UNIVERSITY OF OULU

FACULTY OF TECHNOLOGY

EFFECT OF SURFACE ROUGHNESS ON HEAT TRANSFER IN HEAT EXCHANGERS

Geethanchali Sivanantharaja

PROCESS ENGINEERING

Bachelor Thesis

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Tiivistelmä

Lämmönvaihdin on laite, joka siirtää lämpöä fluidista toiseen tai fluidin ja ympäristön välillä. Viimeisimpien vuosikymmenten aikana lämmönvaihtimien rooli on kasvanut lämmön talteenottoprosesseissa ja uusien energialähteiden käyttöönotossa. Lämmönvaihtimien pinnankarheudella, jolla tarkoitetaan seinämän pintakuvion korkeuden muutosta verrattuna tasaiseen pintaan, on merkittävä rooli lämmönvaihtimen tehokkuudessa. Pinnankarheuden vaikutusta lämmönsiirtoon onkin tarkasteltu useissa tutkimuksissa. Pinnankarheus voi olla osa lämmönvaihdinrakennetta tai johtua ei haluttujen materiaalien kerrostumisesta pinnalle. Tällöin puhutaan likaantumisesta, joka heikentää lämmönvaihtimen lämmönsiirtoa, lisää painehäviötä ja voi aiheuttaa korroosiota. Dimensiottomat korrelaatiot, kuten Nusseltin luku antavat tietoa pinnankarheuden aiheuttamasta vaikutuksen lämmönsiirtoon.

Tässä kandidaatintyössä on tarkasteltu kirjallisuudessa esitettyjä Nusseltin luvun korrelaatioita ja niiden soveltuvuutta eri pinnankarheuden muotoihin sekä tutkittu niiden soveltuvuutta todellisen lämmönvaihtimen tapauksessa. Tästä tutkimuksessa tarkastelluista korrelaatioista Nunnerin korrelaatio soveltui parhaiten likaantuneen lämmönvaihtimen lämmönsiirron tarkasteluun. Sainin ym. korrelaatio arvioitiin soveltuvan paremmin keinotekoisen pinnankarheuden kuin likaantuneen pinnan lämmönsiirron tarkasteluun.

Muita tietoja

ABSTRACT
FOR THESIS

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Degree Programme (Bachelor's Thesis, Master's Thesis)		Major Subject (Licentiate Thesis)	
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Abstract

The heat exchanger is a device that transfers heat from one fluid to another or between fluid and the environment. Over the last few decades, the role of heat exchangers has increased in the process of heat recovery and introduction of new energy sources. Surface roughness of heat exchanger wall plays a vital role in the efficiency of heat transfer. Therefore, significance of surface roughness is examined by many researchers applying different shapes of roughness. Roughness is the variation in the height of a surface. It could be either a part of the geometry or due to deposition of undesired materials (which decreases the thermal function of the heat exchanger, increases the pressure drop and could cause corrosion). Dimensionless heat transfer correlations such as Nusselt number provides a clear view about the effect of heat transfer by surface roughness.

This thesis combines different Nusselt correlations for distinct shapes of surface roughness and investigates the suitability of them on a test experiment by comparing the values gotten. From this investigation it was clear that the value of Nunner correlation delivers the most reasonable results for a fouled layer formed by means of crystallization. Also the Nusselt correlation by Saini et al. could be more suitable for artificial surface roughness than for a fouled surface.

Additional Information

PREFACE

In order to accomplish my Bachelor's Degree I am submitting the thesis on "THE EFFECT OF SURFACE ROUGHNESS ON HEAR TRANSFER IN HEAT EXCHANGERS". The objective of this work was to combine and compare the heat transfer correlations for different surface roughness shapes. This thesis was carried out during the period of November 2016 and November 2017. Regardless the limitation resource, every effort was made to achieve the objective of this thesis. The supervisor Tiina Pääkkönen assisted in exploring different resources and also provided pictures and test experiment values required for the work.

Oulu 27.10.2017 Geethanchali Sivanantharaja

CONTENT

UNITS

Latin Symbols

A	area [m^2]
A_c	area of absorber plate [m^2]
A_m	mean temperature of area [k]
A_T	throat area of the orifice [m^2]
A_w	transfer area/ total area of side wall of microchannel [m^2]
C_d	coefficient of discharge for the orifice meter (dimensionless)
C_f	friction coefficient (dimensionless)
c_p	specific heat capacity [$J\ kg^{-1}\ K^{-1}$]
D_h	relative roughness (dimensionless)
dT/dx	temperature gradient (dimensionless)
e	roughness height [m]
e/D	relative roughness height [m]
f_r	Friction factor (dimensionless)
h	heat transfer coefficient [$W\ m^{-2}K^{-1}$]
H	height [m]
k	thermal conductivity [$W\ m^{-1}\ K^{-1}$]
k_s	sand surface roughness (dimensionless)
L	length of the tube [m]
l	characteristic length [m]
l_x	distance from leading edge [m]
M	mass flow rate [m^3s^{-1}]
N	number of micro channels in the chip (dimensionless)
Nu	Nusselt number (dimensionless)
p/e	relative pitch (dimensionless)
Pr	Prandtl number (dimensionless)
Q	total heat removed [W]
q_x''	heat flux [Wm^{-2}]
q_x	heat rate [Wm^{-1}]

R	radius [m]
Ra	average roughness (dimensionless)
Re	Reynolds number (dimensionless)
R_f	fouling resistance
Rp	maximum peak height [m]
T	temperature [K]
U_f	over all heat coefficient for fouled surface [$Wm^{-2}K^{-1}$)]
u_∞	velocity at infinity [ms^{-1}]
v	flow rate [m^3s^{-1}]
V^*	friction velocity [ms^{-1}]
W_b	bottom width [m]
W_t	top width [m]
x	deposit thickness [m]

Greek Letters

β	diameter ratio (dimensionless)
τ_s	shear stress [Pa]
ρ	density of fluid [kg m^{-3}]
$\frac{\partial u}{\partial y}$	velocity gradient (dimensionless)
μ	dynamic viscosity [m^2s^{-1}]
δ	boundary layer thickness [m]
Θ	tilt angle of the manometer
ε	effectiveness (dimensionless)

Sub and super scripts

a	average temperature of air
f	fluid temperature
in	temperature of fluid outlet
m	mean temperature
out	temperature of fluid inlet

p	average temperature of the absorbing plate
s	surface temperature

1 HEAT EXCHANGERS

1.1 Introduction of heat exchangers

Heat exchangers are used to transfer thermal energy from one medium to another and used in most of the industrial process to move energy from one place to another. Therefore, heat exchanger increases the energy efficiency of the process. There are numerous types of heat exchangers, such as shell and tube heat exchangers, plate/fin heat exchangers etc. important factors affecting the heat and mass transfer are temperature of fluid, fluid flow rate, conductivity of the material use in exchangers and heat transfer area. In addition, surface roughness or fouling effect the heat and mass transfer. (Shah 1981, p.10)

The most common classifications of heat exchangers are based on the nature of the heat exchange process, the physical state of the fluids, the heat exchanger's flow arrangement and the design/construction of the heat exchanger. The simplest form is where hot and cold fluid flows in same or opposite direction in a pipe. Figure 1(a) shows the parallel flow where both hot and cold fluid enter and leave at the same end. In counter flow, as shown in Fig 1(b), both fluids enter and leave in opposite directions.(Shah 1981, p.10)

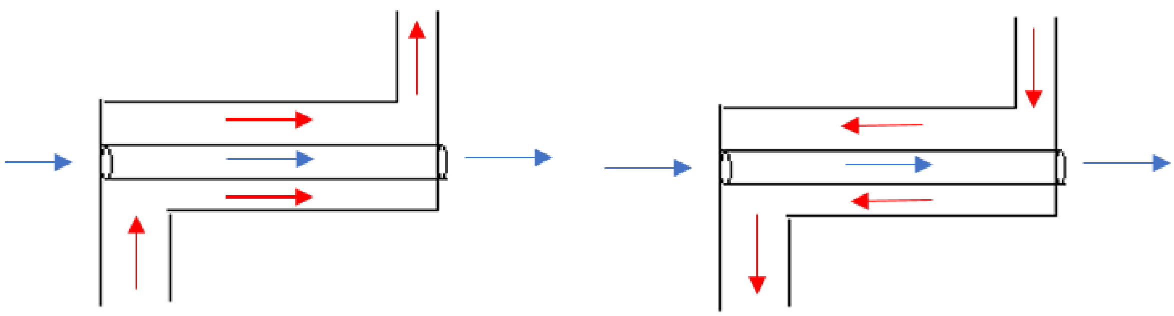


Figure 1. Parallel flow (a), and counter flow (b) of heat exchanger configuration.

Heat exchangers are broadly used in industries e.g. as components of air conditioning and cooling systems. Heat exchangers keep machinery, chemicals, water, gas, and other substances in their desired and safe temperature. Heat exchangers are also used to eliminate byproducts such as steam, heat exhausted from the system by transferring it to other valuable product. This way industries expand the energy efficiency and improve economical profitability. (Shah 1981, p.10).

Function of heat exchangers depends mainly on the purpose and arrangement of the system. Different models of heat exchangers are used in different flow arrangement. However, all heat exchangers function is (either direct or indirect) to provide warmer medium to cooler medium by exchanging heat. The basic classification of heat exchangers refers to, whether the fluids, which involve in heat exchanging are in direct contact with the each other or they are separated by a physical barrier such as tube walls.

Direct contact heat exchangers are where the fluids are not separated by any physical barriers. In this case convection is not required and fluids are let to interact directly. This method is extremely efficient, however all fluids cannot be let to interact directly because of e.g. safety purpose. Direct contact heat exchangers are suitable for same fluids with distinct temperature or if the fluid mixture is a desired or irrelevant part of the industrial process. (Shah 1981, p.12)

In indirect contact heat exchangers, the fluids are separated by a physical barrier, commonly in different pipes, and continued convection and conduction are used as the mode of thermal energy transfer. This method is suitable and recommendable for a fluid that can pollute the other fluid. (Shah 1981, p.12)

2 HEAT TRANSFER

Heat transfer can be defined as the transition of thermal energy due to a spatial temperature difference in a medium or within two or more media. The modes of heat transfer can be classified into three categories, which are convection, conduction and radiation. Conduction occurs through a solid medium and convention occurs within a moving fluid. (Bergman et al. 2002, p.1)

2.1 Conduction

Conduction occurs by the exchange of heat from one part of the substance to another parts OR one particle to another, which are in physical contact with each other. The heat transfer occurs by means of two mechanisms. (Bergman et al. 2002, p.3)

I. Good conductors, such as metals, have free electrons. The heat transfer occurs through these free electrons.

II. Those atoms/molecules with high energy, which associates with high temperature will pass that energy to its adjacent atoms or molecules by means of lattice energy.

Molecular collations transfer energy from high energy molecules to lower energy molecules.

Amount of conduction can be assessed by suitable rate equation. Rate equations evaluates the amount of energy that has been transferred per unit time. Fourier's Law is used to measure the amount of conduction. (Bergman et al. 2002, p.3)

$$q_x^{"} = -\mathrm{k}\frac{dT}{dx}, \tag{1}$$

where, $q_x^{"}$ is heat flux [Wm^{-2}] and

$\frac{dT}{dx}$ is temperature gradient [Km^{-1}].

Heat flux [W/m^2] expresses the amount of heat transferred per unit area in the x-direction. It is essential to indicate that the area is perpendicular to the direction of heat transfer and energy that is transferred is directly proportional to the gradient $\frac{dT}{dx}$. Thermal conductivity of the material is denoted by letter k [W/m k] in the equation. The minus sign in the equation represents that the convection travels from higher temperature to lower temperature.

In steady state condition, where the temperature is linearly distributed, the temperature gradient can be expressed as $\frac{dT}{dx} = \frac{T_{out}-T_{in}}{L}$, where T_{out}, T_{in} are the temperatures at the outlet and the inlet of the heat exchanger, respectively. (Bergman et al. 2002, p.3)

Therefore, at steady state the Fourier's Law for the system described in Fig. 2 is

$$q_x'' = -k\frac{T_2-T_1}{L} = -k\frac{\Delta T}{L}, \tag{2}$$

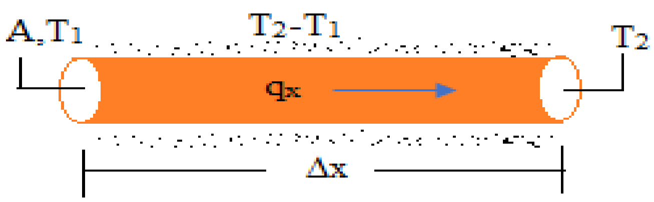


Figure 2. Heat conduction at steady state.

Heat rate q_x by conduction is directly proportional to the product of area A and temperature gradient $\frac{dT}{dx}$. Therefore, heat rate is given by the equation. (Bergman et al. 2002, p.68)

$$q_x = -kA\frac{dT}{dx}. \tag{3}$$

Therefore, heat flux can be expressed as

$$q_x^{"} = \frac{q_x}{A} = -k\frac{dT}{dx}\ , \tag{4}$$

and futher the heat rate as

$$\mathrm{q_x} = q_x^{"}\mathrm{A} = \ -kA\frac{dT}{dx}. \tag{5}$$

2.2 Convection

Convection occurs due to the random movement of molecules, which is defined as diffusion. However, energy is also transferred by bulk or macroscopic motion, which is called advection (Bergman et al. 2002, p. 6). In the other words, convection can be defined as the "study of heat transport process affected by fluid flow process" (Beyan 2013). Convection could happen only in fluids where molecules can move easily. Therefore, the rate of convection mainly depends on the flow rate. Convection can be classified into two mechanisms: forced and natural convection. Natural convection takes place by the difference in densities of fluids. As the temperature increases the density of the fluid decreases, which causes the fluid to rise to the liquid surface. When work is done to move the fluid through the volume, the mechanism is known as forced convection. (Bergman et al. 2002, p.8)

The energy transferred by convection is expressed as a function of heat transfer coefficient h, difference of temperature between surface, and fluid and the surface area A (Bergman et al. 2002, p.6):

$$q = hA\left(T_s - T_f\right), \tag{6}$$

where, T_s is the surface temperature [K] and
T_f is the fluid temperature [K].

2.3 Radiation

Radiation is defined as the thermal energy emitted from a matter where the temperature of the energy is not zero. In contrast to conduction and convection, medium is not required for radiation to occur. Instead, radiation occurs most efficiently in vacuum. The energy emitted by radiation is transmitted by electromagnetic waves (Bergman et al. 2002, p.3). Radiation of heat is complicated and has usually minor effect in heat exchanges, where the temperatures remain relatively low.

3 FLUID FLOW

3.1 Boundary layer

Consider a flow over a flat plate in Fig. 3. When the fluid particles interact with the surface of the plate, their velocity becomes relatively smaller. Therefore, the flow velocity can be considered almost as zero. The particles with negligible velocity then act to delay the motion of the particles present in adjoining fluid layer, which then act to delay the motion of particles in the next layer. This pattern repeats until the distance y =δ. After this, the effect becomes negligible. (Bergman et al. 2002, p.382)

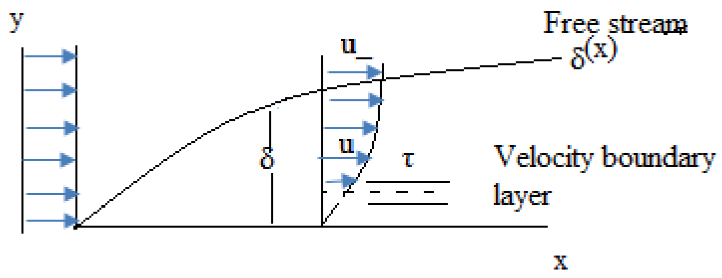


Figure 3. Velocity boundary layer of flat plate.

The cause of the retardation of fluid is associated with the quantity called shear stress τ, which acts on the plane, parallel to the fluid motion. Moving to the right, the x velocity component u increases until the free stream outside the boundary layer. The boundary layer thickness δ is typically the value of y when $u=0.99u_\infty$. (Bergman et al. 2002, p.382)

Velocity boundary layer will be present for any flow over any surface. Hence a surface friction will develop. For the external flow, the local friction coefficient is determined from

$$C_f = \frac{\tau_s}{\rho u_\infty^2/2}, \tag{7}$$

where, C_f is friction coefficient, τ_s is shear stress[Pa],
u_∞ is velocity at infinity[ms^{-1}] and
ρ is density of fluid [kg m^{-3}].

Friction coefficient is a dimensionless quantity from which the surface friction drag is calculated. For Newtonian fluid, the surface shear stress can be calculated from the velocity gradient at the surface with known fluid viscosity μ. (Bergman et al. 2002, p.382)

$$\tau_s = \mu \frac{\partial u}{\partial y}\Big|_{y=0,} \tag{8}$$

where, $\frac{\partial u}{\partial y}$ is velocity gradient and
μ is dynamic viscosity [m^2s^{-1}] .

3.2 Thermal and concentration boundary layers

Similar to velocity boundary layers, a thermal boundary layer is formed when there is a temperature difference between the fluid free stream and the surface temperature. Likewise, the concentration boundary layers develop by the changes in concentration decay. (Bergman et al. 2002, p.382)

3.3 Laminar and turbulent flow

In both lamina and turbulent flow, the fluid motion is defined by the x and y components of velocity. In the laminar segment, the fluid flow is well organized and streamlines could be identified clearly. (Bergman et al. 2002, p.390)

In turbulent flow regime, the flow is highly disordered and represents the three dimensional movement of large fluid particles. Figure 4 shows the laminar segment, transition segment and turbulent segment. In the transitional segment, the fluid flow changes with time. Both laminar and turbulent flows occur in transitional segment. (Bergman et al. 2002, p.390)

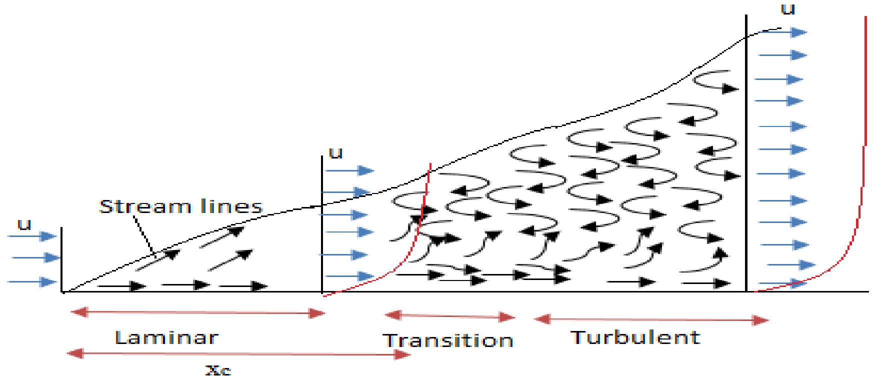


Figure 4. Boundary layer segments on a flat plate.

Figure 4 also shows the free stream velocity of laminar and turbulent flow. The velocity in laminar boundary layer increases linearly but there is a steep increase in the velocity in turbulent boundary layer. Fluid flow transfer from laminar to turbulent by the "triggering mechanisms such as the interaction is unsteady flow structures that develop naturally within the fluid or small disturbance that exist within many typical boundary layers". Disturbance may occur due to the fluctuation in free stream or by surface roughness or even by minute surface vibration. (Bergman et al. 2002, p.391)

3.4 Reynolds number

The Reynolds number is the most common dimensionless factor that determines the weather a flow is laminar or turbulent. The Reynolds number is a ratio of inertial forces to viscous forces. (Munson et al. 2013)

$$Re = \frac{\rho V l}{\mu}, \tag{9}$$

where, V is the velocity of fluid flow [ms^{-1}] and l is the characteristic length [m].

It is defined that for a tube flow, the Re lower than 2300 falls in the laminar flow and Re over10000 falls in turbulent flow. Between 2300 and 10000 there is a transition region in which the flow is not any more totally laminar, but not fully turbulent either. The origination of turbulence rest on whether the triggering mechanisms are increased or decreased in the direction of fluid flow, which depend Reynolds number, defined as

$$Re = \frac{\rho u_\infty l_x}{\mu}, \tag{10}$$

where, l_x is the distance from the leading edge [m]. (Bergman et al. 2002, p.390)

For smaller Reynolds number the inertia forces are negligible relative to the viscous flow. Therefore, the disturbance raised is then dissipated and the flow remains laminar. When the Reynolds number is high the inertia forces cause the increase in triggering mechanisms, which turns the transitional flow to turbulent flow. (Bergman et al. 2002, p.390)

Reynolds number is the key parameter to determine the characteristic of flow. Assuming that the transitional flow at point x_c as shown in Fig. 4, the location is calculated by the critical Reynolds number, $Re_{x,c}$. In case of flat plate, the value on $Re_{x,\ c}$ varies between 10^5 to $3*10^6$. The exact value depends on the surface roughness and the level of turbulence. (Bergman et al. 2002, p.390)

3.5 Nusselt number

One of the most vital quantities in heat and mass transfer is the heat transfer coefficient h, and its non-dimensional correlation is known as Nusselt number (Nu) (Herwig 2016). Nusselt number is defined as:

$$Nu = \frac{q_x L}{\mathrm{k}\Delta T}. \tag{11}$$

Since the heat transfer coefficient h is,

$$h = \frac{q_x}{\Delta T}, \tag{12}$$

it follows that

$$Nu = \frac{q_x L}{\mathrm{k}\Delta T} = h\frac{L}{\mathrm{k}}. \tag{13}$$

Nusselt number of flow, which aids to measure the ratio of convective heat transfer to the conduction heat transfer on the boundary. As the friction coefficient is there to velocity boundary layer, Nusselt number is there for thermal boundary layer. (Bergman et al. 2002, p. 401).

In order to predict the Nusselt number for turbulent flow, the Gnielinski's correlation for smooth tubes can be used (Subramanian 2015). Gnielinski's correlation is defined as,

$$Nu = \frac{\left(\frac{f_r}{2}\right)(Re_D - 1000)Pr}{1 + 12.7(Pr^{\frac{2}{3}} - 1)\sqrt{f_r/2}}, \tag{14}$$

where, f_r is friction factor for smooth tube,
Re is Reynolds number and
Pr is Prandtl number.

The equation for Prandtl number is

$$Pr = \frac{C_p \rho}{k}, \tag{15}$$

where, c_p is specific heat capacity [J kg^{-1} K^{-1}].

4 SURFACE ROUGHNESS

This thesis focuses on the effect caused by the surface roughness on heat and mass transfer in heat exchangers. Surface roughness is defined as “the measurement of small scale variation in the height of a physical surface.” Roughness impact the thermal and mechanical performance and causes friction, wear, drag, fatigue and heat loss of heat exchangers. However, it also helps to trap lubricants and avoid bonding to each other. (Maisuria 2013) Figure 5 shows the major causes of surface roughness.

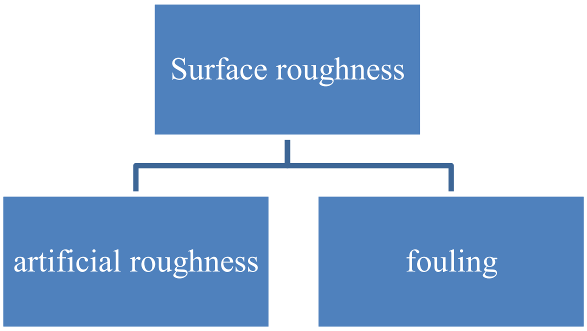


Figure 5. Classification of surface roughness on heat exchanger walls.

4.1 Artificial roughness

Heat transfer could be improved by artificial surface roughness. Artificial surface roughness could be created by fixing wires, ribs, and wire mesh and by providing different geometry. (Verma 2007)

Figure 6 shows an example of artificial surface roughness or texture of a heat exchanger wall. The artificial roughness is designed to be equally distributed diamond shape with equal roughness height.

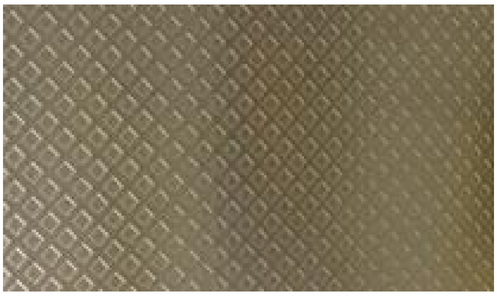


Figure 6. Artificial surface roughness.

Considering the roughness characterization, roughness parameters that affect the micro fluids were introduced by Kandlikar et al. (2005). In the experiment it was found that the different roughness profiles with equal values of R_a (average roughness), may have different effects of flow with variations in other profile characteristics. For instance, a rough surface with double pitch but same average roughness could have distinct pressure drop. In other word, regardless the same quantitative value for the average roughness R_a for different roughness profiles, they may have distinct effect on flows with variations in other profile characteristics. Roughness can be calculated in terms of maximum peak height R_p. (Kandlikar et al. 2005)

4.2 Roughness due to fouling

Surface roughness of heat exchangers may be also caused by fouling. One of the main problem, which affects the mass and heat transfer in heat exchanger, is the deposition of undesired product residing on the surface of exchanger which is known as fouling. Fouling of heat exchangers takes place e.g. by means of crystallization and scale formation, liquid solidification (liquid freezing), corrosion, chemical reaction and biological growth (Bott 1995, p. 2). Fouling takes place usually due to the temperature difference that occurs in side heat exchangers. These types of fouling leads to a drastic increase in thermal resistance, which then decrease the efficiency of heat exchangers significantly. (Albert et al. 2010)

Figure 7 demonstrates the idealized asymptotic graph of the growth of contaminated layer on a heat exchanger surface. In section A the deposition is initiated. The time taken for this depends on several factors such as temperature, concentration of fluid, purity of fluid etc. Section B shows the steady growth of the deposit layer. In this stage there is a competition between deposition and removal processes. The rate at which deposition of contaminants takes place gradually decreases while removal of contaminants grows. At the final stage C the removal and deposition of contaminants become equal so that the asymptote is reached. At this stage the thickness of contaminant layer stays constant. (Bott 1995, p. 4)

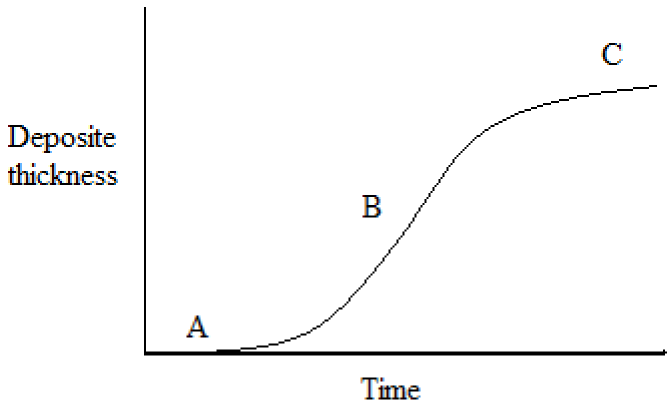


Figure 7. Growth of contaminated layer on a heat exchanger surface.

Figure 8 represents the effect of undesired materials accumulation on the temperature distribution. T_1 and T_6 indicate the temperature of hot and cold bulk fluids respectively. Since the heat is transferred physically, when the flow is turbulent these temperatures spread approximately to the boundary layer. The area in-between foulant layer and fluid (boundary layer) causes lower heat flow in the system. This is because of the relatively lower thermal conductivity of the accumulated layer than the metal walls. Therefore, the temperature difference within the fouled layers is higher than the metal walls. (Bott 1995, p. 7)

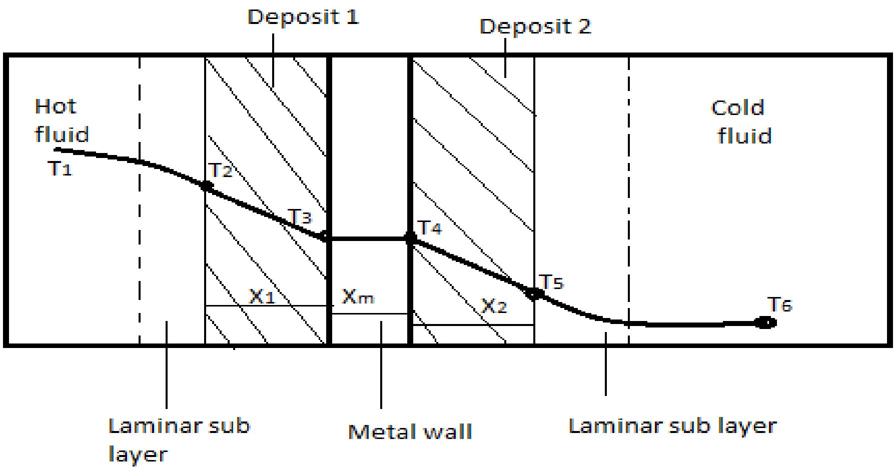


Figure 8. Temperature distribution across the fouled layers.

The quantitative significance of surface roughness relies on the factors such as, size, shape, orientation and distribution of roughness elements. Enlargement of the surface area relative to smooth wall and increase in near wall turbulence by surface roughness, may be the factors that enhances the heat transfer. (Albert et al. 2010)

Fouling of the heat exchanger surface by means of crystallization generally faces three different stages such as induction period, roughness controlled phase and finally crystal growth phase. (Albert et al. 2010)

I. Induction phase is where the mechanism starts with nucleation on the surface caused by supersaturation. The time required for this stage to complete depends on the salt concentration, energetic characteristics and topography of heat surface.
II. In the roughness controlled phase the heat transfer coefficient increases due to the increase in turbulent convective heat transfer, that formed by the initial perceptible scale, which dominates the thermal resistance effect of the scale.
III. Crystal growth keeps increasing in the final stage with time. The fouled layer become thicker and thicker as well as the overall heat transfer coefficient decreases. (Albert et al. 2010)

Figure 9 shows an example of a fouled surface, in which the fouling is caused by calcium carbonate ($CaCO_3$) deposition, which has formed by crystallization on the heat transfer surface. The craters or holes in the deposition layer are formed due to nucleation of dissolved air, which takes place due to solubility difference of air between the bulk fluid and the heated surface.

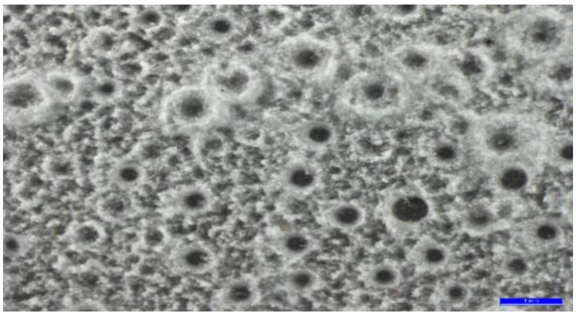


Figure 9. Example of fouled layer by crystallization.

4.3 Reynolds number for surface roughness

Surface roughness in a heat exchanger with a turbulent boundary layer increases the value of the friction factor and the local heat transfer coefficient. For flat surface, the relative roughness parameter is defined as e/ δ where, e is the roughness height and δ is the boundary layer thickness. Relative roughness parameter of pipe is e/R where R is the radius of pipe. For a flat surface, the relative roughness parameter decreases along the surface. The relative level of the roughness Reynolds number, Re_k, presenter in Eq. 16, determines the relative effect of the roughness. (Abuaf et al. 1997)

$$Re_k = \frac{V^* k_s}{\mu}, \tag{16}$$

where, V^* is friction velocity which is equals to $\sqrt{\tau_0/\rho}$ and
k_s is equivalent to sand surface roughness.

If Re_k is less than 5 the flow is defined to be smooth. For Re_k greater than 70, the flow is rough and in-between 5 and 70 flow is in transitional stage. (Abuaf et al. 1997)

5 SURFACE ROUGHNESS AND HEAT TRANSFER CORRELATIONS

5.1 Heat transfer in micro channels with different surface condition

When the laminar convective heat transfer and pressure drop of water, in distinct trapezoidal silicon micro channels were investigated, it revealed that both Nusselt number and apparent friction constant depend on the geometric parameters. Both correlations increase with the surface roughness and surface hydraulic properties. Hydrophobic and hydrophilic characters of a surface could be altered by changing the thickness of the oxide layer on a silicon surface. (Wu et al. 2003)

The experiment conducted by Wu et al. (2003) concentrated on the study of convective heat transfer in silicon micro channels with different surface condition. They chose to use trapezoidal microchannel because of its flexibility to use it in various microsystems. In this experiment the surface roughness of the microchannel was determined by the atomic force microscope. Wu et al. (2003) used 13 different micro channels and were able to derive the heat transfer correlation for the deionized water flowing through the micro channels. The cross-sectional area of trapezoidal microchannel used in the experiment by Wu et al. (2003) is given in Fig. 10.

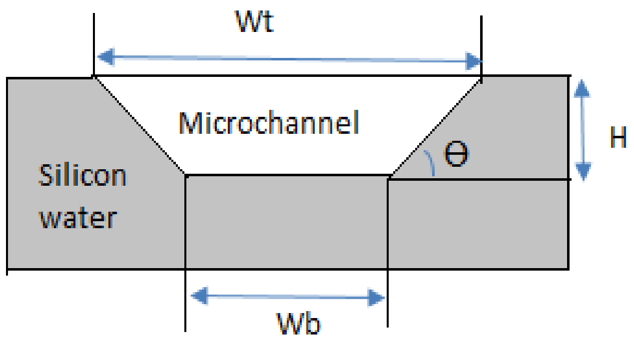


Figure 10: Cross-sectional profile of the micro channels.

In Fig. 10, the top width, bottom width and depth is defined as W_t, W_b and H, respectively. The special triangular channel is with width W_b=0 or rectangular channel with ϴ=90°. The total heat removed by the deionized water Q, is determined by the heat balance equation:

$$Q = Mc_p(T_{out} - T_{in}), \tag{17}$$

where, M is mass flow rate [m^3s^{-1}],
c_p is specific heat capacity[J kg^{-1} K^{-1}],
T_{out} is bulk temperature of outlet[k] and
T_{in} is bulk temperature of inlet[k].

The mean temperature T_m, within the wall and water is calculated from,

$$\Delta T_m = \frac{1}{5}(T_1 + T_2 + \cdots T_5) - \frac{1}{2}((T_{out} - T_{in}), \tag{18}$$

where, T_1, $T_{2...}T_5$ are the temperature of five walls.

Heat transfer coefficient h, was defined as:

$$h = \frac{Q}{NA_w\Delta T_m}, \tag{19}$$

where, N is number of micro channels in the chip and
A_w is transfer area/ total area of side wall of microchannel [m^2].

From the heat transfer coefficient, the Nusselt number is defined as:

$$Nu = \frac{hD_h}{\mathrm{k}}, \tag{20}$$

where, D_h is relative roughness.

Substituting Eqs. (15), (16) and (17) on the Eq. (18), the Nusselt number can be defined as;

$$Nu = \frac{Mc_pD_h(T_{out}-T_{in})}{NkA_w\Delta T_m}. \tag{21}$$

Even though it is believed that the surface roughness does not disturb the laminar flow in macro channels, as the size of channels decline in terms of few microns, the effect of surface roughness becomes vital.

Figure 11 presents the effect of the surface roughness on Nusselt number in the experiment of Wu et al. (2003). Channels 1 and 2 are trapezoidal and have same geometric parameters but the relatively distinct roughness, 5.87×10^{-3} and 3.26×10^{-5}, respectively. Also channels 3 and 4 are triangular and have the same geometric parameters but the roughness differs (1.09×10^{-2} and 3.62×10^{-5} for channels 3 and 4, respectively). For the same Reynolds number, the Nusselt number and apparent friction factor for the trapezoidal cross-sectional channel 2 is significantly larger than channel 1, although the surface roughness for channel 2 is lower than channel 1. Similarly, Wu et al. (2003) showed that there was an increase in Nusselt number and apparent friction constant for channels 3 and 4. Also, the correlations increase faster for more roughened surface than less roughened surface.

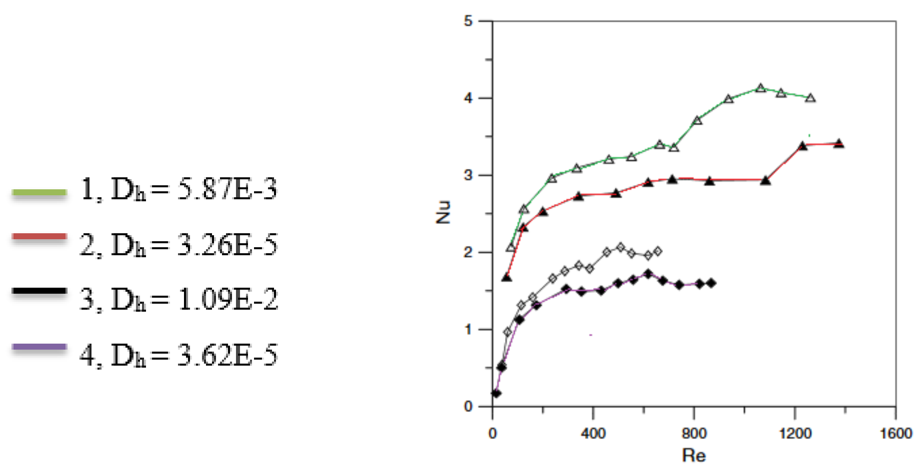


Figure 11. Effect of surface roughness on Nusselt number (modified from Wu et al. 2003).

5.2 Dimple shaped artificial roughness

Saini et al. (2007) investigated the impact on the application of dimple shaped roughness in solar air heater ducts. Dimple shaped roughness does not require complex manufacturing process and the structure reduces the weight of the plate. Figure 12 demonstrates the dimpled shape plate and its profile view. (Saini et al. 2007)

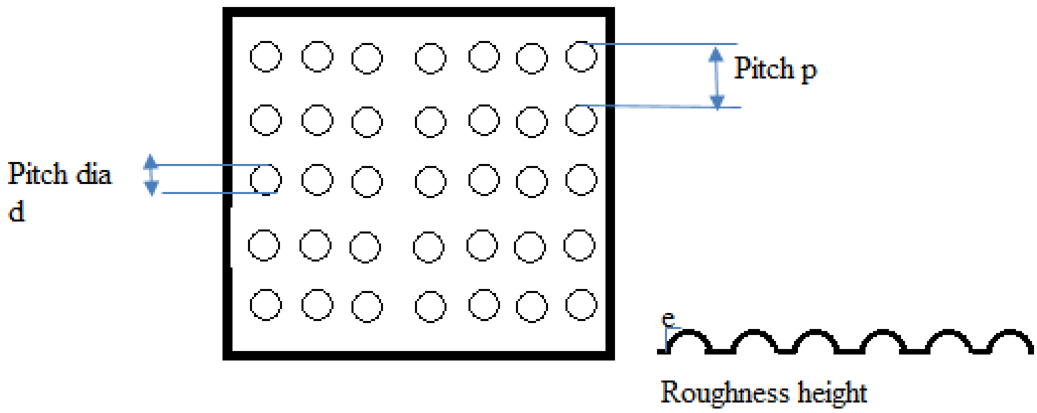


Figure 12. Dimple shaped plate (modified from Saini et al. 2007).

From the experiment, the temperatures of the absorber plate, air at inlet and outlet were measured. The mass flow rate is defined as,

$$M = C_d A_T \left[\frac{2\rho\Delta P sin\theta}{1-\beta^4}\right]^{1/2}, \tag{22}$$

Where, C_d is coefficient of discharge for the orifice meter,

A_T is throat area of the orifice,

Θ is tilt angle of the manometer and

β is diameter ratio. (Saini et al. 2007)

Heat transfer coefficient is defined as

$$h = \frac{MC_p(T_{out}-T_{in})}{A_c((T_p-T_a)}, \tag{23}$$

Where, A_c is area of the absorber plate [m^2],

T_p is average temperature of the absorbing plate [K] and

T_a is average temperature of air [K].

The Nusselt number correlation is derived by substituting the values in the Eq. (24)

$$Nu = \frac{hD_h}{\mathrm{k}}, \tag{24}$$

$$Nu = \frac{C_p M \mathrm{k}(T_{out} - T_{in})}{A_c((T_p - T_a)}. \tag{25}$$

Since the Nusselt number and friction factor depends on the parameters such as Reynolds number, roughness dimension of relative pitch (p/e) and relative roughness height (e/D), the functional relation could be represented as, (Saini et al. 2007)

$$Nu = f_n\left(Re, \frac{p}{e}, \frac{e}{D_h}\right), \tag{26}$$

$$f_r = f_n\left(Re, \frac{p}{e}, \frac{e}{D_h}\right). \tag{27}$$

The graph of Nusselt number versus the Reynolds number is shown in Fig. 13

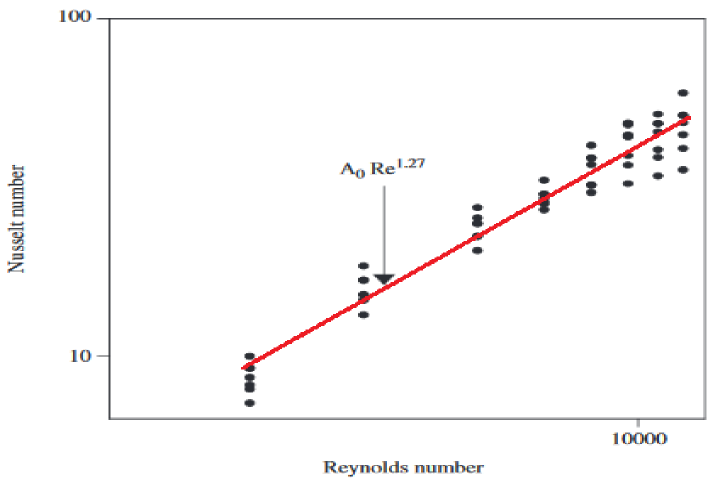


Figure 13. Nusselt number vs Reynolds number (modified from Saini et al. 2007).

The best fit for the data gathered is represented by the linear equation

$$Nu = A_0(Re)^{1.27}, \tag{28}$$

Where A_o is the coefficient that will be a function of other parameters that influences the experiment. Similarly, the graph A_o versus (p/e) is drawn and the regression analysis for the best fit delivers the quadratic curve, (Saini et al. 2007)

$$\log\left(\frac{Nu}{Re^{1.27}}\right) = logB_o + B_1\left(\log\left(\frac{p}{e}\right)\right) + B_2\left(\log\left(\frac{p}{e}\right)^2\right), \tag{29}$$

By further arrangement, B_o gets the form of

$$\frac{Nu}{Re^{1.27}} = \left[\frac{\left(\frac{Nu^{1.27}}{Re}\right)}{\left(\frac{p}{e}\right)^{3.15}[\exp(-2.12)\,(\log\left(\frac{p}{e}\right))^2}\right]. \tag{30}$$

Final step is to draw B_o versus (e/D), which is a second quadratic curve.

$$\left(\frac{Nu}{Re^{1.27}}\left(\frac{p}{e}\right)^{3.15}\right)\left[\exp(-2.12)(log\frac{p}{e}))^2\right] =$$
$$C_0\left(\frac{e}{D_h}\right)^{0.033}\left[\exp(-1.30)\left(\log\left(\frac{e}{D_h}\right)\right)^2\right]. \quad 31)$$

From Eqs. (29),(30) and (31) the values on A_o, B_o and C_o can be calculated to be;

$$Ao = 8.2 \times 10^{-4},$$

$$Bo = 3.1 \times 10^{-4},$$

$$Co = 1.98 \times 10^{-3}.$$

Values of theses coefficient result the correlation of Nusselt number as

$$Nu = 5.2 \times 10^{-4} Re^{1.27}\left(\frac{p}{e}\right)^{3.15} \times [\exp(-2.12(\log\left(\frac{p}{e}\right))^2]\left(\frac{e}{D_h}\right)^{0.033} \times$$
$$\left[\exp(-1.30\left(\log\left(\frac{e}{D_h}\right)\right)^2\right]. \tag{32}$$

Similarly, the equation for the friction factor is defined as

$$f_r = 0.642Re^{-0.456}\left[\exp(0.054)\left(\log\left(\frac{p}{e}\right)\right)^2\right] \times$$
$$(\frac{e}{D_h})^{-0.0214}[\exp(0.840)\left(\log\left(\frac{e}{D_h}\right)\right)^2. \qquad (33)$$

From this experiment, Saini et al. (2007) concluded that the transfer of heat can be improved significantly by providing dimple shaped roughness on the absorber plate of the solar heater duct. Saini et al. (2007) also defined that the maximum value for heat correlation such as Nusselt number corresponds to relative roughness height (e/D) which is in this case 0.0379 and relative pitch (p/e) of 10. They were able to figure out the minimum value of Nusselt number corresponding to e/D =0.0289 and p/e=10. Hence the roughness parameters of geometry can be chosen by taking net heat gain and power required to propel air through the duct in to account. (Saini et al. 2007)

5.3 Surface roughness by fouling

An experiment was conducted by Albert et al. (2011), which produces a fouling layer by crystallization of $CaSO_4$. From the fouled layer created, they were able to analyze the constriction effects caused by fouling, using different correlations, which eventually gave different heat transfer coefficient. They were also able to obtain a graphical comparison between asymptotic fouling resistance and corresponding fouling thickness based on constant heat transfer coefficient and heat transfer coefficients taking into account the roughness and constriction effects by, Nunner (1956), Burck (1969), Hughmark (1975) and Ceylan and Kelnaliyev (2003) correlations. Table 1 shows the equations for these correlations with respective shapes of their surface roughness (Albert et al. 2010). Eq. (34) was used to calculate the fouling resistance in the experiment

$$R_f(t) = \frac{1}{U_f} - \frac{A}{(h_{i,0}\varepsilon_{Nu,f_r}\varepsilon_{Nu,\omega})} - \left(\frac{xA}{f_r A_m} + \frac{A}{h_a A_a}\right), \qquad (34)$$

where, R_f is the fouling resistance,

U_f is the overall coefficient for fouled layer,

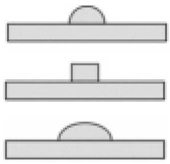
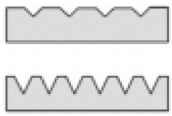
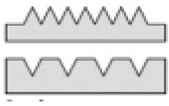
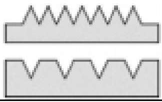
A_m is the mean temperature of area,

ε is the effectiveness,

x is the deposit thickness and

f_r is the friction factor.

Table 1. Heat transfer correlations with respective surface roughness.

Author	Roughness shape	Nusselt correlation
Nunner (1956)		$Nu = \frac{\left(\frac{f}{8}\right)RePr}{1+1.5Re^{-1/8}Pr^{-1/6}(\Pr(\frac{f}{fo}-1)}$ (35)
Burck (1969)		$Nu = Nu_0\left(log\frac{Pr^{0.33}}{k^{0.243}}0.32\times10^{-3}k^{+}logPr + 1.25\right)\frac{f}{f_0}$ (36)
Hughmark (1975)		$Nu = \sqrt{\frac{f}{8}}Re[\frac{1}{0.0303+0.0615Pr^{\frac{1}{2}}} + \frac{1}{0.625+0.062Pr^{\frac{1}{3}}} + \frac{1}{\frac{7.16}{\sqrt{\left(\frac{f}{8}\right)Re}}+2\sqrt{\left(\frac{f}{8}\right)}Pr}]^{-1}$ (37)
Ceylan and Kelnaliyev (2003)		$Nu = 1.15Nu_0Pr^{\frac{1}{7}}(1-0.106K^{+\left(\frac{1}{4}\right)}\frac{f}{f_o}$ (38)

After running the experiment for approximately 250h the fouling reaches its asymptotic resistance. From this the final deposit thickness x_f can be calculated. From the Fig 15. The average thickness of the fouling layer, considering the roughness and constriction effects, can be seen. As a conclusion the Nunner's (1965) equation gives 65% larger deposition thickness when roughness effect was considered. Hughmark delivers 144% increase in the thickness whereas Burk (1969) and Ceylan and Kelnaliyev (2003) gives the highest value resulting 16% lower total thickness than the measured value. Therefore, taking into account the roughness effect in the calculation of the total thickness and the asymptotic fouling resistance is crucial in order to prevent under estimation of them. (Albert et al. 2011)

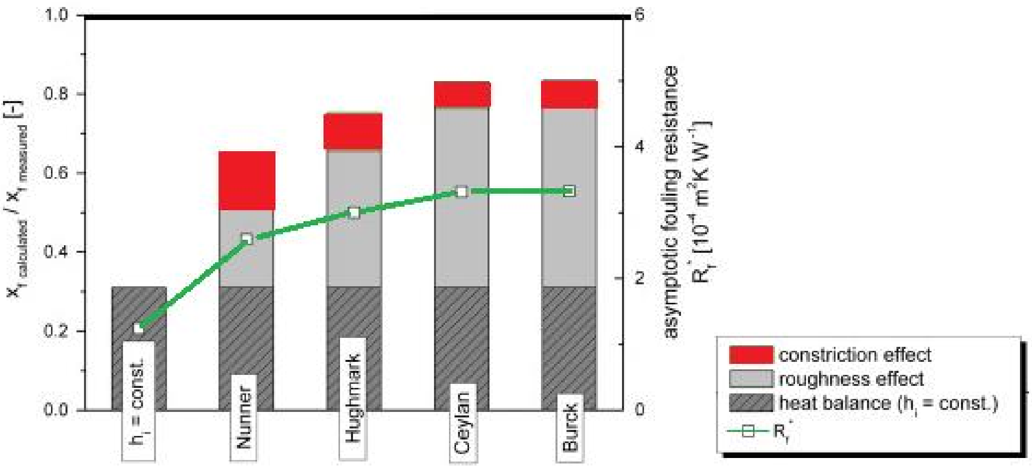


Figure 15. Asymptotic fouling resistance and corresponding deposit thickness based of different correlations (modified from Albert et al. 2011).

5.4 Comparison of different Nusselt correlation for a fouled surface by crystallization

Experiments for the crystallization fouling of $CaCO_3$ on a flat plate heat exchanger were conducted by Pääkkönen et al. (2015). Table 2 shows the data collected from these experiments.

Table 2. Parameters and their corresponding values.

Parameter	Unit	Value
D_h	[m]	0.03
q	[W/m^2]	59200
k	[W/mK]	0.66
T_s	[^{o}C]	30
T_b	[^{o}C]	67-82

Cp		4200
μ	[kg/(ms)]	$8.3 \cdot 10^{-4}$
ρ	[kg/m^3]	995.7
f_rough		0.0255-0.0240
e/D_h		0.001667
p/e		0.4

Experiment data (in Table 2) is used to calculate the Nusselt number by the correlations presented in Eqs. (13), (30), (33) and (35). From the data in Table 2 values of Re_D were calculated utilizing Eq. (9). The values of Pr are calculated from the Eq. (12). Friction factor for the rough surface can be gained from the Moody diagram (Munson 2013, p. 430).

Quantitative values of the obtained Nusselt numbers are presented in Fig. 16 as a function of Reynolds number. Nunner´s correlation seems to give the most reasonable result, since it gives higher Nu than the correlation for the smooth surface (Gnielinski), as should be since the fouling layer is not smooth. Hughmark instead gives quite low values. It might be that the roughness shape of Nunner's corresponds best with quite thin and porous fouling layers of these experiments. Therefore, Nunner's correlation seems to be the most appropriate to use for fouling layers from the correlations compared here. The correlation of Saini et al (2007) provides very low Nu. This might be caused by the fact, that in crystallization fouling $p<e$ ($p/e \sim 0.4$) meaning that relatively long crystals are located near to each other's, whereas in Saini et al. (2007) had p/e value as 10. For higher p/e values the correlation Saini et al. (2007) would provide acceptable Nu values. The Nusselt correlation by Saini et al. (2007) would be more appropriate for artificial roughness than fouled surface.

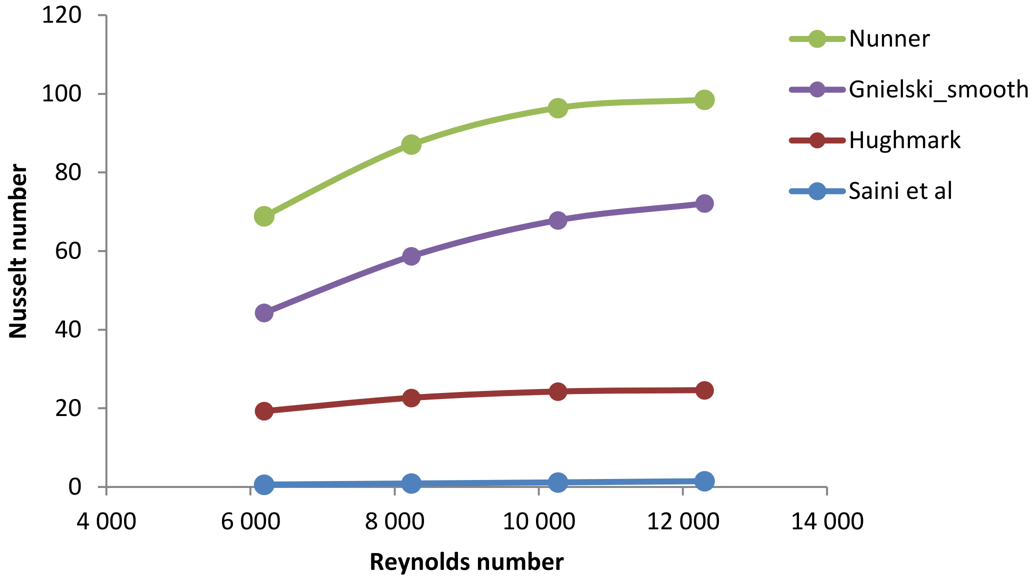


Figure 16. Nusselt correlation as a function of Reynolds number

6 SUMMARY

The beginning of this thesis introduces the function of heat exchangers and their uses. Then it concentrated on the modes of heat transfer and their mathematical presentation. Following that comes the concept of fluid flow which is explained in detail in terms of boundary layer, thermal and concentration boundary layer, laminar and turbulent flow. Afterwards the heat transfer correlations required for this work are presented.

Following the theory necessary for the thesis the concept of surface roughness is introduces. The reason for surface roughness and the effect of surface roughness in heat transfer is discussed in detail in this section. After this the Nusselt correlations for different surface roughness are presented from various resources.

Finally the Nusselt number vs. Reynolds number graph is derived for a test experiment which was conducted by Pääkkönen et al. (2015), utilizing the correlations introduced in the previous section. The investigation on the test experiment revealed that the Nunner's correlation is more suitable for a fouled surface and Saini et al. (2007) correlations is suitable for artificial surface roughness.

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