Design and flow rate improvement of a small aeroengine turbine

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Abstract. The need of small high thrust-to-weight ratio aeroengine is growing. One way to meet this is higher through flow rate of engine. Higher engine mass flow means higher turbine mass flow. In the paper, a small turbine is designed and then promoted with higher flow rate. First the definition of universal turbine through flow capability is discussed and formulated. Then the design and flow rate promotion process of a turbine is introduced. And three design results are obtained and compared, marked MI, MII, and MIII. Among them, MI is original design result with highest efficiency of 0.8899 but doesn’t meet mass flow requirement. MII and MIII are two retrofitted turbines based on MI both with improved mass flow rate by about 3.40%. Different flow rate improvement methods are implemented for the two turbines. S shape endwall is applied to MII stator shroud and MIII turns down stator blade exit angle a bit. MII has higher efficiency than MII but through flow capability is in fact lowered. Efficiency decline has been seen in both two redesigned turbines compared to MI. The reason possibly lies in the increased rotor shock loss found by detailed CFD flow analysis. Finally, MII is regarded as the best result of which efficiency is 0.8884.

1. Introduction
For the recent decades, the demand for small and light aerial vehicles has been growing gradually, like unmanned aerial vehicles (UAVs), rotorcrafts, combat drones, cruise missiles, and scaled aircraft models [1-3]. In the future aerial combat, small disposable drones and intelligent missiles would play a more important role in the field. And to improve a drone or missile’s maneuverability and defense penetration, small high thrust-to-weight ratio aeroengine is needed. There hasn’t been a rigorous definition of small engine somehow till now yet. Usually, as Buysschaert [4] and Huang [5] have concluded, a gas turbine engine of thrust class from 1 kN to 10 kN is called small turbine engine. Engine with lower thrust then is designated as micro turbine engine (MTE). And others whose thrust is higher than 10 kN are regarded as middle or large turbine engines. To increase engine thrust, one possible way is to increase mass flow rate or through flow of the engine given the constant exhaust velocity [6]. However, in small aeroengine development, turbine design work will meet new difficulties different from large engine [7, 8].

At present, the major difficulties to design a small turbine come from the size effect [9]. Since turbine blade height is constrained, to maintain the turbine output work ability, the spool rotation speed has to increase for sake of higher rim speed. Generally, the rotation speed can be up to 30,000 to 70,000 r/min...
in a small engine, some MTE’s speed can even be more than 100,000 r/min [10]. Higher rotation speed results in larger disc and blade stress which presents a big challenge for structure and material, and causes complex rotor dynamics problems [11]. Meanwhile, present fabrication technology can’t decrease rotor tip clearance, blade trailing thickness, and hub rounding radius anymore, so the leakage flow and flow blockage will be stronger [10, 12]. Furthermore, blade height to chord ratio of small turbine is usually close to 1, thus larger proportion of secondary flow will generate [10]. Also, during high altitude flight for some reconnaissance UAVs, low Reynolds number condition (approximately under 25,000) in small turbine makes the internal flow more unstable, and easy to separate [13, 14]. Hence engine efficiency will be reduced. And it is more difficult to design cooling configuration and place cooling holes as well [10].

In order to increase aeroengine through flow, it is inevitable to promote mass flow of turbomachinery component. In small aeroengine, radial and axial turbomachineries are both used in practice [15-18]. How to choose the type of turbine to increase mass flow rate is needed to be considered. Ewen [12] points out that axial turbine is preferred now than radial turbine in advanced small aeroengine. Radial turbine on the other hand is more suitable for low flow and high pressure ratio condition. Xia [18, 19] has compared one radial and two axial turbines for a MTE, and concluded that radial turbine has advantage of smaller size when engine flow rate is lower than 0.5kg/s, while on the contrast axial turbine will gain the advantage when flow rate is over 0.5kg/s. However, he thinks that design difficulty of micro turbine stands in the way to implement axial turbine. And Xie [20] as well says axial micro turbine is lighter, easier to realize multi-stage configuration with higher mass flow than radial turbine. But sometimes, as he suggests, turbine flow rate is constrained by compressor. Expect these findings, however, there has not been much more detailed information in open literature about small turbine through flow improvement methods till now. And turbine through flow capability itself is still a vague concept with no accurate and universal definition.

In this paper, the aerodynamic design and through flow improvement of axial turbine for a small short life aeroengine has been introduced in order to explore a way to improve turbine mass flow rate and maintain comprehensive performance as good as possible at the same time. Chapter 1 discusses the definition of universal turbine through flow capability. Chapter 2 introduces turbine design and improvement method. Chapter 3 is the analysis of flow field. Finally, Chapter 4 is conclusion.

2. Definition of universal turbine through flow capability

Before the presentation of turbine design and flow improvement process, it is necessary to find a common and standardized way of turbine mass flow capacity measurement at first. Since different turbines have different scale size, work condition, and configuration, the absolute mass flow rate (mass versus time) can’t be a generalized standard to assess flow capacity. Therefore, the paper introduces a parameter of corrected mass flow rate divided by maximum turbine aerodynamic windward area or corrected flow per unit area, \( M \sqrt{T_0^3(P_0^*A_{max})^{-1}} \), to evaluate through flow capability. Note that the phrase “through flow capability” from now on becomes a specific term in this paper, which stands for this special normalized flow rate that contains the influences of turbine inlet condition and radial geometry. Through flow capability can better reflect turbine physical characteristics than mass flow rate (through flow rate). For convenience, the symbol \( \tilde{M} \) is used to stand for the parameter. Comparison of \( \tilde{M} \) of 8 different aeroengine turbines at design point is shown in Tab. 1 to display the function of the parameter. In the table, each turbine is represented by a sequence number. And as turbine number grows, the maximum tip radius of flow path rises. Here are some briefs of these turbines. No. 1 is the turbine of micro aeroengine MTE-A developed by Nanjing University of Aeronautics and Astronautics [19]. No. 2 is an ordnance turbojet engine turbine [21]. No. 3 is a subsonic experimental turbine developed by Pratt and Whitney Aircraft Division of United Aircraft Corporation [12]. No. 4 is a small low-cost turbofan engine turbine [22]. No. 5 is the high through flow turbine design result MII the paper introduces. No. 6 is the high pressure turbine of NASA/GE E3 aeroengine [23]. No. 7 is a cold-air experimental turbine for high temperature aeroengine research [24]. Finally, No. 8 is the low pressure turbine of NASA/GE E3 aeroengine [25]. Besides, number of turbine stages N and stage average
corrected specific work $L(NT'_0)^{-1}$ are also presented in the table. Still, here use the symbol $\tilde{L}$ to represent $L(NT'_0)^{-1}$.

**Table 1.** Turbine through flow capability.

|   | $N$ | $R_{T,max}$ (m) | $\tilde{M}$ (kg√$K$(s · kPa · m$^2$)$^{-1}$) | $\tilde{L}$ (J/(kg · K)$^{-1}$) |
|---|-----|-----------------|---------------------------------|---------------------------------|
| 1. | MTE-A turbine | 1 | 0.033 | 9.452 | 139.13 |
| 2. | Ordnance turbine | 1 | 0.1235 | 8.589 | 146.28 |
| 3. | Subsonic turbine | 1 | 0.1259 | 1.923 | 153.22 |
| 4. | Low-cost turbofan turbine | 2 | 0.1260 | 6.578 | 137.06 |
| 5. | MII turbine | 1 | 0.15 | 6.629 | 256.05 |
| 6. | E$^3$ HPT | 2 | 0.15 | 12.234 | 111.27 |
| 7. | Experimental turbine | 1 | 0.381 | 6.764 | 132.35 |
| 8. | E$^3$ LPT | 5 | 0.59 | 3.731 | 63.64 |

Generally, it can be seen from the table that $\tilde{M}$ of most turbines is over 6.5 kg√$K$(s · kPa · m$^2$)$^{-1}$, and $\tilde{M}$ of different turbines can be quite different from each other with the highest of over 12 kg√$K$(s · kPa · m$^2$)$^{-1}$ and lowest below 2 kg√$K$(s · kPa · m$^2$)$^{-1}$. Even if two turbines have basically the same $R_{T,max}$, their $\tilde{M}$ can differ a lot in the table. So, it can be assured that some other factors than radial size possibly like engine overall requirement or compressor will limit turbine through flow. Another important factor that is connected with $\tilde{M}$ is $\tilde{L}$. As a matter of fact, stator throat area $A_N$ decides turbine flow capacity and influences specific work at the same time. As $A_N$ grows up (with unchanged number of blades), $\tilde{M}$ increases. However, flow turning and acceleration in stator cascade essentially becomes insufficient, and $\tilde{L}$ decreases in the meantime. Most turbines in the table have $\tilde{L}$ around 140 J/(kg · K)$^{-1}$, while No. 5 turbine has a much higher $\tilde{L}$ over 250 J/(kg · K)$^{-1}$. This high $\tilde{L}$ certainly brings profits. For instance, $\tilde{M}$ of No. 5 turbine is near to No. 4, but its $\tilde{L}$ is almost two times of the latter. Therefore, number of stages can be reduced and so can be the turbine weight. Thus thrust-to-weight ratio will be improved. Also, from this instance comparison, it can be found that $N$ can be thought as the result of $\tilde{M}$ and $\tilde{L}$ combination. In all, with the help of $\tilde{M}$, turbine mass flow rate is translated to some intrinsic physical property.

3. **Turbine design and through flow improvement**

3.1. **Design requirements**

A turbine is asked to be developed mainly to be applied to a small short life aeroengine. Its biggest design difficult is to maintain mass flow rate under high specific output work requirement. Based on the discussion in Chapter 0, the turbine is decided to be axial with one stage. The design corrected flow rate is 0.460 kg√$K$(s · kPa)$^{-1}$, design total pressure ratio $\pi^*$ is 2.895, and total efficiency $\eta^*$ is required to be no less than 0.85. The stator blade needs cooling while rotor does not.

3.2. **Overall design process**

The whole design process herein still follows the traditional system from 1D, through flow, and blade design with the in-house code and CFD computation later [26]. And it takes several loops to finish the job. As Lu [27] advises, once overall performance has been achieved in lower dimensional job, detailed analysis and optimization effort should be spent in 3D phase. And the paper turbine design work basically agrees with this idea. The CFD simulation and local geometry adjustment have cost the majority of time during the design process.

In preliminary design, meridian flow path and three important parameters need to be decided, i.e. flow coefficient $\psi$, loading coefficient $\mu$, and mean rotor reaction $\Omega$. Definitions of them are shown from formula (1) to (3). Some initial geometry and aerodynamic design parameters are inputted manually into
the in-house code. The 1D code uses energy loss $\zeta$ models (including profile loss $\zeta_p$, secondary loss $\zeta_s$, and tip clearance leakage loss $\zeta_k$, shown in formula (4)) [28] to solve inviscid angular momentum equation based on mean flow theory. In formulas (4) to (7), $\beta_1$ and $\beta_2$ are stator and rotor exit relative flow angle, $h/b$ is aspect ratio, and $k/b$ is relative tip clearance height. Through flow code generally uses the same loss models to solve inviscid radial equilibrium equation with streamline curvature method neglecting blade force effect. Meanwhile, flow path is adjusted during through flow design for better performance. Some changes have been made on secondary and tip clearance leakage loss model. And consequently, $\zeta_s$ is divided into tip loss term $\zeta_{st}$ and hub loss term $\zeta_{sh}$ both varying with radius $r$, as formulas (9) to (11) show. And also becomes a function of $r$, shown in formula (12). This way radial flow loss distribution can be more accurate. Next, with energy loss coefficient known, blade row total pressure recovery $\sigma$ can be calculated by formula (13). Since shock loss is not included in the loss model, some discrepancy may be produced when designing transonic turbines. Nevertheless, design practice shows that error won’t be very large. Next with blade inlet and outlet angles, blade design code utilizes Pritchard 11 parameters profiling method to design blade sections. Leading and trailing edge circle radius and wedge angle, maximum thickness and its chord wise location, metal angle, and chord length need to be inputted into the code. The blade section profiles are represented by Bezier curve.

$$\psi = \frac{v_{1s}}{U_m}$$  \hspace{1cm} (1) \\
$$\mu = \frac{\Delta h}{U_m^2}$$  \hspace{1cm} (2) \\
$$\Omega = \frac{h_{1-h_2}}{h_0-h_2}$$  \hspace{1cm} (3) \\
$$\zeta = \zeta_p + \zeta_s + \zeta_k$$  \hspace{1cm} (4) \\

$$\zeta_p = \frac{0.003}{\cos \beta_2 + 0.46 (\cos \beta_1 - \cos \beta_2) + 0.085} + 0.017$$  \hspace{1cm} (5) \\

$$k_{p1} = \begin{cases} 
\cos \beta_1 \cos (\beta_1 + \beta_2), & \beta_1 + \beta_2 > 90^\circ \\
\cos \beta_1 \cos (\beta_1 + \beta_2), & \beta_1 + \beta_2 \leq 90^\circ 
\end{cases}$$  \hspace{1cm} (6) \\

$$\zeta_s = \begin{cases} 
0.0474 \frac{\cos \beta_2}{\cos \beta_1} \left( \frac{\cot \beta_1 + \cot \beta_2}{2} \right) + 0.0118, & h/b \geq 2 \\
0.0474 \frac{\cos \beta_2}{\cos \beta_1} \left( \ln \left(2 \frac{h}{b} \right) \right) + 0.0118, & h/b < 2 
\end{cases}$$  \hspace{1cm} (7) \\

$$\zeta_k = a_k \frac{k}{h}$$  \hspace{1cm} (8) \\

$$\zeta_s = \zeta_{st} + \zeta_{sh}$$  \hspace{1cm} (9) \\

$$\zeta_{st} = a_{st} r^2$$  \hspace{1cm} (10) \\

$$\zeta_{sh} = a_{sh} r^2$$  \hspace{1cm} (11) \\

$$\zeta_k = a_{ak1} \frac{k}{h} r^2$$  \hspace{1cm} (12)
\[ \sigma = \left[ 1 - \left( \frac{1 + \frac{\gamma^2 - 1}{2} M_a^2}{1 - \frac{\gamma^2 - 1}{2}} \right)^{\frac{\gamma}{\gamma - 1}} \right] \]  

(13)

3.3. CFD simulation method

After design work finished, CFD calculation plays a role in analysing design result with commercial software package NUMECA. Many researchers have used this software in turbine numerical simulation, and proven its reliability [14, 20, 27, 29]. The RANS method is a reliable and relatively simple tool in turbomachinery 3D calculation [30]. And steady RANS approach of FINE/Turbo is used to solve the flow field, and mixing plane method is used in stator/rotor interface flow solution. Spalart-Allmaras turbulence model is used, second order central difference scheme is used for spatial discretization, and fourth order Runge-Kutta scheme is used for time discretization. And O-H topology is adopted to construct computational mesh by AutoGrid with solid wall \( y^+ \) less than 5, minimum skewed angle greater than 30°, maximum aspect ratio less than 400, and maximum expansion ratio less than 2. Single turbine blade row channel mesh structure is show in figure 1, and number of grid points is around 2 million. Solver precision is double, and convergence criteria level is 10e-6 during computation. Inlet boundary condition is set according to total pressure and temperature, and outlet boundary condition is settled by static pressure imposed with radial equilibrium. To confirm the mesh level validation, independence test of mesh has been carried out with 8 different meshes and the paper design turbine MI. Among them, grid level varies from 440,000 to 4,200,000. The corrected mass flow of each meshes is displayed in figure 2. Clearly, very little change can be found of the mass flow after number of grids is higher than 1,600,000. Thus, 2,000,000 level mesh should be reliable for calculation.

Combustion gas property is used in CFD simulation which incorporates empirical functions of constant pressure specific heat \( C_p \) and heat conduction coefficient \( \lambda \). Both \( C_p \) and \( \lambda \) are regarded as polynomial functions of temperature. Total efficiency \( \eta^* \) is then computed by equation (14) and (15), where \( T_{2i}^* \) is the isentropic rotor exit total temperature given the same total pressure ratio \( \pi^* \). Yet, for simplification, some researchers still use equation (16) based on constant specific ratio \( \gamma \) to evaluate efficiency \( \eta_c^* \) even if some of them are using variant \( C_p \) during simulation at the same time [20, 27, 31]. Without a doubt, this will cause discrepancy. Figure 3 exhibits an example of efficiency calculation result with varying \( \gamma \). The \( \eta_c^* \) changes in a near linear fashion, and every time \( \gamma \) grows by 0.005, \( \eta_c^* \) will drop by about 0.01. Therefore, the error introduced by \( \gamma \) can be big.

\[ \eta^* = \frac{\int_{T_{2i}^*}^{T_{1i}^*} C_p(T) dT}{\int_{T_{2i}^*}^{T_{1i}^*} C_p(T) dT} \]  

(14)

\[ \int_{T_{2i}^*}^{T_{0i}^*} \frac{C_p(T)}{T} dT = R \ln \pi^* \]  

(15)

\[ \eta_c^* = \frac{T_{1i} - T_{2i}}{T_{1i} \left( 1 - \frac{1}{\pi^* \gamma} \right)} \]  

(16)
4. Results comparison

4.1. Overall performance

After many trials, three design results have been accomplished, marked MI, MII, and MIII separately. Of them, MI is initial result which has in reality not met the mass flow rate requirement, even though it shows satisfactory pressure ratio and efficiency. MII and MIII, next, are two retrofitted plans of MI to raise mass flow. However, some performance characteristics are reduced as a sacrifice. MII has adjusted stator shroud endwall profile to S shape and MIII has reduced stator tip blade section exit angle by around 2° keeping axial chord length unchanged. Stator shroud S shape endwall is suggested as an easy and effective method to control endwall secondary, reduce loss, and improve efficiency [9, 11, 12, 32-34]. Also, it can increase flow rate [12, 33]. According to reference [34], annular cascade loss reduction is related to relative convergence, \((h_1 - h_2)/h_2\) and aspect ratio \(h_1/b\) \((h_1\) is inlet blade height, \(h_2\) is exit blade height, and \(b\) is blade chord length). But in design practice as Shi [34] implies, \((h_1 - h_2)/h_2\) can be chosen within a relatively wide range. Hass [33] has tested two turbines with different contoured stators, and both turbines have higher efficiency compared to cylindrical wall stage due to lower stator loss. Due Jr. [35] has also conducted experimental survey on one cylindrical wall and three different contoured wall stators, and found different outer wall contouring could all improve cascade efficiency. Thus, basically almost any kind of S endwall can be beneficial. Finally, the \((h_1 - h_2)/h_2\) of MII is 0.156 and MIII is 0.751. Referred to the picture provided by reference [34] (figure 4), it could be found that the match of these two parameters should be appropriate. And definitely smaller blade exit angle of MIII shall enlarge stator throat area, and then flow rate rises. MI and MIII share the same meridian flow path.
Flow path comparison of MI and MII is shown in figure 5. Stator tip blade section profile comparison of MII and MIII is shown in figure 6.

Because stator blade is in need of cooling, its thickness is designed to be higher. Stators blade profiles are designed at hub at tip section on the cylindrical surface, and rotor blade profiles are designed at hub, mean, and tip section. Stator blades sections are stacked along leading edge and rotor sections are stacked through centroids. Rotor blade stacking line has a small lean angle circumferentially to compensate some aerodynamic force on blade pressure surface to improve blade strength. For MII, when stacking stator blade sections, tip profile line is extrapolated radially to intersect with S endwall. Since the turbine is going to be used in short life engine, AN^2 is increased above conventional standard to improve through flow capability. As reference [9] indicates, usually AN^2 of high pressure turbine is within (19–32) m²/(r/min)². While the paper turbine’s AN^2 is 36 m²/(r/min)². Empirically, it is believed this compromise could be acceptable. And meanwhile better material or fabrication can if necessary, make up for rotor strength flaw.

Overall aerodynamic characteristic parameters obtained by NUMECA of MI, MII, and MIII are displayed in Tab. 2. And three different total efficiency computation results of the turbines are demonstrated in Tab. 3. In the table, \( \eta^* \) is calculated by equation (14), \( \eta_{c1}^* \) is calculated with different \( \gamma \) at rotor/stator interface of each turbine respectively by equation (16), and \( \eta_{c2}^* \) is also obtained by equation (16) given the same \( \gamma \) (1.318) which is the mean value of the three turbines at rotor/stator interface. And among them, \( \eta^* \) is thought as the most accurate result.

|            | \( \psi \) | \( \mu \) | \( \Omega \) | \( \bar{M} \) (kg \( \sqrt{K} \)(s \cdot kPa)^{-1}) | \( \bar{M} \) (kg \( \sqrt{K} \)(s \cdot kPa \cdot m^2)^{-1}) | \( \bar{L} \) (J/(kg \cdot K)^{-1}) |
|------------|------------|------------|------------|---------------------------------|---------------------------------|---------------------------------|
| MI         | 0.7846     | 1.675      | 0.2624     | 0.4531                           | 6.859                           | 256.51                          |
| MII        | 0.7794     | 1.672      | 0.2949     | 0.4685                           | 6.629                           | 256.06                          |
| MIII       | 0.7600     | 1.668      | 0.3040     | 0.4685                           | 7.093                           | 255.52                          |
Table 3. Total efficiency.

|     | $\eta^*$  | $\eta_{c1}$ | $\eta_{c2}$ |
|-----|----------|-------------|-------------|
| MI  | 0.8899   | 0.9084      | 0.9092      |
| MII | 0.8884   | 0.9086      | 0.9077      |
| MIII| 0.8866   | 0.9071      | 0.9058      |

From the first table above, it can be found that MII and MIII have effectively increased flow rate $\bar{M}$ by around 3.40% from MI, however both of them have suffered specific work deterioration at the same time. With consideration of manufacture error and blade corner rounding induced flow blockage, $\bar{M}$ of MII and MIII is a little higher than design value. $\bar{M}$ of MIII is the highest. Although MII has improved $\bar{M}$, its shroud radius goes up at the same time causing through flow capability to get lower. So, S endwall may not necessarily be effective in $\bar{M}$ promotion. Different design attempts have shown that incorporating low $\psi$ and $\mu$ can be beneficial for turbine efficiency, but turbine radial size will increase at the same time. And to achieve high enough stator expansion work and required $\pi^*$, $\Omega$ should be low. Compared with MI, both MII and MIII has reduced $\psi$, and increased $\Omega$. The reason of this change could be the larger stator throat of MII and MIII. And then flow acceleration in stator decreases as stated in Chapter 1. Consequently, $V_{1z}$ and $\Delta h$ turns smaller with higher bigger proportion of expansion work loaded on rotor. Higher $\Omega$ and increased stator exit gaging area attributed by S endwall is also found by Ewen [12].

Next from the second table, it shows that $\eta^*$ of MI is the highest, MII is 0.0015 lower, and MIII is 0.0033 lower when compared. The efficiency decline is the cause of $\bar{L}$ decrease. This does not accord with other researchers’ findings where turbine efficiency is successfully promoted with implementation of S endwall [12, 32]. The reason will be explored the next section. Compared with $\eta^*$, $\eta_{c1}$ and $\eta_{c2}$ are higher by around 0.02. Since $\eta_c$ is close to a linear function of $\gamma$ as shown in figure 2, therefore the suitable value of $\gamma$ might be higher by 0.01. And compared to $\eta_{c1}$, $\eta_{c2}$ is more accurate and its change tendency from MI to MIII is very close to $\eta^*$. $\eta_{c1}$ of MI is lower than MII, this is the result of different $\gamma$ induced discrepancy. $\eta_{c2}$ is very sensitive to $\gamma$, the small change of $\gamma$ can lead to a large error of absolute value of efficiency covering the real physical quality. But equation (15) could still be used to compare relative efficiency among different turbines as soon as their operation conditions and configurations are similar, and the same $\gamma$ is applied.

4.2. Flow field analysis

Since efficiency has dropped in two redesigned turbines, it is necessary to find out the reason. Secondary flow loss takes a big park of overall loss in small turbine, and alterations are made in stator of MII and MIII. Then first of all, secondary flow in stator is analyzed. figure 6 shows limiting streamlines at stator shroud and hub endwall respectively of MI, MII, and MIII. Clearly, S endwall has promoted the flow condition near stator shroud found from figure 7 (a). For MI and MIII, crossflow is stronger with pressures side separation line starting from saddle point moves toward and ends at the adjacent blade suction side. Pressure separation line of MII on the other hand moves downstream without intersection with the other blade. Thus, lower transverse pressure gradient should there be near stator tip of MII. Transverse pressure gradient is the source of secondary flow [34], therefore secondary flow is then effectively controlled. However, figure 7 (b) shows no obvious difference of flow near hub that can be seen among three turbines. Influence of S shape endwall is mainly focused on shroud area.
Then blade loading is analysed here. Stator and rotor blade section static pressure coefficient distribution $P/P^*$ at different span locations of the three turbines is shown in figure 8 and 9. The benefit of S endwall on stator blade is remarkable as figure 8 shows. Stator blade loading distribution of MI and MIII are close. At 90% blade span location there is an abrupt pressure rise at suction side of MI and MIII from 0.35 to 0.55 axial position, behind the maximum curvature point of suction profile line. This is the result of strong acceleration on frontal suction side surface with pressurized waves formed, as depicted in figure 10. And at 10% and 50% span section, there is also a phenomenon of over speeding of flow on frontal half of suction side for MI and MIII. While for MII, stator blade pressure side $P/P^*$ is raised especially after 0.5 axial position, and loading is successfully put back by reducing flow pressurization near high curvature profile area on suction side. In the meantime, figure 8 clearly displays the lower pressure difference between pressure and suction side of MII, which is the reason of weaker secondary flow as discussed before. And S shroud wall improvement is most effective at 90% span. However, the pressure rise near trailing edge is not restrained on all turbines.

As for rotor blade, MII and MIII have both managed to optimize the loading distribution at different levels compared to MI depicted by figure 9. Generally, $P/P^*$ on pressure side is raised for MII and MIII. At 10% span location, there near the leading area of MI suction side exists reverse pressures gradient. MIII has reduced the pressure gradient and MII basically has eliminated it. Meanwhile, pressure rise after 0.7 axial position is also decreased in MII and MIII. The pressure distribution at 50% span section
of MII and MIII are very close. And both two retrofitted turbines have improved suction side pressure gradient from near leading to trailing edge as well as eliminating the pressure rise at trailing edge. At 90% span, the pressurization of MI at trailing of suction side is also removed in MII and MIII. Compared to MI, positive pressure gradient along suction side surface is improved in MII and is further improved in MIII. Pressure side $P/P^*$ rise is also increased more in MIII than MII.

![Stator blade section $P/P^*$](image)

(a) 10% span.  
(b) 50% span.  
(c) 90% span.

**Figure. 8** Stator blade section $P/P^*$

![Rotor blade section $P/P^*$](image)

(a) 10% span.  
(b) 50% span.  
(c) 90% span.

**Figure. 9** Rotor blade section $P/P^*$

![Stator Mach number contour at 90%](image)

**Figure. 10** Stator Mach number contour at 90%

Flow loss decides the efficiency of turbine, and the cause of efficiency drop of two redesigned turbines can be explored from loss analysis. Stator and rotor total pressure loss coefficient of three turbines is illustrated in figure 11. And blade inlet and exit Mach number is displayed in figure 12. It can be found that MIII stator loss is basically reduced compared to MI and MIII. This is possibly the result of effective control of secondary flow and blade suction side reverse pressures gradient. Also, it could be the consequence of decreased inlet and exit Mach number, shown in figure 12 (a). On the contrary, loss is hardly decreased in MIII, and though stator exit Mach number is reduced, inlet Mach
number turns higher. In spite of the optimized rotor blade loading distribution, effective overall loss reduction is not seen in MII and MIII. Loss near rotor hub has been reduced in the two retrofitted turbines, and this could be the effect of endwall secondary flow control, since hub transverse pressure gradient of these turbines is lower. Also, the high Mach zone with more complex expansion and pressurized waves near leading edge are in MI rotor hub as figure 13 exhibits could bring extra flow loss. Lower rotor inlet Mach number could be part of the reason of lower hub loss in MII and MIII. But loss above 40% span area becomes higher in MII and MIII. The reason could be the higher rotor exit Mach number which produces stronger shock wave and more shock loss as shown in figure 12 (b) and figure 14. Therefore, increased rotor loss should possibly be blamed for the descended efficiency. Also, it can be explained why S shroud endwall does not promote stage efficiency like other researchers have found. Because in Ewen’s experiment [12], both contoured and straight wall turbines testing S endwall function are subsonic. The three turbines tested in Hass’ work [33] are also substantially subsonic, with only one contoured turbine having tip exit relative Mach number slightly over 1 (1.023) on design point. While in this study here, MII turbine is basically transonic with higher exit relative Mach number.

![Blade row total pressure loss](image1.png)  
**Figure. 11** Blade row total pressure loss.

![Blade row inlet and exit Mach number](image2.png)  
**Figure. 12** Blade row inlet and exit Mach number.
4.3. Design summary
After comparison of MI, MII, and MIII design results, it is decided that MII should be the best result. Because it meets the flow rate requirement and has higher efficiency than MIII. Although $\bar{M}$ is not promoted in MII but in MIII, it is still believed that efficiency improvement of 0.0018 from MIII is more important. The higher efficiency of MII among the two retrofitted turbines is achieved by S shape endwall. The S endwall is another way to enlarge stator throat area rather than turning down the blade exit angle. And it is more profitable in improving flow characteristics. Unfortunately, on the other hand, overall stage efficiency promotion has not been seen in MII. The reason could be in the larger rotor shock loss.

5. Conclusions
The paper discussed the definition of universal turbine through flow capability firstly, and then mainly introduces the design and flow rate improvement process of a turbine prospectively for application in small short life aeroengine. Original and two retrofitted design results are shown and compared, and finally turbine MII is decided as the best result. Conclusions can be summarized in 3 points.

(1) A precise and generalized assessment method of turbine through flow capability has been proposed with the introduction of the parameter, corrected flow per unit area $\bar{M}$. This way, through flow capability becomes a clear and special concept that can be measured quantitatively and better reflects intrinsic physical property.

(2) The final turbine design result MII is basically satisfactory with corrected mas flow of 0.4685 kg/$\sqrt{\bar{K}}$($s\cdot kPa$)$^{-1}$, through flow capability of 6.629 kg/$\sqrt{\bar{K}}$($s\cdot kPa\cdot m^2$)$^{-1}$, efficiency of 0.8884, and corrected specific work of 256.06 J($kg\cdot K)^{-1}$ at design point and its exit flow is transonic. Its efficiency
is higher by 0.0016 than MIII. The biggest characteristic of the turbine is considerably large specific work much higher than many others with lowered strength requirement.

(3) Stator shroud S shape endwall is effective in stator secondary flow control, blade loading optimization and loss reduction. It can also improve loading distribution on rotor blade. It is as well a way to improve turbine mass flow rate. However, in transonic turbine, it might increase rotor exit Mach number and lower efficiency. And S endwall does not necessarily improve $\dot{M}$.

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