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Exergy Destruction Assessment of Heat Transfer and Fluid Flow Through Spiral Passage Subjected to Constant Wall Temperature

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Abstract — **Exergy destruction of heat transfer and turbulent convective fluid flow through spiral passage subjected to constant wall temperature is analyzed. Constant as well as variable thermo physical properties models of process fluid have been adopted. Heat transfer characteristics and both thermal as well viscus fluid friction exergy destruction are investigated along the spiral passage. It is found that the total exergy destruction is dominated by thermal influence due to temperature difference between narrowing passage wall and fluid flow. It is also concluded that secondary flow phenomenon plays a significant role in fluid mixing and conversion of viscous dissipation into thermal energy. Furthermore variable fluid properties have considerable effect on exergy destruction.**

Keywords -- exergy destruction, turbulent flow, spiral passage, secondary flow, Bejan number

I. AN INTRODUCTION AND LITERATURE REVIEW

Forced convection heat transfer in a flow passage is affected by two types of exergy destruction namely, destruction associated with heat transfer through a temperature difference and that associated with fluid friction. Exergy destruction minimization has been proposed as a criterion for the design of flow passages in internal flow forced convection with heat transfer. Fluid flow and heat transfer of experimental and theoretical studies on spiral heat exchanger commonly used in chemical, food industry, and energy storage systems found less investigation. However ,the following literature reviews analysis some of important research results regarding entropy generation of heat transfer ad fluid flow of different configuration. **Bejan [1, 2, 3, 4, 5]** analyzed entropy generation in internal conduit flow, external flow, and in insulation system using lamped parameters .He showed that the geometrical parameters of the flow can be selected in such a way to minimize irreversibility. **Sekulic et al[6]** investigated the entropy generation in different duct geometries under constant wall temperature of fully developed flow with constant thermo physical properties. The geometries of the ducts included circular, triangular, parallel plates, rectangular and square. Different parameters were evaluated in this analysis, namely Reynolds number, inlet to wall temperature ratio and a specific duct length. **Sahin[7]** analyzed the laminar fluid flow according to second law analysis through different cross-sectional geometries, such as circular, rectangular, square, triangle, and sinusoidal. He concluded that triangular and rectangular duct shapes are in general the worst choices for both entropy generation and pumping power requirement. **Mansour Talebi [8]** studied entropy generation of a convection fluid flow in a helical tube. It was reported that the constant viscosity assumption would overestimate the entropy generation rate due to both heat transfer and friction loss. It was revealed that there were optimum values of curvature and fluid inlet temperature and these values were certain functions of viscosity. **Milovancevice and El-Sagier [9]**, recently analyzed spiral plate heat exchanger using Thermoeconomics approach with constant wall temperature. The result showed that the model can be used for design of an optimal spiral plate heat exchanger (SPHE) without performing economic analysis and calculation of the minimal annual costs. **El-Sagier [10]** studied from entropy point of view the influences of a various parameters of spiral channels with different cross-sectional geometry on the performance a turbulent forced convective fluid flow of ideal gas at a constant wall temperature. The findings show that the narrowing passage consumes more pump power to overcome viscous dissipation. However, the convential exergy analysis of heat transfer and fluid flow through spiral passage found fewer attentions in the literatures from the knowledge of the author. Thus the main task of the present paper is to assist the exergy destruction due to the inevitable irreversibility of real processes of heat transfer and forced convected turbulent fluid flow through spiral passage configuration subjected to constant wall temperature..

For better understanding of this subject still more detailed assessment is required. The study included two models, constant and variable thermophysical properties of a working heat transfer process fluid (air).

II. CONCEPTUAL GEOMETRY AND THERMAL QUANTITIES

A spiral heat exchanger is made from two separate channels by welding two straight sheets of copper of a widthδ, to gather while maintaining a constant distance between them. This structure makes the total surface areas of the heat exchanger higher, and the heat loss smaller compared to the corresponding straight passages. Two coaxial passages are created. Connected to the headers, they can be viewed as a pair of concentric spiral passages whose axial cross-section is rectangular. The flow passages in spiral heat exchangers are in complete contrast to the shell and tube or double pipe exchangers. The paths in the former are rectangular in a cross-section and spiral in its shapes. Figure.1 shows a conceptual of a mid-height rectangular cross-section of Archimedes spiral geometry. Geometrical data of the spiral passages under consideration are given in Table1. The operational data and the constant thermo physical properties model are given in table.2 The geometrical of the heat exchanger is determined by many parameters such as minimum passage radius r_{min}, passage length L(x), and passage width (S- δ). The passage of a spiral heat exchanger is constructed as an Archimedean spiral in which the radius of curvature varies proportionally with the angle of revelation.

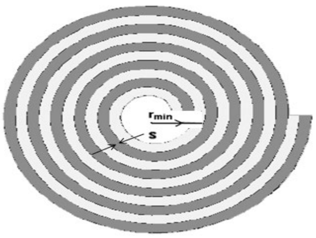

Fig.1.Conceptual geometry of a mid-height cross-section Of the spiral passages [10].

Table 1. Geometrical data of the spiral passages unit.

Height, b	0.65 m
Minimum Radius, R_{min}	0.05 m
Maximum Radius, R_{max}	0.38 m
Wall Thickness, δ	0.001 m
Channel width,(S- δ)	0.03 m

Table 2 Transport properties of the fluid and conditional temperatures used [10].

Parameter	Value
Air inlet temperature$T_i(K)$	293.0
Wall passage temperature $T_w(K)$	373.0
Air density $\rho(kg/m^3)$	1.02
Air specific heat $c_p(J/kg\ K)$	1.007×10^3
Air thermal conductivity $k\ (W/m\ K)$	0.027
Air dynamic viscosity μ [kg/ m s^2]	2.0×10^{-5}
Mass flow rate $\dot{m}$(kg/s)	0.062
Reference states, $T_0(K)$, $P_0\ (kPa)$	298.15, 101.325

The explicit empirical correlations for variable property model were taken from [11]. The hydraulic diameter of the spiral passage is given as

$$D_h = \frac{2(S-\partial)b}{(S-\partial+b)} \qquad (1)$$

And the cross-sectional of the rectangular passage is given as

$$A_c = (S-\delta)\,b \qquad (2)$$

The pumping power required to derive a fluid flow is given as

$$\dot{W} = A_c\, u_{ave}\Delta P = \frac{\dot{m}}{\rho}\Delta P \quad (3),$$

$Where, u_{ave} = averge\ velocity, and\ \Delta P = pressure\ drop$

The heat transfer to the bulk of the flow of fluid takes place through the average heat transfer coefficient, h_{ave} is given by

$$\dot{Q} = \dot{m}c_p(T_w - T_i)exp\left(1 - \frac{4h_{ave}}{\rho\, u_{ave}D_h c_p}x\right) \ (4)$$

III. CONVENTIAL EXERGY ANALYSIS

Although entropy generation is the measure of energy degradation in any system, it is not consisted to compare between entropy generation and energy. However the linkage between energy and exergy is straight forward. Since both are thermodynamic sate properties and have the same units.

Therefore the real loss in energy of the system can be evaluated via convential exergy analysis, due to direct relationship between exergy destruction and entropy generated via the environmental temperature (or dead state) [12,13].Exergy destruction of a system could be calculated either by the exergy balance equation or by the Gouy-Stodola relation [4, 5, 12].The Gouy-Stodola theorem is adopted in this work.

$$\dot{E}x_d = T_0\dot{S}_{g,Tot} \qquad (5)$$

The details study of both energy and entropy points of view were found in the work given by El-sagier as in [10]:

$$\dot{S}_{gen,Tol} = \dot{m}c_p\left[\ln\left(\frac{1-\tau e^{-Bx}}{1-\tau}\right) - \tau(1-e^{-Bx}) + \frac{1}{8}f\frac{Ec}{St}\left(ln\left(\frac{1-\tau e^{-Bx}}{1-\tau}\right) + Bx\right)\right] \ (5.1)$$

$$Where, \tau = \frac{T_w - T_i}{T_w}, Ec = \frac{u_{av2}^2}{c_pT_w}, and\ B = 4\frac{St}{D_h},$$

$$St = Stanon\ number, and\ f = friction\ factor$$

Splitting equation (5.1) into two parts

$$\dot{S}_{gen}(\Delta T) = mc_p\left[\ln\left(\frac{1-\tau e^{-Bx}}{1-\tau}\right) - \tau(1-e^{-Bx})\right] \quad (5.2)$$

Exergy destruction as a result of thermal effect is given as:

$$\dot{E}x_d(\Delta T) = T_0\dot{S}_{gen}(\Delta T) \quad (5.2a)$$

$$\dot{S}_{gen}(\Delta P) = \dot{m}c_p\left[\frac{1}{8}f\frac{Ec}{St}\left(ln\left(\frac{1-\tau e^{-Bx}}{1-\tau}\right) + Bx\right)\right] \quad (5.3)$$

Exergy destruction as a result of frictional effect is given as:

$$\dot{E}x_d(\Delta P) = T_0\dot{S}_{gen}(\Delta P) \quad (5.3a)$$

Another criterion describing the contribution of entropy generation due to energy exchange by heat on the overall entropy generation is the *Bejan number* (Be)[1,2]. This nondimensional number is defined as:

$$Be = \frac{\dot{S}_{gen}(\Delta T)}{\dot{S}_{gen}(\Delta T) + \dot{S}_{gen}(\Delta P)} \quad (5.4)$$

Note that Be (Bejan Number) has three limiting cases as.

$$\mathrm{Be} = \begin{cases} 0 \ \ (\text{irreversibility due } \Delta P) \\ 0.5 \ (\text{irreversibility due } \Delta P \text{ and } \Delta T \text{ are identical}) \\ 1.0 \ (\text{irreversibility due } \Delta T) \end{cases}$$

Following Bejan [2, 3], the most frequently used entropy generation number N_{gen} which is obtained by dividing the total entropy generation rate $\dot{S}_{gen,Tot}$ by the capacity flow rate $\dot{m}c_p$ .That is

$$N_{gen} = \frac{\dot{S}_{gen}}{\dot{m}c_p} \quad (5.5)$$

This is composed of heat transfer and viscous friction contributions as follows:

$$N_{gen} = (N_{gen})_T + (N_{gen})_P \quad (5.6)$$

$$(N_{gen})_T = \ln\left(\frac{1 - \tau e^{-Bx}}{1 - \tau}\right) - \tau(1 - e^{-Bx}) \quad (5.7)$$

$$(N_{gen})_P = \frac{1}{8} f \frac{Ec}{St}\left(\ln\left(\frac{1 - \tau e^{-Bx}}{1 - \tau}\right) + Bx\right) \quad (5.8)$$

IV. ANALYSIS AND DISCUSSIONS

A. Exergy Destruction Profiles along Spiral Passage

The spiral passages are curved ducts with varying curvature; therefore in spiral flow, the radius of curvature varies along the spiral passage, and hence, the flow does not have a single Reynolds number [14]. In this work Reynolds number varies between 8181.and 9576.

In the present study for constant thermo physical properties model $Re \approx 8826.$, for specified mass flow rate, and the corresponding Dean Number $De \approx 4557.$ that induces secondary flow due to curvature. While for variable physical properties model Re is varying. Figure 4.1 shows the thermal, $\dot{E}x_d(\Delta T)$ and viscous, $\dot{E}x_d(\Delta P)$ components of exergy destruction along the passage. It is observed that the variation of thermal component is increasing exponentially for both models of the working fluid until the middle of the curved passage then the behavior attained constant. While the variation of viscous dissipation component or frictional exergy destruction has a linear trend along the passage. However, Figure4.2 shows the behavior of total exergy destruction along the spiral passage. The behavior is dominated by thermal component of exergy destruction for both models. This behavior is of a particular significance in the heat transfer and fluid flow in spiral passages where secondary flow phenomenon plays an important role in thermal mixing.

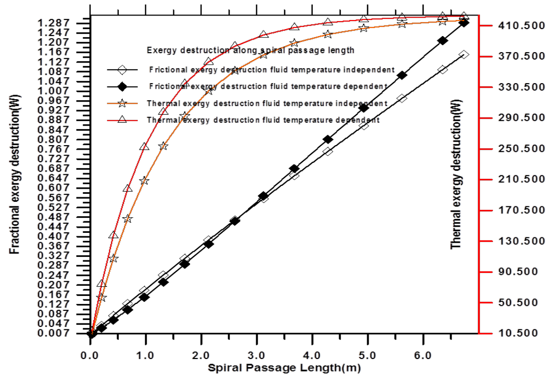

Fig4.1.Thermal and frictional exergy destruction profiles a long spiral passage.

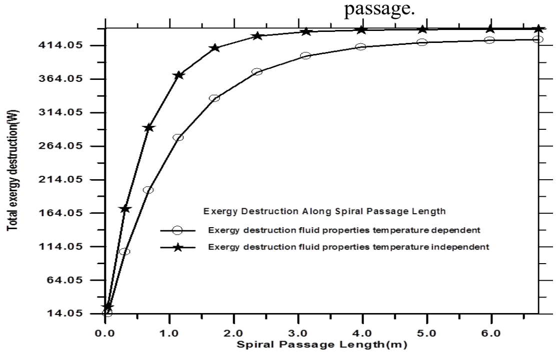

Fig4.2.Total exergy destruction a long spiral passage for both models

B. Exergy Destruction and Pumping Power along Spiral Passage

The total exergy destruction patterns and pumping power required to keep a fluid flow through spiral passage were evaluated in the case of constant fluid properties. The result of this evaluation is depicted in Figure4.3

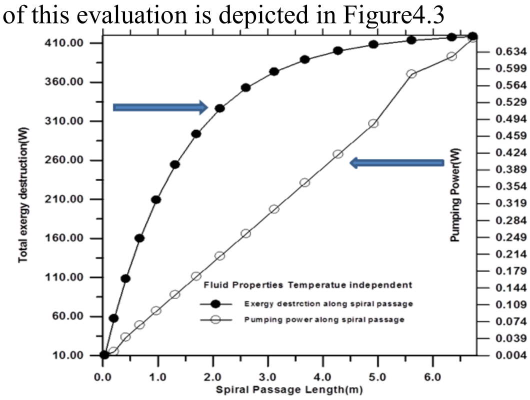

Fig4.3.Exergy destruction and pumping power required versus spiral passage

It is showed that total exergy destruction increase in exponential pattern .But pumping power behaves linearly along the passage. The influence of temperature dependent thermophysical properties of working fluid on exergy destruction patterns and pumping power is investigated. The behavior of both quantities is plotted in Figure4.4

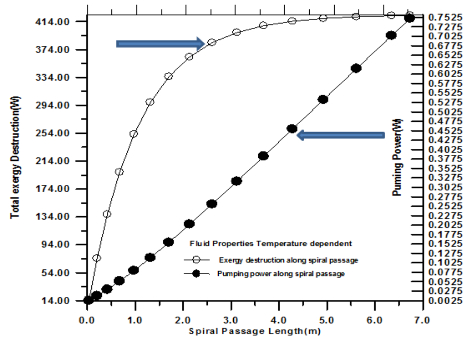

Fig4.4. Exergy destruction and pumping power required versus spiral passage for variable thermo physical properties.

A comparison between Figure4.3 and Figure4.4 shows that the total exergy destruction $\dot{Ex}_{d,tot}$ for variable thermophysical model was higher than that for constant thermophysical model.

C. Exergy Destruction and Heat Transfer Rate

The essential concern of heat exchanger design from the energetic point of view is the quantity of heat transfer. In order to assist the effect of the interaction between the thermal system and the environment on process performance, patterns of exergy destruction and heat transfer rate are analyzed. This is the most important concern from the exergetic point of view.

The variation of the total exergy destruction (thermal and frictional) and the rate of heat transfer along the passage are plotted in Figure4.5.

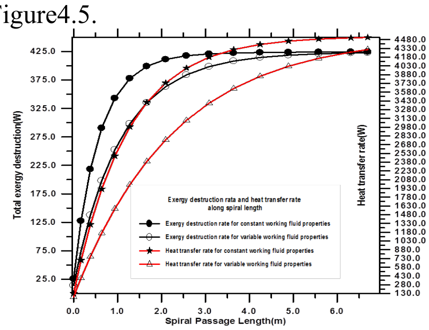

Fig4.5. Exergy destruction and heat transfer rate along spiral passage

The exponential behavior of both quantities is observed. In the investigation of both quantities, the influence of fluid properties temperature dependent is significant. Furthermore it is found that the calculated heat transfer coefficient due to swirling motion in spiral passage is more than two times of that in straight passage having the same variation of Re number.

***D.** Effect of Secondary Flow on Exergy Destruction*

Figure4.6 visualizes the trend of Bejan (Be) number and entropy generation number N_{gen} versus (DE) number as a measure of the secondary flow due to centrifugal force created by spiraling motion of the convective flow in the range of $(9576 \geq Re \geq 8181)$ along the spiral passage. It is shown that the irreversibity pattran in terms of Bejan number (Be) is increasing with increasing Den number which is a function of Reynolds number (Re). While in terms of N_{gen} is decreasing with increasing De. It is worthwhile to mention that secondary flow phenomenon plays opposite roles in a compact passage for the same two number of entropy generation as a direct measure of exergy destruction.

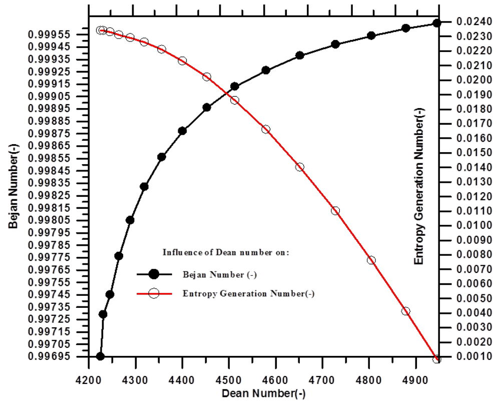

Fig 4.6.Bejan and entropy generation numbers versus De Number for variable thermo physical properties model.

The engineering importance of these two numbers is that the Bejan number (Be) performs as a visual map signifying the relative strength of thermal effects within the fluid domain interacted by the secondary flow and Dean vortices. This approach is then utilised for thermally optimizing the forced convection process in spiral passage. While entropy generation number N_{gen} represents a high or low exergy destruction rate.

V. CONCLUSIONS and RECOMENDATION

Exergy destruction of turbulent forced convective of air flow through a spiral passage subjected to isothermal boundary condition has been analyzed.

Two models of thermo physical properties have been considered, constant and variable properties models. The following conclusions have been extracted from the current investigation:

A. The enhancement of heat transfer process that characterizing by high Reynolds number fluid flow as well as secondary flow phenomenon in narrowing passage is affected by exergy destruction which dominates by temperature difference between fluid and passages walls.

B. In the assessment of heat transfer rate and exergy destruction, the influence of fluid properties temperature dependent is significant.

C. It is recommended that advanced exergy theory is very important to apply for evaluation exergy destruction minimization of fluid flow and heat transfer through such type of energy exchange and storage device.

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