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细微通道内低温 CO_2 流动沸腾换热特性研究

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摘 要:用 CO_2 作为制冷剂,对内径为0.6和1.5 mm的细微通道内的低温流体流动沸腾换热特性进行了实验研究,定量分析了实验测试工况下低温流体流经管路时不同工况参数对换热系数的影响。研究表明:参考文献[7]中提出的流动沸腾换热模型具有较高的预测精度,且将误差控制在30%的范围时,流体发生干涸前的换热系数理论预测精度比(实验数据与模型数据之比)可达79.8%,平均偏差可达21.8%;流体发生干涸后的换热系数理论预测精度比为18.4%,平均偏差为59.9%。

关 键 词:二氧化碳;换热系数;干涸;流动沸腾换热

中图分类号:TK121 **文献标识码**:A

引 言

虽然对 CO_2 在低温系统如复叠制冷系统的研究早已进行,但传统的复叠制冷系统中仍旧采用管翅式换热器,现代制冷系统为了满足制冷剂零泄漏、最小制冷剂充注以及高效换热等要求,微细通道换热器将成为制冷系统换热器发展趋势。微细通道换热器在制冷系统应用对换热系数及机理研究提出了新的挑战。同时,随着换热器通道的变小,其换热过程中导致的制冷剂压降也成为研究的重要课题。

本研究针对 CO_2 在低温工况时的流动沸腾换热特性进行研究,分析热流密度、质量流量、蒸发温度和压降对换热系数的影响,并对其间出现的干涸现象进行了阐述,为实现紧凑、高效、低压降的 CO_2 制冷系统微细通道换热器研发提供理论支持。

1 实验测试原理

实验研究不仅要获得 CO_2 在微细通道内流动沸腾换热过程中不同质量流量、热流密度、蒸发温度和管径等参数对换热系数的影响,同时要能判断干度、流态等对于干涸现象的影响。因此需要综合考虑以上因素确定实验装置能够测试的范围,如表1所示。

表1 实验测试工况表

Tab. 1 Table of the operating conditions under the test

测试参数	测试范围
测试温度/℃	24-60
测试管径/mm	0.6,1.5
测试段长度/m	0.3
测试流量/kg·(m^2·s)$^{-1}$	300-600
测试热流密度/kW·m^{-2}	7.5-30
测试干度	0-1

实验系统由循环系统、低温载冷剂循环系统和数据采集系统3部分构成。实验系统原理如图1所示,实验系统实物图如图2所示。

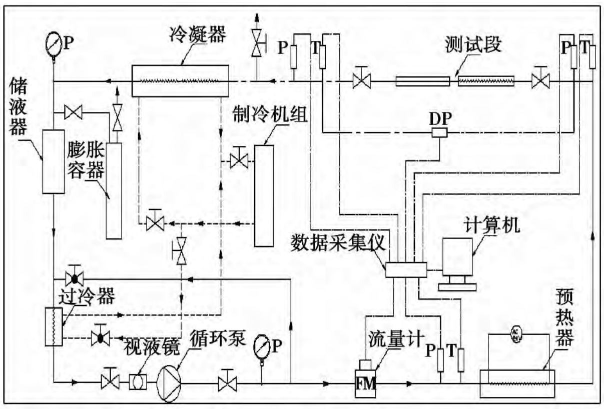

图1 实验系统原理图

Fig. 1 Schematic diagram of the test system

CO_2 循环系统主要由测试段、冷凝器、储液器、过冷器、预热器、膨胀容器、阀件以及连接管路构成,采用 CO_2 液体柱塞泵为系统内部介质提供循环动力。CO_2 制冷系统运行压力通常要比目前常用的制冷剂高的多,系统运行压力可达到R22制冷系统的10倍,利用 CO_2 液体柱塞泵取代制冷系统循环中的压缩机作为介质流动驱动装置能够有效降低整个实验装置在运行过程中的系统压力。实验系统运行时

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储液器中液态 CO_2 经过冷凝器冷却后由柱塞泵输送至测试段内,经过预热器时通过调节输入预热量功率可以控制测试段入口 CO_2 液体焓值。采用 Coriolis 质量流量计测得 CO_2 液体循环流量,通过调节柱塞泵的行程调节旋钮精确控制系统测试时所需的 CO_2 流量。对测试段不锈钢钢管两端施加大电流、低电压,通过调节施加电压控制测试段管路输入的热流密度,施加电功率由功率计直接测得,由测试段排出的高干度 CO_2 气体进入套管式换热器中被冷凝成液体后再次进入储液器完成一个测试循环,分别采用热电偶与电容式压力传感器获测得温度、压力。

图2 实验系统图

Fig. 2 Diagram of the test system

测试段内局部换热系数 h 主要由换热过程中测得的热流密度 q,计算所得的微细通道内制冷剂温度 T_r 以及管内壁温度 $T_{w,i}$ 决定:

$$h = \frac{q}{T_{w,i} - T_r} \tag{1}$$

式中: $T_{w,i}$—管内壁面温度,K; T_r—制冷剂温度,K。

为便于实现对测试段加热量调节与制冷剂流态观测,采用两端施加电压的方式对测试段进行加热,因此对于内部测试制冷剂可假设为常物性,对测试圆管换热过程具有内热源的一维稳态导热模型[2],其导热微分方程为:

$$\frac{1}{r}\frac{d}{dr}\left(r\frac{dt}{dr}\right) + \frac{\dot{\phi}}{\lambda} = 0 \tag{2}$$

其中,$\dot{\phi} = \dfrac{Q}{\pi(r_{w,o}^2 - r_{w,i}^2)L}$。

式中: r—测试管半径,m; λ—导热系数,W/(m·K)。

边界条件:

$$r = r_{w,i}, \frac{dt}{dr} = \frac{Q}{2\pi r_{w,i}\lambda L}, r = r_{w,o}, t = t_{w,o}$$

由式(2)可得到测试圆管壁温度及管内外壁温差:

$$t = -\frac{\phi r^2}{4\lambda} + C_1 \ln r + C_2 \tag{3}$$

其中,$C_1 = \dfrac{Q}{2\pi r_{w,i}\lambda L} + \dfrac{Q r_{w,i}^2}{2\pi(r_{w,o}^2 - r_{w,i}^2)\lambda L}$。

$$\triangle t = T_{w,o} - T_{w,i} = -\frac{Q(r_{w,o}^2 - r_{w,i}^2)}{4\lambda} + C_1 \ln\frac{r_{w,o}}{r_{w,i}} \tag{4}$$

管内壁温度 $T_{w,i}$ 为:

$$T_{w,i} = T_{w,o} - \left(\frac{QD_i}{4\lambda}\right)\left(\frac{\eta(1-\ln\eta) - 1}{1-\eta}\right) \tag{5}$$

其中,$\eta = \left(\dfrac{r_{w,o}}{r_{w,i}}\right)^2$。

式中: D—测试管内径,m; L—测试管长,m; Q—热量,W。

微细通道内制冷剂温度 T_r 由测试段内制冷剂饱和压力所对应饱和温度计算,其中测试段内位置 z 处 CO_2 饱和压力采用线性内插法计算得到:

$$P(X) = P_{p1} - \frac{\triangle P(L_{p1} + z)}{L_{dp}} \tag{6}$$

对于测试段内位置 z 处,基于 CO_2 的焓值建立能量平衡方程,z 处制冷剂焓值为:

$$h(z) = h_{h,i} + \frac{Q_h + Q_{leak} + q\pi D_{w,i} z}{q_m} \tag{7}$$

式中: $h_{h,i}$—进入预热器入口过冷液体制冷剂焓值,kJ/kg,可由实验装置中测得的温度与压力求得; Q_h、Q_{leak}—预热器对于 CO_2 的加热量、预热器与测试段入口进入 CO_2 的热量。

2 实验结果分析

实验工况: 内径为 0.6 和 1.5 mm,热流密度为 7.5 和 30 kW/m^2,质量流量为 300 、400、500 和 600 kg/m^2·s,蒸发温度为 0、-10、-20、-30 和 40 ℃。图3 为管径 d = 1.5 mm、质量流量 m = 300 kg/m^2·s 热流密度 q = 7.5 kW,不同蒸发温度时换热系数随干度的变化曲线,比较表明: 该工况下随着蒸发温度降低换热系增大,到达某一干度后换热系数发生突降。

图4 为内径 d = 1.5 mm、蒸发温度 T = -10 ℃、热流密度 q = 7.5 kW/m^2,不同质量流量时的换热系数随干度的变化曲线。结果表明: 随着质量流量增加换热系数也相应增加,且同样存在换热系数转折干度,随着质量流量的增加其换热系数发生突降的转折干度相应增大。

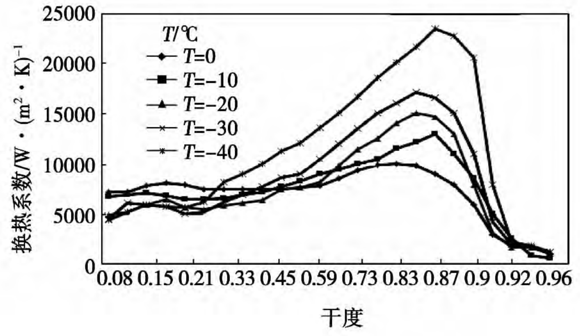

图 3　不同蒸发温度时换热系数随干度变化

Fig. 3 Heat exchange coefficient vs. dryness at various evaporation temperatures

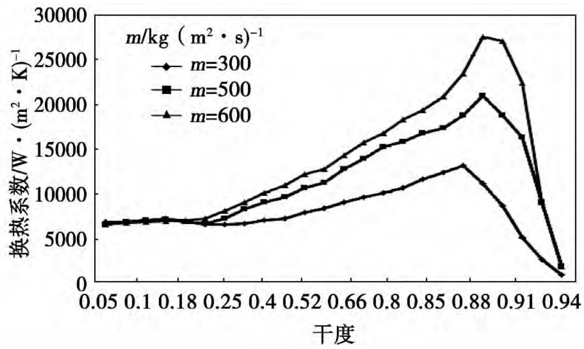

图 4　不同质量流量时换热系数随干度变化

Fig. 4 Heat exchange coefficient vs. dryness at various mass flow rates

图 5 为内径 $d = 1.5$ mm、质量流量 $m = 300$ kg/m^{2}s、热流密度 $q = 7.5$ 和 30 kW/m^2，不同热流密度时的换热系数随干度变化曲线。结果表明：热流密度增加对低干度区域换热起到了明显强化作用，同时导致发生换热系数突降的干度相应减小，干涸发生后换热系数并没有明显强化。

图 6 为蒸发温度 $T = -10$℃、质量流量 $m = 400$ kg/m^{2}s、热流密 $q = 30$ kW/m^2，不同管内径时换热系数随干度的变化曲线，随着管径的减小换热系数转折干度大幅降低，但在低干度区域其换热能力得到了较大提升。

3　理论换热模型比较

通过实验发现 CO_2 在微细通道内低温沸腾换热过程中不同参数对其换热系数有很大影响，且随干度变化呈现出突变性。目前，叠加模型是微细通道内换热系数宏观预测最常采用的模型，针对大管径换热及池沸腾换热研究的关联式在新的实验基础上被重新拟合用于微细通道内换热系数预测，其中文献[3]将微细通道内换热机理归结为由核态沸腾与强制对流换热组成，因此将核态沸腾换热系数与流动沸腾换热系数进行叠加用于计算两相换热系数：

$$h = Sh_{\mathrm{hbc}} + Fh_{\mathrm{f}} \tag{8}$$

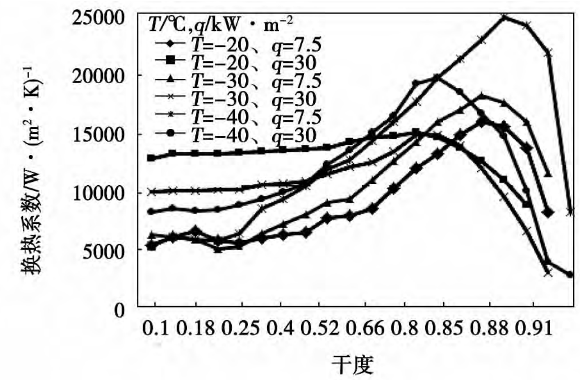

图 5　不同热流密度时换热系数随干度变化

Fig. 5 Heat exchange coefficient vs. dryness at various heat flux densities

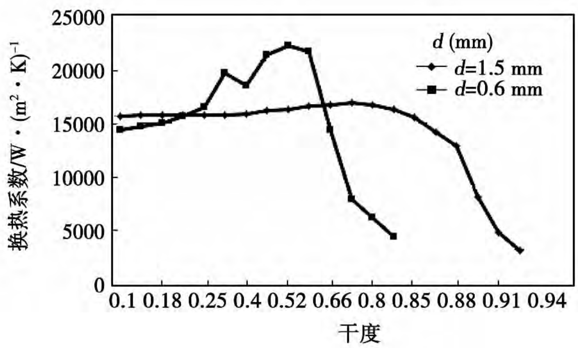

图 6　不同管径时换热系数随干度变化

Fig. 6 Heat exchange coefficient vs. dryness at various tube diameters

目前，核态沸腾换热系数计算通常采用基于文献[4]的池沸腾关联式：

$$h_{\mathrm{nbc}} = 55Pr^{0.12-0.2\log R_p}(-\log_{10}^{P_r})^{-0.55}M^{-0.5}q^{0.67} \tag{9}$$

由于 CO_2 热物性的特殊性，目前的上述数学模型应用于 CO_2 进行换热及压降等预测时偏差较大，现有研究在此基础上进行了进一步进。文献[5]提出，当约束数 Co 大于 0.5 时换热及流态特性发生了显著变化，当约束数 Co 大于 1 时完全可以忽略重力作用，见式(10)。CO_2 作为换热工质，当换热管径小

于 2 mm 时其 Co 值已大于 0.5，文献[6]在以往研究的基础上提出针对 CO_2 的 Kattan - Thome - Favrat 模型，见式(11)，其本质仍是采用了模型叠加的方法进行换热系数预测，在模型中对于局部换热系数的预测引入了干涸角因子，当管内为环状流时定义为干涸角零，当管内全部为层流时干涸角为 2π，因此换热过程中管内润湿周长为 $\frac{D(2\pi-\theta)}{2}$，润湿部分换热系数定义为 h_1，采用核态沸腾与强制对流模型叠加进行计算见式(12)。

$$CO = \left(\frac{\sigma}{g\triangle\rho D^2}\right)^{1/2} \tag{10}$$

$$h = \frac{\theta h_v + (2\pi-\theta)h_1}{2\pi} \tag{11}$$

$$h_1 = ((Sh_{nbc})^3 - h_f^3)^{1/3} \tag{12}$$

文献[8]在上述的研究基础上提出了液体换热系数、气体换热系数、孔隙率以及综合换热系数的理论预测模型，且与当前的实验研究数据进行对比，表明该模型具有较高的预测精度[7-11]，与 1 124 个实验数据比较表明，±30% 预测偏差内精度为71.4%，高于文献[12-17]。因此，选用本研究低温工况下的实验测试数据与参考文献[8]中的 CO_2 沸腾换热系统理论预测模型进行了对比分析，如图 7 - 图 10 所示，在换热系数发生突变干度前也即当前理论研究提出的干涸点前，上述理论模型能够获得较好的预测精度，然而当干涸发生后其换热系数预测精度则明显降低，分别对实验测得所有数据、干涸前数据、干涸后数据的理论预测精度进行了统计分析，如表 2 所示。

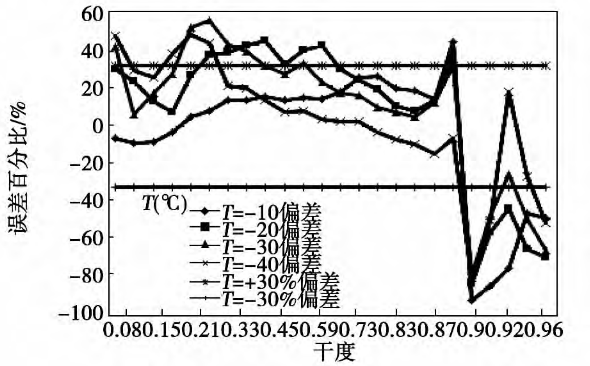

图 7　不同饱和温度时换热系数预测值与实验值比较

Fig. 7 Comparison between the heat exchange coefficient prediction value and the test one at various saturation temperatures

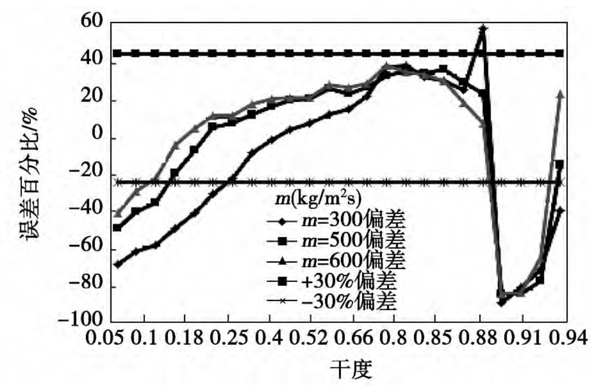

图 8　不同质量流量时换热系数预测值与实验值比较

Fig. 8 Comparison between the heat exchange coefficient prediction value and the test one at various mass flow rates

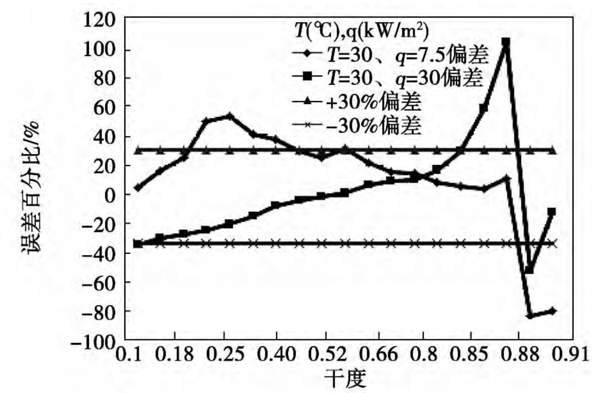

图 9　不同热流密度时换热系数预测值与实验值比较

Fig. 9 Comparison between the heat exchange coefficient prediction value and the test one at various heat flux densities

表 2　换热系数预测统计

Tab. 2 Statistics of the heat exchange coefficients predicted

比较数据类型	λ	ME
所有数据	66.8	27.4
干涸前数据	79.8	21.8
干涸后数据	18.4	59.9

注：λ -30% 误差带内预测精度(%)，ME - 平均误差(%)

4　结　论

针对低温工况时 CO_2 在微细通道管内流动沸腾

换热过程进行了实验研究,对比分析了当前该领域主要理论模型特点,并对实验结果与理论预测模型进行了对比分析,得出结论:

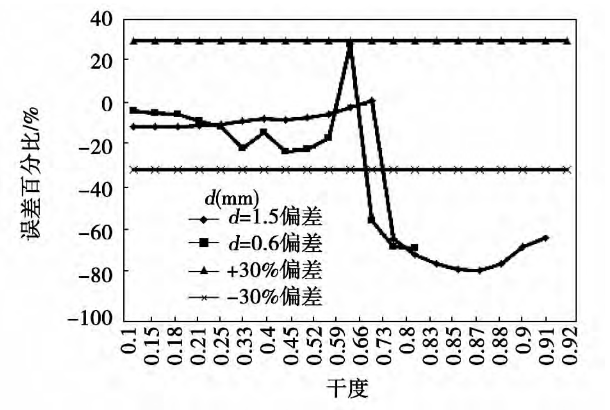

图10　不同管径时换热系数预测值与实验值比较

Fig. 10 Comparison between the heat exchange coefficient prediction value and the test one at various tube diameters

(1) 与常规制冷工质相比 CO_2 具有较优的热物性,使其在再微细通道内中低干度时能够获得较高的换热系数,随着干度增加换热过程会发生干涸现象导致换热性能急剧下降,使其整个换热过程中平均换热系数降低。

(2) 本研究工况换热过程中,在中低干度时换热系数随质量流量、热流密度增加而增加,随管径、蒸发温度减小而增加;在高干度区域随热流密度增加其干涸发生干度降低,干涸现象提前发生导致换热系数下降,且相同工况时管径越小干涸干度越低。

(3) 研究对比表明参考文献[7]中提出的理论模型具有较高预测精度,能够较好预测干涸发生前换热系数,在30%误差带内其预测精度比能够达到79.8%,干涸发生后的预测精度则不理想,现有理论与实验研究结果相比,仍未能获得较好预测干涸发生干度的模型。

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(丛 敏 编辑)

recuperation, exhaust steam recuperation, combined recuperation, thermal performance

EHD/脉动流混合强化传热换热器的传热与阻力特性实验研究 = **Experimental Study of the Heat Transfer and Resistance Characteristics of an EHD (Electrohydrodynamics) /Pulsating Flow Hybrid Enhanced Heat Transfer-based Heat Exchanger** [刊, 汉] YANG Xia, LIU Feng-liang, XIONG Hui, YANG Qing (College of Electromechanical Engineering, Wuhan Engineering University, Wuhan, China, Post Code: 430073) //Journal of Engineering for Thermal Energy & Power. – 2014, 29(3). –256 –261

With water serving as the working medium, experimentally studied was the influence of the electric field and pulsating flow on the heat transfer and resistance characteristics of a single tube heat exchanger. During the test, the voltage value is set at 0 – 40 kV, the frequency of the pulsating flow at f = 1, 2 and 3 Hz and its amplitude A = ii, the time – averaged flow rate inside the tube under the condition of the pulsating flow q = 0.1 – 0.5 m^3/h. The test results show that under the single action of the pulsating flow in the tube path and an identical flow rate and with an increase of f, the heat transfer coefficient α has no conspicuous change, i. e. the single action of the pulsating flow plays an inconspicuous role in enhancing the heat transfer inside the round tube. The single action of EHD can remarkably enhance the heat transfer. When the voltage is relatively small, α grows slowly and when the voltage U > 30 kV, α grows relatively quick but with a continuing increase of the voltage, α tends to become slow and smooth and can maximally increase by 0.12 times. Both the pulsating flow and EHD can play a certain combination role. f has a relatively big influence on α under the condition of the electric field being intensified, especially, when U > 30 kV, the bigger f is, the greater α in the tube path will be. The corresponding α can increase by about 0.25 times. Under the combination action of EHD and the pulsating flow, with an increase of Re, the resistance coefficient in the tube path λ will gradually decline and in case of an identical Re, λ enhanced by the pulsating flow will be conspicuously higher than that without the pulsating flow. Under the condition that both Re and the frequency are identical, whether or not the voltage increases has no influence on λ, i. e. the resistance loss mainly comes from the action of the pulsating flow while the electric field has a relatively small influence on λ. **Key words:** EHD (electrohydrodynamics), pulsating flow, hybrid enhancement, convection-based heat exchange

细微通道内低温 CO_2 流动沸腾换热特性研究 = **Study of the Boiling Heat Exchange Characteristics of a Low Temperature CO_2 flow inside a Micro-channel** [刊, 汉] ZHANG Liang, LIU Jian-hua, DING Yang (Refrigeration Technology Research Institute, Shanghai University of Science and Technology, Shanghai, China, Post Code: 200093), YE Fang-ping (Zhejiang Xinjin Air Conditioning Equipment Co. Ltd., Longquan, China, Post Code: 323700) //Journal of Engineering for Thermal Energy & Power. – 2014, 29(3). –262 –266

With CO_2 serving as the refrigeration agent, experimentally studied were the boiling heat exchange characteristics of a low temperature fluid inside a micro-channel having an inner diameter of 0.6 mm and 1.5 mm respectively and quantitatively analyzed was the influence of various parameters under various operating conditions on the heat exchange coefficient when the low temperature fluid flows through the pipeline under the test condition. The research results show that the model proposed in the literature[7] has a relatively high prediction precision. When the error was controlled in a range of 30%, the theoretical prediction precision ratio (ratio of the test data and those obtained from the model) of the heat exchange coefficient before the dry-out takes place can be up to 79.8% and the average deviation can be up to 21.8% while after the dry-out has taken place, the theoretical prediction precision ratio of the heat exchange coefficient can be up to 18.4% and the average deviation can be up to 59.9%. **Key words:** carbon dioxide, heat exchange coefficient, dry-out, boiling heat exchange of a flow

600 MW **低质量流速垂直管圈超临界煤粉锅炉设计开发 = Development of the Design of a 600 MW Low Mass Flow Speed Supercritical Pulverized Coal-fired Boiler With a Vertical Tube Coil [刊,汉]** ZHANG Man, ZHANG Hai, LU Jun-fu, WU Yu-xin, ZHANG Da-long (Education Ministry Key Laboratory on Thermal Science and Power Engineering, Department of Thermal Energy Engineering, Tsinghua University, Beijing, China, Post Code: 100084) //Journal of Engineering for Thermal Energy & Power. - 2014, 29(3). -267 -273

Analyzed were commonly seen specific features of several water wall arrangement modes of supercritical pulverized coal-fired boilers, tangentially and wall opposed combustion modes and principles for choosing the key parameters of a vertical tube coil water wall and designed and developed was a low mass flow speed vertical tube coil 600 MW supercritical pulverized coal-fired boiler. Under the full load operating condition, the design mass flow speed of the water wall was 940 kg/m^2s. The version under discussion adopted the low mass flow speed vertical tube technology developed by the Siemens Company. In combination with such merits as simple in the structure of vertical water walls and the self-compensated characteristics of the working medium at a low mass flow speed, the throttle orifice was removed, thus avoiding the complex structure of both water walls and lower headers, and at the same time, eliminating the safety hazard as the tube walls have exceeded the allowable temperature, created by any clogging in the structure of the throttle orifice during operation. An intermediate mixing header was provided between the upper and lower water wall in the furnace to mix with the working medium coming from the lower part of the furnace and minimizing the temperature deviation at the working medium side caused by non-uniform heat absorption and difference in the structure of the furnace. Finally, the authors described the structure of the boiler under discussion and predicted its performance. **Key words:** 600 MW, supercritical, low mass flow speed, vertical tube coil, design

掺水燃油中水珠粒径对锅炉热效率影响 = Influence of the Water Drop Particle Diameter in Fuel Oil Dilu-