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# Investigation and Experimental Analysis of Gasketed Plate Type Heat Exchanger

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**Abstract—***In this work, a model of compact heat exchanger, Gasketed Plate type heat exchanger set up has been investigated and analysed. The plate type heat exchanger made of SS 316 having chevron angle 30$^o$ has been designed and fabricated to for experimental analysis. The hydraulic parameters have been changed to observe its effect on thermal performance of Plate Heat Exchanger as the Reynolds Number is varied from 600 to 6500. The mass flow rate, dimensions of plate, pressure difference between the plates and temperature at the inlet and outlet port of Gasketed Plate Type Heat Exchanger are measured. The Kumar Correlation for Nusselt Number which is the function of Prandtl Number and Reynolds Number has been used to calculate the convective heat transfer coefficient and the Overall heat transfer coefficient. The mathematical simulation of physical dimensions of plate type heat exchanger such as plate length, width, spacing and thickness has been done using simulation and its effect on overall heat transfer coefficient and pressure drop has been analyzed. The outcomes of these volumetric flow variations on test set-up and mathematical simulation has been verified with theoretical analysis.*

**Keywords**: *Plate Type Heat Exchanger(PHE), Nusselt Number, Reynolds Number, Overall Heat Transfer coefficient.*

## 1. INTRODUCTION

Heat exchangers are devices used to transfer heat energy from one fluid to another. A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. In heat exchangers, there are usually no external heat and work interactions. Heat exchangers are broadly classified on the basis of construction as:-

1. Tubular
2. Plate Type
3. Extended Surface
4. Regenerative

Plate Type Heat Exchangers are broadly classified into PHE, Spiral, Plate Coil and Printed Circuit. The PHE is further classified into Gasketed, Welded and Brazed on the basis of sealing. The corrugations on successive plates contact or cross each other to provide mechanical support to the plate pack through a large number of contact points. The resulting flow passages are narrow, highly interrupted, and tortuous, and enhance the heat transfer rate and decrease fouling resistance by increasing the shear stress, producing secondary flow, and increasing the level of turbulence. The corrugations also improve the rigidity of the plates and form the desired plate spacing. Plates are designated as hard or soft, depending on whether they generate a high or low intensity of turbulence. There are several plate patterns available for PHE as washboard, zigzag, chevron or herringbone, protrusions and depressions, washboard with secondary corrugations and oblique washboard. We have considered Chevron pattern of PHE with 30$^o$ chevron angle.

## 2. LITERATURE REVIEW

Abubaker E.M Elbalsohi, Ruoxu Jia and Junling Hu (1) studied heat exchangers are used widely in many industries for heat recovery or cooling purposes. This paper developed a numerical model to simulate a counter flow parallel heat exchanger. A representative repeating unit cell of the multichanneled heat exchanger was taken as the computational domain, which includes a cold channel and a hot channel separated by plates. The model was simulated in COMSOL for an oil to water heat exchanger. Higher temperature oil and relatively lower temperature water entered two separate parallel channels in opposite directions. The detailed distributions of temperature, velocity, and pressure were used to analyze the performance of the heat exchanger. It was found the model can be used to provide guidance for designing an optimal heat exchanger.

Angela Pleșa, Oana Giurgiu and Lavinia Socaciu (2) presented a Computational Fluid Dynamics (CFD) numerical study for two different models of mini channels, included in plate heat exchangers structure. The influence of geometric characteristics of the two studied plates on the intensification process of heat transfer was studied comparatively. For this purpose, it was examined the distribution of velocity, temperatures fields and distribution of convection coefficient along the active mini channel. The analyzed mini channels had the inclination angles

A. Bellagi H. Dardour and S. Mazouz (3) presented a theoretical analysis of a cocurrent plate heat exchanger and the results of its numerical simulation. Knowing the hot and the cold fluid streams inlet temperatures, the respective heat capacities mCp and the value of the overall heat transfer coefficient, a 1-D mathematical model based on the steady flow energy balance for a differential length of the device is developed resulting in a set of N first order differential equations with boundary conditions where N is the number of channels.For specific heat exchanger geometry and operational parameters, the problem is numerically solved using the shooting method. The simulation allows the prediction of the temperature map in the heat exchanger and hence, the evaluation of its performances. A parametric analysis is performed to evaluate the influence of the R-parameter on the NTU values. The simulation allows the prediction of the temperature map in the heat exchanger and hence, the evaluation of its performances. A parametric analysis is performed to evaluate the influence of the R-parameter on the NTU values.

Jogi Nikhil G., Assist. Prof. Lawankar Shailendra M (4) experimented heat transfer data will obtained for single phase flow (water-to-water) configurations in a corrugated plate heat exchanger for symmetric 45°/45°, 45°/75° chevron angle plates. The effect of variation of chevron angles with other geometric parameter on the heat transfer coefficient will be study. Reynold number ranging from 500 to 2500 and Prandtl number ranging from 3.5 to 6.5 will be taken for given experiment.Based on the experimental data, a correlation will estimate for Nusselt number as a function of Reynolds number, Prandtl number and chevron angle.

J.M.Pinto and J.A.W.Gut (5) determined the best configuration of gasketed plate heat exchangers is presented. The objective is to select the configuration with the minimum heat transfer area that still satisfies constraints on the number of channels, the pressure drop of both fluids, the channel flow velocities and the exchanger thermal effectiveness. The configuration of the exchanger is defined by six parameters, which are as follows: the number of channels, the numbers of passes on each side, the fluid locations, the feed positions and the type of flow in the channels. The resulting configuration optimization problem is formulated as the minimization of the exchanger heat transfer area and a screening procedure is proposed for its solution. In this procedure, subsets of constraints are successively applied to eliminate infeasible and nonoptimal solutions. Examples show that the optimization method is able to successfully determine a set of optimal configurations with a minimum number of exchanger evaluations. Approximately 5 % of the pressure drop and channel velocity calculations and 1 % of the thermal simulations are required for the solution.

Fatih AKTURK, Nilay SEZER-UZOL, Selin ARADAG and Sadik KAKAC (6) studied the thermal and hydrodynamic performance analyses of selected gasketed-plate heat exchanger with different number of plates are performed experimentally. A gasketed-plate heat exchanger (GPHE) test set-up is designed and constructed to perform experimental measurements for thermal and hydrodynamic performance analyses of plate heat exchangers. The temperatures at the inlet and outlet ports, the volumetric flow rates of the hot and cold fluids, and the pressure drops between the inlet and outlet ports are measured during the experiments. By using the experimental data, Nusselt number correlation for heat transfer analysis and a friction factor correlation for pressure drop calculations are developed for the chevron-type plate heat exchanger tested as a function of Reynolds number and Prandtl number. The results obtained from these new correlations are compared with several existing correlations in the literature, which were developed for different plates. Although the trends of the new correlations are similar with the ones in the literature, this study shows that every plate design needs its specific correlations for heat transfer and pressure drop calculations, since the specific correlations developed for the specific plates tested are different than the ones given in literature. Obtaining these kinds of correlations for local products is especially important for the related industry to improve with the help of local test set-ups.

Tisekar Salman W, Mukadam Shakeeb A, Vedpathak Harshad S, Rasal Priyanka K, Khandekar S. B (7) researched work deals with experimental investigation of corrugated type of PHE with evaluation of convective heat transfer coefficient, overall heat transfer coefficient and exchanger effectiveness. Heat exchanger for carrying out this work consists of thin metal brazed plates of stainless steel with 0.5mm thickness, plates with rectangular geometry and corrugations on plate surface with chevron angle 450/450, plate spacing of 2.24mm and with 14 number of plates. Tests are conducted by varying operating parameters like mass flow rate of hot water, inlet temperatures of hot water. The main objective of this work is to find out effects of these parameters on performance of heat exchangers with parallel and counter flow arrangements. Results show that convective heat transfer coefficient increases with increase in mass flow rate and Reynolds number. Also the effectiveness varies with the mass flow rate of hot water. In this study, maximum effectiveness achieved for parallel flow arrangement with water as a working fluid is 0.67 and that of counter flow arrangement is 0.82. Use of plate heat exchanger is more advantageous than the tube type heat exchanger with same effectiveness as it occupies less space.

NTU graphs are elaborated for specific heat exchanger geometry and different operating conditions. For chevron angle of 30° respectively 60° and the Reynold flow number was 3500. Also a session of experimental measurements have been carried out on the two types of analyzed plates for the heat exchangers, confirming the results obtained through numerical simulation that the plate heat exchanger model using mini channels with inclination angle of $\beta = 60^{o}$ provides best heat transfer.

## 3. METHODOLOGY

**THERMODYNAMIC MODEL**:-A heat exchanager is designed on the basis of various parameters depending on optimization of weight, volume of heat transfer surface, initial cost, pressure drop, Loga Mean Temperature Difference and effectiveness. The basic optimization problem includes heat transfer, output temperature, pressure drop and size problem. These are specified by all the physical parameters and flow conditions of process and cooling fluids and heat exchanger selected.For designing the Plate heat exchanger following base line parameters are considered:-

**Table 1: Design and Simulation Base Line Parametrs**

| | |
|---|---|
| HEIGHT OF PLATE(L) | 300mm |
| WIDTH OF PLATE(W) | 60mm |
| THICKNESS OF PLATE(t) | 5mm |
| SPACING BETWEEN PLATES(b) | 2.9mm |
| MASS FLOW RATE OF HOT FLUID($m_h$) | 0.1kg/s |
| MASS FLOW RATE OF COLD FLUID($m_c$) | 0.1kg/s |
| ENHANCEMENT FACTOR(ϕ) | 1.16 |
| HOT FLUID INLET TEMPERATURE($T_{h1}$) | $85^0$C |
| HOT FLUID OUTLET TEMPERATURE($T_{h2}$) | 75 |
| NUMBER OF PLATES | 12 |
| CHEVRON ANGLE(β) | $30^0$ |
| COLD FLUID INLET TEMPERATURE($T_{c1}$) | $25^0$C |
| MATERIAL OF PLATE | SS 316 |
| PLATE CONDUCTIVITY | 17.5W/mK |

**Plate geometry as specified in Table 1 were fabricated for experimental analysis of and obtaining a set of data for thermal and hydraulic performance analysis.**

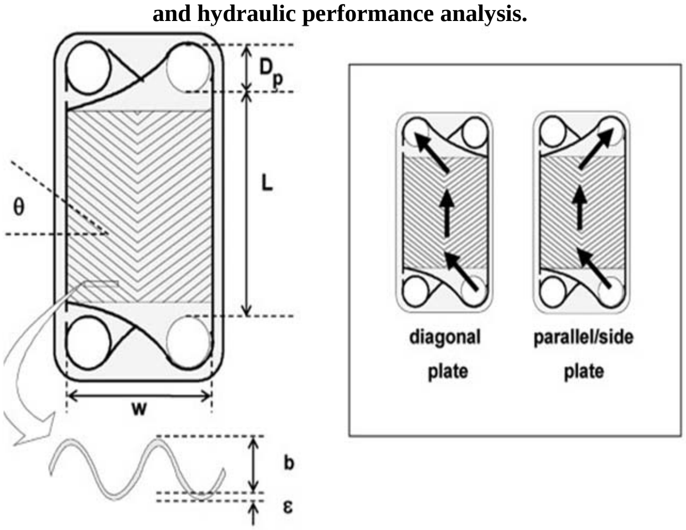


**Fig. 1: Dimensions of chevron type plate of PHE.**

**MATHEMATICAL MODEL**:- To calculate Heat Duty, volumetric flow rate is obtained by rotameter and then First law of Thermodynamics is applied to obtain the output temperature of cold fluid
$Q = \dot{m}C\Delta T$

The hydraulic diameter is used for calculation of Reynolds Number and Nusselt Number and is given by
$D_e = \frac{2b}{\emptyset}$

The flow area of both the fluids where N is the number of channels of fluid flow is given by
$A_f = NWb$

The velocity of fluid flow depends upon the mass flow rate and flow area and is given by
$V = \frac{m}{A_f \rho}$

Reynolds Number is defined by hydraulic diameter, mass flow rate and dynamic viscosity and for both the fluids is given by

$$Re = \frac{\rho V D_e}{\mu}$$

Prandtl's Number for a particular fluid at a given temperature is fixed, depending on dynamic viscosity, thermal conductivity, specific heat and given by
$Pr = \frac{\mu C_p}{k}$

Nusselt Number can be calculated by
$Nu = \frac{h D_e}{k}$

Kumar Correlation for Re>10 and Chevron Angle $30^0$ is used for calculating the value of convective heat transfer coefficient and is given by

$$Nu = .348 Re^{0.663} Pr^{\frac{1}{3}} \left(\frac{\mu}{\mu_w}\right)^{0.17}$$

Overall Heat Transfer Coefficient without considering the fouling factor can be calculated from $\frac{1}{U} = \frac{1}{h_p} + \frac{t}{k_p} + \frac{1}{h_c}$

LMTD (Log Mean Temperature Difference) which is used to determine the temperature driving force for heat transfer for counter flow heat exchanger is given by
$\theta_m = \left[\frac{(T_{h1}-T_{c2})-(T_{h2}-T_{c1})}{\ln\left(\frac{T_{h1}-T_{c2}}{T_{h2}-T_{c1}}\right)}\right]$

Heat Duty can be used to calculate the actual heat transfer area and can be calculated from
$Q = UA\theta_m$

Pressure Drop in the plates forming the channels of heat transfer in Plate type heat exchanger can be given by
$\Delta P = \frac{64}{Re \times D_e} \times \rho \times \frac{V^2}{2D_e} \times L$

**EXPERIMENTAL SETUP**:- The experimental setup consists of corrugated herringbone type PHE consisting 12 plates of dimensions given in table 1, rotameter, pressure gauge, centrifugal monoblock pumps, reservoir and temperature gauges in form of digital thermometer with probes. The fluid flow is regulated by valves and reading is taken accordingly. The working fluids used in the PHE are hot and cold water. The configuration of heat exchanger is counter flow for better heat transfer. Various experiments were conducted keeping all parameters constant and varying a single parameter. For getting better outcome the procedure is repeated. The physical parameters are simulated keeping all other physical and hydraulic parameters constant.

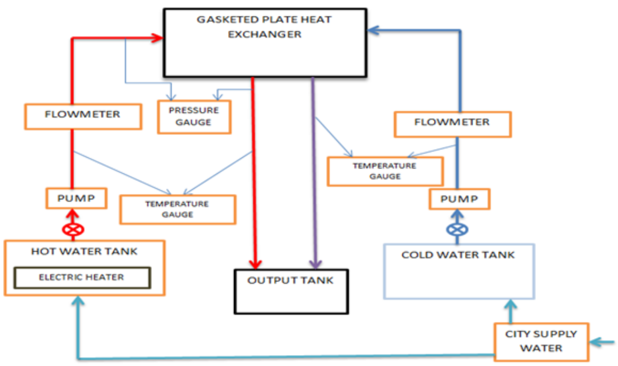


**Fig. 2 Schematic diagram for experimental setup of PHE**

**Fig. 3 Experimental Setup of PHE**

## 4. RESULT AND DISCUSSION

For water to water heat transfer different results are plotted based on experimental outcomes and simulation results:-

A) For varying mass flow rate of fluid:-

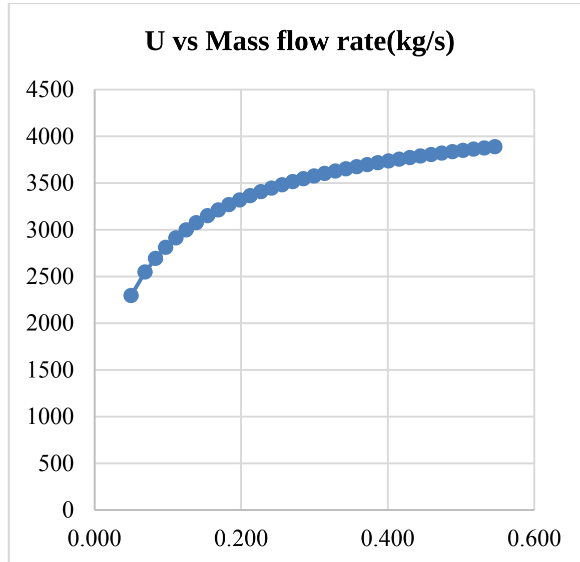


**Chart 1:- Heat transfer coefficient Vs Mass flow rate**

This graph shows that heat transfer coefficient with mass flow rate increases logarithmically resulting in higher heat transfer rate.

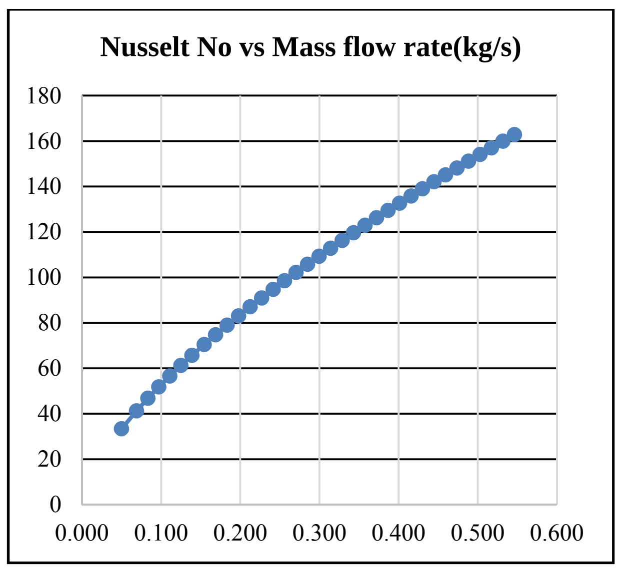


**Chart 2:-Nusselt No Vs Mass flow rate**

This graph shows that Nusselt number increases linearly with mass flow rate of fluid resulting in higher convective heat transfer coefficient.

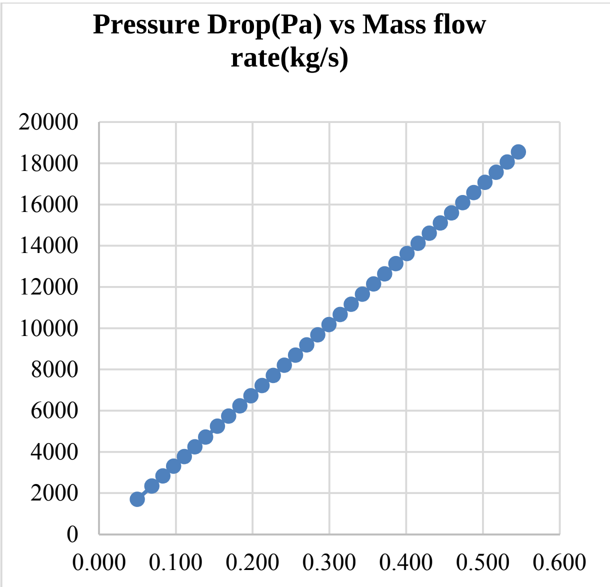


**Chart 3:-Pressure drop Vs Mass flow rate**

This graph shows that pressure drop inside the PHE increases with increasing mass flow rate resulting in higher pumping loss which is not desirable.

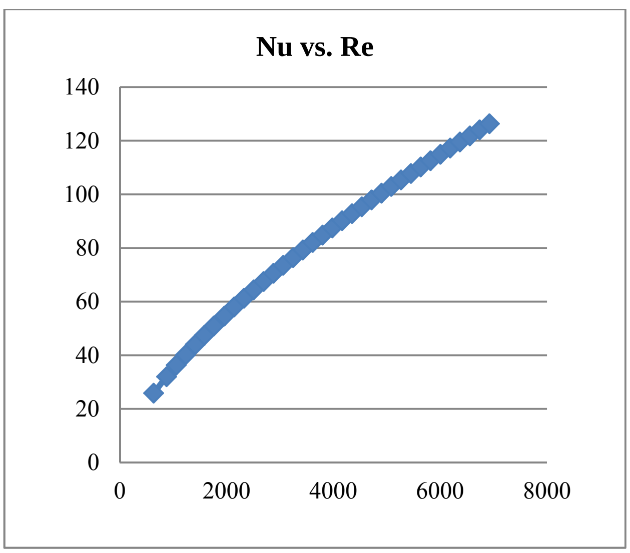


**Chart 4:-Nusselt Number Vs Reynolds Number**

This graph shows that with increase in Reynolds number that is higher turbulence the convective heat transfer increases resulting in higher heat transfer.

B) For varying thickness of plate

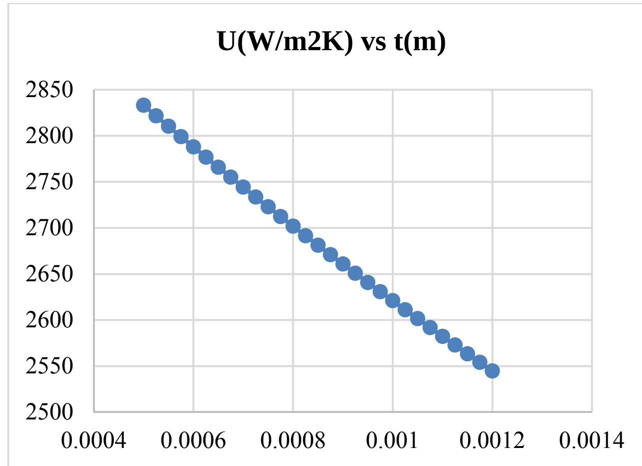


**Chart 5:-Heat transfer Coefficient Vs Thickness of plate**

This graph shows that overall heat transfer coefficient decreases with increase in thickness resulting in lower heat transfer.

C) For varying spacing between plates

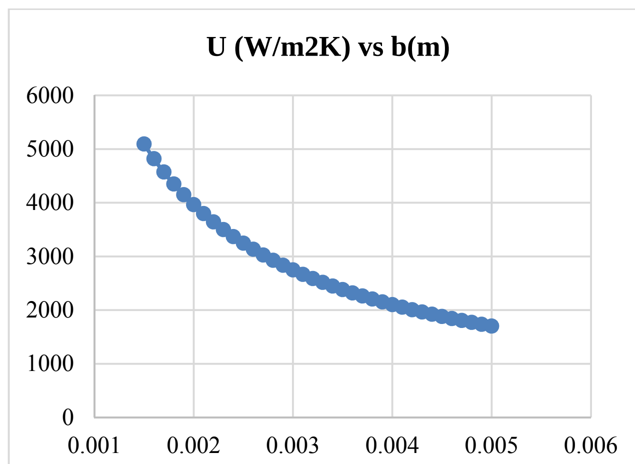


**Chart 6:-Heat transfer coefficient Vs spacing between plates**

This graph shows that overall heat transfer coefficient decreases exponentially with increase in spacing between plates resulting in very low heat transfer

D) For varying width of plate

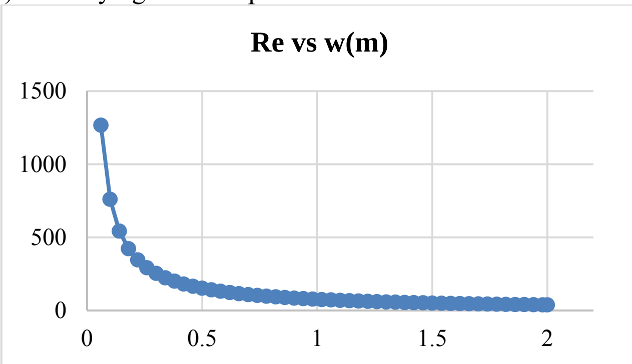


**Chart 7:-Reynolds Number Vs width of plate**

This graph shows that with increase in width of plate Reynolds Number decreases very sharply leading to lesser turbulence which leads to very sharp decline in heat transfer rate in PHE.

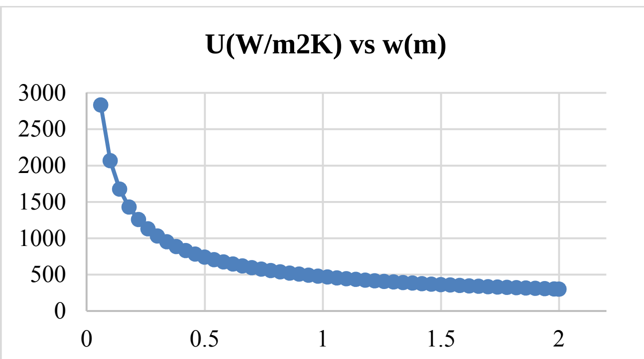


**Chart 8:-Heat Transfer Coefficient Vs width of plate**

This graph shows that there is sharp decline in heat transfer between plates of PHE with increase in width of plate which can be attributed to lower turbulency.

E) For varying Length of plate

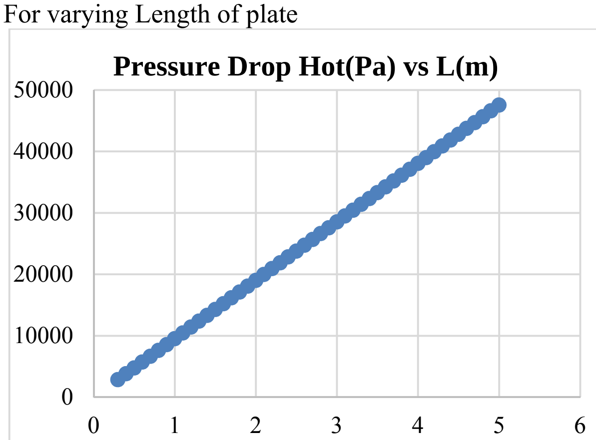


**Chart 9:-Pressure drop Vs Length of Plate**

This graph shows that there is linear increase in pressure drop in increase with length of plate. This leads to higher pumping loss but plate length should be high for more heat transfer area so plate length should be kept corresponding to pumping power and required heat duty.

## 5. CONCLUSION

Gasketed Plate type Heat Exchanger has been used for thermal and hydrodynamic performance analysis as discussed in graphs above:-

- For maximum heat transfer at lower pumping loss mass flow rate should be low.
- Plate thickness should be minimum to survive thermal and working stress.
- Plate spacing can be adjusted for increased mass flow rate but it should be kept minimum.
- Plate width should be kept minimum as per the given discharge of process fluid.
- Plate length should be as high as possible for maximizing heat transfer rate.
- The outcomes of this analysis and simulation has been validated by analysis of Selin and Sadik [6].

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