Shell and Double Concentric Tube Heat Exchanger Calculations and Analysis

Basma Abbas Abdulmajeed
Assistant Professor
College of Engineering - University of Baghdad
e-mail:basma1957@yahoo.com

Fadhil Abed Allawi
College of Engineering - University of Baghdad
e-mail:fadel_eng2009@yahoo.com

ABSTRACT

This study concerns a new type of heat exchangers, which is that of shell-and-double concentric tube heat exchangers. The case studies include both design calculations and performance calculations.

The new heat exchanger design was conducted according to Kern method. The volumetric flow rates were 3.6 m^3/h and 7.63 m^3/h for the hot oil and water respectively. The experimental parameters studied were: temperature, flow rate of hot oil, flow rate of cold water and pressure drop.

A comparison was made for the theoretical and experimental results and it was found that the percentage error for the hot oil outlet temperature was (- 1.6%). The percentage errors for the pressure drop in the shell and in the concentric tubes were (17.2%) and (- 39%) respectively. For cold water outlet temperature, the percentage error was (- 3.3%), while it was (18%) considering the pressure drop in the annulus formed. The percentage error for the total power consumed was (-10.8%).

A theoretical comparison was made between the new design and the conventional heat exchanger from the point of view of, length, mass, pressure drop and total power consumed.

Key Words: new heat exchanger, conventional heat exchanger, length of heat exchanger, mass of heat exchanger, total power expenditure

الحسابات والتحاليل للمبادل الحراري ذو القشرة والانابيب المتداخلة المتمركزة

بسمة عباس عبد المجيد
استاذ مساعد
كلية الهندسة-جامعة بغداد

فاضل عبد علاوي
كلية الهندسة-جامعة بغداد

الخلاصة

ان هذه الدراسة تعنى بنوعية جديدة من المبادلات الحرارية , هو المبادل الحراري ذو القشرة والانابيب المتداخلة المتمركزة.هذا البحث يتضمن الحسابات التصميمية وحسابات كفاءة المبادل. ان المبادل الحراري الجديد صمم على طريقة(kern method) حيث تم استخدام دهن ساخن ذو تدفق حجمي3.6m^3/h وماء بارد ذو تدفق حجمي7.63m^3/h. المتغيرات التي تم قياسها من التجارب العملية هي درجة الحرارة و معدل الجريان للدهن الساخن ومعدل الجريان للماء.

تم اجراء مقارنة بين النتائج العملية والنظرية للمبادل الجديد حيث وجد ان نسبة الخطأ بالنسبة لدرجة حرارة الدهن الساخن الخارجة من المبادل هي (1.6%-), ونسبة الخطأ بالنسبة لهبوط الضغط في ال(shell)هي (17.2%), وكانت نسبة الخطأ بالنسبة لهبوط الضغط في الانابيب المتداخلة هي (39%-) .اما بالنسبة للماء البارد فقد كانت نسبة الخطأ هي (3.3%-) بالنسبة لدرجة حرارة الماء الخارج من المبادل ونسبة الخطأ بالنسبة لهبوط ضغط الماء المار في التشكيل الجديد الحاصل من الانابيب المتداخلة هي (18%), اما بالنسبة لمقدار الطاقة الكلية المصروفة فقد كانت نسبة الخطأ (10.8%-).

هذا البحث تضمن مقارنة نظرية بين تصميم المبادل الحراري الجديد والمبادل التقليدي من حيث الطول, الوزن ,هبوط الضغط ومقدار الطاقة المصروفة.

الكلمات الرئيسة: المبادل الحراري الجديد, المبادل الحراري التقليدي, طول المبادل الحراري, وزن المبادل الحراري, الطاقة الكلية المصروفة.

1. INTRODUCTION

A shell and double concentric tubes heat exchanger is a new invention in heat transfer devices that is used for transfer of internal thermal energy between three fluids at different temperatures (two hot flows and a cold flow H-C-H or opposite C-H-C).

These heat exchangers can be used in many specific applications such as air conditioning, waste heat recovery, chemical processing, power production, distillation, food processing, etc.

The use of these new heat exchangers will enhance the heat exchange and will increase compactness. It means the decrease of the cost and the weight of this type of heat exchanger.

The new heat exchangers are similar to the prior one; the differences are that the simple (or corrugated) tubes with or without fines are now replaced by double concentric tubes. The outside diameter of the envelope of the double concentric tube is the same order that tubes used in the shell and tube heat exchangers.

Principally, the second tube (inner tube) improves heat transfer through an additional flow passage and a larger heat transfer area per heat exchanger length unity.

Also, two new tube sheets are added to serve as flanges for attachment of the channels and their respective channel covers and two distribute a fluid in inner tubes of double concentric tubes. The old two tubes sheets are always used as flanges for attachment of the two channels and distribution of the fluid passing in the annuals passage formed by the concentric tubes.

The typical shell-and-tube heat exchanger is not ideal in terms of its size**.** This contributes to an increased cost of manufacturing and installation, and on top of that, consumes a lot of space.

In this new heat exchanger the application of two different streams is the main one. In this case, the new heat exchanger is compact; it has a less exchanger length (volume) than a shell-and-tube heat exchanger. The new heat transfer area is equal to the heat transfer area of shell-and tube heat exchanger plus of inner tubes heat exchange area.

2. DESCRIPTION OF THE HEAT EXCHANGER

In shell and double concentric tubes heat exchanger as shown in **Fig. 1**, one sees the shell, the three distributors, the three collectors and the channel covers with four tube sheets. The two fluids of same temperature level enter by the first and the third distributor and goes out by the third and the first collector, respectively. The fluid of different temperature level of two other fluids, pass by the intermediate collector (or distributor).

First fluid (the same temperature level or the same nature as the third fluid) enters by the first distributor and passes by the first tube sheet and goes out by the fourth tube sheet and the last collector. Whereas second fluid penetrates into the heat exchanger by the second distributor and crosses the passages of annulus shape formed with the inner tubes and the second tube sheet then it goes out the heat exchanger by the third tube sheet and the last front collector. Third fluid enters the heat exchanger by the third distributor and crosses the heat exchanger at outside of double envelopes (shell side) and goes out the heat exchanger by the first collector in the same way as classic shell-and-tube heat exchangers.

These shell-and-double concentric-tubes heat exchangers are conceived in a different ways, according to present fluids. Tubes can be corrugated tubes with fins. Generally, fluid circulating towards the shell can circulate in multi pass because of the baffles presence. It allows irrigating better all the tubes. There are several types of baffles: segmental baffle, disc and doughnut baffle. Tubes can be arranged in the bundle following a staggered or aligned arrangement. The two fluids of same nature or same temperature level enter by the same heat exchanger side. The different fluid of the two other fluids flowing in the annulus section of the concentric tubes passes generally in the opposite direction of the global circulation of the two other fluids.

3. THEORY OF DESIGN AND ANALYSIS

3.1 Design Consideration

In designing heat exchangers, a number of factors that need to be considered are:

1. Resistance to heat transfer should be minimized.
2. Contingencies should be anticipated via safety margins; for example, allowance for fouling during operation.
3. The equipment should be sturdy.
4. Cost and material requirements should be kept low.
5. Corrosion should be avoided.
6. Pumping cost should be kept low.
7. Space required should be kept low.
8. Required weight should be kept low.

Design involves trade-off among factors not related to heat transfer. Meeting the objective of minimized thermal resistance implies thin wall separating fluids. Thin walls may not be compatible with sturdiness. Auxiliary steps may have to be taken, for instance, the use of support plates for tubing, to realize sturdiness, **Saunders, 1988**.

The optimum thermal design of a shell and double concentric tubes heat exchanger involves the consideration of many interacting design parameters which can be summarized as follows:

Processes

1. Process fluid assignments to three streams shell side or inner tubes side and annulus side.
2. Selection of stream temperature specifications for three streams.
3. Setting shell side, inner tubes side and annulus side pressure drop design limits.
4. Setting shell side, inner tubes side and annulus side velocity limits.
5. Selection of heat transfer models and fouling coefficients for shell side, annulus side and inner tubes side.

Mechanical

1. Selection of heat exchanger TEMA layout and number of passes.
2. Specification two types of tube parameters - size, layout, pitch and material for bundle of inner tubes and bundle formed annulus passages.
3. Setting upper and lower design limits on inner tubes and annulus tubes length.
4. Specification of shell side parameters – materials, baffles cut, baffle spacing and clearances.
5. Setting upper and lower design limits on shell diameter baffle cut and baffles spacing**, John, 1998**.

3.2 Analysis of New Heat Exchanger

The analysis of heat new exchanger is simplified through a number of reasonable and realistic assumptions:

1. Steady- flow.
2. Kinetic and potential energy changes are negligible.
3. The specific heat of a fluid (oil and water) is constant.
4. The axial heat conduction along the tube (inner tubes and annulus passages) is negligible.

5. The outer surface of the heat exchanger is perfectly insulated.

Based on these assumptions, it can be shown through the first law of thermodynamics that the rate of heat transfer from the hot fluid be equal to the rate of heat transfer to the cold one.

The transfer rate to the cold fluid:

$$q_c = m_c Cp_c \Delta T_c \tag{1}$$

The transfer rate to the hot fluid:

$$q_h = m_h Cp_h \Delta T_h \tag{2}$$

The heating and cooling loads of a heat exchanger under operating conditions can be calculated from the above equations. The temperatures of the fluids in a heat exchanger are generally not constant but vary from point to point as heat flows from the hotter to the colder fluid. Even for a constant thermal resistance, the rate of heat flow will therefore vary along the path of the exchanger because its value depends on the temperature difference between the hot and cold fluid in the test section, **Ramesh, et al., 2003.** The evaluation of the logarithmic mean temperature difference (LMTD) for counter and parallel flow streams, respectively by using Eq. (3):

$$LMTD = \frac{\Delta T_a - \Delta T_b}{\ln \dfrac{\Delta T_a}{\Delta T_b}} \tag{3}$$

ΔT_a difference between the hot and cold fluid streams at the inlet to the heat exchanger, while ΔT_b is the temperature difference at the outlet from the heat is the temperature exchanger.

It is convenient to use an average effective temperature difference (ΔT_m) for the entire heat exchanger, defined by (Eq.1):

$$q = U \ A \ \Delta T_m \tag{4}$$

The average temperature difference $\left(\Delta T_m\right)$ can be taken as the same value of the logarithmic mean temperature difference for one tube pass. For more than one tube pass, the average temperature difference $\left(\Delta T_m\right)$, can be calculated by multiplying the logarithmic mean temperature difference by the temperature correction factor as Eq.(5), **Coulson,** and **Richardson, 1998**:

$$\Delta T_m = \left(F\right)\left(LMTD\right) \tag{5}$$

4. EXPERIMENTAL WORK

4.1 Manufacturing and Description

The test section was a shell- and -double concentric tube heat exchanger with dimensions of 1.3 m in length and 1.08 m effective tube length.

The tubes of the conventional shell and tube heat exchanger were replaced with double concentric tubes, to improve the heat transfer through an additional flow passage which gives larger heat transfer

area. The shell and double concentric tube heat exchanger was designed to work with three streams of fluids (two hot flows and a cold one H-C-H or the opposite C-H-C).
The shell and double concentric tube heat exchanger was designed for counter flow configuration, in which the hot oil flows in the inner tubes and also in the shell in opposite direction to the cold water which flows in the annulus section of the concentric tubes side. Thermometers and pressure gauges are connected to the tubes and shell sections.
The heat exchanger constituted with:

4.2 Tubes and Tube Sheet

A circular plate of carbon steel with 10 mm thick was used as the tube sheet. Tube holes were drilled with 6.25mm clearances in dimensions of the holes and hole pitches, with a tip of 45° cut in each hole. Tubes were welded to the tube sheet.

A bundle of 16 carbon steel tubes of 20 mm inside diameter and 25 mm outside diameter was used; the tubes are distributed as a triangular 30° tube pattern. The clearance between two adjacent tubes is 6.25 mm, and the tubes pitch is 31.25 mm.

A second bundle of 16 carbon steel tubes of 6 mm inside diameter and 10 mm outside diameter, were added concentrically in each of the mentioned above. **Fig. 2** shows the concentric tubes.

4.3 Shell

Plate of carbon steel with 8 mm thickness was used to construct the shell. Six flanges were welded, one in each of the ends of shell cylinder, one of them is to close the shell and the other flange has two holes for inlet and outlet flows to the inner tubes. Four flanges were in the ends of both bundles of the tubes, in each end two flanges. Shell had been drilled from points on top and bottom to insert the nozzle. The shell inner diameter is 203 mm, and the shell outer diameter is 220 mm. Baffles of thickness 6 mm were spaced by a distance of 100 mm. The free section left was of 25%.

4.4 System of Fluids

Two fluids are used to complete the cycle of the heat exchanges. The first one which passes through the shell side and the inner tubes side is forty stock oil (lube oil) from Dorra Refinery. The experimental working range of this oil is 120˚C to 80˚C.The other fluid is water. It passes in the annulus section of the concentric tubes. The temperature range for the water is 20˚C to 30˚C.The flow is accomplished counter currently.

4.5 Cooling Circulation Unit

Water was used as the cooling fluid for this unit .The circuit consisted of a pump of 3 HP to circulate cold water, and to pump water to the annulus passages in the heat exchanger.
The cold water is supplied by a constant head tank of 250 liters capacity. Water is pumped from the above mentioned tank through the test section.
The cooling water leaving the heat exchanger will flow to a vessel of 100 liters capacity as a container to measure the temperature of outlet water and drainage to the sewage.

4.6 Heating Circulation Unit

Oil was used as the heating media for this unit, it consisted of a cubical tank equipped with an immersed two electric heaters each of 3000 W to heat oil to the desired temperature and the two heaters were equipped with a thermostat to adjust the oil temperature. The flow was measured using a flow meter in the range of 70 l/min and the hot oil flow is controlled by gate valve.

The hot oil is pumped by a single stage centrifugal pump from 250 liters capacity heating tank through the test section and it returns back to the second heating tank.

4.7 Rig Construction

4.7.1 Construction

A lay out of experimental setup is shown in **Fig. 3**.All piping system used in this rig were supported firmly and easily dismantled. Also supports were used to prevent piping from breakage due to vibration and load. All piping was joined by using threaded joints. A gate valve was used as a bypass valve for the heating and cooling circulation pumps .All pipes were cleaned before use to avoid fouling problem. The test rig was closed circulated for hot media and open for cold media.

4.7.2 Measuring Instrumentation

Temperature, pressure and flow rate were measured in the piping system, using thermo couples, pressure gauges and flow meters have been used as the main measuring instruments.

The parameters to be measured during the test are:

1. The inlet and outlet temperatures of the tube side (inner and annulus) and shell side.
2. The inlet and outlet pressures of the tube (inner and annulus) side and shell side.
3. The flow rates of the tube side and the shell side.

4.7.3 Test Procedure

After completing checking steps, the test process begins by switching on the circuit breaker that supplies power to the whole system.Then, switching on the individual switches of the two heaters will transfer the electrical power to the heaters which will rise the temperature of oil in the reheater as required to the desired temperature fixed by setting off the thermostat.This process takes 60 to 90 minutes depending upon the required temperature from the thermostat and the temperature of oil before starting. After that, the hot oil pump will start and the gate valve that controls flow rate of hot oil to in the inner tube side and in the shell side in the heat exchanger is opened.

On the cold water side, the cold water pump is switched, at the same time of the hot oil pumping and the flow is set according to the required flow rate.

After reaching the steady state condition, flow rate is fixed in the annulus side cold water at 2.4 m^3/h and at the required temperature of the hot oil set by the thermostat ,the hot oil temperature is regulated and changed from 120˚C to 80˚C. On the hot oil side (inner tube side and shell side), flow rate of oil is regulated and changed from 1.2, 1.5, 1.8, 2.1, 2.4, 2.7, 3.0, 3.3 and 3.6 m^3/h.

Water temperatures are constantly measured during flow rate variation. The pressures are also measured at the inlet and the outlet of the exchanger.

The procedure was repeated for flow rate of cold water in the annulus side as 2.4, 3.6, 4.8 m^3/h with fixed thermostat setting.

Tests were repeated after changing setting of the thermostat by 10˚C step from temperature setting level of 80˚C up to 120˚C. During each step of test flow rate of water is regulated and temperature and pressure are taken. Analysis and comparison of readings results are carried out at different conditions.

5. RESULTS AND DISCUSSION

5.1 Comparison between the Results from Theoretical and Experimental Work for New Heat Exchanger

5.1.1 Exit Temperature of Hot Oil

The theoretical design for new heat exchanger represents that the exit temperature of hot oil is at 60˚C. **Table 1** shows the experimental results obtained from the tested new heat exchanger and it can be seen from these results that the exit temperature of hot oil is 59˚C.

The difference between the two temperatures is 1˚C therefore, the experimental result is better than the theoretical design and the error ratio from these results is about -1.6%.

5.1.2 Exit Temperature of Cold Water

The temperature of cold water theoretically designed in new heat exchanger is at 20˚C inlet and at 30˚C outlet as shown in Appendix. The experimental results in **Table 1** represent that the exit temperature of cold water is at 29˚C.

The experimental result for the exit temperature of cold water is also close to the theoretical result and the error ratio between these results is about -3.3%.

5.1.3 Pressure Drop in Shell Side

Calculations in Appendix shown the theoretically value of pressure drop for hot oil in shell side of a new heat exchanger is 1.25 kPa. Experimentally the result of pressure drop in shell side is 1.51 kPa as shown in **Table 1**. The error ratio between the two values is about 17.2%.

5.1.4 Pressure Drop in Inner Tubes Side

The theoretical design of new heat exchanger represents the value of pressure drop for hot oil in inner tube side is 35 kPa as shown in Appendix, but the experimental result given in **Table 1** shows a lower value of pressure drop in inner tube side of 21.4 kPa and the error ratio between the values of pressure drop is about -39%.

5.1.5 Pressure Drop in Annulus Side

The value of pressure drop for water in annulus passage is 9 kPa for theoretical design as shown in Appendix. From **Table 1** the pressure drop for water in annulus passage is 11 kPa for experimental work.The error ratio for these results is about 18%.

5.1.6 Total power expenditure

From Calculations in Appendix the value of total power expenditure is 0.037 W for theoretically designed heat exchanger and from the experimental result the value of total power expenditure is 0.033 W.It can be seen that the experimental result is better than theoretical result and the error ratio for these results is about -10.8%.

5.2 Comparison between the New and Conventional Heat Exchangers

5.2.1 Length of heat exchanger

Table 2 represents the variation of the heat exchanger length with the volumetric flow rate of the hot oil for the new and conventional heat exchanger. The results in this table were obtained from the thermal design equations for the new and conventional heat exchangers.

Comparison between the lengths of two heat exchangers is presented in **Fig. 4**.It can be seen that at a flow rate of $3.6m^3/h$, the length of the conventional heat exchanger designed at $3.6m^3/h$ is higher about 63% than the length of the new heat exchanger.

Using a very small wall thermal resistance (copper tubes), the length of the new heat exchanger is L=1.03m and the percentage difference in the volumes of the two heat exchangers is about 70%.
Table 2 shows also that the length of shell and double concentric-tube heat exchanger is reduced still, by increasing the hot oil volumetric flow rate. This length is smaller of about 75% in the case where the two heat exchangers work with an industrial oil mass flow rate equal to $30m^3/h$.

5.2.2 Mass of heat exchanger

The mass of the new heat exchanger falls to about 46% regarding the shell-and-tube heat exchanger for the volumetric flow rate $3.6m^3/h$ and 6/10mm inside /outside diameter for the inner tubes. **Table 3** shows the differences in the mass between the two heat exchangers. The results in this table were obtained from the thermal design equations for the new and conventional heat exchangers.
This is earning decreases with the increase of inner tubes diameter and reaches 20 to 30%, and increases with increasing the length of heat exchanger and reaches to 89% in case where the two heat exchanger work with $30m^3/h$ volumetric flow rates of hot oil.
Fig.5; represents comparison between the masses of the two heat exchangers. The mass of new heat exchanger is always lower than the conventional heat exchanger in all volumetric flow rates of hot oil.

5.2.3 Pressure drop in heat exchanger

Fig. 6 shows the relation between the shell side pressure drop and volumetric flow rate of hot oil for new heat exchanger and conventional heat exchanger. The trend of this figure shows that increasing the hot oil flow rate causes an increase in its pressure drop component in two heat exchangers, but the pressure drop in the shell of the new heat exchanger is almost-still and falls about 85% with regard to that of shell-and-tube heat exchanger, as shown in **Table 4**.
The results in this table were obtained from the hydraulic design equations for the new and conventional heat exchangers.

5.2.4 Total power expenditure

Fig .7 shows the variation of the total power expenditure with volumetric flow rate of hot oil in the new heat exchanger.
The total power expenditure is negligible in new heat exchanger. The total friction power expenditure increases with the mass flow rate of the hot oil. The total power in the heat stream is minimal for the inner tube diameter 6/10 mm and do not exceed 0.037W of the heat exchanged by the hot oil mass flow rate designed $3.6m^3/h$ and reached to 7 W for $20m^3/h$.

6. CONCLUSION

1- The new heat exchanger is characterized with the heat transfer between three fluids, with its compactness and can be widened to the cross flow heat exchangers (simple-flow, parallel-flow and counter flow).
2- Optimizing a shell-and-double concentric tube heat exchanger lengthwise provides a considerable amount of savings in space and material when compared with a shell and tube heat exchanger with the same outer tube diameter of the double concentric-tubes and the shell diameter.
3- The length and mass of new heat exchanger are strongly dependent upon the diameters of tubes especially the inner tubes.
4- It is demonstrated that the relative diameter sizes of two tubes with respect to each other are the most important parameters that influence the new heat exchanger size.

5- The pressure drop in shell side of the new heat exchanger is lower than in the conventional heat exchanger.
6- The total power expenditure increases with increasing the flow rate of hot oil and its value in the new heat exchanger is very small and considered negligible.

REFERENCES

- Binay K.Dutta., 2009, *Heat Transfer Principles and Applications*, published by Asoke K. Ghosh, PHI Learning Private Limited.

- Bougriou, C., and Baadache, K., 2010, *Shell-and-Double Concentric Tube Heat Exchangers*. Heat and Mass Transfer, 46. PP.315-322.

- Chapman, A. J., 1984, *Heat Transfer*, Macmillan Publishing Company, 4th edition.

- Coulson, J. M., and J. F. Richardson, 1998, *Chemical Engineering*, Vol. **1**, Butterworth-Heinemann, 5th Edition.

- Emad, Y. M., 2005, *Heat Transport Through a Corroding Metal Wall of concentric tube Heat Exchanger",* Ph.D. Thesis, Chem. Eng.,Al-Nahrain University.

- Garci´a-Valladares, O., 2004, *Numerical Simulation of Triple Concentric Tube Heat Exchangers*, Privada Xochicalco S/N, Temixco, 62580 Morelos, Mexico, pp 980–991.

- Hewitt, G.F., et al 1994, *Process Heat Transfer*,CRC Press.

- John, E., Edwrads , 1998, *Design and Rating of Shell and Tube Hear Exchanger*, www.pidesgn.co.uk.pdf.

- Kern, D., 1999, *Process Heat Transfer,* McGraw Hill.

- Nelson, W.L., 1958, *Petroleum Refinery Engineering,* McGraw-Hill, Inc.

- Perry, R.H., and Green, D., 1997, *Perry's Chemical Engineers Handbook*, 7th edition McGraw Hill.

- Ramesh, K., Shah and Dusan, P., Sekulic, 2003, *Fundamentals of Heat Exchanger Design*, John Wiley & Sons, Inc.

- Ray Sinnot, and Gavin Towler, 2009, *Chemical Engineering Design*, Butterworth-Heinemann, Vol.6, 5th Edition.

- Saunders, E.A.D, 1988, *Heat Exchanger Selection, Design and Construction*, John Wiley & Sons, New York, NY, USA.

➢ Shlűnder, E., 1989, *Heat Exchangers Design Handbook*, Published by Hemisphere, New York, USA.

NOMENCLATURE

A_c = cross sectional area of the tube, m^2
a_s =cross flow area at the shell, m^2
B =baffle spacing, m
Cp =specific heat, J/(kg K)
D = diameter, m
D_e= equivalent diameter on the shell-side, m
D_s = shell inside diameter, m
d = inner tube diameter, m
d_{hs} = hydraulic diameter of the shell, m
d_h =hydraulic diameter of the annulus, m
F =corrective factor
f =friction factor
G_s =shell side mass velocity, kg/ m^2.s
h =heat transfer coefficient, W/ m^2.K
k =thermal conductivity, W/ m K
U =overall heat transfer coefficient, W/ m^2.K
L =length of tube, m
M =mass, kg
m =mass flow rate, kg/s
N_b =number of baffles
Nt =total number of tubes
Ntp =number of tubes per pass
p =tube pitch, m
P_T =total power expenditure, W
Δp =pressure drop, Pa
Q =volumetric flow rate, m^3/h
q =heat transfer rate, W
R =dimensionless temperature ratio
S =dimensionless temperature ratio
S_a =exchange surface, m^2
T =temperature, ˚C
LMTD = log-mean temperature difference, ˚C
u =fluid velocity, m/s

GREEK SYMBOLS

δ Thickness, m
μ Dynamic viscosity, Pa s
ρ Density, kg/m^3

DIMENSIONLESS NUMBERS

Nu Nusselt number: Nu = h d/k
Pr Prandtl number: Pr = μ Cp/k
Re Reynolds number: Re = ρ u d/ μ

SUBSCRIPTS

1 Hot oil (shell side), outer
2 Water, inner
3 Hot oil (inner tube)
12 Shell and annulus
23 Annulus and inner tube
s Shell
h Hydraulic
i Inlet
io Hot oil
o Outlet
st Shell-and-tube heat exchanger
sdct Shell-and-double concentric-tube heat exchanger
w wall

Table 1. At 110˚C inlet temperature of oil, 80 l/min flow rate of cold water and 20˚C inlet temperature of cold water.

Flow of oil inlet (l/min)	Temp of oil outlet (˚C)	Temp of water outlet (˚C)	P_i of oil shell (kPa)	P_o of oil shell (kPa)	P_i oil of inner tube (kPa)	P_o oil of inner tube (kPa)	P_i water Annulus (kPa)	P_o water Annulus (kPa)
60	59	29	52	50.49	50.3	28.9	83	72
55	57	28	49.3	48.13	45.5	27.5	83	72
50	55	27	46	45.11	40.7	26.6	83	72
45	53	26	42	44.32	36.6	25	83	72
40	51	26	37.2	36.68	33.1	24.1	83	72
35	50	25.5	32	31.62	28.25	21.35	83	72

30	49	25	26.5	26.21	25.2	19.7	83	72
25	47	24	21.8	21.59	20.3	16.2	83	72
20	45	23.5	17	16.83	15.2	12.4	83	72

Table 2. Length of heat exchanger values.

Volumetric flow rate (m^3/h) of hot oil	Length of new heat exchanger (m)	Length of conventional heat exchanger (m)
2	0.88	1.37
3.6	1.08	1.76
4	1.18	1.86
6	1.33	2.16
8	1.46	2.43
10	1.62	2.71
20	2.12	3.68
30	2.53	4.47
40	2.96	5.08

Table 3. The masses of the two heat exchangers.

Volumetric flow rate (m^3/h) of hot oil	Length of conventional heat exchanger (m)	Mass of conventional heat exchanger (kg)	Length of new heat exchanger (m)	Mass of new heat exchanger (kg)
2	1.37	81	0.88	52
3.6	1.76	93.4	1.08	63.9
4	1.86	100	1.18	69.8
6	2.16	128	1.33	78.7
8	2.43	144	1.46	86
10	2.71	160	1.62	95
20	3.68	218	2.12	125
30	4.47	265	2.53	140
40	5.08	301	2.96	175

Table 4. The pressure drop in shell side for the two heat exchangers.

Volumetric flow rate (m^3/h) of hot oil	Pressure drop in shell in new H.E (kPa)	Pressure drop in shell in conventional H.E (kPa)
2	0.3	1.6
3.6	1.25	7.2
4	1.5	9.5
6	3.2	19.2
8	6.4	37.2
10	10.5	61.3
20	47.5	287
30	117.3	703

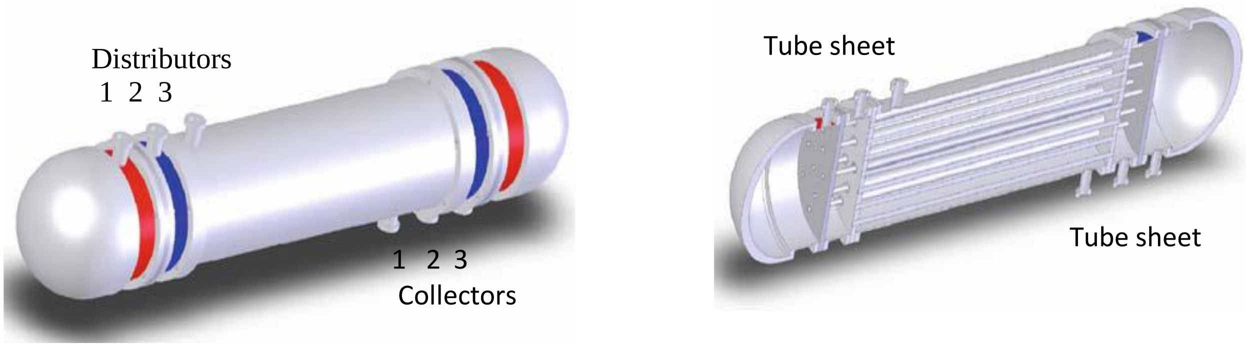

Figure 1. Perspective view and longitudinal section of the shell-and-double concentric-tube.

Figure 2. The concentric tubes.

Figure 3. All components of test rig.

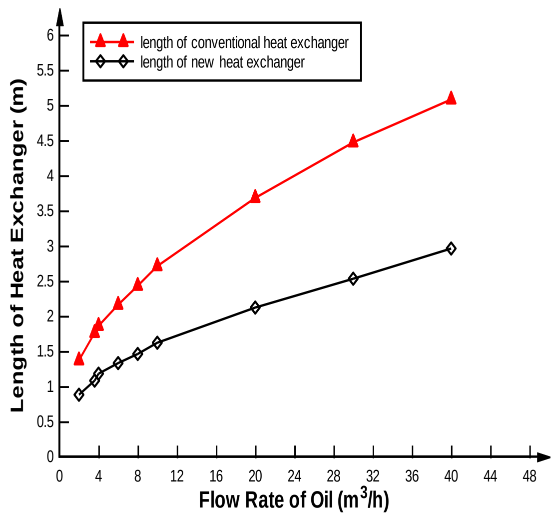

Figure 4. Length variation of the two heat exchangers with volumetric flow rate for hot oil.

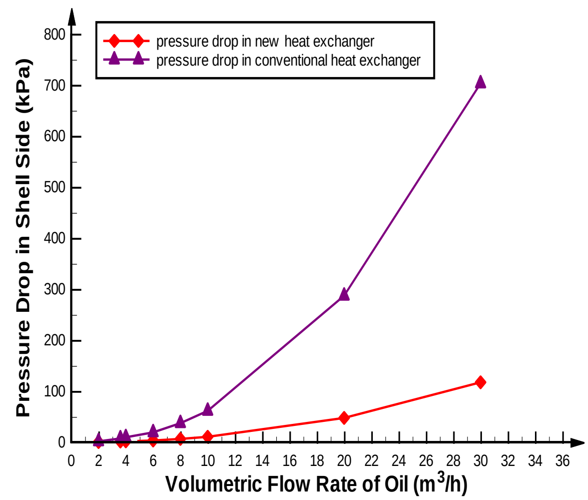

Figure 6. Variation of the shell side pressure drop with volumetric flow rate of hot oil for the new and conventional heat exchangers.

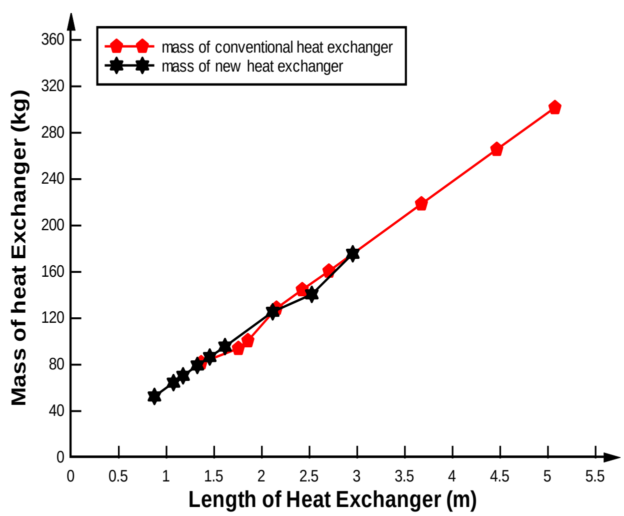

Figure 5. Variation the masses of the two heat exchangers with its length.

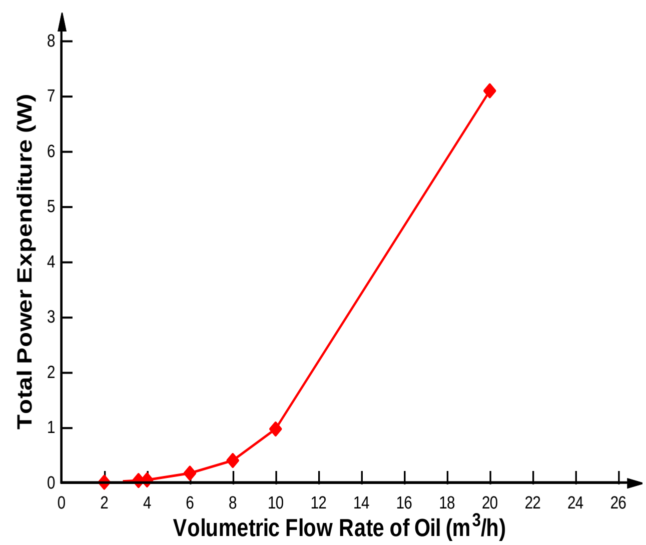

Figure 7. Variation of the total power Expenditure with volumetric flow rate of hot oil for the new heat exchanger.

APPENDIX
SAMPLE OF CALCULATIONS

Conventional Heat Exchanger

The heat exchanger serves for cooling a flow of oil (forty stock) $Q_1 = 3.6 m^3/h$ of $T_{i1} = 110^oC$ to $T_{o1} = 60^oC$ with water flowing in the tubes of $T_{i2} = 20^oC$ to $T_{o2} = 30^oC$ .The thermo physical properties of the oil for an average temperature of 85°C are as follows: $\rho_1 = 822 kg/m^3$, $Cp_1 = 2135 J/kg.k$, $K_1 = 0.1299 W/m.k$ and $\mu_1 = 3.97 \times 10^{-4} pa.s$, **Nelson, 1958**.

The thermo physical properties of water for an average temperature of 25°C are as follows: $\rho_2 = 1000 kg/m^3$, $Cp_2 = 4180 J/kg.k$ $K_2 = 0.607 W/m.k$ and $\mu_2 = 8.9 \times 10^{-4} pa.s$. **Perry,** and **Green, 1997**.

The heat exchanger is constituted of a bundle of $N_t = 16$ steel tubes of thermal conductivity $K_w = 50 W/m.k$, of diameters inside/outside (D2/D1) of 20/25 mm, in the normal triangular pitch p = 31.25 mm.

The heat exchanger has two passes. The shell has a diameter $D_s = 203 mm$ and possesses baffles of thickness $\delta = 6mm$ spaced by a distance B = 60 mm. The free section left with baffles is of 25%.

To determine the tubes length:

Mass flow rate m_1 of the hot oil is:

$$m_1 = \frac{3.6 \times 822}{3600} = 0.822 Kg/s \tag{6}$$

The exchanged heat flux is:

$$q = m_1 Cp_1 (T_{i1} - T_{o1}) = 87748.5 W \tag{7}$$

Mass flow rate m_2 of the water is:

$$m_2 = \frac{q}{Cp_2 (T_{02} - T_{i2})} = 2.10 Kg/s \tag{8}$$

Volumetric flow rate of water is:

$$Q = 7.56 m^3/h \tag{9}$$

For counter flows the logarithmic mean temperature difference from Eq. (3) as:

$$LMTD = \frac{(110-30)-(60-20)}{\ln\frac{(110-30)}{(60-20)}} = 57.7^oC$$

The values of temperature ratio are:

$$R = \frac{(110 - 60)}{(30 - 20)} = 5 \tag{10}$$

$$S = \frac{(30 - 20)}{(110 - 20)} = 0.11$$

The corrective factor F of the logarithmic mean temperature difference, corresponding to the calculated values of R and S and from, **Kern, 1999** is:

$$F = 0.98$$

The cross sectional area of the tube is, **Binay, 2009**:

$$A_c = \frac{3.14}{4} \times (0.02)^2 = 3.14 \times 10^{-4}\, m^2 \tag{11}$$

The velocity of the water in tubes is:

$$u_2 = \frac{2.10}{1000 \times 8 \times 3.14 \times 10^{-4}} = 0.835 m/s \tag{12}$$

The calculation of the Reynolds number and the Prandtl number:

$$\mathrm{Re}_2 = \frac{1000 \times 0.835 \times 0.02}{8.9 \times 10^{-4}} = 18764$$

and

$$\mathrm{Pr}_2 = \frac{8.9 \times 10^{-4} \times 4180}{0.607} = 6.13$$

By using the Colburn Equation, the Nusselt number is:

$$Nu_2 = 0.023 (18764)^{0.8} (6.13)^{0.33} = 109.7 \tag{14}$$

The heat transfer coefficient from Equation below is:

$$h_2 = \frac{109.7 \times 0.607}{0.02} = 3329.4 W/m^2.k \tag{15}$$

The shell equivalent diameter for triangular pitch is, **Ray Sinnot, and Gavin Towler, 2009**:

$$D_e = 0.0176\, m$$

The bundle cross flow area is:

$$a_s = \frac{0.203 \times 0.06 \times 0.00625}{31.25 \times 10^{-3}} = 0.0024 m^2 \qquad (16)$$

The shell side mass flow rate is calculated from Equation as:

$$G_s = \frac{0.835}{0.0024} = 347.9 kg/m^2.s \qquad (17)$$

The heat transfer coefficient in shell side is calculated as:

$$h_1 = 992.2 W/m^2 k \qquad (18)$$

The overall heat transfer coefficient is, **Binay, 2009**:

$$U = \frac{1}{\frac{0.02}{0.025 \times 992.2} + \frac{0.02}{2 \times 50} \ln \frac{0.025}{0.02} + \frac{1}{3329.4}} = 874.12 W/m^2 k \qquad (19)$$

The surface area of the heat exchanger is:

$$A = 1.77 m^2 \qquad (20)$$

The length of the heat exchanger is calculated:

$$L = 1.76 m \qquad (19)$$

The mass of conventional heat exchanger is, **Bougriou,** and **Baadache, 2010**:

$$M_{st} = 93.4 kg \qquad (20)$$

4.2 The New Heat Exchanger

The same data of the conventional heat exchanger are used. In this case, one adds concentrically in each tube of diameters (D_2/D_1) of 20/25 mm a tube of diameters (d_2/d_1) of 6/10 mm.

Inner Tubes Side Calculation

The inner flow cross sectional area of the inner tubes is calculated as:

$$A_{c3} = \frac{3.14}{4}(0.006)^2 = 0.000028 m^2 \qquad (22)$$

The mass flow rate of oil circulating inside the inner tubes is:

$$m_3 = 0.822 Kg/s \qquad (23)$$

The velocity of oil inside inner tubes is:

$$u_3 = 2.23 m/s \tag{24}$$

Inner Tubes Side Nusselt Number and Heat Transfer Coefficient

By using the Colburn Equation, the Nusselt number is, **Hewitt, G.F., et al 1994**:

$$Nu_3 = 0.023(27702.7)^{0.8}(6.52)^{0.33} = 153 \tag{25}$$

The heat transfer coefficient is:

$$h_3 = \frac{153 \times 0.1299}{0.006} = 3312.45 W/m^2k \tag{26}$$

Shell Side Calculation

The shell side mass flow rate is calculated from Eq.(17) as:

$$G_s = \frac{0.411}{0.0024} = 171.25 kg/m^2.s$$

Shell Side Heat Transfer Coefficient

The heat transfer coefficient is:

$$h_1 = \frac{91 \times 0.1299}{0.0176} = 671.64 W/m^2k \tag{27}$$

Annulus Side Calculation

The flow cross sectional area of the annulus passages is calculated from equation below:

$$A_{c2} = \frac{3.14}{4}\left((0.02)^2 - (0.01)^2\right) = 2.35 \times 10^{-4} m^2 \tag{28}$$

The velocity of water in annulus flow passages is:

$$u_2 = \frac{2.10}{1000 \times 2.35 \times 10^{-4} \times 8} = 1.117 m/s \tag{29}$$

The equivalent diameter of the annulus is calculated as, **Emad, 2005**:

$$d_h = 0.02 - 0.01 = 0.01 m \tag{30}$$

Annulus Side Nusselt Number and Heat Transfer Coefficient

By using the Colburn equation, the Nusselt number is, **Chapman, 1984**:

$$Nu_2 = 0.023(12550)^{0.8}(6.13)^{0.33} = 79.5 \tag{31}$$

The heat transfer coefficient is:

$$h_2 = \frac{79.50 \times 0.607}{0.01} = 4825.65 W/m^2.k \tag{32}$$

Overall Heat Transfer Coefficient

Overall heat transfer coefficient (U_{12}) between (the fluid in the shell side and fluid in the annulus passage) is calculated as:

$$U_{12} = \frac{1}{\frac{0.020}{0.025 \times 671.64} + \frac{0.02}{2 \times 50}\ln\frac{0.025}{0.02} + \frac{1}{4825.65}} = 744 W/m^2 k \tag{33}$$

Overall heat transfer coefficient (U_{23}) between (the fluid in the annulus passage and the fluid in the inner tube side) is calculated as:

$$U_{23} = \frac{1}{\frac{0.006}{0.01 \times 4825.65} + \frac{0.006}{2 \times 50}\ln\frac{0.01}{0.006} + \frac{1}{3312.45}} = 2222.2 W/m^2 k \tag{34}$$

Length of the Heat Exchanger

The double concentric tubes length in the shell is calculated as:

$$L = 1.08 m \tag{35}$$

Inner Tubes Side Pressure Drop Calculation

The inner tube side pressure drop is calculated as, **Shlűnder, E., 1989**:

$$\Delta P_3 = \left(4 \times 0.024 \times \frac{1.08 \times 2}{0.006} + 4 \times 2\right)\frac{822 \times (2.23)^2}{2} = 35 kPa \tag{36}$$

Shell Side Pressure Drop Calculations

The hydraulic diameter of the shell is calculated as, **Bougriou,** and **Baadache, 2010**:

$$d_{hs} = 0.025\left(\frac{3.46}{3.14}\left(\frac{31.25\times10^{-3}}{0.025}\right)^2 - 1\right) = 0.018m^2 \tag{37}$$

The pressure drop is calculated as:

$$\Delta P_1 = \frac{0.32\times(171.25)^2(18+1)\times0.203}{2\times822\times0.018} = 1.25kPa \tag{38}$$

Annulus Side Pressure Drop Calculation

The annulus side pressure drop is calculated as:

$$\Delta P_2 = \left(4\times0.029\times\frac{1.08\times2}{0.01} + 4\times2\right)\frac{1000\times(1.117)^2}{2} = 9kPa \tag{39}$$

The Total Power Expenditure

The total power expenditure of the new heat exchanger is, **Garci´a-Valladares, 2004**:

$$P_T = \frac{0.411\times1.25}{822} + \frac{2.10\times9}{1000} + \frac{0.411\times35}{822} = 0.037W \tag{40}$$

The Mass of Heat Exchanger

The mass of shell and double concentric tubes heat exchanger is:

$$M_{sdct} = 63.9kg \tag{41}$$