管程转子组合式强化传热装置工业试验研究

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: 介绍了当前节能降耗的迫切需要和转子组合式强化传热装置——"洁能芯"的技术原理。论述了 2 号机组甲、乙侧凝汽器性能试验内容, 试验方法和试验计算结果, 计算分析了安装转子组合式强化传热装置后凝汽器性能及机组经济性。安装转子组合式强化传热装置后, 凝汽器端差降低 2.79 ℃, 真空提高 2.97 kPa, 冷却水流量减小 9.8%; 凝汽器在相同进水温度、相同真空下, 机组带负荷能力提高 19.7%; 凝汽器水阻增加 19.52 kPa。工业试验和计算分析结果表明, 转子组合式强化传热装置应用于电厂凝汽器可提高其性能。

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: TK263. 6　　　　　: A

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1.1

[1]

[2]:

10

33. 8%,　　　　　　6~7

2010　　　　　　　　360 g/(kWh)

[3]

1 kPa,

0.7% ~ 1%;

2. 16 mm　, 　　　　　　54%,

50%。

[4]

——"　　　"

U

[5]

: (1)

; (2)　　　　; (3)

; (4)

[6]

[7],

49%。　　　　　"

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1 。

图 1 转子组合式强化传热与自清洁装置

2 000 mm ,

" " , ,

3 ~4 ℃, " "

。 , 125 MW

5 ℃, 95.12

kJ/(kWh), 3.66 kg/(kWh)。 ,

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1.2

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[8]

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2007 4 25 ,4 27

8 : 40 MW

; 50 MW,

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、 ; 30 MW ,

、 ; 40 MW ,

; 40 MW ,

; 47 MW

; 48 MW

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2.1

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60 min, ;

30 s; 2 min。

《 》B/T3344—93 《 》

(1978) 。

(1) 。

(2) ,

, (3) 。

,

, (4) (5) , 《

》B/T3344—93。

2.1.1 对数平均温差 θ_{mT}

$$\theta_{mT} = \frac{\theta_{1T} - \theta_{2T}}{\ln\left(\dfrac{\theta_{1T}}{\theta_{2T}}\right)} = \frac{(t_{sT} - t_{1T}) - (t_{sT} - t_{2T})}{\ln\left(\dfrac{t_{sT} - t_{1T}}{t_{sT} - t_{2T}}\right)} =$$

$$\frac{t_{2T} - t_{1T}}{\ln\left(\dfrac{t_{sT} - t_{1T}}{t_{sT} - t_{2T}}\right)} \tag{1}$$

: θ_{mT} — , ℃; θ_{1T} — , $\theta_{1T} =$ $t_{sT} - t_{1T}$, ℃; θ_{2T} — , $\theta_{2T} = t_{sT} - t_{2T}$, ℃; t_{sT} —

, ℃; t_{2T}—

, ℃; t_{1T}—

, ℃。

2.1.2 凝汽器热负荷 Q_T

$$Q_T = G_{wT} \cdot C_{PT} \cdot (t_{2T} - t_{1T}) \tag{2}$$

: Q_T— , kW; G_{wT}—

, kg/s; C_{PT}—

, kJ/(kg·℃)。

2.1.3 凝汽器总体传热系数 K_T

$$K_T = \frac{Q_T}{A \times \theta_{mT}} = \frac{G_{wT} \cdot C_{PT} \cdot 10^3}{A \times (t_{2T} - t_{1T})} \times \ln\left(\frac{\theta_{1T}}{\theta_{2T}}\right) \tag{3}$$

: K_T— , W/(m²·℃);

A— , m²。

2.1.4 冷却水流量对试验传热系数 K_T 的修正系数 F_v

$$K_C = K_T F_v \tag{4}$$

$$F_v = \sqrt{v_D / v_T} \tag{5}$$

: K_C— ; F_v—

; v_T— , m/s; v_D—

, m/s。

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50、40 30 MW , 、

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2.2.1 机组功率 49.66 MW, 凝汽器压力 5.15 kPa, 50 MW 负荷工况

(1) (

) 4 215.86 m³/h, 10.73 ℃,

16.70 ℃, 294 717.61 MJ/h,

2.63 kW/(m²·℃);

(2) (

) 3 468.50 m³/h, 11.82 ℃,

15.61 ℃, 226 685.83 MJ/h,

1.94 kW/(m²·℃);

(3) 、

3 468.50 m³/h ,

1.29 kPa。

2.2.2 机组功率 40.64 MW, 凝汽器压力 4.58 kPa, 40 MW 负荷工况

(1) (

) 4 164.47 m³/h, 11.07 ℃,

14.04 ℃, 244 921.18 MJ/h,

2.27 kW/(m²·℃);

(2) (

) 3 419.77 m³/h, 11.91 ℃,

13.20 ℃, 189 100.16 MJ/h,

1.70 kW/(m²·℃);

(3) 、

3 419.77 m³/h ,

1.03 kPa。

2.2.3 机组功率 30.90 MW, 凝汽器压力 4.10 kPa, 30 MW 负荷工况

(1) (

) 4 155.148 m³/h, 11.97 ℃,

11.14 ℃, 194 036.57 MJ/h,

1.82 kW/(m²·℃);

(2) (

) 3 378.68 m³/h, 13.18 ℃,

9.93 ℃, 140 647.84 MJ/h,

1.26 kW/(m²·℃);

(3) 、

3 378.68 m³/h ,

0.86 kPa。

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2.3 、

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40 MW

, 40、47

48 MW 3 。

2.3.1 甲侧凝汽器 40 MW 单侧运行

40.36 MW, 14.92 kPa,

3.94 ℃, 4 725.94 m³/h,

29.08 ℃, 20.86 ℃,

574 320.97 MJ/h,

2.3.2 乙侧凝汽器 40 MW 单侧运行　　　　　　　　　　　　　　　　　　　　　　　　　　　。、
　　　　　　40.44 MW,　　　　　　12.02 kPa,　　　　　　　　　　　　　　　1　　。
　　　　5.55 ℃,　　　　　　4 262.66 m³/h,　　，　　　　　，　　　　　　　　　　　　19.52 kPa,
　　　25.87 ℃,　　　18.07 ℃,　　　　　　　　　23.28。
460 702.54 MJ/h。
　　　　　　　　　　　　　　　　3.94　　　　1　　、

℃,　　　　　　　11.95 kPa。

	（　　　）	（　　　）
/m³·h⁻¹	4 215.86	3 468.50
/m·s⁻¹	1.80	1.48
/m	1.87	3.86
	11.37	34.65
/m	0	1.99
	0	23.28

2.3.3　在 40 MW 负荷下,甲乙凝汽器单侧运行比较
　　　　　　　　　　　　1.61 ℃,
　　　　9.8%(　　463.27 m³/h),
2.79 ℃,　　　　　　　　　　　　　2.9
kPa。　　　　　　　　　　　　　,
　　　　11.95 kPa,　　　　　　　2.97 kPa。

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2.3.4　乙侧凝汽器 47 MW 单侧运行
　　　　　　47.0 MW,　　　　　14.57 kPa,
　　　6.07 ℃,　　　　4 196.07 m³/h,
　　　29.74 ℃,　　　17.58 ℃,
520 974.70 MJ/h。

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　　　　　。　　　　　　　　　　，
　　　2.79 ℃,　　　　　2.97 kPa;
　　9.8%;
,

℃,　　　　　14.47 kPa。
2.3.5　乙侧凝汽器 48 MW 单侧运行
　　　　　48.31 MW,　　　　15.38 kPa,
　　　7.00 ℃,　　　　4 183.35 m³/h,
　　　30.11 ℃,　　17.41 ℃,
525 621.04 MJ/h。
　　　　　　　　　　　　　　　3.94

　　　　　19.7%;
,　　　　　　　19.52 kPa,
　　。

℃,　　　　　　15.20 kPa。
2.3.6　单侧运行试验结果分析结论
　　（1）　　　　　　　　　,

2.97 kPa;
　　（2）　　　　　　　　　　　　9.8%;
　　（3）　、　　　　　　　　　　（　15
kPa),
　　　48.31 MW,
　　　　　　　40.36 MW,
　7.95 MW,　　19.7%。
2.4

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　　（编辑　单丽华）
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/ ＝**A Study of Methanol Synthesis, Distillation Simula-
tion and Load Variation of a Methanol/Power Cogeneration System**[，] / WANG Ming-hua, LI Zheng, FENG
Jing, et al (Thermal Energy Engineering Department, Tsinghua University, Beijing, China, Post Code: 100084)//Journal
of Engineering for Thermal Energy &Power. — 2008, 23(4). —363～368

As one of the most promising clean-coal-fired power generation technologies, an integrated gasification combined cycle
(IGCC) system coupled with a methanol synthesis system not only can enhance the load regulating ability of the system
but also simultaneously improve the cost-effectiveness of the IGCC power plants. The flow path in the process of C301
type LP tube row gas-phase methanol synthesis and three-tower distillation was simulated through the use of software AS-
PEN, and a tactic for load-variation regulation based on the divided flow ratio and circulation ratio has been proposed.
With the help of a four-quadrant chart, the methanol load variation range and the ability of regulating power loads under
the condition of different divided flow ratios and circulating ratios were shown. In addition, a load variation regulation of
distillation processes was accomplished through a change of the reboiling and condensing loads of various distillation tow-
ers. **Key words**: methanol/power cogeneration, IGCC (integrated gasification combined cycle), methanol synthesis,
methanol distillation, load variation

＝**A Study of the Active Balancing Technology for the Far End Optimiza-
tion of a Cantilever Rotor**[，] / SU Yi-ru, HE Li-dong, FENG Wei (Diagnosis and Self-healing Engineering Re-
search Center, Beijing University of Chemical Technology, Beijing, China, Post Code: 100029)//Journal of Engineering
for Thermal Energy &Power. — 2008, 23(4). —369～372

To guide the application of an active balancing device in industrial cantilever rotors, a cantilever rotor test stand was set
up to perform an experimental study of active balancing technology. The results of finite-element simulation calculation of
a test stand rotor show that the balancing device installed on a coupling can effectively reduce the vibration caused by any
imbalance of a flying wheel. At two rotating speeds, both amplitudes of 2♯ bearing can be reduced by over 25%. How-
ever, during the vibration reduction of 2♯ bearing, the vibration of 1♯ bearing will increase rapidly. An optimized con-
trol over 1♯ and 2♯ bearing vibration was proposed to make the vibration of both bearings at the operating speed not ex-
ceed alarm value 50μm. Based on the idea of an optimized control, hydraulic automatic balancing experiments at several
rotating speeds were performed on the cantilever rotor test stand. The test results verified the conclusion of the simulation
calculation, and the amplitude reduction of 2♯ bearing amounts to over 30%, achieving a relatively good balancing effec-
tiveness. The numerical simulation and experimental study have laid a foundation for the application of active balancing
technology to cantilever rotors in engineering practice. **Key words**: cantilever rotor, active balance, vibration, optimized
control

＝**A Study of the Supporting Modes for a Long Overhanging Ro-
tor During a High-speed Dynamic Balancing Process**[，] / QI Nai-bin, YUAN Qi, RAO Jin-yang (College of En-
ergy Source and Power Engineering, Xi'an Jiaotong University, Xi'an, China, Post Code: 710049)//Journal of Engineer-
ing for Thermal Energy &Power. — 2008, 23(4). —373～377

For large capacity steam turbines, the overhanging section of a LP rotor must be lengthened due to the restrictions of an
exhaust hood structure. This may bring about a great difficulty for the shop high-speed dynamic balance of the rotor owing
to the influence of the rotational inertia of the long overhanging section. To study the supporting modes for the rotor in
question during the high-speed dynamic balance, set up was a test rig for a model rotor with a long overhanging section.
Tests were performed respectively with the bearing points being moved outward and an auxiliary bearing being added. The
test results show that for rotors with a long overhanging section, the method of adding an auxiliary bearing can effectively
reduce the first order resonant amplitude of rotors at the main bearing points. Furthermore, there exists an optimum choice
for the auxiliary bearing position. **Key words**: long overhanging section, rotor, high speed dynamic balance, supporting
mode

＝ **Commercial and Experimental Study of a Rotor-assembly
Type Intensified Heat-transfer Device at the Tube Side**[，] / YANG Wei-min, LI Feng-xiang, YAN Hua (College

of Electro-mechanical Engineering, Beijing University of Chemical Technology, Beijing, China, Post Code: 100029), CHEN Sheng-li (Xi' an Thermodynamics Academy Co. Ltd., Xi' an, China, Post Code: 710032)// Journal of Engineering for Thermal Energy &Power. — 2008, 23(4). —378~381

Current urgent demands for energy-saving and consumption reduction were described along with the technical theory of a rotor-assembly type intensified heat-transfer device—"Clean-energy Core". The above device has been installed on the 2 ♯ turbo-generator unit of Datang Jiamusze No. 2 Power Plant. The condenser at one side of the turbo-generator unit was provided with the device in question while the condenser at another side of the unit was not. The performance test content, testing method and the calculation results of the two condensers of the No. 2 power plant were discussed. The performance of the two condensers and the cost-effectiveness of the No. 2 unit after installation of the device on one condenser were calculated and analyzed. After the installation of the above-cited device, the terminal temperature difference of the relevant condenser drops by 2.79 ℃, its vacuum increases by 2.97 kPa and its cooling water flow rate decreases by 9.8%. Under the condition of the condensers having the same inlet water temperature and vacuum, the load bearing capacity of the turbo-generator unit will increase by 19.7%, and the water resistance of the condensers increase by 19.52 kPa. The results of the industrial experiments and analytic calculation show that the rotor-assembly type intensified heat-transfer device can be used for condensers in power plants to improve their performance. **Key words**: rotor-assembly type, intensified heat transfer, industrial experiment, turbo-generator unit

=A Study on the Improvement of Balancing Target Selection and Holographic Dynamic Balancing Method

[,] / LIAO Yu-he, LANG Gen-feng, QU Liang-sheng (Intelligent Instrument and Monitoring/diagnosis Research Institute, College of Mechanical Engineering, Xi' an Jiaotong University, Xi' an, China, Post Code: 710049)// Journal of Engineering for Thermal Energy &Power. — 2008, 23(4). —382~386

Discussed were the problems existing during the description of the balancing state of a rotor by using a working-frequency trajectory initial-phase vector, which serves as a balancing target in a holographic dynamic-balancing method. Through a precession decomposition of the working-frequency trajectory of the rotor, the different influences of the unbalanced mass of the rotor on its positive and reverse precession component were analyzed in detail. It has been shown that the reverse precession component is not a direct reflection of the balancing state of the rotor. Under the condition of a trajectory with a large eccentricity, the interference on the estimation of the unbalanced mass caused by the reverse precession component should not be neglected. On this basis, presented was an improved holographic dynamic-balancing method with the positive-precession component trajectory initial-phase vector of the rotor serving as a balancing target. Under the condition of not influencing the balancing accuracy of the original holographic dynamic-balancing method, the counterweight version calculation process has been simplified. Compared with the traditional balancing methods, the method under discussion is more accurate and effective. On-site practical applications have verified the reliability and validity of the method. **Key words**: rotor, holographic dynamic-balancing, precession, initial phase vector

=A Study of the Characteristics of Average Heat Transfer Between a Big-particle Dominated Vibrating Fluidized Bed and Horizontal Tubes

[,] / ZHU Xue-jun, YE Shi-chao, SHI Jin-xia, et al (College of Chemical Engineering, Sichuan University, Chengdu, China, Post Code: 610065)// Journal of Engineering for Thermal Energy &Power. — 2008, 23(4). —387~390

In a two-dimensional fluidized bed (240 mm× 80 mm), with glass beads of average diameters d_p of 0.71 mm and 1.83 mm and millet of 1.66 mm diameter serving as materials, studied was the heat transfer law between the vibrating fluidized bed and submerged horizontal tubes. The influence of such factors as gas flow speed, vibration frequency, bed height and diameters of horizontal tubes etc. on the average heat transfer coefficient was investigated. The results of the study show that with an increase of the vibration frequency, the optimum gas flow speed will decrease, and with an increase of the gas flow speed, the optimum vibration frequency will also go down. The average heat transfer coefficient will increase with a decrease of the particle diameter. The particle thermo-physical properties and the tube diameter also have a relatively big influence on the average heat transfer coefficient. From the test data, a correlation formula for the calculation of the average heat transfer coefficient has been obtained. The calculation values have been in relatively good agreement with the test ones, and the calculation error is within a range of ±10%. The above results can serve as reference data for the design and study of vibrating fluidized beds fitted with submerged horizontal tubes. **Key words**: vibrating fluidized bed, average