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换热器特性参数与热力性能熵产分析

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摘 要: 引入无量纲熵产数 N_S 表示换热器热力完善程度, 对换热器进行性能熵产分析与评价, 研究了表征换热器换热性能的特性参数: 进口温度比 α、预热温度比 β、水当量比 W、效能 ε、传热单元数 NTU 及流型对换热性能的影响及相关关系。研究表明, N_S 值随 α 增大而增大; 提高 β 值(大于临界值), 可降低 N_S 值; NTU 值应大于 1, 考虑经济性, 其值不宜过大; $W<1$ 的非平衡流是造成有效能损失的原因; 应使 W 趋于 1; ε 值大于 0.5 而趋于 1, 可减少不可逆性及提高换热率。通过熵产分析, 可揭示换热器能耗产生的原因, 确定热力参数的优化匹配, 达到节能目的。

关 键 词: 换热器; 热力特性参数; 熵产分析; 热力性能评价

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引 言

能源消耗的飞速增长要求人们更有效地利用能源, 减少能量损失显得尤为重要。换热器作为能源、化工、轻工和动力等众多工业领域中使用最为广泛的热力设备之一, 对其进行优化以提高性能, 对于改进热力系统用能过程, 降低能量消耗具有非常重要的意义。熵产分析基于热力学第二定律, 从能量利用的质量角度出发, 是评价换热器热力性能完善程度的指标[1]。用它指导换热器设计, 使其接近热力学上的理想状况, 还可从能源合理利用角度比较不同换热器传热和流动性能的优劣。本文根据表征换热器换热性能的诸因素, 提出了换热器热力特性参数的概念, 同时引入无量纲熵产数 N_S, 对基本流型换热器进行热力过程不可逆性分析和完善程度评价, 探讨了热力特性参数对换热性能的影响及相互关系, 为工程应用提供参考。

1 换热器熵产数分析

Bejan 将熵产数的概念应用于换热器热力不可逆分析[2], 它是换热器热力性能完善程度的评价指标, 其值大表明换热器热力不可逆性高, 有效能损失大, 热力完善程度低; N_S 值低则相反。

以换热器为控制容积系统, 进出口有工质流动, 外表面视为绝热, 并与环境无功交换, 冷热流体间只有热传递。对于这样的换热器模型, 其熵产为:

$$\dot{s}_{irr}=\Delta\dot{s}_h=\dot{m}_h\Delta s_h+\dot{m}_c\Delta s_c \tag{1}$$

由热力学第一定律知: $Tds=dh-(1/\rho)dp$, 得到:

$$\dot{s}_{irr}=(\dot{m}c_p)_h\ln\frac{T_{h2}}{T_{h1}}+(\dot{m}c_p)_c\ln\frac{T_{c2}}{T_{c1}}-\dot{m}_hR\ln\frac{P_{h2}}{P_{h1}}-\dot{m}_cR\ln\frac{P_{c2}}{P_{c1}} \tag{2}$$

以冷流体水当量为基准, 换热器的熵产数为:

$$N_S=\frac{\dot{s}_{irr}}{W_c}=\frac{1}{W}\ln\frac{T_{h2}}{T_{h1}}+\ln\frac{T_{c2}}{T_{c1}}-\frac{R}{Wc_{p_h}}\ln\frac{P_{h2}}{P_{h1}}-\frac{R}{c_{p_c}}\ln\frac{P_{c2}}{p_{c1}} \tag{3}$$

式中: W—水当量比。下标: h 和 c—热、冷流体; 1、2—进口和出口。式右边第一、二项是传热引起的不可逆损失; 第三、四项是粘性流动引起的不可逆损失, 对换热器不可逆性的分析应考虑传热与粘性流动两方面。Hany 对平衡流和非平衡流换热器[3], 在不同 Re 数、换热通道几何参数($4L/D$)、进出口参数(T_c/T_h)条件下, 进行传热和粘性流动与熵产关系的研究, 发现换热器中流动引起的熵产和由传热引起的熵产相比很小, N_S 主要取决于传热因素。也可通过对下式分析得出:

$$G=u\rho=Re\mu/D_h \tag{4}$$

$$\dot{m}=G\cdot A \tag{5}$$

$$f=f(Re) \tag{6}$$

$$\Delta p=0.5f(G^2/\rho)(L/D_h) \tag{7}$$

由式(4)~式(7)看出, 当 Re 数为一定值时质量流率和阻力系数可得到确定, 压降也相应确定。式(3)中, N_S 值只和温度比有关。Re 数增加会使 N_S 相应地增加, 但与温度变化导致 N_S 值的增加相比

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是非常小的。式(3)显示,流动熵产项乘了因子 R/c_p($R/c_p<1$)。鉴于流动引起的熵产和由传热引起的熵产相比很小,可忽略不计,式(3)变为:

$$N_S=(1/W)\ln(T_{h2}/T_{h1})+\ln(T_{c2}/T_{c1}) \tag{8}$$

2 换热器特性参数

式(8)中,温度 T_{h1}、T_{h2}与效能 ε 相关,为深入对换热器熵产数分析和换热性能的认识,定义以下无量纲数(换热器热力特性参数):

有效度 $\varepsilon=\frac{(\dot{m}c_p)_h}{(\dot{m}c_p)_c}\frac{(T_{h1}-T_{h2})}{(T_{h1}-T_{c1})}=\frac{T_{c2}-T_{c1}}{T_{h1}-T_{c1}}$,流体进口温度比 $\alpha=T_{h1}/T_{c1}$,预热温度比 $\beta=T_{c2}/T_{c1}$,水当量比 $W=w_c/w_h=(\dot{m}c_p)_c/(\dot{m}c_p)_h$,传热单元数 $NTU=KF/w_c$。式(8)由热力特性参数表示为:

$$N_S=\frac{1}{W}\ln\left(1-W\frac{\beta-1}{\alpha}\right)+\ln\beta$$
$$=\ln[1+\varepsilon(\alpha-1)]+\frac{1}{W}\ln[1-\varepsilon W(1-1/\alpha)] \tag{9}$$

2.1 特性参数 α、β、W 对换热性能的影响

由式(9)可得熵产数 N_S 与水当量比 W 以及进出口温度之间的关系。

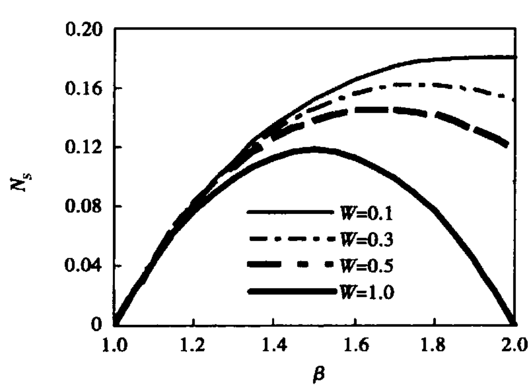

图 1 N_S 和β、W 关系曲线($\alpha=2.0$)

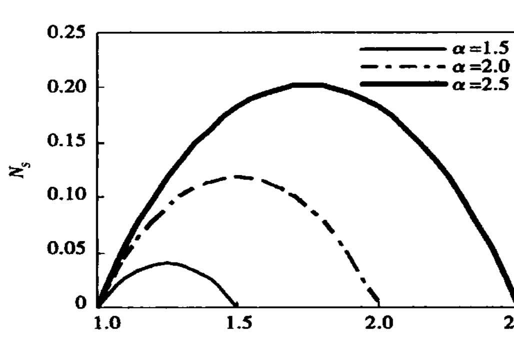

图 2 N_S 和β、α关系曲线($W=1.0$)

由图1和图2可见,在水当量比 W 和进口温度比 α 一定情况下,N_S 随着温度比β的增加而呈抛物线状发展,从 N_S 低值增至最大值后递减。N_S 最大值对应的β值可由式(9)得到:临界 $\beta_c=(W+\alpha)/(W+1)$,β_c 对应的温度 T_{c2}为临界温度,$T_{c2}=(T_{h1}+WT_{c1})/(1+W)$。水当量比 W 为1.0时,当 $\alpha=\beta$(理想情况),可使 $N_S=0$;W 小于1.0时,即使理想情况下,即 $\alpha=\beta$,N_S 也不为零,在 α 和 W 一定时,尽可能使 β 大于β_c,而趋近于 α 可减小N_S。适当选择 W 和β可以减少传热过程中的N_S。而对于 W 和β一定情况下,热流体的进口温度愈高,α 愈大,N_S 愈高,可用能损失就愈大。

2.2 特性参数 NTU、W 及流型对换热性能的影响

熵产数 N_S 不仅取决于 W、α、β,还取决于传热单元数 NTU、流动型式等。下面分析了换热器的流动型式、NTU 及 W 对换热性能的影响。

顺流换热器的对数平均温差:

$$\Delta T_m=\frac{(T_{h1}-T_{c1})-(T_{h2}-T_{c2})}{\ln[(T_{h1}-T_{c1})/(T_{h2}-T_{c2})]} \tag{10}$$

代入 $Q=w_h(T_{h1}-T_{h2})=w_c(T_{c2}-T_{c1})=KA\Delta T_m$。其中:$T_{h2}=T_{c1}(\alpha-\theta)$; $T_{c2}=T_{c1}(1+\theta/W)$,得到:$N_S=\frac{1}{W}\ln(1-\frac{\theta}{\alpha})+\ln(1+\frac{\theta}{W})$。

其中:

$$\theta=\frac{W(\alpha-1)}{1+W}[1-e^{-(1+W)NTU}] \tag{11}$$

由上式可得不同水当量比 W 和传热单元数 NTU 与熵产数N_S 的关系(见图3和图4)。

由图3可知,顺流换热器在 α 和 W 一定情况下,N_S 随着NTU 单调递增,在 NTU 小于0.5范围内递增过程非常明显;当 NTU 趋于1,N_S 缓慢趋于一稳定值。NTU 是换热强度的无量纲度量,在 NTU 小于0.5的区域,熵产数虽甚小,但换热能力差,对于实际工程应用是不可行的。由图4可知,对于 NTU 和α一定情况下,W 愈大,其传热过程的熵产数愈小。当 W 远大于1,其 N_S 的确较小,但在工程应用中,这种非平衡流配置不合理,因此从能量合理匹配与利用角度,使 W 趋于1,相应取得较小的熵产数 N_S。

逆流换热器平均温差代入能量平衡式,得:

$$T_{h2}=T_{c1}(\alpha-\theta);\quad T_{c2}=T_{c1}(1+\theta/W) \tag{12}$$

由此,$N_S=\frac{1}{W}\ln(1-\frac{\theta}{\alpha})+\ln(1+\frac{\theta}{W})$,式中:

$$\theta=\frac{W(\alpha-1)(1-e^{(W-1)NTU})}{1-We^{(W-1)NTU}} \tag{13}$$

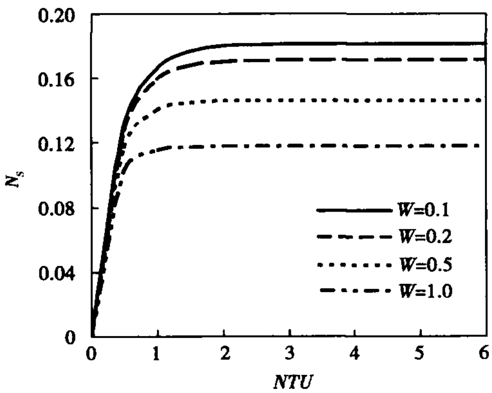

图3　N_S 和 NTU 关系曲线(顺流, α=2.0)

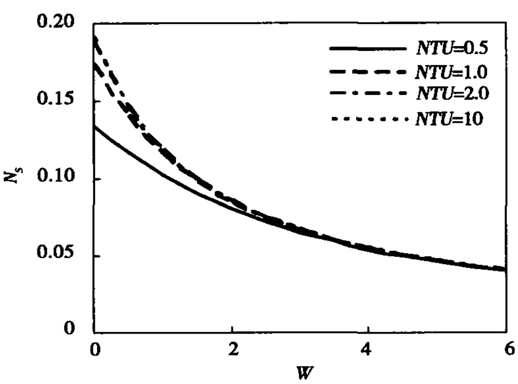

图4　N_S 和 W 关系曲线(顺流, α=2.0)

可得不同水当量比 W 和传热单元数 NTU 与熵产数 N_S 的关系(见图5和图6)。

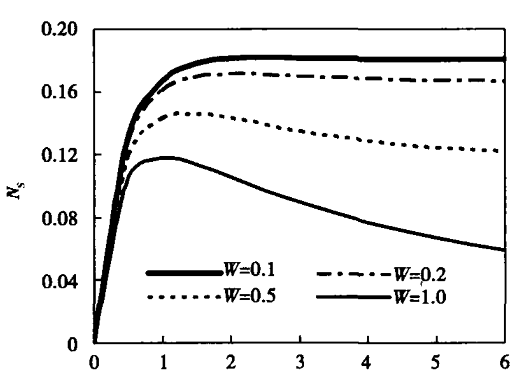

图5　N_S 和 NTU 关系曲线(逆流, α=2.0)

逆流情况下 N_S 随 NTU 不像顺流那样单调变化, N_S 随 NTU 增加而增加,在 NTU 为1附近缓慢增加到一最大值,当 $NTU=1$ 时, $N_{S,\max}=\ln\times\left[\frac{(T_{1,\mathrm{in}}+T_{2,\mathrm{in}})^2}{4T_{1,\mathrm{in}}T_{2,\mathrm{in}}}\right]$,数值上和顺流换热器的熵产极限值相等[4],但随着 NTU 的增加而逐步减少。逆流换热器当 NTU 很大时,熵产数可以很小,即可用能损失很小。因此在逆流换热器设计时, NTU 应远大于1,这样不仅换热能力量大,而且有用功损失明显减少, NTU 愈大,熵产数愈小。图6显示, NTU 取较大值时, N_S 随 W 增加而减少,在 $W=1$ 处取得最小值。

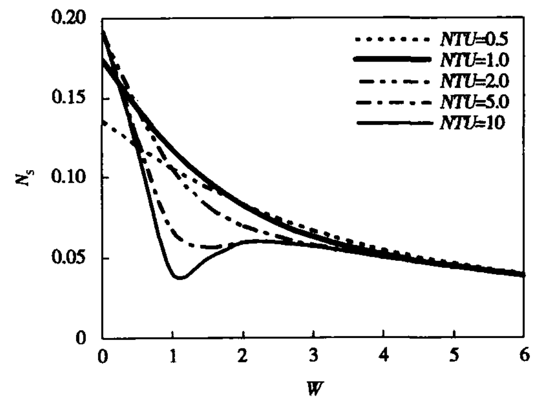

图6　N_S 和 W 关系曲线(逆流, α=2.0)

叉流换热器,因难以求得局域温度的分析式,可采用数值计算法。文献[5]给出交叉流换热器的熵产数表达式和数据表。限于篇幅,本文只探讨冷热流体均不混合的型式:

$$N_S=\frac{1}{W}\ln\left(1-\frac{\theta}{\alpha}\right)+\ln\left(1+\frac{\theta}{W}\right),\ \theta=W(\alpha-1)\times\{1-\exp\{WNTU^{0.22}[\exp(-W^{-1}NTU^{0.78})-1]\}\}\quad(14)$$

由图7和图8知,叉流换热器的热力性能介于顺流和逆流换热器之间,在 NTU 小于1范围内, N_S(叉流)和 N_S(逆流), N_S(顺流)差别不大, N_S(逆流)略高于 N_S(顺流)。NTU 大于1时, N_S(逆流)总是低于 N_S(顺流),差别非常明显,可见,逆流和叉流换热器总体上是优于顺流换热器的。NTU 越大,一定程度上传热量也越大,对于叉流和逆流换热器,却可

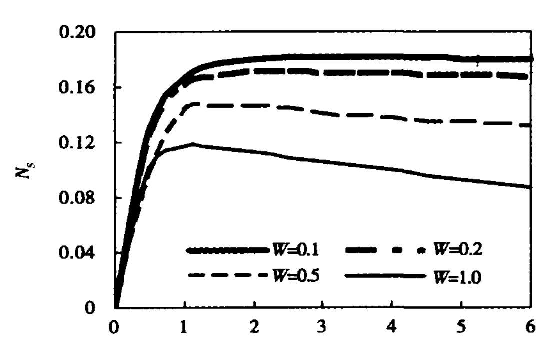

图7　N_S 和 NTU 关系曲线(叉流, α=2.0)

以得到更小的熵产数, 能量损失也越少, 但随 NTU 的增加, N_S 减小趋势相应减缓, 所以不宜采用过高的 NTU 值, 否则投资费用的增加大于熵产数 N_S 降低所获得的效益。

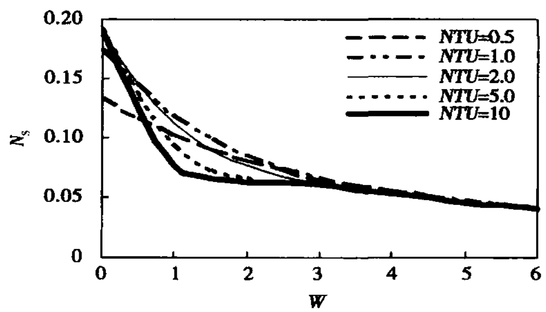

图 8　N_S 和 W 关系曲线(叉流, $\alpha=2.0$)

2.3　熵产数 N_S 与效能 ε 的关系

文献[6] 指出, 在进出口温度均匀条件下换热器的效能与熵产数之间存在对应关系, 它们都是表征热力学完善程度的。这种对应关系对于换热器优化与设计的影响, 相关文献所提供的信息很少。本文对效能与熵产数的对应关系进行了探讨。

经推导得:

$$N_S=\ln\{[(\alpha-1)\varepsilon+1][1-(1-1/\alpha)\varepsilon W]^{1/W}\} \tag{15}$$

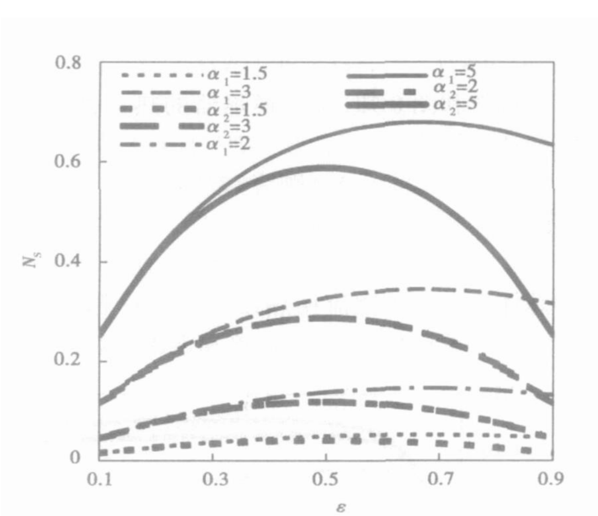

图 9　N_S 和 ε 关系曲线 α_1($W=0.5$), α_2($W=1$)

图 9 为 $W=1$(平衡流)和 $W=0.5$(非平衡流)时, 熵产数 N_S 随着不同温度比 α 与效能 ε 的关系, 图中 1、2 分别代表在 $W=0.5$ 和 $W=1$ 情况。对于平衡流, $N_S\sim\varepsilon$ 呈抛物线型, N_S 随着 ε 增加, 增加到最大值(此时 $\varepsilon=0.5$, $dN_S/d\varepsilon=0$)然后逐渐减小, N_S 最小值出现在 ε 最小或最大处。ε 的减小意味着传热的减弱, N_S 明显地减小而趋于零。对 ε 值小的换热器, 因其较差的工况和不合理的设计是没有存在的必要。由图可见, 换热器的效能应大于 0.5, 这样在较大 ε 的情况下, 不仅工况合理且可实现较小的不可逆损失。

对于非平衡流换热器, N_S 随着 α 增大而增大, 随着 W 增大而减小。相同 α 情况下, N_S(非平衡流)与 N_S(平衡流)曲线相比, 随 ε 增加过程中有所不同, 其 N_S 值在 $\varepsilon=0.5$ 后, 继续增大至最大值, 然后减小, 减小幅度也没有平衡流换热器那么明显。

在 $\varepsilon=1$, 忽略压降理想情况下, 式(15)可简化为:

$$N_S=\ln\{\alpha[1-(\alpha-1)W/\alpha^{1/W}\} \tag{16}$$

对于平衡流 $W=1$, $\varepsilon=1$, 压降为零, $N_S=0$, 可视为不存在不可逆损失; 对于非平衡流 $W\neq1$, 其不可逆性随温度比增大、水当量比减小而增大。可见, 流动的非平衡配置造成换热器不可逆性的发生[3]。综上分析, ε 的增加会使 N_S 增加, 直至 N_S 最大值(取决于 α 的不同)然后减小。对于平衡流换热器, 其最大值位于 $\varepsilon=0.5$ 处。非平衡流换热器则在大于 0.5 处。鉴于低效能意味着低换热率, 建议换热器的 ε 大于 0.5 而趋于 1, 可兼顾低不可逆性及高换热率。提高 W 使其趋于 1 的平衡流配置, 可降低换热器不可逆损失。

为验证上述理论, 本文对一用于水源热泵的逆流式套管换热器进行实验研究。在热流体水当量和进口温度比 α 恒定, 冷流体不同流量情况下, 对换热器熵产数 N_S(由式(13)得到)和实验计算值(根据进出口参数由式(1)和工质热物性得到)进行比较。

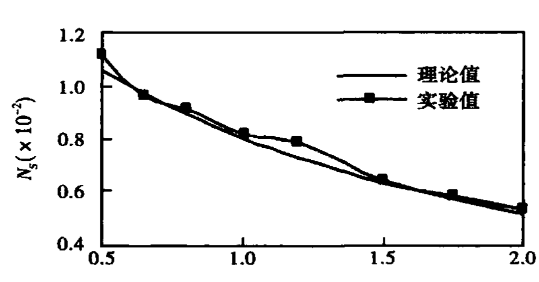

图 10　N_S 理论与实验比较(逆流, $\alpha=1.2$)

由图 10 可见, 理论和实验计算值相当吻合, 证明本文的数学模型及方程的建立是正确的。

3　结　论

(1) 熵产数 N_S 值大表明换热器热力不可逆性高,热力完善程度低,有效能损失大,能量未得到有效利用;N_S 值低则相反。

(2) 对于 W 和 β 一定情况下,α 愈大,N_S 愈高,可用能损失就愈大。在 α 和 W 一定时,尽量提高预热比 β,使其大于临界值,相应 N_S 减小。

(3) N_S 随 NTU 的变化视流动型式的不同而不同。N_S 在 NTU 小于0.5范围变化显著,在 NTU 为1趋于最大值。NTU 值小于0.5时,对工程应用没有意义,NTU 值应大于1。当 NTU 增加到一定范围,N_S 降低趋势减缓,鉴于投资经济性,换热器不宜采用过大的 NTU。

(4) N_S 随 α 增大而增大,随 W 增大(趋于1)而减小。在其它条件相同情况下,平衡流传热过程中 N_S 最小。水当量比 $W<1$ 的非平衡流是造成有效能损失的一大原因,应使 W 趋于1可降低换热器不可逆损失。

(5) N_S 随 ε 增加呈抛物线趋势达到最大值(取决于 α 的不同)后逐渐变小,对于平衡流换热器,其 N_S 最大值位于 ε 为0.5,非平衡流换热器的 N_S 最大值在 ε 大于0.5处。换热器的 ε 应高于0.5而趋于1,可兼顾低不可逆性及高换热率。

在工程技术领域中,换热器的应用非常广泛,通过熵产分析可揭示换热器能耗产生的原因,确定热力参数的优化匹配,提供热力优化手段,达到节能目的。

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(编辑　辉)

新 工 艺

循环流化床煤泥锅炉木炭点火新工艺

兖州矿业(集团)公司鲍店煤矿煤泥热电厂在循环流化床煤泥锅炉运行初期使用原先设计的由点火系统点火,点火时间为90~100 min,点火时间长、耗资大、易结焦。改用木炭点火以后,点火时间缩短为30~50 min,耗资仅为原来的1/3,极有推广应用的价值。

该厂规模为三炉三机,主要设备为JG—35/3.82—Mn型循环流化床煤泥锅炉、抽凝式汽轮机机组和汽轮发电机组。木炭点火工序为:全面检查将启用的锅炉,主辅机、炉膛、风帽均良好后,向炉膛内加入10~20目细石英砂,厚度为230~250 mm,石英砂可以由部分细渣代替;在一燃烧床上加入300~400 kg木炭,均匀散开,并且用少量柴油洒匀,便于引燃;启动鼓、引风机,调整负压为50~100 Pa,用棉纱将炉膛内的木炭引燃;适当地增加鼓引风量,使石英砂逐步达到微沸腾状态;待到床温不断升高时,加入少量原煤,并且适当地加大风量,火焰稳定以后开启螺旋给煤机,使燃烧床燃起;调整快速风门,使火焰转向邻近的燃烧床,等到该床燃起以后投入螺旋给煤机,并且使火焰引向另外一床,最终3个燃烧床全部正常燃起,通过了并床阶段;炉膛稳定燃烧一段时间以后,向炉内加入8~10目较大颗粒的石英砂,使得料层的厚度不断加大,风室的风压增大到6~8 kPa燃烧比较理想;炉膛温度达到850 ℃左右,风室的风压达到要求以后开启锅炉顶部的螺旋给料机,少量给入煤泥,逐步提高锅炉出力,加大风量和煤泥给料量,最终达到正常运转。

(李剑峰　供稿)

tion Monte Carlo' s method

煤粉锅炉膜法富氧局部助燃技术开发及应用＝**Development and Applied Research of Local Combustion-supporting Technology Involving a Membrane-method-based Oxygen Enrichment for Pulverized Coal-fired Boilers** [刊，汉] /ZHANG Jia-yuan, ZHOU Jie-min (College of Energy Science and Engineering under the Central South University, Changsha, China, Post Code: 410083), YANG Shao-wei (Changsha Nonferrous Metallurgical Design Institute, Changsha, China, Post Code: 410011), CHEN Qiao-ping (Thermal Power Plant of Henan Subsidiary under China Aluminum Industry Company, Zhengzhou, China, Post Code: 450041) // Journal of Engineering for Thermal Energy & Power. －2007, 22(4). －391～394

In the light of various problems existing in a 150 t/h pulverized coal-fired boiler, such as high-temperature corrosion, slagging, low thermal efficiency and inferior combustion stability at low loads when no oil is used for combustion support etc., the authors have by adopting membrane method-based oxygen enrichment techniques developed local combustion-supporting technology featuring oxygen enrichment and designed a combustion-supporting system based on the above technology. Industrial experiments applying the technology under discussion were conducted for a pulverized coal-fired boiler. Practice has shown that with a reduction in combustible content in large slags and in fly ash, the thermal efficiency of the boiler has been increased by over 2.5%, the NO_x emission concentration lowered (627～768 mg/m^3 at a load of 120～150 t/h) and the combustion stability at low operating loads (50% of rated load) enhanced with no oil being provided for combustion support. As a result, various problems, such as slagging in furnace and high-temperature corrosion, have been effectively solved, blazing a new path for the safe, economical and environment-friendly operation of pulverized coal-fired boilers. **Key words**: pulverized coal-fired boiler, membrane method-based oxygen enrichment, local combustion support, energy saving, environmental protection

螺旋槽管换热过程的三维速度场与温度场耦合数值模拟＝**Numerical Simulation of a Three-dimensional Velocity Field Coupled With a Temperature Field for the Heat Exchange Process in a Spirally Grooved Tube** [刊，汉] /PENG Jie, YU En-lin (College of Mechanical Engineering under Yanshan University, Qinhuangdao, China, Post Code: 066004), JIANG Wei (College of Mechanical Engineering under Heilongjiang Institute of Science and Technology, Harbin, China, Post Code: 150027) // Journal of Engineering for Thermal Energy & Power. －2007, 22(4). －395～398

On the basis of structural features and heat transfer characteristics of a spiral-grooved tube heat exchanger, a three-dimensional geometric model was established for the flow and heat transfer in a heat exchanger with water serving as its working medium. By employing a finite-element analysis software ANSYS, simulated were the conditions of the velocity and temperature fields in the heat exchanger during its heat exchange process and obtained were the convective heat exchange coefficients for inner and outer walls of the spiral-grooved tubes respectively. The simulation results show that the deeper the groove, with an increase in Reynolds number, the better the heat-exchange performance. When the Reynolds number is relatively small, the greater the fin pitch, the poorer the heat exchange effectiveness. It has been found that the heat exchange coefficient of the spiral-grooved tubes is approximately 2.5 times that of the bare tubes when compared with the bare tube heat exchangers of the same kind, thus intensifying the heat transfer and providing a basis for the further theoretical research and widespread application of such products. **Key words**: spiral-grooved tube, intensification of heat exchange, numerical simulation, ANSYS, convective heat exchange coefficient

换热器特性参数与热力性能熵产分析＝**Entropy Production Analysis of Heat Exchanger Characteristic Parameters and Thermodynamic Performance** [刊，汉] / YU Min, MA Jun-jie, YANG Mo, et al (Shanghai University of Science and Technology, Shanghai, China, Post Code: 200093) // Journal of Engineering for Thermal Energy & Power. －2007, 22(4). －399～403

On the basis of the second law of thermodynamics, introduced was a non-dimensional entropy production number Ns to express the thermodynamic perfection degree of a heat exchanger. Through an analysis and evaluation of entropy production studied was the impact of heat exchanger characteristic parameters, such as inlet temperature ratio α, preheat temperature ratio β, water equivalent ratio W, effective energy ε, heat transfer units NTU and flow pattern, on heat exchange performance and mutual relations. The results of the study indicate that the entropy production number Ns will increase with the increase of inlet temperature ratio α. The increase of preheat temperature ratio β to a value greater than a critical one can decrease the value Ns. The number of heat transfer units NTU should be greater than 1. On account of cost-effectiveness, this value should not be excessively large. The non-equilibrium flow representing water equivalent ratio W being less than 1 is a major cause leading to a loss of effective energy, therefore efforts shall be made to enable W tend to be 1. The value ε should be greater than 0.5 and tend to be 1, thus reducing the irreversibility and enhancing the heat exchange rate. Through an analysis of the entropy production, the cause of energy consumption in a heat exchanger can be revealed and an optimized matching of thermodynamic parameters identified, thereby attaining the energy-saving objective. **Key words**: heat exchanger, thermodynamic parameters, entropy production analysis, thermodynamic performance evaluation

椭圆形封头大开孔结构强度分析=**Structural Strength Analysis of an Elliptic Head with a Large Opening**[刊, 汉] /XU Yan, LIANG Hai-dong, ZHANG Zhong-lian (Harbin No. 703 Research Institute, Harbin, China, Post Code: 150036), ZHENG Hong-tao (College of Power and Energy Source under the Harbin Engineering University, Harbin, China, Post Code: 150001) // Journal of Engineering for Thermal Energy & Power. —2007, 22(4). —404 ~408

In the light of the structural features and operating conditions of elliptic heads, by employing a finite-element analysis method, a calculation and analysis has been performed of the stress distribution on an elliptic head structure with a large opening. Measurements of static stress and experimental verification analyses have also been undertaken on a reduced-scale simulation test piece. The calculation and test results show that the dangerous part of the structure is located at the inner top side of the connected portion between the thin-walled large nozzle and the head, which can be taken as the design main control point. The stresses in the external-side high stress zone calculated by using the finite element method correspond with the actually measured stress results with an error of only 0.7% being recorded. A slightly bigger error exists between the calculated and actually measured values in the welding seams and the inner surface but it does not exceed 11.3%. This shows that the use of the three-dimensional finite-element analytic method to resolve the structural design of heads with a large opening is feasible and reliable. The analysis of causes leading to a relatively great error can serve as a helpful reference for the design, manufacture and tests of real products. **Key words**: finite element, large opening, head, stress distribution, stress evaluation

热力学㶲及其普遍化表达式的动力学特征=**Thermodynamic Exergy and Dynamic Characteristics of its Generalized Expressions**[刊, 汉] /HAN Guang-ze, LI Shao-xin (College of Physical Science and Technology under the South China University of Science and Technology, Guangzhou, China, Post Code: 510640), GUO Ping-sheng (College of Physics and Electronics Engineering under the Guangxi Normal University, Guilin, China, Post Code: 541004), HUA Ben (Education Ministry Key Laboratory on Heat Transfer Intensification and Process Energy-saving of the South China University of Science and Technology, Guangzhou, China, Post Code: 510640) // Journal of Engineering for Thermal Energy & Power. —2007, 22(4). —409 ~413

Analyzed were the physical meaning of work, heat, energy and exergy as well as their relations with thermodynamic laws. Doing work and transferring heat are the two ways for transferring and converting energy and exergy. The energy defined from the first law of thermodynamics has only a relative meaning. Exergy represents the ability of a system to do maximal useful work with respect to an environment. Relative to a specified environment, exergy is the status parameter of a system. The conventional calculation formula of exergy is derived from the first and second law of thermodynamics. From the viewpoint of dynamics discussed was the physical meaning of exergy and its generalized expression. The exergy originates from the non-equilibrium of a system with its environment. If any (or several kinds of) intensive property difference exists between a system and its environment, then under the driving force of such an intensive property difference, the system may automatically change to a state featuring a balance with its environment (dead state). During this process, the system can do work to the outside world and such an ability to do maximal useful work is defined as the exergy of the system. On the basis of energy postulation, the differential of exergy is generally expressed as a product of the intensive property difference and the differential of its conjugated extensive variables. The generalized expressions of exergy can fully reflect the physical meaning and dynamic characteristics of exergy. By employing the generalized expressions of energy and exergy, derived was the generalized expression of exergy losses. **Key words**: energy postulation, thermodynamic law, thermodynamic effective energy, intensive property, generalized expression of exergy