转角正方形管束降膜蒸发时的传热温差损失

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摘要 实验测量并获得饱和蒸汽水平方向流经铝黄铜管束的压降数据, 基于实验数据拟合出来的压降系数, 以转角正方形排列的管束为物理模型, 计算水平管束降膜蒸发时压降造成的传热温差的变化, 分析饱和温度、喷淋密度和管列数等对传热温差损失的影响. 在管束排列方式一定、蒸汽流量相同下, 计算结果表明, 传热温差损失随喷淋密度的增加而增大, 喷淋密度为 $0.02 \text{ kg}/(\text{m s})$ 时的传热温差损失值约是 $0.08 \text{ kg}/(\text{m s})$ 时的一半; 饱和温度越低, 传热温差损失越大, 50°C 条件下的传热温差损失值约是 70°C 时的3.5倍. 计算还表明, 当管束为转角正方形排列时, 传热温差损失随管列数的增加呈抛物线型升高.

关键词 降膜蒸发, 温差损失, 管束排列

因在低温、小喷淋密度、小温差工况下具有高传热系数的优点^[1,2], 低温多效降膜蒸发技术^[3]在石油化工、食品冷冻等领域有着广泛的应用; 尤其在海水淡化领域, 被称为最有前景的热法技术之一^[4,5]. 关于降膜蒸发的流动形态、传热特性等研究领域, 学者们开展了深入研究^[6,7]. 张颖等人^[8]分析了布液方式对水平管管间降膜流型的影响; 何茂刚等人^[9]运用激光诱导荧光法测量了水平光管的管外液膜厚度. 沈胜强等人^[10]给出了水平管降膜蒸发器内的总换热系数分布, 他们考虑了喷淋密度和管内冷凝的影响. 但在流动特性尤其是压降这一方面, 学者们还需要开展更加深入的研究; 比如, 降膜蒸发的流动阻力对传热影响的细致研究还很罕见. 尽管对于传统两相流动, Ishihara等人^[11]提出了两相流压降公式, 其他学者在此基础上添加新的影响因子如管间距^[12]、体积分数^[13]等对其进行丰富; 但降膜蒸发条件下, 管

束间气体和液体是呈近似垂直流动的, 其与同向流动的两相流有着极大的不同. 本实验室设计建造了水平降膜管束压降测试实验台, 研究大型降膜蒸发管束中的流动阻力, 在实验数据基础上给出了压降预测关系式^[14]. 在低温多效蒸发装置内, 每一效蒸发器的表观传热温差很小, 加之低温降膜蒸发多是在低于 70°C 的饱和条件下进行, 因而较小的温度变化都会对传热造成较大的影响. 流动阻力会造成压力的降低, 从而引起管内外传热温差降低, 造成温差损失. 流动阻力造成的温差损失已经引起了相关学者的重视, 研究方法多为理论分析^[15].

为深入分析传热温差对降膜蒸发传热效率的影响, 本文在实验研究的基础上^[14], 计算蒸汽流过管束的压降所造成的传热温差损失. 期望研究结果对降膜蒸发器的优化设计及传热效率的提高具有指导意义.

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1 计算模型

1.1 物理模型

图1(a)为测试台核心的简化模型. 铝黄铜管, 管外径 D 为25.4 mm, 每根管长500 mm, 管束的相对管间距为 $1.3D$, 管束为转角正方形排列. 实验流体为水和水蒸气, 水喷淋到管束最上排, 在重力作用下沿管子外表面形成液膜并作降膜流动; 水蒸气横向冲刷有降膜流动的管束, 模拟降膜蒸发器内蒸汽和液体两相分离流动的过程; 测量蒸汽横掠有降膜流动管束的压降, 并根据实验数据给出了压降系数的关系式. 图1(b)为计算模型, 采用一维模简化计算, 取沿管列数变化的方向, 计算蒸汽横向流过降膜管束时的压力变化及压降造成的温差损失.

1.2 数学模型

采用叠加的方法计算转角正方形排列管束的管外二次蒸汽的压降及对应的温差损失^[16]. 蒸汽流过单列管的压降计算式为^[17]

$$\Delta p_i = 0.5f \cdot G_i^2 / \rho, \quad (1)$$

式中, f 是由实验数据拟合得出的压降系数^[14]:

$$f = 6.8Re_g^{-0.77} Re_r^{0.69} + 1.4Re_g^{-0.15}, \quad (2)$$

式中, Re_g 和 Re_r 分别为蒸汽雷诺数和降膜雷诺数

$$Re_g = \frac{G_i \cdot D}{\eta}, \quad (3)$$

图1 (网络版彩色)实验和计算模型

Figure 1 (Color online) Experimental and physical model

$$Re_r = \frac{4\Gamma}{\mu}. \quad (4)$$

假定每根管单位管长蒸汽产率是相同值, 且作为一个变量 m 添加到计算中.

$$G_i = i \cdot m / A. \quad (5)$$

计算沿管列方向的第 i 列管处的压力值采用式(6):

$$p_i = p_{i-1} + \Delta p_i. \quad (6)$$

$$T_{\text{sat}} = F(p), \quad (7)$$

式(7)参考“水和水蒸气热力性质公式IAPWS-IF97”, 计算时, 相关物性参数直接引用至该公式. 根据第 i 列管处的压力值 p_i , 可以求得对应的饱和温度 T_i , 根据每列对应的饱和温度进而求得沿管列变化方向的温差损失 ΔT_i .

$$\Delta T_i = T_i - T_1. \quad (8)$$

1.3 计算说明及验证

计算流程图见图2, 计算时输入管列数、单位管长蒸汽量、边界饱和温度和喷淋密度作为初始值和边界值, 先根据公式(1)~(6)计算一列管造成的压降, 再根据公式(7), (8)计算一列管流动阻力造成的温差损失; 最后通过迭代累加的计算方法得出总列数(初始值给出的管列数)管束的流动阻力造成的温差损失.

说明如下: 计算中涉及的水和水蒸气的密度、黏

图2 计算流程图

Figure 2 The flowchart of calculation

度等物性等参数均调用水和水蒸气热力性质公式 IAPWS-IF97 国际标准。

对管列数 N 为 32 的转角正方形排列水平管束进行模拟计算, 并与实际测量值进行比较, 验证计算的可信度。图 3 表明, 在设定与实验相同的工况和入口参数条件下, 计算值与实验值误差小于 5%, 验证了计算程序可信。

2 结果与分析

2.1 喷淋密度对传热温差损失影响分析

图 4 为管束 32 排、饱和温度 60°C 时不同喷淋密度下的传热温差损失。由图 4 可知, 传热温差损失随喷淋密度的增加而增大, 喷淋密度为 $0.02\text{ kg}/(\text{m s})$ 时的传热温差损失值约是 $0.08\text{ kg}/(\text{m s})$ 时的一半。这是因为降膜蒸发时, 液体在管外形成液膜, 相当于增加了管径, 减小了管间距离。随着喷淋密度的增加, 管间液体的流动形态会由滴状流逐渐变成柱状流^[18], 蒸汽的流通面积进一步减小; 此外, 液滴及液柱的存在会额外增加蒸汽和液体的接触面积, 也导致了阻力的增加。总之, 随着喷淋密度的增加, 压降逐渐变大, 较大的流动阻力导致传热温差损失的增大。

图 4 中不同曲线代表了不同蒸汽流量时的传热温差损失和喷淋密度的关系。比较不同曲线可知, 热负荷对流动阻力的影响非常明显, 每根管子蒸汽流量越大传热温差损失越大。

2.2 饱和温度对温差损失影响分析

由图 5 可知, 随着饱和温度的升高, 温差损失降低。在饱和温度为 50°C 时, 同样蒸发量下, 由压降造成的温差损失约为 70°C 下的 3.5 倍。由饱和蒸汽性质可知, 饱和温度越低, 单位压力变化造成的温降越大; 在蒸汽质量流量确定的条件下, 温度越低, 对应的蒸汽流速越大, 压降也越大, 相应的传热温差损失越大。仅从饱和温度的因素考虑, 在温度较高的工况可以减小传热温差损失。低温多效蒸发设计的运行温度多在 $70\sim 40^\circ\text{C}$ 范围内, 在传热温差确定的条件下, 为更大效率地使用热源, 应考虑高温段和低温段传热温差损失的差异, 在蒸发量分布、蒸发器传热面积分布等方面合理优化最大效率的使用表现传热温差。图 5 中不同曲线代表着不同的喷淋密度; 喷淋密度越大, 温差损失越大。蒸汽质量一定时, 压降造成

图 3 (网络版彩色) 计算值与实验结果的对比

Figure 3 (Color online) Comparison of calculated value and experimental results

图 4 (网络版彩色) 传热温差损失受喷淋密度的影响

Figure 4 (Color online) The influence of spray density on temperature drop

图 5 (网络版彩色) 传热温差损失受饱和温度的影响

Figure 5 (Color online) The influence of saturated temperature on temperature difference drop

的温差损失随着饱和温度的升高而减小,随着喷淋密度的增加而升高。

2.3 管列数对传热温差影响分析

传热温差损失与管列数的关系如图6所示:传热温差损失随管列数增加呈抛物线型升高,即对传热温差损失的影响是“后增加的”管列数比其前一列管束的影响更加明显。随着管列数的增加,总的蒸汽量呈线性加大,压降升高,流动阻力造成的传热温差损失也随之升高。

由本文计算可知,水平管降膜蒸发器内压降造成的传热温差损失相当可观。传热温差损失使蒸发器管内外实际传热温差降低,影响传热效果。因此,控制各效蒸发器的传热温差损失,有利于提高传热温差利用率。

在小温差传热的条件下,即使不大的传热温差损失都会造成非常明显的影响。确定传热温差损失数值的条件下,可计算其对应的最大管列数 $N_{\max}$ 。本文在转角正方形排列结构下,选定传热温差损失为 0.3°C 时计算的最大管列数值。图7(a)所示为最大管列数 $N_{\max}$ 在不同饱和温度下与蒸汽产率之间的关系,图7(b)所示为最大管列数在不同饱和温度下与喷淋密度的关系曲线。由数据可知,最大管列数随喷淋密度和蒸汽蒸发量的增加而减小,但随着饱和温度的增加而升高。由前所述,可以理解最大管列数喷淋密度、蒸发温度和蒸发量等参数的关系。

低温多效蒸发装置的每一效蒸发器中传热温差很小,较小的温差损失对传热过程的影响很大,若某

图6 (网络版彩色)管列数对传热温差损失的影响

Figure 6 (Color online) The influence of tube column number on temperature drop

图7 (网络版彩色)最大管列数与操作参数的关系。(a) 蒸汽产率与饱和温度的影响;(b) 喷淋密度与饱和温度的影响

Figure 7 (Color online) The relationship of $N_{\max}$ and operation parameters. (a) The effect of steam production rate and saturated temperature; (b) the effect of the spray density and saturated temperature

一效的蒸发器温差损失过大,将大大降低其蒸发量,还将影响系统的整体性能。因此能够对流动阻力引起的传热温差损失给予准确的认识极其重要。

3 结论

计算了转角正方形排列管束降膜蒸发时,流动阻力造成的温度变化;分析了饱和温度、喷淋密度及管列数等因素对传热温差损失影响,可得如下结论:

- (1) 降膜蒸发时传热温差损失随饱和温度的增加而减小。相同蒸汽质量流量条件下,蒸汽在 50°C 的传热温差损失比 70°C 的传热温差损失约增加了2.5倍。
- (2) 降膜蒸发时传热温差损失随喷淋密度的增加而增大。相同蒸汽质量流量条件下,喷淋密度 $0.08\text{ kg}/(\text{m s})$ 时的传热温差损失约是 $0.02\text{ kg}/(\text{m s})$ 时的2倍。
- (3) 降膜蒸发时传热温差损失随管列数的增加呈抛物线型升高。

符号说明:

A	为计算流通面积, m^2 ;	V	为流速, m/s ;
f	为压降系数;	Δp	为压降, Pa ;
G	为质量流率, $\text{kg}/(\text{m}^2 \text{ s})$	ΔT	为温差损失, $^\circ\text{C}$
m	为质量流量, kg/s	ρ	为蒸汽密度, kg/m^3 ;
N	为管列数;	Γ	为喷淋密度, $\text{kg}/(\text{m s})$;
$N_{\max}$	为最大管列数;	η	为饱和蒸汽动力黏度, Pa s ;
p	压强, Pa ;	μ	为饱和水动力黏度, Pa s ;
T	温度, $^\circ\text{C}$;	下标	
		i	整数

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Summary for “转角正方形管束降膜蒸发时的传热温差损失”

Heat transfer temperature difference drop on falling film evaporation of horizontal tube bundle in rotated square-arranged

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Low-temperature multi-effect distillation that employs horizontal tube falling film evaporation technology is one of the most promising desalination methods with advantage of high heat transfer coefficient at low temperature, low water-spray density and small heat transfer temperature difference. Falling film evaporators have been widely used in absorb-type chillers, desalination devices and other areas. In a horizontal tube falling film evaporator, water and steam were flowed in different directions. The vertical falling film flow was driven by gravity, whereas the steam flow across the tube bundle was driven by pressure difference. Hence, the pressure drop and the consequent temperature drop of the steam are not only caused by the tube bundle but also by the falling water. In this paper, the effects of parameters such as spray density were determined. Experimental measurement of pressure drop data during the saturated steam flowing across aluminum brass tube bundles was carried out. The pressure drop coefficient was obtained from the experimental data. The rotated square-arranged tube bundle was chosen as a physical model to calculate the steam temperature difference drop on falling film evaporation. Based on the experimental data, the fitting pressure drop coefficient was employed in this calculation. The numerical codes were developed to simulate the steady-state performance of temperature difference drop in horizontal tube falling film evaporation. The iterative procedure was required based on the initial values with a close chain of equations and parameters. The solution was obtained through a step-by-step method. This quantitative temperature difference drops caused by the pressure drop can be calculated by the solution. The influences of spray density, saturated temperature, and tube column number on vapor temperature difference drop were analyzed respectively. The temperature difference drop caused by the pressure drop will decrease the temperature difference. The effect of temperature difference drop can be analyzed in two aspects. On the one hand, the temperature difference drop directly reduces the difference in heat transfer temperature, and thus, decreases heat transfer rate; on the other hand, the temperature difference drop generally increases the heat transfer coefficient under a certain heat transfer flux. When temperature difference drop is considered in a calculation, the heat transfer coefficient can be derived more accurately. Moreover, an accurate calculation of the temperature difference drop can provide a more detailed analysis on the influence factors of the heat transfer coefficient. At a certain tube arrangement and the same steam flow flux, the results showed that the temperature difference drop increases with the increase of spray density. It can be showed that the temperature difference drop at 50°C is 3.5 times of that at 70°C. However, the temperature difference drop decreases with the increase of saturated temperature. The temperature difference drop at spray density of 0.08 kg/(m s) is almost double of that at spray density of 0.02 kg/(m s). The calculations also show that the temperature difference drop is in parabolic rise with the increase of tube column number. When the temperature difference drop is set at lower than 0.3°C, the maximum tube column ($N_{\max}$) can be obtained. The results showed that the $N_{\max}$ increases with increasing saturated temperature, but decreases with increasing spray density. The $N_{\max}$ can be employed to determine the heat transfer area for optimization design of the heat exchanger.

falling film evaporation, temperature difference drop, tube bundle layout

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