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.

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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 inclusions (bifurcations) and presented the energy losses inclusions 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].

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 \( \delta \). 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:

$$Re = \frac{Ud}{v}, \quad De = Re \sqrt{\delta}, \quad He = \frac{De}{\sqrt{1 + \gamma^2}}$$  \hspace{1cm} (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

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

where $b$ is a positive fluid dependent dimensional constant and $\tau$ 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

$$\mu (T) = \mu_{ref} \left( \frac{T}{T_{ref}} \right)^{a - \frac{B}{3}}$$  \hspace{1cm} (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

$$Nu = \left[ 3.657 + \frac{4.343}{(1 + \frac{957}{PrHe})^{2}} \right]^{3/2} + 1.158 \left( \frac{He}{Pr} \frac{0.477}{1 + \frac{477}{Pr}} \right)^{3/2} \times \left( \frac{\mu_{w}}{\mu_{w}} \right)^{n/3}$$  \hspace{1cm} (4)

where $n$ is equal to 0.11 for heating and 0.25 for cooling. The bulk and wall $\mu$ 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:

$$f = \frac{16}{Re} \left[ \left( 1 - \frac{0.18}{1 + \left( \frac{35}{He} \right)^{2}} \right)^{m} + \left( 1 + \frac{\delta}{3} \right)^{\frac{He}{88.33}} \right]^{1/2} \times \left( \frac{\mu_{w}}{\mu_{w}} \right)^{-0.25}$$  \hspace{1cm} (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]:

$$\dot{S}_{gen} dx = m dS - \frac{\dot{Q}}{T_w}$$  \hspace{1cm} (6)

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

$$ds = \frac{dh}{T} - \frac{1}{\rho T} dP$$  \hspace{1cm} (7)

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

$$\dot{S}_{gen} = m \left( \frac{dh}{dx} \frac{1}{T} - \frac{1}{\rho T} \frac{dP}{dx} \right) + \frac{m}{\rho T} \left( - \frac{dP}{dx} \right)$$  \hspace{1cm} (8)

Solving $\dot{Q} = mC_{p}dT = \overline{h}A(T_{w} - T)$, the bulk temperature at a cross-section is obtained as:

$$T = T_{w} - (T_{w} - T) \exp \left( -\frac{4\overline{h}x}{\rho UdC_{p}} \right)$$  \hspace{1cm} (9)

The dimensionless temperature $\Theta$ can be obtained from Eq. (9) as:

$$\Theta = \frac{\tau - \tau_{w}}{\tau_{i} - \tau_{w}} = \exp \left( -\frac{4\overline{h}x}{\rho UdC_{p}} \right)$$  \hspace{1cm} (10)
The pressure drop in Eq. (8) is evaluated using the relation,
\[-\frac{dP}{dx} = \rho \mu \left( \frac{dU}{dx} \right)^2 \]  
(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:

\[ N_i = \frac{S_{\text{gen}}}{mC_p} = \ln \left( \frac{e^{\theta / \delta} - 1}{1 - \theta} \right) - 4A_n + \theta \left( e^{\theta / \delta} - 1 \right) + \frac{\Lambda_n}{8} \ln \left( \frac{e^{\theta / \delta} - 1}{1 - \theta} \right) \]  
(12)

here, the dimensionless values \( \theta \), \( \Lambda_1 \) and \( \Lambda_2 \) are defined as follows:

\[ \theta = 1 - \frac{\tau_w}{\tau_w} \]

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

\[ \Lambda_2 = \frac{1}{Nur_w C_p T_{\text{ref}}} \left[ \frac{He^2 \left( 1 + \gamma^2 \right)}{\nu^2 \left( 1 + \gamma^2 \right) \delta} \right]^{\frac{3}{2}} \]

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 (\( \tau_w \)) is varied from 0.8–1.2, 1 representing no difference between wall temperature and bulk fluid temperature.

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:

\[ Re_c = 2100 \left( 1 + 12 \sqrt{\delta} \right) \]  
(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 \( \Lambda_1 \) for cooling and heating. The parameter \( \Lambda_1 \) can be viewed as three groups of variables comprising heat transfer, fluid, and geometrical kind. The parameter \( \Lambda_2 \) 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 \( \Lambda_1 \) by only a small amount, hence the most influencing parameter is the passage length of the helical pipe. As the \( \Lambda_1 \) increases the total irreversibilities increase.

Table 1. Thermophysical properties

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

Figure 2. Entropy generation variation with \( \Lambda_1 \) number for \( \tau_w = 1.1 \) (cooling), \( \tau_w = 0.92 \) (heating) and \( He = 80 \).
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 \( \text{Be} \), which is defined as:

\[
\text{Be} = \frac{N_{\text{s,fr}}}{N_{\text{s}}} \quad (14)
\]

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 \( \Lambda_1 = 0.5 \).

Figure 5. Bejan number vs wall temperature ratio for \( \Lambda_1 = 0.2 \) and \( He = 80 \).
Figure 5. At the adiabatic condition ($\tau_w = 1$), 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

\[
\text{Irreversibility} = \text{total exergy inflow} - \text{total exergy outflow}
\]

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

\[
\text{Irreversibility} = T_0 \times \text{entropy generation rate},
\]

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:

\[
\dot{E}_D = \dot{E}_D^{AV} + \dot{E}_D^{UN}
\]  
(15)

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

\[
\dot{E}_D = i_{\text{int}} = T_0 S_{\text{gen}},\quad \dot{E}_D^{UN} = i_{\text{int, min}} = T_0 S_{\text{gen, min}}
\]  
(16)

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

\[
\chi_D^{AV} = \frac{\dot{E}_D^{AV}}{\dot{E}_D^{AV}} = \frac{\dot{S}_{\text{gen}} - \dot{S}_{\text{gen, min}}}{S_{\text{gen}}} = \frac{N_s - N_{s, \text{min}}}{N_s}
\]  
(18)

The value $N_{s, \text{min}}$ 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 $\chi_D^{AV}$ is considerable in wall temperatures that are either side from adiabatic value $\tau_w = 1$. 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.

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 6.** Potential of improvement with wall temperature for $\Lambda_1 = 0.2$ and $He = 80$.  

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

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

\[
\chi_D^{AV} = \frac{\dot{E}_D^{AV}}{E_D^{AV}}
\]  
(17)
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

\[ \Psi = \frac{Q}{\Delta P} \]  \hspace{1cm} (19)

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

\[ \Psi = \frac{2\theta(1 - e^{-\Lambda\tau})}{\Lambda_1\Lambda_2} \]  \hspace{1cm} (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 \( \Theta \).

**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

\( h \) 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

\[ \chi \] Potential of improvement
\[ \delta \] Curvature ratio
\[ \gamma \] Pitch to coil diameter ratio
\[ \mu \] Dynamic viscosity, kg m\(^{-1}\) s\(^{-1}\)
\[ \nu \] Kinematic viscosity, m\(^2\) / s
\[ \rho \] Density kg / m\(^3\)
\[ \phi \] Volume flow rate m\(^3\) / s
\[ \Psi \] Heat transfer rate to pumping power ratio
\[ \tau \] Temperature ratio
\[ \Theta \] Ratio of dimensionless temperature difference
\[ \theta \] 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.

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