RADIAL AND MIXED FLOW TURBINE OPTIONS FOR HIGH BOOST TURBOCHARGERS

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Presented at
7TH INTERNATIONAL CONFERENCE ON
TURBOCHARGERS AND TURBOCHARGING
May 14-15 2002

ABSTRACT

This paper demonstrates that there are definite and predictable limits to the boost pressure obtainable when the turbocharger compressor is driven by a conventional single stage radial turbine. Present engine requirements go beyond these limits. The usual solution is a series turbocharging arrangement which brings additional complexity and parts count. Alternative single stage turbines for high boost pressures are investigated. It is shown that mixed flow turbine concepts can achieve stage loadings that are about 20% greater than those of a conventional radial turbine, without any increase in blade speed and maintaining structural integrity. The design of such a turbine is described. Current developments in new turbine rotor materials offer an alternative route to this goal, and the prospects for this are also considered.

NOTATION

C	absolute velocity
h	enthalpy
U	blade speed
β	relative flow angle
ϕ	flow coefficient
ψ	stage loading coefficient

Subscripts

m	meridional
θ	tangential
0	stagnation state
2	rotor inlet
3	rotor exit

INTRODUCTION

There is a continuing upward trend in boost pressure in order to achieve higher BMEP and reduce emissions levels. The boost pressure that can be achieved in a simple turbocharger is usually considered to be limited by the compressor, but it may also be limited by the turbine. A single stage centrifugal compressor is capable of working efficiently at high pressure ratios, but this is usually only achieved at the expense of range. For applications where a wide range is not required, the limiting factor may then be the ability of the turbine to provide sufficient power to drive the compressor while maintaining an adequate service life and competitive manufacturing cost.

Where high compressor pressure ratio and wide range are required simultaneously, the common solution is the series, or two stage, turbocharger. Most often this is implemented by coupling two single stage turbochargers. This has the advantage of requiring no new components, but the installation volume can be significant, as can the parasitic pressure losses in the ducting between the turbochargers. In a few limited cases integrated two stage turbochargers have been developed, but these have not completely overcome the problems of two, single stage machines.

An alternative solution where high boost pressure is required together with wide range is a two stage compressor driven by a single stage turbine. This can be made considerably more compact than a series turbocharger arrangement. The limitation that the two compressor stages must rotate at the same speed is not as serious as is sometimes supposed. Any turbocharger is optimally matched to an engine at one condition only and a series turbocharger is only slightly better than a single turbocharger at other conditions. The solution to the matching problem that is increasingly being adopted today is variable turbine geometry. Optimal operation of a series turbocharger requires that both turbines be variable geometry, which is considerably more expensive than any turbocharging scheme that uses only a single turbine.

The power required to drive the compression system increases with boost pressure. When high boost pressure is used, the focus of attention falls on the power that the turbine can effectively deliver. In this paper we examine the limits of conventional radial inflow turbines and show how these limits can readily be predicted. Where higher output power is necessary, modifications to the conventional turbine geometry can extend the maximum power and delay the point at which a two stage turbine is required.

POWER REQUIREMENTS

The performance of a radial-inflow turbine can simply be described in terms of two nondimensional parameters: the stage loading coefficient $\psi = \Delta h_0/U^2$ and the flow coefficient $\phi = C_m/U$. Chen and Baines (1) showed that these parameters could be used to correlate the turbine efficiency very effectively. Figure 1 shows the measured efficiencies of a large number of radial turbines of various sizes and configurations, Clearly the maximum efficiency is obtained for stage loading coefficients in the range 0.9–1.0 and flow coefficients between about 0.2 and 0.3.

These data establish a link between the specific work output of the turbine Δh_0 and the rotor blade tip speed U. By making some simple assumptions about the stage loading coefficient, the mechanical efficiency of the turbocharger, and the efficiency of the compressor, it is possible to relate the blade speed to the boost pressure ratio of the compressor. This is shown in Fig. 2 for a turbine stage loading coefficient of 0.95 and a range of compressor efficiencies. For many applications the maximum compressor efficiency that is realistically achievable today is about 0.85, and in this case a boost pressure ratio of 6, for example, requires a turbine blade tip speed of approximately 500 m/s. Higher boost pressures, or lower compressor efficiencies, require larger turbine blade tip speeds.

The maximum blade speed that can be tolerated is limited by the turbine rotor stress. The largest component of this is centrifugal and hence is a function of the square of the blade speed. Other stress components include pressure, thermal and vibratory loads and so the actual relation between peak stress and blade speed is not a simple one, but peak stress does increase rapidly with blade speed. The maximum allowable stress is a function of the rotor material, the quality of the casting, and the duty cycle of the engine. For engines that undergo many cycles, low cycle fatigue limits the rotor life, and with the commonly used Inconel alloys, the blade speed limit is usually found to be in the region of 500 m/s. For engines that run continuously for long periods this limit may be extended somewhat.

Further increases in boost pressure therefore require an increase in either the turbine stage loading coefficient or the blade tip speed. The former is a function of the turbine design and it is clearly important to understand what it is about the conventional radial-inflow turbine that prevents higher stage loadings from being achieved with good efficiency. The latter requires developments in blade materials and manufacturing technology, and the prospects for this are discussed later in this paper.

LOADING LIMITS

The specific work output of the turbine can be written in terms of the blade speed U and the tangential component C_θ of the gas velocity at the inlet (station 2) and exit (station 3) of the rotor using the Euler turbomachine equation:

$$\Delta h_0 = U_2 C_{\theta 2} - U_3 C_{\theta 3}$$

Because of the radius change in the rotor, the blade speed at exit is smaller than that at inlet, and for a turbine working at its design point the flow leaving the rotor is usually in, or close to, the axial direction. The second term on the right hand side of the equation is therefore much smaller than the first, and to a good approximation the equation can be written as

$$\Delta h_0 = U_2 C_{\theta 2}$$

The stage loading coefficient can therefore be written as

$$\psi = C_{\theta 2} / U_2$$

and then, using the geometry of the rotor inlet velocity triangle, it can be shown that the inlet flow angle of gas relative to the rotor is

$$\beta_2 = \tan^{-1}\left(\frac{\psi - 1}{\phi}\right)$$

This equation is plotted in Fig. 3. Because the inlet blade angle of a radial-inflow turbine is zero (measured with the radial direction), the rotor inlet flow angle is numerically equal to the incidence angle. Considering the region of maximum efficiency in Fig. 1, it can be seen in Fig. 3 that this occurs at negative incidence. This is consistent with the common observation that radial turbines achieve their best efficiency when the incidence is negative (2). Zero incidence occurs at a stage loading coefficient of unity irrespective of the flow coefficient, and for stage loading coefficients greater than unity the incidence becomes positive. Positive incidence very quickly leads to separation of the flow from the blade suction surface and poor performance ensues.

It would appear that in order to achieve higher stage loadings, therefore, the blades must be set, not in the radial direction, but curved forward to match a positive inlet flow angle, see Figs. 4a and 4b. Unfortunately this solution must be ruled out because it introduces a large degree of non-radial character into the blade sections near the inlet and greatly increases the blade stresses. However, radial sections can be maintained in a mixed flow turbine as illustrated in Fig. 4c. Such a solution, combining radial blade sections and forward curvature, introduces considerable lean into the blade geometry which, if taken too far, can cause manufacturing difficulties. The amount of lean depends on the cone angle at the inlet and the blade inlet angle, and the actual limit depends on the manufacturing process, but generally it is found that inlet blade angles up to 20–30° are acceptable. Figure 3 indicates that the maximum stage loading coefficient can be extended to about 1.15 by doing so.

DESIGN OF A HIGHLY LOADED MIXED FLOW TURBINE

A turbine was required to drive a compressor in order to achieve a high boost pressure ratio. The compressor design speed was fixed at 31,000 rpm and the target power output for the turbine was 670 kW. The application required an intensive cyclic duty, and based on fatigue tests on material specimens, the engine exhaust gas temperature, and experience of the likely levels of stress in the rotor, a maximum blade speed of 440 m/s was arrived at. This leads to a stage loading coefficient of 1.29 which is well in excess of anything achieved with a conventional radial-inflow turbine design.

Many trial calculations using the mean line analysis described in (3) were made in order to obtain the best possible turbine efficiency within the limits of what was feasible, and the final result is shown in column A of Table 1. The rotor inlet relative flow angle is large and positive at 43.2°, and this rules out the use of radial section blades. A mixed flow turbine design was therefore selected. This choice also meant that the ratio of the exit tip to inlet tip radii could be increased to high values without introducing excessive shroud curvature. The specific speed of this application is quite low, and together these meant that the exducer tip radius was relatively unconstrained. The exducer hub radius however is limited by crowding of the blades, and it was anticipated that the blade hub thickness would be large to confer adequate strength.

The design was then detailed and subjected to flow field and structural analyses. The flow field analysis was a three dimensional viscous analysis method that has been extensively tested for many types of turbomachine (4). This revealed that this level of stage loading could not be supported in reality, because with any acceptable number of blades the flow separated early on the suction surface near the hub (Fig. 5a) and late on the same surface near the tip (Fig. 5b), leading to considerable underturning. The structural analysis indicated that the blade and hub stresses were close to their limiting values, confirming the initial expectations for this blade speed.

The design therefore had to be re-examined. A change of material was made to one of slightly higher strength (with a cost penalty), and with that and some further refinement to the rotor design it was decided that an increase in blade tip speed to 470 m/s could be managed. This reduced the stage loading to 1.12 and the approach flow angle to 13.9°. This meant that the rotor inlet blade angle could be set at about 25° to maintain the incidence adequately negative. Additional work output to achieve the design target was gained by allowing the rotor exit whirl to increase to just over 20°, which is about the limit for effective pressure recovery in an exhaust diffuser. The rotor exit hub radius had to be increased to allow the blades to be thickened. The result is summarized in column B of Table 1. The total-to-total efficiency of the new design was predicted to be just over 1% higher than that of the old, but the total-to-static efficiency is slightly lower because the more restricted exducer increases the rotor exit kinetic energy. The effects of incidence have not been completely eliminated and there remains a small region of separation in the hub-suction surface corner (Fig. 6a), this is much smaller than before and does not affect the remainder of the flow field. The flow near the tip of the blade (Fig. 6b) is quite uniform and the deviation is reduced to a very small amount.

ALTERNATIVE TURBINE ROTOR MATERIALS

The alternative route to high levels of turbine output power is through the development of new rotor materials. The industry standard for many years has been the nickel-based alloys Inconel 713C or 713LC, which combine excellent creep resistance at high temperatures with good casting properties and an acceptable cost. Other materials are worthy of consideration and in some cases are under active consideration in the quest, not only for higher blade speeds, but also reductions in mass and inertia. Some of these are listed, together with relevant properties, in Table 2.

The baseline is IN713LC. IN718 is of similar composition, but it is a wrought alloy that offers significantly higher fatigue strength at high temperatures. It is widely used for disks in gas turbines that are exposed to moderate temperatures, but is not usually considered to offer any advantages for turbochargers except for a few specialty applications where cost is not a principal consideration.

A number of other material have also been developed for high temperature gas turbine applications. CM MAR-M-247LC is one such material that is used in at least one turbocharger application. The specific strength of this material is similar to that of IN713LC but the specific stiffness is much higher. This suggests that thinner blades might be used to aerodynamic advantage while maintaining an adequate high cycle fatigue life. The material is slightly more dense than IN713LC but with thinner blade sections the overall rotor mass and inertia are likely to be about the same. The principal drawback of this material is the additional cost.

Titanium alloys offer high specific strength and have long been used in the last stages of high pressure gas turbine engine compressors at temperatures up to about 450°C. Their application is limited to turbochargers of relatively low exhaust gas temperature. Since the specific stiffness is similar to that of IN713LC, the advantage of a titanium alloy would be the reduction in mass and inertia.

The intermetallic titanium aluminide is currently receiving considerable attention. It has a specific strength similar to that of IN713LC, but the specific stiffness is much higher. It would therefore allow complex blade shapes to be formed in thin sections while retaining an adequate fatigue life. Compared with conventional metal alloys it is quite brittle as the low value of elongation to rupture shows. This material was originally developed for military applications, and many of the key properties remain uncertain and the details of manufacture are confidential. The available information suggests that in practice blades in this material have to be made slightly thicker than optimum for castability. Failures are easier to contain that those of metal alloys because the material tends to fragment into many small particles rather than a few large pieces, and so the weight of the turbine casing can be substantially reduced.

CONCLUSIONS

Turbocharger turbines can be effectively characterized by stage loading and flow coefficients. For a conventional radial inflow turbine with radial section blades, the maximum stage loading consistent with high efficiency is about 0.95. Higher stage loadings require forward swept blades, which require a mixed flow turbine geometry if radial sections are to be retained, The amount of forward sweep obtainable is limited by the blade lean angle, but in practice rotor inlet blade angles of 20–30° are feasible.

By this means it is possible to increase the stage loading coefficient to about 1.15, or about 20% greater than that of a radial turbine. A design example shows that a high efficiency can be obtained at this level of loading, but a higher level of loading than this cannot be supported.

An alternative route to high turbine power output is in developing better rotor materials. The intermetallic titanium aluminide shows promise but its durability in service has yet to be proved.

REFERENCES

CHEN H, BAINES N C 1994 The aerodynamic loading of radial and mixed flow turbines. *Int Journ Mech Sci* **36**: 63-79.

BAINES N C 1996 Flow development in radial turbine rotors. ASME Paper no. 96-GT-65.

BAINES N C 1998 A meanline prediction method for radial turbine efficiency. In *Turbochargers and Air management Systems*, Inst Mech Engrs, pp. 45–56.

TSUEI H H, OLIPHANT K N, JAPIKSE D 1998 The validation of rapid CFD modelling for turbomachinery. In *CFD Technical Developments and Future Trends*, Inst Mech Engrs.

Table 1. Summary of highly loaded turbine designs.

	A	*B*
Stage loading coefficient	1.29	1.12
Flow coefficient	0.323	0.356
Nozzle exit blade angle (deg)	76.1	76.9
Rotor inlet mean radius (mm)	135.5	145
Rotor exit blade angle (deg)	-62	-60
Rotor inlet blade speed (m/s)	440	470.7
Rotor inlet relative flow angle (deg)	43.2	13.9
Rotor exit swirl angle (deg)	-11.5	-21.5
Rotor exit tip radius (mm)	107.24	101.16
Rotor exit hub radius	33.72	35.55
T-S expansion ratio	3.946	3.855
T-T expansion ratio	4.191	4.231
T-S efficiency	0.868	0.862
T-T efficiency	0.899	0.911
Blade speed ratio *U*/*C*	0.591	0.617

Table 2. Properties of turbine rotor materials.

Material (temperature)	***IN713LC (538°C)***	***IN718 (538°C)***	***CM MAR-M-247LC (427°C)***	***Ti6-4 (370°C)***	***γTiAl***
Yield stress (MPa)	760	1060	860	565	400-425
UTS (MPa)	895	1280	1020	680	450-600
Elongation at rupture (%)	11		9	18	3
Density (kg/m^3)	8000	8200	8500	4430	4400
Young's modulus *E*	183	200	700	114	159
Specific strength UTS/ρ	0.112	0.156	0.12	0.156	0.12
Specific stiffness *E*/ρ	0.023	0.024	0.062	0.026	0.036

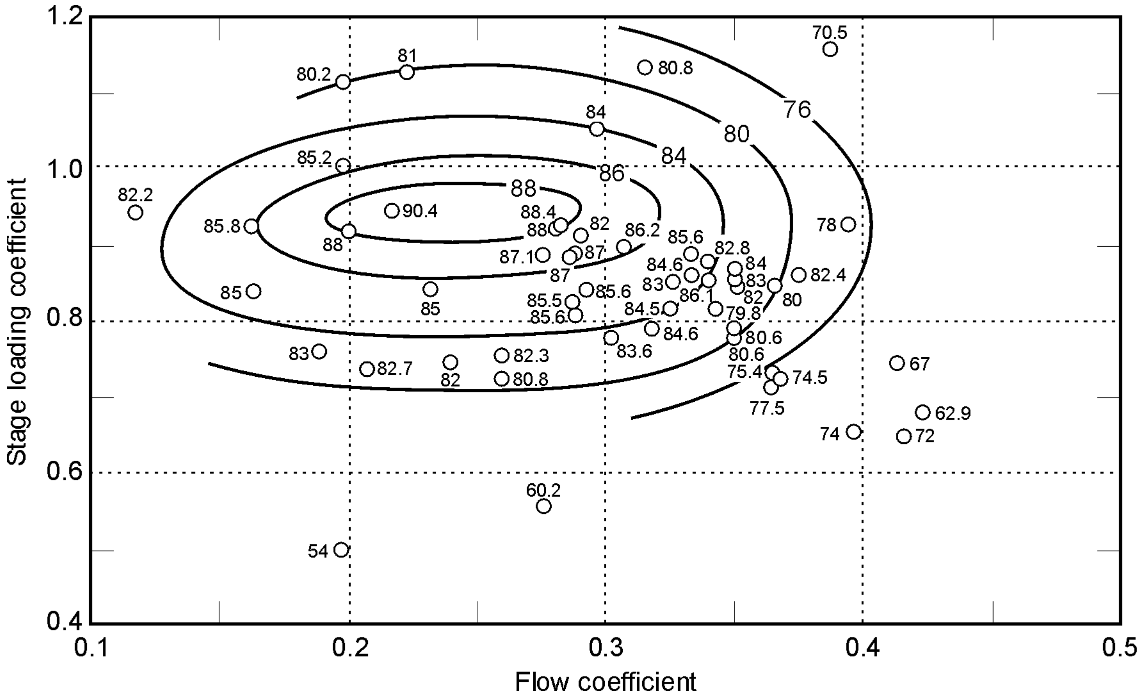

Figure 1. Correlation of measured efficiency of a range of turbine designs with stage loading and flow coefficients (Chen and Baines 1998).

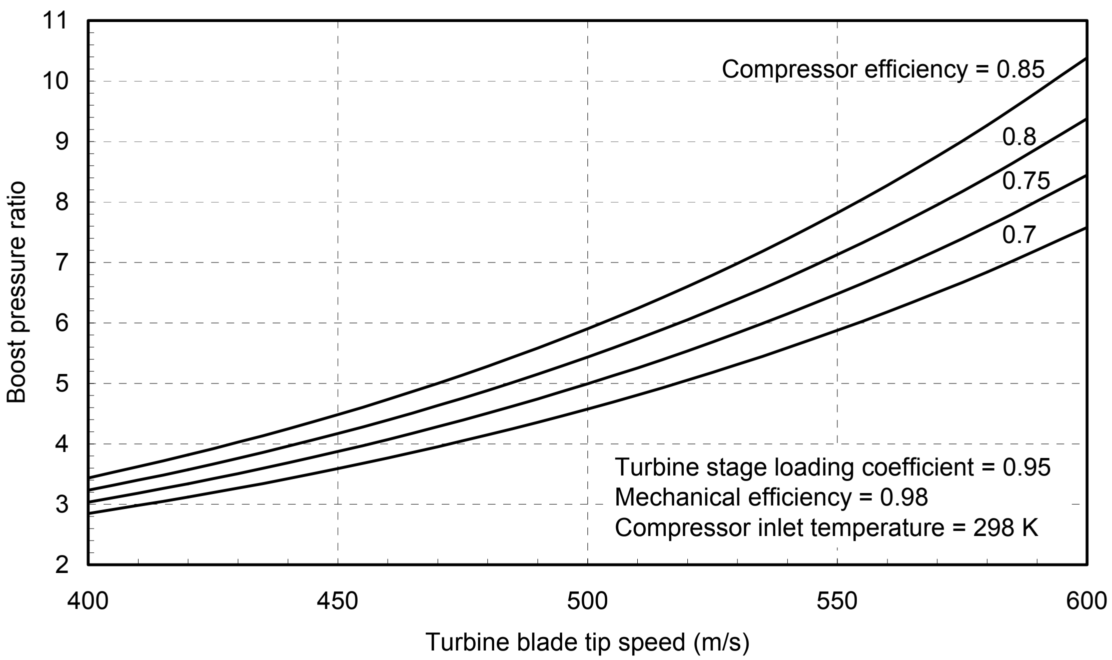

Figure 2. Compressor pressure ratio as a function of turbine blade speed and compressor efficiency.

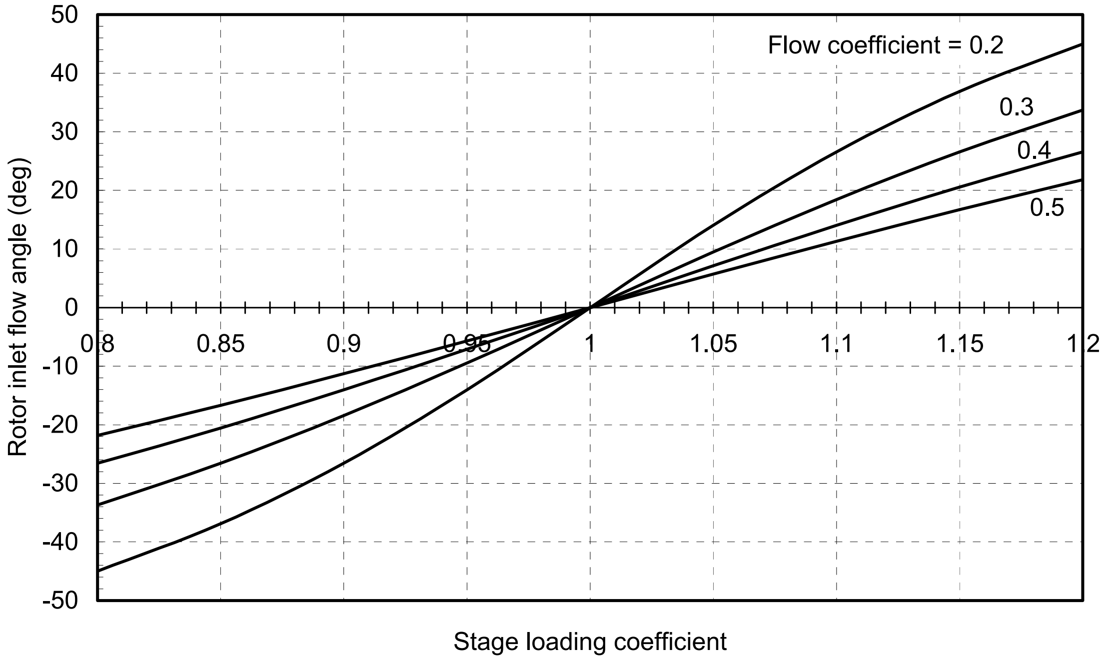

Figure 3. Rotor inlet flow angle as a function of stage loading and flow coefficients at the notional turbine design point.

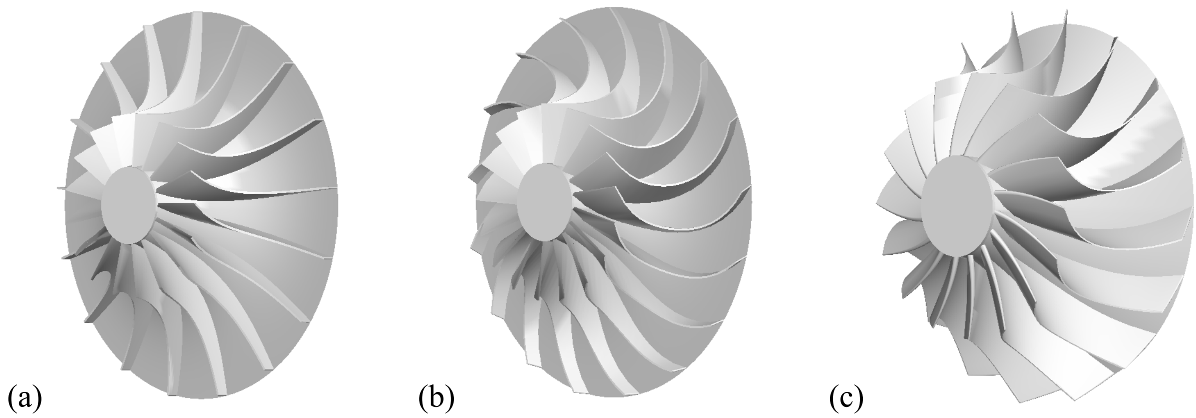

Figure 4. (a) Radial turbine with zero inlet blade angle and radial sections (b) Radial turbine with positive inlet blade angle and non-radial sections (c) Mixed flow turbine with positive inlet blade angle and radial sections.

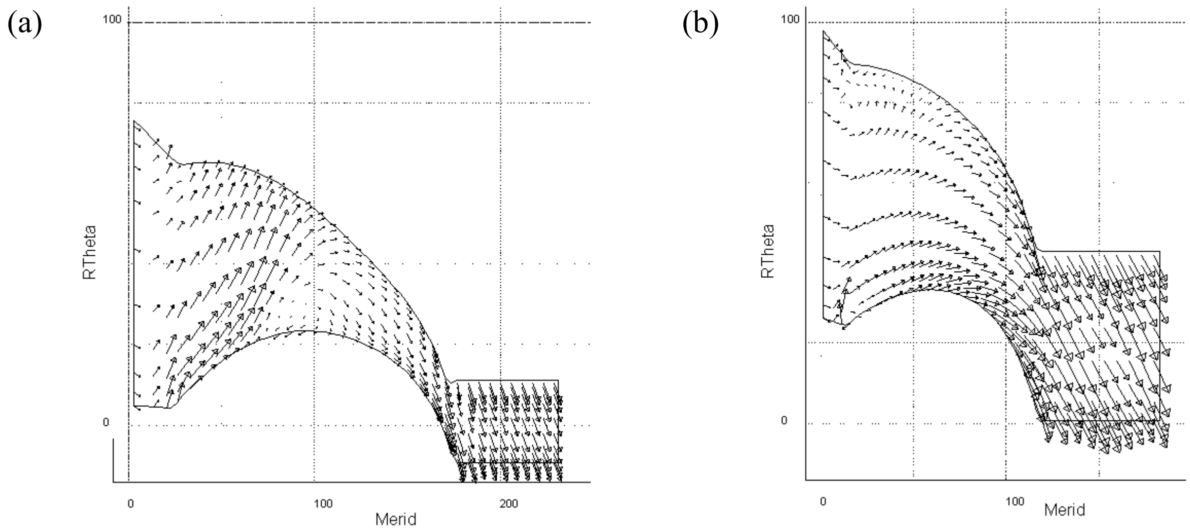

Figure 5. CFD solutions of highly loaded mixed flow turbine A on stream surfaces (a) near hub (b) near tip.

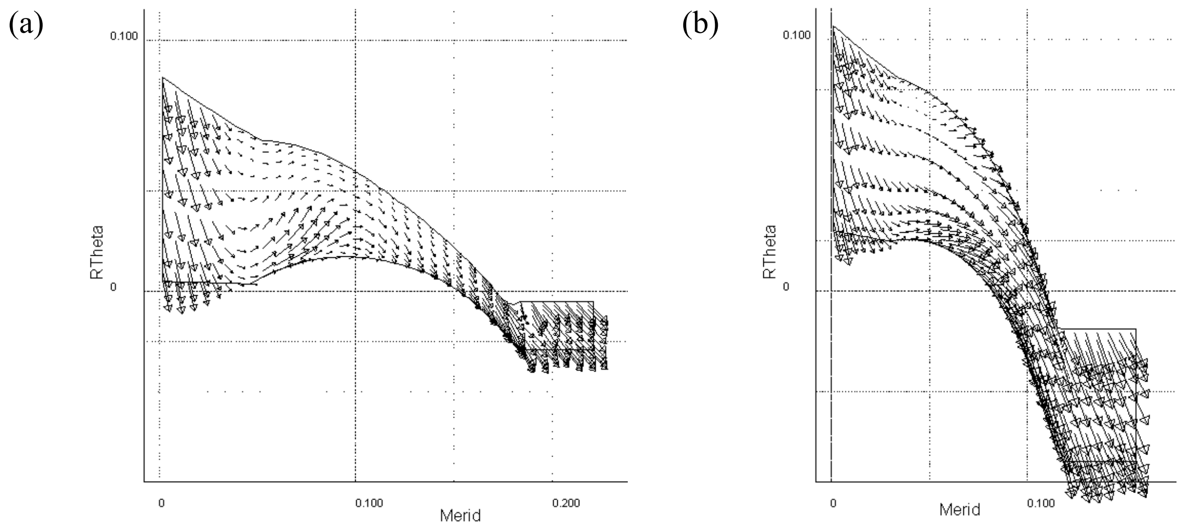

Figure 6. CFD solutions of highly loaded mixed flow turbine B on stream surfaces (a) near hub (b) near tip.