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板式换热器单边流动与对角流动数值模拟

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摘 要:基于传热学控制方程,采用数值计算方法,对板式换热器单边流动和对角流动时的流动与换热特性进行分析。在分析过程中保持换热器的结构参数不变,只改变进出口的流动方式,结果发现:在相同的流速下,单边流动的总对流换热系数要高于对角流动,而总压降单边流动要低于对角流动,在流速 $u=0.6$ m/s 工况下,努谢尔数单边流动比对角流动高出 10.87%,压降对角流流动比单边流动高出 5.13%。随着进口流速的增大,单边流动与对角流动的冷热流体进出口温差均减小,而且减小的趋势对角流动要大于单边流动,摩擦因子 f 和传热因子 j 逐渐减小。单边流动的流动和传热特性要优于对角流动。

关 键 词:单边流动;对角流动;数值模拟;流动;换热

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引 言

板式换热器在化工、动力、食品、核能等领域得到了普遍的应用。由于换热器波纹板片交错叠置,其内部流体流动较为复杂,目前对其流动特性和传热性能的分析研究主要用实验和数值模拟两种方法。所做的研究主要体现在换热特性和压降两个方面,目的是提高换热效果和降低压降。

实验研究虽然能够直接得到换热器的整体性能[1~2],但要花费大量人力、物力,研究开发周期较长、费用高,也无法满足大型换热器、复杂物性等工况条件的要求。数值模拟方法作为一种高效、经济的研究手段,在板式换热器流动与传热研究中逐渐得到应用。阴继翔等人利用数值模拟方法探讨了上下波纹板交错角 θ 波纹相对节距 P/H(波长/波高)及雷诺数 Re 对流动与换热性能的影响[3]。崔立祺,曲宁和吴华新等人分别截取人字形波纹板式换热器50 mm×100 mm,50 mm×110 mm 和128 mm×128 mm 主流区域进行模拟计算分析[4~6],得到了波纹夹角,波高和波距对换热器性能的影响,并利用计算结果拟合各几何参数与努谢尔特数、压力降之间的关系曲线。仇嘉等人对在分配器作用下,单相流动和两相流动时[7],板式换热器并联流道间的流动分配性能进行分析发现分配器对板式换热器流道间流体分配的均匀性有很大的影响。Mehrabian 等人采用 CFD 研究板式换热器[8],截取最小的一个单元作为计算区域,认为波纹形状对换热和压降有十分重要的影响。Galeazzo 等人根据板片的完整结构建立了四个流道的三维计算模型[9],但把人字板波纹简化成平板,这样就不能模拟出流体在流道内的湍流情况。

由于上述所做的数值模拟研究的计算区域和换热器的实际结构存在一定的差别,未能完整地反映换热器整体的流动特性和换热性能,因此,本研究建立了包含了进出口分配区和波纹换热区在内的完整的冷热双流道换热几何模型,重点研究对角流动和单边流动换热器的流动和换热特性,得到相关准则关联式,为换热器的设计和安装提供理论依据。

1 研究对象

目前,流体在板式换热器板片上的流动方式有两种:单边流动和对角流动,如图1所示。冷热流体在各自的流道中流动,冷流体流经换热器波纹板片上侧两个角孔,左角孔进入,右角孔流出,热流体流经换热器波纹板片下侧两个角孔,右角孔进入,左角孔流出,这种流动方式称作单边流动。冷热流体在各自的流道中流动,冷流体由波纹板片上侧左角孔流入,沿着波纹板片对角线的方向由下侧右角孔流出;热流体由波纹板片下侧右角孔流入,沿着板片对角线的方向由下侧左角孔流出,这种流动方式称作对角流动。

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图 1 板式换热器波纹板片

Fig. 1 Corrugated plates for plate type heat exchangers

研究对象为 BR0.015F 型人字形板式换热器，换热器特征参数如表 1 所示；换热器流体流动通道结构图如图 2 所示。利用 Pro/e 3.0 软件构建的冷热双流道的几何模型，如图 3 所示。单边流动：热流体由 A 流入，由 B 流出，冷流体由 D 流入，由 C 流出；对角流动：热流体由 A 流入，由 C 流出，冷流体由 B 流入，由 C 流出。

表 1 BR0.015F 型板式换热器特征参数

Tab. 1 Characteristic parameters of a BR0.015F type plate type heat exchanger

特征参数	数值	特征参数	数值	特征参数	数值
外形尺寸/mm × mm	258 × 100	单片有效面积/m^2	0.015	试件片数	12
试件流程	(1 ×5) /(1 ×6)	试件传热面积/m^2	0.15	板片厚度/mm	0.6
板间距/mm	2	当量直径/mm	4	角孔直径/mm	20
波纹角度/(°)	120	波纹深度/mm	2	波纹法向节距/mm	6
波纹形状	人字形	板片材质	不锈钢 304	单流道截面积/m^2	0.000166

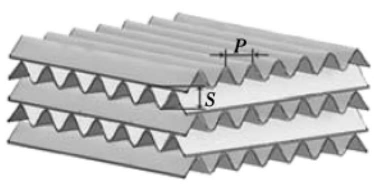

图 2 换热器流动通道结构图

Fig. 2 Structural drawing of the flow path of a heat exchanger

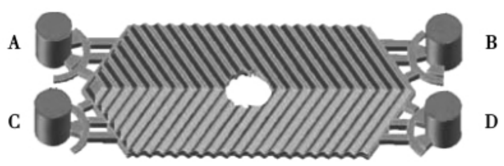

图 3 换热器冷热双流道模型

Fig. 3 Model for the cold-hot dual flow path of a heat exchanger

2 数学模型和边界条件

由于研究单相流体传热，且流道内的温差较小，特做如下假设：

(1) 流体的各物理量不随时间变化，为单相流体，定常流动；

(2) 流体为不可压缩的牛顿流体；

(3) 重力和浮升力的影响忽略不计；

(4) 忽略流体流动时的粘性耗散作用所产生的热效应。

对于单相不可压缩流体，换热器模型的控制方程表达式为[10]：

连续性方程：

$$\partial u_j / \partial x_j = 0 \tag{1}$$

动量方程：

$$\frac{\partial u_i}{\partial t} + \frac{\partial u_i u_j}{\partial x_j} = -\frac{1}{\rho}\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left(\mu \frac{\partial u_i}{\partial x_j}\right) \tag{2}$$

能量守恒方程：

$$\frac{\partial T}{\partial t} + \frac{\partial u_j T}{\partial x_j} = \frac{\partial}{\partial x_j}\left(\frac{\mu}{Pr}\frac{\partial T}{\partial x_j}\right) \tag{3}$$

$RNG\ k-\varepsilon$ 模型：

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_j}\left(p_{r_k}\mu_{eff}\frac{\partial k}{\partial x_j}\right) + G_k - \rho\varepsilon \tag{4}$$

$$\frac{\partial(\rho\varepsilon)}{\partial t} + \frac{\partial(\rho\varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_j}\left(p_{r_\varepsilon}\mu_{eff}\frac{\partial \varepsilon}{\partial x_j}\right) + \frac{C^*_{1\varepsilon}}{k}G_k - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} \tag{5}$$

式中：u_i—i 方向上的速度分量；μ—湍动粘度，G_k—由于平均速度梯度引起的湍动能 k 的产生项，$C_{1\varepsilon}$、$C_{2\varepsilon}$经验常数；p_{r_k}、p_{r_ε}—湍动能 k 和耗散率 ε 对应的 Prandtl 数；$C_{1\varepsilon}=1.42$，$C_{2\varepsilon}=1.68$；$C_\mu=0.0845$。

边界条件：

(1) 进出口边界条件

进口采用速度入口条件，进口温度采用实验测得数据，速度由流量计算得到；出口采用压力出口条件，具体值由实验测量得到。

(2) 壁面条件

外部边界为无滑移速度边界条件,冷热流道相接触的面设为换热面,其余各面设为绝热边界条件。换热面和其它壁面的物理参数如下: 板片的材料为 304 不锈钢[11], $c = 6.44 \times 10^2$ J/(kg·K), $\rho = 7600$ kg/m^3, $\lambda = 644$ W/(m·K)。

通过 Pro/e 3.0 软件进行模型建立,运用 Gambit 软件,采用非结构华四面体网格单元进行网格划分。。由于换热器内部结构复杂,不同流动区域选择不同的网格步长。先将模型按照进出口,分流区和波纹换热区分割成 10 部分,各自填充网格。梯次加密网格,波纹换热区域网格步长达到 0.5 mm(网格数在 140 万)时,平均 Nu 数开始稳定,不再发生变化,此时网格密度已经足够满足模拟精度的要求。

板间流体为单相流动,流动为湍流,采用 $k-\varepsilon$ RNG 湍流模型,保持模型中的各参数值不变。计算采用分离变量隐式法求解,速度和压力耦合采用 SIMPLE 方法[10],二阶精度的迎风格式离散。

3　模拟结果分析

为使流体在板间流动时,处于充分的湍流状态[12],研究了板间流速在 0.2 ~ 0.6 m/s 工况下单边流动与对角流动的流动与换热过程,在模拟计算过程中热流道进口温度为 51 ℃,冷流道进口温度为 35 ℃。

对于人字形板式换热器,冷热流道由相同的板片组装而成,结构相同,而且是在相同的流速下,则冷热流体对流换热系数可以用相似准则方程式为:

$$Nu = C(Re)^m(Pr)^n \tag{6}$$

式中: C、m 与板片、流体和流动类型有关。流体被加热, n 取 0.4; 流体被冷却, n 取 0.3。

努塞尔数 $Nu = hd_e/\lambda_w$; 雷诺数 $Re = ud_e/\nu$; 当量直径 $d_e = \dfrac{4bs}{2(b+s)}$; 普朗特数 $Pr = \nu/\alpha$; b—板片有效宽度,m; s—板间距,m; λ_w—波纹板片导热系数, W/(m·K); u—流道间流速,m/s; ν—流体的运动粘度,m^2/s; α—流体的热扩散率,m^2/s。

分别用下角标 1 和 2 表示热流体和冷流体,两流体的对流换热系数可以进一步表示为:

$$h_1 = CRe_1^m Pr_1^{0.3}(\lambda_1/d_e) \tag{7}$$

$$h_2 = CRe_2^m Pr_2^{0.4}(\lambda_2/d_e) \tag{8}$$

传热热阻为:

$$k = (1/h_1 + \delta/\lambda_w + 1/h_2)^{-1} \tag{9}$$

由于模拟计算过程中冷热流道是在等流速条件下进行的,即 $u_1 = u_1 = u$,则根据式(7)、式(8)和式(9)可得:

$$Cu^m = \left[1/\left(\frac{1}{k}-\frac{\delta}{\lambda_w}\right)\right]\left[1/\left[\left(\frac{d_e}{\nu_1}\right)^m\left(\frac{\lambda_1}{d_e}\right)Pr_1^{\ 0.3}\right]+\left[1/\left(\frac{d_e}{\nu_2}\right)^m\left(\frac{\lambda_2}{d_e}\right)Pr_2^{\ 0.4}\right]\right] \tag{10}$$

令式(10)右边以 P 表示,等号两边取对数,可得:

$$\ln P = \ln C + m\ln u \tag{11}$$

模拟计算过程中,冷热流体按等流速计算,冷热流体温差较小,流体的运动粘度变化较小,而且流道结构参数相同,两侧流体的雷诺数相对误差 $|\eta'| \leqslant 9\%$,按 $Re_1 \approx Re_2 \approx Re$ 对数据进行处理,线性回归,可得到 C 和 m 的数值,进而得到冷热流体对流换热系数准则方程:

单边流动: $Nu = 0.066Re^{0.9491}Pr^n$

当 0.2 m/s $\leqslant u \leqslant 0.6$ m/s 时,准则计算值和实验结果相关性为 99.21%

对角流动: $Nu = 0.069Re^{0.9567}Pr^n$

当 0.2 m/s $\leqslant u \leqslant 0.6$ m/s 时,准则计算值和实验结果相关性为 98.54%

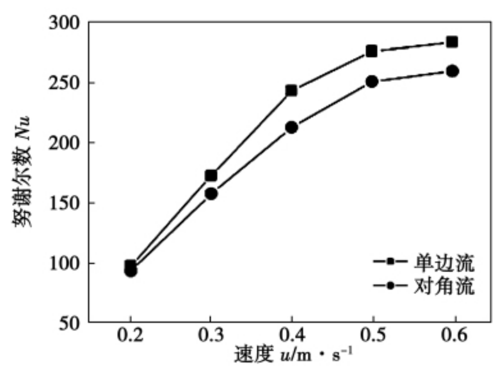

图 4　努谢尔数与流速变化关系

Fig. 4 Variation relationship between the Nusselt number and flow speeds

图 4 和图 5 分别给出 Nu 数和压降 ΔP 随流速的变化关系。由图看出,液体的流速对换热器的性能影响较大,随热液体流速的增大,努谢尔特数和换热器冷热流体通道的压力降均增大,但是努谢尔特数增大趋势随着流速增大逐渐变缓,而压降将增大的趋势加剧。在相同的流速下,单边流动的努谢尔特数要高于对角流动,单边流动传热效果要优于对角流动,但是压力降单边流动低于对角流动。在流速 $u = 0.6$ m/s 工况下,努谢尔特数单边流动比对角

流动高出 10.87%，压降对角流流动比单边流动高出 5.13%。

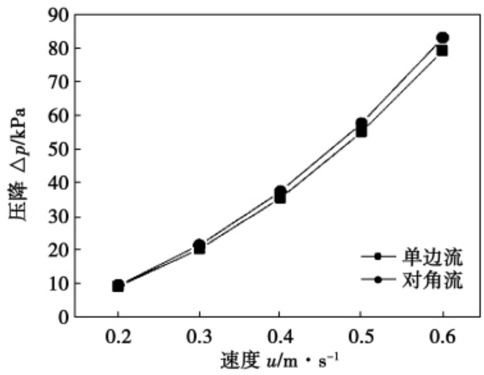

图 5　冷流道压降与流速变化关系

Fig. 5 Variation relationship between the pressure drop of the cold flow path and the flow speed

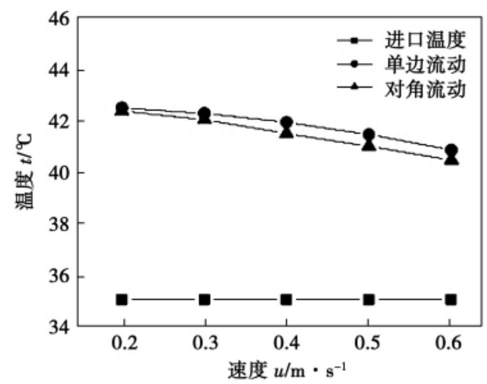

图 6　冷流道出口温度与流速的变化关系

Fig. 6 Variation relationship between the temperature at the outlet of the cold flow path and the flow speed

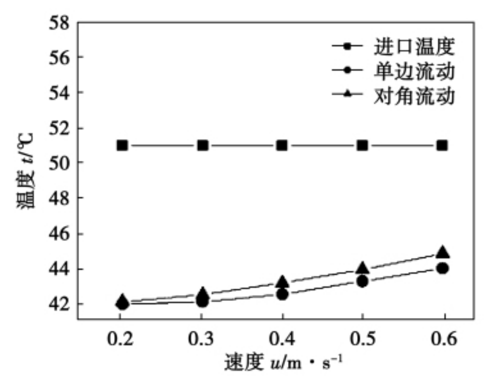

图 7　热流道出口温度与流速的变化关系

Fig. 7 Variation relationship between the temperature at the outlet of the hot flow path and the flow speed

图 6 和图 7 给出单边流动与对角流动冷热流体通道出口温度与流道内流体流速的变化关系。从图上，可以看出在相同的流速下，冷热流体的进出口温差，单边流动要大于对角流动；随着流速的增加，单边流动与对角流动的冷热流体进出口温差均减小，而且减小的趋势对角流动要大于单边流动。

目前，对换热器的研究主要体现在压降和换热特性两个方面，本研究计算了不同流动速度工况下的摩擦系数和传热因子。板式换热器的摩擦因子 $f = \dfrac{2\Delta p d_e}{4\rho \Delta L \bar{u}^2}$，（$L$ 为流道长度）和传热因子 $j = \dfrac{Nu}{Re\, Pr^{\frac{1}{3}}}$ 的具体数值如表 2 所示，随着进口流速的增大，f 和 j 逐渐减小。流体的传热量单边流动大于对角流动，随着流速的增加，增大的趋势逐渐变大。

表 2　单边流动与对角流动的比较

Tab. 2 Comparison of the single-side flow and the diagonal one

流速/m · s⁻¹	单边流动摩擦系数 f	对角流动摩擦系数 f	单边流动传热因子 j	对角流动传热因子 j	单边流动传热量 Φ/W	对角流动传热量 Φ/W	差值/%
0.2	1.7645	1.7952	0.0485	0.0464	6193.54	6098.77	1.55
0.3	1.7585	1.7742	0.0478	0.0458	9061.56	8886.85	1.97
0.4	1.7411	1.7581	0.0471	0.0451	11796.47	11539.21	2.23
0.5	1.7352	1.7493	0.0465	0.0441	13235.52	12606.4	4.99
0.6	1.7199	1.7321	0.0458	0.0431	14458.07	13645.22	5.96

4　计算方法验证

对 BR0.015F 型人字形板式换热器进行实验研究的原理和方案等见文献[13]。试验中，板片之间按照单流程，单边流动，冷热流体交替布置，板片之间实际流动过程如图 8 所示。在实验过程中，采用便于控制的等流速法[14]，即保持冷热流体进出口的流速相等。模拟过程中，根据温度的数值来确定流体的物性参数，使数值模拟的结果能够准确的反映人字形板式换热器内部的真实流动和换热特性。实验和模拟计算结果对比分析如表 3 所示，其中每一个工况，上侧为热流体通道，下侧为冷流体通道。进出口温差和压降的计算值与实验所得值误差均小于 6%，能够比较准确地反应换热器流道内部流体的流动与换热特性。

在研究过程中虽然按照波纹板式换热器的实际结构参数建立了冷热双流道流体流动与换热物理模型，但是与实验结果还存在约 45% 的误差。误差来

源主要来自两个方面: 一是实验过程中数据采集所用仪器带来的,这是不可避免的; 一个是在数值模拟计算过程中,建立物理模型时,板式换热器分流区域由于制图软件的限制与实际换热器的结构存在一定差别,导致流体流动阻力变化,影响传热效果。

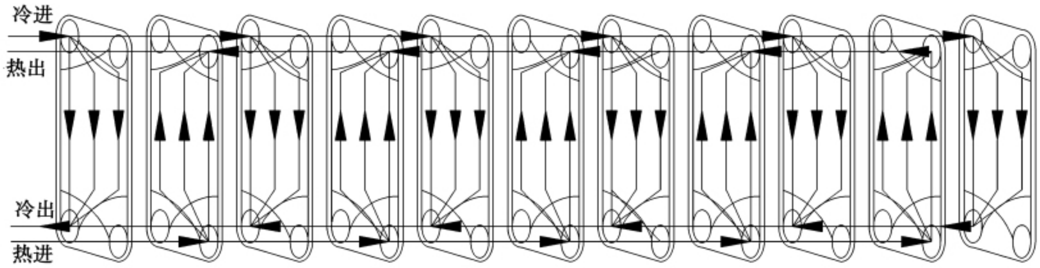

图 8　人字形板式换热器板间实际流程图

Fig. 8 Chart showing the actual flow path between the plates in a herringbone plate type heat exchanger

表 3　实验数据与模拟数据

Tab. 3 Test data and simulation ones

进口流量/ ($m^3 \cdot h^{-1}$)	实验进口温度/℃	实验出口温度/℃	模拟出口温度/℃	误差/%	实验测量压降/Pa	模拟计算压降/Pa	误差/%	实验传热系数/$W \cdot m^{-2} \cdot K^{-1}$	模拟传热系数/$W \cdot m^{-2} \cdot K^{-1}$	误差/%
0.308	51.02	41.16	41.34	-1.83	2320	2392.16	3.11	3494.70	3323.57	-4.90
0.371	35.00	43.28	43.07	-2.57	2570	2487.58	-3.74			
0.266	51.12	40.84	40.96	-1.17	1730	1809.59	4.60	3239.57	3129.69	-3.39
0.321	34.93	43.53	43.38	-1.74	1960	1890.90	-3.53			
0.235	51.14	40.72	40.80	-0.77	1350	1425.88	5.6	3064.23	2928.46	-4.43
0.281	35.08	43.95	43.68	-3.04	1540	1476.48	-4.12			
0.201	51.26	40.52	40.63	-0.99	1010	1068.34	5.78	2756.81	2631.89	-4.53
0.242	35.13	44.12	43.87	-2.78	1190	1121.77	-5.73			
0.168	51.15	40.29	40.41	-1.11	720	760.11	5.57	2450.38	2319.26	-5.35
0.202	35.03	44.36	44.06	-3.18	850	803.41	-5.48			

5　结　论

采用数值模拟计算方法,对 BR0.015F 型人字形板式换热器单边流动和对角流动时的流动和换热性能分别进行研究,得到结论:

(1) 在相同的流速下,单边流动的总对流换热系数要高于对角流动,总压降单边流动要低于对角流动;

(2) 随着进口流速的增大,f 和 j 逐渐减小;

(3) 冷热流体(单相流体) 的进出口温差,单边流动高于对角流动;

(4) 随着流速的增加单边流动与对角流动的冷热流体进出口温差均减小,而且减小的趋势对角流动要大于单边流动;

(5) 单边流动的流动和传热特性要优于对角流动。

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(丛 敏 编辑)

新技术、新产品

新型燃气发动机

往复式燃气发动机能达到1∶1功热比,而燃气轮机可以达到1∶1.5到1∶2功热比,并且具有更高的高温热量利用率,使其成为生产蒸汽和的理想机型,并用于联合循环。对于简单循环,燃气发动机具有更高的发电效率,平均高于46%。从排放的观点出发,燃用天然气时,燃气发动机二氧化碳排放低,大约0.202 t/(MWh),至于燃气轮机,在不采用后处理,如选择性催化还原和注水或蒸汽,可以达到更低的氮氧化物和一氧化碳排放。

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better reaction characteristics and sustainable circulating capacity than the Co-base oxygen carrier. **Key words**: chemical chain combustion, CO_2 separation, metallic oxygen carrier

湍流射流冲击移动平板的流动和传热分析 = **Analysis of the Flow and Heat Transfer on a Moving Flat Plate Impinged by a Turbulent Jet Flow** [刊,汉] YE Chun-jie, PAN Hong-liang (College of Mechanical and Power Engineering, East China University of Science and Technology, Shanghai, China, Post Code: 200237) // Journal of Engineering for Thermal Energy & Power. - 2011, 26(6). -669 ~ 674

By using the Reynaulds stress turbulent flow model, numerically analyzed was a moving flat plate impinged by a semi-closed turbulent jet flow with the configuration of its flow and temperature field under various speeds of the plate as well as the near-wall-surface turbulent flow intensity and the curves showing the Nusselt number distribution on partial plate surface being obtained. The analytic results show that to increase the speed of the moving plate will invariably result in an asymmetry of both flow and temperature field relative to the jet flow center and form a secondary vortex zone at a side of the flow field. The value of the turbulent flow intensity on the plate surface will increase while the peak value of the local Nusselt number in the zone under impingement will decrease with an increase of the speed of the plate in motion. When the speed of the plate is higher than the inlet jet flow speed, the plate surface average Nusselt number will gradually increase with an increase of the speed of the plate. When the speed of the plate increases to a value two times higher than the inlet jet flow speed, the plate surface turbulent flow intensity at the impingement point will increase by about 40% and the peak value of the Nusselt number will decrease about 60% but the plate surface average Nusselt number will increase by above 30%. The research findings can offer important guidance for the intensified heat and mass transfer during a continuous operation. **Key words**: jet flow, moving flat plate, turbulent flow intensity, heat transfer

板式换热器单边流动与对角流动数值模拟 = **Numerical Simulation of the Single-side and Diagonal Flow of a Plate Type Heat Exchanger** [刊,汉] XU Zhi-ming, WANG Yue-ming, ZHANG Zhong-bin (College of Energy Source and Power Engineering, Northeast University of Electric Power, Jilin, China, Post Code: 132012) // Journal of Engineering for Thermal Energy & Power. - 2011, 26(6). -675 ~ 680

Based on the control equation in heat transfer and by using the numerical calculation method, analyzed were the flow and heat exchange characteristics of a plate type heat exchanger when a single-side and diagonal flow was adopted. In the process of the analysis, the structural parameters of the heat exchanger were kept unchanged, only the flow mode at the inlet and outlet was changed. It has been found that at a same flow speed, the total convection heat exchange coefficient of the single-side flow is higher than that of the diagonal flow while the total pressure drop of the single-side flow is lower than that of the diagonal flow. Under the condition of the flow speed $u = 0.6$ m/s, the Nusselt number of the single-side flow is 10.87% higher than that of the diagonal flow and the pressure drop of the diagonal flow is 5.13% higher than that of the single-side flow. With an increase of the inlet flow speed, the

inlet and outlet temperature differences of the cold and hot fluid in both single-side and diagonal flow will invariably decrease. Furthermore, such a decreasing tendency of the diagonal flow is bigger than that of the single-side flow while the friction factor f and heat transfer factor j will gradually decrease. The flow and heat transfer characteristics of a single-side flow is superior to that of a diagonal flow. **Key words**: single-side flow, diagonal flow, numerical simulation, flow, heat exchange

管程组合转子传热及阻力特性的实验研究 = **Experimental Study of the Heat Transfer and Resistance Characteristics of a Tube-side Combined Rotor** [刊,汉] YANG Wei-min, HAN Chong-gang, ZHANG Zhen (Beijing University of Chemical Technology, Beijing, China, Post Code: 100029), LI Feng-xiang (Patent Bureau, State Intellectural Property Office of PRC, China, Post Code: 100191) //Journal of Engineering for Thermal Energy & Power. - 2011, 26(6). -681 ~ 686

Tube-side combined rotor intensification technology has its functions to intensify heat transfer and self clean. With the help of the test means and oil and water serving as the working medium respectively in the tube side, studied were the heat transfer and resistance characteristics of a tube-side combined rotor. In the meantime, the influence of the self structural parameters of the rotor under discussion on its heat transfer and resistance characteristics was compared and analyzed. The analytic results show that in the laminar flow and transitional zone where $Re = 500-8000$, to increase the spiral angle of the rotor can remarkably intensify the heat transfer. However, in the torrent turbulent flow zone where $Re = 10^4 \sim 10^5$, to increase the outer diameter of the rotor can obviously intensify the heat transfer, thus, laying a foundation for application of rotor combination type heat transfer intensification devices in various industries. **Key words**: tube-side combined rotor, intensified heat transfer, resistance, structural parameter

单回路紫铜—水脉动热管传热性能的实验研究 = **Experimental Study of the Heat Transfer Performance of a Single-loop Copper-water Pulsation Heat Pipe** [刊,汉] SU Lei, ZHANG Hong, DING Lei-jiang, et al (College of Energy Source, Nanjing Institute of Technology, Nanjing, China, Post Code: 210009) // Journal of Engineering for Thermal Energy & Power. -2011, 26(6). -687 ~ 693

Experimentally studied was the law governing the influence of four factors, namely, cooling water flow rate, inclination angle, tube diameter and liquid filling rate, on the heat transfer performance of a single-loop copper-water pulsation heat pipe, including its tube wall temperature at measuring points, mean temperature in the cold and hot section, heat transfer temperature difference, heat transfer resistance and temperature fluctuation. As a result, some measures for enhancing the heat transfer performance were obtained. The research results show that the single-loop pulsation heat pipe arranged horizontally can not be started up with any possible methods and at an inclination angle of over 30 degrees, oscillation may be produced inside the pipe and to increase the inclination angle can lower the heat transfer resistance. At a given thermal power, there exists an optimum value of the cooling water flow rate and