# Technical Notes

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## Novel Passive Thermal Mixer

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### Introduction

WHEN a liquid in a container is heated from below, it is well known that convection eventually takes place, causing mixing, and the entire liquid is heated up uniformly. On the other hand, if the liquid in a container is heated from the top, there is stratification; the top portion of the liquid becomes hot, but the bottom portion of the liquid remains cold. Even when the top portion of the liquid starts boiling, the bottom portion will still remain cold. One way to cause mixing is by mechanical stirring. This has been a major problem for the manufacturers of thermal storage tanks. From the point of view of good mixing and uniform heating, it is desirable to locate the heating coils at the bottom of the tank. However, in this configuration, it is not easy to replace a damaged coil without emptying the tank first. If, on the other hand, the heating coil is located at the top for ease of maintenance, the thermal performance is poor.

It is possible to improve the thermal performance of a heating tank with heating elements at the top by using a passive thermal mixer. This paper presents a novel concept for a passive thermal mixer. Proof of concept experiments have been successfully conducted, and the results are presented in this paper.

The proposed passive thermal mixer is shown in Fig. 1. The heating element located at the top is attached to the top lid of the heating tank. The heating element is enclosed by a cylindrical shroud extending to the bottom without touching it. Discharge tubes extending to the bottom are connected to the shroud, as shown in Fig. 1. The principle of operation is very simple. As the liquid surrounding the heating element is heated, pressure builds up locally, forcing the hot liquid down the discharge tubes. The hot liquid is then replaced by cold liquid entering from the bottom of the shroud. Thus, a convective process is set up, improving the thermal performance of the heating tank. The flow circulation through the mixer requires that the fluid pressure at the heating element be balanced by the static pressure due to the liquid level in the shroud and the pressure drop through the shroud. Hence, mixer geometry is expected to have a strong influence on the mixing process.

A large volume of research has been devoted to examination of natural convection in enclosures with differentially heated vertical walls; see, for example, Catton[1] and Ostrach.[2] Such research has been motivated by interest in predicting energy transfer in buildings, solar collectors, and fluid thermal storage systems. The natural-circulation solar water heater, which consists of a flat-plate collector, a storage tank, and connecting plumbing, is the most commonly used solar energy system. It has been investigated by Zvirin et al.[3] Gnafakis and Manno[4] studied transient destratification in a rectangular enclosure, in which a destabilizing heat source was introduced after the initial stratification.

Thermosiphons are natural circulation systems wherein density differences induced by heat transfer sustain the convective flow. Single-loop systems have been investigated by Mertol and Greif.[5] Multiple-loop thermosiphon networks are also used in many applications. Recent works include that of Chato,[6] Zvirin et al.,[7] Sen and Fernandez,[8] and Sen et al.[9] Salazar et al.[10] studied a loop system in which a number of natural circulation loops were physically separated, but thermally coupled through heat exchangers. Greif[11] presented a review of recent studies on loop thermosiphon systems. Yilmaz[12] investigated horizontal shell-side thermosiphon reboilers, which are used in the chemical processing industry to vaporize process fluids. The passive thermal mixer design investigated in this paper is unique in the sense that the shroud and discharge tubes that enclose the heating element are internal to the mixer tank.

### Experimental Program

Several mixer designs were tested in order to determine which configuration would yield the most uniform water temperature distribution. The mixer consisted of a central tube (shroud) constructed from polyvinylchloride (PVC). Four smaller PVC discharge tubes (arms) extended from the shroud. The arms were situated symmetrically about the axis of the central tube. Various shroud and arm lengths were considered.

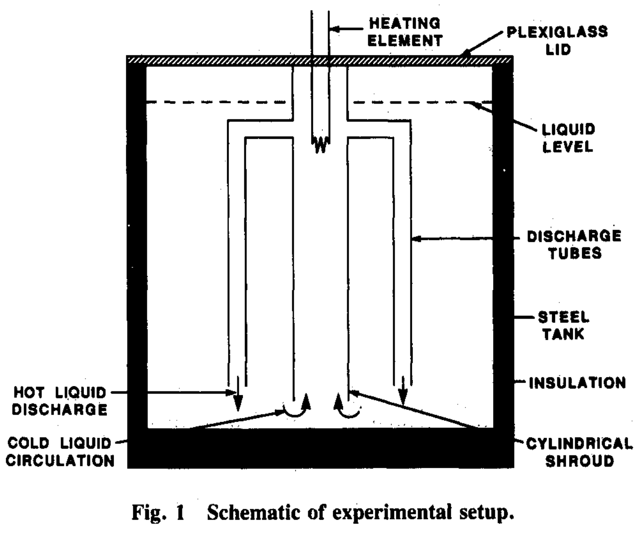


**Fig. 1 Schematic of experimental setup.**

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The experimental setup consisted of a steel tank 35.6 cm (14 in.) diameter and 38.1 cm (15 in.) deep. The lateral surface of the container was wrapped with 5.1 cm (2 in.) of glass-wool insulation. The tank rested on 5.1 cm (2 in.) thick insulation material. The top of the tank was covered by a plexiglass lid, thereby minimizing evaporation. The 1500 W heating coil was held by the plexiglass lid and projected approximately 7.6 cm (3 in.) below the water surface. The 7.6 cm (3 in.) inside diameter shroud was also attached to the plexiglass lid. Four discharge tubes, each 1.75 cm (11/16 in.) inside diameter and situated symmetrically, were attached to the shroud, as shown in Fig. 1. The horizontal distance from the center line of the shroud to that of the discharge tubes was 12.7 cm (5 in.). The plexiglass lid also held the guide wires used to support thermocouples located at various levels in the tank.

Water temperatures were measured by means of copper-constantan thermocouples, with expected measurement accuracies of $\pm 0.6°C$ ($\pm 1°F$). The thermocouples were placed at four depths in the tank, namely, 2.5, 12.7, 22.9, and 33.0 cm (1, 5, 9, and 13 in.). At each level, four thermocouples were arrayed symmetrically about the central axis of the mixer in order to determine the temperature at any depth. Each thermocouple signal was read by a Molytek 4702-4A1 data logger. The data logger was programmed to average the temperatures at each depth in the tank. It was found that the temperatures varied by no more than 0.8°C (1.5°F) at any level in the container. The RS-232 serial port of the data logger was used to route the data to a computer for analysis and plotting.

The heating element consisted of six 250 W electrical resistance heaters placed at the same depth as the inlets to the mixer arms. A voltage transformer and wattmeter allowed the power to the heaters to be accurately monitored and controlled.

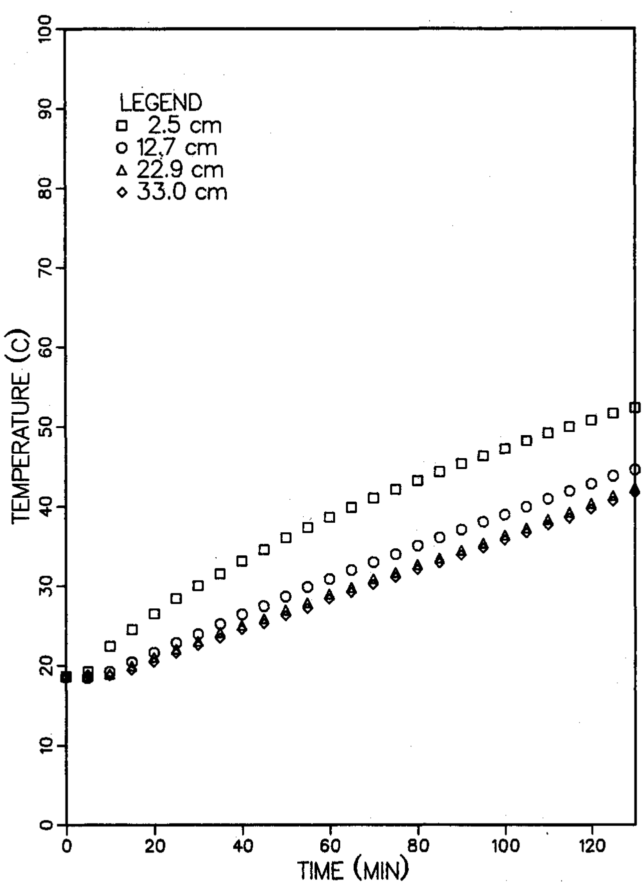


**Fig. 3 Average temperature as function of time and depth with mixer: $C$ = 26.7 cm; $L$ = 33 cm.**

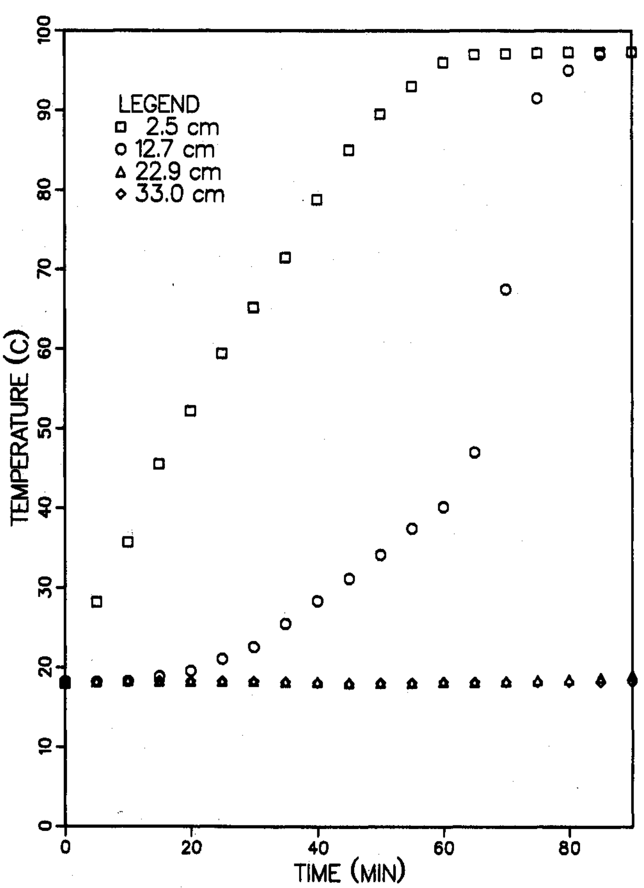


**Fig. 2 Average temperature as function of time and depth without mixer.**

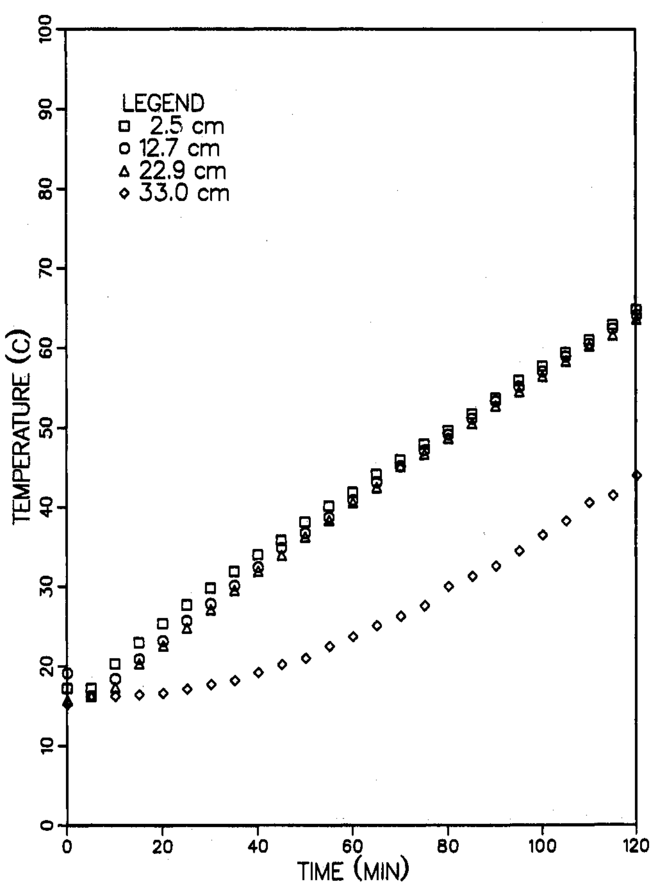


**Fig. 4 Average temperature as function of time and depth with mixer: $C$ = 26.7 cm; $L$ = 22.9 cm.**

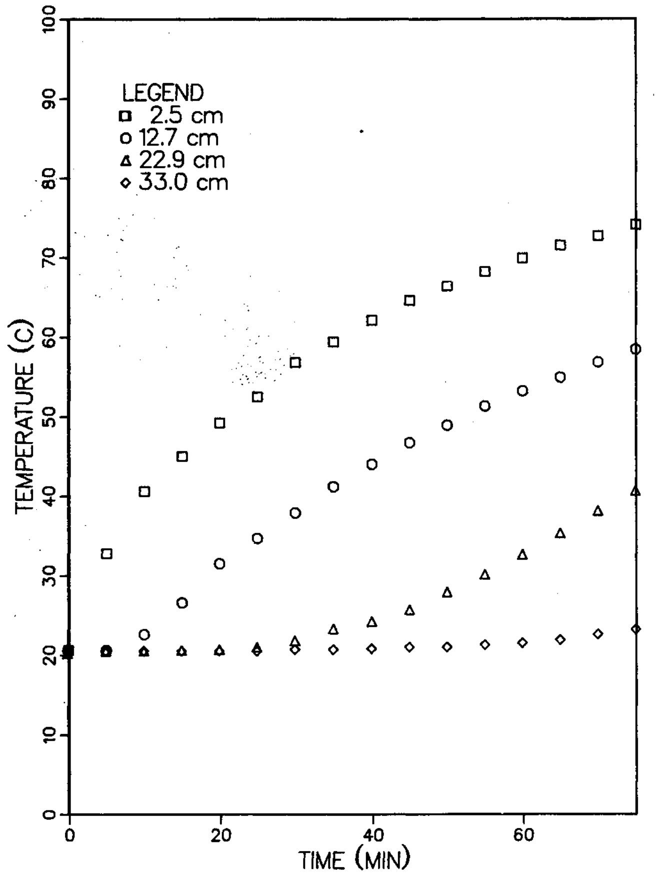


**Fig. 5 Average temperature as function of time and depth with mixer: $C$ = 26.7 cm; $L$ = 12.7 cm.**

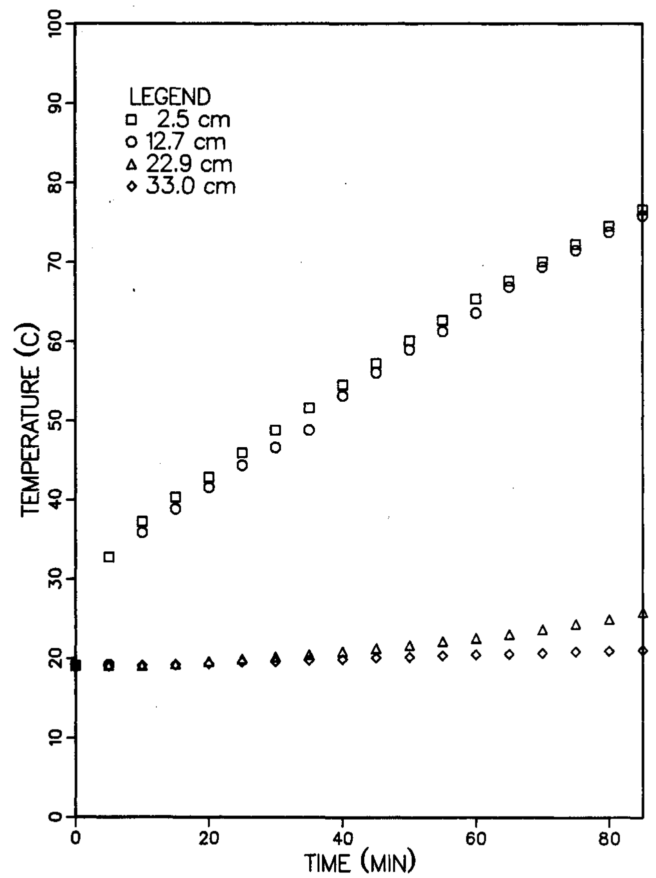


**Fig. 6 Average temperature as function of time and depth with mixer: $C$ = 11.4 cm; $L$ = 33 cm.**

The water was heated from approximately room temperature up to the boiling point. The data logger sampled the temperature once every minute. Data collection for a given test was terminated when the average temperature at any depth in the tank was observed to change less than 0.6°C (1°F) per minute. Typical test periods were up to 2 h. An identical procedure was followed for each mixer design that was tested.

## Results

Figure 2 presents the average water temperature as a function of depth when no mixer was used. The liquid in the vessel displayed a highly stratified temperature distribution. The temperatures at depths of 22.9 cm (9 in.) and 33.0 cm (13 in.) changed very little, even after heating for 90 min. Boiling was observed only in the immediate vicinity of the heating elements during the course of the experiment. Figures 3–6 demonstrate the performance enhancement due to the passive thermal mixer. In each case, the average water temperature as a function of depth is presented for a different ratio of shroud length to arm length. All other mixer dimensions were held constant. Each plot is in dimensional form, as suitable scaling parameters are not currently known.

## Conclusions

A novel passive thermal mixer is proposed, wherein the heating element is contained within a shroud and discharge tubes. Proof of concept experiments were conducted. A small volume of water in the vicinity of the heating element was induced to boil when the mixer was absent. In this case, the majority of the liquid volume remained virtually unaffected by the heat addition, even for an extended period of time.

Insertion of the passive thermal mixer yielded a more uniform temperature distribution throughout most of the liquid volume. The data show that thermal performance can be improved by almost an order of magnitude by use of the mixer. It was found that the mixing effect depended strongly on the dimensions of the shroud and tubing. Several combinations of shroud and discharge tube length were considered. In general, the water temperature distribution became more uniform as the ratio of shroud length to discharge tube length approached unity. This was true regardless of whether the shroud was longer or shorter than the discharge tubes.

The performance of the mixer is thought to depend on several other geometrical parameters in addition to the shroud/arm length ratio, namely, the diameter of the discharge tubes and the shroud, the radial distance between the discharge tubes and the shroud, and the gap between the discharge tubes and the bottom of the tank. Tests are in progress varying other geometric parameters in order to develop design criteria.

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# Heat Transfer to a Power Law Non-Newtonian Falling Liquid Film

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## Nomenclature

$B$ = width of plate
$Br$ = Brinkman number
$g$ = gravitational acceleration
$Gz$ = Graetz number
$Gz^*$ = modified Graetz number
$h$ = heat transfer coefficient
$K_f$ = thermal conductivity
$n$ = power law index
$Nu$ = Nusselt number
$Pe$ = Peclet number
$T$ = temperature
$u$ = velocity component in $x$-direction
$x, y$ = parallel and normal coordinates
$\xi, \psi$ = dimensionless axial coordinates
$\mu$ = viscosity coefficient for power law fluids
$\alpha$ = thermal diffusivity
$\eta, \phi$ = dimensionless normal coordinates
$\delta$ = film thickness
$\theta$ = dimensionless temperature
$\rho$ = density

*Subscripts*
$i$ = inlet conditions
$w$ = surface conditions

*Superscript*
$\bar{}$ = average conditions

## Introduction

THIN falling film with heat transfer is important in modern technology. Non-Newtonian fluid falling film shell and tube exchangers are used in the food and polymer processing applications.

Experimental and theoretical studies of the heat transfer under thermally fully developed conditions were reported by Chun and Seban.[1,2] The conjugate heat transfer in falling liquid films has been recently studied by Gorla et al.[3,4] These investigations were concerned with Newtonian fluids. Murthy and Sarma[5] analyzed heat transfer in the entrance region of a non-Newtonian power law model laminar falling film under constant surface temperature conditions by means of an approximate integral method. The heat transfer from an isothermal inclined plate to non-Newtonian fluid falling films was studied by Stuckheli and Widmer.[6]

In the present work, we have studied the heat transfer in the thermal entrance of a laminar Ostwald-de-Waele type power law model non-Newtonian falling liquid film. The velocity field will be taken as fully developed, whereas the thermal field will be assumed to be developing. The effect of heat generation due to viscous dissipation is considered in the formulation of the problem.

## Analysis

Consideration will be given to a vertical plate placed in a parallel stream of a hydrodynamically fully developed non-Newtonian laminar falling liquid film. The liquid flow is characterized by the power law rheological model. The total shear stress distribution in the liquid film is given by

$$\tau = \mu\left(\frac{\mathrm{d}u}{\mathrm{d}y}\right)^n = \rho(\delta - y)g \tag{1}$$

Using the boundary condition of no slip at the wall and zero interfacial shear at the gas-liquid interface, one may obtain an expression for the velocity distribution in the following form:

$$\frac{u(\eta)}{U_0} = 1 - (1 - \eta)^{(n+1)/n} \tag{2}$$

where

$$\eta = (y/\delta); \qquad U_0 = \left(\frac{n}{n+1}\right)\left(\frac{\rho g}{\mu}\right)^{1/n}\delta^{(n+1)/n}$$

The governing energy equation may be written as

$$u\frac{\partial T}{\partial x} = \alpha\frac{\partial^2 T}{\partial y^2} + \frac{\mu}{\rho C_p}\left(\frac{\mathrm{d}u}{\mathrm{d}y}\right)^{n+1} \tag{3}$$

with boundary conditions given by

$$x = 0\colon T = T_i \text{ (inlet condition)}$$

$$y = 0\colon q_w = -K_f\left(\frac{\partial T}{\partial y}\right)$$

$$y = \delta\colon \frac{\partial T}{\partial y} = 0 \text{ (zero interfacial heat flux)} \tag{4}$$

Proceeding with the analysis, define

$$\xi = \frac{x}{L}, \qquad \eta = \frac{y}{\delta}, \qquad \theta = \frac{T - T_i}{\left(\dfrac{q_w \delta}{K_f}\right)}$$

$$Pe = \frac{\rho C Q}{B K_f}, \qquad Gz = \frac{\delta}{L}Pe, \qquad Gz^* = Gz\left(\frac{2n+1}{n+1}\right)$$

$$Br = \frac{\mu(Q/B)^{n+1}[(2n+1)/n]^{n+1}}{Q_w\,\delta^{2n+1}}$$

$$\phi = \eta[2GZ^*/9\xi]^{1/3}, \qquad \psi = (9\xi/2Gz^*)^{1/3} \tag{5}$$

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