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PHASE SHIFT METHODOLOGY ASSESSMENT OF AN AUTOMOTIVE MIXED FLOW TURBOCHARGER TURBINE UNDER PULSATING FLOW CONDITIONS

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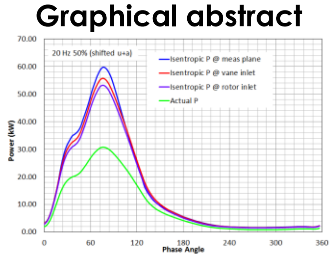
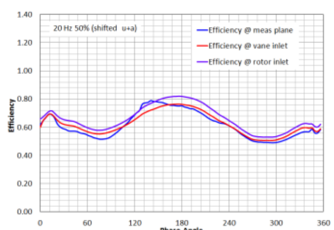
Graphical abstract





Abstract

The reciprocating nature of an Internal Combustion Engine (ICE) inevitably results in unsteady flow in the exhaust manifold. In a turbocharged engine, it means that the turbine is subjected to highly pulsating flows at its inlet. The finite time taken by the travelling pressure waves necessitates the need for phase-shifting method before any instantaneous parameter can be analyzed. In a turbocharger test-rig where the instantaneous isentropic power is evaluated upstream of the instantaneous actual power, one of the parameter has to be time-shifted in order to obtain meaningful instantaneous turbine efficiency. This research aims to compare two different methods of phase shifting which are by peak power matching and summation of sonic and bulk flow velocity. In achieving this aim, Computational Fluid Dynamics (CFD) models of full stage turbine operating at 20 Hz, 40 Hz, 60 Hz and 80 Hz have been developed and validated. Instantaneous efficiency was calculated at different locations and the order of calculated efficiency throughout the pulse is analyzed. Results have shown that phase shift using summation of sonic and bulk flow velocity indicated more reasonable efficiency values, thus the method could be used with high confidence for analysis involving unsteady turbine performance.

Keywords: Phase-shift, mixed-flow turbine, isentropic power, actual power, sonic flow

1.0 INTRODUCTION

Although there are many CFD works conducted to simulate the turbocharger turbine and enhance its development, relatively little work has been done to simulate and analyse its behaviour under pulsating flow conditions. Perhaps the earliest successful pulsating flow CFD simulation work was that of Lam *et al.* [1] in 2002. The main aim of the work was to demonstrate the capability of CFD to provide a sufficient description of the flow within the nozzles and the turbine passage. Lam *et al*. utilized the Multiple Rotating Frames (MRFs) method which is also known as the `frozen rotor approach' that assumes no relative movement between stationary and rotating frames of reference during the simulations. They also indicated that the MRFs method was only valid for steady state conditions where the flow at the interface between stationary and rotating domain is assumed to be relatively uniform. For unsteady calculations, the assumption of relatively uniform flow was only valid if the time scale of the transient effect (pulsating frequency) is relatively large compared to the timescale of the rotating body (rotor rotation). Therefore Lam *et al.* justified this method for their unsteady calculation since they applied a pulse frequency of 53.33Hz which was much smaller than the turbine frequency of 2267Hz. This work also highlighted a few difficulties in order to perform and evaluate unsteady turbine performance. One of the issues was the difficulty of getting the solution to converge adequately, therefore only qualitative comparisons could be made. Furthermore, the complex geometry of the turbine itself presented a difficulty in defining the exact entry point of unsteady flow at the rotor inlet. Despite that, the work of Lam *et al.* proved that it was possible to further understand the complexity of pulsating flow by means of full 3-D CFD.

Palfreyman and Martinez-Botas [2] later continued the analysis [3] by including the pulsating pressure profile as the boundary condition at the domain inlet. The main feature of this simulation is that the interface between stationary and rotating domain was defined as `sliding-plane' where there were relative motion between the domains on both sides of the interface at each time steps. Moreover, Palfreyman and Martinez-Botas indicated that the employment of this method ultimately improved the model capability to capture the hysteresis loop of mass flow rate and efficiency appropriately without the degree of damping observed by Lam *et al.* [1]. The predicted static pressure at several locations within the domain also compared well with the experimental data. The results of this work indicated that the flow field in the turbine passages is highly disturbed, and that the effect of the blades passing the volute tongue had most effect in the inducer region. Furthermore, Palfreyman and Martinez-Botas also concluded that the poor flow guidance observed particularly at the inlet and exit of the turbine was due to the assumption of quasi-steady conditions during the design stage of the particular turbine rotor.

More recent work on pulsating flow 3D simulation was conducted by Copeland *et al.* [4] in 2012 on a double entry turbine. This work attempted to characterize the level of `unsteadiness' within the turbocharger turbine stage by defining a parameter, lambda, λ that represents the ratio of the time-averaged rate of change of the mass flow within the domain to the time-averaged of the through flow. Therefore the flow inside the domain is considered steady when the λ value is 0 and that it is unsteady when λ is 1. This work also hinted that the rotor stage is not wholly quasi-steady but is insignificant enough as compared to the volute stage. The definition of λ is particularly useful since it depends not only on the flow frequency but also its amplitude.

Building on the work of Copeland [5], Newton [6] in 2014 investigated different vane geometries to obtain the best arrangement to suit unsteady flow operation, through a combination of numerical and experimental work. The 3D models were fully validated with excellent agreement in instantaneous mass flow predictions. Perhaps the most interesting part of this work is the comparison between steady and unsteady entropy generation characteristics. Newton concluded that the overall entropy generation in the pulsating case was 1.66% higher than the corresponding steady state condition, and could be attributed primarily to an increase in entropy generation in the nozzle-rotor interspace region. Newton also supported the findings of Copeland et al. that the rotor is not completely quasi-steady but remains very close to it.

The limited number of 3D simulation on turbocharger turbine under pulsating flow conditions is likely due to the high computational cost as well as the difficulties in obtaining experimental data for validation purposes. Without further full 3D simulations to investigate the phenomena, it is unlikely that the design of turbocharger turbines that take advantage of pulsating flow will improve.

Another major issue in the analysis of turbine performance under pulsating flow conditions is phase-shifting. Since the pressure pulse travels through the turbocharger turbine in a finite amount of time, an appropriate phase-shifting technique has to be applied in order to ensure that the ratio between the isentropic power available at the turbine inlet and the actual power extracted at the turbine shaft is calculated at the same phase instant. Over the years, different phase-shifting techniques were utilized by different researchers. For instance, Dale and Watson [7] and Arcoumanis *et al.* [8] used the sonic velocity for phase-shifting. Meanwhile Winterbone *et al.* [9] and Baines *et al.* [10] used the bulk flow velocity on which to base their phase-shifting adjustment. Recent studies by Rajoo and Martinez-Botas [11] and Szymko *et al.* [12] indicated that the sum of sonic and bulk flow velocities can also be used as the pulse travelling speed.

Another way of phase-shift is to simply match the peak of isentropic and actual power. As phase-shifting is central to obtaining accurate instantaneous parameters such as efficiency and pressure ratio and that the improvement of turbine design would depend on this issue, the current research aims to compare two phase-shifting methods which are phase-shifts using peak power matching and using summation of sonic and bulk flow velocities. The first method is chosen due to its simplicity while the latter is chosen as it is the most recent technique available in the literature.

To achieve this, a full 3D computational model of turbocharger turbine under pulsating flow conditions are modelled and experimentally validated using the turbocharger cold flow test rig available at Imperial College London.

2.0 METHODOLOGY

2.1 Experimental Setup

Figure 1 shows the schematics of the turbocharger test facility used in this research. The facility is located at Imperial College London for cold-flow testing and can be used for steady and pulsating flow testing. The compressed air for the test rig is supplied by three screw-type compressors with capacity of up to 1 kg/s at maximum absolute pressure of 5 bars. The air is heated by a heater-stack to temperatures between 330K and 345K to prevent condensation during gas expansion in the turbine. The flow is then channelled into two 81.40mm diameter limbs, namely outer and inner limb due to its relative position. This enables testing not only for single entry turbines but also for double or twin entry turbines. The mass flow rate in both limbs is measured using orifice plates. Downstream of the orifice plates is a pulse generator originally designed by Dale and Watson [7] in 1986. The pulse generator enables actual pressure pulses in the exhaust manifold to be replicated in the facility with frequencies up to 80Hz. Downstream the pulse generator is the 'measurement plane', where all the parameters for the turbine inlet were acquired. This includes instantaneous total and static pressure sensors and thermocouples. A hotwire anemometer is also installed at the measurement planet to measure instantaneous mass flow rate.

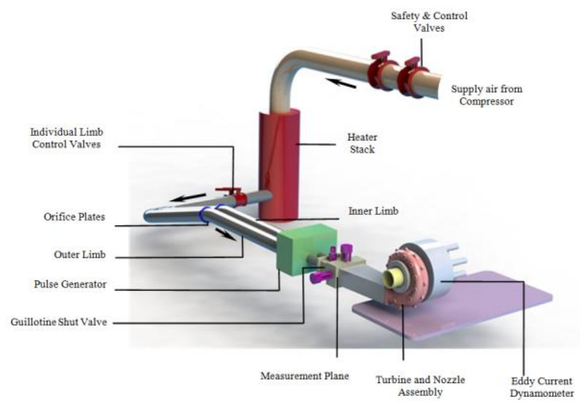
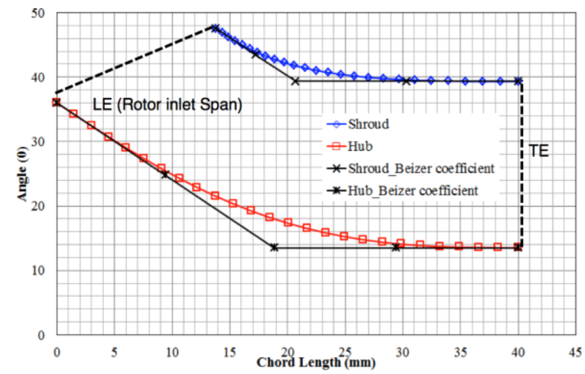


Figure 1 Imperial College 'cold flow' turbocharger test facility

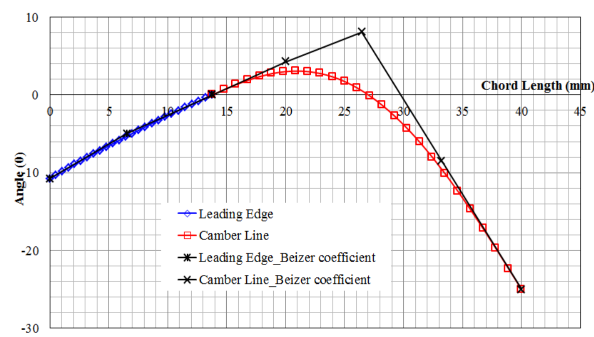
The turbine is attached to a 60kW eddy current dynamometer where it is placed on a gimbal bearing. The details of the dynamometer design are discussed by Szymko [12]. The reaction force on the dynamometer assembly is measured by a 20kg load cell where the torque is then calculated. The dynamometer also places a high flow rate water cooling system to disperse excessive heat absorbed by the magnetic plate. An optical sensor for instantaneous speed measurement is also installed within the dynamometer assembly. The design condition of the current turbine is 60000 rpm.

2.2 Numerical Setup

The simulation works conducted in this research were executed using commercial software Ansys CFX 14.1. The 3-Dimensional turbocharger turbine geometry consists of 4 main components which are the inlet duct, a modified Holset H3B turbine volute [13], 15 NACA 0015 profiled vanes and a mixed flow turbine rotor with 40mm chord length. The inlet duct and the volute were constructed using Solidworks and meshed using Ansys ICEM CFD. For the nozzle stage, 15 lean vanes were constructed by importing 3 profile lines into TurboGrid software where structured hexahedral mesh is automatically generated. Similar method is used to mesh the mixed flow turbine except 8 profile lines are needed due to its more complex geometry. The profile lines of the turbine blade were created using Bezier polynomial where its control points are shown in Figure 2. The turbine used is an in-house turbine designated Rotor A created by Abidat [14] in 1991 for applications in high loading operations. Figure 2(a) shows the resultant polynomial lines that form the hub (red line) and shroud (blue line) of the rotor wheel. The dotted line in Figure 2(a) indicates the imaginary position of the leading and trailing edge of the rotor. It can be seen that the overall chord length is 40 mm. Figure 2(b) shows the curvature for leading edge and camber line of the blade.



(a)



(b)

Figure 2 Development of turbine geometry using Bezier Polynomial

Subsequently, all meshed components were assembled in Ansys CFX-Pre as shown in Figure 3. The total number of mesh elements is 4.2 million. Distribution of nodes and its type is shown in Table 1. The interfaces between each component are specified during this stage. The interfaces between inlet duct and volute, and also between volute to vane are specified as general connection as shown in Figure 3. Transient rotor-stator interface is specified between stationary and rotating domain to enable physical movement of the rotor for each time step. The specified time step is 1 degree of rotor rotation per time step. Data is extracted for every 90° turbine rotation.

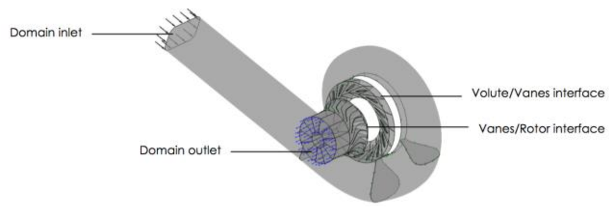


Figure 3 Assembly of domain in CFX-Pre

At the domain inlet (inlet duct), time varying total pressure and total temperature are specified with the turbine speed of 30000 rpm. 4 different frequencies are simulated, namely 20 Hz, 40 Hz, 60 Hz and 80 Hz. The original values for the boundary conditions were extracted from the experimental results. The direction of inlet flow is defined so that the only velocity component that exists is normal to the inlet plane. The outlet boundary condition requires the static pressure value. For this purpose, a constant atmospheric pressure is applied at the domain outlet. No-slip boundary condition is specified at all walls including vanes and rotor blades. Furthermore, k-ε turbulent model is utilized due to its balance on accuracy and efficiency of the simulation.

Table 1 Distribution of nodes and type of mesh

Domain	Number of Nodes	Type of mesh
Inlet Duct	242 320	Unstructured Hexahedral
Volute	730 016	Unstructured Hexahedral
Nozzle	560 160	Structured Hexahedral
Rotor	2 600 436	Structured Hexahedral
Total	4 162 932	

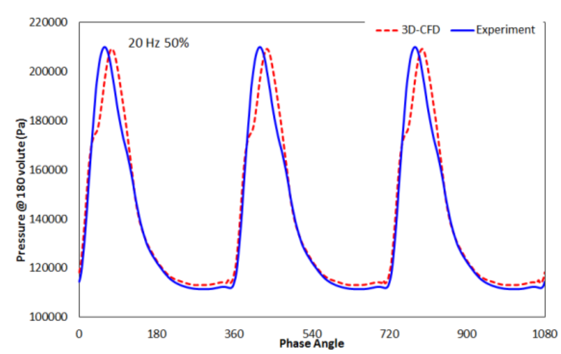
3.0 RESULTS AND DISCUSSION

3.1 Validation Exercise

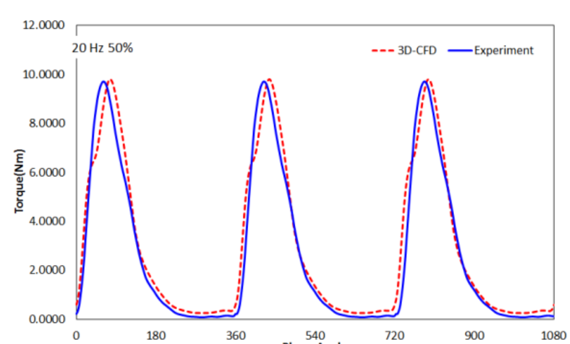
The simulation involving pulsating flow with transient interface are very extensive and computationally expensive where more than 2 terabyte of post-processing data were gathered for total 4 simulations. All pulsating conditions were validated with available experimental results. To demonstrate the validation exercises undertaken for these simulations, a single turbine operating frequency of 20 Hz is selected. The validation for pulsating turbine condition is focused on fundamental parameters rather than derived parameters. In this case, two parameters namely static pressure at 180° volute circumference and the turbine rotor torque have been selected. The selection of fundamental parameters instead of the turbine performance parameters is done in order to validate the parameters obtained by CFD to the parameters that are measured during the experimental works. This ensures that any deviations between CFD and experiment can be seen in time domain and a potential source of error could be determined.

Figure 4(a) shows the plot of static pressure at the centroid 180° volute circumference for both experiment as well as CFD calculation. It can be seen that CFD calculation is able to pick up the pressure trace well during the pressure increment and decrement instances, as well as the trough region of the pulse. In addition the calculated static pressure range also matches the experimental data well. However, at the pulse peak, CFD calculation shows some deviation in terms of the phase of the peak pressure. The peak pressure as indicated by CFD occurs about 10° phase angle later than that of experimental plot. Nevertheless, the magnitude of both peaks is in good agreement to each other.

Following the validation of pressure trace in the volute stage, another parameter which is directly related to the turbine wheel, namely torque is plotted in Figure 4(b). Overall, the CFD calculation agrees well with experimental data. The shifting of peak position that is seen previously in Figure 4(a) can also be seen in the torque trace while the magnitude of peak torque is still in agreement to each other. The validation of both parameters, despite slight shifting in the peak region still indicate that the developed model is able to capture the time dependant pulsating flow within the turbocharger turbine stage. Therefore, the viability of the model for pulsating flow environment is confirmed and could be used for further analysis.



(a)



(b)

Figure 4 Comparison between CFD and Experimental data of (a) Static pressure at 180° volute circumference and (b) rotor torque

3.2 Phase-Shift Analysis

The need for phase-shifting is due to the different measurement location of the parameters required to evaluate the turbine instantaneous performance parameter. For instance, in order to obtain the turbine efficiency, the parameters needed are the isentropic and actual power. In the current research, the isentropic power is obtained at the measurement plane upstream the volute inlet, whereas the actual power is obtained at the rotor (see Figure 1). Therefore, in a pulsating flow environment, the finite time required for both pressure and mass to propagate from the measurement plane to the rotor needs to be taken into account during calculation of the turbine instantaneous efficiency. Otherwise, the calculated instantaneous efficiency does not correspond to the exact instance of when the available energy is being converted to the rotor actual power. This ultimately results in not only abnormal but also misleading efficiency values. The efficiency is calculated using equations 1 – 3.

$$\eta = \frac{W_{act}}{W_{isen}} \tag{1}$$

$$W_{act} = 2\pi N\tau \tag{2}$$

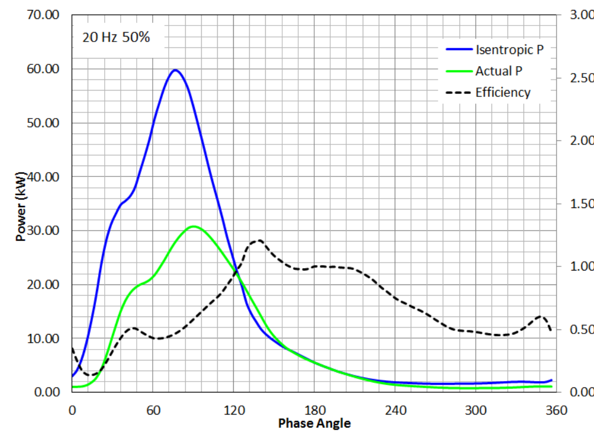
$$W_{isen} = \dot{m}c_pT_{01}\left[1-\left(\frac{P_5}{P_{01}}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{3}$$

Where W_{act} and W_{isen} is actual and isentropic power respectively, N is rotational speed, τ is torque, $\dot{m}$ is mass flow rate, T_{01} is total temperature at inlet, P_5 is exit static pressure and P_{01} is total pressure at inlet.

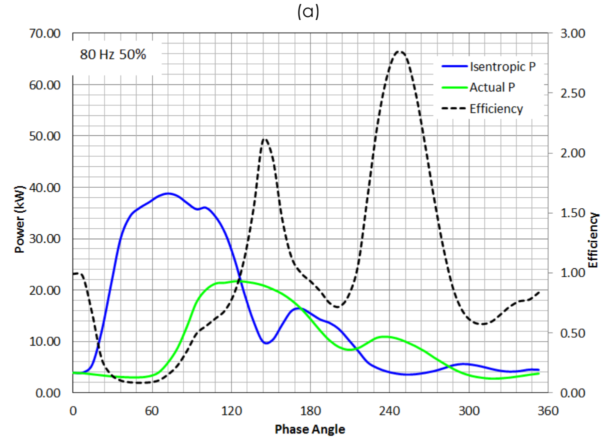
The effect of shifted phase during data acquisition of isentropic and actual power is clearly demonstrated in Figure 5, which shows the instantaneous isentropic and actual power of the turbine for 50% design speed (30000 rpm) at 20 Hz and 80 Hz without any phase-shifting methods applied. The black dotted line plotted on similar axis in Figure 5 is the corresponding instantaneous turbine efficiency. It is obvious in Figure 5(a) that the actual and isentropic power is not in the same phase where the actual power is shifted to the right due to the measurement delay of the travelling pressure waves. Furthermore, it can also be seen from the efficiency plot that the value is far greater than unity especially in Figure 5(b) as a result of inconsistencies in the phase on which the efficiency is evaluated. This is a clear indication that phase-shifting is needed in order to obtain accurate instantaneous turbine performance characteristics.

At the moment, there are already a few methods of phase-shifting techniques introduced by different researchers as detailed in the introduction section. However, a more simplistic method to correct for the phase different is to evaluate the time phase by matching the peak point of isentropic and actual power. The other promising phase-shifting approach is by using the summation of sonic and bulk flow velocity which has been used by Rajoo [13] and Szymko [15]. In this section, these two methods of phase-shifting will be evaluated.

In order to evaluate both methods, evaluation of isentropic power is made at three different planes, namely the measurement plane, the vane inlet and the rotor inlet as indicated in Figure 6. These evaluations are made possible by employing the available validated CFD results. The instantaneous isentropic powers available at different planes are area-averaged and serve as a comparison basis for calculation of instantaneous efficiency. At these



(a)



(b)

Figure 5 Isentropic and actual power as well as its instantaneous efficiency without phase-shifting at 30000 rpm for (a) 20 Hz and (b) 80 Hz

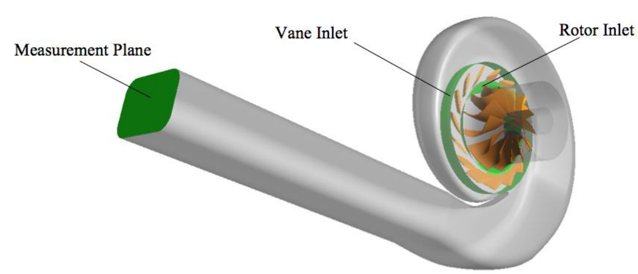


Figure 6 Isentropic power measurement locations

planes, the magnitude of calculated instantaneous efficiency should therefore be highest when evaluated at rotor inlet, followed by vane inlet and finally at the measurement plane. These efficiency values should be consistently in descending order ($\eta_{RotorInlet} > \eta_{VaneInlet} > \eta_{MeasurementPlane}$) throughout a whole pulse cycle if the phase-shifting is conducted properly.

Figure 7(a) shows the calculated isentropic and actual power at different locations without any phase shifting. It is worthwhile to note that the plot of actual and isentropic power at the measurement location is similar to that of Figure 5(a). The right hand side of Figure 7(a) shows the calculated instantaneous efficiency evaluated at different locations. It can be seen that not only the efficiency value is too high, but the values are also not arranged according to the descending order based on the locations of evaluation. If the phase-shifting procedure is done using the peak power matching (Figure 7(b)), the value and order of instantaneous efficiency at different locations improved greatly as indicated in the right hand side of Figure 7(b). However, there are still a few instances which are not in order. This irregularities are labelled X and Y in Figure 7(b). For the second method, where the phase-shifting is done using the summation of bulk and sonic velocity, the instantaneous efficiency plot improved even more as can be seen in Figure 7(c). Although, the irregularities of efficiency order at different location still exist.

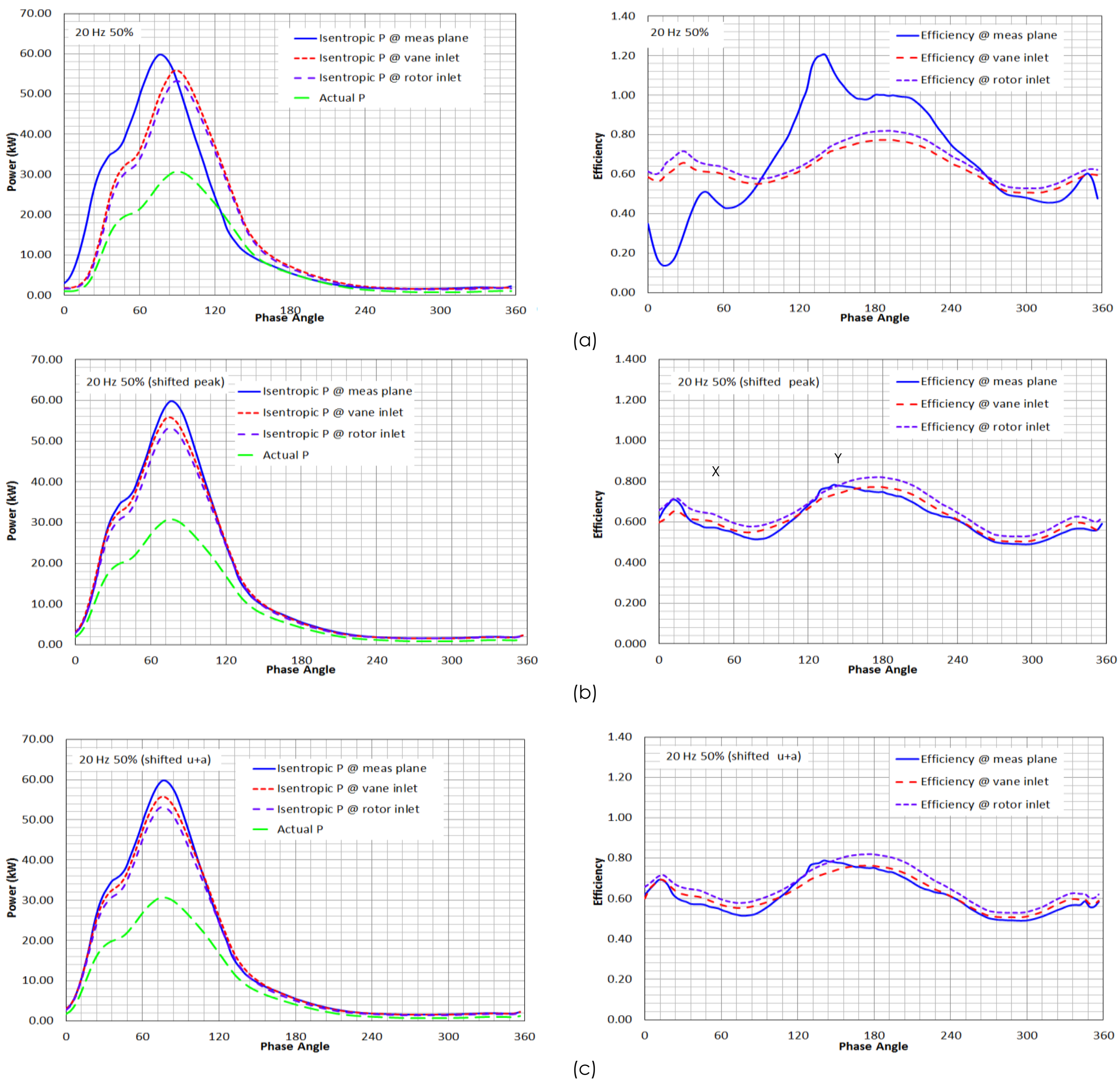


Figure 7 Total-to-static efficiency evaluated at different locations (a) without phase shifting, (b) phase shifted using peak power matching and (c) phase shifted using summation of bulk flow and sonic velocity.

There are several drawbacks that can potentially be deduced from both techniques. The use of the first method (peak power matching) can only be employed when the peak of isentropic and actual power is clearly visible. Therefore as the pulsating frequency increases, the interferences originated from pressure wave reflection and superposition could result in difficulties in locating this peak. The example of difficulties can be clearly seen in Figure 5(b) where at the frequency of 80 Hz, no obvious peak can be selected. For the second method (summation of bulk flow and sonic velocity), in order to be able to calculate the time shift, additional information is needed which is the location of entry point at the turbine inlet. At the moment, this point is assumed to be at 180° from the volute tongue. The assumption is based on the information from previous researches as well as evidence found in the current research where phase-shifting using this location as entry point yielded reasonable efficiency values [15]. Furthermore, a quick calculation using peak power matching Figure 7(b) revealed that the location of this entry point is 230°. However, this value seems not to be consistent throughout all tested frequencies.

Comparison between the two methods results in a definitive conclusion that phase-shifting using summation of bulk and sonic velocity is able to provide better results with less ambiguity. Therefore, this method could be used throughout the entire analysis involving instantaneous efficiency parameters.

4.0 CONCLUSION

The numerical model for a full stage turbocharger turbine has been successfully modelled and validated. For the pulsating flow conditions, the location on which the isentropic and actual powers are evaluated at different locations and therefore, phase shifting is necessary. The assessment of the phase shifting procedures is conducted where the shifting technique using the summation of bulk flow and sonic velocity has been proven to be superior then simply matching the peak of isentropic and actual power. This method is therefore recommended for future research particularly research involving turbine performance under unsteady flow conditions.

Acknowledgement

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