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Thermodynamic investigation and optimization of laminar forced convection in a rotating helical tube heat exchanger

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ABSTRACT

Based on the second law of thermodynamics, an entropy generation investigation is carried out under given dimensionless parameters, i.e. heat exchanger duty, heat flux, with respect to heat transfer and frictional pressure drop in a rotating helical tube heat exchanger with laminar convective flow. The entropy generation from heat transfer across a finite temperature difference – Ψ_h decreases with increasing Dean number which represents the impact of centrifugal force induced secondary flow in enhancing heat transfer. Another aspect of increasing Dean number is that intensified momentum transfer in the radial direction also raises the entropy generation from frictional pressure drop – Ψ_f, the superposed effect of which yields a decreasing–increasing trend of the total entropy generation-Ψ, a local minimum located in between. The rotation of the helical tube in streamwise (co-rotation) or counter streamwise (counter-rotation) direction leads to a decrease in Ψ_h and a increase in Ψ_f which complicates the situation that whether or where the minimum of total entropy generation exists is dependent on whether Ψ is dominated by Ψ_h or Ψ_f or somewhere in between. No difference is discerned between pairs of cases with constant wall temperature and uniform wall heat flux but the same set of variables and parameters. A multi-objective optimization targeting Ψ_h and Ψ_f simultaneously is implemented using the non-dominated sorting genetic algorithm II (NSGA II). Five solution sets are selected and compared with the conventional optimization in regard of Ψ distinguishing the Ψ_h-dominated region from the Ψ_f-dominated region, the dimensionless variable η_1 is found to be the most suitable representative in describing the trade-off between Ψ_h and Ψ_f. The Pareto solution sets is dominated by Ψ_h within the variable and parameter space under discussion. On the Pareto frontier, the counter rotational cases are distributed where the impact of Ψ_f is relatively higher while co-rotational cases dominate almost all the rest part. The proposed investigation procedure is a synthetic analysis concerning optimization of both Ψ and its components Ψ_h and Ψ_f, via which the dominating compartment and the key impact factors for irreversibility minimization can be obtained as a guidance for practical design of rotating helical tube heat exchangers.

1. Introduction

Flow and heat transfer in rotating helical channel has drawn many attentions not only because of the additional momentum and energy transportation by cross-sectional convection due to coexistence of centrifugal force and Coriolis force but also has many industrial applications, especially for heating or cooling in rotating machinery [1,2]. Since heat transfer and pressure drop characteristics, sometimes associated with maximum wall temperature [3–8] and pumping power consumption [9–14] respectively, are two vital factors in evaluating the performance of heat exchangers, previous investigators did a lot of work on the impact of geometry, flow rate, rotation speed, etc. on vortex perpendicular to axial flow and its resultant effect on heat transfer and pressure drop [15–26].

However, from the perspective of thermodynamics the overall efficiency of heat exchanger can be evaluated by the total irreversibility from heat transfer over a finite temperature difference and flow friction. The key point for optimizing heat exchanger then becomes minimization of entropy generation, which was initially proposed by Bejan [27–29]. Makkawi et al. [30] presented a numerical study on a rectangular duct with a constant heat flux at top wall and thermally insulated condition at all other walls. An even distribution of entropy generation between the asymmetric thermal boundary condition was reported. Balkan [31], taking

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Nomenclature

B_0	dimensionless heat exchanger duty
d	diameter of helical tube, m
De	Dean number
F	ratio of Coriolis force to centrifugal force
f	fanning friction factor
H	helical pitch
Nu	Nusselt number
Pr	Prandtl number
Q	dimensionless heat flux
q'	heat flux per unit tube length, W/m
R	curvature radius, m
Re	Reynolds number
Ro	Rossby number
$\dot{m}$	mass flow rate, kg/s

Greek symbols

β	the angle between helix and horizontal plane
δ	curvature ratio
ζ	dimensionless factor when Ro is small, see Eq. (16)
η_1	dimensionless parameter depicting heat transfer and flow friction increase due to combined effect of centrifugal force and Coriolis force
η_2	dimensionless parameter based on η_1, considering the effect of Pr
μ	dynamic viscosity, Pa s
ρ	density, kg/m^3
Ψ	dimensionless entropy generation in total
Ψ_f	dimensionless entropy generation due to flow friction
Ψ_h	dimensionless entropy generation due to heat transfer over a finite temperature difference
Ω	angular rotation speed, rad/s

Subscripts

0	stationary straight pipe flow
b	baseline case
q	uniform wall heat flux
T	constant wall temperature

the variation of overall heat transfer coefficient due to changes of flow rates into consideration, improved the effectiveness of equipartition of entropy production (EoEP) [32] method. Many other studies on thermodynamic analysis of conventional types of heat exchangers, e.g. shell-and-tube, tube-in-tube, etc., can be found in literature [33–40]. On the other hand, Hajmohammadi et al. [41,42] reported an opposite developing trend of the optimized heat flux distribution numerically obtained for minimizing the peak wall temperature compared with the one from entropy generation minimization method. The author [43] also proposed a new configuration of bending tubes comprised of three straight segments and two orthogonally curved bends on a basis of numerically obtained entropy generation and pressure drop compared by the cases with semi-circularly-curved bend. Satapathy [44] derived the optimal dimensionless flow rate and curvature ratio for laminar and turbulent flow respectively. With constant Prandtl number and heat exchanger duty parameter, the optimal curvature ratio is found to increase with increasing Dean number for laminar flow under constant wall heat flux condition. Shokouhmand and Salimpour [45] compared the effect of Reynolds number, curvature ratio and coil pitch, etc. on the entropy generation rate for air and water with uniform wall temperature. It has been concluded that the total entropy generation rate is more sensitive to Reynolds number and curvature ratio while less sensitive to helical pitch for low pitch values. The optimal Reynolds number decreases as curvature ratio increases in laminar flow region. The impact of other dimensionless parameters in the formulae of total entropy generation rate is further discussed with a correlation predicting optimal Reynolds number proposed [46]. Ko and Ting [47] derived the total entropy generation rate in a different way. Again the helical pitch is found to have minor effect on entropy generation rate. The increase of curvature ratio raises the fluid friction irreversibility but attenuates the entropy generation due to heat transfer. The optimal curvature ratio increases to the upper limit as Reynolds number is below 5000 while for Reynolds number greater than 5000 smaller curvature ratio should be selected. The ratio of entropy generation by fluid friction to that by heat transfer increases with Reynolds number. When Reynolds number goes up to 6000 and 7000, the total entropy generation rate is dominated by frictional irreversibility [48]. Under constant wall heat flux, the numerical simulation indicates that the primary entropy generation are located at the next-to-wall region where both velocity and temperature gradients are of greatest magnitude. The optimal Reynolds number increases with the wall heat flux while the relationship between entropy generation and wall heat flux is dependent on Reynolds number [49]. Ko [50] further explored the curved pipe with rectangular cross-section. The effects of Dean number, wall heat flux and rectangular aspect ratio are discussed in detail. Frictional irreversibility is found to be favored by Dean number and increasing wall heat flux will bring increment of heat transfer irreversibility. The optimal aspect ratio

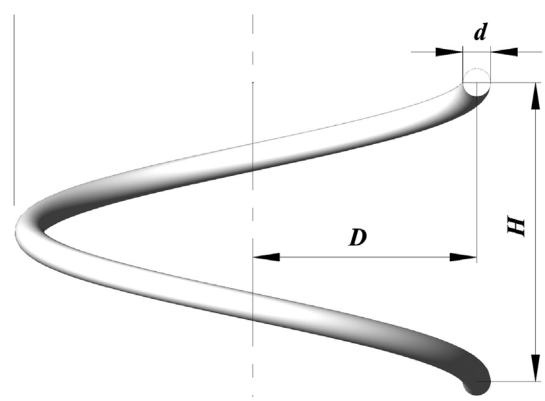

Fig. 1. Schematic representation of the helical pipe.

Table 1
Boundary conditions for dimensionless variables.

Variables	Lower boundary	Upper boundary
Q	1.05	31.4
B_0	9.84E+08	9.84E+09
Re	100	10,000
δ	5.0E−04	0.0625
Ro	8	100
Pr	0.7	1000