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基于场协同理论的重力式热管新设计

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摘 要: 为了优化重力式热管内流场和温度场的协同性、改善其内部传热特性,基于场协同理论,根据重力式热管内部的实际几何尺寸及场协同角度等因素,在重力式热管内部设置一种带有锥度的螺旋翅片扰流结构。建立了带有内螺旋翅片的热管模型,圆管长度 $L_{model}=500$ mm,管径 $D=20$ mm,内螺旋翅片长度 $L_2=100$ mm,装配尺寸 $L_1=200$ mm,管内工质为水,通过计算确定螺旋翅片的螺旋锥角为12.33°,螺旋升角为25.91°。利用流体力学软件对管内的传热与阻力特性进行的数值模拟,结果表明:重力式热管径向传热性能得到改善,在 $Re=1\ 800$ 时,热通量增加18.7%,沿程阻力损失 h_f 增加24.88倍,阻力系数 f 增加23.33倍。

关 键 词: 重力式热管;场协同;内螺旋翅片;产品设计;模型简化;计算流体力学

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符号说明:

Nu—努谢尔特数
Re—雷诺数
Pr—普朗特常量
u—流动速度/m·s^{-1}
F_c—场协同数
W_e—韦伯数
ρ—密度/kg·m^{-3}
T_{wa}—热管管内径向截面中心温度/K
T_{wi}—热管绝热段壁温/K
T_{wc}—热管蒸发段壁温/K
h_e—换热系数/W·(m^2·K)$^{-1}$
β—场协同角/rad
α—管内温度梯度与径向截面的夹角/rad
Ψ—螺旋锥角/rad
φ—螺旋升角/rad
σ—液体表面张力,10^{-3} N/m
z—与汽液交界面几何形状有关的定性尺寸/m
l—传热面的几何特征长度/m
λ—工质导热系数,W/(m·K)
$\triangle T_m$—热管管内对数平均温度
C—修正系数
H—原换热管蒸发段长度/m
q—单根热管的热通量/W
A—换热面积/m^2
P_{sat}—饱和蒸汽压/Pa
P_a—大气压力/Pa
q_e—蒸汽段热流密度/W·m^{-2}
h_{fg}—汽化潜热/J·kg^{-1}
μ_l—液体黏度/Pa·S
c_{pl}—液体定压比热 J·(kg·K)$^{-1}$
λ_l—液体导热系数 W·(m·K)$^{-1}$
μ_v—蒸汽黏度/Pa·s
T_v—蒸汽温度/K
L_a—热管总长/m
下角标
z—轴向
x,y—径向
e—蒸发段
a—全部,总和
v—蒸汽

引 言

热管以其优良的轴向导热特性被广泛地应用于宇航、军工以及电子等能源行业,随着能源行业的发展,能量的高效利用成为世界关注的焦点。热管换热器的传热强化作为学界研究的热点,数十年来的科学研究,使热管得到长足的发展。目前,对热管内部传热性能的研究上,一部分文献对热管传热特性的数值模拟分析过程中,发现热管在沿其轴向传热性能优异[1-4],这一观点在基于场协同理论的文献[5-18]中得到进一步论证;热管内部径向的热阻对传热特性影响较轴向大,对于如何改善其内部的径向传热特性鲜见报道。

为了优化重力式热管内部流场和温度场的协同性,本研究通过数值模拟法研究重力式热管径向热阻对传热性能的影响,运用场协同理论对建立的热管模型进行数值分析,根据重力式热管内部的实际几何尺寸和场协同角,通过设置一种带有一定锥度的螺旋上升翅片的扰流件,改变传统重力式热管的

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内部蒸汽流动方向，优化重力式热管传热特性，借助计算机辅助设计，采用智能化设计方法，提出了一种新式的重力式热管内部结构，该设计已经申报国家发明专利[19]。

1　现有重力式热管内部传热性能

1.1　场协同理论

场协同理论的层流对流换热模型为：

$$Nu_x = Re \cdot Pr \cdot \int_0^1 (\vec{u} \cdot \nabla \vec{T}) \cdot \mathrm{d}y \tag{1}$$

或，

$$Nu_x = Re \cdot Pr \cdot \int_0^1 (|\vec{u}| \cdot |\nabla \vec{T}| \cdot \cos\beta) \cdot \mathrm{d}y \tag{2}$$

由式(2)可以得到改善传热的 3 个途径：(1)提高雷诺数，表征流体流动特性和流体粘性影响的相似准则数；(2)提高普朗特数，普朗特数由参与换热介质的物理性质决定；(3)增加积分项的数值，可以通过改变热边界层厚度内的热源强度以及改变场协同角 β 来实现。清华大学过增元等人为此定义了一个场协同数[20]：

$$Fc = \int \bar{\vec{U}} \cdot \nabla \bar{T} \mathrm{d}\bar{y} = \frac{Nu}{Re \cdot Pr} \tag{3}$$

无论是在层流、湍流，稳态还是非稳态换热过程中，描述换热强度通常采用兰州理工大学卢小平等人提出的场协同散度方程[21]。

$$\nabla \vec{J} = \nabla \vec{J}_d + \rho \cdot C_p \cdot |\vec{u}| \cdot |\nabla \vec{T}| \cdot \cos\beta \tag{4}$$

1.2　重力式热管的携带极限

判断出现携带传热极限的准则是 weber 数等于 1。weber 数是蒸汽流动的惯性力与吸液芯表面液体的表面张力之比：

$$W_e = \frac{\rho_v w_v^2 z}{\sigma} = 1 \tag{5}$$

提高携带极限的途径之一就是降低定型尺寸 z，在重力式热管中，从冷凝段回流的液体直接与蒸发段高温壁面接触，更容易使得冷凝回流液被蒸汽夹带进反向蒸汽流而到达冷凝段。

1.3　重力式热管传热性能分析

重力式热管内部温度梯度 T 如图 1 所示，传统热管蒸汽速度 U' 与温度梯度 T 夹角为 θ，温度场与速度场的协同性能需要改善。

因此，新式热管设计一个带有锥度的内螺旋翅片作为扰流件，使蒸汽速度方向 U' 调整为速度方向 U。由式(3)可以得到热管蒸发段吸收的热量 Q 与场协同角度 θ 的关系式式(6)，x、y 为实验系数，根据场协同数定义为 1。

$$Q = c\frac{\lambda}{l}|\vec{U}||\vec{T}|\cos\beta \cdot Re^x Pr^y \cdot A \cdot \Delta T_m \tag{6}$$

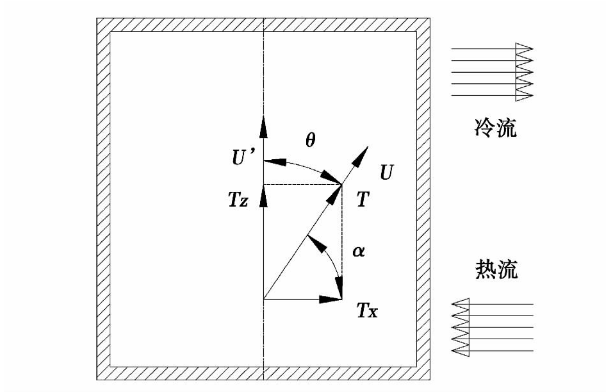

图 1　热管内速度和温度平面矢量

Fig. 1 Plane vector of the speed and temperature inside the heat pipe

在笛卡尔坐标系下，重力式热管内部温度梯度 T 如图 2 所示，得到张量式：

$$Q_i = c\frac{\lambda}{l}|\vec{U}_i||\vec{T}_i|\cos\beta_i \cdot Re^x Pr^y \cdot A \cdot \Delta T_m \tag{7}$$

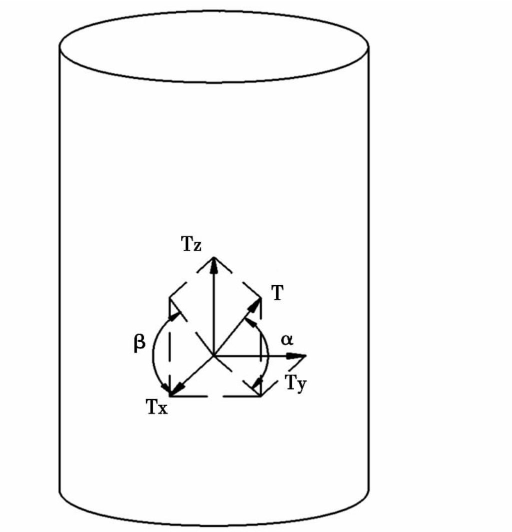

图 2　热管内速度和温度三维矢量

Fig. 2 3 – D vector of the speed and temperature inside the heat pipe

(1)现有热管管内径向蒸汽速度为零，$U_x = U_y = 0$，$\cos\theta_z = 1$，得到现有热管蒸发段吸热量：

$$Q = Q_z = c\frac{\lambda}{l}|\vec{U}_z||\vec{T}_z|Re^x Pr^y \cdot A \cdot \Delta T_m \tag{8}$$

(2) 新式热管内螺旋翅片试着改变管中蒸汽流向，使得 $\partial Q/\partial\theta = 0$，此时传热量 Q 达到最大，且 $\theta = 0$，即流场与热场很好地协同。该新式热管结构中蒸发段的吸热量为:

$$Q = |Q_x| + |Q_y| + |Q_z| \tag{9}$$

热管热通量在场协同角 θ 最小时，热通量最大; 如图 3 所示，若使新式内螺旋翅片的螺旋升角 $\varphi = \alpha$，螺旋锥角 $\Psi + \beta = 90°$时，可以使得这个角度最小，此时换热量得以强化。

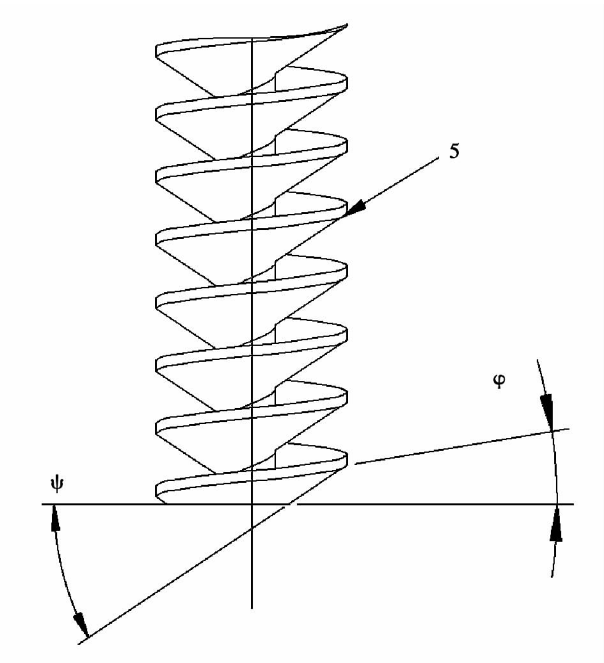

图 3　内螺旋翅片升角和锥角

Fig. 3 Lead angle and cone angle of the inner helical fins

$$\alpha = \mathrm{arctg}(T_z/\sqrt{T_y^2 + T_x^2}) \tag{10}$$

$$\beta = 90^{\circ} - \mathrm{arctg}(T_z/T_x) \tag{11}$$

根据式(10) 和式(11)，在确定轴向温度梯度与径向温度梯度的条件下，按式(12) 和式(13) 确定新式热管内螺旋翅片的螺旋升角、螺旋锥角。为保证扰流效果，内螺旋翅片高度与热管内径相等，翅片间距为一定值。按照场协同理论增加的内螺旋翅片对于热管蒸发段具有强化换热作用，提高了热管蒸发段的热流密度，新型内螺旋翅片的长度数值由式(14) 给定，在原热管冷凝段长度不变前提下，调整原热管蒸发段长度为式(14) 中 L，使蒸发段和冷凝段热流密度相当。

$$\varphi = \mathrm{arctg}(T_z/\sqrt{T_y^2 + T_x^2}) \tag{12}$$

$$\psi = \mathrm{arctg}(T_z/T_x) \tag{13}$$

$$L = H \cdot \sin\{\mathrm{arctg}[T_z/(\sqrt{2}T_x)]\} \tag{14}$$

1.4　带锥度的内螺旋扰流件的锥角和升角

在热管内径确定条件下，根据加入的介质性质及其相应参数，采用以下两种方法均可确定内螺旋翅片的螺旋升角 φ、螺旋锥角 ψ 和内螺旋翅片的长度 L，确定热管的结构。

1.4.1　实验室测定

温度参数 T_z、T_y、T_x 以《GBT 14812 - 2008 热管传热性能试验方法》[22] 中的温度测量方式进行测量，标准所列出的测量点缺少对热管内工质中心处温度的测量，为获取轴向温度梯度，除按标准设置蒸发段测量点之外，另布置两处测量点，一处测量蒸发段底部液相工质径向截面中心处温度 T_{wi}，另一处测量绝热段汽相工质径向截面中心温度 T_{wa}，则最终轴向温度为 $T_z = |T_{wa} - T_{wi}|$，径向温度为 $T_x = |T_{we} - T_{wi}|$。

通过测定的管内温度梯度，利用式(12)、式(13)、式(14) 可确定螺旋角 φ、螺旋锥角 ψ 和内螺旋翅片的长度 L，确定热管的结构。

1.4.2　理论计算

径向温度梯度 T_x 通过热阻 R_x 确定:

$$R_x = T_x/q \tag{15}$$

径向热阻 R_x，由式(16) 确定:

$$R_x = \frac{1}{h_e \cdot \pi \cdot d_i \cdot l_e} \tag{16}$$

对于热管蒸发段换热系数 h_e，采用 Imura[23] 建议公式计算:

$$h_e = 0.32 \times \left(\frac{\rho_l^{0.65} \cdot \lambda_l^{0.3} \cdot c_{pl}^{0.7} \cdot g \cdot q_e^{0.4}}{\rho_v^{0.25} \cdot h_{fg}^{0.4} \cdot \mu_l^{0.1}}\right)\left(\frac{P_{sat}}{P_a}\right)^{0.3} \tag{17}$$

对于管内蒸汽流动引起的轴向热阻 R_z，由式[18] 确定[24]:

$$R_z = \frac{128 \cdot L_a \cdot \mu_v \cdot T_v}{\pi \cdot d_i^4 \cdot \rho_v{}^2 \cdot h_{fg}{}^2} \tag{18}$$

在原热管结构参数已知、热管内工质确定的条件下，由式(16)、式(18) 得到轴向和径向温度比值。比值带入式(12)、式(13) 和式(14) 确定新型结构中的螺旋升角、螺旋锥角和内螺旋翅片结构长度。

2　新式热管的结构设计

现有的热管散热翅片结构仅仅是为了弥补两相传热系数相差较大的缺陷而设计，且此类翅片结构一般设计在热管外侧，并未达到场协同强化换热的

效果，因此新式热管内螺旋翅片结构按照场协同理论做如下的结构设计：

（1）整体结构为带有角度的等螺旋翅片；（2）翅片螺旋角度的给定；（3）翅片为冷凝回流液提供了低温回流通道，可以有效减少蒸汽夹带；

以气－气式热管换热器中的热管为例，其结构设计如图 4－图 6 所示。

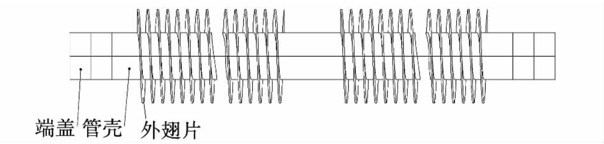

图 4　气—气换热器热管

Fig. 4 Gas－gas heat exchanger heat pipe

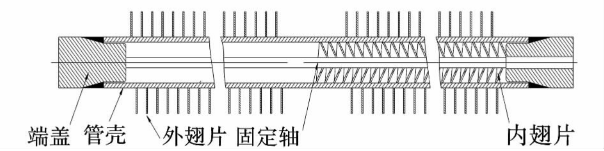

图 5　具有内螺旋翅片的热管剖视图

Fig. 5 Sectional view of a heat pipe with inner helical fins

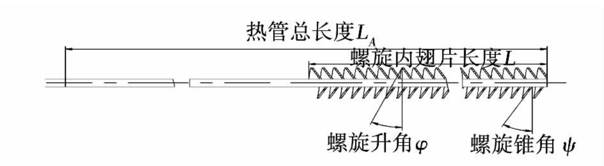

图 6　内螺旋翅片剖视图

Fig. 6 Sectional view of inner helical fins

3　数值计算分析

为了对前述改进后的模型进行计算验证，内螺旋扰流件改变上升流速方向，热管冷凝回流液沿着翅片上侧向下行流动，回流液对上升液体的剪切力较现有热管小，强化管携带极限较高，不考虑强化管回流影响，为了简化问题，建立如图 7 所示的内螺旋翅片模型，圆管长度 $L_{model}=500$ mm，管径 $D=20$ mm，内螺旋翅片的长度 $L_2=100$ mm，装配尺寸 $L_1=200$ mm。热管管内工质为水，最终确定螺旋翅片的锥角为 $\psi=12.33°$。

强化管增加了流体换热时间，因此对式（18）中的 L_a 除以一个矫正系数 $\sin\varphi$，按本模型得到升角和锥角的关系式：

$$\sin\varphi\cdot \mathrm{tg}\psi=0.147/\sqrt{2} \tag{19}$$

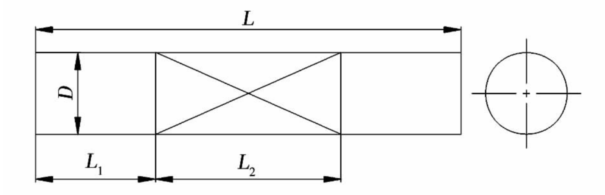

图 7　简化模型示意图

Fig. 7 Schematic diagram of the simplified model

为保证设置扰流片前后管内上升蒸汽体积流量不变，建立一个热管内部蒸汽脱离守恒方程，体积流量等于流道截面 S 与流速 v 的乘积，强化管流道截面如图 8 所示，脱离守恒等式式（20）左侧为光管体积流量，右侧为强化管体积流量。

$$\pi d_i^2 v/4=\pi d_i r_i v\mathrm{tg}\lambda/\sin\varphi \tag{20}$$

对式（20）简化后得到式（21）：

$$\sin\varphi/(\sin\psi\cdot\sqrt{1-\sin^2\varphi})=1/2 \tag{21}$$

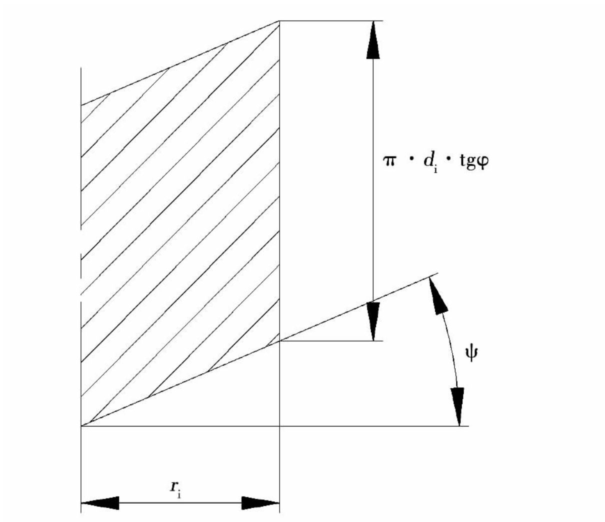

图 8　强化管流道截面

Fig. 8 Cross section of the flow passage inside a heat-transfer intensified heat pipe

最终可求得升角 $\varphi=25.91°$。热管内部流体是在封闭空间内作循环流动，因此简化模型计算得到的流动阻力和压力差只作为热管壁厚校核的参考依据。

利用 CFD 软件计算时，压力与速度耦合均采用 SIMPLEC 算法；差分格式均采用二阶迎风格式；进口为速度进口，出口为自由出流，翅片设为绝热。计算中设定：壁温 $T_w=523$ K；流体来流温度 $T_\infty=433$

K；计算流体为水蒸气。数值计算结果如图 9 – 图 12 所示。

图 9、图 10 显示了新式强化热管内部的阻力特性。一方面，在雷诺数 300 – 1800 的范围内，强化管沿程阻力损失增幅较大；另一方面，强化管阻力系数增幅随雷诺数增大而减少，阻力系数较之光管增加 10.4 – 23.3 倍。

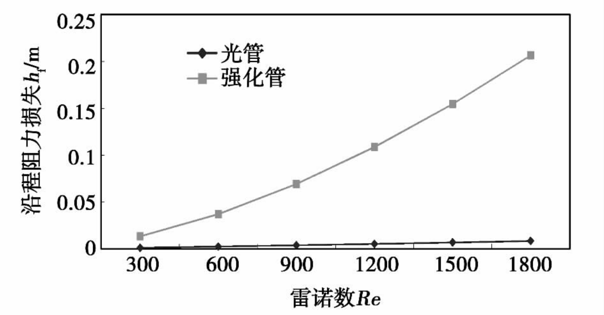

图 9　光管与强化管的沿程阻力损失 h_f 随雷诺数 *Re* 数的变化

Fig. 9 Changes of the resistance loss along the flow passage of a bare tube and a heat-transfer intensified heat pipe hf with Re

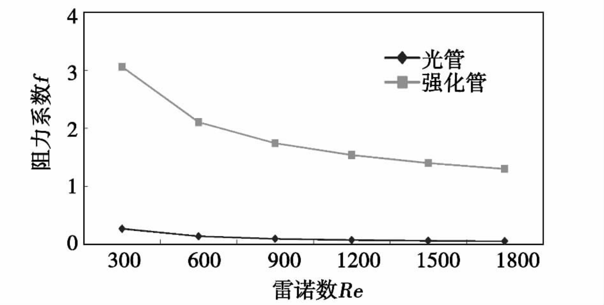

图 10　光管与强化管的阻力系数 *f* 随 *Re* 数的变化

Fig. 10 Changes of the resistance coefficient of a bare tube and a heat-transfer intensified heat pipe f with Re

图 11 为光管与强化管的热通量 Q 随 Re 数的变化。在 $Re = 1800$ 时，强化管导热性能比光管增加 18.7%。图 12 显示了光管与强化管的压降 Δp 随 Re 数的变化，相对应传热性能的提高，管内的流动阻力也在增加，增加幅度为 11.68 – 24.88 倍，强化管的 $Re - \Delta p$ 曲线近似直线。图 13 显示的是光管与强化管的效能评价系数 EEC 值[25] 随 Re 的变化 $(EEC = (Q/Q_0) \times (P_w/P_{w0})^{-1})$，强化管的效能评价系数 EEC 最大只有 0.09，随着 Re 数的增大，效能评价系数 EEC 出现下降趋势，一般来说，EEC 值应该接近甚至大于 1。虽然强化管能够改善热管的换热性能，但是与光管相比，强化传热热流量的增加倍数小于功耗增加的倍数。因此，这种片面追求场协同性的强化管不值得推荐。

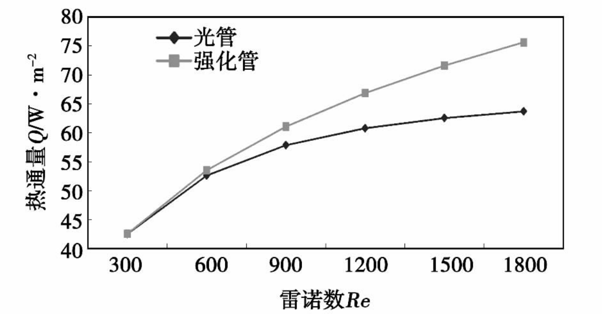

图 11　光管与强化管的热通量 Q 随 Re 数的变化

Fig. 11 Changes of the heat flux of a bare tube and a heat-transfer intensified heat pipe Q with Re

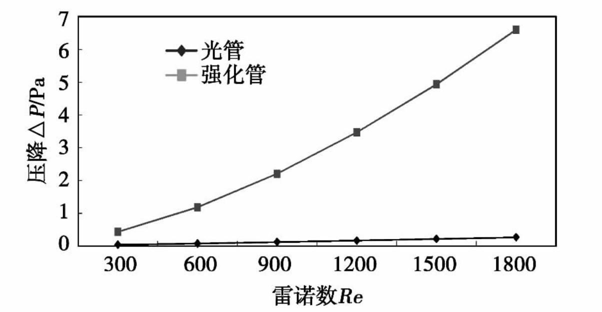

图 12　光管与强化管的压力降 Δp 随 Re 数的变化

Fig. 12 Changes of the pressure drop of a bare tube and a heat-transfer intensified heat pipe ΔP with Re

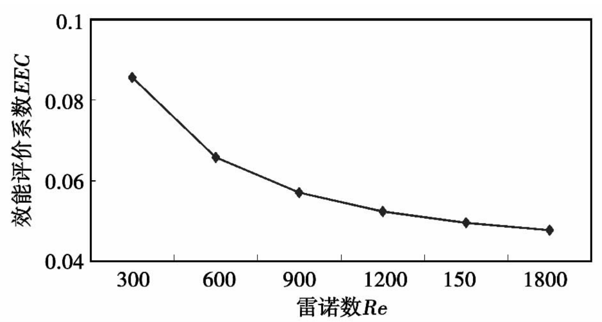

图 13　光管与强化管的 EEC 值随 Re 数的变化

Fig. 13 Changes of the *EEC*(efficiency evaluation criterion) of a bare tube and a heat-transder intensified heat pipe with *Re*

4 结 论

（1）根据对流换热场协同理论，增加内翅片来强化径向传热，给出了内螺旋翅片的设计参数，螺旋翅片的锥角 $\psi = 12.33°$ 升角 $\varphi = 25.91°$。

（2）强化后的热管与普通重力式热管相比，对流换热系数得到提高，在 $Re = 1800$ 时，热通量 Q 增加 18.7%。Re 由 300 到 1 800 的过程中，管内的流动阻力 h_f 增加 11.68 – 24.88 倍。

（3）阻力系数较之光管增加 10.4 – 23.3 倍，相应会使强化后重力式热管工作压力上升，设计中应考虑增加强化后重力式热管的壁厚。

（4）强化传热热流量的增加倍数小于功耗增加的倍数，强化传热效能评价系数 EEC 为 0.086，简化模型的强化效果不理想。

（5）本研究所给出的螺旋翅片锥角仅仅考虑了场协同性的影响，应综合考虑扰流件角度参数和流动阻力的关系，最终确定新式热管的最优设计。

致谢：西华大学能源与环境学院符杰、石建伟等人对文中数值模拟部分提供了必要的技术支持，因文章篇幅所限，对提供帮助的所有未列人员在此一并表示感谢。

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（丛　敏　编辑）

新技术、新工艺

地热载热介质杂质引起的地热电站运行问题

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据《Теплоэнергетика》2013 年 7 月刊报道，地热电站的运行证明在地热电站工作回路内侵蚀 - 腐蚀破坏和形成沉积物的主要原因是地热载热介质含有腐蚀 - 侵蚀性杂质和盐。

建立的计算模型“地热电站 B - M 回路”考虑地热电站的运行条件来确定在工艺回路不同地点的地热载热介质液相和汽相内杂质和盐的浓度。

按照模型“地热电站 B - M 回路”计算的结果可以用来定性确定不同运行工况下地热电站工艺回路内形成沉积物的可能性、强度和机理。

计算结果证明从地热载热介质液相得到十八烷所需浓度的可能性，以便保证其在地热电站整个工艺回路内的冲洗 - 抑制性能。

证明利用表面活化抑制剂来防止地热电站设备侵蚀 - 腐蚀的可能性。

（吉桂明　摘译）

show that the Nusselt number, and pressure drop of the fluid inside the spiral tube are all higher than those inside the straight tube and will increase with an increase of the curvature ratio and Reynolds number. The influence of the twist rate on the Nusselt number is not evident, however, to increase the twist rate can lead to a decrease of the pressure drop. The enhanced heat transfer comprehensive performance evaluation coefficient of the spiral tube is invariably greater than 1 at any Reynolds number, curvature ratio and twist rate. Under the condition of low Reynolds numbers, the spiral tube has very good enhanced heat transfer performance. **Key Words:** spiral tube, enhanced heat transfer, Dean vortex, Nusselt number, pressure drop

锯齿螺旋翅片管束换热与阻力特性关联式研究及比较 = Comparison of the Correlation Formulae for Calculating the Heat Exchange and Resistance Characteristics of Serrated Spirally-finned Tube Bundles 刊, 汉 PEI Yu-feng (Northeast Electric Power Designing Institute, China Electric Power Engineering Consultancy Group, Changchun, China, Post Code: 130021), MA You-fu (College of Energy Source and Power Engineering, Shanghai University of Science and Technology, Shanghai, China, Post Code: 200093), LIU Hong-wei (Sanhe Power Generation Co. Ltd., Sanhe, China, Post Code: 065201) //Journal of Engineering for Thermal Energy & Power. -2014, 29(6). -651-656

The currently available correlation formulae for calculating the heat exchange and resistance characteristics of serrated spirally-finned tubes were first sorted and summarized. Afterwards, based on the wind tunnel test results of 12 serrated spirally-finned tube bundles, the correlation formulae for calculating relevant heat exchange and resistance characteristics were verified and compared. It has been found that 1) the results predicted by using correlation formulae proposed by various researchers differ greatly, 2) the results predicted by using Weierman 1976, ESCOA1979 and Chen1998 correlation formula are in relatively good agreement with the test results, all the deviations are within 20%, among which the results predicted by using the ESCOA1979 correlation formula is in best agreement with the test ones, its deviation being within 10%, 3) under the same conditions, compared with the results predicted by using the correlation formula for continuous spirally-finned tubes, the increase in the heat quantity exchanged inside and outside the serrated spirally-finned tubes is relatively more but the change in the resistance is not big. **Key Words:** enhanced heat exchange, heat recovery steam generator, waste heat boiler, serrated spirally-finned tube, correlation formula

基于场协同理论的重力式热管新设计 = New Design of a Gravity Type Heat Pipe Based on the Field Synergy Theory 刊, 汉 SUN Xue-min, SONG Wen-wu, LIU Yu, TIAN Chao-chao (College of Energy and Environment, West China University, Chengdu, China, Post Code: 610039) //Journal of Engineering for Thermal Energy & Power. –2014,29(6). –657 –663

To optimize the synergy of the flow and temperature field in a gravity type heat pipe and improve the heat transfer characteristics inside the tube, based on the field synergy theory and according to the actual geometric dimensions and the field synergy angle etc. factors, a spiral fin flow disturbance tapered structure was mounted inside the gravity type heat pipe. A model for inner spirally-finned heat pipe was established: the round tube length $L_{model} = 500$ mm, tube diameter $D = 20$ mm, inner spiral fin length $L_2 = 100$ mm, the assembly dimension $L_1 = 200$ mm, the working medium in the tube was water. Through calculation, it has been determined that the cone angle of the spiral fin is 12.33 degrees and the lead angle is 25.91 degrees. By using the CFD software, the heat transfer and resistance characteristics of the flow inside the pipe were numerically simulated. It has been found that the radial heat transfer performance of the gravity type heat pipe has been improved and when Re = 1800, the heat flux Q of the enhanced heat pipe increases by 18.7% and the flow resistance inside the tube hg increases by 24.88 times as compared with those of common gravity type heat pipe. **Key Words:** gravity type heat pipe, field synergy, inner spiral fin, product design, model simplification, computational fluid dynamics

高瑞利数条件下竖排管束对原油的换热特性研究 = Study of the Heat Exchange Characteristics of Crude Oil in a Vertical Tube Bundle at High Rayleigh Numbers 刊,汉 ZHAO Jian, LIU Yang, DONG Hang, WEI Li-xin (National Key Laboratory on Production Ratio Enhancement, Northeast Petroleum University, Daqing, China, Post Code: 163318) //Journal of Engineering for Thermal Energy & Power. –2014,29(6). –664 –670

By using the standard turbulent flow model and based on the finite volumetric method, numerically studied were the natural convection heat exchange characteristics of crude oil outside a vertical tube bundle with the Rayleigh number and Pr number being in a range from $1.12 \times 10^6 - 1.02 \times 10^8$ and 101 – 127 respectively. It has been found that with an increase of the centerline distance between any neighboring two heating tubes, the tube bundle as a whole will experience in turn various stages, i. e. the heat exchange worsening, enhancing, stabilizing and declining stage. The fluid flow induced by the natural convection of the crude oil outside the bottom heating tubes enhances the speed of the fluid outside the upper tubes and intensifies the heat exchange of the upper tubes and at the same time, changes the temperature distribution in the fluid surrounding the upper tubes, leading to a deterioration of the heat exchange of the upper tubes and a fluctuation of Nu number with time. Moreover, the critical centerline distances of the highest heat exchange intensity and heat exchange enhancement all decrease with an increase of the Ra number and the role in enhancing the heat exchange will weaken with an increase of Pr number, thus to add the number of the tubes at the top can enhance the average heat exchange intensity of the tube bundle. **Key Words:** numerical simulation, heat transfer, natural convection, vertical tube bundle, crude oil