



膜式热湿调控原理与技术进展

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摘要 基于膜的热湿传递过程作为一种新型的温度、湿度处理技术, 在建筑环境节能高效控制领域取得了积极进展. 膜技术具有高效紧凑的特点, 并且它可以选择性地只允许水蒸气通过膜表面, 从而避免了液体除湿过程中溶液小液滴对新风的污染. 近年来, 该技术从热湿传递原理到实际应用都取得了新进展. 本文介绍了目前应用的选择性透湿膜材料, 分析了平板膜全热换热器、板翅式膜全热换热器、交叉三角形波纹板全热换热器和中空纤维膜组件等膜设备的传热传质过程. 它们的传热传质分析, 同时考虑了膜两侧热湿耦合的自然边界条件、流体在组件内流动的不均匀性、管束随机分布等实际运行因素对传热传质的影响. 这些研究对膜组件的设计和膜系统的优化提供了理论基础. 此外, 还介绍了各种新型膜式除湿系统, 当它们与热泵或太阳能等系统联合应用时, 可以扩大系统的热湿负荷适应范围, 增加系统的能量利用效率. 今后, 随着新型膜材料、内冷型膜组件、多级除湿系统以及瞬态动态参数模拟技术的出现, 膜式热湿传递技术将会在实际工程中发挥更大作用.

关键词

膜式热湿调控
膜材料
膜组件
除湿系统
传热传质

人们一生有80%左右的时间在室内度过, 随着人们生活水平的提高, 室内空气品质得到了越来越广泛的重视^[1-3]. 空调作为热湿环境调控的主要设备, 已经成为人们的生活必需品. 据统计, 目前我国空调能耗已经占到能源总消耗量的15%左右^[4]. 特别是在我国南方高温高湿气候地区, 除湿的能耗占20%~40%^[5]. 建筑环境热湿节能高效调控措施正受到重视, 其中新型的空气除湿技术是关键.

独立除湿技术代表了空调发展趋势. 目前的空气除湿技术主要包括冷却盘管除湿、盐溶液除湿、固体吸附床除湿、转轮除湿和电化学除湿等^[5,6]. 在这些除湿方式中, 液体除湿适合用作独立除湿技术. 它没有冷凝液滴, 避免了细菌滋生对室内环境的破坏. 同时可以利用太阳能等低品位能进行溶液再生, 能源综合利用效率高. 除湿溶液还可以作为一种能量储存介质适应不同条件下湿负荷的要求, 达到为电能

削峰填谷的作用^[5,6]. 因此液体除湿是一种节能环保的新型独立除湿技术.

在传统的填料塔内进行的盐溶液除湿过程, 溶液与湿空气直接接触, 气流不可避免夹带盐溶液中的小液滴, 从而会污染室内空气品质. 为了克服这种缺陷, 人们提出利用膜式组件来代替填料塔实现液体空气除湿^[7]. 它可以选择性地只允许水蒸气透过而对溶液有拦截作用. 因此可以有效防止气液夹带现象. 另一方面, 组件具有较高的填装密度, 确保较大的气液传质有效面积, 壳层空气流动阻力也得到降低.

膜法空气除湿作为一种新技术, 近年来, 很多学者在新型膜材料的制备、膜组件的结构、热湿调控原理和新型除湿系统的建立等方面做了大量的研究工作并获得了许多进展. 本文将对上述工作进行介绍和总结, 并指出该技术未来的发展方向.

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1 新型膜材料

膜材料作为水蒸气选择性渗透的核心部分,在膜式传热传质过程中起重要作用.它们通常采用聚合物材料^[8,9],主要有全氟磺酸、再生纤维素、三乙酸纤维素、磺化聚酯纤维、聚苯醚、聚醚聚氨酯、硅氧烷酰胺共聚物、聚磺苯乙烯、聚偏二氟乙烯、聚醚砜和玻璃纸.它们一般为均匀致密膜,主要利用水蒸气在膜中的传递速率和溶解度的不同进行分离.其主要受扩散率的影响,但是它们的水蒸气扩散系数都非常小,约 $10^{-12} \sim 10^{-13} \text{ m}^2/\text{s}$,因此限制了它们的应用.后来,Zhang等人提出亲水/憎水复合膜^[9,10]、复合支撑液膜^[11-14]和指状高孔隙膜等新型膜^[15]来改进水蒸气扩散系数.其中亲水/憎水复合膜弥补了传统均质膜水蒸气传递效果差的缺点.如图1所示,很薄的亲水表皮层附着在憎水的多孔支撑层上,表层为活性膜,复合膜的分离性能主要由表皮的孔径和性质决定,同时也受微孔支撑层的结构、孔径、孔分布和孔隙率的影响^[16].皮层可以吸收大量的水蒸气,并通过选择性吸收使得只有水蒸气透过.多孔层给复合膜提供一定的机械强度,所以皮层可以很薄.由于总体传质阻力的降低,传质量可以大幅提高.当选择层附着了不同的组分后还可以适用于不同组分的分离.如硅酮涂层可应用于挥发性有机物与硅油的分离^[17],聚四氟乙烯涂层可用于 CO_2 , SO_2 , H_2S , NH_3 与乙二醇或胺类物质的分离^[18].这些新型膜材料已经应用于空气的热湿交换,今后将研制透湿性更好而传质阻力更小的膜.

2 新型膜组件

膜组件结构紧凑,根据不同的流道形式分为平板膜组件、板翅式膜组件、交叉波纹板和中空纤维膜组件等^[5].很多学者对其中的热质交换过程进行了研究.

2.1 平板膜组件

如图2所示的平行平板膜组件,它是最常见的膜

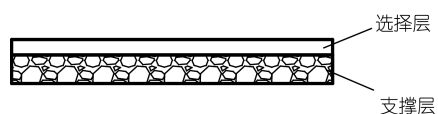


图1 亲水/憎水复合膜结构示意图

Figure 1 Composite hydrophobic-hydrophilic membrane structure

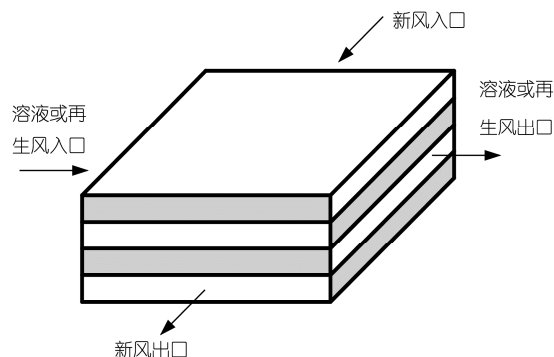


图2 平行平板膜组件示意图

Figure 2 Structure of a parallel-plate membrane module

式组件.它由平板膜叠加组成,相邻的平板膜组成流动流道.新风与再生风或除湿溶液以错流的形式流经组件完成热湿交换.它首先应用于空气-空气的热湿换热器上,Zhang和Jiang^[19]讨论了单层膜的热湿传递机理.Zhang等人^[20-22]进一步研究了膜材料的性质对膜式全热交换器性能的影响.他们推导出了显热效率、潜热效率和全热效率的关联式,并研究了全热交换器的节能潜力.其研究发现,实际的平板膜壁面既不是简单的等温度也不是等热流边界条件,而是膜两侧流体在膜两侧表面自然形成的边界条件.Zhang^[23]提出的共轭边界条件如下:

$$q = -\lambda_a \frac{\partial T}{\partial Z} \Big|_{\text{surface}} = -\lambda_m \frac{\partial T}{\partial Z} \Big|_{\text{surface}}, \quad (1)$$

$$J = -\rho_a D_{va} \frac{\partial \omega}{\partial Z} \Big|_{\text{surface}} = -\rho_m D_{vm} \frac{\partial \omega}{\partial Z} \Big|_{\text{surface}}, \quad (2)$$

式中, q , λ , T , J , ρ , ω , D 和 Z 分别为热量、导热系数、温度、传质量、密度、含湿量、质扩散率和膜厚方向.下标a, m, va和vm分别代表空气、膜、空气侧水蒸气和膜侧水蒸气.在此共轭边界条件下,膜两侧的热质交换实现耦合求解.

表1列出了平板膜在不同的膜/流体导热比和流

表1 平板膜在不同的膜/流体导热系数比和流动方式下充分发展段的努塞尔特数和舍伍德数^{[23]a)}

Table 1 Fully developed Nusselt and Sherwood numbers in the parallel-plate membrane channels for various aspect ratios and types of contactor^[23]

τ	顺流		逆流		错流	
	Nu	Sh	Nu	Sh	Nu	Sh
0.1	8.01	7.03	8.17	7.22	8.12	7.13
1	8.04	7.05	8.19	7.51	8.14	7.23
10	8.06	7.06	8.21	7.33	8.16	7.26

a) τ 为膜与流体导热系数比

动方式下充分发展段努谢尔特数(Nu)和舍伍德数(Sh), 它们比等温边界条件的 $Nu_T=7.54$ 和等热流边界条件的 $Nu_H=8.23$ 都大. 因为一般操作过程的Peclet数都大于50, Zhang等人^[24]把流动充分发展的假设扩展到流动全过程, 可以更加真实地反映组件内的传热传质过程.

由于平板膜组件结构简单、组装方便, 它同时也被用在盐溶液除湿领域. Mahmud等人^[25]提出了一种膜式液体除湿-再生系统, 除湿与再生组件都由微孔膜组成. Vail等人^[26]建立了逆流膜式液体除湿-再生系统的稳态传热模型, Seyed-Ahmadi等人^[27,28]建立了该系统的耦合传热传质模型, 并在不同的工况下进行了数值计算和实验验证. Huang等人^[29]研究了共轭边界条件下错流平板膜的耦合传热传质过程, 发现在充分发展段空气侧的 Nu 在 Nu_T 和 Nu_H 之间, 由于组分边界层在出口仍没有充分发展, 因此溶液侧 Nu 比空气侧 Nu 大15%.

2.2 板翅式膜组件

虽然平板膜组件结构简单, 但是它的传热传质能力有限. 为了提高空气侧的传热传质系数, 有人在平板流道中加入翅片形成板翅式膜组件^[30,31], 如图3所示. 相比平板膜, 它的结构更紧凑、机械强度高、传热传质效率高.

Zhang等人^[30,31]建立了板翅式膜组件的传热方程, 针对物理模型中非规则的边界条件造成数值求解困难的问题, 通过贴体坐标转化为规则边界条件进行求解. 为了将板翅式膜组件用于热湿传递领域, Zhang^[32-35]把翅片材料从传统的金属材料扩展到非金属高分子材料. 发现全金属板翅式换热器中的平板和翅片都极大地促进了传热, 但用于膜组件时, 高分子材料做翅片, 翅片虽然也强化了传热, 但是对传

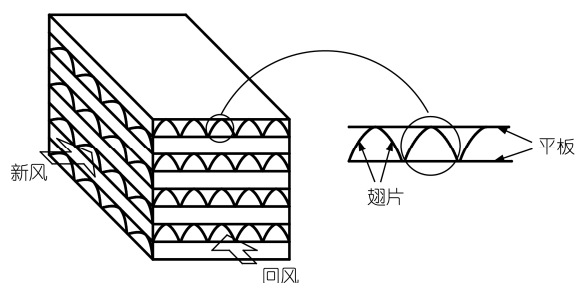


图3 板翅式膜组件示意图

Figure 3 Schematic of a plate-fin structure

质起主要起作用的还是平板膜. 由于翅片的导热系数不高, 所以翅片效率和 Nu 都偏低^[33,34]. 为了优化组件并提高其传热传质性能, Zhang^[36]提出了一种适合实际应用的准逆流组件, 计算了传热传质效率, 其数值相比较错流方式有较大的提高.

2.3 交叉波纹板流道膜组件

为了进一步增强膜组件的传热传质能力, Zhang等人^[37,38]提出交叉三角形波纹板膜组件用于新风与回风的换热. 如图4所示, 平板膜被压制成为等边三角形流道, 每2个相邻的流道按照90°交叉排列, 风道入口处如图中阴影部分进行填充. 由于流体在非均匀流道内的转向、扩张和收缩, 组件的传热传质能力大大提高. Zhang和Chen^[37]计算了在等热流边界条件下流体的传热特性, 并给出了压降和 Nu 的计算关联式. Zhang^[38]对交叉波纹板膜内三维湍流流动进行了计算, 得出在增加了2倍流动阻力的情况下, 流体的传热传质能力可以加强40%~60%. Zhang和Che^[39]通过对交叉三角形波纹板流道内三维湍流流动和传热的模拟, 比较了不同湍流模型的优缺点, 指出低雷诺数 $k-\epsilon$ (LBKE)模型和剪切压力传输(SST)模型和实验吻合得最好. Zhang^[40,41]利用低雷诺数模型分别在等热流(质流)和等温度(浓度)边界条件下研究了流动和传热传质充分发展段的阻力因子、平均 Nu 以及流动和传质充分发展段的阻力因子、平均 Sh , 得出随着雷诺数(Re)的增加, 湍流中心会加强并从波纹板的上部向下部移动, 从而增强传热传质能力. Chen等人^[42]利用场协同理论分析了不同角度下的波纹板顶角对流动

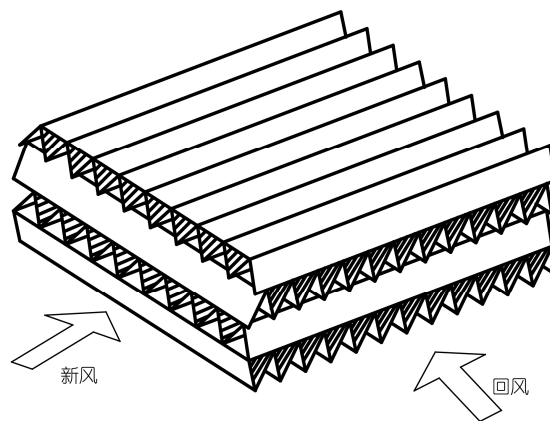


图4 交叉三角形波纹板膜组件示意图

Figure 4 Schematic of the cross-corrugated triangular ducts

和换热的影响, 得出 90° 下阻力系数和 Nu 最大. Zhong等人^[43]通过实际制备不同型式膜组件并进行测试, 研究了不同结构和材料的膜组件性能, 得出交叉三角形波纹板相比平板膜组件和板翅式膜组件能有效地增强传热传质能力, 一步法制得的醋酸纤维素膜传热传质阻力最低.

2.4 中空纤维膜组件

由于装填密度高, 中空纤维膜组件近年来得到了越来越广泛的应用. 如图5所示, 组件由一束中空纤维膜填装而成, 结构如常见的管壳式换热器^[44]. 除湿溶液走管内(管程), 空气走壳体(壳程), 两股流体逆流通过膜进行热湿交换. 中纤维膜外表面采用亲水聚乙烯醇(PVA), 内表面采用憎水聚偏氟乙烯(PVDF)膜复合制成, 膜表面微孔直径约 $0.1\ \mu\text{m}$, 膜直径 $1\sim 3\ \text{mm}$, 因此可以得到高达 $100\ \text{m}^2/\text{m}^3$ 的装填密度^[45].

Bergero和Chiari^[46]利用中空纤维膜对空气进行除湿和加湿研究, 分析了空气侧和溶液侧的热量和水蒸气传递机理, 得出了空气流量和溶液流量对组件性能的影响. Kneifel等人^[47]利用实验研究了错流膜组件内的空气压降和水蒸气的传输特性, 证实了膜支撑层和涂层对渗透速率有决定性的影响. 由于传热传质问题对膜组件设计和除湿系统的优化有重要作用, 许多学者对组件的传热传质特性进行了研究.

(i) 逆流中空纤维膜组件. 由于纤维管的直径很细, 管内流体的雷诺数通常小于2300, 认为是层流流动^[48,49]. Graetz^[50]研究了不同流动条件下, 管内

流动的平均舍伍德数. 由于众多纤维管管径及排列分布不均匀, 壳程的流动要比管程复杂. Happel^[51]认为纤维管束由一系列的自由表面单元组成. 如图5(b), 每个自由表面包括一个纤维管和包围它的假想流体, 纤维管的管心和自由表面的中心重合, 在自由表面的外部没有动量、热量和质量的传递, 该模型已经得到广泛的应用^[52~56]. 同平板膜组件类似, 实际的中空纤维膜边界条件也是两侧流体在膜表面自然形成. Zhang等人^[57]建立了基于自由表面假设的膜表面共轭传热传质模型, 计算得到了局部和平均努塞尔特数和舍伍德数并用实验进行了验证. 研究发现, 溶液侧平均努塞尔特数($Nu_{C,s}$)接近于等 Nu_T . 充分发展段空气侧的努塞尔特数和舍伍德数列于表2^[58]. 从中可以看出充分发展段空气侧的努塞尔特数($Nu_{C,a}$)在 Nu_T 和 Nu_H 之间, 并且 $Nu_{C,a}$ 更接近 Nu_H , 舍伍德数($Sh_{C,a}$)略小于相应的努塞尔特数($Nu_{C,a}$).

该项研究得到后续跟进, Huang等人^[59,60]利用椭圆形中空纤维膜管研究了逆流组件的流动和传热特性, 并得到了阻力因子、努塞尔特数和舍伍德数, 研究发现, 传热和传质能力都有不同程度的下降, 在实际应用中应避免使用.

(ii) 错流中空纤维膜组件. 当流体流速较大时, 逆流中空纤维膜组件中的压力损耗增大. 为了避免膜组件中过大的压降, 通常使得空气横掠纤维管束, 形成错流中空纤维膜组件. 相比较逆流换热器, 错流中空纤维膜组件内壳侧流体的流动和传热传质比较复杂. Ahmad^[61]研究了普朗特数(Pr)为0.7下空气横掠单根长圆柱体的对流换热. Lange等人^[62]对此问题给出了更加精确合理的结果, 但是单根纤维管模型不能反映纤维管束管与管之间的相互影响. 为了解决这一问题, 目前主要采用自由表面或周期性边界条件模型.

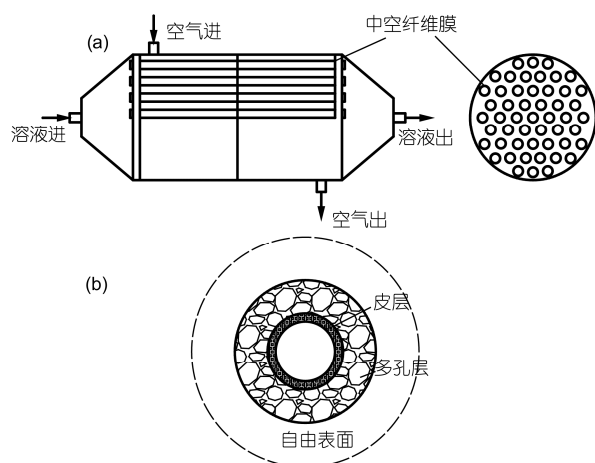


图5 中空纤维膜组件示意图. (a) 管壳式组件结构; (b) 单根膜结构
Figure 5 Schematic of the hollow fiber membrane module. (a) The shell and tube structure; (b) structure of a single membrane fiber

表2 自由表面环形流道内充分发展的努塞尔特数和舍伍德数^{[58]a)}

Table 2 Fully developed Nusselt and Sherwood numbers for the annular channel in the counter flow hollow fiber membrane module with the free surface model^[58]

填充率	r_o/r_i	Nu_T	Nu_H	$Nu_{C,a}$	$Sh_{C,a}$
0.162	0.40	5.51	5.97	5.94	5.84
0.253	0.50	4.99	5.43	5.40	5.34
0.360	0.60	4.66	5.10	5.07	4.98
0.491	0.70	4.41	4.81	4.80	4.75
0.641	0.80	4.24	4.63	4.60	4.55

a) r_o 与 r_i 分别为管内径与自由表面模型

(1) 自由表面模型. 如上节所述, 自由表面模型是模拟除湿组件管与管之间相互影响的一种简单有效的方法. 自由表面单元的填充率等于整个管束的填充率^[51]. Dhotkar等人^[63]和Mandhani等人^[64]利用该模型得出了流体横掠圆管自由表面单元中的压降阻力系数、摩擦阻力系数和努塞尔特数. 错流换热器的努塞尔特数的关联式大多在等壁温或等热流边界下得到^[64,65]. Zhang等人^[66]研究了在膜表面共轭传热传质边界条件下层流($50 < Re < 300$)流体的流动和热质传递特性. Huang等人^[67]发现文献[66]中的结果在 $Re > 300$ 情况下不适用, 并利用低雷诺数的 $k-\varepsilon$ 模型计算了湍流流动及传热传质规律. 溶液侧的 $Nu_{C,s}$ 接近于等热流 $Nu_H(=4.36)$, 大于等温度 $Nu_T(=3.66)$. 空气侧的平均努塞尔特数和舍伍德数见表3. 从中可以看出, 当管束填充率比较低时, 空气侧的 $Nu_{ave,a}$ 数值在 Nu_T 与 Nu_H 之间, 当填充率较大时, $Nu_{ave,a}$ 大于 Nu_H . 当空气为层流时, $Sh_{ave,a}$ 大于相应的 $Nu_{ave,a}$; 而空气为湍流时, 两者基本相等.

(2) 周期性单元模型. 自由表面模型没有考虑纤维之间的相互影响, 而周期性边界模型可以很好地描述管束间的分布情况. 该模型在纤维管束间选择一个典型的周期性流道, 在该流道内求解动量和能量方程. Wung和Chen^[68,69]在普朗特数为0.1, 1和10, 雷诺数为40, 120, 400和800的情况下研究了流体横掠三角形和四边形排列管束的流动与传热. Martin等人^[70]在填充率小于0.2, 雷诺数在3~160条件下, 对错流排列管束内的流动与传热进行了数值模拟, 得出努塞尔特数在等热流密度边界条件时大于等壁温条件. Wilson和Bassiouny^[71]在空气流动雷诺数为50000时, 对湍流下错流排列管束内的流动进行了数值模拟, 计算了等壁温条件下的努塞尔特数. Yoo等人^[72]研究了不同的管间距、位置和雷诺数的局部传热系

数, 并得到了平均传热准则数的关联式. Buyruk^[73]得到了在雷诺数小于400时, 管束不同布置条件下的传热特性.

相比自由表面模型, 周期性单元模型可以更加真实地表达相邻管束对流动的影响. Zhang等人^[74,75]利用周期性单元模型分析了层流和湍流条件下的耦合流动和传热传质特性. 所选取的单元模型如图6所示, 得到的流动与传热传质系数见表4. 表4中的数据与表3中采用自由表面得到的计算数据相比较, 可看出自由表面模型得到的数据仅适合于填充率小于0.2的条件.

3 膜式热湿调控过程的关键科学问题

3.1 流动的分布不均匀性

风管和进出口等部件也是膜式热湿交换系统的必要组成部分. 如图7所示, 空气受入口风道尺寸变化和进入膜组件角度等因素的影响, 两股流体出现不均匀流动. Zhang^[76~78]模拟了不均匀流动对平行板膜组件和板翅式膜组件的传热传质影响, 指出当板间距大于2 mm时, 组件的显热和潜热效率会分别下降10%~20%. Li和Zhang^[79]和Zhang等人^[80]分析了流动不均匀分布对逆流和错流中空纤维膜传热传质性

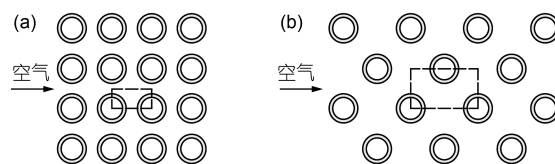


图6 错流中空纤维膜组件示意图. (a) 四边形排列; (b) 三角形排列 (周期性单元模型为图中虚线所围面积)

Figure 6 Schematic of a cross flow hollow fiber membrane bundle. (a) In-line; (b) staggered (the area surrounded by the dash line is the representative calculation domain)

表3 空气横掠单根纤维管的平均努塞尔特数和舍伍德数^[66,67]

Table 3 Mean Nusselt and Sherwood numbers for the air stream across a fiber^[66,67]

填充率	r_o/r_i	$Re_a=100$				$Re_a=200$			
		$Nu_{ave,T}$	$Nu_{ave,H}$	$Nu_{ave,a}$	$Sh_{ave,a}$	$Nu_{ave,T}$	$Nu_{ave,H}$	$Nu_{ave,a}$	$Sh_{ave,a}$
0.160	0.4	6.23	7.18	7.11	7.36	8.57	9.88	9.61	9.87
0.362	0.6	7.79	8.99	9.26	10.00	10.45	12.12	12.39	12.71
0.642	0.8	11.93	13.57	14.21	17.70	15.48	17.92	19.20	21.27
填充率	r_o/r_i	$Re_a=400$				$Re_a=600$			
		$Nu_{ave,T}$	$Nu_{ave,H}$	$Nu_{ave,a}$	$Sh_{ave,a}$	$Nu_{ave,T}$	$Nu_{ave,H}$	$Nu_{ave,a}$	$Sh_{ave,a}$
0.160	0.4	12.53	14.31	13.79	13.83	15.52	17.64	17.05	16.94
0.362	0.6	14.74	17.07	17.04	17.10	18.07	20.89	20.79	20.65
0.642	0.8	21.06	24.41	26.61	26.24	25.25	29.37	31.47	30.74

表4 四边形和三角形排列下空气横掠纤维管束的平均努塞尔数和舍伍德数^[74,75]Table 4 Mean Nusselt and Sherwood numbers for the air stream across the hollow fiber membrane bundle with in-line and staggered arrangement^[74,75]

	$S_L/(2r_0)$	填充率	四边形排列				三角形排列			
			$Nu_{ave,T}$	$Nu_{ave,H}$	$Nu_{ave,a}$	$Sh_{ave,a}$	$Nu_{ave,T}$	$Nu_{ave,H}$	$Nu_{ave,a}$	$Sh_{ave,a}$
$Re_a=200$	1.5	0.349	6.27	9.68	6.62	9.34	15.63	18.79	14.38	18.29
	2.0	0.196	4.46	5.52	5.52	5.43	11.14	14.15	9.79	13.92
	2.5	0.125	3.31	3.78	4.96	4.02	9.41	12.71	7.31	12.56
$Re_a=400$	1.5	0.349	10.07	12.52	8.98	11.72	21.62	24.57	20.88	23.74
	2.0	0.196	7.45	8.93	7.03	7.36	15.21	20.42	14.53	19.63
	2.5	0.125	5.84	6.72	6.62	5.32	13.62	17.51	12.56	16.94

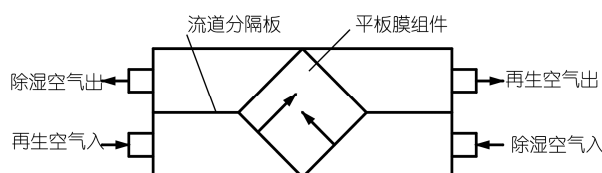


图7 膜式换热器内空气流动不均匀示意图

Figure 7 Schematic of the air flow maldistribution for total heat exchanger

能的影响,得出膜组件的填充率越低,组件内的流动分布越不均匀,组件的传热和传质效率下降也越明显。

3.2 管束的随机排布

由于纤维管数众多,纤维管之间不再是严格的三角形或四边形排列,而是随机排布的形式。Chen和Hlavacek^[81]提出利用Voronoi模型将随机分布纤维管附近的横截面积划分为一系列的多边形,每根纤维处于一个多边形的中心。Wu和Chen^[82]与Bao和Lipscomb^[83]等利用Voronoi模型分别计算了等壁面浓度和等壁面质量流边界条件下,组件逆流时的传质系数。相比较Voronoi模型,分形理论可以更好地反映随机分布的程度对管束传热传质的影响。Zhang提出了描述气体在多孔膜渗透的分形模型^[84],进一步,建立了随机分布的中空纤维膜组件在逆流时的分形模型^[85],确定了管束分布的分形维数与填充密度之间的联系,提出基于分形理论的传质准则方程式:

$$Sh = (14.06\varphi^4 - 29.21\varphi^3 + 22.59\varphi^2 - 7.71\varphi + 1.03) Re^{0.33} Sc^{0.33} \psi_k, \quad (3)$$

式中, φ , Sc 和 ψ_k 分别为填充率、施密特数和分形维数的校正系数,其中 ψ_k 表示为

$$\psi_k = 0.882D_f - 0.535, \quad (4)$$

式中, D_f 为分形维数。

Jiang等人^[86]研究了空气横掠随机分布椭圆纤维

管束的流动与传热过程,得出随机分布椭圆纤维管束比均匀分布纤维管束的流动与传热性能有不同程度的降低。

3.3 热量、质量积在膜式热湿交换机理研究中的应用

近几年,流体的传热传质机理在热力学方向又有了新的突破。Guo等人^[87]通过与导电现象的类比,提出了热量积的概念来描述物体向外界传递热量的能力,定义了热量积耗散来衡量传热过程中该物体传热能力的损失。指出熵产是热功转换过程不可逆性的量度,与热功转换无关的传热过程不可逆性的量为热量积耗散。Liu等人^[88]用热量积的形式描述了热力学第二定律,并对热传递过程进行了优化。Guo等人^[89]与Cheng和Liang^[90]分别利用热阻和热量积耗散原理对换热器进行分析,指出冷热流体的温差场越均匀,热量积耗散损失越小。江亿等人^[91]将热量积引入空气与水热湿转换过程中,定义了湿空气的质量积,指出在满足参数匹配的情况下,为达到质量积损失最小,应使湿空气尽量接近饱和线与水进行热湿交换。陈林等人^[92]导出了衡量溶液除湿过程中水分传递不可逆性的湿阻的表达式,并利用湿阻的概念对溶液除湿系统的除湿性能进行分析和优化。刘晓华等人^[93]利用热量积和质量积耗散对逆流除湿塔进行热湿传递分析,得出在溶液进口状态等浓度线上进行的热湿传递过程不匹配系数最小,而溶液进口状态等焓线上发生的过程不匹配系数较大。张宁等人^[94,95]用热量积耗散和质量积耗散对错流中空纤维膜除湿器的性能进行了研究,由于它偏离平衡逆流传热传质形式,相比较逆流除湿器的传热和传质不可逆损失都增大,并通过结构设计使质量积耗散最小。

4 新型的膜式湿度处理系统

膜式湿度处理系统除了除湿器外还有再生器、换热器、水泵和风机等组件. 近年来, 很多学者对系统各组件的参数优化以及系统的集成等进行了研究. Seyed-Ahmadi等人^[27,28]提出一种利用平板膜进行空气除湿的系统. 该系统由2个平板膜作为除湿器和再生器, 用输液管把它们与储液罐和溶液泵相连, 系统结构简单. 但是该系统对热湿负荷变化的适应性较差. Liang等人^[96]把膜式除湿系统与热泵系统联合应用, 并建立了相应的数学模型. 如图8(a)所示, 先用再生空气通过膜式全热回收器对新风进行除湿冷却, 当处理后的新风湿度达不到送风要求时, 用蒸发器对新风进行深度除湿并通过辅助冷凝器再热后送入室内. 该系统在额定工况下的制冷能效比(COP)达到6.8. 从上述介绍中可以看出, 热泵系统的加入可以更好地提高系统的负荷适用范围. Zhang等人^[97]开发了热泵驱动的中空纤维膜除湿系统, 并对整个系统的除湿过程进行了模拟. 如图8(b)所示, 除湿与再生器从平板膜换为填充密度更大的中空纤维膜组件, 并利用热泵的蒸发器与冷凝器分别对除湿与再生前的溶液进行冷却与加热, 热泵的制冷量与产热量都

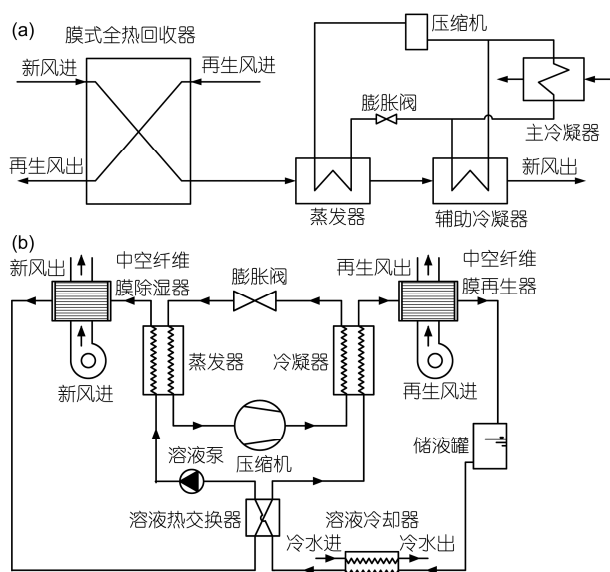


图8 热泵与膜式接触器联合除湿系统示意图. (a) 膜式全热回收制除湿系统; (b) 热泵驱动的中空纤维膜除湿系统

Figure 8 Schematic of the heat pump combined with membrane-based air dehumidification system. (a) Independent air dehumidification system with membrane-based total heat recovery; (b) heat pump driven and hollow fiber membrane-based liquid desiccant air dehumidification system

得到了综合利用, 可以大幅提高系统的能量利用系数, 可达4.0.

当有太阳能等低品位的可再生能源可以利用时, 张立志等人^[98]设计了一种太阳能驱动的中空纤维膜除湿系统. 如图9所示, 利用太阳能吸收装置对除湿后的稀溶液进行加热浓缩, 可以减少加热装置的能耗, 从而提高系统的能量综合利用效率.

由于除湿溶液吸收水蒸气时会释放出大量的潜热, 使溶液的温度不可避免的升高, 这样膜两侧的空气与溶液的水蒸气分压力差会减少, 从而降低除湿效果. 江亿等人^[99]提出一种利用喷淋塔的多级除湿系统, 将原有的一个除湿器用多个小型除湿器代替, 并在每个除湿器前对溶液进行降温从而降低除湿溶液的温升. 与同条件下的吸收式制冷相比较, 该系统的能源利用效率可以提高1.55~2.33倍. 类似地, 该方法可应用到膜式除湿系统, 如图10所示, 中空纤维膜除湿器代替了喷淋塔, 通过多级的溶液冷却和空气除湿过程, 相比较现有的单级系统可以获得

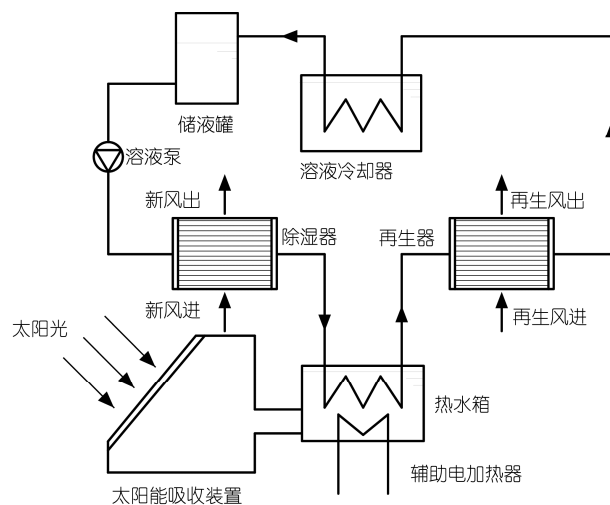


图9 太阳能驱动的中空纤维膜除湿系统示意图

Figure 9 Schematic of the solar energy driven and hollow fiber membrane-based liquid desiccant air dehumidification system

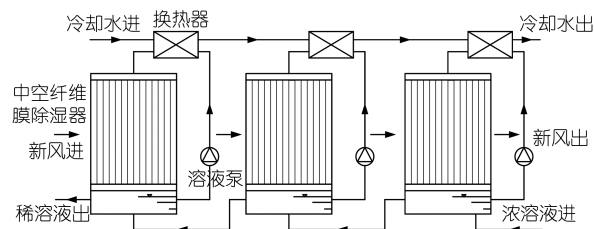


图10 多级除湿系统示意图

Figure 10 Schematic diagram of the multi-stage dehumidifier

更好的除湿效果。

5 结论与展望

从以上介绍可以看出,膜技术可以实现选择性地透过水蒸气而避免再生空气或除湿溶液中的小液滴对新风的污染。该技术可以较好地实现湿度控制,保证室内空气品质。当它与热泵或可再生能源利用相结合时,可以大幅升高能源利用效率,是一种节能环保的新技术,特别为解决高温高湿地区的建筑环境控制提供一条节能高效的途径。本文介绍了膜式热湿调控原理及其技术的进展,列举了这方面近年来在材料的制备、组件的流动与传热传质过程、系统的结构与优化等方面的研究进展。为了实现膜式热湿调控技术在工程实际中大规模的应用,今后还需要在以下方面进一步研究。

(1) 研究新型的膜材料。复合膜中多孔支撑层的传质阻力较大,约占全部阻力的73%^[100],同时多孔材料也加大了由于膜破损导致的流体泄露的风险。

因此需要开发新型材料以改进现有的膜材料。

(2) 膜材料结构的微介观研究。现有有关膜材料的研究大多立足于宏观传热传质特性,下一步可以从微介观角度利用格子玻尔兹曼等方法对膜材料结构进行深入研究,寻找提高传热传质能力的新途径。

(3) 膜组件的结构设计。目前膜组件基本采用绝热型结构,随着水蒸气被吸收后散发潜热,再生空气或除湿溶液的温度不断升高,限制了传质效率的提高。需要设计内冷型等除湿组件来减轻流体温升的影响。

(4) 膜式热湿控制系统的优化设计。膜组件与热泵或太阳能系统的联合运行可以扩大系统的热湿负荷适用范围,并提高系统的能源利用率。当系统从单级优化为多级的冷却-除湿模块后,系统的性能将进一步提高。

(5) 膜式热湿控制系统的运行控制。空气的热湿负荷随着室外气象条件而时时变化,需要建立系统的动态模型为运行控制提供理论指导。

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Membrane-based heat and moisture control: Progress on theory and technology

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Membrane-based heat and moisture treatment is a novel temperature and humidity control technology for energy conservation in buildings. Its moisture control effectiveness is high, and it is compact. Moreover, the traditional problem of desiccant droplet crossover is prevented. In recent years, much progress has been made in the development of this technology, from fundamentals to applications. In this review, advances are introduced, including novel membrane materials, new findings in conjugate heat and mass transfer in membrane modules, and new systems that combine air dehumidification with renewable energy use. A membrane made of a uniform material is weak on mass transfer and not suitable for engineering applications. A composite hydrophobic–hydrophilic membrane can substitute for it because of the reduction in mass transfer resistance. The heat and moisture transfer mechanisms in parallel plates, plate fins, and cross-corrugated membrane-based total heat exchangers are discussed. The heat and mass transfer properties in hollow fibers modules are described. The conjugate heat and mass transfer on the surface membrane, flow mal-distribution in the membrane modules and the randomly distributed nature of heat in the tube banks are ideal for application in dehumidification systems. Their effects on heat and mass transfer are presented. The correlations for heat and mass transfer, and the detailed Nusselt and friction data for module design, are summarized. They provide the fundamentals for system design and optimization. Moreover, the heat and mass entransy dissipation, which is a new concept in thermodynamics, is introduced to analyze the system performance. It is found that the larger the heat and mass entransy dissipation, the more heat is lost and the more irreversible the mass transfer for air dehumidification. In addition, the system set-up and the applications of this new technology are introduced. The heat pump driven liquid dehumidification systems perform well because the cooling and heating before dehumidification and regeneration can be supplied simultaneously and efficiently. The energy-use efficiencies are greatly improved if they are combined with solar energy use. The single-stage desiccant system can be improved by separation into a multi-stage system, where the desiccant is inter-cooled after absorbing moisture. In future, to extend this technology to commercial applications, the following research needs to be addressed: new low-cost membrane materials, internally cooled membrane-based modules, multi-stage membrane liquid desiccant air dehumidification systems, and real-time dynamic simulation technology.

membrane-based heat and moisture control, membrane material, membrane module, dehumidification system, heat and mass transfer

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