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# Determination of Refrigerant Path Number for Fin-tube Condenser Considering Heat Transfer Performance and Pumping Power

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## ABSTRACT

Fin-tube heat exchangers are widely used in air-conditioners and heat pumps, which are constructed with a lot of tubes. Refrigerant circuit of heat exchanger with numerous pipe can be constructed by many methods. Refrigerant circuit design is usually determined designer's experience and case by case test without guides. The number of path affects largely on heat exchanger performance. In this paper, design methodology for optimum number of path is suggested by relating convective thermal resistance and pumping power. Suggested methodology is described through an example and verified by various refrigerant circuit simulation results.

## 1. INTRODUCTION

The fin-tube heat exchanger widely used for an air-conditioner or a heat pump is comprised of a number of heat transfer tubes, on the outside of which fins are installed. It is difficult to design a heat exchanger comprised of a number of heat transfer tubes as there are innumerable combinations that can compose a refrigerant circuitry. In general, as design of a refrigerant circuitry is conducted relying on the experience of the designer and a case by case test, it requires much time and expense.

The optimization of fin-tube heat exchanger's refrigerant circuitry was attempted in various methods. Wang et al. (1999) experimentally studied the effect of the refrigerant circuitry on the performance in a condenser which uses air as the heat source. They concluded that, though the performance increases when the refrigerant circuitry is counter-cross arranged, the increase in the performance may be offset by the thermal conduction phenomenon through the fins. There was a study which applied the genetic algorithm to optimization of a refrigerant circuitry (Wu et al., 2008). This method has a disadvantage that the optimization process should be carried out again when the operational condition or the shape of the heat exchanger is changed. Also, as this algorithm looks for the optimum refrigerant circuitry among the refrigerant circuitries constructed in advance, it not only takes much time but also is difficult to review whether the circuitry can be actually produced or not. Liang et al. (2000) proposed a model which can be used to study the heat transfer characteristics and performance evaluation of a refrigerant circuitry through the exergy destruction analysis. Though the exergy destruction analysis has been utilized in determining which refrigerant circuitry is good for a certain operational condition, it can not give the answer as to how a refrigerant circuitry should be designed. A model that can be utilized for minimization of the refrigerant-side exergy destruction of a condenser was presented using the entropy generation minimization theory (Ye and Lee, 2012). And the method to find the optimum number of paths and the mergence point using the proposed model was presented. Though this method is useful, the calculation process is complicated and the refrigerant circuitry cannot be intuitively designed.

Studies on heat exchangers have long been conducted. But, the experiment data of the studies related to refrigerant circuitry is still insufficient. Also, it is not yet clear how to achieve optimization. In particular, there is a serious shortage of the studies which consider divergence/mergence of refrigerant circuitry. In this study, a methodology is proposed where the optimum number of paths is determined by an intuitive method which has a physical meaning considering the pumping power and the heat transfer performance of the heat exchanger. And, methodology is also proposed which can be used to review where to diverge/merge the paths and into how many paths. The proposed methods are verified through analysis of various refrigerant circuitries.

## 2. METHODOLOGY TO DETERMINE THE NUMBER OF PATHS

The heat transfer capacity of a heat exchanger is determined by the overall heat transfer coefficient, heat transfer area, and the temperature difference between the two working fluids, which can be expressed as Equation (1):

$$\dot{Q} = UA \cdot (T_r - T_a)_{LM} \tag{1}$$

Here, $(T_r - T_a)_{LM}$ represents the log mean temperature difference between the refrigerant and the air. If the heat exchanger matrix is fixed as the size and number of the heat transfer tubes and fins of the fin-tube heat exchanger are determined, the total heat transfer area does not change even when the refrigerant circuitry is changed. But, if the refrigerant circuitry is changed, the overall heat transfer coefficient and the temperature difference between the two working fluids change. The mean temperature difference between the refrigerant and the air and the local distribution of the temperature difference can be a little changed by changing the refrigerant circuitry.

Meanwhile, the overall thermal resistance is comprised of the sum of convective, conductive, contact and fouling thermal resistances, and the convective thermal resistance accounts for most of the overall thermal resistance. As the number of these paths changes, the mass flow rate of the refrigerant flowing inside the tube and the length of each flow path change. As the change in the mass flux varies the Reynolds number, which has an effect not only on the convective thermal resistance but also on the pressure loss of the refrigerant, the effect of the number of paths on the performance of the heat exchanger is very large. Accordingly, the number of paths should be selected first before designing a refrigerant circuitry. In this chapter, we intend to suggest a methodology to optimize the number of paths for a refrigerant circuitry with no divergence/mergence.

The overall thermal resistance in Equation (1) can be expressed as Equation (2). The terms of the right side represent the thermal resistances related to convection inside the tube, conduction of the tube wall, convection outside the tube, contact, and fouling, respectively.

$$\frac{1}{UA} = \frac{1}{\eta_i h_r A_i} + \frac{\ln(D_o / D_i)}{2\pi \cdot L \cdot k} + \frac{1}{\eta_o h_a A_o} + R_{cont} + R_{foul} \tag{2}$$

As the conductive, contact and fouling thermal resistances can be thought to be constant at a certain point in time, the above equation can be expressed as the following equation:

$$\frac{1}{UA} = \frac{1}{\eta_i h_r A_i} + \frac{1}{\eta_o h_a A_o} + C \tag{3}$$

As convective thermal resistances account for most of the overall thermal resistance, the Eq. (3) is suitable for review of the heat transfer phenomenon. $\eta_i h_r A_i$ and $\eta_o h_a A_o$ represent the heat transfer performances of the refrigerant side and the air side, respectively. If the heat exchanger matrix is fixed, the heat transfer performance of the air side depends on the air velocity and is not affected by the refrigerant circuitry construction. In order to examine the effects of the change in the number of paths on the heat transfer performance of the refrigerant side and the pumping power, the pumping power and the heat transfer coefficient were calculated changing the mass flux in a heat transfer tube, of which the result is shown in Figure 1. The condensation heat transfer coefficient on the internal wall of the heat transfer tube and the pumping power per unit length of the heat transfer tube are calculated when the quality of the refrigerant flowing inside the heat transfer tube is fixed. The equations used for the calculation are shown in Table 1, and the conditions of the calculation are shown in Table 2. Figure 1 represents that, though the relation between the heat transfer coefficient and the pumping power is proportional, it is not a linear relation. The

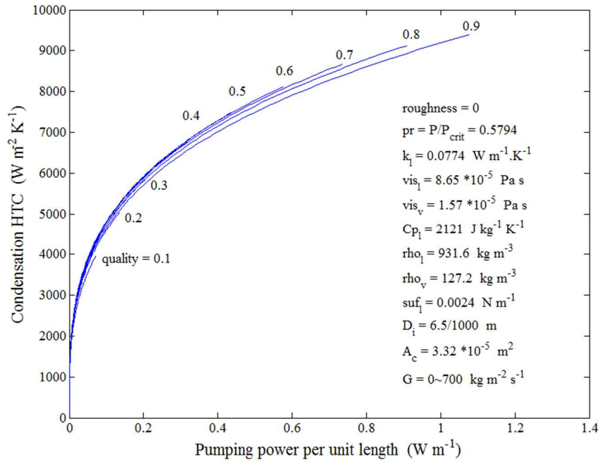


**Figure 1:** Relation of HTC(ref) and pumping

**Table 1:** Correlation to investigate HTC and pumping power

| List | model |
|---|---|
| Condensation heat transfer coefficient | Shah(1979) |
| Pumping power | $\dot{V}\Delta P_{f,tp} = \emptyset_{lo}{}^2 \dfrac{f_{lo}\, G_r{}^2}{\rho_l} L \left(\dfrac{G_r}{\rho_{tp}}\right) A_c$ |
| Two-phase frictional multiplier | Friddel(1992) |
| Single-phase friction factor | Churchill(1983) |
| Two-phase density | $\rho_{tp} = \left(\dfrac{x}{\rho_v} + \dfrac{1-x}{\rho_l}\right)^{-1}$ |

pumping power increases as the mass flux increases, and the increase rate of the heat transfer coefficient reduces as the pumping power increases.

In order to enhance the heat transfer performance of a fin-tube heat exchanger, additional cost or work is required such as widening of the area by installing more fins or provision of more flow work by increasing the rpm of the fan or the compressor. If the work (cost) which can increase $\eta_i h_r A_i$ or $\eta_o h_a A_o$ is the same and the total cost which can be invested in $\eta_i h_r A_i$ or $\eta_o h_a A_o$ is limited, the relation can be expressed in the following equation:

$$\eta_i h_r A_i + \eta_o h_a A_o = constant \tag{4}$$

In the condition of Equation (4), the overall heat transfer coefficient ($UA$) of Equation (3) is the biggest when $\eta_i h_r A_i = \eta_o h_a A_o$. That is to say, in the aspect of the number of paths, when the number of paths is selected to make $\eta_i h_r A_i$ and $\eta_o h_a A_o$ equal, the maximized heat transfer performance can be achieved with small work, and this number of paths can be said to be optimum.

Let's consider the case wherein the shape and the number of the fins and the heat transfer tubes are same and only the number of the refrigerant paths is changed by connecting the heat transfer tubes. If the heat transfer tubes are connected to make the number of paths smaller than the optimum number of paths determined to make the $\eta_i h_r A_i = \eta_o h_a A_o$ state as above, the heat transfer performance of the refrigerant side ($\eta_i h_r A_i$) in each path will increase and results in an increase in the overall heat transfer performance ($UA$). However, as the heat transfer performance of the air side ($\eta_o h_a A_o$) is fixed and $\eta_i h_r A_i > \eta_o h_a A_o$, the heat transfer performance of the air side dominates the overall heat transfer performance. Accordingly, the contribution of the improvement in the heat transfer performance of the refrigerant side to the overall heat transfer performance is reduced. Nevertheless, as the contribution to the pumping power of the refrigerant ($= \Delta P \cdot \dot{V}$) increases in proportion to the volume flow rate ($\dot{V}$), the increase in the cost (pumping power) grows bigger than the increase in the benefit (heat transfer performance).

## 3. OPTIMUM PATH NUMBER OF FIN-TUBE CONDENSER

The optimum number of paths was determined by applying the methodology explained in the previous chapter to a fin-tube heat exchanger of which tube configuration is 22 steps and 2 rows. The object was a fin-tube heat exchanger with a rated capacity of 3.5 kW widely used for an air-conditioner or a heat pump. The detailed specification of this heat exchanger

**Table 2:** Condition to investigate HTC and pumping power

| List | Value | Unit |
|---|---|---|
| Refrigerant | R410A | |
| Pressure | 2840 | kPa |
| Inner diameter | 6.5 | mm |
| Roughness | 0 | mm |
| Mass flux | 0~700 | kg $m^{-2}$ $s^{-1}$ |

is shown in Table 3. As to the operation condition, the median value in the operation range required to satisfy the load was selected, which is put in order and shown in Table 4.

When it is assumed that there is no fin of the refrigerant side and the overall surface efficiency of the air side is 1 in Equation (2), the relation can be simplified into Equation (5) if the conductive, contact and fouling thermal resistances are disregarded.

$$\frac{1}{UA} = \frac{1}{h_r A_i} + \frac{1}{h_a A_o} \tag{5}$$

The total area inside the tubes of this heat exchanger is 0.5867 $m^2$ and the total external area of the tubes including the fins is 9.8655 $m^2$. The heat transfer coefficient of the refrigerant side is calculated by taking the mean value over the tube length in order to find the value which can represent the whole heat exchanger. In this study, the condensation heat transfer model of Shah (1979) was used. The model of Shah (1979) is as shown in Equation (6). Assuming negligible change in transport properties of liquid phase and pressure along the length of condenser, the mean heat transfer coefficient over the whole length of the heat transfer tube is obtained as shown in Equation (7).

$$h_r = h_l\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{x^{0.38}}\right] \tag{6}$$

$$h_l = 0.023\,\mathrm{Re}_l^{\,0.8}\,\mathrm{Pr}_l^{\,0.4}\,k_l / D \tag{6a}$$

$$\mathrm{Re}_l = \frac{G_r D_i}{\mu_l} \tag{6b}$$

$$\mathrm{Pr}_l = \frac{c_p \mu_l}{k_l} \tag{6c}$$

$$\bar{h}_r = \frac{h_l}{(L_2 - L_1)}\int_{L_1}^{L_2}\left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{pr^{0.38}}\right]dL \tag{7}$$

Assuming that the tube length and the vapor quality are in a linear relation, Shah (1979) proposed the equation of the mean heat transfer coefficient as follows:

$$\bar{h}_r = \frac{h_l}{(x_2 - x_1)}\left[-\frac{(1-x)^{1.8}}{1.8} + \frac{3.8}{pr^{0.38}}\left(\frac{x^{1.76}}{1.76} - \frac{0.04x^{2.76}}{2.76}\right)\right]_{x_1}^{x_2} \tag{8}$$

The main area of the condenser where phase change takes place is when the quality is between 0 and 1. Substituting $x_1$=1 and $x_2$=0 in equation (8) yields:

**Table 4:** Specification of fin-tube HX

| List | Value |
|---|---|
| Rated Capacity | 3.5 kW |
| Coil type | fin-tube HX |
| Tube configuration | Staggered<br>22 Step x 2 Row |
| Tube length | 653.2 mm |
| Tube OD | 7 mm |
| Tube thickness | 0.25 mm |
| Tube horizontal spacing | 12.7 mm |
| Tube vertical spacing | 21 mm |
| Tube material | copper |
| Fin type | louver |
| Fin thickness | 0.1 mm |
| Fin spacing | 18 fpi |
| Fin height | 0.7 mm |
| Fin material | Aluminum |

**Table 3:** Operation condition used for simulation

| List | Value | Unit |
|---|---|---|
| Refrigerant | R410A | |
| Pressure at inlet | 2840 | kPa(A) |
| Superheat at inlet | 35 | ℃ |
| Refrigerant mass flow rate | 71.8 | kg $hr^{-1}$ |
| Air temperature | 35 | ℃ |
| Air humidity | 40 | % |
| Air velocity | 1.27 | m $s^{-1}$ |

$$\bar{h}_r = h_l\left(0.55 + 2.09 / pr^{0.38}\right). \tag{9}$$

The mean heat transfer coefficient of the refrigerant side can be obtained by applying the operation condition in Table 4 to equation (9) while the heat transfer coefficient of the air side can be obtained using the model of Wang (1999). For the case where all heat transfer tubes are connected in series to form a single refrigerant circuitry, the heat transfer performances of the refrigerant side and the air side can be compared as follows:

$$h_r A_i (= 3750) > h_a h_o (= 1627) \tag{10}$$

The above equation shows that the heat transfer performance of refrigerant side is superior to that of the air side when the refrigerant flows through one path. It implies that the increase in the overall heat transfer performance will not be large in comparison to the cost invested in the refrigerant side since the performance of the refrigerant side is excessively superior to that of the air side.

As the total mass flow rate of the refrigerant is divided into the number of paths, the refrigerant mass flow rate of each path decreases in reverse proportion to the number of paths. In addition, the length of each flow path decreases also in reverse proportion since the whole length of the heat transfer tube is divided into the number of paths,. As shown in equation (6), the heat transfer performance inside the tube is also affected by the mass flux of the refrigerant. If the mass flux is $G$ in a one-path system, the mass flux of each path in a multiple path system is $G / N_p$. When the path number factor ( $N_p$ ) is added to equation (5) using equation (6), we obtain equation (11).

$$\frac{1}{UA} = \frac{N_p^{\,0.8}}{h_r A_i} + \frac{1}{h_a A_o} \tag{11}$$

As the number of paths which makes the two terms in the right side the same is optimum, the optimum number of paths can be calculated as shown in equation (12).

$$N_p = \left(\frac{h_r A_i}{h_a A_o}\right)^{\frac{1}{0.8}} = 2.84 \tag{12}$$

The above calculation is made by assuming that the overall surface efficiency of the air side is 1. If the overall surface efficiency of the air side is 0.9, the optimum number of paths increases to 3.24. Also, it is assumed that $UA$ is constant, and the conductive, contact and fouling thermal resistances are very small. Accordingly, the value produced by equation (12) is an approximate value so that the optimum number of paths for the above condenser is speculated to be 3. The heat transfer coefficient of the refrigerant side which falls under the case of the path number 2.84 obtained from equation (12) is 2775 $Wm^{-2}K^{-1}$. That is to say, the most effective design is to make the investment in the cost (pumping power) to allow the heat transfer coefficient of the refrigerant side to be 2775 $Wm^{-2}K^{-1}$ under the given conditions of air-side heat transfer performance and the internal area of the heat transfer tube.

In order to verify the methodology to determine the number of paths which is applied above, the performance of the condenser with the specification presented in Table 3 was analyzed in the operation condition presented in Table 4. The refrigerant circuitries organized are shown in Figure 2. The equations used for the analysis is shown in Table 5. The analysis is conducted assuming the followings: 1) the effect of the oil inside the tube is disregarded; 2) the internal surface of the tube is smooth; 3) the flow of the two working fluid is in steady state; 4) the conductive heat transfer between tubes and rows is disregarded; and 5) pressure drops only take place on curved tube parts (U-bends) without having any heat transfer. The analysis is conducted such that the refrigerant flow of each path is determined to make the outlet pressure at each path the same, setting the total refrigerant flow and the inlet pressure constant. The result of the analysis is shown in Figure 3. The abscissa and the ordinate represent the pumping power and the heat transfer capacity, respectively. It shows a trend wherein the pumping power decreases and also the heat transfer capacity decreases as the number of paths increases. It can be seen that the change in the reduction rates of the pumping power and the heat transfer capacity following the increase in the number of paths are similar to the change rate trend explained in the previous section. While the pumping power of the heat exchanger with path numbers of 1 or 2 is much higher than that of the heat exchanger with a path number of 3, its heat transfer capacity

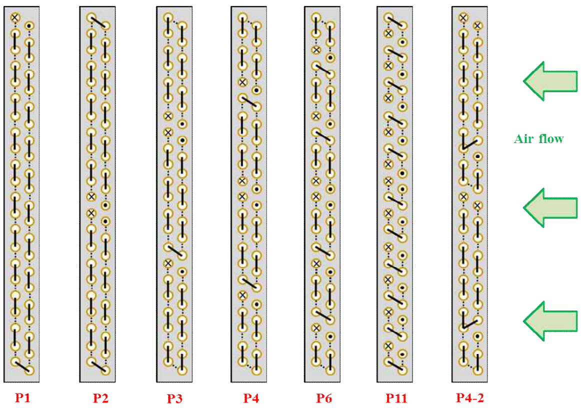


**Figure 2:** Refrigerant circuits examined

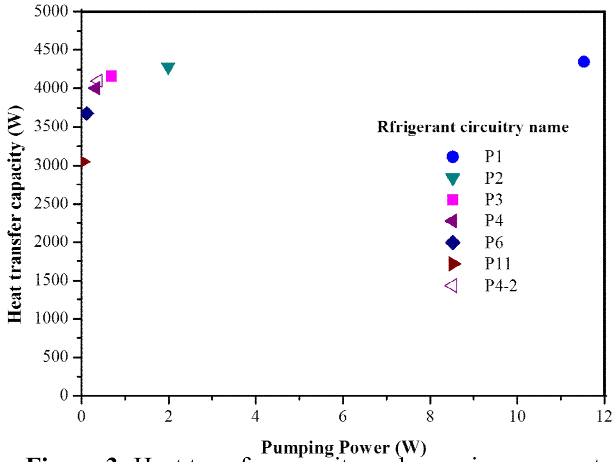


**Figure 3:** Heat transfer capacity and pumping power at condenser condition

is not much higher. Also, though the pumping power of the heat exchanger with the path number of 6 or 11 is a little lower than that of the heat exchanger with the path number of 4, the heat transfer performance is shown to be much lower. Path number 3, from which the increase rate of the heat transfer performance starts sharply decreasing in comparison to increase in the pumping power, can be regarded as the optimum number of paths. Path number 3 is quite similar to the path number 2.84 deduced by applying the methodology to determine the number of paths.

## 4. DIVERGED/MERGED PATH

In the previous section, the design of paths with no divergence/mergence was discussed. At this time, the number of paths was deduced using the mean heat transfer coefficient of the refrigerant side of the whole heat transfer tube. But, it is required to investigate the suitability of divergence/mergence considering the point at which the heat transfer coefficient changes as the refrigerant flows through the heat transfer tube of the heat exchanger.

The performance of the refrigerant side can be controlled by changing the mass flux through divergence/mergence even though the quality changes as refrigerant flows into the tube. Figure 4 shows the condensation heat transfer coefficient and the pressure loss inside the heat transfer tube for the same heat exchanger considered in the previous section with a path number of 2. The analysis in Section 3 suggests the mean condensation heat transfer coefficient of refrigerant-side is about 2775 $Wm^{-2}K^{-1}$ when the number of paths is at the optimum. When this heat transfer coefficient is considered, we can see that the heat transfer coefficient in the quality range from 0.6 to 0.9 is excessively large while those in the ranges from

0.0 to 0.2 and from 1.0 to 1.3 are too small. Accordingly, it is presumed to be advantageous to reduce the pumping power by increasing the number of paths in the quality region between 0.6 and 0.9, and reasonable to increase the heat transfer coefficient of the refrigerant side by reducing the number of paths in the region between 0.0 and 0.2. But the quality region between 1.0 and 1.3 is a superheated vapor area, in which the heat transfer coefficient of the refrigerant side is low and the pressure loss is large. Though the heat transfer performance of this section can be considerably improved by reducing the number of paths, it is not desirable to adopt a design of reducing the number of paths as the pressure loss may rapidly increase and a problem of vibration and noise may occur.

**Table 5:** Correlations used for simulation

| Items | Zone | Correlations |
|---|---|---|
| Heat transfer coefficient | air side<br>liquid refrigerant<br>two-phase refrigerant<br>vapor refrigerant | Wang (1999)<br>Ditttus-Boelter (1930)<br>Shah (1979)<br>Gnielinski (1976) |
| Pressure drop in matrix | air side<br>liquid refrigerant<br>two-phase refrigerant side<br>vapor refrigerant | Wang (1999)<br>Churchill (1977)<br>Friedel (1979)<br>Churchill (1977) |
| Pressure drop in U-bend | liquid refrigerant<br>two-phase refrigerant<br>vapor refrigerant | Ito (1959)<br>Chen (2004)<br>Ito (1959) |

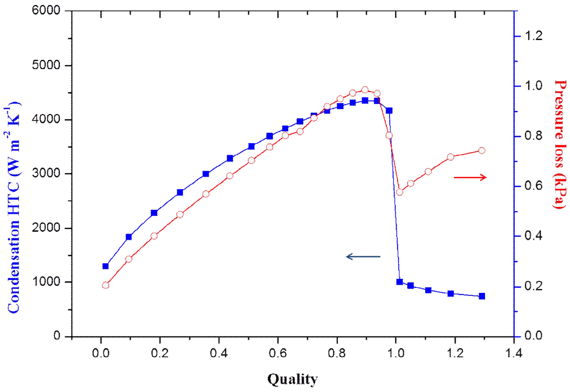


**Figure 4:** Heat transfer coefficient and pressure loss of 2-path condenser

**Table 6:** Optimum path number at each quality

| Quality | Condensation HTC | $N_p$ |
|---|---|---|
| 0 | 2048 | 0.68 |
| 0.1 | 3540 | 1.36 |
| 0.2 | 4506 | 1.83 |
| 0.3 | 5321 | 2.26 |
| 0.4 | 6037 | 2.64 |
| 0.5 | 6676 | 3.00 |
| 0.6 | 7245 | 3.32 |
| 0.7 | 7741 | 3.61 |
| 0.8 | 8144 | 3.84 |
| 0.9 | 8386 | 3.99 |

In order to determine the number of paths for divergence/mergence, local heat transfer coefficient should be considered rather than the mean heat transfer coefficient. Table 6 shows the optimum number of paths calculated for a different quality using equation (6) and equation (12). It shows that the refrigerant circuitry should be designed such that the number of path is increased where the local quality is high. That is to say, divergence/mergence of paths can be applied by predicting the behavior of the quality and selecting the number of paths suitable for it.

Divergence/mergence was applied to a condenser with the specification presented in Table 3 under the operation condition presented in Table 4. The path number of 4-2 means the case wherein the number of refrigerant tubes is 4, which are merged into 2 in the middle to make two outlets. The points at which the paths are merged are selected as the points with the quality of about 0.5 referring to Table 6. A schematic diagram for the refrigerant circuitry is shown in P4-2 of Figure 2. The analysis result is shown in Figure 3. While the pumping power of P4-2 is much smaller than that of P2, the heat transfer performance is a little lower. Also, while the heat transfer performance is much higher than that of P4, the pumping power is only a little increased. Accordingly, the performance of the heat exchanger is found to have been improved by applying divergence/mergence.

## 5. CONCLUSION

There are innumerable methods of organizing the refrigerant circuitry of a fin-tube heat exchanger comprised of a number of tubes. The number of paths which is one of the factors composing a refrigerant circuitry changes the pumping power and the heat transfer coefficient of the refrigerant side. In the relation between the two convective thermal resistances, when the number of paths is selected to make the two thermal resistances equal, the overall heat transfer performance is maximized in comparison to the pumping power. The optimum number of paths is deduced considering such a point. The validity of the methodology to determine the optimum number of paths is verified by applying the newly proposed methodology to a condenser of an air-conditioner. Furthermore, as the heat transfer coefficient changes as the refrigerant flows into the tube, the divergence or mergence of path can also be determined by considering the optimum number of paths corresponding to local vapor quality

## NOMENCLATURE

| | | |
|---|---|---|
| A | hotel cost | (US$/night) |
| $A$ | area | (m$^2$) |
| $D$ | diameter | (m) |
| $G$ | mass flux | (kg m$^{-2}$ s) |
| $L$ | length | (m) |
| $N_p$ | path number | |
| Pr | Prandtl number | |
| $\dot{Q}$ | heat transfer rate | (W) |

| | | |
|---|---|---|
| $R$ | thermal resistance | ($m^2\ K\ W^{-1}$) |
| Re | Reynolds number | |
| $T$ | temperature | (K) |
| $U$ | overall heat transfer coefficient | ($W\ m^{-2}\ K^{-1}$) |
| $\dot{V}$ | volume flow rate | ($m^3\ s^{-1}$) |
| $c_p$ | specific heat at constant pressure | ($J\ kg^{-1}\ K^{-1}$) |
| $h$ | convective heat transfer coefficient | ($W\ m^{-2}\ K^{-1}$) |
| $k$ | thermal conductivity | ($W\ m^{-1}\ K^{-1}$) |
| $pr$ | actual pressure/critical pressure | |
| $x$ | thermodynamic vapor quality | |
| $\eta$ | overall surface efficiency | |
| $\mu$ | viscosity | (Pa s) |

**Subscripts**

| | |
|---|---|
| $a$ | air |
| $cont$ | contact |
| $foul$ | fouling |
| $i$ | inner |
| $l$ | liquid |
| $o$ | outer |
| $r$ | refrigerant |

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