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EXPERIMENTAL STUDIES ON FRICTION FACTOR AND HEAT TRANSFER CHARACTERISTICS THROUGH WIRE-WOVEN BULK KAGOME STRUCTURE

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Periodic cellular metals with open, periodic cell topologies have received much attention owing to their potential for multi-functionality such as load bearing, thermal dissipation, and actuation. Recently, a new technique, known as wire-woven bulk Kagome, has been introduced, which is used for fabricating multi-layered Kagome with truss periodic cellular metal. The fabrication of the wire-woven bulk Kagome is based on a concept where continuous helical wires are systematically assembled in six directions. Besides its excellent load-bearing capability with light weight, the wire-woven bulk Kagome has potential for a heat dissipation media because of the high ratio of surface area to volume and low flow resistance. This article presents the experimental results of the fluid flow and heat transfer characteristics of the multi-layered wire-woven bulk Kagome composed of aluminum 1100 helix wires. Under forced-air convection conditions, the friction factor and heat transfer rate of the wire-woven bulk Kagome specimen were investigated for two specimen orientations. The results showed that the friction factor of the wire-woven bulk Kagome was mainly affected by the flow blockage area, and the heat transfer characteristics depended on the open-area ratio. In addition, the results were compared with other heat dissipation media (e.g., open-celled foams, woven screens, lattice-frame materials, and cast Kagome structures). The results showed that the heat transfer performance of the multi-layered aluminum wire-woven bulk Kagome competed favorably with the best heat dissipation media currently available.

Keywords forced convection, open area ratio, periodic cellular metal, porosity, surface area density, thermal efficiency

INTRODUCTION

Cellular metals are well known for their high mechanical strength and stiffness with low weight advantage. They are increasingly being used for multi-functional applications. Over the past few years, several approaches for mechanical and thermal properties and for optimal design of cellular metals have been performed. However, commercially available closed- and open-celled aluminum foams were the first cellular metal whose engineering applications have been limited because of its high cost and relatively low mechanical

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NOMENCLATURE

A	heating area of copper plate (m^2)
c_p	specific heat (J kg^{-1} K^{-1})
f	friction factor
h	heat transfer coefficient (W m^{-2} K^{-1})
H	total channel height (m)
k_f	thermal conductivity (W m^{-1} K^{-1})
$\dot{m}$	mass flow rate (kg s^{-1})
Nu_H	Nusselt number based on channel height
q	heat flux (W m^{-2})
Q	rate of heat transfer ($= \dot{m}c_p(T_{m,out} - T_{m,in})$) (W)
Re_H	Reynolds number based on channel height
$T_{f,ave}$	average bulk mean temperature of air at inlet and outlet (K)
$T_{m,out}$	bulk mean temperature of air at outlet (K)
$T_{m,in}$	bulk mean temperature of air at inlet (K)
T_w	average temperature of copper plate (K)
U_c	mid-height velocity (m s^{-1})
U_m	mean velocity (m s^{-1})
y	height measured from the bottom along the centerline (m)
α_A	surface area density (m^{-1})
Δp	static pressure drop (Pa)
ε	porosity
ρ_f	density (kg m^{-3})
ρ_{rel}	relative density

Subscripts

A	area
c	mid-height
f	fluid
H	height
in	inlet
m	mean
out	outlet
w	wall

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strength due to the random patterns of its cell structure. On the other hand, a periodic cellular metal (PCM) significantly enhances mechanical performances at a given relative density. Specifically, the truss PCMs have been developed because of several advantages, namely, it light weight, excellent specific strength and stiffness, and potential for multi-functional applications [1]. The truss PCMs have been reported to be promising in many applications including heat dissipation [2, 3], fluid storage [4], and impact energy absorption [1, 4]. Because they have a large surface area density, these structures are expected to be effective in dissipating heat with a minimum volume. Several types of truss PCMs have been introduced, examples of which include the pyramid [5, 6], the octet [7, 8], and the Kagome [9]. For the Kagome truss PCM, Hyun et al. [9] showed that it has low anisotropy and stable plastic behavior under compression and shear loading compared to other PCMs.

Recent fabrication developments for multi-layered truss PCMs are based on the node-to-node stacking of single-layered structures and bonding [3, 10]. A new technique, called wire-woven bulk Kagome (WBK), for fabricating multi-layered Kagome truss PCMs has been introduced [11]. During the fabricating of the WBK, continuous helical wires were systematically *assembled* in six different directions. Through this fabrication method, a uniform structure of a WBK can be obtained, which guarantees a high specific strength. It is expected that this PCM can be mass-produced because the fabrication is carried out by continuous assembling of helically formed wires. Lee et al. [12] documented the detailed assembly process for this technique.

For thermal management applications, performances of a heat dissipation medium are evaluated by three key characteristics, i.e., high conductivity ligaments, low-pressure drop, and turbulent fluid flow (high heat transfer rate). Especially, the turbulent fluid flow and low pressure drop are closely affected by the cell geometries (e.g., porosity,

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relative density, open-area ratio, surface area density, etc.) of the heat dissipation medium. Namely, several mechanisms contributing to heat transfer have been identified, including low thermal resistance conduction paths from a thermal source through the medium, high surface area contact between the medium and a fluid coolant, and high heat transfer between the surface of the medium and the fluid coolant. Consequently, the optimum heat exchanger media balance the heat conduction through interconnected ligaments (maximized by increasing the relative density) against the need to create easy flow paths (maximized by minimizing relative density).

By using analytical models, CFD (computational fluid dynamics) simulations, and experimental measurements, the heat transfer performance of a variety of the PCMs, including woven-screen cores [3], metal honeycombs [13], open-celled metal foams [14, 15], and lattice-frame materials (LFMs) [16–18], have been evaluated. A geometrical effective thermal conductivity model of a saturated porous metal foam was developed by Boomsma and Poulikakos [14], based on the idealized three-dimensional basic cell geometry of a foam, the tetrakaidecahedron. Detailed expressions for foam effective thermal conductivity were derived. The results showed that changing the fluid conductivity (in vacuum, air, and water) has a relatively small effect on increasing the effective thermal conductivity. Under forced water convection, Boomsma et al. [15] also performed experimental investigations on the hydraulic characterization and the heat transfer performance in open-cell aluminum foams (6106-T6 aluminum alloy, two different porosities of $\varepsilon = 0.92$ and 0.95). The results showed that the metal foam heat exchangers decreased the thermal resistance by nearly half when compared to currently used heat exchangers designed for the same application.

Under steady-state forced air convection, Tian et al. [10] experimentally characterized the fluid flow and heat transfer of the PCMs with plane weave copper meshes along different orientations. The result was compared with that of metallic forms, micro-truss materials, packed beds, louvered fins, and other heat dissipation media. Thermally, the copper meshes are as good as the stochastic metal foams, both having large surface area densities. However, the pumping power required is significantly lower for the copper meshes due to the periodic and uniform patterns of its cell structure. Lu [13] conducted theoretical investigations with micro-cell aluminum honeycombs that were subjected to convective cooling. Simple analytical solutions were derived from thermal fields and for the overall heat transfer coefficient as a function of cell morphological parameters and heat transfer conditions. The results were analyzed in order to find the optimal morphology for the maximum overall heat transfer efficiency. Kim et al. [17, 18] investigated the heat transfer and pressure drop for a compact heat sink made of fully triangulated, lightweight (porosity $\sim$ 0.938), LFMs made of LM25 aluminum casting alloy. The specimens were produced by a process of rapid prototyping and subsequent investment casting as described in [7]. The results were compared with those predicted by an analytical model based on fin analogy. The experimental results revealed that the pressure drop was strongly dependent on the orientation of the structure, primarily because of the flow blockage effect.

Although a number of works have been carried out on mechanical performances of a Kagome truss PCMs [9, 12, 19]. Performance of the Kagome truss PCM as a heat dissipation medium remains to be explored. In this work, the performances of the WBK as a multi-layered Kagome PCM are investigated. Thus, the fluid flow and heat transfer characteristics of the multi-layered aluminum WBK structures were experimentally inves-

tigated. Under forced-air convection conditions, the friction factor and heat transfer rate were measured for two specimen orientations with the same relative density (porosity). In addition, the performance of the multi-layered aluminum WBK were compared with that of other heat dissipation media including copper wire screens, metallic foams, aluminum LFMs, corrugated ducts, and louvered fins.

EXPERIMENTS

Specimen

A specimen of a WBK was fabricated as follows. First, wires of aluminum 1100-O (http://www.matweb.com) with a diameter of 1 mm were formed into a helical shape with a pitch of 14.7 mm and helical radius of 0.6 mm. The helical wires were assembled in six directions to create five-layered WBK specimens. Figure 1 illustrates the finished specimen of a five-layered aluminum WBK. The dimensions are 150 mm in width, 150 mm in length, and 60 mm in height. Figure 2 illustrates two different orientations of side views. Orientation B is rotated 90° from Orientation A. The chemical composition and the thermal properties of the aluminum 1100-O are summarized in Tables 1 and 2.

Relative Density and Porosity

First, direct measurements of the relative density (ρ_{rel}, which is defined as the ratio of the volume occupied by the solid metal to the entire external volume) were performed. Namely, the weight and volume of the specimen were measured to obtain the relative density. Also, if the WBK is idealized as a classical truss structure composed of straight

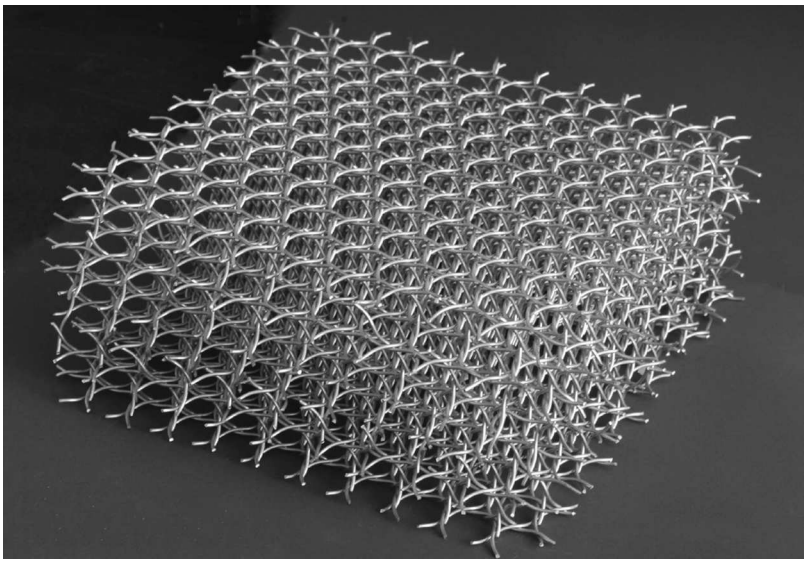

Figure 1. Photograph of the five-layered aluminum WBK specimen after assembling.

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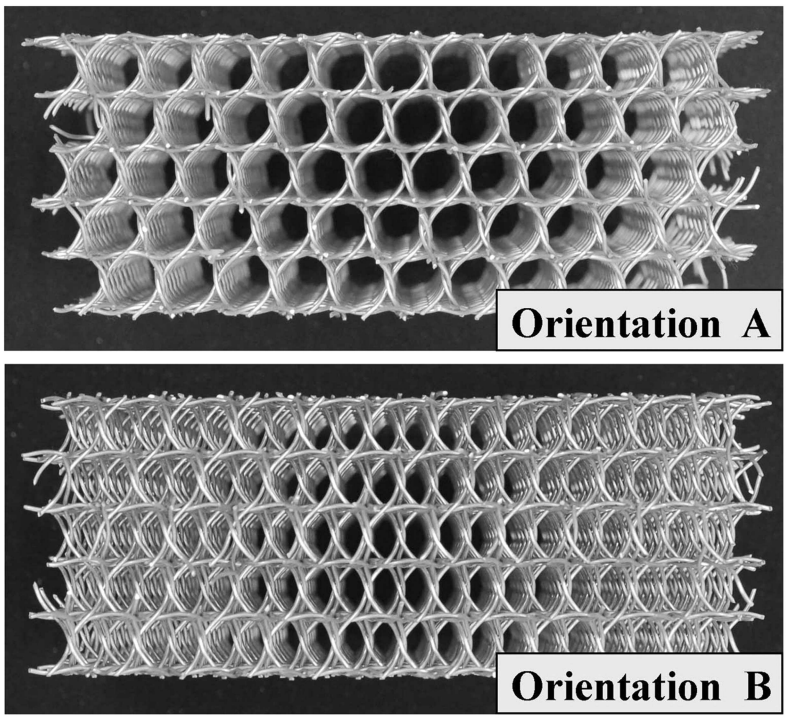

Figure 2. Photographs of side views of the five-layered WBK specimen at two different orientations.

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Table 1. Composition of aluminum 1100-O

Component	Al	Cu	Mn	Other, total	Si + Fe	Zn
Wt %	Min 99	0.05–0.2	Max 0.05	Max 0.15	Max 0.95	Max 0.1

Table 2. Thermal properties of aluminum 1100-O

Coefficient of thermal expansion, linear 250°C	25.5 μm/m°C
Specific heat capacity	0.904 J/g°C
Thermal conductivity	222 W/m K
Melting point	643–657.2°C

struts, the relative density is expressed as

$$\rho_{rel} = \frac{3\sqrt{2}\pi}{8}\left(\frac{d}{a}\right)^2, \tag{1}$$

where d and a are the diameter and length of the truss element, respectively. Correspondingly, the porosity is expressed as

$$\varepsilon = 1 - \rho_{rel}. \tag{2}$$

Lee et al. [19] showed that idealization has little error in estimation of the topological parameters such as the relative density.

Surface Area Density and Open-Area Ratio

Another important parameter for heat transfer analysis is the surface area density, α_A, which is defined as total surface area per unit volume. With the above idealization, the surface area density is given by

$$\alpha_A = \frac{3\sqrt{2}\pi}{2}\left(\frac{d}{a^2}\right) = \rho_{rel}\left(\frac{4}{d}\right). \tag{3}$$

A large surface area density is desirable for enhanced thermal transport in heat exchangers. However, pressure loss likely increases with the surface area density. The aluminum WBK specimen used in the present study has the relative density $\rho_{rel} = 0.03$; that is, the porosity $\varepsilon = 0.97$ and the surface area density $\alpha_A = 123.36\ \mathrm{m^2/m^3}$.

The WBK is intrinsically anisotropic, and therefore, flow passages vary along different orientations, even though the porosity and the surface area density remain unchanged. Therefore, parameters in addition to the porosity must be introduced. To compare the effect of blockage on the pressure drop and heat transfer, two orthogonal orientations (Figure 2) are examined. An additional key parameter, namely, the open-area ratio, is defined as the ratio of the projected void area to the whole area in the flow direction. A geometric model using the commercial code MSC PATRAN2005 was used to valuate the open-area ratio of the WBK specimen with two selected orientations. The open area ratios of the Orientations A and B are 0.75 and 0.62, respectively.

Forced Convection Apparatus

A schematic diagram of the test system for pressure drop and heat transfer measurements as well as its photograph is shown in Figure 3. An open-circuit type wind tunnel was used for these measurements. Air was drawn into a rectangular Perspex channel (150-mm width, 2,400-mm length, and 60-mm height), which contained the test section. A constant heat flux (fixed for all tested cases at 7,080 $\mathrm{W/m^2}$) through a 2-mm thick copper plate; a silicone rubber etched foil (Code no. F060050C8; Watlow Inc., California, USA) with AC electricity was used to heat the bottom face-sheet (copper plate), while the top, side, and bottom walls were thermally insulated with gypsum. Three T-type thermocouples were mounted on the copper plate, and two additional T-type thermocouples were used at the inlet and the outlet of the test section in order to measure

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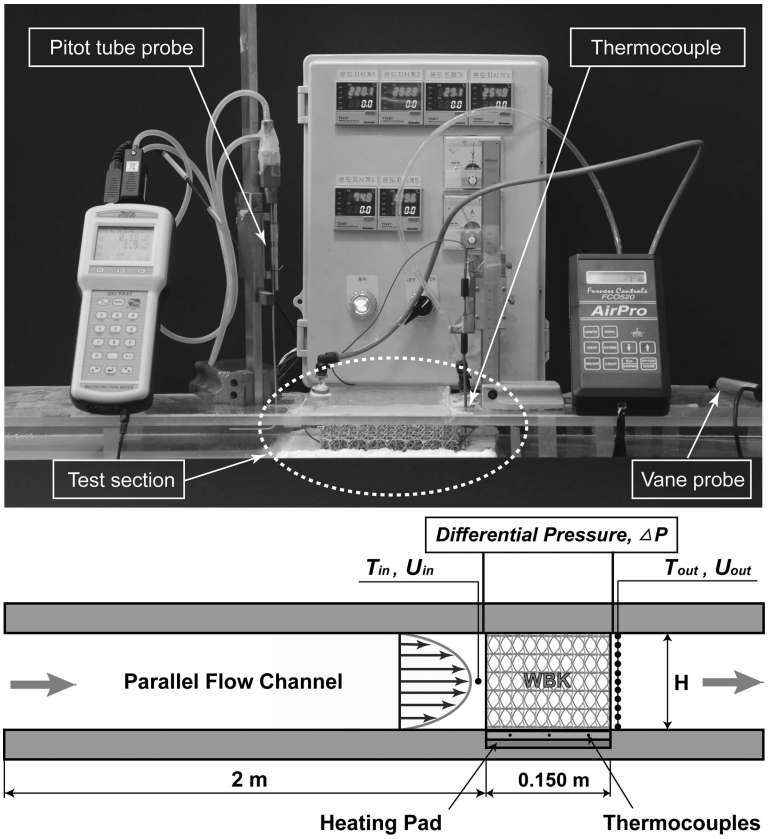

Figure 3. A schematic diagram and photograph of the test rig for pressure drop and heat transfer measurements.

the air temperature along the channel height. Two high-resolution digital manometers (FCO520 AirPro [Furness Controls Ltd., East Sussex, England] and DO9847 [Delta Ohm Inc., Padua, Italy]), were used for static pressure drop and velocity measurements at the inlet and the outlet of the test section. The experimental conditions are listed in Table 3. All measurements were performed under a steady-state condition and were repeated in order to minimize random errors.

The velocity profiles for the laminar and turbulent flow that were fully developed in a channel can be expressed as follows [20]:

for laminar flow,

$$U = U_c \left(1 - \frac{y^2}{H^2}\right), \tag{4a}$$

and for turbulent flow,

$$U = U_c \left(1 - \frac{y}{H}\right)^{1/7}, \tag{4b}$$

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Table 3. Experimental conditions

No.	Mid-height velocity at inlet U_c (m/s)	Mean velocity U_m (m/s)			Reynolds number Re_H ($\times 10^4$)		Bulk mean temperature			
			Outlet				$T_{m,in}$ (K)		$T_{m,out}$ (K)	
		Inlet	O-A	O-B	O-A	O-B	O-A	O-B	O-A	O-B
1	2.0	1.3	0.8	1.7	0.54	0.52	289.6	297.3	303.8	308.1
2	2.7	2.0	1.2	1.9	0.82	0.79	289.6	297.1	299.7	306.3
3	3.5	2.7	1.6	2.0	1.1	1.1	288.9	297.1	296.9	304.5
4	3.8	3.0	1.7	2.1	1.2	1.2	288.9	296.9	296.2	303.4
5	4.5	3.7	2.2	2.2	1.5	1.5	288.9	296.9	295.6	303.3
6	5.2	4.3	2.3	2.4	1.8	1.7	288.4	297.6	293.9	303.5
7	6.9	5.9	3.3	3.0	2.5	2.3	288.6	298.8	293.3	302.7
8	7.5	6.5	3.4	3.3	2.7	2.5	288.8	298.9	293.0	302.5
9	8.3	7.3	3.9	3.6	3.0	2.8	289.1	300.8	293.4	304.3
10	9	7.9	4.3	4.0	3.3	3.1	289.1	300.7	293.8	304.1

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where y is the height measured from the bottom along the centerline, and H is the total channel height. Figure 4 shows the measured inlet coolant velocity profiles along height in the channel. When the velocity at the mid-height, U_c, is 2.6 m/s, the velocity profile is typical for the laminar (fully developed and parabolic). When increasing the mid-height velocity, the velocity profile approaches that for the turbulent flow, which is relatively flat over the height of the channel, rapidly falling to zero velocity at the bottom and top. In this work, we also limit consideration to incompressible flow. This limitation means that the Mach number of the flow must be low ($\mathrm{Ma} \ll 1$), which usually applies to liquid and to gas flow in which the maximum velocity of the gas is less than about 100 m/s. In the present experiments, the highest velocity occurring at maximum $\mathrm{Re}_H = 33{,}000$ is less than about 10 m/s. Therefore, all tested flows can be considered as incompressible flow.

In order to determine the mean velocity of airflow in a rectangular channel, measurements were made at various positions in the channel cross-section by traversing the probe, and an area weighted-average velocity was determined as

$$U_m = \frac{1}{A_{total}} \sum_{i=1}^{N} A_i U_i, \tag{5}$$

where each U_i is measured at the midpoint of the area A_i, and A_{total} is the channel cross-sectional area. Accordingly, the mean velocity, U_m, was empirically correlated as a function of the mid-height velocity, U_c, as

$$U_m = 0.9426 U_c - 0.5636. \tag{6}$$

Measurement Uncertainties

All measurements were performed under a steady-state condition and were repeated in order to obtain significant results. The external surface of the heating element and the

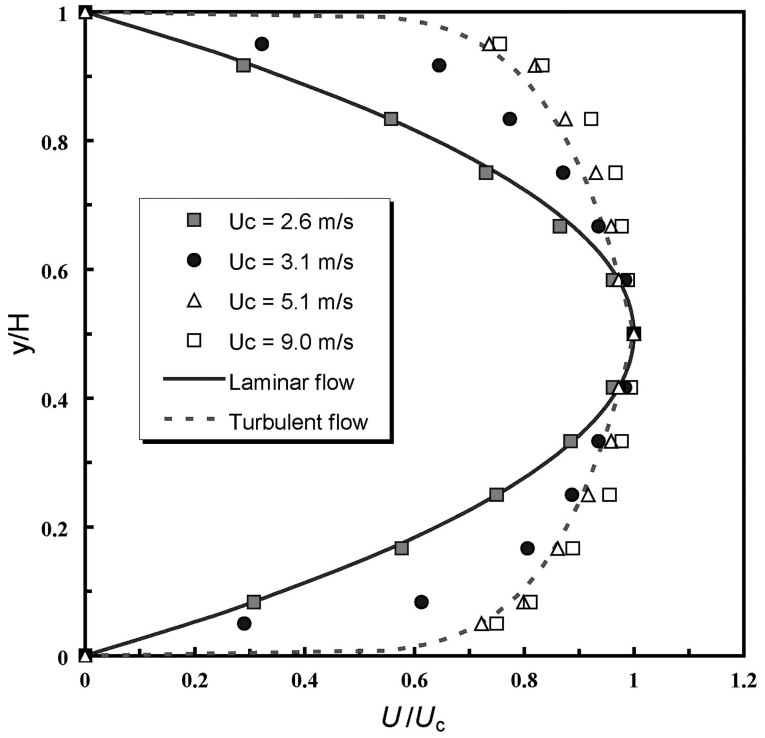

Figure 4. Measured inlet flow velocity profile along the channel height.

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rectangular channel of the test section were covered with a thermal insulation material in order to minimize heat loss. The thermal properties of the coolant are given as a function of temperature in order to minimize errors. The variation of thermal conductivity, k_f, in the air is approximately 3.5% at the operating temperature range of 288.4–300.8 K, while the density of air, ρ_f, varies by about 3.2%. Consequently, the conductivity and density of air are considered to be unequal at the coolant temperature of the inlet and the outlet.

By using the Coleman and Steele approach [21], an uncertainty analysis of measurements was performed. The uncertainty in the measurements of pressure drop was less than 3.8%. The uncertainties calculated from the root-square method for the heat transfer coefficients and Nu_H numbers were estimated to be less than 3.5% and 4.6%, respectively.

FRICTION FACTOR

The design of high-performance heat exchangers requires a high rate of heat transfer and a low mechanical pumping power to overcome the pressure drop across the heat exchanger media. During the evaluation of the pressure drop, a non dimensional pressure

coefficient, the so-called "friction factor" is used, defined typically as

$$f = \left(\frac{\Delta p}{L} \cdot H\right) \cdot \left(\frac{1}{\rho_f U_m^2/2}\right), \tag{7}$$

where $\Delta p/L$, ρ_f, and U_m are the static pressure drop per unit length, the fluid density, and the mean air velocity at the inlet of the test section of length L, respectively. In this study, the channel height, H, was selected as a length scale to compare the experimental results obtained from the WBK specimen with those from other conventional heat dissipation media. The coolant velocity can also be expressed in a dimensionless form by the Reynolds number, Re_H, defined as

$$\mathrm{Re}_H = \frac{\rho_f U_m H}{\mu_f}, \tag{8}$$

where μ_f is the viscosity of air.

The static pressure drops when the flow enters the test section and rises when it exits. The pressure changes include reversible (flow-area change) as well as irreversible (boundary layer separation) processes [16]. In this work, the relative density of the WBK specimen is only 4%. Accordingly, it is expected that the entrance and exit effect is minimal, although it has never been proved. Static pressure drop measurements across the multi-layered WBK specimen were performed first. In Figure 5, the experimental results of pressure drop in Orientations A and B are represented by symbols (circles and rectangles) together with curve fitting. The corresponding friction factors are presented

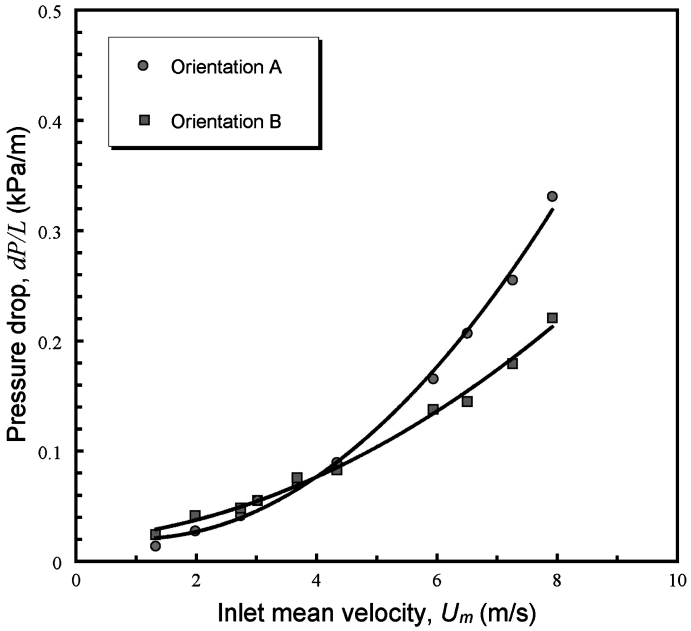

Figure 5. Pressure drop per unit length of the multi-layered WBK as a function of the inlet mean velocity for two selected orientations.

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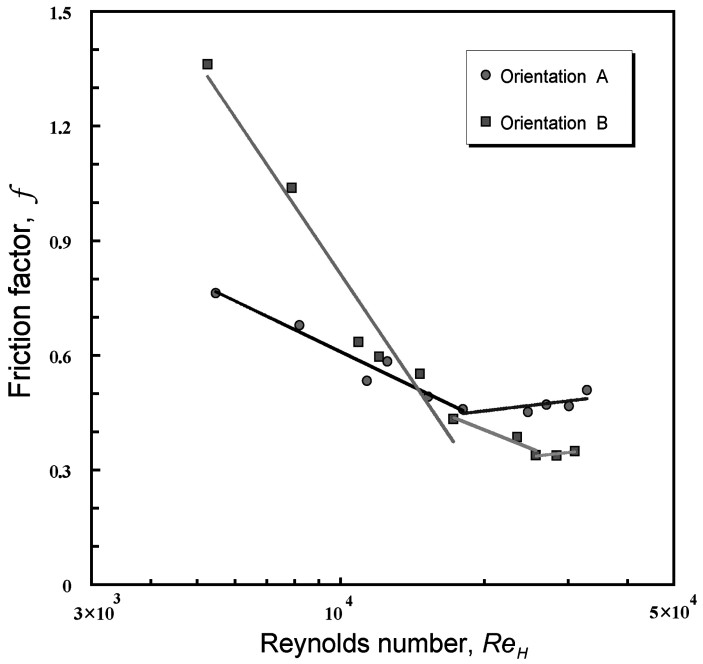

Figure 6. Measured friction factor as a function of the Re number based on the channel height.

in Figure 6. The friction factor, f, can be correlated with the Reynolds number, Re_H, as follows: for Orientation A,

$$f = -0.2608 \ln(\mathrm{Re}_H + 3.0116, \qquad 5.5 \times 10^3 \le \mathrm{Re}_H \le 1.8 \times 10^4, \tag{9a}$$

$$f = 0.0645 \ln(\mathrm{Re}_H) - 0.1847, \qquad 1.8 \times 10^4 \le \mathrm{Re}_H \le 3.3 \times 10^4, \tag{9b}$$

and for Orientation B,

$$f = -0.8049 \ln(\mathrm{Re}_H) + 8.2254, \qquad 5.2 \times 10^3 \le \mathrm{Re}_H \le 1.7 \times 10^4, \tag{10a}$$

$$f = -0.2165 \ln(\mathrm{Re}_H) + 2.5497, \qquad 1.7 \times 10^4 \le \mathrm{Re}_H \le 2.5 \times 10^4, \tag{10b}$$

$$f = 0.0585 \ln(\mathrm{Re}_H) - 0.2570, \qquad 2.5 \times 10^4 \le \mathrm{Re}_H \le 3.1 \times 10^4. \tag{10c}$$

The friction factor of the multi-layered WBK ranged from 0.36 to 1.36, depending on the Reynolds numbers. The plot of the friction factor levels off a permeability based Re_H of approximately 17,000. In this range, the pressure drop over the multi-layered WBK specimen is dominated by the form drag. In both orientations, the friction factor rapidly decreases with the increase of the Re number (i.e., $\mathrm{Re}_H < 17{,}000$) followed by slight increases ($\mathrm{Re}_H > 17{,}000$). At a low Re number, e.g. $\mathrm{Re}_H > 15{,}000$, the friction factor of Orientation B is higher than that of Orientation A, because the flow resistance by Orientation B is larger than that of Orientation A. These results indicate that the pressure drop is mainly contributed by the flow blockage effect expressed by the open-area ratio.

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HEAT TRANSFER PERFORMANCE

The heat transfer rate mainly depends on the balance of both thermal conduction and convection capability. To evaluate heat transfer characteristics, the heat transfer coefficient, h, and the corresponding Nusselt number, Nu_H, were defined as

$$h = \frac{Q}{A}\frac{1}{\Delta T} = \frac{q}{T_w - T_{f,ave}}, \qquad \mathrm{Nu}_H = \frac{h \cdot H}{k_f}, \tag{11}$$

where A is the heating area of the copper plate, Q is the rate of heat transfer ($= \dot{m}c_p (T_{m,out} - T_{m,in})$), T_w is the average temperature of the copper plate, k_f is the thermal conductivity of air, and $T_{f,ave}$ is the average of the bulk mean temperature of air at inlet and outlet. In order to determine the bulk mean temperature of airflow at the test section outlet, flow temperature and velocity were measured at various longitudinal locations along the direction of the flow, and area weighted-mean temperature (bulk mean temperature) was determined as

$$T_{m,out} = \frac{1}{A_{total}U_{m,out}} \sum U_i T_i dA_i, \tag{12}$$

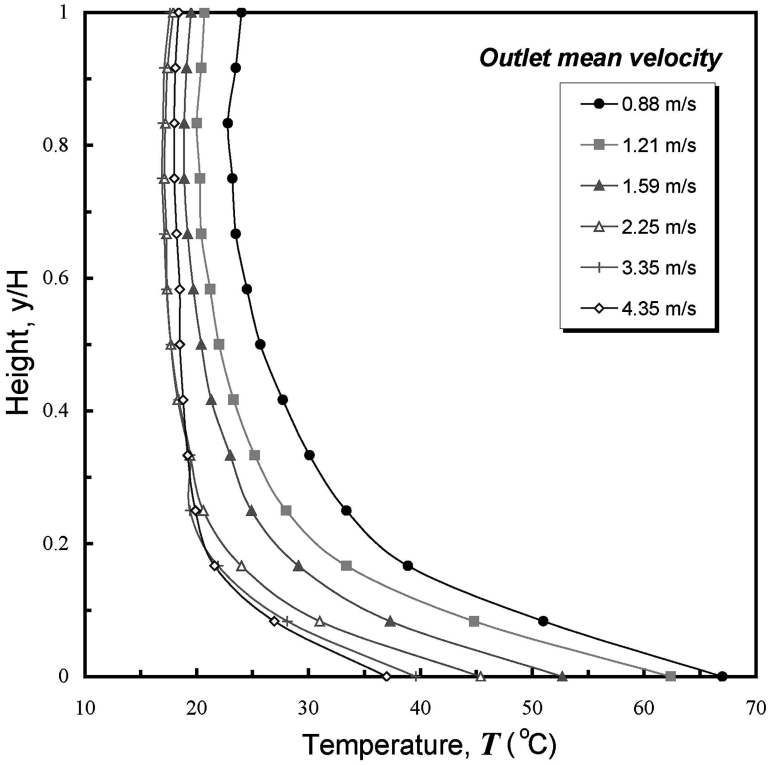

Figure 7. Temperature profile along the channel height at outlet (Orientation A).

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where each U_i and T_i were measured at the midpoint of the area A_i, and A_{total} is the channel cross-sectional area. For Orientation A, experimental results of the temperature profile along the channel height at the outlet and the bulk mean temperatures at the inlet and the outlet are shown together with curve fittings in Figures 7 and 8, respectively. The results show that the bulk mean temperature at the outlet decreases with the increase of inlet mean velocity under the forced air convection. In addition, the rate of decrease changes at the inlet mean velocity of approximately 2.6 m/s, which seems to be attributed to the change from a laminar to a turbulent flow pattern. Incidentally, a laminar flow can be transformed to a turbulent flow during air passes through the test section (WBK sample). In fact, the change from a laminar to a turbulent flow pattern due to formation of horseshoe- and arch-shaped vortices around truss elements of a truss type cellular metal was already reported [17, 18], even though this aspect is not considered in this study. Also, from the heat transfer point of view, the flow pattern change enhances convective heat transfer, and the good performance of the multi-layered WBK is partly attributed to the flow pattern change.

Figure 9 shows the Nu_H numbers of the multi-layered aluminum WBK, which is correlated as a function of the Re_H number and the Pr number: for Orientation A,

$$\mathrm{Nu}_H = 0.183\,\mathrm{Re}_H^{0.8}\,\mathrm{Pr}^{0.4}, \qquad 5.4 \times 10^3 \le \mathrm{Re}_H \le 1.8 \times 10^4, \tag{13a}$$

$$\mathrm{Nu}_H = 0.205\,\mathrm{Re}_H^{0.8}\,\mathrm{Pr}^{0.4}, \qquad 1.8 \times 10^4 \le \mathrm{Re}_H \le 3.3 \times 10^4, \tag{13b}$$

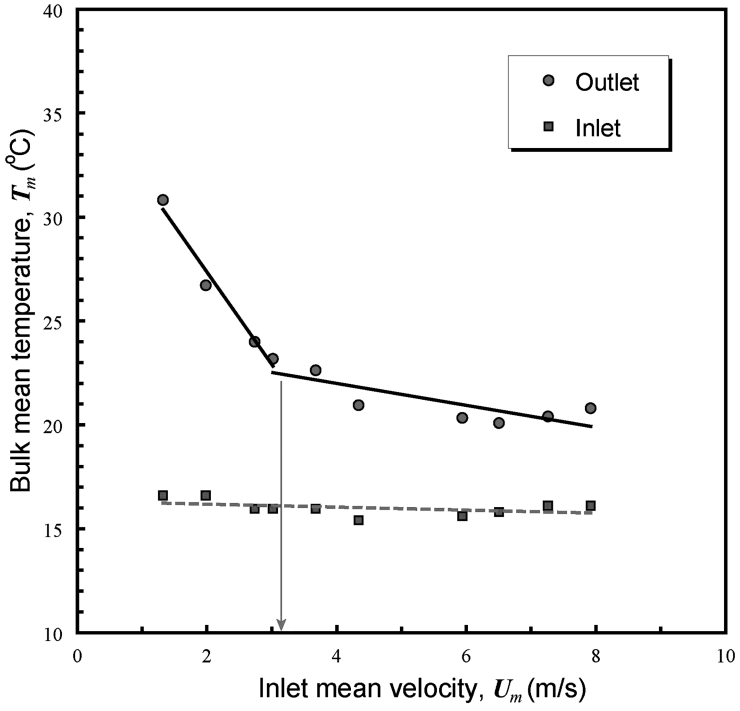

Figure 8. Bulk mean temperatures at inlet and outlet (Orientation A).

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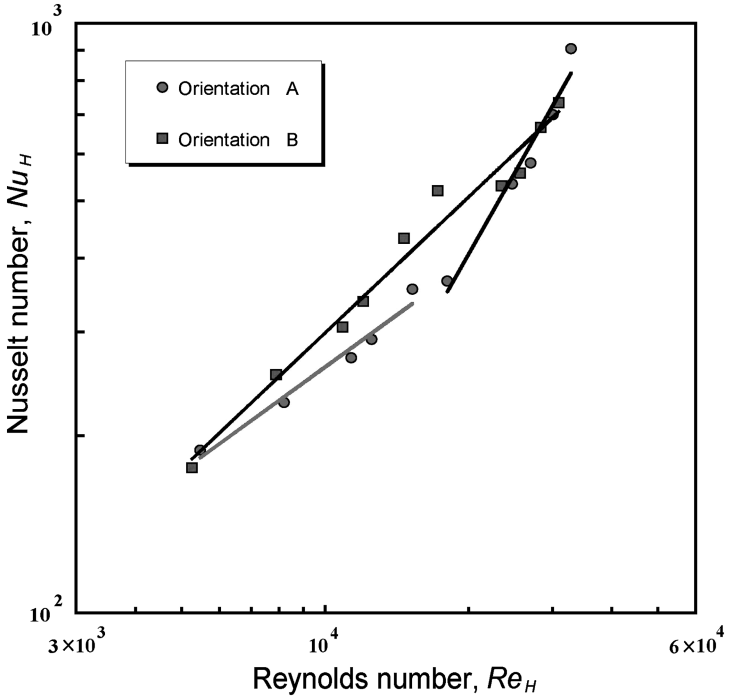

Figure 9. Measured Nu number as a function of the Re number based on the channel height.

and for Orientation B,

$$\mathrm{Nu}_H = 0.212\,\mathrm{Re}_H^{0.8}\,\mathrm{Pr}^{0.4}, \qquad 5.2\times10^3 \le \mathrm{Re}_H \le 3.1\times10^4, \tag{14}$$

where the Pr number is assumed to be constant (= 0.7). Over the whole measured range of Reynolds number, the Nusselt number of the multi-layered WBK ranged from 171 to 906. The Nu_H number of the WBK in Orientation B is marginally higher than that in Orientation A, while the open-area ratio of Orientation B is marginally lower than that of Orientation A. This implies that Orientation B has more blockage area, where the WBK struts impact on the mainstream cooling flow, resulting in a higher heat transfer rate. That is, an increase in the turbulence of the cooling flow causes an increase in heat transfer by open-area ratio or blockage effect. Therefore, these results infer that the rate of heat transfer depends on an open-area ratio that changes with specimen orientation.

COMPARISONS

Numerous studies have been carried out to characterize the cross-flow heat exchanger performance of the various cellular metals. For a wide range of the Re_H number, Figures 10 and 11 show the friction factor and the heat transfer performance of the multi-layered aluminum WBK, respectively, in comparison with those of other heat dissipation media (aluminum, copper and iron base alloy foams, bronze packed beds, copper-woven screens, LFMs, and Kagome truss structures). They also show data of a corrugated duct

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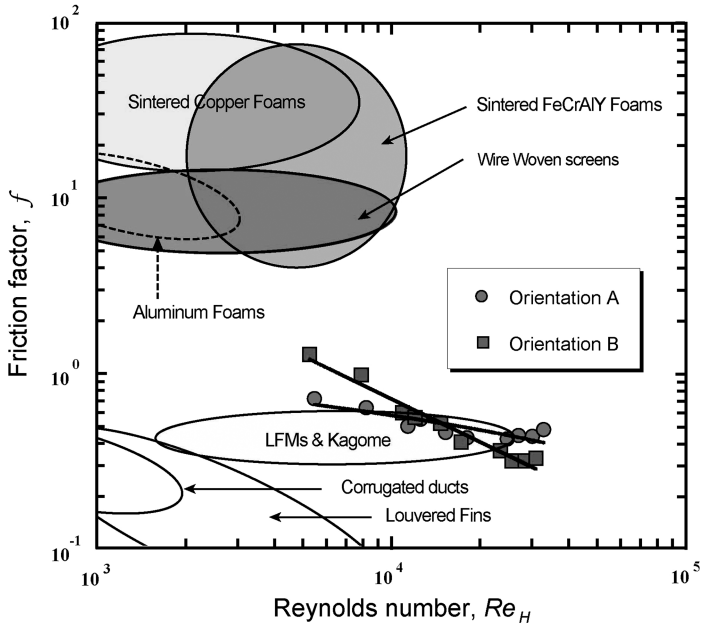

Figure 10. Friction factor of WBK with variation of Re number in comparison with those of other dissipation media [10 16].

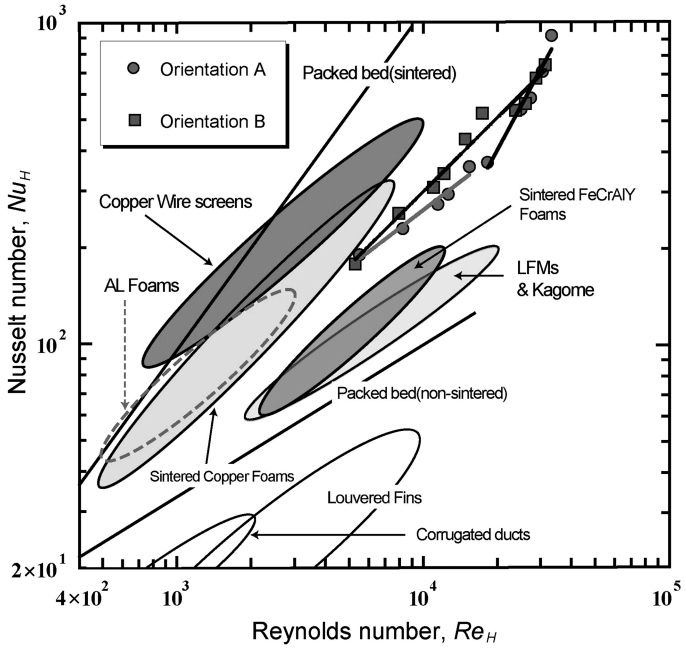

Figure 11. Nu number of WBK with variation of Re number in comparison with those of other dissipation media [10, 16].

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and a louvered fin structure [10, 16]. The friction factor of the WBK is slightly higher than that of louvered fins but much lower than that of metal foams and wire-woven screens. The friction factor of the WBK is similar to that of aluminum LFMs and cast Kagome structures (porosity ~ 0.9). This is because the relative density or the porosity of the material is a primary factor to affect the friction factor, that is, the pressure drop. Namely, a similar density yields a similar friction.

Meanwhile, the multi-layered aluminum WBK has a higher Nu_H number than the aluminum LFMs and cast Kagome structures, because its surface area density is 14.9% higher than that of the Kagome structures, although its relative density is 7.7% lower. On the other hand, the Nu_H number of the WBK is lower than that of copper wire screens, which have 1.7 times higher thermal conductivity than that of aluminum. The Nu_H number of the WBK is similar to that of sintered copper and aluminum foams, because its main flow mixes fully owing to WBK topology, although its porosity is higher.

A thermal efficiency index can be used [10, 16] in order to compare the performance of heat transfer for different heat dissipation media at a fixed pumping power condition. The index is expressed as

$$\text{Thermal Efficiency Index} = \frac{\mathrm{Nu}_H}{f^{1/3}}. \tag{15}$$

Figure 12 compares the thermal efficiency indices for various heat dissipation media that were calculated by using the data illustrated in Figures 10 and 11. This figure physically ranks the heat transfer performance of these heat dissipation media at a fixed

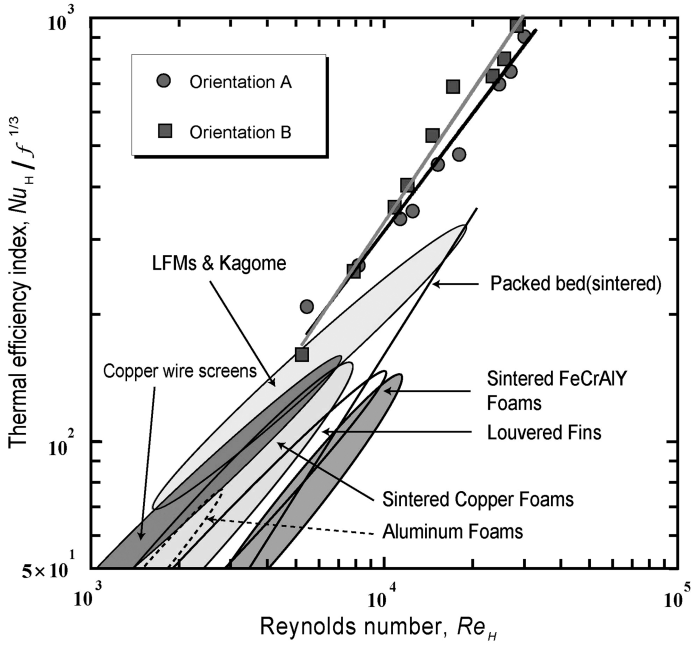

Figure 12. Thermal efficiency index of WBK with variation of Re number in comparison with those of other dissipation media [10, 16].

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pumping power condition [10]. The performance of heat transfer of the multi-layered aluminum WBK is superior to the competitors. This is attributed to a combination of the lower pressure drop and higher heat transfer of the WBK due to the lower relative density and the higher surface area density than those of other heat dissipation media. Consequently, the WBK have excellent heat dissipation characteristics at an acceptably low pressure drop. These results can be used to guide the design and fabrication of optimum WBK structures that would be able to maximize the heat transfer per unit pumping power.

CONCLUSIONS

The friction factor and heat transfer characteristics of a multi-layered aluminum WBK were experimentally investigated under a forced-air convection condition for two selected orientations. The friction factor of the multi-layered WBK ranged from 0.36 to 1.36, depending on the Reynolds numbers. When $\mathrm{Re}_H < 15{,}000$, the friction factor of Orientation B is higher than that of Orientation A, which is attributed to the higher blockage effect of Orientation B. At higher Reynolds numbers, e.g., $\mathrm{Re}_H > 15{,}000$, however, the friction factor of Orientation B is marginally lower. The Nusselt number of the multi-layered WBK ranged from 171 to 906. From the results, the fiction factor and the Nusselt number of the WBK are mainly affected by the open-area ratio, and it has excellent heat dissipation characteristics at acceptably lower pressure drop compared to other PCMs. Consequently, the performance as a heat dissipation medium, scaled by the thermal efficiency index, is comparable in both orientations. In comparison with other heat dissipation media, the fiction factor of the WBK is much lower than that of metal foams and comparable to that of LFMs and cast Kagome structures. The Nusselt number of the WBK is slightly lower than that of metal foams and higher than that of LFMs and cast Kagome structures. As a result, the WBK outperforms the competitors in the thermal efficiency indices. In order to evaluate the performance of the WBK in the lower range of Reynolds numbers, some additional experiments are needed.

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