PENGOPTIMUMAN SIRIP PENYEJUK JENIS WAYAR DAN TIUB DALAM KITAR PENYEJUKAN

(OPTIMIZATION OF WIRE AND TUBE CONDENSER IN A REFRIGERATION CYCLE)

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Nomenclature

A	area, m^2
C_p	specific heat capacity at constant pressure, W/kg K
D	diameter, m
H	specific enthalpy of saturated vapor, kJ/kg
h	specific enthalpy of liquid, kJ/kg
hf	specific enthalpy of saturated liquid, kJ/kg
L	length, m
m	mass flow rate, kg/s
N	number, shape function
P	perimeter, m
S	pitch, m
T,t	temperature, K
U	overall heat transfer coefficient, W/m^2K

Subscripts

R	refrigerant
Sat	saturated condition
Sup	superheated condition
t	tube
ti	inside tube
to	outside tube
w	wire
∞	ambient temperature

ABSTRACT

This thesis presents the experimental results of condenser that are commonly used in vapor compression cycle based on domestic refrigerators. The type of condenser that been used in this experiment is wire and tube condenser. A condenser was experimentally tested in a real refrigerator for some operating conditions.

Based on an ideal performance of a wire and tube condenser, this experiment allowed a comparison between the actual result and the simulation one. The most important parameters that been measured is the temperature of the condenser surface at the point selected. This temperature will show whether the condenser operating at the required performance or not.

Some change can be made to the condenser to make it operating at the maximum efficiency. In condenser optimization, the performance efficiency of the condenser can be change by changing the length and diameter of the wire and tube condenser. Pressure in the refrigeration cycle also influences the performance of the refrigerator. These things must be in a perfect set up in order to get a perfect operating refrigerator.

OBJECTIVES

1. Determine the length of tube required for complete condensation.
2. Determine the number of tube required for complete condensation
3. Determine the effect of ambient temperature to the condensation.
4. Determine the effect of mass flow rate of refrigerant to the condensation.

ABSTRAK

Kajian ini menerangkan tentang keputusan eksperimen yg dijalankan terhadap sirip penyejuk yang digunakan pada peti sejuk domestik. Sirip penyejuk yang digunakan di dalam kajian adalah jenis "wire and tube". Sirip penyejuk tersebut diuji melalui satu eksperimen dengan menggunakan sebuah peti sejuk domestik.

Berdasarkan kepada kemampuan ideal sirip penyejuk pada sebuah peti sejuk, eksperimen membolehkan kita membandinkan di antara kecekapan operasi di antara sirip penyejuk ideal dan sebenar. Parameter penting yang diambil kira adalah suhu permukaan sirip penyejuk pada setiap titik-titik yang telah ditetapkan. Bacaan suhu tersebut akan memberitahu tahap operasi sirip penyejuk sama ada ia beroperasi dalam keadaan yang diingini atau tidak.

Berdasarkan ekspermen ini, beberapa perubahan boleh dilakukan kepada sirip penyejuk bagi memastikan ia berfungsi dalam kecekapan yang maksima.

Pengubahsuaian yang biasa dilakukan adalah dengan mengubah panjang dan diameter tiub sirip penyejuk. Tekanan di dalam sistem penyejukan turut mempengaruhi tahap operasi sesebuah peti sejuk. Factor-faktor ini mesti dikawal bagi menghasilkan sebuah peti sejuk yang beroperasi dalam keadaan yang terbaik.

CHAPTER 1: INTRODUCTION

1.1 SCOPE

Condenser is used as a heat exchanger device to reject heat to the surroundings. Type of condenser that been analyzed was wire-and-tube condenser which attached to a home refrigerator. The wire-and-tube condenser is predominantly a natural convection heat exchanger. It consists of a single copper tube and solid steel wires that serve as extended surfaces. The tube, which carries the refrigerant, is bent into a single passage serpentine shape with wires symmetrically spot-welded to both sides in a direction normal to the tubes.

Example of a wire-and-tube condenser shown in Fig 1 below;

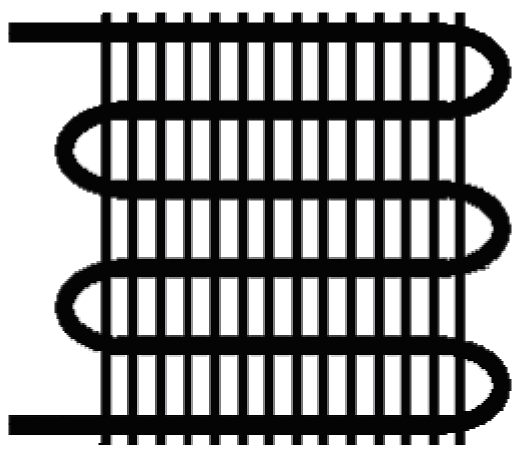

Figure 1: Schematic of a wire-and-tube condenser

By two way of getting data from the condenser; experimental method and simulation method, some comparison can be made between these two data. There may be some difference between these data because data that got from experiment may influence by some factor like environment and instrument condition.

Finite element method also been used during the analysis. Some equations from finite element method can be simplified in C++ program. By using that program some parameter can be varied which is involved in the performance of condenser. Some dimension and refrigerant properties filled into the program to complete the requirement of the program. Two critical regions can be determined using that program; they were phase change region and sub-cooling region. That program can also determine sub-cooling temperature of the refrigerant.

Computer model is a very efficient tool for analyzing the performance of the condenser for different design parameters such as tube diameter, wire diameter, tube spacing, wire spacing, number of wire and tube length. Designing a condenser using this method will develop a good condenser.

Generally modifications that can be made to a refrigerator condenser were by changing the tube length, tube and wire diameter, wire gab, and tube material. These changes will result a huge influence to the refrigerator operation.

1.2 LITERATURE REVIEW

1.2.1 MODELLING AND OPTIMIZATION OF WIRE AND TUBE CONDENSER I

The rate of heat transfer from an element of tube length, Δz can be expressed as

$$Q_{ele} = UA_{ele}.(T_{ref} - T_{\infty})_{ele} \qquad (1)$$

The variable conductance, UA_{ele} applied to each element is expressed as

$$\frac{1}{UA_{ele}} = R_i + R_t + R_o = \left(\frac{1}{h_i A_i} + \frac{\ln\left(r_o/r_i \right)}{2\pi.k.\Delta z} + \frac{1}{h_o A_o} \right)_{ele} = R_{TOT} \qquad (2)$$

It is primarily a function of the inner heat transfer; radial heat conduction (in the tube) and outer heat transfer resistances. The elemental tube length is equal to the pitch of wire, $\Delta z = p_w$ and the elemental outer area of heat transfer, A_o is given as

$$A_o = A_t + A_w = \pi.d_{t,o}.p_w + 2.\pi.d_w.p_t \qquad (3)$$

The computation of the convective heat transfer requires the knowledge of the fin efficiency of the wire, η_w which depends on the temperature distribution along the wire due to conduction and convection. Assuming the heat transfer coefficient to be constant along the elemental segment of the wire, the one-dimensional fin efficiency, η_w is expressed as

$$\eta_w = \frac{\left[\tanh\left(\frac{m p_t}{2}\right) \right]}{\left[\frac{m p_t}{2} \right]} \text{ where } m = \sqrt{\frac{4 h_w}{k_w d_w}} \qquad (4)$$

Initially, the convective heat transfer coefficient of the wire, h_w is guessed in a way that yields the fin efficiency (η_w) of about 0.9 to start the iterations. By definition, the fin efficiency is the ratio of temperature difference between the wire and the ambient, and the tube and ambient as given by:

$$\eta_w = \frac{(T_w - T_\infty)}{(T_{t,o} - T_\infty)} \therefore T_w = \eta_w.(T_{t,o} - T_\infty) + T_\infty \qquad (5)$$

Initially, the outer tube temperature $T_{t,o}$, is assumed to be 0.5 °C lower than the refrigerant temperature. This value is iterated after the elemental heat transfer has been computed. The calculation of the convective and radiative heat transfer requires the evaluation of the mean surface temperature of the heat exchanger, T_{ex}. This can be expressed as a function of the surface temperature of the tubes, $T_{t,o}$ and the mean surface temperature of the wires, T_w as:

$$T_{ex} = \frac{(A_t T_{t,o} + A_w T_w)}{A_e} \qquad (6)$$

Substituting (3) and (5) into (6), T_{ex} can be written as:

$$T_{ex} = \frac{(T_{t,o} + GP.\eta_w.(T_{t,o} - T_\infty) + GP.T_\infty)}{(1 + GP)} \qquad (7)$$

Where *GP*, the geometric parameter is expressed as

$$GP = 2.\left(\frac{p_t}{d_{t,o}}\right).\left(\frac{d_w}{p_w}\right) \qquad (8)$$

To find a more economical solution for an improved condenser design, Bansal and Wich introduced a useful parameter called the *optimization factor*, f_o. This relates the optimum heat exchanger design with the maximum capacity per unit weight. This factor can also be applied to condenser design and is defined as the ratio of the condenser capacity per unit weight of the optimized design ($\dot{Q}_{opt}/w_{opt}$) and the present design ($\frac{\dot{Q}_{100\%}}{w_{100\%}}$) as:

$$f_o = \frac{\dot{Q}_{opt}/w_{opt}}{\dot{Q}_{100\%}/w_{100\%}} \qquad (9)$$

Where $\dot{Q}_{opt}$ and $\dot{Q}_{100\%}$ are, respectively, the condenser capacity of the optimized and the present design, while w_{opt} and $w_{100\%}$ are respectively the condenser weight of the optimized design and the present design.

It is an interesting concept to reduce the metal weight (i.e. cost reduction to the manufacturer) instead of the refrigerant volume. For the optimum condenser design, $f_o>1$ is desired for better capacity per unit weight of the condenser. To optimize the condenser, the parameters including the wire pitch and diameter, and tube pitch and diameter were varied while maintaining the width and height of the condenser at current values. The optimization process was carried out using the refrigerant condensing temperature of 40 °C and mass flux of 110 kg/m^2 s

The condenser capacity increases with increasing wire diameter as well as number of rows due to increased heat transfer area as shown in. However, the optimization factor, f_o decreases with increasing wire diameter and number of rows, indicating a reduction in condenser capacity per unit weight over current condenser design. The largest f_o value is achieved with the minimum wire diameter of 0.007 m and tube pitch of 0.0532 m (22 rows) where the condenser capacity is 150 W (8% less than the current condenser design).

Alternatively, the wire pitch and tube diameter can be altered for optimization purposes. Generally, condenser capacity is maximized with increasing tube diameter and wire density. However, the effect is less than 1% when the wire pitch, *p*wi is reduced beyond 4.3 mm (90 pairs), or the diameter is increased beyond 0.045 m.

Optimization factor (f_o) decreases with increasing tube diameter or wire density, indicating that the condenser capacity gain is relatively small compared with the corresponding increase in condenser weight. As the wire pitch p_{wi} approaches zero, the condenser behaves much like a double array of vertical wires. In this condition, the thermal behavior of the external side is similar to that of a constant-temperature flat plate. The gain in fins area is offset by the fact that heat transfer from the internal side is completely inhibited. The resulting Nusselt number is given by heat transfer correlation for vertical plate. For laminar flow, the average Nusselt Number applied to each surface can be approximated as:

$$\overline{Nu} = 0.68 + 0.67\left[Ra.\left[1+\left(\frac{0.492}{Pr}\right)^{9/16}\right]^{-16/9}\right]^{1/4} \tag{10}$$

On the other hand, as p_{wi} becomes very large, the geometry becomes a vertical array of horizontal tubes where the Nusselt number is given by as:

$$Nu_H = 0.66 \times Ra_{dt,o}^{0.25} \tag{11}$$

These are the two limiting cases for the wire-and-tube geometry, and the corresponding equations (10) and (11) provide the upper and lower limits for the Nusselt number.

1.2.2 MODELLING AND OPTIMIZATION OF WIRE-AND-TUBE CONDENSER II

Abstract

This paper presents the modeling and experimental results of wire-and-tube condensers that are commonly used in vapor compression cycle based domestic refrigerators. A condenser was experimentally tested in a real refrigerator for some operating conditions. A simulation model was developed using the finite element and variable conductance approach, along with a combination of thermodynamic correlations. The condenser capacity per unit weight was optimized using a variety of wire and tube pitches and diameters. An *optimizations factor*, f_o was defined as ratio of the condenser capacity per unit weight of the optimized design and the present design. The application of this factor led to an improved design with 3% gain in capacity and 6% reduced condenser weight

Introduction

Currently, refrigeration research focuses on improving energy efficiency, reducing manufacturing cost and introducing innovative designs of heat exchangers (compact, functional, user-friendly). A well-designed condenser will not only improve the energy efficiency, but will also reduce the space and material for a specific cooling capacity. One of the commonly used condensers in domestic refrigerators is the wire-and-tube condenser.

The wire-and-tube condenser (denoted as W&T hereon) is predominantly a natural convection heat exchanger. It consists of a single copper tube and solid steel wires that serve as extended surfaces. The tube, which carries the refrigerant, is bent into a single-passage serpentine shape with wires symmetrically spot-welded to both sides in a direction normal to the tubes.

It is envisaged that a computer model would act as a convenience tool for analyzing the performance of the condenser for different design parameters such as tube diameter and mass flow rate, and reducing the cost of testing and prototype manufacturing of new or modified condensers. These models were based on refrigerant flow inside the tube and cross-flow or counter-flow forced convective heat transfer on the outside.

However, the W&T condenser is a natural draft type heat exchanger with relatively different tube geometry and flow direction, and hence, a distinct model needs to be developed for its performance assessment. The aim of this paper is to present the development and application of such a model for design and optimizations purposes of the W&T condensers in domestic refrigerators. The model was developed using the finite element and variable conductance approach, along with a combination of thermodynamic correlations. It was written in FORTRAN 90 programming language.

Modeling of the condensers

The rate of heat transfer from an element of tube length, Δz can be expressed as

$$Q_{ele} = UA_{ele} \cdot (T_{ref} - T_{ca})_{ele} \qquad (1)$$

The variable conductance, UA_{ele} applied to each element is expressed as

$$\frac{1}{UA_{ele}} = R_i + R_t + R_o = \left(\frac{1}{h_i A_i} + \frac{\ln(r_o/r_i)}{2\pi k . \Delta z} + \frac{1}{h_o A_o} \right)_{ele} = R_{WKT} \qquad (2)$$

The elemental outer area of heat transfer, A_o is given as

$$A_o = A_t + A_w = \pi . d_{t,o} . p_w + 2 . \pi . d_w . p_t \qquad (3)$$

The one-dimensional fin efficiency, η_w is expressed as

$$\eta_w = \frac{\left[\tanh\left(\frac{m p_t}{2}\right)\right]}{\left[\frac{m p_t}{2}\right]} \text{ where } m = \sqrt{\frac{4 h_w}{k_w d_w}} \qquad (4)$$

The fin efficiency is the ratio of temperature difference between the wire and the ambient, and the tube and ambient as given by:

$$\eta_w = \frac{(T_w - T_\infty)}{(T_{t,o} - T_\infty)} \text{ or } T_w = \eta_w . (T_{t,o} - T_\infty) + T_\infty \qquad (5)$$

In the mean surface temperature of the wires, T_w as:

$$T_{ex} = \frac{(A_t T_{t,o} + A_w T_w)}{A_o} \qquad (6)$$

Substituting (3) and (5) into (6), T_{ex} can be written as:

$$T_{ex} = \frac{(T_{t,o} + GP . \eta_w . (T_{t,o} - T_\infty) + GP . T_\infty)}{(1 + GP)} \qquad (7)$$

Where *GP*, the geometric parameter is expressed as

$$GP = 2 . \left(\frac{p_t}{d_{t,o}}\right) . \left(\frac{d_w}{p_w}\right) \qquad (8)$$

Experimental validation of the results

To validate the model, experiments were conducted on a wire-and-tube condenser using a real refrigerator. The condenser was tested on a two-door vapor compression cycle based domestic refrigerator-freezer (model E406B). The capacity of the Provision Compartment (PC) and the Freezer Compartment (FC) was respectively 271 and 133 l. The fridge had a reciprocating compressor (model Embraco FGS90HAW), a defrost heater of 350 W, an egg-crate type evaporator, a non-adiabatic capillary tube along with a wire-and-tube condenser.

Closed-door experiments were carried out. Measurements of refrigerant temperature and pressure were taken during the experiments. The modeling results were compared with the experimental results for the specified ambient conditions where the refrigerant was fully condensed at the condenser outlet. A fair agreement can be realized when comparing the simulation with test results.

As the state of the refrigerant is determined by the measured temperatures and pressures, a minor error in the measurement may lead to the refrigerant condition being taken as fully condensed, while in fact, the refrigerant may still be in the two-phase state. The capacity for these two conditions could differ considerably. This condition may specifically be relevant to situations where the degree of sub-cooling is less than 0.5 °C.

CHAPTER 2: THEORY AND EXPERIMENTS SET UP

2.1 GENERAL

The fundamental reason for having a refrigerator is to keep food cold. Cold temperatures help food stay fresh longer. The basic idea behind refrigeration is to slow down the activity of bacteria (which all food contains) so that it takes longer for the bacteria to spoil the food.

For example, bacteria will spoil milk in two or three hours if the milk is left out on the kitchen counter at room temperature. However, by reducing the temperature of the milk, it will stay fresh for a week or two. The cold temperature inside the refrigerator decreases the activity of the bacteria that much.

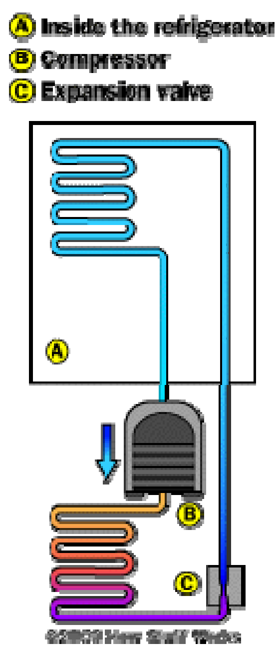

Figure 2: Illustration of refrigerator system[6]

Refer to Fig 2, the basic mechanisms of a refrigerator are;

1. The compressor compresses the refrigerant gas. This raises the refrigerant's pressure and temperature (orange), so the heat-exchanging coils outside the refrigerator allow the refrigerant to dissipate the heat of pressurization.
2. As it cools, the refrigerant condenses into liquid form (purple) and flows through the expansion valve.
3. When it flows through the expansion valve, the liquid refrigerant is allowed to move from a high-pressure zone to a low-pressure zone, so it expands and evaporates (light blue). In evaporating, it absorbs heat, making it cold.
4. The coils inside the refrigerator allow the refrigerant to absorb heat, making the inside of the refrigerator cold. The cycle then repeat.

2.2 PARTS OF A REFRIGERATOR

(a) Compressor.

Compressor is the motor (or engine) of the cooling system. Obviously it placed at the bottom of a refrigerator in the back. It's almost always black and about the size of a football. If the refrigerator is self-defrosting, the compressor may be behind a thin panel. The compressor runs whenever the refrigerator thermostat calls for cooling (and the defrost timer is not in a defrost cycle, for self-defrosting units). It is normally very quiet. When running, it is compressing a refrigerant that is in a low-pressure gaseous state to a high-pressure gas.

(b) Condenser

The condenser is a series of tubes with fins attached to together similar to a radiator. It's always somewhere on the outside of the refrigerator. The main function for a condenser is to release heat that absorbs from the inside of a refrigerator. It may be:

- A large black grid mounted to the back of the refrigerator
- Folded and placed under the refrigerator
- Coiled up and placed near the compressor
- Integrated in the liner of the refrigerator

The high-pressure refrigerant gas coming from the compressor flows through the condenser and becomes a liquid. As this occurs the refrigerant release heat. The heat is conducted away from the tubes by the fins. The famous type of condenser that usually used for a domestic refrigerator is wire and tube condenser.

(c) Metering Device (Capillary Tube)

The metering device in most household refrigerators is a capillary tube, a tiny copper tube. The capillary tube is attached from the end of the condenser to the beginning of the evaporator. The capillary tube controls the pressure and flow of the refrigerant as it enters the evaporator. Once the liquid refrigerant has travelled the length of the condenser, it is forced through the capillary tube.

(d) Evaporator

The evaporator is always located on the inside of the refrigerator, usually inside the freezer compartment. When the liquid refrigerant comes out of the small capillary tube,

it's injected into the larger tubes of the evaporator causing a pressure drop. This pressure drop allows the refrigerant to expand back into a gaseous state. This change of state from liquid to gas absorbs heat. The gaseous refrigerant travels through the evaporator tubes, back out of the refrigerator and down to the compressor to begin the circulation process again.

(e) Temperature controller

All refrigerators have a thermostat to maintain the proper temperature. These are usually very simple devices. When the refrigerator reaches the set temperature, the thermostat interrupts the electricity flow to the compressor, which stops compressor.

2.3 REFRIGERATION CYCLE

2.31GENERAL

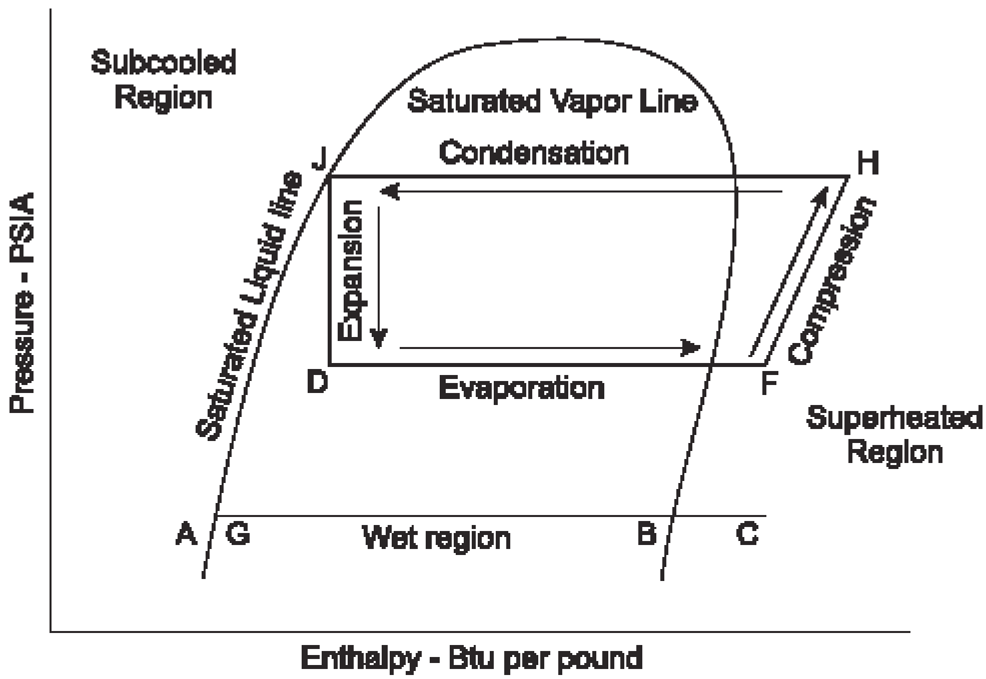

Figure 3: P-H chart for general refrigeration cycle

The P-H chart is an important tool in understanding the property changes that take place during each phase of the cycle to give graphical view in the cycle. Referring to Fig 3, horizontal lines on the P-H Chart are lines of constant pressure and vertical lines are lines of constant enthalpy or heat energy. The line labeled “Saturated liquid line” and “Saturated Vapor Line” is plots of the pressure versus enthalpy for the saturated state of a refrigerant.

The chart is divided into three regions. The area to the left is the sub-cooled region, to the right is the superheated region and in the middle is the wet region or mixture state. The constant temperature lines are horizontal in the mixture region indicating that phase change occurs at constant pressure. Likewise, expansion of the gas takes place at constant enthalpy. Following the chart, if refrigerant liquid at point A absorbs heat at constant pressure, it will begin to boil. Evaporation takes place with no change in temperature.

As heat is added, the enthalpy increases and it enters a mixture state of vapor and liquid. At point B, the mixture becomes a saturated vapor. Any additional heat applied at constant pressure causes the refrigerant to enter the superheat region indicated by point C. In evaporation, the refrigerant enters the evaporator as a mixture of vapor and liquid at point D of the chart. It enters the evaporator by being metered through a thermostatic expansion valve (TXV), which lowers its pressure and therefore its temperature.

Because the refrigerant is at a temperature below the process fluid, it absorbs heat from the process fluid, and boils, and changed phase from a liquid to a gas. In order for the refrigerant to change state, it must take in heat energy. During this transfer of heat energy, only latent heat is absorbed resulting in the refrigerant remaining at a constant temperature.

In theory, it leaves the evaporator as a vapor at point E, however in application additional heat called “superheat” is added to prevent liquid condensation in the lines that can damage the compressor. After absorbing the latent heat during evaporation and superheating, the refrigerant gas is compressed from a low-pressure gas to a high-pressure gas. During the compression process, the refrigerant gas absorbs additional heat known as the Heat of Compression, which is merely the friction of molecules being rapidly forced into a confined space.

The additional heat energy caused by compression is represented by the line between points F and H. Note that point H is to the right of point F, indicating the additional enthalpy resulting from the Heat of Compression. The hot and high-pressure gas is passed through a condenser to remove the heat of compression plus the latent heat of evaporation, collectively known as the Total Heat of Rejection (THR). This heat is typically rejected to a water source in the case of a water-cooled chiller package, or to ambient air in an air-cooled condenser package.

From the P-H chart, it can be seen that condensation takes place at constant pressure. The heat transfer is represented by the difference in enthalpy between points H and J. At point J the refrigerant is totally condensed into a liquid and remains at constant pressure.

2.32 SUPERHEATED

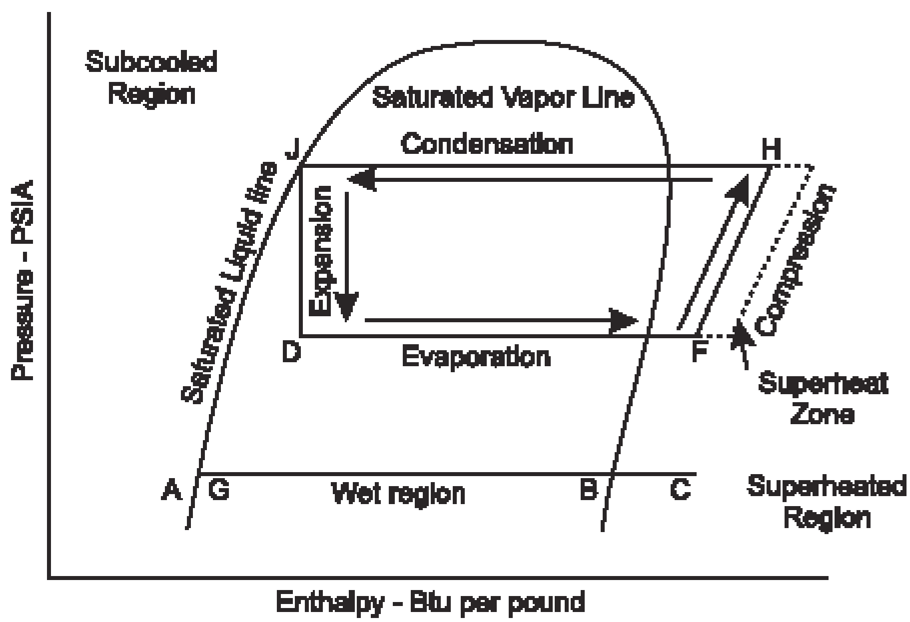

Figure 4: P-H chart for superheated refrigeration cycle

Superheat is the heat added to the vapor beyond what is required to vaporize all of the liquid. Superheat therefore is not latent, but sensible heat and is measured in degrees. From Fig 4, it can be seen that superheat from the evaporation phase has a corresponding increase in the total heat of rejection at the condenser and results in the compressor operating at higher temperature.

While some amount of superheat is required to protect the refrigeration system and prevent liquid entering the compressor, too much superheat can contribute to oil breakdown and increased system downtime.

2.33 SUB COOLING

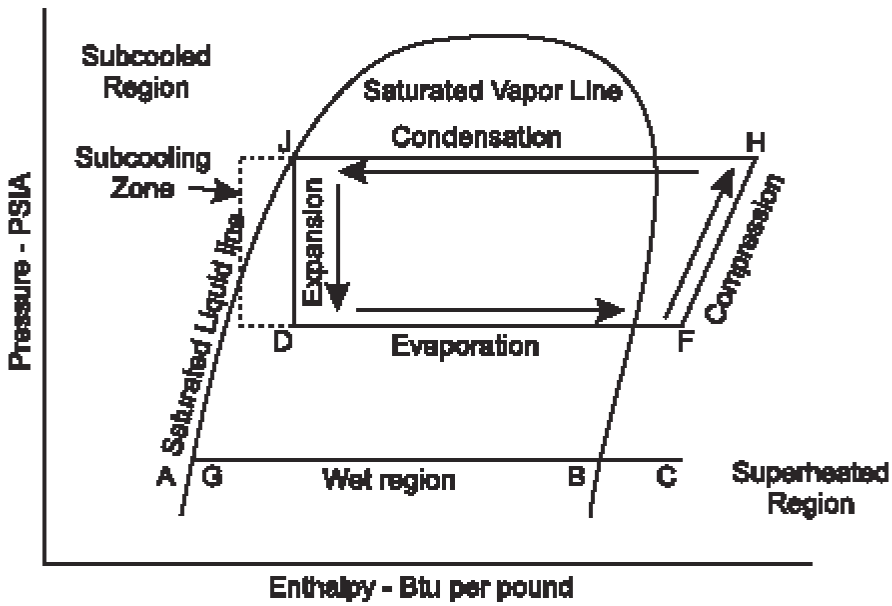

Figure 5: P-H chart for general refrigeration cycle

Sub cooling is the process of cooling condensed gas beyond what is required for the condensation process. Sub cooling is sensible heat and is measured in degrees. Sub cooling can have a dramatic effect in the capacity of a refrigeration system by increasing the capacity of the refrigerant to absorb heat during the evaporation phase for the same compressor kW input.

Sub cooling assures that no gas is left at the end of the condensing phase, thus assuring maximum capacity at the TXV. Sub cooling is best accomplished in a separate sub cooler or a special sub cooling section of a condenser because tube surface must be submerged in liquid refrigerant for sub cooling to occur.

Refrigerant is forced across the condensing section and then downward through the submerged tube section. It is important to pipe the condenser with the coldest water inlet countercurrent to the sub cooling section. As the high-pressure cooled liquid from the condenser is reduced in pressure at the TXV, its corresponding temperature was reduced and the cycle is complete. It can be seen from the P-H chart that the heat load on the condenser is greater than that of the evaporator, or process load.

2.4 MODELING OF THE CONDENSER

Governing Equation.[1]

For computer modeling purpose, three equations have been used for three regions; de-superheating, phase change and sub-cooling regions.

De-superheating region

The heat balance for a small element can be written as,

$$m_r C_{pr} \frac{dT}{dA} = -u(T - T_\alpha) \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (1)$$

The differential of the equation over an elemental area dA in terms of enthalpy H is

$$\frac{dH}{dA} + \frac{U}{m_r}\left[\frac{H}{C_{pr}} - T_\alpha\right] = 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (2)$$

The enthalpy for the refrigerant in the element is assumed to vary linearly as,

$$H = N_1 H_1 + N_2 H_2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (3)$$

Where, the shape functions are given by

$$N_1 = 1 - \frac{A}{\Delta A} \quad \text{and} \quad N_2 = \frac{A}{\Delta A} \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

Applying the approximation to governing equation (2) two equations for the two unknowns H_1 and H_2 are obtained as follows:

$$\int N_1 \left[\frac{dH}{dA} + \frac{U}{m_r}\left(\frac{H}{C_{pr}} - T_\alpha\right)\right] dA = 0 \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

$$\int N_2 \left[\frac{dH}{dA} + \frac{U}{mr}\left(\frac{H}{C_{pr}} - T_\alpha\right)\right] dA = 0 \ldots\ldots\ldots\ldots\ldots\ldots (6)$$

After solving the above equations we got the following element matrix,

$$\begin{bmatrix} 2C-0.5 & C+0.5 \\ C-0.5 & 2C+0.5 \end{bmatrix} \begin{Bmatrix} H_1 \\ H_2 \end{Bmatrix} = \begin{Bmatrix} U.\Delta A.T_\alpha/(2m_r) \\ U.\Delta A.T_\alpha/(2m_r) \end{Bmatrix} \ldots\ldots(7)$$

where, $C = \dfrac{U.\Delta A}{6m_r.C_{pr}}$

The above formulation is valid till the enthalpy of the refrigerant equals to that corresponding to dry saturated condition.

Phase change region

For this region, the differential equation governing heat transfer is written as;

$$\frac{dH}{dA} + \frac{U}{m_r}(T_{sat} - T_\alpha) = 0 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(8)$$

By assuming linear variation in enthalpy

$$H=N_1H_1+N_2H_2 \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(9)$$

And following the same procedure as that in earlier region the following element matrix is found:

$$\begin{bmatrix} 1 & -1 \\ 1 & -1 \end{bmatrix} \begin{Bmatrix} H_1 \\ H_2 \end{Bmatrix} = \begin{Bmatrix} U.\Delta A(T_{sat} - T_\alpha)/m_r \\ U.\Delta A(T_{sat} - T_\alpha)/m_r \end{Bmatrix} \ldots\ldots\ldots\ldots..(10)$$

The above formulation will be valid till the enthalpy equals to that of saturated liquid enthalpy at the operating condenser pressure.

Sub-cooling region

For this region, the differential equation governing heat transfer is written as;

$$m_r C_{pr} \frac{dT}{dx} + U.P(T - T_\alpha) = 0 \quad \text{………………………(11)}$$

Assuming linear variation for the temperature of the refrigerant in the tube,

$T=N_1T_1+N_2T_2$ ………………………………………(12)

Applying the same procedure, the following element matrix is found:

$$\begin{bmatrix} 2C - 0.5 & C + 0.5 \\ C - 05 & 2C + 0.5 \end{bmatrix} \begin{Bmatrix} T_1 \\ T_2 \end{Bmatrix} = \begin{Bmatrix} 3.C.T_\alpha \\ 3.C.T_\alpha \end{Bmatrix} \quad \text{……………...(13)}$$

Where, $C= U.P.\Delta l/(6m_r.C_{pr}) =U.\Delta A/(6m_r.C_{pr})$

Variable	
Tube diameter (mm)	5.0
Wire diameter (mm)	1.7
Tube gap (mm)	52.4
Wire gap (mm)	5.0
Wire portion length (mm)	325.0
Straight bare portion (mm)	30.0
Tube length (straight portion) (mm)	385.0
Curved bare portion (mm)	48.08
Number of wire	104
Tubing Material	**Stainless steel coated with copper**

Table 1: Geometrical data of the wire and tube condenser.

2.4 EXPERIMENTAL SETUP

Figure 6: Experimental apparatus.

This project presents the experimental results of condenser that are commonly used in vapor compression cycle based on domestic refrigerators. A condenser was experimentally tested in a real refrigerator for some operating conditions. The type of condenser that been used in this experiment is wire-and-tube condenser.