Heat Transfer Characteristics in a Heat Exchanger for Turbulent Pulsating Water Flow with Different Amplitudes

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Abstract: The effect of pulsation on the heat transfer rates, for turbulent water stream with upstream pulsation of different amplitudes, in a double-pipe heat exchanger for both parallel and counter flows, with cold water on the shell side, was investigated. Pulsation frequencies, with using a reciprocating device, ranged up to 260 cycles per minute (up to 4.3 Hz) and 5 different displacement amplitudes were used, (Stroke length of the reciprocating piston was varied from 60 to 185 mm), at different Reynolds numbers 3855-11570. The experimental results indicate that the heat exchanger with a reciprocating piston inserting upstream the flow provides a considerable improvement of the heat transfer rate. The heat transfer coefficient was found to increase with pulsation, with the highest enhancement observed in the transition flow regime. The convective heat transfer coefficient for the turbulent flow was found to increase with pulsation for all pulsation frequencies and amplitudes with the highest enhancement of about 10 times. The maximum enhancement in Nusselt number for the parallel flow was about 8 times while it was about 10 times for the counter one. The improvement in Nusselt number was peaked with the amplitude for both parallel and counter flows.

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1. Introduction

The operation of modern power-producing facilities and industrial equipments used in metallurgy, aviation, chemical, food technology, and other technologies are governed largely by pulsating flows. Cavitations in hydraulic pipelines, pressure surges and flow of blood are some of familiar instances of pulsating flows. The applications include refrigerating systems, reciprocating compressors, internal combustion engines, and pulsing combustion systems. Therefore, recently there has been a growing interest in the effects of pulsating flow on convective heat transfer. In addition, improving the heat exchangers performance is an important application to pulsating flows. In the present work, pulsating flow at different Reynolds numbers was produced by a reciprocating piston, which was located upstream the inlet of the heat exchanger. From literature survey, it may normally be expected that the heat transfer to or from the flow would be changed, the heat transfer will increase or decrease, because the pulsation would alter the thickness of the boundary layer especially near the wall and hence thermal resistance [1-5].

The influence of pulsating flows in a heat exchanger has been studied previously by many investigators [6-27], with conflicting conclusions according to the results. **Baird *et al.*** [6] used steam-water heat exchanger to test the heat transfer performance of a pulsating water flow for Reynolds number ranged from 4300 to 16,200. The cold water was passed upward through a steam jacketed copper tube. Sinusoidal pulsation was produced by an air pulsar, located upstream of the test section. Frequency of pulsation ranged from 0.8 to 1.7 Hz while pulsation amplitude varied from 0.0274 to 0.335 m. The results showed a maximum enhancement of about 41 % based on the overall heat transfer coefficient for Reynolds number of 8000. **Karamercan and Gainer** [7] studied experimentally the effect of pulsation on heat transfer for water stream in a double-pipe heat exchanger, with steam on the shell side. Two heat exchangers, differed in length, were used. Pulsation frequencies ranged up to 5 Hz. Five different displacement amplitudes were used at each flow rate investigated. Reynolds number varied from 1000 to 50000. The pulsator (a reciprocating pump) was located upstream of the heat exchanger and then located downstream. The heat transfer coefficient was found to increase with pulsation, with the highest enhancement observed in the transition flow regime. This was in agreement with the experimental results obtained by **Baird *et al.*** [6] & **Lemlich** [8]. **Lemlich** [8] investigated the effect of pulsation on heat transfer coefficient of water in a double pipe, steam-water heat exchanger. An electrical-hydraulic pulsator consisting of a solenoid valve triggered by an adjustable pressure switch was employed. The valve was installed at a short distance upstream in the water line. Frequency of 1.5 Hz was attained and Reynolds number varied from 500 to 5000. The pulsation increased the overall heat transfer coefficient by 80 %, depending on the upstream location of the solenoid valve, at Reynolds number equal to 2000. The closer the valve to the test section inlet, the better improvement in the overall heat transfer coefficient was achieved. **Lemlich and Hwu** [9] studied the effect of acoustic vibration on forced convective heat transfer of air following in the core of a horizontal, double pipe stream-air heat exchanger. Frequencies of 198, 256, and 322 Hz were imposed on air flowing at Reynolds number of 560 to 5900. The sound vibration frequency was introduced by an electromagnetic driver, actuated through audio amplifier by variable sinusoidal audio signal generator, located upstream of the test section. The results showed an increase of about 51 % in Nusselt number in the nominally laminar regime and up to 27 % in the nominally turbulent regime. The improvement is peaked with both amplitude and frequency. **Shuai *et al.*** [10] studied experimentally the effect of pulsed perturbation on convective heat transfer for laminar flow, in a co-axial cylindrical tube heat exchanger for a viscous fluid. Reynolds number ranged from 150 to 1000 and frequency ranged from about zero to 2 Hz. The amplitude of pulsation varied from 155 mm to 400 mm. The flow pulsation was introduced by a reciprocating pump, located upstream of the heat exchanger. The pulses significantly increased the heat transfer coefficient by more than 300 %, obtained with strong-pulsed perturbations. **West and Taylor** [11] studied experimentally the effect of the pulsation on heat transfer coefficient of water in a long horizontal tube of a steam-water heat exchanger. The pulsating stream of water was pulsated by a reciprocating pump, located upstream, with a variable air chamber, through a 50-mm ID and 6130-mm length galvanized iron pipe of the test section. Reynolds number was varied from 30,000 to 85,000 and pulsation frequency was fixed at 1.6 Hz. The amplitude ratio varied from one to 1.56. They recorded an increase between 60 to 70 % in the heat transfer coefficient at amplitude ratio of 1.42. **Darling** [12] reported an increase of 90% in the heat transfer coefficient, at a Reynolds number of 6000 and a pulsation rate of 160 cycles/min, when pulses were introduced upstream of the heaters. No improvement in the heat transfer coefficient was observed with the interrupter value downstream of the heater.

In the turbulent flow regime, **Baird *et al.*** [6] concluded that pulsations would improve heat transfer, particularly if the flow could be made to reverse in direction for part of the cycle. The maximum improvement they found was 225% (Re = 8000 and pulsated amplitude of 0.335 m) for the waterside heat transfer coefficient in a double pipe, steam-to-water vertical heat exchanger. Experiments conducted by **West and Taylor** [11], on pulsating turbulent flow showed as much as a 70% increase in inner film heat transfer coefficients inside tubes at Reynolds numbers of 30000 to 55000. The pulsation frequency was 100 cycles/min. **Havemann and Rao** [13], using pulsations at 5-33 cycles/s obtained an increase of 42% in the heat flux over the steady flow value under turbulent flow conditions. **Keil and Baird** [14] observed as much as a 100% increase in the overall heat transfer coefficient using pulsating frequencies of 24 to 66 cycles/min, in a commercial shell and tube heat exchanger, with steam in the shell. **Ludlow** [15], using a double pipe heat exchanger with hot water in the annulus, obtained a 500% increase in the tube-side heat transfer coefficient in the transition flow regime. His pulsation frequencies were between 10 and 170 cycles/min.

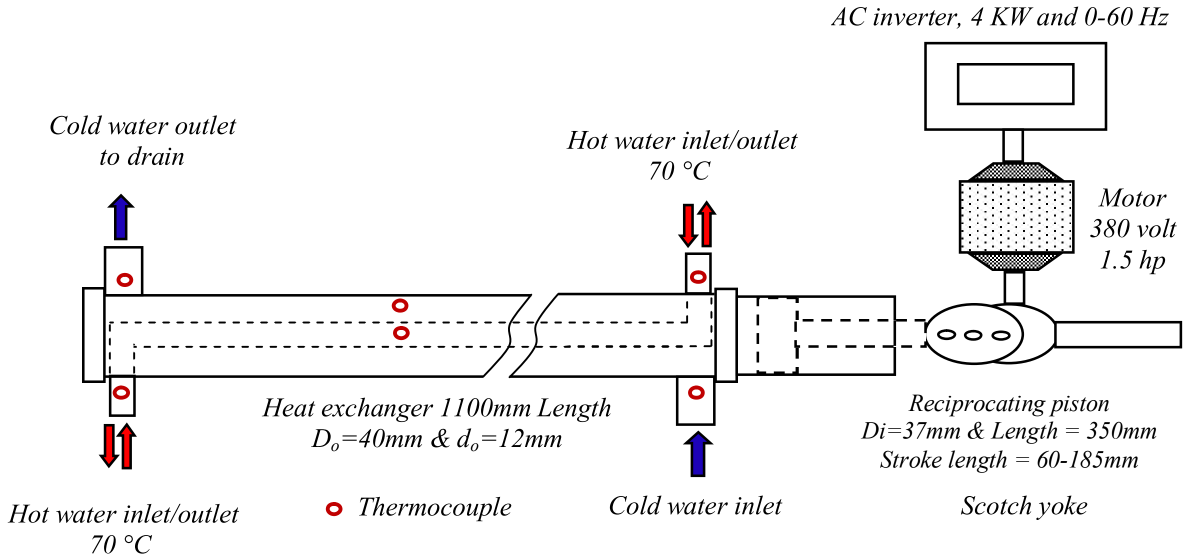
The conflicting conclusions of the effect of pulsated flow in a heat exchanger indicate a lack of proper understanding of the pertinent variables involved. There is an evidence in the literature (**Baird *et al.,*** [6]; **West and Taylor,** [11]; **Darling** [12]; **Havemann and Rao** [13]; **Keil and Baird** [14]; **Ludlow** [15]; and **Martinelli *et al.,*** [18]) that pulsating the flow in a heat exchanger enhances the heat transfer. Furthermore, most of the previous investigators considered only a small number of operating variables in their studies and usually confined their studies to relatively narrow ranges of these variables. Various researchers have investigated the effect of pulsed flow on heat and mass transfer. In the area of heat transfer, theoretical and experimental studies of **Mueller** [16] showed that, for pulsating flows which encompassed a frequency range of 2.3 to 14.9 cycles/min and a Reynolds number range of 53000 to 76000 the average Nusselt number was found to be less than the corresponding steady flow Nusselt number. **McMichael and Hellums** [17] have presented a theoretical development, which concludes that for laminar flow, pulsations cause a decrease in the rate of heat transfer at low amplitudes, which do not allow flow reversal to take place. **Martinelli *et al.*** [18], using semi-sinusoidal velocity disturbances on the tube side fluid in a concentric tube heat exchanger, conducted experimental studies for laminar flow conditions. They found that overall heat transfer coefficient was increased over the steady flow coefficient by 10% at most cases. Experiments performed by **Lemlich and Armour** [19] on a double-pipe, steam-to- water heat exchanger showed increases up to 80% in the overall heat transfer coefficient when the pulsator was installed upstream. However, when installed downstream, a decrease in the overall heat transfer coefficient was observed. The Reynolds numbers investigated were between 500 and 5000 and the pulsation frequencies ranged from 30:200 cycles/min.

From the previous work, it can be observed that, due to variety of control parameters of heat transfer, previous work showed conflicting results for the effect of pulsation on heat transfer. Some investigators reported increases in heat transfer from pulsated flow [6-15]. As seen, there are considerable variations in the quantitative results of pulsating flow experiments. However, this is not surprising in view of the many possible combinations of pulsation variables, pulse generation mechanisms, flow regimes, etc. As a result, some investigators report increases in heat transfer from pulsating flow, and others report little increase, no increase, and even decrease in heat transfer, [16-22]. The aim of this study, therefore, was to investigate the effects of a combination of independent variables on the performance of pulsed-parallel and pulsed-counter flow heat exchangers. These variables included Reynolds number, the frequency of pulsation, and the amplitude of pulsation through a concentric double pipe heat exchanger with cold water inside the annular space.

2. Test rig and Instrumentation

An experimental facility was designed and constructed to investigate the heat transfer characteristics of the turbulent pulsating water flow through a concentric tube heat exchanger. The test rig, as shown in figure 1, consists mainly of a pulsator mechanism, temperatures measuring devices and a horizontal water-to-water concentric tubes heat exchanger with parallel or counter water flows. To minimize the heat losses in the system, the hot water is fed through the inner pipe, with cooling water in the outer annulus. The heat losses to the atmosphere from the outer tube are minimized by insulating the heat exchanger. The heat exchanger of 1100 mm in length, 40 mm outer tube diameter with 1.5 mm wall thickness while the inner tube has 1000 mm length and 12 mm outer diameter with 1 mm wall thickness, (0.037 m^2 heat transfer area), was used in the experiments. The outer and inner tubes of heat exchanger were made of brass (k = 109 W/m^2 K). Six type-K thermocouples are installed through holes in both the inside and outside tubes, to measure the fluid temperatures accurately at the middle and end caps of the heat exchanger. Heat transfer coefficient (h_o) is determined from the overall heat transfer coefficient and then Nusselt number can be calculated, **Pethkool *et al.*** [23] and **Meter** [29]. The cold water entering the system, through outer annulus, was drawn by a 3/4 hp pump from cold water supply tank, and passed to the drain out at downstream. The pulsator device was located upstream the cold-water annulus flow. Control valves are incorporated in each of the two streams to regulate the flow. The flow rates are measured using independent Rota-meters that installed in each line. A hot storage tank (50 liter) is equipped with an immersion type heater and adjustable temperature controller, which can maintain a temperature to within ± 0.5 °C. Circulation of the hot water to the heat exchanger is provided by a 3/4 hp pump and water returns to the storage tank via a baffle arrangement to ensure adequate mixing. The temperature of the hot water was kept at about 65 °C ± 0.5°C. The pulsator mechanism was constructed of three main parts: an electric motor (380 volt and 1.5 hp), a variable speed transmission (AC inverter, 4 KW and 0-60 Hz), and a reciprocating piston (Scotch-Yoke type). The output of the inverter was used as the input to the AC motor. The output of the motor could be adjusted to any value within the range of 0-1550 rpm, which leads to different piston frequencies (140, 170, 200, 230, and 260 cycles/min). The pulsation was imposed to the cold water by the reciprocating piston mechanism, which consisted of a stainless steel cylinder 37 mm inner diameter and 350 mm in length containing an aluminum piston inside. In order to convert the cyclic output of the transmission into a back-and-forth movement for the piston, a coupling mechanism was devised. Its mode of operation was based upon the Scotch yoke principle. The heat losses are the losses of heat through the insulation to the atmosphere and the axial conduction heat losses due to tube thickness. The major heat losses are assumed to be through insulation only with neglecting other losses as concluded by **Baughn *et al.*** [24] and **Incropera and Dewitt** [26].

Nomenclature:			
A	heat transfer area (m^2)	Q	heat transfer rate (W)
C	specific heat capacity (kJ/kg K)	Re	Reynolds number
D	diameter (m)	T	temperature (°C)
F	friction factor	U	overall heat transfer coefficient (W/m^2K)
f	pulsation frequency (Hz)	ΔT_m	logarithmic mean temperature difference (K)
h	convective heat transfer (W/m^2 K)	V	average axial velocity (m/s)
k	thermal conductivity (W/m K)	Greek Symbols	
m	mass flow rate (kg/s)	μ	dynamic viscosity (N s/m^2)
Nu	Nusselt number	ν	kinematic viscosity (m^2/s)
Pr	Prandtl number ($\mu Cp/k$)	ε	heat exchanger effectiveness
Subscripts			
C	cold fluid	o	outlet/outer/un-pulsated
H	hydraulic	p	pulsated
h	hot fluid	s	steady
i	inlet/inner	t	total



3. Experimental Procedure

An experimental program was designed to study the effect of upstream pulsation with different amplitudes on the heat transfer through a concentric tube heat exchanger for the turbulent water flow. The studied values of Reynolds numbers of cold water are 3855, 5780, 7710, 9640, and 11570. These values correspond to mass flow rates of 0.0665, 0.0997, 0.1329, 0.1662 and 0.1994 kg/s, where the mass flow rate of the hot water was kept at 0.1144 kg/s. The inlet hot water temperature was kept at about 65 °C. The outer surface of the test section was insulated to minimize heat losses, and necessary precautions were taken to prevent leakages in the system. The frequency of the pulsations could be varied up to 260 cycles per minute (140 cpm $\leq f \leq$ 260 cpm) where the amplitude was varied by the reciprocating piston mechanism located upstream the cold water inlet. Five different displacement amplitudes were used, (Stroke length was varied as: 6 cm $\leq L_s \leq$ 18.5 cm). Reynolds number is defined as ($4m_c/\pi D_H.\mu_{ci}$) based on inlet cold-water flow conditions. Measurements were carried out with the pulsator located upstream of the outer annulus of concentric heat exchanger. The value of the heat transfer coefficient of pulsated flow was normalized with the corresponding un-pulsated one. Since the temperatures of the hot and cold water vary over the length of the tubes, the temperature difference, $\Delta T = T_h - T_c$, is not constant over the length. To account the temperatures variations, a log mean temperature difference (ΔT_m) is used.

3.1 Theoretical analysis

The heat given by the hot fluid (i.e. water) at any Reynolds number is:

$$Q = m_h C_{ph}(T_{hi} - T_{ho}) = U_i A_i \Delta T_{mi} \qquad (1)$$

While the heat transferred to the cold fluid, (i.e. water) is:

$$Q = m_c C_{pc}(T_{ci} - T_{co}) = U_o A_o \Delta T_{mo} \qquad (2)$$

As usual, this heat may be expressed in terms of a heat transfer coefficient and tube logarithmic mean temperature difference ΔT_m:

$$Q_t = UA\Delta T_m \qquad (3)$$

In the experiments, the tube-wall temperature was not measured, with negligible losses to surrounding air from the cold water, by equalizing the energy loss of hot fluid and the energy received by the cold fluid, convective and overall heat transfer coefficients were deduced and Nusselt numbers were acquired as follows, [25-30]:

$$\frac{1}{UA} = \frac{1}{h_o A_o} + \frac{ln(d_o/d_i)}{2\pi kL} + \frac{1}{h_i A_i} \qquad (4)$$

Where, h_i and h_o are heat transfer coefficients for cold and hot water respectively. The areas, A_i and A_o are the inner and the outer surface areas of each tube. The diameters, D_i and D_o are the inner and the outer tube diameters. U is the overall heat transfer coefficient, k is the thermal conductivity of the tube material and L is the total tube length. For fully developed, turbulent flow in tubes where the Reynolds number is between 2300 and $5x10^6$ and the Prandtl number is between 0.5 and 2000, an empirical correlation to determine h_i proposed by Gnielinski, V. [30], is widely used and hence you can get h_o & Nu_{Do}.

$$Nu_{D_i} = \frac{h_i D_i}{k} = \frac{(F/8)(Re_{D_i} - 1000)Pr}{1+127(F/8)^{1/2}(Pr^{2/3}-1)} \qquad (5)$$

The tube-side heat transfer coefficient could be evaluated from Gnielinski correlation,

$$h_i = = \frac{(F/8)(Re_{Di} - 1000)Pr}{1+12.7(F/8)^{1/2}(Pr^{2/3}-1)} x \frac{k}{D_i} \qquad (6)$$

Where, for smooth tubes, the friction factor is given by:

$F = [0.79 \ln(Re_{D_i}) - 1.64]^{-2}$ (7)

For the hot and cold fluids, the Reynolds numbers are:

$Re_D = VD_H/\nu$ (8)

Heat exchanger effectiveness, ε, is defined as,

$\varepsilon = m_hC_h(T_{hi} - T_{ho})/mC_{min}(T_{hi} - T_{ci}) = m_cC_c(T_{co} - T_{ci})/(mC_{min})(T_{hi} - T_{ci})$ (9)

Where $(mC)_{min}$ *= minimum of either* m_hC_h *or* m_cC_c.

The enhancement in heat transfer was then calculated as the ratio of the heat transfer coefficient obtained in pulsed flow, h_p to that obtained in unpulsated steady flow, h_o. All fluid properties were determined at the overall bulk mean temperature. A more precise method of estimating uncertainty in experimental results has been presented by **Kline and McClintock**, which is described in **Holman** [28]. The method is based on careful specification of the uncertainties in the various primary experimental measurements, suppose that the result dependant variable (R) is a given function of the independent variables x_1, x_2, x_3, ……………, x_n. Thus: R = R (x_1, x_2, x_3, ……………, x_n). Let u_R is the uncertainty in the result and u_1, u_2, u_3, …….., u_n are the uncertainties in the independent variables. The uncertainty in the result is given as:

$$u_R = \left[\left(\frac{\partial R}{\partial x_1}u_1\right)^2 + \left(\frac{\partial R}{\partial x_2}u_2\right)^2 + \ldots\ldots\ldots\ldots + \left(\frac{\partial R}{\partial x_n}u_n\right)^2\right]^{1/2} \quad (10)$$

The following table (3.1) summarizes the calculated values of the uncertainty of the measured quantities.

Table 3.1, values of the uncertainty of the measured quantities

Parameter	Absolute Uncertainty	Relative Uncertainty
Mass flow rate		± 1.2 %
Reynolds number		± 1.2 %
Mean flow temperature	± 0.5 °C to ± 1.0 °C	
Mean Nusselt number		± 4.5 %

4. Results and Discussions

The main target of the present work is to investigate experimentally the influence of pulsation frequency and amplitude in a water-to-water concentric heat exchanger with counter and parallel flows, on heat transfer. With the values obtained from the experimental data (255 experiments), the changes in Nusselt numbers against Reynolds numbers were drawn for six different pulsation frequencies and different amplitudes, as shown in the figures (2-11). Figure (2) and figure (3) show the variation of average Nusselt number versus Reynolds number at different pulsation frequencies for 6 cm stroke length. All curves of the average Nusselt number variation against Reynolds number, for both parallel and counter flows, gives a nearly the same fashion trend. In the case of the parallel flow for the pulsating flow, Nusselt number decreases or increases according the value of pulsation frequency, pulsation amplitude and Reynolds number. It is seen from the figures that the Nusselt number increases as the pulsation frequency, created by the reciprocating piston, increases especially at high Reynolds number. In the parallel flow (Figure 2), for the highest pulsation frequency (260 cycle/minute), the increase in heat transfer rate was about 400% to 500 % according to Reynolds number with maximum enhancement at Reynolds number of 9640. More enhancements in heat transfer rates were obtained for higher amplitudes up to 8 times. In the counter flow (Figure 3), for high pulsation frequency (230 cycle/min), the increase in heat transfer rate was about 500% to 600% according to Reynolds number with maximum enhancement at Reynolds number of 9640. More enhancements in heat transfer rates were obtained for higher amplitudes up to 10 times.

The influence of both pulsation frequency and amplitude on the heat transfer rates, for both parallel and counter flows, is shown in figures (4-9). Figures (4-7) show the variation of the average relative Nusselt number (Nu_{mp}/Nu_{mo}), versus Reynolds number for various pulsation frequencies and various pulsation amplitudes. Figures (8 and 9) show the variation of the average relative Nusselt number (Nu_{mp}/Nu_{mo}), versus pulsation frequency for various Reynolds numbers at stroke length of 12.5 cm. The results of heat transfer show that, more enhancements in relative Nusselt number are obtained for the counter flow than that obtained for the parallel flow. More enhancements in heat transfer rates are obtained at Reynolds number of 3855 and 9640. The heat transfer coefficient was found to increase with pulsation, with the highest enhancement observed in the transition flow regime. This was in agreement with the experimental results obtained by **Baird *et al.*** [6], **Karamercan and Gainer** [7] & **Lemlich** [8]. The heat transfer improvement for counter flow is about 200% higher than that for the parallel flow. The maximum enhancement in Nu_{mp}/Nu_{mo} for the parallel flow was about 8 times. While the maximum enhancement in Nu_{mp}/Nu_{mo} for the counter flow was about 10 times. These enhancements in heat transfer are related to the increased level of turbulence, to the introduction of forced convection in the boundary layer and cavitations [7-26]. The increases in heat transfer with pulsation are due to the higher pulse intensity imparted to the flow through the completely external pipe. The pulsation motion of the fluid results in a pressure gradient being created in the radial direction, thus affecting the boundary layer development. The increased rate of heat transfer in such flows is a consequence of the renewing and reducing the boundary layer thickness and increased resultant velocity. From the figures (4-9), it is also seen that the effect of pulsation flow with different amplitudes on the heat transfer is significant for all Reynolds numbers. This effect is related to the high turbulence that comes from the interacting between pulsation frequency and the bursting frequency of turbulent of water flow, which results from the breakdown of the boundary layer in a shorter time [11-17]. In the counter flow, the heat transfer rates are somewhat greater than that in the parallel flow [27].

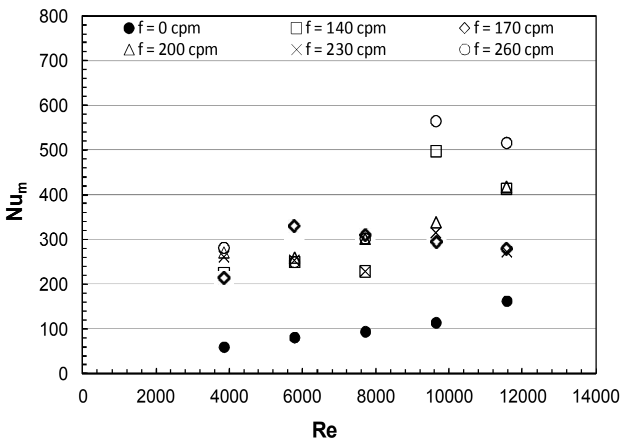


Fig. 2 Nusselt number in the **parallel flow** as a function of Reynolds number for different pulsation frequencies (L_{Stroke} = 6 cm)

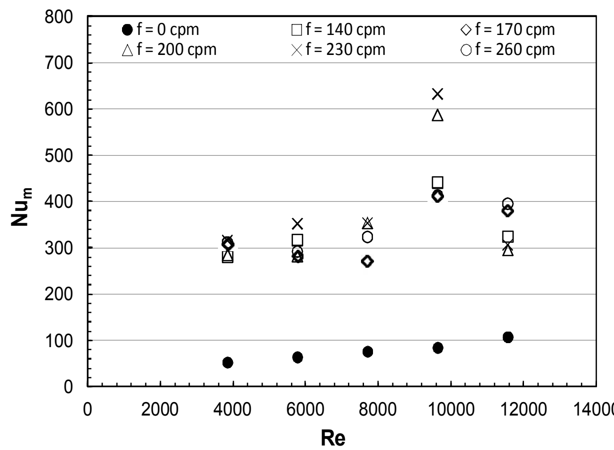


Fig. 3 Nusselt number in the **counter flow** as a function of Reynolds number for different pulsation frequencies (L_{Stroke} = 6 cm)

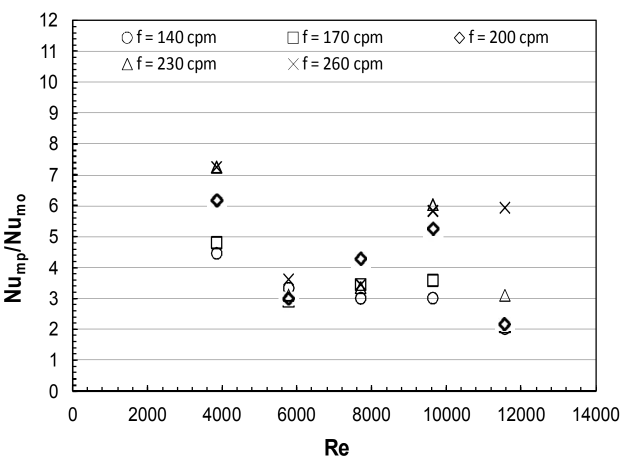


Fig. 4 Average Nusselt number ratio in the **parallel flow** as a function of Reynolds number for different pulsation frequencies (L_{Stroke} = 9.5 cm)

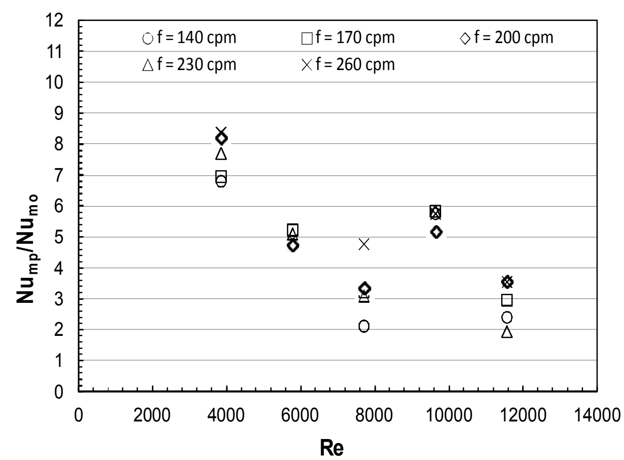


Fig. 5 Average Nusselt number ratio in the **parallel flow** as a function of Reynolds number for different pulsation frequencies (L_{Stroke} = 18.5 cm)

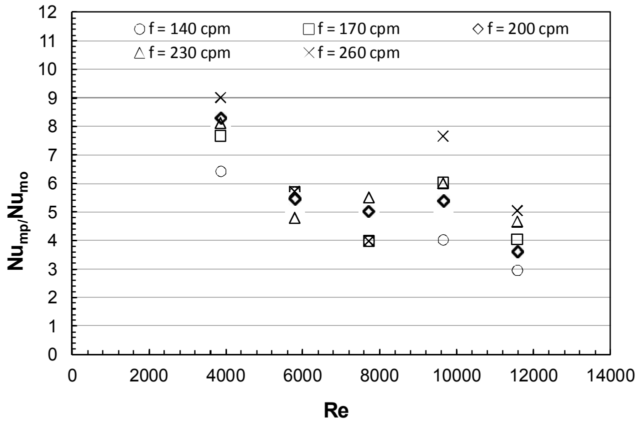


Fig. 6 Average Nusselt number ratio in the **counter flow** as a function of Reynolds number for different pulsation frequencies (L_{Stroke} = 9.5 cm)

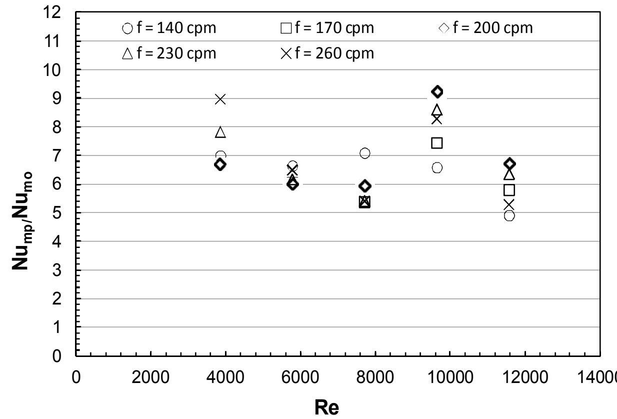


Fig. 7 Average Nusselt number ratio in the **counter flow** as a function of Reynolds number for different pulsation frequencies (L_{Stroke} = 18.5 cm)

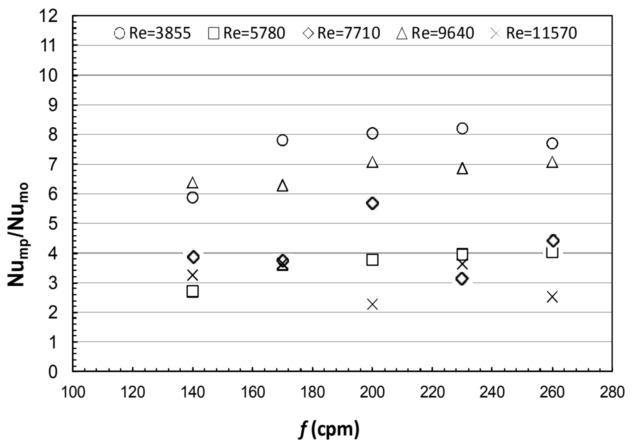


Fig. 8 Average Nusselt number ratio in the **parallel flow** as a function of pulsation frequency for different Reynolds numbers (L_{Stroke} = 12.5 cm)

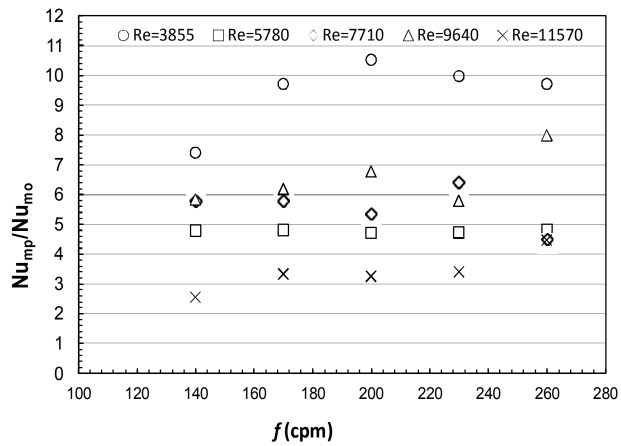


Fig. 9 Average Nusselt number ratio in the **counter flow** as a function of pulsation frequency for different Reynolds numbers (L_{Stroke} = 12.5 cm)

Because the temperature effectiveness of parallel flow is limited with respect to counter-flow, the thermal capacity of the counter heat exchanger can be higher than that of the parallel flow heat exchanger. In addition, the difference between parallel and heat transfer results of counter flow is due probably to later development of a thermal boundary layer. As concluded by many investigators [7-11], a great increase in pressure losses occurs when pulsated or swirl flow generators is mounted at inlet or at outlet of the pipe, in comparison with inner pipe entrance without pulsation or swirling. This results mainly from the dissipation of the dynamical pressure of the fluid (i.e. water) due to very high viscous losses near the pipe wall, and to the extra forces exerted by rotation. Pressure losses will be minimized with low Reynolds number and well designed pulsation generator. Moreover, the pressure drop increase is probably due to the secondary flows occurring because of the interaction of pressure forces with inertial forces in the boundary layer. At relatively low Reynolds numbers (4,000-12,000), as in the present study, and high pulsation frequencies, namely there is an improvement in heat transfer in mentioned range of Reynolds numbers and pulsation frequency at all amplitudes.

Figures 10 and 11 show a variation of relative effectiveness of the heat exchanger versus Reynolds number at different pulsation frequencies, for both parallel and counter flows respectively. Similar fashion trends were obtained as the relative mean Nusselt number. The figures show that more enhancements (about 300%) in the effectiveness were obtained for the counter flow. Figures 12 and 13 show a comparison of the present results, of maximum enhancement values of heat transfer rates, with that obtained by **Karamercan and Gainer** [7]. With comparing the results of the counter flow using upstream pulsation, the maximum enhancement in h_p/h_o given by **Karamercan and Gainer** [7] ranged from 500% to 800% (Re = 4000 – 12,000). While for the present results, the maximum enhancement in h_p/h_o ranged between 700% to 10 times at the range of Reynolds number (Re = 4000 – 12,000).

In addition, the experimental results of **Karamercan and Gainer** [7] showed that the highest enhancements in the heat transfer coefficient obtained within a Reynolds number range of 7500 to 9500. These enhancements in heat transfer are related to the increased level of turbulence, to the introduction of forced convection in the boundary layer [7-9]. Pulsations of sufficient frequency and amplitude can improve heat transfer in such a way as to increase the longitudinal flow for part of the cycle, which, in turn, decreases the film thickness in the tube. Hence, pulsated flow is associated with a periodic pressure gradient reversal, which causes an increase in radial and longitudinal mixing.

In addition, it can be concluded that, more improvements in heat transfer were obtained for higher amplitudes for all pulsation frequencies at any Reynolds number. The reciprocating piston causes the fluid to reverse its direction by pulling on the fluid for a portion of the pulse cycle. Thus, cavitations as well as flow reversal probably occurred in those experiments. The experimental studies conducted by **Baird *et al.*** [6)], **Lemlich** [8], **Darling** [12], **Keil and Baird** [14], and **Ludlow** [15] for the most part covered different values of the variables involved, they all showed that the greatest improvements in heat transfer due to flow pulsations were obtained within a Reynolds number range of 7500 to 9500. The improvements in heat transfer to a flowing fluid in a heat exchanger, by pulsing the flow with sufficient amplitudes has been variously ascribed to cavitations (**Lemlich**, [8], **Darling,** [12] and **Ludlow,** [15] and **Lemlich and Armour**, [19]), and to the increased level of turbulence (**Lemlich**, [8] and to the introduction of forced convection in the boundary layer (**Marchant,** [22]). The influence of each of these factors on the enhancement in the heat transfer coefficient will be discussed below in the following paragraphs.

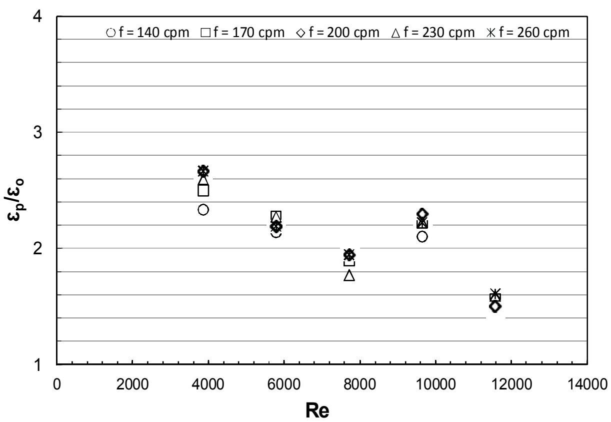


Fig. 10 Effectiveness ratio in the parallel flow as a function of Reynolds number for different pulsation frequencies (L_{Stroke} = 15.5 cm)

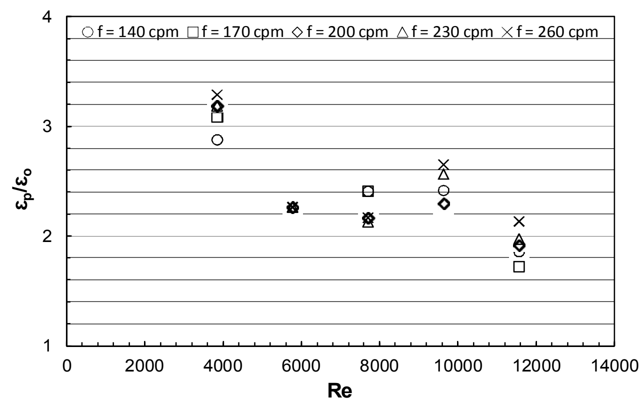


Fig. 11 Effectiveness ratio in the counter flow as a function of Reynolds number for different pulsation frequencies (L_{Stroke} = 15.5 cm)

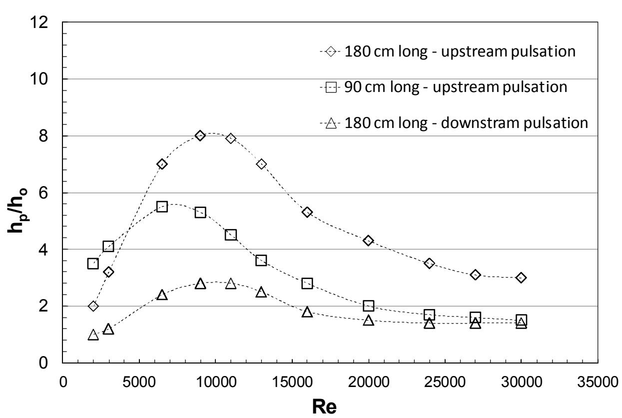


Fig. 12 Maximum enhancement values Counter flow using reciprocating pump, [7]

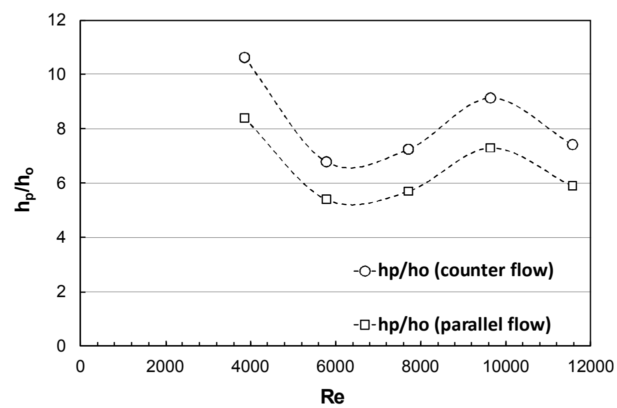


Fig. 13 Maximum enhancement values of relative heat transfer coefficient for both parallel and counter flows

The effect of cavitations is due to bubbles formation, which may be formed in the fluid. These bubbles are produced during the low-pressure portion of the pulse cycle when the flow is suddenly interrupted or forced to change direction and the pressure in the liquid drops below the local vapor pressure. Since the static pressure in the tube is the same across the whole diameter, the bubbles must originate where the vapor pressure is reached first as the pressure drops. This occurs at the hottest point, namely in the boundary layer next to the tube wall, [7]. The periodic formation and collapse of these bubbles agitates the boundary layer and thus increases the rate of heat transfer. According to **Lemlich and Armour** [19], the bubbles also act as carriers of latent heat by moving toward the bulk and giving up their latent heat as sensible heat upon collapsing. They observed a periodic growth and collapse of new bubbles with pulsation, growing from "virtually nothing" to perhaps 0.2 in. in diameter before collapsing. Although they did not degas their water stream before passing it through the exchanger, they noted that the relative growth and collapse of the bubbles far exceeded the comparatively small relative expansion and contraction of stray air bubbles. They felt that this indicated that the bubbles were formed by cavitations rather than merely the expansion of a gas. Like **Lemlich and Armour** [19], we did not degas our entering water, and we observed numerous bubbles leaving through the plastic tube fitted to the down-stream end of the heat exchanger.

In convective heat transfer, axial flow through the tube is a very important parameter. Pulsations of sufficient frequency and amplitude can improve heat transfer in such a way as to increase the longitudinal flow for part of the cycle, which, in turn, decreases the film thickness in the tube. The subsequent reversed axial flow then results in radial diversion of much of the kinetic energy. Hence, pulsated flow is associated with a periodic pressure gradient reversal, which causes an increase in radial and longitudinal mixing. Furthermore, each cycle of motion can be viewed as acting as a disturbance, which increases the level of turbulence in the tube. Higher amplitudes and higher frequencies mean correspondingly larger or more frequent disturbances, hence improved turbulence, and a higher coefficient for heat transfer subsequently; the effect of pulsating on the enhancement in the heat transfer coefficient becomes less at higher flow rates because it must compete with a higher level of turbulence already present in the fluid flow.

Another effect, which may well be related to the ones already discussed, is the introduction of forced convection in the boundary layer when impose pulsation into the flow. It is generally conceded that the heat transfer through the laminar film of the waterside of the tube occurs by means of conduction. In steady flow, convection plays no part in the transfer of heat through this inner boundary layer, which is effectively at rest with respect to the tube wall. However, when pulsations are introduced into the flow, the periodic pressure variations produce forced circulations in the fluid and increase the effective heat transfer by promoting the formation of eddies, thus introducing convection in the boundary layer. Among all the effects discussed above, cavitations seem to be the most dominant factor in the enhancement of heat transfer by flow pulsations.

Conclusions

Observations of behavior of the heat transfer coefficient under the influence of upstream pulsation, for all studied amplitudes, revealed that the heat transfer coefficient was strongly affected with pulsation frequency, amplitude and Reynolds number. In the counter flow, the enhancements in heat transfer rates are somewhat greater than that in the parallel flow. The heat transfer coefficient was found to increase with pulsation, with the highest enhancement observed in the transition flow regime. The results showed that an enhancement in relative average Nusselt number of counter flow up to 10 times was obtained for higher amplitude and higher pulsation frequencies. While, an enhancement in relative average Nusselt number of parallel flow up to 8 times was obtained for higher amplitude and higher pulsation frequency. The maximum enhancements in the heat transfer rates were obtained at Reynolds number of 3855 and 11570. The improvement in the heat transferred to a flowing fluid in a heat exchanger by pulsing the flow with sufficient amplitudes has been variously ascribed to cavitations, the increased level of turbulence and the introduction of forced convection in the boundary layer as concluded by many investigators.

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References

1. Togun H., S.N. Kazi, Badarudin A. "A Review of Experimental Study of Turbulent Heat Transfer in Separated Flow". Australian Journal of Basic and Applied Sciences, 2011, 5(10): 489-505.
2. Zheng, J., Zeng, D., Wang, P. and Gao, H. "An experimental study of heat transfer enhancement with a pulsating flow". Heat transfer-Asian Research, 2004, 33(5): 279-286.
3. Hatami, R., "Pulsating flow in a heat exchanger". Journal of Warme and Stoffubertragung, 1980, (14): 109-118.
4. Krishnan, K. N. and Sastri, V. M. K., "Pulsating flows in heat exchangers - an experimental Study". Warme - und Sroffubertragung, 1982, (16): 169-173.
5. El-Shazly, K.M., Zohir, A.E., Abdel Aziz, A.A., and Abdel Mohimen, M., "Heat transfer characteristics of pulsated flow downstream of

abrupt expansion through pipes", 2nd International Conference on Advances in Engineering Science & Technologies, 2005, November 12-14.
6. Baird, M. H. I, Duncan, G. J., Smith, J. I. and Taylor, J., "Heat transfer in pulsed turbulent flow". Chemical Engineering Science, 1966, (21): 197-199.
7. Karamercan, O. E. and Gainer J. L., "The effect of pulsation on heat transfer". Ind. Eng. Chem. Fundam., 1979, 18(1): 11-15.
8. Lemlich, R. (1961): "Vibration and pulsation boost heat transfer". Chemical Engineering, 1961, 68(10): 171–176.
9. Lemlich, R. and Hwu, C., "The effect of acoustic vibration on forced convective heat transfer". A. I. Ch. E. J., 1961, (21): 197-199.
10. Shuai, X., Cheng, S. and Antonini, G., "Pulsation effects on convective heat transfer in the laminar flow of a viscous fluid". The Canadian Journal of Chemical Engineering, 1994, (72): 468 - 475.
11. West, F. B. and Taylor, A. T.: "The effect of pulsation on heat transfer in turbulent flow of water inside tubes". Chemical Engineering Progress, 1952, (48): 39-43.
12. Darling, G. B., "Heat Transfer to Liquid in Intermitted Flow Petroleum"., London, 1959, (22):177-182.
13. Havemann, H. A., Rao, N. N., "Heat transfer in pulsating flow". Nature 7(4418): 41.
14. Keil, A. H,. Baird, M. H. I., "Enhancement of heat transfer by flow pulsations". Ind., Eng. Chem. Process Des. Dev., 1971, 10:473.
15. Ludlow, J. C., MS. Thesis, University of Virginia, Charlottesville, Va., 1975.
16. Muller, W. K., "Proceedings fifth Midwestern conference of fluid mechanics". p 146, Ann Arbor, Mich., 1957.
17. McMichael, W. J., Hellums, J. G., AIChE J., "The Effect of Pulsations on Heat Transfer for Laminar Flow". 1975, 21:743.
18. Martinelli, R. C., Boelter, L. M. K., Weinberg, E. B. and Yakal, S., "Heat transfer to a fluid flowing periodically at low frequencies in a vertical tube". Transaction of ASME, 1943, 789-798.
19. Lemlich, A., Armour, J. C., "Enhancement of Heat Transfer by Flow Pulsations"., Chern. Eng. Prop. Symp. 1965, (61): 83.
20. Sieder, E. N., Tate, G. E., "Heat Transfer and Pressure Drop of. Liquid in Tubes"., Ind. Eng. Chethe., 1936, (28): 1429-1436.
21. West, A. B., Taylor, A. T., "The effect of pulsation on heat transfer in turbulent flow of water inside tubes". Chem. Eng. Prog., 1952, 48(1): 39-43.
22. Marchant, J. H., "The Effect of Pulsations on Heat Transfer". Trans. ASME, 1943, (65): 786.
23. Pethkool,S., Eiamsa-ard, S., Kwankaomeng, S. and Promvonge, P. "Turbulent heat transfer enhancement in a heat exchanger using helically corrugated tube". International Communications in Heat and Mass Transfer, 2011, (38): 340–347.
24 Baughn, J. W., Hoffman, M. A., Launder, B. E., and Takahashi, R. "Turbulent Heat Transport in Circular Ducts with Circumferentially-Varying Heat Flux". ASME Journal of heat Transfer, 1984, (106): 64-70.
25 Kays, W.M., and Crawford, M.E. (1993) "Convective Heat and Mass Transfer". Third ed., McGraw-Hill, Singapore.
26 Incropera, F., and Dewitt, P.D. (1996) "Introduction to Heat Transfer". Third edition, John Wiley & Sons Inc.
27 Durmus, A., and Esen, M. "Investigation of heat transfer and pressure drop in a concentric heat exchanger with snail entrance". Applied Thermal Engineering, 2002, (22): 321–332.
28 Holman, J. P. (1984) "Experimental Methods for Engineers". 4Th Editions, McGraw- Hill Book Company, New York.
29 Meter, J. V. (2008) "Experimental investigation of a printed circuit heat exchanger using super critical carbon dioxide and water as heat transfer media". MS. Thesis, Kansas, University, Manhattan, Kansas.
30 Gnielinski, V. "New Equation for Heat and Mass Transfer in Turbulent Pipe and Channel Flow". International Chemical Engineering, 1976, (16): 359-368.

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