



# Analytical and experimental studies of power-law fluids in double-pass heat exchangers for improved device performance under uniform heat fluxes

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## ABSTRACT

Considerable improvement of heat transfer efficiency for power-law fluids in flat-plate heat exchangers is obtainable by inserting in parallel an impermeable resistless sheet to divide an open duct into two channels for conducting double-flow operations. The influence of the power-law index on average Nusselt numbers in double-pass parallel-plate heat exchangers under uniform wall fluxes has been investigated theoretically. The complete solution was obtained using an eigenfunction expansion technique in terms of power series for the homogeneous part and an asymptotic solution for the non-homogeneous part. The theoretical predictions show that the effect of double-pass operations for power-law fluids results in the heat-transfer efficiency enhancement as compared with those in an open conduit (without an impermeable resistless sheet inserted). The effects of impermeable-sheet position and power consumption increment for power-law fluids have also been presented.

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## 1. Introduction

Most of the interest devoted to the heat transfer in engineering applications is the study of the thermal response of the conduit wall and fluid temperature distributions to the Newtonian fluid. The problem of laminar heat transfer in bounded conduits at steady state with negligible axial conduction is known as the Graetz problem [1,2]. It should be noted that many materials of high molecular weight encountered in industrial practice (food, polymeric systems, biological process, pulp and paper suspensions, etc.) [3], which exhibit a range of non-Newtonian fluid behavior features and display shear-thinning and/or shear thickening behavior [4,5]. Those non-Newtonian materials are generally processed in laminar flow conditions with negligible viscoelastic effects due to their high viscosity levels. Restated, the flow of purely viscous power-law type fluids with the power-law constants in the appropriate shear rate range is evaluated in pursuing analytical solutions.

A new device in aiming for increasing the fluid velocity and the heat transfer area but keeping the aspect ratio unchanged was designed to insert an impermeable sheet between the parallel plates to conduct a double-pass operation. In addition, the concept of recycle-effect was implemented in such a double-pass device with external recycle to improve the heat or mass transfer efficiency in absorption, fermentation and polymerization such as distillation

[6], extraction [7], adsorption [8], mass diffusion [9], thermal diffusion [10], loop reactors [11], air-lift reactor [12] and draft-tube bubble column [13], which are widely used in separation processes and reactors design. The mathematical modeling of double-pass operations results in a system of partial differential equations, as referred to conjugated Graetz problems [14–16], was solved analytically by means of orthogonal expansion techniques [17–20].

The present study is an extension of our previous work [21] to apply the case of the Neumann boundary condition for the conjugated Graetz problem of which the heat fluxes at the walls were specified. There are many studies [22,23] devoted to the heat transfer problem in a parallel-plate channel with uniform heat flux (Neumann problem). The heat transfer efficiency of multi-stream or multiphase problems coupling mutual conditions at the interface in double-pass operations, as compared that to the single-pass operation, was analyzed in a straightforward manner lumping with the wall Nusselt number. The simplifying mathematical model for power-law fluids was investigated theoretically and could be utilized to the design double-pass operations with power-law index as a parameter. The purposes of the present study are: (a) to obtain the wall temperature distribution in the axial direction under uniform wall heat fluxes based the superposition technique; (b) to study the device performance improvement in double-pass parallel-plate laminar countercurrent heat exchangers; (c) to discuss the influence of the power-law index and impermeable-sheet location on the heat-transfer efficiency enhancement. The solution methodology developed here in a simplified form of a double-pass parallel-plate heat exchanger for power-law fluids is accomplished and could be applied to the

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**Nomenclature**

$B$	conduit width, m	$X_i$	coefficients defined in Eq. (4)
$c$	consistency index of power-law model, $\text{Pa s}^\omega$	$x$	longitudinal coordinate
$c_p$	specific heat at constant pressure, $\text{J/g K}$	$z$	transversal coordinate
$d_{mn}$	coefficient in the eigenfunction $F_{a,m}$	<b>Greek symbols</b>	
$e_{mn}$	coefficient in the eigenfunction $F_{b,m}$	$\alpha$	thermal diffusivity of fluid, $\text{m}^2/\text{s}$
$f(\eta_i)$	function defined by Eq. (4)	$\alpha_1$	constant defined by Eq. (15)
$F_m$	eigenfunction associated with eigenvalue $\lambda_m$	$\alpha_2, \alpha_3$	integration constants in Eq. (17)
$G_z$	Graetz number, $VW/\alpha BL$	$\beta_1$	constant defined by Eq. (16)
$G_m$	function defined by Eq. (25)	$\beta_2, \beta_3$	integration constants in Eq. (18)
$G_z$	function defined by Eq. (15)	$\dot{\gamma}$	shear rate
$H_b$	function defined by Eq. (16)	$\Delta$	impermeable-sheet position, $W_a/W$
$\bar{h}$	average heat transfer coefficient, $\text{kW/m K}$	$\eta$	dimensionless longitudinal coordinate, $x/W$
$I_h$	heat transfer enhancement based on single-pass devices, defined by Eq. (48)	$\theta$	defined by Eq. (9)
$I_p$	power consumption increment, defined by Eq. (50)	$\phi$	defined by Eq. (9)
$k$	thermal conductivity of the fluid, $\text{kW/m K}$	$\lambda_m$	eigenvalue
$L$	conduit length, m	$\delta$	impermeable-sheet thickness, m
$\ell w_h$	friction loss in conduit, $\text{kJ/kg}$	$\xi$	dimensionless transversal coordinate, $z/LG_z$
$n$	the terms of an extended power series in Eqs. (35) and (36)	$\theta$	dimensionless temperature, $k(T - T_i)/q''W$
$\bar{Nu}$	the average Nusselt number defined by Eq. (46)	$\tau$	shear stress
$P$	Hydraulic dissipated energy,	$\omega$	power-law index
$q''$	heat transfer rate, $\text{kW}$	$\rho$	density of the fluid, $\text{kg/m}^3$
$s$	function defined by Eq. (49)	<b>Subscripts</b>	
$S_m$	expansion coefficient associated with eigenvalue $\lambda_m$	0	at the inlet or for the single pass
$T_i$	inlet temperature of fluid in conduit, $\text{K}$	$a$	in forward flow channel
$V$	input volumetric flow rate of conduit, $\text{m}^3/\text{s}$	$b$	in backward flow channel
$v$	velocity distribution of fluid, $\text{m/s}$	$F$	at the outlet of a double-pass device
$\bar{v}$	average velocity of fluid, $\text{m/s}$	$L$	at the end of the channel
$W$	conduit height, m	$w$	at the wall surface

heat- and mass-transfer problem of any arbitrary wall flux distribution with more general geometry.

**2. Theoretical formulations**

Consider the heat transfer of parallel-plate heat exchangers with height  $W$ , length  $L$ , and width  $B$  ( $\gg W$ ) with inserting an impermeable sheet of negligible thickness ( $\delta \ll W$ ) and thermal resistance into a parallel conduit to conduct a double-pass operation, subchannel  $a$  and subchannel  $b$ , with thickness  $\Delta W$  and  $(1 - \Delta)W$ , as shown in Fig. 1. Under this design condition, the fluid with volumetric flow rate  $V$  can first flow through subchannel  $a$  and then turn back to flow through subchannel  $b$  with the aid of a conventional pump.

**2.1. Temperature distribution in a double-pass operation**

After the following assumptions were made: fully-developed laminar flow with power law index  $\omega$   $\tau = -c\dot{\gamma}^\omega$  in each subchannel, constant physical properties of fluid, neglecting end effects and impermeable sheet thermal resistance, ignoring the entrance length and axial heat conduction, well-mixing at the inlet and outlet of each subchannel; the energy balance equations of energy and the velocity distributions in dimensionless form of a double-pass heat exchanger under constant wall heat flux may be obtained as

$$\frac{\partial^2 \psi_i(\eta_i, \xi)}{\partial \eta_i^2} = \left( \frac{v_i(\eta_i) W_i^2}{\alpha L G_z} \right) \frac{\partial \psi_i(\eta_i, \xi)}{\partial \xi}, \quad i = a, b \quad (1)$$

$$v_i = \bar{v}_i \left[ \frac{1 + 2\omega}{1 + \omega} \right] \left( 1 - |2\eta_i - 1|^{\frac{1}{\omega+1}} \right), \quad i = a, b \quad (2)$$

in which

$$\begin{aligned} \bar{v}_a &= [V/W_a B], \quad \bar{v}_b = -[V/W_b B], \quad \xi = \frac{z}{LG_z}, \\ G_z &= \frac{VW}{\alpha BL}, \quad W_a = \Delta W, \quad W_b = (1 - \Delta)W, \\ \frac{W_a}{W_b} &= \frac{\Delta}{1 - \Delta}, \quad \psi_a = \frac{k(T_a - T_i)}{q''W}, \quad \psi_b = \frac{k(T_b - T_i)}{q''W}, \\ \eta_a &= \frac{x_a}{W_a}, \quad \eta_b = \frac{x_b}{W_b} \end{aligned} \quad (3)$$

The exponential term on the right-hand side in Eq. (2) was approximated using the fifth polynomial of Eq. (4) fitted at the points as shown in Appendix A as follows:

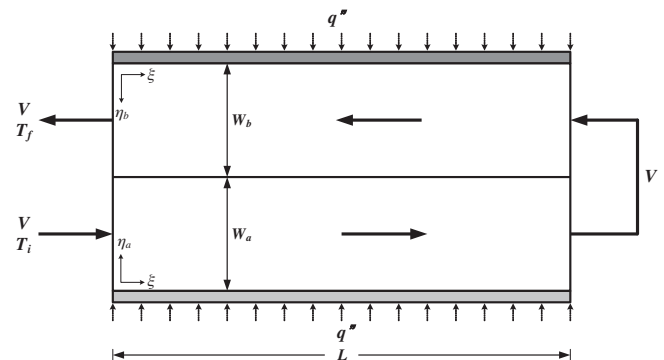


Fig. 1. The double-pass parallel-plate heat exchanger.

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