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Technical Note

Heat transfer enhancement with impinging free surface liquid jets flowing over heated wall coated by a ferrofluid

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1. Introdution

Magnetic fluid is used widely as a new working fluid that is made from magnet nano-particles dispersed and suspended uniformly in base liquid. The development of ferrofluid is perceived to open new technological frontiers. The basic and applied study of magnetofluid has long ago attracted scientists and engineers worldwide [1]. More than 5000 scientific papers have been published and more than 2000 patents have been issued until 1993 [2]. However, study of the influence of ferrofluid on heat transfer was very scarce up till now. Natural convection of magnetofluid was studied by Kikura et al. [3]. Takahashi et al. [4] investigated the nucleate pool boiling heat transfer of magnetofluid. As summarized in Ref. [5], the drag reduction appeared simultaneously with heat transfer enhancement in forced convection. It was found experimentally by Bashtovoi et al. [6] that heat transfer enhancement and drag reduction occurred simultaneously in the case of oil flowing over a rectangular channel with a wall coated by ferrofluid, the heat transfer increased to about 90% while drag coefficient reduced to about 20% in comparison with those without coating. It may be the sole report concerning the heat transfer enhancement and drag reduction with magnetofluid. The present experimental results give reports on heat transfer in free surface liquid jet impingement with magnetofluid coating on the heated surface. To our best knowledge, it is not found in any open literature that magnetofluid is used in heat transfer of jet impingement.

2. Experiments

Transformer oil was used as the working liquid and a kerosene-based ferrofluid C1-20B was coated on the heated surface. The free surface slot jets of transformer oil were adopted in this study. The experimental apparatus and method in this work are similar to those described in Refs. [7–9]. Only a brief description will be given here.

The test liquid transformer oil is circulated in a closed loop which has provision for filtering, metering, preheating and cooling. The test section assembly, described in Fig. 1, is vertically fixed on one side of the chamber made of stainless steel, a permanent magnet which is 25 mm in diameter and 40 mm in length is fixed on the backside of the test section. The magnet provides strong magnetic field which keeps ferrofluid film coated on the heat transfer surface. The main part of the test section is a strip of 10 μm thick constantan foil with a heated section of 13×5 mm^2 exposed to the liquid jets. This active section of the foil is used as an electrical heater as well as a heat transfer surface. The temperature on the center of the inner surface of

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Nomenclature

A	area of heated surface	T_{aw}	adiabatic wall temperature
B	slot width of nozzle	T_f	jet static temperature at nozzle exit
h	local heat transfer coefficient	T_w	local wall temperature
I	current intensity	u	mean fluid velocity at nozzle exit
K	heat transfer enhancement coefficient	x	lateral distance from stagnation point
q	heat flux	Z	nozzle-to-plate spacing
R	electrical resistance		

the heater is measured by a 40 gage iron–constantan thermocouple which is electrically insulated from the foil yet in close thermal contact. The heat transfer surface is sustained at constant heat flux conditions. The surface heat flux q is calculated from the electrical power supplied to the heater as:

$$q = I^2R/A \tag{1}$$

where electrical current I and electrical resistance R are measured with electric meter, A is the heated surface area.

The slot nozzle employed in the present study is made of plexiglass. Fig. 2 shows the configuration of the jet nozzle. The rectangular ducts have a streamwise length of 35 mm and are 0.25 mm in width, and 12 mm in height. The large aspect ratio between height and width eliminates the end effect and leads to a situation of truly two-dimensional jets. The shape and size of the nozzle were precisely measured with a tool maker's microscope of 0.001 mm resolution.

The local heat transfer coefficient for jet impingement is defined by

$$h = q/(T_w - T_{aw}) \tag{2}$$

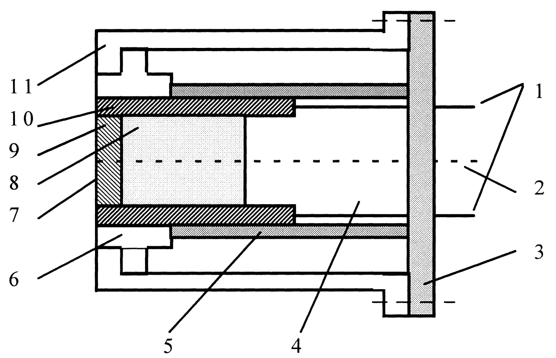


Fig. 1. Detailed structure of test section: 1 — electrical pole; 2 — thermocouple; 3 — plastic cover; 4 — fiberglass; 5 — presure tube; 6 — plexiglass; 7 — constantan foil; 8 — magnet; 9 — bakelite; 10 — copper block; 11 — assembly wall.

where T_w is wall temperature and T_{aw} is adiabatic wall temperature according to the Refs. [7–11]. However, in the range of experimental parameters encountered, the difference between T_{aw} and T_f can be neglected. The wall temperature, T_w, and liquid jet temperature at the exit of nozzle, T_f, are all measured by iron–constanstan thermocouples. The general research results on heat transfer of transformer oil jet impingement have been reported in Refs. [7–10]. It is quoted here to compare the experimental results on heat transfer of liquid jet impingement with magnetofluid coating to that without magnetofluids.

3. Results and discussions

The liquid jet temperature, T_f, and heat flux, q, are kept nearly constant in the experiment. The wall temperature distributions along heat transfer surface for different jet velocities with and without coated magnetic fluid, in Fig. 3, show its minimum at stagnation point and increase smoothly with the lateral distance. The wall temperature has an obvious change since one point in the wall jet zone has magnetic fluid coating with a mean thickness of less than 0.5 mm. The magnetofluid coating would be unstable at stagnation zone and is pushed away with strong impinging of the

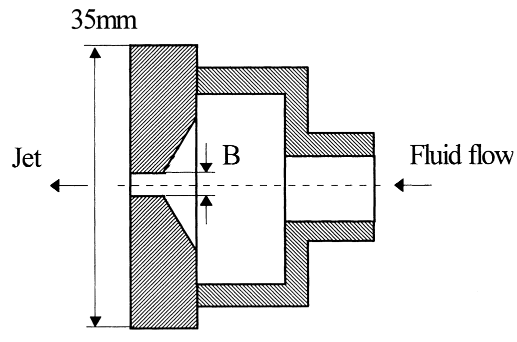


Fig. 2. Jet nozzle assembly.

working liquid. We can decide thereby the presence of the magnetic fluid layer from the wall temperature changing point.

The local heat transfer coefficient distributions plotted in Fig. 4(a) and (b) demonstrate clearly the effect of the heat transfer enhancement with magnetic fluid. Larger the jet velocity, more effective would be the heat transfer. It makes clear that the presence of the magnetofluid intensifies the jet impingement heat transfer to higher mean coefficients, especially in the wall jet area. It is well known that the boundary layer of flow near wall is the main thermal resistance to the heat transfer between the wall and the flowing fluid. The viscosity of kerosene-based magnetofluid coating, instead of the flow boundary layer of transformer oil, is much less than that of transformer oil. This is consistent with the conclusion in Ref. [6] that the effect essentially depends on the fluid viscosity ratio in the main flow and in the liquid coating, the smaller the coating viscosity relative to the main flow more noticeable would be the heat transfer enhancement. Such a viscosity ratio reached 16.8, with water-based magnetofluid at 25°C in Ref. [6], which in present experiments is 10.6 smaller than that in Ref. [6].

To express the effect of the heat transfer enhancement, a coefficient K is defined as: $K = (h - h_0)/h_0$, h and h_0 being the heat transfer, coefficient for coated and non-coated jet impingement heat transfer, respectively. The local heat transfer enhancement ratios with magnetic fluid coating are plotted in Fig. 5. With the distance away from the stagnation point, the heat transfer enhancement ratio increases continuously, and it is also proportional to jet velocity. The maximum value of K reaches 32% at jet velocity $u = 8.5$ m/s in the ranges of the experiments. The trend is similar to that of Bashtovoi et al. [6]. Heat transfer is improved due to the presence of magnetic fluid, which can be qualitatively explained by the novel concept for convective heat transfer enhancement [12]. When the oil flows over the magnetic fluid, the interface shows wave-like behavior. This implies that both the axial and radial components of oil velocity near the interface are enlarged, and consequently, the thermal resistance

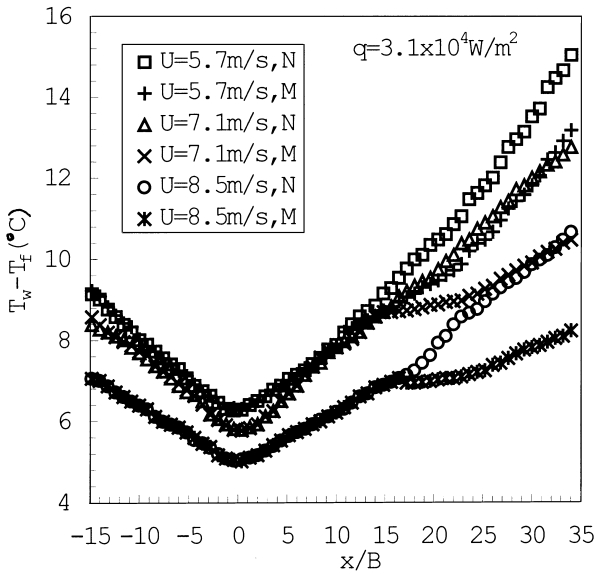
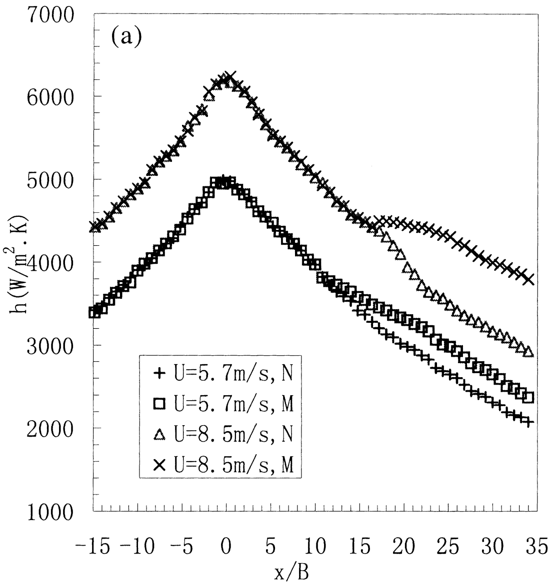


Fig. 3. Lateral distributions of local wall temperature of heat transfer surface M — with ferrofluid film, N — without ferrofluid film.



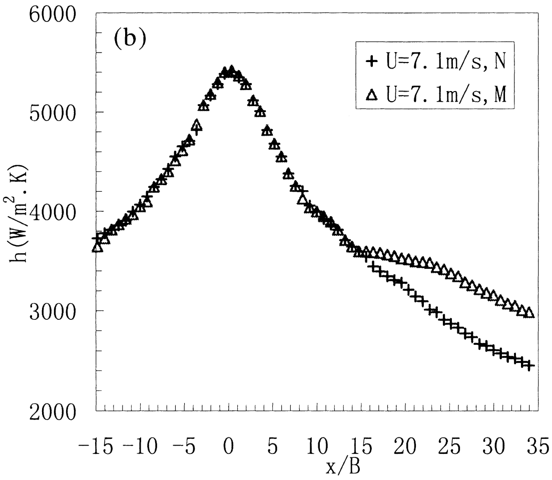


Fig. 4. Lateral distributions of local heat transfer coefficient (a) $u = 5.7$ m/s, $u = 8.5$ m/s; (b) $u = 7.1$ m/s.

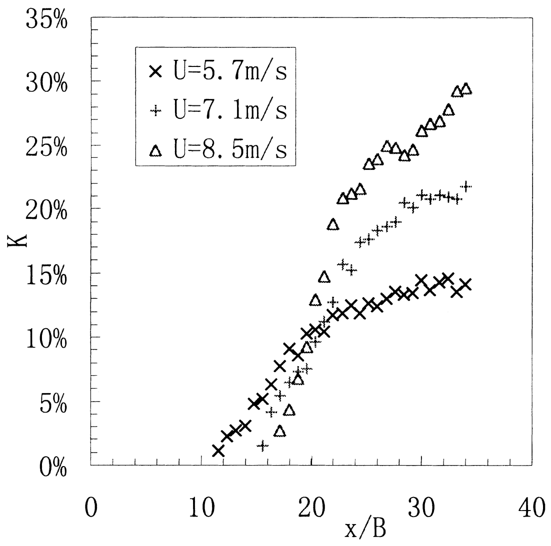


Fig. 5. Lateral distributions of local heat transfer enhancement coefficient.

of oil is reduced, and the thermal resistance of magnetic fluid is comparatively very small due to the existence of vortexes in pairs. The higher the jet velocity, the larger the interfacial wave amplitude, and hence, stronger the effect of heat transfer enhancement.

To conclude, with the magnetic fluid coating, the heat transfer enhancement for jet impingement is clearly observed through the present experiment. The enhancement ratio gets up to 32% in the present study and is influenced by the jet velocity. The result can be explained by the existence of vortexes inside ferrofluid.

Acknowledgements

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