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An Experimental Study of Minimum Thermal Energy Loss or Gain using a Coil-in-Shell Heat Exchanger

Minh-Man Pham*
Department of Mechanical Engineering,
Danang College of Technology,
48 Cao Thang Street, Danang City, Viet Nam
* Department of Mechanical Engineering,
Graduate School, Inje University,
197 Inje-ro, Gyeongnam 621-749, Korea

Tra Qui Phan
Head of Office of Research Management
and International Cooperation
Danang College of Technology
The University of Danang
48 Cao Thang Street, Danang City, Viet Nam

***Abstract* —The present experimental study investigates a coil-in-shell heat exchanger which is the most common and used widespread in numerous industrial applications. Besides, this highly efficient heat exchanger helps minimize a temperature difference between the shell-side fluid and the coil-side fluid, thereby enhancing the transference of the thermal energy between two or more fluids at different temperatures and in thermal contact. The experimental results show that the higher coil-in-shell diameter, coil pitch and mass flow rate in shell and tube can enhance the heat transfer rate in these types of heat exchangers. On the other hand, the Nusselt number in the shell-side of the present heat exchanger with its Rayleigh numbers of $3.1E^9$ at the LMTD of 36.2°C was 539. Moreover, the effectiveness of the coil-in-shell heat exchanger with its mass flow rate ratio R_m of 1.17 and 2 at the LMTD of 33.6°C and 36.2°C were 0.75 and 0.65 respectively.**

Keywords—Thermal energy, Coil-in-Shell Heat Exchanger, Nusselt number, Heat Exchanger Effectiveness, Rayleigh number

I. INTRODUCTION

The heat exchangers have been widely utilized, for example in heat recovery systems, power plants, nuclear reactors, food industries, chemical processing, refrigeration and air conditioning systems. It is an essential unit in heat extraction and recovery system. The scarcity of natural resources and increase of energy consumption has highlighted the important role of heat exchangers in utilizing waste heat and saving energy. It has also become important in implementing the concept of pollution prevention, helping to make these new processes economically feasible. In general, industrial heat exchangers are classified in many different ways according to construction, transfer process, degrees of surface compactness, pass arrangements, phase of the process fluids and heat transfer mechanisms.

The coil-in-shell type heat exchanger is the most widely used in the industry because of its relatively simple construction and multi-purpose application possibilities for gaseous and fluid media in a very large temperature and pressure range. Especially, it is virtually designed for any capacity and operating conditions such as high pressure, high temperature, highly corrosive atmosphere and fluid, etc. Moreover, the coil-in-shell heat exchanger has a considerable flexibility in the design because the core geometry can be varied easily by changing parameters and arrangement. When the performance of a coil-in-shell heat exchanger is enhanced, its improvement enables the size of the heat exchanger to be decreased, Salimpour [1].

In the coil-in-shell heat exchangers, the flow and heat transfer processes on the shell side are highly complicated since they are influenced by geometrical characteristics such as shell-tube size, pitch ratio, tube arrangement, and coil-tube clearances. Among them, the effect of coils type on heat transfer and pressure drop is very significant. Since the coil is primarily used in the coiled heat exchanger to support the tubes, to prevent the flow induced vibration and to induce the counter-flow over the shell, which helps improve heat transfer performance by increasing turbulence and laminar or intensity in the flow and local unmixed on the shell side of the heat exchanger.

Much research has been undertaken to study a natural convection shell-and-coil heat exchanger which consists of a cylindrical shell with helical coils placed inside it. On the tube side the flow is forced by a pump through the coils while buoyancy forces are the cause of flow on the shell side, Taherian et al. [2]. Some studies have indicated that coiled tubes are superior to straight tubes when employed in heat transfer applications, Shokouhmand et al. [3]. Moreover, these systems have lower initial cost and less maintenance than the alternatives. Srinivasan et al. [4] performed experiments on a shell and coil heat exchanger in the case where the buoyancy driven flow occurs in the shell. Other works include those of Tagliafico and Tanda [5] and Parent et al. [6], which involve shell and tube natural convection heat exchangers where buoyancy driven flow occurred inside the tubes. Dravid et al. [7] numerically investigated the effect of secondary flow on laminar flow heat exchanger in coiled tubes both in the fully developed and in the thermal entrance regions. Besides, Parakar et al. [8] discussed the effect of

Dean Number on friction factor and heat transfer in the developing and fully developed region of coiled pipes. Xin and Ebadian [9] studied the effect of Prandtl number and geometric parameter on Nusselt number and helical and straight tubes were compared by Prabhanjan et al. [10]. Their results showed that a helical coil heat exchanger increased the heat transfer coefficient and temperature rise of fluid depends on the coil geometry and flow rate. Also, several review and summary papers in the shell-tube and shell-coil heat exchanger studies have been published by Gray and Williams [11], Kottl and Li [12], Edward and Gnielinski [13], Pekdemir et al. [14], Cole [15],

According to the literature review no experimental work has been done to predict the effect of different design parameters on heat transfer rate in coil-in-shell heat exchanger. The present experimental research investigates the effectiveness of the heat exchanger. The coil-in-shell heat exchanger has been studied in which heat energy is transferred from one fluid stream to another through contact with the carbon steel material of walls separating the fluid streams. The present experiment also describes the performance characteristics of such a heat exchanger where the hot fluid flow rate ratios are maintained at several values of R_m.

II. EXPERIMENTAL APPARATUS AND PROCEDURE

A schematic diagram for the testing of the coil-in-shell heat exchanger is shown in Fig. 1. The experimental apparatus consists of a coiled heat exchanger, refrigerating bath circulators, cooling system, water and fluid pumps, thermocouples and data acquisition device, flow-meters, pressure gauges, in-line valves and computer system. The working fluid (produced water inside the coil-side and water inside the shell-side) is shown with different colors lines. All the thermocouple wires were connected to the data acquisition device, and from this point to the computer system for monitoring data. Further, a cooling system (not shown) was mounted outside of each of the streams inlet and outlet to allow a fluid of shell to be smoothly cooling. On the other hand, the thermocouple wires and the electrical connection between the power station and all of the components were not shown in this figure. The coil-in-shell heat exchanger was installed in the laboratory on the Inje University campus in the Republic of Korea.

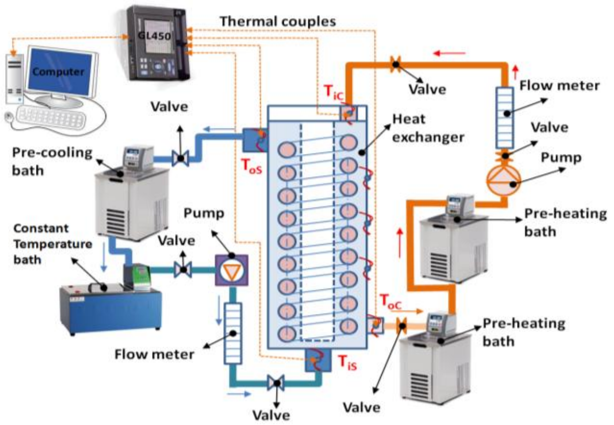

Fig. 1. The schematic view of the experimental installation of the coil-in-shell heat exchanger

The present coil-in-shell heat exchanger is shown in detail in Fig. 2. This heat exchanger is made of a carbon steel material of coil having an inner diameter of 13.9-mm and 21.3-mm outer diameter. Coils were arranged in the shell and had 9 turns. The coil diameter *(D_c)* and pitch *(p_c)* were also shown detail in Fig. 2. The shell-side of heat exchanger had 55-mm inner and 165-mm outer diameters and 442-mm length. To minimize a heat loss to the surrounding, the outside of the shell is tightly insulated with several layers of fiber glass. In other words, a hot fluid stream flowing inside the coiled tube was cooled by a cold stream flowing in the shell-side of the heat exchanger system. In order to maintain hot fluid temperature from inlet nearly constant, two refrigerating bath circulators were used. One bath was used for preheating fluid, and the other bath was used for maintaining the hot fluid temperature. While the cooling fluid temperature was maintained at a constant value by the cooling system, two fluid pumps were used to circulate the fluid stream in the system. The flow rate of fluid was measured and controlled by two flow meters and two gate valves.

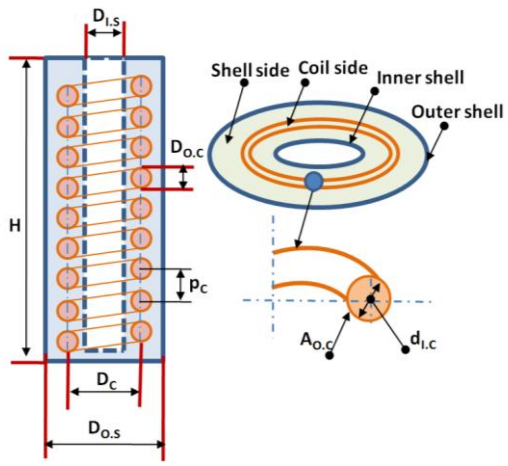

Fig. 2. A schematic detail diagram of a coil-in-shell heat exchanger

The fluid temperatures at two inlets and outlets in the heat exchanger were measured with four K-type thermocouples. These thermocouples were calibrated against PRT (Platinum Resistance Thermometer) in the constant temperature bath to within ±0.1℃ accuracy.

For the preparation of the experiment, constant temperature baths and pumps are turned on. When hot fluid and cold fluid temperatures reached the desired experimental temperature, using gate valves, the flow rates of hot fluid and cold fluid were adjusted to desired values. After a period of about 10 minutes, the test apparatus reached a steady state. And, the stop-wrist watch was operated while a digital data acquisition system started recording inlet and outlet fluid temperatures for a period of one minute. The data acquisition system continuously recorded fluid temperature for 30 minutes. The experiments were conducted with varying different parameters such as different flow rate ratios between in the coil-side and shell-side to study the effect of the parameters on heat transfer rates and effectiveness in the coil-in-shell heat exchanger.

III. DATA REDUCTION

If the heat loss to the surroundings in the heat exchanger is assumed to be negligibly small, the heat transfer rate between the hot fluid and cold fluid can be expressed as:

$$Q = \dot{m}_C c_C (T_{Ci} - T_{Co}) \cong \dot{m}_S c_S (T_{So} - T_{Si}) \tag{1}$$

where $\dot{m}$ is the mass flow rate (kg/s) through the heat exchanger and c is the specific heat of the air (kJ/kg°C), where the indices C and S refer to the coil-side and shell-side flows. T_{Ci}, T_{Co}, T_{So}, and T_{Si} are the temperatures of the fluid inside coil inlet, the fluid inside coil outlet, the fluid inside shell outlet, and the fluid inside shell inlet (°C), respectively.

According to the research by Schmidt [16], the critical Reynolds number for the helical coil flow, which determines the flow is laminar, is related to the curvature ratio as follows:

$$R_{crit} = 2300\left[1 + (8.6)\left(\frac{d_{I.C}}{D_C}\right)^{0.45}\right] \tag{2}$$

In the figure 2, $d_{I.C}$ is the inner diameter of the coiled tube, D_c is the curvature diameter of coil, and p_c is the coil pitch. The other important dimensionless parameters of coiled tube namely, Reynolds number (Re_C), Dean number (D_e), and Helical number are defined as Eq. (3-5).

$$Re_C = \frac{4\dot{m}_C}{\pi d_{I.C}\mu_h} \tag{3}$$

$$De = Re_C\left(\frac{d_{I.C}}{D_C}\right)^{0.5} \tag{4}$$

$$He = \frac{De}{(1+\gamma^2)^{0.5}} \tag{5}$$

According to the research of Gnielinski [17], the Nusselt number (Nu_C) for the helical coiled tube flow is as follows:

$$Nu = \left(3.66 + 0.08\left[1 + 0.8\left(\frac{d_{I.C}}{D_C}\right)^{0.9}\right] Re^m Pr^{1/3}\right)\left(\frac{Pr}{Pr_{wall}}\right)^{0.14} \tag{6}$$

where, $m = 0.5 + 0.2903\left(\frac{d_{I.C}}{D_C}\right)^{0.194}$

The overall heat transfer coefficient U between the two flows was calculated from following Eq. 7 as [18]:

$$Q = FUA\Delta T_m \tag{7}$$

where, U is the overall heat transfer coefficient, A is the surface area through which the heat transfer would occur. F is the correction factor which depends on temperature effectiveness, heat capacity rate ratio and flow arrangement.

The overall heat transfer coefficient can be related to the inner and outer heat transfer coefficient by following equation [21].

$$UA = \left(\frac{1}{A_I h_I} + \frac{\ln(r_O/r_I)}{2\pi k\ L} + \frac{1}{A_O h_O}\right)^{-1} \tag{8}$$

where r_I, r_O are inner and outer diameters of the tube respectively; k is the thermal conductivity of the wall; and L is the length of the coiled tube.

Since all heat exchanger operated with counter flow, mean logarithmic temperature difference is:

$$LMTD = \Delta T_m = \frac{(T_{C.o} - T_{S.i}) - (T_{C.i} - T_{S.o})}{\ln[(T_{C.o} - T_{S.i})/(T_{C.i} - T_{S.o})]} \tag{9}$$

The basic idea of dimensionless parameters of shell-side is to use the hydraulic diameter (D_{eq}) as a characteristic length for shell-side heat transfer coefficient. The other important parameters namely; Reynolds number (Re_S), Rayleigh number (Ra), and Nusselt number (Nu_S) are defined as Eq. (10-13) [2,19].

$$D_{eq} = \frac{4V}{\pi D_{O.C}L} = \frac{(D^2{}_{O.S} - D^2{}_{I.S})H - D^2{}_{O.C}L}{D_{O.C}L} \tag{10}$$

$$Re_S = \frac{\rho_w u_o D_{eq}}{\mu_o} \tag{11}$$

$$Ra = \frac{g\ \beta \Delta T\ D_{eq}^3}{\alpha\nu} \tag{12}$$

$$Nu_o = 0.0041 Ra^{0.4533} Re_S^{0.2} Pr_w^{0.3} \tag{13}$$

The heat-exchanger effectiveness was determined by following the definition in [18].

$$\varepsilon = \frac{\text{Actual heat transfer}}{\text{Maximum possible heat transfer}} = \frac{Q_{act}}{Q_{max}} \tag{14}$$

An uncertainty analysis for the coiled heat exchange effectiveness on the basis of 20:1 odds (i.e., 95% confidence level of errors) was conducted using the method of Pimenta et al. [19]. The uncertainty for the effectiveness was estimated to be 2.06%. On the other hand, the uncertainty of experimental data results from measuring errors of parameters such as volume flow rate and temperature are following Moffat [20]. The precision of the thermocouple were $\pm 0.1^0$C, and the precision of the volumetric flow meter were ±0.5 l/min. As a result, the uncertainty of Reynolds number and heat transfer experiment were less than 5.2%.

IV. DISCUSSION OF RESULTS

Figure 3 shows the effect of the temperature difference of the fluids inside the shell-side ($T_{S.i}$–$T_{S.o}$) and coil-side ($T_{C.o}$–$T_{C.i}$) between the two flows stream outlet and inlet and the rate of heat transfer *Q of* the heat exchanger for fixed condition (R_m=2, $T_{S.i}$ =*22*°C and $T_{C.i}$ =*84*°C).

As the temperature difference of the fluids decreased, the heat transfer rate between the two streams increased. It can be seen that the smaller value of the temperature difference of the fluids, the higher the value of the rate of heat transfer *Q*, which in a thermodynamic sense corresponds to reduced the value of thermodynamic irreversibility and smaller entropy generation. The figure 3 clearly showed that when the temperature difference of the fluids ($T_{S.i}$–$T_{S.o}$) and ($T_{C.o}$–$T_{C.i}$) changed from 37°C to 25°C and 24°C to 17°C respectively, the rate of heat transfer Q sharply increased from 5.75 kW to 7.86 kW. Beyond ($T_{S.i}$–$T_{S.o}$) = 31°C and ($T_{C.o}$–$T_{C.i}$) = 20°C, the rate of heat transfer Q remained at around 6.43 kW.

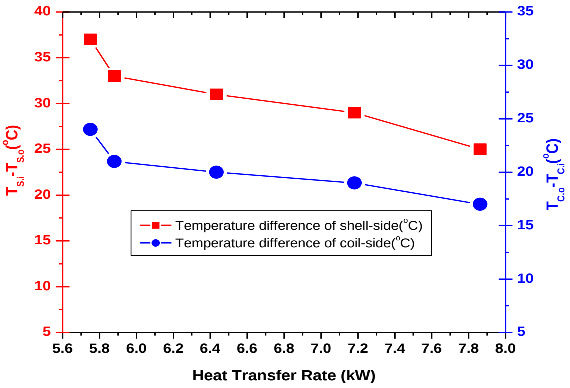

Fig. 3. The heat transfer rate versus the temperature different of coil-side and shell-side inside heat exchanger

This indicates that the present heat exchanger was very sensitive to the difference between the outlet and the inlet temperatures difference of the fluids.

TABLE I. DIMENSION OF THE COIL-IN-SHELL HEAT EXCHNAGER

Dimension	Average of the measured values	
Internal diameter of the coil	13.9 ± 1.1	mm
External coil diameter	21.3 ± 1.2	mm
Length of the coil tube	2.9 ± 0.2	m
Coil pitch	35 ± 0.01	mm
The number of coil turns	9.0	turns
Inner diameter of the shell	55 ± 1.5	mm
Outer diameter of the shell	165 ± 2.1	mm

Figure 4 shows the rate of heat transfer Q based on the overall heat transfer coefficient of the heat exchanger versus various Rayleigh numbers *(Ra_S)*. During the calculations, the two fluid flows were considered (i.e., 84°C of hot fluid inlet temperature and 22°C of cold fluid inlet temperature inside the heat exchanger), and the values of the specification of the coil-in-shell heat exchanger were analysed in Table 1. The experiment was conducted for the laminar flow inside the coil-side and shell-side of the heat exchanger. It is shown from Fig. 4 that the rate of heat transfer Q and the overall heat transfer coefficient of the heat exchanger increased with the Rayleigh numbers, up to 7.86 kW and 995 W/m^{2o}C for a Rayleigh numbers of 3.6E^9. Due to practical limits, we decided to use a heat exchanger Rayleigh number of 3.1E^9, at which the rate of heat transfer Q was calculated to be 6.43 kW with an overall heat transfer coefficient of 964 W/m^2°C.

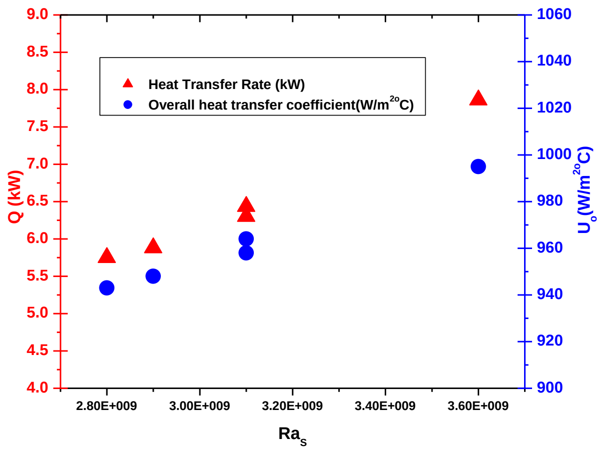

Fig. 4. Heat transfer rate based on overall heat transfer coefficient versus Rayleigh numbers

It could be deduced that the overall heat transfer coefficient of the heat exchanger increased with the increasing rate of heat transfer, because the both the shell-side and coil-side heat transfer coefficient increased as a result of heat transfer rate increase based on increasingly Rayleigh numbers.

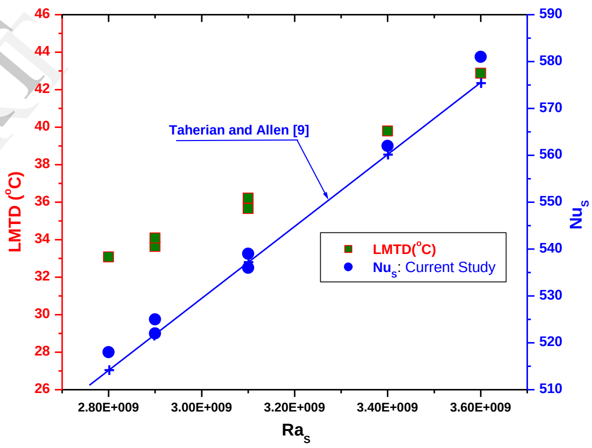

Fig. 5. The Nusselt number based on logarithmic mean temperature difference versus Rayleigh numbers

Figure 5 shows the Nusselt number data based on logarithmic mean temperature difference (LMTD) versus Rayleigh numbers. It could be deduced that Nusselt number in shell-side and LMTD of the heat exchanger increased with the Rayleigh numbers, up to 581 and 42.9°C for a Rayleigh numbers of 3.6E^9. Due to practical limits, we decided to use a heat exchanger Rayleigh numbers of 3.1E^9, at which Nusselt number was calculated to be 539 with an LMTD of 36.2°C. In the coil-in-shell heat exchanger, a complex flow pattern existed in laminar as well as turbulent flow regimes responsible for the increasing heat transfer coefficient, because both the coil-side and shell-side Nusselt number increased as a result of heat transfer rate increase. Neither the Reynolds number nor the Rayleigh numbers could characterize the hydrodynamics of flow through the coil-in-shell heat exchanger. Therefore, the agreement between recent results from experimental and Taherian and Allen's correlations is satisfactory. As shown in Fig. 5 the values of the Nusselt number calculated in the current study were slightly higher than those in [2, 21].

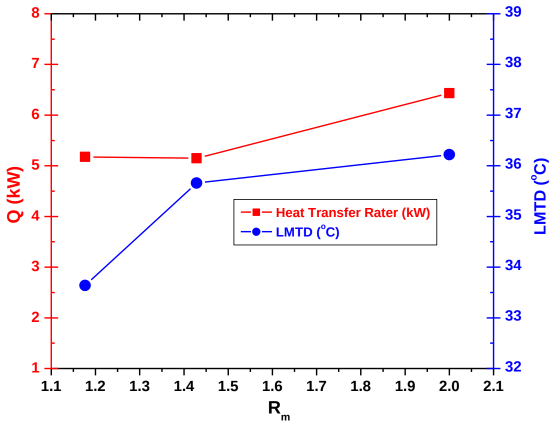

Fig. 6. **T**he rate of heat transfer based on LMTD versus the mass flow rate ratio

Figure 6 shows the effect of the mass flow rate ratio R_m of the two fluid streams on the rate of heat transfer Q based on logarithmic mean temperature difference (LMTD). As the mass flow rate ratio R_m increased, the heat transfer rate between the two fluid streams increased. It was found that the heat transfer rate Q data was correlated to the mass flow rate ratio for $1.1 < R_m < 2.1$, which means that the two fluid flows were considered (i.e., 84°C of hot fluid inlet temperature and 22°C of cold fluid inlet temperature inside the heat exchanger, the value of $R_m \cong 2$ seems to be the critical point). As illustrated in the figure, the smaller the value of the mass flow rate ratio, the lower the value of the rate of heat transfer Q and LMTD. When the mass flow rate ratio changed from 1.17 to 2.0, the heat transfer rate sharply increased from 5.2kW to 6.43kW and LMTD enhanced from 33.6°C to 36.2°C respectively. Beyond the mass flow rate ratio of $R_m \cong 2$, the heat transfer rate remained at around 6.43 Kw. This indicates that the present heat exchanger was very sensitive to the differences in the mass flow rates ratio in the controller process. In other words the shell-side fluid mass flow rate had a positive effect while the coil-side mass flow rate had an adverse effect on the heat transfer rate Q and LMTD of the heat exchanger.

The mass flow rate ratio R_m of the coil-in-shell heat exchanger is the factor eventually influencing the heat transfer of the exchanger. It can be seen in Fig. 7 that an effectiveness of 0.75 was obtained when the mass flow rate ratio R_m of heat exchanger was approximately 1.17 at the LMTD of 33.6°C. The results indicate that with increasing mass flow rate ratio, the logarithmic mean temperature difference decreased and the values of the effectiveness of the heat exchanger also decreased. Due to practical limits, this paper decided to use a mass flow rate ratio of 2, at which point the effectiveness of the heat exchanger was calculated to be 0.65 at an overall heat transfer coefficient of 964 $W/m^{2O}C$ and the logarithmic mean temperature difference of 36.2°C.

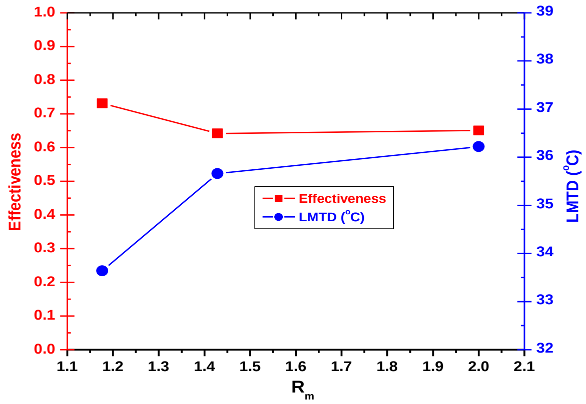

Fig. 7. The effectiveness of heat exchanger based on LMTD versus the mass flow rate ratio

V. CONCLUSIONS

In this study the coil-in-shell heat exchanger for various Rayleigh numbers, various mass flow rate ratio and temperature difference was experimentally investigated. The present study examined the thermal energy inside the coil-side or shell-side of the heat exchanger, and evaluated the effectiveness of the heat exchanger. Several conclusions can be summarized as follows.

(1) The rate of heat transfer Q was calculated to be 6.43 kW with an overall heat transfer coefficient of 964 W/m^{2}°C based on the Rayleigh numbers of $3.1E^9$, when the mass flow rate ratio of $R_m \cong 2$ at the LMTD was 36.2°C and The temperature difference of the fluids inside the shell-side ($T_{S.i}$–$T_{S.o}$) 31°C and coil-side ($T_{C.o}$–$T_{C.i}$) 20°C.

(2) The Nusselt number in the shell-side of the present heat exchanger with its Rayleigh numbers of $3.1E^9$ at the LMTD of 36.2°C was 539.

(3) The effectiveness of the coil-in-shell heat exchanger with its mass flow rate ratio R_m of 1.17 and 2 at the LMTD of 33.6°C and 36.2°C were 0.75 and 0.65 respectively.

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