INVESTIGATION OF TWISTED TAPE TURBULATOR FOR FIRE TUBE BOILER
Part I. Heat Transfer

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ABSTRACT:

The present work presents a new experimental study of the enhancement of turbulent convection heat transfer inside tubes for combined thermal and hydrodynamic entry length of one popular “turbulator” (twisted tape with width slightly less than internal tube diameter) inserted for fire tube boilers. Cylindrical combustion chamber was used to burn (1.6 to 7kg/h) fuel oil #2 to deliver hot gases with ranges of Reynolds number (10500 to 21700), and (11400 to 24150) for both empty and inserted tube respectively.A uniform wall temperature technique was used by keeping approximately constant water temperature difference (25ºC) between inlet and exit cooling water in parallel flow shell and tube heat exchanger. The test tube consisted of smooth carbon steel tube of (2400mm) long and (52mm) internal diameter. This test tube instrumented to derive local heat transfer coefficient and local flue gasses static pressure.The experimental results show that for the same fuel consumption, twisted tape insert with (H/D = 11.15) enhanced the mean Nusselt number in (75.2%), (68.8%), (49.8%), (40.3%), and (16.7%) for fuel consumption (7kg/h), (6.16kg/h), (4.5kg/h), (3.24kg/h), and (1.6kg/h) respectively.A set of empirical correlations that permit the evaluation of the mean Nusselt number (for developing and fully developed region), and average Nusselt number (for developed region) for empty and inserted tube are generated for engineering applications.

Keywords: fire tube boiler; twisted tape; heat transfer enhancement

الخلاصة:

هذا البحث يقدم دراسة عملية جديدة لتحديد التحسين الحاصل في معامل انتقال الحرارة لغازات الاحتراق المارة داخل انابيب مراجل انابيب النار عند ادخال شريط معدني ملتوي بعرض يقل قليلا عن قطر الانبوب.تم تجهيز غازات الاحتراق من غرفة احتراق اسطوانية الشكل يتم فيها حرق كمية من الوقود السائل الخفيف رقم 2 تتراوح من (1.6 – 7 كغم/ساعة) بحيث يعطي رقم رينولدز من (10500-21700) للانبوب الفارغ و(11400-24150) للانبوب المدخل فيه الشريط الملتوي.تمت المحافظة على درجة حرارة جدار الأنبوب منتظمة وذلك بالسيطرة على فرق 25 درجة مئوية بين درجة حرارة الماء الداخل والخارج في مبادل حراري نوع (الغلاف والانبوب) بجريان متوازي بطول انبوب 2400 ملم وقطر داخلي 52ملم . ربطت على هذا الانبوب مجموعة من اجهزة القياس التي ساعدت على حساب معامل انتقال الحرارة والضغط الاستاتيكي الموقعي على طول الانبوب .بينت نتائج الحسابات ان الشريط الملتوي يُحسن القيمة المتوسطة لرقم نسلت بمقدار (75.2%) ، (68.8%) ، (49.8%) ، (40.3%) ، (16.7%) لنفس الكمية من صرف الوقود (7. 6.16 4.5 3.2 1.6 كغم /ساعة) على التوالي .تم استنتاج مجموعة من المعادلات لحساب القيمة المتوسطة لرقم نسلت على طول الانبوب فضلا عن احتساب قيمة متوسطة لرقم لنسلت في منطقة الجريان التام التطور للانبوب الفارغ والمدخل فيه الشريط الملتوي .

INTRODUCTION

Fire-Tube Boilers (FTB) are the most common heating devices that transfer heat from the combustion gases (flue gases) to water in order to have hot-water up to 3000 kW or saturated steam ranging up to 25 ton/h at 25 bar. In fire tube boiler, the furnace is filled with flame and hot gases, while tubes are filled with hot gases. These tubes and furnace are submerged in same water, giving the boiler name - fire tube boiler.

The efficiency of the first design of FTB until 1985 is very low, up to 70% due to utilizing too many tubes, too much refractory, and in many cases too small furnace. All of this as a direct consequence of poor knowledge of heat transfer inside FTB. After 1985 new FTB design starts, energy saving and reducing fuel consumption has been studied extensively, taken into consideration air pollution. Modes of Heat Transfer in Fire Tube Boilers are divided into: (1-) radiation which is the main mode of heat transfer in furnace, (2-) convection which represent up to 95% of heat exchange in hot gases tubes (the influence of radiation will be lower due to lower temperature and smaller diameter) (Advanced Boiler Technology Group, 2002).

Normally enhanced HTC techniques in FTB is used with the reversed flame furnace, because: (1) temperature of exit flue gases from the reversed furnace ranged between (600-700ºC) which assist to use turbulator material actually cheaper than if the flue gases exit temperature from furnace was (900-1100ºC) in the other type design. (2) Low overall pressure drop in two pass reversed furnace boiler than the other types of boilers.

Turbulators used inside tubes of FTB to improve the turbulent convective heat transfer coefficient in the gas side, since the heat transfer coefficient on the outside is very high with boiling water. The overall objective in this application is to improve the boiler efficiency, although other factors such as (pressure drop), (air-fuel ratio), (changes in the water side heat transfer coefficient), (fouling), and (manufacturing cost) are also important.

Turbulators were appeared in different shapes, like: twisted tape (helical), coiled-wire (spiral), bent-strip, bent-tab, louvered strip, conical ring, and truncated half-cylindrical surface…etc.

An inappropriate assessment of a turbulators' impact on pressure drop can cause the choking of the burner because its fan would be unable to overcome the increased pressure drop in the boiler due to an inaccurate assessment of the turbulators' combined effect on heat exchanger and pressure drop. Today twisted tape and coiled wire turbulators are the most widely used in FTB.

The kind of turbulator that used in the present work is twisted tape insert. Twisted tape inserts can be used in all tubes sizes of FTB (from 38mm OD to 89mm OD).

LITERATURE REVIEW

Junkhan *et al.* (1985) studied experimentally three commercial turbulator inserts to determine the thermal – hydraulic performance in fire-tube boiler. Two types of bent-strip inserts, and one twisted tape with width slightly less than tube diameter. The twisted tape has width (66mm), thickness (1.4mm), length (1.892m), and pitch (H=0.712m) for one full twist (360º), (twisted ratio y= (H/D) = 10.48). Test tube of (76mm) OD, (67.9mm) ID, and 1.823m long is made of carbon steel. The water calorimeter for cooling the gas is made of (6.35mm) ID copper tubing wound around the tube and soldered at the outside. An electrically heated flow facility was developed to deliver fully developed velocity profile hot air at a temperature about (170ºC) before entering the cooled steel tube instrumented to derive sectional average heat transfer coefficients for four regions of tubes. The calorimeter tubing was connecting as four separate segments in series, with water temperature measured at inlet and outlet of each segment. The measurable temperature rise of (8ºC) a cross each segment (455mm), (42ºC between inlet and exit water), and nearly isothermal tube wall. The heat transfer enhancements for these three inserts were measured to be (135%, 157%, and 65%) over a plain empty tube at Reynolds number of (10,000), while the corresponding increase in pressure drop were (1100%, 1000%, and 160%) at the same Reynolds number. For twisted tape (y = 10.48) this correlation was predicted:

$$(Nu_{avg}) = 0.122\ Re^{0.649}\ (T_b/\ T_s)^{0.45} \quad (1)$$

In order to identify the effect of the inserts on the flow characteristics in a fire-tube boiler application , Nirmalan *et al.* (1986a) described the initial thermal-hydraulic and flow visualization studies of seven different geometrical variation of one type of bent-strip insert to enhance turbulent heat transfer in tubes, with particular application to fire-tube boilers. The same test rig that adopted by Junkhan *et al.* (1985) was used. Heat transfer coefficient increases from (175% to 285%) at Reynolds numbers of (10,000) with corresponding pressure drop increase of (400% to 1800%). A preliminary correlation of these data was given. Increasing the number of contacting points would appear to increase the heat transfer coefficient, however with larger increase in pressure drop due to:

a- In the visual observations indicate that the flow disturbance is most sever where the bent strip comes in contact with the tube wall while the flow remains relatively intact in the region where the bent-strip does not touch the wall.
b- Conduction heat transfer from turbulator to tube wall in contact points because turbulator temperature can be consider equal to flue gases temperature.

In a subsequent study, Nirmalan *et al.* (1986b) tested experimentally three new geometrical variations of one type of bent strip, also with the same test rig used by Junkhan *et al.* (1985). He studied the insert entrance region and find that an insert length of 1.5 times pitch is necessary to obtain fully developed enhanced heat transfer conditions. Correlations to predict the average heat transfer and pressure drop are given. To differentiate the wall and core regions, one insert was cut apart to provide core and wall inserts that were tested separately. This investigation also indicates that the core region of the insert is responsible for the major part of the heat transfer augmentation.

Junkhan *et al.* (1988) studied two configurations of inserts, bent – tab inserts, and bent –strip insert, used the same test of (Junkhan, *et al.* 1985). The maximum heat-transfer enhancement for bent strip inserts of about (300%) was achieved, but this was accompanied by a pressure–drop increase of about (1800%). Junkhan suggest a new insert, termed a “bent tab” insert, was design based on results of the core and wall regions test from the partial–insert studies by Nirmalan *et al.* (1986b). The favorable enhancement is available in Reynolds number range of (3000 to 30,000) under a constant pumping power constraint. However, under a constant pressure drop constraint, favorable enhancement is available only in the lower Reynolds number range of about (3000 to 5000).

Ayhan and Demirtas (2001) studied experimentally five different types of turbulator inserts for FTB contain 200 tubes. Four new types of turbulator consisted of truncated conical ring are tested, it was found that turbulators increase the boiler efficiency from (8% to 12%). A fifth new turbulator, consisting of a truncated half-cylindrical surface and placed in tandem with flow direction, provided a (4%) increase in the boiler efficiency. It was also shown that there was no need to use an excess fan for the flue gas in the chimney because of the very low pressure drop in the new types of turbulator.

Neshumayev *et al.* (2004) investigated the heat transfer of the twisted tape, straight tape and combined turbulator in the gas heated tubes of fire-tube boilers by measuring the temperature of the flue gas at the input and output of the tubes array, and also the temperature of cooling water at the input and output from boiler as well as the volume flow rate of the cooling water are recorded. Comparison of the experimental data for the twisted tape with correlation by Manglik and Bergles shows the agreement within (18%). The mean heat transfer of the combined turbulator is higher than the mean heat transfer for the twisted tape and the helical-wire-coil insert cases.

EXPERIMENTAL APPARATUS AND PROCEDURES

A novel model of Fire-Tube boiler design is built up to operate a single hot gases tube in order to study the actual process system that existed in each part of Fire Tube-Boiler separately, for fuel consumption between (1.6 kg/h to 7 kg/h) that produce range of flue gases tube specific gas flow rate from (5 to 15kg/m².s) which was represent the popular range from (5 to 25kg/m².s) that was specified in BS standard 2790 (1989) for tube of FTB.

This model is named as (single tube FTB) as shown in Fig. (2). In this work only the tube part for both empty and inserted plain tube will be studied.

The test rig consists of a burner, combustion chamber, test tube, smoke chamber and chimney, and cooling water devise, each will be described separately.

Description of Test Rig Parts

Referring to Insayif, 2008 the test rig can be classified as follow:

The Burner

Burner is a device, which produces multiple flames' dimensions in the combustion chamber according to the way of burning injected fuel in the furnace. The test rig burner consists of a centrifugal air blower delivered maximum volume flow rate (3.5 m³/min), (13000 rpm), (650W), and 50mm outlet nozzle diameter. The volume flow rate can be adjusted by varying motor speed. The burner case also contains a circular shadow sight glass of (40mm) diameter in order to inspect the flame inside the chamber. An oil pump was operated by a motor of 2750 rpm, 220V, and 150W named (OLBRENNER MOTOR) SUNTEG oil pump France made contains adjustable screw to control the fuel oil pressure in the discharge line and hence the fuel mass flow rate.

Combustion Chamber

The combustion chamber was designed as a reversed flame chamber type. Depending on the flame dimensions furnace dimensions (780mm) length, and (360mm) inside diameter were specified from (RIELLO S.P.A. 2001), and (8mm) wall thickness from (ASME, 1989). The cylindrical shell of furnace was surrounded with a jacket filled with water flow to cool the furnace shell as a tube and shell heat exchanger. The cooled door was designed from two ellipsoidal different size head welded on the ring forming a space filled with water enter and exit from specified opening and nozzle in the upper and lower part from cold head. Exit hole (D = 110mm) was made on each of these two heads to exhaust the reversed gases from the combustion chamber through two concentric 90º elbows to the hot gases tube. Concentric elbows conserve the internal elbow (contact with hot gases) from thermal damage, the cold water coming from cold door is allowed to flow through the annuals space of the elbow.

Test Tube Assembly

The basic part in the present work is the test tube, which was divided to the hot gases tube, and shell. The hot gases coming from the combustion chamber flow inside hot gases tube which is inserted inside a shell tube. (15mm) annulus space was filled with water to simulate the control volume around one tube of tube bundle distribution in FTB (staggered arrangement).

A seamless carbon steel tube (A192) according to ASME standard was used as a hot gases tube (same material and dimension as actual FTB). Dimensions of hot gases tube were (52mm ID, 60mm OD, 2470mm L) and that of shell were (90mm ID, 94mm OD, 2440mm L).

The test tube was connected to the chamber and smoke chamber by a flanges gave the ability to separate the test tube from cold door and smoke chamber while in the actual the hot gases tube ends welded directly with front and rear tube sheet. The tube was welded from front face of flange such that to simulate the actual case in FTB where the tube was welded to the front face of tube sheet.

Temperature and pressure measurements were taken at 13 selected positions along the test tube by inserting the instruments through right and left bushes at these positions. The bushes were made from carbon steel of dimensions (12mm ID, 16mm OD, and 16mm L).

As shown in Fig. (2) bushes that used for thermocouples fixed at angle 180 degree along the tube, while static pressure bushes fixed at angle (0°) along the tube. At the centre point of static pressure bushes 1mm diameter drilled and flashing from the inner and outer of the tube wall.

A carbon steel shell tube (2mm) wall thickness, (90mm) ID, and (2440mm) length was made from two semicircular parts welded together to permit welding bushes upper end on the shell tube in which the temperature and pressure measuring instrument were inserted.

On each of the thirteen axial positions, at angle 90º fixed thirteen thermocouples to measure the water temperature at each of these thirteen points in direct contact with water.

Inlet and outlet water flow nozzles (ID = 17mm) were welded to the beginning and ending of the shell respectively. Finally, the test tube shell was brushed, painted and insulated with 2-inch glass wool.

Smoke Chamber and Chimney

A carbon steel A283 smoke chamber (460mm length, 160mm OD, and 6mm thickness.) was manufactured to provide approximately the same space to the exit gases per tube in the actual rear smoke chamber for range of fuel consumption (1.6kg/h to 7kg/h) per tube at FTB. A tube 52mm ID, and 150mm length was welded to this smoke chamber at one end and to the flange at the other end. The chimney consists of two parts, pipe of (OD =114mm, 1500mm height) as a base connected on it through reducer a tube (50mm ID, 2500mm height) to gave Reynolds number ranges accepted with the actual chimney in the fuel consumption from (1.6kg/h to 7kg/h) per tube.

Cooling System Device

Water is used as a cooling fluid. Cooling system in this work was designed as that for hot water FTB. The calculations showed that it is more economic to preserve constant temperature difference between inlet and exit water flow by blow down part of hot water from furnace shell exit and compensate of it with fresh water at 25 ºC in the storage tank to maintain a constant inlet temperature during steady state operation.

25m³/h, 32m head water pump was used to circulate the water between the storage tank and combustion chamber and test tube. Pressure gage fixed on the furnace water shell. There was four branches out from the water pump discharge line and one suction line connected with lower part from the storage tank. Every branch from the discharge line is connected with inlet nozzle that fixed in different positions depended on the cooled part from the test rig.

Twisted Tape

Carbon steel tape of (50.5 mm) width, (2470mm) long, and (3mm thickness) is twisted by machine to twisted ratio (y = H/D = 11.15), where H is the pitch of full twist (360º) and D is the tube diameter as shown in Fig. (1).

Measurement Instruments

Mass Flow Rate Measurements

Air Mass Flow Rate Measurements

A square edge orifice plate with flange tapping is used in the test rig to measure the pressure drop across the orifice plate to calculate air mass flow rate. The specification of this orifice plate are: Stainless steel square edge flange taping orifice from CRANE company, flange type, DN50 PN16 with ß equal to 0.735 where ß = d/D and (d) is the throat diameter and (D) is the inside upstream pipe diameter.

On each pressure tap, needle valve was used to control the pressure tap and hence the manometer reading. ISO 5167, 2003, parts 1, and 2 are used to specify the orifice recommendations. (1050mm) upstream length, (960mm) down stream length from the orifice device and 90 º elbows was fixed between (790mm) blower tube and upstream line as shown Fig. (2).

Based on ISO 5167, 2003, parts (1, and 2) a bundle of 19 tube straighter is fixed inside the upstream line to reduce the recommended upstream tube length required to reach developed flow. The straighter was manufactured from 19 copper tube (each of 8mm ID), and 134 mm length soldered together as a bundle.

Fuel Mass Flow Rate

The fuel mass flow is an important parameter, since it affects the air mass flow rate required in combustion, then the flue gases Reynolds number in the test tube. A float meter type (HEINRICHS MESSGERATE) was used to measure fuel volume flow rate in the range (1 L/h to 10 L/h water at 20 ºC).

This float meter was fixed between fuel pump and fuel tank. Also it was calibrated using different cylinders (from 10mL to 500mL), and a stop watch of accuracy (0.01s).

Water Mass Flow Rate

Float type flow meter (BLUE – WHITE F- 400), (2 to 20 LPM, Sp.Gr.1) installed on the outlet nozzle at the end of test tube assembly in vertical position is used to measure water volume flow rate and hence water mass flow rate is obtained through multiply the volume flow rate by water density at exit temperature. Calibration for float meter scale was done by using cylinder of volume (1.5 L) and stop watch of accuracy (0.01 sec) at different fuel consumption (1.6 to 7kg/h) depending on exit water temperature.

Temperature Measurements

Thirty-nine type K thermocouples were used in the test rig. Thirteen thermocouples were used to measure water temperature along the shell, and another thirteen thermocouples were used to measure the outside wall temperature along the hot gases tube. Two thermocouples were used to measure hot flue gases temperature, one at inlet and the other on outlet test tube.

Temperature of two Teflon ring faces were measured by two thermocouples fixed on each face for both front and rear Teflon ring. Inlet and exit water temperatures from test tube were measured by installed thermocouple in the inlet and exit nozzle. Water outlet temperature from furnace shell, rear wall, cold door, ambient temperature, and upstream orifice line were measured also.

All these type K thermocouples have the same length (1.5) m and were connected to the selector switch box. The selector switch box have 20 switches, every one have three manual stack position, upper, lower, and middle positions. Nineteen thermocouples were connected to the upper position and other 20 thermocouple connected to lower positions. In the middle position, all the 38 switches were connected in parallel with wire type K which is going to reference junction point at (0 ºC) with copper wire. Positive and negative copper wires are going to the digital voltmeter reading from 0.1 DC mV to 200 DC mV (Zemansky and Richard, 1982).

Depending on using 0ºC (ice and water kept in cold store) and 100ºC (water boiling) the (39) thermocouples were calibrated to temperature range (0 – 100ºC). For temperature large than 100 ºC a Tempilasik made in USA was used to calibrate the thermocouples for temperature range reach (800ºC).

Manometer

Two kinds of manometers were used, inclined and vertical manometer. Inclined manometer used to measure the pressure differential across the orifice, and vertical manometer used to measure the static pressure in the upstream line.

MATHEMATICAL MODEL

This part describes the calculation procedure that is used in order to reach locally thermal – hydraulic calculations, and its average values for developed region and the mean value for developing and fully developed region and for more details go to Insayif, 2008.

The energy balance and the subsequent analysis are performed with the following assumptions:

1- The heat exchanger is insulated from its surrounding, in which case the only heat exchange is between the hot and cold fluids.

2- Axial conduction along the tube is negligible.

3- All fluids temperatures used in the calculations are bulk temperatures.

4- Potential and kinetic energy changes are negligible.

5- Neglecting fouling effect.

Combustion Products

Referring to (Warga, 1999) the chemical composition of light oil #2 is (C= 87.62%, S = 0.12%, H2 = 11.95%, O2 = 0.26%, N2 = 0.05%). The complete combustion process produces a mixture of gases, (Carbon dioxide, Water vapor, Nitrogen, Sulfur dioxide). The weight percentage of these components depends on the weight percentage of the fuel chemical composition (Chattopadhyay, 1998).

$$C + O_2 = CO_2 + \text{heat } (408.8 \text{ kJ/mol}) \quad (2)$$

1mole 1 mole 1 mole

$$S + O_2 = SO_2 + \text{heat } (292.2 \text{ kJ/mol}) \quad (3)$$

1mole 1 mole 1 mole

$$H_2 + 0.5O_2 = H_2O + \text{heat } (242 \text{ kJ/mol}) \quad (4)$$

1mole 0.5 mole 1 mole

Thermo-Physical Properties of Combustion Products:

Calculation of the combustion gases physical properties, (density, dynamic viscosity, thermal conductivity, specific heat) is the first step in calculating the amount of heat released from the combustion gases flow inside heat exchanger tubes. Least squares method (polynomial regression) was used in (Graphing Advantage Plus-Curve Fitting Program) in order to convert the gases physical properties tables data to equations that will be need in the program that build in Microsoft Excel 2003 to perform all the calculation and graphs where the mixture gases physical properties were calculated from TEMA, 1988.

$$\rho_{\text{mixture}} = \sum_{i=1}^{n} \rho_i X_i \quad (5)$$

$$\mu_{mixture} = \frac{\sum_{i=1}^{n} \mu_i Y_i (M_{wi})^{1/2}}{\sum_{i=1}^{n} Y_i (M_i)^{1/2}} \quad (6)$$

$$Cp_{\text{mixture}} = \sum_{i=1}^{n} Y_i Cp_i \quad (7)$$

$$K_{mixture} = \frac{\sum_{i=1}^{n} K_i Y_i (M_{wi})^{1/3}}{\sum_{i=1}^{n} Y_i (M_i)^{1/3}} \quad (8)$$

where the mean combustion gases physical properties can be calculated by integration the local physical properties along tube length.

Water Temperature Gradient

Along the shell side of heat exchanger, thirteen local water temperatures were measured beside the inlet and outlet temperatures of water.

Local thermal calculation is needed to predicate the bulk water temperature in any point along the test tube. The only suitable curve fitting convenient to water temperature along heat exchanger length is the logarithmic curve and to keep the bulk temperature along this curve, only the inlet and exit water temperatures are used to find equation constants (a and b).

$$T = a \,\text{Ln}\,(X/D) + b \quad (9)$$

Tube Wall Temperature Gradient

The outer surface of flue gases tube wall temperature is measured by thirteen thermocouples located along the tube length.

Since the difference between the water inlet and outlet temperatures are approximately (25^{O}C) along cooling tube length, thus uniform wall surface temperature will be considered (Junkhan, *et al.* 1985). Referring to (Kays and London, 1964) the cooling of very high temperature gases by a liquid can usually be approximated by a constant wall temperature due to the relative thermal resistance and relative capacity rates. Also local thermal calculation program is needed to predicate the tube wall temperature at any point along the test tube. Linear curve-fitting fit tube wall temperature as:

$$T_w = a + b(X/D) \quad (10)$$

Air and Combustion Gases Mass Flow Rate

According to (ISO 5167, 2003, part 1,2) air mass flow rate is calculated from:

$$m=[\acute{C}/(1- \beta^4)^{0.5}]*E*(\pi/4)*d^2*(2\Delta p* \rho_1) \quad (11)$$

Square edge orifice flange taping is used in this work. The combustion gases mass flow rate is equal to the summation of fuel and air mass flow rate that supplied to combustion chamber.

Local Thermal Calculation Procedure

In order to calculate the thermal local values (Nusselt number, heat transfer coefficient, heat flux), the test tube length is divided theoretically into segments of (1mm) length each is considered as a heat exchanger connected in serious (McAdams, 1954). Thus when applying the energy balance between the hot and cold side for every 1mm segment, the obtained data in the outlet segment used as the inlet condition to the second segment and so on. Thus segmental heat balance can be written as:

$$Q_w = Q_g$$
$$m_w*Cp_w(T_{wout}-T_{win})= m_g*Cp_g(T_{g\ in}-T_{g\ out}) \quad (12)$$

This calculation is repeated for each segment of the tube to determine local combustion gases bulk temperature.

The gas temperature variation obtained from the above procedure shows a logarithmic temperature distribution along the tube length.

$$T_g = a \ Ln\ (X/D) + b \quad (13)$$

Combustion Gases Heat Flux

The local heat flux can be calculated such as:
From eq.12 [$Q_w = m_w * Cp_w\ (T_{w\ out} - T_{w\ in})$] where Cp_w is calculated at segment average temperature.

According to (Kays, and Perkins, 1973) the constant heat flux Nusselt number is always greater than the constant surface temperature Nusselt number. The difference becomes quite negligible for ($Pr > 1$) and since the length of segment is very small (1mm) and combustion gases Prandtl number have a range from (0.73 to 0.78), thus will assume constant heat flux distribution along the (1mm) length segment, and thus when dividing the segment heat flux that calculated from the upper procedure by the segment heat transfer area (π* D*L) will be obtained local heat flux. By using Microsoft Excel program this procedure will repeated for all the (2351) segment and will specified the local heat flux distribution along the tube length.

Heat Transfer Coefficient

Since the local heat flux is constant across the test tube section, thus using Newton's equation:

$$Q = U_i * A_i * (T_g - T_{SO}) \quad (14)$$

$$h_{xi} = q_x / [(T_g - T_{SO}) - (q_x * R)] \quad (15)$$
where $R = [(r_i / k) * Ln(r_O / r_i)]$.
The derivation of eq. 15 was clarified in Insayif, 2008.

Local Nusselt number can be calculated from:
$$Nu_x = h_x * D / k_x \quad (16)$$
where k_x local thermal conductivity.

Reynolds Number

Combustion gases local Reynolds number (Re_X) can be calculated as:
($Rex = 4\ m_g/ \pi\mu_x D$)

Where m_g is equal to the summation of fuel consumption rate plus air mass flow rate that calculated in eq. 11 and μ_x is calculated in eq. 6, while the mean value is calculated as:

$$Re_{mean.} = (1/L) * \int_{0}^{L} Re_x \, . \, d_x \qquad (17)$$

Mean and Average Combustion Gases Heat Transfer Coefficient

Since the mean heat transfer coefficient in heat exchanger application is more benefit than the local heat transfer coefficient, thus the mean heat transfer coefficient along the tube length and average heat transfer coefficient along the fully developed region is calculated by integrating the local heat transfer coefficient curve as below:

$$h_{mean}=(1/L)* \int_{0}^{L} h_x \, d_X \qquad (18)$$

$$h_{avg.}= (1/L) * \int_{10D}^{L} h_x \, . \, d_X \qquad (19)$$

Experimental Uncertainty

The tests on the test rig were run under the same conditions for every fuel consumptions rates. The average difference between test tube thermal outputs calculated from repeated tests under the same conditions was less than ±0.8 %.

The relative uncertainties in Nusselt number, Prandtl number, and Reynolds number, were estimated from a typical run at fuel consumption rate (7kg/h), according to Moffat (1985), and Kline (1985).The following uncertainties are typical of the uncertainties that can be expected in other test runs: Nu_{mean}= ± 6%, Re_{mean} = ± 3.5%, Pr_{mean} = ± 3.5.

RESULTS AND DISCUSSIONS

From the experimental measurements of (water, wall tube, inlet and exit combustion gases temperatures, water and combustion gases mass flow rate, combustion gases static pressure), the combustion gases local: physical properties, combustion gases temperature, heat flux, and convection HTC, have been predicted.

Since the abrupt contraction (sharp–edged contraction) used as the entrance in the test tube as in the actual FTB without using calming length, combined thermal and hydrodynamic boundary layer will grow starting from zero thickness at ((X/d)=1.173).

From (Kays and Crawford, 1980) the abrupt contraction entrance cause flow contraction and then re expansion during the first diameter of tube length, and from (Kays and Perkins, 1973) when abrupt contraction used, boundary layer separation (stall) and very high convection HTC will gained after the boundary layer reattaches. So that the cooling starting point taken at ((X/D)= 1.173) in this work test tube.

Fig. (3) shows the variation of combustion gases HTC along empty and inserted test tube for different fuel consumptions rates. Referring to this figure, maximum convection HTC will be at the ((X/D) = 1.173), (cooling start point) and decreasing rapidly in exponential curve until arrive ((X/D) =10) from which the decreasing in convection HTC in the flow direction become very smooth. So that ((X/D) =10) point is the end of the thermally developing region and the beginning of the thermally fully developed region. In this work analysis, the convection HTC did not take a constant value in the fully developed region because of the change in combustion gases physical properties along the tube as the combustion gases temperature decreasing in the flow direction.

Referring to (Holman, 1992) the assumption of constant convection HTC throughout the heat exchanger is serious because of entrance effects, fluid viscosity, and thermal conductivity changes, etc. Due to inserted twisted tape (with twisted ratio H/D = 11.15) inside the test tube, the HTC curve for inserted tube is above that for empty test tube for same fuel consumption rate

Since the dimensionless Nusselt number value equal to the convection HTC multiply by inside tube diameter over combustion gases thermal conductivity, thus local Nusselt number will take the same behavior of convection HTC as shown in Fig. (4), because of linear proportionality between Nusselt number and convection HTC and the smaller change in combustion gases thermal conductivity value along the cooling tube length.

Fig. (4) shows the variation of combustion gases Nusselt number along empty and inserted test tube for different fuel consumptions rates. As shown in this figure, local Nusselt number with twisted tape is greater than the empty tube for the same fuel consumptions, because of:

1- The increase of flow path length.
2- Producing rotational and secondary flow, which reduces the thermal resistance.

Fig. (5) shows mean Nusselt number variation with mean Reynolds number for combustion gases flow inside empty and inserted test tube. It is clear in this figure the enhancement in mean Nusselt number increase as Reynolds number increase, where there is linear proportionality between fuel consumption and mean Reynolds number.

The percentage increase for mean Nusselt number of combustion gases due to inserted twisted tape inside the test tube for same fuel consumption rate are (75.2%), (68.8%), (49.8%), (40.3%), and (16.7%) for fuel consumption (7kg/h), (6.16kg/h), (4.5kg/h), (3.24kg/h), and (1.6 kg/h) respectively.

Extracted Thermal Correlations

In order to estimate a correlations for the mean and average Nusselt number, an integral to the curves in Fig. (4), were done to find the mean values (from (X/D) > 1.173 to the end of the tube), and average values (from (X/D) > 10 to the end of the tube) of Nusselt , and Reynolds number, where its values were plotted in Fig.s 5, 6, and 7. Convection HTC (h) is functionally connected with the following quantities (Klaczak, 1973):

$$h = f(\bar{u}, \rho, D, \mu, Cp, k, T'_g, T'_s)$$

The dimensional analysis of the foregoing function gives the dependence of four dimensionless criterion numbers:

$$(hD/k) = a\,(\bar{u}\rho D/\mu)^b \,.\, (\mu\, Cp/k)^c \,.\, (T'_g/T'_s)^d$$

$$Nu = a\,Re^{b}\,Pr^{c}\,(T'_g/T'_s)^d \qquad (20)$$

The linear equations with four unknowns were obtained after taking logarithms of this equation .

$$Log(a)+bLog(Re) + c\,Log(Pr) + d\,Log(T'_g/T'_s) = Log(Nu) \qquad (21)$$

Equation (21) is used to calculate the general correlation equations. Here, method of least squares was used, giving:

$$(Nu_{mean}) = 0.011Re^{0.9114}\,Pr^{0.9467}\,(T'_g/T'_s)^{-0.1302} \qquad (22)$$

for twisted tape inserted,((H/D)=11.15), (X/D> 1.173)

$$(Nu_{avg.}) = 0.0103\,Re^{0.8887}\,Pr^{1.2721}\,(T'_g/T'_s)^{-0.1716} \qquad (23)$$

for twisted tape inserted,((H/D)=11.15) , (X/D > 10)

$$(Nu_{mean}) = 1.3864Re^{0.217}\,Pr^{-3.4816}\,(T'_g/T'_s)^{0.5377} \qquad (24)$$

for empty tube,(X/D > 1.173)

$$(Nu_{avg.}) = 2.411Re^{0.1063}\,Pr^{-3.7672}\,(T'_g/T'_s)^{0.5627} \qquad (25)$$

for empty tube , (X/D> 10)

Comparison With the Other Works

In order to make comparison with other works for developed flow in empty test tube, equations listed below were plotted together with correlation eq. 25 in Fig.6.

Comparison equations are:

Colburn equation, which was modified from Dittus-Boelter equation for fully developed turbulent flow (thermal and hydraulic) in smooth tube that given by Kays, and Perkins (1973):

$$(Nu_{avg}) = 0.023\,Re^{0.8}\,Pr^{1/3} \qquad (26)$$

another modification equation for constant surface temperature was given by Kays, and Perkins (1973):

$$(Nu_{avg}) = 0.021\,Re^{0.8}\,Pr^{0.6} \qquad (27)$$

and a new equation for heat transfer in turbulent pipe and channel flow was given by Gnielinski, (1976) :

$$(Nu_{avg}) = \frac{(f/8)\,(Re-1000)\,Pr}{}\,[1 + (d/L)^{2/3}]\,[(T_b/T_s)^{0.45}] \qquad (28)$$

$1 + 12.7\,(f/8)^{0.5}\,(Pr^{2/3} - 1)$

where drag (Darcy) coefficient was calculated from Filonenko equation for isothermal flow

$$f = (1.82\,(\log Re) - 1.64)^{-2} \qquad (29)$$

or from Blasius equation

$$f = 0.3164 / (Re)^{1/} \qquad (30)$$

In addition, comparison with other work for developed flow inside tube inserted with twisted tape is done by plotting correlation (23) and Junkhan equation (1) in Fig. (7) for (pitch / diameter = 10.48).

In Fig.s (6), and (7), correlation (25), and (23) give lower values than other correlations, the reason is related to the use of the following in the present work:

1- Actual combustion gases fluid flow which cause:

a – Higher inlet temperature with range (400-800°C) to test tube, while the others used air with temperature range 270ºC or liquid and modified its equation to gases.

b – Soot covers the inner tube surface, which added thermal resistance decreases the heat transfer, and hence convection HTC.

2- Local values along tube length of temperature, physical properties, and hence heat flux, Reynolds number, convection HTC, Nusselt number, as explained in 13.

3- Parallel flow shell and tube heat exchanger.

4- Friction factor that used in Genleski equation was derived for isothermal flow.

CONCLUSIONS

The following conclusions can be extracted from this work:

The maximum convection heat transfer coefficient is located at first cooling point and decreasing rapidly in exponential trend until arrive ((X/D)=10) from which the decrease in heat transfer coefficient in the flow direction become very smooth. So that (X/D)=10) is the end of thermally developing region and the beginning of the fully developed region. Also the percentage enhancement in mean Nusselt No. due o the inserted twisted tape was (75.2, 68.8, 49.8, 40.3 and 16.7) for fuel consumption (7 kg/hr, 6.16 kg/hr, 4.5 kg/hr, 3.24 kg/hr and 1.6 kg/hr respectively).

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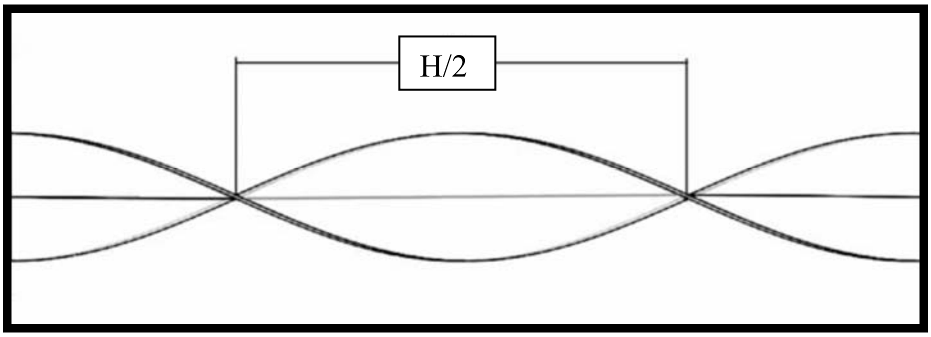
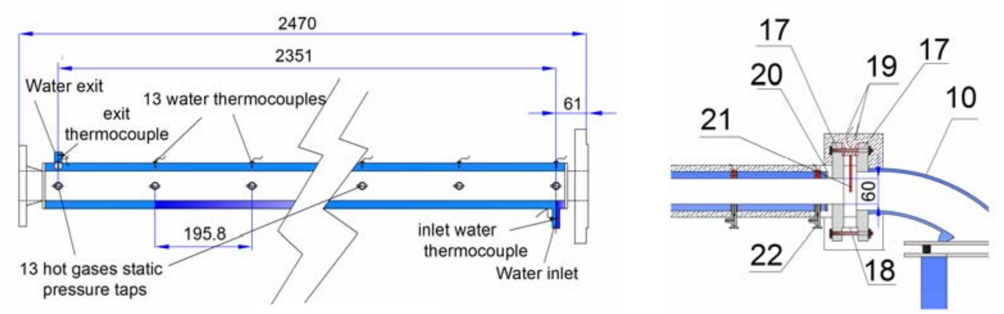
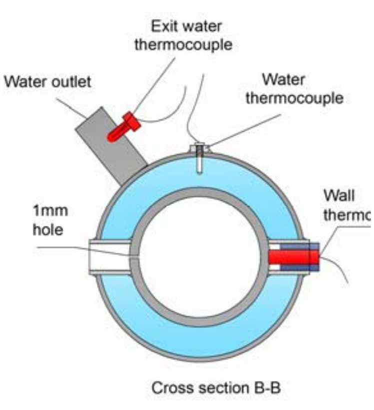


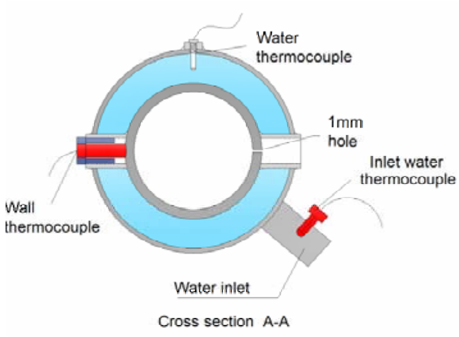
Fig. 1 Schematic Diagram of the Twisted Tape



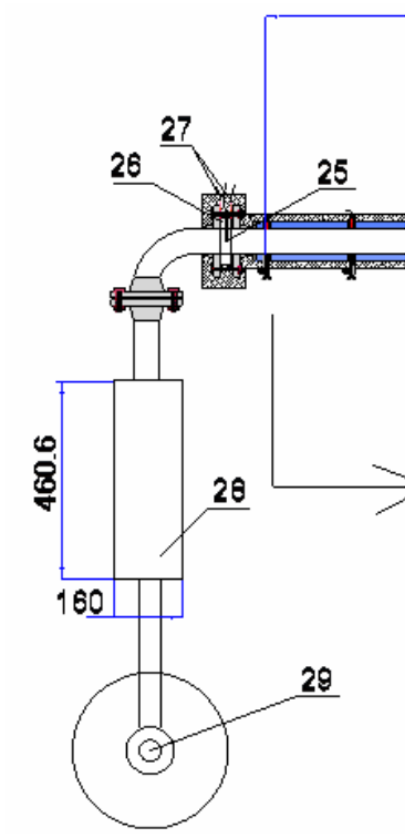
Test tube

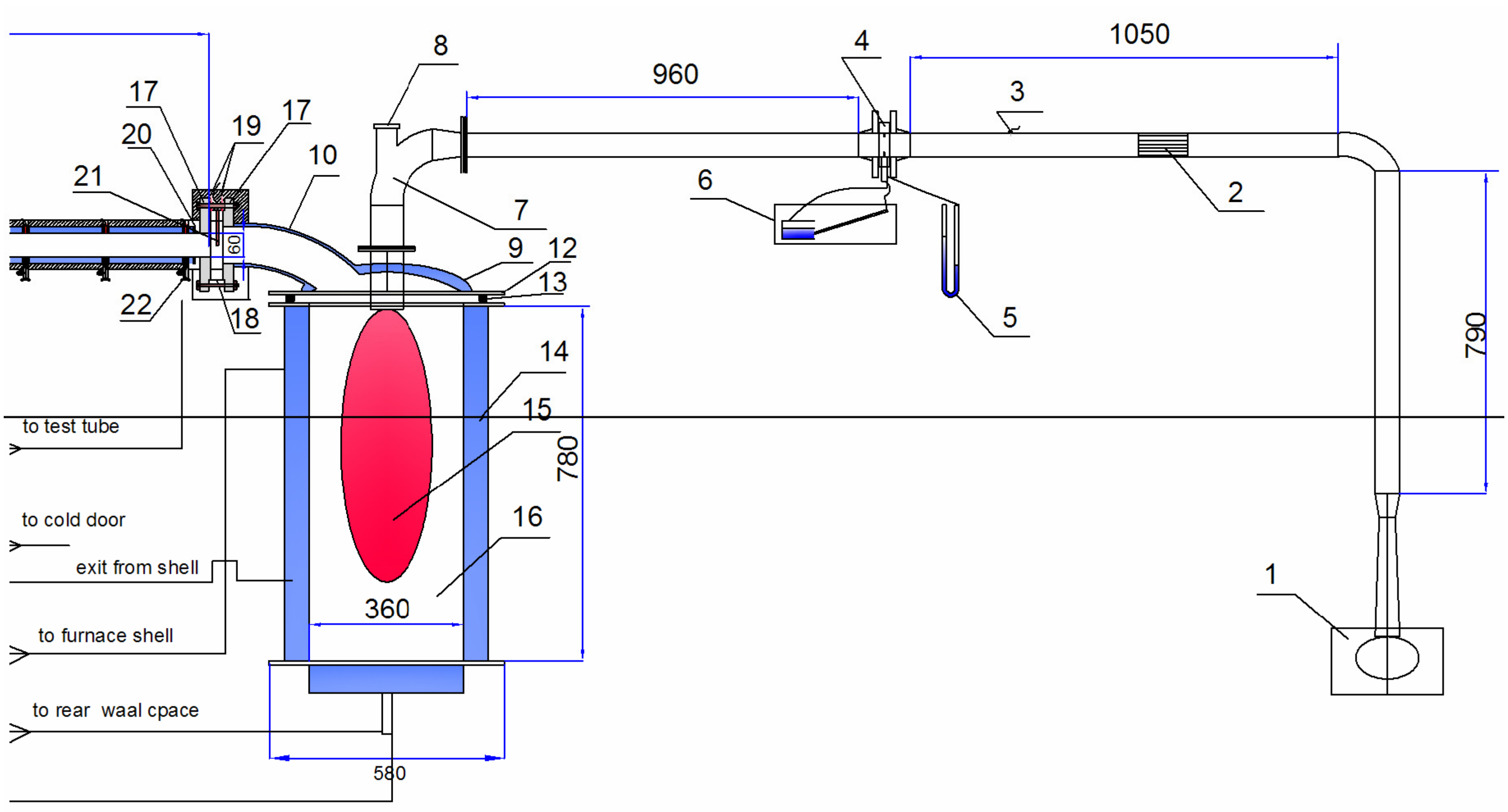


Test tube left connection



Test tube right connection





Note/ All Dimensions are in mm.

Fig. (2)Layout of the Test Rig

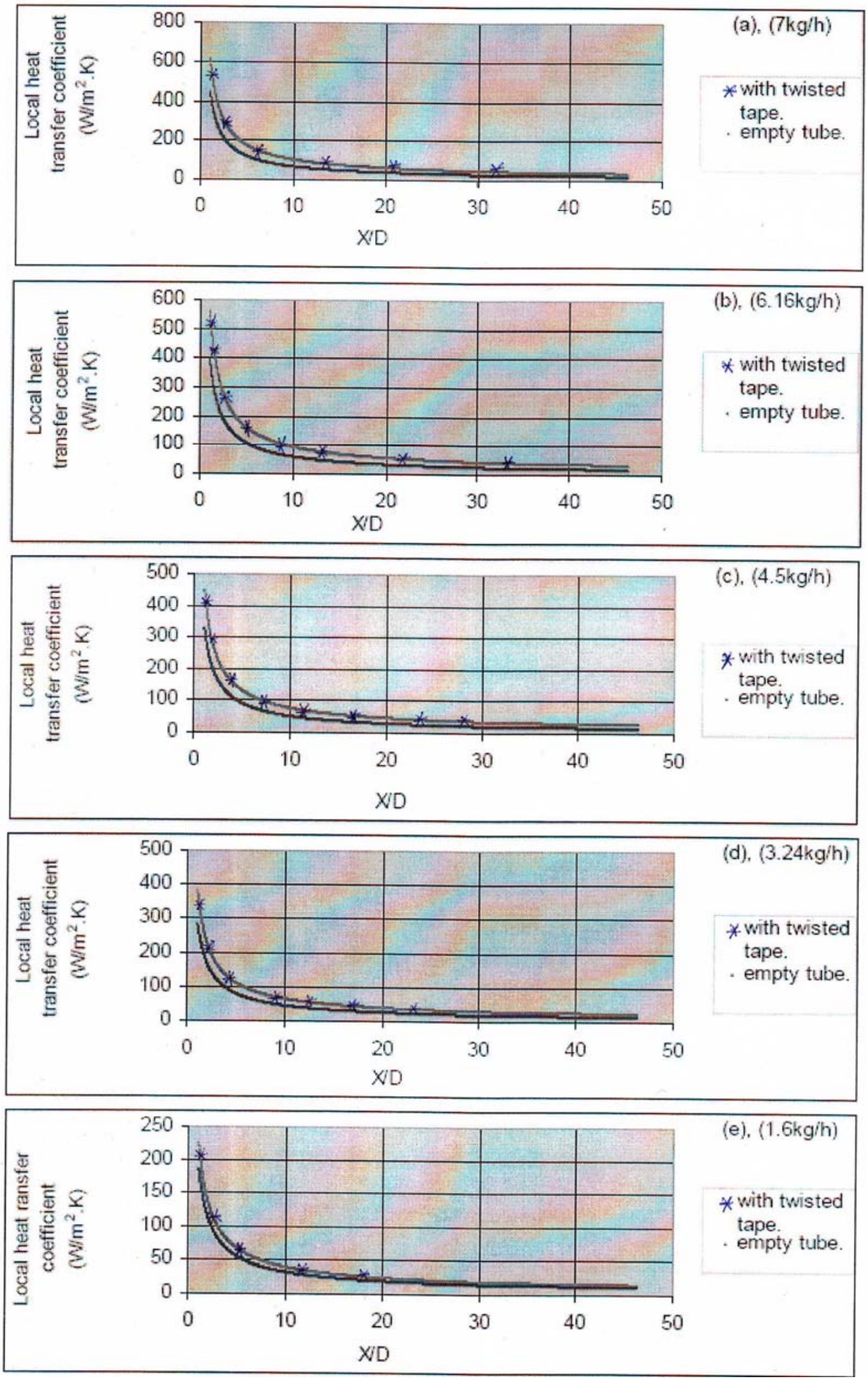


Fig. 3 Variation of Combustion Gases HTC along Empty and Inserted Test Tube for Different Fuel Consumptions Rates.

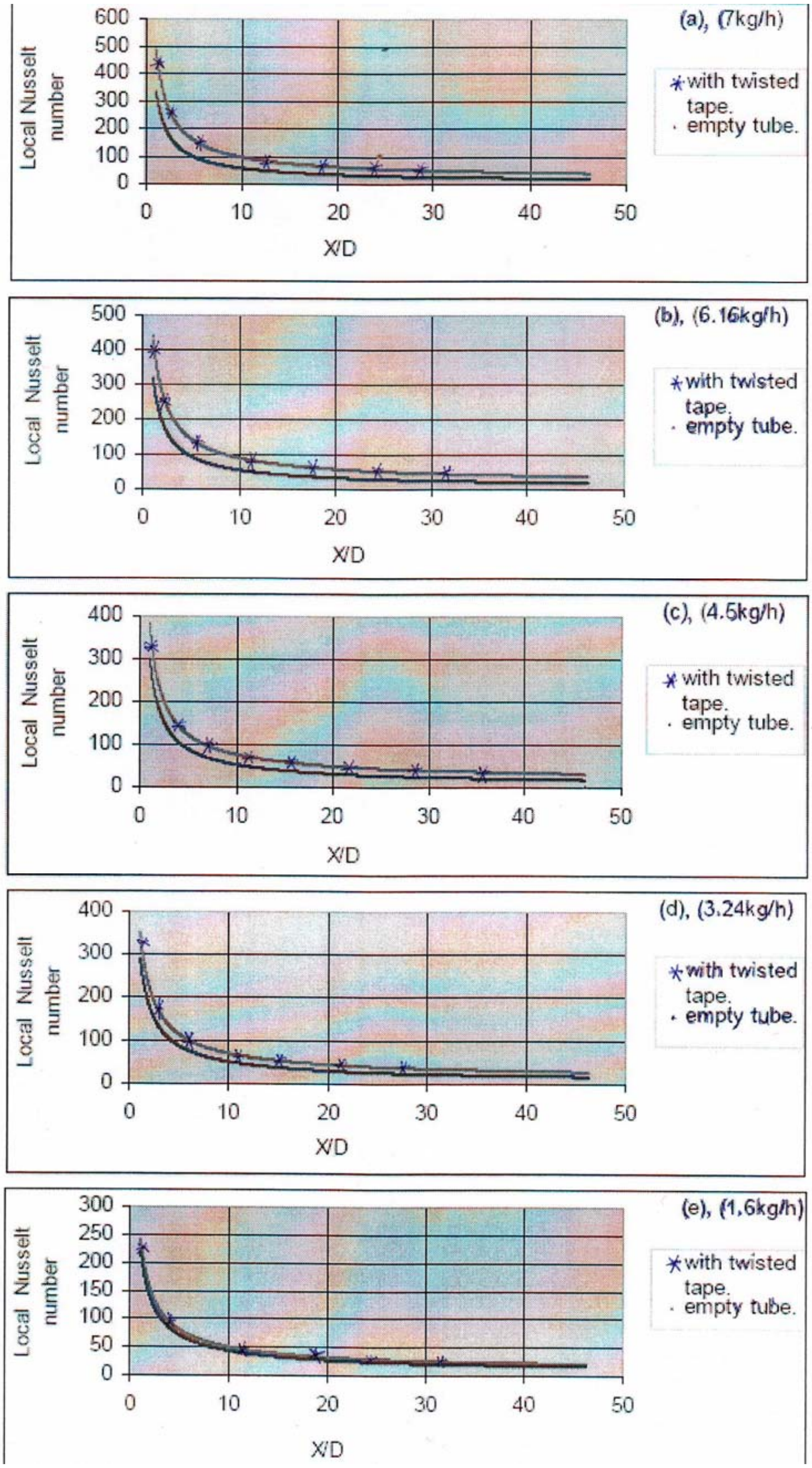


Fig. 4 Variation of Combustion Gases Nusselt Number along Empty and Inserted Test Tube for Different Fuel Consumptions Rates.

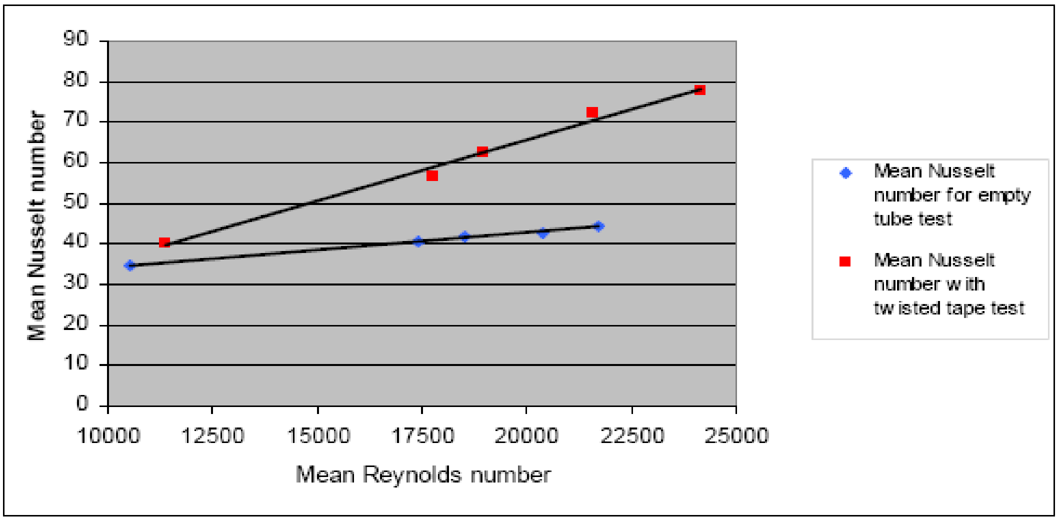


Fig. 5 Mean Nusselt Number Variation with Mean Reynolds Number inside Empty and Inserted Test Tube.

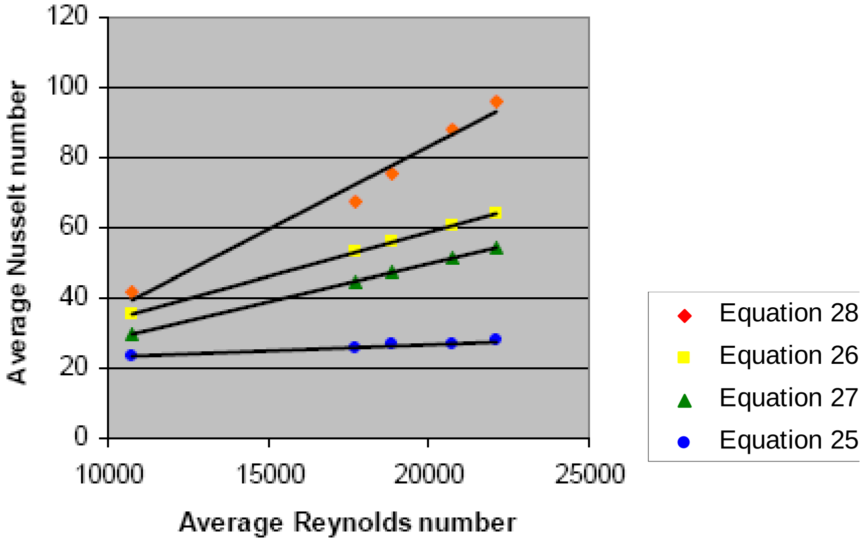


Fig. 6 Variation of Average Nusselt Number with Average Reynolds Number for the Present Empty Test Tube and Others Similar Works.

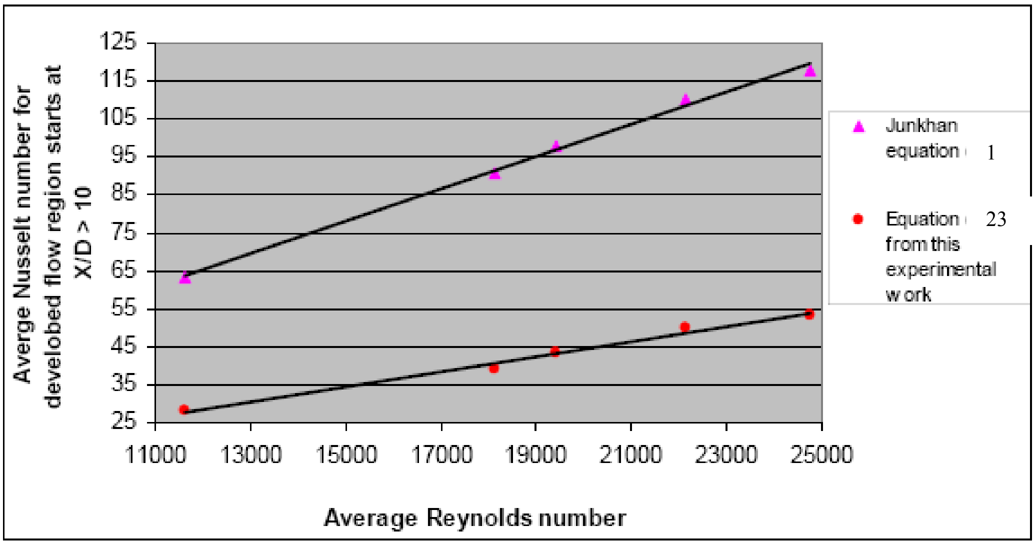


Fig. 7 Variation of Average Nusselt Number with Average Reynolds Number the Present Inserted Test Tube and Other Similar Work.

NOMENCLATURE

$\acute{C}$	Orifice discharge coefficient	Nu	Nusselt number
Cp	Specific heat J/mol.K	Pr	Prandtl number
d	Orifice diameter m	q	Local heat flux W/m²
D	Inside tube diameter m	Q	Heat flow W
E	Expansibility (expansion) factor	r	Tube radius m
f	Darcy Friction factor	R	Thermal resistance K. m²/W
g	Gravitational acceleration m/s²	Re	Reynolds number
G	Tube specific gas flow rate kg/m².s	R_O	Universal gas constant = 8.314 J/mol.K
h	Heat transfer coefficient W/m².K	T	Temperature ºC
H	Pitch of full twist (360º) m	U	Overall heat transfer coefficient W/m².K
k	Thermal conductivity W/m.K	$\bar{u}$	Average cross section velocity m/s
$\acute{K}$	Specific heat ratio	V	gas volume m³
L	Length m	W	Mass of one gas kg
m	Mass flow rate kg/s	X_i	Mass fraction ($X_i = W_i / W_{mixture}$)
mV	Volt/1000 mV	y	Twisted ratio (H/D)
Mw	Molecular weight kg	Y_i	mole fraction ($Y_i = N_i / N_t$)
n	Number of gases content in the flue gases	X	Longitudinal distance m
N	moles number of the gas		

Greek Symbols

ρ	Density kg/m³	β	d/D
μ	Dynamic viscosity N.s/m²	π	22/7

Superscripts & Subscripts

$(\ )_1$	Upstream line	$(\ )_{out}$	outlet
$(\ )_2$	Down stream line	$(\ )_s$	Internal tube surface
$(\ )'$	Absolute value	$(\ )_{SO}$	Outer surface
$(\ )_b$	At bulk temperature	$(\ )_t$	Total gases
$(\ )_D$	For tube	$(\ )_w$	Water
$(\ )_g$	Gases	$(\ )_{w\,in}$	At water inlet
$(\ )_i$	Inside	$(\ )_{w\,out}$	At water outlet
$(\ )_{in}$	Inlet	$(\ )_x$	Local
$(\ )_m$	kind of the gas	$(\ )_{xi}$	Local inside
$(\ )_O$	outside		

Abbreviations

AF	(Air/Fuel) ratio	OD	Outside Diameter
FTB	Fire Tube Boiler	HTE	Heat Transfer Enhancement
ID	Inside Diameter	HTC	Heat Transfer Coefficient

Nomenclatures of Figure (2)

Part No.	Part name	Part No.	Part name
1	Air blower	16	Furnace
2	Straighter	17	4” flange
3	Up stream thermocouple	18	Teflon Ring
4	Orifice plate	19	Thermocouple
5	U-tube manometer	20	Ring
6	Inclined manometer	21	Hot gases thermocouple
7	Burner	22	Static pressure valve
8	Shadow sight glass	23	Insulation
9	Cold door ellipsoidal head	24	Tube wall thermocouple
10	Cold concentric elbow	25	Hot gases thermocouple
11	Flange	26	Teflon ring
12	Cold Door ring	27	Thermocouple
13	Asbestos belt	28	Smoke chamber
14	Water	29	Chimney
15	Flame		

Note/ All Dimensions are in mm.