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Research Article

Gas-liquid two-phase flow pressure drop in flattened tubes: an experimental and numerical study

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ABSTRACT

Experimental, numerical and empirical research is carried out on pressure drop features of air-water two-phase flow in horizontal flattened tubes. Circular tubes of 10.5 mm I.D. made of copper were successively flattened into inner heights of 9, 8, and 6 mm (*AR*=1.27, 1.5, and 2.2, respectively). The experiment operation conditions were 200, 500, and 1000 kg/m^{2}s for mass velocity, 6, 8, 10 LPM for flow rate, and 0 to 0.005 for gas quality. Also, the pressure drop for R134a and R410A was estimated numerically using ANSYS Fluent. The simulation test conditions were for vapor quality of 0.1 to 0.9 and saturation temperature of 40°C, while the conditions for mass velocity and flowrate are taken as that of the experiment test. The experimental data were examined to see how different factors affect on the pressure gradient. According to the outcomes and as compared to the circular tube, the pressure gradient was raised up to 27%, 95%, and 218% for tubes flattened with aspect ratio of 1.27, 1.5, and 2.2, respectively. Moreover, the pressure drop for either air-water or refrigerant fluids is increased dramatically with increasing flow rate, but it decreases with increasing vapor quality. When compared to known circular tube correlations, a good agreement was achieved. Finally, the minimum difference between the experimental, numerical, and correlated results was less than 3% for gas quality of 0.0048 and aspect ratio of 2.2.

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INTRODUCTION

The two-phase flow pressure drop and heat transmission mechanisms in a heat exchangers are affected by a large number of variables. Some investigations have been done for inclined circular pipe during steam condensation [1] and inclined elliptical tube [2]. Others used refrigerant fluids in small diameter tubes with different inclination angles [3]. Also, the pressure drop in circular mini channels [4], micro channels [5, 6], semi triangular micro channels [7], sinusoidal wavy channels [8], and internally finned tube [9, 10] has been investigated.

The majority of refrigerant and air-conditioning systems used circular tubes for their condensers or evaporators. However, using flattened tubes as alternative circular

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tubes will relieve the air-side effectiveness degeneration and also minimize the charging quantity for the refrigerant fluid [11]. Under continuous heat flux, Razi et al. [12] experimentally investigated the behavior of nanofluid flow through flattened tubes regarding heat transmission and pressure gradient. The influence of several factors on flow pressure drop was investigated, including Reynolds number of fluid flow, aspect ratio, and concentration of nano-particles in the base fluid. Yousefi et al. [13] used a model of two-phase mixture to explore the influence of a magnetic field on forced convection heat transmission of ferro-fluids in flattened tubes. They also investigate the impact of tube flattening in a non-uniform magnetic field.

Quibén et al. [14] used four horizontal flattened smooth copper tubes with equivalent diameters of 8.6, 7.17, 6.25, and 5.3 mm and with two different heights of 2 and 3 mm to conduct experiments on pressure gradient during two-phase diabatic flow boiling. The findings were discussed and interpreted under the operation condition of 150 - 500 kg/m^2s for mass flux with R22 and R410A as working fluids. Moreover, a broad experimental investigation was conducted by Thome and Quiben [15] to examine the drops in pressure for refrigerant fluid (R-22 and R-134a) flows in flattened tubes, where circular smooth tubes made of copper with 13.8 mm and 7.17 mm inside diameter were flattened to internal heights of 2 mm and 3 mm, respectively. The results indicated that the pressure gradient penalty of a flattened tube might be as high as seven for a certain mass velocity. As a result, to achieve a balance between transfer of heat and drop in pressure, careful thermal design is required when using flat tubes to reduce gas charging in such condensers and evaporators.

A few investigations can be found in the written works on two-phase flow and heat transmission in flattened tubes. Charging of refrigerant fluid, pressure gradient, and heat transmission in a number of flattened tubes were examined by Wilson et al. [16] during intensification of refrigerant fluids (R134a and R410A). Their findings demonstrate that as a tube is flattened, the refrigerant charge decreases significantly. Similarly, same characteristics for R134a have been performed by Koyama et al. [17] using multi-executor extruded flattened tubes with hydraulic diameters of 1.114 and 0.807 mm. In particular, the constriction of a flattened tube with relatively big aspect ratio would substantially impact two-phase pressure gradient and heat transmission properties [18-21]. Also, 8.7 mm inside diameter circular tubes made of copper were progressively deformed to flattened tubes with internal height of 6.7 mm, 5.2 mm and 3.1 mm by Darzi et al. [22] to predict the behavior of R600a during condensation regarding heat transmission and pressure gradient. Based on the test conditions, their result show that the flattened tube with an internal height of 5.2 mm had the highest overall effectiveness.

Nasr et al. [23] studied experimentally the effect of tube flattening on transfer of heat and pressure gradient for a refrigerant fluid (R-134a) flow. Circular tubes made of copper with 8.7 mm I.D were flattened to four different heights (6.6, 5.5, 3.8, and 2.8 mm). According to the data gathered, they conclude that with decreasing the height, both the coefficient of heat transfer and pressure gradient are increased. Also, based on their experimental results, they achieved a correlation to estimate the pressure gradient in flattened tubes.

Cristiano et al. [24] experimentally investigated the critical heat flux and heat transfer coefficient for flow boiling inside tubes with 2.2 mm identical equivalent inner diameter have been flattened with different aspect ratios. Also, Akhavan et al. [25] empirically investigated the heat transfer characteristics of R600a flow boiling in horizontal tubes that have been flattened. They used circular copper tubes with 8.7 mm I.D to form three tubes that have been flattened to 6.9, 5.5, and 3.4 mm inner heights. They investigated the effects of mass flux, quality of the vapor, and flattened tube inner height on the coefficient of heat transfer. They show that tube flattening improves heat transmission significantly. Also they established a new correlation to predict the coefficient of heat transfer in flattened tubes. Kim et al. [26] used a circular tube of 5.0 mm inside diameter to make flat tubes to study the behavior of R410A regarding evaporation heat transfer coefficients and pressure drops. Mass velocity of 200-400 kg/m^2 s and heat flux from 5 to 15 kW/m^2 were used to cover their test. The results demonstrate that as the aspect ratio increased, so did the evaporation heat transfer coefficients and pressure drops. It has been demonstrated that using equivalent diameter rather than hydraulic diameter results in a more accurate estimate of coefficient of heat transmission. The success of pressure gradient, on the other hand, is reliant on the correlation. For the same input data and test range conditions, similar results have been achieved by Lee and Kim [27], when they used a 7 mm OD round microfin tubes to make flattened tubes with aspect ratio of 2 and 4.

Circular tubes of 8.2 mm I.D. were progressively deformed to flattened tubes with three different inner heights (6.7, 5.2, and 3.1 mm) by Fazelnia et al. [28] to examine the coefficient of heat transmission and pressure gradient during condensation flow of R1234yf. The test operation range was 95 – 380 kg/m^2 s for mass flux and 0.15 – 0.8 for gas quality. They developed two correlations to anticipate the heat transmission coefficient and pressure gradient in flattened tubes depending on their findings and by improving the existing correlations in the written works. The same system has been used by Azarhazin et al. [29] but with different operation conditions to investigate the behavior of R1234yf during boiling flow. Pressure drop and heat transfer of condensation for refrigerant fluids (R410A and R134a) in a round tube of 3.78 mm inner diameter and flattened tubes with various aspect ratio were numerically investigated by Zhang et al. [30]. The findings show that the coefficients of heat transfer and pressure gradients are increased with increase in aspect ratio, vapor quality, and mass flux. Also, the estimated computational results were compared to their empirical equivalents.

The above literature study indicates that researches on non-circular tube heat exchangers is relatively restricted. Furthermore, a limited number of works were subjected to numerical study. Therefore, in the present work, the pressure drops for air-water, R134a and R410A two-phase flow in rounded and four flattened smooth copper passageways are examined. This study uses computational fluid dynamic (CFD) simulation analysis and this section will concentrate on the fluid mechanics side of two-phase flow pressure drop. The purpose of this work is to look at the benefits and drawbacks of two-phase flow in flattened copper tubes. The computational results were validated with experimental data and with empirical models.

EXPERIMENTAL SYSTEM AND PROCEDURES

The air-water test facility was used to conduct the experiment in the Fluid Mechanics Laboratory at Erbil Polytechnic University. In this section the experimental apparatus used in this study is described.

Experimental System

A schematic layout of the system loop is shown in Figure 1. The air-water system consists of a water tank, a pump to circulate water through the system, a 2 m^3 compressed air tank for air supply to the facility, and valves to control the flow rate. The basic instrumentations used are: two flow meters (rotameters), one for water and the other for air, pressure gauges, and a pressure transducer (Comark C9500) to measure the pressure drop along the test section with calibrated accuracy of ±0.2% of full scale. The rotameters were calibrated and based on the standard deviation of the calibration, it has an estimated uncertainty of ±1.3% and ±1.1% of full scale for liquid and gas, respectively.

The one-way valves were used in water and air flow path to prevent mixing. Then the water and air phases are introduced into the test section via a Y-junction (mixer), which placed 1.125 m upstream of the test section in order to ensure fully developed flow. In order to minimize the load on pump and compressor a bypass line for pump and compressor is left open at low flow rate.

Test Section and Test Conditions

The tubes tested had a nominal outside diameter of 12.5 mm with a wall thickness of 1.0 mm and 1.0 m long. The tubes were successively flattened from an initial internal diameter of 10.5 mm to height (H) of 9, 8, and 6 mm, respectively (see Figure 2a). Pressure taps are soldered at the inlet and outlet and are attached to the differential pressure transducer. Experiments were performed using water and air as two-phase flow test fluid. Different total volume flow rates were chosen, which were 100, 133, and 167 cm^3/s (6, 8, and 10 LPM) with mass quality varying from 0 to 0.005.

Experimental Procedures

During the experimental work, the temperature of the laboratory was kept about 20°C and the fluid properties were considered depending on this temperature. For the air-water flow system and for each test, the following procedure was followed:

1. Selecting a known total volume flow rate (e.g. 10 LPM) and adjusting the water and air flow rates that provide this total flow rate (starting with 100% water and 0% air).
2. Measuring the pressure drop for the first test section and for the first mixture percentage.

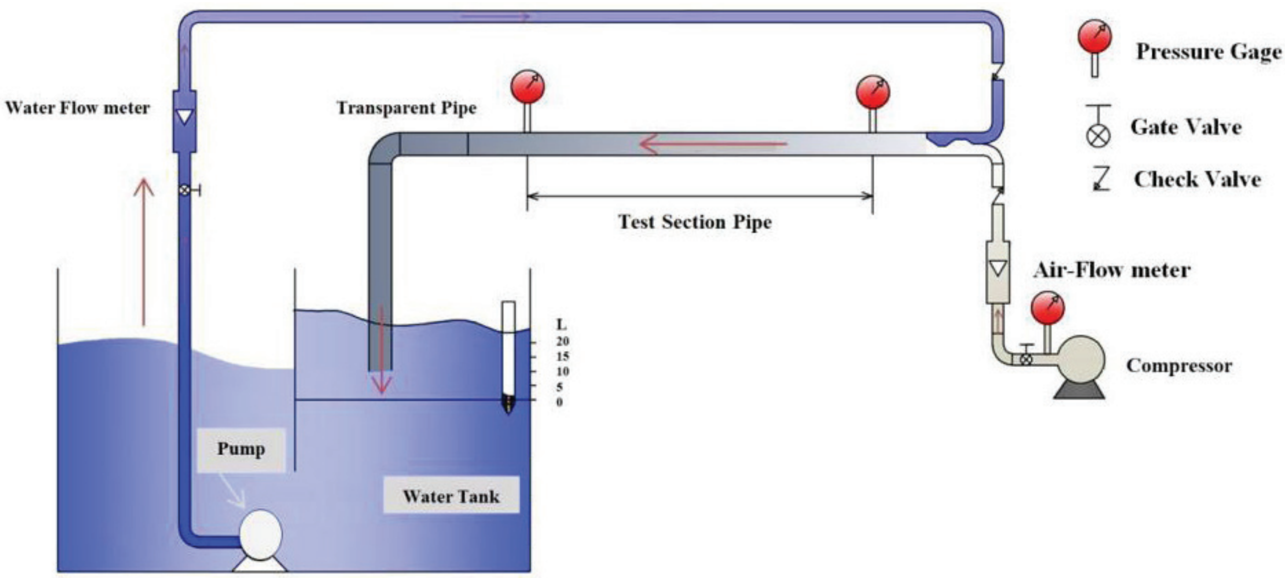

Figure 1. Schematic diagram of the experimental set-up.

3. Increasing air flow rate with decreasing water flow rate while retaining the constant total flow rate (till reaching 0% water and 100% air), and measuring the pressure drop for each condition.
4. Repeating the procedures from 1 to 3 for another total volume flow rate.
5. Repeating the procedures from 1 to 4 for another test section.

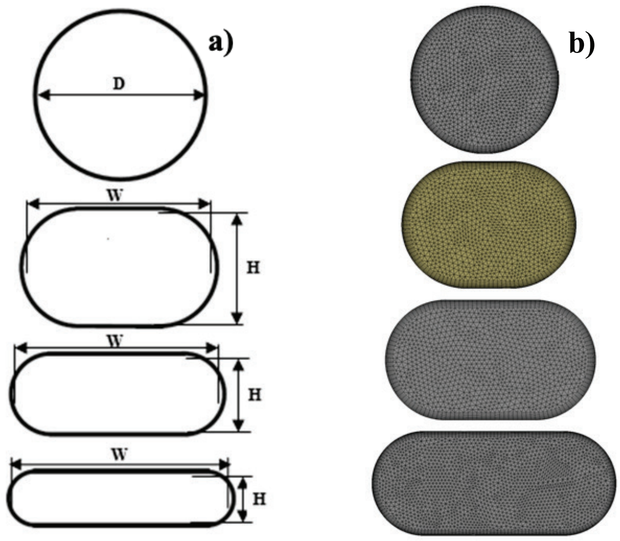

Figure 2. Round and flattened tubes cross-sectional (a) geometry and (b) computational inlet domain.

COMPUTATIONAL MODEL

CFD Governing Equations

In this simulation ANSYS Fluent (Release 19.1) was used to perform the numerical analysis. The computational models are prepared to analyze the interaction of fluids which have specific boundary conditions. A mixture model was adopted in this simulation. This method is reliable to solve problems and applicable for two-phase flow simulation and it is suitable for flows in pipes and channels where phases pass at various velocities. The mixture model analyzes continuity, momentum, and energy equations for mixtures and this model applies the principle of slip velocity because of different velocities of mixture. In this study and based on steady numerical method assumption, the governing equations are given as follows:

The continuity equation

$$\frac{\partial}{\partial t}(\rho_m) + \nabla.(\rho_m \bar{v}_m) = 0 \tag{1}$$

The velocity of the two fluids as a one particle is a mixture velocity ($\bar{v}_m$) and it is given by;

$$\bar{v}_m = \frac{\sum_{k=1}^{n} \alpha_k \rho_k \bar{v}_k}{\rho_m} \tag{2}$$

and the mixture density (ρ_m) of the particle is given by;

$$\rho_m = \sum_{k=1}^{n} \alpha_k \rho_k \tag{3}$$

where (α_k) is the phase volume fraction and (k) represents the phase.

The momentum equation

$$\frac{\partial}{\partial t}(\rho_m \bar{v}_m) + \nabla.(\rho_m \bar{v}_m \bar{v}_m) = -\nabla p + \nabla.[\mu_m(\nabla \bar{v}_m^T)] + \rho_m \bar{g} + \bar{F} + \nabla.\left(\sum_{k=1}^{n} \alpha_k \rho_k \bar{v}_{dr,k} \bar{v}_{dr,k}\right) \tag{4}$$

where ($\bar{F}$) is the force of the body of the fluid particle which is measured in Newton, (n) is the phase number, and (μ_m) is the mixture dynamic viscosity given as:

$$\mu_m = \sum_{k=1}^{n} \alpha_k \mu_k \tag{5}$$

and ($\bar{v}_{dr,k}$) is the average velocity of the particle and known as drift velocity, (k) is the secondary phase;

$$\bar{v}_{dr,k} = \bar{v}_k - \bar{v}_m \tag{6}$$

The slip velocity caused by different velocities of fluids in flow is given by;

$$\bar{v}_{pq} = \bar{v}_p - \bar{v}_q \tag{7}$$

where (p) is the secondary phase and (q) is the primary phase.

The mass fraction for phase (k) in a mixture stream is given by;

$$c_k = \frac{\alpha_k \rho_k}{\rho_m} \tag{8}$$

The Domain of the Computational Model and Boundary Conditions

Four horizontal test sections were used in this work with a computational domain of 1.0 m length and 10.5 mm inside diameter for the round tube. The circular tubes have been flattened to three different aspect ratios (AR) of 1.27, 1.5 and 2.2, respectively. The tetrahedral cells model was adopted for the CFD domain, and a fine boundary layer was used to provide high-quality solution near the wall region and due to capability of tetrahedral mesh to create a model with high number of cells and faces to increase the solution convergence speed. Table 1 presents the complete information of the test section tubes studied in this work, as

Table 1. Dimensions and details of test sections

Tube	W (mm)	H (mm)	AR [W/H]	D_h (mm)	D_e (mm)	No. of nodes	No. of faces	No. of cells
Round	-	-	round	10.5	10.5	627673	3553628	1486404
Flat No.1	11.4	9	1.27	10.33	10.4	634096	3610061	1513065
Flat No.2	12.0	8	1.5	9.97	10.2	678194	3879095	1628502
Flat No.3	13.2	6	2.2	8.59	9.5	637590	3631002	1521947

well as the number of nodes, faces, and cells. The hydraulic (D_h) and equivalent (D_e) diameters are determined using a formula proposed by Quiben et al. [14]. Also, the tube wall perimeter is considered constant for the circular and flat tubes, as reported by other researchers [12, 14, 25].

In this computational model, Fluent solver was adopted and introduced reliable problems solution with appropriate selection of algorithm to prevent divergence of the data available in the created model. Mixture model has been selected instead of volume of fluid since the phases travel at different velocities and this model is more beneficial and accurate for this type of flow and boundary condition. Also, in this computational model, the effect of temperature and its variation has been neglected and the flow was in the steady state assumption.

The effect of shear stress and wall roughness on the fluids flow inside the tubes and channels are important, especially on the pressure of the mixture, but in the present work the effect of shear stress and wall roughness were not investigated. The standard k-ε turbulence model available in the ANSYS Fluent, which is appropriate for a turbulent flow, is used in this simulation to solve the flow characteristics for gas-liquid flow and its conditions. For the momentum, the discretization scheme has been carefully chosen and it was second-order upwind, while for the volume-fraction, the turbulent dissipation-rate, and the turbulent kinetic energy, the scheme used was the first-order upwind to keep the numerical solution stable and accurate. Figure 2b shows the computational geometries and inlet domains for the test sections.

The simulation model was first validated using air-water two-phase flow and with simulation conditions as that for experimental test (i.e., mass velocity of 200, 500, and1000 kg/m^{2}s, flow rate of 6, 8, 10 LPM, and gas quality from 0 to 0.005). Although the simulation test conditions for refrigerant fluids are for vapor quality from 0.1 to 0.9 and saturation temperature of 40°C, the mass velocity and flowrate are taken same as that of the air-water simulation test.

RESULTS AND DISCUSSION

Validation of CFD Results

Figure 3 shows a comparison between computational and experimental results for air-water two-phase flow pressure drop in four different test sections: circular tube with 10.5 mm I.D. and three tubes that have been flattened to aspect ratio (AR) of 1.27, 1.5, and 2.2, respectively. The figure shows that an increase in flow rates increases superficial velocity of both phases, which gives rise to more liquid on the pipe wall resulting in increase in wall resistance. That is why the pressure drop increased with increasing flow rate. On the other hand, the pressure drop also increased as the aspect ratios (ratio of the width to the height of the tube) get larger. This is because an increase in the aspect ratio of the tube narrowing the tube wall lead to rise in fluid particle in contact with the tube wall, thus increasing shear force.

The effect of total flow rate on experimental and computational results with different tube geometries is represented in Figure 4. It can be indicated from this figure that as the total flow rate increased, the difference between experimental and computational results declined. As a verification for the ANSYS software program results, a good acceptance is achieved with the experimental outcomes as shown in Figure 4. The same principle can be observed in Figure 3.

Also a comparison between the experimental findings with the empirical correlation given by Lockhart and Martinelli [31] is presented in Figure 5. In this comparison, the round tube was compared with flattened tubes with three different aspect ratios (1.27, 1.5, and 2.2) for 10 LPM (0.6 m^3/h) total flow rate. A good agreement was achieved from this comparison where the average maximum error of (±10.7%) for the round tube were observed.

A relation between pressure drop and quality of air-water two-phase flow is shown in Figure 6. In this figure, a constant total flow rate of 10 LPM (0.6 m^3/h) was considered for four different tube geometries, one round tube and three flattened tubes with aspect ratio of 1.27, 1.5, and 2.2, respectively. The results are also shown for experimental, computational and empirical using Lockhart-Martinelli [31] correlation. Here it can be concluded that the results of the three methods converge with increase in tube flattening. Also, the pressure drop decreases with increase in gas quality, which is because of increasing the gas flux at the expense of liquid phase for same total flow rate.

Effect of Mass Flux and Mass Quality for R134a and R410A

The pressure decreases as the fluid flows inside the tubes and channels, and its value depends on several factors and parameters. One of these parameters are the tube

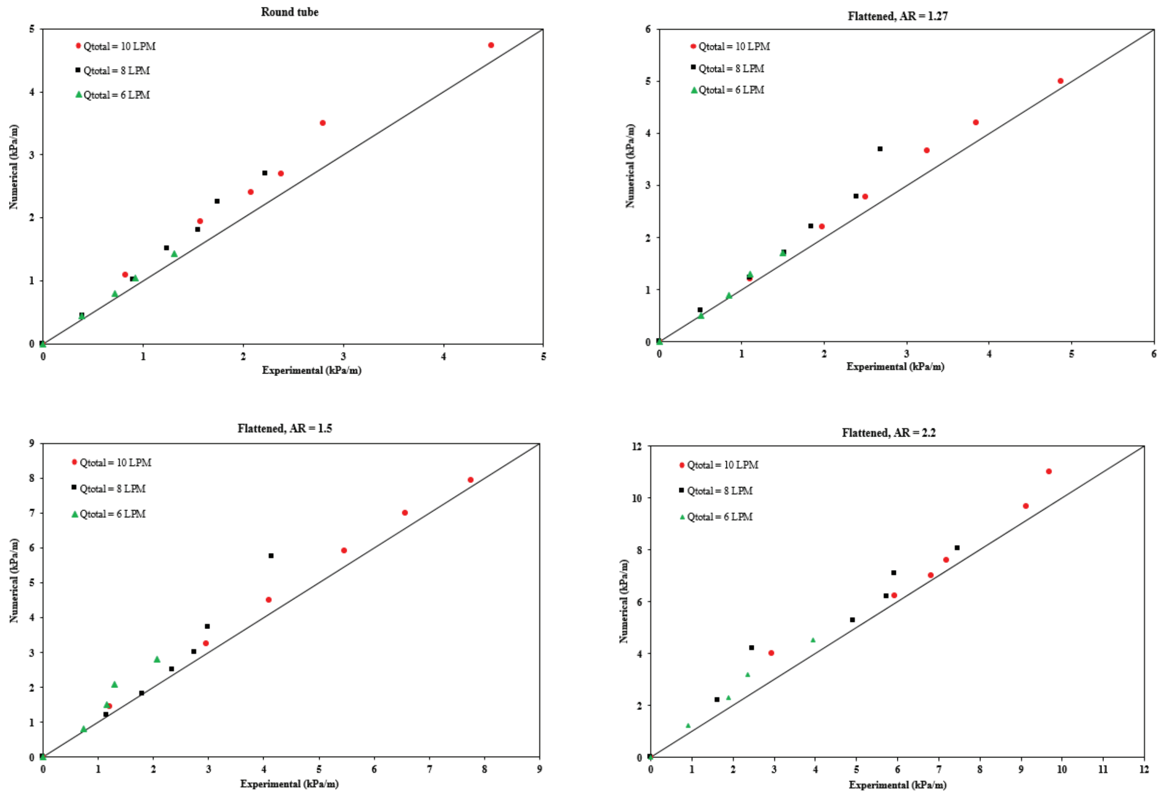

Figure 3. Comparison between experimental and computational air-water two-phase pressure drop for round tube and three flattened tubes with aspect ratio (AR) of 1.27, 1.5, 2.2.

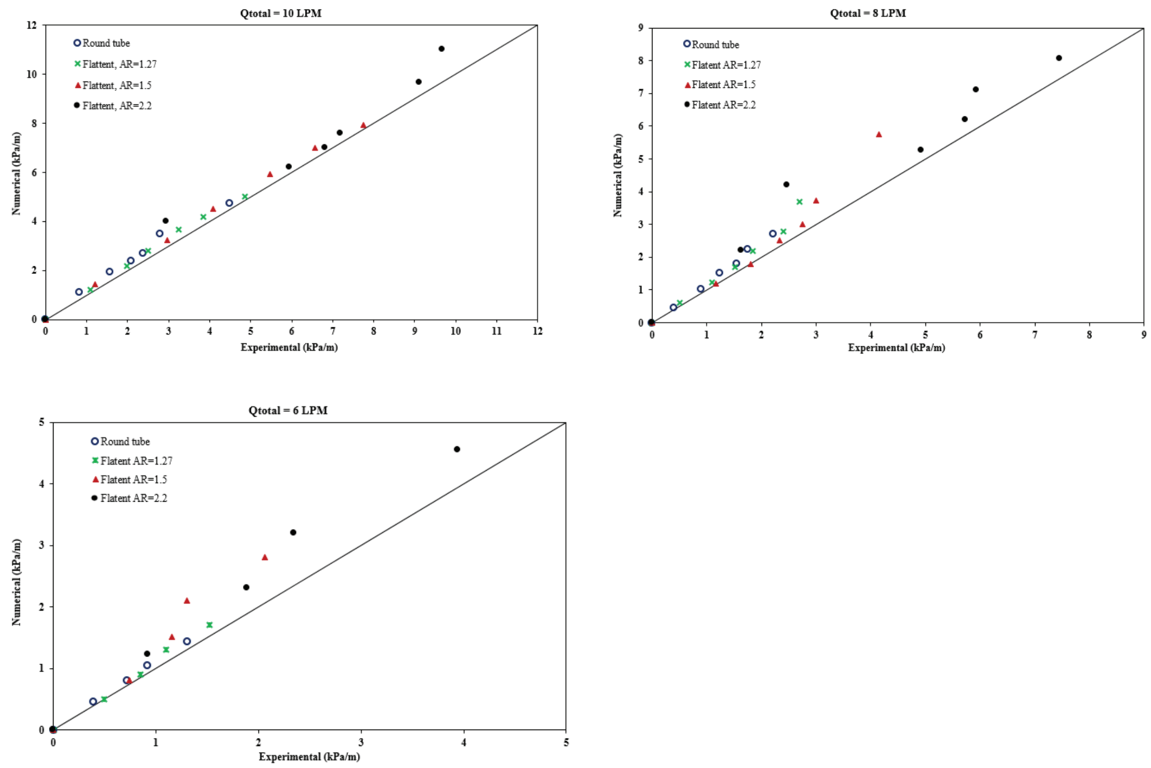

Figure 4. Comparison between experimental and computational pressure gradients for different tube geometries and for total flow rates of 10 LPM, 8 LPM, and 6 LPM.

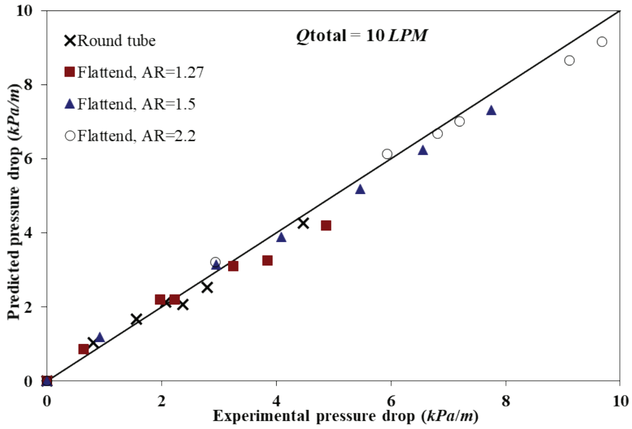

Figure 5. Comparison between experimental and empirical pressure drops using Lockhart-Martinelli (1949) correlation for different tube geometries.

geometry (cross section area). The numerical investigation performed by ANSYS Fluent to compute the frictional pressure gradient of the two-phase refrigerant fluid flow in round and flattened tubes with three different width to height ratios (*AR*) as a function of mass quality and mass flux are illustrated in Figure 7. In this figure, one round and three flattened tubes with *AR* equal to 1.27, 1.5, and 2.2 were considered for R134a and R410A as a working fluid and for three mass fluxes of (200, 500, and 1000 kg/m^2 s). From this figure, it can be notices that the pressure drop increases dramatically with increase in mass flux and also with increase in *AR*. This is because the value of the shear stress intensity between the two phases (liquid and gas) increased with increase in mass flux and vapor quality, resulting in a large frictional pressure drop. Also, the difference in pressure gradient for the rounded and flat tubes converges with increase in mass flux, where the minimum difference can be detected at 500 kg/m^2 s mass flux (see Figure 7b). Then this difference will diverge again because of increase in turbulence flow due to increase in Reynolds number (see Figure 7c). Another point can be indicated here in this figure, where due to the higher working fluid velocity, the drop in pressure for R410A is slightly less than that of R134a at a given vapor quality and mass flowrate per unit area. Also, as gas quality and mass velocity increased, the amount of growth in pressure drops get larger. Similar conclusion was drawn by previous investigators [11, 22, 30].

CFD Plots

The flattening of a round tubes is common in the heat exchangers and mixing processes. As tubes and channels

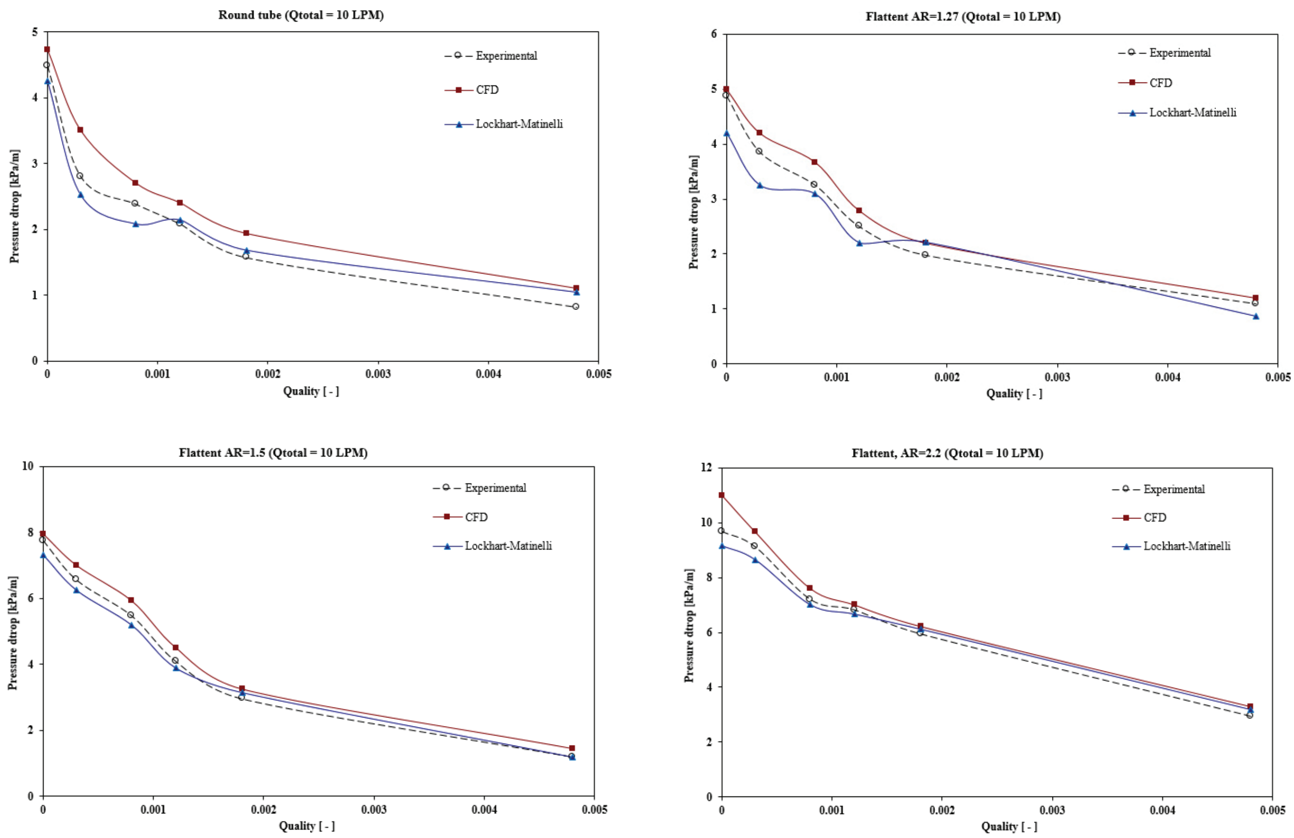

Figure 6. Pressure gradients versus gas quality for round tube and flattened tubes with aspect ratio of 1.27, 1.5, and 2.2 for experimental, computational, and empirical techniques.

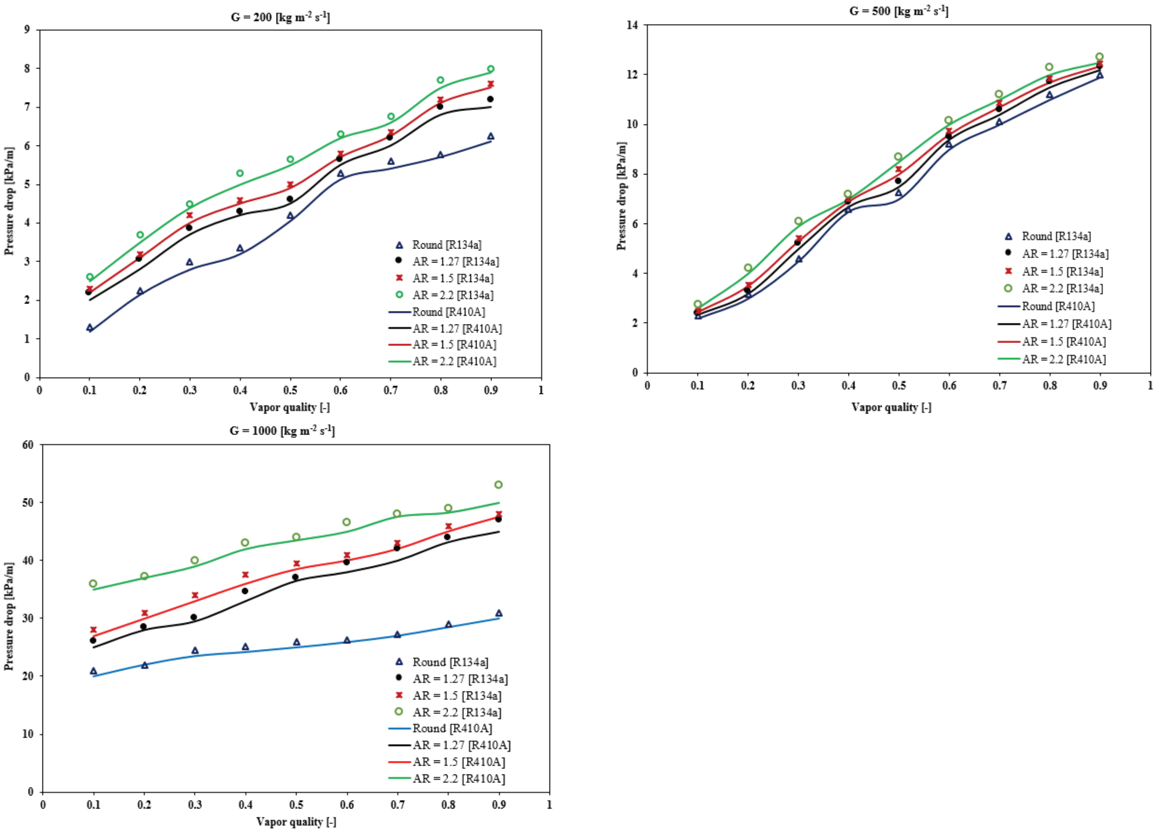

Figure 7. Pressure drop versus vapor quality for R134a and R410A and for round and flattened tubes with three different aspect ratios and at mass flux (G) of 200, 500, and 1000 (kg/m^2 s).

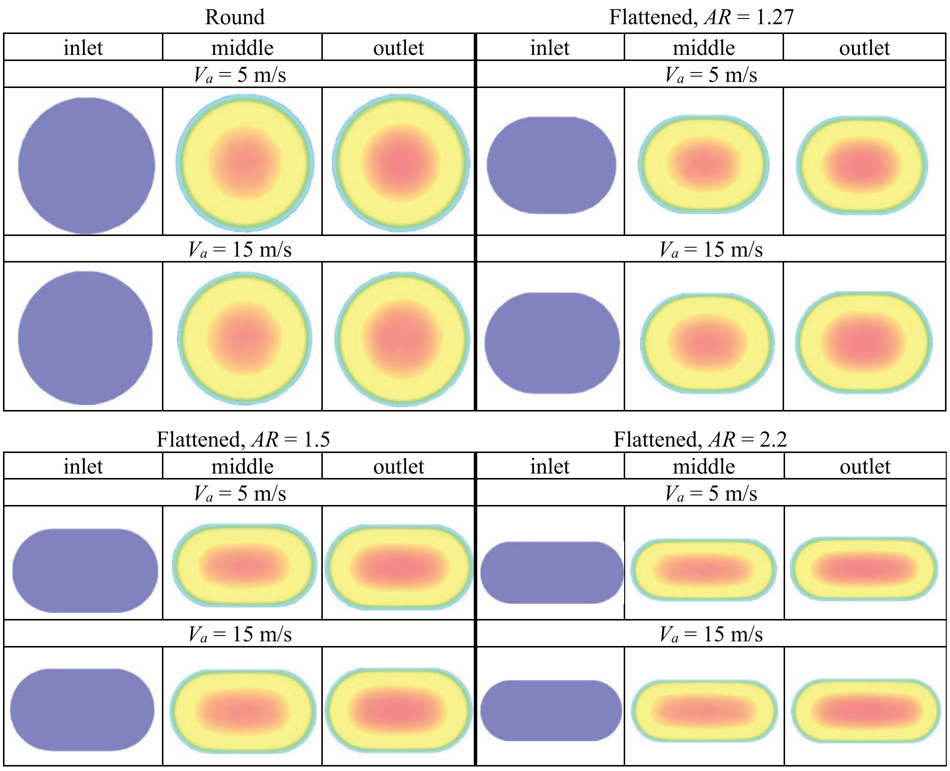

Figure 8. Air-water two-phase radial velocity distribution at V_w = 0.5 m/s for round tube and flattened tubes with aspect ratio (AR) of 1.27, 1.5, and 2.2.

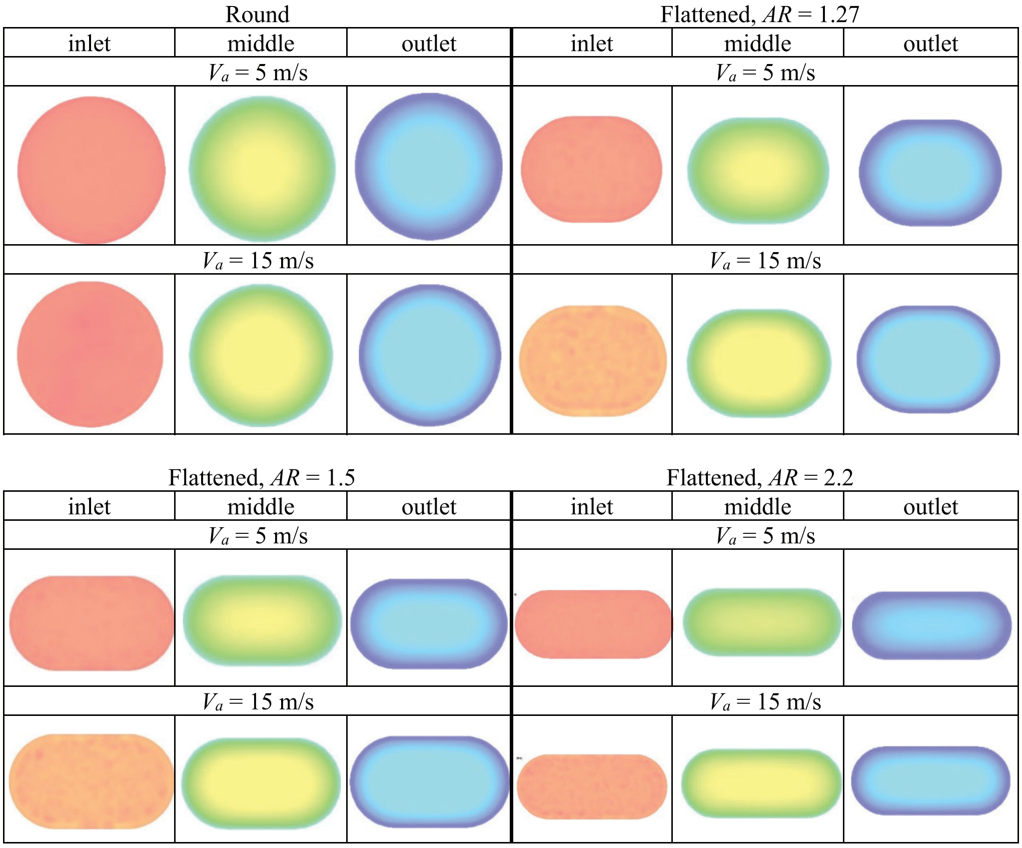

Figure 9. Air-water two-phase total pressure distribution at V_w = 0.5 m/s for round tube and flattened tubes with aspect ratio (AR) of 1.27, 1.5, and 2.2.

become viable in many engineering applications, the interaction of the phases and their flow pattern are significant. The pressure prediction of gas-liquid two-phase for circular and flattened tubes with different aspect ratio were investigated computationally and experimentally. For better understanding of the mixing process and interaction of the fluids, Figures 8 and 9 show air-water two-phase interfaces in these tubes.

In Figure 8, the cross-sectional radial velocity contours are plotted at the entrance, center, and exit of the tubes for constant water velocity of 0.5 m/s and for two values of air velocity, 5 and 15 m/s, respectively. From this figure it can be indicated that the velocity profile increases significantly from inlet location to outlet, and the velocity intensity increased with increasing air velocity.

For the same conditions as in Figure 8, the cross-sectional total pressure contours are plotted at three locations, inlet, middle, and outlet, respectively as shown in Figure 9. The pressure distribution from the contours is irregular without forming any uniform pattern and interface arrangement, but it is obvious that along the tube the scale of the pressure decreases, while there is an increase in pressure with an increase in the air velocity due to increasing the intensity of the shear stress between the two phases.

CONCLUSION

In the present work, experimental, numerical and correlated pressure drop for air-water two-phase flow were conducted in three horizontal flattened copper tubes (AR = 1.27, 1.5, 2.2) and a round tube of 10.5 mm ID (for validation purposes). The test range covered mass flux of 200, 500, and 1000 kg/m^2 s and quality of 0 to 0.005. Also the pressure drop for R134a and R410A are estimated numerically using ANSYS Fluent release 19.2 for same range of mass flux but for vapor quality of 0.1 to 0.9. The following conclusions are drawn from this study:

1. The pressure drops for air-water are increased dramatically with increasing flow rate and also with increasing aspect ratio, while it decreases with increasing vapor quality. The same is true for the refrigerant fluids (R134a and R410A).
2. An acceptable validation was achieved between the simulation and experimental result for air water two-phase flow which was within 10% for all test sections and test range.
3. A good acceptance was achieved between the experimental and empirical correlation given by Lockhart and Martinelli [29], where the average maximum error of (±10.7%) for the round tube was observed. The

difference between experimental and computational results declined as the total flow rate increased.

4. The air-water pressure drop decreases with increase in quality, and the difference between the experimental, numerical, and correlated results are reduced with increase in gas quality and in aspect ratio, where this difference is less than 3% for gas quality of 0.0048 and aspect ratio of 2.2.
5. The difference in pressure gradient for the round and flat tubes converges as mass flux is increased, where the minimum difference, which is about 12%, can be detected at 500 kg/m^{2}s mass flux. Then this difference will diverge again with increase in mass flowrate per unit area.
6. For a given mass flux and vapor quality, the drop in pressure for R410A is slightly less compared to R134a. Also as vapor quality and mass flux increased, the amount of increase in pressure drops get larger.
7. The velocity profile increases significantly from inlet location to outlet of the flattened tubes, and the velocity intensity increased with increasing air velocity.

NOMENCLATURE

AR	Aspect ratio, W/H [-]
Cp	Specific heat capacity [J/kg K]
D	Inner diameter [mm]
E	Specific sensible enthalpy [J/kg]
f	Friction factor for pipe
F	Force [N]
G	Mass flux or mass velocity [kg/m s]
g	Gravitational acceleration [m/s]
H	Tube height [m]
h_{lv}	Latent heat of vaporization [J/kg]
k	Thermal conductivity [W/m K]
P	Pressure [Pa]
ΔP	Two-phase pressure drop [kPa]
P_{rt}	Turbulent Prandtl number
Re	Reynolds number
Tr	Temperature [K]
v	Velocity [m/s]
W	Tube width [m]
x	Dryness fraction or mass quality

Greek symbols

α	Volume fraction
σ	Prandtl coefficient
ρ	Density [kg/m^3]
μ	Dynamic viscosity [kg/m s]
μ_t	Turbulent viscosity [kg/m s]

Subscripts

1ph	Single-phase
2ph	Two-phase
d	Dispersion
dr	Drift
drag	Drag
e	Equivalent
g	Gas
k	Any phase
l	Liquid
m	Mixture
p	Secondary phase
pq	Relative phase
q	Primary phase
T	Total
t	Time

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AUTHORSHIP CONTRIBUTIONS

Authors equally contributed to this work.

DATA AVAILABILITY STATEMENT

The authors confirm that the data that supports the findings of this study are available within the article. raw data that support the finding of this study are available from the corresponding author, upon reasonable request.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

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