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Performance Prediction of Axial Flow Turbomachines Using a Modified One Dimensional Method

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Abstract— A computer code is presented for off-design performance prediction of axial-flow compressors and fans. Stage and compressor performance is obtained by a stage-stacking method that uses representative velocity diagrams at rotor inlet and outlet mean line radius. The code has options for: (1) direct user input or calculation of non-dimensional stage characteristics; (2) adjustment of stage characteristics for off-design speed; (3) calculation of different source of loss in design and off-design condition. New blading concepts as used in modern transonic axial-flow compressors require improved loss and deviation angle correlations. The new model presented in this paper incorporates several elements and treats blade-row flows have both subsonic and transonic inlet conditions .Correlations from experimental data are used to model real flow conditions. Then Calculations are compared with experimental data of two cases. Although this type of flow analysis is a gross simplification of a complex three-dimensional system, which can now be modeled more accurately by many of today's computational fluid dynamics (CFD) packages, it does offer the advantages of simple input requirements and fast convergence time.

Index Terms— Axial Fan, Stage Stacking, Performance, Loss, Off Design, Characteristics, Pressure Ratio, Efficiency.

1 Introduction

Several research in one dimensional methods in turbomachinery performance analysis were done up to now. In the jet engine application, the compressor faces a wide variety of operating conditions. On the ground at takeoff phase, the inlet pressure is high, inlet speed is zero, and the compressor is spun at varying speeds to generate the required pressure rise and mass flow rate and also in industrial gas turbine application operating condition must be variable to produces variety demanded load as well. Once in flight the inlet pressure drops, but the inlet speed increases (due to the forward motion of the aircraft) to recover some of this pressure, the compressor tends to run at a single speed for long periods of time. Clearly, this large variation in both rotor RPM and mass flow rate needs to be considered while designing an axial flow compressor. On the other hand, a gas turbine works under part load conditions most of the time. In this regard, performance prediction of turbomachineries is of interests. Performance prediction of turbomachines can be done by some famous methods in which most of them are three dimensional and require time and state of the art hardware to solve. So, one-dimensional methods called stage stacking have been always considered as the promising and fast methods for early analysis to find turbomachineries characteristics.

A.R. Howell et al. in 1978 [1] introduced a new stage stacking technique for axial flow performance prediction and also Ronald J. Steinke at NASA Lewis research center in 1982 [2] developed a basic code for stage stacking process. All the researches were done were based on numbers of correlations which were revealed in the relevant time. Tonye K. Jack et al.[3] in 2012 introduced a modified method for multistage axial flow compressor calculation with variable stator stagger setting angle. Kesey, developed a method for calculating axial flow compressor performance with a repeating stage model. He used one dimentional analysis method for performance prediction of industrial compressors [4]. Chen et al. [5] considered one- dimensional modeling to predict a single-stage axial flow compressor performance in which absolute flow angles at input and output of rotor may vary and also optimized the compressor efficiency.

To describe the stage-stacking method, first the flow assumptions are discussed and then the stage characteristics and the stacking procedure will be explained. One-dimensional compressible flow is assumed in stage stacking method and Flow continuity equation can therefore be expressed as $\dot{m} = \rho c_a A$ where A is the annulus area. Mass flow continuity is satisfied with axial velocity of c_a at rotor inlet and outlet axial direction of each stage. Thus, for a given mass flow rate, rotational speed and stage inlet flow conditions of total pressure, temperature and absolute inlet flow angle "α_1", mean line velocity diagram can be obtained at the rotor inlet. It's obvious that mean line velocity diagram can be obtained at the rotor exit by assuming that the stage overall pressure ratio and adiabatic efficiency are applied at the rotor exit. If the total pressure and temperature at the rotor exit are assumed as the same as for the next rotor inlet, the mean line velocity diagram can be obtained at rotor inlet and exit using the overall stage performance parameters of pressure ratio "Pr" and adiabatic efficiency "η_D".

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2 Stage Stacking Process and Loss Calculation

2.1 Method

The stage performance characteristics consist of three non-dimensional quantities - adiabatic efficiency η_D, pressure coefficient ψ, and flow coefficient φ. Equation (1) ,(2) and (3) show their relations respectively [6].

$$\phi = \frac{c_a}{u} \quad (1)$$

$$\psi = \frac{c_p \eta_{ad} (T_{02} - T_{01})}{u_1^2} \quad (2)$$

$$\eta_{ad} = \frac{[(\Pr)^{\frac{\gamma-1}{\gamma}} - 1]}{\frac{T_{03}}{T_{01}} - 1} \quad (3)$$

At first, the mentioned three parameters are calculated for all stages at design speed and then incidence and deviation angles for rotor and stator blades in each stage at design speed are determined. Lieblein [7] developed a simple analytical relation between the blade-wake momentum thickness and the blade surface-velocity diffusion for conventional cascade blades which became a basis for most research in correlating losses and deviations with diffusion factor. He also presented experimental evidence to show as the blade camber or blade angle of attack is increased, large velocity gradients will occur on the blade suction surface, but comparatively small changes occur on the pressure surface. Therefore, changes in total wake momentum thickness caused primarily by the diffusion contribution of the suction-surface boundary layer. Equation (4) is the correlation used for calculating the reference incidence angle.

$$\begin{aligned} i^*_{2D} &= k_{sh} k_{it} i_{010} + n\theta(i_{010}, \theta) \\ k_{it} &= f(t/c) \\ i_{010} &= f(\sigma/\beta_1) \\ n &= f(\sigma/\beta_1) \end{aligned} \quad (4)$$

Blade shape factor is a correction which varies for each type of blades. The relevant values are listed in table 1. " i_{010} " is defined as the incidence angle based on 10% blades thickness. The variable n represents the incidence slope factor. k_{sh}, k_{it} are the two correction factors for blade shape and thickness respectively[6].

Table 1 : Shape factor for the calculation of incidence angle

Blade type	K_{sh}
DCA	0.7
65 series	1
C series	1.1

Diffusion factor is calculated based on the maximum suction surface velocity to the inlet or exit velocity ratio. The equivalent diffusion factor "D_{eq}" is defined by (5).

$$D_{eq} = \frac{v_{max}}{v_2} = \frac{v_{max}}{v_1} \times \frac{v_1}{v_2} \quad (5)$$

Daini [8], presented a correlation to calculate the momentum thickness to predict losses. "R" is blade height percentage along span of blade element. Here mean line radius and the momentum thickness are applied as indicated in (6) and (7) respectively.

$$\overline{R} = (1 - [\frac{R_t - R_m}{R_t - R_h}]) * 100 \quad (6)$$

$$(\frac{\theta}{c})^* = (-1.0312 + 0.01722\overline{R}) + (1.396 - 0.0244\overline{R})\, D_{eq} + (-0.467 + 0.00866\overline{R})\, D_{eq}^{\ 2} \quad (7)$$

Two-dimensional profile cascade losses arise primarily from the growth of the boundary layer on the suction and pressure side of the blade. Theoretical analysis shows that the relationship between the wake momentum-defect thickness $(\frac{\theta}{c})$ and the total pressure loss ω^* for conventional un-stalled blade profiles is given approximately by the correlation stated in (8).

$$\omega^* = 2(\frac{\theta}{c})(\frac{\sigma}{\cos\beta_2})(\frac{\cos\beta_1}{\cos\beta_2})^2 \quad (8)$$

Using (8) leads to obtain the value of ω^* at design condition. This equation becomes more doubtful at off-design conditions. If the value of the inlet Mach number is higher than the critical Mach number value, a correction suggested by AI-Daini [8] is applied which is defined as (9).

$$\omega^* = \omega^*[(2M'_1 - M'_{1C}) + 1] \quad (9)$$

2.2 Loss Correlation at Off-Design

The first step is to evaluate an equivalent diffusion ratio at off-design. Eq. 10 shows the equivalent diffusion ratio by AI-Daini [8].

$$D_{eq} = [k_1 + a(i - i^*)^{1.43} + (k_2 \frac{\cos\beta_2}{\sigma}) * (\tan\beta_1 - \frac{V_{a2}}{V_{a1}} * \tan\beta_2)] * (\frac{V_{a1}}{V_{a2}})(\frac{\cos\beta_2}{\cos\beta_1}) \quad (10)$$

The terms "k1" and "k2" indicate the blade shape factor which can be obtained by

$$k_1 = 1.03 + 0.7\frac{t}{c} \quad (11)$$

$$k_2 = 0.4 + \frac{t}{c} \quad (12)$$

The term "a" is a constant which depends on blade profile type as:

a =0.0117 for 65-series blade profile;
a =0.007 for C-series and DCA profiles;

There is a correlation for the off-design deviation angle as a function of $(D_{eq} - D^*_{eq})$ and inlet Mach number. Equation (13) based on experimental data, indicates a relation between

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equivalent diffusion ratio, relative Mach number and deviation angle at design and off-design conditions.

$$\delta - \delta^* = [6.4 - 9.45(M_1' - 0.6)](D_{eq} - D^*_{eq}) \qquad (13)$$

Since D_{eq} is a function of β_2, and on the other hand β_2 is a function of δ deviation, the above two equations have to be solved iteratively for the values of δ and D_{eq}. So, the wake momentum thickness in off-design is given as a function of $(D_{eq} - D^*_{eq})$ and the relative inlet Mach number as in (14).

If $(D_{eq} > D^*_{eq})$ then

$$(\frac{\theta}{c}) - (\frac{\theta}{c})^* = [0.827M_1' - 2.692M_1'^2 - 2.675M_1'^3] \times (D_{eq} - D_{eq}^*)^2 \qquad (14)$$

If $(D_{eq} < D^*_{eq})$ then

$$(\frac{\theta}{c}) - (\frac{\theta}{c})^* = [2.8M_1' - 8.71M_1'^2 - 9.36M_1'^3] \times (D_{eq} - D_{eq}^*)^2$$

The off-design loss parameter "ω" can be evaluated using Eq (8). After the loss calculation and obtaining the results from the stage stacking method which include pressure ratio, incidence and deviation angles and velocity triangles at stages inlet and outlet the compressor characteristic curves are calculated, which indicate the overall compressor pressure ratio and efficiency versus mass flow rate.

3 Validation

Here, to verify the prepared code two comparisons are of interests. First, the two stage axial flow fan of NASA Lewis Research Center [2] and second, applying the code with the three stage axial flow compressors titled "NASA 74-A"[9].

The two-stage axial flow fan results are used to validate the prepared stage stacking code. The test was established in 1982 in NASA Lewis Research Center [2]. The general information about this fan is tabulated in table 2.

Table 2: Input data of the two-stage axial fan [2]

Number of stage	2
Number of speed lines	5
Considered RPM	100% 90% 80% 70% 50%
Computational points	8
Inlet pressure	101.325 [kpa]
Inlet temperature	288 [k]
Design rotational speed	16042 [RPM]
Design Mass flow rate	33.248 [kg / s]
Design pressure ratio	2.4

The results of the prepared code for the two stage axial flow fan have been compared with the test data. The first four graphs of Fig. 1, Fig.2, Fig.3 and Fig. 4, show the characteristic of first and second stage of axial flow fan respectively and the Fig. 5 and Fig. 6 show the overall fan characteristic which compared with the test results of reference 3. Bold lines indicate the analytical results and dotted lines show the test results points which are fitted with a curve approximately. Acceptable agreement between the results from the code analysis and the experimental results is observed according to Fig. 5 and Fig. 6 which indicate compressor characteristics. This agreement is very good at 80%, 70% and 50% RPM of the design speed. The circle points in the last two Fig.s on design speed curve show the predicted design point of compressor.

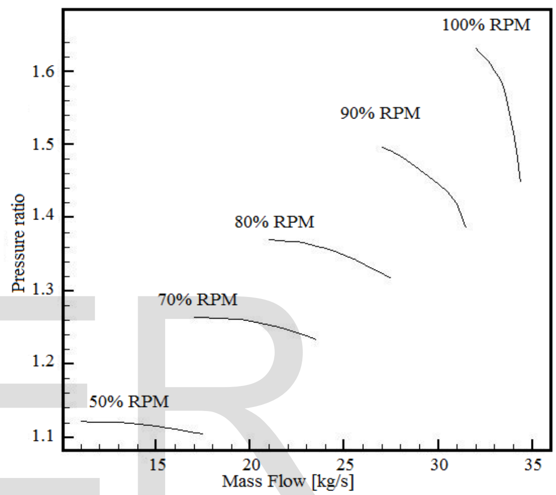

Fig. 1. Predicted pressure ratio characteristics for the first stage of the fan

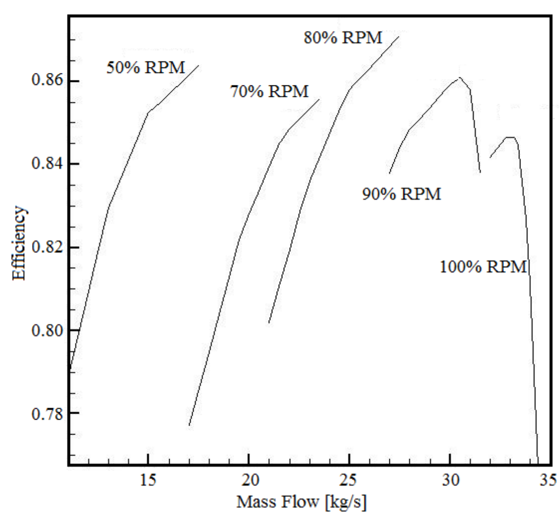

Fig. 2. Predicted efficiency characteristics for the first stage of the fan

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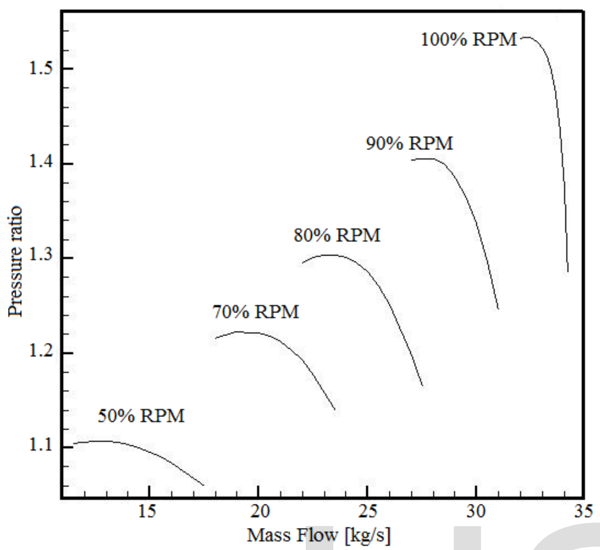

Fig.3. Predicted pressure ratio characteristics for the second stage of the fan

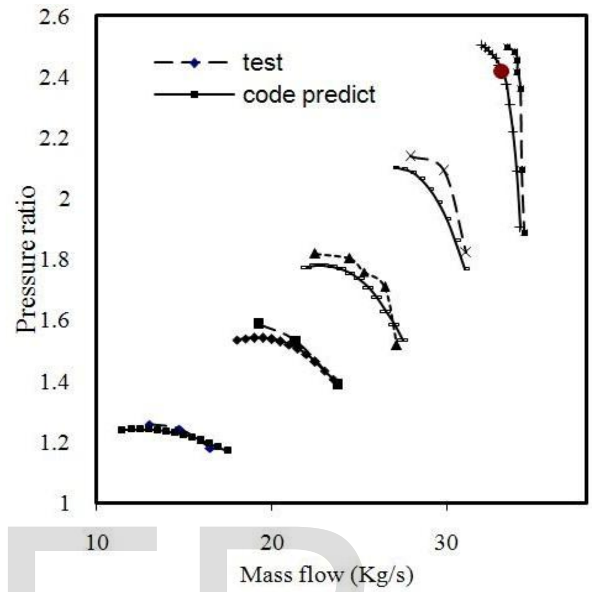

Fig. 5: Overall Pressure ratio characteristics

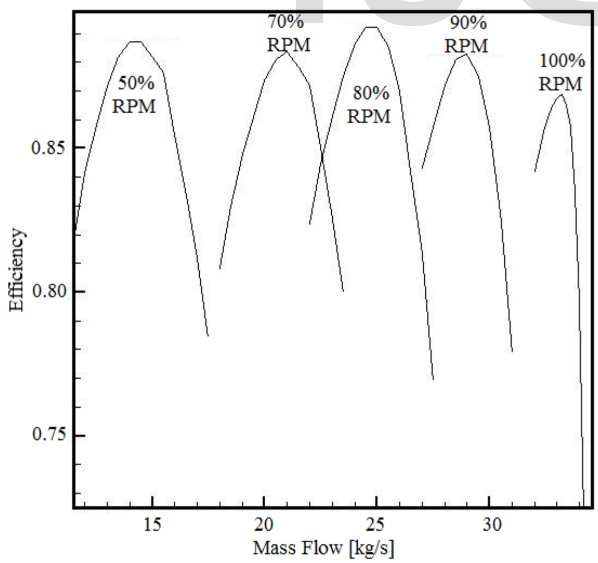

Fig. 4: Predicted Efficiency versus mass flow rate (kg/s) of second stage fan

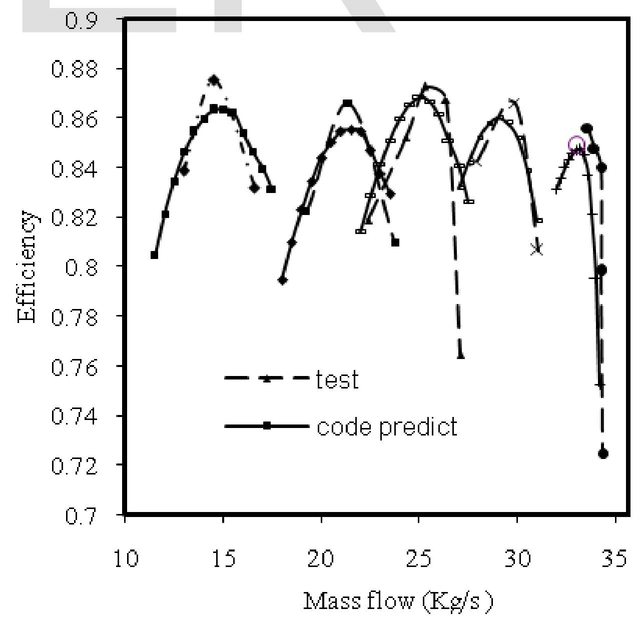

Fig. 6: Overall Efficiency characteristics

On the other hand, the code was applied for the three stage axial flow compressor. The compressor overall pressure ratio and its efficiency are 3.94 and 85.4% respectively. Mass flow rate is 29.71 kg/sec and blade profiles are of double circular arc type. The rotational speed is 16041 RPM. This type of com-

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pressor is transonic and its blades profile type is multiple circular arcs.

Fig. 7 shows the pressure ratio characteristics in five rotational speeds obtained by the prepared stage stacking code in comparison with NASA 74-A data.

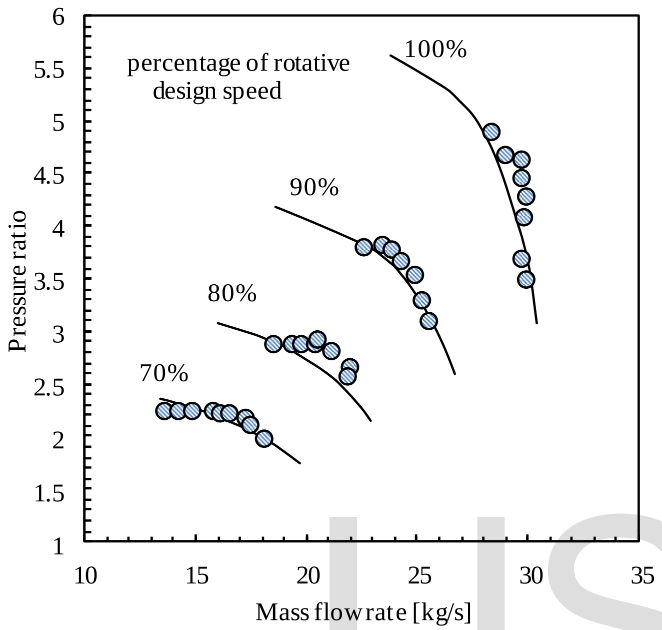

Fig. 7 The three stage axial compressor pressure characteristics and experimental data

6 Conclusion

The aim of this work is to use a suitable loss and deviation model for transonic and subsonic compressors and fans. The blade profiles used in such two stages NASA fan are mainly NACA 65 Series and the experimental data presented are of these profiles. Obviously, there are many factors which would influence losses and deviations such as blade surface roughness, turbulence level, tip clearance effects, secondary flow, annulus loss and etc. which would have an influence on the entire flow. Although the two cases were studied for validations have in agreement with the code approximately, the differences exist between experimental data and predicted values could be due to (a) experimental errors, (b) inadequacy of the prediction procedure and (c) the correlations used which may not match completely with the cases. The method presented for loss and deviation calculation is incorporated into a one dimensional through flow analysis prediction, Ronald J. Steinke [2] method. Calculation of loss, incidence and deviation angle was not available in old version of program. The method would produce accurate predictions of overall performance if the available experimental points are more. Nevertheless, the model presented leads to the following general conclusions:

(a) The estimation of deviation in off-design region for the both fan and compressor considered is good.

(b) The estimation of loss for pressure loss is good for all regions specially for low RPM in the fan.

(c) The estimation of loss needs more experimental correlation in off-design for calculation other source of loss like annulus loss, secondary loss to produce accurate predictions of overall performance specially in high mach number.

Notation

Pr	Pressure ratio
M	Absolute Mach number
c	Blade chord
D_{eq}	Equivalent diffusion ratio
R	Blade height
t	Blade thickness
i	Incidence angle
n	Slope parameter for incidence equation (4)
k_{ti}	Design incidence angle thickness correction factor
k_{sh}	Blade shape parameter
M′	Relative Mach number
i_{010}	Incidence angle for 10% thickness profile
Φ	Flow coefficient
Ψ	Pressure coefficient
ω	Loss parameter
θ	Chamber angle
σ	Solidity
η	Efficiency
δ	Deviation angle
β	Flow angle relative to the blade
α	Absolute flow angle

Subscript

1	Blade inlet
2	Blade outlet
c	Critical value
*	Design value

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