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# Enhancement of laminar convective heat transfer relying on excitation of transverse secondary swirl flow

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ABSTRACT

Laminar forced convective heat transfer is studied for the purpose of getting the best heat transfer performance with the least flow resistance increase. The variation calculus method is employed to establish the equations describing the optimized fluid velocity field and temperature field. Numerical solutions of the equations for a convective heat transfer process in a section-cut of a square duct indicate the optimized flow should have a transverse secondary swirl flow pattern consisting of multiple vortexes with identical swirl direction in the junction region of any two neighboring vortexes. We then propose the convective heat transfer enhancement method relying on excitation of transverse secondary swirl flow. To validate this method, we numerically study the heat transfer and flow resistance characteristics of laminar flows in tubes with four-reverse-vortex-generator (FRVG) inserts, four-homodromous-vortex-generator (FHVG) inserts, or a twisted tape insert. The calculated transverse secondary flow in the tube with the FRVG inserts approximately follows the optimized flow pattern and the tube is thus found to have the best thermo-hydraulic performance, validating the proposed convective heat transfer enhancement method.



## 1. Introduction

Various techniques of convective heat transfer enhancement are widely applied in many industrial fields for efficient energy generation, conversion, and utilization. During the past several decades, many researchers and engineers have expended great efforts to develop advanced convective heat transfer enhancement techniques [1,2] like heat transfer surface modifications, direct flow stirrers and indirect flow stirrers by external electric or magnetic fields, to name a few. Fundamentally, these techniques strengthen the convection/advection of fluid flow and reduce the thickness of thermal boundary layer. For instance, the vortex generators destroy the boundary layer by generating longitudinal vortex near the tube-wall region [3,4], and the twisted tape inserts arouse overall swirl flows and enhance the fluid advection between the near-wall and central regions [5–7]. Therefore, these techniques or measures augment heat transfer rate normally at the cost of increasing the flow resistance, i.e., consuming more external pump work. In other words, there exists an intrinsic contradiction between convective heat transfer enhancement and flow resistance reduction. In order to comprehensively evaluate the overall performance of various heat transfer enhancement techniques, Webb [8] proposed a performance evaluation criterion (PEC), in terms of which a larger heat transfer rate does not necessarily mean a better overall performance as enough attention must be paid to the flow resistance increase after the implementation of the technique. Designing convective heat transfer enhancement techniques with maximized heat transfer enhancement effect at the cost of minimized external pump work consumption is persistently a hot research subject in the thermal science and engineering and relevant fields.

Bejan [9,10] proposed the principle of minimum entropy production for heat transfer optimization based on the idea that the entropy production in an optimized heat transfer process should be minimized. He derived the expression of entropy generation induced by both the heat transfer and viscous fluid flow and analyzed the optimum geometry of heat transfer enhancement devices at the constraint of least total entropy generation. Many researchers [11–17] analyzed the entropy generations in various

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convective heat transfer processes and found optimum geometrical parameters of relevant devices based on the principle of entropy production minimization. Relatively recently, some other objective functions for heat transfer optimization have been proposed, such as the entransy dissipation extreme by Guo et al. [18] and the power consumption minimization by Liu et al. [19,20]. Guo et al. [18] proposed a new concept, "entransy", and regarded it as a physical quantity describing the heat transfer ability and the potential capacity of a usable heat source. The fundamentals of their entransy theory are i) there is some inevitable entransy dissipation during heat transfer process, and ii) the entransy dissipation should be maximized or minimized for the optimization of a heat transfer process. Meng et al. [21] and Chen et al. [22] analyzed the entransy dissipation in some typical convective heat transfer processes and derived the expressions for process optimization. In contrast, Liu et al. [19,20] set the minimum power consumption as the optimization objective constrained by constant entransy dissipation and derived the modified expressions for the optimization of convective heat transfer in circular tube flows. In these works [18–22], the calculus of variations [23–25] was employed for the theoretical derivations.

Despite the aforementioned principles for heat transfer optimization, theories for the design of advanced convective heat transfer enhancement techniques are obviously still needed. This work is aimed to explore the mechanisms of convective heat transfer enhancement for laminar flows and attempts to develop a generic approach that can maximize the convective heat flux at the constraint of constant pump work consumption. We will first identify the objective function of heat transfer enhancement and the constraint function of pump work consumption. We then derive the equations that describe the optimized convective heat transfer in laminar flows based on the principle of the calculus of variations and apply these equations to a special case, convective heat transfer in a 2D section-cut of a square duct. Last, based on the findings from the results of 2D calculations, we propose a novel convective heat transfer enhancement method relying on excitation of transverse secondary swirl flow and numerically examine its effectiveness with respect to 3D tube flows.

## 2. Derivation of optimization equations

### 2.1. Heat transfer enhancement

To facilitate the derivation of equations, we consider a one-dimensional (1D) steady-state heat transfer process, schematic of which is depicted in Fig. 1. The left domain A is solid and the right domain B is fluid. The symbols $n$ and $T$ denote the 1D spatial position and the temperature variable, respectively. A constant temperature $T_w$ is prescribed at the left boundary of domain A; $T_w$ is higher than the temperature in the other regions. The heat is transported from the left boundary of domain A, across domain A by conduction, and into the fluid by convection. Besides maneuvering thermal properties of the solid or fluid, one more important way to enhance this heat transfer process is optimizing the velocity and temperature fields in the fluid domain B to enhance the heat convection.

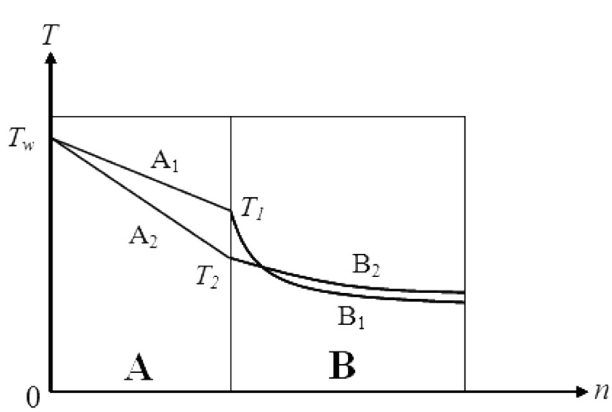


**Fig. 1.** Schematic of a one-dimensional heat transfer process.

The two temperature profiles (labeled with profile $B_1$ and $B_2$, respectively) are different due to the distinct velocity fields in domain B. Profile $B_1$ exhibits larger temperature gradients in the boundary layer, meaning that the heat transfer from the right boundary of domain A to the fluid is more inefficient. In contrast, profile $B_2$ is flatter, indicating more efficient heat transport. The temperature at the interface between the solid and fluid domains for profile $B_2$ is lower than that for profile $B_1$, i.e., $T_2 < T_1$. Since there is only heat conduction in the solid domain A, the temperature profiles $A_1$ and $A_2$, corresponding to $B_1$ and $B_2$ in the fluid domain, respectively, both show linear dependence on position $n$. Moreover, the temperature profile $A_2$ has larger gradients than profile $A_1$, indicating larger heat flux for scenario $A_2$. Therefore, the fluid flow in scenario $A_2$–$B_2$ has better effect of heat transfer enhancement than that in scenario $A_1$–$B_1$.

We turn the above discussion into the following mathematical expression, as

$$\left( \int_n |\partial T/\partial n| \mathrm{d}n \right)_2 < \left( \int_n |\partial T/\partial n| \mathrm{d}n \right)_1 \tag{1}$$

where the subscripts 1 and 2 denote profile $B_1$ and $B_2$, respectively, and the integrals over $n$ are only on the fluid domain. The integral $\int_n |\partial T/\partial n| \mathrm{d}n$ quantifies the convective heat transfer enhancement effect for a fluid flow, and the smaller the value is the greater the effect can be. For two- or three-dimensional heat transfer processes, a generic expression is introduced as

$$J = \int_\Omega |\nabla T| \mathrm{d}\Omega \tag{2}$$

where, $\Omega$ represents the whole fluid domain. The smaller the $J$ is the greater is the effect of convective heat transfer enhancement. Therefore, to enhance convective heat transfer, an effective method is to reduce the absolute value of temperature gradient in the fluid domain by manipulating the velocity distribution in the fluid.

Of course, all heat transfer processes must follow the energy conservation law, which is described by the following Eq. (3) if transient effects and energy sources/sinks are not considered.

$$k\nabla^2 T - \rho c_p \mathbf{U} \cdot \nabla T = 0 \tag{3}$$

### 2.2. External pump work consumption

External pump work is needed for forced convective heat transfer. When the convective heat transfer is enhanced by manipulating fluid velocity field, the external pump work consumption is inevitably increased for real viscous flow. The momentum equation of fluid flow describes the relationship between the forces and the fluid motions. We may identify the pump work consumption of viscous flow by comparing the momentum equations for viscous and inviscid flows.

For steady-state incompressible laminar flows, the momentum equations for inviscid and viscous flows, respectively, are