THE EFFECT OF TEMPERATURE DEPENDENT VISCOSITY VARIATION ON THE PERFORMANCE OF HEAT EXCHANGERS

BY

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Dedication

To My Mother and My Father

To My Wife and My Son

To My Brothers and Sisters

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THESIS ABSTRACT

FULL NAME OF STUDENT: SALEH MAHFOUDH BAGALAGEL

TITLE OF STUDY: THE EFFECT OF TEMPERATURE DEPENDENT VISCOSITY VARIATION ON THE PERFORMANCE OF HEAT EXCHANGERS

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The effect of viscosity on the thermal effectiveness and entropy generation of parallel and counter flow heat exchangers was numerically investigated. The viscosity variations with temperature in addition to the viscous friction were considered in the analysis for different tube fluid inlet temperatures and different heat exchanger heat capacity rate ratios.

The results show that viscosity causes significant deviations from the heat exchanger performance obtained using the assumption that the viscous effects are negligible. As the high viscosity fluid inlet temperature decreases, the effect of viscosity becomes more significant. However, the effect of viscosity becomes smaller as the heat capacity rate ratio decreases. It was found that an optimum heat exchanger size, at which the operation of the heat exchanger becomes most effective, could be determined. The results show also that the effect of viscosity on counter flow heat exchangers is smaller than its effect on parallel flow heat exchangers.

ملخـــص الرســـالـــة

الإســـم: صــالح محفــوظ باقـــلاقـــل

عـــنوان الرســـالـــة: تأثير تغير اللزوجة المعتمد على درجة الحرارة على أداء المبادلات الحرارية

التخصـــص: الهندســة الميــكانيكــية

تاريخ التخـــرج: ربيـــع الأول ١٤٢٢هـــ الموافـــق يونيو ٢٠٠١م

لقد تمت دراسة تأثير اللزوجة على الفاعلية الحرارية و مقدار ازدياد الانتروبي للمبادلات الحرارية المتوازية و المتعارضة التدفق بطريقة التحليل العددي مع اعتبار الأثر الناتج عن تغير اللزوجة المعتمد على درجة الحرارة و الاحتكاك الناشئ عن اللزوجة. كما تمت ملاحظة أثر اختلاف درجة حرارة دخول السائل في الأنبوب و اختلاف نسبة معدل المحتوى الحراري على نتائج الدراســـة.

ولقـــد أوضحت النتائج أن اللزوجة تؤدي إلى حصول انحراف واضح عن أداء المبادلات الحرارية الناشئ عن افتراض إمكانية تجـــاهل أثر اللزوجة. كما بينت الدراسة أن أثر اللزوجة يتزايد كلما نقصت درجة حرارة دخول السائل العالي اللزوجة. و لكن هذا الأثر يتناقص كلما نقصت نسبة معدل المحتوى الحراري. ولقد تبين كذلك أنه يوجد مقاس أفضل للمبادلة الحرارية يمكن عنده الحصـــول على أحسن كفاءة تشغيلية للمبادلة الحرارية. و لقد أوضحت النتائج أيضاً أن أثر اللزوجة على أداء المبادلات الحرارية المتعارضة التدفق أصغر من أثرها على أداء المبادلات الحرارية المتوازية التدفق.

CHAPTER 1

INTRODUCTION

1.1 Background

Convection heat transfer is one mode of heat transfer in which mechanical mass transport in addition to the existence of a temperature difference contribute to the heat transfer process (Kreith and Bohn, 1993). The mechanical mass transport depends on velocity profile of the flow as well as fluid properties such as density and viscosity.

The convection heat transfer rate can be calculated using Newton's Law of Cooling:

$$q_c = h_c A (T_F - T_W) \tag{1-1}$$

where:

q_c = rate of convection heat transfer (W)

A = heat transfer area (m^2)

T_F = temperature of fluid (K)

T_W = wall temperature (K)

h_c = convection heat transfer coefficient ($W/m^2/K$)

Because convection heat transfer depends heavily on the flow field, fluid properties and the temperature gradient, the evaluation of the heat transfer coefficient is considered to be a difficult task (Kreith and Bohn, 1993). For this reason, some assumptions need to be made in order to simplify this task. However, this will lead to some inaccuracy in the evaluation. Therefore, a factor of safety is usually included in design to compensate for any deficiency caused by these assumptions.

One common assumption usually used in heat transfer analysis is that fluid properties such as thermal conductivity and viscosity are independent of temperature. However, the variation of physical properties with temperature can affect the heat transfer rate considerably. For liquids, only the temperature dependence of the viscosity is of major importance (Kreith and Bohn, 1993).

The effect of temperature dependent viscosity variations on the results obtained by utilizing the constant properties assumption is still not well defined yet (Kreith and Bohn, 1993). One correction method, which might be applicable only when certain conditions are met, was proposed by Sieder and Tate for flow of liquids in tubes (Kreith and Bohn, 1993). The correction, according to this method, can be made by multiplying the Nusselt number (hD/k), the dimensionless group for convection heat transfer coefficient h_c, found using the constant properties assumption, by the ratio of viscosity at the bulk temperature to the viscosity at the surface temperature raised to 0.14 power. Some investigators have pointed out that the correction factor depends on viscosity-temperature relationship of a particular fluid (Shome and Jensen, 1995). However, this method is widely used to correlate experimental results (Kreith and Bohn, 1993). Many other researchers studied the effect of considering temperature dependent viscosity on the rate of convection heat transfer for different cases. The literature review given in this proposal presents examples of recent studies and summary of results.

The accurate evaluation of a heat exchanger performance requires also the consideration of temperature dependent viscosity. Some other assumptions might need to be reviewed also. For example, the consideration of viscous frictional heating showed that there is an optimum Number of Transfer Units (NTU) at which the effectiveness of the heat exchanger reaches a maximum value (Şahin, 1997; 1998a).

1.2 Literature Review

As the effect of temperature dependent viscosity variation on heat transfer systems is still not defined clearly enough, increasing number of research studies have been conducted recently. The main purpose of these studies was to find out a reliable and accurate estimation of the heat transfer behavior when viscosity varies from point to point in the system as temperature changes. Another part of research was also conducted recently to establish more accurate viscosity-temperature relationships for different fluids. In this literature review, representative examples from both parts of these studies are summarized.

1.2.1 Effect of Temperature Dependent Viscosity Variation on Heat Transfer Systems

Numerical and experimental analyses were performed to study the effect of large viscosity variation on thermal convection in an enclosure with localized heating strip from below (Chu and Hickox, 1990). The purpose of the study is to simulate the condition in a magma chamber. A correlation for the Nusselt number, Rayleigh number and viscosity at the heating strip and reference value viscosity was obtained. Numerical and experimental results were found to be in good agreement. It was found that the effect of variable viscosity could be estimated using a power law estimation method similar to Sieder and Tate method.

The effect of different heat transfer coefficients, tank aspect ratio and temperature dependent viscosity on the heat transfer rate – during the transient period – from the stored oil to the cold environment was studied using numerical simulator (Cotter and Charles, 1993). Five temperature-viscosity relationships were used, two of them have significant temperature change effect on viscosity, one has small temperature change effect on viscosity and two with two constant values of viscosity. The results show that the rate of heat loss is decreased more in the cases of significant temperature change effect on viscosity since the heat loss rate decreases from the stirred tank (convection in the tank) rate towards the conduction rate. In the case of small temperature dependent viscosity variation and in the two cases of constant viscosity, it was found that the rate of heat loss does not change significantly.

Fowler and Bejan (1994) have obtained the optimal Reynold numbers required to have minimum entropy generation for a given external forced convection heat transfer duty parameter B which is a function of the required heat transfer rate, fluid properties and body dimension. The entropy generation considered in this study consists of both heat transfer and fluid friction contributions. For the case of viscous fluids, the viscosity variation might have some effect on the results obtained. This effect was not addressed in this study and the viscosity was assumed to be uniform.

Numerical study for fully developed laminar flows in a semicircular duct with temperature-dependent viscosity variations has been made using a finite difference scheme (Harms et al., 1998). The arrhenius model for temperature-viscosity relationship was employed to solve for the velocity and temperature profiles. The effect on both the friction factor and Nusselt number was noted to be significant. The study shows that the ratio of Nu number for the case of temperature dependent viscosity to Nu number for the constant viscosity case is greater than one for the case of heating and lower than one for the case of cooling. This indicates that the heat transfer rate gets smaller when viscosity gets higher. This result applies for both axially uniform wall heat flux with peripherally uniform wall temperature boundary condition (H1) and peripherally and axially uniform wall temperature boundary condition (T) investigated in the study.

Numerical analysis of high Prandtl number fluid natural convection with temperature-dependent viscosity for vertical slot with aspect ratio 15 was done using a finite difference method (Jin and Chen, 1996). The study utilized a four parameter expression for the temperature-viscosity relationship:

$$\mu(T) = D\,exp(E/T^3 + FT + G/T) \tag{1-2}$$

where values of D, E, F and G are given by Chen and Pearlstein (1987). The effect of the temperature dependent viscosity on the critical Grashof number (at which the flow pattern changes) and the heat transfer rate was observed. It was found that the heat transfer is slightly smaller in the case of temperature-dependent viscosity suggesting that heat transfer rate gets smaller as viscosity variation becomes larger.

The effect of temperature dependent viscosity on the heat flow between the downward moving ethylene glycol vapor and a horizontal tube with variable wall temperature was studied using numerical solution of the governing differential equations (Memory and Rose, 1994). Four cases were solved for comparison for both free and forced convection. These cases are: (1) constant tube wall temperature with constant viscosity, (2) constant tube temperature with radial variation of viscosity, (3) variable tube wall temperature with constant viscosity and (4) variable tube wall temperature with circumferential variation of viscosity. Results represented in graphical form show that in cases 1 and 2, the effect of variable viscosity can be neglected due to a small temperature difference. However, in cases 3 and 4, the effect of variable viscosity is stronger with higher heat flux since initial viscosity is lower than the viscosity taken at reference film temperature. As temperature difference between film and wall gets smaller, the effect of variable viscosity with temperature becomes smaller.

The effect of viscous friction on the effectiveness of parallel and counter flow heat exchangers was studied analytically by Şahin (1997; 1998a). It was found that there is an optimum heat exchanger size for which the effectiveness is a maximum. The thermo-physical properties of the running fluids were assumed to be constant. The results were presented in curves with dimensionless parameters in order to have wider applicability. Finally, the optimum size of the heat exchanger was obtained as a function of the running fluids viscosity.

The entropy generation for a viscous flow in a duct subjected to constant surface temperature was investigated by Şahin (1998b). An analytical approach was used to study the effect of temperature-dependent viscosity on the ratio of pumping power to the total heat flux and the entropy generation. The constant viscosity case in addition to two temperature viscosity relationships was considered separately for water and glycerol and then the results were compared. It was found that, for high viscosity fluids, an optimum duct length can be determined at which energy loss due to entropy generation resulting from pressure drop and heat transfer is minimum. Similarly, an optimum inlet temperature can also be obtained such that energy loss is minimum.

The entropy generation for a viscous flow in a duct with constant heat flux was investigated analytically with the effect of temperature dependent viscosity variation taken into consideration (Şahin, 1996). The paper also considers the effect of temperature dependent viscosity variation on the pumping power to total heat transfer ratio. The constant viscosity case in addition to two temperature-viscosity relationships was considered separately for water and glycerol and the results were compared. It was found that, for high viscosity fluid, an optimum duct length could be obtained such that energy loss due to entropy generation resulting from pressure drop and heat transfer is minimum. The effect of temperature dependent viscosity variation becomes more significant for low heat flux because entropy generation due to viscous friction becomes dominant. It was also noticed that for fluids with small temperature dependent viscosity variation (as water was considered), the effect of temperature dependent viscosity is much less significant.

Numerical study of thermally developing and simultaneously developing mixed convection flow and heat transfer was carried out to find out the effect of temperature dependent viscosity on the Nusselt number and friction factor of an isothermal horizontal tube (Shome and Jensen, 1995). The study considered the effect of inlet Prandtl number, inlet Rayleigh number, wall-to-inlet temperature difference and inlet axial velocity profile for both cooling and heating conditions. Then, a scaling analysis was performed to find a parameter that can be used in correlating the Nusselt number and friction factor found numerically and experimentally in order to have more accurate correlations. The authors found that the obtained correlations have a smaller error than all experimentally found correlations existing in the literature. It was also found that the effect of temperature dependent variable viscosity is significant, being stronger on the friction factor than on the Nusselt number.

Experimental study of the effect of variable viscosity of mineral oil on the laminar heat transfer in a 2:1 rectangular duct was conducted by Xie and Hartnett (1992). Three cases were considered for an axially constant temperature duct: (1) top wall heated, other walls adiabatic, (2) bottom wall heated, other walls adiabatic and (3) top and bottom walls heated, other walls adiabatic. The authors found that the heat transfer is enhanced when top wall is heated because of increased velocity. In the case in which bottom wall is heated, the heat transfer is enhanced more due to the additional boundary effect. When both top and bottom walls are heated, the effect of variable viscosity is small due to symmetrical variations in the duct cross-section.

Numerical study of the effect of variable viscosity on the laminar heat transfer in a 2:1 rectangular duct was carried out by Shin et al. (1993). The same cases considered by Xie and Hartnett (1992) were used in this study. The result gave excellent agreement with the results obtained by Xie and Hartnett experimental work.

1.2.2 Viscosity Temperature Relationship

Many researchers have studied the behavior of viscosity with temperature for different fluids. Some correlations were developed as a result of this research work. These correlations can be classified into two major types (Orbey and Sandler, 1993):

(1) Simple correlation that uses temperature and pressure with fluid specific adjustable parameters.

(2) Generalized correlations that uses reduced forms of temperature and pressure to predict a reduced viscosity.

From the literature review in section 2.1, it can be seen that the effect of temperature dependent viscosity variation is more significant in the case of systems involving high viscosity fluids. Hence, we shall concentrate more on the temperature viscosity relationships of high viscosity fluids in this section.

Viscosity of refined, bleached and deodorized (RBD) and refined, bleached and winterized (RBW) Canola oil were measured experimentally at temperature from 4 to 100°C. using Cannon-Fenske Routine Viscometer (Lang et al., 1992). The experimental results were compared with the results obtained using the typical arrhenius temperature-viscosity relation:

$$\mu=\mu_{ref}\exp[E/(RT)] \qquad (1\text{-}3)$$

and the modified arrhenius equation for crude Canola oil. However, the results obtained experimentally differed from the results obtained using these two equations. Hence, a temperature-viscosity relation was developed to match the experimental measurements. This relation is:

$$\nu = \exp\left(C_0 + C_1\mathrm{T} + C_2\mathrm{T}\right) \qquad (1\text{-}4)$$

where: C_0 =5.3273, C_1= 0.0519 and C_2=0.0002.

A method of predicting viscosity of liquid hydrocarbons was developed by Orbey and Sandler (1993). This method combines the advantages of both simple correlation with adjustable parameters and generalized correlation / prediction method with reduced parameters. This was achieved by first considering expression for atmospheric pressure viscosity prediction for pure hydrocarbons as a function of temperature. Then, a generalized extension, in the form of a multiplicative factor, was developed to account for the effect of pressure. Finally, mixing rules were used to predict viscosity for mixtures, including mixtures with carbon dioxide. It was found that this approach is surprisingly accurate for the fluids and conditions examined in the study.

A temperature-viscosity correlation that accounts for pressure effect in case of Bitumen was developed (Puttagunta et al., 1993). This correlation, which is an extension of a previously developed correlation in terms of temperature, was found successful in predicting the viscosity of Canadian bitumens and heavy oils. Predictions are made on a new set of data using a single measurement of viscosity at 30 °C. and 101.3 kPa pressure. The procedure was examined and the results were found to have similar accuracy as obtained in the set of data used in developing the correlation.

A temperature viscosity relation that uses adjustable parameters was presented with the related parameters for 355 hydrocarbon liquids (Yaws et al., 1994). These parameters were collected using both experimental and literature data.

1.3 Significance of the Study

From the literature review, we can draw the following conclusions:

1- The effect of temperature dependent viscosity variations on heat transfer, particularly on heat exchanger performance, is not covered enough in the literature.
2- The effect of temperature dependent viscosity variations varies depending on fluids involved and system conditions.
3- The existing correlations that describe the effect of temperature dependent viscosity variation on heat transfer rate can't be generalized for all cases and all fluids. There is a considerable error involved in the results obtained using these correlations.
4- The temperature-viscosity relationship depends on the type of fluid. Moreover, there is no relationship found for many fluids.
5- The effect of temperature dependent viscosity variation is more significant in the case of high viscosity fluid.
6- More investigation is needed to have better understanding of the effect of temperature dependent viscosity variation on heat transfer systems.

In this thesis, the effect of temperature dependent viscosity variation on the effectiveness of parallel and counter flow heat exchangers will be investigated. From the literature review discussed earlier, there is no paper dealing with this effect. Şahin (1997; 1998a) investigated the effect of viscous friction on the effectiveness of parallel and counter flow heat exchangers using dimensionless parameters. However, the viscosity of both fluids was assumed to be constant. The results of this study showed a decrease in the heat exchanger effectiveness for sizes greater than a certain heat exchanger size expressed in NTU as shown in Fig. 1.1 and Fig. 1.2. Therefore, an optimum heat exchanger size can be found as a function of the running fluid viscosity. This optimum size is inversely proportional to the running fluid viscosity. Therefore, the investigation of the temperature

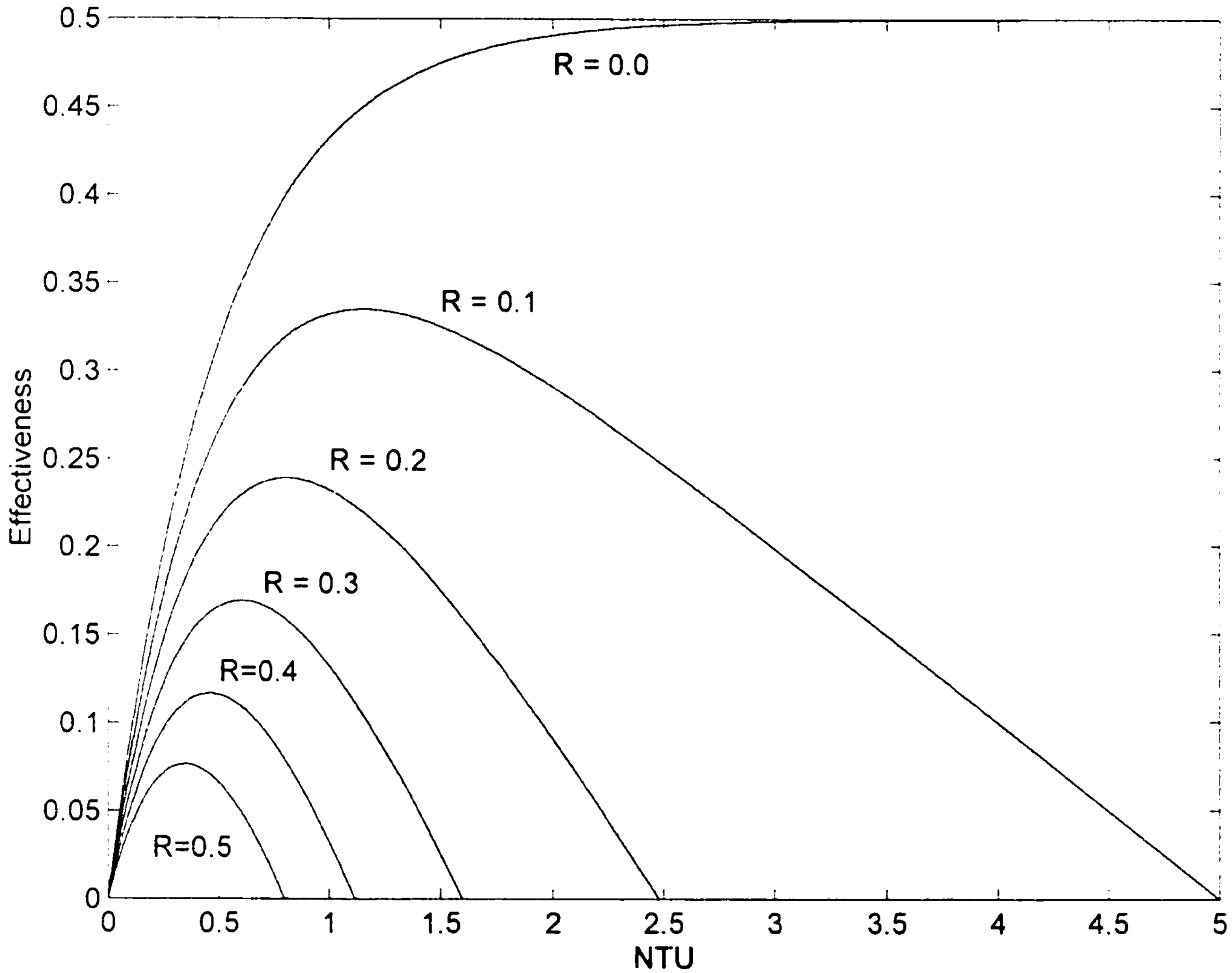


Fig. 1.1 Effect of viscous friction on the thermal effectiveness of parallel flow heat exchangers (Şahin, 1998a), (R = 8 Ec/St Re where Ec = Eckert number, St = Stanton number and Re = Reynolds number).

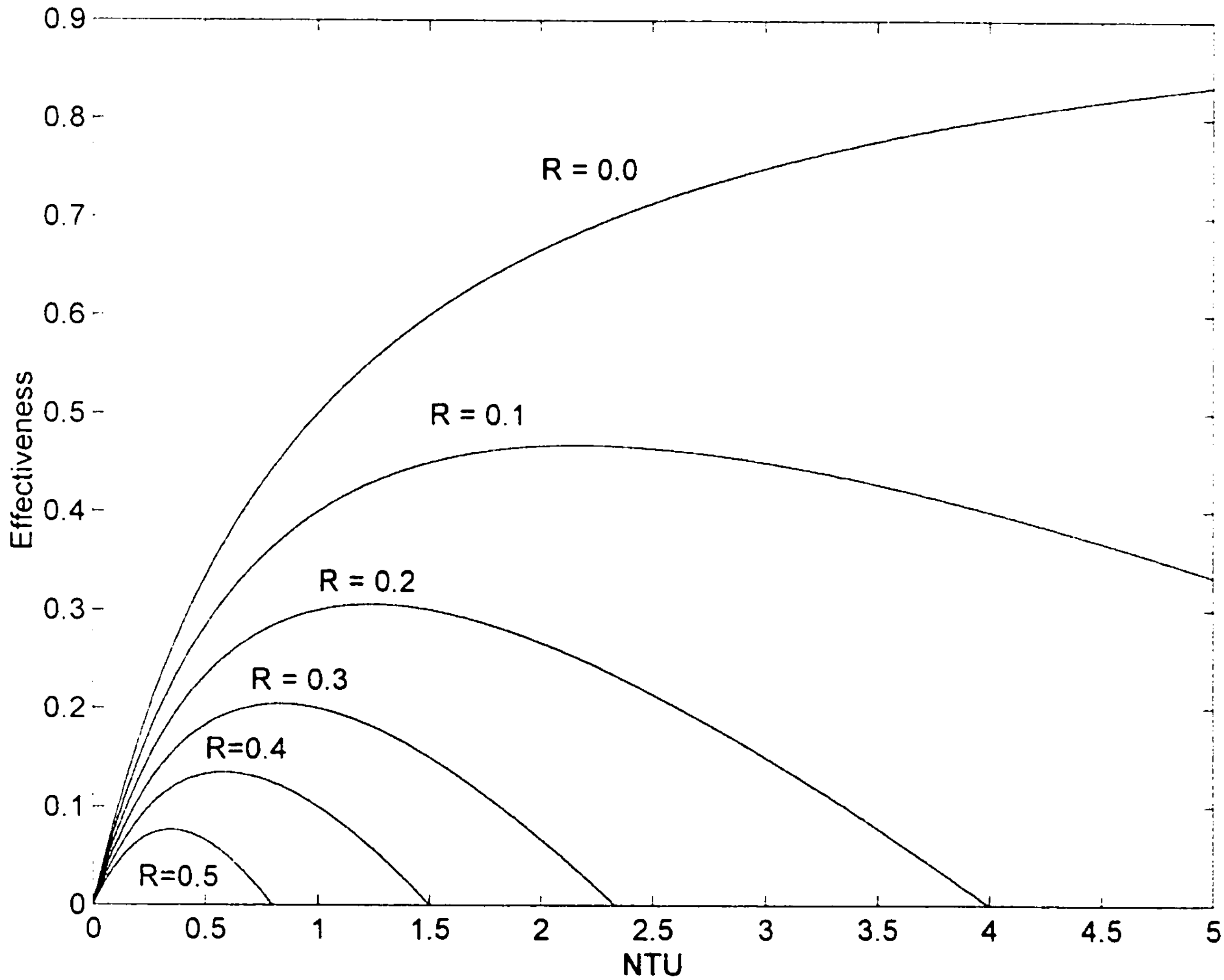


Fig. 1.2 Effect of viscous friction on the thermal effectiveness of counter flow heat exchangers (Şahin, 1997), (R = 8 Ec/St Re where Ec = Eckert number, St = Stanton number and Re = Reynolds number).

dependent viscosity variation is of major importance for design and rating of heat exchangers that handle high viscosity fluids.

The effect of temperature dependent viscosity variation was investigated for a tube subjected to constant heat flux or constant wall temperature by Şahin (1996; 1998b). An optimal tube length, for which energy loss due to entropy generation is a minimum, was obtained. This shows also the need for such investigation for designers as minimization of energy loss is a very important design criterion.

1.4 Industrial Applications

From the results obtained by Şahin (1996; 1998b) for the effect of viscous friction with temperature dependent viscosity variation on a tube subjected to constant heat flux or constant wall temperature, it was concluded that the effect of viscosity on the heat transfer system is more significant in the case of the high viscosity fluid investigated (Glycerol) than it is in the case of the low viscosity fluid investigated (Water). Therefore, this analysis is most useful in the industrial applications where high viscosity fluids are processed. Examples of these applications can be found in food, chemical and polymer industries.

CHAPTER 2

MATHEMATICAL FORMULATION AND

CALCULATION METHODOLOGY

2.1 The Statement of the Problem

The performance of parallel and counter flow heat exchangers is to be investigated when viscosity of the running fluids is considered as function of the running fluids temperatures. Viscous frictional heating is to be also considered in the investigation. This performance can be represented in terms of two performance measures related to the first law and second law of thermodynamics. These measures are the heat exchanger thermal effectiveness (First Law analysis) and the entropy generation (Second Law analysis). With these two measures, the performance of the heat exchanger can be examined and optimum-operating conditions can be identified.

Because of the difficulty encountered in the analysis of the governing equations when the viscosity dependence on temperature is introduced, numerical techniques shall be used in the analysis. For this purpose, a computer program shall be developed to perform the analysis. This will also make it easy for further investigation of different fluids, heat exchanger geometry and mass flow rates. The mathematical formulation and the calculation methodology of the analysis are explained in the following sections.

2.2 Temperature-Viscosity Relationship

One complication encountered is that the temperature-viscosity relationship is not well established for many liquids. Investigators dealing with the heat transfer rate with temperature dependent viscosity have used different models of the temperature-viscosity relation. While some studies utilized the exponential or the arrhenius model (Harms et al., 1998, and Shin et al., 1993), other studies have utilized a four-parameter temperature-viscosity relationship (Jin and Chen, 1996). Sherman (1990) has given the following relationship for temperature dependent viscosity:

$$\mu(T) = \mu(T_{ref})(\frac{T}{T_{ref}})^n \exp[B(\frac{1}{T} - \frac{1}{T_{ref}})] \qquad (2-1)$$

The coefficients n and B used in the relationship depend on the type of the fluid (Sherman, 1990). For glycerol and water, these coefficients are as follows (Sherman, 1990):

For glycerol: n = 52.4, B = 23100.

For water: n = 8.9, B = 4700.

A graphical representation for the viscosity variation with temperature for water and glycerol is given in Fig. 2.1. From this figure, it can be noted that the temperature-dependent viscosity variation for water is smaller than it is for glycerol. Moreover, the viscosity values of water in the shown range are much smaller than the viscosity values of glycerol. In this thesis, this viscosity variation with temperature will be used for the analysis.

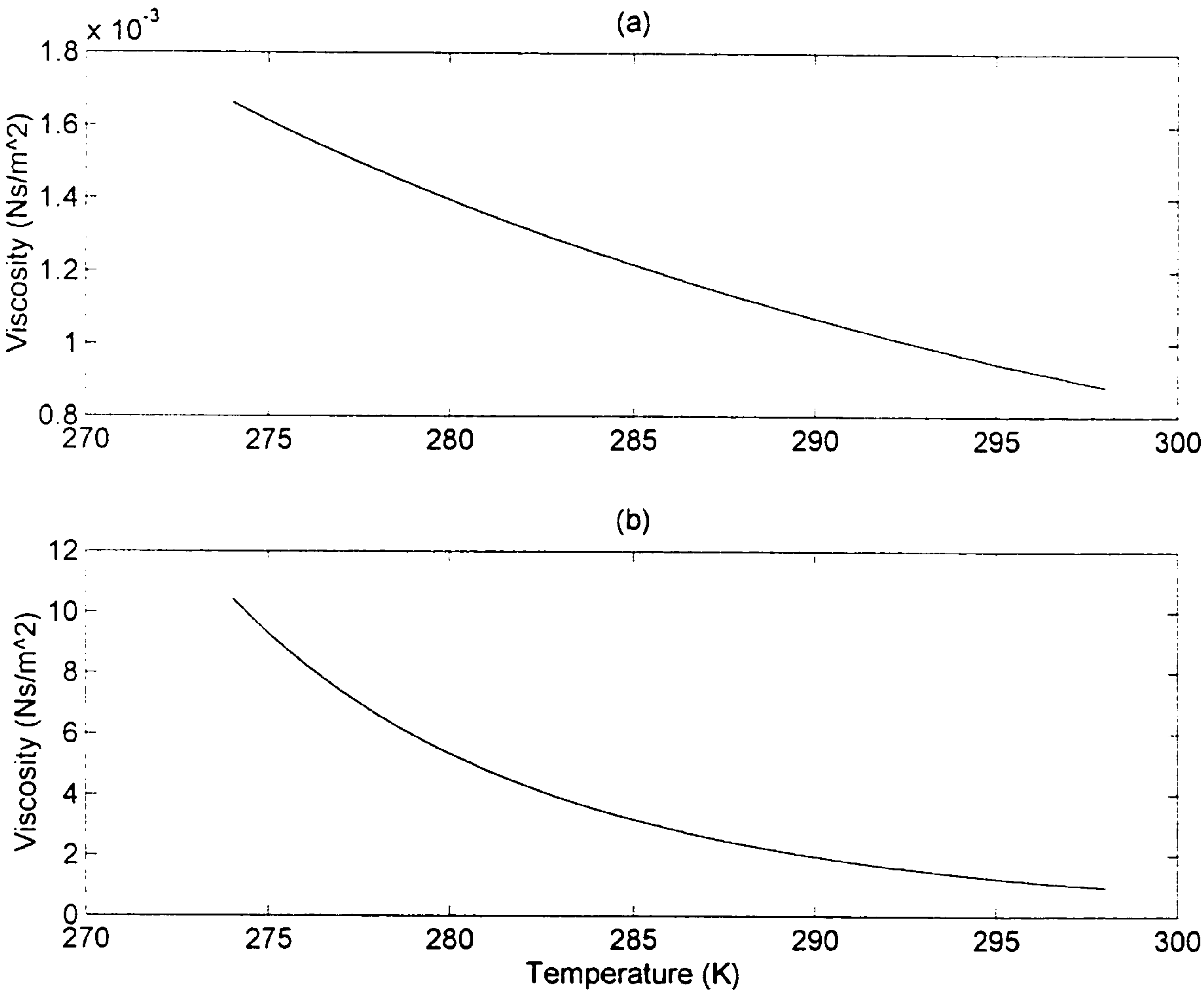


Fig. 2.1 Viscosity variation with temperature for: (a) water and (b) glycerol.

2.3 Mathematical Formulation

The performance of parallel and counter flow heat exchangers shall be performed using the following assumptions:

1- The temperatures of both streams are steady and functions of axial position only.

2- The properties of the fluids, except viscosity, are independent of temperature.

3- Heat transfer from the heat exchanger to surrounding is negligible.

4- No phase change occurs in the heat exchangers.

5- The temperature and velocity are uniform over the flow cross section.

6- Fouling resistance is assumed to be negligible.

7- Axial conduction in the wall is negligible due to very thin wall thickness.

Following the pattern of analysis given by Şahin (1997; 1998a), the effect of temperature dependent viscosity variation can be investigated for parallel and counter flow heat exchangers.

2.3.1 Parallel Flow Heat Exchangers

The basic differential equations governing the temperature distribution of the hot stream (the tube fluid) and the cold stream (the annulus fluid) can be obtained using the system of a parallel flow heat exchanger shown in Fig. 2.2. The application of the conservation of energy principle (the first law of thermodynamics) on the differential element dx for the steady state condition:

$$\dot{E}_{in} = \dot{E}_{out} \qquad (2-2)$$

will lead to the following two differential equations:

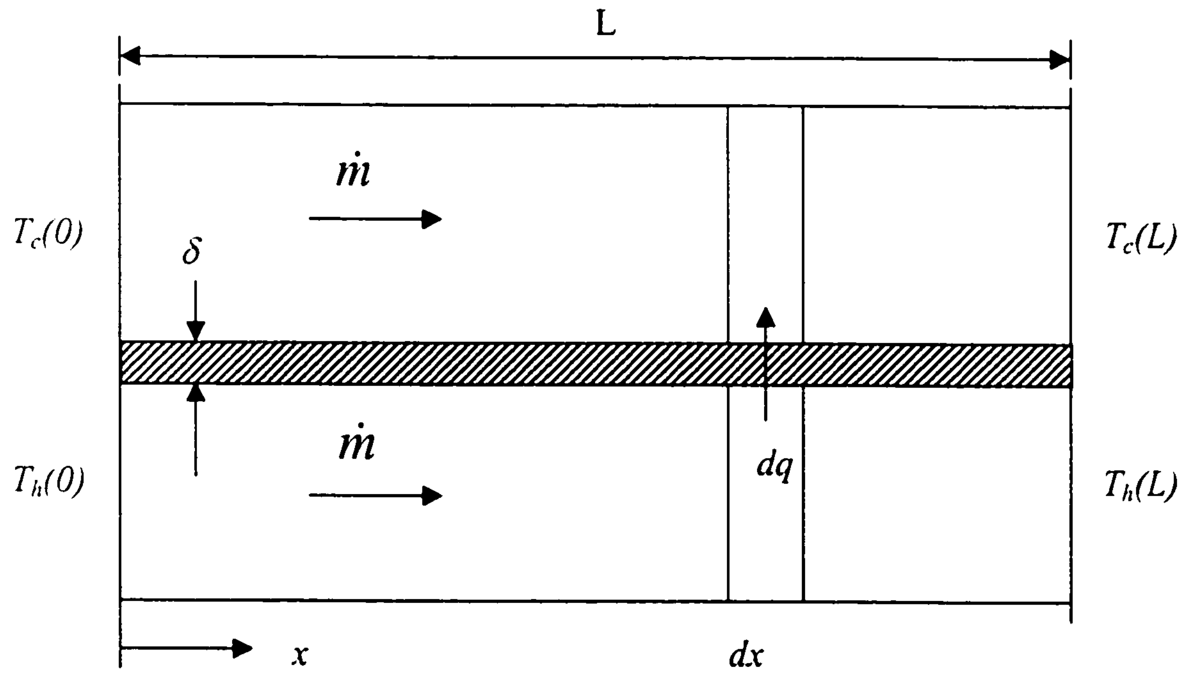


Fig. 2.2 The system of a parallel flow heat exchanger.

$$(\dot{m}C_p)_t \frac{dT_t}{dx} = -pU(T_t - T_a) + \frac{d\dot{W}_{ct}}{dx} \qquad (2-3)$$

$$(\dot{m}C_p)_a \frac{dT_a}{dx} = pU(T_t - T_a) + \frac{d\dot{W}_{ca}}{dx} \qquad (2-4)$$

where p is the perimeter, W_{ct} and W_{ca} are the internal heat generation due to viscous friction in the hot fluid stream and the cold fluid stream respectively; C_p is the specific heat capacity and U is the overall heat transfer coefficient defined as (assuming very small wall thickness δ):

$$\frac{1}{U} = \frac{1}{h_a} + \frac{\delta}{k} + \frac{1}{h_t} \qquad (2-5)$$

The term $pU(T_t - T_a)$ represents the local axial heat transfer between the two streams dq_x. The term representing the viscous frictional heating for the tube stream can be written as:

$$\frac{d\dot{W}_{ct}}{dx} = -\frac{\dot{m}_t}{\rho_t}\frac{dP_t}{dx} \qquad (2-6)$$

where:

$$-\frac{dP_t}{dx} = \frac{4 f_{ct}}{D_{Ht}} \frac{1}{2} \rho_t V_t^{\,2} \qquad (2-7)$$

hence

$$\frac{d\dot{W}_{ct}}{dx} = 2 f_{ct} \frac{\dot{m}_t V_t^{\,2}}{D_{Ht}} \qquad (2-8)$$

for fully developed laminar flow in a circular tube (Churchill, 1988):

$$f_{ct} = \frac{16}{\text{Re}} = \frac{16}{\rho_t V_t D_{Ht}} \mu_{ct}(T_t) \qquad (2-9)$$

using this equation, equation 2-8 can be written as:

$$\frac{d\dot{W}_{ct}}{dx} = \frac{32}{\rho_t V_t D_{Ht}} \frac{\dot{m}_t V_t^{\,2}}{D_{Ht}} \mu_{ct}(T_t) \qquad (2-10)$$

therefore, the temperature rise caused by frictional heating can be written as :

$$(\Delta T)_t = \frac{d\dot{W}_{ct}}{dx} \frac{dx}{(\dot{m}C_p)_t} = \frac{32V\mu_{ct}(T_t)dx}{\rho_t D_{Ht}^2 (C_p)_t} \qquad (2-11)$$

where the hydraulic diameter for the tube is given by:

$$D_{Ht} = D_i \qquad (2-12)$$

Similarly, an expression for the viscous frictional heating term for the annulus fluid can be obtained using the friction factor expression for fully developed laminar flows in a circular annulus (Churchill, 1988):

$$\frac{d\dot{W}_{ca}}{dx} = \frac{32}{\rho_a V_a D_{Ha}} \frac{\dot{m}_a V_a^2}{D_{Ha}} \frac{\mu_{ca}(T_a)(a_2 - a_1)^2}{(a_2^2 + a_1^2 - 2r_{\max}^2)} \quad (2-13)$$

Where:

$$r_{\max}^2 = \frac{(a_2^2 - a_1^2)}{2\ln(a_2 / a_1)} \quad (2-14)$$

The equation for the heat generation due to viscous friction would be:

$$\frac{d\dot{W}_{ca}}{dx} = \frac{32}{\rho_a V_a D_{Ha}} \frac{\dot{m}_a V_a^2}{D_{Ha}} \frac{\mu_{ca}(T_a)(a_2 - a_1)^2}{(a_2^2 + a_1^2 - 2r_{\max}^2)} \quad (2-15)$$

and the temperature rise due to viscous friction would be:

$$(\Delta T)_a = \frac{d\dot{W}_{ca}}{dx} \frac{dx}{(\dot{m}C_p)_a} = \frac{32 V_a dx}{\rho_a D_{Ha}^2 (C_p)_a} \frac{\mu_{ca}(T_a)(a_2 - a_1)^2}{(a_2^2 + a_1^2 - 2r_{\max}^2)} \quad (2-16)$$

where the hydraulic diameter for the annulus is given by:

$$D_{Ha} = 2(a_2 - a_1) \quad (2-17)$$

Therefore, the total temperature difference (due to heat transfer and viscous friction) per dx increment for the tube side and the annulus side can be written as:

$$\left(\Delta T\right)_t = \frac{-dq_x}{(\dot{m}C_p)_t} + \frac{32V_t\mu_{ct}(T_t)dx}{\rho_t D_{Ht}^2 (C_p)_t} \qquad (2-18)$$

$$\left(\Delta T\right)_a = \frac{dq_x}{(\dot{m}C_p)_a} + \frac{32V_a dx}{\rho_a D_{Ha}^2 (C_p)_a} \frac{\mu_{ca}(T_a)(a_2 - a_1)^2}{(a_2^2 + a_1^2 - 2r_{\max}^2)} \qquad (2-19)$$

Therefore, knowing the temperature-viscosity relationship, we can solve equation 2-3 and 2-4 numerically simultaneously from $x = 0$ to L in order to get the temperature distribution for each stream.

2.3.2 Counter Flow Heat Exchangers

Similarly, for the counter-flow heat exchanger, the system can be represented by Fig. 2.3. The governing differential equations that describe the temperature distribution for both streams can be written as:

$$(\dot{m}C_p)_t \frac{dT_t}{dx} = -pU(T_t - T_a) + \frac{d\dot{W}_{ct}}{dx} \qquad (2-20)$$

$$-(\dot{m}C_p)_a \frac{dT_a}{dx} = pU(T_t - T_a) + \frac{d\dot{W}_{ca}}{dx} \qquad (2-21)$$

Equations 2-6 through 2-17 are also applicable for the counter flow heat exchanger. The same technique described for the parallel flow heat exchanger can be followed here to get the temperature distribution. However, one exit temperature needs

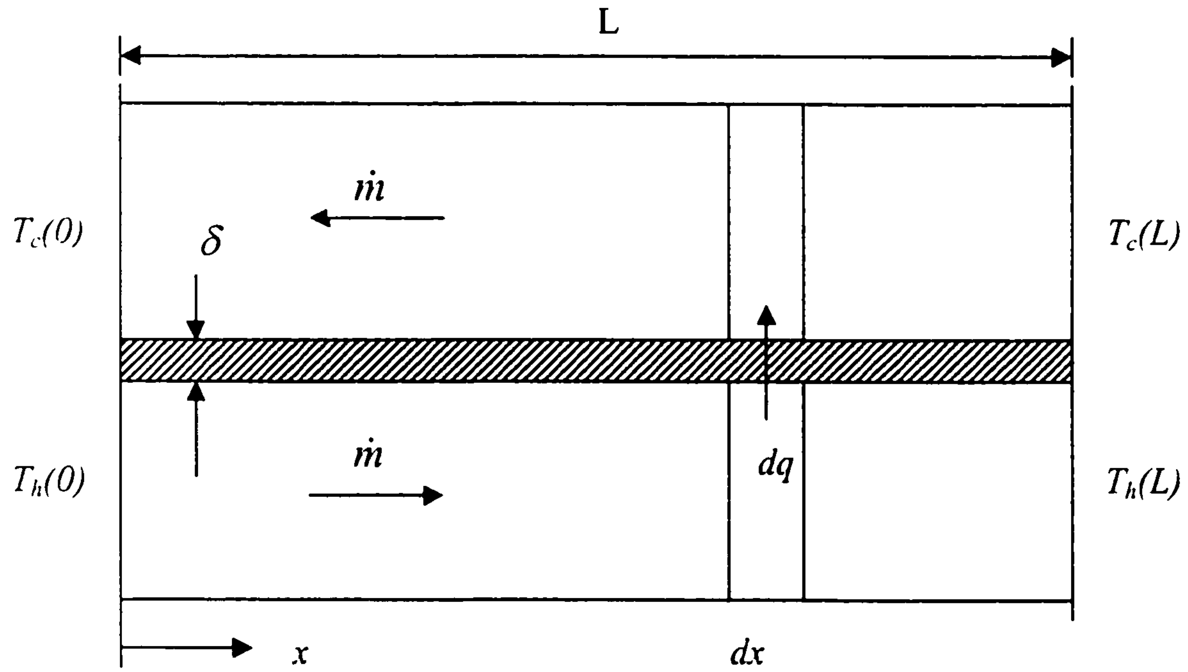


Fig. 2.3 The system of a counter flow heat exchanger.

to be assumed in order to solve for the temperature distribution. With the annulus fluid exit temperature assumed, the numerical solution can be executed starting from the tube fluid inlet side until the annulus fluid inlet temperature is reached. Different annulus fluid exit temperatures shall be assumed in order to get the temperature distribution for different heat exchanger sizes.

The total temperature difference (due to the heat transfer and viscous friction) per dx increment for the tube side and the annulus side can be written as:

$$\left(\Delta T\right)_t = \frac{-dq_x}{(\dot{m}C_p)_t} + \frac{32V_t\mu_{ct}(T_t)dx}{\rho_t D_{Ht}^2 (C_p)_t} \qquad (2-22)$$

$$\left(\Delta T\right)_a = \frac{-dq_x}{(\dot{m}C_p)_a} - \frac{32V_a dx}{\rho_a D_{Ha}^2 (C_p)_a} \frac{\mu_{ca}(T_a)(a_2 - a_1)^2}{(a_2^2 + a_1^2 - 2r_{\max}^2)} \qquad (2-23)$$

2.4 Calculation Methodology

The calculation methodology that would be used throughout this thesis for the calculation of temperature distributions and the heat exchanger performance measures is outlined in this section.

2.4.1 Temperature Distributions

The evaluation of the overall heat transfer coefficient requires the knowledge of the convection heat transfer coefficients (h) for each fluid and the thermal conductivity (k) of the wall material as given by equation 2-5. This equation can be used when the convection heat transfer coefficients (h) of the running fluids are known *a priori* (Hewitt et al., 1997). This is usually the type of problems encountered when the physical properties are assumed to be constant throughout the heat exchanger. If the variations of the physical properties of the running fluids are considered, the convection heat transfer coefficients (h) can't be assumed to be constant. This type of problems is called a *conjugate* problem (Hewitt et al., 1997). This type of problems is usually encountered when the physical properties of the running fluids are considered to be function of temperature.

Since the viscosity is considered to be temperature-dependent in this study, the analysis to be done is of the conjugate type. Thus, the heat exchanger overall heat transfer coefficient can't be determined from equation 2-5. Instead, the convection heat transfer coefficient (h) for each fluid needs to be determined at different axial locations in the heat exchanger based on the temperature of the wall and the bulk temperature of the fluid at the respective location.

As pointed out earlier, a widely used empirical correlation for the Nusselt number (Nu), the dimensionless number for the convection heat transfer coefficients (h), was proposed by Sieder and Tate for laminar flows in circular ducts with uniform wall temperature (Kreith and Bohn, 1993). This empirical correlation accounts for the effect of viscosity variations, which occur due to the temperature difference. This correlation gives the average Nusselt number (Nu) as:

$$\overline{Nu}_{D_H} = 1.86\left(\frac{\mathrm{Re}_{D_H}\,\mathrm{Pr}\,D_H}{L}\right)^{0.33}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} \qquad (2-24)$$

in this equation, a correction factor $(\mu_b/\mu_s)^{0.14}$ is introduced to account for the viscosity variation with temperature. This correlation is applicable in the following ranges:

$$\left(\frac{\mathrm{Re}_{D_H}\ \mathrm{Pr}\ D_H}{L}\right)^{0.33}\left(\frac{\mu_b}{\mu_s}\right)^{0.14} > 2$$

$$0.004 < \left(\frac{\mu_b}{\mu_s}\right) < 10$$

$$0.5 < \mathrm{Pr} < 16{,}000$$

This correlation can be used for the analysis of double pipe heat exchangers by dividing the heat exchanger into a finite number of dx increments with the wall temperature assumed to be uniform for each segment. With the bulk temperature of each stream known at the first segment, the wall temperature can be assumed and the corresponding fluid viscosity at the wall and fluid bulk temperature can be evaluated using the viscosity-temperature relationship given by equation 2-1. Then, the correlation can be used to evaluate the convection heat transfer coefficient. The local convection heat transfer rate for each stream can be calculated from the equations:

$$dq_t = (hA)_t (T_{bt} - T_w) \qquad (2-25)$$

$$dq_a = (hA)_a (T_w - T_{ba}) \qquad (2-26)$$

The assumed wall temperature shall be modified until the calculated convection heat transfer rates are equal:

$$dq_t \approx dq_a \qquad (2-27)$$

Then, the calculated local convection heat transfer rates will be considered as dq_x and will be substituted into equations 2.18, 2.19, 2.22 and 2.23 to get the local temperature difference for the tube and annulus streams. From the calculated temperature difference, the next axial bulk stream temperature can be obtained. The same process shall be repeated until the other side of the heat exchanger is reached. A flow diagram showing this process for parallel and counter flow heat exchangers is given in Appendix A.

2.4.2 Thermal Effectiveness and Number of Transfer Units (NTU)

Once the temperature distribution is known, the heat exchanger Number of Transfer Units (NTU) and the thermal effectiveness (ε) can be calculated. The Number of Transfer Units (NTU) is defined as (Krieth and Bohn, 1993):

$$NTU = UA/C_{\min} \qquad (2-28)$$

Where the quantity UA can be calculated from the equation:

$$q = UA(LMTD) \qquad (2-29)$$

Where:

$$q = (\dot{m}C_p)_a(T_{ae} - T_{ai}) \qquad (2-30)$$

The thermal effectiveness (ε) can be calculated from the equation:

$$\varepsilon = \frac{(\dot{m}C_p)_t (T_{ai} - T_{ae})}{C_{\min}(T_{ti} - T_{ai})} \qquad (2-31)$$

2.4.3 Entropy Generation

The local and total entropy generation can be calculated for each stream as follows:

1- The local entropy generation (due to the heat transfer rate and viscous friction) for the tube side:

$$d\dot{S}_t = (d\dot{S}_t)_{heat\,transfer} + (d\dot{S}_t)_{viscous\,friction}$$

$$= -\frac{dq_x}{T_{xt}} + \frac{d\dot{W}_{ct}}{T_{xt}} \qquad (2-32)$$

The total entropy generation in the tube side would be the sum of all local entropy generation values as calculated from equation 2-32.

2- The local entropy generation (due to the heat transfer rate and viscous friction) for the annulus side:

$$d\dot{S}_a = (d\dot{S}_a)_{heat\,transfer} + (d\dot{S}_a)_{viscous\,friction}$$

$$= \frac{dq_x}{T_{xa}} + \frac{d\dot{W}_{ca}}{T_{xa}} \qquad (2-33)$$

The total entropy generation in the annulus side would be the sum of all local entropy generation values as calculated from equation 2-33.

3- The local entropy generation in the heat exchanger would be the sum of equation 2.32 and equation 2.33.

$$d\dot{S}_{total} = -\frac{dq_x}{T_{xt}} + \frac{d\dot{W}_{ct}}{T_{xt}} + \frac{dq_x}{T_{xa}} + \frac{d\dot{W}_{ca}}{T_{xa}} \qquad (2-34)$$

The total entropy generation in the heat exchanger would be the sum of all local entropy generation value as calculated from equation 2.34. The total entropy generation number would be taken as the total entropy generation divided by the heat capacity rate (C_{min}) :

$$Total\ Entropy\ Generation\ Number = \frac{\dot{S}_{total}}{(\dot{m}C_p)_t} \qquad (2-35)$$

2.5 Validation of calculation results

In order to validate the results obtained by the calculation methodology, the thermal effectiveness-NTU curves for the case of no viscous friction and constant viscosity were generated for both parallel and counter flow heat exchangers using the calculation methodology for different heat capacity rate ratios (C_{min} /C_{max}). The obtained curves were compared with the thermal effectiveness-NTU curves given by Kays and London for parallel and counter flow heat exchangers (Kreith and Bohn, 1993) as shown Fig. 2.4 and Fig. 2.5. It was found that the curves generated using the calculation methodology are in excellent agreement with the curves given by Kays and London.

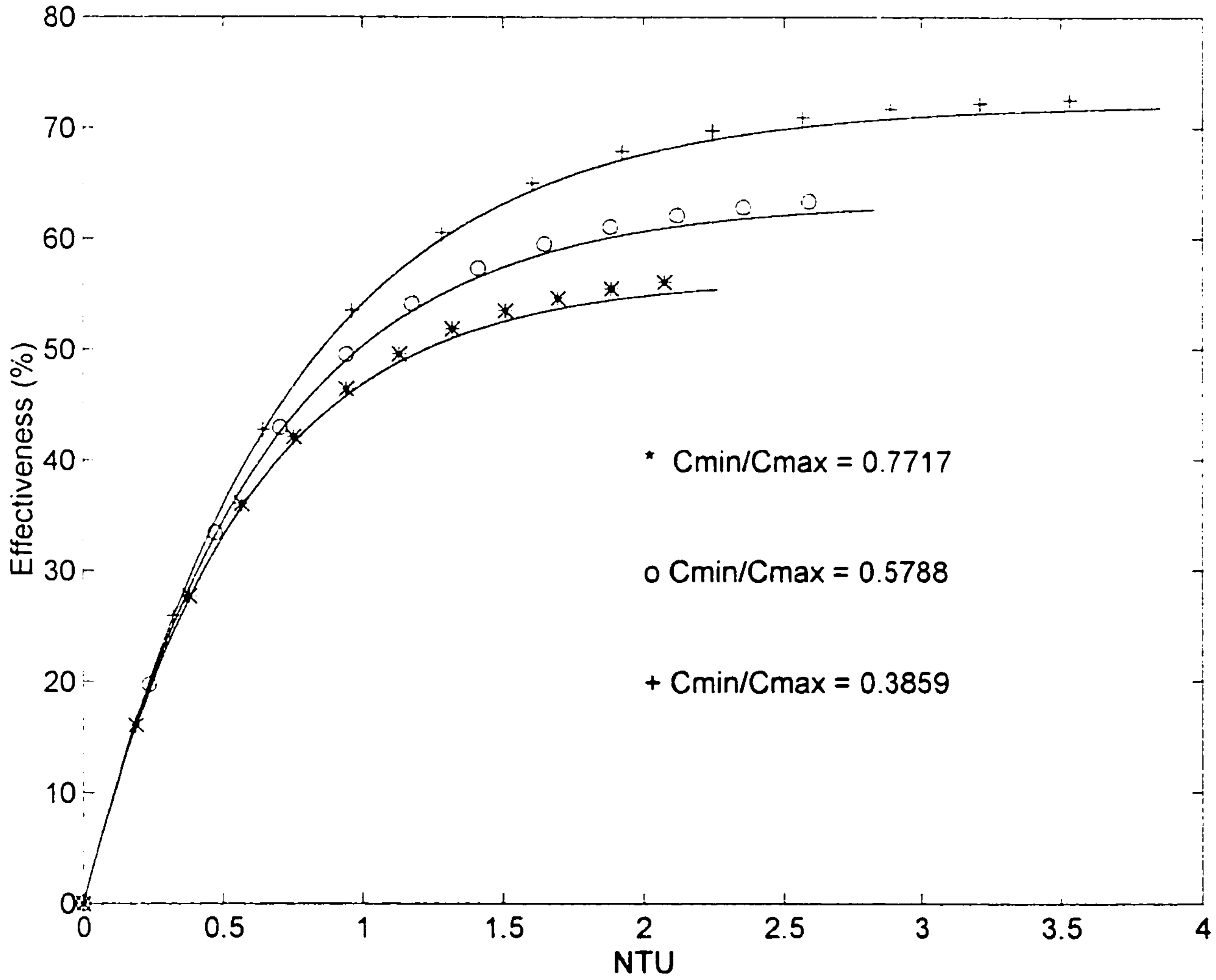


Fig. 2.4 Validation of the calculation methodology for parallel flow heat exchangers (continuous line represents Kays and London thermal effectiveness-NTU curves).

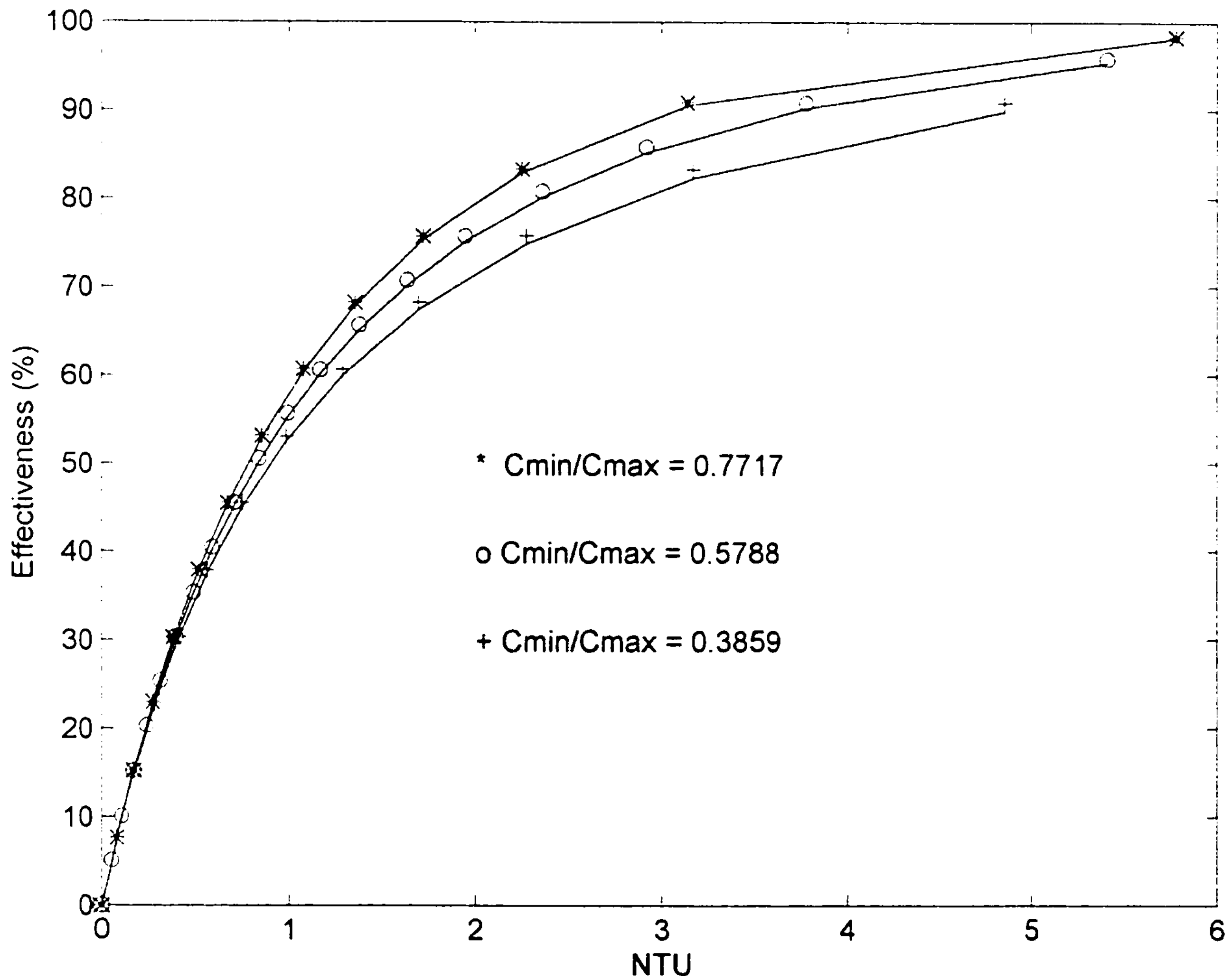


Fig. 2.5 Validation of the calculation methodology for counter flow heat exchangers (continuous line represents Kays and London thermal effectiveness-NTU curves).

CHAPTER 3

INVESTIGATION OF THE EFFECT OF VISCOSITY ON THE PERFORMANCE OF PARALLEL AND COUNTER FLOW HEAT EXCHANGERS

3.1 Introduction

The mathematical formulation and the numerical solution scheme explained in the previous chapter can be used to investigate the effect of the temperature dependent viscosity variation on the performance of parallel and counter flow heat exchangers.

The specific objective of this investigation is to observe the deviation of the heat exchanger performance from its performance obtained for the case of no viscous friction. In the no viscous friction case, the heat exchanger performance is studied with the assumption that the effect of the temperature dependence of viscosity variation and the viscous frictional heating is negligible. The analysis shall be performed for three cases and the behaviors corresponding to different cases shall be compared. These cases are:

1- No viscous friction case.

2- Viscous friction with constant viscosity.

3- Viscous friction with variable viscosity.

As discussed in the previous chapter, the performance of the heat exchanger can be measured by computing the thermal effectiveness and entropy generation at different sizes of the heat exchanger expressed in Number of Transfer Units (NTU). In this way, The critical heat exchanger size range, at which the deviation from the no viscous friction behavior becomes more significant, can be identified. The amount of deviation can also be affected by some other parameters such as the heat capacity rate ratios (C_{min}/C_{max}) and the inlet temperature.

Throughout the analysis, the high viscosity fluid (glycerol) will be considered as the tube fluid and water will be considered as the annulus fluid. The fluid and the mass flow rate of the annulus side will be fixed in order to observe the effect of changing the parameters of the tube side on the analysis.

The heat exchanger geometry and the operating conditions as well as the properties of the running fluids considered in the analysis are outlined in Appendix B. Some parameters in Appendix B will be modified, as indicated in the following sections, in order to study their effect on the analysis.

3.2 Investigation of Parallel Flow Heat Exchangers

The effect of temperature dependent viscosity variation on the performance of parallel flow heat exchangers for different tube fluid inlet temperature and heat capacity rate ratios (C_{min}/ C_{max}) is discussed in this section.

3.2.1 Effect of the Tube Fluid Inlet Temperature

Effect of different tube fluid inlet temperatures on the amount of deviation of the viscous friction performance from the no viscous friction performance will be studied for both cases of constant viscosity and variable viscosity. The data given in table 3.1 will be used in the analysis. The tube fluid inlet temperature will be taken as 283 K first and then modified to study its effect on the analysis.

The thermal effectiveness-NTU curves for the three cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity are given in Fig. 3.1. In this figure, the curve corresponding to no viscous friction case is in excellent agreement with the curve given by Kays and London for the same heat capacity rate ratio (C_{min}/C_{max}) (Kreith and Bohn, 1993).

This curve indicates that effectiveness of the heat exchanger increases as the size expressed in NTU increases. The rate of this thermal effectiveness increase is higher at small NTU values. This is due to the large temperature difference between the cold and hot streams near the inlet of the heat exchanger. This large temperature difference decreases as the fluids travel in the axial direction leading to lower heat transfer rate between the two fluids and consequently lower increase in the thermal effectiveness. The change in the temperature difference between the two fluids can be seen in Fig. 3.2.

The thermal effectiveness-NTU curve for the viscous friction with variable viscosity case (curve 3 in Fig. 3.1) shows a considerable reduction in the thermal effectiveness of the heat exchanger at large NTU values. The thermal effectiveness remains the same as that of the no viscous friction case for small NTU values. This similar behavior for small heat exchanger sizes is due to the relatively low viscosity values of the tube fluid near the inlet where the temperature is relatively high. The viscosity values of water are generally much smaller than that of glycerol and the water viscosity dependence on temperature is also small compared to that of glycerol. Therefore, the effect of the viscosity in the water side on the thermal effectiveness is much less significant. As the fluids travel downstream, the tube fluid temperature

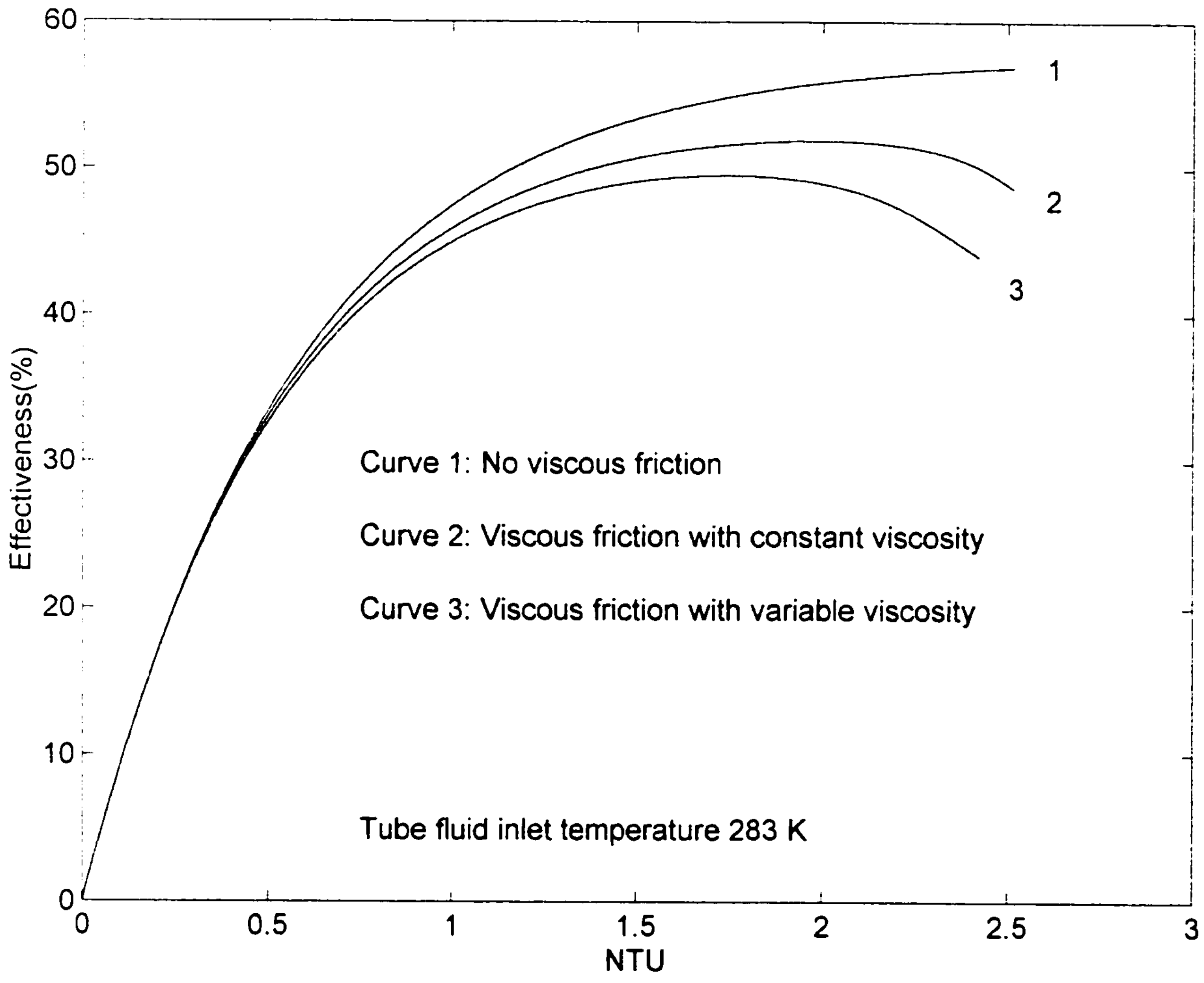


Fig. 3.1 Thermal effectiveness-NTU curves of parallel flow heat exchangers for the cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity (Tube fluid inlet temperature=283 K, Heat capacity rate ratio = 0.7717).

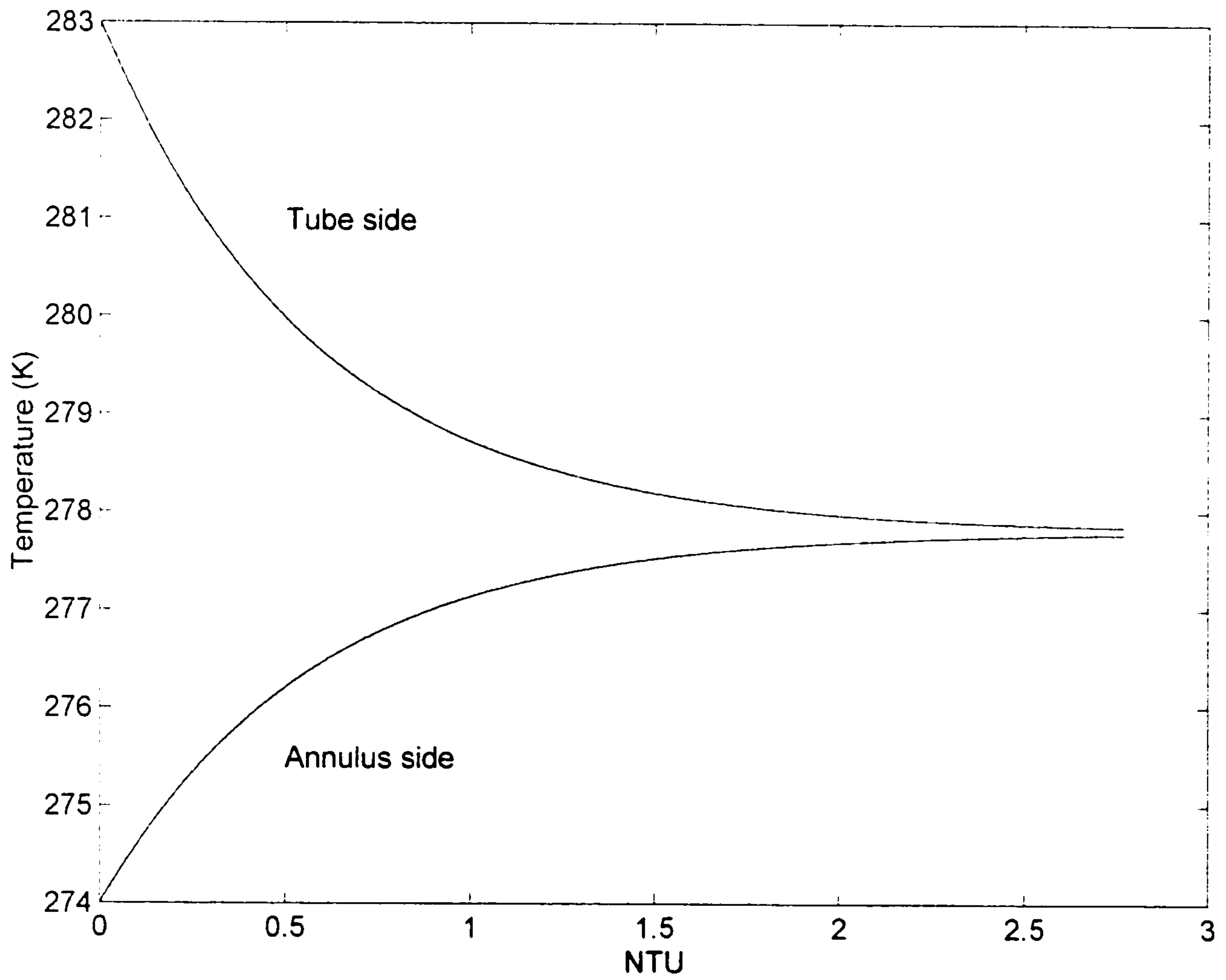


Fig. 3.2 Axial bulk temperature profiles of parallel flow heat exchangers for the case of no viscous friction (Tube fluid inlet temperature = 283 K, Heat capacity rate ratio = 0.7717).

decreases and the viscosity increases. This makes the viscous frictional heating more significant and causes the temperature of the tube fluid to decrease at a slower rate than that of the no viscous friction case as can be seen in Fig. 3.3. This change in the temperature profile tends to reduce the effectiveness of the heat exchanger and initiates the deviation from the no viscous friction behavior at an NTU value of approximately 0.5 (Fig. 3.1).

As the two fluids travel further downstream, the temperature difference becomes smaller and results in smaller heat transfer rate between the two fluids. This leads to less cooling effect for the tube fluid and makes the effect of viscous frictional heating stronger. As a result of this, the deviation from the no viscous friction behavior becomes larger as the size of the heat exchanger increases. As detailed in Fig. 3.1, at NTU=1.75, the thermal effectiveness is 54.9% for the no viscous friction case while at the same value of NTU, the thermal effectiveness is 49.52% for the viscous friction with variable viscosity case. This is equivalent to a 9.8% reduction in the thermal effectiveness.

The effect of viscous frictional heating continues to increase as the two fluids travel in the axial direction until the temperature profile of the tube fluid reaches a critical point of minimum temperature after which the tube fluid temperature start increasing rather than decreasing. This point represents the highest thermal effectiveness value on the thermal effectiveness-NTU curve. The increase in the tube fluid temperature that happens after this point causes the thermal effectiveness to drop as the heat exchanger size increases.

The heat exchanger behavior pertaining to the case of viscous friction with constant viscosity on both streams is shown in Fig. 3.1 as curve 2. In this case, the viscosity of

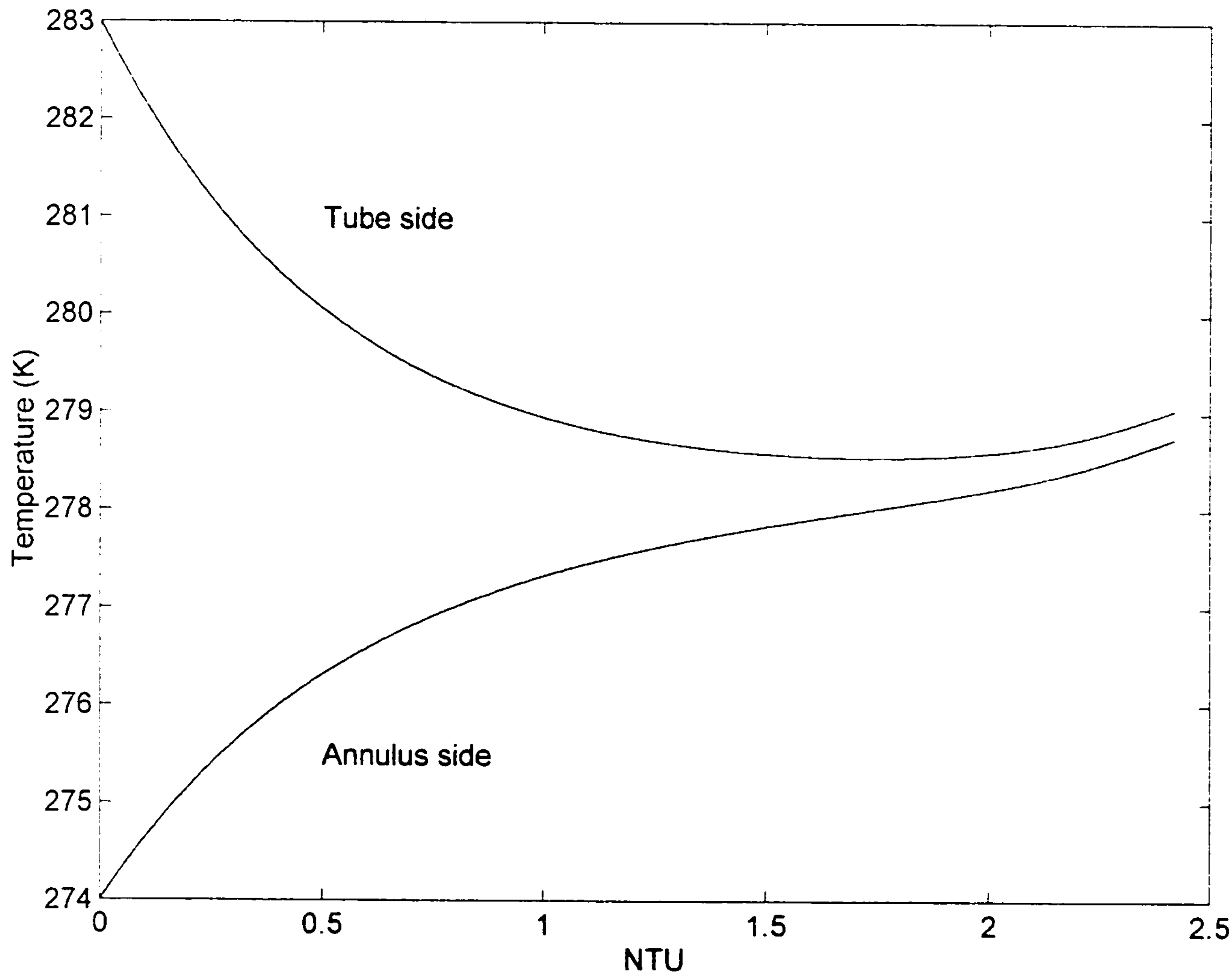


Fig. 3.3 Axial bulk temperature profiles of parallel flow heat exchangers for the case of viscous friction with variable viscosity (Tube fluid inlet temperature = 283 K, Heat capacity rate ratio = 0.7717).

each stream is evaluated at the inlet temperature of that stream and considered to be constant for all points in the heat exchanger. The general trend of the behavior pertaining to this case is similar to that of the viscous friction case with variable viscosity (curve 3) discussed earlier. However, the viscosity of the tube fluid increases for the variable viscosity case as the tube fluid travels downstream resulting in higher viscous frictional heating. For the case of constant viscosity, the viscous frictional heating remains constant as the fluid travels downstream. For this reason, the curve of the viscous friction with constant viscosity case exhibits smaller deviation from the curve of the no viscous friction case. For example, the deviation from the no viscous friction behavior at NTU=1.75 for the viscous friction with variable viscosity case is 9.8% while at the same NTU value, the deviation from the no viscous friction behavior for the viscous friction with constant viscosity is 7.61% only. The highest thermal effectiveness reached for the case of constant viscosity is 51.9% while the highest thermal effectiveness of the variable viscosity case is 49.5% only.

The results obtained for the heat exchanger performance with viscous friction taken into consideration is in good agreement with the results obtained analytically by Şahin (1997) for parallel flow heat exchangers.

In order to observe the effect of tube fluid inlet temperature on the analysis, the same data given in table 3.1 will be used with tube fluid inlet temperatures of 280 K and 277 K instead of 283 K. The corresponding thermal effectiveness-NTU curves and temperature profile behaviors is shown in Fig. 3.4 for the tube fluid inlet temperature of 280 K and Fig. 3.5 for the tube fluid inlet temperature of 277 K. From these two figures, it is clear that as the tube fluid inlet temperature decreases, the deviation from the no viscous

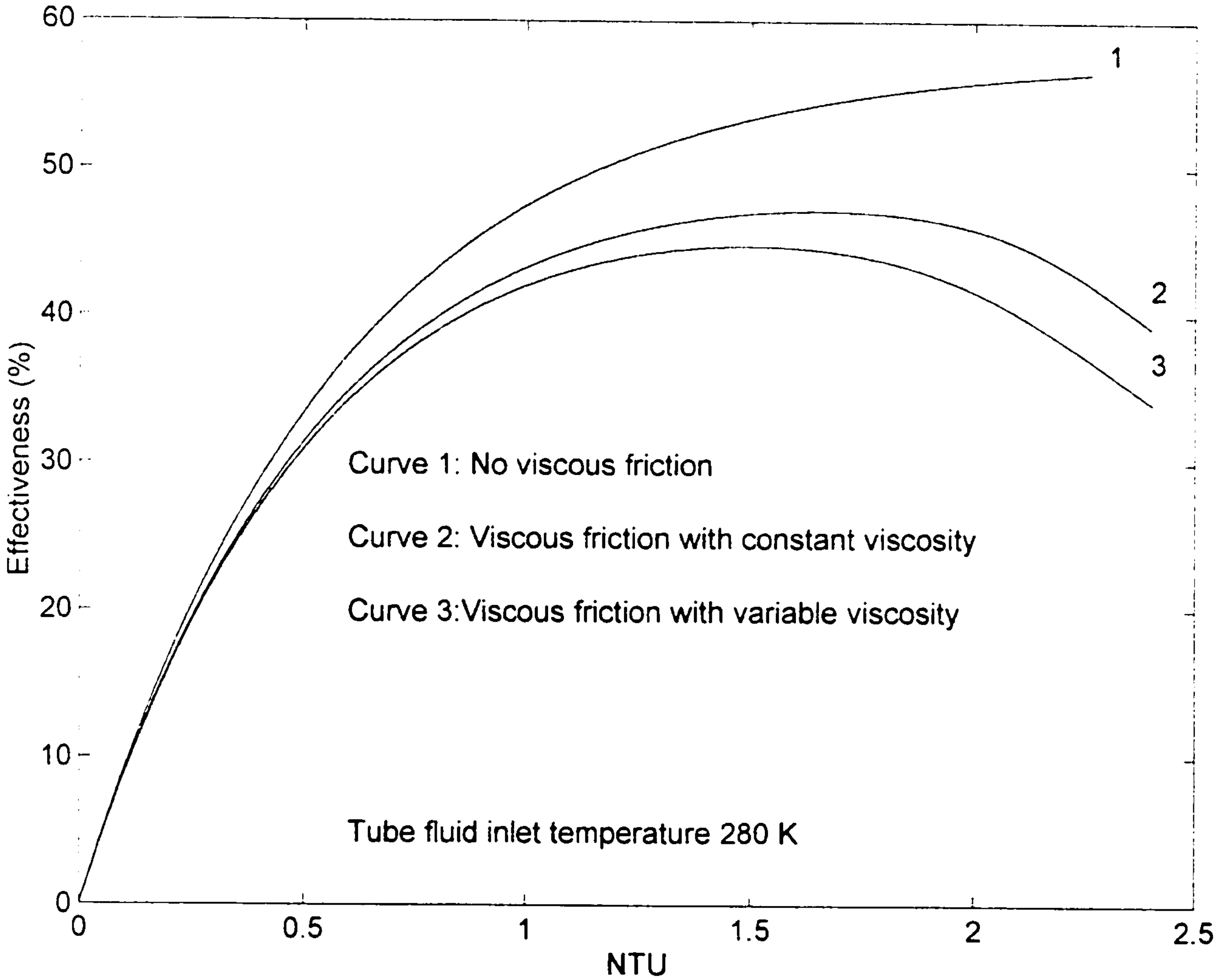


Fig. 3.4 Thermal effectiveness-NTU curves of parallel flow heat exchangers for the cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity (Tube fluid inlet temperature=280 K, Heat capacity rate ratio = 0.7717).

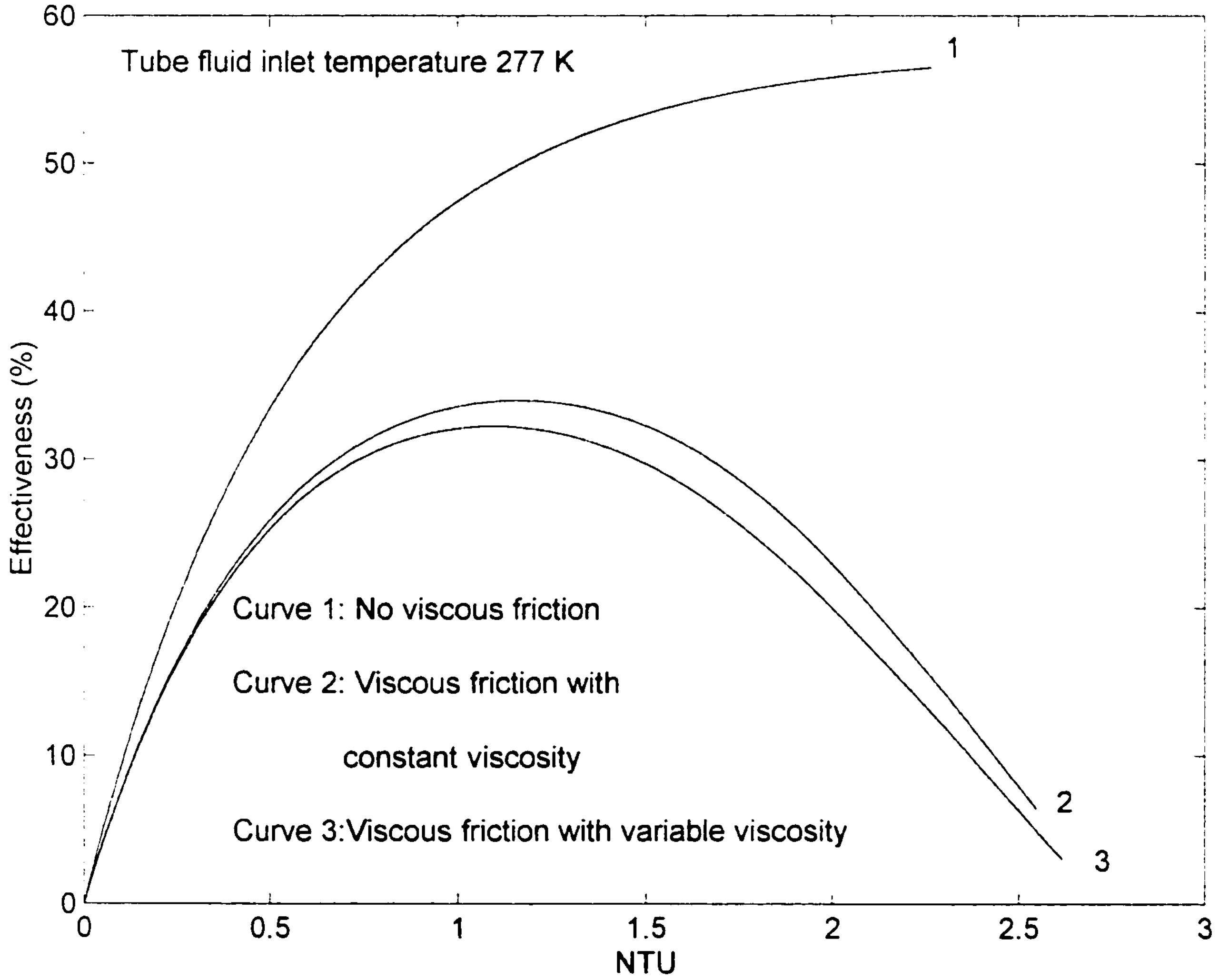


Fig. 3.5 Thermal effectiveness-NTU curves of parallel flow heat exchangers for the cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity (Tube fluid inlet temperature=277 K, Heat capacity rate ratio = 0.7717).

friction curve increases. This is due to the higher fluid viscosity at lower temperatures, which causes higher viscous frictional heating. For the tube fluid inlet temperature of 280 K, the thermal effectiveness of the heat exchanger for the viscous friction with variable viscosity case (curve 3 in Fig. 3.4) at NTU =1.75 is 44.03% whereas the thermal effectiveness of the same case for 283 K (curve 3 in Fig. 3.4) at the same NTU value is 49.52%. This means that the thermal effectiveness corresponding to the tube fluid inlet temperature of 280 K is lower than the thermal effectiveness corresponding to the tube fluid inlet temperature of 283 K by 11.09% at NTU = 1.75. For the tube fluid inlet temperature of 277 K, the thermal effectiveness at the same NTU value for the viscous friction with variable viscosity case (curve 3 in Fig. 3.5) is 25.66% which is lower than the thermal effectiveness of the tube fluid inlet temperature of 283 K by 48.18%.

The local entropy generation within the heat exchanger for the 283 K tube fluid inlet temperature is illustrated in Fig. 3.6. The maximum absolute values of the local entropy generation of the tube side and the annulus side for the case of no viscous friction occur at the inlet and decreases as the fluids travel in the axial direction as can be seen in Fig. 3.6 (a). All values of the local entropy generation of the tube side are negative while all values of the local entropy generation of the annulus side are positive because the heat is transferred from the tube fluid, which has the higher temperature, to the annulus fluid, which has the lower temperature. The local entropy generation for both fluids has its highest absolute values at the inlet because of the large temperature difference between the two fluids. This large temperature difference results in large heat transfer rate. The local entropy generation due to viscous frictional heating for the tube fluid for the cases of constant viscosity and variable viscosity are

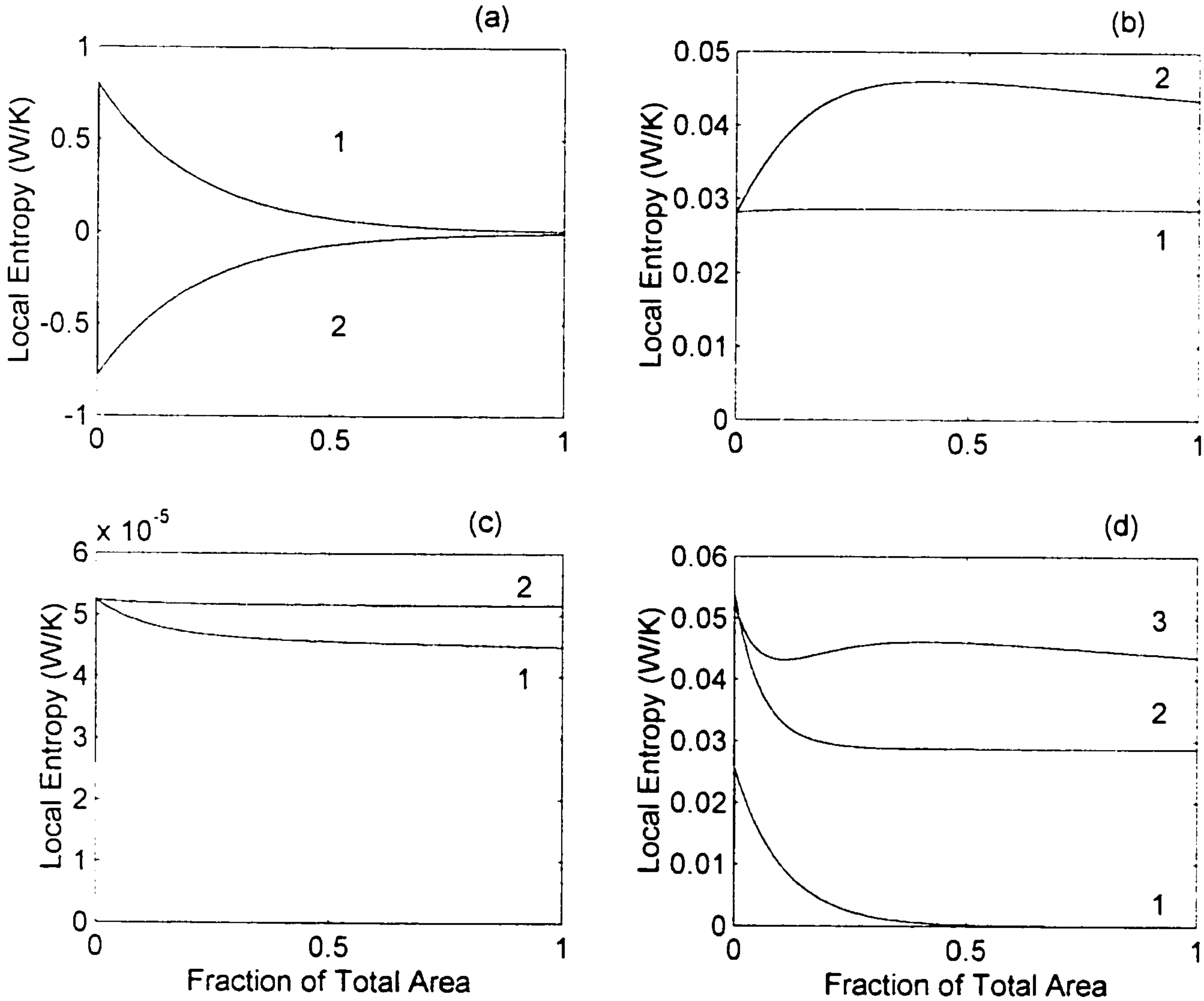


Fig. 3.6 Local entropy generation of parallel flow heat exchangers in: (a) the annulus side (curve 1) and the tube side (curve 2) for the no viscous friction case, (b) the tube side due to viscous friction with constant viscosity (curve 1) and variable viscosity (curve 2), (c) Local entropy generation in the annulus side due to viscous friction with constant viscosity (curve 2) and variable viscosity (curve 1), (d) Total local entropy generation for the cases of no viscous friction (curve 1), viscous friction with constant viscosity (curve 2) and viscous friction with variable viscosity (curve 3); (Tube inlet temperature = 283 K, Heat capacity rate ratio = 0.7717).

shown in Fig. 3.6 (b) and for the annulus fluid in Fig. 3.6 (c). From these two figures, it can be seen that the local entropy generation due to viscous frictional heating in the tube fluid is much higher than the local entropy generation due to viscous frictional heating in the annulus side with positive values for both of them. This difference is a result of the high viscosity of the tube fluid (glycerol) compared to the annulus fluid (water).

The extra local entropy generation caused by the viscous frictional heating leads to higher values of total local entropy generation (the sum of the local entropy generation on both sides) compared to the no viscous friction case as shown in Fig. 3.6 (d). In this figure, the total local entropy generation corresponding to the variable viscosity case (curve 3) has higher values than the total local entropy generation corresponding to the constant viscosity case (curve 2). This difference happens because the tube fluid viscosity increases as the fluid runs in the axial direction and its temperature decreases. This variable viscous friction reaches its maximum value at the minimum value of the tube side temperature. The difference between the constant viscosity case curve and the variable viscosity case curve increases as the temperature of the tube fluid decreases and reaches its maximum at the minimum temperature point.

The total entropy generation number of the heat exchanger is shown in Fig. 3.7 (a). In this curve, the difference in the local entropy generation values along the heat exchanger accumulates resulting in larger total entropy generation number difference as the heat exchanger size increases. This indicates that the consideration of temperature-dependent viscosity variation becomes more important for larger sizes of the heat exchanger. For the large NTU value of 1.75, the difference between the

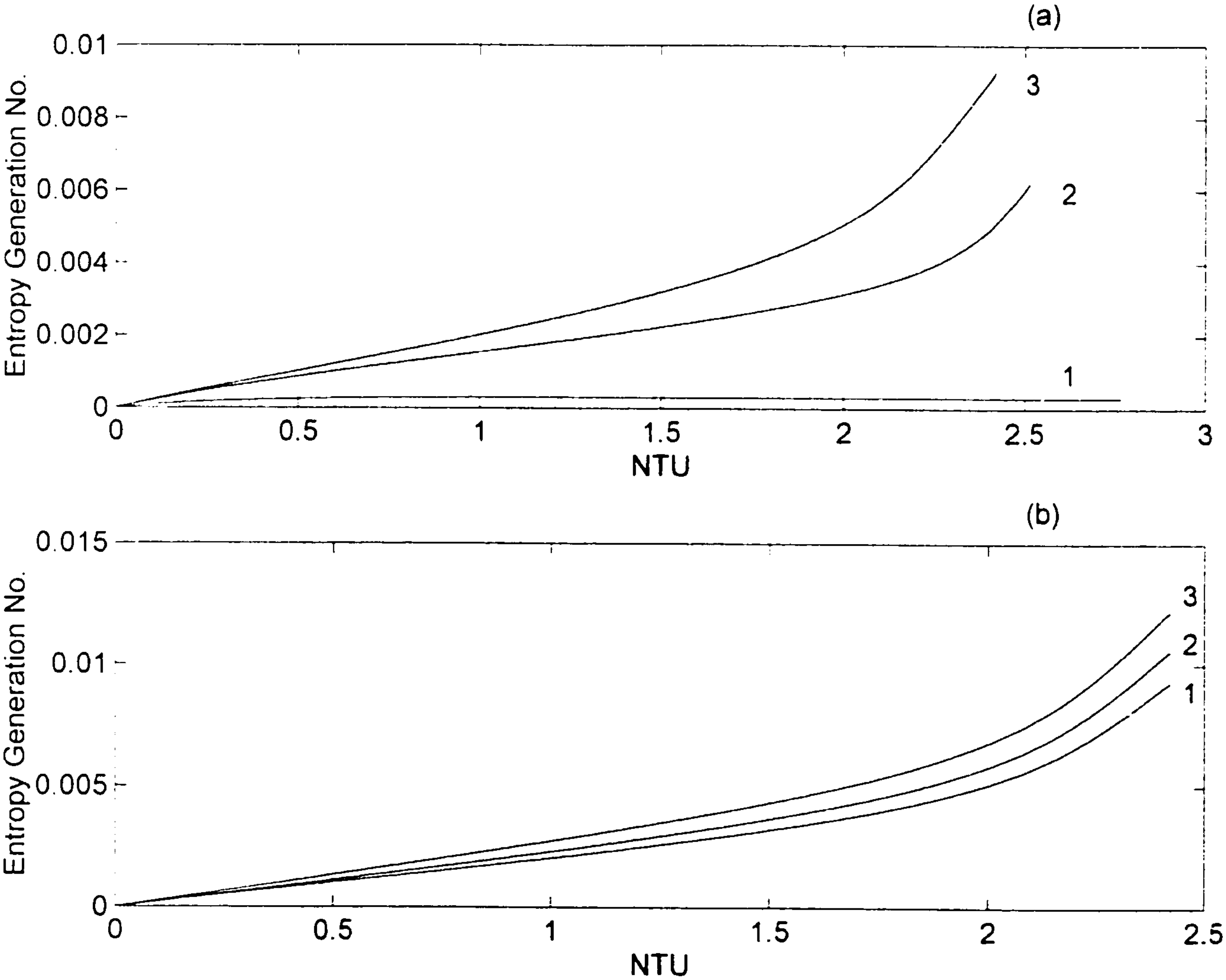


Fig. 3.7 Total entropy generation of parallel flow heat exchangers for: (a) Tube fluid inlet temperature = 283 K, No viscous friction (curve 1), viscous friction with constant viscosity (curve 2) and viscous friction with variable viscosity (curve 3), (b) Viscous friction with variable viscosity, Tube fluid inlet temperature 283 K (curve 1), 280 K (curve 2) and 277 K (curve 3); (Heat capacity rate ratio = 0.7717).

constant viscosity curve and the variable viscosity case is approximately ten times higher than the difference between the two curves at the small NTU value of 0.5.

The total entropy generation number associated with the viscous friction with variable viscosity case is shown in Fig. 3.7 (b) for different inlet tube temperatures. It is clear from this figure that the entropy generation of the heat exchanger increases more, for lower tube inlet temperature, as the heat exchanger size increases.

The good design of heat exchangers should make the most effective use of energy during the operation. For this reason, both the effectiveness of the heat exchanger and the entropy generation associated with its operation should be considered carefully in order to have a good design. From the discussion given in this section, the thermal effectiveness of the heat exchanger increases as the size of the heat exchanger increases. For the case of viscous friction, the effectiveness increases until it reaches a maximum value at a certain size and starts decreasing for larger sizes. On the other hand, the total entropy generation within the heat exchanger increases also as the heat exchanger size increases. Since the viscous frictional heating is the source of both entropy generation and pressure drop in the heat exchanger, the entropy generation can be thought of as a measure of pumping power requirements and consequently the energy required for the operation of the heat exchanger. Therefore, the use of energy is considered to be more effective when the increase in the heat exchanger effectiveness per unit of entropy generation is higher. In this sense, a heat exchanger operation cost effectiveness can be defined as the ratio of the thermal effectiveness to the total entropy generation at a given size of the heat exchanger. This cost effectiveness can be expressed mathematically as:

$$Cost\ Effectiveness = \frac{Heat\ Exchanger\ Thermal\ Effectiveness/10}{Fraction\ of\ Total\ Entropy\ Generation\ Number} \tag{3.1}$$

However, the primary objective of a heat exchanger is to transfer heat between two fluids. This objective is well represented by the heat exchanger thermal effectiveness. Thus, the sum of both the cost effectiveness, as defined above, and the thermal effectiveness should be maximized in order to achieve the heat exchanger optimum operation point. This sum of thermal effectiveness and cost effectiveness can be called the heat exchanger operation factor. The cost effectiveness, thermal effectiveness and the operation factor are shown in Fig. 3.8 for the no viscous friction case and in Fig. 3.9 for the viscous friction with variable viscosity case at different heat exchanger sizes expressed in Number of Transfer Units (NTU).

From Fig. 3.8, it can be seen that for the no viscous friction case there is no optimum size of the heat exchanger at which the operation factor reaches a maximum value since all curves in the figure increase as the size increases. However, for the case of viscous friction with variable viscosity (Fig. 3.9), there is an optimum size of the heat exchanger at which the operation effectiveness reaches a maximum value.

The effect of the tube inlet temperature on the optimum size of the heat exchanger can be seen by plotting where the operation factor curves corresponding to tube inlet temperatures of 283 K, 280 K and 277 K for the viscous friction with variable viscosity case as shown in Fig. 3.10. From this figure, the optimum heat exchanger sizes for 283 K, 280 K and 277 K tube inlet temperatures occur at approximate NTU values of 1.0, 0.75, 0.5 respectively. This indicates that the optimum size of the heat exchanger decreases as the tube fluid inlet temperature decreases.

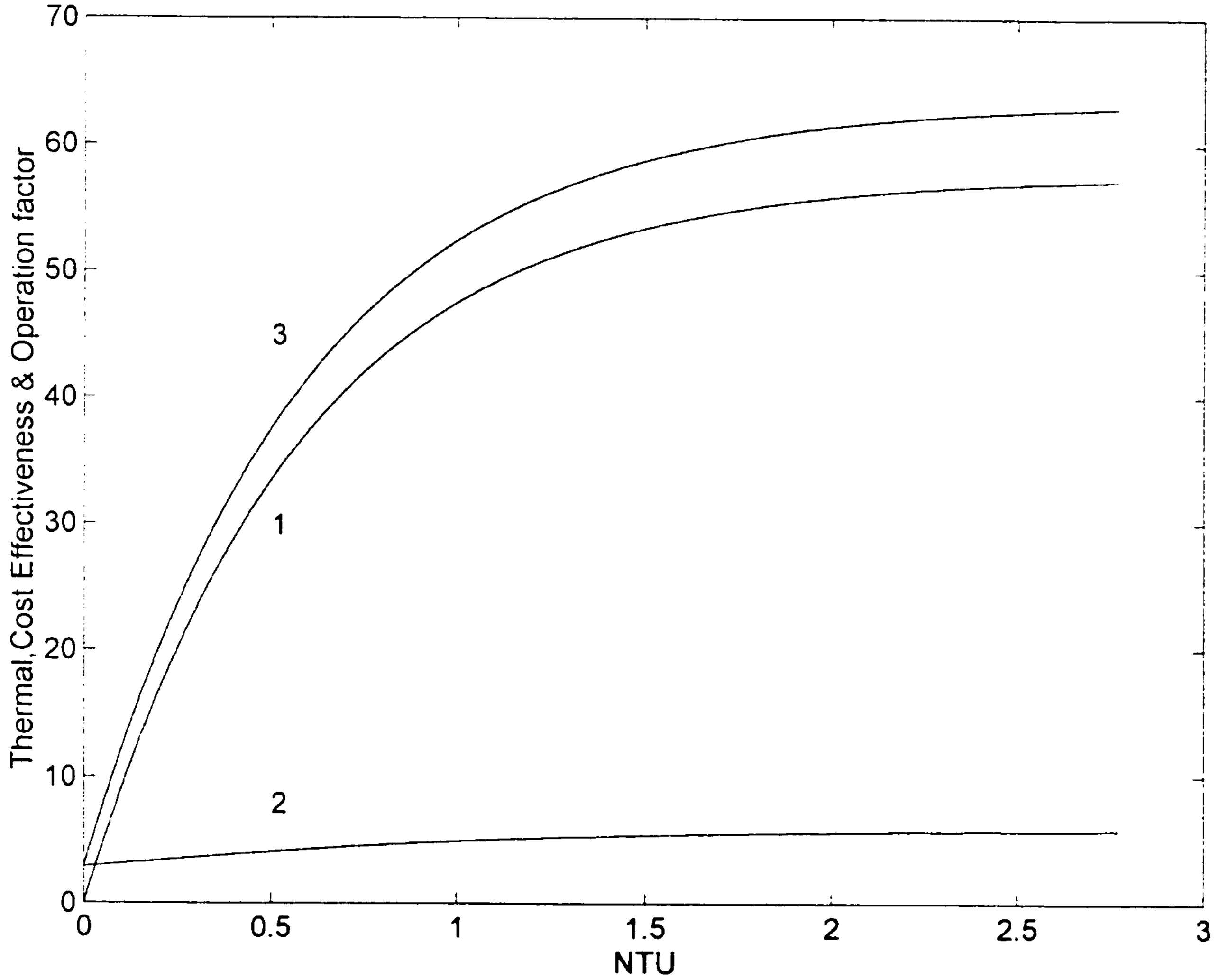


Fig. 3.8 Thermal effectiveness (curve 1), cost effectiveness (curve 2) and operation factor (curve 3) of parallel flow heat exchangers for the no viscous friction case (Tube fluid inlet temperature=283 K, Heat capacity rate ratio = 0.7717).

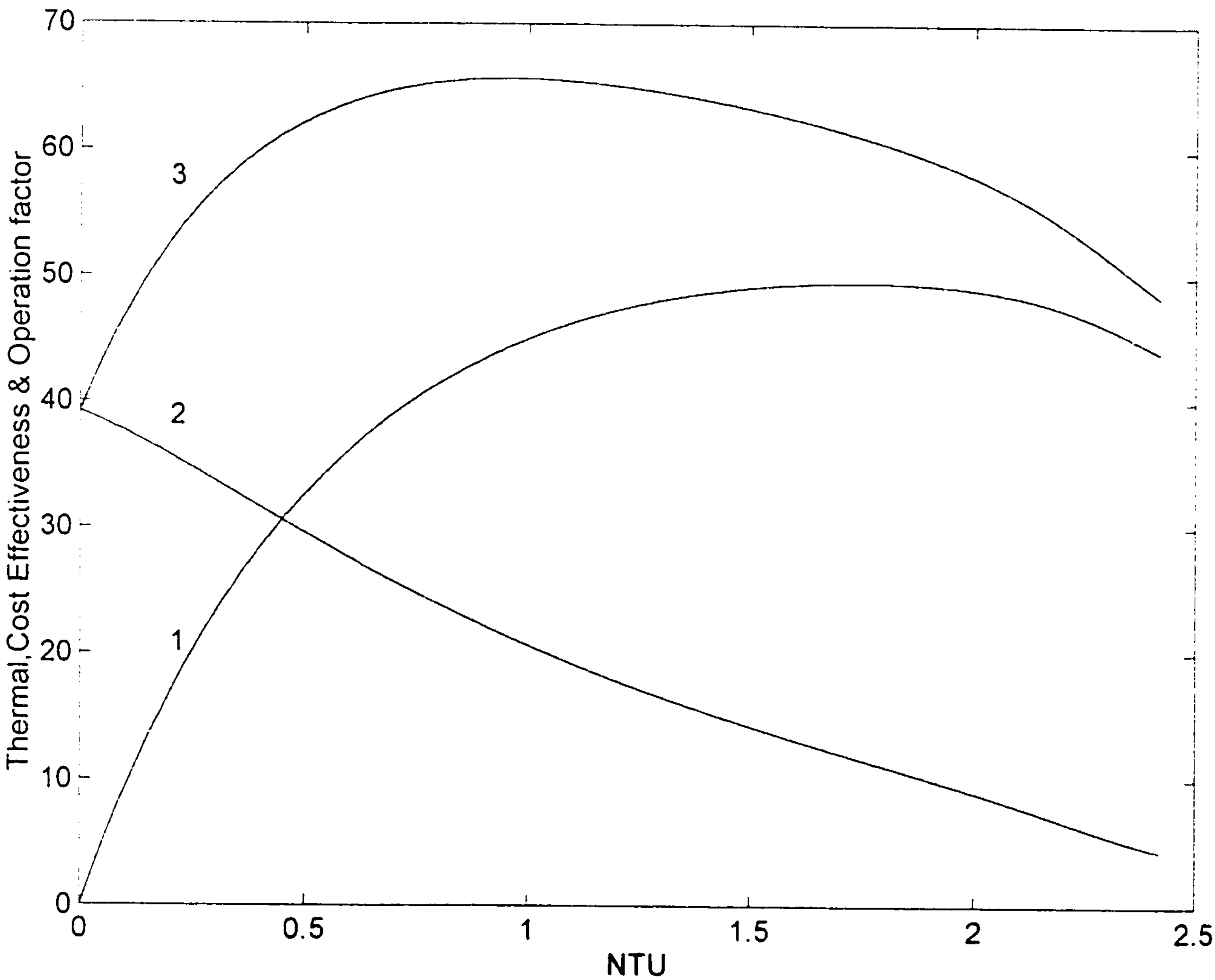


Fig. 3.9 Thermal effectiveness (curve 1), cost effectiveness (curve 2) and operation factor (curve 3) of parallel flow heat exchangers for the viscous friction with variable viscosity case (Tube fluid inlet temperature=283 K, Heat capacity rate ratio = 0.7717).

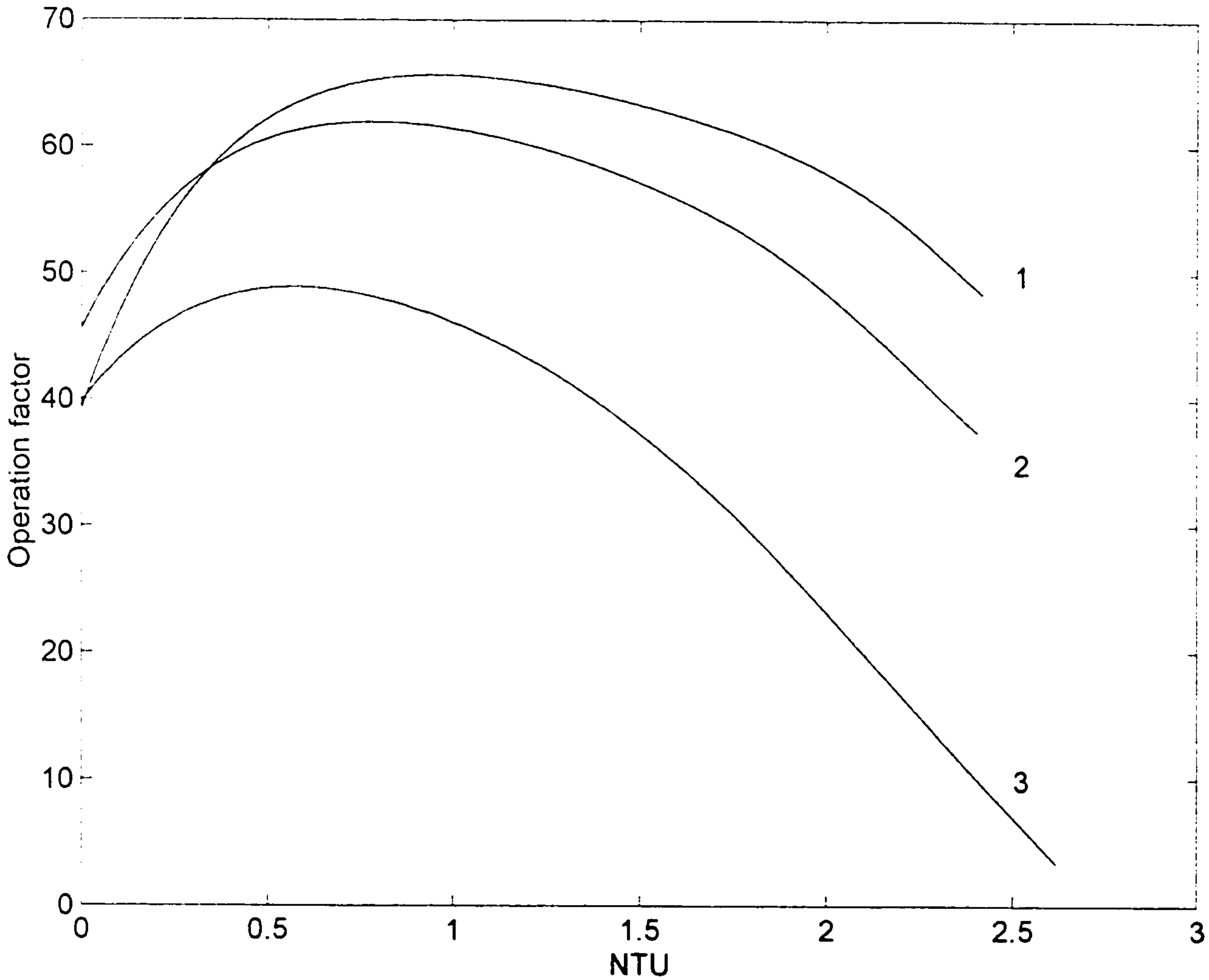


Fig. 3.10 Operation factors of parallel flow heat exchangers for the viscous friction with variable viscosity case, Tube fluid inlet temperature 283 K (curve 1), 280 K (curve 2) and 277 K (curve 3); (Heat capacity rate ratio = 0.7717).

3.2.2 Effect of the Heat Capacity Rate Ratio (C_{min}/C_{max})

In this section, the same heat exchanger performance measures discussed in the pervious section will be investigated for different heat capacity rate ratios (C_{min}/C_{max}). These performance measures are the heat exchanger thermal effectiveness, the total entropy generation number and the operation factor.

In the investigation, three different heat capacity rate ratios (C_{min}/C_{max}) will be used in order to compare the behaviors corresponding to each one of them. These heat capacity rate ratios (C_{min}/C_{max}) are 0.7717, 0.5788 and 0.3859. These ratios correspond to tube fluid mass flow rates of 4 kg/s, 3 kg/s and 2 kg/s respectively. The tube fluid inlet temperature will be fixed at 283 K.

The effectiveness-NTU curves pertaining to the no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity cases for the heat capacity rate ratio (C_{min}/C_{max}) of 0.7717 are shown in Fig. 3.1. The peak value of the thermal effectiveness for the variable viscosity case is 49.5% while the peak value for the constant viscosity case is 51.9%. This shows that the constant viscosity assumption results in a peak thermal effectiveness value overestimation of about 4.85% more than the peak value obtained using variable viscosity. However, this overestimation decreases to 3.48% for the heat capacity rate ratio (C_{min}/C_{max}) of 0.5788 and to 2.07% for the heat capacity rate ratio (C_{min}/C_{max}) of 0.3859. The thermal effectiveness-NTU curves for the heat capacity rate ratios (C_{min}/C_{max}) of 0.5788 and 0.3859 are shown in Fig. 3.11 and Fig.3.12 respectively. This indicates that the consideration of variable viscosity becomes more important for higher heat capacity rate ratios (C_{min}/C_{max}).

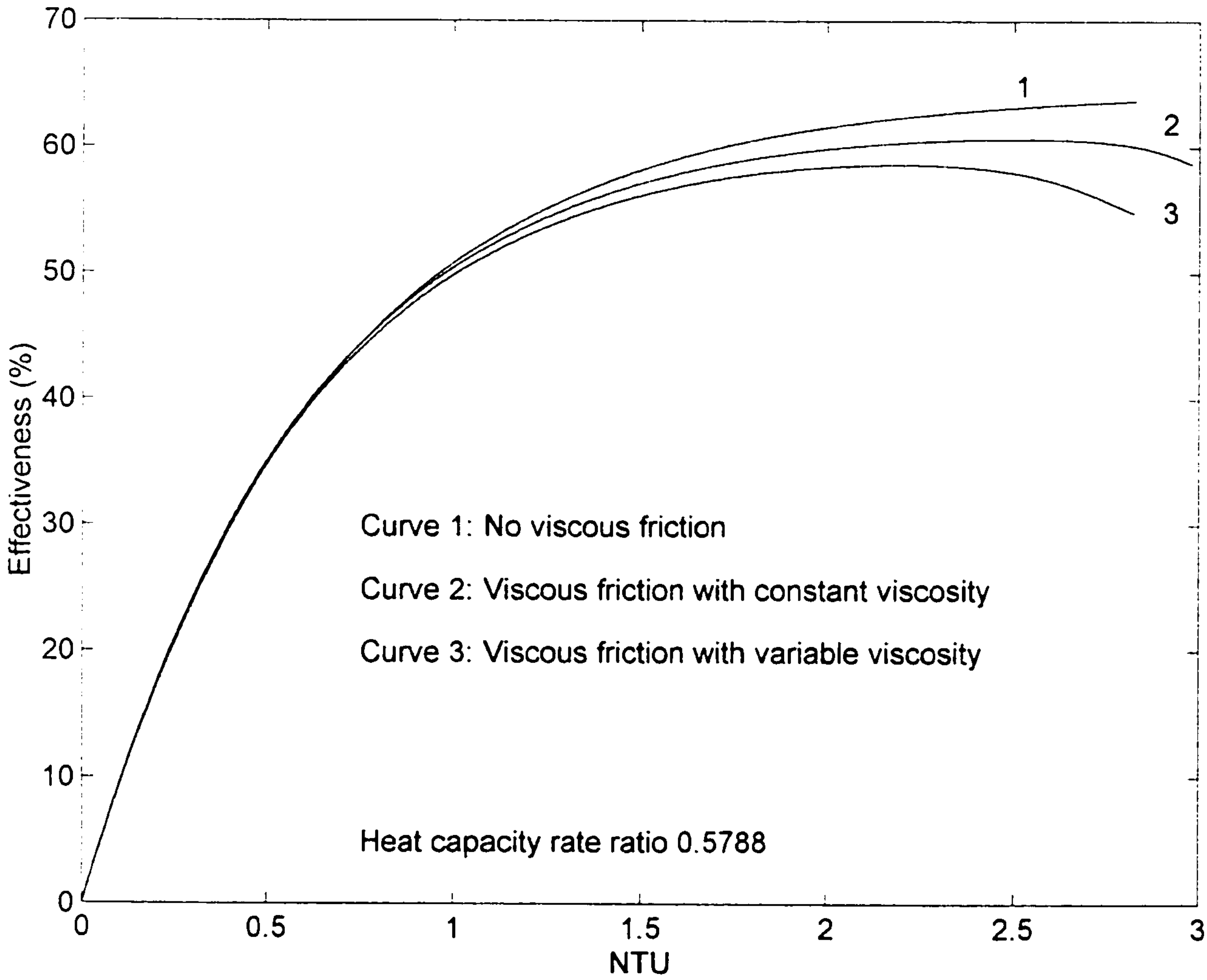


Fig 3.11 Thermal effectiveness-NTU curves of parallel flow heat exchangers for the cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity (Tube fluid inlet temperature=283 K, Heat capacity rate ratio = 0.5788).

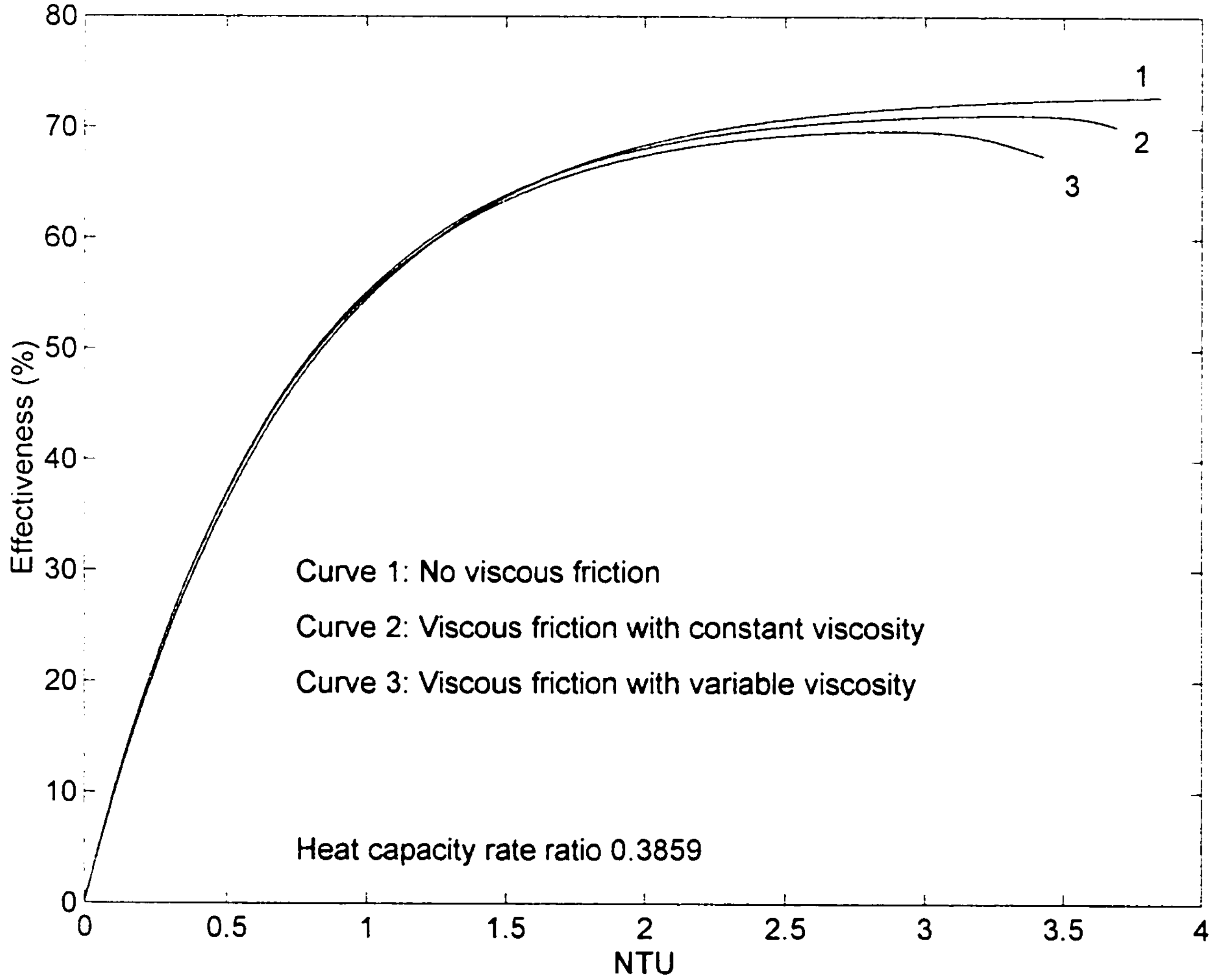


Fig 3.12 Thermal effectiveness-NTU curves of parallel flow heat exchangers for the cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity (Tube fluid inlet temperature=283 K, Heat capacity rate ratio = 0.3859).

The heat capacity rate ratio (C_{min}/C_{max}) affects also the optimum size of the heat exchanger at which the thermal effectiveness reaches a maximum value. For the heat capacity rate ratio (C_{min}/C_{max}) of 0.7717, the peak thermal effectiveness value occurs at an NTU value of 1.77. However, for the heat capacity rate ratios (C_{min}/C_{max}) of 0.5788 and 0.3859, the heat exchanger reaches its maximum thermal effectiveness at approximate NTU values of 2.22 and 2.90 respectively. This shows that the range of practical sizes of the heat exchanger becomes smaller as the heat capacity rate ratio (C_{min}/C_{max}) increases.

In order to investigate the effect of different heat capacity rate ratios (C_{min}/C_{max}) on the total entropy generation number of the heat exchanger, the total entropy generation number, pertaining to the case of viscous friction with variable viscosity, versus the size of the heat exchanger expressed in NTU values are plotted in Fig. 3.13 for different heat capacity rate ratios (C_{min}/C_{max}). In this figure, it is clear that the total entropy generation number tends to increase as the heat capacity rate ratio (C_{min}/C_{max}) increases.

This increase in the heat exchanger total entropy generation number affects the parallel flow heat exchanger cost effectiveness and operation factor curves as can be seen in Fig. 3.14 and Fig. 3.15 for the heat capacity rate ratios (C_{min}/C_{max}) of 0.5788 and 0.3859 respectively. The curve for the heat capacity rate ratio (C_{min}/C_{max}) of 0.7717 is shown in Fig. 3.9. From these figures, it can be observed that the cost effectiveness increases as the heat capacity rate ratio (C_{min}/C_{max}) decreases. As a consequence of this, the heat exchanger operation factors corresponding to different heat capacity rate ratios (C_{min}/C_{max}) differ considerably.

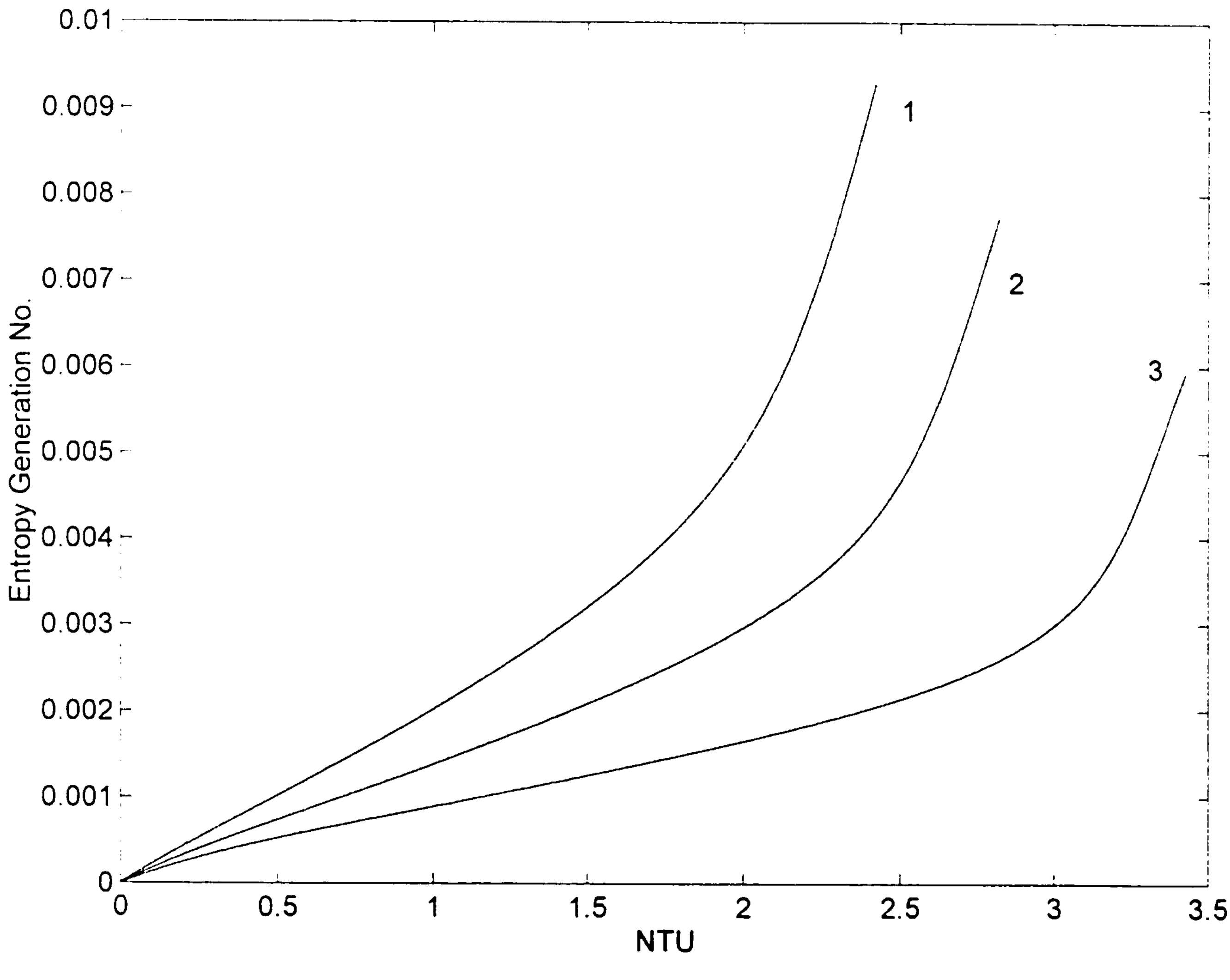


Fig. 3.13 Total entropy generation of parallel flow heat exchangers for viscous friction with variable viscosity and heat capacity rate ratio of 0.7717 (curve 1), 0.5788 (curve 2) and 0.3859 (curve 3); (Tube fluid inlet temperature = 283 K).

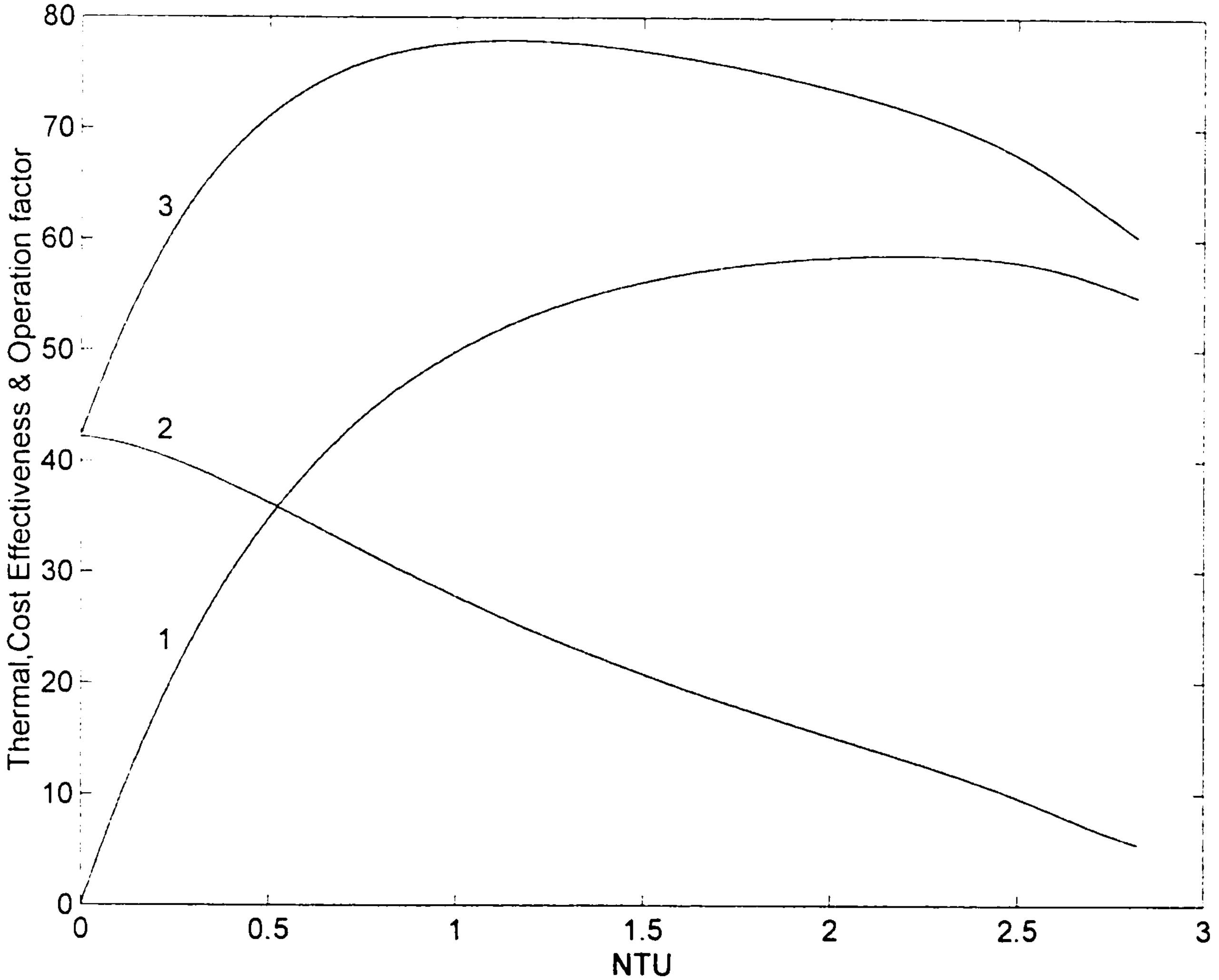


Fig. 3.14 Thermal effectiveness (curve 1), cost effectiveness (curve 2) and operation factor (curve 3) of parallel flow heat exchangers for the viscous friction with variable viscosity case (Tube fluid inlet temperature =283 K, Heat capacity rate ratio = 0.5788).

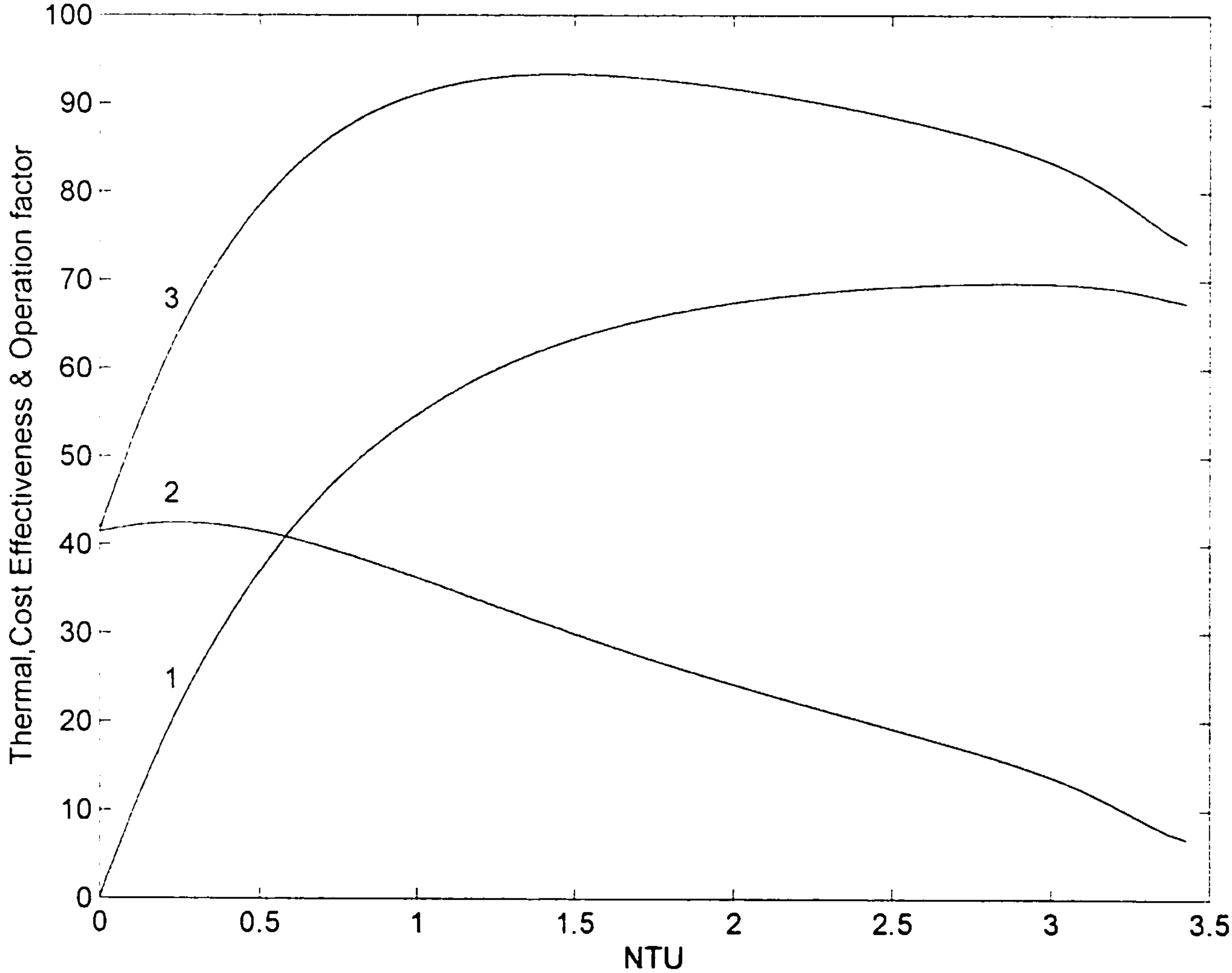


Fig. 3.15 Thermal effectiveness (curve 1), cost effectiveness (curve 2) and operation factor (curve 3) of parallel flow heat exchangers for the viscous friction with variable viscosity case (Tube fluid inlet temperature = 283 K, Heat capacity rate ratio = 0.3859).

As can be observed from these figures, the optimum size the heat exchanger at which the operation factor reaches its maximum value decreases as the heat capacity rate ratio (C_{min}/C_{max}) increases. The maximum heat exchanger operation factor also decreases as the heat capacity rate ratio (C_{min}/C_{max}) increases. For a heat capacity rate ratio (C_{min}/C_{max}) of 0.7717, the heat exchanger maximum operation factor is less than the maximum operation factor corresponding to a heat capacity rate ratio (C_{min}/C_{max}) of 0.3859 by 30.3%. The optimum heat exchanger size for the heat capacity rate ratio (C_{min}/C_{max}) of 0.5788 is less than the optimum size for the heat capacity rate ratio (C_{min}/C_{max}) of 0.3859 by 16.4%.

3.3 Investigation of Counter Flow Heat Exchangers

The same analysis outlined in the previous section will be performed for counter flow heat exchangers. The counter flow heat exchanger is more attractive than the parallel flow heat exchanger because the thermal effectiveness of a counter flow heat exchanger is usually higher than the thermal effectiveness of a parallel flow heat exchanger of the same size. In this section, the effect of viscous friction with constant viscosity and variable viscosity on the performance of the counter flow heat exchanger shall be investigated and compared to the effect on the performance of a parallel flow heat exchanger.

3.3.1 Effect of the Tube Fluid Inlet Temperature

The thermal effectiveness-NTU curves for the counter flow heat exchanger corresponding to no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity cases can be presented as shown in Fig. 3.16.

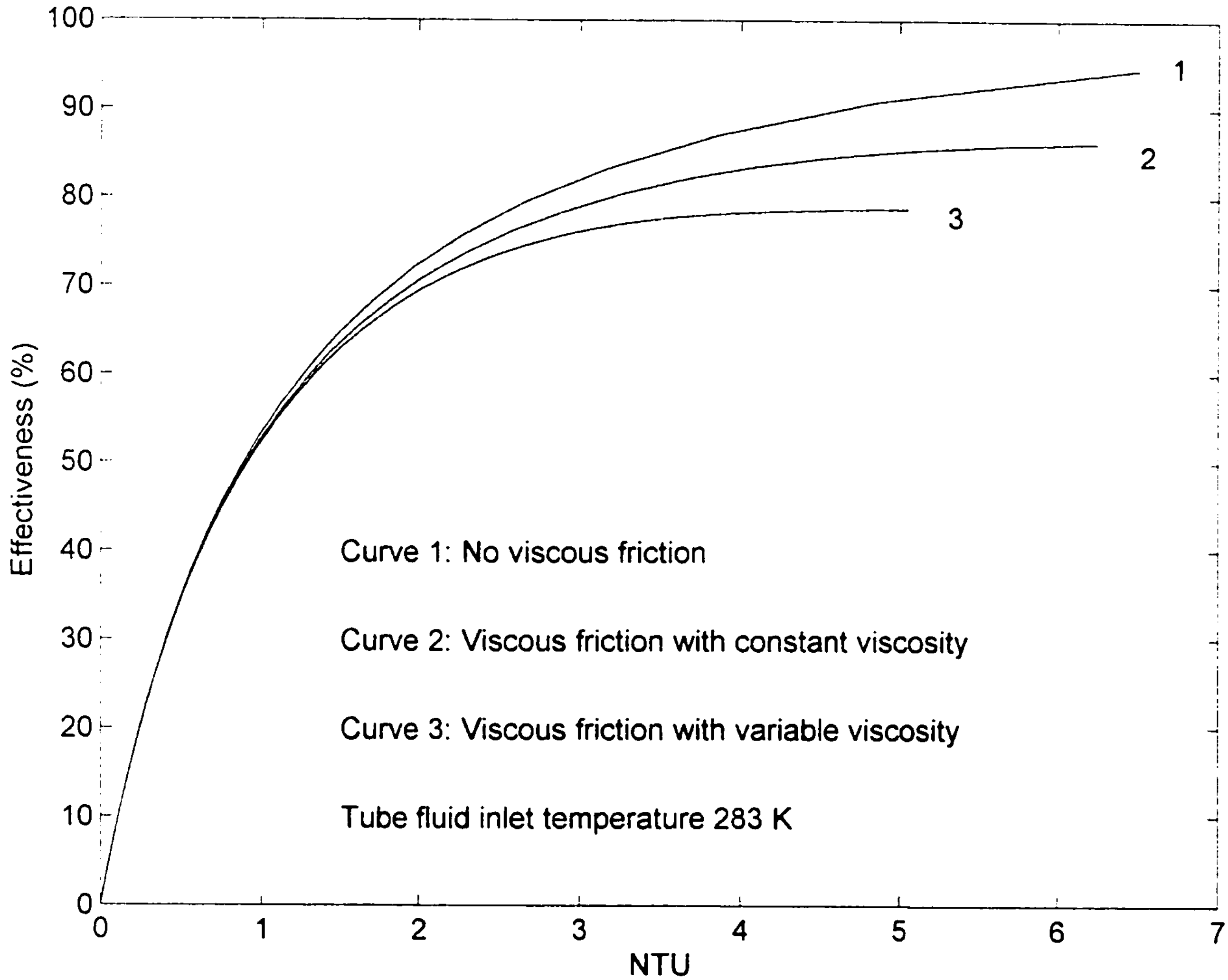


Fig 3.16 Thermal effectiveness-NTU curves of counter flow heat exchangers for the cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity (Tube fluid inlet temperature=283 K, Heat capacity rate ratio = 0.7717).

The tube fluid inlet temperature considered for these curves is 283 K. The thermal effectiveness-NTU behavior corresponding to the no viscous friction case is in excellent agreement with the behavior given by Kays and London (Kreith and Bohn, 1993). From the temperature profile of the tube and annulus fluids for the no viscous friction case shown in Fig. 3.17 (curve 1), it can be seen that the temperature difference between the two streams decreases as the heat exchanger size increases. This decrease in the temperature difference between the two fluids results in a slower rate of increase in the heat exchanger thermal effectiveness.

The thermal effectiveness of the counter flow heat exchanger for the viscous friction case with constant viscosity (curve 2) and variable viscosity (curve 3) behaves in a very similar way as the thermal effectiveness for no viscous case at small heat exchanger sizes as can be seen in the Fig. 3.16. This similar behavior is due to the small temperature drop of the high viscosity fluid. Moreover, the viscous frictional heating accumulation in small heat exchangers is not big enough to make a significant change in the temperature profile.

As the heat exchanger size increases, the effect of viscous frictional heating becomes more significant. This effect can be seen easily at NTU = 1.5 in Fig. 3.16 where a small deviation from the no viscous friction behavior starts. This deviation grows gradually for larger heat exchanger sizes. For an NTU value of 5.0, the thermal effectiveness for the no viscous case is 91%. However, the thermal effectiveness at the same NTU value for the viscous friction with constant viscosity case (curve 2) and the viscous friction with variable case (curve 3) are 85% and 79% respectively. This corresponds to a thermal effectiveness decrease of 6.6% and 13.2% of the no viscous friction effectiveness for the viscous friction with constant viscosity and the viscous friction with variable viscosity

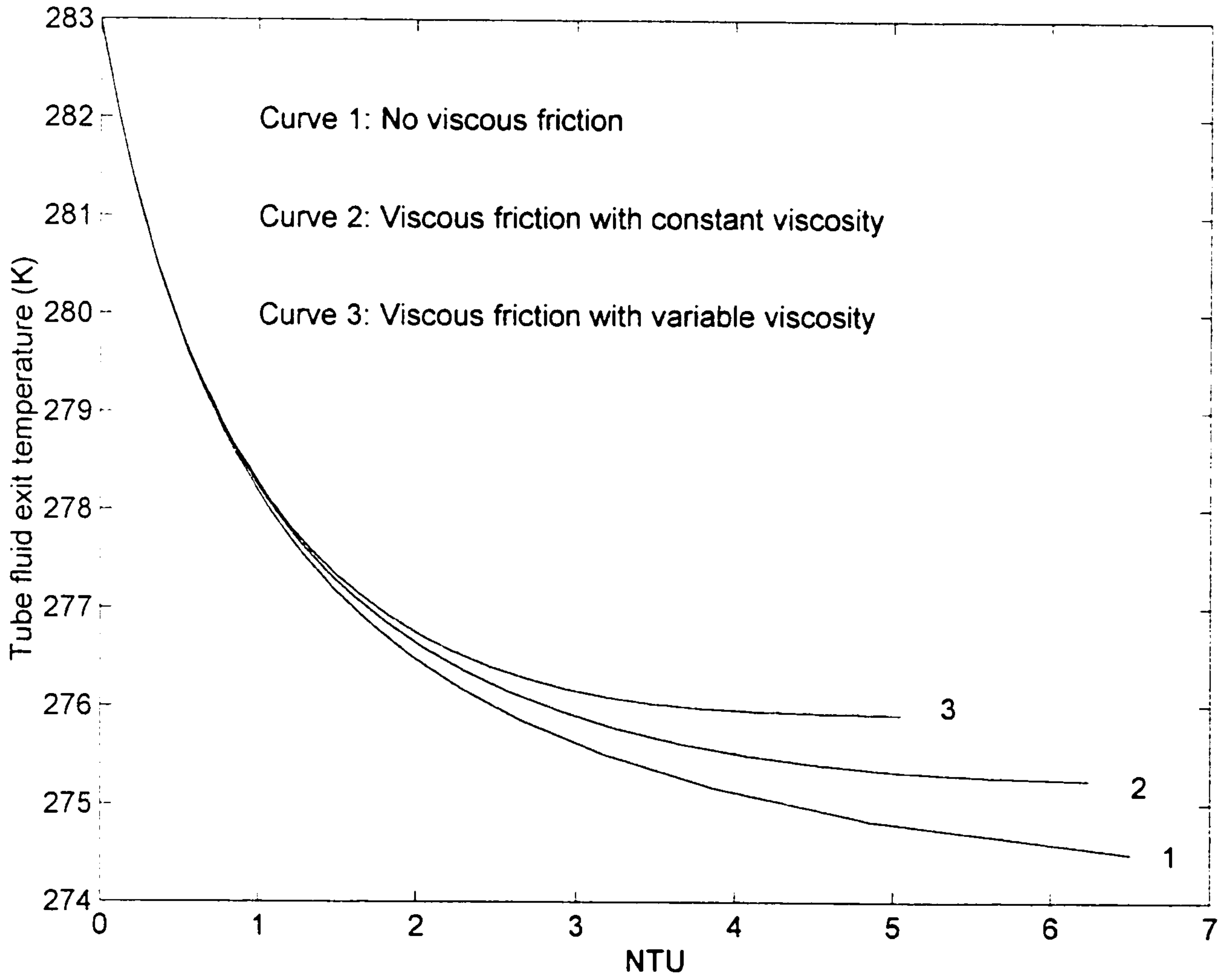


Fig. 3.17 Axial bulk tube fluid exit temperature profiles of counter flow heat exchangers (Annulus fluid inlet temperature = 274 K, Tube fluid inlet temperature = 283 K, Heat capacity rate ratio = 0.7717).

respectively. This reduction in the counter flow heat exchanger thermal effectiveness agrees with the analytical results obtained by Şahin (1998a).

The axial bulk exit temperature profile for each stream is shown in Fig. 3.17 for the tube side and in Fig. 3.18 for the annulus side for the three cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity. Comparing these temperature profiles with the temperature profiles for the no viscous friction case, it can be seen that the viscous frictional heating tends to reduce the tube fluid temperature drop and to increase the annulus fluid temperature rise.

In the case of no viscous friction (Fig. 3.17 curve 1), the temperature difference between the two running fluids at the tube fluid exit reduces as the heat exchanger size increases. However, the temperature difference between the two fluids for the case of viscous friction with constant viscosity (Fig. 3.17 curve 2) exhibits a different behavior. As the tube fluid travels axially, the viscous frictional heating lowers the rate at which the temperature of the tube fluid decreases. This lowering action prevents the temperature profiles of the two fluids from approaching each other as in the case of no viscous friction and results in larger temperature difference between them at the tube fluid exit side. This larger temperature difference causes the annulus fluid to gain more heat transfer and consequently increases the rate at which the annulus fluid temperature rises. This temperature profile behavior initiates the deviation of the thermal effectiveness-NTU curve of the viscous friction with constant viscosity case from the curve of the no viscous friction case.

A similar temperature profile behavior can be seen in Fig. 3.17 (curve 3) for the viscous friction with variable viscosity case. However, the viscous frictional heating has a

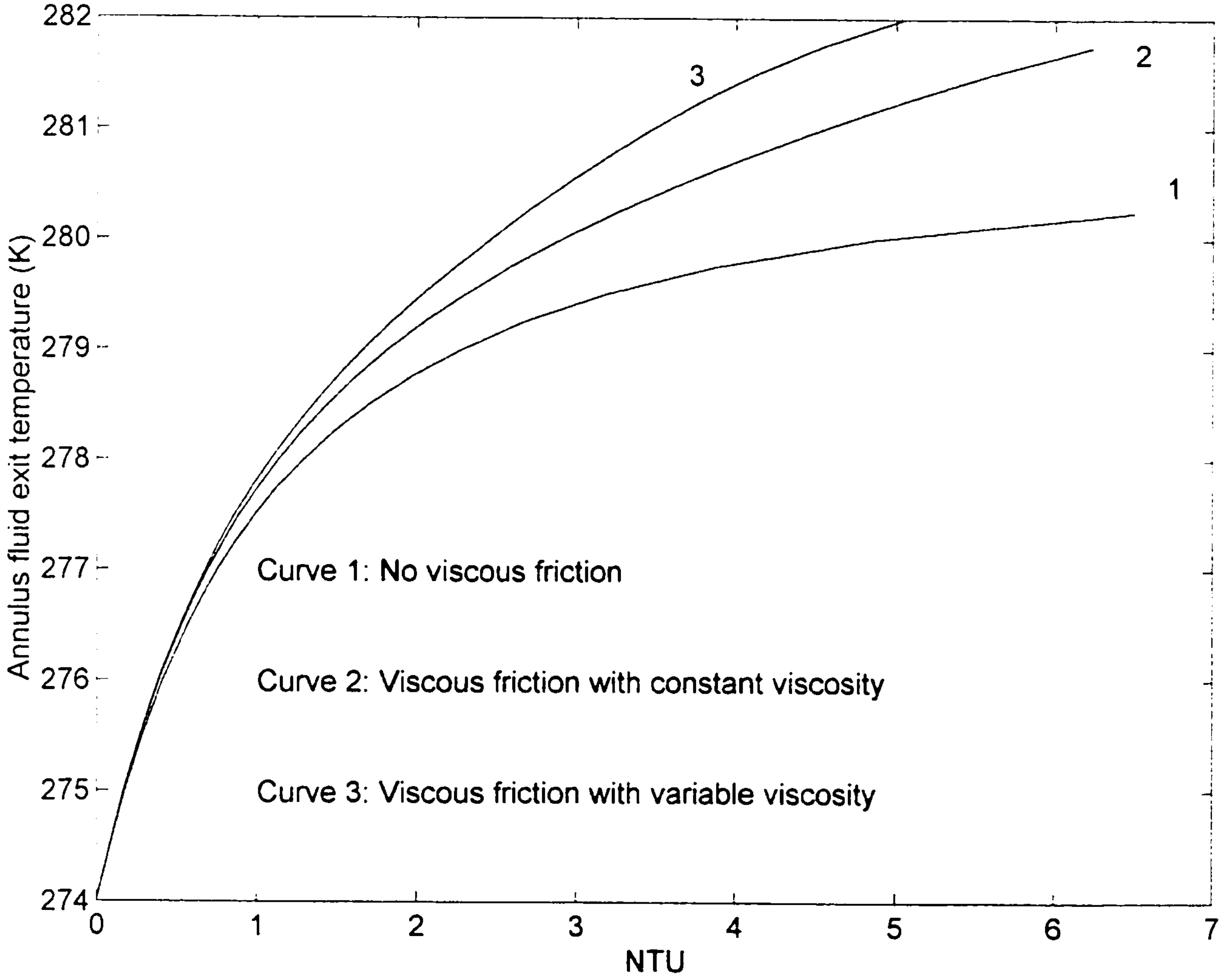


Fig. 3.18 Axial bulk annulus fluid exit temperature profiles of counter flow heat exchangers (Annulus fluid inlet temperature= 274 K, Tube fluid inlet temperature= 283 K, Heat capacity rate ratio = 0.7717).

stronger effect in the case of variable viscosity because the tube fluid viscosity increases as its temperature decreases. This causes a larger temperature difference between the two fluids at the tube fluid exit. This different behavior causes the thermal effectiveness-NTU curve of the variable viscosity case to deviates more from the thermal effectiveness-NTU curve of the no viscous friction case as discussed earlier. This shows that the viscous friction can limit the range of practical heat exchanger sizes because it makes the increase of the heat exchanger thermal effectiveness values smaller for larger heat exchanger sizes.

The amount of the deviation of the thermal effectiveness-NTU curve of the viscous friction case from the curve of the no viscous friction case depends also on the tube fluid inlet temperature. In order to study the effect of the tube fluid inlet temperature, the thermal effectiveness behavior has been investigated for two more different tube fluid inlet temperatures. The thermal effectiveness-NTU curves, pertaining to the cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity, are shown in Fig. 3.19 for the tube fluid inlet temperature of 280 K and in Fig. 3.20 for the tube fluid inlet temperature of 277 K.

From these figures, it can be observed that the deviation from the no viscous friction behavior increases as the tube fluid inlet temperature decreases. This is due to the higher tube fluid viscosity at the lower temperatures, which causes higher viscous frictional heating. A Comparison between the results of the counter flow heat exchanger and the results of the parallel flow heat exchanger shows that the viscous friction effect on the performance of parallel flow heat exchangers is more significant compared to counter flow heat exchangers. For the same operating conditions, the behavior of the parallel flow heat exchanger starts deviating from the no viscous friction behavior at an NTU value of

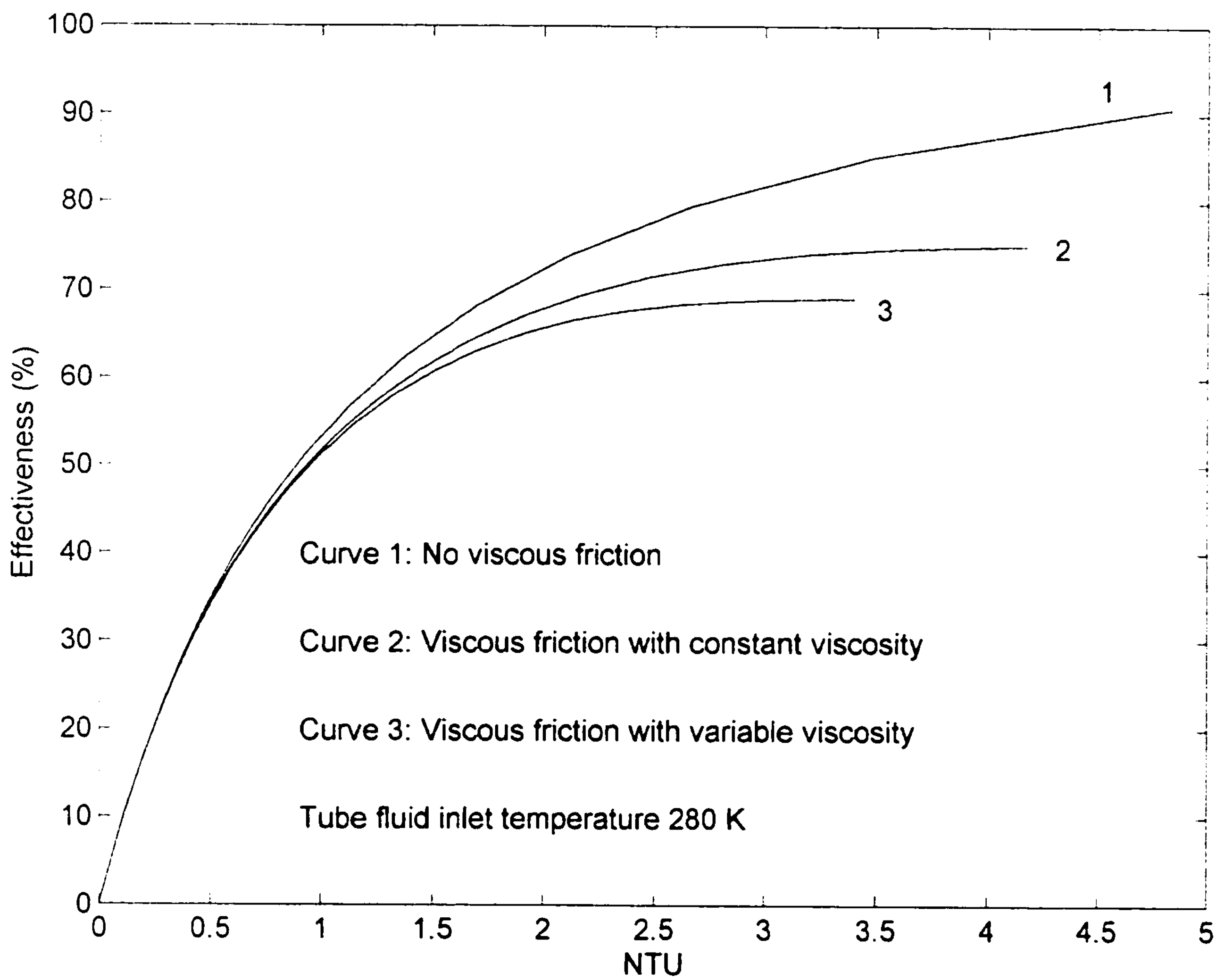


Fig 3.19 Thermal effectiveness-NTU curves of counter flow heat exchangers for the cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity (Tube fluid inlet temperature=280 K, Heat capacity rate ratio = 0.7717).

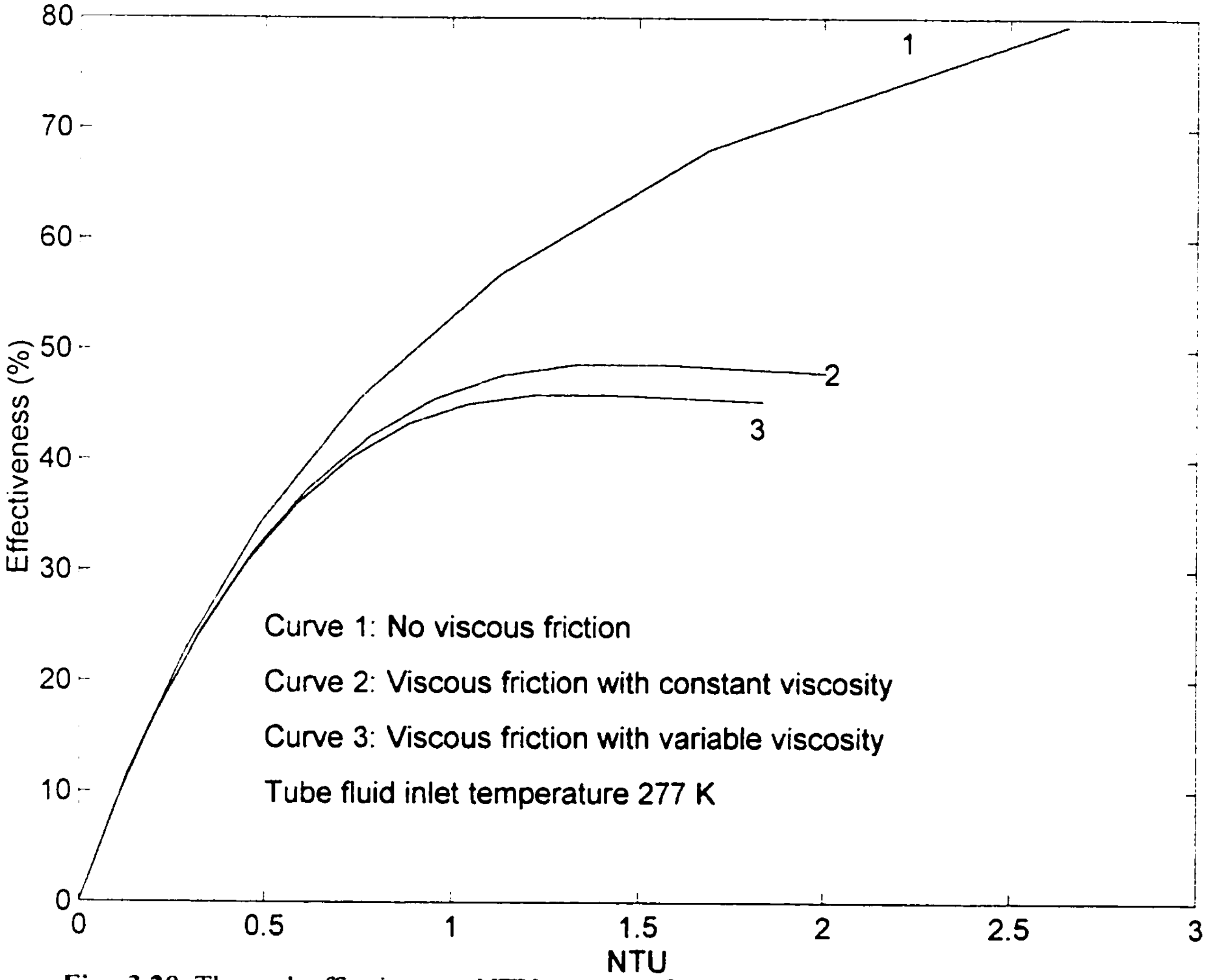


Fig. 3.20 Thermal effectiveness-NTU curves of counter flow heat exchangers for the cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity (Tube fluid inlet temperature=277 K, Heat capacity rate ratio = 0.7717).

approximately 0.5. On the other hand, the performance of the counter flow heat exchanger starts deviating from the no viscous friction behavior at an NTU value of approximately 1.5. At an NTU value of 1.75, the thermal effectiveness of the parallel flow heat exchanger, for the variable viscosity case, is less than the thermal effectiveness for the no viscous friction case by 9.8%. However, for the same NTU value, the thermal effectiveness of the counter flow heat exchanger, for the variable viscosity case, is less than the thermal effectiveness for the no viscous friction case by 2.5% only.

The local entropy generation for each stream in the case of no viscous friction is shown in Fig. 3.21 for four different values of the counter flow heat exchanger size expressed in NTU. This figure indicates that for all sizes of the counter flow heat exchanger, the absolute values of the local entropy generation of each stream decrease in the axial direction. This is due to the smaller temperature difference between the two fluids away from the tube fluid inlet side as can be understood from the temperature profiles shown in Fig. 3.17.

Figure 3.21 shows also that the absolute values of the local entropy generation of each stream, for the counter flow heat exchanger, at the tube fluid inlet side depend on the size of the heat exchanger. As the heat exchanger size increases, the absolute values of the local entropy generation at the tube fluid inlet side ($x = 0$) decreases. This behavior is due to the fact that the temperature difference between the two fluids at the tube fluid inlet side increases as the heat exchanger size increases. Consequently, the heat transfer rate between the two fluids at the tube fluid inlet side decreases as the heat exchanger size increases because of the larger temperature difference.

However, this is not the case for the parallel heat exchanger in which the absolute

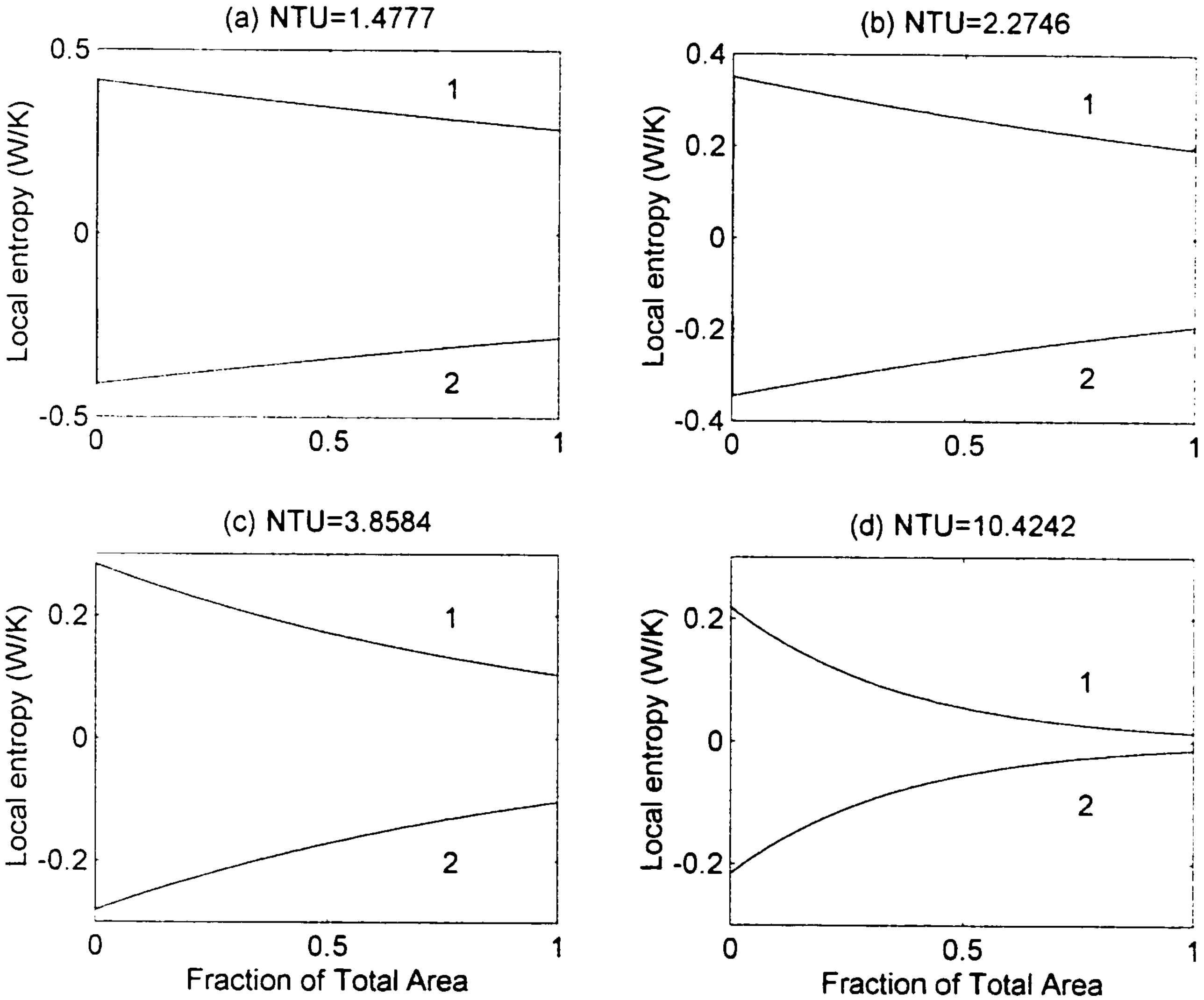


Fig. 3.21 Local entropy generation of counter flow heat exchangers for the case of no viscous friction for the tube side (curve 2) and annulus side (curve 1) at different NTU values (Tube fluid inlet temperature=283 K, Heat capacity rate ratio = 0.7717).

values of the local entropy generation for each stream at the tube fluid inlet side remain the same regardless of the size of the heat exchanger. This is due to the fact that the temperature of each stream is fixed at the inlet side. This difference between the counter flow heat exchanger and the parallel flow heat exchanger results also in a different behavior of the total entropy generation number versus NTU curve.

The total entropy generation number corresponding to the no viscous friction case for the counter flow heat exchanger is shown in Fig. 3.22 for different heat exchanger sizes expressed in NTU. The behavior shown in this figure is in good agreement with the analytically obtained behavior by Bejan for gas-to-gas counter flow heat exchangers (Bejan, 1977). As shown in this figure, the total entropy generation number of the counter flow heat exchanger increases with increasing NTU until it reaches a maximum value and then starts decreasing. However, the total entropy generation number of the parallel flow heat exchanger keeps increasing with increasing size as shown in Fig. 3.7 (a).

The effect of viscous frictional heating on the total entropy generation number of the counter flow heat exchanger is shown in Fig. 3.23. For the variable viscosity case, the effect of viscous frictional heating on the total entropy generation number of the heat exchanger is larger than the effect of viscous frictional heating for the constant viscosity. This due to the fact that the viscosity of the tube fluid increases as the temperature of the tube fluid decreases in the axial direction in the variable viscosity case. In order to investigate the effect of different tube fluid inlet temperatures on the analysis, the total entropy generation number of the heat exchanger, for the viscous friction with variable viscosity case, is shown in Fig. 3.24 for tube fluid inlet temperatures of 283 K, 280 K and

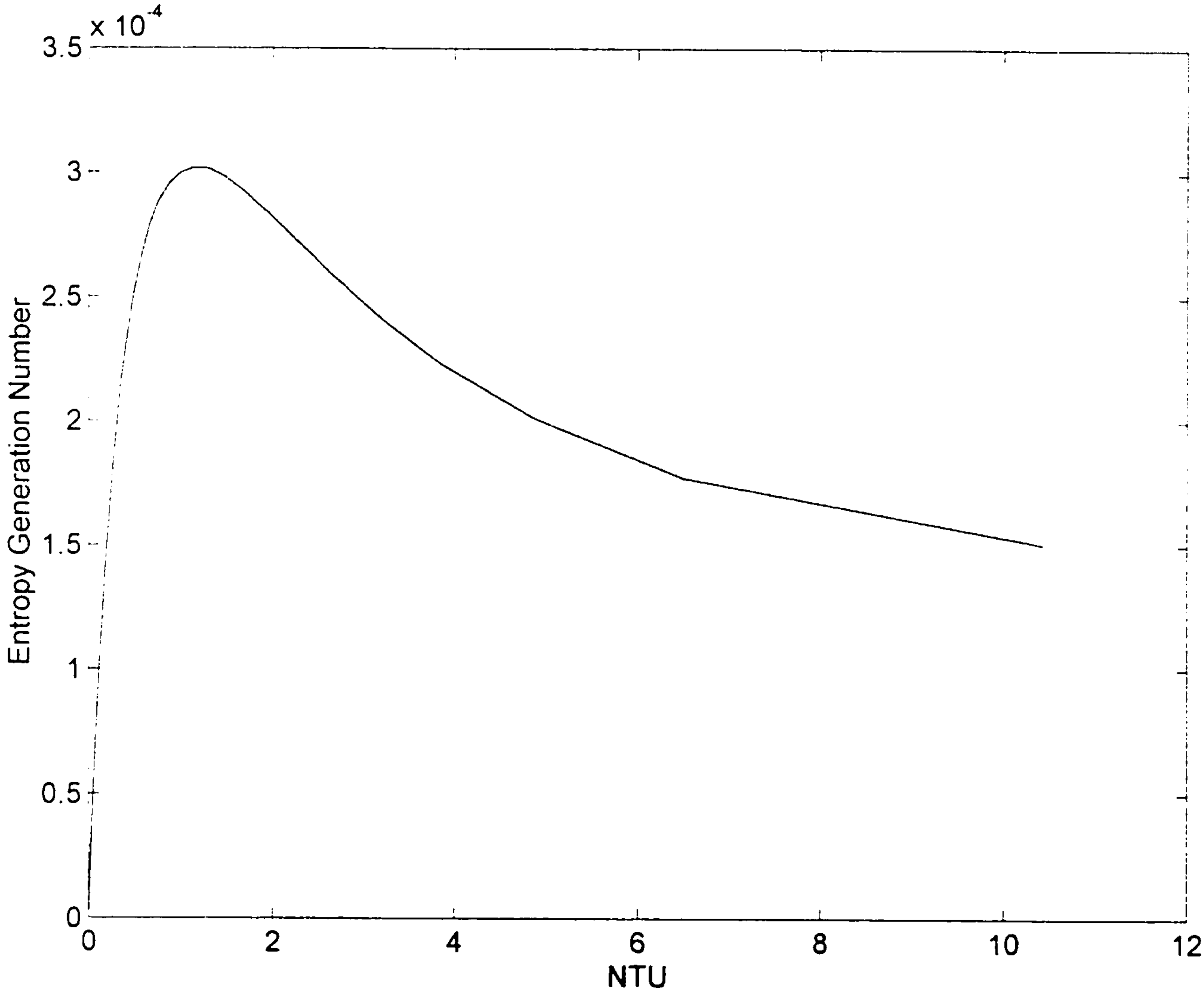


Fig. 3.22 Total entropy generation number of counter flow heat exchangers for the case of no viscous friction (Tube fluid inlet temperature=283 K, Heat capacity rate ratio = 0.7717).

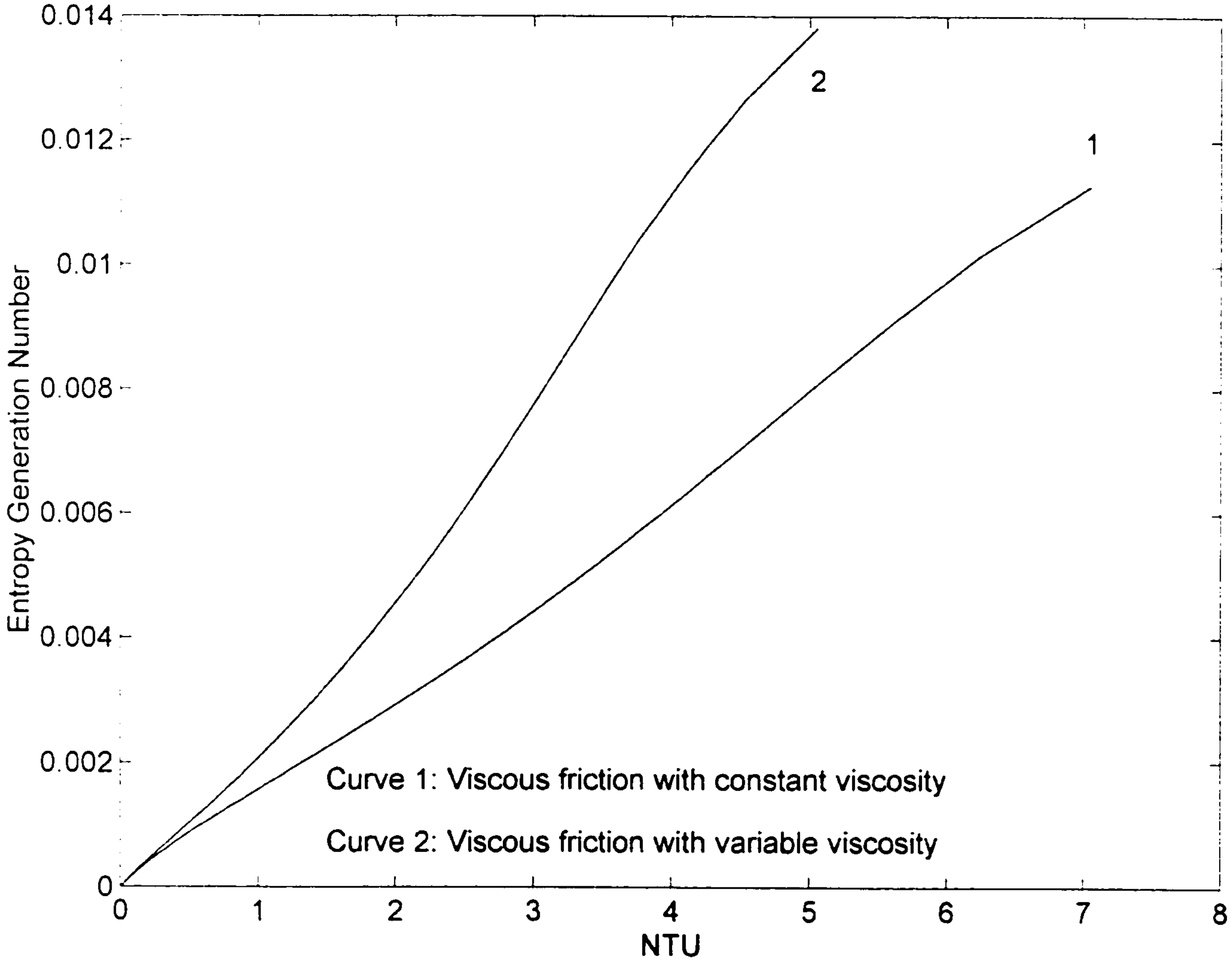


Fig. 3.23 Total entropy generation number of counter flow heat exchangers for the cases of viscous friction with constant viscosity and variable viscosity (Tube fluid inlet temperature = 283 K, Heat capacity rate ratio = 0.7717).

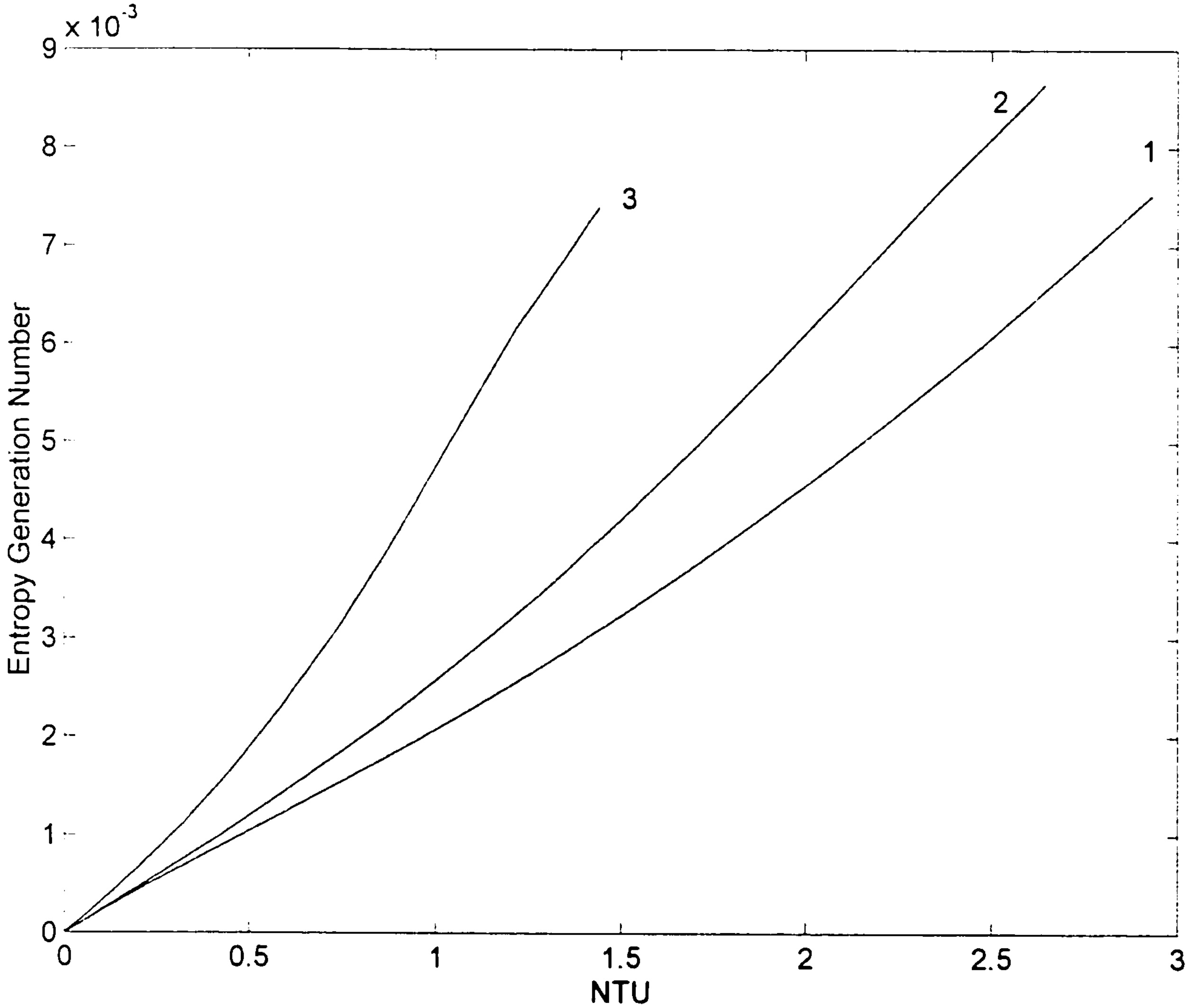


Fig. 3.24 Total entropy generation number of counter flow heat exchangers for the case of viscous friction with variable viscosity for tube fluid inlet temperature of 283 K (curve 1), 280 K (curve 2) and 277 K (curve 3) (Heat capacity rate ratio = 0.7717).

277 K. From this figure, it is clear that the viscous frictional heating effect on the total entropy generation number increases as the tube fluid inlet temperature decreases.

The counter flow heat exchanger cost effectiveness and operation factor, for the tube fluid inlet temperature of 283 K, is shown in Fig. 3.25 for the no viscous friction case. It can be seen from Fig. 3.25 that there is no optimum size of the heat exchanger at which the operation factor reaches a maximum value because both the cost effectiveness curve and the thermal effectiveness curve increases as the heat exchanger size increases. However, the total entropy generation number behavior (Fig. 3.22) suggests that the NTU value should be greater than 2.0 in order to have a smaller total entropy generation.

In order to include the viscous frictional heating effect in the analysis, the counter flow heat exchanger cost effectiveness and operation factor for the case of viscous friction with variable viscosity have been plotted as shown in Fig. 3.26. From this figure, it can be seen that the heat exchanger reaches its maximum operation factor at an NTU value of approximately 2.0. This result is very close to the result obtained from the total entropy generation behavior alone in the case of no viscous friction. The thermal effectiveness of the heat exchanger at this NTU value is approximately 73%. For the same tube fluid inlet temperature (283 K), the optimum size obtained, for the case of the viscous friction with variable viscosity, is at an NTU value of approximately 2.0 with a thermal effectiveness value of approximately 70%. This shows that the optimum size of the heat exchanger, at which the operation factor reaches its maximum value, falls in the range where the deviation from the no viscous friction behavior is very small. No optimum parallel flow heat exchanger size could be found for the no viscous friction case because the total entropy generation keeps increasing with increasing NTU. A comparison of the results for

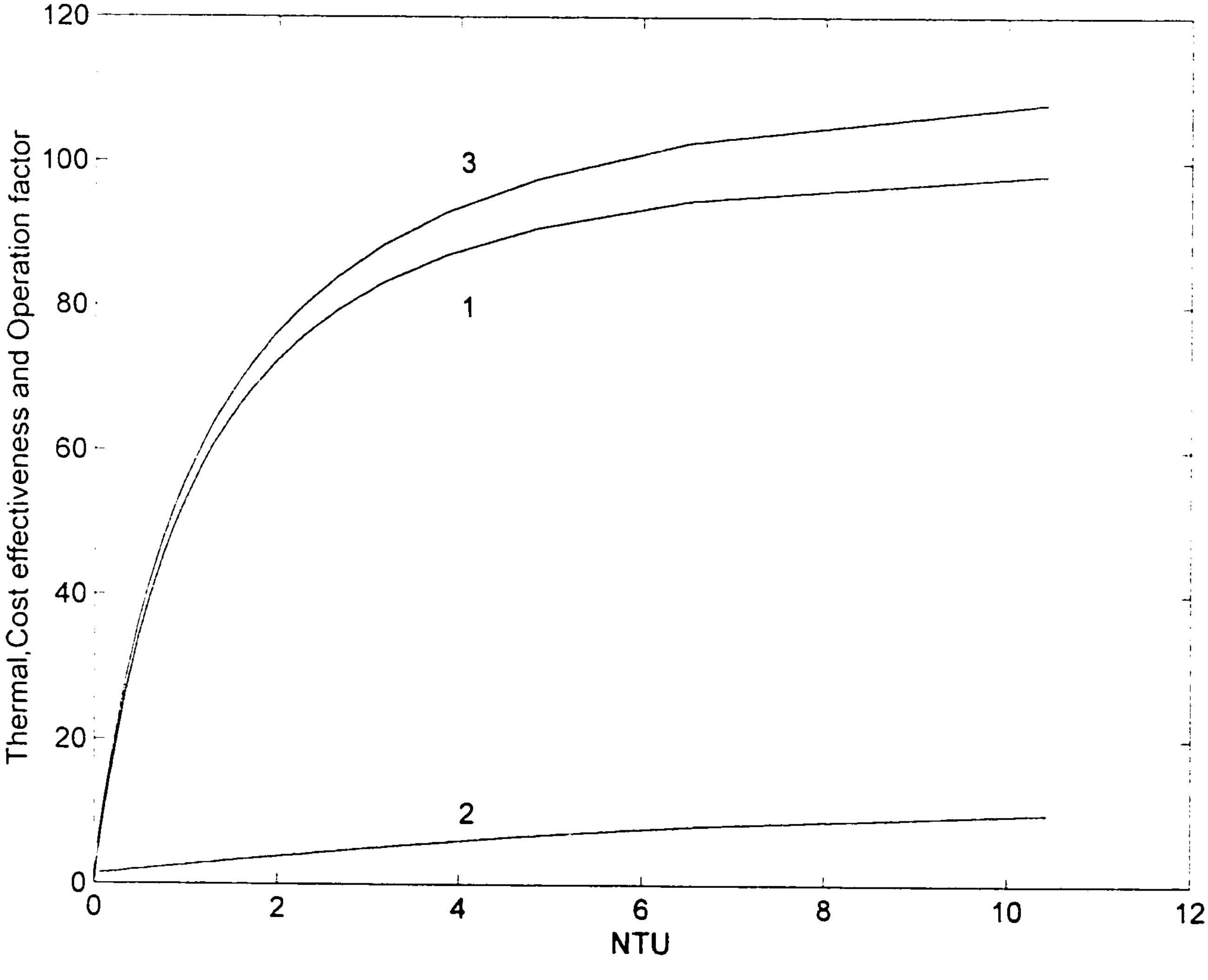


Fig. 3.25 Thermal effectiveness (curve 1), cost effectiveness (curve 2) and operation factor (curve 3) of counter flow heat exchangers for the case of no viscous friction (Tube fluid inlet temperature =283 K, Heat capacity rate ratio = 0.7717).

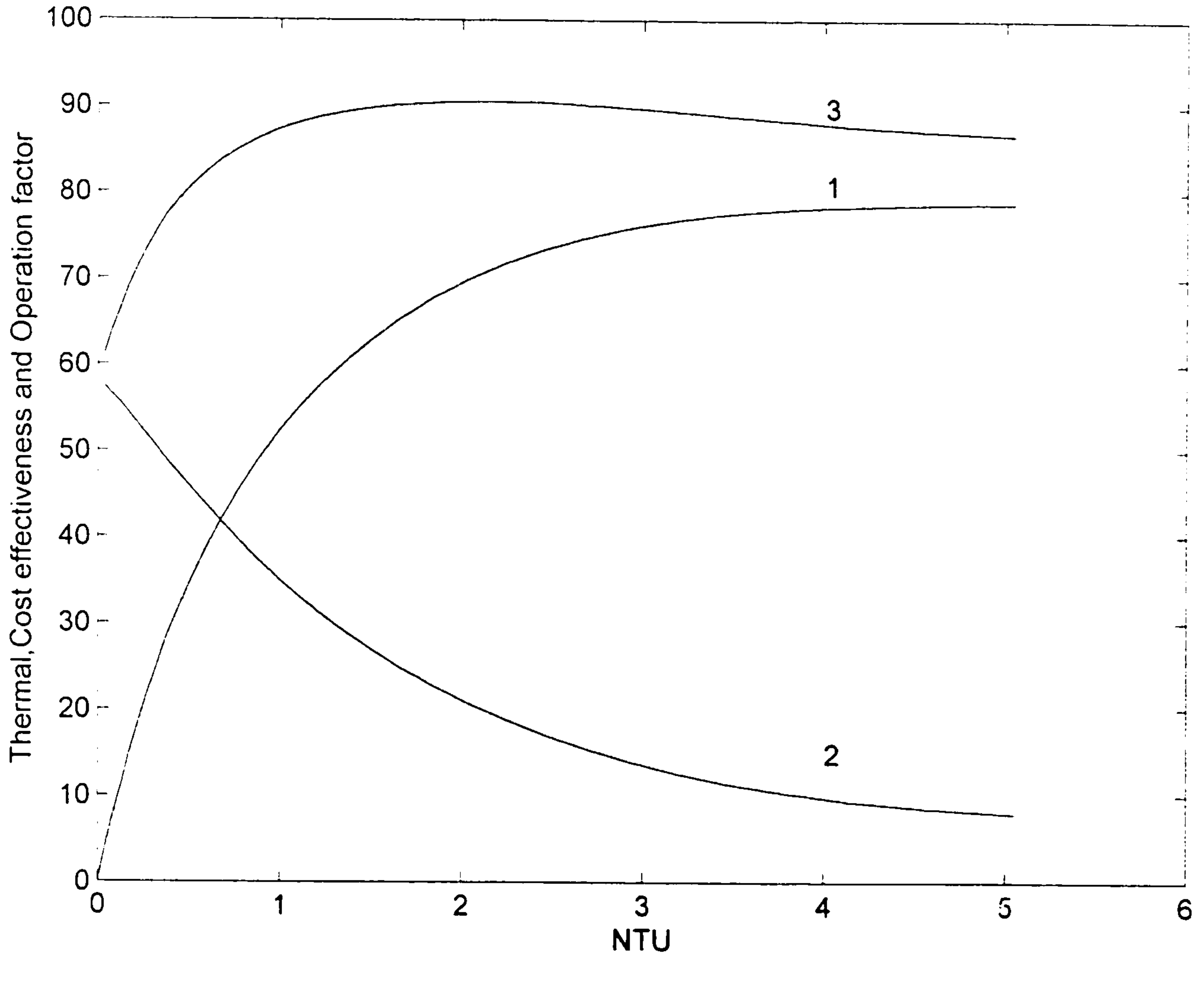


Fig 3.26 Thermal effectiveness (curve 1), cost effectiveness (curve 2) and operation factor (curve 3) of counter flow heat exchangers for the case of viscous friction with variable viscosity (Tube fluid inlet temperature = 283 K, Heat capacity rate ratio = 0.7717).

the counter flow heat exchanger and the parallel flow heat exchanger indicates that the use of counter flow heat exchangers provide higher thermal effectiveness at the optimum size of the heat exchanger, which gives the maximum operation factor.

The counter flow heat exchanger cost effectiveness and operation factor curves for the viscous friction with variable viscosity case is shown in Fig. 3.27 for 280 K and 277 K tube fluid inlet temperatures. From this figure, it can be seen that the optimum size of the heat exchanger decreases as the tube fluid inlet temperature decreases. Therefore, the thermal effectiveness of the heat exchanger at its optimum operation size decreases as the tube fluid inlet temperature decreases.

3.3.2 Effect of the Heat Capacity Rate Ratio (C_{min}/C_{max})

The amount of deviation of the heat exchanger performance, for the cases of viscous friction with constant viscosity and viscous friction with variable viscosity, from the no viscous friction case behavior depends also on the heat capacity rate ratio (C_{min}/C_{max}). In order to investigate the effect of different heat capacity rate ratios (C_{min}/C_{max}) on the analysis of counter flow heat exchangers, the same heat capacity rate ratios (C_{min}/C_{max}) considered in section 3.2.2 for parallel flow heat exchangers will be used with the tube fluid inlet temperature taken as 283 K. These heat capacity rate ratios (C_{min}/C_{max}) are 0.7717, 0.5788 and 0.3859. The effect on the heat exchanger performance measures discussed in the previous sections will be observed.

The thermal effectiveness-NTU curves for the heat capacity rate ratio (C_{min}/C_{max}) of 0.7717 for the cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity are shown in Fig. 3.16 discussed previously.

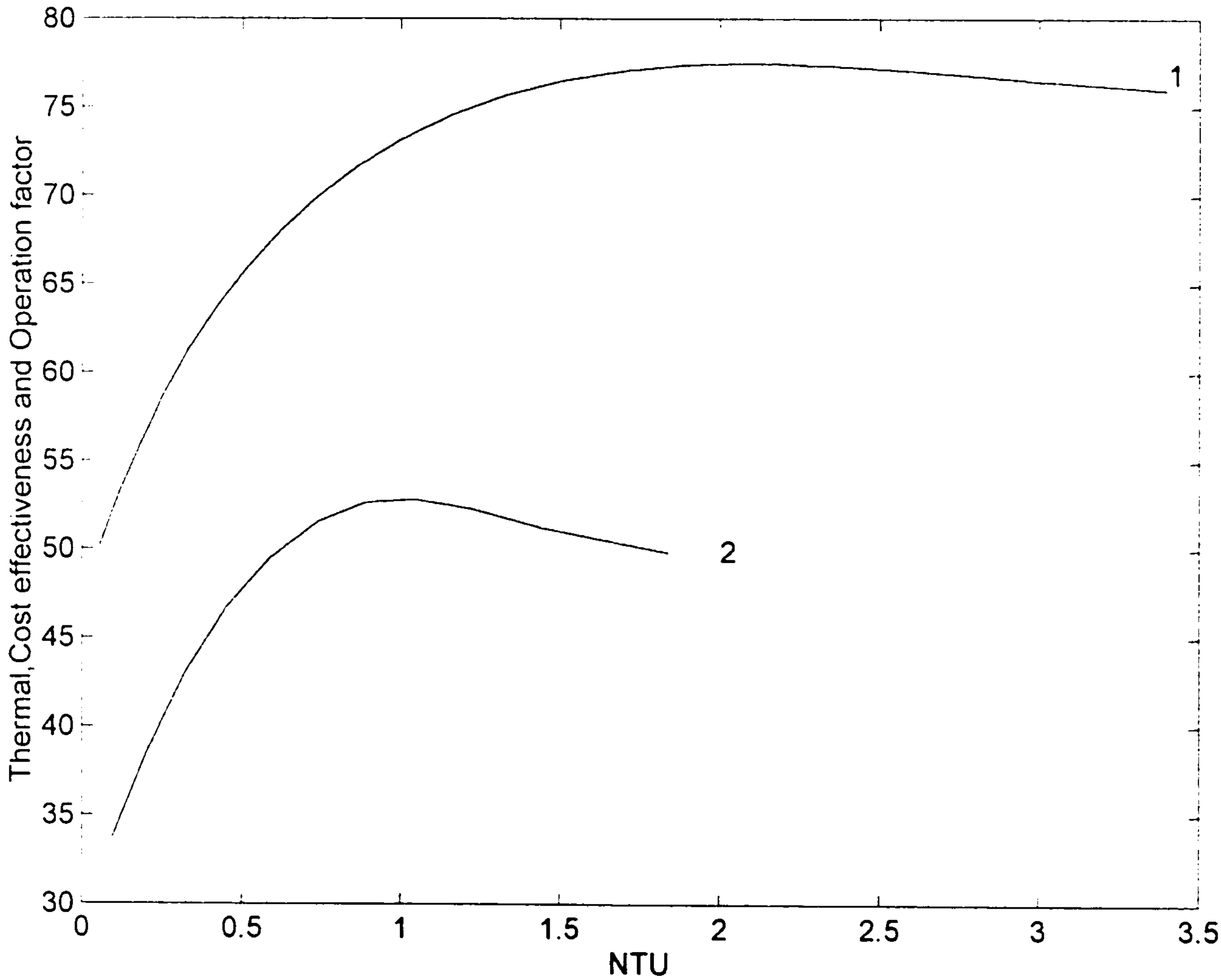


Fig. 3.27 Operation factor of counter flow heat exchangers for the case of viscous friction with variable viscosity for tube fluid inlet temperature of 280 K (curve 1) and 277 K (curve 2) (Heat capacity rate ratio = 0.7717).

The other two sets of curves for the heat capacity rate ratios (C_{min}/C_{max}) of 0.5788 and 0.3859 are shown in Fig. 3.28 and Fig. 3.29 respectively. A comparison of the three sets of curves shown in these figures indicates that the performance of the counter flow heat exchanger, for the viscous friction cases, deviates more from the performance for the no viscous friction as the heat capacity rate ratio (C_{min}/C_{max}) increases.

For example, at an NTU value of 5.0, the thermal effectiveness corresponding to the cases of no viscous friction and viscous friction with variable viscosity are 91% and 79% respectively for the heat capacity rate ratio (C_{min}/C_{max}) of 0.7717. This is equivalent to a 13.2% reduction in the thermal effectiveness. On the other hand, at the same NTU value, the thermal effectiveness for the cases of no viscous friction and viscous friction with constant viscosity are 95% and 89% respectively for the heat capacity rate ratio (C_{min}/C_{max}) of 0.5788. This corresponds to a thermal effectiveness percentage reduction of 6.3% only. For the heat capacity rate ratio (C_{min}/C_{max}) of 0.3859, the deviation of the viscous friction curves from the no viscous friction curve diminishes and the three curves overlap as shown in Fig. 3.29.

The same result was observed for the parallel flow heat exchanger. However, the counter flow heat exchanger exhibits a smaller tendency to deviates from the no viscous friction behavior than the parallel flow heat exchanger for the same heat capacity rate ratio (C_{min}/C_{max}). This is clear because the performance of the parallel flow heat exchanger continues to deviates from the no viscous friction performance at the heat capacity rate ratio (C_{min}/C_{max}) of 0.3859. However, the deviation of the counter flow heat exchanger performance from the no viscous friction performance

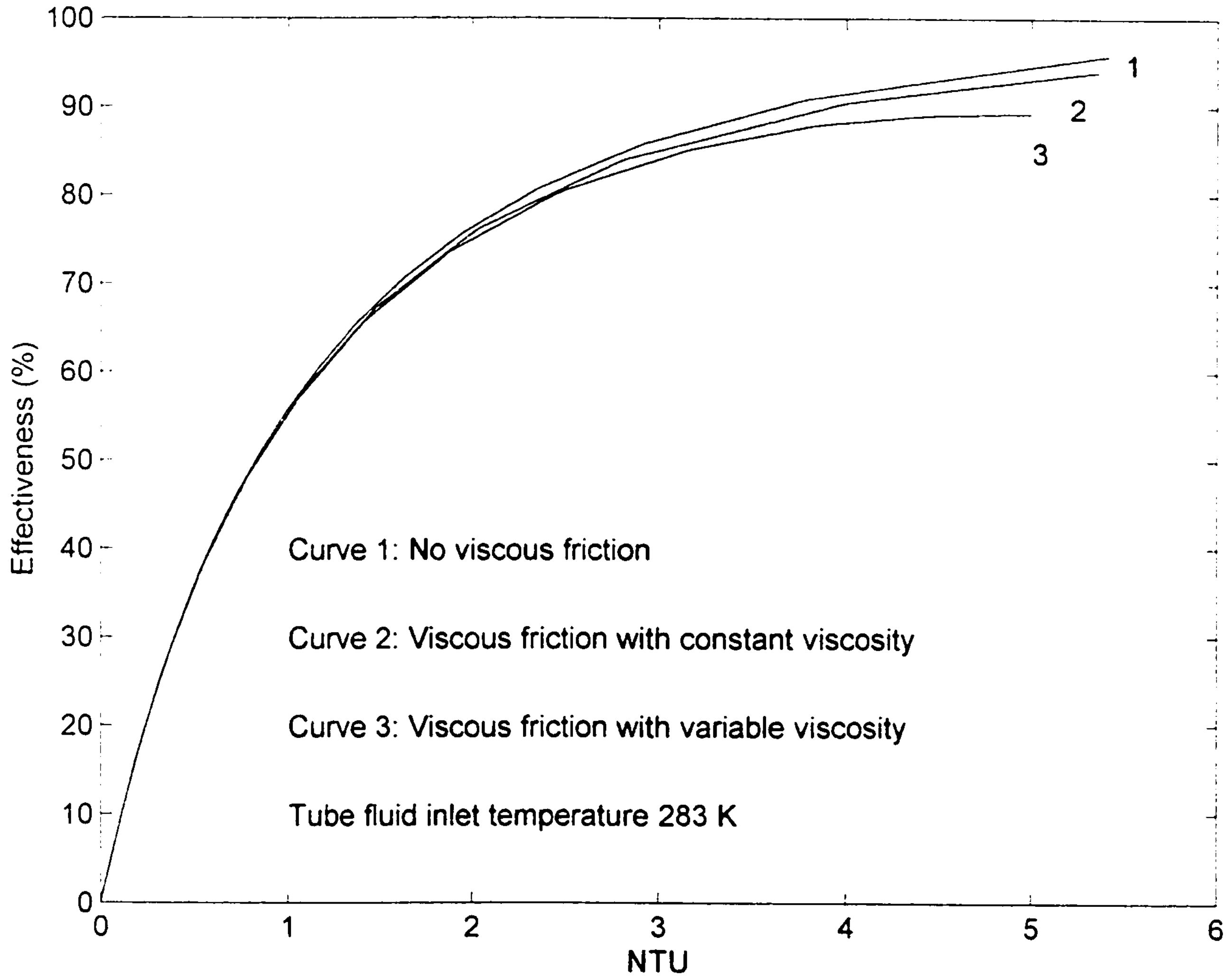


Fig 3.28 Thermal effectiveness-NTU curves of counter flow heat exchangers for the cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity (Tube fluid inlet temperature=283 K, Heat capacity rate ratio = 0.5788).

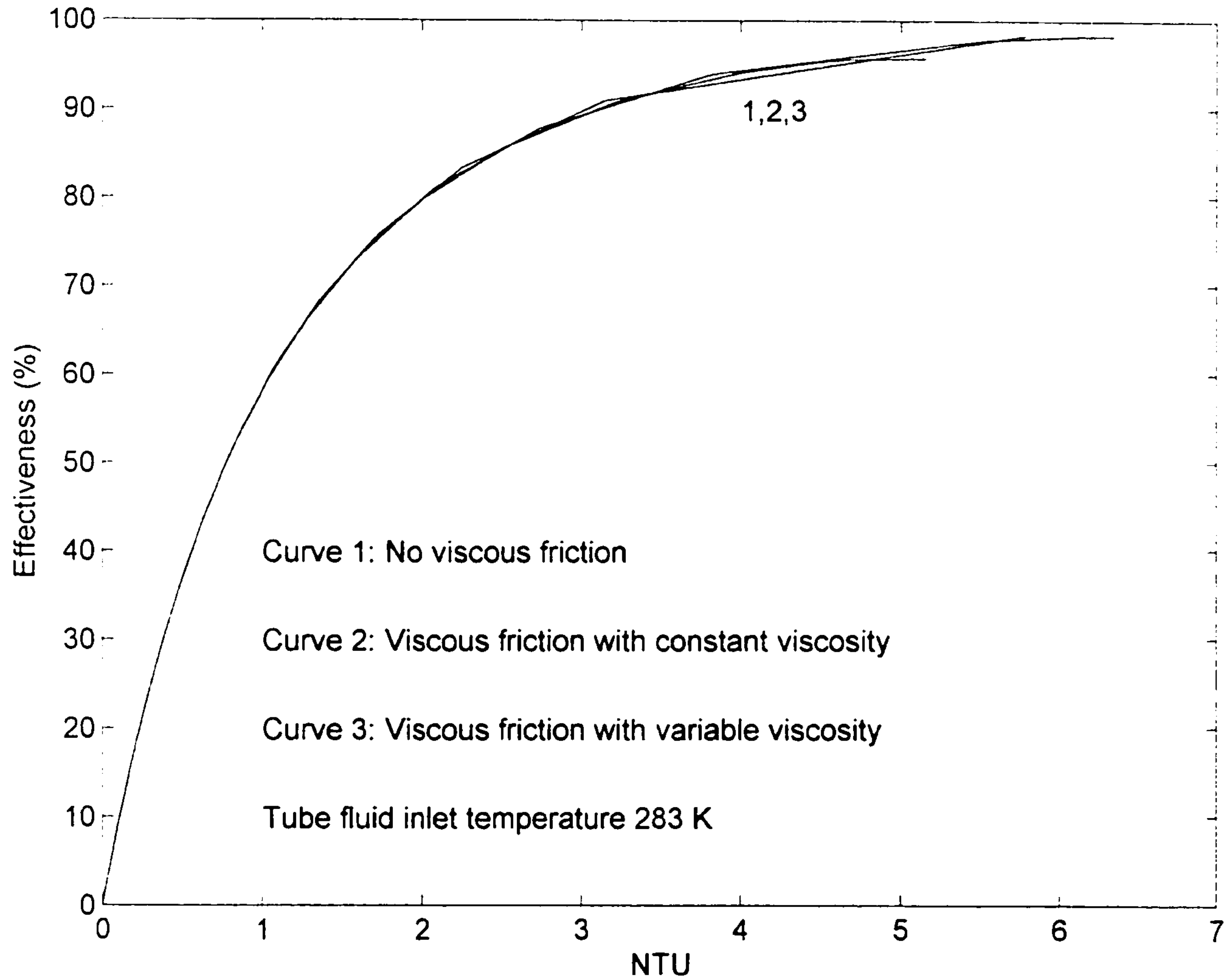


Fig 3.29 Thermal effectiveness-NTU curves of counter flow heat exchangers for the cases of no viscous friction, viscous friction with constant viscosity and viscous friction with variable viscosity (Tube fluid inlet temperature=283 K, Heat capacity rate ratio = 0.3859).

diminishes at the same the heat capacity rate ratio (C_{min}/C_{max}). This shows that the counter flow heat exchanger has more resistance against the viscous effects.

The total entropy generation number of the counter flow heat exchanger for the case of viscous friction with variable viscosity is shown in Fig. 3.30 for the three heat capacity rate ratio (C_{min}/C_{max}). From this figure, it can be seen that the total entropy generation number of the counter flow heat exchanger increases as the heat capacity rate ratio (C_{min}/C_{max}) increases. This increase is due to the higher viscous frictional heating in the tube fluid, which occurs as a result of the higher tube fluid mass flow rate associated with the larger heat capacity rate ratio (C_{min}/C_{max}).

The effect of higher frictional heating extends also to the heat exchanger cost effectiveness and operation factor. The counter flow heat exchanger cost effectiveness curves and the operation factor curves, for the viscous friction case with variable viscosity, are shown in Fig. 3.31 and Fig. 3.32 for the heat capacity rate ratios (C_{min}/C_{max}) of 0.5788 and 0.3859 respectively. The same curves for the heat capacity rate ratio (C_{min}/C_{max}) of 0.7717 is shown in Fig. 3.26 discussed earlier. From these figures, it can be noted that the maximum operation factor of the heat exchanger decreases as the heat capacity rate ratio (C_{min}/ C_{max}) increases. This is due to the higher viscous frictional heating associated with the higher tube fluid mass flow rate in the case of higher heat capacity rate ratios (C_{min}/ C_{max}). This higher viscous frictional heating causes higher entropy generation and consequently lower cost effectiveness.

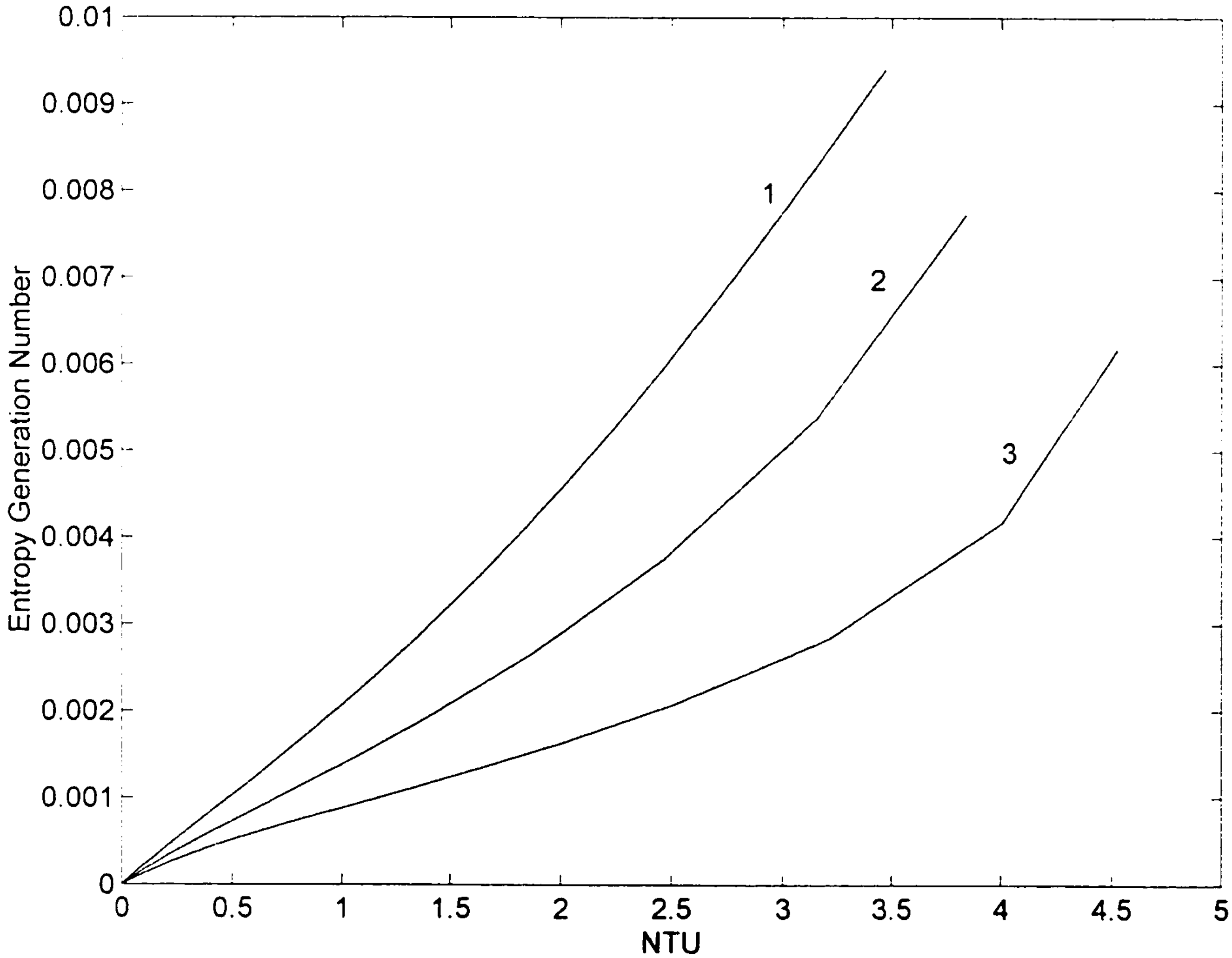


Fig. 3.30 Total entropy generation number of counter flow heat exchangers for the case of viscous friction with variable viscosity for heat capacity rate ratio = 0.7717 (curve 1), 0.5788 (curve 2) and 0.3859 (curve 3) (Tube inlet temperature = 283 K).

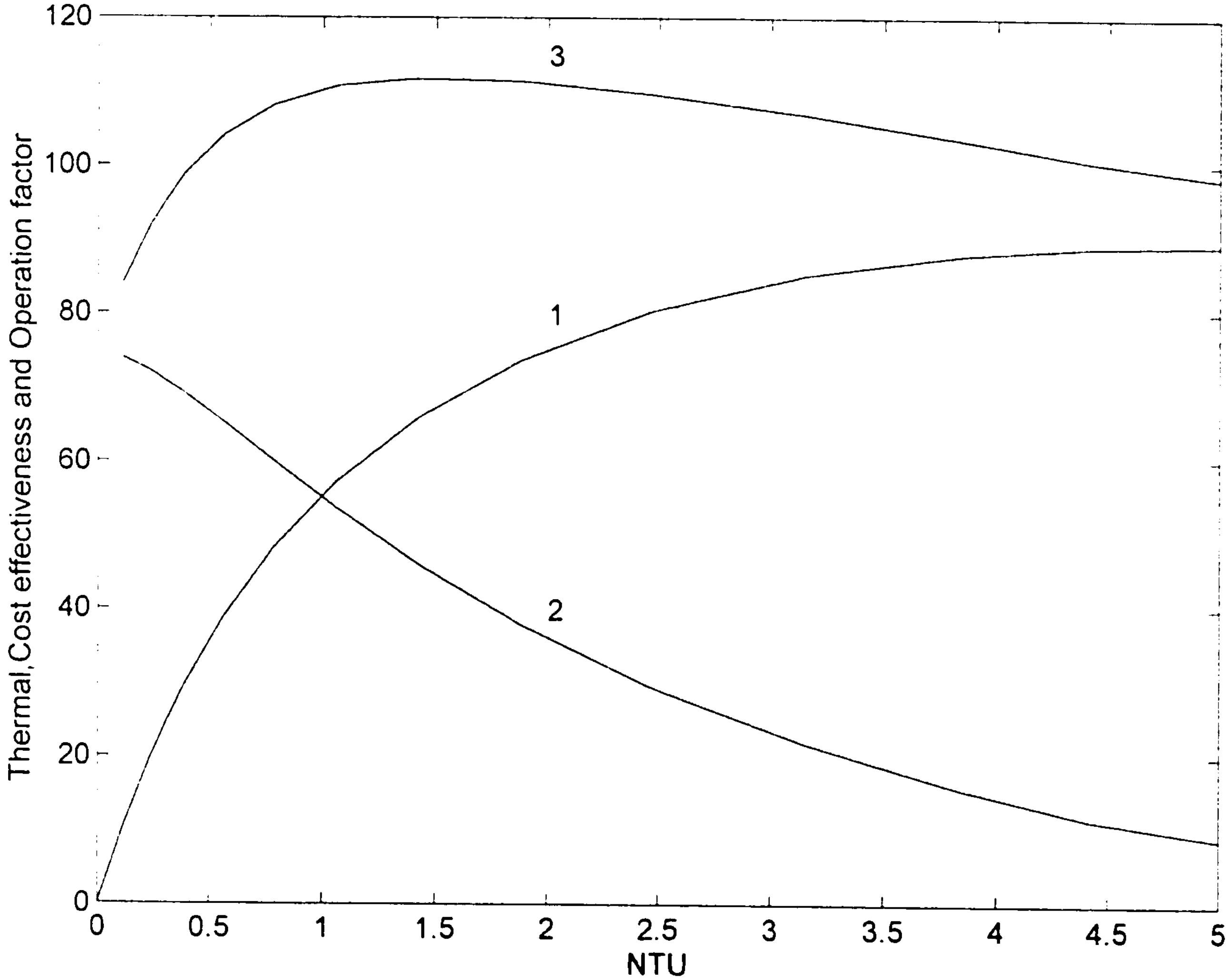


Fig 3.31 Thermal effectiveness (curve 1), cost effectiveness (curve 2) and operation factor (curve 3) of counter flow heat exchangers for the case of viscous friction with variable viscosity (Tube fluid inlet temperature = 283 K, Heat capacity rate ratio = 0.5788).

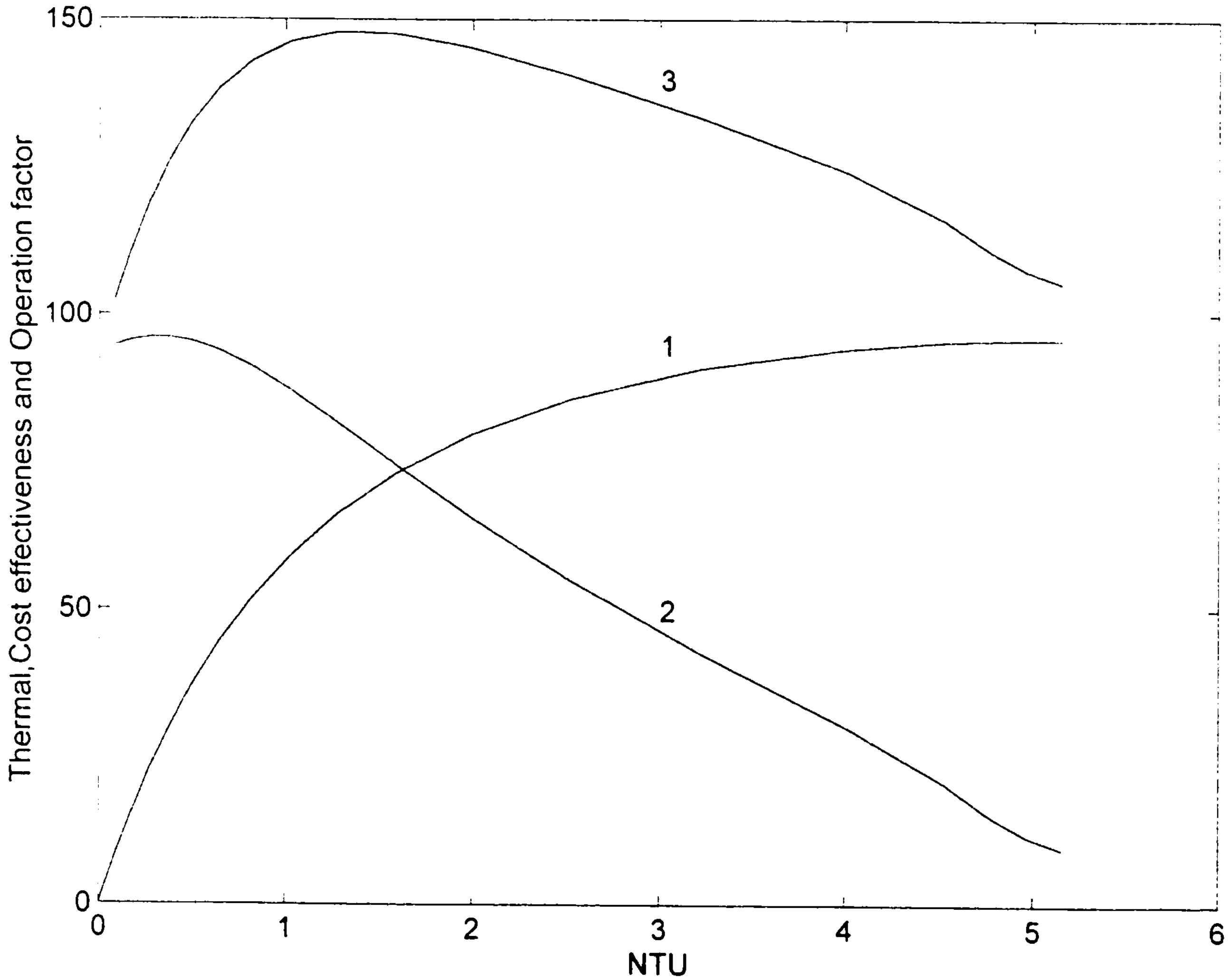


Fig 3.32 Thermal effectiveness (curve 1), cost effectiveness (curve 2) and operation factor (curve 3) of counter flow heat exchangers for the case of viscous friction with variable viscosity (Tube fluid inlet temperature = 283 K, Heat capacity rate ratio = 0.3859).

3.4 Conclusions

From the results and discussion given in this chapter, the following conclusions and recommendations can be derived about the effect of viscosity on the performance of parallel and counter flow double pipe heat exchangers:

1- When the effect of viscosity is introduced in the analysis, the thermal effectiveness-NTU of the parallel flow heat exchanger tends to deviate from the no viscous friction behavior given in the literature by Kays and London. The extent of this deviation depends on the viscosity values of the running fluids at the operating temperature range. As the inlet temperature of the high viscosity fluid decreases, the amount of deviation from the no viscous behavior increases. Thus, the effect of viscosity on the performance of the heat exchanger becomes more significant at low inlet temperatures.

2- It was found also that the consideration of viscous friction only without the variation of viscosity with temperature leads to smaller deviations from the no viscous friction case. Therefore, the exact evaluation of the heat exchanger performance requires the consideration of viscosity variations with temperature.

3- The thermal effectiveness of the parallel flow heat exchanger deviates gradually from the no viscous friction behavior as the NTU value increases until it reaches a maximum value after which the thermal effectiveness starts decreasing rather than increasing as the heat exchanger size increases.

4- The consideration of viscous friction affects also the entropy generation number of the heat exchanger considerably. The entropy generation number becomes much higher when the viscous friction is considered. This effect becomes more significant as the heat exchanger size increases. The assumption of constant viscosity leads to smaller values of entropy generation number. As the entropy generation number increases with increasing NTU values, no optimum size of the heat exchanger could be found to minimize the entropy generation number.

5- In order to find an optimum size of the heat exchanger, an operation factor was defined to reach a compromise between the thermal effectiveness and the entropy generation. Using this definition, no optimum heat exchanger size could be found for the no viscous friction case as the thermal effectiveness, entropy generation and operation factor of the heat exchanger increase with increasing NTU values. For the case of viscous friction, an optimum size at which the operation of the heat exchanger becomes most effective, from both the thermal effectiveness and entropy generation points of view, could be determined.

6- It was found that the effect of viscosity on the performance of the heat exchanger depends also on the high viscosity fluid mass flow rate and consequently the heat capacity rate ratio (C_{min}/C_{max}). It was found that as the mass flow rate of the high viscosity fluid increases, the adverse effects of viscous friction on the performance of the heat exchanger becomes stronger.

7- The effect of viscosity on of counter flow heat exchangers was found to similar to its effect on parallel flow heat exchangers. However, the counter flow heat exchanger performance showed a smaller tendency to deviate from the no viscous friction behavior when the viscous friction is considered. This feature stresses the fact that counter flow heat exchangers are more attractive than parallel flow exchangers.

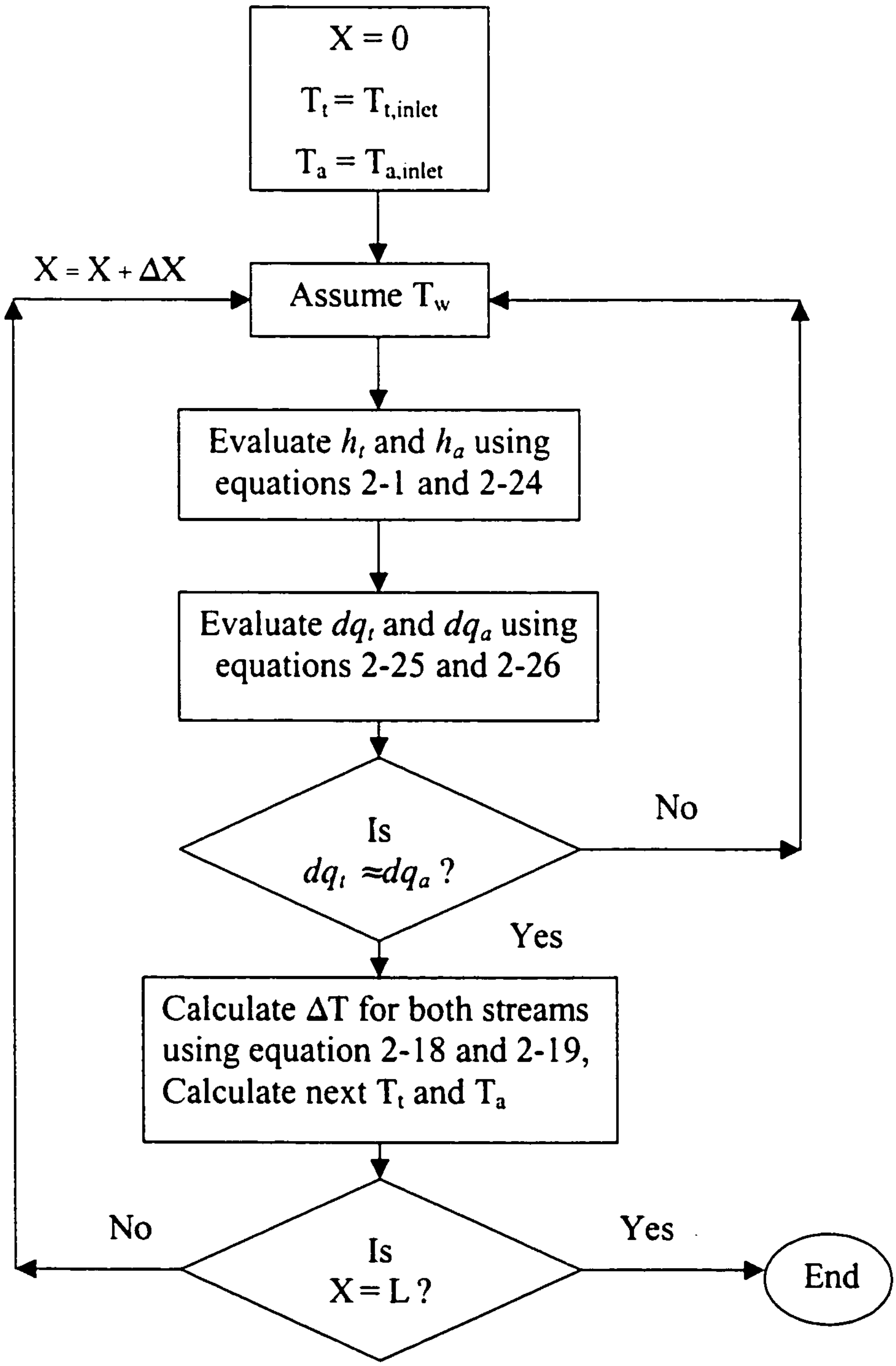
APPENDIX A

Flow Diagrams for

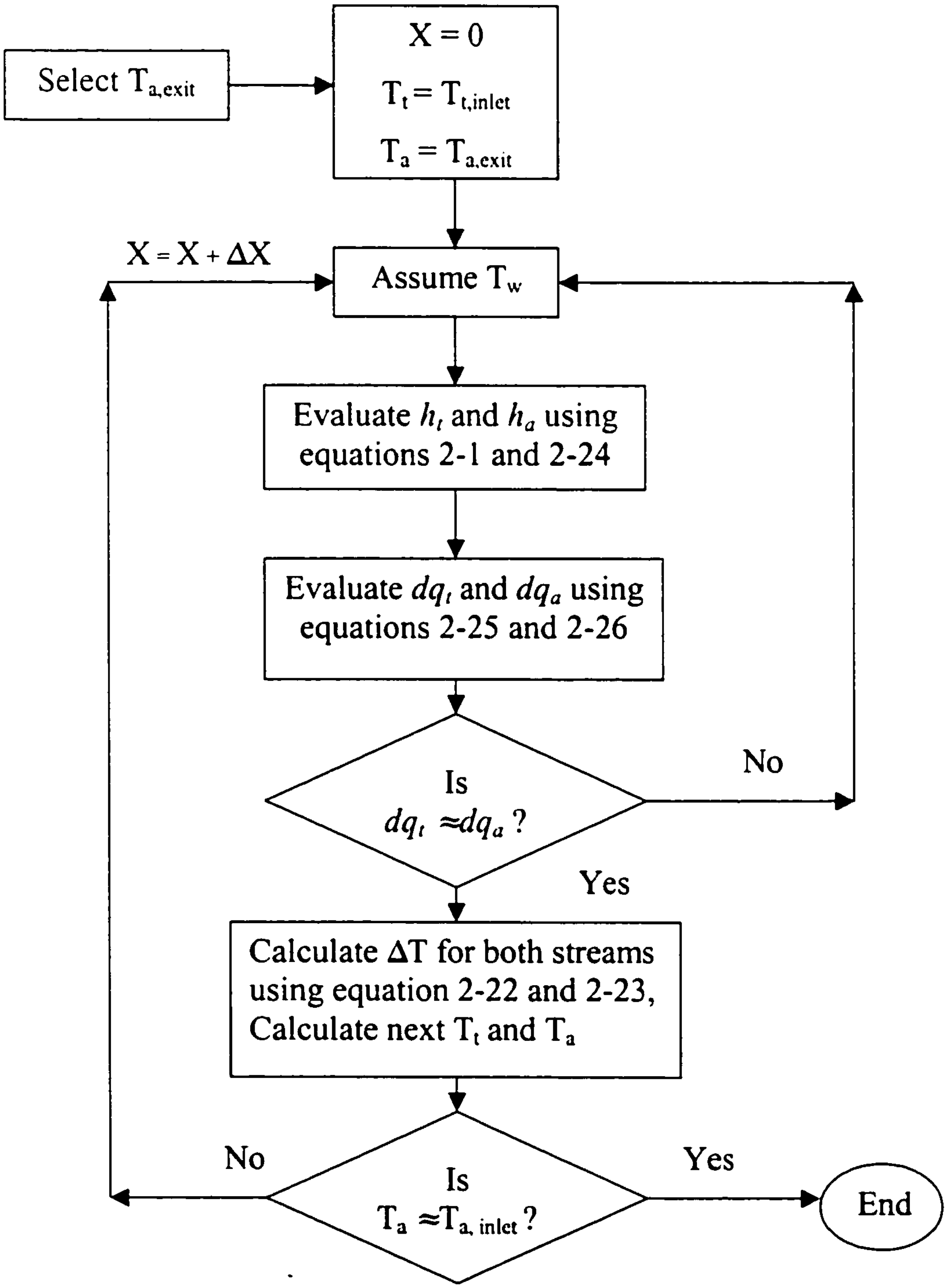
Calculating the Temperature Distribution of

Parallel and Counter Flow Heat Exchangers

Appendix A-1: Flow diagram for the calculation of the temperature distribution of parallel flow heat exchangers



Appendix A-2: Flow diagram for the calculation of the temperature distribution of counter flow heat exchangers



APPENDIX B

Thermophysical properties and parameters considered in the analysis

Appendix B: Thermophysical properties and parameters considered in the analysis.

Parameter	**(Annulus side)** $r_o = 7.6$ cm $r_i = 5.1$ cm	**(Tube side)** $r = 5$ cm
Density (kg/m^3)	1000	1260
C_p (J/kg K)	4195	2428
Thermal Conductivity (W/m K)	0.577	0.285
n	8.9	52.4
B	4700	23100
Mass Rate (kg/s)	3	4
Inlet Temperature (K)	274	283
Viscosity at 293 K (N s/ m^2)	9.93×10^{-4}	148

Note: n and B are the constants used in the viscosity- temperature relationship given by equation 2-1.

NOMENCLATURE

A	heat transfer area (m^2)
a_1	inner radius of the annulus (m)
a_2	outer radius of the annulus (m)
B	a coefficient used in equation 2-1 (K)
C_p	specific heat of fluid (J/kg K)
D_H	hydraulic diameter (m)
D_i	internal diameter of the tube (m)
dq_x	differential heat transfer rate (W)
dx	differential axial distance (m)
Ec	Eckert number
E_{in}	energy entering the system (W)
E_{out}	energy leaving the system (W)
f_c	friction factor
h_c	convection heat transfer coefficient of fluid (W/m^2 K)
k	thermal conductivity of the wall material (W/m K)
L	length (m)
m	mass flow rate of fluid (kg/s)
n	a coefficient used in equation 2-1
Nu	Nusselt number
P	pressure (Pa)
p	perimeter (m)
Pr	Prandtl number

q_c convection heat transfer rate (W)

Re Reynolds number

S entropy generation rate (W/ K)

St Stanton number

T temperature (K)

U overall heat transfer coefficient of the heat exchanger (W/m^2 K)

V mean velocity of the fluid (m/s)

W_c heat generation rate due to viscous friction (W)

Greek Symbols

δ wall thickness (m)

ε thermal effectiveness

μ viscosity of the fluid (N/s m^2)

ρ density of the fluid (kg/m^3)

Subscripts

a annulus

b bulk

F fluid

max maximum

min minimum

ref reference

s surface

t tube

W wall

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