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Application of numerical methods to study the effect of variable fluid viscosity on the performance of plate heat exchangers

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Abstract

Purpose – The purpose of this paper is to investigate numerically the influence of variable fluid viscosity on thermal characteristics of plate heat exchangers for counter-flow and steady-state conditions.

Design/methodology/approach – The approach to fulfill the purpose of the paper is to derive the one-dimensional energy balance equations for the cold and hot streams in the adjacent channels of a plate heat exchange composed of four corrugated plates. A finite difference method has been used to calculate the temperature distribution and thermal performance of the exchanger. Water is used as the hot liquid being cooled in the side channels, while a number of working fluids whose viscosity variation versus temperature is more severe were used as the cold fluid being heated in the central channel.

Findings – The program is run for a combination of working fluids such as water-water, water-isooctane, water-benzene, water-glycerin and water-gasoline. The temperature distributions of both streams have been plotted along the flow channel for all the above combination of working fluids. The overall heat transfer coefficients have also been plotted against both cold and hot fluid temperatures. It is found that the overall heat transfer coefficient varies linearly with respect to either cold or hot fluid temperature within the temperature ranges applied in the paper. The exchanger effectiveness is not significantly affected when either the temperature dependent viscosity is applied or the nature of cold liquid is changed.

Originality/value – This paper contains a new method of numerical solution of energy balance equations for the thermal control volumes bounded by two plates. A comparison of the calculated results with documented experimental results validates the numerical method.

Keywords Heat exchangers, Heat transfer, Numerical analysis, Finite difference methods

Paper type Research paper

Nomenclature

A	= plate heat transfer area, m^2
A'	= heat transfer area per unit length, m
b	= plate spacing, m
C	= constant in equation (8)
C	= heat capacity, W/°C
c_p	= specific heat at constant pressure, J/kg°C
D_e	= equivalent diameter, m
h	= convective heat transfer coefficient, W/m^2°C
j	= axial section designation
k	= plate conductivity, W/m°C
L	= developed plate length, m
m	= exponent in viscosity correction factor
$\dot{m}$	= mass flow rate, kg/s
n	= slope of $Nu\,Pr^{-1/3}$ against Re
NTU	= number of transfer units
Nu	= Nusselt number
Pr	= Prandtl number

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Q	=	local rate of heat transfer, W
Re	=	Reynolds number
r	=	exponent in equation (8)
t	=	time, s
T	=	temperature, °C
u	=	flow velocity, m/s
U	=	local overall heat transfer coefficient, W/m^2°C
$\overline{U}$	=	mean overall heat transfer coefficient, W/m^2°C
V	=	channel volume, m^3
w	=	flow width, m
x	=	coordinate normal to the plates
y	=	axial coordinate
μ	=	fluid dynamic viscosity, kg/m s
ε	=	heat exchanger effectiveness
ρ	=	fluid density, kg/m^3
δ	=	plate thickness, m
ϕ	=	ratio of developed to projected plate area

Subscripts

c	=	cold fluid
cv	=	control volume
h	=	hot fluid
m	=	mean
min	=	minimum
w	=	wall

Introduction

Plate heat exchangers' contribution in different process industries is increasing every day. These exchangers are considered an important priority in engineering applications, because of their advantages and pronounced characteristics such as compactness (occupying small physical space), excellent thermal performance, recovering heat from a small temperature difference, flexibility, lower risk of scale formation, accessibility to the plates for manual cleaning and lower repair and maintenance costs.

Plate heat exchangers have proved their superiority over conventional shell and tube heat exchangers in milk and medicine production and liquid food processing where severe hygienic standards have to be met and accurate temperature controls are required (Dunkley *et al.*, 1961). They are also suitable in rubber and paper industries and petrochemical plants (Reppich, 1999). Plate heat exchangers are being used as evaporators and condensers in heating and cooling systems (Mazza, 1984).

Temperature difference between the hot and cold fluids varies along the exchanger length, making the calculation of heat transfer rate complicated. Haseler *et al.* (1992) conducted local temperature measurements along the chevron region in a plate heat exchanger composed of three channels using single phase fluids at atmospheric pressure. Water and R113 were used as cold fluid in the central channel. Temperatures were measured in the central channel at five local points. The accuracy of the measured temperatures was estimated to be ± 0.2 K. These data were used to validate the HTFS computer program APLE, for the design and simulation of plate heat exchangers.

In counter-current and co-current flow arrangements, if the overall heat transfer coefficient is constant, the logarithmic mean temperature difference can be used as the true temperature difference between the hot and cold streams. However, the overall heat transfer coefficient depends on the fluids' thermal properties, and, therefore, varies with temperature. Experimental investigations conducted by Colburn (1933), and many other researchers since, in a liquid-liquid heat exchanger have also proved that the overall heat transfer coefficient is a function of temperature and, therefore, varies along the exchanger length. Therefore, for fluids whose physical properties vary strongly with temperature, assuming a constant overall heat transfer coefficient is not true. For such fluids, the logarithmic mean temperature difference does not represent the true temperature difference between the hot and cold streams. Investigations have been performed by Foote (1967) to establish correction factors for the logarithmic mean temperature differences in specific flow arrangements. These correction factors are only available for exchangers with unlimited number of plates and a few exchangers with limited number of plates.

Different methods of dealing with variable overall heat transfer coefficient have been proposed in classic heat transfer texts (Kern, 1950), or more recent in (Schlunder, 1989). Mehrabian (2003) developed an analytical-numerical approach to work out the longitudinal temperature changes of flow in the passages of a plate heat exchanger. Uniform heat flux, constant overall heat transfer coefficient, linear relationship between U and T, and linear relationship between U and ΔT were four special cases for which he solved the system of coupled, simultaneous differential equations obtained from the energy balance equation for the control volumes in the hot and cold stream channels.

Unless a simple relationship, such as those considered in (Mehrabian, 2003), between the overall heat transfer coefficient and temperature exists, the variations of overall heat transfer coefficient and temperature distributions of flow in the channels of a plate heat exchanger can only be determined if the analysis is performed on a numerical (finite difference, finite element or finite volume) basis. In this approach, the exchanger channels are divided into a number of axial sections small enough, so that temperature can be assumed to be constant throughout each section but varies from one section to another. A finite difference-based computer program is conducted to determine the overall heat transfer coefficient and the temperature of both fluids in each axial section. It is evident that the accuracy of the results depends on the number of axial divisions.

The purpose of this paper is to investigate how variable viscosity affects the overall heat transfer coefficient, temperature distribution and thermal performance of plate heat exchangers. To validate the numerical results obtained from this investigation, the plate dimensions and flow details used to obtain the experimental data in (Haseler *et al.*, 1992) are incorporated in the computer program. Numerical predictions are then compared with the experimental data.

Mathematical modelling

The plate heat exchanger used for numerical analysis has counter flow arrangement and U-shape configuration. Four standard APV SR3 plates have been used to form three flow channels. The side channels contain hot fluid flowing downwards, while the central channel contains cold fluid flowing upwards. The chevron region in central channel of the exchanger is divided into five axial sections so that fluid exiting from one axial section enters the next. The inlet and outlet ports are at the bottom-left and top-right of the plate. For the side channels, however, the inlet and outlet ports are reversed. It should be noted that the presence of distributor triangles, makes the heat transfer area per unit length to be different in different regions of the exchanger. This difference, however, is not appreciated in this paper because the nodal points are in the main chevron region and thus the axial divisions are assumed to be equal. The same plate geometry and flow arrangement was used in (Haseler *et al.*, 1992) for experimental local temperature measurements. This makes the comparison between two sets of data meaningful.

The mathematical modeling has been developed using the energy balance equation, based on the following assumptions:

(1) axial conduction in flow channels as well as in plates is not significant;

(2) the end plates of the exchanger are insulated;

(3) steady-state conditions prevail;
(4) the hot fluid is equally distributed in two side channels;
(5) heat loss to the ambient is ignored;
(6) phase change (boiling and condensation) does not occur;
(7) physical properties except viscosity are constant;
(8) plug (one-dimensional) flow is assumed; and
(9) temperature variations across the sub-channels are negligible.

Assuming plug (one-dimensional) flow in the channels of heat exchanger (assumption 8) allows a constant mean velocity to be considered in vertical direction in each channel. Assuming equally distributed fluid in hot and cold channels (assumption 4), makes this mean velocity to be constant in channels carrying each fluid. Based on the above assumptions, the energy balance equation for the control volume shown in Figure 1 is:

$$\rho_c V c_{pc}\left(\frac{\partial T_c}{\partial t} + u_{mc}\frac{\partial T_c}{\partial y}\right) = Q_{cv} \tag{1}$$

Applying the steady state conditions (assumption 3), equation (1) reduces to:

$$\dot{m}_c c_{pc} L \frac{dT_c}{dy} = Q_{cv} \tag{2}$$

Symmetry in geometry and flow makes the control volume shown in Figure 1 receive heat equally from both sides and T_h is the same on the side channels. As a result of this, equation (2) becomes:

$$\dot{m}_c c_{pc} \frac{dT_c}{dy} = UA'(T_h - T_c) \tag{3}$$

A similar control volume in either left or right hand side channel, receives heat only from one side (assumption 2). The energy balance equation for a side channel control volume is:

$$\dot{m}_h c_{ph} \frac{dT_h}{dy} = UA'(T_h - T_c) \tag{4}$$

The coupled simultaneous differential equations (3) and (4) are the governing equations controlling the temperature distributions of flow in two adjacent channels of

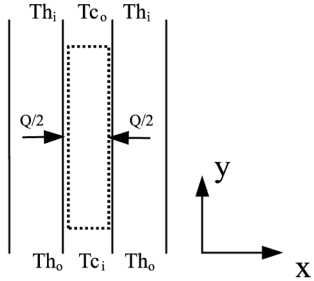

Figure 1.
Thermal control volume

the exchanger. Analytical solution for a general variation in U, except for some special cases such as those studied in (Mehrabian, 2003), becomes so complicated as to be impractical.

Numerical analysis

The numerical method consists of dividing the heat exchanger into a number of axial sections. A typical axial section has a surface area ΔA_j. The hot and cold fluid temperatures for this incremental surface area are T_{hj} and T_{cj}, respectively, and we shall assume that the overall heat transfer coefficient can be expressed as a function of these temperatures. Thus:

$$U_j = U_j(T_{hj}, T_{cj})$$

equation (2) can be applied for the axial section, to give:

$$\dot{m}_c c_{pc} \Delta y_j \left(\frac{dT_c}{dy}\right)_j = \Delta Q_j \tag{5}$$

equations (3) and (4) can also be written for two axial sections in the adjacent channels of the exchanger, giving:

$$\dot{m}_c c_{pc} \left(\frac{dT_c}{dy}\right)_j = U_j A'(T_h - T_c)_j \tag{6}$$

$$\dot{m}_h c_{ph} \left(\frac{dT_h}{dy}\right)_j = U_j A'(T_h - T_c)_j \tag{7}$$

Numerical solution of the above equations is obtained when the spatial derivatives are discretised using forward differences. Tables of temperature dependant viscosities (Yaws, 2003) are read in, and the program calls the viscosity of each fluid at the operating temperature in each axial section. Linear interpolation is done when the operating temperature does not coincide with the tabulated values. The other fluid properties like density and thermal conductivity are assumed to be temperature independent (assumption 7). The values of these properties for each stream are assigned at the mean stream temperature and used as input data. The inlet temperatures of cold and hot streams are used as boundary conditions in numerical analysis.

The dimensionless heat transfer coefficient in the channels of a plate heat exchanger is evaluated assuming a heat transfer correlation of the type (Rao *et al.*, 2002):

$$Nu = C\,Re^n Pr^r \tag{8}$$

Shah and Focke (1988) conducted an experimental investigation to study the heat transfer and pressure drop characteristics of plate heat exchangers. They noticed that constant C depends on the plate type and exchanger geometry, while constant n depends on the fluid flow regime.

Edwards *et al.* (1974) proved that at Reynolds numbers less than about 10, the experimental data plotted as $Nu\,Pr^{-1/3}$against Re for the APV Junior Paraflow plates fall on a line of slope about 1/3 indicating the typical laminar heat transfer relationship:

$$Nu = 0.23 Re^{1/3} Pr^{1/3}$$

At higher Reynolds numbers ($Re > 10$), the slope is about 0.7, giving for transitional and turbulent conditions:

$$Nu = 0.54Re^{0.7}Pr^{1/3}$$

This type of relationship is to be expected for turbulent flow regime between two parallel flat plates at distance b from each other. It is assumed that the Reynolds number based on the equivalent diameter ($D_e = 2b/\varphi$) takes care of heat transfer augmentation due to plate corrugations. The results obtained by Edwards *et al.*, for Newtonian fluids when APV Junior Paraflow plates were used to form the flow channels, can be generalized for any other plate type, provided a modification in the value of constant C is applied.

Mehrabian (1996) performed an extensive study from experimental, theoretical and computational points of view to investigate the hydrodynamic and thermal characteristics of plate heat exchangers. He suggested the following relationship applies for APV SR3 plates in turbulent regime ($Re > 10$):

$$Nu = 0.124Re^{0.7}Pr^{1/3} \tag{9}$$

equation (9) applies to both cold and hot channels of the plate heat exchanger giving, respectively, the film heat transfer coefficients h_{cj} and h. The overall heat transfer coefficient for the axial section j is then:

$$\frac{1}{U_j} = \frac{1}{h_{cj}} + \frac{\delta}{k} + \frac{1}{h_{hj}} \tag{10}$$

As suggested in (Edwards *et al.*, 1974), the flow in plate heat exchanger channels may be turbulent at Reynolds numbers as low as 10. Thus, turbulent flow assumption in present analysis is reasonable and equation (9) holds for water as the hot fluid, as well as, glycerin, benzene and iso-octane as the cold fluid. The cold fluids are selected to have temperature dependent viscosities.

Results and discussion

To get grid-independent results, the program was run for different number of axial divisions. The numerical results obtained for certain grid points were compared with the corresponding experimental data from (Haseler *et al.*, 1992) and the discrepancies were recorded. It was noticed that increasing the number of grid points reduces the discrepancies. However, when the number of axial divisions is 17, the minimum discrepancies are achieved. Using more than 17 axial divisions did not reduce the discrepancies significantly. The results presented in this paper are obtained using 17 axial divisions, those corresponding to the grids; their numbers are multiples of 3 (3, 6, 9, 12 and 15). This is for the purpose of comparison with the experimental results at corresponding points available in (Haseler *et al.*, 1992).

It should be mentioned that at the beginning of numerical solution the outlet temperatures of two fluids are not known, the fluids properties are evaluated at mean inlet temperatures of two fluids. When the outlet temperatures are obtained, the properties of each fluid are evaluated at the mean temperature of that fluid (Khoramabadi, 2004).

The plate dimensions and flow conditions used for numerical analysis are listed in Table I. These are exactly the same dimensions and conditions used for experimental local temperature measurements in (Haseler *et al.*, 1992). Typical experimental local temperatures for water are given in Table II. The Reynolds numbers for hot and cold fluids, based on equivalent diameter $D_e = 6.28\,\text{mm}$, are 879 and 304, respectively. The numerical results for the local temperatures in the central channel of the exchanger are then compared with the experimental temperatures in the chevron region. This comparison is shown in Table II, and quantitative agreement is achieved. The error is within -2.14 and $+1.88$°C, showing that the numerical technique is accurate.

Now that the numerical procedure is validated, a bigger temperature difference between the hot and cold fluids inlet temperatures is applied to highlight the effect of viscosity variations on temperature distribution, the overall heat transfer coefficient and thermal performance of heat exchanger.

The cold and hot fluids inlet temperatures are 5 and 90°C, respectively. The program is run for four sets of cold-hot working fluids, namely water-water, water-glycerin, water-benzene and water-isooctane. Temperature distributions of cold and hot fluids along the exchanger length are shown in Figures 2 and 3, respectively. The temperature distributions for water-benzene are similar to those for water-isooctane and, therefore, are not shown in Figures 2 and 3. Temperature distributions for both constant and variable viscosities are shown for each set of working fluids. It is noticed that the deviation between temperature distributions of water for constant and variable viscosities is more than that for glycerin and isooctane. However, the viscosity variations of water with temperature are less severe than that for glycerin and isooctane. The reason for this behavior is that the convective heat transfer coefficient for glycerin and isooctane are less than that for water and, therefore, they control the overall heat transfer coefficient.

Figures 4 and 5 show the variations of overall heat transfer coefficient with cold and hot fluid temperatures, respectively. The overall heat transfer coefficient varies almost linearly against fluids temperatures. However, when the fluids viscosities are assumed

Table I.
Plate dimensions and flow details

Developed plate length	L	1 m
Flow width	W	0.35 m
Plate heat transfer area	$W \times L$	0.35 m^2
	Hot fluid	*Cold fluid*
Inlet temperature	77.9°C	47.9°C
Outlet temperature	71.7°C	76.9°C
Specific heat	4,191 J/kg K	4,184 J/kg K
Mass flow rate	472.6 kg/hr	101.2 kg/hr

Table II.
Comparison of numerical and experimental results

Numerical results (°C)	Experimental results (Haseler *et al.*, 1992) (°C)	Error (°C)
47.9	47.9	0.0
60.16	61.9	1.74
67.69	66.8	−0.89
72.16	70.15	−2.01
74.74	72.6	−2.14
76.23	74.35	1.88
77.07	76.9	−0.17

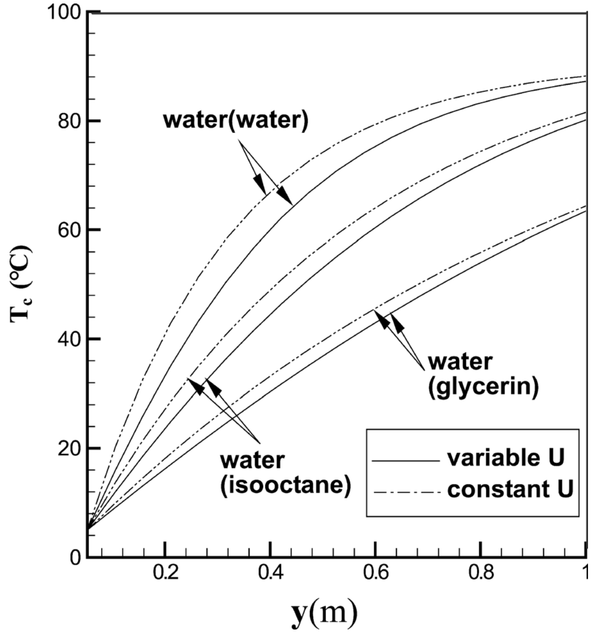

Figure 2.
Temperature distribution of cold fluid along the flow channel

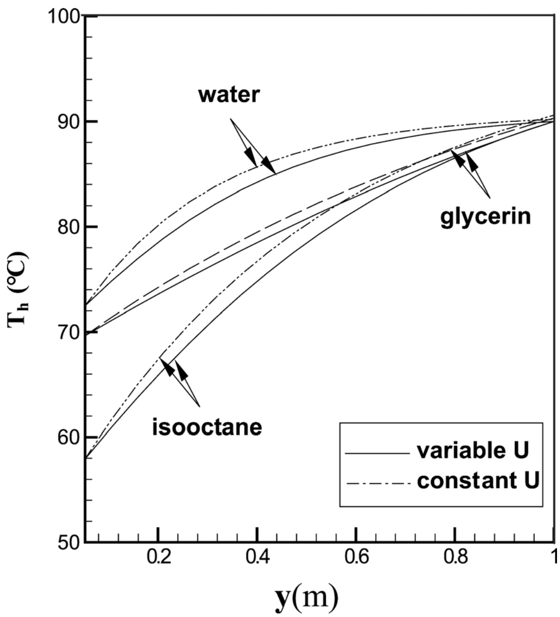

Figure 3.
Temperature distribution of hot fluid along the flow channel

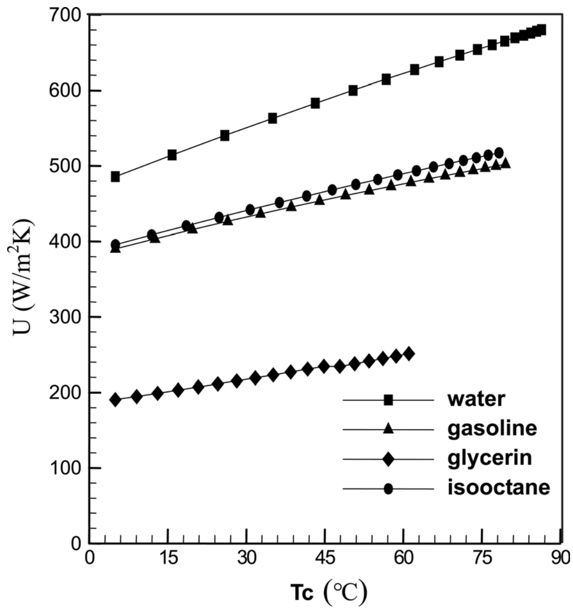

Figure 4.
Overall heat transfer coefficient against cold fluid temperature

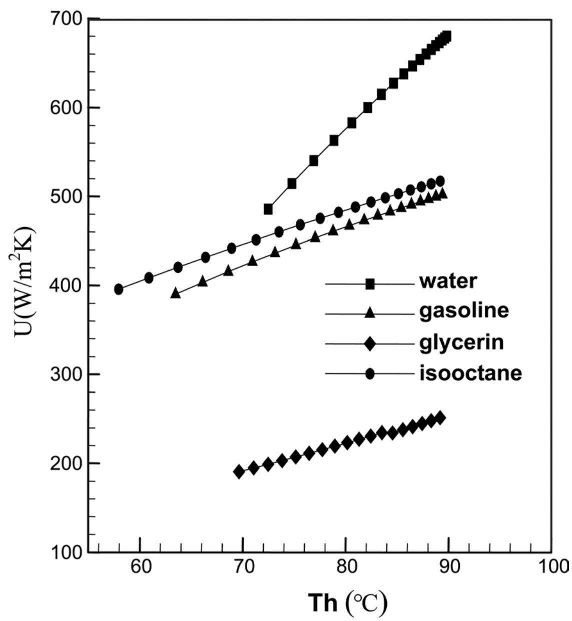

Figure 5.
Overall heat transfer coefficient against hot fluid temperature

constant, the overall heat transfer coefficient is also a constant. This makes the variable viscosity an important factor controlling the overall heat transfer coefficient. The reason for linear relationship between U and T, while viscosity is thermally activated, is shown in Figure 6, in which the viscosity of liquids involved in this investigation are plotted against temperature. The viscosities show linear variations within the temperature ranges stated in Table II. This behavior, however, does not violate the fact that viscosity of most liquids scales as an exponential function versus temperature. Linear viscosities, as discussed above make the hot and cold side heat transfer coefficients vary linearly with temperature as shown in Figures 7 and 8, respectively.

The exchanger $\varepsilon - \mathrm{NTU}$ diagrams for constant and variable viscosities are shown in Figure 9. This figure is for water-isooctane. Similar figures can be drawn for water-water, water-benzene and water-glycerin. The exchanger performance does not change considerably under the influence of temperature dependent viscosity.

Figure 10 shows the $\varepsilon - \mathrm{NTU}$ diagram for different working fluids when viscosity is temperature dependent. The exchanger performance does not depend on the nature of fluid as long as NTU is small ($\mathrm{NTU} < 1$), but as NTU is increased ($\mathrm{NTU} > 1$), the performance is differentiated for different working fluids. The performance calculations related to Figures 9 and 10 are based on the following definitions for NTU and $\overline{U}$:

$$\mathrm{NTU} = \frac{\overline{U}A}{C_{\min}} \tag{11}$$

$$\overline{U} = \frac{1}{\Delta T}\int_T U \, \mathrm{d}T \tag{12}$$

The integration could be with respect to T_c or T_h.

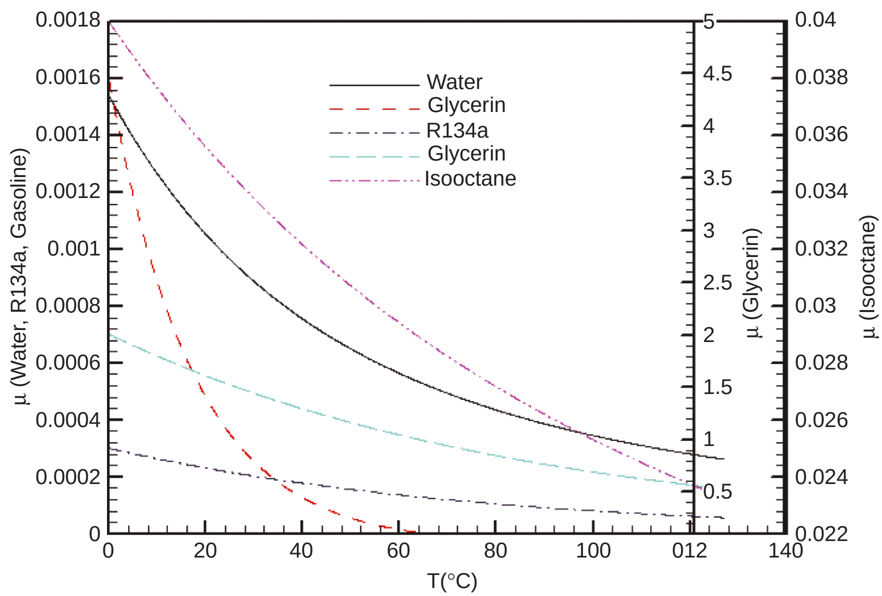

Figure 6.
Variations of the viscosity of working liquids with temperature

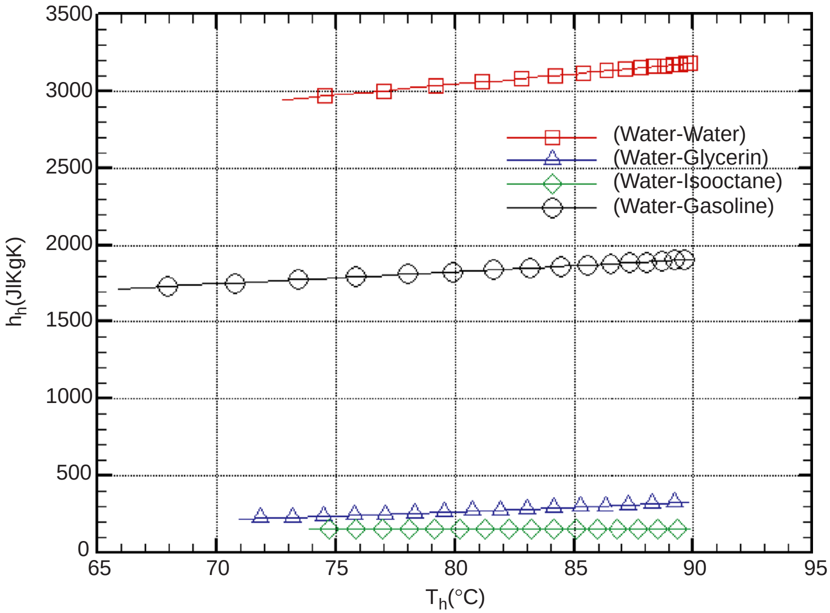

Figure 7.
Hot side heat transfer coefficient against hot fluid temperature

Summary

A finite difference based numerical method has been developed to predict the axial temperature changes for flow in the channels of a plate heat exchanger under the influence of temperature dependent fluid viscosity. A combination of working fluids such as water-water, water-isooctane, water-benzene, water-glycerin and water-gasoline has been examined. In all the above working fluid combinations, water is used as the hot fluid being cooled in the side channels of the exchanger, while the other fluid whose viscosity variation versus temperature is more severe, is used as the cold fluid being heated in the central channel. The exchanger for the numerical analysis consists of four standard APV SR3 plates making three channels. The exchanger geometry and flow conditions have been selected to match the geometry and conditions used in a well established previous experimental investigation. The temperature predictions obtained from the numerical method developed in this paper have been compared with the experimental results obtained with the same kind of plate and similar flow conditions. The error being calculated as the local temperature difference, was between -2.18 and $+1.88$°C. Thus, quantitative agreement is achieved, proving the correctness of the numerical model. When the validation procedure succeeds, the program is run with a different operating condition.

In a previous investigation performed by the first author (Mehrabian, 2003), the effect of variable overall heat transfer coefficient on the performance of plate heat exchanger has been studied. In that investigation, four different models for variations of the overall heat transfer coefficient have been proposed. One of those models is a linear relationship between U and T. The numerical results obtained in the present

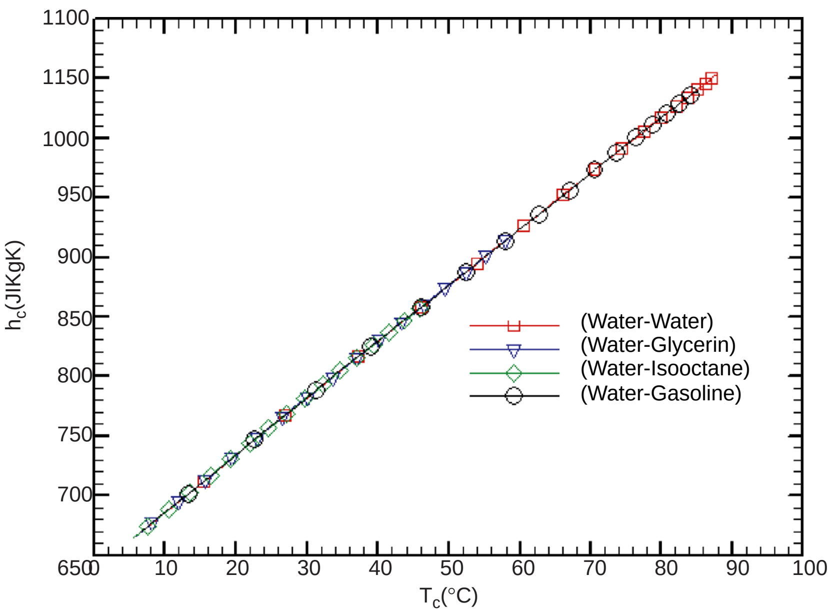

Figure 8.
Cold side heat transfer coefficient against cold fluid temperature

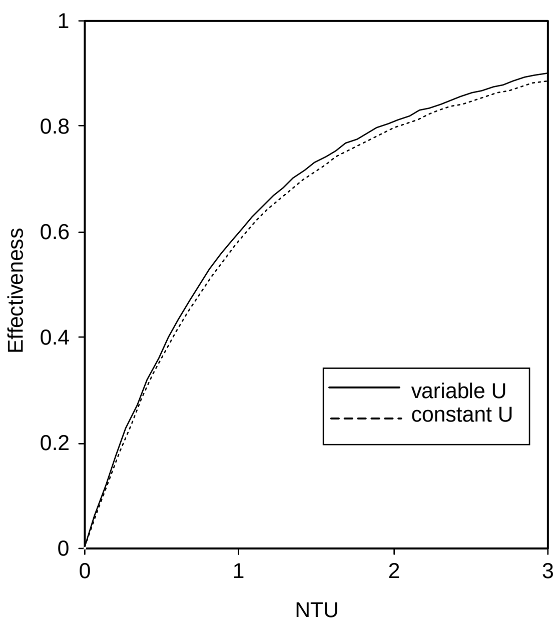

Figure 9.
Effect of variable U on ε – NTU with water-isooctane

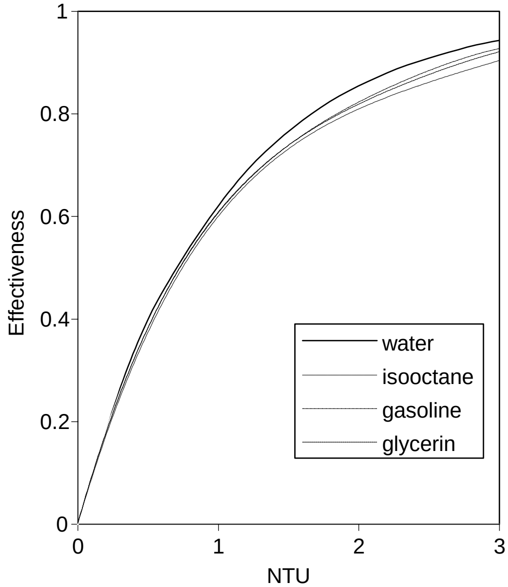

Figure 10.
Effect of fluid kind on $\varepsilon -$ NTU with variable U

paper also indicate a linear relationship between U and T_c as well as between U and T_h, within the temperature ranges adopted in this investigation. The agreement between the present work and the previous work which have been conducted using two different approaches, is of great concern to the design engineers and plate heat exchanger manufacturers. That means, for certain class of liquids whose viscosities change exponentially with temperature a linear relationship between U and T_c or T_h exists within small and moderate operating temperatures.

The results obtained in this analysis show that viscosity variations have only a minor effect on the exchanger effectiveness. This may lead to the conclusion that a correction factor of the kind $(\mu/\mu_w)^m$ or $(Pr/Pr_w)^m$ when incorporated in equation (8) may take care of the viscosity variations. The numerical predictions based on the present computer program can be used to evaluate the exponent m.

Conclusions

The impact of using mean viscosities versus local viscosities in thermal analysis of plate heat exchangers is investigated numerically. For fluids with strong temperature dependent viscosities, the difference between temperature distributions for constant and variable viscosities is less than that for fluids having weak temperature dependent viscosities. With variable viscosities, the overall heat transfer coefficient varies linearly with respect to either cold or hot fluid temperature when the temperature ranges are small and moderate. The exchanger effectiveness is not highly affected when the temperature dependent fluid viscosities are applied. The nature of fluid does not change the performance when NTU is small.

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