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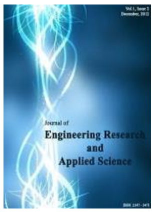

Steady-state simulation of heat pump using milk pasteurization

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Abstract
A simulation model to determine the steady- steady performance of heat pump for specified milk temperatures both source and sink medium and component sizes has been developed. Simple mathematical models were employed for each component of the milk to milk heat pump system with a plate economizer. They resulted in a set of nonlinear equations, which was solved numerically. This simulation program is based on a mathematical model of the thermodynamic cycle and on experimental data from equipment testing. The predicted performance was compared to the experimental result used that of an existing R-114 unit, showing good agreement. As an application, a comparative analysis is made on the thermodynamic performance of a heat pump using milk pasteurization on two different refrigerants: R-114 and R-134a. Although according that same temperature of pasteurization milk discharge pressure ratio of heat pump compressor is to be very high level for R-134a, the predicted model with R-134a was shown that it was been applicable to heat pump system using milk pasteurization.

Keywords: Heat pump; simulation; milk pasteurization; R-134a.

1. Introduction

The increase in the cost of energy has made the use of heat pumps advantageous systems for the production of efficient energy. It is obvious that heat pumps will become more and more competitive as an economical alternative source of energy as long as the price of conventional energy continues to increase [1,2]. Çomaklı *et al.*, [3] and Ozyurt *et al.*, [4] investigated the applicability of heat pumps to milk pasteurization for cheese production and compared the results with classical pasteurization systems. Energy consumption to pasteurize one kilogram milk was presented in Table 1 in the previous study by Ozyurt *et al.* [4]. Table shows energy types including heat energy, electrical energy, equivalent energy of cooling water, and total energy. The heat pump consumes only electrical energy, the plate pasteurizer consumes heat energy and electrical energy, and the double jacket boiler system consumes heat, electrical and equivalent energy of cooling water. The comparison of the total energy shows that the heat pump consumed 182.8 kJ/kg-milk of total energy, while the plate pasteurizer and the double jacket boiler system consumed 346.86 and 509.87 kJ/kg-milk, respectively. These results reveal that the energy consumption to pasteurize one kg milk for the heat pump is significantly less than that of the two classical systems. The plate pasteurizer consumed 1.9 times of the energy of the heat pump, and the double jacket boiler system consumed 2.8 times of the energy of the heat pump. It can be expressed that using the heat pump for pasteurizing milk for cheese production is advantageous on the basis of energy consumption [4].

The use of simulation methods in the study of heat pumps is a relatively recent development. A number of methods may be used to predict the performance of the vapor compression cycle. They may be simplistically divided into five categories as below [5]:

(1) Carnot cycle analysis;
(2) Simple methods based on fundamental observations and principles;
(3) Theoretical and semi-theoretical cycle analysis;
(4) Detailed equipment simulation-programs;
(5) Laboratory test of the vapor compression machine.

Several studies on the simulation of vapor compression systems can be found in the literature [6-8]. The degree of complexity varies considerably such as [6]:

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(a) Some of these models are based simply on empirical curves describing the performance of the components. Simulation, in such cases, consists in finding the balance point between the condensing unit (condenser and compressor) and the evaporator [7]. This can be accomplished graphically or mathematically, with equations representing the performance data of the components [8].

(b) In a second category of simulation models, comprising the majority of existing models, equations are established to describe the behavior of each component and of the system as a whole. The number of equations for each component is generally small and the result ia a system of algebraic non-linear equations. The way the system of equations is solved also varies. It can be solved simultaneously, with an appropriate numerical method, such as successive substitutions or the Newton method for multiple variables [5,6,9-11]. Saraç *et al.*, [12] developed a new algorithm for solution of the non- linear equation systems appearing in the design of air-compression systems.

(c) Components are now modeled in much a greater detail. Heat transfer zones that is superheated vapor, two-phase region, subcooled liquid, each different heat transfer characteristics and balance equations, are created for the condenser and evaporator [13,14].

Table 1. Technical features of the experimental set-up.

Milk cycle information:	
Milk pasteurization temperature	72^0C
Milk coagulation temperature	32-34^0C
Milk pump	Volume flow rate 1,2 m^3/h, mono faze motor (0,75 HP and 2800 rpm.)
Economizer	Plate type and Cr-Ni 304 stainless steel, heat exchange surface: 0.0442 m^2 per plate
Heat pump information:	
Capacity	9 kW (500 liter/hour)
Compressor type	Rotary (semi-hermetic)
Compressor power input	3.14 kWh
Compressor displacement volume	$8x10^{-3}$ m^3 $hour^{-1}$
Compressor rotation speed	Three faze 2880 rpm
Evaporator type	Milk cooled shell and tube
Evaporating temperature	22^0C
Condenser type	Milk cooled shell and tube
Condensing temperature	82^0C
Refrigerant type	R114
Maximum working pressure	9.8 bar

On the other hand, dynamic modeling, i.e. the simulation of the transient behavior, of vapor compression systems could be established. Macarthur developed a theoretical analysis of the dynamic interactions of vapor compression heat pumps [15]. Krakow and Lin [16] developed a computer model for the simulation of multiple source heat pump performance. The model is based on experimental investigations of a solar-source heat pump and a multiple-source heat pump system for cold climates. Çomaklı *et al.*, [17] developed a dynamic simulation program for a solar-assisted heat pump system with energy storage. In their study, experimental results were compared with the dynamic simulation results. In the literature, some of models are aimed to validate by experiment thermodynamic and simulation modeling of solar-assisted heat pump system with latent heat energy storage and to develop a tool for their design. A mathematical model was developed for the energy storage and energy conversion periods. Design parameters were calculated by means of regression analysis using experimental data obtained from the experimental set-up, and simulation studies were performed by means of the model developed [18-21].

More recently, the search for CFC replacement has once again proved the worthiness of simulation models [5, 13]. New refrigerants and mixtures, some of them still only available in restricted quantities, could rapidly be given a first assessment of their performance ratings [14, 22]. The aim of this study was to investigate of using R134a as a working fluid to replace R114 for heat pumps to milk pasteurization. For this purpose a simulation model, for liquid-to-liquid heat pump system with a plate economizer was used, is developed in this study. For a given geometry and set of operating conditions, the model is used to predict system performance. The experimental system used in the present investigation was a milk -to-milk heat pump system as shown schematically in Figure 3 is already given in the previous study by Ozyurt *et al.* [4].

2. The simulation model

The basic cycle of the simulation model is shown schematically in Figure 1. The system consists of five fundamental components namely: compressor, condenser, expansion valve, evaporator and a milk-to-milk plate heat exchanger was used as an economizer. The simulation model is specific for heat pump to milk pasteurization and uses cycle analysis to predict steady – state operating the conditions at points around the system. The pressure–enthalpy diagram shown in Figure 2 represents the heat pump as modeled by the computer simulation. The flow diagram of the simulation is shown in Figure 3. During development of the mathematical model, some assumptions were made to simplify the numerical solution.

These assumptions can be summarized as follows:

(a) Equilibrium thermodynamic relationships may be applied in general.

(b) The system's components are well insulated

(c) The pressure drop is negligible in the evaporator and in the condenser.

(d) The compression is considered polytropic process of constant index, n, in the compressor.

(e) The expansion is considered isenthalpic in the expansion valve.

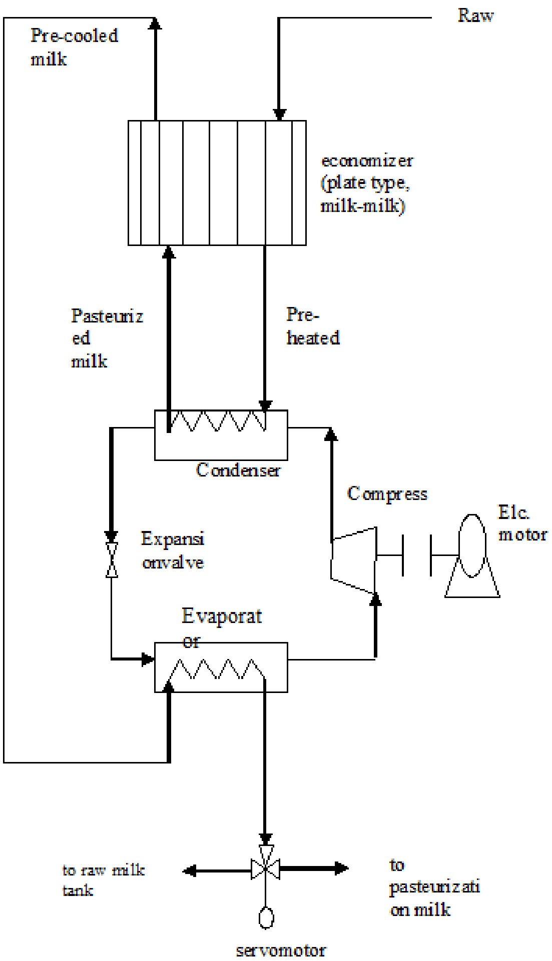

Figure 1. Schematic view of the simulation model.

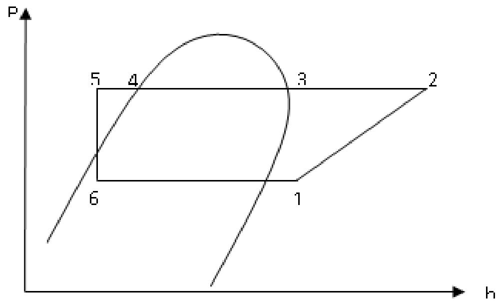

Figure 2. The pressure – enthalpy diagram for heat pump.

The design equations used in the model are given as follows:
Semi-hermetic compressor used in the heat pump system was a pallet rotating type. The refrigerant mass flow rate is obtained from,

$$\dot{m}_r = V_{dep}\frac{\lambda}{v_1} \tag{2.1}$$

where V_{dep} is the displacement volume of the compressor (m^3/s), λ is the volumetric efficiency of the compressor and v_1 is the specific volume of the refrigerant at the compressor inlet.

The ratio of density at outlet and inlet of the compressor as to it is assumed that compression follows a polytropic process of constant index, n can be obtained.

$$\frac{\rho_2}{\rho_1} = \left(\frac{P_{cd}}{P_{ev}}\right)^{\frac{1}{n}} \tag{2.2}$$

In this equation, n is the polytropic exponent and it is calculated by means of the procedure described Stocker and Jones [7].

2.2. Condenser
Three regimes may exist in the condenser:
(1) High pressure superheated refrigerant vapor entering from the compressor (2-3),

$$\dot{Q}_{23} = \dot{m}_r\left(h_2 - h_3\right) \tag{2.3}$$

(2) Refrigerant vapor condensing at saturation conditions (3-4),

$$\dot{Q}_{34} = \dot{m}_r\left(h_3 - h_4\right) \tag{2.4}$$

(3) Subcooled liquid in the exit section (4-5)

$$\dot{Q}_{45} = \dot{m}_r\left(h_4 - h_5\right) \tag{2.5}$$

Total heat transfer rate of refrigerant in the condenser can be obtained from equations 2.3, 2.4 and 2.5.

2.1. Compressor

$$\dot{Q}_{cd} = \dot{m}_r\left(h_2 - h_5\right) \tag{2.6}$$

The heat transfer rate absorbed/released from the condenser is calculated using the following equation:

$$\dot{Q}_{cd} = \dot{m}_m c_{p,m}\left(T_{mo,cd} - T_{mi,cd}\right) \tag{2.7}$$

where $\dot{m}_m$ is the milk mass flow rate, and $T_{mi,cd}$ and $T_{mo,cd}$ are the milk temperatures entering and leaving inside the condenser, respectively.

Effectiveness method for condenser is as follows:

$$\varepsilon_{cd} = 1 - e^{-NTU} \tag{2.8}$$

Equation (2.8), derived for heat exchangers with one of the fluids undergoing phase change exclusively is an approximation as the condenser has also to cope with desuperheating and eventual subcooling of the refrigerant [23,24].

The heat transfer equation refrigerant to milk can be obtained using the effectiveness method;

$$\dot{Q}_{cd} = \dot{m}_m c_{p,m}\varepsilon_{cd}\left(T_2 - T_{mi,cd}\right) \tag{2.9}$$

where $\dot{m}_m$ is the milk mass flow rate, $c_{p,m}$ is milk specific heat at constant pressure, ε_{cd} is effectiveness of condenser, T_2 is discharge temperature and $T_{mi,cd}$ is the milk temperatures entering inside the condenser, respectively.

2.3. Evaporator
The condenser and the evaporator are of shell-tube and hermetic type. The refrigerant flows inside a double tube located inside a smooth outer tube, and the heat input is provided by milk flowing in the shell. The evaporator section may be characterized by two regimes: a mixture of liquid and vapuor leaving the expansion valve enters the evaporator and

is heated to saturated vapor and the vapour is then superheated before entering the compressor.

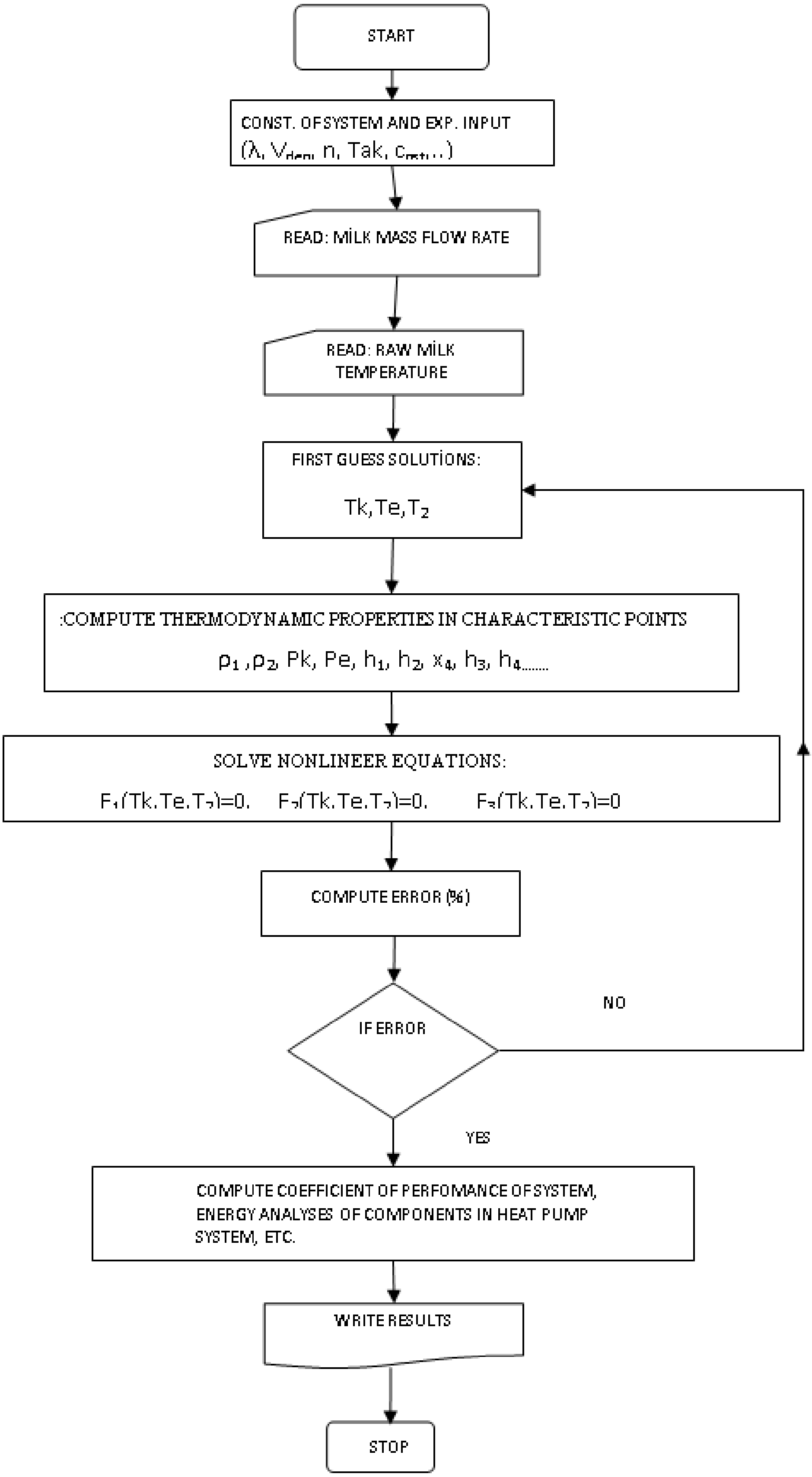

Figure 3. The flow diagram of the heat pump system simulation.

The evaporator equations are as follows,

$$\dot{Q}_{ev} = \dot{m}_r \left[(1-x_6)h_{sbev} + (h_1 - h_{bTev})\right] \quad (2.10)$$

$$\dot{Q}_{ev} = \dot{m}_m c_{p,m} (T_{mi,ev} - T_{mo,ev}) \quad (2.11)$$

Effectiveness method for evaporator is as follows:

$$\varepsilon_{ev} = 1 - e^{-NTU} \quad (2.12)$$

The milk to refrigerant heat transfer equation can be obtained using the effectiveness method;

$$\dot{Q}_{ev} = \dot{m}_m c_{p,m} \varepsilon_{ev} (T_{mi,ev} - T_{ev}) \quad (2.13)$$

where $\dot{m}_m$ is the milk mass flow rate, $c_{p,m}$ is milk specific heat at constant pressure, ε_{ev} is effectiveness of evaporator, $T_{mi,ev}$ is the milk temperatures entering inside the evaporator and T_{ev} is evaporation temperature, respectively.

2.4. Expansion device

The expansion process is based thermostatic expansion valve that modulates the flow of refrigerant to maintain a constant superheat at the evaporator outlet. With no work done and assuming no heat transfer across the device, the enthalpy of refrigerant remains constant.

$$h_5 = h_6 \quad (2.14)$$

2.5. Heat exchanger (economizer)

The economizer used in heat pump system is a plate-frame type heat exchanger as shown in Figure 4. The raw milk is sent into the economizer by the milk pump, and preheated by the hot milk from the condenser. At the same time, the hot milk from the condenser is pre-cooled in the economizer by the raw milk, and then sent into the evaporator.

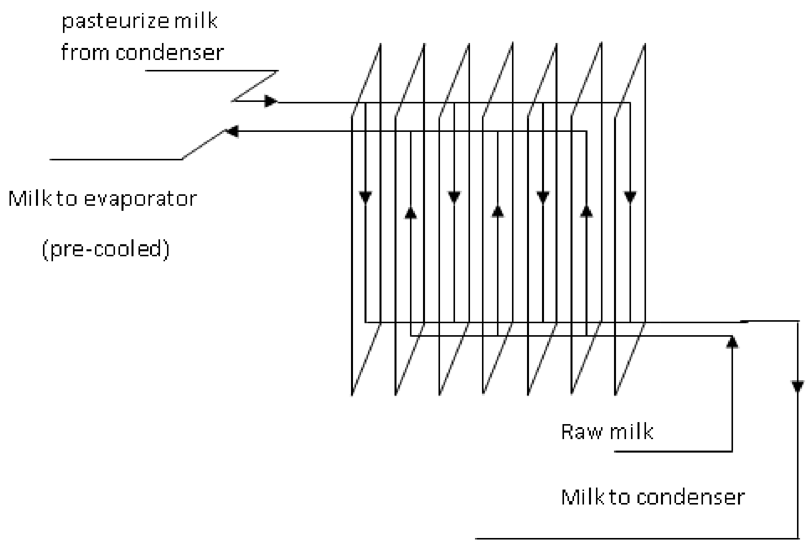

Figure 4. The economizer used in heat pump system is a plate-frame type heat exchanger.

Milk temperatures entering inside the condenser and the evaporator are calculated by the economizer model. Preheated rate to raw milk in the economizer is defined as;

$$\dot{Q}_{ec} = \dot{m}_m c_{p,m} (T_{mi,cd} - T_{raw}) \quad 2.15)$$

Pre-cooled rate to pasteurize milk in the economizer is defined as;

$$\dot{Q}_{ec} = \dot{m}_m c_{p,m} (T_{mo,cd} - T_{mi,ev}) \quad 2.16)$$

In this case no heat loss to ambient is considered. Effectiveness method ($\varepsilon - NTU$) is used for economizer [23-25].

Assumptions made in the economizer model:

(1) The heat exchanger operates under steady-state conditions.
(2) The overall heat transfer coefficient is constant throughout the heat exchanger.
(3) Fluid (milk) temperatures and velocities are uniform across the channel.
(4) Heat is not conducted in the direction of flow.
(5) The fluids are equally distributed between the channels.

The economizer heat capacity;

$$\dot{Q}_{ec} = \dot{m}_m c_{p,m} \varepsilon_{ec} (T_{mo,cd} - T_{raw}) \quad (2.17)$$

the effectiveness, ε_{ec} is defined as;

$$\varepsilon_{ec} = \frac{\exp[(1-R)NTU]-1}{\exp[(1-R)NTU]-R} \tag{2.18}$$

where, the definitions R and NTU are

$$R = \frac{(\dot{m}_m c_{p,m})_{cold}}{(\dot{m}_m c_{p,m})_{hot}} = \frac{T_{mo,cd} - T_{mi,ev}}{T_{mi,cd} - T_{raw}} \quad 2.19)$$

and

$$NTU = \frac{A_{ec} U_{ec}}{\dot{m}_m c_{p,m}} \tag{2.20}$$

Milk temperatures entering to condenser is defined from equation (2.15) as follows

$$T_{mi,cd} = T_{raw} + \frac{\dot{Q}_{ec}}{\dot{m}_m c_{p,m}} \tag{2.21}$$

Milk temperatures entering to evaporator is defined from equations (2.16) and (2.17) as follows

$$T_{mi,ev} = T_{raw} + \frac{\dot{Q}_{ec}}{(\dot{m}_m c_{p,m})\varepsilon_{ec}}(1-\varepsilon_{ec}) \tag{2.22}$$

2.6. Refrigerants property equations

Refrigerant property also should be calculated in the simulation model. In general, the refrigerant property equations can be established as follows:

$$v_2 = v_v(T_2, P_{cd}) \tag{2.23}$$

$$v_1 = v_v(T_1, P_{ev}) \tag{2.24}$$

$$h_2 = h_v(T_2, P_{cd}) \tag{2.25}$$

$$h_1 = h_v(T_1, P_{ev}) \tag{2.26}$$

$$h_{lvc} = h_{lv}(T_{cd}) \tag{2.27}$$

$$h_{lve} = h_{lv}(T_{ev}) \tag{2.28}$$

$$h_4 = h_{4l} + x h_{lvev} \tag{2.29}$$

$$h_{4l} = h_v(T_{ev}, P_{ev}) - h_{lvev} = h_1(T_{ev}) \tag{2.30}$$

$$h_3 = h_v(T_3, P_{sat}(P_3)) - h_{lv}(T_3) = h_1(T_3) \tag{2.31}$$

$$P_{ev} = P_{sat}(T_{ev}) \tag{2.32}$$

$$P_{cd} = P_{sat}(T_{cd}) \tag{2.33}$$

$$c_{v0} = c_{v0}(T) \tag{2.34}$$

Equations for Refrigerant-114 thermodynamic properties were obtained from Downing [26], Stewart *et al*. [27], Yılmaz and Oğulata [28], Mecarik and Masaryk, [29]. For Refrigerant-134a, equations have been provided by Wilson and Basu [30].

2.7. System of equations and solutions

By performing the mathematical model of the heat pump system that is used for milk pasteurization, three non-linear equations were obtained. The relations and numerical method were explained as follows:

In first system equation compression process in the compressor was obtained by assuming that it was a polytropic process,

$$\rho_2 - \rho_1 \left[\frac{P_k}{P_e}\right]^{\frac{1}{n}} = X(1) = (T_{ev}, T_{cd}, T_2) = 0 \tag{2.35}$$

where ρ_1 and ρ_2 are densities *of refrigerant* inlet and outlet of compressor. P_{cd} and P_{ev} are pressures of condensation and evaporation, respectively. There are three independent variables in system. They are temperature of evaporation, T_{ev}, temperature of condensation T_{cd}, and discharge temperature of compressor T_2, respectively. The other thermodynamic properties were described by depending on these three variables.

The process in expanding valve was considered as the second system equation. It is assumed that adiabatic expanding occurs at the expanding valve. According to this assumption, inlet and outlet enthalpies of expanding valve are to be equal.

$$h_5 - h_6 = X(2) = (T_{ev}, T_{cd}, T_2) = 0 \tag{2.36}$$

Where h_5 is specific enthalpy of refrigerant for inlet of expansion device, and h_6 is specific enthalpy of refrigerant for outlet of expansion device.

Third system equation is system energy equation.

$$\dot{W}_c + \dot{Q}_{ev} - \dot{Q}_{cd} = X(3) = (T_{ev}, T_{cd}, T_2) = 0 \tag{2.37}$$

In the system, the temperature of the refrigerant is increased a more little by preheating with heat exchanger before compressor inlet. Besides, a little temperature decreasing takes place on account of heat loss occuring from pipe between compressor outlet and condenser inlet. Depending on these familiars, it can be also seen from P-h diagram, work of compressor;

$$\dot{W}_c = \dot{m}_r\left(h_2 - h_1\right) \quad (2.38)$$

Heat loss amount occuring from pipe between compressor outlet and condenser inlet

$$\dot{Q}_{ls} = \dot{m}_r\left(h_2 - h_3\right) \quad (2.39)$$

If it is denoted the heat amount delivering to milk with $\dot{Q}_{cd}$ and heat amount withdrawing from evaporator with $\dot{Q}_{ev}$, total energy amount can be written as follows;

$$\dot{W}_c + \dot{Q}_{ev} - \dot{Q}_{cd} - \dot{Q}_{ls} = 0 \quad (2.40)$$

From here,

$$\dot{m}_r\left(h_2 - h_1\right) - \dot{m}_r\left(h_2 - h_3\right) + \dot{Q}_{ev} - \dot{Q}_{cd} = 0 \quad (2.41)$$

The equation given in 2.41 is obtained by arranging it;

$$\dot{m}_r\left(h_3 - h_1\right) + \dot{m}_m c_{p,m}\varepsilon_{ev}\left(T_{ev} - T_{mi,ev}\right) - \dot{m}_m c_{p,m}\varepsilon_{cd}\left(T_2 - T_{mi,cd}\right) = X(3) = \left(T_{ev}, T_{cd}, T_2\right) = 0$$

(2.42)

where $\dot{m}_r$ is mass flow rate of refrigerant and $\dot{m}_m$ is mass flow rate of milk. $T_{mi,ev}$ and $T_{mi,cd}$ correspond to inlet temperatures of evaporator and condenser, respectively. ε_{ev} for evaporator and ε_{cd} for condenser are effectiveness. h_3 and h_6 are specific enthalpy for inlet to condenser of refrigerant and specific enthalpy for outlet to evaporator of refrigerant, respectively.

In model, the equation, which was obtained from (2.10) and (2.13) equations, was used for vapor quality in evaporator inlet.

$$x_6 = 1 + \frac{c_{pv}T_s}{h_{lvev}} - \frac{\dot{m}_m c_{p,m}\varepsilon_{ev}\left(T_{mi,ev} - T_{ev}\right)}{\dot{m}_r h_{lvev}} \quad (2.43)$$

where T_s is super heating amount before inlet to compressor.

In model, the equation, which was obtained from (2.3), (2.4), (2.5) and. (2.9), was used for temperature value at condenser exit.

$$T_5 = T_{cd} - \frac{\dot{m}_m c_{p,m}\varepsilon_{cd}\left(T_2 - T_{mi,cd}\right)}{\dot{m}_r c_{pl}} + \frac{h_{lv,cd}}{c_{pl}} + \frac{c_{pv}\left(T_2 - T_{cd}\right)}{c_{pl}}$$

(2.44)

The usage of numerical methods is requiring for solution of three non-linear equations given in above. In this study, Newton method was chosen for solution of three non-linear equations. This method is a numerical analysis method that is applied for solution of non-linear equations. Approximate numerical values of three independent variables, which provide the equivalent of equation, were calculated with solution at the same time of system equations by applying Newton- Raphson method [31].

2.8. Coefficient of performance

The coefficient of performance (*COP*) for a heat pump cycle and heat pump system indicates the overall power consumption for a desired output and were evaluated using the following equations:

$$COP_{hp} = \frac{\dot{Q}_c}{\dot{W}_c} = \frac{\text{Energy delivered or removed}(kW)}{\textit{Energy consumed in compressor}\ (kW)}$$

(2.45)

$$COP_{sys} = \frac{\dot{Q}_c}{\dot{W}_c + \dot{W}_{mp}} = \frac{\text{Energy delivered or removed}(kW)}{\textit{Energy consumed in compresor}\text{ and milk pump }(kW)}$$

(2.46)

3. Results and discussion

The simulation model has been validated by comparison with accurate sets of data obtained from performance experimental systems conducted on R-114 operating in the heat pump to milk pasteurization test facility. The technical specifications of the experimental set-up are given in Table 1. Details of the experimental apparatus are given by Özyurt *et al.* (2004). Experimental and predicted values were compared for a number of operating conditions. For each run, the operating conditions (compressor speed, milk flow rate and raw milk temperatures) and empirical parameters (compressor volumetric coefficient, polytropic index of compression and condenser, evaporator and economizer overall conductance) were entered as input data. The input data used in the simulation model given in Table 2.

Figures 5, 6 and 7 show the comparison between the experimental and predicted values for discharge temperatures (*T_2*), condensation temperatures (*T_{cd}*) and evaporation temperatures (*T_{ev}*) at steady- state conditions. In general a good agreement was obtained.

Table 2. Computer simulation required inputs.

Data	Description	Value
λ	Volumetric efficiency of the compressor	0,7
V_{dep}	Displacement volume of the compressor (m^3/h)	23.04
n	Polytropic index of compression	1,16[a]
T_{sh}	Superheating temperature (˚C)	10
T_{sb}	Subcooling temperature (˚C)	10
$(UA)_{ev}$	Overall conductance of evaporator (W/˚C)	214
$(UA)_{cd}$	Overall conductance of condenser (W/˚C)	143
$(UA)_{ec}$	Overall conductance of economiser (W/˚C)	56
$T_{m,i}$	Milk inlet temperature (˚C)	24
m_m	Milk mass flow rate (kg/sn)	0.05
$c_{p,m}$	Specific heat of milk (kJ/kg ˚C)	3,84

[a] the polytropic index of compression was supposed to be the same for both working fluids (R134a and R114)

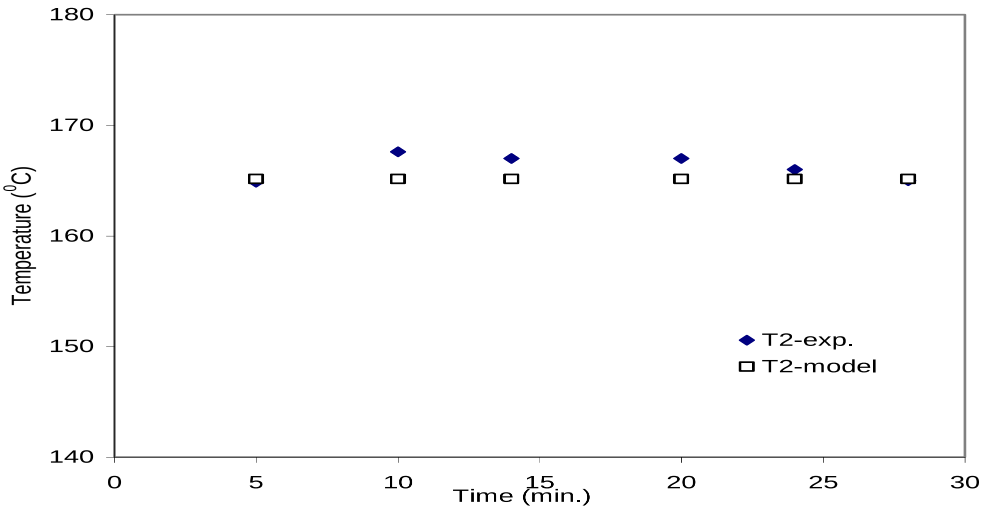

Figure 5. The comparison between the experimental and predicted values for discharge temperatures (T_2) at steady- state conditions.

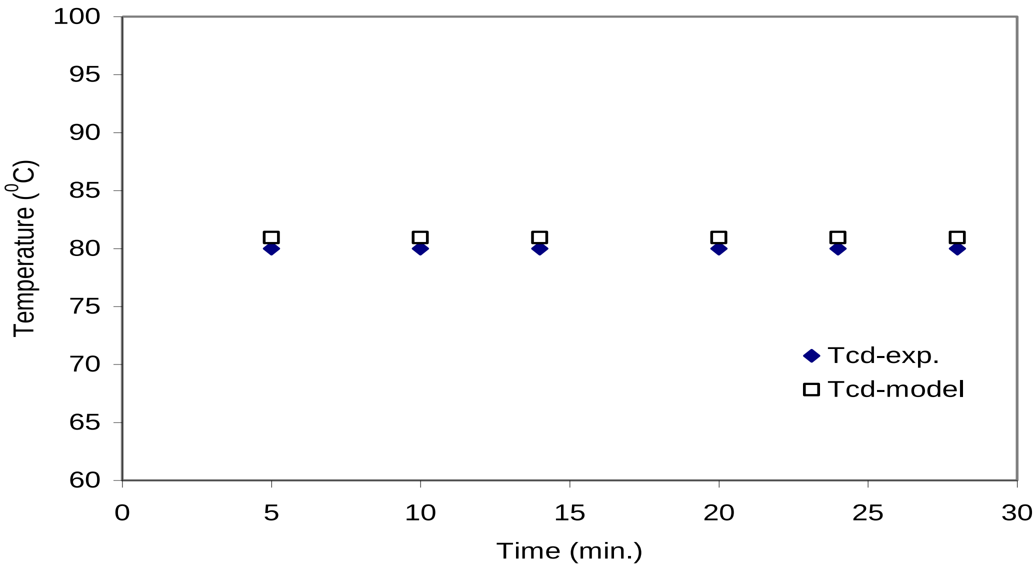

Figure 6. The comparison between the experimental and predicted values for condensation temperatures (T_{cd}) at steady-state conditions.

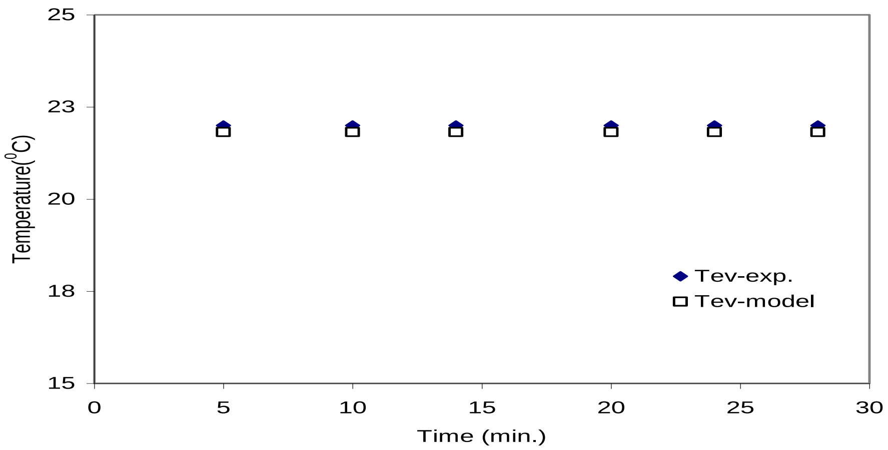

Figure 7 . The comparison between the experimental and predicted values for evaporation temperatures (T_{ev}) at steady-state conditions.

Figures 8 and 9 show the comparison for the variation of the milk temperatures at various points of the system with time for steady-state conditions. It is desirable that the milk temperatures be stable with time during the operation of the heat pump. Particularly, the outlet temperature of the condenser, which is the pasteurization temperature, and the outlet temperature of the evaporator, which is the coagulation temperature, is of much importance and they should be stable. The examination of these figures reveals that these temperatures were nearly stable along the experiments.

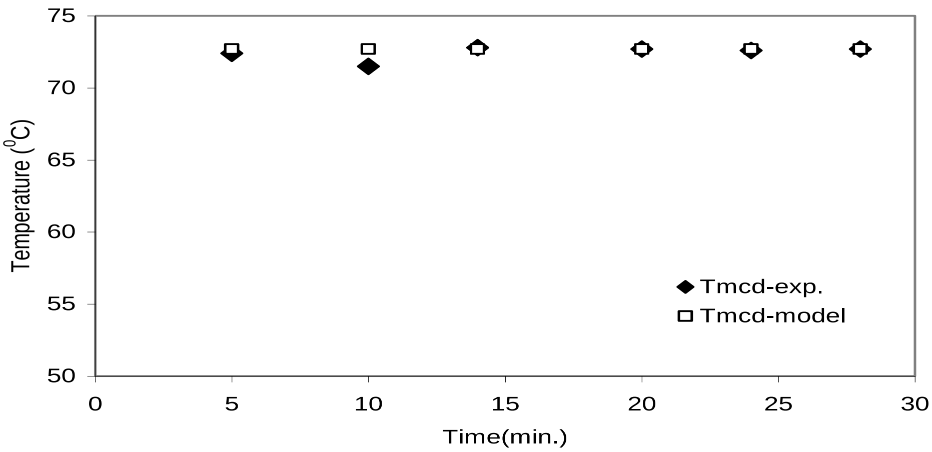

Figure 8. The comparison between the experimental and predicted values for milk temperatures of condenser outlet (T_{mcd}) at steady- state conditions.

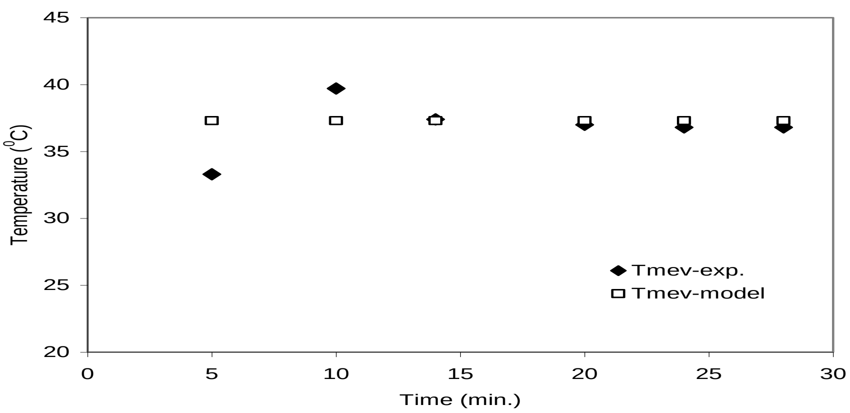

Figure 9. The comparison between the experimental and predicted values for milk temperatures of evaporator outlet (T_{mev}) at steady- state conditions.

The variation of refrigerant temperatures (T_2, T_{cd}, *and* T_{ev}) with milk flow rate for R114 and R134a are compared in Figures 10. It is observed that there is not a significant change for constant raw milk temperature, condensation and evaporation temperatures increases as the flow rate of milk increases for the constant temperature of raw milk.

It is undesired condition that the discharge temperatures of the compressor for R134a is higher than that of R114, as can be seen from Figure 11 in which the condensation pressure for R134a is significantly high.

In Table3 shows the typical experimental and predicted results. The difference between temperatures of the predicted and experimental temperatures for a specific operating point is, in general, within 2%. This indicates that the system equipment for R134a must be selected to endure the high pressure and temperatures. The reason of the increase in the COP can be related to the amount of heat transfer to milk in the condenser.

The R-114, which is the best refrigerant as technique, gas was chosen while building the experimental system. However, its usage was forbidden as it is determined that it is harmful to ozone layer. Therefore, it is required to be used an alternative refrigerant gas that is harmonious environment.

R-134a was used in designed model since it was determined that R-134a was the most suitable for this system in present alternative refrigerants. It was seen in the performed analysis with results of experiment system in analysis that the high pressure is required for R-134a and the outlet temperature of compressor is considerably high. The main precaution, which will be able to remove these inconveniences, is to find compressor that supply that pressure (about 26 bars) and to design the system. The other way is also to investigate the useful of an alternative refrigerant in system.

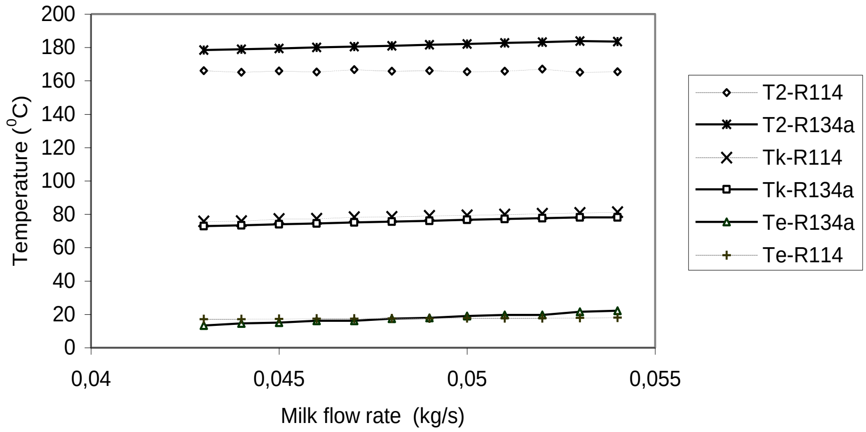

Figure 10. The comparison between variation of refrigerant temperatures (T_2, T_{cd}, T_{ev}) with milk flow rate for R114 and R134a at steady- state conditions.

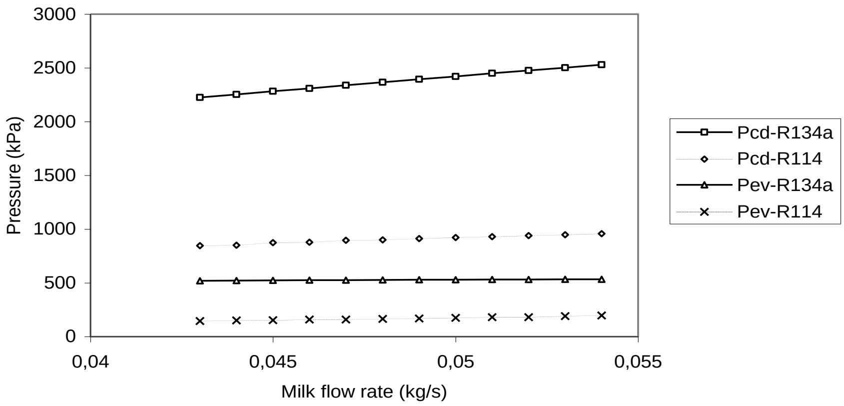

Figure 11. The comparison between condensation and evaporation pressures (P_{cd}, P_{ev}) with milk flow rate for R114 and R134a at steady- state conditions.

Table 3. Typical experimental and predicted results.

Data	R114 (exp.)	R114 (pred.)	R134a (pred.)
T_2 (^{0}C)	166	165.49	**182.12**
T_{cd} (^{0}C)	82	79.47	**76.64**
T_{ev} (^{0}C)	21.10	19.08	**17.68**
P_{cd} (kPa)	980	921.69	**2420.96**
P_{ev} (kPa)	190	176.07	**530.72**
Q_{cd} (kW)	8.14	8.51	**24.38**
Q_{ev} (kW)	4.86	5.46	**16.39**
W_c (kW)	3.21	3.04	**7.98**
COP_{hp}	2.54	2.79	**3.05**
COP_{sys}	**2.13**	**2.33**	**2.83**

4. Conclusions

In this study, a simulation model was developed for heat pump system with milk-to- milk plate heat exchanger in application of milk pasteurization. The system parameters were calculated by using experimental results given in the previous paper [4]. The important results that can be drawn are summarized as follows:

- Comparing the simulation results with the experimental findings, a good agreement was obtained between the simulation and experimental results
- The exit temperature of compressor did not almost exchange. while the temperatures of the condensation and evaporation increased with increasing of milk mass flow rate for raw milk temperature,
- The coefficient of performance of the heat pump and system increased with increasing of milk mass flow rate. The similar increasing was observed for the work of compressor as well as.
- The temperatures of condensation and evaporation decreased with increasing of milk mass flow rate. On the other hand, the decreasing amount was more less according to other temperatures though the exit temperature of compressor decreased.
- With the increasing of entry temperature into condenser, the coefficient of performance of system increased and the work of compressor decreased

Shortly, it may be stated that using the heat pump to pasteurize milk for cheese production is advantageous on the basis of energy consumption. On the other hand, it was found that the model agrees well with the experimental results. From simulation studies, the optimum design of the equipment of heat pump system should be calculated by a system optimization study which considers both the investment cost and system performance according to dairy products.

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