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不可逆闭式布雷顿热电联产装置㶲经济性能优化

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摘 要: 应用有限时间热力学方法, 研究了恒温热源条件下不可逆闭式布雷顿联产装置的㶲经济性能, 导出了利润率及㶲效率解析式。利用数值计算方法, 以利润率为目标, 对热导率分配和压比的选取进行了优化。研究了最优利润率及相应㶲效率特性, 并分析了各种联产设计参数对联产优化性能的影响。结果表明, 对于给定的总热导率, 在高温、低温和用户侧换热器之间, 存在唯一的最佳热导率分配比和唯一的最佳压比, 使得装置的无因次利润率取得最大值; 同时存在最佳用户温度。

关 键 词: 有限时间热力学; 闭式布雷顿热电联产装置; 㶲; 经济性能; 利润率

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引 言

自有限时间热力学理论产生以来, 它在物理和工程领域的应用已取得了很大的进展[1~4]。热电联产装置由于具有提高能源利用效率, 减少污染物排放等优点, 越来越受到更多的关注, 一些学者将有限时间热力学方法应用于分析联产循环, 以确定联产装置的优化设计参数。Bojic 建立了利用余热供热的内可逆 Carnot 热电联产装置模型[5], 进行了热经济优化, Sahin 等人建立了抽汽供热的内可逆 Carnot 热电联产装置模型[6], 并对其进行了㶲优化, Erdil 等人对不可逆 Carnot 联合循环热电联产装置进行了㶲优化[7], Yilmaz 与 Hao 等人对内可逆 Brayton 热电联产装置进行了㶲优化[8~9]。Ust 等人提出了㶲性能系数 EPC (exergetic performance coefficient), 并以 EPC 为目标对不可逆回热式 Brayton 热电联产装置、不可逆 Dual 热电联产装置进行了优化[10~11]。20 世纪 90 年代陈林根等人提出了将有限时间热力学与热经济学相结合, 建立了有限时间㶲经济分析法, 定义利润率为热力循环的输出㶲的收益率与热力循环的输入㶲的成本率之差, 导出了内可逆 Carnot 热机、制冷机、热泵的有限时间㶲经济性能界限、优化关系和参数优化准则[12~15]。此外, 一些学者还将㶲经济分析法推广到量子热机、广义不可逆热机、普适热机、三热源制冷机、热泵的有限时间热力学性能的研究中。但以利润率为目标, 对热电联产装置进行有限时间㶲经济分析与优化, 目前仍是空白, 为此本文将把有限时间㶲经济性能分析法引入到热电联产装置的研究中去。

1 循环模型

图 1 为由不可逆闭式布雷顿循环构成的热电联产装置模型。其中, T_H 和 T_L 分别为高温热源和低温热源的温度, T_K 为用户侧的用热温度, 3 个换热器的热导率(传热系数与传热面积之积)分别为 U_H、U_L 和 U_K, 工质的热容率(质量流率与定压比热之积)为 C_{wf}。过程 1～2 和过程 3～4 分别为工质在压气机和透平内不可逆绝热压缩和不可逆膨胀过程, 过程 2～3 和 5～1 分别为工质从高温热源等压吸热和向低温热源等压排热过程, 过程 4～5 为工质在热回收装置中等压供热过程。1～2s 与 3～4s 为 1～2 与 3～4 相应的可逆绝热压缩和膨胀过程。

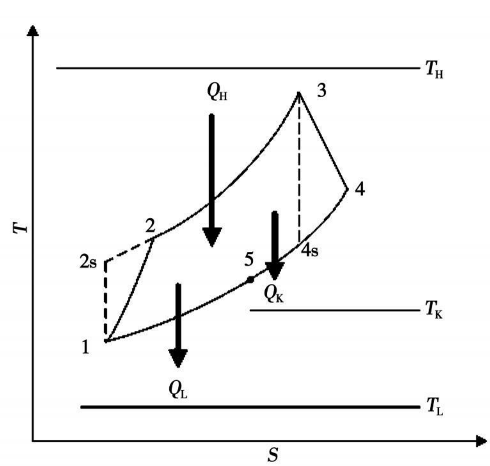

图 1 循环模型

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设工质为定比热的理想气体, 各状态点的温度为 T_i ($i=1, 2s, 2, 3, 4, 4s, 5$), 由换热器理论和工质性质可知, 工质从高温热源的吸热率 Q_H 和向低温热源的放热率 Q_L, 以及向热用户的供热率 Q_K 分别为:

$$Q_H=C_{wf}E_H(T_H-T_2)=C_{wf}(T_3-T_2) \tag{1}$$

$$Q_L=C_{wf}E_L(T_5-T_L)=C_{wf}(T_5-T_1) \tag{2}$$

$$Q_K=C_{wf}E_K(T_4-T_K)=C_{wf}(T_4-T_5) \tag{3}$$

式中: $E_H=1-e^{-N_H}$, $E_L=1-e^{-N_L}$, $E_K=1-e^{-N_K}$—高温、低温和用户侧换热器的有效度; $N_H=U_H/C_{wf}$, $N_L=U_L/C_{wf}$, $N_K=U_K/C_{wf}$—换热器的传热单元数。

2　有限时间㶲经济性能

压气机和透平的内损失用内效率 η_c 和 η_t 来表示, 即有:

$$\eta_c=(T_{2s}-T_1)/(T_2-T_1),\ \eta_t=(T_3-T_4)/(T_3-T_{4s}) \tag{4}$$

对内可逆循环有 $T_1T_3=T_{2s}T_{4s}$, 定义循环等熵温比 x 为:

$$x=T_{2s}/T_1=T_3/T_{4s}=\pi^m \tag{5}$$

式中: π—循环的压比, $m=(k-1)/k$; k—工质的绝热指数。

装置产生的功率(㶲输出率)为:

$$P=Q_H-Q_L-Q_K \tag{6}$$

而对整个装置应用㶲平衡方程有:

$$e_H=P+e_K+T_0\sigma \tag{7}$$

式中: $e_H=Q_H(1-T_0/T_H)-Q_L(1-T_0/T_L)$—装置的㶲输入率; T_0—环境温度; $\sigma=Q_L/T_L+Q_K/T_K-Q_H/T_H$—循环熵产率; $T_0\sigma$—循环㶲损失率。

由上式可得提供给用户热量㶲的输出率为:

$$e_K=Q_K(1-T_0T_K) \tag{8}$$

而计算提供给用户热量㶲的输出率为文献[8~10]:

$$e_K=Q_K(1-T_0/T_a) \tag{9}$$

式中: $T_a=(T_4-T_5)/(\ln T_4/T_5)$—过程 4~5 的平均温度。

比较式(8)与式(9), 可以发现式(9)计算热量㶲的输出率比式(8)多计入了传热过程的㶲损失, 是不完备的, 用式(8)来表征热量㶲的输出率更能体现有限时间热力学的本义, 所以提供给用户热量㶲的输出率应为 式(8)。

装置的㶲输入率为:

$$e_H=Q_H(1-T_0/T_H)-Q_L(1-T_0/T_L) \tag{10}$$

设联产装置输出功率价格为 φ_P, 提供给用户热量㶲的输出率价格为 φ_K, 装置㶲输入率价格为 φ_H, 则装置的利润率为:

$$\Pi=\varphi_PP+\varphi_Ke_K-\varphi_He_H \tag{11}$$

用 $\varphi_HC_{wf}T_0$ 对利润率 Π 进行无因次化($\bar{\Pi}=\Pi/(\varphi_HC_{wf}T_0)$), 并联立式(1)~式(6)、式(8)、式(10)与式(11), 可得装置的无因次利润率为:

$$\bar{\Pi}=\frac{\begin{aligned}&a(x-1)\{\eta_c\eta_tx^{-1}E_H\tau_1+\eta_tx^{-1}(x-1+\eta_c)(1-E_H)[E_L\tau_3+(1-E_L)E_K\tau_2]-E_L\tau_3-(1-E_L)[E_K\tau_2+\\&(1-E_K)(1-\eta_t+\eta_tx^{-1})E_H\tau_1]\}+bE_K(1-\tau_2^{-1})\{(1-\eta_t+\eta_tx^{-1})\}\times\{\eta_cE_H\tau_1+(x-1+\eta_c)(1-E_H)\times\\&[E_L\tau_3+(1-E_L)\tau_2]\}-\eta_c\tau_2-E_H(1-\tau_1^{-1})\{\eta_c\tau_1-(x-1+\eta_c)\{E_L\tau_3+(1-E_L)[E_K\tau_2+(1-E_K)(1-\\&\eta_t+\eta_tx^{-1})\tau_1]\}+E_L(1-\tau_3^{-1})\times\\&\{\eta_cE_K\tau_2-\eta_c\tau_3+(1-\eta_t+\eta_tx^{-1})(1-E_K)[\eta_cE_H\tau_1+(x-1+\eta_c)(1-E_H)\tau_3]\}\end{aligned}}{\eta_c-(x-1+\eta_c)(1-E_L)(1-E_K)(1-E_H)(1-\eta_t+\eta_tx^{-1})} \tag{12}$$

式中: $a=\varphi_P/\varphi_H$、$b=\varphi_K/\varphi_H$—价格比, $\tau_1=T_H/T_0$、$\tau_2=T_K/T_0$ 和 $\tau_3=T_L/T_0$—高温、用户和低温侧热源温度与环境温度之比。

装置的㶲效率为:

$$\eta=(P+e_K)/e_H \tag{13}$$

联立式(1)~式(6)、式(8)、式(10)与式(13)得:

$$\eta=\frac{\begin{aligned}&(x-1)\{\eta_c\eta_tx^{-1}E_H\tau_1+\eta_tx^{-1}(x-1+\eta_c)(1-E_H)[E_L\tau_3+(1-E_L)E_K\tau_2]-E_L\tau_3-(1-E_L)[E_K\tau_2+\\&(1-E_K)(1-\eta_t+\eta_tx^{-1})E_H\tau_1]\}+E_K(1-\tau_2^{-1})\{(1-\eta_t+\eta_tx^{-1})\times\\&\{\eta_cE_H\tau_1+(x-1+\eta_c)(1-E_H)[E_L\tau_3+(1-E_L)\tau_2]\}-\eta_c\tau_2\}\end{aligned}}{\begin{aligned}&E_H(1-\tau_1^{-1})\{\eta_c\tau_1-(x-1+\eta_c)\{E_L\tau_3+(1-E_L)[E_K\tau_2+(1-E_K)(1-\eta_t+\eta_tx^{-1})\tau_1]\}\}-E_L(1-\tau_3^{-1})\times\\&\{\eta_cE_K\tau_2-\eta_c\tau_3+(1-\eta_t+\eta_tx^{-1})(1-E_K)[\eta_cE_H\tau_1+(x-1+\eta_c)(1-E_H)\tau_3]\}\end{aligned}} \tag{14}$$

当时 $\eta_c=\eta_t=1$, 式(12)与式(14)转化成内可逆布雷顿热电联产装置的利润率与㶲效率目标, 由于热量㶲计算方法不同, 又不同于以前的㶲效率目标[8~9]。

一单位价格的㶲输入至少有一单位价格的㶲输出，为保证联产装置盈利，须有 $\varphi_P \geqslant \varphi_H$ 和 $\varphi_K \geqslant \varphi_H$。

当 $\varphi_P = \varphi_K = \varphi_H$ 时，式(11)变成：

$$\Pi = \varphi_P(P + e_K - e_H) = -\varphi_P T_0 \sigma \tag{15}$$

式(12)由不可逆布雷顿热电联产装置的最大利润率目标转化成最小熵产率目标。

当 $\varphi_P = \varphi_K$ 且 $\varphi_P/\varphi_H \to \infty$ 时，式(11)变成：

$$\Pi = \varphi_P(P + e_K) \tag{16}$$

式(12)由不可逆布雷顿热电联产装置的最大利润率目标转化成最大㶲输出率目标。

3 数值算例

由式(12)知，对于一定的价格比和热源温比，无因次利润率与 3 个换热器的热导率和等熵温比(压比)有关，假设总热导率 U_T($U_T = U_H + U_L + U_K$)保持恒定，并定义高温、低温和用户侧热导率分配比为：

$$u_h = U_H/U_T,\ u_l = U_L/U_T,\ u_k = U_K/U_T \tag{17}$$

显然，u_h、u_l、u_k 必须满足：

$$0 < u_h < 1,\ 0 < u_l < 1,\ 0 < u_k < 1,\ u_h + u_l + u_k = 1 \tag{18}$$

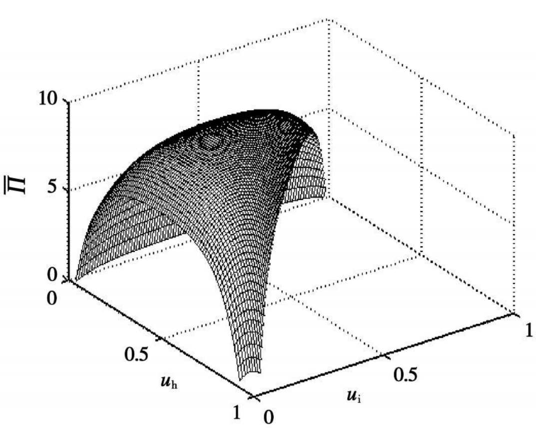

图 2　$\overline{\Pi}$ 与(u_h, u_l)的三维关系

利用式(12)很难通过解析的方法直接分析各种参数对无因次利润率的影响，对热导率分配和压比的优化均采用数值计算的方法进行分析。计算中取 $C_{wf} = 1.0$ kW/K, $k = 1.4$, $\tau_1 = 5$, $\tau_2 = 1.4$, $\tau_3 = 1$, $U_T = 10$ kW/K, $\eta_c = \eta_t = 0.85$，根据文献[16]的分析，取 $a = 10$, $b = 6$。图 2 给出了当压比 $\pi = 8$ 时，无因次利润率($\overline{\Pi}$)与高低温侧热导率分配比(u_h, u_l)之间的三维关系。从图 2 可以看出，当压比给定时，在高温、低温和用户侧换热器之间存在唯一的热导率分配，使得 $\overline{\Pi}$ 达到最优值($\overline{\Pi}_{opt}$)。图 3 给出了最优无因次利润率 $\overline{\Pi}_{opt}$ 与循环压比 π 的关系，从图 3 可以看出，$\overline{\Pi}_{opt}$ 与压比 π 呈类抛物线关系，存在一个最佳的压比($\pi_{\overline{\Pi}}$)使最优利润率 $\overline{\Pi}_{opt}$ 达到最大值($\overline{\Pi}_{max}$)。这即是说，对应一个最大利润率，高温、低温和用户侧换热器之间存在唯一的最佳热导率分配比($(u_h)_{\overline{\Pi}}$, $(u_l)_{\overline{\Pi}}$, $(u_k)_{\overline{\Pi}}$)和一个最佳压比 $\pi_{\overline{\Pi}}$。

3.1　最佳的热导率分配与最佳压比

图 4 显示了最佳热导率分配比($(u_h)_{\overline{\Pi}}$, $(u_l)_{\overline{\Pi}}$, $(u_k)_{\overline{\Pi}}$)与最佳压比 $\pi_{\overline{\Pi}}$ 随 η_c(假设 $\eta_c = \eta_t$)的变化规律，由计算可知，高温侧最佳热导率分配比(u_h)$_{\overline{\Pi}}$ 始终接近 0.5，基本不随 τ_1、τ_2、a、b、η_c 与 U_T 变化；而低温和用户侧最佳热导分配率比$(u_l)_{\overline{\Pi}}$ 和$(u_k)_{\overline{\Pi}}$ 随 τ_2、a、b、η_c 与 U_T 变化，随 τ_2、a、η_c、U_T 的增加与 b 的降低，$(u_l)_{\overline{\Pi}}$ 增加，而$(u_k)_{\overline{\Pi}}$ 降低；随 τ_2、b 的减少与 a、τ_1、η_c、U_T 的增加，$(\pi)_{\overline{\Pi}}$ 增加。

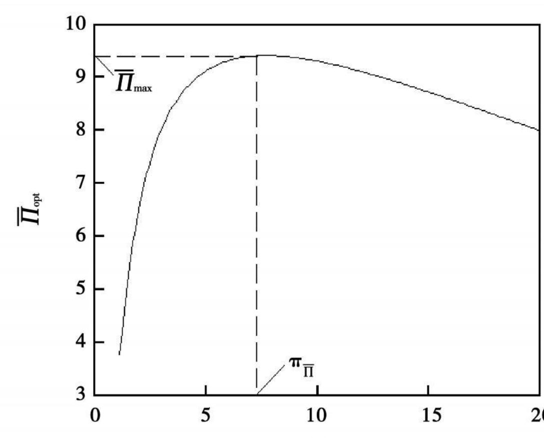

图 3　$\overline{\Pi}_{opt}$ 与 π 的关系

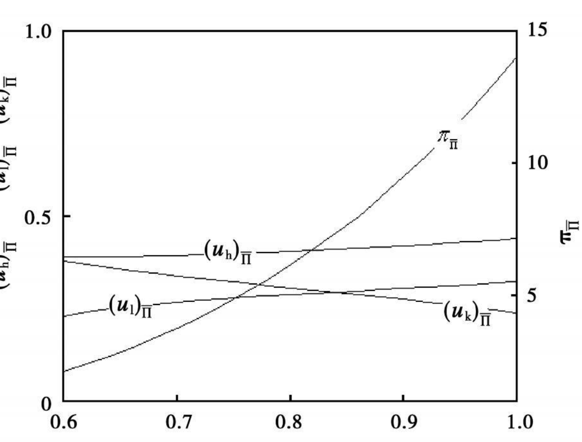

图 4　η_c 对$(u_h)_{\overline{\Pi}}$、$(u_l)_{\overline{\Pi}}$、$(u_k)_{\overline{\Pi}}$ 与 $\pi_{\overline{\Pi}}$ 的影响

3.2　利润率与㶲效率特性

图 5 显示了最优无因次利润率 $\overline{\Pi}_{opt}$ 与最优无因

次利润率条件下的㶲效率（$\eta_{\overline{\Pi}_{opt}}$）之间的变化关系。从图中可以看出，当 $\overline{\Pi}_{opt}$ 等于零时，$\eta_{\overline{\Pi}_{opt}}$ 并不为零，说明装置有功率产出，不一定有利润，跟实际情况吻合。两者的特性关系呈扭叶型，存在最大㶲效率 η_{max} 和相应的利润率 $\overline{\Pi}_{\eta}$，也存在最大利润率 $\overline{\Pi}_{max}$ 和相应的㶲效率 $\eta_{\overline{\Pi}}$，$\eta_{\overline{\Pi}}$ 即为装置的有限时间㶲经济性能界限。

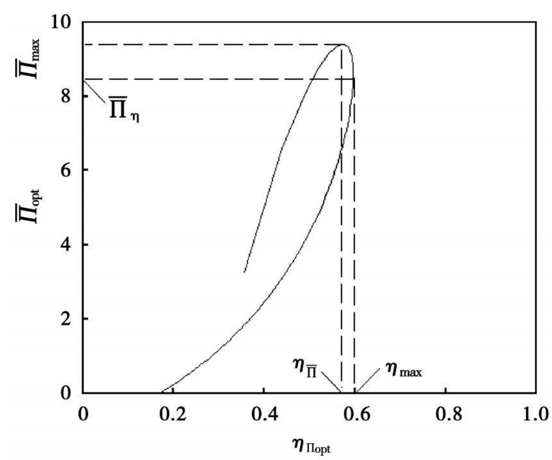

图 5　$\overline{\Pi}_{opt}$ 与 $\eta_{\overline{\Pi}_{opt}}$ 的关系

3.3　最大利润率与㶲经济性能界限

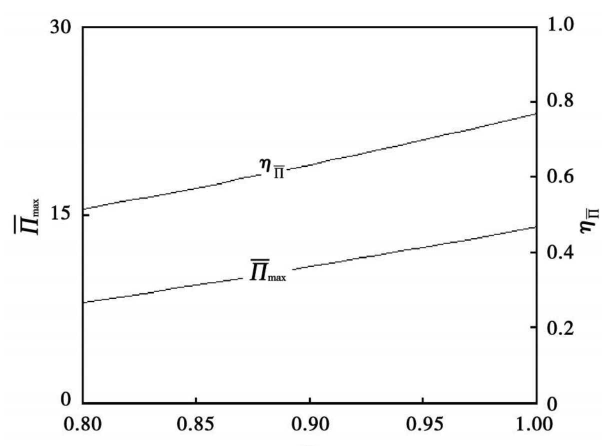

图 6　η_c 对 $\overline{\Pi}_{max}$ 与 $\eta_{\overline{\Pi}}$ 的影响

图 6 显示了最大无因次利润率 $\overline{\Pi}_{max}$ 与㶲经济性能界限 $\eta_{\overline{\Pi}}$ 随 η_c（假设 $\eta_c = \eta_t$）的变化规律，由计算可知，随 η_c 与 U_T 的增加，$\overline{\Pi}_{max}$ 与 η_c 均增加，但随 U_T 的增加，$\overline{\Pi}_{max}$ 与 $\eta_{\overline{\Pi}}$ 增加的幅度越来越小；随 τ_1 与 a 的增加，$\overline{\Pi}_{max}$ 与 $\eta_{\overline{\Pi}}$ 均有所增加，但 $\eta_{\overline{\Pi}}$ 增加的幅度越来越小；随 b 的增加，$\overline{\Pi}_{max}$ 增加，$\eta_{\overline{\Pi}}$ 减小；$\overline{\Pi}_{max}$ 与 $\eta_{\overline{\Pi}}$ 均与 τ_2 呈类抛物线关系，即存在最佳的用户侧温度，利用式（16）以㶲输出率为目标计算，可同样发现存在最佳的用户侧温度，而以往文献[6～11，17～18]对联产装置的优化分析，用户侧温度都是越低越好，显示出修改热量㶲的计算方法后，得出不同以往的优化结论。

3.4　最大㶲效率和相应的利润率

图 7 显示了最大㶲效率 η_{max} 与相应的利润率 $\overline{\Pi}_{\eta}$ 随 η_c（假设 $\eta_c = \eta_t$）的变化规律。由计算可知，η_{max} 与 $\overline{\Pi}_{\eta}$ 均随 η_c、U_T、τ_1 和 τ_2 的增加而增加，但随 U_T 的增加，η_{max} 与 $\overline{\Pi}_{\eta}$ 增加的幅度越来越小。此外，对比图 6 与图 7 可以看出，随压气机与透平效率 η_c 的增加，一直有最大利润率工作点，但最大㶲效率工作点会消失。

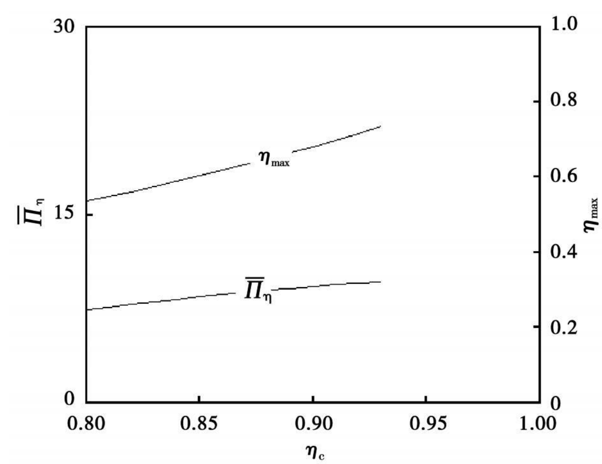

图 7　η_c 对 η_{max} 与 $\overline{\Pi}_{\eta}$ 的影响

4　结　论

应用有限时间㶲经济性能优化方法，对恒温热源条件下的不可逆闭式布雷顿热电联产装置进行利润率优化。研究表明，对于给定的总热导率，在高温、低温和用户侧换热器之间，存在唯一的最佳热导率分配比，同时存在唯一的最佳压比，使得装置的无因次利润率取得最大值。而且，高温侧热导率分配比始终接近于 0.5，低温、用户侧热导率分配比随用户侧温度、总热导率、压气机与透平效率、价格比变化而变化。此外，还对最优的无因次利润率与㶲效率特性，各种因素对最大利润率与㶲经济性能界限的影响，对最大㶲效率和相应利润率的影响进行了分析。对热量㶲的计算方法进行了探讨，提出不同以往热量㶲的计算方法，发现存在最佳用户温度的新规律。本文的优化变量是 3 个换热器的热导率与压比，用户侧温度只是用来分析对联产优化性能的影响。此外，用户侧温度虽然取决于用户的需求，但

是用户的需求是在一定变化范围内,也是多种多样的,本文的结论对寻求用户的需求和联产装置的最优匹配是有一定意义的。

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新技术、新产品

UTR－Ⅱ型锅炉的研制

据《Kawasaki Technical Review》2008 年 1 月号报道, 为了减少 LNG(液化天然气)运输船的燃料消耗, 川崎集团公司计划研制新概念锅炉汽轮机装置, 在该装置中提高了蒸汽的压力和温度, 并采用再热循环。Kawasaki Plant Systems 公司已经结束了该锅炉的研制工作。

介绍了 UTR－Ⅱ型再热式双锅筒水管锅炉的主要参数: 最大蒸发量 55 t/h(再热蒸汽量 43 t/h), 常用蒸发量 41 t/h(再热蒸汽量 37 t/h); 过热器出口压力 12 MPa G(表压)、温度 565 ℃; 再热器出口压力 2.6 MPaG、温度 545 ℃; 再热常用负荷时的锅炉效率 90.2%; 给水温度为 229 ℃; 再热常用负荷时的空气比 1.085。

每艘 LNG 运输船装用 1 台汽轮机, 每台汽轮机配 2 台锅炉, 用重油作为燃料。

还分析了高压引起的问题和解决办法, 以及高温引起的问题和解决办法。

给出了 UTR－Ⅱ型锅炉的剖视装配图、过热器和再热器的支承构件图、不再热和再热运行的切换图、在挡板关闭方式下烟气的自然循环图、以及蒸汽温度控制图。

(吉桂明　摘译)

Journal of Engineering for Thermal Energy & Power. —2009, 24(5). —588 ~591

The temperature of atmospheric environment has a big influence on the performance of a gas turbine. An additional installation of an inlet air atomization and cooling system is of enormous practical value for improving the performance of the gas turbine. Through an analysis of the working principle of an inlet air atomization and cooling system of a gas turbine, proposed were a design version and functional realization of a PLC-based (programmable logic controller) gas turbine inlet air atomization and cooling control system. The operation results show that the control system enjoys a high automation level, a good operating stability and a reliable performance. After the gas turbine inlet air atomization type cooling skid equipped with the control system in question has been put into operation, the power output of a PG6551(B) type gas turbine increased, relatively speaking by 8.35% and the efficiency rose by about 3.24%. **Key words**: gas turbine, inlet air cooling, control technique

冷热电联产系统新评价准则研究＝**A Study of New Evaluation Criteria for Combined Cooling-heating-power Cogeneration Systems**[刊, 汉] / HE Bin-bin, DUAN Li-qiang, YANG Yong-ping (Education Ministry Key Laboratory on Power Plant Equipment Condition Monitoring and Control, College of Energy Source and Power Engineering, North China Electric Power University, Beijing, China, Post Code: 102206) // Journal of Engineering for Thermal Energy & Power. —2009, 24(5). —592 ~596

From the intrinsic characteristics of energy stepped utilization of a cooling-heating-power cogeneration system, presented were the criteria for evaluating energy stepped utilization rates. The criteria were obtained by accumulative adding of energy utilization rates of power generation, refrigeration and heat supply, multiplied by various weighting coefficients respectively. The reference point for comparison was first determined and then a layer-by-layer analytic method was adopted to obtain the weighting coefficients for various energy utilization rates at the reference point. Then, the weighting coefficients at the reference point were corrected by using the temperature of the cold and hot product and the ambient temperature to obtain the weighting coefficients under other circumstances. In conjuction with a calculation case of a practical cogeneration system, the method for using the evaluation criteria was given, and an analysis and comparison with the original evaluation criteria were performed. The research results show that the evaluation criteria under discussion feature rationality, thus adequately serving as a practical method for evaluating and comparing combined cooling-heating-power cogeneration systems. **Key words**: combined cooling-heating-power cogeneration, evaluation criterion, energy stepped utilization rate

燃气机热泵冷热电三联供系统热经济学分析＝**Thermo-economics Analysis of a Cooling-heating-power Cogeneration System for a Gas Engine-driven Heat Pump**[刊, 汉] / FANG Zheng, YANG Zhao, CHEN Yi-guang (Thermal Energy Research Institute, Tianjin University, Tianjin, China, Post Code: 300072) // Journal of Engineering for Thermal Energy & Power. — 2009, 24(5). —597 ~603

A cooling-heating-power cogeneration system for a gas engine-driven heat pump was analyzed by adopting a thermo-economics analytic method. From a calculation and analysis of exergy cost differences and exergy economic factors of subsystems as well as exergy economic coefficient of the whole system under the condition of the following 4 influencing factors, i.e. various rotating speeds, evaporation temperatures, condensing temperatures and natural gas prices, the authors have proposed some improvements necessary for the above cooling-heating-power cogeneration system and problems meriting attention in setting a rational transmission ratio during the design of the system. Moreover, they have also concluded that the system enjoys broad prospects for its application in China. **Key words**: cooling-heating-power cogeneration, gas-turbine driven heat pump, thermo-economics, exergy cost difference, exergy economic factor, exergy economic coefficient

不可逆闭式布雷顿热电联产装置火用经济性能优化＝**Exergy Economic Performance Optimization of an Irre-**

versible Closed Type Brayton Heating-and-power Cogenration Plant[刊,汉]/ TAO Gui-sheng, CHEN Lin-gen, SUN Feng-rui (Postgraduate School, Naval University of Engineering, Wuhan, China, Post Code: 430033) // Journal of Engineering for Thermal Energy & Power. — 2009, 24(5). —604~608

By adopting a finite time thermodynamic method, studied was the exergy economic performance of an irreversible closed type Brayton cogeneration plant under the condition of a constant temperature heat source and derived were its profit margin and exergy coefficient analytic expression. By employing a numerical calculation method, with the profit margin serving as a target, optimized were the distribution of heat conductivity and the choice of pressure ratio. The authors have studied the optimum profit margin and corresponding exergy efficiency characteristics and analyzed the influence of various design parameters of the cogeneration system on its optimized performance. The research results show that for a given total heat conductivity, there exist only one optimum heat conductivity distribution ratio and pressure ratio among heat exchangers at high temperature, low temperature and end-user side, which results in an maximal value of the non-dimensional profit margin of the plant. In the meantime, there is an optimum end-user temperature. **Key words**: finite time thermodynamics, closed type Brayton heating-and-power cogeneration plant, exergy economic performance, profit margin

循环流化床锅炉热惯性分析=**An Analysis of Thermal Inertia of a CFB (Circulating Fluidized Bed) boiler**[刊,汉]/LI Jin-jing, LI Yan, LU Jun-fu, et al (Education Ministry Key Laboratory on Thermal Sciences and Power Engineering, Thermal Energy Engineering Department, Tsinghua University, Beijing, China, Post Code: 100084)// Journal of Engineering for Thermal Energy & Power. —2009, 24(5). —609~613

The thermal inertia of a CFB (Circulating Fluidized Bed) boiler represents an important factor affecting the boiler dynamic characteristics. From the standpoint of a dynamic energy balance, defined was the thermal inertia of the CFB boiler. For boilers rated at 6 different capacities, calculated respectively were their thermal inertia magnitudes in various links of energy transfer. The calculation results show that the total thermal inertia magnitude of a boiler increases with an increase of its capacity, however, its unit evaporative capacity decreases with an increase of its capacity. Thermal inertia of a working medium and refractory materials constitutes a control link in the energy transfer process. As far as an economizer is concerned, metallic thermal inertia is of equal importance to that of a working medium. The thermal inertia of refractory materials in superheaters/reheaters is of the same magnitude order as the metallic thermal inertia. In water walls/panels, the working medium thermal inertia is considered as the biggest. **Key words**: circulating fluidized bed boiler, heat transfer, thermal inertia

电站煤粉锅炉炉内压力信号的混沌特性=**Chaotic Characteristics of In-furnace Pressure Signals in a Pulverized Coal-fired Utility Boiler**[刊,汉]/NIU Wei-ran, QIU Yan, TIAN Mao-cheng (College of Energy Source and Power Engineering, Shandong University, Jinan, China, Post Code: 250061), LIU Zhi-chao (Thermal Energy Research Institute, Shandong Electric Power Academy, Jinan, China, Post Code: 250021)// Journal of Engineering for Thermal Energy & Power. — 2009, 24(5). —614~617

The in-furnace process of a large-sized coal-fired utility boiler features a complex non-linear time-variation one. With a comprehensive consideration of the practical operation of a coal-fired utility boiler and the application of chaotic kinetics theory, studied were the in-furnace chaotic motion characteristics with the in-furnace pressure serving as a parameter-variable. By employing a power spectrum method, Cao method and Kolmogorov entropy, it can be confirmed that the in-furnace pressure signals are chaotic ones involving random signals, of which the fluctuation range is about 50% to 75% of that of the chaotic signals. If the phase space for chaotic motion is restructured by a time delay of 8 s and the number of inserted dimensions totaling 8 in a pulverized-coal boiler under a normal operating condition, the correlative in-furnace motion dimension is ascertained as 6.56 through a calculation and there exists a positive Lyapunov index of 0.019 4 with Kol-