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线性压缩机迷宫密封泄漏特性及结构优化研究

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摘　要　为降低无油线性压缩机活塞与气缸的间隙泄漏,提高压缩机密封性能,本文提出一种适用于线性压缩机自由活塞间隙密封的矩形迷宫结构,建立了包含自由活塞、工作腔、活塞径向间隙及背腔结构的瞬态数值模型,以对迷宫空腔特征尺寸及空腔宽度、深度和间隔对密封性能的影响进行研究。研究结果表明:迷宫空腔特征尺寸存在最优取值,当特征尺寸为0.2 mm时,净泄漏率降至0.798%。针对单一迷宫密封结构参数进行了研究,当空腔宽度为0.2 mm、深度为0.1 mm、间隔为0.2 mm时,迷宫密封综合性能最佳。同时,本文对迷宫密封与间隙密封抑漏性能进行对比分析,结果表明:迷宫密封结构能够显著降低瞬时间隙质量流量,具有更明显的抑制泄漏损失效果,且在高压比下,该效果更为显著。在压比为6.0的工况下,迷宫密封结构的净泄漏率降低约46.52%。研究结果可为无油线性压缩机非接触密封结构设计及参数优化提供理论依据。

关键词　有阀线性压缩机;迷宫密封;间隙泄漏;数值模拟;密封性能
中图分类号:TB61+1;TB652　　　　**文献标识码**:A

Leakage Characteristics and Structural Optimization of Labyrinth Seals for Linear Compressors

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Abstract

Objective　Oil-free linear compressors employ a narrow clearance between piston and cylinder to achieve noncontact operation, thereby eliminating lubrication-related contamination and mechanical wear. However, the unavoidable clearance leakage reduces the volumetric efficiency and becomes increasingly severe under high-pressure conditions, thereby limiting overall compressor performance. Labyrinth seals have been widely adopted in turbomachinery and reciprocating compressors owing to their excellent leakage-suppression capability. Nevertheless, their application in oil-free linear compressors has rarely been reported, and the influence of cavity geometry on transient leakage characteristics has not been systematically clarified. In this study, a rectangular labyrinth seal was proposed for sealing the piston - cylinder clearance, and the effects of cavity geometry on sealing performance were investigated.

Method　A transient computational fluid dynamics (CFD) model was established for an oil-free Oxford-type moving-magnet linear compressor. The numerical model incorporated the compression chamber, piston-cylinder radial clearance, rectangular labyrinth seal, and back chamber, enabling simulation of the transient leakage process throughout one operating cycle. Simulations were performed at pressure ratios of 2, 3, 4, 5, and 6. The characteristic cavity size was first evaluated by assigning identical values of 0.1, 0.2 and 0.4 mm to the cavity width, depth and spacing, respectively. Subsequently, the cavity width, depth, and spacing were varied independently, whereas the remaining parameters were maintained at their reference values. The sealing performance was evaluated using the instantaneous leakage mass-flow rate, net-leakage ratio, and transient pressure, velocity, and refrigerant density distributions. Finally, the optimized labyrinth seal was compared with a conventional smooth-clearance seal under identical operating conditions.

Results and Discussion　Cavity geometry significantly influenced transient-leakage characteristics. An optimum characteristic cavity

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移动阅读

size of 0. 2 mm was obtained, corresponding to a net-leakage ratio of 0. 798%. Independent parametric analyses further indicated that the optimum cavity dimensions were 0. 2 mm width, 0. 1 mm depth, and 0. 2 mm spacing, yielding a minimum net-leakage ratio of 0. 668%, which decreased progressively along the leakage path because of successive throttling through adjacent labyrinth cavities. Meanwhile, the refrigerant density exhibited a layered distribution that reversed with the leakage direction during different stages of the compression cycle. The transient velocity field showed that the local high-velocity regions were primarily concentrated within the throttling gaps between adjacent cavities, whereas the low-velocity recirculation zones were maintained inside the cavities throughout the cycle. Compared with the conventional smooth-clearance seal, the proposed labyrinth seal reduced the net-leakage ratio by 46. 52% at a pressure ratio of 6. Moreover, leakage suppression became increasingly pronounced as the pressure ratio increased, demonstrating the superior sealing capability of the labyrinth configuration under high-pressure ratio conditions.

Conclusion The proposed rectangular labyrinth seal effectively suppressed clearance leakage while maintaining noncontact piston operation. An appropriate cavity geometry is essential for achieving high sealing performance. The optimized structure provided an effective sealing solution for oil-free linear compressors operating under high-pressure ratio conditions.

Keywords valve-controlled linear compressor; labyrinth seal; clearance leakage; numerical simulation; sealing performance

随着新能源汽车热管理、电子冷却以及热泵等领域对高效制冷技术需求的持续增长,压缩机作为蒸气压缩式制冷系统的核心部件,其性能直接影响系统能效和运行可靠性。近年来,线性压缩机因结构简单、效率高及易于容量调节等优势,在制冷领域受到广泛关注[1-2]。面对线性压缩机的高效运行需求,国内外学者围绕动力学特性、容量调节及高压比工况适应性开展了深入研究,明确了活塞运动、气体力耦合及关键结构参数对压缩机性能的影响规律[3-5]。为实现无油运行,线性压缩机采用间隙密封结构,依靠活塞与气缸间的微小径向间隙实现非接触运行与间隙密封,具有摩擦损失小、可靠性高和使用寿命长等优点,已成为无油线性压缩机广泛采用的密封形式,但活塞与气缸壁间的径向间隙也不可避免地造成了间隙泄漏[6]。随着压比提高,间隙泄漏进一步加剧,并导致活塞偏移,降低系统运行稳定性和能效[7-8]。因此,在保证活塞无油运行的前提下有效抑制间隙泄漏,是进一步提升无油线性压缩机性能的关键。

针对间隙泄漏问题,迷宫密封凭借无接触运行、低摩擦及高可靠性等特点,已广泛应用于旋转机械非接触密封领域。迷宫密封通过多个节流单元逐级释放压力,并利用空腔内复杂流动降低泄漏量,其理论模型和设计方法已较为成熟[9-11]。随着数值模拟技术的发展,迷宫密封内部复杂流动机理得到了进一步揭示。Wei Xin等[12]通过数值仿真发现,节流射流进入空腔后形成明显回流结构和涡旋耗散区域,从而实现流体动能衰减。Huo Changjiang等[13]研究表明,不同齿形会显著改变迷宫密封内流场结构及泄漏特性。Xue Wensong等[14]进一步发现,通过优化空腔结构和齿形布置,可增强流体能量耗散能力并降低泄漏量。此外,密封间隙非均匀分布、压比及运行工况变化会进一步影响迷宫密封内部压力分布及泄漏特性[15-16]。国内学者进一步从密封齿优化设计、密封间隙匹配及流固耦合分析等方面开展研究,揭示了迷宫几何结构、结构变形及流动特性间的耦合关系,为迷宫密封结构优化提供了理论依据[17-19]。

虽然迷宫密封结构已在运动机械中得到广泛应用,其流动机理及结构优化研究较为成熟,但针对无油线性压缩机自由活塞间隙的迷宫密封设计及其泄漏特性缺乏系统研究。本文以动磁式有阀线性压缩机为研究对象,建立包含工作腔、自由活塞、活塞径向间隙、迷宫密封结构及背腔的瞬态数值模型,揭示迷宫空腔几何参数对间隙泄漏特性的影响规律,并获得适用于线性压缩机的迷宫密封优化参数,为无油线性压缩机非接触密封结构设计提供理论依据。

1 数值模型

1.1 模型结构及网格划分

本文基于ANSYS Fluent平台建立的数值模型以动磁式有阀线性压缩机为原型[20],如图1所示,压缩机主要由动磁式直线电机、板弹簧支撑系统、自由活塞及气缸组件以及吸排气阀构成。自由活塞与气缸之间采用间隙密封,设计径向间隙为15 μm,从而实现无油运行,压缩机主要部件规参数如表1所示。

本研究采用模块化分区网格划分策略,如图2所示,针对不同区域流动特征采用不同的网格形状和尺寸。工作腔与背腔区域仅发生腔内容积变化,且变形过程较为规律。该区域采用四边形网格,网格尺寸为0.5 mm,使用层铺动网技术模拟活塞运动引起的体积变化。活塞与气缸间隙是影响泄漏特性的关键区域,径向尺寸为15 μm,对计算精度要求极高。同时采用高分辨率四边形静态网格,径向划分5层网格,尺寸为0. 003 mm×0. 1 mm,以确保间隙内流动细

表1 线性压缩机主要部件参数

Tab.1 Parameters of the main components of the linear compressor

参数	数值
活塞直径/mm	25
最大行程/mm	14
运行频率/Hz	30~50
机械弹簧刚度/(N/ mm)	17
电机力常数/(N/A)	30
阻尼系数/(N·s/m)	0.047 5
活塞质量/kg	0.66

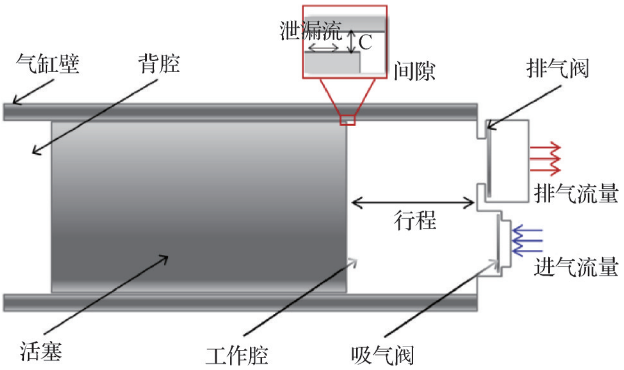


图1 压缩机数值模型几何结构

Fig.1 Geometric structure of the numerical compressor model

节的准确捕捉。各区格通过流体交界面连接,确保流体可在间隙、工作腔、进排气阀以及间隙之间流动。各区域网格划分参数如表2所示。

表2 网格划分参数

Tab.2 Mesh parameters

区域	网格类型	网格尺寸/mm	网格数量
工作腔	四边形网格	0.5	约25 000
背腔	四边形网格	0.5	约20 000
间隙区域	四边形网格	0.003	约3 000
吸排气流道	三角形网格	0.1	约8 000
总计			约56 000

1.2 控制方程与湍流方程

线性压缩机内流体流动、热传递和质量传递过程分别基于连续性方程、动量守恒方程和能量守恒方程:

$$\frac{\partial \rho}{\partial t}+\frac{\partial}{\partial x_i}(\rho u_i)=0 \tag{1}$$

$$\rho\left(\frac{\partial u_i}{\partial t}+u_j\frac{\partial u_i}{\partial x_j}\right)=\frac{\partial \sigma_{ij}}{\partial x_j}-\frac{\partial p}{\partial x_i} \tag{2}$$

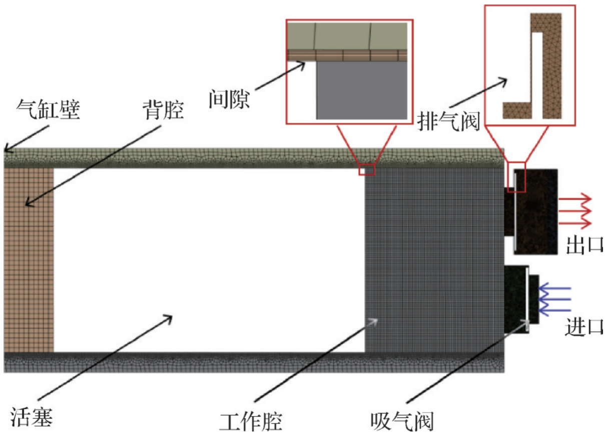


图2 压缩机数值模型网格划分

Fig.2 Mesh of the numerical model of the linear compressor

$$\frac{\partial}{\partial t}(\rho e_t)+\frac{\partial}{\partial x_j}(\rho u_j e_t)=-\frac{\partial}{\partial x_j}(pu_j)+\frac{\partial}{\partial x_j}(\sigma_{ij}u_i)+\frac{\partial}{\partial x_j}\left(\lambda\frac{\partial T}{\partial x_j}\right) \tag{3}$$

式中:ρ为密度,kg/m^3;t为时间,s;u为速度分量,m/s;x为坐标分量,m;p为压力,Pa;σ_{ij}为黏性应力张量,Pa;e_t为单位质量能量,J/kg;T为温度,K;λ为导热系数,W/(m·K);下标i、j表示坐标方向。

本模型采用广泛应用的RNG(k-ε)湍流模型,湍动能(k,m^2/s^2)和耗散率(ε,m^2/s^3)由式(4)和式(5)求得。

$$\frac{\partial}{\partial t}(\rho k)+\frac{\partial}{\partial x_i}(\rho k u_i)=\frac{\partial}{\partial x_j}\left(\alpha_k\mu_{\text{eff}}\frac{\partial k}{\partial x_j}\right)+G_k+G_b-\rho\varepsilon-Y_M+S_k \tag{4}$$

$$\frac{\partial}{\partial t}(\rho\varepsilon)+\frac{\partial}{\partial x_i}(\rho\varepsilon u_i)=\frac{\partial}{\partial x_j}\left(\alpha_\varepsilon\mu_{\text{eff}}\frac{\partial \varepsilon}{\partial x_j}\right)+C_{1\varepsilon}\frac{\varepsilon}{k}(G_k+C_{3\varepsilon}G_b)-C_{2\varepsilon}\rho\frac{\varepsilon^2}{k}-R_\varepsilon+S_\varepsilon \tag{5}$$

式中:μ_{eff}为有效动力黏度,Pa·s;α_k和α_ε分别为湍动能和耗散率对应的有效普朗特数倒数;G_k和G_b分别为平均速度梯度和浮力引起的湍动能生成项,kg/(m·s^3);Y_M为可压缩湍流中脉动膨胀对耗散率的影响项,kg/(m·s^3);S_k为用户定义的湍动能源项,kg/(m·s^3);S_ε为用户定义的耗散率源项,kg/(m·s^4);$C_{1\varepsilon}$和$C_{2\varepsilon}$为耗散率方程模型常数;$C_{3\varepsilon}$为浮力影响修正系数;R_ε为湍流模型中的附加修正项。

与标准k-ε模型相比,RNG k-ε模型最显著的特点是在ε方程中引入了附加项R_ε,其表达式为:

$$R_\varepsilon=\frac{C_\mu\rho\eta^3(1-\eta/\eta_0)}{1+\beta\eta^3}\frac{\varepsilon^2}{k} \tag{6}$$

$$\eta=Sk/\varepsilon \tag{7}$$

式中：C_{μ}为湍流黏度模型常数；S为平均应变率张量模，s^{-1}；η_0和β为RNG模型常数。

1.3 边界条件设置

根据动磁式有阀线性压缩机参数，设置出口压力为1.35 MPa，压比覆盖2.0~6.0。气缸壁面设置为耦合传热壁面，参与流体与固体的传热计算，环境温度恒定为20 ℃。活塞壁面及气缸盖面均设为绝热无滑移壁面。背腔设为封闭腔体，其压力随工作腔泄漏质量交换过程动态变化，与工作腔及活塞径向间隙形成瞬态耦合。

1.4 动网格设置

本模型采用混合动网格策略，包含弹簧光顺、层铺和局部重构。吸排气流道区域使用弹簧光顺和局部重构动网格。工作腔和背腔区域仅产生容积变化，采用层铺动网格。为提高精度，关键区域网格尺寸设定为：阀片附近0.1 mm，活塞附近0.4 mm，以平衡计算精度和效率。

1.5 网格无关性验证

选取R134a工质和压比为4作为典型工况，以排气质量流量作为网格无关性验证的考核指标进行对比分析，相应的验证工况设定参数如表3所示。

表3 网格无关性验证工况

Tab.3 Operating conditions for the mesh independence verification

吸气压力/MPa	吸气温度/K	排气压力/MPa	排气温度/K
0.3375	277.14	1.35	324.11

在无关性验证中，使用3套不同密度的网格进行模型求解，网格数量分别为32 000、56 000、87 000。3套网格均采用相同的分区划分策略，分别进行瞬态计算，结果如表4所示。粗网格的计算结果与细网格的偏差较大，达8.4%，中等网格与细网格的计算结果差异仅为1.1%。综合考虑计算精度与计算效率，采用约5.6万网格数的方案进行仿真模拟。

2 模型验证

本文使用动磁式有阀线性压缩机和R134a作为工质，开展了一系列实验来验证模型的可靠性。压缩机泄漏流量及轴功率的仿真值与实验值对比分别如图3和图4所示，所有工况下两者吻合良好，最大误差在±10%以内。因此，本文建立的压缩机数值模型具有较高的可靠性，适用于分析迷宫密封对线性压缩机间隙泄漏的影响。

表4 网格无关性验证仿真结果

Tab.4 Simulation results of the mesh independence verification

方案	网格数量	质量流量/(g/s)	与细网格的偏差
粗网格	32 000	0.87	8.4%
中等网格	56 000	0.94	1.1%
细网格	87 000	0.95	—

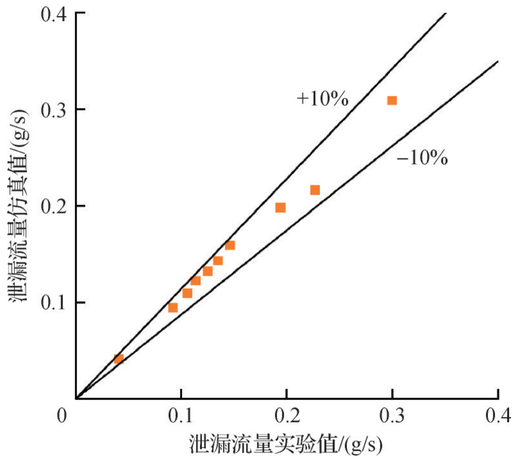


图3 泄漏流量仿真值与实验值对比

Fig.3 Comparison of simulated and experimental leakage flow rates

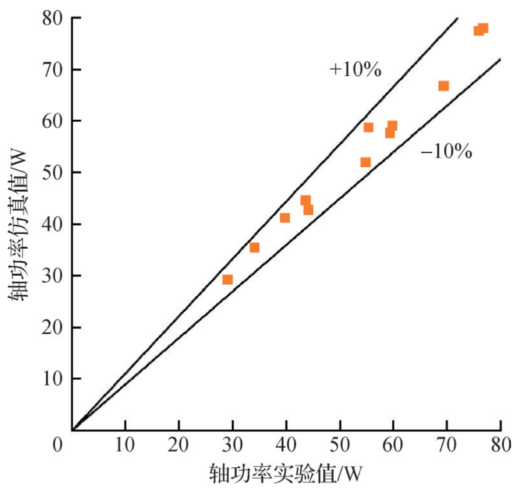


图4 轴功率仿真值与实验值对比

Fig.4 Comparison of simulated and experimental shaft power

3 结果与分析

线性压缩机工作循环内温度场随活塞往复运动呈周期性演化，1个工作循环温度场分布如图5所示。工质依次经历吸气、压缩、排气和膨胀过程，压缩腔温度在压缩过程中逐渐升高，并在排气阶段达到最大值，而背压腔温度变化相对平缓。两腔之间持续存在热力学状态差异，并形成周期性压力梯度，为工质经活塞与气缸径向间隙泄漏提供驱动力。

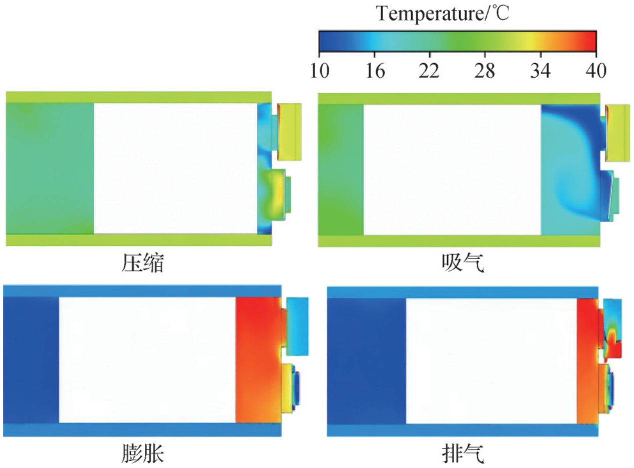


图5 线性压缩机1个工作循环温度场分布

Fig.5 Temperature distribution during one cycle of the linear compressor

在压比为6.0时，分析了矩形迷宫空腔特征尺寸对密封性能的影响，结果如图6所示。为表征迷宫结构尺寸，将空腔宽度、深度和间隔设为相同数值，并定义该数值为迷宫空腔特征尺寸。本文中，质量流量由背腔流入工作腔为正方向，工作腔流入背腔为负方向。由图6(a)可知，不同特征尺寸下瞬时间隙质量流量变化趋势基本一致，均随活塞往复运动呈周期性变化，其中特征尺寸为0.2 mm时，正、负方向瞬时间隙质量流量峰值均最小，净泄漏率约为0.798%。结合图6(b)可知，较小的特征尺寸限制了泄漏流体在空腔内的射流扩散及动能耗散；当特征尺寸增至0.2 mm时，空腔内形成明显的低速回流区，多级节流效应显著，泄漏流体的流动动能得到充分耗散，密封性能最佳；继续增大特征尺寸，虽然空腔内仍存在低速区，但节流效应减弱，导致密封性能下降。因此，合理匹配迷宫空腔几何尺寸有利于增强迷宫密封的抑漏性能。本节确定迷宫空腔特征尺寸的最优值为0.2 mm，并在此基础上进一步分析空腔宽度、深度和间隔对密封性能的独立影响。

在迷宫空腔宽度和间隔均为0.2 mm条件下，分析了空腔深度对迷宫密封性能的影响，结果如图7所示。由图7(a)可知，不同空腔深度下瞬时间隙质量流量变化趋势基本一致，其中空腔深度为0.1 mm时正、负方向瞬时间隙质量流量峰值均最小，说明该结构对泄漏流动的抑制作用最为显著。结合图7(b)可知，当空腔深度较小时，空腔流动空间有限，泄漏流体难以充分扩散，流速衰减不明显；空腔深度增至0.1 mm后，空腔内低速区域明显扩大，泄漏流体流速得到有效衰减，有利于降低泄漏量；继续增大空腔深

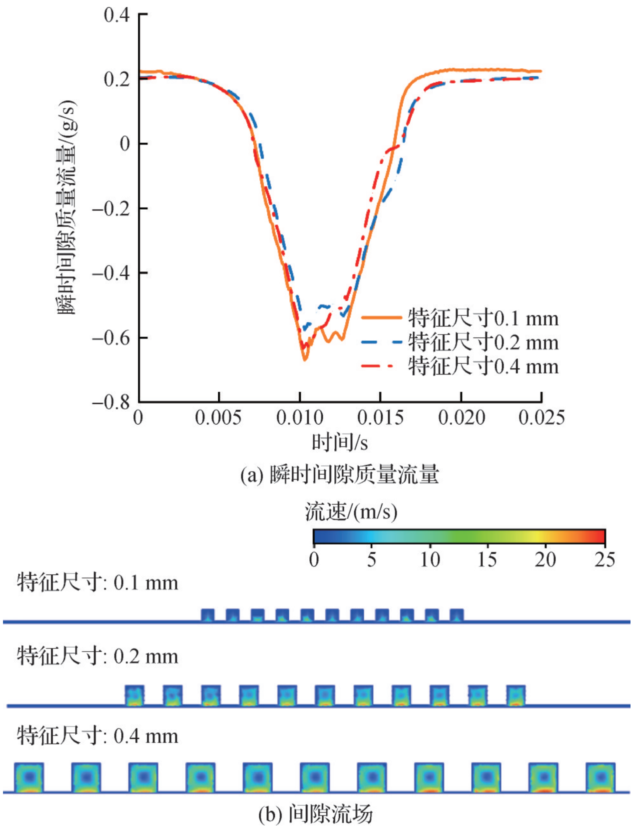


图6 空腔特征尺寸对迷宫密封性能的影响

Fig.6 Effect of cavity characteristic dimension on labyrinth seal performance

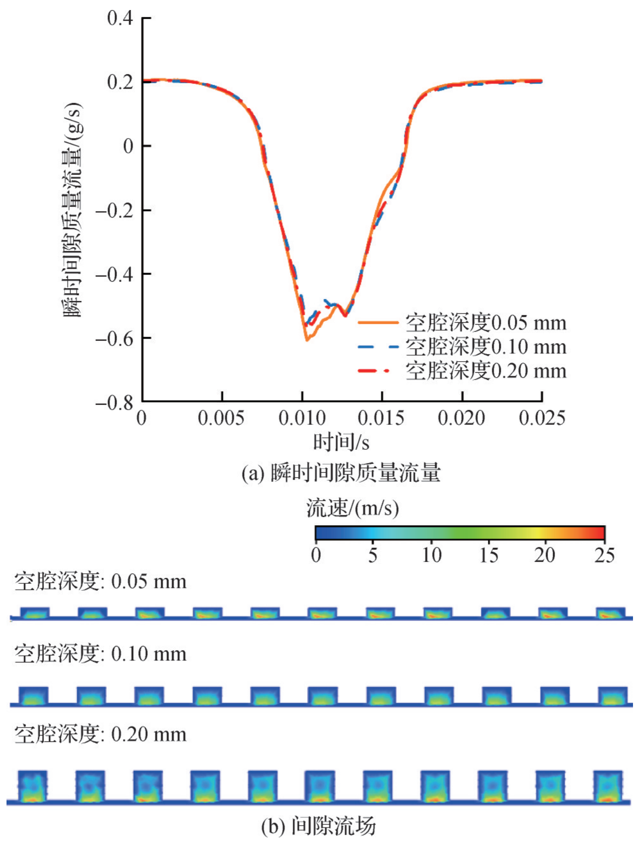


图7 空腔深度对迷宫密封性能的影响

Fig.7 Effect of cavity depth on labyrinth seal performance

度时，低速区域变化趋于有限，而空腔容积进一步增加削弱了相邻节流单元之间的协同节流作用，密封性能反而下降。在本研究工况下，迷宫空腔深度存在适宜取值，合理的空腔深度能够兼顾流速衰减与多级节流效应，从而获得较好的密封性能。

在迷宫空腔宽度为0.2 mm、深度为0.1 mm条件下，分析了空腔间隔对迷宫密封性能的影响，结果如图8所示。由图8(a)可知，空腔间隔为0.1 mm时，瞬时间隙质量流量负向峰值略大；当空腔间隔增至0.2 mm及以上时，瞬时间隙质量流量曲线基本一致。结合图8(b)可知，较小空腔间隔下相邻空腔流动耦合作用较强，不利于空腔内射流扩散和动能耗散；空腔间隔增至0.2 mm后，空腔间流动干扰减弱，空腔内动能耗散更加充分，从而改善了密封性能。当空腔间隔进一步增至0.4 mm时，空腔内流动结构变化不明显，说明空腔间隔超过一定范围后，对迷宫密封抑漏性能的改善作用有限。适当增大空腔间隔有利于削弱相邻空腔间的流动干扰，增强空腔内泄漏流体的动能耗散；当空腔间隔超过临界值后，其对迷宫密封抑漏性能的提升趋于饱和。

选取空腔宽度0.2 mm、深度0.1 mm、间隔0.2 mm的迷宫结构，对空腔内流场进行分析，结果如图9所示。迷宫空腔内流动受工作腔与背腔瞬时压差驱动，并随压缩机工作循环呈现周期性变化。压缩阶段，泄漏流体由工作腔流向背腔，空腔内流速主要分布于0~5 m/s，高速区集中于相邻空腔间的节流间隙，并伴随局部涡旋形成。排气阶段压差进一步增大，节流间隙处流速增至约10 m/s，空腔内回流涡旋更加明显。膨胀与吸气阶段，流动方向反转，流体由背腔回流至工作腔，高速区相应转移至另一侧节流间隙，其中膨胀阶段局部流速为5~8 m/s，吸气阶段随压差增大流速进一步提高。整个工作循环中，迷宫空腔内始终保持多级节流与回流涡旋并存的流动特征，促进泄漏流体动量耗散，从而有效抑制间隙泄漏。

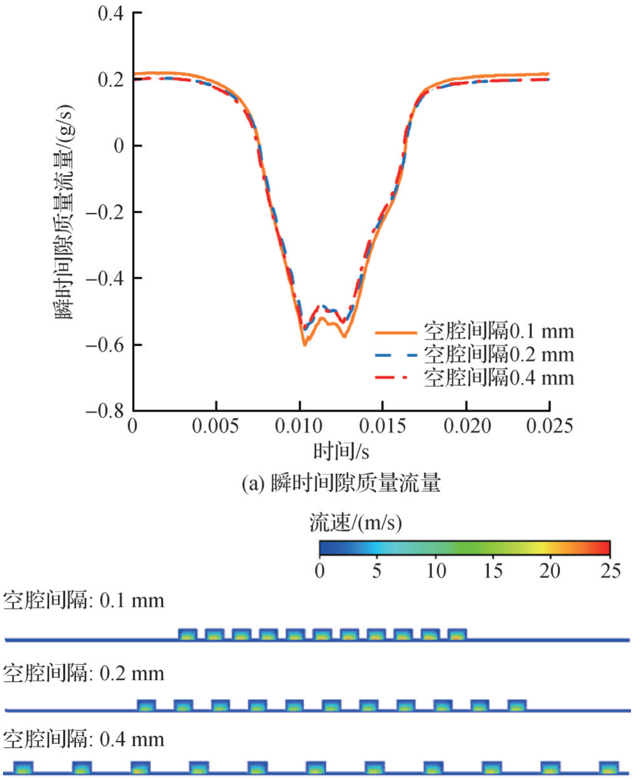


图8 空腔间隔对迷宫密封性能的影响

Fig.8 Effect of cavity spacing on labyrinth seal performance

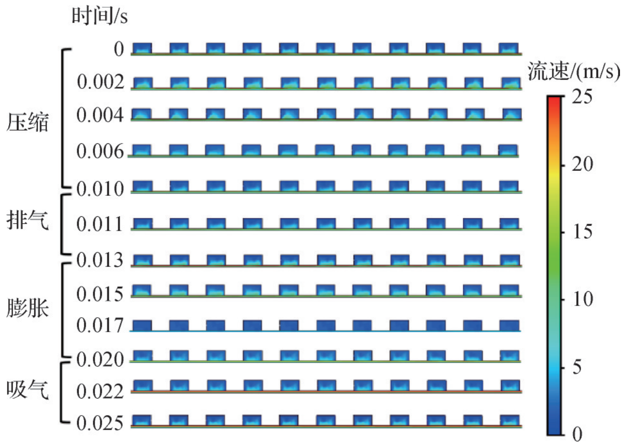


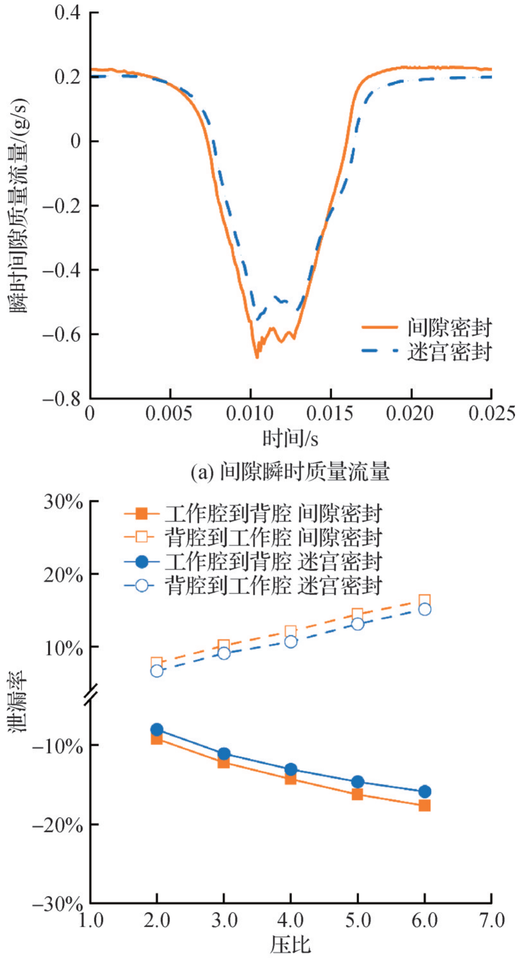
图9 迷宫空腔流场

Fig.9 Flow field in the labyrinth cavity

选取空腔宽度为0.2 mm、深度为0.1 mm、间隔为0.2 mm的矩形迷宫结构，将使用该规格迷宫密封的压缩机与间隙密封的压缩机进行对比，结果如图10所示。压比为6.0时，采用间隙密封和迷宫密封的压缩机间隙瞬时间隙质量流量如图10(a)所示，迷宫密封压缩机的瞬时间隙质量流量峰值均低于间隙密封压缩机，净泄漏率降低46.52%。间隙密封和迷宫密封压缩机双向泄漏率如图10(b)所示。相较于传统间隙密封，迷宫密封在各压比工况下均表现出更低的双向泄漏率。随着压比进一步提高，2种密封结构间泄漏率差异逐渐增大，表明迷宫密封在高压比工况下具有更显著的抑漏优势。

4 结论

本文针对无油线性压缩机间隙泄漏问题，提出了一种矩形迷宫密封结构，并建立了包含工作腔、迷宫密封及背腔的瞬态数值模型，研究了迷宫空腔特征尺寸及宽度、深度和间隔对密封性能的影响，得到如下结论：



注：泄漏率由背腔流入工作腔定义为正值，工作腔流入背腔定义为负值。

(b) 双向泄漏率

图 10 迷宫密封与间隙密封性能对比

Fig.10 Comparison of labyrinth and clearance seal performance

1）与传统间隙密封相比，矩形迷宫密封能够显著降低瞬时间隙质量流量和净泄漏率，且随着压比提高，其抑漏优势进一步增强，在压比为 6.0 时，净泄漏率降低 46.52%。

2）矩形迷宫空腔几何参数对密封性能的影响显著，各参数均存在适宜取值。其中，迷宫空腔特征尺寸为 0.2 mm 时净泄漏率最低，为 0.798%；进一步优化单一结构参数后，空腔宽度为 0.2 mm、深度为 0.1 mm、间隔为 0.2 mm 时，迷宫密封综合性能最佳。

3）合理匹配迷宫空腔几何参数能够增强多级节流作用，提高迷宫密封抑漏能力；当空腔尺寸超过适宜范围后，继续增大对密封性能的改善有限。

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