基于熵理论的有机工质管壳式换热器构形优化

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收稿日期: 2019-07-16; 接受日期: 2020-01-21; 网络版发表日期: 2020-05-11

国家重点研发计划(编号: 2017YFB0603503)和国家自然科学基金(批准号: 51779262, 51506220)资助项目

摘要 基于构形理论和熵理论, 以传热熵耗散率和总泵功率组成的复合函数为优化目标, 在总换热率和总传热面积一定的条件下, 对管壳式换热器有机工质蒸发过程进行构形优化, 得到最小复合函数和最佳换热管外径. 结果表明: 管壳式换热器最优构形与初始设计结构相比, 传热熵耗散率和复合函数分别降低9.54%和1.56%, 总泵功率增加22.40%. 这说明换热器综合性能的提高是通过较大程度地降低流体流动性能和显著减小传热不可逆性来实现的. 增大总换热面积和工质质量流率、减小工质进口温度以及选择合适的总换热管数和热水质量流率均有助于进一步提高换热器的综合性能. 熵耗散率和泵功率复合函数最小的换热器最优构形与换热率和泵功率复合函数最小的最优构形是完全不同的. 将熵理论应用到有机工质管壳式换热器的构形优化中, 为其结构优化设计提供了新的指导.

关键词 构形理论, 熵理论, 管壳式换热器, 熵耗散率, 总泵功率, 广义热力学优化

1 引言

管壳式换热器广泛应用于国民经济的各个领域^[1,2], 对管壳式换热器结构和性能参数进行优化是提高其性能的重要措施. 一些学者采用经典传热优化方法分别以传热率^[3~6]、效率^[5,7,8]、压降^[4,6,7]、总成本^[3,7,9]、熵产率^[7,10,11]、场协同数^[12,13]等为优化目标对其进行了优化, 使得其性能得到显著提高.

构形理论是由美国杜克大学Bejan^[14]于1996年提出, 它为各种传递过程和系统的优化设计提供了理论

支撑^[15~24]. 在构形优化中, 低阻抗通道的面积或体积是给定的, 这与对应问题的工程应用是一致的. 但构形理论自提出后也发现了一些不足, 如在低阻抗通道面积一定的条件下采用上一级最优方法进行构形优化时, 容易出现增加内部复杂度后系统性能降低的情况. Ghodoossi^[25], Kuddusi和Denton^[26], Kuddusi和Eğri-can^[27]对这一问题进行了详细的评论. 他们发现无论基于一维还是二维模型对导热问题进行构形优化, 构造体内部复杂度的增加并不总是使得系统性能得到提高, 由此对构形理论产生了质疑. 伍文君等人^[28]对这

引用格式: 冯辉君, 陈林根, 吴志祥, 等. 基于熵理论的有机工质管壳式换热器构形优化. 中国科学: 技术科学, 2020, 50: 1577–1587

Feng H J, Chen L G, Wu Z X, et al. Constructal optimization for an organic fluid shell-and-tube heat exchanger based on entransy theory (in Chinese). Sci Sin Tech, 2020, 50: 1577–1587, doi: [10.1360/SST-2019-0246](https://doi.org/10.1360/SST-2019-0246)

一问题进行了深入分析,发现虽然“采用上一级最优构造体建立下一级构造体的方法”是构形理论的关键步骤,但该“生长方法”并不能得到全局最优结果.他们由此提出了整体构形优化方法来解决这一问题.Boichot等人^[29]采用元胞自动机方法、Song和Guo^[30]采用仿生优化和模拟退火方法分别对矩形产热体内高导热材料分布进行了优化,发现新的非直角多分支高导热材料分布比传统直角多分支构形分布对应的导热性能更优.针对这一问题,Hajmohammadi等人^[31]采用构形理论对矩形产热体内非直角多分支高导热材料分布进行了优化,所得高导热材料最佳分布的导热性能比传统直角多分支构形分布显著提高.由此可见,文献^[29~31]采用非直角多分支高导热通道结构均使得矩形产热体导热性能得到提高,这也说明原始的基于直角分支的导热构形优化存在较严重的缺陷,对其加以完善是十分必要的.Escher等人^[32]将树形和平行微通道的传热和流动性能进行了比较,发现在一定条件下树形微通道性能并不优于平行微通道.此外,为简化问题,早期构形理论的研究工作采用解析解法将多维问题简化成了一维问题,当传输材料中流动方向不满足假设条件时,得到的优化结果存在一定的误差,这是早期构形研究的一个缺点.针对这一问题,部分学者采用数值计算方法建立了二维和三维模型,对多种传递问题进行了构形优化,一定程度上使得这一缺点得到弥补.综上所述,虽然构形理论在研究一些问题时还存在不足,但通过各种讨论和修正这一理论也将不断完善,从而不断接近或得到各种传递问题的最优结果,这是许多学者正在努力的方向.近年来,构形理论已经成功应用于一般换热器^[33,34]、非均匀产热体^[35~38]、蒸发器^[39,40]、锅炉过热器^[41]、双压透平^[42]等设备部件的最优设计.

一些学者将构形理论引入到管壳式换热器的性能优化研究中.Bonjour等人^[43]以最大温差最小为目标对添加肋片的多孔介质管壳式换热器进行了优化,发现当换热器横截面积较大时换热器肋片最优构形呈树状布置,反之呈辐射状布置.Azad和Amidpour^[44]建立了Y形换热管的管壳式换热器模型,以总费用最小为目标对其进行了构形优化,使得总费用比优化前降低了50%.Mirzaei等人^[45]以热效率最大为目标对Y形换热管的管壳式换热器进行了构形优化,构形优化后其热效率提高了28%,并发现换热器热效率和总费用两种

目标之间不能兼顾.Yang等人^[46,47]进一步建立了串并联Y形换热管的管壳式换热器模型,发现构形优化后混联的换热器总成本最大降低幅度达53.9%.Martins等人^[48]以孔隙率和热流体质量流率为优化变量对回热式管壳式换热器进行了构形优化,得到回热器双重最大效率为45.5%.冯辉君等人^[49]和蔡存光等人^[50]以换热率和总泵功率组成的复合函数最小为目标分别对有机工质和氨水工质的管壳式换热器蒸发过程进行了构形优化,发现所得最优构形设计方案在牺牲一定传热性能的前提下使得换热器流体流动性能大大提高.

为了更科学地开展传热优化研究,过增元等人^[51~53]提出了新的物理量——“焓”.对于定容热容量为 Q_{vh} 和温度 T 的物体,其表达式为^[51~53]

$$E_{vh} = Q_{vh}T/2. \quad (1)$$

由式(1)可得体积 v 内的焓耗散率为^[51~53]

$$\dot{E}_{vh\phi} = \int_v |\dot{q} \cdot \nabla T| dv, \quad (2)$$

式中, $\dot{q}$ 为热流密度矢量, ∇T 为温度梯度.基于焓理论^[54~64],许多学者研究了不同过程和系统的传热优化问题^[65~78],拓展了其应用范围.

在基于焓理论的管壳式换热器研究中,李孟寻等人^[66]对管壳式换热器进行了焓耗散数最小化的研究,优化后其有效度有所提高、泵功率大大降低.魏曙寰等人^[67]对多孔介质管壳式换热器进行了焓耗散率最小的研究,发现适当增加换热器内肋片数目有助于提高换热性能.Guo和Xu^[68]发现传热引起的焓耗散远大于流体流动引起的焓耗散,采用多目标研究方法有效地解决了这一问题,优化后管壳式换热器的泵功有效降低.Wen等人^[70]以当量热阻最小为目标对包含螺旋折流板的管壳式换热器进行了优化,优化后换热器当量热阻降低了7%,并发现壳侧流体流速对换热器当量热阻影响最大.

近年来,Bejan^[79]质疑焓理论为构形理论的翻版,事实上两种理论有明显区别,过增元^[80]对此质疑作了详细回复.在许多已有的研究中,焓理论一般作为一种优化准则,基于焓理论可实现基于焓定义的各种性能指标的优化;而构形理论一般作为一种结构优化设计方法,基于构形理论可实现一定尺寸约束下各种传输系统结构的优化.在文献^[49]研究工作的基础上,本文将焓理论与构形理论相结合,将两种理论引入到有

机工质蒸发问题的研究中, 以考虑传热焓耗散率和总泵功率的复合函数最小为目标, 对管壳式换热器进行构形优化, 并分析工质进口温度、总换热面积、总换热管数、工质和热流体质量流量等参数对构形优化结果的影响。

2 有机工质管壳式换热器模型

有机工质管壳式换热器模型和 T - s 图如图 1 所示。壳程流体为热水, 质量流率为 $\dot{m}_h$, 进出口温度分别为 $T_{h,in}$ 和 $T_{h,out}$ 。有机工质在管程内被加热, 质量流率为 $\dot{m}_{wf}$, 进口温度为 T_1 。工质分别经历预热阶段 ($T_1 \rightarrow T_{1'}$)、等温蒸发阶段 ($T_{1'} \rightarrow T_{2'}$) 和过热阶段 ($T_{2'} \rightarrow T_2$), 最终形成过热蒸气。换热管外径、长度和

数量分别为 $d_{h,out}$, L_e 和 N_{e1} , 管程和壳程数分别为 $N_{e,wf}$ 和 $N_{e,water}$, 忽略换热器外壳和环境的换热。

由能量平衡可知, 换热器在预热、蒸发和过热阶段的换热率分别为

$$\begin{aligned} Q_{h3} &= \dot{m}_{wf}(h_{1'} - h_1) = \dot{m}_h c_{ph}(T_{h,in} - T_{h1}) \\ &= K_{h3} A_{h3} \Delta T_{h3}, \end{aligned} \quad (3)$$

$$\begin{aligned} Q_{h2} &= \dot{m}_{wf}(h_{2'} - h_{1'}) = \dot{m}_h c_{ph}(T_{h1} - T_{h2}) \\ &= K_{h2} A_{h2} \Delta T_{h2}, \end{aligned} \quad (4)$$

$$\begin{aligned} Q_{h1} &= \dot{m}_{wf}(h_2 - h_{2'}) = \dot{m}_h c_{ph}(T_{h2} - T_{h,out}) \\ &= K_{h1} A_{h1} \Delta T_{h1}, \end{aligned} \quad (5)$$

式中, $h_i (i=1, 1', 2', 2)$ 为工质的比焓, c_{ph} 为热流体比热, $\Delta T_{hj} (j=1, 2, 3)$ 为各换热阶段的对数平均温差。

三个换热阶段的总传热系数 $K_{hj} (j=1, 2, 3)$ 分别为

$$K_{hj} = \frac{1}{\frac{d_{h,out}}{d_{h,in} \alpha_{hj,wf}} + \frac{d_{h,out} r_{e,in}}{d_{h,in}} + \frac{d_{h,out} \ln(d_{h,out}/d_{h,in})}{2\lambda_{h,wall}} + r_{e,out} + \frac{1}{\alpha_{h,water}}}, \quad j = 1, 2, 3, \quad (6)$$

式中, r_e 和 $\lambda_{h,wall}$ 分别为管壁污垢热阻和热导率。

壳侧对流换热系数 $\alpha_{h,water}$ 可表示为^[81]

$$\alpha_{h,water} = \frac{\lambda_{h,water}}{d_{h,equ}} \cdot \frac{f_{h,water}/8 \cdot (Re_{h,water} - 1000) Pr_{h,water}}{1 + 12.7 \sqrt{f_{h,water}/8} (Pr_{h,water}^{2/3} - 1)} \left[1 + \left(\frac{d_{h,equ}}{L_e N_{e,wf}} \right)^{2/3} \left(\frac{Pr_{h,water}}{Pr_{hw}} \right)^{0.01} \right], \quad (7)$$

式中, $f_{h,water} = (1.82 \lg Re_{h,water} - 1.64)^{-2}$, $Re_{h,water}$ 和 $d_{h,equ}$ 分别为雷诺数和当量直径, Pr_h 为普朗特数。

对于预热、蒸发和过热阶段, 管侧对流换热系数 $\alpha_{hj,wf} (j=1, 2, 3)$ 可表示为^[81,82]

$$\alpha_{h3,wf} = \frac{\lambda_{h3,wf}}{d_{h,in}} \cdot \frac{f_{h3,wf}/8 \cdot (Re_{h3,wf} - 1000) Pr_{h3,wf}}{1 + 12.7 \sqrt{f_{h3,wf}/8} (Pr_{h3,wf}^{2/3} - 1)} \left[1 + \left(\frac{d_{h,in}}{L_e N_{e,wf}} \right)^{2/3} \left(\frac{Pr_{h3,wf}}{Pr_{hw}} \right)^{0.01} \right], \quad (8)$$

$$\alpha_{h2,wf} = \begin{cases} 3.4 X_{tt}^{-0.45} \alpha_{i,all}, & 0 \leq x \leq 0.85, \\ 3.4 X_{tt}^{-0.45} \alpha_{i,all} \left[1 - \left(\frac{x-0.85}{0.15} \right)^2 \right] + \alpha_{v,all} \left(\frac{x-0.85}{0.15} \right)^2, & 0.85 \leq x \leq 1, \end{cases} \quad (9)$$

$$\alpha_{h1,wf} = \frac{\lambda_{h1,wf}}{d_{h,in}} \cdot \frac{f_{h1,wf}/8 \cdot (Re_{h1,wf} - 1000) Pr_{h1,wf}}{1 + 12.7 \sqrt{f_{h1,wf}/8} (Pr_{h1,wf}^{2/3} - 1)} \left[1 + \left(\frac{d_{h,in}}{L_e N_{e,wf}} \right)^{2/3} \left(\frac{Pr_{h1,wf}}{Pr_{hw}} \right)^{0.01} \right], \quad (10)$$

式中, $X_{tt} = \left(\frac{1-x}{x} \right)^{1.75} \left(\frac{\rho_v}{\rho_l} \right) \left(\frac{\mu_l}{\mu_v} \right)$, $\alpha_{i,all} (i=l, v)$ 为工质在饱和条件下的对流换热系数。

对于预热、蒸发和过热阶段, 其换热面积 $A_{hj} (j=1, 2, 3)$ 分别用如下公式计算:

$$A_{h3} = \pi d_{h,out} N_{e1} L_{e3}, \quad (11)$$

$$A_{h2} = \pi d_{h,out} N_{e1} L_{e2}, \quad (12)$$

$$A_{h1} = \pi d_{h,out} N_{e1} L_{e1}, \quad (13)$$

式中, $L_{ej} (j=1, 2, 3)$ 为 3 个阶段在换热管中的等效长度。

图1 (网络版彩图)管壳式换热器模型和T-s图

Figure 1 (Color online) Model diagram and T-s diagram of the shell-and-tube heat exchanger.

由式(3)~(5)可得换热器的总换热率为

$$Q_h = Q_{h1} + Q_{h2} + Q_{h3} = \dot{m}_{wf}(h_2 - h_1). \quad (14)$$

由式(2)~(5)可知, 管壳式换热器在3个换热阶段对应的传热焓耗散率分别为

$$\begin{aligned} \dot{E}_{vh\phi,h3} &= 0.5c_{p,wf3}\dot{m}_{wf}(T_1^2 - T_{1'}^2) \\ &\quad + 0.5c_{ph}\dot{m}_h(T_{h2}^2 - T_{h,out}^2), \end{aligned} \quad (15)$$

$$\dot{E}_{vh\phi,h2} = \dot{m}_{wf}(h_{1'} - h_{2'})T_{1'} + 0.5c_{ph}\dot{m}_h(T_{h1}^2 - T_{h2}^2), \quad (16)$$

$$\begin{aligned} \dot{E}_{vh\phi,h1} &= 0.5c_{p,wf1}\dot{m}_{wf}(T_2^2 - T_1^2) \\ &\quad + 0.5c_{ph}\dot{m}_h(T_{h,in}^2 - T_{h1}^2). \end{aligned} \quad (17)$$

当流动阻力引起的焓耗散可忽略时, 由式(15)~(17)可得总焓耗散率为

$$\begin{aligned} \dot{E}_{vh\phi} &= \dot{E}_{vh\phi,h1} + \dot{E}_{vh\phi,h2} + \dot{E}_{vh\phi,h3} \\ &= 0.5c_{p,wf3}\dot{m}_{wf}(T_1^2 - T_{1'}^2) \\ &\quad + \dot{m}_{wf}(h_{1'} - h_{2'})T_{1'} + 0.5c_{p,wf1}\dot{m}_{wf}(T_2^2 - T_1^2) \\ &\quad + 0.5c_{ph}\dot{m}_h(T_{h,in}^2 - T_{h,out}^2). \end{aligned} \quad (18)$$

工质侧各换热阶段的压降分别为^[83]

$$\begin{aligned} \Delta P_{h,wfi} &= (\Delta P_{h,Li} + \Delta P_{h,ri})F_i N_{e,wf} N_{e,water} \\ &\quad + \Delta P_{h,ni} N_{e,water} \quad (i = 1, 2, 3), \end{aligned} \quad (19)$$

式中, F_i 为常数, 由管内摩擦阻力、弯管和进出口效应引起的压降损失分别为^[83]

$$\Delta P_{h,Li} = \lambda_{h,i} \frac{L_{ei}}{d_{h,in}} \left(\frac{\rho_{h,i} u_{h,i}^2}{2} \right), \quad (20)$$

$$\Delta P_{h,ri} = 3 \left(\frac{\rho_{h,i} u_{h,i}^2}{2} \right), \quad (21)$$

$$\Delta P_{h,ni} = 1.5 \left(\frac{\rho_{h,i} u_{h,i}^2}{2} \right), \quad (22)$$

式中, $\rho_{h,i}$ 和 $u_{h,i}$ ($i=1, 2, 3$)分别为三个阶段工质的密度和流速, $\lambda_{h,i}$ 为^[83]

$$\lambda_{h,i} = 0.01227 + \frac{0.7543}{Re_{hi,wf}^{0.38}} \quad (i = 1, 2, 3). \quad (23)$$

由于蒸发阶段存在相变, 实际计算中将该阶段划分成若干等份区间, 然后根据工质物性参数的变化求解区间内的压降。

热流体侧的压降为^[83]

$$\Delta P_{h,water} = (\Delta P_{h,1} + \Delta P_{h,2})F_s N_{e,water}, \quad (24)$$

式中, F_s 为常数, 由管内摩擦阻力和折流板缺口引起的压降损失分别为^[83]

$$\Delta P_{h,1} = F_h f_o n_{h,c} (N_{h,b} + 1) \left(\frac{\rho_{h,water} u_{h,water}^2}{2} \right), \quad (25)$$

$$\Delta P_{h,2} = N_{h,b} \left(3.5 - \frac{2l_{h,b}}{D_h} \right) \left(\frac{\rho_{h,water} u_{h,water}^2}{2} \right), \quad (26)$$

式中, F_h 为常数, f_o , $n_{h,c}$, $N_{h,b}$, $l_{h,b}$, D_h , $\rho_{h,water}$ 和 $u_{h,water}$ 分别为摩擦系数、中线上换热管的数目、折流板数目和间距、公称直径、热流体密度和流速。

由式(19)~(26)可得换热器的总泵功率为

$$P_h = \frac{\dot{m}_h \Delta P_{h,water}}{\eta_p \rho_{h,water}} + \sum_{i=1}^3 \frac{\dot{m}_{wf} \Delta P_{h,wfi}}{\eta_p \rho_{h,i}}, \quad (27)$$

式中, η_p 为泵效率。

图2给出了无量纲 $\dot{E}_{vh\phi}$ 和 P_h 与换热管外径 $d_{h,out}$ 的关系, 其中 $\dot{E}_{vh\phi}$ 和 P_h 的无量纲值定义为当前值与初始值的比值, 即: $\tilde{E}_{vh\phi} = \dot{E}_{vh\phi} / \dot{E}_{vh\phi,int}$ 和 $\tilde{P}_h = P_h / P_{h,int}$. 由图可

图 2 $\tilde{E}_{vh\phi}$ 和 $\tilde{P}_h$ 与 $d_{h,out}$ 的关系
Figure 2 The relationships of $\tilde{E}_{vh\phi}$ and $\tilde{P}_h$ versus $d_{h,out}$.

知, 随着换热管外径 $d_{h,out}$ 的增大, 无量纲焓耗散率 $\tilde{E}_{vh\phi}$ 增大, 而无量纲总泵功率 $\tilde{P}_h$ 先急剧减小后略有增大. 这两个目标是传热和流动性能优劣的重要反映, 但单目标优化不能同时反映传热和流动性能. 两个目标的单位也不同, 不能简单相加. 为解决这一问题, 本文在将焓耗散率和泵功率无量纲化的基础上, 采用线性加权法^[49,50,84,85]建立同时考虑传热和流动性能的复合目标函数 F_{EP} :

$$F_{EP} = a_0 \frac{\dot{E}_{vh\phi}}{\dot{E}_{vh\phi, \text{int}}} + (1 - a_0) \frac{P_h}{P_{h, \text{int}}} \\ = a_0 \tilde{E}_{vh\phi} + (1 - a_0) \tilde{P}_h, \quad (28)$$

式中, a_0 为加权系数. 由式(3)~(14)可知, 当换热器总换热率 Q_h 和总换热面积 ($A_h = A_{h1} + A_{h2} + A_{h3}$) 一定时, 复合函数 F_{EP} 是换热管外径 $d_{h,out}$ 的函数, 通过优化 $d_{h,out}$ 可对管壳式换热器开展构形优化. 与传统优化方法相比, 构形优化方法考虑了换热面积约束, 更适用于有限换热材料的换热器优化设计. 换热器产生的蒸气将用来推动透平做功, 在给定工况条件下一般要求透平蒸气供入量一定; 当蒸气参数不变时这就要求换热器传热量一定, 因此总换热率 Q_h 给定是具有工程意义的.

3 有机工质管壳式换热器构形优化

为开展有机工质管壳式换热器构形优化, 取

R245fa 作为研究工质. 计算中取 $T_{h,in}=403$ K, $A_{h1}=4$ m², $A_h=30$ m², $T_l=323$ K, $\dot{m}_h=7.2$ kg/s, $\dot{m}_{wf}=2.3$ kg/s, $Q_h=5 \times 10^5$ W, $a_0=0.75$, $N_{el}=144$, $N_{e,wf}=4$, $N_{e,water}=1$ 和 $d_{h,out}=0.025$ m. 无特殊注明的情况下, 以上初始参数均保持假定值不变. 在初始参数条件下通过计算发现, 管壳式换热器传热焓耗散率约是流动阻力焓耗散率的 330 倍, 因此式(18)中忽略流动阻力引起的焓耗散率是可行的.

图 3 给出了不同热水温度 $T_{h,in}$ 下复合函数 F_{EP} 与换热管外径 $d_{h,out}$ 的关系. 由图可知, 当热水进口温度为 $T_{h,in}=403$ K 时, 复合函数 F_{EP} 存在最小值 ($F_{EP,m}=0.984$), 此时对应的最佳换热管外径为 $d_{h,out,opt}=0.023$ m. 管壳式换热器最优构形与初始设计结构相比, 总焓耗散率和复合函数分别降低了 9.54% 和 1.56%, 总泵功率增加了 22.40%. 这说明换热器综合性能的提高是通过较大程度地降低流体流动性能和显著减小传热不可逆性来实现的. 此外, $F_{EP,m}$ 随着 $T_{h,in}$ 的增大而减小, 因此管壳式换热器的综合性能可通过提高热水进口温度来进一步提升.

图 4 给出了工质进口温度 T_l 、总换热面积 A_h 、工质质量流率 $\dot{m}_{wf}$ 和权重系数 a_0 对最小复合函数 $F_{EP,m}$ 和最佳换热管外径 $d_{h,out,opt}$ 的影响. 由图可知, 随着 A_h 和 $\dot{m}_{wf}$ 的增大、 T_l 的减小, 最小复合函数 $F_{EP,m}$ 减小. 这是由于此时焓耗散率和总泵功率整体上均减小, 最终导致最小复合函数减小. 而随着权重系数 a_0 的增大, 最

图 3 $T_{h,in}$ 对 F_{EP} 与 $d_{h,out}$ 关系的影响
Figure 3 The influence of $T_{h,in}$ on the relationship of F_{EP} versus $d_{h,out}$.

图4 $F_{EP,m}$ 和 $d_{h,out,opt}$ 与 T_1 (a), A_h (b), m_{wf} (c), a_0 (d)的关系

Figure 4 The relationships of $F_{EP,m}$ and $d_{h,out,opt}$ versus T_1 (a), A_h (b), m_{wf} (c), a_0 (d).

小复合函数 $F_{EP,m}$ 先增大后减小。这是由于 a_0 较小时焓耗散率和总泵功率的变化幅度相当, 因此 $F_{EP,m}$ 增大; a_0 较大时虽然总泵功率比焓耗散率的增大幅度大, 但 a_0 较大使得 $F_{EP,m}$ 减小, 最终导致 $F_{EP,m}$ 先增大后减小。此外, 最佳换热管外径 $d_{h,out,opt}$ 随着 a_0 的增大而减小, T_1 , A_h 和 m_{wf} 对最佳换热管外径的影响相对较小。针对设计需求可能变化的情况, 给出不同权重系数下的优化结果, 有利于指导不同设计需求下的换热器优化设计。

图5和6给出了总换热管数 N_{et} 和热水质量流量 $\dot{m}_h$ 对最小复合函数 $F_{EP,m}$ 和最佳换热管外径 $d_{h,out,opt}$ 的影响。由图可知, 随着总换热管数 N_{et} 的增大, 最小复合函数 $F_{EP,m}$ 先减小后增大; F_{EP} 存在双重最小值($F_{EP,m}=0.981$), 对应的最佳总换热管数为 $N_{et,opt}=200$, 二次最佳换热管外径为 $d_{h,out,oo}=0.02$ m, 此时管壳式换热器的综合性能略有提高。随着热水质量流量 $\dot{m}_h$ 的增大, 最小复合函数 $F_{EP,m}$ 先减小后增大, 但二次优化结果和图3中的一

次优化结果相差很小, 其主要原因是最佳热水质量流量与初始热水质量流量比较接近。图6进一步比较了以焓耗散率和泵功率组成的复合函数(本文)以及换热率和泵功率组成的复合函数为优化目标时(文献[49])的换热器最优构形优化结果。由图可知, 基于前者复合函数目标(本文)所得的二次最小复合函数值为 $F_{EP,m}=0.984$, 对应的最佳热水质量流量为 $\dot{m}_{h,opt}=7.3$ kg/s; 而基于后者复合函数目标[49]所得的二次最小复合函数值为 $F_{QP,mm}=0.891$, 对应的最佳热水质量流量为 $\dot{m}_{h,opt}=6.9$ kg/s, 且后者二次最佳换热管外径大于前者。由此可知, 两个复合函数最小化所得的管壳式换热器最优构形是完全不同的。前者有利于降低换热器传热不可逆性和流动阻力, 是换热器传热不可逆性和流动性能的折中; 后者有利于提高换热器热量产出和降低流动阻力, 是换热器传热效能和流动性能的折中。在实际中, 可根据不同的换热器设计需要来

图 5 $F_{EP,m}$ 和 $d_{h,out,opt}$ 与 N_{et} 的关系Figure 5 The relationships of $F_{EP,m}$ and $d_{h,out,opt}$ versus N_{et} .

确定其优化设计方案。

4 结论

本文基于构形理论和焓理论, 对管壳式换热器有机工质蒸发过程进行了研究。以同时考虑换热和流动性能的复合函数为优化目标、换热管外径为优化变量, 考虑了总换热率和总换热面积约束条件, 对管壳式换热器进行了构形优化, 得到了其最优构形, 并分析了不同参数对构形优化结果的影响。得到如下结论。

(1) 复合函数 F_{EP} 存在最小值 ($F_{EP,m}=0.984$), 此时对应的最佳换热管外径为 $d_{h,out,opt}=0.023$ m。管壳式换热器最优构形与初始设计结构相比, 总焓耗散率和复合函数分别降低了 9.54% 和 1.56%, 总泵功率增加了 22.40%。这说明换热器综合性能的提高是通过较大程度地降低流体流动性能和显著减小传热不可逆性来实现的, 这一结论与文献[49]中换热率和泵功率复合函数最小化所得结论刚好相反。由于本文和文献[49]的构形优化的边界条件和优化目标均不同, 使得换热管

图 6 $F_{EP,m}$, $F_{QP,m}$ 和 $d_{h,out,opt}$ 与 $\dot{m}_h$ 的关系Figure 6 The relationships of $F_{EP,m}$, $F_{QP,m}$ and $d_{h,out,opt}$ versus $\dot{m}_h$.

径、换热率、焓耗散率和泵功率的变化趋势也不同, 最终得到了不同的结论。

(2) 随着工质进口温度 T_l 的减小、总换热面积 A_h 和工质质量流率 $\dot{m}_{wf}$ 的增大, 最小复合函数 $F_{EP,m}$ 减小。随着权重系数 a_0 的增大, 最小复合函数 $F_{EP,m}$ 先增大后减小。随着总换热管数 N_{et} 和热水质量流率 $\dot{m}_h$ 的增大, 最小复合函数 $F_{EP,m}$ 先减小后增大, 此时复合函数分别存在双重最小值。

(3) 焓耗散率和泵功率复合函数最小的换热器最优构形与换热率和泵功率复合函数最小的最优构形是完全不同的。前者有利于降低换热器传热不可逆性和流动阻力, 是换热器传热不可逆性和流动性能的折中; 后者有利于提高换热器热量产出和降低流动阻力, 是换热器传热效能和流动性能的折中。在实际中, 可根据不同的换热器设计需要来确定其优化设计方案。

本文将焓理论应用到有机工质管壳式换热器的优化设计中, 使其综合性能得到提高。下一步将建立 CFD 模型以精确地描述蒸发传热和流动过程, 从而更好地指导实际有机工质蒸发设备的优化设计。

致谢 两位审稿人对本文提出了细心、无私和充满建设性的意见, 在此致以诚挚的谢意!

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Constructal optimization for an organic fluid shell-and-tube heat exchanger based on entransy theory

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Based on constructal theory and entransy theory, constructal optimization of a shell-and-tube heat exchanger (STHE) for the organic fluid evaporation process is performed using a complex function as an optimization objective. The complex function is composed of the entransy dissipation rate (EDR) caused by heat transfer and the total pumping power. The total heat transfer rate and heat transfer area are taken as constraints in the optimization. The minimum complex function and optimal external diameter of the heat transfer tube can then be obtained. The results show that, compared to the initial design construct, the optimal construct of the STHE reduces the EDR caused by the heat transfer and complex function by 9.54% and 1.56%, respectively, and increases the total pumping power by 22.40%. This illustrates that an overall performance improvement of the STHE can be realized by substantially reducing the fluid flow performance and significantly decreasing the heat transfer irreversibility. The overall performance of the STHE can be further improved by increasing the total heat transfer area and the mass flow rate of the working fluid and decreasing the inlet temperature of the working fluid, as well as choosing an appropriate total number of heat transfer tubes and an optimal mass flow rate of the hot water. The optimal constructs of the STHE derived from the minimizations of the two complex functions, composed of the EDR and total pumping power, heat transfer rate and total pumping power, are obviously different. New guidelines for the optimal structure designs of STHEs are provided by introducing entransy theory into constructal optimizations of organic fluid STHEs.

constructal theory, entransy theory, shell-and-tube heat exchanger, entransy dissipation rate, total pumping power, generalized thermodynamic optimization

doi: [10.1360/SST-2019-0246](https://doi.org/10.1360/SST-2019-0246)