Modelling and Simulating The Effect of Velocity of Fluid Distribution on Physical Parameters of Stanton and Euler Number on a Steam Heater Tube

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Abstract

This paper is aimed at determining the effect of velocity of fluid distribution on physical parameters of Stanton and Euler number on a steam heater tube. In all cases, the solution for the velocity of fluid distribution was evaluated numerically for various physical parameters of Stanton and Euler number. The developed model was simulated with the acid of C^{++} visual basic software programming and found that the Stanton and Euler number increases with increase in velocity. Similarly, the model developed for effectiveness factor of steam heater was simulated and results obtained showed increase in effectiveness factor with increase in number of transfer unit (NTU) for various incremental step of capacity ratio {$(mc)_{min}/(mc)_{max}$ = 0.1 and $(mc)_{min}/(mc)_{max}$ = 0.2}. The developed models were found useful in monitoring and predicting the effect of velocity of fluid distribution on physical parameters of Stanton and Euler number on a steam heater tube process. The research demonstrates the usefulness of Euler and Stanton number in simulating the characteristics of fluid flow in a steam heater tuber as well as the effect of density, viscosity, temperature pressure, tube diameter, mean temperature and other functional coefficients.

Introduction

The need to understand some of the most important engineering problems involving heat transfer generally associated with two fluids with the effect of physical parameter: Euler and Stanton numbers on the temperature and velocity profit of gas distribution in a steam heater tube is well discussed in this paper. Although considerable progress has been made towards understanding many heat transfer equipment factors that affect temperature and velocity profits, in gas distribution process in a heat exchange and little attention has been given to how differences in Euler and Stanton number affect such processes.

Two fluid flow play an important role in many fields of engineering and technology. One of the important application of fluid flow is in material processing, acronautics, astrophysics, marine and mechanical engineers encounter a multitude of complex flow phenomena. These complex flows are often comprised of two or more phases in which the interaction between them plays a dominant role in controlling transport processes such as heat and mass exchange and reaction kinetics (Eckert, 1950; Coulson, Richardson, Backhurst, & Harker, 1954; Kierkus, 1968; Vajravelu & Hadjinicolaou, 1993; Perry, 1997; Chamkha, 1997 & Ukpaka, Ogoni & Ben, 2005).

Investigation conducted by various research groups revealed that fluid flow through the exchange in the same direction, influence the process and as well as the flow pattern is said to operate in parallel or co-current flow. Flow in opposite directions is known as counter-current flow (Sinnot, 1992, Lustig, Caruthers, & Peppas, 1992; Tortora Vitoria, Galli, Ritrorati & Chiellini, 2002; Ukpaka, Ogoni, & Ikenyiri, 2005). To enhance certain characteristics of the physical parameters, various parameters, that is Euler, number, Stanton number, temperature fields, velocity, mean temperature, density of the substances; friction factor (f) and heat transfer coefficient are measured to enable one determine the required specifications, parameters and materials that can lead to proper design of a steam heater tube exchanger. Investigation on literature shows that temperature and velocity profits on fluid distribution are affected by a steam heat tuber (heat exchanger process). So it is felt that there is a need to model and simulate the performance parameters that influence the Stanton and Euler number in analyzing the velocity of fluid distribution on a steam heater tube processes.

The Model

One of the simplest forms of steam heater exchanger is the so-called double-pipe exchanger. This arrangement is shown schematically in Figure 1. One of the fluids passes through the tube whilst the second fluid flows through the annular space. If the fluids flow through the exchanger in the same direction, then the unit is said to operate in parallel or co-current flow.

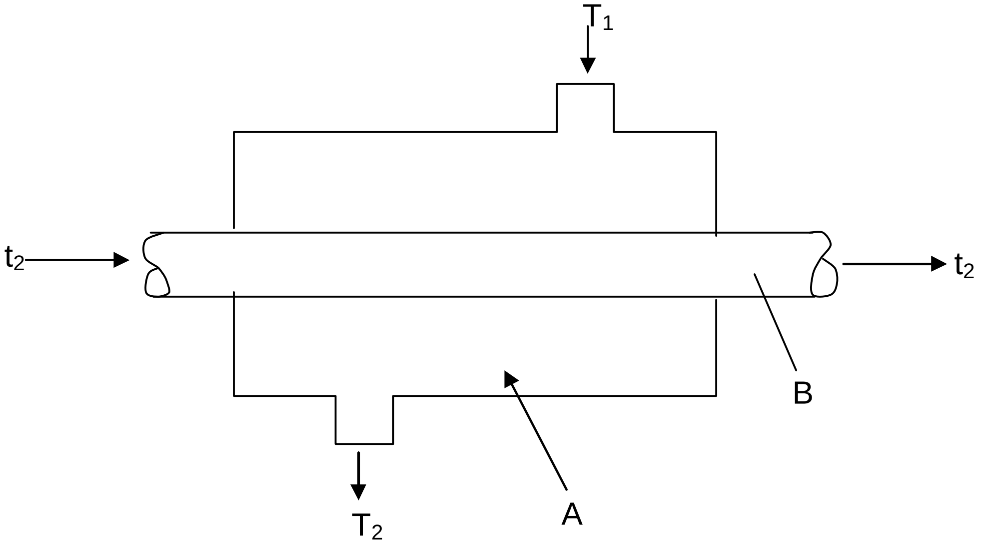

Figure 1: A Simply Double-Pipe Heat Exchanger

The energy balance equation is given as [Heat gained by fluid in flowing through the tube] = [mass flow rate] [specific heat] [rise in temperature] (1)

[Heat transferred from the tube surface to the fluid] = [surface area] [heat transfer coefficient] [mean temperature different] (2)

Defining equations (1) and (2) mathematically yields the expression for heat gained by fluid in flowing through the tube is given as;

$$Q_G = \frac{\pi D^2}{4} V\rho C_P \left(T_2 - T_1\right) \quad (3)$$

And the heat transferred from the tube surface to the fluid is given as

$$Q_F = \pi D L h \theta_m \quad (4)$$

Therefore, considering when heat gained in the system is equal to the heat loss, thus,[heat gained by fluid in flowing through the tube] = [heat transferred from the tube surface to the fluid] (5)

Substituting equations (3) and (4) into equation (5) yields

$Q_G = Q_F$

(6)

$$\frac{\pi D^2}{4} V\rho C_P \left(T_2 - T_1\right) = \pi D L h \theta_m$$

(7)

Stanton Number Model

The Equation (7) can be written in term of Stanton number $\left(h/C_p V \rho\right)$ = $\left(Nu/\mathrm{Re}\,\mathrm{Pr}\right)$, thus

$$\frac{h}{\rho V C_P} = St = \left(\frac{T_2 - T_1}{\theta_m}\right)\left(\frac{D}{4L}\right)$$

(8)

Therefore

$$\mathrm{St} = \left(\frac{T_2 - T_1}{\theta_m}\right)\left(\frac{D}{4L}\right)$$

Recalling the mathematical expression for the friction factor in relation with the Reynolds Number (Re); then the friction factor (f) can be expressed as

$$\mathrm{f} = \tau_w \Big/ \left(\tfrac{1}{2}\rho V^2\right) = \left(\left(R/2\right)\Delta P / \Delta X\right) \Big/ \left(\tfrac{1}{2}\rho V^2\right)$$

(9)

Then,

$$\Delta P = 2f\left(\frac{L}{D}\right)\rho V^2$$

(10)

Similarly, equation (10) can be rearranged to become

$$\frac{D}{L} = \frac{2 f \rho V^2}{\Delta P}$$

(11)

Therefore substituting equation (11) into equation (8) yields

$$\mathrm{St} = \left(\frac{T_2 - T_1}{\theta_m}\right)\left(\frac{f \rho V^2}{2\Delta p}\right)$$

(12)

Or

$$V = \left[\left(\frac{St}{f/2}\right)\frac{\theta_m}{(T_2 - T_1)}\left(\frac{\Delta P}{\rho}\right)\right]^{1/2}$$

(13)

Assuming that ΔP = ΔP_{max} and V = V_{max}, therefore equation (13) can be written as

$$V_{max} = \left[\left(\frac{St}{f/2}\right)\frac{\theta_m}{(T_2 - T_1)}\left(\frac{\Delta\rho_{max}}{\rho}\right)\right]^{1/2}$$

(14)

Euler Number Model

The Euler number which is the ratio of pressure force to that of inertia force that is

$$\text{Euler number EU} = \frac{\rho}{\rho V^2} = \frac{\text{pressure force, } [P_A]}{\text{Inertia force } [\rho V^2 A]}$$

(15)

Rearranging equation (7) and then substituting equation (10) into the derived expression from equation (7) yields

$$\frac{\pi D^2}{4} V\rho\ Cp\ (T_2 - T_1) = \pi DLh\ \theta m$$

(16)

$$\frac{D^2}{DL} = \frac{\pi V\rho Cp}{4}(T_2 - T_1) = \pi\ h\theta m$$

(17)

$$\frac{D}{L} = \frac{4\pi h\theta m}{\pi V\rho Cp(T_2 - T_1)}$$

(18)

$$D\pi V\rho\ Cp(T_2 - T_1) = 4\pi\ h\theta\ mL$$

(19)

$$\frac{L}{D} = \frac{\pi V\rho Cp(T_2 - T_1)}{4\pi\ h\theta m}$$

(20)

$$\Delta p = \frac{2 f\pi V\rho Cp(T_2 - T_1)\rho\mu^2}{4\pi\ h\theta\ m}$$

(21)

The Coconut

$$\frac{\Delta p}{\rho \mu^2} - Eu = \frac{2 f \pi V \rho Cp(T_2 - T_1)}{4\pi h \theta m}$$

(22)

Therefore

$$Eu = \frac{2 f V \rho Cp(T_2 - T_1)}{4 h \theta m} \quad (23)$$

The Mean Temperature Model

The mean temperature difference for a number of simple heat exchange is shown in figure 2. Consider a heat balance over the complete heat exchange and also over an element T_1 the exchanger.

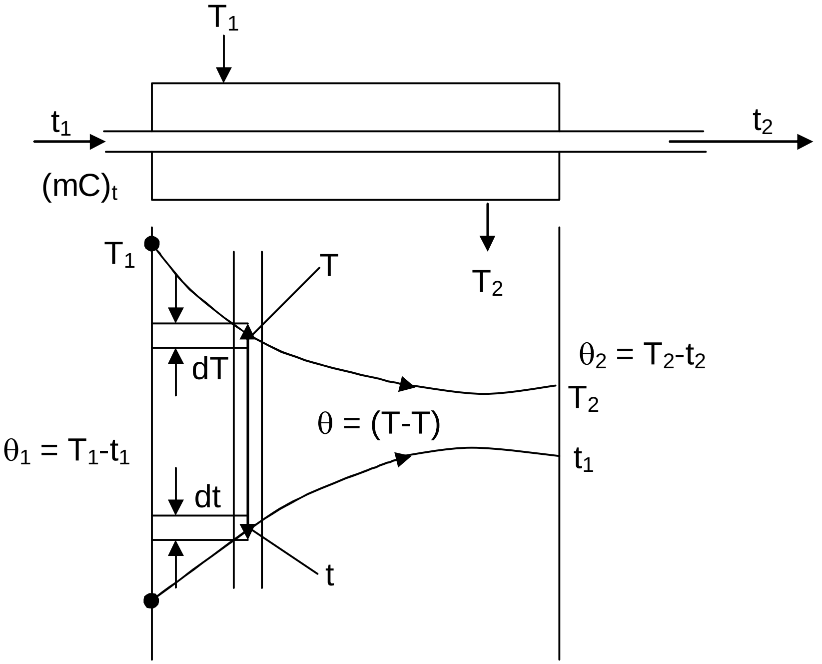

Figure 2. Simple parallel flow of heat exchange

For $(mc)_T$ = mass flow rate x specific heat

(24)

Therefore

(25)

$Q = (mc)_t (t_1 - t_2)$

(26)

$Q = UA\theta_m$

(27)

and

$dQ = -(mc)_T dT$ (29)

$dQ = (mc)_t dt$ (29)

$dQ = UdA (T-t) = UdA\theta$ (30)

$$dQ\left(-\frac{1}{(mC)_i} - \frac{1}{(mC)_t}\right) dT - dt = d(T-t) = d\theta \qquad (31)$$

From equation (30) we have

$$U\theta dA \left(-\frac{1}{(mC)_T} - \frac{1}{(mC)_t}\right) = d\theta$$

(32)

$$U \left(-\frac{1}{(mC)_T} - \frac{1}{(mC)_t}\right) = dA = \frac{d\theta}{\theta}$$

(33)

Integrating equation (33) yields

$$U \left(-\frac{1}{(mC)_T} - \frac{1}{(mC)_t}\right) A = \text{In} \frac{\theta_2}{\theta_1}$$

(34)

where $\theta_2 = T_2 - t_1$, $\theta_1 = T_1 - t_1$ (35)

Therefore from equations (25) and (26)

$$Q \left(-\frac{1}{(mC)_T} - \frac{1}{(mC)_t} = -(T_1 - T_2) - (t_2 - t_1)\right)$$

(36)

$$= (T_2 - t_2) - (T_1 - t_1) = \theta_2 - \theta_1$$

(37)

From equations (35) and (8)

$$\theta_m = (\theta_2 - \theta_1)/\text{In} \ (\theta_2 / \theta_1)$$

(38)

Model for Correlation of Stanton Number with Mean Temperature

Therefore, substituting equation (38) into equation (12) yields the expression of Stanton number in relation with the mean temperature difference

$$St = \left(\frac{T_2 - T_1}{\theta_2 - \theta_1 / \text{In} \left(\theta_2 / \theta_1 \right)} \right) \left(\frac{f \rho V^2}{2 \Delta P} \right)$$

(39)

Therefore

$$St = \left(\frac{\left(T_2 - T_1 \right) \text{In}\, \theta_2 / \theta_1}{\theta_2 - \theta_1} \right) \left(\frac{f \rho V^2}{2 \Delta P} \right)$$

(40)

Model for Correlation of Euler Number with Mean Temperature

Similarly, substituting equation (38) into equation (23) yields the expression of Euler number in relation with the mean temperature difference.

$$\text{EU} = \frac{2 f V \rho C_P \left(T_2 - T_1 \right)}{4h \left(\theta_2 - \theta_1 \right) / \text{In} \left(\theta_2 / \theta_1 \right)}$$

(41)

Equation (38) is the logarithmic mean temperature different (LMTD). The same expression applies for the mean temperature difference for the counter-current flow configuration as shown in figure 2.

Equation (25), (26) and (38) can be used either to predict the performance of a given exchange, or to size an exchanger for a given process. However, unless the four terminal temperature of the exchange are known, the use of the logarithmic error solution. A more convenient procedure results if a number of dimensionless parameters is given as follows:

Heat Exchanger Effectiveness (∈) Model

This is the ratio of the actual rate of heat transfer Q to the maximum rate of heat transfer permitted by the second law of thermodynamics. Thus, the maximum rate is given as

$$Q_{max} = (mc)_{min} (T_1 - t_1)$$

(42)

where $(mc)_{min}$ is the smaller of the two heat capacity rates, and T_1 and t_1 are the inlet temperatures of the hot and cold fluid, respectively.

Then

$$\in \; = \frac{(mc)_T (T_1 - T_2)}{(mc)_{\min}(T_1 - t_1)} \; = \frac{(mc)_t (t_2 - t_1)}{(mc)_{\min}(T_1 - t_1)}$$

(43)

The number of transfer unit NTU is given as

$$NTU \; = \frac{U_A}{(mc)_{\min}}$$

(44)

where NTU is the measure of the size of a heat exchange from the point of view of heat transfer.

Ratio of Heat Capacity Rates (Capacity Ratio) Model

The ratio of heat capacity is given as $(mc)_{min}/(mc)_{max}$ for a particular flow arrangement, the effectiveness is a function of the number of transfer unit and the capacity ratio only.

Considering equation (36) for the case of parallel flow then, $(mc)_T \; = (mc)_{\min}$

Therefore,

$$\frac{(T_2 - t_1)}{(T_1 - t_1)} \; = \; \exp\left[- NTU \left(1 + (mc)\min/ (mc)_{\max}\right)\right] \tag{45}$$

Also equation (44) becomes

$$\in \left[\frac{1}{(mc)_{\min}} \; + \frac{1}{(mc)_{\max}}\right] \; = \frac{(T_1 - T_2 + t_2 + t_1)}{(T_1 - t_1)/(mc)_{\min}} \tag{46}$$

where

$$\left[\frac{1 + (mc)_{\min}}{(mc)_{\max}}\right] \in = 1 - \exp\left[- NTU \; \left(1 + (mc)_{\min}\right)/(mc)_{\max}\right] \tag{47}$$

making $\in$ is the subject formala from equation (47) yields

$$\in \; = \; \frac{1 - \exp\left[- NTU \left(1 + (mc)_{\min}\right)/(mc)_{\max}\right]}{(1 + mc)_{\min} / (mc)_{\max}}$$

(48)

It can be shown that for a counter-current exchanger the appropriate expression for heat exchanger effectiveness $\in$ is given

$$\in = \frac{1 - \exp\left[- NTU\left(1 - (mc)_{\min}\right)/(mc)_{\max}\right]}{\left(1 + mc\right)_{\min}/(mc)_{\max}}$$

(49)

It can be shown that for a counter-current exchanger the appropriate expression for heat exchanger effectiveness $\in$ is given

$$\in = \frac{1 - \exp\left[- NTU\left(1 - (mc)_{\min}\right)/(mc)_{\max}\right]}{1 - \left[(mc)_{\min}/(mc)_{\max}\right] \exp\left[- NTU\left(1 - (mc)_{\min}\right)/(mc)_{\max}\right]}$$

(50)

Boundary Condition of Capacity Ratio Model

For convenience, equation (50) is used in the determination of heat exchanger effectiveness ($\in$) considering the following boundary condition

at $(mc)_{min} = 0, \quad (mc)_{max} = 0$ (51)

at $(mc)_{min}/(mc)_{max} = 0$ (52)

at $(mc)_{min} = 1, \quad (mc)_{max} = 1$ (53)

at $(mc)_{\min} / (mc)_{\max} = 1$

(54)

General Solution Model for the Determination of Effectiveness Factor

Considering the boundary condition as stated in equation (3), (4) then this ratio is zero, one of the fluids passing through the exchanger is undergoing a change of phase, thus equation (50) becomes

E = 1-exp (-NTU)

(55)

Considering the boundary condition as stated in equation (54) when this ratio is one, for parallel or co-current flow

$$\in = \frac{1 - \exp\left(-2NTU\right)}{2}$$

(56)

But for counter-current flow

$$\in = \frac{NTU}{1 + NTU}$$

(57)

Computational Procedure

Physical properties of crude oil sample obtained from one oil well in Niger Delta area of Nigeria was used for the investigation. The viscosity and density was measured using the aid of viscometer and hygrometer respectively.

The following operational parameters were used oil side heat transfer coefficient = 5670W/m^{2}k, specific heat of oil = 2.18kJ/kgk, specific heat of water = 4.19kJ/kgk, water flow rate =151kg/s, oil flow rate = 0.19kg/s, mean diameter of tube = 12.7m, inlet oil temperature = 422k, exit oil temperature = 325k, inlet water temperature = 327k, exit water temperature = 373k, friction factor of 0.011813 and $(T_2 - T_1)$ = 60k, viscosity = 22.98kg/m.h, density = 880kgm^3, oil/water inlet temperature difference θ_1 = (352-327) = 25k, oil/water inlet temperature different θ_2 = (422-337) = 85k. The data obtained were fed into the model computer using the developed mathematical expressions in equations (41), (42), (57) and (58) by considering various incremental steps in velocity from 1 to 10m/s and as well as various incremental steps on capacity ratio of 0.1 and 0.2.

Results and Discussion

The program was run with the insertion of the appropriate parameters as given above. The computation was carried out to determine the effect of velocity and NTU (number of transfer unit) on Stanton number, Euler number, and heat exchange effectiveness.

Table 1: Determination of Some Dimensionless Parameters at Various Velocity and Number of Transfer Unit (NTU)

V (m/s)	St	EU	$\in$	NTU
1	0.00194274	1.02701E-10	0	0
1.5	0.00437116	1.54051E-10	0.2753355	0.4
2	0.00777095	2.05402E-10	0.3990517	0.8
2.5	0.01214211	2.56752E-10	0.454641	1.2
3	0.01748464	3.08103E-10	0.4796189	1.6
3.5	0.02379854	3.59453E-10	0.4908422	2
4	0.03108381	4.10804E-10	0.4958851	2.4
4.5	0.03934045	4.62154E-10	0.4981511	2.8
5	0.04856845	5.13505E-10	0.4991692	3.2
5.5	0.05876783	5.64855E-10	0.4996267	3.6
6	0.06993857	6.16205E-10	0.4998323	4
6.5	0.08208068	6.67556E-10	0.4999246	4.4
7	0.09519417	7.18906E-10	0.4999661	4.8

7.5	0.10927902	7.70257E-10	0.4999848	5.2
8	0.12433524	8.21607E-10	0.4999932	5.6
8.5	0.14036283	8.72958E-10	0.4999969	6
9	0.15736178	9.24308E-10	0.4999986	6.4
9.5	0.17533211	9.75659E-10	0.4999994	6.8
10	0.19427381	1.02701E-09	0.4999997	7.2

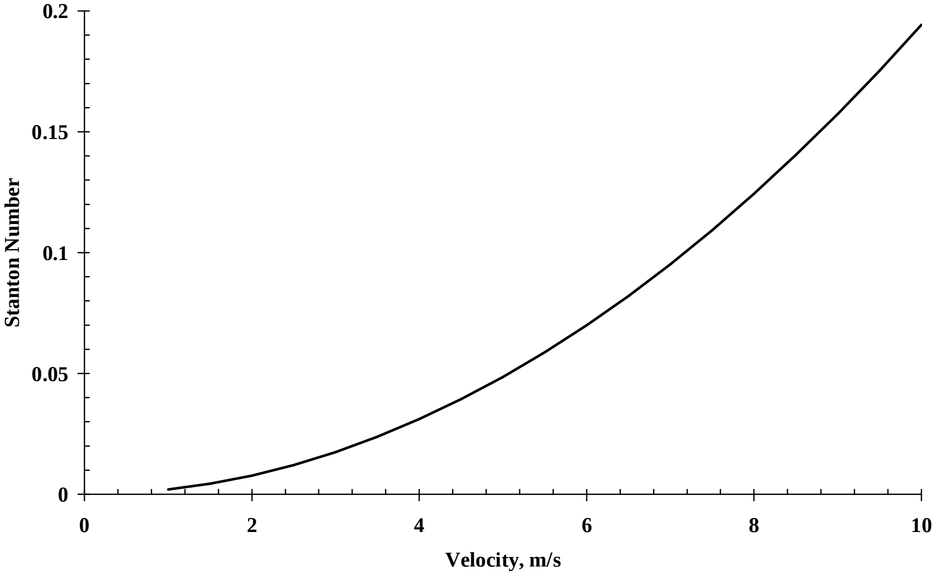

Figure 3: Graph of Stanton number versus velocity

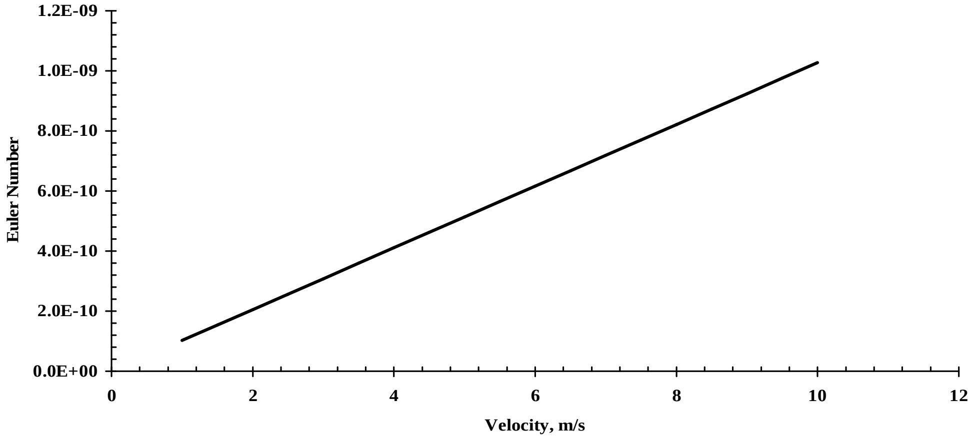

Figure 4: Graph of Euler number versus velocity

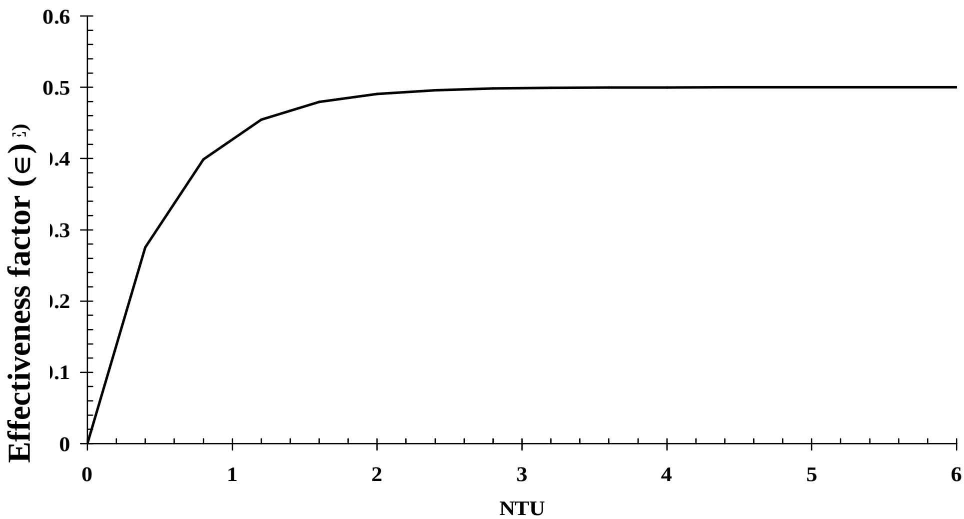

Figure 5: Graph of Effectiveness factor (∈) versus NTU for a constant parameter of capacity ratio

Problems in heat and mass transfer

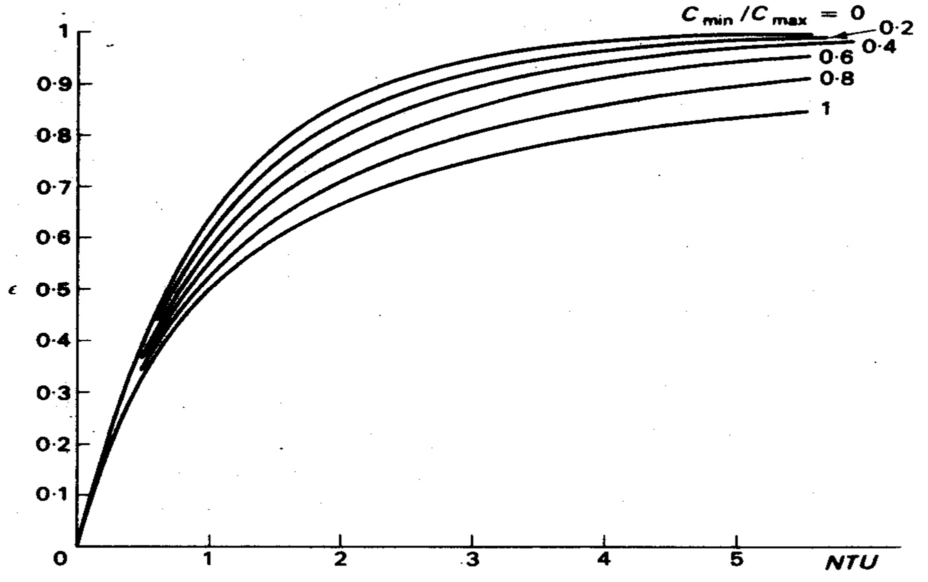

Fig. 6-3

Figure 6: Graph of Effectiveness Factor Versus NTU on Capacity Ratio of Incremental Step of 0.2

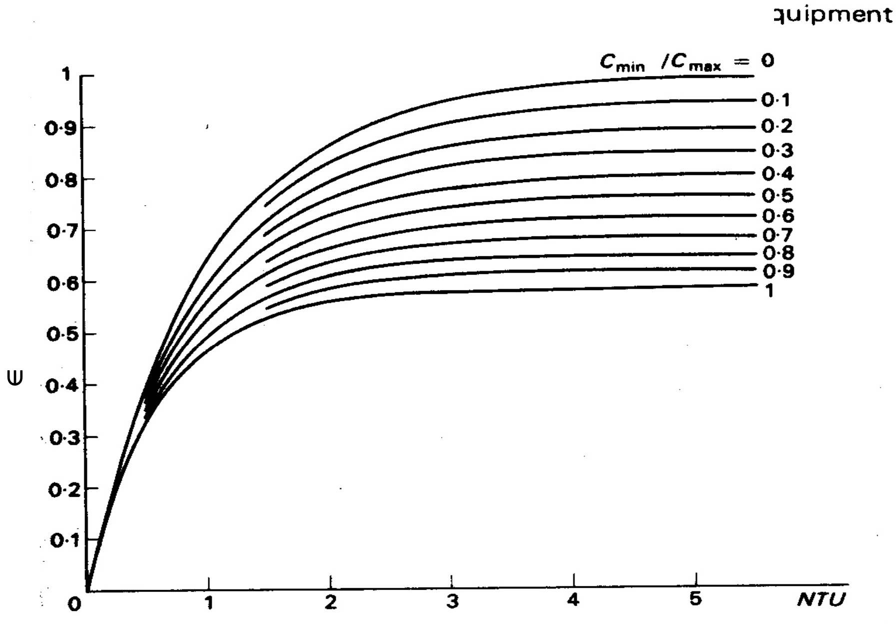

Figure 7: Graph of Effectiveness Factor Versus NTU on Capacity Ratio of Incremental Step of 0.1

Effect of Velocity on Stanton Number

The Stanton number increases with increase in velocity as shown in figure 3. The behaviour of the graph in figure 3 can be attributed on characteristics of the functional parameters such as temperature, mean temperature, friction factor, density, velocity and as well as change in pressure. Hence the developed model equation is presented in equation (41) for the Stanton number under consideration yields a solution of

$$St = \left[\frac{(T_2 - T_1)}{\theta_2 - \theta_1} \text{In } \theta_2/\theta_1\right]\left[\frac{f\rho V^2}{2\Delta P}\right]$$

The increase in Stanton number shows a parabolic shape on the behaviour of the system with increase in velocity.

Effect of Velocity on Euler Number

The Euler number increases with increase in velocity as shown in figure 4. The increase in Euler number as shown in figure 4 illustrated that the graph is linear in behaviour with increase in velocity. Similarly, one can also attribute the behaviour of the graph as on the characteristics of the functional parameters such as friction factor, velocity, density, specific heat capacity at a constant pressure, temperature, heat transfer coefficient and as well as mean temperature. Hence the developed model equation is presented in equation (42) for the Euler number under consideration yields a solution of

$$EU = \frac{2fV\rho C_p\left(T_2 - T_1\right)}{4h\left(\theta_2 - \theta_1\right)/\text{In }\left(\theta_2/\theta_1\right)}$$

Effect of NTU on Effectiveness Factor

The result shown in figure 5 illustrated the behaviour of effectiveness factor with incremental step on NTU. From Figure 5, it is observed that for the range of 0 to 2.8 of incremental step on NTU yielded increase in effectiveness factor. From 2.8 to 6 as shown in graph the effectiveness factor shows a slight difference in values with increase in NTU. The behaviour of the graph as shown in figure 5 can be attributed on the characteristics of the functional parameters as presented in the equation developed in this research work.

Similarly, from figure 6 and 7 the behaviour pattern of the graph is the same with the one of figure 5. but the effectiveness factor was simulated with NTU at various incremental step on capacity ratio of 0.2 as shown in figure 6 and incremental step on capacity ratio of 0.1 as shown in figure 7. From Figures 6 and 7 it is observed that the effectiveness factor decreases with increase in capacity ratio for various incremental step on NTU.

Conclusion

In modeling the effect of physical parameters Euler and Stanton number of fluid distribution on a steam heater tube of a given density is influenced by the degree of velocity, and NTU. The extent of Stanton and Euler numbers were observed to be increased with increasing velocity and NTU values. The above study brings out the following conclusions on the velocity, temperature, NTU,

1. The effect of velocity on the Stanton and Euler numbers on the characteristics of the functional parameter.
2. The effect of NTU on the effectiveness factor as it influences the characteristics of the functional parameter
3. The operating temperature of the system also influences the behaviour of Stanton and Euler number of functional parameters.
4. the boundary conditions of the capacity ratio parameters also influences the characteristics of Stanton and Euler number

5. The boundary conditions of the capacity ratio also influence the characteristics of the effectiveness factor.

Recommendations

The research carried out illustrates the effect of NTU on effectiveness factor for the range of 0 to 2.8 incremental step of NTU. Further studies can be carried out to examiner the behaviour of the system when the incremental step of NTU is 0 to 1 or 0 to 3; this will enable us evaluate the characteristics of the effectiveness factor and the usefulness of Stanton and Euler number. To achieve accuracy on the true result of this system, a laboratory analysis should be conducted at intervals to know the actual composition of the sample in the steam heater tuber.

Nomenclature

St	=	Stanton number (h/cvρ) = (Nu/Repr)
Re	=	Reynolds number (DVρ/μ)
Q	=	exchanger duty (w)
D	=	diameter (m)
V	=	velocity (m/s)
ρ	=	density (kg/m^3)
Cp	=	specific heat capacity at constant pressure
L	=	length (m)
h	=	heat transfer coefficient (w/m^2k)
Q_G	=	heat gained by gas (w)
Q_F	=	heat transferred from the tube surface to the gas (w)
T,t	=	temperature (k)
U	=	overall heat transfer coefficient
E	=	heat exchanger effectiveness factor (-)
θ_m	=	logarithmic mean temperature
μ	=	viscosity (kg/m.h)
P	=	pressure (N/m^2)
Δp	=	pressure drop (N/m^2)
m	=	mass flow rate (kg/s)
K	=	thermal conductivity (w/mk)
f	=	friction factor (-)
Nu	=	Nusselt number (hD/k)
Pr	=	parandtl number Cμ/k
A	=	area
NTu	=	number of transfer unit (-)

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