in [120...500] seconds. The condensing temperature isn't a constant value as it is considered in most cases. The simulation shows (see Figure-3) the new values for condensing temperature depending on input variable changes and the new solution resulted from solving differential system equation with new initial conditions.

In the case of study in first 500 sec. the stationary regime was reached in 140 sec. with condensing temperature at +33.16°C, condensing pressure value 8.44 bar, water outlet temperature value +30.73°C, sections heat flow values 32.40/16.50/0.85kW, refrigerant heat transfer coefficient value 2750W/m$^2$K, mean flow section temperature values +30.73/29.9/28.3°C.

For next time range of 500 sec., when the water temperature is step decreasing variable the new stationary regime was reached in 180 sec. with condensing temperature at +24.0°C, condensing pressure value 6.47 bar, water outlet temperature value +21.15°C, sections heat flow values 33.30/17.00/1.00kW, refrigerant heat transfer coefficient value 2835W/m$^2$K, mean flow section temperature values +21.15/20.20/18.35°C, and water heat transfer coefficients on each section values 1480/1460/1420W/m$^2$K.

Next simulation was the step variation of water mass flow beginning with the 1000 sec. moment of time and the step variation of refrigerant mass flow beginning with the 1500 sec. moment of time. As it can be seen in figures, from 3 to 8, the main and substantial influence on the simulated parameters is the cooling water temperature of condenser. This variation step of 10°C determines the decreasing of condensing temperature with 9.16°C, respectively a pressure decreasing from 8.44 bar to 6.47 bar. This fact points to a stage of designing of refrigeration plants and also thermal plants which are using these types of heat-exchangers when the design condenser pressure or temperature it is taken as a constant, and this is going to a bigger constructive shell-tube values. The variation of the two mass flows considered in simulation are smaller than the influence of water temperature regarding condensing temperature, finding a -1.16°C change considering the water mass flow step change, and a +1.46°C change considering the refrigerant mass flow step change.

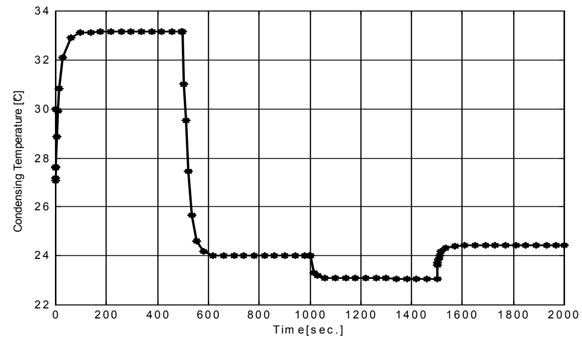


**Figure 3** Condensing temperature[°C] vs. Time[sec.]

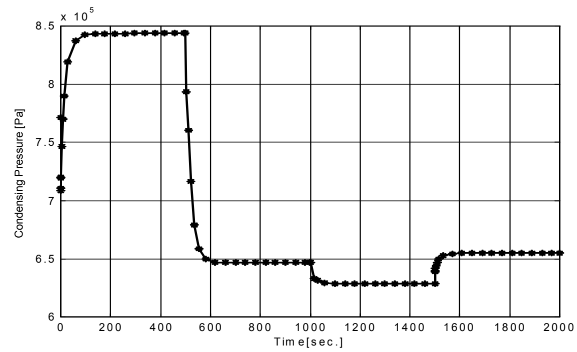


**Figure-4** Condensing pressure[bar] evolution vs. time[sec.]

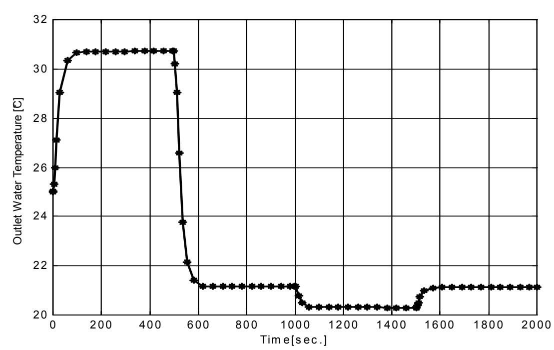


**Figure-5** Outlet water temperature[°C] evolution vs. time[sec.]

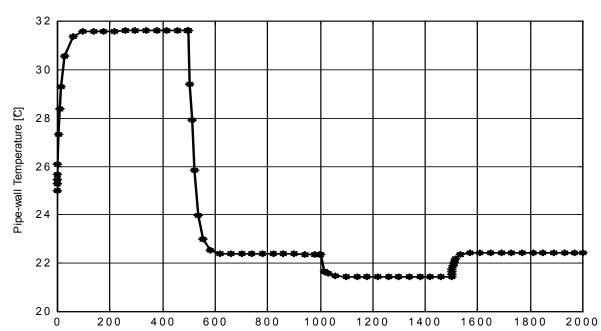


**Figure -6** Pipe-wall temperature[°C] evolution vs. Time [sec.]

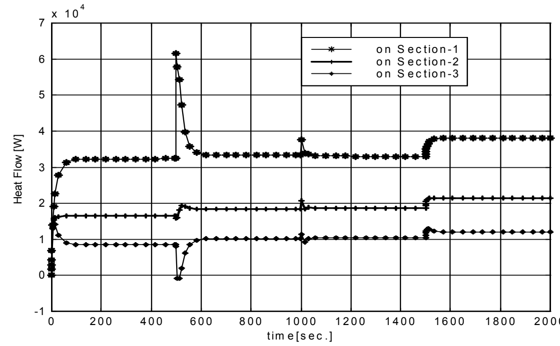


**Figure-7** Heat flow [$10^4$W] evolution on each flow section vs. time [sec.].

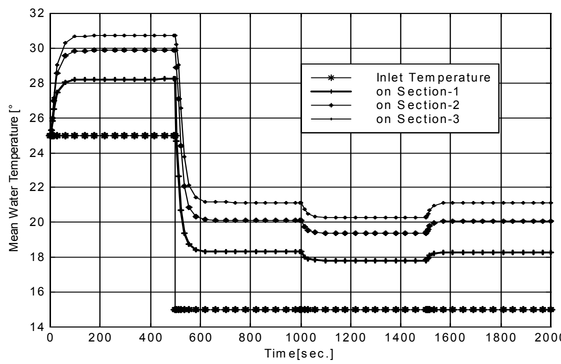


**Figure-8** Section mean water temperature[°C] vs. time[sec.].

## 5.CONCLUSIONS

The advantage brought by the developing a proper environment for the functioning designing and analysis on dynamic regime of the thermal systems are: the possibility of developing the advanced models for different component units of the refrigeration system with mechanical vapor compression and of complex models, easy to configure, for the whole system with mechanical vapor compression; elaborating of a computerized data base for the usual and for the new ecological refrigerating agents, in order to extend the model utilization to different types of refrigeration systems applications; computer implementation of the refrigeration system model functioning in dynamic regime and the developing of an interface for the users that allows the modification, visualizing and checking out the values important for the designing and the functioning optimization of the whole plant or of its component units.

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