**HYBRID TECHNIQUE OF ANALYSIS AND DEVELOPMENT OF HEAT EXCHANGERS FOR VUILLEUMIER HEAT PUMPS.**

A Thesis Presented

by

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Abstract of the Thesis

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Vuilleumier heat pumps (VHPs) are heat driven heat pumps working on the Vuilleumier cycle. The Vuilleumier heat pump is a closed system that captures heat at lower temperatures and deposits the captured heat energy to some intermediate temperature at the expense of high temperature heat energy. The heat energy is the input to the system by burning fuel externally. Due to the interfacing between the heat pump and the outside heat sources and heat sinks, heat exchangers represent critical components in the VHP and have a major impact on overall coefficient of performance.

High operating pressures, minimal dead volume and minimal pressure drop are several key challenges that need to be addressed during the down-selection of heat exchangers for Vuilleumier heat pumps. This thesis focuses on the analysis and configuration selection of the VHP heat exchangers using simplified 2D models and tailored analysis techniques to evaluate the performance of heat exchangers.

Cross- flow heat exchangers are generally well suited for their application in VHPs because of their robust structure and compact in size. Because the fluid flow in cross flow heat exchangers predominantly covers all three dimensions, simulating the fluid flow in 3D becomes an integral part of CFD modelling and thermal analysis. 3D CFD models are particularly computational intensive due to the involvement of large number of elements in the mesh. Appropriately developed 2D models can replicate 3D models efficiently in many ways (particularly for cross flow heat exchangers).They tend to save considerable computational time and can also maintain good levels of accuracy. The appropriate simplifying of 3D models, however, require good approximation techniques and use of analytical and numerical methods to converge on a solution. The literature will describe efficient techniques of blending analytical and numerical methods to derive the solution.

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I would like to dedicate this work to my mother whose immense cooperation and guidance has helped me throughout my college life and the values which she imparted in me would be carried throughout my engineering career. I want to express a special thanks to my father without whom none of this would have been possible.

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# Chapter 1 INTRODUCTION

## 1.1 –Electricity prices in the U.S.

Electrical energy is most widely used form of energy in Residential, Commercial and Industrial sectors. Energy transmission, storage and convertibility to other forms of energy are the advantages that electrical energy has over other forms of energies. Electrical energy, however is not the cheapest form of energy in the market. US electricity prices are at hike. On an average, since the year 2003 the annual growth in US residential electricity price has been around 3.2 %, Fig.1.1. The factors affecting the cost of the electricity include the growth in the prices of fuel, needs of new infrastructure and environmental concerns.
U.S. residential electricity consumption, in past years more or less has shown an increasing trend Fig.1.4. In residential sector, space heating, water heating and cooling demand comprise of around 42% of the total electrical energy requirement.

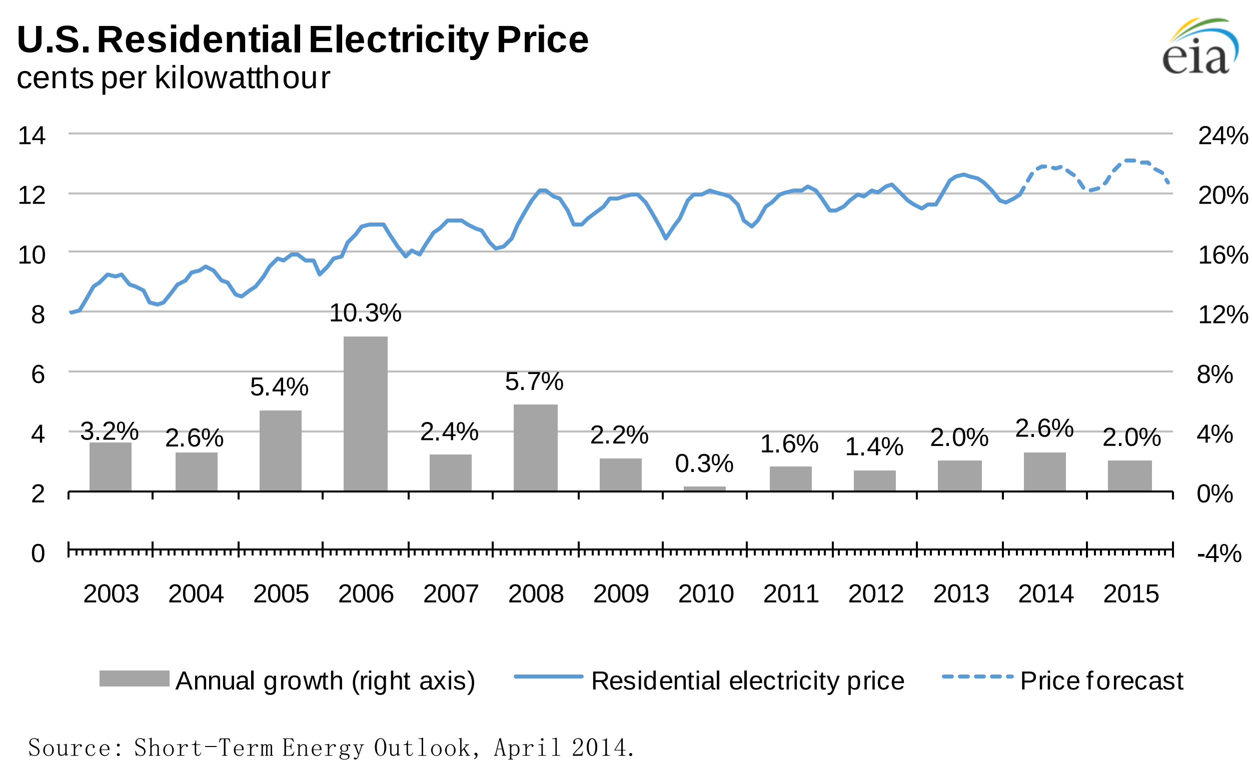
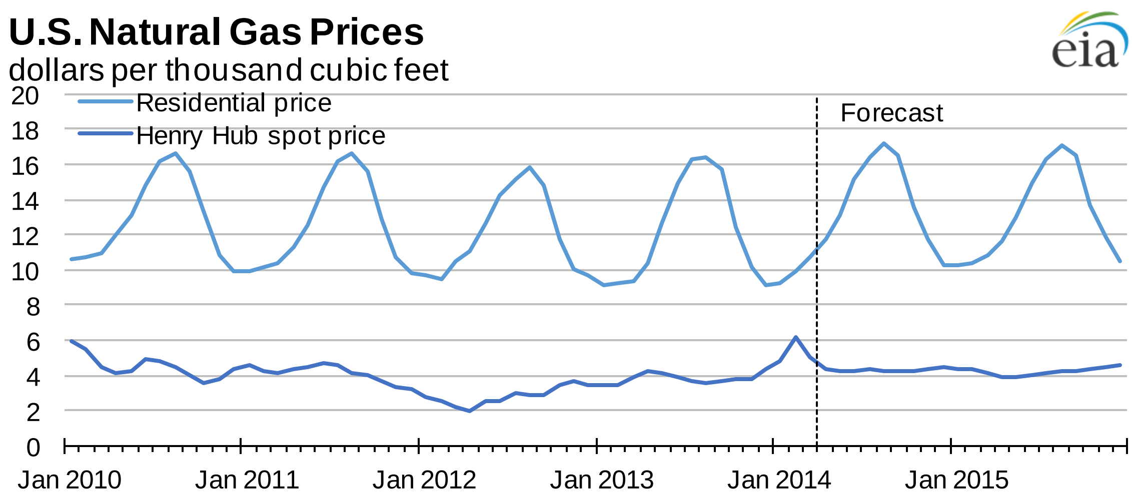


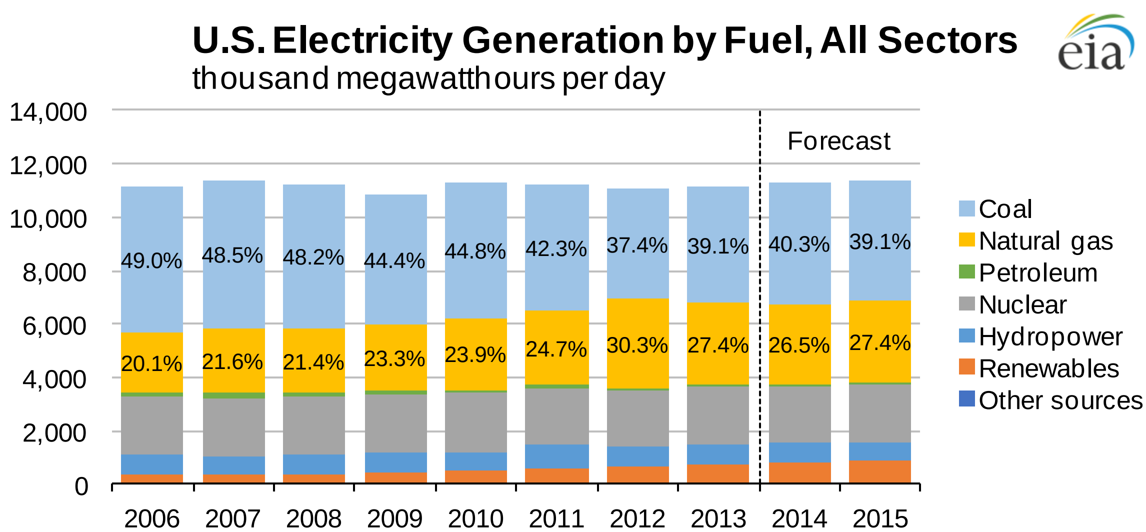

*Figure 1-1-Residential Electricity Price*




Source: Short-Term Energy Outlook, April 2014.

*Figure 1-2-U.S. Natural Gas Prices*

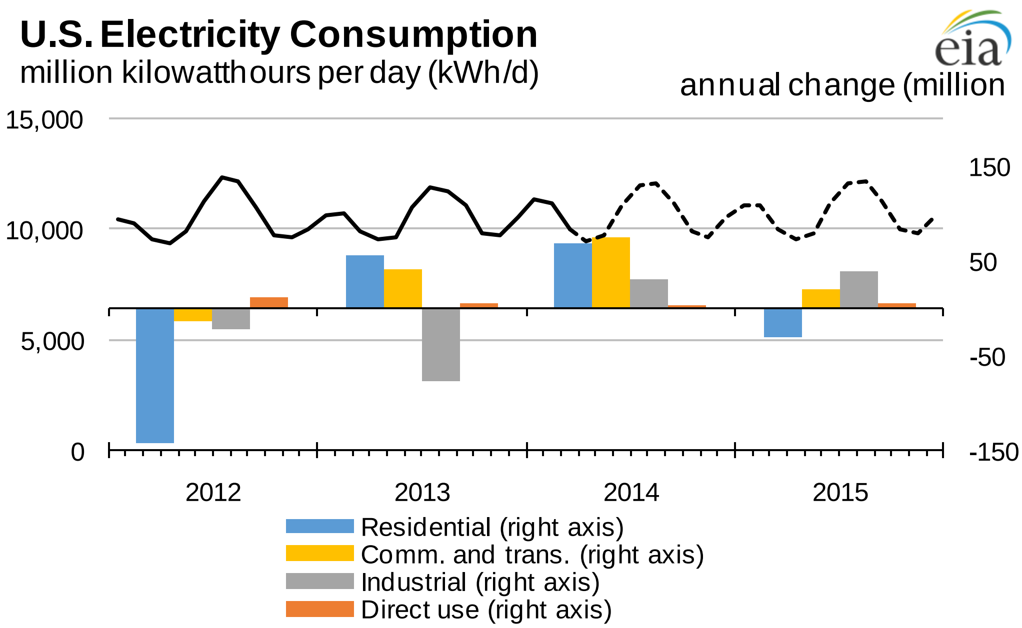
Coal as a fuel is still the highest contributor in the generation of electrical energy Fig 1.3. This is because of its availability and cheap cost. However, coal is not the cleanest source of energy and has higher environmental impacts as compared to the burning of other fuels such as Natural gas. The annual growth in the carbon dioxide emissions due to electrical energy production by burning coal is higher than any other fuel, refer Fig. 1.5.




Note: Labels show percentage share of total generation provided by coal and

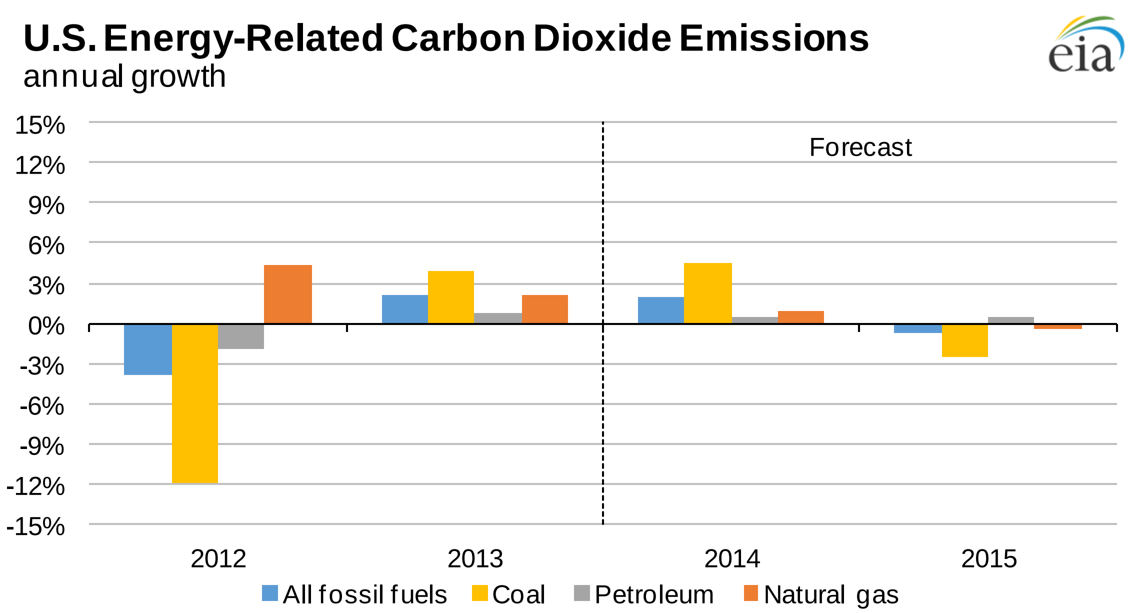
Source: Short-Term Energy Outlook, April 2014.

*Figure 1-3-U.S. Electricity Generation by Fuel*




Source: Short-Term Energy Outlook, April 2014.

*Figure 1-4-U.S. Electricity Consumption*




Source: Short-Term Energy Outlook, April 2014.

*Figure 1-5-U.S. Energy Related Carbon Dioxide Emissions*

The rise in electrical peak demand during the summer, increase in the electricity prices and pollution caused during the production of electrical energy, these all factors demand for an alternative technology which would take the load off the electrical grid. As described earlier, in the residential sector 42 % of the electrical energy consumption is for the heating, cooling and hot water. A Vuilleumier heat pump can be the futuristic alternative technology which would run completely off grid to supply the heating, cooling and hot water requirements in residential, commercial and industrial sectors.

## 1.2- Introduction to Vuilleumier Heat Pump Technology

**Vuilleumier Heat Pump:**

Vuilleumier heat pump (VHP) is heat engine driven heat pump (VHP) working on Vuilleumier cycle which was patented back in the year 1918 by Rudolph Vuilleumier. It is a type of a heat pump which can be used for heat and cryogenic temperature cooling at the same time.
Vuilleumier cycle is a closed cycle, i.e. the total volume of the gas in the cycle remains constant at all times. VHPs have three operating chambers and working fluid (usually Hydrogen or Helium) moves in between these three chambers in a sequential manner.
The cycle works between three discrete temperature zones viz. Hot, Warm and Cold. The working gas captures heat in the hot chamber and cold chamber and dissipates heat to the atmosphere at the warm chamber. Fig 1.6 is a schematic diagram of the Vuilleumier heat pump. Hot, Warm and Cold chambers are interconnected to each other. VHPs have two displacers which move in different phase angles and move the working gas along three different chambers viz. Hot, Warm and Cold. Here the displacers function differently than a piston in an Internal Combustion Engine. The main function of the displacer is to help gas move from one chamber to another. Displacers do not play a role in compression of gas inside the system. The displacer and the cylinder liner should make a perfect contact with each other so as to avoid any bypass of working gas from one chamber to another.

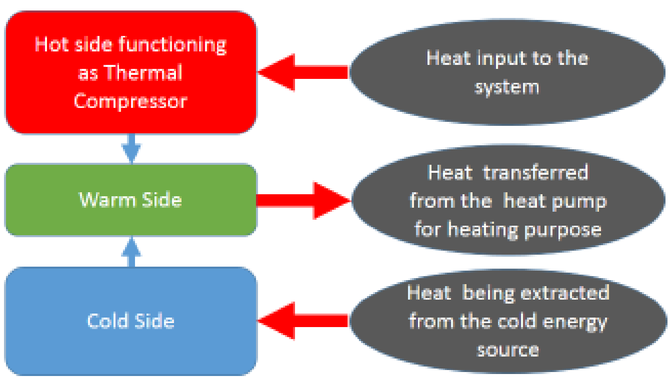


*Figure 1-6- Schematic Diagram of VHP*

Since the three chambers are at very different temperature ranges, the displacers should also be long and well insulated so that they allow minimal heat to migrate from one chamber to another. This can be achieved by using materials with low thermal conductivity for the displacers.
Vuilleumier heat cycle is somewhat similar to the sterling cycle. Modifications can be made to the Vuilleumier heat pump and the displacers may or may not be linked with each other thus making it different than a sterling cycle. Further the Vuilleumier heat pump can be modified to run using electromechanical system mechanism.

Thermal regenerators or the recuperators are the critical components of the VHPs. Thermal regernator are porous structures which ideally should provide minimum resistance to the flow of the working fluid and at the same time capture maximum heat energy from the working fluid.

There are different types of regenerators available Balls, stacked screens and wool made up of Stainless Steel, Monel, Bronze, Copper or Lead.

In Vuilleumier heat pump the three chambers are separated by two regenerators. The gas moving between the chambers has to pass through these thermal regenerators.

The working gas which is heated up and is moving from hot chamber to warm chamber passes through hot regenerator and deposits heat at the hot regenerator. Similarly gas moving from warm chamber to cold chamber deposits heat at the cold regenerator. While the gas returns back from cold chamber to warm chamber it recollects the deposited heat energy from the regenerator and similarly for the gas moving from warm to hot chamber, collects heat energy from the hot chamber.

The regenerators have temperature gradient along the line of the gas flow. The more the temperature gradient the better is the performance of the regenerators.

## 1.3- Motivation to develop efficient heat exchangers for VHP

The function of Vuilleumier heat pump is to move heat from the cold region to the intermediate temperature region. Vuilleumier cycle is a closed cycle where the working fluid should interact with other external fluid without mixing, so as to transfer heat in and out of the system. This heat transfer is achieved using heat exchangers. Heat exchangers are the critical components of the VHP's. Three discrete temperature regions in the VHP have three different heat exchangers operating under three different temperature conditions. The properties of the working fluid as well as the external fluid changes with the temperature and the pressure. The three heat exchangers used in Vuilleumier heat pumps can be classified as,

1. Hot heat exchanger
2. Warm heat exchanger
3. Cold heat exchanger

Vuilleumier heat pumps are highly sensitive to the flow restrictions. COP values of these heat pumps are significantly affected by the pressure drops in the working gas as it moves between different chambers of the machine. One of the other parameters which affects the performance of the VHP is the dead volume. Pressure drop and the dead volume are inversely related. The heat exchangers need to be designed appropriately so as to have minimal pressure losses while maintaining lower dead volumes.

This is a challenging a aspect in designing the heat exchangers for their applications in Vuilleumier heat pumps.

## 1.4-Objective and outline of thesis

Objective of this work is to device a technique to analyze a heat exchanger suitable to satisfy heat transfer requirements in the closed loop Vuilleumier heat pump. For cross flow heat exchanger design two participating fluids occupy all three dimensions. 3D CFD analysis becomes an integral part for the heat transfer analysis for these type of heat exchangers. 3D analysis is computational intensive.

In the Chapter 1, Vuilleumier heat pump concept has been introduced.

Chapter 2 discusses about the calculation of COP in determining the performance of the Vuilleumier heat pump. Different heat exchangers used in the Vuilleumier heat pump and their operating conditions has been mentioned here. This chapter also discusses the Theory of operation of Vuilleumier Heat Pump (VHP), explains in detail the components used in the VHP and their functioning. And also lists few of the advantages and disadvantages of the VHP.

Chapter 3 briefly introduces the different potential heat exchangers that may work with the VHP, gives the description of the problem and explains the reason behind selecting the model that has been analyzed in the subsequent chapters.

Chapter 4 presents the fundamentals of the COMSOL software and explains about different turbulent models that were considered and the one selected.

Chapter 5 presents the technique of analysis which is blend of analytical and numerical procedure to measure the performance of the heat exchanger. This chapter discusses about the evaluation of quantities

# Chapter 2 -LITERATURE REVIEW

## 2.1-COP calculation for VHP

The cooling COP is given as the ratio of amount of heat energy absorbed at the cold end to the amount the heat energy supplied at the hot end. The heating COP is the ratio of amount of heat energy dissipated at the atmosphere to the amount of energy supplied at the hot end.

The performance of the Vuilleumier heat pumps depend on several parameters like swept length of the displacers, cylinder volumes, operating temperature ranges, working fluid, efficiency of heat exchangers etc. However, the Carnot COP is a direct function of the temperatures in the three chambers.
COP heating value is usually greater than 1 as the heat energy absorbed by the pump at the cold end in addition to the heat energy supplied to the machine on the hot end is rejected at the ambient temperature condition.

$$Cooling\ COP = \frac{Heat\ Extracted\ from\ the\ atomsphere\ at\ lower\ temperature(Qcold)}{Heat\ supplied\ to\ the\ Vuilleumier\ Heat\ Pump(Qhot)}$$

$$Heating\ COP = \frac{Heat\ deposited\ to\ the\ atmosphere\ (Qwarm)}{Heat\ supplied\ to\ the\ Vuilleumier\ Heat\ Pump(Qhot)}$$

## 2.2-Different heat exchangers used in the Vuilleumier heat pump and their operating conditions.

As described earlier, there are three different temperature zones in the Vuilleumier heat viz, 1. Hot 2. Cold and 3. Warm.
These three chambers have 3 different heat exchangers working under different operating conditions.

**Hot heat Exchanger**: The heat exchanger in the hot zone is gas-gas heat exchanger. This heat exchanger on its outer surface captures heat energy from the radiation which is being emitted out from the natural gas burner and transmits that heat energy to the gas flowing inside the tubes. Operating temperature usually range is between 700-1200 ℃

**Warm Heat Exchanger**: Warm heat exchanger is a heat exchanger which operates in the intermediate temperature regions which range from 50-100℃. This heat exchanger is a gas-liquid heat exchanger which transfers heat energy from gas to the liquid which then is circulated to through the external heat exchanger for space heating.

**Cold Heat Exchanger:** Cold heat exchangers operate at a much lower temperature ranges. This is again gas-liquid heat exchanger but it transfer heat energy from liquid at higher to the gas inside the Vuilleumier heat pump.

## 2.3- Advantages and Disadvantages of Vuilleumier cycle.

The following are the advantages and disadvantages of the Vuilleumier cycle,

**Advantages:**

- Closed cycle with few moving parts, less amount of mechanical input energy required.
- External heat input to a closed cycle through a NOx free combustion in wire mesh burner thus is environmental friendly.
- Potential for high COP despite the disadvantages of the Vuilleumier cycle. Few moving parts and the low mechanical friction makes the obtainable COP very competitive for smaller heat driven heat pumps compared to the absorption type and the gas or diesel engine driven heat pump
- Low noise because of continuous heat supply. Electromechanical synchronization without gear mechanisms.
- Low mechanical forces on moving parts. Increased life.
- Only axial forces on the displacers due to the electromechanical system.
- Significant weight reduction due to careful selection of materials for the displacers.

**Disadvantages:**

- Cylinder volumes are not isothermal but nearly adiabatic.
- As the changes in the temperature in the cylinder volumes are proportional with the changes of pressure, the indicated COP will decrease with the increasing change of pressure over the cycle.
- For cooling COP, the temperature difference between the cylinder volumes is often smaller than the change of temperature in the cylinder volumes themselves, and the COP for cooling cycle is seriously reduced.
- Specific output is low.

## 2.4-Theory of Operation of Vuilleumier Heat Pump:

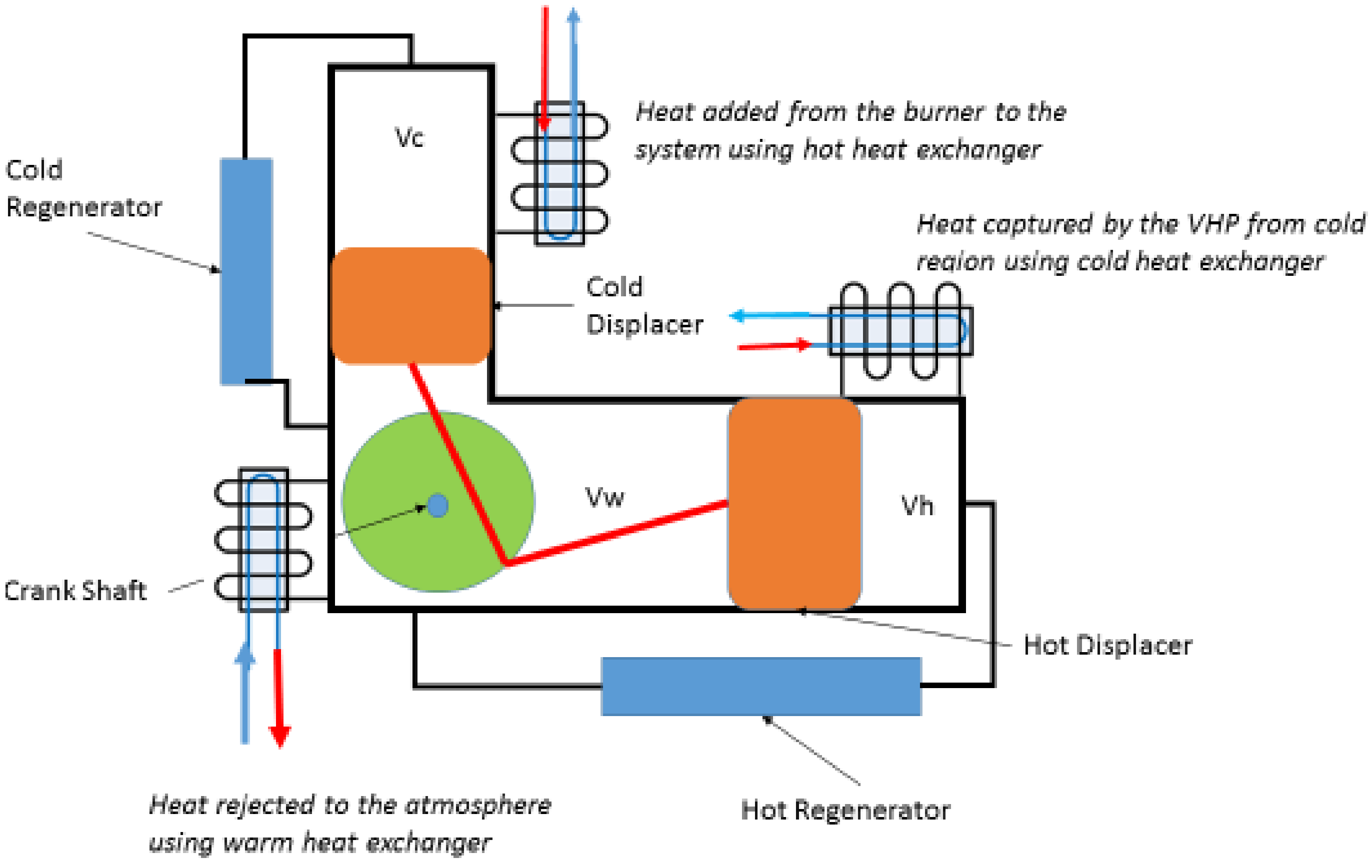



*Figure 2-1-Schematic Representation of VHP*

Fig. 2.1 is a schematic representation of the Vuilleumier heat pump. As described earlier the Vuilleumier heat pump primarily comprises of three volumes, hot (Vh), Warm (Vw) and Cold (Vc). These three volumes change dynamically with the motion of the displacers. We use the term 'active volume' to describe the volume swept by the displacers.

When the hot displacer moves towards its bottom dead center, the hot active volume increases. As heat is being constantly supplied to the hot chamber, the temperature of the gas in the hot chamber increases. This causes the pressure to increase in the hot chamber. The rise in pressure is reflected in all three chambers as the chambers are interconnected through the flow passages containing regenerators and heat exchangers. On the other hand, when the hot displacer travels to its top dead center the hot active volume decreases. This decreases the volume of the gas carrying heat energy at high temperature and thereby decreasing the pressure throughout the system. The motion of the hot displacer in the VHP causes this pressure in the entire system to fluctuate from peak to the minimum. Thus it acts as a thermal compressor.

Referring to the Fig. 2.2, as the hot displacer moves from its bottom dead center to the top dead center the high pressure high temperature gas causes compression of the gas in the warm chamber which is also being cooled by the warm heat exchanger. As the gas moves from hot chamber to the warm chamber it deposits heat energy to the thermal regenerator and exits the regenerator with a lower temperature. The gas is further cooled by the warm heat exchanger.

The cold displacer is then moved to its bottom dead center thus increasing the cold active volume. The gas in the warm chamber moves to the cold chamber after passing through the cold regenerator. Gas deposits heat at the cold thermal regenerator. When all the all the gas is moved to the cold active volume the hot displacer moves to the top dead center decreasing the pressure in the overall system. This causes expansion of the gas in the cold chamber further decreasing the temperature of the gas in the cold chamber. This gas is now at lower temperature and at lower pressure. This gas at low temperature now can pick up heat from the ambient heat source and thus can act as a refrigerator. After gas has absorbed the heat it and as the cold displacer moves from the bottom dead center to the top dead center, the gas moves from cold chamber to warm chamber after passing through the cold regenerator. At the cold regenerator, the gas picks up the heat which was deposited earlier and it rejects the heat at some intermediate temperature. After this cycle repeats itself.

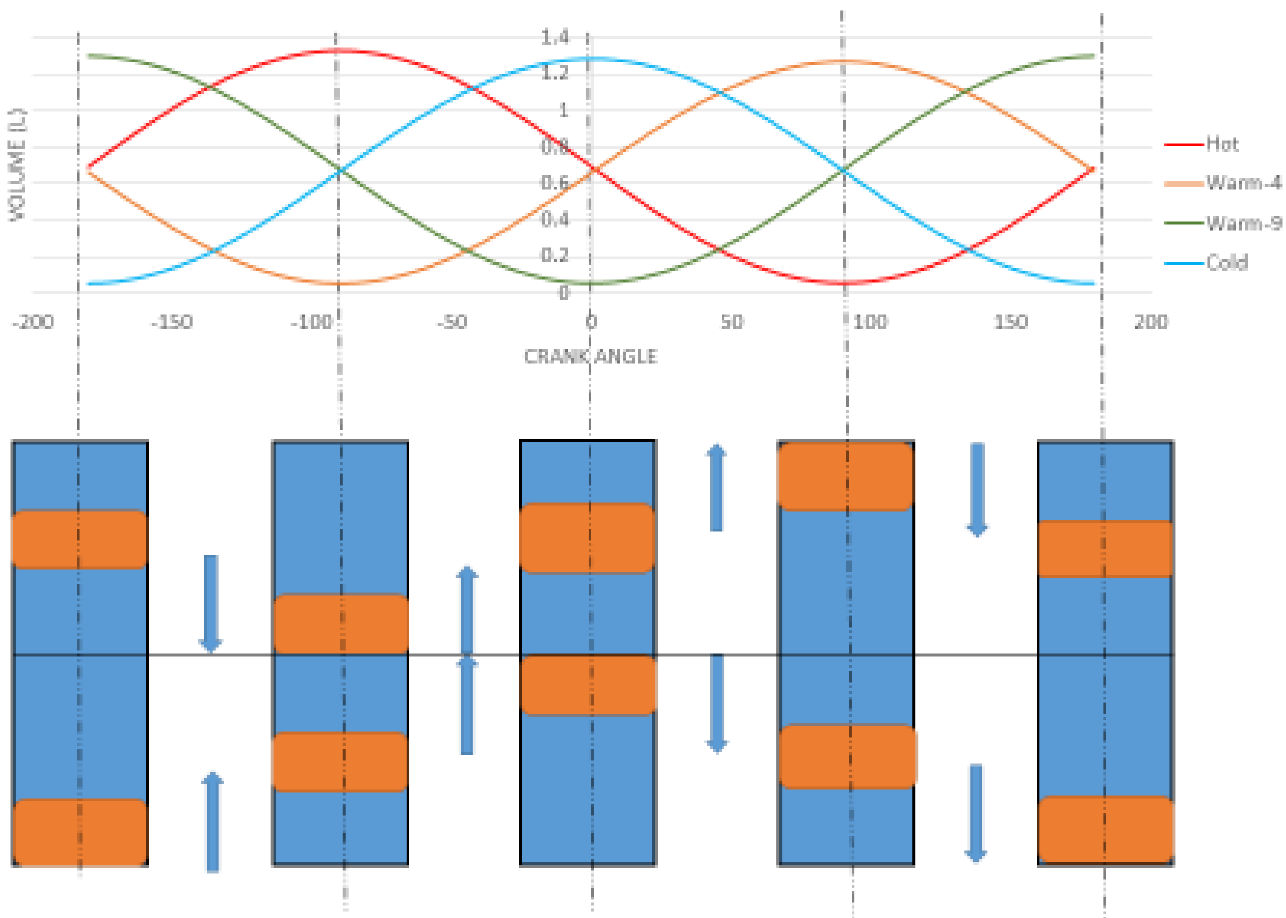



*Figure 2-2-Displacer Position*

As can be seen from the Fig. 2.2 the motion of the displacers are in 90 degrees out of phase where the hot displacer leads.

The Fig. 2.3 shows the Pressure Vs Volume diagram for combined three chambers.

GT-Power software was used to run the 1D thermodynamic cycle of the Vuilleumier heat pump. The three chambers in the Vuilleumier heat pump (Hot, Warm and Cold) can be also looked upon as four

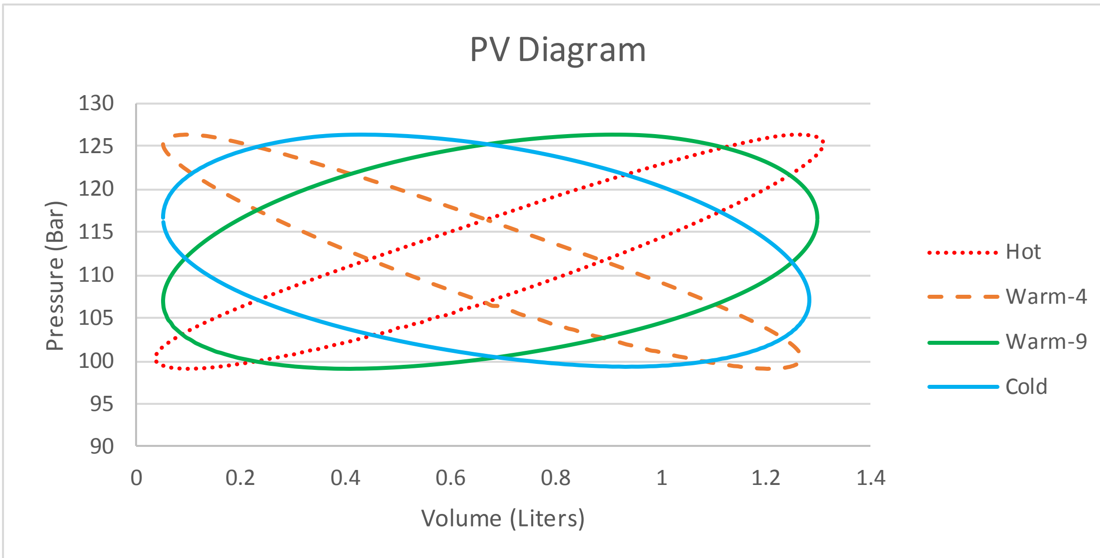



*Figure 2-3-PV Diagram*

volumes the warm volume comprising of two volumes, Warm-4 and Warm-9 as designated in the GT-Power map.

A pure sinusoidal waveform has been applied to the hot and cold displacers.

Various waveforms can be applied to define the dynamic position for the displacers. The conventional VHP can be modified to have electromechanical synchronization of the displacers instead of using gear mechanism.

As described earlier the function of the displacers is just to move gas between chambers and they play no role in compression of the gas inside the system. Here the displacers have to overcome minimal forces caused because of the frictional resistances on the gas as it passes through the regenerators and heat exchangers.

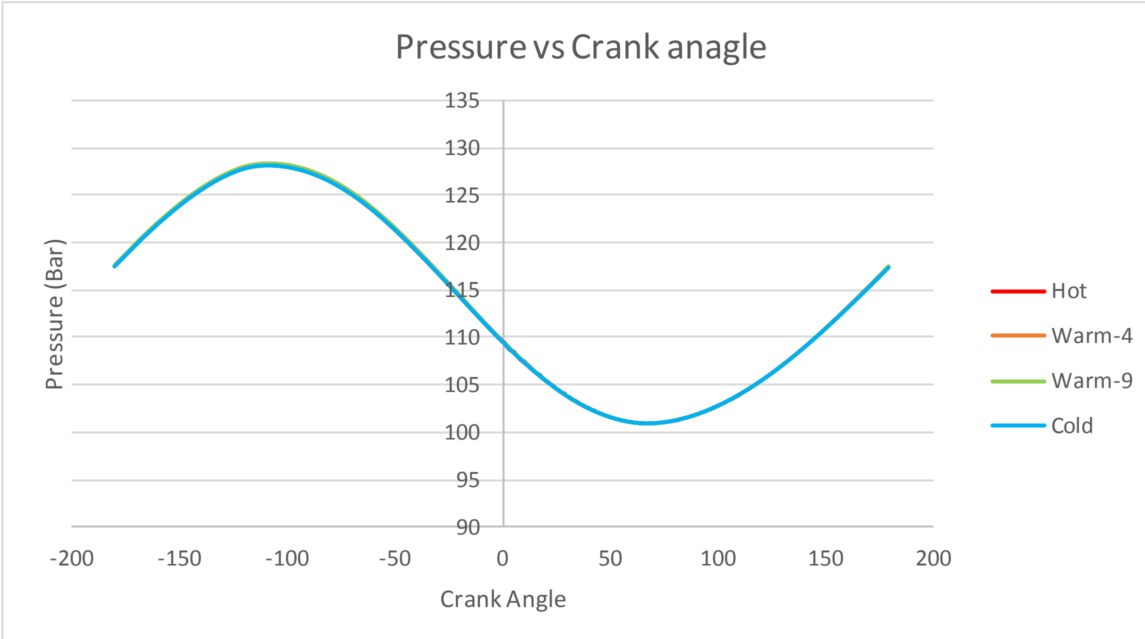



*Figure 2-4-Pressure Vs Crank Angle*

The diagram shown in the figure 2.4 shows the pressure variation in the chambers with the change in the crank angle. It can be observed that at particular crank angle pressure is almost the same in all the chambers.

# Chapter 3 -HEAT EXCHANGER THEORY

## 3.1- Different types of heat exchangers

**Shell and tube heat exchangers:**

Shell and tube heat exchanger are most commonly used heat exchanger. These type of heat exchangers carry several tubes which run across the heat exchanger carrying one of the two fluids which needs to be heated or cooled. Shell and tube heat exchangers are particularly used for high pressure applications as these types of heat exchangers are robust in their design.

The major parameters in these types of heat exchangers are:
1. Tube inner diameter and number of tubes: Tube inner diameter and number of tubes will influence the heat transfer coefficient values and the pressure drop of the fluid which flows inside these tubes.
2. Tube Thickness: The tube thickness should be appropriately selected so as have enough strength in tubes to handle the pressure. The tube thickness also influences the thermal resistance so the tube thickness should be minimal but at the same time should ensure enough strength to handle the pressure.

3. Tube Pitch: Tube pitch should be selected in away that the overall heat exchangers remain compact.

4. Tube Corrugation: Corrugated tubes have extended surfaces on the inside of the tubes which help in increasing the turbulence of the fluid flow and thereby increasing the heat transfer coefficient.

5. Tube layout: Tubes can be arranged in different layouts,
   i. Triangular
   ii. Rotated triangular
   iii. Square
   iv. Rotated square.

6. Baffles: Baffles create partition in the heat exchanger chamber and also help to minimize the vibration in tubes. They can also control the velocity of the fluid flow on the shell side. Thus the baffle positioning along with the tube layout is a critical parameter determining the heat transfer coefficient and pressure drop for the fluid flow on the shell side.

**Helical coil heat exchangers:**

Helical coil heat exchangers have one fluid flowing inside the helical coil which is circular or rectangular in the cross section while the other fluid flows over the outside surface of the helical tube. The mass transfer and heat transfer characteristics are enhanced by the Dean Vortex effect.

**Plate heat exchangers:**

Plate heat exchangers are compact heat exchangers where thin corrugated plates are stacked in parallel with each other. Plate heat exchangers have added advantage of having high surface area of contact for the two fluids participating in the heat transfer. The chamber created between the two parallel plates has minimal volume which allows good contact surface area for both of the fluids. These heat exchangers can have applications where the approach temperature is as low as 1 ℃ whereas shell and tube heat exchangers require approach temperature to be greater than 5 ℃ or more. Plate heat exchangers are customizable as we can increase and decrease the number of plates in this heat exchanger system.

## 3.2-Problem Description

Aim is to design and analyze a central heat exchanger also known as the warm heat exchanger which would be used in the Vuilleumier. This warm heat exchanger would transfer heat from the working fluid (Helium gas) to the water. This heat exchanger should be capable of transferring 24 kW of heat energy.

*Figure 3-1-Warm Heat Exchanger*

The Donut design of heat exchangers which is similar to the shell and tube heat exchanger shown in the Fig. 3.1 can be incorporated in the design of the Vuilleumier heat pump system quite readily. This type of annular heat exchanger design allows the space for passage of the shaft or the connecting rod of two displacers. This type of design is also easier to manufacture and has lower cost of manufacturing as compared to helical coiled tube and parallel plate heat exchangers. In the following part of the literature the thermal performance analysis technique of this warm heat exchanger is discussed. The following are the operating conditions:

| Fluids Participating in heat transfer | **Water** | | **Helium** | |
|---|---|---|---|---|
| Mass flow rate | 0.287 kg/s | | 0.59 kg/s | |
| Operating Temperature conditions | Tin | Tout | Tin | Tout |
| | 333 K | 353 K (Expected) | 373 K | Can be evaluated using energy balance |
| Operating Pressure conditions | 1.013 bar | | 120 bar | |

# Chapter 4 -CAE FUNDAMENTALS

## 4.1-Overview of COMSOL software and CFD turbulent models.

The CAD Import Module in the COMSOL software supports the import of a variety of formats of the CAD files like STEP and IGES. These file formats are supported by all CAD packages, and can be readily imported to the COMSOL Multiphysics by saving file in any of these formats.

All CAD files are automatically converted to a Parasolid geometry, using the Parasolid geometry engine that is included with the module. These geometries can subsequently be changed by a number of tools within COMSOL Multiphysics. This can include geometry repair or defeaturing (both explained below).

Producing CAD models is a robust, while not exact, science. The interfaces between the surfaces is may not be perfect in case of CAD files. The discontinuities in the surfaces make it difficult for meshingVery small and hardly noticeable anomalies can occur throughout the CAD model, creating non-physical objects or regions that can lead to difficulties when meshing within COMSOL Multiphysics. It is therefore required to repair the geometry before the simulation is started. COMSOL has an algorithm designed to repair the anomalies in the geometrical structure.

The CAD Import Module provides features that allow you to automatically detect and, either manually or automatically, repair your CAD models. While it may be preferable for the design engineer to perform some of the repair job in his or her CAD tool, the CAD Import Module will identify where these anomalies exist as they may not be immediately discernible from within the CAD program. Alternatively, you can manually select adjacent faces within the CAD Import Module and knit them together to form a solid, or specify tolerances in the import process, allowing the CAD Import Module to perform this automatically.

**Defeaturing Streamlines the Simulation Process**

After the repairing the geometry still the CAD file may have certain sub geometrical structures. Such structures take high amount of grid points after it has been meshed. For example spikes, thin wires all these structures which are extremely small size have to be meshed even finer. These structures are however might be irrelevant in the simulation phase. They may not contribute much to the physics of the entire geometry. Such small geometrical structures can be defeatured.

The COMSOL CAD module has the provision to define tolerances for these structures and they module can recognize and discard them from the overall geometry. This significantly saves time in the computation of the entire problem.

**Meshing Techniques:**

Tetrahedral meshing is used as a default while meshing any imported geometry on COMSOL platform.

The other three element types (bricks, prisms, and pyramids) should be used only when it is motivated to do so. It is first worth noting that these elements will not always be able to mesh a particular geometry. The meshing algorithm usually requires some more user input to create such a mesh. Usually at the interfaces between fluid and solid we select boundary layer mesh. This mesh type gives more grid points for computation in the regions where there is large gradient of velocity and temperature.

The element size can be selected on the basis of the material for example, finer mesh is preferred for the fluid as compared to mesh size for the solid. Also the regions of the interaction between the fluid and the wall should be given more grid points for improving the computational accuracy.

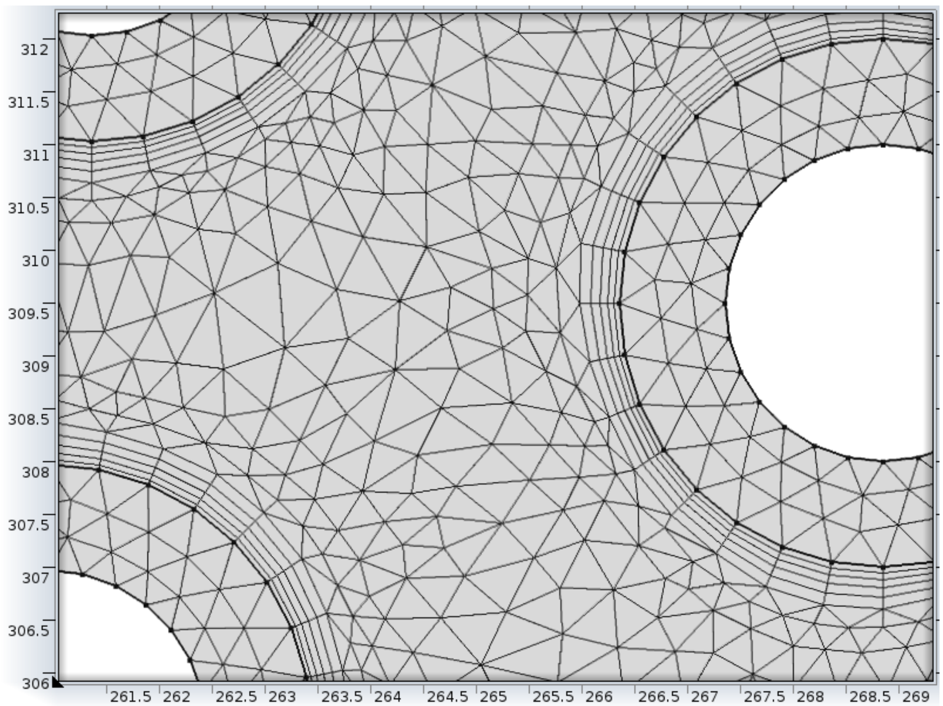



*Figure 4-1-Geometry Mesh*

## 4.2-Different CFD turbulent models considered:

Types of two equation models:

**K-epsilon ($K - \varepsilon$):**

K-epsilon model consist of two variables, k which defines the turbulent kinetic energy and epsilon is the rate of dissipation of kinetic energy. This is the most commonly used model while solving the flow problems with turbulence. Widely used commercially because of its faster convergence rates and relatively low memory requirements. This model however does not function well for high pressure gradient, strong curvatures to the flow or jet flow.

**K-omega model ($\boldsymbol{K-\omega}$):**

This model is similar to $(K-\varepsilon)$ model, however in the palace of rate of dissipation of energy omega is considered which the specific rate of dissipation of kinetic energy.

K-epsilon model is used first and tested for convergence because memory requirements are higher for the k-omega model as it also uses wall functions.

K-Omega model is however more accurate for internal flows, flows having strong curvatures.

**Low Reynolds number k-epsilon model ($\boldsymbol{K-\varepsilon}$):**

Similar to k-epsilon model but does not use wall functions it solves for the flow elsewhere.

**SST**

SST model is a combination of the k epsilon in the free stream and k omega models near the walls.

It does not use wall functions and tends to be most accurate when solving the flow near the wall.

SST model takes time to converge.

The flow through the geometry was initially tested using $(K-\varepsilon)$ model for turbulence however, due to the type of geometry which includes large number of curvatures, the $(K-\omega)$model was preffered.

$(K-\omega)$ turbulence model does solve for the wall functions and is usually preferred in internal flows as mentioned earlier.

# Chapter 5 -ANALYSIS TECHNIQUE

The hybrid analysis technique takes advantage of some of the empirical correlations which have been derived already and have been validated. Such empirical correlations can be used to find some of the missing parameters in solving the heat transfer problem.

In the shell and tube heat exchanger design, there are large number of tubes. These tubes carry helium gas which is compressible and interacts with another liquid which is water to transfer heat energy. The Nusselt number correlations can help in determining the heat transfer coefficient for the flow of the gas through these tubes. Further this heat transfer coefficient values can be given as the boundary conditions to the CFD model. So the requirement is to numerically analyze only one fluid participating in the heat transfer. This saves the memory requirement to solve the problem computationally. In this thesis, the heat transfer coefficient values for the flow of the helium gas has been calculated using analytical correlations as discussed in the following sub chapters. Several correlations were considered before down selecting a particular correlation. The pressure drop correlations are also available for flow of compressible and incompressible gases and liquids flowing through tubes.

## 5.1-Empirical relations for finding heat transfer coefficients for the gas flowing through tubes:

1. Dittus Boelter equation:

   Dittus Boelter equation (5.1) is in explicit form of finding Nusselt number as a function of Reynolds (5.2) and Prandtl number (5.3) of the fluid flowing through the tubes. This equation is simple equation mainly used to evaluate Nusselt number for the flow through smooth tubes. The equation gives fairly accurate results when there is

$$Nu_D = 0.023 Re_D^{4/5} Pr^n \tag{5.1}$$

Where,

$$Re_D = \rho U D/\mu \tag{5.2}$$

$$Pr = \mu Cp/k \tag{5.3}$$

Here,

$Nu_D$ is the Nusselt Number for internal diameter D,

$Re_D$ is the Reynolds number for the flow of gas through the pipe with diameter D.

$Pr$ is the Prandtl number of the fuild

$n$ is 0.3 for cooling of the fluid and 0.4 for heating of the fluid.

$Cp$ is specific heat at constant pressure for helium gas.(J/KgK)

$k$ is the thermal conductivity of the gas (W/mK)

$\rho$ is the density of fluid $(\frac{kg}{m^3})$

$U$ is the velocity of fluid flow (helium gas flow inlet velocity) (m/s)

2. Sider Tate Relations:

Sider Tate equation (5.4) is another form of empirical relation to find Nusselt number for fluid flow through tubes. This equation can be more accurate as compared to the Dittus Boelter equation if the temperature varies significantly for the fluid flowing through the tube. Sider-Tate equation has the term $\mu/\mu_s$ which takes into account the change in the viscosity of the fluid due to its temperature and pressure variations

$$Nu_D = 0.027 Re_D^{\frac{4}{5}} Pr^n (\frac{\mu}{\mu_s})^{0.14} \quad (5.4)$$

Where,

$\mu$ is the fluid viscosity at the bulk fluid temperature

$\mu_s$ is the fluid viscosity at the heat transfer boundary surface temperature.

Most of the analysis for determining heat transfer coefficient on the helium side has been done using Dittus Boelter equation as the temperature variation are not significant.

Dittus Boelter equation can be applied for the turbulent flow with Reynolds number extending from 2500 ≤ Re ≤ 1.24x10^5

The explicit form of Dittus Boelter is programmed to plot Heat transfer coefficient as a function of internal diameter and Number of tubes.

## 5.2-Pressure Drop calculation for the flow inside the cylindrical tubes:

The Weymouth, Panhandle A, and Panhandle B equations were developed to simulate compressible gas flow in long pipelines. The Weymouth is the oldest and most common of the three. It was developed in 1912. The Panhandle A was developed in the 1940s and Panhandle B in 1956 (GPSA, 1998). The equations were developed from the fundamental energy equation for compressible flow, but each has a special representation of the friction factor to allow the equations to be solved analytically. The Weymouth equation is the most common of the three - probably because it has been around the longest. The equations were developed for turbulent flow in long pipelines. For low flows, low pressures, or short pipes, they may not be applicable.

If the pressure drop in a pipeline is less than 40% of inlet pressure, then our Darcy-Weisbach incompressible flow calculation may be more accurate than the Weymouth or Panhandles for a short pipe or low flow. The Darcy-Weisbach incompressible method is valid for any flowrate, diameter, and pipe length. Crane (1988) states that if the pressure drop is less than 10% of pressure at the inlet and you use an incompressible model, then the gas density should be based on either the upstream or the downstream conditions. If the pressure drop is between 10% and 40%, then the density used in an incompressible flow method should be based on the average of the upstream and downstream conditions. If the pressure drop exceeds 40% of inlet pressure, then use a compressible model, like the Weymouth, Panhandle A, or Panhandle B.

Darcy Weisbach equation:

Darcy-Weisbach is an explicit form

$$\Delta p = f_D \left(\frac{L}{D}\right)\rho\frac{V^2}{2} \tag{5.5}$$

Here,

$\Delta p$ is the pressure drop for the fluid flow through pipe (Pa),

$f_D$ Darcy's friction factor,

L is the length of the tube,

D is the diameter of the tube,

$\rho$ is the density of the fluid (helium gas),

$V$ is the velocity of gas flow through each tube.

Darcy-Weisbach equation has Darcy frictional factor which needs to be determined either through Moody's diagram or by using explicit or implicit correlations which are described below:

Following are some equations for determining the Darcy's frictional factor

**1. Colebrook Equation:**

Colebrook equation (5.6) is an implicit form of equation widely used to determine the Darcy frictional factor.

$$\frac{1}{\sqrt{f_D}} = -2\, log_{10}(\frac{\in}{3.7 D_h} + \frac{2.51}{Re\sqrt{f_D}}) \tag{5.6}$$

Here,

$f_D$ is the Darcy friction factor

Roughness height, ∈ (m, ft)

Hydraulic diameter, $D_h$ (m, ft) – For fluid-filled, circular conduits, $D_{\mathrm{h}}$ = D = inside diameter

$Re$ is the Reynolds number.

**2. Blasius Equation:**

Blasius equation (5.7) is a simple solution for finding the Darcy Fricitonal factor which does not take into account the roughness of the pipe. This equation can be used when the

$$f_D = 0.079 Re^{-1/4} \tag{5.7}$$

**3. Halland Equation:**

Halland equation (5.8) is used to find the darcys fricional factor. It is quite accurate over 3000 Reynolds number. It is an explicit formula hence straight forward to compute. It is an approximation of the Colebrook equation.

$$\frac{1}{\sqrt{f_D}} = -1.8 \log_{10}[\left(\frac{\in/D}{3.7}\right)^{1.1} + \frac{6.9}{Re}] \tag{5.8}$$

Here,

$f_D$ is the Darcy friction factor

$\in/D$ is the relative roughness

$Re$ is the Reynolds number.

**4. Swamee-Jain:**

Swamee-Jain equation to determine Darcy's fricition factor is another approximation of Colebrook's equation.

$$f_D = 0.25[\log_{10}(\frac{\in}{3.7D_h} + \frac{5.74}{Re^{0.9}})]^{-2}$$
(5.9)

Where,

$f_D$ is a function of:

Roughness height, ε (m, ft)

Pipe diameter, $D$ (m, ft)

Reynolds number, $Re$ (unitless).

Halland equation has been preferred in this thesis as the Reynolds number for the flow ranges from 7000 to 1200. It is also an explicit equation hence easier to model in a numerical program and further the values obtained for the frictional factor can be used in the Darcy Weisbach equation (5.5).

## 5.3-Thermal resistance model

In shell and tube heat exchangers or for the Donut design which has been mentioned earlier in this thesis the exchange of heat energy between two fluids takes place by combined convection and conduction heat transfer. For the warm heat exchangers which are being analyzed in this work, the hot Helium gas heats up the interior walls of the heat exchanger by convection heat transfer. Heat energy then gets transferred through the walls through conduction and then water picks up the heat from the outer surface of the walls through convection heat transfer.

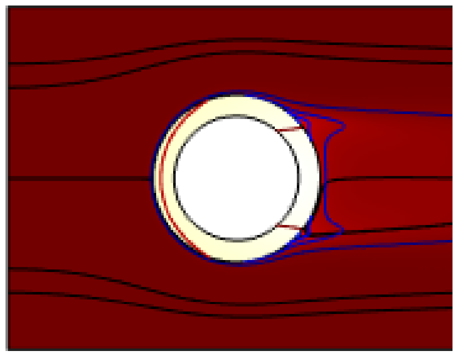


*Figure 5-1-Fluid Wall Interface*

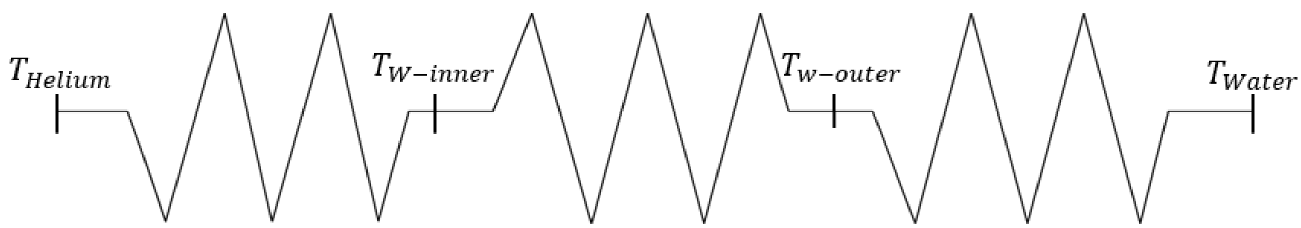



*Figure 5-2- Thermal Resistor Model*

Combined 2D and analytical calculation procedure can be brought down to 1D by building a thermal resistor model with three thermal resistances or heat flow barriers caused by convective heat transfer from the Helium gas to tube walls, Water to tube walls and conduction heat transfer across the tube wall. The primary objective to increase the heat transfer is to decrease these thermal resistances $(1/h_{He}A_{in})$, $(\frac{L}{A_{avg}\,K_{Wall}})$ and $(1/h_{H2O}A_{out})$. Here as the wall is very thin and for the purpose of finding the total averaged surface area $(\frac{L}{A_{avg}\,K_{W\,all}})$ has been approximated for thermal resistance of the wall instead of $[(\ln(\frac{r_2}{r_1}))/2\pi Lk]$ .Water would flow across the heat exchanger at a steady state however for helium steady state flow condition and the RMS value of the total mass flow rate has been considered. Heat transfer coefficient of helium is calculated using empirical relations and water side heat transfer coefficient has been calculated for varying inlet velocities using 2D CFD program. Resistor model equation suggest that the thermal resistance caused by the wall is far less insignificant as compared to the convective thermal resistances. The effects due to fouling of the tubes have been disregarded in this analysis and also the radiant heat transfer from the helium to the wall has been neglected for purpose of analysis.

## 5.4-Devising the equation to evaluate the heat transfer coefficient on the water side:

For calculating the average heat transfer coefficient on the water side, 2D simulations were run after supplying following boundary conditions to solve the problem,

1. Water inlet temperature :
2. Water inlet velocity
3. Temperature of the inner side of the tubes.
4. Heat transfer coefficient at the inner side of the tube

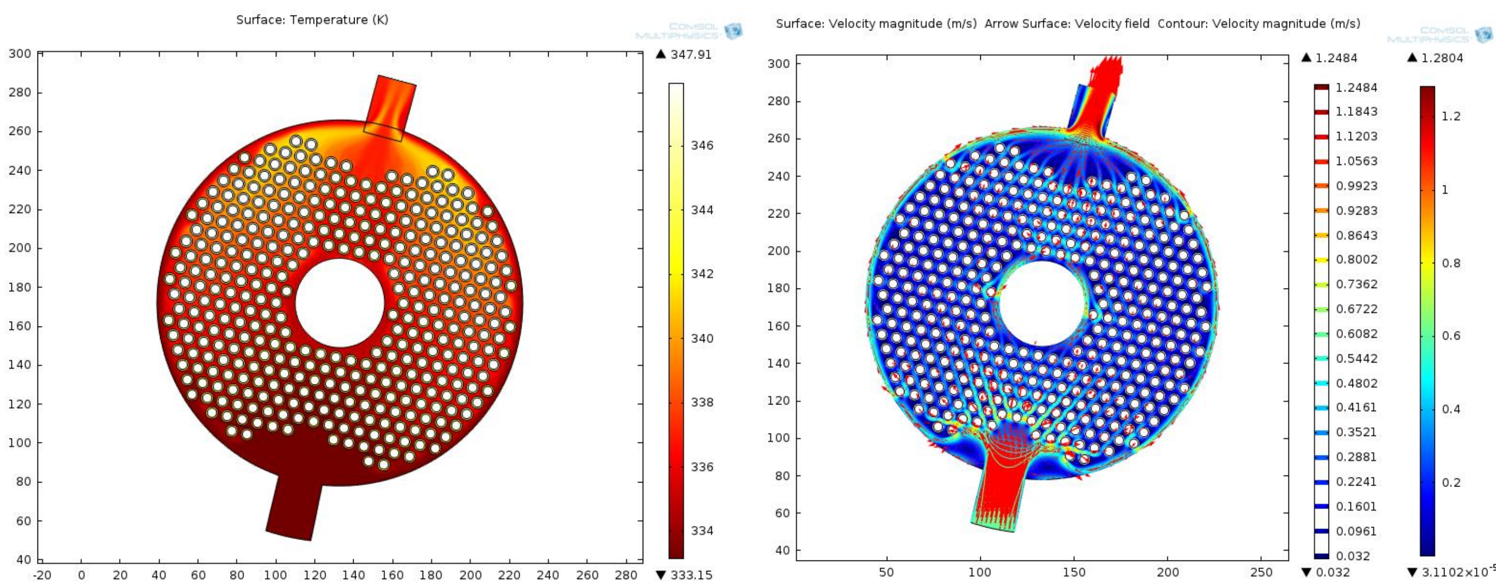



*Figure 5-3-Temperature and Velocity Contours*

Sequential parametric sweep operation has been performed to run the model for different inlet conditions. The following parameters were recorded which were further utilized for the purpose of analysis,

1. Temperature at the outlet.
2. Average surface temperature of the water.

As the steady state condition has been assumed, the first law of thermodynamics can be used to calculate the total amount of heat energy being picked up by the water after accounting for the temperature difference at the outlet and inlet.

$$\Delta Q = \dot{m} Cp_{H2O} \Delta T$$

Here,

$\Delta Q$ is the heat energy being picked up by the water,

$\dot{m}$ is the mass flow rate of water,

$Cp_{H2O}$ Specific heat at constant pressure for water,

$\Delta T$ is difference in temperature of outlet and inlet.

The energy equation can be rearranged to give an explicit equation for determining the heat transfer coefficient on the water side,

$$\Delta Q_{\text{Thermal Resistance}} = \frac{T_{He,avg} - T_{H2O,avg}}{\frac{1}{h_{He}A_{in}} + \frac{L}{K_{Cu}A_{avg}} + \frac{1}{h_{H2O}A_{out}}} \tag{5.10}$$

$$\dot{m} Cp_{H2O}(T_{out} - T_{in}) = \frac{T_{He,avg} - T_{H2O,avg}}{\frac{1}{h_{He}A_{in}} + \frac{L}{K_{Cu}A_{avg}} + \frac{1}{h_{H2O}A_{out}}}$$

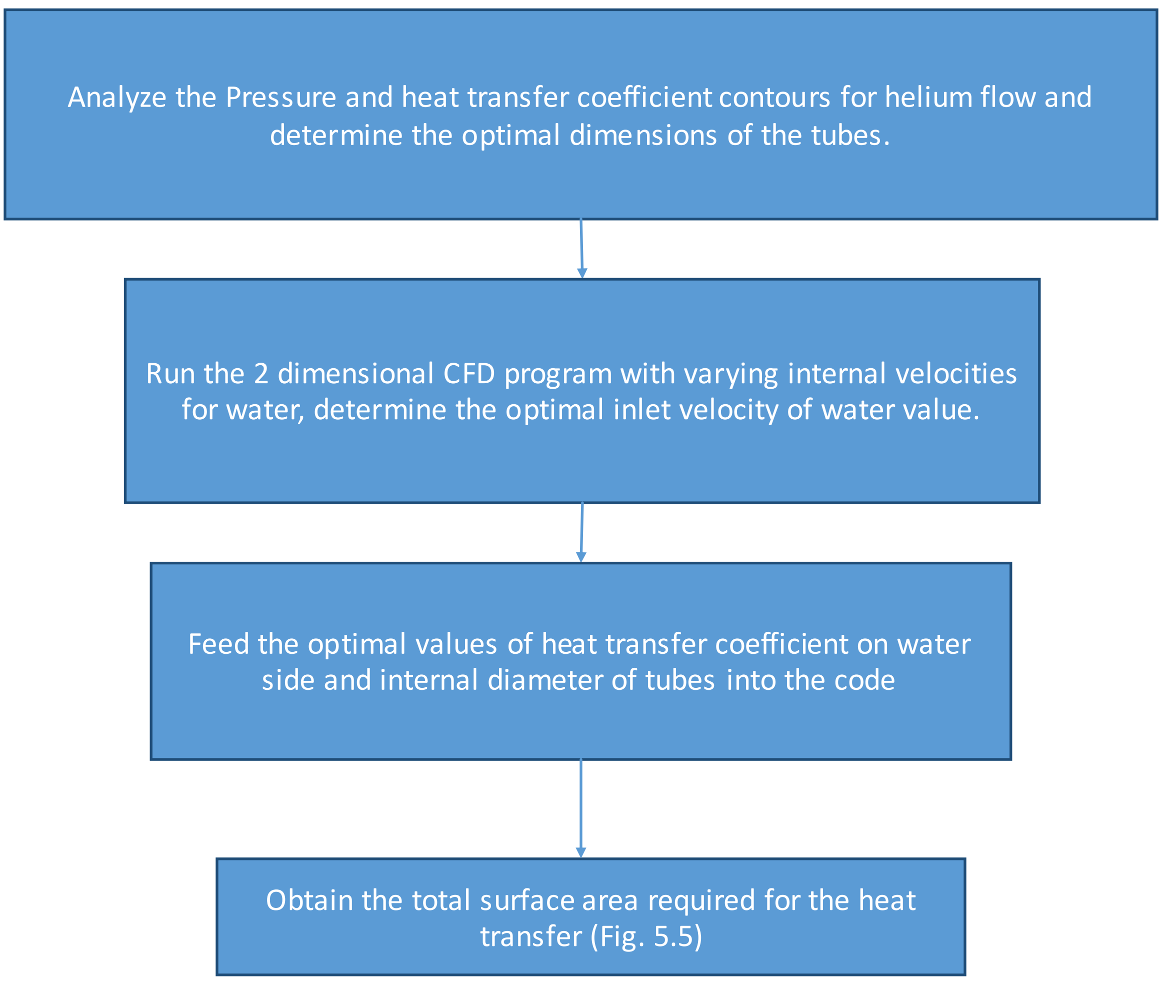
$$\frac{1}{h_{H2O}A_{out}} = \frac{(T_{He,avg} - T_{H2O,avg})}{[\dot{m} Cp_{H2O}(T_{out} - T_{in})]} - \left(\frac{1}{h_{He}A_{in}} + \frac{L}{K_{Cu}A_{avg}}\right)$$

$$\frac{1}{h_{H2O}} = A_{out}\left\{\frac{(T_{He,avg} - T_{H2O,avg})}{[\dot{m} Cp_{H2O}(T_{out} - T_{in})]} - \left(\frac{1}{h_{He}A_{in}} + \frac{L}{K_{Cu}A_{avg}}\right)\right\} \tag{5.11}$$

The equation (5.11) can be used to evaluate the heat transfer coefficient values for the water flowing across the heat exchanger.

## 5.5-Evaluation of the total average surface area required for the heat transfer:

A program has been developed which calculates the total surface area that is required for a specified heat transfer load requirement.



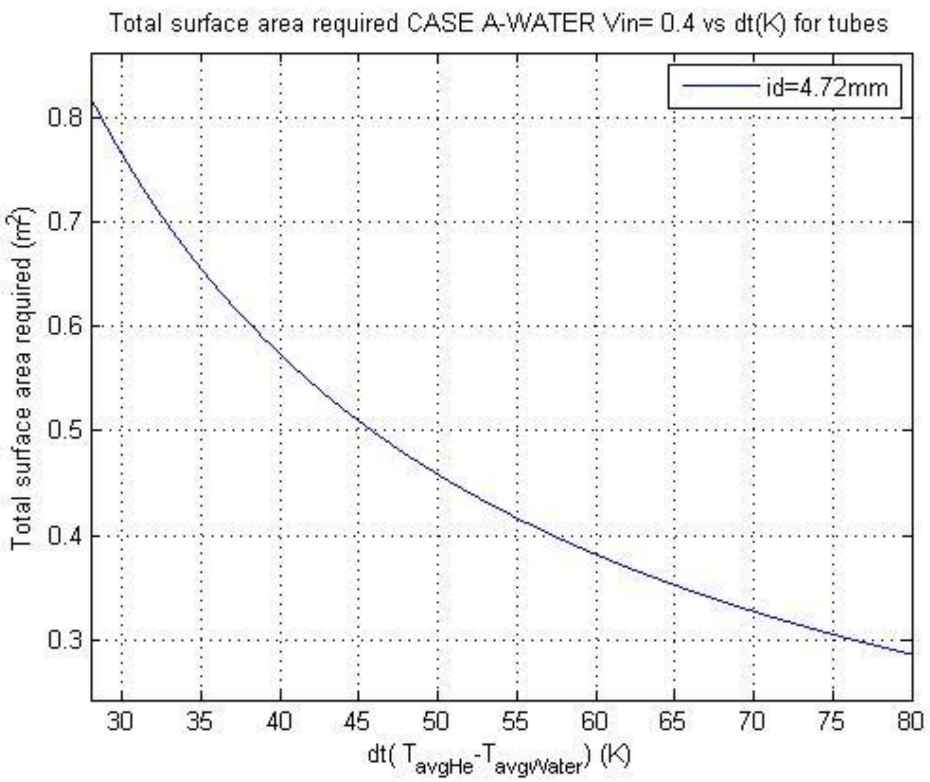



*Figure 5-4-Total Surface required plot*

The graph in Fig.5.5 shows the averaged surface area required to achieve the required amount of heat transfer. The heat transfer coefficients on the helium and water side are the inputs provided to the code which uses the equation (5.11) to calculate the averaged surface area required for the heat trasnfer.

$$\Delta Q = \frac{T_{He,avg} - T_{H2O,avg}}{\frac{1}{h_{He}A_{avg}} + \frac{L}{K_{Cu}A_{avg}} + \frac{1}{h_{H2O}A_{avg}}}$$

$$\Delta Q = \frac{T_{He,avg} - T_{H2O,avg}}{\frac{1}{A_{avg}}\left(\frac{1}{h_{He}} + \frac{L}{K_{Cu}} + \frac{1}{h_{H2O}}\right)}$$

$$A_{avg} = \frac{\Delta Q\left(\frac{1}{h_{He}} + \frac{L}{K_{Cu}} + \frac{1}{h_{H2O}}\right)}{T_{He,avg} - T_{H2O,avg}} \tag{5.11}$$

### 5.6-Surface area and the Dead volume:

The surface area available for the heat transfer can be varied by varying,

1. Number of tubes
2. Tube internal and external diameter
3. Length of the tubes

The code evaluates the total surface area available for varying internal diameters and varying lengths of the tubes.

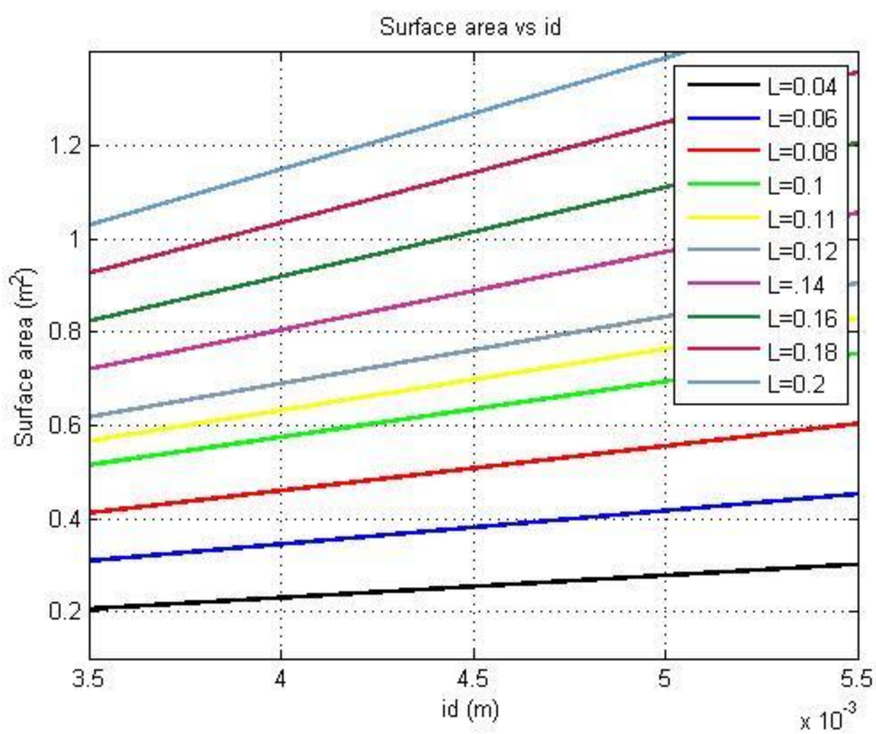



*Figure 5-5-Surface Area Available Plot for different lengths and ID*

Once the optimal value of internal diameter has been selected, the length can be determined from graph which would satisfy the surface area requirement.

**Dead Volume:**

In an idealized system it is assumed that there is no dead volume, i.e. all the volume in the VHP is the active volume present in Hot, Warm and Cold chambers of the VHP. The dead volume as earlier discussed has significant influence on the performance of the Vuilleumier heat pump. Heat exchangers, regenerators and the flow passages comprise of significant amount of dead volume present in the VHP

The parameters that influence the dead volume in the heat exchangers are,

1. Number of tubes.
2. Cross sectional area of tubes.
3. Length of tubes.

## Chapter 6 RESULTS AND DISCUSSION

### 6.1-Heat transfer coefficient variations for the Helium gas as a function of internal diameter and number of tubes:

As described earlier chapter, the heat transfer coefficient for the gas flow can be evaluated using the Ditus boelter equation

$$Nu_D = 0.023 Re_D^{4/5} Pr^n \tag{6.1}$$

$$Re_D = \rho U D/\mu$$

$$Nu_D = hD/k$$

$$\left(\frac{hD}{k}\right) = 0.023 Re_D^{4/5} Pr^n$$

$$h = (k * 0.023 \left(\frac{\rho U D}{\mu}\right)^{4/5} Pr^n)/D$$

$$h = (k * 0.023 \left(\frac{\rho U}{\mu}\right)^{\frac{4}{5}} Pr^n)/D^{1/5} \tag{6.2}$$

Here,

$Nu_D$ is the Nusselt Number for internal diameter D,

$Re_D$ is the Reynolds number for the flow of gas through the pipe with diameter D.

$Pr$ is the Prandtl number of the fuild

$n$ is 0.3 for cooling of the fluid and 0.4 for heating of the fluid.

$Cp$ is specific heat at constant pressure for helium gas.(J/KgK)

$k$ is the thermal conductivity of the gas (W/mK)

$\rho$ is the density of fluid $(\frac{kg}{m^3})$

$U$ is the velocity of fluid flow (helium gas flow inlet velocity) (m/s)

For the Donut design of heat exchanger model the mass flow rate through each tube will be $\dot{m}/n$, where $\dot{m}$ is the total cycle averaged mass flow rate of the gas and n denoting the total number of tubes.

As it can be seen from the equation 6.2 the heat transfer coefficient increases with the inlet velocity of gas and decreases with increase in the internal diameter. Velocity of the flow of the gas is influenced by the number of tubes.

The explicit formula is used to plot the heat transfer coefficient as a function of the internal diameter and total number of tubes.

It can be observed from the Fig. 6.2, there is a steep rise in the heat transfer coefficient values as internal diameter and number of tubes decrease. Also it can be noted that the intermediate higher values of the heat transfer coefficients can be attained with the combination of larger number of tubes with smaller internal diameters and smaller number tubes with larger diameter

| Case | Number of tubes | Internal Diameter (m) | Heat Transfer Coefficient (W/m^2K) | Pressure Drop (Pascal) | Pressure Drop (kPa) | Total surface area ($m^2$) | Dead Volume (Liters) |
|---|---|---|---|---|---|---|---|
| 1 | 4000 | 0.001 | 5500 | 952 | 0.952 | 0.753 | 0.188 |
| 2 | 3000 | 0.0012 | 5512 | 847 | 0.847 | 0.678 | 0.203 |
| 3 | 1920 | 0.0014 | 5502 | 680 | 0.68 | 0.506 | 0.177 |
| 4 | 373 | 0.003 | 5496 | 497 | 0.497 | 0.210 | 0.158 |
| 5 | 108 | 0.005 | 5513 | 412 | 0.412 | 0.101 | 0.127 |

*Table 6-1-Case Study*

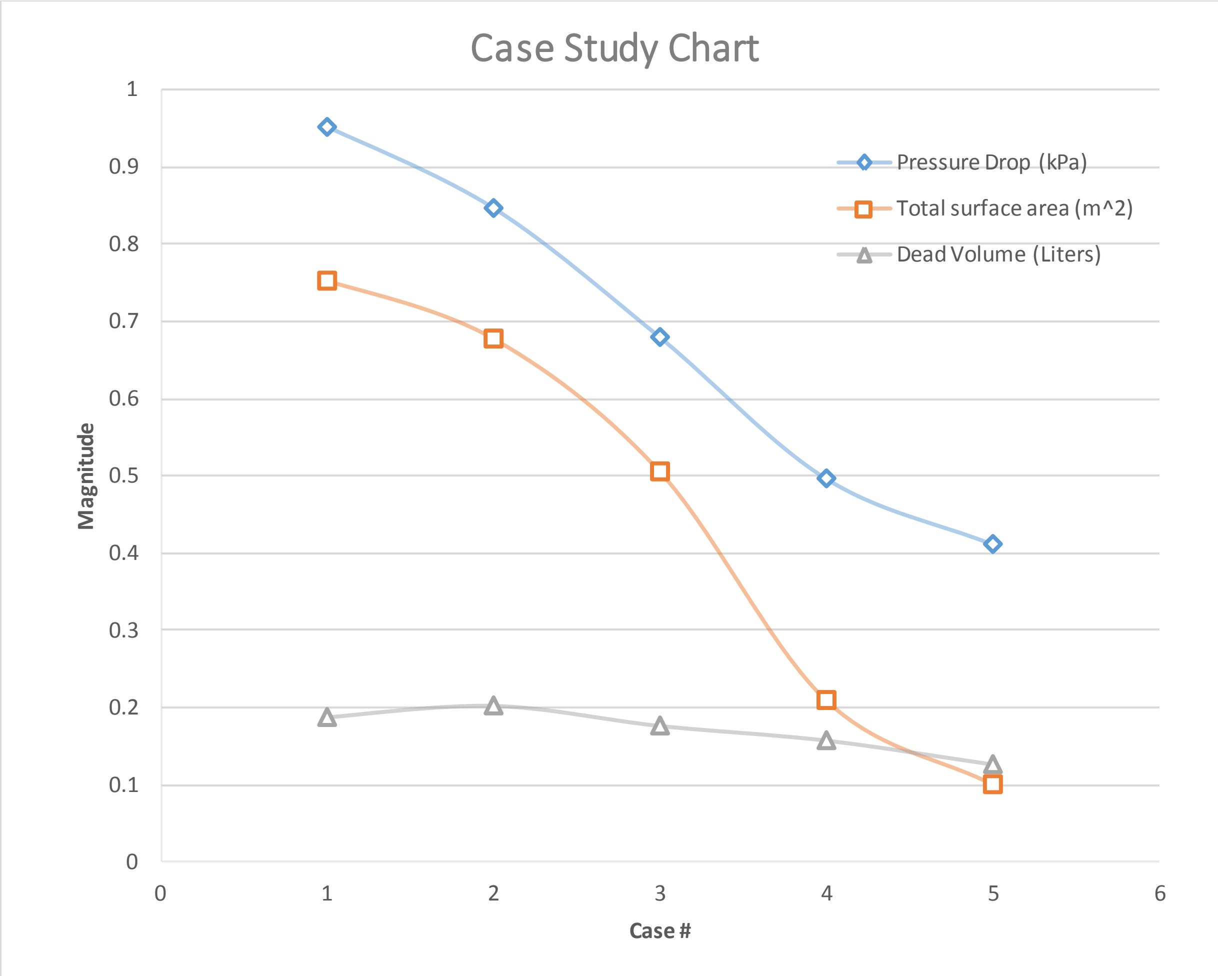



*Figure 6-1- Case Study Chart*

Five different cases for different configurations of number of tubes and internal diamteres have been anayzed in this case study. 5 different configurations have been selected in a way that they provide the same heat transfer coefficeint on the inside wall (helium side). The objective of this study is to compare the surface area, pressure drop and the dead volume against the cases. This would help to acknowledge the advantage of having larger number of tubes but with smaller diameter. All the cases have been compared keeping the length of the tubes as a constant parameter. Looking at Fig. 6.1, it can be observed that for same heat transfer coefficent without loosing much on the pressure drop and the dead volume higher surface areas can be attained for larger number of tubes with smaller diameters. The added advantage of having higher suface area could result in reduction in the required length for heat transfer. This suggest that the heat exhanger overall size can be further reduced thus reducing the dead volume and pressure drop which are linear functions of the length.

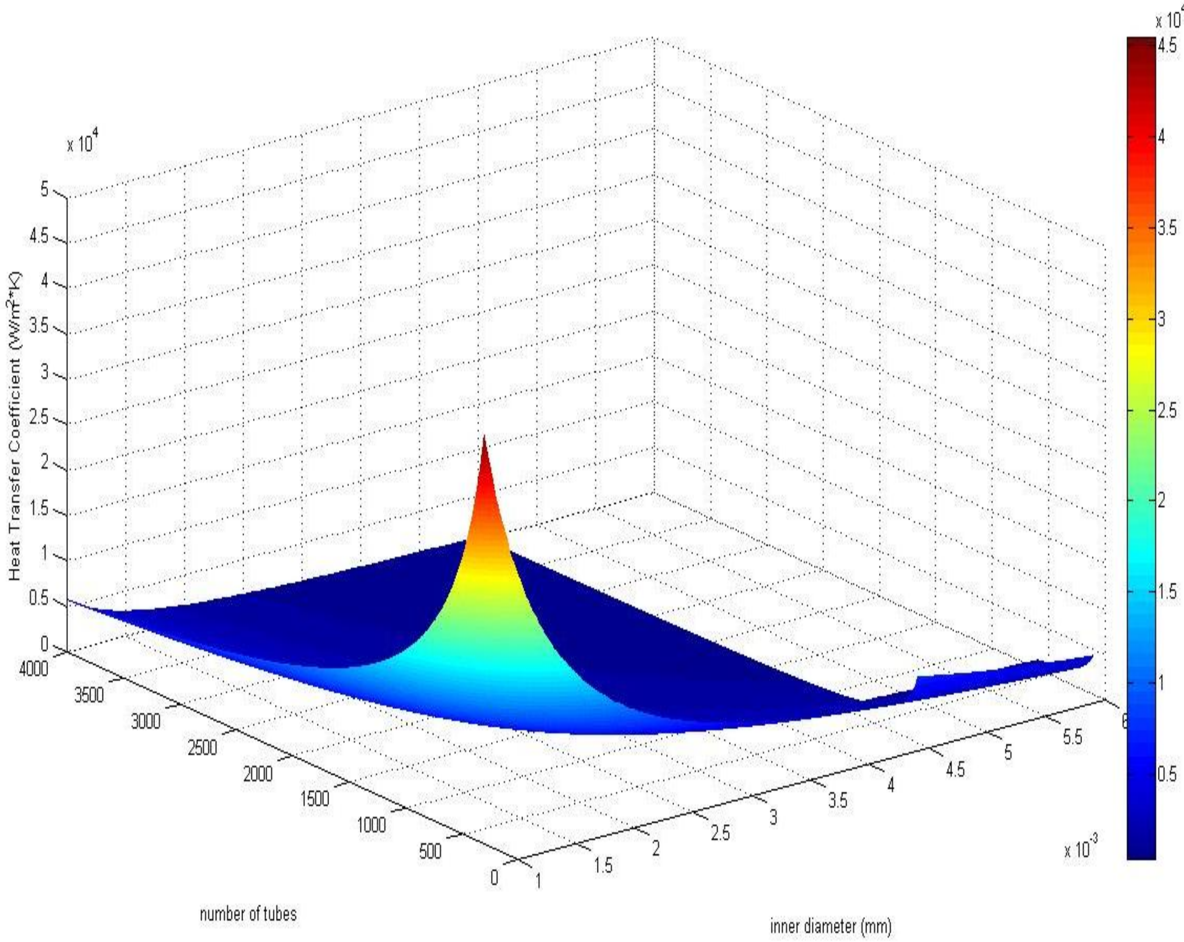



*Figure 6-2-Heat Transfer Coefficient on Helium Side.*

**Pressure Drop:**

In every heat exchanger model there always exists a tradeoff between heat transfer coefficient and pressure drop. Before determining the best suitable value for the internal diameter of the pipes which would give maximum heat transfer coefficient it is always recommended to check the pressure drop that the particular configuration would offer. It is usually a synergy of all the factors which influence the overall performance of the machine and the pressure drop is one of the critical factor which influences the performance of the machine invariably. Thus it is advised to consider the optimal value of the pressure drop while sizing the heat exchanger tubes. A program developed in was in which would evaluate the pressure drop as a function of number of tubes and internal diameter.

Pressure drop as earlier described is a function of Reynolds number for flow through each tube and length of tubes. 3D plots in help to determine the regions for certain configurations with relatively lower pressure drop values and further helps to determine the dimensions and the number of the pipes that can be considered for further investigating the heat transfer coefficients.

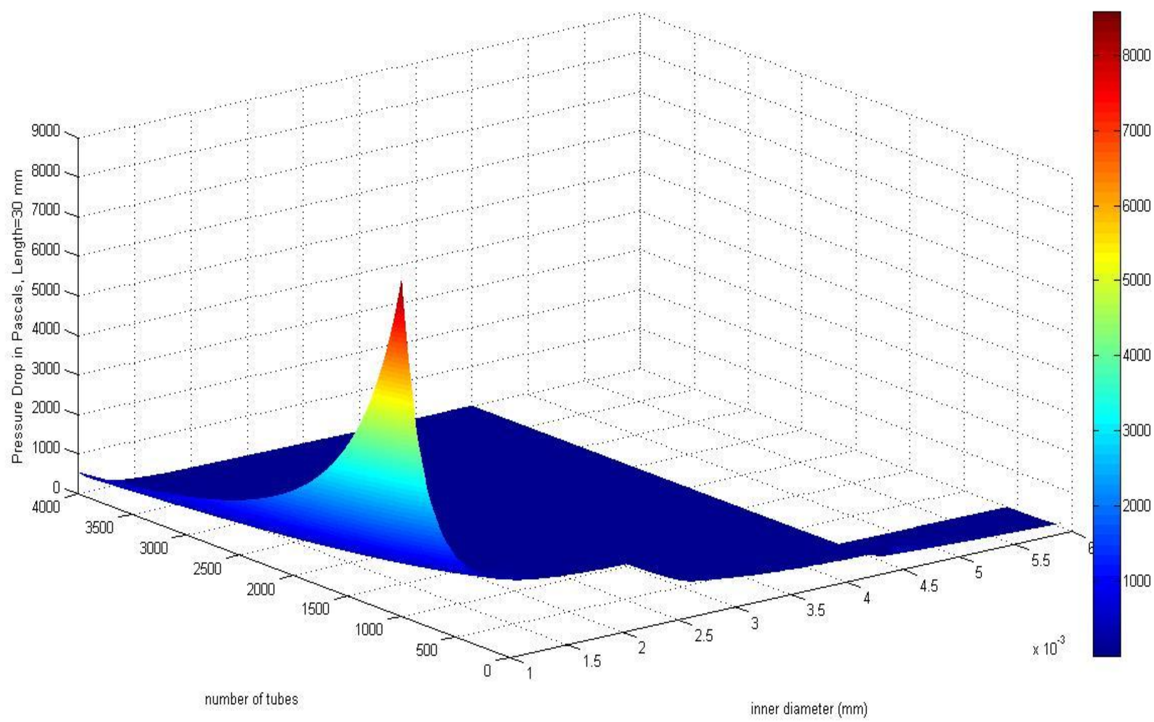



*Figure 6-3-Pressure Drop L=30 mm*

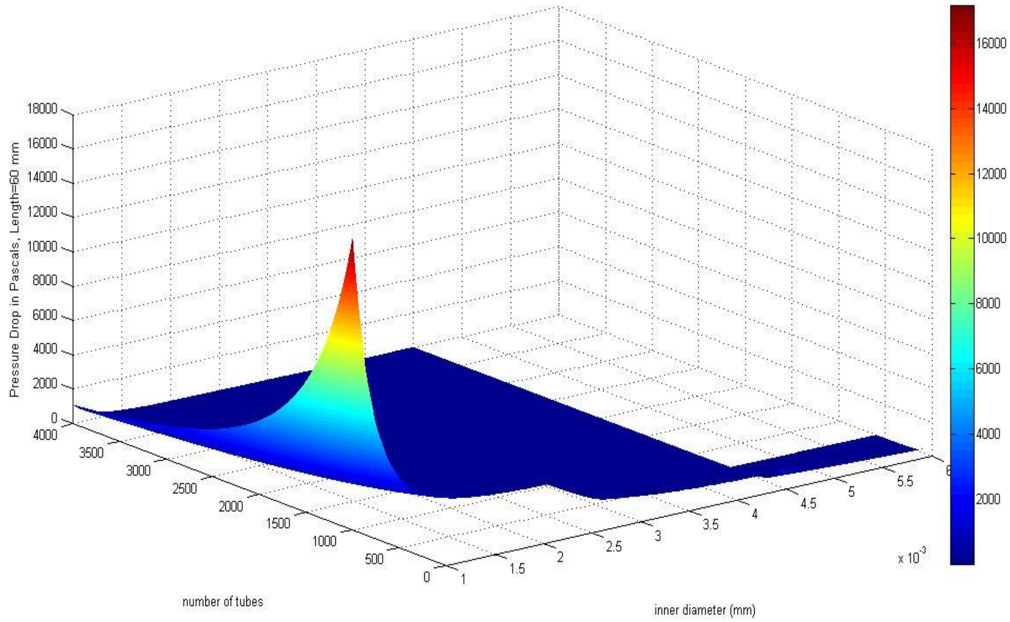



*Figure 6-4-Pressure Drop L=60 mm*

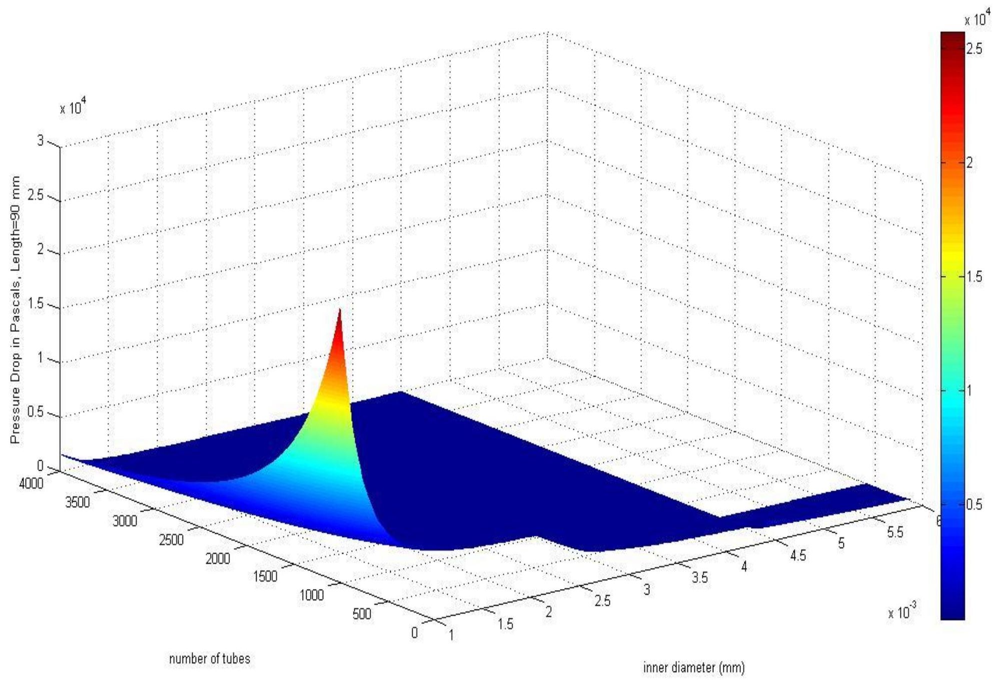



*Figure 6-5-Pressure Drop L=90 mm*

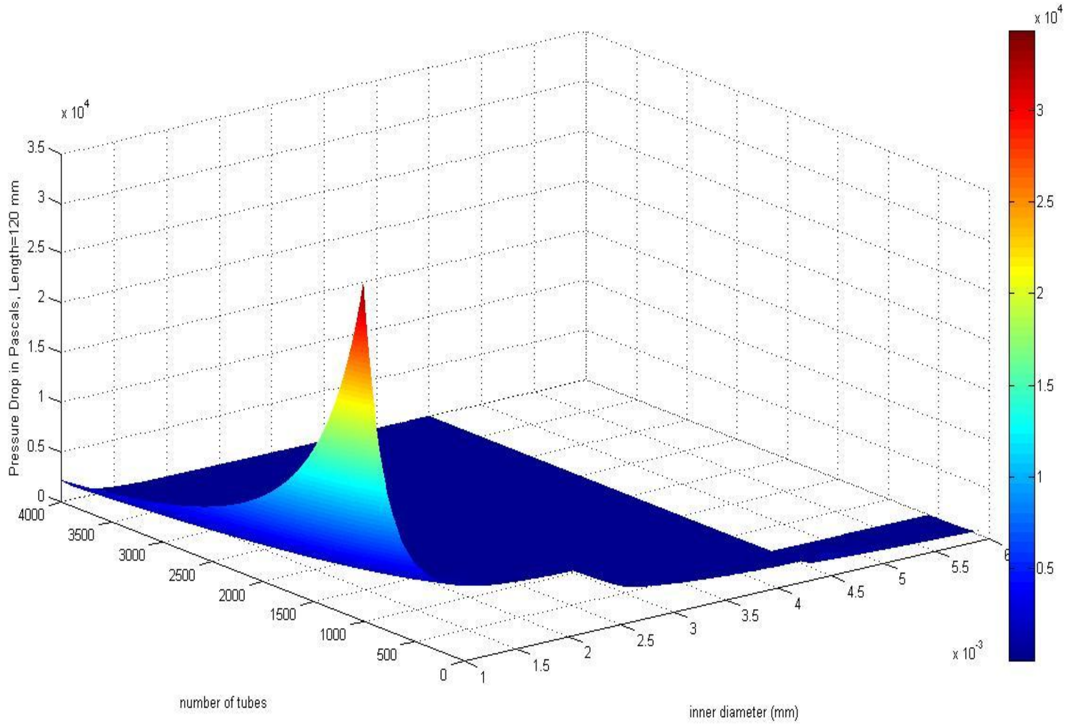



*Figure 6-6-Pressure Drop L=120 mm*

The pressure drop is the linear function of length. A limiting value for the pressure drop can be set and corresponding configuration can be decided. Again, it can be observed from the plots above that the configuration with larger number of tubes with smaller ID which offered good heat transfer coefficients in addition to larger surface area can be favored. The advantage of having higher surface areas for the heat transfer can further reduce the total length required for the heat transfer which has been proven beneficial to decrease the pressure drop.

## 6.2 - Comparison of two different models, surface area requirement evaluation procedures:

In this chapter the model evaluation procedure is discussed considering two different cases.

Two different configurations (Case A and Case B) of the donut type heat exchanger are discussed below:

| Case | Number of tubes | Internal diameter of tubes (mm) | Outer diameter of tubes (mm) |
|---|---|---|---|
| A | 380 | 4.72 | 6.35 |
| B | 360 | 3 | 5 |

*Table 6-1-Comparison of two models*

**Case A:**

As discussed in the earlier chapter, the following steps will followed to analyze the performance and evaluate the total surface area required for the 24 kW of heat transfer.

Step 1: Evaluating the heat transfer coefficient on the helium side:

A code has been written to evaluate the explicit form of the Dittus Boelter equation which finds the Nusselt number and the heat transfer coefficient for the gas flowing insider the cylinder. The equation takes the input of the parameters like the internal diameter, number of tubes, averaged mass flow rate of the gas flowing inside the tubes and the gas inlet temperature.

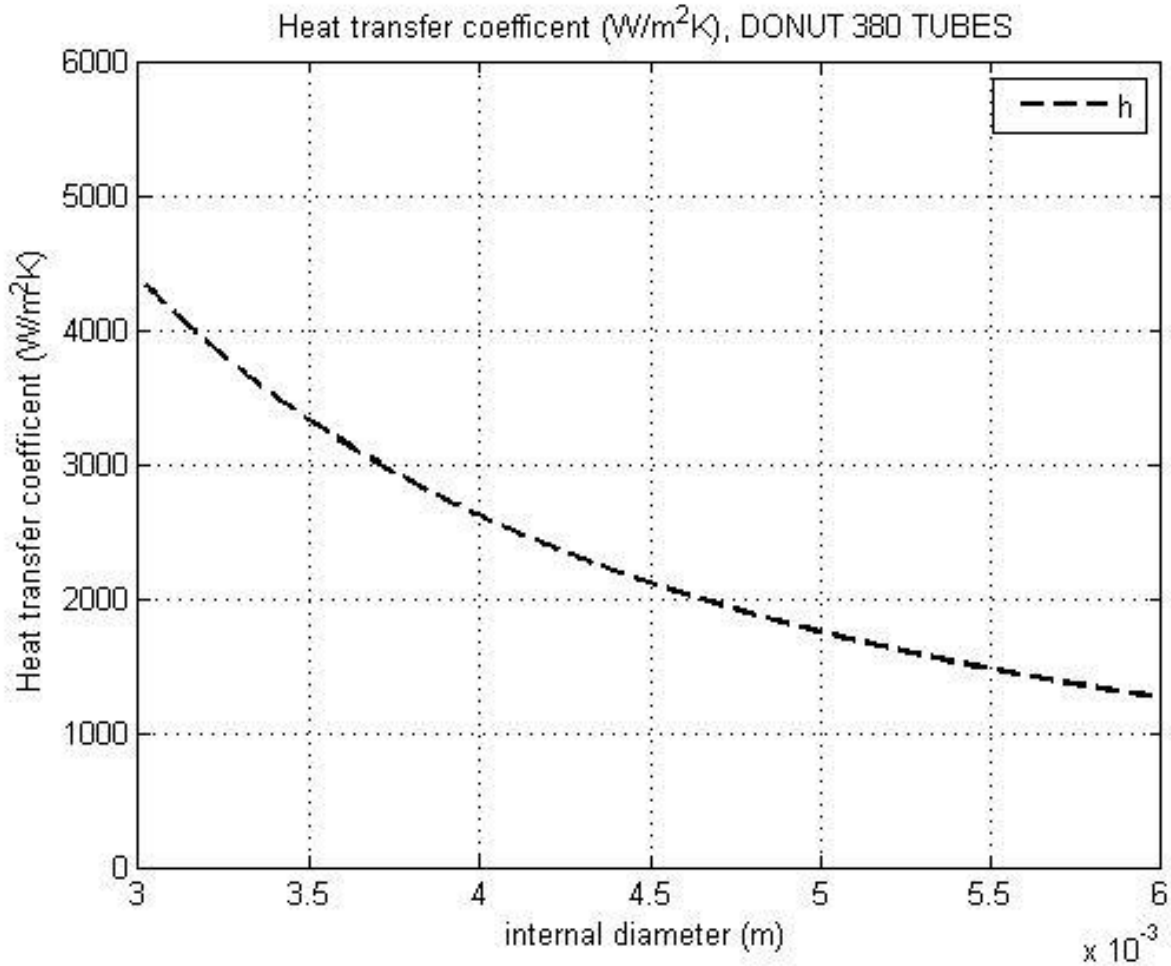



*Figure 6-7-Heat Transfer Coefficient on Helium Side-Case A*

**Step 2:**

Evaluating heat transfer coefficient on the water side:

A 2D CAD model is generated using Pro-E software and the model is imported to the COMSOL module.

The following boundary conditions are given to the model;

1. Inlet water temperature: 333 K
2. Inlet water velocity: Varied in a range of 0.2 m/s to 0.9 m/s
3. Heat transfer coefficient on the internal wall of the tubes found in the step 1
4. Internal wall temperature of tubes: 369

| Inlet Velocity (m/s) | Average inside wall Helium Temperature (K) | Water Temperature at the outlet (K) | Water Temperature at the inlet (K) | Average Water Temperature (K) | Heat Transfer Coefficient Water Side $(W/m^2K)$ | ΔT (K) | ΔQ (kW) | Water side ΔP (Pa) |
|---|---|---|---|---|---|---|---|---|
| 0.2 | 369 | 343.30 | 333.15 | 339.48 | 1567 | 10.15 | 6.79 | 213 |
| 0.3 | 369 | 341.02 | 333.15 | 338.11 | 1985 | 7.87 | 7.90 | 412 |
| 0.4 | 369 | 339.62 | 333.15 | 337.10 | 2299 | 6.47 | 8.65 | 620 |
| 0.5 | 369 | 338.46 | 333.15 | 336.57 | 2354 | 5.31 | 8.88 | 917 |
| 0.6 | 369 | 337.79 | 333.15 | 336.14 | 2584 | 4.64 | 9.31 | 1232 |
| 0.7 | 369 | 337.20 | 333.15 | 335.79 | 2640 | 4.05 | 9.48 | 1732 |
| 0.8 | 369 | 336.79 | 333.15 | 335.53 | 2789 | 3.64 | 9.74 | 2109 |
| 0.9 | 369 | 336.47 | 333.15 | 335.32 | 2958 | 3.32 | 9.99 | 2660 |

*Table 6-2-Heat Transfer Characteristics Case A*

The heat transfer coefficient values can be found out using the method described earlier. Along with the heat transfer coefficient values the pressure drop values are also recorded.

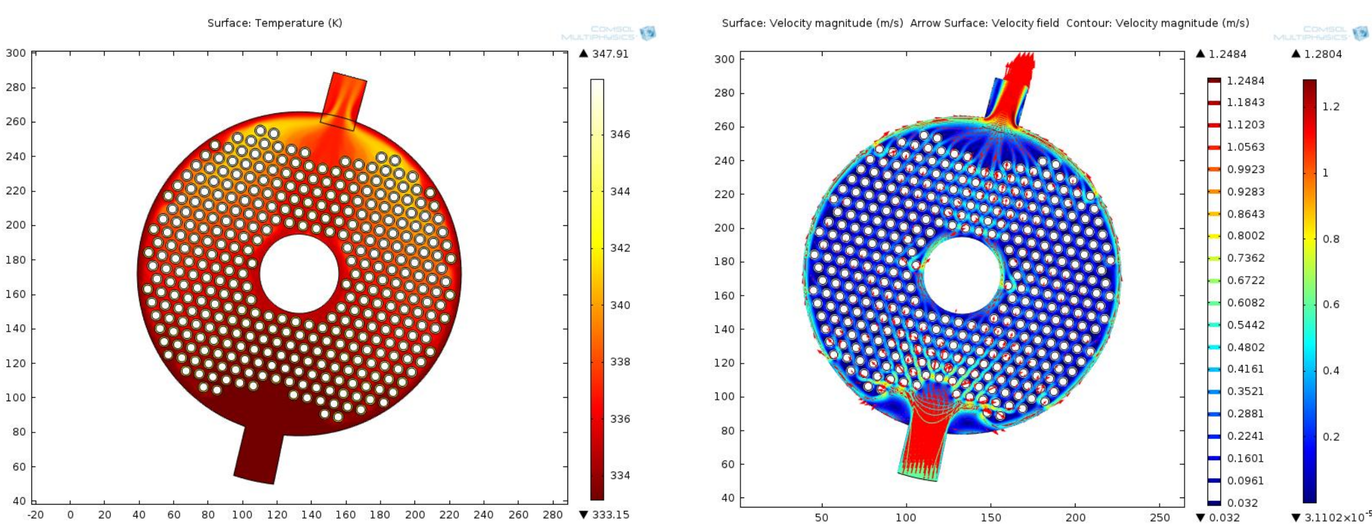



*Figure 6-8-Temperature and Velocity Contours-Case A*

The heat transfer coefficient values primarily depend on the Reynolds number for the flow of the water. Parameters like outer diameter, pitch length, flow velocity and temperature of water all influence the Reynolds number of the flow.

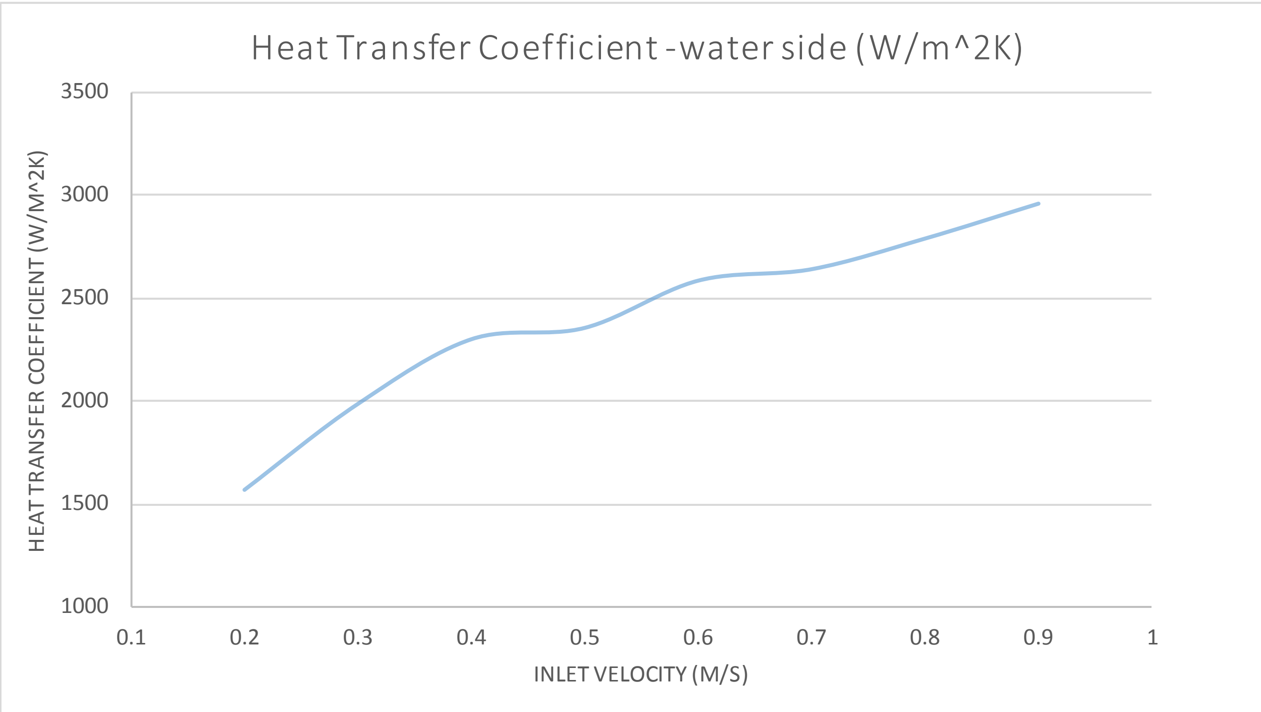



*Figure 6-9-Heat Transfer Coefficient Water side-Case A*

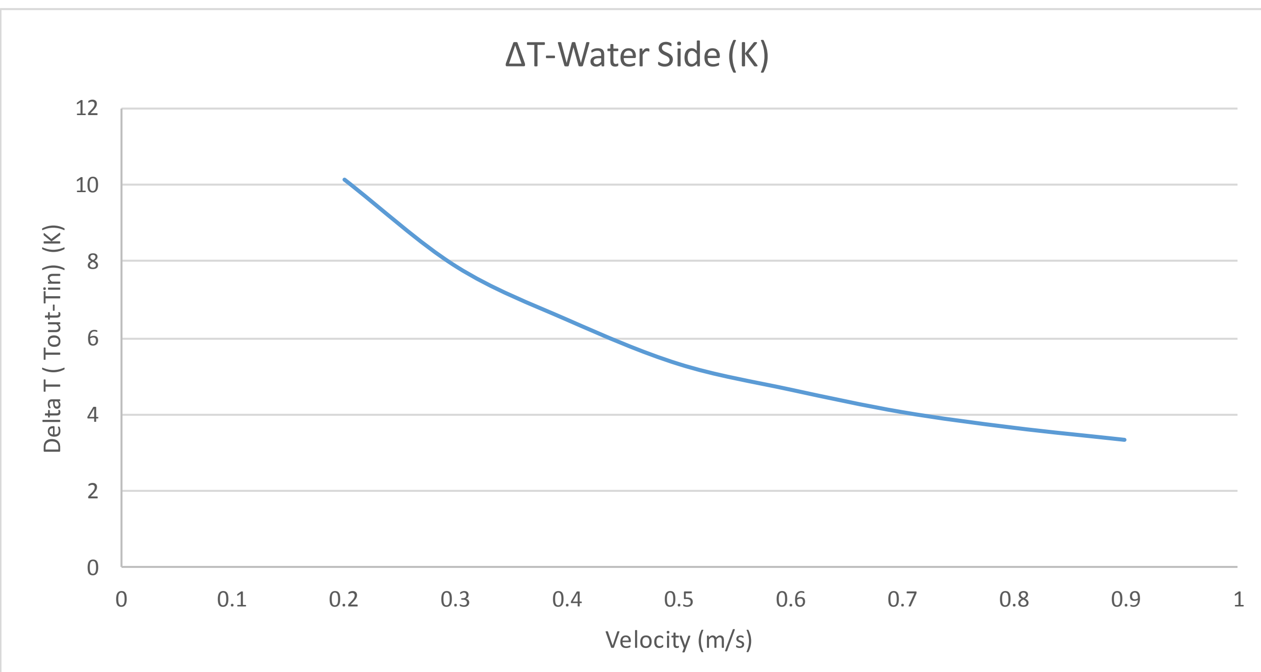



*Figure 6-10-Delta T Water side Case A*

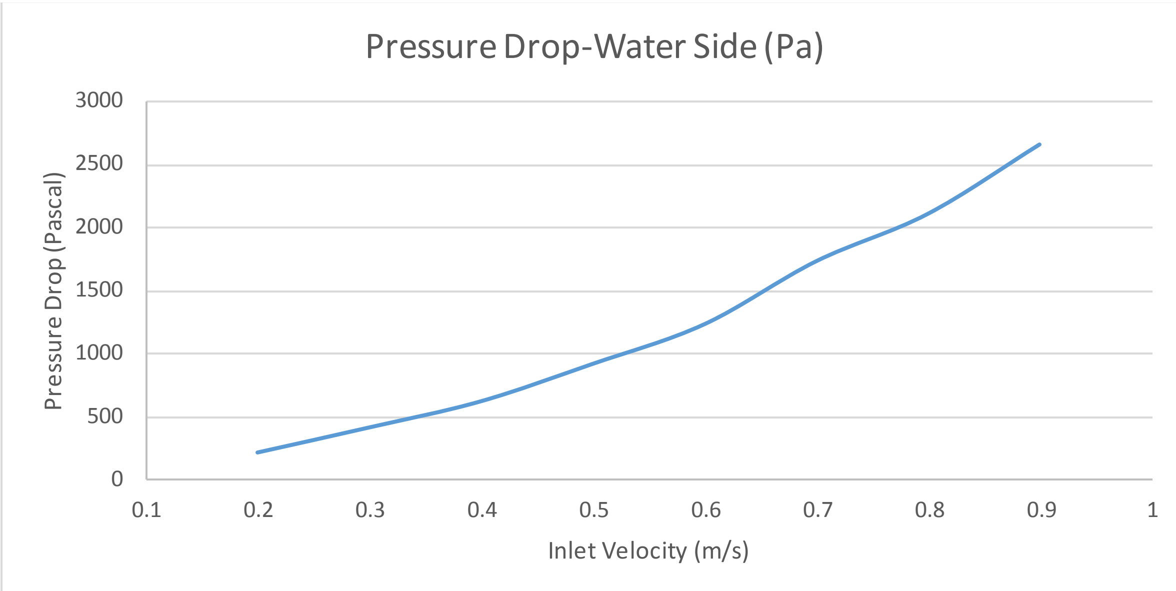



*Figure 6-11-Pressure Drop Water Side Case A*

Step 3:

To find the total surface area required for specified amount of heat transfer (24 kW in this case): In this step we input values of heat transfer coefficients on the helium side and on the water side in the program written to evaluate the total surface area required.

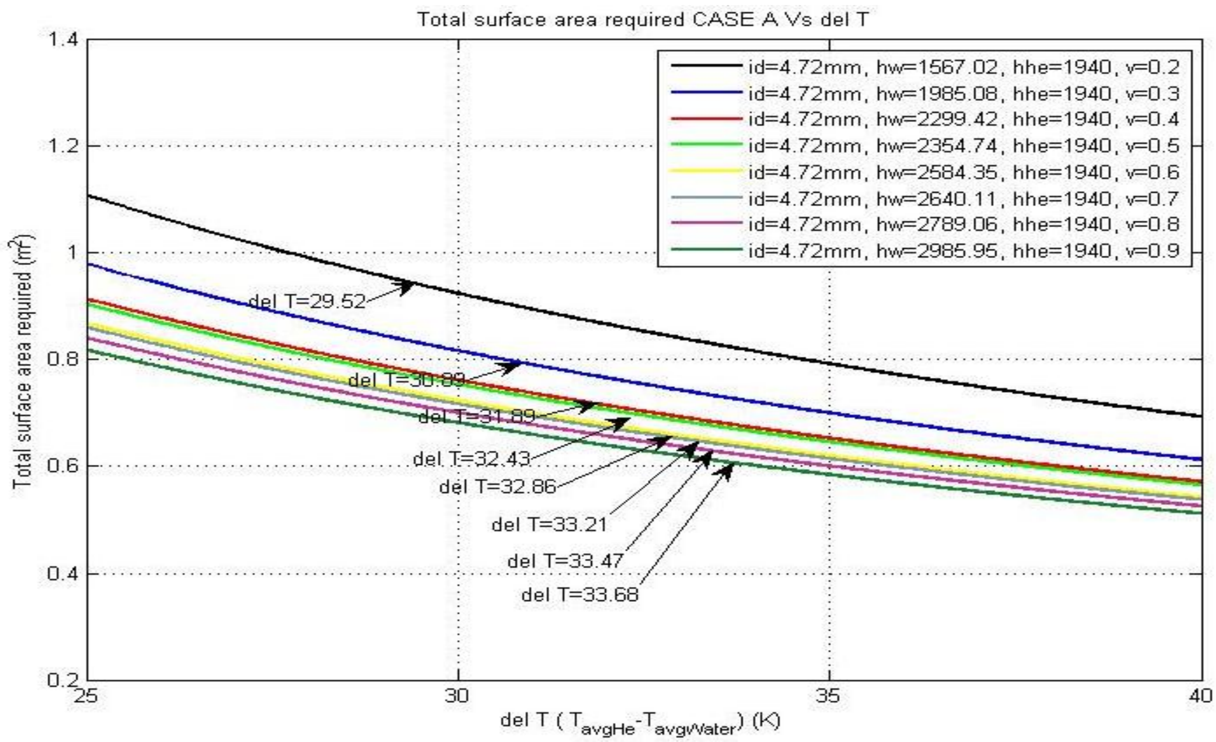



*Figure 6-12-Total Surface Area Required*

| Velocity | Heat Transfer Coefficient Water Side $(W/m^2K)$ | Heat Transfer Coefficient Helium Side $(W/m^2K)$ | Temperature Difference (K) | Area Required (m^2) | ΔT (K) | Required Length (m) | Helium side pressure drop (Pascal) |
|---|---|---|---|---|---|---|---|
| 0.2 | 1567 | 1940 | 29.52 | 0.94 | 10.15 | 0.142 | 352 |
| 0.3 | 1985 | 1940 | 30.89 | 0.79 | 7.874 | 0.120 | 265 |
| 0.4 | 2299 | 1940 | 31.90 | 0.71 | 6.47 | 0.108 | 182 |
| 0.5 | 2354 | 1940 | 32.43 | 0.69 | 5.31 | 0.105 | 169 |
| 0.6 | 2584 | 1940 | 32.86 | 0.66 | 4.64 | 0.100 | 145 |
| 0.7 | 2640 | 1940 | 33.21 | 0.65 | 4.05 | 0.098 | 143 |
| 0.8 | 2789 | 1940 | 33.47 | 0.62 | 3.64 | 0.094 | 140 |
| 0.9 | 2958 | 1940 | 33.68 | 0.6 | 3.321 | 0.091 | 139 |

*Table 6-3-Length of Surface Area Required-Case A*

**Case B:**

Step 1: Evaluating the heat transfer coefficient on the helium side:

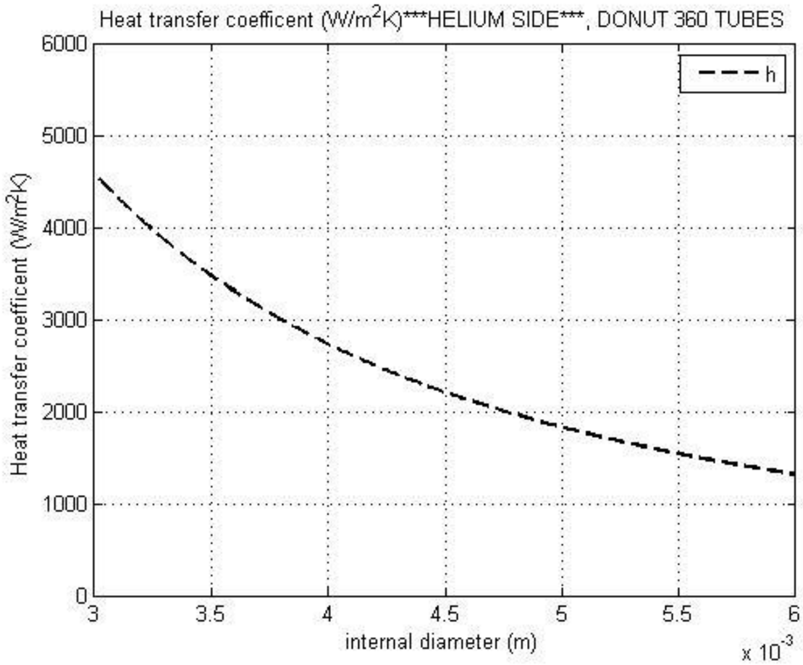



*Figure 6-13-Heat Transfer Coefficient Vs Internal Diameter Case B*

Step 2:

Evaluating heat transfer coefficient on the water side:

| Inlet Velocity (m/s) | Average inside wall Helium Temperature (K) | Average Water Temperature (K) | Water Temperature at the outlet (K) | Water Temperature at the inlet (K) | Heat Transfer Coefficient Water Side $(W/m^2K)$ | ΔT (K) | ΔQ (kW) | ΔP (Pa) |
|---|---|---|---|---|---|---|---|---|
| 0.2 | 369 | 339.15 | 343 | 333.15 | 1641 | 10.41 | 6.88 | 317 |
| 0.3 | 369 | 337.96 | 341 | 333.15 | 2182 | 8.53 | 8.45 | 489 |
| 0.4 | 369 | 337.29 | 340 | 333.15 | 2745 | 7.37 | 9.74 | 758 |
| 0.5 | 369 | 336.80 | 339 | 333.15 | 3233 | 6.47 | 10.69 | 1211 |
| 0.6 | 369 | 336.43 | 338 | 333.15 | 3754 | 5.82 | 11.54 | 1597 |
| 0.7 | 369 | 336.15 | 338 | 333.15 | 4272 | 5.3 | 12.26 | 2210 |
| 0.8 | 369 | 335.93 | 338 | 333.15 | 4785 | 4.87 | 12.87 | 2699 |
| 0.9 | 369 | 335.75 | 337 | 333.15 | 5334 | 4.52 | 13.44 | 3352 |

*Table 6-4-Heat Transfer Characteristics-Case B*

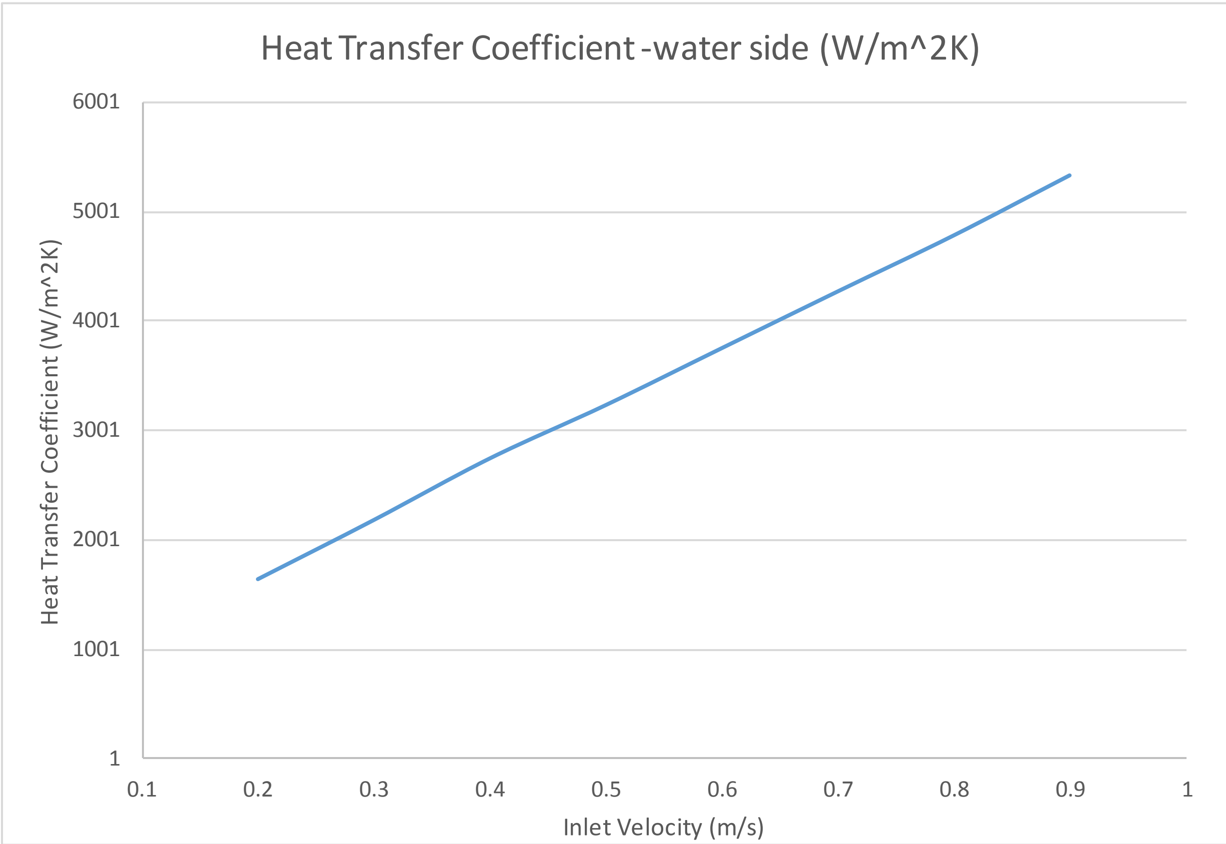



*Figure 6-14-Heat Transfer Coefficient-Water Side Case B*

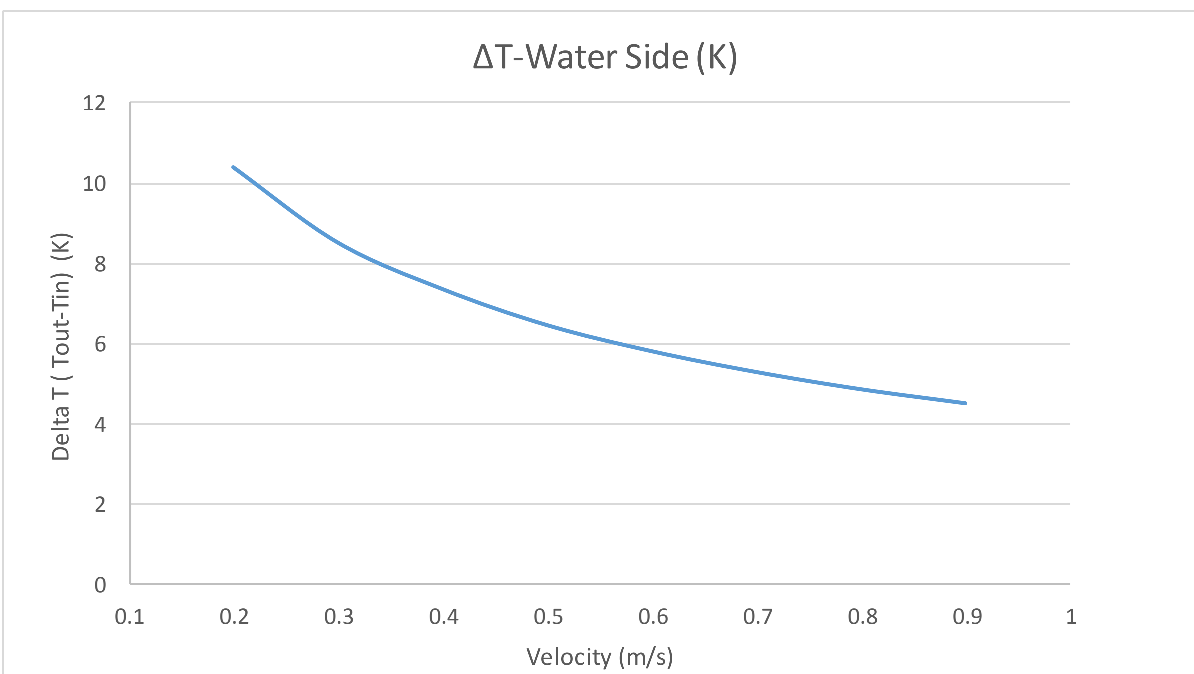



*Figure 6-15-Delta T on the Water side Case B*

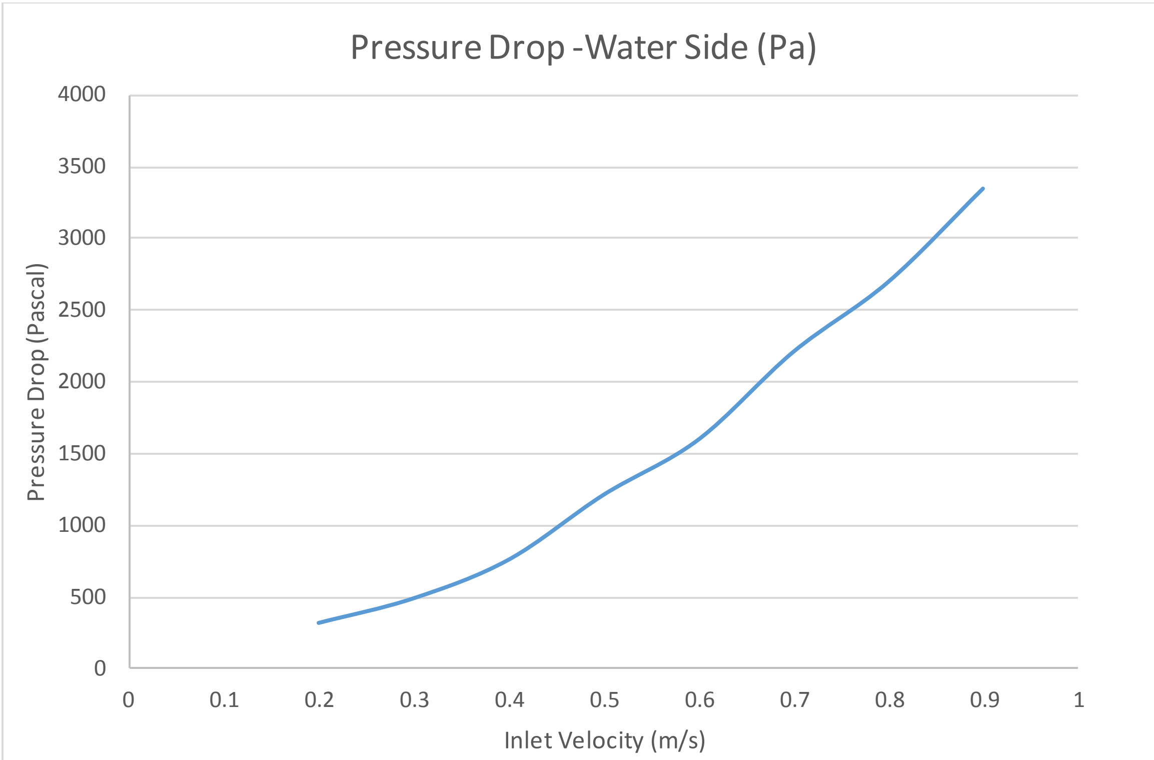



*Figure 6-16-Pressure Drop on the Water side Case B*

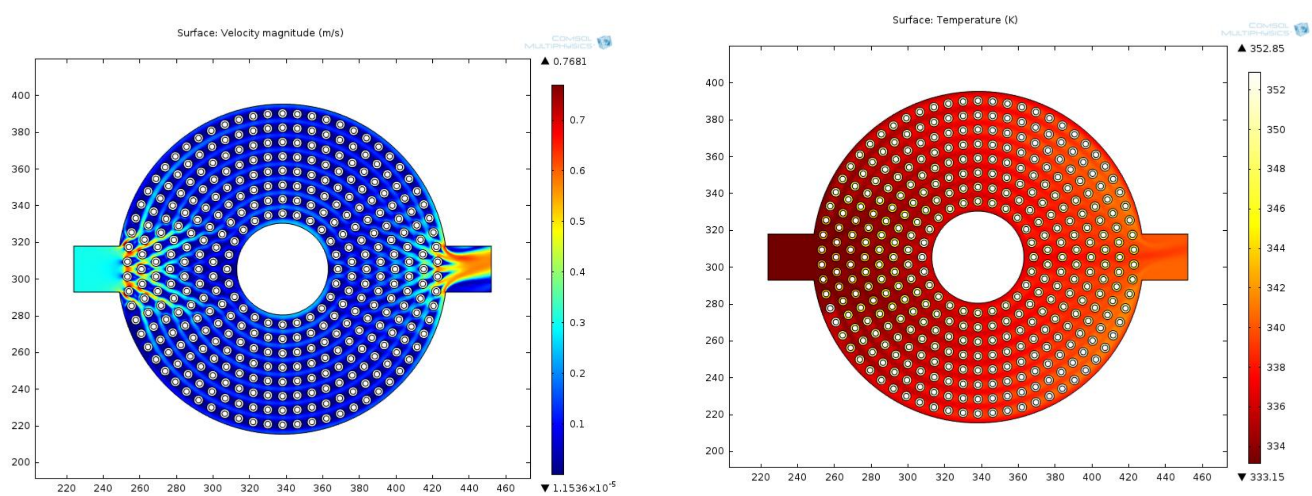



*Figure 6-17-Velocity and Temperature Plots Case B*

Step 3:

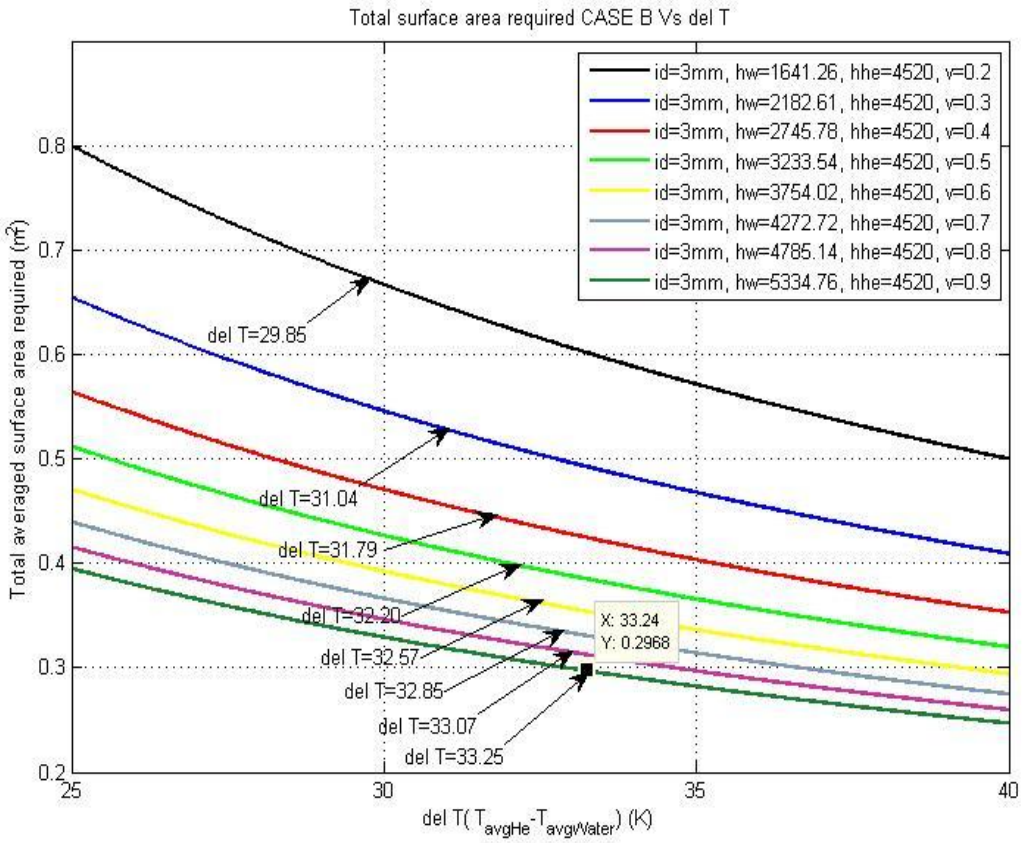



*Figure 6-18-Surface Area Required Case B*

| Velocity (m/s) | Heat Transfer Coefficient Water Side $(W/m^2K)$ | Heat Transfer Coefficient Helium Side $(W/m^2K)$ | Average Temperature Difference (K) | Area Required $(m^2)$ | ΔT (Tout-Tin) (K) | Requir ed Length (m) | Helium side pressure drop (Pascal) |
|---|---|---|---|---|---|---|---|
| 0.2 | 1641 | 4520 | 29.85 | 0.66 | 10.41 | 0.15 | 2143 |
| 0.3 | 2182 | 4520 | 31.04 | 0.53 | 8.53 | 0.12 | 1837 |
| 0.4 | 2745 | 4520 | 31.71 | 0.44 | 7.37 | 0.10 | 1531 |
| 0.5 | 3233 | 4520 | 32.20 | 0.40 | 6.47 | 0.09 | 1255 |
| 0.6 | 3754 | 4520 | 32.57 | 0.36 | 5.82 | 0.08 | 1225 |
| 0.7 | 4272 | 4520 | 32.85 | 0.33 | 5.3 | 0.07 | 1196 |
| 0.8 | 4785 | 4520 | 33.07 | 0.32 | 4.87 | 0.07 | 1182 |
| 0.9 | 5334 | 4520 | 33.25 | 0.30 | 4.52 | 0.07 | 1176 |

*Table 6-5-Length and Surface Area Required*

## 6.3-Selection of water inlet velocity

Heat transfer coefficient on the water side also depends on the average value of Reynolds number for the flow through the shell side of the heat exchanger. It is important to analyze the effect of the changes in the velocity has on the total surface area requirement for the heat transfer. There is an interesting trend which can be observed in Fig. 6.19. We can observe that, up to a certain limit of inlet velocity we get substantial decrement in the requirement for surface area and dead volume. However, the graph becomes more stable in terms of magnitude as we increase the value of the inlet velocity. Also, increasing the inlet velocity of water also increases the pressure drop on the water side. The rise in the pressure drop causes increase in pumping losses which is unfavorable. This analysis helps in determining the optimal velocity of flow especially when it important to consider the dead volume and pressure drop at the same time.

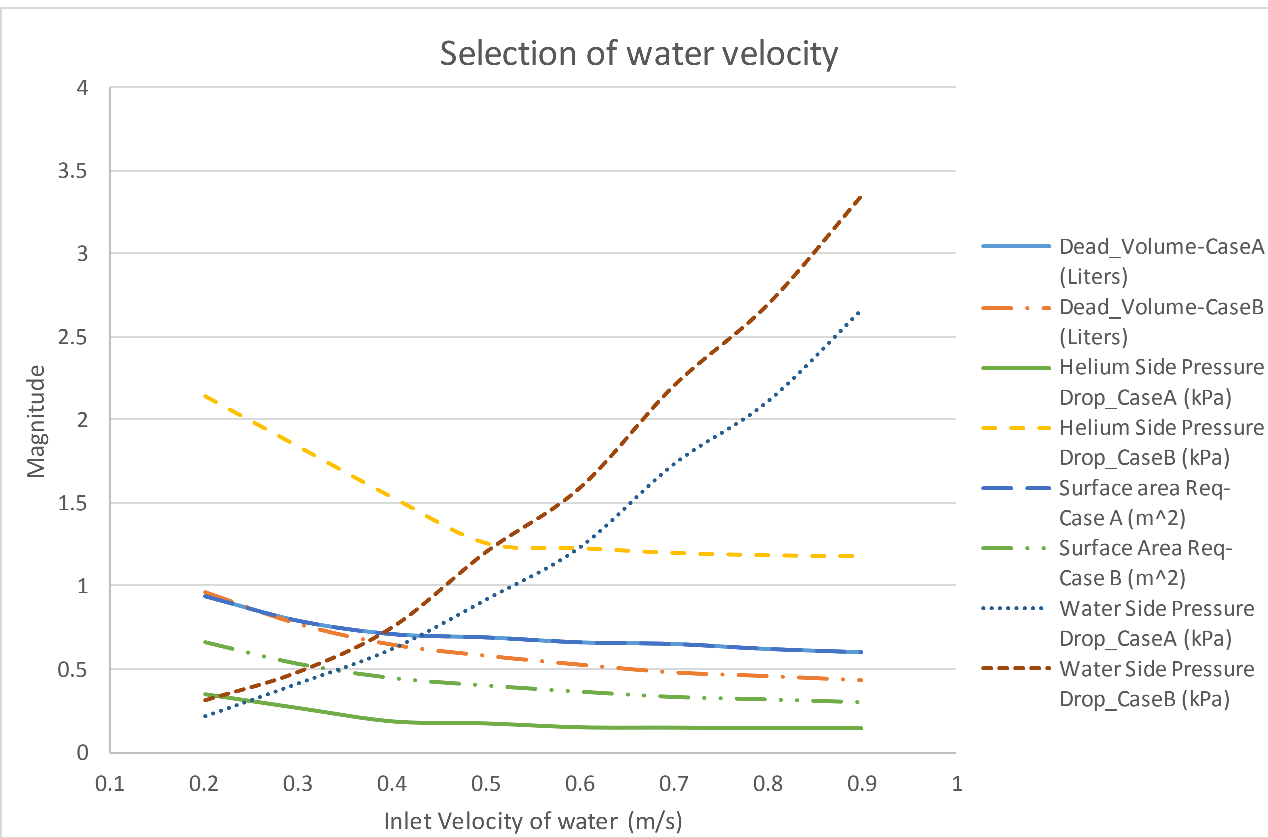



*Figure 6-19-Selection of Water Inlet Velocity*

## 6.4 -Comparison of different material for tubes:

In this chapter the change in the requirement for total surface area required for the specified amount of heat transfer with change in materials for the tubes is discussed.

Copper has a very high thermal conductivity because of which it is most favored as a material for heat exchangers. However, copper is expensive as compared to the other alternatives like Aluminum and Stainless Steel. Figure 33 shows the surface area requirements for Stainless Steel, Copper and Aluminum as a function of difference in the average temperatures of Helium and Water.

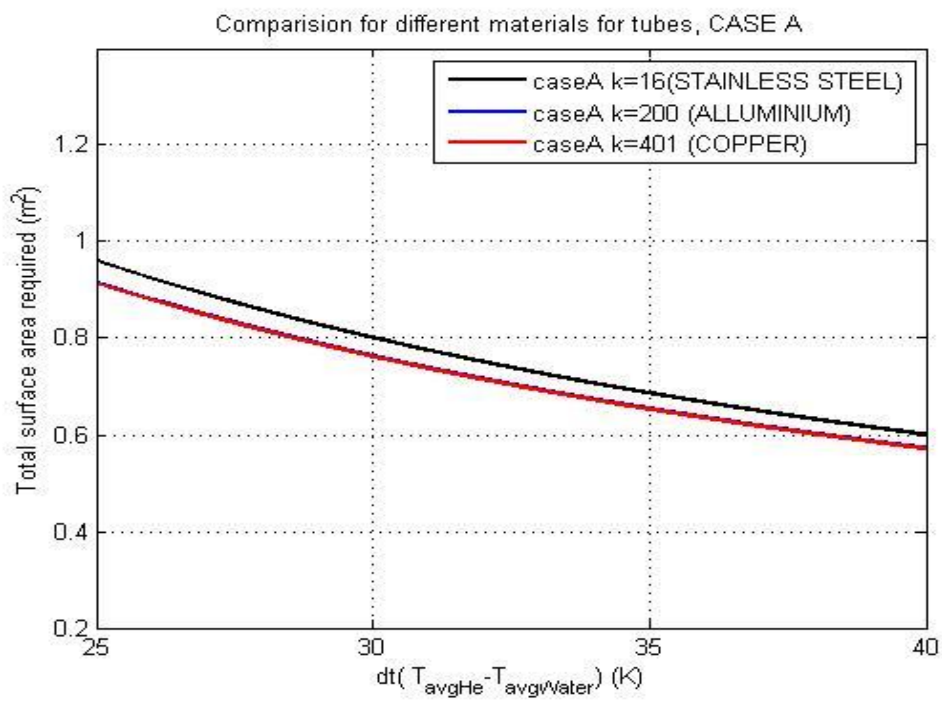



*Figure 6-20-Total Surface Area Required*

Therm
al conductivity of Stainless Steel is 16 W/mK is very low as compared to the Copper thermal conductivity which is 400 W/mK. However, as the thickness of the wall of the tubes is in the around 1 millimeter range, the thermal resistance induced on the overall thermal circuit is of the less magnitude.

| Material (Thermal conductivity kW/mK at ATP) | Total surface area required for 24 kW of heat transfer (m^2) | Percent increase in the total surface area required compared to the base line. |
|---|---|---|
| Base Line -COPPER (401) | 0.7184 | 0.000 |
| Aluminum (200) | 0.7196 | 0.167 |
| Stainless Steel (16) | 0.7965 | 10.871 |

*Table 6-6-Material Analysis*

## 6.5- COP study for VHP

As a part a part of the overall performance analysis of the Vuilleumier heat pump, the Coefficient of Performance (COP) was tested for different input parameters. COP of the Vuilleumier heat pump is determined and influenced by the several parameters, few of which are Stroke length, bore size and operating temperature ranges.

Studies were done using GT-Power software which is one dimensional thermodynamic simulation tool. The VHP model was analyzed by performing several case studies.

**Stroke length study**

| Case | Engine Speed | Stroke length- Hot Cylinder | Stroke Length-Cold Cylinder | Wall temperature Hot | COP-Hot | COP-Cold |
|---|---|---|---|---|---|---|
| 1 (50,50) | 450 | 50 | 50 | 700 | 1.816 | 0.818 |
| 2 (45,55) | 450 | 45 | 55 | 700 | 1.784 | 0.782 |
| 3 (60,40) | 450 | 60 | 40 | 700 | 1.801 | 0.806 |
| 4 (65,35) | 450 | 65 | 35 | 700 | 1.779 | 1.779 |
| 5 (70,30) | 450 | 70 | 30 | 700 | 1.695 | 0.782 |

*Table 6-7-Stroke length study*

Different combinations of the stroke lengths for warm and cold sides were considered for this analysis

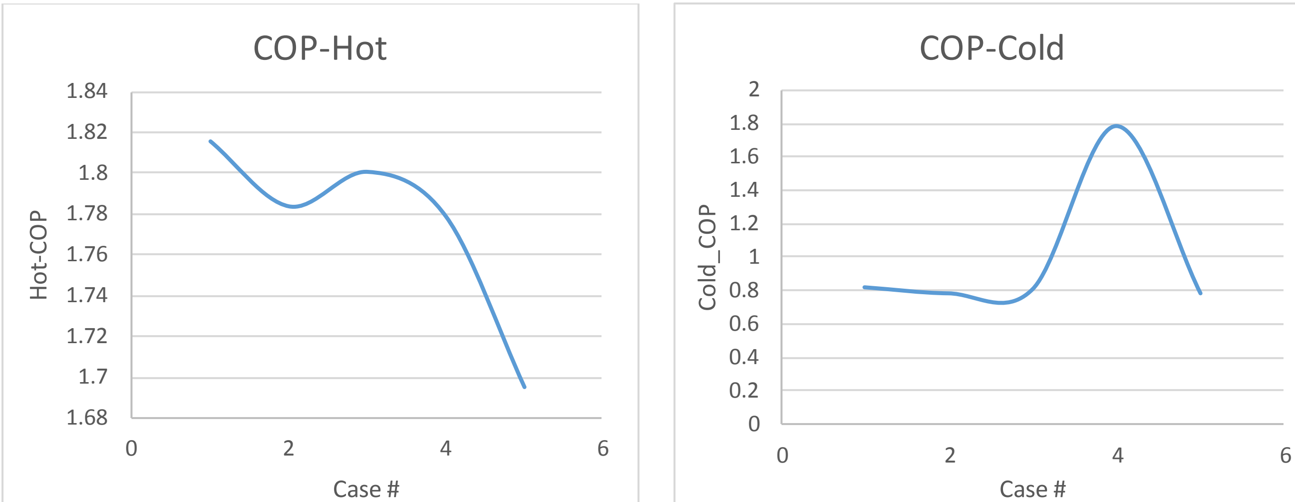



*Figure 6-21-Stroke length Study*

**Bore size diamter**

The COP of the machine is also dependent on the bore diameter size for the hot and the warm chambers. The Volume inside the gas has an influence on the amount of the heat energy picked up on the cold side and dissipated on the warm size. It has been noted that the COP of the machine decreases with increase in volume of the gas inside the system. Another contributing factor to this decrease in the COP with

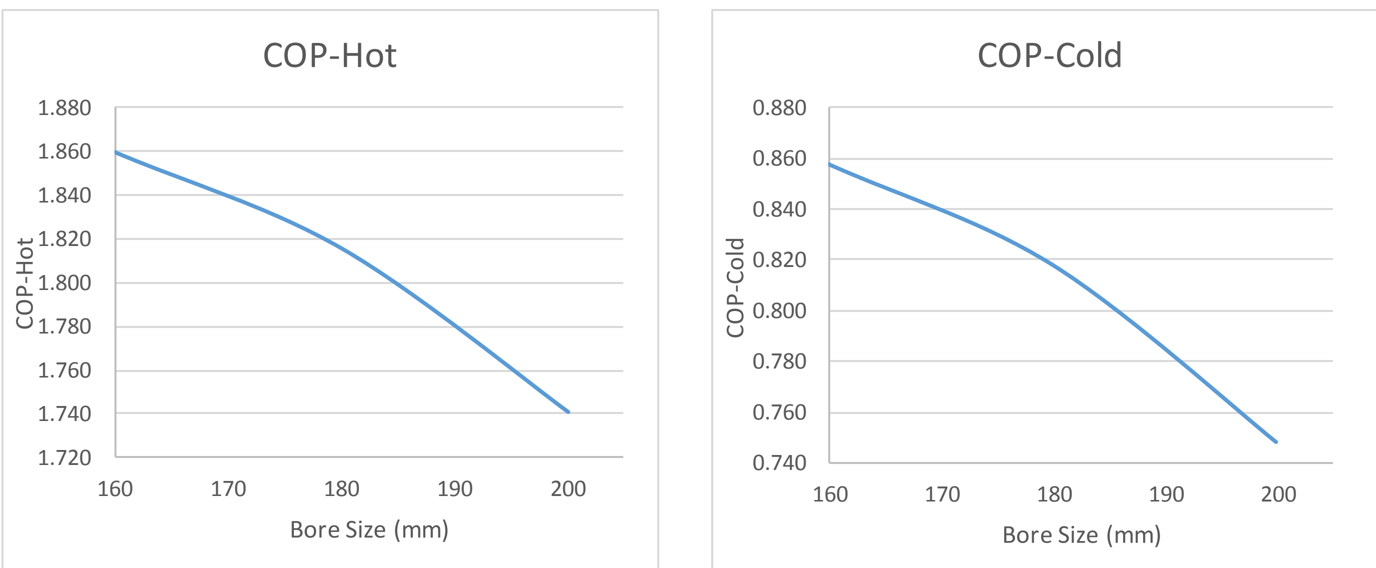



*Figure 6-22-Bore size diameter study*

increase in the bore size is the increase in the amount of energy required to drive the displacers.

**Cold heat exchanger wall temperature**

The temperature differences between hot side and the cold side has a major influence on the COP of the machine. The VHP COP can be relate to the Carnot heat pump COP which is defined as,

$$COP_{heating} = \frac{T_{hot}}{T_{hot} - T_{Cold}}$$

| Case | Bore Size | Engine Speed | Stroke length-Hot cylinder | Stroke Length - Cold Cylinder | Wall temperature Hot | Wall Temperature Cold | COP-Hot |
|---|---|---|---|---|---|---|---|
| 1 | 180 | 450 | 50 | 50 | 700 | -5 | 1.816 |
| 2 | 180 | 450 | 50 | 50 | 700 | 5 | 1.917 |
| 3 | 180 | 450 | 50 | 50 | 700 | 10 | 2.015 |
| 4 | 180 | 450 | 50 | 50 | 700 | 20 | 2.11 |
| 5 | 180 | 450 | 50 | 50 | 700 | 25 | 2.15 |

*Table 6-8- Cold heat exchanger wall temperature*

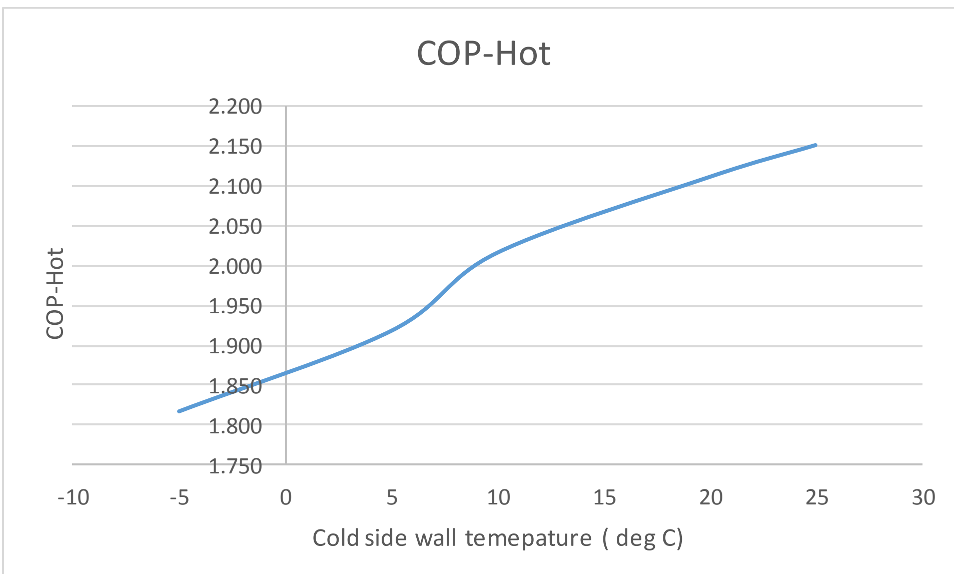



*Figure 6-23- COP vs Cold heat exchanger wall temperature*

The COP of the VHP does relate to the Carnot COP and is influenced by the temperature difference between the hot and the cold regions.

## 6.6- Conclusion and Future Work

### Conclusion:

Heat exchangers are critical components of the Vuilleumier heat pump. It is of high importance to design heat exchangers that are optimal in their functionality to enhance the performance of the Vuilleumier heat pump.

In this thesis a robust technique of analysis was developed for modified design of shell and tube heat exchangers which can be then installed in Vuilleumier heat pumps. This analysis technique significantly reduces the time required to output the results as compared to the 3D CFD analysis procedure. Breaking up the analysis in sequential steps helps to study the heat transfer characteristics for the participating fluids individually in the heat exchanger. The analysis procedure generates enough data which can be further validated using experimental procedures.

Through this study it was observed that tubes with smaller internal diameters and large in number perform much better in terms of heat transfer capabilities. This configuration helps in reduction of dead volume when compared to the configuration of small number of tubes with larger internal diameters.

Two cases with different configurations were analyzed and performance characteristics were measured and plotted. It was observed that pressure drop is strong inverse function of heat transfer coefficient for the helium flow. Significant amount of dead volume and surface area requirement can be reduced by concentrating on improvement of heat transfer coefficient by wisely choosing pressure drop as a limiting factor.

The configuration with smaller tube IDs and large in number does give good heat transfer performance however the manufacturability and cost are the limiting factors. It was also observed through this analysis that there is a definite range of optimal values that can be selected for the inlet velocity of water.

### Future Work:

The technique described in this literature is not computationally intensive as compared to 3D CFD analysis with single step input and output operation but it a multistep operation which requires user to record the data at every step. After the data is recorded in each step it acts as an input for further steps. This process however can be automated by creating a link between the CFD software, MATLAB and MS Excel.

A program can be built in MATLAB which would give maximum heat transfer coefficient on the helium side after accepting maximum allowable pressure drop as an input.

Fouling on the outer surfaces of the heat exchanger pipes is another factor which would degrade the performance of the heat exchanger which needs to be addressed. Fouling is the accumulation of the unwanted material on the surface of the solid which is being exposed the liquid containing impurities.

Finally, the numerical results should be validated by performing experiments.

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