Experimental Study for Double Pipe Heat Exchanger with Rotating Inner Pipe

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ABSTRACT
The present study aims to investigate experimentally fluid flow and performance characteristics of a double-pipe heat exchanger with rotating inner tube. Parameters that can be used to measure the performance of this type of heat exchanger are also presented, investigated and estimated. The experimental results are reported for the effect of cold and hot water mass flow rates, the heat exchanger arrangement (parallel or counter) and the rotation speed on NTU and effectiveness of the heat exchanger. This study was done for $0 \le N \le 1000$ R.P.M, $0.022 \le m_c \le 0.09$ kg/s and $0.022 \le m_h \le 0.09$ kg/s.

Key words: Heat exchanger, Effectiveness, NTU, Rotating Pipe.

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INTRODUCTION

Heat exchanger is a piece of equipment built for efficient heat transfer from one medium to another. They are widely used in space heating, refrigeration, air conditioning, power plants, chemical plants, petrochemical plants, petroleum refineries, natural gas processing, and sewage treatment. The classic example of a heat exchanger is found in an internal combustion engine in which a circulating fluid known as engine coolant flows through radiator coils and air flows past the coils, which cools the coolant and heats the incoming air. Heat transfer in rotating systems has been the subject of many experimental and theoretical studies. Many engineering applications involve rotating machinery components with flow in an annulus formed between two concentric cylinders where one or both of the cylindrical surfaces is or are rotating one. Kim and Hwang [1] studied the experimental concerns the characteristics of vortex flow in a concentric annulus with a diameter ratio of 0.52, whose outer cylinder was stationary and inner one was rotating. Taylor [2, 3] performed analytical and experimental works to predict flow, and thermal fields, stability, heat and mass transfer characteristics, etc., inside the concentric annular space. Mathew and Hegab [4] developed an analytical solution for counter flow micro channel heat exchanger subjected to external heat flux while operating under balanced and unbalanced flow conditions. With the addition of heat from the external heat source, the effectiveness of hot and cold A heat transfer characteristic of laminar flow in a circular annulus with a rotating inner cylinder in the presence of a laminar axial flow was investigated using sublimation techniques by Molki et al. [5]. The work focused on the entrance region of the annulus with simultaneous development of velocity and temperature profiles. Lei et al. [6] studied the existence of hydrodynamic instabilities leads to the formation of Taylor vortices in flows in the annulus between two concentric cylinders with one or both cylinders rotating.

Bouafia et al. [7] studied the convective heat transfer between the walls of an annular gap with a rotating inner cylinder. Two geometrical configurations were analyzed: the surfaces of the cylinders were either smooth, or the moving wall was smooth or the other is axially grooved. Hot-wire measurements were presented by Greaves et al. [8] of the onset of instability in developed axial flow and in both developing and developed tangential flow caused by inner cylinder rotation in concentric annuli of radius. Numerical experiments are performed by Lee [9] to study the effects of the convective fluid motion of air enclosed between the annuli of eccentric horizontal cylinders. Pfitzer and Beer [10] studied experimentally and analytically the effects of the rotating inner and outer tube on the turbulent fluid flow and heat transfer in a concentric annulus. In the experimental investigations the heat transfer rate in the hydrodynamic and thermal entrance region of the rotating annulus and the velocity and temperature profiles at the end of the test section were determined. Al-Sadah [11] studied numerically the convective heat transfer in a gap formed by vertical concentric cylinders. Wan and Coney [12] studied in spiral vortex flow, between concentric cylinders with the inner cylinder rotating and the outer stationary, the addition of a thermal gradient across the gap is a known complicating factor. The heat transfer characteristics and the modes of transition have been investigated together with the relationship between them. Hsu [13] studied the intended to provide an inverse approach for estimating the viscosity of fluid and thermal behavior of the concentric cylinders. Finite-difference methods are first employed to discredit the problem domain and then a linear inverse model is constructed to identify the condition for the viscosity of fluid and thermal behavior of the concentric cylinders. The transient buoyant rotating convective flow and heat transfer in a tall vertical annulus containing cold water near the density inversion have been investigated via a finite difference procedure by Ho and Tu [14]. Simulations are carried out by solving axi-symmetric Navier-Stokes equations adhering to the Boussinesq approximation coupled to the energy equation. Numerical results demonstrate that the transient mixed convective flow and heat transfer may evolve into sustained oscillation over a certain range of Rayleigh number. Nouri and Whitelaw [15] measured the three mean velocity components and the associated Reynolds stress tensor of the flow subjected to an axial superimposed through flow in a concentric annulus with or without rotation of the inner cylinder. Three velocity components of a Newtonian and a weakly elastic shear-thinning non-Newtonian fluid have been measured in an annulus by Nouri and Whitelaw [16]. They showed that the rotation had similar effects on the Newtonian and non-Newtonian fluids, with a more uniform axial flow across the annulus and the maximum tangential velocities in the narrowest gap in both cases. Mahmud and Fraser [17] presented analytical the first and second laws of thermodynamics characteristics of fluid flow and heat transfer inside a cylindrical annulus. Dou et al [18] presented distribution of energy loss due to viscosity friction in plane Couette flow and Taylor–Couette flow between concentric rotating cylinders. Lepiller et al [19] investigated the influence of a weak radial temperature gradient in a wide gap and large aspect ratio Couette-Taylor system. The inner cylinder is rotating and can be heated or cooled, the outer cylinder is at rest and immersed in a large thermal bath. Jeng et al [20] presented experimentally the heat transfer characteristics of Taylor–Couette Poiseuille flow in an annular channel by mounting longitudinal ribs on the rotating inner cylinder. Michiyoshi et al [21] presented heat transfer in a fluid, with uniformly distributed internal heat source, flowing upwards through a vertical tube. Measured were made of the temperature distribution in both laminar and turbulent flow, and both with and without heat transfer at wall. Heat generation within the fluid was brought about by passing an electrical current through the working fluid, which was an aqueous solution of sodium chloride. Poncet et al [22] presented consider turbulent flows in a differentially heated Taylor–Couette system with an axial Poiseuille flow. Dou et al [23] used the energy gradient theory to study the instability of Taylor–Couette flow between concentric rotating cylinders. Ravanchi et al [24] derived an approximate analytical solution for the steady state, purely tangential flow of a viscoelastic fluid obeying the Giesekus constitutive equation in a concentric annulus with inner cylinder rotation. Monde et al [25] measured critical heat flux during natural circulation boiling of water on uniformly heated outer tube in vertical annular tube. The experiment was carried out using water at atmospheric pressure. Yildiz et al [26] gave heat transfer correlations in terms of Nusselt, Reynolds and Rossby numbers, the flow mode (parallel or countercurrent) and the inner rotating-pipe surface area. Li et al [27] studied transition to Taylor vortex flow between two conical cylinders, with the inner one rotating and the outer one stationary by the numerical method. Escudier et al [28] presented a computational and experimental study of fully developed laminar flow of a Newtonian liquid through an eccentric annulus with combined bulk axial flow and inner cylinder rotation. Batten et al [29] presented turbulent Taylor vortices between two concentric at a very high radius ratio. Venkatachalappa et al [30] presented a numerical study to understand the effect of rotation on the axi-symmetric flow driven by buoyancy in an annular cavity formed by two concentric vertical cylinders which rotate about their axis with different angular velocities. The present study aims to experimentally investigate fluid flow and performance characteristics of a double-pipe heat exchanger with rotating inner tube. Parameters that can be used to measure the performance of this type of heat exchanger are also presented, investigated and estimated. The effects of fluid flow parameters on the performance of the heat exchanger are also investigated. We hope that we can present some guidelines that will be helpful in the design of such kind of heat exchangers in the different engineering applications.

EXPERIMENTAL SET UP AND MEASURING TECHNIQUE

A schematic view of the experimental setup is shown in Fig.1.Basically; the setup consists of two vertical pipe of different diameters and same lengths, and 4 tanks. Hot water was passed though the inner pipe, while cold water was passing though the annulus space. The hot fluid is hot water, which is obtained from an electric geyser. Hot water flows through the inner tube, in one direction. Cold fluid is cold water, which flows through the annulus. Control valves are provided so that direction of cold water can be kept parallel or opposite to that of hot water. Thus, the heat exchanger can be operated either as parallel or counter flow heat exchanger. The temperatures are measured by thermocouples. The temperatures of water at the inlets of the hot and cold sides respectively are $T_{h,in}$ and $T_{c,in}$, whereas $T_{h,out}$ and $T_{c,out}$ are the temperatures at the outlets. The outer surface of the heat exchanger was insulated by rubber. The experimental set up is designed to enable varying and controlling the water flow rates and the variables speed inner pipe. Different instruments were incorporated in the experimental set up to enable measuring the different parameters such as temperatures, flow rate, and heat transfer rate. These measurements enabled studying the effect of different flow parameters such as Reynolds number, Prandtl number as well as the heat transfer rate input and heat transfer characteristics as well as on the performance of the vertical a double pipe heat exchanger. The experimental set up can be divided into three main sections; namely the water circuit and the test section and the electrical section.

WATER THERMODYNAMIC PROPERTIES

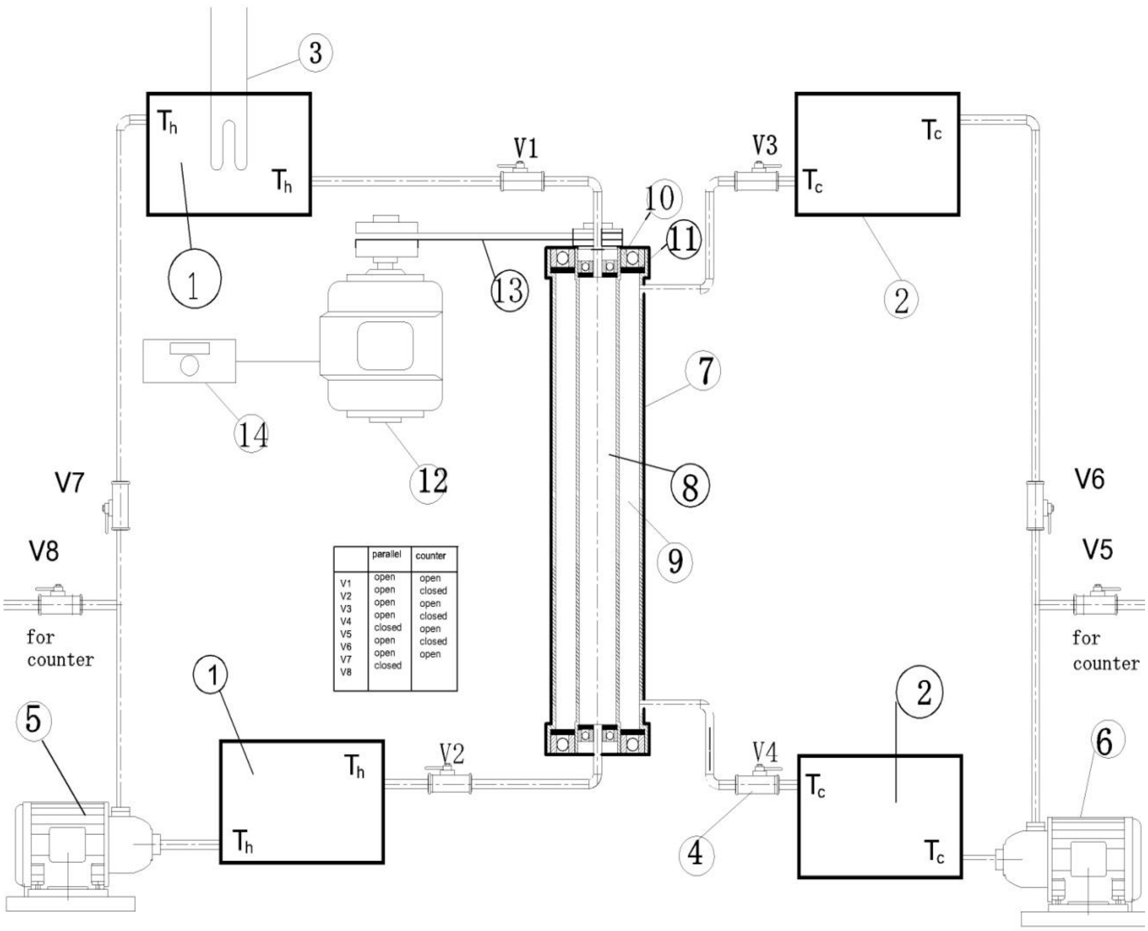
In the experimental study the working fluid is water for both hot and cold fluids and so for simplistic of the analysis, computer program was developed to calculate the all parameters that led to the heat transfer results such that Reynolds number, Prandtl number and Nusselt number, so the thermodynamic properties of water were correlated by us as a function of the average water temperature.

Table1. Correlations of Thermodynamic Properties of Water

property	*correlation*	*% error*
C_p	$4222 - 3.179T + 0.07059\,T^2 + -0.0006647\,T^3 + 3.231*10^{-6}\,T^4$	*0.068*
μ	$9.602/(5365 + 190.7\,T + T^2)$	*5*
k	$0.5655 + 0.002093\,T - 1.093*10^{-5}\,T^2 + 1.498*10^{-8}\,T^3$	*0.35*
ρ	$1002 - 0.1636\,T - 0.002642\,T^2$	*0.224*

While the Prandtl number is calculated from equation

$$Pr = \frac{\mu\, C_p}{k}$$



1	Hot Water Tank	6	Cold Water pump	11	Oil seal
2	Cold Water Tank	7	Insulation	12	Electric motor 3 ph
3	Heater	8	Inner pipe	13	Pelt drive
4	Ball valve	9	Outer Pipe	14	Inverter
5	Hot Water Pump	10	Bearing		

Fig.1: Schematic Diagram for Experiment Setup

SOLUTION PROCEEDING AND DATA REDUCTION

- Fixation of the hot water mass flow rate at 0.022 kg/s.

- Changing the cold water mass flow rate from 0.022 kg/s to 0.09 kg/s
- Changing the inner pipe speed rotation from 0 to 1000 R.P.M.
- Measurement of the hot and cold fluid temperatures.
- Check on the heat transfer between hot and cold one as a heat balance.
- Evaluation of the hot and cold water properties from correlations in Table 1 as function of the water temperatures.
- Calculation of heat transfer coefficient for both hot and cold fluids and hence overall heat transfer coefficient.
- Calculation of minimum and maximum heat capacity and hence the heat capacity ratio between hot and cold fluids.
- Calculation of NTU number.
- Calculation of heat exchanger effectiveness as a function of NTU.
- The previous scenario is done for both parallel and counter flow.
- Repeat the above procedures with fixation of the cold water mass flow rate at 0.022 kg/s and changing the hot water mass flow rate from 0.022 kg/s to 0.09 kg/s.
- Drawing the relation between mass flow rate for both cold and hot fluid and both NTU and effectiveness for parallel and counter flows.

RESULTS AND DISCUSSION

PARALLEL FLOW HEAT EXCHANGER WITH VARIABLE SPEED ROTATION

The results of the experimental data for variable speed rotation from 0 to 1000 R.P.M are represented in Figs.2 to 27; the results mainly include the effectiveness and the NTU number.
m_h=0.022 kg/s: The results of NTU and effectiveness (ε), as a function of cold fluid mass flow rate m_c are represented as m_c increases from 0.022 kg/s to 0.09 kg/s, with constant value of R.P.M the NTU is affected by two values, overall heat transfer coefficient (U) and minimum heat capacity $(mc)_{min}$, as the cold water mass flow (m_c) increases both overall heat transfer coefficient (U) and minimum heat capacity $(mc)_{min}$ increase , but the effect of cold water mass flow (m_c) is significant on $(mc)_{min}$ rather than the effect on overall heat transfer coefficient (U) this is because that the increase in mass flow is very small from 0.022 kg/s to 0.09 kg/s. If $m_c \leq 0.022$ kg/s the value of $(mc)_{min}$ is equal to $(mc)_{cold}$ with small heat transfer coefficient and hence high NTU value decreasing with the increasing of m_c till 0.022 kg/s . If $m_c \geq 0.022$ kg/s the value of $(mc)_{min}$ is equal to $(mc)_{hot}$ which is constant due to the hot fluid mass flow is held constant at 0.022 kg/s with small heat transfer coefficient and hence NTU value is held constant with the increasing of m_c till 0.09 kg/s. it must be observed that the minimum value of NTU take place if m_c is equal to m_h with value of 0.022 kg/s i.e. C_r equal to unity. The same observation is found with increasing the inner pipe revolution (R.P.M) but the NTU increased with noticeable values, this is due to the successive increase in overall heat transfer coefficient because the successive increase in Reynolds number which take into account the rotational velocity due to rotation and linear velocity due to the flow inside the channels. The relation between the effectiveness and NTU is proportional relation, If $m_c \leq 0.022$ kg/s the value of $(mc)_{min}$ is equal to $(mc)_{cold}$ and the heat capacity of the hot fluid is constant lead to decreasing in the effectiveness . If $m_c \geq 0.022$ kg/s the value of $(mc)_{min}$ is equal to $(mc)_{hot}$ which is constant due to the hot fluid mass flow is held constant at 0.022 kg/s but the cold fluid heat capacity increased lead to low temperature difference between hot and cold fluids and hence high effectiveness. The same observation is found with increasing the inner pipe revolution (R.P.M) but the effectiveness increased with noticeable values, this is due to the successive increase in overall heat transfer coefficient because the successive increase in Reynolds number which take into account the rotational velocity due to rotation and linear velocity due to the flow inside the channels.

m_c=0.022 kg/s: The same results are found as the case of changing the cold fluid mass flow rate, but the division between the results due to the change of heat transfer coefficient with the same heat capacities, the inside heat transfer coefficient is held constant with constant mass flow rate but the outside heat transfer coefficient changing with the changing of mass flow rate but not identical with the case of changing the cold water mass flow. In the case of changing the cold water flow, the hydraulic diameter is equal to the inside pipe diameter but in the case of changing the hot fluid flow with the same flow, the heat transfer changes due to the change of the hydraulic diameter which is equal to the difference between the outside pipe diameter and inside pipe diameter (annuals).

COUNTER FLOW HEAT EXCHANGER WITH VARIABLE SPEED ROTATION

The results of the experimental data for variable speed rotation from 0 to 1000 R.P.M are represented in Figs.28 to 53; the results mainly include the effectiveness and the NTU number. The same discussion as in parallel flow but with some division will be discussed in the comparison.

COMPARISON BETWEEN PARALLEL AND COUNTER FLOW HEAT EXCHANGER

Comparison between parallel and counter flow arrangement is represented in Figs.54 to 57 for NTU and effectiveness at 0 R.P.M. The comparison shows that the counter flow is most efficient than parallel one due to the temperature difference between hot and cold fluids for counter flow is small than parallel flow and hence the high heat transfer rate with the same surface area. The same behavior is for different revolutions of the inner pipe and so, the using of rotating pipe heat exchanger is favorable especially if the arrangement is counter one.

CONCLUSION

The results of the experimental data for variable speed rotation from 0 to 1000 R.P.M are represented; the results mainly include the effectiveness and the NTU number. The hot water mass flow rate was held constant at 0.022 kg/s and changing the cold water mass flow rate from 0.022 kg/s to 0.09 kg/s. Measurement of the hot and cold fluid temperatures was done. Good correlations for water properties to simplify the calculations were executed. The results showed that:

- For small mass flow rates for both hot and cold fluids the NTU is high with decreasing value with the increasing of the mass flow rate.
- At equal heat capacity the NTU values are minimum values.
- The NTU value is constant after the minimum value until if one of the hot or cold fluids increases.
- The effectiveness for both parallel and counter flow has minimum value at heat capacity ratio equal to unity but with higher values else.
- The speed of rotation increases the Reynolds number and hence heat transfer rate.
- The speed of rotation increases the NTU and effectiveness values.

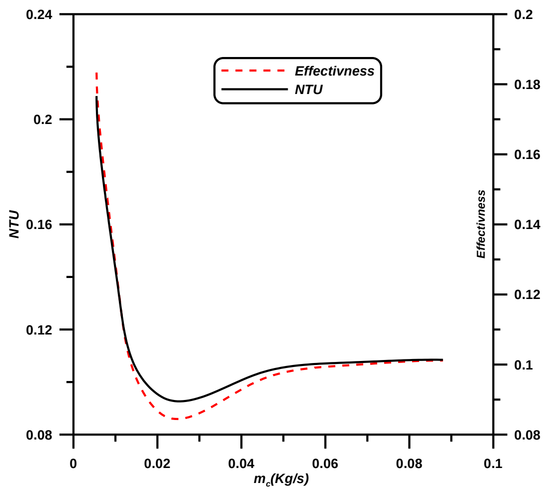


Fig.2:Parallel Flow m_h=0.022 kg/s at 0 R.P.M

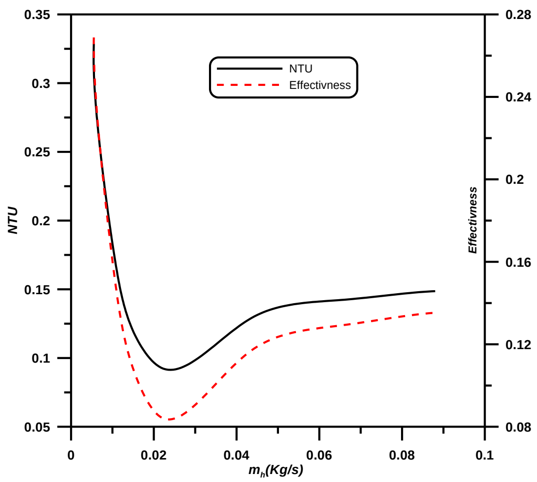


Fig.3:Parallel Flow m_c=0.022 kg/s at 0 R.P.M

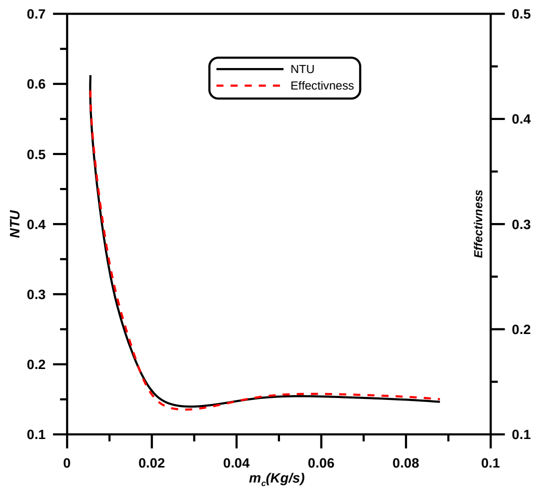


Fig.4:Parallel Flow m_h=0.022 kg/s at 25 R.P.M

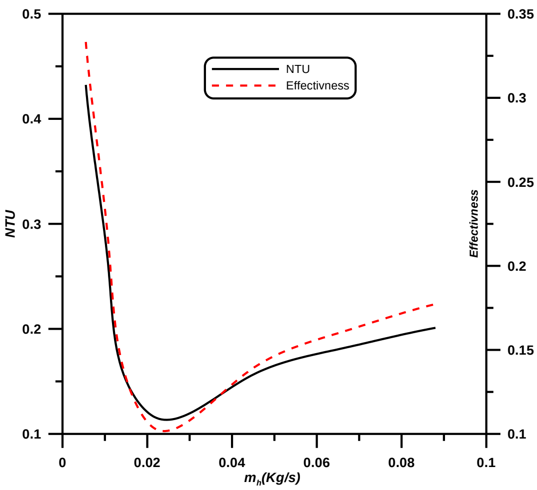


Fig.5:Parallel Flow m_c=0.022 kg/s at 25 R.P.M

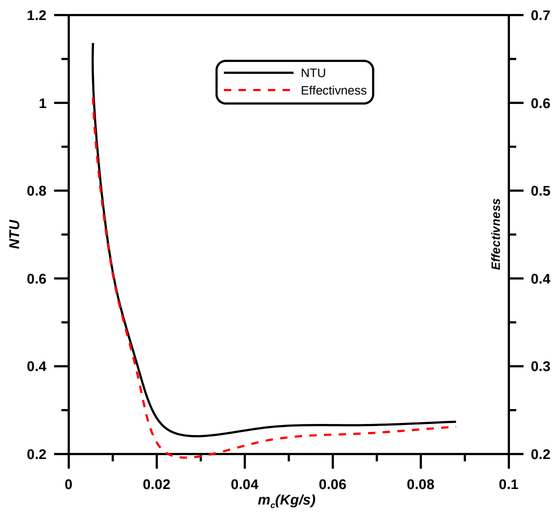


Fig.6:Parallel Flow m_h=0.022 kg/s at 50 R.P.M

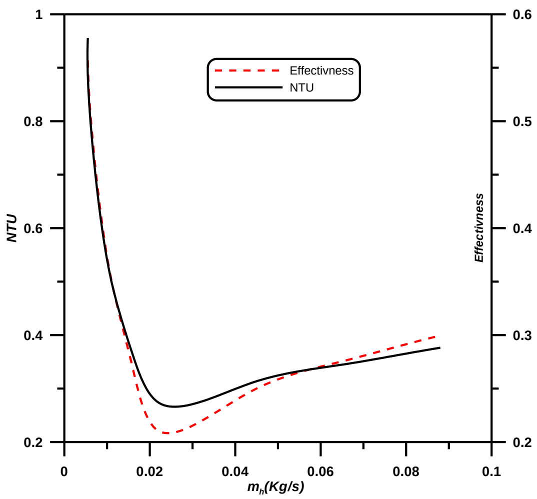


Fig.7:Parallel Flow m_c=0.022 kg/s at 50 R.P.M

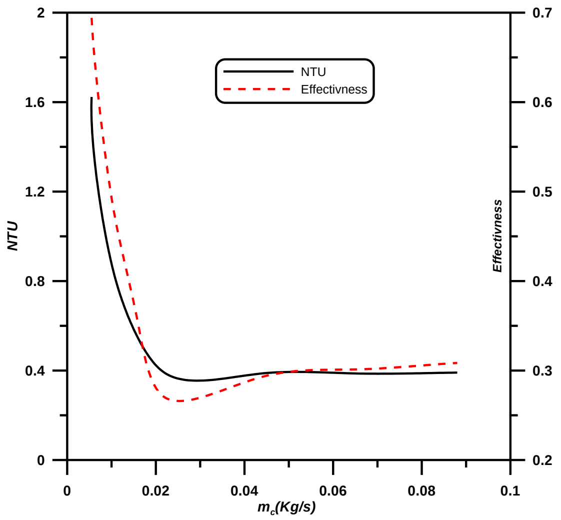


Fig.8:Parallel Flow m_h=0.022 kg/s at 75 R.P.M

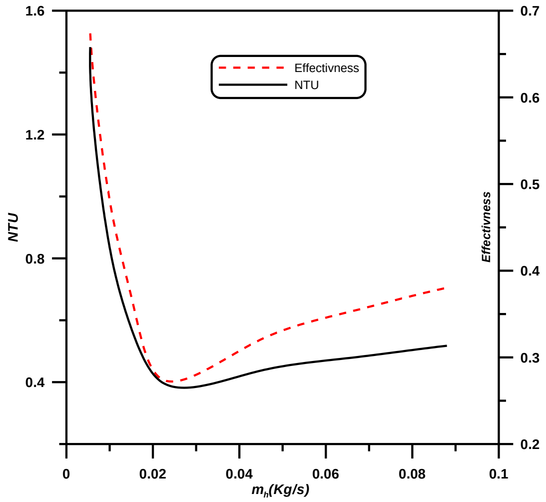


Fig.9:Parallel Flow m_c=0.022 kg/s at 75 R.P.M

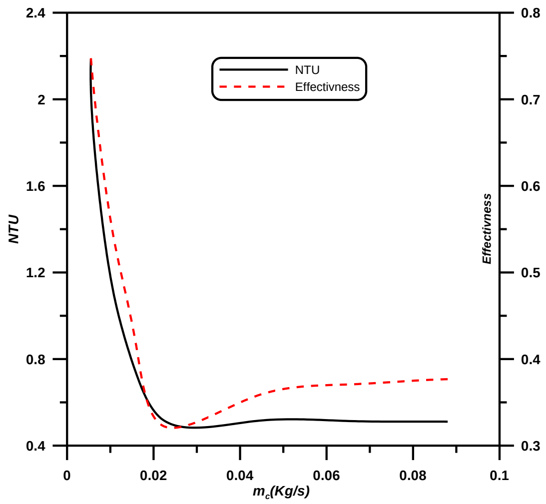


Fig.10:Parallel Flow m_h=0.022 kg/s at 100 R.P.M

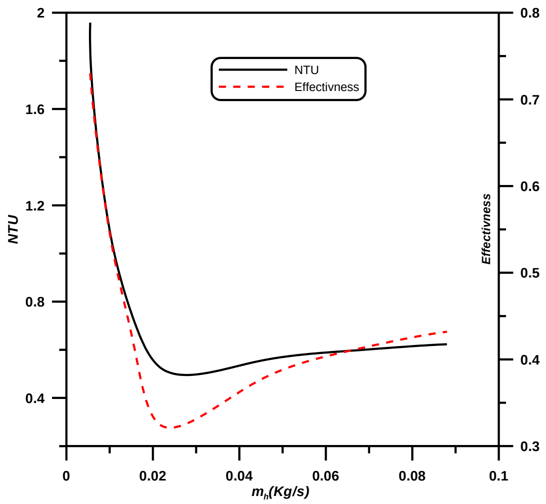


Fig.11:Parallel Flow m_c=0.022 kg/s at 100 R.P.M

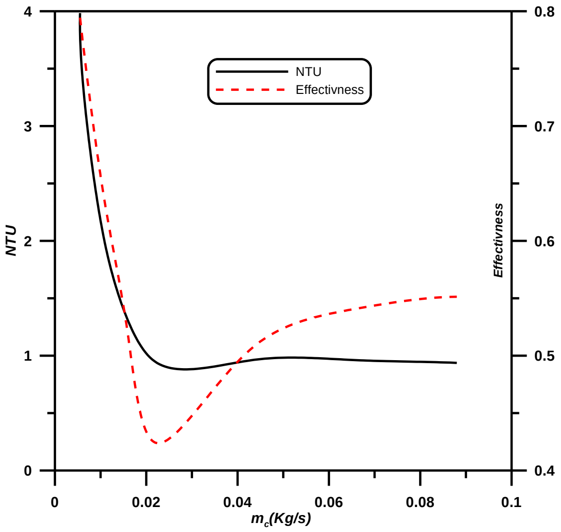


Fig.12:Parallel Flow m_h=0.022 kg/s at 200 R.P.M

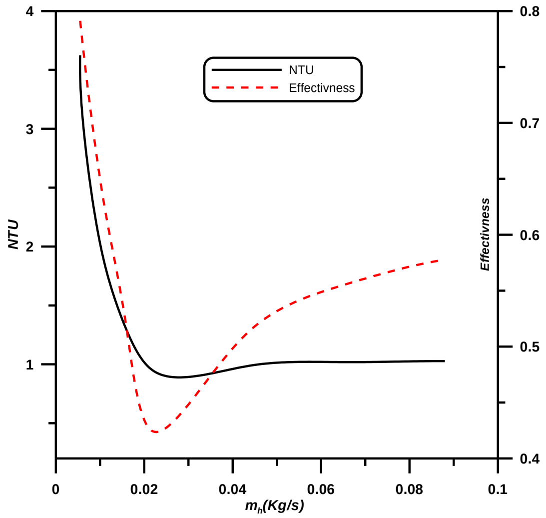


Fig.13:Parallel Flow m_c=0.022 kg/s at 200 R.P.M

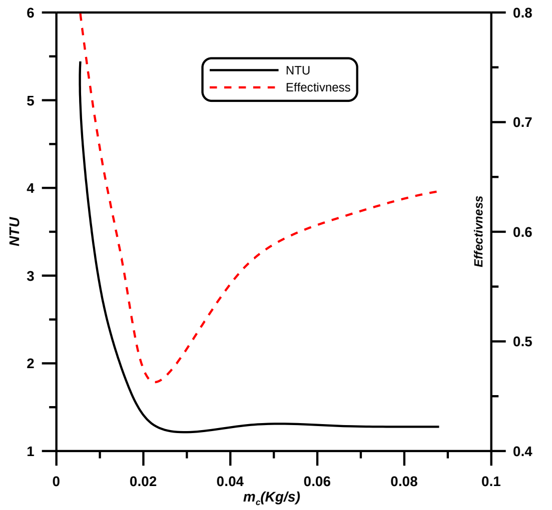


Fig.14:Parallel Flow m_h=0.022 kg/s at 300 R.P.M

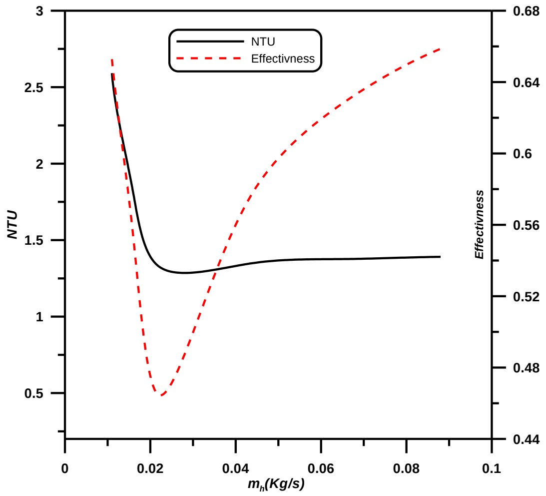


Fig.15:Parallel Flow m_c=0.022 kg/s at 300 R.P.M

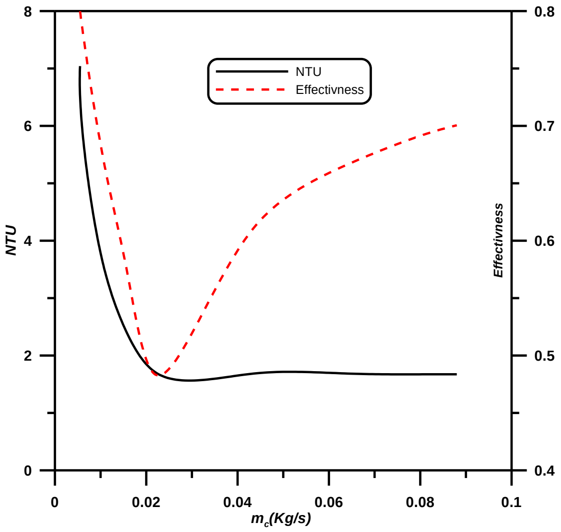


Fig.16:Parallel Flow m_h=0.022 kg/s at 400 R.P.M

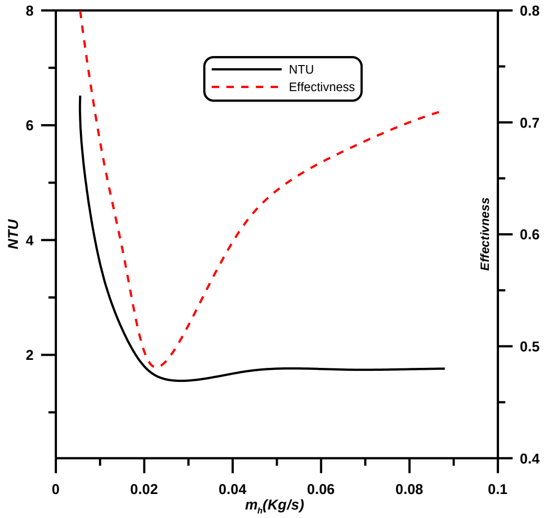


Fig.17:Parallel Flow m_c=0.022 kg/s at 400 R.P.M

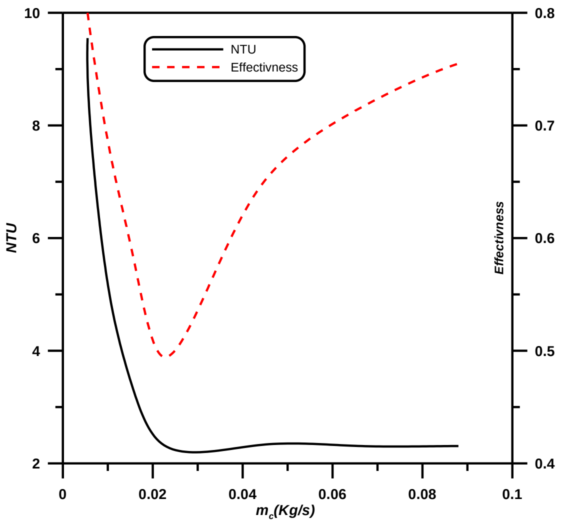


Fig.18:Parallel Flow m_h=0.022 kg/s at 600 R.P.M

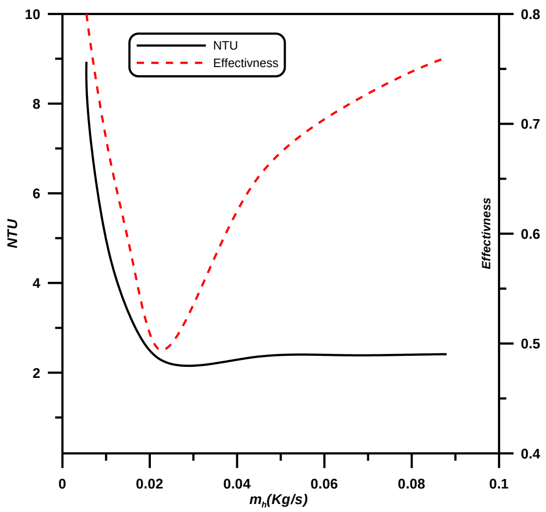


Fig.19:Parallel Flow m_c=0.022 kg/s at 600 R.P.M

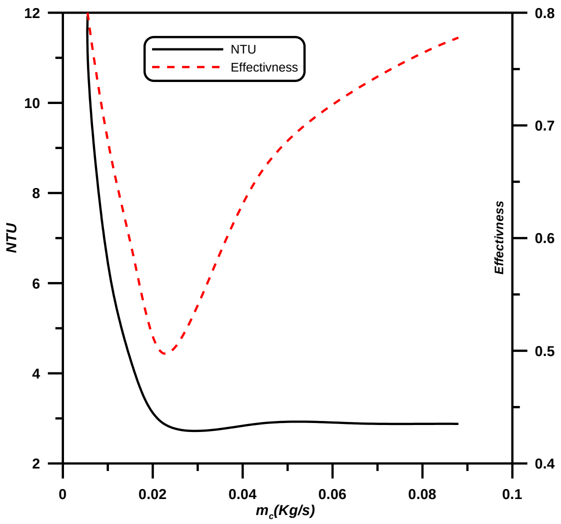


Fig.20:Parallel Flow m_h=0.022 kg/s at 800 R.P.M

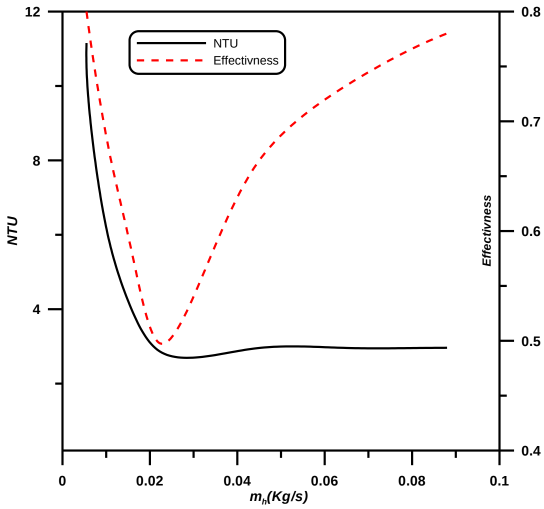


Fig.21:Parallel Flow m_c=0.022 kg/s at 800 R.P.M

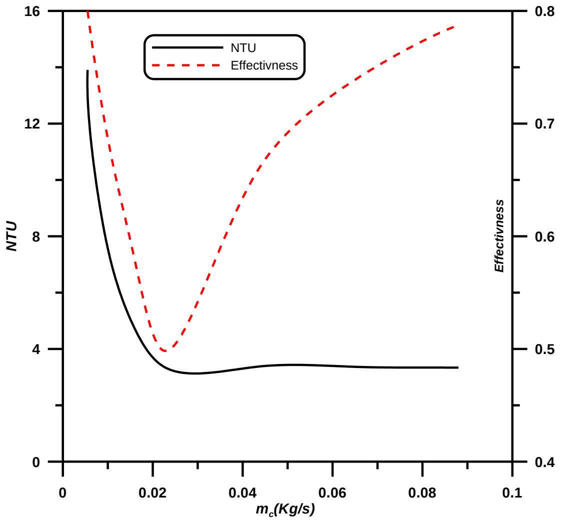


Fig.22:Parallel Flow m_h=0.022 kg/s at 1000 R.P.M

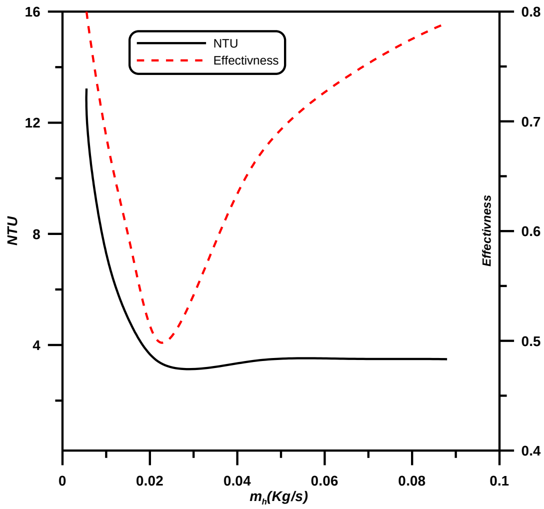


Fig.23:Parallel Flow m_c=0.022 kg/s at 1000 R.P.M

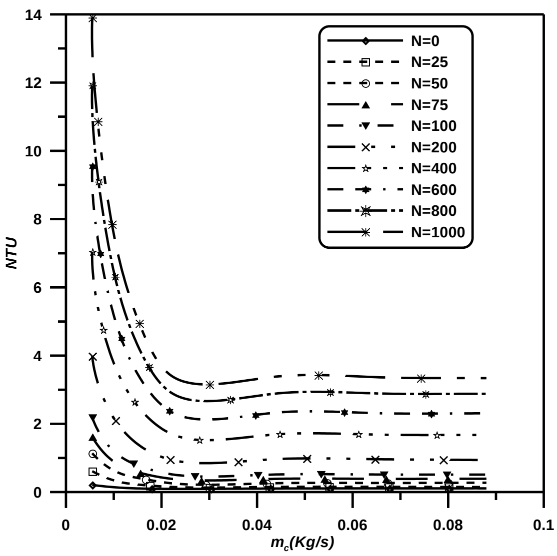


Fig.24:Parallel Flow m_h=0.022 kg/s

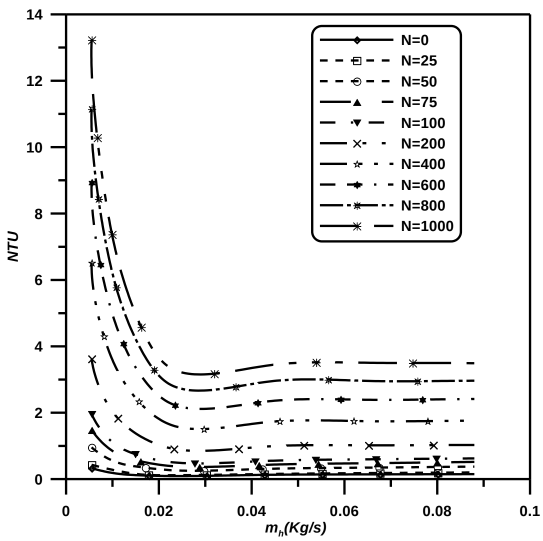


Fig.25:Parallel Flow m_c=0.022 kg/s

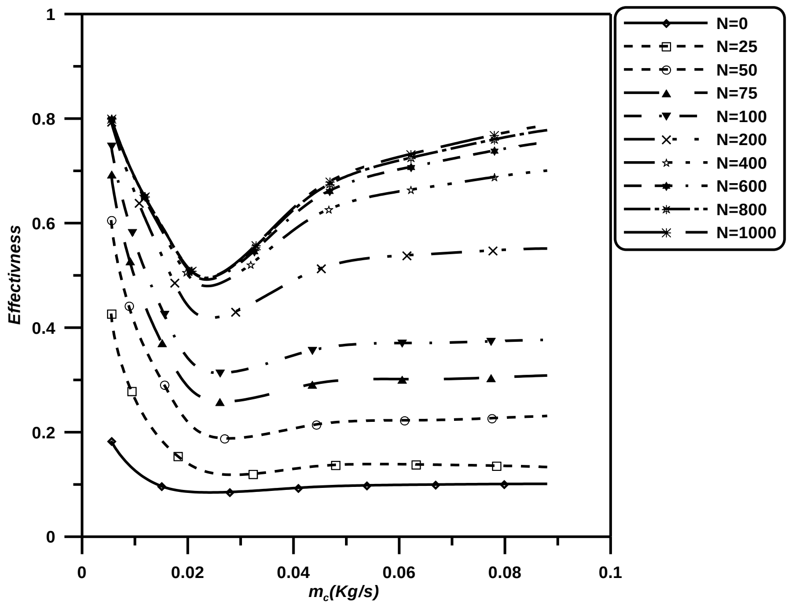


Fig.26:Parallel Flow m_h=0.022 kg/s

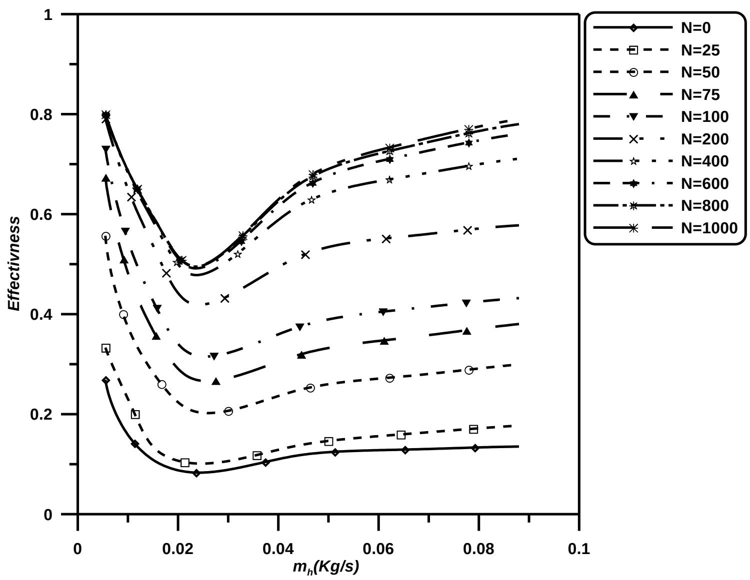
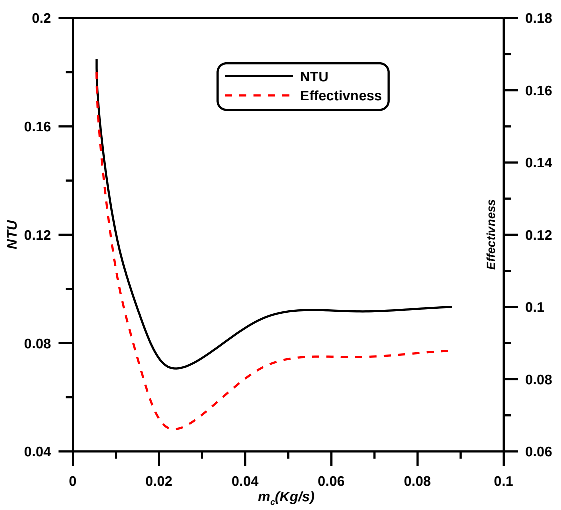


Fig.27:Parallel Flow m_c=0.022 kg/s



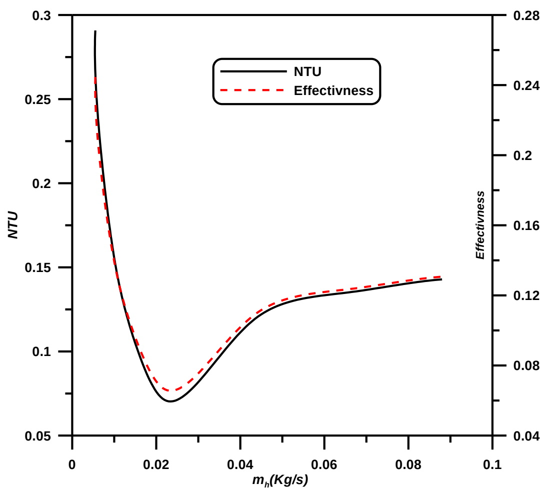
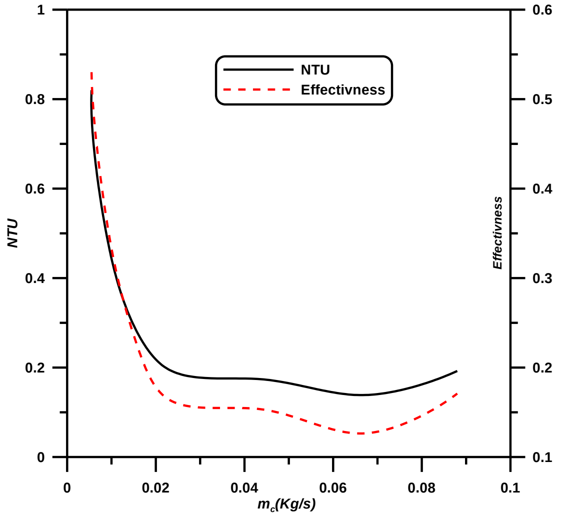


Fig.28:Counter Flow m_h=0.022 kg/s at 0 R.P.M

Fig.29: Counter Flow m_c=0.022 kg/s at 0 R.P.M



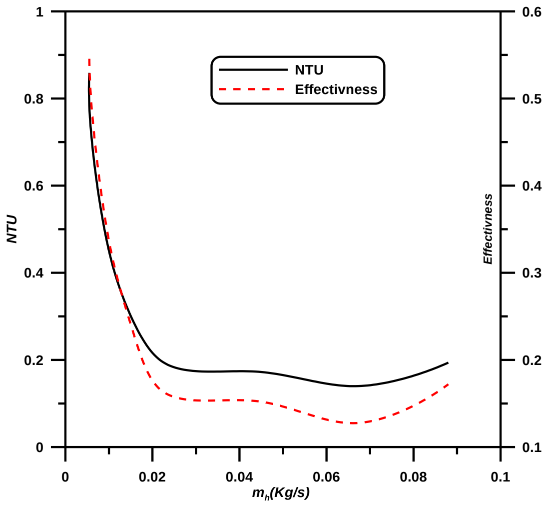
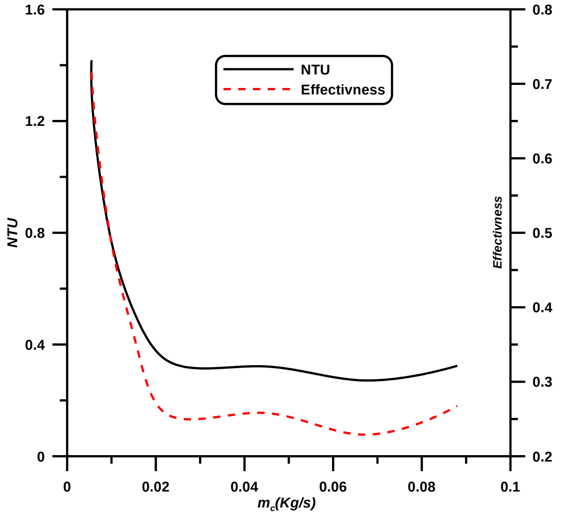


Fig.30: Counter Flow m_h=0.022 kg/s at 25 R.P.M

Fig.31: Counter Flow m_c=0.022 kg/s at 25 R.P.M



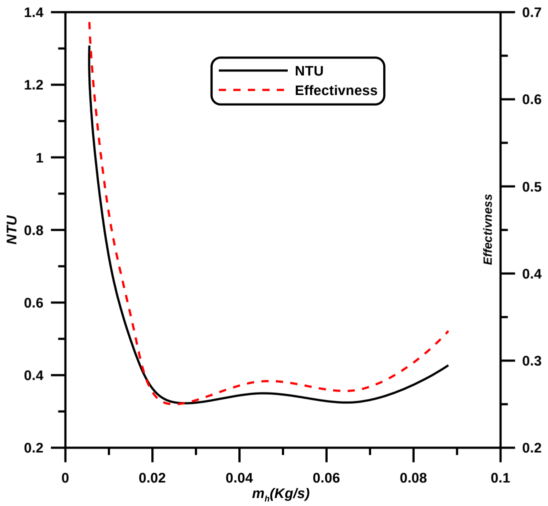
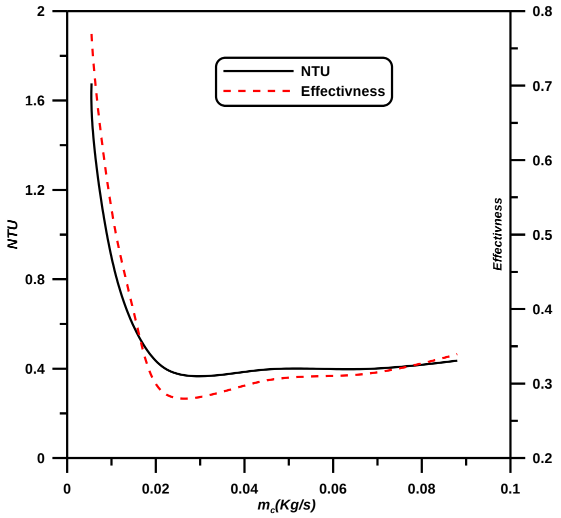


Fig.32: Counter Flow m_h=0.022 kg/s at 50 R.P.M

Fig.33: Counter Flow m_c=0.022 kg/s at 50 R.P.M



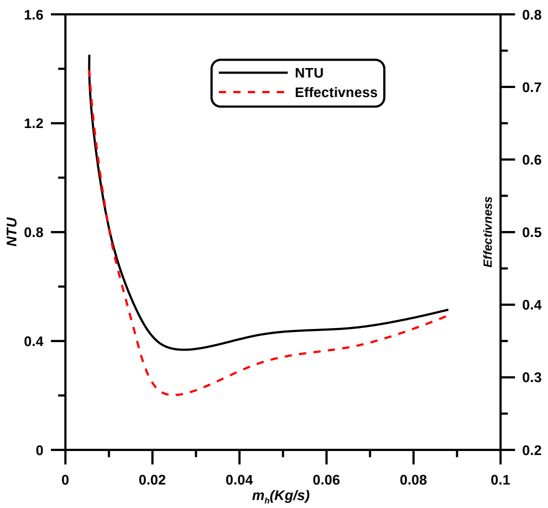


Fig.34: Counter Flow m_h=0.022 kg/s at 75 R.P.M

Fig.35: Counter Flow m_c=0.022 kg/s at 75 R.P.M

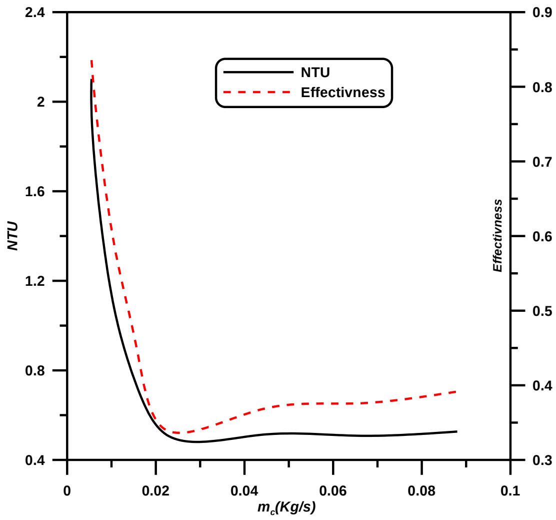


Fig.36: Counter Flow m_h=0.022 kg/s at 100 R.P.M

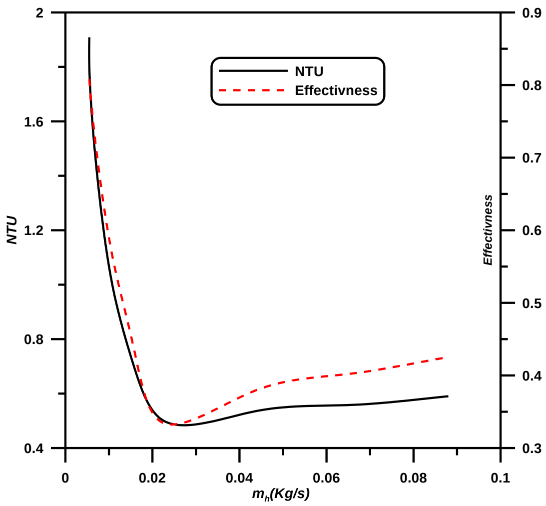


Fig.37: Counter Flow m_c=0.022 kg/s at 100 R.P.M

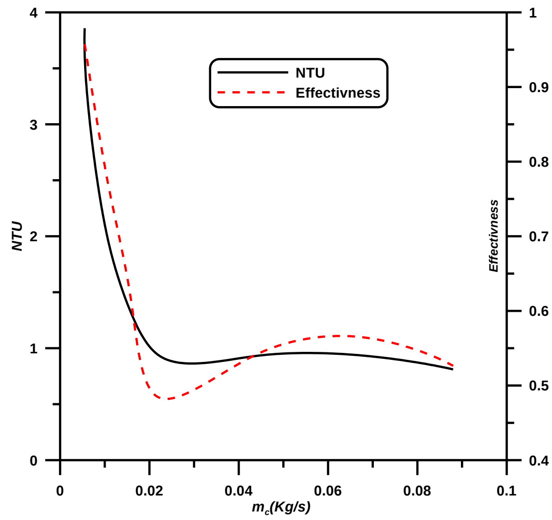


Fig.38: Counter Flow m_h=0.022 kg/s at 200 R.P.M

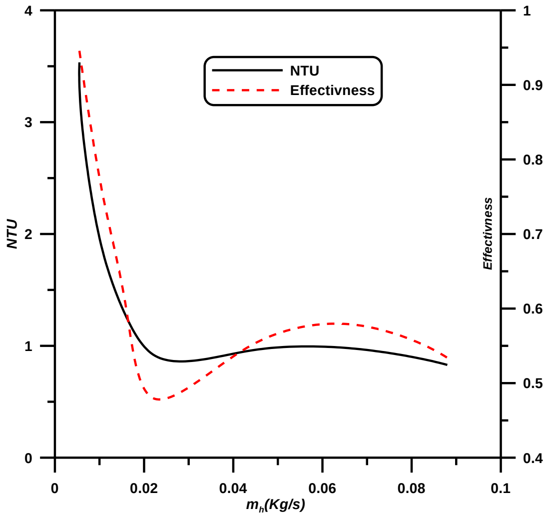


Fig.39: Counter Flow m_c=0.022 kg/s at 200 R.P.M

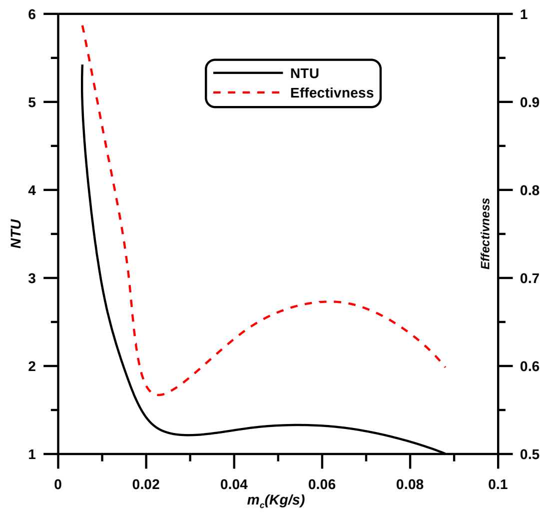


Fig.40: Counter Flow m_h=0.022 kg/s at 300 R.P.M

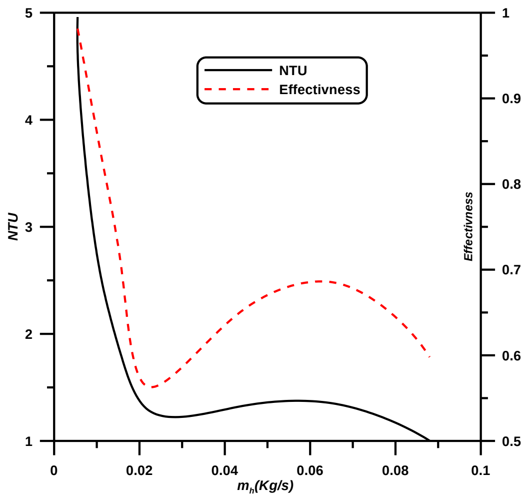
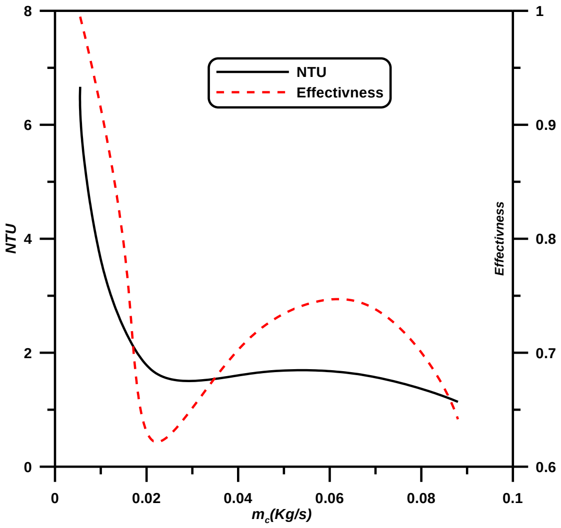


Fig.41: Counter Flow m_c=0.022 kg/s at 300 R.P.M



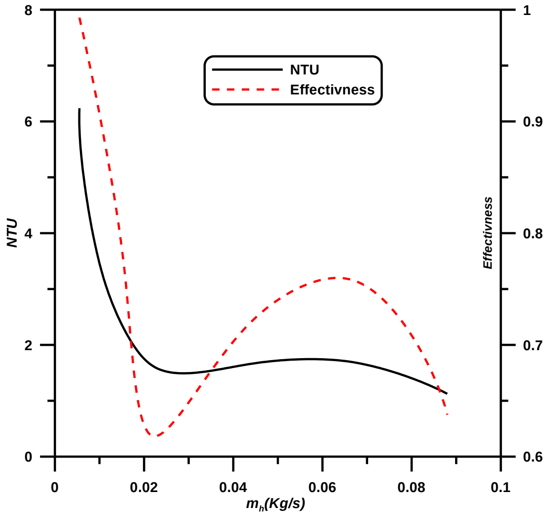
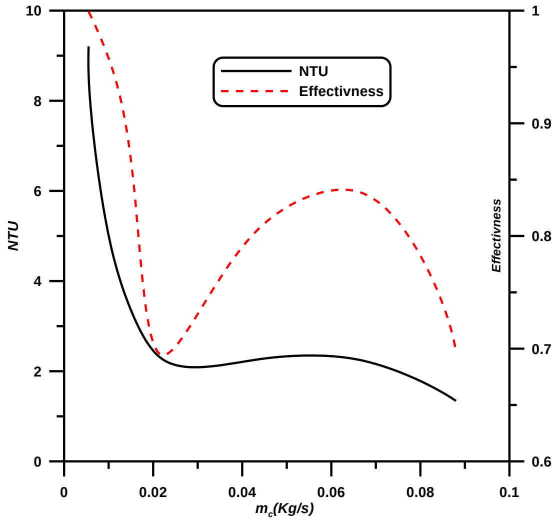


Fig.42: Counter Flow m_h=0.022 kg/s at 400 R.P.M

Fig.43: Counter Flow m_c=0.022 kg/s at 400 R.P.M



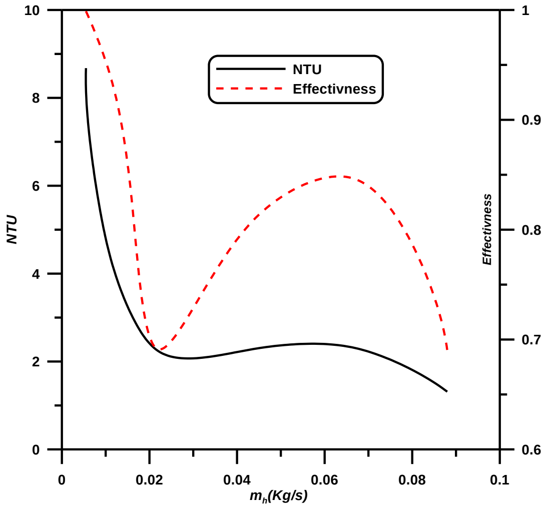
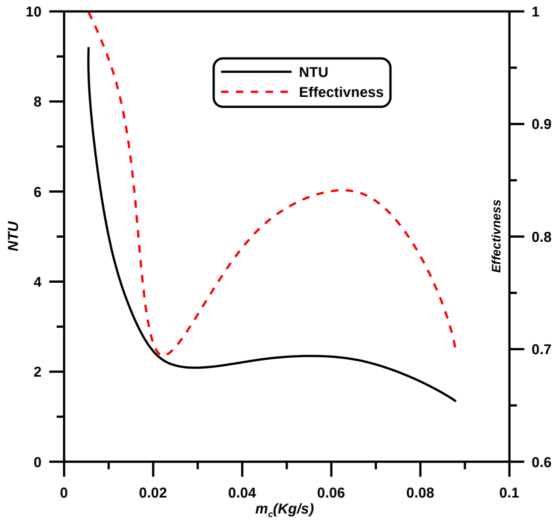


Fig.44: Counter Flow m_h=0.022 kg/s at 600 R.P.M

Fig.45: Counter Flow m_c=0.022 kg/s at 600 R.P.M



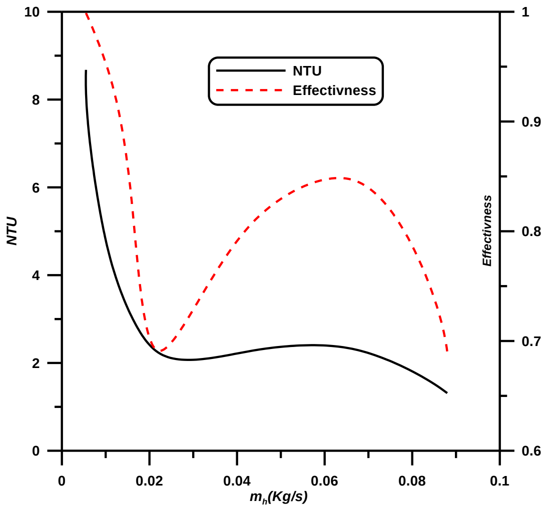


Fig.46: Counter Flow m_h=0.022 kg/s at 800 R.P.M

Fig.47: Counter Flow m_c=0.022 kg/s at 800 R.P.M

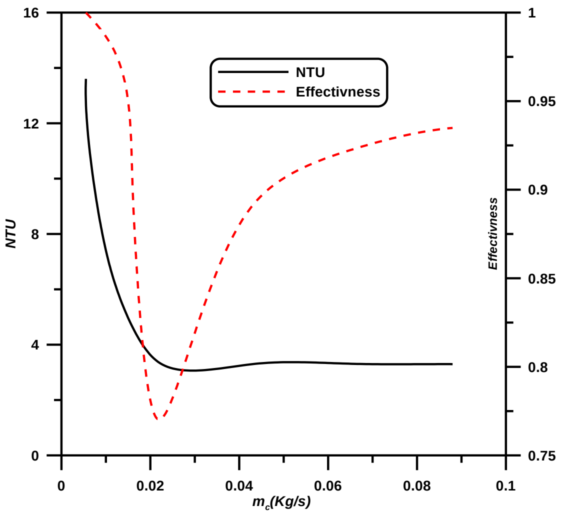


Fig.48: Counter Flow m_h=0.022 kg/s at 1000 R.P.M

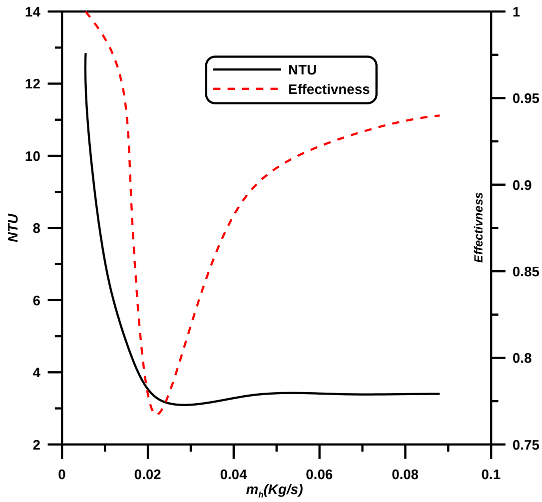


Fig.49: Counter Flow m_c=0.022 kg/s at 1000 R.P.M

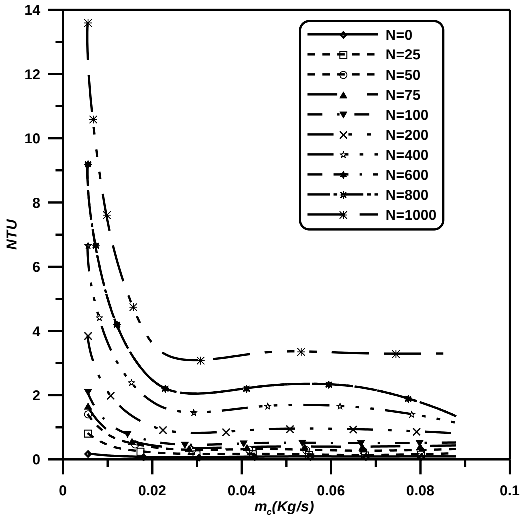


Fig.50: Counter Flow m_h=0.022 kg/s

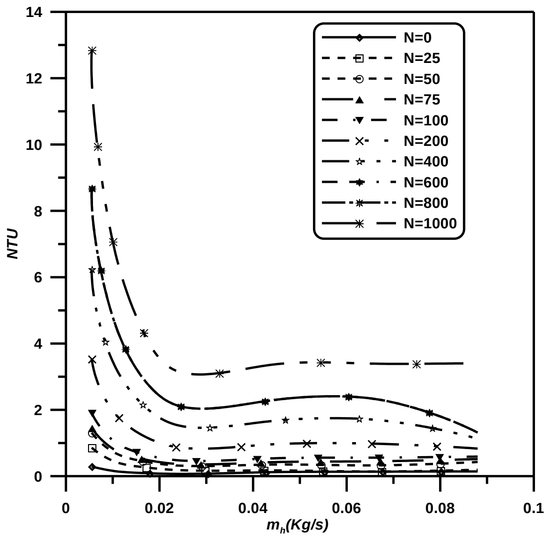


Fig.51: Counter Flow m_c=0.022 kg/s

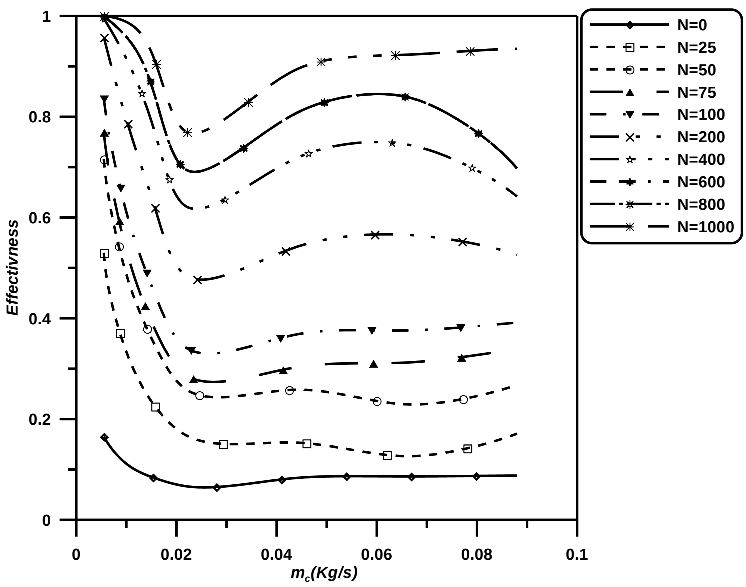


Fig.52: Counter Flow m_h=0.022 kg/s

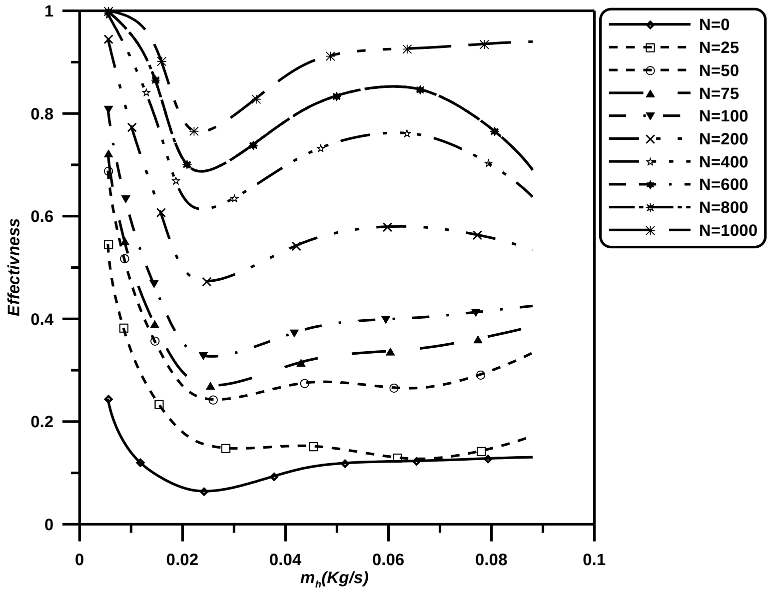
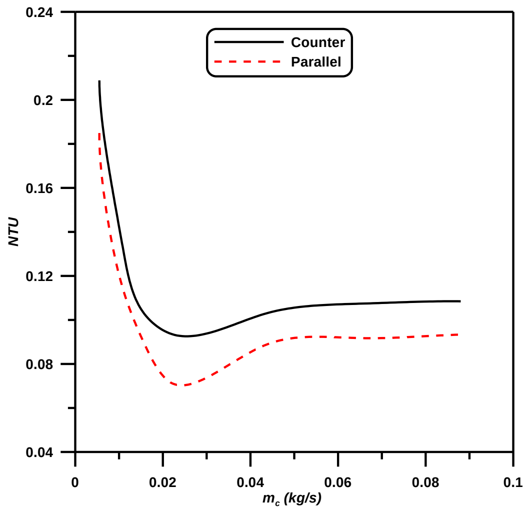


Fig.53: Counter Flow m_c=0.022 kg/s



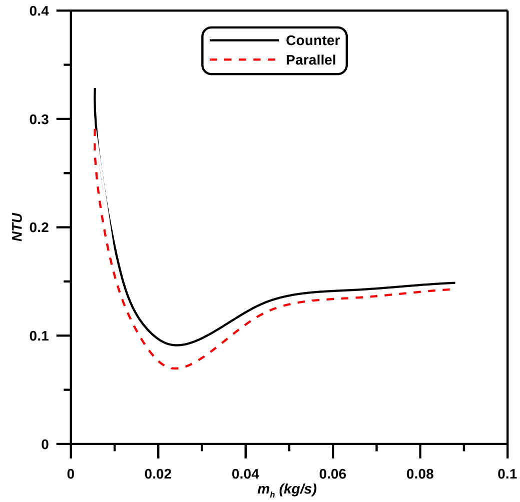
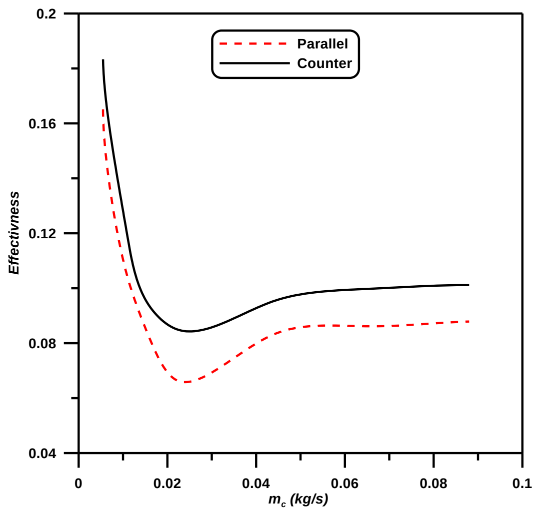


Fig.54: NTU, m_h=0.022 kg/s, N=0

Fig.55: NTU, m_c=0.022 kg/s, N=0



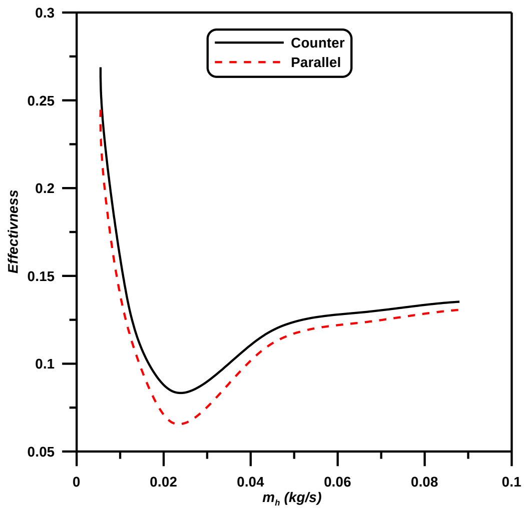


Fig.56: Effectiveness, m_h=0.022 kg/s, N=0

Fig.57: Effectiveness, m_c=0.022 kg/s, N=0

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