

Numerical evaluation of tandem rotor for highly loaded transonic fan

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Abstract: Transonic tandem rotor was designed for highly loaded fan at a corrected tip speed of 381 m/s and another conventional rotor was designed as a baseline to evaluate the loading superiority of tandem rotor with three-dimensional (3-D) numerical simulation. The aft blade solidity and its impact on total loading level were studied in depth. The result indicates that tandem rotor has potential to achieve higher loading level and attain favorable aerodynamic performance in a wide range of loading coefficient 0.55~0.68, comparing with the conventional rotor which produced a total pressure ratio of 2.0 and loading coefficient of 0.42.

Key words: transonic fan; tandem rotor; highly loaded; loading coefficient; numerical simulation

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Nomenclature

C	chord	s	pitch-wise spacing between forward blade and aft blade for tandem rotor
C_x	axial chord	t	pitch-wise
D	diffusion factor	U	blade tip speed
M	mass flow		

Greek

β	relative flow angle to axial coordinate	σ	solidity
ϕ	loading coefficient, $\Delta h_0 / U_{tip}^2$	π	total pressure ratio
θ	cascade camber angle	η	isentropic efficiency
χ	metal angle relative to axial coordinate		

Subscripts

ov	overall	2	outlet section
1	inlet section		

Introduction

In past decades, it is always a hot research aspect to increase loading level for compressor design, which is helpful to have lightweight and fewer parts count. The main limit to promote loading level originates from separation due to shock/boundary layer interaction in transonic compressor. Tandem blading technology is considered as an effective method to control bounda-

ry layer, therefore, it is helpful to do more work in per stage. The basic concept is that a new boundary layer reforms on the aft blade, allowing for greater overall loading without larger flow separation.

Actually, tandem blading was used to design low-subsonic axial blower in 1950s and 1960s firstly, whose excellent performance indicated that tandem blade could attain higher efficiency and pressure ratio than conventional blade. It is

the very motive to bring tandem blading technology into the high speed compressor design.

In 1970s, NASA sponsored a series of program to evaluate the potential of extending loading limit and stable operating range for tandem rotor including both the transonic and subsonic test, which was conducted by Pratt and Whitney. Burger and Keenan^[1] described the transonic compressor stage test. It was designed with an inlet hub-to-tip ratio of 0.50, a tip speed of 488 m/s, and inlet tip relative Mach number of 1.60. The experiment results indicated that the test rig attained the design total pressure ratio of 1.92, and the stage loading coefficient was 0.30. The subsonic test, which was a single stage of 0.8 hub-to-tip ratio and about 0.8 tip relative inlet Mach number, was summarized by Brent et al.^[2] The loading coefficient and total pressure ratio of this tandem rotor were 0.495 and 1.30 respectively. Experimental results reveal that tandem blading is superior to conventional blading at this loading level.

After that, research mainly focused on subsonic core compressor for middle and rear stage. Bammert et al.^[3-5] tested a multistage compressor consisted of a conventional rotor at first stage and three tandem rotors at rear stage. The average tip speed was 189 m/s, the inlet tip relative Mach number was 0.85, and whole-machine loading coefficient was about 0.77. Compared to conventional blade, the flow loss was reduced approximately 18%. Additionally, Wu et al.^[6], Sachmann et al.^[7], Roy and Saha^[8] also did a great deal of work in low speed condition.

Most recently, Hasegawa et al.^[9] tested a transonic tandem rotor for the air turbo ramjet propulsion system which was analyzed by fully three-dimensional (3-D) viscous solver. The design inlet tip speed is 431 m/s, and tip inlet Mach number is 1.40. The rotor achieved a loading coefficient of 0.51, total pressure ratio of 2.3 and isentropic efficiency of 88%. Surge margin of this tandem rotor was approximately 10%. McGlumphy et al.^[10-12] investigated two-dimensional (2-D) and 3-D tandem airfoils for

subsonic axial compressor rotors basing on computational fluid dynamics (CFD) technology. The 3-D isolated rotor was targeted to produce a loading coefficient of 0.56, whose tip speed is 213 m/s and relative Mach number is 0.62.

The conclusion could be summarized from previous works that subsonic tandem rotor can increase the loading level effectively, but the performance of transonic tandem rotor is somewhat disappointed. The main reason is that the loading coefficient chosen by Burger and Keenan^[1] is not high enough to present loading capacity for transonic tandem rotor and Hasegawa et al.'s work^[9] focused on ramjet turbofan, and the inlet axial velocity is higher than that in typical transonic compressor, which results in low efficiency.

The goal of this paper is to evaluate the feasibility of transonic tandem rotor in fan design for military turbofan engine. The structure of this paper is as follows. The aerodynamic design of transonic tandem rotor and another baseline conventional rotor is summarized firstly. Their performances are compared and analyzed based on CFD results. In addition, the solidity of aft blade in tandem rotor and its influence on overall loading level are investigated either. Lastly, the significant conclusion is given.

1 Comparison of performance between transonic tandem and conventional rotor

1.1 Basic design concept of transonic tandem rotor

Fig. 1 is the illustration of flow difference between the transonic tandem rotor and conventional rotor. Conventional rotor has the single blade but tandem rotor consists of forward blade (FB) and aft blade (AB). In transonic fan, loading increase will enhance the interaction between passage shock wave and the boundary layer on blade surface, and result in strong flow separation. However, in transonic tandem rotor, the forward and aft blade share the overall loading, thereby the loading of transonic for-

ward blade could be chosen appropriately to minimize the shock/boundary layer interaction. The subsonic aft blade could be free from separation and has capacity to do more work efficiently. Consequently, the transonic tandem rotor could obtain higher loading level and pressure ratio without the increasing in excessive loss.

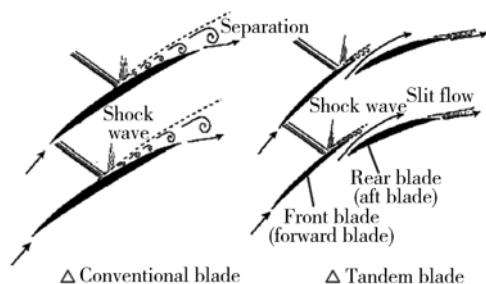


Fig. 1 Comparison of flow characteristic between transonic conventional and tandem rotor^[9]

1.2 Tandem blade geometrical configuration

It's necessary to define the geometrical configuration parameter at first. Tandem rotor consists of two single blades, including FB and AB (Fig. 2). The blade-to-blade geometry parameter definition of single blade is as the same as that of conventional blade, but some overall geometry parameters need to be redefined such as overall solidity and camber angle of tandem blade.

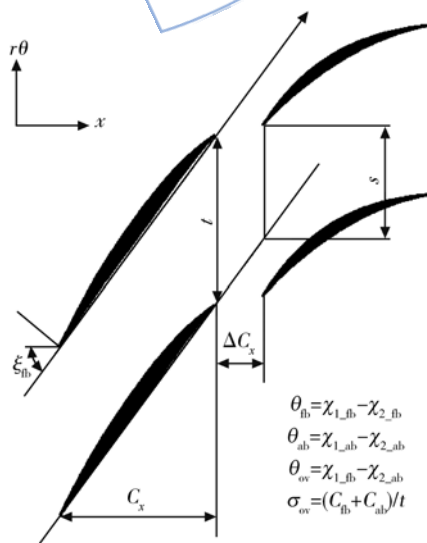


Fig. 2 Tandem geometry configuration

1.3 Design parameters of first tandem rotor and baseline conventional rotor

For most contemporary highly loaded transonic fan rotors, the design loading coefficient ranged from 0.30 to 0.35^[13-17]. Compared to those above, a conventional rotor was designed as a baseline, which produces a total pressure ratio of 2.0 and loading coefficient of 0.45 and is considered as the baseline to estimate the loading superiority of tandem rotor.

The tandem rotor marked A was designed to achieve a design total pressure ratio of 2.2. The loading coefficient is up to 0.55, which is 22% higher than that of conventional rotor (Table 1). The geometry parameters of both conventional and tandem rotor are given in Table 2.

Table 1 Aerodynamic design parameters of conventional and tandem rotor

Design parameters	Conventional rotor	Tandem rotor A
Physical tip speed/(m/s)	442	442
Corrected tip speed/(m/s)	381	381
Axial Mach number	0.5	0.5
$M/(kg/s)$	120	120
π	2.0	2.2
$\eta/\%$	>93.0	>92.0
ψ	0.45	0.55
Surge margin/%	>12	>12

Table 2 Geometry design parameters of conventional and tandem rotor

Design parameters	Conv.	Tandem rotor A		
		FB	AB	OV
Tip solidity	1.70	1.30	0.96	2.31
Tip camber angle/(°)	30.2	7.3	22.3	25.0
Hub solidity	2.14	1.58	1.17	2.81
Hub camber angle/(°)	79.6	24.7	65.7	76.2
Aspect ratio	1.128	1.560	1.889	0.849
Number of blades		37		

The methodology to design transonic tandem rotor is to keep enough tip solidity for FB, which makes sure of passage shock stability and sufficient stall margin. Meanwhile, the choice of solidity for AB mainly depends on the diffusion factor and mechanism structure. Consequently, the overall solidity of transonic tandem rotor would be higher if single conventional rotor were split into forward and aft blade individually. The design of transonic tandem rotor differs from that in subsonic condition just because of the shock existence in forward blade. Compared to conventional rotor, the axial length of tandem rotor was increased slightly to make FB and AB have appropriate solidity they need (Fig. 3). The abbreviation of LE and TE represent the leading and trailing edge respectively.

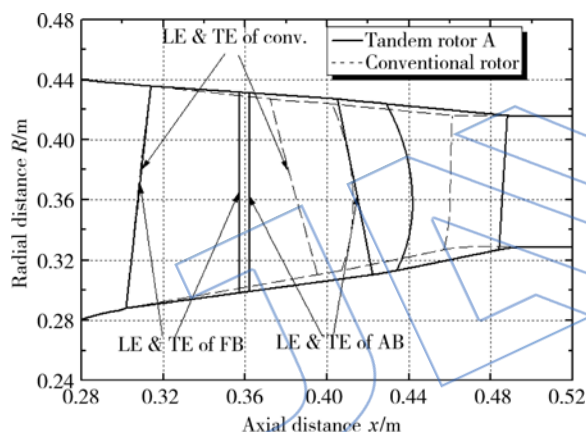


Fig. 3 Comparison of flow path for conventional and tandem rotor

1.4 CFD setup and procedure

The computational H-mesh of tandem rotor was generated integrally, which consist of two circumferential blocks and five blocks in stream-wise direction, shown in Fig. 4, the pressure side (PS) and suction side (SS) of aft blades are marked in the mesh. Approximately 528 thousands of grid points were used to calculate the domain. Stator was not included in simulation.

The CFD solver employed in this paper is Denton code^[18], which was imported by our department in 1990s. Viscous and turbulence effect are included in this code using a thin shear

approximation to the Navier-Stokes equations and an eddy viscosity mixing length model. To make simulation more reliable, Denton code was verified to identify its mixing length limits through comparing with simulation results of commercial CFD solver NUMECA.

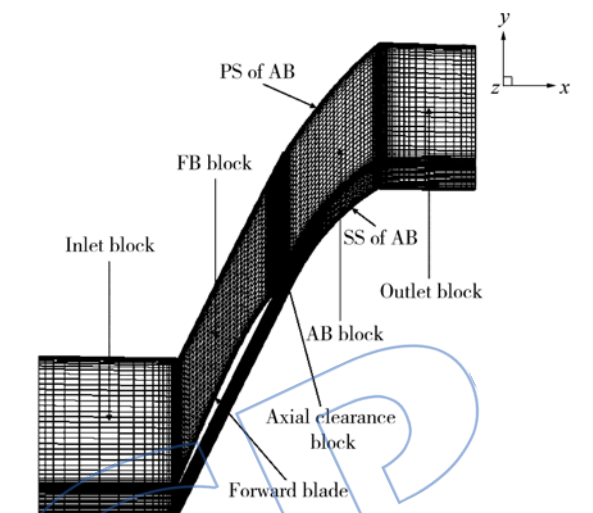


Fig. 4 Tandem rotor CFD mesh

1.5 Comparison of calculation results

The characteristic comparison between the tandem rotor A and conventional rotor is shown in Fig. 5, which indicates that tandem rotor A attains higher total pressure ratio and loading level. Meantime, the efficiency of tandem rotor A is obviously better, and its peak efficiency is approximately 1.2 points higher than that of conventional rotor at design speed. Additionally, the choke mass flow of these two rotors has some discrepancy, probably resulted from the difference of throat area.

Fig. 6 shows the span-wise distribution of outlet metal angle and relative flow angle for baseline conventional rotor and tandem rotor at design point, from which the deviation angle of conventional rotor and tandem rotor A could be deduced and compared. It is apparent that the deviation angle of conventional rotor is larger even though its loading level is lower. The average value of deviation angle for conventional rotor is about 30 degrees, and it is the best illustration to prove that separation is the determina-

tive restriction to promote loading level with traditional design method. Tandem rotor A attained higher loading level but with lower blade camber angle, which means the flow through tandem rotor is more effective and the flow separation as well as huge deviation angle is extensively eliminated by tandem blading especially at near tip section.

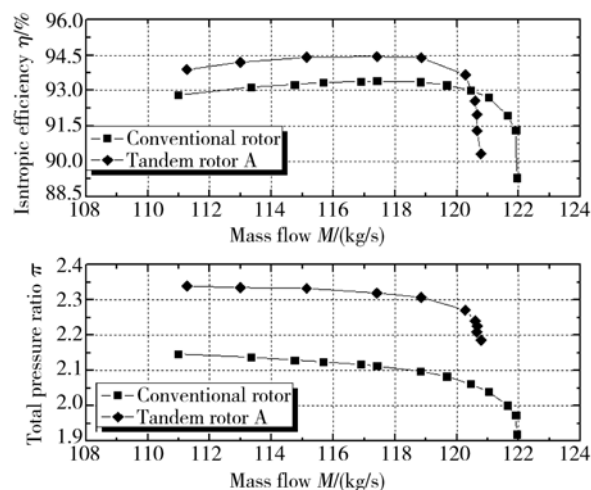


Fig. 5 Comparison of total pressure ratio and isentropic efficiency characteristic for the baseline conventional rotor and tandem rotor A

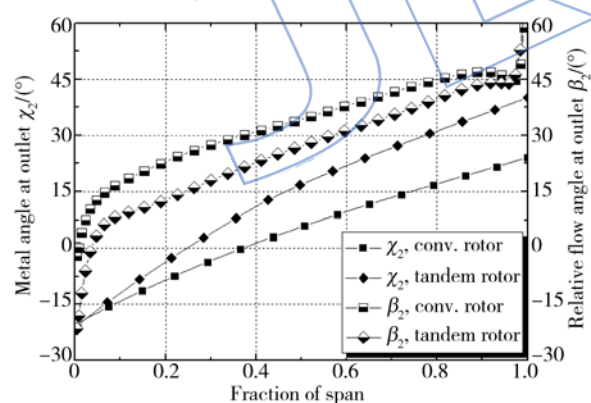


Fig. 6 Span-wise distribution of outlet metal angle and relative flow angle for baseline conventional rotor and tandem rotor at design point

Fig. 7 illustrates the solidity distribution of both rotors in span-wise direction, as well as the diffusion factor at design point (DP) and near stall point (NSP). The solidity of tandem rotor

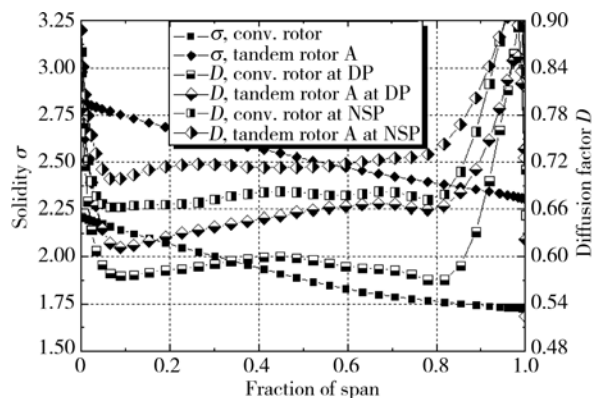


Fig. 7 Span-wise distribution of solidity and diffusion factor for baseline conventional rotor and tandem rotor at design point and near stall point

is higher, which has been explained above this paper. However, the promotion of loading level for tandem rotor A does not only originate from solidity increasing but the breakthrough of aerodynamic limit, which could be represented by diffusion factor. Diffusion factor takes account of aerodynamic loading and solidity. It can be learned from Fig. 7 that the diffusion factor attained by tandem rotor A is much higher at both the design and near stall point due to the larger flow turning angle. Besides, the diffusion factor of tandem rotor A exceeds 0.7 at near stall point, and the value in traditional design is 0.65 generally.

The comparison of blade-to-blade Mach number contours for both the two rotors at design and near stall point are illustrated respectively in Figs. 8 and 9, in which 94%, 60%, and 15% fraction in span-wise direction are chosen as typical sections to reveal the flow field details of tip, mean and hub section.

Double-shock system exists at design point, and pre-shock Mach number is well controlled by using pre-compression at tip for both rotors. The flow separation near trailing edge in conventional rotor, resulted from the excessive sonic diffusion after passage shock, is severe but tolerable, which is the origin of deviation angle shown in Fig. 7.

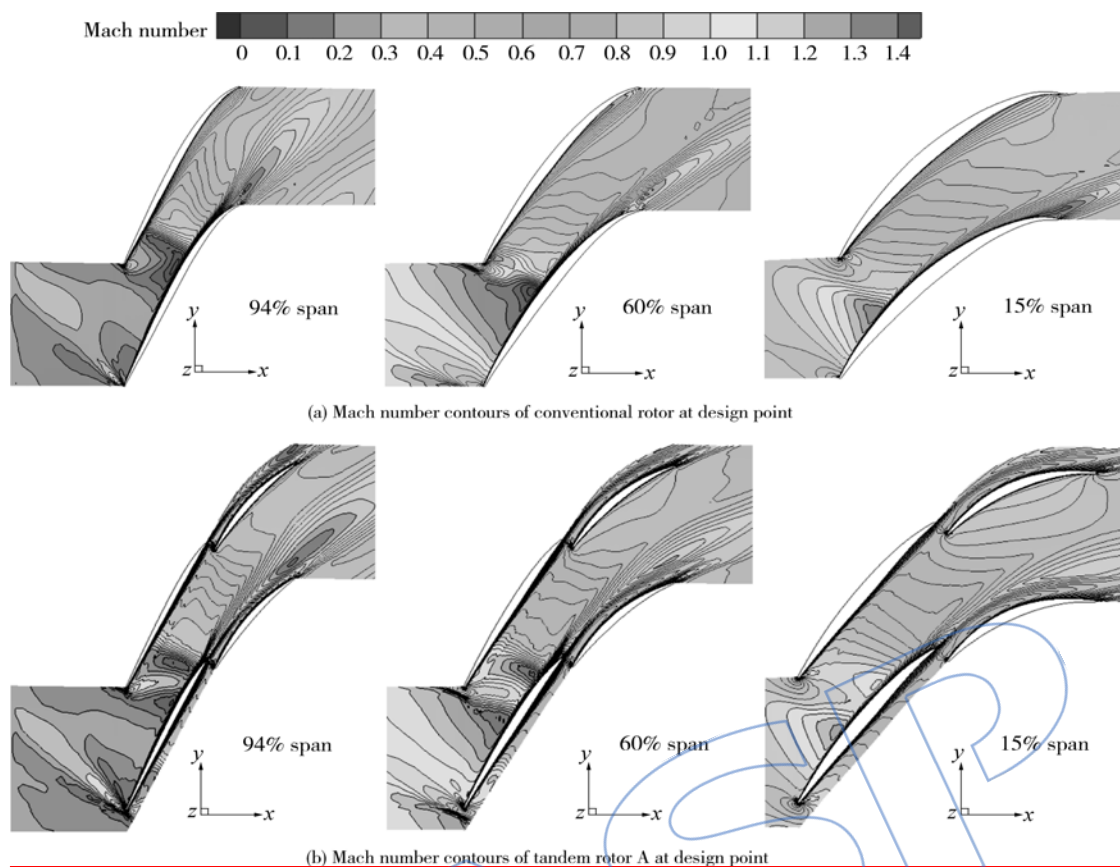


Fig. 8 Comparison of Mach number for conventional rotor and tandem rotor A at design point

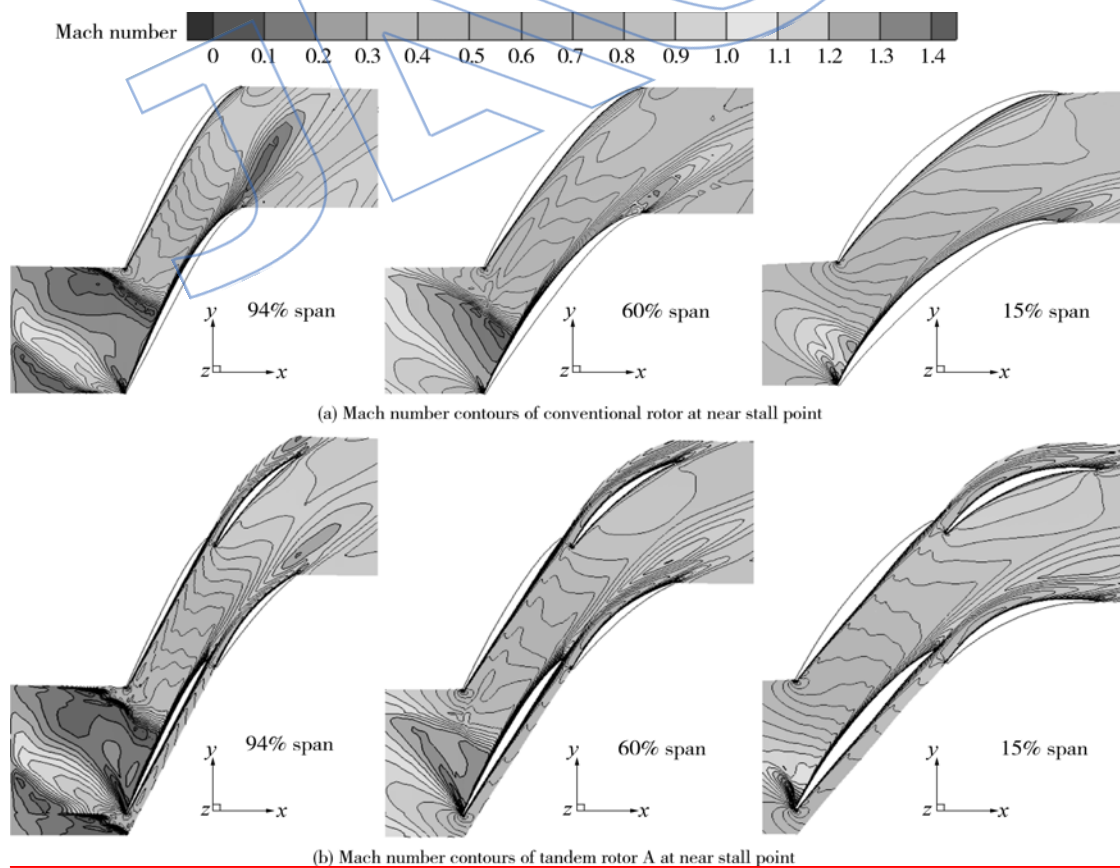


Fig. 9 Comparison of Mach number for conventional rotor and tandem rotor A at near stall point

Compared with conventional rotor, the passage shock over-expands at trail edge in tandem rotor A, hence the design point of forward blade is matched to choke condition closely, which is in order to provide sufficient stall margin for tandem rotor. The wake of forward blade flows through the aft blade and mixes with mean flow. It is apparent that the low-energy region is deeply influenced by the wake from forward blade. The area of the low-energy region is extensive at tip section due to the flow separation resulted from shock/boundary layer interaction. In addition, it should be emphasized that the radial migration of wake due to the static pressure field established by aft blade could not be neglected.

At near stall point, the tip shock structure of both rotor degenerates from double-shock system to normal single-shock system. With the forward movement of passage shock, the aft-shock separation flow has enough space to re-attach, therefore the weak of forward blade alleviates and the low-energy area in aft blade decreases, which could be seen obviously in Fig. 9.

2 Influence from average solidity of aft blade on loading level

One-dimensional fundamental aerodynamic design parameters, such as flow coefficient, loading coefficient and reaction degree, could be chosen perfectly for the forward blade based on the sophisticated experience of compressor design. Meantime, the parameter selection of aft blade is also important to the integral performance of tandem rotor but lacks study. Aft blade shares a considerable loading fraction, hence the average solidity of aft blade and its influence on overall loading level will be investigated in the following part of this paper.

2.1 Design parameters of another two tandem rotors

Another two tandem rotors, marked tandem rotors B and C, were designed with larger average solidity of aft blade, and its loading level

increased correspondingly. Meanwhile the forward blades of them were kept as the same as that of tandem rotor A. The inlet condition and tip speed for all the three tandem rotors is identical, and the design details of these two tandem rotors are shown in Table 3. The loading coefficient attained by tandem rotors B and C are 0.61 and 0.66 respectively, which is closed to high-speed aspirated compressor tested in Massachusetts Institute of Technology reported by Merchant et al.^[19] The outlet metal angle of aft blade at hub is over-turned excessively relative to axial direction for both of these two tandem rotors.

Table 3 Design parameters of tandem rotors B and C

Design parameters	Rotor B	Rotor C
Tip solidity of AB	1.04	1.32
Tip camber angle of AB/(°)	25.7	36.4
Hub solidity of AB	1.32	1.45
Hub camber angle of AB/(°)	91.5	106.4
Average solidity of AB	1.16	1.32
Aspect of AB	1.760	1.540
π	2.39	2.60
ψ	0.61	0.66

2.2 Comparison of calculated results

Characteristic of all the three tandem rotors is shown in Fig. 10, which indicates that tandem rotor is competent to achieve higher loading level and it is necessary to increase average solidity of aft blade to control deviation angle. The aerodynamic design parameters of forward blade are identical for all the three tandem rotors, therefore the loading proportion of forward blade lowers alone with the increasing of the overall loading, that is why the total pressure ratio characteristic line is mild for rotor C and steep for rotor A. The maximum isentropic efficiency of rotor C is highest in three tandem rotors, resulting from the more work produced by sonic aft blade whose efficiency is much higher than

that of forward blade.

Fig. 11 shows the span-wise distribution of outlet metal angle and relative flow angle for all the three tandem rotors at design point. Compared with tandem rotor A, the outlet metal angle of tandem rotor C increased 9.30 and 36.0 at tip and hub section separately. Benefited from the augment of average solidity for aft blade, the deviation angle just amplified a little without obvious performance breakdown.

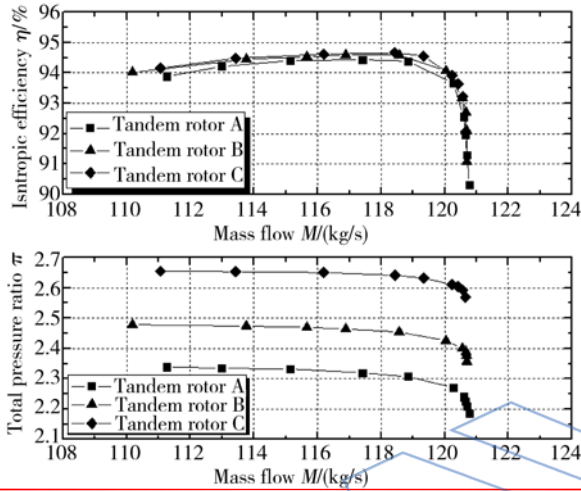


Fig. 10 Comparison of total pressure ratio and isentropic efficiency performances for all the three tandem rotors

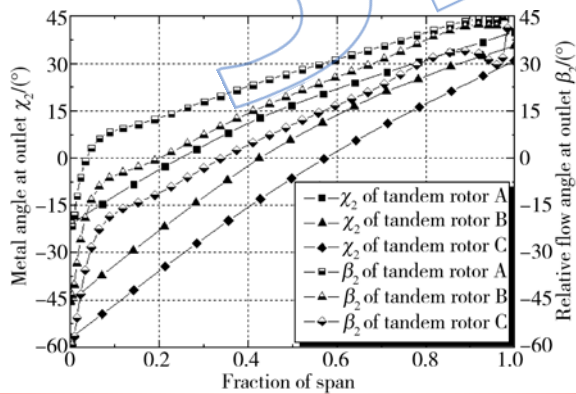


Fig. 11 Span-wise distribution of outlet metal angle and relative flow angle for all the three tandem rotors at design point

Fig. 12 shows the span-wise distribution of overall diffusion factor for three tandem rotors at design and near stall point, which is significant

to assess aerodynamic loading. At near tip section, diffusion factor achieved by rotor C is highest among all the three tandem rotors, and exceeds 0.8 at near stall point. The diffusion factor at hub section is approximately close for all the three tandem rotors at design point, and at near stall point, diffusion factor of tandem rotor C is lower owing to the less static pressure rise caused by more flow overturning.

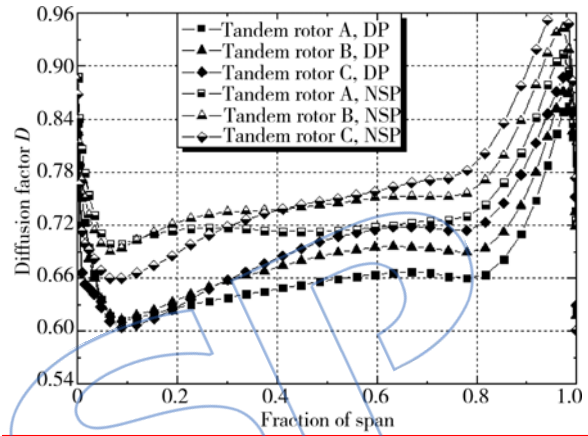


Fig. 12 Span-wise distribution of diffusion factor for all the three tandem rotors at design and near stall point

Fig. 13 illustrates the blade-to-blade Mach number near tip and hub section for tandem rotors B and C at design and near stall point. The enormous camber angle at hub is shown for rotor C, and the flow field near hub is in good condition without severe separation. Meantime, the large flow turning angle produces huge swirl velocity at hub as well as high total pressure ratio. Compared to tandem rotors A and B, the low-energy region in aft blade for tandem rotor C enlarged because of the more intense radial wake migration within the strong static pressure gradient in radial direction established by highly loaded aft blade. It reveals that aft blade with larger average solidity could achieve higher loading level without abrupt deviation angle increase, therefore it is convenient to design tandem rotor in wide range of loading level just through changing solidity of aft blade.

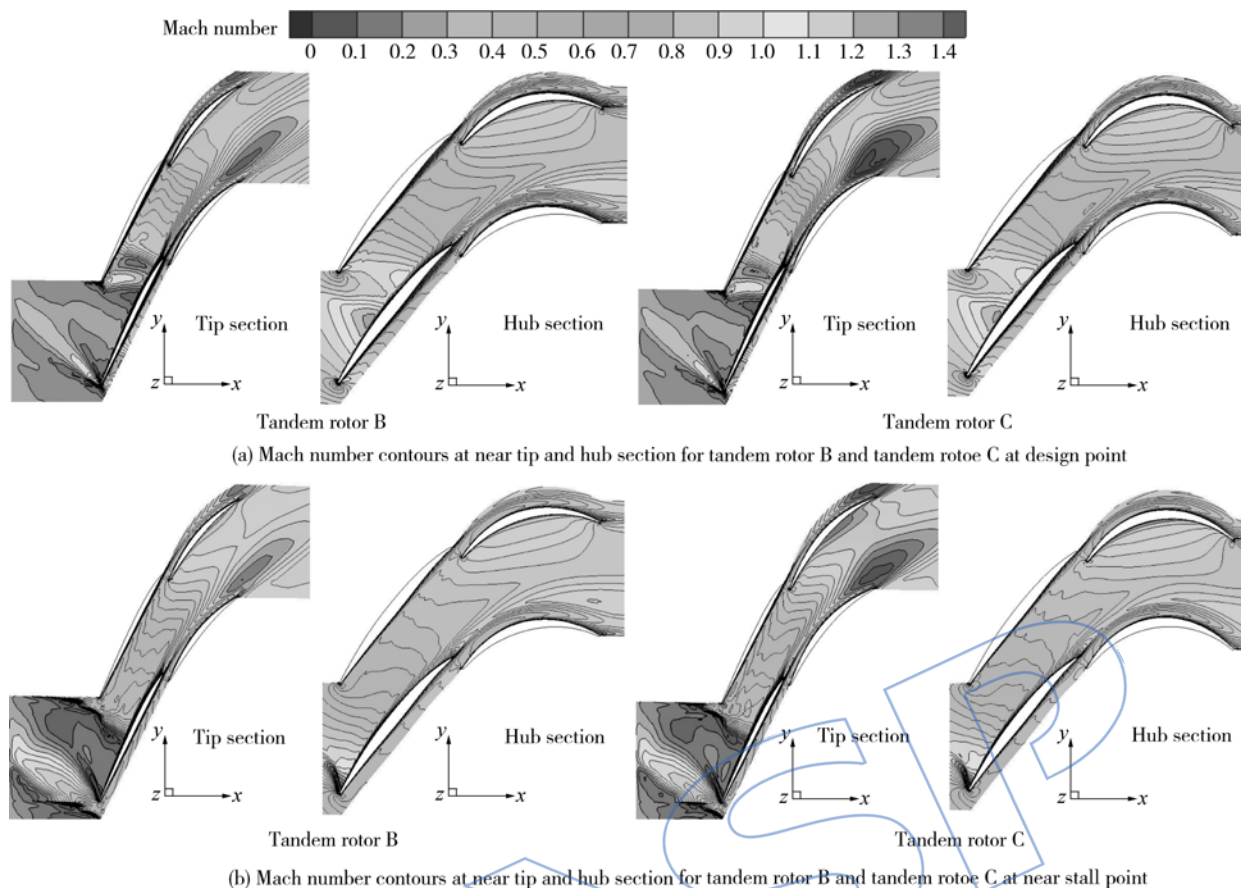


Fig. 13 Near tip and hub section Mach number contours for tandem rotors B and C, at design point and near stall point

3 Conclusions

Three transonic tandem rotors and a conventional rotor were designed in this paper. The loading superiority of tandem rotor was presented comparing to the conventional rotor. The average solidity of aft blade and its influence on overall loading level were studied in depth. The following conclusions can be drawn from analysis of numerical simulation results:

- 1) Deviation angle of conventional rotor is excessive while the loading coefficient attains 0.45, which is the main limit to promote the loading level of transonic rotor.
- 2) Tandem rotor has potential to achieve higher loading level without the restriction from excessive deviation angle due to severe flow separation, and the overall efficiency even increases.
- 3) Tandem rotor can attain higher loading level through increasing solidity of aft blade, and has excellent performance within the range

of loading coefficient from 0.55 to 0.66, but wakes from forward blade accumulates much more with the loading increasing of aft blade.

Acknowledgment

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References:

- [1] Burger G D, Keenan M J. Single-stage evaluation of highly-loaded high-Mach-number compressor stages III: data and performances tandem rotor[R]. NASA-CR-72772, 1971.
- [2] Brent J A, Cheatham J G, Nilsen A W. Single-stage experimental evaluation of tandem-airfoil rotor and stator blading for compressors[R]. NASA-CR-120803, 1972.
- [3] Bammert K, Staude R. Optimization for rotor blades of tandem design for axial flow compressors[R]. ASME Paper 79-GT-125, 1979.
- [4] Bammert K, Beetle H. Investigation of an axial flow compressor with tandem cascade[R]. ASME Paper 80-GT-93,

- 1980.
- [5] Bammert K, Staude R. New features in the design of axial-flow compressors with tandem blades[R]. ASME Paper 81-GT-113, 1981.
- [6] Wu G, Zhang B, Guo B. Experimental investigation of tandem blade cascade with double circular arc profile[R]. ASME Paper 85-GT-94, 1985.
- [7] Sachmann J, Fother L. Highly loaded tandem compressor cascade with variable camber and stagger[R]. ASME Paper 93-GT-225, 1993.
- [8] Roy B, Saha U K. Experimental analysis of controlled diffusion compressor cascade with single and tandem airfoils[R]. ASME Paper 95-CTP-41, 1995.
- [9] Hasegawa H, Hiroaki A. Development of highly loaded fan with tandem cascade[R]. AIAA Paper 2003-1065, 2003.
- [10] McGlumphy J, Ng W F, Wellborn S R, et al. Numerical investigation of tandem airfoils for subsonic axial-flow compressor blades[R]. ASME Paper 2007-GT-43929, 2007.
- [11] McGlumphy J, Ng W F, Wellborn S R, et al. 3D numerical investigation of tandem airfoils for a core compressor rotor[R]. ASME Paper 2008-GT-50427, 2008.
- [12] McGlumphy J. Numerical investigation of subsonic axial-flow tandem compressor blades[D]. Virginia: Virginia Polytechnic Institute & State University, 2007.
- [13] Law C H, Wadia A R. Low aspect ratio transonic rotors Part 1: baseline design and performance[J]. ASME Journal of Turbomachinery, 1993, 115(2): 218-225.
- [14] Wadia A R, Szucs P N, Crall DW. Inner workings of aerodynamics sweep[J]. ASME Journal of Turbomachinery, 1998, 120(4): 671-682.
- [15] Neuber R J, Hobbs D E, Weingold H D. Application of sweep to improve the efficiency of transonic fan Part I: design[J]. Journal of Propulsion and Power, 1995, 11(1): 49-54.
- [16] Chen B S, Hu G R, Wei Y B, et al. Design and experimental researches on single stage fan with high Mach number and highly loaded[J]. Aeroengine, 2000(3): 28-35. (in Chinese)
- [17] Yu Q, Li Q S, Li L. The experimental researches on improving operating stability of single-stage transonic fan[R]. ASME Paper 2002-GT-30640, 2002.
- [18] Denton J D. The calculation of three dimensional viscous flow through multistage turbomachines[R]. ASME Paper 97-GT-345, 1997.
- [19] Merchant A, Kerrebrock J L, Adamczyk J J, et al. Experimental investigation of a high pressure ratio aspirated fan stage[J]. ASME Journal of Turbomachinery, 2005, 127(1): 43-51.