

油涡轮流道的几何参数对性能影响分析及验证

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摘 要 “主油泵-油涡轮增压泵”型式的油系统已广泛应用于 600 MW 及以上等级大型汽轮机组。根据油涡轮各部件流道几何特点、工作原理及系统对于油涡轮流体动力性能要求,在一定假设条件下推导出油涡轮设计过程中主要几何参数与外特性的关系方程。结合某 1 000 MW 汽轮机组油系统研制要求设计出了两个油涡轮方案,采用 CFD 软件进行多方案、多工况数值模拟和性能预测,揭示了所推导出的流道几何参数对油涡轮流体动力性能影响关系。通过油系统试验获得了两个方案的外特性,验证性能预测的可靠性,进一步证明分析方法的可行性,为完善这类油涡轮的设计方法奠定了一些基础。

关 键 词 汽轮机组;油涡轮;几何参数;性能预测

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引 言

油系统是火电和核电用汽轮机组的重要组成部分之一,是保证汽轮机组能够正常运行的关键系统。单机 300 MW 及以下的汽轮机组曾广泛采用“主油泵-多孔射油器”供油系统,对于 600 MW 及以上的机组,射油器的工作流量已接近其临界流量,已不能满足容量进一步增大的机组用油量。为此,国内外广泛采用“主油泵-油涡轮增压泵”型式的油系统。该类油系统具有效率高、供油量大、体积小等优点,而且在非设计工况下仍可以维持较高效率。多孔射油器供油系统效率约为 25%,而油涡轮增压泵供油系统的总效率超过 50%。油涡轮增压泵系统具有明显的节能效果,提高了机组经济性。油涡轮工作流体动力性能对整个油系统起着至关重要的作用,其稳定性直接决定汽轮机组的安全可靠性^[1]。

目前,国内所使用的“主油泵-油涡轮增压泵”型式油系统成套设备均为国外进口或引进国外设

计。迄今为止,我国对按照流体动力性能要求来确定油涡轮流道几何参数等关键技术都未进行深入研究,仅采用模化设计方法对油涡轮过流部件的流道进行多方案设计。因此,有必要借鉴水轮机的设计理论^[2~3],借助数值模拟和性能预测来探索油涡轮的流体动力设计理论和方法^[4]。

结合某 1 000 MW 汽轮机组“主油泵-油涡轮增压泵”型式油系统研究开发,分别设计出两个油涡轮(以下称为方案 A 和方案 B),利用 CFD 软件进行多方案、多工况数值模拟和性能预测,揭示了所推导出的流道几何参数对油涡轮流体动力性能的影响。通过油系统试验获得了两个方案的外特性,验证性能预测的可靠性,进一步证明其分析方法的可行性,为完善油涡轮的设计方法奠定了一定基础。

1 油涡轮的主要几何参数确定

油涡轮是油系统的“心脏”,它直接把油流的能量转换为机械能。转轮流道的形状及几何参数对油涡轮的过流能力有很大影响,而且还直接影响着油涡轮的水力效率、工作稳定性以及对变工况的适应性,其基本结构如图 1 所示。

1.1 叶片反动度

假设过流介质为理想流体^[5],通过油涡轮的流体所输出的能量被完全转换为机械能,则:

$$N = M_z \omega = P_T Q_s \quad (1)$$

式中: M_z —油涡轮的输出力矩, $N \cdot m$; ω —转轮的旋转角速度, rad/s ; Q_s —通过油涡轮的体积流量, m^3/s ; P_T —流体压力减少量, Pa 。

图 2 为叶片进出口的速度三角形,其中: U —圆周速度; W —相对速度; V —绝对速度; α —叶片绝对

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速度方向角; β —叶片相对进出口角; 下标 1 表示进口, 下标 2 表示出口。

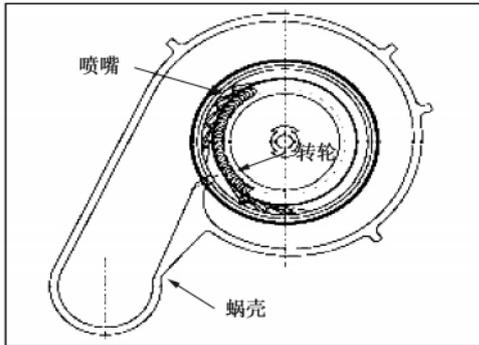


图 1 油涡轮的基本结构
Fig. 1 Basic structure of an oil turbine

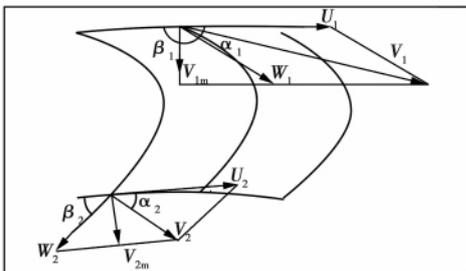


图 2 叶片进出口速度三角形
Fig. 2 Velocity triangle at the inlet and outlet of a blade

根据动量矩定理, 流道进口流体对轴心的动量矩与流道出口流体对轴心的动量矩之差, 等于油涡轮输出的力矩, 即:

$$M_z = \rho Q_s (V_{1u} R_1 - V_{2u} R_2) \quad (2)$$

式中: ρ —所输送流体的密度 kg/m^3 。

将式(2)代入式(1)化简得:

$$P_T = \rho (U_1 V_{1u} - U_2 V_{2u})$$

由图 2 可知, 根据余弦定理得:

$$W_1^2 = V_1^2 + U_1^2 - 2V_1 U_{1m}$$

$$W_2^2 = V_2^2 + U_2^2 - 2V_2 U_{2m}$$

$$P_T = \frac{\rho}{2} (V_1^2 - V_2^2) + \frac{\rho}{2} (U_1^2 - U_2^2) + \frac{\rho}{2} (W_2^2 - W_1^2) \quad (3)$$

式中右边第一项表示流体进入转轮后动压变化量, 第二项和第三项之和表示静压变化量。

由此可得到油涡轮的反动度:

$$\Omega = \frac{U_1^2 - U_2^2 + W_1^2 - W_2^2}{2(U_1 V_{1u} - U_2 V_{2u})} \quad (4)$$

根据叶片进出口速度三角形, 得:

$$V_{1u} = U_1 - W_1 \cos \beta_1$$

$$V_{2u} = U_2 - W_2 \cos \beta_2$$

将上式代入式(4)化简得:

$$\Omega = \frac{1}{2} (U_1^2 - U_2^2 + W_1^2 - W_2^2) \times [(U_1^2 - U_2^2) + (U_2 W_2 \cos \beta_2 - U_1 W_1 \cos \beta_1)]^{-1} \quad (5)$$

然而, $U_1 = \frac{n\pi D_1}{60}$, $U_2 = \frac{n\pi D_2}{60}$ 由此化简式(5)

得:

$$\Omega = \psi (D_1^2 - D_2^2 + W_1^2 - W_2^2) \times [(D_1^2 - D_2^2) + (D_2 W_2 \cos \beta_2 - D_1 W_1 \cos \beta_1)]^{-1}$$

式中: D_1 、 D_2 —叶片进出口处转轮直径; ψ 与转速 n 相关, 即 $\psi = f(n)$ 。

根据油系统性能调节要求, 油涡轮的出力与通过的流量基本成线性关系, 与转速在一定范围内变化基本无关。主要靠喷嘴喷出的高速流体冲动转轮做功。因此油涡轮的叶栅反动度必须设计得很小, 但是又不能没有反动度。根据油涡轮反动度表达式, 要使 Ω 取值较小, 则应满足以下条件:

- (1) $D_1^2 - D_2^2 \approx W_2^2 - W_1^2$
- (2) $(D_2 W_2 \cos \beta_2 - D_1 W_1 \cos \beta_1)_{\max}$

如果以 t 表示栅距, 则叶片通道进口垂直于 W 的宽度为 $t \sin \beta_1$, 出口垂直于 W 的宽度为 $t \sin \beta_2$ 。对于油涡轮叶道, 应满足 $t \sin \beta_2 < t \sin \beta_1$ 时叶道收敛, 相对速度不断增加, 即 $W_2 > W_1$ 。然而 W_2 的增大应满足 $D_1^2 + W_1^2 \approx D_2^2 + W_2^2$ 。在 β_1 不断趋近于 180° 时, $-\cos \beta_1$ 取最大值; β_2 不断趋近于 0° 时, $\cos \beta_2$ 取最大值; 此时满足条件 2 使得 Ω 取值较小。如若 $\beta_1 + \beta_2 \approx 180^\circ$ 时, 叶片为强前弯型, 不仅满足反动度较小的要求, 而且性能更加良好。

1.2 叶片进口角 β_1

根据叶片进口速度三角形, 得出:

$$V_{1u} = U_1 + V_{1m} \cot \beta_1$$

$$V_{2u} = U_2 + V_{2m} \cot \beta_2$$

根据反动度对叶片相对进出口角 $\beta_1 + \beta_2 \approx 180^\circ$ 的要求, 则:

$$P_T = \rho (U_1 V_{1u} - U_2 V_{2u}) = \rho [U_1^2 + U_2^2 + (U_1 V_{1m} - U_2 V_{2m}) \cot \beta_1]$$

油涡轮的理论功率为:

$$N_T = P_T Q_T$$

然而 $Q_T = \pi D_1 b_1 V_{1m}$, 则:

$$N_T = \rho (U_1^2 + U_2^2) Q_T + \rho \left(\frac{U_1}{\pi D_1 b} - \frac{U_2}{\pi D_2 b} \right) \frac{Q_T^2}{\tan \beta_1}$$

式中: b —叶片高度, mm。

当 $\beta_1 = 90^\circ$ 时, 油涡轮的功率线是通过坐标原点的一条直线。当 $\beta_1 > 90^\circ$ 时, 功率线则为直线下方的抛物线。且随着 β_1 的不断趋近于 180° , 抛物线开口增大, 流量-功率线更加趋于平坦。

1.3 叶片高度 b

叶片高度 b 的大小与油涡轮流量、转轮叶片堵塞系数有关。转轮进口处的子午速度为 V_{1m} , 则叶片高度 b 由下式决定:

$$b = Q_T / \pi D_1 V_{1m} \quad (6)$$

τ_1 是由于叶片厚度使流道进口截面堵塞的系数, 其表达式为:

$$\tau_1 = \frac{\pi D_1 \sin \beta_1 - Z \delta_1}{\pi D_1 \sin \beta_1} \quad (7)$$

式中: Z —叶片数; δ_1 —叶片进口厚度, mm。

合并式(6)和式(7)得:

$$b = \frac{(1 - \tau_1) Q_T \sin \beta_1}{Z \delta_1 V_{1m}} \quad (8)$$

在式(8)中其它参数都已知的情况下, 叶片高度 b 主要由叶片数和叶片厚度决定, 也就是应该选取合适的叶片数和叶片形状。

1.4 叶片数 Z

叶片数 Z 对油涡轮的性能有较大影响, 叶片数和流动损失密切相关。叶片数增加时, 旋涡损失减少, 摩擦损失增大, 但其中必有最佳叶片数。

叶栅稠密度:

$$\sigma = l_m / t_m \quad (9)$$

式中: l_m —叶片中线的长度; t_m —转轮平均半径处的栅距。

当叶片型线用单圆弧绘制且 β_1 和 β_2 相差不大时, 叶片平均长度为:

$$l_m = (R_2 - R_1) / \sin \frac{\beta_1 + \beta_2}{2}$$

将上述两关系式代入式(9)得:

$$Z = \pi \sigma \frac{D_1 + D_2}{D_1 - D_2} \sin \frac{\beta_1 + \beta_2}{2}$$

故选择合适的叶片数即为选择最佳叶栅稠密度, 需根据经验得到。

2 1 000 MW 汽轮机组油涡轮的设计

油涡轮的运行工况较为复杂, 必须动态适应汽轮机组的工况变化要求, 针对于某 1 000 MW 汽轮机组配套油系统要求, 按照上述的理论确定流道几何参数, 设计出了 A、B 两个方案的油涡轮。其中, 油涡轮工作介质为 32 号透平油, 设计流量 Q_T 为 6 900 L/min, 油涡轮设计转速 n 为 1 650 r/min, 油涡轮进出口压差 0.855 MPa。油涡轮的几何参数如表 1 所示, 全流道三维模型如图 3 所示。

表 1 油涡轮关键几何参数

Tab. 1 Key geometrical parameters of an oil turbine

	A 方案	B 方案
喷嘴数目/个	20 组	18 组
喷嘴直径/mm	11	10.5
射流角 $\beta_1 / (^\circ)$	144	147.5
转轮直径/mm	280	270
叶片高度/mm	29	32
叶片数目/个	58	58

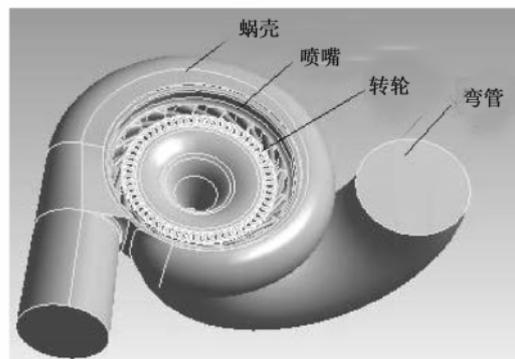


图 3 油涡轮全流道三维模型

Fig. 3 Full flow pass 3D model for oil turbines

3 数值模拟及性能预测

3.1 数值模拟验证方法

基于 CFX 对 A、B 两个油涡轮进行数值模拟分析, 创建蜗壳、喷嘴、转轮和弯管 4 个计算区域; 采用

CFX - mesh 对模型进行前处理得到四面体混合网格,两模型的网格数在 $(2.6 \sim 2.9) \times 10^6$ 之间。工作介质为 32 号透平油;湍流模型采用 RNGk - ε 模型^[6];固体壁面采用无滑移边界条件,喷嘴和转轮交界面选取转子冻结模型,通过改变参考系处理动静之间的流动,同时保持静止区域的相对位置。

3.2 数值模拟结果对比与理论分析

在汽轮机油系统中,油涡轮与增压泵同轴,油涡轮将油流能量转换为旋转机械能驱动增压泵向主油泵入口供油。由此可见,油涡轮应该在流量 - 功率 - 转速关系曲面图上的某一点工作。

经过迭代计算后,得到 A、B 两个油涡轮在不同工况下转速、流量和输出轴功率之间的关系,如图 4 所示。

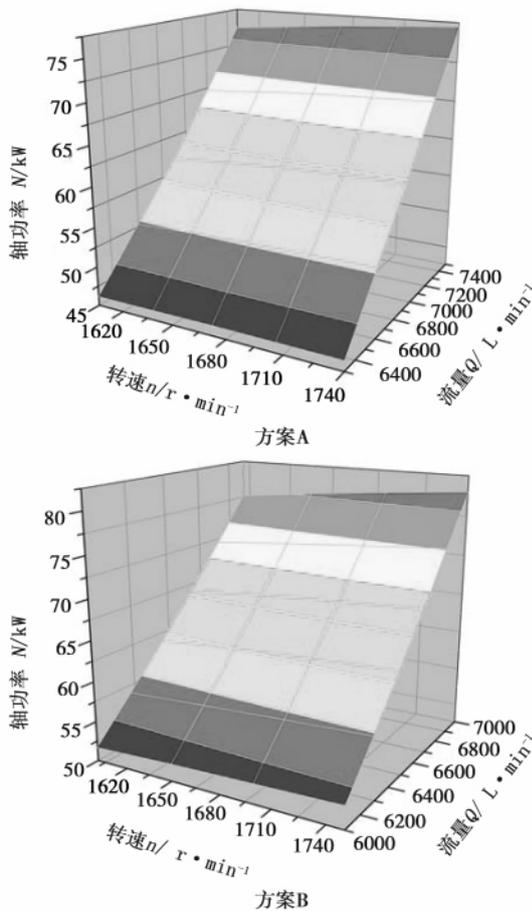


图 4 流量 - 功率 - 转速关系图

Fig. 4 Chart showing the relationship of flow rate-power-rotating speed

从图中可以看出,流量 - 功率 - 转速关系曲面图接近于平面,主要由于油涡轮的调节特性:在固定

流量下,油涡轮的“n - P”曲线较平坦;在固定转速下,油涡轮的“Q - P”曲线尽可能线性变化。由此可见,A、B 油涡轮均满足其调节特性要求。

在设计工况点处,增压泵所需轴功率为 53.5 kW,而 A 油涡轮输出轴功率为 59.8 kW,B 油涡轮输出轴功率为 72.03 kW,均能满足增压泵的输入轴功率要求。B 油涡轮输出轴功率较 A 油涡轮高,主要由于在相同流量下,B 油涡轮的喷嘴数和喷嘴射流直径都相对减少,导致流体从喷嘴流出的绝对速度较大,更多的动能转换为转轮的旋转机械能。

图 5 为 A、B 油涡轮的流量 - 效率 - 转速关系图。流体动力效率(不包括机械损失、容积损失):

$$\eta_h = \frac{M_z \omega}{\Delta p Q} \times 100\%$$

式中: η_h - 流体动力学效率; Δp - 油涡轮的进出口压力差,MPa。

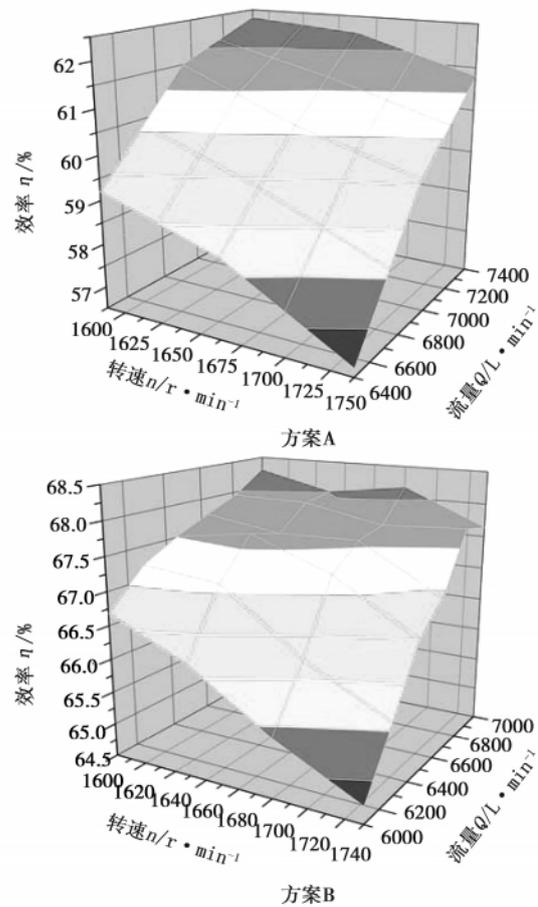


图 5 流量 - 效率 - 转速关系图

Fig. 5 Chart showing the relationship of flow rate-efficiency-rotating speed

由图5可以看出,在油系统工作流量范围内,A油涡轮的效率随流量变化较大,57%~62%之间。而B油涡轮则不同,流量在6 200~7 000 L/min范围内变化时,其效率值稳定在67%~68%之间,性能更好。

4 试验验证

为了更好地了解以上两个方案油涡轮性能,制造厂在汽轮机油系统试验台上更换A、B两个油涡轮,并与相同的主油泵和增压泵进行实机联动试验。通过调节油涡轮的流量来保持油涡轮的进出口压差为0.85 MPa不变,监测增压泵出口压力和主油泵流量判断其能否满足油系统调节性能要求。数值模拟和真机试验所得的油涡轮“转速-输出轴功率”对比曲线如图6所示。

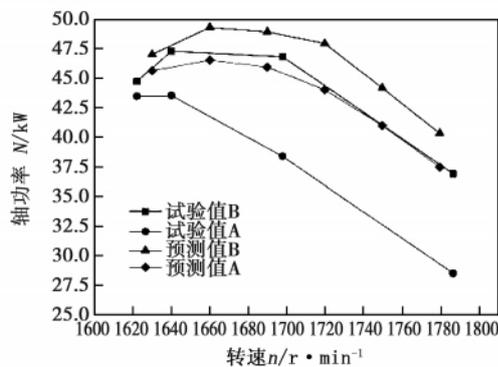


图6 油涡轮转速-功率对比曲线

Fig. 6 Curves showing a comparison of the rotating speed and power of the oil turbine

从转速-功率曲线来看,随着转速的增大,油涡轮输出轴功率先增大后减小,转速在1 600~1 700 r/min时输出轴功率较稳定。根据喷嘴与叶片的耦合关系,喷嘴射流角 β_1 的增大会引起油涡轮输出轴功率的增大,从而印证了射流角 β_1 增大后叶片为强前弯型,涡轮反动度较小,油涡轮输出轴功率随转速变化趋于平缓,即“ $n-P$ ”曲线更加平坦。由图6可知,在油涡轮整个运行范围内,输出轴功率的预测值与试验值吻合较好,输出轴功率的预测值误差最大为8%。由此说明,所提出的理论和方法设计出的两个方案的油涡轮均能为增压泵提供足够的轴功

率。同时也验证了所采用的性能预测方法是正确的,是可以借鉴的。

5 结论

(1) 根据油涡轮各部件流道几何特点、工作原理及系统对油涡轮流体动力性能要求,推导出油涡轮设计过程中主要几何参数与外特性的关系方程。分析油涡轮关系方程可知,在叶片进口角 β_1 不断趋近于 180° , β_2 不断趋近于 0° 时,油涡轮的反动度较小,且随着 β_1 的不断增大,流量-功率曲线开口增大更加趋于平坦,能更好的满足油系统性能调节要求。

(2) 对方案A、B两个油涡轮进行实机试验,工程实践表明A、B油涡轮都能为增压泵提供足够的轴功率,但是方案B油涡轮更优。由此验证了所研究性能预测方法的可靠性,进一步证明理论和方法的可行性,为油涡轮的设计方法奠定一定基础。

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(丛敏 编辑)

生物质燃料在燃煤锅炉脱硝中的应用 = **Application of Bio-mass Fuels in the Denitration of Coal-fired Boilers** [刊, 汉] DING Xu-hong, ZHAO Zeng-li, CHANG Sheng, LI Hai-bin (Chinese Academy of Sciences Key Laboratory on Renewable Energy and Natural Gas Hydrate, Guangzhou Energy Source Research Institute, Chinese Academy of Sciences, Guangzhou, China, Post Code: 510640) // Journal of Engineering for Thermal Energy & Power. - 2013, 28(3) . - 223 ~ 228

The denitration technology is regarded as an important measure for reducing NO_x and in recent years, as a high efficiency low cost denitration technology, the bio-mass combustion denitration has attracted wide-ranging attention. The fundamentals of the bio-mass combustion denitration were described and compared with those of the traditional selective non-catalytic reduction (SNCR) method and selective catalytic reduction (SCR) method. In combination with the recent advances in the study of bio-mass and its derivative fuels applied in the denitration of coal-fired boilers both at home and abroad, the authors have concluded that the combustion of both bio-mass and its derivative fuels could all achieve a relatively good denitration effectiveness and the cost of doing so is lower than that of the traditional denitration methods. Finally, they summarized the merits and demerits of various bio-mass combustion denitration modes, thus providing reference for their further research and applications. **Key words:** bio-mass, combustion, denitration

油涡轮流道的几何参数对性能影响分析及验证 = **Analysis and Verification of the Influence of the Geometrical Parameters of the Flow Passage of an Oil Turbine** [刊, 汉] GOU Qiu-qin, LAI Xi-de, ZHANG Xiang, SONG Dong-mei (College of Energy Source and Environment, West China University, Chengdu, China, Post Code: 610039) // Journal of Engineering for Thermal Energy & Power. - 2013, 28(3) . - 229 ~ 233

“Main oil pump-oil turbine-based booster pump” type oil systems have been widely used in 600 MW and above class large-sized steam turbine units. In the light of the geometrical features and the working principle of the flow passages of various parts and components of an oil turbine and requirements of the system for the fluid dynamic performance of the oil turbine, under certain presumption conditions, the correlation equations between the main geometrical parameters and the external characteristics of the oil turbine during its design stage were derived. In combination with the requirements for development of the oil system of a 1000 MW steam turbine unit, two versions of the oil turbine were designed and the CFD software was employed to conduct a numerical simulation of multiple versions and operating conditions, and performance prediction to reveal the relationship of the geometrical parameters of the flow passages influencing the fluid dynamic performance of the oil turbine thus derived. Through the test of the oil system, the external characteristics of the two versions were obtained and the reliability in predicting the performance

was verified. It has further proven that the analytic method is feasible ,thus laying a foundation for improving the methods for designing the oil turbine and its kind. **Key words:** steam turbine unit ,oil turbine geometrical parameter ,performance prediction

三轴燃气轮机喷水冷循环研究 = **Study of a Three-shaft Gas Turbine-based Water-spraying Inter-cooling Cycle** [刊 汉] DONG Bin ,LIN Feng ,MA Zheng-jun ,LI Dong-ming (CSIC No. 703 Research Institute ,Harbin ,China ,Post Code: 150078) //Journal of Engineering for Thermal Energy & Power. - 2013 28(3) . - 234 ~ 240

Calculated was the performance of a three-shaft gas turbine-based water spraying inter-cooling cycle and studied was the variation tendency of the main parameters when the ratio of the inter-cooling water quantity sprayed and the inlet air quantity was between 0% to 2% . On this basis ,a test of which the ratio of the inter-cooling water quantity sprayed and the inlet air quantity was 0.45% was conducted. The test results show that with 0.8 to 1.0 Ne of the rated load serving as the reference ,when the exhaust gas temperature of the LP turbine is kept unchanged ,the output power will increase by about 7.5% ,the efficiency will go up by around 1.8% (relative value) ,the rotating speed of the HP shaft will be basically constant while that of the LP shaft will increase about 1.8% ,the NO_x emissions will decrease by about 10% and the non-uniformity of the temperature field at the tail portion of the turbine and the vibration parameter of the unit will be normal. It can be seen that the inter-stage water spraying and inter-cooling will become an effective means for improving the performance of a unit. **Key words:** gas turbine ,water-spraying inter-cooling ,test

烟气热源有机朗肯循环系统工质选择 = **Working Medium Selection for Flue Gas Heat Source Organic Rankine Cycle Systems** [刊 汉] LIU Guang-lin ,CHEN Qi-cheng ,ZHANG Bing(National Key Laboratory on New Energy Source Power Systems ,Beijing City Key Laboratory on Low Grade Energy Sources Multi-phase Flow and Heat Transfer ,North China University of Electric Power ,Beijing ,China ,Post Code: 102206) //Journal of Engineering for Thermal Energy & Power. - 2013 28(3) . - 241 ~ 245

Under the condition of the heat source of flue gases being at different temperatures ,studied was the relationship between the thermal efficiency and the organic working medium of a subcritical organic Rankine cycle power generation system and applications of the organic working medium thus optimized in a recuperator system. It has been found that for a subcritical organic Rankine cycle system ,at given heat source parameters ,when the temperature of the organic working medium at the inlet of the expander is kept constant ,the system will attain its maximal efficien-