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Liquid-Metal Pin-Fin Pressure Drop by Correlation in Cross Flow

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Introduction

The pin-fin configuration is widely used as an heat transfer enhancement method in high-heat-flux applications. Recently, the pin-fin design with liquid-metal coolant was also applied to synchrotron-radiation beamline devices. This paper investigates the pressure drop in a pin-post design beamline mirror with liquid gallium as the coolant. Because the pin-post configuration is a relatively new concept, information in literature about pin-post mirrors or crystals is rare, and information about the pressure drop in pin-post mirrors with liquid metal as the coolant is even more sparse. Due to this we considered the cross flow in cylinder-array geometry, which is very similar to that of the pin-post, to examine the pressure drop correlation with liquid metals over pin fins. The cross flow of fluid with various fluid characteristics or properties through a tube bank was studied so that the results can be scaled to the pin-fin geometry with liquid metal as the coolant. Study lead to two major variables to influence the pressure drop: fluid properties, viscosity and density, and the relative length of the posts. Correlation of the pressure drop between long and short posts and the prediction of the pressure drop of liquid metal in the pin-post mirror and comparison with an existing experiment are addressed.

Background

Historically, one can list two categories of variables that affect the pressure drop across a tube bank [1]: primary variables that have a relatively large influence on the pressure drop and

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secondary variables that do not have as large an influence as do the primary variables. Table 1 lists the variables affecting pressure drop across a tube bank.

Table 1. Variables affecting pressure drop across tube banks

I. Primary Variables
A. Operating conditions
1. Fluid-flow rate
2. Fluid characteristics or properties
B. Tube-bank arrangement
1. Tube spacing (transverse and longitudinal)
2. Tube alignment characteristics (in-line and staggered)
3. Number of rows of tubes
II. Secondary Variables
A. Operating conditions
1. Nonisothermal or nonisobaric fluid conditions
B. Tube-bank arrangement
1. Fluid approach configuration
2. Tube-surface characteristics: roughness
3. Wall leakage past tubes
4. Wall frictional effect
5. Mutual effect of tube rows on each other
6. Tube-bundle shell configuration
7. Tube-baffle clearance leakage

It is generally possible to take into account all the primary variables by means of a friction factor-Reynolds number relationship with tube spacing and alignment as parameters. Secondary variables are too numerous and have been insufficiently investigated to permit any quantification, however their effect is usually deemed to be of minor nature.

Brandt and Dingler [2] found that, for turbulent flow, a given arrangement yielded substantially a straight line when plotted as friction factor vs. Reynolds number on a log-log basis. They proposed, therefore, to represent the friction factor as a power function of the Reynolds number and presented families of curves relating the required proportionality coefficient and exponent to the tube spacing. Gaddis and Gnielinski [3] discuss the most common methods of calculating the pressure drop in cross flow across a tube bank. Generally the pressure drop Δp in cross flow across a tube bank is given by

$$\Delta p = \xi N \frac{\rho w^2}{2}, \quad (1)$$

where ξ is a function of geometrical constants and Reynolds number, N characterizes the number of rows of tubes in the bundle, ρ is the fluid density, and w is the velocity. For turbulent flow, Various methods can be used [3] to calculate the different parameters used in the above formula. A new way is suggested [3] to calculate the friction factor. The equation has the same form as equation (1), and ξ is a parameter dependent upon Reynolds number and geometry. For turbulent flow, the friction factor of an in-line configuration is

$$\xi = \frac{f_{a,t,f}}{\text{Re}^{0.1(b/a)}}, \quad (2)$$

and the friction factor of a staggered configuration is

$$\xi = \frac{f_{a,t,v}}{\text{Re}^{0.25}}, \quad (3)$$

where $f_{a,t,f}$ and $f_{a,t,v}$ are geometry-dependent coefficients. To take into account the heating or cooling of the fluid in nonisothermal flow, a factor F_{zn} should be multiplied with ξ when calculating the pressure drop. Gaddis and Gnielinski [3] suggested that, for isothermal flow and a tube bank with $N \geq 10$, $F_{zn} = 1$, and, for nonisothermal flow at high Reynolds numbers, $F_{zn} = (\eta_w/\eta)^{0.14}$, where η is the viscosity of fluid at its mean temperature, and η_w is the viscosity of fluid at mean wall temperature.

Some isothermal (and approximately isothermal) data for turbulent flow as reported in the literature are summarized by Boucher and Lapple [1]. All pressure-drop data have been reduced to a correlated form of the friction factor and Reynolds number for uniformity. In most cases, a straight line would adequately represent the data when plotting friction factor vs. Reynolds number on log-log scale. Boucher and Lapple [1] found that the wall can affect measured results by introducing frictional effects of its own, as well as by influencing the general velocity distribution and flow pattern within the tube bank. This effect is of importance only for cases in which the tube length, expressed as a number of tube diameters, is relatively short or the bank is not many rows wide. Outstanding in this respect is the arrangement tested by Brandt and Dingler [2] in which the tubes are only 1.25 to 2.5 diameter long. These results show that the wall-friction effect is small,

and the pressure drop through a short tube bank can be approximated by the results from a long tube bank. More importantly, because the wall-friction effect is a secondary variable, the relation between the friction factor and Reynolds number can still be expressed as an exponential function.

Metzger [6] compared the overall flow friction behavior of a short pin-fin with that of a long cylinder array using data from other authors. Metzger's results for short pin-fin geometry [7] are in good agreement with the long cylinder results over the entire test range. When the pressure drop is calculated, the long cylinder results have a different Reynolds number dependence in the range $10^3 < Re < 10^4$ than they do in the range $10^4 < Re < 10^5$. Metzger [7] found that this also carried over to the short pin-fin case, which is represented to within 15% by the following two equations:

$$f = 0.317 Re^{-0.132} \quad \text{for } 10^3 < Re < 10^4 \quad (4)$$

and

$$f = 1.76 Re^{-0.318} \quad \text{for } 10^4 < Re < 10^5 \quad (5)$$

Olson's test [8] shows that the pressure drop of a compact pin-fin heat exchanger can be expressed as $f = 0.8561Re^{-0.216}$.

The influence of viscosity and density on the pressure drop

The thermophysical properties of fluids, such as density and viscosity, have been taken into account in the friction factor-Reynolds number correlation. Both theoretical and experimental results from various sources [1-11] show that the exponential function can be used to express that relationship no matter what kind of coolant is used. The nonisothermal effect can be taken into account by a correction factor, such as F_{zn} , as proposed by Gaddis and Gnielinski[3].

Prediction of the pressure drop for gallium in a pin-post design and comparison with experimental results

From the above review for both short and long cylinders with different coolant properties, it can be seen that the wall-friction effect is a secondary variable, and the pressure drop for short cylinders can still be predicted by the results for long cylinders. The fluid characteristics can be taken into account by the friction factor-Reynolds number relationship. The fluid nonisothermal effect can be taken into account by a correction factor, e.g., $F_{zn} = (\eta_w/\eta)^{0.14}$. Therefore, the pressure drop through a pin-post mirror or crystal with gallium as the coolant can be predicated by the following equation:

$$\Delta p = \xi F_{zn} \frac{\rho w^2}{2}, \quad (6)$$

where ξ is a function of geometry and Reynolds number is defined by:

$$\xi = \frac{f(a,b,N)}{Re^m}, \quad (7)$$

f is a geometry-dependent constant, and m is a constant.

By assuming the value for water with a subscript "h" and for gallium with "g", one can predicate the pressure drop of gallium versus that of the water (assuming the same geometry of the pins and the same flow velocity):

$$\begin{aligned} \Delta p_g &= \frac{\xi_g F_{zn,g} \rho_g w_g^2 / 2}{\xi_h F_{zn,h} \rho_h w_h^2 / 2} \Delta p_h = \frac{(f_g / Re_g^m) F_{zn,g} \rho_g w_g^2}{(f_h / Re_h^m) F_{zn,h} \rho_h w_h^2} \Delta p_h \\ &= \frac{Re_h^m F_{zn,g} \rho_g w_g^2}{Re_g^m F_{zn,h} \rho_h w_h^2} \Delta p_h = \frac{(w_h d \rho_h / \eta_h)^m F_{zn,g} \rho_g w_g^2}{(w_g d \rho_g / \eta_g)^m F_{zn,h} \rho_h w_h^2} \Delta p_h \\ &= \left(\frac{\rho_g}{\rho_h} \right)^{1-m} \left(\frac{\eta_g}{\eta_h} \right)^m \frac{F_{zn,g}}{F_{zn,h}} \Delta p_h \end{aligned} \quad (8)$$

If an isothermal turbulent flow is assumed, F_{zn} approximately equals unity. Substituting $\rho_h = 1$, and $\rho_g = 6$ into equation (8) and assuming $m = 0.25$, we have

$$\Delta p_g = \left(\frac{6.0}{1.0} \right)^{1-m} \left(\frac{2.04 \times 10^{-3}}{9.8 \times 10^{-4}} \right)^m \Delta p_h = 4.61 \Delta p_h.$$

If $m = 0.2$ is used, we have

$$\Delta p_g = \left(\frac{6.0}{1.0} \right)^{1-m} \left(\frac{2.04 \times 10^{-3}}{9.8 \times 10^{-4}} \right)^m \Delta p_h = 4.86 \Delta p_h.$$

Smither and others [9] have performed experiments with liquid-gallium-cooled crystals and compared those results with the results from experiments with water-cooled crystals as listed in Tables 2-3 [10]. When the mass flow rates are the same, the experiments show that the pressure drop of gallium is about 0.17 times that of water.

If we assumed that the mass flow rates are the same, that is,

$$\frac{w_h}{w_g} = \frac{\rho_g}{\rho_h},$$

from equation (6), the following results are obtained:

$$\begin{aligned} \Delta p_g &= \frac{\xi_g F_{zg} \rho_g w_g^2 / 2}{\xi_h F_{zh} \rho_h w_h^2 / 2} \Delta p_h = \frac{(f_g / \text{Re}_g^m) F_{zg} \rho_g w_g^2}{(f_h / \text{Re}_h^m) F_{zh} \rho_h w_h^2} \Delta p_h \\ &= \frac{\text{Re}_h^m F_{zg} \rho_g w_g^2}{\text{Re}_g^m F_{zh} \rho_h w_h^2} \Delta p_h = \frac{(w_h d \rho_h / \eta_h)^m F_{zg} \rho_g w_g^2}{(w_g d \rho_g / \eta_g)^m F_{zh} \rho_h w_h^2} \Delta p_h \quad (9) \\ &= \left(\frac{\eta_g}{\eta_h} \right)^m \frac{\rho_h}{\rho_g} \Delta p_h \end{aligned}$$

Substituting the material properties of gallium and water, as well as the constant $m = 0.2$, into equation (9), the pressure drop of gallium can be found to be 0.19 times that of water [10]. This is reasonably close to the experimental result, 0.17, given by Smither and others [9]. Because no geometry is given in Ref. [10], it is impossible to know whether the flow in the tests described in references [9] and [10] was laminar or turbulent. Because the velocity of gallium is relatively low, the flow might be in the laminar range. Therefore, if the value of m is taken according to equation (4) (i.e., $m = 0.132$ for laminar flow), the ratio of the pressure drop of gallium to that of water is 0.18, still very close to the experimental results. In reality, the measured pressure drop with water is 2.8 psi and with gallium is 16.21 psi for a flow rate of 120 lb/h [10].

The analysis offered above explains this increase in the actual pressure drop with gallium from the fundamental considerations. Here relation (8) or (9) can be confidently used to predict pressure drop of mirror/crystals with pin-post fins for a variety of coolant used.

Discussion

Equation (6) applies to both low Reynolds number and high Reynolds number cases. However, the value of m are different for the two cases. Also, the value of the correction factor, F_{zn} , for nonisothermal conditions should be decided differently. When scaling the pressure drop from one coolant to another, one should remember that m may be different even if the two coolants have the same velocity or the same mass flow speed.

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Table 2

Liquid gallium and Water comparison (microchannel flowrate = 120 lb/h)

Average heat load [W/cm ²]	Coolant	Distortion [μ m] 1-D maximum	ΔT surface [°C]	ΔP channels [psi]	ΔT coolant [°C]
100	liquid gallium	2.613	0.85	2.8	8.02
	water	0.505	10.47	16.21	0.741
1000	liquid gallium	26.123	8.26	2.8	80.2
	water	5.045	104.46	16.21	7.41

Table 3

Liquid gallium and water comparison (microchannel flowrate = 2400 lb/h)

Average heat load [W/cm ²]	Coolant	Distortion [μ m] 1-D maximum	ΔT surface [°C]	ΔP channels [psi]	ΔT coolant [°C]
100	liquid gallium	0.21	0.80	613.5	0.41
	water	0.325	3.92	3550	0.037
1000	liquid gallium	1.50	7.77	613.5	4.10
	water	1.23	38.91	3550	0.37

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