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Determination of Lockhart-Martinelli Parameter Using CFD in 2D Vertical Rectangular and offset Mini-Channels with R717

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Abstract

Study of mini-channels has become more popular in the current scenario due to the increasing developments in the field of compact heat exchangers due to the inherent property of increased heat transfer coefficients. Two types of vertical mini-channels, rectangular and offset channels are studied at six different configurations. The flow properties are analyzed using commercial CFD code by varying hydraulic diameters from 1 – 3 mm and mass flux from 5 – 75 kg/m^{2}s. The friction factor is determined using CFD in two dimensional steady state conditions. The results are compared with available empirical correlations. Modified correlations are proposed for friction factor to determine Lockhart-Martinelli parameter based on CFD results. The Lockhart-Martinelli parameters are determined from modified friction factor correlations.

Keywords: CFD, Friction factor, Lockhart-Martinelli parameter, Mini-channels, Pressure drop

NOMENCLATURE

- m Mass flux, kg/m^{2}s
- D_h, d_h Hydraulic diameter, mm
- f Fanning friction factor
- H Free flow height, mm
- I Turbulence intensity
- l Fin Length, mm
- P Saturated Pressure, M Pa
- Re Reynolds number
- S Lateral fin spacing, mm
- T Fin thickness, assumed as 0.5 mm
- x Vapor quality
- X Lockhart-Martinelli number
- $\frac{\Delta P}{\Delta Z}$ Pressure gradient

Greek letters

- ρ Density, kg/m^3
- μ Dynamic viscosity, Pa s

Subscripts

- l Liquid side
- g Vapor side

INTRODUCTION

An increasing attention is being paid to heat transfer from small channels in current scenario, which generates large amounts of heat. The methods used till now of gaseous or liquids mediums do not give very large heat transfer coefficients. Because of this reason, the potential ability of heat exchange during phase change of the refrigerants in the flow in channels are used. The channels in which the cooling process is done have very small dimensions and hence the flow of the boiling medium occurs in the so-called mini-channels. According to Kandilkar (2003), mini-channels are single tubes or multi-port extruded aluminum channels with inner hydraulic diameter in the range 0.2–3 mm. As stated by Cavallini et al. (2009), no correlations are able to satisfactory predict the frictional pressure drop during liquid vapor flow in different mini-channels over a wide range of test conditions. Dutkowisky (2008) stated that because of increase in tube wall friction, the pressure drop with the mini-channel is larger than that of a conventional tube.

Heat transfer and pressure drop characteristics for liquid single phase flow over an array of micro pin fins in a mini-channel are investigated by Tullius et al. (2012). Six pin fin shapes circle, square, triangle, ellipse, diamond and hexagon are used in a staggered array and attached to the bottom heated surface of a rectangular mini-channel and analyzed. Correlations of Nusselt number with respect to Reynolds number and Darcy friction factor are obtained and compared to previous work. With decrease in fin width and spacing, the Nusselt number increases; but the increase in pressure drop is also significant. Pressure drop increases significantly with a 15% increase to the unfinned channel. To study the flow behavior, Saad et al. (2011) conducted experiments in a compact heat exchanger with two-phase inlet conditions and vertical up flow. The test section consisted of offset strip fin heat exchanger with rectangular cross-section. Single phase pressure drop has been analyzed numerically using CFD simulations for air and water. Manglik and Bergles correlation is used to calculate the friction factor. Brix et al. (2010) studies have shown that the correlations applied for calculating single-phase pressure drop for conventional channels performs well when applied to small channels.

Cavallini et al. (2009) performed single phase pressure drop experiments to achieve critical insight of the test section hydraulic performance. The experiment is carried out for R134a inside a single horizontal mini tube, with 0.96 mm inner diameter at conditions of single-phase flow and adiabatic two-phase flow. The surface roughness is considered not negligible. New experimental frictional pressure gradient data are obtained to calculate new friction factors. The new single phase friction factors are successfully compared against predictions by Hagen–Poiseuille, Blasius and Teplovas; and that by Churchill model over a range of Reynolds numbers. Matkovic et al. (2009) reported the local heat transfer coefficients acquired from the measurement of the local heat flux, saturation and wall temperatures during condensation of R134a and R32 inside a single circular mini-channel test section with 0.96 mm diameter. Tests are conducted with single phase R134a fluid in the test section. The measured friction factor is in good agreement with the Hagen–Poiseuille correlation in the laminar flow. In the turbulent flow regime, it follows the Churchill correlation using the measured value of roughness in the model. Dutkowski (2008) conducted experimental investigations of pressure drop in mini-channels using working fluids - water and air. The test section is made from stainless steel pipes having hydraulic diameters of 0.55, 0.64 and 1.10 mm, respectively. The experiments are conducted in range from Re = 30 till transition to the turbulent flow. An increase in Re number of the water flow, resulted in an increase in flow resistance. Mini-channel with diameter 0.55 mm gave the flow resistance value four times greater than for a mini-channel with a diameter of 1.10 mm, with the same Re number.

Reynau et al. (2005) measured the friction and heat transfer coefficients in two dimensional mini-channels of 1.12 mm to 300 micrometers in thickness. The friction factor is calculated from the measured pressure drop along the channel. Experimental results are in good agreement with classical correlations similar to the channels of conventional size. Wang et al. (2004) conducted experiments to determine the frictional characteristics inside mini-channels with diameter varying from 0.198 to 2.01 mm with water and lubricant oil as the working fluids. Tests are conducted in both rectangular and round configurations. The test results gave a negligible influence of viscosity on the friction factor, if the hydraulic diameter is greater than 1.0 mm. Further, the measured data are well predicted by the conventional correlation in both laminar and turbulent flow conditions. For rectangular configuration with a gap distance of 0.1 mm, the measured friction factors for water are well predicted by the conventional correlation. But the measured friction factors for lubricant oil are under predicted.

According to Choi and Kim (2011), two-phase flow pressure drop correlations is classified as either homogeneous flow models (HFMs) or separated flow models (SFMs). HFMs consider a two-phase flow as a single phase flow with mixed properties. In HFMs, the two-phase density and viscosity should be defined. But SFMs are based on a two-phase frictional multiplier, which is defined as the ratio of the pressure gradient of the liquid phase to that of the gas phase which is the square of Lockhart-Martinelli parameter. The term Lockhart-Martinelli parameter was first introduced by Lockhart and Martinelli (1949). It is defined as the square root of the ratio of pressure gradient of liquid to pressure gradient of gas or vapour. Chisholm (1967) developed equations correlating groups for friction pressure gradient from flow of gas - liquid or vapour - liquid mixtures in pipes in terms of Lockhart-Martinelli.

The following study is carried out to modify the conventional friction factor correlations by validating them with CFD results. Zhang et al. (2003) used FLUENT to predict the fluid flow distribution in plate-fin heat exchangers. The numerical simulation confirms that CFD is a suitable tool for predicting the flow distribution and optimizing the design of the plate-fin heat exchanger. The modified friction factor is used to determine the Lockhart-Martinelli parameter for R717 fluid in rectangular and offset configurations of mini-channels. The Lockhart-Martinelli parameter is essential to determine the two phase multipliers required to calculate the two phase heat transfer coefficients for flow boiling. The numerical analysis of the flow through mini-channels is done to determine the Lockhart-Martinelli parameter using GAMBIT as the modelling, meshing software and computational code ANSYS FLUENT as the solver.

CFD SIMULATION OF THE SINGLE-PHASE FLOW

Modelling and meshing

A vertical single channel, 90^0 to horizontal is considered for both rectangular and offset channels. The offset channel is considered in a single flow path neglecting the diversion of fluid to the neighboring channels. Three variant models of rectangular and offset mini-channels with the hydraulic diameter (d_h) varied from 1–3 mm at steps of 1 mm. A total of six configurations are used. The lengths of the rectangular and offset channels are taken as 100 mm ensuring the flow is fully developed. The length of the hydrodynamic developing (L_h) region is given by equation (2.1), Kandilkar (2014).

$$\frac{L_h}{D_h} = 0.05 \times \mathrm{Re} \quad (2.1)$$

The fin lengths of offset channels are taken as 3, 8 and 9 mm respectively. The meshing for rectangular channels carried out in structured rectangular meshing with boundary layer and offset channels are carried out in unstructured triangular meshing. Figure 2.1 – 2.4 represents a few models created in GAMBIT.

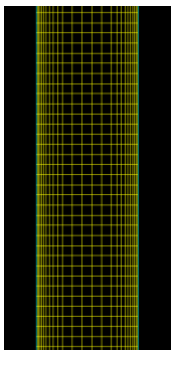

Figure 2.1. Rectangular channel meshing at D_h = 1 mm

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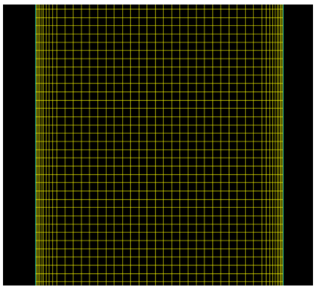

Figure 2.2. Rectangular channel meshing at D_h = 3 mm

Figure 2.3. Offset channel meshing at D_h = 1 mm

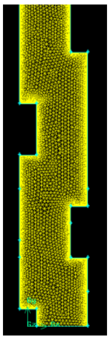

Figure 2.4. Offset channel meshing at D_h = 3 mm

Boundary conditions

Commercial CFD code ANSYS FLUENT is used as the solver. Numerical simulations were conducted for both liquid phase and vapor phase at saturated conditions. For rectangular channels, the laminar model was considered for analysis for liquid phase since the Reynolds number was within the laminar range of Re < 2000. Vapor phase were analysed in both laminar and turbulent flow conditions depending on Re < 2000 for laminar conditions and Re > 2000 for turbulent conditions; McNeil et al. (2013), Zhang (2004), Choi (2014). For offset channels, the laminar model was considered for analysis for liquid phase since the Reynolds number was within the laminar range of Re < 1000. Vapor phase were analysed in both laminar and turbulent flow conditions depending on Re < 1000 for laminar conditions and Re > 1000 for turbulent conditions; Teruel et al. (2009). The K - omega model was used for the turbulence effects. This model gives a more accurate near wall treatment with an automatic switch from a wall function to a low Reynolds number formulation based on grid spacing. The model gives a superior performance for wall bounded and low Reynolds number flows. The specification method used for the model is turbulence intensity and hydraulic diameter method. This specification is required to reduce the reverse flow effects. Turbulence intensity was obtained using the equation 2.2

$$I = 0.16(\mathrm{Re})^{-\frac{1}{8}} \tag{2.2}$$

The fluid properties are inducted manually and fluid properties are assumed constant. Wall material is set to aluminium as default. The inlet boundary conditions are set to mass flow conditions. The outlet conditions are set to pressure outlet conditions. In all set of analysis, the gauge pressure is set to zero and the direction specification is set normal to boundary. In mass flow inlet conditions, the superficial/initial gauge pressure is set to zero since there is no supersonic flow. Mass flux inlet boundary is selected for mass flow specification. The mass flux is varied between 5 – 75 kg/m^2s for each individual model in steps of 5, 10, 15, 30, 45, 60 and 75 kg/m^2s. Wall motion is set to stationary and no slip condition avoiding shear conditions. Analysis is done on pressure based and steady state condition with absolute velocity formulation. Operating pressure is set at saturated conditions of 28^0C. SIMPLE scheme is opted for pressure velocity coupling with least squares cell based gradient; with second order pressure and second order upwind momentum and spatial discretization for accurate results. Then the iterations were done until the solutions converged. Table 2.1 represents the fluid properties of R717.

Table 2.1: R717 properties

Liquid properties		
Density	598.2	kg/m^3
Dynamic viscosity	127.9×10^{-6} Pa s	
Vapour properties		
Density	8.537	kg/m^3
Dynamic viscosity	9.93×10^{-6}	Pa s

The basic continuity equation, ANSYS FLUENT theory guide (2011) is given by

$$\frac{\delta\rho}{\delta t} + \nabla\left(\rho\vec{V}\right) = S_m \tag{2.3}$$

The momentum equation, ANSYS FLUENT theory guide (2011) is given by

$$\frac{\delta\left(\rho\vec{V}\right)}{\delta t} + \nabla\left(\rho\vec{V}\vec{V}\right) = -\nabla p + \nabla\left(\bar{\bar{\tau}}\right) + \rho\vec{g} + F \tag{2.4}$$

$$\bar{\bar{\tau}} = \mu\left[\left(\nabla\vec{V} + \nabla\vec{V}\right) - \frac{2}{3}\nabla\vec{V}I_t\right] \tag{2.5}$$

where

μ = Molecular viscosity

I_t = Unit tensor

S_m = Source term

p = Static pressure

$\overline{\overline{\tau}}$ = Stress tensor

$\rho\vec{g}$ = Gravitational body force

$\vec{F}$ = External body forces

$\vec{V}$ = Velocity vector

The grid independency check is conducted. The number of elements for offset channels for 1 mm, 2 mm and 3 mm are varied from 100587 - 124563, 148952 - 162921, 178768 - 195494 elements respectively. The results showed consistency at 118138, 157644, 190386 elements. Similarly for rectangular channel the results showed consistency from 25194, 35682 and 45978 elements respectively.

ANALYTICAL ANALYSIS

The Lockhart-Martinelli parameter is shown in equation (3.1).

$$X = \sqrt{\frac{\left(\frac{\Delta P}{\Delta Z}\right)_l}{\left(\frac{\Delta P}{\Delta Z}\right)_g}} \tag{3.1}$$

Frictional factor for laminar flow in rectangular flat channel, equation (3.2) is given by Kakac (1987).

$$f = \left(\frac{24}{\text{Re}}\right) \tag{3.2}$$

Pamitran et al. (2008), Piasecka (2013) and Choi et al. (2009) used the Blasius dependency equation to determine the frictional factor for turbulence flow were the Blasius equation is given by equation (3.3).

$$f = 0.0791(\text{Re})^{-0.25} \tag{3.3}$$

Kim and Sohn (2006), Saad et al. (2011) and Teruel (2009) used the Manglik and bergles correlation (1990) to determine the frictional factor for offset strip fins. Equation (3.4) and (3.5) stands for laminar and turbulent region respectively.

$$f = 9.6243(\text{Re})^{-0.7422}\beta^{-0.1856}\partial^{0.3053}\gamma^{-0.2659}\left[1+\left(7.669\times10^{-8}\right)(\text{Re})^{4.429}\beta^{0.92}\partial^{3.767}\gamma^{0.236}\right]^{0.1} \tag{3.4}$$

$$f = 1.8699(\text{Re})^{-0.2993}\beta^{-0.0936}\partial^{0.6820}\gamma^{-0.2423} \tag{3.5}$$

where

$$\beta = \frac{S}{H} \tag{3.6}$$

$$\partial = \frac{t}{l} \tag{3.7}$$

$$\gamma = \frac{t}{S} \tag{3.8}$$

Since Manglik and bergles correlation is meant for full model channel, an assumption is made where the free flow height is taken same as lateral fin spacing (hydraulic diameter) to reduce error. A modified hydraulic diameter is found using the equation (3.9).

$$D_h = \frac{4\times S\times H\times l}{2\left[(S\times l)+(H\times l)+(t\times H)\right]+(t\times S)} \tag{3.9}$$

A standard deviation error of 3.5 % and a mean error of 7 % are obtained between the taken hydraulic diameters and the modified hydraulic diameters which are accounted for during modification of friction factor correlations.

RESULTS AND DISCUSSION

Modification of empirical friction factor correlations

Saad et al. (2011) conducted experiments in a compact heat exchanger with two-phase inlet conditions and vertical up flow in order to study the flow behavior. As an initial study, the single phase flow study in 3D numerical simulations for friction factor was conducted in offset channel with air and water as the medium; which gave at most 15% different from the experimental results. But the difference with Manglik and Bergles correlation (1990) is about 45%. The results obtained by Saad et al. (2011) evidently show the relation between friction factor and Reynolds number follows power law distribution. The power law distribution is shown in figure 4.1.

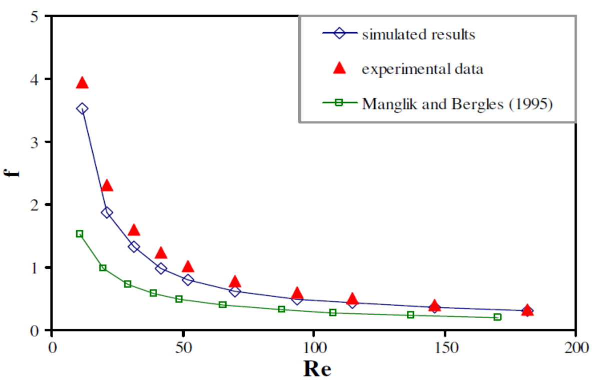

Figure 4.1. Friction factor: comparison between experimental, numerical results and Manglik and Bergles correlation, Saad et al. (2011).

The general form of power law distribution is given by equation 4.1

$$y = C_1 \times x^{C_2} \tag{4.1}$$

where C_1 and C_2 are constants.

The friction factor "*f*" is generally presented by the equation 4.2, Kandilkar (2014)

$$f = \frac{Po}{\mathrm{Re}} \tag{4.2}$$

where Po is the Poiseuille number.

Accordingly power law regression analysis is done to modify the friction factor correlations.

Manglik and Bergles correlation is modified as equations 4.3 and 4.4.

For laminar region

$$f = 13.474(\mathrm{Re})^{-0.7417} \beta^{-0.1856} \partial^{0.3053} \gamma^{-0.2659} \left[1 + \left(7.669 \times 10^{-8}\right)(\mathrm{Re})^{4.429} \beta^{0.92} \partial^{3.767} \gamma^{0.236}\right]^{0.1} \tag{4.3}$$

For turbulent region

$$f = 2.6048(\mathrm{Re})^{-0.2984} \beta^{-0.0936} \partial^{0.6820} \gamma^{-0.2423} \tag{4.4}$$

The Blasius equation is modified as equation 4.5.

$$f = 0.1143(\mathrm{Re})^{-0.2531} \tag{4.5}$$

The new correlations are validated with CFD results. The error for newly obtained friction factors is within ±5%. The empirical rectangular laminar friction factor correlation is within the error ±7% of CFD results. Hence the particular empirical correlation is not modified. Fig 4.2 represents the total pressure distribution across offset mini-channels and rectangular at d_h = 1mm and m = 15 kg/m^{2}s during liquid phase flow. Fig 4.3, 4.4 and 4.5 represents the comparison of CFD and modified empirical correlation friction factor versus Reynolds number.

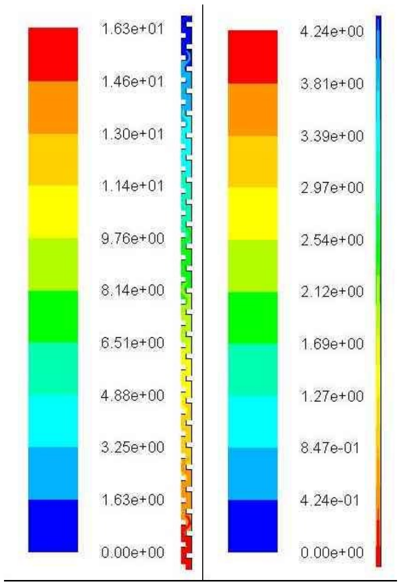

Figure 4.2. (a) Offset (b) Rectangular pressure distribution at d_h = 1mm, m = 15 kg/m^{2}s during liquid phase flow.

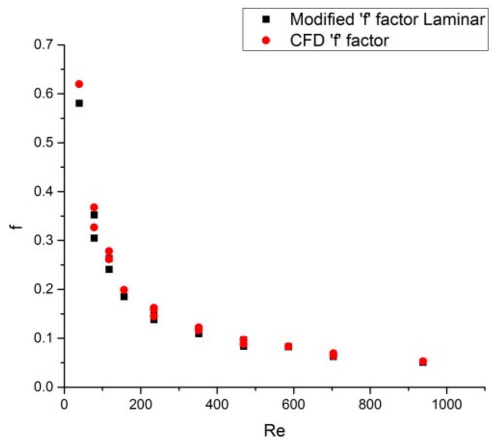

Figure 4.3: Comparison of CFD and modified Manglik and Bergles correlation friction factor at laminar flow

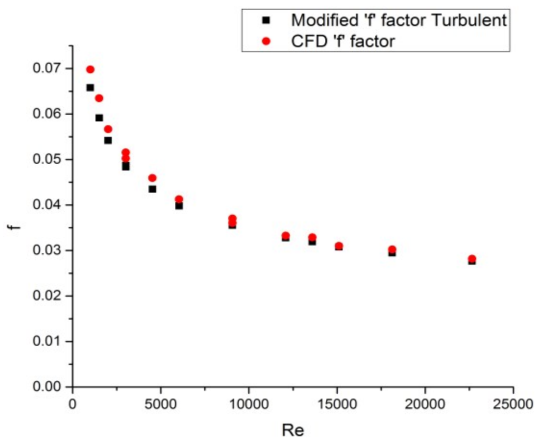

Figure 4.4. Comparison of CFD and modified Manglik and Bergles correlation friction factor at turbulent flow

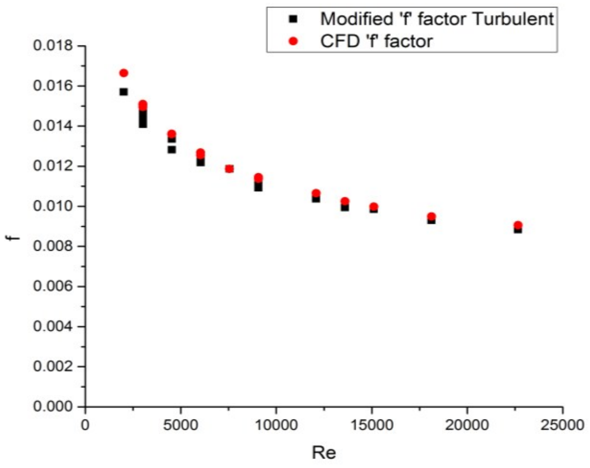

Figure 4.5. Comparison of CFD and modified Blasius correlation friction factor at turbulent flow

Determination of Lockhart-Martinelli parameter using modified friction factor

The classical Lockhart-Martinelli parameter equation can be modified to the equation 4.6.

$$X = \left(\frac{f_l}{f_g}\right)^{\frac{1}{2}} \left(\frac{\rho_g}{\rho_l}\right)^{\frac{1}{2}} \left(\frac{(1-x)}{x}\right) \tag{4.6}$$

Table 4.1: Lockhart-Martinelli parameter without considering vapor quality (x) effects calculated using modified friction factors at m = 15 kg/m^{2}s.

Rectangular mini-channels			Offset mini-channels		
Mass flux, kg/m^{2}s	Hydraulic diameter, mm	$X \times \frac{(1-x)}{x}$	Mass flux, kg/m^{2}s	Hydraulic diameter, mm	$X \times \frac{(1-x)}{x}$
15	1	0.24755	15	1	0.18367
	2	0.22232		2	0.18581
	3	0.21017		3	0.18112

Table 4.1 represents the values of Lockhart-Martinelli parameter calculated considering vapor quality value as variable using modified friction factors at a mass flux of 15 kg/m^{2}s for all varied hydraulic diameters. The vapour quality values lies between 0 and 1 according to experimental conditions.

CONCLUSIONS

Empirical friction factor correlations are modified. The error for newly obtained friction factors is within ±5 % of CFD results. Lockhart-Martinelli parameter is determined using modified friction factor correlations.

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