Journal of Thermal Engineering
Web page info: https://jten.yildiz.edu.tr
DOI: 10.18186/thermal.977960

Journal of Thermal Engineering

Research Article

Effect of viscosity on entropy generation for laminar flow in helical pipes

R. PRATTIPATI[1], V. K. NARLA[2], and Srinivas PENDYALA[1,*]

[1]*Department of Mechanical Engineering, School of Technology, GITAM - Hyderabad, TS, India*
[2]*Department of Mathematics, School of Sciences, GITAM - Hyderabad, TS, India*

ARTICLE INFO

Article history
Received: 16 July 2019
Accepted: 22 November 2019

Key words:
Laminar flow; Viscosity; Helical pipe; Entropy; Exergy

ABSTRACT

Entropy generation for fully developed laminar flow in a helical pipe carrying high viscous fluid under constant temperature boundary conditions is investigated analytically. This work focuses on geometrical, fluid, and thermal aspects and their influence on irreversibilities in helical coils. The effect of viscosity on the irreversibilities and its influence on the operating parameters of the helical coil are studied with the second law of thermodynamics. The most commonly used relationships for estimating viscosity change due to temperature are selected for analysis. The entropy generation and avoidable exergy destruction in each case are presented. Bejan number is plotted for varying viscosities under different wall temperatures for both heat transfer to and from the fluid. The thermodynamic potential of improvement based on avoidable and unavoidable exergy destruction concepts showed that the potential of improvement for heating and the cooling condition is considerable for a given operating condition in helical tubes. The selected model for estimating viscosity influences the optimum operating wall temperature, thereby giving an insight into a selection of a proper viscosity model. The optimum helical number is not affected by fluid properties and wall temperature. The heat transfer to pumping ratio is evaluated and it is found that the optimal value is influenced by the change in viscosity.

Cite this article as: Prattipati R, Narla VK, Pendyala S. Effect of viscosity on entropy generation for laminar flow in helical pipes. J Ther Eng 2021;7(5):1100–1109.

INTRODUCTION

Flow-through coiled tubes is of theoretical and practical importance because of their secondary flow phenomenon and wide usage in engineering systems [1, 2]. Due to curvature and pitch of coil, buoyancy and centrifugal forces generate complex cross-stream motion increasing momentum and energy transport from the tube walls at the same time decreasing axial dispersion [3]. Diabatic flow creates density differences at the wall and influences the degree of coupling between the axial velocity and temperature distributions [4]. The numerical solution has been obtained for laminar flow by taking into consideration the effect of pitch and the

*__Corresponding author.__
*E-mail address: pendyala76@yahoo.com

This paper was recommended for publication in revised form by Regional Editor Mohammad Ghalambaz

Published by Yıldız Technical University Press, İstanbul, Turkey
Copyright 2021, Yıldız Technical University. This is an open access article under the CC BY-NC license (http://creativecommons.org/licenses/by-nc/4.0/).

results indicated that pitch is significant for coils where it exceeds the coil radius [5]. The influence of temperature on the fluid thermophysical properties along the length of the pipe as well as on the secondary flow has been studied by Kumar et al. [6]. A model has been developed for friction factor and heat transfer in helical pipes with varying thermophysical properties and reported heat transfer value increases up to 25%. This change necessitates the need for understanding the role of thermophysical properties to evaluate the heat transfer and pressure drop in thermal systems.

Bejan introduced entropy generation minimisation (EGM) as a method of modelling and optimization of devices accounting for irreversibilities in thermal systems [7]. Heat transfer and fluid friction are the prominent sources of irreversibilities associated with fluid flow in a pipe. The finite temperature difference between the fluid and wall generates thermal irreversibilities and the viscosity of fluid during flow causes frictional losses. Entropy generation analysis of fluid flow in a straight duct subjected to constant wall temperature has been investigated by Şahin [8]. Chamka [9] analysed fluid flow under oscillating and ramp pressure gradients, the transient flow and heat transfer of a particulate suspension in an electrically conductive channel fluid, and a circular pipe with an applied transverse magnetic field. Ko [10] explored the impacts of longitudinal ribs in a curved rectangular duct on laminar forced convection and entropy generation. Sanchez et al. [11] focused on laminar energy losses injunctions (bifurcations) and presented the energy losses injunctions by considering the entropy generation. Pendyala et al. [12] studied turbulent flow in helical coils through second-law analysis. Mehryan et al. [13] studied the impact of a periodic magnetic field on the natural convection and entropy generation of nanofluid flowing in a square enclosure. One of the primary objectives in designing any thermodynamic system is the efficient utilisation of exergy. The concepts of avoidable/unavoidable exergy destruction and investment cost analysis are combined with an exergoeconomic evaluation technique which is very useful in designing cost-effective energy systems. Tsatsaronis and Park [14] described a procedure to calculate the avoidable part of exergy destruction rate in a system component and the avoidable part of investment cost. This procedure was extended by Cziesla et al. [15] to analyse the exergoeconomic evaluation of a conceptual design of an advanced externally fired combined cycle (EFCC) power plant. The framework has been used and extended by Bahiraei et al. [16] to investigate the potential of improvement of helical coils based on avoidable and unavoidable exergy destruction concepts.

Transport of high viscosity liquids is encountered in chemical process and pharmaceutical industries. The effect of flow parameters and curvature ratio on the total entropy generation for laminar flow in helical pipe subjected to constant wall temperature has been analysed by Shokouhmand et al. [17]. Further, the work has been extended to find the optimum Reynolds number for air and water flow in helical pipes [18]. More recently first law analysis for flow in helical channels with varying viscosity has been performed [19]. It is found that viscosity is the most sensitive property among all thermophysical properties which may influence the heat transfer and pressure drop by a large amount. The effects of viscosity on entropy generation have been studied in smooth ducts for laminar flows [20]. The same formulation has been used for entropy generation analysis of laminar and turbulent flows in ducts subjected to a constant temperature, heat flux, and heat exchangers [21–24]. More rigorous work has been carried out by Chamka for various flow and geometrical conditions [25–32].

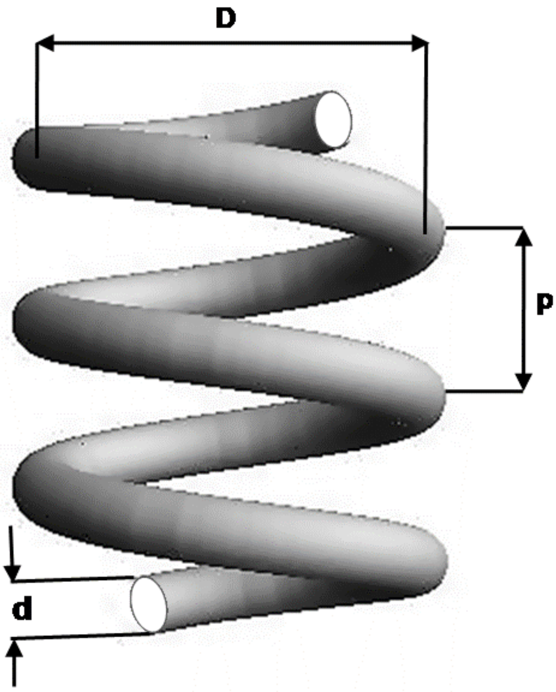

Figure 1. Schematic view of helical coil.

In the present study, the entropy generation rate and the thermodynamic potential of improvement of helically coiled tubes under cooling and heating conditions have been investigated. The flow velocity in the helical pipe is verified to be in the laminar region under constant wall temperature boundary conditions. In addition, the influence of viscosity variation is also discussed.

MATHEMATICAL ANALYSIS

The geometry of the system under consideration in this present study is a helically coiled tube as shown schematically in Figure 1. It consists of unperturbed tube diameter *d*, coil diameter *D*, and pitch of the coil *p*. The ratio of tube diameter to the coil diameter is diameter ratio of curvature ratio δ. The other important non-dimensional parameters in helically coiled tubes include Reynolds number

Re, Dean number *De* and helical number *He*, which are defined as:

$$Re = \frac{Ud}{\nu}, De = Re\sqrt{\delta}, He = \frac{De}{\sqrt{1+\gamma^2}} \tag{1}$$

where U is average velocity and $\gamma = \frac{p}{\pi D}$.

Effect of Viscosity on Friction Factor and Heat Transfer

The viscosity of fluids is affected by the change in bulk temperature. The viscosity variation due to temperature in some fluids may not necessitate a re-evaluation of heat transfer and pressure drop but in fluids such as glycerol, the change of viscosity is considerable. As a first approximation, a linear relationship is assumed between viscosity and temperature

$$\mu(T) = \mu_{ref} - bT_{ref}(\tau - 1) \tag{2}$$

where b is a positive fluid-dependent dimensional constant and τ is T/T_{ref} which is evaluated at wall and bulk conditions. This is a reasonable approximation if the variation of the viscosity due to bulk temperature is small. For highly viscous liquids, a more accurate empirical correlation is given by Sherman [33], where the viscosity varies exponentially with temperature

$$\mu(T) = \mu_{ref}(\tau)^a \exp\left[\frac{B}{T_{ref}}\left(\frac{1}{\tau} - 1\right)\right] \tag{3}$$

where a and B are fluid dependent constant parameters.

For evaluating the total entropy generation, the proposed correlations for Nusselt number and friction factor with different parameters of flow and geometry of the helical tube by Manlapaz and Churchill [5] have been used. Nusselt numbers obtained from correlations are satisfactory for small variations, but for large variations, the Nusselt number is multiplied by the ratio of viscosity at the bulk temperature to the viscosity at wall temperature, raised to a certain power, to correct for the variation of properties. It is given by

$$Nu = \left[\left(3.657 + \frac{4.343}{\left(1+\frac{957}{PrHe^2}\right)^2}\right)^3 + 1.158\left(\frac{He}{1+\frac{0.477}{Pr}}\right)^{3/2}\right]^{1/3} \times \left(\frac{\mu_b}{\mu_w}\right)^n \tag{4}$$

here, n is equal to 0.11 for heating and 0.25 for cooling. The bulk and wall μ values are obtained by substituting the respective temperature ratios in Eq. (2) for linear and Eq. (3) for exponential variation.

Similarly, the variation in physical properties effect on the friction factor is given by:

$$f = \frac{16}{Re}\left[\left(1 - \frac{0.18}{\left[1+\left(\frac{35}{He}\right)^2\right]^{0.5}}\right)^m + \left(1+\frac{\delta}{3}\right)^2 \frac{He}{88.33}\right]^{1/2} \times \left(\frac{\mu_b}{\mu_w}\right)^{-0.25} \tag{5}$$

values of $m = 2$, 1 and 0 were recommended for $De < 20$, $20 < De < 40$, and $De > 40$, respectively.

Entropy Generation

A fully developed incompressible laminar flow in a helical coil subject to uniform wall temperature is considered. According to the second law of thermodynamics applied to a control volume of the helical tube passage length dx, the relation is given by [34]:

$$\dot{S}'_{gen} dx = \dot{m} ds - \frac{\dot{Q}}{T_w} \tag{6}$$

Where, $\dot{Q} = \dot{m} dh$ is the rate of heat transferred into the control volume. The entropy change for an incompressible fluid is

$$ds = \frac{dh}{T} - \frac{1}{\rho T} dP \tag{7}$$

Therefore, $\dot{S}'_{gen}$ becomes

$$\dot{S}'_{gen} = \dot{m}\frac{dh}{dx}\left(\frac{1}{T} - \frac{1}{T_w}\right) + \frac{\dot{m}}{\rho T}\left(-\frac{dP}{dx}\right) \tag{8}$$

Solving $\dot{Q} = \dot{m} C_p dT = \bar{h} A(T_w - T)$, the bulk temperature at a cross-section is obtained as:

$$T = T_w - (T_w - T_i) exp\left(-\frac{4\bar{h}x}{\rho U d C_p}\right) \tag{9}$$

The dimensionless temperature Θ can be obtained from Eq. (9) as:

$$\Theta = \frac{\tau - \tau_w}{\tau_i - \tau_w} = exp\left(-\frac{4\bar{h}x}{\rho U d C_p}\right) \tag{10}$$

The pressure drop in Eq. (8) is evaluated using the relation,

$$-\frac{dP}{dx} = f\frac{\frac{1}{2}\rho U^2}{d} \tag{11}$$

The exact form of entropy generation can be obtained by integrating Eq. (8) along the helical tube passage length, using the Eqs. (9), (11). The dimensionless form of the entropy generation after substituting helical coil parameters can be written as:

$$N_S = \frac{\dot{S}'_{gen}}{\dot{m}C_p} = \underbrace{ln\left(\frac{e^{4\Lambda_1}-\theta}{1-\theta}\right) - 4\Lambda_1 + \theta\left(e^{4\Lambda_1}-1\right)}_{heat\ transfer\ contribution\ (N_{S,T})} + \underbrace{\frac{\Lambda_2}{8} ln\left(\frac{e^{4\Lambda_1}-\theta}{1-\theta}\right)}_{pressure\ drop\ contribution\ (N_{S,P})} \tag{12}$$

here, the dimensionless values θ, Λ_1 and Λ_2 are defined as follows:

$$\theta = 1 - \frac{\tau_i}{\tau_w}$$

$$\Lambda_1 = Nu\frac{1}{Pr}\sqrt{\frac{\delta}{He^2 d^2(1+\gamma^2)}}L$$

$$\Lambda_2 = \frac{1}{Nu\tau_w}\frac{\nu^2 Pr}{C_p T_{ref}}\left[\frac{He^2}{d^{4/3}}\left(\frac{1+\gamma^2}{\delta}\right)\right]^{\frac{3}{2}}$$

RESULTS AND DISCUSSION

In this analysis, the inherent irreversibilities in the flow for high viscous fluids and the influence of temperature on the fluid in a helical tube are investigated. Validation of the current analysis has been given by Prattipati et al. [35]. The effects of various geometric and fluid parameters are analysed for both heating and cooling conditions. The cooling condition is that the wall temperature is higher and the liquid flows through the pipe cooling the surface and vice-versa for heating conditions. The flow velocity in the helical pipe is assumed to be in the laminar region and the inlet and wall temperatures are specified. Water and glycerol are considered as working fluids and thermophysical values are shown in Table 1 [21]. The reference temperature is taken as 293 K at which the properties of water and glycerol are taken. The dimensionless wall temperature (τ_w) is varied from 0.8–1.2, 1 representing no difference between wall temperature and bulk fluid temperature.

Table 1. Thermophysical properties

Variable	Water	Glycerol
b	8.943×10^{-6}	0.0182
B	4700	23100
C_p	4182	2428
k	0.6	0.264
a	8.9	52.4
m_{ref}	9.93×10^{-4}	1.48

The curvature ratio has been varied from 0.026 to 0.3 and pitch from 0.05 to 2. By keeping the Reynolds number in the laminar regime, the Helical number ranged from 100–300. The Reynolds number is ensured to be below the critical Reynolds number which is given by Srinivasan et al. [36], as:

$$Re_{cr} = 2100\left(1+12\sqrt{\delta}\right) \tag{13}$$

Irreversibility Analysis

Thermodynamic irreversibility in the thermal process is calculated through entropy generation. The total entropy generation rate is the sum of two entropy generation rates $N_{S,T}$ and $N_{S,P}$, where each is associated with a specific source of irreversibility as shown in Eq. (12). Figure 2 shows the variation of entropy generation versus Λ_1 for cooling and heating. The parameter Λ_1 can be viewed as three groups of variables comprising heat transfer, fluid, and geometrical kind. The parameter Λ_2 is a combination of heat transfer and geometric kind group of variables. For a given wall temperature, three viscosity models namely constant, linear and exponential are used for heating and cooling to make a total of six cases.

These models influence the Nusselt number term in Λ_1 by only a small amount, hence the most influencing parameter is the passage length of the helical pipe. As the Λ_1 increases the total irreversibilities increase.

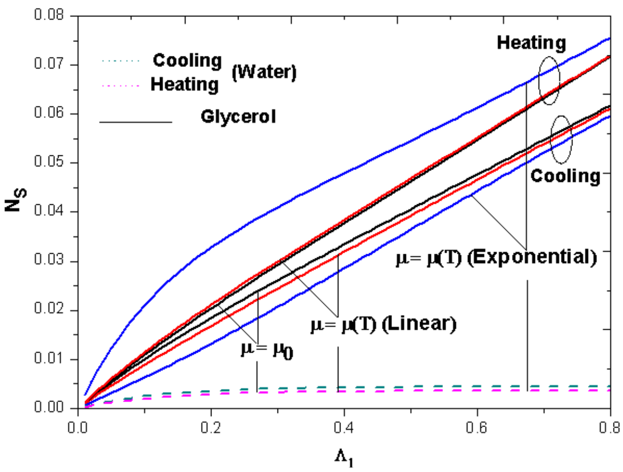

Figure 2. Entropy generation variation with Λ_1 number for $\tau_w = 1.1$ (cooling), $\tau_w = 0.92$ (heating) and $He = 80$.

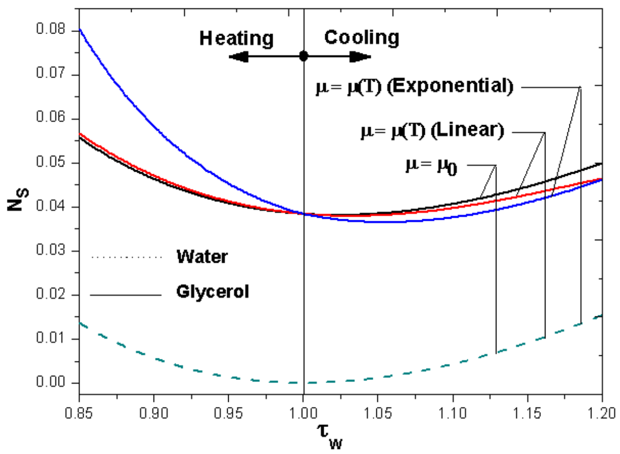

Figure 3. Entropy generation variation with wall temperature ratio for $\Lambda_1 = 0.2$ and $He = 80$.

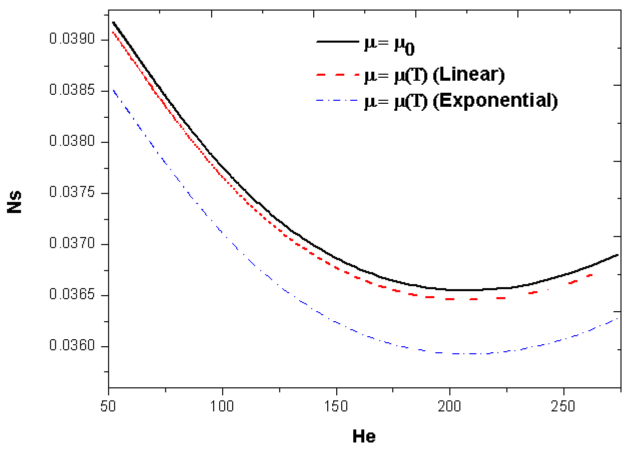

Figure 4. Entropy generation variation with Helical number for $\tau_w = 1.15$ (cooling) and $\Lambda_1 = 0.5$.

This can be attributed to increasing frictional losses along the length of the pipe. For cooling conditions, the viscosity of a liquid decreases thereby reducing the irreversibilities due to friction. The viscosity change is more when exponential variation is considered and the entropy generation number falls further. The same can be observed with a heating conditions where the viscosity value is more than reference viscosity that increases the frictional losses in the total entropy generation. The irreversibilities caused due to the viscosity change with temperature in the water is almost negligible, but in glycerol, it varies significantly. This is due to the large difference in viscosity and viscosity change. The pressure drop contribution to the total entropy generation in glycerol is much higher than water.

Figure 3 shows the influence of wall temperature on entropy generation rate. The effect of viscosity change is almost negligible for water. However, the effect of the assumed variation viscosity on entropy generation is apparent in the case of glycerol for cooling and heating. The adiabatic flow condition is satisfied when the wall temperature equals the reference temperature, which is $\tau_w = 1$. The irreversibilities based on the constant viscosity model is higher than those evaluated for the models of viscosity dependent on temperature to the right side of the adiabatic value in Figure 3. The curve corresponding to the linear viscosity model eventually approaches the exponential viscosity model curve. For heating, the entropy generation evaluated based on the exponential viscosity model produces higher values than the values obtained for the other two viscosity models considered. Water and glycerol vary in a similar manner when viscosity is constant with a change in magnitude. When the viscosity is corrected for change in temperature, the entropy generation increases for heating and it is much slower to increase for cooling.

The entropy generation number is found for different geometrical parameters of the helical coil under heating and cooling conditions. The entropy generation magnitude however differs from cooling to heating where the high viscous liquid generates maximum entropy. In many engineering designs and industrial problems, the ratio of the entropy generation due to heat transfer to the total entropy generation is needed. As an alternative irreversibility distribution parameter, Paoletti et al. [37] presented Bejan number Be, which is defined as:

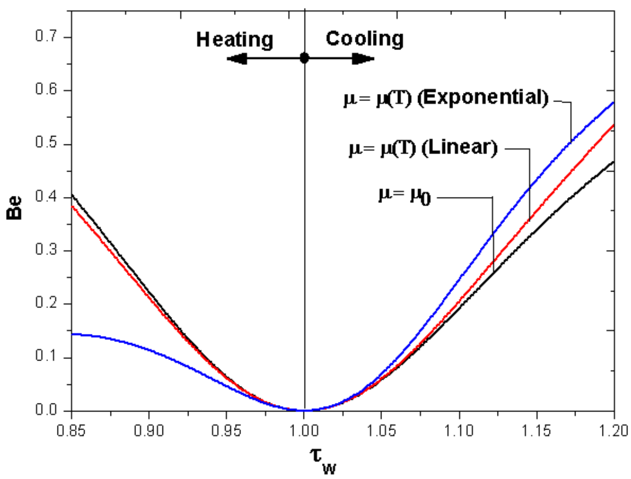

Figure 5. Bejan number vs wall temperature ratio for $\Lambda_1 = 0.2$ and $He = 80$.

$$Be = \frac{N_{S,T}}{N_S} \tag{14}$$

The Bejan number value lies in a range of $0 \le Be \le 1$. If $Be \to 0$, then the irreversibility is dominated by the effect of fluid friction, but if $Be \to 1$, then the irreversibility due to heat transfer dominates the flow system by the virtue of finite temperature differences. The effect of different wall temperature ratios on Bejan number is illustrated in

Figure 5. At the adiabatic condition (τ_w = 1), there is no heat transfer and the total irreversibilities are caused by flow only making Be = 0 for all viscosity models as shown in Figure 5. When the wall temperature is lower than the bulk temperature, the heating condition prevails and the exponential viscosity variation gives the lowest Be owing to the highest viscosity. The linear viscosity relationship mostly lies in between the constant value and the exponential variation.

Avoidable Exergy Destruction

The irreversibility of any thermal process can be calculated by two different approaches, one approach is exergy balance using

Irreversibility = total exergy inflow – total exergy outflow

Another alternative approach is the Gouy-Stodola relationship that is given [38] as:

$$Irreversibility = T_0 \times entropy\ generation\ rate,$$

where T_0 is the absolute temperature of the appropriate environment. The potential of improvement of any thermal system can be obtained by utilising the above two relations. The total exergy destruction rate $\dot{E}_D$ of any thermal system can be written into two components:

$$\dot{E}_D = \dot{E}_D^{AV} + \dot{E}_D^{UN} \tag{15}$$

According to the relation between exergy balance and Gouy-Stodola relationship the exergy destruction can be written as:

$$\begin{aligned} \dot{E}_D &= \dot{I}_{tot} = T_0 \dot{S}_{gen}, \\ \dot{E}_D^{UN} &= \dot{I}_{tot,min} = T_0 \dot{S}_{gen,min} \end{aligned} \tag{16}$$

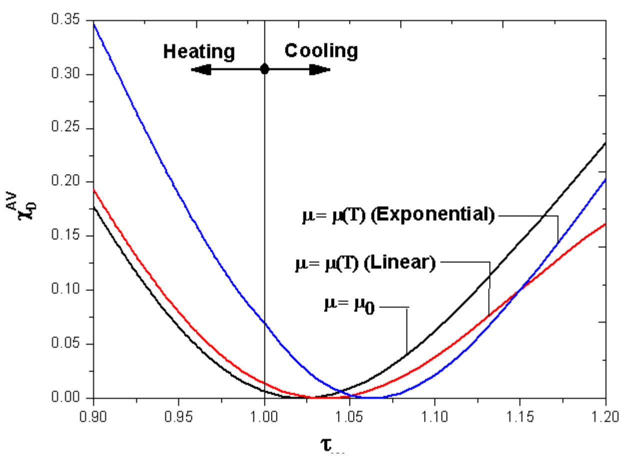

Figure 6. Potential of improvement with wall temperature for Λ_1 = 0.2 and He = 80.

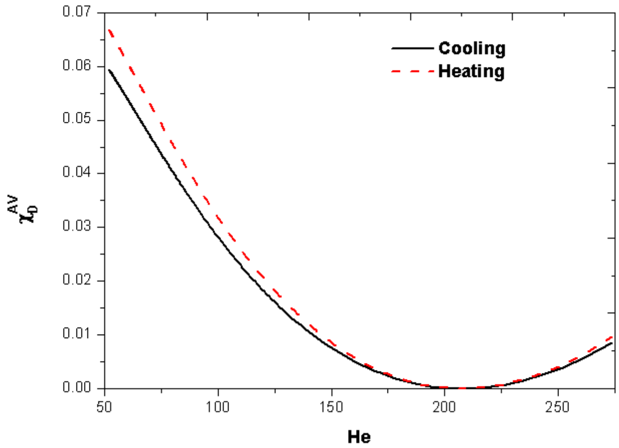

Figure 7. Potential of improvement with Helical number for τ_w = 1.1 (cooling), τ_w = 0.92 (heating) and He = 80.

A thermodynamic measure for the potential of improvement χ_D^{AV} of a thermal system component is introduced in Cziesla et al. [15] as:

$$\chi_D^{AV} = \frac{\dot{E}_D^{AV}}{\dot{E}_D} \tag{17}$$

Combine Eqs. (15), (16) and (17), we get

$$\chi_D^{AV} = \frac{\dot{E}_D - \dot{E}_D^{UN}}{\dot{E}_D} = \frac{\dot{S}_{gen} - \dot{S}_{gen,min}}{\dot{S}_{gen}} = \frac{N_S - N_{S,min}}{N_S} \tag{18}$$

The value $N_{S,min}$ can be calculated through N_S of that particular parameter optimum value. Figures 6 and 7 show the variation of the potential of improvement for glycerol with three cases of viscosity dependence. As shown in Figure 6, the effect of the assumed variation of viscosity on χ_D^{AV} is considerable in wall temperatures that are either side from adiabatic value τ_w = 1. For cooling, it can be observed that almost 20–25% of total exergy destruction can be avoided in the case of the constant viscosity assumption. Whereas for heating, the potential of improvement value is high for the case of the exponential viscosity model. It can be observed that up to 35% of the total exergy destruction can be avoided.

The potential optimisation in helical number He for cooling and heating scenarios is shown in Figure 7. Whenever the contribution of heat transfer dominates, the potential of improvement is possible up to 6–7% in helical numbers which are less than the optimum helical value (around 200).

Whenever, friction contribution dominates, the avoidable exergy destruction is as little as 1% in helical numbers which are greater than the optimum value around 200. The optimum helical number was calculated for the case of laminar flow operating conditions. The thermodynamic

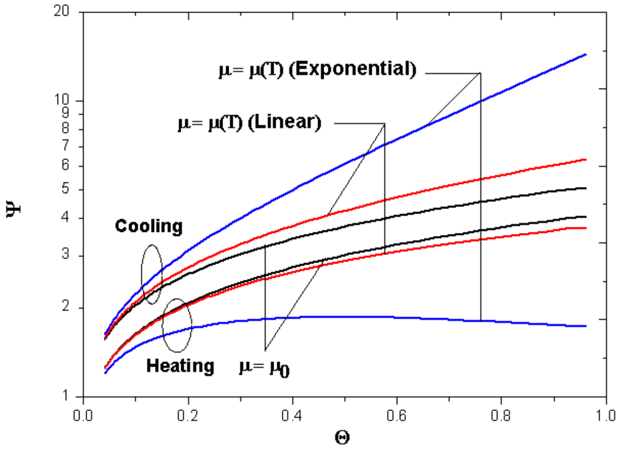

Figure 8. Heat transfer ratio with dimensionless temperature difference for $\tau_w = 1.1$ (cooling), $\tau_w = 0.92$ (heating) and $He = 80$.

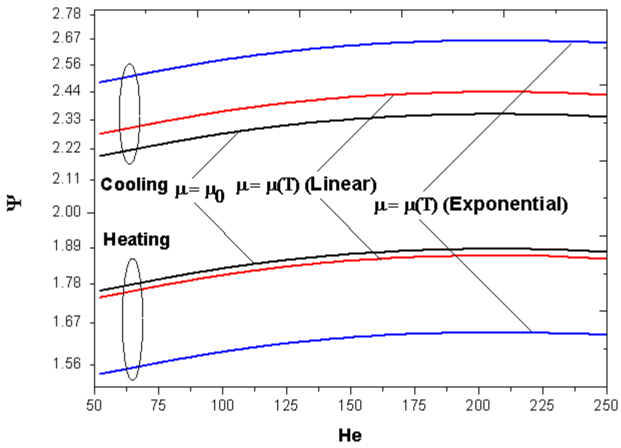

Figure 9. Heat transfer ratio with Helical number for $\tau_w = 1.1$ (cooling), $\tau_w = 0.92$ (heating) and $He = 80$.

performance of helical coils thus can be improved by selecting appropriate design parameters.

Heat Transfer Rate to Pumping Power Ratio

The heat transfer enhancement is usually associated with increase in friction factor. The rate of heat transfer per volumetric flow rate to pressure drop can give an estimate for different configurations

$$\Psi = \frac{\dot{Q}}{\phi \Delta P} \tag{19}$$

After substituting $\dot{Q} = \dot{m} C_p dT = \bar{h} A (T_w - T)$ and Eq. (11) in Eq. (19), the heat transfer rate to pumping power ratio can be obtained by introducing helical parameters as:

$$\Psi = \frac{2\theta(1 - e^{-\Lambda_1})}{\Lambda_1 \Lambda_2} \tag{20}$$

A measure of heat transfer rate to pumping power ratio is given in Eq. (20) compares the heat transfer enhancement to the input power consumed by changes in viscosity. Figure 8 shows the change of the heat transfer rate to pumping power ratio with dimensionless inlet wall to fluid temperature difference for cooling and heating with the three cases of viscosity dependence. The heat transfer to pumping power ratio is a higher for cooling conditions and lower for heating conditions as expected. The exponential viscosity assumption gives higher heat transfer rate to pumping power ratio for cooling. It is observed that the behaviour of the curves in the heating condition is opposite to that in the cooling condition. In both cases the viscosity variation is apparent for high values of Θ.

It is interesting to notice that the exponential relation of viscosity tends to increase the ratio which suggests that for operation under the ranges of values considered, this particular temperature may be the best operating condition. For all other cases, as the bulk temperature nears the inlet temperature, the ratio reduces thereby suggesting that inlet temperature being low in fact is desirable in terms of power being consumed. Variation of Ψ with helical numbers is shown in Figure 9. The maximum heat transfer rate to pumping power ratio can be observed at the optimum helical number for cooling and heating with the three cases of viscosity dependence.

CONCLUSION

An analytical study has been made for fully developed laminar flow in helical pipes subjected to constant wall temperature. The two cases of heating and cooling have been analysed for the changes in viscosity effects on the entropy generation rate. Three viscosity relations, namely constant, linear and exponential variations with temperature are taken for estimating the viscosities. The conclusions are summarized as follows:

- The entropy generation for water is almost the same for all the relations since the viscosity changes little with temperature. However, for glycerol, the viscosity effect shows considerable difference in the entropy generation where the colder liquid showed large irreversibility due to friction.
- The exponential viscosity model gives a more accurate value for high viscous liquids. If a linear model of viscosity is being chosen, it is recommended to carefully select the limits of the linear relationship as it may not be valid beyond a certain range.
- The thermodynamic potential of improvement analysis revealed that up to 20–25% of total exergy

destruction can be avoided for heating conditions based on a constant viscosity model. Whereas for heating conditions, up to 35% of total exergy destruction can be avoided based on the exponential viscosity model for the selected range of variables considered in this analysis.

- Maximum value of heat transfer to pumping power ratio is influenced by the change in viscosity and is obtained at optimum helical value and the ratio tends to decrease for heating condition based on exponential viscosity model.
- Furthermore, relating the concepts of avoidable exergy destruction that is presented in this work and avoidable investment cost analysis can be very useful in designing cost-effective energy systems.

NOMENCLATURE

$\bar{h}$	Heat transfer coefficient, W / m^2 K
C_p	Specific heat, kJ / kg K
D	Coil diameter, m
d	Pipe diameter, m
De	Dean number
f	Friction factor
He	Helical number
k	Thermal conductivity, W / m K
N_s	Dimensionless entropy
Nu	Nusselt number
p	Pitch of the coil, m
Pr	Prandtl number
Re	Reynolds number
T	Temperature, K
U	Velocity, m / s

Greek symbols

χ	Potential of improvement
δ	Curvature ratio
γ	Pitch to coil diameter ratio
μ	Dynamic viscosity, kg m^{-1} s^{-1}
ν	Kinematic viscosity, m^2 / s
ρ	Density kg / m^3
ϕ	Volume flow rate m^3 / s
Ψ	Heat transfer rate to pumping power ratio
τ	Temperature ratio
Θ	Ratio of dimensionless temperature difference
θ	Ratio of dimensionless temperature with reference to wall

Subscripts

b	Refers to bulk fluid
i	Refers to inlet
min	Refers to minimum
ref	Refers to reference conditions
tot	Refers to total
w	Refers to wall

AUTHORSHIP CONTRIBUTIONS

R. Prattipati: Design, Analysis, Literature search, Writing. V.K. Narla: Concept, Data, Materials. Srinivas Pendyala: Supervision, Analysis, Critical revision

DATA AVAILABILITY STATEMENT

No new data were created in this study. The published publication includes all graphics collected or developed during the study.

CONFLICT OF INTEREST

The author declared no potential conflicts of interest with respect to the research, authorship, and/or publication of this article.

ETHICS

There are no ethical issues with the publication of this manuscript.

REFERENCES

[1] Naphon, P., Wongwises, S. A review of flow and heat transfer characteristics in curved tubes. Renewable and Sustainable Energy Reviews 2006;10: 463–90. https://doi.org/10.1016/j.rser.2004.09.014.

[2] Prasad, B.V., Das, D.H., Prabhakar, A.K. Pressure drop, heat transfer and performance of a helical coil tubular exchanger. Heat Recovery Systems and CHP 1989;9: 249–56. https://doi.org/10.1016/0890-4332(89)90008-2.

[3] Goering, D.J., Humphrey, J.A.C., Greif, R. The dual influence of curvature and buoyancy in fully developed tube flow. Int. J. Heat Mass Transfer 1997;40: 2187–99. https://doi.org/10.1016/S0017-9310(96)00248-7.

[4] Ciofalo, M., Arini, A., Liberto, M.D. On the influence of gravitational and centrifugal buoyancy on laminar flow and heat transfer in curved pipes and coils. Int. J. Heat Mass Transfer 2015;82: 123–34. https://doi.org/10.1016/j.ijheatmasstransfer.2014.10.074.

[5] Manlapaz, E., Churchill, S.W. Fully developed laminar convection from a helical coil. Chem Eng Commun 1981;9: 185–200. https://doi.org/10.1080/00986448108911023.

[5] Kumar, V., Gupta, P., Nigam, K.D.P. Fluid flow and heat transfer in curved tubes with temperature-dependent properties. Ind. Eng. Chem. Res. 2007;46: 3226–36. https://doi.org/10.1021/ie0608399.

[6] Bejan, A. Second-law analysis in heat transfer and thermal design. Adv. Heat Transfer 1982;15: 1–58. https://doi.org/10.1016/S0065-2717(08)70172-2.

[7] Şahin, A. Z. A second law comparison for optimum shape of duct subjected to constant wall temperature and laminar flow. Heat and Mass Transfer 1998;33: 425–30. https://doi.org/10.1007/s002310050210.
[8] Chamkha, A.J. Unsteady laminar hydromagnetic fluid particle flow and heat transfer in channels and circular pipes. International Journal of Heat and Fluid Flow 2000;21: 740–746. https://doi.org/10.1016/S0142-727X(00)00031-X.
[9] Ko, T.H. A numerical study on entropy generation and optimization for laminar forced convection in a rectangular curved duct with longitudinal ribs. International Journal of Thermal Sciences 2006;45: 1113–25. https://doi.org/10.1016/j.ijthermalsci.2006.03.003.
[10] Sanchez, M., Henderson, A.W., Papavassiliou, D.V., Lemley, E.C. Entropy generation in laminar flow junctions. In: ASME 2012 Fluids Engineering Division Summer Meeting collocated with the ASME 2012 Heat Transfer Summer Conference and the ASME 2012 10th International Conference on Nanochannels, Microchannels, and Minichannels, pp. 325–30. Fluids Engineering Division, ASME. https://doi.org/10.1115/FEDSM2012-72334.
[11] Pendyala, S., Narla,V. K., Prattipati, R. Second law analysis for turbulent flow in helical pipes subject to variable viscosity. AIP Conference Proceedings. 2020;2246: 020038. https://doi.org/10.1063/5.0014559.
[12] Mehryan, S., Izadi, M., Chamkha, A.J., Sheremet, M.A. Natural convection and entropy generation of a ferrouid in a square enclosure under the effect of a horizontal periodic magnetic field. Journal of Molecular Liquids 2018;263: 510–25. https://doi.org/10.1016/j.molliq.2018.04.119.
[13] Tsatsaronis, G., Park, M.H. On avoidable and unavoidable exergy destructions and investment costs in thermal systems. Energy Conversion and Management 2002;43: 1259–70. https://doi.org/10.1016/S0196-8904(02)00012-2.
[14] Cziesla, F., Tsatsaronis, G., Gao, Z. Avoidable thermodynamic inefficiencies and costs in an externally fired combined cycle power plant. Energy 2006;31: 1472–89. https://doi.org/10.1016/j.energy.2005.08.001.
[15] Bahiraei, F., Saray, R.K., Salehzadeh, A. Investigation of potential of improvement of helical coils based on avoidable and unavoidable exergy destruction concepts. Energy 2011; 36:3113–9. https://doi.org/10.1016/ j.energy.2011.02.057.
[16] Shokouhmand, H., Salimpour, M.R. Entropy generation analysis of fully developed laminar forced convection in a helical tube with uniform wall temperature. Heat Mass Transfer 2007;44: 213–20. https://doi.org/10.1007/s00231-007-0235-x.
[17] Shokouhmand, H., Salimpour, M.R. Optimal Reynolds number of laminar forced convection in a helical tube subjected to uniform wall temperature. Int Comm Heat Mass Transf. 2007;34: 753–61.https://doi.org/10.1016/j.icheatmasstransfer.2007.02.010.
[18] Wang, C., Liu, S., Wu, J., Li, Z. Effects of temperature-dependent viscosity on fluid flow and heat transfer in a helical rectangular duct with a finite pitch. Brazilian Journal of Chemical Engineering 2014;3: 787–97. https://doi.org/10.1590/0104-6632.20140313s00002676.
[19] Şahin, A.Z. Thermodynamics of laminar viscous flow through a duct subjected to constant heat flux. Energy 1996;21: 1179–89. https://doi.org/10.1016/0360-5442(96)00062-X
[20] Şahin, A.Z. Thermodynamic design optimization of a heat recuperator. Int. Comm. Heat Mass Transfer 1997;24: 1029–38. https://doi.org/10.1016/S0735-1933(97)00088-2.
[21] Şahin, A.Z. Effect of viscosity on effectiveness of parallel flow heat exchanger. Energy Convers. Mgmt 1998;39: 1233–8. https://doi.org/10.1016/S0196-8904(98)00013-2.
[22] Şahin, A.Z. Second law analysis of laminar viscous flow through a duct subjected to constant wall temperature. ASME Journal of Heat Transfer 1998;120: 76–83. https://doi.org/10.1115/1.2830068.
[23] Şahin, A.Z. Entropy generation in turbulent liquid flow through a smooth duct subjected to constant wall temperature. Int. J. Heat Mass Transfer 2000;43: 1469–78. https://doi.org/10.1016/S0017-9310(99)00216-1.
[24] Chamkha, A. J. On laminar hydromagnetic mixed convection flow in a vertical channel with symmetric and asymmetric wall heating conditions, International Journal of Heat and Mass Transfer 2002;45: 2509–25. https://doi.org/10.1016/S0017-9310(01)00342-8.
[25] Chamkha, A. Unsteady laminar hydromagnetic flow and heat transfer in porous channels with temperature-dependent properties, International Journal of Numerical Methods for Heat & Fluid Flow 2001;11: 430–48. https://doi.org/10.1108/EUM0000000005529.
[26] Chamkha, A. J., Grosan T., Pop I. Fully developed free convection of a micropolar fluid in a vertical channel, International Communications in Heat and Mass Transfer, 2002; 29: 1119–27. https://doi.org/10.1016/S0735-1933(02)00440-2.
[27] Chamkha, A. J., Grosan T., Pop I. Fully Developed Mixed Convection of a Micropolar Fluid in a Vertical Channel, International Journal of Fluid Mechanics Research 2003;30: 251–63. DOI: 10.1615/InterJFluidMechRes.v30.i3.10.

[28] Chamkha, A. J. Flow of Two-Immiscible Fluids in Porous and Nonporous Channels. ASME. J. Fluids Eng. March 2000; 122: 117–24. https://doi.org/10.1115/1.483233.
[29] Mansoor, S. Entropy generation rate in a microscale thin film. Journal of Thermal Engineering 2019;5: 405–13.
[30] Kurtulmuş N., Bilgili M., Şahin B. Energy and Exergy analysis of a vapor absorption refrigeration system in an intercity bus application. Journal of Thermal Engineering 2019;5: 355–71. https://doi.org/10.18186/thermal.583316.
[31] Kaşka, Ö., Bor, O., Tokgöz, N., Aksoy, M. First and second law evaluation of combined Brayton-Organic Rankine power cycle. Journal of Thermal Engineering 2020;6: 577–91. https://doi.org/10.18186/thermal.764299.
[32] Sherman, F.S. Viscous Flow. McGraw-Hill Co., New York; 1990.
[33] Bejan, A.: Entropy generation minimization: the new thermodynamics of finite size devices and finite-time processes. Applied Physics Reviews 1996;79: 1191–218. https://doi.org/10.1063/1.362674.
[34] Prattipati, R., Narla, V.K., Pendyala, S., Prasad, B. Performance comparison of straight and helical pipes subjected to constant heat flux. In: Proceedings of the 6th International and 43rd National Conference on Fluid Mechanics and Fluid Power. 2016, MN NITA, MNNIT, Allahabad, India.
[35] Srinivasan, P.S., Nandapurkar, S.S., Holland, F.A. Pressure drop and heat transfer in coils. Chem. Res. 1968;218: 113–9.
[36] Paoletti, S., Rispoli, F., Sciubba, E. Calculation of exergetic losses in compact heat exchanger passages. ASME AES 1989;10: 21–9.
[37] Dincer, I., Rosen, M.A. Energy, Environment And Sustainable Development. Elsevier, Kidlington, UK; 2013.