Type	d_p	Model results			Experiment	deviation
	Mm	t_d, s	t_c, s	t_t, s	t_e, s	%
Bagasse	4	7.22	28.62	35.84	37	-3.13
	10	39.90	93.25	133.15	127	4.84
	15	86.83	155.56	242.39	254	-4.57
	20	151.63	222.34	373.97	385	-2.86
Manure	4	8.07	33.21	41.28	43	-4.00
	10	45.56	99.18	144.74	152	-4.78
	15	99.78	160.45	260.24	256	1.65
	20	151.63	225.06	376.69	390	-3.41
Robinia	4	4.81	26.60	31.41	33	-4.81
	10	27.32	88.56	115.88	121	-4.23
	15	59.94	148.79	208.74	199	4.89
	20	105.15	213.56	318.71	328	-2.83

Table 3. Particle total burn-out time, a comparison between model and experiments.

5. CONCLUSION

Mechanism of combustion of highly wet biomass is studied and investigated. New data is obtained on the different stages of combustion, drying, devolatilization and char combustion. According to the present work the following concluding remarks may be drawn.

1. Volatiles and water vapour are found to be yield simultaneously from the biofuel particle under FBC conditions.
2. The devolatilization time, t_v, decreases as expected with an increase of the particle diameter, the bed temperature, T_b, and decrease of moisture content
3. The char combustion time is higher with increase of both fixed carbon and ash content.
4. The ratio t_v/t_c increases with:
   - increase of the biomass particle diameter
   - increase of the oxygen concentration, since t_v is virtually independent of oxygen concentration.
5. A simplified analytical model of heat transfer controlled is established. The model describes well the experimental volatiles release. Empirical correlations (31) and (32) are obtained for the volatiles release time and kinetics of particle char combustion.

6. NOMENCLATURE

Ar_i Archimedes number of inert particle, $gd_i^3(\rho_i - \rho_g)/(\nu_g^2 \rho_g)$

B_c $(Bi_c-1)/Bi_c$

B_d $(Bi_d-1)/Bi_d$

Bi_c Biot number, hr_o/k_c

Bi_d Biot number, hr_o/k_d

C_f fixed carbon content

C_{O2} oxygen concentration

c_p specific heat capacity
D diffusion coefficient of oxygen
d_i diameter of inert (bed) particle
d_p diameter of fuel particle
d_{po} Initial diameter of fuel particle
d_{pore} pore diameter
E activation energy
g acceleration of gravity
h heat transfer coefficient
h_m mass transfer coefficient
k thermal conductivity
k_c chemical reaction rate constant
k_{ov} overall reaction rate coefficient
k_v kinetic of devolatilization constant
Pr Prandtl number, $\rho_g C_{pg} v_g / k_g$
q latent heat of evaporation
R dimensionless radial co-ordinate, r/r_o
r radial co-ordinate
Sc Schmidt number, vg/D
Sh Sherwood number, $h_m d_p/D$
T_a time-average surface temperature of a fuel particle
T_b bed temperature
T_v volatile release temperature
t_c char burnout time
t_{vd} delay period before volatile ignition
t_{vf} volatile flaming (release) time
W_o initial moisture content
Z $k_c(T_b - T_v)/k_d(T_b - T_{wc})$

Greek symbols

α Thermal diffusivity, $k/\rho c_p$
ε_a, ε_b emissivity
ε_{mf} bed voidage at minimum fluidization
ε_v porosity
Nu Nusselt number, hd_p/k_g
μ stoichiometric coefficient, (kg carbon)/(kg oxygen)
ν kinematic viscosity
ρ density
τ dimensionless time
θ dimensionless temperature
θ_v $(T_v-T_{wc})/(T_b-T_{wc})$
σ Stefan-Boltzmann constant

Subscripts

a active particle
b bed
c char combustion; convective; conductive
d dry
g gas

i inert particle
o initial
r radiative
s surface
v devolatilized
vd delay before devolatilization
vf volatile flaming
wc wet core

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AN EXPERIMENTAL STUDY FOR HEAT TRANSFER IN LAMINAR PULSATING ANNULAR FLOW

"دراسة معملية لانتقال الحرارة فى السريان الرقائقى النبضى الحلقى"

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خلاصة

هـــذا البحث يقدم دراسة معملية لانتقال الحرارة لسريان نبضي رقائقي حلقي مقارنة بالسريان المنتظم في مـــبادل حـــراري أنبوبـــي. وقد تم تصميم وتنفيذ دائرة اختبار معملية مزودة بنظام إلكتروني لتجميع البيانات الخاصــــة بدائـــرة الاختـــبار حيث يتم تجميع الإشارات الواردة من حساسات قياس درجات الحرارة والضغط والـــتدفق وتوصـــيلها بالحاســـب الآلـــي وتخزينها لمعالجتها وتحليلها. تم حساب رقم نوسلت للسريان الحلقي الرقائقي والفعالية للمبادل الحراري عند ظروف تشغيل مختلفة، حيث أجريت التجارب لسريان نبضي ذو تردد واحد هرتز وعند قيم مختلفة لرقم رينولدز، وسعة النبضة للسريان النبضي ونسبة خلط كل من سريان نبضي وأخر منتظم وذلك لدراسة تأثير العوامل السابقة علي انتقال الحرارة في السريان النبضي الرقائقي الحلقي.

وقـــد أظهـــرت النـــتائج أن رقم نوسلت للسريان النبضي يزيد بمقدار ٢٠% مقارنة بالسريان المنتظم في المـــدى مـــن ١٠٠ الي ١١٠٠ لرقم رينولدز. مما أدي لزيادة بنفس النسبة في معامل انتقال الحرارة الإجمالي للمبادل الحراري المستخدم في حالة السريان النبضي مقارنة بالسريان المنتظم.

ABSTRACT:

Heat transfer in laminar pulsating flow is experimentally investigated. Water flow through an annulus of a tubular heat exchanger is considered. An experimental set-up equipped with a computerized data acquisition system is designed and constructed to carry out this work. The test section is a tubular heat exchanger formed by two concentric tubes. The pulsating flow of cold water in the annulus is heated by a steady flow of hot water through inner tube of the tubular heat exchanger. The experimental measurements of temperature, pressure and mass flow rate for both the hot and cold fluids are eollected by a data acquisition system, which is connected to a personal computer for further data analysis. The operating parameters considered are Reynolds number, mixing ratio between pulsating and steady flows and the change in amplitude for pulsating flow The frequency of the pulsating flow considered is 1 Hz. The influence of flow pulsation in the annulus on Nusselt number and its impact on the overall heat transfer coefficient and effectiveness of a tubular heat exchanger are considered.

The results show that, flow pulsation increases Nusselt number of laminar annular flow compared with the steady flow. Such increase in Nusselt number is slightly dependent on Reynolds number. For the considered range of Reynolds number, Re=100-1100, flow pulsation causes an increase in Nusselt number by 20% in average compared to steady laminar flow.

KEY WORDS: Laminar, annular, pulsating flow.

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1. INTRODUCTION

Pulsating flow has received a considerable attention for many years due to its importance for many industrial, practical and biological applications. Examples of such applications are blood flow, respiratory systems, Stirling engine heat exchangers, reciprocating pumps and compressors, and many others. Many investigations are carried out, but little attention was given to heat transfer in laminar pulsating flow. Even for the very little work available, its results are contradicting.

Kim et. al. (1993) made a theoretical analysis to study heat transfer characteristics for fully developed pulsating flow in a channel. The effects of pulsation on the Nusselt number are pronounced for large amplitudes and in the upstream entrance region. Changes in Nusselt number due to pulsation are appreciable thought much of the channel length when frequency is low and moderate. At high pulsation frequencies, changes in Nusselt number are generally minor. In comparison to the case of a non-pulsating flow, reduction in local Nusselt number is expected at low pulsation frequencies and in the extreme up stream region, and enhancement in local Nusselt number is anticipated at moderate downstream locations. At locations further downstream, Nusselt number remains virtually unchanged.

Ligrani et. al. (1996) presents experimental results to describe the effect of bulk flow pulsation on film cooling from a single row of holes. The pulsations are in the form of sinusoidal variations of static pressure and streamwise velocity. The results provide clear evidence of the dramatic impact of bulk flow pulsations on film cooling heat transfer.

The effect of pulsation in a tube with constant heat flux at the wall is considered by Moschandreou et al. (1997) to determine how pulsation affects on the heat transfer rate and how the phenomena depend on the Prandtl number and on pulsation frequency. The results show that in a range of moderate values of the frequency there is a positive peak in the effect of pulsating flow whereby the bulk temperature of the fluid and the Nusselt number are increased But the effect is reversed when the frequency is outside this range (very low or very high) The peaks (the mechanism for change in the rate of heat transfer) are higher at lower Prandtl numbers due to lower momentum and higher heat diffusivity.

Many versions of the Nusselt number definitions have been tested by Guo et. al. (1997) to clarify the conflicting results in the heat transfer characteristics for pulsating flow in a pipe. For small amplitude both heat transfer enhancement and reduction were detected depending on the pulsation frequency. But for large amplitudes, the heat transfer due to pulsation is always enhanced.

Hafez et. al. (2000) made a theoretical model, to show the effect of pulsations on the heat transfer coefficient, for the flow inside a circular tube with constant heat flux. This model show that, pulsation improvement was obtained with increasing frequency up to certain limit also, with decreasing tube length.

Hemeada et. al. (2000) analyzed heat transfer in laminar incompressible pulsating flow in a duct. An analytical solution of the thermally fully developed case and a numerical solution were obtained. This study shows that, the overall heat transfer coefficient increases with increasing the amplitude and decreases with increasing frequency and Prandtl number.

Tubular heat exchanger is the simplest form of heat exchangers and consists of two concentric tubes carrying the hot and cold fluids. Flow pulsation in heat exchangers is frequently encountered in many applications. Knowledge of the influence of flow pulsation on heat transfer is essential. Data available are limited and contradicting. Therefore the objective of the present work is to study, experimentally, the effect of flow pulsation on the heat transfer in laminar annular flow

2. EXPERIMENTAL SET-UP

The experimental set-up consists mainly of a tubular heat exchanger, hot water circuit, cold water circuit and plunger pump A schematic diagram for the experimental set-up is shown in Fig. (1). The experimental set-up is provided with a data acquisition system, which is connected to a personal computer to collect experimental data for storage and further analysis. The system is provided with sensors to measure the temperatures, pressures and mass flow rate for both hot and cold fluids.

Tubular heat exchanger consists of two concentric tubes Hot water passes through the inner tube and cold water passes through the annulus. The inner tube is made of stainless steel with 9.5 mm outer diameter, 0 6 mm wall thickness and 660 mm length. The outer tube is made of clear acrylic tube with 12 mm inner diameter and 3mm thickness. The outer surface of the outer tube is insulated by a 3 cm thickness glass wool layer covered with an aluminum foil to minimize heat loss. Heat transfer area is 184.5 cm^2 The temperatures at inlet and outlet for both the cold and hot water are measured using K type thermocouples and connected to a data acquisition system.

Hot water circuit consists of a circulating pump, 2 kW electric water heater and pipe connections. A thermostat controller is used for controlling the temperature of hot water entering the heat exchanger.

Cold water flows through the annulus may be pulsating or steady. The pulsation equipment comprises of a plunger pump, which is driven by an electric motor. The frequency of the pulsating pump is 1 Hz Referring to figure (1), water inlet to the plunger pump passes through a constant volume tank (680 cm^3) to measure the volume flow rate. The tank is provided with two optical level sensors fixed at the top and bottom of the tank. The time taken for the water level in the tank to fall from the upper switch to the lower one is measured and the volume flow rate of cold water is calculated by the data acquisition system. Pressure of cold water at outlet from plunger pump and at the inlet to the test section are measured by two pressure sensors, which are connected to the data acquisition system. The flow rate of both hot and cold water flows are measured by two electronic turbine flow meters connected to the data acquisition system. To control the pressure fluctuation at the discharge of the plunger pump, a pressure accumulator (or pulsation damper) was fitted, as shown in Fig. (1). Also, mixing between pulsation and steady flows is performed during the experimental work. Control valves are used to obtain the required mixing ratio between pulsating and steady flow rates Pressure and mass flow rate for the mixture are measured before and after mixing

3. EXPERIMENTAL MEASUREMENTS TECHNIQUE

Prior to the start of the experiments, the experimental set-up was allowed to equilibrate for approximately one hour until steady state condition had been reached (the fluctuating in temperatures was about ± 0.1 °C). Once the system reached the desired steady state condition, the required measurements were taken. These measurements are temperature, mass flow rate and pressure at different positions, as shown in Fig (1).

The uncertainties are calculated for the measured and calculated quantities. The quantities measured directly included test section dimensions, pressure, temperature and flow rate for both hot and eold water flows. The accuracy of the thermocouples used is ± 0.1 °C, thus, the uncertainty of temperature difference was estimated to be about 0.142 °C. The accuracy of the electronie turbine flow meters is ±0.01 Lit/min, which is equivalent to 0.3 % of the full seale. Also, the accuracy of the geometric dimensions is estimated to be 0.55 %. The largest calculated uneertainties in the current investigation, were less than 2.5 % for the overall heat transfer coefficient and 2.65 % for the effectiveness of the tubular heat exchanger. Also, the largest calculated uncertainties in the Reynolds number was found to be about 3.5% and that for Nusselt number was 2.5 %.

4. DATA REDUCTION

The hot and cold fluid streams flow in opposite directions (counter-current flow). The measured temperatures and mass flow rates for the hot and cold fluids are analyzed to calculate the overall heat transfer coefficient, outer tube side (cold side) heat transfer coefficient and heat exchanger effectiveness using a computer data reduction program. Cold fluid properties are calculated at the mean cold fluid temperature along the test section (tubular heat exchanger) Also, hot fluid properties are calculated at the mean hot fluid temperature.

Hot water flowing inside the inner tube of a tubular heat exchanger was kept at the same condition throughout this work. Where, hot water flow was turbulent at Reynolds number approximately fixed at Re_h = 14000 Therefore, the inner side convection heat transfer coefficient was calculated from the following correlation:

$$h_i = 0\ 023\ Re_h^{0.8}\ Pr^{0.3}\ (k / d_i)$$

Assuming no fouling resistance, the outer side convection heat transfer coefficient, for the cold fluid in the annulus, is determined as follow.

$$h_o = \frac{1}{\frac{1}{U} - \frac{A_o}{A_i h_i} - \frac{A_o \ln(d_o / d_i)}{2\pi k L}}$$

Therefore, Reynolds number and Nusselt number for cold fluid, which flows in the annulus, can be obtained as:

$$Re_c = \rho\ u_c\ D / \mu \quad \& \quad Nu = h_o\ D / k$$

5. RESULTS AND DISCUSSIONS

The pressure of the pulsating flow is measured at the inlet of the test section using a pressure transducer. The sampling rate of pressure data is 1000 samples/sec. A typical plot of the pressure for the pulsating flow as function of time is presented in figure (2). It shows that the frequency of the flow pulsation is 1 Hz. The difference between the maximum and minimum pressures divided by the average value for the pressure represents the dimensionless amplitude for pulsating flow.

The hot fluid flowing inside the inner tube of a tubular heat exchanger was kept at constant conditions throughout this work. The variation of the inner heat transfer coefficient was small. Therefore, the behavior of the overall heat transfer coefficient (U) is the same as that of the outer heat transfer coefficient for the cold fluid (h_o). In order to detect the changes in heat transfer due to flow pulsation, the Nusselt number for the cold fluid (Nu) is plotted for both steady and pulsating flows in figure (3) The range of Reynolds number considered for the annulus flow is 100-1100. It has been clearly demonstrated that Nusselt number (Nu) for laminar annulus flow depends on Reynolds number (Re_c) for both steady and pulsating flows. Curve fitting of the experimental results in the considered range of Reynolds number shows that Nusselt number is related to Reynolds number as:

$$Nu = 0.215 . Re_c^{0.55} \qquad \text{for steady laminar annulus flow}$$

$$Nu = 0.145 . Re_c^{0.65} \qquad \text{for pulsating laminar annulus flow (at A=0.66)}$$

Figure (3) shows that laminar flow pulsation increases Nusselt number compared with that of steady laminar flow in the annular cross section. This increase may be attributed to the local turbulent eddies generated by the flow pulsation Therefore, heat is transferred by this eddying motion as well as by the molecular diffusion in laminar flow.

The percentage change in Nusselt number $((Nu)_p-(Nu)_s)/(Nu)_s)$ is plotted as function of Reynolds number in figure (4). It shows that, for the considered range of Reynolds number, Re_c=100-1100, flow pulsation causes an increase in Nusselt number by 20% in average

Mixing between pulsating and steady flows was earried out to give higher values for Reynolds number. The results of mixing between steady and pulsating flows are considered in figure (5). The variation of Nusselt number with Reynolds number for two mixing ratios X=1, and 0.5 compared with steady flow (X=0.0) is shown in figure (5). A noticeable increase in Nu for pure pulsating flow (with mixing ratio, X=1) over steady flow (X=0.0) is found. However, it is hardly to detect any change in Nu for the mixing ratio of X=0.5 compared with steady flow. This may be due to the damping of flow pulsation with mixing.

Figure (6) presents a plot of the dimensionless parameter $Nu/Re_c^{0.55}$ versus Reynolds number. It clearly demonstrates Nusselt number enhancement obtained in case of pulsating flow compared with steady flow in laminar annular flow.

The effect of pulsating dimensionless amplitude (A) on the dimensionless parameter $Nu/Re_c^{0.65}$ for pulsating laminar flow is shown in figure (7). It is observed that, such variation in $Nu/Re_c^{0.65}$ is slightly dependent on the dimensionless amplitude The highest value for $Nu/Re_c^{0.65}$ was performed at dimensionless amplitude equal to about 0.4.

Experimental measurements are taken to obtain the performance behavior for a tubular heat exchanger with different operating parameters. Heat exchanger performance can be determined by calculating the overall heat transfer coefficient and effectiveness. Overall heat transfer coefficient for different values of Reynolds numbers is shown in Fig. (8) for pulsating and steady flows in laminar flow region. It is observed that U for pulsating flow increases by about 20% than steady flow. This increase can be explained as, due to increasing in the temperature difference for the hot fluid (decrease in the outlet hot temperature for the same inlet temperature and mass flow rate for the hot fluid) in pulsating flow than steady flow Then the amount of heat transferred increased This leads to an expected improve in the U was obtained.

Figure (9) shows that, the effectiveness for the tubular heat exchanger (ε) versus Reynolds number in laminar flow region for pulsating and steady flows. It is found that, pulsating flow increases the amount of heat transferred, consequently ε increases, according to the definition of heat exchanger effectiveness.

6. CONCLUSIONS

An experimental investigation was carried out to determine the overall heat transfer coefficient and effectiveness for a tubular heat exchanger, in case of pulsating laminar flow compared with steady flow. Water was used as working fluid for both hot and cold fluids. The studied operating parameters are cold fluid mass flow rate, mixing ratio between pulsating and steady flows and amplitude

The obtained results from the experimental investigation verified that, Nusselt number for pulsating laminar flow increases by an average value 20% compared with steady flow. Also, higher values for Nusselt number was obtained at dimensionless amplitude equal to about 0 4.

Additional investigations are still necessary to discover different flow regimes, pulsating frequencies, and other flow configurations.

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NOMENCLATURE

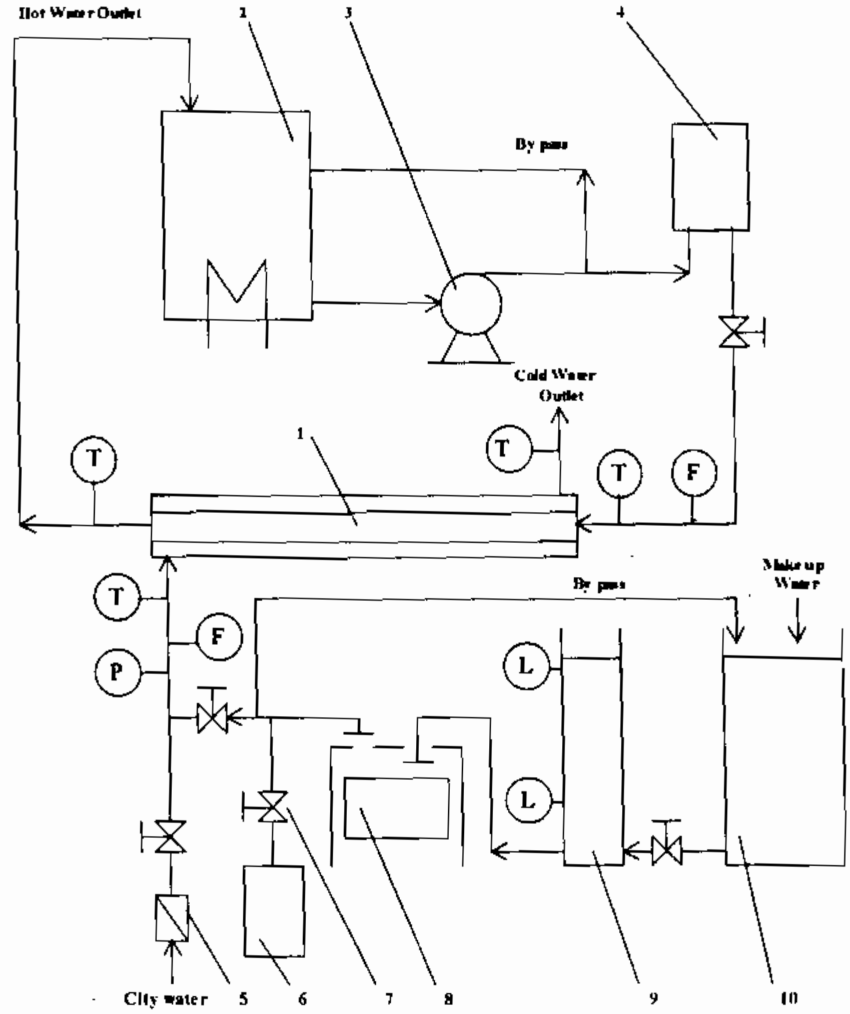
a	: Amplitude	(bar)
A	: Dimensionless amplitude ($A=a/P_{av}$)	-
A_i	: Inner heat transfer surface area ($A_i = \pi d_i L$)	(m^2)
A_o	: Outer heat transfer surface area ($A_o = \pi d_o L$)	(m^2)
d	: Tube diameter	(m)
D	: Hydraulie diameter for the annulus	(m)
h	: Heat transfer coefficient	($W/m^2.K$)
k	: Thermal conductivity	(W/m.K)
L	:Length	(m)
m	:Mass flow rate	(kg/s)
Nu	: Nusselt number	-
P	: Pressure	(Pa)
Pr	: Prandtl number	-
Q_c	: Heat transfer rate to cold fluid	(W)
Q_{max}	: Maximum possible heat transfer rate	(W)
Re	: Reynolds number	-
T	: Temperature	(°C)
u	: Velocity	(m/s)
U	: Overall heat transfer coefficient	($W/m^2.K$)
X	: Mixing ratio [$X= m'_p/(m'_p + m_s)$]	-

Greek symbols

ε	: Effectiveness [$\varepsilon = Q_c/Q_{max}$]	(%)
μ	: Dynamic viscosity of the fluid	(kg/m.s)
ν	: Kinematic viscosity of the fluid	(m^2/s)
ρ	: Density	(kg/m^3)

Subsripts

av	average
c	cold
h	hot
i	inner
o	outer
p	pulsate
s	steady



1 Tubular heat exchanger
2 Electric water heater
3 Hot water circulating pump
4 Hot water circulator
5 Filter
6 Pulsation damper
7 Control valve
8 Plunger pump
9 Volumetric tank
10 Water tank
F Flow rate
L Level sensor
P Pressure sensor
T Temperature sensor

Fig. (1) Schematic diagram for the experimental set-up