

低温蒸汽-太阳能双热源 ORC 发电系统热力性能分析

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摘要: 为充分回收矿藏热采过程尾端低温蒸汽余热, 提出一种通过利用太阳能热量补充预热器中热源显热以缩小换热温差的新型低温蒸汽-太阳能双热源 ORC 发电系统。根据热力学第一、第二定律, 建立其热力学模型, 编制计算程序并进行了热力性能分析及比较。计算结果表明: 采用热源补助可有效减小换热温差, 进而显著提升系统热力性能。当采用 R245fa 作为循环工质时, 与基本的 ORC 系统相比, 选择冷端温差较小的预热器可使双热源系统焓效率显著增加; 在预热器冷端温差为 30 K、两系统分别采用 5 种不同循环工质时, 双热源 ORC 系统的热力性能均高于基本 ORC 系统, 且以 R236fa 为工质的双热源 ORC 系统热力性能最佳。

关键词: 有机朗肯循环; 冷端温差; 热力性能; 低温蒸汽-太阳能双热源; 循环工质

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引言

国内外研究学者在 ORC 系统理论和实验方面取得了大量的研究成果^[1-5]。为进一步提升 ORC 系统热力性能以及利用新能源, 近几年新能源(如太阳能)与 ORC 系统综合利用的研究日益受到重视。文献[6]设计了一种以太阳能为热源的 100 kW 的中低温有机朗肯循环系统, 并利用 CFD 软件进行了向心透平的设计与造型; 文献[7]研究了分级抽汽回热式太阳能有机朗肯循环系统的热力性能; 文献[8]利用 TRNSYS 软件对槽式太阳能有机朗肯循环进行了热力性能分析; 文献[9]给出了一种利用太阳能增加蒸发器级数的太阳能 ORC 低温热发电系统, 并指出工质蒸发温度、集热器倾斜角及蒸发器级数是系统优化的主要影响因素。文献[10]利用 PR(对比参数)状态方程对适合太阳能热力发电系统的工质进行了筛选。综上所述, 目前对太阳能结

合 ORC 发电系统的研究多集中在采用单一太阳能作为 ORC 系统热源进行热力发电。

在灌注蒸汽原位对流开采油页岩油气过程中尾端可产生大量的低温余热蒸汽。前期研究成果表明, 基本 ORC 循环能够有效回收低温蒸汽余热, 但理论和实验结果均显示, 采用基本 ORC 循环热力性能较低, 低温蒸汽热能利用仍有提升空间^[11-12]。这是由于蒸汽在释放潜热过程中温度恒定且潜热量相对较多, 换热温差较大导致热源输入的潜热被大量浪费从而降低系统性能。因此, 为充分回收低品位蒸汽热能以及合理利用新能源, 提出一种新型低温蒸汽-太阳能双热源 ORC 系统(以下简称双热源-ORC 系统), 该系统通过采用双热源使工质在预热阶段吸收蒸汽全部显热及太阳能补助热水热量, 在蒸发阶段吸收蒸汽潜热而使蒸汽热量充分回收。双热源系统可避免在换热过程中出现换热温差较大、系统输入热量浪费较多的问题, 同时也弥补了低温余热蒸汽显热供给不足问题, 从而实现热源回收利用最大化, 为其它低品位余热回收利用提供借鉴和参考。

1 双热源-ORC 发电系统原理

1.1 系统基本组成和工作原理

图 1、图 2 分别是双热源-ORC 热力发电系统示意图及热力循环 $T-s$ 图。如图 1 所示, 系统主要由太阳能集热器热水回路、蓄热回路以及有机工质回路组成。热井中的排水经处理后由泵 2 加压, 进入太阳能集热器中将太阳辐射能转化为自身热量后, 根据需要, 调节阀门 2 控制进入预热器内与有机工质换热的水流量, 调节阀门 3 控制进入蓄热罐

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中的热水流量。蓄热回路可在环境条件达不到换热要求时保证整个发电系统的稳定运行,如图1中虚线所示。当日照条件较好时,打开阀门2、阀门3,关闭阀门1,运行泵2,调节阀2控制进入预热器中换热的水量,多余的水则通过阀门3进入蓄热罐内与蓄热材料换热。当日照条件较差时,集热器内的热水热量不能满足系统换热要求,此时打开阀门1,关闭阀门2、阀门3,停止运行泵2,调节阀1控制进入预热器内与有机工质换热的水流量。有机工质通过泵1加压后首先在预热器内吸收热量达到液相饱和点,之后继续在蒸发器内吸收低温蒸汽潜热达到气相饱和态后进入汽轮机做功,乏气进入冷凝器冷凝成饱和液体,并通过泵1送回预热器完成循环。

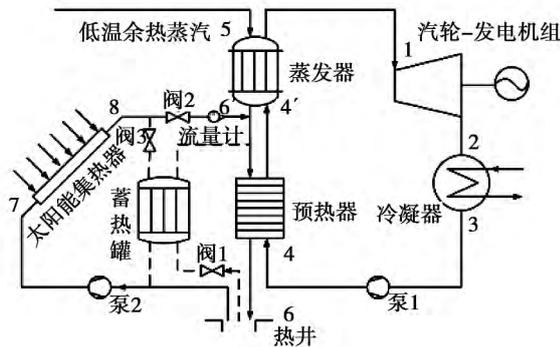


图1 双热源 - ORC 发电系统原理图
Fig. 1 The schematic diagram of double heat source - ORC system

循环系统包括5个热力过程:1-2为理想状态下工质在汽轮机中绝热膨胀做功过程;2-3为工质在冷凝器中定压放热过程;3-4为工质在工质泵中绝热压缩过程;4-4'-1为工质分别在预热器、蒸发器中定压吸热过程。

1.2 预热器内换热过程

图3为有机工质在预热器内温差变化示意图。其中,虚线及实线分别代表系统采用单一热源、双热源时,工质在预热器中的换热过程。如图3所示,若单独采用余热蒸汽流做为系统热源,尽管余热流自身存在大量潜热,但由于换热过程中存在较大换热温差 ΔT_{max} ,使余热流输入系统热量不能有效被工质吸收利用,进而影响系统热力性能。使用双热源后,太阳能集热器补充的热量弥补了余热流显热不足,使工质在蒸发器中吸收余热流潜热,而在预热器内

则依靠余热流显热以及集热器辅助热水热量加热有机工质,且使系统换热温差减小到 ΔT_{pin} 并保持稳定。这样在缩小换热温差的同时也大大提高了余热流热量利用率,使系统热力性能大幅提升。

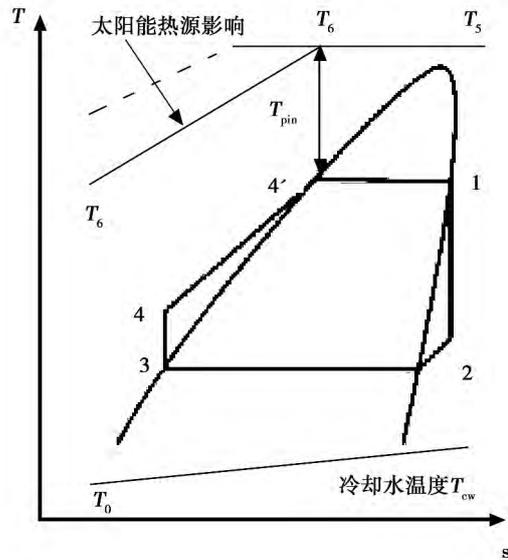


图2 双热源 - ORC 发电系统热力循环温熵图
Fig. 2 T - s diagram of double heat source - ORC system

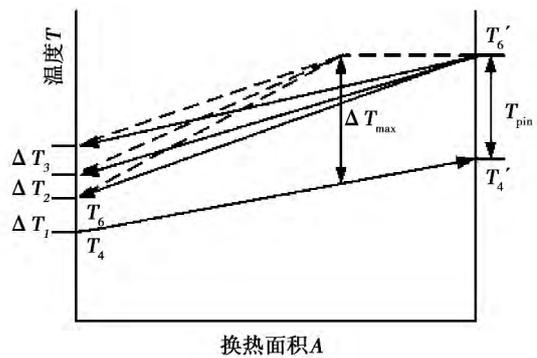


图3 预热器换热温差变化示意图

Fig. 3 The schematic diagram of variation of temperature difference in preheater

2 系统热力分析理论模型

蒸发器:

$$Q_{4'-1} = m_{wf}(h_1 - h_{4'}) = m_g(h_5 - h_6) \quad (1)$$

预热器:

$$Q_{4-4'} = m_{wf}(h_{4'} - h_4) = cm_g(t_{6'} - t_6) + Q_s \quad (2)$$

集热器补助的换热量:

$$Q_s = m_{wf}(h_1 - h_4) - m_g(h_5 - h_6) - cm_g(t_{6'} - t_6) \quad (3)$$

集热器吸收的热量:

$$Q_{7-8} = \frac{Q_s}{\eta_v} \quad (4)$$

集热器补助换热的热水质量流量:

$$m_s = \frac{Q_s}{c \cdot (t_{6'} - t_6)} \quad (5)$$

集热器内热水质量流量:

$$m_{7-8} = \frac{Q_{7-8}}{c \cdot (t_{6'} - t_6)} \quad (6)$$

系统总功输出:

$$W_T = m_{wf}(h_1 - h_2) \eta_T \eta_m \eta_G \quad (7)$$

泵耗功:

$$W_P = \frac{m_{wf}(h_4 - h_3)}{\eta_p} \quad (8)$$

系统净输出功:

$$W_{net} = W_T - W_P \quad (9)$$

集热器补助热水的热量焓:

$$E_s = m_s((h_{6'} - h_6) - (t_0 + 273)(s_{6'} - s_6)) \quad (10)$$

输入系统总热源焓:

$$E = m_g((h_5 - h_6) - (t_0 + 273)(s_5 - s_6)) + E_s \quad (11)$$

系统热效率:

$$\eta = \frac{W_{net}}{Q_{吸}} = \frac{W_T - W_P}{Q_{4-4'} + Q_{7-8} + cm_g(t_{6'} - t_6)} \quad (12)$$

系统焓效率:

$$\eta_E = \frac{W_{net}}{E} \quad (13)$$

式中: m_{wf} 、 m_g —有机工质和余热源质量流量, kg/s; h —比焓, kJ/kg; W —功, kW; Q —热流量, kJ/s; η_v —太阳能集热器效率; s —比熵, kJ/(kg · K); E —热量焓, kJ/s; t —温度, K; c —水比热容, kJ/(kg · K); 角标 1-6、4'和 6'代表工质和热源各状态点; t_0 —环境温度, K; 角标 net—净值; 角标 T、m、G、P—汽轮机、机械传动、发电机和工质泵。

3 双热源 - ORC 与 ORC 系统热力性能比较

针对原位对流开采油页岩油气技术, 其尾端低温余热蒸汽温度为 383.15 K, 压力为 0.14 MPa, 单井筒余热流量为 27.8 kg/s。计算过程中设定汽轮机以背压 0.2 MPa 运行, 环境温度设定 298 K, 窄点温差 $T_{pin} = 5$ K, 各设备效率分别为 $\eta_T = 0.65$, $\eta_p = 0.8$, $\eta_m = 0.85$, $\eta_G = 0.85$, $\eta_v = 0.45$ 。为使两系统在相同条件下进行比较, 将集热器补助热量加入基本 ORC 系统理论模型中, 同时假定系统中各换热器热力过程均为等压过程。工质物性参数来源于 REFPROP 8.0 程序。

3.1 蒸发压力、预热器冷端温差对系统性能影响

对于双热源系统, 工质在蒸发器中吸收热源全部潜热, 换热量固定, 当蒸发温度确定时, 由式(1)可知, 工质质量流量为定值, 从而其在预热器中换热量亦被确定。预热器冷端温差 ΔT 的变化对太阳能需要补助水量以及系统的热力性能具有重要影响。

图 4 所示为双热源 - ORC 系统采用 R245fa 为循环工质时, 集热器补助热水质量流量随预热器冷端温差以及蒸发压力的变化规律。从图中可以看出, 随着蒸发压力逐渐增大, 预热器内工质需要的吸热量逐渐增加, 且预热器冷端温差升高, 使集热器补助热水质量流量不断提高。

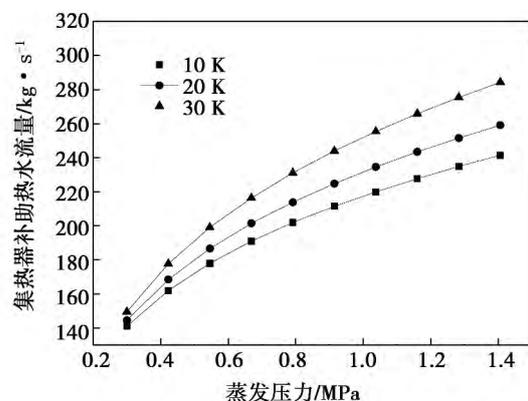


图 4 补助热水流量与蒸发压力、预热器冷端温差关系

Fig. 4 The relationship between added hot water flow and evaporation pressure and cold-side temperature difference in preheater

图5为采用R245fa工质时,双热源-ORC和ORC系统在不同蒸发压力下,系统焓效率随预热器冷端温差的变化规律。从图中可以看出,蒸发压力增大,两系统焓效率均随预热器冷端温差的减小而增大,且增幅逐渐增加。相比于基本ORC系统,减小预热器冷端温差对双热源-ORC系统焓效率的提升更为明显。当预热器冷端温差一定时,随蒸发压力不断上升,双热源-ORC系统焓效率明显高于ORC系统且增幅越来越大。例如在冷端温差20 K、蒸发压力为1.16 MPa时,双热源-ORC、ORC系统焓效率分别为60.72%和37.27%,增加了23.45%,蒸发压力为1.28 MPa时,双热源-ORC、ORC系统焓效率分别为63.86%和38.39%,增加了25.47%。由于ORC系统在预热过程中存在较大换热温差,导致系统焓效率较小,系统采用双热源后,相对于采用单一热源的基本ORC系统,可更充分利用余热流潜、显热能量,使热源间热量得到更好分配。因此在实际工程中,选取合理的双热源-ORC系统运行压力以及预热器冷端温差可明显提高系统热力性能。

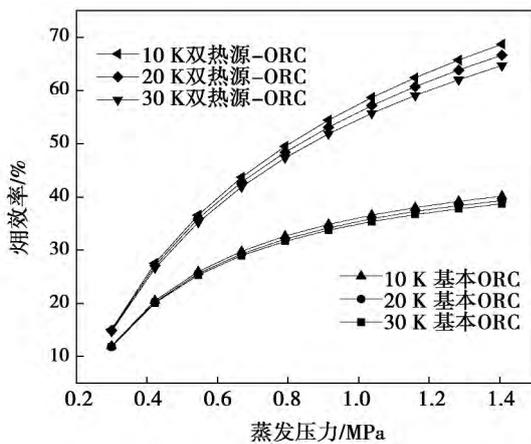


图5 系统焓效率与蒸发压力、预热器冷端温差关系

Fig. 5 The relationship between system efficiency and evaporation pressure and cold-side temperature difference in preheater

3.2 工质对不同系统性能影响

工质筛选主要从工质热物性、安全性、环保性和稳定性4方面综合考虑,根据文献[10, 13]推荐,选取R245fa、R114、R123、R236fa和R600 5种有机工质,并设定预热器冷端温差为30 K进行系统热力性

能比较。图6为双热源-ORC系统采用不同工质时,蒸发压力对太阳能集热器辅助热水质量流量的影响。由图可知,对于不同工质,随蒸发压力不断提高,辅助热水质量流量均增大。这是由于蒸发压力增大,单位质量工质吸热量减少,由式(1)可知,工质质量流量增大,此时则需要采用双热源满足工质在预热器段对换热量的需求,因此集热器内热水质量流量逐渐增大。其中R236fa系统增幅最明显,工质临界温度、临界压力等物性参数的不同是导致这一现象的原因。

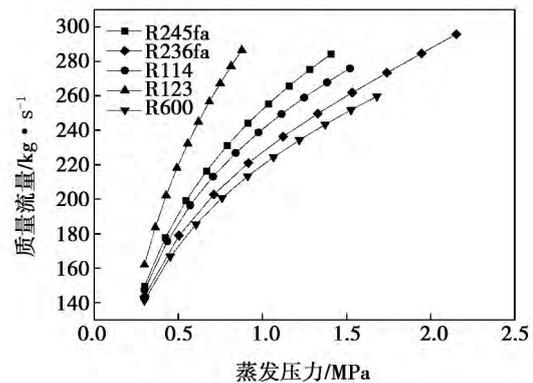


图6 不同工质、辅助热水流量与蒸发压力关系
Fig. 6 The relationship between added hot water flow and evaporation pressure in different working fluids

图7为工质流量随蒸发压力的变化关系。由图可知,随蒸发压力不断增加,双热源-ORC系统工质流量与ORC系统工质流量变化趋势截然相反。这是因为系统采用双热源后,工质在蒸发器阶段质量流量由热源潜热及单位工质潜热决定。随着蒸发压力逐渐提高,工质由饱和液态到饱和气态吸热量减小,其质量流量呈不断增加趋势。对于基本ORC循环,窄点温差位置具有不确定性,其质量流量则需由换热器整体换热量及单位工质由未饱和液态吸热至饱和气态数值决定,随蒸发压力提高,单位工质吸热量不断增加,质量流量呈下降趋势。

根据式(1)、式(9)、式(12)可知,工质质量流量的变化不仅对换热过程中工质吸热量和汽轮机净输出功率产生直接影响,而且还会间接影响系统热效率以及焓效率。图8分别为双热源-ORC、ORC系统净输出功同循环工质、蒸发压力的变化规律。从

图 8 可以明显看出, 随蒸发压力的提高, 双热源 - ORC 系统工质质量流量随蒸发压力逐渐增大, 系统净输出功增加显著且增幅逐渐增大。例如: 蒸发压力为 0.49 MPa, 以 R123 为循环工质的双热源 - ORC、ORC 系统净输出功分别为 2 938 kW、2 651 kW, 增加了 10.83%; 当蒸发压力增大到 0.81 MPa, 双热源 - ORC、ORC 系统的净输出功分别为 5 024 kW 和 3 887 kW, 增加了 29.04%。

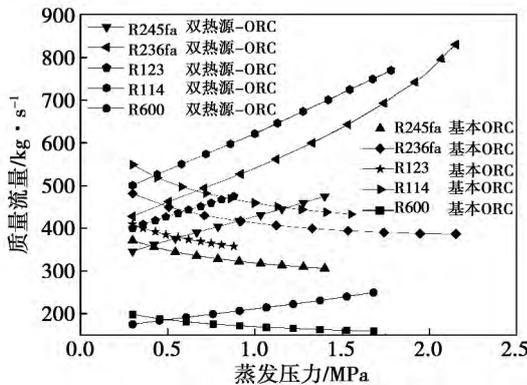


图 7 工质流量随蒸发压力变化关系

Fig. 7 The relationship between working fluid flow with evaporation pressure in different working fluids

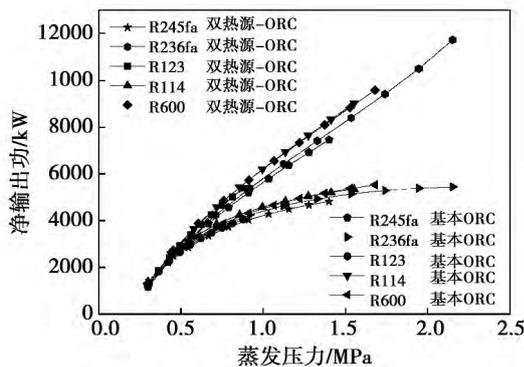


图 8 不同工质、系统净输出功随蒸发压力变化关系

Fig. 8 The relationship between net work and evaporation pressure in different working fluids

图 9 - 图 10 分别为两系统在使用不同循环工质时, 系统热效率、焓效率随蒸发压力的变化规律。如图 9 所示, 随蒸发压力的不断提高, 采用不同工质的双热源 - ORC 系统热效率均比 ORC 系统有所提升, 增幅则呈现先增大后减小的趋势。如图 10 所

示, 双热源 - ORC 系统焓效率在使用 5 种不同工质时均比 ORC 系统焓效率有所增加, 且增幅越来越明显, 这也是由于工质质量流量变化趋势不同所导致。相比于采用单一热源的 ORC 系统来说, 采用双热源 - ORC 系统在减小换热温差的同时保证了对余热流的充分利用。因此不仅焓效率增加明显, 也大大提高了热源回收使用率。其中以 R236fa 为工质的双热源 - ORC 系统在其达到允许最大蒸发压力时, 系统焓效率增幅最大, 增加 26.33%。

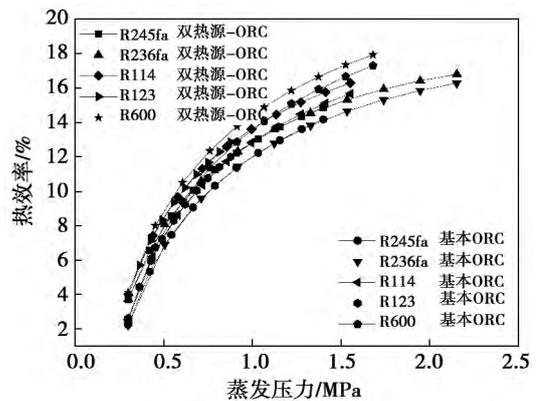


图 9 不同工质 系统热效率随蒸发压力变化关系

Fig. 9 The relationship between thermal efficiency and evaporation pressure in different working fluids

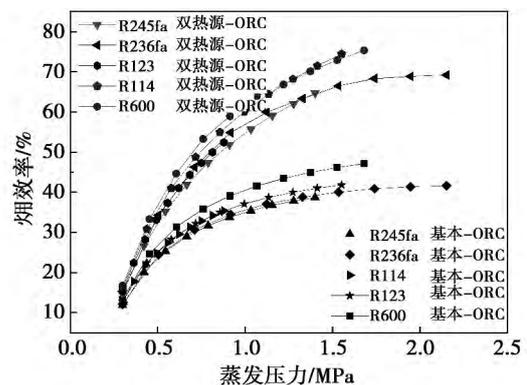


图 10 不同工质 系统焓效率随蒸发压力变化关系

Fig. 10 The relationship between exergy efficiency and evaporation pressure in different working fluids

4 结 论

(1) 采用双热源可有效减小工质在换热过程中的换热温差, 提高热源使用率。采用 R245fa 为循环

工质 根据系统蒸发压力的不同 选择适合系统的预热器冷端温差可大幅提升系统热力性能。

(2) 随系统蒸发压力的不断增加,在分别采用 5 种工质时,双热源 - ORC 系统的热力性能均大于基本 ORC 系统。在工质达到允许最大蒸发压力时,选择 R236fa 作为双热源 - ORC 系统的循环工质最具优势,最大可使系统焓效率增加 26.33%。

参考文献:

[1] Hung T. C. Waste heat recovery of organic rankine cycle using dry fluids[J]. Energy Conversion and Management 2011 42(5): 539 - 553.

[2] Galanis N ,Cayer E ,Roy P. Electricity generation from low temperature source[J]. Journal of Applied Fluid Mechanics 2009 2(2): 55 - 67.

[3] Philippe Roy ,MartinDesilets ,Nicolas Galanis. Thermodynamic analysis of a power cycle using a low-temperature source and a binary NH₃ - H₂O mixture as working fluid[J]. International Journal of Thermal Science 2009 49: 48 - 58.

[4] WANG Zhi-qi ,ZHOU Nai-jun ,GUO Jing ,et al. Fluid selection and parametric optimization of organic Rankine cycle using low temperature waste heat[J]. Energy 2012 37(40): 107 - 115.

[5] Wei Dong-hong ,Lu Xue-sheng ,Lu Zhen ,et al. Performance analysis and optimization of organic Rankine cycle(ORC) for waste heat recovery[J]. Energy Conversion and Management 2007 48(4): 1113 - 1119.

[6] 宋 健 徐俊杰 李 艳 等. 太阳能中低温有机朗肯循环系统的设计与分析 [J]. 工程热物理学报, 2014, 35(7): 1309 - 1312.

SONG Jian ,XU Jun-jie ,LI Yan ,et al. Design and analysis of an organic rankine cycle system using medium and low grade solar energy[J]. Journal of Engineering Thermophysics 2014 35(7): 1309 - 1312.

[7] 韩中合 叶依林 王 璟. 分级抽汽回热式太阳能低温有机朗肯循环系统的热力性能分析[J]. 汽轮机技术 2012 54(2): 81 - 84.

HAN Zhong-he ,YE Yi-lin ,WANG Jing. Thermodynamic analysis of graded regenerative extraction low-temperature power generation

based on solar-heated organic rankine cycle[J]. Turbine Technology 2012 54(2): 81 - 84.

[8] 刘怀亮 何雅玲 程泽东 等. 槽式太阳能有机朗肯循环发电系统模拟[J]. 工程热物理学报 2010 31(10): 1631 - 1634.

LIU Huai-liang ,HE Ya-ling ,CHENG Ze-dong ,et al. Simulation of parabolic trough solar thermal generation with organic rankine cycle[J]. Journal of Engineering Thermophysics 2010 31(10): 1631 - 1634.

[9] 李 晶 裴 刚 季 杰. 太阳能有机朗肯循环低温发电关键因素分析[J]. 化工学报 2009 60(4): 826 - 832.

LI Jing ,PEI Gang ,JI Jie. Analysis of key factors in low temperature solar-thermal electric power generation with organic rankine cycle [J]. CIESC Journal 2009 60(4): 826 - 832.

[10] 王辉涛 王 华. 低温太阳能热力发电有机朗肯循环工质的选择 [J]. 动力工程 2009 29(3): 287 - 291.

WANG Hui-tao ,WANG Hua. Selection of working fluids for low-temperature solar thermal power generation organic rankine cycle [J]. Journal of Power Engineering 2009 29(3): 287 - 291.

[11] 杨新乐 赵阳升 冯增朝 等. 对流热采油页岩过程低温余热 ORC 系统热力分析 [J]. 热能动力工程 2012 27(6): 664 - 668.

YANG Xin-le ,ZHAO Yang-sheng ,FENG Zeng-chao ,et al. Thermal analysis of low temperature waste heat ORC system in the process of mineral mining with steam injection [J]. Journal of Engineering for Thermal Energy and Power 2012 27(6): 664 - 668.

[12] 杨新乐 黄菲菲 赵阳升 等. 低温余热蒸汽 ORC 发电系统热力性能实验研究 [J]. 热能动力工程 2013 28(3): 465 - 470.

YANG Xin-le ,HUANG Fei-fei ,ZHAO Yang-sheng , et al. Experimental research on the thermal performance of low temperature steams ORC power generation system [J]. Journal of Engineering for Thermal Energy and Power 2013 28(3): 465 - 470

[13] Saleh B ,Koglbauer G ,Wendland M ,Fischer J. Working fluids for low temperature organic rankine cycles [J]. Energy 2007 32(7): 1210 - 1221.

(丛 敏 编辑)

the design parameters of the axial flow fan, a fan model with a rotating speed of 9 000 r/min was designed based on the original blade profile and a fan model with a rotating speed of 8 000 r/min was also designed based on a bowed blade profile, both blade profile lifting force coefficients being 0.72 and 1.02 respectively. A study of the flow field and performance of both fans was performed through numerical simulation and tests. It has been found that under the operating condition nearing the design load, the performance of both prototype and bowed blade profile model is basically identical and the error of the total pressure is about 6%. Under the condition of the fan models being operating at the half rotating speeds, the aerodynamic noise of the bowed blade profile fan model will decrease by 1.4 dB as compared with that of the original blade profile fan model while the aerodynamic performance of both fan models is basically kept unchanged. **Key words:** axial flow fan, bowed blade profile, fan performance, noise

低温蒸汽-太阳能双热源 ORC 发电系统热力性能分析 = **Thermal Performance Analysis of a Low Temperature Steam-Solar Energy Dual Heat Source ORC (Organic Rankine Cycle) Power Generation System** [刊], 汉 J YANG Xin-le, DONG Si-han, HUANG Fei-fei (College of Mechanical Engineering, Liaoning Engineering Technology University, Fuxin, China, Post Code: 123000), ZHANG Bo (Huayou Industry Company, Liaohe Petroleum Prospecting Bureau, Panjin, China, Post Code: 124010) // Journal of Engineering for Thermal Energy & Power. - 2016, 31(1). - 105 - 110

To fully recover the low temperature waste heat steam in the tail end of the mineral reserve hot mining process, proposed was a novel type low temperature steam-solar energy dual heat source ORC system utilizing the heat quantity from solar energy to replenish the apparent heat of the heat source in the preheater to reduce the heat exchange temperature difference. According to the first and second law of thermodynamics, a thermodynamic model for the system in question was established, a program was prepared and an analysis and comparison of the thermal performance of the system were conducted. The calculation results show that to use a supplementary heat source can effectively reduce the heat exchange temperature difference, thus remarkably enhancing the thermal performance of the system. When R245fa is used as the working medium in the dual heat source cycle system, compared with the basic ORC system, to choose a preheater with its temperature difference at the cold end being relatively small in the system can make its exergy efficiency remarkably increase. When the temperature difference at the cold end of the preheater is 30 K and both systems adopt respectively five kinds of cyclic working medium, the thermal performance of the dual heat source systems are all higher than that of the basic ORC system and the thermal performance of the dual heat

source ORC system using R236fa as its working medium is optimum. **Key words:** dual heat source ,organic Rankine cycle ,cold-end temperature difference ,working medium ,thermal performance

百万千瓦燃煤机组烟气超低排放设计及应用 = **Design and Applications of the Ultra Low Emissions of Flue Gases From a 1 000 MW Coal-fired Boiler Unit** [刊 汉] MENG Wei ,LI Qing-yi ,HU Da-qing (Zhejiang Heaven-and-earth Environmental Protection Engineering Co. Ltd. ,Hangzhou ,China ,Post Code: 310003) ,GAO Xiang (National Key Laboratory on Clean Utilization of Energy Sources ,Thermal Energy Engineering Research Institute , Zhejiang University ,Hangzhou ,China ,Post Code: 310027) //Journal of Engineering for Thermal Energy & Power. -2016 ,31(1) . -111 -116

To realize the ultra low emissions of flue gases from a 1 000 MW coal-fired boiler unit in a power plant ,the following flue gas ultra low emission technical version was applied: low nitrogen burners were used in the denitration process , catalysts were added into the SCR (selective catalytic reduction) reactors ,low-low temperature ESPs (electronic static precipitator) and wet type electronic precipitators were employed to remove the dust and the interactive sprinkling technology was used to remove the sulfur. After this technical version has been put in practice ,the flow path for discharging the flue gases from the boiler unit can be given as follows: the flue gases outgoing from the outlet of the boiler adopting low nitrogen burners will in turn flow through the economizers ,SCR reactors ,air preheaters ,the desuperheating section of tube type heat exchangers and the low-low temperature ESPs and then enter into the absorption towers ,afterwards ,pass through the wet type ESPs and the temperature rise sections of the tube type heat exchangers and go into the smoke stacks. After the technical version had been executed ,the emissions concentrations of the main pollutants tested at the inlets of the smoke stacks ,including NO_x ,soot and SO_2 ,arrived at 25.83 mg/Nm^3 ,1.61 mg/Nm^3 and 22.08 mg/Nm^3 at an oxygen concentration of 6% respectively ,thus making the pollutant emissions concentrations of the coal-fired boiler unit meet the requirements stipulated in the emission standard for natural gas turbines. **Key words:** ultra low emissions ,pollutant ,natural gas turbine unit ,standard

1 045 MW 超超临界贫煤锅炉燃用高挥发分烟煤的燃烧调整研究及工程实践 = **Study of the Adjustment to the Combustion in a 1 045 MW Ultra-supercritical Boiler Burning Bituminous Coal With a High Volatile Content Instead of Lean Coal and Its Engineering Practice** [刊 汉] LI De-bo (Electric Power Science Research Institute ,Guangdong Power Grid Company ,Guangzhou ,China ,Post Code: 510060) ,DI Wan-feng ,LI Xin ,GUO