文章编号: 1001－2060(2004)01－0042－03

等截面直肋传热简化计算的适用条件

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摘　要: 工程上在进行等截面直肋计算时, 常用肋端绝热计算公式代替肋端对流换热计算公式计算传热。采用理论分析方法, 将这样近似的误差表示为截(面积与)侧面积比 $f/(Uh)$ 和毕渥数 Bi 的函数, 并通过可能取值范围的计算, 表明当等截面直肋的截侧面积比 $f/(Uh)<0.5$ 或 $Bi>7$ 两个条件满足一个这种计算方法的误差小于 1%。

关　键　词: 假想肋高; 毕渥准则; 截侧面积比

中图分类号: TK124　　　文献标识码: A

1　引　言

肋片是一种常用的强化换热方法。由于加工方便, 又以等截面直肋应用最广。如将与周围流体的对流换热系数取为常数, 则直肋的传热计算可以用分析法求得[1]。肋片肋端的换热是第三类边界条件, 但这样得到的公式比较复杂, Harper 和 Brown 提出对于等截面直肋, 采用肋端绝热的边界条件公式代替肋端对流换热的公式来计算直肋的传热量[2], 因忽略了肋端传热, 故计算结果会有一定的误差。为了减少这种误差, Harper 和 Brown 建议用假想肋高代替实际肋高, 然后使用端部为第二类边界条件的计算公式进行计算。Harper 和 Brown 的建议被广泛采用[1,3~7], 计算结果表明, 如果以半厚度作为特征尺度的毕渥数 $Bi_{\delta/2}=\alpha\delta/2\lambda<0.25$ 则这样做的误差小于 8%。文献[6] 指出"实践表明, 只要 $\sqrt{\alpha\delta/\lambda}\leq 1/4$, 这样计算的误差小于 1%"。本文通过理论分析, 将这种计算方法的误差用两个无量数; 截侧面积比 $f/(Uh)$ 和毕渥数 Bi 表示, 在这两个参数的可能取值范围进行计算, 找出了误差的分布规律, 并指出这种计算方法的适用范围。

2　直肋传热计算方法

在对流换热系数 α 为常数时, 一维、稳态、无内热源、肋断为第三类边界条件的等截面直肋导热问题的分析解为[1]:

$$Q_3=m\lambda f\theta_0\frac{\alpha/m\lambda+\tanh(mh)}{1+(\alpha/m\lambda)\tanh(mh)}\tag{1}$$

式中: λ 为肋的导热系数; θ_0 是肋基的过余温度; h 为肋的高度; f 是肋的截面面积; U 是肋截面周边的长度; $m=\sqrt{\alpha U/\lambda f}$。采用第二类边界条件, 即肋端绝热, 则传热量 Q_2 的计算公式为[1~7]:

$$Q_2=\frac{\alpha U}{m}\theta_0\tanh(mh)\tag{2}$$

式(1) 的计算比较麻烦, 为此, Harper 和 Brown 提出采用式(2) 来计算直肋的传热量[2], 因忽略了肋端传热, 故计算结果会有一定的误差。为了减少这种误差, Harper 和 Brown 建议用假想肋高 $h'=h+\delta/2$ 代替实际肋高。按照这个思路, 其它截面形状的等截面直肋的假想肋高应为[1]:

$$h'=h+\delta=h+\frac{f}{U}\tag{3}$$

式中: δ 是截面积 f 与截面周长的长度 U 之比:

$$\delta=f/U\tag{4}$$

用假想肋高代替式(2) 中的实际肋高 h, 得到通过肋的热流量 Q'_2:

$$Q'_2=\frac{\alpha U}{m}\theta_0\tanh(mh')=\frac{\alpha U}{m}\theta_0\tanh\left[m\left(h+\frac{f}{U}\right)\right]\tag{5}$$

3　计算误差分析

为了分析误差, 引入了毕渥准则[4]:

$$Bi=\frac{\alpha l}{\lambda}=\frac{\alpha}{\lambda}\cdot\frac{Uh^2}{f}=(mh)^2\tag{6}$$

可见 Bi 数是以 Uh^2/f 为特征尺度的导热热阻与对流换热热阻的比值。将式(6) 分别代入式(1) 和式(2), 化简得:

$$Q_3=(Bi)^{\frac{1}{2}}\frac{\lambda f}{h}\theta_0\left[\frac{\sinh(Bi)^{\frac{1}{2}}+(Bi)^{\frac{1}{2}}(f/Uh)\cosh(Bi)^{\frac{1}{2}}}{\cosh(Bi)^{\frac{1}{2}}+(Bi)^{\frac{1}{2}}(f/Uh)\sinh(Bi)^{\frac{1}{2}}}\right]\tag{7}$$

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$$Q_2 = (Bi)^{\frac{1}{2}}\frac{\lambda f}{h}\theta_0\tanh(Bi)^{\frac{1}{2}} \tag{8}$$

若用假想肋高 $h' = h + \frac{f}{U}$ 替代式(6)中的实际肋高 h，则得：

$$Bi' = m^2\left(h + \frac{f}{U}\right)^2 \tag{9}$$

所以

$$(Bi')^{\frac{1}{2}} = m\left(h + \frac{f}{U}\right) = mh\left(1 + \frac{f}{Uh}\right) = (Bi)^{\frac{1}{2}}\left(1 + \frac{f}{Uh}\right) \tag{10}$$

用 Bi'，h' 分别替代式(8) 中的 Bi，h 得：

$$Q'_2 = (Bi')^{\frac{1}{2}}\frac{\lambda f}{h'}\theta_0\tanh(Bi')^{\frac{1}{2}} = (Bi)^{\frac{1}{2}}\frac{\lambda f}{h}\theta_0\tanh\left[(Bi)^{\frac{1}{2}}\left(1 + \frac{f}{Uh}\right)\right] \tag{11}$$

用假想肋高计算时的相对误差为：

$$e = \left|\frac{Q'_2 - Q_3}{Q_3}\right| \times 100\% = \left|\frac{Q'_2}{Q_3} - 1\right| \times 100\% \tag{12}$$

将式(7) 和式(11) 代入式(12) 得：

$$e = \left|\frac{\cosh(Bi)^{\frac{1}{2}} + (Bi)^{\frac{1}{2}}(f/Uh)\sinh(Bi)^{\frac{1}{2}}}{\sinh(Bi)^{\frac{1}{2}} + (Bi)^{\frac{1}{2}}(f/Uh)\cosh(Bi)^{\frac{1}{2}}}\tanh\left[(Bi)^{\frac{1}{2}}(1 + f/Uh)\right] - 1\right| \times 100\% \tag{13}$$

由式(13) 知 $e = f(Bi, f/Uh)$，即相对误差只是 Bi 数和截面侧面比的函数。在 Bi 数和截面侧面比的可能取值范围内 $Bi \in (0.000\,01, 10\,000)$，$f/Uh \in (0.000\,01, 1\,000)$(实际上 $f/Uh = 1\,000$ 已经非常大，这时就意味着一个大平板贴在基面上，很难看出是肋片)进行计算机详算，计算结果如图 1 ~ 图 3 所示。由图可见，在给定 Bi 数时，其相对误差随截面侧面比 $f/(Uh)$ 的增加而增加，而在 $f/(Uh)$ 一定时，随 Bi 数增加有极大值。表 1 和表 2 分别给出相对误差最大值 e_{max} 随 $f/(Uh)$ 和随 Bi 的变化规律的几组典型数值。由表 1 可以看出，对于不同的 $f/(Uh)$ 值，相对误差的最大值多处于 $Bi = 0 \sim 3$ 之间，且其对应的 Bi 随 $f/(Uh)$ 的增大而减小。相对误差的最大值随着 $f/(Uh)$ 的增大而增大。由图 2 和表 1 可见，当 $f/(Uh) < 0.5$(截面小于侧面一半)时，最大相对误差 $e < 1\%$，当 $f/(Uh) < 0.1$ 时，最大相对误差 $e < 0.02\%$。由表 2 可见，相对误差的最大值也随着 Bi 的减小而增大。当 $Bi > 7$ 时，最大相对误差 $e < 1\%$，当 $Bi > 10$ 时，最大相对误差 $e < 0.36\%$。由此可以推论，只要 $f/(Uh) < 0.5$，或 $Bi > 7$ 两个条件满足一个，即可以保证相对误差 $e < 1\%$。

表 1　e_{max} 随 $f/(Uh)$ 的变化规律

$\frac{f}{Uh}$	e_{max}/ %	Bi
0.001	0.000 000 02	2.22
0.005	0.000 003	2.21
0.01	0.000 022	2.18
0.05	0.002 432	2.03
0.1	0.017 049	1.87
0.5	0.940 057	1.16
1.0	3.657 749	0.8
5.0	27.718 012	0.22
10.0	43.393 524	0.11
13.0	49.106 209	0.08
15.0	52.088 638	0.07
20.0	57.737 355	0.05
30.0	64.803 397	0.03
50.0	72.356 951	0.02

表 2　e_{max} 随 Bi 的变化规律

Bi	e_{max}/ %	$\frac{f}{Uh}$
0.001	93.628 514	999.91
0.005	91.534 924	999.99
0.05	77.579 492	999.91
0.1	69.104 457	999.95
0.5	39.025 140	999.95
1.0	23.781 389	708.75
5.0	2.252 472	316.41
7.0	0.998 979	267.20
10.0	0.356 702	223.40
13.0	0.147 143	195.85
15.0	0.086 215	182.26
20.0	0.026 019	157.71
30.0	0.003 486	128.58
50.0	0.000 144	99.37

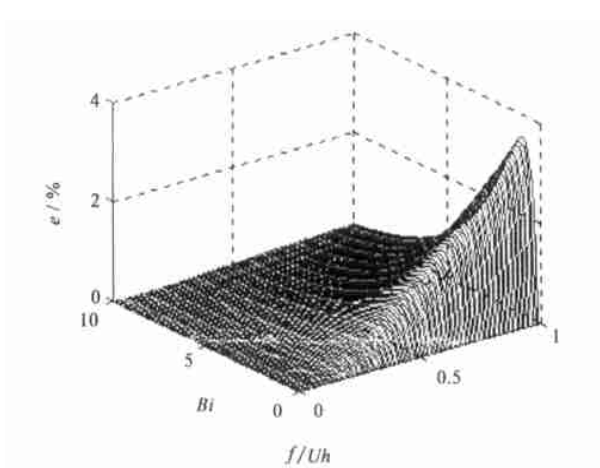

图 1　相对误差随 $f/(Uh)$ 和 Bi 的变化规律

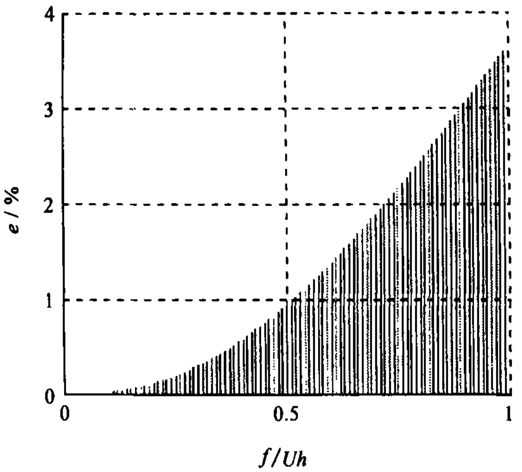

图 2 截侧面积比 $f/(Uh)$ 对相对误差的影响

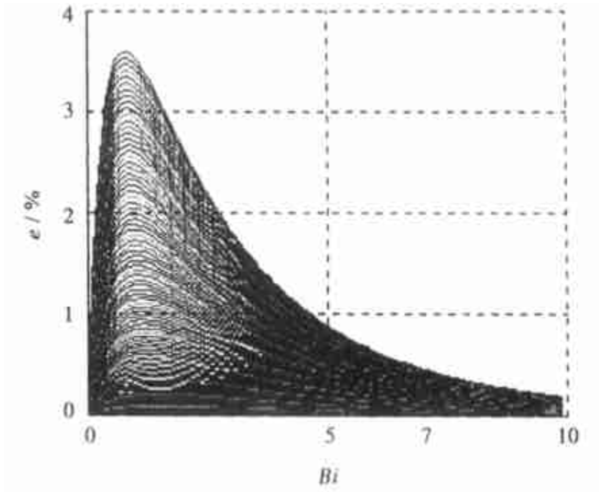

图 3 Bi 对相对误差的影响

4 结 论

利用假想肋高 $h' = h + f/U$ 代替实际肋高, 当等截面直肋的截侧面积比 $f/(Uh) < 0.5$, 或 $Bi > 7$ 两个条件满足一个, 用肋端绝热公式代替肋端对流换热公式计算肋的换热量的误差小于 1%。

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(何静芳 编辑)

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(3) 实验设计中准则数 7 可以按照单相流的方法进行考虑; 准则数 8 和 9 反映了传质潜热同气相内能间的关系, 考虑到饱和器研究的重点, 该准则数也是需要考虑的; 至于准则数 10 可以在实验设计中不予考虑; 准则数 11 ~13 可以根据实验的具体目的和研究的重点区域, 进行适当取舍。

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(何静芳 编辑)

non-dimensionalized treatment. On the basis of an invariance principle of differential equations similarity criteria were deduced, which the humidifier shall comply with during the experimental research. Some explanations are given concerning the role being played by these criteria during experiments. Moreover, some major issues requiring due attention during the tests of the humidifier are also presented. **Key words**: humid air turbine cycle, humidifier, heat and mass transfer, similarity analysis

等截面直肋传热简化计算的适用条件＝**Applicable Conditions for the Simplified Calculation of Heat Transfer for Straight Fins of Uniform Cross-section** [刊, 汉] / XU Zhi-ming, ZHOU Li-qun, BU Yu-bing, et al (Northeast Electric Power Institute, Jilin, China, Post Code: 132012) //Journal of Engineering for Thermal Energy & Power. -2004, 19(1). —42～44

With the help of a theoretical analysis method a fin-end adiabatic calculation formula is often used instead of a formula based on a fin-end convection heat exchange to calculate the heat transfer of straight fins of uniform cross-section. The approximate error thus obtained can be expressed as a function of the ratio of cross-sectional area to lateral area f/(Uh) and also as a function of Biot number Bi. Through a calculation of the possible range of selected values it has been found that when one of the following two conditions is met, the error of the above calculation method will be less than 1%. The two conditions are 1. The ratio of f/(Uh) of the straight fins is less than 0.5; 2. Number Bi is greater than 7. **Key words**: assumed fin height, Biot criteria, the ratio of cross-sectional area to lateral area

相变材料相变点温度热物性的测试及误差分析＝**Test Measurements and Error Analysis of Thermo-physical Properties of Phase-change Materials at a Phase-transition Point Temperature** [刊, 汉] / LI Chang-geng, ZHOU Jie-min (Institute of Physical Sciences under the Zhongnan University, Changsha, China, Post Code: 410083) //Journal of Engineering for Thermal Energy & Power. — 2004, 19(1). —45～47

The moving phase-interface curves during a solid-liquid phase-transition process are closely related with such a variety of two-phase thermo-physical properties as specific heat, density, thermal conductivity and phase-transition latent heat. The authors have come up with a method for determining several thermo-physical parameters, among others, the thermal conductivity of phase-change materials at a solid-liquid phase transition temperature. The above determination was carried out through the measurement of phase-interface moving rates. A test measurement device was designed and a quantitative analysis of measurement errors perfomed of the test measurement system. It was found that the error of the measurement system based on a combination of numerical calculations and experimental tests would not exceed 3%. The thermal conductivity and thermal diffusion factor of several kinds of materials were measured by using the above-mentioned test measurement system with satisfactory results being obtained. This shows that the measurement method proposed by the authors is trustworthy. **Key words**: phase change materials, thermo-physical properties, measurement, error analysis

圆管状内壁面管口辐射传递的方向分布特性＝**Direction Distribution Characteristics of Radiation Transmission from a Cylindrical Inner-wall Surface Tube-end**[刊, 汉] / LU Yi-ping, LI Bing-xi, YUAN Li-ming, et al (Institute of Energy Science and Engineering under the Harbin Institute of Technology, Harbin, China, Post Code: 150001) // Journal of Engineering for Thermal Energy & Power. — 2004, 19(1). —48～51

To obtain the direction distribution of radiation transmission through the tube end of a cylindrical inner-wall surface the authors have introduced a Monte Carlo method for solving the radiation transmission factor RD among cylindrical tube inner-wall surface elements. With the inner wall being an isothermal gray body, of a diffuse emission and diffuse reflection the impact was studied of the change of tube inner-wall emission rate, and of the ratio of tube length to radius on the equivalent directional emission rate of a tube-end surface. The study results indicate the following general tendency. With the increase in tube length-to-radius ratio the maximum value point of the equivalent directional emission rate of the tube-end surface will shift in the direction of a small-angled zenith angle. When the ratio of tube length to radius is relatively great, the tube inner-wall emission rate will decrease with an increase in tube length. With a relatively small tube outlet