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直接接触式蒸汽发生器传热性能分析

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摘　要: 本文建立了直接接触式蒸汽发生器的数理模型, 模拟分析了直接接触式蒸发器的独立变量, 如: 初始换热温差、工质流率和导热油流率及它们对容积换热系数、总换热体积、工质蒸汽发生量及工质蒸汽出口温度等蒸发器主要换热性能的影响, 同时通过试验手段对所建立的数理模型进行了验证, 结果表明: 蒸汽发生器性能理论曲线与试验值一致性较好, 且各独立变量与换热性能间存在复杂的非线性函数关系, 为了获得最佳换热性能, 有必要对换热系统进行多参数并进行优化。

关 键 词: 直接接触式蒸汽发生器; 数理模型; 容积换热系数; 含气率

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符号说明

ρ——流体密度/kg·m^{-3};
ε——含气率;
μ——动力粘度/Pa·s;
x——汽化分率;
P——压强/kPa;
m——质量流率/kg·s^{-1};
g——重力加速度/m·s^{-2};
A——换热面积/m^2;
Q——换热量/W;
T——温度/K;
M_{RL}——单位体积混合相中的工质的质量/kg;
$\overline{D}$——平均直径/m, $\overline{D} = \frac{D^2 + D_0^2}{2DD_0}$;
k——蒸发速率常数;
h——容积换热系数/kW·m^{-3}·K^{-1};
α——体积分数;
B——定常系数;
η——连续相传递给分散相热量的比率;
F——合力/N;
f——摩擦系数;
U——速度/m·s^{-1};
Z——高度/m;
上标:
L——两相泡滴中的液相;
V——两相泡滴中的气相;
下标:
c——连续相流体;
d——分散相流体。

引　言

在直接接触式蒸汽发器中, 分散相液滴群在连续相中受热、膨胀、蒸发、脱离, 其流场的流动和传热特性较为复杂, 直接接触式蒸汽发生器的传热性能(如分散相蒸汽出口温度、蒸汽发生率和容积换热系数等)受众多因素的综合影响, 这些因素与蒸汽发生器的传热性能存在着复杂的函数关系[1-3]; 李彦博对直接接触式环流换热器气-液-液三相换热过程进行了计算机模拟[4], 推导出适用其换热器及实验物系的气含率分布模型和体积传热系数模型, 模型计算结果与实验结果基本吻合; 张鹏在全面考虑了直接接触换热器中汽体的膨胀、连续相水的蒸发、界面张力的作用及汽泡破碎等因素影响的基础上[5], 建立了整个换热器的一维两流体模型, 并通过模型的求解, 对换热器进行了数值模拟, 得到了与实验数据吻合较好的结果。

为了研究直接接触式蒸汽发生器传热性能的影响规律, 本文拟在综合考虑分散相的上升和膨胀、含气率、汽液两相流的相互作用和滑移、相间作用力等因素的影响下, 建立直接接触式蒸汽发生器性能评价指标的数理模型, 并通过试验手段对数理模型进

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行验证,希望通过本文的研究为直接接触式蒸汽发器性能的模拟和优化提供依据。

1　数理模型

集中参数建模方法主要关注于整体,不关注内部参数的相互影响,且建模过程中参数采用平均值。由于沿直接接触式蒸汽发生器轴向方向温度场一直变化,且分散相液滴群热物性参数如粘度、导热系数等随温度的变化较大,因此若采用集总参数法进行建模将会产生较大的误差。为提高精度,对模型采用分段模拟的方法,即将直接接触式蒸汽发生器沿轴向方向划分若干微元,再对每一个微元按分布参数模型进行建模。本文模拟的直接接触式蒸汽发生器结构示意图如图 1 所示,微元段示意图如图 2 所示。

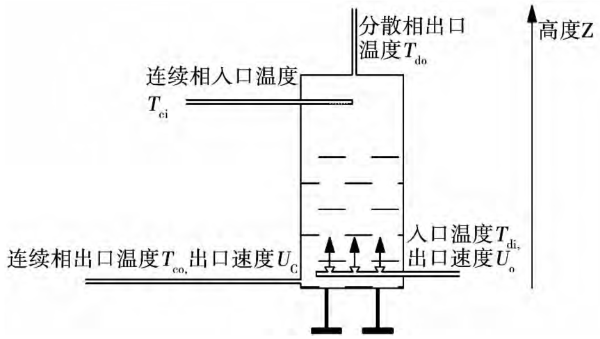

图 1　直接接触式蒸汽发生器结构示意图

Fig. 1 Schematic diagram of the structure of a direct contact type steam generator

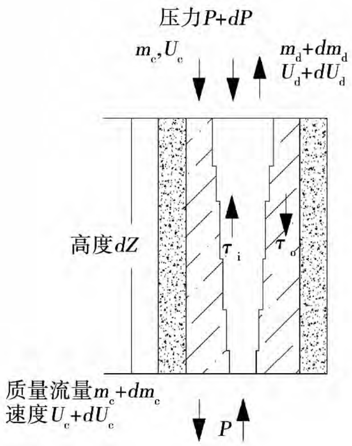

图 2　微元段示意图

Fig. 2 Schematic diagram of of a microelement section

1.1　基本方程

将直接接触式蒸汽发生器中的流动视为稳态一维流动,根据多相流理论将泡滴内汽液两相流按均相流计算,而连续相和分散相泡滴之间按分散流计算。在模型推导过程中,全面考虑了蒸发器中泡滴的膨胀和蒸发、含气率、汽液两相流的相互作用和滑移、相间作用力、界面张力作用等因素的影响。

(1) 连续性方程

对直接接触式蒸汽发生器 dz 微元段内逆向流模型进行质量衡算[4]:

连续相连续性方程:

$$\frac{\mathrm{d}}{\mathrm{d}z}\left[\rho_c A(1-\varepsilon_g)u_c\right]=0 \tag{1}$$

分散相连续性方程:

$$\frac{\mathrm{d}}{\mathrm{d}z}\left[\rho_d A\varepsilon_g u_d\right]=0 \tag{2}$$

(2) 动量方程

根据动量守恒定律,相动量变化等于作用在该相上的力之和。对于分散相,其在 z 方向上能动量方程为[4]:

$$(m_c+\mathrm{d}m_c)(u_c+du_c)-m_c u_c=P(1-\varepsilon_g)A-\left\{P(1-\varepsilon_g)A+\mathrm{d}z\frac{\mathrm{d}}{\mathrm{d}z}\left[P(1-\varepsilon_g A)\right]\right\}+\left(P+\frac{\mathrm{d}P}{2}\right)\mathrm{d}z\frac{\mathrm{d}}{\mathrm{d}z}\left[(1-\varepsilon_g)A\right]-g\rho_c(1-\varepsilon_g)A\mathrm{d}z-\tau_0 C\mathrm{d}z+\tau_i C_i\mathrm{d}z \tag{3}$$

式中等号左边的动量相未引入分界面上质量交换所产生的动量项,因为它在 z 方向上的分量为零。而等号右边前两项是作用在微元段两端的压力; 第三项是作用在微元体表面上的压力,其是由于连续相截面面积的变换产生的。简化并略去高阶微分项,便可得分散相动量方程:

$$\rho_h u_d\varepsilon_g\frac{\mathrm{d}u_d}{\mathrm{d}Z}=-\varepsilon_g\frac{\mathrm{d}P}{\mathrm{d}Z}-g\rho_h\varepsilon_g-F_{wd}-F_1 \tag{4}$$

同样对于连续相有:

$$\rho_c u_c(1-\varepsilon_g)\frac{\mathrm{d}u_c}{\mathrm{d}Z}=-(1-\varepsilon_g)\frac{\mathrm{d}P}{\mathrm{d}Z}-g\rho_c(1-\varepsilon_g)-F_{wc}+F_1 \tag{5}$$

(3) 能量方程

根据能量守恒,连续相与分散相流体的一维能量方程为[5]:

连续相:

$$\frac{\mathrm{d}}{\mathrm{d}z}\left[\rho_c(1-\varepsilon_g)u_c h_{Vc}\right]=-\frac{\eta Q_d}{V} \tag{6}$$

分散相：

$$\frac{\mathrm{d}}{\mathrm{d}z}\left[\rho_{\mathrm{f}}\varepsilon_{\mathrm{g}}u_{\mathrm{d}}h_{\mathrm{Vd}}\right]=\frac{Q_{\mathrm{d}}}{V}\tag{7}$$

1.2 分散相流体含气率分布模型

对于图 2 所示的微元段来说，通过该微元段的含气率变化为 $\mathrm{d}\varepsilon$，所以工质蒸汽流量的变换是：

$$\mathrm{d}m_{\mathrm{g}}=u_{\mathrm{g}}\rho_{\mathrm{g}}A\mathrm{d}\varepsilon=\dot{\Gamma}(1-\varepsilon)A\mathrm{d}z\tag{8}$$

式中：$\dot{\Gamma}$—工质的体积蒸发率[7]：

$$\dot{\Gamma}=kM_{\mathrm{RL}}^{n}\tag{9}$$

$\dot{\Gamma}$ 是随气化分率 x 变化的，根据其定义两者的关联为：

$$\dot{\Gamma}=\frac{xm_{\mathrm{d}}}{A\int_0^z(1-\varepsilon)\mathrm{d}z}\tag{10}$$

联立式(8)和式(10)得：

$$\frac{\mathrm{d}\varepsilon}{\mathrm{d}z}=\frac{xm_{\mathrm{d}}(1-\varepsilon)}{A\rho_{\mathrm{g}}u_{\mathrm{g}}\int_0^z(1-\varepsilon)\mathrm{d}z}\tag{11}$$

由于分散相泡滴在气化过程中质量保持不变，同时根据连续性方程(2)，泡滴的密度为：

$$\rho_{\mathrm{d}}=\frac{m_{\mathrm{b0}}}{V_{\mathrm{b}}}=\frac{m_{\mathrm{b0}}}{\dfrac{xm_{\mathrm{b0}}}{\rho_{\mathrm{d}}^{V}}+\dfrac{(1-x)m_{\mathrm{b0}}}{\rho_{\mathrm{d}}^{L}}}=\frac{\rho_{\mathrm{d}}^{L}\rho_{\mathrm{d}}^{V}}{(1-x)\rho_{\mathrm{d}}^{V}+x\rho_{\mathrm{d}}^{L}}\tag{12}$$

而 $U_{\mathrm{g}}=U_{\mathrm{b}}$，$\rho_{\mathrm{d}}^{V}<<\rho_{\mathrm{d}}^{L}$

所以有：

$$\rho_{\mathrm{d}}\doteq\frac{\rho_{\mathrm{d}}^{\mathrm{L}}\rho_{\mathrm{d}}^{V}}{x\rho_{\mathrm{d}}^{\mathrm{L}}}\doteq\frac{\rho_{\mathrm{d}}^{\mathrm{V}}}{x}=\frac{\rho_{\mathrm{g}}}{x}\tag{13}$$

即 $\rho_{\mathrm{g}}=x\rho_{\mathrm{d}}$，带入式(11)得到含气率随轴向高度变化的关系式：

$$\frac{\mathrm{d}\varepsilon}{\mathrm{d}z}=\frac{\varepsilon(1-\varepsilon)}{\int_0^z(1-\varepsilon)\mathrm{d}z}\tag{14}$$

1.3 分散相流动换热的计算

在计算分散相换热时，根据分散相含气率模型及分散相进口参数：分散相流量 m_{r}、蒸汽发生器进口温度 t_{in} 和压力 p_{in}，则分散相容积换热系数为[8]：

$$h_{\mathrm{vd}}=\frac{2h_{\mathrm{b0}}}{D_0Bz}\Bigg[3\alpha_0Bz_{\mathrm{a}}+\frac{\alpha_{\max}}{1-\alpha_{\max}}\left(\left(1+\frac{7}{2}Bz_{\mathrm{a}}+21(1-\alpha_{\max})B(z-z_{\mathrm{a}})\right)^{1/7}-\left(1+\frac{7}{2}Bz_{\mathrm{a}}\right)^{1/7}\right)\Bigg]\tag{15}$$

1.4 连续相流动换热的计算

在直接接触式蒸发器中，连续相流体的流动状态为湍流状态，其换热系数为[8]：

$$h_{\mathrm{vc}}=2\frac{h_{\mathrm{b0}}}{D_0}\frac{\alpha_0(r_{\mathrm{a}}^3-1)+\dfrac{\alpha_{\max}}{1-\alpha_{\max}}(r^{1/2}-r_a^{1/2})}{\dfrac{r_a^3-1}{3}+\dfrac{1}{21r_{\mathrm{a}}^{1/2}}\dfrac{r^{7/2}-r_{\mathrm{a}}^{7/2}}{1-\alpha_{\max}}}\tag{16}$$

1.5 辅助方程

(1) 分散相液滴初始直径 d_0

分散相液滴初始直径与连续相物性和喷嘴的直径有关[9]：

当 $d_i/\left(\dfrac{\sigma}{g\Delta\rho}\right)^{\frac{1}{2}}<0.785$ 时，

$$\frac{d_i}{d_0}=0.485\left[\frac{d_i}{(\sigma/g\Delta\rho)^{\frac{1}{2}}}\right]+1\tag{17}$$

当 $d_i/\left(\dfrac{\sigma}{g\Delta\rho}\right)^{\frac{1}{2}}>0.785$ 时，

$$\frac{d_i}{d_0}=1.51\left[\frac{d_i}{(\sigma/g\Delta\rho)^{\frac{1}{2}}}\right]+0.12\tag{18}$$

(2) 分散相液滴的瞬时速度 U

分散相液滴的瞬时速度由 Raina 给出[10]：

$$U=\frac{1.91\left[\left\{1-\dfrac{\rho_{\mathrm{d}}}{\rho_{\mathrm{c}}}\left(\dfrac{D_0}{D}\right)^3\right\}\left\{\dfrac{\sigma}{\rho_{\mathrm{c}}D}\right\}\right]^{\frac{1}{2}}\overline{D}^{\left[\frac{5}{6}-\frac{\overline{D}}{T_{\mathrm{C}}}\right]}}{\left[\dfrac{T_{\mathrm{C}}^2+T_{\mathrm{L}}^2}{2T_{\mathrm{C}}T_{\mathrm{L}}}\right]^{\overline{D}}\times\left[\dfrac{C_{\mathrm{pc}}\mu_{\mathrm{c}}}{k_{\mathrm{c}}}\right]^{\frac{D_0}{1.6D}}}\tag{19}$$

(3) 壁面摩擦力

单相流壁面摩擦力[11]：

$$f_{\mathrm{r}}=4f_{\mathrm{w}}(1-\varepsilon)\tag{20}$$

汽液两相流的壁面摩擦力采用与单相流相似的计算方法，只是摩擦系数 f_{w} 计算方法不同：

$$f_{\mathrm{w}}=0.0468\left(\sqrt{gD_{\mathrm{r}}}/u_{\mathrm{c}}\right)^{1.1}\sqrt{\varepsilon}\tag{21}$$

适用条件为：$\varepsilon>0$，$0<u_{\mathrm{c}}/\sqrt{gD_{\mathrm{r}}}<1$。

(4) 相间相互作用力

连续相与分散相泡滴之间的相互作用力 F_i 由相间界面摩擦阻力 F_{d}、表观质量力 F_{vm} 和 Basset 力组成[12]。其中 Basset 力很小，可以忽略不计，则：

$$F_i=F_{\mathrm{d}}+F_{\mathrm{vm}}\tag{22}$$

其中：

$$F_{\mathrm{d}}=\frac{3C_{\mathrm{D}}}{4D}\rho_{\mathrm{c}}\varepsilon\,|u_{\mathrm{d}}-u_{\mathrm{c}}|\,(u_{\mathrm{d}}-u_{\mathrm{c}})\tag{23}$$

且 C_D 为拖曳系数[8]:

$$C_D = \frac{2}{3}\bar{D}\sqrt{\frac{g(\rho_c - \rho_d)}{\sigma(1-\varepsilon)}} \tag{24}$$

联系式(23)和式(24)得:

$$F_d = \frac{\varepsilon\rho_c}{2D}\sqrt{\frac{g(\rho_c - \rho_d)}{\sigma(1-\varepsilon)}}\,|u_d - u_c|\,(u_d - u_c) \tag{25}$$

表观质量力为:

$$F_{vm} = \frac{\varepsilon(1+2\varepsilon)}{2(1-\varepsilon)}\rho_c\left(u_d\frac{\mathrm{d}u_d}{\mathrm{d}z} - u_c\frac{\mathrm{d}u_c}{\mathrm{d}z}\right) \tag{26}$$

(5) 两相分界面的周界 C_i

在高度为 $\mathrm{d}z$ 的微元段内,连续相和分散相的相接触面积 A_i 为:

$$A_i = A\pi D^2 n_b \mathrm{d}z \tag{27}$$

两相分界面周界 C_i 的定义为:

$$A_i = C_i \mathrm{d}z \tag{28}$$

联立得:

$$C_i = \pi D^2 n_b A \tag{29}$$

2 数理模型求解

2.1 初始条件设定

模拟计算初始条件参照试验过程中实际运行条件进行设定:

试验过程中,分散相流体选 R245fa 作为分散相;连续相流体选用导热油;直接接触蒸气发生器的几何结构为蒸发器内径 0.5 m,高 1.2 m,喷头喷孔直径 4 mm,喷孔个数 30 个。

同时忽略分散相工质进口的欠焓,即 $\Delta T_{app} = 0$;

环境条件:空气干球温度 $t_{ain} = 15$ ℃,相对湿度 $\varphi_{in} = 55\%$,大气压力 $p_{atm} = 101.325$ kPa,本文环境温度统一取 $t_{surr} = 25$ ℃。

2.2 数理模型求解过程

对于直接接触式蒸汽发生器数理模型的求解,首先将基本方程的各微分项展开,然后将与其相关的辅助方程代入,即可得到一组微分方程。根据初始设定条件的初值,即可用四阶 Rong - Kutta 对微分方程组进行数值求解。数理模型计算程序流程如图 3 所示。

3 试验

试验台主要由直接接触式蒸气发生器、电加热器、齿轮油泵、工质离心泵及板式换热器组成,如图 4 所示。电加热器连接交流调压器的温控装置,温控调节范围为 80 ~ 120 ℃;工质泵和齿轮油泵均安装变频器调速器,调速器均有三个档位,分别对应工质流率的调速为 0.04、0.08、0.12 m/s,导热油流率的调速为零(导热油为静止状态)、0.36、0.72 kg/s;直接接触式蒸气发生器为圆柱形罐体,罐顶安装压力表及安全阀;质量流量计、压力表、温度表按图 4 所示位置安装,各测试仪表的误差在 ±5% 以内。

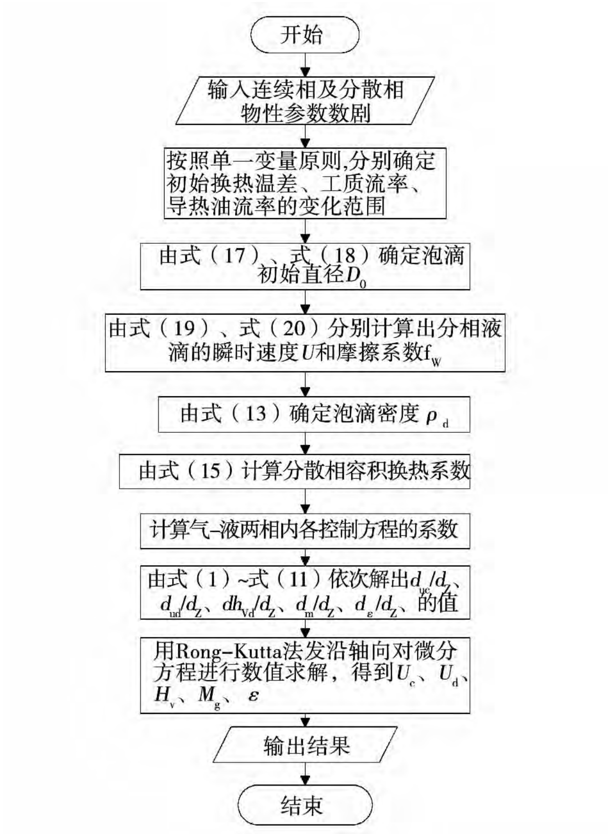

图 3　数理模型计算程序流程图

Fig. 3 Flow chart of a mathematical model-based calculation program

4 性能分析

本文主要模拟直接接触式蒸汽发生器的主要性能,包括容积换热系数、工质蒸汽发生量、总换热体积及工质蒸汽出口温度随初始换热温差、工质流率和导热油流率的变化规律。最后在试验平台客观运行范围内得到蒸气发生器性能的试验数据,通过理论值与试验值的对比,验证理论模型的精确度。

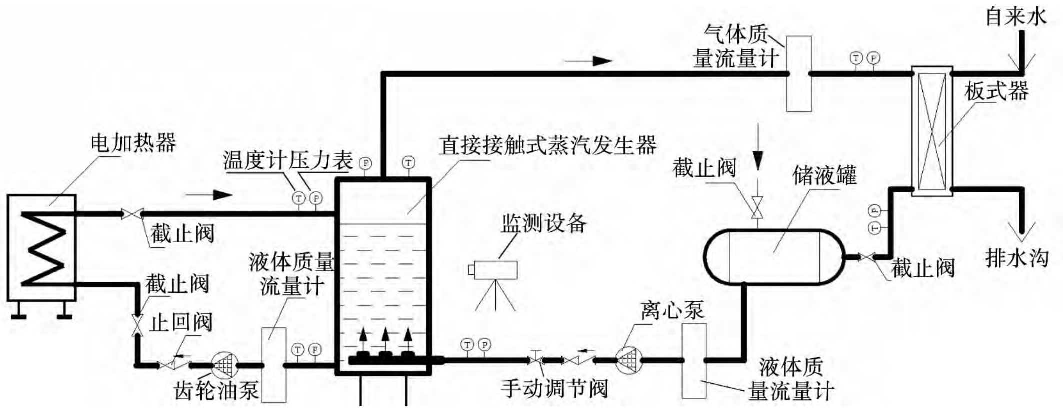

图 4　直接接触式蒸气发生器试验平台管路系统

Fig. 4 Piping system on the test platform of the direct contact type steam generator

4.1　初始换热温差对直接接触式蒸汽发生器主要性能的影响

如图 5 ~ 图 6 所示,在工质流率和导热油流率不变的情况下,随着初始换热温差的增加,工质蒸汽出口温度呈增加的趋势,而工质蒸汽发生量随初始换热温差的增加先呈上升趋势,然后呈下降趋势,而总换热体积变化趋势则相反。

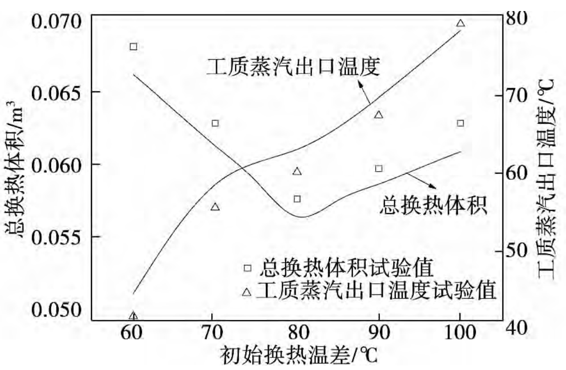

图 5　总换热体积及工质蒸汽出口温度随初始换热温差的变化

Fig. 5 Changes of the total heat exchange volume and the temperature of the working medium steam at the outlet with the initial heat exchange temperature difference

容积换热系数随着初始换热温差的增加先呈上升趋势再呈下降趋势,在上升阶段,其是由于工质蒸汽发生量与总换热体积对容积换热系数的影响占主导作用,但当初始换热温差增加到一定值后,容积换热系数受初始换热温差的影响又占主导作用,所以导致容积换热系数呈下降趋势。且工质蒸汽发生量与容积换热系数均存在一个最高峰值点,其对应的初始换热温差的值在邻近区域。最后从图 5 ~ 图 6 中可以看出蒸气发生器理论曲线与试验值一致性较好。

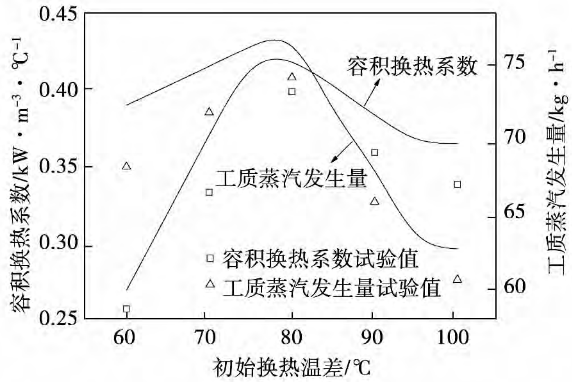

图 6　容积换热系数及工质蒸汽发生量随初始换热温差的变化

Fig. 6 Changes of the volumetric heat exchange coefficient and the production capacity of the working medium steam with the initial heat exchange temperature difference

4.2　工质流率对直接接触式蒸汽发生器主要性能的影响

如图 7 ~ 图 8 所示,随着工质流率的增加,工质蒸汽出口温度呈下降趋势,而工质蒸汽发生量呈上升趋势,同时在初始阶段,容积换热系数呈上升趋势,这是由于工质蒸汽发生量的增加及总换热体积的下降所导致的,而当工质流率增加到一定程度后,由于蒸发器内加热负荷的增加,所需的总换热体积

迅速增大，容积换热系数呈下降趋势，即容积换热系数的值存在一个最高峰值。因此，工质流率对直接接触蒸发器的性能的影响主要表现在导热油热量传递与工质流率匹配上，当工质流率较低时，最直接的影响是使工质的蒸汽发生量降低，同时会导致导热油的出口温度较高，导热油的热量不能充分的传递给分散相工质。而当工质流率较高时，由于热负荷增大导致所需热量的不匹配，使得工质蒸汽的出口温度较低。

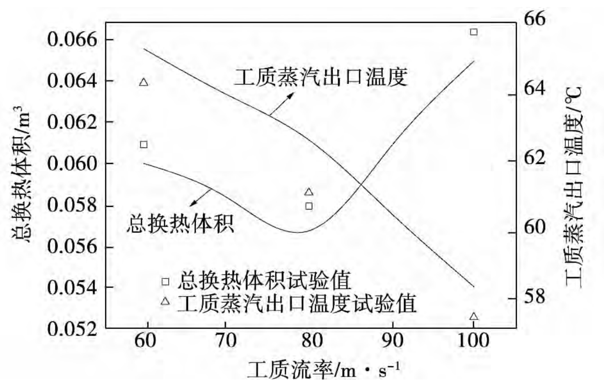

图 7　总换热体积及工质蒸汽出口温度随工质流率的变化

Fig. 7 Changes of the total heat exchange volume and the temperature of the working medium steam at the outlet with the flow rate of the working medium

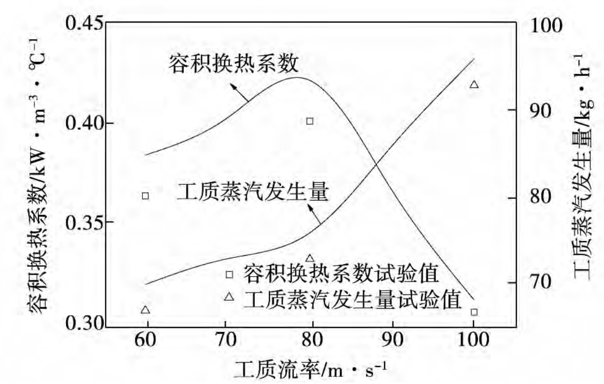

图 8　容积换热系数及工质蒸汽发生量随工质流率的变化

Fig. 8 Changes of the volumetric heat exchange coefficient and the production capacity of the working medium steam with the flow rate of the working medium

4.3　导热油流率对直接接触式蒸汽发生器主要性能的影响

从图 9 ~ 图 10 可以看出，在初始换热温差和工质流率不变的情况下，随着导热油流率的增加，直接接触换热器工质蒸汽出口温度和总换热体积均呈上升趋势，而工质蒸汽发生量呈下降趋势。由于工质蒸汽发生量对容积换热系数的影响，使得容积换热系数随导热油流率的增加一直呈现下降趋势。

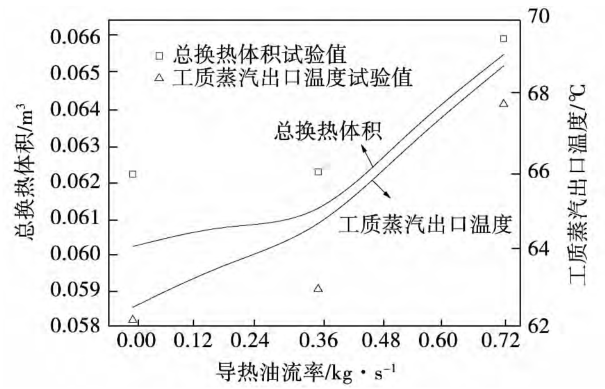

图 9　总换热体积及工质蒸汽出口温度随导热油流率的变化

Fig. 9 Changes of the total heat exchange volume and the temperature of the working medium steam at the outlet with the flow rate of the heat conduction oil

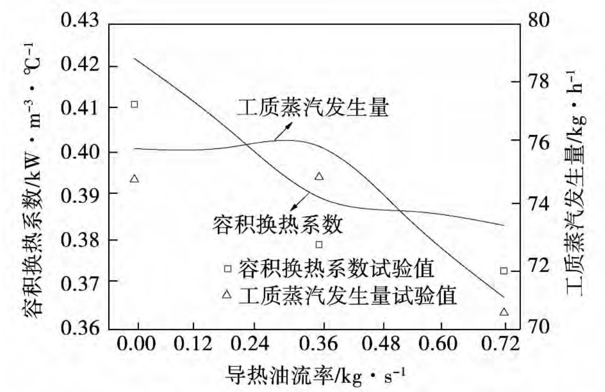

图 10　容积换热系数及工质蒸汽发生量随导热油流率的变化

Fig. 10 Changes of the volumetric heat exchange coefficient and the production capacity of the working medium steam with the flow rate of the heat conduction oil

5 结 论

(1)工质蒸汽发生量及容积换热系数随初始换热温差的增加先呈上升趋势,然后呈下降趋势,而总换热体积变化趋势则相反;

(2)随着工质流率的增加工质蒸汽出口温度呈下降趋势,而工质蒸汽发生量呈上升趋势,容积换热系数在所模拟及试验测试范围内存在极大值;

(3)工质蒸汽出口温度和总换热体积均随着导热油流率的增加而增加,而工质蒸汽发生量呈下降趋势;

(4)通过试验验证发现蒸汽发生器性能理论曲线与试验值一致性较好。

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(刘 瑶 编辑)

新技术、新设计

UGT 6000 + 将驱动中国海军气垫登陆艇

据《Gas Turbine World》2014 ~ 2015 年度手册报道,中国海军已于 2013 年向乌克兰 Zorya-Mashproekt(曙光-机器设计)燃气轮机科研生产联合体订购了 20 台 UGT 6000 + 船舶燃气轮机,用来驱动中国海军气垫登陆艇。

各气垫登陆艇将安装 5 台 UGT 6000 + 燃气轮机,3 台用于推进,2 台用于垫升。20 台燃气轮机将装备 4 艘气垫登陆艇。

在该档功率的船舶燃气轮机中,UGT 6000 + 是一型性能较好的机组。

UGT 6000 + 船舶燃气轮机的技术规范和规定性能:

推出年份	ISO 连续额定功率/kW	耗油率/kJ·(kW·h)$^{-1}$	效率/%	压比	质量流量/kg·s^{-1}	动力涡轮转速/r·min^{-1}	排气温度/℃	约重/kg	总体尺寸 长×宽×高/m
1997	8 825	0.255	33.0	16.0	34.0	7 000	470	3 502	3.0×1.5×1.8

(吉桂明 摘译)

基于两相流的微通道冷却技术研究进展及展望 = Research Proqress and Prospects of Micro-channel Cooling Technology Based on Two-phase Flow [刊，汉] ZHOU Yun-long, SUN Zhen-guo (School of Energy and Power Engineering, Northeast Dianli University, Jilin, China, Post Code: 132012) //Journal of Engineering for Thermal Energy & Power. -2016, 31(7). -1 ~ 6

With the development of the science and technology, a relatively high temperature may gradually limit the power output of a large power component and the studies currently focus on the dimensions of the channels and structural optimization etc. In addition, the working media are relatively singular and the most majority of the phase change emerge in the form of boiling. The study of the cooling in micro-channels relating to non-phase change two-phase flows is relatively less. The flow patterns serve as the basis for probing into the behavior of a two-phase flow and may produce a direct influence on the heat and mass transfer characteristics. Upon the completion of a summing-up of the research results of researchers both at home and abroad, the authors have viewed the prospects of the micro-channel cooling technology in the actual applications of the PV batteries. One may attempt to apply the multi-phase flows in the cooling of the solar power cells, therefore enhancing its power generation efficiency. **Key words:** micro-channel, two-phase flow, flow pattern, cooling technology

池沸腾临界热流密度关系式分析研究 = Analysis and Study of the Pool Boiling Critical Heat Flux Density Correlation Formula [刊，汉] DONG An-qi, FANG Xian-de (College of Aerospace Engineering, Nanjing University of Aeronautics and Astronautics, Nanjing, China, Post Code: 210016), HUANG Yong-kuan (Hongdu No. 650 Research Institute, China Aviation Industry Corporation, Nanchang, China, Post Code: 330024) //Journal of Engineering for Thermal Energy & Power. -2016, 31(7). -7 ~ 14

With respect to the calculation of the pool boiling critical heat flux density, a great many of correlation formulae were proposed. In the design of engineering projects, it is necessary to be aware of the adaptability of these correlation formulae and to analyze the currently-available calculation methods in the study of a new calculation model. The test data and the number of the correlation formulae involved in the evaluation and analysis of the currently available formulae are limited, leading to a great difference between the evaluation results. A total of 468 groups of data were collected from 27 literatures and used to compare the accuracy and applicable range of the commonly-used 20 critical heat flux density calculation formulae and the formulae in comparatively good agreement with the test data were sifted out. It has been found that El-Genk-Guo formula, which is the best of all, has an average absolute error of 26.2%. To this end, it is necessary to find out a more accurate formula. Moreover, the factors influencing the accuracy of the formulae were analyzed, thus offering reference for proper design of relevant equipment items. **Key words:** pool boiling, critical heat flux density, heat transfer, correlation formula

直接接触式蒸汽发生器传热性能分析 = Analysis of the Heat Transfer Performance of a Direct Contact Type Steam Generator [刊，汉] HUANG Jun-wei (College of Electromechanical Engineering, Yunnan Agricultur-

al University, Kunming, China, Post Code: 650201), WANG Hui-tao (College of Metallurgical and Energy Source Engineering, Kunming University of Science and Technology, Kunming, China, Post Code: 650093), LI Hong-bo (State Grid Jilin Provincial Electric Power Co. Ltd. Training Center, Changchun, China, Post Code: 130011), XU Jian-xin (National Key Laboratory on Complex Nonferrous Metal Resource Clean Utilization Established by the Province and Ministry, Kunming University of Science and Technology, Kunming, China, Post Code: 650201) //Journal of Engineering for Thermal Energy & Power. -2016, 31(7). -15~21

Established was a mathematical model for direct contact type steam generators, simulated and analyzed was the influence of such independent variables of a direct contact type steam generator as the initial heat exchange temperature difference and the working medium flow rate and heat conduction oil flow rate on the main performance of a steam generator such as its volumetric heat exchange coefficient, total heat exchange volume, working medium steam production capacity and the temperature of the working medium at the outlet and at the same time verified was the mathematical model thus established through employing the test means. It has been found that the values obtained from the theoretical curves of the performance of the steam generator has a relatively good agreement with the test values and a complicated non-linear function relationship does exist between various independent variables and the heat exchange performance. In order to obtain the optimum heat exchange performance, it is necessary to perform a parallel optimization of the heat exchange system. **Key words**: direct contact type steam generator, mathematical model, volumetric heat exchange coefficient, gas content

非能动余热排出系统C型管换热器数值模拟 = Numerical Simulation of a C-shaped Tube Heat Exchanger in a Passive Residual Heat Removal System [刊,汉] LIU Ai-qiong, ZHANG Xiao-ying (College of Sino-France Nuclear Engineering and Technology, Zhongshan University, Zhuhai, China, Post Code: 519082), LI Zhi-wei (College of Electric Power, South China University of Science and Technology, Guangzhou, China, Post Code: 510640) //Journal of Engineering for Thermal Energy & Power. -2016, 31(7). -22~29

To guarantee the effective heat conduction of residual heat from a passive residual heat removal system under the accidental operating conditions, studied were the heat exchange characteristics of the main equipment item, i. e. the PRHR heat exchanger and established was a model for analyzing the inner and outer coupled heat transfer of C-shaped tube heat exchangers in a passive waste heat discharging system. In this connection, a one-dimensional homogeneous phase flow model was employed to calculate the condensation heat exchange inside the tubes and the CFD program was used to analyze the natural convection in the space of a water pool. In addition, the influence of the mass flow rate at the inlet, gas content of the fluid at the inlet, the inclination angle of the tubes and the temperature inside the water box on the heat exchange performance of the C-shaped tube heat exchanger was also studied. It has been found that a saturated two-phase flow always exists inside the tubes in the inclined section at the inlet of the C-type tube heat exchanger, the temperature of the fluid inside the tubes in the vertical section and the inclined section at the outlet will gradually decline, the pressure inside the tubes, enthalpy value of the fluid and the heat exchange coefficient will drop step by step along the tube length direction, the parameters of the fluid in