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多级轴流压气机端壁边界层中叶片力亏损的研究

西北工业大学　刘松龄

摘　要

本文基于对有径向间隙和无径向间隙两种情况下叶栅二次流造成边界层内叶片力的偏转的分析，建立了计算叶片力亏损的模型。文中给出了由端壁边界层叶片力亏损计算压气机效率的方法。用上述方法对两台轴流压气机和一个进口导向叶片排计算了端壁边界层厚度，得到了环面堵塞系数，计算了一台压气机的效率。对计算结果的分析表明，用这种方法可以得到较合理的计算结果。

一、数 学 模 型

可压缩流端壁边界层动量积分方程为[1]：

$$\frac{1}{\rho_e r}\frac{d}{dm}(\rho_e r V^2{}_{me}\theta_m) + H\theta_m V_{me}\frac{dV_{me}}{d_m} = D'_m + \frac{\tau_{wm}}{\rho_e} \tag{1}$$

$$\frac{1}{\rho_e r}\frac{d}{dm}(\rho_e r V_{me} V_{ue}\theta_u) + H\theta_m V_{me}\frac{dW_{ue}}{d_m} = D + \frac{\tau_{wu}}{\rho_e} \tag{2}$$

m代表子午面端壁方向，u代表周向，r代表径向e为边界层外缘，V和W分别为绝对速度和相对速度，式中各项为：

$$D_m = \frac{d}{dm}\left(\frac{1}{2}V_e^2 F_m\right) \tag{3}$$

$$D_u = \frac{d}{dm}\left(\frac{1}{2}V_e^2 F_u\right) \tag{4}$$

$$V_e^2 = V^2{}_{me} + W_{ue}^2 \tag{5}$$

$$\frac{1}{2}V_e^2 F_m = \int_0^\delta \frac{1}{\rho}(f_e - f)_m dy \tag{6}$$

$$\frac{1}{2}V_e{}^2 F_u = \int_0^\delta \frac{1}{\rho}(f_e - f)_u dy \tag{7}$$

式中f是叶片力，F是叶片力亏损厚度，D_m、D_u分别是叶片力亏损在m和u方向的分量。f可表示为

$$\frac{\partial fm}{\partial m} = \frac{\Delta p}{s} t_g \beta \tag{8}$$

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$$\frac{\partial f_u}{\partial_m} = -\frac{\Delta p}{s} \tag{9}$$

Δp是同一m值的叶片上下表面压差。边界层动量厚度θ和位移厚度δ^*为：

$$\theta_m = \int_0^\delta \frac{\rho V_m}{\rho_e V_{me}}\left(1 - \frac{V_m}{V_{me}}\right)dy \tag{10}$$

$$\delta_m^* = \int_0^\delta \left(1 - \frac{\rho V_m}{\rho_e V_{me}}\right)dy \tag{11}$$

$$\theta_u = \int_0^\delta \frac{\rho V_m}{\rho_e V_{me}}\left(1 - \frac{W_u}{W_{ue}}\right)dy \tag{12}$$

$$H = \frac{\delta_m^*}{\theta_m} \tag{13}$$

为使方程组封闭，Mellor等提出下列补充方程：

$$V_{me}D_m + (1-\varepsilon)W_{ue}D_u = 0 \tag{14}$$

$$W_{ue2}(\theta_{u2} - \theta_{m2}) = 0.55\frac{t_c}{\cos\lambda}\left|\frac{s}{c}V_e(W_{ue2} - W_{ue1})\right|^{\frac{1}{2}} \mathrm{sgn}(W_{ue2} - W_{ue1}) \tag{15}$$

式中λ为叶片安装角，s为栅距，t_c为径向间隙，c为弦长。在式(14)中$\varepsilon = 0$ 则叶片力亏损$\vec{D}$与主流速度$\vec{V}_e$垂直，Mellor等认为ε是个小量。

文献〔4〕的实验表明，ε并非为零，而是量级为1的量。在主流区可以认为叶片力与气流方向垂直，在端壁边界层内由于二次流的影响，叶片力会有偏转，而叶片力的微小偏转都会造成叶片力亏损显著偏离主流速度的法向。

本文作者曾取$\varepsilon = 0$对一台六级轴流压气机和一台两级风扇的端壁边界层做了计算，发现轮毂边界层厚度增长过快，在后面的级出现了不合理的结果。从文献〔3〕中给出的计算结果也可以看到类似的情况。

本文提出一种估计叶片力亏损的模型，其基本思想是：

（1）端壁边界层内流速方向的变化引起叶片力方向的改变，造成边界层内流速方向改变的因素是叶栅通道内的二次流和尖部间隙的漏气，这两个因素造成流速方向变化的趋势相反，前者造成过转折，后者造成欠转折（见图1）。过转折使边界层内叶片力的方向如图2a所示方向偏转，叶片力亏损与主流速度间的夹角为锐角，欠转折时叶片力的方向如图2b 所示，叶片力亏损与主流速度间的夹角为钝角。在有径向间隙时，上述两种因素同时存在。文献

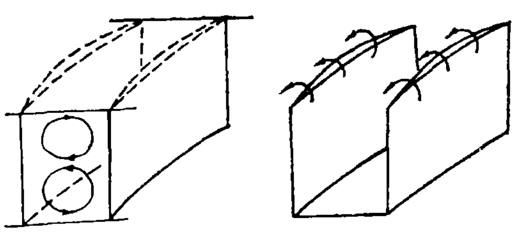

图 1　叶栅中的二次流和漏气

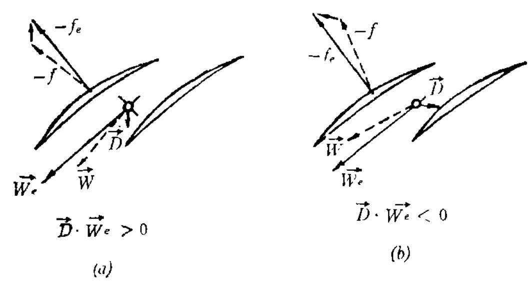

图 2　边界层内的叶片力亏损

〔5〕的研究表明，叶片排出口截面与进口截面之差Δ与叶高h之比Δ/h为0.017时后者的影响较前者大得多，甚至可以忽略前者的影响，Δ/h的上述数值和一般压气机内径向间隙值相当，因此在目前的径向间隙水平下，不妨假设有间隙时可以略去其他二次流对叶片力亏损方向的影响。目前正在发展的主动间隙控制技术可以大幅度减小间隙值，在这种情况下上述假设能否成立则有待进一步研究。

（2）边界层中叶片力的亏损使得叶片在端壁区要额外多做功，这部分功大体上就是叶栅中的二次损失功，可以假设叶片力亏损与主流速度的点积和叶栅二次流引起的阻力的做功率成正比，即：

$$\rho_e \vec{D}\cdot\vec{V}_e = \pm K_D \frac{1}{2} C_D \rho_e V_e{}^3 \frac{h}{2s}\frac{1}{\cos\lambda} \tag{16}$$

C_D是二次流阻力系数，K_D是待定常数其数值应当是1的量级，实际计算表明K_D的取值范围大约是1.0—1.5。由式(1)知，$\vec{D}$的定义是端壁上单位面积上的叶片力亏损，将叶片上的二次流阻力分别折算到上下端壁便得到式(16)。

阻力系数C_D由两部分组成：

$$C_D = C_{Ds} + C_{Dt} \tag{17}$$

C_{Ds}是无间隙时二次流阻力系数，C_{Dt}是径向间隙漏气造成的阻力系数。文献〔2〕和〔5〕分别建议用：

$$C_{Ds} = 0.01 C_L^2 \frac{C}{h} \tag{18}$$

$$C_{Dt} = 0.7 C_L^2 \frac{t_c}{h}\frac{c}{s} \tag{19}$$

不难看出，在叶片负荷很轻即C_L值很小时，式(16)和(14)是一致的。在静叶根部，径向间隙的结构与转子叶片尖部不同，此时仍可用式(16)，只是K_D的取值可略有不同。

二、端壁边界层与压气机效率的关系

由端壁边界层厚度和叶片力亏损厚度可以估计对应于端壁损失的压气机效率（不包括端壁区以外的损失），实际上也就是压气机二次损失对应的效率。转子的做功率$\dot{W}$可表示为：

$$\frac{\dot{W}}{2\pi} = \triangle\int_{r_h}^{r_t} \rho V z H^* r dr$$
$$= \int_{r_h}^{r_t} f_u u r dr + \frac{\dot{w}_f}{2\pi} \tag{20}$$

其中$\dot{w}_f$是根部摩擦阻力的功率，f_u是周向叶片力，H^*为滞止焓，z表示轴向。

在主流区的计算中，为考虑端壁边界层的堵塞作用，可认为经过一个叶排时气体流量有变化。由质量守恒关系可写出加入叶排的质量流量为：

$$\sum_{h,t}(\rho V n)\Delta m = \sum_{h,t}\Delta(\rho_e V m e \delta^*) \tag{21}$$

Vn是假想的沿壁面法向的速度。由此无粘流能量方程为：

$$\triangle\int_{r_h}^{r_t}\rho_e V z_e H e^* r dr - \sum_{h,t} r H_e^*(\rho_e V_{me}\delta^*) = \int_{r_h}^{r_t} f_{ue} u r dr \tag{22}$$

考虑到边界层很薄可以写出：

$$\int_{r_h}^{r_t}\rho_e Vze\, He^* rdr \doteq \int_{r_h+\delta_h^*}^{r_t-\delta_t^*}\rho_e Vz_e H_e^* rdr + \sum_{h,t}\rho_e V_{me} He^* r\delta^* \tag{23}$$

由式（20）（22）得：

$$\triangle\int_{r_h}^{r_t}\rho V_z H^* rdr = \triangle\int_{r_h}^{r_t}\rho_e Vz_e H_e^* rdr + \int_{r_h}^{r_t}(f_u - f_{ue})urdr - \sum_{h,t} rH_e^*\Delta(\rho_e V_m e\delta^*) + \frac{\dot{W}_f}{2\pi} \tag{24}$$

由式（7）有

$$\int_{r_h}^{r_t}(f_u - f_{ue})urdr = -\sum_{h,t}\frac{1}{2}\rho_e V_e^2 urF_u \tag{25}$$

由以上诸式可写出：

$$\eta_s = \frac{\Delta\int_{r_h+\delta_h^*}^{r_t-\delta_t^*}\rho_e V_{ze} H_e^* rdr}{\Delta\int_{r_h}^{r_t}\rho V_z H^* rdr}$$

$$= \frac{\Delta\int_{r_h}^{r_t}\rho_e V_{ze} H_e^* rdr - \Delta\sum_{h,t}\rho_e V_m e H_e^* r\delta^*}{\Delta\int_{r_h}^{r_t}\rho_e V_{ze} H_e^* rdr - \sum_{h,t} rH_e^*\Delta(\rho_e V_{me}\delta^*) - \sum_{h,t}\frac{1}{2}\rho_e V_e^2 urF_u + \frac{\dot{W}_t}{2\pi}} \tag{26}$$

其中

$$\frac{\dot{W}_f}{2\pi} = (\tau_{wu} ur)_h \Delta m \tag{27}$$

τ_w是壁面摩擦应力，可按文献〔1〕的方法计算，τ_{wu}是τ_w沿周向的分量。

三、计算结果和分析

应用式（1）（2）(15)(16)和文献〔1〕的求解方法，可求出θm、θu、Dm(Fm)和Du(Fu)。在求解时取边界层形状因子H为常数1.4，也可以再补充一个方程（例如掺混方程）以求出形状因子的变化，文献〔3〕曾用此方法做过计算，得到的边界层厚度和取H为常数相差不大，从文献〔4〕的实验数据看，H值在各叶排内交替增减，每一个叶排H的平均值与上述数值相差不多。本文的主要目的是探讨边界层内叶片力亏损问题，故取H为常数。

1. 六级双轴轴流压气机

图3表示计算得到的轮毂和机匣上边界层动量厚度θm的变化情况，在轮毂除第一级转子外其余各级转子内θm均增大，在静子内则相反。在机匣除第一、二级外其余各叶排内θm的变化规律与轮毂相反。

图4、5表示叶片力亏损厚度在各叶片排的变化情况，在转子的轮毂和机匣周向叶片力亏损大部分为正，表示在端壁区叶片对气流要多做些功，同时也产生了端壁损失，这个结果是符合实际的。转子和静子轴向叶片力亏损在轮毂和机匣方向大体相反。计算结果还表明，叶

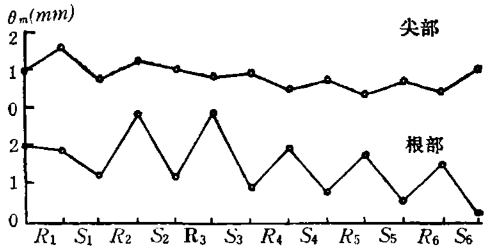

图 3　边界层动量厚度θm

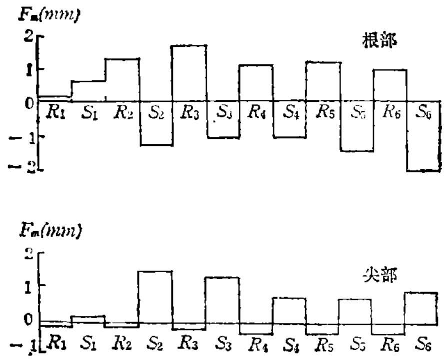

图 4　子午面端壁方向叶片力亏损厚度

片力亏损值比端壁摩擦阻力大得多。

图6表示由边界层位移厚度求出的环面堵塞系数K_G，沿轴向堵塞系数并非单调增长，在转子出口减小，静子出口增大，后三级进出口堵塞系数几乎不变。

图7表示计算得到的对应于端壁损失的压气机效率，显然后面级的端壁损失所占比例比前面级大得多。

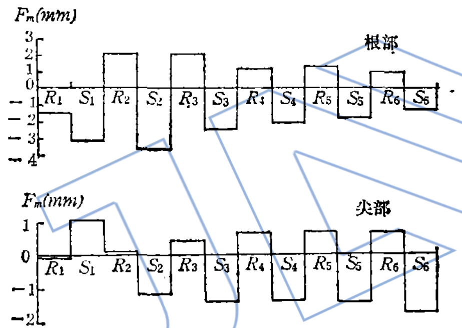

图 5　周向叶片力亏损厚度

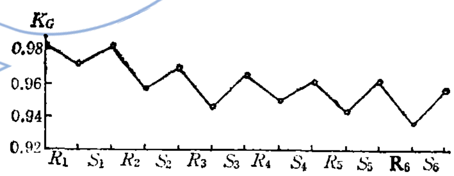

图 6　环面堵塞系数

2. 二级风扇

文献〔7〕给出二级风扇的气动参数和叶片几何参数，用本文所述方法计算所得的边界层动量厚度和叶片力亏损厚度在各叶片排中的变化趋势和六级轴流压气机相似。图8给出了计

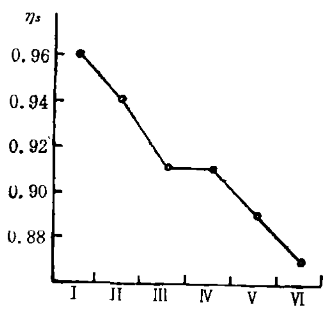

图 7　压气机效率

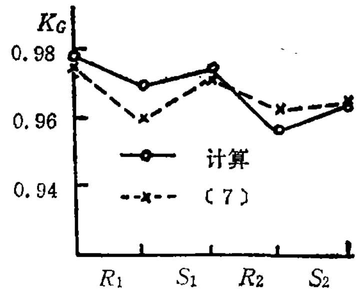

图 8　环面堵塞系数

算的环面堵塞系数和文献〔7〕中给出的设计该风扇时选择的值，第一级转子有防振凸台，因而计算值较设计值偏高，在其他位置上计算值与设计值均符合良好。

3． 进口导向叶片排

文献〔6〕给出该叶栅的气流参数和几何参数，还给出了实验测定的进出口边界层动量厚度，进口子午方向动量厚度为0.45mm，出口为0.90mm。计算的出口子午方向动量厚度为0.92mm，符合程度也是令人满意的。

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characteristics of three types of aeroengine have been analyzed with this program and the results are satisfactory.

A CALCULATION OF A ROTOR INTEGRAL STRENGTH WITH FINITE ELEMENT METHOD FOR FLEXIBILITY

Sheng Huiyu, Sun Botao, Li Fei, Niu Xiuyun

(*Marine Boiler anb Turbine Research Institute*)

Abstract

A calculation method is presented to determine the integral strength of a complex rotor with both radial interference and axial pretension. The method is characterized by followings: calculating the flexibilities and free deformations at the connection nodal points of rotor substructures with finite element method at first; determining the inner forces at these points then; obtaining the deformations and stresses of the individual rotor substructures at last. The method is verified by using a conventional finite element method for a modal stage in good agreement. The present method has been applied to the final design of a disk-drum detachable rotor with bidirectional interferences, which consists of profile wheels, annulus spacers and conical shafts. The rotor is designed for a six-stage compressor and is divided into 2504 annular elements of triangular type by 2020 nodal points. The rotor integral strength calculation is performed on a small computer of only 16K memories. The rotor deformation and iso-stress line diagrams are drawn by computer. The mechanical relationship of the rotor with its individual parts is quite clear.

INVESTIGATION ON BLADE FORCE DEFECT IN END WALL BOUNDARY LAYER FOR MULTISTAGE AXIAL COMPRESSOR

Liu Songling

(*Northwestern Polytechnical University*)

Abstract

In order to predict the blade force defect a new model is proposed which is based on the analysis of the deflection of the blade force in boundary layer due to the secondary flows in cascade both with and without tip clearance. It is assumed that the scalar product of blade force defect and mainstream velocity is proportional to the work done by the cascade secondary flow drag and can be expressed as equation(16) in the present paper. This equation approaches to the relation suggested by Mellor et al. as the lift coefficient value of the blade is very low. A method using the values of blade force defect in boundary layer is presented to estimate the compressor efficiency. The end wall boundary layer thicknesses are calculated and the blockage in meridional plane is obtained for two axial compressors and an inlet guide vane. The efficiency of a compressor is also estimated. The analysis indicates that reasonable results can be obtained by the use of this method.

SECONDARY FLOW LOSSES IN PLANE TURBINE CASCADES AND ITS MATHEMATICAL PREDICTION MODEL

Li Caixiu and Ni Zhong

(*Shanghai power Equipment Research Institute*)

Abstract

The phenomenon and cause of the secondary flow in plane turbine cascades are discussed and a mathematical prediction model of losses is presented. It is stressed that the secondary flow in turbine cascades is a dynamic but not a kinematic effect and thus should be studied from the viewpoint of dynamics. Based on the results of plane cascade tests, a model that predicts the secondary flow loss distribution is suggested. The effect of inlet boundary layer has been considered in the model. Precision obtained from the proposed model conforms more satisfactorily with experimental results than those given in previous literatures.

A STUDY ON DEPLOYMENT RULE OF SINGLE EXTENDIBLE EXIT CONE BY MASSCENTER MOVEMENT THEOREM

Ma Guobao

(*Shanxi Power machinery Institute*)

Abstract

The deployment rule of the single extendible exit cone is investigated in this paper. The cone is divided into four pieces in the direction of meridian. As a result, the actuators are just located in the middle of each piece. Each piece of the cone can thus be simplified into a moving mechanism. Based on the force balance between the mechanisms, a second ordes ordinary differential equation is derived by applying the mass center movement theorem. The numerical solution of the equation can be obtained by means of the fourth order Runge-Kutte method. The time-dependent parameters, such as position, velocity and acceleration of the external sleeve of the actuator relative to the internal tube, and the time needed for fully deploying the cone can then be obtained by using other related equations.