

端弯叶片通道出口二次流的研究

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【摘要】 实验并且采用考虑粘性的涡传输方程计算端弯压气机叶片通道出口二次流。给出了 3 种端弯叶片及对应的原始直叶片通道出口气流偏转角的实验值和计算预测值。研究表明: 采用上述方程能比较准确地预测出口气流偏转角; 叶片后缘减弯可减小流向二次涡强度。

主题词: 叶片 二次流 流场

分类号: V 231.3 V 232.4

1 前 言

国内外大量研究表明, 采用端弯叶片可明显减小端区流动损失。实施端弯后, 叶片通道出口流场将随之改变。在此主要考察端弯对出口二次流的影响, 以及直叶片通道能比较准确地预测出口气流二次流的涡传输方程对端弯叶片通道的适用性。本文采用有粘的二次涡传输方程对 3 种不同端弯参数以及对应的没有实施端弯的压气机叶片通道出口二次流进行了计算预测, 并且对上述 4 种叶片通道出口流场进行了实验测量, 分析比较了测量值和计算值。

2 通道二次流计算

对于不可压流, 且转动角速度为零, 根据文献[1]在 $(\vec{b}, \vec{n}, \vec{s})$ 坐标系下(见图 1), 可得主流方向涡量 ω_s 和流线副法向涡量 ω_b 的传输方程如下:

$$(\partial/\partial s)(\omega_s/V) = 2\omega_b/R + (1/\rho V^2)(\partial/\partial s^2)(\mu\omega_s) \quad (1)$$

$$(\partial/\partial s)(\omega_b/V) = -(\omega_s V/a_b)(\partial a_b/\partial s) + (1/\rho)(\partial/\partial s^2)(\mu\omega_b) \quad (2)$$

$\mu = \mu_l + \mu_t$, R 为流线的曲率半径, a_b 为相邻流线间间距。

要通过方程(1)、(2)求得 ω_s 、 ω_b 首先要给出主流速度 $V(s, b)$ 及流面的坐标。在此假设主流沿叶片中弧线构成曲面流动; 用欧拉方程计算出势流区速度 $V_e(s)$, 根据

$$V(s, b)/V_e(s) = (b/\delta)^{1/7} \quad (3)$$

求得 $V(s, b)$, δ 为附面层厚度。

涡粘性系数 μ_t 采用代数模型方程^[2]计算。

通常二次涡计算不考虑端壁附面层沿流向发展, 即壁面剪切应力 τ_w 和方程(3)中 δ 沿流向近似假设不变。但对于压气机叶片通道, 尤其是高压比压气机通道, 端壁附面层发展较快。在此本文提出计算附面层发展的近似方法, 忽略主流流线弯曲的影响, 根据 Head 公式有:

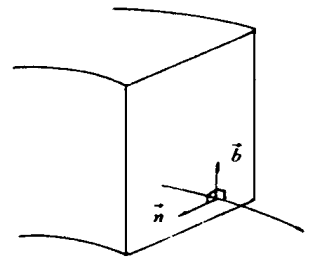


图 1 坐标系示意图

$$d\theta/ds = C_f/2 - (\theta/R_e)(dV_e/ds)(2+H), \quad E = (d/ds)(V_e\theta_{H_1}) = F(H_1) \quad (4)$$

由(4)式得:

$$dH_1/ds = (1/V_e\theta)F(H_1) + (H_1/V_e)(dV_e/ds)(1+H) = H_1C_f/2\theta \quad (5)$$

其中, $H_1 = G(H)$, $H = \delta^*/\theta$

$$F(H_1) = 0.0306(H_1 - 3.0)^{-0.6196} \quad (6)$$

$$G(H) = \begin{cases} 0.8234(H - 1.1)^{-1.287} + 3.3 & H \leq 1.6 \\ 1.5501(H - 0.6778)^{-3.064} + 3.3 & H > 1.6 \end{cases} \quad (7)$$

假设^[3]:

$$C_f = \frac{\tau_w}{1/2\rho V_e^2} = 0.246Re^{-0.268}10^{-0.678H} \quad (8)$$

此外, 二次流速度和涡量之间存在下列数学关系。假设 u, v 分别为 $\vec{b}, \vec{n}$ 方向的二次流速度, 则有:

$$u = -(\partial\psi/\partial n), \quad v = -(\partial\psi/\partial b), \quad \nabla^2\psi = -\omega_1 \quad (9)$$

ψ 为二次流函数。如图 2, $\omega_1 = \omega\cos\theta, \omega_2 = \omega\sin\theta$

$$\Delta a = \tan^{-1}(v/V) \quad (10)$$

Δa 为气流偏转角。

涡量边界条件。进口边界 ω_{b0}, ω_{n0} 给定, 半叶高处 ω_{be}, ω_{ne} 为零, 端壁面 $\omega_{bw} = (\partial\psi/\partial b)|_w = \tau_w/\mu_w$, ω_{bw} 采用迭代法求得, 具体为: 先给定一个初值, 由(1)式计算出内点值, 再由内点值外插求出新的 ω_{bw} , 反复迭代至收敛。

二次流函数边界条件。在端壁处和叶片表面, $\psi_w = 0$, 在半叶高处 $\partial\psi/\partial b = 0$ 。

计算过程: (1) 由欧拉方程求得 $V_e(s)$; (2) 根据给定的通道进口截面 δ^*, θ , 采用四阶龙格—库塔法求解(4) - (8), 得 $\theta(s), \delta^*(s), \tau_w(s)$; 根据(3)式可推得

$$\delta/\delta^* = 1/0.125 \quad (11)$$

由此计算出 $\delta(s)$; (3) 由(3)式计算出 $V(s, b)$; (4) 计算紊流粘性系数 μ_t ; (5) 对方程(1)、(2)进行差分离散, 采用追赶法沿流向推进求得 $\omega(s, b), \omega_1(s, b)$; (6) 由(9) - (11)式计算出 Δa 沿壁面法向分布。

3 计算和实验结果

实验设备及实验叶栅几何参数见文献[4]。3种端弯叶片参数见表 1。

表 1 叶片的端弯参数及出口涡强度

叶片编号	N00	N01	N02	N03
l_t/mm	0.0	4.1	8.3	12.5
l_h/mm	0.0	20.0	20.0	20.0
$\bar{\omega}_1/(1/s)$	334.0	331.1	294.1	281.1

N00 对应于未实施端弯的直叶片, l_t 和 l_h 见图 3。

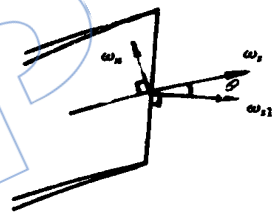


图 2 涡分量之间的关系

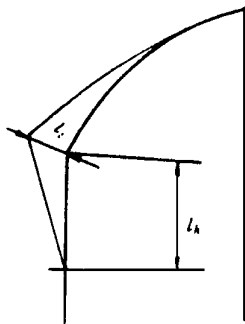


图 3 端弯叶片示意图

图 4 为在叶栅出口额线平面内实验和计算出的平均气流偏转角 $\Delta\bar{\alpha}$ 沿壁面法向分布 (图中 $\circ$ 实验值, — 计算值)。由图可知, 当端弯程度增加时, 在靠近壁面处, 预测值与实验值偏离较大。造成这种偏差增大的原因可能是, 当端弯程度增大时, 叶型中弧线构造的曲面扭曲相应增大, 假设主流流面与此曲面重合与实际存在的偏差也较大。但总体来看, 气流偏转角预测值和实验值吻合较好。

在此引入

$$\bar{\omega}_1 = \frac{\int \omega_1 V db}{\int V db} \quad (12)$$

h 为半叶高, ω_1 为出口二次涡平均强度。则由表 1 可看出后缘部份减弯, 可减小出口流向二次涡强度。这是因为后缘部份减弯, 使得气流流过叶片通道转折角减小, 进而使得流向二次涡强度减小。

4 结 论

(1) 采用有粘二次涡传输方程能够比较准确地预测端弯叶片通道出口气流偏转角。(2) 叶片后缘部份减弯可减小出口流向二次涡强度。

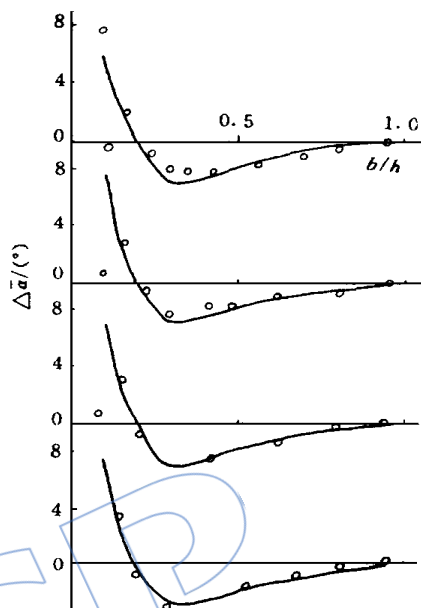


图 4 平均气流偏转角 $\Delta\bar{\alpha}$ 分布

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very effective and applicable for practical engineering

Key Words Rocket engines Fault detection Autocorrelation functions Real time

SURVEY PAPERS

APPLICATION AND LIFE RESEARCH OF POWDER METALLURGY TURBINE DISKS

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ABSTRACT Since powder metallurgy (PM) turbine disks have high yield strength and excellent property of resistance to fatigue and creep, they have been applied broadly in the most advanced engines at present in the world. As the thrust-to-weight ratio increases, the PM turbine disks will be developed in our country. At first, this paper makes a general survey of the application of the PM turbine disks. Then a review is made for the material properties, fatigue life, crack propagation, and stress analysis of PM turbine disks. Finally, some research projects of the PM turbine disks fatigue life, which should be conducted in our country, are suggested.

Key Words Powder metallurgy Turbines Discs Application Lifetimes

TECHNICAL NOTES

RESEARCH ON BLADE EXCITATION FREQUENCY DURING ROTATING STALL OF AXIAL FLOW COMPRESSOR

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ABSTRACT This paper presents an investigation of blade excitation during rotating stall in a single stage axial flow compressor with various geometrical parameters, such as blade stagger and cascade solidity at the mean radius varying from 32.9 to 47.9 and from 0.583 to 1.165, respectively, and the axial distance between rotor and stator from b to $5.7b$. The blade vibration was measured with strain gages on the blade. During rotating stall as the cause of blade excitation, the flow field was simultaneously determined by fast response dynamic pressure transducers. The rotational speed of the compressor varied from 8130 to 12500 rpm. The characteristics of unsteady flow during rotating stall were studied in detail. The prediction criterion of the blade excitation frequency during rotating stall has been set up. The agreement between theoretical and experimental results indicates that the criterion is feasible.

Key Words Rotation stall Blade vibration Frequency Prediction

INVESTIGATION OF OUTLET SECONDARY FLOWS IN CASCADES WITH END-BEND BLADES

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ABSTRACT The outlet secondary flows in compressor cascades with end-bend blades have been studied experimentally and numerically. The viscous secondary vorticity e-

equations are applied to prediction. The experimental and predicted data of the outlet angle due to the secondary flow are presented for three cascades with end-bend blades and one with the corresponding unbended blades. The investigation shows that the change in outlet angle can be accurately predicted by the presented method. Reducing curvature of the blade rear part makes the averaged intensity of the stream line component of the outlet secondary vorticity decrease, but does not change the distance from the center of the vorticity to the end wall.

Key Words Blades Secondary flow Flow fields

*) A MODEL TO CORRELATE FLOW SEPARATION WITH FLOW LOSS IN A TRANSONIC ROTOR

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ABSTRACT A model correlating flow separation with flow loss is proposed for predicting the incipient location and size of the flow separation in a transonic rotor. The flow fields of several rotors have been investigated with the model. The results show that the flow separation mainly occurs in the tip area owing to strong shock boundary interaction, and the incipient separation point is behind the shock location. The tip leakage flow vortex has a strong effect on the interaction between shock wave and boundary layer.

Key Words Flow separation Losses Rotors

A PRELIMINARY INVESTIGATION ON CONTROLLING JET STRUCTURE BY HYDRODYNAMIC EXCITATION

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ABSTRACT Jets were excited by Kaman Vortex Streets under three conditions of Jet Mach Number 0.206, 0.361 and 0.831. Three vortex generators with different geometries were tested. Jet Velocity distributions with and without excitation were measured by stagnation pressure probe. The turbulent characteristics in the jet boundary layer were detected by a single hot wire probe. It is shown that excited jet velocity decay faster than that of unexcited jet, and turbulent energy in jet boundary layer is significantly increased by vortex excitation. Therefore, jet structure can be effectively controlled by hydrodynamic excitations.

Key Words Excitation Jets Boundary layer Kaman vortex Control

A STUDY ON SUPERSONIC 6- NOZZLE EJECTOR PERFORMANCE

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ABSTRACT The supersonic 6- nozzle ejector performance has been investigated experimentally for the given geometric dimensions. The experiment ranges over $1.98 \leq P_p^*/P_s^* \leq 7.46$, $0.971 \leq T_p^*/T_s^* \leq 1.02$ and $L_m/D_m = 0.80$. The entrainment ratio x varies from 6.96 to 11.66 for the ejector with a diffuser, and its corrected value $x \sqrt{T_s^*/T_p^*}$ varies from 6.88 to 11.62; on the other hand, the entrainment ratio varies from 2.82 to 6.28 for the ejector without the diffuser, and its corrected value varies from 2.78 to 6.24. The experimental results show that the entrainment ratio and its corrected value decrease with the increase