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SINGLE PASSAGE CFD ANALYSIS FOR NON-RADIAL FIBRE ELEMENT OF LOW PRESSURE TURBINE

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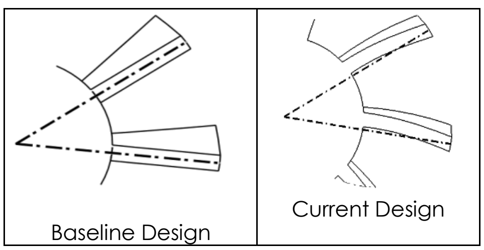
Graphical abstract



Abstract

Low Pressure Turbine (LPT) is a mixed-flow low pressure turbine meant for extracting energy from the exhaust of internal combustion engine. It converts the expanded exhaust energy into mechanical energy to drive an electric generator. The current available design of the LPT is only able to recover the exhaust energy efficiently with a pressure ratio range of 1.04 to 1.30. However, the performance efficiency deteriorates significantly when the pressure ratio exceeds 1.25. In the previous studies, flow field analysis has shown that the entropy is largely generated at the exit due to bigger vorticity. This vorticity can be minimized by optimizing the exit flow direction. This can be done by adjusting the exit camberline which reduces the deflection angle of the flow. This will effect exit flow of the fluid; subsequently reduces the exit loss as stipulated in the 1-Dimensional analysis of the turbine. Results have shown that the overall efficiency of the turbine has been improved as much as 7% at pressure ratios of 1.20. Its swallowing capacity is not largely affected at this point and its velocity ratio has shifted slightly from its design point of 0.70 to 0.65.

Keywords: Mixed-flow turbines, radial, non-radial, energy recovery, turbocompounding, low pressure turbine

1.0 INTRODUCTION

A mixed flow turbine is defined as a hybrid design between radial and axial turbine [1]. The schematic drawing that is shown in Figure 1 illustrates the difference that of a radial and a mixed flow turbine. The working fluid enters and meets the leading edge of the turbine at an angle between the radial and the axial turbine. Both radial and axial turbines share a common properties, that is the blades are modelled based upon radial fiber element. This means that the blades are radial in any section when it is cut normal to the rotation axis of the turbine [2]. This design is illustrated in Figure 1, of Baseline Design. This design is utilized widely in radial turbine due to the mechanical constraints in manufacturing a turbine with required strength. This leads to zero angle on the blade inlet which then leads to unfavorably high incidence angle [3, 4]. To improve the overall efficiency of the turbomachinery system, mixed flow is now being employed. The key advantage that the mixed-flow turbine has over radial turbine is that, it is able to have an additional degree of freedom towards the incidence angle [5]. Chen and Baines [6] found that, the mixed flow turbine can be optimized simply by introducing a positive swirl and reduce the exit swirl angle. This was concluded after they have taken the internal losses into calculation.

Below the Baseline Design in Figure 1, is the Current Design which utilizes non-radial fiber element of the turbine. At the leading edge (A), the blade design is based on radial-fibre component. The blade becomes progressively non-radial midway (B) of the blade. Figure 1 (b) shows the trailing edge (C) of the non-radial design has become obvious, of which the blades are swept back. The purpose of this design is to address the issue of vorticity generated at the trailing edge since it is directly linked to the entropy generation of the turbine [5, 4]. The entropy generation is often linked to the overall efficiency of the turbine. This is achieved by reducing the blade angle at the shroud of the rotor blade.

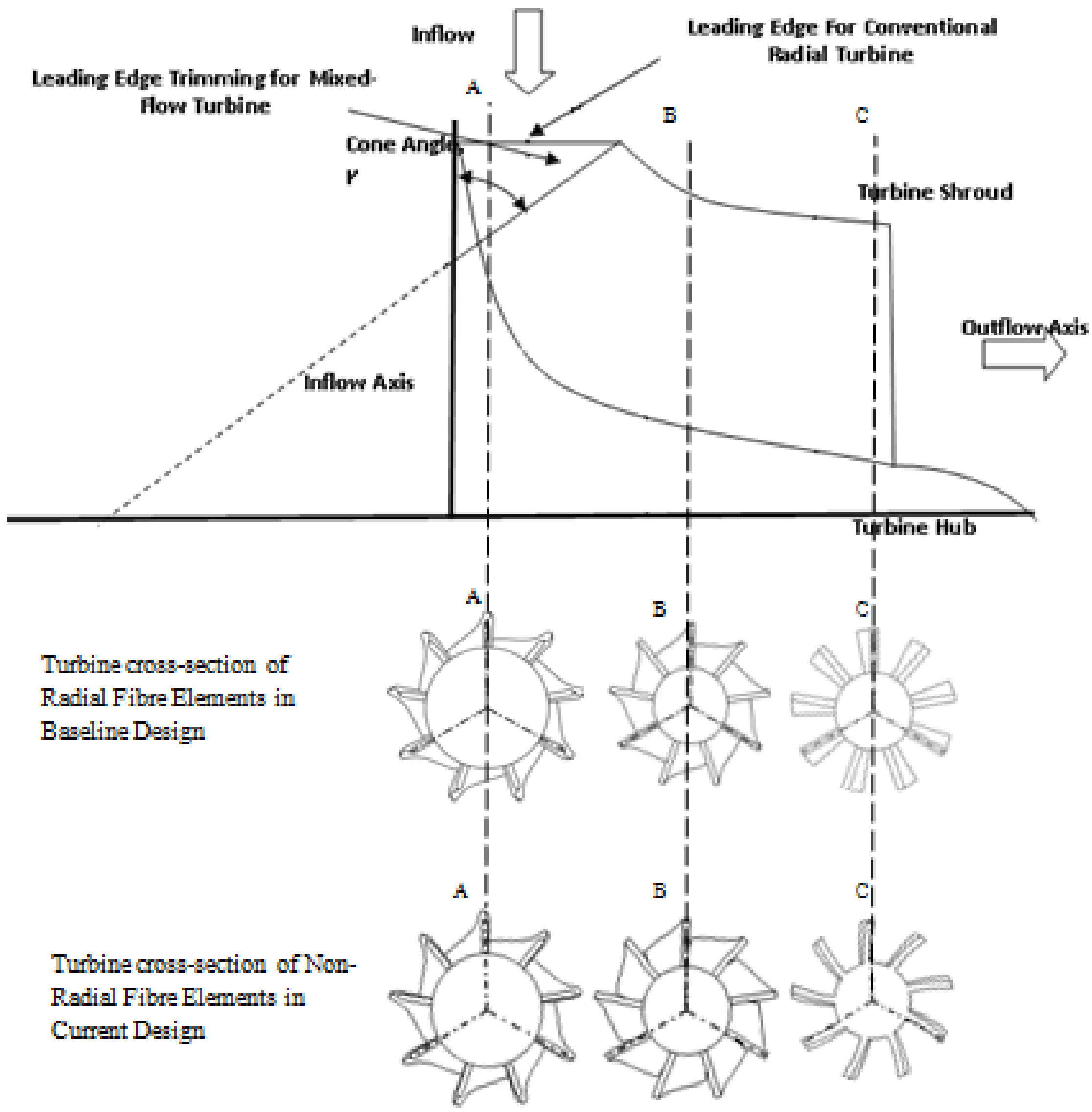


Figure 1(a) Schematic diagram of the fibre elements utilized in the Baseline Design and Current Design [3]

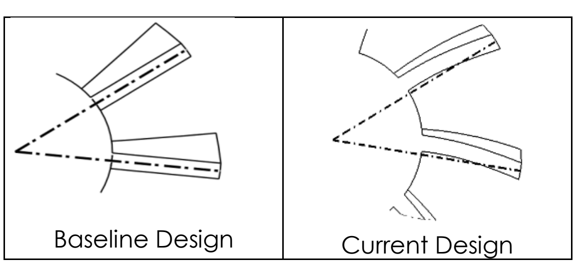


Figure 1(b) Enlarged cross-sectional view of the blades at trailing edge (C)

The turbine performance characteristics are commonly based on the following functional equation:

$$f\left(PR, \eta_{t-s}, \frac{\dot{m}\sqrt{T_{01}}}{P_{01}}, \frac{U_3}{C_{is}}\right) \qquad (1)$$

The PR refers to the total-to-static pressure ratio, of the turbine stage. The $\eta_{t\text{-}s}$ refers to the total-to-static efficiency of the turbine. The $\dot{m}\sqrt{T_{01}}/P_{01}$ is regarded as "pseudo non-dimensional" mass-flow parameter (MFP). While the U_3/C_{is} is the velocity ratio (VR). The turbine performances are usually plotted on the performance map. The turbine efficiency, $\eta_{t\text{-}s}$ is mapped against the pressure ratio, PR and also the velocity ratio, VR. As for the mass-flow parameter, MFP, it is mapped against the PR. The performance maps are useful for mapping the turbine corresponding applications such as 'engine-turbocompounding'.

2.0 CFD SIMULATION SETUP

2.1 Turbine Model Design

Currently, the turbine is designed with seven profiles to make the turbine blade as shown in Figure 1. The leaning of the turbine blade is achieved by varying the camberline angle from hub to shroud as shown in Figure 2. The leading edge is largely unaffected and it is maintained at -10°. The trailing edge of the turbine blade is varied from -45° at the hub to -36° at the shroud. Table 1 is the specification of the turbine used in this simulation.

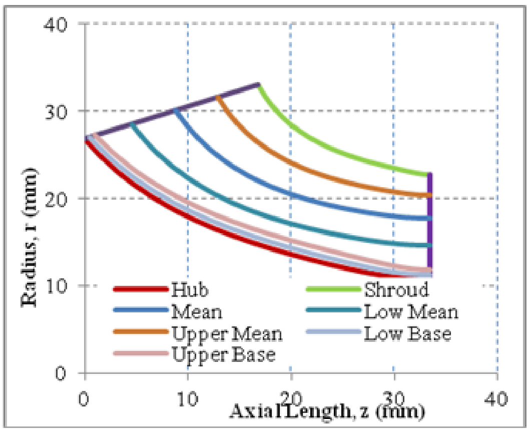


Figure 2 Profiles of the LPT camberline

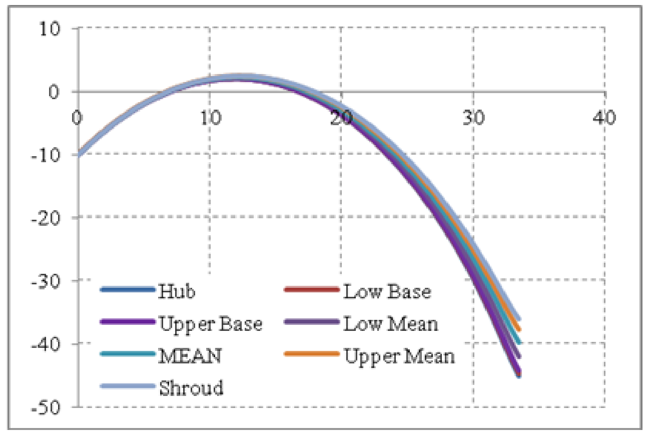


Figure 3 Camberline angle for blade profile

Table 1 Turbine characteristics

Characteristics	Quantity
Number of Blades, Z	9
Leading Edge Root Mean Square Radius, $c_{3,rms}$	42.2 mm
Trailing edge tip radius, d_4	22.7 mm
Cone angle, γ	20^0
Rotor blade length, l	33.5mm

Table 2 Single passage mesh statistics

Domain	Nodes	Elements
Inlet	9936	8580
Passage	92974	84048
Outlet	15309	13520

2.2 Turbine Meshing Element

The turbine meshing element is modeled based upon the Turbogrid. The meshing method used is APM Optimized. This meshing method is optimized at the wall using the y+ method. Table 2 shows the amount of nodes and elements used in each component which is used in this analysis. The total number of elements used in this simulation is 106,304 elements.

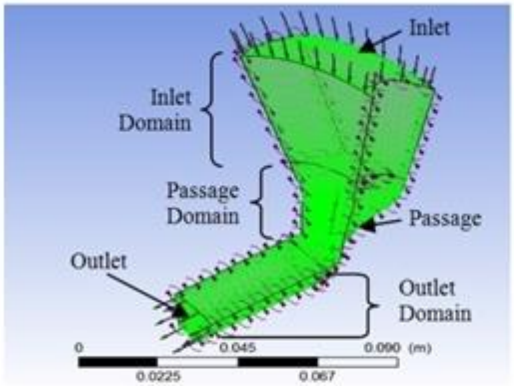


Figure 4 Meshing of the turbine and boundary interface

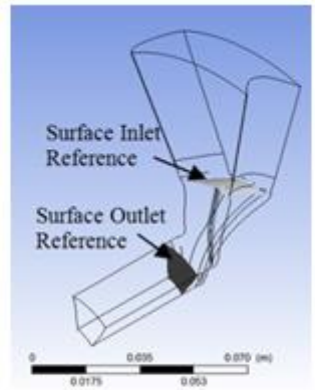


Figure 5 CFX post processing

2.3 Computer Modelling

In this setup, the boundary conditions are set accordingly. For the inlet boundary condition, the swirling angle is set to 77°. Here, it is assumed that there are only radial and tangential components. The axial component is assumed to be 0. The working fluid is assumed to be an ideal gas with the specific heat at constant pressure, C_p to be at the average value of $C_{p,average}$ = 1533 J/kg/K. The pressure range that was simulated varied from 1.03 to 1.80 by manipulating the mass flow rate at the inlet. The exit condition of the turbine was set to P_{exit} = 100 kPa and T_{exit} = 300K. The analysis was conducted at 3 operating speeds; 40,000 rpm, 50,000 rpm and 60,000 rpm. This corresponds to 80%, 100% and 120% of the designed speed. The turbulence model used in this setup is the k-epsilon because the fluid flow in the passage is turbulent [7]. Figure 4 shows the setup used in this simulation. The number of iterations used in this analysis was set to 300 and the control variables are set to 1×10^{-7} for greater precision. The results were then exported using the export command to analyze the performance of the turbine rotor. The point of where the data was taken is shown in Figure 5.

3.0 RESULTS AND DISCUSSION

The results have shown that there are improvements in this term, when they are compared against the same speed of the previous design as shown in Figure 6. Although there is limited difference at the lower PR, the difference becomes significant as the PR increased by as much 7%. The total-to-static efficiency, $\eta_{t\text{-}s}$ of the turbine varies according to the operating speed and the Pressure Ratio, PR as indicated by the graphs in Figure 7. As for the velocity ratio, VR where it is shown in Figure 8, the turbine peak efficiency has shifted slightly from 0.70 to 0.65. This shows that the turbine performance has changed slightly. This result coincides with the increase of efficiency at higher PR. The change is due to the leaning of the blade at the trailing edge have effected the blade close to the leading edge where it has swept itself instead of having radial fibre element [8]. Figure 9 shows the peak efficiency of the Turbine at various points of VR. It moves towards the left as the turbine speed is increased.

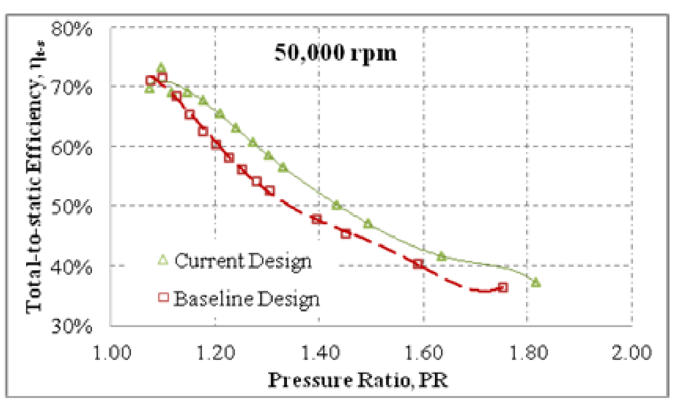


Figure 6 Comparison of $\eta_{t\text{-}s}$ between current and previous design at 100% design speed

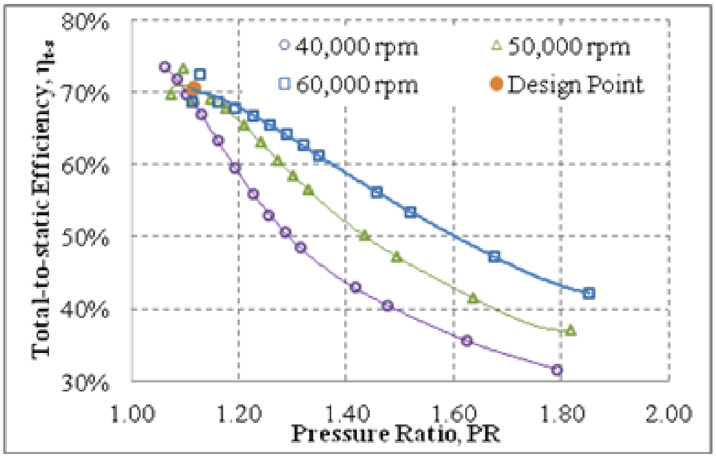


Figure 7 η_{t-s}vs PR for current design

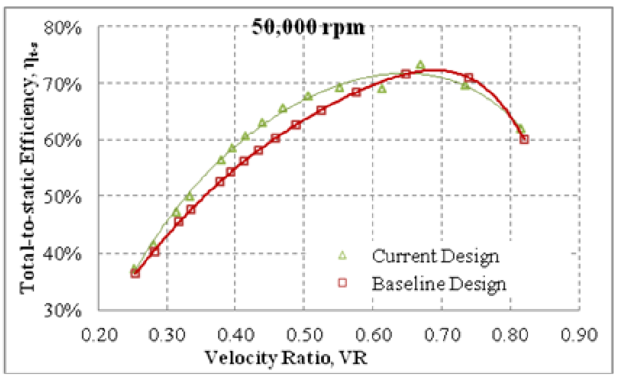


Figure 8 Comparison of VR between current and previous design at 100% design speed

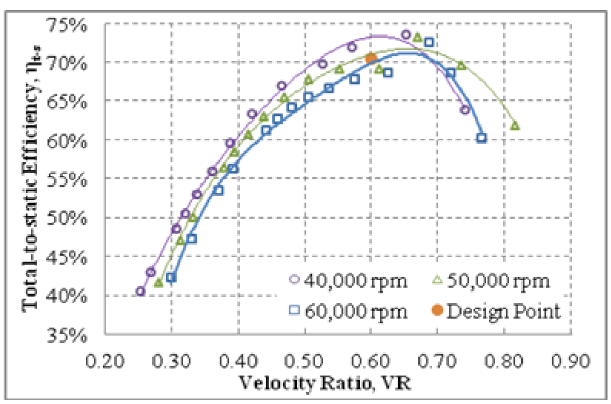


Figure 9 η_{t-s}vsVR for current design

Another aspect of turbine performance map is the Mass Flow Parameter, MFP. The MFP is plotted against the PR. Figure 10 and 11 show that the MFP reaches its swallowing capacity limit as the pressure ratio is increased. This indicates that the turbine reaches its capacity limit and it is experiencing choking whereby it is defined as a limiting condition of mass flow [9]. Despite the increase in the PR, the mass flow would not increase because the fluid velocity is approaching the speed of sound at the exit plane. Comparing between the current design and previous design, it has lower swallowing capacity. This indicates that, the trailing edge of the turbine has slightly increased the effects of choking, but improves the overall efficiency of the turbine at higher operating range.

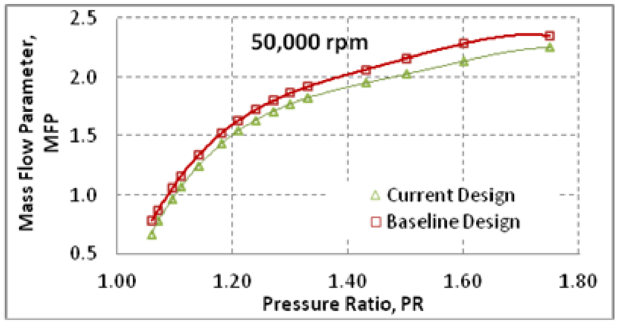


Figure 10 Comparison of MFP vs PR between current and previous design

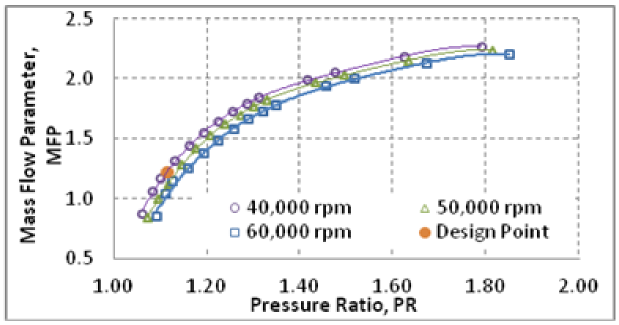


Figure 11 MFP vs PR for current design

4.0 CONCLUSION

An analysis of performance has been carried out to compare the new design with the old design of the turbine. The simulation is carried out at 3 operating speed of the turbine; 40 000 rpm, 50 000 rpm and 60 000 rpm. The mass flow of the turbine is varied to simulate the pressure ratio of the turbine. The performance maps have shown that the overall efficiency of the turbine is improved at higher pressure ratios by as much as 7.02%. This shows that the trailing edge has mitigated the exit loss of the turbine. Its MFP, however, has reduced slightly when compared against the previous design. This could be attributed to the leaning of the blade has changed the MFP characteristics of the new turbine.

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