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Experimental and Theoretical Investigation of Hydrodynamics Characteristics and Heat Transfer for Newtonian and Non-newtonian Fluids

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Abstract

This work studies the hydrodynamic characteristics and heat transfer for both oil Newtonian and non-Newtonian fluids. Therefore the comparison between velocity and temperature distribution Profiles, pressure drop and Nusselt numbers for heating and cooling of these fluid were achieved. The fluids flow into tubes with laminar flow condition. The search leads to shown the effects of rheological properties of non-Newtonian fluids and their variation with temperature. For the theoretical work, the fluid flow through circular tube was taken under the conditions of constant wall temperature, fully developed flow and a uniform temperature with neglecting axial conduction heat transfer axial conduction heat transfer. Temperature-dependent power law relationship used to describe the rheological properties. It was found that the hydrodynamic characteristics and pressure drop were affected by the fluid temperature. For the experimental work, the Newtonian fluid is oil with specific heat of (2 KJ/Kg °C at 50°C), thermal conductivity of (0.14 W/m °C at 50°C) and non-Newtonian fluid is carbopol polymer with (2.8 KJ/Kg °C at 20°C) and thermal conductivity of (0.3 W/m °C at 20°C). Tests covered a range of Graetz numbers (Gz) from (160 to 1600) also the temperature differences from (50°C to 90°C) for heating and (70°C to 30°C) for cooling.

Keywords

Newtonian Fluid, Non-newtonian Fluid, Fluid Velocity Profile, Fluid Temperature Profile

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1. Introduction

The hydrodynamics and heat transfer problems for Newtonian and non-Newtonian fluids are different but it can identify through two cases, these are heat exchanging and maintaining temperature at predetermined value. This paper deals with viscous Newtonian fluids and variable non-Newtonian fluids which their rheological properties are sensitive to the temperature. A. K. Datta, [1], studied the heat transfer theoretical model for a flow of a non-Newtonian fluid and the importance of this process into food processes, he includes that the laminar flow is better than turbulent flow for these processes. Timothy J. R. and Vijaya G. S. [2], used A double-pipe helical heat exchanger to determine the effects of thermally dependent viscosity and non-Newtonian flows on heat transfer and pressure drop for laminar flow. The Pressure drop data were compared using ratios of the pressure drop of the non-Newtonian fluid to a Newtonian fluid at identical mass flow rates and consistency indices. C. Hsu and S. V. Patankar, [3], solved numerically the case of flow into curved tubes and these results compared with the experimental data for velocity and temperature distributions. A. B. Metzner, et. al, [4], present a theoretical and experimental study for variables controlling temperature distribution for non-Newtonian fluids, the results compared with the results of Newtonian fluids and the experimental

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data taken for Graetz number (100-2000). J. Schenk and J. Van Laar,[5], studied the theoretical heat transfer for non-Newtonian fluids flow into cylindrical pipes, the flow was laminar. They introduced the velocity and temperature distributions; also they studied the effect of wall tube thermal resistance. R. M. Cotta, et. al, [6], solved analytically the governing equations for non-Newtonian fluid of laminar forced convection into pipe and parallel plate channel, the wall temperature was uniform. Nusselt number (local and average) obtained for the Graetz numbers. Y. Xie, et. al, [7], solved numerically the non-Newtonian fluid flow performance into rectangular pipes with protrusions. The thermal performance of non-Newtonian fluid was enhancement. The comparing between Protrusion and smooth channels were achieved, they concluded that protrusion structure can effectively enhance the heat transfer. F. Mônica. et. al, [8], examined non-Newtonian fluids heat transfer to flow through rectangular ducts. They solved numerically the three basic equations; they obtain the optimal combinations of aspect ratio and Reynolds numbers, depending on mechanical behavior of the fluids. S. S. Pawar, et. al, [9], studied experimentally Newtonian and non-Newtonian fluids flow under isothermal steady state and non-isothermal unsteady state conditions. The experiments performed for helical coil in laminar and turbulent flows. It was found that overall heat transfer coefficient and Nusselt numbers for Newtonian are higher than non-Newtonian fluids. They observed that when helix diameter increases, overall heat transfer coefficient and Nusselt numbers of both fluids decreases. M. Shojaeian and A. Kosar, [10], investigate the flow between parallel plates for newtonian and non-newtonian fluids with fully developed laminar flows. They solved analytically the governing equations. The effects of slip coefficient, power-law index and viscous heating were examined on the heat transfer hydrodynamic characteristics. Key parameters effect on nusselt number and the rate of entropy generation is more pronounced for shear-thickening fluid. T. A. Pimenta, J. B. L. M. Campos, [11], achieved an experimental work to obtain heat transfer coefficients for Newtonian and non-Newtonian fluids. The fluid flow is fully developed laminar flow inside a helical coil at constant wall. The non-Newtonian fluids had shear thinning behavior and different values of elasticity. The Nusselt numbers of non-Newtonian fluids are, on average, slightly higher than those for Newtonian fluids. Zhenbin He, et. al., [12] investigated an experimental and numerical study for an aqueous solution non-Newtonian fluid. They studied the heat transfer and flow resistance characteristic in the vertical heat exchanger combined helical baffles with elliptic tubes. Experimental work achieved to prove the numerical simulation model accuracy. They conclude that the heat transfer rate of the elliptic tube increases with the Reynolds number increase and at the same Reynolds number; the heat transfer rate of the helical baffle heat exchanger with elliptic tubes is higher than that of the heat exchanger with circular tubes.

2. Theoretical Analysis

The solutions of basic equations depend on initial and boundary conditions. There are some assumptions can be made:

- Steady state fluid.

- The fluid flow is laminar.

- The flow pattern is rectilinear.

-Specific heat of fluid (c_p)($J / Kg^{\circ}C$), Thermal conductivity (k)($W / m^2\,^{\circ}C$) and fluid density (ρ)(Kg / m^3) are independent of temperature and pressure.

- Fully developed flow.

- Uniform flow.

- Constant wall temperature.

- No conduction heat transfer in radial and axial directions.

The continuity, momentum and energy equations formulated in cylindrical coordinates, as shown in figure (1) below:

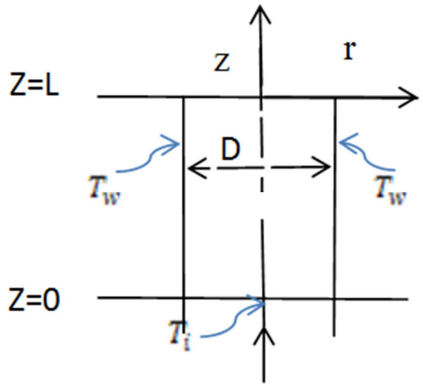


Figure 1. Problem Formulation.

Continuity equation:

$$\frac{\partial u_z}{\partial z} = 0 \ (\rho = \text{constant}) \qquad (1)$$

Where; (u_z) ($m / \sec$) is the axial velocity and (z) is the axial coordinate

Momentum equation:

$$-\frac{\partial P}{\partial z} = \frac{1}{r}\frac{\partial}{\partial r}(r\tau_{rz}) \qquad (2)$$

Where; (p)(N / m^2) is the pressure, r is the radial coordinate and (τ_{rz})(N / m^2) is the shear stress.

And Energy equation:

$$c_p\, u_z \rho \frac{\partial T}{\partial z} = \frac{k}{r}\frac{\partial}{\partial r}\left(\frac{r\partial T}{\partial r}\right) - \tau_{rz}\frac{\partial u_z}{\partial r} \quad (3)$$

Where; (T)($^\circ C$) is the temperature.

Also, the governing equations in dimensionless form are:

$$2\pi\rho u_m {R_w}^2 \int_0^1 RVdR = \pi {R_w}^2 \rho u_m = \dot{m} \quad (4)$$

$$\left(-k/\rho u_z {R_w}^2 c_p\right)\frac{\partial p}{\partial Z} = \frac{1}{R}\frac{\partial}{\partial R}(R\tau_{rz}) \quad (5)$$

Where; (u_m)($m/\sec$) is the mean velocity, (R_w)(m) is the tube radius and ($R = r/R_w$) is Dimensionless radial coordinate.

$$\text{And } V\frac{\partial \theta}{\partial Z} = \frac{1}{R}\frac{\partial}{\partial R}\left(\frac{R\partial\theta}{\partial R}\right) - \left(\frac{R_w u_m}{kT_i}\right)\frac{\partial V}{\partial R}\tau_{rz} \quad (6)$$

Where; ($V = u_z/u_m$) is the dimensionless axial velocity, (T_i)($^\circ C$) is the Inlet temperature and ($\theta = (T - T_i)/T_i$) is the dimensionless temperature; Respectively.

Equation (6) solved through substitution of an appropriate constitutive equation into equation (5) with chosen initial and boundary conditions.

The temperature-dependent power law relationship to describe rheological properties is:

$$\tau = K(T)\,\dot{\gamma}^n \quad (7)$$

Where; (τ)(N/m^2) is the shear stress; (K)($Pa_\bullet s^n$) is the consistency index, ($\dot{\gamma}$)(s^{-1}) is the shear rate and (n) is the Flow behaviour index for power law fluids.

If K and μ independent of temperature, the solution of equation (2) is:

$$u = u_m \frac{2n+1}{n+1}\left[1 - R^{(n+1)/n}\right] \quad (8)$$

Where; (μ)($Kg/m\ \sec$) is the dynamic viscosity and (u)($m/\sec$) is the axial velocity.

$$\text{And } \log K(T) = A + BT + CT^2 + DT^3 + \cdots \quad (9)$$

Where; ($A, B, C \text{ and } D$) are the constants characterising the viscosity temperature dependence.

By rearranging (9) and inserting into (7), we obtain constitutive equation:

$$\tau_{rz} = 10^{\emptyset(T)}\,\dot{\gamma}^n, \left[\varphi(T) = A + BT + CT^2 + DT^3 + \ldots\ldots\right] \quad (10)$$

Also in dimensionless temperature θ then:

$$\emptyset(\theta) = A + BT_i(\theta+1) + CT_i{}^2(\theta+1)^2 + DT_i{}^3(\theta+1)^3 + \cdots \quad (11)$$

Since $\frac{\partial P}{\partial z} = 0$ then equation (2) becomes after integrated it:

$\tau_{rz} = -\frac{r}{2}\frac{dP}{dz}$, also for shear stress wall; $\tau_w = -\frac{R_w}{2}\frac{dP}{dz}$; therefore:

$$\tau_{rz} = \frac{r}{R_w}\tau_w \quad (12)$$

Substitute (12) into (11); then

$$\tau_{rz} = 10^{\emptyset(\theta)}\,\dot{\gamma}^n = (r\,\tau_w/R_w)^{1/n}\,1/10^{\emptyset(\theta)} \quad (13)$$

Also $\dot{\gamma} = f(\tau_{rz})$ and $u_z = \int_r^{R_w} f(\tau_{rz})dr$; therefore:

$$u_z = \int_r^{R_w} (r/R_w)^{1/n}\,(\tau_w)^{1/n}\,1/10^{\emptyset(\theta)}\,dr \quad (14)$$

Where; (τ_w) is the wall shear stress.

For $R = r/R_w$, then (14) becomes:

$$u_z = R_w(\tau_w)^{1/n}\int_R^1 (R)^{1/n}\,1/10^{\emptyset(\theta)}\,dR = R_w(\tau_w)^{1/n} I_1 \quad (15)$$

Where; ($I_1 = \int_R^1 (R)^{1/n}\,1/10^{\emptyset(\theta)}\,dR$)

But $\dot{m} = 2\pi\rho\int_0^{R_w} r u_z dr = 2\pi\rho {R_w}^2 \int_0^{R_w} R(\tau_w)^{1/n} R_w I_1 dR = 2\pi\rho {R_w}^3 (\tau_w)^{1/n} I_2$

$$\text{Therefore } u_z = \dot{m}/[2\pi R^2 \rho(I_1/I_2)] \quad (16)$$

Where; ($\dot{m}$) is the fluid mass flow rate and ($I_2 = \int_0^1 I_1 dR$).

Equation (16) is the fluid velocity distribution; and can be represented as dimensionless group as

$V = u_z/u_m$ where $u_m = 2I_2$; then:

$$V = I_1/2I_2 \quad (17)$$

Now for fluid temperature distribution; heat transfer radially through conduction and along tube through convection; insert equation (17) into equation (6) yields:

$$I_1/2I_2\frac{\partial\theta}{\partial Z} = \frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial\theta}{\partial R}\right) - \tau_{rz}(R_w u_m/kT_i)\frac{\partial V}{\partial R} \quad (18)$$

Substitute wall shear stress τ_w and fluid velocity gradient $\frac{\partial V}{\partial R}$ to obtain:

$$I_1/2I_2\frac{\partial\theta}{\partial Z} = \frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial\theta}{\partial R}\right) + \frac{{u_m}^{n+1} R^{(n+1)/n}}{kT_i {R_w}^{n-1} {I_2}^{n+1} * 2^{n+1} * (1/10^{\emptyset(\theta)})^{1/n}} \quad (19)$$

For viscous dissipation; (Br) Brinkman number defined as:

$Br = {u_m}^2\,\mu(T_i)/kT_i \ldots$ For Newtonian fluids

$Br = {u_m}^{n+1}\,K(T_i)/kT_i\,{R_w}^{n-1}$ … For Non-Newtonian fluids

Therefore energy equation becomes:

For Newtonian fluids;

$$(I_1/2I_2)\frac{\partial\theta}{\partial Z} = \frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial\theta}{\partial R}\right) + BrR^2/2^2\,{I_2}^2 * 10^{\emptyset(\theta)}\mu(T_i) \quad (20)$$

For Non-Newtonian fluids;

$$(I_1/2I_2)\frac{\partial\theta}{\partial Z} = \frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial\theta}{\partial R}\right) + BrR^{n+1}/2^{n+1}\,{I_2}^{n+1}K(T_i)(1/10^{\emptyset(\theta)})^{1/n} \quad (21)$$

Initial and boundary conditions require to solve equation (20) to obtain $\theta = \theta(R,Z)$ are:

Z=0; $0 \le R \le 1$; $\theta = 0$ and $V = 2(I - R^2)$

Z≥0; $R = 1$; $\theta = \theta_w = (T_w - T_i)/T_i = \emptyset - 1$ and $V = 0$

Z≥0; $R = 0$; $\frac{\partial\theta}{\partial R} = 0$ and $\frac{\partial V}{\partial R} = 0$

Initial and boundary conditions require to solve equation (21) to obtain $\theta = \theta(R,Z)$ are:

Z=0; $0 \le R \le 1$; $\theta = 0$ and $V = (3n+1)/(n+1)[1 - R^{(n+1)/n}]$

Z≥0; $R = 1$; $\theta = \theta_w = \emptyset - 1$ and $V = 0$

Z≥0; $R = 0$; $\frac{\partial\theta}{\partial R} = 0$ and $\frac{\partial V}{\partial R} = 0$

Also $\emptyset = T_w/T_i$ and $\emptyset > 1$ for heating; $\emptyset < 1$ for cooling; where (T_w) is the wall temperature.

The mean Nusselt number (Nu_m) is:

$$Nu_m = (1/L)\int_0^L Nu_z dz \quad (22)$$

Where (Nu_m) is the Mean Nusselt number.

And the energy equation can be written as:

$$\dot{m}c_p(T_o - T_i) = h_m\pi DL[T_w - 1/2\,(T_i + T_o)] \quad (23)$$

Where; (T_o) is the mean outlet temperature, (h_m) is the mean heat transfer coefficient, (D) is the Tube

Diameter and (L) is the Length of the tube.

But $T_o = \int_0^{R_w}\frac{u_z T(r,L) r dr}{u_z r dr}$

By substituting $\theta = (T - T_i)/T_i$; $\theta_o = (T_o - T_i)/T_i = \int_0^1 (I_1/I_2)\theta R dR = I_3$; $u = u_m I_1/2I_2$ then:

$$Nu_m = (Gr/\pi)[\theta_o/(\theta_w - \theta_o/2)] = (Gr/\pi)[I_3/(\theta_w - I_3/2)] \quad (24)$$

Where; (Gr) is the Graetz number (θ_o) is the dimensionless mean outlet temperature and (θ_w) is the dimensionless wall temperature.

Pressure drop $\frac{dp}{dz}$ can be written as:

$$\frac{dP}{dz} = (u_m/2^{n-1}){R_w}^{n+1}{I_2}^n \quad (25)$$

By integration of equation (25) yields:

$$P_i - P_z = \frac{1}{2^{n-1}}\int_0^Z \frac{dz}{{I_2}^n} \quad (26)$$

Where;

In dimensionless form;

$$P = p{R_w}^{n-1}k/{u_m}^{n+1}c_p \quad (27)$$

3. Experimental Work

Experimental work leads to determine fluid velocity and temperature distributions and measurements both of overall heat transfer coefficients and pressure drop. Physical and thermal properties for fluids like density, specific heat, thermal conductivity were achieved as shown in table (1). Figure (2) shows schematic chart for heat transfer loop. The fluid was pumped from the reservoir tank (1) into the test sections where the fluid either heated or cooled to a constant wall temperature. Tank storage tank (2) equipped with a $(100\,rpm)$ agitator contains two copper coils placed concentrically in it used for either heating or cooling the test fluid with required conditions. The fluid flow rate from the tank (1) measured by flow rate gage (3). Tank (4) similar to tank (2) used as a storage tank for supplying hot water to the heating coil in tank (2). A pump (5) used to circulate the working fluids. Another pump (6) supplied the heating water from the tank (4) into the tank (2). The heat exchanger (7) (heat transfer section test) consisted of two concentric pipes. To measuring wall temperature of the heat transfer section, the thermocouples (8) were used. The thermocouples placed in the upper and lower sections of the jacket to measure the temperatures of the heating water in the jacket of the exchanger (3) and that of the cooling water. By using tappings at the test, the pressure drop can be determined. Pressure gauge used to measure the pressure inside the jacket of the heat exchanger. At the exit of the heat transfer section into the measurement unit (9), Pitot tube (10) and thermocouple (11) used to measure the local velocities and temperatures. A small hole drilled in the wall used to measure static pressure located at the opposite side of Pitot tube probe. The differential pressure obtained by the difference between a pressure gage reading from the Pitot tube and that from the wall static tap.

Firstly, the fluid mixed and heated to a predetermined temperature in the tank (2), then flowed at a large flow rate. At the same time, heating or cooling water supplied to the Jacket of the test section to obtain constant wall temperatures through governing of pressure from (3 to 1 bar). Steady state achieved when all temperatures remain constant for 30 minutes. Velocity and temperature profiles determined Pitot tube and thermocouple situated at the exit section.

Table 1. Physical and thermal properties for fluids.

Property	Newtonian	Non-Newtonian
Specific heat	$2.2\ at\ 50^{\circ}C$ $2.5\ at\ 90^{\circ}C$	$2.18\ at\ 20^{\circ}C$
Thermal conductivity	$0.143\ at\ 50^{\circ}C$ $0.145\ at\ 80^{\circ}C$	$0.304\ at\ 20^{\circ}C$

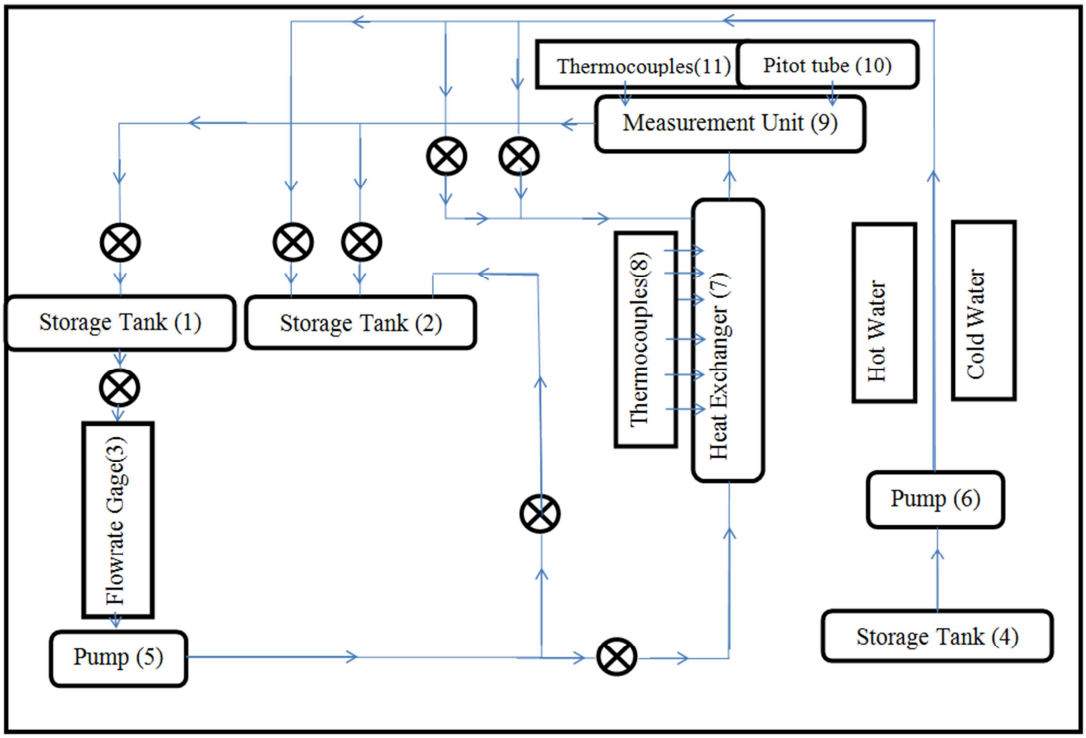


Figure 2. Schematic chart for heat transfer loop.

4. Results and Discussion

Experimental and theoretical works will be compared. When (T_w/T_i) is greater than one (heating), a temperature gradient established between the fluid with a high temperature at the wall and a low temperature at the axis. Therefore we obtain a viscosity gradient between a low viscosity at the wall and a high viscosity at the axis depending on the relation between temperature and viscosity. Viscosity gradient arise to velocity gradient, velocity at wall greater than velocity at tube middle. The observed velocity distribution becomes much flatter than the parabolic one.

For Newtonian fluids, the distortion of the parabolic velocity profile depends on the value of the ratio (T_w/T_i) and on the axial distance. we note figures (3a) and(3b) when $(T_w/T_i) = 1.3$ at $90^{\circ}C$, radial temperature difference of the dimensionless maximum velocity has a value of (1.17) and it remains constant up to the dimensionless radial position of $((r/R_w) = 0.85$, which means uniform flow. figures (3-5) show that as this ratio (T_w/T_i) decreases the departure from the parabolic profile decreases. figure (9) shows the development of the velocity profiles along the heat transfer section under different conditions. The maximum deviation from the isoviscous case occurs at high values of the Graetz number (Gz) and this deviation diminishes gradually until it may be neglected at values of Graetz number (Gz) less than 2. When (T_w/T_i) is less than one (cooling), figures (6-8), the velocity profiles are much sharper than the parabolic velocity distribution since viscosity near the wall increases giving lower velocity gradients at the wall and higher centre line velocities. If (T_w/T_i) decreases the departure of the isoviscous distribution increases and vice versa. We note from figure (9a) that the velocity profiles for cooling are more influenced by the axial distance than the velocity profiles for heating.

Temperature distributions profiles obtained at the exit of the heat transfer section are plotted as dimensionless temperature, (θ), (positive for heating and negative versus ratio (r/R_w) for different Graetz numbers and ratios (T_w/T_i) in figures (3-8). There are high changes in the temperature of

radial point comparison with the isoviscous values. The temperature distributions are flat until (r/R_w) is greater than (0.6) for cooling or (0.7) for heating depending on Graetz number. The temperature gradients at the wall are smaller than in the isoviscous case and therefore the rate of the heat transfer is decreased.

The arithmetic mean Nusselt number is dependent on the the mean outlet temperature (T_0) value and consequently on the Graetz number, For high Graetz numbers the outlet temperature (T_0) is very near to the inlet temperature (T_i) due to the small amount of heat transferred to the fluid. Also for small Gractz numbers the (T_0) is near to (T_w) where Graetz number value depends upon the thermal properties of the fluid $(c_p, \rho, Q$ and $L)$. The Nusselt number increases with raising the temperature ratio (T_w/T_i) and the Graetz number.

For pressure drop, the values of the variables (T_w/T_i) and (Gz) were the same as those used to measure the velocity and temperature profiles and the heat transfer rates. It is noted that the pressure drop are dependent on the Graetz number, as the Graetz number decreases the departure from the isoviscous case gradually decreases and diminishes at small Graetz number as shown in figure (10).

For Non-Newtonian fluids, the changes which occurred on both temperature and velocity profiles are similar to those of the Newtonian oil. Figures (11-14) show that the deviation from the isoviscous solution is almost the same as that with the Newtonian oil. Figure (15), similar to figure (9) give the center line velocities as a function of the Graetz number, showing the development of the velocity profiles during the heat transfer process. For the arithmetic mean Nusselt numbers. For non-Newtonian fluid, the deviation from the isoviscous case is greater in character, the decrease in heat transfer coefficients is much greater than in the Newtonian flow, because that the deviations in velocity gradient at the wall from the isoviscous gradient are more appeared as non-Newtonian behaviour increases. For the pressure drop, the non-isoviscous pressure drop deviates from the isoviscous pressure drop but this deviation is not as great as that in the case of the Newtonian oil. Figure (16) shows the relation of the ratio of the non-isoviscous pressure drop and the isoviscous pressure drop with a consistency index evaluated at wall temperature of solution against the Graetz number and the temperature ratio. For heating and cooling, the deviation from the isoviscous case is greater as the temperature ratio increases and decreases respectively.

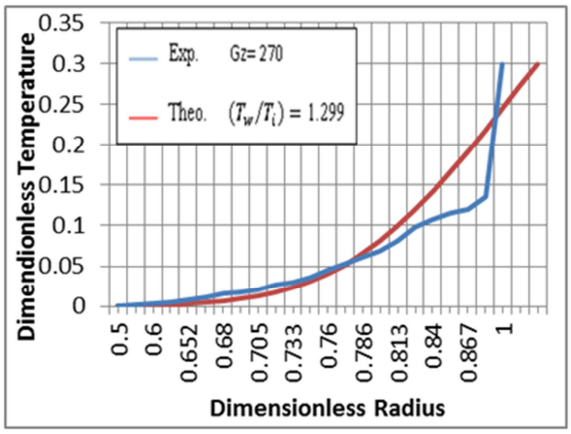


Figure 3a. Newtonian temperature distributions (Heating).

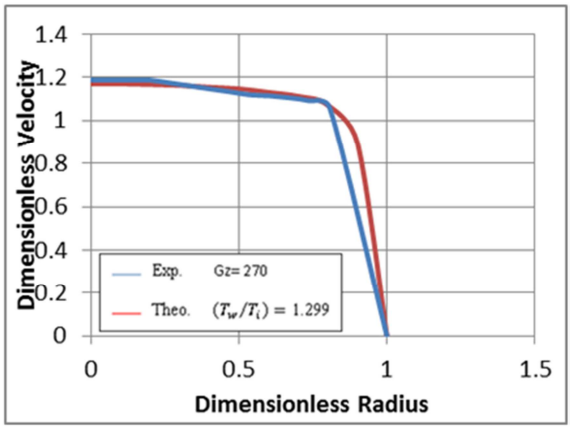


Figure 3b. Newtonian Velocity distributions (Heating).

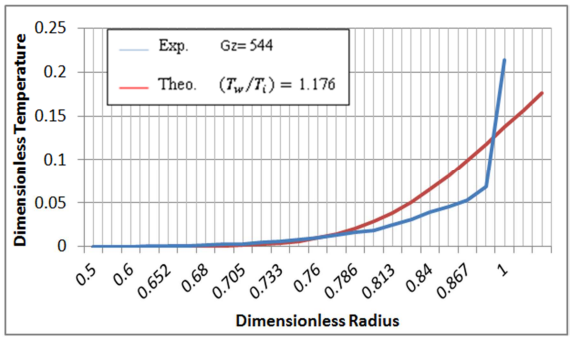


Figure 4a. Newtonian temperature distributions (Heating).

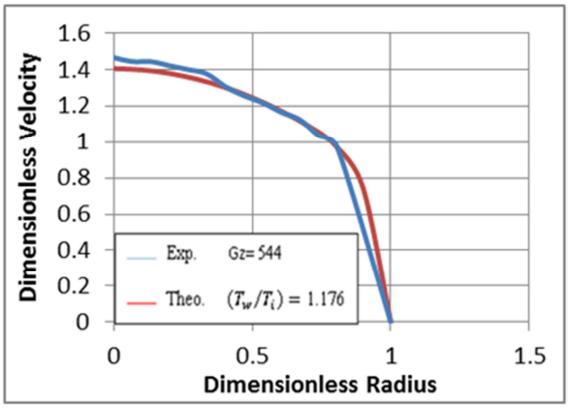


Figure 4b. Newtonian Velocity distributions (Heating).

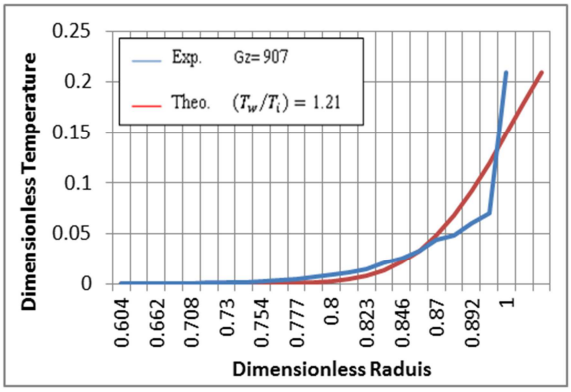


Figure 5a. Newtonian temperature distributions (Heating).

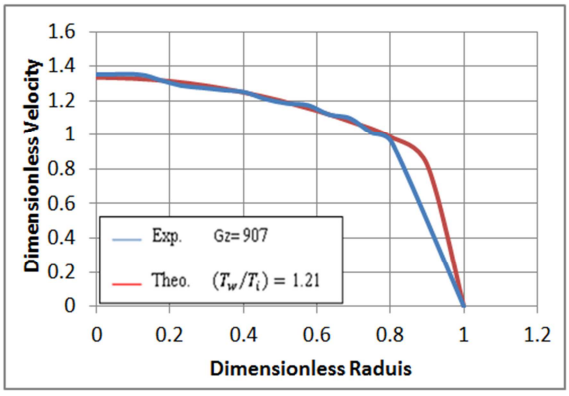


Figure 5b. Newtonian Velocity distributions (Heating).

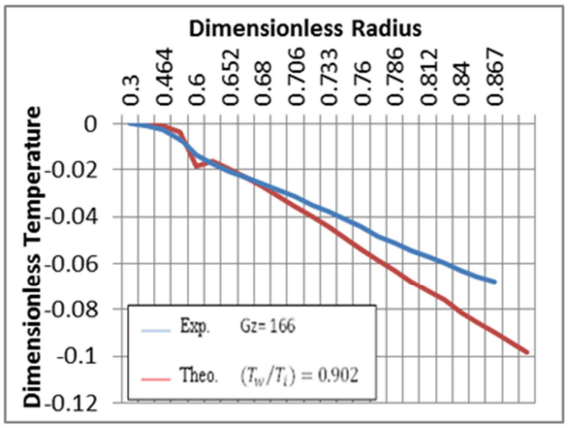


Figure 6a. Newtonian temperature distributions (Cooling).

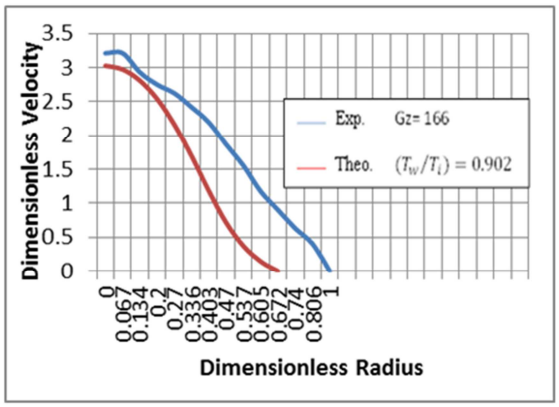


Figure 6b. Newtonian Velocity distributions (Cooling).

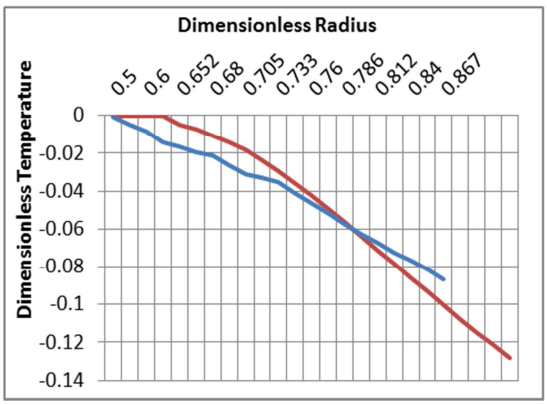


Figure 7a. Newtonian temperature distributions (Cooling).

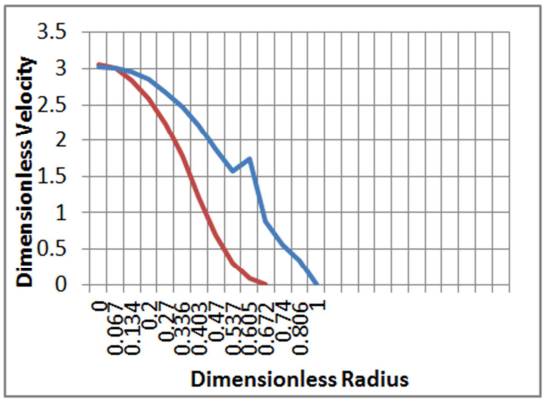


Figure 7b. Newtonian Velocity Distributions (Cooling).

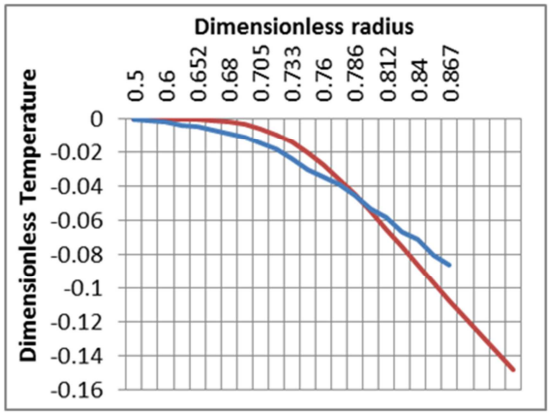


Figure 8a. Newtonian temperature distributions (Cooling).

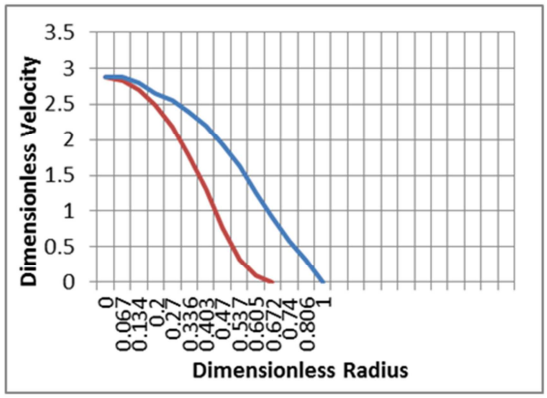


Figure 8b. Newtonian Velocity Distributions (Cooling).

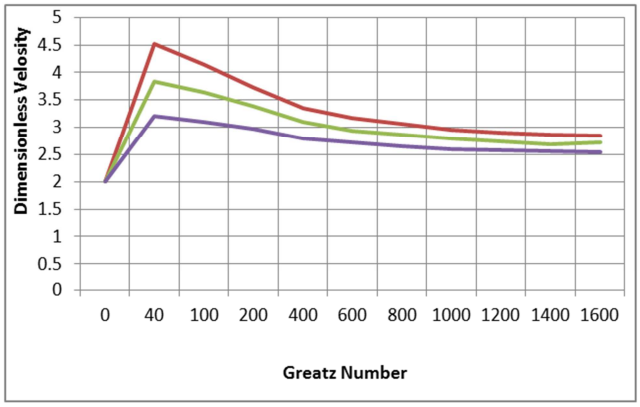


Figure 9a. Newtonian Velocity distributions (Cooling).

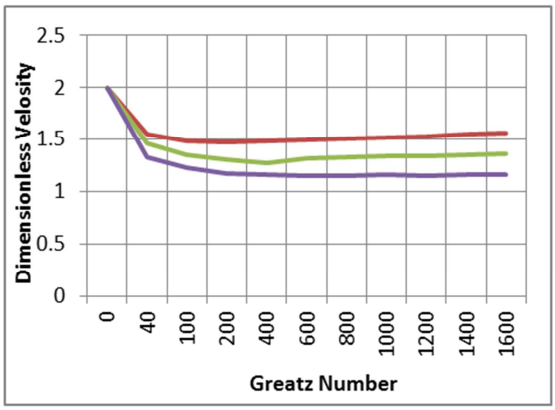


Figure 9b. Newtonian Velocity distributions (Heating).

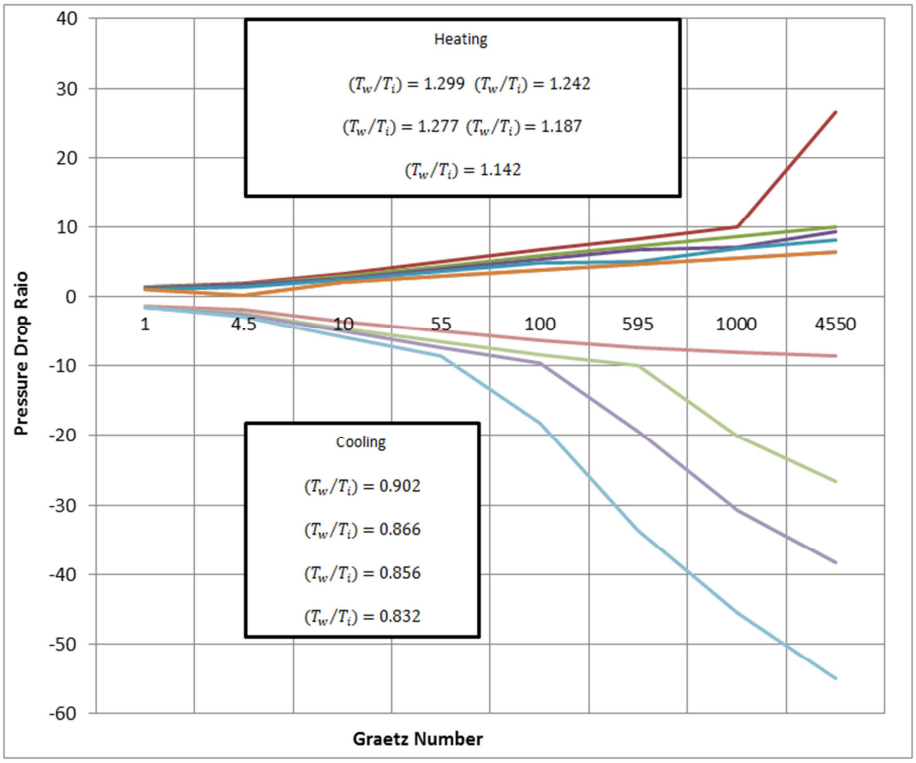
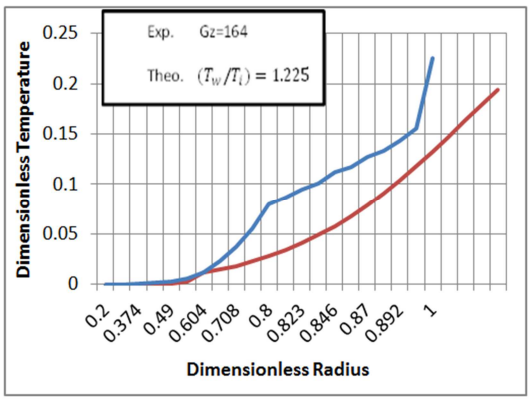


Figure 10. Pressure Drop for Different Temperature Ratios.



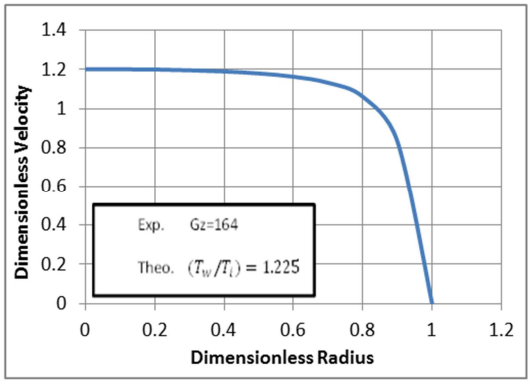


Figure 11a. Non-Newtonian Temperature Distributions (Heating). **Figure 11b.** Non-Newtonian Velocity Distributions (Heating).

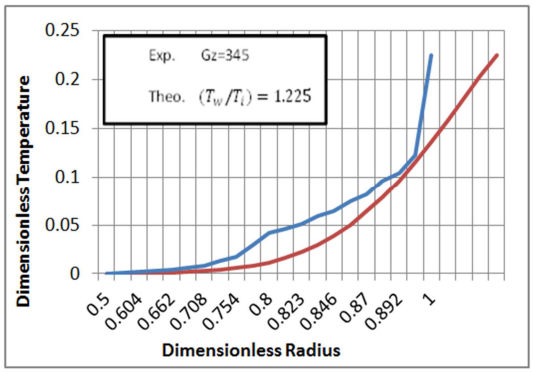


Figure 12a. Non-Newtonian Temperature Distributions (Heating).

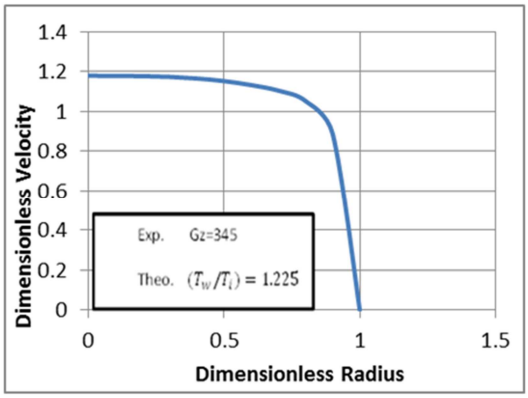


Figure 12b. Non-Newtonian Velocity Distributions (Heating).

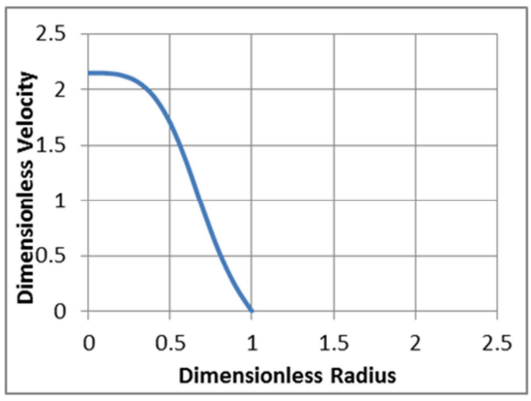


Figure 13a. Non-Newtonian Temperature Distributions (Cooling).

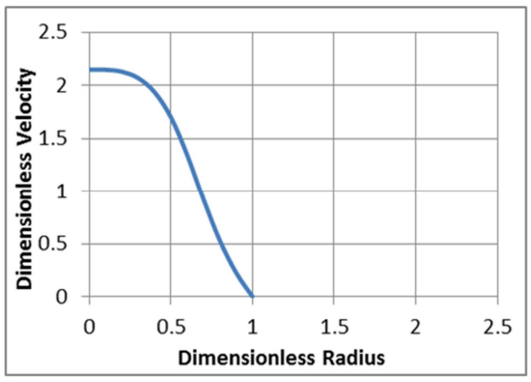


Figure 13b. Non-Newtonian Velocity Distributions (Cooling).

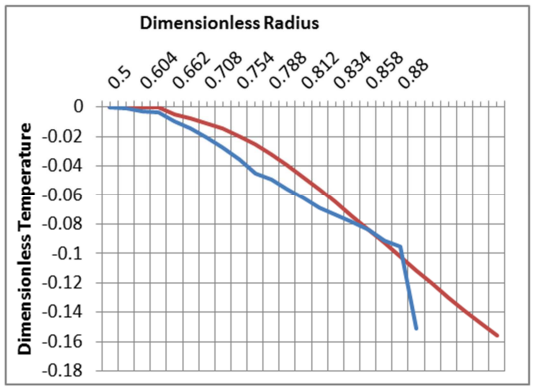


Figure 14a. Non-Newtonian Temperature Distributions (Cooling).

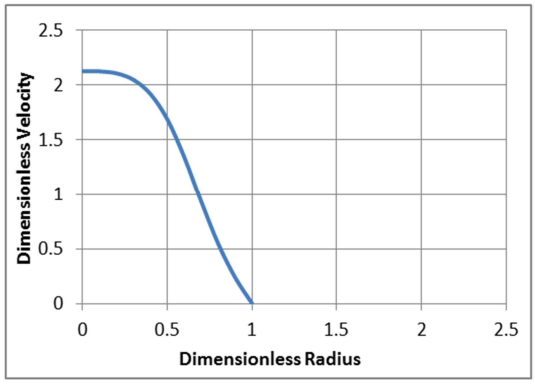


Figure 14b. Non-Newtonian Velocity Distributions (Cooling).

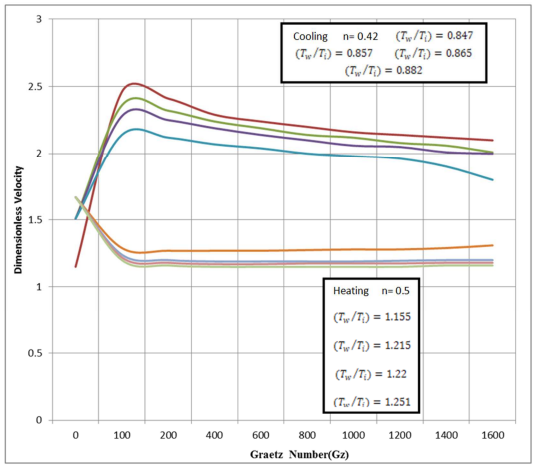


Figure 15. Center velocity variation with temperature gradient.

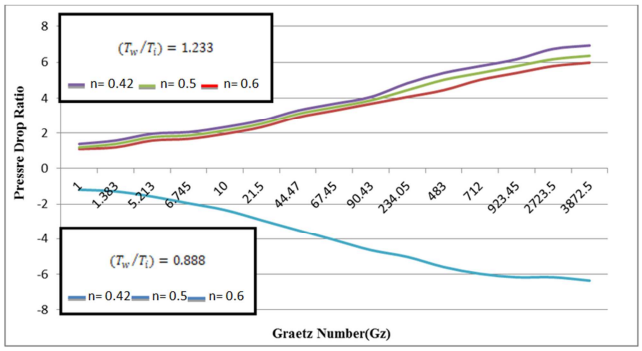


Figure 16. Pressure drop ratio variation with power law index.

5. Conclusions

a- The viscosity-temperature relationship nature is important to prediction of the hydrodynamics and heat transfer rates of fluids, especially if they are non-Newtonian.

b- The dimensionless centre axis velocities change by a factor of (2), in comparison with the isoviscous solution.

c- Temperature profiles for Newtonian fluid deviate from those for Non-Newtonian fluid. For heating, larger gradient of temperature and for cooling, smaller gradient of temperature are noted for Newtonian fluid comparing with Non-Newtonian fluid.

d- There is large deviation for pressure drop of Newtonian fluid from Non-Newtonian fluid.

e- Nusselt numbers for Newtonian fluid are higher at heating while they are lower for cooling.

f- All above differences between Newtonian and Non-Newtonian fluids depend on Graetz number values and the thermal conditions used.

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