Theoretical Model for Condenser of Miniature LiBr-H_2O Vapor Absorption Refrigeration System

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ABSTRACT

The microchannels/minichannels are being used in industry to yield compact geometry for heat transfer in a wide variety of application. The channels provide high heat transfer coefficients and the large surface area to volume ratios, has led to their use in compact condensers for air-conditioning and refrigerant systems. In order to achieve the compactness the present work attempts to develop an innovative design for condensers by the use of minichannel and it is done by providing a series of parallel mini-slots and they are cut on both sides of the plate. The refrigerant is passed on the bottom surface and the coolant is passed on the top surface. The top surfaces of the channels are covered by the cover plates. The work also attempts to develop theoretical modeling of the condensers for both refrigerant and coolant. The modeling is based on the three parameters like geometric, operating and thermal parameters. These parameters that describes the dimensions, input parameters and output parameter of the system.

Keywords – **Condenser, Heat transfer coefficient, Minichannel, Pressure drop, Refrigerant**

I. INTRODUCTION

The recent advancement in the electronic field is miniaturization by providing high functionality, high reliability and compactness at the low cost. This miniaturization increases the power density, high packaging in small area and also increases the heat flux. A host of alternative cooling approaches have been investigated and developed by the researchers in the past few years like mini/micro channels, heat pipes, liquid immersion, jet impingement & sprays, miniature refrigeration. The compact condenser in air conditioning and miniature refrigeration utilizes mini/micro channels for achieving high heat transfer rate. If the condenser does not dissipate the heat at the required rate, the discharge pressure would build up resulting in the high pressure ratio and subsequently an increase in compressor power. One way of increasing the condenser effectiveness is to increase the surface area of condenser with the use of micro/minichannels. Through much research efforts, flow condensation phenomenon in horizontal/vertical tube are well understood and a few correlations have proposed. Shah [15] proposed the simplest correlation for heat transfer coefficient based on the data from the multiple researchers. Breber et.al [3] recommended convective two phase multiplier for Annular flow, a Nusselt –type correlation for stratified flow, due to the lack of appropriate models, annular flow correlations was introduced in the intermittent and bubbly flow. Gairmella and Bandhauever [7] conducted heat transfer experiments using the tubes 0.4<Dh>4.9mm, they addressed the problems in heat transfer coefficient determination due to high heat transfer coefficient and low mass flow rates in microchannels. Thome et.al [19] analyzed data from multiple researcher and developed condensation models for a wide range of mass fluxes, diameter and fluids. Davide Del Colv et.al [5] developed new heat transfer coefficient correlation for square minichannel and compared with the circular channel. The results showed that, the enhancement in the heat transfer for lowest mass velocity due to the effect of surface tension. The author confirms that there was no increase in the heat transfer coefficient for the highest value of mass velocity and condensation was shear stress dominated for the predicted correlation. Hoo-Kyu oh et.al [13] experimentally investigated the heat transfer coefficients for R-22, R-134a and R-410A in a single circular micro tube. The results showed that annular flow is dominated in condensation flow in the small diameter tube. R-410A showed higher heat transfer coefficient than others. The authors compared the correlations of different researchers and concluded that the correlation is not enough to consider the heat transfer characteristics for a small channel. So, it is necessary to develop accurate & reliable correlation to predict the heat transfer characteristics for small channels. Sapali et.al [18] studied the heat transfer coefficient of condensation in micro-fin tubes and smooth tube. The results showed heat transfer coefficient increases with the increasing mass flux and decreases with increasing condensation temperature for both micro-fin tubes and smooth tube. A.Cavallini et.al [2] critically reviewed the correlations to compute heat transfer coefficient for refrigerant condensing inside the tube and for enhanced surface of the tubes. H. Louahlia et.al [10] studied the flow characteristics of the condensation in the miniature tube. The photographs showed that there was a reverse of annular flow at the end of each periodic flow when the bubble leaves the channel and also there is no effect on stratification during the condensation in miniature tube. Akhil Agarwal

[1] & Michael M. Ohadi [16] conducted the experiments on condensation in micro channel, the results showed that the heat transfer increases with the increasing of vapor quality, increasing mass flux and decreasing saturation temperature. Garimella et.al [8] developed the correlation to predict the pressure drop with modification represented the total pressure drop as the summation of the friction pressure drop in the slug and bubble regions and the pressure drop associated with the transition between these regimes. Akhil Agarwal [1] measured the pressure drop by accounting for expansion and contraction terms and acceleration or deceleration pressure drop. The result concluded that with increasing of vapor quality, increasing mass flux and decreasing of saturation temperature increases pressure drop. As shown in literature reviews mentioned earlier, there are many studies related to the heat transfer in micro/mini channels, but only few studies releated to the heat transfer in microtubes. More studies are necessary to develop heat transfer database because some theoretical and experimental works different from each other. So, the present work aims at systematic design of condenser.

II. Cycle description

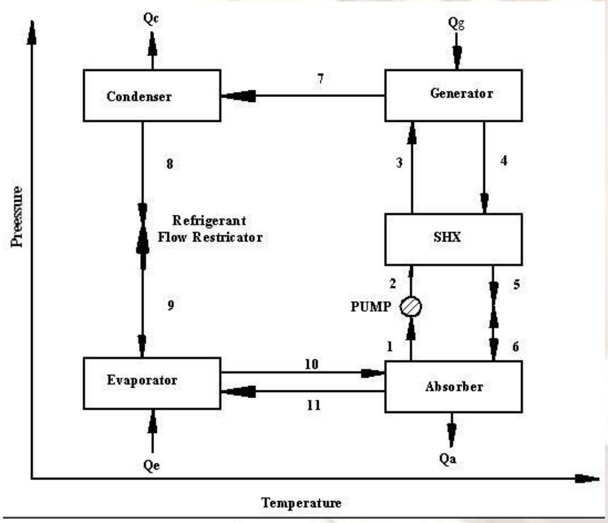

Fig.1: Shows cycle LiBr-H_2O Vapor Absorption Refrigeration System

A single effect LiBr- water absorption refrigeration system is illustrated in Fig.1 and its schematic presentation on a pressure temperature diagram is also illustrated in Fig 1.With reference to numbering system shown in Fig. 1 at point (1), the solution is rich in refrigerant and pump (2) forces the liquid through a Solution heat exchanger (SHX) to the generator (3). The temperature of the solution in the heat exchanger (SHX) is increased. In the generator, thermal energy is added and the refrigerant boils off the solution. The refrigerant vapor (7) flows to the condenser, where heat is rejected as a refrigerant condenses. The condensed liquid (8) flows through a flow restrictor to the evaporator (9). In the evaporator the heat from the load evaporates the refrigerant, which flow back to the absorber (10). A small portion of the refrigerant leaves the evaporator as liquid spillover (11). At the generator exit (4), the fluid consists of the absorbent-refrigerant solution, which is cooled in the heat exchanger. From points (6) – (1), the solution absorbs refrigerant vapor from the evaporator and rejects heat through a heat exchanger. This paper aims and limits to theoretical design of miniature condenser for LiBr-H_2O Vapor Absorption Refrigeration System

III. Theoretical Modeling

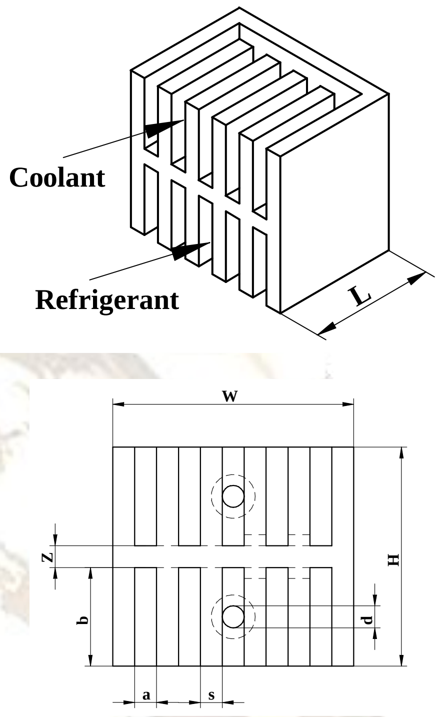

Fig.2: Shows Conceptual Model of Miniature of Condenser

The present work aims to design the miniature condensers with miniature coolant system using minichannels. The Fig. 2 illustrates the construction of a minichannel condenser i.e. like minichannel heat sink, additional with a coolant system. The condenser is composed of a base and two cover plates. The base is usually made from a high thermal conductivity material such as aluminum. In order to obtain the compactness, series of parallel mini-slots of different ranges hydraulic diameters are cut on both sides as shown in the Fig.2. The cover plates are made from a low thermal conductivity material such as glass to ensure that all the heat is removed from the coolant. In operation, the refrigerant is passed into the minichannel on the bottom surface of the base as shown in Fig.2. Heat is generated due to the rejection of latent heat by the Refrigerant. The heat is conducted through the base and removed by the coolant flowing through the other side of the base. Since the condensation process is a two-phase, the various system parameters are introduced in the designing a miniature condenser.

The main parameters that can be grouped are as follows:-1) Operating parameters 2) Geometric parameters and 3)

Thermal/Fluid parameters based on Weilin and Issam Mudwar for the heat sink.

3.1Refrigerant side

3.1.1 Geometric Parameters

The geometric parameters included in miniature condenser are illustrated in Fig.2. The overall condenser dimensions are length (L), width (W), and height (H). The height represents the sum of the height of channel and thickness of cover plate. The thickness between two channels (Z) as shown in the fig.2. The distance between the two channels is proportional to the thermal conduction resistance between the refrigerant side and the coolant side due to this, the thickness always should be less so that all the heat will be absorbed by the coolant. The cover plate can be treated as thermally insulating; thickness has no bearing on the performance of the condensers. Fixing of the hydraulic diameter(D_r), Aspect ratio of channel (β_r) and Fin aspect ratio (α_r) of the channel on both side i.e. refrigerant and coolant side provides the complete geometry like Depth of channel (b_r), Width of channel (a_r), Thickness of wall separating minichannel (s_r), Number of channel (N_r) as follows:-

$$\text{Aspect ratio } (\beta_r) = \frac{a_r}{b_r} \tag{1}$$

$$\text{Fin spacing ratio } (\alpha_r) = \frac{s_r}{b_r} \tag{2}$$

Hydraulic diameter for rectangular channel is given by:

$$\text{Hydraulic Diameter } (D_r) = \frac{2a_r b_r}{a_r + b_r} \tag{3}$$

$$\text{Number of channels } (N_r) = \frac{W}{a_r + s_r} \tag{4}$$

$$\text{Cross sectional area } (A_r) = a_r b_r \tag{5}$$

3.1.2 Operating Parameters

The operating parameters represent condition under which the condenser is expected to operate. They include material, type of refrigerant, Saturation temperature of refrigerant (T_{sat}), inlet (x_i) and outlet (x_o) quality of refrigerant, Outlet pressure of refrigerant (P_o), total mass flow rate of refrigerant (m_r), total heat (Q) represents the heat removal requirement from the refrigerant:

$$Q = m_r(x_i hfg) \tag{6}$$

$$\text{Massflux in channel } (G_r) = \frac{m_r}{A_r N_r} \tag{7}$$

3.1.3 Thermal/Fluid Parameters

The parameters are dependent transport parameters that determine the performance of condensers under given operating and geometric parameters. Thermal parameters includes heat transfer coefficient, pressure drop etc…

3.1.3.1 Heat transfer coefficient during condensation

The heat transfer coefficient of liquid predicted by Dittus-Boelter [6] by assuming all the mass flows inside the channel is given by:

Reynolds number of the liquid refrigerant is given by:

$$Re_L = \frac{G_r D_r}{\mu_L} \tag{8}$$

$$h_L = 0.023(Re_L)^{0.8}(Pr_L)^{0.4}\frac{k_L}{D_r} \tag{9}$$

Shah [15] proposed the simplest correlation for two phase heat transfer coefficient for condensation of steam in small channel based using Dittus-Boelter is given by:

$$h_{tp} = h_L[(1-x_i)^{0.8} + \frac{3.8(x_i)^{0.76}(1-x_i)^{0.04}}{(P_{red})^{0.38}}] \tag{10}$$

Where $P_{red}=(P_{sat}/P_{ct})$

By considering, the adiabatic fin tip condition and the width of the fin is much smaller than its length, Fin efficiency is given by:

$$\eta_r = \frac{\tanh(mb_r)}{mb_r} \tag{11}$$

$$\text{Where, } mb_r = \sqrt{\frac{2h_{tp}}{Ks_r}}b_r$$

In consideration of fin efficiency, the surface heat flux is given by:

$$q'' = \frac{Q}{(2b_r\eta_r + a_r)N_r L} \tag{12}$$

By the consideration of heat flux and two-phase heat transfer coefficient, the surface temperature of the refrigerant side channel is given by:

$$q'' = h_{tp}[T_{sat.r} - T_{w.r}] \tag{13}$$

Since the condensation process takes under constant temperature and pressure. By assuming the surface temperature of the refrigerant side is also remains constant throughout the length. Hence, by assuming coolant exit temperature is equal to the maximum surface temperature of the refrigerant side. In this aspect, the properties of coolant side are evaluated at the average coolant temperature.

3.1.3.2 Pressure drop

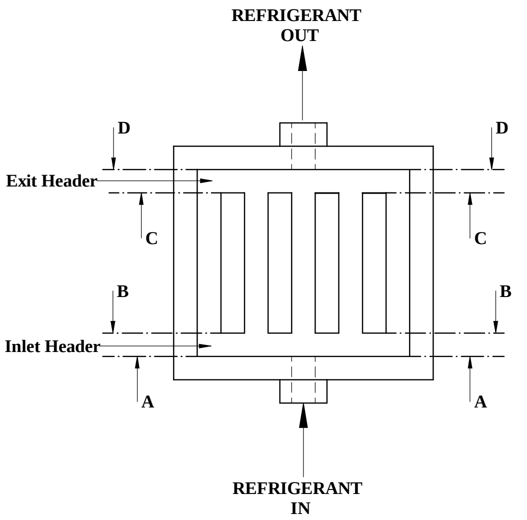

Fig.3: Various sections at which pressure drops are evaluated

The pressure drop is sum of pressure drop across two phase region (ΔP_{tp}), as well as pressure losses and recoveries associated with the expansion pressure drop from inlet tube to inlet header $(\Delta P_{exp.i})$, contraction pressure drop from intake header to channel $(\Delta P_{cot.i})$, expansion pressure drop from channel to exit header $(\Delta P_{exp.o})$, Contraction pressure drop from exit header to outlet $(\Delta P_{cot.o})$

Expansion pressure drop from inlet tube to inlet header $(\Delta P_{exp.i})$ at section A-A is evaluated based on the separated flow model as recommended by Hewitt [11,12] is given by:

$$A_{h.i}=0.004\left[b_r\left(N_r-1\right)+N_r a_r\right] \tag{14}$$

Expansion area ratio is given by

$$\gamma_{exp.i}=\frac{A_t}{A_{h.i}} \tag{15}$$

$$\psi_{exp.i}=\left[1+\left(\frac{\rho_L}{\rho_G}-1\right)\left(0.25x_i\left(1-x_i\right)+x_i^2\right)\right] \tag{16}$$

$$\Delta P_{exp.i}=-\frac{G_t^2\,\gamma_{exp.i}\left(1-\gamma_{exp.i}\right)\psi_{exp.i}}{\rho_L} \tag{17}$$

Contraction pressure drop from inlet header to channel $(\Delta P_{cot.i})$ at section B-B is determined based on the separated flow model as recommended by Hewitt [11, 12] is given by:

Contraction area ratio is given by

$$\gamma_{cot.i}=\frac{A_{h.i}}{A_r N_r} \tag{18}$$

$$C_{cot.i}=\frac{1}{0.639\left(1-\frac{1}{\gamma_{cot.i}}\right)^{0.5}+1} \tag{19}$$

$$\psi_{cot.i}=\left[1+\left(\frac{\rho_L}{\rho_G}-1\right)x_i\right] \tag{20}$$

$$\Delta P_{cot.i}=\frac{G_r^2\psi_{cot.i}}{2\rho_L}\left[\left(\frac{1}{C_{cot.i}}-1\right)^2+1-\frac{1}{\gamma_{cot.i}^2}\right] \tag{21}$$

Garimella et.al [9] predicted the correlation for pressure drop during the condensation of refrigerant between the section B-B and C-C by using Barcozy [4] correlation for void fraction and Lockhart and Martinelli [14] parameters as follows,

Void fraction is determined by using the Barcozy [4] as follows:

$$J=[1+\frac{(1-x_i)^{0.74}}{(x_i)^{0.74}}(\frac{\rho_V}{\rho_L})^{0.65}(\frac{\mu_L}{\mu_V})^{0.13}]^{-1} \tag{22}$$

Liquid and vapor Reynolds number is given by:

$$Re_L=\frac{G_r D_r(1-x_i)}{(1+\sqrt{J})\mu_L} \tag{23}$$

$$Re_V=\frac{G_r D_r x_i}{\mu_V\sqrt{J}} \tag{24}$$

Friction factor for the laminar film and turbulent vapor friction factor is given by:

$$f_L=\frac{64}{Re_L} \tag{25}$$

$$f_V=0.316Re_V \tag{26}$$

With these friction factors, the corresponding single-phase pressure gradients are given by:

$$(\frac{dP}{dz})_L=\frac{f_L G_r^2(1-x_i)^2}{2D_r\rho_L} \tag{27}$$

$$\left(\frac{dP}{dz}\right)_V = \frac{f_v G_r^2 x_i^2}{2 D_r \rho_V} \quad (28)$$

The Lockhart and Martinelli [14] parameter is determined from the definition as follows:

$$x_{tt} = \left[\frac{\left(\frac{dP}{dz}\right)_L}{\left(\frac{dP}{dz}\right)_V}\right]^{0.5} \quad (29)$$

The superficial velocity is given by:

$$j_L = \frac{G_r (1-x_i)}{\rho_L (1-J)} \quad (30)$$

This velocity is used to evaluate the surface tension is given by:

$$\omega = \frac{j_L \mu_L}{\sigma} \quad (31)$$

The interfacial friction factor is then determined as follows:-

$Re_L < 2100$, $A = 1.308 \times 10^{-3}$,
$a = 0.4273$, $b = 0.9295$, $c = -0.1211$

$$\frac{f_i}{f_L} = A x_i^a \, Re_L^{\,b} \, \omega^c \quad (32)$$

The pressure drop during the condensation of refrigerant as follows:

$$\frac{\Delta P_{tp}}{L} = \frac{0.5 f_i G_r^2 x_i^2}{\rho_V J^{2.5} D_r} \quad (33)$$

The pressure drop can be expressed as:

$$\Delta P = \Delta P_{exp.i} + \Delta P_{cot.i} + \Delta P_{tp} \quad (34)$$

Outlet Pressure is given by:

$$P_o = P_i - \Delta P \quad (35)$$

The temperature at the outlet (T_o) and quality can be determined based on the outlet pressure.

Expansion pressure drop from channel to exit header $(\Delta P_{exp.o})$ at section C-C is evaluated based on the separated flow model as recommended by Hewitt [11, 12] is given by:

$$A_{h.o} = 0.004\left[b_r (N_r - 1) + N_r a_r\right] \quad (36)$$

Expansion area ratio is given by

$$\gamma_{exp.o} = \frac{A_r N_r}{A_{h.o}} \quad (37)$$

$$\psi_{exp.o} = \left[1 + \left(\frac{\rho_L}{\rho_G} - 1\right)\left(0.25 x_o (1 - x_o) + x_o^2\right)\right] \quad (38)$$

$$\Delta P_{exp.o} = -\frac{G_r^2 \gamma_{exp.o} (1 - \gamma_{exp.o}) \psi_{exp.o}}{\rho_L} \quad (39)$$

Contraction pressure drop from exit header to channel (ΔP_c) at section D-D is determined based on the separated flow model as recommended by Hewitt [11, 12] is given by:

Contraction area ratio is given by

$$\gamma_{cot.o} = \frac{A_{h.o}}{A_t} \quad (40)$$

$$C_{cot.o} = \frac{1}{0.639\left(1 - \frac{1}{\gamma_{cot.o}}\right)^{0.5} + 1} \quad (41)$$

$$\psi_{cot.o} = \left[1 + \left(\frac{\rho_L}{\rho_G} - 1\right) x_o\right] \quad (42)$$

$$\Delta P_{cot.o} = \frac{G_t^2 \psi_{cot.o}}{2 \rho_L}\left[\left(\frac{1}{C_{cot.o}} - 1\right)^2 + 1 - \frac{1}{\gamma_{cot.o}^2}\right] \quad (43)$$

The total pressure drop can be expressed as:

$$\Delta P = \Delta P_{exp.i} + \Delta P_{cot.i} + \Delta P_{tp} + \Delta P_{exp.i} + \Delta P_{cot.o} \quad (44)$$

Outlet Pressure is given by:

$$P_o = P_i - \Delta P \quad (45)$$

The temperature at the outlet (T_o) and quality can be determined based on the outlet pressure.

Length of single phase region is given by:

$$L_{sp} = \frac{L m_r c_{pr} (T_{out.r} - T_{sat.r})}{(-Q)} \quad (46)$$

Length of two phase region is given by:

$$L_{tp} = L - L_{sp} \quad (47)$$

The length of two phase region and single phase region should not exceed the length of condenser.

3.2Coolant side

3.2.1Geometric Parameters

The geometric parameters include miniature condenser are illustrated in Fig.3. The overall condenser dimensions are length (L), width (W), and height (H). The height represents the sum of the height of channel and thickness of cover plate and the distance between the two channels (H). The distance between the two channels is proportional to the thermal conduction resistance between the refrigerant side and the coolant side due to this, the thickness always should less so that all the heat will be absorbed by the coolant. The cover plate can be treated as thermally insulating; thickness has no bearing on the performance of the condensers. Fixing of the hydraulic diameter(D_c), Aspect ratio of channel (β_c) and Fin aspect ratio (α_c) of the channel on both side i.e. hot and cold side provides the complete geometry like Depth of channel (b_c), Width of channel (a_c), Thickness of wall separating minichannel (s_c), Number of channel (N_c) as follows:-

$$\text{Aspect ratio } (\beta_c) = \frac{a_c}{b_c} \quad (48)$$

$$\text{Fin spacing ratio } (\alpha_c) = \frac{s_c}{b_c} \quad (49)$$

Hydraulic diameter for rectangular channel is given by:

$$\text{Hydraulic Diameter } (D_c) = \frac{2a_c b_c}{a_c + b_c} \quad (50)$$

3.2.2 Thermal/Fluid Parameters

The parameters are dependent transport parameters that determine the performance of condensers under given operating and geometric parameters. Thermal parameters includes heat transfer coefficient, pressure drop etc…

3.2.2.1 Heat transfer coefficient for coolant side

Satish.G.Kandlikar et.al [17] proposed a simplest correlation for predicting the Nusselt number based on aspect ratio for three side heated channel for constant wall temperature.

$$Nu_c = 7.541(1-2.610\beta_c + 4.970\beta_c^2 - 5.119\beta_c^3 + 2.702\beta_c^4 - 0.548\beta_c^5) \quad (51)$$

Heat transfer coefficient (h_c) in coolant side is given by:

$$h_c = \frac{Nu_c k_c}{D_c} \quad (52)$$

By neglecting the conductive resistance, the Overall heat transfer coefficient is given by:

$$\frac{1}{U} = \frac{1}{h_{tp}} + \frac{1}{h_c} \quad (53)$$

Logarithmic mean temperature difference is given by

$$LMTD = \frac{(T_{sat.r} - T_{c.i}) - (T_{sat.r} - T_{c.o})}{\ln \frac{(T_{sat.r} - T_{c.i})}{(T_{sat.r} - T_{c.o})}} \quad (54)$$

Surface area of the coolant side (A_c) is given by

$$A_c = 2L(a_c + b_c) \quad (55)$$

Number of channels in coolant side is given by:

$$Q = UA_c N_c LMTD \quad (56)$$

Thickness of the fin (s_c) is given by:

$$\text{Number of channels } (N_c) = \frac{W}{a_c + s_c} \quad (57)$$

By considering the heat transfer coefficient of coolant side and adiabatic fin tip condition and the width of the fin are much smaller than its length, Fin efficiency is given by:

$$\eta_c = \frac{\tanh(m_c b_c)}{m_c b_c} \quad (58)$$

Where, $m_c b_c = \sqrt{\frac{2h_c}{Ks_c}}\, b_c$

By relating the conductive and convective heat transfer for a constant heat flux, the average temperature difference between the surface and the fluid is given by:

$$\Delta T = \frac{Q}{h_c(2b_c\eta_c + a_c)N_c L} \quad (59)$$

The Coolant outlet temperature, surface inlet temperature and the distance between the two channels is given by:

$$T_{co} = T_{w.r} - \Delta T \quad (60)$$

$$T_{si} = T_{ci} + \Delta T \quad (61)$$

Average coolant temperature at the inlet should be equal to average temperature of the coolant, to maintain the coolant exit temperature same as maximum surface temperature.

$$T_{avg} = \frac{T_{c.i} + T_{c.o}}{2} \quad (62)$$

The distance between the two channels is proportional to the thermal conduction resistance between the refrigerant side and the coolant side due to this, the thickness always should less so that all the heat will be absorbed by the coolant.

$$Q = \frac{KLW(T_{w.r} - T_{avg})}{Z} \quad (63)$$

3.2.2.1 Pressure drop for coolant side

Satish.G.Kandlikar et.al [17] proposed a simplest correlation for predicting the Pressure drop is evaluated by assuming, the core of the channel includes pressure drop due to only frictional losses in the fully developed region and the loss due to the developing region. Reservoirs are large so the Area of the reservoir >> the Area of microchannels. The total pressure drop between the inlet an outlet manifolds would include the pressure drop in the core of the channel plus the minor losses at the entrance and exit. The minor loss is defined by:

$$\Delta P_m = \frac{K_z \rho_c u_m^2}{2} \quad (64)$$

Where, K_z is a loss coefficient related to area changes at the entrance or exit by assuming, area ratio as zero. The core of the channel includes pressure drop due to only frictional losses in the fully developed region and the loss due to the developing region.

The Hagenbach factor $K(\infty)$, which is defined by:

$$K(\infty)=0.676+1.276\beta_C+3.3089\beta_C^2-9.5921\beta_C^3+8.9089\beta_C^4-2.9959\beta_C^5 \quad (65)$$

The f Re term is given by:

$$f\,Re_c = 24(1\,.\,1.3553\beta_C + 1.9467\beta_C^2-1.7012\beta_C^3+0.9564\beta_C^4-0.2537\beta_C^5) \quad (66)$$

The total pressure drop is obtained by:

$$\Delta P_c = \frac{2(fRe_c)\mu_c u_m L}{D_c^2} + K(\infty)\frac{\rho_c u_m^2}{2} + z_{cot}\frac{\rho_c u_m^2}{2} + z_{exp}\frac{\rho_c u_m^2}{2} \quad (67)$$

Outlet Pressure is given by:

$$P_{co} = P_{ci} - \Delta P_c \quad (68)$$

3.1.3 Operating Parameters

The operating parameters represent condition under which the condenser is expected to operate. They include material, type of coolant, Inlet temperature (T_{ci}), Outlet pressure of coolant (P_{co}), total mass flow rate of coolant (m_c), total heat (Q) represents the heat absorbed by the coolant. By assuming coolant only absorbs the heat and the complete surface is insulated.

$$Q = m_c C_{p.c}(T_{co} - T_{ci}) \quad (69)$$

$$\text{Massflux in channel } (G_c) = \frac{m_c}{A_c N_c} \quad (70)$$

Reynolds number is given by:

$$(Re_c) = \frac{G_c D_c}{\mu_c} \quad (71)$$

IV. THEORETICAL RESULTS

The above model theoretically simulated and the results are tabulated as below:-

Table 1: Geometric Parameter

Geometric Parameter		
Parameter	**Refrigerant**	**Coolant**
Length (L) m	0.05	0.05
Width (W) m	0.05	0.05
Hydraulic Diameter (D) m	0.003	0.003
Aspect ratio (β)	0.25	0.25
Fin spacing ratio (α)	0.25	2.44
Material	Aluminum	
Channel Geometry		
Depth (b) m	0.0075	0.0075
Width (a) m	0.001875	0.001875
Thickness (s) m	0.001875	0.004575
No. of channel (N)	13	8
Area of header (A_h) m^2	0.000458	0.000458
Tube Diameter (d) m	0.002	0.002

Table 1: Operating Parameter

Operating Parameter		
Parameter	**Refrigerant**	**Coolant**
Fluid	Steam	Water
Heat dissipated (Q) W	100	100
Input temperature (T) °C	110	30
Input pressure (P) bar	1.4	1.4
Mass flux (G) kg/m^2s	0.490173	5.351102
Inlet Quality (x_i)	0.5	

Table 2: Thermal/ Fluid Parameters

Thermal/ Fluid Parameters	
Refrigerant	**Values**
Reynolds number (Re)	5.656
Two phase heat transfer coefficient (h) w/m^2K	392
Fin efficiency (η)	96.33
Surface temperature(T_w) °C	85.96
Total Pressure drop (ΔP) pa	8.757
Outlet Quality (x_o)	0.015
Coolant	**Values**
Reynolds Number (Re)	23.84
Heat transfer coefficient (h) w/m^2K	1130
Overall Heat transfer coefficient (U) w/m^2K	291
Fin efficiency (η)	95.7
Surface Inlet temperature (T_s) °C	44.07
Pressure drop (ΔP) pa	0.783

V. CONCLUSION

In order to achieve the compactness the paper attempts to develop an innovative design for condensers by the use of minichannel and it is done by providing series of parallel mini-slots of and they are cut on both sides of the plate.The top surfaces of the channels are covered by the cover plate. The paper also presents the simplest and systematic procedure for the designing of miniature condenser for the low heat flux that can applicable for miniature refrigeration system. System parameters and predictive tools are arranged in the chronological order of the design procedure. System parameters are grouped as Geometrical, Operating and Thermal/fluid parameters. The geometric parameters includes condenser and channel dimensions for both refrigerant and coolant. The operating parameters include input parameters for refrigerant side and coolant like saturation temperature of refrigerant and inlet temperature of the coolant etc…and are often specified beforehand. Thermal/fluid parameters are the output parameters for refrigerant and coolant that describe the transport behavior of the channel like pressure drop, heat transfer coefficient etc… and whose magnitude are determined by the designer in selecting channel dimensions. The paper limits the experimental validation that to be needed to support of our design.

Nomenclature

a	Width of channel in m
A	Area in m^2
b	Depth of channel in m
Cp	Specific heat in kJ/kgK
d	Diameter of inlet tube in m
Dh	Hydraulic diameter in m
dP/dz	Pressure gradient
f	Friction coefficient
fi	Interfacial friction factor
G	Mass flux in kg/m^2s
h	Heat transfer coefficient in W/m^2K
H	Height in m
J	Void fraction
j	Superficial velocity in m/s
k	Thermal conductivity of fluid in W/mK
K	Thermal conductivity of material in W/mK
K_z	Loss coefficient
L	Length in m
LMTD	Logarithmic Mean temperature difference
m	Mass flow rate in kg/s
N	No of channels
Nu	Nusselt number
P	Pressure in bar
Pr	Prandtl number
Q	Heat load in kW
q``	Heat flux in w/m^2
Re	Reynolds number
s	Thickness of fin in m
T	Temperature in K
U	Overall heat transfer coefficient in W/m^2K
u	Mean velocity in m/s
W	Width in m
x	Quality
x_{tt}	Lockhart and Martinelli parameter
Z	Thickness in m
z	Loss coefficient

Subscript

avg	Average
c	Coolant
cot	Contraction at inlet
ct	Critical
e1	Expansion at inlet
exp	Expansion
G	Gas
h	Header
i	Inlet
L	Liquid
m	Minor
o	Outlet
r	Refrigerant
red	Reduced
s	Surface
sat	Saturation
sp	Single phase
t	Tube
tp	Two phase
V	Vapor
w	Wall

Greek letters

μ	Viscosity in kg/ms
α	Fin Aspect ratio
β	Aspect ratio
γ	Expansion area ratio
Δ	Difference
η	Efficiency
ρ	Density in kg/m^3
ψ	Parameter
ω	Velocity in m/s

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