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Performance Comparison of Different Microfine Tubes During Flow Boiling of Azeotropic Refrigerants

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Nomenclature

Latin Symbols

A

A_C [m²] cross section area;

A_H [m²] heat transfer area;

B

Bo Boiling number;

Bon Bond number;

C

C Chisholm coefficient;

c_p [J/kgK] specific heat;

C_0 distribution parameter;

D

D [m] tube diameter;

D_h [m] hydraulic diameter;

D_r [m] root diameter;

$\left(\frac{dp}{dz}\right)$ [Pa/m] pressure gradient;

E

e [J/kg] total energy;

F

Fr Froude number;

f friction number;

G

G [kg/m²s] mass flux;

G^* [kg/m²s] apparent mass flux;

Ga Galileo number;

g [m/s²] gravity acceleration;

H

h [W/m²K] heat transfer coefficient;

I

i [J/kg] enthalpy;

i_{lv} [J/kg] phase change enthalpy;

J

j [m/s] volumetric flux;

J [m/s] superficial velocity;

Ja Jakob number;

K

k [J/kg] kinetic energy;

k	$[W/mK]$	thermal conductivity;
L		
L	$[m]$	tube length;
La		Laplace number;
M		
M	$[kg]$	mass;
MRD		mean relative deviation;
$MARD$		mean absolute relative deviation;
N		
N		recorded values;
N_p		number of data points;
Nu		Nusselt number;
P		
P	$[W]$	power provided by the electric evaporator;
Pr		Prandtl number;
p		pressure;
Q		
$\dot{Q}$	$[W]$	heat exchanged;
Q	$[m^3/s]$	volume flow rate;
q	$[W/m^2]$	heat flux;
R		
r		proration factor;
R_b	$[J/kg]$	mechanical energy dissipation;
S		
S	$[m^2]$	heat transfer surface area;
$\bar{s}$		average slip ratio;
s	$[J/kgK]$	entropy;
T		
T	$[K]$	temperature;
t	$[m]$	thickness;
U		
U	$[m/s]$	local velocity;
$\bar{U}$	$[m/s]$	cross-section averaged velocity;
u	$[J/kg]$	internal energy,
V		
V_{gj}	$[m/s]$	drift velocity;
V	$[m^3]$	volume;
v	$[m^3/kg]$	specific volume;
$\bar{v}$	$[m^3/kg]$	average specific volume;
$\bar{v}_m$	$[m^3/kg]$	momentum weighted average specific volume;

W*We* Weber number;***X****X* Lockhart-Martinelli parameter $\bar{x}$ average mass quality; $\bar{x}_v$ average volumetric quality;30% percentage number of points between $\pm 30\%$ deviation lines;**Greek Symbols** α local void fraction; $\bar{\alpha}$ cross-section averaged void fraction; ΔT [K] temperature difference; Δp [Pa] pressure drop; Δx quality variation; δ [m] liquid film thickness; Γ [kg/s] mass flow rate; λ [m/s] mass transfer coefficient; μ [Pas] viscosity; Ω [m²] cross-sectional area; ϕ two-phase multiplier; π [J/kg] potential energy; ψ enhancement factor; ρ [kg/m³] density; $\bar{\rho}_m$ [kg/m³] momentum weighted average specific density; $\bar{\rho}^*$ [kg/m³] photographic density; σ [N/m] surface tension; τ [Pa] shear stress; θ [degree] angle subtended by the control volume;**Subscripts***A* annular flow;*a* accelerative component;*b* bulk;*bp* bubble;*c* correlation;*cr* critical;*exp* experimental;*eq* equivalent;*eva* evaporator;

<i>f</i>	frictional component;
<i>film</i>	liquid film;
<i>gj</i>	superficial vapor velocity;
<i>g</i>	gas phase (gas-alone);
<i>go</i>	gas-only;
<i>g</i>	gravitational component;
<i>i</i>	inlet;
<i>l</i>	liquid phase (liquid-alone);
<i>lo</i>	liquid-only;
<i>mf</i>	micro-fin;
<i>ml</i>	mean logarithmic;
<i>nb</i>	nucleate boiling;
<i>out</i>	outlet;
<i>pool</i>	pool boiling;
<i>r</i>	reduced;
<i>ref</i>	reference;
<i>sat</i>	saturation;
<i>sc</i>	subcooled-saturation
<i>sub</i>	subcooled;
<i>theo</i>	theoretical;
<i>tp</i>	two-phase;
<i>w</i>	water;
<i>wall</i>	wall;

Abstract

The purpose of this thesis is to get familiar with the two-phase flow theory, heat transfer coefficient and frictional pressure drop per unit length in the process of flow boiling in horizontal tubes by using the experimental data which is obtained in the labs of the Energy Department of Politecnico di Milano. It also focuses on the behavior on the microfine. In this sense, smooth tube and four different types of microfine tubes are taken into consideration. The microfine tubes are named as Tube V, Tube W, Tube VA and Tube HVA. Each have their own unique fin type and fin alignment.

Nowadays, two-phase vapor-liquid flows are commonly encountered in many industrial and engineering applications such as heat exchangers, oil and gas exploitation, refrigeration and air conditioning systems, environmental control and life support systems and pipeline network systems. Therefore, the behavior of fluids in terms of heat transfer coefficient and pressure drop have a significant importance. In this respect, many correlations have been proposed throughout the history both for heat transfer coefficient and pressure drop.

In the process of theoretical calculation, Kandlikar, Shah, Fang, Saitoh, Gungor & Winterton models are used for heat transfer coefficient while Muller-Steinhagen and Heck, Gronnerud, Friedel, Chen, Tran and Haraguchi methods are used for frictional pressure drop per unit length. After the frictional pressure drop per unit length is calculated, the behavior of both heat transfer coefficient and frictional pressure drop per unit length are considered as a function of mass flux and mean quality. As it can be seen later in this thesis, they provide a linear increase with increasing mass flux and mean quality.

An error band of $\pm 30\%$ is checked for each method by using mean absolute relative deviation (MARD) and mean relative deviation (MRD). Then the results of experimental and theoretical values are compared by using parity plots for each type of tube. Process is repeated for each method mentioned above. By making this comparison, the best method for each type of microfine tube is obtained both for heat transfer coefficient and pressure drop per unit length. All of these methods are proposed by using their own set of data. In other saying, they are mostly valid for a specific data set. That is why each microfine tube has a different method that represents the best result.

K E Y W O R D S :

Two-Phase Flow, Boiling, Microfine Tubes, Heat Transfer Coefficient, Kandlikar, Shah, Fang, Saitoh, Gungor & Winterton, Frictional Pressure Drop, Muller-Steinhagen and Heck, Gronnerud, Friedel, Chen, Tran, Haraguchi

Chapter 1

1 Introduction

Nowadays, technology has been developing day by day and micro-fin tubes are also part of this improvement. The micro-fin tubes are widely used in commercial air conditioning applications, heat pump, pipeline network systems and heat exchanger [1, 2]. The calculation of pressure drops and heat transfer coefficients in two-phase channel are significant consideration in order to increase efficiency of the system and save the cost. However, the development of micro-fin tubes of performance is still required in order to have high efficiency. In theory, two-phase flow is synchronically flow of two any different phases such as solid, liquid and gas. The two stages may be of diverse component because of phase changing due to evaporation and condensation of a single fluid [3].

In the literature, there are lots of research papers about the calculation of the pressure gradient and heat transfer coefficient due to importance of the topic. Many correlations and models were recommended in order to calculate the frictional pressure drop and heat transfer coefficient in the system.

The methods developed for pressure drop calculation can be divided in two groups, first one is homogeneous, second one is separated approaches. The calculation of frictional pressure gradient can be made by using these methods. The majority of correlations introduced based on two-phase frictional multipliers recommended by Lockhart and Martinelli (1949) and the relation of multipliers with Chisholm (1967) [4]. Also, there are lots of difference models and correlations in the literatures such as Friedel, Chen, Tran, Gronnerud and Haraguchi, but results may be changeable depends on the experimental data base. Sometimes, correlations do not work out comparing with the other correlations because of this reason it is not easy to say that this correlation is always the best one. In addition, the calculation of heat transfer coefficient is an important consideration in order to reduce the cost and avoid the drastic results due to the design of evaporators, boilers and other two-phase process equipment [5]. In recent years, many correlations and models are developed in order to calculate the heat transfer coefficient such as Kandlikar, Shah, Gungor and Winterton, Saitoh and Fang.

According to experimental studies for two-phase flow boiling ensured the real values of heat transfer coefficients for several fluids under specific ranges of operating conditions. These are the foundations for the development and testing of any correlation. In order to improve a general correlation, it is primary to have a large data base by using different fluids with an extensive range of operation conditions such as mass flux, heat flux, pressure, quality and tube diameter. In the absence of a comprehensive data bank in the literature, every researcher wants to work in this topic has to keep in mind this task before beginning any correlation work [5]. The purpose of this thesis is to get familiar with the two-phase flow and to see the results of different methods on both heat transfer coefficient and frictional pressure drop per unit length in the process of flow boiling for horizontal tubes. Also, it focuses on the behavior on the microfine. In this sense, smooth tube and four different types of microfine tubes are taken

into consideration. In this sense, there exists “Smooth Tube, Tube V, Tube W, Tube VA and Tube HVA.

			Smooth Tube	Tube V	Tube W	Tube VA	Tube HVA
Outer Diameter	D_{out}	[m]	0,00952	0,00952	0,00952	0,00952	0,00952
Inner Diameter	D_{in}	[m]	0,00892	0,00892	0,00884	0,00892	0,00862
Wall Thickness	t	[m]	0,0003	0,0003	0,00034	0,0003	0,00045
Length of Pressure Drop	$L_{\Delta P}$	[m]	2,6	2,6	2,6	2,6	2,6
Length of Heat Transfer	$L_{heat-transfer}$	[m]	2,24	2,24	2,24	2,24	2,24
Fin (Type 1)							
Number of Fin		[-]	-	60	65	27	41
Height		[m]	-	0,0002	0,00015	0,00023	0,0002
Apex Angle		[degree]	-	53	90	40	40
Base		[m]	-	1,99E-04	3,00E-04	1,67E-04	1,46E-04
Helix Angle		[degree]	-	18	25	18	18
Fin (Type 2)							
Number of Fin		[-]	-	-	-	27	41
Height		[m]	-	-	-	0,00016	0,00017
Apex Angle		[degree]	-	-	-	40	40
Base		[m]	-	-	-	1,16E-04	1,24E-04
Helix Angle		[degree]	-	-	-	18	18

Table 1: Tube and Fin Properties

All of the tubes are considered in *counter flow* arrangement except Tube V. Tube V is examined both for *counter flow* and *parallel flow*. Some of the tubes contains one type of microfine while the others have two types of microfine on the inner wall of the circular cross-section. Please keep in mind that all of the tubes have their own unique fin type and alignment. The properties of tubes and fins are given in Table 1.

The test section mainly consists of three loops which are the refrigerant circuit, the demineralized water circuit and the cooling water circuit. The main components of the refrigerant circuit, whose schematic layout is reported in Figure 9, are: the boiler, the flow metering sections, the mixer, the test section, the visualization apparatus, the condenser, the variable speed gear pump and the filter dryer. For more of a detail please see part “Experimental Setup”. The data set used through the thesis is taken during the flow boiling in a horizontally oriented tube for fluid R22. Each data set is the average of 10-time measurement.

As mentioned earlier in this section, Kandlikar, Shah, Fang, Saitoh, Gungor & Winterton models are used for heat transfer coefficient. On the other hand, Muller-Steinhagen and Heck, Gronnerud, Friedel, Chen, Tran and Haraguchi methods are used for frictional pressure drop per unit length. The comparison between the predicted data and the experimental data is performed by using parity plots for that shows the distribution of the data on a 45° line together with $\pm 30\%$ deviation lines.



Chapter 2

2 Two-phase Flow Theory

Two-phase flow might be thought as the flow of two phases together out of three existing phases which are solid, liquid and gas. In this sense, two-phase flow made of liquid and vapor is taken into account.

2.1 Basic Quantities

In this part of the thesis, the typical parameters that defines the two-phase flow are presented.

2.1.1 Mass Flow Rate

The amount of mass passing through a surface per unit time is called mass flow rate.

$$\Gamma = \Gamma_g + \Gamma_l \quad \text{Eqn 1}$$

where; Γ [kg/s] total mass flow rate,
 Γ_g [kg/s] gas phase mass flow rate,
 Γ_l [kg/s] liquid phase mass flow rate.

2.1.2 Volume Flow Rate

Volume of a fluid passing through a surface for unit time.

$$Q = Q_g + Q_l \quad \text{Eqn 2}$$

where; Q [m³/s] volume flow rate,
 Q_g [m³/s] gas phase volume flow rate,
 Q_l [m³/s] liquid phase volume flow rate.

the density links the mass flow rate and the volume flow rate;

$$Q = \frac{\Gamma}{\rho} \quad \text{Eqn 3}$$

where; ρ [kg/m³] density.

2.1.3 Mass Flux

It is defined as mass flow rate per unit cross-sectional area.

$$G = \frac{\Gamma}{\Omega} \quad \text{Eqn 4}$$

where; G [kg/m²s] mass flux,
 Ω [m²] cross-sectional area.

2.1.4 Apparent Mass Flux

It is the mass flux of the i^{th} phase flowing alone.

$$G_g^* = \frac{\Gamma_g}{\Omega} \quad \text{and} \quad G_l^* = \frac{\Gamma_l}{\Omega} \quad \text{Eqn 5}$$

where; G_g^* [kg/m²s] gas phase apparent mass flux,
 G_l^* [kg/m²s] gas phase apparent mass flux.

2.1.5 Superficial Velocity

It is the apparent velocity of the i^{th} phase flowing alone.

$$J_g^* = \frac{Q_g}{\Omega} \quad \text{and} \quad J_l^* = \frac{Q_l}{\Omega} \quad \text{Eqn 6}$$

where; J_g^* [m/s] gas phase apparent velocity,
 J_l^* [m/s] liquid phase apparent velocity.

2.1.6 Local Void Fraction

Normally, in considering void fraction, the time-average value (taken over a long period of time) is used, but it should be appreciated that void fraction is fluctuating with time and instantaneous values are also of interest. At a given point in the flow, the local fluid is either gas or one of the other phases. The probability of finding gas at a given point may be determined using local probes and is referred to as the local void fraction.

$$\alpha = \frac{d\Omega_g}{d\Omega} \quad \text{and} \quad 1 - \alpha = \frac{d\Omega_l}{d\Omega} \quad \text{Eqn 7}$$

where; α [-] local void fraction,
 $d\Omega$ [m²] area of the cross-section element,
 $d\Omega_g$ [m²] proportion of the $d\Omega$ occupied by the gas phase,
 $d\Omega_l$ [m²] proportion of the $d\Omega$ occupied by the liq. phase.

2.1.7 Cross-Section Averaged Void Fraction

The fraction of the channel cross-sectional area that is occupied by the gas phase. This void fraction is known as the cross-sectional void fraction.

$$\bar{\alpha} = \frac{\Omega_g}{\Omega} \quad \text{and} \quad 1 - \bar{\alpha} = \frac{\Omega_l}{\Omega} \quad \text{Eqn 8}$$

where; $\bar{\alpha}$ [-] local void fraction,
 Ω_g [m²] proportion of the Ω occupied by the gas phase,
 Ω_l [m²] proportion of the Ω occupied by the liq. phase.

2.1.8 Local Velocity

The local velocity of the blade elements residing near to outboard part of a normally operating propeller is in the high subsonic range. Consequently, these elements should be corrected for compressibility effects. These effects call for correlations of the drag that differ from those for lift. The corrections are applied to the drag and lift coefficients.

$$U_g = \frac{dQ_g}{d\Omega_g} = \frac{dQ_g}{\alpha d\Omega} \quad \text{Eqn 9}$$

$$U_l = \frac{dQ_l}{d\Omega_l} = \frac{dQ_l}{(1 - \alpha) d\Omega} \quad \text{Eqn 10}$$

where; U_g [m/s] gas phase local velocity,
 U_l [m/s] liquid phase local velocity.

2.1.9 Cross-Section Averaged Velocity

It is obtained by integrating the local velocity over the cross-sectional area. The fluid velocity in a pipe changes from zero at the surface because of the no-slip condition to a maximum at the pipe center. In fluid flow, it is convenient to work with an average velocity, which remains constant in incompressible flow when the cross-sectional area of the pipe is constant.

$$\bar{U}_g = \frac{Q_g}{\bar{\alpha} \Omega} \quad \text{and} \quad \bar{U}_l = \frac{Q_l}{(1 - \bar{\alpha}) \Omega} \quad \text{Eqn 11}$$

where; $\bar{U}_g$ [m/s] gas phase averaged velocity,
 $\bar{U}_l$ [m/s] liquid phase averaged velocity.

2.1.10 Actual Mass Flux

The term flux refers to the unit cross section area. It means that, the mass of a given mixture component passing per unit time across any surface.

$$G_g = \rho_g \bar{U}_g = \rho_g \frac{Q_g}{\bar{\alpha} \Omega} = \frac{\Gamma_g}{\bar{\alpha} \Omega} = \frac{G_g^*}{\bar{\alpha}} \quad \text{Eqn 12}$$

$$G_l = \rho_l \bar{U}_l = \rho_l \frac{Q_l}{\alpha \Omega} = \frac{\Gamma_l}{\alpha \Omega} = \frac{G_l^*}{\alpha} \quad \text{Eqn 13}$$

where; G_g [kg/m²s] gas phase max flux,
 G_l [kg/m²s] liquid phase mass flux.

2.1.11 Local Slip-Ratio

In two-phase fluid flow it is convenient to use the slip ratio. The slip ratio in two-phase flow is defined as the ratio of the velocity of the vapor phase to the velocity of the liquid phase.

$$s = \frac{U_g}{U_l} \quad \text{Eqn 14}$$

where; s [-] local slip ratio.

2.1.12 Average Slip Ratio

The slip ratio in two-phase flow is defined and showed before as the ratio of the average velocity of the vapor phase to the average velocity of the liquid phase.

$$\bar{s} = \frac{\bar{U}_g}{\bar{U}_l} \quad \text{Eqn 15}$$

where; $\bar{s}$ [-] average slip ratio.

2.1.13 Average Mass Quality

Quality is the mass fraction of vapor in a liquid/vapor mixture. In thermal equilibrium, the quality of a two-phase mixture is directly related to heat input and is sometimes called the thermodynamic quality.

$$\bar{x} = \frac{\Gamma_g}{\Gamma} \quad \text{and} \quad 1 - \bar{x} = \frac{\Gamma_l}{\Gamma} \quad \text{Eqn 16}$$

where; $\bar{x}$ [-] average mass quality.

2.1.14 Thermodynamic Average Quality

It can be used only for single-component mixtures (e.g. water with steam) and can take values <0 (for sub-cooled fluids) and >1 (for super-saturated vapors).

$$\bar{x}_{th} = \frac{i - i_l}{i_{lv}} \quad \text{Eqn 17}$$

where; i	[J/kg]	is the specific enthalpy of the mixture,
i_l	[J/kg]	specific enthalpy of the saturated mixture,
i_{lv}	[J/kg]	the specific enthalpy of vaporization,
$\bar{x}_{th}$	[-]	thermodynamic average quality.

2.1.15 Average Volumetric Quality

Volumetric quality is the volume fraction of vapor in a liquid/vapor mixture.

$$\bar{x}_v = \frac{Q_g}{Q} \quad \text{and} \quad 1 - \bar{x}_v = \frac{Q_l}{Q} \quad \text{Eqn 18}$$

where; $\bar{x}_v$ [-] average volumetric quality.

REMARKS:

1. Please note that $\bar{x}_{th} = \bar{x}$ only if the system is under thermodynamic equilibrium, ie, if the properties are uniform over the cross-section.
2. $0 \leq \bar{x} \leq 1$
3. $\bar{x}_v < 0$; refers to subcooled liquid
 $\bar{x}_v > 1$; refers to superheated vapor
4. Relation between $\bar{x}$ and $\bar{x}_v$ is given as follows;

$$\bar{x}_v = \frac{1}{1 + \frac{1-\bar{x}}{\bar{x}} \frac{\rho_g}{\rho_l}} \quad \text{Eqn 19}$$

5. Relation between $\bar{\alpha}$ and $\bar{x}$ is given as follows;

$$\bar{\alpha} = \frac{1}{1 + \bar{s} \frac{1-\bar{x}}{\bar{x}} \frac{\rho_g}{\rho_l}} \quad \text{Eqn 20}$$

6. Particular cases;
 - $\bar{x} = 0 \rightarrow \bar{x}_v = \bar{\alpha} = 0$
 - $\bar{x} = 1 \rightarrow \bar{x}_v = \bar{\alpha} = 1$
 - $\bar{s} = 1 \rightarrow \bar{x}_v = \bar{\alpha} \quad \forall \bar{x}$ and this is referred to as homogenous flow.
7. $\bar{x}_v$ and $\bar{\alpha}$ are both increase with increasing $\bar{x}$.

2.1.16 Photographic Density

It is the density of a fluid that at a certain time occupies an elementary portion of a duct. It is then the ratio of the mass contained in the element and its volume. There is generally intermittent flow pattern so, the photographic density varies in time. In this sense, it is better

to refer to a time average quantity. Referring to the whole cross-section, cross-section averaged photographic density is given by Eqn 21 as follows;

$$\bar{\rho}^* = \frac{dM}{dV} = \bar{\alpha}\rho_g + (1 - \bar{\alpha})\rho_l \quad \text{Eqn 21}$$

where; dV [m³] infinitesimal volume,
 dM [kg] infinitesimal mass inside dV ,
 $\bar{\rho}^*$ [kg/m³] photographic density.

2.1.17 Bulk Density

Bulk density is the density that would pertain to an equivalent single-phase fluid flowing with the same mass and volume flow rates of the two-phase mixture. Referring to the whole cross-section, cross-section averaged bulk density is given by Eqn 22, Eqn 23, Eqn 24.

To calculate the average bulk density $\bar{\rho}_b$ it is needed the average mass quality $\bar{x}$ and the densities of the two phases ρ_l and ρ_g . For evaporating or condensing flows of pure fluids (e.g. water), if thermodynamic equilibrium holds, ρ_l and ρ_g can be determined from the pressure (or temperature) whereas the average mass quality coincides with the thermodynamic quality, $\bar{x}_{th}$ and can be calculated from the enthalpy. In the less frequent case of non-equilibrium recourse has to be made to determine $\bar{x}$.

$$\bar{\rho}_b = \frac{\Gamma}{Q} \quad \text{Eqn 22}$$

$$\bar{\rho}_b = \bar{x}_v\rho_g + (1 - \bar{x}_v)\rho_l \quad \text{Eqn 23}$$

If $\bar{x}_v$ is replaced by its definition in Eqn 19, the new definition can be obtained as follows;

$$\bar{\rho}_b = \frac{1}{\frac{\bar{x}}{\rho_g} + \frac{1-\bar{x}}{\rho_l}} \quad \text{Eqn 24}$$

where; $\bar{\rho}_b$ [kg/m³] bulk density,
 ρ_g [kg/m³] gas phase density,
 ρ_l [kg/m³] liquid phase density.

2.1.18 Bulk Specific Volume

It is the reciprocal of blk density. It is defined as the number of cubic meters occupied by a one kilogram of a particular substance.

$$\bar{v}_b = \frac{1}{\bar{\rho}_b} = \bar{x}v_g + (1 - \bar{x})v_l \quad \text{Eqn 25}$$

where; $\bar{v}_b$ [m³/kg] bulk specific volume,
 v_g [m³/kg] gas phase specific volume,
 v_l [m³/kg] liquid phase specific volume.

2.2 Dimensionless Groups

The following dimensionless groups are introduced, in alphabetic order, here because of their useful importance. Dimensionless numbers reduce the number of variables that describe a system, thereby reducing the amount of experimental data required to make correlations of physical phenomena to scalable system [6]. Later on in this thesis, these dimensionless groups will have significant effect in the process of calculation for both pressure drop and heat transfer coefficients.

2.2.1 Boiling Number

Boiling number is basically the ratio between mass flux of vapor generated at the pipe surface to the total mass flux parallel to the pipe surface [7]. It is argued that it represents the stirring effects of the bubbles upon the flow the boiling number is used in convective boiling. Shah (1982) uses an empirical boiling number correction to the single-phase liquid heat transfer coefficient in order to predict the two-phase heat transfer. This approach avoids the introduction of a nucleate pool boiling term into the correlation. Boiling number has the following expression;

$$Bo = \frac{q}{G i_{lv}} \quad \text{Eqn 26}$$

where; Bo [-] Boiling number,
 q [W/m²] heat flux,
 i_{lv} [J/kg] specific enthalpy of vaporization.

2.2.2 Bond Number

The Bond number represents the ratio between order of magnitude of the buoyancy and the order of magnitude of the surface tension forces. A high Bond number indicates that the system is relatively unaffected by surface tension effects; a low number (typically less than one is the requirement) indicates that surface tension dominates. Intermediate numbers indicate a non-trivial balance between the two effects. It is also known as Eötvös number and it has the following formula;

$$Bon = \frac{g (\rho_l - \rho_g) D^2}{\sigma} \quad \text{Eqn 27}$$

where; Bon [-] Bond number,
 σ [N/m] surface tension,
 g [m/s²] gravitational acceleration.

2.2.3 Froude Number

the Froude number represents as the ratio between inertial and gravitational forces. For a single-phase flow, it is defined as in Eqn 28

$$Fr = \frac{U}{\sqrt{gD}} \quad \text{Eqn 28}$$

where; Fr [-] Froude number,
 g [m/s²] gravitational acceleration

There is also relation between Fr and enhancement factor, ψ . It may further be noted that for vertical pipes the pipe surface will always remain wetted until liquid deficiency occurs. Hence Fr is not required as similarity parameter for vertical pipes [7]. For liquid only, gas only, liquid alone and gas alone cases there exist the following equations;

$$Fr_{lo} = \frac{G}{\rho_l \sqrt{gD}} \quad \text{Eqn 29}$$

$$Fr_{go} = \frac{G}{\rho_g \sqrt{gD}} \quad \text{Eqn 30}$$

$$Fr_l = \frac{G (1 - \bar{x})}{\rho_l \sqrt{gD}} \quad \text{Eqn 31}$$

$$Fr_g = \frac{G \bar{x}}{\rho_g \sqrt{gD}} \quad \text{Eqn 32}$$

where; Fr_{lo} [-] liquid only Froude number,
 Fr_{go} [-] gas only Froude number,
 Fr_l [-] liquid alone (or liquid) Froude number,
 Fr_g [-] gas alone (or gas) Froude number.

2.2.4 Galileo Number

The Galileo number represents as the ratio between gravitational and viscous forces. It is useful in viscous flow and thermal expansion calculations. Galileo number is described as follows in Eqn 33;

$$Ga = \frac{g \rho^2 D}{\mu} \quad \text{Eqn 33}$$

where; Ga [-] Galileo number.

2.2.5 Jakob Number

It is the ratio between the sensible heat and the latent energy absorbed during liquid-vapor phase change. It is shown by Eqn 34;

$$Ja = \frac{c_p(T_{sat} - T_{sc})}{i_{lv}} \quad \text{Eqn 34}$$

where; Ja [-] Jakob number,
 c_p [J/kgK] constant pressure specific heat,
 T_{sat} [K] saturation temperature,
 T_{sc} [K] subcooling temperature,
 i_{lv} [J/kg] the specific enthalpy of vaporization.

2.2.6 Laplace Number

The Laplace number (La), also known as the Suratman number, is a dimensionless number used in the characterization of free surface fluid dynamics. It represents a ratio of surface tension to the momentum transport (especially dissipation) inside a fluid. It is shown as follows;

$$La = \frac{\sigma \rho L}{\mu^2} \quad \text{Eqn 35}$$

where; La [-] Laplace number,
 σ [N/m] surface tension,
 μ [pas] viscosity,
 L [m] length.

2.2.7 Nusselt Number

Nusselt number is the ratio of heat transferred by convection to the heat transferred by conduction. It shows how much is the heat is transferred due to fluid motion as compared to the heat transfer by fluid by the process of conduction. In addition, the physical interpretation of Nusselt number is the enhancement of heat transfer due to convection over conduction alone. If $Nu=1$, means that the fluid is stationary, and all heat transfer is by conduction. With $Nu>1$, the fluid motion enhances heat transfer by advection. The Nusselt number is of paramount importance because it contains the heat transfer coefficient

information. Indeed, one interpretation of Nusselt number is simply that of dimensionless heat transfer coefficient. In simplest terms, Nu is simply a dimensionless heat transfer coefficient. It has the following definition;

$$Nu = \frac{h D}{k} \quad \text{Eqn 36}$$

where; Nu [-] Nusselt number,
 h [W/m²K] heat transfer coefficient,
 k [W/mK] thermal conductivity.

2.2.8 Prandtl Number

It is the ratio of the momentum diffusivity to thermal diffusivity. For a single-phase flow, it is defined as it is shown in Eqn 37.

$$Pr = \frac{c_p \mu}{k} \quad \text{Eqn 37}$$

where; Pr [-] Prandtl number,
 c_p [J/kgK] constant pressure specific heat,
 k [W/mK] thermal conductivity.

Liquid and vapor Prandtl numbers are given in Eqn 38 and Eqn 39, respectively.

$$Pr_l = \frac{c_{pl} \mu_l}{k_l} \quad \text{Eqn 38}$$

$$Pr_v = \frac{c_{pv} \mu_v}{k_v} \quad \text{Eqn 39}$$

where; Pr_l [-] liquid Prandtl number,
 Pr_v [-] vapor Prandtl number,
 c_{pl} [J/kgK] constant pressure liquid specific heat,
 c_{pv} [J/kgK] constant pressure vapor specific heat,
 k_l [W/mK] liquid thermal conductivity,
 k_v [W/mK] vapor thermal conductivity.

2.2.9 Reynolds Number

The Reynolds number is the most significant dimensionless number in fluid mechanics. It is an input parameter for all forced flows and a criterion used for classification the laminar and turbulent flow regimes [8]. The Reynolds number is normally explicated in most research paper and it can be defined as the physical process with governs the fluid flows which are

advection and diffusion fluxes, it provides a clear understanding of fluid mechanics. Finally, the Reynolds number is the ratio between inertial and viscous forces. For a single-phase flow, it is defined as in Eqn 40.

$$Re = \frac{\rho U D}{\mu} = \frac{G D}{\mu} \quad \text{Eqn 40}$$

where; Re [-] Reynolds number,
 μ [kg/ms] viscosity,

Here there is an important distinction between only and alone cases. In other words, Reynolds number for liquid alone and Reynolds number for liquid only stands for different formulas. On the other hand, gas alone and gas only have different meanings. In liquid only case Reynolds number is defined as the liquid phase flows as the total mass flux of the mixture. Moreover, in gas only case Reynolds number is defined as the gas phase flows as the total mass flux. Reynolds number for liquid only case and Reynolds number for gas only case is defined in Eqn 41 and Eqn 42, respectively.

$$Re_{lo} = \frac{G D}{\mu_l} \quad \text{Eqn 41}$$

$$Re_{go} = \frac{G D}{\mu_g} \quad \text{Eqn 42}$$

where; Re_{lo} [-] liquid only Reynolds number,
 Re_{go} [-] gas only Reynolds number.

In liquid alone and gas alone cases, liquid and gas phases flow with their own mass flow rate. For this situation we have the following two equations;

$$Re_l = \frac{G (1 - \bar{x}) D}{\mu_l} \quad \text{Eqn 43}$$

$$Re_g = \frac{G \bar{x} D}{\mu_l} \quad \text{Eqn 44}$$

where; Re_l [-] liquid alone (or liquid) Reynolds number,
 Re_g [-] gas alone (or gas) Reynolds number.

2.2.10 Weber Number

It is the ratio between the momentum transport inside and the surface tension. It is also a dimensionless group occurring in the analysis of bubble formation and some other interfacial phenomena and the Weber number indicates whether the kinetic or the surface tension energy is dominant. Weber number can be expressed as;

$$We = \frac{G^2 D}{\sigma \rho} \quad \text{Eqn 45}$$

where; We [-] Weber number.

2.3 Two-Phase Flow Models

Two-phase flow models are divided into groups as follows;

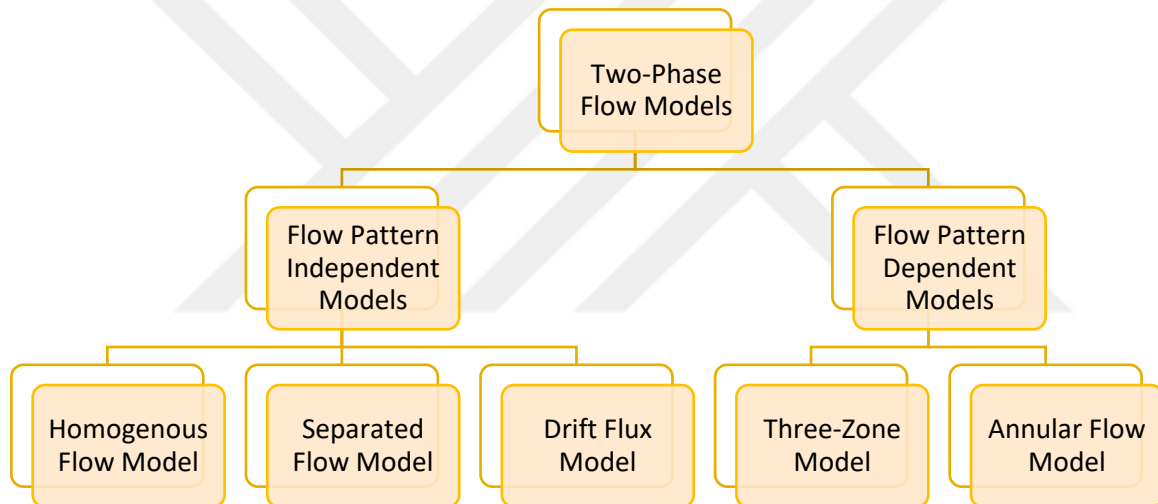


Figure 1: Two-Phase Flow Models

2.3.1 Flow Pattern Independent Models

2.3.1.1 Homogenous Flow Model

The easiest technique in order to analyze two-phase (or multiphase) flows can be provided by the homogeneous flow model. Liquid and vapor phases flow at the same velocity (slip ratio=1) in the homogenous model. Eventually, the homogenous model has been identified as the zero-slip model. The homogeneous model takes into consideration the two-phase flow as a single-phase flow having average fluid properties that depend on mixture quality [9]. In this way, the frictional pressure drop is determined by considering a constant frictional coefficient between the inlet and outlet sections of the tube. The void fraction for homogenous model can be written as follows;

$$\alpha_m = \frac{1}{1 + \left(\frac{1-x}{x}\right) \left(\frac{\rho_g}{\rho_l}\right)} \quad \text{Eqn 46}$$

When the ratio between liquid density and vapor density is much larger than 1, the void fraction based on the homogeneous model (α_m) goes up sharply once the mass quality (x) increases even slowly above zero. The relation between the void fraction and the mass quality is shown in Figure 2. In the opposite case, the ratio between liquid density and vapor density is near to 1 (critical state), the void fraction based on the homogeneous model (α_m) approaches the mass quality (x) and the homogeneous model is practicable at this case.

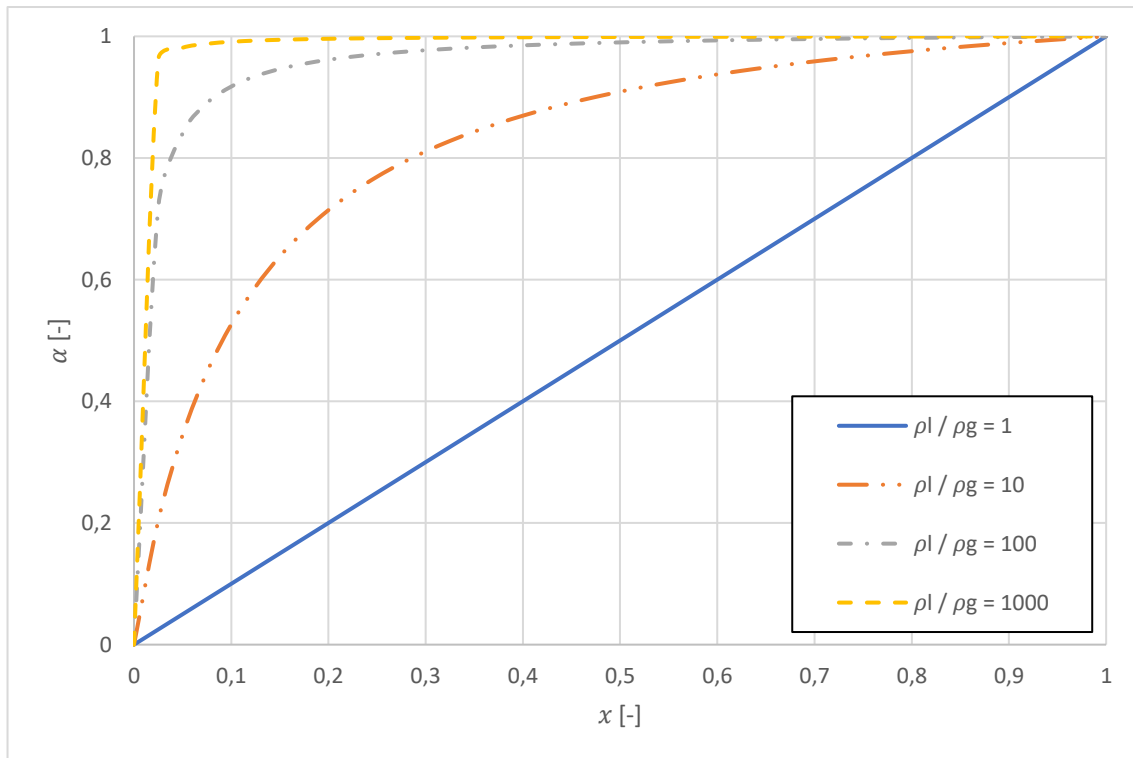


Figure 2: Void Fraction as a Function of Mass Quality for Various Density Ratios

From the Figure 2, the result is obviously clear that the void fraction is a function of mass quality for various density ratios. It is shown according to four different density ratios such as 1, 10, 100 and 1000.

2.3.1.2 Separated Flow Model

Separated flow model can be shown as it is in the Figure 3. In this schematic the flow is imagined as if it is separated into two groups flowing on its own tubes. The main idea of the separated flow model is to examine the fluid in liquid and vapor phases differently. The Figure 3 makes a clear understanding of the model.

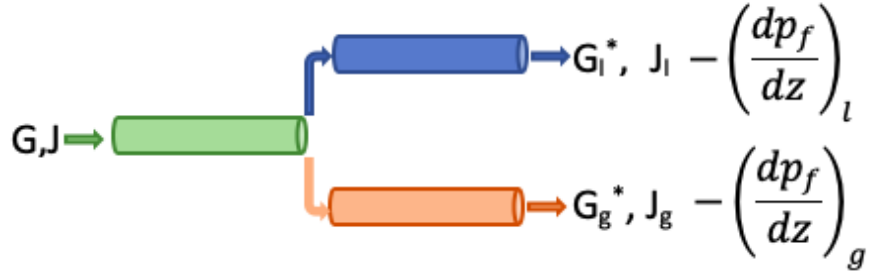


Figure 3: Schematic of Separated Flow

Separated flow model can also be divided into two sub-groups based on the methods used by the scientists. The first one is ϕ_l^2, ϕ_g^2 based models and the second sub-group is called as ϕ_{lo}^2, ϕ_{go}^2 based models. ϕ_l^2 can be defined as the ratio of the two-phase frictional pressure gradient to the frictional pressure gradient that could consist if the liquid phase is supposed to flow alone and it is named as two-phase frictional multiplier for liquid alone. ϕ_g^2 can be identified as the ratio of the two-phase frictional pressure gradient to the frictional pressure gradient that could consist if the gas phase is supposed to flow alone and it is named as two-phase frictional multiplier for gas alone. Moreover, ϕ_{lo}^2 is identified as the ratio of the two-phase frictional pressure gradient that could consist if the total mixture is supposed to be liquid and it is named as two-phase frictional multiplier for liquid only. Finally, ϕ_{go}^2 can be identified as the ratio of the two-phase frictional pressure gradient that could consist if the total mixture is supposed to be gas and it is named as two-phase frictional multiplier for gas alone [4] [10]. As it is mentioned before, this paper will deal with the second sub-group in order to calculate the pressure drop. In addition, Lockhart and Martinelli developed a correlation which the slip ratio denoted as a function of the given quantity, $s = f(X)$.

$$X^2 = \frac{\left(\frac{dp_f}{dz}\right)_l}{\left(\frac{dp_f}{dz}\right)_g} \quad \text{Eqn 47}$$

where; X^2 [-] Lockhart-Martinelli parameter,
 $\left(\frac{dp_f}{dz}\right)_l$ [Pa/m] single phase liquid pressure gradient,
 $\left(\frac{dp_f}{dz}\right)_g$ [Pa/m] single phase gas pressure gradient.

In the Eqn 47, it is written the frictional pressure gradient of the liquid and vapor as flow alone. Usually, the dp_f/dz is determined from beginning of the friction factor. It can be examined two different approaches such as Fanning and Dimensionless analysis and shown below, respectively. Fanning's definition:

$$f = \frac{\tau_w}{\frac{1}{2}\rho U^2} \quad \text{Eqn 48}$$

where; f [-] Fanning fraction factor,
 τ_w [Pa] wall shear stress.

According to Eqn 48, it is possible to identify the wall shear stress τ_w as a function of the frictional pressure gradient:

$$\tau_w = -\frac{\Omega}{S} \frac{dp_f}{dz} \quad \text{Eqn 49}$$

where; Ω [m²] cross sectional pipe area,
 S [m] perimeter.

In the Eqn 49, S represents the duct's perimeter. It can be shown like Ω/S for a cylindrical pipe and it is equal to $D/4$.

Dimensionless analysis:

$$f = ARe^{-n} \quad \text{Eqn 50}$$

when:

$$\begin{cases} \text{Poiseuille:} & A = 16 \quad n = 1, \text{ laminar flow } Re < 2300 \\ \text{Blasius:} & A = 0.079 \quad n = 0.2, \text{ turbulent flow } Re > 2300 \end{cases} \quad \text{Eqn 51}$$

The final version of the Eqn 47 is written below:

$$X^2 = \frac{\rho_v}{\rho_l} \left(\frac{\mu_l}{\mu_v} \right)^n \left(\frac{1-x}{x} \right)^{2-n} \quad \text{Eqn 52}$$

To sum up, depending on the variation of the Reynolds number, the Lockhart-Martinelli parameter has an important role in order to calculate the frictional pressure gradient as shown above.

2.3.1.3 Drift-Flux Model

The method of drift-flux theory is developed by Zuber and Findlay in 1965, and later it is modified by Wallis (1969), Ishii (1977) and finally Joshi et al. in 1990, respectively. The drift-flux is focused on the effect of non-uniformity of the holdup distribution across a cross-sectional area with the local relative velocity between liquid and vapor phases. In order to make the drift-flux model simple, many authors have applied this theory to provide the gas

holdup over a large range of operating conditions and column diameters. Latterly, the most of experimental and modeling activities have been tested for small diameter columns, and the equations for the drift-flux model have been improved for vertical upward two-phase flows in small diameters range from 25 to 50mm [11]. Finally, the kinematic relation of the drift-flux model is shown as follows:

$$\frac{\langle j_g \rangle}{\langle a \rangle} = \langle \langle V_g \rangle \rangle = C_0 \langle j \rangle + \langle \langle V_{gj} \rangle \rangle \quad \text{Eqn 53}$$

where; j_g [m/s] superficial gas velocity,
 V_g [m/s] gas velocity,
 j [m/s] mixture volumetric flux,
 C_0 [-] distribution parameter,
 V_{gj} [m/s] drift velocity,
 a [-] local void fraction.

In addition, the parenthesis of $\langle \rangle$ and $\langle \langle \rangle \rangle$ show the area-averaged and void-weighted area-averaged mean quantities [12]. The distribution parameter and drift velocity can be written as follows:

$$C_0 = \frac{\langle aj \rangle}{\langle a \rangle \langle j \rangle} \quad \text{Eqn 54}$$

$$\langle \langle V_{gj} \rangle \rangle = \frac{\langle \alpha V_{gj} \rangle}{\langle a \rangle} \quad \text{Eqn 55}$$

where; V_{gj} [m/s] local drift velocity with $V_{gj} = V_g - j$

According to Ishii, the distribution parameter can be affected by the density ratio and Reynolds number and shown below:

$$C_0 = C_\infty(Re_l) - [C_\infty(Re_l) - 1] \sqrt{\frac{\rho_g}{\rho_l}} \quad \text{Eqn 56}$$

where C_∞ is the asymptotic value of the distribution parameter calculated by Reynolds number and the density ratio between liquid and vapor. The liquid Reynolds number is shown in Eqn 41. When subcooled boiling happens, the distribution parameter goes down against zero as the void fraction approaches zero because of the bubbles concentrates close to the channel wall. Therefore, Ishii is developed the correlation for subcooled boiling in 1977. It is written below:

$$C_0 = \left[C_\infty - (C_\infty - 1) \sqrt{\frac{\rho_g}{\rho_l}} \right] (1 - e^{-18(\alpha)}) \quad \text{Eqn 57}$$

The following correlation in order to determine the drift velocities in different two-phase flow regimes can be examined three groups for small diameter tubes.

Bubbly flow:

$$\langle\langle V_{gj} \rangle\rangle = \sqrt{2} \left[\frac{\sigma g \Delta \rho}{\rho_l^2} \right]^{1/4} (1 - \langle a \rangle)^{1.75} \quad \text{Eqn 58}$$

Slug flow:

$$\langle\langle V_{gj} \rangle\rangle = 0.35 \sqrt{\frac{g D_H \Delta \rho}{\rho_l}} \quad \text{Eqn 59}$$

Churn flow:

$$\langle\langle V_{gj} \rangle\rangle = \sqrt{2} \left[\frac{\sigma g \Delta \rho}{\rho_l^2} \right]^{1/4} \quad \text{Eqn 60}$$

where; σ [N/m] surface tension,
 g [m/s²] gravitational acceleration,
 D_H [m] hydraulic diameter,
 $\Delta \rho$ [kg/m³] density difference.

On the other hand, when the liquid recirculation has a low mixture volumetric flux state, that results in a totally different drift velocity and void fraction distribution by comparing with those in small-diameter tubes [12]. Afterwards, Kataoka and Ishii are improved a drift-flux model in order to calculate the pool void fraction under stagnant liquid flow states. Consequently, depending on the channel diameter, pressure, superficial vapor velocity and fluid physical properties, Correlation of the drift velocity can be written as follows:

For $N_{uf} < 2.25 \times 10^{-3}$,

$$\langle\langle V_{gj}^+ \rangle\rangle = 0.0019 (D_H^+)^{0.809} \left(\frac{\rho_g}{\rho_l} \right)^{-0.157} N_{uf}^{-0.562}, \text{ for } D_H^+ \leq 30 \quad \text{Eqn 61}$$

$$\langle\langle V_{gj}^+ \rangle\rangle = 0.030 \left(\frac{\rho_g}{\rho_l} \right)^{-0.157} N_{uf}^{-0.562}, \text{ for } D_H^+ \geq 30 \quad \text{Eqn 62}$$

For $N_{uf} > 2.25 \times 10^{-3}$,

$$\langle\langle V_{gj}^+ \rangle\rangle = 0.92 \left(\frac{\rho_g}{\rho_l} \right)^{-0.157}, \text{ for } D_H^+ \geq 30 \quad \text{Eqn 63}$$

where N_{uf} , $\langle\langle V_{gj}^+ \rangle\rangle$ and D_H^+ are defined like the viscosity number, non-dimensional drift velocity and non-dimensional hydraulic diameter, respectively, that are shown below:

$$N_{uf} = \frac{\mu_f}{\left(\rho_l \sigma \sqrt{\frac{\sigma}{g \Delta \rho}} \right)^{1/2}}, \quad \langle\langle V_{gj}^+ \rangle\rangle = \frac{V_{gj}}{\left(\frac{\sigma g \Delta \rho}{\rho_l^2} \right)^{1/4}}, \quad D_H^+ = \frac{D_H}{\sqrt{\frac{\sigma}{g \Delta \rho}}} \quad \text{Eqn 64}$$

where μ_f and D_H are shown as liquid dynamic viscosity and channel hydraulic diameter, respectively. Hibiki and Ishii are provided a new drift velocity correlation from stagnant liquid flow to high flow states for large-diameter tubes.

$$\langle\langle V_{gj}^+ \rangle\rangle = \langle\langle V_{gj,B}^+ \rangle\rangle e^{-1.36 \langle j_g^+ \rangle} + \langle\langle V_{gj,P}^+ \rangle\rangle (1 - e^{-1.36 \langle j_g^+ \rangle}) \quad \text{Eqn 65}$$

where $\langle\langle V_{gj,B}^+ \rangle\rangle$ and $\langle\langle V_{gj,P}^+ \rangle\rangle$ are determined by Eqn 58, Eqn 61 and Eqn 63, respectively. $\langle j_g^+ \rangle$ is defined as the non-dimensional superficial vapor velocity shown below:

$$\langle j_g^+ \rangle = \frac{\langle j_g \rangle}{\left(\frac{\sigma g \Delta \rho}{\rho_l^2} \right)^{1/4}} \quad \text{Eqn 66}$$

2.3.2 Flow Pattern Dependent Models

2.3.2.1 Three-Zone Model and Pattern-Oriented Model

Kattan developed a new pattern-oriented model in order to calculate the local heat transfer coefficient for two-phase flow boiling of pure ammonia in a horizontal pipe based on huge database. According to Kattan, the different heat transfer models exist, such as the superposition, the enhancement and the asymptotic models [13]. However, their range is limited, basically only to annular flows. Finally, the method proposed by Kattan is significant progress compared to those previous methods. In addition, the new flow pattern-based model is improved by using the three-zone model according to Thome et al. for the annular flow region [14]. The intermittent flow regime is a combination of the isolated bubble flow regime and coalescing bubble flow regime. These two methods are modified in order to fit the high-aspect ratio microchannel and a new flow pattern transition. Moreover, the three-zone model is used to calculate the Nusselt number. The calculation of the Nusselt number for liquid and vapor slugs are modified. This method works better when the film thickness has a minimum

value. The improving flow equations, that would guide to non-sensical results in small channels, are switched by fully developed. All the related equations are shown below:

$$Nu_{lam} = 4.36 \quad \text{Eqn 67}$$

$$Nu_{turb} = \frac{Pr \frac{f}{8} (Re - 1000)}{1 + 12.7 \left(\frac{f}{8}\right)^{0.5} (Pr^{2/3} - 1)} \quad \text{Eqn 68}$$

$$\text{For } Re \geq 1500. \quad \text{Eqn 69}$$

To sum up, the liquid film heat transfer is used. In order to avoid computational problems, Dupont improved the method by adding one nanometer in the subtraction when $\delta_0 \cong \delta_{end}$. It is shown below:

$$\alpha_{film} = \frac{\lambda_1}{\delta_0 - \delta_{end} + 1 \times 10^{-9}} \ln \left(\frac{\delta_0}{\delta_{end}} \right) \quad \text{Eqn 70}$$

2.3.2.2 Annular Flow Model

Cioncolini and Thome developed the annular flow method for non-circular tubes in 2011. It equilibrates forces in the channel to determine the radial film thickness in a circular channel and this film has to be redistributed to the rectangular perimeter in order to keep the same area ratio between the shapes and to preserve the liquid cross sectional area. [14]. After that, the equivalent diameter can be used. In short, the annular flow model can be used at vapor qualities higher than the transition zone. The vapor quality transition zone is identified below:

$$x_{CB-AF} \pm \frac{x_{exit}}{5} \quad \text{Eqn 71}$$

where; x [-] vapor quality.

The two methods which are three-zone and annular flow model combined in order to form the first flow pattern-based estimation method for flow boiling in micro channels with a smooth transition in the heat transfer coefficient made by the inclusion of a buffer zone, thenceforward the three-zone (α_{3Z}) and annular (α_{AF}) do not mate in value at the CB-AF transition [14]. According to all data, the CB-AF transition zone are processed by the three-zone model. The x_{exit} is determined from an energy balance from inlet to outlet of the evaporator. The pressure losses and the effect of flashing are negligible. The proration factor r is shown in Eqn 72.

$$r = \frac{x - x_{CB-AF}}{0.4x_{exit}} + 0.5 \quad \text{Eqn 72}$$

At $r = 0.5$ (the middle), there should be a balance for heat transfer coefficient between both models. The heat transfer coefficient in the transition zone can be written below:

$$\alpha_{CB-AF} = (1 - r)\alpha_{3Z} + \frac{r\alpha_{AF}}{(1 - r)\alpha_{3Z} + r\alpha_{AF}} (r\alpha_{AF} - (1 - r)\alpha_{3Z}) \quad \text{Eqn 73}$$

This new method is correct when the hydraulic diameters have a larger range. They cover microchannel a wide range of hydraulic diameter from 148 mm to 3 mm, four different refrigerants (R-134a, R-236fa, R-245fa and R-1234ze(E)) and were reported with accuracy going from 2 to 6% in the case of Ong and Thome (2011b) and 3 to 15% for Costa-Patry et al. (2011, 2012) [14].

2.4 Pressure Drop

A single-phase flows or two-phase mixture flowing in a pipe is restricted to a pressure variation. This pressure drop consists of three components which are named as accelerative, frictional and gravitational. Frictional pressure drop calculation in two-phase flow is an important aspect for various industrial applications such as heat exchangers, pipeline network systems, refrigeration and air conditioning systems, environmental control and life support systems, and oil and gas exploitation [4] [10].

In this sense, pressure drop in two-phase flow is studied by many scientists throughout the history. Based on their studies, many models and so the correlations are proposed for different types of data sets and conditions. The models developed depend on the type of the fluid (R134a, R113, R12, R22, water-vapor etc.) and the conditions under which the data sets are obtained. Depending on the condition there might be evaporation or condensation inside the tube. In order to simplify there exists some assumptions in the process of analyzing the pressure drop.

- ⇒ Steady-state conditions
- ⇒ One-dimensional flow
- ⇒ Uniform cross-section
- ⇒ Constant properties

An infinitesimal element is shown in Figure 4 in order to see the free body diagram for pressure drop. In this figure there are the following parameters to be defined;

τ_w	[Pa]	wall shear stress,
dM_g	[kg]	mass of the infinitesimal element,
P_z	[Pa]	pressure at position z,
P_{z+dz}	[Pa]	pressure at position z+dz,
S	[m]	perimeter,

Ω	[m ²]	cross-sectional area,
θ	[radian]	angel with horizontal
H_z	[m]	height at position z,
H_{z+dz}	[m]	height at position z+dz.

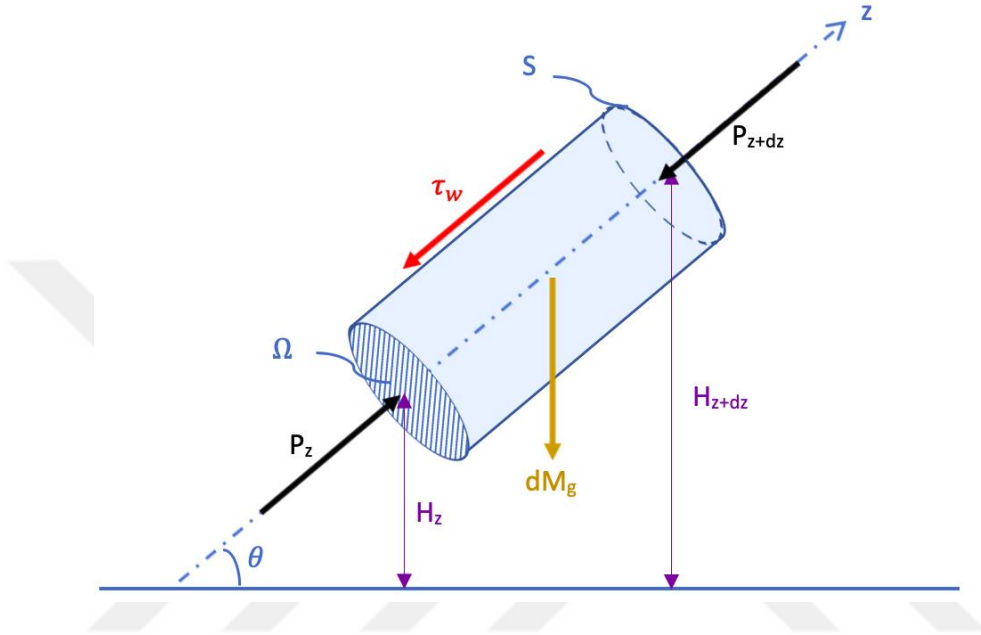


Figure 4: Control Volume for Balance Equations

According to Figure 4, the mass balance can be written as follows:

$$\Gamma_{g,z} + \Gamma_{l,z} - \Gamma_{g,z+dz} - \Gamma_{l,z+dz} = 0 \quad \text{Eqn 74}$$

Considering the assumption of uniform cross-section, the equation is reduced to:

$$\frac{d\Gamma}{dz} = 0 \quad \text{and eventually} \quad \frac{dG}{dz} = 0 \quad \text{Eqn 75}$$

Momentum variation is equal to the resultant of the applied forces to the control volume as shown above and can be written as follows:

$$\begin{aligned} \Gamma_{g,z+dz} U_{g,z+dz} + \Gamma_{l,z+dz} U_{l,z+dz} - \Gamma_{g,z} U_{g,z} - \Gamma_{l,z} U_{l,z} = \\ = (p_z - p_{z+dz})\Omega - \tau_w S dz - dM g \sin(\theta) \end{aligned} \quad \text{Eqn 76}$$

where dM can be written by using related equations:

$$\begin{aligned}
dM &= dM_g + dM_l = \rho_g dV_g + \rho_l dV_l = \rho_g \Omega_g dz + \rho_l \Omega_l dz = \\
&= \rho_g \Omega \bar{\alpha} dz + \rho_l \Omega (1 - \bar{\alpha}) dz = \\
&= [\rho_g \bar{\alpha} + \rho_l (1 - \bar{\alpha})] \Omega dz = \bar{\rho}^* \Omega dz
\end{aligned}
\tag{Eqn 77}$$

After the Taylor's expansion, the Eqn 76 can be written as:

$$\frac{d(\Gamma_g \bar{U}_g + \Gamma_l \bar{U}_l)}{dz} = -\frac{dp}{dz} \Omega - \tau_w S - \bar{\rho}^* g \sin(\theta) \Omega
\tag{Eqn 78}$$

The first term of the previous equation can be reformulated by taking advantage of the initial assumptions.

$$\frac{d(\Gamma_g \bar{U}_g + \Gamma_l \bar{U}_l)}{dz} = G^2 \Omega \frac{d}{dz} \left[v_g \frac{\bar{x}^2}{\bar{\alpha}} + v_l \frac{(1 - \bar{x})^2}{(1 - \bar{\alpha})} \right]
\tag{Eqn 79}$$

$$\bar{v}_m = \frac{1}{\bar{\rho}_m} = v_g \frac{\bar{x}^2}{\bar{\alpha}} + v_l \frac{(1 - \bar{x})^2}{(1 - \bar{\alpha})}
\tag{Eqn 80}$$

where; $\bar{v}_m$	[m ³ /kg]	momentum weighted avg specific vol,
$\bar{\rho}_m$	[kg/m ³]	momentum weighted average density,
v_g	[m ³ /kg]	gas phase specific volume,
v_l	[m ³ /kg]	liquid phase specific volume,
$\bar{\alpha}$	[-]	average void fraction,
$\bar{x}$	[-]	mass quality,
Γ_g	[kg/s]	gas mass flow rate,
Γ_l	[kg/s]	liquid mass flow rate,
$\bar{U}_g$	[m/s]	gas cross-section average velocity,
$\bar{U}_l$	[m/s]	liquid cross-section average velocity,
G	[kg/m ² s]	mass flux.

The pressure gradient is shown clearly below:

$$-\frac{dp}{dz} = G^2 \frac{d}{dz} \left(\frac{1}{\bar{\rho}_m} \right) + \tau_w \frac{S}{\Omega} + \bar{\rho}^* g \sin(\theta)
\tag{Eqn 81}$$

The Eqn 81 is used to get the pressure variation along the duct and $\bar{\alpha}$ can be calculated by using the related correlations. In right side of the Eqn 81 there is the summation of the three terms affecting the pressure drop per unit length: the first is accelerative in nature, the second is related to frictional and the third is connected to gravitational acceleration. The equations written from the momentum equation can be examined in three groups such, accelerative, fractional and gravitational effects.

In order to write to energy variation, the shaft work is negligible and only the work/heat flows are considered. The total energy specific per unit mass (e) is made by internal (u), kinetic (k) and potential (π) energy.

$$\begin{aligned}\Gamma_{g,z+dz}e_{g,z+dz} + \Gamma_{l,z+dz}e_{l,z+dz} - \Gamma_g e_g - \Gamma_l e_l &= \\ &= \Gamma_{g,z}p_z v_{g,z} - \Gamma_{g,z+dz}p_{z+dz}v_{g,z+dz} + \Gamma_{l,z}p_z v_{l,z} \\ &\quad - \Gamma_{l,z+dz}p_{z+dz}v_{l,z+dz} + \delta\dot{Q}\end{aligned}\quad \text{Eqn 82}$$

where; e [J/kg] internal energy,
 p [Pa] pressure,
 $\dot{Q}$ [W] heat exchanged.

The internal energy can be shown below:

$$\begin{aligned}\Gamma_{g,z+dz}u_{g,z+dz} + \Gamma_{l,z+dz}u_{l,z+dz} - \Gamma_g \bar{u}_g - \Gamma_l \bar{u}_l &= \\ &= G \frac{d}{dz} [u_g \bar{x} + u_l (1 - \bar{x})] \Omega\end{aligned}\quad \text{Eqn 83}$$

The bulk energy for internal, kinetic and potential can be obtained as follows:

$$\bar{u}_b = u_g + u_l (1 - \bar{x}) \quad \text{Eqn 84}$$

$$\bar{k}_b = k_g + k_l (1 - \bar{x}) = \frac{\bar{U}_g^2}{2} \bar{x} + \frac{\bar{U}_l^2}{2} (1 - \bar{x}) \quad \text{Eqn 85}$$

$$\bar{\pi}_b = \pi_g \bar{x} + \pi_l (1 - \bar{x}) = gH \quad \text{Eqn 86}$$

where; $\bar{u}_b$ [J/kg] bulk internal energy,
 $\bar{k}_b$ [J/kg] bulk kinetic energy,
 $\bar{\pi}_b$ [J/kg] bulk potential energy,

In addition to this, the right term of the Eqn 82 can be written as:

$$-\frac{d}{dz} \left[p \left((\Gamma_g v_g) + (\Gamma_l v_l) \right) \right] dz + \delta\dot{Q} = -G \frac{d(p\bar{v}_b)}{dz} \Omega dz + \delta\dot{Q} \quad \text{Eqn 87}$$

After dividing the equation to Γdz , this new expression is obtained:

$$\frac{d\bar{u}_b}{dz} + \frac{d\bar{k}_b}{dz} + g \frac{dH}{dz} = -\frac{d}{dz} (p\bar{v}_b) + \frac{\delta q}{dz} \quad \text{Eqn 88}$$

with:

$$\frac{d}{dz}(p\bar{v}_b) = p \frac{d\bar{v}_b}{dz} + \bar{v}_b \frac{dp}{dz} \quad \text{Eqn 89}$$

and

$$\frac{dH}{dz} = \sin(\theta) \quad \text{Eqn 90}$$

Lastly, considering to the first principle of the thermodynamic, it can be written:

$$d\bar{u}_b = Tds_b - pd\bar{v}_b \quad \text{Eqn 91}$$

According to Eqn 91, Tds_b shows the sum of the mechanical energy's dissipation (frictions) and the heat exchange. Then, Eqn 91 is divided to $\bar{v}_b$ and the last version of the expression is:

$$\bar{\rho}_b \frac{\bar{k}_b}{dz} + \bar{\rho}_b \frac{dR_b}{dz} + \bar{\rho}_b g \sin(\theta) = -\frac{dp}{dz} \quad \text{Eqn 92}$$

where R_b is the dissipation of mechanical energy per unit mass because of the frictional effect. The pressure drop is made of three terms and the accelerative pressure drop is shown below:

$$\begin{aligned} -\frac{dp_a}{dz} &= \frac{1}{2} \bar{\rho}_b \frac{d}{dz} [\bar{U}_g^2 \bar{x} + \bar{U}_l^2 (1 - \bar{x})] = \\ &= \frac{1}{2} \bar{\rho}_b G^2 \frac{d}{dz} \left[\frac{\bar{x}^3}{\rho_g^2 \bar{\alpha}^2} + \frac{(1 - \bar{x})^3}{\rho_l^2 (1 - \bar{\alpha})^2} \right] \end{aligned} \quad \text{Eqn 93}$$

For homogenous flow, the accelerative pressure drop can be written as:

$$-\frac{dp_a}{dz} = G^2 \frac{d}{dz} \left(\frac{1}{\bar{\rho}_b} \right) \quad \text{Eqn 94}$$

As shown before, the momentum balance approach is applied in order to get the result. Also, shear stress and void fraction must be calculated. It is related with the third flow pattern independent model that is the drift flux model. The void fraction can be determined as:

$$\alpha = \frac{J_g}{U_{gj} + C_0 J} \quad \text{Eqn 95}$$

where; J_g [m/s] superficial gas velocity,
 U_{gj} [m/s] local drift velocity,
 J [m/s] mixture volumetric flux,
 C_0 [-] distribution parameter,
 a [-] local void fraction.

In this thesis, the void fraction is determined by using the Rouhani correlation because it should be better for horizontal pipes, since it accounts for the effect of both gravity and inertia. After that, accelerative pressure drop is calculated by adopting and integrating the momentum balance. All the related equations for the Rouhani correlation are shown below as:

$$\alpha = \frac{x}{q_G} \left[C_0 \left(\frac{x}{q_G} + \frac{1-x}{q_L} \right) + \frac{U_{GM}}{G} \right]^{-1} \quad \text{Eqn 96}$$

$$U_{GM} = \left(\frac{1.18}{\sqrt{\rho_L}} \right) (g\sigma(\rho_L - \rho_G))^{0.25} \quad \text{Eqn 97}$$

$$C_0 = 1 + 0.2(1-x)(gD)^{0.25} \left(\frac{\rho_L}{G} \right)^{0.5} \quad \text{Eqn 98}$$

After integrating the momentum balance, the accelerative pressure drop can be written as:

$$\Delta P_a = G^2 \left[\frac{v_g \bar{x}_{out}^2}{\bar{\alpha}_{out}} + \frac{(1 - \bar{x}_{out})^2 v_l}{1 - \bar{\alpha}_{out}} - \frac{v_g \bar{x}_{in}^2}{\bar{\alpha}_{in}} + \frac{(1 - \bar{x}_{in})^2 v_l}{1 - \bar{\alpha}_{in}} \right] \quad \text{Eqn 99}$$

Finally, the total experimental pressure drop values are known from the data taken from the laboratory and it is equal to summation of the accelerative, frictional and gravitational pressure drop. In order to calculate the experimental frictional pressure drop values, the accelerative pressure drop shown from the Eqn 103 must be subtracted from the total experimental pressure drop values because of the gravitational pressure drop is equal to zero for horizontal tubes.

In addition, the other significant point is the fractional pressure gradient and it is written below as:

$$-\frac{dp_f}{dz} = \tau_w \frac{S}{\Omega} \quad \text{Eqn 100}$$

The definition of the friction factor:

$$f_{tp} = \frac{2\tau_w \bar{\rho}_b}{G^2} \quad \text{Eqn 101}$$

After τ_w is left alone on one side of Eqn 101 we obtain the definition of wall shear stress. Then τ_w is put in Eqn 100 and the final version of the frictional pressure gradient is shown as:

$$-\frac{dp_f}{dz} = \frac{2f_{tp}G^2\bar{v}_b}{D} \quad \text{Eqn 102}$$

According to dimensionless analysis, the friction factor is a function of the Reynold number and it can be presented below:

$$Re_{tp} = \frac{GD}{\mu_{tp}} \quad \text{Eqn 103}$$

Several methods are developed for the calculation of the two-phase mixture apparent viscosity such as Owens, McAdams and Cicchitti.

Owens:

$$\mu_{tp} = \mu_l \quad \text{Eqn 104}$$

The liquid phase wet the wall, then:

$$Re_{tp} = \frac{GD}{\mu_l} = Re_{lo} \quad \text{Eqn 105}$$

The Re_{lo} is defined before the liquid-only Reynold number. It means that, the liquid phase is flowing with the total mass flow rate of the mixture inside of the tube.

McAdams:

$$\frac{1}{\mu_{tp}} = \frac{\bar{x}}{\mu_g} + \frac{(1 - \bar{x})}{\mu_l} \quad \text{Eqn 106}$$

Cicchitti:

$$\mu_{tp} = \bar{x}\mu_g + (1 - \bar{x})\mu_l \quad \text{Eqn 107}$$

In order to make it simple, the Blasius equation for the friction can be shown below:

$$\begin{aligned}
f_{tp} &= 0.079 \left(\frac{\mu_{tp}}{GD} \right)^{1/4} = 0.079 \left[GD \left(\frac{\bar{x}}{\mu_g} + \frac{1-\bar{x}}{\mu_l} \right) \right]^{-1/4} \\
&= 0.079 \left[\frac{GD}{\mu_l} \left(1 + \bar{x} \frac{\mu_{lg}}{\mu_g} \right) \right]^{-1/4}
\end{aligned}
\tag{Eqn 108}$$

The result of the friction factor depends on x and dominated by the liquid-only Reynold number. Also, the two-phase multiplier can be written as follows:

$$\phi_{lo}^2 = \frac{-\frac{dp_f}{dz}}{-\left(\frac{dp_f}{dz}\right)_{lo}}
\tag{Eqn 109}$$

$$\phi_{lo}^2 = \frac{f_{tp}}{f_{lo}} \frac{\bar{v}_b}{v_l} = \left(\frac{Re_{lo}}{Re_{tp}} \right)^n \frac{v_g + v_l(1-\bar{x})}{v_l} = \left(\frac{\mu_{tp}}{\mu_l} \right)^n \left(1 + \bar{x} \frac{v_g - v_l}{v_l} \right)
\tag{Eqn 110}$$

In order to consider this equation, two assumptions must be done. First, the compressibility of the gas phase is negligible. Second; specific volume of the gas, liquid phases and the two-phase friction factor are constant. The pressure drop could be evaluated in both cases of homogenous and separated flow models. In particular, in the separated flow model friction component is calculated from two phase multiplier which could be correlated by means of the Lockhart-Martinelli parameter. A further expansion of this quantity was provided by Chisholm as shown in Eqn 111 Eqn 112.

$$\phi_g^2 = 1 + CX + X^2
\tag{Eqn 111}$$

$$\phi_l^2 = 1 + \frac{C}{X} + \frac{1}{X^2}
\tag{Eqn 112}$$

where C is defined the constant parameter, which depends on the flow regime of the single-phase. The values of constant parameters are listed in Table 2.

Lockhart-Martinelli Parameter	X_{tt}	X_{lt}	X_{tl}	ll
C	20	12	10	5

Table 2: The Values of Parameter C for Lockhart-Martinelli

2.5 Flow Patterns in Horizontal Flow

In the study of multi-phase flow the viscous or turbulent behavior is important as well as for a single phase flow; however; the topology on geometry of the flow plays a greater importance [15]. From a practical engineering point of view one of the major design difficulties in dealing with multiphase flow is that the mass, momentum, and energy transfer rates and

processes can be quite sensitive to the geometric distribution or topology of the components within the flow. For example, the geometry may strongly affect the interfacial area available for mass, momentum or energy exchange between the phases [16]. The variety of configurations taken up by the phases during flow is named as flow patterns.

Flow patterns for co-current flow of gas and liquid in a horizontal tube are strongly influenced by gravity that acts to stratify the liquid to the bottom of the tube and the gas to the top. The liquid and gas phases distribute themselves into several recognizable flow structures [17]. It is useful to provide some examples of flow regime maps along with the definitions that help distinguish the various regimes. There exist the following flow patterns which are formed as a function of void fraction and vapor velocity;

- **Bubbly Flow:** In bubbly flow there are high concentration of the bubbles in the upper part (or half) of the tube because of the buoyancy force. In this type of flow the gas bubbles are dispersed in the liquid. The bubbles tend to disperse uniformly through the tube if there is dominant shear force. In the horizontal flow, this regime occurs only if there is high mass flow rates [17].
- **Stratified Flow:** Complete separation of the gas and liquid phases occur at low velocities. The gas goes to the top while the liquid stays on the bottom of the tube. Both liquid and gas phases are separated by a continuous flat interface. In other words, gas and liquid are stratified, totally.

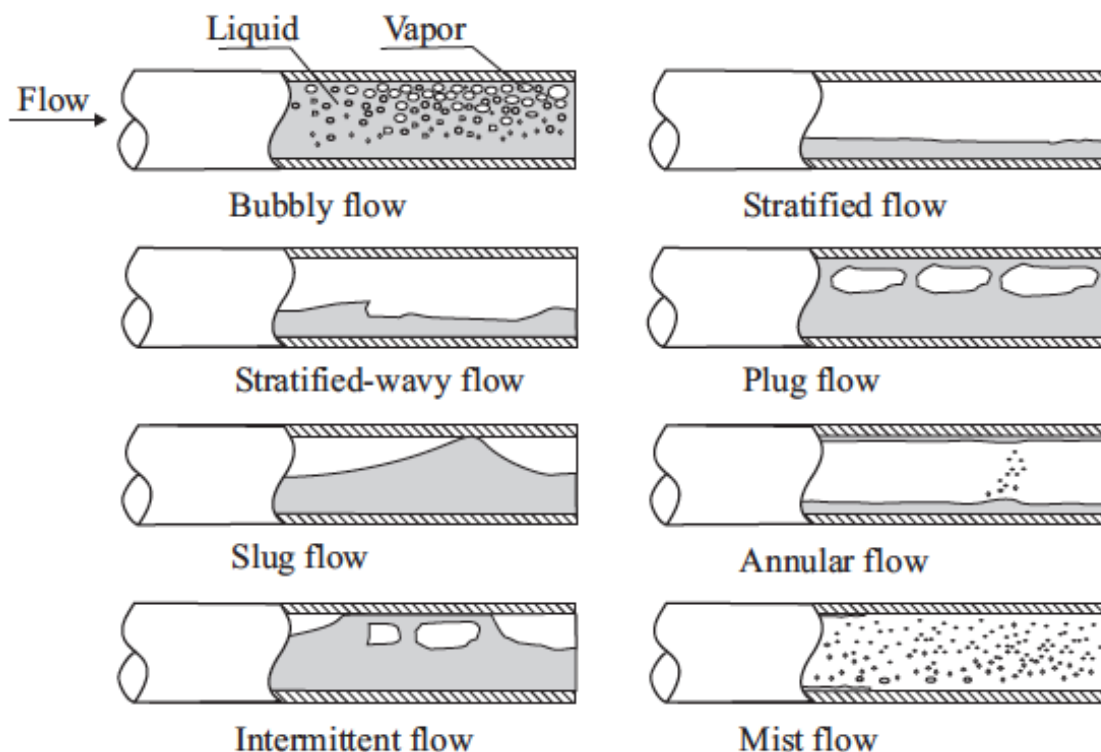


Figure 5: Two-Phase Flow Patterns in Horizontal Flow

- **Stratified-Wavy Flow:** Increasing the gas velocity in a stratified flow, waves are formed on the interface and travel in the direction of the flow [17]. The amplitude of the waves depends on the relative velocity of gas and liquid phases. However; as it can be seen in Figure 5 their tips do not reach the top of the tube. While the waves climb up the side of the tube, they leave films of liquid on the wall.
- **Intermittent Flow:** Increasing the gas velocity more, the interfacial waves become larger so that they can reach the top of the tube. This type of flow can be characterized with the large amplitude waves that reaches the top of the tube intermittently. In between these large amplitude waves, there exists smaller amplitude waves. Top of the tube is almost continuously wet due to the large amplitude waves and so there is a thin liquid film left on the wall. Intermittent flow is can also be described as the combination of the plug flow and slug flow. The definition of plug flow and slug flow is as follows;
 - ⇒ **Plug Flow:** In this type of the flow, there are the liquid plugs which are separated by the elongated bubbles [17]. The diameter of these elongated bubbles are smaller than the tube diameter and also, these bubbles are formed on the upper part of the tube cross-section. In other saying, there is a continuous liquid flow on the lower part of the tube cross-section.
 - ⇒ **Slug Flow:** Slugs are kinematic or concentration waves which are sometimes called continuity waves [16].The diameters of elongated bubbles become similar to the size of the channel height with higher gas velocities. Liquid slugs that separates these elongated bubbles can also be described as large amplitude waves [17].
- **Annular Flow:** When there is even a larger gas flow rate, the liquid phase stays on the wall of the tube. In other words, there is a continuous liquid film on the inner perimeter of the tube. The interface between liquid annulus and the vapor core is disturbed by small amplitude waves and droplets may be dispersed in the gas core as it is seen in Figure 5. When there is a high gas fraction, annular flow transforms in to stratified-wavy flow because the top of the tube with thinner film becomes dry first.
- **Mist Flow:** When the velocities are very high, most of the liquid is entrained as spray by the gas. In other words, mist flow occurs at very high flow rates and very high flow quality. The droplets might make the wall wet; however, this situation occurs intermittently and only locally.

2.6 Flow Pattern Maps in Horizontal Flow

Different flow regimes are generally observed at the different positions of the duct along its length when phase change occurs during the two-phase mixture flow. The position and the sequence of different types of flow depend flow rate, fluid properties, distribution and the

magnitude of the heat flux and also on the channel characteristics. Figure 6 shows the flow regimes during evaporation and condensation;

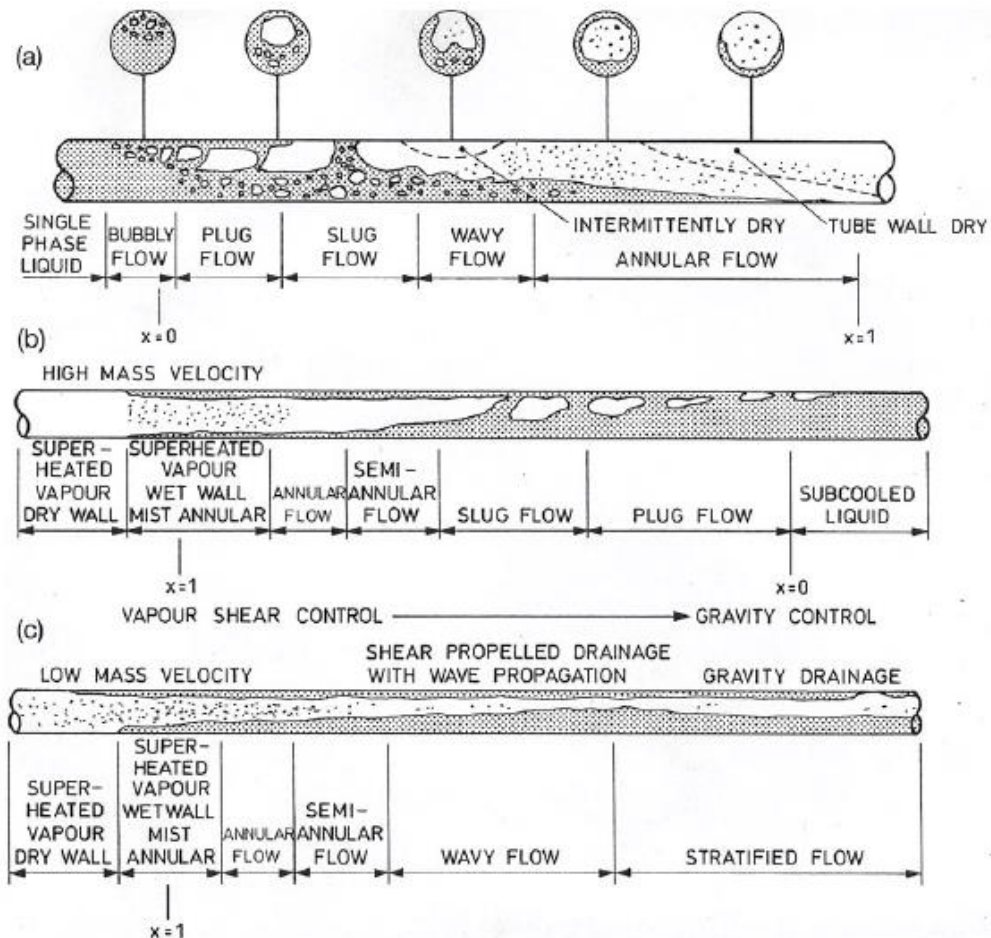


Figure 6: Flow Regimes During Evaporation and Condensation

- Schematic representation of a horizontal tubular heated channel by a uniform low heat flux fed with liquid just below the saturation temperature,
- Condensation with high liquid loading,
- Condensation with low liquid loading.

In order to be able to predict the local flow pattern, a flow pattern map is used. These are two-dimensional graphs and aim to separate the space into areas corresponding to the various flow regimes [17]. Baker flow pattern map can be an example for horizontal flow in a tube.

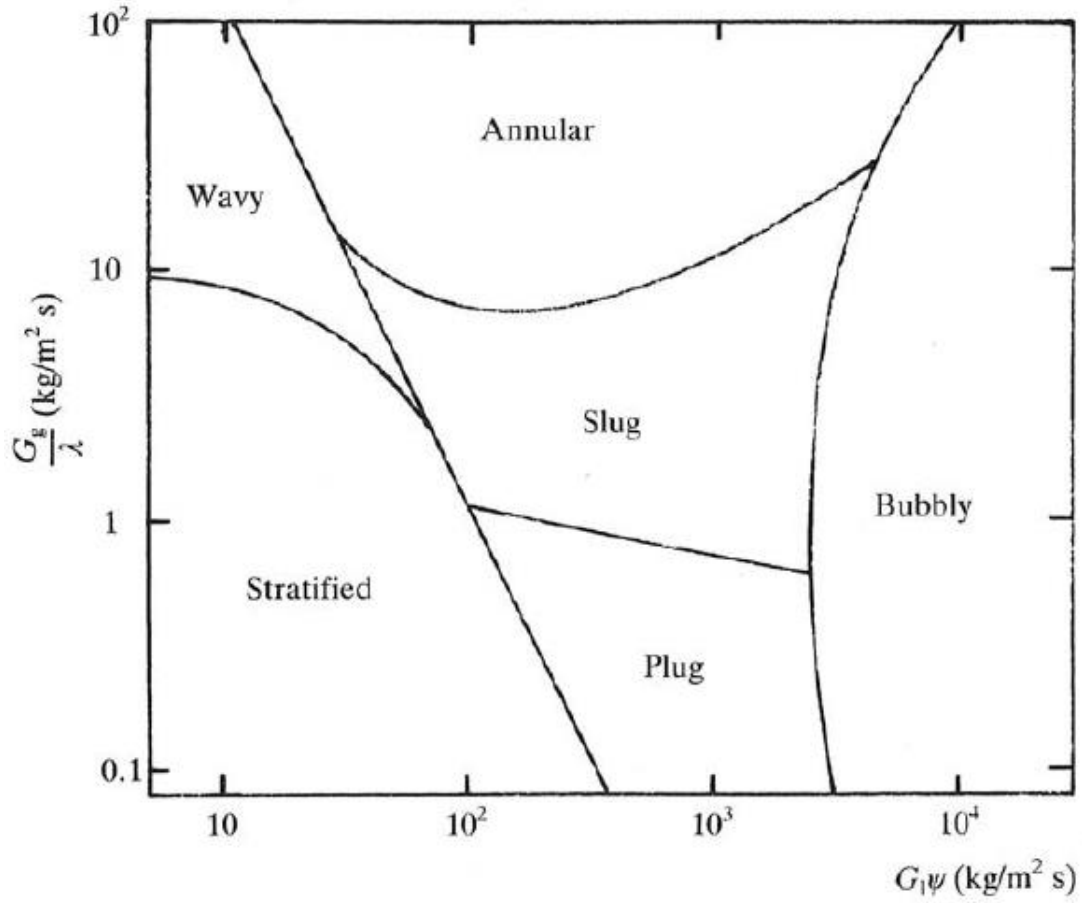


Figure 7: Baker (1954) Flow Pattern Map for Horizontal Flow in a Tube

It should be noted that this map is developed based on air-water data. λ and φ are dimensionless parameters and they are defined as follows [17];

$$\lambda = \left(\frac{\rho_g}{\rho_{air}} \frac{\rho_l}{\rho_{water}} \right)^{1/2} \quad \text{Eqn 113}$$

$$\varphi = \left(\frac{\sigma_{water}}{\sigma} \right) \left[\left(\frac{\mu_l}{\mu_{water}} \right) \left(\frac{\rho_{water}}{\rho_l} \right)^2 \right]^{1/3} \quad \text{Eqn 114}$$

Chapter 3

3 Frictional Pressure Drop Empirical Correlations

3.1 Muller-Steinhagen and Heck

Muller-Steinhagen and Hack proposed this method in 1986 and it is essentially an empirical two-phase extrapolation between single phase liquid flow and single phase vapor flow [4]. Based on a research made by Ribatski, Muller-Steinhagen and Hack provides the best predictions [18] [19]. The method is a ϕ_{lo}^2 based model and the two-phase frictional multiplier of liquid only, ϕ_{lo}^2 is calculated with the following correlation;

$$\phi_{lo}^2 = Y^2 x^3 + (1 - x)^{1/3} [1 + 2x(Y^2 - 1)] \quad \text{Eqn 115}$$

$$Y^2 = \frac{\left(-\frac{dp_f}{dz}\right)_{go}}{\left(-\frac{dp_f}{dz}\right)_{lo}} \quad \text{Eqn 116}$$

After all the parameters are calculated the pressure drop is calculated as follows;

$$-\frac{dp_f}{dz} = \phi_{lo}^2 \left(-\frac{dp_f}{dz}\right)_{lo} \quad \text{Eqn 117}$$

$$\text{where; } \left(-\frac{dp_f}{dz}\right)_{lo} = \frac{2 f_{lo} G^2 v_l}{D} \quad \text{Eqn 118}$$

where; v_l	[m ³ /kg]	liquid specific volume,
D	[m]	diameter of the duct,
G	[kg/m ² s]	mass flux,
f_{lo}	[-]	liquid only friction factor.

As it is also stated by Muller-Steinhagen and Heck, this method is simple and more convenient to use compared to other methods. In order to check the reliability of this method, it is compared with fourteen other correlations by using a data bank of 9300 measurements [20]. Best agreement between predicted and measured values was obtained with the correlation by Bandel which, however, is quite lengthy and complicated to use. The correlation suggested by Muller-Steinhagen and Hack is more convenient, and still predicts the frictional pressure drop with reasonable accuracy. It included single-phase liquid and gas pressure drop and predicts correctly the influence of flow parameters [20].

3.2 Gronnerud Correlation

The correlation was proposed by Gronnerud in 1979. Gronnerud correlation can provide acceptable predictions under restricted circumstances. It is effective when there exists evaporation data. It also does not perform worse in vertical flow compared to the horizontal flow [4]. However, based on the comparison of Xu and Fang it shows larger errors under entire scope of vapor quality. The method is a ϕ_{lo}^2 based model and the two-phase frictional multiplier of liquid only, ϕ_{lo}^2 is calculated with the following correlation;

$$\phi_{lo}^2 = 1 + \left(-\frac{dp}{dz}\right)_{Fr} \left[\left(\frac{\rho_l}{\rho_g}\right) \left(\frac{\mu_g}{\mu_l}\right)^{0.25} - 1 \right] \quad \text{Eqn 119}$$

$$\left(-\frac{dp}{dz}\right)_{Fr} = f_{Fr} [x + 4(x^{1.8} - x^{10} f_{Fr}^{0.5})] \quad \text{Eqn 120}$$

$$f_{Fr} = \begin{cases} 1, & Fr_{lo} \geq 1 \\ Fr_{lo}^{0.3} + 0.0055 \left[\ln \left(\frac{1}{Fr_{lo}} \right) \right]^2, & Fr_{lo} < 1 \end{cases} \quad \text{Eqn 121}$$

$$Fr_{lo} = \frac{G^2}{gD\rho_l^2} \quad \text{Eqn 122}$$

$$-\frac{dp_f}{dz} = \phi_{lo}^2 \left(-\frac{dp_f}{dz}\right)_{lo} \quad \text{Eqn 123}$$

After all the parameters are calculated the pressure drop is calculated as follows;

$$\left(-\frac{dp_f}{dz}\right)_{lo} = \frac{2 f_{lo} G^2 v_l}{D} \quad \text{Eqn 124}$$

3.3 Friedel Correlation

In 1979, Friedel performed a predictive model for both horizontal and vertical smooth tubes [1]. Based on 25,000 data points it considers the effects of the gravity and surface tension. The method was propounded on the bases of the two-phase friction multipliers proposed by Lockhart and Martinelli (1949) and the fitting correlation of the multipliers from Chisholm (1967) [4]. Predictions of the model are in satisfactory agreement with data for bubbly and slug flow patterns at high Re_l [21]. The method is a ϕ_{lo}^2 based model and the two-phase frictional multiplier of liquid only, ϕ_{lo}^2 is calculated with the following correlation;

$$\phi_{lo}^2 = (1-x)^2 + x^2 \frac{\rho_l f_{go}}{\rho_g f_{lo}} + \frac{3.24 x^{0.78} (1-x)^{0.224} H}{Fr_{tp}^{0.045} We_{tp}^{0.035}} \quad \text{Eqn 125}$$

$$H = \left(\frac{\rho_l}{\rho_g}\right)^{0.91} \left(\frac{\mu_g}{\mu_l}\right)^{0.19} \left(1 - \frac{\mu_g}{\mu_l}\right)^{0.7} \quad \text{Eqn 126}$$

$$Fr_{tp} = \frac{G_{tp}^2}{gD\rho_{tp}^2} \quad \text{Eqn 127}$$

$$\frac{1}{\rho_{tp}} = \frac{x}{\rho_g} + \frac{1-x}{\rho_l} \quad \text{Eqn 128}$$

After all the parameters are calculated the pressure drop is calculated as follows;

$$-\frac{dp_f}{dz} = \phi_{lo}^2 \left(-\frac{dp_f}{dz}\right)_{lo} \quad \text{Eqn 129}$$

$$\left(-\frac{dp_f}{dz}\right)_{lo} = \frac{2 f_{lo} G^2 v_l}{D} \quad \text{Eqn 130}$$

where; v_l	[m ³ /kg]	liquid specific volume,
D	[m]	diameter of the duct,
G	[kg/m ² s]	mass flux,
f_{lo}	[-]	liquid only friction factor.

3.4 Chen Correlation

The Chen correlation was proposed in 2001 and it is basically the modified version of the Friedel correlation, the two-phase frictional pressure gradient is the frictional pressure drop provided by the Friedel correlation multiplied by a constant of Ω . The Friedel correlation is basically a ϕ_{lo}^2 based method. The details of this correlation were discussed in the previous section. Xu and Fang [4] also proposed the details of Friedel method in their published paper. It is important to bear in mind that correlation of Ω is obtained from the data measured for air-water and R410a [4]. The method calculated the two-phase frictional drop directly by using the following equations;

$$\left(-\frac{dp}{dz}\right)_{tp} = \Omega \left(-\frac{dp}{dz}\right)_{tp, \text{Friedel}} \quad \text{Eqn 131}$$

where; $x = \bar{x}$

$$\Omega = \begin{cases} \frac{0.0333 Re_{lo}^{0.45}}{Re_g^{0.09}[1 + 0.4 \exp(-Bo)]}, x < 0 \\ \frac{We_{tp}^{0.2}}{2.5 + 0.06 Bo}, x \geq 0 \end{cases} \quad \text{Eqn 132}$$

$$Bo = \frac{g(\rho_l - \rho_g) \left(\frac{D}{2}\right)^2}{\sigma} \quad \text{Eqn 133}$$

$$We_{tp} = \frac{G^2 D}{\rho_{tp} \sigma} \quad \text{Eqn 134}$$

$$\frac{1}{\rho_{tp}} = \frac{x}{\rho_{tp}} + \frac{1-x}{\rho_{tp}} \quad \text{Eqn 135}$$

3.5 Tran Correlation

Tran proposed this correlation in 2000. It is applicable for R134a, R113 and R122 in smooth tubes with $p \in [138; 864]$ kPa and $G \in [33; 832]$ kg/m²s, $q \in [2.2; 90.8]$ kW/m² and $x \in [0; 0.95]$. The method is a ϕ_{lo}^2 based model and the two-phase frictional multiplier of liquid only, ϕ_{lo}^2 is calculated with the following correlation;

$$\phi_{lo}^2 = 1 + (4.3 Y^2 - 1)(x[1 - x]^{0.875} La + x^{1.75}) \quad \text{Eqn 136}$$

$$La = \frac{\sqrt{\sigma / g(\rho_l - \rho_g)}}{D} \quad \text{Eqn 137}$$

After all the parameters are calculated the pressure drop is calculated as follows;

$$-\frac{dp_f}{dz} = \phi_{lo}^2 \left(-\frac{dp_f}{dz} \right)_{lo} \quad \text{Eqn 138}$$

$$\text{where; } \left(-\frac{dp_f}{dz} \right)_{lo} = \frac{2 f_{lo} G^2 v_l}{D} \quad \text{Eqn 139}$$

3.6 Haraguchi Correlation

Haraguchi developed a series of condensation heat transfer measurements for R134, R134A, and R22 horizontally smooth tube and a horizontally micro-fin tube, and presented a correlation based on vapor-phase two-phase multiplier (ϕ_g) and the Martinelli parameter X_{tt} [1]. The method is a ϕ_g^2 based model and the two-phase frictional multiplier of gas alone, ϕ_g^2 is calculated with the following correlation;

$$\phi_g = 1.1 + 1.3 \left\{ \frac{X_{tt} G}{[g d_e \rho_g (\rho_l - \rho_g)]^{0.5}} \right\}^{0.35} \quad \text{Eqn 140}$$

$$d_e = \sqrt{\frac{4A_c}{\pi}} \quad \text{Eqn 141}$$

$$f_{e.g} = 0.046 Re_{e.g}^{-0.2} \quad \text{Eqn 142}$$

$$Re_{e.g} = \frac{G d_e x}{\mu_g} \quad \text{Eqn 143}$$

$$X_{tt} = \left(\frac{1-x}{x} \right)^{0.9} \left(\frac{\rho_g}{\rho_l} \right)^{0.5} \left(\frac{\mu_l}{\mu_g} \right)^{0.1} \quad \text{Eqn 144}$$

After all the parameters are calculated the pressure drop is calculated as follows;

$$\left(\frac{dP}{dz} \right)_f = \phi_g^2 \left(\frac{dP}{dz} \right)_g = \frac{2 \phi_g^2 f_{e.g} (Gx)^2}{\rho_g d_e} \quad \text{Eqn 145}$$

Chapter 4

4 Heat Transfer Coefficient Empirical Correlations

Nowadays, technology has been developing day by day and micro-fin tubes are also parts of this improvement. The micro-fin tubes are generally used in industrial applications. The calculation of heat transfer coefficient is a significant consideration in order to reduce the cost and avoid the drastic results due to the design of evaporators, boilers and other two-phase process equipment [5]. In recent years, many correlations and models are developed in order to calculate the heat transfer coefficient such as Kandlikar and Shah.

4.1 Kandlikar Model

A simple correlation was developed earlier by Kandlikar (1983) for predicting saturated flow boiling heat transfer coefficients inside horizontal and vertical tubes. The correlation was based on a model utilizing the contributions due to nucleate boiling and convective mechanisms. It incorporated a fluid-dependent parameter F_{fl} in the nucleate boiling term.

After reviewing the literature for available correlations and models describing the flow boiling mechanism, it was clear that the ability of a correlation to apply to new fluids is an important feature [5]. Equally important are the ability to correlate additional data with any test fluids used in the correlation development, and the ability to predict correctly the dependency of heat transfer coefficient on various system parameters. The Kandlikar (1983) correlation was able to correlate the data better than the Shah (1982) correlation and was able to be extended to new fluids. This correlation was based on only 1100 data points, while a larger data base seemed necessary to arrive at a general correlation [5] [15].

The two-phase flow boiling heat transfer coefficient, h_{TP} was expressed as the sum of the convective and the nucleate boiling terms and it is shown below:

$$h_{TP} = C_1 Co^{C_2} h_l + C_3 Bo^{C_4} h_l \quad \text{Eqn 146}$$

where; h_{TP}	[W/m ² °C]	two-phase heat transfer coefficient,
$C_1 - C_4$	[-]	constants to be determined,
Co	[-]	convection number,
Bo	[-]	boiling number,
h_l	[W/m ² °C]	single-phase heat transfer coefficient liquid fraction.

In this thesis, the Dittus-Boelter equation is used in order to calculate h_l . The mean error for each data set was also calculated. Initially, the data was divided into two regions as follows:

$Co < 0.65$ convective boiling region,

$Co > 0.65$ nucleate boiling region.

In the convective boiling region, the heat transfer is predominantly by a convective mechanism. Similarly, in the nucleate boiling region, the heat transfer is predominantly by nucleate boiling mechanism. Influence of heat flux is particularly different in the two regions, as will be discussed later.

The correlation was then extended to other fluids by incorporating a fluid-dependent parameter F_{fl} . Eqn 146 is rewritten in a slightly different form, and F_{fl} is included in the second term as follows:

$$\frac{h_{TP}}{h_l} = C_1 Co^2 + C_3 Bo^{C_4} F_{fl} \quad \text{Eqn 147}$$

The F_{fl} value that resulted in the lowest combined mean error for all data sets for that fluid under both regions, convective and nucleate boiling, was finally selected [5] [15]. The influence of stratification at low flow rates in horizontal tubes was correlated by introducing the Froude number in the nucleate boiling in the convective boiling terms as follows:

$$\frac{h_{TP}}{h_l} = C_1 Co^2 (25 Fr_{lo})^{C_5} + C_3 Bo^{C_4} (25 Fr_{lo})^{C_6} F_{fl} \quad \text{Eqn 148}$$

The limiting value of 0.04 was found to be most suitable with all the horizontal-flow data sets. The entire process from the beginning was repeated to include the horizontal-flow data points with $Fr_{lo} > 0.04$ along with the vertical-flow data sets for evaluating the constants. The final form of the equation obtained in this analysis is

$$\frac{h_{TP}}{h_l} = C_1 Co^2 (25 Fr_{lo})^{C_5} + C_3 Bo^{C_4} F_{fl} \quad \text{Eqn 149}$$

The single-phase liquid-only heat transfer coefficient h_l is given by

$$h_l = 0.023 Re_l^{0.8} Pr_l^{0.4} (k_l / D) \quad \text{Eqn 150}$$

The values of constants $C_1 - C_5$ are given in Table 3 [15]. The constant C_6 in Eqn 148 was found to be zero, and therefore the Froude number multiplier in the nucleate boiling term in Eqn 149 is missing.

Constant	Convective Region	Nucleate Boiling Region
C₁	1.136	0.6683
C₂	-0.9	-0.2
C₃	667.2	1058
C₄	0.7	0.7
C₅(horizontal)	0.3	0.3
C₅(vertical)	0	0

Table 3: Constants for Kandlikar

The two sets of values given in Table 3 correspond to the convective boiling and nucleate boiling regions, respectively. The heat transfer coefficient at any given condition is evaluated using the two sets of constants for the two regions, and since the transition from one region to another occurs at the intersection of the respective correlations, the higher of the two heat transfer coefficient values represents the predicted value from the proposed correlation. Also, the fluid dependent parameters F_{fl} are shown below in Table 4. This method provides a continuity between the convective and nucleate boiling regions.

Fluid	F_{fl}
R-22	2.2
R-134a	1.63

Table 4: Fluid-surface Parameter

▪ **Nucleate Boiling Term**

The constant C_4 in the proposed correlation, Eqn 149, is the exponent to the boiling number $Bo = q/(Gi_{lg})$ in the nucleate boiling term and is seen to be 0.7 for both convective and nucleate boiling regions. The dependence of the nucleate boiling contribution on heat flux q is therefore proportional to $q^{0.7}$ [5]. The underlying mechanism for the nucleate boiling in two-phase flow can be related to the pool boiling, and it seems reasonable to expect similar dependence on q in both cases. The dependence of the heat transfer coefficient on q in pool boiling can be expressed as $h \propto q^n$.

▪ **Convective Boiling Term**

The dependence of the convective boiling term on quality is investigated in the two regions. In the convective boiling region, the exponent of Co is -0.9, and along with the $Re_l^{0.8}$ in the h_l expression, the convective boiling term varies as $x^{0.72}(1-x)^{0.08}$. However, this dependence should not be viewed in isolation, since the nucleate boiling contribution varies as $(1-x)^{0.8}$. Combining these two contributions may result in a different dependence of h_{TP} on x depending on the fluid and other system and operating parameters.

The exponent of Co in the nucleate boiling region is -0.2, which is much weaker than the value -0.891 found in the earlier work by Kandlikar (1983). The main reason for this is that very few data points in this region were used in the earlier work. This weak dependence on Co is to be expected in the nucleate boiling region where convection is not the dominant mechanism of heat transfer [5]. As Bo increases, the percentage contribution due to convective boiling decreases. At high heat fluxes the heat transfer is predominantly by the nucleate boiling mechanism. However, at low heat flux values, the convective contribution may be quite significant. The proposed correlation can cover both these cases.

4.2 Shah Model

The calculation of the heat transfer coefficients for boiling fluids has an important part in order to design of boilers and evaporators [7]. In 1976, the author is performed in graphical form a general correlation to predict the heat transfer coefficients during saturated boiling. Approximately, the 780 data points are used from 19 independent experimental studies to have low mean deviation value [22]. This correlation has been tested many times against to a larger amount of extra data for tubes by other researchers and the author. In order to calculate the heat transfer coefficient according to Shah, the Eqn 151 shows below the correlation in graphical form. The determination of ψ_{bs} has two different equations that depend on convection number. The procedure and remaining steps are written below, respectively.

$$\psi = \frac{h_{tp}}{h_l} \quad \text{Eqn 151}$$

The dimensionless Shah parameter can be found by the following equations:

$$\psi_{bs} = \begin{cases} FBo^{0.5} \exp(2.74N^{-0.1}), & 0.1 < N \leq 1.0 \\ FBo^{0.5} \exp(2.47N^{-0.15}), & N \leq 0.1 \end{cases} \quad \text{Eqn 152}$$

when the boiling and Froud number are:

$$\begin{aligned} Bo &\geq 11 \times 10^{-4}, & F &= 14.7 \\ Bo &< 11 \times 10^{-4}, & F &= 15.43 \end{aligned} \quad \text{Eqn 153}$$

The dimensionless parameter N is defined as follows; For vertical tubes at all values of Fr_L and for horizontal tubes with $Fr_L \geq 0.04$,

$$N = Co \quad \text{Eqn 154}$$

For horizontal tubes with $Fr_L \leq 0.04$,

$$N = 0.38 Fr_L^{-0.3} Co \quad \text{Eqn 155}$$

Finally, the heat transfer coefficient is calculated by using all these equations for Shah model according to collected data.

4.3 Gungor and Winterton Correlation

The correlation is performed by Gungor and Winterton in 1986. It is essentially the modification of the Chen correlation, and is improved by using 3700 data points [5]. The authors tried to improve a model, it can also be extended to subcooled boiling for both vertical and horizontal flow. Eventually, the last form of the correlation is shown below and the heat transfer coefficient for two-phase is determined by using Eqn 156.

$$\frac{h_{TP}}{h_l} = 1 + 3000Bo^{0.86} + [x/(1-x)]^{0.75}(\rho_l/\rho_g)^{0.41} \quad \text{Eqn 156}$$

4.4 Fang Correlation

In the literature, there are lots of research papers about the calculation of heat transfer coefficient due to importance of the topic. Many correlations and models are recommended in order to calculate the heat transfer coefficient. Fang performed a new correlation for R134a two-phase flow boiling heat transfer. Totally, 2286 experimental data points are used for R134a, the least squares and error analysis have been applied for improving of a new correlation [23]. After that, the new correlation is shown as follows;

$$Nu = \frac{0.00061(S + F)Re_l Pr_l^{0.4} Fa^{0.11}}{\ln\left(\frac{1.023\mu_{l,f}}{\mu_{l,w}}\right)} \quad \text{Eqn 157}$$

$$S = \begin{cases} 30000Bo^{1.13}, & Bo < 0.0026 \\ 36x, & Bo \geq 0.0026 \end{cases} \quad \text{Eqn 158}$$

$$F = \left(\frac{x}{1-x}\right)^{0.95} \left(\frac{\rho_l}{\rho_v}\right)^{0.4} \quad \text{Eqn 159}$$

where; $\mu_{l,f}$ [Pas] liquid viscosity at the fluid temperature,
 $\mu_{l,w}$ [Pas] liquid viscosity at the wall temperature.

Afterwards, the two-phase heat transfer coefficient is determined by Eqn 160.

$$h_{tp} = Nu\lambda/D_h \quad \text{Eqn 160}$$

Fa number is a dimensionless parameter developed by Fang. The Fa was introduced recently in 2013 by Professor Xiande Fang to provide great facilitation in describing flow boiling heat transfer coefficients. It is defined as the product of two terms. The first term is the ratio of buoyancy force to gravitational force, which has effects on bubble departure. The second term is the ratio of surface tension force to inertial force, which affects bubble formation. As a result, Fa is associated with the formation and departure of bubbles:

$$Fa = \frac{(\rho_l - \rho_v)\sigma}{G^2 D_h} \quad \text{Eqn 161}$$

The liquid Reynolds number is shown as:

$$Re_l = \frac{(1-x)GD_h}{\mu_l} \quad \text{Eqn 162}$$

4.5 Saitoh Correlation

The authors are performed a correlation for flow boiling heat transfer based on R13a in horizontal tubes. The correlation is improved in order to make a great contribution in the design of compact heat exchangers which utilize small tubes [24]. After the modification of Chen correlation for saturated flow boiling, the heat transfer coefficient arises from a convective boiling contribution and a nucleate boiling contribution. Then, the heat transfer coefficient is shown as:

$$h = Fh_l + Sh_{pool} \quad \text{Eqn 163}$$

where the liquid phase of heat transfer coefficient is determined as:

$$h_l = \begin{cases} 0.023 \frac{k_l}{D} Re_l^{0.8} Pr_l^{1/3}, & Re_l > 1000 \\ \frac{4.36k_l}{D}, & Re_l < 1000 \end{cases} \quad \text{Eqn 164}$$

The collected 2224 flow boiling data points are used for modification of the correlation for 134a flowing in horizontal tubes with $0.5 < D < 11\text{mm}$ as inner diameter [24]. The factor F which is dimensionless parameter for forced convective heat transfer is expressed as a function of X and We_g . Finally, the procedure and remaining steps are written below, respectively.

$$F = 1 + \frac{\left(\frac{1}{X_{tt}}\right)^{1.05}}{1 + We_g^{-0.4}} \quad \text{Eqn 165}$$

$$We_g = G_g^2 D / \sigma \rho_g \quad \text{Eqn 166}$$

$$S = \frac{1}{1 + 0.4(Re_{tp} 10^{-4})^{1.4}} \quad \text{Eqn 167}$$

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_g}{\rho_l}\right)^{0.5} \left(\frac{\mu_l}{\mu_g}\right)^{0.1} \quad \text{Eqn 168}$$

where the two-phase Reynolds number is shown as:

$$Re_{tp} = Re_l F^{1.25} \quad \text{Eqn 169}$$

After all, Saitoh et al. are determined the nucleate pool boiling term according to Eqn 170.

$$h_{pool} = 207 Pr_l^{0.533} \frac{k_l}{D_b} \left(\frac{q D_b}{k_l T_{sat}} \right)^{0.745} \left(\frac{\rho_v}{\rho_l} \right)^{0.581} \quad \text{Eqn 170}$$

where the D_b is defined as the bubble departure diameter:

$$D_b = 0.51 \left[\frac{2\sigma}{g(\rho_l - \rho_v)} \right]^{0.5} \quad \text{Eqn 171}$$



Chapter 5

Explanation made in this section is directly taken by “International Journal of Refrigeration 35 (2012), 2150-2165”. For more of a detail please see reference number [25] in Bibliography.

5 Experimental Setup

The experimental apparatus consists of three loops, namely: the refrigerant circuit, the demineralized water circuit and the cooling water circuit. Figure 8 depicts a simplified flow diagram of the test facility. The main components of the refrigerant circuit, whose schematic layout is reported in Figure 9, are: the boiler, the flow metering sections, the mixer, the test section, the visualization apparatus, the condenser, the variable speed gear pump and the filter dryer. The boiler, positioned about 3 m above the test section, is a 58 dm³ stainless-steel pressure vessel of cylindrical shape with welded ellipsoidal heads provided with electrical cartridges placed in its lower part. Liquid and vapor are drawn from the boiler along two separate lines leading the fluids to the flowmeters. They are two, with a parallel arrangement, for each line: a Coriolis type and a variable area type (it is a backup system to handle mass flux larger than the operating range of the former, during the experiment that case never occurred and it remained unused). To assure a single-phase flow in the flowmeters the vapor is superheated by an electric resistance, wrapped around the line, while the liquid is subcooled by a concentric tube heat exchanger. The vapor line and the liquid line unite in the mixer and a two-phase flow enters in the test section, then in the visualization apparatus, in the end in the condenser. The last device is a pressure vessel of the same shape and size as the boiler and it is provided with coil heat exchangers in its upper part and with an

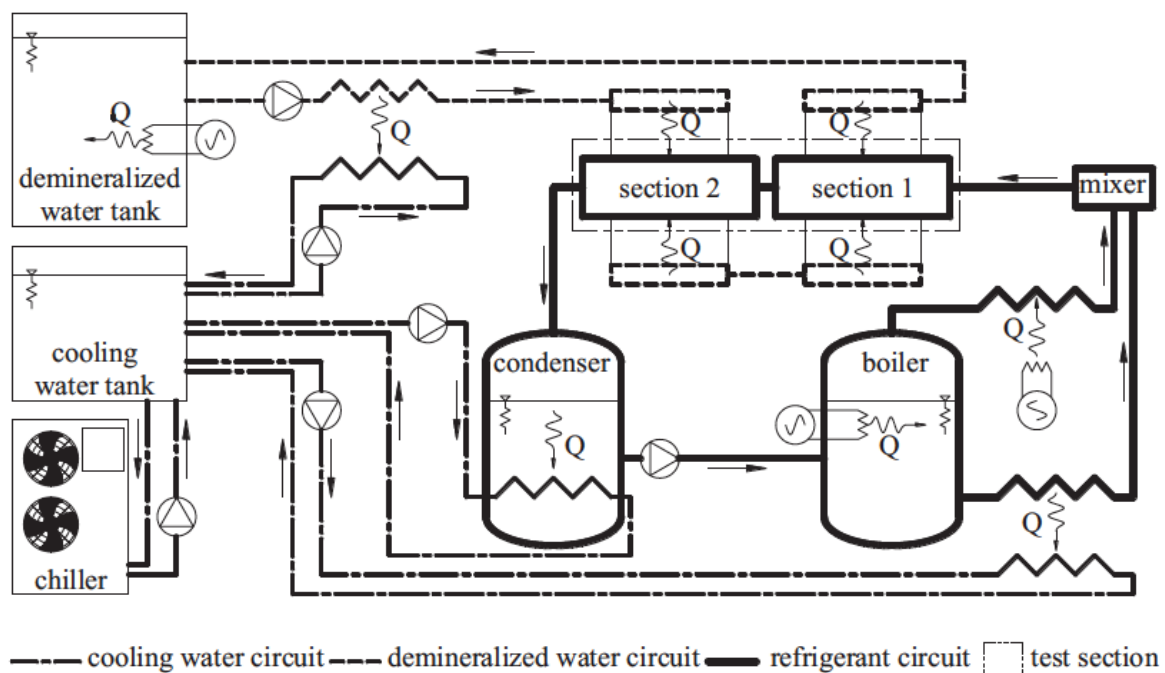


Figure 8: Experimental Set Up

electrical cartridge in its lower part, to set the test section outlet pressure at a fixed value.

A line conveys the liquid refrigerant into the variable speed gear pump whose suction is equipped with a concentric tube heat exchanger that cools the saturated liquid refrigerant and prevents cavitation. PID control systems are used for the cartridges placed in the boiler and in the condenser to control the pressure in these devices. To be sure of the operating condition the refrigerant temperature is measured upstream of the flow meters, downstream of the controlling valves, and at the inlet and outlet of the test section by thermocouples inserted in 60 mm long, 3 mm outer diameter, stainless steel L-shaped wells located on the duct axis. Rubber foam(10mmthick) is used for the thermal insulation of the refrigeration circuit.

The test section is made of three components in line, namely a calming section followed by two tube-in-tube heat exchangers with the refrigerant flowing in the inner tube and water flowing in a counterflow direction in the annulus. Each heat exchanger constitutes a subsection on which heat transfer and pressure drop measurements are performed. In between the subsection a short sight-glass, cylindrical in shape, was mounted, to visualize the flow pattern at this intermediate position. The inner diameter of the sight-glass is the same as the inner diameter of the heat exchangers tubes in order not to affect the flow inside the tubes.

Calming section and subsections are insulated by placing a 1 cm thick foam plastic annulus over the copper tube and 10 cm thick glass-wool annulus over the foam plastic annulus.

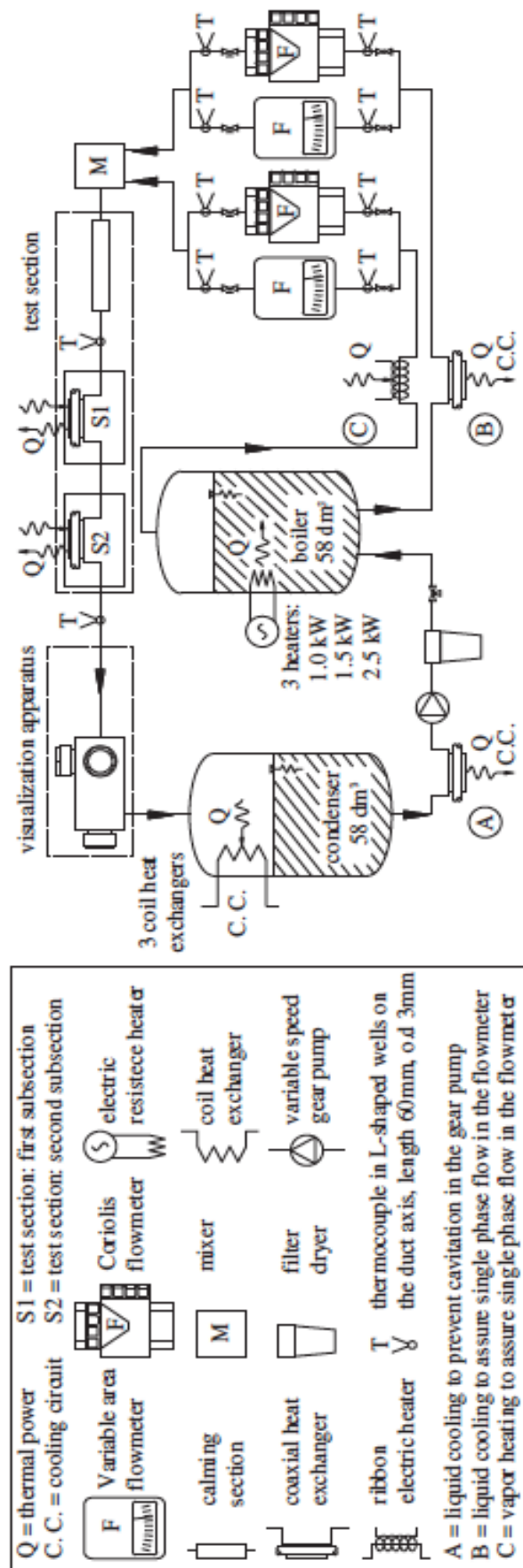


Figure 9: Refrigerant Circuit

The adiabatic calming section is a copper tube of inner diameter 8.92 mm and length 1.5 m, aimed to allow the flow to develop and to attain thermal equilibrium between the phases. The outer tube of the heat exchangers is made of copper and has inner diameter of 14 mm. The overall length of each exchanger is 1.3 m. However, the spacing between inlet and outlet of the demineralized water flowing in the annulus is 1.12 m, which is considered as the active length of heat exchange. The heat exchangers are connected upstream and downstream and without interruption of the inner tube via 4-way cross fittings. Two pressure taps are drilled in each junction, the gap between the inner surface of the fitting and the outer surface of the tube serves as a pressure annular chamber and a manifold connects them with the pressure transducers. The refrigerant inlet pressure is measured with an absolute strain-gauge transducer whereas pressure drop along the two heat exchangers is measured with two inductive differential pressure transducers. The refrigerant temperature is measured at the beginning of the first heat exchanger and at the end of the second one by means of K type thermocouple placed in stainless steel L-shaped wells (60 mm long, 3 mm outer diameter) located on the duct axis. However, because of lack of thermodynamic equilibrium, the reading of the thermocouple placed at the end of the test section, is not reliable and will not be used in the calculation of the heat transfer coefficients, as it will be explained later. Each heat exchanger is equipped with four T-type thermocouples to measure the wall temperature of the tested tube. They are placed in pairs at 50 mm from either heat transfer ends; in each pair, thermocouples are 180 degrees apart, the wires (0.15 mm diameter) are cemented in longitudinal grooves (80 mm long, 0.5 mm large and 0.2 mm deep) machined in the outside wall of the tube. Three couples of heat exchangers were used, namely a smooth tube and two microfine tubes. The microfine tubes are manufactured by Trefimetaux and named: Metofin 952-30VA40/54A (denoted as VA-tube), Metofin 952-45HVA40/82 (denoted as HVA-tube). Their main distinguishing feature from other microfine tubes is that fins alternate with two different heights. Between each other they differ essentially in the fin number and fin heights. The visualization apparatus is placed at the end of the test section and it is the assembly of an adiabatic connection tube and of a visualization chamber. The former is made by the same tube under test insulated by a 1 cm thick, foam plastic annulus on the copper tube, 10 cm thick, glass-wool annulus on the foam plastic annulus. Its ending part is inserted in the chamber and the discharge cross-section is cut along a vertical plane inclined of 45° with respect to the tube axis. The latter is a steel box (400 mm × 150 mm × 150 mm) with three windows 1 cm thick, two for illumination, and one for video recording with an high speed CCD camera (640 × 480 px, 200 Hz frame rate, 7.4 × 7.4 mm pixel size).

From there it enters a plate heat exchanger in counter flow with the chilled coolant, and then it is discharged into the tank. The volume flow rate of water is measured by an inductive flow meter. The bulk temperature of the water stream is measured by three thermal probes, located at the inlet of each subsection and at the end of the last one, respectively. Each probe consists of three T-type thermocouples cemented in three fine wells drilled in a copper cylinder with an outer diameter of 13 mm and a height of 20 mm. Such cylinder reduces the area of stream cross-section, thus promoting mixing and hence equalization of the liquid

temperature. Furthermore, since the thermocouples are connected in series, the measured voltage is proportional to the sum of the temperatures taken in three points of the cylinder. This measurement is considered representative of the bulk temperature of the flow.

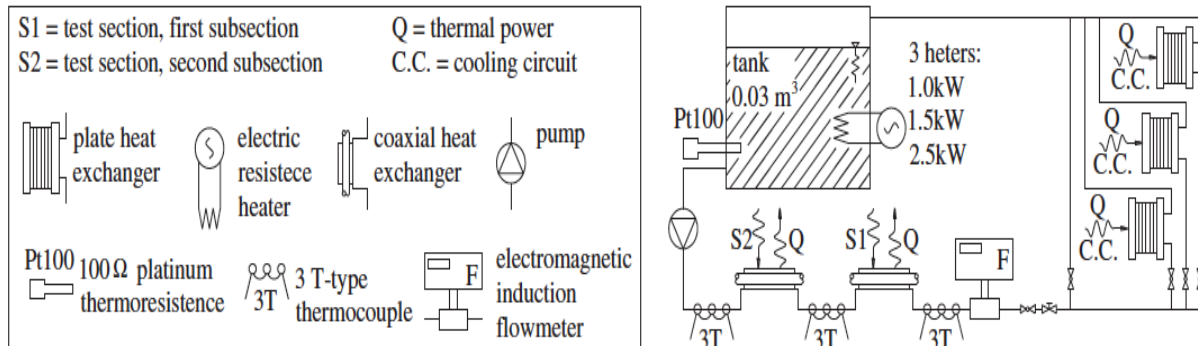


Figure 10: Demineralized Water Circuit

Finally, the chilled coolant circuit, represented in Figure 11, is filled with water e glycol solution and consists of a commercial refrigeration unit and a centrifugal pump. This circuit provides the cold fluid circulated in the heat exchangers placed in the refrigerant condenser or mounted on both the refrigerant and the water circuits.

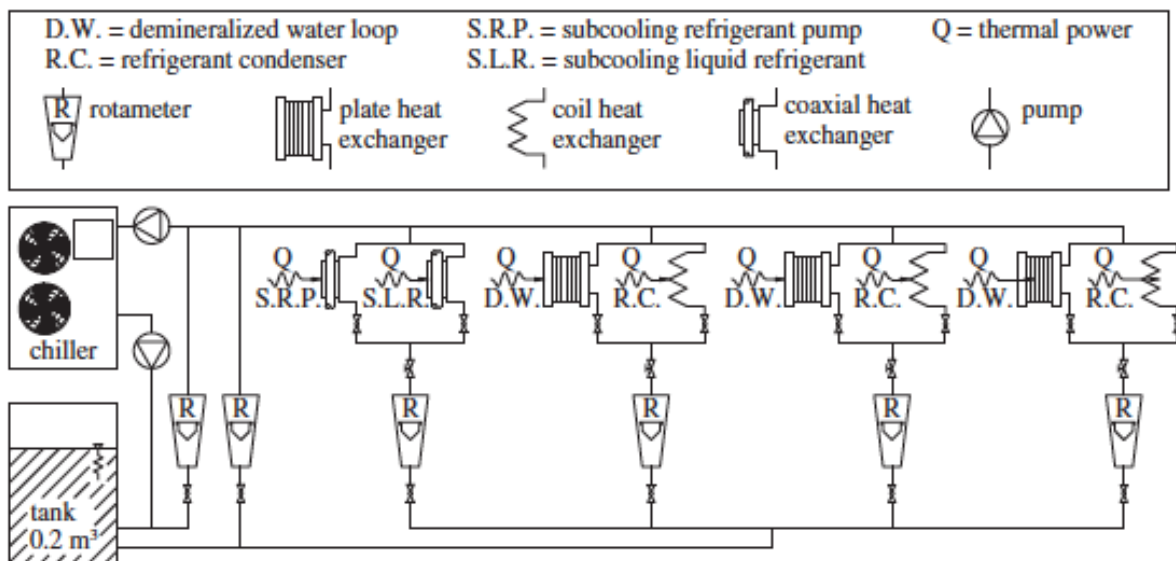


Figure 11: Cooling Circuit

Chapter 6

6 Heat Transfer Coefficient and Frictional Pressure Drop as a Function of G and x_m During Flow Boiling in Horizontal Tube

6.1 Smooth Tube- Counter Flow

6.1.1 Results for Heat Transfer Coefficient

This section aims to check the effects of the mass flux (G) and the mean quality (x_m) on the pressure drop and the heat transfer coefficient in the process of flow boiling. In Figure 12 heat transfer coefficient is given as a function of mass flux for three different conditions as it can be seen in Table 5. Here, Δx and T_{sat} are constant for all three cases while x_m and q vary in between.

Cases	Δx [-]	T_{sat} [°C]	x_m [-]	q [W/m ²]
Case 1	0.2	5	0.45	5500
Case 2	0.2	5	0.39	11000
Case 3	0.2	5	0.49	5500

Table 5: Properties of Figure 12

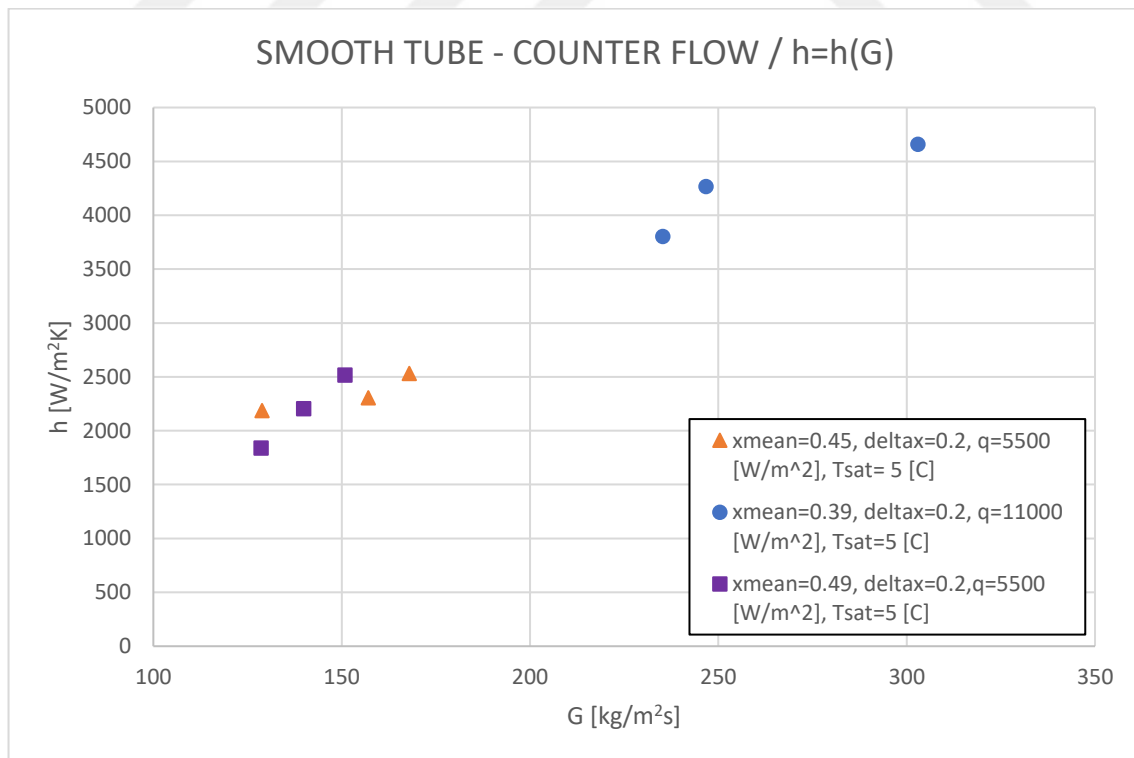


Figure 12: Smooth Tube - Counter Flow / $h=h(G)$

Based on the behavior of the orange triangles and purple squares in Figure 12, it is observed that the heat transfer coefficient tends to increase with increasing mass flux. The same behavior can also be obtained by observing the trend provided by the blue dots. It can even be said that the heat transfer coefficient is linearly variable with increasing mass flux.

Heat transfer coefficient is given as a function of mean quality x_m in Figure 13 for $G=150$ [kg/m²s], $\Delta x = 0.2$, $q=5600$ [W/m²] and $T_{sat}=5^\circ\text{C}$. There is an increasing trend for heat transfer coefficient as the mean quality increases.

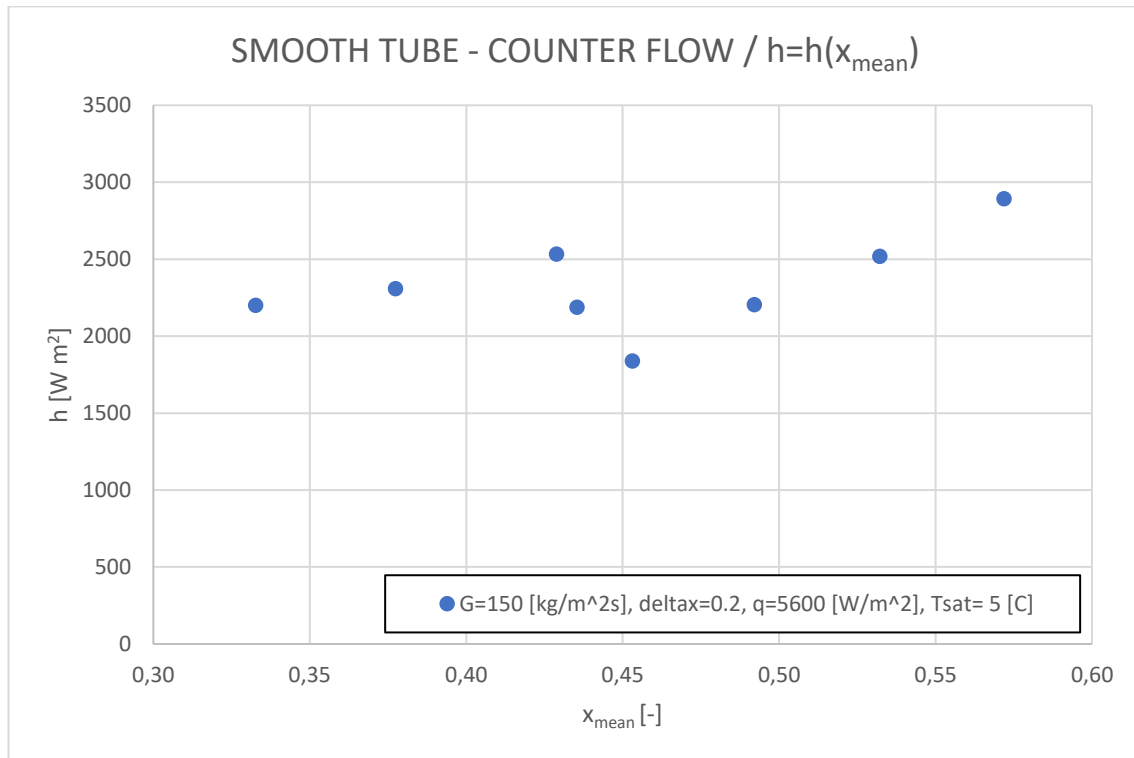


Figure 13: Smooth Tube - Counter Flow / $h=h(x_m)$

6.1.2 Results for Frictional Pressure Drop per Unit Length

The operating conditions for Figure 14 and Figure 15 are the same as they are for Figure 12 and Figure 13, respectively. This is also showed on the legends of these graphs. By observing, Figure 14 one can obtain that frictional pressure drop per unit length increases with increasing mass flux in all of the three cases. On the other hand, Figure 15 shows that frictional pressure drop per unit length is positively proportional with increasing mean quality.

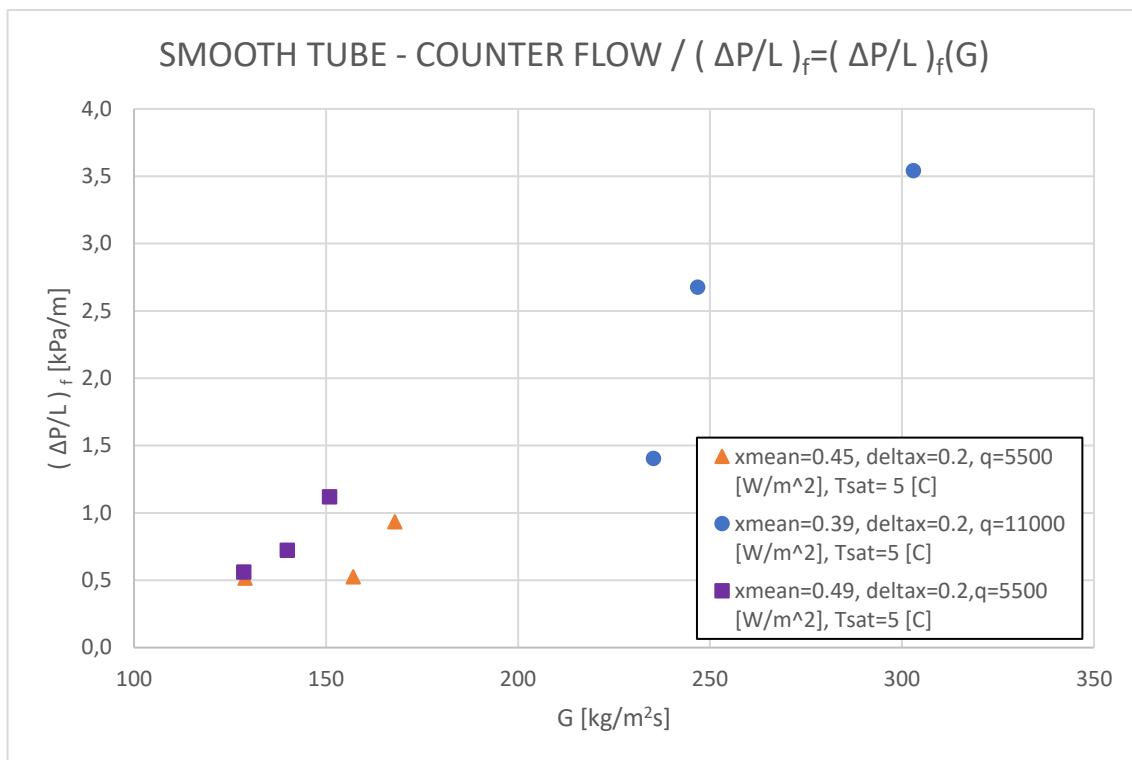


Figure 14: Smooth Tube - Counter Flow / $(\Delta P/L)_f = (\Delta P/L)_f(G)$

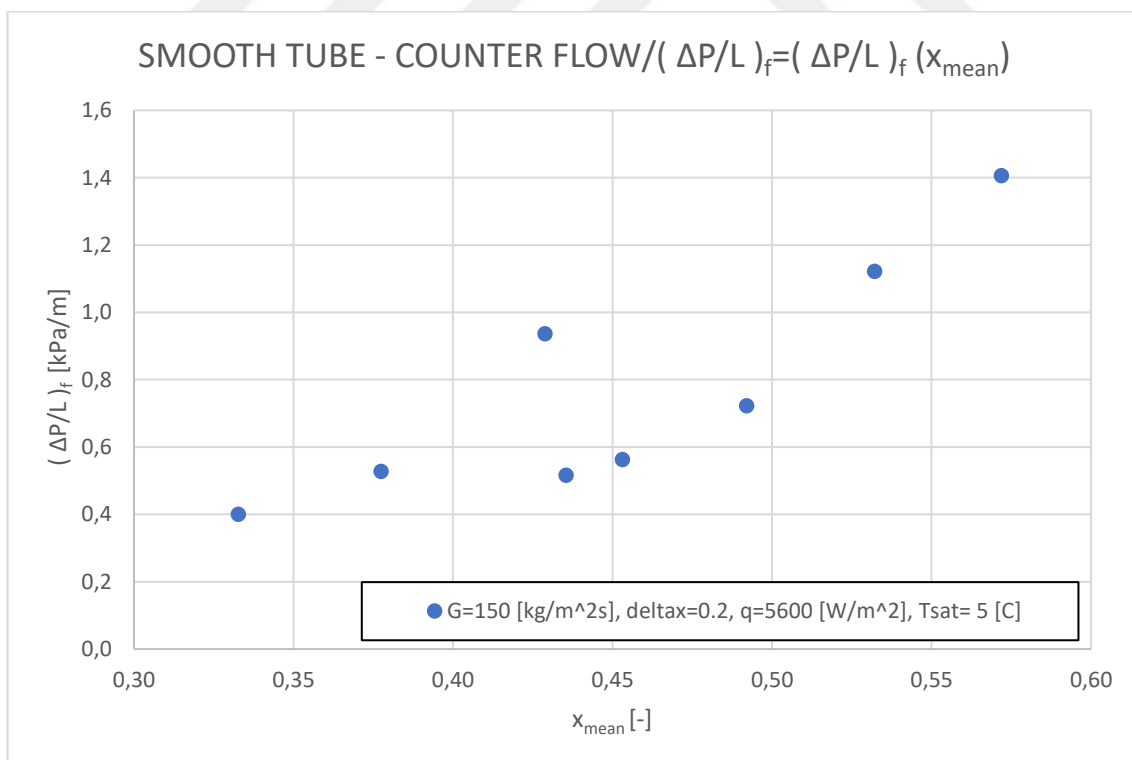


Figure 15: Smooth Tube - Counter Flow / $(\Delta P/L)_f = (\Delta P/L)_f(x_m)$

6.2 Tube V-Parallel Flow

This part aims to provide the behavior of both heat transfer coefficient and frictional pressure drop as a function of mass flux (G) and mean quality (x_m). Tube V contains only one type of fin on the inner walls of the tube with the following parameters in Table 6;

Properties	Value	Dimension
Number of Fin	60	[-]
Height	0,0002	[m]
Apex Angle	53	[degree]
Base	1,99E-04	[m]
Helix Angle	18	[degree]
Area of a Single Fin	1,99E-08	[m ²]
Total Fin Area	1,20E-06	[m ²]

Table 6: Properties of Tube V – Parallel Flow

6.2.1 Results for Heat Transfer Coefficient

In Figure 16 heat transfer coefficient is given as a function of mass flux. Here, $\Delta x = 0.3$, x_m is set to 0.45 while heat flux is 4000 [W/m²]. Moreover, the saturation temperature $T_{sat} = 10^\circ\text{C}$ for all three cases. For this type of micro-fin tube heat transfer coefficient shows an increase with increasing mass flux. This is also the theoretical expected behavior.

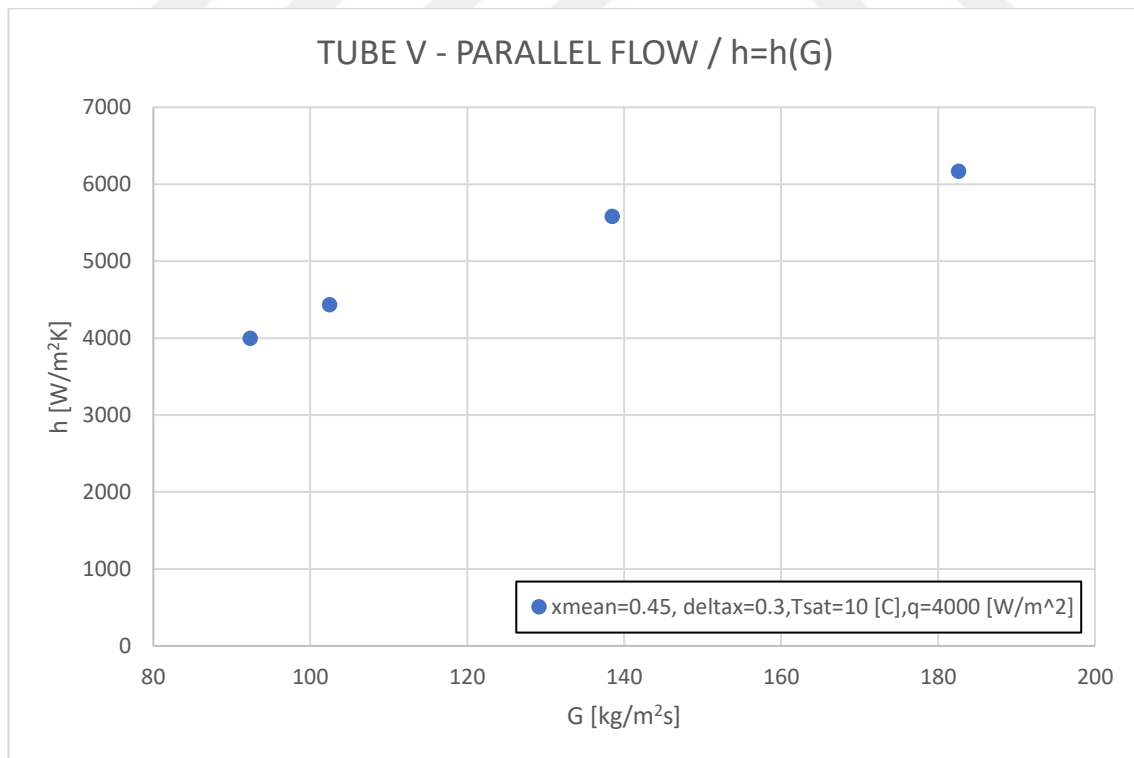


Figure 16: Tube V - Parallel Flow / $h=h(G)$

Heat transfer coefficient is given as a function of mean quality x_m in Figure 17 for $G=220$ [kg/m²s], $\Delta x = 0.3$, $q=8300$ [W/m²] and $T_{sat}=10^\circ\text{C}$. The chart shows an increasing trend for heat transfer coefficient as the mean quality increases, while the dry out phenomenon was not observed in the explored range of mean qualities.

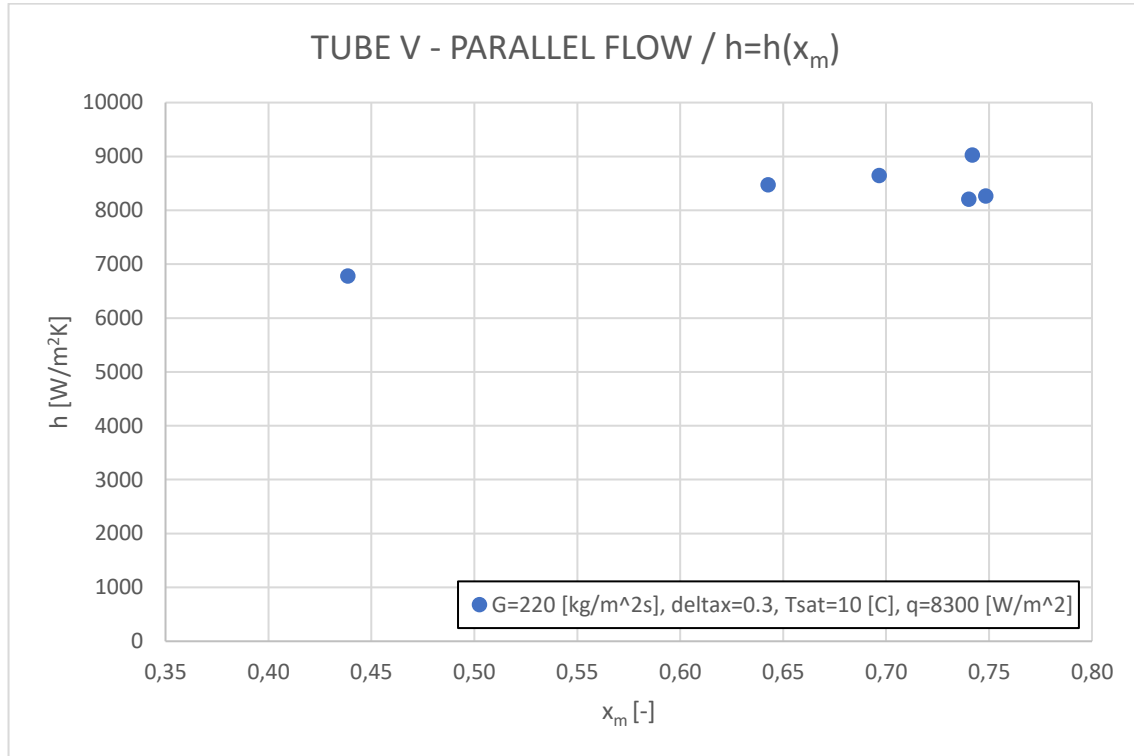


Figure 17: Tube V - Parallel Flow / $h=h(x_m)$

6.2.2 Results for Frictional Pressure Drop per Unit Length

The operating conditions for Figure 18 and Figure 19 are the same as they are for Figure 16 and Figure 17, respectively. This is also showed on the legends of these graphs. By observing, Figure 18 one can obtain that frictional pressure drop per unit length increases with increasing mass flux. Figure 19 shows that frictional pressure drop per unit length is positively proportional with increasing mean quality even if there is a slight mismatch in the last two data.

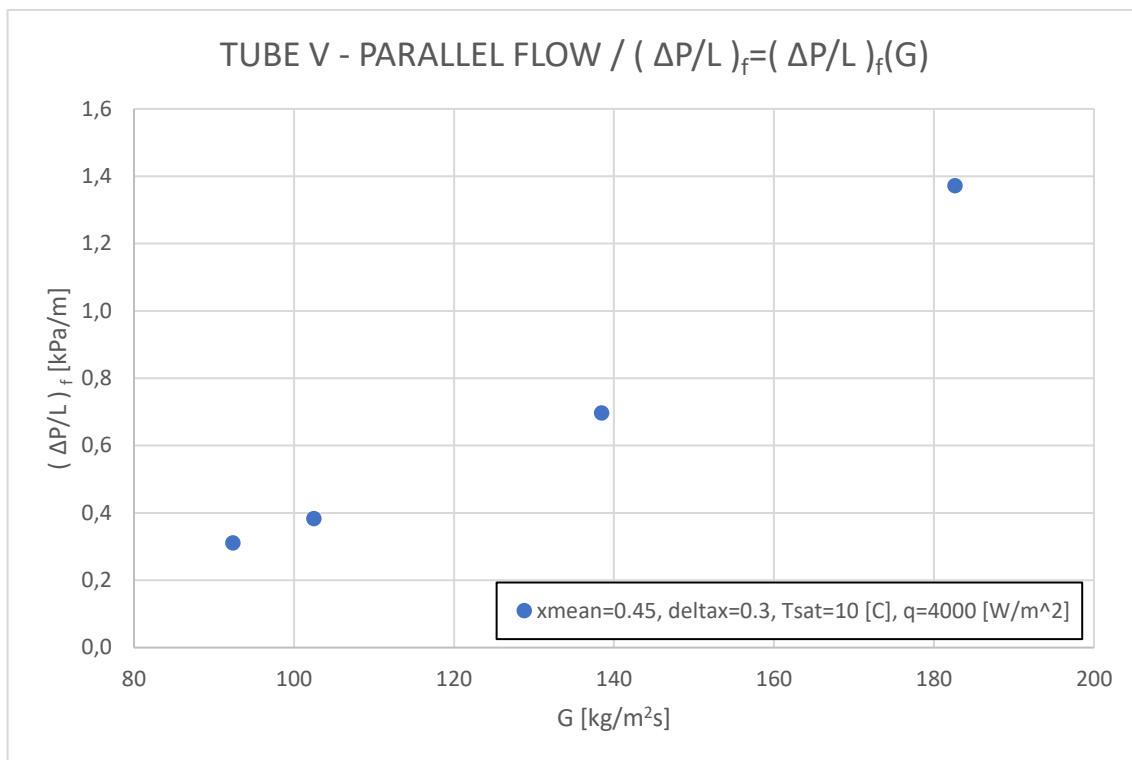


Figure 18: Tube V - Parallel Flow / $(\Delta P/L)_f = (\Delta P/L)_f(G)$

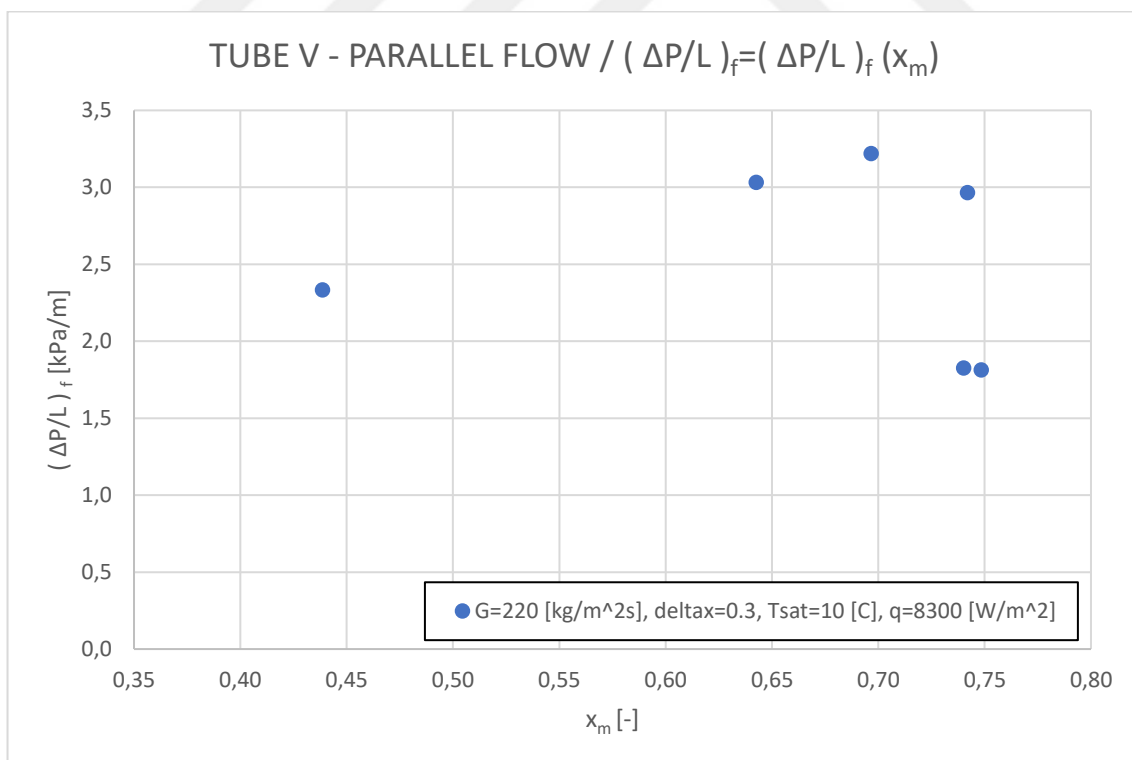


Figure 19: Tube V - Parallel Flow / $(\Delta P/L)_f = (\Delta P/L)_f(x_m)$

6.3 Tube V - Counter Flow

This part aims to provide the behavior of both heat transfer coefficient and frictional pressure drop as a function of mass flux (G) and mean quality (x_m) in the process of flow boiling. Tube V contains only one type of fin on the inner walls of the tube with the following parameters in Table 7;

Properties	Value	Dimension
Number of Fin	60	[-]
Height	0,0002	[m]
Apex Angle	53	[degree]
Base	1,99E-04	[m]
Helix Angle	18	[degree]
Area of a Single Fin	1,99E-08	[m ²]
Total Fin Area	1,20E-06	[m ²]

Table 7: Properties of Tube V – Counter Flow

6.3.1 Results for Heat Transfer Coefficient

In Figure 20 heat transfer coefficient is given as a function of mass flux. Here, $\Delta x = 0.3$, x_m is set to 0.45 while heat flux is 13500 [W/m²]. Moreover, the saturation temperature $T_{sat} = 5^\circ\text{C}$ for all three cases. For this type of micro-fin tube heat transfer coefficient shows an increase with increasing mass flux. This is also the theoretical expected behavior.

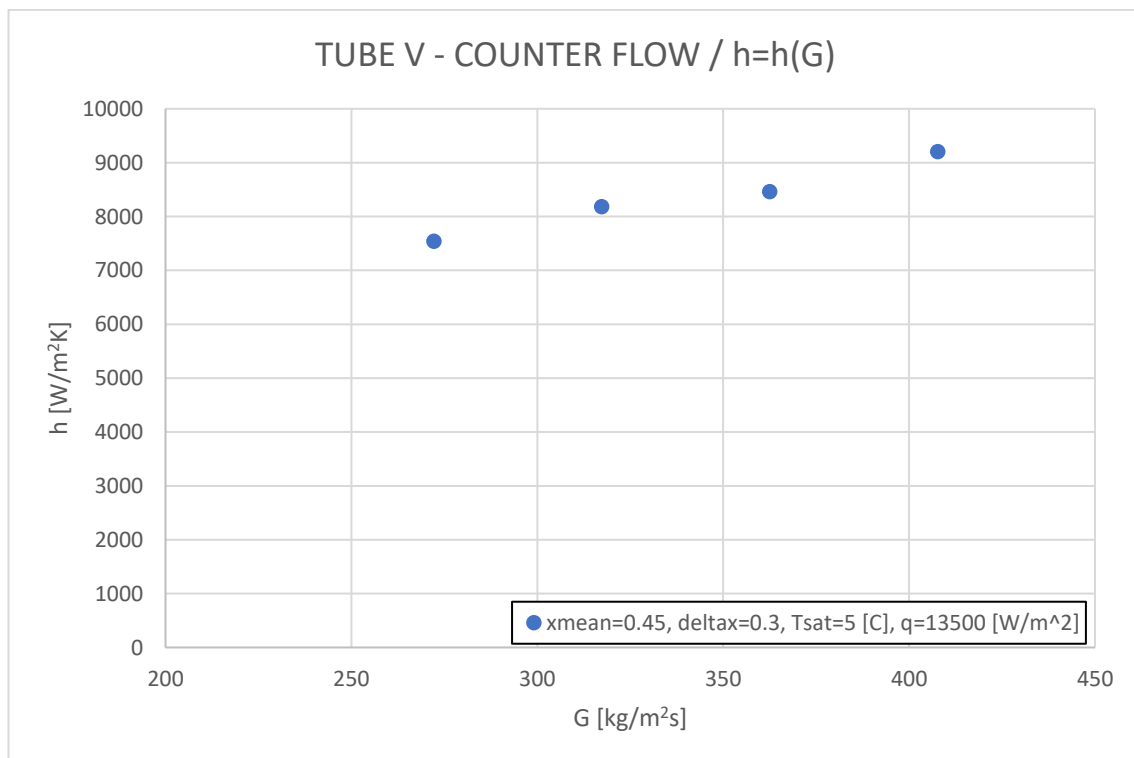


Figure 20: Tube V - Counter Flow / $h=h(G)$

Heat transfer coefficient is given as a function of mean quality x_m in Figure 21 for $G=222$ [kg/m²s], $\Delta x = 0.3$, $q=8500$ [W/m²] and $T_{sat}=5^\circ\text{C}$. The heat transfer coefficient increases as the mean quality increases till $x_m=0.8$ where it seems the dry out phenomenon takes place.

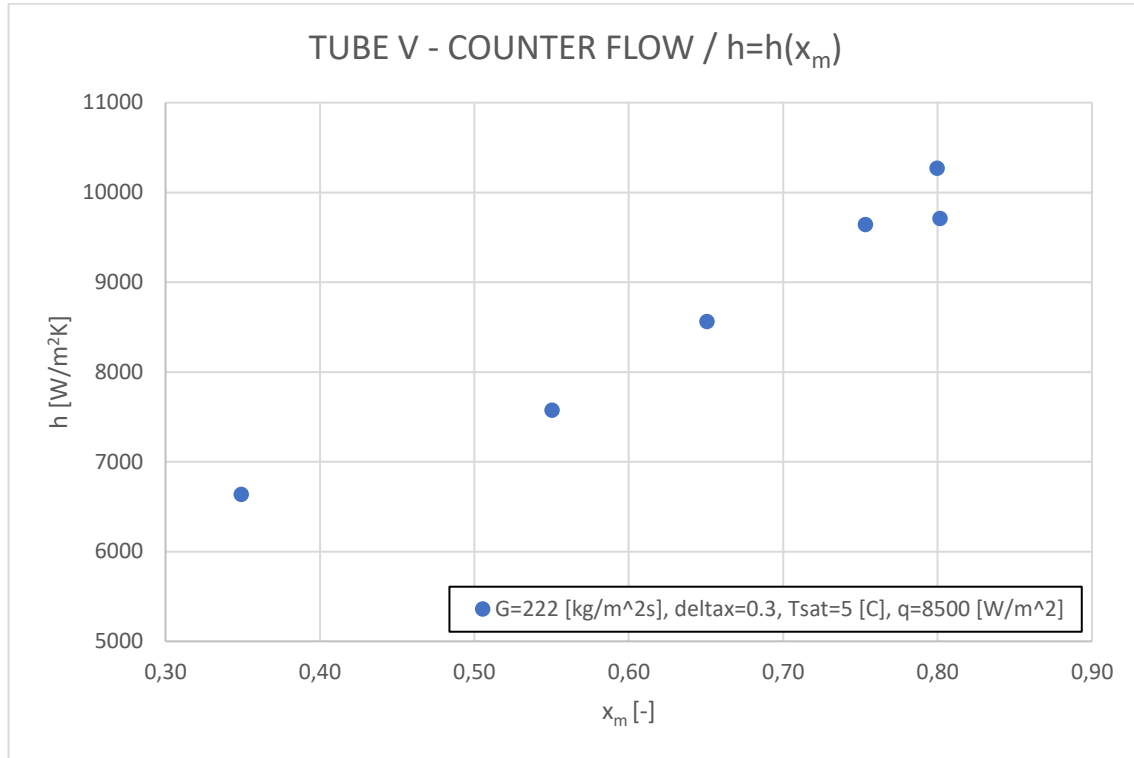


Figure 21: Tube V - Counter Flow / $h=h(x_m)$

6.3.2 Results for Frictional Pressure Drop per Unit Length

The operating conditions for Figure 22 and Figure 23 are the same as they are for Figure 20 and Figure 21, respectively. This is also showed on the legends of these graphs. By observing, Figure 22 one can obtain that frictional pressure drop per unit length increases with increasing mass flux. Figure 23 shows that frictional pressure drop per unit length is positively proportional with increasing mean quality even if there is a slight mismatch in the last data.

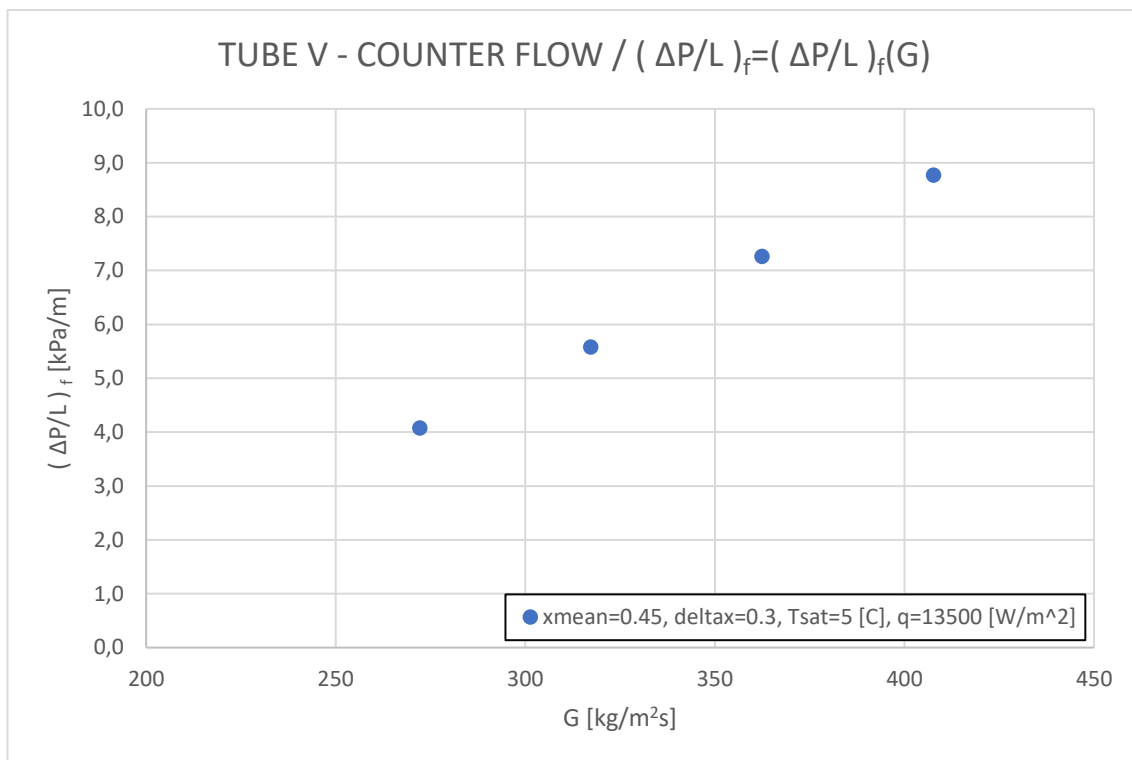


Figure 22: Tube V - Counter Flow / $(\Delta P/L)_f = (\Delta P/L)_f(G)$

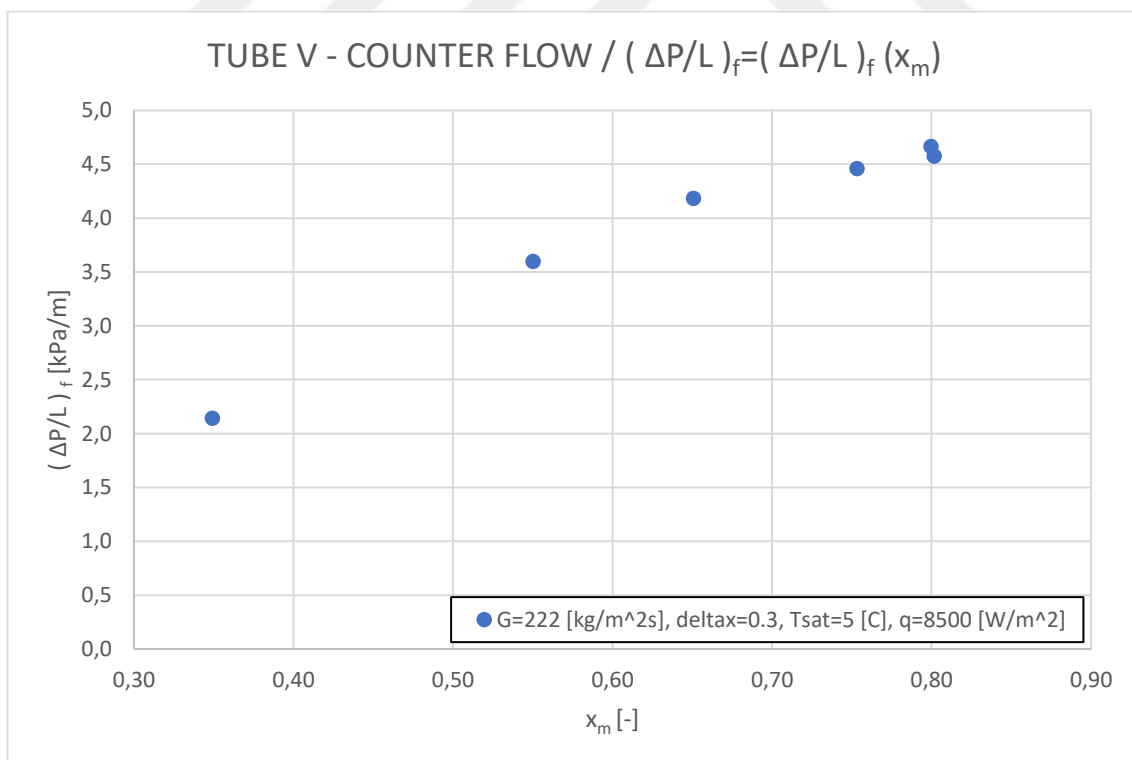


Figure 23: Tube V - Counter Flow / $(\Delta P/L)_f = (\Delta P/L)_f(x_m)$

6.4 Tube W - Counter Flow

This part aims to provide the behavior of both heat transfer coefficient and frictional pressure drop as a function of mass flux (G) and mean quality (x_m) in the process of flow boiling. Tube W contains only one type of fin on the inner walls of the tube with the following parameters in Table 8;

Properties	Value	Dimension
Number of Fin	65	[-]
Height	0.00015	[m]
Apex Angle	90	[degree]
Base	3.00E-04	[m]
Helix Angle	25	[degree]
Area of a Single Fin	2.25E-08	[m ²]
Total Fin Area	1.46E-06	[m ²]

Table 8: Properties of Tube W – Counter Flow

6.4.1 Results for Heat Transfer Coefficient

In Figure 24 heat transfer coefficient is given as a function of mass flux. Here, $\Delta x = 0.3$, x_m is 0.45 while heat flux values are 7000 [W/m²] and 14000 [W/m²]. Moreover, the saturation temperature $T_{sat} = 5^\circ\text{C}$ for all cases. For this type of micro-fin tube heat transfer coefficient shows an increase with increasing mass flux.

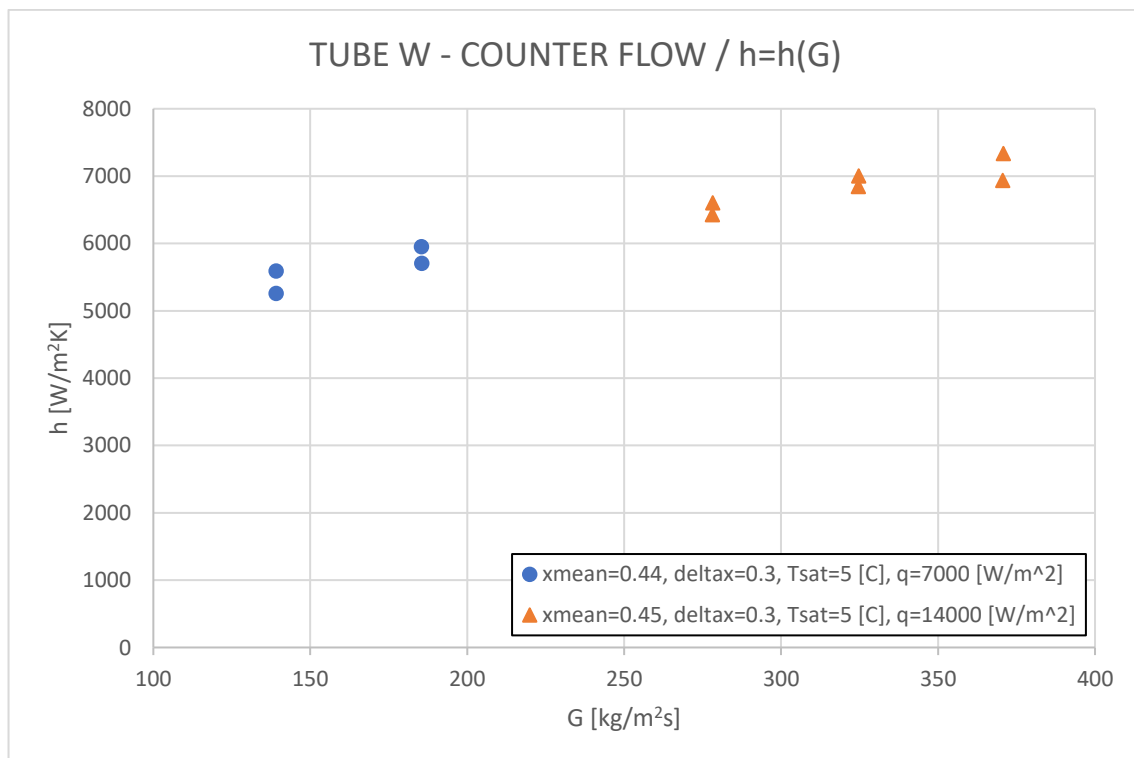


Figure 24: Tube W - Counter Flow / $h=h(G)$

Heat transfer coefficient is given as a function of mean quality x_m in Figure 25 for $G=232$ [kg/m²s], $\Delta x = 0.3$, $q=10000$ [W/m²] and $T_{sat}=5^\circ\text{C}$. there is an increasing trend for heat transfer coefficient as the mean quality increases. There are some mismatched data, but it is acceptable because the trend is going up clearly according to Figure 25.

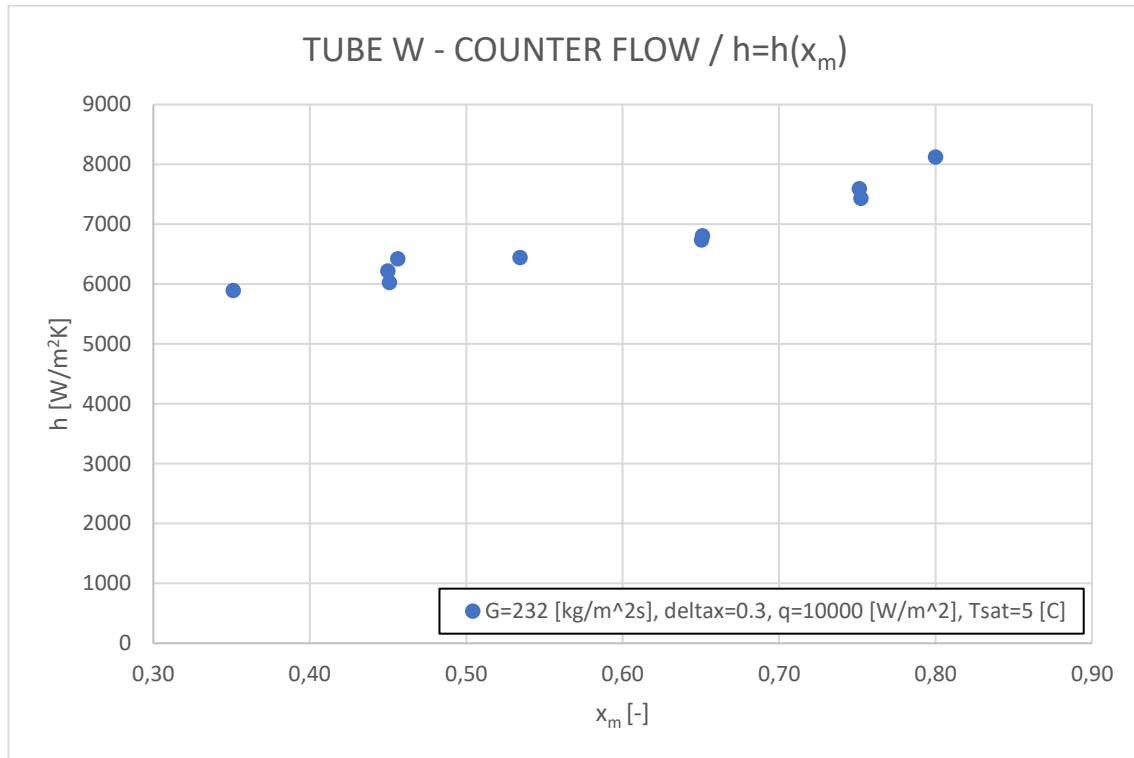


Figure 25: Tube W - Counter Flow / $h=h(x_m)$

6.4.2 Results for Frictional Pressure Drop per Unit Length

The operating conditions for Figure 26 and Figure 27 are the same as they are for Figure 24 and Figure 25, respectively. This is also showed on the legends of these graphs. By observing, Figure 26 one can obtain that frictional pressure drop per unit length increases with increasing mass flux. Figure 27 shows that frictional pressure drop per unit length is positively proportional with increasing mean quality even if there is a slight mismatch in the last data.

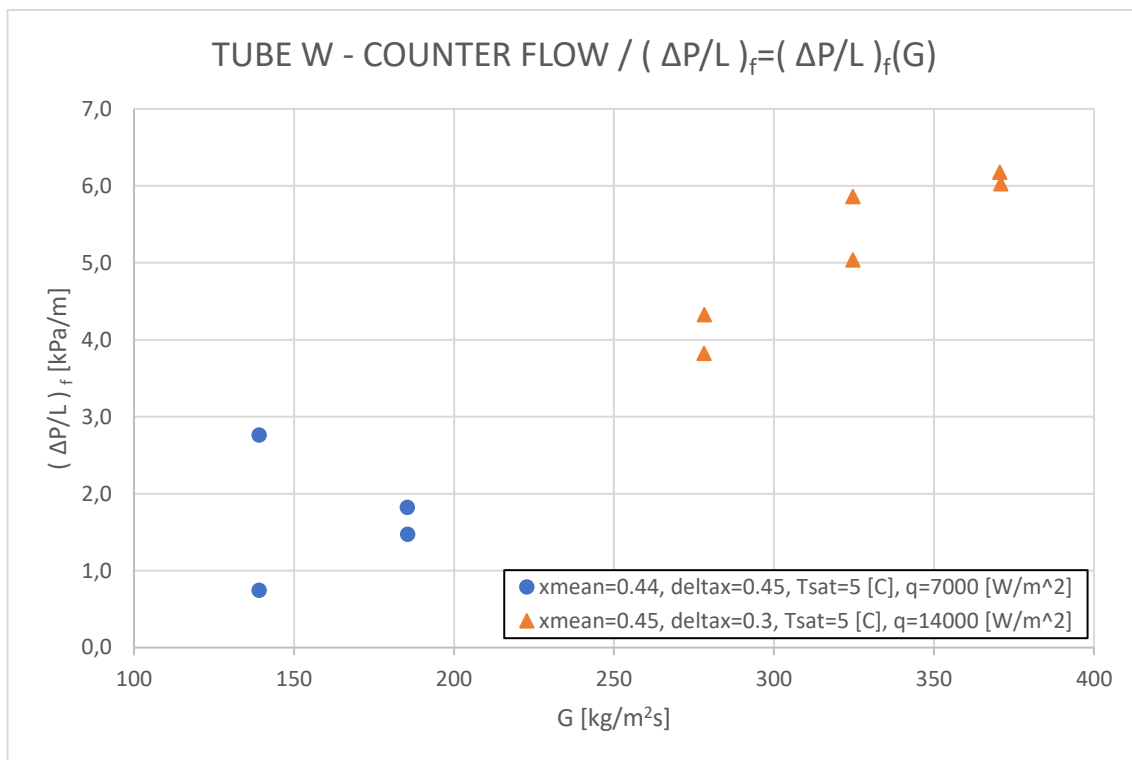


Figure 26: Tube W - Counter Flow / $(\Delta P/L)_f = (\Delta P/L)_f(G)$

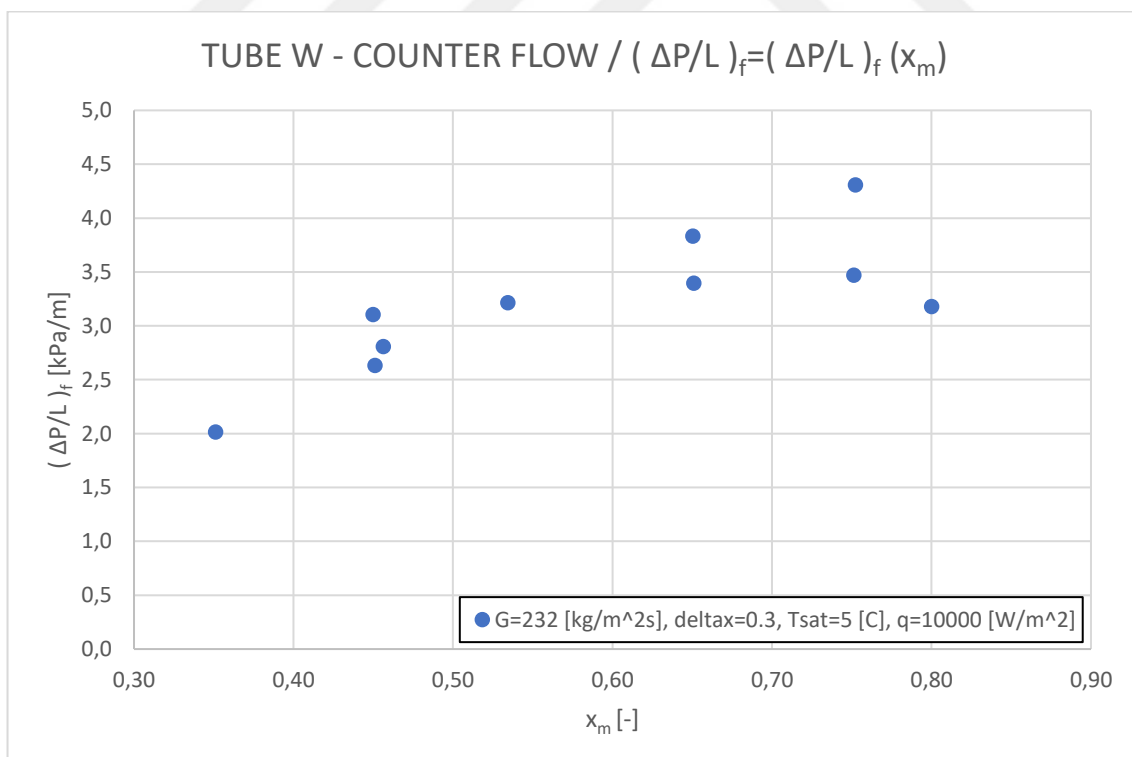


Figure 27: Tube W - Counter Flow / $(\Delta P/L)_f = (\Delta P/L)_f(x_m)$

6.5 Tube VA - Counter Flow

This part aims to provide the behavior of both heat transfer coefficient and frictional pressure drop as a function of mass flux (G) and mean quality (x_m) in the process of flow boiling. Tube VA contains two types of fin on the inner walls of the tube with the following parameters in Table 9 and Table 10;

Properties	Value	Dimension
Number of Fin	27	[-]
Height	0.00023	[m]
Apex Angle	40	[degree]
Base	1.67E-04	[m]
Helix Angle	18	[degree]
Area of a Single Fin	1.93E-08	[m ²]
Total Fin Area	5.20E-07	[m ²]

Table 9: Properties of Tube VA – Counter Flow Fin Type 1

Properties	Value	Dimension
Number of Fin	27	[-]
Height	0.00016	[m]
Apex Angle	40	[degree]
Base	1.16E-04	[m]
Helix Angle	18	[degree]
Area of a Single Fin	9.32E-09	[m ²]
Total Fin Area	2.52E-07	[m ²]

Table 10: Properties of Tube VA – Counter Flow Fin Type 2

6.5.1 Results for Heat Transfer Coefficient

In Figure 28 heat transfer coefficient is given as a function of mean quality for three different conditions as it can be seen in Table 11. Here, T_{sat} is constant for all four cases while G and q vary in between. Also, Δx is constant for three cases.

Cases	Δx [-]	T_{sat} [°C]	G [kg/m ² s]	q [W/m ²]
Case 1	0.3	5	225	8600
Case 2	0.2	5	150	4000
Case 3	0.2	5	200	5000
Case 4	0.2	5	320	8000

Table 11: Properties of Figure 28

There is an increasing trend for heat transfer coefficient as the mean quality increases, the experiments did not show the dry out phenomenon.

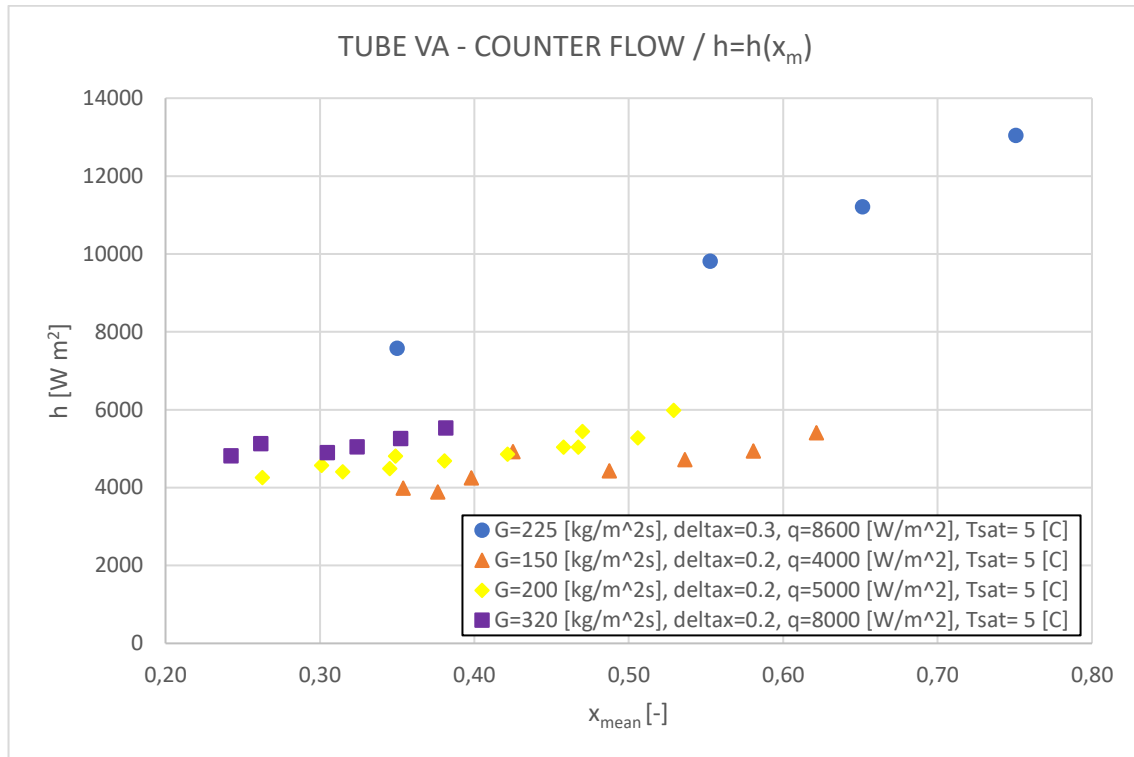


Figure 28: Tube VA - Counter Flow / $h=h(x_m)$

6.5.2 Results for Frictional Pressure Drop per Unit Length

The operating conditions for Figure 30 is the same as they are for Figure 28. In Figure 29 frictional pressure drop per unit length is given as a function of mass flux for two different conditions as it can be seen in Table 12. Here, T_{sat} , Δx and q are constant for all both cases while x_m vary in between.

Cases	Δx [-]	T_{sat} [$^{\circ}\text{C}$]	x_m [-]	q [W/m^2]
Case 1	0.2	5	0.23	9100
Case 2	0.2	5	0.28	9100

Table 12: Properties of Figure 29

This is also showed on the legends of these graphs. By observing, Figure 29 one can obtain that frictional pressure drop per unit length increases with increasing mass flux. Figure 30 shows that frictional pressure drop per unit length is positively proportional with increasing mean quality even if there is a slight mismatch data.

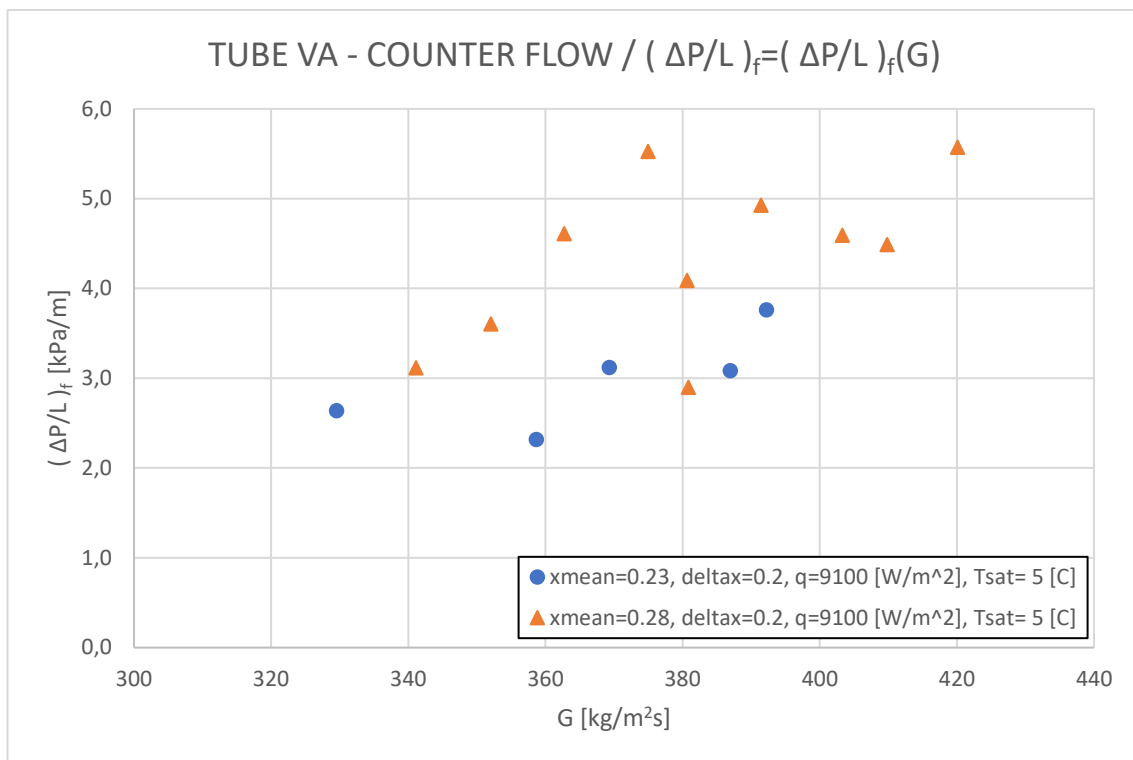


Figure 29: Tube VA - Counter Flow / $(\Delta P/L)_f = (\Delta P/L)_f(G)$

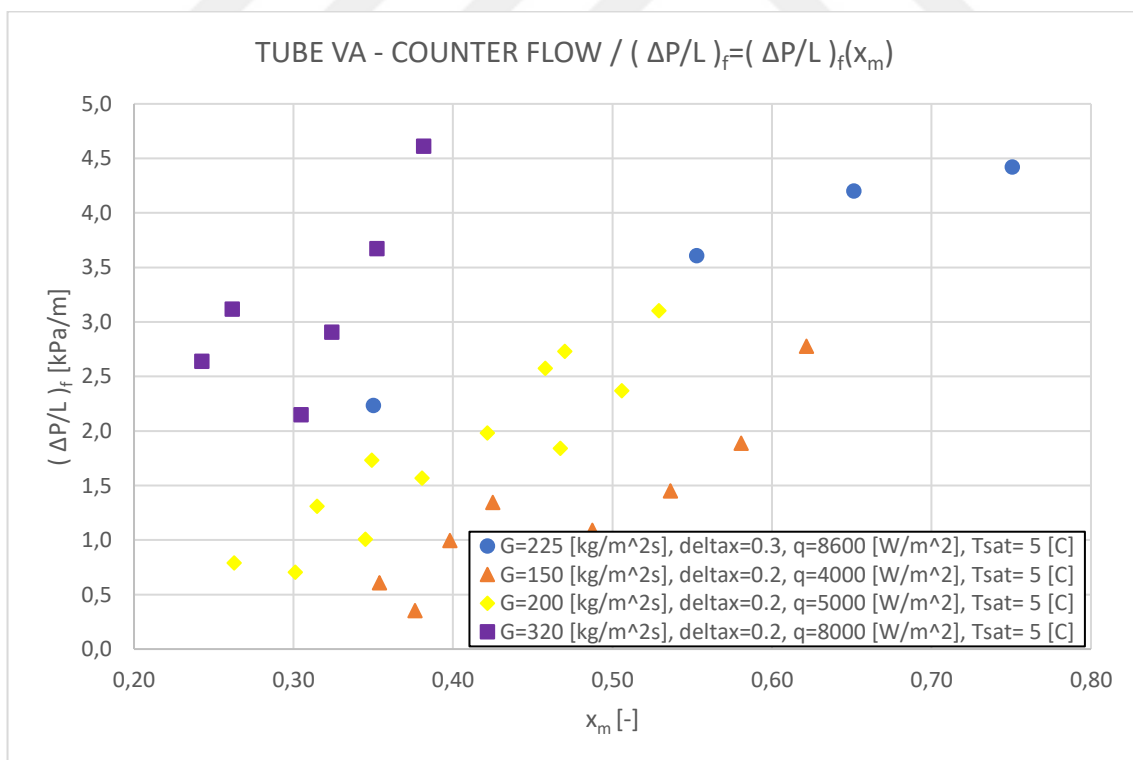


Figure 30: Tube VA - Counter Flow / $(\Delta P/L)_f = (\Delta P/L)_f(x_m)$

6.6 Tube HVA - Counter Flow

This part aims to provide the behavior of both heat transfer coefficient and frictional pressure drop as a function of mass flux (G) and mean quality (x_m) in the process of flow boiling. Tube HVA contains two types of fin on the inner walls of the tube with the following parameters in Table 13 and Table 14;

Properties	Value	Dimension
Number of Fin	41	[-]
Height	0.0002	[m]
Apex Angle	40	[degree]
Base	1.46E-04	[m]
Helix Angle	18	[degree]
Area of a Single Fin	1.46E-08	[m ²]
Total Fin Area	5.97E-07	[m ²]

Table 13: Properties of Tube HVA – Counter Flow Fin Type 1

Properties	Value	Dimension
Number of Fin	41	[-]
Height	0.00017	[m]
Apex Angle	40	[degree]
Base	1.24E-04	[m]
Helix Angle	18	[degree]
Area of a Single Fin	1.05E-08	[m ²]
Total Fin Area	4.31E-07	[m ²]

Table 14: Properties of Tube HVA – Counter Flow Fin Type 2

6.6.1 Results for Heat Transfer Coefficient

In Figure 31 heat transfer coefficient is given as a function of mass flux. Here, $\Delta x = 0.3$, x_m value is set to 0.45 while heat flux is 11000 [W/m²] for two cases. Moreover, the saturation temperature $T_{sat} = 5^\circ\text{C}$ for all cases.

For this type of micro-fin tube heat transfer coefficient shows an increase with increasing mass flux. This is also the theoretical expected behavior and it is easy to see this trend in the Figure 31.

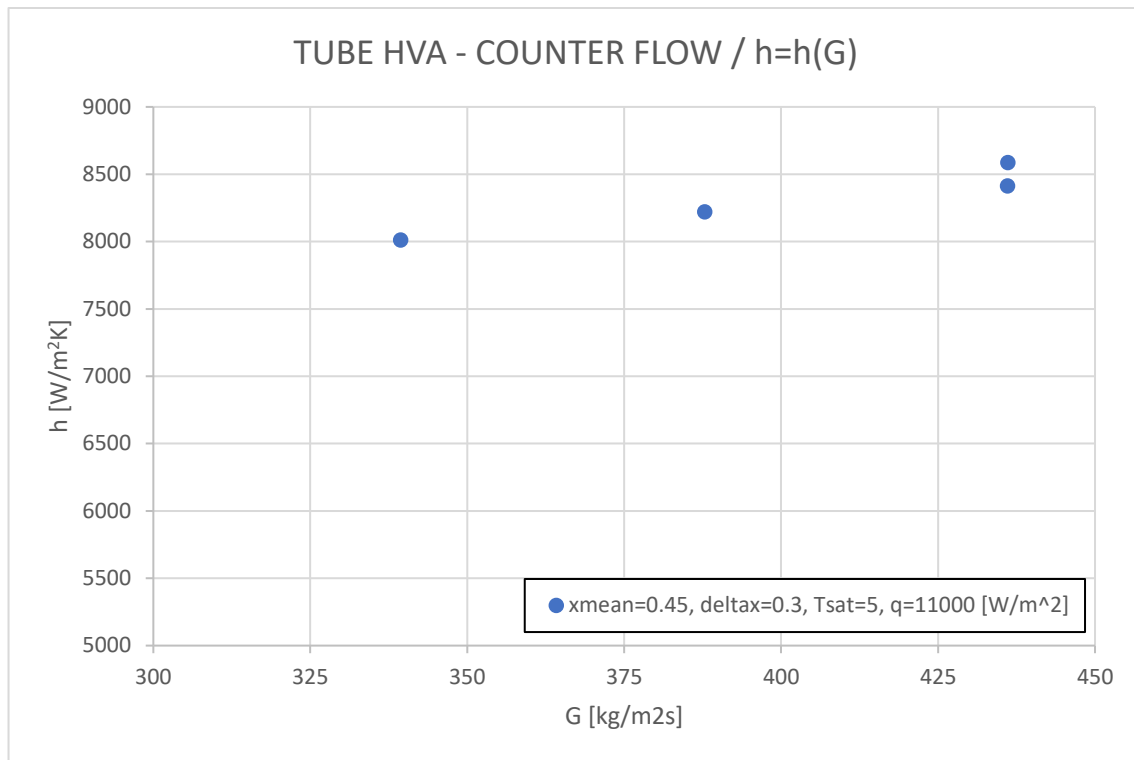


Figure 31: Tube HVA - Counter Flow / $h=h(G)$

Heat transfer coefficient is given as a function of mean quality x_m in Figure 32 for two different conditions as it can be seen in Table 15. Here, T_{sat} and Δx are constant for both cases while G and q vary in between. There are also some mismatched data but, it is acceptable because these data do not affect the increase trend.

Cases	Δx [-]	T_{sat} [°C]	G [kg/m²s]	q [W/m²]
Case 1	0.3	5	145	4500
Case 2	0.3	5	242	7500

Table 15: Properties of Figure 32

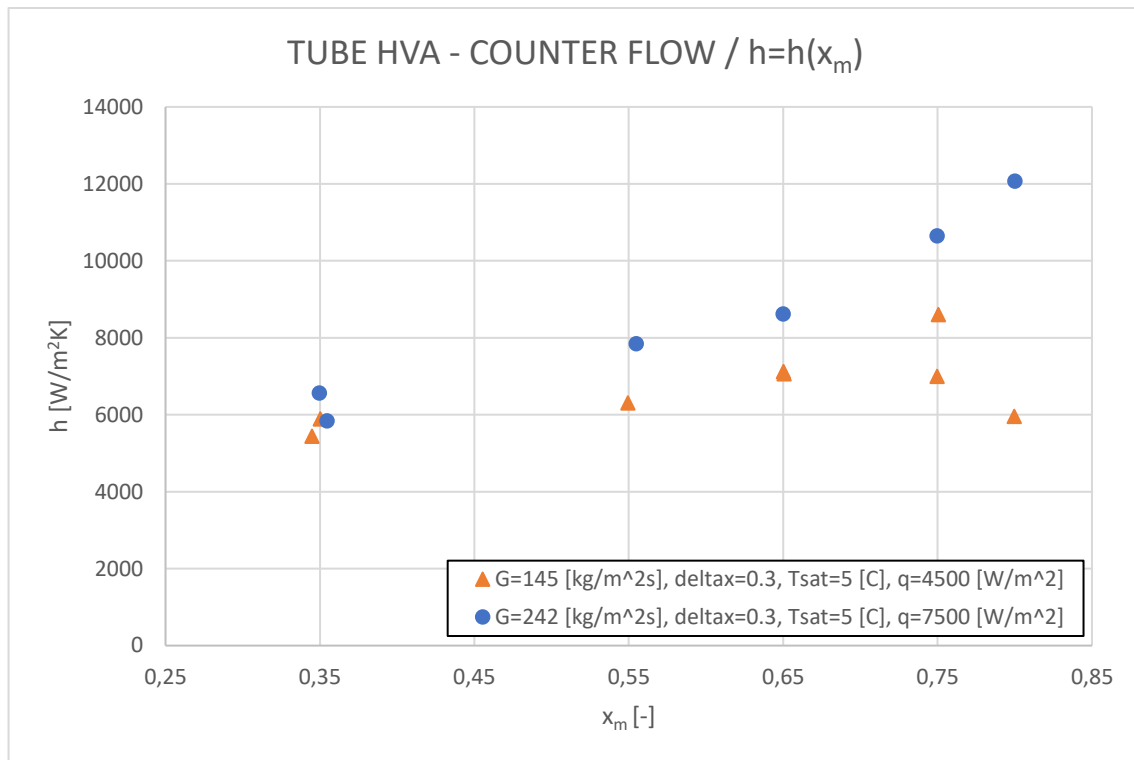


Figure 32:Tube HVA - Counter Flow / $h=h(x_m)$

6.6.2 Results for Frictional Pressure Drop per Unit Length

The operating conditions for Figure 33 and Figure 34 are the same as they are for Figure 31 and Figure 32, respectively. This is also showed on the legends of these graphs. By observing, Figure 33 one can obtain that frictional pressure drop per unit length increases with increasing mass flux. Figure 34 shows that frictional pressure drop per unit length is positively proportional with increasing mean quality even if there is a slight mismatch data.

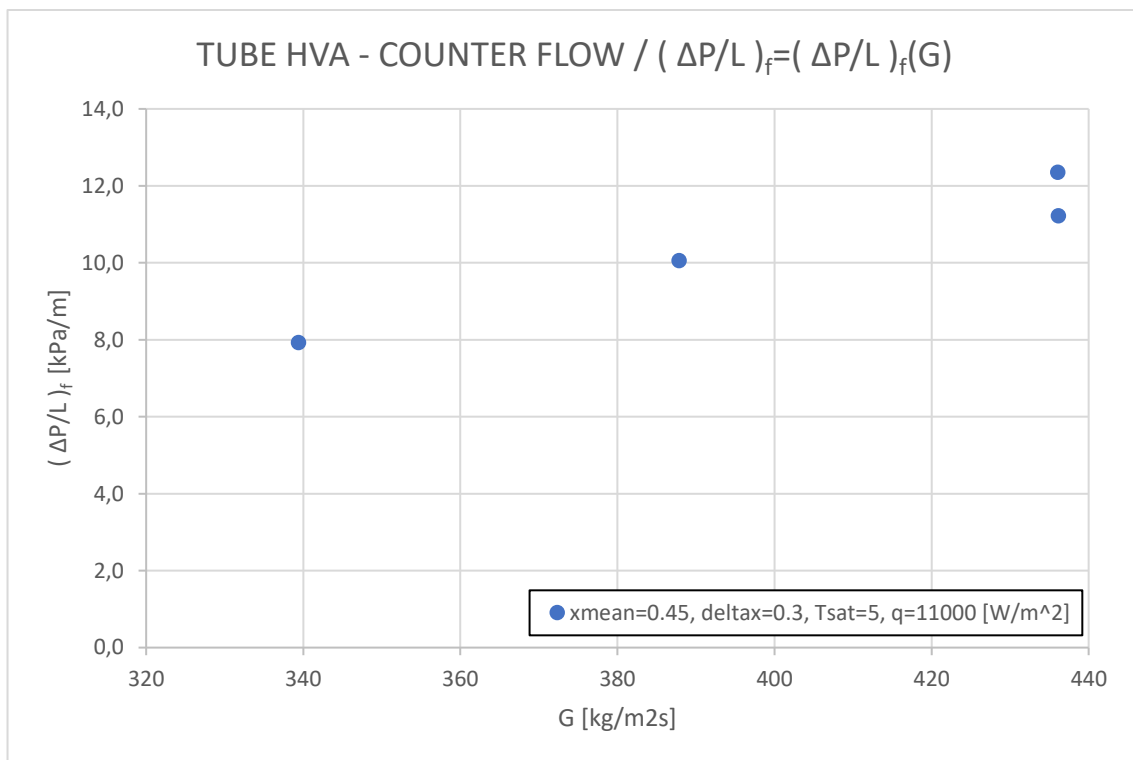


Figure 33: Tube HVA - Counter Flow / $(\Delta P/L)_f = (\Delta P/L)_f(G)$

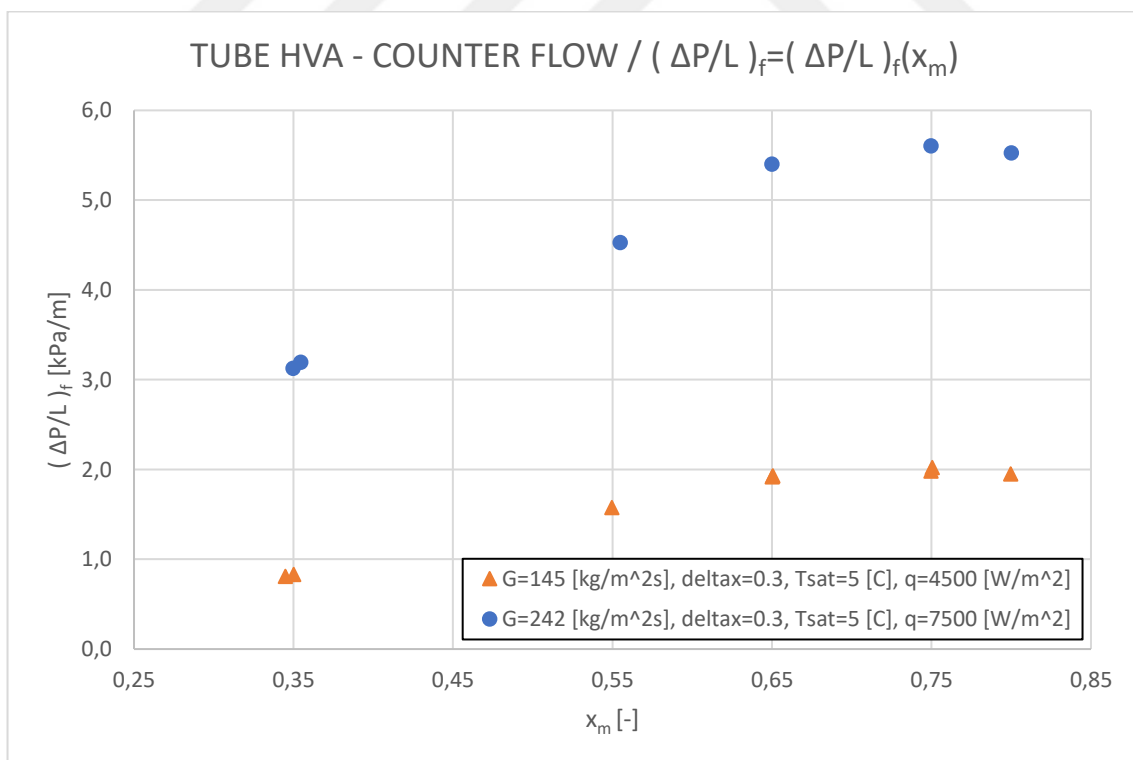


Figure 34: Tube HVA - Counter Flow / $(\Delta P/L)_f = (\Delta P/L)_f(x_m)$

The Shah diagram can be used in order to see four regimes of boiling are shown below in Figure 35 and Figure 36. At $Co > 0.1$ is the nucleate boiling regime in which Ψ is independent of Co and depends only Bo [26].

$$\Psi = 1.8/Co^{0.8} \quad \text{Eqn 172}$$

In this regime, the two-phase convective effects are negligible and heat transfer enhancement factor is calculated only by intensity of bubble nucleation [26]. The Eqn 172 is shown above, it represents the condition of pure convective boiling in which bubble nucleation has been completely suppressed with the tube surface fully wet [7]. The surface is fully wet for vertical tubes at any value of Fr_l , while for horizontal tubes, the surface is fully wet only if $Fr_l \geq 0.04$. For horizontal tubes with $Fr_l < 0.04$, part of the tube surface is dry, and the heat transfer coefficient is lower than in vertical tubes [26]. In between the convective boiling line which is calculated by using the Eqn 172 and the nucleate boiling regime is the bubble suppression regime in which both bubble nucleation and convective effects are important. With decreasing values of Co , in this case, increasing vapor quality, bubble nucleation is more and more suppressed until it completely disappears at line. In shortly, the data taken for all tube types are included in convective boiling with partly dry surface according to Shah diagram.

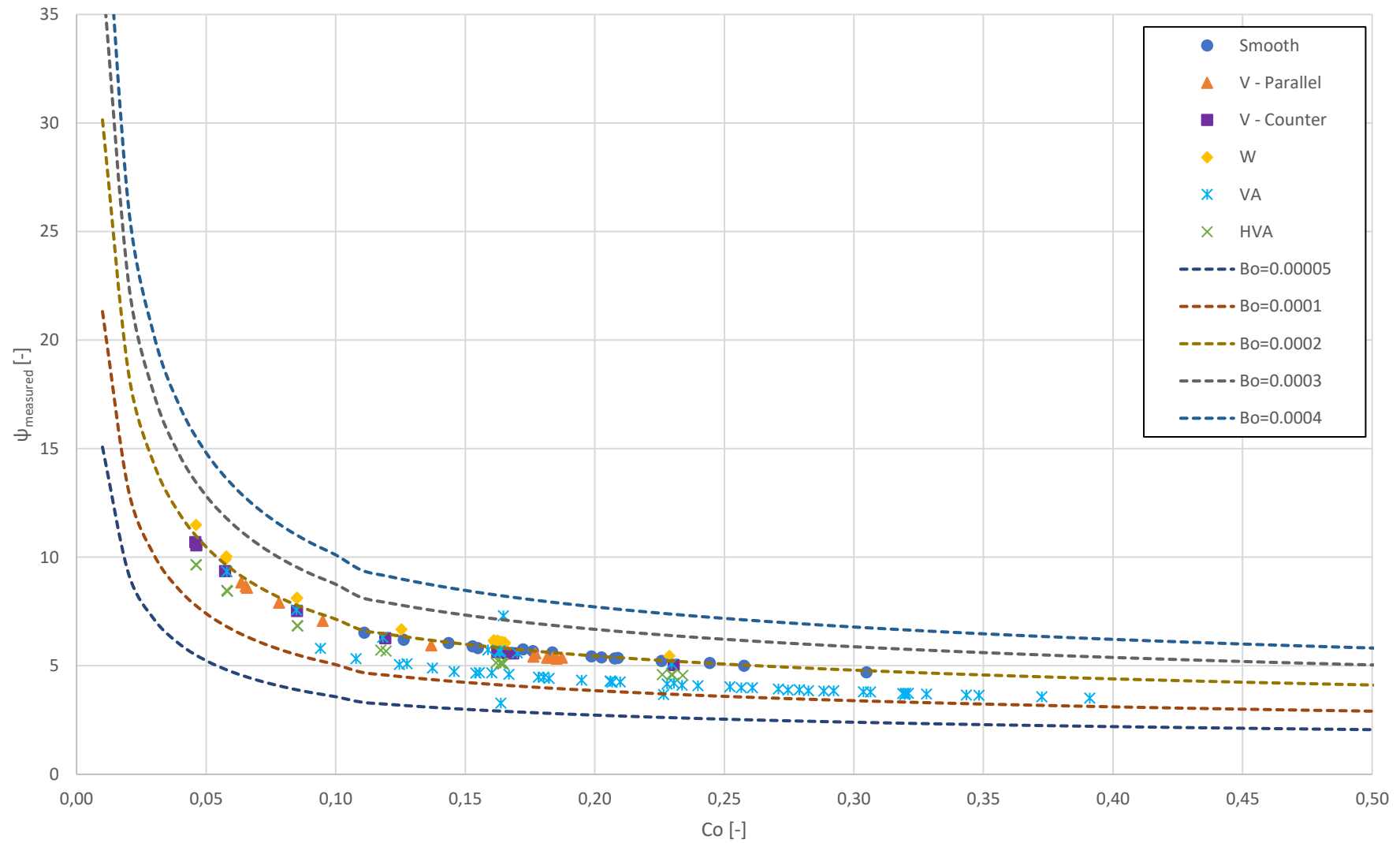


Figure 35: Enhancement Factor ψ vs Convection Number Co for Shah

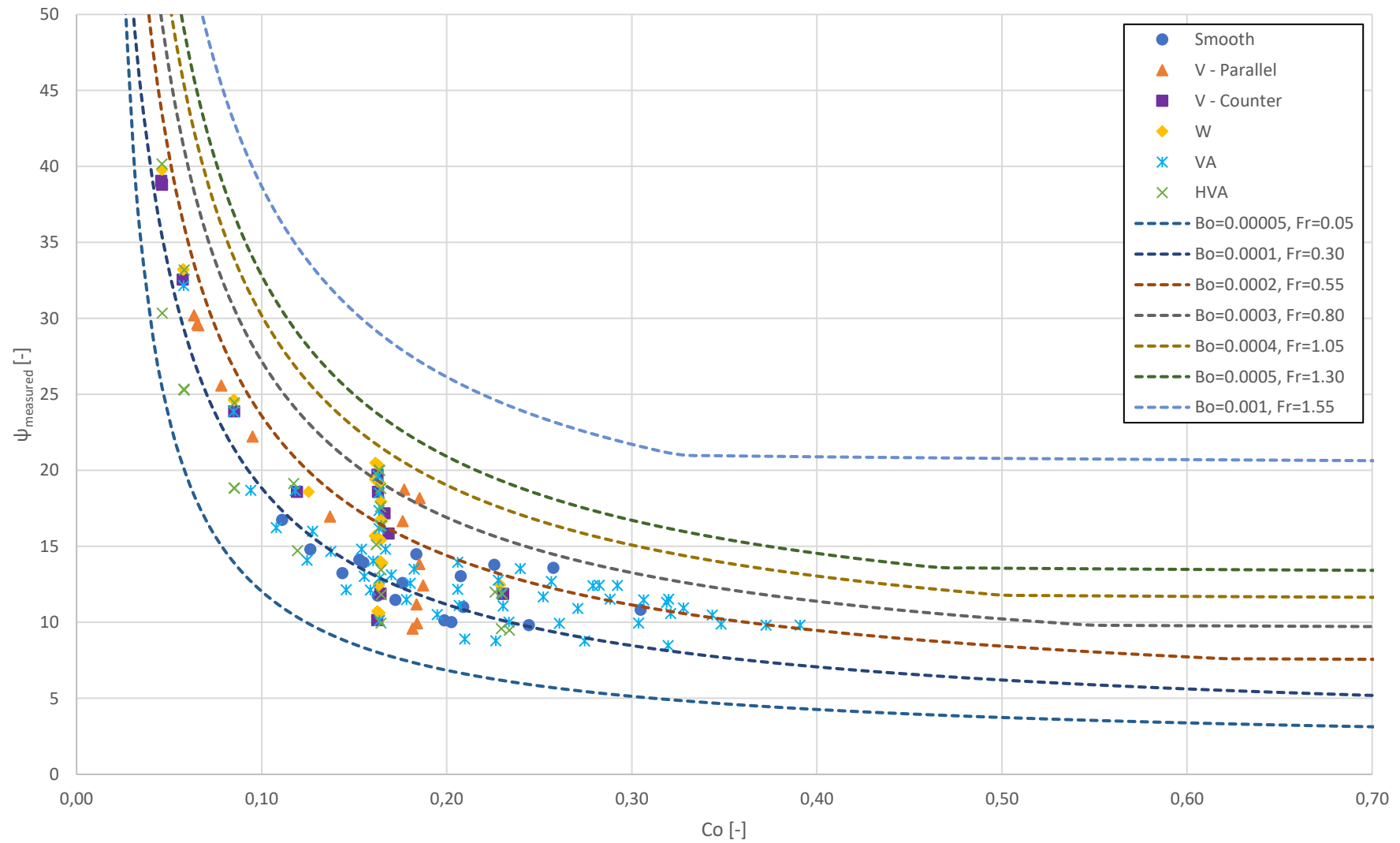


Figure 36: Enhancement Factor ψ vs Convection Number Co for Kandlikar

Chapter 7

7 Comparison Between Experimental Data and Theoretical Results

In this part of the thesis data distribution and the error values for both for heat transfer coefficient and frictional pressure drop per unit length will be observed for different types of tubes. Kandlikar, Shah, Fang, Saitoh, Gungor & Winterton models are used for heat transfer coefficient. On the other hand, Muller-Steinhagen and Heck, Gronnerud, Friedel, Chen, Tran and Haraguchi methods are used for frictional pressure drop per unit length. The reason behind using several models is to see which one provides the best result.

The deviation between the predicted data and experimental data is compared by using mean absolute relative deviation (MARD) and mean relative deviation (MRD). The MARD and MRD is calculated with the following formulas;

$$MARD = \frac{1}{N} \sum_{i=1}^N \left| \frac{pred - exp}{exp} \right| \quad \text{Eqn 173}$$

$$MRD = \frac{1}{N} \sum_{i=1}^N \frac{pred - exp}{exp} \quad \text{Eqn 174}$$

$$X_{\pm 30\%} = \frac{N_{\pm 30\%}}{N} \quad \text{Eqn 175}$$

where; <i>MARD</i>	mean absolute relative deviation,
<i>MRD</i>	mean relative deviation
pred	predicted value,
exp	experimental value,
<i>N</i>	total number of data,
<i>N</i> _{±30%}	number if data within ±30%
<i>X</i> _{±30%}	percentage of data within ±30%

The comparison between the predicted data and the experimental data is performed by using parity plots for that shows the distribution of the data on a 45° line together with $\mp 30\%$ deviation lines. Data closer to the 45° line show lower error.

7.1 Smooth Tube - Counter Flow

7.1.1 Results for Heat Transfer Coefficient

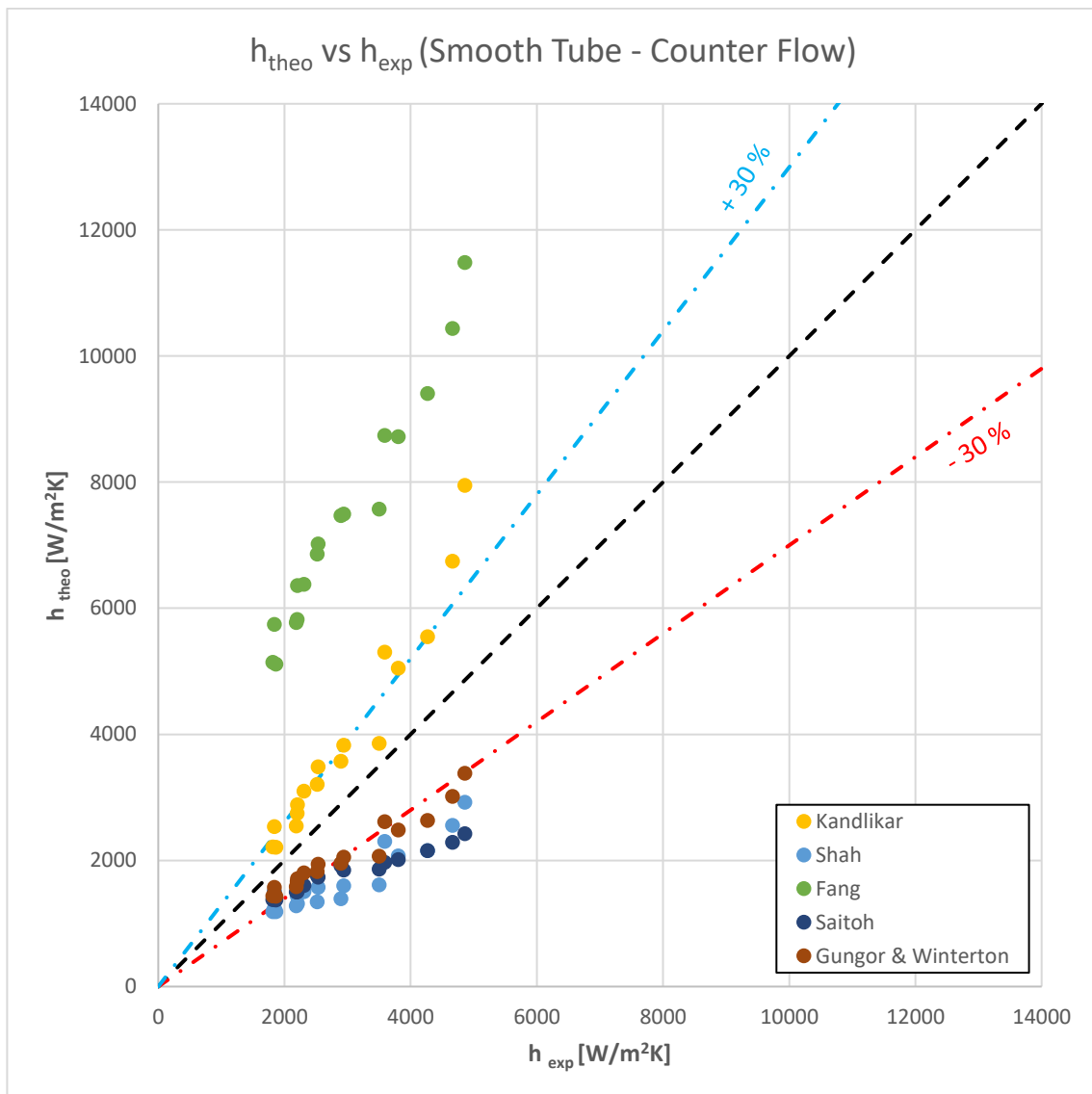


Figure 37: Smooth Tube - Counter Flow / h_{theo} vs h_{exp} for All of the Model

The best result is provided by Gungor & Winterton correlation with 58.82 % of the data set staying within the boundary of $\pm 30\%$. Besides, MRD and MARD value of this correlation are -27.95% and 27.95% (see Table 16), respectively and the distribution of all the points is shown in Figure 42. On the other hand, the worst result is provided by Fang with MRD and MARD values of 158.61 %. It is important to denote that 0.0% of the data set stays within the boundary of $\pm 30\%$ and this can be seen in Figure 40. Although Shah model provides no data within the boundary of $\pm 30\%$ its MRD and MARD values (-41.57% and 41.57%, respectively) is much lower compared to Fang model. In addition, Shah has a better data distribution as it can be seen in Figure 39.

	Kandlikar	Shah	Fang	Saitoh	Gungor & Winterton
MRD	31.32 %	- 41.57 %	158.61 %	- 35.87 %	- 27.95 %
MARD	31.32 %	41.57 %	158.61 %	35.87 %	27.95 %
Number of Data within $\pm 30\%$	8	0	0	4	10
Total Data	17	17	17	17	17
[%] within $\pm 30\%$	47.06 %	0.0 %	0.0 %	23.53 %	58.82 %

Table 16: Error of Heat Transfer Coefficient for Smooth Tube - Counter Flow

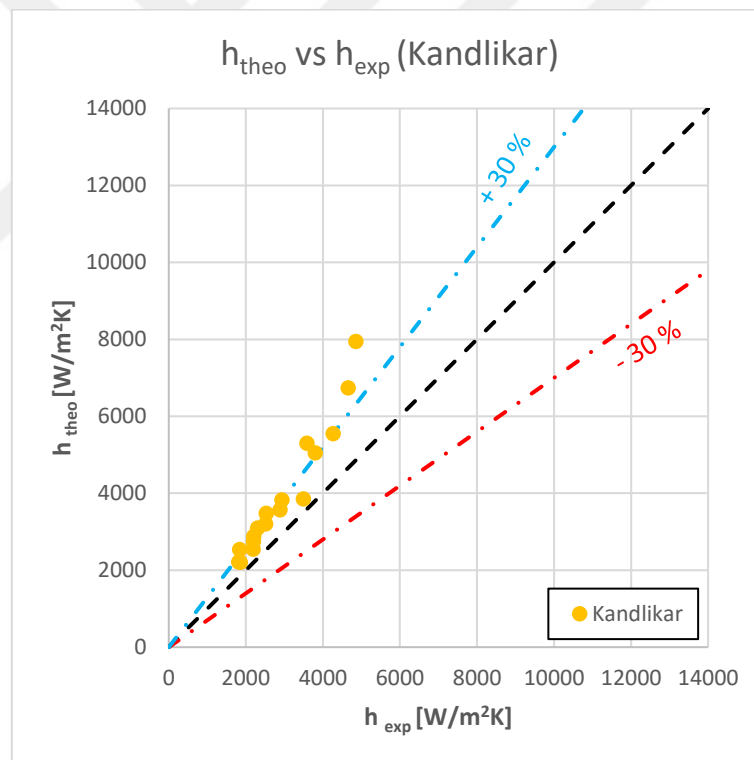


Figure 38: h_{theo} vs h_{exp} for Kandlikar

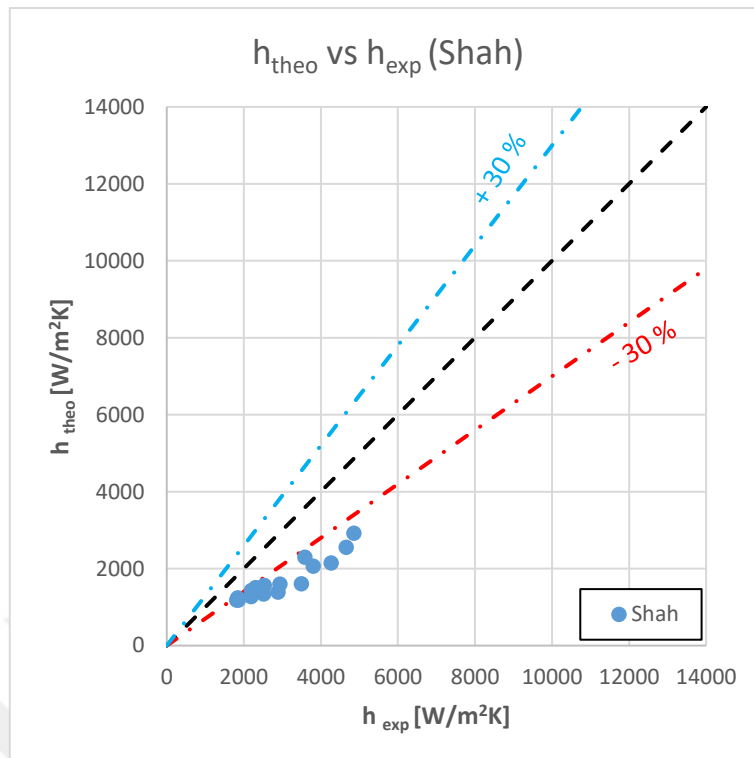


Figure 39: h_{theo} vs h_{exp} for Shah

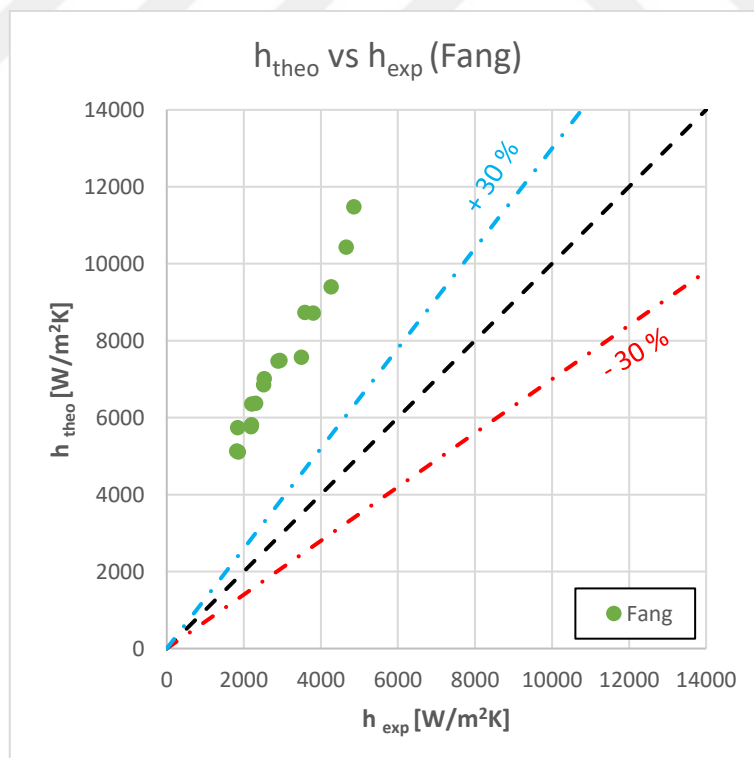


Figure 40: h_{theo} vs h_{exp} for Fang

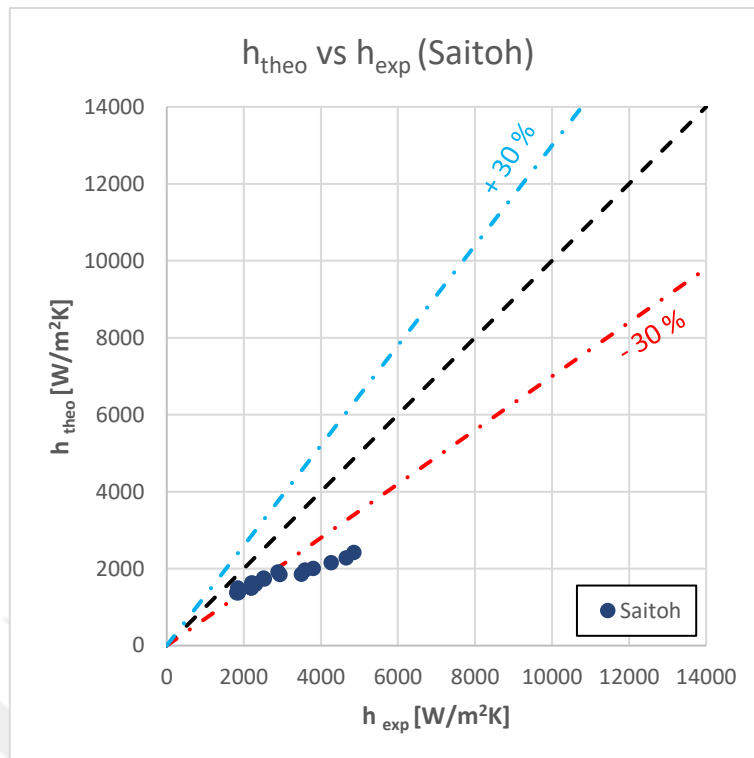


Figure 41: h_{theo} vs h_{exp} for Saitoh

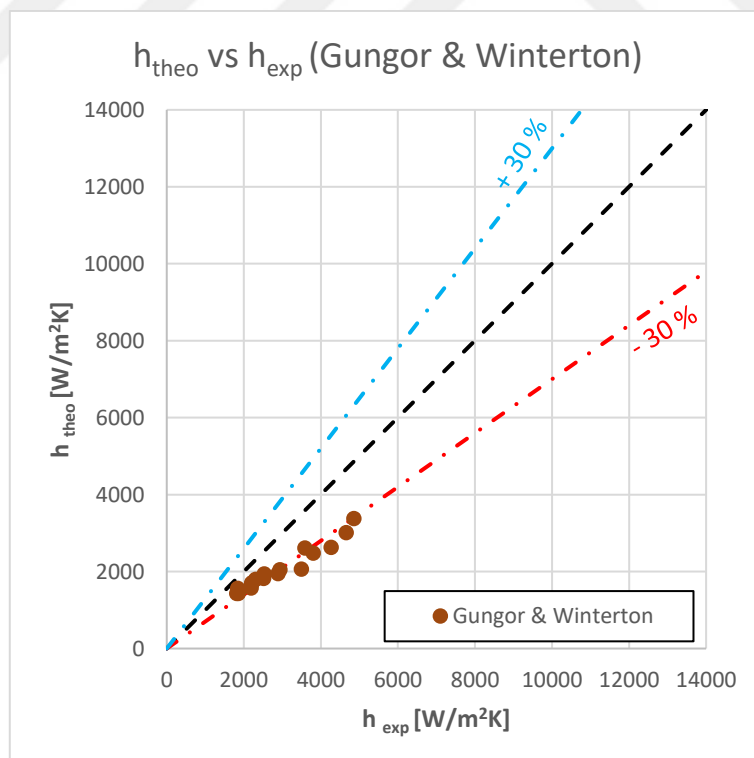


Figure 42: h_{theo} vs h_{exp} for Gungor & Winterton

7.1.2 Results for Frictional Pressure Drop per Unit Length

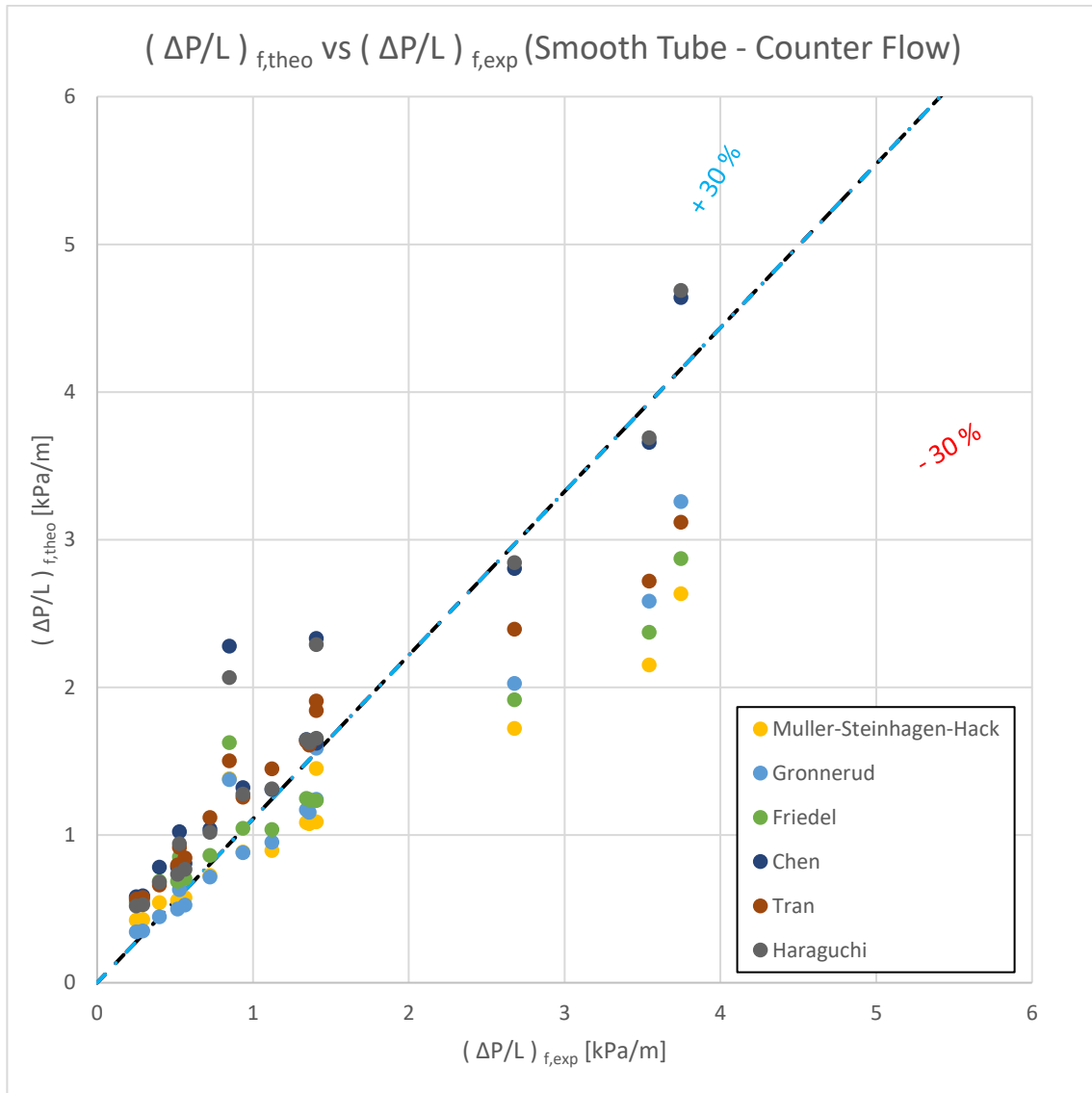


Figure 43: Smooth Tube - Counter Flow / $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for All of the Models

The best result is provided by Gronnerud correlation with 88.24% of the data set staying within the boundary of $\pm 30\%$. Besides, MRD and MARD value of this correlation are 1.68% and 17.63% (see Table 17), respectively and the distribution of all the points is shown in Figure 45. On the other hand, the worst result is provided by Chen with MRD and MARD values of 55.59%. It is important to denote that 41.18% of the data set stays within the boundary of $\pm 30\%$ and this can be seen in Figure 47. Although Tran model provides less data within the boundary of $\pm 30\%$ its MRD and MARD values (42.28% and 48.23%, respectively) is lower compared to Chen model. In general, all of the models for frictional pressure drop per unit length provides acceptable results for “Smooth Tube – Counter Flow”. However, the best three can be given as Gronnerud, Muller-Steinhagen-Heck and Friedel as it can be seen in Table 17.

	Muller- Steinhagen- Heck	Gronnerud	Friedel	Chen	Tran	Haraguchi
MRD	4.11 %	1.68 %	24.47 %	55.59 %	42.28 %	47.83 %
MARD	26.74 %	17.63 %	38.63 %	55.59 %	48.23 %	47.83 %
Number of Data within $\pm$ 30%	10	15	10	7	6	7
Total Data	17	17	17	17	17	17
[%] within $\pm$ 30%	58.82 %	88.24 %	58.82 %	41.18 %	35.29 %	41.18 %

Table 17: Error of Fric. Pressure Drop per Unit Length for Smooth Tube – Counter Flow

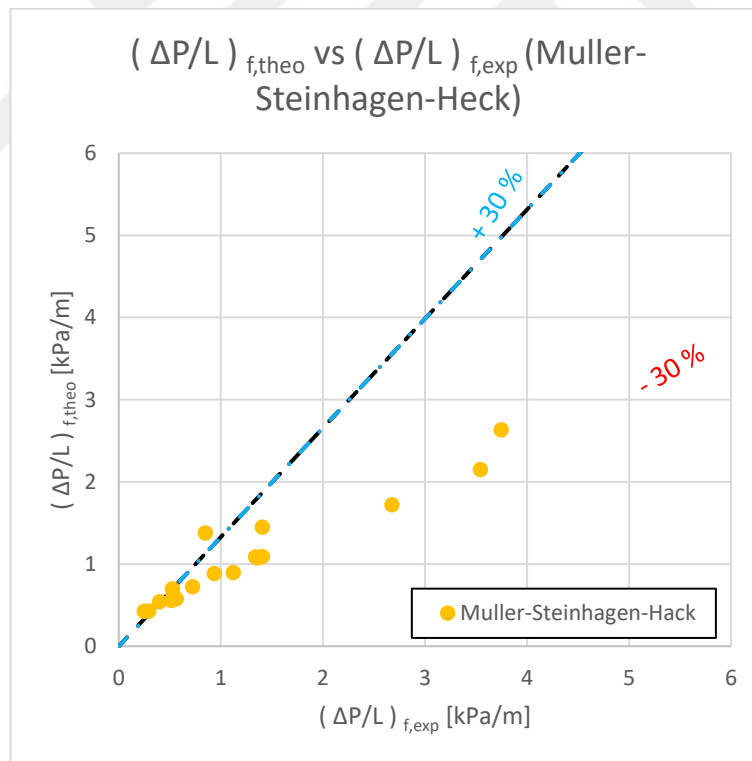


Figure 44: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Muller-Steinhagen-Heck

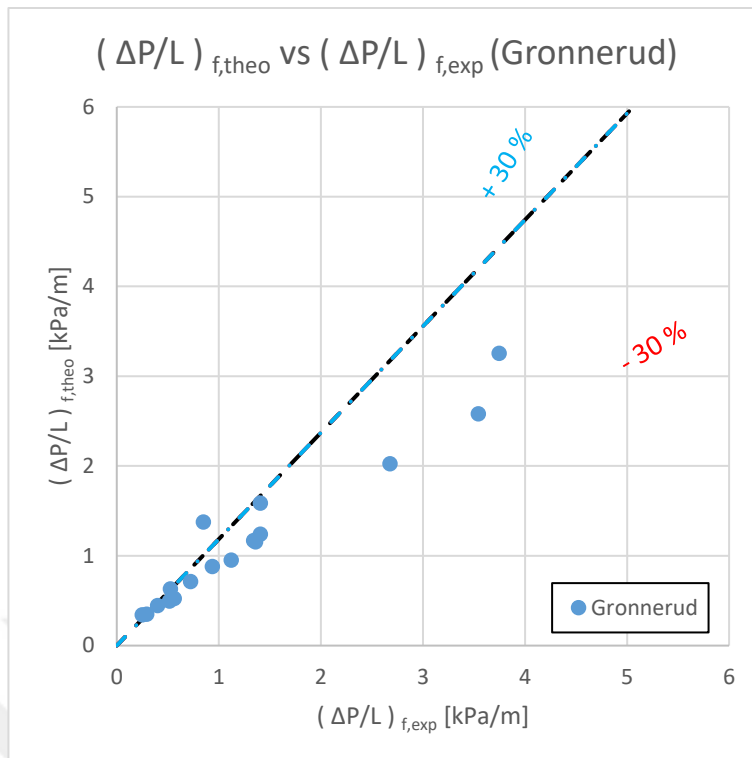


Figure 45: ($\Delta P/L$)_{f,theo} vs ($\Delta P/L$)_{f,exp} for Gronnerud

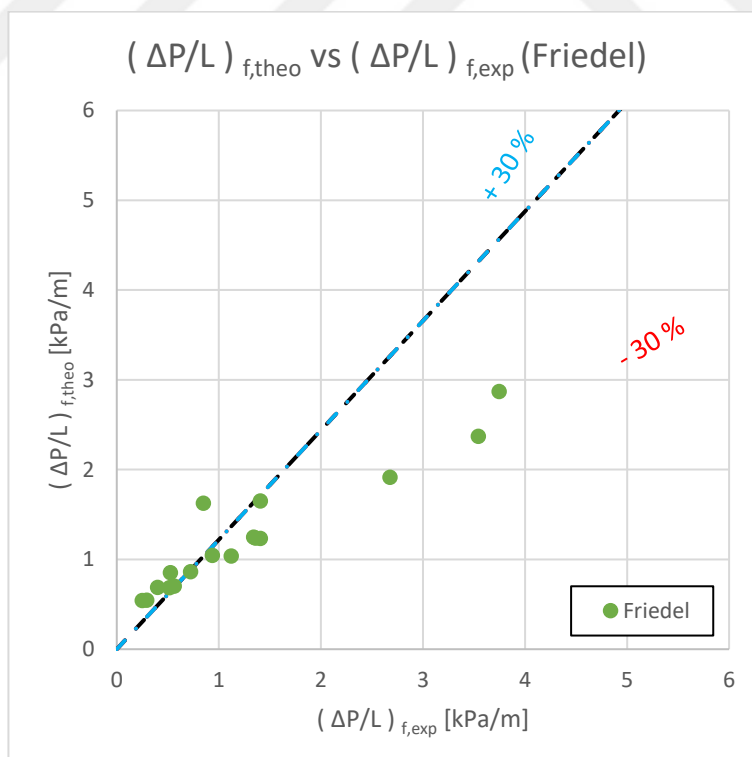


Figure 46: ($\Delta P/L$)_{f,theo} vs ($\Delta P/L$)_{f,exp} for Friedel

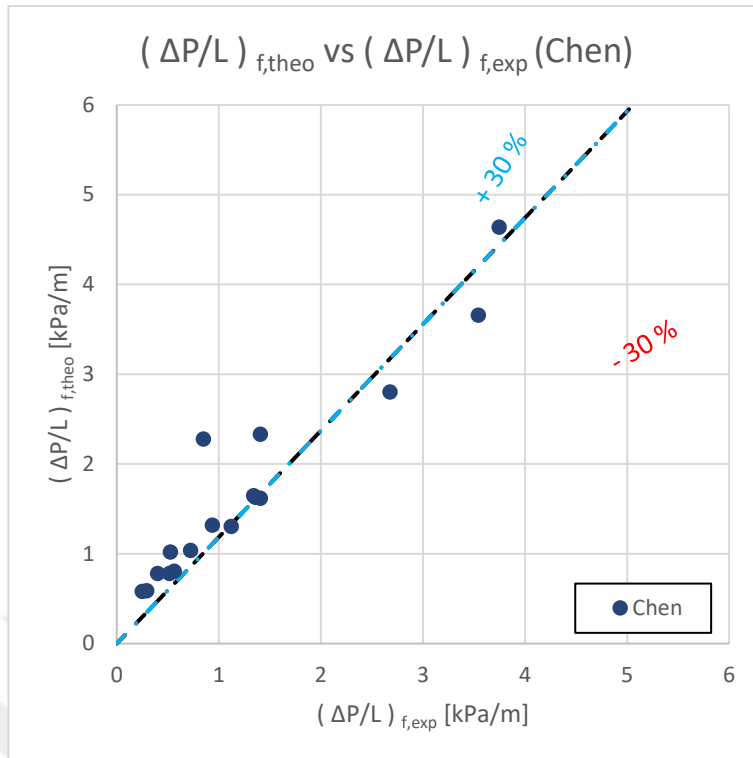


Figure 47: $(\Delta P/L)_{f,theo} \text{ vs } (\Delta P/L)_{f,exp}$ for Chen

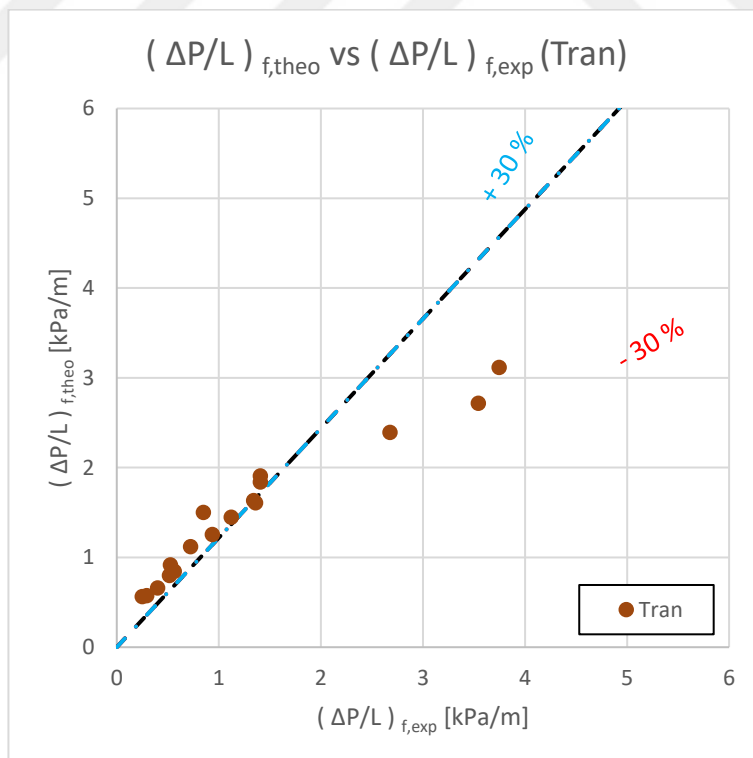


Figure 48: $(\Delta P/L)_{f,theo} \text{ vs } (\Delta P/L)_{f,exp}$ for Tran

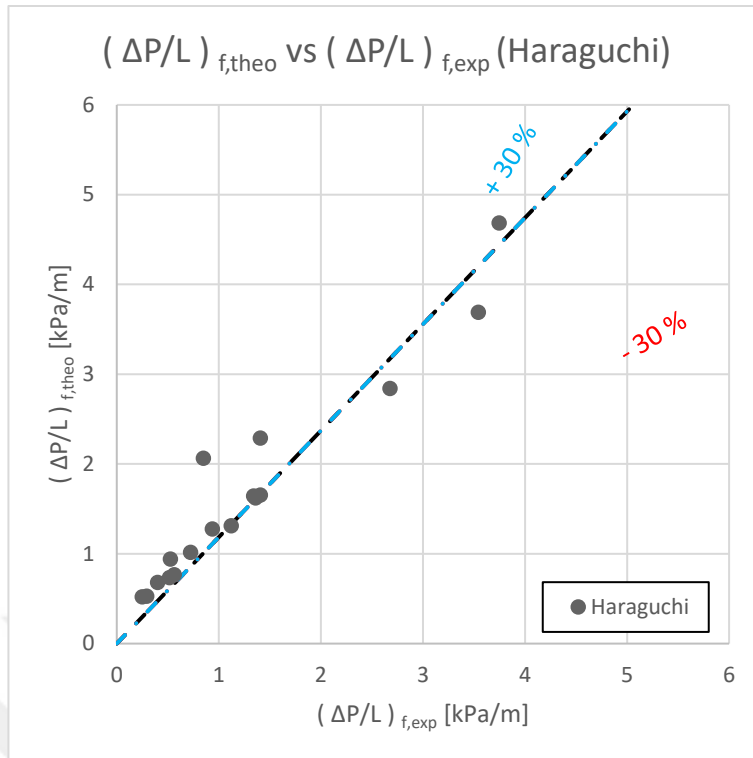


Figure 49: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Haraguchi

7.2 Tube V - Parallel Flow

7.2.1 Results for Heat Transfer Coefficient

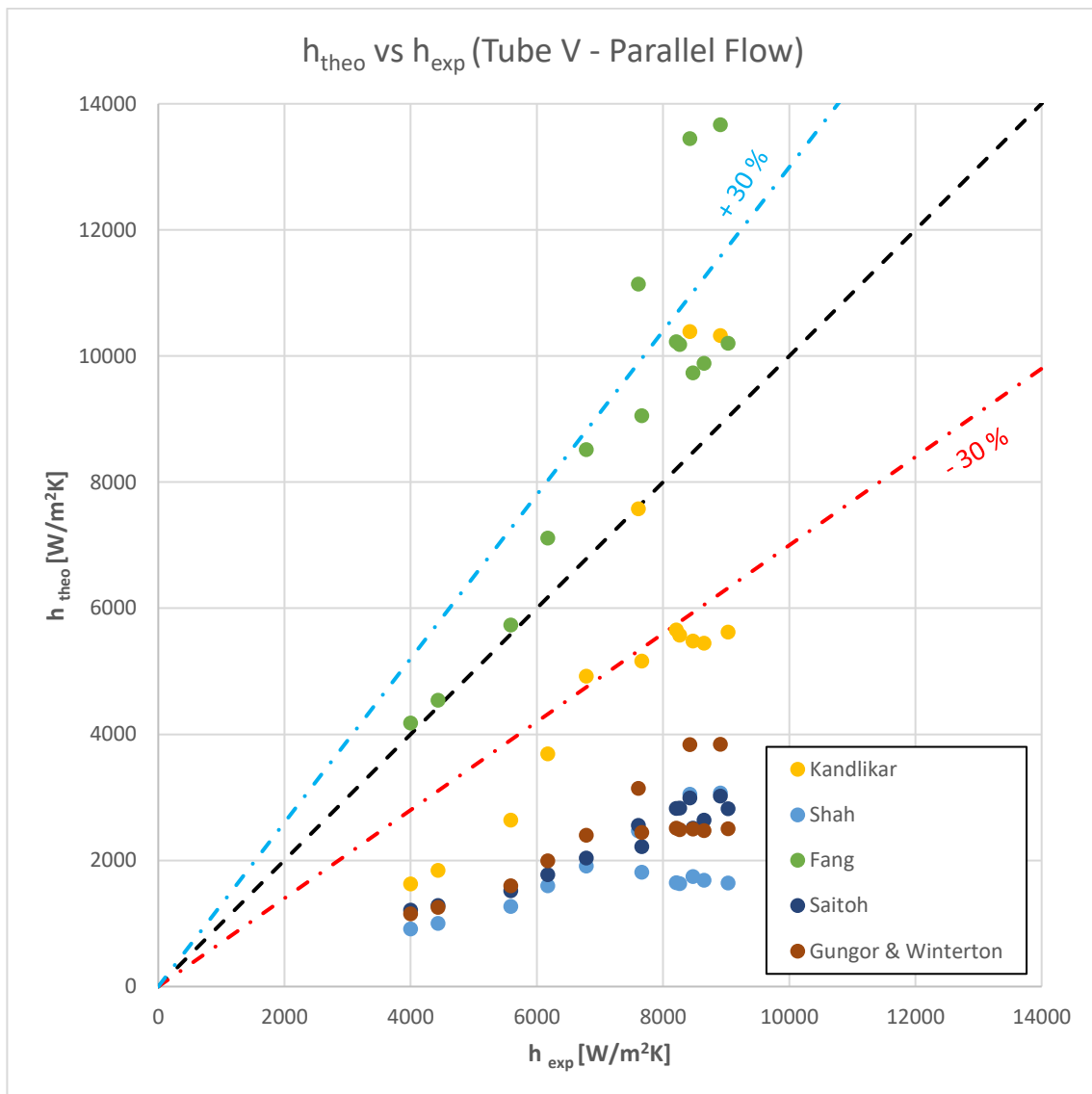


Figure 50: Tube V - Parallel Flow / h_{theo} vs h_{exp} for All of the Model

The best result is provided by Fang correlation with 78.57 % of the data set staying within the boundary of $\mp 30\%$. Besides, MRD and MARD value of this correlation are 22.72%(see Table 18) for both of them and the distribution of all the points is shown in Figure 53. On the other hand, the worst result is provided by Shah with MRD and MARD values of -75.22% and 75.22%, respectively. It is important to denote that 0.0% of the data set stays within the boundary of $\mp 30\%$ and this can be seen in Figure 52. In addition, Gungor & Winterton and Saitoh also provides no data within the boundary of $\mp 30\%$.

	Kandlikar	Shah	Fang	Saitoh	Gungor & Winterton
MRD	- 28.98 %	- 75.22 %	22.72 %	- 68.77 %	- 67.02 %
MARD	34.58 %	75.22 %	22.72 %	68.77 %	67.02 %
Number of Data within $\pm 30\%$	4	0	11	0	0
Total Data	14	14	14	14	14
[%] within $\pm 30\%$	28.57 %	0.0 %	78.57 %	0.0 %	0.0 %

Table 18: Error of Heat Transfer Coefficient for Tube V - Parallel Flow

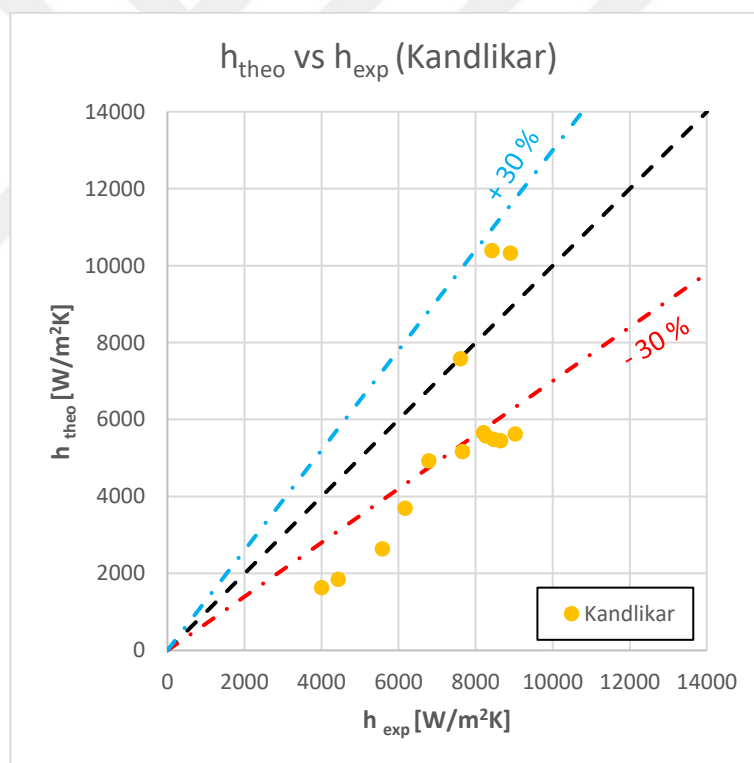


Figure 51: h_{theo} vs h_{exp} for Kandlikar

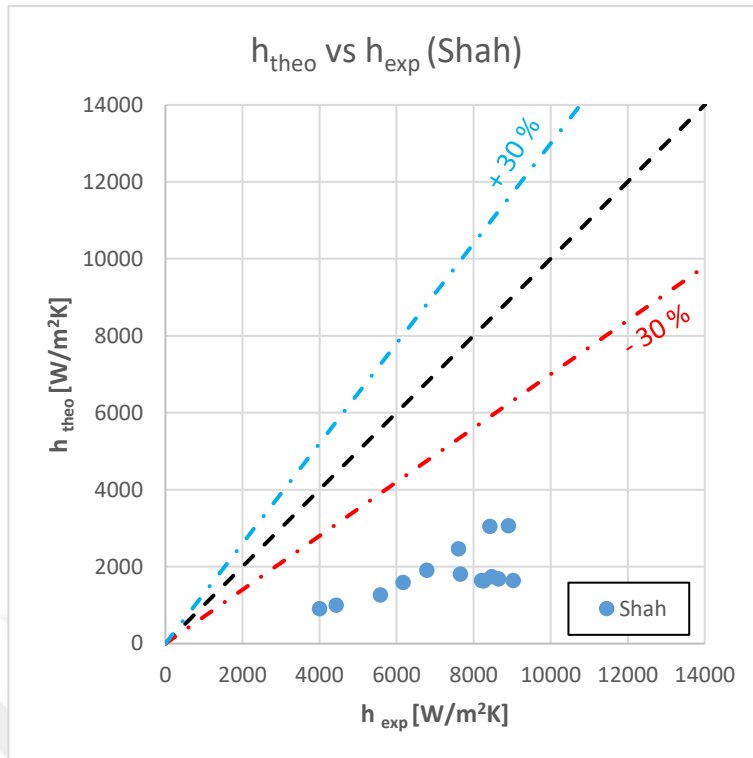


Figure 52: h_{theo} vs h_{exp} for Shah

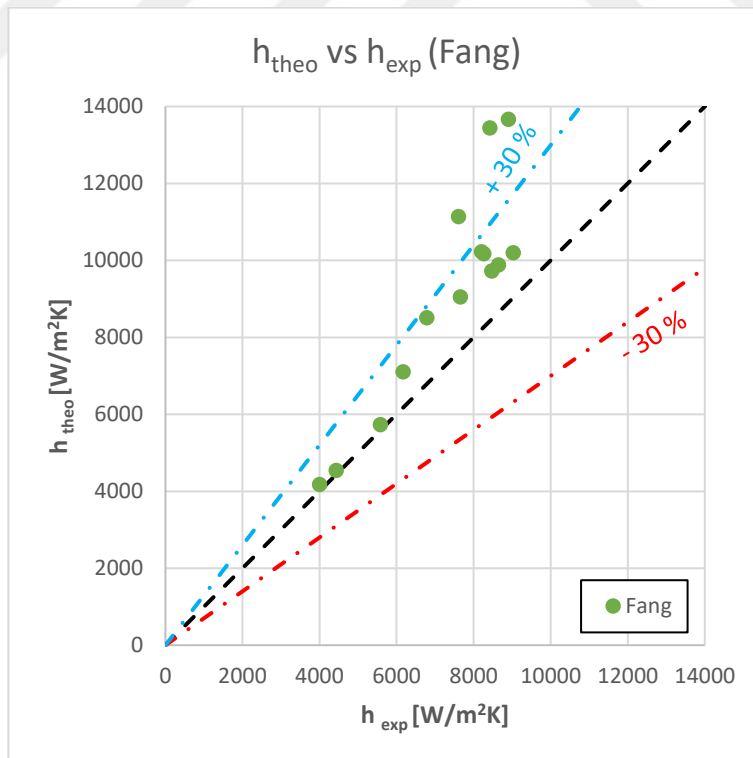


Figure 53: h_{theo} vs h_{exp} for Fang

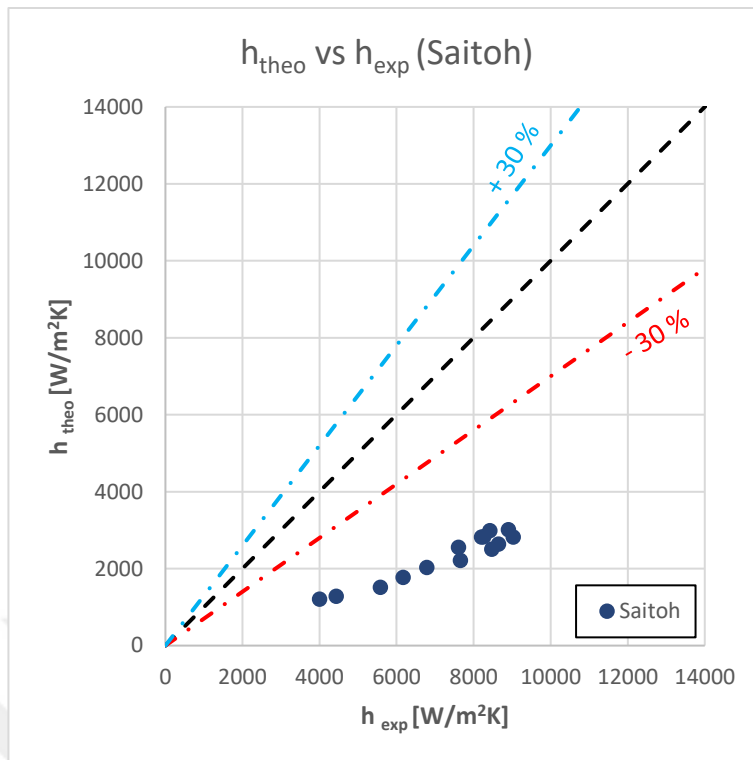


Figure 54: h_{theo} vs h_{exp} for Saitoh

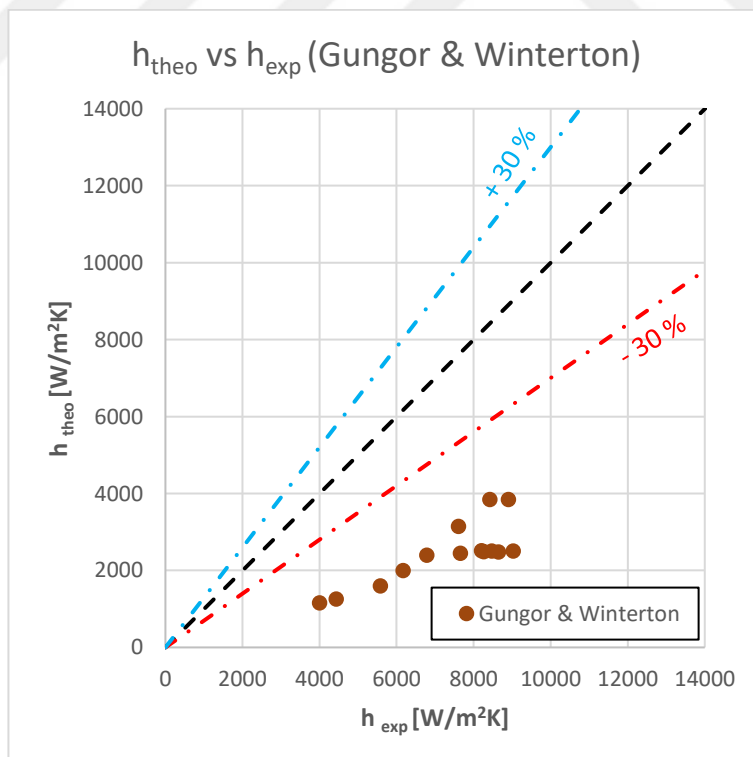


Figure 55: h_{theo} vs h_{exp} for Gungor & Winterton

7.2.2 Results for Frictional Pressure Drop per Unit Length

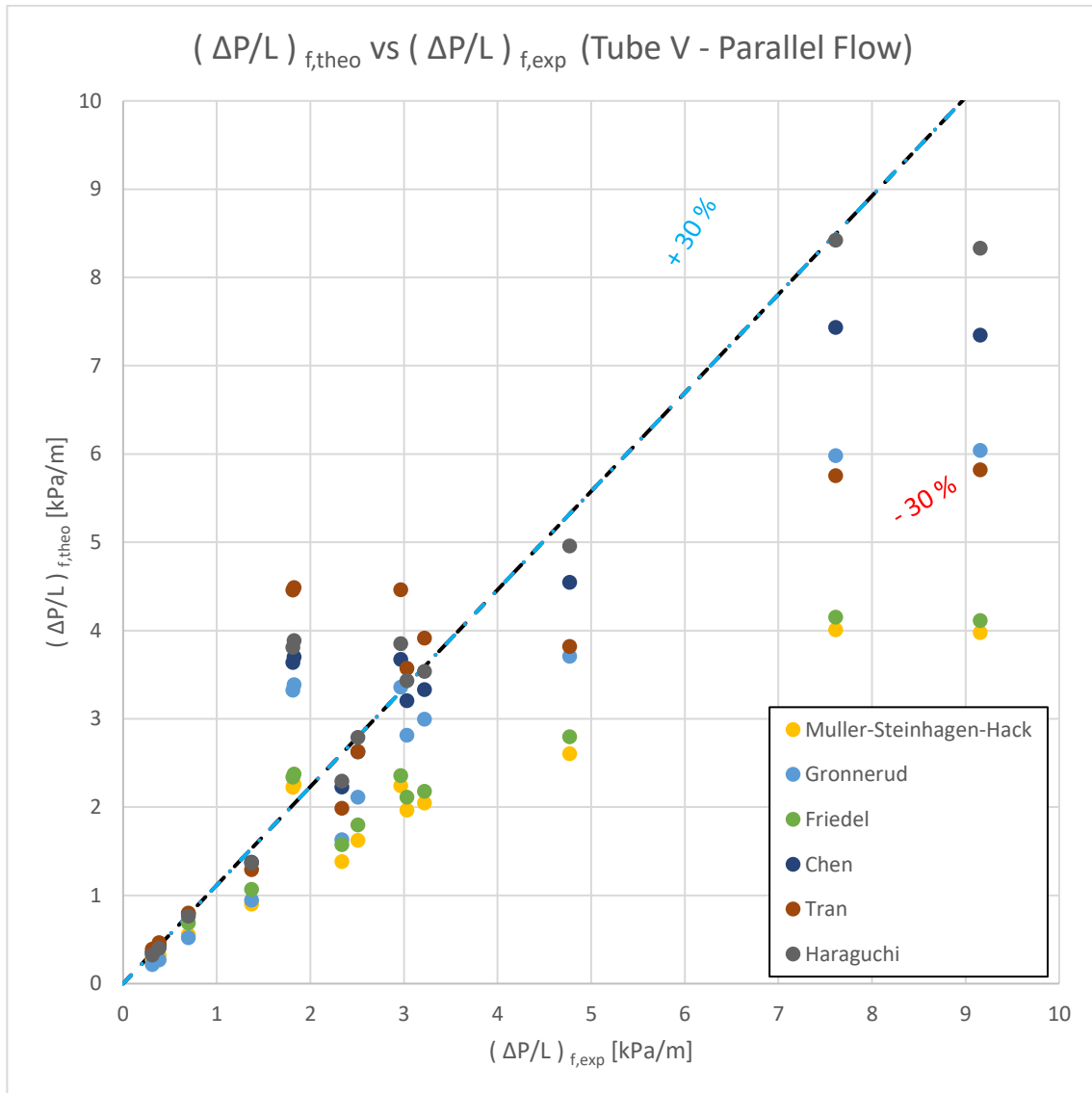


Figure 56: Tube V - Parallel Flow / $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for All of the Models

The best result is provided by Chen correlation with 85.71% of the data set staying within the boundary of $\pm 30\%$. Besides, MRD and MARD value of this correlation are 17.78% and 22.26% (see Table 19), respectively and the distribution of all the points is shown in Figure 60. Also, the results provided by Haraguchi are so close what is provided by Chen with the same percentage of data set staying within the boundary of $\pm 30\%$. On the other hand, the worst result is provided by Tran with MRD and MARD values of 24.79% and 39.29%, respectively. It is important to denote that 71.43% of the data set stays within the boundary of $\pm 30\%$ and this can be seen in Figure 61. Muller-Steinhagen-Heck has lower number of data staying within the boundary of $\pm 30\%$ with lower MARD value. This means Tran has better distribution with higher error. Besides these, all of the results provided by different models are acceptable.

	Muller- Steinhagen- Heck	Gronnerud	Friedel	Chen	Tran	Haraguchi
MRD	-25.61 %	- 5.14 %	- 16.12 %	17.78 %	24.79 %	22.13 %
MARD	32.22 %	31. 19 %	28.05 %	22.26 %	39.29 %	25.72 %
Number of Data within $\pm$ 30%	6	7	7	12	10	12
Total Data	14	14	14	14	14	14
[%] within $\pm$ 30%	42.86 %	50.00%	50.00 %	85.71 %	71.43 %	85.71 %

Table 19: Error of Frictional Pressure Drop per Unit Length for Tube V – Parallel Flow

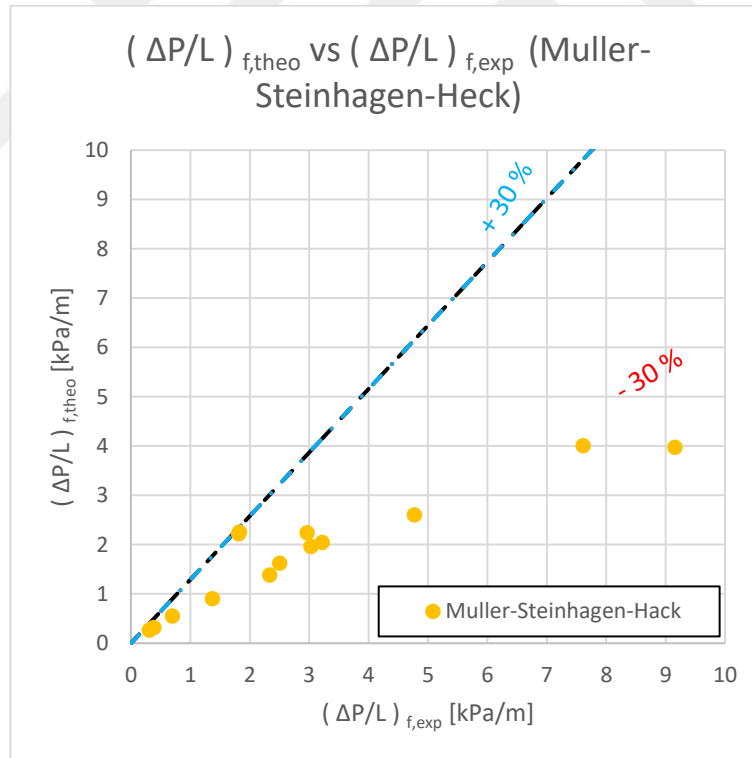


Figure 57: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Muller-Steinhagen-Heck

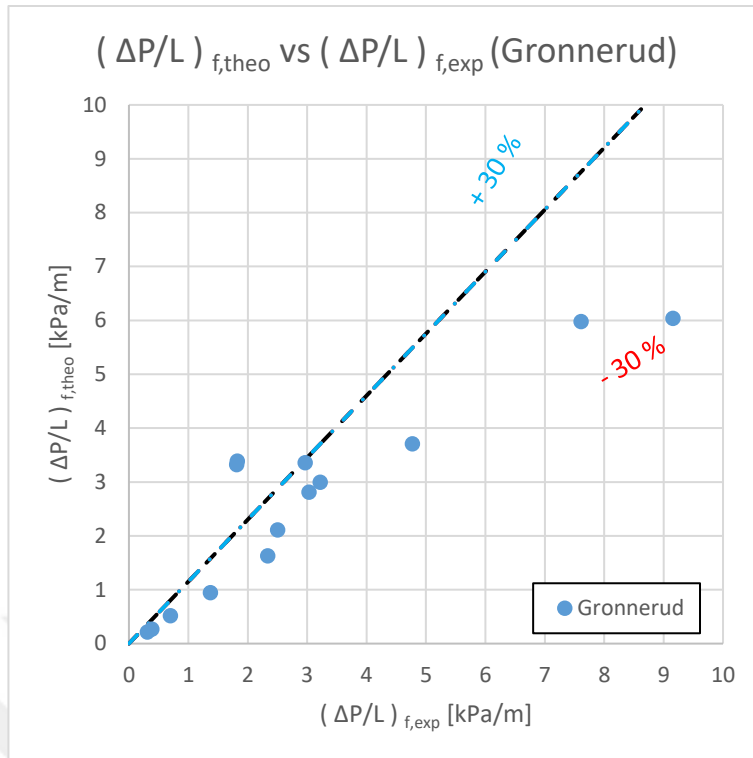


Figure 58: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Gronnerud

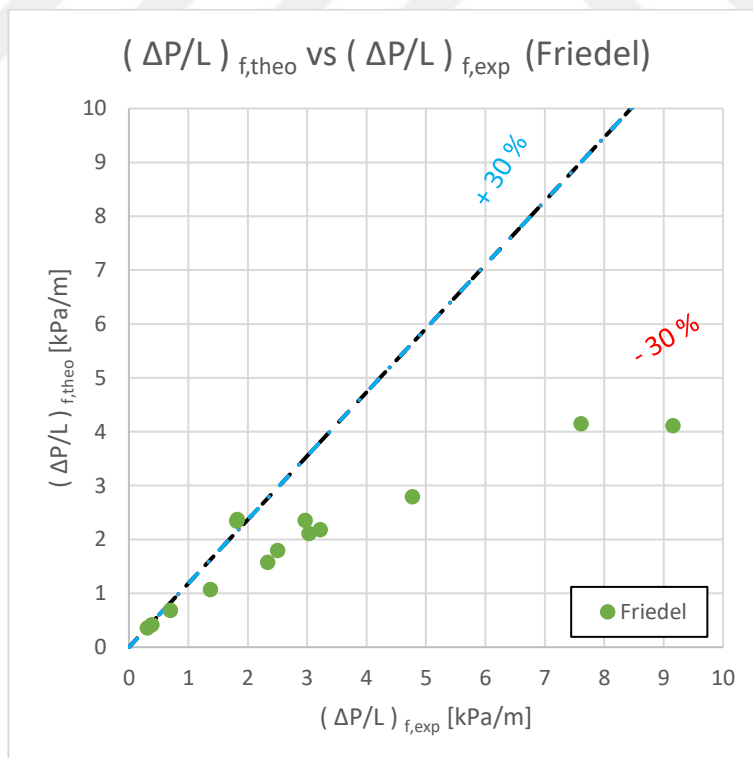


Figure 59: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Friedel

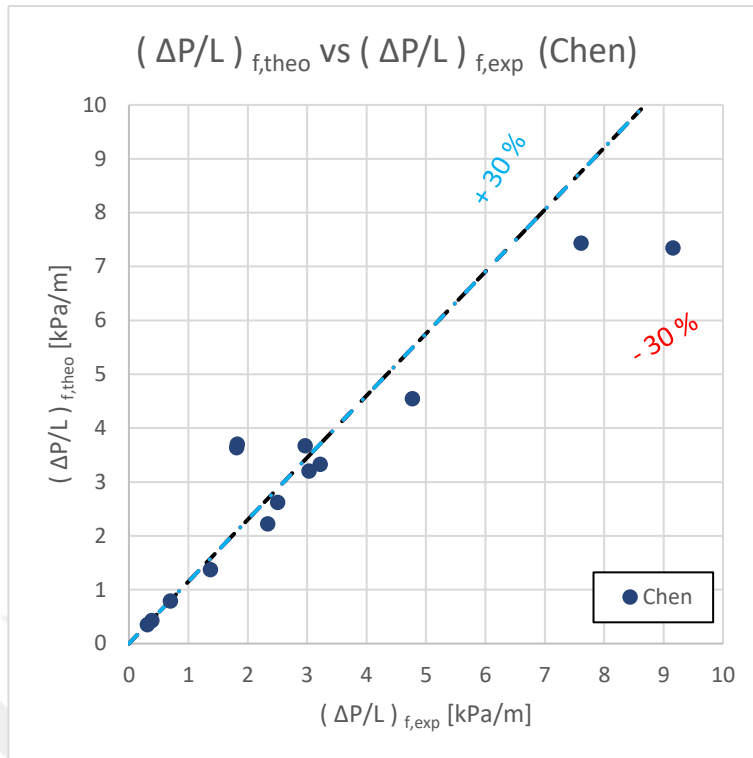


Figure 60: ($\Delta P/L$)_{f,theo} vs ($\Delta P/L$)_{f,exp} for Chen

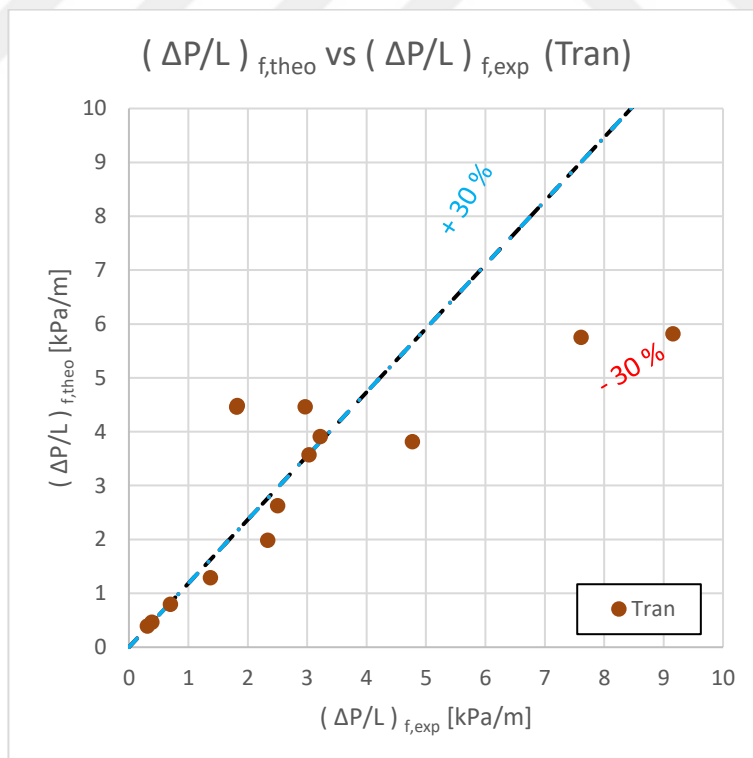


Figure 61: ($\Delta P/L$)_{f,theo} vs ($\Delta P/L$)_{f,exp} for Tran

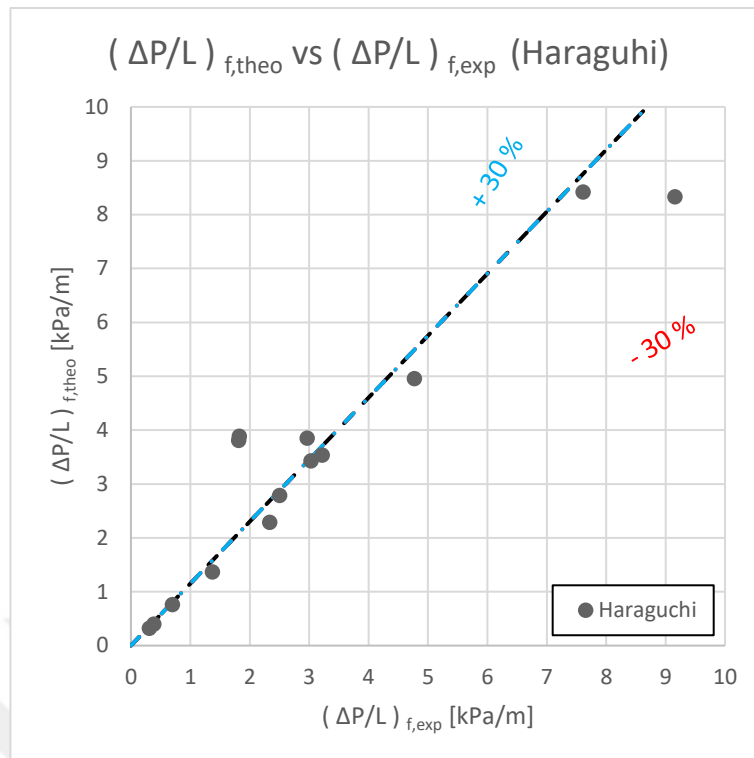


Figure 62: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Haraguchi

7.3 Tube V - Counter Flow

7.3.1 Results for Heat Transfer Coefficient

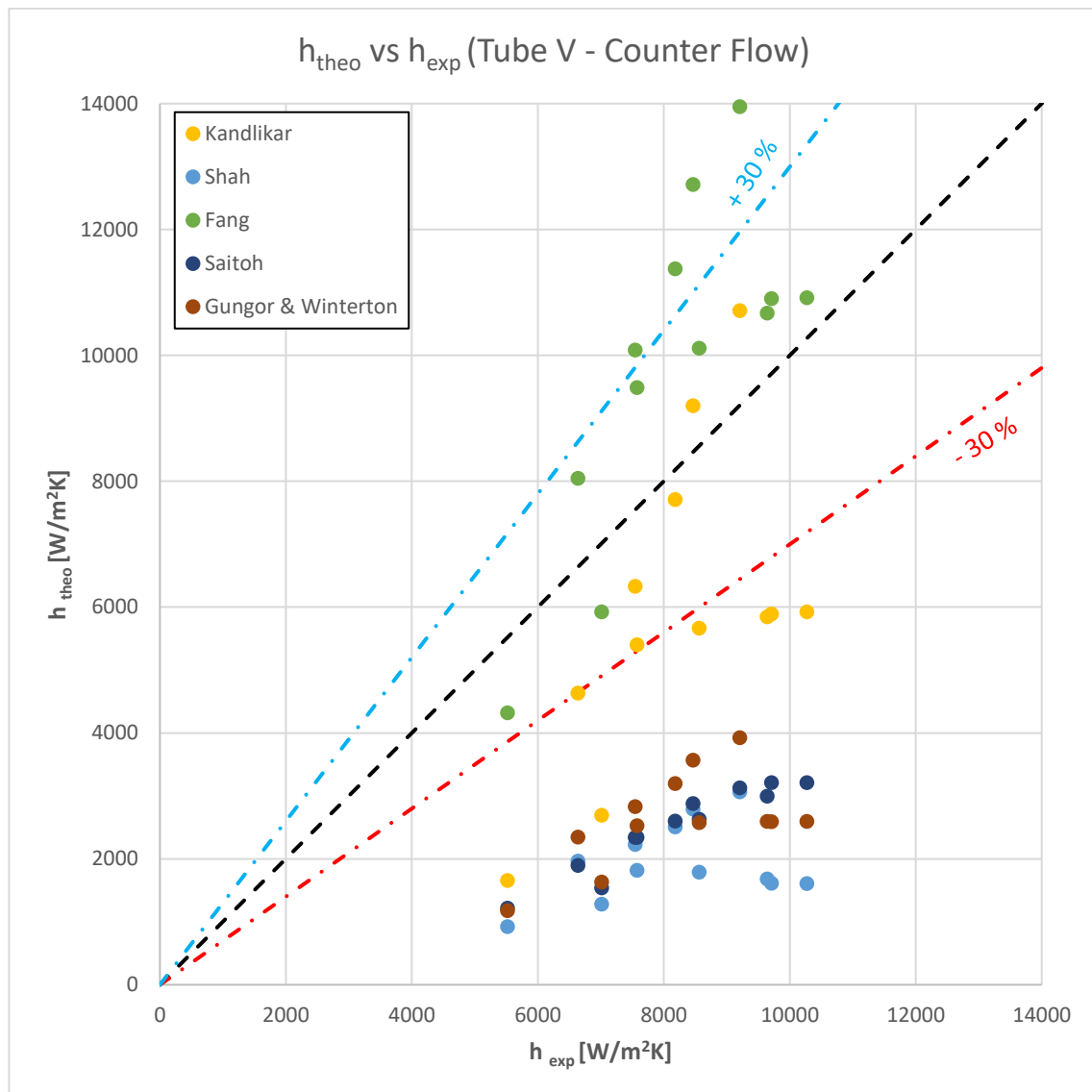


Figure 63: Tube V - Counter Flow / h_{theo} vs h_{exp} for All of the Model

The best result is provided by Fang correlation with 66.67% of the data set staying within the boundary of $\pm 30\%$. Besides, MRD and MARD value of this correlation are 19.28% and 25.46% (see Table 20), respectively and the distribution of all the points is shown in Figure 66. On the other hand, the worst result is provided by Shah with MRD and MARD values of -76.19% and 76.19%, respectively. It is important to denote that 0.0% of the data set stays within the boundary of $\pm 30\%$ and this can be seen in Figure 65. Although Saitoh and Gungor & Winterton models provide no data within the boundary of $\pm 30\%$ its MRD and MARD values are much lower compared to Shah model.

	Kandlikar	Shah	Fang	Saitoh	Gungor & Winterton
MRD	- 28.50 %	- 76.19 %	19.28 %	- 69.96 %	- 68.03 %
MARD	32.68 %	76.19 %	25.46 %	69.96 %	68.08 %
Number of Data within $\pm 30\%$	5	0	8	0	0
Total Data	12	12	12	12	12
[%] within $\pm 30\%$	41.67 %	0.0 %	66.67 %	0.0 %	0.0 %

Table 20: Error of Heat Transfer Coefficient for Tube V - Counter Flow

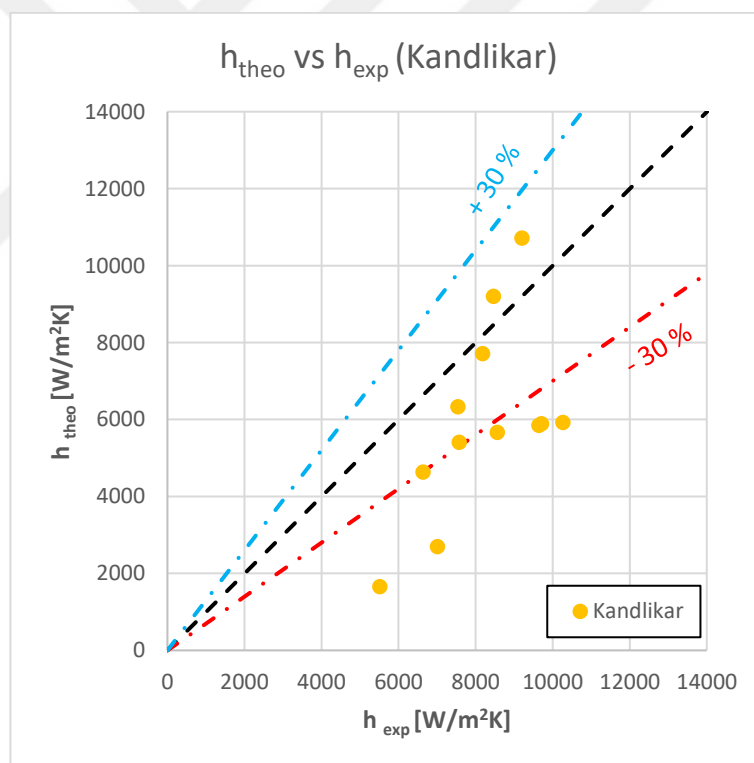


Figure 64: h_{theo} vs h_{exp} for Kandlikar

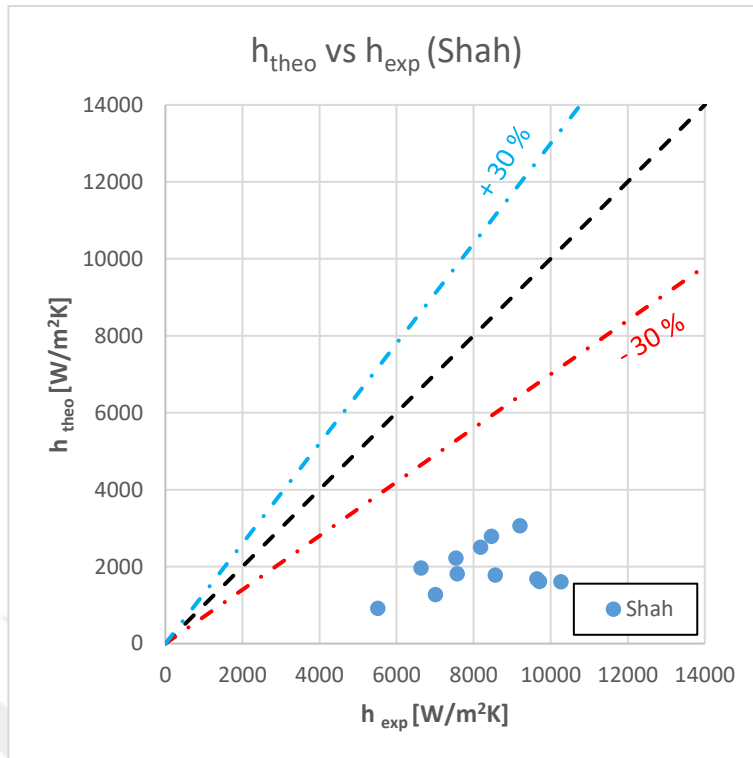


Figure 65: h_{theo} vs h_{exp} for Shah

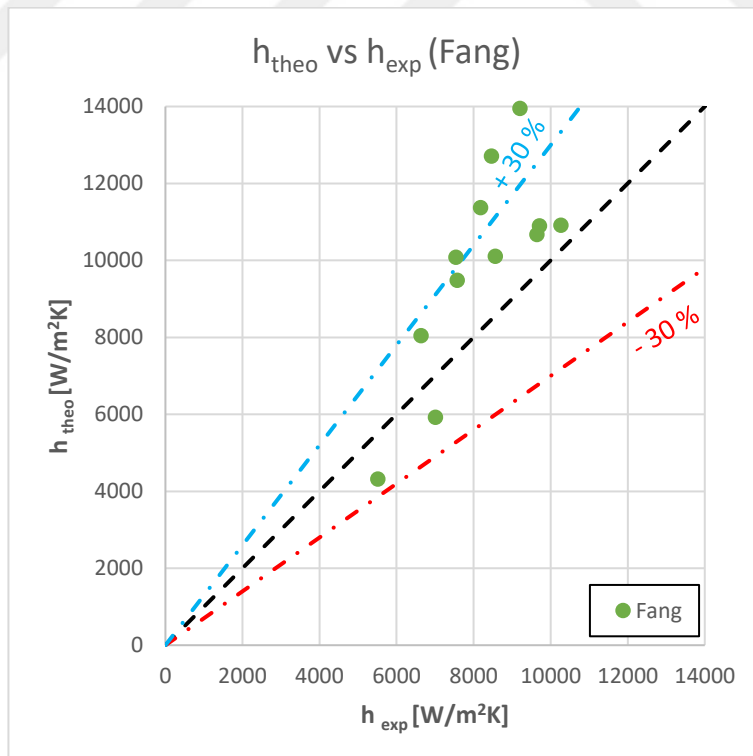


Figure 66: h_{theo} vs h_{exp} for Fang

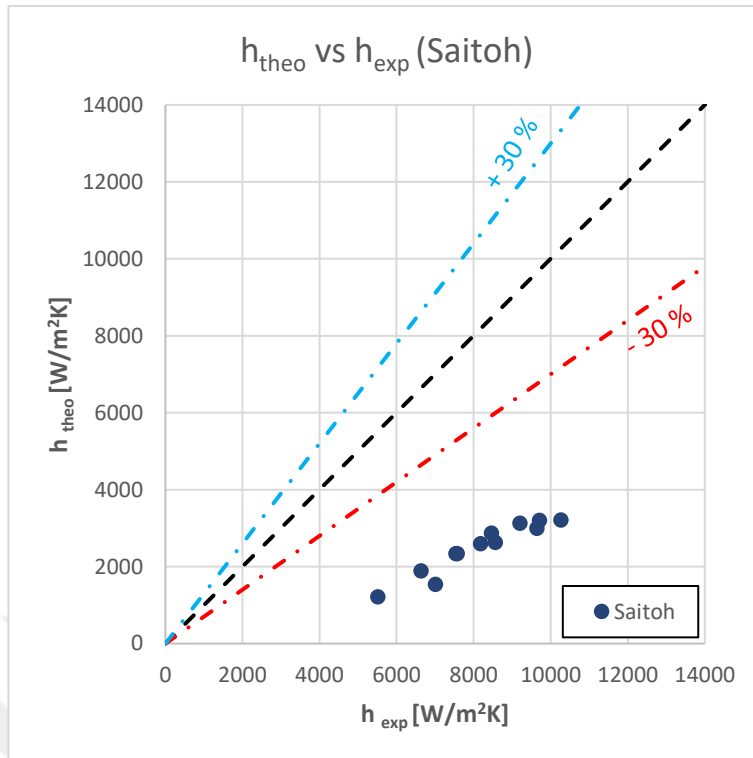


Figure 67: h_{theo} vs h_{exp} for Saitoh

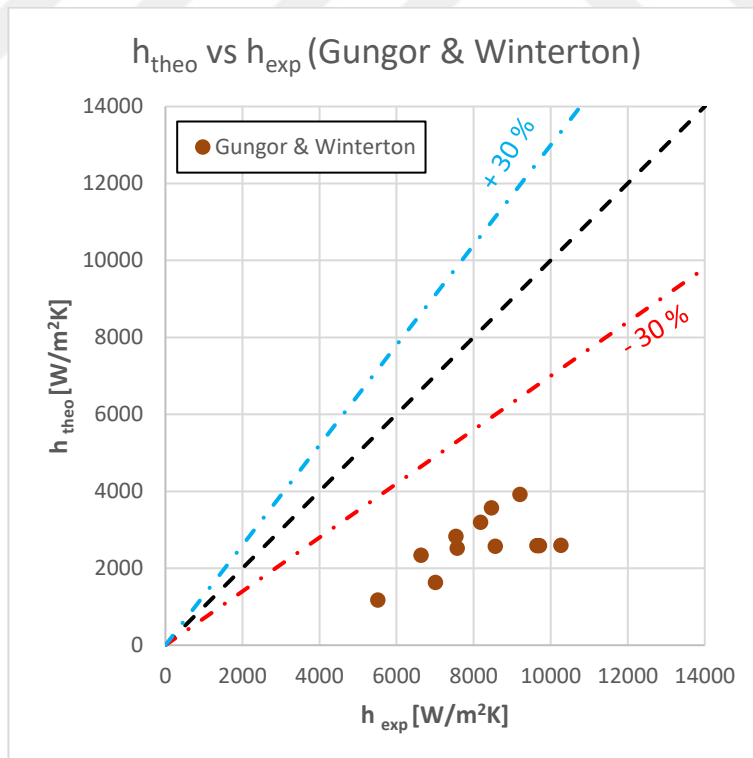


Figure 68: h_{theo} vs h_{exp} for Gungor & Winterton

7.3.2 Results for Frictional Pressure Drop per Unit Length

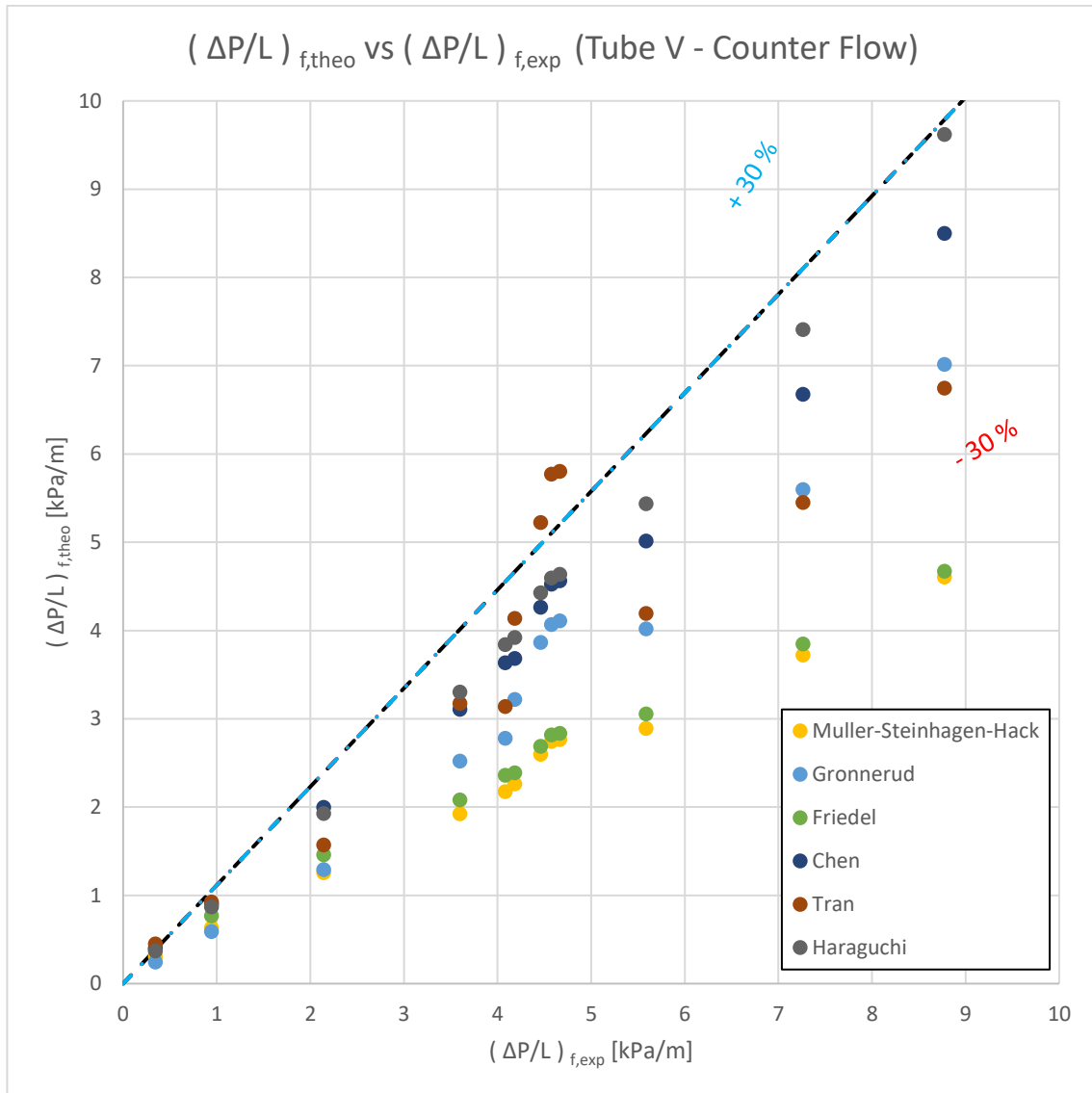


Figure 69: Tube V - Counter Flow / $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for All of the Models

The best result is provided by Haraguchi with 100.00% of the data set staying within the boundary of $\pm 30\%$. Besides, MRD and MARD value of this correlation are -1.85% and 5.14% (see Table 21), respectively and the distribution of all the points is shown in Figure 75. Here, Chen also provides 100.00% of the data set staying within the boundary of $\pm 30\%$ but it has a slightly higher MRD and MARD values. On the other hand, the worst result is provided by Muller-Steinhagen-Heck with MRD and MARD values of -40.89% and 40.89%, respectively. It is important to denote that only 8.33% of the data set stays within the boundary of $\pm 30\%$ and this can be seen in Figure 70. All of the models provide acceptable results for this type of micro-fin tube except Muller-Steinhagen-Heck and Friedel.

	Muller- Steinhagen- Heck	Gronnerud	Friedel	Chen	Tran	Haraguchi
MRD	- 40.89 %	- 24.81 %	- 34.84 %	- 5.11%	- 3.19 %	- 1.85 %
MARD	40.89 %	24.81 %	37.49 %	7.70 %	19.72 %	5.14 %
Number of Data within $\pm$ 30%	1	9	2	12	11	12
Total Data	12	12	12	12	12	12
[%] within $\pm$ 30%	8.33 %	75.00 %	16.67 %	100.00 %	91.67 %	100.00 %

Table 21: Error of Frictional Pressure Drop per Unit Length for Tube V – Counter Flow

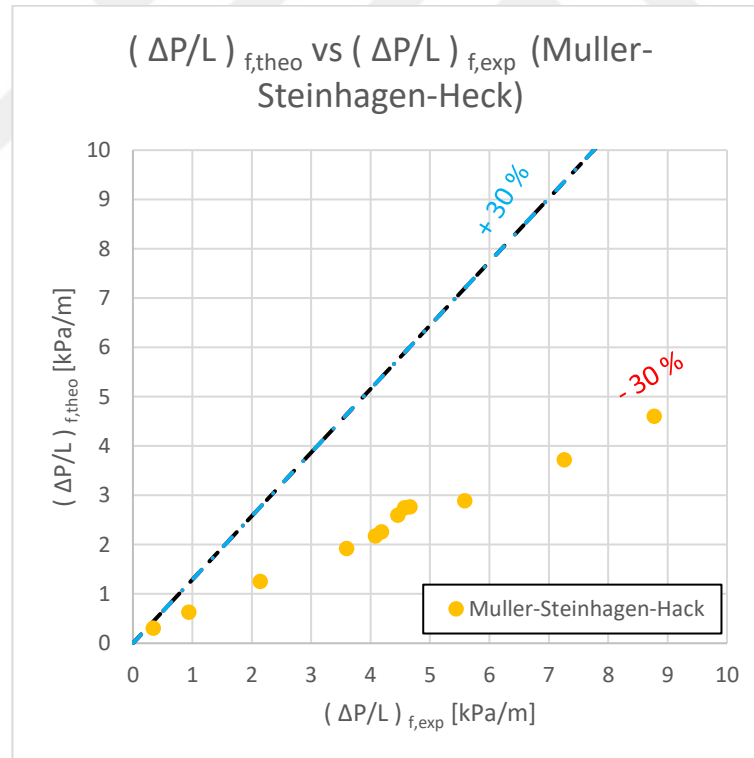


Figure 70: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Muller-Steinhagen-Heck

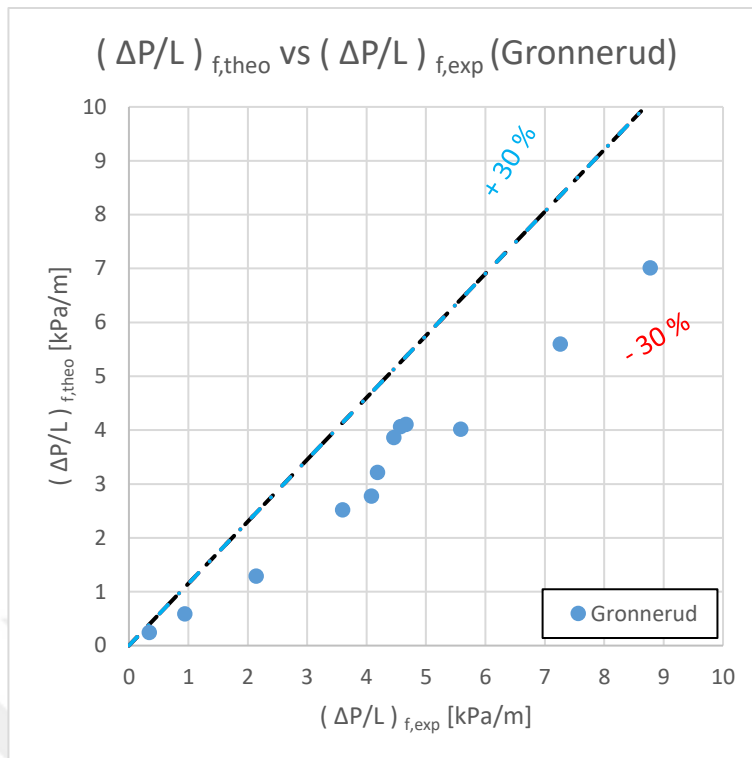


Figure 71: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Gronnerud

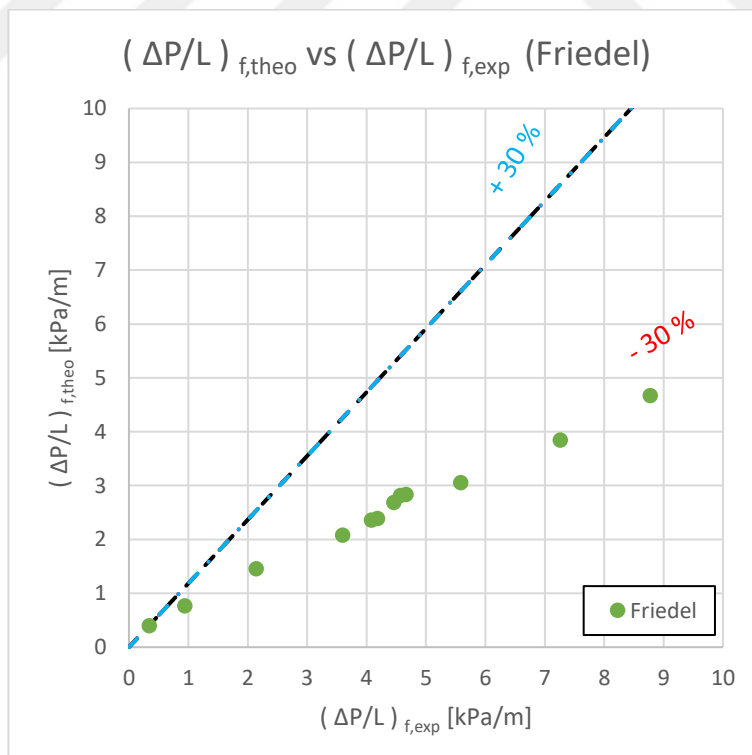


Figure 72: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Friedel

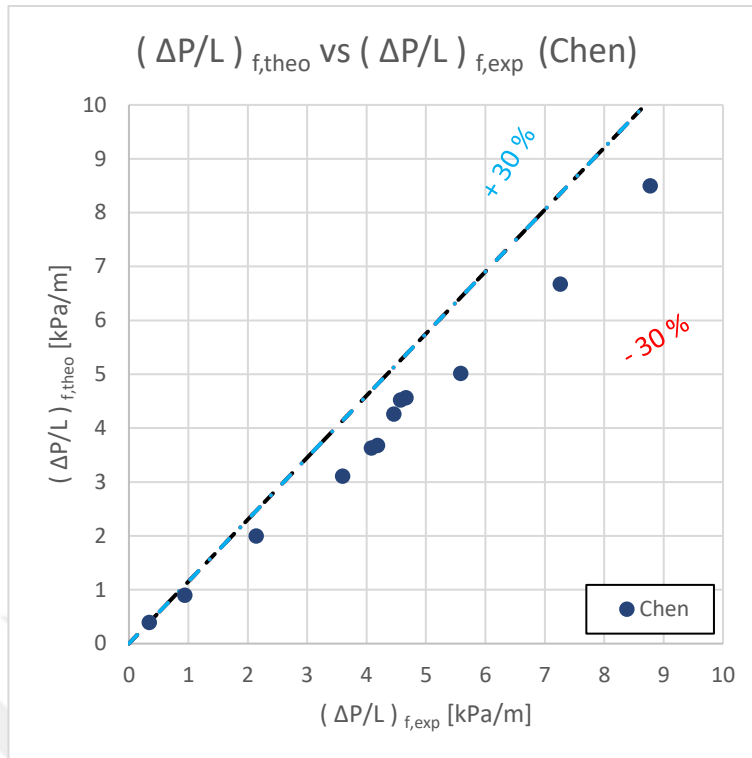


Figure 73: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Chen

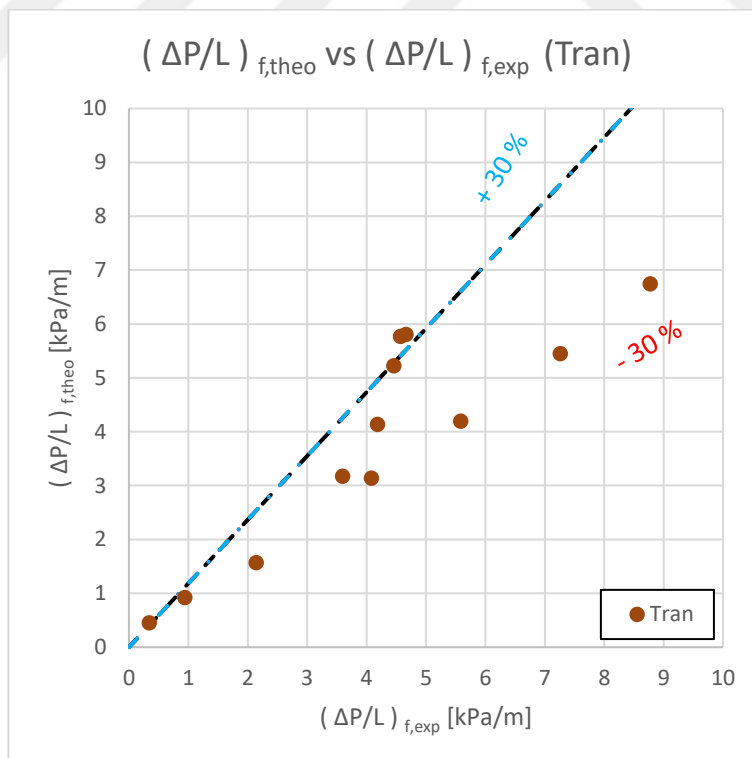


Figure 74: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Tran

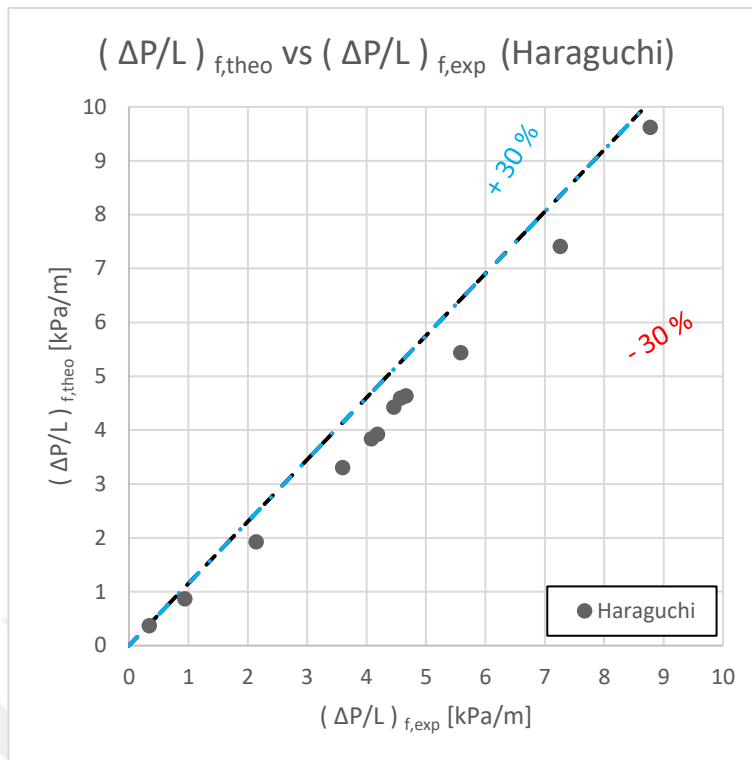


Figure 75: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Haraguchi

7.4 Tube W - Counter Flow

7.4.1 Results for Heat Transfer Coefficient

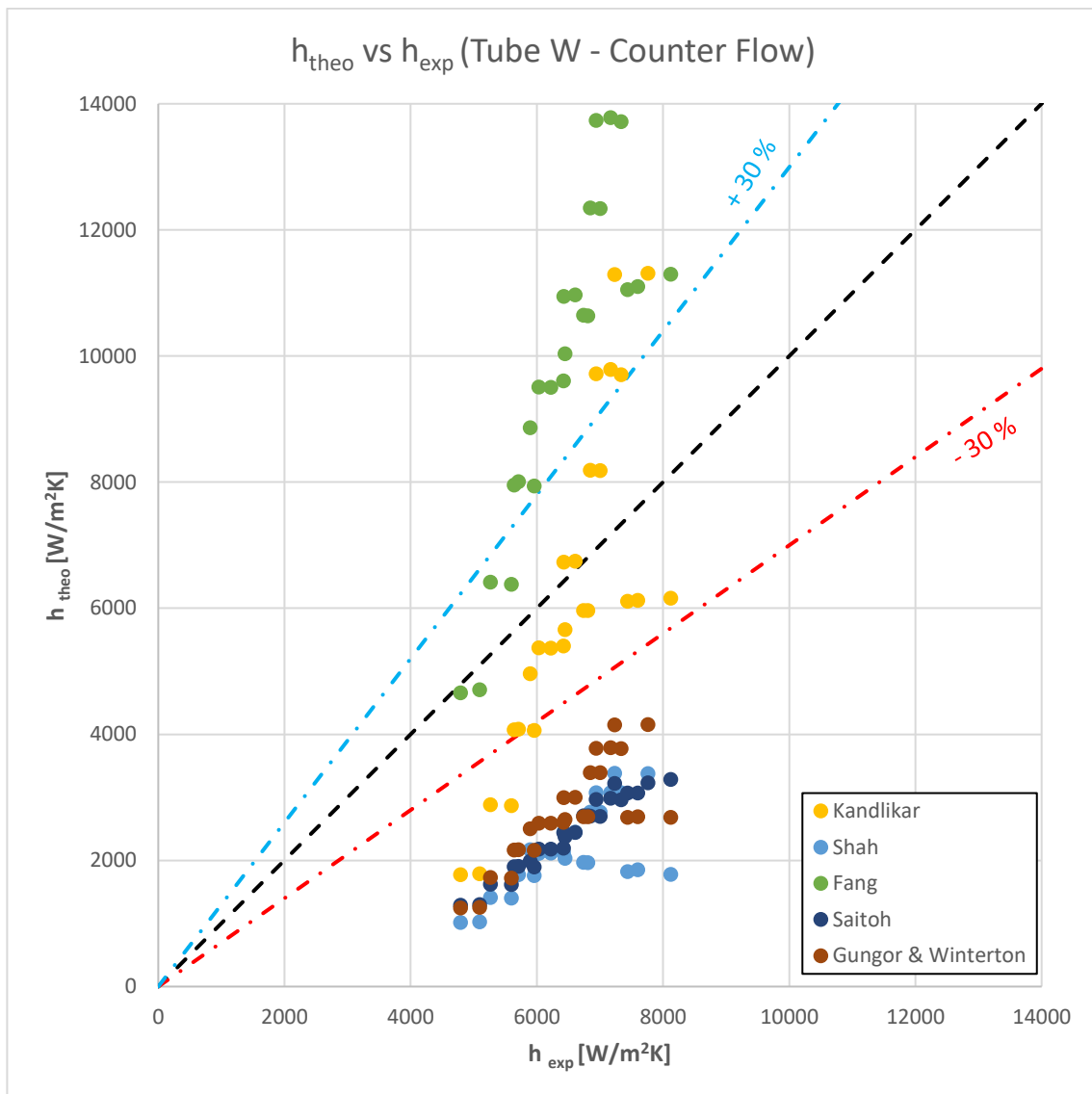


Figure 76: Tube W - Counter Flow / h_{theo} vs h_{exp} for All of the Model

The best result is provided by Kandlikar correlation with 61.54 % of the data set staying within the boundary of $\pm 30\%$. Besides, MRD and MARD value of this correlation are -8.07% and 27.62% (see Table 22), respectively and the distribution of all the points is shown in Figure 77. On the other hand, the worst result is provided by Shah with MRD and MARD values of -66.97 %. It is important to denote that 0.0% of the data set stays within the boundary of $\pm 30\%$ and this can be seen in Figure 78. Although Saitoh and Gungor & Winterton models provide no data within the boundary of $\pm 30\%$ its MRD and MARD values (-63.34%, 63.34%, -58.49% and 58.49%, respectively) are much lower compared to Shah model. In addition, Kandlikar has a better data distribution as it can be seen in Figure 77.

	Kandlikar	Shah	Fang	Saitoh	Gungor & Winterton
MRD	- 8.07 %	- 66.97 %	54.89 %	- 63.34 %	- 58.49 %
MARD	27.62 %	66.97 %	55.68 %	63.34 %	58.49 %
Number of Data within $\pm 30\%$	16	0	4	0	0
Total Data	26	26	26	26	26
[%] within $\pm 30\%$	61.54 %	0.0 %	15.38 %	0.0 %	0.0 %

Table 22: Error of Heat Transfer Coefficient for Tube W - Counter Flow

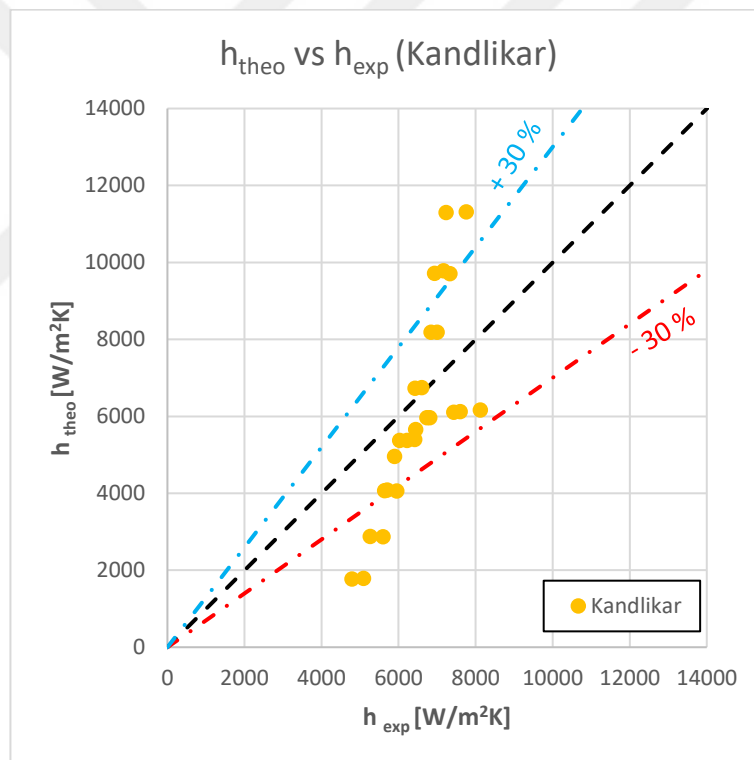


Figure 77: h_{theo} vs h_{exp} for Kandlikar

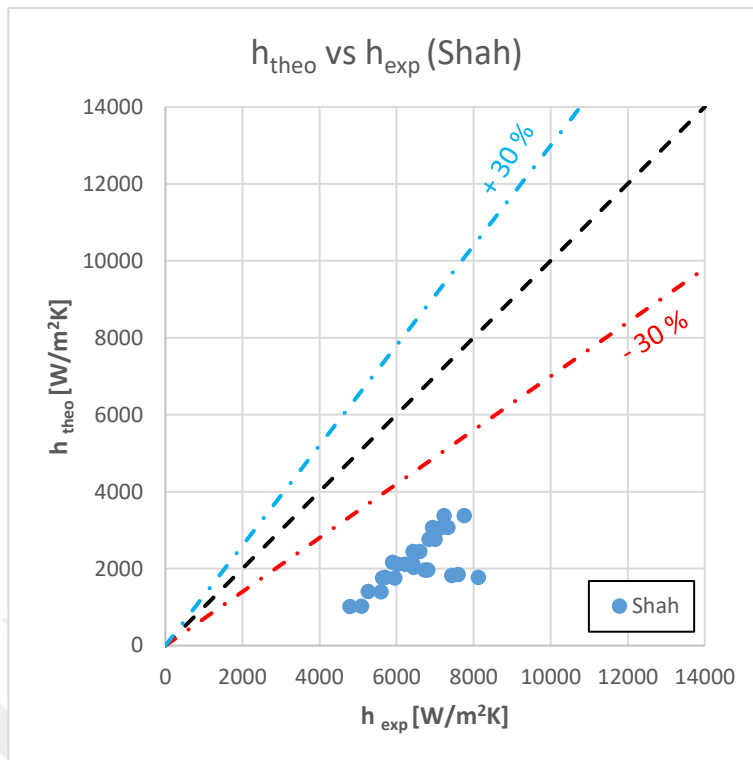


Figure 78: h_{theo} vs h_{exp} for Shah

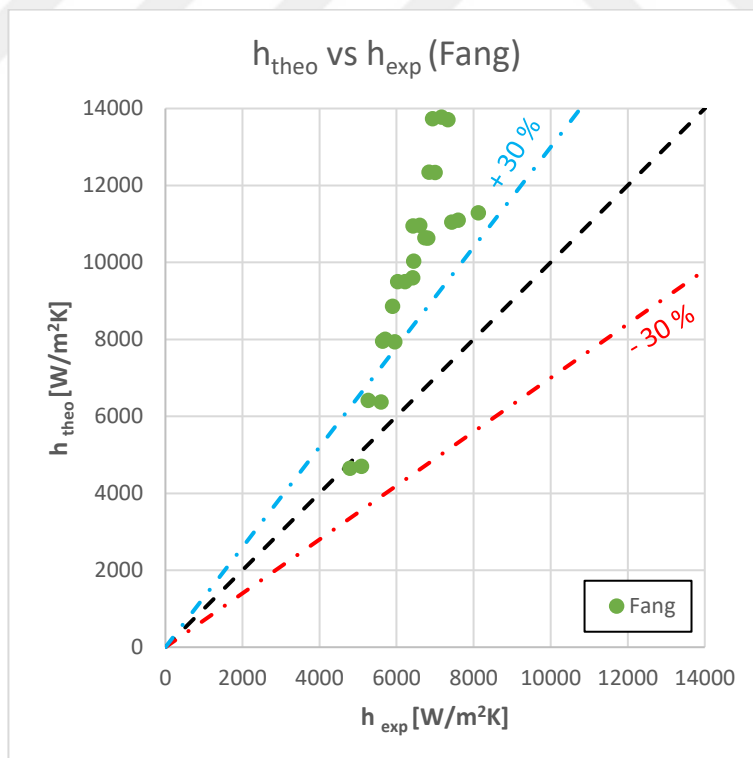


Figure 79: h_{theo} vs h_{exp} for Fang

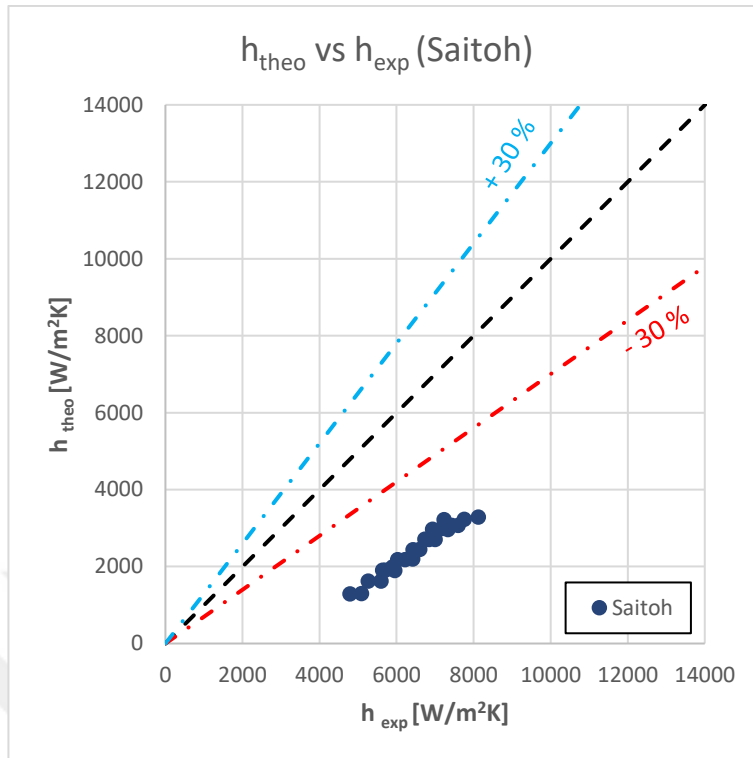


Figure 80: h_{theo} vs h_{exp} for Saitoh

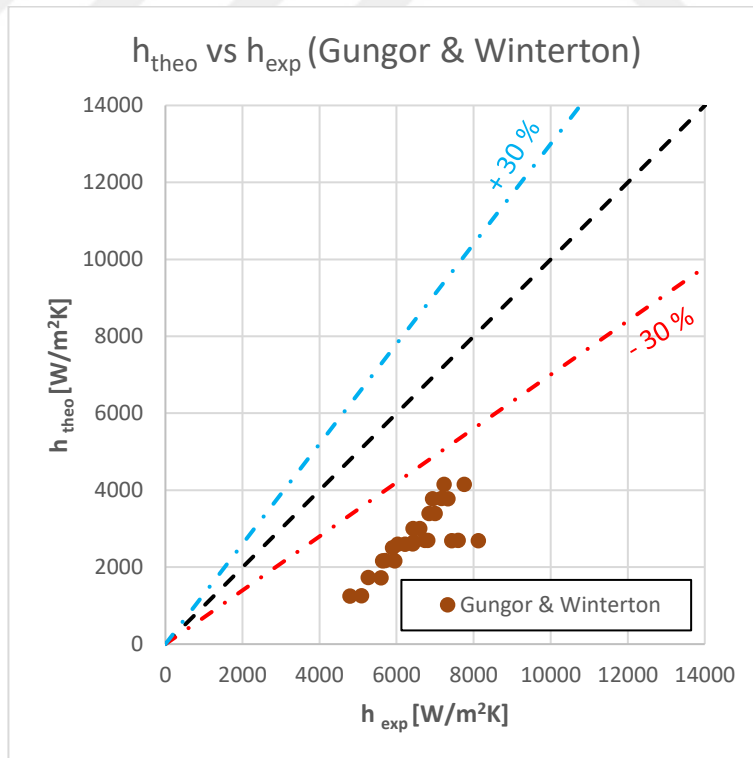


Figure 81: h_{theo} vs h_{exp} for Gungor & Winterton

7.4.2 Results for Frictional Pressure Drop per Unit Length

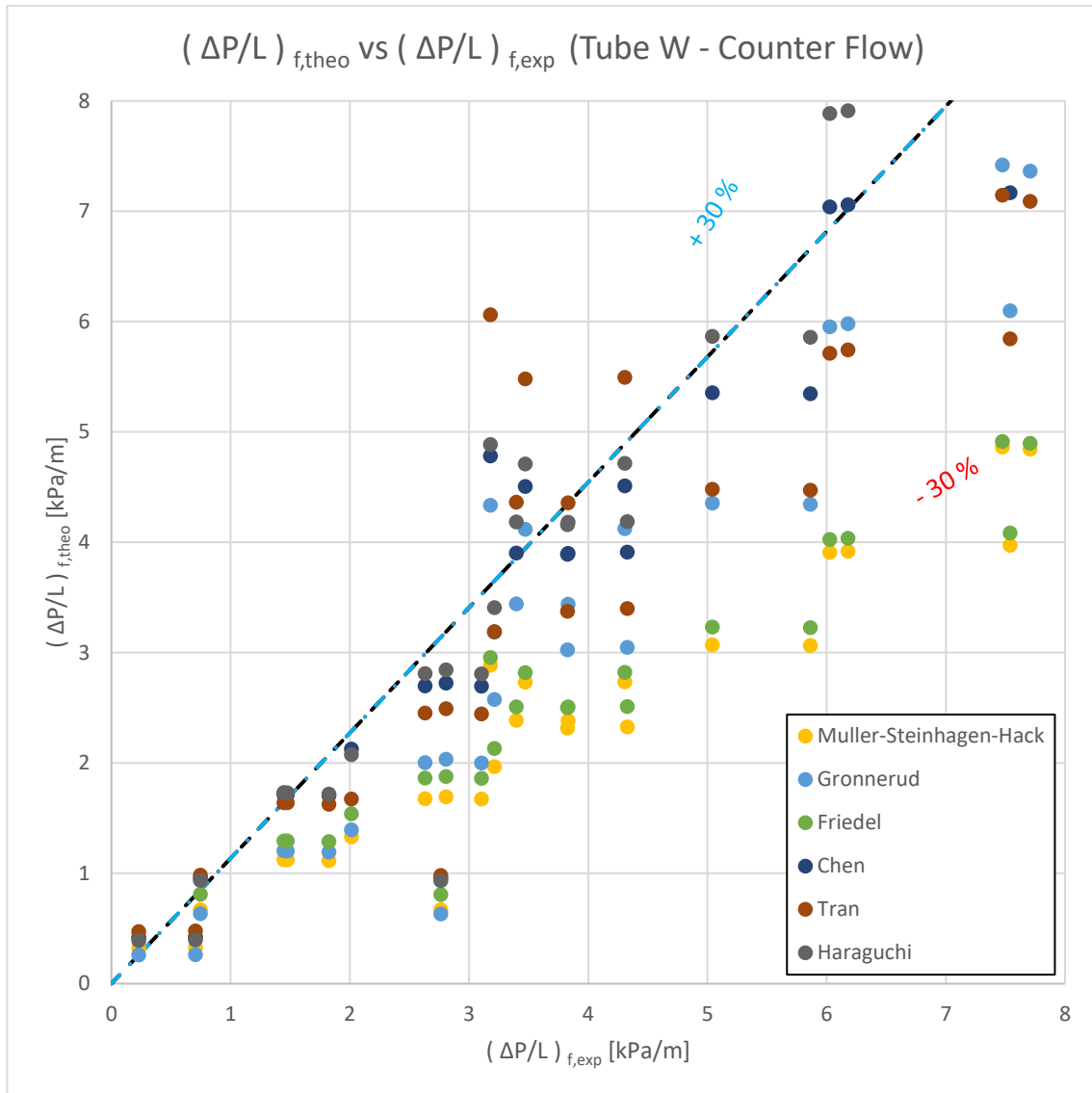


Figure 82: Tube W - Counter Flow / $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for All of the Models

The best result is provided by Chen with 84.62% of the data set staying within the boundary of $\pm 30\%$. Besides, MRD and MARD value of this correlation are 6.93% and 18.54% (see Table 23), respectively and the distribution of all the points is shown in Figure 86. Here, Gronnerud also provides 79.95% of the data set staying within the boundary of $\pm 30\%$ but it has a slightly higher MRD and MARD values. On the other hand, the worst result is provided by Muller-Steinhagen-Heck with MRD and MARD values of -33.70% and 36.88%, respectively. It is important to denote that only 23.08% of the data set stays within the boundary of $\pm 30\%$ and this can be seen in Figure 83. All of the models provide acceptable results for this type of micro-fin tube except Muller-Steinhagen-Heck and Friedel.

	Muller- Steinhagen- Heck	Gronnerud	Friedel	Chen	Tran	Haraguchi
MRD	- 33.70 %	- 16.36 %	- 26.72 %	6.93 %	3.93 %	12.16 %
MARD	36.88 %	21.76 %	33.74 %	18.64 %	25.43 %	22.05 %
Number of Data within $\pm$ 30%	6	20	9	22	20	18
Total Data	26	26	26	26	26	26
[%] within $\pm$ 30%	23.08 %	79.92 %	34.62 %	84.62 %	76.92 %	69.23 %

Table 23: Error of Frictional Pressure Drop per Unit Length for Tube W – Counter Flow

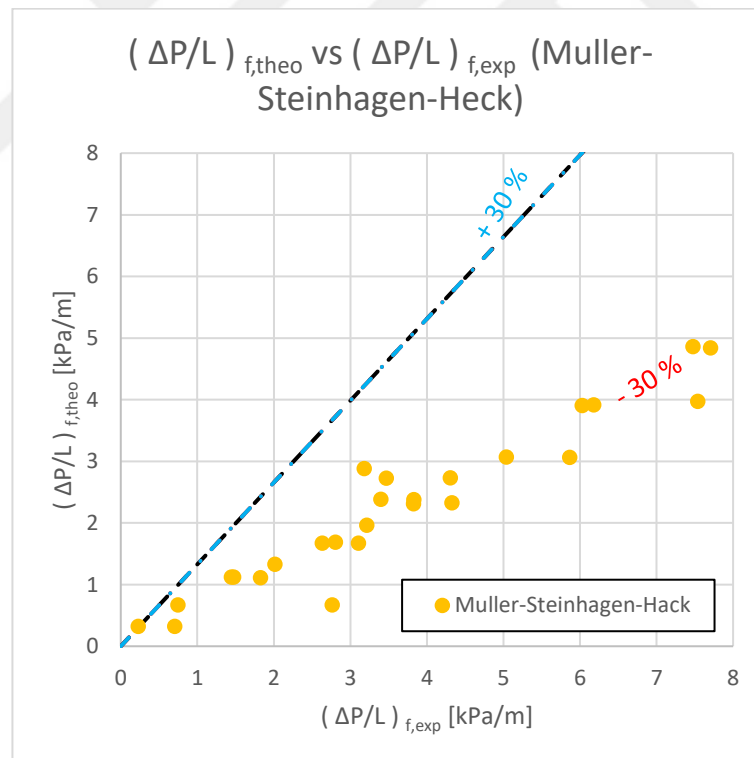


Figure 83: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Muller-Steinhagen-Heck

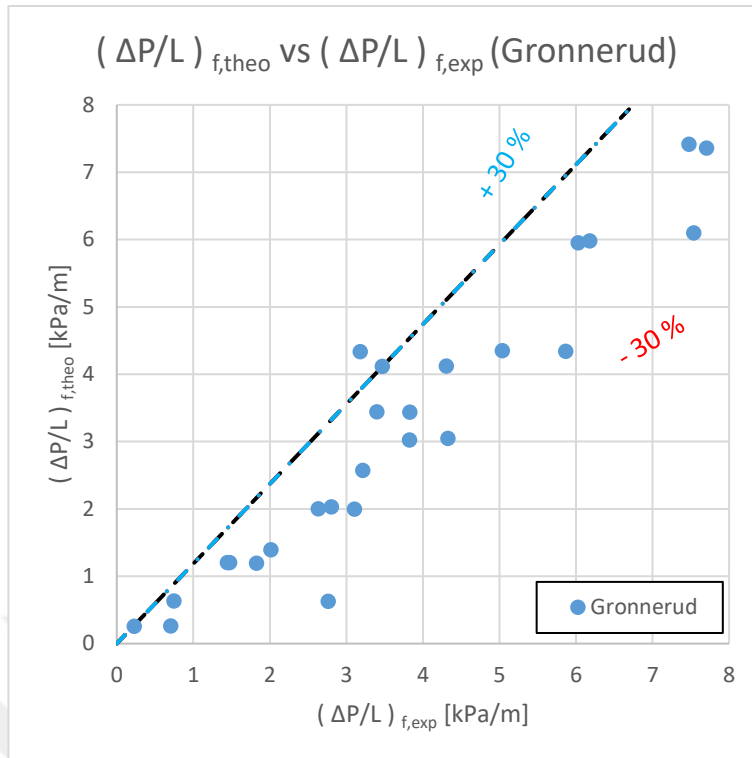


Figure 84: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Gronnerud

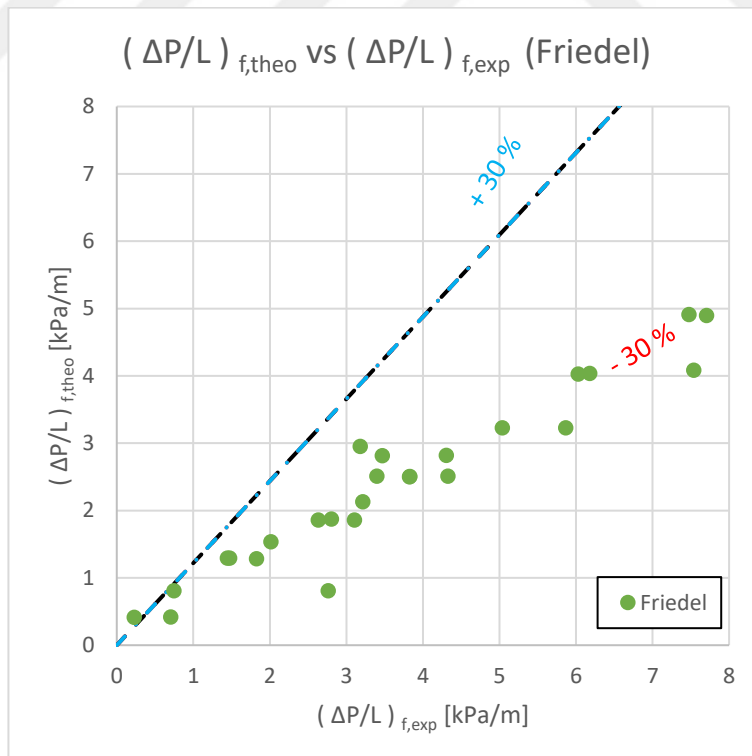


Figure 85: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Friedel

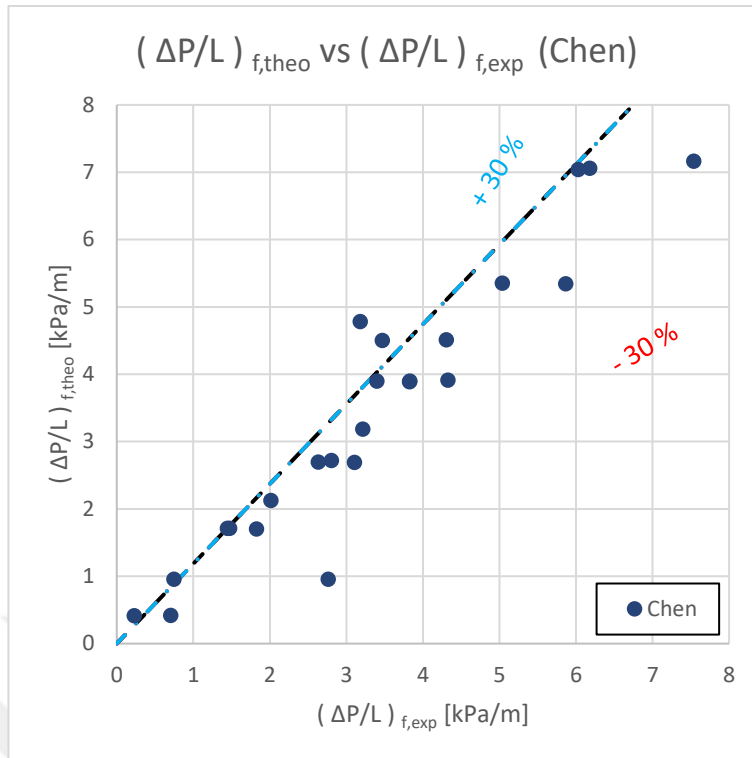


Figure 86: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Chen

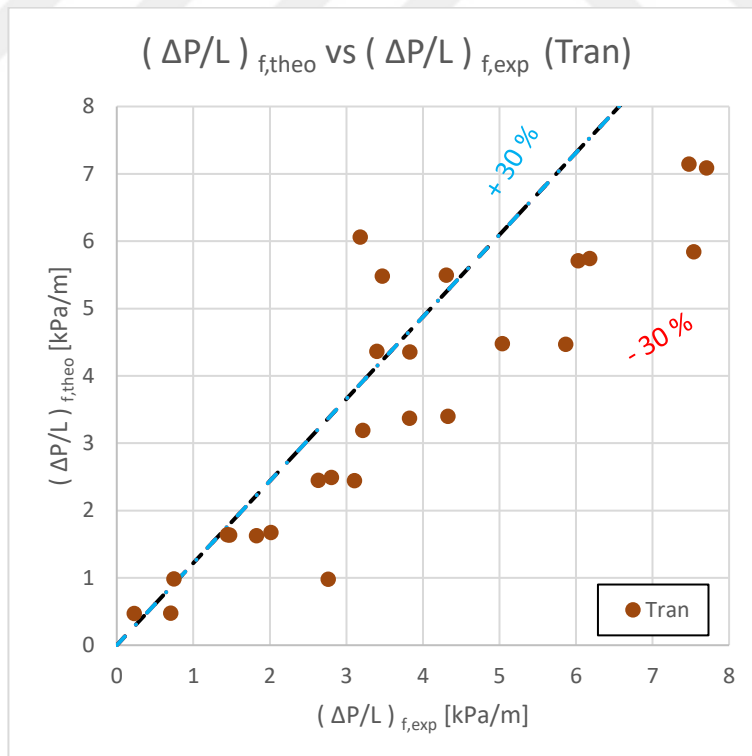


Figure 87: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Tran

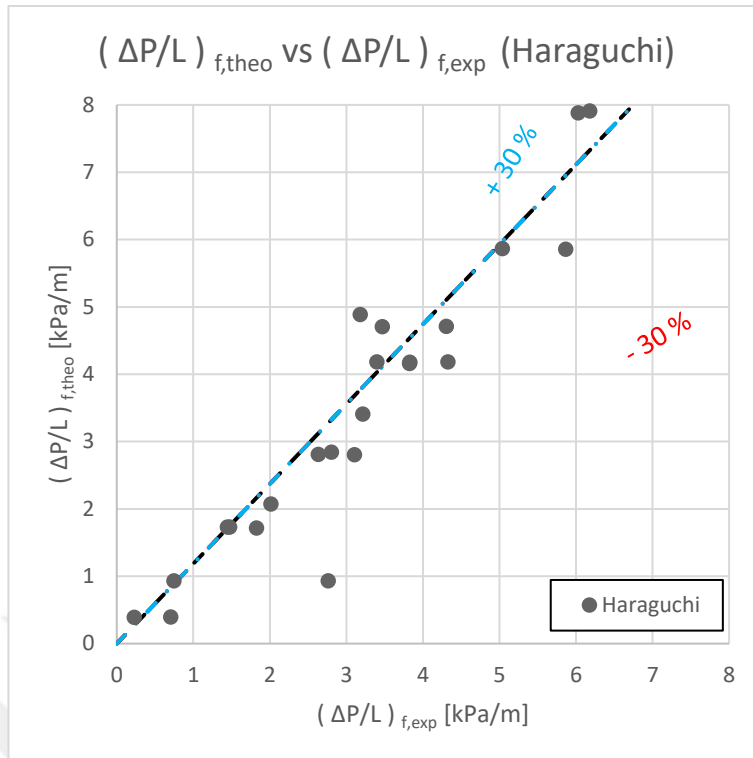


Figure 88: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Haraguchi

7.5 Tube VA - Counter Flow

7.5.1 Results for Heat Transfer Coefficient

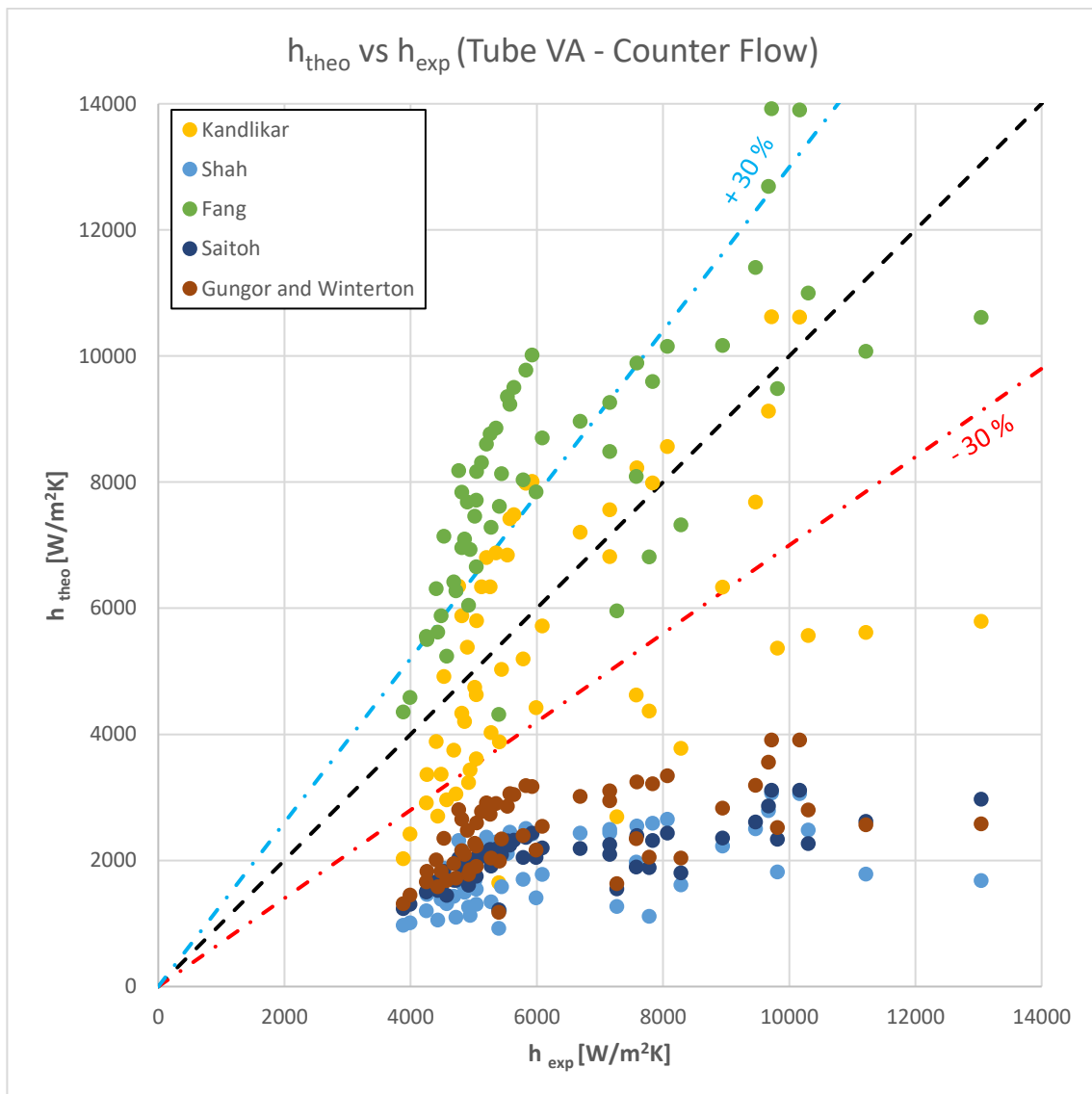


Figure 89: Tube VA - Counter Flow / h_{theo} vs h_{exp} for All of the Model

The best result is provided by Kandlikar correlation with 59.65% of the data set staying within the boundary of $\pm 30\%$. Besides, MRD and MARD value of this correlation are -11.62% and 25.61% (see Table 24), respectively and the distribution of all the points is shown in Figure 90. On the other hand, the worst result is provided by Shah with MRD and MARD values of -69.22 % and 69.22%. It is important to denote that 0.0% of the data set stays within the boundary of $\pm 30\%$ and this can be seen in Figure 91. Although Saitoh and Gungor & Winterton models provide no data within the boundary of $\pm 30\%$ its MRD and MARD values (-66.11%, 66.11%, -59.18% and 59.18%, respectively) are lower compared to Shah model. In addition, Kandlikar has a better data distribution as it can be seen in Figure 90.

	Kandlikar	Shah	Fang	Saitoh	Gungor & Winterton
MRD	- 11.62 %	- 69.22 %	34.05 %	- 66.11 %	- 59.18 %
MARD	25.61 %	69.22 %	37.36 %	66.11 %	59.18 %
Number of Data within $\pm 30\%$	34	0	21	0	0
Total Data	57	57	57	57	57
[%] within $\pm 30\%$	59.65 %	0.0 %	36.84 %	0.0 %	0.0 %

Table 24: Error of Heat Transfer Coefficient for Tube VA - Counter Flow

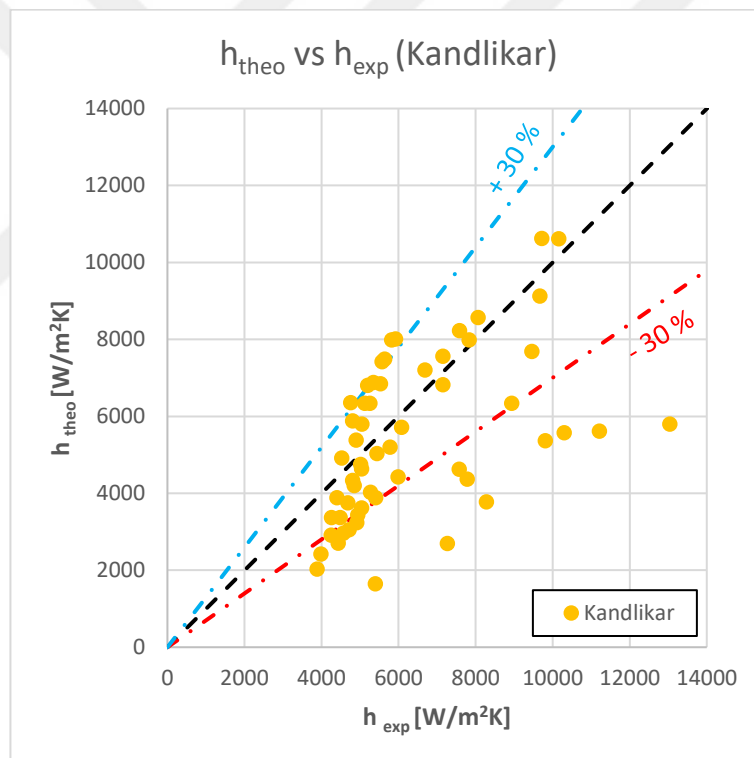


Figure 90: h_{theo} vs h_{exp} for Kandlikar

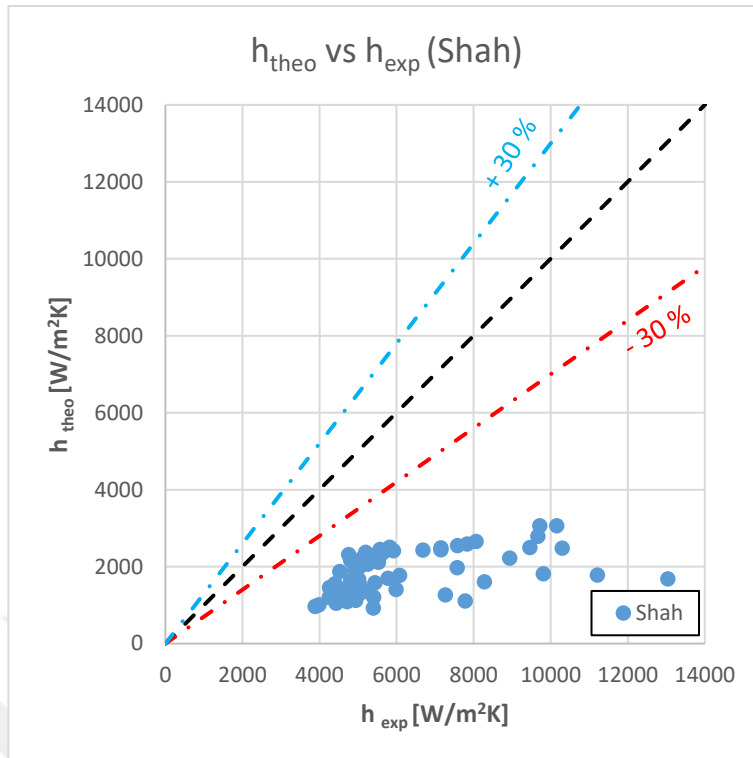


Figure 91: h_{theo} vs h_{exp} for Shah

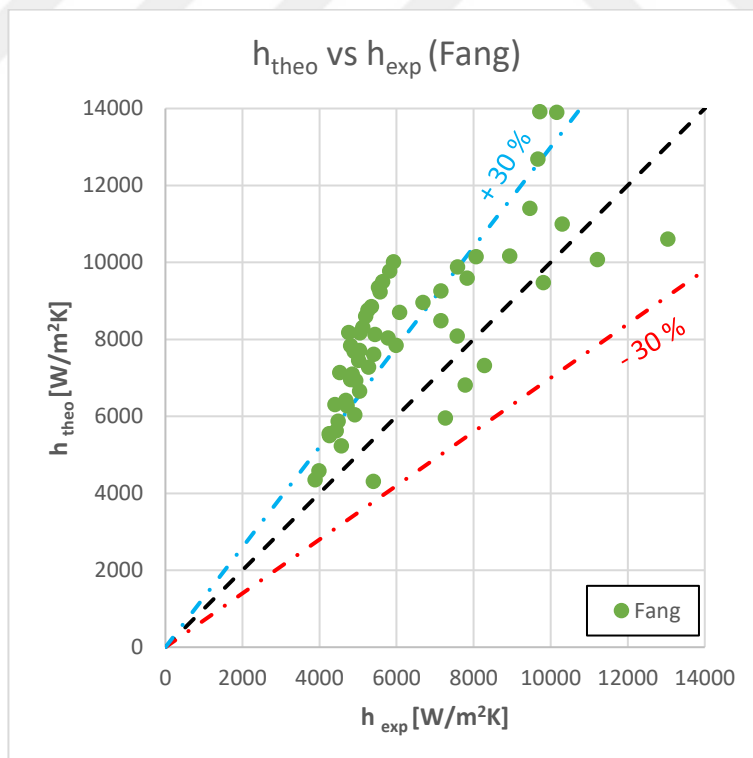


Figure 92: h_{theo} vs h_{exp} for Fang

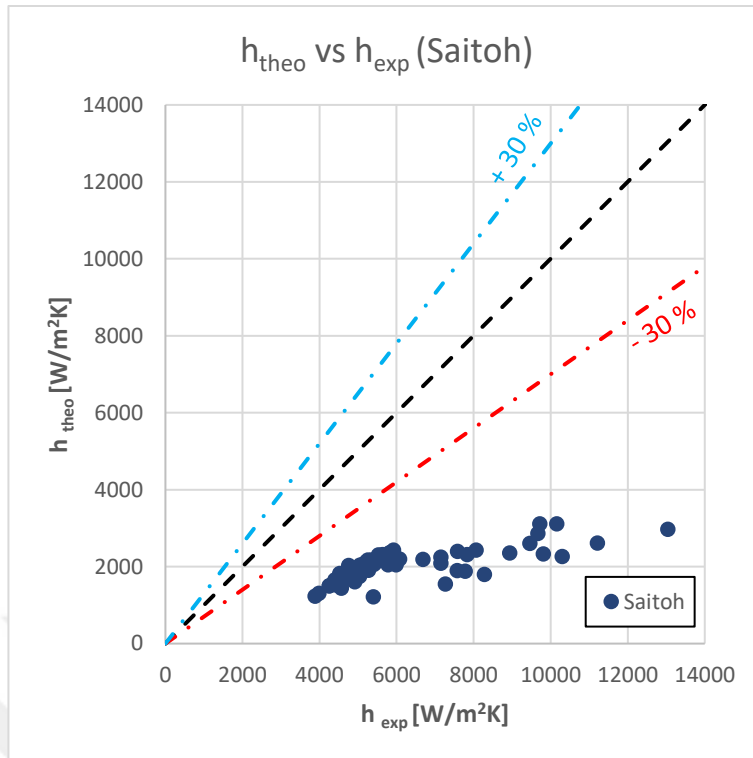


Figure 93: h_{theo} vs h_{exp} for Saitoh

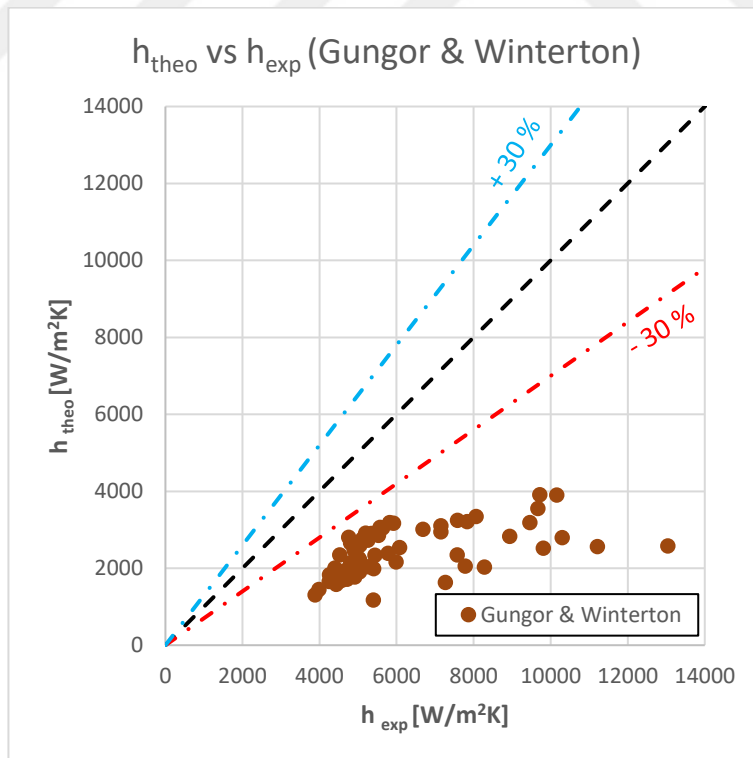


Figure 94: h_{theo} vs h_{exp} for Gungor & Winterton

7.5.2 Results for Frictional Pressure Drop per Unit Length

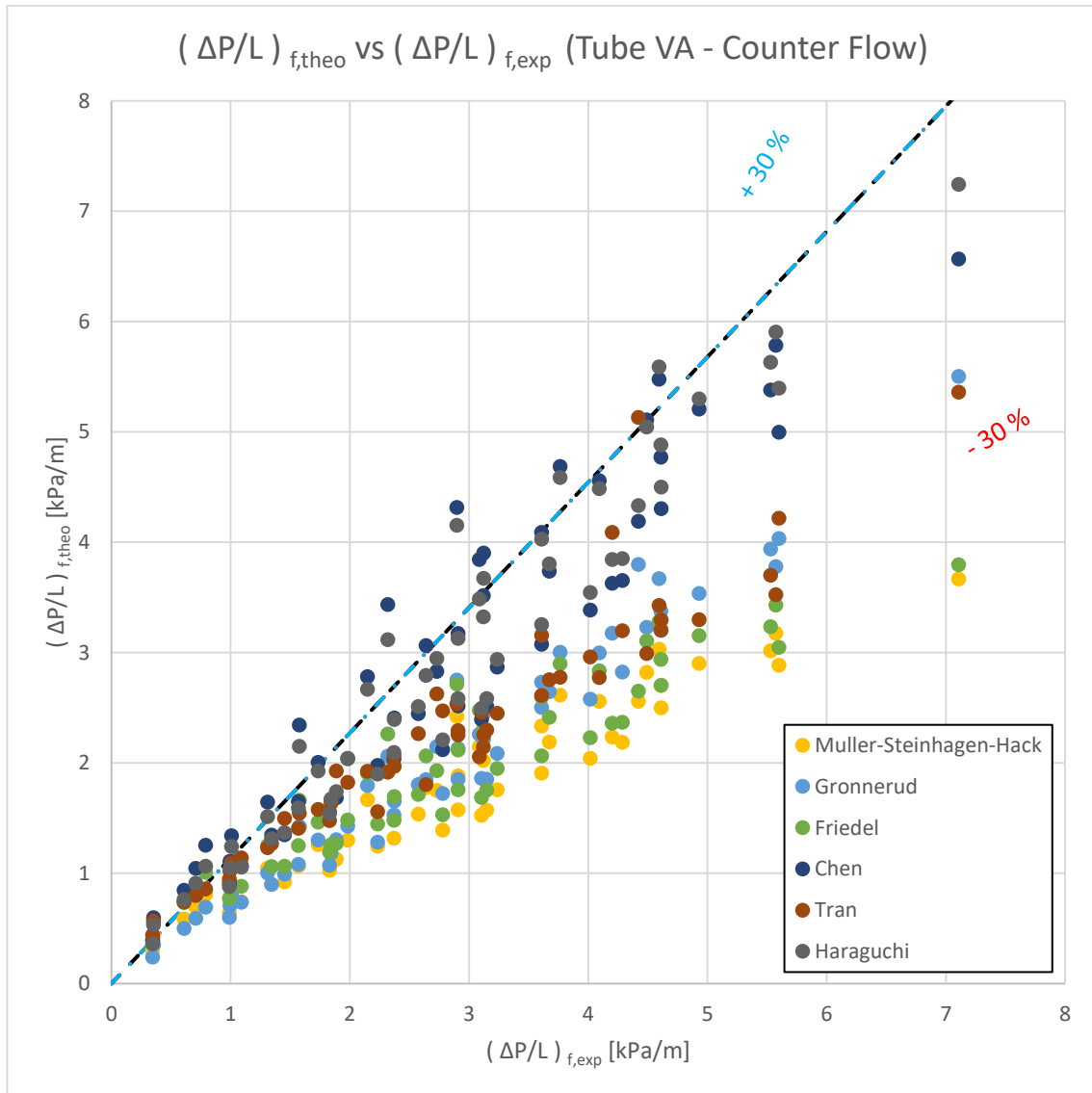


Figure 95: Tube VA - Counter Flow / $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for All of the Models

The best result is provided by Haraguchi with 91.23% of the data set staying within the boundary of $\pm 30\%$. Besides, MRD and MARD value of this correlation are 5.45% and 12.81% (see Table 25), respectively and the distribution of all the points is shown in Figure 101. Here, Chen also provides 85.96% of the data set staying within the boundary of $\pm 30\%$ but it has a slightly higher MRD and MARD values. On the other hand, the worst result is provided by Muller-Steinhagen-Heck with MRD and MARD values of -33.93% and 34.82%, respectively. It is important to denote that only 24.56% of the data set stays within the boundary of $\pm 30\%$ and this can be seen in Figure 96. All of the models provide acceptable results for this type of micro-fin tube except Muller-Steinhagen-Heck.

	Muller- Steinhagen- Heck	Gronnerud	Friedel	Chen	Tran	Haraguchi
MRD	- 33.93 %	- 26.81 %	- 24.44 %	7.23 %	- 13.49 %	5.45 %
MARD	34.82 %	26.81 %	29.83 %	16.98 %	19.49 %	12.81 %
Number of Data within $\pm$ 30%	14	36	28	49	46	52
Total Data	57	57	57	57	57	57
[%] within $\pm$ 30%	24.56 %	63.16 %	49.12 %	85.96 %	80.70 %	91.23 %

Table 25: Error of Frictional Pressure Drop per Unit Length for Tube VA – Counter Flow

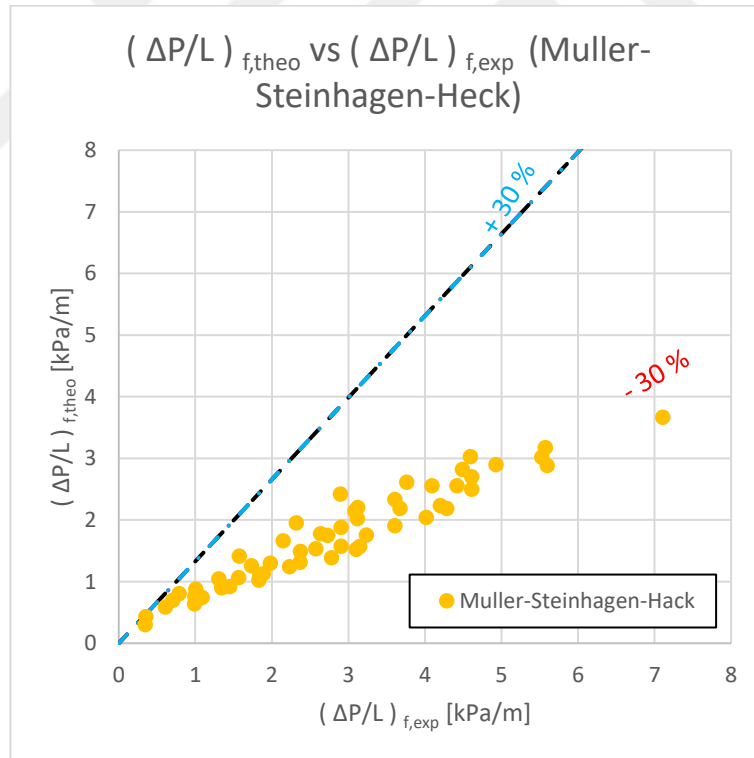


Figure 96: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Muller-Steinhagen-Heck

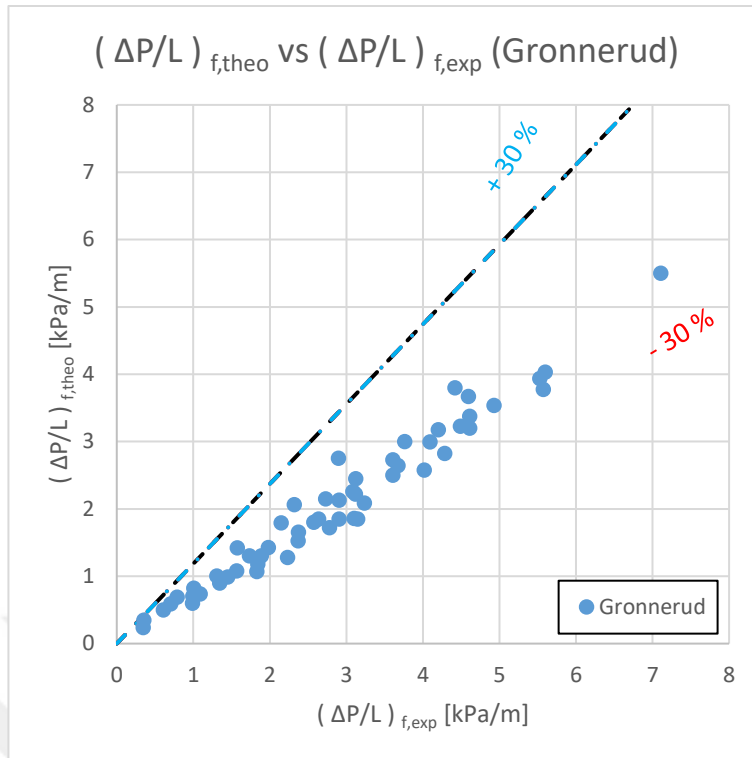


Figure 97: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Gronnerud

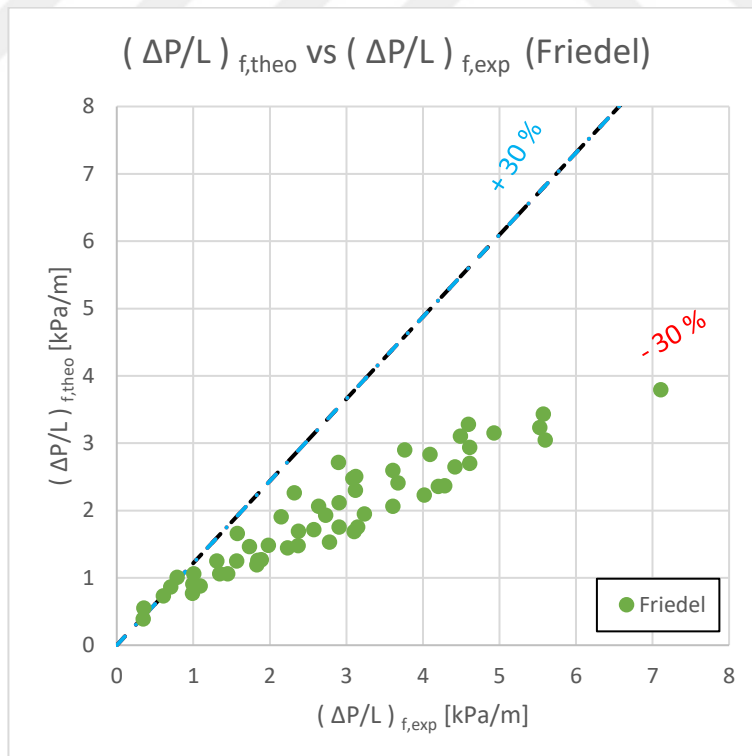


Figure 98: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Friedel

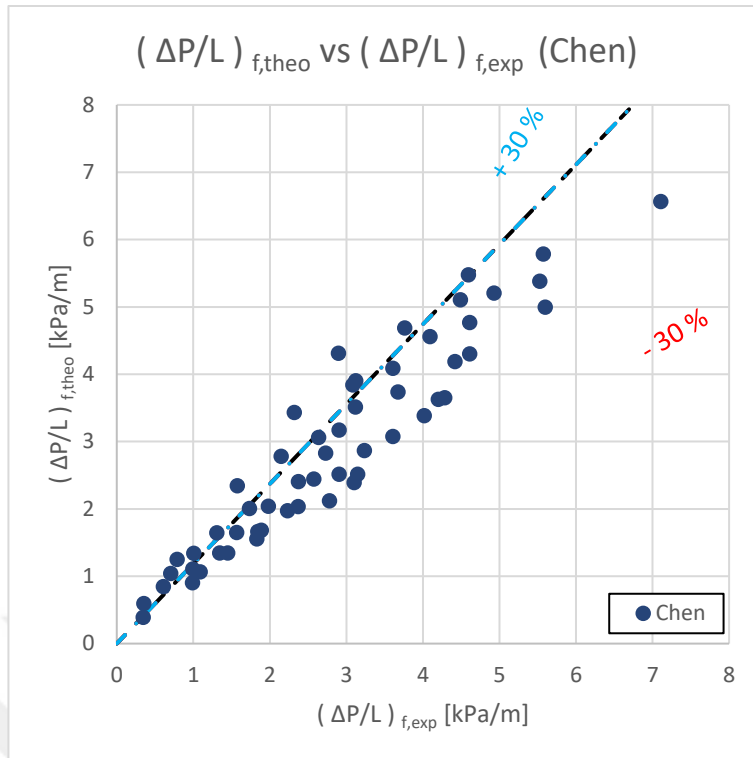


Figure 99: $(\Delta P/L)_{f,theo} \text{ vs } (\Delta P/L)_{f,exp}$ for Chen

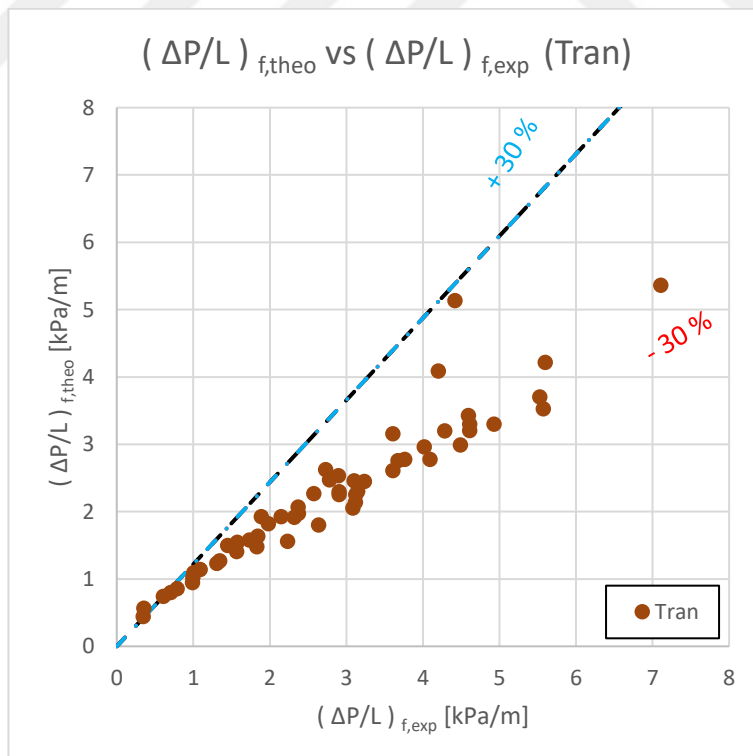


Figure 100: $(\Delta P/L)_{f,theo} \text{ vs } (\Delta P/L)_{f,exp}$ for Tran

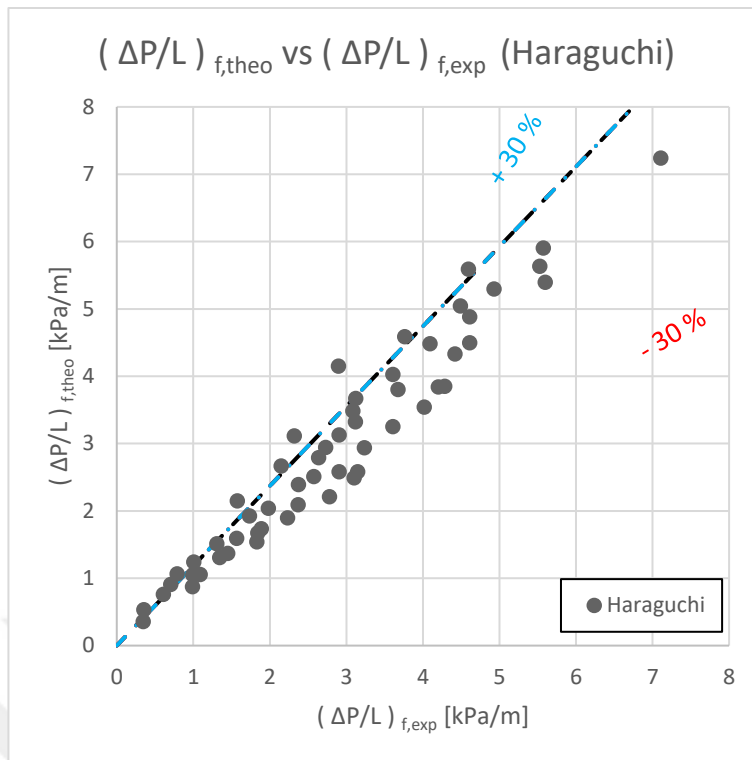


Figure 101: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Haraguchi

7.6 Tube HVA - Counter Flow

7.6.1 Results for Heat Transfer Coefficient

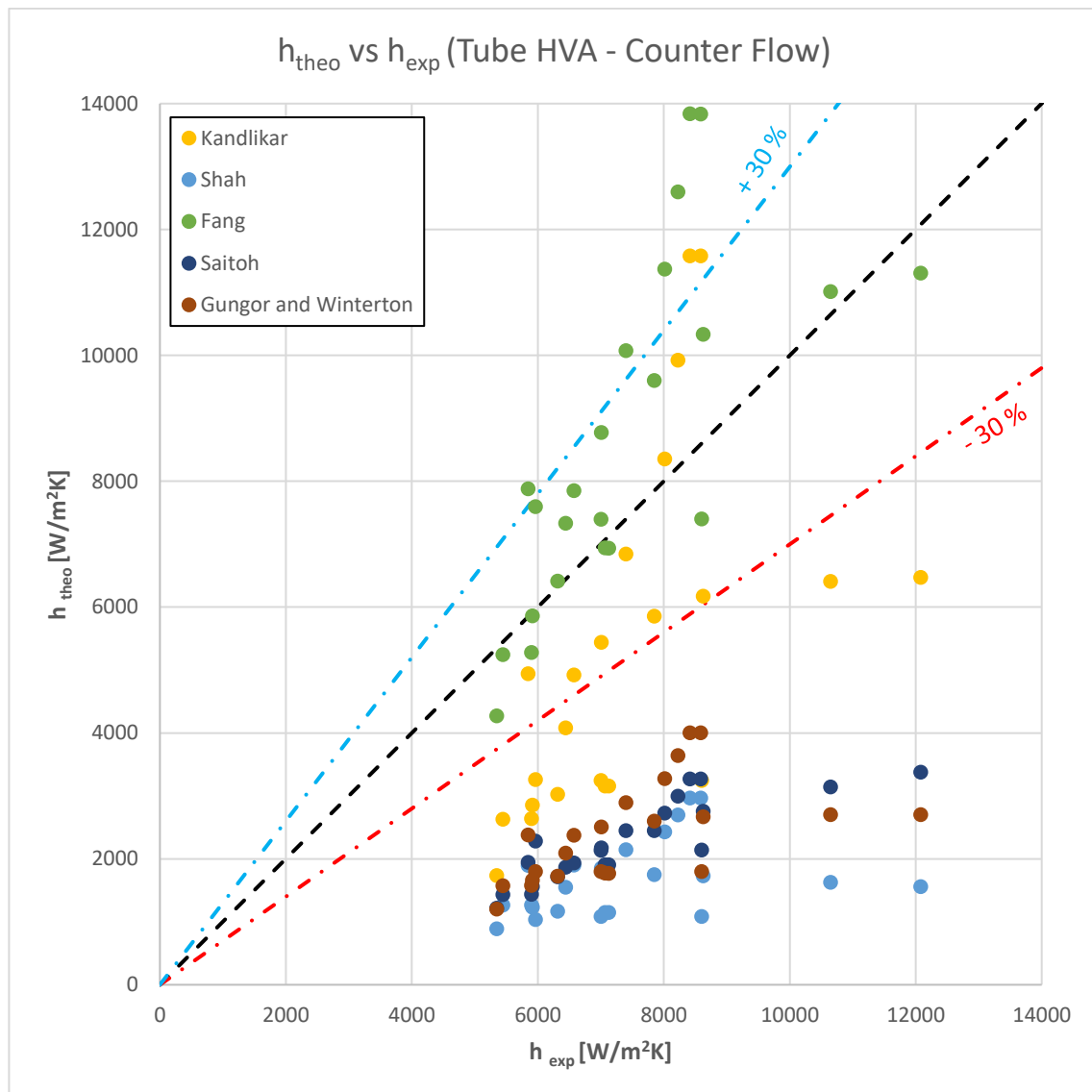


Figure 102: Tube HVA - Counter Flow / h_{theo} vs h_{exp} for All of the Model

The best result is provided by Fang correlation with 73.91% of the data set staying within the boundary of $\pm 30\%$. Besides, MRD and MARD value of this correlation are -16.11% and 21.32% (see Table 26), respectively and the distribution of all the points is shown in Figure 105. On the other hand, the worst result is provided by Shah with MRD and MARD values of -77.28% and 77.28%. It is important to denote that 0.0% of the data set stays within the boundary of $\pm 30\%$ and this can be seen in Figure 104. Although Saitoh and Gungor & Winterton models provide no data within the boundary of $\pm 30\%$ its MRD and MARD values (-69.65%, 69.65%, -68.02% and -68.02%, respectively) are much lower compared to Fang model. In addition, Fang has a better data distribution as it can be seen in Figure 105.

	Kandlikar	Shah	Fang	Saitoh	Gungor & Winterton
MRD	- 30.45 %	- 77.28 %	16.11 %	- 69.65 %	- 68.02 %
MARD	38.92 %	77.28 %	21.32 %	69.65 %	68.02 %
Number of Data within $\pm 30\%$	8	0	17	0	0
Total Data	23	23	23	23	23
[%] within $\pm 30\%$	34.78 %	0.0%	73.91 %	0.0 %	0.0 %

Table 26: Error of Heat Transfer Coefficient for Tube HVA - Counter Flow

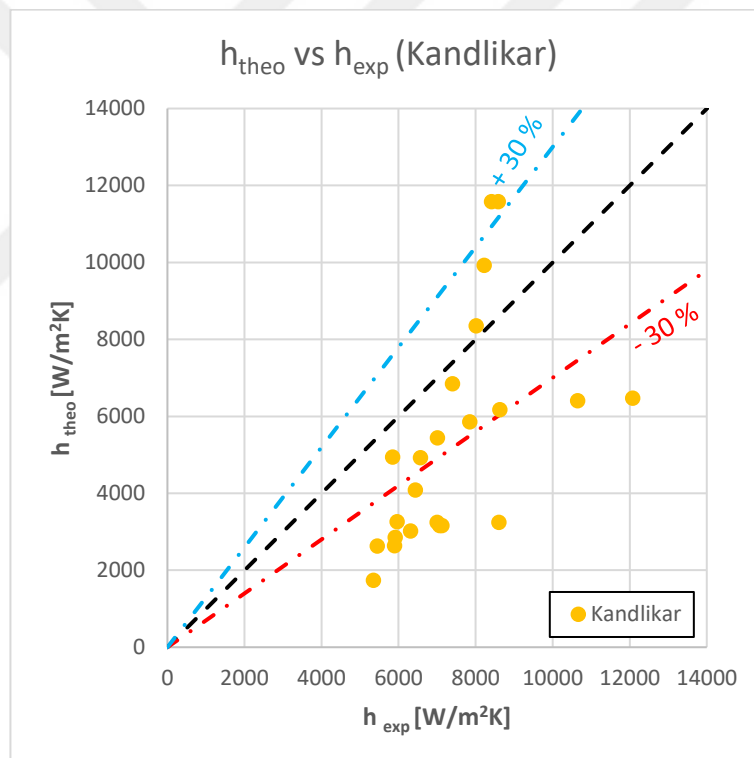


Figure 103: h_{theo} vs h_{exp} for Kandlikar

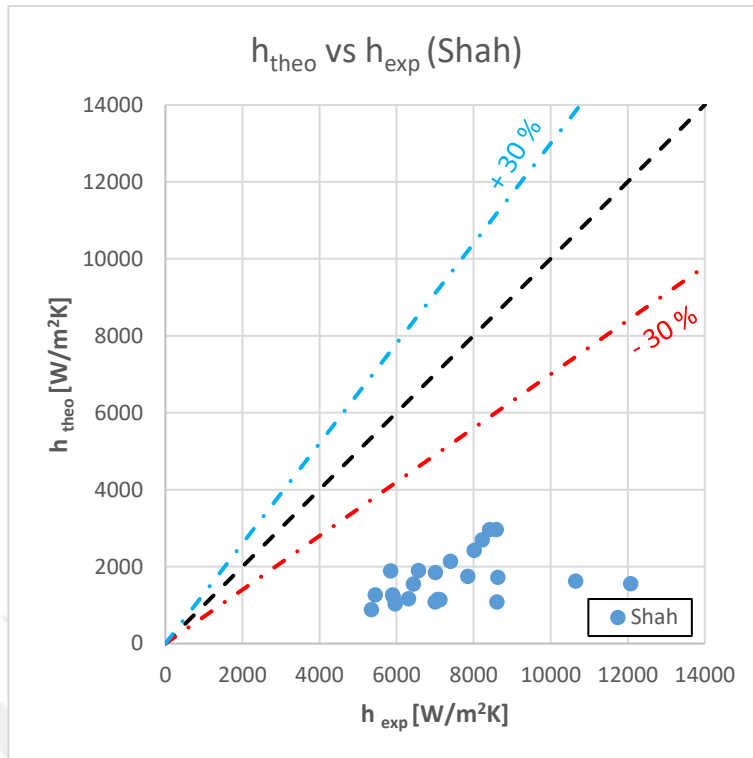


Figure 104: h_{theo} vs h_{exp} for Shah

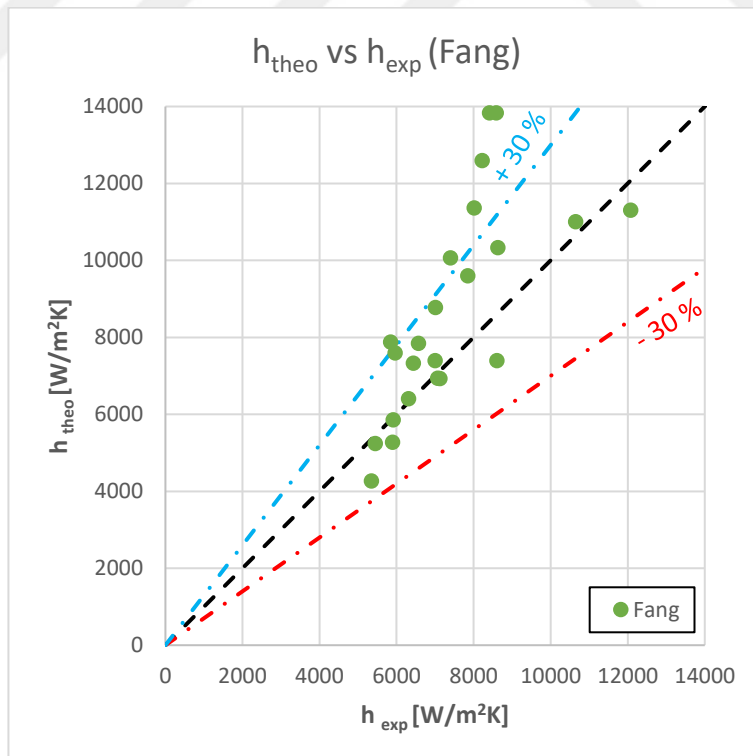


Figure 105: h_{theo} vs h_{exp} for Fang

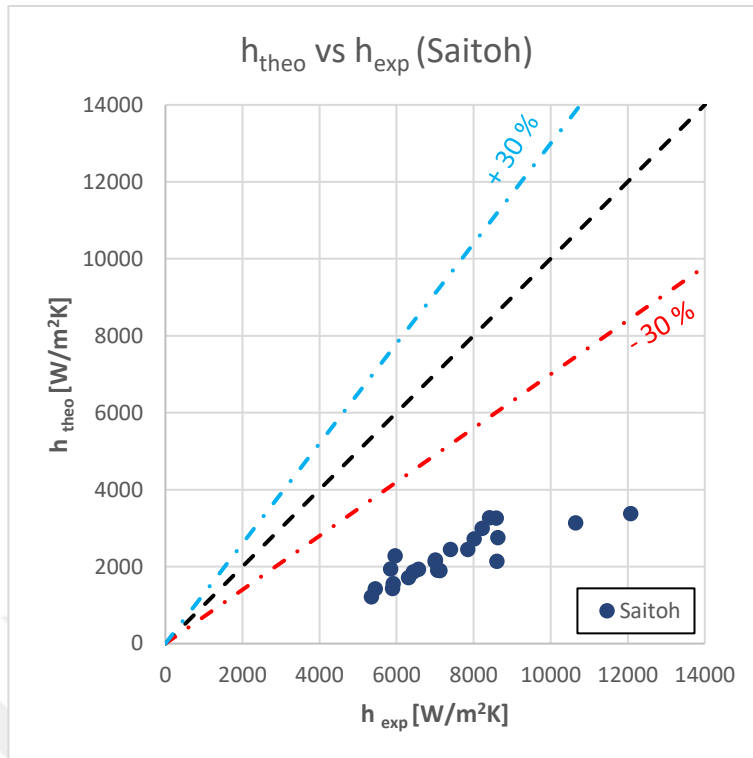


Figure 106: h_{theo} vs h_{exp} for Saitoh

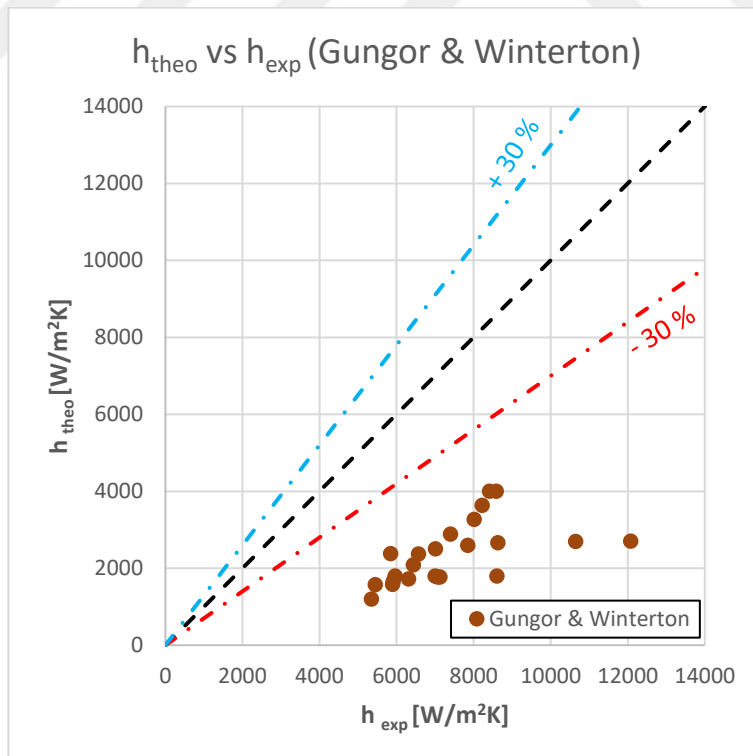


Figure 107: h_{theo} vs h_{exp} for Gungor & Winterton

7.6.2 Results for Frictional Pressure Drop per Unit Length

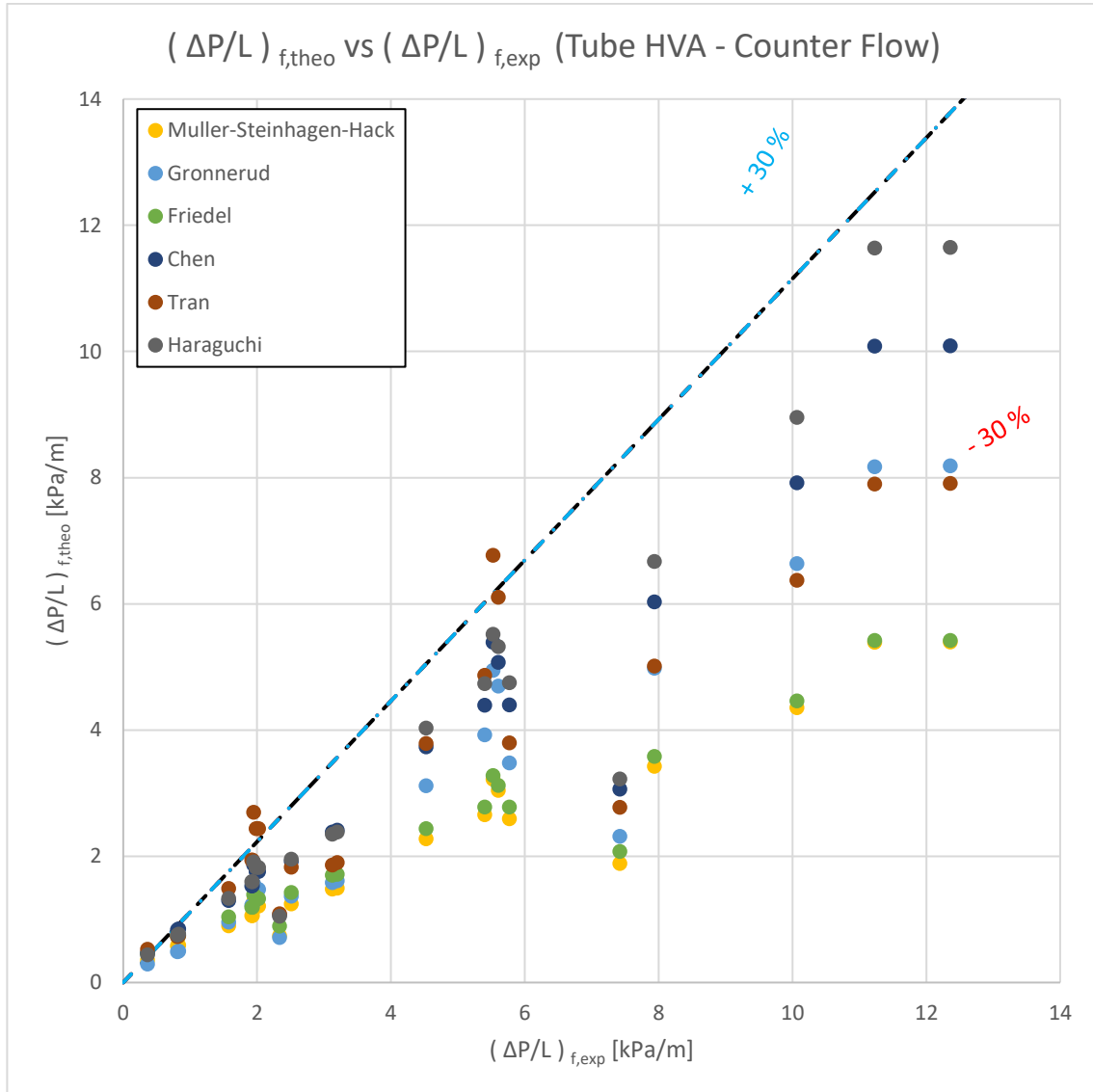


Figure 108: Tube HVA - Counter Flow / $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for All of the Models

The best result is provided by Haraguchi with 91.30% of the data set staying within the boundary of $\pm 30\%$. Besides, MRD and MARD value of this correlation are -13.81% and 16.16% (see Table 27), respectively and the distribution of all the points is shown in Figure 114. Here, Chen also provides 86.96% of the data set staying within the boundary of $\pm 30\%$ but it has a slightly higher MRD and MARD values. On the other hand, the worst result is provided by Muller-Steinhagen-Heck with MRD and MARD values of -46.31% and 46.31%, respectively. It is important to denote that only 13.04% of the data set stays within the boundary of $\pm 30\%$ and this can be seen in Figure 109. All of the models provide acceptable results for this type of micro-fin tube except Muller-Steinhagen-Heck and Friedel.

	Muller- Steinhagen- Heck	Gronnerud	Friedel	Chen	Tran	Haraguchi
MRD	- 46.31 %	- 35.66 %	-39.76 %	- 16.50 %	- 12.46 %	- 13.81 %
MARD	46.31 %	35.66 %	42.21 %	19.68 %	26.55 %	16.16 %
Number of Data within $\pm$ 30%	3	8	4	20	13	21
Total Data	23	23	23	23	23	23
[%] within $\pm$ 30%	13.04 %	34.78 %	17.39 %	86.96 %	56.52 %	91.30 %

Table 27: Error of Frictional Pressure Drop per Unit Length for Tube HVA – Counter Flow

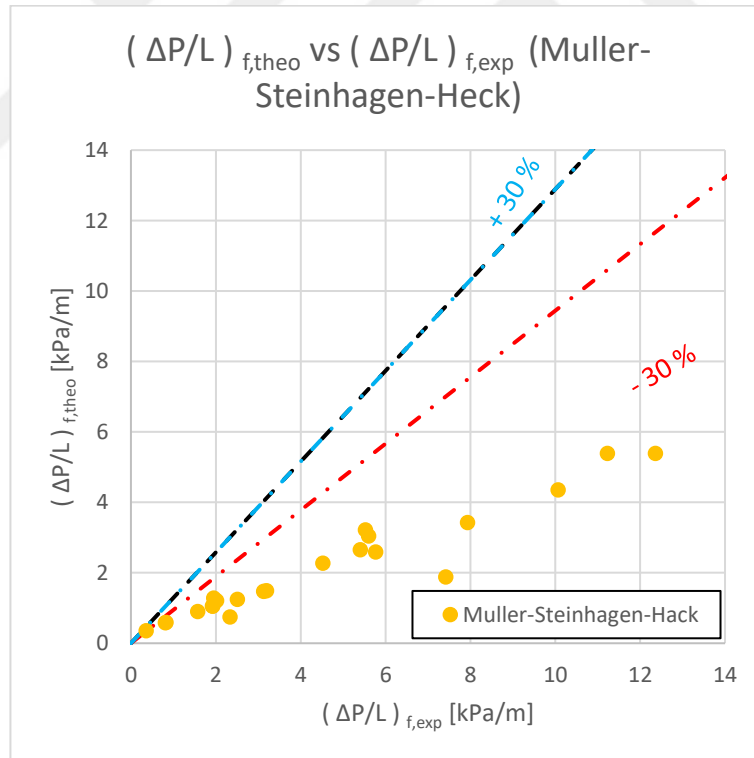


Figure 109: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Muller-Steinhagen-Heck

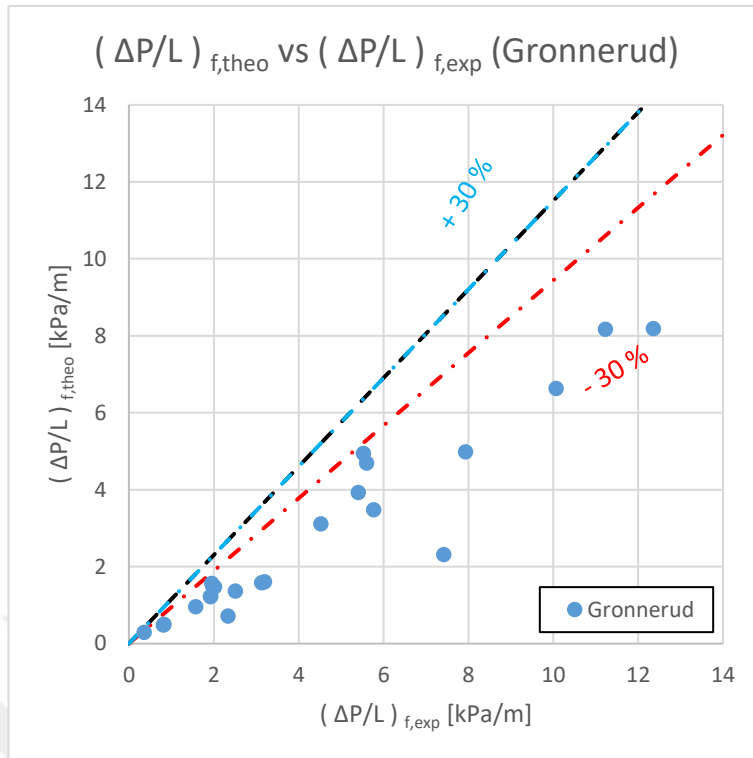


Figure 110: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Gronnerud

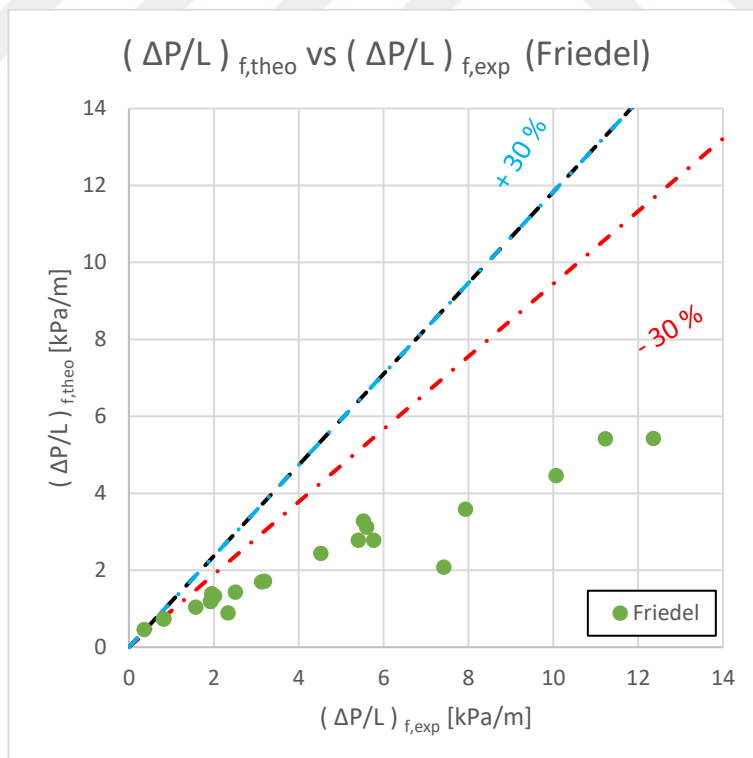


Figure 111: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Friedel

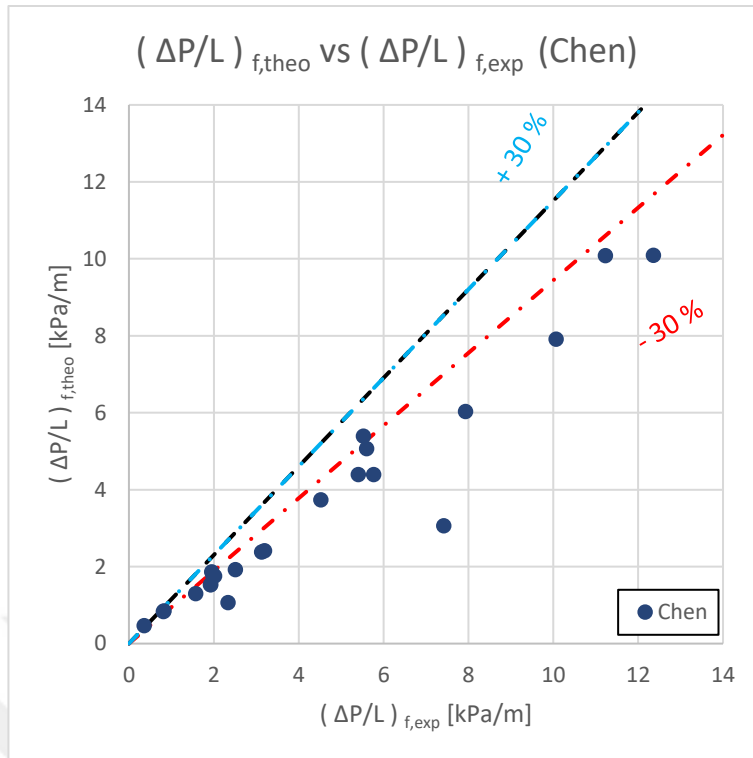


Figure 112: $(\Delta P/L)_{f,theo} \text{ vs } (\Delta P/L)_{f,exp}$ for Chen

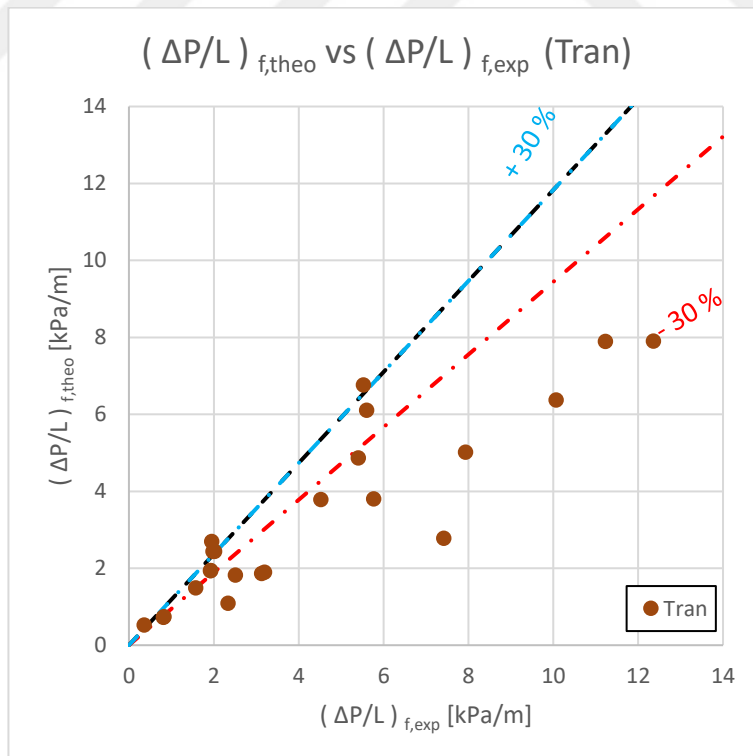


Figure 113: $(\Delta P/L)_{f,theo} \text{ vs } (\Delta P/L)_{f,exp}$ for Tran

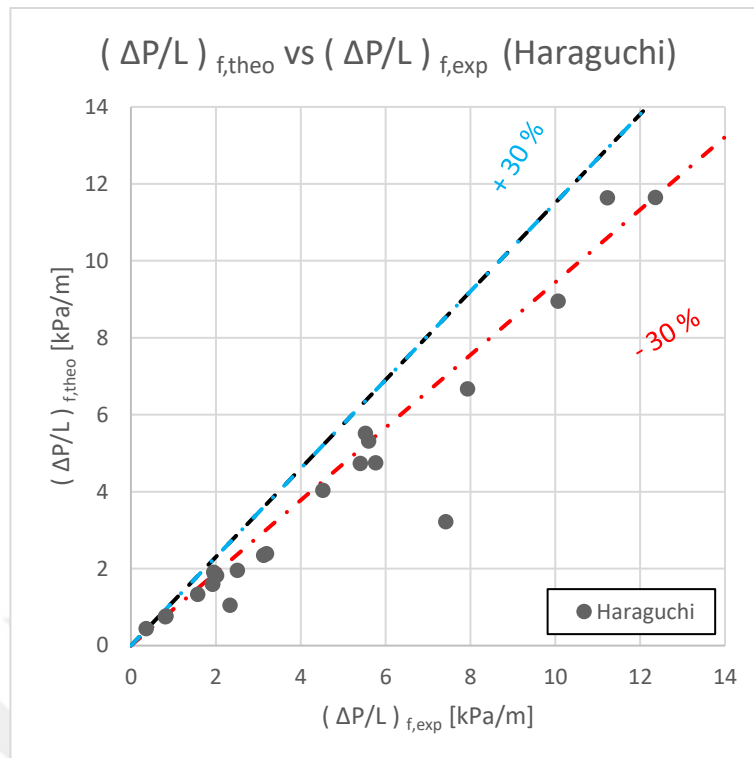


Figure 114: $(\Delta P/L)_{f,theo}$ vs $(\Delta P/L)_{f,exp}$ for Haraguchi

8 Conclusion

This research aimed to focus on the two-phase flow theory, heat transfer coefficient and frictional pressure drop per unit length in the process of flow boiling in horizontal tubes for fluid R22. Therefore, in the labs of the Energy Department of Politecnico di Milano, experimental data is gathered in order to acquainted with the purpose of the research. Moreover, the thesis concentrates on the behavior on the microfine. In this regard, smooth tube, Tube V, Tube W, Tube VA, and Tube HVA are taken into account.

Based on the behavior of fluids in terms of heat transfer coefficient and pressure drop, it can be concluded that two-phase vapor-liquid flow provides a linear increase with increasing mass flux and mean quality. The results indicate that after the frictional pressure drop per unit length is calculated, the behavior of both heat transfer coefficient and frictional pressure drop per unit length are considered as a function of mass flux and mean quality. The comparison between the predicted data and the experimental data is performed by using parity plots for that shows the distribution of the data on a 45° line together with $\pm 30\%$ deviation lines. Parity plots are made for both heat transfer coefficient and pressure drop per unit length. The results are shown in Table 28 and Table 29 as follows;

Tube Type	Kandlikar	Shah	Gungor & Winterton	Fang	Saitoh
Smooth	$X_{\pm 30\%}=47.06\%$	$X_{\pm 30\%}=0.0\%$	$X_{\pm 30\%}=58.82\%$	$X_{\pm 30\%}=0.0\%$	$X_{\pm 30\%}=23.53\%$
	MARD=31.32%	MARD=41.57%	MARD=27.95%	MARD=158.61%	MARD=35.87%
V (Parallel)	$X_{\pm 30\%}=28.57\%$	$X_{\pm 30\%}=0.0\%$	$X_{\pm 30\%}=0.0\%$	$X_{\pm 30\%}=78.57\%$	$X_{\pm 30\%}=0.0\%$
	MARD=34.58%	MARD=75.22%	MARD=67.02%	MARD=22.72%	MARD=68.57%
Tube V (Counter)	$X_{\pm 30\%}=41.67\%$	$X_{\pm 30\%}=0.0\%$	$X_{\pm 30\%}=0.0\%$	$X_{\pm 30\%}=66.67\%$	$X_{\pm 30\%}=0.0\%$
	MARD=32.68%	MARD=76.19%	MARD=68.08%	MARD=25.46%	MARD=69.96%
Tube W	$X_{\pm 30\%}=61.54\%$	$X_{\pm 30\%}=0.0\%$	$X_{\pm 30\%}=0.0\%$	$X_{\pm 30\%}=15.38\%$	$X_{\pm 30\%}=0.0\%$
	MARD=27.62%	MARD=66.97%	MARD=58.49%	MARD=55.68%	MARD=63.34%
Tube VA	$X_{\pm 30\%}=59.65\%$	$X_{\pm 30\%}=0.0\%$	$X_{\pm 30\%}=0.0\%$	$X_{\pm 30\%}=36.84\%$	$X_{\pm 30\%}=0.0\%$
	MARD=27.61%	MARD=69.22%	MARD=59.18%	MARD=37.36%	MARD=66.11%
Tube HVA	$X_{\pm 30\%}=34.78\%$	$X_{\pm 30\%}=0.0\%$	$X_{\pm 30\%}=0.0\%$	$X_{\pm 30\%}=73.91\%$	$X_{\pm 30\%}=0.0\%$
	MARD=38.92%	MARD=77.28%	MARD=68.02%	MARD=21.32%	MARD=69.65%

Table 28: Best Method for Heat Transfer Coefficient, h

As it can be seen from the Table 28, the best estimated correlations are highlighted for each tube types. Based on the results, Fang and Kandlikar methods yield the lowest values of mean absolute relative deviation for R22. In the case of R22, all correlations perform almost quite well except Shah and Saitoh. The large errors are noted for these two correlations. As noticed from Table 28, they never provided the best results. In this sense, another important observation was made that correlation of the Shah and Saitoh generally underestimate which is expected. Finally, as clearly seen in the Table 28, the remaining correlations such as

Kandlikar, Gungor & Winterton and Fang are mostly within the 30-percentage error range for almost all types of tubes.

Tube Type	Muller-Steinhagen and Heck	Gronnerud	Friedel	Chen	Tran	Haraguchi
Smooth	$X_{\pm 30\%}=58.82\%$	$X_{\pm 30\%}=-88.24\%$	$X_{\pm 30\%}=58.82\%$	$X_{\pm 30\%}=41.18\%$	$X_{\pm 30\%}=35.29\%$	$X_{\pm 30\%}=41.18\%$
	MARD=26.74%	MARD=17.63%	MARD=38.63%	MARD=55.59%	MARD=48.23%	MARD=47.83%
V (Parallel)	$X_{\pm 30\%}=42.86\%$	$X_{\pm 30\%}=50.00\%$	$X_{\pm 30\%}=50.00\%$	$X_{\pm 30\%}=85.71\%$	$X_{\pm 30\%}=71.43\%$	$X_{\pm 30\%}=85.71\%$
	MARD=35.22%	MARD=31.19%	MARD=28.05%	MARD=22.26%	MARD=39.29%	MARD=25.72%
Tube V (Counter)	$X_{\pm 30\%}=8.33\%$	$X_{\pm 30\%}=75.00\%$	$X_{\pm 30\%}=16.67\%$	$X_{\pm 30\%}=100.00\%$	$X_{\pm 30\%}=91.67\%$	$X_{\pm 30\%}=100.00\%$
	MARD=40.89%	MARD=24.81%	MARD=37.49%	MARD=7.70%	MARD=19.72%	MARD=5.14%
Tube W	$X_{\pm 30\%}=23.08\%$	$X_{\pm 30\%}=79.92\%$	$X_{\pm 30\%}=34.62\%$	$X_{\pm 30\%}=84.62\%$	$X_{\pm 30\%}=76.92\%$	$X_{\pm 30\%}=69.23\%$
	MARD=36.88%	MARD=21.76%	MARD=33.74%	MARD=18.64%	MARD=25.43%	MARD=22.05%
Tube VA	$X_{\pm 30\%}=24.56\%$	$X_{\pm 30\%}=63.16\%$	$X_{\pm 30\%}=49.12\%$	$X_{\pm 30\%}=85.96\%$	$X_{\pm 30\%}=80.70\%$	$X_{\pm 30\%}=91.23\%$
	MARD=34.82%	MARD=26.81%	MARD=29.83%	MARD=16.98%	MARD=19.49%	MARD=12.81%
Tube HVA	$X_{\pm 30\%}=13.04\%$	$X_{\pm 30\%}=34.78\%$	$X_{\pm 30\%}=17.39\%$	$X_{\pm 30\%}=86.96\%$	$X_{\pm 30\%}=56.52\%$	$X_{\pm 30\%}=91.30\%$
	MARD=46.31%	MARD=35.66%	MARD=42.21%	MARD=19.68%	MARD=26.55%	MARD=16.16%

Table 29: Best Method for Frictional Pressure Drop per Unit Length, ($\Delta P/L$)_f

The best predicted correlations are shown in the Table 29 for each tube types. According to results, best agreement between predicted and measured values was obtained with the correlation by Chen and Haraguchi. They have the lowest values of mean absolute relative deviation. In addition, the remaining correlations such as Friedel, Tran, Gronnerud and Muller-Steinhagen Heck are usually within the 30% error range for all types of tubes. However, average deviation of more than $\pm 30\%$ between predicted and measured values can still be accepted because they are not so much bigger than 30%. To allow quantitative assessment of the correlations given in the Table 29, all correlation were checked against the data set obtained for frictional pressure drop, ($\Delta P/L$)_f. As characteristic values for the performance of the respective correlations, the mean absolute relative deviation and 30% error deviation lines were used as well as the percentage of estimated values within 30% of the measured data.

The Shah diagram (see Figure 35) can be used in order to see four regimes of boiling which are pure convective boiling with surface fully wet, bubble suppression regime, nucleate boiling regime and convective boiling with partly dry surface. It has been demonstrated that the chart correlation is in reasonable agreement with a wide variety of experimental data. It may therefore be stated that the choice of Co , Bo , Fr_L and Ψ as correlating parameters has been adequately justified and that the chart can be recommended as a reliable tool for the prediction of boiling heat transfer coefficient in both horizontal and vertical tubes.

All these methods are proposed by using their own set of data. In other saying, they are mostly valid for a specific data set. To put it differently, the experimental data that they use is not abundant. This might be the one of the main reasons behind the large deviations. Another

equally significant factor in the comparison of the correlations is their ability to predict correct parametric trends among the important system variables. On the other hand, there may exist some human based errors such as setting pressure into a wrong value and perceiving the number on the indicator wrong.



9 Future Developments

In this thesis, the experimental data focused on the performance evaluation of the refrigerant fluid which is R22. The results related to the performances at constant saturation temperatures have been collected along with a final assessment of the fins beneficial effects on the heat exchanged. As final evaluation, the experimental data were compared with the predictions resulting from the available correlations for pressure drop and heat transfer coefficient. Generally, the pressure drop models provide similar outcomes, while, as concern the heat transfer coefficient, the models supply contrasting results.

The data distribution and the error values for both for heat transfer coefficient and frictional pressure drop per unit length observed for different types of tubes. Kandlikar, Shah, Fang, Saitoh, Gungor & Winterton models were used for heat transfer coefficient. On the other hand, Muller-Steinhagen and Heck, Gronnerud, Friedel, Chen, Tran and Haraguchi methods were used for frictional pressure drop per unit length.

In addition to this, for R22, the best performing correlation is developed by Kandlikar and fang in order to calculate heat transfer coefficient. However, for pressure drop calculation, the best performing correlation is developed by Gronnerud, Chen and Haraguchi. Thus, it will be possible to demonstrate the higher mixture performances both for the heat transfer coefficient and pressure drop, as already emerged from this initial study.

This thesis was also examined the effects of the mass flux (G) and the mean quality (x_m) on the pressure drop and the heat transfer coefficient. it was observed that the heat transfer coefficient trends to increase with increasing mass flux. In order to improve a general correlation, it is primary to have a large data base by using different fluids with an extensive range of operation conditions such as mass flux, heat flux, pressure, quality and tube diameter. To have a large data base can be an advantage in order to get good results in the experimental studies. Also, according to experimental studies for two-phase flow boiling ensured the real values of heat transfer coefficients for several fluids under specific ranges of operating conditions.

To sum up briefly, the calculation of pressure drops and heat transfer coefficients in two-phase channel are significant consideration and different microfine tubes have an important role in affecting of the results.

Further developments may consist in the analysis of the heat transfer characteristics of other microfine tubes in order to study the influence of a different number and geometry of the fins. Additionally, other refrigerants, maybe characterized by higher performances and lower global warming potential and new microfine tubes configurations can be investigated in the future.

10 Bibliography

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