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# Numerical simulation of modulated heat transfer tube in laminar flow regime

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A B S T R A C T

The flow field modulation concept was proposed. A mesh cylinder was suspended in a tube, dividing the tube cross section into an annular region and a core region. The invention is called the modulated heat transfer tube (MHTT). The numerical simulation was performed in laminar flow regime at constant heat flux boundary condition with water as the working fluid. An equal flow area criterion was proposed for the conversion of 3D to 2D mesh pores. The non-uniform grids link micron scale of mesh pores with meter scale of tube. The results show double-peak velocity distribution over tube cross section. The near wall region has larger velocity and velocity gradient, accounting for the heat transfer enhancement mechanism. For any specific Reynolds number, there is a critical length beyond which heat transfer is deteriorated. Therefore, a set of short mesh cylinders was suspended in the tube. The configuration is called the improved modulated heat transfer tube (IMHTT). It is shown that the IMHTT ensures significant heat transfer enhancement over the whole tube length. This study provides a new enhancement mechanism and tube configuration for heat transfer, having a wide engineering application potential.


## 1. Introduction

Heat transfer tubes are widely used in many energy conversion and power generation systems. Heat transfer in tubes, with various geometry shapes, dimensions and material has been investigated by more than one century. According to the thermodynamic law, the exergy efficiency is increased by decreasing temperature differences across the two sides of the tube. This requires significantly higher heat transfer coefficients with acceptable pumping power.

The worldwide energy shortage and environment problems demand the energy utilization efficiency to be increased for fossil energy systems. Meanwhile, renewable energy such as solar energy, ocean energy etc. has been put into use. Miniaturization of heat exchangers increases system efficiencies and decreases investment cost. Under many situations, one needs to extract low grade energy from extremely low temperature resource. An example was cited from Ref. [1]. A heat exchanger for an ocean thermal energy conversion (OTEC) plant requires a heat transfer surface area in the order of $10^4$ $m^2$/MW.

The classical fluid mechanics tells us that the non-slip boundary condition on wall is acceptable to predict flow and heat transfer behavior in tubes. This implies a stationary thin fluid layer on the wall. Besides, velocity and velocity gradient near the wall are limited, no matter for laminar or turbulent flows. Such flow behavior directly limits the temperature gradient near the wall to determine the heat transfer coefficient. Besides, after a long time operation of heat exchangers, the thermal resistance is increased due to fouling. This problem becomes more serious when a system is operating at dirty environment or an impurity working fluid flows in tubes. Conventionally, heat transfer performance can be improved by introducing a disturbance in fluid flow (breaking the viscous and thermal boundary layers).

Various techniques have been proposed for heat transfer enhancement in the literature. These can be found in consecutive review articles such as Dewan et al. [1] and Liu and Sakr [2], etc. Heat transfer tubes with many kinds of inserts such as twisted tape, wire coil, swirl flow generator, etc. enhance the thermal efficiency. The twisted tape inserts are popular researched and used to strengthen the heat transfer efficiency [3–8]. Twisted tape inserts perform better in laminar flow regime than in turbulent flow regime. But other several passive techniques such as ribs, conical

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**Nomenclature**

| | |
|---|---|
| $A$ | flow area of the square mesh pores, $m^2$ |
| $A'$ | flow area of the stripe-type mesh pores, $m^2$ |
| BT | bare tube |
| $C_p$ | specific heat, J/kg K |
| $D$ | bare tube diameter, m |
| $D_m$ | mesh cylinder diameter, m |
| $D_o$ | outer tube diameter, m |
| $g$ | gravity acceleration ($g = 9.81$), $m/s^2$ |
| IMHTT | improved modulated heat transfer tube |
| $L$ | bare tube length, m |
| $L_b$ | bare tube section length for the modulated heat transfer tube, m |
| $L_{b'}$ | distance between two neighboring mesh cylinder, m |
| $L_m$ | modulated flow section length for the modulated heat transfer tube, m |
| $L_{m'}$ | a single short mesh cylinder length for the improved modulated heat transfer tube, m |
| $L_t$ | the transition flow length, m |
| MHTT | modulated heat transfer tube |
| $Nu_{ave}$ | average Nusselt number |
| $Nu_x$ | local Nusselt number |
| $p$ | pressure, pa |
| $q$ | heat flux, $W/m^2$ |
| $q_w$ | the heat flux added to the wall, $W/m^2$ |
| $r$ | radial coordinate, m |
| $Re$ | Reynolds number |
| $T_f$ | the fluid temperature, K |
| $T_{f,ave}$ | the average fluid temperature, K |
| $T_{grid}$ | the temperature within the first fluid layer, K |
| $T_{in}$ | the inlet temperature, K |
| $T_{out}$ | the outlet temperature, K |
| $T_w$ | the wall temperature, K |
| $T_{w,ave}$ | the average wall temperature, K |
| $T_{wo}$ | the outer wall temperature, K |
| $u$ | the axial velocity, m/s |
| $u_{ave}$ | the average velocity, m/s |
| $u_w$ | the velocity on the wall, m/s |
| $v$ | the radial velocity, m/s |
| $w$ | square mesh pore width, m |
| $w'$ | stripe-type mesh pore width, m |
| $x$ | axial coordinate, m |
| $y$ | first fluid layer thickness, m |

*Greek symbols*

| | |
|---|---|
| $\delta$ | square mesh wire thickness, m |
| $\delta'$ | stripe-type mesh wire thickness, m |
| $\lambda$ | fluid thermal conductivity, W/m K |
| $\lambda_w$ | tube wall thermal conductivity, W/m K |
| $\mu$ | dynamic viscosity, Pa s |
| $\rho$ | density, $kg/m^3$ |
| $\phi$ | enhanced heat transfer factor |

*Subscript*

| | |
|---|---|
| ave | average |
| b | bare tube |
| f | fluid |
| in | inlet |
| m | modulated tube |
| o | outer |
| out | outlet |
| w | tube wall |
| wo | outer tube wall |
| x | axial position |

nozzle, and conical ring, etc. are generally more efficient in turbulent flow regime than in laminar flow regime [9–12].

The primary objective of this paper is to propose a new heat transfer tube. A mesh cylinder was suspended in the tube, dividing the tube into an annular region and a core region. The second objective of the paper is to verify the fresh idea by the numerical simulation together with the experimental data. The numerical results do show the larger velocity and velocity gradient near the wall. The numerical and experimental works are being continued. The future optimized heat transfer tube is expected to not only have significantly high heat transfer coefficients, but also delay the fouling, having wide engineering applications.

## 2. The modulated heat transfer tube

Fig. 1(a) shows the modulated heat transfer tube (MHTT). A mesh cylinder is suspended in a tube, dividing the tube cross section into an annular region and a core region. Supporting structures may be needed for a practical application. The mesh cylinder consists of a flat bottom mesh surface and a side circular mesh surface. Inside of the mesh cylinder is empty. When a fluid stream strikes the mesh cylinder, part of the flow rate enters the core region (mesh cylinder inside), but most of the flow rate flows in the annular region. The configuration creates larger velocity and velocity gradient near the wall in the annular region, which helps to enhance heat transfer. With flow evolution in the tube downstream, flow from the annular region to the core region is generated to decrease the velocity and velocity gradient in the annular region. The flow and heat transfer behavior in MHTT is thoroughly different from that in a concentric tube annular. The later configuration does not include mesh pores.

## 3. Numerical simulation and method

### 3.1. The 3D to 2D conversion of mesh pores

Numerical simulation of laminar flow and heat transfer in MHTT was performed. Initially, the 3D to 2D conversion of mesh cylinder was performed. MHTT is a multiscale system with three-levels of sizes: (1) micron scale of mesh pores; (2) miniature scale of the annular region; and (3) macroscale of the tube diameter and length (meter scale). Fig. 1(b) shows mesh cylinder with square mesh pores. The mesh cylinder diameter and length are recorded as $D_m$ and $L_m$ respectively. Mesh pores have a pore width of $w$ and mesh wire thickness of $\delta$. Assuming $D_m = 10.0$ mm, $L_m = 1000.0$ mm, $w = 165$ μm and $\delta = 85$ μm, the number of mesh pores attains 502,655 (million magnitude). Numerical simulation of such a huge number of mesh pores in micron scale is impossible at this stage. Thus a two-dimensional equivalent system should be developed and an equivalent criterion for mesh pores is necessary. Fig. 1(b) can be simplified to a 2D system by converting square mesh pores into stripe-type mesh pores (Fig. 1(c)), with a metal wire thickness of $\delta'$ and stripe-type mesh pore size of $w'$. At the mesh cylinder diameter of $D_m$ and length of $L_m$, the total flow area of square mesh pores is

$$A = \frac{\pi D_m}{w+\delta} \times \frac{L_m}{w+\delta} \times w^2 \quad (1)$$