

Bachelor's Thesis
Information Technology
Internet Technology
2016

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GASKETED PLATE TYPE HEAT EXCHANGER DESIGN SOFTWARE



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The purpose of this thesis was to make the design of gasketed plate type heat exchangers easier, simple, and accurate by reducing human error. Properly designed heat exchangers can provide more benefits and better safety in wide range of applications. Since the design of heat exchangers is so complicated and involves several steps, computer-aided design has come to be widely used. In this design, along with the knowledge of heat exchangers, the Visual Studio 2013 Professional and the programming language (C#) have been used to complete the software design. The software calculates important design parameters based on the overall heat transfer requirement or the pressure drop requirement. It also compares the design for the percentage over the surface area which will help to know how large or small the physical size of the exchanger is. The manufacturing and maintenance cost can also be predicted based on the size data and other design selections. The software has been tested using real data and results show that it generates accurate data throughout all calculation steps. This will help design engineers to attain a more effective and efficient design.

KEYWORDS:

Heat transfer, heat exchanger, heat exchanger design, heat exchanger design software, accuracy, simplicity

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LIST OF ABBREVIATIONS (OR) SYMBOLS

A_t	total effective plate heat transfer area, m^2
A_1	single- plate effective area, m^2
A_{1p}	single-plate projected area, m^2
b	mean mass channel gap
C_p	specific heat of fluid, (J/kgK)
CF	cleanliness factor
C_h	constant
C_s	safety factor
D_e	channel equipment diameter
D_p	port diameter, m
f	friction factor
K_p	constant
k	thermal conductivity, (W/mK)
G_c	mass velocity through a channel, $kg/m^2.s$
G_p	heat transfer coefficient, W/mK
L_e	length of compressed plate pack, m
L_h	horizontal port distance, m
L_p	projected plate length, m
L_v	vertical port distance (flow length in one pass), m

L_w	plate width inside gasket
L_{eff}	effective flow length between inlet and outlet port ($\approx L_v$), m
\dot{m}	mass flow rate, kg/s
P	plate Pitch
Pr	prandtl number, $C_p\mu/k$
P_w	wetted perimeter
Q_c	heat load under clean conditions, W
Q_f	heat load under fouled conditions, W
Q_r	heat transfer required, W
R_{fc}	fouling factor for cold fluid, $m^2 \cdot K/W$
R_{fh}	fouling factor for hot fluid, $m^2 \cdot K/W$
Re	Reynolds number, $(G_c D_e / \mu, \text{ or } G_p D_p / \mu)$
t	plate thickness, m
TEMA	The Tubular Exchanger Manufacturers' Association
U_c	clean overall heat transfer coefficient, $W/m^2 \cdot k$
U_f	fouled (service) overall heat transfer coefficient, $W/m^2 \cdot k$
β	chevron angle, degree
ΔP_c	channel flow pressure drop, kPa
ΔP_p	port pressure drop, kPa
ΔT_m	mean temperature difference, °C, °K
ρ	fluid density, kg/m^3

μ dynamic velocity at average inlet temperature, N.s/m² or Pa.s

1 INTRODUCTION TO HEAT EXCHANGERS

A heat exchanger is a device in which heat is transferred from a fluid at a high temperature to a fluid at a low temperature. (Cengel 1997, 701) The main objective of this kind of heat transfer is to control the temperature of one of the fluids momentarily or for a given period of time for some technological purpose. Heat transfer from one fluid to another can be accomplished either by mixing the fluids directly or by passing the fluids through a partition between them. Almost all kinds of heat exchangers require a partition because of the need to keep one fluid separated from the other. The heat transfer between the two fluids takes the form of convection on the fluid side and conduction through the partition walls. (Cengel 1997, p. 701) "The principles of heat transfers are involved in the design of many forms of industrial and commercial equipments" (Brown and Marcos 1958, pp. 1.4).

1.1 Types of Heat Exchangers

Heat exchangers according to the requirement of either fluid mixing or not can be classified as

- Regenerator heat exchangers: they are heat exchangers in which the hot and cold fluids flow alternatively through the same space.
- Open type heat exchangers: the physical mixing of the two streams is required in this kind of devices.
- Closed type heat exchangers: they are the one and the most common type of heat exchangers in which heat transfer occurring between fluids streams is attained without mixing or physical contact of fluids.

According to the flow of one fluid relative to the other fluid, heat exchangers can be classified into three types: (Cengel 1997, p. 703)

- Parallel flow heat exchangers: where the flow of both fluids is in the same direction.

- Counter flow heat exchangers: in which the flow of fluids is in opposite directions.
- Cross flow heat exchangers: in which both fluids flow at right angle to each other.

Heat exchangers can also be classified in three groups based on their geometry of construction. (Kakac and Liu 1998, pp. 6-22)

- Tubular heat exchangers: circular tubes are used to build the flow channels in these exchangers. The double pipe, the shell and tube type and the spiral tube type exchangers are included in this category.
- Plate type heat exchangers: the flow channels for these devices are made of thin plates. The Gasketed plate, Spiral plate, and Lamella type heat exchangers are part of this group.
- Extended surface heat exchangers: are devices having fins or appendages on the primary heat transfer surface (tubular or plate). Plate fins and Tube fins are the most common types of Extended Surface Heat exchangers. Fins are mainly used in gas to gas or gas to liquid exchangers to increase the heat transfer area on the gas side or to meet the need of having compact heat exchanger.

1.2 Heat Exchanger Design Considerations

During the process of selection and purchasing of heat exchangers, many process and power industries mainly base their decision on the specification and the cost of the device. However, when the exchanger is required for a more specialized application, a particular design is frequently used. In such cases, a special design of heat exchangers could be ordered. A heat exchanger for a certain duty can be also selected by comparing their efficiency on the basis of their effectiveness or by using counter flow exchangers as a basis for comparison. (Janna 2000, p. 502)

The heat transfer requirements, the cost, the physical size and the pressure drop requirements need to be considered during the process of designing a

heat exchanger. For example, the designer can meet the heat transfer requirements by forcing the fluids through the tubes at higher velocities. This will increase the overall heat transfer coefficient but it will result in a large pressure drop through the exchanger and correspondingly a large pumping cost. If the surface area of the exchanger is increased, the overall heat transfer coefficient will also increase and hence the pressure drop needs to be solved; however, there also might be limitations on the physical size of the exchanger which can be accommodated. (Gebremariam and Abebe 2002, p. 2) It should be remembered that a large physical size also means a larger cost. So a designer should consider all the design consideration factors properly. For example, if plate type heat exchangers are considered, the cost may depend on the cost of the frame, liquid connection, plates, and gaskets used. The working pressure is the main factor to affect the cost of the frame and the plate. The thickness of the main structural members and the tie bar diameter also depend on the working pressure. The process liquid and process requirements also have an impact on the cost of liquid connection, plates, and gaskets that need to be used. (Dennis Usher 1983, p. p.4.8.4-1)

In addition to considering the basic requirements, there are several issues that need to be taken in to account, such as the fouling factor. The performance of any kind of heat exchanger may decline through time due to the accumulation of deposits on the heat transfer surface area which in turn affect the rate of heat transfer negatively. (Cengel 1997, p. 709) So TEMA (The Tubular Exchanger Manufacturers' Association) published a table of fouling factor to help the design in preventing any kind of the fouling on duty. However, it should be noticed that the fouling factor only partially controls fouling. Moreover, it can be said that the entire concept of fouling factor is somehow indefinite. (Kraus 2003, p. 894) The fouling of heat transfer surfaces will give rise to high economic penalties that are currently dealt with the TEMA approach during the design of a heat exchanger. (Taborek, Hewitt, & Afgan 1983, pp. 795-796)

1.3 Computer-Aided Design of Heat Exchanger

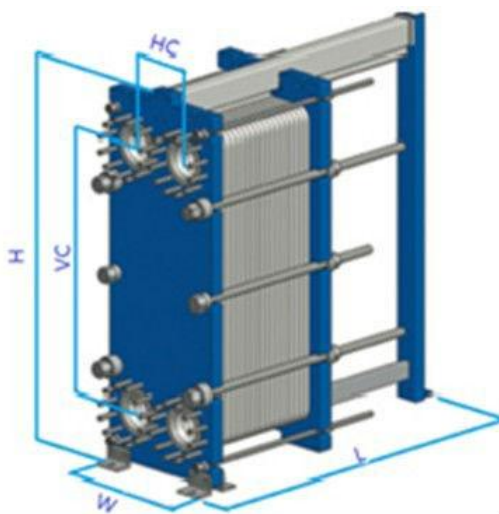
Nowadays, different software programs are developed for various heat exchangers with the aim of reducing human errors, tiresome tasks and increasing the efficiency of the design. The main reason why computer-aided design has to be used is that the design of heat exchangers involves several calculations and it is done by assuming certain proposed construction data. The heat transfer and pressure drop requirement should be calculated for the proposed data. In case the heat transfer and pressure drop requirement is not yet met, it has to be done again. The percentage over surface area requirement also needs to be considered during the design process which makes it difficult to do it manually. (Geberemariam & Abebe 2006, pp. 2-3)

In this design the gasket plate type heat exchanger is considered. While developing the software for Gasketed plate type heat exchangers, an assumption is made to the effect that the user is aware of the terms, the tables, and other factors that are and need to be used in the design.

2 GASKATED PLATE TYPE HEAT EXCHANGER

Plate type heat exchangers are made of thin plates forming flow channels. These thin plates are usually used to build heat exchangers that are either smooth or corrugated and they are either flat or round in an exchange. (Picture1). Normally, these exchangers cannot hold very high temperatures, pressures and temperature differences. Plate heat exchangers can further be classified as Gasket plate, Spiral plate, and Lamella. (Naik & Matawala 2013, p. 2249. Kakac & Liu 1998, p. 323-326)

The Gasketed plate heat exchangers (Picture 2) became to be known in the 1930s mainly in the food industries due to the ease they brought to maintenance and cleaning. They became more common during the 1960s with the development of effective plate geometries, assemblies, and better quality gasket material. At present, the use of Gasketed plate type heat exchangers in many industries for low and medium pressure liquid to liquid heat transfer requirement has become more common. They are now widely used for water treatment, oil cooling, food processing, and paper making industry and so on. Their range of possible applications has widened and they are also being used as an alternative for the shell and tube type heat exchangers in some industries. (Kakac & Liu 1998, p. 323)



Picture 2. Plate type heat exchanger (Jiangsu Boade heat exchanger 2016)



Picture 2. Gasketed plate heat exchangers (Every China 2016)

The Gasketed plate type heat exchanger has many advantages but the most important one is that it can be opened for inspection or maintenance. The following are additional advantages of the heat exchanger. (Kakac & Liu 1998, 328-330)

- The exchanger use gaskets to minimize internal leakage.
- Design can be done by making use of different plate size and pass arrangements.
- There is efficient heat transfer due to turbulence and small hydraulic diameters.
- These exchangers are very compact and have low weight.
- There is minimum heat loss.
- There is less chance of gaskets failure and thus, less chance of fluid mixing during operation.

2.1 Mechanical Features

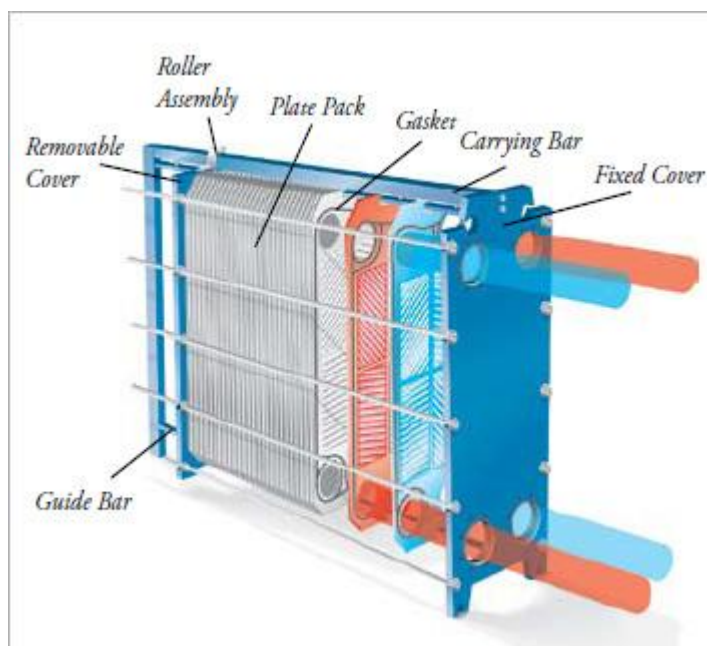
2.1.1 Plate pack, frames and plate type

The most common type of plate used for Gasketed plate type heat exchanger are the Chevron type. Nevertheless, other kinds of plates with a corrugated

surface pattern are also used that are called the washboard types. (Kakac & Liu 1998, pp. 323-326)

The metal plates are arranged alternately and bolted together between the end frames to form continuous tunnels or manifolds through which hot and cold fluids flow. Mechanical or hydraulic tightening devices that can control the pressure to the required level are used to tighten the plate pack and the pack is compressed between the head plates using bolts. These bolts are also used to keep the several hundred plates together in a frame. (Kakac & Liu 1998, pp. 323-326)

The hot fluid and the cold fluid flow on alternate side of the plate through the passage formed between the plate and the corner ports with the plate itself providing the most effective means to transfer heat from one fluid to the other. The Gaskets on the plates are used to seal the channels and guide the flow direction. Due to this flow, the cold fluid becomes warmer as the hot liquid becomes cooler. Many of the designs are usually one-pass/one-pass flow arrangement. (Kakac & Liu 1998, pp. 323-326) (Picture 3)



Picture 3. Parts of Gasketed plate type heat exchanger (Systemcorp 2016)

2.2 Mechanical Design Calculations

During the design different equations have been used. (Kakac & Liu 1998, pp. 337-347)

The surface enlargement factor (ϕ) is the ratio of developed length to the projected length and is used to calculate the effective flow path.

$$\phi = \frac{\text{developed length}}{\text{projected length}} \quad (1)$$

$$\phi = \frac{A_1}{A_{1P}} \quad (2)$$

where A_1 is the effective plate area specified by the manufacturer and

A_{1P} is the projected plate area.

The enlargement factor can be also approximated between 1.15 and 1.25 or simply the average value which is 1.17 can be also taken without calculating.

The projected area A_{1P} can be calculated from

$$A_{1P} = L_p \cdot L_w \quad (3)$$

And L_p (the projected plate length) and L_w (the effective channel width) are estimated from the vertical and horizontal port distance L_v and L_h and port diameter D_p as:

$$L_p = L_v - D_p \quad (4)$$

$$L_w = L_h + D_p \quad (5)$$

The mean channel spacing (b) which is formed by the two adjacent plates between the gaskets is calculated as:

$$b = p - t \quad (6)$$

Where p is the plate pitch, t is the plate thickness and b is the thickness of a fully compressed gasket. The Channel spacing is very important as it is used for the calculation of the mass velocity and Reynolds number.

The plate pitch is the ratio of compressed plate pack length (L_c) to the total number of plates (N_t)

$$p = \frac{L_c}{N_t} \quad (7)$$

The equivalent diameter of the channel D_e is defined as:

$$D_e = \frac{4 \times \text{channel flow area}}{\text{wetted surface}} = \frac{4A_c}{P_w} \quad (8)$$

$$D_e = \frac{4(b)(L_w)}{2(b+L_w\phi)} \approx \frac{2b}{\phi} \quad (9)$$

Based on the approximation that $b \ll L_w$

The heat transfer coefficient is obtained from the equation

$$\frac{hD_e}{k} = C_h \left(\frac{D_e G}{\mu} \right)^n \left(\frac{C_p \mu}{k} \right)^{1/3} \left(\frac{\mu_b}{\mu_w} \right)^{0.17} \quad (10)$$

The values of C_h and n which are constants in the equation (10) can be determined based on the flow characteristics and chevron angles.

The Reynolds number, Re , is calculated from the channel mass velocity and the equivalent diameter, D_e , of the Channel is defined as:

$$Re = \frac{G_c D_e}{\mu} \quad (11)$$

Where, G_c is the channel mass velocity and μ is the dynamic viscosity of the fluid at average inlet temperature.

The Channel mass velocity is calculated from the flow rate (\dot{m}), the number of channel per pass (N_{cp}), the mean flow gap (b), the effective channel width (L_w).

$$G_c = \frac{\dot{m}}{N_{cp} b L_w} \quad (12)$$

The number of channel per pass (N_{cp}) is obtained from the total number of plates (N_t) and the number of passes (N_p)

$$N_{cp} = \frac{N_t - 1}{2N_p} \quad (13)$$

The channel pressure drop is the sum of the frictional channel pressure drop (ΔP_c) and the port pressure drop (Δp_p). The frictional channel pressure drop is calculated as follows:

$$\Delta P_c = 4f \frac{L_{eff} N_p}{D_e} \frac{G_c^2}{2p} \left(\frac{\mu_b}{\mu_w} \right)^{-0.17} \quad (14)$$

Where, (L_{eff}), is the effective length of the fluid between inlet and outlet ports and is equal to the vertical port distance (L_v).

The friction factor can be obtained using the Reynolds number (Re) and the constant (K_p) and (m).

$$f = \frac{K_p}{Re^m} \quad (15)$$

The port pressure drop estimated from the port mass velocity (G_p), the number of passes (N_p) and density of the fluid used.

$$\Delta p_p = 1.4 N_p \frac{G_p^2}{2\rho} \quad (16)$$

The port mass velocity (G_p) is obtained by dividing the flow rate to the port diameter area as follows:

$$G_p = \frac{\dot{m}}{\frac{\pi D_p^2}{4}} \quad (17)$$

Where, \dot{m} is the total flow rate in the port opening and D_p is the port diameter.

The total pressure drop can now be calculated by adding together the frictional channel pressure drop and the port pressure drop.

$$\Delta P_t = \Delta p_c + \Delta p_p \quad (18)$$

From the above formulas, it is now possible to calculate the total pressure drop. The next important parameter that needs to be defined during the design step is the overall heat transfer coefficient for both clean and fouled surfaces.

The overall heat transfer coefficient for the clean surfaces is given by:

$$\frac{1}{U_c} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{t}{k_w} \quad (19)$$

Where (h_h) is the hot fluid heat transfer coefficient, (h_c) is the cold fluid heat transfer coefficient, (t) is the plate thickness and (k_w) is thermal conductivity of the plate material.

The overall heat transfer coefficient under fouling condition can be obtained:

$$\frac{1}{U_f} = \frac{1}{h_h} + \frac{1}{h_c} + \frac{t}{k_w} + R_{fh} + R_{fc} \quad (20)$$

Where (h_h) is the hot fluid heat transfer coefficient, (h_c) is the cold fluid heat transfer coefficient, (t) is the plate thickness, (k_w) is thermal conductivity of the plate material, (R_{fh}) is the fouling factor for the hot fluid and (R_{fc}) is the fouling factor for the cold fluid.

The relation between the cleanliness factor, (CF) and the overall heat transfer coefficient for both clean and fouled surfaces can be defined by (equation 21).

$$U_f = U_c (CF) = \frac{1}{\frac{1}{U_c} + R_{fh} + R_{fc}} \quad (21)$$

When considering, the heat transfer, the parameters that need to be defined are the heat duties for the cold and hot streams and the actual heat duties for both clean and fouled surfaces.

The required heat duties for cold and hot streams can be calculated as:

$$Q_r = (\dot{m}c_p)_c (T_{c_2} - T_{c_1}) = (\dot{m}c_p)_h (T_{h_1} - T_{h_2}) \quad (22)$$

Where (\dot{m}) is the mass flow rate, (c_p) is the specific heat capacity and (T) is temperature. The term (c) and (h) are to represent the cold and hot streams respectively.

The heat duty for clean surfaces is calculated as:

$$Q_c = U_c A_e \Delta T_m \quad (23)$$

Where (U_c) is the overall heat transfer coefficient for clean surfaces, (A_e) is the total effective area of the plate, and (ΔT_m) is the true mean temperature difference.

The heat duty for the fouled surfaces is calculated from the overall heat transfer coefficient for fouled surfaces (U_f), the total effective area of the plate (A_e), the correction factor (F), and the true mean temperature difference (ΔT_m).

$$Q_f = U_f A_e F \Delta T_m \quad (24)$$

The true mean temperature difference is calculated as

$$\Delta T_{lm,cf} = \frac{\Delta T_1 - \Delta T_2}{\ln \frac{\Delta T_1}{\Delta T_2}} \quad (25)$$

The safety factor can be obtained from the ratio of heat duties Q_f and Q_r

$$C_s = \frac{Q_f}{Q_r} \quad (26)$$

Additional formulas that have used during the designing steps are included here.

The total effective number of the plates, (N_e), has been obtained from:

$$N_e = N_t - 2 \quad (27)$$

One channel flow area, (A_{ch}), is calculated as

$$A_{ch} = b \times L_w \quad (28)$$

The single plate heat transfer area is the ratio of the total effective area of the plate to the total effective number of plates.

$$A_1 = \frac{A_e}{N_e} \quad (29)$$

Reynolds number for hot and cold streams can be obtained:

$$Re_h = \frac{G_h D_e}{\mu_h} \quad (30)$$

$$Re_c = \frac{G_c D_e}{\mu_c} \quad (31)$$

Finding the Reynolds number is important to find the different constants for various values of Chevron angles. But Reynolds number could be also specified or given by plate manufacturers.

The cleanness factor (CF) is the ratio of the heat transfer coefficient compared to the clean value.

$$CF = \frac{U_f}{U_c} \quad (32)$$

The cleanliness factor could be different for different plate material. It is very important data as it shows the percentage of how clean or fouled the heat exchanger surface could be under the assumed design condition.

It is good to include some form of safety factor in the designing process to allow some uncertainties. The safety factor (C_s) is the ratio of fouled heat duty to that of clean

$$C_s = \frac{Q_f}{Q_r} \quad (33)$$

The percentage over surface design (OS) is calculated as:

$$OS = 100U_c R_{ft} \quad (34)$$

The percentage over surface design lets the designer know the size of the exchanger size for the required duties.

3 DESIGN PROCEDURE

The design practices of heat exchanger consider a number of requirements to be taken in account. In heat exchanger design, the rating and sizing problems which are the most common problem need to be addressed properly.”(Kakac and Liu,p 31). In addition to this, the structural and economical considerations have to be also included. Several heat transfer equations were used for the design of heat exchanger.

3.1 Basic Design Procedure

The heat exchanger must satisfy the following main requirements.

- Heat transfer requirements
- Pressure drop requirements

The following steps are used in designing a heat exchanger.

- Identify the problem
- Select a heat exchanger type
- Calculate/Select initial design parameters
- Rate the initial design.
- Calculate the overall heat transfer coefficient and the total pressure drop.

3.2 Rating an Exchanger

In the rating process of a heat exchanger, the geometry (constructional design parameters) is used to check the thermal performance of a fully specified heat exchanger. Process conditions (flow rate, temperature, and pressure) and material/ fluid properties (density, thermal conductivity) are also used.

The first output from the rating process is either the outlet temperature for fixed tube length or the tube length itself to meet the outlet temperature requirement.

Second output from the rating process is the pressure drop for both fluid streams hence the pumping energy requirements and size are important.

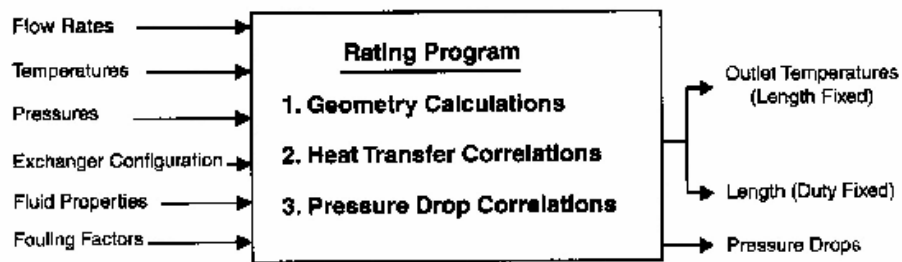


Figure 1. Rating procedures of heat exchanger (Abebe & Gebremariam 2006, p. 21)

The thermal rating is insufficient when the output of the rating analysis for the thermal performance is not acceptable, and this leads to geometrical modification of the design. If the required amount of heat cannot be transferred to satisfy specific outlet temperature, one should find a way to increase the heat transfer coefficient or increase the exchanger surface area.

It is said that the rating of the pressure drop is insufficient when the pressure drop calculated is greater than the allowable pressure drop, this can be adjusted by decreasing the number of tube passes or increasing the tube diameter or by just increasing the tube diameter and the number of tubes.

3.3 The Design Software of Gasketed Plate Type Heat Exchanger

In the design of gasketed type heat exchanger, the following flow chart was used. Figure 2 illustrates the very simplified version of the work flow.

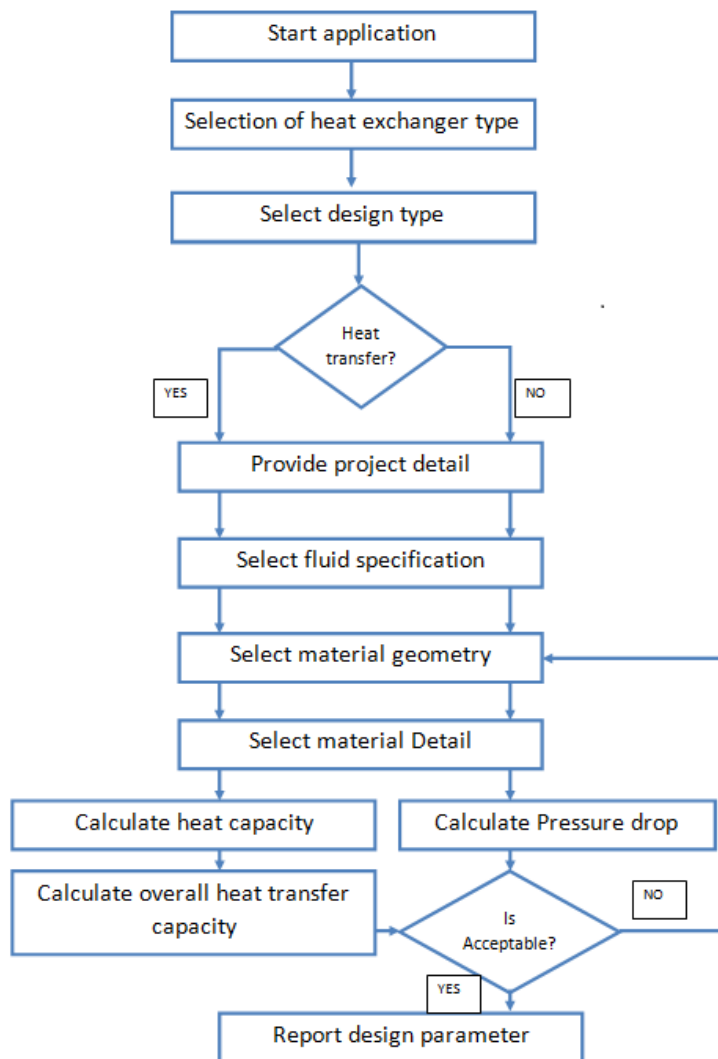


Figure 2. Work flow chart

When the application starts, the user selects the heat exchanger type and chooses the design type. The design type asks the user to choose either the heat transfer requirement or the pressure drop requirement. If the user chooses one of them, the application will still proceed to the common work flows that provide the project details, select the fluid specification, select the material geometry, and select the material details. After this step, it will proceed depending on the choice of the user. For example, if the user chooses the heat transfer requirement, then it calculates the heat capacity, the overall heat transfer capacity and checks if the results are acceptable. Finally, if the results are acceptable, it reports the design parameters. If not, it returns the user back to the selection of the material geometry for another trial. The same goes for the

pressure drop requirement. The application calculates the result and checks if it is acceptable or not and then displays the design parameters if the results are within the acceptable range. In the case where the result is not acceptable, it will return the user to the material geometry phase for a new attempt.

3.4 The Software development tools

The Visual Studio 2013 Professional and C# were used to program the design. Visual studio was chosen because of its learning curve and productivity advantages. (Sharp 2013)

Microsoft Visual Studio is an integrated development environment (IDE) from Microsoft. It can be used to develop console and graphical user interface applications along with Windows Forms applications, websites, web applications and web services in both native code together with managed code for all platforms supported by Microsoft Windows, Windows phone, Windows CE, .NET Framework, .NET Compact Framework and Microsoft Silverlight. (Halvorsen 2014, p. 5)

C# (pronounced "C sharp") is a programming language that is designed for building a variety of applications that run on the .NET Framework. C# is simple, powerful, type-safe, and object-oriented. The many innovations in C# enable rapid application development while retaining the expressiveness and elegance of C-style languages.

Visual C# is an implementation of the C# language by Microsoft. Visual Studio supports Visual C# with a full-featured code editor, compiler, project templates, designers, code wizards, a powerful and easy-to-use debugger, and other tools. The .NET Framework class library provides access to many operating system services and other useful, well-designed classes that speed up the development cycle significantly. (Microsoft 2016)

In this design, there are two projects and one solution. The two projects are the HED and HEDcore. The HEDcore (.dll) is the dynamic link library where all the formulas are written (sample code is attached at Appendix 2 to show how the equations are written) and the HED is the executable (.exe). This functional separation makes it possible to add another project if it is required.

The whole project is divided into two main components as described above. Each project is in a separate assembly as shown below. The external dependencies that are found in the dot net framework are also shown in (Figure 3) as Externals. The executable (exe) shown as HED.exe has few classes that are used as user interfaces to interact with the user. The executable assembly

loads the dll (dynamic link library), shown as HECore.dll, dynamically at run time whenever it needs to use the functions included. Both of these assemblies depend on the dot net framework. (see Figure 3)

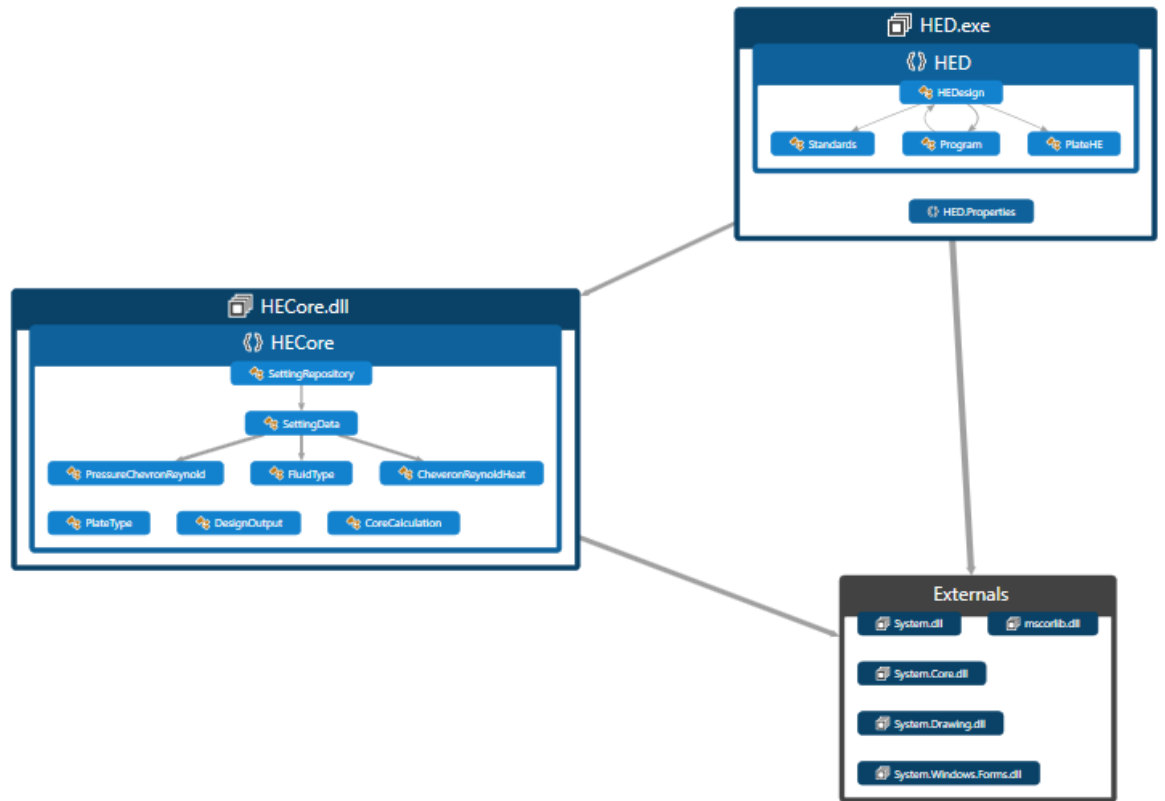


Figure 3. Assembly dependency of heat exchanger design software

4 IMPLEMENTATION OF THE DESIGN

The main purpose of the software design of gasketed plate type heat exchanger is simply to increase the accuracy and simplicity of the design. In the process of this design, different requirements from different perspectives have been considered. Previous experience and enough knowledge obtained from books and experts have been used in finding and dealing with the problem. This software has targeted mainly engineers mechanical or chemical or anyone who have interest in the design and rating of heat exchangers specifically gasketed plate type heat exchanger. This design has been tested with sample data and has shown to be working very well. To explain the implementation of the whole design and the different sections of the design procedure well, screenshots have been taken and explained below.

4.1 The Welcome Page

In this form, the designer or user of the software needs to choose the SI unit option. Basically, the design only supports SI units at the moment. In the future, there is a plan to include other measurement units to make it more unit friendly so that designers can use the software with any kind of measuring units which they are familiar with.

This form provides additional selection for the design type. The user needs to select the design for either the heat transfer capacity or the pressure drop requirements and then proceed to the design by just clicking the start design icon on the right side (Figure 4).

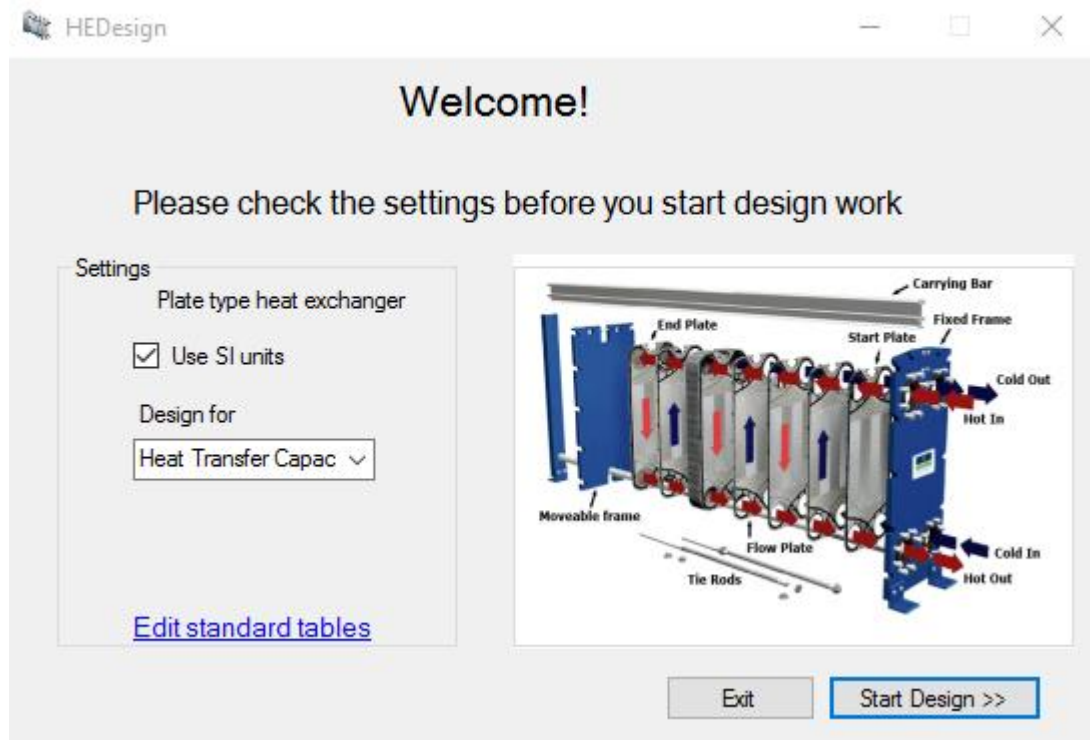


Figure 4. Welcome page

4.2 The Project Information Page

The Project Information Page or project description page provides the designer to fill the details of the project such as the project name, project number, project description, the designer name, date of the design and also some remarks. These pieces of information are very important and need to be displayed on the summary page of the whole design as they are required for information purposes and keeping records.(Figure 5)

Figure 5. Project description page

4.3 The Fluid Specification Form

The fluid specification form provides the standard process specifications of the fluid types that are used. In this design, the standard data for the waste water and cooling water are included. If the designer wants to use another kind of fluid, the designer is required to provide the details manually from the standard tables. (See Appendix 1). (Kakac & Liu 1998, p. 406- 413)

The inlet or outlet temperature of the fluid depending on which one is known and which one is required are calculated. The software also calculates the flow rate of each fluid provided that the temperatures of the hot and cold side fluids are predetermined or given. Basically, it can generate any of the missing temperatures or flow rate making use of the given data. (Figure 6)

Figure 6. Specification page

4.4 The Geometry Form

This is the most important area of the design where the designer uses tentative constructional data of the proposed heat exchanger type until the heat transfer or pressure drop requirements are met. The vertical port length (L_v), horizontal port width (L_h), all port diameter (D_p) plate thickness, channel spacing and compressed plate pack length (L_c) can be taken from the plate manufacturers.

For channel spacing and enlargement factor, the designer can also use the average value from the software by just clicking the use icon or can choose any intermediate value from the data given on the message box.

The material for the thermal conductivity depends on the kind of plate that the designer want to use and can be found from a standard table. The chevron angle should be chosen from the alternatives.

The number of plates or the total effective area can be calculated given that one of them is known or predetermined first. This means the engineer can first decide the number of plates needed to be used and then calculate the total

effective area or if the area is already known then the number of plates that are required can be calculated from the given and proposed data.(Figure 7)

PlateHE

Project Description Specification **Geometry** Material Details Heat Calculations Pressure Drop Reporting

Geometry

Number of plates Calculate

Total effective area m² Calculate

Data from plate manufacturer

Vertical port length (Lv) m

Horizontal port width (Lh) m

Port diameter (Dp) m

Plate thickness m

Channel spacing m Use values between 1.5mm - 5 mm

Compressed plate pack length m

Material thermal conductivity W/m²K

Number of passes

Chevron Angle, Beta Degrees

Enlargement factor (Use values between 1.15 - 1.25)

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Figure 7. Geometry page

4.5 The Material Detail Form

The material detail form provides the option to select the plate material, gasket material, the connection and frames that will be used.

Below the material selection, there is another form on the same page where the overall heat transfer coefficient for both fouled and clean surfaces will be calculated. In addition, the cleanliness factor and the percentage over surface design are also calculated here.

The overall heat transfer is the main important data of the design. Nevertheless, it is very important to also calculate the percentage over surface design as it gives the idea how large or small the heat exchanger could be. For example, if the percentage over surface area is more than 30%, it means that the exchanger is larger in size for the required duty. Due to this, there will be an increase in the investment and maintenance cost. The most acceptable

percentage over surface area is (10-30) %. If the value is above 30% the designer needs to go back and modify the proposed design data and calculate for a new one until both the heat requirement and the percentage over surface area are met. (Figure 8)

Figure 8. Material Detail page

4.6 Heat Calculation Form

This form provides the heat details that need to be calculated. It calculates the average temperature, the total heat load and the heat duties both for the clean and fouled surfaces. It also calculates the safety factor internally and finally gives one of the most important data which is the percentage over surface design. (Figure 9)

PlateHE

Project Description Specification Geometry Material Details Heat Calculations Pressure Drop Reporting

Average Temperature

Hot side *K Cold side *K

Heat load

Q= KW Calculate

Log mean temperature difference

Parallel Flow Counter flow *K Calculate

Heat duties

Heat duty for Clean surface (Qc) KW Heat duty for fouled surface (Qf) KW

Calculate

Percentage over surface design % Calculate

A good design has 10% - 30%

<< Back Next >>

Figure 9. Heat duty and heat load page

4.7 Pressure Drop Analysis Form

The pressure drop calculation is very important as it is one of the main considerations in the design of any kind of heat exchangers.

"The total pressure drop is composed of the channel pressure drop, ΔP_c , and the port pressure drop, ΔP_p " (Kakac & Liu 1998, p. 340).

Firstly, the fluid friction coefficients are calculated for both fluids by reading data from the standard table as a function of the Reynolds number for the value of the chevron angle chosen during the design. Then the frictional pressure drop is obtained for the cold and hot streams and also the port duct pressure drop is calculated for both fluids. Finally, the overall pressure drop is the sum of the port and frictional pressure drop, $(\Delta P_c + \Delta P_p)$, of the cold and hot side separately. (Figure 10)

PlateHE

Project Description Specification Geometry Material Details Heat Calculations Pressure Drop Reporting

Pressure drop analysis

Friction factor - Hot Friction factor - Cold Calculate

Frictional Pressure drop

Hot side KPa Cold side KPa Calculate

Port duct Pressure drop

Hot side KPa Cold side KPa Calculate

Overall total pressure drop

Hot side KPa Cold side KPa Calculate

<< Back Next >>

Figure 10. Pressure drop analysis page

4.8 The Summary Page

The summary page will provide the designer with data that are required for drafting and for manufacturing purposes. The final results of the most important data that are required for manufacturing of the heat exchangers, which have been found through different calculation, are displayed in this page. In addition, other data that have been provided during the design steps for information purposes are also displayed here. (Figure 10)

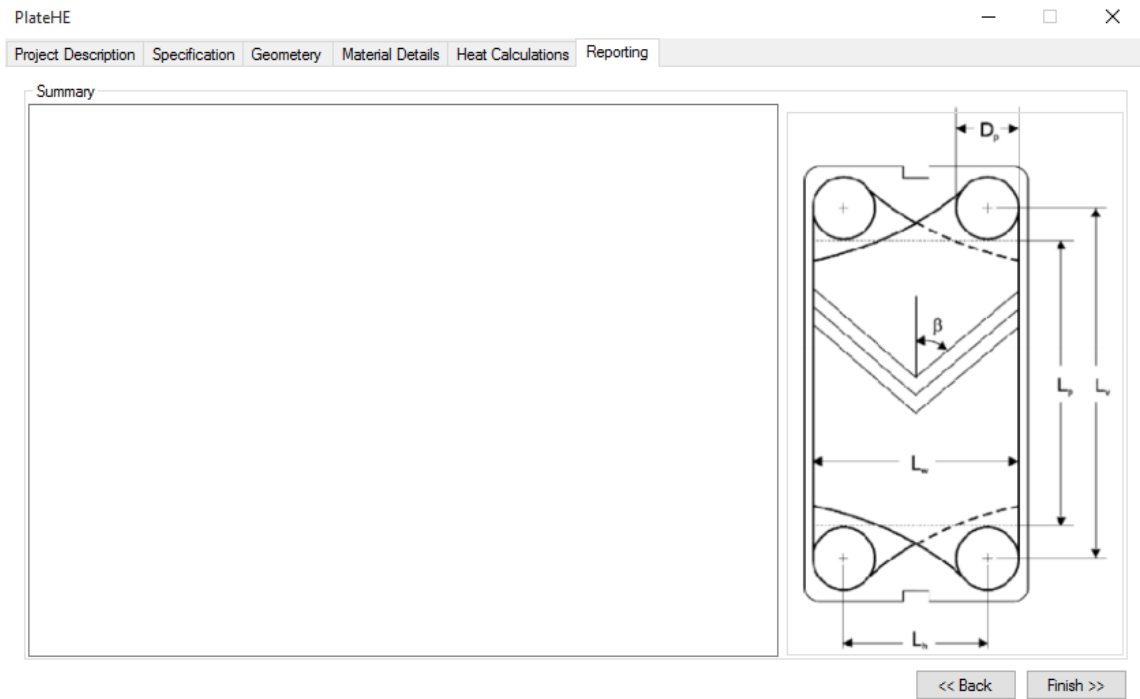


Figure 11. Summary page

5 CONCLUSION AND RECOMMENDATION

The design process of heat exchangers is complicated and it cannot be fully automated as it needs subjective decisions at intermediate steps. The main reason is that the designing process involves many steps and designers need to try a number of times making use of tentative data until they meet the targets of the design requirements.

The software developed for the gasketed plate type heat exchanger was made based on the concept of heat exchanger design and the knowledge of the programming language (C#). The main task was to meet the heat transfer and pressure drop requirements of the exchanger in consideration, in this case the Gasketed plate type exchanger. In addition to this, physical size and cost were also considered.

This software helps in providing better accuracy along with simplicity of the design by reducing the multiple tasks of calculating different trials to meet the desired value. It enables the designer to compare different results by changing different tentative parameters until he/she gets the desired values which they are looking for. In addition, the software program provides an option of selecting either the heat transfer requirement or the pressure drop requirement of the design. Based on this selection, it displays the constructional design data for manufacturing.

The Visual Studio 2013 Professional, the programming language (C#) along with the concept of heat exchangers design made the software possible.

In the future, there is a plan to include other measurement units to make the software more unit-friendly so that designers can use the software with any kind of measuring units which they are familiar with.

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Standard Tables

TABLE B.2
Thermophysical Properties of Saturated Ice-Water-Steam

P (bar)	T (K)	v_f^* (10^{-3} m ³ /kg)	v_g (m ³ /kg)	h^* (kJ/kg)	h_g (kJ/kg)	μ_g (10^{-4} Pa·s)	k_f^* (W/m·K)	k_g (W/m·K)	Pr_f^*	Pr_g
0.001	252.84	1.0010	1167	-374.9	2464.1	0.0723	2.40	0.0169		
0.002	260.21	1.0010	600	-360.1	2477.4	0.0751	2.35	0.0174		
0.003	265.11	1.0010	408.5	-350.9	2486.0	0.0771	2.31	0.0177		
0.004	267.95	1.0010	309.1	-344.4	2491.9	0.0780	2.29	0.0179		
0.005	270.74	1.0010	249.6	-337.9	2497.3	0.0789	2.27	0.0180		
0.006	273.06	1.0010	209.7	-333.6	2502	0.0798	2.26	0.0182		
0.0061	273.15	1.0010	206.0	-333.5	2502	0.0802	2.26	0.0182		
0.0061	273.15	1.0002	206.0	0.0	2502	0.0802	0.566	0.0182	13.04	0.817
0.008	276.73	1.0001	159.4	21.9	2508	0.0816	0.568	0.0184	11.66	0.823
0.010	280.13	1.0001	129.2	29.4	2513.4	0.0829	0.578	0.0186	10.39	0.828
0.02	290.66	1.0013	67.00	73.5	2532.7	0.0872	0.595	0.0193	7.51	0.841
0.03	297.24	1.0028	45.66	101.1	2544.8	0.0898	0.605	0.0195	6.29	0.854
0.04	302.13	1.0041	34.80	121.4	2553.6	0.0918	0.612	0.0198	5.57	0.865
0.05	306.04	1.0053	28.19	137.8	2560.6	0.0933	0.618	0.0201	5.08	0.871
0.06	309.33	1.0065	23.74	151.5	2566.6	0.0946	0.622	0.0203	4.62	0.877
0.08	314.68	1.0085	18.10	173.9	2576.2	0.0968	0.629	0.0207	4.22	0.883
0.10	318.98	1.0103	14.67	191.9	2583.9	0.0985	0.635	0.0209	3.87	0.893
0.20	333.23	1.0172	7.65	251.5	2608.9	0.1042	0.651	0.0219	3.00	0.913
0.30	342.27	1.0222	5.23	289.3	2624.6	0.1078	0.660	0.0224	2.60	0.929
0.40	349.04	1.0264	3.99	317.7	2636.2	0.1105	0.666	0.0229	2.36	0.941
0.5	354.50	1.0299	3.24	340.6	2645.5	0.1127	0.669	0.0233	2.19	0.951
0.6	359.11	1.0331	2.73	359.9	2653.0	0.1147	0.673	0.0236	2.06	0.961
0.8	366.66	1.0385	2.09	391.7	2665.3	0.1176	0.677	0.0242	1.88	0.979
1.0	372.78	1.0434	1.6937	417.5	2675.4	0.1202	0.6805	0.0244	1.735	1.009
1.5	384.52	1.0530	1.1590	467.1	2693.4	0.1247	0.6847	0.0259	1.538	1.000
2.0	393.38	1.0608	0.8854	504.7	2706.3	0.1280	0.6866	0.0268	1.419	1.013
2.5	400.58	1.0676	0.7184	535.3	2716.4	0.1307	0.6876	0.0275	1.335	1.027
3.0	406.69	1.0735	0.6056	561.4	2724.7	0.1329	0.6879	0.0281	1.273	1.040
3.5	412.02	1.0789	0.5240	584.3	2731.6	0.1349	0.6878	0.0287	1.224	1.050
4.0	416.77	1.0839	0.4622	604.7	2737.6	0.1367	0.6875	0.0293	1.185	1.057
4.5	421.07	1.0885	0.4138	623.2	2742.9	0.1382	0.6869	0.0298	1.152	1.066
5	424.99	1.0928	0.3747	640.1	2747.5	0.1396	0.6863	0.0303	1.124	1.073
6	432.00	1.1009	0.3155	670.4	2755.5	0.1421	0.6847	0.0311	1.079	1.091
7	438.11	1.1082	0.2727	697.1	2762.0	0.1443	0.6828	0.0319	1.044	1.105
8	445.57	1.1150	0.2403	720.9	2767.5	0.1462	0.6809	0.0327	1.016	1.115
9	448.51	1.1214	0.2148	742.6	2772.1	0.1479	0.6788	0.0334	0.992	1.127
10	453.03	1.1274	0.1943	762.6	2776.1	0.1495	0.6767	0.0341	0.973	1.137
12	461.11	1.1386	0.1632	798.4	2782.7	0.1523	0.6723	0.0354	0.943	1.156
14	468.19	1.1489	0.1407	830.1	2787.8	0.1548	0.6680	0.0366	0.920	1.175
16	474.52	1.1586	0.1237	858.6	2791.8	0.1569	0.6636	0.0377	0.902	1.191
18	480.26	1.1678	0.1103	884.6	2794.8	0.1589	0.6593	0.0388	0.889	1.206
20	485.53	1.1766	0.0995	908.6	2797.2	0.1608	0.6550	0.0399	0.877	1.229
25	497.09	1.1972	0.0799	962.0	2800.9	0.1648	0.6447	0.0424	0.859	1.251
30	506.99	1.2163	0.0666	1008.4	2802.3	0.1684	0.6347	0.0449	0.849	1.278
35	515.69	1.2345	0.0570	1049.8	2802.0	0.1716	0.6250	0.0472	0.845	1.306
40	523.48	1.2521	0.0497	1087.4	2800.3	0.1746	0.6158	0.0496	0.845	1.331
45	530.56	1.2691	0.0440	1122.1	2797.7	0.1775	0.6068	0.0519	0.849	1.358
50	537.06	1.2858	0.0394	1154.5	2794.2	0.1802	0.5981	0.0542	0.855	1.386
60	548.70	1.3187	0.0324	1213.7	2785.0	0.1854	0.5813	0.0589	0.874	1.442
70	558.94	1.3515	0.0274	1267.4	2773.5	0.1904	0.5653	0.0638	0.901	1.503
80	568.12	1.3843	0.0235	1317.1	2759.9	0.1954	0.5499	0.0688	0.936	1.573
90	576.46	1.4179	0.0205	1363.7	2744.6	0.2005	0.5352	0.0741	0.978	1.651
100	584.11	1.4526	0.0180	1408.0	2727.7	0.2057	0.5209	0.0798	1.029	1.737
110	591.20	1.4887	0.0160	1450.6	2709.3	0.2110	0.5071	0.0159	1.090	1.837
120	597.80	1.5268	0.0143	1491.8	2689.2	0.2166	0.4936	0.0925	1.163	1.963
130	603.98	1.5672	0.0128	1532.0	2667.0	0.2224	0.4806	0.0998	1.252	2.126
140	609.79	1.6106	0.0115	1571.6	2642.4	0.2286	0.4678	0.1080	1.362	2.343
150	615.28	1.6579	0.0103	1611.0	2615.0	0.2373	0.4554	0.1307	1.502	2.571
160	620.48	1.7103	0.0093	1650.5	2584.9	0.2497	0.4433	0.1280	1.688	3.041
170	625.41	1.7696	0.0084	1691.7	2551.6	0.2627	0.4315	0.1404	2.098	3.344
180	630.11	1.8399	0.0075	1734.8	2513.9	0.2766	0.4200	0.1557	2.360	3.807
190	634.58	1.9260	0.0067	1778.7	2470.6	0.2920	0.4087	0.1749	2.951	8.021
200	638.85	2.0370	0.0059	1826.5	2410.4	0.3094	0.3976	0.2007	4.202	12.16

Standard Tables

TABLE B.2 (CONTINUED)

Thermophysical Properties of Saturated Ice-Water-Steam

P (bar)	s_f^* (kJ/kg·K)	s_g (kJ/kg·K)	$c_{p,f}^*$ (kJ/kg·K)	$c_{p,g}$ (kJ/kg·K)	μ (10^{-4} Pa·s)	γ	γ_g	\bar{v}_f (m/s)	\bar{v}_g (m/s)	σ^* (N/m)
0.001	-1.378	9.848	1.957							
0.002	-1.321	9.585	2.015							
0.003	-1.280	9.456	2.053							
0.004	-1.260	9.339	2.075							
0.005	-1.240	9.250	2.097	1.851						
0.006	-1.222	9.160	2.106	1.854						
0.0061	-1.221	9.159	2.116	1.854						
0.0061	0.0000	9.159	4.217	1.854	17.50					0.0756
0.008	0.0543	9.0379	4.206	1.856	15.75					0.0751
0.010	0.1059	8.9732	4.198	1.858	14.30					0.0747
0.02	0.2605	8.7212	4.183	1.865	10.67					0.0731
0.03	0.3543	8.5756	4.180	1.870	9.09					0.0721
0.04	0.4222	8.4724	4.179	1.874	8.15					0.0714
0.05	0.4761	8.3928	4.178	1.878	7.51					0.0707
0.06	0.5208	8.3283	4.178	1.881	7.03					0.0702
0.08	0.5925	8.2266	4.179	1.887	6.35					0.0693
0.10	0.6493	8.1482	4.180	1.894	5.88					0.0686
0.20	0.8321	7.9065	4.184	1.917	4.66					0.0661
0.30	0.9441	7.7670	4.189	1.935	4.09					0.0646
0.40	1.0261	7.6686	4.194	1.953	3.74					0.0634
0.5	1.0912	7.5928	4.198	1.967	3.49					0.0624
0.6	1.1454	7.5309	4.201	1.978	3.30					0.0616
0.8	1.2330	7.4338	4.209	2.015	3.03					0.0605
1.0	1.3027	7.3598	4.222	2.048	2.801	1.136	1.321	438.74	472.98	0.0589
1.5	1.4336	7.2234	4.231	2.077	2.490	1.139	1.318	445.05	478.73	0.0566
2.0	1.5301	7.1268	4.245	2.121	2.295	1.141	1.316	449.51	482.78	0.0548
2.5	1.6071	7.0520	4.258	2.161	2.156	1.142	1.314	452.92	485.88	0.0534
3.0	1.6716	6.9909	4.271	2.198	2.051	1.143	1.313	455.65	488.36	0.0521
3.5	1.7273	6.9392	4.282	2.233	1.966	1.143	1.311	457.91	490.43	0.0510
4.0	1.7764	6.8943	4.294	2.266	1.897	1.144	1.310	459.82	492.18	0.0500
4.5	1.8204	6.8547	4.305	2.298	1.838	1.144	1.309	461.46	493.69	0.0491
5	1.8604	6.8192	4.315	2.329	1.787	1.144	1.308	462.88	495.01	0.0483
6	1.9308	6.7575	4.335	2.387	1.704	1.144	1.306	465.23	497.22	0.0468
7	1.9918	6.7052	4.354	2.442	1.637	1.143	1.304	467.08	498.99	0.0455
8	2.0457	6.6596	4.372	2.495	1.581	1.142	1.303	468.57	500.55	0.0444
9	2.0941	6.6192	4.390	2.546	1.534	1.142	1.302	469.78	501.64	0.0433
10	2.1382	6.5821	4.407	2.594	1.494	1.141	1.300	470.76	502.64	0.0423
12	2.2161	6.5194	4.440	2.688	1.427	1.139	1.298	472.23	504.21	0.0405
14	2.2837	6.4651	4.472	2.777	1.373	1.137	1.296	473.18	505.33	0.0389
16	2.3436	6.4175	4.504	2.862	1.329	1.134	1.294	473.78	506.12	0.0375
18	2.3976	6.3751	4.534	2.944	1.291	1.132	1.293	474.09	506.65	0.0362
20	2.4469	6.3367	4.564	3.025	1.259	1.129	1.291	474.18	506.98	0.0350
25	2.5543	6.2536	4.640	3.219	1.193	1.123	1.288	473.71	507.16	0.0323
30	2.6455	6.1837	4.716	3.407	1.143	1.117	1.284	472.51	506.65	0.0300
35	2.7253	6.1229	4.792	3.593	1.102	1.111	1.281	470.80	505.66	0.0200
40	2.7965	6.0685	4.870	3.781	1.069	1.104	1.278	468.72	504.29	0.0261
45	2.8612	6.0191	4.951	3.972	1.040	1.097	1.275	466.31	502.68	0.0244
50	2.9206	5.9735	5.034	4.168	1.016	1.091	1.272	463.67	500.73	0.0229
60	3.0273	5.8908	5.211	4.582	0.975	1.077	1.266	457.77	496.33	0.0201
70	3.1219	5.8162	5.405	5.035	0.942	1.063	1.260	451.21	491.31	0.0177
80	3.2076	5.7471	5.621	5.588	0.915	1.048	1.254	444.12	485.80	0.0156
90	3.2867	5.6820	5.865	6.100	0.892	1.033	1.249	436.50	479.90	0.0136
100	3.3606	5.6198	6.142	6.738	0.872	1.016	1.244	428.24	473.67	0.0119
110	3.4304	5.5595	6.463	7.480	0.855	0.998	1.239	419.20	467.13	0.0103
120	3.4972	5.5002	6.838	8.384	0.840	0.978	1.236	409.38	460.25	0.0089
130	3.5616	5.4408	7.286	9.539	0.826	0.956	1.234	398.90	453.00	0.0076

Standard Tables

TABLE B.2 (CONTINUED)

Thermophysical Properties of Saturated Ice-Water-Steam

P (bar)	s_f^* (kJ/kg·K)	s_g (kJ/kg·K)	c_{pf}^* (kJ/kg·K)	c_{pg} (kJ/kg·K)	μ (10^{-4} Pa·s)	γ_f	γ_g	\bar{v}_f (m/s)	\bar{v}_g (m/s)	σ^* (N/m)
140	3.6243	5.3803	7.834	11.07	0.813	0.935	1.232	388.00	445.34	0.0064
150	3.6859	5.3178	8.529	13.06	0.802	0.916	1.233	377.00	437.29	0.0053
160	3.7471	5.2531	9.456	15.59	0.792	0.901	1.235	366.24	428.89	0.0043
170	3.8197	5.1855	11.30	17.87	0.782	0.867	1.240	351.19	420.07	0.0034
180	3.8765	5.1128	12.82	21.43	0.773	0.838	1.248	336.35	410.39	0.0026
190	3.9429	5.0332	15.76	27.47	0.765	0.808	1.260	320.20	399.87	0.0018
200	4.0149	4.9412	22.05	39.31	0.758	0.756	1.280	298.10	387.81	0.0011

* Above the solid line, solid phase; below the line, liquid.

From Liley, P. E. (1987) Thermophysical properties, in *Boilers, Evaporators, and Condensers*, S. Kakaç (Ed.), Wiley, New York.

TABLE B.3

Thermophysical Properties of Steam at 1-bar Pressure

T (K)	v (m ³ /kg)	h (kJ/kg)	s (kJ/kg·K)	c_p (kJ/kg·K)	c_v (kJ/kg·K)	γ	Z	\bar{v} (m/s)	μ (10^{-4} Pa·s)	k (W/m·K)	Pr
373.15	1.679	2676.2	7.356	2.029	1.510	1.344	0.9750	472.8	1.20	0.0248	0.982
400	1.827	2730.2	7.502	1.996	1.496	1.334	0.9897	490.4	1.32	0.0268	0.980
450	2.063	2829.7	7.741	1.981	1.498	1.322	0.9934	520.6	1.52	0.0311	0.968
500	2.298	2928.7	7.944	1.983	1.510	1.313	0.9959	540.3	1.73	0.0358	0.958
550	2.531	3028	8.134	2.000	1.531	1.306	0.9971	574.2	1.94	0.0410	0.946
600	2.763	3129	8.309	2.024	1.557	1.300	0.9978	598.6	2.15	0.0464	0.938
650	2.995	3231	8.472	2.054	1.589	1.293	0.9988	621.8	2.36	0.0521	0.930
700	3.227	3334	8.625	2.085	1.620	1.287	0.9989	643.9	2.57	0.0581	0.922
750	3.459	3439	8.770	2.118	1.653	1.281	0.9992	665.1	2.77	0.0646	0.913
800	3.690	3546	8.908	2.151	1.687	1.275	0.9995	685.4	2.98	0.0710	0.903
850	3.921	3654	9.039	2.185	1.722	1.269	0.9996	705.1	3.18	0.0776	0.897
900	4.152	3764	9.165	2.219	1.756	1.264	0.9996	723.9	3.39	0.0843	0.892
950	4.383	3876	9.286	2.253	1.791	1.258	0.9997	742.2	3.59	0.0912	0.886
1000	4.614	3990	9.402	2.286	1.823	1.254	0.9998	760.1	3.78	0.0981	0.881
1100	5.076	4223	9.625	2.36			0.9999	794.3	4.13	0.113	0.858
1200	5.538	4463	9.384	2.43			1.0000	826.8	4.48	0.130	0.837
1300	5.999	4711	10.032	2.51			1.0000	857.9	4.77	0.144	0.826
1400	6.461	4965	10.221	2.58			1.0000	887.9	5.06	0.160	0.816
1500	6.924	5227	10.402	2.65			1.0002	916.9	5.35	0.18	0.788
1600	7.386	5497	10.576	2.73			1.0004	945.0	5.65	0.21	0.735
1800	8.316	6068	10.912	3.02			1.0011	999.4	6.19	0.33	0.567
2000	9.263	6706	11.248	3.79			1.0036	1051.0	6.70	0.57	0.445

From Kakaç, S. and Yener, Y. (1995) *Convective Heat Transfer*, 2nd ed., CRC Press, Boca Raton, FL. With permission.

Standard Tables

TABLE B.4

Thermophysical Properties of Water-Steam at High Pressures

T (K)	v (m ³ /kg)	h (kJ/kg)	s (kJ/kg·K)	c_p (kJ/kg·K)	c_v (kJ/kg·K)	γ	Z	\bar{v}_1 (m/s)	μ (Pa·s)	k (W/m·K)	Pr
<i>P = 10 bar</i>											
300	1.003-3 ^a	113.4	0.392	4.18	4.13	1.01	0.0072	1500	8.57-4 ^b	0.615	5.82
350	1.027-3	322.5	1.037	4.19	3.89	1.08	0.0064	1552	3.70-4	0.668	2.32
400	1.067-3	533.4	1.600	4.25	3.65	1.17	0.0058	1509	2.17-4	0.689	1.34
450	1.123-3	749.0	2.109	4.39	3.44	1.28	0.0054	1399	1.51-4	0.677	0.981
500	0.221	2891	6.823	2.29	1.68	1.36	0.957	535.7	1.71-5 ^c	0.038	1.028
600	0.271	3109	7.223	2.13	1.61	1.32	0.987	592.5	2.15-5	0.047	0.963
800	0.367	3537	7.837	2.18	1.70	1.28	0.994	686.2	2.99-5	0.072	0.908
1000	0.460	3984	8.336	2.30	1.83	1.26	0.997	759.4	3.78-5	0.099	0.881
1500	0.692	5224	9.337	2.66			1.000	917.2	5.35-5	0.18	0.80
2000	0.925	6649	10.154	3.29			1.002	1050	6.70-5	0.39	0.57
<i>P = 50 bar</i>											
300	1.001-3	117.1	0.391	4.16	4.11	1.01	0.0362	1508	8.55-4	0.618	5.76
350	1.025-3	325.6	1.034	4.18	3.88	1.08	0.0317	1561	3.71-4	0.671	2.31
400	1.064-3	536.0	1.596	4.24	3.64	1.16	0.0288	1519	2.18-4	0.691	1.34
450	1.120-3	751.4	2.103	4.37	3.43	1.27	0.0270	1437	1.52-4	0.681	0.975
500	1.200-3	976.1	2.575	4.64	3.24	1.43	0.0260	1246	1.19-4	0.645	0.856
600	0.0490	3013	6.350	2.85	1.94	1.47	0.885	560.5	2.14-5	0.054	1.129
800	0.0713	3496	7.049	2.31	1.74	1.32	0.966	674.5	3.03-5	0.075	0.929
1000	0.0911	3961	7.575	2.35	1.85	1.27	0.987	756.5	3.81-5	0.102	0.880
1500	0.1384	5214	8.589	2.66			1.000	918.8	5.37-5	0.18	0.81
2000	0.1850	6626	9.398	3.12			1.002	1053	6.70-5	0.33	0.64
<i>P = 100 bar</i>											
300	9.99-4	121.8	0.390	4.15	4.09	1.01	0.0722	1516	8.52-4	0.622	5.69
350	1.022-3	329.6	1.031	4.17	3.87	1.08	0.0633	1571	3.73-4	0.675	2.31
400	1.061-3	539.6	1.590	4.23	3.64	1.16	0.0575	1532	2.20-4	0.694	1.34
450	1.116-3	754.1	2.097	4.35	3.43	1.27	0.0537	1452	1.53-4	0.685	0.975
500	1.193-3	977.3	2.567	4.60	3.24	1.42	0.0517	1269	1.21-4	0.651	0.853
600	0.0201	2820	5.775	5.22	2.64	1.97	0.726	502.3	2.14-5	0.073	1.74
800	0.0343	3442	6.685	2.52	1.82	1.38	0.929	662.4	3.08-5	0.081	0.960
1000	0.0449	3935	7.233	2.44	1.88	1.30	0.973	753.3	3.85-5	0.107	0.876
1500	0.0692	5203	8.262	2.68			1.000	921.1	5.37-5	0.18	0.82
2000	0.0926	6616	9.073	3.08			1.003	1057	6.70-5	0.31	0.67
<i>P = 250 bar</i>											
300	9.93-3	135.3	0.385	4.12	4.06	1.02	0.1792	1542	8.48-4	0.634	5.50
350	1.016-3	341.7	1.022	4.14	3.84	1.08	0.1572	1599	3.78-4	0.686	2.28
400	1.053-3	550.1	1.578	4.20	3.62	1.16	0.1426	1568	2.24-4	0.704	1.33
450	1.105-3	762.4	2.078	4.30	3.41	1.26	0.1330	1496	1.57-4	0.696	0.969
500	1.175-3	981.9	2.541	4.50			0.1273	1331	1.24-4	0.666	0.838
600	1.454-3	1479	3.443	5.88	4.22	1.40	0.1313	896.9	8.63-5	0.532	0.952
800	0.0120	3261	6.086	3.41			0.813	627.3	3.29-5	0.109	1.03
1000	0.0173	3845	6.741	2.69	1.97	1.36	0.935	745.9	3.98-5	0.125	0.856
1500	0.0277	5186	7.827	2.73			1.000	929.1	5.40-5	0.18	0.819
2000	0.0372	6608	8.642	3.04			1.008	1068			
<i>P = 500 bar</i>											
300	9.83-4	157.7	0.378	4.06	3.98	1.02	0.3549	1583	8.45-4	0.650	5.28
350	1.005-3	361.8	1.007	4.10	3.81	1.08	0.3112	1644	3.87-4	0.700	2.27
400	1.041-3	567.8	1.557	4.14	3.59	1.15	0.2820	1623	2.31-4	0.719	1.33
450	1.088-3	776.9	2.050	4.23	3.39	1.25	0.2618	1561	1.62-4	0.714	0.960
500	1.151-3	991.5	2.502	4.37			0.2493	1418	1.29-4	0.689	0.822
600	1.362-3	1456	3.346	5.08	3.72	1.37	0.2459	1080	9.34-5	0.588	0.808
800	4.576-3	2895	5.937	5.84	2.79	2.10	0.620	597.8	4.04-5	0.178	1.33
1000	8.102-3	3697	6.302	3.17	1.81	1.76	0.878	742.1	4.28-5	0.150	0.905
1500	0.0139	5157	7.484	2.82			1.004	943.6			
2000	0.0188	6595	8.310	3.04			1.018	1086			

^a Notation -3 signifies $\times 10^{-3}$.^b Notation -4 signifies $\times 10^{-4}$.^c Notation -5 signifies $\times 10^{-5}$.

Standard Tables

TABLE B.5

Properties of Liquid Metals

Liquid Metal and Melting Point	Temperature, °C	k W/(m·K)	ρ kg/m ³	c kJ/(kg·K)	$\mu \times 10^4$ kg/(m·s)
Bismuth (288°C)	315	16.40	10000	0.1444	16.22
	538	15.70	9730	0.1545	10.97
	760	15.70	9450	0.1645	7.89
Lead (327°C)	371	18.26	10500	0.159	24.0
	482	19.77	10400	0.155	19.25
	704	—	10130	—	13.69
Mercury (-39°C)	10	8.14	13550	0.138	15.92
	149	11.63	13200	0.138	10.97
	315	14.07	12800	0.134	8.64
Potassium (64°C)	315	45.0	804	0.80	3.72
	427	39.5	740	0.75	1.78
	704	33.1	674	0.75	1.28
Sodium (98°C)	93	86.0	930	1.38	7.0
	371	72.5	860	1.29	2.81
	704	59.8	776	1.26	1.78
56% Na, 44% K (19°C)	93	25.6	885	1.13	5.78
	371	27.6	820	1.06	2.36
	704	28.8	723	1.04	5.94
22% Na, 78% K (-11°C)	93	24.4	850	0.95	4.94
	400	26.7	775	0.88	2.07
	760	—	690	0.88	1.46
44.5% Pb, 55.5% Bi (125°C)	315	9.0	10500	0.147	—
	371	11.9	10200	0.47	15.36
	650	—	9820	—	11.47

From Kakaç, S. and Yener, Y. (1995) *Convective Heat Transfer*, 2nd ed., CRC Press, Boca Raton, FL. With permission.

Sample Codes

```

using System;
using System.Collections.Generic;
using System.Linq;
using System.Text;
using System.Threading.Tasks;

namespace HECore
{
    public class CoreCalculation
    {
        public double CalculateHeatTransfer(double flowRate, double specificHC,
double changeTemp)
        {
            double heatTransfer = flowRate * specificHC * changeTemp;
            return heatTransfer;
        }

        public int GetEffectiveNumberOfPlates(int numberOfPlates)
        {
            return numberOfPlates - 2;
        }

        public double SinglePlateHTArea(double totalEffectiveArea, double
totalNumberOfEffectivePlate)
        {
            return totalEffectiveArea / totalNumberOfEffectivePlate;
        }

        public int CalculateEffectiveNumber(double a1, double ae)
        {
            return Convert.ToInt32(ae / a1);
        }

        public double CalculateLw(double Lh, double D) //Lw
        {
            return Lh + D;
        }

        public double CalculateLp(double Lv, double D)
        {
            return Lv - D;
        }

        public double CalculateA1P(double Lp, double Lw)//projected plate area
        {
            return Lp * Lw;
        }

        public double CalculateA1(double enlargementFactor, double A1p)//one plate
area
        {
            return A1p * enlargementFactor;
        }

        public double CalculateEffectiveArea(double numberOfEffectivePlates,
double A1)//Ae
    }
}

```

Sample Codes

```

    {
        return A1 * numberOfEffectivePlates;
    }

    public double CalculatePlatePitch(double totalPlateNumber, double
compressedPlatePackLength)//p
    {
        return compressedPlatePackLength / totalPlateNumber;
    }

    public double CalculateChennelFlowGap(double platePitch, double
tickness)//b
    {
        return platePitch - tickness;
    }

    public double CalculateCEDiameter(double flowGap, double
enlargmentFactor)//De enlargementfactor = phi
    {
        return (2*flowGap)/enlargmentFactor;
    }

    public double CalculateNumberChannelPass(double totalPlateNumber, double
numberOfPass)//Ncp numberofpass=np
    {
        return (totalPlateNumber -1)/(2*numberOfPass);
    }

    public double CalculateMassflowRatePerChannel(double flowrate,double
numberOfChannelsPerPass)//mch
    {
        return flowrate/numberOfChannelsPerPass;
    }

    public double CalculateReynoldNumber(double massVelocity, double
channelEquiDiameter, double viscosity) //R
    {
        return (massVelocity * channelEquiDiameter) / viscosity;
    }

    public double CalculateMassFlowRate(double flowrate,double ncp) //mch
    {
        return flowrate / ncp;
    }

    public double CalculateMassVelocity(double mch, double ach) //gch
    {
        return mch / ach;
    }

    public double CalculateOneChannelFlowArea(double b, double lw) //ach
    {
        return b * lw;
    }

    public double CalculateNu(double rey, double prand, double ch, double n)

```

Sample Codes

```

    {
        double res = 0;
        res = ch * (Math.Pow(rey, n)) * (Math.Pow(prand, 0.333));
        return res;
    }

    public double CalculateH(double nu, double thermalcon, double ced)
    {
        return ((nu * thermalcon) / ced);
    }

    public double CalculateU(double hh, double hc, double tickness, double
plateThermal)
    {
        return ((hh * hc * plateThermal)/(hh * plateThermal + hc *
plateThermal + hh * hc * tickness));
    }

    public double CalculateUf(double uc, double fouling)
    {
        return (uc / (1 + uc * fouling));
    }

    public double CalculateLogMeanTemprature(double hotChange, double
coldChange)
    {
        if(hotChange == coldChange)
        {
            return hotChange;
        }
        else
            return ((hotChange - coldChange) / (Math.Log(hotChange /
coldChange)));
    }

    public double CalculateCleannessFactor(double uc, double uf)//cf
    {
        return uf/uc;
    }

    public double CalculateHeatDuty(double ucf, double ae, double logmeanT)
    {
        return ucf * ae * logmeanT;
    }

    public double CalculatePercentageOverSurface(double uc, double fouling)
    {
        return 100*uc*fouling;
    }

    public double CalculateFrictionFactor(double kp, double reynold, double m)
    {
        return (kp / (Math.Pow(reynold, m)));
    }

```

Sample Codes

```
    public double CalculateFrictionPD(double ff, double lv, double np, double
ced, double gc, double density)
    {
        return ((4 * ff * lv * np * gc * gc) / (2 * density * ced));
    }

    public double CalculatePortDuctPD(double np, double gp, double density)
    {
        return ((1.4 * np * gp * gp) / (2 * density));
    }

    public double CalculatePortMassVelocity(double flowrate, double d)
    {
        return ((4 * flowrate) / (Math.PI * d * d));
    }
}
```