Laminar flow heat transfer of a dilute viscoelastic solution in flattened tube heat exchangers

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ABSTRACT

Results of studies on heat transfer of viscoelastic solution in flattened tube exchangers were presented. Effect of aspect ratio on the heat transfer performance in exchangers with 0.635 cm and 1.27 cm original diameters and 50 cm to 76 cm lengths were carried out. Five flattened-tube heat exchangers with four thermocouples soldered at regular intervals on the outside wall were placed in turn in the experimental circuit to determine the heat transfer coefficients. Hot water was used as the heating medium; and dilute solution of polyacrylamide in water was used as the viscoelastic solution. Heat transfer increase as a result of flattening the tubes could be as high as 101% while the effect due to secondary flow had a maximum increase of about 86% at an aspect ratio of 1.6.

Paper Description: Heat transfer of dilute viscoelastic solution in flattened tube exchangers is presented. Heat transfer enhancement was at a maximum at an aspect ratio of 1.6.

Keywords: aspect ratio, dilute viscoelastic solution, flattened tube, heat transfer increase, secondary flow

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Nomenclature:

a	major axis of cross-section
Ar	Aspect ratio
b	minor axis of cross-section
c_p	specific heat
d	original tube diameter
h_c	heat transfer coefficient, circular tube
h_f	heat transfer coefficient, flattened tube
k	thermal conductivity
K	material constant
L	length
q	shear rate, flow rate
q_1	flow rate for central region, flattened tube
q_2	flow rate for semi-circular ends, flattened tube
Q	volumetric flow rate
U	overall heat transfer coefficient
v_s	secondary flow velocity
W	Watt
ρ	density
μ	viscosity

Φ Geometric factor

1. INTRODUCTION

Refreshing the boundary layer at the wall of an exchanger improved heat transfer performance. Bott [1, 2], Bott and Romero [3], Chong [4] and Tahti [5] demonstrated this by scraping of the exchanger surface. The heat exchanger could also be vibrated as demonstrated by Song [6], or electrostatic forces applied to it as shown by Nassauer and Kessler [7]. Pattenden and Richards [8] rotated the exchanger tubes to achieve the same effect. Liu and Masliyah [9] and Yang and Chung [10] showed that secondary flows created in curved tube exchangers also improved heat transfer. For viscoelastic solutions, secondary flow could also be induced from differences in normal forces by using suitable sections as demonstrated by Green and Rivlin [11]. Carlson and Irvine Jr [12], Deissler and Taylor [13], Mitsuishi et al. [14], Middleman [15] and Sparrow [16] studied flows in isosceles triangular ducts. The more acute the triangle the more pronounced was the secondary flow created. Flow in rectangular ducts could also generate secondary flows as demonstrated by Beyer and Towsley [17], Hartnett and Kostik [18], Rao [19] and Wheeler and Wissler [20]. Mitsuishi and Aoyagi [21] conducted studies of non-Newtonian flow in both triangular as well as rectangular cross-section ducts using dilute solutions of 1-7% carboxymethyl cellulose (CMC) and similar results were obtained. Studies using polyacrylamide solutions by Ray and Date [22] showed significant improvements with the square duct, particularly at higher Prandtl numbers. Oliver and Karim [23] showed that secondary flow enhanced heat transfer in flattened tubes. Siginer and Letelier [24] conducted a study on straight tubes of arbitrary cross-section including the triangle, the square, the rectangle and the ellipse. A relationship between the Nusselt number and the Weissenberg number was obtained. The flow properties can be described for various conditions and geometries as illustrated by Metzner et al. [25] and Metzner and White [26]. Heat transfer in non-circular tubes has a main application in compact heat exchangers.

2. MATERIAL AND METHODS

Five heat exchangers (Ex1 – Ex5) were constructed by flattening copper tubes of diameter 0.00635 m (quarter-inch) and 0.0127 m (half-inch) between rollers to obtain tubes of different aspect ratios. The tubes were 12.7 m to 19.4 m long. The circuit included a feed tank, a gear pump, the test section, a flow-mixer, and a return heat exchanger. The feed tank was made of glass with a metallic frame, and the dimensions were 0.50 m, by 0.25 m and 0.25 m high. A stirrer was used to even out the temperature of the solution. The gear pump was a calibrated 2.5 x 10^{-5} l/rev Slack and Parr type with a variable speed to regulate the flow rate. The maximum flow rate obtainable for water using this pump was about 1.1 x 10^{-3} l/min. A 'Crawley' type pressure gauge with scale ranging from 0 to 689.5 x 10^{3} Pa was used to calibrate the gear pump for various values of back-pressure. A water manometer was also used to measure the pressure drops across the test-section. A tank similar to the feed tank was used as the heating water storage tank. A 2-KW Griffin-Grundy Temperostat with an attached stirrer was used to maintain constant temperatures in the tank. The heating water was circulated through the jacket of the heat exchanger using a Stuart Turner pump. A return heat exchanger is used to cool back the test solutions to the ambient temperature before flowing back into the storage tank. A closed system was adopted in order to save on chemicals. A 2-mV Honeywell-Brown ‘Electronik’ recorder was calibrated to record all temperatures. All thermocouples used were connected to this recorder via a rotary switch. Model 35 Fann V-G concentric cylinder viscometer was used to determine the shear rate-shear stress relationship for the test solutions. The test rig was connected by 0.0127 m and 0.00635 m copper and rubber tubes. All electrical apparatus was connected to a common panel. The schematic diagram of the experimental circuit is given in Fig 1. Three or four copper-constantan thermocouples, with temperature range 0 C-100 C, were soldered on the outside wall of each exchanger for the measurement of average wall temperatures. The section between the heated section and the flow-mixer was lagged. The flow-mixer was constructed from a 0.075 m diameter brass cylinder 0.085 m long with four paddles and four baffles. A Voss S30, 0.025 KW variable speed stirrer was attached. Four thermocouples were placed at various locations in the mixer to record the exit temperature. The stirrer shaft was connected to the drive motor by a short rubber tubing to accommodate for any misalignment. It also reduced heat conduction effects from the motor casing. During the experiment the exchanger section was connected to a manometer. Table 1 gives the dimensions of the exchangers used in the experiments. The experiments were performed with dilute solution (250 ppm in water) of a polyacrylamide as the viscoelastic medium. Various weights of the polymer were dissolved in tap water. The material was added in small amounts with constant stirring to avoid agglomeration. Heating could enhance the dissolution, but this was not done to avoid unnecessary thermal degradation. Polyacrylamide was chosen for its resistance to mechanical as well as thermal degradation. Effect of degradation was minimized by preparing fresh solutions on a daily basis. Degradation was further reduced by maintaining low flows and small temperature differences. The densities of the solutions were unaffected by the dissolved polymer. The physical properties were taken as that of water. The heat exchangers were heated with hot water from a well-stirred water bath which formed part of the experimental circuit. The circuit comprised of a feed tank, a gear pump, the flattened tube test section, a flow-mixer, and a return heat exchanger before flowing back into the feed tank. The flow rate for each run ranged from 1.5×10^{-4} -1.15×10^{-3} l/min. The bath temperature was first set and maintained at a desired value. The feed solution was circulated; the flow-mixer and the return line heat exchanger were started. Once equilibrium was reached, temperatures and flow-rates were obtained. The series of experiments were repeated for different test solutions at different flow rates; and for different exchangers of different diameters and aspect ratios. The bath temperature was then changed, and the above procedure repeated for different temperature differences and different heat exchangers. All the experiments were carried out at room temperature.

3. THEORY

Overall heat flux through the flattened tube heat exchanger is given by,

$$q = UA\Delta T_{LM} \tag{1.1}$$

where

q = overall heat flow = volumetric flow x density x specific heat

U = overall heat transfer coefficient = Watt / (area. Kelvin) = $W/(m^2 \cdot K)$

A = heat transfer surface area (m^2) = πdL

ΔT_{LM} = log mean temperature difference (K)

$= (\Delta T_A - \Delta T_B)/\ln(\Delta T_A / \Delta T_B)$

where ΔT_A is the temperature difference between the two streams at inlet, and ΔT_B is the temperature difference between the two streams at outlet. At equilibrium results of temperatures and flow rates for exchangers Ex1 to Ex5 were recorded and the necessary graphs plotted to obtain the individual heat transfer coefficients. These are shown in the plots of q/A vs ΔT_{LM} for all exchangers as given in Fig. 2. The theoretical increase in heat transfer coefficients due to flattening the tubes were calculated and recorded. The difference between these values and the experimental values gave the heat transfer enhancement due to secondary flow. The percentage increase was obtained by comparing these increases with the theoretical values. A graph of additional heat transfer and the percentages was plotted against the aspect ratio.

At low shear rates, the constitutive equation used was the one suggested by Oldroyd [27]:

$$\tau + \lambda_1(d\tau/dt) = (\mu/g_c)(d\gamma/dt) + \lambda_2(d^2\gamma/dt^2) \quad (1.2)$$

where,

γ = strain

λ_1 = relaxation time

λ_2 = retardation time

The flow equation for a Newtonian laminar flow in circular tubes can be reduced to:

$$Q = \pi R^4 P / 8\eta L \quad (1.3)$$

where,

Q = volumetric flow rate

P = pressure gradient

R = tube radius

For similar flow in a uniform non-circular tube the equation can be represented by:

$$Q = CD^4P/ \eta L \tag{1.4}$$

where,

D = some characteristic dimension

C = constant

Secondary flow is inversely proportional to $D^{1.5}$. At a given flow rate, and for 0.00635 m and 0.0127 m tubes of equal aspect ratios;

$$\text{Normal stress/Shear stress})_{0.635cm} / \text{Normal stress/Shear stress})_{1.27cm} = 2^{1.5}$$

$$= 2.82 \tag{1.5}$$

T low flow rates secondary flow developed more readily in the tube of smaller diameter. The following approach using original diameter was proposed. Assuming that heat transfer is proportional to the one-third power of wall shear rate, then the relative increase in heat transfer could be derived. The ellipse can also produce secondary flow in the case of viscoelastic materials. This was shown by Semjonow [28] with high polymer melts using a brass tube and a dye-injection technique. The secondary flow velocity was shown to be given by Eqn. (1.6):

$$v_s = K\Phi P^4(x^2/a^2 + y^2/b^2 - 1) \{(x^2/a^2 + 5y^2/b^2 - 1)^2x^2 + (5x^2/a^2 + y^2/b^2 - 1)^2y^2\} \tag{1.6}$$

where,

K is the material constant given by,

$$K = (1/4) (G_0/H_0^5) (H_1/H_0 - G_1/G_0) \tag{1.7}$$

Φ is the geometric factor given by,

$$\Phi= a^6b^6(a^2-b^2) / \{(a^2+b^2) (5a^4+6a^2b^2+5b^4)\} \tag{1.8}$$

The material constant was obtained from shear rate (q) relationships by experiment,

$F (q^2) = F_0 + F_1q^2 + \dots$

$G (q^2) = G_0 + G_1q^2 +\dots$

$H (q^2) = H_0 + H_1q^2 +\dots$

From the above, it could be seen that at any point in the ellipse, for a particular material, the secondary flow was only a function of the ellipse geometry. Differentiation showed that the secondary flow velocity was a maximum at any point when the aspect ratio was about 1.5. Visual studies of the secondary flow patterns were carried out by Oliver [29] using flattened tubes, which could be approximated to an ellipse. However, no conclusive evidence of secondary flow was encountered. Harris and Goldschmidt [30] studied the heat transfer of combustion gases in inclined elliptical tube heat exchangers and found increased heat transfer performance. Studies on laminar film condensation heat transfer in and on inclined elliptical tubes by Fieg [31] also showed enhanced heat transfer especially for shorter tubes. Studies using refrigerants in inclined flattened tubes by Wilson et al. [32] also showed improved heat transfer performance. However, little experimental work was been done on heat transfer of non-Newtonian fluids in non-circular tubes; and less work has been done on viscoelastic liquids in flattened tubes.

4. RESULTS AND DISCUSSION

The heat transfer coefficients were computed for all exchangers and the polymer solution from Fig. 2. Flattening the exchanger tubes has the effect of increasing wall shear rates which in turn increase heat transfer coefficients. This increase of heat transfer due to increased shear rates were calculated to range from 6 % for a heat exchanger with an aspect ratio of 1.4 to about 101% for a heat exchanger of aspect ratio of 5.7. For the case of viscoelastic solutions the flattening of the heat exchanger tubes has also the effect of promoting secondary flow which also enhances heat transfer performance. The maximum additional values due to secondary flow for the 250 ppm polymer solution increased with increasing aspect ratio hitting a maximum at about 86% and reducing to about 50% with increasing aspect ratios. The maximum effect was seen at an aspect ratio of 1.6. The details are as shown in Fig. 3. This agreed well with results of 1.5 obtained by Semjonow [28] on exchangers with elliptical cross-sections.

Several factors interact to influence the overall heat transfer performance of dilute viscoelastic solutions in a flattened tube exchanger system. Increase in concentration should increase normal stresses and improve heat transfer performance from improved secondary flow but there could be more resistance from possible increase in viscosity. As mentioned earlier, increased wall shear rates from flattening enhances heat transfer performance but the penalty of increased pressure drop must be borne in mind. The velocity profile would also change. Increasing the aspect ratio improves heat transfer performance up to a point but when the aspect ratio is too high the secondary flow would be restricted by the tight geometry. Increasing the tube diameter could help but there is natural convection effects to reckon with.

Results of studies by Hartnett and Kostic [18] and Rao [19] for heat transfer of non-Newtonian liquids in rectangular channels, and in arbitrary cross-sectioned tube including the rectangle, the triangle and the ellipse by Siginer and Letelier [24] all exhibited enhanced heat transfer which was attributed to the presence of secondary flow.

Oliver [29] observed the presence of such secondary flow in flattened tubes by using a dye to trace the flow and Wilson et al. [32] obtained enhanced heat transfer working with refrigerants in flattened tubes. To further support these findings, it is recommended that optimization studies be conducted dealing with a greater range of tube diameters, shapes, orientation and aspect ratios and solution concentrations.

5. CONCLUSIONS

Heat transfer increases due to increased wall shear rate was found to be as large as 101%, while increases due to secondary flow was found to be about 86% for the 250 ppm solution. The maximum effect due to secondary flow occurred around aspect ratio of 1.6. Further studies over a greater range of viscoelastic properties are recommended. Use of different tube diameters and use of other non-circular tubes to investigate similar secondary flow effects are also recommended.

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FIGURE CAPTIONS

List of Figures

List of Tables