THERMAL ANALYSIS OF TRANSIENT OPERATION OF A SMALL SCALE REFRIGERATION SYSTEM

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ABSTRACT
An unsteady- state model of vapor compression refrigeration system is presented. Thermal and flow processes occurring in a small-scale refrigeration system have been discussed. A theoretical description of the processes in the system components has been proposed in order to develop a theoretical model of the entire system. Components model include the heat exchangers (condenser and evaporator), expansion valve and compressor. For the condenser and evaporator mathematical model, assume two approaches for homogenous refrigerant, the first approach is the lumped model and the second is distributed model.
Experimental investigations are carried out on the laboratory scale system model RCT/EV.
The results obtained from the two models prediction are in a good agreement with the experimental results. The system approaches its final steady-state operating condition in a few minutes.

Keywords: Refrigeration System, Transient, .

الخلاصة
يتناول البحث دراسة التغيرات الفيزياوية التي تحدث لمائع الشغل في كل مكونة من مكونات وحدة تثليج تعمل بدورة تبريد انضغاطية واعتمدت الحالة الغير المستقرة (unsteady) في هذه الدراسة. حيث تمت مناقشة عمليات الجريان وانتقال الحرارة مع وصف نظري للعمليات التي تحدث داخل مكونات المنظومة من اجل تطوير نموذج رياضي للمنظومة وتشمل الدراسة الأجزاء التالية: المبادل الحراري (المكثف والمبخر) وصمام الخنق والضاغط . إن النموذج الرياضي الذي تم عمله إلى المكثف و المبخر اعتمد على المعادلات الفيزياوية لتوازن الكتلة والطاقة في حاله الجريان غير المستقر بالاعتماد على تغير الزمن. أما في للضاغط وصمام الخنق فان العمليات اعتبرت مستقرة وذلك لأنها تحدث بسرعة اكبر من المبادل الحراري .
تمت الدراسة النظرية على أساس الجريان المتجانس في المكثف والمبخر وعلى اعتماد الحالة غير المستقرة . وتم عمل نموذجين منفصلين باستخدام فرضيتين: الفرضية الأولى: هي طريقة السعات المجمعة (Lumped) وتم في هذه الطريقة توقع معدل التغيرات لدرجات الحرارة كتوزيع حقيقي للمسافات والزمن. و الفرضية الثانية هي طريقة التوزيع (Distributed) لتوقع التغير ألموقعي لدرجة الحرارة والمحتوي الحرارى في كل نقطة كدالة للزمن . هذين النموذجين أعطيا وصفا نظريا جيدا للظواهر الفيزيائية التي تحدث للمائع في كل مكونة من مكونات وحدة تثليج.النتائج ألعمليه تم إنجازها بشكل حقيقي على وحدة تثليج نموذجيه مختبريه إيطالية الصنع.
تم إعداد برنامج حاسوبي لكل طريقة لحساب الاستجابة العابرة لوحدة تثليج انضغاطية مبردة بالهواء تعمل بمائع التثليج R134a. النتائج العملية تمت مقارنتها مع النتائج النظرية للطريقتين وأعطت نتيجة جيدة ومتقاربة.

INTRODUCTION

Refrigeration in the engineering sense means the transfer of heat from a lower temperature region to a high temperature one. Devices that produce refrigeration are called refrigerators [Cengel, and Boles]. A refrigerator is designed to cool an enclosed space to a set temperature. Many refrigerators consist of two compartments that are maintained at different temperature. The freezer compartment is usually kept at temperature of about (-15) ºC, fresh food compartment at (5-10) ºC [Klein, and Reindl]. In household refrigerators the temperature control in freeze chamber is carried out by means of a temperature controller performing an on-off control. The peculiar character of the refrigerating unit design (including a compressor, engine unit) and of the geometry of the refrigerating chambers and the temperature filed required in them impose a necessity to control temperature in a special way.
Some properties of the system result from such a way of control, i.e.
-Temperature and pressure are selected, important places of the refrigerating unit are highly variable in time during the on off procedure.

-Heat exchange and flow processes of the medium in the refrigerating unit are unsteady.
It leads to an obvious conclusion that the analysis of energy of such a refrigerating unit, cannot be made if classical methods for refrigeration cycles in steady and quasi-steady states are used. The problems connected with the unit performance from the viewpoint of its design and actual operation, which are difficult to determine quantitatively and often also qualitatively, impose a certain procedure, which, in general, can be presented as a set of two-way activities:
Model tests that consist in a theoretical description of physical phenomena occurring in individual components of the theoretical system (model) on the basis of quantitative law and relations, and experimental investigation carried out on the actual system [Dlugoszewki].

MATHEMATICAL ANALYSIS

A theoretical model of the system must comprise component models, owing to which it will be possible to describe quantitatively, the physical phenomena corresponding to the effects occurring in component of the actual system [Dlugoszewki].
Thus, the theoretical model will consist of the models of a condenser an evaporator and brief description for the expansion valve and the compressor.
The purpose of the present work is to expand the body of heat mechanism solutions to include the cases of spatial and temporal distribution within the component system by two models distribution and lumped with different assumptions. Anew technique of numerical solution (the explicit and implicit methods) used to give more realistic, unsteady-state description of the temperature profile induced by vapor compression refrigeration system.

Heat Exchanger (Condenser , Evaporator).

The condenser and evaporator are wire and tube heat exchanger with air-cooled. The relevant physical laws applicable to the phenomenon of the refrigerant movement within the system are those that govern compressible fluid flow mechanics, and are, in the most general case, the law of the conservation of mass, energy and momentum.

Since both condenser and evaporator are governed by the same physical, a generic heat exchanger model is developed that could be specified to operate as a condenser or as an evaporator at the time of system execution [Oskarssion et. al.].

First Model (Lumped Model).

A simplified model of the condenser (evaporator) is shown in fig.(1). In this model, it is assumed that the homogenous refrigerant, has thermodynamic properties correspond to average parameters from an actual distribution in time and space occurs in the whole inner volume of condenser (evaporator).

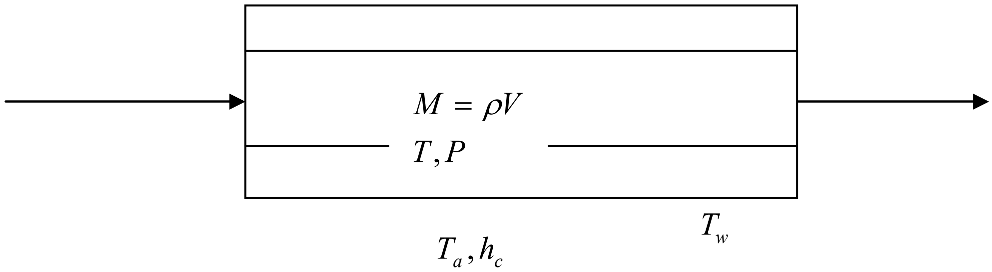

Fig. (1): Model Of The Exchanger-Condenser (Evaporator) [Dlugoszewki]

Model Assumptions

Owing to a complex design structure (from the view point of geometry) and physical processes occurring in a condenser (evaporator), which are difficult to describe mathematically, certain simplifications are assumed [Dlugoszewki].

1- Refrigerant flow inside the tube is one dimensional along the tube axis.
2- Pressure drop in the heat exchanger is negligible.
3- In a cross section of the tube (of the condenser, evaporator), the refrigerant is homogenous (physical properties of the refrigerant are invariable in the cross section).
4- Energy and mass transfer occurs only by convection.

To solve the equations for such a model, the boundary and initial conditions should be introduced; which is a difficult task due to the complicated geometrical structure of the exchanger (the shape, cooling fans of the flow ducts). These conditions are most often a very simplified representation of the actual conditions.

The variation of the refrigerant mass in the inner volume of the heat exchanger (condenser, evaporator) is expressed by the relation:

$$dM = V.d\rho \tag{1}$$

The mass balance in the heat exchanger is written as:

$$\dot{m}_{in} - \dot{m}_{out} = \frac{dM}{dt} = V\frac{d\rho}{dt} \tag{2}$$

The heat balance for the heat exchanger is given by [Dlugoszewki].

$$(\dot{m}_{in} H_{in} - \dot{m}_{out} H_{out})dt = cMdT + c_r M_r d\overline{T} + UA(T_w - T_a) \tag{3}$$

It can be assumed in the first approximation that:

$$T = \overline{T} = T_w$$

$T -$ average temperature of the heat exchanger tube.
Assuming that

$$T - T_a = \theta$$

and T_a constant

$$\dot{m}_{in} H_{in} - \dot{m}_{out} H_{out} = (cM + c_r M_r)\frac{d\theta}{dt} + UA\theta \tag{4}$$

$$\dot{m}_{in} H_{in} - \dot{m}_{out} H_{out} = (cV\rho(T) + c_r M_r)\frac{d\theta}{dt} + UA\theta \tag{5}$$

where:

- T_a - Ambient temperature
- T_w - Wall temperature.
- c - The specific heat of refrigerant at vapor phase.
- M - Mass of the refrigerant at vapor phase.
- c_r - The specific heat of refrigerant at liquid phase.
- M_r - Mass of the refrigerant at liquid phase.

These parameters were represented the phase change happened in the condenser and evaporator.
Assuming that the heat resistance of the boundary layer on the refrigerant side and the heat resistance of the heat exchanger wall are negligible with respect to the heat resistance of the boundary layer on the airside, as:

$$\dot{m}_{in} H_{in} - \dot{m}_{out} H_{out} = (cV\rho(T) + c_r M_r)\frac{d\theta}{dt} + hA\theta \tag{6}$$

$$h = 0.314\frac{k_a}{D_o} Re^{0.68} Pr_{air}^{1/3} (\frac{H_f}{s})^{-0.12} (\frac{T_f}{s})^{-0.12} \tag{7}$$

Next, if we assume that the pressure pulsation of the refrigerant caused by the compressor operation is highly damped on its way towards the condenser, we can determine the mass flow rate at the condenser inlet, which is equal to the mass flow rate at the compressor outlet.
Equation (6) is non-linear, which is caused by the fact that the dependence of the variability of its parameters in the function of temperature and pressure are entangled in this equation (by compressing the refrigerant in the closed system). This variability is strongly influenced by the heat exchange taking place in the condenser. The refrigerant parameters at the condenser outlet depend on the refrigerant parameters in the condenser and the operation condition in the throttling element.
In the case of the evaporator, it is difficult to determine the inlet parameters as the correct determination of the throttling process in the expansion device, which is strongly affected by variable thermodynamic properties of the refrigerant at the expansion device inlet, is very laborious [Dlugoszewki].

Second Model (Distributed Model)
The distributed model allows spatial variations to be monitored and the governing equations for all the nodes are identical [Oskarssion et at] .

Model Assumptions:
Some of the common assumptions are [Bendapudi and Wang]:

1-The flow inside the tube is one dimensional along the tube axis.
2-The two phase region of refrigerant is assumed to be a homogeneous mixture of liquid and vapor.
3-The heat transfer coefficient is uniform on the air side.
4-The variation of refrigerant kinetic and potential energy is ignored.
5-The tube is so thin and the thermal conductivity of copper is so high, the conduction resistance in the radial direction is negligible.
6-Axial heat conduction in the tube wall is negligible.

Conservation Equation
By applying the conservation equations of mass and energy on the refrigerant and tube wall, the following basic equation can be obtained.

The mass conservation is represented by:

$$\frac{d(\rho V)_i}{dt} = \dot{m}_{i-1} - \dot{m}_i \tag{8}$$

The energy conservation is given by:

$$\frac{d(\rho Vh)_i}{dt} = \dot{m}_{i-1} H_{i-1} - \dot{m}_i H_i - q_{r,i} \tag{9}$$

The rate of heat transfer is determined using the rate of equation:

$$q_{r,i} = h_i A_i (T_{r,i} - T_{t,i}) \tag{10}$$

The temperature of the tube is determined from an energy balance across the tube material in that node.

$$M_t c_t \frac{dT_{t,i}}{dt} = q_{r,i} - q_{a,i} \tag{11}$$

The airside heat rate is determined as.

$$q_{a,i} = h_o A_O (T_{t,i} - T_a) \tag{12}$$

EXPERIMENTAL WORK
Theoretical investigations of an actual system have been carried out, from which the relation of: temperature fields in individual components of the refrigerating system, pressure in characteristic places of the system, and active electrical power of the compressor engine as a function of time have been obtained. These results of measurements make it is possible to correlate the model investigations.

The aims of the experimental work are:

1. To test the performance of refrigeration cycle using R134a.

2. To study mainly variables, affect the performance of refrigeration cycle.

3. Experimental results are used to verify the developed model.

Test Rig Layout

The test rig layout is consists of general refrigeration cycle trainer Mod. RCT/EV. The educational panel of this equipment provided with mimic diagram and control board shows an exploded view of all its components allowing an easy study of the thermodynamic cycle of compression refrigeration. This trainer also constitutes the necessary introduction to a complete laboratory for studying and testing refrigeration and air-conditioning equipment [Electtronicaveneta].

Test Procedure

Refrigeration cycle Trainer Mod. RCT/EV was used to perform the measurements requirement for individual components of refrigeration system in order to verify the theoretical models of these components.
Initially, the ambient temperature of the laboratory was measured by digital thermometer which was fixed in the system. Also the initial temperatures of all components reading from the screen of these digital thermometers, which were the same in individual position of the test rig. This temperature in all tested conditions, for the condenser and evaporator wall, and refrigerant in the coil is 26°C. At this temperature the condenser and evaporator are both at the saturation pressure and in equilibrium as indicated by the pressure gauge?
The test is standard including measured of, the temperatures at (14) locations
Compressor inlet. -Condenser inlet
Five nodes at different location in the condenser.
Condenser outlet. - Evaporator inlet
Five nodes at different location in the evaporator.
the discharge and section pressures. The volume flow rate of refrigerant was also measured through calibrated flow meter. These measured values are recorded at 30 seconds step time until steady state is achieved.
To simulate the operation with the dynamic refrigerator model, a preliminary run is made from the initial conditions.
In the laboratory, the refrigerator runs to a steady state condition and then shut off with on/off switch and left switch fan on through the condenser and evaporator to reach an equilibrium state. After that the refrigerator is turned on, and data recorded from different measuring devices.
The above tested conditions are repeated many times to obtain a good accuracy of reading recorded.

RESULTS AND DISCUSSION.

Figure (2) represents the theoretical results of lumped model (in this model it is assumed that the temperature of the refrigerant and the tube wall are equal). The figure shows the temporal distribution of condenser and evaporator average temperature with including the effect of phase change. It has been shown that the condenser average temperature rises very rapidly in the first 30 seconds of the cycle, while the evaporator average temperature decreases. The increase in the condenser average temperature and the decrease in evaporator average temperature are due to increases in the condenser pressure and decreased in evaporator pressure as shown in fig.(3) (the condenser pressure is taken from laboratory data only). The increase in the condenser pressure is due to the decrease in volume available for the vapor as the volume required by the liquid increases. During the sudden drop in evaporator pressure the refrigerant will begin to flash. This vaporization will rapidly cool the evaporator. Then the average temperature approaches its final steady-state operating condition in a few minutes.

In the second model refrigerant temperature distribution is illustrated in figs (4),(5) and (6). The refrigerant enters the condenser in a superheated state. The condensation occurs when the tube wall temperature falls to or below the refrigerant saturation temperature whiles the bulk temperature of the refrigerant remains above the saturation temperature.

Figures (7), (8) and (9) illustrated that the temperature of the tube wall increases to a high value at the beginning of the two-phase region due to a sudden increase in heat transfer coefficient. The tube wall temperature in the two-phase region decreases with distance as the condensation heat transfer coefficient decreases with quality reduction. The sharp changes in tube temperature at the inception of the two-phase flow region are attributed to neglecting the conduction heat in the axial direction of the tube [Lockhart et. al.].

CONCLUSIONS

In particular, the following conclusion can be deduced.

1- The results obtained from the models prediction are in a good agreement with the experimental results.
2- The models developed here provide a powerful tool for predicting both steady and transient performances of refrigerators.
3- The system approaches its final steady-state operating condition at five minutes.

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NOMENCLATURE:

Symbols	Description	Units
A	Cross section area	m^2
D_o	Outside diameter	m
H_f	Fin height from root to tip	m
H	heat transfer coefficient	W/m^2. ºC
h	Specific enthalpy	J/kg
M	Mass	kg
$\dot{m}$	Mass flow rate	kg/S
P	Pressure	bar
Pr	prandtle number	-
Re	Reynolds number	-
S	Distance between fins surface to surface	m
T	Temperature	ºC
U	Overall heat transfer coefficient	W/m^2. ºC
V	Volume	m^3
v	Specific volume	m^3/kg
T_f	Thickness of fin	m

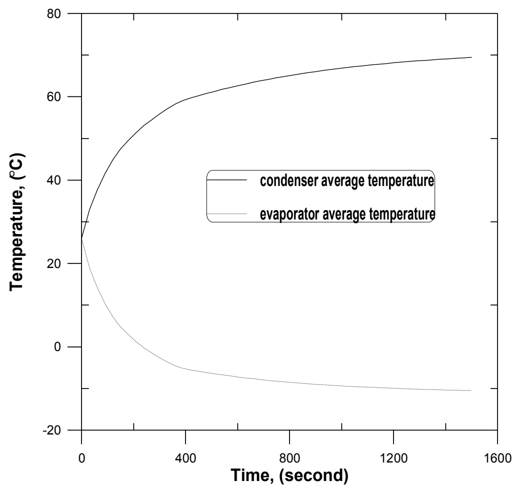

Fig. (2) Theoretical Temperature Distribution of

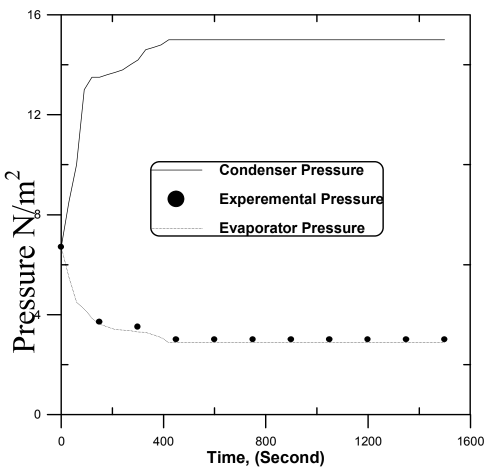

Fig. (3) condenser and evaporator pressure

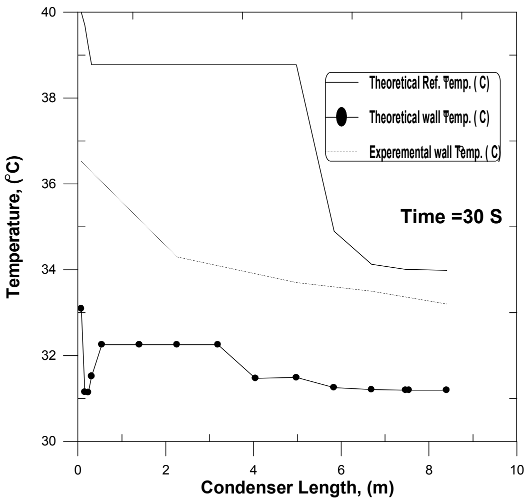

Figure (4) Spatial Temperature Distribution of Refrigerant in the Condenser using different time

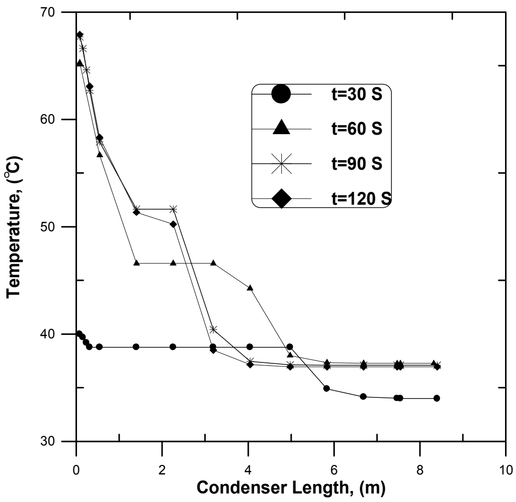

Figure (5) Theoretical Spatial Temperature distribution of Refrigerant in the Condenser using Distributed model at different time

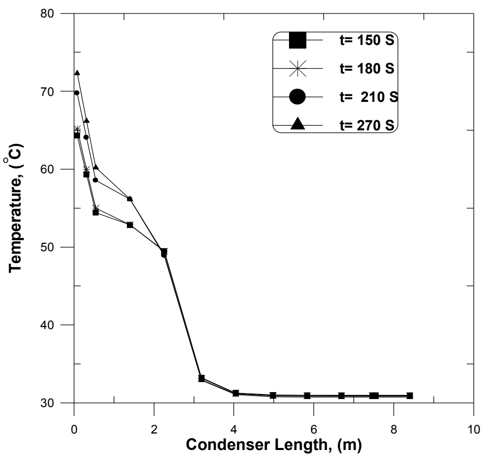

Fig. (6) Theoretical Spatial Temperature Distribution of Refrigerant in the Condenser using Distributed model at different time

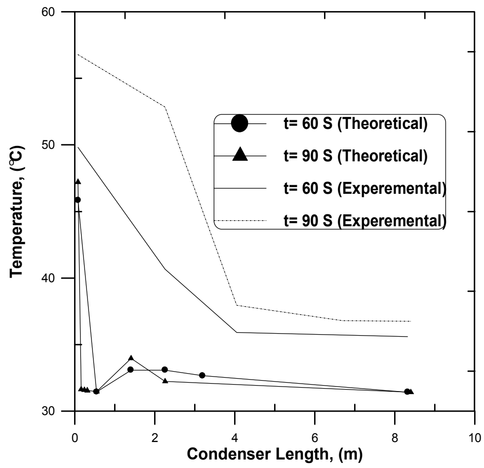

Fig. (7) Theoretical and Experimental Spatial Temperature Distribution of tube Wall in the Condenser using Distributed model at different time

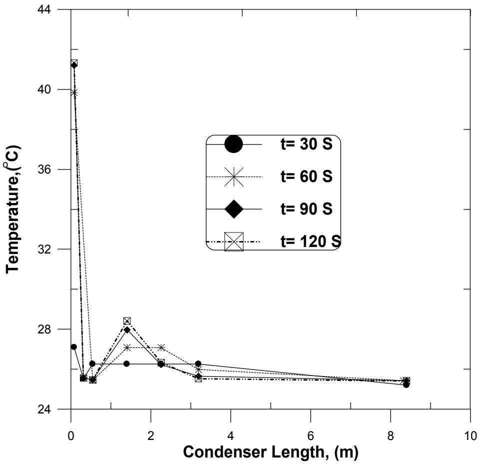

Figure (8) Theoretical Spatial Temperature Distribution of tube Wall in the Condenser using Distributed model at different time

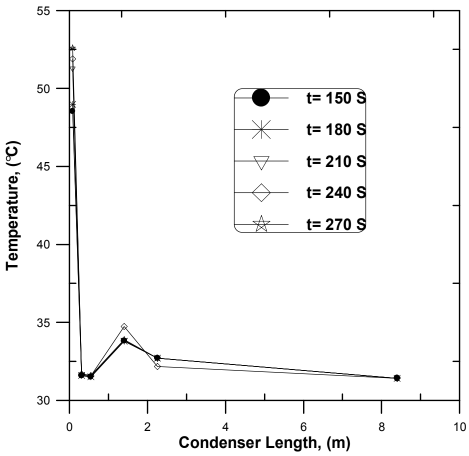

Figure (9) Theoretical Spatial Temperature Distribution of tube Wall in the Condenser using Distributed model at different time