

Imperial College London

The Evaluation of Steady and Pulsating Flow Performance of a Double-entry Turbocharger Turbine

This thesis is submitted for the award of Doctor of Philosophy (Ph.D.)
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December 2009

I declare that the work in this thesis is my own and relevant citations are included to acknowledge the work of others

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Abstract

The turbocharger remains one of the best means available to the engine developer to satisfy the power density demands on a modern internal combustion engine. This simple device uses the otherwise waste exhaust gas energy to provide significant improvements in the volumetric efficiency or 'breathing capacity' of an engine. In order to maximize the energy of the exhaust driving the turbine, most applications utilize *pulse turbocharging* where a compact exhaust manifold feeds the highly pulsating exhaust flow directly into the turbine wheel. This thesis considers the influence that this pulse-charging has on a double-entry turbocharger turbine.

The design of this turbine plays an important role in much of the research presented in this thesis. The turbine is equipped with a mixed-flow rotor with 12 blades that are fed by a 24 blade nozzle ring. The circumferentially divided volute is designed with two gas inlet passages that each feed a separate 180° section of the nozzle ring. Thus, there is no communication between the entries from the volute inlet to the exit of the nozzles. At the exit to the nozzle, the fluid from both inlets expands into an interspace that spans the circumference of the rotor inlet. This small volume that is formed between the nozzle and the mixed flow rotor is the first area where interaction between the flows can occur.

The core of this report contains three main divisions: Steady flow experimental results, CFD modelling, and unsteady flow experimental results. These sections are preceded by an introduction explaining the background of the research study, and an essential outline of the equipment and the method of experimentation. The aim of this work is to use a combination of experiments and computational modelling to build up a picture of the performance of the turbine under a wide variety of flow conditions that will eventually lead to further insight into its unsteady performance.

First, a comprehensive steady-state experimental data set was obtained to establish the base-line turbine performance. Steady, equal admission tests yielded excellent performance, peaking at 80% efficiency. Owing to the double-entry arrangement, steady flow could also be introduced in the two inlets unequally. During unequal, steady-state operation a notable decrease in performance was observed. The correlation between the ratios of entry pressures and the efficiency of operation was apparent but essentially independent of which flow was varied. In the extreme, when the turbine was only partially supplied with air, the consequence was a 28 point decrease in performance at the optimal velocity ratio. Despite the division between the two entries, the experiments showed that the flows through each inlet were interdependent. Compared to full flow,

when the pressure in one entry was low, the second entry could swallow more mass, and when it was high, the second entry swallowed less.

A three-dimensional CFD model was constructed in order to permit a detailed study of the flow in the double-entry design and answer specific questions regarding the observed steady-state performance. For both equal and unequal admission simulations, the model showed close agreement with the experimental mass flow behaviour and reproduced the measured efficiency trends quite well. The interdependence of the swallowing capacity of the two inlets was also predicted by the model, thereby allowing the analysis of the physical flow effects that drive this trend. It was found that the interspace region near the tongues was the site of much of the interaction between inlets. A major emphasis of this modelling work was also to discover areas of loss generation that could lead to the decrease in performance. By focussing on partial admission, this study found that the windage loss in the interspace region of the non-flowing entry proved to be one of the more significant areas of loss generation.

Pulsating air flow was then introduced using the range of frequencies typically produced by an internal combustion engine. The operating point of the turbine, traced an orbit within a 3-D space defined by three non-dimensional parameters: velocity ratio, pressure ratio across inlet one, and pressure ratio across inlet two. Direct comparison between steady and unsteady values at the same pressure ratios and velocity ratio was possible due to the large amount of steady data measured. Thus, a quasi-steady versus unsteady comparison was made on the basis of efficiency, mass flow and output power. In general, under pulsating flow conditions, the turbine behaved quite differently than that predicted by the quasi-steady assumption. Lower frequency, higher amplitude pulsations produced the lowest unsteady cycle-averaged efficiency and also produced the most significant departure from quasi-steady behaviour.

Acknowledgments

I wish to thank my supervisor Dr. Ricardo Martinez-Botas for his support throughout my PhD. I also wish to thank the host of helpers at Imperial College particularly John Laker and Harminder Flora.

I wish to thank ABB Turbo Systems for funding this project and providing support throughout my PhD. Particular thanks to Martin Seiler for his advice and direction as well as Dietmar Filsinger for his help at the start of the project .

Very special thanks to my sister Jennifer for helping me with my punctuation and my parents Mike and Nancy for their continual support.

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Nomenclature

Abbreviations

BDC	Bottom dead centre
CAD	Computer aided design
CFD	Computational Fluid Dynamics
ICE	Internal Combustion Engine
TDC	Top dead Centre

M	Mach Number	$\frac{U}{a}$
MP	Mass parameter	$\frac{\dot{m}\sqrt{T_0}}{P_0}$
MSt	Modified Strouhal number	$\frac{fL}{\bar{U}} \cdot \frac{1}{\phi}$
PMSt	Pressure Modified Strouhal number	$\frac{fL}{\bar{U} + \bar{a}} \cdot \frac{1}{\phi}$
Pr	Prandtl number	$\frac{\mu c_p}{k}$
PR	Pressure ratio	$\frac{P_{01}}{P_{04}}$
RP	Ratio of inlet pressures	$\frac{P_{0,inner}}{P_{0,outer}}$
SP	Speed parameter	$\frac{N}{\sqrt{T_0}}$
VR	Velocity ratio	$\frac{U_2}{C_{is}}$

English

A	Area	[m ²]
a	Speed of sound	[m/s]
A _s	Effective isentropic flow area (Also use A _{eff})	[m ²]
C	Absolute Flow velocity	[m/s]
C _d	Discharge coefficient	
cp	Specific heat capacity for constant pressure	[J/kg/K]
d	diameter	[m]
D	Diameter	[m]
d ₂	Average turbine blade inlet diameter	[m]
E	Constant temperature anemometer voltage	[V]
f	frequency	[Hz]
h	Specific Enthalpy	[J/kg]
I	Moment of inertia	[kg.m ²]
I _m	Usteady Influence factor based on mass	-
I _p	Usteady Influence factor based on power	-
k	Thermal conductivity, Turbulent kinetic energy	[W/m/K], [J]
L	Length scale	[m]
$\dot{m}$	Mass flow rate	[kg/s]
n	Coefficient of wall roughness, Number of samples	-
N	Turbine speed	[rev/s]
P	Pressure	[Pa], [Bar]
q	Rate of heat conduction	[W/m ²]
$\dot{Q}$	Rate of heat input	[W/kg]
r	Radius, Recovery factor	[m], -
R	Specific gas constant, Radius of pipe	[J/kg/K]
S	Source term, Entropy	-, [J/K]
s	Specific Entropy	[J/kg/K]
T	Temperature, Total time interval	[K], [s]
U	Blade speed, Velocity	[m/s], [m/s]
u*	Mass flux	[kg/s/m ²]
U*	Total discharge mass flux	[kg/s/m ²]
U ₂	Average inlet blade speed, Blade tip speed	[m/s], [m/s]
V	Volume	[m ³]
v	velocity	[m/s]
W	Relative velocity	[m/s]
$\dot{W}$	Work rate	[W]

Greek

α	Absolute flow angle, Angular acceleration	[Degrees],[Rad/s ²]
β	Relative flow angle, Diameter ratio	[Degrees], -
γ	Ratio of specific heats	-
Γ	Diffusivity	[Pas]
δ	Gas deviation angle	[Degrees]
Δ	A small change	-
ϵ	Expansion factor, Turbulence eddy dissipation	-, [J/s]
κ	von Karman constant for wall bounded shear flow	-
θ	Azimuth angle	[Degrees]
λ	Bulk viscosity	[Pas]
μ	Dynamic viscosity	[Pas]
π	Ratio of circle diameter to circumference, Average pressure ratio	[3.14159],-
ρ	Density	[kg/m ³]
τ	Torque, Stress tensor	[Nm], [Pa]
Φ	Pulse length as a fraction of total wavelength	-
ω	Rotational speed	[Rad/s]

Non-Greek

$\emptyset$	Segment angle	[Degrees]
∇	Gradient operator	[1/m]
$\nabla \cdot$	Divergence operator	[1/m]
$\nabla \times$	Curl operator	[1/m]

Subscripts

0	Stagnation conditions, Volute inlet station
1	Nozzle inlet station, First inlet to volute
2	Rotor inlet station, Second inlet to volute
3	Rotor exit station
4	Exducer exit station
act	Actual value
avg	Average value
bulk	Bulk flow value
cal	Calibrated value
calib	Calibration coefficient
cold	At cold conditions
corr	corrected value

E	Energy
E1	First Entry
E2	Second Entry
eff	Effective
entry	At entry
f	Film value
fluct	Fluctuating component
gen	Generated
hot	At hot conditions
in	Inner limb entry
inner	Inner limb entry
inst	Instantaneous value
is	Isentropic process
isen	Isentropic process
M	Momentum
m	Meridional direction, measure value
meas	Measured value
o	Reference value
orif	Conditions at orifice plate
out	Outer limb entry
outer	Outer limb entry
QS	Quasi-steady
s	Static condition
T	Total
t	Turbulent
t-s	Total to static
US	Unsteady
w	Wire value
θ	Tangential direction

Superscripts

...	Ensemble averaged value
'	Fluctuating component
.	Derivative with respect to time
—	Average component
*	Non-dimensional value

Chapter 1

Introduction

1.1 Perspective

Recognition of the environmental impact of greenhouse gases has led to policy initiatives aimed at tightening the transport emission requirements in recent years. Road transport is the second largest greenhouse-gas emitting sector in the European Union, with emissions continuing to rise [1]. Thus, any serious effort to tackle climate change and move toward a sustainable future must include engineering solutions aimed at reducing transportation CO₂ emissions. The electrification of transport provides one of the greatest potential for substantial decarbonisation in the transport sector. However, as has been pointed out by even the biggest proponents of the electric vehicles, there are currently a number of significant hindrances to their wide-spread adoption (technological, economical and infrastructural) [2]. For example, any shift towards on-board chemical storage currently poses many challenges, namely - current battery technologies are expensive, heavy and cannot provide the energy density needed for a range beyond city driving. Therefore, in the immediate future, CO₂ emission reduction must come about by *downsizing* the internal combustion engine (ICE). Aggressive downsizing is generally only possible by increasing the boost pressures of the intake system. Turbocharging and supercharging are both proven technologies in this area - with long histories of being applied to increase the power to weight ratio of the ICE. It is only more recently, however, that the motivation of climate change has led to an ever increasing interest in the importance of air management for producing a more efficient, and ultimately, a more sustainable transport solution.

1.2 The Turbocharger

The turbocharger remains one of the best means available to the engine manufacturer to achieve significant power gains without a large financial investment. Unlike a supercharger which consumes usable power from the ICE via the crankshaft, a turbocharger extracts energy from the hot, high pressure exhaust gases that are normally simply expelled from the tailpipe. The exhaust valves at the top of the cylinders open and close once every two revolutions of the crankshaft,

releasing a highly pulsating flow into a turbine that drives the rotor blades before expanding into the exhaust outlet. Although some larger turbochargers are designed with an axial turbine stage, it is more common that the exhaust flow is introduced into the wheel in a predominantly radial direction as shown in Figure 1. Thus, both the change in radius and the expansion of the gas through the turbine rotor result in the shaft torque needed to drive the compressor. The rotating compressor wheel then draws in ambient air through the inlet and pressurizes the air charge entering the internal combustion engine. The increase in air density that results thereby increases combustion intensity causing greater power per stroke. The exhaust manifold back pressure introduced by the presence of a turbine will not have an adverse effect on the engine operation as long as the intake boost pressure is maintained above the exhaust pressure. Hence, this simple device uses the otherwise waste exhaust gas energy to provide significant improvements in the volumetric efficiency of the internal combustion engine.

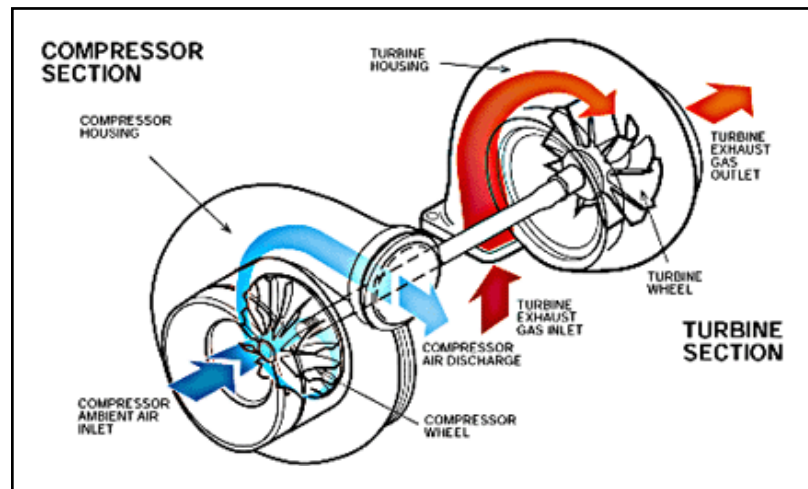


Figure 1.1: Radial Turbocharger Schematic

Although the principle of operation is really quite simple, there are a number of challenges in turbocharger design that are unique in turbomachinery. Since the operating range of an internal combustion engine can be variable, the turbocharger must be able to operate efficiently over the broad range of flow conditions that it encounters. Unfortunately, the efficiency of the turbocharger is highly dependent on the angle with which the gas enters the compressor impeller and turbine rotor. The blades are designed to be most efficient at a specific gas inlet angle, which in turn, depends on the speed of rotation and the design of the collector (volute). Therefore, the choice of turbocharger characteristics is driven by the operating characteristics of a particular application. If, for example, a large marine engine generally operates at a constant speed and load, the turbocharger will be selected such that its optimum efficiency corresponds to the primary operating

point. In most cases, however, the design point of the turbocharger will be a compromise to obtain the best all-around performance over the entire engine operating range.

The transient response of the system is also important since for many applications, a sudden power increase is often needed. In this situation, the lack of mechanical connection with the ICE is a disadvantage since the turbine rotor will require time to overcome inertia and spin up to speed. While this '*turbo-lag*' can be improved by simply installing a smaller turbine, the engineer must then contend with risks of rotor over-speed or intake manifold over-boosting at highest engine load. As the turbine area is reduced, specific available energy increases causing an increase in the boost pressure that could be damaging to engine parts and efficiency [3]. The two primary methods of controlling turbine speed and boost pressure is via an exhaust gas by-pass valve (waste gate) or by adjusting the geometry of the turbine inlet area. The waste gate is the simpler of the two options, but is also less efficient since the level of boost is simply controlled by by-passing the turbocharger completely. Variable geometry turbines are generally considered the most effective approach to engine matching since the inlet geometry of the turbine is adjusted in-situ to achieve the best performance over the range of operating conditions from a typical passenger vehicle. This thesis, however, will focus on a turbocharger with a fixed geometry made for much larger diesel engines.

Another challenge to the turbocharger engineer is how to control the temperature rise associated with the compression of the intake air. It is clearly advantageous to provide the engine with low temperature air since this will result in a denser air charge in the cylinder. Therefore, it is common to first pass the flow through an intercooler which uses water or air flow to cool the compressed air before delivery into the intake manifold. Although this additional item in the air intake system will cause a small pressure drop, the increase in air density due to air cooling should more than compensate.

1.3 The Turbocharger Turbine

1.31 Components

Figure 2 shows an exploded view of the main components that make up a typical turbocharger turbine. The turbine volute or scroll accepts the exhaust gases from the exhaust manifold in the tangential direction and turns the fluid to give it a radial velocity component using the characteristic spiral shape. The decreasing area of the scroll is necessary to distribute the flow equally around the circumference of the rotor inlet. Nozzle guide vanes are often added to further

accelerate and direct the flow into the turbine rotor. They also allow a convenient method for the manufacturer to tailor the turbine swallowing characteristics to the particular application without changing the entire volute. As mentioned, these nozzles can be either fixed or variable depending on the application, but the discussion in this thesis will be limited to a fixed nozzle design only. Finally, the work transfer from the fluid flowing through the turbomachine occurs in the rotor blade passages as it expands from inlet to exit.

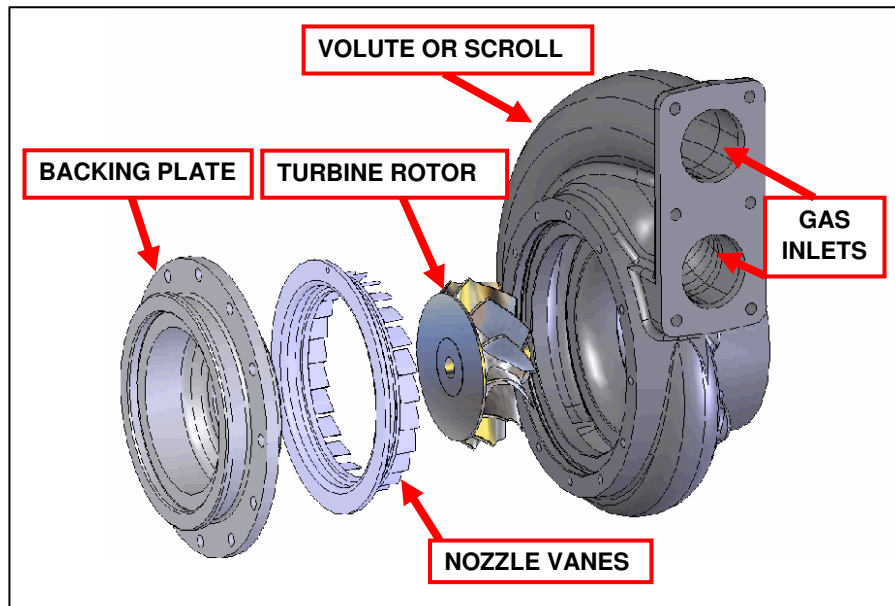


Figure 1.2: ABB radially divided volute

1.32 Turbine Volute

As indicated in section 1.2, the air enters the turbine in a tangential direction and is turned by the volute as a free vortex centred about the axis of rotation. Under a free vortex, the term rC_θ in Figure 1.3 will be constant at any azimuth angle θ and the mass flow will be: $\dot{m}_\theta = \rho_\theta A_\theta C_\theta$, where A_θ is the cross-sectional area. Thus, with a uniform distribution of mass flow around the circumference desirable, it follows that:

$$\frac{A_\theta}{r_\theta} = \frac{\dot{m}}{\rho_\theta r C_\theta} \left(1 - \frac{\theta}{2\pi}\right) \quad (1.1)$$

Thus, with the assumption of uniform density, the area-to-radius ratio A_θ/r_θ will be a linear function of the azimuth angle θ and the resulting area reduction produces the characteristic scroll shape. This simple analysis forms the basis of volute design and can be easily applied to each half of a double-

entry volute as well. A simple relationship can also be derived between the area-to-radius ratio at the volute inlet and the flow angle that results at the exit of the volute [4]. From simple conservation of mass in and out of the volute: $\rho_1 A_1 C_{m1} = \rho_0 A_0 C_0$ and the free vortex relation: $r_1 C_{\theta 1} = r_0 C_0$, the angle of incidence is:

$$\alpha_1 = \tan^{-1} \left(\frac{C_{\theta 2}}{C_{m2}} \right) = \tan^{-1} \left(\frac{\rho_2 A_2 / r_2}{\rho_0 A_0 / r_0} \right) \quad (1.2)$$

Equation 1.2 is interesting because it suggests that the incidence angle is only a function of the area-to-radius ratio at the inlet and outlet of the volute and any change in density through the volute. Thus, if we assume that the density ratio does not change appreciably, the absolute angle of incidence into the nozzle ring will remain constant irrespective of the mass flow. Thus if the volute and nozzle are matched, any small difference in flow angle should be corrected in the nozzles without significant loss. Therefore, the most important function of the volute in a nozzled turbine is to deliver a uniform mass flow to the entire circumference of the nozzle ring inlet.

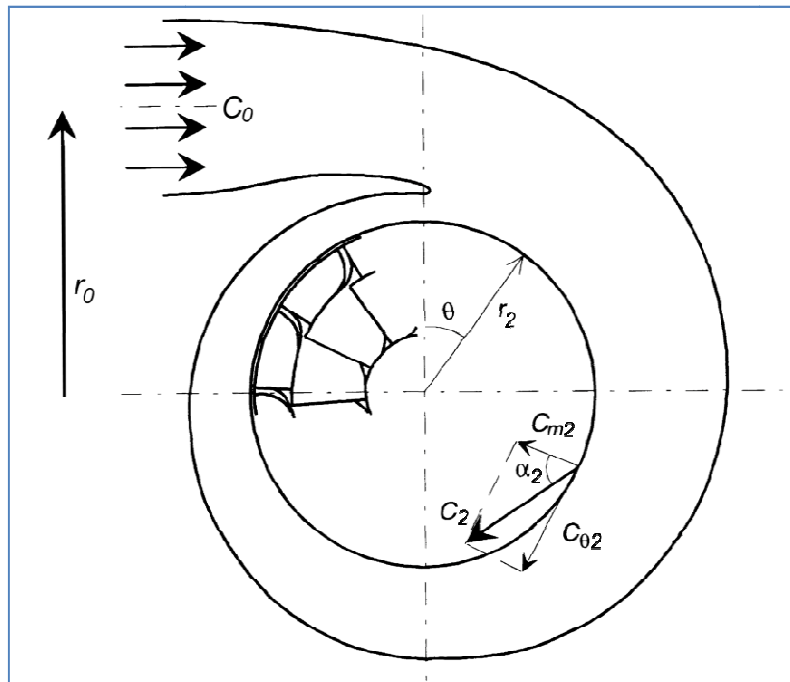


Figure 1.3: Basic volute design [4]

Most applications of radial turbocharging today utilize *pulse turbocharging* to maximize the energy extraction by the turbine. This means that the pulses produced from the opening and closing of the exhaust valve are directly fed into the turbocharger to impart the full energy in the exhaust

gases to the rotor blades. However, as in apparent in Figure 1.4, an internal combustion engine with more than four cylinders has a significant overlap between the exhaust pulses. Unless isolated, the interference between pressure waves can decrease the overall pulse energy and increase the engine pumping loss as two cylinders try to expel exhaust gas into the same pipe. Thus, it is advantageous to isolate a set of cylinders and introduce the exhaust gases into two or more entries separately. The design of the volute therefore plays a second important role, that is, to isolate two or more exhaust streams up to the turbine wheel.

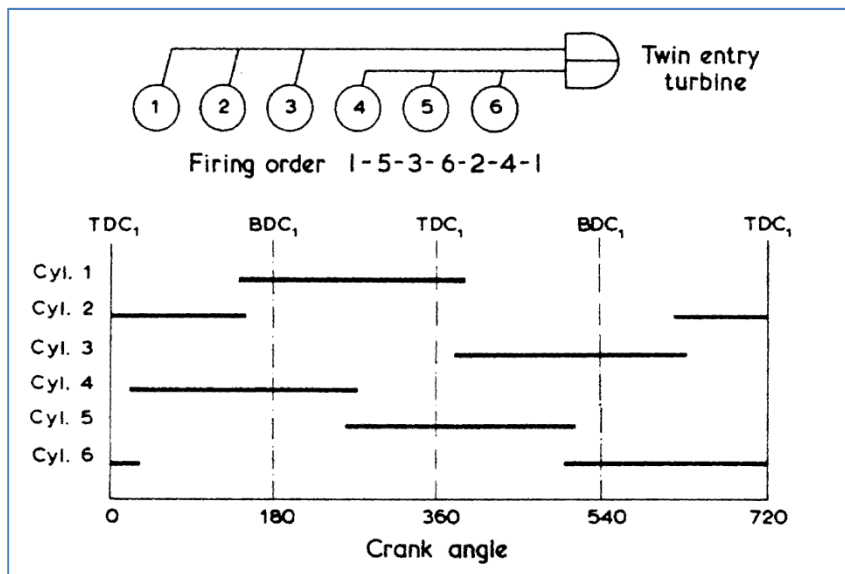


Figure 1.4: Exhaust Pulse Timing [4]

There are two main types of dual entry turbine volutes depending on the method of flow division. Meridionally divided or *twin-entry* has a single divider around the entire perimeter of the housing as shown on the left hand side of Figure 1.5. In this design, each entry feeds the entire circumference of the rotor or stator inlet. However, the space between the end of the divider and rotor inlet can permit a significant amount of flow interaction prior to rotor entry which can cause mixing loss or flow angle disturbances. Despite this, the twin-flow nozzleless turbine is the most common multi-entry geometry due to its simple, inexpensive design.

The circumferentially divided or double-flow scroll has a divider which separates the flow such that each entry feeds a separate section of the rotor as shown on the right hand side of Figure 1.5. Unlike the twin-entry turbine, this design permits more than two entries and thus is typical for larger multi-cylinder applications. When equipped with nozzle vanes, the volute tongue created at the end of each dividing wall must be designed to match one of the nozzle vanes to prevent loss in

these regions as shown in Figure 1.6. The advantage to this design is that, with the exception of areas around the tongue, there should be little interaction between the two flows – especially when the space between the exit of the nozzle vanes and the inlet to the turbine rotor is small.

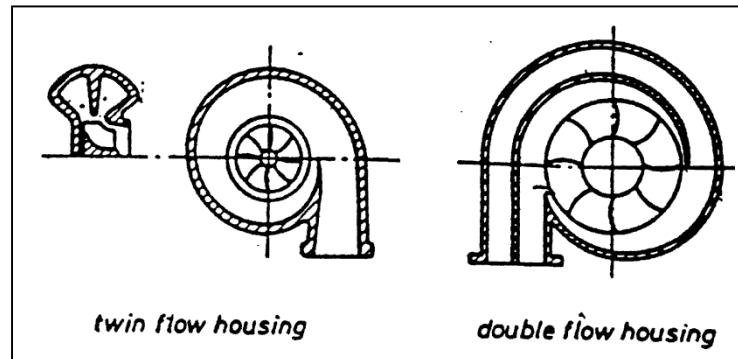


Figure 1.5: Dual entry volute designs [5]

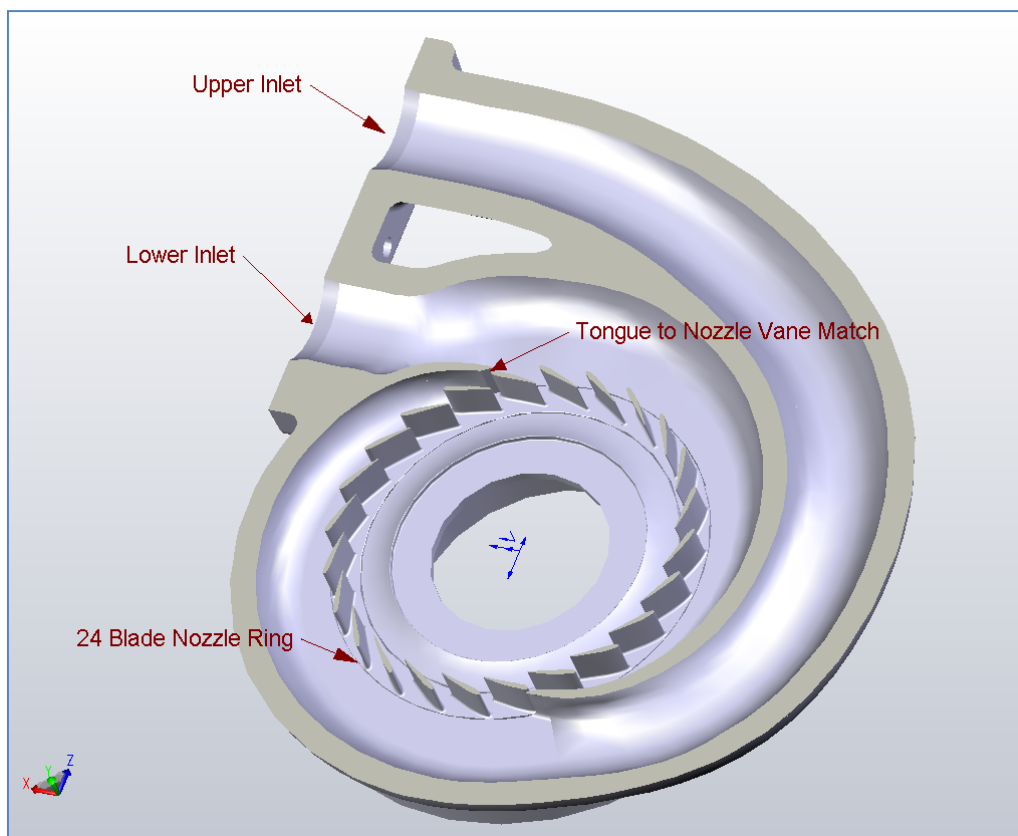


Figure 1.6: Double-entry Volute and Nozzle Ring

Although both of these multiple-entry designs serve a similar purpose, the choice of volute geometry can significantly influence the behaviour of the turbine. The differences between the two

types are most obvious when the flows in each inlet are not equal. Figure 1.7 shows a steady flow performance comparison between the two designs published by Pischinger and Wunsche [5] and discussed in more depth in the Chapter 2. Although the two types of volutes have the same equal admission effective area A_s (demonstrated by the common intersection of the curves in the left hand figure), this plot highlights the differences in how the gases in each volute passage interact under unequal admission. The right hand plot in Figure 1.7 also suggests that the efficiency loss under unequal admission is dependent on the volute geometry. The research presented in this thesis will entirely focus on the double-entry turbine and the resulting performance implications of this design.

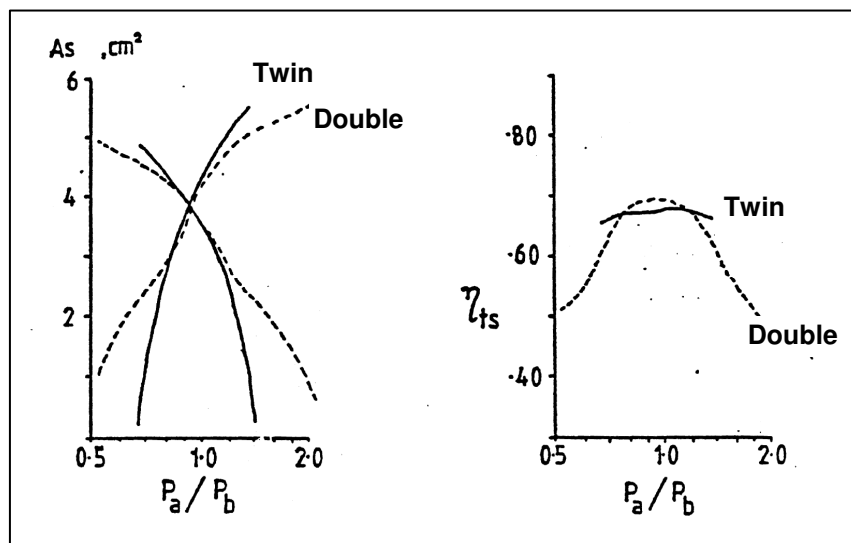


Figure 1.7: Twin entry versus double volute performance [5]

1.33 Nozzle Vanes

In a turbocharger turbine equipped with a volute, the volute provides most of the swirl needed for correct incidence into the rotor. Therefore, the nozzles shown in Figure 1.6 are not designed with any camber, but are straight aerofoils that distribute and accelerate the flow equally around the turbine wheel inlet. As mentioned, the nozzle ring also plays an important role to the manufacturer in providing an effective method to trim the turbine inlet area to match the flow rate requirements of the engine. The nozzle opening will be sized to pass the range of engine mass flows without choking occurring in the throat of the passage (near the trailing edge for a typical convergent nozzle) at high pressure ratios. The nozzle is not the only limiting area in the turbine, since the rotor exducer may also choke in some nozzle-rotor arrangements. Wherever it occurs, however, choking in the turbine will limit the flow capacity and may produce an efficiency loss due to the induced shock waves in the passage and should be avoided.

If one now considers the off-design conditions, it may be tempting to think that a different mass flowing through the volute will cause an incorrect absolute incidence angle at nozzle inlet thereby leading to a mismatch and possible flow separation as the vanes attempt to turn the flow. However, it is important to note that equation 1.2 indicates that apart from the ratio of densities ρ_1/ρ_0 , the absolute incidence angle is purely a function of the volute geometry. This is an especially important point to consider under unsteady admission where there are large fluctuations in flow conditions over a pulse cycle. Experimental work by Karamanis [6] showed that the absolute flow angle exiting a nozzleless turbine volute will, in fact, remain relatively constant over a flow pulse despite the large change in inlet mass flow and pressure. The small change in absolute flow angle shown in Figure 1.8 could be attributed to the change in the density ratio ρ_1/ρ_0 that inevitably results from a varying Mach number. This then leads to the conclusion that the loss associated with off-design conditions should be largely due to a non-ideal *relative* flow angle introduced to the turbine wheel. This is not to say the nozzle has little effect during pulsating flow. Since the size of the nozzle throat plays a very important role in controlling the turbine mass flow characteristics, it could have a significant influence on how the volute fills and empties itself. This influence will be discussed in more depth in the ensuing chapters.

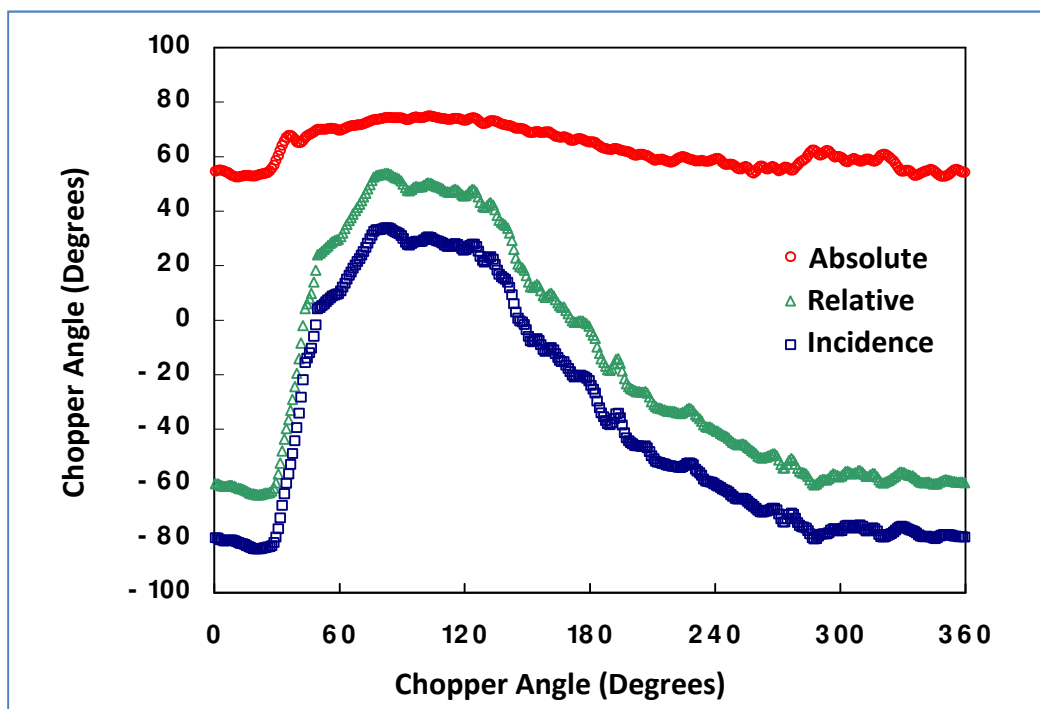


Figure 1.8: Flow angles measured at rotor inlet [6]

Between the exit to the nozzle and the inlet to the rotor there is a gap termed here the *nozzle-rotor interspace* whose size and shape will be determined by the design of the turbine blades and the difference in radii between the nozzle and rotor. A cross section of the double-entry turbine

presented in this thesis is shown in Figure 1.9. Since the rotor is mixed-flow in design (between axial and radial), there will be quite a sizable vaneless interspace which spans the circumference of the rotor inlet area. This space gains some importance in the double-entry design since this it joins the two entries prior to rotor inlet. Thus, in the case where the two inlets are fed with two different pressures and mass flows, it seems reasonable to expect that there will be some interaction between the flows via this interspace.

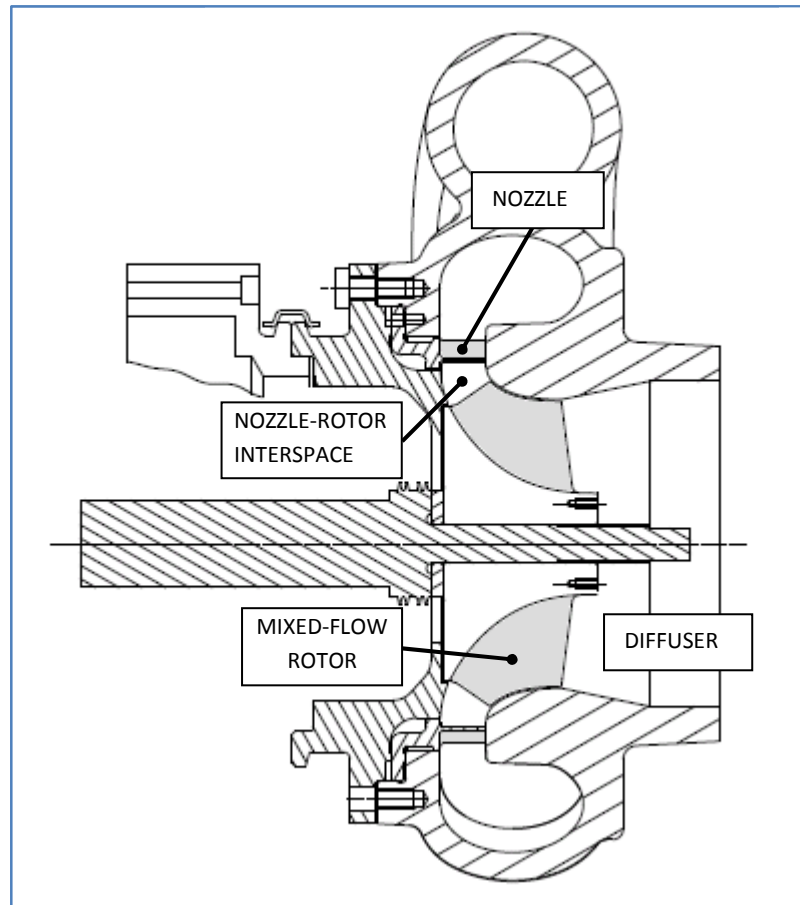


Figure 1.9: Cross section of a double-entry turbine with a mixed-flow rotor

1.34 Turbine Wheel

It is important to first outline the terminology that will be used in this thesis. The rotor is a complicated, three dimensional shape and therefore difficult to illustrate in a single two-dimensional figure. Therefore, the description and the terminology are often important in conveying the point that is being made. Figure 1.10 below illustrates both a meridional view of the rotor in the casing as well as an oblique 3-D view of the turbine wheel. The rotor consists of a solid *hub* with blades which expand radially outwards normal to the axis of rotation. Thus, despite the complex shape of the

blade profile, a radial fibre is largely maintained to alleviate bending stresses that will result from high-speed rotation. Moving along the radial fibre, the blade terminates just prior to the stationary *shroud*, thus forming a small *tip clearance* whose width can have a significant influence on turbine efficiency. So a single blade passage is bound radially by the *hub* and the *shroud*, and moving in the direction of rotation, by the suction and pressure surface of the two blades forming the passage.

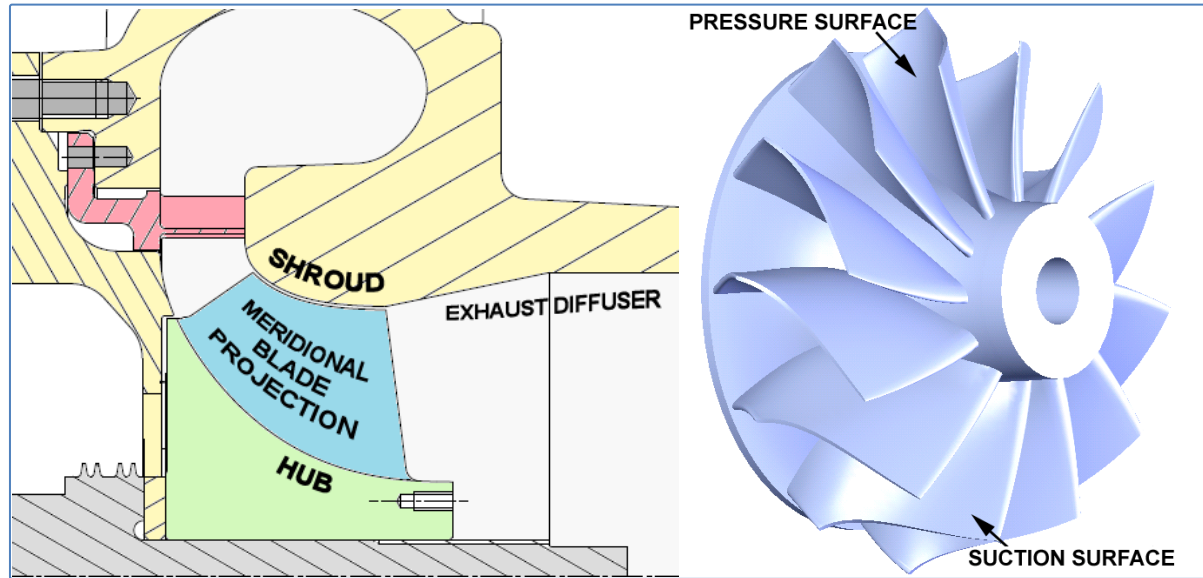


Figure 1.10: Mixed flow turbine rotor from a meridional and 3-D viewpoint

The basic purpose of the turbine wheel is to extract energy from the exhaust gases and thus perform useful work in turning the compressor wheel. The Euler turbomachinery equation is the simplest way to understand the source of the work transfer which occurs in the rotor. The rate of change of angular momentum of a steady flow moving from rotor inlet (station 3) to outlet (station 4) will result in a torque:

$$\tau = \dot{m}(r_3 C_{\theta 3} - r_4 C_{\theta 4}) \quad (1.3)$$

where the tangential velocities at inlet $C_{\theta 3}$ and outlet $C_{\theta 4}$ are defined in Figure 1.11. The Euler turbomachinery equation then simply defines the specific work transfer as:

$$\dot{W}/\dot{m} = (U_3 C_{\theta 3} - U_4 C_{\theta 4}) \quad (1.4)$$

The velocity triangles relations in Figure 1.11 can then be substituted into equation 1.4 to give:

$$\frac{\dot{W}}{\dot{m}} = \frac{1}{2} [(U_3^2 - U_4^2) - (W_3^2 - W_4^2) + (C_3^2 - C_4^2)] \quad (1.5)$$

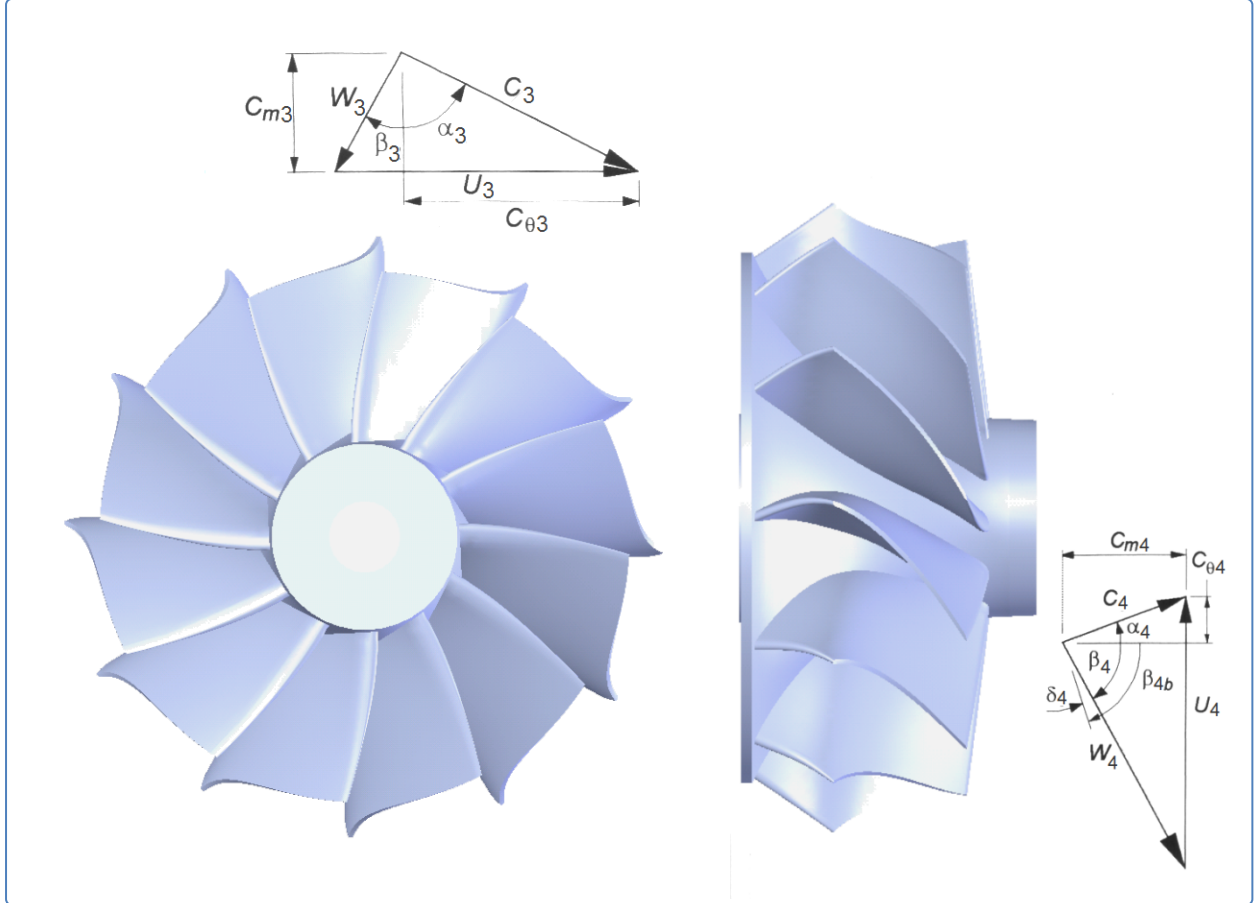


Figure 1.11: Inlet and exit rotor flow angles

Equation 1.5 thus gives an indication of how the velocities in and out of the rotor must be optimized to maximize the energy transfer. The first term $(U_3^2 - U_4^2)$ illustrates the advantage of a radial machine since the change in radius from inlet to outlet results in a contribution to the shaft power. The relative velocity term $(W_3^2 - W_4^2)$ must be minimized, meaning that the flow must accelerate through the passage in the relative frame of reference. Lastly, the difference between the inlet and exit absolute velocities $(C_3^2 - C_4^2)$ can be maximized by reducing exit swirl $C_{\theta 4}$ and adding an exhaust diffuser to recover some of the exit kinetic energy.

It is desirable for a purely radial turbine to be constrained to a zero blade inlet angle, and in this way maintain the radial fibre and avoid the bending stresses. This requirement also limits the

point of maximum efficiency to a certain area in the turbines operating range since the optimum flow inlet angle will also be constrained. If, for example, the relative flow inlet angle is zero ($C_{\theta 3} = U_3$) and the exit swirl is also zero ($C_{\theta 4} = 0$), equation 1.5 will be simplified to:

$$\frac{\dot{W}}{\dot{m}} = U_3^2 \quad (1.6)$$

The velocity ratio is a non-dimensional number that is frequently used by the turbocharger designer to match the compressor wheel to the turbine wheel. It also is very useful to define the operating point of the turbine since the peak efficiency occurs at approximately the same velocity ratio regardless of speed. It is defined as the rotor tip velocity U_3 divided by the velocity of the gas that would result from an isentropic expansion between the inlet and outlet:

$$\frac{U_3}{C_{is}} = \frac{U_3}{\sqrt{2[(h_{01} - h_{4,is})]}} = \frac{U_3}{\sqrt{\{2c_p T_{01}[1 - (P_4/P_{01})^{(\gamma-1)/\gamma}]\}}} \quad (1.7)$$

Assuming complete exhaust recovery, equation 1.6 can be substituted into equation 1.7 to give:

$$\frac{U_3}{C_{is}} = \frac{U_3}{\sqrt{(2U_3^2)}} = 0.707 \quad (1.8)$$

This illustrates that the purely radial inflow turbine, with all the assumptions made above, has an optimum efficiency at a velocity ratio around 0.7. However, it would be advantageous to design a turbine that is able to extract the power most effectively at the high pressure ratios (lower velocity ratios) where the exhaust energy is greatest. It is therefore sometimes advantageous to have the design freedom to lower the velocity ratio value where peak performance occurs. To do this without sacrificing the radial fibre requirement, the turbine blade could be of *mixed-flow* geometry as shown in Figure 1.12. This allows a non-zero blade inlet angle thus permitting some adjustment of the optimum relative flow inlet angle β_2 . The mixed-flow turbocharger turbine that is used for the work in this thesis is shown in Figure 1.11.

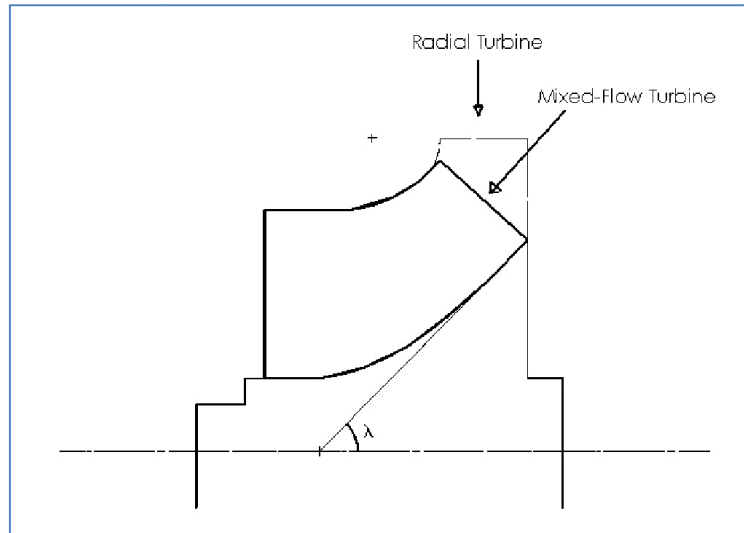


Figure 1.12: Radial versus mixed flow blade geometry

1.4 Pulse charging

Most applications of turbocharging today use a process of pulse charging whereby a compact exhaust manifold delivers a highly pulsating flow resulting from the opening and closing of the exhaust valve to the inlet of the turbocharger turbine. As alluded to earlier, the motivation of introducing this highly unsteady flow is to preserve as much of the pulse energy to the turbine blades as possible. Although it is possible to damp-out the pulses in a large volume exhaust manifold to produce a constant pressure system, this approach is less common both due to the space requirements and energy lost in this process. The caveat to a pulse charged turbocharger arrangement is that the flow driving the turbine is no longer steady in nature, but a very complex flow with large variations in pressure, mass flow and temperature. Recall that much of the discussion thus far has been in aiming at optimizing the various components of the turbine to produce the most efficient operation at a design point that best matches the engine operating range. However, the reality is that the conditions driving the turbine wheel will be non-ideal for much of the pulse cycle. This will ultimately result in an overall performance penalty. One of the largest contributors to off-design efficiency loss is incidence effects, since they determine much of the flow structures in the rotor passage. Laser measurements by Karamanis [6] at the turbine inlet shown in Figure 1.8 clearly demonstrate the impact of a pulsating flow on the relative incidence angle of the flow entering the turbine. With such a wide variation of inlet conditions, it is not surprising that the pulsating efficiency of the turbine will not match that predicted by the steady state performance.

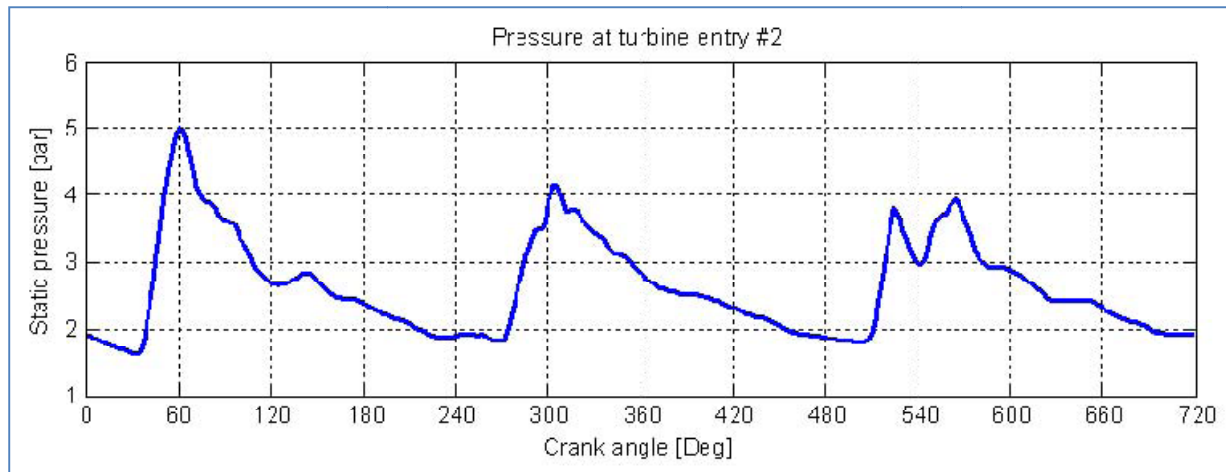


Figure 1.13: Pressure pulse shape at turbine inlet [7]

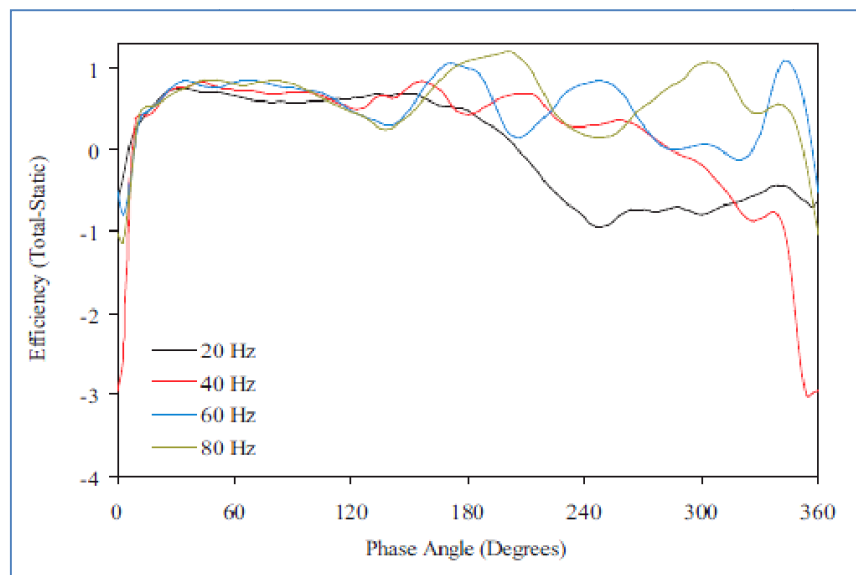


Figure 1.14: Instantaneous turbine efficiency [8]

As the exhaust valve just starts to open, the pressure in the exhaust manifold increases dramatically as the gases expand into a passage feeding the turbine. The pressure in the manifold will quickly peak as shown in Figure 1.13 followed by a slower decrease resulting from the piston moving up the cylinder expelling the exhaust gases into the passage. From the time the piston has reached top-dead-centre (TDC) and the exhaust valve has closed, the mass that has accumulated in the manifold and volute will be consumed by the turbine until the point where there is insufficient energy in the flow to drive the turbine wheel. Under these conditions, the inertia of the wheel means it will start to ‘windmill’ and that in the extreme it will be acting as a compressor. Evidence of this is apparent when attempting to measure the instantaneous efficiencies in the lab and negative values result (Figure 1.14) [8]. To reduce this effect, 3 or more cylinders are often grouped into a

single manifold as shown in Figure 1.4 to superimpose the opening of the exhaust valves so that the time lag between one pulse 'event' and the next is small thereby resulting in a more steady flow without pulse interference. As already mentioned, care must be taken in this process to avoid significant pulse overlap by grouping appropriate cylinders and selecting appropriate pipe lengths to mitigate any engine pumping loss.

Thus, the unsteady, highly pulsating nature of the exhaust gases is clearly an important feature that differentiates the turbocharger from other forms of turbomachinery. Yet, although researchers are clearly aware of the highly unsteady nature of turbocharger operation, a large proportion of the experimental knowledge accrued over the past 40-50 years has been based on treating the turbine as a steady-flow, aerodynamic device. This is largely due to the inherent difficulty in making accurate time resolved unsteady flow measurements in such a challenging, high temperature environment. In addition, there appears to be a general feeling in some of the early research that the turbine does not significantly depart from quasi-steady behaviour. This presumes that over a pulse cycle, the unsteady performance of the turbine at every instance in time can be described without considering the previous history of the flow. Were this assumption valid, the turbocharger's unsteady performance could be predicted as an integration of steady behaviour on an instant-by-instant basis. Thus most of the software used today to predict the on-engine turbocharger behaviour is based on steady-state performance maps. One of the primary aims of this thesis is to provide a fuller understanding of the implications of unsteady flow, the flow physics that may be driving the observed performance, and to understand where the quasi-steady assumption breaks down.

An additional level of complexity is introduced with a multiple entry turbocharger since the pulses that enter each entry are typically out-of-phase. This is clearly shown by measurements published by Dale and Watson [9] (Figure 1.15) that illustrate a 180° phase difference in the pulse shapes resulting from two groupings of 3 cylinders in a six- cylinder twin-turbocharged application. The turbine rotor that is fed with a dual-entry volute will therefore not only experience a large variation of inlet conditions resulting from a pulse cycle, but also instantaneous conditions of unequal admission. If the turbine does indeed operate in a quasi-steady manner, the out-of-phase nature of the flow should translate into an additional loss in turbine efficiency. Thus one of the aims of the work presented in this thesis is to study the effect that phase of the pulse has on double-entry turbocharger performance.

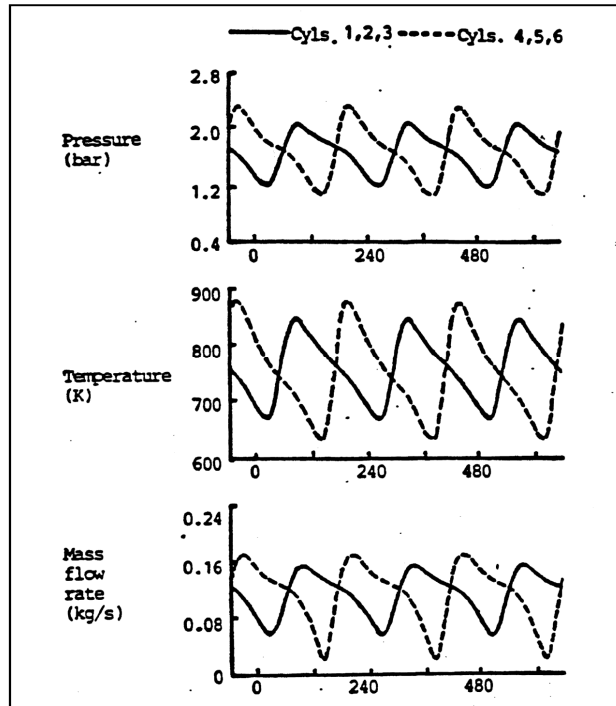


Figure 1.15: Flow pulses into a twin-entry turbine from a six cylinder engine [9]

Thesis Objectives

The main objectives of this thesis are as follows:

1. To quantify the steady-state performance of a double-entry turbocharger turbine using a wide selection of performance tests on the turbocharger test stand. The second gas inlet adds another degree of freedom since the tests must fully cover all the equal and unequal states that the turbine might experience. This step was necessary to establish base-line turbine performance and lead to further understanding of the impact of unequal flows in the absence of a pulsating flow field.
2. To analyse the flow features that lead to the observed steady-state performance with the help of a fully transient 3-D CFD code. This is needed to understand the reasons behind the significant loss in efficiency always associated with unequal admission.
3. To investigate the influence of unsteady flow in a double-entry turbocharger turbine by introducing a variety of pulsating flow conditions. Both the performance implication of pulse frequency and the phase difference between the pulse cycles at each inlet is investigated.

4. To assess the validity of the quasi-steady assumption. By matching the instantaneous unsteady operating condition from objective 2 to the corresponding steady-state performance from objective 1, a quasi-steady 'equivalent' performance is calculated. This is then compared to the measured unsteady performance to assess the discrepancy in the quasi-steady assumption for different pulse frequencies.

Thesis Outline

Chapter 1: Introduction

The first chapter introduces the topic of turbocharging and lays the groundwork for the research. The components of the turbocharger turbine are introduced and discussed, as well as a brief overview of the important topics of this thesis. A summary of the objectives of the thesis is also provided, as well as a brief outline.

Chapter 2: Literature Survey

The purpose of this chapter is to outline the published literature that is most pertinent to the discussion in the subsequent chapters. The topics include: double-entry and twin-entry performance, pulsating flow performance, and quasi-steady analysis. Each work is discussed as it relates to the research that is presented in this thesis.

Chapter 3: Experimental Procedure

This chapter contains an outline of the facility and testing methods that are used to produce the experimental data that is presented in Chapters 4 and 5. First, an outline of the dimensionless and pseudo-dimensionless parameters that will be used to represent the turbocharger performance is presented. This is followed by a comprehensive summary of the facility and the turbocharger turbine that is tested exclusively in this thesis. Lastly, an outline of the experimental and data processing procedure is also provided.

Chapter 4: Steady Flow Experimental Results

In this chapter, the steady-state experimental data will be presented and discussed. First, the equal admission performance maps will be presented and considered. The most in depth discussion will result from the unequal admission performance. Here, the loss of turbine performance due to the unequal flows, and the interaction between the two limbs that is suggested by the data will be considered. The chapter also discusses the impact that unequal admission flows might have on the flow in the turbine wheel as a result of the unique way flow is introduced by the double-entry design.

Chapter 5: Computational Fluid Dynamics Model

To compliment the experimental analysis in the previous chapter, Chapter 5 presents results from a 3-D CFD model of the double-entry turbine. The computational domain (mesh) is reviewed along with a discussion of the appropriate boundary conditions for modelling the steady-state behaviour in Chapter 4. Since the influence of unequal admission is of particular interest, a number of different cases are analysed to attempt to understand some of the flow physics driving the observed performance.

Chapter 6: Unsteady Flow Experimental Results

This chapter contains all the unsteady experimental test data. The performance of the turbocharger under both in-phase and out-of-phase pulsations is presented and the influence of turbine speed and pulse frequency is analysed in some depth. The steady and unsteady data is then combined to test whether the turbine behaves in a quasi-steady manner under pulsating flow conditions. To interpret the results of the quasi-steady analysis, a comparison of the different time scales involved in unsteady flow by means of a Strouhal number is necessary.

Chapter 7

The final chapter simply discusses some of the more unique findings of this thesis, and makes recommendations for future research.

Chapter 2

Literature Survey

2.1 Introduction

As outlined in the introductory section, the aim of the research work presented in this thesis is to evaluate the unsteady performance of a double-entry ABB turbine and provide a physical insight for future design options. Although a number of researchers have sought to study the effect of unsteady flow in a turbocharger turbine, there are a very limited number of experimental papers that deal with unsteadiness in a double-entry design where the consequence of unequal admission is considered [10,13,16]. In addition, the research on turbochargers with a circumferentially divided turbine was primarily done before 1980 and thus many of the experimental techniques to measure performance and unsteady behaviour fall short of what is capable with current instrumentation. Our understanding of unsteady behaviour and the ability to measure it accurately has advanced significantly in recent years. With this in mind, it seems clear that there is an opportunity to contribute to the knowledge in this area.

As was discussed in the introduction, the design of the volute, specifically how the two entries are divided, has a significant impact on the flow dynamics and ultimately on the performance of the turbocharger. Thus, the few papers that are the most applicable to the current research are those which feature a similar design to the circumferentially divided turbine. These papers will therefore be considered in more depth in the first section of this review. Following this, the second section will contain a short review of research based on the twin-entry, meridionally divided turbine. Much of the work in this area was done at Imperial College in the late 80's and early 90's, primarily by A. Dale under Neil Watson and J. Yeo under Nick Baines [9,17-19,24]. These publications are useful insofar as issues of unequal and unsteady admission are examined for their effect on turbine performance. However, many of the conclusions are limited to the twin-entry case and therefore cannot be directly applied to the current turbine configuration. The third section will consider the range of published literature that investigates the effect of an unsteady flow in a turbocharger turbine. Particularly relevant to the present research, are those papers which compare the steady and

unsteady relationship through the quasi-steady assumption. Earlier authors were limited by lack of time-resolved quantities. Starting with Dale and Watson [9], however, the later authors were able to measure all the instantaneous quantities and therefore contributed a great deal to the physical understanding of the unsteady flow and its effects.

2.1 Double-entry Performance

Mizumachi, Yoshiki and Endoh [10] investigated the flow in a double-entry turbine using both numerical and experimental means. The authors' report is split up into two main parts: steady flow and unsteady flow. To begin, steady, partial admission performance in a radial turbine was measured and compared to full admission performance. The authors then sought to model this flow computationally to gain some insight into the influence of partial admission. In the second section, unsteady in-phase and out-of-phase turbine performance was tested experimentally with a different turbine and experimental apparatus. A 1-D model was developed that was then compared to the experimental data.

For steady partial admission flow tests, a *single* entry turbine with nozzle vanes was used as shown in Figure 2.1. To achieve the partial admission condition, a 180° section of the nozzle entry was blocked as shown. The turbine was coupled to an electric dynamometer through a set of reduction gears to allow testing beyond the range of the dynamometer limitations. In addition to the standard instruments for measuring mass flow, pressure and temperature, a set of 38 pressure taps was installed around the periphery of the nozzle exit. This data permitted an interesting insight into partial admission behaviour and a validation for the computational analysis. A plot of the pressure distribution at nozzle exit under partial admission is shown in Figure 2.2. It is interesting to note the rise in pressure that occurs in the direction of rotor rotation in the non-admission region. This can be attributed to the centrifugal head that is created in the non-flowing section by a rotating rotor. For comparison, the theoretical centrifugal pressure rise at rotor inlet (p_c) was plotted for each velocity ratio. This plot may, however, be influenced by the decision to simulate partial admission in a double-entry design by simply blocking off a section of a single entry turbine. Despite this puzzling choice, the authors do make some interesting observations concerning the impact of unequal admission in this design. They point out the possibility that a partial admission turbine may create a pulse in the rotor as it rotates from the flowing to the non-flowing section.

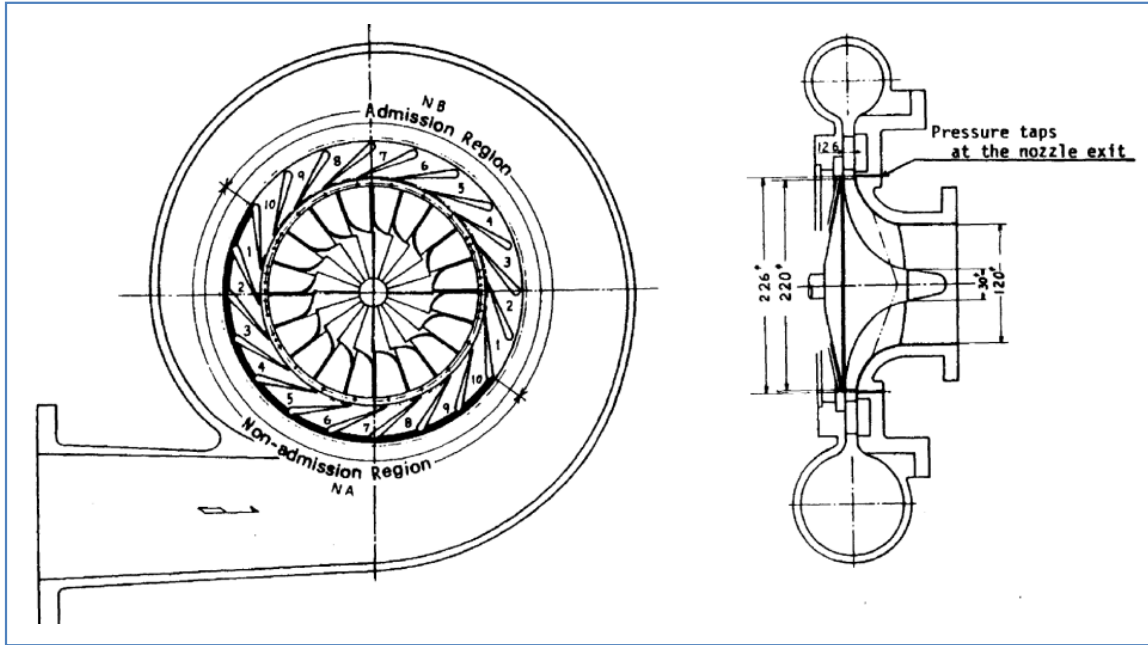


Figure 2.1: Single entry turbine set up for partial admission [10]

In terms of performance, the authors recorded a significant drop in efficiency between full and partial admission conditions as shown in Figure 2.3. This dramatic drop in efficiency is hardly surprising considering that a 180° sector of the rotor inlet was blocked. The mass flow rate comparison is slightly more unexpected. As shown in Figure 2.3 for a constant pressure ratio of 2.0, the mass flow characteristics of the partial admission turbine was approximately equal to half of the full admission mass parameter. This may be related to the decision to simulate a partially admitted double-entry volute by blocking a single-entry turbine. Instead of having some interaction between the two entries as one might expect from a true double-entry design, the results simply show a turbine swallowing capacity that has been reduced by half since half of the inlet is blocked.

A computational compressible flow analysis was executed using the method of characteristics, using Riemann variables. Losses in the nozzle and rotor were not taken into account and thus the large drop in efficiency was not successfully predicted. The mass flow rate prediction for partial and full admission fared well but the partial admission pressure distribution shown in Figure 2.2 did not demonstrate a close agreement.

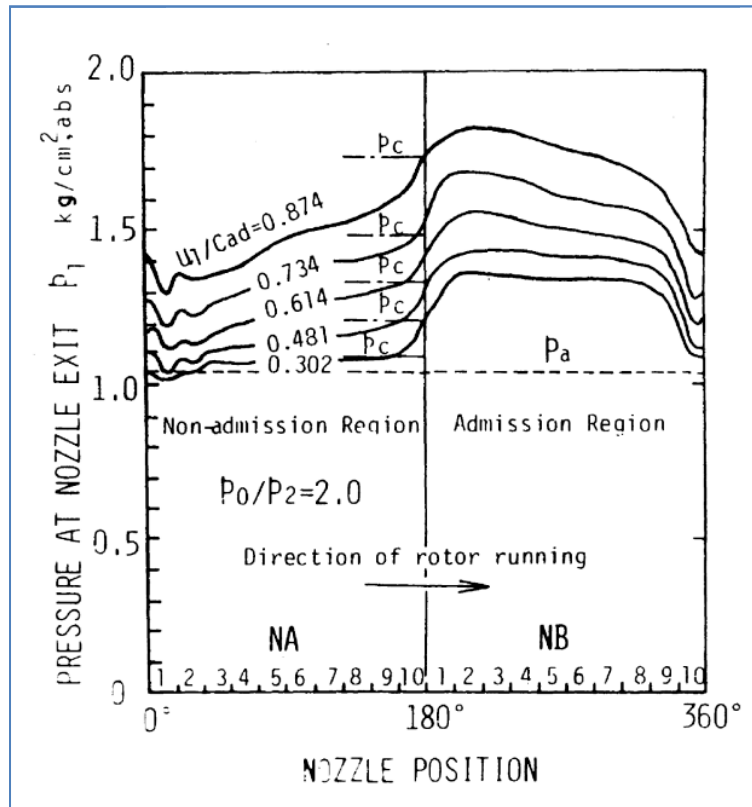


Figure 2.2: Pressure distribution between the nozzles and rotor [10]

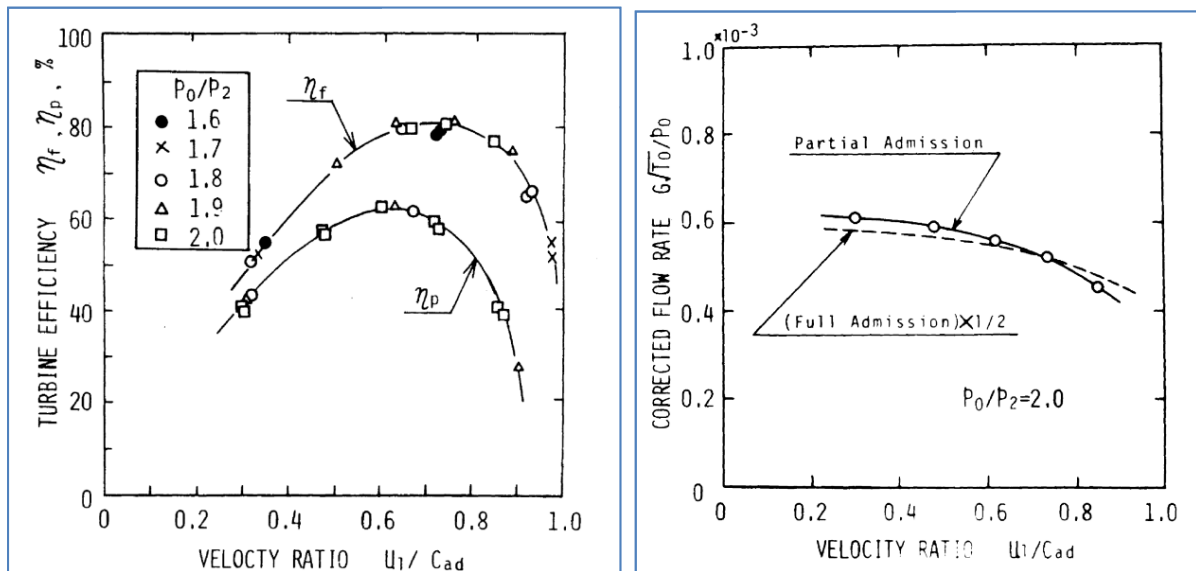


Figure 2.3: Full & Partial Admission Efficiency (LH plot) and Mass Flow (RH plot) [10]

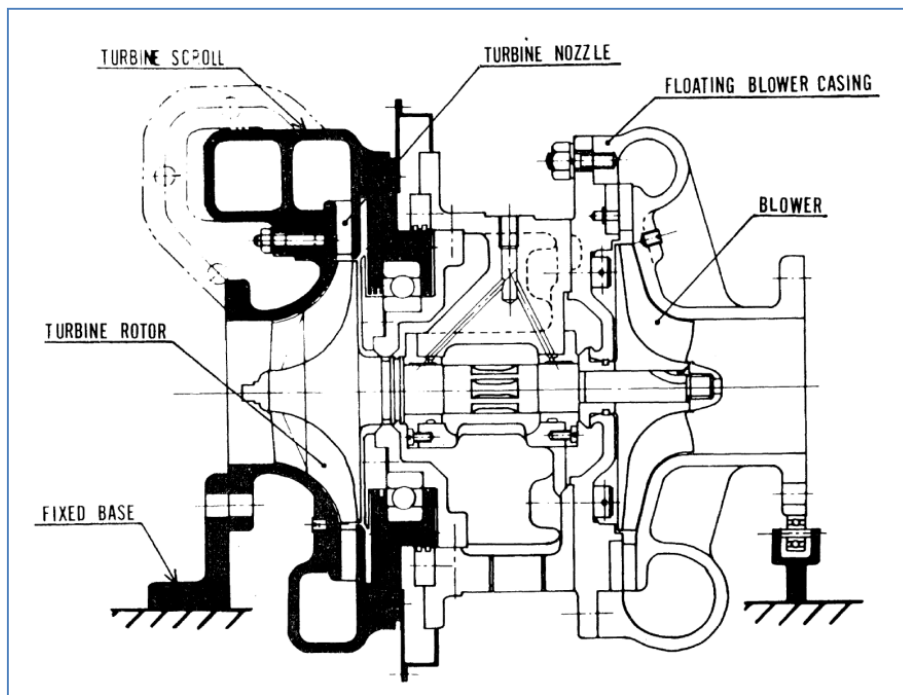


Figure 2.4: Double entry test arrangement [10]

The authors also sought to examine the influence of unsteadiness in a dual entry flow turbine. A different turbine and test facility was used for unsteady flow testing. The turbine was coupled to a compressor as shown in Figure 2.4 which was supported by a set of bearings to permit torque measurement directly from the floating casing. An air source fed a rotary valve used to produce pulsed flow which approximates engine exhaust flow. Two separate rotary valves were used to produce in-phase and out-of-phase pulsed flows into the turbine entries. Although the turbine shown is meridionally divided, each entry feeds a separate nozzle section similar to circumferentially divided volutes. Pressures throughout the rig are measured dynamically, but the speed, temperature, torque and mass flow are all limited to time-averaged values. The main thrust of the unsteady section of this publication seems to be to develop and validate a 1-D flow model that adequately predicts the unsteady flow performance of a double entry turbine. Thus, although some experimental results are reported, it is more with the view to validate the model than to understanding the flow physics. For in-phase and out-of-phase flow, the 1-D model does well to predict the instantaneous pressure fluctuation but over-predicts average torque and efficiency.

Overall, this work aligns with the research in this thesis; however, there are a number of areas that needed significant improvement. With regard to steady-state experimentation, the tests were limited to full and partial admission, with no tests performed in between to allow a true quasi-steady comparison. Also, the use of a single entry turbine with half of the nozzles blocked to

simulate partial admission is an unrealistic case for two reasons. First, the true double-entry turbine will behave quite differently due to areas of mixing between the entries. Secondly, the data from this single-entry turbine was then compared with the unsteady data, despite coming from a totally different turbine. Concerning the unsteady experiments, since they were performed using largely time-averaged measurements, very little real insight into unsteady effects can be gained. Lastly, it is also disappointing to note that the in-phase and out-of-phase unsteady tests were performed at totally different conditions, thus preventing any direct comparison.

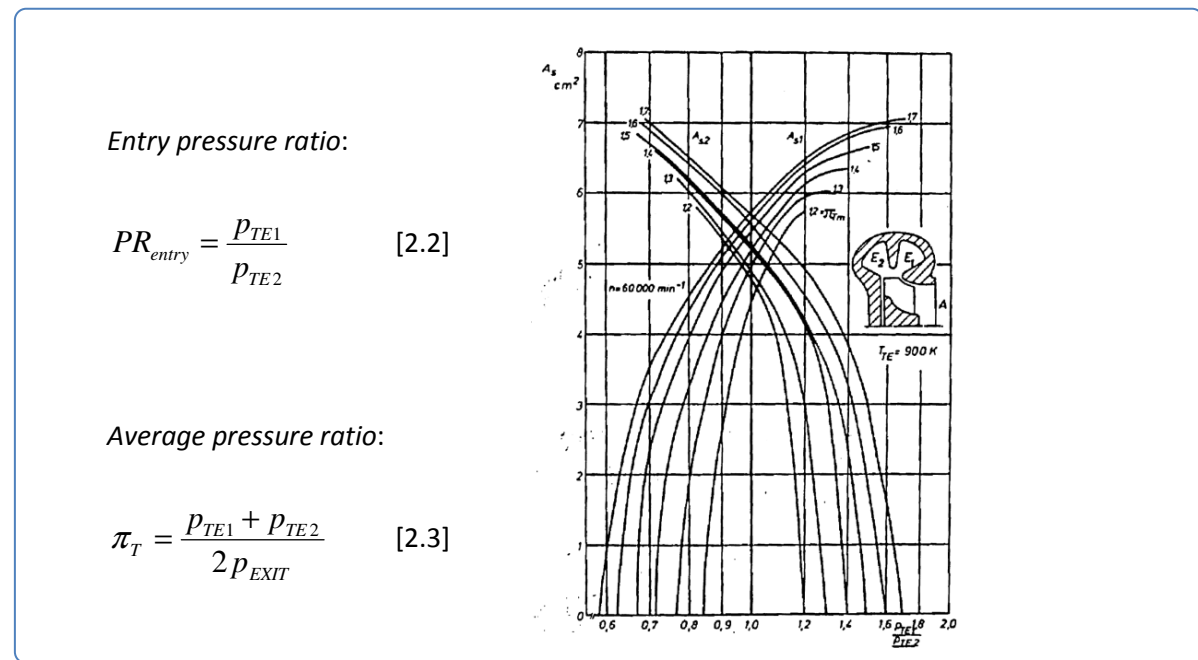


Figure 2.5: Isentropic flow area versus entry pressure ratio [5]

Pischinger and Wunsche [5] were among the first authors to comprehensively examine the issue of unequal admission in a multiple entry turbine. They recognized the importance of studying the influence of unequal flow in pulse turbocharging applications given that out-of-phase pulses cause conditions of temporary unequal admission. Testing was done in steady flow, with the stated assumption that this data would be applicable to the unsteady, pulsating case through a quasi-steady assumption. Most of the testing was performed with a vaneless, twin-flow housing with a radial inflow rotor. To present the flow characteristics, the turbine entries were treated as a pair of nozzles with a varying isentropic flow area A_{eff} defined as:

$$A_{eff} = \frac{\dot{m}}{\rho_s \cdot C_{is}} \quad (2.1)$$

where $\dot{m}$ is the mass flow, ρ_s is the density and C_{is} is the isentropic flow velocity. This treatment is ideal for modelling the turbine as a pair of simple isentropic nozzles with varying areas. Figure 2.5 is a typical plot of this turbine showing the relationship between the isentropic flow area of each entry (A_{s1} and A_{s2}) and the ratio of entry pressures (p_{TE1} and p_{TE2}). The authors here plot lines of constant average pressure ratio and conclude that the flow area is highly dependent on the ratio of pressures between the entries. These plots also indicate the level of symmetry in the flow behaviour between entries.

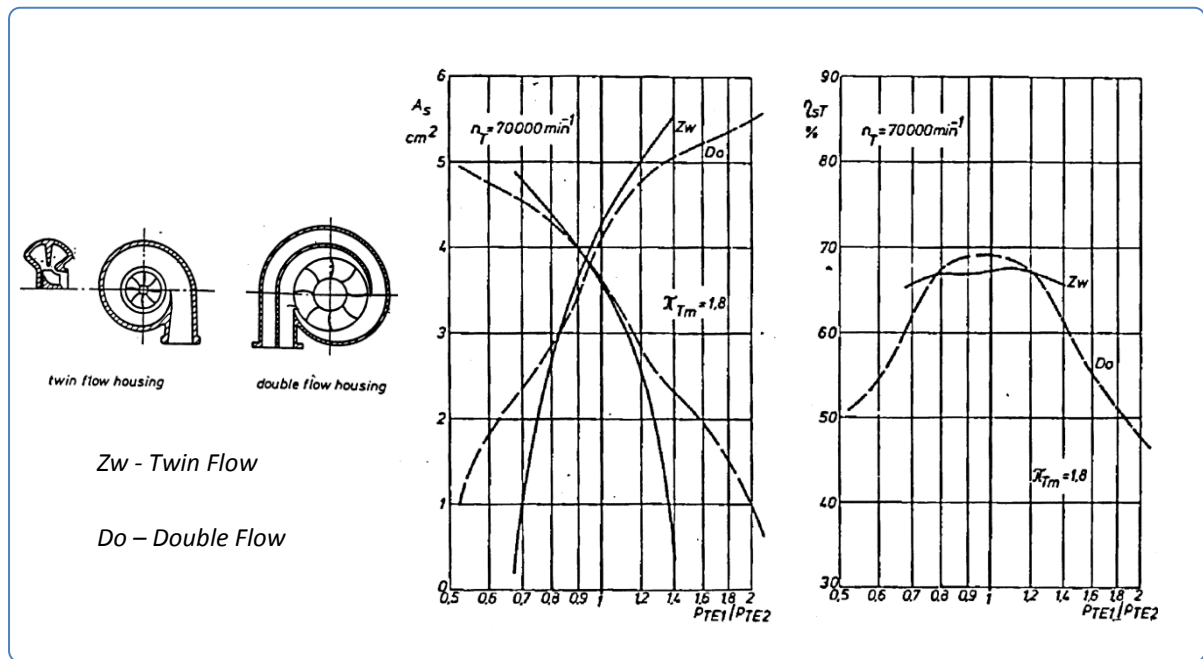


Figure 2.6: Isentropic area (LH plot) and efficiency (RH plot) versus entry pressure ratio [5]

When plotting efficiency versus entry pressure ratio, the authors found that the loss attributable to unequal admission can be quite large. A maximum efficiency loss of 12% was seen between equal and partial admission (at $\pi_T = 1.2$ & $n_T = 51600 \text{ rev min}^{-1}$). The authors also go on to compare the unequal admission behaviour of a twin-entry and double-entry turbine volute. The rotor was identical, and the two housings had the same equal admission isentropic flow area, yet the authors did not comment whether the turbines had a similar area-to-radius ratio (A/R) at inlet. Yet the comparison is useful in a qualitative sense since it highlights the vastly different performance characteristics between volute designs. Figure 2.6 shows the plots of isentropic flow area and efficiency versus the entry pressure ratio for the two types of volutes. The left hand plot demonstrates that the flow area change under unequal admission is more severe with the twin-flow housing than with the double-flow housing. The efficiency plot, on the other hand, displays a more

expected response to unequal admission. Since the double-entry housing will only feed one half of the rotor under partial admission, the rotor is essentially mismatched to the volute, and the loss of efficiency should reflect this. In contrast, the twin flow housing allows all 360° of the rotor inlet to be fed regardless of entry conditions, thus reducing the severity of the loss. The last section of the paper provides a short description of some preliminary computer modelling results using the experimental turbine data. The authors do not comment on the applicability of a quasi-steady approach but do indicate that the implementation of unequal admission into engine behaviour codes is necessary to improve the prediction methods.

Benson and Scrimshaw [11] considered a number of the issues that are pertinent to the current thesis. Like Mizumachi et.al [10], the turbine volute that was used in this study had a dividing wall that was arranged circumferentially in which each entry feeds a separate section of the nozzle ring. The volute is shown in Figure 2.7. This turbine was first tested using a steady flow test facility under equal and partial admission conditions. Two compressor sizes were used for power absorption to extend the range of the performance map, but extrapolation was still deemed necessary. The work done in the compressor was measured by thermodynamic energy balance. The test facility was operated cold at 400K from 30,000 to 60,000 rpm. Plots of torque and mass flow parameter versus pressure ratio are supplied for full and partial flow, but efficiency plots were not included.

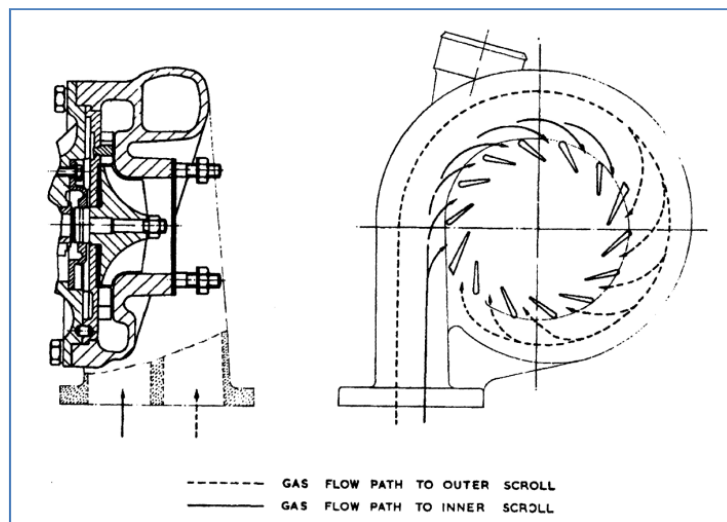


Figure 2.7: Volute design tested by Benson and Scrimshaw [11]

Transient flow testing was also performed with the turbine coupled to a six cylinder pulse generator (3 cylinders for each entry) which replicated the firing order of a typical internal

combustion engine. This produced a complicated pulse waveform with unequal, out-of-phase behaviour. The speed of the pulse generator was varied between 300 to 700 rev/min. A total of seventeen tests were carried out at a variety of turbine and pulse generator speeds. The only time-resolved parameters measured were inlet static temperature and pressure. This undoubtedly limits the ability to analyse the unsteady performance of the turbine. However, based on these limited measurements the authors sought to make a quasi-steady calculation of the unsteady behaviour of the turbine using full and partial steady flow data. Using the unsteady temperature and pressure measurements at 4° crank angle intervals, the corresponding steady flow operations were selected to give instantaneous mass flow rate, torque and efficiency. Integrating over the cycle yielded an estimation of the average quasi-steady performance based on steady full admission data and steady partial admission data. The ratio of efficiencies between the quasi-steady cycle averaged efficiency and the unsteady efficiency was plotted for a variety of pulse frequencies to show the validity of the quasi-steady analysis. These quasi-steady results will be discussed further in section 2.3, but it will be observed here that neither the partial nor full admission steady data produced a good quasi-steady-unsteady agreement. This led the authors to conclude that treating the pulsating flow turbocharger as quasi-steady would not lead to an accurate performance prediction. Overall, despite this study predating the present report by forty years, many of the issues considered here are still very relevant. However, the authors were limited in their ability to accurately analyse unsteady flows due to limitations in experimental equipment. Since the only transient parameters that could be measured were temperature and pressure, it was impossible to calculate any instantaneous performance parameters.

Wallace and Blair [13] tested two and three-entry turbines under a wide variety of conditions. This was the first in a series of papers looking at the performance of these turbines, an example of which is shown in Figure 2.8. The two-entry turbine is unusual because one entry of the volute produces a reverse swirl direction and thus must rely on the nozzles to reverse the flow and match the direction of the rotor. The unique shape of the volute should have a significant influence on the performance of the turbine - especially under unequal admission conditions. Therefore it is likely that the results from this work will not be suitable comparison to a turbine with a more conventional volute design. However, since this paper is one of only a few which report partial and unsteady flow performance of a multiple entry turbine with nozzles, it is useful to provide a brief review.

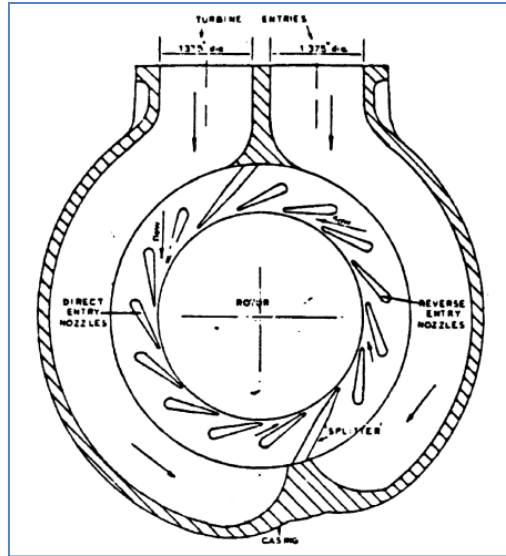


Figure 2.8: CAV two entry 01 turbine [13]

The test facility consisted of an air compressor to supply cold flow to the turbine and a throttled turbocharger compressor to control the speed and power absorption. Power was estimated by measuring the compressor pressure and mass flow in conjunction with known compressor efficiency characteristics. The authors acknowledge a large uncertainty associated with the power measurement due to unaccounted bearing loss. The unreliability of the power measurements makes it difficult to draw concrete conclusions from some of the performance data. Partial admission tests were also performed by blanking off one of the three entries at a time. This resulted in a significant efficiency loss of 34 to 42.5 percent when compared to the steady flow equivalent. The authors comment that these numbers may be pessimistic owing to the large bearing losses associated with unequal admission. Like Benson et. al. [11], the authors also point out that were they to use these efficiency values to predict unsteady flow using the quasi-steady assumption, it would lead to considerable efficiency underestimation. Following the steady-state tests, a rotary valve was installed to supply unsteady, pulsed flow to the 3-entry turbine. The pulsating pressures from these tests were used to make a quasi-steady comparison. The results will be briefly discussed in Section 2.3.

Wallace is also the lead author on a number of other papers extending the work outlined above using similar turbines and experimental techniques. Wallace, Cave, and Miles [15] looked at the performance of inward radial flow turbines under steady flow conditions. This paper describes an extension to a 1-D code to predict the steady flow performance of a turbine with partial admission as well as rotor choking, variable nozzle angle and loss coefficients. The computational results were compared to the steady flow results of a double entry turbine. Full admission prediction

faired well for both mass flow and torque. However, the predictive capabilities of the code did not perform as well under steady partial admission conditions. The subsequent paper by Wallace Adgey and Blair [14] sought to extend this model further to predict unsteady flow performance. A 3-entry turbine coupled to a high-speed dynamometer was fitted to a cold flow test facility capable of supplying unsteady, pulsed flow through a rotary valve. However, the flow pulses were supplied 120° out-of-phase and thus produced conditions of unsteady and partial admission. Thus, since the model did not adequately account for partial admission, the code vastly overestimated the available power when compared to experimental data.

The last paper to be published on this continuing topic was aimed at improving unsteady prediction for multiple entry turbines [16]. The authors performed tests on one, two and three entry turbines under unsteady conditions with pulses out-of-phase with respect to the number of entries. An interesting comparison between the three turbines could be made to show the change in performance when the number of entries is changed. The results showed that for a given average unsteady pressure ratio and pulse frequency, there was an increase in torque and mass flow as the number of entries increased. The authors comment that this is attributable to a reduction in the amount of time the rotor experiences no-flow in a pulse cycle as the number of entries increases. The authors also report a good agreement between the results of an improved quasi-steady and unsteady computational model with experimental data. For the double entry turbine, both the pulse and quasi-steady programmes gave good agreement in terms of mass flow, but power was underestimated by the latter and over estimated by the former.

2.2 Twin-entry Performance

In general, all the above papers which look at multiple-entry radial turbines where each entry feeds a separate part of the rotor fail to provide conclusive results. This is likely due to the fact that none were able to measure instantaneous mass flow and power due to the limitations in available equipment. It was with this in mind that Dale and Watson [9] sought to develop a test facility at Imperial College capable of measuring these instantaneous parameters in a twin-entry turbine. The instantaneous mass flow was measured using hotwires, and the torque was derived from the average load cell reading and instantaneous acceleration. The turbine was tested under a range of equal and unequal admission steady flow conditions. Given that an extra inlet condition introduces an extra degree of freedom, it becomes difficult to represent the range of unequal admission conditions on a single plot. The authors therefore chose to represent the unequal states

by producing many different figures, each corresponding to a single non-dimensional speed. This representation therefore requires plenty of paper space but has the advantage that the result will be in a familiar form. The authors also show some preliminary in-phase, unsteady tests which show the hysteresis between pressure and mass parameter that is discussed in later works.

Yeo and Baines produced two papers [18] [19] on the strength of a PhD thesis [17]. These papers build on the work of Dale and Watson [9] since the same turbine was used with the same test facility. The objective was to measure the velocity triangle at rotor entry under steady, unequal, and out-of-phase pulsed flow. Unlike the double-entry design, the twin-entry volute mixes the two streams in the vaneless space just prior to rotor inlet. The authors found that the flow angle into the rotor varied significantly when the two inlets were fed unequally. This would seem to indicate that a significant amount of unequal entry loss is attributable to incidence losses in the twin-flow housing. While these observations do not directly relate to the double-entry design in this thesis, they do explain why Pischinger and Wunsche [5] measured very different performance characteristics between the two different volutes shown in Figure 2.6. The authors were also able to measure the instantaneous flow angles under pulsating flow. As one might expect, these measurements showed a significant variation in the flow magnitude and direction relative to the rotor at inlet. This data was therefore a first in demonstrating experimentally that the result of an unsteady flow is a wide variation of inlet conditions directly relating to performance loss. The authors also point out that the unsteady flow characteristics were quite similar to the steady flow cases with a similar mass flow. This remark, along with the observation that the pulsation frequency is much less than the rotor blade passing frequency, leads them to also suggest that the fluid in the rotor can be considered quasi-steady.

The publication by Baines, Hajilouy-Benisi and Yeo [24] sought to build further on the unsteady investigation of Dale and Watson [9] with an in-depth analysis of unsteady turbine performance. By increasing the time resolution of the instantaneous data acquisition equipment, they could obtain a more precise picture of the operating point of the turbine over a single pulse cycle. They also took the argument of Yeo et.al [17,18,19] further by suggesting that while the rotor may be considered quasi-steady, the volute will not act in a quasi-steady manner due to a filling and emptying of its volume. Based on this assumption, they published results from a very basic computational model that treats the volute as a reservoir that can transiently fill and empty over a pulse cycle. Nevertheless, the main thrust of this work was to analyse the unsteady behaviour of a twin-entry turbine with particular emphasis on the influence of the out-of-phase pulsations (Figure 2.9).

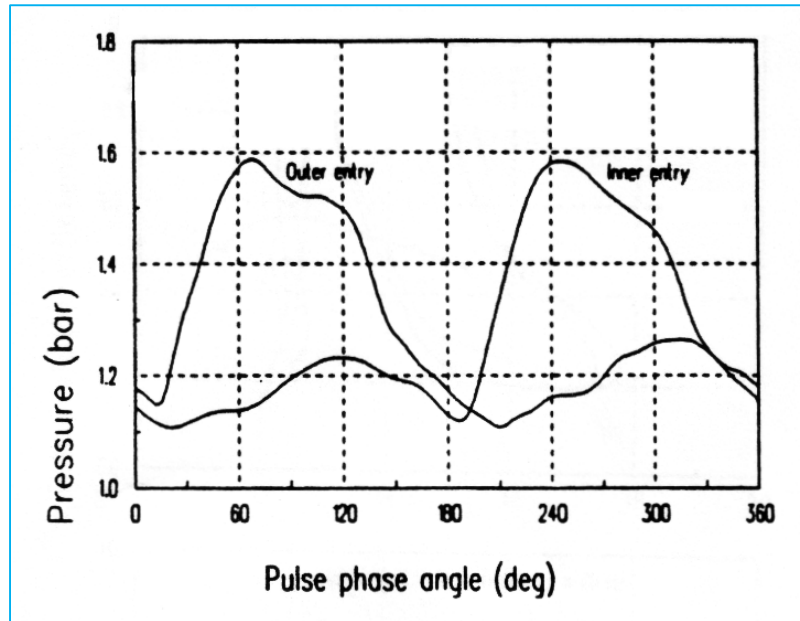


Figure 2.9: Out-of-phase pulses introduced into the twin-entry turbine [24]

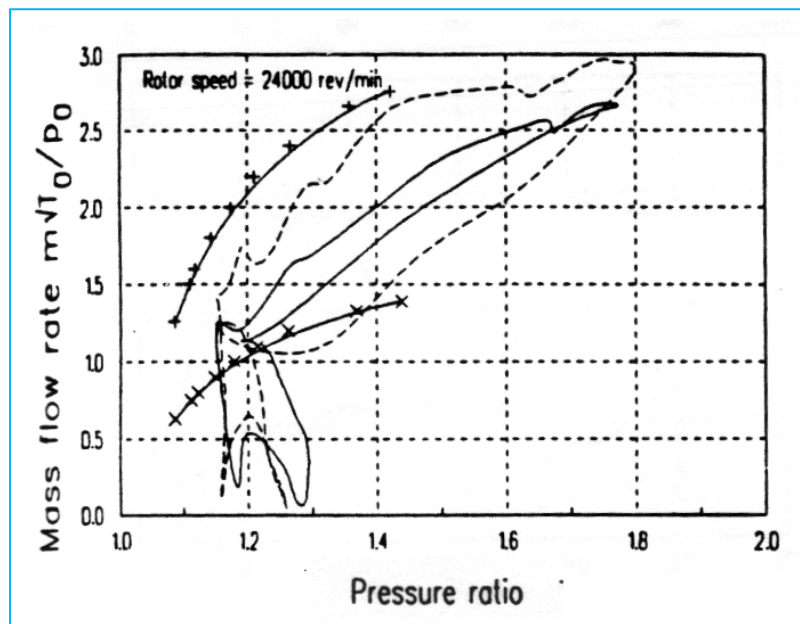


Figure 2.10: Unsteady out-of-phase mass parameter vs pressure ratio [24]

The instantaneous efficiency, mass parameter and pressure ratios were plotted in the standard steady-state representations. The mass parameter versus pressure ratio plot such as the one shown in Figure 2.10 are analysed in some depth by the authors. The most striking feature of these orbits (two - one for each entry) compared to a single-entry turbine is the presence of a

smaller secondary loop. Since this smaller loop occurs when the pulse generator for this inlet is closed, the change in conditions indicated by this loop must be due to an interaction with the second inlet where the pulse generator is open. This effect will be particularly pronounced in this twin-entry, nozzle-less design due to the large space available for flow interaction when there is a large pressure difference between inlets. The same effect is also suggested in Figure 2.9 by the small rise and fall in pressure in the stagnant entry that coincides with the main pulse in the other entry.

Hajilouy-Benisi went on to contribute to a number of papers on the subject of unsteady flow in a twin-entry turbine using both numerical and experimental techniques [36-38]. These works will not be reviewed in great depth due to their limited relevance to the current thesis. A brief overview is provided here for completeness. Ghassemi et.al [36] used the experimental data of Yeo and others to form the basis of a 1-D model that included loss coefficients modified to reflect the character of the twin-entry turbine. This work reported some success in replicating the steady-flow characteristics. Aghaali and Hajilouy-benisi [37] further improved this 1-D model and compared the results with new equal and unequal steady-flow tests on a twin-entry turbine. Shahhosseini et.al [38] is the latest paper in this line of research, but instead of using 1-D modelling, a full 3-D CFD model of the turbine was developed to examine the complex three-dimensional flow field resulting from the twin-entry volute. By comparing results from full and partial admission 3-D simulations, they were able to understand some of the sources of loss attributable to unequally admitted flows. The results from these tests were compared to experimental performance data as well as exit velocities from a 5-hole probe.

In addition to the various works of Hajilouy-benisi, there have been other authors seeking to build a viable model of the twin-entry turbine. Katrasnik [39] published work outlining a model of the twin entry turbine intended to be used as a boundary condition within an engine wave action simulation. The code does not appear to take into account the pulsating nature of the flow, but the authors claim good agreement with experimental engine transient data. One of the most complete and promising attempts at a fully unsteady 1-D, twin-entry turbine model is that of Costall, et. al. [40]. These authors sought to extend the single-entry turbine model reported in previous publications [7,35] to model the pulsating flow in a twin-entry turbine. They were able to use steady and unsteady experimental results from the test facility at Imperial College to validate the simulations. They demonstrated excellent agreement between the single-entry model and the twin-entry data while operating in-phase. This is hardly surprising since a twin-entry turbine will act very much like a single-entry turbine when the flow is introduced into each gas inlet equally. However, since the turbocharger rarely encounters in-phase pulsations on a real engine, out-of-phase pulses

are much more important to be able to model accurately. A new twin-entry 1-D domain was constructed to take into account some of the physical characteristics of the twin-entry turbine. This updated model improved the in-phase agreement but significantly underestimated the mass flow and power from the out-of phase experiments. The authors suggest that this is due to an incorrectly calibrated rotor loss coefficient attributed to insufficiency steady, unequal and partial admission data. The work is ongoing, but this first paper is certainly promising.

2.3 Pulsating Flow and the Quasi-Steady Assumption

The turbocharger engineer has long been aware of the highly unsteady nature of the flow driving the turbine wheel yet many of the improvements in turbomachinery design have been based on the assumption of steady flow through the turbine. Even in the past 15-20 years when the ability to measure all the instantaneous quantities of a pulsating flow has been proven by a number of researchers [9, 17-19, 24-34], there has been a reluctance to include the true pulsating nature of the flow in the design process. There are a few reasons why engineers might shy away from the unsteady issue. First, there is the complexity that an unsteady flow introduces to the aerodynamic problem. Even steady flows through the turbocharger are complex three-dimensional flows with secondary effects, clearance leakages, etc. Thus, to reduce the problem to a manageable size, the steady-state assumption has largely been adopted. In addition, there is a stated assumption in some of the earlier research that the turbocharger should behave in a quasi-steady state manner. This assumes that the behaviour of the turbine at each instant in the pulse cycle will be identical to the behaviour if the same set of conditions were introduced in a steady-state manner. Thus, given a wide enough steady-flow turbine map, the unsteady pressure pulse can be discretized, and each set of instantaneous conditions can be matched to a corresponding steady-state operating point in the map to provide a prediction of overall pulsating performance.

With this in mind, early researchers into the unsteady performance of turbochargers such as Benson *et. al* [11,12], Wallace *et. al.* [13,14,16], Kosuge, *et al* [20] and later Capabianco *et. al* [21,22,23] all sought to quantify the degree to which the turbocharger acted in a quasi-steady manner. Since fast response pressure transducers are easily implemented, most researchers were able to directly measure instantaneous pressure at various locations along the turbine stage. The unsteady pressure ratio data in conjunction with averaged turbine wheel speed could then be matched with the corresponding steady-flow conditions to obtain a dynamic, quasi-steady 'equivalent'. However, since instantaneous unsteady torque, mass flow and temperature were not

measured, the quasi-steady data could only be compared with unsteady performance on the basis of time-mean values. The most convenient method of comparison was to calculate the quotient of the mean unsteady and quasi-steady values, thereby producing parameters such as I_M , and I_P which compare mass flow and output power respectively (Equations 2.4 and 2.5). Note that some authors chose to use the inverse of these equations, but the intent is the same.

$$I_M = \frac{\dot{m}_{US}}{\frac{1}{T} \int_0^T \dot{m}(t)_{QS} dt} \quad (2.4)$$

$$I_P = \frac{\dot{W}_{US}}{\frac{1}{T} \int_1^T \dot{W}(t)_{QS} dt} \quad (2.5)$$

Wallace and Blair [13] were among the first to attempt to test whether a turbocharger turbine behaves quasi-steadily. A rotary valve was installed to supply unsteady, pulsed flow to the 3-entry turbine similar to the one shown in Figure 2.8. Apart from the pressure, no other instantaneous measurements were made, thus making the pressure ratio the basis of the quasi-steady comparison. The other major limitation to these earlier authors was the availability of a sufficient quantity of steady test data for making a true quasi-steady comparison. Since Wallace et. al [13,14,16] and Benson et.al [11] were both studying multiple entry turbines, a true quasi-steady comparison should take into account that the two or three inlets will not be fed equally when there is a phase difference between the pulses. For Wallace and Blair [13], the unequal admission was simply ignored and the quasi-steady calculation was made using steady, full admission data as a basis for the prediction. The authors report an error between the quasi-steady prediction and measured unsteady power of up to 25 percent but report a reasonable prediction of the general trends. They also comment that there are indications that the quasi-steady estimation becomes less accurate as the pulse frequency is increased but provide little detail.

At a similar time, Benson and Scrimshaw [11] studied a double-entry turbine as outlined in Section 2.1. They produce complete equal and partial admission (one inlet blocked) steady performance maps as a basis for the quasi-steady comparison. However, the pulse shape produced by their pulse generator produced a range of instantaneous unequal conditions through the turbine. Therefore, their use of the equal or partial admission performance maps as the basis of the quasi-

steady analysis was not strictly appropriate. The result was that the quasi-steady averaged power based on *partial* admission was much less than the measured unsteady power owing to the large losses associated with partial admission. The quasi-steady values based on *full* admission data fared better but still underestimated the mass flow and the power produced under unsteady admission. The ratio of unsteady to quasi-steady efficiency suggested that the unsteady operation was more efficient than the quasi-steady equivalent. Although this result is unexpected, it is also interesting to note that they show a trend of increasing unsteady efficiency (compared to the quasi-steady prediction) as the pulse frequency *increases*.

Benson [12] went on to publish a second paper looking into the validity of the quasi-steady assumption under a greater variety of conditions and sought to establish factors that could be used to correlate the unsteady pulsed flow to the steady flow data. The test facility was fed by cold flow through a single rotating pulse generator with two ports per revolution, thus creating equal, in-phase pulses into the double-entry, nozzleless turbine. Like the previous papers, the only dynamic data that was measured was the pressure fluctuation. Using just this pressure trace, the process for calculating a quasi-steady 'equivalent' was outlined as follows: first, the transient temperature is calculated using an isentropic relationship with pressure. Then, at a given instant in time, by using the transient pressure ratio and speed parameter (which is related to velocity ratio) the corresponding values of steady mass flow, power and efficiency can be interpolated from the steady map. These were then averaged to calculate the correlations similar to equations 2.4 and 2.5. In addition, a *form factor* of the pulse profile was calculated by dividing the root mean square of the pulse shape by the mean value. The author felt that this value, which ranges from a value of 1 (steady flow) to 1.11 (half sine wave), may prove to be a good indicator of the deviation from the quasi-steady assumption and unsteady test results. The correlations I_m and I_p were plotted against the form factor to test the relationship between the form of the pulse and the domination of unsteady effects. Their data suggested that the form factor had a greater influence on the departure from quasi-steady than the pulsation frequency. This implies that as the pressure waves approach a sinusoid, the quasi-steady assumption breaks down and unsteady effects predominate. The magnitude of the correlations I_m and I_p are somewhat puzzling since they are all below unity suggesting the opposite to his previous work: that the quasi-steady calculation *over-estimates* mass flow and power. However, the author simply states that the intent of the work is to look at qualitative effects, not quantitative.

Kosuge et.al [20] published one of the most comprehensive studies looking at the comparison between averaged quasi-steady versus unsteady performance. Like Wallace et.al and

Benson et.al, the authors relied solely on instantaneous pressure to derive the average quasi-steady - unsteady quotients of mass and power, $\bar{G}_q/\bar{G}_m$ and $\bar{L}_{eq}/\bar{L}_{em}$ (the inverse of equations 2.4 and 2.5). Figure 2.11 show the results plotted against pulse frequency. This figure demonstrates a tendency of the quasi-steady prediction to under-predict both mass flow and power, but interestingly also reveals a contrasting influence of pulse frequency. The top figure clearly shows that the quasi-steady mass flow predictions approached the unsteady value as the frequency increased, but the measured power yields the opposite trend, with the quasi-steady and unsteady averages diverging with increasing frequency.

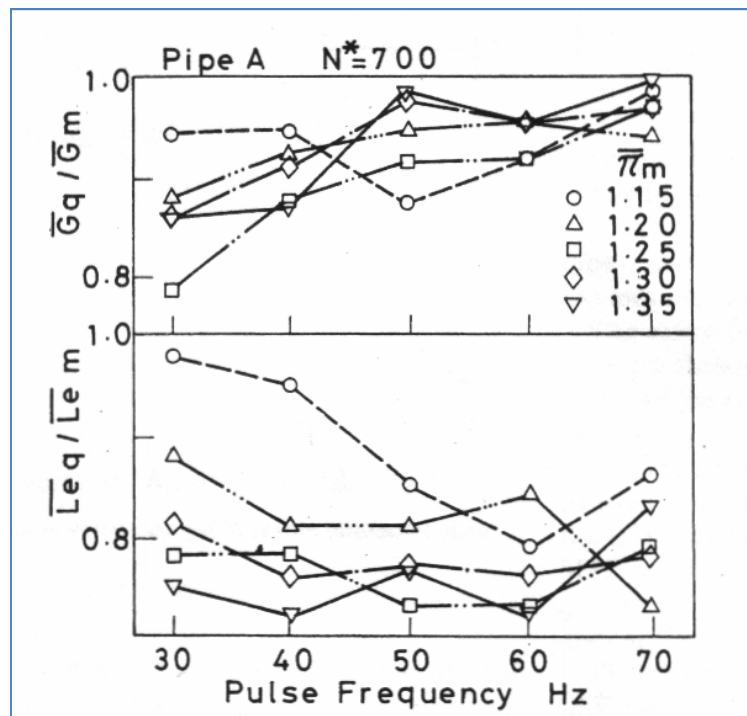


Figure 2.11: Averaged quasi-steady – unsteady quotient of mass flow and power plotted against pulse frequency [20]

These observations lead the authors to conclude that the frequency is not sufficient to fully describe the source of the departure from quasi-steadiness. They suggest that in addition to frequency, the pulse shape and amplitude must also have an impact on the unsteadiness in the turbine. To encapsulate these additional features, they derive an “equivalent amplitude” by non-dimensionalizing the path integral of the pressure pulse profile. The result is equation 2.6 below:

$$Equiv. Amp. = \int_0^{2\Delta t} \sqrt{\left(\frac{\Delta p}{\Delta t}\right)^2 + \left(\frac{dp}{dt}\right)^2} \cdot dt / \bar{p} \quad (2.6)$$

The authors then plot the quasi-steady – unsteady ratios against this new parameter in Figure 2.12. It would seem from this plot that in comparison with a frequency dependent parameter such as the Strouhal number, the equivalent amplitude does seem to provide a clearer correlation to the departure from quasi-steady operation. The difficulty with the equivalent amplitude presented in equation 2.6 is that it carries no real physical meaning. The Strouhal number, on the other hand, compares the travel time of a particle over a selected length scale to the length of time of an unsteady event. Thus, despite the fact that the Strouhal number seems ineffective here, it is derived from a physical process that is linked to the quasi-steady problem.

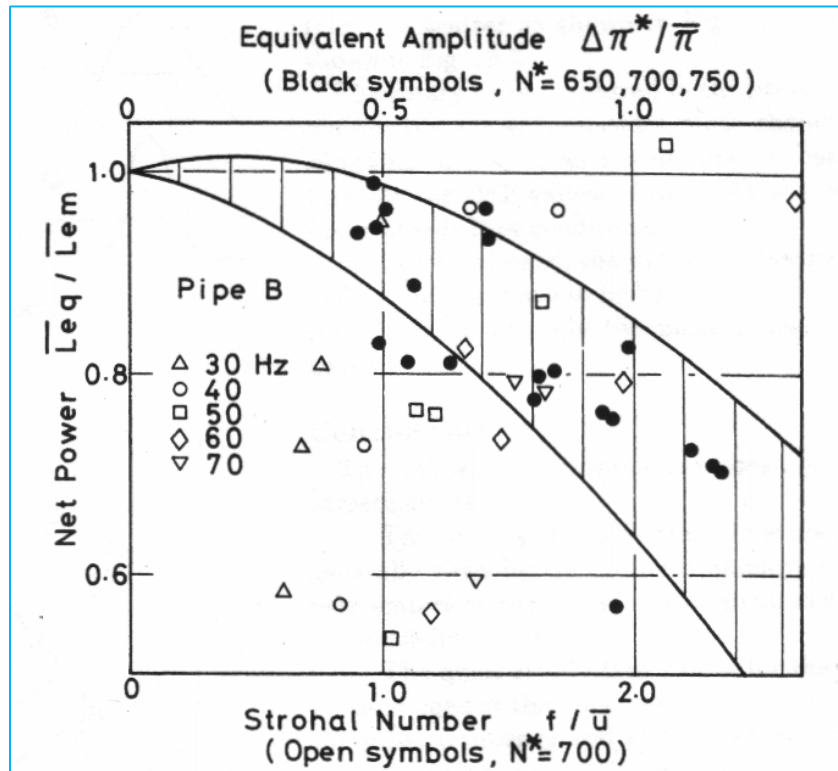


Figure 2.12: Averaged quasi-steady – unsteady quotient of power plotted against Strouhal number and equivalent amplitude [20]

Similar to Kosuge, Capobianco et. al [22] did considerable work on the differences between steady and unsteady behaviour. By using a single-entry turbocharger connected to a pulse generator capable of adjusting pulse amplitude and frequency separately, they could study the effect of each

of these properties on an individual basis. First, they compared the unsteady and steady flow performance on the basis of a single averaged set of parameters. In other words, the value of pressure ratio and speed which was averaged over the entire pulse was compared to the steady state that corresponded to these values. Since this was not a quasi-steady analysis and takes no account of the pulse shape at all, it is not surprising that the steady and unsteady performances based in this comparison did not match. He also performed a quasi-steady analysis using the same technique as Benson [12] and calculated the influence factors I_m and I_p . Like other authors, this data showed that if the turbine is assumed to be quasi-steady, the result would be an underprediction of the true averaged mass flow and power. The data published did not, however, demonstrate any appreciable influence of pulse frequency. Instead, the authors suggest that the amplitude of the unsteady pulse appeared to have a greater impact on the departure from quasi-steady behaviour than the frequency.

To sum up this set of early works [11-16, 20-23], all the authors sought to only use instantaneous pressure measurements to assess if, when averaged over a pulse cycle, the turbocharger turbine performs in a quasi-steady manner. Although there is some disagreement, most of the research produces values of I_m , and I_p above unity, thus indicating that the quasi-steady assumption tends to under-predict averaged mass flow and power. There is, however, conflicting data concerning the reason for this deviation. Some authors show an influence of pulse frequency but most have the opinion that there must also be an influence of pulse shape and amplitude. Benson [12] and Kosuge et.al [20] therefore suggest using a pulse-shape parameter to obtain a better correlation with I_m and I_p . However, there is no clear case for linking these parameters with the nature of unsteady flow in a turbine. These ‘shape’ parameters merely stem from an intuition that the pulse amplitude must have an influence on the accuracy of the quasi-steady assumption.

Since the approach of comparing time-mean performance data has not been able to provide a definitive consensus on the underlying cause of the departure of the unsteady behaviour from the steady case, to begin to understand the physical processes which drive unsteady behaviour, time-resolved flow data must be acquired over a pulse cycle. Dale and Watson [9] were among the first to comprehensively measure all necessary instantaneous unsteady quantities, followed by Baines et al. [24], Arcoumanis et.al [25,26], Karamanis et.al [27,28], Szymko et.al [8,29], Rajoo et. al [30,31,32] and Copeland et.al [33,34]. This body of work has yielded a number of insights into the effect of a pulsating flow field on turbine operation. Some of the more fundamental discoveries are as follows:

1. The volume contained in the volute prior to rotor entry can act as a reservoir which can accumulate and empty mass over a pulse cycle. In addition to this ‘filling and emptying’, there is a

further influence of wave dynamics along the passage length. The balance of these two effects determines the dynamic behaviour of the flow entering the rotor and therefore the power it produces.

2. For the range of frequencies typically encountered in an IC engine, the rise in pressure is too rapid to allow the mass accumulation to follow. This hysteresis (or lag) between pressure and mass flow generates an operating orbit that may or may not encircle the quasi-steady operating line (Figure 2.11). This effect has been replicated in 1D wave action simulations by Costall et.al [7,35].

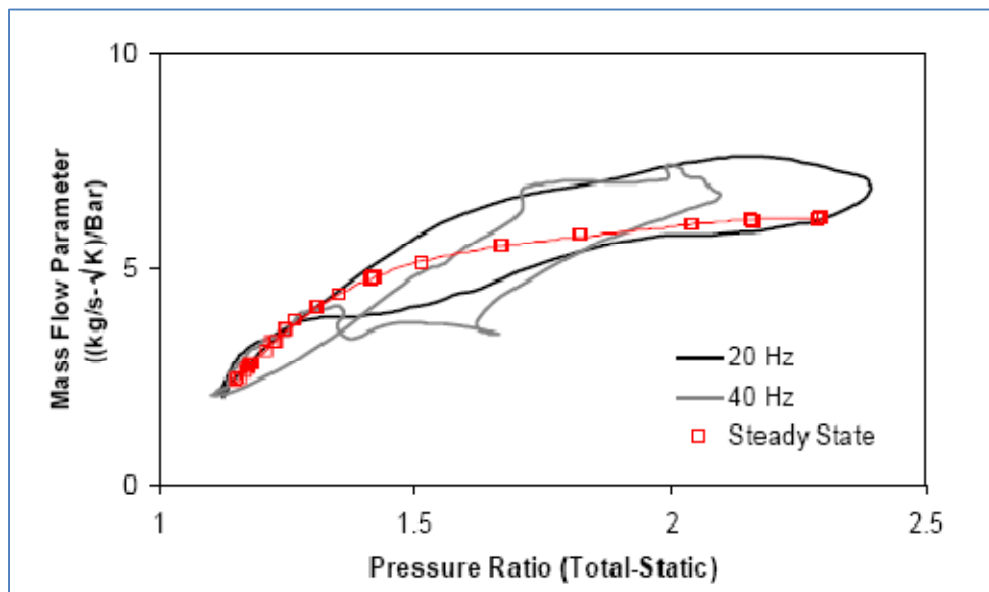


Figure 2.13: Mass flow parameter vs pressure ratio. Comparison between the steady operating line and the unsteady instantaneous operation orbit. [29]

The luxury of measuring all the time-resolved flow and torque data means the instantaneous performance of the turbine can now be directly compared to the steady-state operation without having to rely on averaging techniques. Thus, Figure 2.13 compares the mass parameter (MP) versus pressure ratio (PR) characteristics of a single entry, nozzleless turbine operating with a steady flow to the unsteady behaviour under a 20Hz and 40Hz pulse [29]. This plot clearly demonstrates the hysteresis mentioned in point 2 above but also shows that on an instantaneous basis, the turbine does not operate in a quasi-steady manner when exposed to a pulsating flow. If it were a strictly quasi-steady process, the unsteady operating orbit would collapse onto the steady-state line. This departure from the steady-state relationship between pressure and mass flow has been a topic of some debate in these investigations. One of the clearest attempts to explain this phenomenon was

that of Szymko et.al [29]. They proposed three modes to describe the relationship between pressure and mass flow over a pulse cycle. The first involves purely quasi-steady behaviour where the pressure change over a pulse is sufficiently slow to permit a state of equilibrium to exist between mass and pressure. In reality, the frequency of the pulse must be very slow for this to occur in a turbine. As the increase in frequency begins to produce the hysteresis between pressure and mass flow described earlier, a second mode is proposed where an operating orbit results which encapsulates the steady flow behaviour. If the frequency is increased still further, the final mode is reached where the turbine will start to depart from steady operation altogether, and the operating loop will no longer encapsulate the quasi-steady line. To distinguish between these three modes of operation, the authors suggested the Strouhal number as a useful indicator. The Strouhal number defines the rate of localized changes of a moving particle in comparison to its rate of propagation. Szymko et.al [29] modified the basic definition slightly (Equation 3) to include the parameter ϕ which accounts for pulse lengths which are a fraction of the total wavelength and also included a factor of $\frac{1}{2}$ which normalizes the frequency such that the pulse event constitutes half of the wavelength. The result was the *Modified Strouhal Number (MSt)* as defined by equation 2.7.

$$MSt = \frac{fL}{U} \cdot \frac{1}{2\phi} \quad (2.7)$$

Since the Strouhal number is, in essence, a comparison of time scales, Szymko et.al [29] realized that the time of the pulse event could also be compared to the rate of propagation of the pressure wave. This led to the definition of the *Pressure Modified Strouhal Number (PMSt)* in equation 2.8:

$$PMSt = \frac{fL}{U + a} \cdot \frac{1}{2\phi} \quad (2.8)$$

The authors suggested that the unsteady effects start to dominate above a Strouhal number of 0.1. Although this is somewhat subjective, the authors felt that this value was most appropriate on the basis of their experimental results. Applying this criterion to the *MSt*, all of the pulsating test cases from 20Hz to 80Hz were judged to be unsteady with respect to the bulk flow propagation. However, the same criterion applied to the *PMSt* indicated that the pressure wave only became unsteady above a frequency of 40Hz. This suggests the three modes of ‘unsteadiness’ [8]:

1. $MSt < 0.1$: Quasi-steady operation can be assumed
2. $PMSt < 0.1 < MSt$: No longer quasi-steady with filling and emptying effects dominating
3. $0.1 < PMSt$: Unsteady effects dominating both the bulk flow and pressure waves

Szymko based the validity of this approach by noting that an increase in the Strouhal number caused an increasing difference between the trend of the unsteady MP versus PR orbit and the trend of the steady line. Costall et.al [35] took this approach further by identifying the transition to a fully unsteady regime using the slope of the unsteady orbit calculated from his 1-D numerical code ONDAS. Figure 2.14 shows that there is a definite increase in the slope of the unsteady orbit in comparison to the steady operating line as frequency is increased.

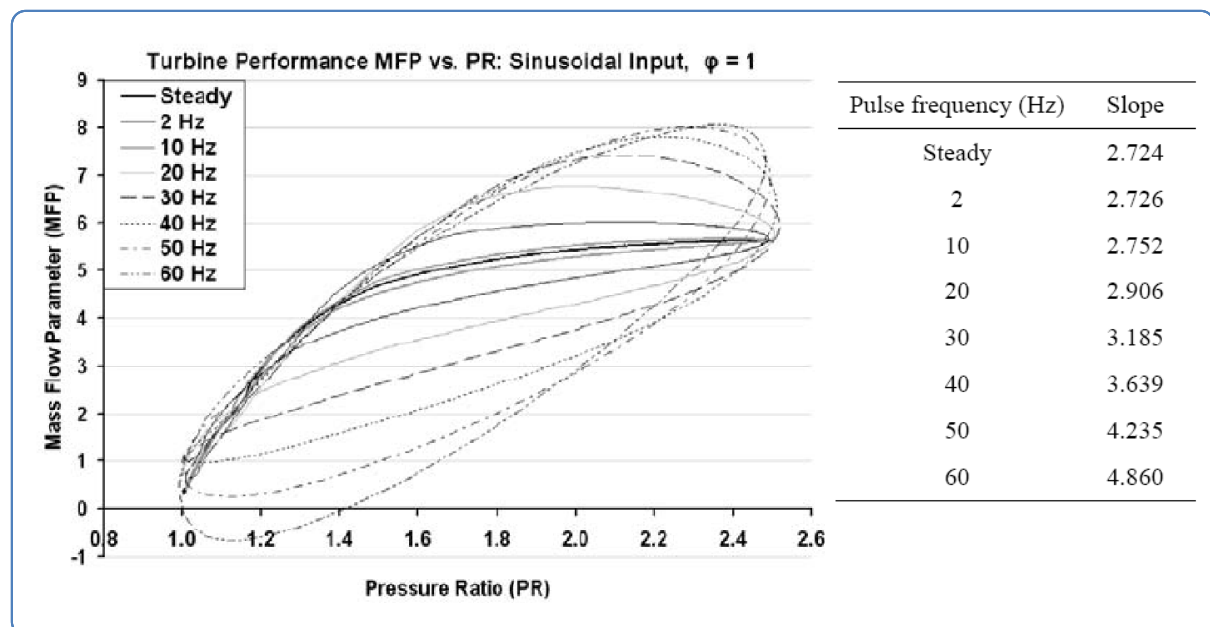


Figure 2.14: LHS: Unsteady operating orbits from a sinusoidal waveform at different frequencies. RHS: Values of the slope calculated by linear regression [35]

These results seem to show that the increase in pulse frequency (and Strouhal number) will lead a departure from the steady-state relationship between mass parameter and pressure ratio. However, these authors do not expressly address whether or not the Strouhal number can be used to describe an average performance deficit that is caused by a pulsating flow [29-32]. In fact, if the unsteady, cycle-averaged turbine efficiency is calculated and compared to the quasi-steady cycle-averaged efficiency, this appears to show the opposite trend than predicted by the Strouhal number. Figure 2.15 shows the ratio of unsteady to quasi-steady, cycle-averaged efficiency calculated by Szymko [8] and Rajoo [32], plotted against the frequency of pulsation. The trend from this plot is

clear: the lower frequency pulsations produce a lower efficiency than predicted by the quasi-steady analysis. This is puzzling since the foregoing discussion put forward by Szymko suggested that lower frequencies (lower Strouhal numbers) should demonstrate a closer agreement between unsteady and quasi-steady operation. These authors do not address these seemingly contradictory results in the use of the Strouhal number but simply suggest that further investigation is necessary.

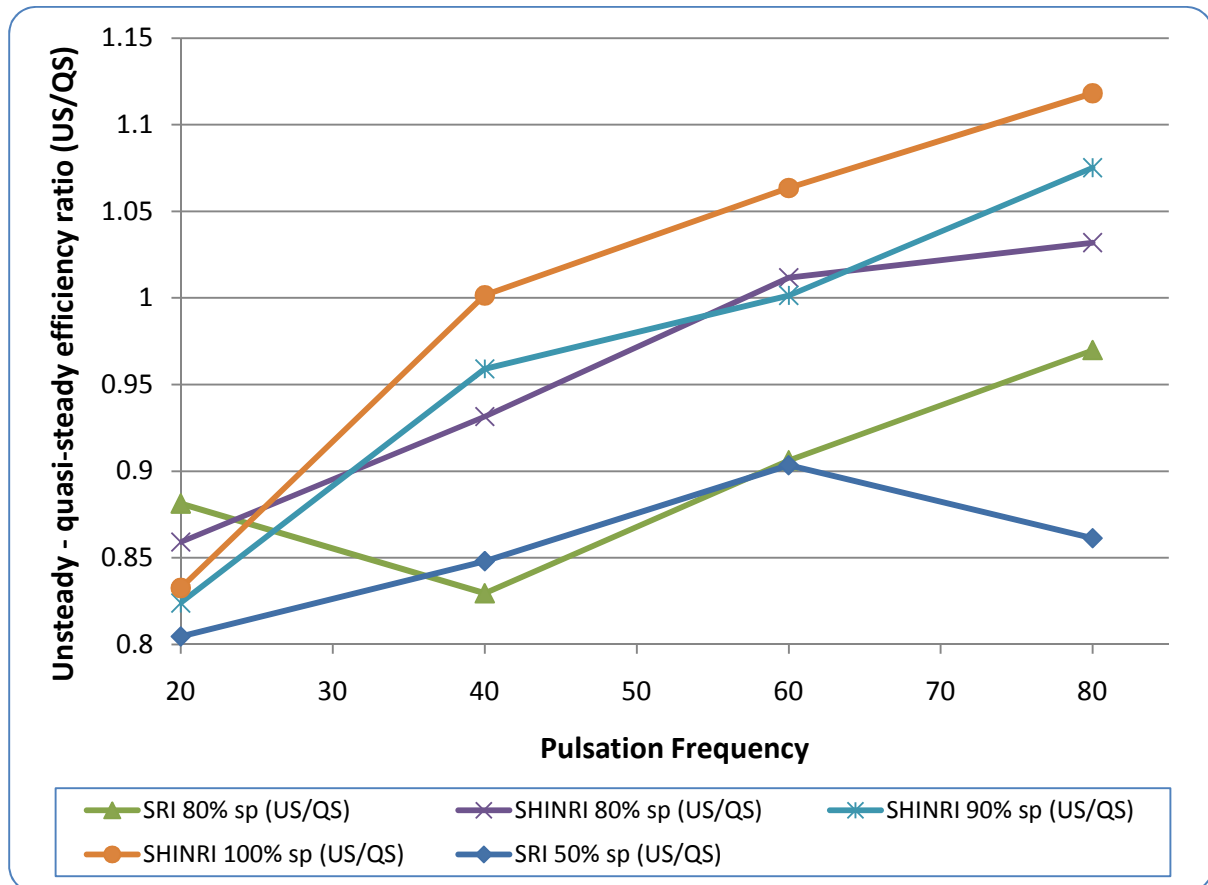


Figure 2.15: Ratio of Unsteady to Quasi-steady cycle averaged efficiencies at various pulse frequencies [8, 32]

In a standard fixed-nozzle turbine, the throat area is designed to pass the necessary mass flow for a particular application [4]. Thus, the influence of nozzle opening is quite straightforward to predict for steady flow. However, under unsteady flow, the addition of a nozzle ring produces some interesting effects that have yet to be clearly understood. Rajoo and Martinez-Botas [30,31] were among the first to comprehensively test unsteadiness in a nozzled turbine by recording all the instantaneous quantities. They had the advantage of a variable nozzle turbine where the effect of different nozzle openings could be measured under steady and unsteady conditions. They noted that

the nozzle acted to isolate some of the unsteadiness in the volute passage from the interspace volume upstream of the turbine rotor. Thus, there was less of a match between the input and output power time-traces compared to results shown in Szymko, et. al [29]. Also, since the nozzle area controls the swallowing capacity of the turbine, it therefore determines how quickly the scroll volume can empty the mass that has accumulated over a pulse. Thus, the nozzle geometry has quite a large impact on the relationship between mass flow and pressure measured at the turbine inlet. Most interestingly for the current discussion, if steady and unsteady operating lines for a particular nozzle opening were plotted together, the degree to which these behaviours coincided strongly depended on the nozzle opening area. This effect can be clearly seen in Figure 2.16 where the same pulsating conditions produced very different unsteady operating curves for two different nozzle stagger angles. Previous to this, steady–unsteady disagreement was generally believed to be frequency or amplitude dependent, but this research suggested that the nozzle might also have some effect.

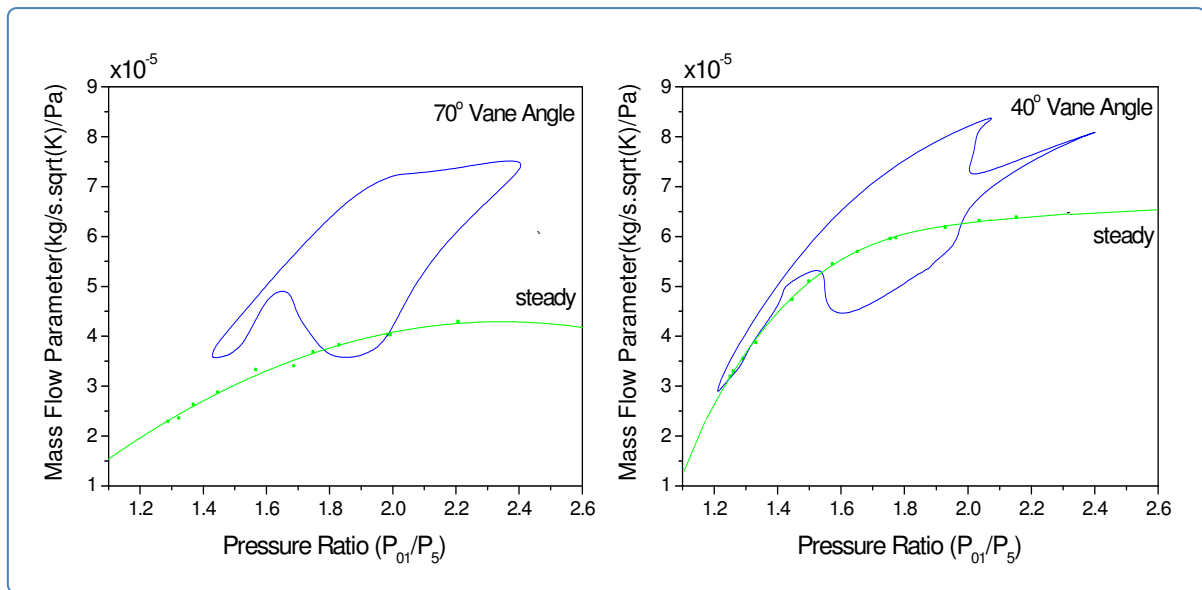


Figure 2.16: Turbine Swallowing Capacity at 40Hz Pulsating Flow with 70° and 40° Nozzle Vane Angles [32]

The quasi-steady assumption becomes much harder to evaluate in a two-inlet scroll. Unless the unsteady pulsations arrive at the turbine completely in-phase, there will be two distinct flow conditions driving the turbine wheel over much of the pulse cycle. Since this unequal flow has been shown to produce significant performance implications under steady flow conditions [5,10], such test data should be used in the quasi-steady analysis. Benson and Scrimshaw [11] tested a double-entry, nozzled turbine exposed to two pulsating inlet conditions that were not in-phase.

Nevertheless, as indicated earlier, they could only calculate averaged quasi-steady parameters based on either equal or partial admission steady flow maps. Their analysis was therefore not a true quasi-steady comparison. Work by Dale and Watson [9] in a twin-entry turbine was restricted to in-phase pressure pulsations, essentially simulating a single-entry turbine. Further work by Baines, et.al [24] did cover out-of-phase pulsations, but did not publish a quasi-steady comparison likely due to the prohibitive amount of steady data required. In fact, for this reason, it appears as if a dual-entry, quasi-steady comparison using the full range of unequal admission data has not been attempted.

2.4 Summary

Table 2.1 provides a full summary of the literature discussed in this chapter, organized by the design of the volute (double, twin and single entry) and by the type of research presented (computational, experimental, steady and unsteady). From this table, note that the literature available on the topic of unequal and unsteady flows in a double-entry volute is somewhat limited. With the exception of the work published on the basis of the research contained in this thesis [33-34], all of the research on the double-entry turbine is 30-40 years old. While this in itself does not permit immediate dismissal, it does suggest that modern experimental measurements techniques should be able to provide significant progress in this area. In addition, with the advent of fully transient, three-dimensional CFD, the ability to study the flow features that drive turbine performance can prove invaluable. Also, there is really very little available research where the double-entry turbine has been fully characterized by measuring the full range of unequal admission inlet conditions. Pischinger and Wunsche [2] did provide some unequal performance data but with the primary intent to compare this performance with the twin-entry turbine. Mizumachi, et.al [10] simulated the partial admission flow conditions by physically blocking half of the entry to the nozzle vanes in a single entry volute. This resulted in some interesting pressure measurements around the circumference of the volute, but it is difficult to know whether a partially blocked single-entry turbine is very representative of the double-entry design. On the whole, all the published research in this area suggests that a significant drop in efficiency should result from unequal, steady flows. Therefore, there appears to be a clear need to study the steady-state characteristics of the double-entry turbine, especially focused on the effect of unequal admission in view of the unsteady, out-of-phase on-engine pulsations.

Table 2.1 also shows there has been some attempt both to model and test the double-entry geometry under pulsed flow. However, as outlined, the approach in these papers is limited since

instantaneous measurements could not be made. Dale and Watson [9] began measuring instantaneous mass flow, pressure, temperature and torque in a *twin-entry* turbine in 1986 which was a big step forward. The papers that followed measured the turbine performance under a range of steady, unequal conditions, as well as in-phase and out-of-phase pulsations. In general, this data cannot be directly applied to the current thesis due to the differences in the volute design. These differences will shape the way the turbine behaves under both steady and unsteady flow. However, there are a number of useful observations that come out of this work. First, the presence of a second entry means that there are many different combinations of inlet conditions that must be tested in steady-state to directly compare with the unsteady performance. The authors also found that for a twin-entry design, there is considerable interaction between entries, especially when the flows into each inlet are unequal. Furthermore, work by Costall [40] recommended that these unequal states must be accounted for if trying to model the on-engine pulsating operation due to the predominance of out-of-phase pulsations.

One question that is yet to be clearly answered in the available literature is whether or not the turbine behaves quasi-steadily, and if not, what drives the turbine to depart from steady operation. Earlier authors sought to compare unsteady and quasi-steady data based on *time-averaged* parameters. Most reported that the quasi-steady values underestimated the unsteady mass flow and power. A number of these authors felt that the amplitude of the pulse had the most significant influence on unsteadiness. However, this opinion was generally not taken up by the later authors as they began to measure all the instantaneous unsteady data. Instead, the Strouhal number was suggested by Szymko et.al [29] as one of the best means of assessing the departure from steady operation during a highly pulsating flow. These questions will be examined in this thesis.

In conclusion, in reviewing the available research, there appears to be a real opportunity to contribute to the scientific knowledge in this area by studying both the steady-state, unequal and unsteady flows in a double entry design. Using the unique test facility at Imperial College, there is an opportunity to produce fresh experimental results and provide a new insight into this unique design with the ultimate aim of improving the performance of the turbocharger.

Table 2.1: Literature Review Comparison

									
Author(s)	Year	Computational	Experimental	Twin entry	Double entry	Single entry	Partial Steady	Unequal Steady	Unsteady
Pischinger and Wunche [5]	1977								
Benson and Scrimshaw [11]	1965								
Benson [12]	1974								
Mizumachi et. al. [10]	1979								
Wallace and Blair [13]	1965								
Wallace, Adgey and Blair [14]	1969								
Wallace, Cave, Miles [15]	1969								
Wallace and Miles [16]	1970								
Copeland et.al [33]	2008								
Copeland et.al [34]	2009								
Dale and Watson [9]	1986								
Yeo (PhD Thesis) [17]	1990								
Yeo and Baines [18]	1990								

									
Author(s)	Year	Computational	Experimental	Twin entry	Double entry	Single entry	Partial entry	Unequal entry	Unsteady
Yeo and Baines [19]	1994								
Baines, Hajilouy and Yeo [24]	1994								
Ghassemi et.al [36]	2005								
Aghaali et.al [37]	2007								
Shahhosseini et.al [38]	2008								
Katrasnik et.al [39]	2007								
Costall et. al [40]	2009								
Kosuga et. al [20]	1976								
Capobianco and Gambaraotta [21]	1989								
Capobianco and Gambaraotta [22]	1990								
Capobianco and Gambaraotta [23]	1992								
Arcoumanis et.al [25]	1995								
Arcoumanis et.al [26]	1999								
Karamanis (PhD Thesis) [6]	2000								

									
Author(s)	Year	Computational	Experimental	Twin entry	Double entry	Single entry	Partial entry	Unequal entry	Unsteady
Karamanis et.al [27]	2001								
Karamanis et.al [28]	2002								
Szymko et.al [29]	2005								
Szymko (PhD Thesis) [8]	2006								
Rajoo et.al [30]	2006								
Rajoo et.al [31]	2007								
Rajoo, S. (PhD Thesis) [32]	2007								
Costall et. al [35]	2006								
Costall et. al [7]	2006								

Chapter 3

Experimental Procedure

3.1 Equivalent Conditions

3.1.1 Dimensionless Analysis

It is good practice in turbomachinery performance analysis to reduce the basic turbine measurements and fluid properties to a set of dimensionless parameters through the principle of dimensional analysis. This allows a comparison of the data from different turbochargers and inlet conditions. This is especially important in the laboratory testing since the inlet conditions must be set to match the driving non-dimensional parameters found in real on-engine conditions. The basic relevant parameters in a turbine are as follows, expressed in a functional relation:

$$f(d_2, N, \dot{m}, P_{01}, P_4, T_{01}, T_4, R, \gamma, \mu) = 0 \quad (3.1)$$

Where d_2 is the characteristic length which, for the rotor, is the mean inlet diameter, N is speed of rotation, $\dot{m}$ is the mass flow, P is the pressure, T is the temperature, R is the gas constant, γ is the specific heat ratio and μ is the dynamic viscosity. Note that the total state is used at the inlet to the turbocharger (subscript 1: P_{01} and T_{01}), but the static state is used at the exit (subscript 4: P_4 and T_4) since the energy in the exhaust gases after the turbocharger is generally not recovered. Since there are four fundamental units (mass, length, time and temperature), the Buckingham Pi theorem reduces equation 3.1 to six non-dimensional groups as follows:

$$f\left(\frac{P_{01}}{P_4}, \frac{T_{01}}{T_4}, \frac{\dot{m}\sqrt{RT_{01}}}{P_{01}d_2}, \frac{Nd_2}{\sqrt{RT_{01}}}, \frac{\dot{m}}{\mu \cdot d_2}, \gamma\right) = 0 \quad (3.2)$$

Since the total-to-static pressure ratio is a function of the temperature ratio and total-to-static efficiency η_{t-s} , the efficiency can replace the temperature ratio. The Reynolds number term $\dot{m}/\mu \cdot d_2$ is usually dropped since the flow is typically above the transitional value of Reynolds number (laminar to turbulent) and thus no longer plays a significant role in the overall performance [4]. Also, for a given working fluid, the ratio of specific heats γ and the ideal gas constant R are assumed

constant and are also dropped. Finally for a given rotor geometry, the rotor diameter is fixed so that this dependency can also be removed in equation 3.2. This means that the analysis is restricted to the current working fluid and turbine. The result is the following non-dimensional and pseudo non-dimensional groups that can be used to assess turbine performance:

$$f\left(\frac{P_{01}}{P_4}, \eta_{t-s}, \frac{\dot{m}\sqrt{T_{01}}}{P_{01}}, \frac{N}{\sqrt{T_{01}}}\right) = 0 \quad (3.3)$$

In the subsequent analysis, an additional group has been considered: the velocity ratio. This parameter does not contribute to the non-dimensional analysis directly as it is a function of the pressure ratio and speed parameter. This term is very useful in turbomachinery and it is critical in compressor-to-turbine matching considerations. It is defined as the ratio between blade tip speed U_3 and the isentropic or spouting velocity C_{is} . This velocity is defined as the velocity of the fluid that would be achieved if the flow expanded isentropically over the working pressure ratio. As explained in section 1.34, the velocity ratio is also related to the inlet flow angle and therefore is a useful parameter to plot against the efficiency. Thus, the performance parameters that will be used in this thesis are listed in table 3.1 below, given that we have removed the diameter and gas property variations the speed and mass flow parameter are now quasi-non-dimensional:

Table 3.1: Performance Parameters

NAME	FORMULA	UNITS
Velocity Ratio (VR)	$\frac{U_3}{C_{is}}$	Non-dimensional
Speed parameter (SP)	$\frac{N}{\sqrt{T_{01}}}$	$\frac{rev \cdot s^{-1}}{K^{1/2}}$
Pressure Ratio (PR)	$\frac{P_{01}}{P_4}$	Non-dimensional
Mass Parameter (MP)	$\frac{\dot{m}\sqrt{T_{01}}}{P_{01}}$	$\frac{kg}{s} \cdot \frac{K^{1/2}}{Pa}$
Total to Static Efficiency	$\eta_{t-s} = \frac{\dot{W}_{act}}{\dot{W}_{isen}}$	Non-dimensional

Note that the derivation of the performance parameters has been based on a single set of inlet conditions: $\dot{m}$, P_{01} , and T_{01} . For a single-entry turbine or a multiple-entry turbine operating with

equal admission conditions, there will not be a problem. However, for a turbine where there are two or more flows entering the turbine at different mass flows, temperatures or pressures, all the parameters in Table 3.1 must be re-evaluated to take into account the different inlet conditions. This will be discussed further in the forthcoming sections.

3.1.2 Equivalent Conditions

One of the most important aspects of the parameters listed in Table 3.1 is their use in scaling the hot, on-engine environment to a more manageable set of conditions conducive to gas stand testing. The test facility used in this thesis, first introduced by Dale and Watson [9] is a cold flow test facility where the air is typically heated to 323-343K to avoid condensation during expansion across the turbine. Thus, the range of appropriate speeds in the laboratory must be derived using the principle of similitude and the speed parameter in Table 3.1:

$$\left(\frac{N}{\sqrt{T_{01}}} \right)_{cold} = \left(\frac{N}{\sqrt{T_{01}}} \right)_{hot} \rightarrow N_{cold} = N_{hot} \cdot \frac{(\sqrt{T_{01}})_{cold}}{(\sqrt{T_{01}})_{hot}} \quad (3.4)$$

As an example, if a turbine has an on-engine design speed of 100,000 rpm at an exhaust temperature of 900K, the speed of the turbine for a cold flow test at 324K will be 60,000rpm. These calculations led to an eddy-current dynamometer designed by Szymko [8] with a maximum speed limit of 60,000 rpm which will be used in this thesis as the reference speed. In this way all other turbine speeds can be expressed as a percentage of the maximum speed for convenience (example: 50% speed = 30,000 rpm). Similarly, since the mass parameter is also temperature dependent, the mass flow rate in the laboratory will be appropriately scaled to represent on-engine conditions:

$$\left(\dot{m} \frac{\sqrt{T_{01}}}{P_{01}} \right)_{cold} = \left(\dot{m} \frac{\sqrt{T_{01}}}{P_{01}} \right)_{hot} \rightarrow \dot{m}_{cold} = \dot{m}_{hot} \cdot \frac{(\sqrt{T_{01}})_{hot}}{(\sqrt{T_{01}})_{cold}} \quad (3.5)$$

In this case, since the ratio of temperatures is an inverse square root function, the mass flow through the turbine must be set to a greater value on the gas test stand than would occur on the engine environment.

3.2 Test Facility

3.2.1 ABB Turbocharger Turbine

The turbocharger turbine under investigation in this research was supplied by ABB Turbo Systems as a consequence of a close collaboration. ABB specializes in diesel and gasoline engines rated above 500kW, thus making their production units beyond the capability of the Imperial College test facility. Therefore, to ensure that the turbine could be tested using the laboratory high speed dynamometer and match the mass flow capacity of the facility, the turbine scaled down. The resulting turbocharger turbine rotor, volute and nozzle ring are shown in Figure 3.1 and 3.2. These components are assembled together as shown in Figure 1.2.

The turbine is equipped with a double-entry, circumferentially divided volute where each engine exhaust feeds 180° of the mixed flow rotor inlet. The arrangement of the inlet dividing walls and their positioning with respect to the nozzle blades is clearly shown in Figure 1.6. Since the inlet plane is the same for both passages, the volute must be designed with different passage lengths. This is a unique characteristic inherent to the ABB double-entry design and must be taken into account when analyzing the unsteady flow since the travel time of a pressure wave will be different in each inlet. Both ducts in the volute are designed with an equal area-radius-ratio to produce a uniform rotor inflow. Figure 1.6 shows that there is no communication between the gas inlets in the volute until the flows reaches the nozzle –rotor interspace. Figure 1.9 illustrates that due to the 55.1° cone angle of the mixed-flow rotor, there will be a reasonably sized gap between the nozzle exit and rotor inlet. Although perhaps difficult to see from these figures, this inter-space spans the entire circumference of the turbine, thereby permitting a link between the set of cylinders that feed each inlet with exhaust gases. This will prove to be an important area which could cause mixing loss or even flow reversal via this interspace.

Table 3.2 lists the basic design features of the turbine and nozzle blades. The nozzle ring is equipped with 24 blades that are set at a constant vane angle of 70°. A locating pin ensures that two nozzle vanes are matched with the end of the dividing wall (tongue) to provide a smooth flow in this area. The mixed-flow rotor has 12 blades with a radially swept leading edge that helps to produce peak efficiencies at higher pressure ratio. Since it is a cold flow facility, the rotor was manufactured in aluminum. At the exit of the rotor, an exhaust diffuser is integrated into the volute design to recover some of the exhaust kinetic energy (Figures 1.9 and 1.10). In addition, a 425mm long exit duct of the same diameter as the volute exit was added to prevent immediate expansion to

atmosphere. In this way, the pressure ratio across the turbine will be more representative of the on-engine condition where the exhaust gases are ducted through an exhaust treatment system.

Table 3.2: Basic Turbine Design Features

ROTOR	DESIGN	Mixed-flow
	NUMBER OF BLADES	12
	INLET CONE ANGLE	55.1°
	EXIT CONE ANGLE	7.2°
	LEADING EDGE TIP DIAMETER	92.95 mm
	LEADING EDGE HUB DIAMETER	78.7 mm
	TRAILING EDGE TIP DIAMETER	81.5 mm
	TRAILING EDGE HUB DIAMETER	28.93 mm
NOZZLE	DESIGN	Fixed
	NUMBER OF BLADES	24
	NOZZLE VANE ANGLE	70°
	LEADING EDGE DIAMETER	116.406 mm
	TRAILING EDGE DIAMETER	103.718 mm
	THROAT DIAMETER	4.405 mm
	BLADE WIDTH	12.963 mm

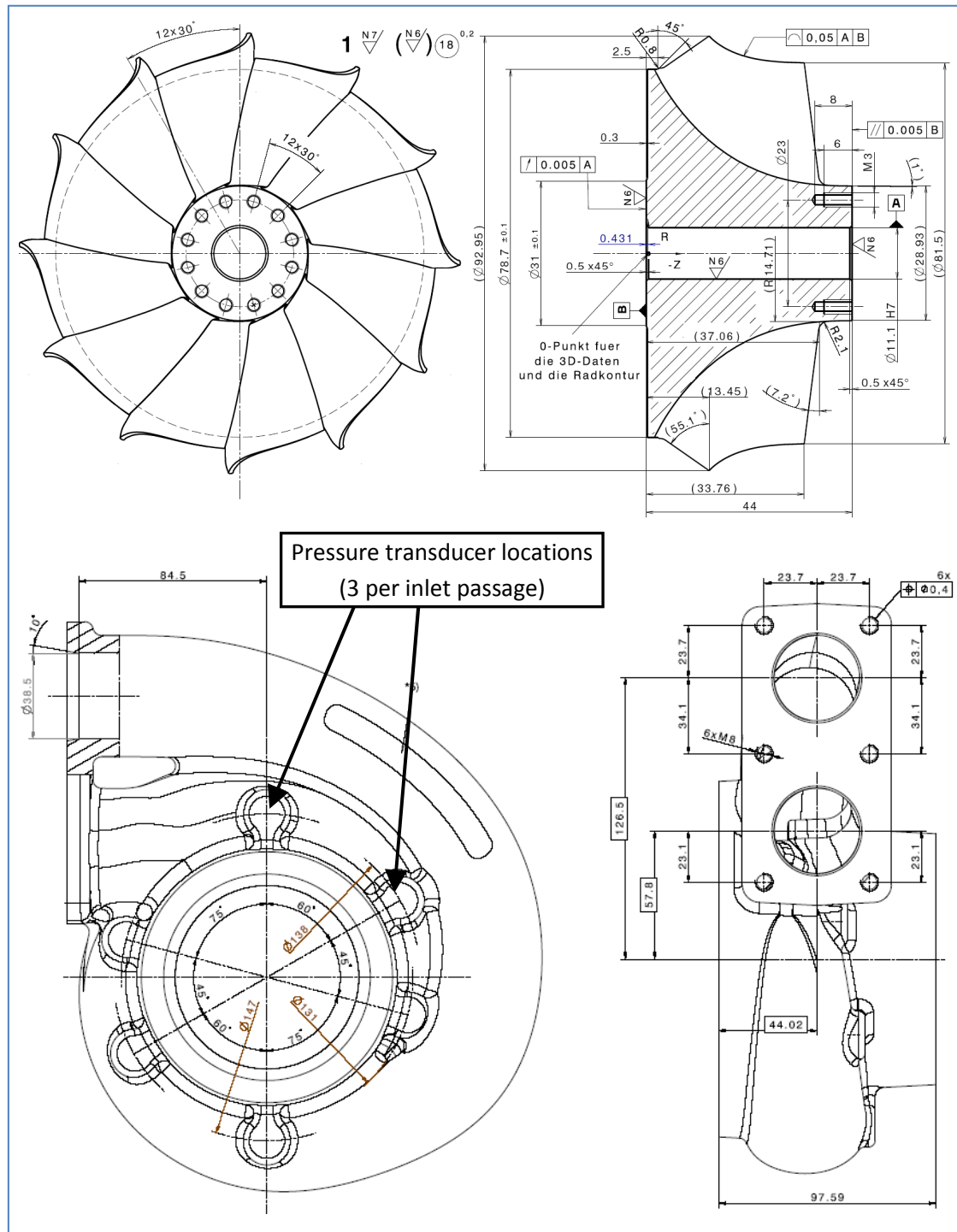


Figure 3.1: ABB double-entry turbocharger turbine volute and rotor wheel

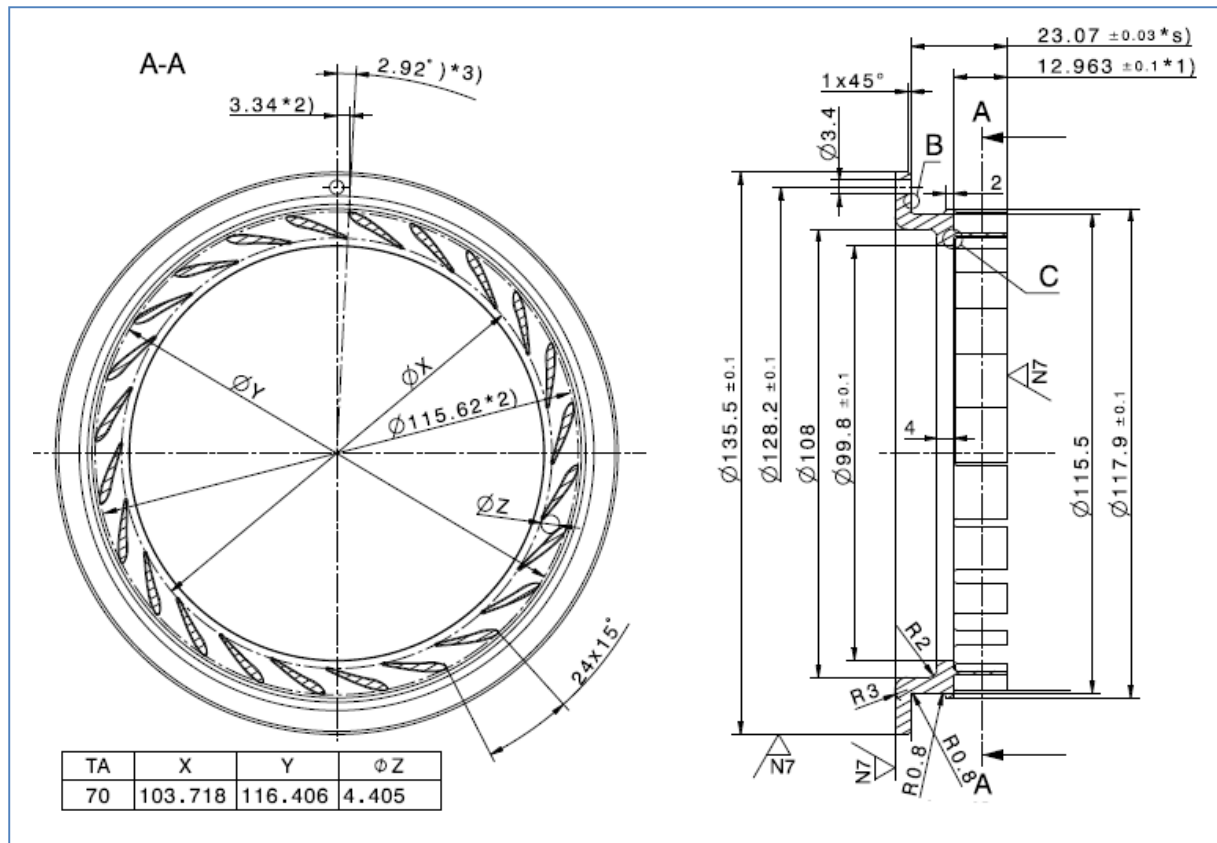


Figure 3.2: ABB double-entry turbocharger nozzle ring

3.2.2 Summary of Turbocharger Test Facility

The turbocharger turbine test facility located at Imperial College in London has been described by many other researchers [8,9,17,32] but has evolved over the years with continuous improvements and adaptations for each research application. The schematic of the turbocharger test facility is shown in Figure 3.3. The facility consists of an air supply and heating system, a pulse generator, and a high speed dynamometer, as well as a data acquisition and control equipment. One of the main advantages of the test facility is its versatility for testing under a wide variety of conditions. As shown in Figure 2, the main air compressors supply two independent ducts thus allowing testing of dual-entry turbines. The mass flow rate in each limb can be controlled independently through two valves, thus allowing full or partial admission testing under steady or unsteady conditions. To simulate a typical exhaust pulse from an internal combustion engine, a set of rotating plates periodically cut the flow, thus producing a pulsating flow of desired frequency. The

freedom to vary the flow conditions independently permits a more complete characterization of the turbine performance under a wide range of conditions.

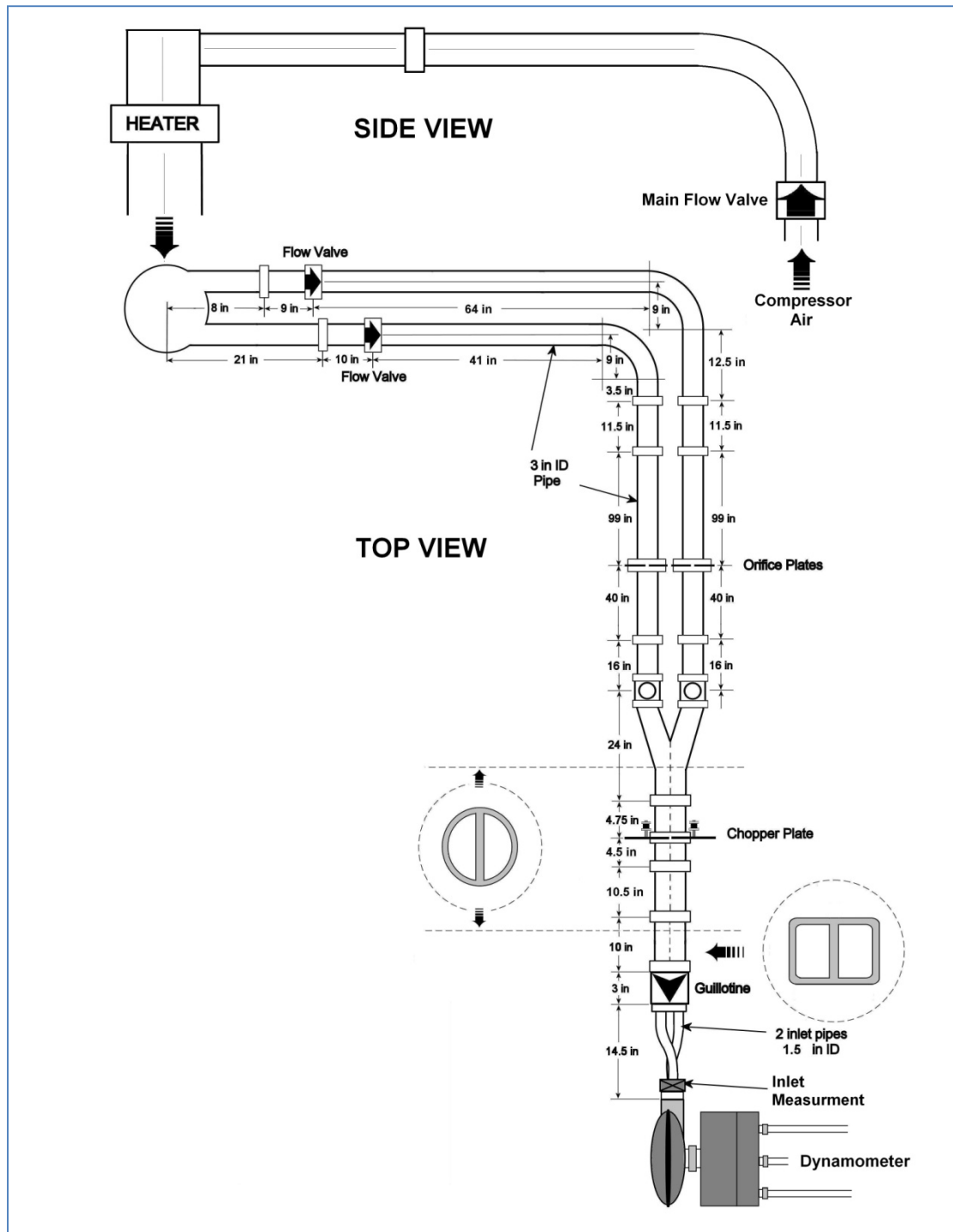


Figure 3.3: Schematic of test facility components

Although the air system also includes a heater, this is used to prevent condensation across the turbine, not to replicate the exhaust gas temperatures. This 'cold flow' allows the researcher to

measure instantaneous flow parameters that would be very difficult to measure in the harsh conditions of a high temperature exhaust flow. The turbine is coupled to a purposely built eddy-current dynamometer allowing testing of the turbine over a wide range of velocity ratios unavailable using standard compressor loading. Since it is very important to have a broad steady performance map in order to carry out an unsteady analysis, this ability is indispensable to the research here.

3.2.3 Air Supply, Heating System and Ductwork

Air from three Ingersoll-Rand air compressors enters the laboratory cell through a 4 inch diameter pipe. A combined mass flow rate of 1.2 kg/s can be supplied at room temperature and at a pressure of 5 bar absolute. A remotely controlled main valve adjusts the total air flow rate in the test facility. The metered air flow passes through a heater stack that is regulated by a PID controller capable of maintaining a constant, user-defined air temperature in the 320K~350K range. The heater construction is shown in Figure 3.4 and reveals the location of the stack of heating tubes that hang in-line with the 8 inch pipe-work. Since the heating elements are designed to operate in cross-flow, the inlet pipe is mounted off-centre to generate some inlet swirl, thereby improving the heat transfer. Downstream of the heater, the pipe work splits into two smaller 3 inch diameter *limbs* with a set of remotely controlled flow globe valves that adjust the balance of flow fed into each turbine entry. A set of 59.85mm diameter orifice plates are incorporated into each limb to measure the air mass flow rate according to the British Standard, BS 5167-1:1997 [41]. After passing through the pulse generating section, the two limbs combine into a single 4 inch diameter split duct which maintains a separate flow to the entry of the turbine.

In order to adapt the ABB dual entry configuration to the experimental rig, two problems had to be addressed. First, the direction of rotation of previous tests at Imperial College has typically been in the anti-clockwise direction with respect to the turbine exit. Since the ABB turbine spins in the opposite direction, the flow must be redirected to the top of the volute. Secondly, the radially divided volute as seen in Figure 3.1 means that the rig needed to be modified from the horizontally divided flow configuration to produce two entries in the vertical plane. This required a new set of pipes feeding the volute. Figure 3.5 shows the orientation of the conversion ducts that were designed to produce the appropriate flow into the ABB volute.

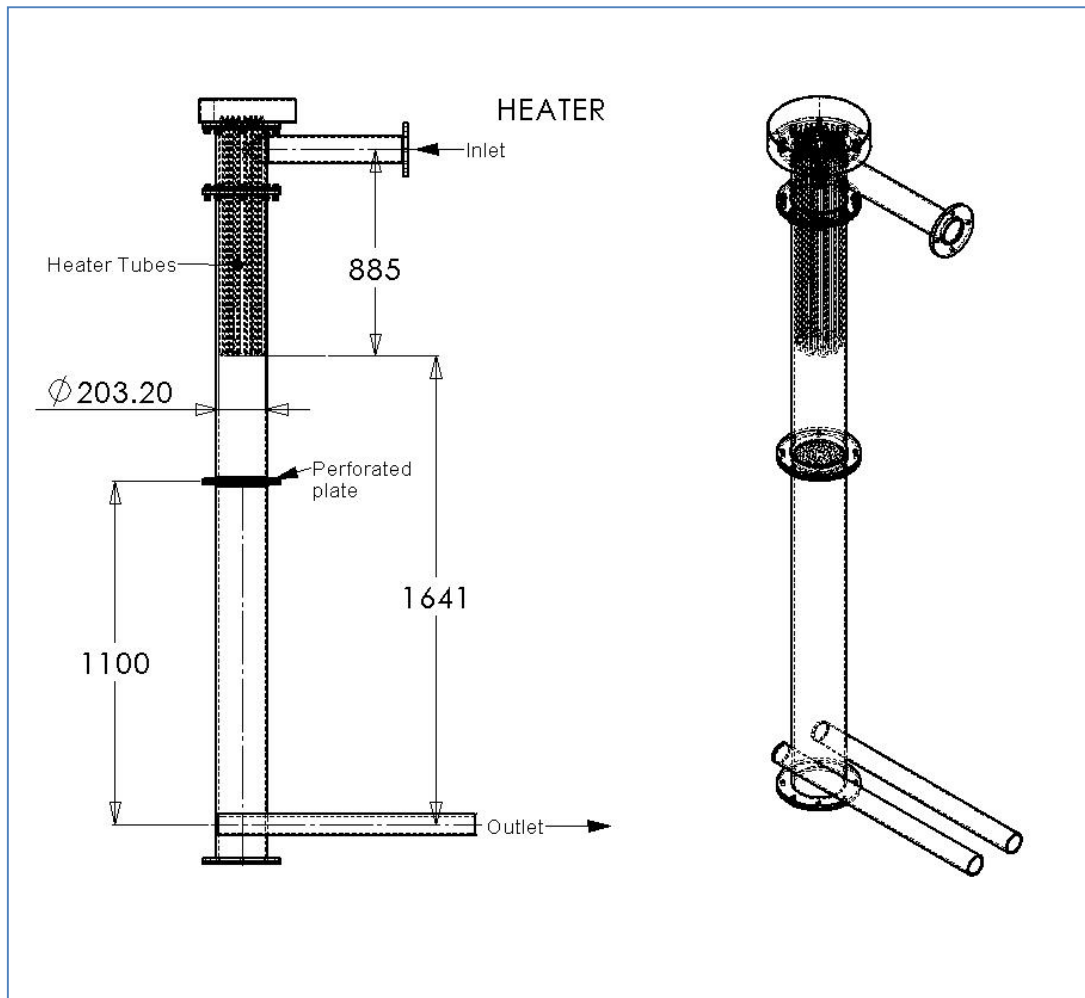


Figure 3.4: Heater Construction

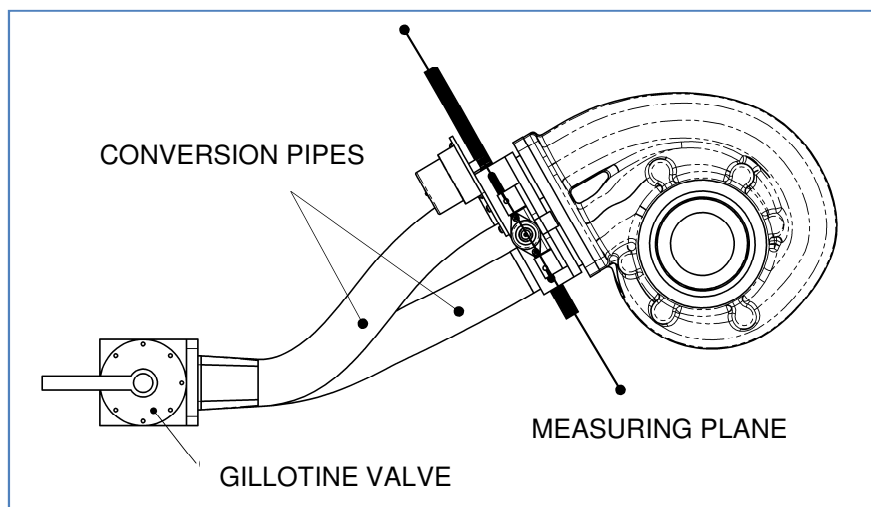


Figure 3.5: Configuration to adapt ABB turbine to existing rig

3.2.4 Pulse Generating Plates

One of the primary advantages of the test facility is the ability to study the effect of unsteady, pulsed flow on the performance of a turbocharger turbine. This pulse flow is created by a set of rotating chopper plates with a cut out portion covering 120° of rotation. A disassembled view is shown in Figure 3.6 below:

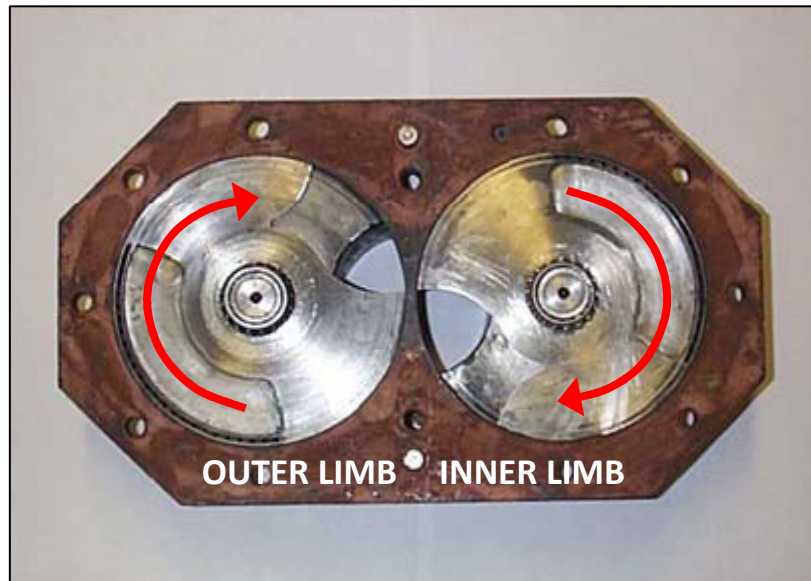


Figure 3.6: Disassembled view of rotating chopper plates

The profile of the opening in the chopper plates are designed to produce a linear opening and closing area as shown in Figure 3.7. Although the closed chopper plate is shown with a zero opening area in this plot, there is some inevitable leakage through small clearances when the chopper plate is closed when exposed to high upstream pressures. By pressurizing the upstream pipe work and suddenly closing the main valve, the leakage through a closed chopper plate was measured. The resulting data for three such tests are shown in Figure 3.8.

The plates are driven by a toothed drive-belt connected to a variable speed electrical DC motor. The operator can remotely control the speed of the motor to achieve a pulse frequency between 0 Hz and 85 Hz. The position of the plates with respect to each other can be adjusted by repositioning the belt drive to obtain different phase angles between the two pulses. A digital magnetic pickup connected to the belt idler pulley provides chopper plate speed information and a trigger signal for unsteady data acquisition.

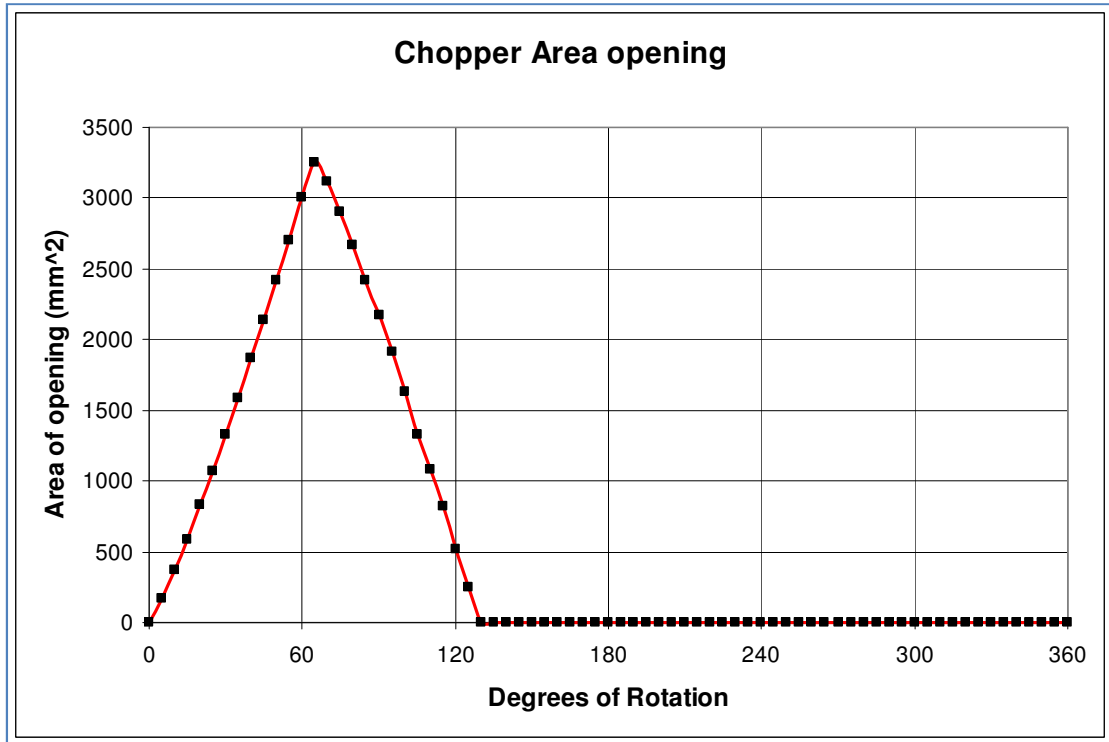


Figure 3.7: Chopper Plate Opening Area versus Rotational Angle

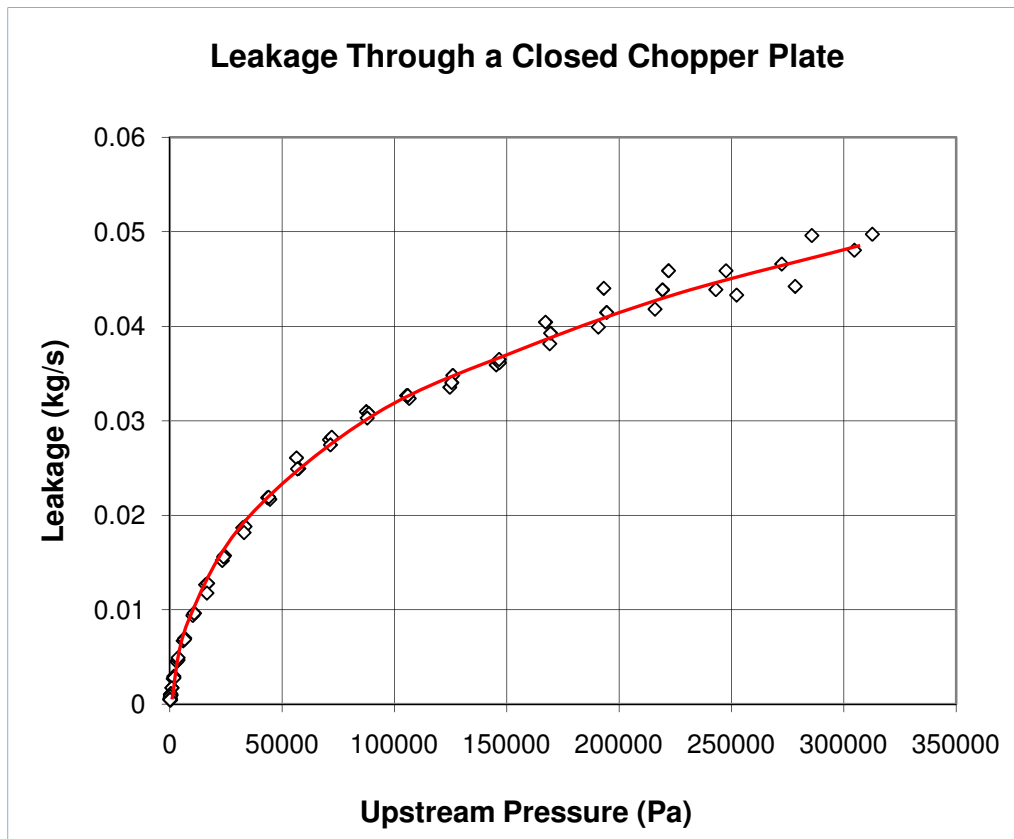


Figure 3.8: Chopper plate leakage characteristics

3.2.5 Dynamometer

A detailed sectioned view of the dynamometer assembly that was designed and built by Shinri Szymko [8] is shown in Figure 3.10 with a single-entry volute attached. The main shaft is supported by three roller bearings and connects the magnetic rotor inside the dynamometer to the turbine rotor within the volute. The entire dynamometer housing is suspended on a set of gimble bearings which allow the assembly to react against a load cell to measure the torque produced by the spinning turbine. The level of power absorption is remotely adjusted by two stepper motors that adjust the gap between the stator plates and magnetic rotor. Figure 3.9 shows a CAD drawing of the stator plates and magnetic rotor. The range of air gaps between stator and magnetic rotor is limited between 0.8mm and 10mm. With a small gap, the eddy currents induced in the stator plates generate a significant amount of heat and therefore requires high velocity cooling water flow to prevent overheating and eventual damage. At the largest gap, the eddy-current load is minimal, thereby leaving the friction in the shaft bearings as the only load on the rotor. The air gap is calibrated each time the dynamometer is set up for a series of tests. This is an automated procedure which uses four limit switches to provide the maximum and minimum distances in travel from which the stator gaps can be referenced.

In order to install the ABB turbine onto the dynamometer, most of the main components need to be disassembled. The rotor is then installed on the main shaft, and the entire unit consisting of the shaft, turbine rotor, bearings and magnetic rotor are balanced together to keep the vibration to a minimum during operation. Following balancing and cleaning, the entire dynamometer is reassembled.

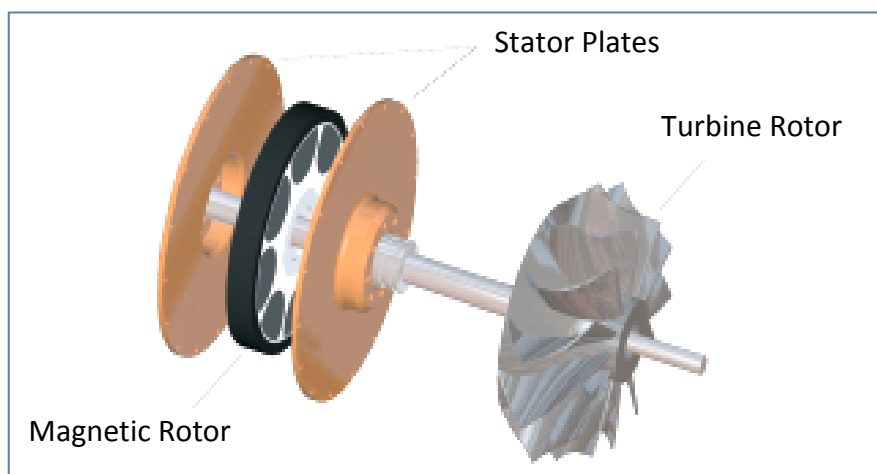


Figure 3.9: CAD model of the magnetic model and stator plates [8]

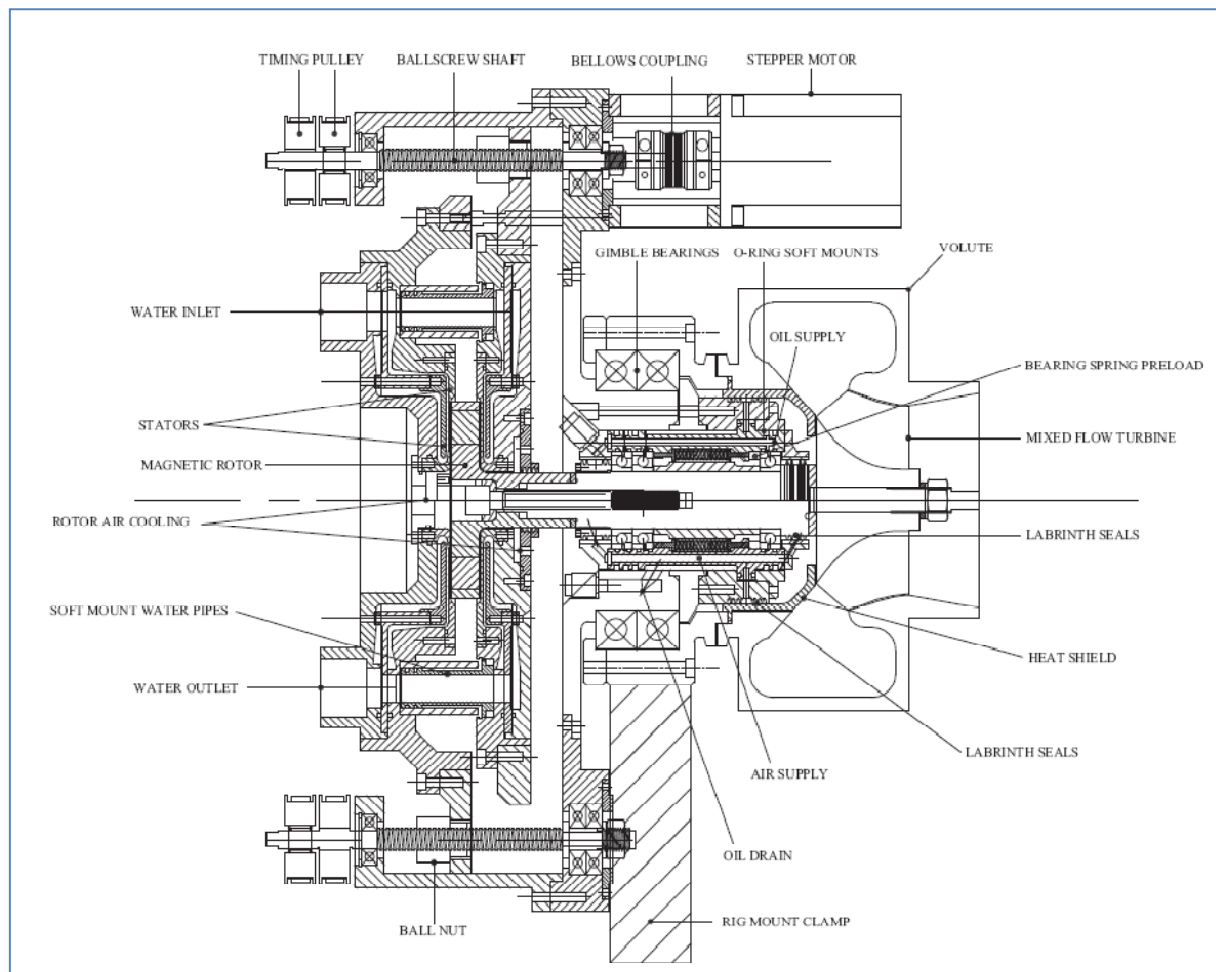


Figure 3.10: Sectioned view of the eddy current dynamometer [8]

3.2.6 Measurement Instrumentation

To perform comprehensive turbocharger performance tests with both steady and unsteady flow, there are many different quantities which must be measured accurately. A short outline of the main instruments will be provided here with a more in-depth explanation in sections 3.3 and 3.4. Table 3.3 provides a comprehensive list of the data collected for steady and unsteady testing. All steady flow pressure tapings are fed into a rotary Scanivalve which switches from one channel to another remotely, recording pressure measurements from the selected channel with a high or low pressure transducer. All temperatures are measured using an array of thermocouples that records time-averaged values. The steady mass flow rate through each limb is calculated from the pressure drop across a calibrated orifice plate. Average torque is measured from a load cell that measures the reaction of the dynamometer to the spinning turbine rotor. In addition to these, there are a number of safety and monitoring quantities that are collected for each test.

Table 3.3: Test facility measured quantities and instrumentation

Measuring Location	Physical Property	Instrumentation	
		Steady-flow	Pulsating-flow
Electric Heater	Temperature	E-type and K-type Thermocouples	
Orifice Plates on each limb	Upstream Static Pressure	Scanivalve	
	Pressure difference	Scanivalve	
	Upstream Temperature	E-type	
Pulse Generator	Rotational Speed		Shaft encoder
Turbine Inlet	Static Pressure	Scanivalve	
	Temperature	2 T-type thermocouples	
	Instantaneous Static Pressure		Strain gauge pressure transducers
	Instantaneous mass flow		CTA probe
Turbine volute	Instantaneous Static Pressure		Strain gauge pressure transducers
Turbine exit	Static Pressure	Scanivalve	Strain gauge pressure transducer(s)
Dynamometer	Rotational speed	Optical sensor	
	Torque	Load cell	Load cell + Optical sensor (acceleration)
	Stator and Bearing Temperature	K-type Thermocouples	
	Water Temperature	T-type Thermocouple	
	Vibration	Velometer	
Atmosphere	Temperature	Wall Thermometer	
	Pressure	Mercury Barometer	

Measuring the unsteady flow quantities adds an extra level of complexity as they must be time-resolved. Mass flow is measured using a pair of CTA (constant temperature anemometer) sensors at the volute entry. Pressure is measured using high-response strain gauge pressure transducers. Temperature is derived from the average temperature and instantaneous pressure

assuming isentropic compression/expansion. Instantaneous torque is calculated from the average torque plus a fluctuating torque derived from acceleration. Lastly, pulse frequency is recorded by a shaft encoder on the chopper plate pulley.

3.2.7 Control System and Safety

Apart from the cooling water flow and lubrication oil flow, the entire test facility is computer controlled from outside the test cell. Rig control is all done using Labview, thereby providing the operator with full control and monitoring capabilities during operation. The desired testing condition is set by the operator via computer as follows:

- The dynamometer air gap is set using the two stepper motors pictured in figure 3.10.
- For unsteady flow, the speed of the DC motor is set to give the desired pulse frequency.
- The main flow valve shown in figure 3.3 is adjusted to provide an overall flow rate of air entering the turbocharger turbine.
- The two limb valves are then adjusted to set the balance of flow rates between each entry.
- The temperature of the air is set and maintained via the PID controlled heater bank.

Since the facility is designed to test high-speed turbomachinery, a sophisticated safety feedback system must be included to protect the operator and prevent equipment damage in the event of a failure. The centre piece of the safety system is the guillotine valve. During operation this valve is held open by an electromagnet working against a heavy-duty coil spring. Table 3.4 lists the safety limits of the various parameters monitored by the main Labview program that result in a rig shutdown. If these limits are reached during operation, the guillotine electro-magnet is released and the air flow to the turbine is shut off in a fraction of a second. In addition, the flow valves are closed, the heaters disabled and the chopper plates shut down automatically. Rig shutdown can also be initiated remotely by the operator using a button on the Labview control panel or a physical stop button next to the computer terminal. In addition to these programmed safety limits, the vibration signal is fed into an FFT analyser for constant monitoring to insure that safe limits are not exceeded. This is especially important during unequal admission where axial loads on the turbine rotor wheel are prominent.

Table 3.4: Safety Limits

Parameter	Safety Limits		Action
	Min	Max	
Turbine Speed (<i>rps</i>)	-	1,050	Rig Shut-down
Turbine Bearing Temperatures (<i>K</i>)	-	373	Rig Shut-down
Magnetic Rotor Temperatures (<i>K</i>)	-	323	Rig Shut-down
Vibration Level (<i>mm/s</i>)	-	1.5	Rig Shut-down
Cooling Water Flow Rate (<i>L/min</i>)	150	-	Rig Shut-down
Bearing Oil Flow Rate (<i>L/hr</i>)	11	-	Rig Shut-down
Air Mass Flow Rate (<i>kg/s</i>)	0.05	-	Heater disabled
Heater Air Temperature (<i>K</i>)	-	373	Heater disabled

3.3 Steady Flow Measurements

3.3.1 Mass Flow

The mass flow rate through each limb is measured using two sharp edged orifice plates as shown in Figure 3.3. In order to calculate mass flow in accordance to British Standards BS 5167-1:1997 [41], the density of the fluid must first be calculated from upstream temperature and pressure. The pressure drop across the orifice is also measured from pressure tappings $D/2$ upstream and D downstream of the orifice, where D is the duct diameter. The mass flow can then be calculated from the expansion factor ϵ , the diameter ratio $\beta = d/D$ and the discharge coefficient C_d . Since the discharge coefficient is a function of the Reynolds number and therefore mass flow, a solution to equation 3.6 must be found by iteration.

$$\dot{m} = \frac{C_d}{\sqrt{1-\beta^4}} \epsilon \frac{\pi}{4} d^2 \sqrt{2\Delta p_{orif} \rho_{orif}} \quad (3.6)$$

Although the air supply system is well sealed from the compressor to the turbine entry, there will be a certain amount of air leakage from the orifice plate measurement. A new set of conversion ducts were installed and a new hotwire traversing mechanism constructed for the ABB turbine, thus making it was important to quantify this leakage for the current setup. A leakage test was therefore performed by disconnecting the volute and sealing the exit with a flat plate. The main valve was opened to pressurize the system to approximately 3 to 3.5 bar before quickly closing the air supply. By measuring the loss of air density (from pressure and temperature measurements) in the pipework with time, the leakage mass flow rate could be measured as a function of pressure. The results for each limb are given in Figures 3.11 and 3.12. A third order polynomial fit does well in predicting the behaviour of the mass leakage with changes in pressure. Thus, for a given steady state pressure entering the turbine, the leakage could be calculated and subtracted from the mass flow measured at the orifice plate.

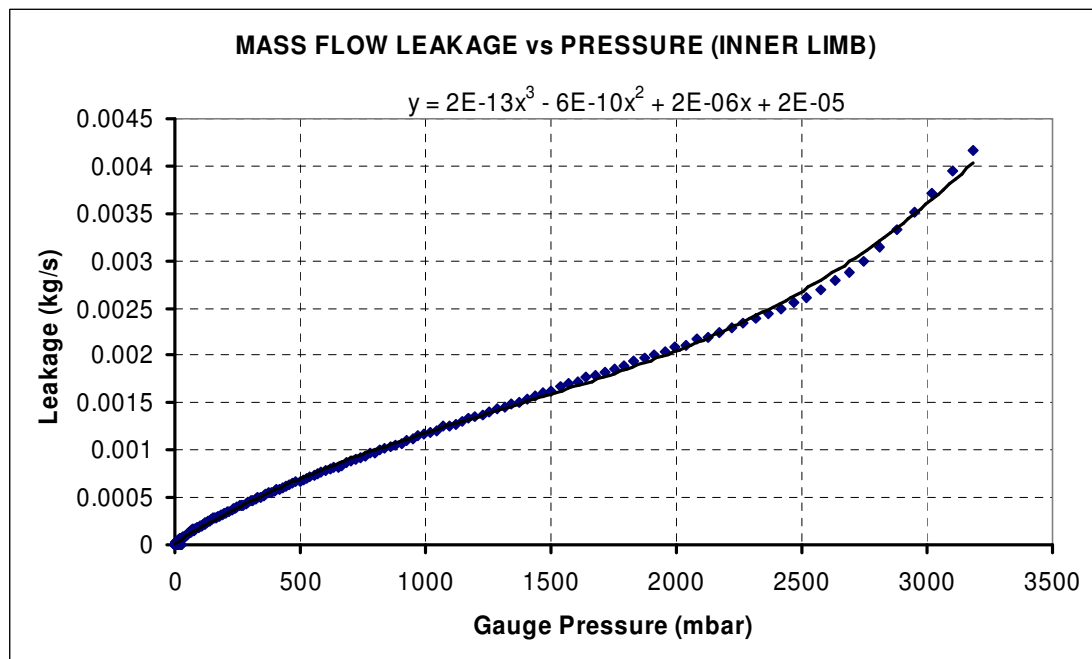


Figure 3.11: Inner limb air leakage with pressure

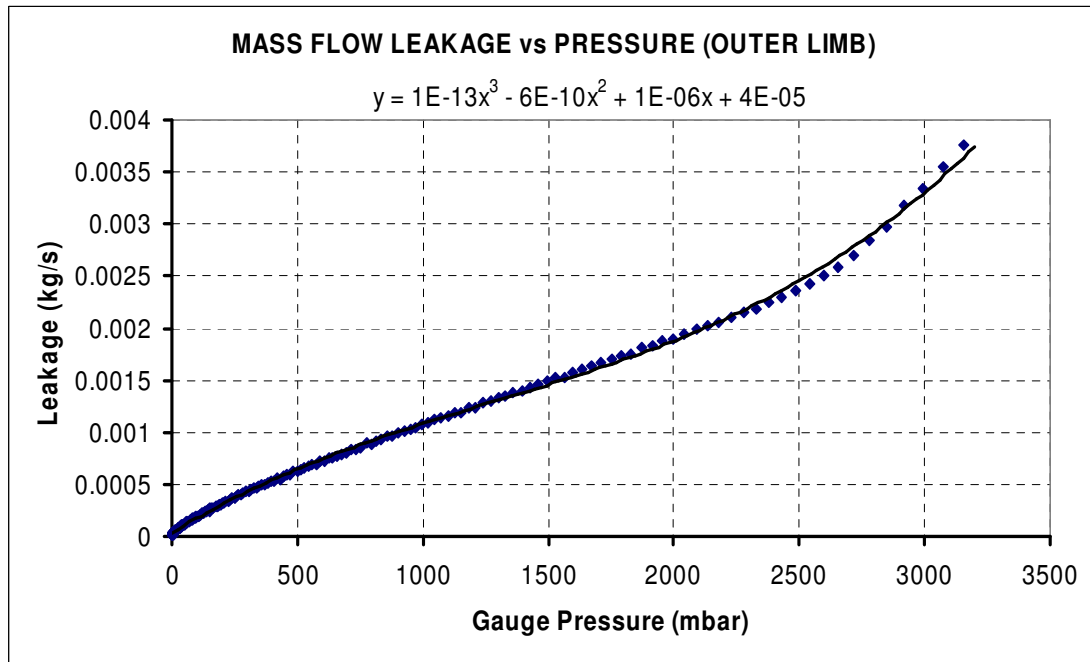


Figure 3.12: Outer limb air leakage with pressure

3.3.2 Pressure

For steady flow, static pressure tapings in the wall of the pipe work are connected to a 24 channel Scanivalve system through pneumatic tubing. The Scanivalve system is remotely controlled using Labview through a *National Instruments FieldPoint* digital output module. A rotary switch allows the operator to select between a high or low pressure transducer and connect it to any one of the measurement channels. The high pressure transducer (*Druck PDCR 23D*) has a range of ± 3.5 bar, and the low pressure transducer (*Druck PDCR 22*) has a range of ± 0.35 bar. The calibration of these transducers was re-checked for the measurements in this thesis using a Druck portable transducer calibration system designed especially for this purpose. The output of the transducers is connected to a *Fylde FE-492-BBS Mini-Bal* and *Fylde FE-351-UA Uni-Amp* to condition and amplify the signal. A total of seven channels were used in the steady flow regime: two to measure the pressure drop across each orifice plate, two to measure static pressure upstream of the orifice plates, two tapings at the turbine inlet, and one measurement on the exit duct. Acquisition of the pressure data from the Fyldes occurred via an analogue channel on the *FieldPoint* module that was connected to the computer terminal through an Ethernet connection. The Labview program was designed such that after steady-state operation was reached, a logging algorithm was activated that automatically cycled through the various pressure measurements that were necessary for the performance calculation.

3.3.3 Temperature

The temperature measurements for steady flow are all measured using thermocouple probes at various positions as specified in Table 3.3. These thermocouples were calibrated at three temperatures: the freezing point of water (273 K), atmospheric pressure corrected boiling point of water (373 K), and lastly at room temperature (measured by a thermometer). Repeatability of the calibration was less than $\pm 4\text{K}$ [8].

The relationship between static (T_s) and total (stagnation) temperature (T_0) can be expressed in terms of the Mach number (M) and the specific heat ratio (γ) according to the isentropic relation expressed in equation 3.7.

$$\frac{T_0}{T_s} = 1 + \frac{\gamma - 1}{2} M^2 \quad (3.7)$$

The measured temperature (T_m) in a flowing fluid will be subject to a competing effect of heating due to the stagnation effect and heat transfer from the probe to the flowing fluid. Therefore, the measured temperature from a probe in a high velocity flow will fall in between the static and stagnation temperatures. A recovery factor (r) can be introduced into equation 3.7 to account for this effect as follows:

$$\frac{T_s}{T_m} = 1 + r \left(\frac{\gamma - 1}{2} \right) M^2 \quad (3.8)$$

The recovery factor depends on the probe and flow properties and is calculated for each probe from equation 3.9. A special calibration test section in the lab is used for this purpose and yields a recovery factor which is a weak function of the Mach number.

$$r = \frac{T_m - T_s}{T_0 - T_s} \quad (3.9)$$

The four temperatures important for steady performance evaluation are two located upstream at the orifice plates and two mounted on the inlet flange of the volute. Orifice plate temperatures are used to calculate the density, and ultimately the mass flow, of the air through the orifice plates. For these and the other thermocouples upstream, the Mach number is low enough that a recovery factor correction was not necessary. However, for the turbine inlet thermocouples, since the Mach number at turbine inlet can rise above 0.3, the effects of compressibility and heat transfer cannot be neglected and thus require a recovery factor correction.

3.3.4 Rotational Speed

Turbine speed is measured from an infra-red optical sensor triggered by a 10 toothed wheel integrated into the magnetic rotor in the eddy current dynamometer. The sensor itself is of type *Omron EE-SX4101* with integrated amplifier. The circuitry used for steady-state testing reduces the 10 pulses per revolution to a single pulse which is then used as a digital gate for a 16MHz clock in a 16-bit counter. In this way, the time for one revolution is measured and converted into a voltage that can be related to turbine speed. This voltage signal is fed into the National Instruments Fieldpoint module responsible for analogue input and recorded by the Labview program. Shinri Szymko [8], provides a detailed outline of the calibration of the speed sensor using a 5kHz square-wave signal generator. No changes have since been made to this sensor, making it unnecessary to recalibrate. However, the accuracy of the calibration was easily verified by comparing the speed signal displayed in the Labview program with the first harmonic of the turbine vibration signal displayed by the FFT analyser (Section 3.3.7).

3.3.5 Torque

The method used in the Imperial facility to measure power is unique for turbocharger test rigs. Traditionally, a compressor was used as the loading device, and thus power was measured indirectly by thermodynamic analysis. However, compressors suffer from margins of surge and choke, thus limiting the range of test conditions that is available to the researcher. The eddy-current dynamometer developed at Imperial College does not have these aerodynamic limitations, and the power can be measured directly. The main dynamometer shaft connects the spinning turbine rotor to the magnetic rotor which induces a rotational torque on the stationary stator plates. The stator plates are bolted to the body of the dynamometer which is supported on a set of gimble bearings and is therefore free to react onto a load cell as shown in Figure 3.13. Any bearing friction in the dynamometer will add to the loading on the shaft, thereby making it a true turbine torque measurement. The arm shown in Figure 3.13 preloads the load cell and eliminates backlash in the load cell link. Since the ABB turbocharger rotates in an opposite direction to most other turbochargers, it was necessary to reposition the loading arm and modify the load cell link to accept tensile loading. This arm was also used to calibrate the sensor by adding a series of weights at a fixed distance along the arm and recording the voltage. An example of this calibration is shown in Figure 3.14 where the data is plotted to obtain the slope of the line. This slope, along with the offset at the time of the experiment is used to obtain the torque and thus the power absorbed at a given speed.

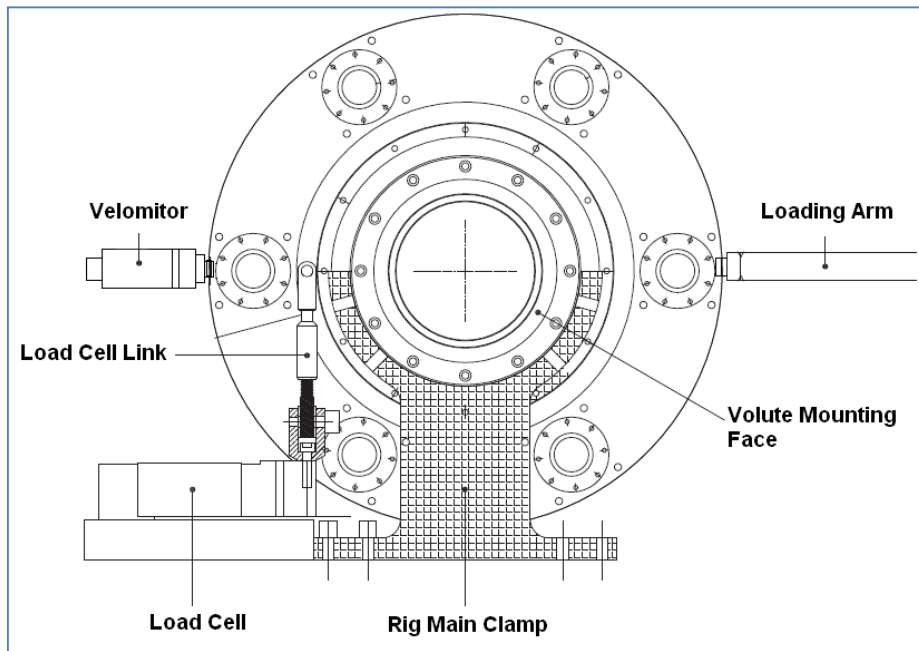


Figure 3.13: Main body and load cell arrangement [8]

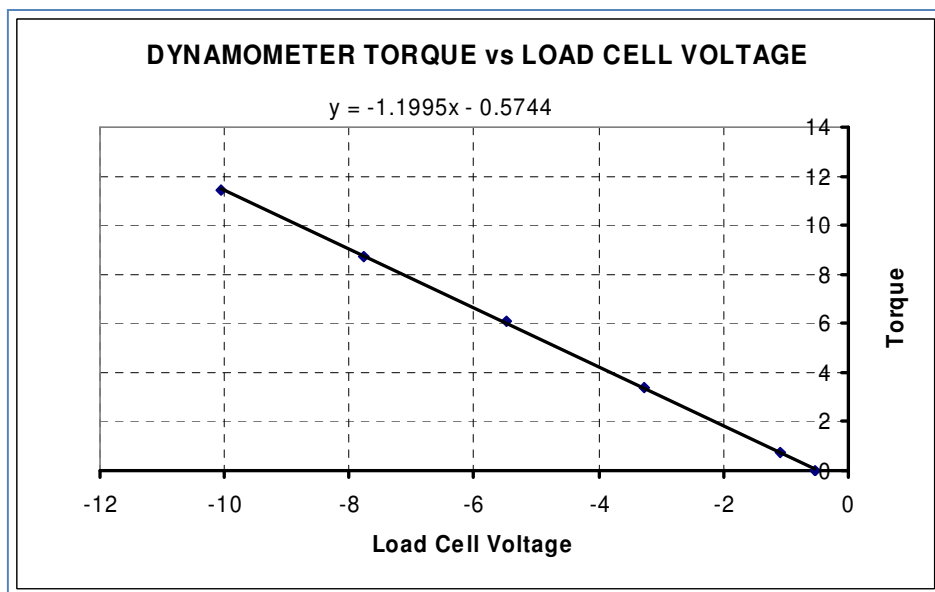


Figure 3.14: Load cell calibration curve

3.3.6 Vibration

The vibration is monitored by a *Bently-Nevada Velomitor* transducer mounted on the body of the dynamometer. The sensor contains a piezoelectric accelerometer with integrated electronics to convert the signal to a velocity output. The radial mounting shown in Figure 3.13 provides a signal which is sensitive to radial vibrations from the rotating assembly. The sensor has a wide operating

range capable of measuring frequencies from 4.5 Hz to 5 kHz. The RMS of the signal is integrated into the main Labview monitoring program and used as a safety limit as outlined in section 3.2.7. The velometer signal was also wired to a DI-2200 Real Time Fast Fourier Transform (FFT) Analyser which processes the frequency domain of the signal, thus providing the operator with the magnitude of the signal at each discrete frequency. This is especially important in unsteady flow operation since the magnitude of the vibration from the turbine rotation must be distinguished from that of the chopper plate which operates at a much lower frequency.

3.4 Unsteady Flow Measurements

3.4.1 Introduction

Unsteady flow measurements are more challenging than their steady-state counterparts due to the time resolution required. The instruments measuring unsteady flow must accurately resolve any small scale fluctuations in pressure, mass flow, and torque in a minimum pulse period of 0.012 seconds (85Hz). Therefore, with the exception of rotor speed, all the unsteady measurements are sampled at 20kHz via a high speed data acquisition card. This results in a minimum of 235 data points per pulse, sufficient to capture small scale signal variations. In addition, a total of 50 pulse cycles are recorded in order to further increase the accuracy of the measurement through ensemble averaging. The sampling rate of the rotor speed sensor will be constrained by the number of speed sensor shutters per revolution. The signal from the speed sensor was therefore post-processed and re-sampling at 20kHz to maintain consistency with the other measurements. These post-processing methods are outlined in greater depth in Section 3.5.

Figures 3.15 and 3.16 show the ABB turbine mounted on the eddy current dynamometer and connected to the conversion ducts. A series of instruments are mounted on these ducts as close as possible to the inlet to the volute. This was important in order that the unsteady response of the turbine could be considered in isolation from the volume of air contained in the pipework. This should become clearer in later discussions, especially in comparison to the work of Rajoo [32] and Syzmko [8] where the unsteady measurements were made a distance upstream of the turbine inlet. It was also important to measure the unsteady pressure and mass flow at the same stream-wise location to prevent any phase difference between the two measurements. The thermocouples that provide turbine inlet temperature are also shown, but the location is not as critical since they are not dynamic measurements. Pressure transducers that are omitted here are discussed in Section 3.4.4.

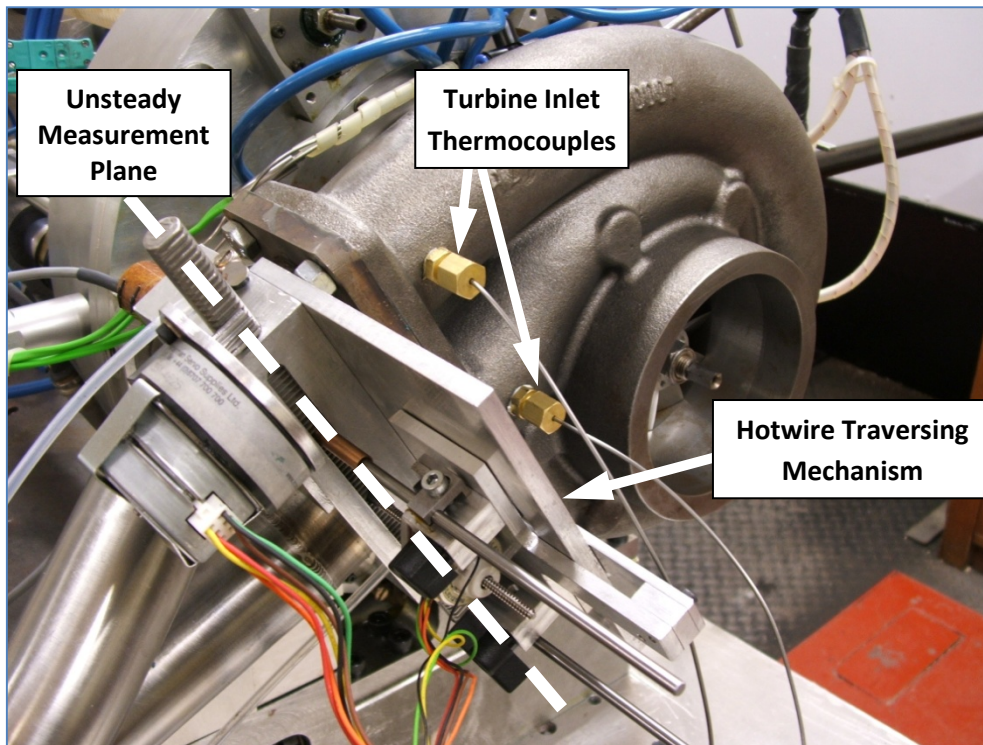


Figure 3.15: ABB turbine installed showing location of unsteady measuring plane

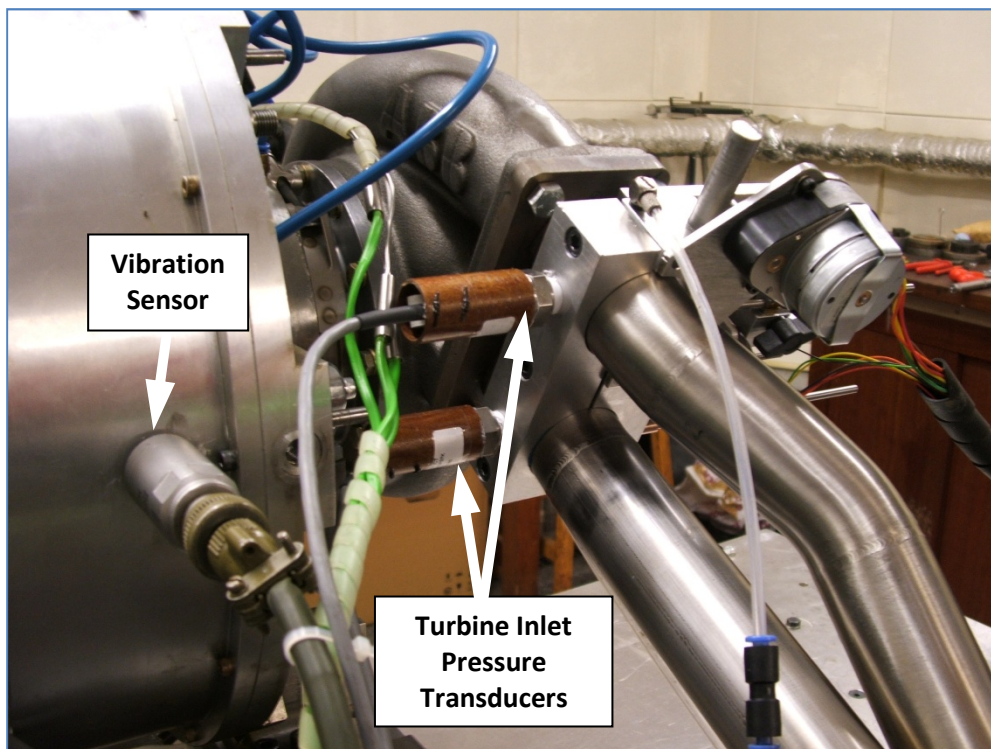


Figure 3.16: ABB turbine installed showing location of inlet pressure transducers

3.4.2 Mass Flow

Apart from expensive laser methods, there are very few velocity measurement techniques that have the temporal resolution to capture a highly pulsating flow. One technique that is able to provide both spatial and temporal resolution is Constant Temperature Anemometry. This approach has been widely used to study such challenging areas of experimental research as turbulence, boundary layers, fluid structures, etc. As the name implies, constant temperature anemometry works by maintaining the temperature of a heated wire while a fluid is flowing past it. The amperage that is necessary to maintain the wire temperature can then be related to convective cooling and hence, the velocity. A very small diameter wire is used where a good temporal resolution is required, and a small length is used where spatial resolution is important.

One of the more important aspects of the research presented in this thesis is the ability to measure time-resolved mass flow entering the turbocharger. This ability means that the instantaneous energy of the flow at the turbine inlet can be measured which, as outlined in Chapter 2, is a great advantage over other research on similar turbines. However, the use of a hotwire probe for this application does require careful calibration due to the large variation in flow conditions being measured. Hotwire probes have been used for unsteady flow measurements at Imperial College since Dale and Watson, but the technique has been further refined by both Szymko [8] and Rajoo [32]. Therefore, the hotwire technique presented in this thesis will largely follow the approach of these authors with the exception of a few differences attributable to changes in the experimental setup. Figures 3.17 show the new hotwire probe design used in the current work. Two 10 μ m tungsten hotwire probes are angled 90° to permit easy near-wall access. Each one is supported by a 235 mm straight probe support shown in Figure 3.18.

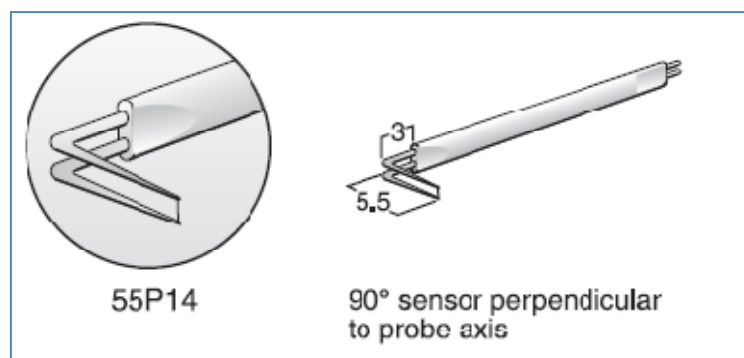


Figure 3.17: Model 55P14, 90°angled hotwire probe

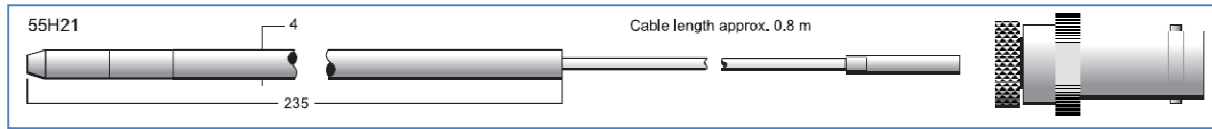


Figure 3.18: Model 55H21, straight, single sensor probe support

The new inlet pipes shown in Figure 3.5 required a new hotwire traversing mechanism to position two hotwire probes in each duct remotely. The design of the hot wire traversing mechanism was based on the following requirements:

1. The position of two hot wire probes needed to be on a plane parallel to the volute entry and in the same location in each duct to facilitate data logging of the fluctuating mass flow at the same point before entry into the volute.
2. The hotwire probes must be traversed from point to point to log the velocity profile in two axes according to BS-1042 [42].
3. Stepper motors must be used to remotely traverse the probes within the duct from outside the test cell for safety reasons.
4. A circumferential limit switch is needed to prevent the probes from colliding with the pipe wall.

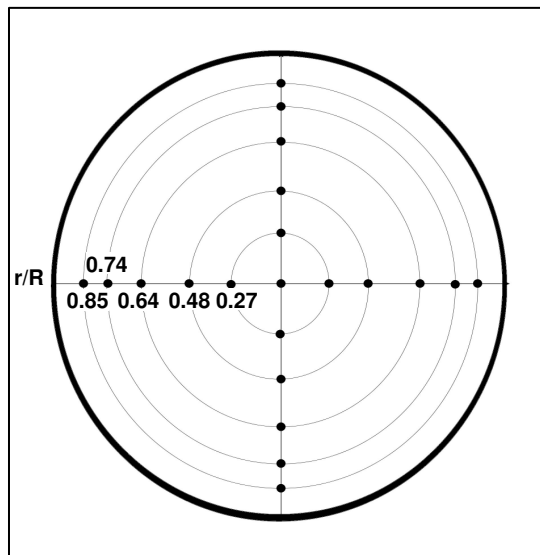


Figure 3.19: CTA measuring points across the entry duct profile

The British Standards guidelines BS-1042 [42] for fluid flow measurement in a duct state that for circular cross-sections, measuring points are located at the intersection of a number of circumferences concentric with the pipe axis and at least two mutually perpendicular diameters. The traversing routine for the current application has been programmed with 21 points across two axes in each duct as shown in Figure 3.19. The resulting design incorporates a rack and pinion arrangement with two stepper motors to provide two-axis movement. The movement of the probes is coordinated in such a way as to ensure that each measurement position is matched. A contact sensor is mounted into a circular cutout which corresponds to the limits of travel within the duct. This prevents damage to the delicate $10\mu\text{m}$ CTA wire. The design is shown in Figure 3.20.

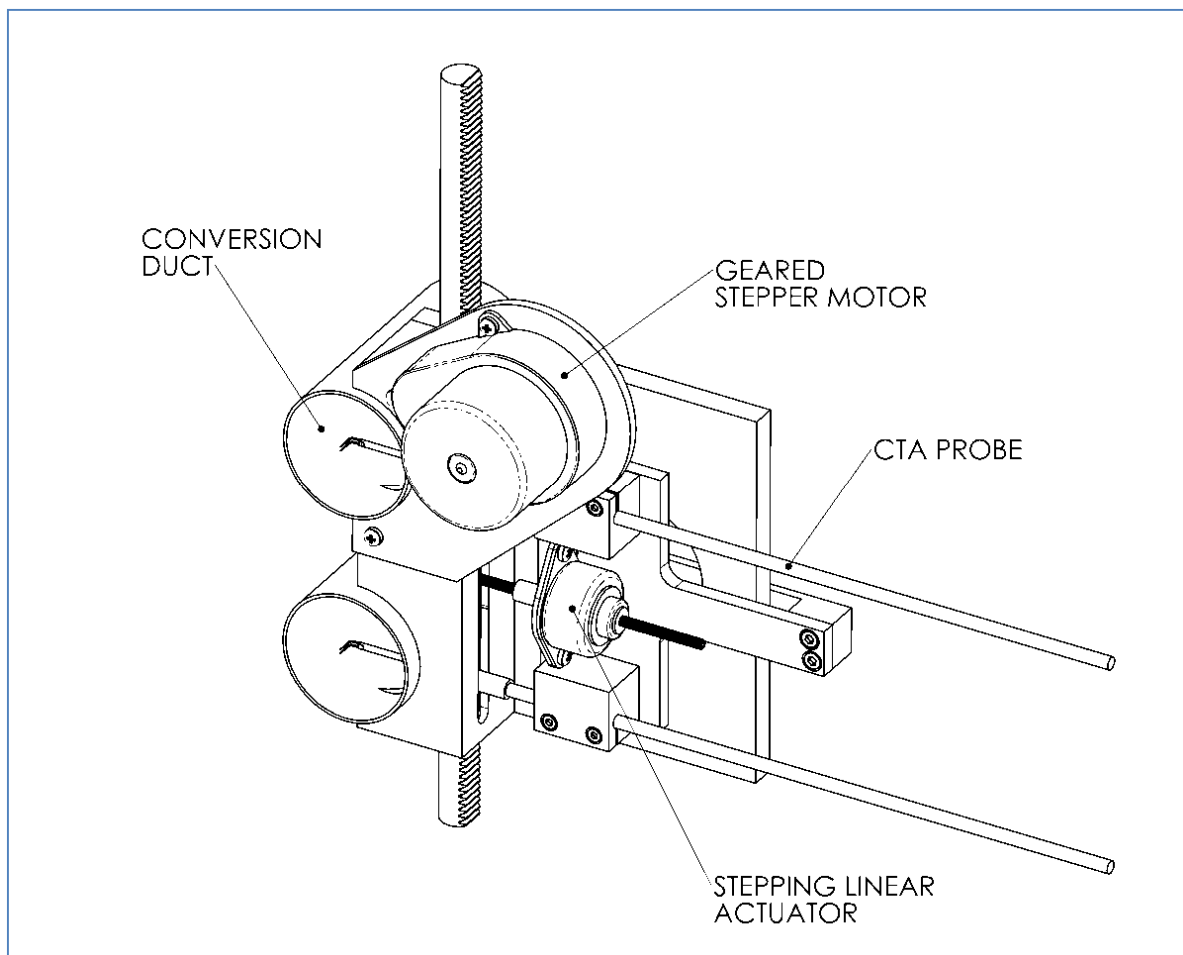


Figure 3.20: Constant temperature anemometer (CTA) probe traversing mechanism

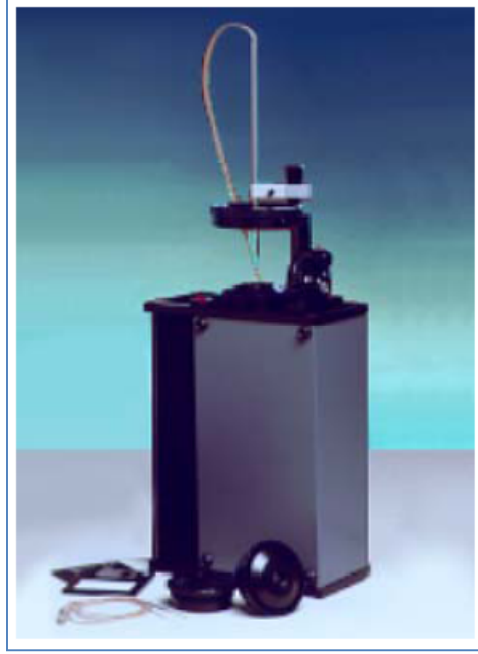


Figure 3.21: StreamLine hotwire calibration

The hotwires were connected to a Dantec Dynamics StreamLine CTA system which contains a Wheatstone bridge circuit for each hotwire channel. The hotwire is maintained at a constant temperature by keeping the resistance across the bridge constant despite any change in heat transfer that results from a change in velocity. To do this, the system must vary the current through the sensor to keep the bridge in balance. The resulting voltage drop E across the bridge circuitry will therefore be a function of the velocity of the flowing fluid. In addition, the low thermal inertia of the $10\mu\text{m}$ diameter CTA wire and the high gain servo loop amplifier in the StreamWare system provides excellent dynamic response to fluctuations in the fluid velocity. This dynamic response is especially important for measuring the pulsating flow in this thesis. Assuming constant fluid properties, the square of the voltage will be related to the velocity of the fluid through a power law equation commonly referred to as Kings Law [8]:

$$E^2 = A_{calib} + B_{calib} (U)^n \quad (3.10)$$

Where E is the CTA voltage, A_{calib} , B_{calib} and n are power law coefficients and U is the velocity of the flow. A calibration curve for the hotwire probes was generated using a Dantec StreamLine calibration unit shown in Figure 3.21. It automatically meters velocities from 5 to 300 m/s through a calibrated orifice. By positioning the hotwire into the flow with velocity U and recording the voltage E , a power law curve will result with the form given by equation 3.10. Figure 3.22 plots the results of the calibration test, and Table 3.5 gives the power law constants resulting from the curve fit.

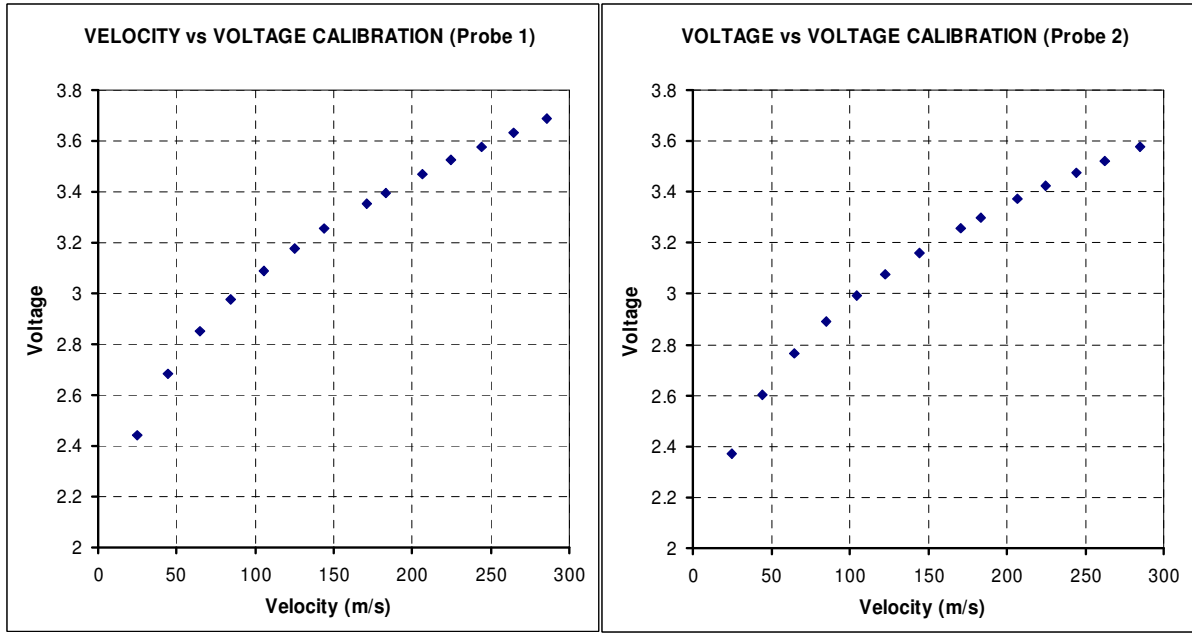


Figure 3.22 CTA probe calibration

Table 3.5: Power law fit for CTA probe calibration

Probe 1 Power law fit		Probe 2 Power law fit	
<i>A</i>	1.24439	<i>A</i>	1.244113
<i>B</i>	1.331774	<i>B</i>	1.222942
<i>n</i>	0.392953	<i>n</i>	0.396453

Unfortunately, due to the highly pulsating nature of the flow being measured, the calibration of the hotwire becomes more complicated. The basic principle of the hotwire relies on the heat transfer from a fixed body (wire) to calculate the velocity of the fluid flowing across it. Thus, the heat transfer will not only be a function of the velocity but also of the temperature and density of the flowing fluid, both of which can vary widely over a pulse cycle. The calibration coefficients obtained from the Dantec calibration unit will be limited to the ambient conditions under which they were obtained. In order to account for changes in heat transfer due to fluid property changes, the generally accepted approach is to correct these coefficients A_{calib} and B_{calib} according to the ratio of fluid properties evaluated at the reference calibration temperature T_{calib} and the measured fluid temperature T_{meas} . The technical note supplied by Dantec Dynamics [43] recommends using

equations 3.11 and 3.12 to obtain corrected coefficients A_{corr} and B_{corr} based on the fluid and wire properties k , Pr , ρ and μ . In order to better represent the conditions immediately around the wire, these properties are evaluated at a *mean film temperature* defined by equations 3.13 and 3.14. These two equations simply average the temperature of the fluid T_{meas} and T_{calib} with the temperature of the wire T_w .

$$A_{corr} = A_{calib} \left(\frac{T_w - T_{meas}}{T_w - T_{calib}} \right)^{(1 \pm m)} \left(\frac{k_{f,meas}}{k_{f,calib}} \right) \cdot \left(\frac{Pr_{f,meas}}{Pr_{f,calib}} \right)^{0.2} \quad (3.11)$$

$$B_{corr} = B_{calib} \left(\frac{T_w - T_{meas}}{T_w - T_{calib}} \right)^{(1 \pm m)} \left(\frac{k_{f,meas}}{k_{f,calib}} \right) \cdot \left(\frac{Pr_{f,meas}}{Pr_{f,calib}} \right)^{0.33} \left(\frac{\rho_{f,meas}}{\rho_{f,calib}} \right)^n \left(\frac{\mu_{f,meas}}{\mu_{f,calib}} \right)^{-n} \quad (3.12)$$

$$T_{f,meas} = \frac{(T_w - T_{meas})}{2} \quad (3.13)$$

$$T_{f,calib} = \frac{(T_w - T_{calib})}{2} \quad (3.14)$$

$$Pr_f = \frac{\mu_f c_{p_f}}{k_f} \quad (3.15)$$

Once a corrected velocity is obtained, it is multiplied by the local film density to give the mass flux u^* . To obtain total mass flow in each duct according to the British Standard BS-1042 [42], the 21 points shown in Figure 3.15 must be integrated and multiplied by the duct area. The data around the circumference of each circle concentric with the axis of the duct are averaged. The mean values $u^*_1, u^*_2 \dots u^*_n$ along the circumferences with increasing relative radii $r'_1, r'_2 \dots r'_n = r_i/R$ are then used to calculate the total discharge mass flux U^* by interpolating between measuring points using the equation 3.16.

$$\begin{aligned}
U^* = & u_0^* \left[-\frac{1}{12} r_2'^2 + \frac{5}{12} r_1'^2 + \frac{1}{12} \frac{r_1'^3}{r_2'} \right] + u_1^* \left[\frac{1}{6} r_1'^2 + \frac{2}{3} r_2'^2 - \frac{1}{12} r_3'^2 \right] - u_2^* \left[\frac{r_1'^3}{12 r_2'} \right] \\
& + u_2^* \left[-\frac{1}{12} r_4'^2 + \frac{2}{3} r_3'^2 - \frac{2}{3} r_1'^2 \right] + u_3^* \left[-\frac{1}{12} r_5'^2 + \frac{2}{3} r_4'^2 - \frac{2}{3} r_2'^2 + \frac{1}{12} r_1'^2 \right] \\
& + u_4^* \left[\frac{1}{2} r_5'^2 + \frac{1}{12} r_4'^2 - \frac{2}{3} r_3'^2 + \frac{1}{12} r_1'^2 \right] \\
& + u_5^* \left[\frac{n}{n+1} (1 - r_5'^2) + \frac{(r_5'^2 - r_4'^2)^2}{12n(1 - r_5'^2)} + \frac{7}{12} r_5'^2 - \frac{2}{3} r_4'^2 + \frac{1}{12} r_3'^2 \right]
\end{aligned} \tag{3.16}$$

where u_0^* is the velocity at the centre of the pipe, R is the radius of the pipe, and n is the coefficient of wall roughness.

The final step in the calibration process is to obtain a value of the temperature loading factor m in equations 3.11 and 3.12 with an in-situ hotwire calibration. Following the initial ambient calibration using the Dantec unit, the hotwires were reinstalled in the turbine pipework and a series of steady flows introduced. The value of the temperature loading factor was selected to provide the best possible agreement between the integrated hotwire measurement and the orifice plate. The result of a series of these tests is shown in Figure 3.23. These in-situ calibration tests were checked on a daily basis to insure that any small changes in wire resistance due to contaminants in the flow were accounted for in the calibration.

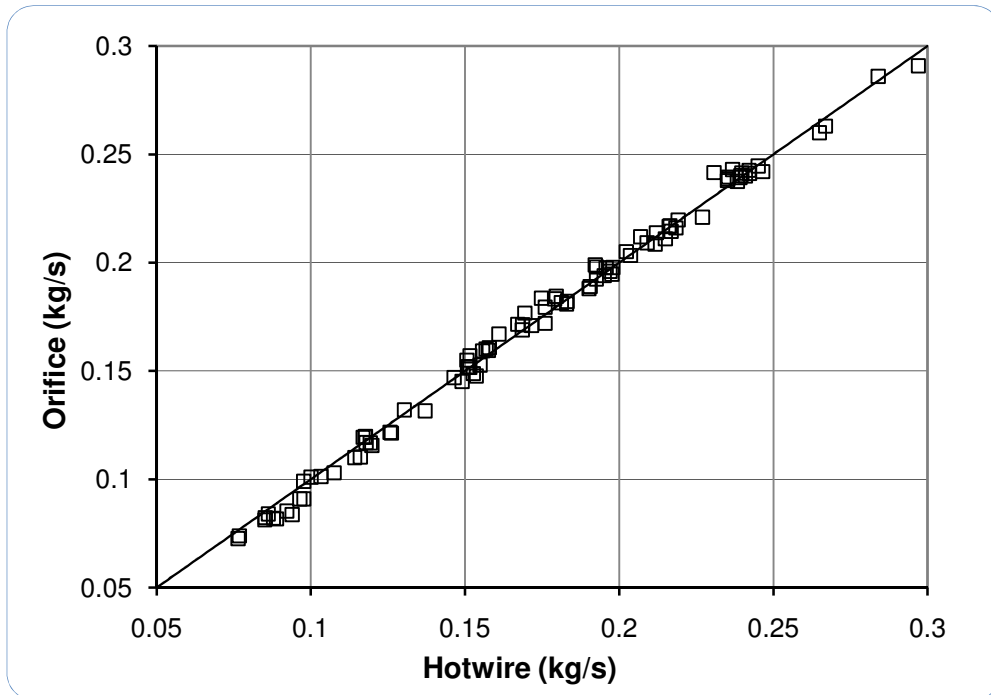


Figure 3.23: Hotwire versus orifice plate mass flow

3.4.3 Temperature

All the thermocouples listed in table 3.3 for steady-state measurements are also used to provide average temperatures during unsteady flow. However, owing to their high thermal mass, they do not have the temporal resolution to provide instantaneous temperatures. Therefore, the fluctuating temperature of a pulsating flow must be deduced using the instantaneous pressure P_{inst} and the average pressure $\bar{P}$ and temperature $\bar{T}$ measured at turbine inlet using equation 3.17. This approach assumes that the rise and fall of temperature over a pulse cycle is due to the isentropic expansion and compression of the air.

$$T_{inst} \approx \bar{T} \left(\frac{P_{inst}}{\bar{P}} \right)^{\frac{\gamma-1}{\gamma}} \quad (3.17)$$

The validity of this assumption was examined by Szymko [8] by measuring the instantaneous temperature using a dual hotwire probe and comparing it to values obtained from equation 3.17. An example result from his analysis in Figure 3.24 demonstrated that the calculated temperature agreed very well with the measured values. This work therefore demonstrated that the isentropic compression assumption replicated the amplitude and main features of the instantaneous temperature trace and therefore can be considered reasonable in the absence of instantaneous data. On the strength of this work, the instantaneous temperatures in this thesis will be calculated using equation 3.17.

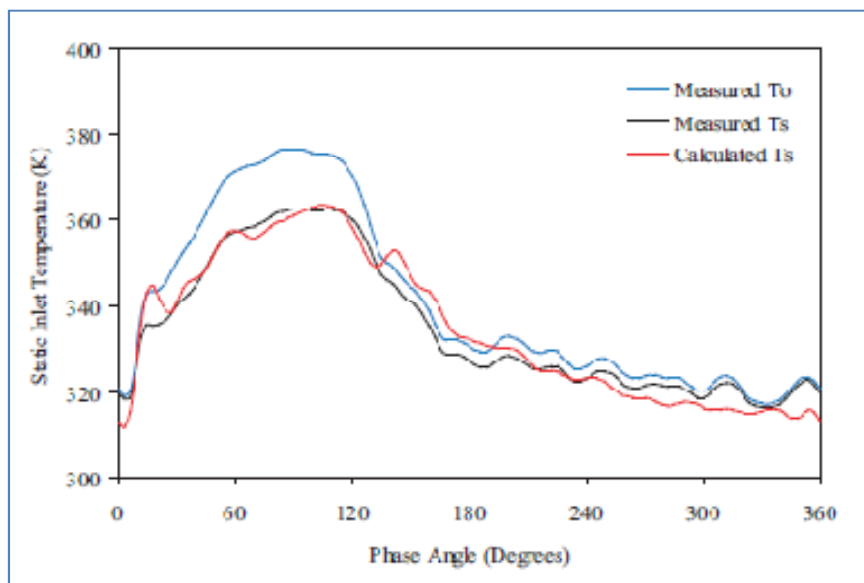


Figure 3.24: Measured versus calculated instantaneous temperature [8]

3.4.4 Pressure

Inlet pressures are measured by two strain gauge, temperature compensated, *SenSym 19C Series* pressure transducers mounted on the pipes prior to volute entry. The response time of 0.1 ms was ample to temporally resolve the pressure pulse. Also, allowances were made by ABB to install an additional six pressure transducers around the circumference of the volute as shown in Figure 3.1. Once installed, these pressure transducers provided a series of direct pressure measurements prior to the entry to the nozzle ring. Finally, two additional pressure transducers were placed on the exit duct 30 mm and 210 mm from the exit of the volute. Thus, a total of 10 pressure transducers were installed as listed in Table 3.6. Note from this table that some *Schaevitz type P704-0001* strain gauge pressure transducers were also used, but their use was limited to three locations due to their large and bulky construction.

Table 3.6: Pressure transducer measurements

Location	Number	Type
Volute Inlet (measuring plane)	2	SenSym 19C
Upper Volute Passage	3	SenSym 19C Schaevitz P704-0001
Lower Volute Passage	3	SenSym 19C Schaevitz P704-0001
Exit Duct	2	SenSym 19C Schaevitz P704-0001

Each transducer was connected to a *Fylde FE-492-BBS Mini-Bal* conditioner and *Fylde FE-351-UA Uni-Amp* amplifier. The output of the conditioner-amplifier modules was connected directly to the high-speed National Instruments analogue PCI card *NI 6034E*. The Labview program used for unsteady data logging began recording the conditioned signal after receiving a trigger pulse from the encoder signal on the chopper plate pulley. This trigger signal synchronized all the pressure signals to ensure an identical time period for comparability between all the unsteady measurements. Prior to installation, all of the strain gauge pressure transducers were calibrated using a *Druck DPI 610* portable pressure calibration unit to apply pressures from 0 to 3 bar and record the voltage. Figure

3.25 shows an example pair of calibration plots from this calibration unit. Similar to the load cell calibration, these curves provide the slope of the sensor response needed for post-processing along with a value of the offset (atmospheric pressure) taken prior to each test.

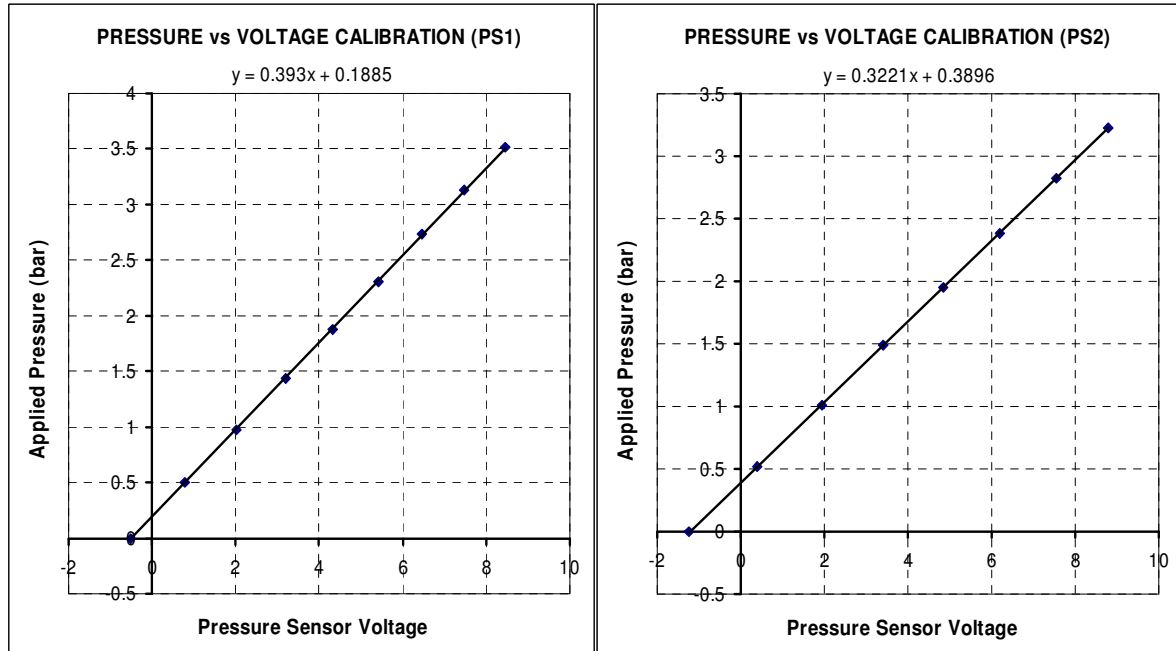


Figure 3.25: Pressure transducer calibration

3.4.5 Torque

Since this test facility is used primarily to study unsteadiness in turbocharger turbines, it is important to be able to accurately measure power fluctuations which results from pulsed flow. Since the load cell only provides an average torque measurement, the fluctuating component of torque must be derived from the variation in speed. Thus, the instantaneous torque τ_{inst} is first defined as the addition of a mean component $\bar{\tau}$ and fluctuating torque τ_{fluct} :

$$\tau_{inst} = \tau_{fluct} + \bar{\tau} \quad (3.18)$$

The fluctuating torque component is defined as the product of the rotor's angular acceleration α and its polar moment of inertia I , as given by equation 3.19. The derivative of the velocity signal $d\omega/dt$ is calculated using the first central difference technique.

$$\tau_{fluct} = I \cdot \alpha = I \cdot \left(\frac{d\omega}{dt} \right) \quad (3.19)$$

The polar moment of inertia of the ABB turbine rotor was measured using the *tri-filar suspension technique* where the rotor is mounted on a plate suspended by three strings. By timing the oscillations of the suspended mass, the rotor polar moment of inertia was calculated to be $1.456 \times 10^{-4} \text{ kg m}^2$. This value was also confirmed by the polar moment inertia calculated from a SolidWorks CAD model of the rotor which gave a value of $1.469 \times 10^{-4} \text{ kg m}^2$. The polar moment of inertia of the rotor was then added to the value for the shaft, bearings and magnetic rotor given by Rajoo [32]: $3.5772 \times 10^{-4} \text{ kg m}^2$. Thus, the total value of I used in equation 3.19 was $5.033 \times 10^{-4} \text{ kg m}^2$.

3.4.6 Turbine Speed

Since for unsteady flow, the variation of turbine speed over the pulse cycle is an important measurement for the torque calculation, precise speed sensor data was essential. Therefore, a full 10 pulses per revolution from the *Omron EE-SX4101* infra-red optical sensor are fed directly into a National Instruments PCI counter card, NI 6602. Similar to the steady-state approach, the pulses from the sensor are used as a gate for a 20MHz clock in the counter card. Thus, the time taken to rotate between each shutter can be recorded and converted into a speed signal. A trigger pulse from the chopper plate synchronizes the speed sensor acquisition with the other unsteady measurements. Unfortunately, the shutters on the rotating magnetic rotor were not machined to a sufficiently high standard to insure that an identical rotational angle corresponds to each optical sensor pulse. Therefore, Shrinri Syzmko [8] devised a simple method to account for any small inconsistencies in shutter-gap width. Since the 10-toothed sensor wheel will produce an identical repeating pattern for each rotation, the angle of the individual segments ϕ_i is evaluated using the following equation:

$$\phi_i = \frac{\omega_i}{\bar{\omega}} \cdot \frac{2\pi}{10} \quad i = 1, 2, \dots, 10 \quad (3.15)$$

where the ratio $\omega_i/\bar{\omega}$ is the ratio of the angular velocity of an individual shutter spacing to the average. Once the angle of the individual segments was calculated and divided by the time from the counter, the instantaneous speed results. This technique also has the advantage of correcting minor cyclic movement in the turbine shaft that may introduce further inaccuracies in the speed

measurement. The error associated with the speed measurement has been estimated to be ± 0.16 revolutions per second [8].

3.4.7 Pulse Generator Speed

The speed of the chopper plate must also be measured in order to determine the frequency of the pulses that are being produced. A shaft encoder: *Kubler Type 3720* was used for this purpose. It has the secondary function of providing a reference pulse that triggers the data acquisition of the unsteady hotwire (mass flow), pressure, and speed measurements. Thus, the encoder produces a single trigger pulse per revolution followed by 4096 pulses for the speed measurement. Like the turbine speed, the signal is fed into the National Instruments PCI counter card to record the time per revolution that results in the frequency of pulse generation.

3.5 Control, Data Acquisition and Processing

3.5.1 FieldPoint Control and Monitoring System

As alluded to in the previous sections, the control and data acquisition for the entire test rig was all managed via two personal computers located outside the test room. These two computers were divided such that one was used for main rig control and steady-state data acquisition, and the second for unsteady, high-speed data acquisition and control. Most of these tasks were performed with a series of Labview programs designed for different stages of control, data acquisition and post-processing. Many of these required significant modification to adapt to the changes needed to test the double-entry ABB turbine.

The *National Instruments FieldPoint* system is a modular data acquisition arrangement where a series of input or output modules can be installed to a base unit that is controlled via an Ethernet connection to a personal computer. This setup has the great advantage of having the flexibility to adapt to changes in the instrumentation without constantly having to re-hardware the connection to the computer terminal. However, the Ethernet connection also restricts the data transfer speeds, meaning that this system must be limited to steady-state data acquisition and lab control. For a sufficient number of I/O channels, two banks of FieldPoint modules are connected to

the computer terminal as shown in Figure 3.26. Table 3.7 lists the output channels necessary to run the facility from the computer terminal, and Table 3.8 lists the input data channels.

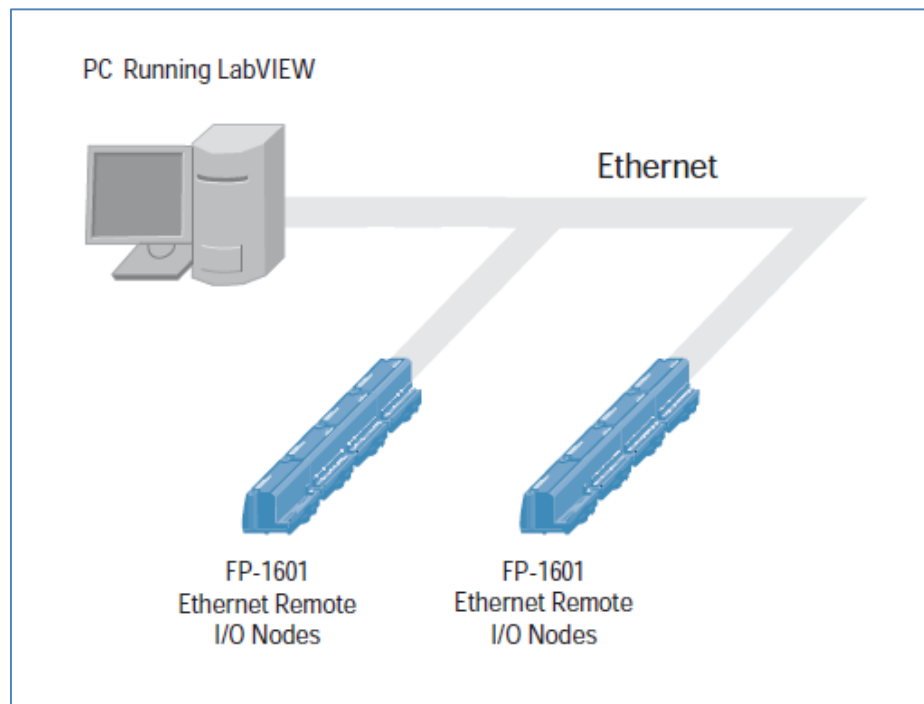


Figure 3.26: FieldPoint module setup for rig control and steady state data acquisition

Table 3.7: FieldPoint output control channels

Output	Control
Scanivalve	Selection of: - the pressure channel - the high or low pressure transducer
Chopper Plate	Speed of DC motor (Pulse Frequency)
Heater	Temperature Control
Air Control Valves	Main valve: Total air flow Limb valves (2): Flow imbalance
Dynamometer	Stator plate position (load)
Guillotine valve	Release of the magnetic gate in emergencies

Table 3.8: FieldPoint Input control channels

Input Source	Data Acquisition
Scanivalve	1. Pressure 2. Scanivalve position
Chopper Plate	Speed of rotation
Thermocouples	1. Heater Stack temperature 2. Orifice Plate temperatures 3. Turbine inlet temperatures 4. Dynamometer water, bearing, and stator plate temperatures
Air Control Valves	Valve position
Dynamometer	1. Load cell 2. Rotational Speed 3. Water flow rate 4. Stator plate limit switches 5. Vibration
Guillotine valve	Open/Close Position information

For a single steady-flow test, the dynamometer gap was set and the main valve opened to bring the speed of the turbine up to the desired level. After ensuring that the turbine had settled into a steady state condition, the data logging can begin. A purposely designed, steady-state data logging program was used to automatically cycle through the ScaniValve channels and log all the necessary pressures and temperatures. The resulting data file could be used to calculate the performance parameters for this particular condition.

3.5.2 Unsteady Data Logging

The second computer terminal was used exclusively for high speed, unsteady data acquisition. Two high speed cards were connected to the terminal via the PCI slot: a National Instruments analogue PCI card Type NI 6034E and a National Instruments PCI counter card, Type NI

6602. The former is a 16-Bit, Analog-Input Multifunction DAQ with 16 single ended or 8 differential analogue inputs. To reduce electronic noise in the signal, all the channels were set up with differential input meaning only 8 channels were available. However, referring to Table 3.9 with the exception of turbine speed, there were a total of 12 unsteady measurements needed. To get around this, a series of solid state switches were connected to each of the eight DAQ channels so that two sensors could be connected to each channel. The series of solid state switches could then be switched between sensors from computer terminal via a FieldPoint output channel. Although this method required additional time for data logging, it did succeed in doubling the number of high speed measurements that could be made on a single DAQ card.

Table 3.9: High-speed data acquisition channels

Quantity	High-speed Data Acquisition	Number
Pressure	Volute Circumference	6
	Volute inlet	2
	Exit Duct	2
Mass Flow	Hot wires	2
Speed	Turbine Speed	1

3.5.3 Unsteady Ensemble Averaging

A single unsteady test was set up in a similar way to a steady-flow test except that the chopper plate speed was selected to obtain the desired frequency prior to spinning the turbine up to speed. Once the turbine had reached a stable operating speed, a Labview logging program especially designed for the ABB turbine was started. The unsteady data logging process centred around the need to traverse the hotwire through a 21 point pattern. At each of these points, the chopper plate reference pulse triggered the analogue data logging of all the pressure and mass flow sensors at a 20kHz sample rate for a full 50 pulse cycles. For the pressure data, this corresponds to a total of 1050 pulse cycles recorded at 20kHz. For the mass flow, each hotwire was also recorded for 50 cycles at each point in the traversing pattern, but these needed to be integrated together according to the British Standard BS-1042 (equation 3.16) to obtain a final mass flow at each point in time. The

unsteady speed signal could not be sampled at the same rate as explained in Section 3.4.6 since this depended on the number of shutters and speed of rotation. Therefore, the speed signal was re-sampled at 20kHz in the post-processing phase to match the pressure and mass flow data.

It should be immediately apparent from this discussion that the post processing of the unsteady data quickly becomes a very complex and involved process due to the volume of data and complexity of the measurements. The first stage of the procedure was to split the 50 pulses into single pulse lengths and average them across each time (ensemble averaging). The ensemble average $\bar{\pi}$ of a property π at time t_j over n cycles will take the form defined by equation 3.21. The value of n was 1050 for the pressure and speed signals, and 50 for each of the hotwire points.

$$\bar{\pi}(t_j) = \frac{\sum_{i=1}^n \pi(t_j)_i}{n} \quad (3.21)$$

This averaging method is one of the most effective means of reducing noise and any other variations in the signal that are not cyclic. This increases the signal to noise ratio by a factor of $\sqrt{n}$. As an example, Figure 3.27 shows the result of ensemble averaging the exit pressure signal.

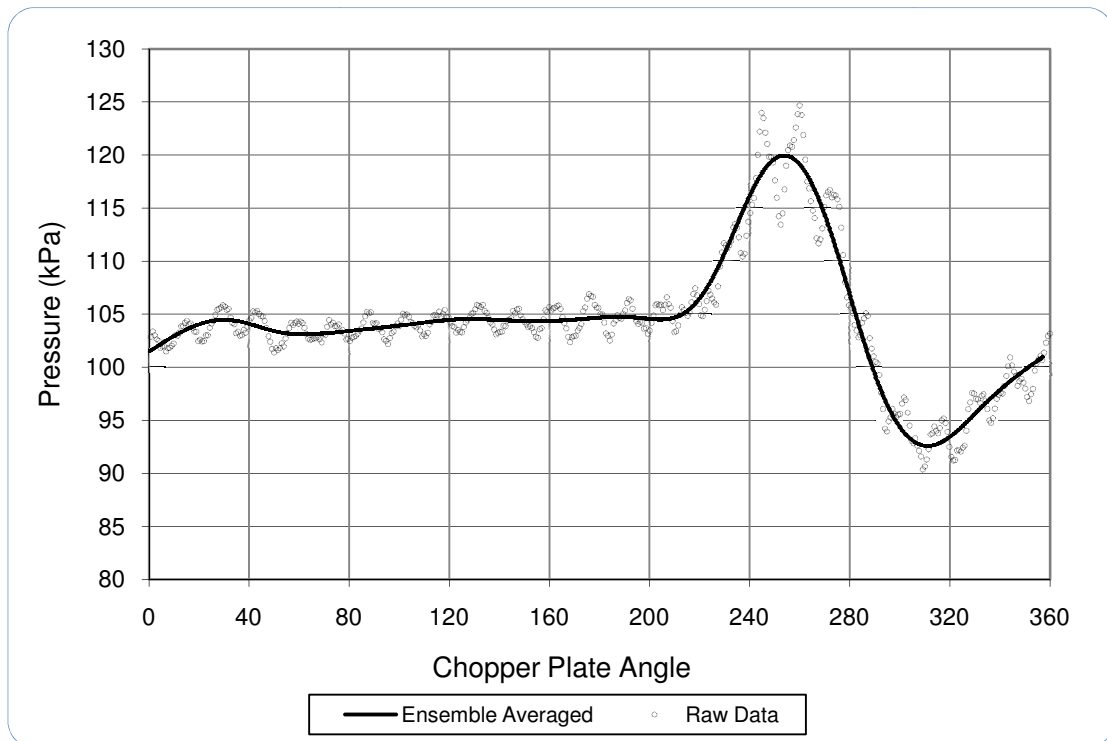


Figure 3.27: Ensemble Averaging of the Exit Pressure

3.5.4 Unsteady Filtering

Ensemble averaging was not always sufficient to remove all of the noise from the unsteady signal. For instance, vibration from the rotating turbine can produce small harmonic fluctuations in the speed signal that may not be completely removed by ensemble averaging due to their cyclic nature. The hotwire is also susceptible to unwanted noise due to small scale turbulence in the flow or vortex shedding from sharp edges in the ductwork. Thus, to remove these features and smooth the pulse profile without removing important physical features, the ensemble-averaged pulse was passed through a low-pass Finite Impulse Response (FIR) filter. The FIR filter is a digital, linear response function that is built into the Labview database with the freedom to vary its characteristics and frequency cut-off. In general, it was found that cyclic events above 10 times the frequency of pulsations could be effectively filtered out without loss of pulse features. Figure 3.28 shows the variation of the turbine speed signal resulting from a 42Hz pulse before and after the FIR filter has been applied. Note that the small-scale cyclic fluctuations have been removed, resulting in a smooth speed curve ready to be processed for the torque calculation.

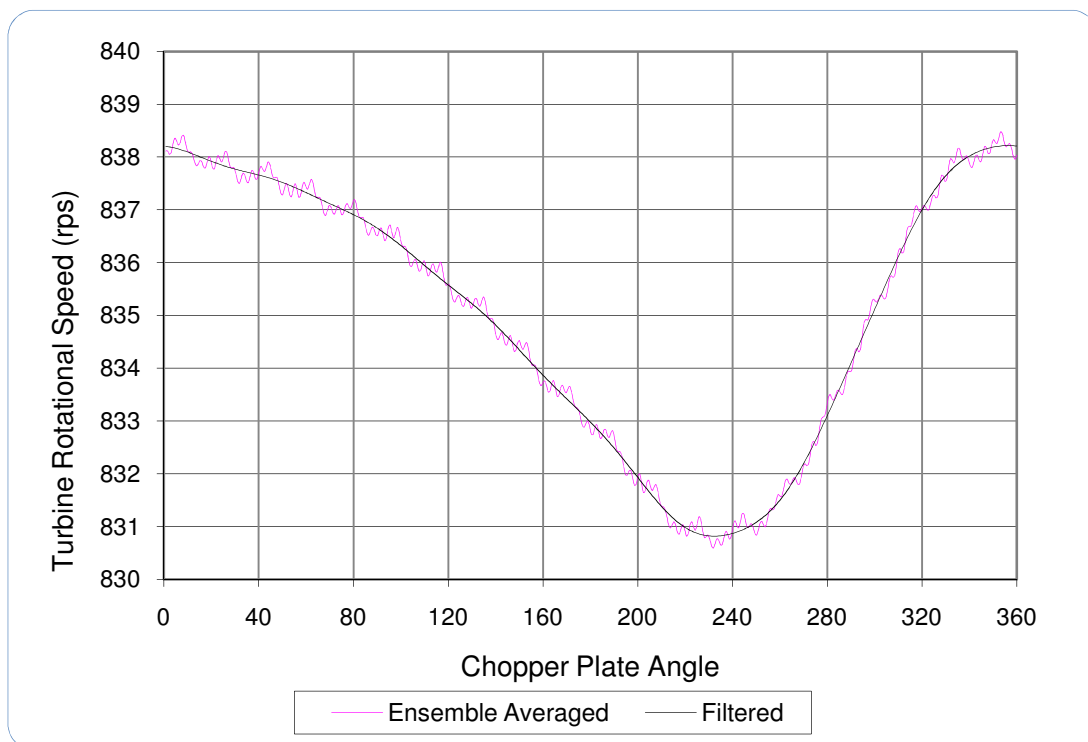


Figure 3.28: Filtering the Turbine Wheel Speed Signal

As a summary, Figure 3.29 lays out all the steps necessary to refine the data from start to finish. It begins with a series of 21 files, each containing 50 pulses of raw voltage data and ends with a single file containing the pressure, mass flow, temperature, speed, and torque information for a single pulse length. This information can then be passed onto a routine that calculates all the necessary performance parameters such as pressure ratio, velocity ratio, power and efficiency.

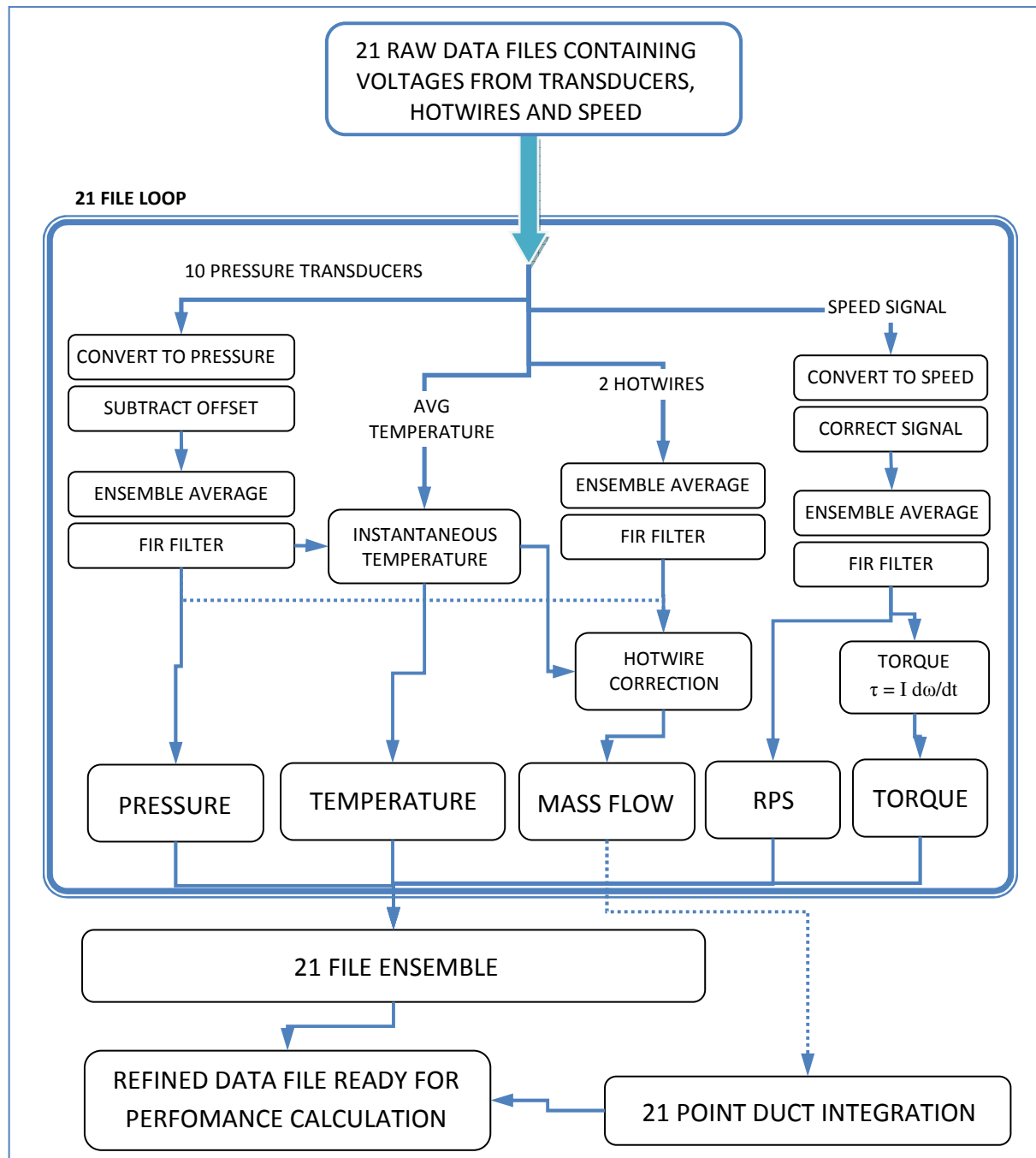


Figure 3.29: Refinement Procedure for Unsteady, Time Resolved Data

3.6 Uncertainty of the Individual Variables

Error in a measurement can be defined as the difference between the measured value and the true value. The difficulty, of course, is to know the true value of any measurement. Although often used interchangeably, the term *uncertainty* therefore refers to the estimate by the researcher of the interval Δx in which the true value x_{true} is believed to lie [44]. In this section, the uncertainty associated with the individual measurements of pressure, temperature, mass flow, torque and speed will be estimated. Since the majority of the measurement techniques and instrumentation in this thesis remain unchanged from the work of Shrinri Syzmko, many of these estimates are based on his exhaustive review of the uncertainty in reference [8]. How the individual variables propagate this uncertainty to the performance calculations in the next two chapters will be considered separately in these sections.

One method of estimating uncertainty in a sensor or transducer is through multiple calibrations against a known standard. For instance, Figure 3.23 shows the in-situ calibration of the hotwire against the orifice plate for a range of different conditions. To assess the uncertainty interval resulting from the measurements x_i , the root mean square deviation from the calibrated standard values $x_{i,cal}$ is used as defined by equation 3.22. Here the 1.96 multiplier is added to this equation to calculate a 95% confidence interval. As the number of samples increases, this equation should move from a measure of the precision toward an estimate of the bias in the measurements.

$$\pm\Delta x = \pm \sqrt{\frac{1.96^2}{n-2} \sum_{i=1}^n (x_{i,cal} - x_i)^2} \quad (3.22)$$

3.6.1 Steady Measurement Uncertainty

PRESSURE

There are a few possible sources of steady pressure measurement uncertainty. These include the pressure tapping error, non-linearity of the transducer or the calibrator, temperature drift, and signal/amplifier voltage drift. The ScaniValve unit contains a high pressure transducer (*Druck PDCR 23D*) with a range of ± 3.5 bar and the low pressure transducer (*Druck PDCR 22*) with a range of ± 0.35 bar. These have rated full-scale uncertainties of $\pm 0.02\%$ and $\pm 0.008\%$ respectively. However, Szymko estimated the non-linearity of these transducers from the calibration to be $\pm 0.07\%$ for the high pressure and $\pm 0.1\%$ for the low pressure transducer. Since these were calibrated against

a third standardized transducer, certified for calibration within $\pm 0.025\%$, this will add a small additional uncertainty. The error associated with a static pressure tapping was considered with reference to Benedict [45]. This error arises from an aerodynamic flow interaction with the hole in the pipework and scales with the flow Reynolds number. The tapping uncertainty in the high pressure transducer at full scale was estimated to be ± 100 Pa but negligible for the low-pressure transducer.

Putting these uncertainties together for a worst case uncertainty estimate gives ± 470 Pa for the high pressure transducer and ± 86 Pa for the low pressure transducer. This uncertainty includes an estimate of the voltage drift from the signal amplifier. Note that the temperature drift is considered negligible here since the transducers are mounted in a separate cabinet and not physically connected to the rig.

TEMPERATURE

The uncertainty of the average turbine inlet temperature was estimated on the basis of a three point calibration procedure. Within the temperature range 273-373K, the uncertainty was considered to be no more than ± 0.4 K.

MASS FLOW

The uncertainty in the steady mass flow measurement will be related to the pressure and temperature uncertainties as is clear from equation 3.6. The propagation of these uncertainties to the final mass flow measurement is outlined by the British Standard BS 5167-1:1997 document [41]. The maximum uncertainty of the mass flow measurement is estimated to be $\pm 2\%$ under low flow conditions.

TURBINE TORQUE AND SPEED

The turbine torque is calculated from a direct measurement of the force applied onto a load cell located at a given distance from the dynamometer gimbal axis. There are a few sources of uncertainty in this measurement, some of which are difficult to assess quantitatively. Although the load cell is equipped with temperature compensating circuitry, there is always some risk of bias error due to temperature drift. To mitigate this error, the turbine was brought up to a stable operating temperature before data acquisition. Also, before each test, the voltage offset at zero torque was recorded and subtracted from the data. The uncertainty due to voltage drift was therefore considered low in comparison to the influence of external frictional torque on the

dynamometer body. This second source of error stems from the need for a completely free movement of the entire dynamometer mounted on the gimbal bearings. Thus, any resistance from the water hoses, air/oil pipes, and wiring attachments could produce an unwanted bias error. This error is, to some degree, taken into account from the calibration shown in Figure 3.14. This calibration was performed at regular intervals and always with all the fluids running through the pipes. By substituting 40+ calibration points into equation 3.21, an estimate of the uncertainty in the torque measurement was found to be ± 0.071 Nm to a 95% confidence interval. To obtain the power, the value of torque must be multiplied by the value of turbine speed obtained from the calibrated *Omron EE-SX4101* infra-red optical sensor. Uncertainty associated with the speed measurement has been estimated to be on the order of ± 2 rps [8].

3.6.2 Unsteady Measurement Uncertainty

PRESSURE

There were a total of 10 strain gauge pressure transducers installed at the various locations listed in Table 3.6. Two types were used, a compact designed SenSym 19C050PJ7K rated for 0-50psi and a bulkier Schaevitz Type P704-0001 rated up to 3.45 bar. Both have temperature compensation circuitry which helps to reduce temperature drift error. The SenSym transducers are rated for a maximum, full scale non-linearity of $\pm 0.25\%$ (± 863 Pa), but the RMS non-linearity measured from the calibration data was ± 1012 Pa. The Schaevitz transducer is rated for a maximum combined nonlinearity of $\pm 0.18\%$ (± 621 Pa), and the calibration confirms this value with RMS deviation of ± 604 Pa according to equation 3.21. The transducer used for the calibration is the same as for the steady measurements, thus adding an additional uncertainty of 0.025% (86Pa). Thus, the maximum pressure measurement uncertainty is ± 1098 Pa for the SenSym and ± 707 Pa for the Scaevitz transducers. Since all these measurements were ensemble averaged over 1050 pulse lengths, this will significantly reduce the signal to noise ratio in the final data arising from a variety of unsteady effects.

MASS FLOW

Due to the complexity of the measurement and its dependence on a large number of parameters, the best estimate of the uncertainty associated with the mass flow measurement is gained from the in-situ calibration against the orifice plate. Using equation 3.21 and over 100 data points, the RMS uncertainty compared to the orifice plate was calculated to be ± 0.018 kg/s to a 95% confidence interval. Due to the large number of calibration points, these values should give a

reasonable estimate of the bias error. Like the pressure, the ensemble average will help to further reduce the precision error in the final mass flow measurement.

TEMPERATURE

It is quite difficult to assess the temperature error associated with the isentropic compression-expansion assumption (Equation 3.17) without measuring the 'true' unsteady temperature as shown in Figure 3.24 by Szymko [8]. One way could be to assume that equation 3.17 is completely accurate and judge the uncertainty based on a propagation of error from the averaged temperature and the fluctuating pressure. However, since the instantaneous temperature data is calculated in the same manner as the work of Szymko [8], his assessment of the error in this approach is likely the most accurate for the work in this thesis. Using the data from the dual hotwire experiments, he prescribes an unsteady temperature uncertainty interval of $\pm 3\text{K}$.

TURBINE SPEED

The unsteady turbine speed was obtained by measuring the time between pulses generated by the optical speed sensor as explained in Section 3.4.6. The data generated from the high-speed counter card was corrected to account for the non-uniformity in the shutters of the speed sensor. This improved the uncertainty of the measurement to an estimated ± 0.16 rps. However, considering that the turbine speed is also ensemble averaged, this should decrease the signal to noise ratio significantly. Assuming the uncertainty will scale with $1/\sqrt{n}$, where n is number of repeated data points, the resulting uncertainty in the speed measurement will be ± 0.0072 rps. This does not include the additional influence of noise reduction due to digital filtering.

Chapter 4

Steady Flow Experimental Results

4.1 Introduction

This chapter presents the performance of the double-entry, ABB turbocharger turbine under wide range of steady-flow test conditions. It might seem somewhat surprising that an extensive series of steady-state tests are relevant, considering the reality of the highly pulsating flow when attached to an engine. However, there are a number of reasons why this first step is so important. To begin with, the standard selection and design process in the turbocharger industry still relies on treating the turbine as a constant-flow device. Note, for example, that many of the equations in Chapter 1 dealing with basic turbocharger design have been derived with the assumption of a steady flow. In addition, most of the research and development in turbocharger turbine design is based on steady fluid dynamics. Furthermore, the steady treatment of the turbine is also extends to the engine-turbocharger matching procedures. Even the more sophisticated engine simulation codes in use today rely on steady-state performance maps. Since these maps are often quite narrow due to testing limitations, significant extrapolation and interpolation is needed. Given these considerations, it is therefore important to investigate how far the unsteady performance differs from the constant-flow performance and thus provide some physical insights. One of the questions in this thesis is to understand if the unsteady performance can be treated as a succession of steady states (so called *quasi-steady calculation*). Since the turbine experiences a wide variation of conditions during a single unsteady pulse, in order to make a full comparison between steady and unsteady performance, the scope of available steady-flow test data must be broad enough to produce a reliable comparison.

In addition to the quasi-steady question, it is also important to study the behaviour of this turbine when the flows are introduced unequally. While this situation occurs instantaneously when the pulses enter each inlet out-of-phase, the best way to begin to understand the influence of two different flows driving the turbine wheel is by introducing them in a steady-state manner. This research is especially interesting for this particular design of turbocharger given that there will be two separate flows driving the same turbine wheel. Unfortunately, the extra inlet also brings an

extra degree of freedom to the number of possible test conditions meaning a large amount of experimental data is required to fully characterize the turbine. The results from these unequal tests are therefore provided in Section 4.3 of this Chapter.

One final incentive for steady-state experimentation is that it is quite straightforward to model in computation fluid dynamic (CFD) software. A three-dimensional CFD model gives the engineer the ability to resolve the complete fluid domain and there study some of the possible causes of performance observed from the steady-state tests. The results from a steady-state CFD analysis of the turbocharger turbine are provided in Chapter 5.

4.2 Equal Admission

4.2.1 Performance Parameters

Efficiency

The efficiency is one of the most important performance parameters in any turbomachinery application, so a clear understanding of its definition is vital. The basic form is familiar to every engineer, namely, the ratio between output and input energy. More precisely for a turbine, it is the amount of shaft work compared to a theoretical amount that would result from the working fluid following an ideal process. How this basic definition is derived will depend to some degree on convention and what is appropriate for a particular application. The efficiency definition that will be used exclusively in this thesis is the *total-to static isentropic efficiency*:

$$\eta_{ts} = \frac{h_{01} - h_{04}}{h_{01} - h_{4, is}} = \frac{\dot{W}_{act}}{\dot{W}_{is, ts}} \quad (4.1)$$

where the difference between the total enthalpy at inlet h_{01} and the isentropic *static* exit state $h_{4, is}$ is the theoretical amount of power assumed available to the turbine. The theoretical and actual processes are shown on the enthalpy-entropy diagram in Figure 4.1. The *total* inlet state is used here since the kinetic energy at the inlet of the turbocharger will be available for use. However, since this is a single stage turbine where the working fluid is simply expelled to atmosphere after exiting the

wheel, the *static* state at exit is most appropriate. This definition of isentropic energy is defined by equation 4.2.

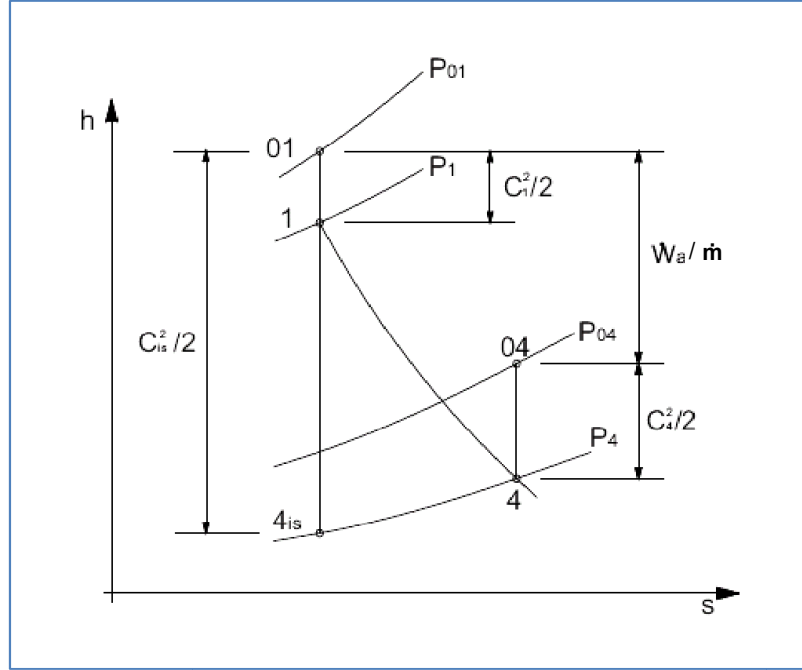


Figure 4.1: Turbine enthalpy – entropy diagram

$$\begin{aligned}\dot{W}_{t-s,isen.} &= \dot{m}(h_{01} - h_{4,is}) \\ &= \dot{m} \left[\int_{4,is}^1 c_p dT + \frac{1}{2} U_1^2 \right] = \dot{m} \frac{C_{is}^2}{2}\end{aligned}\quad (4.2)$$

where the inlet dynamic enthalpy $1/2 U_1^2$ and specific heat integral $\int_{4,is}^1 c_p dT$ sum to the total-to-static enthalpy across the stage. This equation also provides the definition of isentropic velocity C_{is} .

The specific heat integral in equation 4.2 was evaluated as a function of the static air temperature at the measuring plane (station 1) and the exit (station 4).

$$\int_{4,is}^1 c_p dT = \left(\begin{aligned} &A_0 T_1 + \frac{A_1 T_1^2}{2} + \frac{A_2 T_1^3}{3} + \frac{A_3 T_1^4}{4} + \frac{A_4 T_1^5}{5} + \frac{A_5 T_1^6}{6} \\ &- A_0 T_{4,isen} - \frac{A_1 T_{4,isen}^2}{2} - \frac{A_2 T_{4,isen}^3}{3} - \frac{A_3 T_{4,isen}^4}{4} - \frac{A_4 T_{4,isen}^5}{5} - \frac{A_5 T_{4,isen}^6}{6} \end{aligned} \right) \quad (4.3)$$

where:

$$c_p = A_0 + A_1 T + A_2 T^2 + A_3 T^3 + A_4 T^4 + A_5 T^5$$

$$\begin{aligned} A_0 &= 0.10831 \times 10^4 & A_1 &= -0.68388 \times 10^0 \\ A_2 &= 0.17875 \times 10^{-2} & A_3 &= -0.11324 \times 10^{-5} \\ A_4 &= -0.82943 \times 10^{-9} & A_5 &= -0.11100 \times 10^{-11} \end{aligned} \quad (4.4)$$

The static T_I and total temperatures T_{0I} are calculated from the measured temperatures using the recovery factor from equations 3.8 and 3.9. Exit static temperature $T_{4,isen}$ is evaluated using the isentropic relation:

$$T_{4,isen} = T_{0I} \left[\frac{1}{P_{0I}/P_4} \right]^{\frac{\bar{\gamma}_{(1-4)} - 1}{\bar{\gamma}_{(1-4)}}} \quad (4.5)$$

Where total pressure is evaluated as:

$$P_{0I} = P_I \left[\frac{T_{0I}}{T_I} \right]^{\frac{\gamma_I}{\gamma_I - 1}} \quad (4.6)$$

Equation 4.5 requires an iterative solution since the average specific heat ratio between inlet and exit $\bar{\gamma}_{(1-4)}$ is dependent on the exit specific heat calculated from equation 4.4, which in turn is dependent on the exit static temperature.

The inlet dynamic enthalpy $1/2 U_1^2$ can be simply evaluated using the velocity of the flow at inlet U_I or the total and static temperatures as per equation 4.7.

$$\frac{1}{2} U_1^2 = c_{p,1} (T_{0I} - T_I) \quad (4.7)$$

This finally allows us to evaluate the inlet isentropic energy $W_{I-s,isen}$ and the isentropic velocity C_{is} from equation 4.2.

The difference between the total enthalpy states at inlet and exit ($h_{01} - h_{04}$) represent the specific work extracted by the turbine. This can be evaluated on a turbine gas stand by measuring the total temperature difference at the inlet and outlet of the turbine. However, there is a major drawback with this approach since any heat transfer to the surroundings must be carefully accounted for to prevent an overestimate of the power produced. By far the best measure of actual power from a turbine is therefore to measure it directly from a dynamometer as outlined in section 3.3.6. By measuring the reaction torque τ of the floating dynamometer body and multiplying it by the angular velocity ω , the actual power will result:

$$\dot{W}_{act} = \tau \cdot \omega = \tau \cdot 2\pi \cdot N \quad (4.8)$$

It is important to understand what is included in this measurement of torque. As outlined in the previous chapter, the Imperial College eddy-current dynamometer is mounted on a set of gimbal bearings which allow it to react onto the load cell from which torque is measured. A free-body diagram of this arrangement would demonstrate that any frictional parasitic loads on the shaft (bearings, etc) will only contribute to the total load on the turbine. Therefore, the resulting torque measurement is a direct evaluation of the full output power of the turbine and will *not* include bearing loss. This differs somewhat from a real turbocharger where the power delivered to the compressor will be the turbine power less the bearing losses. Nonetheless, the dynamometer has the clear advantage of reporting only aerodynamic performance of the turbine.

Velocity Ratio

The velocity ratio is a useful parameter for compressor-turbine matching and can be related to the turbine wheel flow angles as demonstrated in Section 1.34. Therefore, it is a convenient parameter to plot against turbine efficiency since the peak efficiency typically occurs at a similar velocity ratio, regardless of speed. It is defined by equation 4.9 below:

$$VR = \frac{U_3}{C_{is}} = \frac{\pi \cdot d_3 N}{\sqrt{\frac{2W_{t-s,isen}}{\dot{m}}}} = \frac{\pi \cdot d_3 N}{\sqrt{\left(2 \int_{4,is}^1 c_p dT + U_1^2\right)}} \quad (4.9)$$

To calculate the velocity ratio, the value of isentropic energy from equation 4.2 is simply substituted into this equation along with the speed of the turbine N .

Pressure Ratio, Mass Parameter and Speed Parameter

The remainder of the performance parameters used to assess turbine performance are all quite straightforward to calculate for steady, equal admission operation. These three parameters are listed below:

$$PR = \frac{P_{01}}{P_4} \quad (4.10)$$

$$MP = \frac{\dot{m}\sqrt{T_{01}}}{P_{01}} \quad (4.11)$$

$$SP = \frac{N}{\sqrt{T_{01}}} \quad (4.12)$$

Note that the velocity ratio, speed parameter and pressure ratio can all be related to each other in a single equality as shown in equation 4.13:

$$\frac{U_3}{C_{is}} = \left(\frac{N}{\sqrt{T_{01}}} \right) \cdot \frac{\pi \cdot d_2}{\sqrt{2 \cdot c_p \cdot \left(1 - (P_4/P_{01})^{\frac{\gamma-1}{\gamma}} \right)}} \quad (4.13)$$

4.2.2 Graphical Representation

When it comes to presenting the performance of the turbocharger operating with an equal inlet flow (or a single entry), there are a number of conventional means of displaying the relationship between the performance parameters. For a turbocharger compressor, the most common way is to plot the pressure (or expansion) ratio against the mass parameter for lines of constant speed. Lines of constant efficiency can be added on the same plot since the compressor map is broad enough to make these clear. However, Figure 4.5 demonstrates that this same approach for the turbine produces a very narrow map, meaning that efficiency must be plotted

separately. Thus, the turbine efficiency is most commonly plotted against the velocity ratio with lines of constant speed.

4.2.3 Experimental Method

The range of flow conditions that can be tested using the eddy-current dynamometer is more extensive than can be achieved by a typical gas stand since it is not limited by the surge and choke of the compressor. However, the dynamometer does still have physical limitations. Speed is limited to approximately 60,000 rpm by the specifications of the high-speed roller bearings. There is also a limitation on the maximum power absorption at this speed. Figure 4.2 illustrates the power absorption characteristics of the dynamometer predicted by Szymko [8] which shows a maximum power rating of 62kW at the narrowest rotor-to-stator air gap.

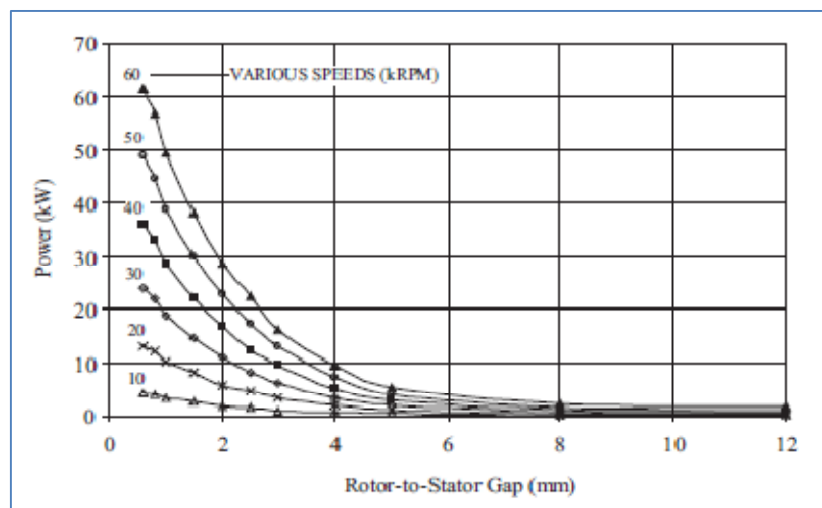


Figure 4.2: Eddy-current dynamometer power absorption [8]

In order to obtain a live pressure ratio reading during a test, simultaneous measurements of mass flow (requiring 2 static pressure measurements for each limb) and inlet pressures are needed. Since the Scanivalve unit used to measure steady flow pressures can only read a single channel at a time, live calculation of the performance parameters was not possible. It therefore proved to be a lengthy process to adjust the mass flow and dynamometer gap incrementally to obtain a precise pressure ratio or velocity ratio. Thus, the easiest approach to obtain a full steady performance map was to test at constant speed. Not only was this simpler experimentally, but it also affords a better comparison with the unsteady data since the rotor speed is essentially invariant over a pulse cycle. For each dynamometer air gap, the turbine wheel speed was set by metering the air flow through

the system. This process was repeated for different air gaps and a range of turbine speeds between 50% and 100%. Approximately 12-13 different stator-to-rotor gaps ranging between 0.8-10mm provided a good distribution of data for each speed line.

4.2.4 Experimental Results and Discussions

Table 4.1 lists the range of turbine speeds used to generate the steady-state turbine performance map. Six equivalent speeds were tested ranging from 50% to 100% speed in 10% increments. This corresponds to an approximate speed range of 30,000 to 60,000 rpm or a speed parameter range of 26.9 to 53.7 $rps/K^{1/2}$. The standard performance maps are provided in Figures 4.3 to 4.5 along with a tabulated summary of the test data in Table 4.2. The figures show, in order, the efficiency versus velocity ratio (Figure 4.3), the efficiency versus pressure ratio (Figure 4.4), and the mass flow parameter versus pressure ratio (Figure 4.5). Table 4.2 lists the performance parameter data corresponding to three points along the steady speed line: the locations of minimum velocity ratio, peak efficiency, and maximum velocity ratio.

Table 4.1: Full admission test conditions

Percentage Speed	Average Turbine Speed (RPM)	Speed Parameter $N/\sqrt{T_{01}}$ (rps/ $K^{1/2}$)	Maximum $\dot{W}_{act}$ (kW)	Minimum $\dot{W}_{act}$ (kW)
50%	28,697	26.9	20.7	0.52
60%	34,600	32.3	26.8	0.79
70%	40,426	37.8	33.0	1.13
80%	45,832	43.0	43.4	1.46
90%	52327	48.3	43.4	2.07
100%	58010	53.7	45.9	2.15

Figure 4.3 shows the relationship between the total-to-static efficiency of the turbine and the velocity ratio under which it is operating. As equation 1.7 reveals, the velocity ratio is a parameter that relates the speed of the rotor, to the fluid that is driving it. It is therefore not

surprising that this plot produces a curve with a distinct maximum as this is the condition where the fluid flow is best matched to the spinning rotor. For the range of speeds tested here, these equal admission efficiencies all fall in between 75-80% at peak, demonstrating the excellent performance of the ABB turbine design. The maximum measured efficiency (from Table 4.2) of 80.4% occurs at a turbine speed parameter of $48.3 \text{ rps}/\sqrt{K}$. Although the magnitude of the optimum efficiency is clearly important, the location where it occurs is also significant to the designer. The location of peak efficiency should correspond to the predominant operating condition of the turbocharger for the particular engine application.

The test conditions that produced the peak efficiency are provided in Table 4.2 (with respect to the three parameters: velocity ratio, pressure ratio and mass parameter). The optimum velocity ratio ranges between 0.62-0.67 with a mean value of 0.643. However, since these tests are only discrete points along a constant speed line, the location of peak efficiency is better evaluated by calculating the first derivative of a third-order polynomial curve fit. This leads to a much more consistent optimum velocity ratio over the range of speeds tested. From 50-90% speed, this method suggests an optimum velocity ratio range of 0.661 -0.669 and an average of 0.663. The results show that there is no correlation between speed and the optimum velocity ratio, as this value stays relatively constant for the range of speeds tested.

It has been shown in Chapter 1 (equations 1.6-1.8) that the peak velocity ratio for a *radial* turbine with zero exit swirl is typically constrained to a value of 0.7. The result for this turbine with an optimum velocity ratio of 0.66 highlights one of the main advantages of the mixed-flow turbine design. Note from equation 4.13 that a lower velocity ratio corresponds to a higher pressure ratio for a given speed, meaning that there is a greater energy available in the fluid under these conditions. It is therefore advantageous to be able to produce a turbine that operates most efficiently where the energy content in the flow is greater. This is especially true in a pulsating flow where the high pressure ratios that occur at the peak of the pulse contain much of the energy in the flow. A mixed flow rotor should therefore operate more efficiently under these conditions than the equivalent radial turbine due to the shift in optimum velocity ratio. Figure 4.4 also demonstrates that for this turbine operating at 90% speed, a very flat efficiency curve (near optimum) results from any pressure ratios above 1.6.

Figure 4.3 shows the large range of velocity ratios (0.362 to 1.184) that is achievable using the eddy-current dynamometer as a loading device. As the speed of the turbine increases, the range of test conditions is reduced due to power absorption limitation at high pressure ratios. For the sake of comparison, the extent of the 50% and 100% speed lines published by Karamanis et. al [27] using

a standard compressor as a loading device are included in Figure 4.3. It is clear from this comparison that the dynamometer significantly improves the width of the data available to the researcher. This additional, off-design data is most important for a steady-unsteady comparison since a pulsating flow produces a wide variation of flow conditions over the pulse cycle.

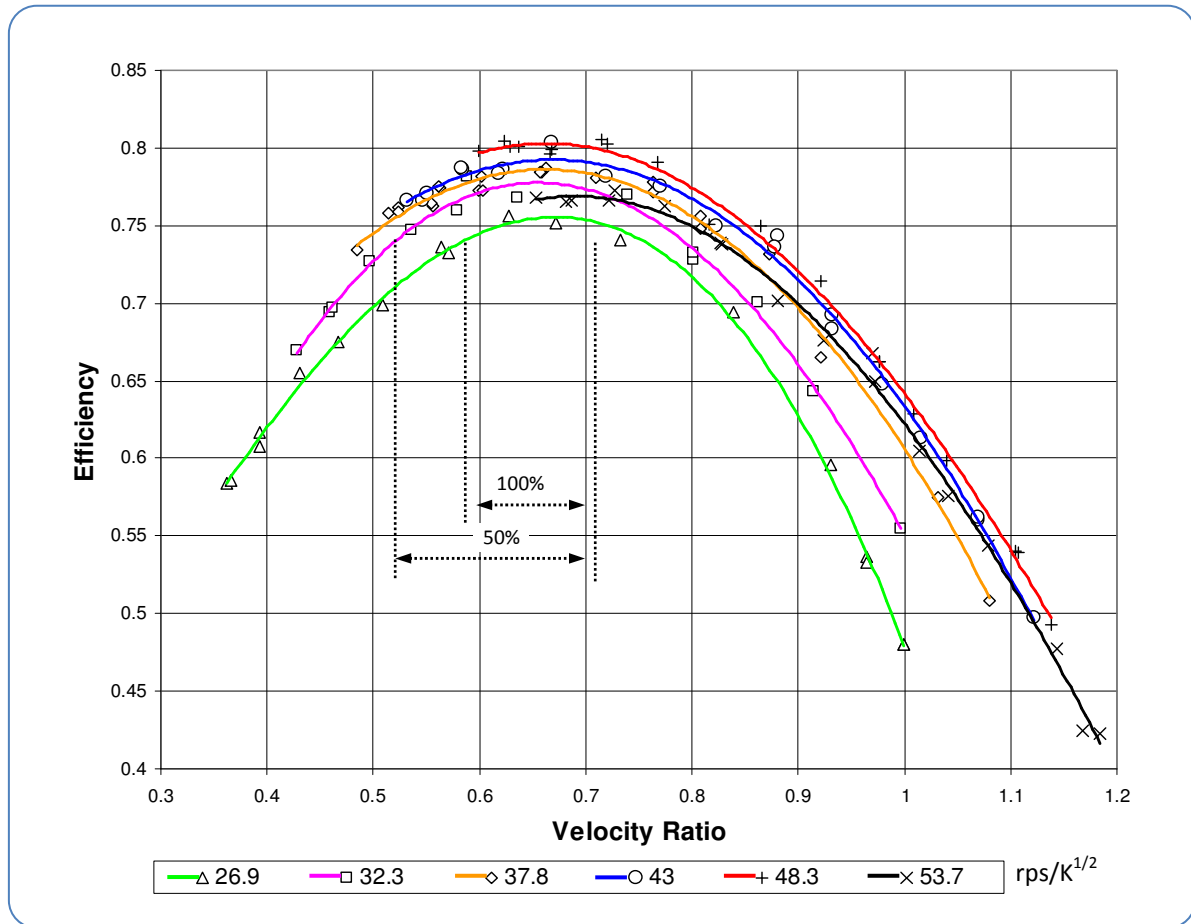


Figure 4.3: Efficiency versus velocity ratio (constant speed parameter)

Figure 4.5 shows a standard plot of mass parameter versus total-to-static pressure ratio for the dual entry turbine. As with the previous figures, lines of constant speed parameters ranging from $26.9 \text{ rps}/\sqrt{K}$ to $53.7 \text{ rps}/\sqrt{K}$ are plotted. The wide range of pressure ratio and mass parameter test conditions is also clear from the figure and specified by Table 4.2 as 1.099 - 2.679 and 2.023×10^{-5} - $4.78 \times 10^{-5} (\text{kg/s})/\sqrt{K/\text{Pa}}$ respectively. The trend of a decreasing rise in mass flow with an increase in the pressure ratio is typical of a turbocharger turbine. At a fixed turbine speed parameter, this behaviour is similar to an adiabatic nozzle where the rise in mass flow through the nozzle increases with pressure ratio until an asymptotic limit is reached that corresponds to sonic conditions in the throat. This condition is referred to as the *choking* condition. Unlike a simple nozzle, however, the

swallowing capacity of a turbine shows a speed dependency due to the centrifugal pressure field created by the rotating turbine wheel. This speed dependent pressure field will oppose the motion of the gas through the rotor, causing the turbine to choke at a higher pressure ratio than a simple convergent nozzle. Other analogies for the radial turbine can be consider beyond the single nozzle statement above, for instance it be considered as two nozzles in series relating to the stator and the rotor, both of which have locations of minimum area.

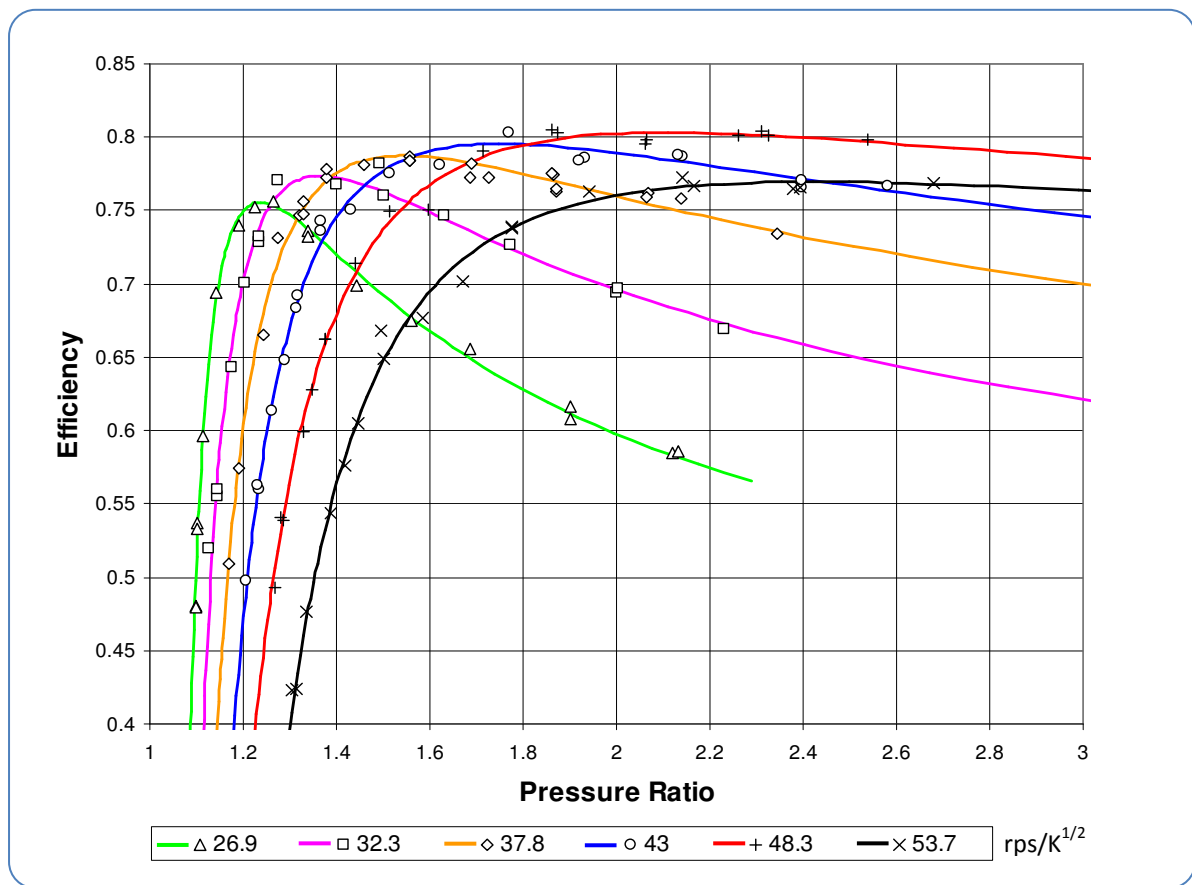


Figure 4.4: Efficiency versus pressure ratio (constant speed parameter)

Since the blade exit diameter of the current rotor (TPS-IC) is slightly greater than previously tested at the Imperial facility, the rotor exducer area will be larger. This explains why the constant speed lines in Figure 4.5 do not reach a horizontal asymptote (i.e. they do not show the choking condition) as per other published work on smaller turbines. Also, the rotor is not the only minimum area in the stage – choking could also occur in the nozzle blade throat location. The flow capacity through both these critical areas (rotor and nozzle throats) will also depend on the fluid condition in the passages. Effects such as flow separation and boundary layer growth can serve to produce an

aerodynamic blockage, thereby further accelerating the flow. In general, the nozzle and rotor passages are designed to avoid sonic conditions due to the limitations on the mass flow and the shock wave losses that might be induced.

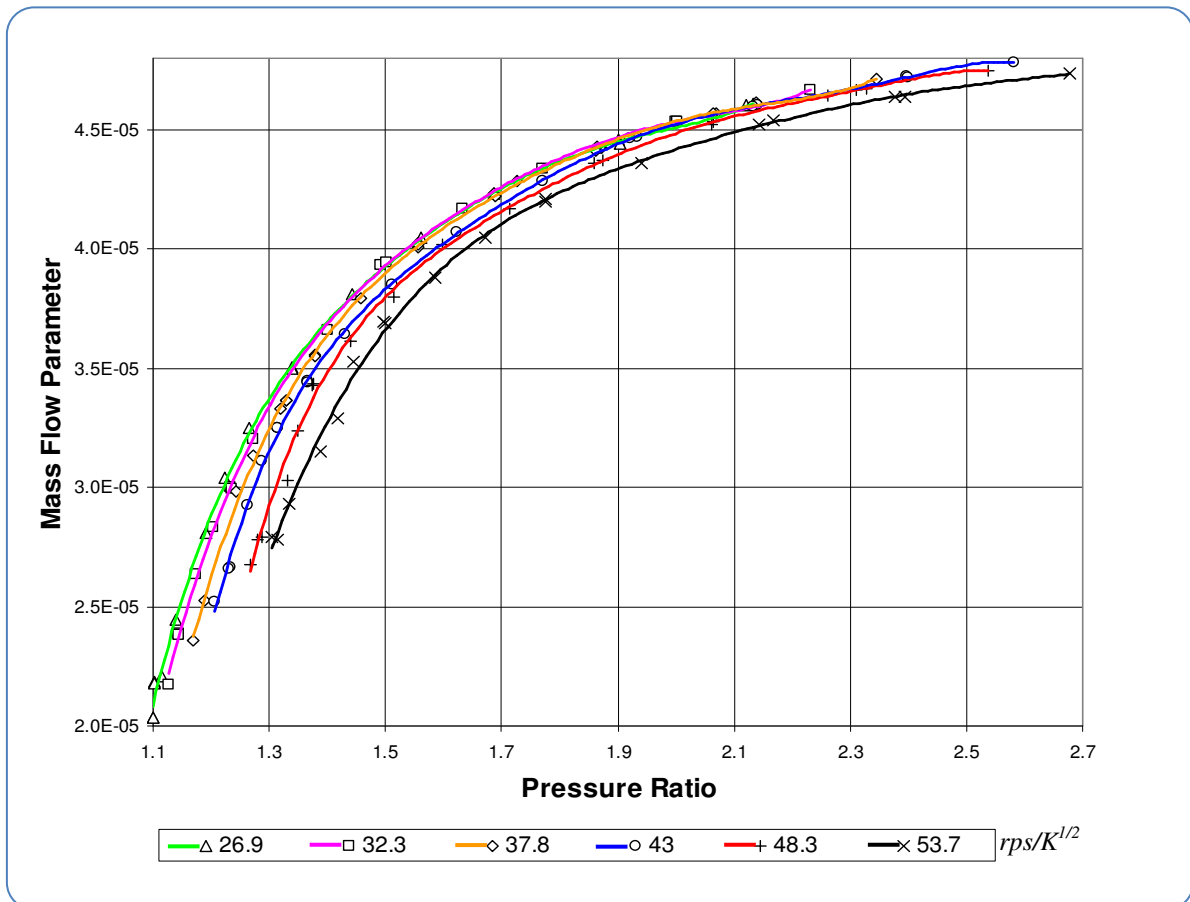


Figure 4.5: Mass flow parameter versus pressure ratio (constant speed parameter)

On the other end of the flow rate spectrum, the turbine does not strictly have a minimum mass flow but there comes a point where the energy of the flow can no longer sustain the momentum of the rotor, causing the turbine to act as a compressor. This point is reached when the static pressure at inlet matches the centrifugal pressure field caused by the rotating wheel. Due to the design of this turbine, this situation could arise once the flows are introduced unequally into each inlet. These low flow conditions can also occur during pulsating flow when a closed exhaust valve causes the turbine to free-wheel for a small amount of time.

Table 4.2: Full admission test results

$N/\sqrt{T_{01}}$ ($rps/K^{1/2}$)	Location on Steady Map	U_3/C_{is}	η_{t-s}	$\dot{m}\sqrt{T_{01}}/P_{01}$ [(kg/s) $\sqrt{K/Pa}$]	P_{01}/P_4
26.9	Minimum VR	0.3623	0.584	4.601×10^{-5}	2.1208
	Peak η_{t-s}	0.6276	0.756	3.247×10^{-5}	1.264
	Maximum VR	1.00	0.480	2.023×10^{-5}	1.099
32.3	Minimum VR	0.428	0.669	4.667×10^{-5}	2.232
	Peak η_{t-s}	0.635	0.77	3.663×10^{-5}	1.400
	Maximum VR	1.0272	0.52	2.174×10^{-5}	1.127
37.8	Minimum VR	0.485	0.734	4.717×10^{-5}	2.346
	Peak η_{t-s}	0.662	0.787	4.006×10^{-5}	1.557
	Maximum VR	1.0803	0.5088	2.361×10^{-5}	1.170
43.0	Minimum VR	0.5322	0.766	4.780×10^{-5}	2.580
	Peak η_{t-s}	0.667	0.803	4.286×10^{-5}	1.770
	Maximum VR	1.1215	0.4974	2.522×10^{-5}	1.206
48.3	Minimum VR	0.5996	0.798	4.750×10^{-5}	2.537
	Peak η_{t-s}	0.6230	0.804	4.667×10^{-5}	2.26
	Maximum VR	1.1375	0.493	2.674×10^{-5}	1.2665
53.7	Minimum VR	0.6527	0.7685	4.735×10^{-5}	2.679
	Peak η_{t-s}	0.6527	0.7685	4.735×10^{-5}	2.679
	Maximum VR	1.1844	0.422	2.795×10^{-5}	1.304
		Maximum Value			Minimum

Although lines of constant speed are the most common representation, there are some advantages to figures showing lines of constant pressure ratio or velocity ratio. To do this, the experimental data was interpolated between the constant speed lines. First, a third degree polynomial curve was fitted to the data shown in Figure 4.3, thereby providing efficiency as a function of velocity ratio for each speed. By substituting equation 4.13 into this polynomial, the efficiency trend at a constant total-to-static pressure ratio could be determined. The result for

pressure ratios of 1.25, 1.5, 2, and 2.5 is shown in Figure 4.6. Six discrete data points for each line in this figure correspond to each of the turbine speeds that were tested. Note that three of the points at a pressure ratio of 2.5 have been extrapolated from the trend line.

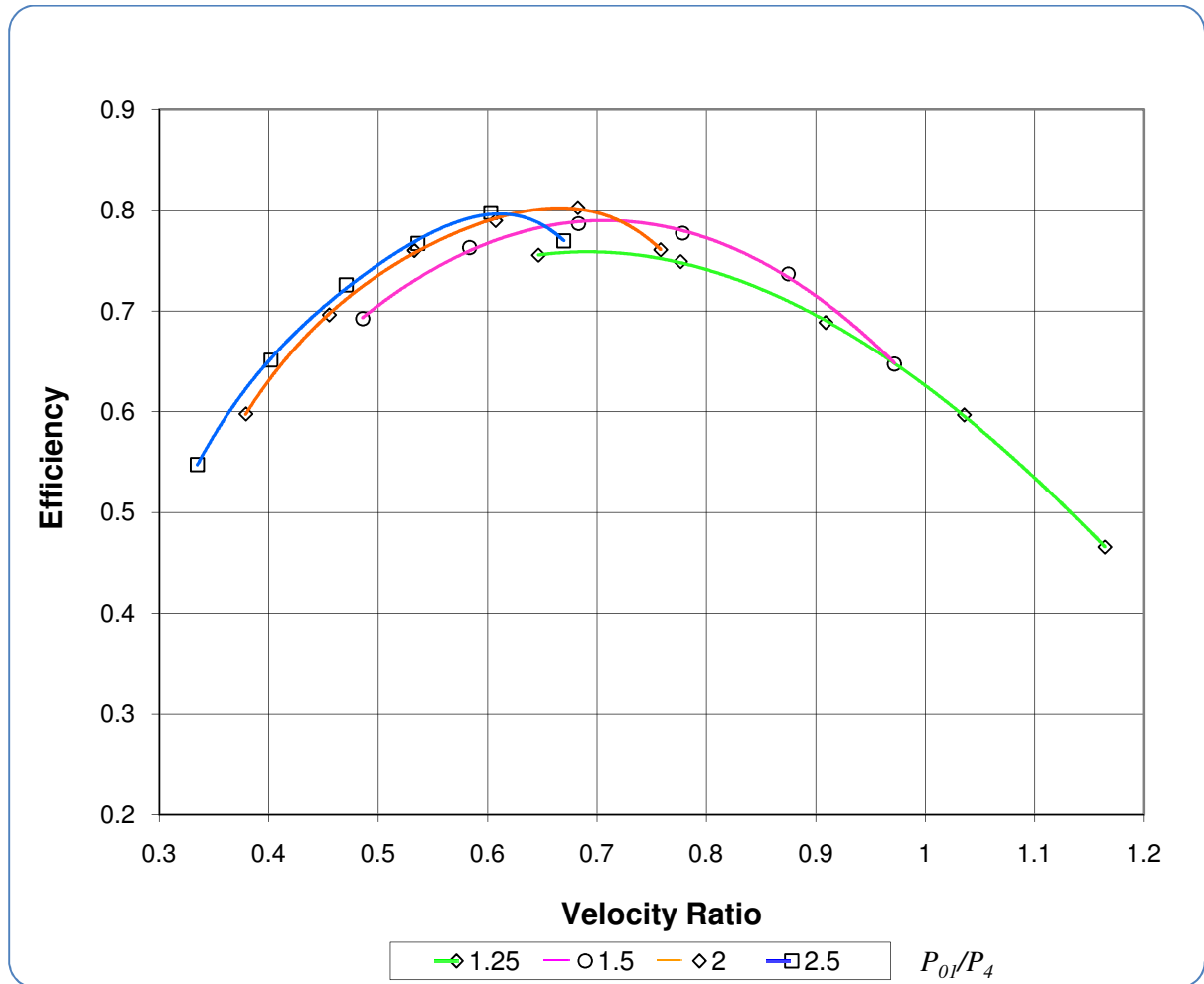


Figure 4.6: Efficiency versus velocity ratio (constant pressure ratio)

Figure 4.6 shows a similar trend as Figure 4.3 where the efficiency peaks at a velocity ratio around 0.65. However, closer examination reveals that the optimum velocity ratio seems to shift from 0.7 to 0.6 as the pressure ratio is increased. This is a useful trend to bear in mind when designing and matching the turbine to the appropriate compressor using the velocity ratio. If, for example, the predominant turbine inlet pressure is 2.5 bar, this information along with the turbine inlet temperature can be used to calculate the isentropic velocity C_{is} . The rotor diameter can then be sized such that, for a given speed, the velocity ratio coincides with the optimum value (~ 0.6) shown in Figure 4.6. Similar conclusions can be drawn from lines of constant velocity ratio shown in

Figure 4.7. These were produced by interpolating in a similar manner as described for the previous figure. Here it is clear that for the turbine operating at a pressure ratio above 2.2, the turbine operates at its most efficient at a velocity ratio closer to 0.6. The interpolated and extrapolated data that corresponds to these figures is given in Tables 4.3 and 4.4.

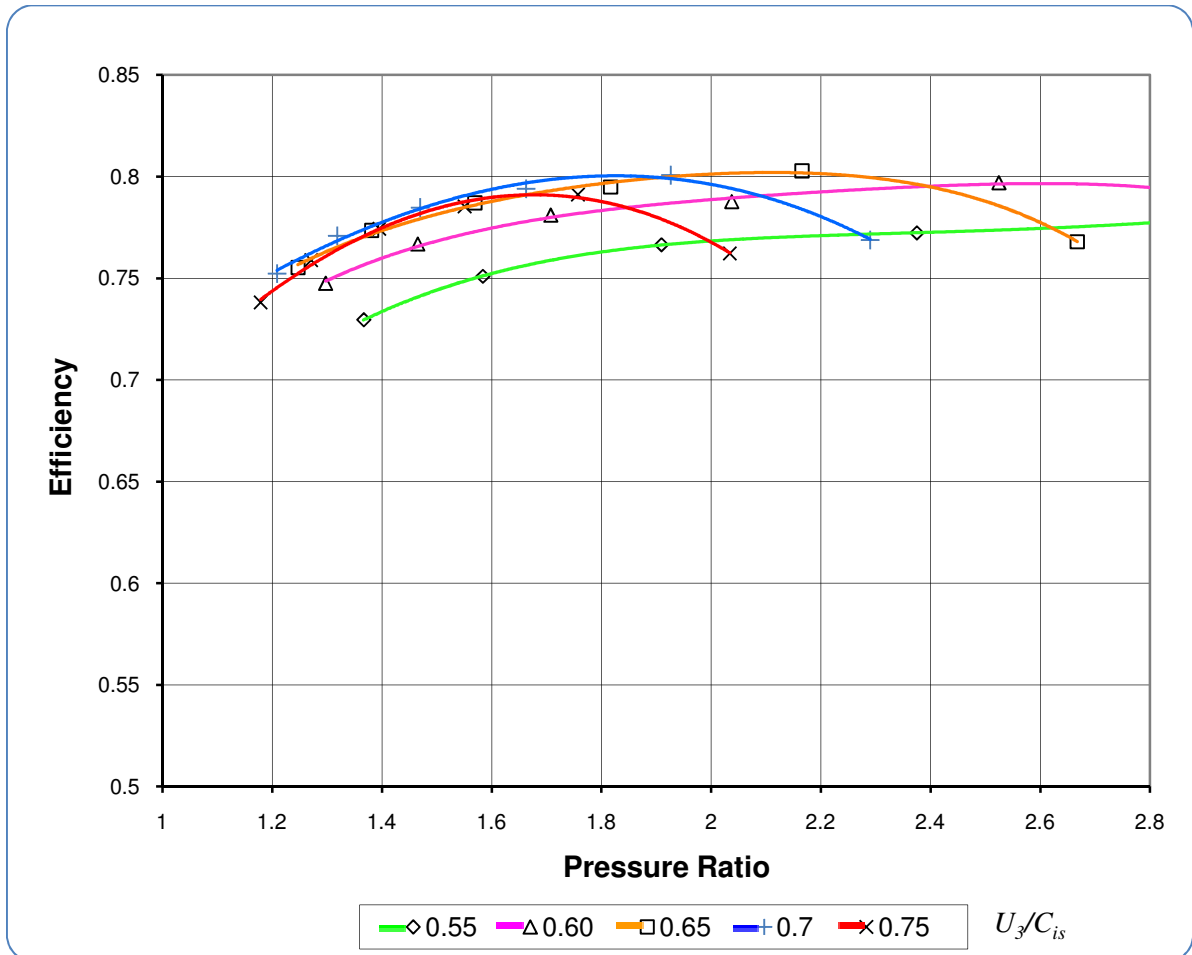


Figure 4.7: Efficiency vs pressure ratio (constant velocity ratio)

Table 4.3: Test data for constant pressure ratio curves

Pressure Ratio	Speed	26.85	32.25	37.75	43.01	48.34	53.70
1.25	Velocity Ratio	0.647	0.776	0.909	1.036	1.164	
	Efficiency	0.755	0.749	0.688	0.597	0.465	
1.5	Velocity Ratio	0.486	0.583	0.683	0.778	0.875	0.972
	Efficiency	0.692	0.763	0.786	0.777	0.737	0.647
2	Velocity Ratio	0.379	0.455	0.533	0.607	0.683	0.758
	Efficiency	0.598	0.696	0.760	0.789	0.802	0.760
2.5	Velocity Ratio	0.335	0.402	0.471	0.536	0.603	0.670
	Efficiency	0.547	0.651	0.726	0.767	0.798	0.769

* Grey areas are calculated from the extrapolation of performance curves.

Table 4.4: Test data for constant velocity ratio curves

Velocity Ratio							
0.55	Pressure Ratio	1.366	1.584	1.910	2.375	3.107	
	Efficiency	0.730	0.751	0.767	0.772	0.783	
0.6	Pressure Ratio	1.298	1.465	1.708	2.038	2.524	
	Efficiency	0.747	0.767	0.781	0.788	0.797	
0.65	Pressure Ratio	1.247	1.381	1.569	1.816	2.165	2.667
	Efficiency	0.755	0.774	0.787	0.795	0.803	0.768
0.7	Pressure Ratio	1.209	1.319	1.470	1.663	1.926	2.290
	Efficiency	0.752	0.771	0.785	0.794	0.801	0.769
0.75	Pressure Ratio	1.179	1.271	1.395	1.550	1.757	2.034
	Efficiency	0.738	0.759	0.774	0.785	0.791	0.762

* Grey areas are calculated from the extrapolation of performance curves.

One further method to represent turbine performance is to plot colour-coded contours of total-to-static efficiency with velocity ratio on the ordinate and pressure ratio on the abscissa as shown in Figure 4.8. Unlike the Figures 4.3-4.7, this depiction of efficiency is not typically available in the literature owing to the somewhat more involved process to produce such a plot. It does, however, provide a very simple way to immediately identify the conditions of optimum efficiency with respect to these two parameters. Note, for example, the wide band of high efficiency operation between velocity ratios 0.55 and 0.75. The usefulness of this type of representation will also become apparent in the forthcoming sections when dealing with unequal and unsteady flows. The plot was created using a Matlab algorithm which performs a linear interpolation between the data points. Since the velocity ratio, pressure ratio and speed parameter are all connected through equation 4.13, lines of constant equivalent speed from 50-100% are also visible on this plot.

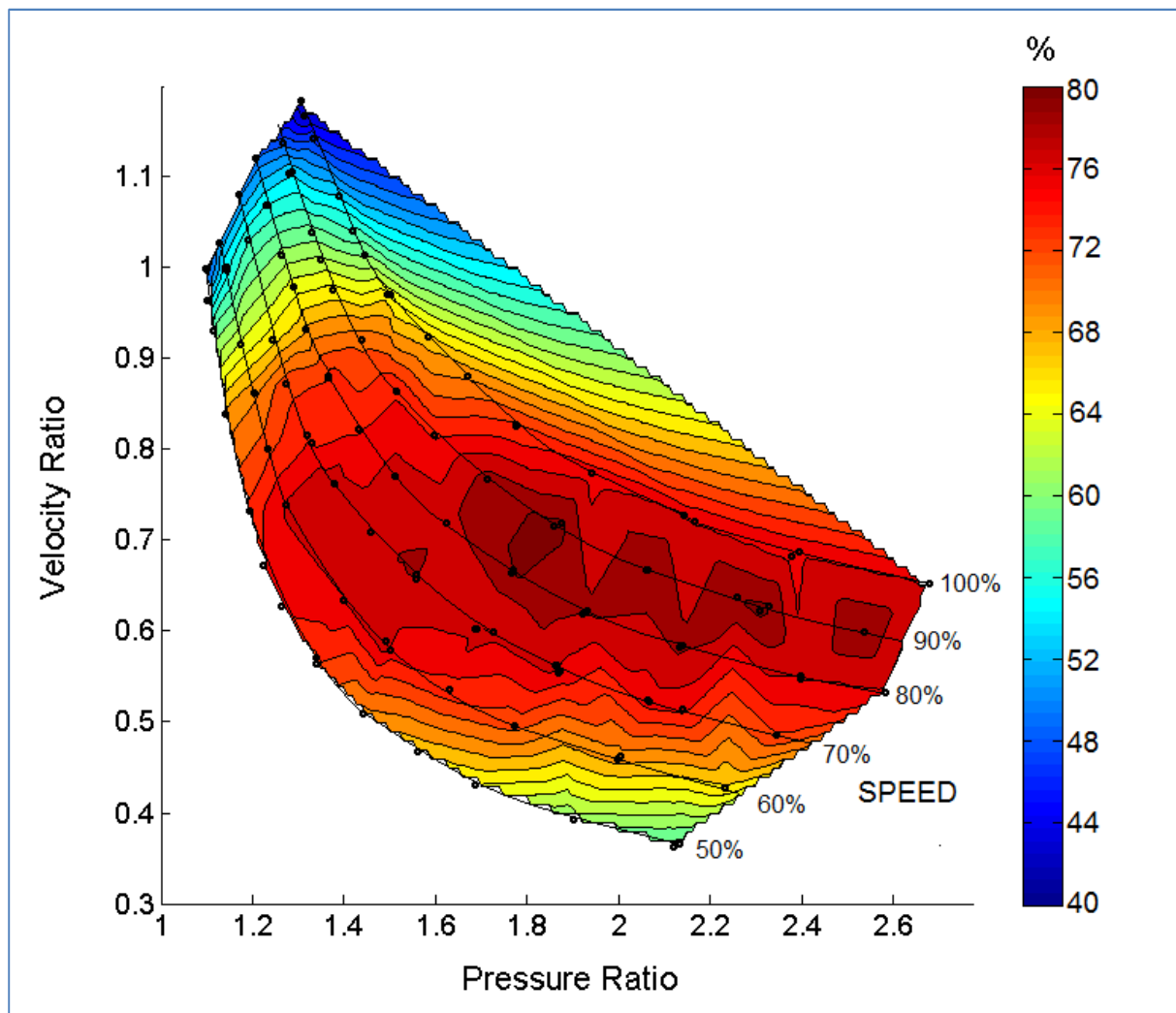


Figure 4.8: Contours of total-to-static efficiency against velocity and pressure ratios.

4.3 Unequal Admission

4.4.1 Introduction

A sectioned view of the double entry ABB volute is shown in Figure 1.6 which illustrates that the double-entry design independently feeds two 180° sectors of nozzle vanes (12 vanes/inlet). These vanes then direct the flow into the spinning rotor via the vaneless space shown in Figure 1.9. Under the equal flow conditions in the previous section, the turbine operates much the same as a single-entry turbine with the exception that there are two volute tongues which could create additional loss. However, if the two flows entering the gas inlet passages are unequal in magnitude, the spinning rotor will be driven by an imbalanced flow around its periphery. In the extreme, when flow is only fed into one entry (partial admission), the volute is essentially mismatched with the rotor and at any instant in time, only half of the rotor experiences mass flow. This encourages the turbine to act as a compressor in half its rotation since the rotational force generated by the flowing section will be dissipated in the non-flowing section. The unequal state can either exist instantaneously in the case of out-of-phase pulsating flow or continually with steady-state, unequal flow. In this section, the performance of the turbine in the latter case will be investigated using a number of unequal flow experimental tests. This will help to build up a complete picture of the turbine behaviour for a wide range of steady-state conditions.

4.4.2 Performance Parameters

If the flows introduced into each entry are equal (as in the last section), all of the standard dimensionless and pseudo-dimensionless performance parameters such as efficiency, pressure ratio, mass parameter and velocity ratio are identical to a single entry turbine. Once the flow conditions between entries differ, these values must be re-evaluated to account for each flow appropriately. Some of the typical performance parameters listed in table 3.1 become somewhat ambiguous when applied to the unequal admission case. The difficulty stems from trying to define a single inlet pressure that is representative of the entire turbine. Note from Figure 2.5, that some authors such as Pischinger and Wunsche [5] have used an averaged pressure ratio (equation 2.3) to plot their results for a twin entry-turbine. While this is, in itself, a perfectly legitimate dimensionless number, the difficulty is whether this mean pressure ratio is comparable to the corresponding equal admission pressure ratio. This approach is likely more appropriate for a twin-entry turbine where

there is a significant amount of flow interaction in the space between the dividing wall and the entry to the rotor. The resulting pressure field is then introduced around the entire circumference of the rotor. However, in the case of a double entry turbine, each flow is fed into the turbine wheel largely independently of each other. Half of the turbine inlet will then be at a different pressure than the other half, making a mean inlet pressure inappropriate for evaluating a single turbine pressure ratio. Therefore, it seems correct to treat the two entries in a double-entry design separately and calculate independent pressure ratios for each entry as follows:

$$PR_{inner} = \frac{(P_{01})_{inner}}{P_4} \quad PR_{outer} = \frac{(P_{01})_{outer}}{P_4} \quad (4.14)$$

where the *inner* and *outer* subscripts refer to the lower and upper inlets respectively. Similarly, the value of mass parameter for each inlet can be calculated separately as in Equation 4.15. However, the mass parameter is a mixture of *intensive* quantities (pressure and temperature) and mass flow which is an *extensive* quantity. Thus, to compare the values in equation 4.15 to equal admission values, the equal admission mass parameters must also be evaluated on a per-entry basis.

$$MP_{inner} = \left(\frac{\dot{m}\sqrt{T_{01}}}{P_{01}} \right)_{inner} \quad MP_{outer} = \left(\frac{\dot{m}\sqrt{T_{01}}}{P_{01}} \right)_{outer} \quad (4.15)$$

An alternative method of representing the swallowing capacity of a dual entry turbine is to treat each turbine entry as a simple adiabatic nozzle as suggested by the discussion in section 4.2. This approach is especially useful to the modeller where a simple nozzle treatment is often used as a boundary condition to a 1-D wave action code. For a double-entry turbine, a pair of nozzles is required, each with an area calculated to provide an isentropic expansion of the gas (calculated from the pressure ratio) for the measured mass flow in the corresponding entry. Equation 2.1 provides the basic concept of an effective flow area A_{eff} as given in the work of Pischinger and Wunsche [5]. To expand on this, one must recall the fundamental compressible flow equations for an ideal convergent nozzle. First, assume that the measured pressure ratio across the turbine is applied across this ideal nozzle. This pressure ratio is then used to check inequality 4.16 and if it is *true*, the nozzle will be subsonic, and equation 4.17 must be used to calculate the effective area A_{eff} needed to swallow the measured mass flow $\dot{m}_1$.

$$\frac{P_{01}}{P_4} < \left(\frac{\gamma + 1}{2} \right)^{\frac{\gamma}{\gamma - 1}} \quad (4.16)$$

$$A_{eff} = \frac{\dot{m}_1}{P_{01}} \cdot \frac{\sqrt{RT_{01}}}{\sqrt{\frac{2\gamma}{\gamma-1} \left[\left(\frac{P_{01}}{P_4} \right)^{\frac{-2}{\gamma}} - \left(\frac{P_{01}}{P_4} \right)^{\frac{-(\gamma+1)}{\gamma}} \right]}} \quad (4.17)$$

If inequality 4.16 is *false*, the ideal convergent nozzle is choked and equation 4.18 must be used to calculate the effective area required to pass the measured mass flow.

$$A_{eff} = \frac{\dot{m}_1/P_{01}}{\left(\frac{2}{\gamma+1} \right)^{\frac{1}{\gamma-1}}} \sqrt{\frac{RT_{01}}{\frac{2\gamma}{\gamma+1}}} \quad (4.18)$$

Since energy is also an extensive quantity, it is additive for a series of different inlet conditions. Thus, the approach used to calculate total-to-static efficiency and velocity ratio of the turbine is much more apparent since the isentropic energy at each volute entry can be simply summed. The measurement of inlet rotor tip speed U_3 and output power $\dot{W}_{act}$ will be unaffected from the definitions in equations 4.9 and 4.8. The isentropic power can be calculated in a similar way as equation 4.2 and summed to give a total inlet isentropic energy as follows:

$$\begin{aligned} \dot{W}_{T,is} &= \dot{m} \left[\int_{4,is}^m c_p dT_s + \frac{1}{2} U_1^2 \right] \\ \dot{W}_{T,is} &= \dot{m}_{inner} \left(C + D + \frac{1}{2} U_1^2 \right)_{inner} + \dot{m}_{outer} \left(C + D + \frac{1}{2} U_1^2 \right)_{outer} \end{aligned} \quad (4.19)$$

$$C = A_0 T_1 + \frac{A_1 T_1^2}{2} + \frac{A_2 T_1^3}{3} + \frac{A_3 T_1^4}{4} + \frac{A_4 T_1^5}{5} + \frac{A_5 T_1^6}{6} \quad (4.20)$$

$$D = -A_0 T_{4,is} - \frac{A_1 T_{4,is}^2}{2} - \frac{A_2 T_{4,is}^3}{3} - \frac{A_3 T_{4,is}^4}{4} - \frac{A_4 T_{4,is}^5}{5} - \frac{A_5 T_{4,is}^6}{6} \quad (4.21)$$

Where D , C and $\frac{1}{2} U_1^2$ are evaluated separately using measurements taken from the inner and outer limbs which feed the upper and lower entries of the volute. The evaluation of total-to-static turbine efficiency is then simply a matter of summing the inlet isentropic energy.

$$\eta_{t-s,total} = \frac{\dot{W}_{act}}{\dot{W}_{is,inner} + \dot{W}_{is,outer}} \quad (4.22)$$

Lastly, equation 4.23 provides the definition for the overall velocity ratio under equal or unequal admission.

$$\frac{U_3}{C_{is}} = \frac{\pi \cdot d_2 N}{\sqrt{2 \left(\frac{\dot{W}_{is,inner} + \dot{W}_{is,outer}}{\dot{m}_{total}} \right)}} \quad (4.23)$$

This definition of velocity ratio, while perfectly correct, does lose some of the meaning discussed in Section 4.3.4. For a single entry or equal admission flow, it can be linked to the relative inlet flow angle of the flow, and when this is well matched to the rotor design, the turbine will operate efficiently. Equation 4.23, however, combines two different isentropic energies in the valuation of velocity ratio. While this should be suitable for comparison and matching purposes, it is no longer correlated to a single gas inlet angle entering the rotor since it is actually a combination of two flows with different driving conditions.

It is also useful to develop a non-dimensional parameter to assess the “amount” or severity of unequal admission. The two most obvious choices are the ratio of inlet mass flows and the ratio of inlet total pressures, both of which can be found in the available literature. For example, Dale and Watson [9] used both parameters – the ratio of mass flows when comparing efficiency and the ratio of pressures when comparing mass parameters. For the purposes of presenting the data in this thesis, the ratio of total pressures at inlet (RP_{inlet}) as provided in equation 4.24 was sufficient.

$$RP_{inlet} = \frac{(P_{01})_{inner}}{(P_{01})_{outer}} = \frac{PR_{inner}}{PR_{outer}} \quad (4.24)$$

4.4.3 Graphical Representation

Unlike equal admission, there does not appear to be a standard method of plotting the performance of a multiple entry turbine under unequal admission conditions. The difficulty stems from trying to represent the large number of possible flow combinations in a double-entry turbine on a single plot. Dale and Watson [9] tried to maintain the approach of plotting equal admission

performance (figures 4.3 and 4.5) to the unequal admission data. Figure 4.9 demonstrates this approach with two plots of the turbine efficiency versus the velocity ratio. Each plot represents a single non-dimensional speed, and multiple lines are shown which represent the different amounts of unequal admission assessed on the basis of the ratio of inlet mass flows. A similar approach is used to represent the mass flow characteristics in Figure 4.10. To include unequal admission in such a plot, the pressure ratios across each inlet were used for different functions. The pressure ratio across the first inlet is plotted on the ordinate, and the pressure ratio of the second inlet is reserved for the lines showing the degree of unequal admission. The result is a figure that is quite confusing to decode as the authors try to use the more familiar full-admission representation.

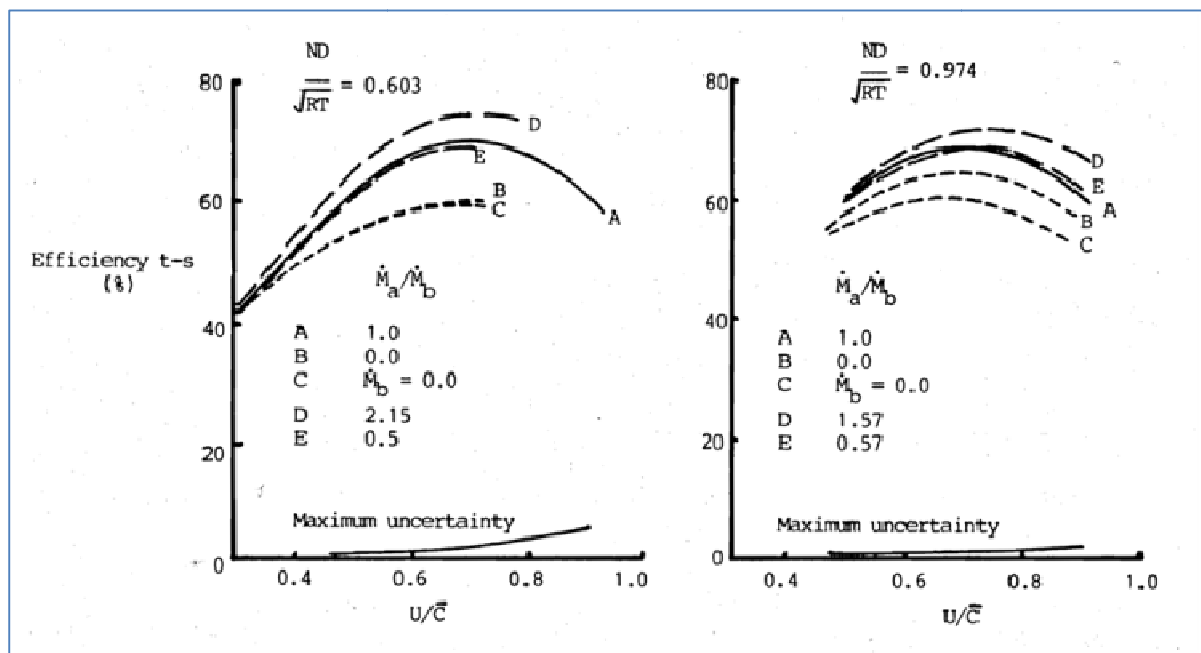


Figure 4.9: Total-to-static efficiency against velocity ratio –equal and unequal admission [9]

Pischinger and Wunsche [5] chose a different approach to plotting the unequal admission results. As shown in Figure 2.6, the efficiency of the turbine was plotted against the ratio of inlet pressures (equation 4.24). A log scale is used on the abscissa to allow a clear indication of the symmetry of behaviour between the two turbine entries. Similarly, the swallowing characteristics of the turbine can be demonstrated by plotting the effective flow areas of each inlet against the ratio of inlet pressures (Figure 2.5). Pischinger and Wunsche [5] chose to show lines of constant *average* pressure ratio. As discussed in the previous section, this parameter is less appropriate for a double-entry design due to the greater degree of isolation between inlets. However, this approach is much clearer and easier to understand than the method of Dale and Watson. Thus, this type of plot will be used to present the unequal admission performance in this thesis.

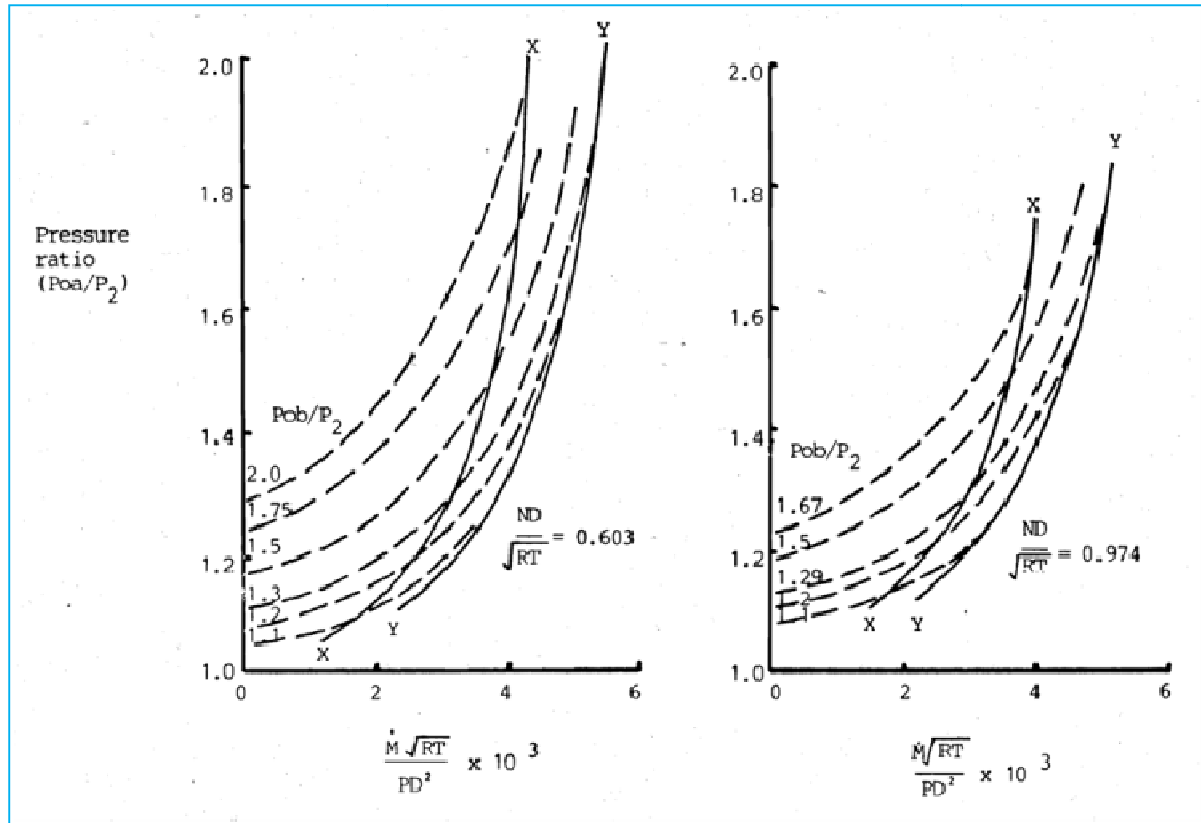


Figure 4.10: Pressure ratio against non-dimensional mass flow—equal and unequal admission [9]

Plotting lines on a 2-D plot to demonstrate behaviour of a dependent variable such as efficiency for a dual-entry turbine requires a minimum of three independent variables (inner pressure ratio, outer pressure ratio and velocity ratio or speed). This not only makes it complicated to visualize but also means that multiple plots are necessary to present all the data. Therefore, a new approach will be developed in this section to be able to simplify these plots and ultimately produce a single plot which shows the entire performance of the turbine on a diagram. The aim is to be able to look at a plot of turbine performance and immediately get a picture of the level of optimum performance and the conditions under which this optimum performance occurs.

4.4.4 Experimental Method

In order to execute the unequal admission testing in an organized manner, one of the performance parameters must be held constant while the others are varied. For full admission testing, the *speed parameter* was held constant since this was the simplest approach and served the purpose well. However, for unequal admission testing, the aim was to compare the behaviour of the

turbine under unequal admission conditions to a baseline (equal) condition. Thus, by maintaining a constant *velocity ratio* that matched the peak of the full-admission performance, the deficit attributable to the degree of unequal admission could be ascertained. Maintaining a constant parameter such as velocity ratio, however, was more difficult to test since live values of the pressure ratio and velocity ratio were unavailable during testing due to the reliance on the Scanivalve pressure measurement system. Using a full-admission case as a starting point, the main flow valve, two limb valves and dynamometer gap all had to be adjusted to achieve the desired test condition. Thus, multiple data logging and calculation was often necessary before the particular set of inlet conditions was found that resulted in the desired performance parameters.

4.4.5 Results and Discussions

Unequal Test Conditions

Table 4.5 lists the range of dimensionless parameters selected to test the unequal admission performance of the double-entry turbine. As mentioned in the previous section, the velocity ratio calculated from equation 4.23 was held at three constant values (cases 1-3) to allow ease of comparison with equal admission performance. In addition, this data will also be used in the unsteady performance analysis of Chapter 6. Therefore, since the turbine can experience a wide variation of flow conditions during a single unsteady pulse, in order to make a full comparison between steady and unsteady performance, the scope of available steady-flow test data must be broad enough to produce a reliable interpolation with little required extrapolation. Therefore, additional unequal admission data (beyond the optimum velocity ratio of 0.65) were tested at velocity ratios equal to 0.5 and 0.8. For cases 1 and 2, the pressure ratio across one inlet was held at an approximately constant value of 2.0. The flow in the second limb could then be varied from zero flow (referred to as partial admission) to the maximum pressure ratio achievable with full dynamometer loading. In this way, the loss of efficiency compared to the peak condition under full flow could be noted. To test the symmetry of the performance between the two entries, the process was repeated by varying the pressure at the first inlet and maintaining the pressure at the second. At velocity ratio of 0.8 (case 3), the pressure ratio range was limited to below 1.8 by the maximum speed of the dynamometer. Thus, one entry was maintained at a constant pressure ratio of 1.5, instead of 2.0.

Table 4.5: Unequal admission test conditions

Unequal Case	Velocity Ratio (average of the data points)	Inner Pressure Ratio	Outer Pressure ratio
1	0.506 $\pm$ 0.0058	2.04 $\pm$ 0.0346	1.176 $\rightarrow$ 2.670
	0.504 $\pm$ 0.0073	1.166 $\rightarrow$ 2.687	2.07 $\pm$ 0.021
2	0.653 $\pm$ 0.00356	2.04 $\pm$ 0.013	1.135 $\rightarrow$ 2.648
	0.652 $\pm$ 0.00287	1.143 $\rightarrow$ 2.611	2.02 $\pm$ 0.011
3	0.800 $\pm$ 0.0106	1.507 $\pm$ 0.015	1.099 $\rightarrow$ 1.771
	0.802 $\pm$ 0.007	1.097 $\rightarrow$ 1.785	1.510 $\pm$ 0.015

* Grey areas are the pressure ratios across the non-flowing inlet for partial admission

Since this data set is made up of a number of separate tests, and owing to the trial-and-error approach to obtaining the desired operating conditions, the velocity ratio and pressure ratio values that were held 'constant' were not always precisely consistent. Table 4.5 therefore also shows the 95% confidence intervals for the parameters that were held approximately constant. The confidence interval that describes the uncertainty in the precision of these multiple experiments is equal to $\pm 1.96 \cdot \sigma / \sqrt{n}$, where σ is the standard deviation and n is the number of samples in the data set.

Unequal Admission Efficiency

Figures 4.11 and 4.12 demonstrate the total-to-static efficiency of the turbine operating under a variety of unequal admission conditions ranging from full flow to partial admission. The first figure shows the turbine operating under a constant velocity ratio equal to 0.65 that matches the peak operation of the turbine in the equal admission condition. The point of intersection of the two lines that represents equal admission therefore reveals this optimal case. The efficiency is plotted against the ratio of inlet pressures (equation 4.24) on a logarithmic scale to demonstrate symmetry between the inner and outer entries. The partial admission condition where one of the inlets is completely shut off, corresponds to the lower extremities of these lines. Interestingly, these partial admission tests result in a ratio of pressures (RP_{inlet}) greater than 0.5 and less than 2.0. Since one of the inlets is being held at a constant pressure ratio equal to 2.0, this means that the pressure in the non-flowing inlet is always above atmospheric despite being closed. There are two possible

explanations for a pressure ratio above unity across the non-flowing entry. First, the pressure in this entry could be caused by the centrifugal head created by the rotor acting as a compressor as it rotates through the stagnant sector. To test this, compare the partial admission results in cases 1 and 2. Since case 1 is rotating approximately 1,000 rpm slower than case 2, one would expect that this would create less centrifugal head in the non-flowing entry. However, case 1 actually results in a higher pressure ratio across the stagnant inlet, thus suggesting that this effect is not the leading cause of this phenomenon (grey highlights in Table 4.5). The second possible explanation simply stems from the observation that a large difference in the pressures in the volute passages could lead to a pressure exchange via the interspace as the two inlets seek to equalize around the tongues. This explanation is supported by the rise in the pressure ratio across the non-flowing inlet from case 3 to case 2. Since the pressure ratio in the flowing inlet is less in the former case, this should produce less pressure rise in the stagnant entry. When the pressure ratio is increased to 2.0, the pressure in the stagnant entry also rises. This is the first indication of some interaction between the two inlets despite a design intent on feeding separate sections of the rotor inlet.

Table 4.6: Unequal Admission Performance

Unequal Case	Velocity Ratio	Location	PR_{inner}	PR_{outer}	η_{t-s}	A_{eff}
1	0.5	Minimum	1.17	1.18	0.60	0
		Equal	2.06	2.06	0.76	0.00056
		Maximum	2.69	2.67	0.76	0.00063
2	0.65	Minimum	1.14	1.14	0.52	0
		Equal	2.06	2.06	0.80	0.00056
		Maximum	2.61	2.65	0.83	0.00062
3	0.8	Minimum	1.10	1.10	0.45	0
		Equal	1.52	1.52	0.77	0.00049
		Maximum	1.77	1.79	0.79	0.00056

The significant loss in turbine performance due to unequal inlet flows is also clear from Figure 4.11 and 4.12. Even at the optimum equal admission velocity ratio of 0.65, Table 4.6 reveals a loss of 28 percentage points of efficiency from full to partial admission. The shape of the three plots suggests that the sensitivity to unequal flows increases as velocity ratio increases. The lowest velocity ratio tested ($U_3/C_{is} = 0.5$) results in a relatively flat curve, whereas the highest ($U_3/C_{is} = 0.8$) demonstrates a steep penalty in turbine performance due to unequal conditions. It is also apparent from the symmetrical nature of the plots that the losses introduced by unequal admission are, in the main, independent of which inlet condition is varied but clearly correlated to the ratio of pressures between them. This finding contrasts the behaviour of a meridionally-divided volute which rarely produces similar symmetric behaviour under unequal admission conditions [5]. This result speaks to the precise design of the ABB double-entry volute in attempting to deliver similar flow characteristics from each of the entries.

The work by Pischinger and Wunsche [5] suggests that unequal conditions in a twin-entry design does not result in as great a loss in performance as the double-entry design. Consider the extreme case where flow is only fed into one entry (partial admission) leaving the second limb stagnant. In the twin-entry design, the entire circumference of the rotor inlet will still be fed by the single entry that is flowing. The asymmetric efficiency characteristics mentioned above are caused by the two inlets feeding different sections of the meridional inlet area (hub-side or shroud-side). Thus, the response to unequal conditions will differ depending on whether the shroud-side or hub-side inlet is flowing. In contrast, the double-entry volute will *not* feed the entire circumference of the rotor when partially admitted. From a global perspective, it is easy to see that since only half of the rotor circumference is fed, the volute is completely mismatched to the rotor, and significant loss can result. To move from this global perspective, consider a single blade passage as it rotates in the unequally admitted, double-entry design. This blade passage will be subjected to an abrupt change in flow velocity and direction as it rotates from one 180° sector to the next. In fact, this abrupt change in flow produces an unsteady flow whose amplitude will be a function of the ratio of inlet pressure (RP_{inlet}) and whose frequency will depend on the speed of rotor rotation. The frequency of this pulsating will be quite high, however. With the turbine operating at 50,000rpm, the frequency due to unequal admission will be 1667 Hz.

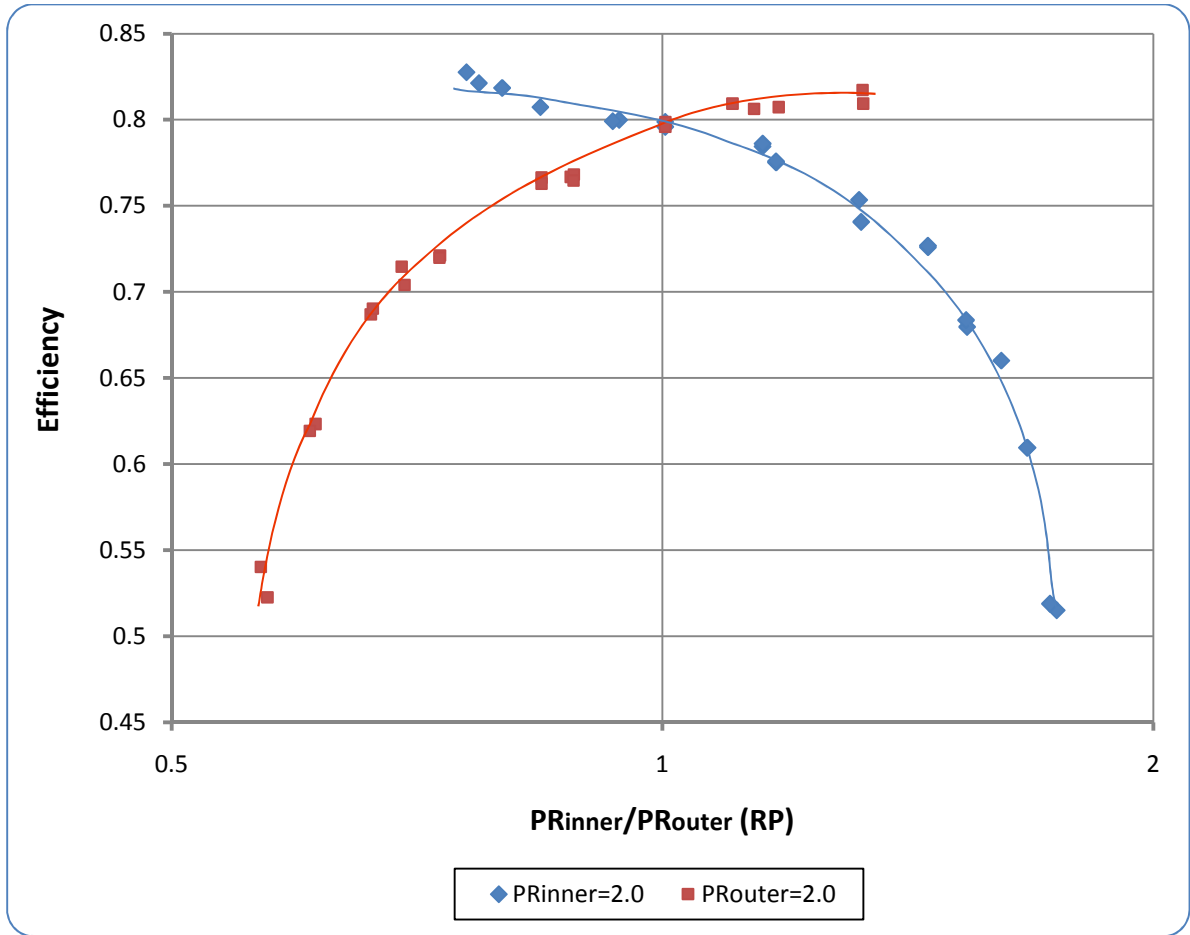


Figure 4.11: Total-to-static efficiency versus inlet ratio of pressures ($U_3/C_{is}=0.65$)

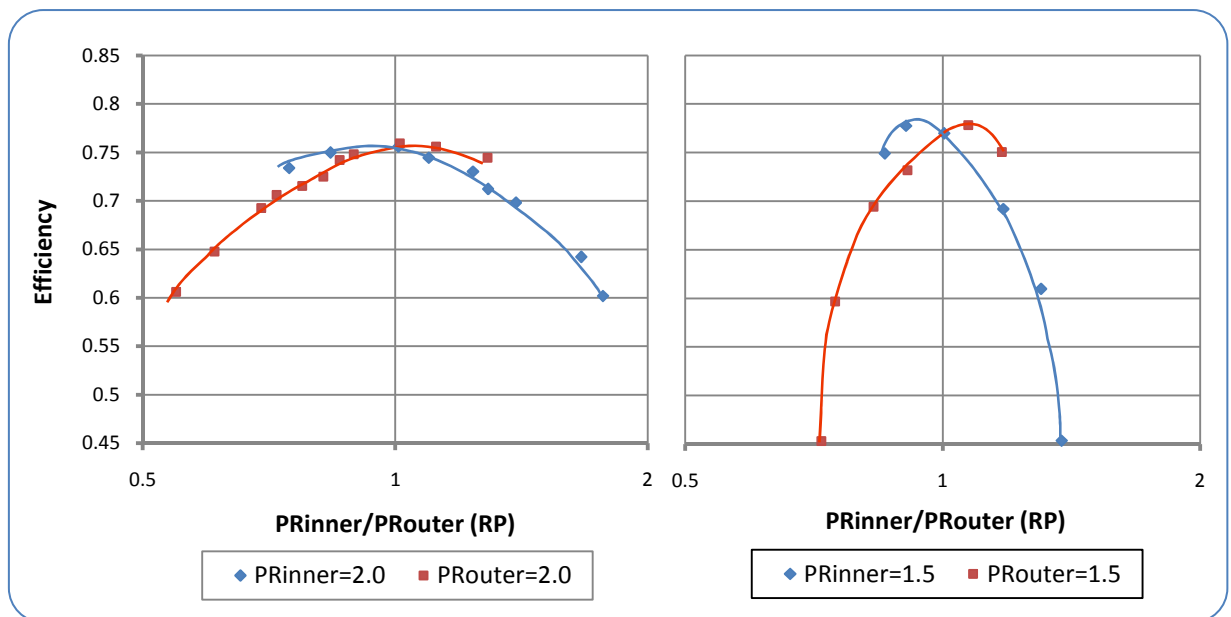


Figure 4.12: Total-to-static efficiency versus inlet ratio of pressures
 (Left figure: $U_3/C_{is}=0.5$; Right figure: $U_3/C_{is}=0.8$)

Effective Flow Area

The values of the effective flow area of the two turbine entries calculated using equations 4.16 to 4.18 are shown in Figures 4.13, 4.14 and 4.15. For each of the three velocity ratios listed in Table 4.6, there were two test cases per inlet -- one where the inlet pressure was held constant and a second where it was varied. This resulted in a total of twelve data sets (2 inlets $\times$ 2 PR test cases $\times$ 3 velocity ratios) as demonstrated by the twelve lines in these figures. Figure 4.13 shows the effective flow area of the turbine inlet where the flow rate is being varied (pressure ratio range between 1.14 and 2.69). Here the turbine is operating at a constant velocity ratio of 0.5 or 0.65, and the second inlet is being held at a constant pressure ratio of 2.0. As expected, the effective flow area ranges from 0 m² when there is no flow, to 0.00063 m² at the highest pressure ratio. There are two main points of interest in this figure. First, the two lines that represent the two inlets are symmetrical about the equal admission intersection point. This shows that, like the loss generation in the previous section, the swallowing characteristics of the two inlets show an almost identical response to unequal admission conditions. This differs from the behaviour of the twin-entry design for the same reasons as outlined in the previous section. The second point of interest in Figure 4.13 is the slight difference between the effective area characteristics for the turbine operating at two different velocity ratios. For a given position on the x-axis, the inlet pressure ratios will be the same since the second inlet in cases 1 and 2 is held at a constant pressure ratio equal to 2.0. Thus, the increase in effective area must be due to a rise in the mass flow as the turbine velocity ratio drops from 0.65 to 0.5. This effect is caused by the variation in centrifugal head caused by the motion of the rotor blades. As speed is decreased, the centrifugal pressure field relents, thus allowing a greater flow of gas for a given pressure ratio. This effect has also been observed in Figure 4.5 where a rise in the mass flow parameter resulted from a decrease in turbine speed.

Figure 4.14 shows the variation in the effective area of the turbine passage with a constant pressure ratio of 2.0. Since along each line in this figure the velocity ratio and the expansion ratio are maintained at a single value, one would expect no change in the swallowing capacity of this inlet. The observation that these lines are *not* horizontal therefore demonstrates a change in the mass flow through this entry as a result of a change in pressure in the second entry. This is a significant discovery as it shows conclusively that there must be a significant amount of flow interaction between the inlets when fed with unequal flows. The deviation of this line from horizontal represents the influence that the flow in the varying entry has on the entry with a constant pressure ratio. Compared to full flow, when the pressure in one entry is low, the second entry can swallow more mass, and when it is high, the second entry swallows less. While a detailed analysis of the

causes behind this effect can only be made through flow field analysis using 3-D computational fluid dynamics, some initial observations can be suggested here. First, note that each volute passage from the turbine inlet to the exit of the nozzles (Figure 1.6) is completely isolated. The two flows only have a chance to interact after exiting the nozzle. Therefore, in order for the flow rate to change in a isolated passage (where the upstream pressure is constant), the average nozzle exit pressure in the inter-space volume must have changed due to unequal conditions.

Figure 4.15 shows the effective area for the varying and constant pressure entries with the turbine operating at $U_3/C_{is} = 0.8$. Since one of the inlets was held at a constant pressure ratio of 1.5, the values of effective area will differ from figure 4.13, but the trend is clearly similar. The symmetry in this data confirms that the two turbine inlets display similar flow characteristics over the range of unequal flows. The RHS plot in Figure 4.15 also follows the trends observed in figure 4.14 showing that despite a constant pressure ratio, the swallowing capacity will be changed by a varying pressure in the second inlet.

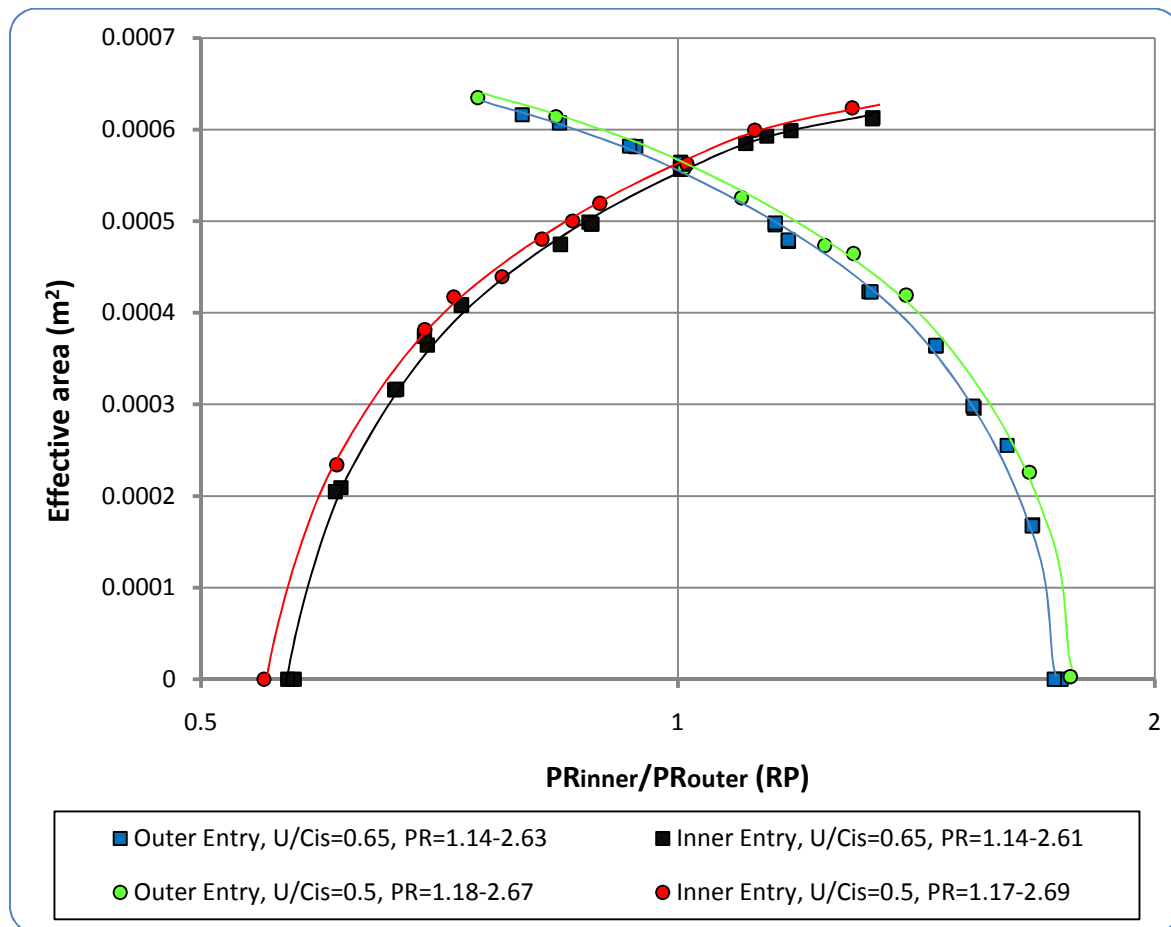


Figure 4.13: Effective area of the inlet with **varying** pressure versus inlet ratio of pressures ($U_3/C_{is}=0.65$ and 0.5)

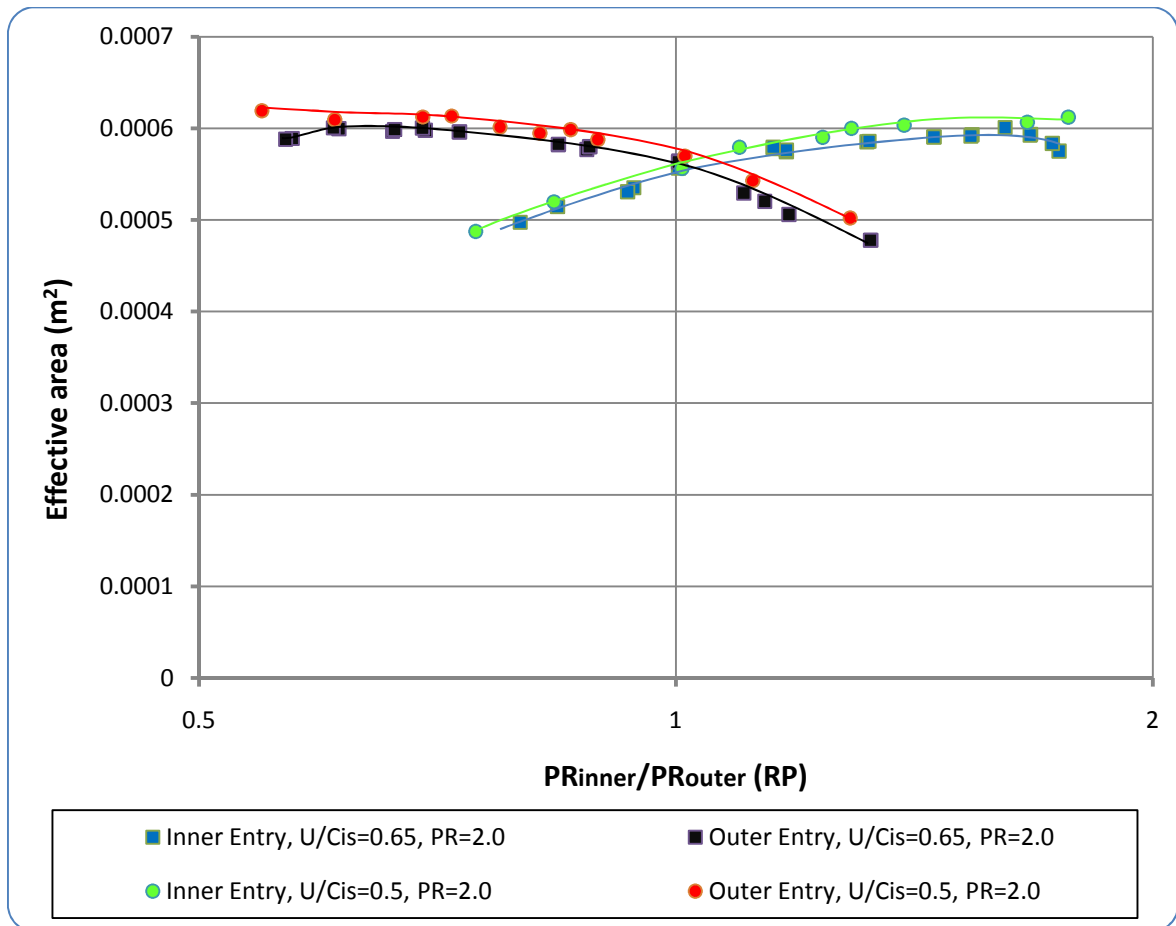


Figure 4.14: Effective area of the inlet with **constant** pressure versus inlet ratio of pressures ($U_3/C_{is}=0.65$ and 0.5)

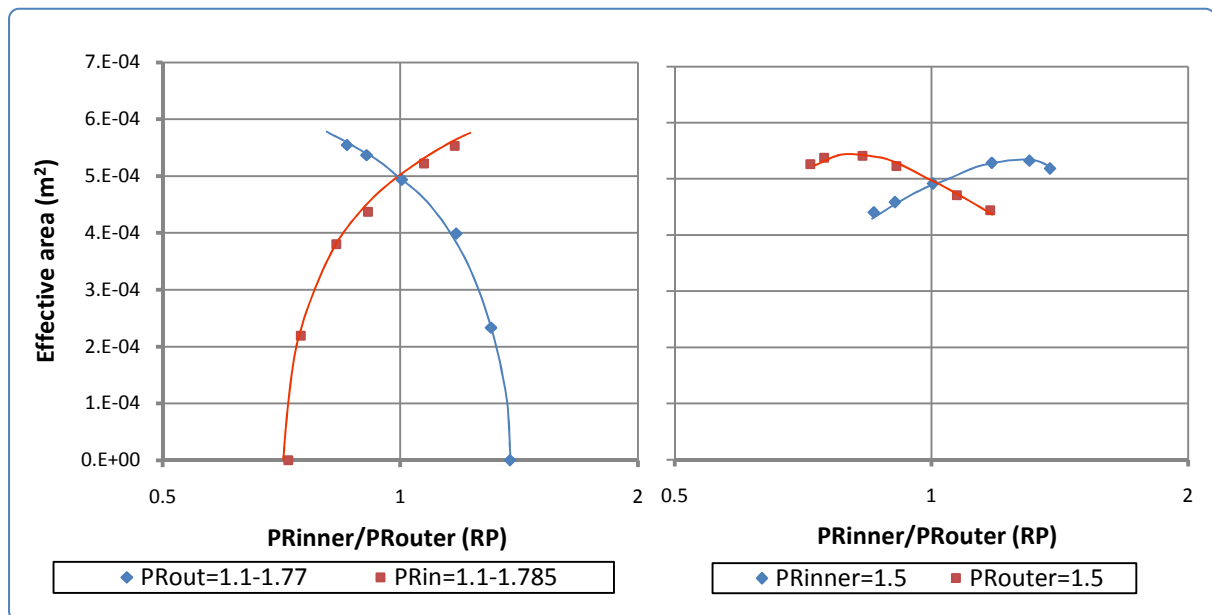


Figure 4.15: Effective area versus inlet ratio of pressures at $U_3/C_{is}=0.8$
(Left figure: Inlet with varying pressure; Right figure: Inlet with constant pressure)

Contour Plotting

Like the equal admission data presented in Figure 4.8, it would be useful to collate the unequal admission performance and interpolate between the data to produce 2-D contours. However, to fix the operating point of a turbine driven by *unequal* flows, the expansion across each of the inlets must be accounted for. Thus, three parameters are necessary to describe a dependent variable such as efficiency: the two pressure ratios and the velocity ratio (or speed parameter). Since testing was performed at a constant velocity ratio, this invites the creation of three 2-D, interpolated plots that show how the performance varies with the inner and outer expansion ratios. These plots are shown in figures 4.16, 4.17 and 4.18 for velocity ratios equal to 0.5, 0.65 and 0.8 respectively.

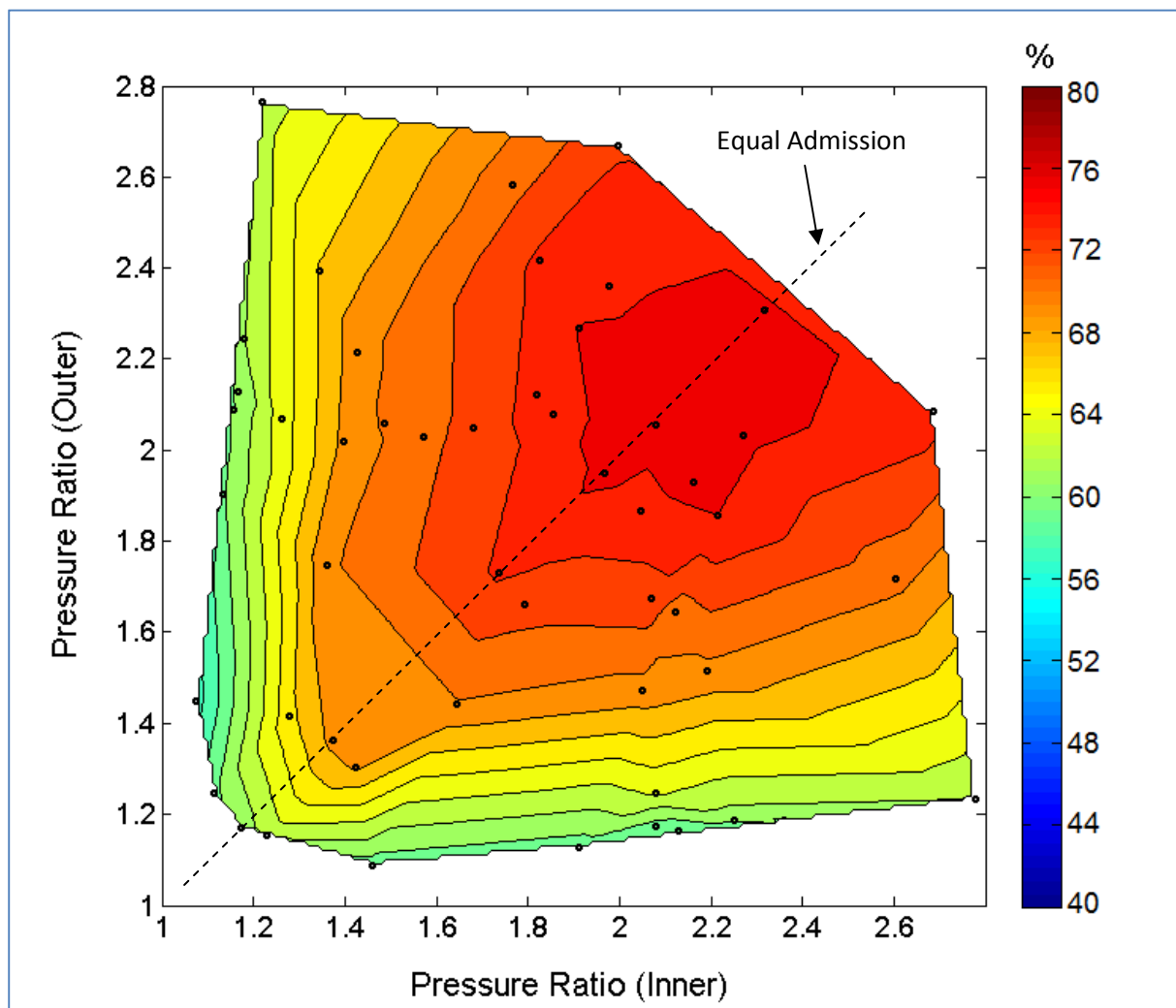


Figure 4.16: Total-to-static efficiency versus the pressure ratios across inner and outer inlets ($U_3/C_{is}=0.5$)

This approach to plotting unequal performance is unique among the available literature and allows quick and easy interpretation in contrast to some of the other representations. To provide a more reliable interpolation, additional unequal test data to that provided in Figures 4.13-4.15 is included in these plots at various combinations of pressure ratios. These plots also include equal admission data along the diagonal (shown as a broken line in Figure 4.16). These diagonals correspond to the constant velocity ratio lines shown in Figure 4.7. Moving off this diagonal represents a greater pressure difference between the entries, leading to an increased penalty in the turbine performance. Since all these figures use the same colour map to represent turbine efficiency, one can clearly observe areas of optimum turbine performance.

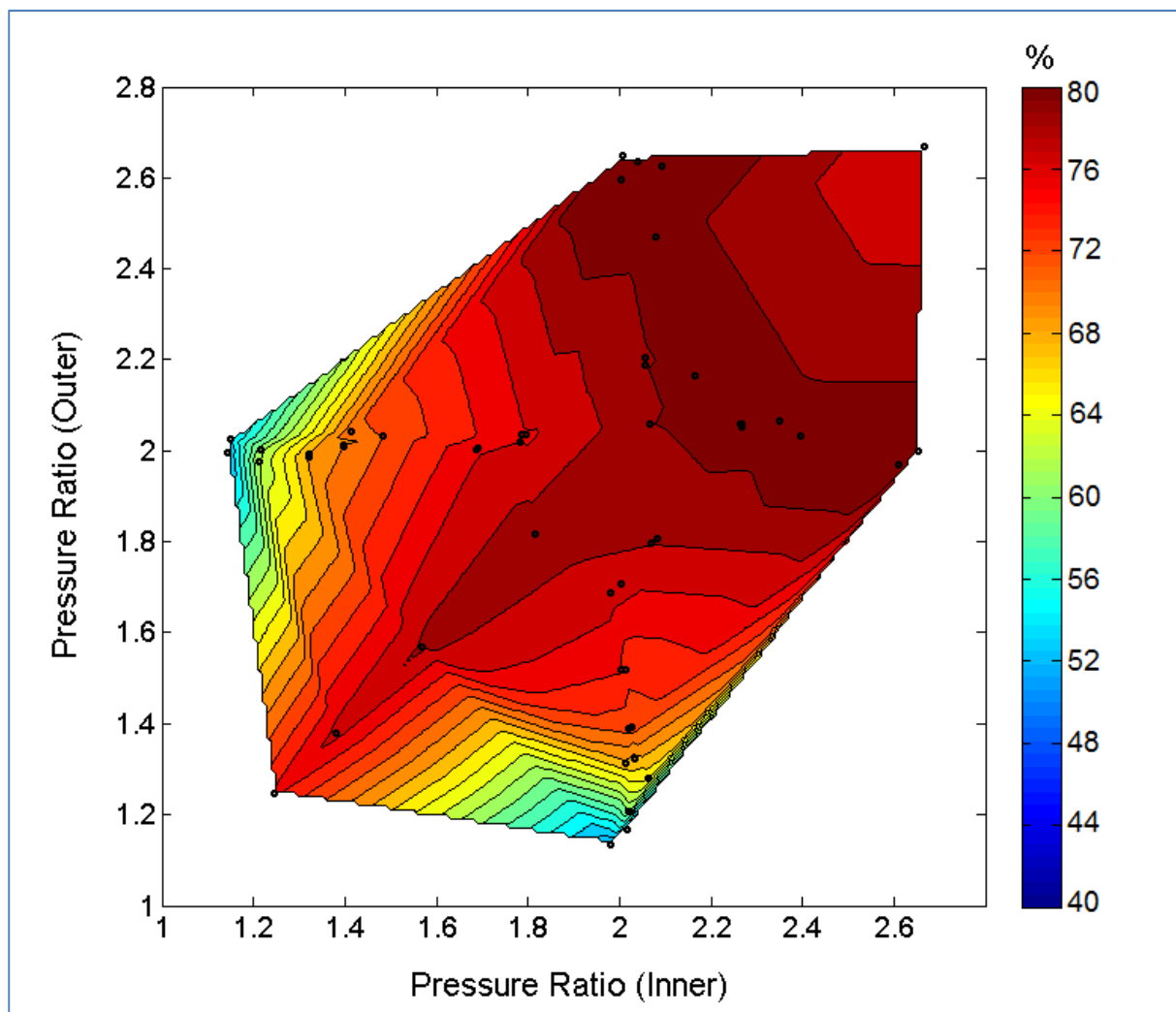


Figure 4.17: Total-to-static efficiency versus the pressure ratios across inner and outer inlets
($U_3/C_{is}=0.65$)

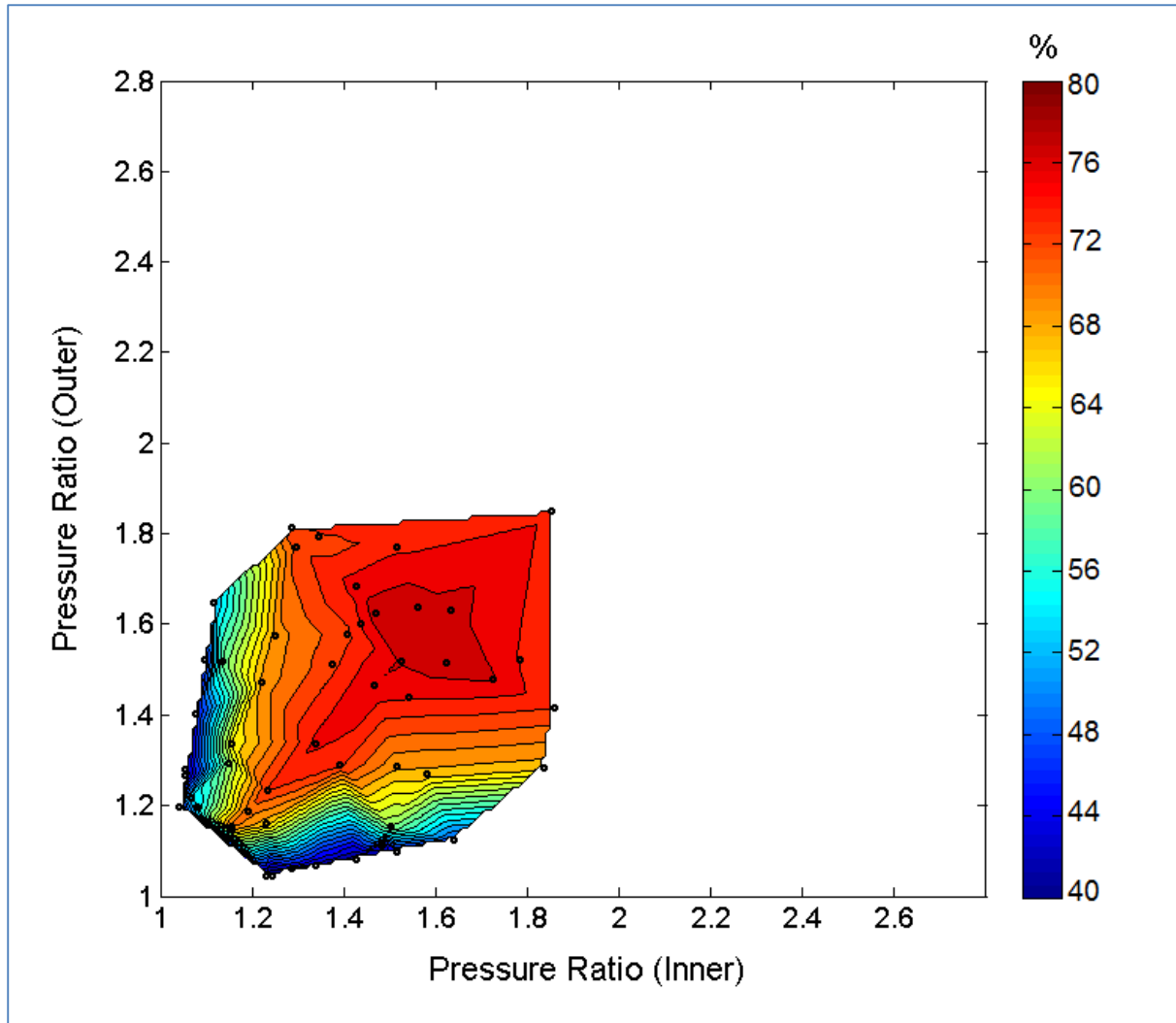


Figure 4.18: Total-to-static efficiency versus the pressure ratios across inner and outer inlets ($U_3/C_{is}=0.8$)

As witnessed by the number of plots displayed in this section, a double-entry turbine is much more complex to visualize than its single-entry equivalent. Therefore, it would be desirable to be able to combine all the steady-state performance tests into a single plot (as can be done for a single-entry turbine). However as discussed, the addition of an extra entry means that three independent parameters are required to fix the operating point for a double-entry turbine: pressure ratio across the first inlet, pressure ratio across the second inlet and a speed based parameter (either the velocity ratio or the non-dimensional speed). This invites the addition of a third axis such that the equal and unequal steady-flow characteristics can be combined on a single, three-dimensional plot as shown in Figure 4.19. The x and y axes scale according to the pressure ratios across each inlet as in Figures 4.16-4.18 and the z-axis scales with the velocity ratio. The four planes of data representing the equal admission and the three unequal admission planes have been

labelled. Although this plot shows the efficiency of the turbine, contours of any dependent variable can be plotted in this way (mass parameter, turbine power, etc).

This representation allows the engineer to immediately see the set of operating conditions (with respect to the parameters PR_{inner} , PR_{outer} and U_3/C_{is}) that produce optimum turbine performance. It also clearly demonstrates the sensitivity of the turbine to the range of off-design conditions. Furthermore, since the velocity ratio and entry pressure ratios vary independently over the course of an *unsteady* pulse, the operating point of the turbine will vary in time within this three-dimensional space. This type of plot is therefore very useful in comparing the unsteady operating range to the performance of the turbine under equivalent steady-flow conditions.

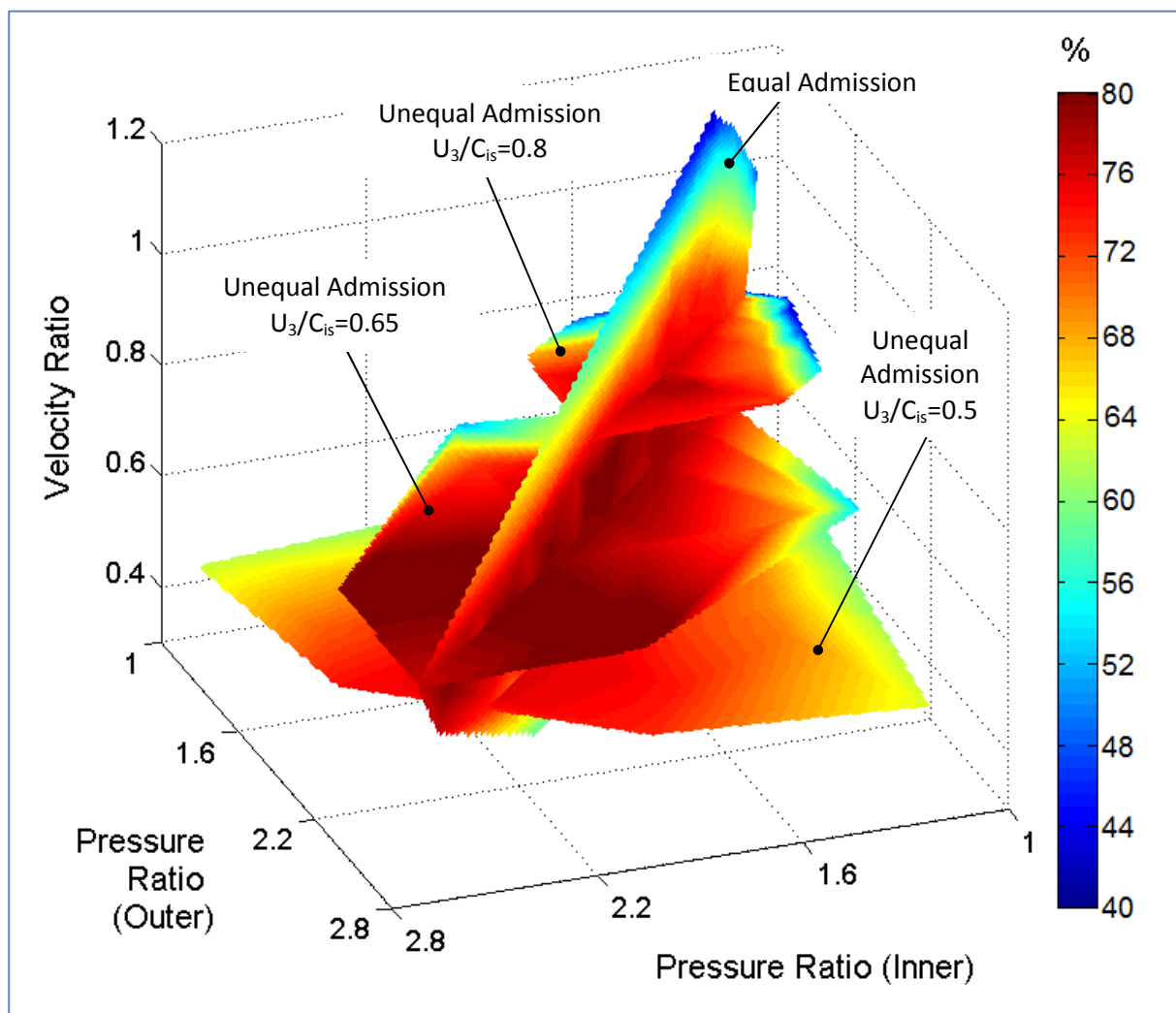


Figure 4.19: Total-to-static efficiency as a function of inner pressure ratio, outer pressure ratio and velocity ratio

4.4 Conclusions

4.4.1 Equal Admission

Testing the operation of the turbine when driven by equal inlet flows was an important first step in establishing the baseline turbine performance. A wide range of equal admission tests were possible due to the range of loading conditions available using the Imperial College eddy-current dynamometer. Five speed parameters were tested ranging from $26.9 \text{ rps}/T^{1/2}$ - $53.7 \text{ rps}/T^{1/2}$. At each speed, the width of the available data was limited by the maximum dynamometer loading (minimum magnetic rotor gap) at high pressure ratio and the friction of the bearings at the lowest pressure ratio. This resulted in a range of velocity ratios between 0.362 and 1.184 and a range of pressure ratios between 1.099 and 2.679.

Plotting the total-to-static efficiency of the turbine revealed a clear optimal operating condition at a velocity ratio of approximately 0.66. The maximum efficiency was measured to be 80.4% at a speed parameter of $48.3 \text{ rps}/T^{1/2}$, but this value was relatively insensitive to rotor speed. In fact, the equal admission efficiencies all fell between 75% and 80% *at the peak velocity ratio*, thus demonstrating the excellent performance of the turbine design. Off-design flow conditions resulted in a minimum turbine efficiency of 43%, but this corresponds to the highest velocity ratio and hence a very low power situation. These values of efficiency measured at constant speed were interpolated to provide efficiency data at constant velocity ratio and pressure ratio. This representation could be useful in matching the turbine to the appropriate compressor to ensure the operation of the turbine matches the optimal conditions for a given inlet pressure ratio.

The swallowing characteristics of the turbine operating under equal admission conditions were also presented by plotting the mass parameter versus the pressure ratio. The decreasing rise in the mass parameter with an increasing pressure ratio was compared to the nozzle characteristics approaching asymptotic choking conditions in the exducer of the turbine or in the nozzle blades at constant speed. Unlike a simple nozzle, however, the centrifugal head created from the spinning rotor means that the swallowing characteristic of the turbine was speed-dependent. Also, the turbine did not appear to reach an asymptote at the maximum test pressure ratio, thus suggesting that fully choked conditions were not reached due to the maximum capacity of the air supply system.

Finally, a new, concise representation of turbine performance was suggested that shows the turbine efficiency as a function of velocity ratio and pressure ratio. This produced a 2-D contour plot of total-to-static efficiency calculated from the linear interpolation of all the equal admission test results. It showed the conditions of peak turbine performance and also revealed lines of constant speed operation.

4.4.2 Unequal Admission

Testing the performance of the turbine when unequal flows are driving the rotor wheel is not just an academic exercise but a practical and important investigation given the reality of the on-engine application. Due to the firing order of an internal combustion engine, the exhaust pulses flowing into each of turbine passage typically arrive at the wheel out-of-phase, thereby producing unequal admission on an instantaneous basis. Thus, the first step in understanding the performance implications was to test the different combinations of steady-state conditions at the two inlets. Owing to the extra degree of freedom that an additional turbine inlet brings, a large range and combination of flow conditions were possible. To ensure direct comparison with unequal admission operation, the turbine was tested at three different velocity ratios: 0.5, 0.65 and 0.8. At each of these velocity ratios, a range of pressure ratios were tested. In order to present the data on a two-axis Cartesian representation, one of the inlets was held at a constant pressure ratio while the second inlet was varied. This process was then reversed for the second inlet to test the response of each inlet separately. The plots of efficiency versus the ratio of inlet pressures ('level' of unequal admission) revealed a marked reduction in performance due to unequal admission. At the most extreme, partially admitting the turbine resulted in a maximum efficiency drop of 16 to 34 points of efficiency depending on the velocity ratio. In contrast, when the pressure ratio in the varying inlet rose above the value in the constant inlet, there was a slight increase in efficiency compared to the equal admission state.

To demonstrate the influence of unequal admission on the swallowing characteristics of the turbine, the concept of effective area was used. This approach allowed the calculation of the area of an ideal nozzle required to pass the measured mass flow for the pressure ratio measured across the turbine. Two such areas were calculated for each of the turbine inlets. Plotting the effective area of the varying inlet revealed an expected curve that showed the variation of swallowing capacity that resulted from a change of pressure ratio applied to this inlet. The flow areas ranged between 0 m^2 and 0.00063 m^2 for the lowest and highest pressures respectively. Some dependence on the velocity

ratio was noted and could be attributed to the change in the centrifugal head created by the spinning turbine rotor. Plotting the effective area of the constant pressure inlet revealed some very interesting characteristics of this turbine. Despite a constant inlet condition, the effective area changed due to the changes in pressure in the other inlet. This clearly showed the influence that one inlet has on the other. Compared to full flow, when the pressure in one entry is low, the second entry can swallow more mass, and when it is high, the second entry swallows less. Both the swallowing capacity and the efficiency of the two turbine inlets demonstrated remarkably symmetric lines under unequal admission. This indicates that, unlike a twin-entry turbine, the two inlets of a double-entry design are very similar in their response to changes in the flow conditions.

Like the equal admission results, the performance results were collated into a series of contour plots showing the total-to-static efficiency of the turbine versus the two inlet pressure ratios. This allowed the easy identification of the influence that unequally admitted flows have on the performance of the turbine. Three such interpolated contour plots were created, showing the unequal admission efficiency at the three velocity ratios under investigation. The fact that the turbine operating point could be described by three dimensionless numbers -- two inlet pressure ratios and the velocity ratio -- suggests a plot of the entire range of steady-state tests on a single three-dimensional plot. In this plot, the four planes of efficiency contours were appropriately placed to give a sense of the areas of optimum turbine efficiency with respect to the three independent parameters. One of the main aims of this research was to completely characterize the steady-state operation of the turbine. While this is relatively simple for a single-entry turbine, the additional entry in a dual-entry turbine greatly increased the set of possible inlet conditions that drive the turbine. In all, a total of 272 individual tests were performed on this turbine, measuring a broad range of both equal and unequal steady-tests. It does not appear as if such an exhaustive range of tests has been attempted before in the published research. This was critical if a direct comparison between the steady and unsteady performance was to be made.

In summary, this research clearly shows that a double-entry turbine driven by unequal admission can result in significant efficiency loss even if the flow in one of the inlets is optimal for equal admission performance. One of the most interesting cases analysed was that of partial admission where a severe loss in performance results as a consequence of blocking one inlet. If the turbine were to behave as if it were two separate turbines, no additional loss of efficiency (beyond equal admission) would result. In this case the only thing that would decrease the efficiency of the turbine would be the frictional loss produced by the turbine as it spins through the stagnant entry. Since, for the most part, the passages in a double-entry design are isolated from each other, it might

be tempting to conclude that the fluid in the flowing entry should simply drive one half of the turbine wheel in a manner that is unaffected by the flow in the other inlet. Clearly, however, there are a number of things that lead to the turbine experiencing more loss than indicated by this ideal situation. First, the research presented in this Chapter has revealed evidence that there is a significant amount of interaction between the inlets, and these effects increase with the pressure difference between them. It seems plausible to suggest on this basis that the difference in pressure leads to a significant flow interaction in the inter-space that joins the two inlets -- especially near the two tongues. This imbalance would not only create mixing loss, but more importantly, create a non-ideal inlet flow angle entering the turbine. The second thing that must be considered with this turbine is the effect of a rotor passage being fed periodically from the two 180° nozzle sectors. To understand this effect, one must be in the rotor frame of reference as it moves from the non-flowing section to the flowing section. At the beginning, the flow in the passage will be largely stagnant as a result of rotation through the non-flowing sector. As it moves into the flowing section, the flow will start to fill the passage. If the filling is very fast in relation to the speed of the rotor, the desired fluid aerodynamics would be established very quickly, leading to an efficient transfer of power in the passage. However, if the flow requires much of the 180° rotation to establish, the result is a blade passage that never quite reaches ideal flow conditions since it is not given sufficient time to develop. The only real way to study this type of phenomenon is by solving the fluid flow in three-dimensional CFD. This will be done in the Chapter 5.

4.5 Uncertainty

The instruments that have been used to obtain the data in this Chapter, as well as the measurement uncertainties of each, have been outlined in Chapter 3. It is now important to be able to assess how these individual uncertainties propagate to the final valuation of the parameters used to judge the turbine performance. The calculation of the propagated error can be made using the root-sum-square (RSS) method outlined by Moffat [44] and used by Syzmko [8] and Rajoo [32]. This method is described by equation 4.24 for a given parameter $P = f(x_1, x_2, x_3 \dots)$:

$$\pm \Delta P_{RSS} = \sqrt{\sum_{i=1}^n \left(\Delta x_i \frac{\partial P}{\partial x_i} \right)^2} \quad (4.24)$$

where the sensitivity parameter ($\partial P / \partial x_i$) is used to account for the influence that each variable Δx_i has on the performance parameter P . Equations 4.25 to 4.30 provide the expansion of equation 4.24 for the speed parameter (SP), mass flow parameter (MFP), velocity ratio (VR), pressure ratio (PR), efficiency (η_{ts}), and effective area respectively.

The uncertainty in the speed parameter for the steady-state measurements is shown in Figure A1 against the pressure ratio. As one might expect, the main source of uncertainty was the speed signal itself, with the variation in the total temperature measurement playing a minor role. Thus, the highest uncertainty was $\pm 0.14 \text{ rps} / \sqrt{K}$ at a speed parameter of $53.7 \text{ rps} / \sqrt{K}$ which works out at approximately $\pm 0.3\%$.

$$\Delta SP = \sqrt{\left(\Delta T \frac{\partial SP}{\partial T}\right)^2 + \left(\Delta N \frac{\partial SP}{\partial N}\right)^2 + \left(\Delta \dot{m} \frac{\partial SP}{\partial \dot{m}}\right)^2} \quad (4.25)$$

The uncertainty in the mass flow parameter is plotted against the pressure ratio as shown in Figure A2. This allows a comparison to the performance curve in Figure 4.5. The uncertainty is relatively small with a maximum value of $\pm 1.2\text{E-}7 \text{ (kg/s)} / \sqrt{\text{K/Pa}}$ at the highest pressure ratio. This corresponds to $\pm 0.25\%$ of the measurement of the mass flow parameter at this point. The measurement of the mass flow rate was the largest contributor to this uncertainty.

$$\Delta MFP = \sqrt{\left(\Delta T \frac{\partial MFP}{\partial T}\right)^2 + \left(\Delta P_1 \frac{\partial MFP}{\partial P_1}\right)^2 + \left(\Delta \dot{m} \frac{\partial MFP}{\partial \dot{m}}\right)^2} \quad (4.26)$$

The uncertainty in the pressure ratio is shown in the appendix in Figure A3. The uncertainty rises with increasing pressure ratio, recording a maximum value of ± 0.008 at a pressure ratio of 2.68. This represents a $\pm 0.3\%$ of the pressure ratio measurement.

$$\Delta PR = \sqrt{\left(\Delta \dot{m} \frac{\partial PR}{\partial \dot{m}}\right)^2 + \left(\Delta P_1 \frac{\partial PR}{\partial P_1}\right)^2 + \left(\Delta P_4 \frac{\partial PR}{\partial P_4}\right)^2} \quad (4.27)$$

The uncertainty of the velocity ratio was calculated using equation 4.28 for each of the range of equal admission tests and displayed in Figure A4. The uncertainty in the velocity ratio rose to a

maximum of ± 0.0126 at a maximum velocity ratio of 1.18. The maximum percentage uncertainty actually occurred at the lowest speed: $\pm 1.2\%$.

$$\Delta VR = \sqrt{\left(\Delta \dot{W}_{isen} \frac{\partial VR}{\partial \dot{W}_{isen}}\right)^2 + \left(\Delta N \frac{\partial VR}{\partial N}\right)^2 + \left(\Delta \dot{m} \frac{\partial VR}{\partial \dot{m}}\right)^2} \quad (4.28)$$

The efficiency is one of the more important performance parameters but unfortunately is also subject to the greatest error due to the inherent difficulties in making very small torque measurements. Figure A5 shows the uncertainty in the efficiency value evaluated for a range of different speeds and plotted against velocity ratio. To give some sense of the uncertainty in relation to the calculated values, Figure 4.20 plots the uncertainty characteristics onto the turbine performance map. It is clear from this figure that the uncertainty in the efficiency measurement is greatest at the highest velocity ratios where the power absorption is lowest. This behaviour in the uncertainty in the efficiency is very similar to the characteristics shown on the plot of Dale and Watson [9] in Figure 4.9. The maximum uncertainty of ± 9.35 percentage points occurs at the lowest speed and highest velocity ratio where power absorbed is only 0.5 kW. Measuring such a small amount of power means that over 90% of the uncertainty in the efficiency at this point is due to the inaccuracies in the torque. It is important to note, however, that these values of uncertainties do not take into account repeated measurements. In order to mitigate the error at the highest velocity ratio, the tests were often repeated to produce a greater degree of confidence.

$$\Delta \eta_{ts} = \sqrt{\left(\Delta N \frac{\partial \eta_{ts}}{\partial N}\right)^2 + \left(\Delta \tau \frac{\partial \eta_{ts}}{\partial \tau}\right)^2 + \left(\Delta \dot{W}_{isen} \frac{\partial \eta_{ts}}{\partial \dot{W}_{isen}}\right)^2} \quad (4.29)$$

Finally, equation 4.30 was used to evaluate the uncertainty propagated to the evaluation of the unequal admission effective area. This uncertainty is shown in the appendix in figure A6 for the unequal case with a velocity ratio of 0.65. The maximum uncertainty of $\pm 5.27 \times 10^{-5} \text{ m}^2$ corresponds to the highest pressure ratio case.

$$\Delta A_{eff} = \sqrt{\left(\Delta \dot{m} \frac{\partial A_{eff}}{\partial \dot{m}}\right)^2 + \left(\Delta T \frac{\partial A_{eff}}{\partial T}\right)^2 + \left(\Delta P \frac{\partial A_{eff}}{\partial P}\right)^2} \quad (4.30)$$

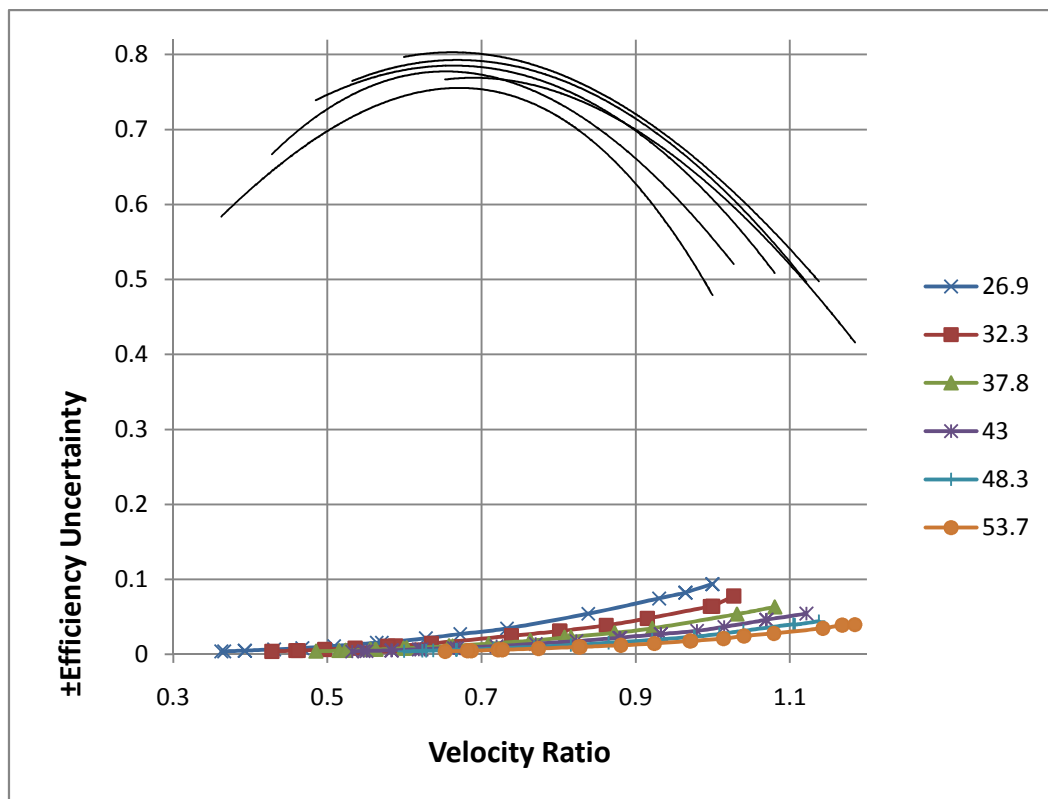


Figure 4.20: Uncertainty in total-to-static turbine efficiency.

Chapter 5

Computational Fluid Dynamics Model

5.1 Introduction

Computational Fluid Dynamics (CFD) is a generic term used to describe a numerical approach to solving approximations of the governing fluid dynamical equations. Since these equations encompass a set of coupled, non-linear, differential equations, direct analytical solutions are only possible in a few, very simplified cases. Computational Fluid Dynamics therefore approximates the solution of a fluid problem through a discretization method whereby the differential equations are represented by a system of algebraic relationships that represent fluid behaviour at discrete locations. Since computers excel at performing a large number of repeated calculations, it is therefore possible to divide the fluid domain into a number of control volumes over which these algebraic equations can be balanced numerically. CFD thus relies on spatial discretization of the fluid domain in either two or three dimensions and the definition of a set of conditions specified at its boundaries. Non-steady solutions also require temporal discretization, whereby the solution to the fluid problem is solved in time by repeatedly advancing and solving the instantaneous state at discrete time intervals. The engineering significance of a full 3-D CFD solution to a problem is clear. Such a solution provides a complete data set of all the physical quantities of the fluid flow at each discretized location in the domain. This wealth of data can then be used to study the physical phenomena that drive the problem in a greater depth than possible by experimental means. Of course, the caveat is that CFD numerical solutions will always be approximate, and their accuracy can vary greatly from case to case and model to model.

Due to the complexity of the flows that dominate the blade passages in high-speed turbomachinery, a very fine mesh density is often required to resolve the small scale flow structures and produce accurate flow field results. However, even with the current state-of-the-art computing power, modelling an entire turbocharger domain with such fine mesh density can become very computationally expensive. A finely discretized model that may take days to arrive at a steady-state solution can take months for a fully transient, unsteady solution. Thus, there are two common compromises to enable a study of the fluid flow in turbochargers without taking a prohibitive length

of computational time. One of the more common methods is to reduce the problem to a single blade passage operating in steady-state. This approach is especially useful in axial turbomachinery modelling since the flow at the inlet is largely steady and evenly distributed around the circumferential of the stage. Thus, a very refined computational mesh can be used without making it too time-consuming to solve on a typical workstation. However, for the double-entry, mixed-flow turbocharger considered here, a single passage model can only be used when the flows into each inlet are equal. In any other condition (when the flows are introduced unequally in a steady or pulsating manner), there will be non-uniform inflow around the circumference of the moving rotor, thereby producing an inherently unsteady problem. Thus, the second compromise (adopted here), is simply to model the entire turbine with a decreased mesh density. This will decrease the accuracy of the solution as well as average out many of the small-scale fluid structures. However, the main purpose of such a computational approach is to study the predominant *macro* flow features that are difficult to measure experimentally. Even this approach can be quite computationally expensive since the rotor must turn a minimum of one full revolution before the flow in the passages reaches periodic convergence.

The computational results presented in this section were all obtained using a commercially available numerical CFD code: ANSYS CFX. One of the major advantages of this code over other packages is its support for turbomachinery modelling. ANSYS includes two additional turbomachinery-specific programs (BladeGen and TurboGrid) as well as some additional support in the pre and post-processing stages. BladeGen significantly reduces the effort in producing the complex three-dimensional blade geometries that are needed for the meshing programs. TurboGrid then takes this geometry and produces a high-quality, structured mesh of the desired density in the blade passage. In addition, CFX supports complex mesh topologies and sliding mesh interface solutions suitable for fully transient turbine stages. The detailed numerical methodology of the CFD code will not be considered in any great depth in this Chapter due to the wealth of available literature on the subject [47]. Also, the main aim here is not to study the use of CFD or the code itself (as this has been well established) but its use as tool to compliment the experimental work in Chapter 4. The compromise of a reduced mesh density to allow for a fully transient 3-D model means that precise agreement with the experiments is not expected.

5.2 Numerical Methodology

5.2.1 Governing Equations

The transport equations that represent the instantaneous mass, momentum and energy conservation are as follows:

Continuity Equation

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{U}) = 0 \quad (5.1)$$

Momentum Equation

$$\frac{\partial(\rho \mathbf{U})}{\partial t} + \nabla \cdot (\rho \mathbf{U} \otimes \mathbf{U}) = -\nabla p + \nabla \cdot \boldsymbol{\tau} + \mathbf{S}_M \quad (5.2)$$

where the stress tensor is given by:

$$\boldsymbol{\tau} = \mu \left(\nabla \mathbf{U} + (\nabla \mathbf{U})^T - \frac{2}{3} \nabla \cdot \mathbf{U} \right) \quad (5.3)$$

Total Energy Equation

$$\frac{\partial(\rho h_{tot})}{\partial t} - \frac{\partial p}{\partial t} + \nabla \cdot (\rho \mathbf{U} h_{tot}) = \nabla \cdot (\lambda \nabla T) + \nabla \cdot (\mathbf{U} \cdot \boldsymbol{\tau}) + \mathbf{U} \cdot \mathbf{S}_M + \mathbf{S}_E \quad (5.4)$$

where the total enthalpy is defined as:

$$h_{tot} = h + \frac{1}{2} \mathbf{U}^2 \quad (5.5)$$

and $\nabla \cdot (\mathbf{U} \cdot \boldsymbol{\tau})$ is the work due to viscous stresses and $\mathbf{U} \cdot \mathbf{S}_M$ is the work term due to external momentum sources (neglected by CFX).

In addition, the equation of state for an ideal gas must also be included:

Ideal Gas Equation of State

$$\rho = \frac{(p + p_{ref})}{R_{air} T} \quad (5.6)$$

5.2.2 Spatial Discretization

Computational Fluid Dynamics requires dividing the spatial domain into discrete volumes over which the governing equations can be evaluated such that the mass, momentum and energy are conserved. Consider the generic form of the conservation of a scalar ϕ as an example:

$$\frac{\partial(\rho\phi)}{\partial t} + \frac{\partial}{\partial \mathbf{x}_j}(\rho \mathbf{U}_j \phi) = \frac{\partial}{\partial \mathbf{x}_j} \left(\Gamma_{eff} \left(\frac{\partial \phi_i}{\partial \mathbf{x}_j} \right) \right) + S_\phi \quad (5.7)$$

This equation must be integrated over each of the control volumes in the domain. Gauss's Divergence Theorem can be used to convert some of the volume integrals into surface integrals as follows:

$$\frac{\partial}{\partial t} \int_V (\rho\phi) dV + \int_s \rho \mathbf{U}_j \phi \, dn_j = \int_s \Gamma_{eff} \left(\frac{\partial \phi_i}{\partial \mathbf{x}_j} \right) dn_j + \int_V S_\phi dV \quad (5.8)$$

Where V and s represent volume and surface integrals respectively and dn_j are the differential Cartesian components of the normal surface vector. Note that the time derivatives are taken outside the integrals if the control volumes do not deform in time.

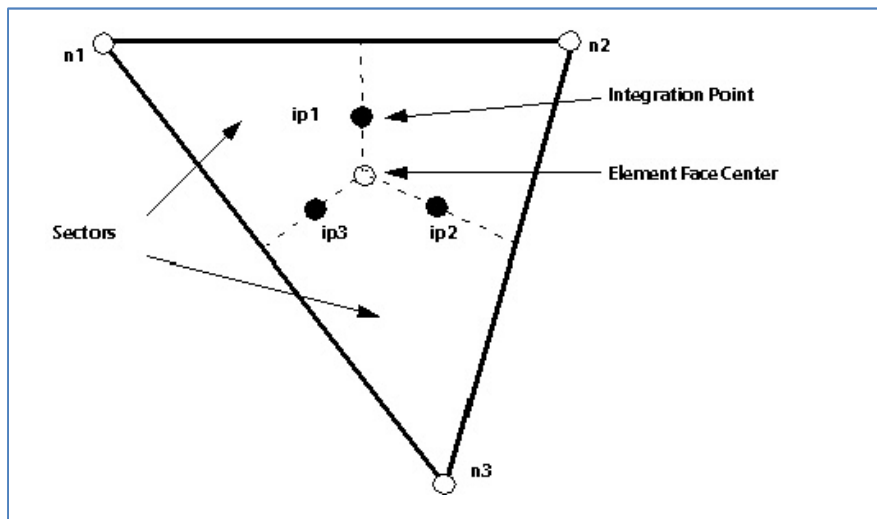


Figure 5.1: Mesh element showing nodes and integration points in CFX

The integrals must then be converted into the discrete form to allow the CFD solver to evaluate the conservation equations for the control volume. Figure 5.1 shows the points of

integration (subscript ip) where the discrete form of the conservation equations is evaluated in CFX. Equation 5.9 gives the discretized form of equation 5.8.

$$V \left(\frac{(\rho\phi)^{n+1/2} - (\rho\phi)^{n-1/2}}{\Delta t} \right) + \sum_{ip} \dot{m}_{ip} \phi_{ip} = \sum_{ip} \Gamma_{eff} \left(\frac{\partial \phi_i}{\partial \mathbf{x}_j} \right) \Delta n_j + \overline{S_\phi} V \quad (5.9)$$

where V is the control volume, Δt is the timestep, Δn_j is the normal surface vector, and the superscript $n + 1/2$ and $n - 1/2$ denotes the value of the variable at the start and end of the timestep respectively. The mass flow through a surface of the control volume is:

$$\dot{m}_{ip} = (\rho U_j \Delta n_j)_{ip} \quad (5.10)$$

and the advection of the variable ϕ_{ip} at the integration point is approximated in terms of the value of ϕ at the nodes. Equation 5.11 shows the general form of the approach taken by CFX:

$$\phi_{ip} = \phi_{up} + \beta \nabla \phi \Delta \zeta \quad (5.11)$$

where ϕ_{up} is the value at the upwind node, and ζ is the vector joining the upwind node to the integration point. The *high resolution scheme* used for the modelling in this thesis computes β locally and sets $\nabla \phi$ equal to the control volume gradient at the upwind node.

5.2.3 Temporal Discretization

For the transient simulations presented in this Chapter, a *Second Order Backward Euler Scheme* is used to discretize the time derivative. This method uses the two previous time steps (superscript 'o' and 'oo') to calculate the time derivative in equation 5.9 as shown below:

$$(\rho\phi)^{n-1/2} = (\rho\phi)^o + \frac{1}{2}((\rho\phi)^o - (\rho\phi)^{oo}) \quad (5.11)$$

$$(\rho\phi)^{n+1/2} = (\rho\phi) + \frac{1}{2}((\rho\phi) - (\rho\phi)^o) \quad (5.12)$$

This results in the final discretization of the time derivative as follows:

$$V \left(\frac{(\rho\phi)^{n+1/2} - (\rho\phi)^{n-1/2}}{\Delta t} \right) \approx V \frac{1}{\Delta t} \left(\frac{3}{2}(\rho\phi) - 2(\rho\phi)^o + \frac{1}{2}(\rho\phi)^{oo} \right) \quad (5.13)$$

5.2.4 Turbulence Model

Turbulence is an important characteristic of the fluid in a turbocharger as it can have a significant influence on the characteristics of the flow. The Navier Stokes equations provided earlier, in principle, describe all the fluid regimes from laminar to turbulent, but the lengths scales that are involved in a typical turbulent flow can be very small. Thus, a prohibitively fine mesh would be required to resolve such detailed structures. Therefore, to greatly reduce the computational cost compared to Direct Numerical Simulation (DNS), *turbulence models* are introduced that modify the governing equations so as to approximate the mean flow quantities while still taking into account the turbulence effects. In these types of models, variables are evaluated as the sum of an averaged term and a fluctuating term to produce the Reynolds Averaged, Navier-Stokes (RANS) Equations. The addition of these terms increases the number of unknowns, thereby making it necessary to introduce further relations to close the problem. These additional equations form the ‘turbulence model’ that has been selected for the engineering problem. To completely address the subject of turbulence modelling would require a long and in-depth treatment that is not appropriate here. Therefore, a brief outline of the two-equation, eddy-viscosity turbulence model that was used for the modelling in this chapter will be provided. Further information is widely available in the available texts and journal publications on this subject. [47]

The addition of average and fluctuating components of a variable ($\bar{\phi}$ and ϕ') into the governing equations will produce additional terms such as Reynolds stresses ($\overline{\rho u'_i u'_j}$) and Reynolds scalar fluxes ($\overline{\rho u'_i \phi'_j}$) due to the presence of the quadratic, non-linear terms. To resolve these additional terms, an eddy-viscosity turbulence model is often introduced that suggests the following relations:

$$-\overline{\rho u'_i u'_j} = \mu_t \left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) - \frac{2}{3} \left(\mu_t \frac{\partial U_k}{\partial x_k} + \rho k \right) \quad (5.14)$$

$$-\overline{\rho u'_i \phi'_j} = \Gamma_t \frac{\partial \bar{\phi}}{\partial x_j} \quad (5.15)$$

where μ_t is the turbulent viscosity and Γ_t is the turbulent eddy diffusivity. These additional variables can be related through the turbulent Prandtl number Pr_t as given in equation 5.16 [47]:

$$\Gamma_t = \frac{\mu_t}{Pr_t} \quad (5.16)$$

The value of the “effective” diffusivity Γ_{eff} in equation 5.7 will then become a summation of the molecular diffusivity and the turbulent diffusivity defined by equation 5.16.

The treatment in equations 5.14-5.16 defines the eddy-viscosity approach to turbulence but has resulted in an additional unknown: the turbulent viscosity μ_t . One of the more common approaches to calculate this additional term is through the use of a *two equation turbulence model*. These models offer a compromise between numerical cost and simulation accuracy. A brief outline of the *k – ε model* that is used in this thesis is provided here. The main assumption is that the turbulence viscosity is linked to the turbulent kinetic energy k and the turbulent eddy dissipation ε through the following relation:

$$\mu_t = C_\mu \rho \frac{k^2}{\varepsilon} \quad (5.17)$$

where C_μ is a constant. The turbulent kinetic energy and the turbulent eddy dissipation can be derived from their transport relations [46]:

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial}{\partial \mathbf{x}_j}(\rho U_j k) = \frac{\partial}{\partial \mathbf{x}_j} \left(\Gamma_{eff,k} \left(\frac{\partial k_i}{\partial \mathbf{x}_j} \right) \right) + P_k - \rho \varepsilon \quad (5.18)$$

$$\frac{\partial(\rho \varepsilon)}{\partial t} + \frac{\partial}{\partial \mathbf{x}_j}(\rho U_j \varepsilon) = \frac{\partial}{\partial \mathbf{x}_j} \left(\Gamma_{eff,\varepsilon} \left(\frac{\partial \varepsilon_i}{\partial \mathbf{x}_j} \right) \right) + \frac{\varepsilon}{k} (C_{s1} P_k - C_{s2} \rho \varepsilon) \quad (5.19)$$

where the values of effective diffusivity are given by: $\Gamma_{eff,k} = \mu + \mu_t / Pr_k$ and $\Gamma_{eff,\varepsilon} = \mu + \mu_t / Pr_\varepsilon$. The values of C_{s1} , C_{s2} , Pr_k and Pr_ε are all constants contained in the CFX code. This finally results in a closed solution to the discretized Navier Stokes equations that estimates the effects of turbulence using the *k – ε*, eddy-viscosity turbulence model.

5.2.5 Convergence

As illustrated in the previous section, the CFD solver discretizes the governing equations to produce a set of algebraic relations that must be solved iteratively to arrive at a solution for the fluid in and out of each control volume throughout the domain. The solver must then converge to a solution, either in a time-independent sense in the cases of steady-state, or at each timestep in the case of transient simulations. If the set of linearized equations describing the flow are written in a matrix form as $[A][\phi]=[b]$ (where A is the coefficient matrix, ϕ is the solution vector and b is the right hand side of the equality), this system can be solved iteratively by a method of successive correction.

At each new iterative step n , a residual or imbalance between the left and right hand side of the equation is calculated from the previous iteration ($r^{n-1} = b - A\phi^{n-1}$). From this residual, a correction ϕ^{corr} to the previous solution ϕ^{n-1} can be calculated as: $A\phi^{corr} = r^{n-1}$. Thus the new solution will be the old solution plus the correction: $\phi^n = \phi^{n-1} + \phi^{corr}$ and the process is repeated. The convergence of this iterative process is therefore judged on the basis of the size of the *residual* that represents the level of imbalance in various transport equalities. Since the equations are solved for each control volume in the entire fluid domain, the overall convergence is typically defined on the basis of a root mean square (RMS) residual value. The value of the RMS residual that demonstrated convergence in this thesis was typically $5E-6$ as shown in Figure 5.2. This is regarded by CFX as “tight convergence”.

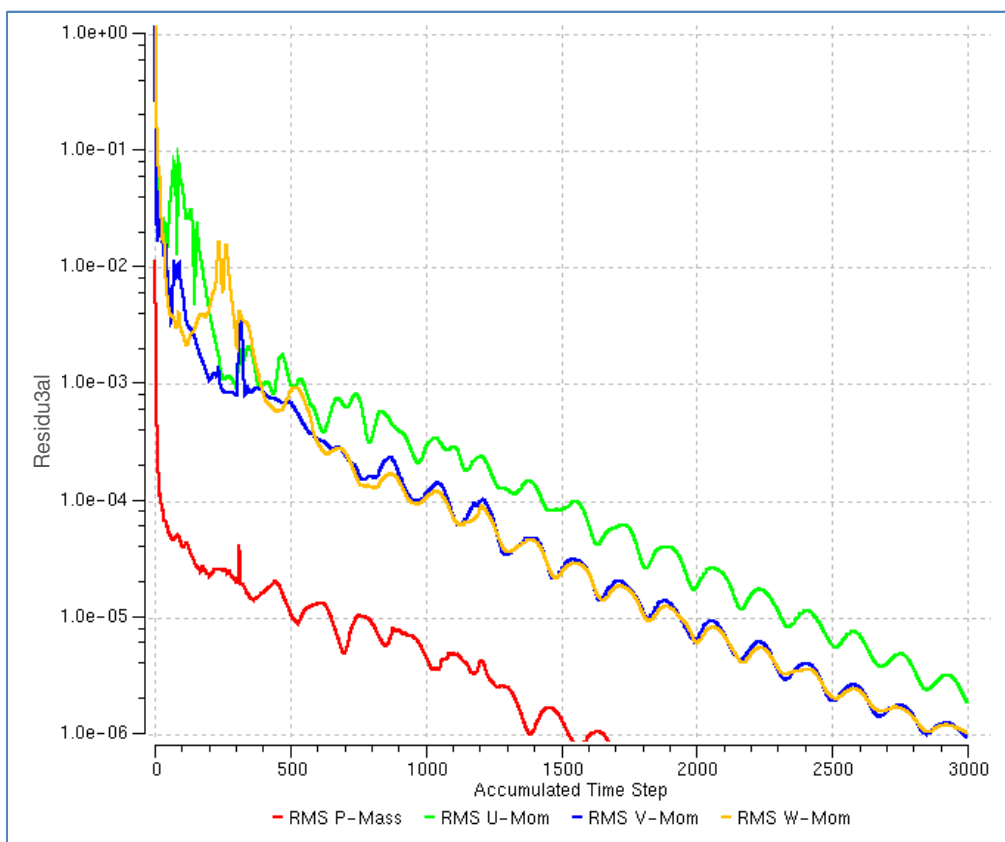


Figure 5.2: Convergence of a typical steady-state simulation showing the mass and momentum RMS residuals

For the transient simulations, convergence must be judged differently. Although all the boundary conditions presented in this thesis are time invariant (implying steady state operation), the flow field in each of the blade passages can change dramatically with time as they rotate from one inlet sector to the next. Thus, to achieve a final solution to a transient case required two separate simulations. First, a preliminary ‘frozen rotor’ solution was obtained using a non-transient, steady state approach. The solution to this first step formed the initial condition to the second step: the

fully transient (moving mesh) model. This transient simulation thus begins with a flow field from the frozen rotor solution and ‘marches’ through time at a fixed rate set by the user. At each timestep, the equations are solved to a specific residual accuracy in a similar way as the non-transient case. It should be evident, therefore, that the RMS residuals calculated at each timestep cannot be used to judge overall convergence of the transient model. A certain length of time is needed before the flow field (initialized by the frozen rotor solution) arrives at a *periodic* solution that represents the actual transient operation of the turbine. Judging the *periodic convergence* of a transient model must be based on indications that the solver has reached a stable, cyclic state. First, it is generally safe to assume that the rotor must make a *minimum* of one full rotation before convergence is achieved. For the transient cases in this thesis, over two full rotor rotations were typically needed before periodic conditions could be reliably assumed. To confirm this, quantities such as mass flow rate, pressure and temperature must have reached a stable, even reading. In addition, one of the best indicators of periodic convergence is the value of instantaneous torque on the turbine blades. This value will transition from the initial frozen-rotor value before eventually reaching a stable, cyclic reading as shown in Figure 5.3. This observation and the stability in the other readings reliably indicate that the transient solution has converged.

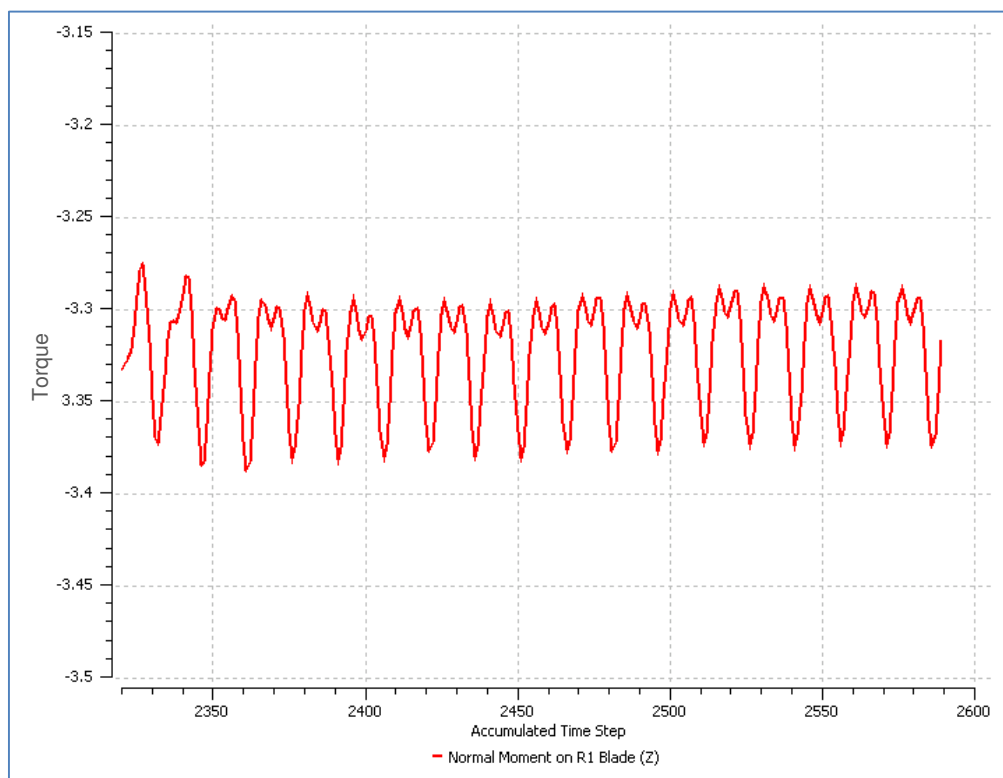


Figure 5.3: Convergence of a typical transient simulation showing torque variation reaching periodic convergence.

5.3 Computational Domain

5.3.1 Mesh

The main components of the computational domain are shown in an exploded view in Figure 5.1. The four main components of the turbocharger turbine (the volute, nozzle, rotor and exit duct) were meshed separately and assembled in the pre-processor. Mesh statistics for each of these domains are provided in Table 5.1. ANSYS TurboGrid was used to obtain structured hexahedral meshing in both the nozzle and rotor. Due to the relatively complex, oblique shape of the volute passages, an unstructured mesh was used here with boundary wall refinement.

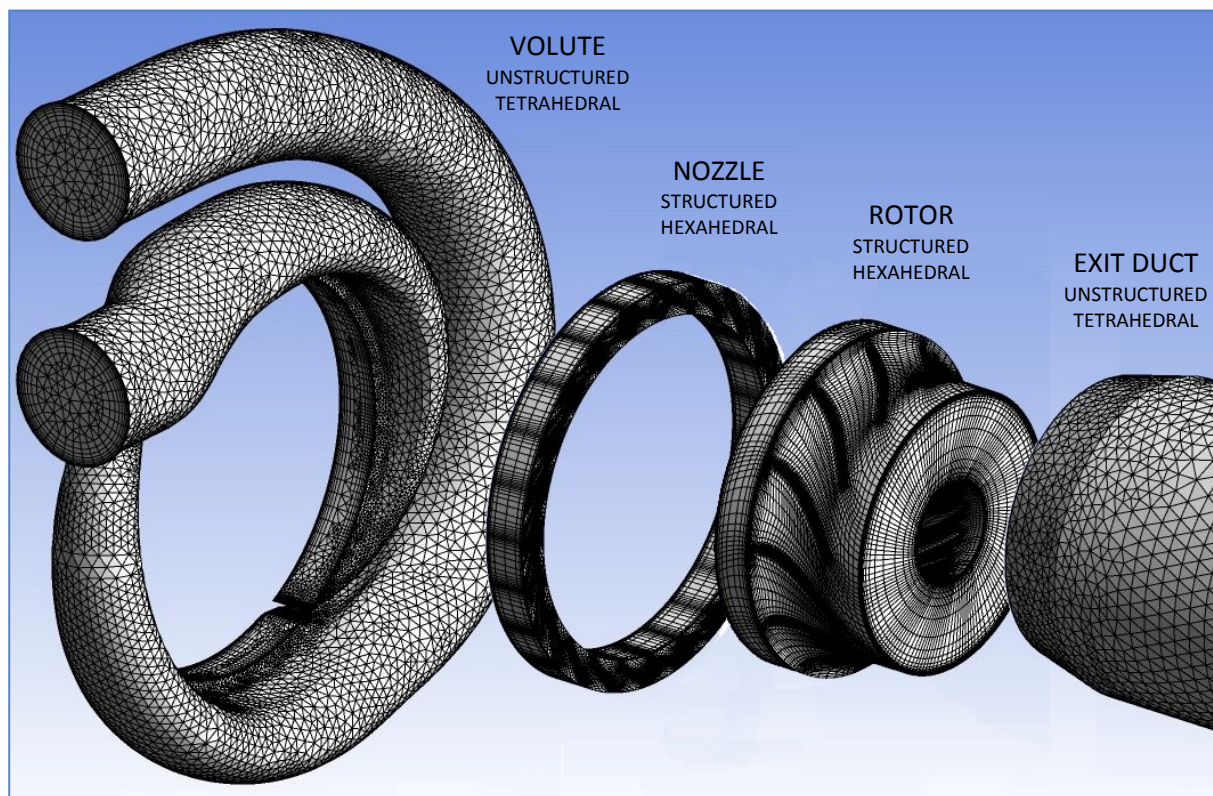


Figure 5.4: Exploded view of the computational domain

The structured mesh of the rotor and nozzle passages is shown in Figure 5.5 with detailed enlargements of the areas of interest. Cell biasing was applied to refine the mesh near the wall regions, both near the blade surfaces but also at the hub and shroud. The air gap between the tip of the blade and the shroud of the volute was modelled with a constant 0.5mm wide mesh containing 7 cells spanning this width. The near wall mesh spacing is often judged on the basis of the

dimensionless distance from the wall to the first node (y^+). The standard definition used by the CFX solver is given in equation 5.20.

$$y^+ = \frac{\sqrt{\tau_n/\rho} \Delta n}{\nu} \quad (5.20)$$

where Δn is the near wall mesh spacing and τ_n is the wall shear stress. Since the value of wall shear can vary significantly over the blade passages, the value y^+ will also vary. The typical range of values in the turbine rotor (calculated from the wall shear at the hub, shroud and blade) was $5 \leq y^+ \leq 50$.

Table 5.1: Mesh characteristics of the turbocharger domain

Region	Element Type	Number of Elements	Number of Nodes	Maximum edge to length ratio
Volute	Unstructured (Tetrahedral, Pyramids, Wedges)	339,123	126,416	27.17
Nozzle	Structured (Hexahedral)	281,664	329,132	352.065
Rotor	Structured (Hexahedral)	495,984	552,696	206.25
Ducting	Unstructured (Tetrahedral, Pyramids, Wedges)	363,018	121,254	42.32
<i>Total</i>		<i>1,479,789</i>	<i>1,129,498</i>	

The geometry of the full CFD model shown in Figure 5.6 required a series of improvements before a final mesh was selected that could converge to a solution for the range of required boundary conditions. For example, it might seem unusual to include the full length of the exit duct, considering the desire to reduce mesh density in this model. Preliminary trials were run with a smaller exit duct, but this was abandoned due to problems with inflow under unequal admission conditions. If, for example, only one limb is flowing, the pressure distribution at any exit plane close to the rotor exducer will show a significant non-uniformity. Areas of low pressure at the exit (due to the stagnant entry flow) would encourage backflow into the domain. For an outlet boundary condition, the simulation will prevent this recirculation by erecting a wall in the region of low pressure. To solve this problem, the location of the exit boundary was extended to the full length of the ductwork used in the experiment. This allowed the unequal flows to mix sufficiently before exiting the domain.

Note in Figure 5.5 that the rotor hub (just downstream of the rotor exducer) was left to continue into the exit duct. In the lab, the rotor is bolted onto a threaded shaft that sticks out beyond this point with a nut attached. Although it is possible to reproduce this geometry in the exit duct mesh, it would require a sharp right-angle to represent the nut. Flow over such a feature would create vortex shedding and boundary layer separation, thereby making solver convergence in this area much more difficult and of little overall importance to model. Therefore, to simplify the transition and ease convergence difficulties, a conical shape was used to close the rotor hub as shown in Figure 5.6.

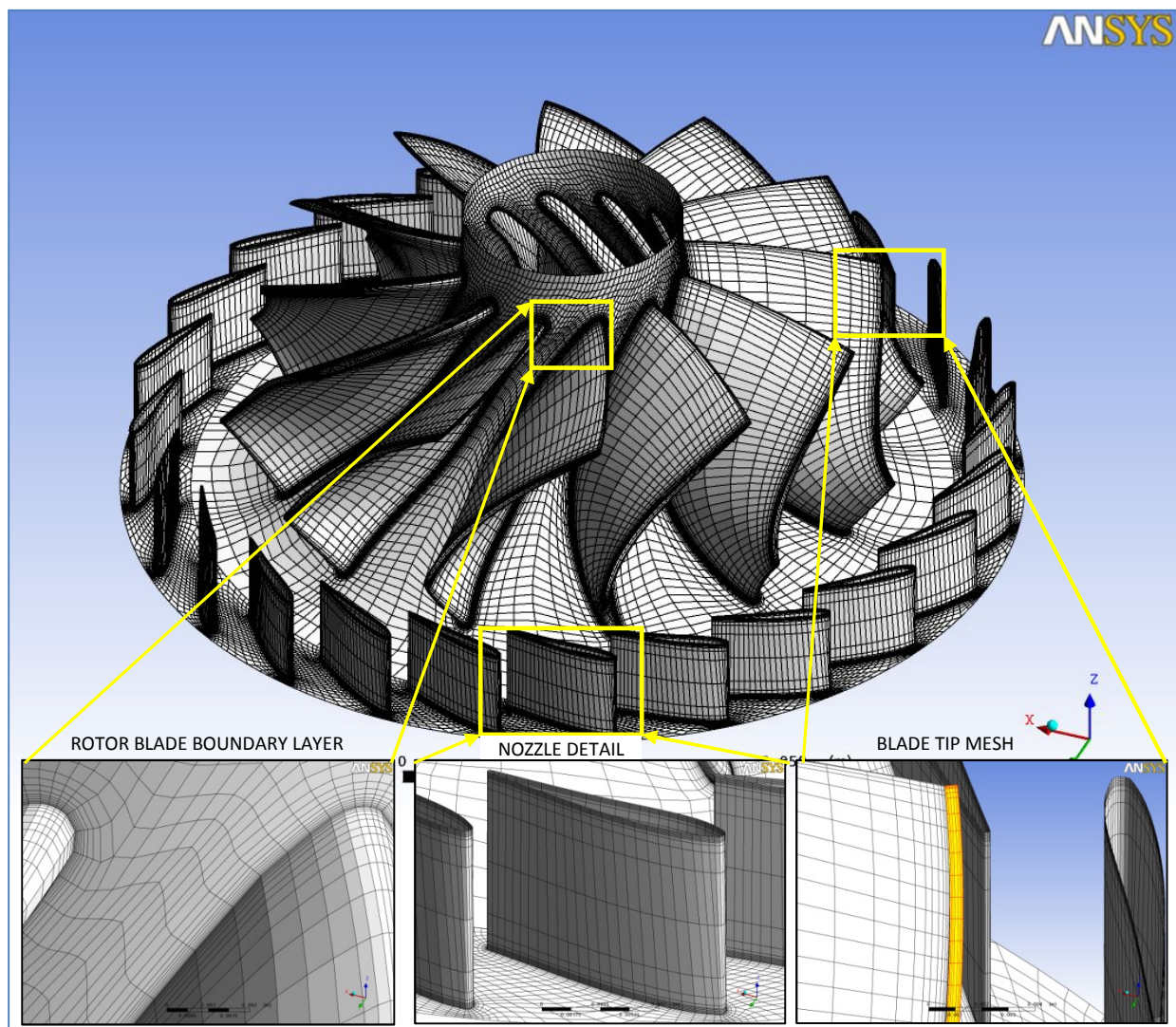


Figure 5.5: Nozzle and rotor mesh with detailed views of interest

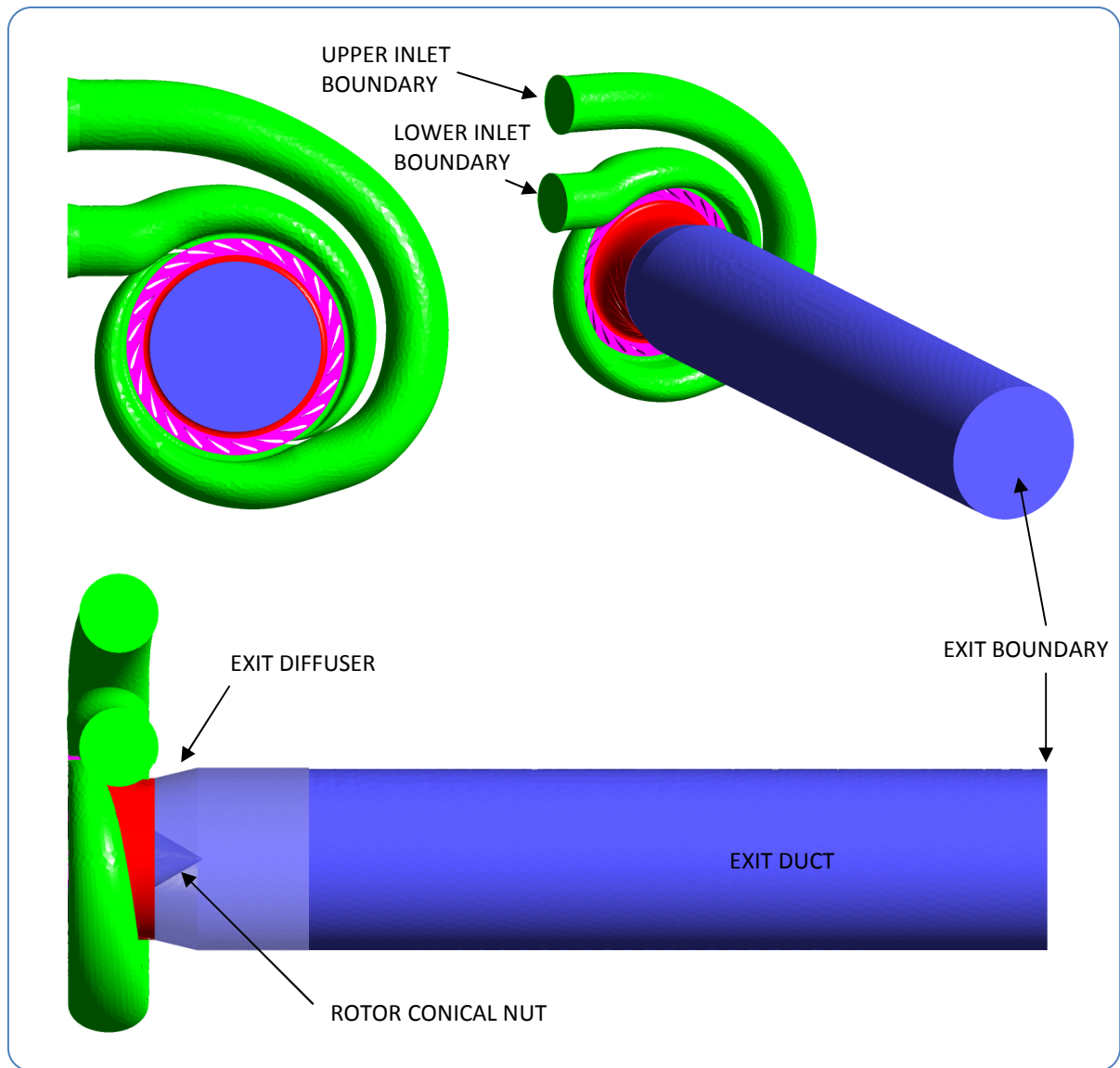


Figure 5.6: Entire fluid domain with inflow and outflow boundaries

5.3.2 Boundary Treatment

As the name implies, boundary conditions set the bounds of the domain, thereby constraining the problem to represent the set of conditions being modelled. Their selection is therefore very important as it can determine how representative the model will be of the real experimental case. All of the boundary conditions in this Chapter were selected to represent the condition in the laboratory – not to simulate on-engine conditions. As such, CFD is used here strictly as a research tool to compliment the experimental data.

INLET

As indicated in Figure 5.6, there are two inflow boundary conditions which must be specified at the upper and lower entries. Each of these boundary conditions requires a temperature specification as well as a value of velocity, mass flow or pressure. Although a mass-flow inlet boundary condition results in a more robust simulation, *total-pressure inlet* boundaries were used instead so that the resulting mass flow through the domain could form part of the validation. By comparing the mass flow in the model to the experimentally measured values, some confidence in the model should result. This might be considered as a virtual experiment where a pressure ratio is applied across the domain resulting in a flow of fluid that should match the laboratory measured value.

OUTLET

ANSYS CFX differentiates between an *opening* and an *outlet* boundary condition. An opening boundary allows the flow to travel in and out of the domain freely but requires additional information in the form of temperature. An outlet, however, constrains the flow out of the domain but only requires a value of mass flow or pressure. The following combinations of inlet and outlet boundary specifications can be made (ordered from the most robust to the least):

1. Mass flow/velocity in – Static pressure out,
2. Total pressure in – Mass flow/velocity out,
3. Total pressure in – Static pressure out.
4. Static pressure in – Static Pressure out

of which, the third combination of boundary conditions was used exclusively in this thesis. The static pressure at the outlet was specified as an *average* over the exit area. Thus, although the pressure at each boundary node was different, the nodal average was strictly enforced. Atmospheric pressure was used as the exit boundary condition since the length and width of the exit duct in the model was identical to the duct used in the lab.

NO-SLIP WALLS

All of the remaining surfaces in the model that were not used for inflow or outflow were set as walls with a non-slip condition. This simply means that the velocity relative to the motion of the wall is set to zero. Modelling the boundary layer that results from the non slip condition requires

more consideration, however. ANSYS CFX makes use of wall functions to impose conditions near the wall on the basis of empirical equations without fully resolving the boundary layer. Since resolving the boundary layer requires a very fine mesh, the wall function approach eases computational time. In boundary layer theory, a wall function is a logarithmic relation that describes how the tangential velocity of the fluid is related to the wall shear stress in the ‘log-law region’ of the boundary layer. Equation 5.21 provides the typical formulation of the wall function that describes the profile in the logarithmic layer.

$$u^+ = \frac{U_t}{u_\tau} = \frac{1}{\kappa} \ln(y^+) + B \quad (5.21)$$

where u^+ is the dimensionless velocity profile, y^+ is the dimensionless distance from the wall given in equation 5.20, U_t is the mean velocity parallel to the wall, and u_τ is the shear velocity given by $\sqrt{\tau_n/\rho}$. The constant B is a log layer constant that depends on the wall roughness [46]. The relationship of equation 5.21 is shown graphically in Figure 5.7 with a dashed line. The experimental data (solid lines) demonstrate that a wall function approach does well to accurately model the reality of the flow near the wall.

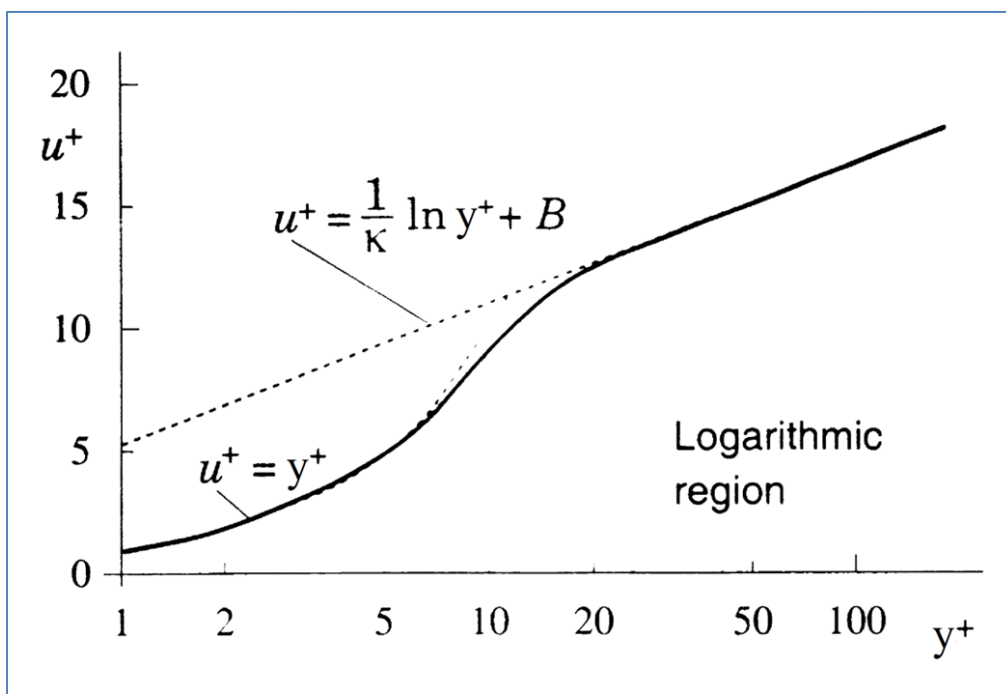


Figure 5.7: Log-law in the turbulent boundary layer [47]

DOMAIN INTERFACE

In CFX, a domain interface provides a method of connecting two fluid regions together to permit an exchange of flow without affecting conservation. Thus, ideally the interface should be entirely invisible, having no influence on the flow transferred from one domain to the next. A *periodic interface* is used when a geometric feature such as a rotor blade can be repeated around an axis (rotational periodicity) to form the full rotor. In the current model, the rotor and nozzle vane meshes were produced by repeating the individual blades 12 and 24 times respectively. The TurboGrid meshing program is designed to produce nodes on the periodic surfaces that match up when the mesh is repeated. This not only provides a seamless mesh but also avoids any error in solving the equations at the interface connection.

The *General Grid Interface* (GGI) in CFX permits a fluid-fluid connection between two domains with different element types, node locations, and grid spacing while still maintaining conservation and solver robustness. This algorithm also allows for small gaps between domains, as well as any minor mismatch between the surfaces. A list of the different locations in the model where a GGI connection is used is provided in Table 5.2. Note that there are two locations where an interface is made between a stationary and moving component. The nozzle/rotor interface and the rotor/exit-duct interface must therefore include a *frame change model* to account for the fluid moving from a stationary frame to a moving frame (or vice-versa). Two main frame change models were utilized in this thesis. The simplest model is a 'frozen rotor' model where the relative orientation of the meshes across the interface is fixed. In CFX, this does not mean that the fluid properties are averaged at the interface. The flow is simply translated across the stationary interface while adjusting the frame of reference of the fluid to represent the domain movement. Clearly, this is only useful where the flow in the rotor passages is quasi-steady and largely unaffected by the rotation of the movement. This can only be safely assumed if the flow into each of the turbine inlets is constant (steady-state) and equal in magnitude. Thus, the frozen rotor approach was used to model the flow under such conditions.

Once the flows into each inlet start to differ appreciably, the frozen rotor method is no longer appropriate. The flow in each blade passage will be influenced by its physical rotation relative to the two different feeding sectors. Therefore, a fully transient model is needed for the unequal admission cases. This is provided in CFX by a sliding mesh interface referred to as the *transient rotor-stator model*. The user sets the length of the timestep in the solver which, along with the speed of rotation, determines the rotational distance travelled between time steps. This model is the closest to reality as it simulates the true operation fluid interaction with the moving rotor blades.

Unfortunately, it also greatly increases the computational time since convergence is sought at each timestep, and the rotor must turn approximately two revolutions before a periodic solution is achieved.

Table 5.2: Summary of Boundary Conditions

Boundary Type	Location	Specified Condition
Inlet	Upper Entry Lower Entry	Total Pressure Total Temperature
Outlet	Exit Duct Exit	Static Pressure (average enforced)
Wall	Volute Wall Nozzle/Rotor Hub, Shroud & Blades Rotor Conical Nut Exit Duct Wall	Smooth, No Slip
Periodic Interface	Between Rotor Blades Between Nozzle Vanes	Conservation Enforced
GGI Interface	Volute Exit- Nozzle Inlet Nozzle Exit – Rotor Inlet Rotor Exit – Exit Duct Inlet	Conservation Enforced

5.4 Equal Admission, Steady-Flow Modelling

The first task was to run a series of steady-state, equal admission simulations that covered the range of experimental test conditions. This was necessary for two main reasons. First, it was important to be able to compare the performance predicted from the model with the values measured in the lab. This provides some level of model verification, but also should indicate where the simulation fails to predict the observed performance. Secondly, the simulation results from these runs will be used as a basis of comparison with the unequal admission simulations. In situations where a comparison of loss generation of flow development is needed, it is useful to have some base-line performance state with which the comparison can be made.

5.4.1 Performance Prediction

Table 5.3 provides a summary of the general parameters set for the equal admission simulations. Most of these selections have been discussed in sections 5.2 and 5.3.

Table 5.3: Equal Admission General Simulation Settings

Simulation Type	Steady-state
GGI Interface Model	Frozen-Rotor
Turbulence Model	k-epsilon
Wall Functions	Scalable
Working Fluid	Air, Ideal Gas
Advection Scheme	High Resolution
Heat Transfer Model	Total Energy
Residual Target for Convergence	5E-06

A total of five experimental operating points were selected for the CFD simulation. Since it was important to be able to directly compare the simulation with the experiments, the pressure and temperature measurements made in the lab were used to specify the boundary conditions in the simulation. The boundary conditions for each of these five simulations are listed in Table 5.4. All the simulations corresponded to tests made at a single speed parameter of 48.3 rps/VK. This speed produced the optimal turbine operating condition at a pressure ratio of ~ 2.0 (Case 2 below). Recall that this operating point also formed the basis of comparison with the unequal admission experimental testing (point of intersection in Figure 4.11). It was therefore important to obtain a CFD solution under this optimal set of conditions as a baseline state from which off-design comparison can be made.

Figures 5.8 and 5.9 compares the results of the CFD simulation with the experimental performance parameters measured in the lab. To begin, consider the swallowing capacity characteristics of the CFD model compared to the actual turbine. Total pressure inlet boundary conditions were used so that mass flow in each passage that resulted could be used to provide some confidence in the model. It is apparent from this figure that the CFD predicts the mass parameter characteristics with excellent accuracy. This immediately suggests that the geometry of the turbine in the CFD model accurately represents the real turbine. For example, if the nozzle or rotor passage inlet and exit areas were significantly different, this should be reflected in the level of agreement in Figure 5.8. Secondly, this result also suggests that the model itself performs very well as a predictive tool – at least in terms of the mass flow characteristics. Although this figure is not a rigorous validation, it does provide a promising indication that the CFD does well to model overall turbine behaviour.

Table 5.4: Equal admission simulation boundary conditions

	Case 1	Case 2 Design point	Case 3	Case 4	Case 5
Total Pressure (Pa) (Lower Inlet)	259358	210792	159073	146303	134649
Total Temperature (K) (Lower Inlet)	356.2	337.9	318.34	313.3	315.85
Total Pressure (Pa) (Upper Inlet)	258085	209949	158700	146002	134605
Total Temperature (K) (Upper Inlet)	357.6	342.12	320.34	314.6	317.23
Exit Static Pressure (Pa)	101912	101525	101832	101832	102525
Turbine Speed (rpm)	54827	53344	51779	51224	51825

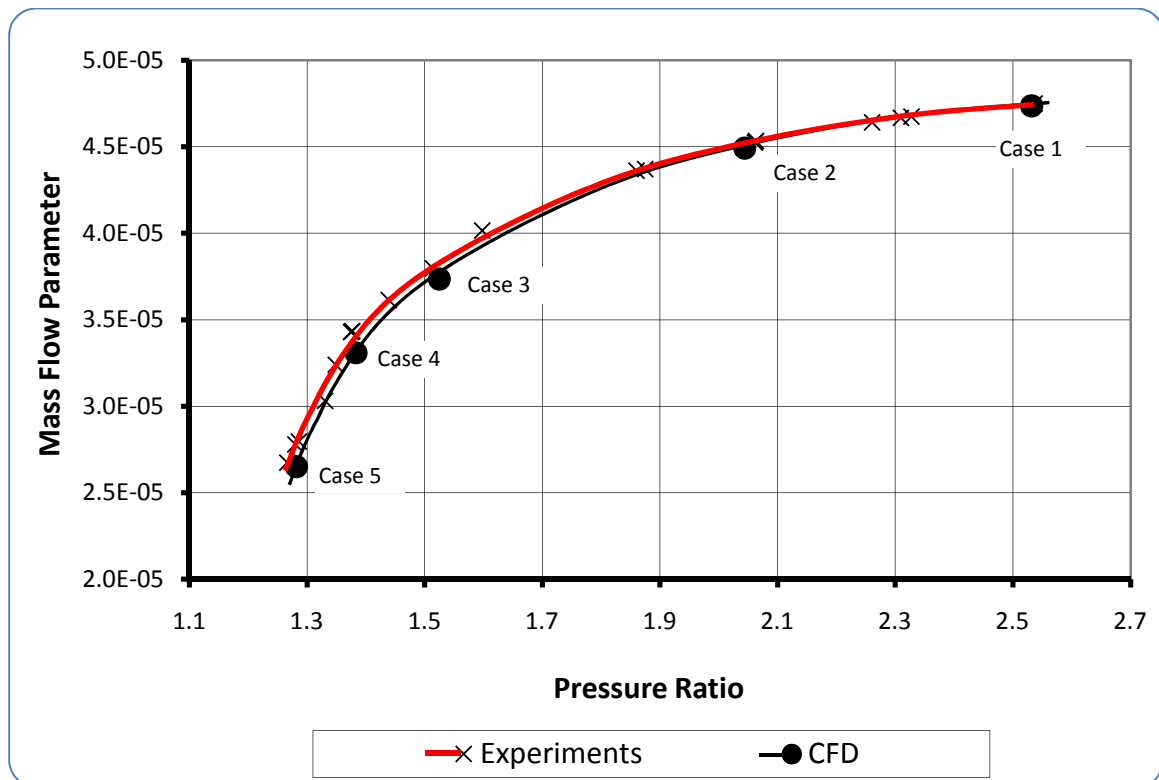


Figure 5.8: Mass Parameter versus Pressure Ratio for the CFD simulations

In addition to the mass flow prediction, the torque on the turbine wheel can be calculated by integrating the pressure forces around each blade surface. Power and efficiency can then be derived and compared to the experimental values. Figure 5.9 shows the total-to-static efficiency predicted by the CFD simulation in comparison to the experimentally measured efficiency. The level of agreement is not as close as noted for the mass flow characteristics, but the simulation does show the trend of the experimental efficiency very well. The simulation shows a peak in the performance of the turbine at the same velocity ratio as the experiments but with some over-prediction. As velocity ratio increases, the CFD simulations show a drop in performance similar to the behaviour measured in the lab. At the highest velocity ratio, the model under-predicts the efficiency by approximately ten percentage points. The relatively high discrepancy can be partially attributed to the greater sensitivity of the efficiency calculations under these low power conditions.

Nonetheless, it is apparent that the CFD does not completely model the loss generation in the turbine. This is, to some degree, expected since there are a number of areas where precise loss modelling is difficult, or simply not included in this simulation. The following are a few areas that could lead to a discrepancy between the CFD and experimental turbine efficiency:

1. The constant pressure inlet boundary condition results in a uniform inlet velocity in the model. In reality, the inlet velocity profile is not symmetrical about the axis of the inlet duct due to the presence of bends in the pipework. Although this non-uniformity is not severe, work by Hellstom [48, 49] has demonstrated that inlet flow disturbances such as Dean Vortices due to manifold bends can impact the entire flow field in the turbine.
2. The gap that exists between the rotor and the backplate of the turbine is not included in the mesh. Therefore, disk windage loss as a result of the friction of the fluid present on the back face of the turbine disk was not modelled.
3. Although the rotor-tip clearance in the model is representative of the real turbine, to fully resolve the tip leakage flow losses in this region would require a very fine mesh density in this area.
4. Frictional loss in a turbine passage is a strong function of surface roughness. However, the no slip condition used in this model assumes a smooth surface, which could lead to an under prediction of passage loss.

Thus, there are a few specific instances where the CFD model is likely to over predict the performance of the turbine. This appears to be the case for all but the final point at the highest velocity ratio. Here, the large values of the inlet flow angle often lead to flow separation and boundary layer growth which are more difficult to precisely model on a course mesh.

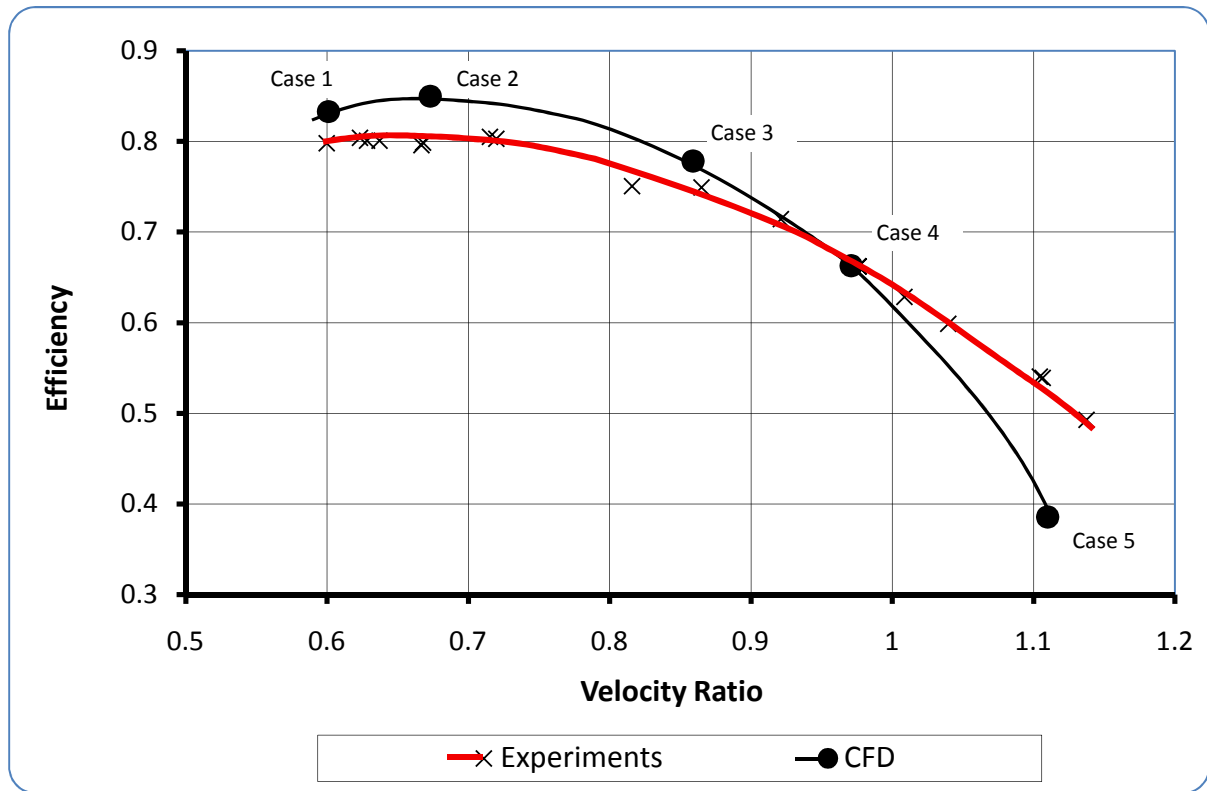


Figure 5.9: Total-to-static efficiency of the CFD simulations

Losses in an equal admission (or single entry) turbine are usually split up into the following categories: Incidence loss, passage loss, tip clearance loss, windage loss and exit losses. *Passage loss* is quite a generic term that typically includes all the losses that occur in the blade passage. This includes secondary flows, skin friction and boundary layer growth as well as other disturbances such as flow separation. As such, incidence loss and passage loss are clearly quite closely related since a non-ideal relative inlet flow angle can lead to such flow disturbances in the blade passage. Due to the importance of the angle of incidence in determining the performance of the turbine, it is useful to study the flow angles at various locations along the passage from inlet to outlet. Four monitoring locations were selected as shown in Figure 5.10 along the mid-span of the nozzle and rotor passages. Points A and B monitor the inlet and exit to the nozzle respectively, and points C and D monitor the inlet and exit to the rotor respectively. In fact, these points represent a line formed from a full revolution around the axis so that the distribution of flow angle along this line could be plotted as in Figure 5.11 or averaged in Table 5.5.

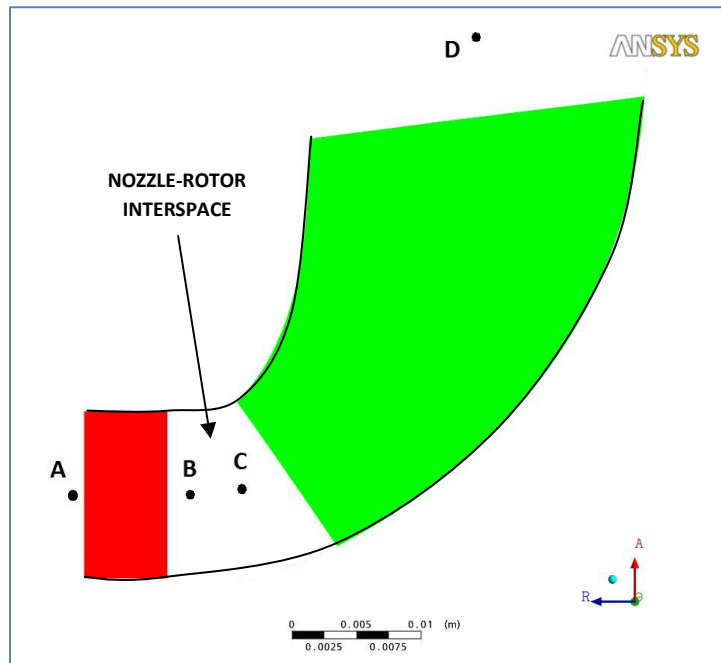


Figure 5.10: Points all the mid-span line flow angle analysis

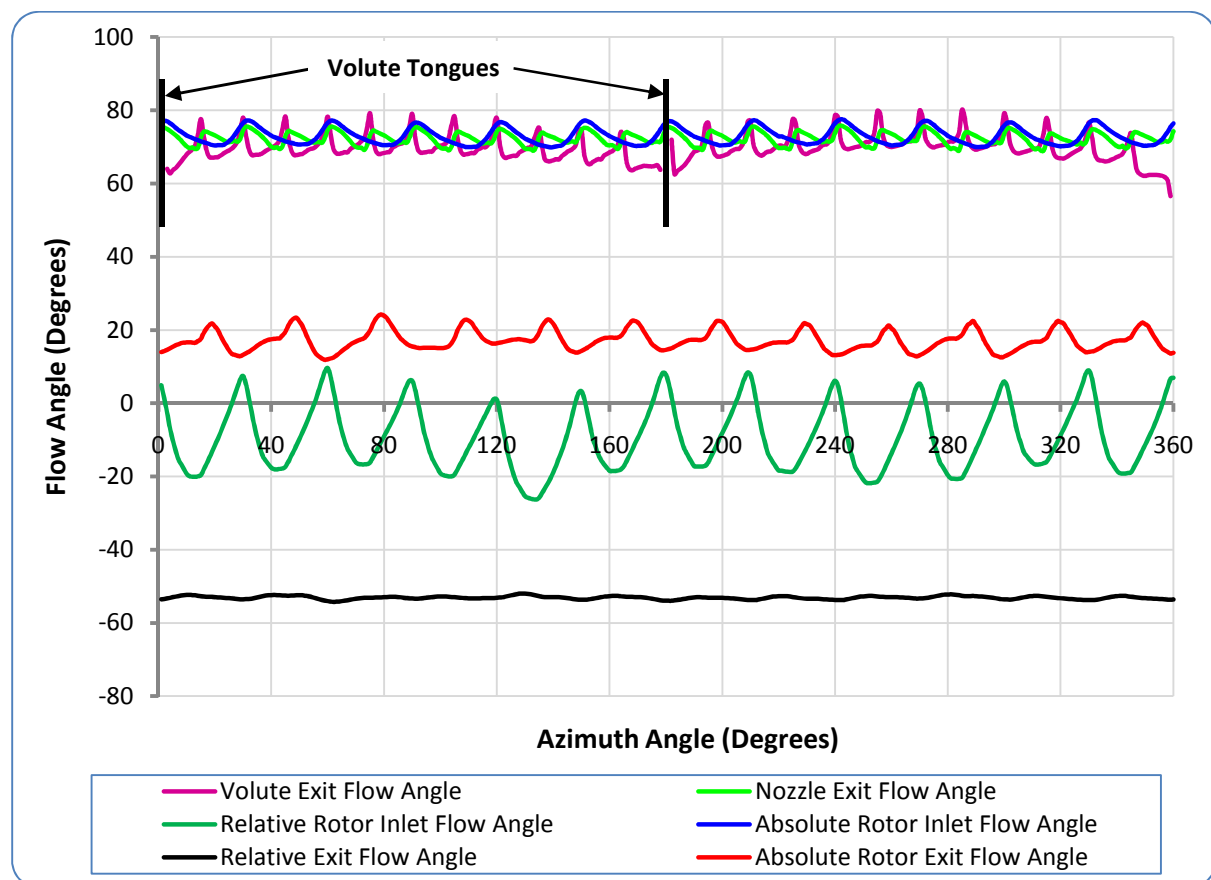


Figure 5.11: Velocity flow angles around the circumference of the turbine (Case 2)

Figure 5.11 plots the velocity flow angles against the azimuth angle for the optimal condition shown in Figure 5.9 (Case 2). The small periodic variations in data demonstrate the influence of the physical presence of the turbine and nozzle blades on the flow through the turbine. The data has been presented such that the volute tongues occur at an azimuth angle of 0° and 180°. These two tongues appear to have little effect on the flow angle in these regions due to the close match with the corresponding nozzle vanes (Figure 1.6). Also, note the close agreement between the flow angles in the inlet section of the turbine. The volute exit angle, the nozzle exit angle and the absolute rotor inlet angle all lie within a few degrees of 70°. This is not entirely surprising, considering that the flow at the exit from the volute should match the nozzle stagger angle of 70° (Table 3.2). The relative inlet flow angle β_3 at this optimal operating point varies quite significantly around the circumference due to the presence of the rotor blades but has a circumferentially average value of -9.34° at midspan. Recall from the turbomachinery equation (eq. 1.5) that in order to maximize the available energy, the relative velocity at exit W_4 should be maximized, and the absolute velocity at the exit C_4 must be minimized. This can be achieved by reducing the exit swirl angle α_4 . However, Figure 5.9 shows that there is a small amount of exit swirl (17°) at the midspan location under these conditions (this value can vary quite significantly from the hub to shroud). Some authors have suggested that a small amount of exit swirl can be beneficial in aiding the exhaust diffuser to recover some of the exit kinetic energy [50].

Table 5.5 provides the circumferentially averaged flow angles for all of the equal admission cases. The absolute inlet angles at points A, B and C all remain approximately constant, irrespective of the three different steady-state conditions. This proves the findings of Karamanis et. al. [6]: that the absolute flow angle at the rotor inlet is purely a function of the volute design, not a result of the inlet flow conditions. Thus, the nozzles do not provide any significant turning of the flow in the turbine. However, there does seem to be a slight change of flow angle across the nozzle vanes of approximately 2-3 degrees. This is consistent with a physical effect known as *nozzle turning*. Due to the boundary layer growth in the nozzle passage followed by a sudden expansion at the exducer, the flow leaving the nozzle vanes will not follow the nozzle angle precisely. There are a few different empirical relations for this effect, but the equation by Mizumachi et.al. [51] seems most widely accepted:

$$\alpha_3 = 7.2 + 0.92 \cos^{-1}(o/s) \quad (5.22)$$

where α_3 is the nozzle exit angle, o is the throat opening and s is the vane spacing. If the nozzle geometry measurements are substituted into this equation, an exit nozzle angle of 72.6° is suggested which is reasonably consistent with the average values in Table 5.5.

Seeing that the absolute inlet flow angle is independent of inlet conditions, the incidence loss must be due to a non-ideal *relative* inlet flow angle. Table 5.5 shows the large change in the average relative inlet flow as the pressure ratio across the turbine decreases. Over the range of conditions tested, the CFD predicts an 80.4° swing in the angle of the flow driving the turbine wheel. It is therefore not surprising that this should result in a turbine that is unable to operate efficiently with such a variation. The absolute exit flow angle α_4 also increases quite significantly as the velocity ratio is increased. From equation 1.5, this will increase the exit swirl $C_{\theta 4}$, leading to further loss in the available energy.

Table 5.5: Circumferentially Averaged Equal admission Flow Angles (50% span)

	Case 1	Case 2 Design point	Case 3	Case 4	Case 5
Volute Exit Flow Angle (Average: Circumference A)	69.60	69.76	67.988	68.102	68.34
Nozzle Exit Flow Angle (Average: Circumference B)	72.10	72.35	72.378	72.364	72.27
Relative Rotor <i>Inlet</i> Flow Angle (Average: Circumference C)	7.38	-9.34	-52.46	-63.135	-73.00
Absolute Rotor <i>Inlet</i> Flow Angle (Average: Circumference C)	72.66	73.02	73.4	72.882	71.587
Relative Rotor <i>Exit</i> Flow Angle (Average: Circumference D)	-55.64	-53.11	-52.18	-51.60	-80.69
Absolute Rotor <i>Exit</i> Flow Angle (Average: Circumference D)	-6.69	17.15	50.275	58.675	72.08

5.5 Unequal Admission, Steady-Flow Modelling

5.5.1 Unequal Admission Boundary Conditions

One of the main aims of this Chapter is to study the detailed flow physics that result from introducing two different flows into each inlet. 3-D CFD simulation is a powerful tool to do this as it provides flow details at each point in the discretized turbine domain. Like equal admission, the

boundary conditions selected to model the unequal admission performance were taken directly from measurements made in the experiments. In this way, the results from the CFD could be directly compared to the experimentally measured results. Due to the unsteady nature of the flow in each rotor passage, each of the unequal admission cases required a fully transient solution. Thus, most of the pre-processing settings listed in Table 5.3 for equal admission were used for the unequal admission simulations with the exception of transience. The interface between the nozzle and the rotor was set as “Transient Rotor Stator” which explicitly rotates the mesh a fixed distance at each timestep. Since this type of simulation requires an initial fluid condition from which to start the transient run, a converged frozen rotor solution with the same boundary conditions was used for this purpose.

Table 5.6: Unequal admission simulation boundary conditions

	Case 6	Case 7	Case 8	Case 9
Total Pressure (Pa) (Lower Inlet)	199407	211063	197334	204222
Total Temperature (K) (Lower Inlet)	320.66	335.05	331.61	348.03
Total Pressure (Pa) (Upper Inlet)	<i>WALL</i>	130799	149485	264643
Total Temperature (K) (Upper Inlet)	333.52	334.99	334.88	355.05
Exit Static Pressure (Pa)	98925	101031	98125	101658
Turbine Speed (rpm)	49431	48120	47559	55984

Table 5.6 lists the set of boundary conditions applied for each of the four unequal admission simulations labelled cases 6-9. These cases all correspond to conditions on the blue line shown in Figure 4.11 with a constant velocity ratio equal to 0.65. In order to simulate partial admission (case 6), the upper inlet was simply set as a non-slip, wall boundary condition. This choice was not immediately clear since a pressure above atmospheric was measured at this location in the real turbine despite being completely blocked upstream. By blocking this inlet in the simulation, this allowed the pressure to develop in the stagnant entry in a similar way as the experimental case. If a standard pressure boundary had been applied here, this would have resulted in an unrealistic mass flow induced through this inlet. Finally, in order to test the less severe unequal conditions, Cases 7

and 8 correspond to unequal admission conditions with ratios of inlet pressure below one and Case 9, above one.

5.5.2 Effective Area

Figure 5.12 and 5.13 show the effective area of the turbine calculated from the mass flow through the upper and lower inlets in the simulation. Like the equal admission runs, the CFD simulation seems to model the mass flow characteristics very well. Figure 5.13 also shows that the interaction of the two inlets that results from the unequal admission also seems to be effectively modelled. The trend of a reduced swallowing capacity as the pressure in the second inlet is increased can be clearly seen in this plot. This was an important effect to be able to model and therefore allows us to examine the physical effect that is driving this trend.

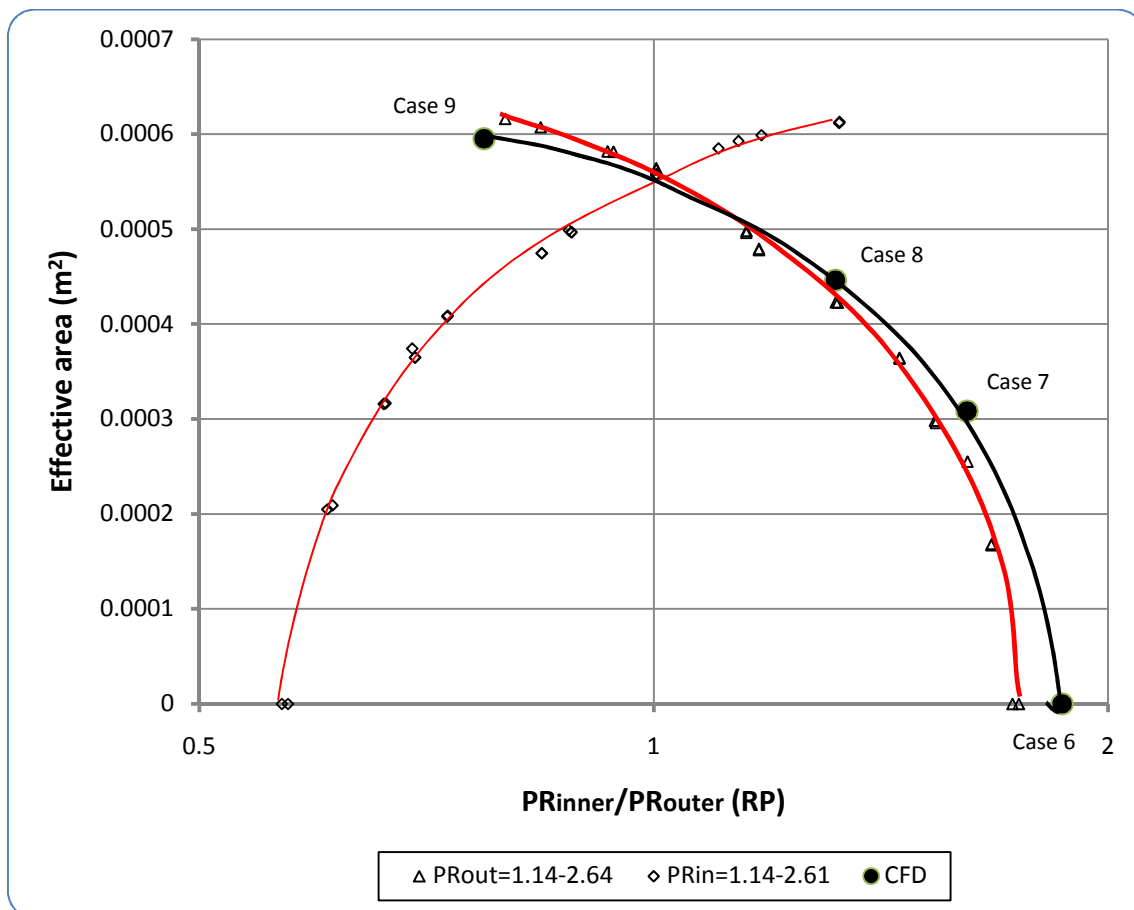


Figure 5.12: Effective area of the inlet with **varying** pressure calculated from the CFD simulation ($U_3/C_{is}=0.65$)

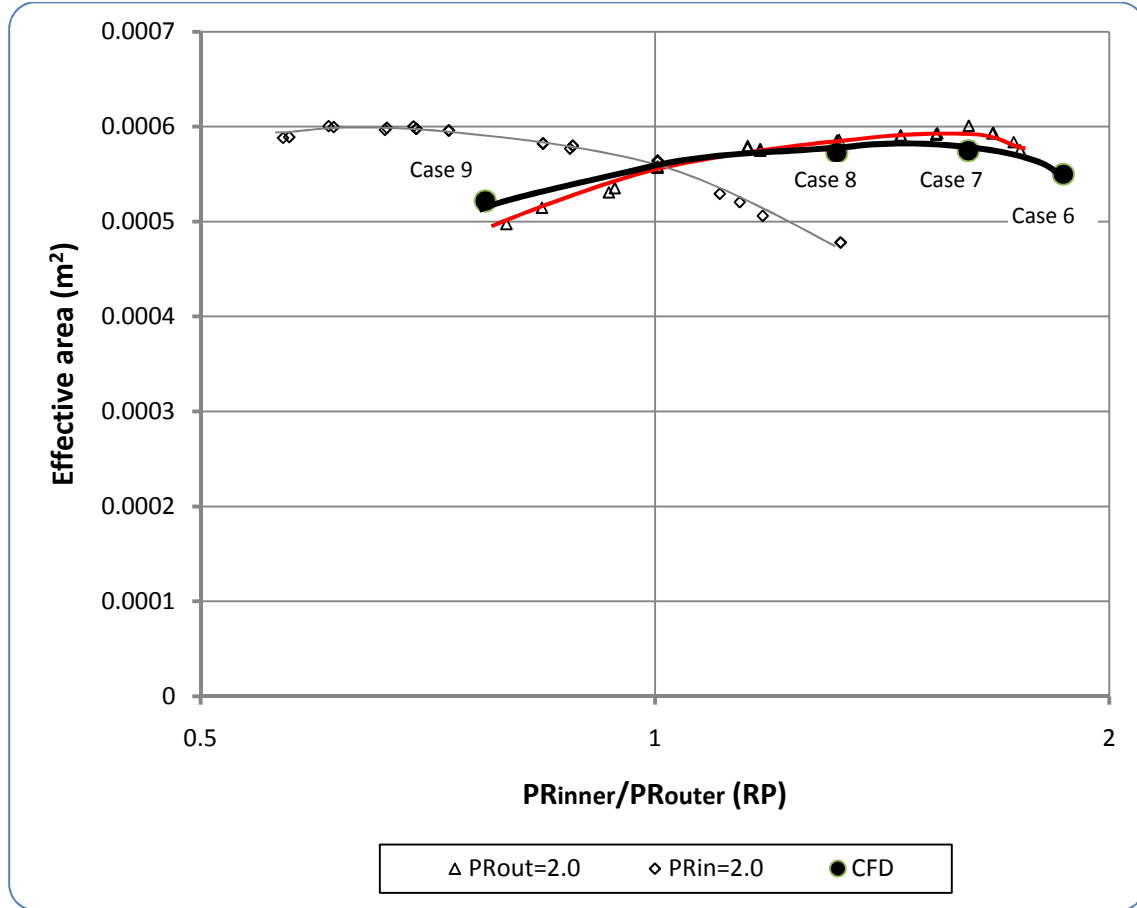


Figure 5.13: Effective area of the inlet with **constant** pressure calculated from the CFD simulation ($U_3/C_{is}=0.65$)

Each of the passages in a double-entry turbine is completely isolated from the inlet of the volute to the exit of the nozzles. At the exit to the nozzle, the fluid from both inlets expands into an interspace that runs the circumference of the rotor inlet. This small volume formed between the nozzle and the mixed flow rotor (see Figure 5.10) is the first area where interaction between the flows can occur. In order for the mass flow in one passage to increase while maintaining a constant inlet pressure (Cases 6-8), the static pressure at the exit to the nozzles (that are connected to this inlet) must decrease. Conversely, if the mass flow decreases while maintaining a fixed inlet pressure (Case 9), the pressure at the exit to the nozzles must have increased. To test this, the pressure around the circumference of the interspace (point B in Figure 5.10) was plotted against the azimuth angle for unequal cases 7 and 9. The first 180 degrees represents the nozzle sector connected to the inlet with a 'constant' upstream pressure (lower inlet). In fact, since these pressures were slightly different in each case, the values provided in Figure 5.14 were normalized against the lower inlet pressure for comparison purposes. Focusing on this first 180° sector with the constant upstream pressure, there are two areas where the pressures at the exit to the nozzles are appreciably

different. Both of these areas (labelled features Y and Z) occur around the tongues where the two flows are most likely to interact. From this plot of normalized pressure, it is clear that the decrease in pressure in the upper inlet (from Case 9 to Case 7) causes the static pressure to drop in the first and last 3 nozzle passages in the lower inlet. This is especially significant at the first tongue (feature Y) which shows an approximately 0.4 bar difference in static pressure in the first nozzle passage. Averaging around the 180° sector, the drop in static pressure due to the influence of the upper inlet pressure is less than this (~6500 Pa), but the critical areas that appear to drive the difference in the swallowing capacity are around the two tongue regions.

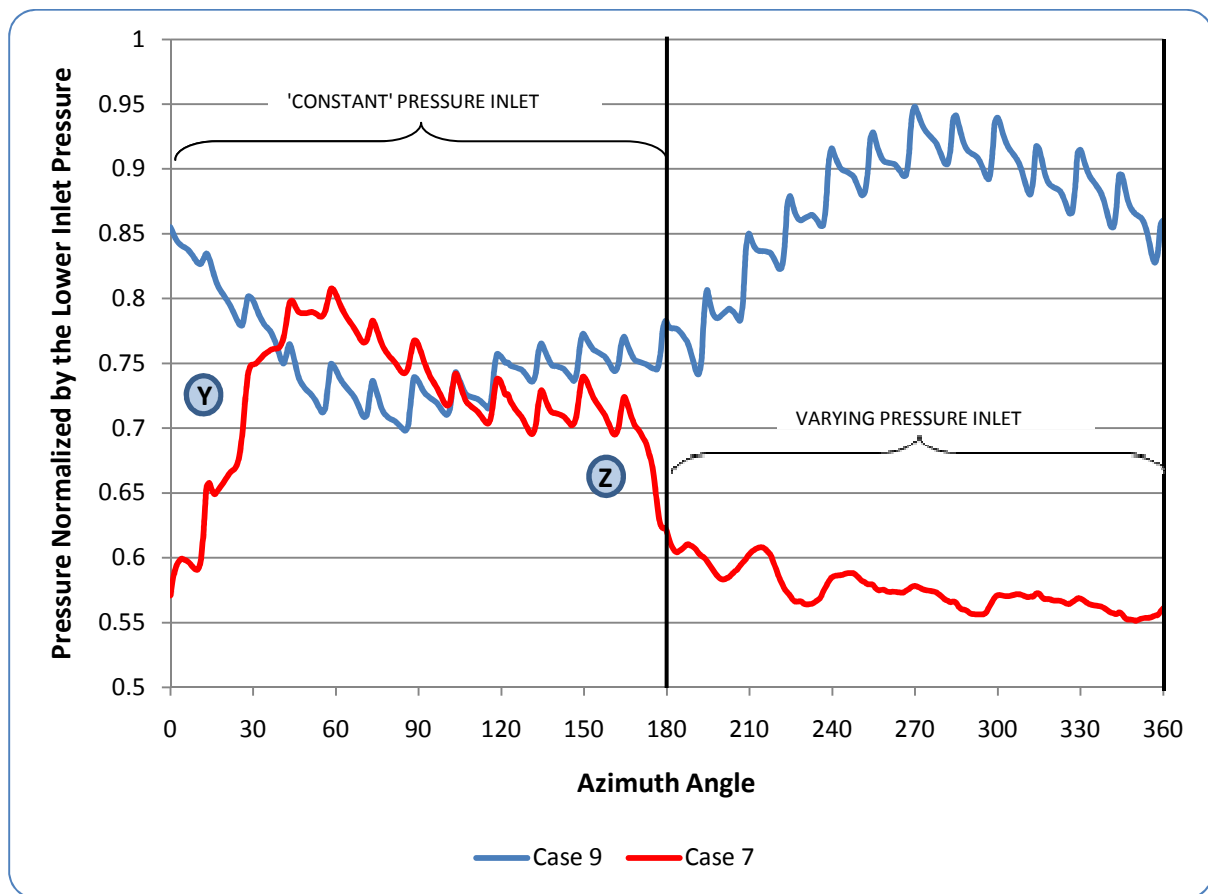


Figure 5.14: Nozzle exit static pressure (normalized by the total pressure of the lower inlet) as a function of the azimuth angle for unequal cases 7 and 9.

Figure 5.15 provides further insight into the interaction between inlets in the interspace. Here streamlines of the flow are superimposed on contours of constant normalized pressure on a flat plane located at the nozzle midspan. The streamlines are in the absolute frame of reference since the intent here is to show the flow in the interspace, not the moving turbine rotor. The streamlines where the upper inlet is at a lower pressure (Case 7) show that the flow in the lower

inlet is able to expand into the interspace to a much greater degree as a result of this reduced pressure. The streamlines in Case 9, however, show that the expansion of the flow into the interspace from the lower inlet is restricted by the high pressure field present in the upper inlet.

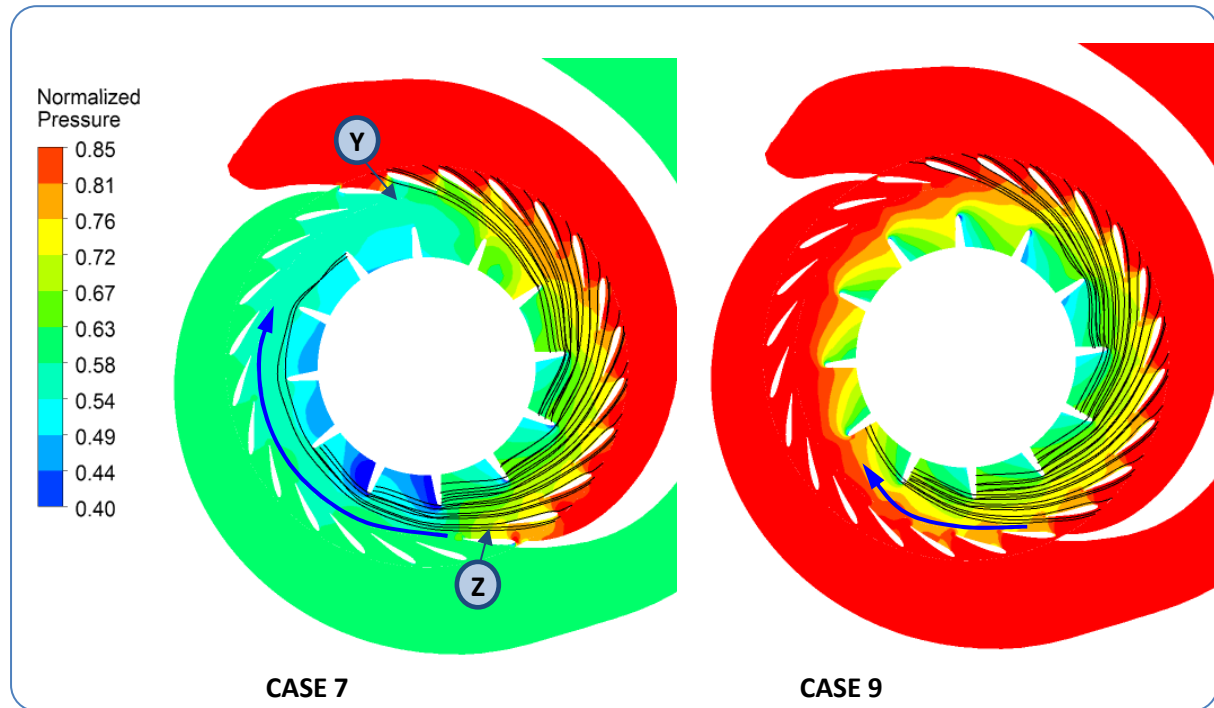


Figure 5.15: Pressure contours normalized against the inlet pressure of the lower entry

In conclusion, the CFD simulations are able to correctly predict the swallowing characteristics of the turbine when exposed to unequal admission flows. In particular, the interdependence of the swallowing capacity of the two inlets was predicted by the model, thereby allowing the analysis of the physical flow effects that drive this trend. It was found that the flow conditions in the interspace around the region of the two tongues are affected by both inlet conditions. Thus, even if one inlet is kept at a constant pressure, the flow through this inlet will depend on the fluid condition in the second inlet. In our case, lowering the pressure in the upper inlet caused the static pressure in the interspace region near the tongues to decrease, therefore leading to an increased expansion through the nozzles in this region and hence, an increased effective area.

5.5.3 Quasi-steady/Time scales

An interesting observation was made at the end of the previous chapter that leads to a question unique to the double-entry design. Since the flow from each volute passage drives a separate section of the rotor, it might be tempting to conclude that, in the main, this design can be considered as two separate turbines. Yet, it has been demonstrated thus far through experimental and computational means that there *is* some flow interaction in the interspace between nozzle and rotor – especially if there is a large pressure difference between inlets. However, a second significant question still remains. That is, what is the effect of each rotor passage being fed with a periodic variation in flow conditions as it rotates in a double-entry volute? When the double-entry turbine is fed by two steady flows that are not equal in magnitude, a given turbine wheel passage will be subject to an abrupt change in pressure, velocity and flow direction as it rotates from one 180° sector to the next. Here, the time for the fluid to travel through the rotor passage gains importance. If the fluid travels very quickly relative to the rotor rotation, the flow will be rapidly established and holds little ‘memory’ of the fluid conditions in the previous section. However, if the fluid requires a significant amount of the 180° rotation to travel through the rotor wheel, the flow in each blade passage may never have the opportunity to establish itself to an optimal condition. In the former case, the flow in the passage can be assumed to be quasi-steady, and in the latter, fully unsteady.

It is therefore significant how quickly the flow will be convected through the rotor passage. However, determining the time needed for a particle to traverse the rotor in a highly unsteady flow is somewhat challenging. First, it depends on what path the flow takes through the rotor since the flow velocity changes from hub to shroud and across the blade passage. Also, the flow velocity and direction will vary a great deal depending on the flow condition in each inlet. In order to get a sense of the range of bulk flow to travel times through the rotor, three equal admission cases were considered (Cases 1-3). The CFD results from these cases were used to calculate the circumferentially-averaged, axial velocity through the rotor as a function of the axial distance of a midspan flow path. This velocity function was then integrated to estimate the average time needed for a particle to travel through the rotor. The results of these calculations are provided in Table 5.7 along with the time taken for the turbine wheel to rotate $\frac{1}{2}$ of a revolution. This analysis suggests that the flow requires between 69% and 119% of the $\frac{1}{2}$ wheel rotation to completely travel through the wheel. Note that this fraction is actually a measure of the *Strouhal Number* (St) associated with the ‘pulses’ that are produced as the rotor is periodically fed with two different flows as it rotates. The estimate of this number implies that a steady-state, unequal admission condition in a double-entry turbine leads to fully unsteady (non-quasi-steady) behaviour in the rotor passages.

Table 5.7: Equal admission simulation boundary conditions

	Case 1 (mass flow=0.325 kg/s)	Case 2 (mass flow=0.26 kg/s)	Case 3 (mass flow=0.17 kg/s)
Blade passage bulk travel time (s)	0.000381	0.000462	0.000688
½ Rotor Rotation Time (s)	0.00054718	0.0005624	0.0005794
Fraction of ½ rotor rotation (St)	0.696	0.821	1.187

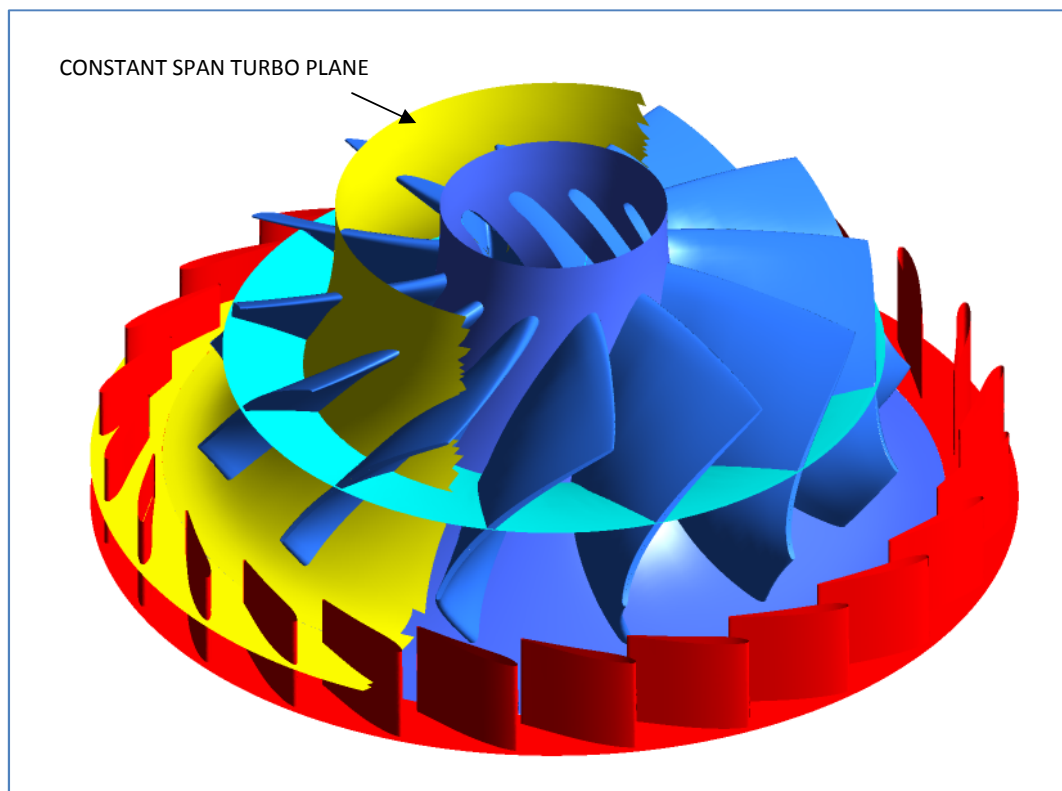


Figure 5.16: Viewing planes for studying the flow through the rotor

In order to analyse the influence of an abrupt change of flow conditions as the rotor rotates, it is useful to consider the most extreme unequal case – partial admission. Here, with one entry completely blocked, any given rotor passage will rotate 180° in a region of near stagnant flow before suddenly being exposed to a high pressure flowing fluid. By studying how the flow develops as it

moves into this flowing region, some insight can be gained in the time needed for flow establishment in the turbine blade passage. This in turn should lead to some insight into where non-ideal flow conditions lead to losses in the turbine.

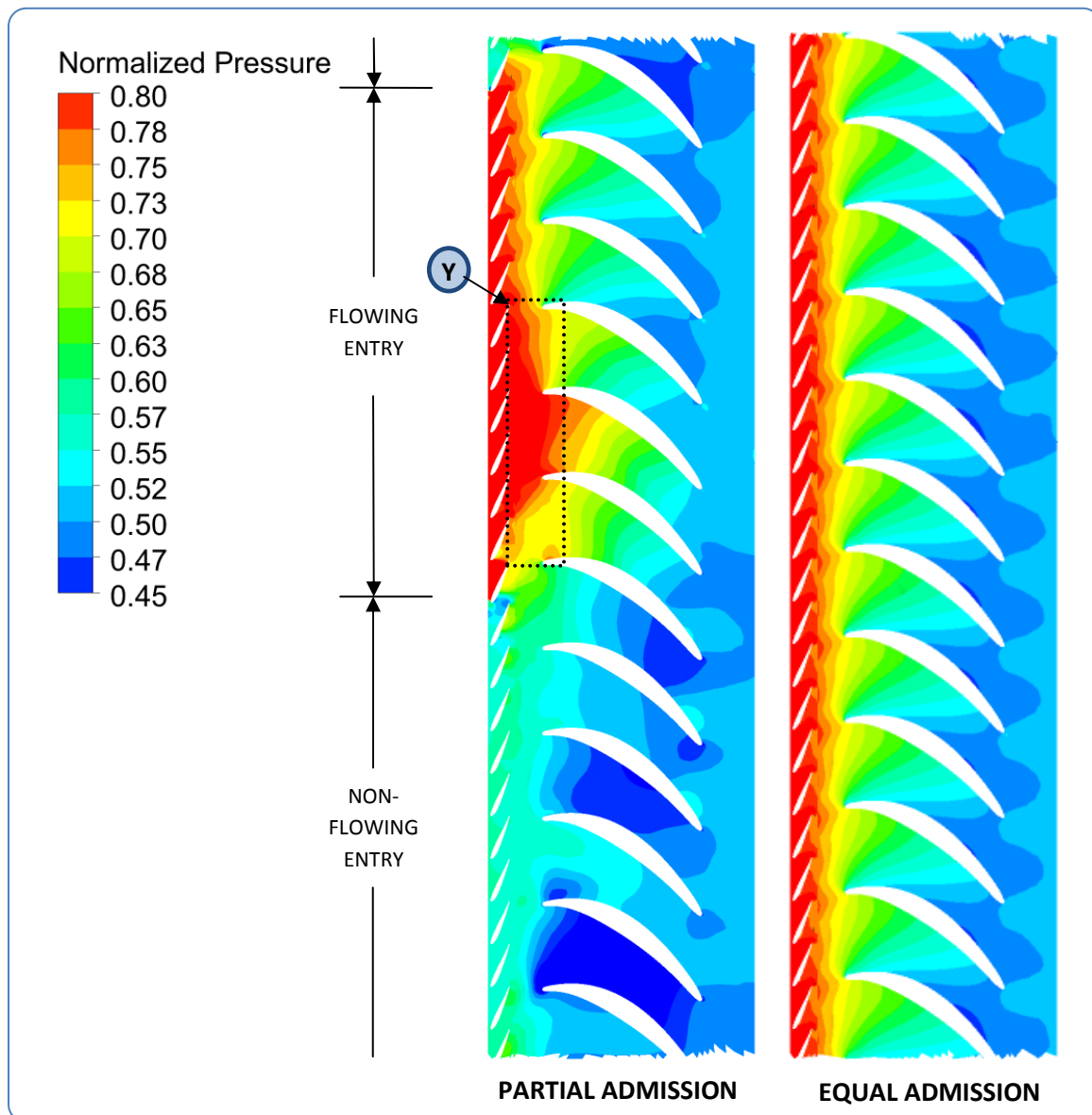


Figure 5.17: Unwrapped constant 50% span pressure contours normalized against the inlet pressure of the lower entry

One way to do this would be to study the flow in a single passage at each time step and take a series of snapshots at these instances. This is time-consuming and would require many pages of figures to demonstrate a single line of reasoning. Instead, a snapshot can be taken of all the rotor passages, making it possible to study the development of the flow in the 6 passages that are being

fed by each entry. However, representing a complex three-dimensional flow with a series of 2-dimensional figures can be difficult. The method chosen to represent the flow in this thesis will be to largely focus on the constant span plane shown in Figure 5.16. The constant-span plane (shown in yellow), creates a curved, rotated surface at a constant blade span to show the thru-flow through the rotor and nozzle passages. In order to show this plane as a two dimensional figure, it can be 'unwrapped' and flattened as shown in Figures 5.17-5.19. Note that in these constant-span plots, the rotor is rotating *up the page*.

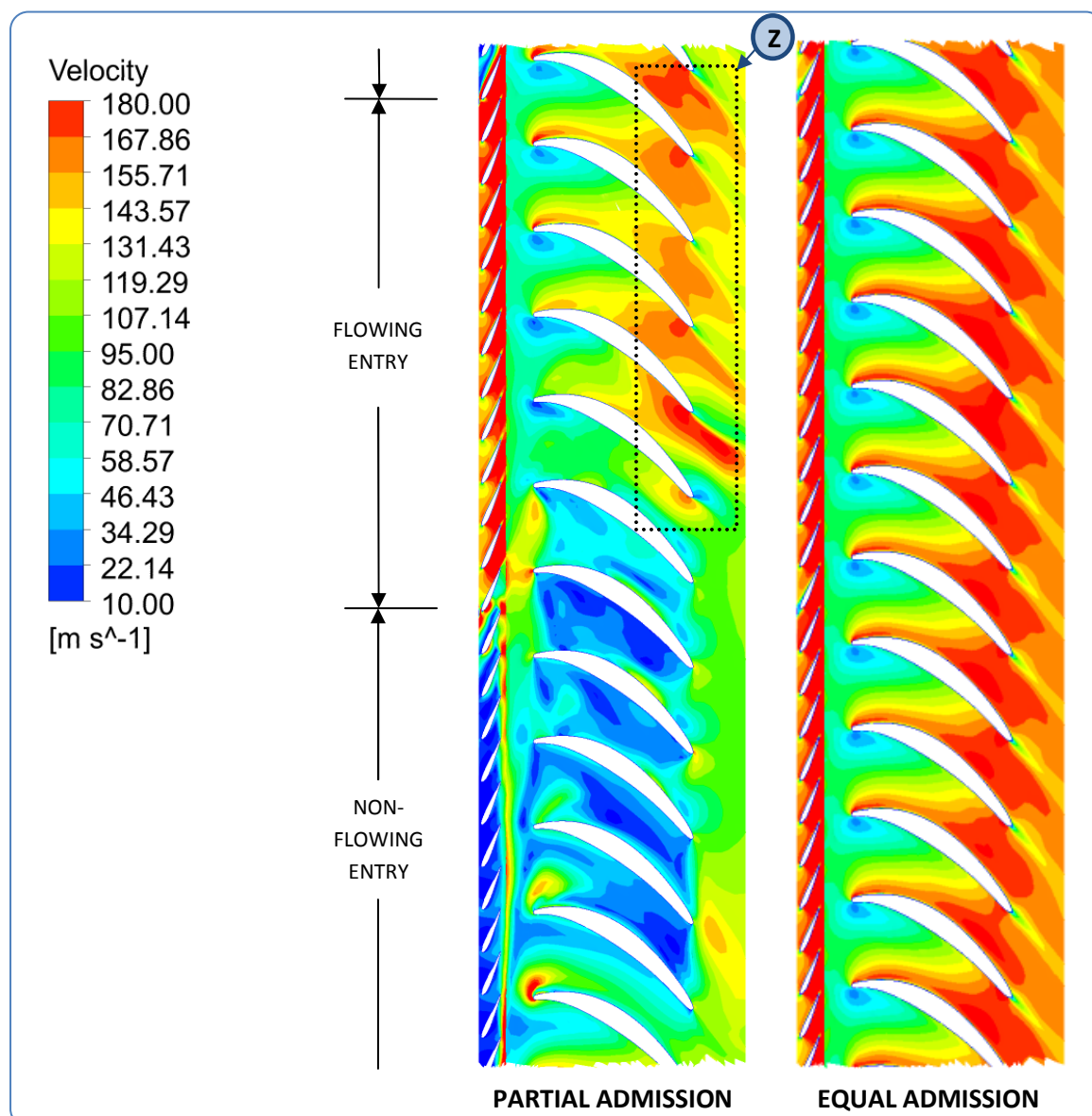


Figure 5.18: Unwrapped constant 50% span relative velocity contours

Figure 5.17 and 5.18 show the pressure and velocity contours at constant mid-span planes for equal and partial admission. First, consider the pressure contours shown in Figure 5.17 that are normalized with the lower inlet pressure to afford comparison. It is immediately apparent from this comparison that the pressure profile at the inlet of the rotor for the first three passages is significantly different than the equal admission case (see feature Y). Therefore, the pressure field rotor takes at least $\frac{1}{2}$ of the flowing section to establish conditions that are close to the equal admission conditions. Yet this plot also suggests that for the pressure field, this is largely due to a highly disturbed *inlet* pressure field located in the rotor interspace. This, therefore, is the first sign that one of the influences hindering the establishment of the flow through the rotor is the presence of a disturbed pressure field in the inlet region where the two unequal flow fields interact.

Figure 5.18 shows the contours of relative velocity in the nozzle and rotor mid-span plane. In this plot, consider the development of the partial admission exducer velocity profile (feature Z) in comparison with the equal admission case. Moving from blade to blade in the direction of the spinning rotor, none of the velocity profiles appear completely consistent. The top-most blade passage shows the greatest similarity with the equal admission case, but this has passed into the non-flowing section. This evidence supports the argument that the fluid does not have sufficient time to reach a fully developed state in each of the 180° feeding sectors. To visualize the fluid flow in the flowing section more clearly, the velocity streamlines at 25% and 50% span are plotted in Figure 5.19. Again, we will consider this plot by moving from the bottom of the figure to the top in the direction of rotor rotation. As the first blade passage arrives from the non-flowing region, there is a strong vortex resulting from the movement of the rotor in the stagnant entry (Feature W). At 50% span, this recirculation remains visible in the first blade passage in the flowing region. However, at 25% span, this recirculation still persists into the second blade passage (feature X). In addition, in line with what was noted with the pressure field in Figure 5.17, evidence of significant flow disruption in the interspace region that neighbours the non-flowing region is clearly visible. Due to the low pressure in the non-flowing interspace region, the streamlines show reverse flow (*relative* to the rotor rotation) via this vaneless space (feature Y). Importantly, this causes a disturbance in the direction of the flow entering the rotor passage which appears to persist for at least $\frac{1}{4}$ of the rotor rotation. In addition to the disturbance in the inlet region, feature Z points to a significant backflow (*relative* to the rotor rotation) at the exit to the turbine in the first three blade passages. All of these features contribute to the observation that the streamlines only really reach a smooth flow profile in the final blade passage before moving into the non-flowing section. In fact, at 25% span where the slower moving fluid must travel a longer distance, there are still indications in the final passage that the fluid has failed to achieve a fully developed flow profile.

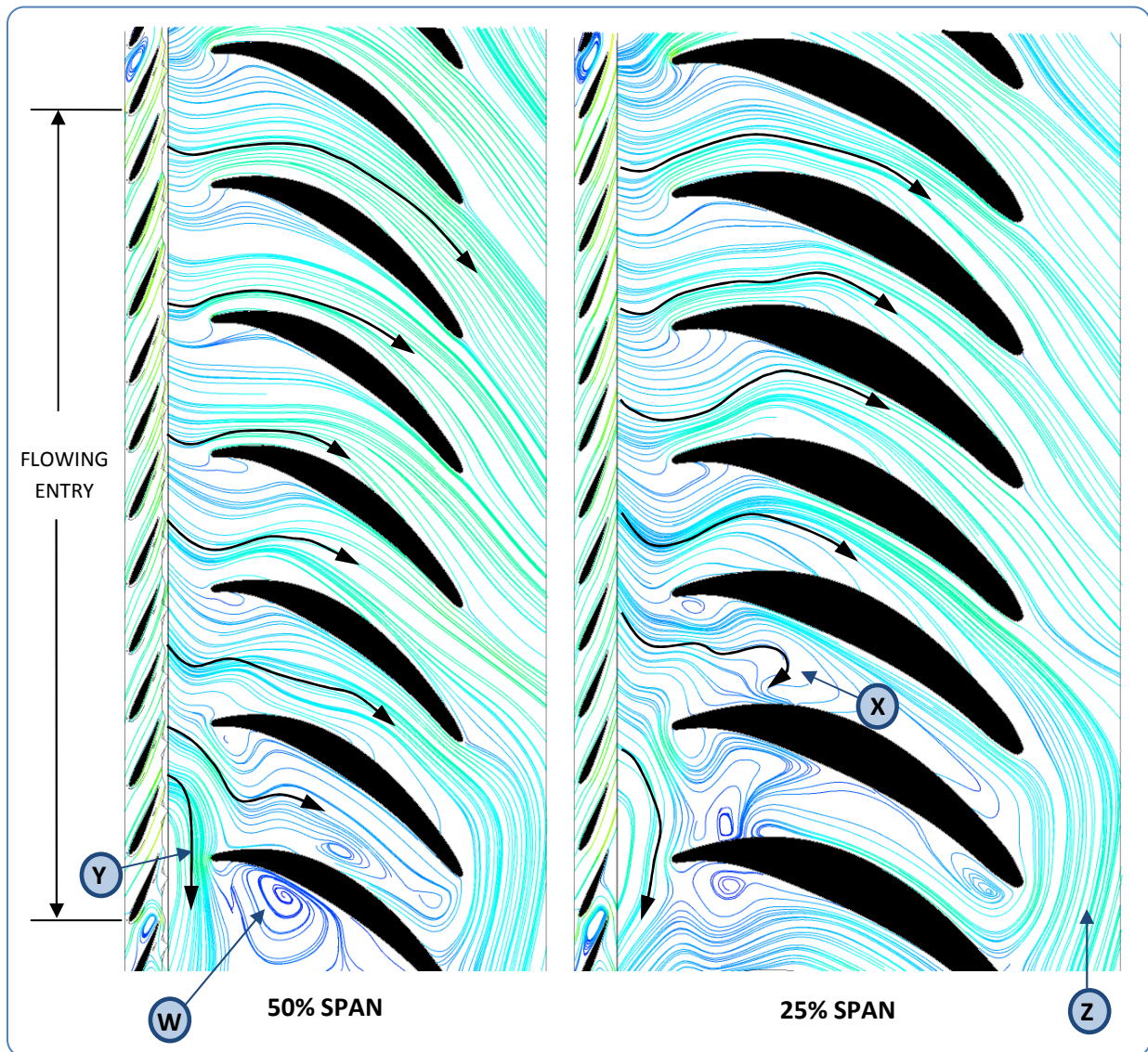


Figure 5.19: Unwrapped constant span velocity streamlines in the flowing section (partial admission)

This analysis therefore confirms that a double-entry turbine that is fed with two unequal flows will result in a fully unsteady problem. There are two main influences that drive this observation. First, under unequal admission conditions, the flow in the interspace region surrounding the tongues will be influenced by the mixing of the two flow conditions. If one of the inlets is completely blocked (the extreme case), this will result in significant flow disturbances at the start and end of each of the nozzle sectors which can adversely affect the rotor inlet conditions for a significant proportion of the rotation. Secondly, a certain length of time needed for the bulk flow in each blade to pass through the rotor. There is evidence to suggest that this through-flow time is on the same order as the time needed to rotate 180° . Thus, even without the flow disturbances in the

interspace, the flow in a blade passage may not have sufficient time to establish before moving into the next sector.

5.5.4 Efficiency

Figure 5.20 shows the efficiency of the turbine calculated by the four unequal admission CFD simulations in comparison to the experimental efficiency. Like the equal admission efficiency prediction, the CFD fails to predict the magnitude of the losses generated due to unequal admission but does demonstrate the trend quite well. At the higher pressure ratios, the model fairs very well in comparison with the experiments, but as the pressure in one inlet is reduced, the model shows an increasing difference with the experiments. This is not surprising, considering the highly complex flow features noted in the previous section that must be resolved on a relatively coarse mesh density.

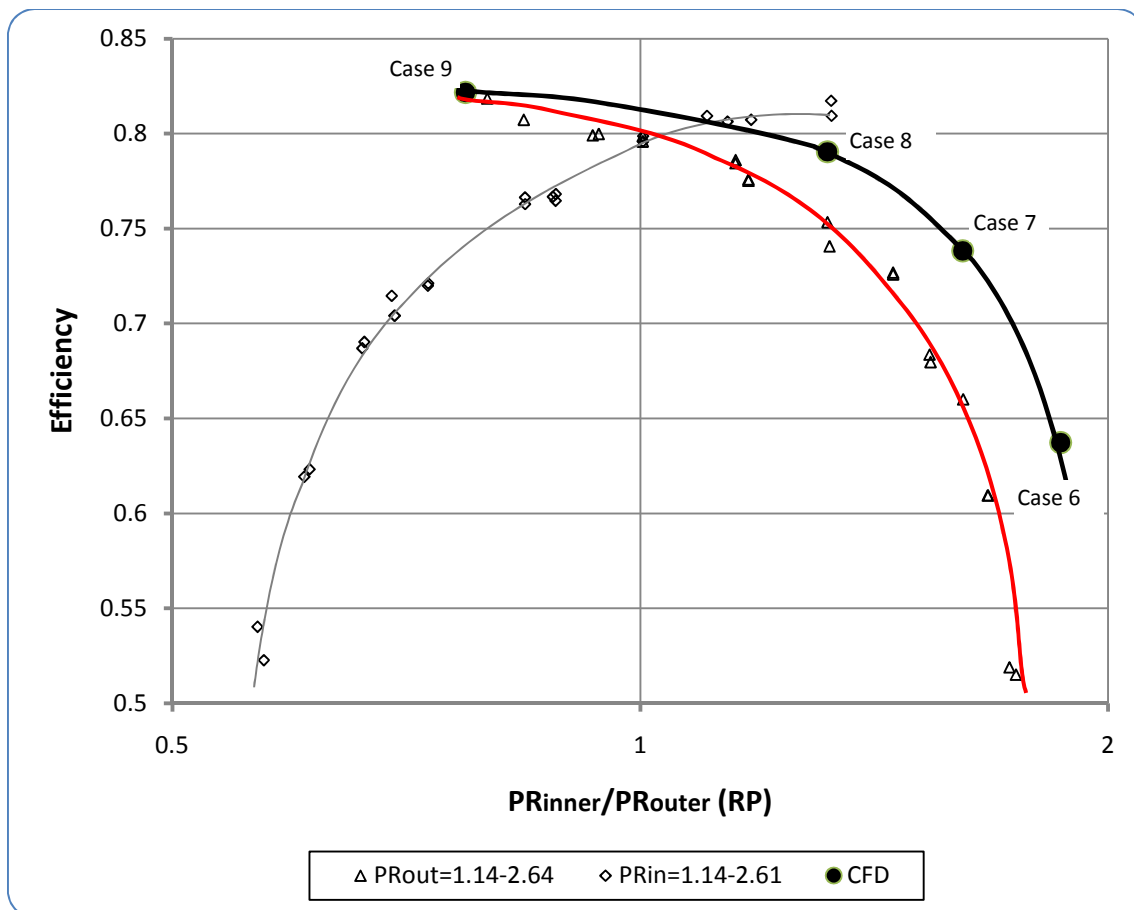


Figure 5.20: Unequal admission efficiency predicted by the CFD in comparison to the experimental efficiency

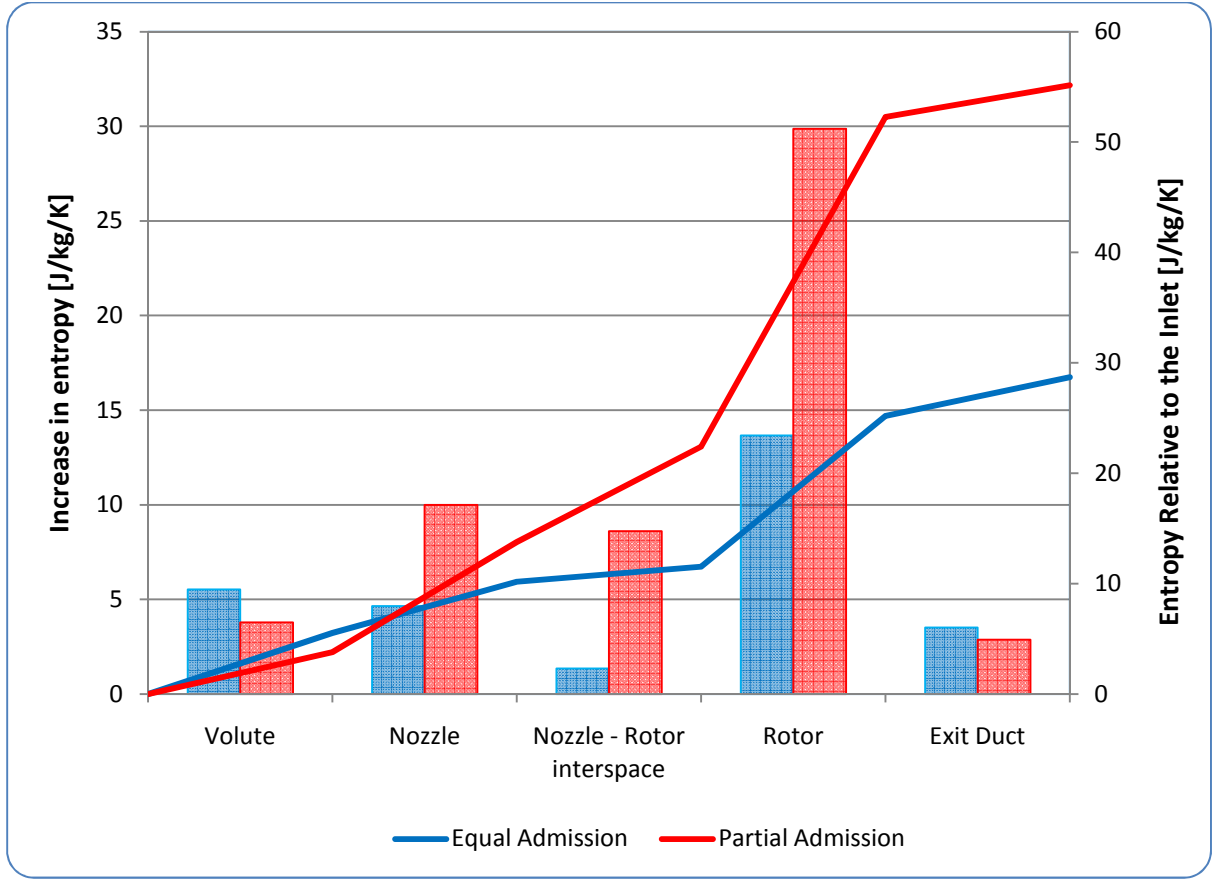


Figure 5.21: Relative static entropy increase in different areas of the turbine domain

In order to gain some insight into the loss generated under unequal admission conditions, the concept of entropy production is very useful. Equation 5.23 can be used to calculate static entropy with respect to a reference temperature and pressure (T_{ref} and p_{ref}).

$$s - s_{ref} = c_p \ln\left(\frac{T}{T_{ref}}\right) - R \ln\left(\frac{p}{p_{ref}}\right) \quad (5.23)$$

However, this equation only provides the entropy relative to a reference value s_{ref} . Thus, the best use of this equation is to calculate the difference in relative entropy in and out of a fluid domain to demonstrate the rise in entropy within the volume. The turbine domain was therefore divided into five main areas: the volute, the nozzle, the interspace, the rotor and the exit duct. The mass averaged relative entropy was calculated in and out of each of these volumes for equal and partial admission. Figure 5.21 demonstrates the result of these calculations by showing the rise in entropy with respect to the inlet (lines) and the magnitude of entropy production in each domain (bars). Moving along the abscissa from left to right follows the fluid path from inlet to exit. Starting at the

inlet, the increase in entropy through the volute is greater in the equal admission case simply because *two* flowing passages will produce a greater loss than a single flowing entry. Passing through the nozzle, however, the unequal flow shows a 53% greater entropy increase compared to the equal admission case. This difference is even larger in the interspace volume where the increase in entropy jumps by 84% from the equal to the unequal admission condition. However, by far the largest increase in the entropy occurs in the blade passages themselves. Given the strong recirculation zones noted in the streamlines in Figure 5.19, this is to be expected.

This entropy audit across the five main areas in the turbine therefore shows that the most significant entropy increase occurs between the nozzle inlet and the rotor exit. While this is valid insight, it would be even more valuable to be able to use CFD to visualize the loss creation on a local basis to pinpoint areas in the fluid domain where irreversibility is generated. Denton [51] suggests the concept of an “entropy generation rate” to calculate the loss generated in different regions of an axial turbine guide vane. The rate of entropy production in a fluid can be analytically derived from the momentum and energy equations as shown by Greitzer [52]. This derivation results in the following equation:

$$T \frac{Ds}{Dt} = \dot{Q} - \frac{1}{\rho} \frac{\partial q_i}{\partial x_i} + \frac{1}{\rho} \tau_{ij} \frac{\partial u_i}{\partial x_j} \quad (5.24)$$

where τ_{ij} is the viscous stress tensor:

$$\tau_{ij} = \mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) + \delta_{ij} \lambda \nabla \cdot \mathbf{u} \quad (5.25)$$

Although this formulation is a wholly analytical derivation from the governing equations, it cannot be directly applied to the CFD model without taking into account the turbulence modelling assumptions. Moore and Moore [53] propose the following equation for the mean entropy production in a turbulent, eddy-viscosity CFD model:

$$\bar{T} \bar{S} = \frac{k}{\bar{T}} \left[\left(\frac{d\bar{T}}{dx_i} \right)^2 + \overline{\left(\frac{dT'}{dx_i} \right)^2} \right] + \bar{\tau}_{ij} \frac{\partial \bar{u}_i}{\partial x_j} + \overline{\tau'_{ij} \frac{\partial u'_i}{\partial x_j}} \quad (5.26)$$

where the turbulent viscous dissipation term can be expressed as:

$$\overline{\tau'_{ij} \frac{\partial u'_i}{\partial x_j}} = \frac{\mu_t}{\mu} \bar{\tau}_{ij} \frac{\partial \bar{u}_i}{\partial x_j} \quad (5.27)$$

For the flow in the turbine, the entropy generation from the thermal diffusion term was small in relation to the viscous dissipation and therefore could be ignored. This resulted in the following equation used here to model the local entropy generation rate per unit volume due to viscous dissipation:

$$\dot{s}_{gen} = \frac{1}{T} \left(\bar{\tau}_{ij} \frac{\partial \bar{u}_i}{\partial x_j} + \frac{\mu_t}{\mu} \bar{\tau}_{ij} \frac{\partial \bar{u}_i}{\partial x_j} \right) \quad (5.28)$$

It should be noted here, however, that the accuracy of this approach does rely on two main assumptions. First, it assumes that the relevant reversibilities are fully captured by the simplification inherent in an eddy-viscosity turbulence model. For this reason, Naterer and Camberos [53] suggest that equation 5.26 will result in inconsistencies close to a wall. Also, this method relies on the accurate resolution of velocity gradients calculated from cell to cell. For a relatively coarse mesh, sharp gradient of velocity that occur in the boundary layers may not be resolved with sufficient accuracy. To test the quantitative accuracy of this formulation, the local entropy generation calculated from equation 5.28 was integrated across a fluid domain and compared to the results given in Figure 5.21. Supporting the claim of Naterer and Camberos [53], these values did not match. Nevertheless, owing to its common usage to analyse loss generation in a CFD model, it will be used here to qualitatively identify regions of increased entropy production.

Figure 5.22 shows a contour plot of the entropy generation rate at the nozzle midspan. This type of plane demonstrates the large entropy generation that occurs as the rotor blades move in the interspace region of the non-flowing entry (feature X). A large proportion of this loss is as a result of the turbine spinning in a region of low pressure, thereby compelling the blades to act as a rather inefficient compressor. This effect is commonly called ‘windmilling’ since the movement of the rotor blades act to churn the fluid in this region and result in energy transfer from the rotor to the fluid. Estimates of this type of windage loss by different authors range from 11 percent [11] to 20 percent [10] for a partially fed turbine. There are also locations where entropy is generated as a result of viscous shearing between two flows travelling at different speeds. Feature Y shows this free-shear mixing layer near the first tongue, and feature Z shows lengthy free-shear layer that results from mixing at the second tongue. Both these cases arise where the flow in the lower inlet leaves the nozzle and mixes with the stagnant fluid in the non-flowing inlet. This effect, along with other loss that occurs in these regions, is commonly referred to as *end of sector loss*.

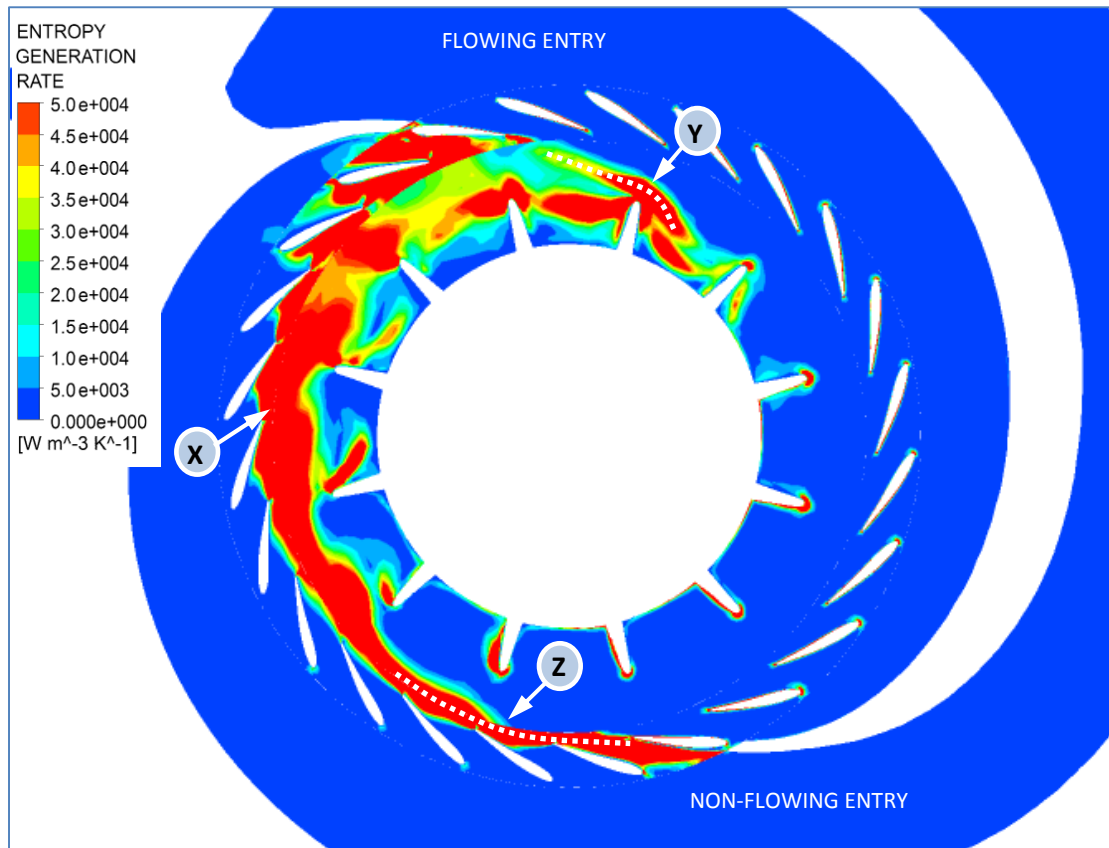


Figure 5.22: Entropy Generation Rate on Nozzle Midspan Plane

Figure 5.23 shows the contours of entropy generation in three constant span locations through the turbine: 25%, 50% and 75% for the partial admission simulation. The same contours are also shown for equal admission at 50% span for comparison (using the same scale). These views show the area of high entropy generation in the blade passages at an instance in time. Owing to the higher blade speed at the tip, it would appear as though the windage loss in the interspace of the non-flowing interspace increases from hub to shroud. What is slightly surprising in this figure is the amount of entropy generation that is present in the blade passages in the flowing region. Unlike the windage loss, this effect *decreases* from hub to shroud. This is related to the time required for the flow to re-establish in the blade passage. As noted earlier, near the hub region the slower moving flow must travel a greater distance, leading to greater time required to mitigate the loss mechanism and establish the flow. In fact, the contours of entropy in these plots seem to provide an additional indication of the time required for flow development in the rotor passage. The dashed black line suggests an 'advancing front' of flow establishment as the flow from the lower inlet moves through the rotor passages. Near the shroud region, it is clear that the flow develops much faster with little remnants of loss generation showing in the final two passages.

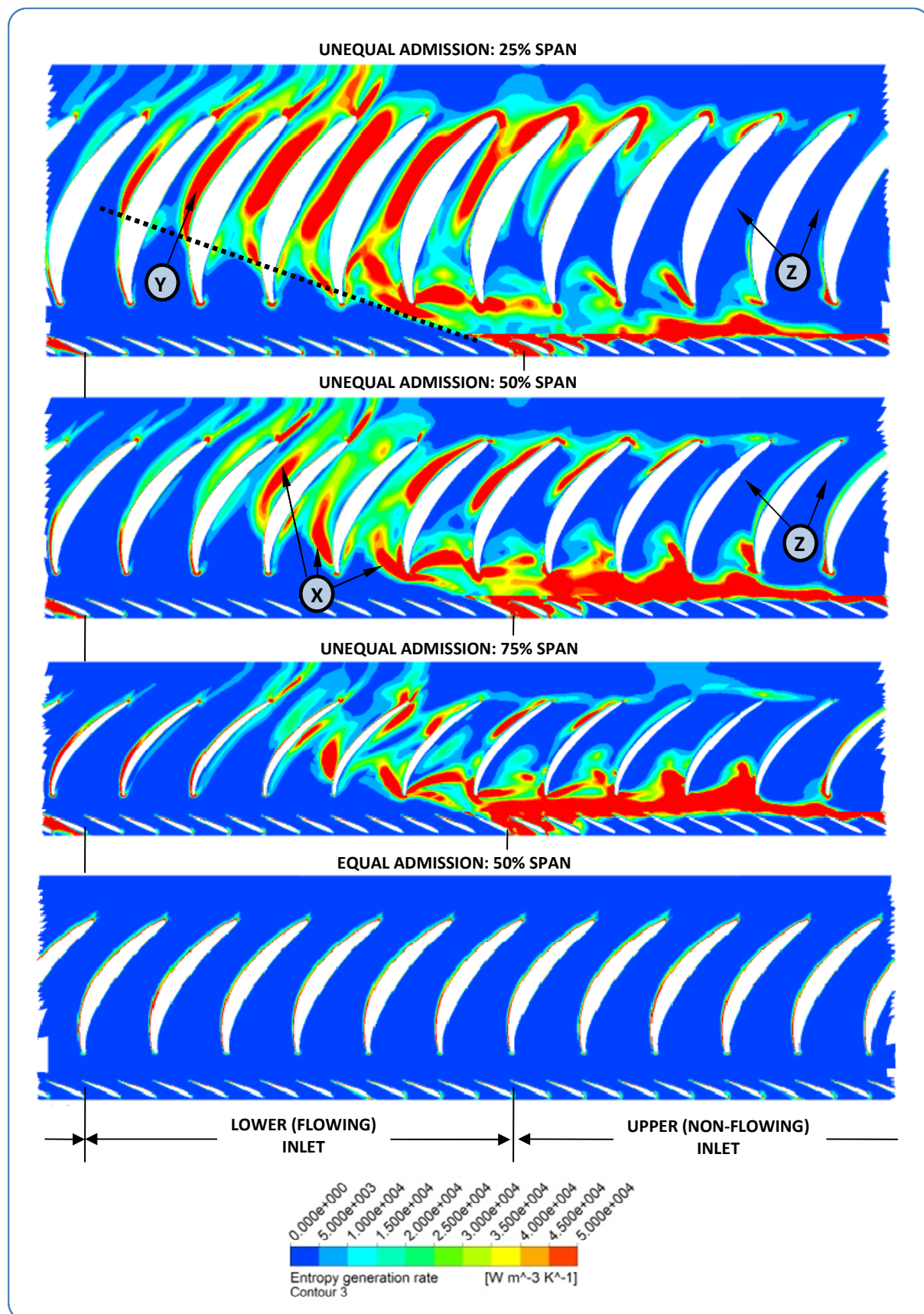


Figure 5.23: Entropy Generation rate contours on a constant span

It would appear as if a large proportion of the entropy generation in the blades as it moves from the flowing region to the non-flowing region occurs as a result of free shear as the fast flowing fluid is introduced into the passage filled with a largely stationary fluid. Consider, for example the development of the entropy generation zone indicated by feature X. As the high velocity fluid from the flowing sector enters the rotor, it will immediately push through the passage on the pressure surface, thus forming a free shear layer with the stagnant fluid left on the suction surface of the blades. Thus, what perhaps might look like a boundary layer separation (feature Y) is more likely the result of a free shear layer forming between the fast moving fluid on the pressure surface and the remaining slow moving fluid. However, focusing on the development of the flow in the first few blade passages in the flowing region is not entirely representative of the entirety of the loss. It is especially important to note that while the passages in the flowing region show significant areas of loss generation, there are at least two passages in the non-flowing sector that exhibit loss profiles close to the equal admission case (Feature Z). Clearly, the fluid remaining in these passages from the flowing region has not had a chance to develop significant entropy generation mechanisms.

Thus, there are clear signs that entropy is generated in the rotor as the fluid fills the blade and shears with the stagnant flow from the upper entry. However, it would also seem that the largest single area of entropy generation occurs in the interspace volume on the non-flowing sector. Here the rotor blades windmill in a region of largely stationary fluid resulting in a highly turbulent region of loss generation. In addition to viscous heating, the rotor will also act to raise the pressure of the fluid as it changes from a turbine into a compressor. While strictly not 'loss' as such, this region will remove some of the power generated in the flowing sector, thereby lowering the efficiency of the machine. It is therefore interesting to consider how this ultimately impacts the power generated by the turbine in the flowing and non-flowing regions. To do this, the torque can be calculated as a function of time by integrating the pressure forces around each blade surface at each timestep. This therefore provides an indication of the changes in torque that occur as the blade rotates through the each of the two sectors. The result of this CFD calculation is provided in Figure 5.24 for equal and partial admission. From this data, the torque can be integrated to give the power according to equation 5.29:

$$P = \frac{\omega}{T} \int_{t_1}^{t_2} \tau dt \quad (5.29)$$

Table 5.8 provides the result of this calculation for one turbine blade travelling through ½ of a full rotor rotation. Note that deciding exactly where to define the start and end of a particular sector is somewhat uncertain for the rotor since the flow in each ½ nozzle sector will rotate a certain amount

before reaching the rotor inlet. Thus, a judgement of flow offset was made here. In any case, the main purpose was to show how the power is distributed between each of the 180 degrees and how this compares with the equal admission case. The first surprising discovery is that, despite the fluctuations, the average torque in the first 180° is not too dissimilar to the equal admission case. This is a somewhat unexpected finding since one might think that owing to the heavy regions of loss shown in the blade passages in the flowing region Figure 5.23, this should translate into a decreased turbine efficiency in the flowing sector. However, the data in Table 5.8 suggests that the power production in the first 180° is comparable to the equal admission case. What is perhaps even more interesting is the magnitude of the power that is required to move the blade through the non-flowing region. The numbers in Table 5.8 show that 24% of the power generated in the flowing sector is required to move the blade through the non-flowing sector. This is not only a significant finding for steady state operation, but also for unsteady, pulsating flow. Over a pulse cycle, the turbine can be exposed to regions of low mass flow where the turbine is essentially “freewheeling” - carried by its own momentum. This suggests that in these conditions, the turbine can experience a significant loss of efficiency due to the windage losses.

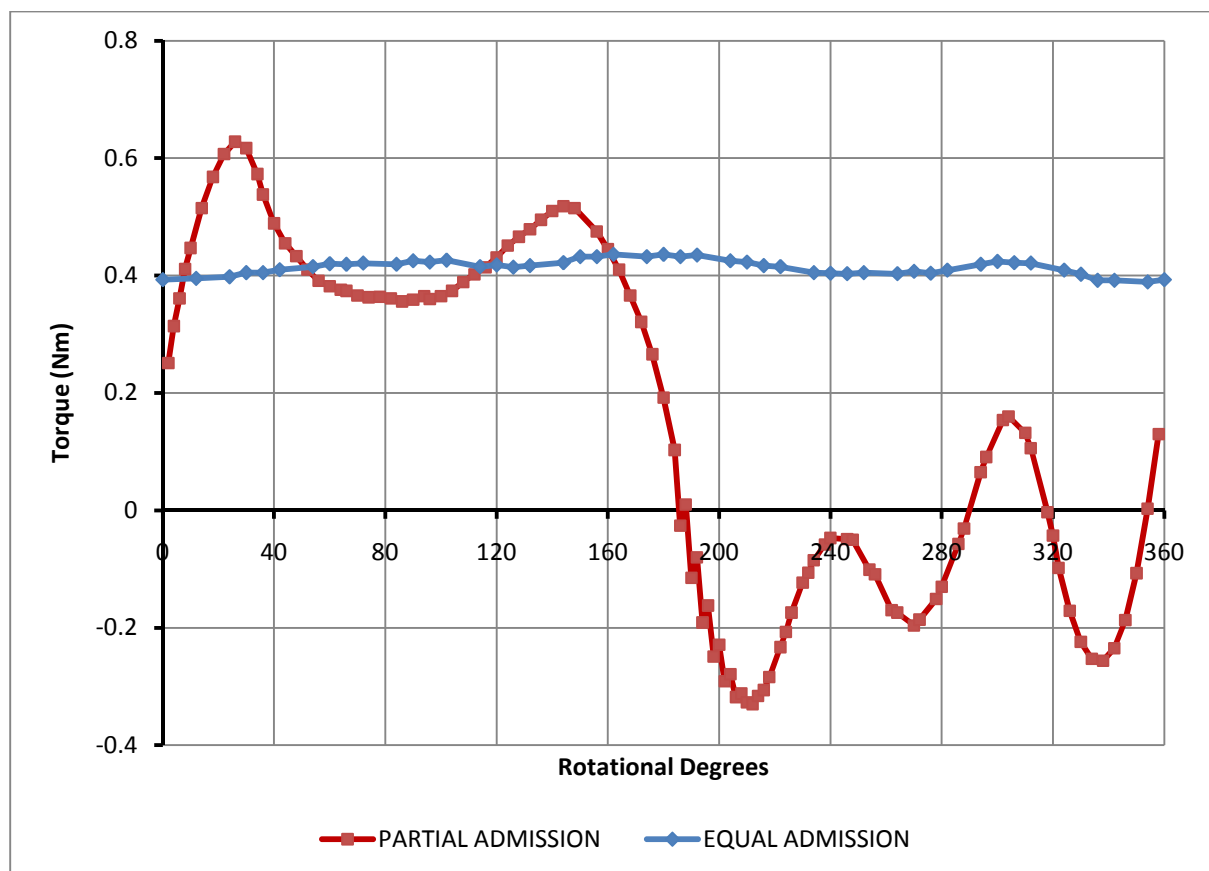


Figure 5.24: Transient torque on a single blade passage

Table 5.8: Power transferred to a given blade in each flowing sector

Rotation	Unequal Admission Power (W)	Equal Admission Power (W)
0 → 180°	2211	2329
180 → 360°	-525	2297

5.6 Conclusions

A full three-dimensional CFD model of the turbocharger turbine has been developed in order to aid the interpretation of the steady-flow experimental results. A commercially available RANS CFD solver was used for this purpose, namely ANSYS CFX, due to its turbo-friendly tools and common usage in the turbomachinery field. Since the impetus for building a CFD model was to use it to interpret and analyse the experiments, the pressure and temperature measurements made in the lab were used directly to specify the boundary conditions in this simulation. To begin, a total of five equal admission simulations were modelled, covering a broad range of conditions at a constant speed of 48.3 rps/VK. This initial step was necessary, first in order to provide some preliminary verification that the model is a reasonable representative of the performance measured in the lab, and secondly to form the basis of comparison with the unequal admission analysis. The model was supplied with a set of pressure-pressure boundary conditions (total inlet to static outlet) in order that the solution would contain a prediction of mass flow through the domain. The result was an excellent prediction of mass flow in comparison with the experimentally measured values. This not only confirmed that the geometry of the model was an accurate representation of the real turbine but also confirmed the predictive capabilities of the CFD, thereby lending some confidence in the model. The model also produced values of the torque on the rotor, thus leading to a prediction of the efficiency of the turbine. The CFD model did well to predict the trend of the experimental efficiency but displayed a tendency to over-predict the performance in most cases. The CFD was used to make a brief study of the inlet flow angles over the range of equal admission conditions. With the flows introduced equally, the disturbances near the tongue region appeared quite low, with no significant evidence of flow angle distortions in these regions. It was also found that in all cases, the exit flow angle from the volute matched the nozzle stagger angle of 70°, reducing any turning in the nozzle to a few degrees. The fact that the exit angle from the volute is essentially

invariant with the flow conditions is an important thing to note in view of a pulsating flow situation. However, the change in the magnitude of the inflow as velocity ratio is increased causes the average relative inlet flow angle β_3 to swing by 80.4° over the range of test conditions. This, along with an increase in the swirl exit angle leads to a decrease in the power available and the introduction of passage losses in the rotor.

The unequal admission performance of the turbine was modelled with four unequal conditions tested in the lab. Each of these four conditions maintained approximately the same lower inlet pressure ratio while the second boundary condition varied from 2.6 bar (total) to a zero flow condition. This latter partial admission case was set by specifying a wall boundary on the inlet of the upper volute passage. Like the equal admission simulations, the CFD does well to predict the swallowing capacity over the range of unequal flow boundary conditions tested here. The simulations also reproduced the interaction between the volute passages that was noted in the experiments. This trend, where the flow rate in one entry is influenced by the pressure in the second, is accurately predicted by the CFD model. This was an important effect to be able to model and therefore allowed the examination of the physical effect driving this trend. By studying the pressure around the circumference of the nozzle exducer, it was found that the static pressure near the two tongues was significantly influenced by the pressure in the second entry. For example, if pressure in the upper inlet was low, this lowered the static pressure in the interspace of the lower inlet, thereby increasing the expansion ratio across the stator. Thus, the lower inlet was able to expand more freely into the nozzle interspace as the pressure in the upper inlet was reduced. Conversely, if the pressure in the upper inlet was increased, this increased the pressure in the interspace region, thereby reducing the expansion ratio across the stator and restricting the effective area.

One of the more interesting aspects of the double-entry design is the way in which a rotor passage is fed by two separate stator sectors as it rotates. Therefore, when fed unequally, the distribution of pressure and mass flow around the circumference of the rotor inlet will not be uniform. This, in effect, leads a situation where each rotor passage experiences a pulsating flow as it rotates from sector to sector. The pulsating frequency will depend on the speed of rotation, and the amplitude will be determined by the ratio of inlet pressures. To understand how this will affect the turbine operation, the time required for the flow to pass through the rotor must be compared to the time to rotate one half of a rotation. If the bulk thru-flow time is on the same order as the rotor transit time, this will result in a flow field that is never able to fully establish itself in the blade passage. Thus, in order to get a sense of this effect, the CFD results from three equal admission cases

were used to calculate the circumferentially-averaged, axial velocity through the rotor as a function of the axial distance along a midspan flow path. The analysis suggested that the flow requires between 69% and 119% of the $\frac{1}{2}$ wheel rotation to completely travel through the wheel. The influence that this had on the flow field was studied by looking the pressure and velocity profiles as they developed in the flowing region at an instance in time for the partial admission case. The pressure appeared most affected in the interspace region prior to the leading edge of the first few blade passages in the flowing region. Furthermore, contours of relative velocity at the trailing edge suggested that the flow never completely established itself to the profile seen in the equal admission equivalent. Finally, by studying the streamlines in each of the six blades in the flowing region, it was possible to observe the flow features as they developed over time. The passages closest to the hub, where the slower moving fluid must travel a longer distance, showed signs that the vortices and shear layers set up near the first tongue were not completely dispersed by the end of its travel in the flowing section. All this analysis supported the argument that the flow in the blade passages of a double-entry turbine that is unequally fed with steady flow cannot be considered quasi-steady.

The unequal admission efficiency of the turbine predicted by the CFD model was also considered. The CFD failed to quantitatively predict the full efficiency loss due to unequal admission but did demonstrate the trend quite well. To gain further insight into where the model predicts loss generation, the difference in the static entropy was calculated across five main areas in the turbine for the partial and equal admission cases. As might be expected, the volute and the exit duct showed little influence due to unequal admission. The nozzle and the vaneless space prior to rotor entry, however, showed a significant increase in loss generation under partial admission conditions. However, by far the largest increase in the entropy occurred in the rotor. In order to visualize the loss creation on a local basis, the concept of an *entropy generation rate* was used as formulated by Moore and Moore [53]. This led to a series of contour plots highlighting the areas of loss creation as the flow moved through the turbine. In the regions near the end of each sector, the high velocity flow from the lower entry formed a free shear layer with the stagnant flow from the upper entry, thereby resulting in regions of high loss. Interestingly, this same effect was noted in the blade passages at the start of the flowing region as they filled with high velocity flow. The largest single area of entropy generation occurred in the interspace volume on the non-flowing sector. Here the rotor blades windmill in a region of largely stationary fluid, resulting in a highly turbulent region of loss generation. To study the ultimate effect that these losses have on the power generated by the turbine, the torque developed from a turbine blade was calculated as a function of time over a single revolution. Somewhat surprisingly, considering the areas of entropy generation shown in the flowing sector, the values of torque in the first 180° were comparable to the equal admission condition. Yet

in contrast, the data also showed that a power *deficit* resulted from the rotation of the blade in the second 180°. This negative value of torque indicated the significance of the loss generation seen in the interspace region of the non-flowing entry but also the fact that the turbine was acting as a compressor in this region, transferring power to the stagnant fluid in the form of pressure and temperature. This is significant because it shows that regions of very low flow in a turbine can result in very high windage losses as the turbine is obliged to consume power to move the blades through such conditions.

Chapter 6

Unsteady Flow Experimental Results

6.1 Introduction

This chapter will seek to present the performance of the double-entry, ABB turbocharger turbine under a wide range of pulsating-flow test conditions. These unsteady flow tests are in some ways the culmination of the experimental investigation that is presented in this thesis. The pulsating flow tests are the most representative of the operation of a real turbocharger driven by an internal combustion engine. In the real turbocharger, a compact exhaust manifold delivers a highly pulsating exhaust stream produced by the opening and closing once of the exhaust valve. This situation is simulated in the laboratory by a pulse generator valve that periodically opens and closes the flow path into each turbocharger inlet. In this way, a pulse is produced that can be varied in frequency and amplitude. By analysing the various time-resolved measurements, one can achieve an understanding of the impact unsteady flow on performance. In addition, since the present turbine geometry is equipped with two inlets, the pulses introduced in each of these inlets do not necessarily need to be in-phase. On a twin entry turbine, the pulses are out-of-phase at the inlet, since these two inlets will be connected to different banks of cylinders.

To begin, the individual measurements from the various instruments in the laboratory will be discussed. The transient mass flow, temperature, speed and torque will all be considered to observe how these conditions vary over a pulse cycle. The data from these individual measurements will then be used to calculate some of the standard performance parameters: velocity ratio, mass parameter, pressure ratio and efficiency. First, the swallowing characteristics of the turbine under various unsteady flow conditions will be investigated by plotting the mass parameter versus pressure ratio. This standard representation also allows a comparison with the steady-state behaviour. Next the efficiency of the turbine will be considered in some depth. Although the instantaneous power and efficiency traces are presented, particular attention is paid to the cycle averaged data due to difficulties in matching the isentropic and actual power traces. Finally, the variation of the unsteady operating point can be traced with respect the three parameters: PR_{INNER} , PR_{OUTER} and U/C_{IS} . Thus by plotting the unsteady operation in 3-D space, a comparison between the steady and unsteady operation results.

6.2 Unsteady Performance Parameters

6.2.1 Instantaneous Condition

The term “instantaneous” will often be used in this thesis to differentiate between the transient (time-resolved) measurements and the steady state (time-averaged) measurements. However, in reality the final instantaneous values were generated by processing 50 cycles of data acquired when the turbine was operating at a stable speed. Consider pressure as an example. The value of static pressure was sampled by the high speed data acquisition system at a rate of 20kHz. This provides between 235-800 data points for each pulse cycle (depending on frequency). Each of these data points represents the pressure at an instance in the pulse.

Unsteady measurements of the flow entering the turbine were made at a *measuring plane* very close to the turbine inlet. This location is shown in Figure 6.1. At this position, two pressure transducers and two hotwire probes provided instantaneous pressure and mass flow data for each entry. It was important to obtain these measurements as close as possible to the turbine inlet so that the data would only reflect the effect of the volume of air contained in the turbine. This instantaneous data could then be used to calculate the unsteady, time-resolved values of the standard performance parameters listed in Section 4.2.1. The definitions of these performance parameters remain largely unchanged for unsteady pulsating flow. There are, however, some complications associated with applying these performance parameters on an instantaneous basis. Therefore these performance parameters definitions must be considered in more detail from an unsteady perspective.

Pressure and Pressure Ratio

In addition to the two pressure transducers at the inlet, three pressure transducers were added on each of the volute passages just prior to the nozzle blades as shown in Figure 6.1. Thus, with a total of four pressure measurements on each passage, the progression of the pressure pulse through the volute could be measured. In addition, two pressure transducers were mounted on the exit duct at 75 mm and 250 mm from the trailing edge of the rotor to record the instantaneous pressure of the flow exiting the turbine.

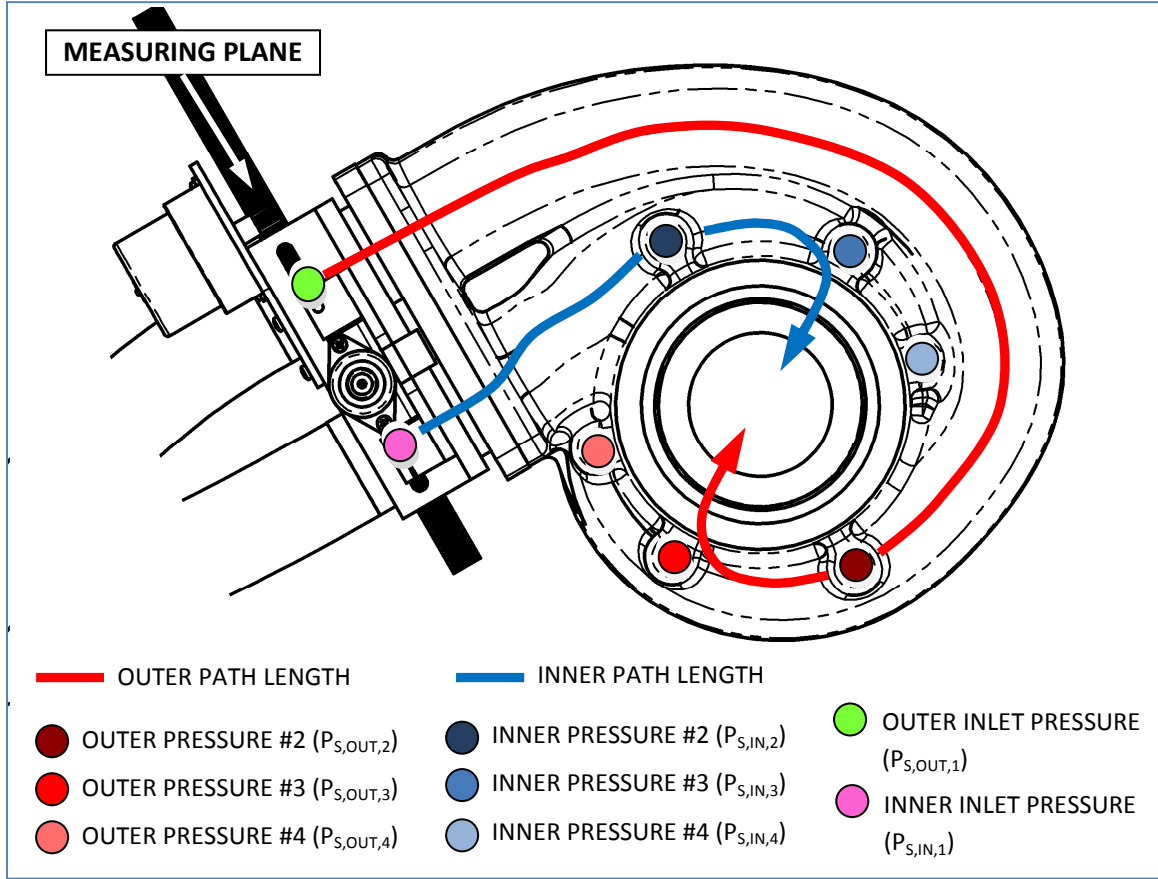


Figure 6.1: Static pressure transducer measuring locations and assumed pulse path length

The total pressure at the inlet to the turbine was needed in order to calculate the instantaneous pressure ratio. The stagnation condition at inlet was calculated from the well known relation:

$$\frac{P_0}{P} = \left(1 + \frac{\gamma - 1}{2} M^2\right)^{\gamma/(\gamma-1)} \quad (6.1)$$

where the Mach number M was calculated from the mass flow measured from the hotwire. Thus, the fluctuating inlet pressure for each of the inlets could be used to calculate a separate value of the pressure ratio across each inlet as follows:

$$PR'_{inner} = \frac{(P_{01})'_{inner}}{P_4} \quad PR'_{outer} = \frac{(P_{01})'_{outer}}{P_4} \quad (6.2)$$

where the single prime mark indicates a unsteady measurement. Note that the exit pressure has not been specified as a fluctuating quantity in equation 6.2. It is not immediately clear whether the

pressure ratio should be calculated using the instantaneous or average exit pressure. Theoretically, the expansion force that is driving the mass flow through the turbine at each instance in time could be evaluated using the instantaneous pressures at the inlet and exit. However, one must then take care to ensure that the phase difference between the measurements is appropriate.

Mass Parameter

The definition of mass parameter given by equation 4.15 for the unequal admission, steady state experiments is identical to the definition given below for the unsteady experiments. The only difference is the use of transient measurements of mass flow, temperature and pressure.

$$MP'_{inner} = \left(\frac{\dot{m}' \sqrt{T'_{01}}}{P'_{01}} \right)_{inner} \quad MP'_{inner} = \left(\frac{\dot{m}' \sqrt{T'_{01}}}{P'_{01}} \right)_{outer} \quad (6.2)$$

The unsteady temperature used in these parameters is derived using the value of instantaneous pressure, average temperature and by assuming the fluid is expanding and contracting isentropically over a pulse cycle. Equation 3.17 provides the formula used to calculate instantaneous temperature.

Efficiency

The total-to-static isentropic efficiency definition has been provided in equation 4.1 for steady state operation. In principle, the transient valuation of efficiency can be simply calculated from the fluctuating value of actual and isentropic power as follows:

$$\eta_{ts} = \frac{\dot{W}'_{act}}{(\dot{W}'_{is,ts})_{total}} \quad (6.3)$$

Here the instantaneous shaft power is calculated using the sum of average and fluctuating torque:

$$\dot{W}'_{act} = 2\pi N(\tau_{fluct} + \bar{\tau}) \quad (6.4)$$

where N is the rotor speed in revolutions per second and τ_{fluct} is the fluctuating torque component obtained from the product of the rotor's angular acceleration α and its polar moment of inertia I (given by equation 3.19). The instantaneous isentropic inlet power can be calculated as a summation of the conditions across each inlet as follows:

$$\begin{aligned}
(\dot{W}'_{is,ts})_{total} = & \dot{m}'_{inner} c'_{p,inner} T'_{01,inner} \left(1 - \left(\frac{P_4}{P'_{01}} \right)_{inner} \right)^{\gamma-1/\gamma} \\
& + \dot{m}'_{outer} c'_{p,outer} T'_{01,outer} \left(1 - \left(\frac{P_4}{P'_{01}} \right)_{outer} \right)^{\gamma-1/\gamma}
\end{aligned} \tag{6.5}$$

where the specific heats are evaluated based on the transient inlet temperatures.

The difficulty with measuring instantaneous stage efficiency in a turbine is the necessary incompatibility between the measurement locations of the input and output power. In other words, since the torque is measured at the shaft of the turbine, and the input power is measured at the volute inlet, a pulse of energy entering the turbine will transmit its energy to the turbine wheel a moment later. The generally accepted approach is to shift the phase of the inlet power (equation 6.5) to ensure a common time frame with output power (equation 6.4). Thus, one must consider the amount of phase shift that is appropriate to ensure a common time frame between the power extraction and the quantities at the measurement plane. Previous approaches generally involve calculating the time lag using an assumed pulse velocity and the length of the flow path from volute inlet to a representative wheel entry point. This is typically taken to be 180° from the volute tongue; however, the present configuration is unique in that the passage length of each entry is different. Thus, the path selected for time lag correction in each entry was measured from turbine inlet to a point 90° from the corresponding tongue. These two path lengths are shown in Figure 6.1 for the upper and lower volute passages. In terms of pulse speed, early authors have used either the sonic or bulk flow velocities [24-28]. More recently, success has been achieved by adding these velocities together to obtain the isentropic pressure wave velocity [29-35]. Like these authors, this latter method of phase shifting seemed to produce the best match for the results presented in this thesis.

There are two main concerns in attempting to match the phase between input and output power. The first is the necessary assumption that the waveform measured at the turbine inlet will not change on-route to the turbine wheel. Therefore, the energy measured in terms of pressure, mass and temperature must not be subject to dissemination or pressure wave superposition within the volute. In addition, it has been shown by S. Rajoo [32] by measuring the pressure waveform before and after the nozzle that the presence of a nozzle ring can act as a damper to isolate the wave action occurring in the volute from the interspace volume upstream of the rotor. Thus, there is some ambiguity in attempting to match the phase of the input isentropic trace to the output power measured at the turbine wheel, especially in conditions where wave dynamics becomes significant. For this reason, a cycle averaged efficiency is often employed to avoid the uncertainties associated with the phase shift between power measurements.

Velocity Ratio

The equation of velocity ratio is very similar to the unequal steady state case where the total isentropic power summed over the two inlets is used (equation 6.5). This definition of velocity ratio will suffer from similar phase shift issues discussed for efficiency since the speed is measured at a different location to the isentropic power. However, the overall effect will be much smaller since the speed only fluctuates by a few rpm over the course of the pulse cycle.

$$\frac{U_3}{C_{is}} = \frac{\pi d_2 N'}{\sqrt{2 \left(\frac{\dot{W}'_{is,ts}}{\dot{m}'} \right)_{total}}} \quad (6.6)$$

6.2.2 Time-Averaged Condition

In addition to time-resolved parameters of turbine performance, it is also useful to calculate parameters that are averaged over a cycle. These are useful for a few different reasons. First, cycle averaged values often avoid the issue of an incongruity due to pulse phasing inherent to some of the unsteady parameters. Also, it is useful to have averaged values that can represent the operating point of the turbine under pulsating flow conditions. This is crucial to allow appropriate comparisons to be made between different conditions.

Cycle Averaged Efficiency

Karamanis [27] suggested the use of a cycle-averaged efficiency definition that was independent of the phase difference between input and output power. By dividing the total energy of an output pulse by the total energy of an input pulse as in Equation 6.7, the efficiency of the turbine operating over a pulse cycle could be calculated. Here the summations are made over n equally spaced data acquisition samples.

$$\eta_{t-s,avg} = \frac{\sum_1^n \dot{W}'_{act}}{\sum_1^n \dot{W}'_{is,ts}} \quad (6.7)$$

This value is independent of the time delay between input and output power since the values of total power are summed before dividing. Therefore, this provides a single value of unsteady turbine efficiency regardless of unsteady effects occurring with the stage. Thus, this is an important

parameter in comparing the steady versus unsteady performance of the turbine as well as noting the effect of pulse frequency, amplitude, and phase.

Cycle Averaged Velocity Ratio

Syzmko [8] suggested calculating a cycle-averaged velocity ratio by weighting the instantaneous value with the isentropic power as shown in Equation 6.8:

$$\left(\frac{U_3}{C_{is}}\right)_{avg} = \frac{\sum_1^n \left[\left(\frac{U_3}{C_{is}}\right)' \cdot \dot{W}'_{is}\right]}{\sum_1^n \dot{W}'_{is}} \quad (6.8)$$

Note that if the value of instantaneous efficiency η'_{t-s} is substituted into Equation 6.8 in place of the velocity ratio, this will result in the definition of cycle-averaged efficiency given in Equation 6.7. Thus the weighting techniques in the two equations are consistent and hence compatible. The cycle averaged velocity ratio is useful since it can be used to pinpoint the operating point under unsteady state operation and therefore form a basis of comparison between different conditions.

6.3 Experimental Method

6.3.1 Unsteady Test Conditions

The range of unsteady-flow test cases that are presented in this thesis are outlined and discussed here. Tables 6.1 and 6.2 provide the test conditions in terms of turbine speed and pulse frequency. The frequency range of 25-83Hz corresponds to a four-stroke, 6-cylinder engine (3 cylinders per entry) running at 1000-3320 rpm. For each condition, the turbine was tested with in-phase and out-of-phase pulsations. As indicated in these tables, the ratio between turbine speed and frequency (N/f) is held at a three constant values: 10, 15, and 20. This parameter specifies the number of times the turbine wheel rotates in the time it takes the pulse generator to produce one complete pulse cycle. This is useful because it gives some sense of how the rate of change in flow conditions due to a single pulse compares with the rotor process. In addition to keeping this parameter constant, all 18 tests were tested at the same cycle averaged velocity ratio of 0.65 corresponding to the peak operating condition in steady-state testing. In the main, this thesis will focus on the 85% speed condition since this speed, in combination with a velocity ratio of 0.65, produces unsteady operation that best corresponds to optimal steady-state operation.

Table 6.1: In-phase Unsteady Tests

Percentage Speed (Approximate)	$N/\sqrt{T_0}$	Frequency (Hz)		
50	27.0	24.9	33.3	50.5
70	37.7	34.9	45.35	68.3
85	44.24	42.1	56.3	83.3
	N/f	20	15	10

Table 6.2: Out-of-phase Unsteady Tests

Percentage Speed (Approximate)	$N/\sqrt{T_0}$	Frequency (Hz)		
50	27.3	25.16	33.2	49.5
70	37.8	34.84	45.15	68.5
85	44.8	42.2	56.5	83.3
	N/f	20	15	10

As the description implies, the ‘in-phase’ tests corresponded to conditions where the pulse generator plates were set with zero offset between them. This therefore produces pulses that arrive at the turbine inlet completely in phase with each other since the length of pipework feeding each inlet are essentially equal. However, it is helpful to note that owing to the different volute passage lengths (Figure 6.1) of a double entry design, pulses which are in-phase at the inlet plane will arrive at the turbine rotor with a slight offset. The ‘out-of-phase’ tests were set up by positioning the chopper plates with a 180° offset prior to testing. Thus, the influence of a phase difference between pulses could be tested. This condition is also the most representative of the on-engine conditions since the cylinder firing order in an IC engine will be staggered, thereby producing pulses with different phase in the exhaust manifold.

6.3.2 Graphical Representation

Conveying graphically the operation of a turbine under a pulsating flow can be challenging. A single unsteady operating condition will produce a data set consisting of at least 230 values of each measurement made at the inlet, exit and shaft of the turbine. By collating this data and calculating the performance parameters at each instance, the behaviour of the turbine associated with each

instantaneous flow condition can be found. However, these instantaneous parameters must then be presented graphically to the reader. This has commonly been approached in a similar manner to the steady state condition. In other words, the instantaneous operating condition of the turbine can be plotted on mass parameter versus pressure ratio curves, efficiency versus velocity ratio curves, etc. In this way, the result is in a familiar form and allows for a swift visual comparison with the steady-state data.

None-the-less, as explained in section 4.4.3, the introduction of a second inlet flow condition means there is an additional degree of freedom necessary to prescribe the operating point of the dual entry turbine. Unfortunately, there is no real point of reference as to how best represent the instantaneous unsteady performance of a double-entry turbine due to the lack of available literature in this area. The research of Baines, Hajilouy-Benisi and Yeo [24] is one of the few experimental works containing instantaneous performance data of a turbine with more than one inlet. Figure 2.10 demonstrates their approach with a plot of the instantaneous mass parameter versus instantaneous pressure ratio for each of the inlets separately. This type of plot, however, cannot represent the unequal inlet condition. Thus, one cannot identify the phase difference that may exist between the pulses entering the inlets.

This thesis will therefore contain a combination of different approaches. First, the measurement data (pressure, mass flow, speed, torque, etc) will be presented as a function of the chopper plate rotational angle. This will give some sense of the change of flow conditions from an Eulerian point of view and the response from the turbocharger. This also provides a chance to discuss the issue of a phase difference between measurements. Secondly the performance parameters will be presented in the standard 2-D representations that are the norm for most of the available literature. Finally, to provide a complete picture of the operation of a double-entry turbine, the trace of the performance can be shown in the 3-dimensional representation introduced earlier in this thesis.

6.4 Experimental Results

6.4.1 Phase Shift

A pulse of fluid that passes the measurement plain will transfer its energy to the turbine wheel a moment later. This will result in a time lag between the conditions at the inlet (pressure, temperature and mass flow), and the speed and torque measured at the outlet. The best way to

illustrate this effect is by comparing the two pressure signals from an out-of-phase unsteady case with the torque measurement that results. This is shown in Figure 6.2 below. There is a lag between the two signals which must be corrected if the instantaneous values of efficiency are to be calculated. Szymko [8] and others since then [29-35] found that the most consistent approach was to assume that the energy is transported in an isentropic pressure wave travelling at bulk flow plus sonic velocity. Therefore, the average bulk speed ($\bar{U}_{bulk} = \bar{m}\rho A$) and the sonic velocity (a) was calculated to shift the inlet data for each entry. In order to calculate the time lag, an appropriate path length must be assumed. This was taken as the length of the flow path between volute inlet plane and a midway nozzle entry point for each entry. Figure 6.1 illustrates the two different path lengths from the turbine inlet to the rotor wheel. The end result of this phase shift between the entries is shown in Figure 6.2. It is clear that the different paths lengths require slightly different amounts of phase shift: something that is unique to this turbine.

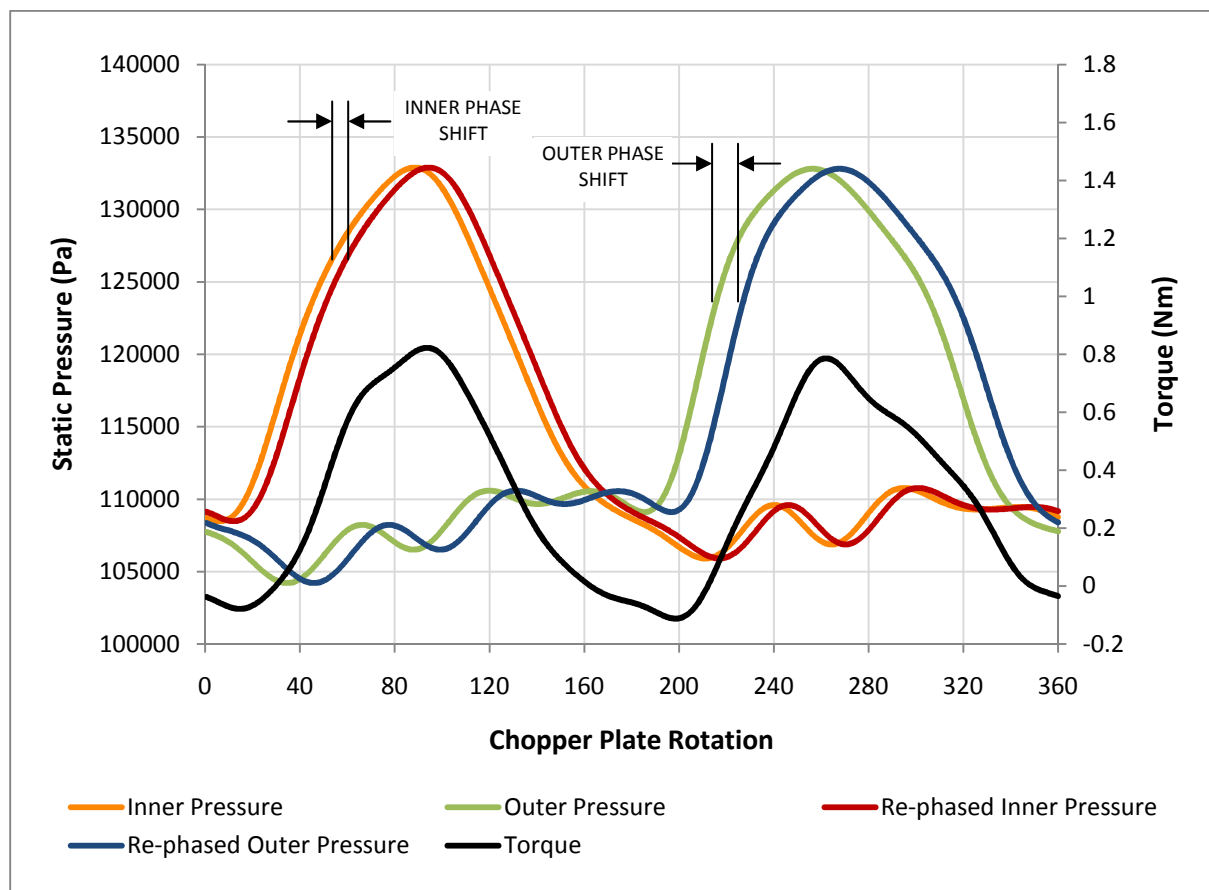


Figure 6.2: Out-of-phase static pressure inlet pressures and the resulting torque signal. Showing the necessity of re-phasing (50% speed, 25 Hz)

As highlighted earlier, there are some significant assumptions engrained in the phase shifting philosophy. The first is that the path lengths of fluid driving the rotor from a full 180° nozzle sector can be adequately represented by a single point 90° from the tongue. Secondly, one also must assume that the isentropic velocity ($U_{isen} = U_{bulk} + a$) averaged over the pulse will correctly predict the development of the mass flow, pressure and temperature in the volute that eventually drives the rotor wheel. This latter point highlights the inherent difficulty in translating the energy measured at one point to a second point in a fully dynamic system. The way in which this unsteady pulse of energy is translated through the volute will be subject to all the perturbations that can occur in an unsteady fluid domain. For instance, the pressure pulse measured at the inlet to the turbocharger will be a superposition of incoming and reflected waves. Nonetheless, the amount of phase shift will only impact the analysis in this thesis where an equation involves both the inlet (isentropic) and the actual instantaneous power.

6.4.2 Individual properties

Figures 6.3 and 6.4 show the measurements taken from the turbine operating at 85% speed exposed to in-phase and out-of-phase pulsations respectively. Measurements from the 50% and 70% speeds are included in the appendix. The following measurements are included in these figures: the inlet static pressure, mass flow, turbine wheel speed, and torque. The conditions measured at each inlet are plotted on separate plots with three different lines showing the three different frequencies. At 85% speed, these frequencies were approximately 42 Hz, 56 Hz, and 84 Hz corresponding to N/f value of 20, 15 and 10 respectively. Each of these measurements are plotted against the chopper plate phase angle in order to compare the different frequencies on a single plot. However, this can be somewhat misleading since each of these pulses will cycle in different lengths of time. For example, in the same time that the turbine responds to a full 42Hz cycle, the turbine will be exposed to *two* 84 Hz cycles. Evidence of this can be seen in the speed signal (refer to Figure 6.19) where the turbine reacts to the slower, 42 Hz pulse with a larger change in speed (~7 rps) compared with the speed response to a 84Hz pulse (~4 rps).

Each of the different unsteady measurements will be discussed in brief before moving on to the performance parameters such as efficiency and mass parameter. In general, the discussion will mainly focus on the 85% speed condition. As will be shown, this speed was found to produce unsteady operation that best coincided with areas of optimum steady-state efficiency.

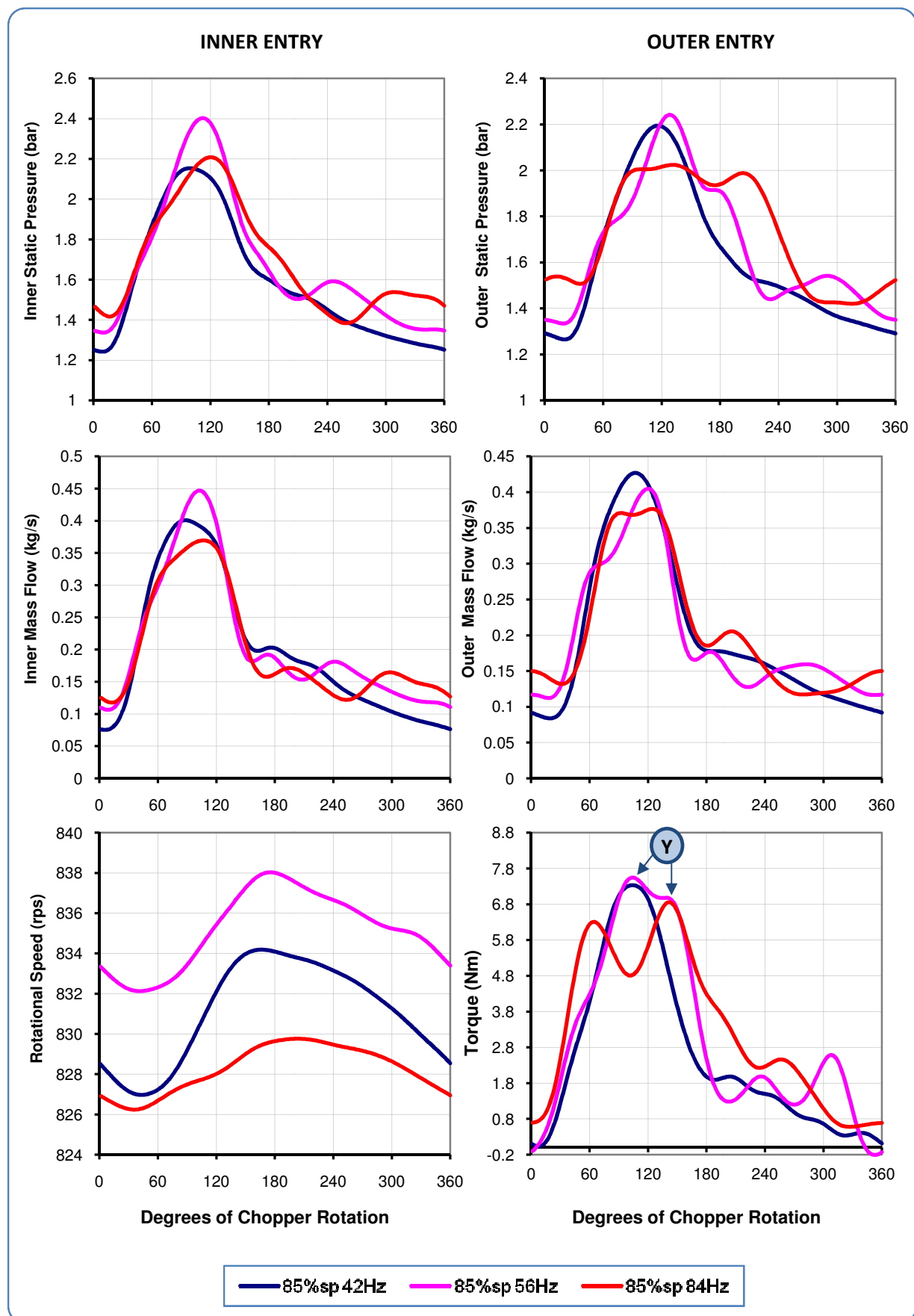


Figure 6.3: Pressure, Mass Flow, Turbine Speed and Torque (85% speed, in-phase)

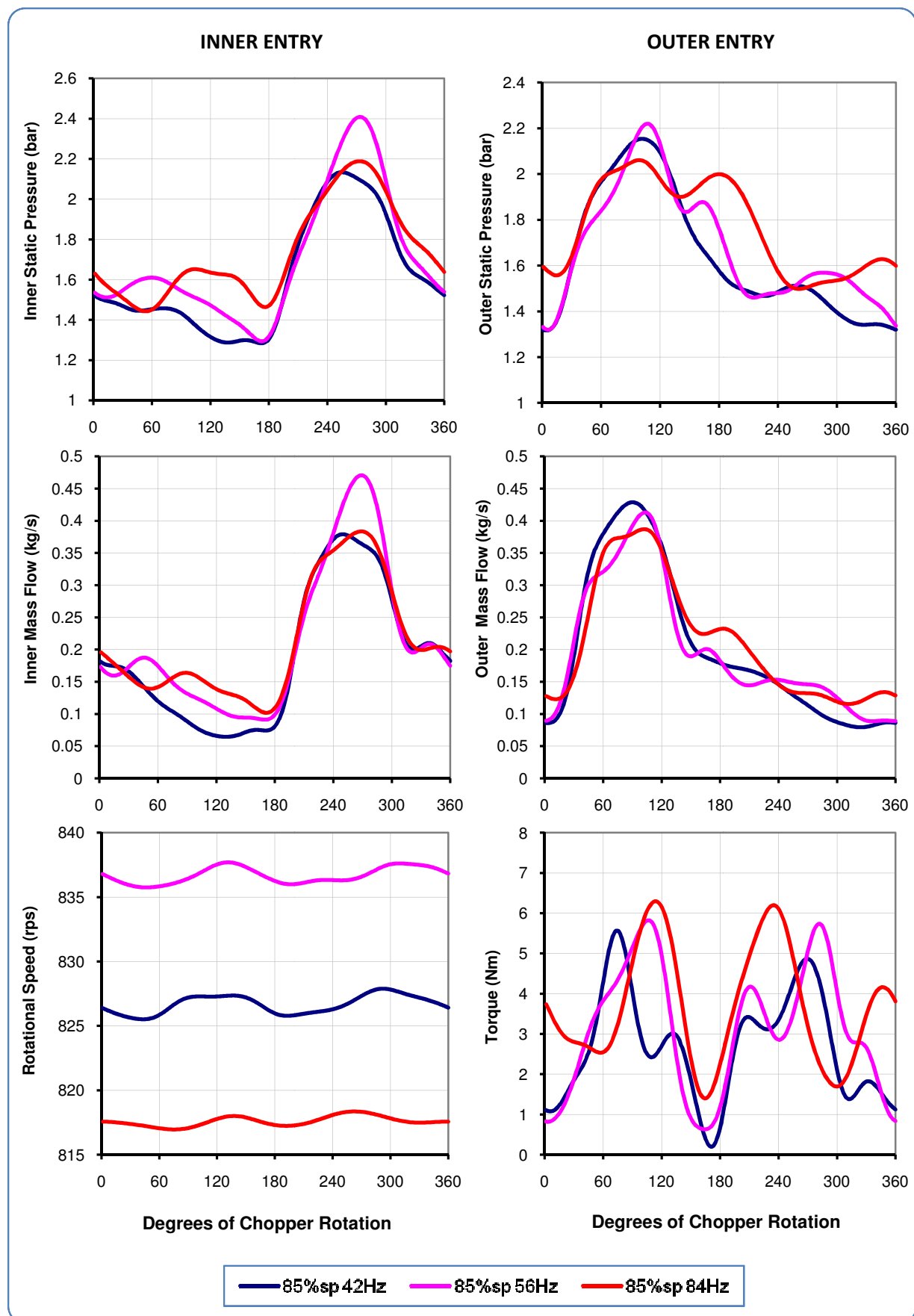


Figure 6.4: Pressure, Mass Flow, Turbine Speed and Torque (85% speed, out-of-phase)

Pressure

The static pressure pulses at the inlet to the turbine for the 85% speed cases are plotted at the top of Figures 6.3 and 6.4. Starting with the inner entry, both the in-phase and out-of-phase pressure pulses show similar characteristics at the three different frequencies. In contrast with the outer entry, these pressure traces do not widen and develop secondary peaks as the frequency is increased. This suggests that both the frequency and length difference between the two passages have some influence on the development of secondary peaks at the inlet. The frequency effect can likely be explained as an increase in the wave dynamics occurring in the stage as frequency is increased. The wave dynamics is more visible in the longer passage since a greater phase difference can develop between the incident and reflected wave along its length. Figure 6.5 shows a further confirmation of this effect by showing change in the pressure traces from the inner inlet ($P_{s,IN,1}$) to the nozzle midpoint ($P_{s,IN,3}$) at the highest frequency (84 Hz). Thus, in general, this suggests that the longer, outer limb is a more dynamic system than the inner, shorter entry.

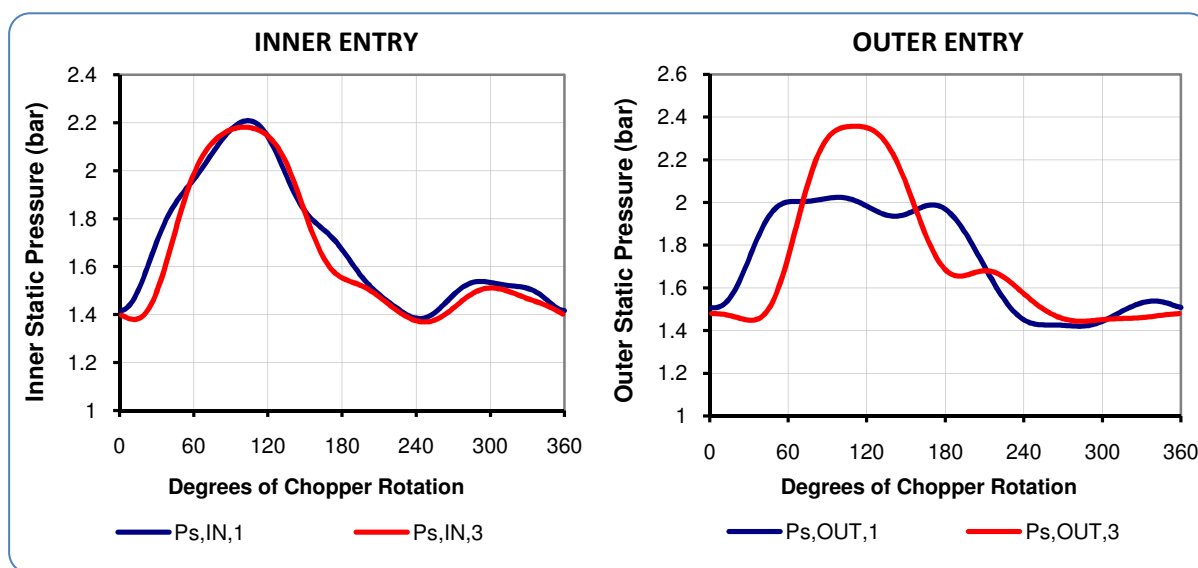


Figure 6.5: Comparing pressure at inlet and nozzle for the inner and outer entry (85% speed, 84 Hz)

Interestingly, the 180° phase difference between Figures 6.3 and 6.4 appears to have little influence on the pressure traces. This is clearly shown in Figure 6.6 which compares the in-phase and out-of-phase pressure pulses measured at the outer inlet. There is one notable exception, however. Both show an off-peak pressure pulse approximately 180° from the main pulse, but the out-of-phase case shows a greater rise in pressure in this area. While not plotted here, this effect can be clearly seen in other cases as well. Since for out-of-phase admission, the onset of this secondary peak in the

outer passage corresponds to the primary peak of the inner passage, this suggests that the small rise in pressure is due to a pressure exchange between entries. This sort of interaction between limbs has been seen in the unequal, steady state admission cases. Since the out-of-phase pulses will produce a condition of unequal flow in the turbine for much of the pulse cycle, it is not surprising that this instantaneous difference in pressure between entries would result in some interaction.

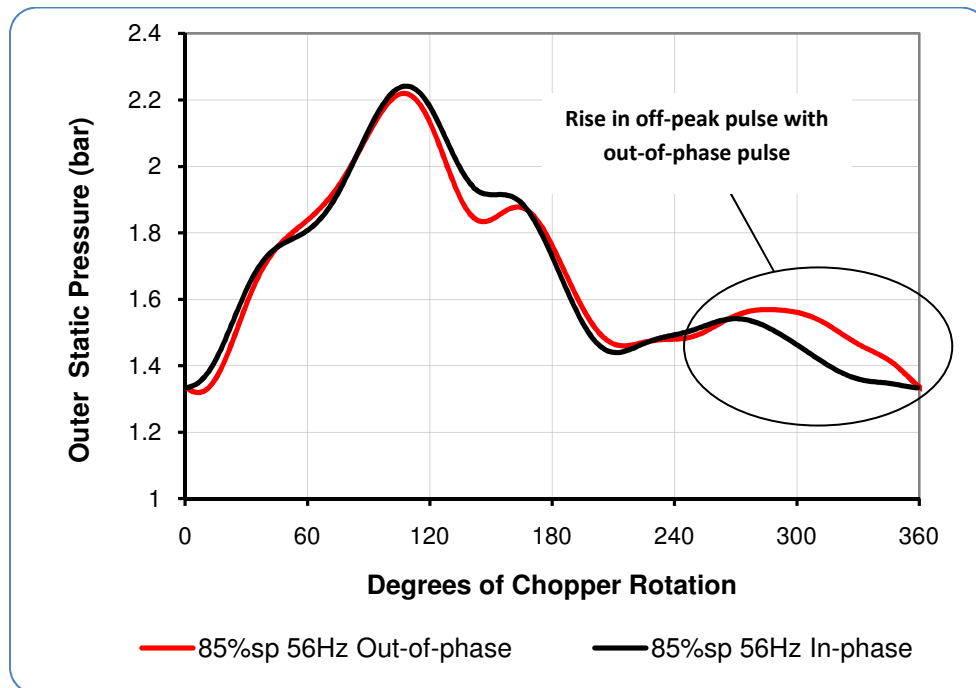


Figure 6.6: Comparing inlet pressure signals between in-phase and out of phase pulses (85% speed, 56 Hz)

Figure 6.7 shows the pressure traces recorded on the exit duct 250 mm downstream from the trailing edge of the rotor. Even though the amplitude of the *inlet* pulse decreases as frequency increases, the opposite is seen at the exit to the rotor. There appears to be a small increase in the amplitude of the pressure pulse in the exit duct as the frequency increase. This implies that the rotor itself is more dynamic under high frequencies.

In evaluating the instantaneous pressure ratio across the turbocharger, a decision was required regarding the appropriate value of the turbine exit pressure. One way could be to phase the pressure measured at the inlet to the rotor inlet and use the instantaneous exit pressure shown in Figure 6.7 to estimate the pressure expansion across the nozzle and rotor at each instant. However, this relies on an accurate phasing technique to match the inlet and exit pressure traces in an appropriate manner. In addition, it is clear from the pressure traces in the outer limb in Figure 6.5 that the pressure changes significantly depending on where it is measured along the passage. Thus,

the approach that is taken in the analysis in this thesis will be to use a time-averaged exit pressure to evaluate the pressure ratio.

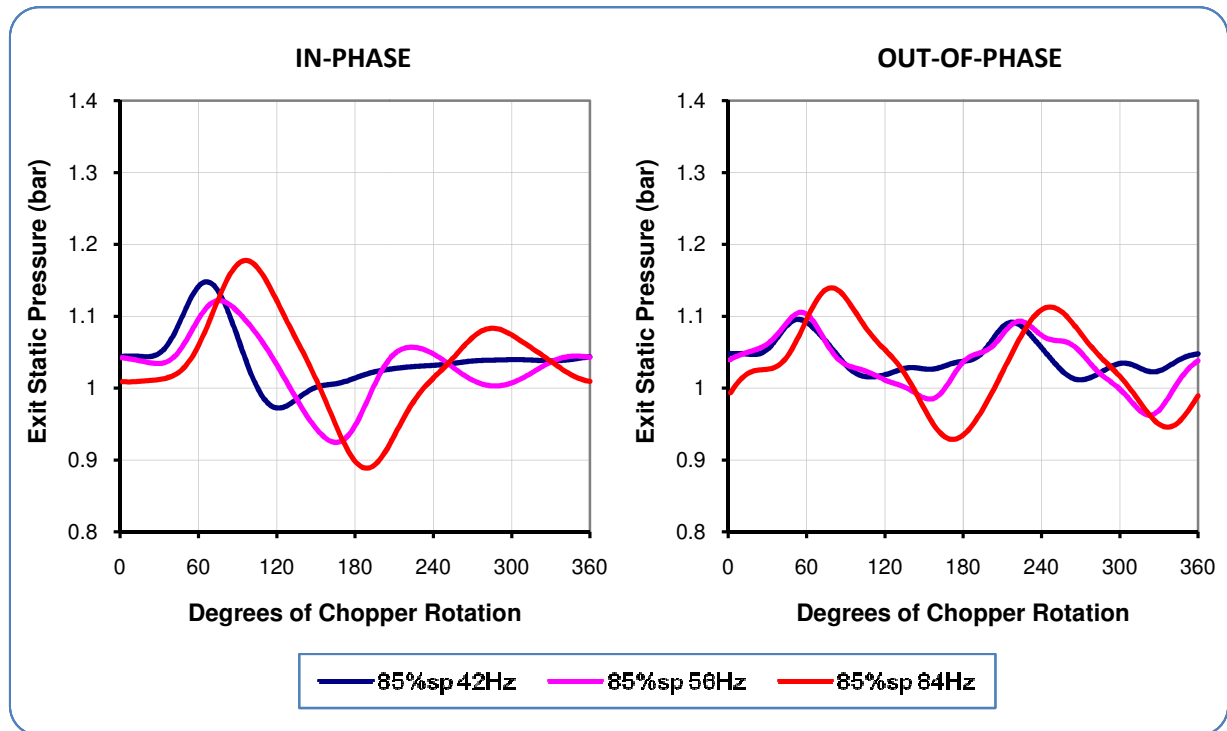


Figure 6.7: Comparing exit pressure signals between in-phase and out of phase pulses (85% speed)

Mass Flow

The mass flow traces shown in Figures 6.3 and 6.4 do not show an appreciable difference between the two inlets or between the three frequencies. Figure 6.8 compares the in-phase and out-of-phase mass flow entering the turbine under the influence of an 84Hz pulse. From this, the mass flow appears to show very little influence on the 180° phase offset. In light of the experiments made on steady, unequal admission, one might expect that *unsteady* unequal admission (out-of-phase pulse) should result in a similar change in the turbine swallowing capacity behaviour depending on the unequal pressure difference. Of course, the unsteady situation is a very different case since, unlike the steady-state case, the mass flow entering the volute is not only driven by the nozzle and turbine swallowing capacity, but also by the volume contained in the volute. The mass of fluid contained in this volume will fluctuate with the pulse. The arrival of a pulse should first act to fill up the volume, followed by a dormant time period where the mass in the volute can empty through the rotor. Thus suggests that the mass flow measurement at the inlet to the turbine will be partially driven by this filling-and-emptying action and partially by the rotor and nozzle characteristics.

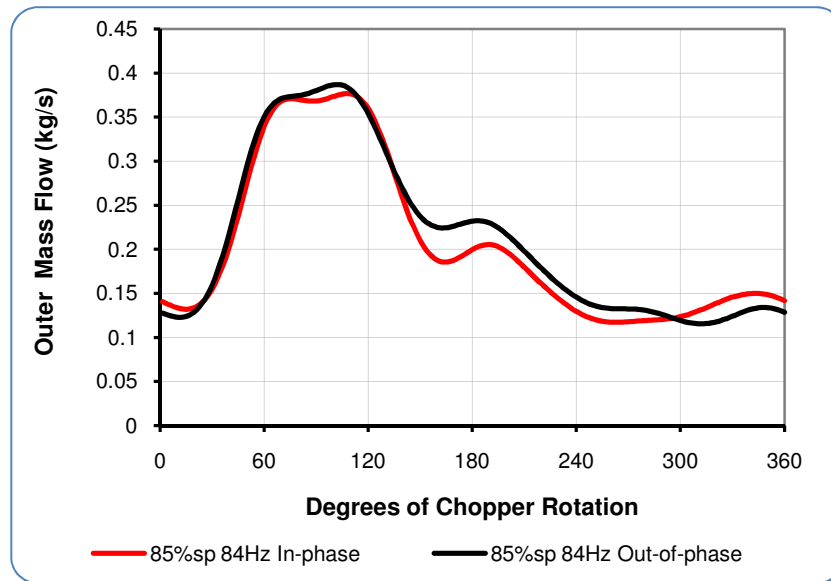


Figure 6.8: Comparing inlet mass flow between in-phase and out of phase pulses (85% speed, 84 Hz)

Rotor Speed and Torque

The ensemble-averaged and filtered signal from the infra-red sensor shows that the turbine experiences a small change in speed due to the change in conditions driving the rotor. The in-phase pulse produces a speed change of 4-7 rps or approximately 0.5% to 0.8% of the average speed depending on frequency. Since the out-of-phase pulsations drive the rotor at staggered intervals, the change of speed is much less (1.4-2.3 rps). The rotor acceleration is then calculated by integrating the small changes in speed to provide the value of fluctuating torque using Equation 3.19. Adding this fluctuating component to the average torque from the load cell results in the torque signal provided in Figures 6.2 and 6.3. The first thing to notice in these figures is the fact that the torque goes slightly negative during the latent period of the in-phase pulse. This effect is even more prominent at the lower speeds. Figure 6.9 shows the torque generated as the result of in-phase pulsations at 50% speed. In this case, all three torque signals drop below zero for a large proportion of the latent pulse period. This clearly indicates that some of the power produced in the peak of the pulse is lost as the turbine freewheels in a largely empty volume. It was found in Chapter 5 that such an effect can significantly decrease the overall performance of the turbocharger. The same effect is not seen in the out-of-phase torque traces since there is no combined latent period where both entries are bereft of flowing fluid. This is not to say that the same windmilling loss does not exist when the pulsations are out-of-phase just that it will occur on alternate 180° sectors of the rotor.

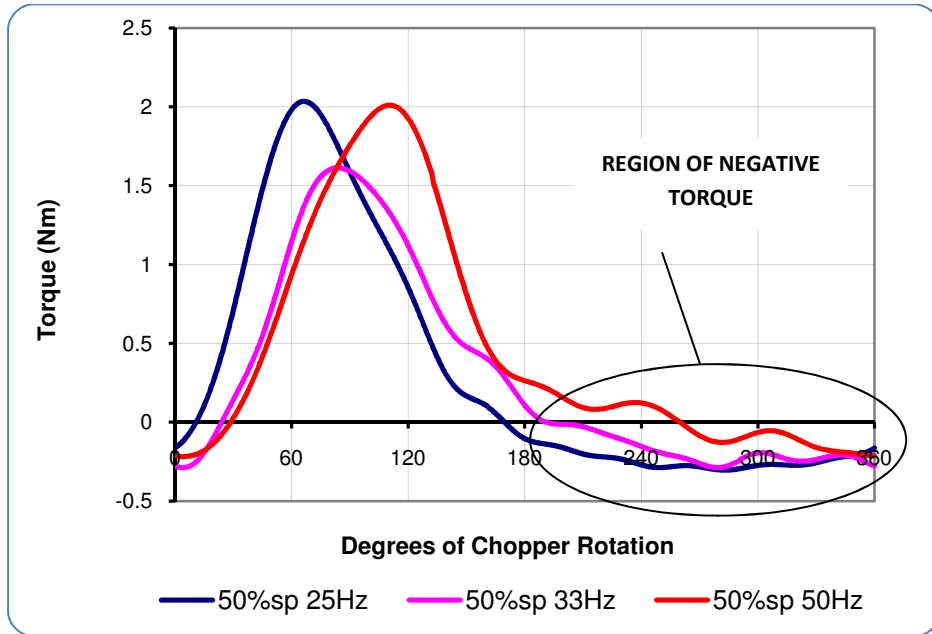


Figure 6.9: Comparing torque resulting from the in-phase pulsations (50% speed)

The rise in frequency also appears to influence the behaviour of the torque produced at 85% speed (Figure 6.3). At 42Hz, the in-phase torque trace shows a smooth rise and fall in response to the change of inlet conditions. However, as the frequency increases, a secondary peak appears that bares no obvious correlation with the inlet pressure or mass flow. None-the-less the way in which the 56Hz torque curve straddles the 42 and 84 Hz (see feature Y in Figure 6.3) indicates a definite progression in the development of the torque as frequency increases. This suggests that the dual peak feature that develops as the frequency increases is not an accidental by-product of the post-processing techniques, but a real effect that has been captured. It is, however, difficult to know what physical event is driving this dual-peak feature without modelling the dynamic response of the entire system.

Temperature

The fluctuating temperature of a pulsating flow was deduced by assuming that the rise and fall of temperature over a pulse cycle is due to the isentropic expansion and compression of the air. Equation 3.17 was therefore used to calculate the instantaneous temperature using the average thermocouple measurement and the transient pressure measurement. The resulting temperature fluctuation calculated using this approach is shown in Figure 6.10. The validity of this method was

examined by Szymko [8] by measuring the instantaneous temperature using a dual hotwire probe. This work demonstrated that the isentropic compression assumption replicated the amplitude and main features of the instantaneous temperature trace and therefore was considered reasonable in the absence of instantaneous data.

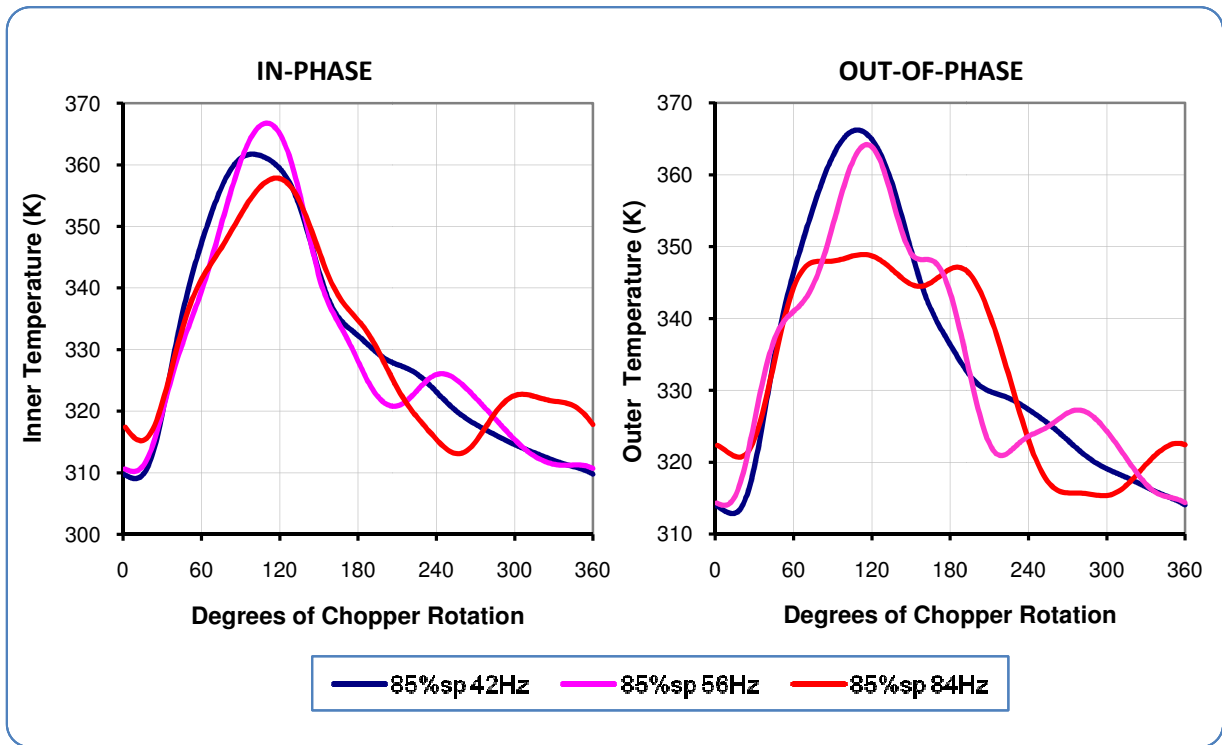


Figure 6.10: Comparing inlet temperature signals between in-phase and out of phase pulses (85% speed, 56 Hz)

6.4.3 Performance Parameters

The following section will cover the unsteady performance of the turbine using the standard performance parameters. As above, the discussion will mainly focus on the 85% speed in order to study the influence of frequency and pulse phasing under optimum operating conditions. First, the swallowing characteristics of the turbine will be considered by studying the mass parameter versus pressure ratio characteristics in comparison with equal admission. The instantaneous power and efficiency of the turbine will then be discussed, bearing in mind the difficulties in the phasing that this evaluation brings. However, by then discussing the cycle-averaged efficiency, a better understanding of the true effect of unsteady flow should result. Finally, the 3-D representation

introduced in Chapter 4 will be used to shed further light on how the instantaneous operation of the turbine compares with the steady-state operation.

Mass Parameter versus Total-to-Static Pressure Ratio

In order to begin to compare the behaviour of the turbine with the steady state operation, it is useful to consider how the swallowing capacity of the turbine reacts to changes in the pressure ratio. To this end, the instantaneous traces of mass parameter versus pressure ratio at 85% speed are plotted with the steady, equal admission operating curves in Figures in 6.11. These same plots are provided in the Appendix for the 50% and 70% speeds. To calculate these parameters, the mass flow, pressure and temperature data were taken from each individual entry. In order to compare these plots to steady-state curves provided in Figure 4.5, the values of equal steady state mass parameter were divided by two. Thus the black curves provided in Figure 6.11 show mass parameter of a one of the entries operating under equal admission, steady state conditions.

The first striking characteristic of the unsteady pulsating curves is that a loop is formed as the pressure rises and falls over the course of the pulse. The rotation around this loop occurs in a clockwise manner as shown by the arrow in Figure 6.11. This loop signifies that the rate of change of mass parameter will be different when the pressure is rising than when the pressure is falling. One of the challenges in interpreting these figures is the difficulty in representing the time taken to move along this line. Each point along the unsteady trace represents the flow conditions of the turbine at an instance in time, but the speed with which the operating point moves along this line cannot be represented without animating these plots. None-the-less, this representation is useful in comparing steady and unsteady operation.

Most of the unsteady orbits exhibit a smaller, secondary loop at the base of the orbit where the pressure ratio is low in the latent period of the pulse. As pointed out in Figure 6.11, this secondary loop is much more pronounced in the out-of-phase case. The larger secondary loop is caused by a small rise in pressure in the dormant entry when there is a pressure rise in the 'primary' entry (when the pulses are out-of-phase). In other words, these small loops indicate the same conclusion as Figure 6.6: that a rise in pressure due to the peak of the pulse in the first entry can be felt in the latent entry. A more striking version of this same effect was noted in out-of-phase testing by Baines, et.al [24]. In their case, the secondary loops were likely more pronounced due to the large amount of interaction that can occur between entries in a twin-entry volute.

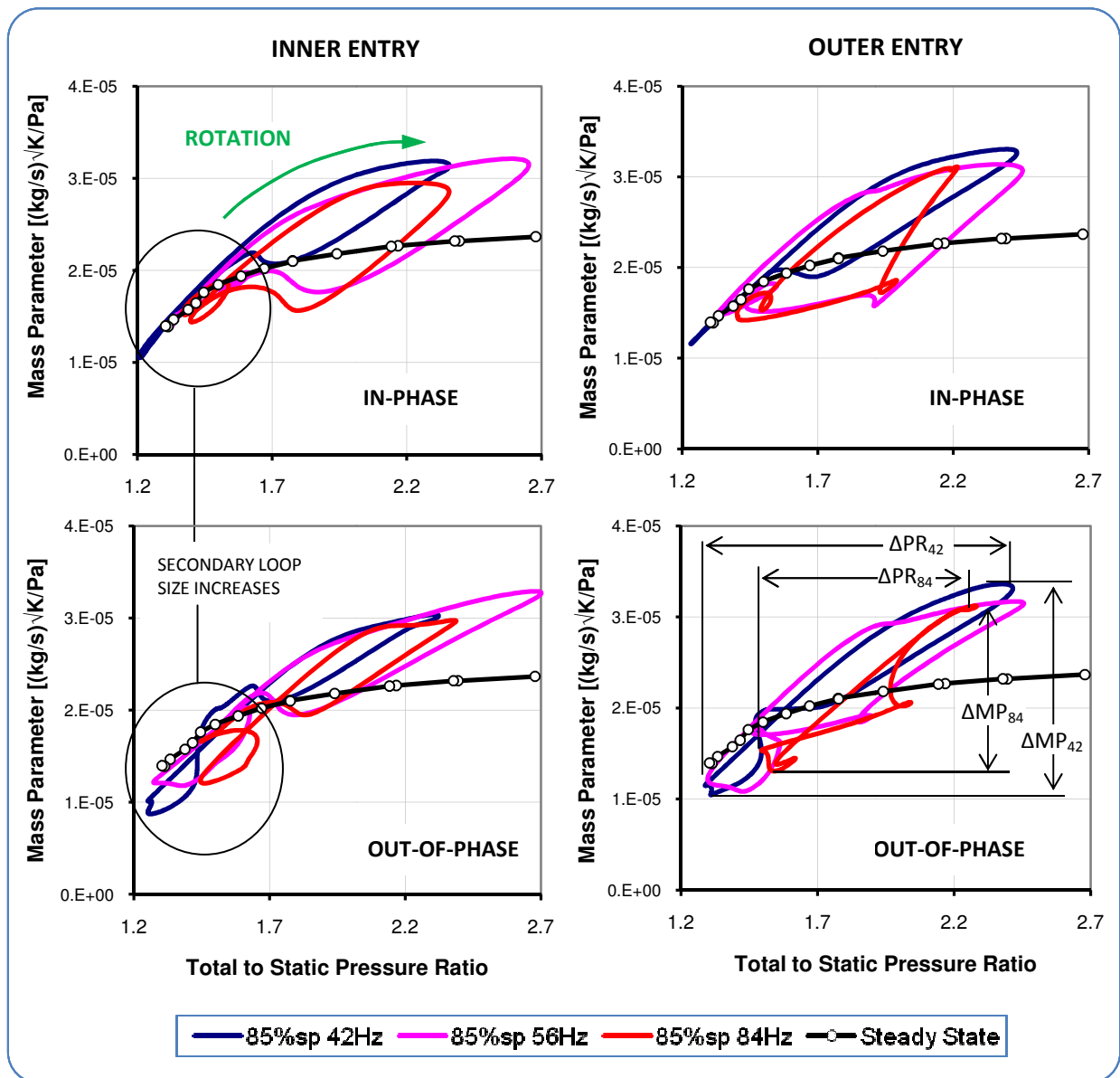


Figure 6.11: Mass Parameter versus pressure ratio across each inlet passage (85% speed)

Szymko, et.al. [29] proposed three modes to describe the relationship between pressure and mass flow within the volute. The first mode occurs when the pressure rise in a pulse is sufficiently slow to permit a state of equilibrium between mass and pressure to exist incrementally. Thus, over the pulse cycle, the rise and fall in pressure would mass parameter change that is consistent with the steady-flow behaviour. The turbine would then be operating in a purely quasi-steady mode causing the unsteady traces to lie directly on the black steady-state curves shown in Figure 6.11. However, it is apparent from the figures that this first mode does not apply here since the instantaneous unsteady operating point traces an orbit above and below the steady line. Thus, a second mode exists when the rise in pressure is too rapid to allow the mass accumulation to follow. As the pressure rises at the start of the pulse, the mass flow measured at the inlet to the volute is *greater*

than the steady-state equivalent since this pressure is driving mass into a more 'empty' volume. The contrary is then true as the pressure of the pulse begins to decrease. The hysteresis between pressure and mass that results therefore produces an operating loop which encapsulates the steady curve. This second mode can be seen in the 50% speed condition shown in Figure 6.12.

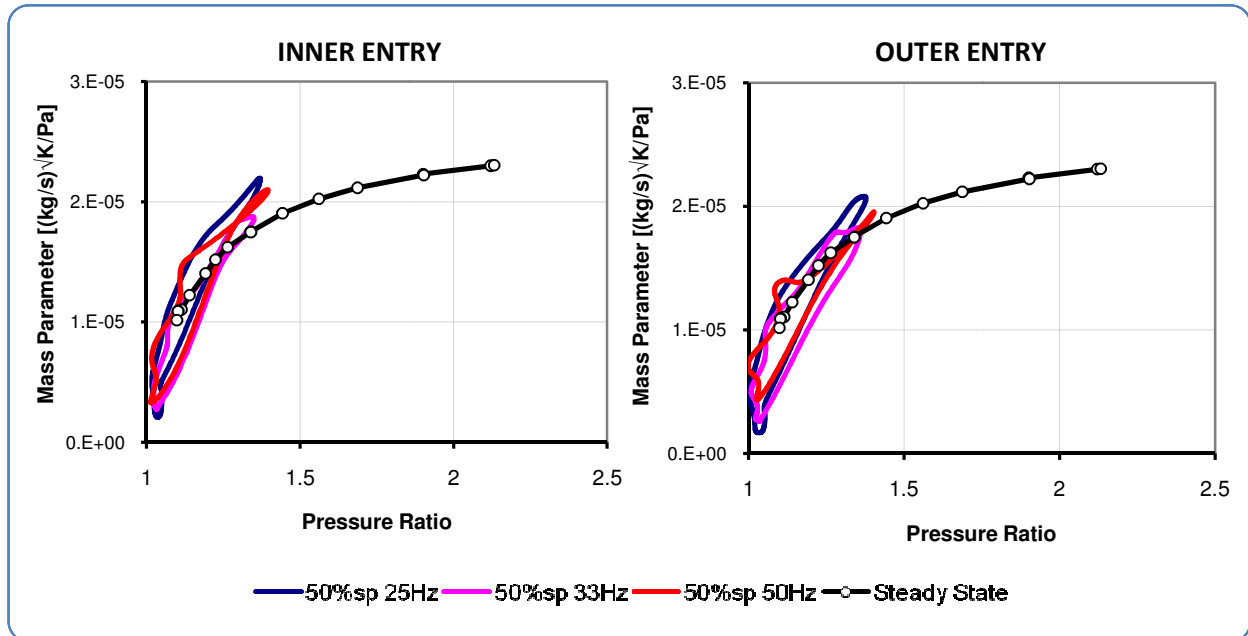


Figure 6.12: Mass Parameter versus pressure ratio across each inlet passage (in-phase, 50% speed)

Szymko et.al [16] then suggested that a further increase in frequency results in the final mode where the turbine starts to depart from steady operation altogether and the operating loop will no longer encapsulate the quasi-steady line. A wave-action simulation by Costall et.al. [7] also suggested that an increase in the pulse frequency caused the shape of the mass parameter loop to widen and rotate away from the steady curve. This seems to be the case for all three frequencies in Figure 6.11 where the angle of the unsteady orbits does not follow the steady-state behaviour. However, these works all surmised that the observed departure between the steady and unsteady swallowing characteristics was largely frequency dependent. However, in spite of this, it is difficult to see from the experiments presented here that frequency has a large influence on the departure from quasi-steady operation. Consider the orbits in Figure 6.11. With a frequency range between 42Hz and 84Hz, the overall slope of unsteady operating loop does not change appreciably. What is noticeable is that a rise in frequency causes a decrease in the amplitude of the pressure ratio (ΔPR) and mass parameter (ΔMP) as shown in the lower right hand plot in Figure 6.11. The increase in

frequency therefore seems to create a more compact unsteady orbit where the overall variation in parameters is less dramatic.

It is interesting to note that despite the performance implications of unequal admission noted in Chapter 4 and 5, the characteristics of the swallowing capacity between in-phase and out-of-phase do not show an appreciable difference. Of course, as pointed out earlier, there are indications from the more prominent 'secondary loop' that suggest an increased interaction between entries. However, the steady-state curves of mass parameter divided by two (to represent a single entry) compares equally to both the in-phase and out-of-phase results. This suggests, with regard to the swallowing capacity of the turbine, the phasing between the pulses entering the turbine is not the driving factor that determines the departure from quasi-steady operation.

Total-to-Static Efficiency

The instantaneous isentropic power calculated from the mass flow, pressure and temperature at 85% speed is shown in Figure 6.13. This data has been phase-shifted to match the rotor frame of reference using the technique outlined in section 6.4.1. The instantaneous output power is also shown as calculated from the torque and speed signals. These plots are useful as they show the match between the power measured at the two locations, and therefore suggest the validity of evaluating the instantaneous efficiency. At the lowest frequency (42Hz) the match seems to be quite good - especially for the in-phase case. Here a smooth rise and fall of the input conditions is matched by a similar characteristic in the output power. However, the out-of-phase torque signal begins to develop a series of double peaks which bear no obvious resemblance to the inlet conditions. At the highest frequency (84 Hz), the out-of-phase power trace shows three maxima that are not matched by the inlet conditions. The in-phase traces also show a similar mismatch as the frequency is increased. At 84Hz, a double-peak has developed in the torque signal that is not replicated by the isentropic power curve. This disparity between the trend of the isentropic and the actual power means that the two curves cross each other at some points, resulting in a greater than 100% instantaneous efficiency. This is unlikely to be a real effect and highlights the difficulty in assessing instantaneous efficiency of the turbine operating with a pulsating flow. Nonetheless, the instantaneous efficiency has been calculated and plotted against the instantaneous velocity ratio and the degrees of chopper plate rotation in Figure 6.14. The instantaneous efficiency at 70% and 50% speeds are also provided in the appendix.

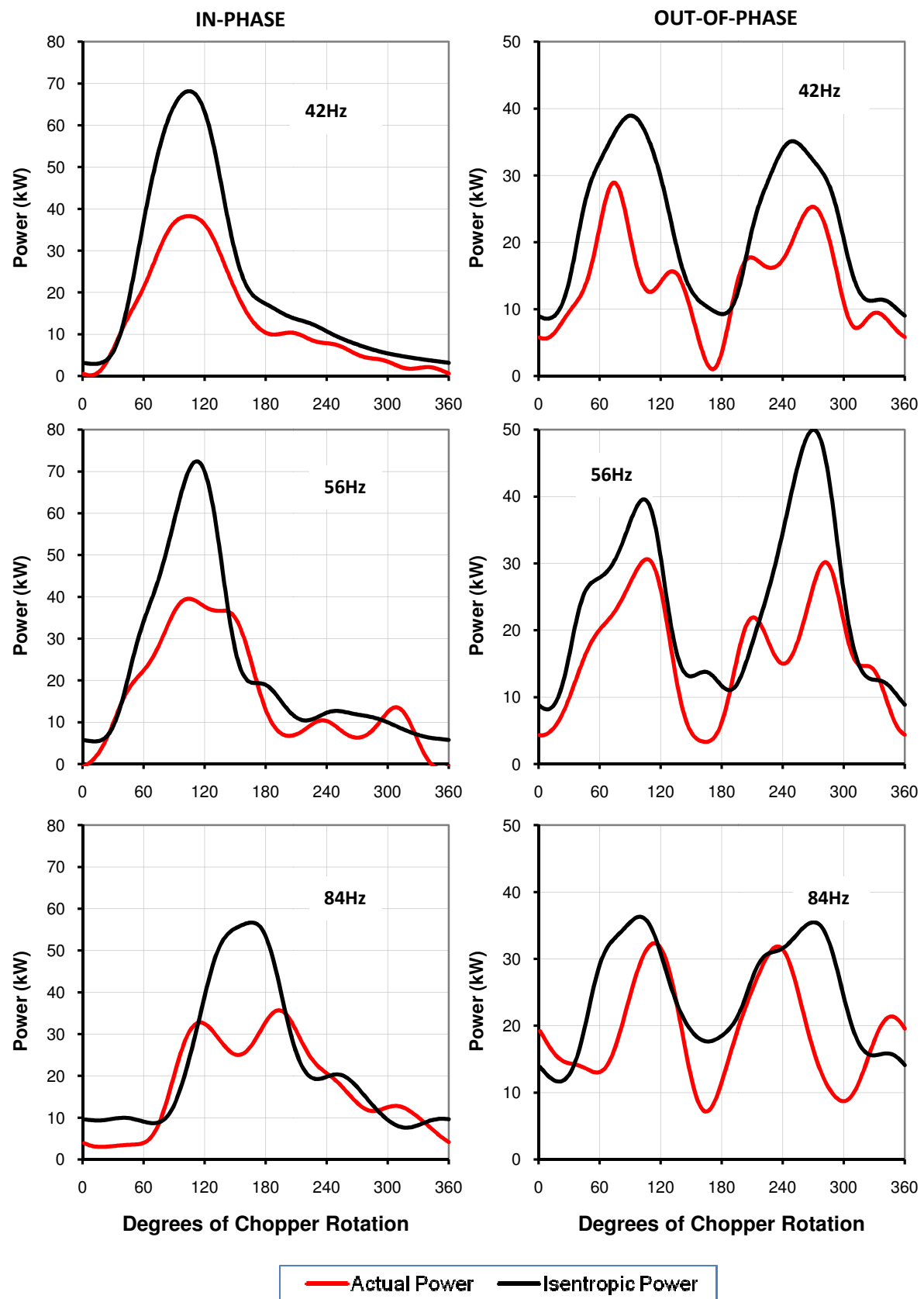


Figure 6.13: Isentropic and Actual Power traces (85% speed)

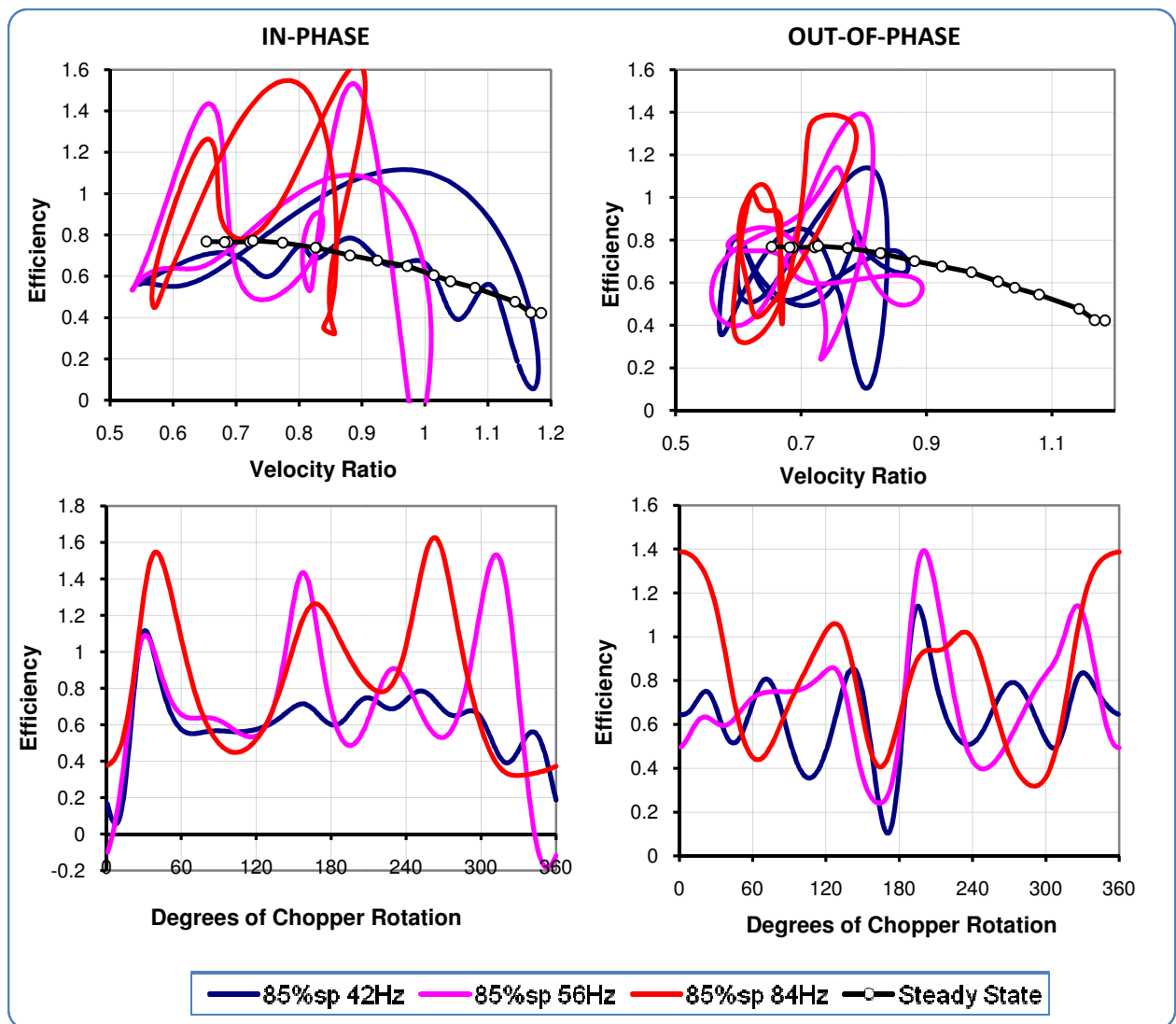


Figure 6.14: Instantaneous turbine efficiency plotted against velocity ratio and chopper rotation (85% speed)

Although the instantaneous efficiency plots are quite difficult to follow, they are not without some merit. For example, consider Figure 6.15 which shows the comparison between the steady-state efficiency and the instantaneous efficiency at 70% speed when driven by an out-of-phase 46Hz pulse. It is clear from this figure that the unsteady performance of the turbine is well below the steady-state operation. To investigate this disparity further and avoid the difficulties of the phase difference between isentropic and actual power, the cycle-averaged efficiency can be calculated using equation 6.7. Since the power quantities are averaged first and then divided, this parameter provides a value of turbine efficiency regardless of the dynamics occurring within the stage. These values have been provided in Table 6.3 to 6.5 for all of the test conditions from 50 to 85% speed. In addition, the cycle -averaged velocity ratio is provided as calculated from equation 6.8. This has been kept at an approximately constant value of 0.65 which corresponds to the location of the peak,

steady-state efficiency. This approach, of maintaining a constant value of cycle-averaged velocity ratio, has not been attempted by other researchers [24-32] due to the practical difficulty in keeping this value constant from experiment to experiment. The cycle averaged velocity ratio is the product of a large amount of post-processing, making it very time-consuming to test and adjust the operating point to achieve a given value. However, it was done here due to the importance in being able to compare between the different unsteady operating conditions and to compare with the steady-state operation as well.

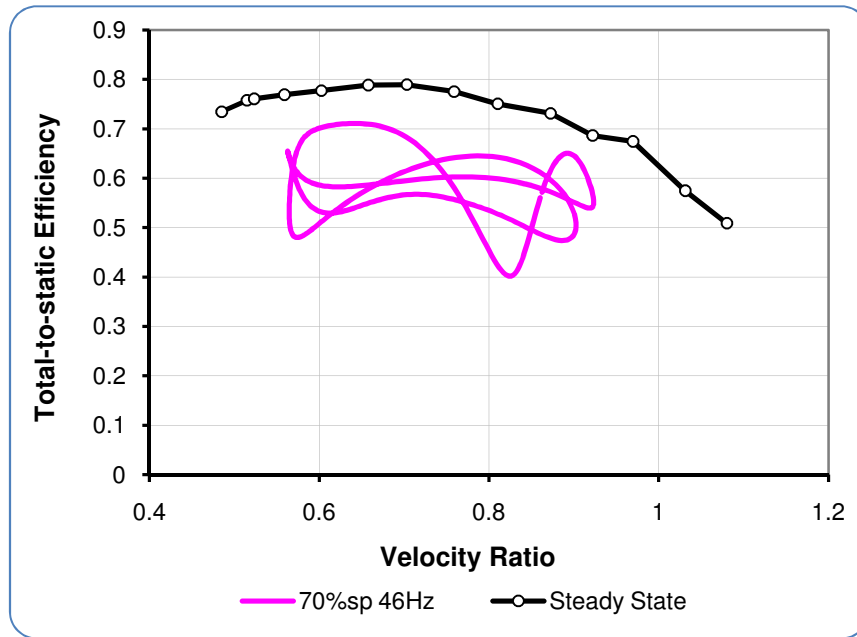


Figure 6.15: Instantaneous turbine efficiency plotted against velocity ratio (Out-of-phase, 56Hz, 70% speed)

Table 6.3: Cycle-Averaged Performance Data at 50% Speed

PHASE BETWEEN INLETS	FREQUENCY (Hz)	$N/\sqrt{T_0}$	CYCLE AVERAGED U_3/C_{is}	N/f	CYCLE-AVERAGED EFFICIENCY
IN-PHASE	24.91	27.74	0.6682	20.3902	0.4236
OUT-OF-PHASE	25.16	27.93	0.6794	20.33983	0.3937
IN-PHASE	33.3	26.82	0.667	14.68829	0.402
OUT-OF-PHASE	33.2	27.24	0.67	14.9759	0.355
IN-PHASE	50.5	26.49	0.655	9.611881	0.521
OUT-OF-PHASE	49.5	26.65	0.655	9.846465	0.545

Table 6.4: Cycle-Averaged Performance Data at 70% Speed

PHASE BETWEEN INLETS	FREQUENCY (Hz)	$N/\sqrt{T_0}$	CYCLE AVERAGED U_3/C_{is}	N/f	CYCLE-AVERAGED EFFICIENCY
IN-PHASE	34.9	37.64634	0.658	19.80802	0.5383
OUT-OF-PHASE	34.84	37.9982	0.6622	19.97704	0.5463
IN-PHASE	45.35	37.61785	0.646	15.16648	0.589
OUT-OF-PHASE	45.15	37.03374	0.646	15.0598	0.584
IN-PHASE	68.3	37.83969	0.646	10.14934	0.646
OUT-OF-PHASE	68.5	38.27905	0.652	10.29197	0.638

Table 6.5: Cycle-Averaged Performance Data at 85% Speed

PHASE BETWEEN INLETS	FREQUENCY (Hz)	$N/\sqrt{T_0}$	CYCLE AVERAGED U_3/C_{is}	N/f	CYCLE-AVERAGED EFFICIENCY
IN-PHASE	42.1	43.90397	0.658	19.7399	0.613
OUT-OF-PHASE	42.2	44.6376	0.654	19.59005	0.628
IN-PHASE	56.3	44.42592	0.656	14.8341	0.707
OUT-OF-PHASE	56.5	45.28064	0.655	14.80885	0.688
IN-PHASE	83.3	44.38343	0.6598	9.942377	0.776
OUT-OF-PHASE	83.3	44.55073	0.644	9.815126	0.772

Figure 6.16 shows the cycle-averaged, unsteady efficiencies plotted against the pulsation frequency for all eighteen unsteady conditions. First, it is apparent from this data that the cycle averaged efficiency drops quite dramatically at lower speeds. This is, to some extent, expected since a decrease in the speed at a constant velocity ratio means that the average pressure ratio is decreasing. This has shown to have a detrimental effect on the efficiency in steady-state testing. In order to compare the steady and unsteady performance on the basis of *average* parameters, the values of unsteady efficiency at a constant frequency of 50Hz have been obtained by linear

interpolation. This has been plotted against the value of the speed parameter in Figure 6.17. In addition, the values of steady-state efficiency as a function of the speed parameter (at a constant velocity ratio of 0.65) are also plotted in this figure. It is immediately apparent from this comparison that the efficiency of the turbine operating under pulsating conditions displays a significant reduction in performance. Yet it is critical to clearly understand the comparison that is being made here. This is not a quasi-steady comparison since the instantaneous variation of conditions has been ignored in favour of a comparison based on averaged parameters only. In other words, one can obtain an averaged value of velocity ratio and speed parameter from a turbine operating in pulsating flow. These two parameters can then be taken to the steady-state map to estimate the performance of the turbine. However, Figure 6.17 shows that such a comparison would result in a significant over-prediction of the unsteady performance. Figure 6.17 also shows the decrease in turbine performance as the speed parameter is reduced. At 50Hz, this reduction is slightly more severe than the decrease in steady performance at lower speeds.

The second interesting trend that can be taken from Figure 6.16 is the increase in cycle-averaged unsteady efficiency as the pulse frequency is increased. With the exception of a small dip at 50% speed (33 Hz), this increase can be seen for all three speeds and appears independent of the phase difference between inlet pulses. From the pressure traces shown in Figures 6.3 and 6.4, it would seem from the emergence of the secondary and tertiary peaks, that a rise in frequency produces an increase in the effect of reflective wave dynamics in the passages. It has also been suggested by some authors that a rise in frequency should represent a departure from quasi-steady operation. Thus, in light of this, the improvement in efficiency with frequency represented in Figure 6.16 may seem somewhat counter-intuitive. This influence of frequency is discussed in much further depth in comparison with the quasi-steady predicted performance in Section 6.5.

The final item of interest from Figure 6.16 is the comparison between the efficiency of the in-phase and out-of-phase cases. Although the 25Hz and 33Hz frequencies at 50% speed do show a small drop in efficiency due to the out-of-phase pulses, all others cases show very little influence to a phase difference. The right-hand plot in figure 6.18 shows the pressure in each passage of the turbine when the pulses arrive at the turbine inlet 180° out-of-phase. It is clear from this figure that this phase difference will result in a state of unequal admission for much of the pulse cycle. Bearing in mind that the turbine wheel rotates 10-20 times over a pulse cycle, it seems reasonable to expect that the out-of-phase efficiency should be lower than the in-phase case as a result of the unequal flows driving the rotor. However, this is clearly not the case in Figure 6.16. Unfortunately, it is difficult to state conclusively how the unsteady out-of-phase flow may differ from the steady,

unequal flow in Chapter 5 without of similar detailed CFD analysis. Nonetheless, a few suggestions can be made. First, the comparison between the out-of-phase pulsating flow and the unequal, steady flow assumes the development of the flow will occur in a quasi-steady manner. In other words, the same loss mechanisms seen in the CFD analysis in Chapter 5 will have time to develop in the pulsating flow. Thus, whether or not the turbine can be considered ‘quasi-steady’ will be examined in greater detail in Section 6.5. A second intriguing argument can also be suggested. A large proportion of the power lost in the partial admission case in Chapter 5 appeared to result from windmilling in the stagnant sector. If one assumes that the trough of the pulse corresponds to a similar stagnant region, then both the in-phase and out-of-phase pulses will produce similar regions of loss at different times. This is best illustrated visually in Figure 6.18. In the in-phase case, the second period of the pulse will result in low pressure around the entire circumference of the rotor. In the out-of-phase case, half of the rotor inlet will have low pressure over the whole pulse cycle. Thus the combination of windage loss in the rotor may result in a similar overall loss in efficiency. However, this argument is still largely conjecture until a fully unsteady CFD model is constructed.

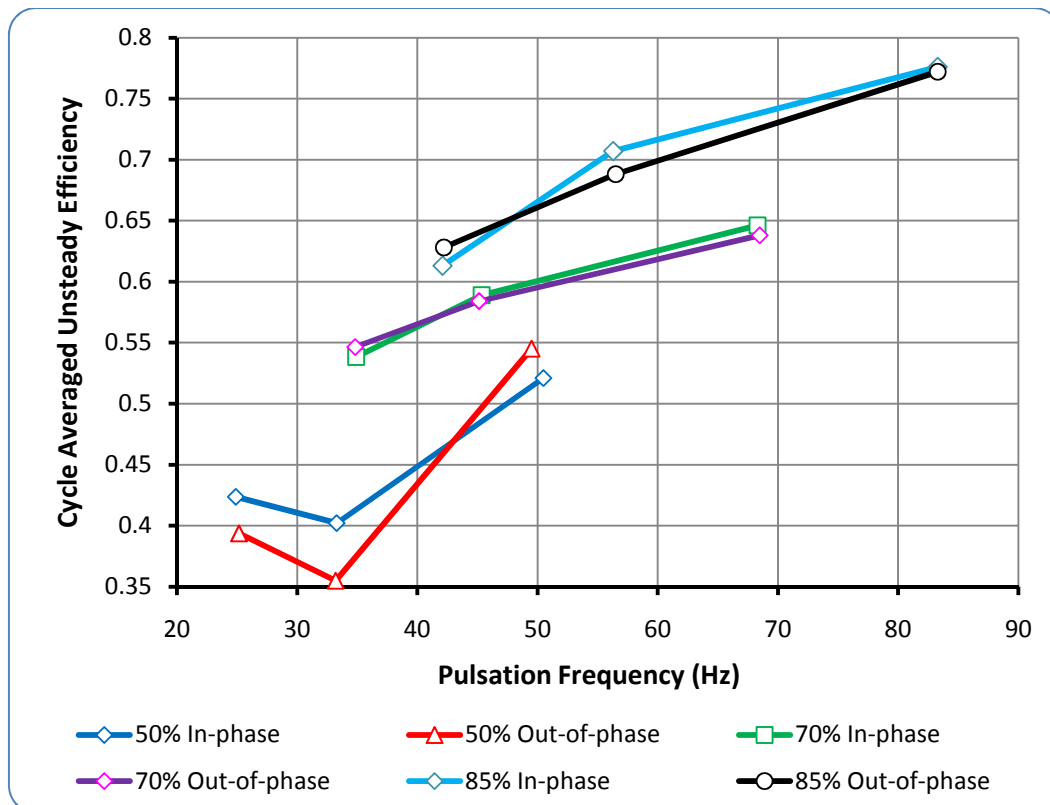


Figure 6.16: Cycle averaged efficiency as a function of frequency for 50%, 70% and 85% speed

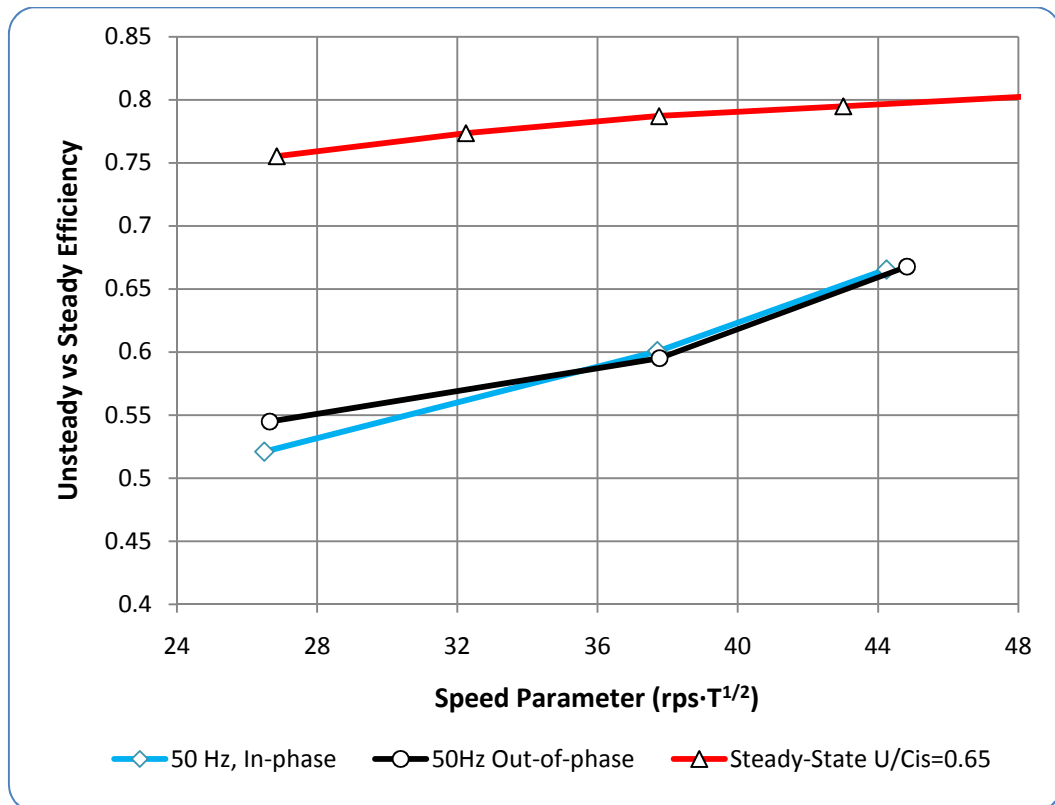


Figure 6.17: Cycle averaged efficiency as a function of speed parameter for 50% speed

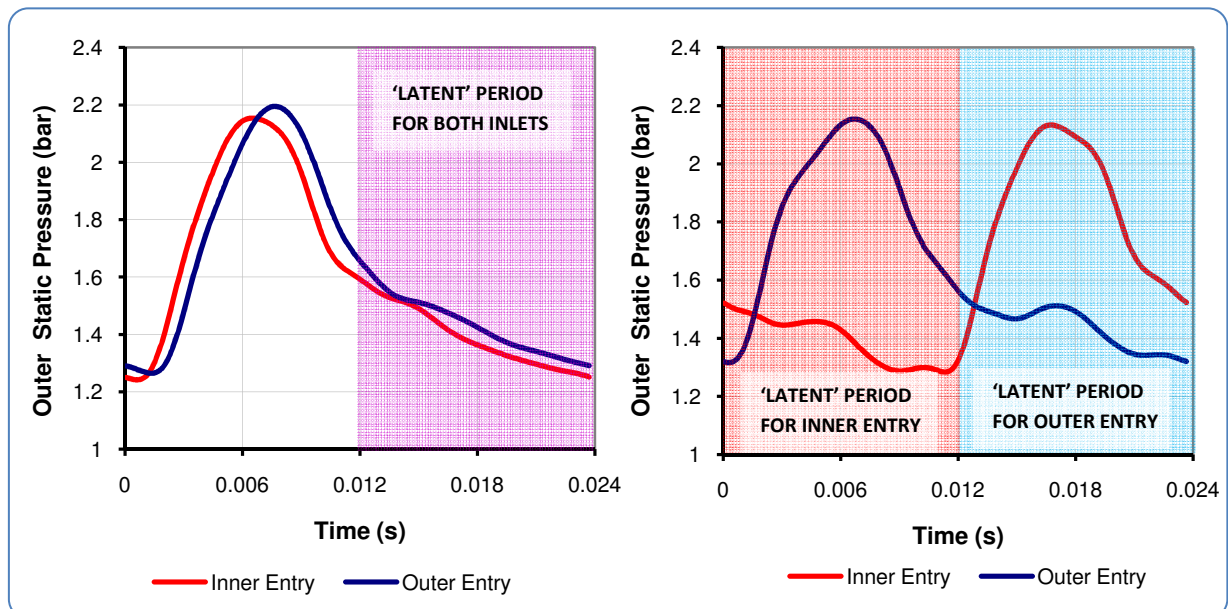


Figure 6.18: In-phase and out of phase 'latent' period comparison (85% speed, 42Hz)

Three-dimensional Operating Cycles

As illustrated in Chapter 4, in order to fix the operating point of the turbine in a double-entry turbine, three non-dimensional parameters are needed: PR_{INNER} , PR_{OUTER} and U/C_{IS} . This facilitated the creation of a colour-coded efficiency map interpolated from all the equal and unequal admission steady data (Figure 4.19). Also, since the velocity ratio and entry pressure ratios vary independently over the course of an unsteady pulse, the instantaneous operating point of the turbine will vary in time within this three-dimensional space. Thus by superimposing the unsteady traces onto the steady-state contours developed in Chapter 4, a better understanding of the unsteady operation is possible.

Figures 6.20 and 6.23 show the unsteady operating orbits in three-dimensional space for in-phase and out-of-phase operation respectively. Since these traces can be somewhat difficult to follow in three dimensions, they are reproduced from an equal and unequal admission perspective in Figures 6.21, 6.22, 6.24 and 6.25. The steady-state efficiency characteristics are also plotted in the background with a different colour scheme than used in early figures to provide a clear view of the unsteady operating traces. From these figures, as the frequency is increased, the variation of both velocity ratio and pressure ratio over a pulse is reduced, resulting in a more compact unsteady orbit. This decrease in overall variation of conditions was also noted in the mass parameter curves (Figure 6.11). Figure 6.19 demonstrates this more clearly by showing a reduction in amplitude of both the pressure ratio and wheel speed for the 42 and 84 Hz in-phase pulses. In addition, the dynamics of a reflecting pressure wave at 84Hz appears to create secondary peaks in the pulse shape. Since the length of the two volute passages is different, the influence of such pressure wave dynamics will differ for each inlet resulting in an asymmetric operating trace at higher frequencies (compare the 42Hz and 84Hz traces in Figure 6.25). These diagrams therefore suggest that the increase in frequency is linked to an increase in wave dynamics but also a reduction in the variation of pressure ratio and velocity ratio over a pulse cycle (in terms of amplitude). This suggests that despite the gas dynamics at higher frequency, the turbine is, in fact, less dynamic in terms of the overall variation of instantaneous quantities.

For the in-phase case, it is interesting to note that an orbit is formed in the unequal perspective in Figure 6.22 which indicates that the flow at rotor inlet is actually operating out-of-phase. This is simply caused by the phase difference that results from the dissimilar path lengths in this double-entry design. However, this is only a slight deviation compared to the out-of-phase trace in Figure 6.25 where the turbine appears to be operating in a region which, in steady flow, is most susceptible to unequal flow efficiency loss.

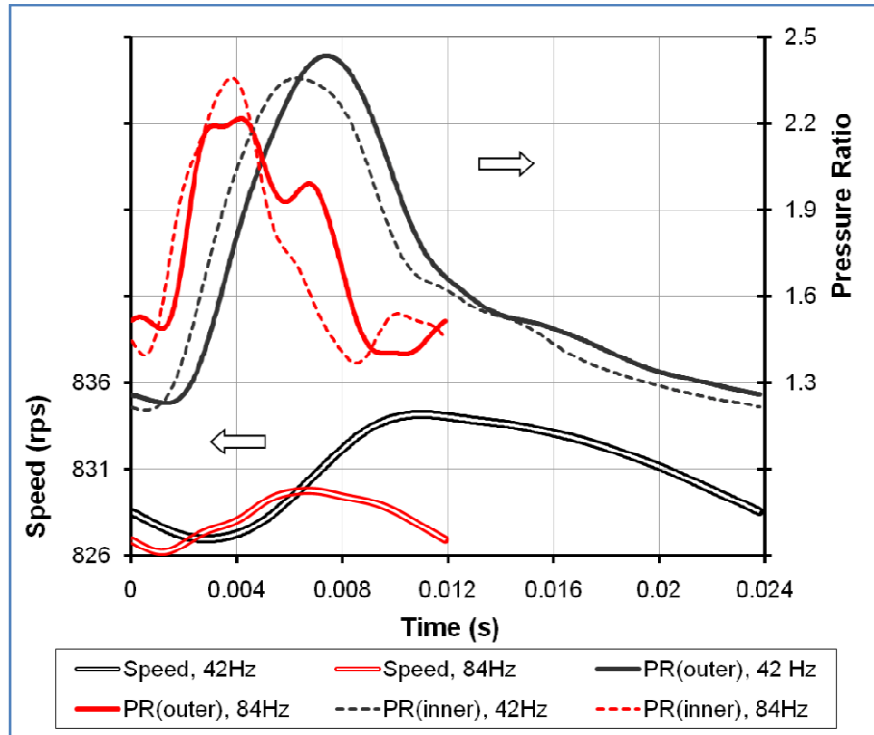


Figure 6.19: Unsteady pressure ratio and speed as a function of time (85% Speed, 42H, 84Hz)

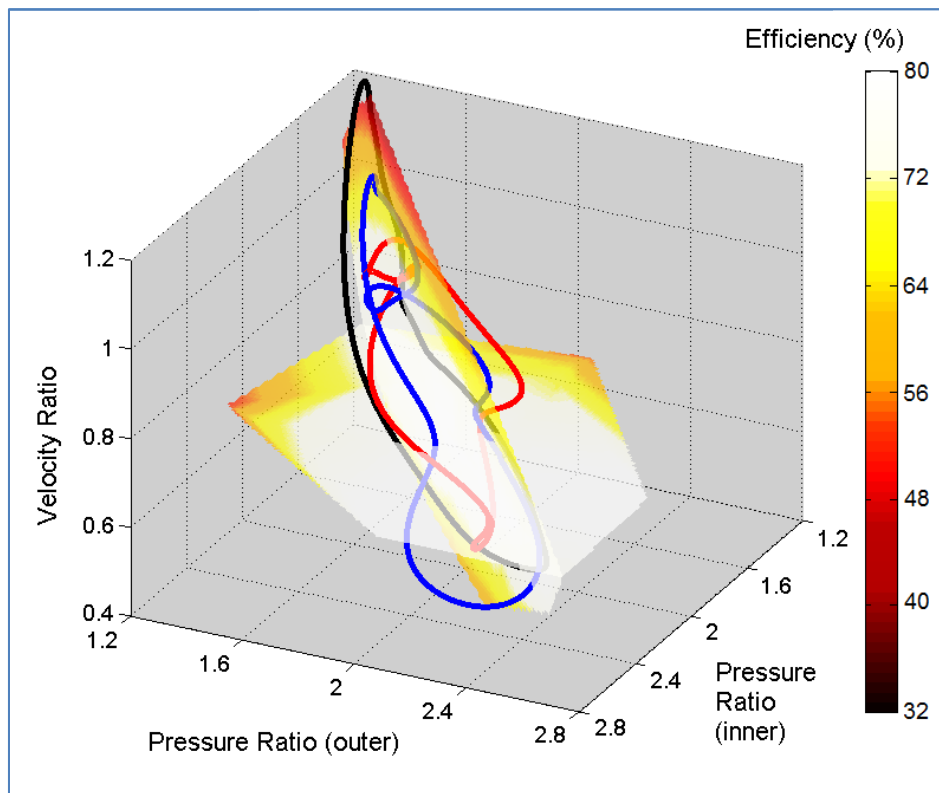


Figure 6.20: Steady-state efficiency contours with in-phase unsteady orbits superimposed (3D representation)

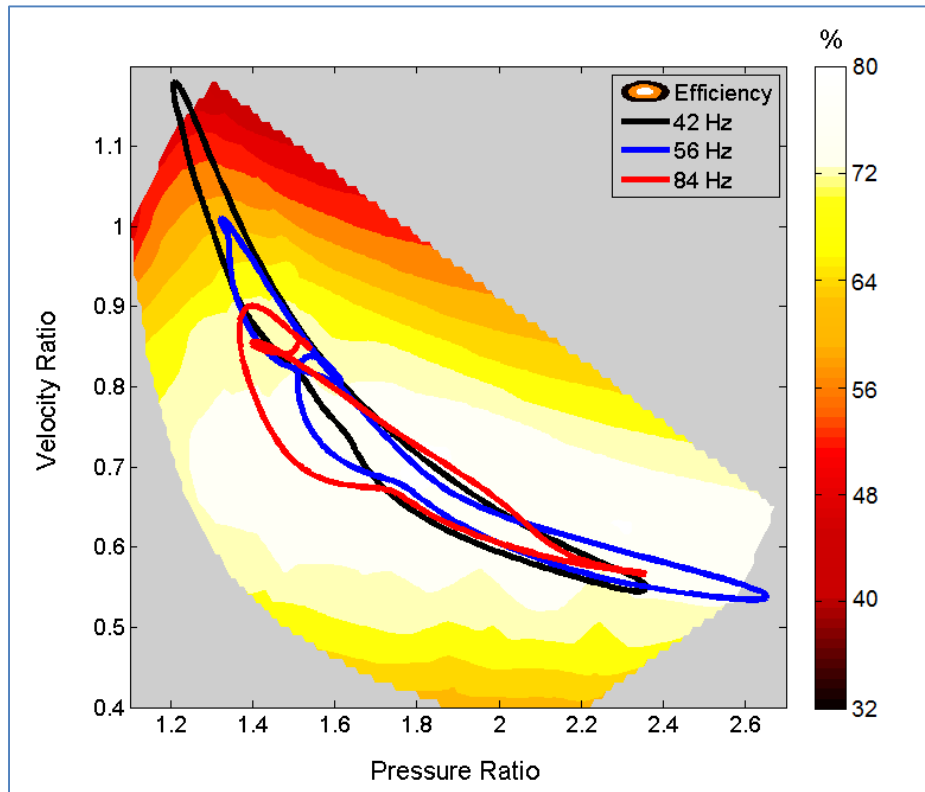


Figure 6.21: Steady-state efficiency contours with in-phase unsteady orbits superimposed (Equal admission perspective)

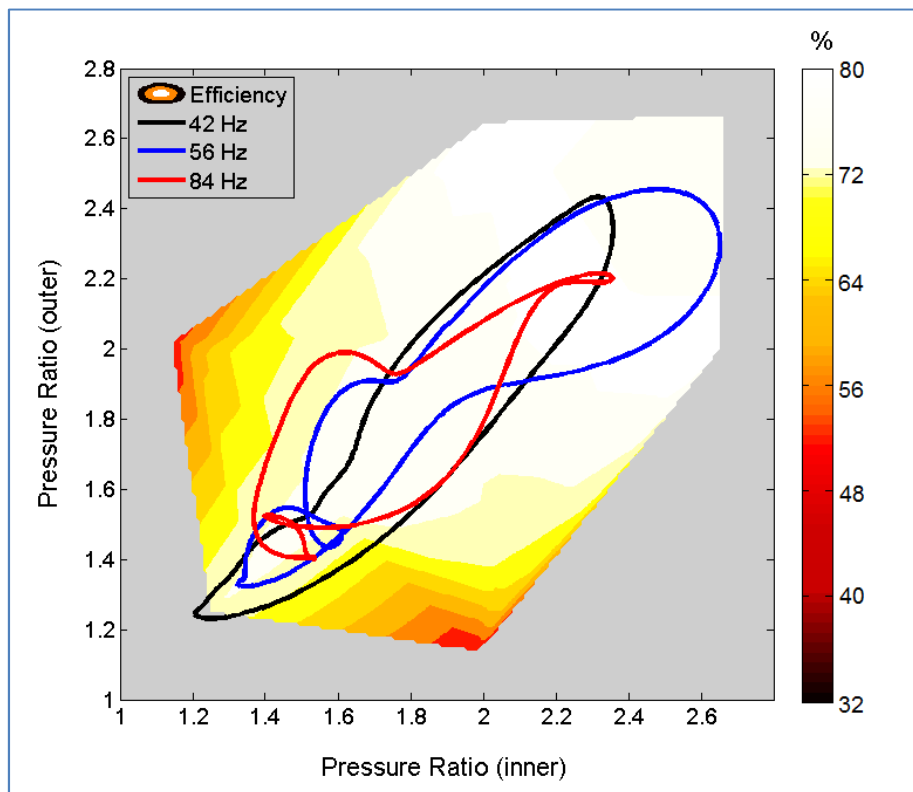


Figure 6.22: Steady-state efficiency contours with in-phase unsteady orbits superimposed (Unequal admission perspective)

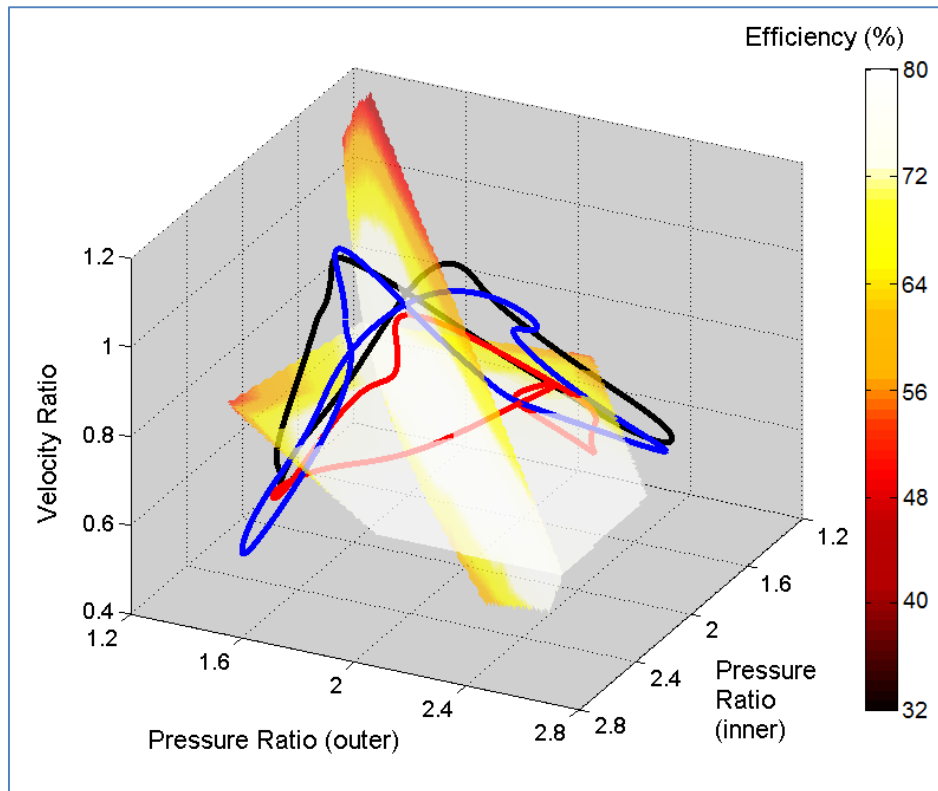


Figure 6.23: Steady-state efficiency contours with out-of-phase unsteady orbits superimposed (3D representation)

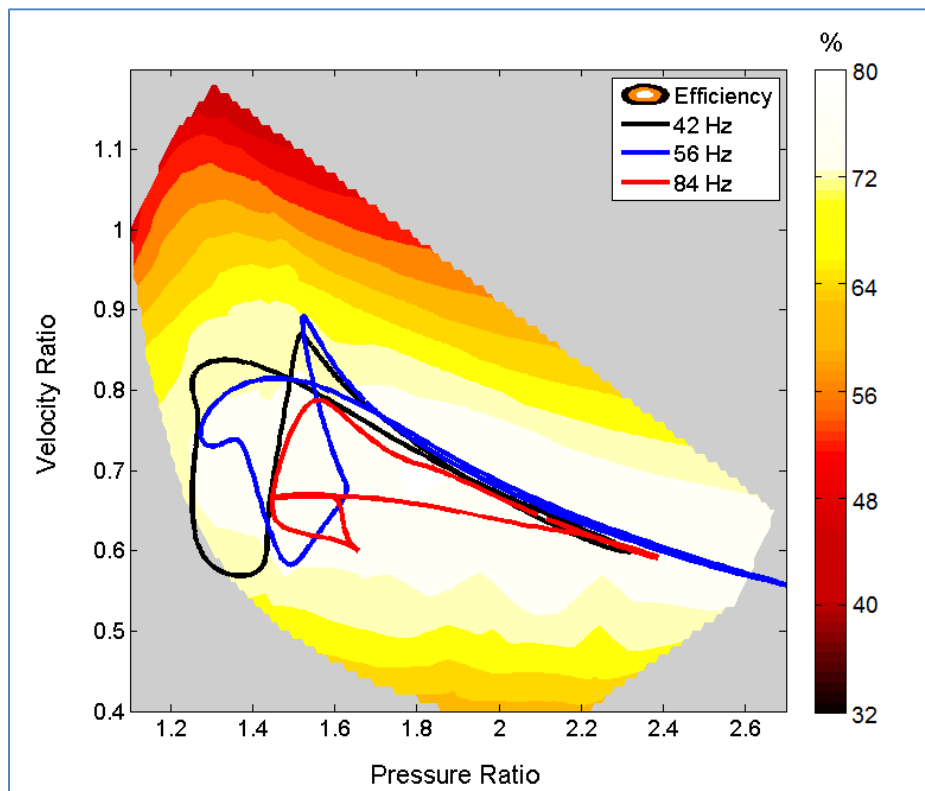


Figure 6.24: Steady-state efficiency contours with out-of-phase unsteady orbits superimposed (Equal admission perspective)

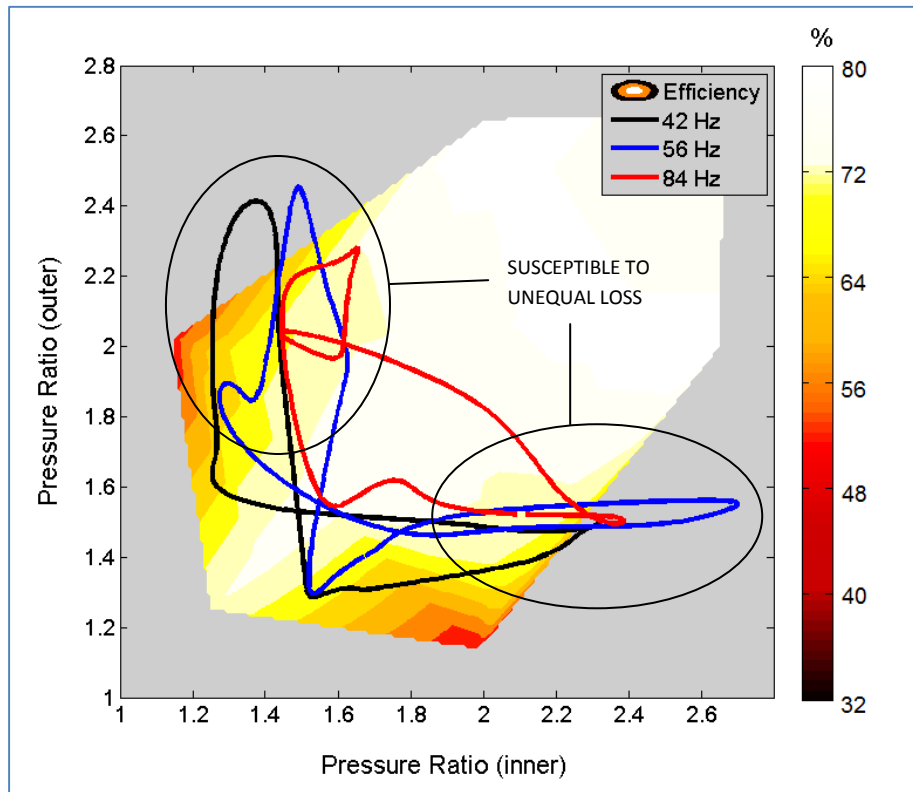


Figure 6.25: Steady-state efficiency contours with out-of-phase unsteady orbits superimposed (Unequal admission perspective)

6.5 Quasi-Steady Comparison

6.5.1 Introduction

The impact of a highly pulsating flow in a turbocharger has been demonstrated from the experimental data in the previous sections. This highly unsteady, pulsating flow causes a large variation in flow conditions in the turbine over a short amount of time. How the turbine will respond to this change in flow conditions has been a matter of some debate in research over the years. A few researchers have considered whether the turbine can be considered quasi-steady. This assumes that the behaviour of the turbine at each instant in the pulse cycle will be identical to the behaviour if the same set of conditions were introduced in a steady-state manner. If these instantaneous steady states are integrated together over the range of conditions encountered in pulse cycle, a quasi-steady prediction of unsteady performance can result. This requires an extensive steady turbine map so that each instantaneous unsteady operating condition can be matched to a corresponding steady-state operating point.

6.5.2 Steady-Unsteady Efficiency Comparison

Thus, a quasi-steady analysis involves comparing the instantaneous operating conditions of the turbine under pulsating flow to the performance data measured under steady-flow. For a single entry turbine, this is quite a simple procedure. Since the turbine speed does not vary significantly over a pulse cycle, a match between steady and unsteady data can be done simply on the basis of the velocity ratio at a single speed. Any interpolation or extrapolation needed is simply done along the speed line. Unfortunately, with a double entry turbocharger, the quasi-steady comparison is made much more difficult. The addition of a second entry means that there are three non-dimensional numbers that are required to fix the turbine operating point. A trace of the operating conditions that result from a single pulse cycle has been shown earlier in Figures 6.20- 6.25. Here the two pressure ratios across each of the inlets are plotted on the x and y axes, with the values of velocity ratio forming the third axes. Interpolation between the steady operating points therefore requires a three dimensional interpolation scheme to calculate a dependent variable such as efficiency as a function of the three dimensionless parameters. Only then can a point-by point comparison between steady and unsteady operation be made.

The accuracy of the interpolation procedure depended on the amount of data available from the steady tests. Therefore a total of 272 steady-state test points were measured encompassing a broad range of equal and unequal admission conditions (Figure 4.19). A program was then used to fit a hypersurface of the form $w = f(P_{Rinner}, P_{Router}, U/Cis)$ to these non-uniformly spaced experimental points using a Delaunay Tessellation-based linear interpolation. The algorithm interpolates the hypersurface at the points specified by a uniform Cartesian grid with a resolution of 0.01. Since at each test point, steady-state mass flow, power, and efficiency were measured or calculated, any of these variables could be interpolated as required. Some of the unsteady cases produced operating conditions outside the bounds of the steady-flow data making some extrapolation necessary. Fortunately, there was very little unsteady operation that was not covered by the steady data thereby considerably reducing the uncertainty of the analysis. The validity of the interpolation was checked against the original steady efficiency data to assure consistency. The interpolated data yielded a root mean square deviation of 0.3% from the measured points, thus providing sufficient confidence in the method.

Figure 6.26 shows a representation of the final interpolated and extrapolated efficiency data in three-dimensional space. An interpolated contour plot of isentropic power is also provided in the

appendix to demonstrate the availability of all the steady-state performance as a function of these three parameters. From this analysis it is possible to select a point in time during an unsteady pulse and compare it to the steady operation under the same conditions, giving quantitative indication of departure from quasi-steady operation. By substituting these interpolated *steady* values into Equation 6.7 that match the instantaneous unsteady conditions, a value of the cycle-averaged quasi-steady efficiency will result.

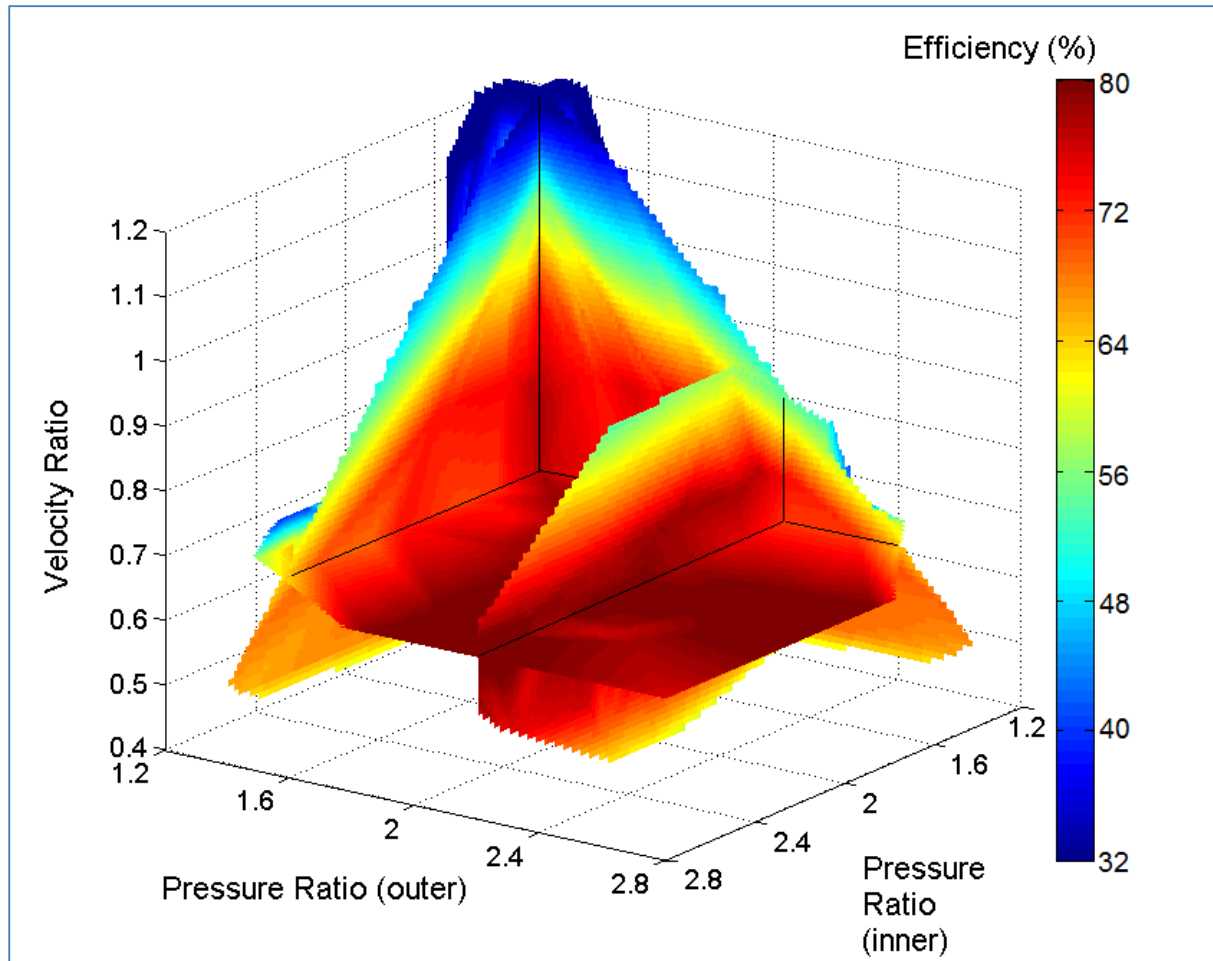


Figure 6.26: Interpolated steady-state efficiency as a function of velocity ratio, PR_{inner} and PR_{outer}

Figure 6.27 displays the results of cycle-averaged efficiency calculations for the 85% speed and plots the values against the frequency parameter N/f (turbine speed divided by frequency). The dashed light blue and green lines show the cycle-averaged, quasi-steady efficiency calculated from the interpolated steady-state data. The solid blue and red lines show the true unsteady cycle-averaged efficiency taken from Table 6.5. Bearing in mind that the values of λ equal to 10, 15 and 20 represent frequencies 84Hz, 56Hz, and 42Hz respectively, this figure compares the effect of frequency on the true unsteady turbine efficiency and the quasi-steady predicted efficiency. As

noted from Figure 6.16, there is a reduction in the unsteady efficiency of the turbine as the frequency decreases. Although the quasi-steady efficiency shows a similar trend, the slope of the decrease is much less than the measured unsteady values.

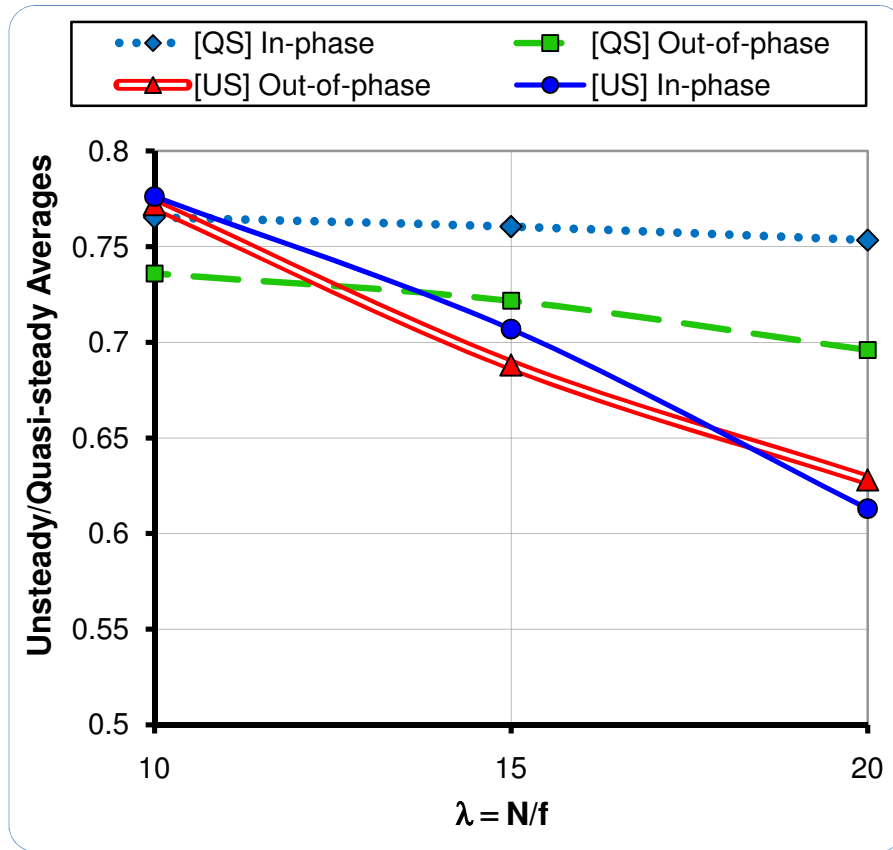


Figure 6.27: Unsteady [US] and Quasi-steady [QS] Efficiency versus Frequency Parameter

To understand the source of these efficiency trends, the unsteady values of mass flow and power were time-averaged and compared to the corresponding time-averaged quasi-steady values in Figures 6.28 and 6.29 using parameters I_M and I_p (Equations 2.4 and 2.5). With the exception of mass flow at 84 Hz, these figures show that the quasi-steady averages generally under-predict the corresponding unsteady values. Similar to the results of Kosuge et.al [20], there appears to be a contrasting influence of pulse frequency on the output power parameter I_p and the mass flow parameter I_M . The former decreases with decreasing frequency, while the latter increases. Thus one might explain the reason for the low unsteady efficiency at the low frequencies (high lambda) by the high mass flow rate of the inlet pulse when compared to the quasi-steady value; the turbine then seems unable to convert this energy into output power.

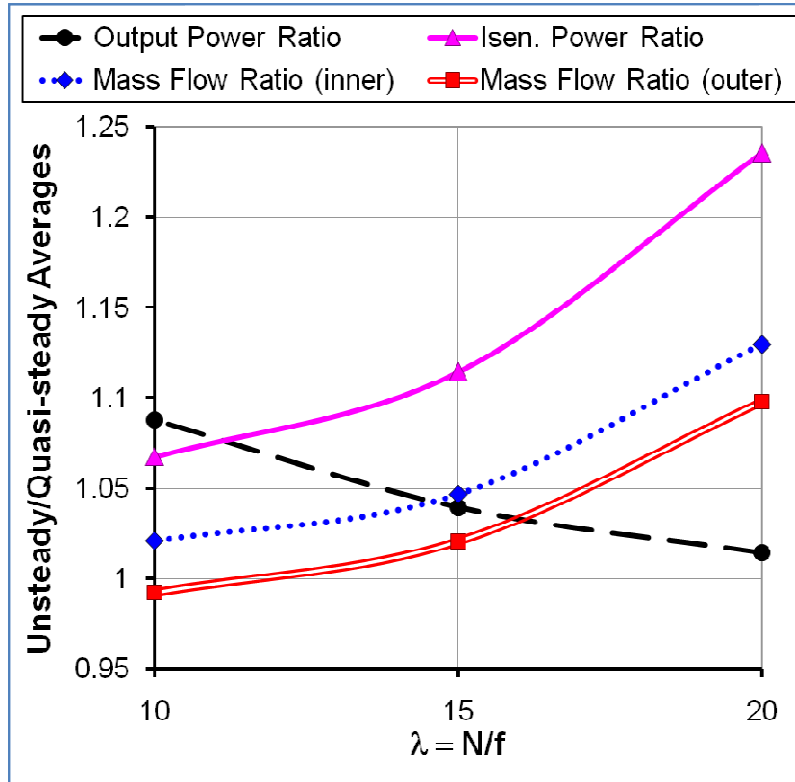


Figure 6.28: Ratio of unsteady and quasi-steady averaged power and mass flow (I_p and I_m)
IN-PHASE

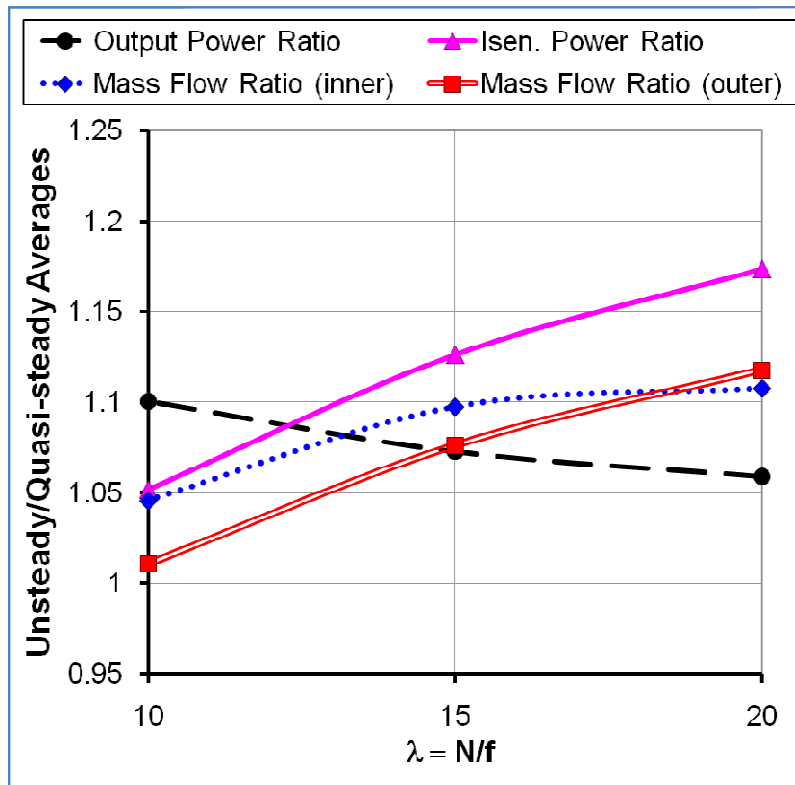


Figure 6.29: Ratio of Unsteady and quasi-steady averaged power and mass flow (I_p and I_m)
OUT-OF-PHASE

Figures 6.28 and 6.29 therefore show that, averaged over a cycle, the quasi-steady assumption under predicts the isentropic energy derived from the inlet conditions. To gain a greater understanding of this discrepancy, the instantaneous isentropic and output power traces used to calculate the averages in Figure 6.28 and 6.29 are shown in Figure 6.30. From this Figure it is clear that the difference between unsteady and quasi-steady turbine efficiency at the lowest frequency can be traced to a significant difference in the isentropic power at the peak of the pulse cycle. Since the quasi-steady data is obtained from matching the steady and unsteady operating points on the basis of pressure and velocity ratios, this difference in isentropic power must be the result of a difference of mass flow. Figure 6.31 therefore compares the trace of unsteady mass flow pulse measured from the hotwire to the mass flow predicted by the quasi-steady assumption. In a sense, this proposes the following thought experiment. At an instant in the pulse cycle, if the same entry pressure ratios are applied to a turbine operating with the same velocity ratio and allowed to reach steady-state, what would the swallowing capacity be? Figure 6.31 shows that the mass flow of this hypothetical quasi-steady case is less than the true mass flow measurement - especially at the peak of the pulse. This difference could be attributed to the 'filling and emptying' of the turbine volume during an unsteady cycle. Upon the arrival of a pulse, the volute volume will have had time to empty some of the mass it accumulated from the previous pulse. Thus, the turbine can swallow more mass compared to the steady-state case in order to fill this 'empty' volume. Although this is conjecture, this could explain why the turbine can swallow much more mass during an unsteady pulse than in the steady flow equivalent. It might also be able to explain why there is greater agreement between unsteady and quasi-steady mass flow at a higher frequency. As pulse frequency increases, the time available to empty the volume through the nozzles/rotor gets shorter, leading to less of a variation of mass contained in the volute. This decrease in the density variation could lead to flow behaviour that is closer to steady-state operation. This interpretation suggests a close link between the pulse frequency, the nozzle swallowing capacity, and the significance of the filling and emptying. In any case, as Figure 6.28 and 6.29 demonstrate, the end result is an unsteady, averaged mass flow value that is closer to the quasi-steady value ($I_m \rightarrow 1$) at higher frequencies. Thus, it is apparent that the turbine is able to swallow more mass than the quasi-steady assumption predicts at lower frequencies but does not translate into greater shaft power. This ultimately leads to a greater departure from quasi-steady performance at the lowest frequency.

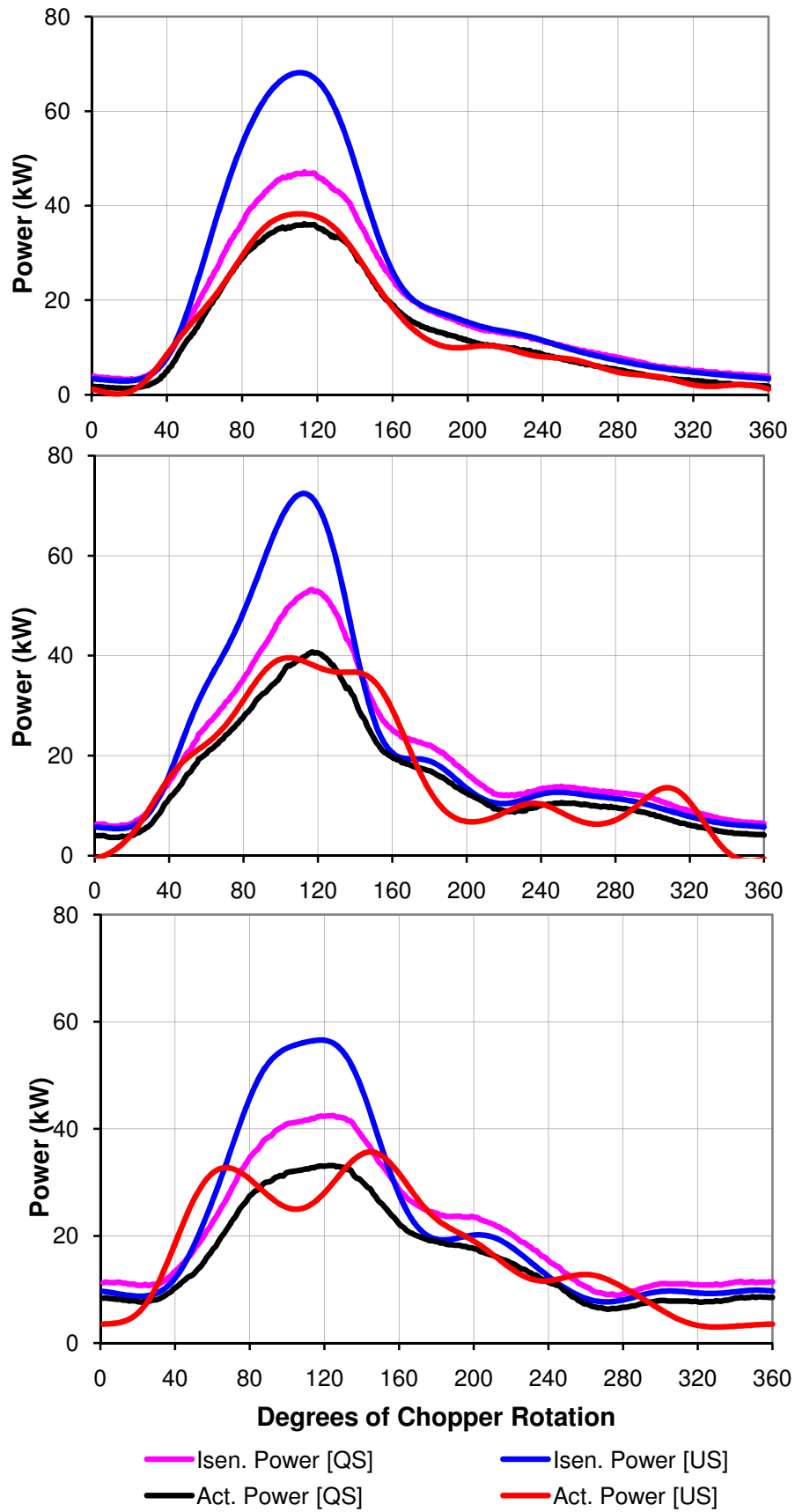


Figure 6.30: Unsteady [US] and quasi-steady [QS] power over one in-phase pulse (TOP: 42Hz, MIDDLE: 56Hz, and BOTTOM: 84Hz)

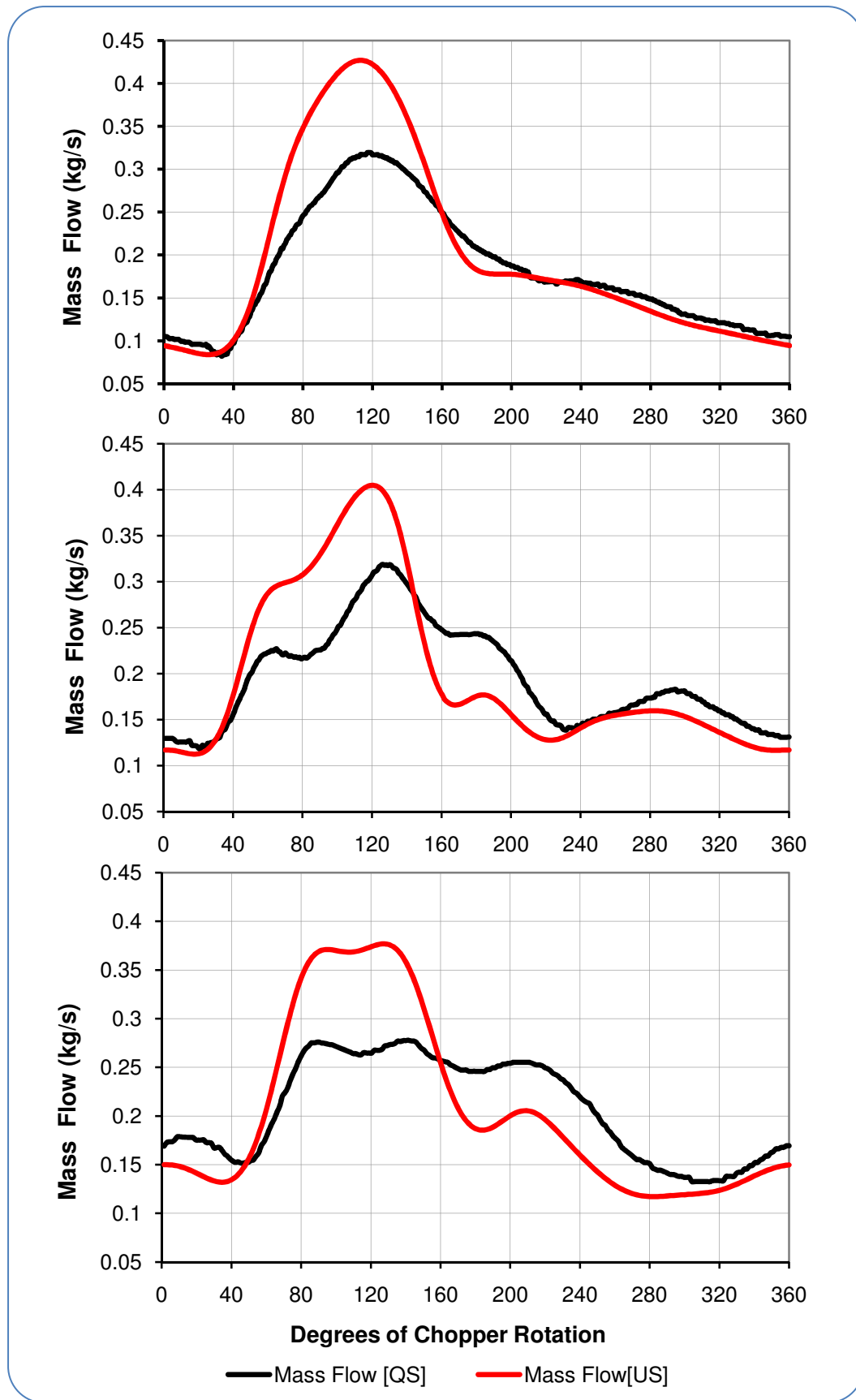


Figure 6.31: Unsteady [US] and quasi-steady [QS] mass flow over one in-phase pulse (TOP: 42Hz, MIDDLE: 56Hz, and BOTTOM: 84Hz)

6.5.3 Time scales and the Strouhal Number

The Strouhal number is a dimensionless parameter that has been used by a number of authors [7,8,32] as a criterion to define the onset of unsteady effects in a flow. In order to understand the source of this non-dimensional parameter, consider the dimensionless form of the Navier Stokes Equations. Time, velocity, pressure and temperature can be non-dimensionalized as follows:

$$t^* = \frac{t}{t_o}; \quad x_i^* = \frac{x_i}{L_o}; \quad u_i^* = \frac{u_i}{v_o}; \quad \rho^* = \frac{\rho}{\rho_o}; \quad p^* = \frac{P}{\rho_o v_o^2} \quad (6.9)$$

Where:

$\rho_o = \text{reference density}$

$t_o = \text{reference time}$

$L_o = \text{reference length}$

$v_o = \text{reference velocity}$

$\rho_o v_o^2 = \text{reference pressure}$

The continuity and momentum equations for compressible flow in dimensionless form thus become:

$$St \frac{\partial \rho^*}{\partial t^*} + \mathbf{1} \frac{\partial(\rho^* u_i^*)}{\partial x_i^*} = 0 \quad (6.10)$$

$$\begin{aligned} St \frac{\partial(\rho^* u_i^*)}{\partial t^*} + \mathbf{1} \frac{\partial(\rho^* u_j^* u_i^*)}{\partial x_j^*} \\ = \frac{\mathbf{1}}{Re} \left(\frac{\partial}{\partial x_j^*} \left(2 \left(\frac{\partial u_i^*}{\partial x_j^*} + \frac{\partial u_j^*}{\partial x_i^*} \right) - \frac{2}{3} \frac{\partial u_k^*}{\partial x_k^*} \delta_{ji} \right) \right) - \mathbf{1} \frac{\partial p^*}{\partial x_i^*} + \frac{\mathbf{1}}{Fr^2} \gamma_i \end{aligned} \quad (6.11)$$

Where the dimensionless numbers that result are defined as:

$$St = \frac{L_o}{v_o t_o}; \quad Re = \frac{\rho_o v_o L_o}{\mu}; \quad Fr = \frac{v_o}{\sqrt{L_o g}} \quad (6.12)$$

These dimensionless numbers are helpful to consider how the different terms scale in equation 6.10 and 6.11. For this particular discussion, the Strouhal number (St) helps to compare how the unsteady, time dependent term $\frac{\partial(\rho^* u_i^*)}{\partial t^*}$ scales in comparison to the other terms in Equation 6.11. If a characteristic frequency f_o is used instead of t_o , the definition of reduced frequency results:

$$\text{Reduced Frequency } (St) = \frac{L_o f_o}{v_o} \quad (6.13)$$

This term can be calculated using different definitions of reference properties that correspond to the particular case. Although it can sometimes be difficult to select an appropriate reference time, length and velocity, it is good to bear in mind what a reduced frequency represents. It is really a ratio of the time needed to travel a distance L_o to the time associated with the disturbance being considered ($1/f_o$). If the travel time is comparable to the time for a local change of condition due to some unsteady event, the Strouhal Number will be on the order one. This indicates that the unsteady term in equation 6.11 is on the same order as the convective term, thus suggesting significant unsteady effects.

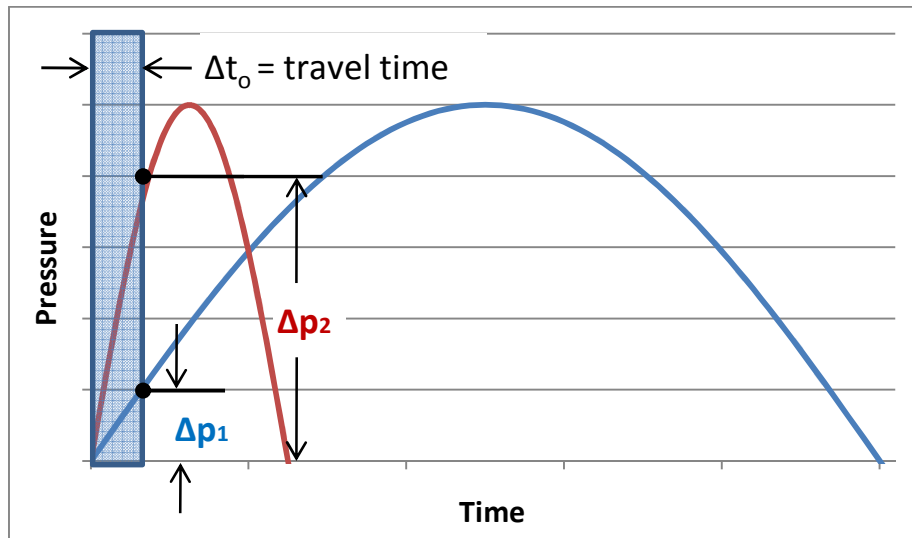


Figure 6.32: Time-to-travel versus pulse time with different frequencies

Figure 6.32 shows two cases with different pulse frequencies but the same flow travel time Δt_o over a distance L_o . Clearly, the higher frequency pulse will cause the flow to ‘feel’ a much larger change in pressure Δp_2 as it moves the distance L_o . Conversely, the lower frequency should be closer to quasi-steady behaviour since the flow only experiences a small change of pressure over the same

distance. The value of the Reduced Frequency/Strouhal number reflects this intuition since its value will be much higher in the second, higher frequency pulse.

Since the Strouhal number determines the importance of the transient term in the Navier Stokes Equations, it could suggest the pulsating flow conditions in a turbocharger that lead to a departure from quasi-steady operation. For a turbine, the basic definition of the reduced frequency in equation 6.13 is modified to include the parameter ϕ which accounts for pulse events which are a fraction of the total wavelength. This results in the definition of the Modified Strouhal Number (MSt) in equation 6.14

$$MSt = \frac{fL_o}{v_o} \frac{1}{\phi} \quad (6.14)$$

where v_o is the bulk flow velocity, ϕ is pulse fraction of wavelength (1/3 in current case), f is the pulsation frequency and L is the characteristic length. If the pressure wave velocity is used as the reference velocity instead of the bulk flow, one can compare the travel time of a pressure wave to the time of the pulse event. Thus, one can envisage a Pressure Modified Strouhal Number (PMSt) with the following definition:

$$PMSt = \frac{fL_o}{v_o + a} \frac{1}{\phi} \quad (6.15)$$

where a is the sonic velocity of the flow.

The choice of characteristic lengths used in Equations 6.14 and 6.15 will depend on the particular turbine domain that is being considered. Here the turbine stator and rotor will be considered separately with different values of the characteristic lengths and flow velocities. For the stator, Figure 6.1 shows the difference in lengths between the two volute passages. Also, as with any turbine, there is no single wheel entry point meaning that any length scale from volute inlet to rotor inlet is somewhat subjective. As before, the distance from the volute inlet, to an entry point 90° from the corresponding volute tongue will be used to evaluate the Strouhal number. The average time for the bulk flow to travel from gas inlet to the rotor is given in Table 6.6 and this can then be divided by the pulse cycle time $\times \phi$ to give the modified Strouhal number. These values are also supplied Table 6.6. The values for the Modified Strouhal number illustrate the unsteady term typically plays quite a significant role in defining the flow in the volute passages. This suggests significant unsteadiness in the volute. In the time taken for a particle to traverse the volute, 30% to 139% of the pulse ‘event’ (1/3 of the pulse cycle) will have been completed. The values of the

Strouhal number based on pressure wave travel in the volute are also given in Table 6.6. In the shorter inlet passage, the travel time of the pressure pulse is only ½ of a millisecond, meaning only 6.5% to 13% of the pulse event has been completed in this time.

Table 6.6: Stator flow time scales with Strouhal numbers

	42Hz, 85% Speed	56Hz, 85% Speed	84Hz, 85% Speed
Pulse cycle time $\times \phi$	0.00794	0.00595	0.00396
Inner bulk flow travel time (avg)	0.0026	0.0024	0.0025
Outer bulk flow travel time (avg)	0.0054	0.0058	0.0055
Inner pressure wave travel time	0.00052	0.00052	0.00052
Outer pressure wave travel time	0.00116	0.00118	0.00117
MSt (inner, average bulk flow)	0.327	0.403	0.631
MSt (outer, average bulk flow)	0.68	0.975	1.389
PMSt (inner, average bulk flow)	0.0655	0.0874	0.1313
PMSt (outer, average bulk flow)	0.146	0.198	0.295

Table 6.7 shows the flow travel times and the Strouhal numbers associated with the turbine wheel blade passage. The time for the bulk flow to travel through the rotor was estimated from the steady-state flow times given in Table 5.6 for the unequal admission analysis. In reality, this value will vary significantly over the course of the pulse cycle. The Strouhal numbers in Table 6.7 are small enough to suggest that the short distance of a blade passage produces a predominantly quasi-steady flow in the wheel— at least with respect to the chopper plate pulsations.

Table 6.7: Rotor flow time scales (in seconds) with Strouhal numbers

	42Hz, 85% Speed	56Hz, 85% Speed	84Hz, 85% Speed
Cycle time $\times \phi$	0.00794	0.00595	0.00396
Blade passage bulk travel time	0.00038 – 0.00046 – 0.00069		
Blade passage MSt	0.0479	0.0773	0.116

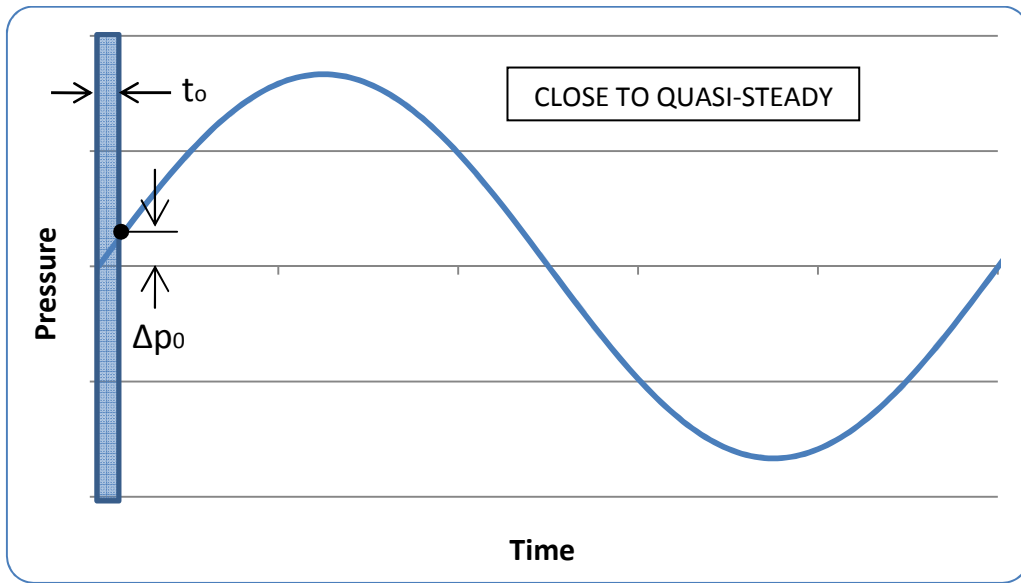


Figure 6.33: Low frequency, high bulk flow, high amplitude pulsation

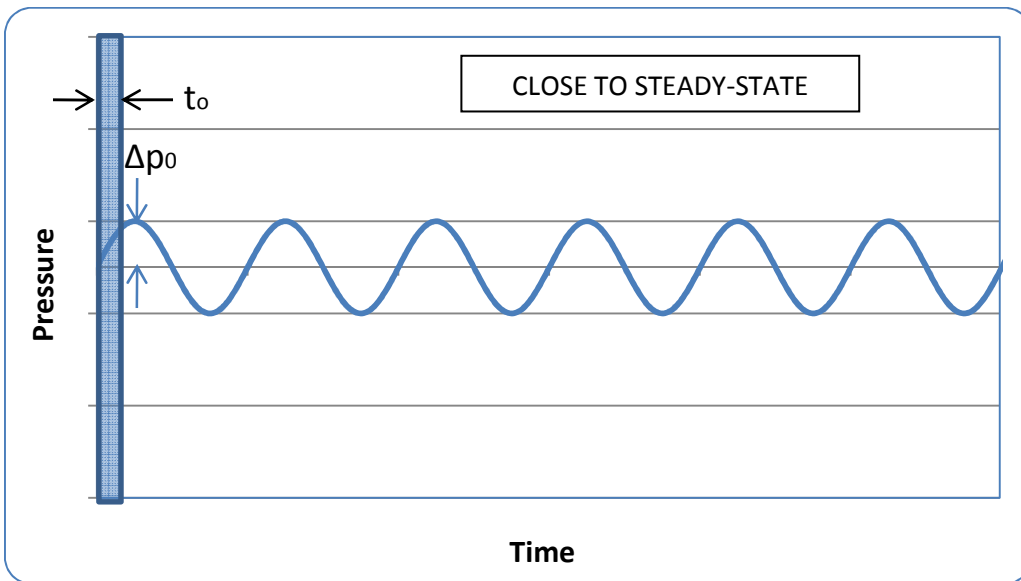


Figure 6.34: High frequency, high bulk flow, low amplitude pulsation

However, this leads to an intriguing question. If the Strouhal number is a good measure of ‘unsteadiness’, this would suggest that the higher frequencies (larger Strouhal number) should lead to a greater departure from quasi-steady operation. However, with regard to efficiency, the

experimental tests have shown the opposite. Figure 6.27 shows that the efficiency of the turbine is closest to the quasi-steady at the highest frequency. This suggests that the Strouhal number can only part of the picture. To illustrate, consider the following two cases:

1. Figure 6.33: Low frequency, high bulk flow pulsation. This will lead to a low value of the Strouhal number, indicating that the flow can be considered quasi-steady. Even though the amplitude of the pulsation is high, the flow is fast enough that the flowing fluid will not ‘feel’ the unsteadiness of the flow field.
2. Figure 6.34: High frequency, high bulk flow pulsation. The Strouhal number will be much greater than case 1 which would seem to indicate a flow where the unsteady effects are significant. However, as shown, the change in conditions Δp_o during the time to traverse the domain t_o , is comparable to case 1 due to the much lower amplitude. Thus as amplitude is reduced, the flow could be considered to be closer to a steady flow.

This example demonstrates what one might expect intuitively: there must be an effect of amplitude and pulse shape on how ‘unsteady’ a flow will be. Since in the current research the frequency, amplitude and pulse shape all change with chopper plate rpm, the Strouhal number may not provide a complete picture of the importance of unsteadiness in the turbine. Clearly, when the Strouhal number is very low, the flow can be considered quasi-steady since the flow will only feel a small change of upstream conditions. However, as the Strouhal number approaches unity, more information may be needed to determine how the unsteadiness will influence the overall performance.

6.5.4 Out-of-phase Pulsations

Figure 6.25 shows that during out-of-phase unsteady operation, the turbine is being fed unequally at the peak of each of the two pulses. The turbine is therefore operating in a region corresponding to decreased performance due to unequal admission. This observation is reflected in the lower out-of-phase quasi-steady efficiency value in Figure 6.27 (dashed green line). Bearing in mind that the turbine wheel rotates 10-20 times over a pulse cycle, it seems reasonable to expect that the measured out-of-phase, unsteady efficiency should similarly suffer. Interestingly however, the measured out-of phase values show very little difference from the in-phase unsteady efficiency. This would seem to suggest that the losses associated with steady, unequal admission are not as prevalent in conditions of steady, out-of-phase admission - or, as suggested using Figure 6.18, that the unsteady in-phase and out-of-phase loss generation may turn out to be very similar.

It is apparent that evaluating the ‘unsteadiness’ in a out-of-phase unsteady case becomes much more difficult due to the combined effect of unequal and unsteady, pulsating flow. The CFD analysis in Chapter 5 showed that unequal flows in a double entry design leads to an inherently unsteady flow phenomenon in the rotor due to the periodic change in the driving flow conditions as the wheel rotates between nozzle sectors. On top of that, the Strouhal number analysis suggests that for the range of frequencies tested here, the flow in the volute passages cannot be considered quasi-steady. Considering this combined effect, it is perhaps not surprising that the unsteady and quasi-steady out-of-phase results do not agree. Nonetheless, it is useful to visualize the different time scales involved in the out-of-phase case at 42Hz and 84Hz. Figure 6.35 superimposes the time for the rotor rotations on an out-of-phase 42Hz pulse cycle. It is immediately clear that since the time for a rotor rotation Δt_o is 1/20th of the pulse cycle, the predominant source of unsteadiness in *the rotor passage* will be caused by the unequal admission, not the chopper plate pulsations. As the frequency increases, there is a greater change in the pulse conditions during the time taken for a single rotor rotation. The highlighted blocks in Figure 6.36 show that in two rotor rotations, the rotor can see a dramatic change in unequal conditions. So not only will there be unsteadiness in the rotor caused by unequal admission, there may also be some unsteadiness in the rotor caused by the high frequency pulsations.

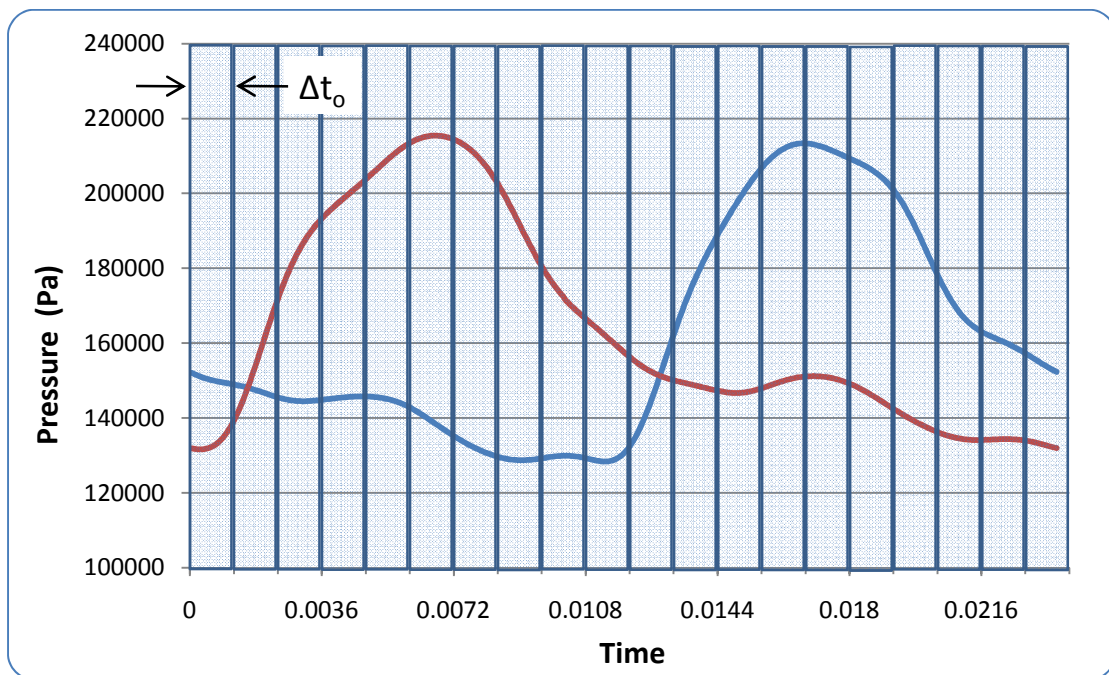


Figure 6.35: Rotor rotations completed during a 42Hz out-of-phase pulse cycle

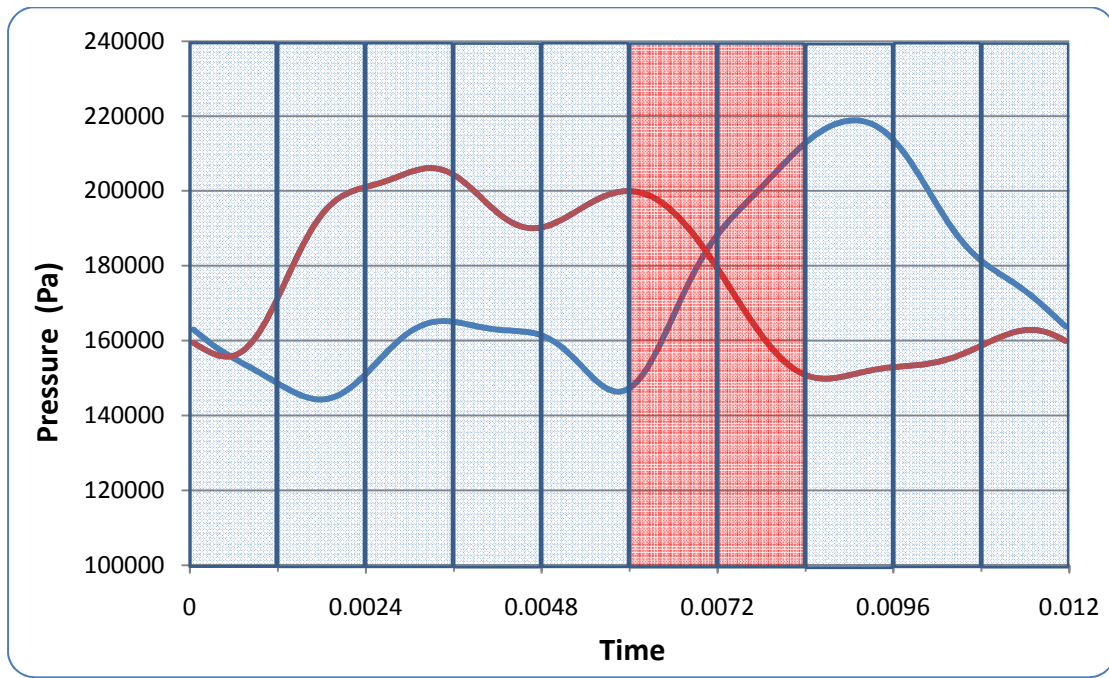


Figure 6.36: Rotor rotations completed during a 84Hz out-of-phase pulse cycle

6.6 Conclusions

The results from a range of unsteady pulsating flow experimental tests have been presented in this section. Three turbine speeds were tested ranging from 50% to 85% speed all maintaining a constant velocity ratio of 0.65. At each speed, three frequencies were tested that corresponded to N/f values of 10, 15, and 20 (turbine speed divided by pulse frequency). At each of the frequencies, the pulses were introduced at the two inlets either in-phase or 180° out-of-phase. Thus, in all, a total of 18 unsteady test cases were considered. To begin, the individual measurements of mass flow, pressure, speed and torque were all provided. As the frequency increased, the pressure profile in the longer, outer entry appeared to widen and develop secondary peaks to a greater extent than the shorter, inner entry. This suggests that both the frequency and length difference between the two passages have some influence on the development of inlet pressure profile. The frequency effect can likely be explained as an increase in the wave dynamics occurring in the stage as frequency is increased. The out-of-phase pressure pulse matched the in-phase profile very closely except in the trough of the pulse where the out-of phase case showed a second, small rise in pressure. Since this ‘secondary peak’ in the outer passage corresponds to the primary peak of the inner passage, this suggested that this small rise in pressure might be the result of a pressure exchange between entries. The same effect was also noted on the instantaneous mass parameter versus pressure ratio

plots. A primary loop was formed due to the hysteresis between the rise of pressure and mass flow in the main pulse. However, a secondary loop was also visible in the out-of-phase case that corresponded to the change in conditions in the latent entry during a pulse event occurring in the flowing entry. This once again suggested some additional interaction between entries in the out-of-phase case. The mass flow versus pressure ratio curves were discussed in comparison to the steady-state curves. At the lowest speed (50%), the orbit of the unsteady operation fully encapsulated the steady-state curve. This suggested a behaviour that was reasonably close to quasi-steady operation. At the 85% speed, however, the orbits show a trend that is not completely in line with the steady-state operation. The turbine was able to sustain a higher mass parameter at the peak of an unsteady pulse than the equivalent steady-state condition. The orbits also demonstrated that as frequency rose, they became more compact - showing less of a variation of inlet conditions at higher frequencies.

The attempt to evaluate the instantaneous efficiency of the unsteady pulses at 85% speed highlighted the difficulties inherent in evaluating the appropriate phase shift between the inlet conditions and the torque measurement. Since the pulse energy at the measuring plane will impact the wheel a moment later, the evaluation of instantaneous efficiency required a phase shift to match these two properties. Plotting the re-phased isentropic power with the actual power showed that output power derived from torque developed a series of double-peaks at higher frequencies that bore no obvious resemblance to the inlet conditions. This disparity between the trends in the power traces meant that the evaluation of instantaneous turbine efficiency must be taken with some reservations. Nonetheless, the trace of instantaneous efficiency compared to the steady-state curve did suggest that the turbine suffered a performance loss as a result of pulsating flow. Owing to the difficulty in evaluating instantaneous efficiency, the cycle-averaged efficiency was calculated for each unsteady condition. Here, the input power integrated over a pulse cycle was divided by the total shaft power to give a value of efficiency that was independent of pulse phasing. The values of cycle-averaged efficiency for all 18 test conditions were plotted against the frequency. This demonstrated a general trend of increased unsteady efficiency as frequency was increased. This analysis also showed that the in-phase and out-of-phase efficiency proved to be very similar in most cases. This was a notable result considering that out-of-phase pulsations produce conditions of unequal admission that are linked to a loss in efficiency.

In order to compare the steady and unsteady performance on the basis of *average* parameters, the values of unsteady efficiency at a constant frequency of 50Hz were plotted against the value of the speed parameter. These values could then be compared to the values of steady-

state efficiency operating at the same average speed parameter and velocity ratio of 0.65. It was apparent from this comparison that the efficiency of the turbine operating under pulsating conditions displays a significant reduction in performance. While not a quasi-steady comparison, this did show that a steady-unsteady comparison on the basis of averaged parameters will result in a significant discrepancy.

The three-dimensional representation introduced in Chapter 4 to show steady-state performance was also useful in representing the instantaneous operating trace of an unsteady pulse cycle. This allowed a greater understanding of the areas of unsteady turbine operation with respect to the steady-state efficiency map. The increase in pulse frequency had two visible effects on the unsteady operating traces. First, the amplitude of variation for both pressure ratio and velocity ratio decreased with increasing frequency, resulting in a more compact unsteady orbit. Second, there appeared to be an increase in the pressure wave unsteadiness resulting from the increasing influence of wave dynamics in the volute passages. Due to the two different passage lengths, the effect of this pressure wave interaction was noted to differ for each entry thus resulting in an increasingly asymmetric operating trace at higher frequency. The complete steady-state turbine map was interpolated in three dimensions allowing a direct point-by-point comparison between unsteady and steady-state turbine performance. This comparison was based on the quasi-steady assumption which presumes that the behaviour of the turbine at each instant in the pulse cycle will be identical to the behaviour if the same set of conditions were introduced in a steady-state manner. These instantaneous steady states were integrated together over the range of conditions encountered in pulse cycle to obtain a quasi-steady prediction of unsteady performance. Despite displaying a similar decreasing trend, the quasi-steady efficiency could not replicate the magnitude of unsteady efficiency drop with frequency. The reason for this departure at lower frequency was investigated further by calculating the quotient of unsteady and quasi-steady, time averaged mass flow and output power (I_M , and I_P). These parameters showed that the quasi-steady assumption under-predicted the magnitude of mass flow, especially at lower frequency. Therefore, the energy of the input waveform was greater than the quasi-steady value at lower frequency, but this did not result in additional output power.

For further understanding, the waveforms of quasi-steady and unsteady isentropic input power and mass flow were plotted in time. It was apparent from these plots that the isentropic power measured from the instantaneous inlet quantities was greater than the quasi-steady equivalent at the peak of the pulse. This difference could only be attributed to a difference in mass flow, which was suggested to result from the effect of ‘filling and emptying’. It was surmised that at lower

frequencies, the time available to swallow the mass introduced from a pulse cycle is greater, thus encouraging a greater flow field variation in the volute. At higher frequencies, the time for the nozzle and turbine to accept the mass contained in a pulse is less, conceivably resulting in a volume that exhibits less of a variation of total mass in the volume. Whatever the interpretation, the experimental data demonstrated that a significant efficiency loss can be attributed to high amplitude, low frequency exhaust pulsations.

The Strouhal number compares the time to travel a distance L_o to the time associated with the disturbance being considered ($1/f_o$). If the travel time is comparable to the time for a local change of condition due to some unsteady event, the Strouhal Number will be on the order one. The Strouhal numbers calculated for the stator domain suggested that the bulk flow could not be considered quasi-steady due to the importance of the unsteady term. In the rotor, however, the short passage length produced smaller values of the Strouhal number thus suggested a predominantly quasi-steady behaviour albeit only with respect to the pulsating flow. The CFD analysis in Chapter 5 showed that unequal flows in a double entry design leads to an inherently unsteady flow phenomenon in the rotor due to the periodic change in the driving flow conditions as the wheel rotates. This Strouhal number analysis led to an intriguing question. If the Strouhal number is a good measure of 'unsteadiness', this would suggest that the higher frequencies pulses should lead to a greater departure from quasi-steady operation. However, with regard to efficiency, the experimental tests have shown the opposite. Figure 6.27 shows that the efficiency of the turbine is closest to the quasi-steady at the highest frequency. Thus through a series of simple examples, it was suggested that there may also be a role of amplitude in determining the departure from quasi-steady operation. Since in the current research the frequency, amplitude and pulse shape all change with chopper plate rpm, the Strouhal number may not provide a complete picture of the importance of unsteadiness in the turbine. Clearly, when the Strouhal number is very low, the flow can be considered quasi-steady since the flow will only feel a small change of upstream conditions. However, as the Strouhal number approaches unity, more information may be needed to determine how the unsteadiness will influence the overall performance.

Chapter 7

Conclusions

7.1 Thesis Summary

This thesis presents the experimental and computation evaluation of the performance of a mixed-flow, circumferentially divided, turbocharger turbine for a wide range of full, unequal and unsteady (pulsating) flow conditions. The experimental tests were used to evaluate the response of the turbine when exposed to this wide range of flow conditions and the computational fluid dynamics was used to study the detailed flow dynamics that drive the observed experimental performance. The intent of this research was to further the understanding of the influence of unequal and pulsating flow on this type of turbine.

The core of the thesis contains three main divisions: Steady flow experimental results, CFD modelling, and unsteady flow experimental results. To begin, a comprehensive steady-state experimental data set was necessary to establish the base-line turbine performance. This data formed the basis of comparison with all other experimental data – both unequal admission and unsteady, pulsating flow. In order to gain an improved physical understanding of the flow driving the performance of the turbine measured in the laboratory, a full three-dimensional CFD model of the turbine was constructed using the ANSYS CFX software package. The intent of this step was not to model the on-engine turbine performance, but to replicate the laboratory conditions – specifically focussing on the unequal admission, steady conditions. The model was used to study the loss generation in the turbine due to the influence of unequal flow interaction. Pulsating air flow was then introduced using the range of frequencies typically produced by an internal combustion engine. The individual measurements such as temperature, pressure and mass flow were studied to note the effect of turbine speed and frequency. The standard performance parameters were also calculated on an instantaneous and cycle-averaged basis. The operating point of the turbine traced an orbit within a 3-D space defined by three non-dimensional parameters: velocity ratio, pressure ratio across inlet one, and pressure ratio across inlet two.

7.2 Experimental and Computational Conclusions

7.3.1 Steady, Equal Admission

A turbocharger that is attached to an engine will be driven by a highly pulsating flow resulting from the opening and closing of an engine exhaust valve. However, the first step in furthering our understanding of this unsteady, transient flow was to test the performance of the turbine under steady-flow conditions. A wide range of equal admission tests were possible due to the range of loading conditions available using the Imperial College eddy-current dynamometer. Five speed parameters were tested ranging from $26.9 \text{ rps}/T^{1/2}$ - $53.7 \text{ rps}/T^{1/2}$.

Plotting the total-to-static efficiency of the turbine revealed an optimal operating condition at a velocity ratio of approximately 0.66. The maximum efficiency was measured to be 80.4% at a speed parameter of $48.3 \text{ rps}/T^{1/2}$. However, the peak efficiency was relatively insensitive to rotor speed. In fact, the equal admission efficiencies all fell between 75% and 80% at the optimal velocity ratio, thus demonstrating the excellent performance of the ABB turbine design at the design point. Off-design flow conditions resulted in a minimum turbine efficiency of 43%, but this corresponded to the highest velocity ratio, and hence a very low power situation. A new, concise representation of turbine performance was suggested that shows the turbine efficiency as a function of velocity ratio and pressure ratio. This produced a 2-D contour plot of total-to-static efficiency calculated from the linear interpolation of all the equal admission test results. This clearly showed the areas where the inlet conditions produced peak turbine performance.

A three-dimensional CFD model was constructed in order to permit a detailed study of the flow in the double-entry design and answer specific questions regarding the observed steady-state performance. Since the impetus was to use the model to interpret and analyse the experiments, the pressure and temperature measurements made in the lab were used directly to specify the boundary conditions in this simulation. A total of five equal admission simulations were modelled, covering a range of velocity ratios at a constant speed of $48.3 \text{ rps}/\sqrt{K}$. This initial step was necessary, first in order to provide some preliminary verification that the model is a reasonable representative of the performance measured in the lab, and secondly to allow direct comparison with the unequal admission analysis.

The model provided an excellent prediction of the swallowing capacity in comparison with the experimentally measured values. This confirmed that the geometry of the model was an accurate representation of the real turbine thereby giving some confidence in the model. However,

the calculation of turbine efficiency did not fare as well as the mass flow prediction - showing a general over-prediction of the turbine performance. Nonetheless, the CFD model did predict the trend of the turbine efficiency by showing a peak in performance at a velocity ratio of 0.66 - followed by a drop as the velocity ratio was increased. The CFD was used to make a brief study of the inlet flow angles over the range of equal admission conditions. With the flows introduced equally, the disturbances near the tongue region appeared quite low, with no significant evidence of flow angle distortions in these regions. Over the range of steady-state conditions, the absolute exit flow angle from the volute matched the nozzle stagger angle of 70° , reducing any turning in the nozzle to a few degrees. The change in the magnitude of the inflow as velocity ratio is increased causes the average relative inlet flow angle β_3 to swing by 80.4° over the range of test conditions. This, along with an increase in the swirl exit angle leads to a decrease in the power available and the introduction of passage losses in the rotor.

7.3.2 Steady, Unequal Admission

The first step in understanding the performance implications of out of phase admission was to test the different combinations of unequal flow conditions at the two inlets in steady-state. Owing to the extra degree of freedom that an additional turbine inlet brings, a large range and combination of inlet conditions were possible. The turbine was tested at three different velocity ratios: 0.5, 0.65 and 0.8 - each with a range of pressure ratios. In order to present the data on a two-axis Cartesian representation, one of the inlets was held at a constant pressure ratio, while the second inlet was varied. This process was then reversed for the second inlet to test the response of each inlet separately. The plots of efficiency versus the ratio of inlet pressures revealed a marked reduction in performance due to unequal admission. At the most extreme, partially admitting the turbine resulted in a maximum efficiency drop of 16 to 34 points of efficiency depending on the velocity ratio. In contrast, when the pressure ratio in the varying inlet rose above the value in the constant inlet, a slight increase in efficiency resulted.

The effective area of each turbine inlet was evaluated so as to show the swallowing characteristics under unequal admission. The turbine entry with the varying pressure ratio showed an expected drop in effective area as the inlet pressure was reduced. Some dependence on the velocity ratio was noted and is likely attributable to the change in the centrifugal head created by the spinning turbine rotor. However, the change in effective area of the *constant pressure inlet* led to some very interesting analysis. Despite holding this entry at a constant pressure ratio, the

effective area varied due to the changes in pressure in the other inlet. This clearly showed the influence that one inlet had on the other. Compared to full flow, when the pressure in one entry is low, the second entry can swallow more mass, and when it is high, the second entry swallows less. Both the swallowing capacity and the efficiency of the two turbine inlets demonstrated remarkably symmetric lines under unequal admission. This indicates that, unlike a twin-entry turbine, the two inlets of a double-entry design are very similar in their response to changes in the flow conditions.

The performance results were combined to produce a series of two-dimensional contour plots showing the total-to-static efficiency of the turbine versus the two inlet pressure ratios. This allows the easy identification of conditions that lead to low turbine performance. Three such interpolated contour plots were created, showing the unequal admission efficiency at the three velocity ratios under investigation. Combined with equal admission, these novel two-dimensional representations of efficiency suggested that all possible flow conditions in a dual-entry turbine could be defined using three dimensionless numbers: two inlet pressure ratios and the velocity ratio. This therefore suggested a three-dimensional plot showing the entire range of steady-state performance. In this plot, the four planes of efficiency contours were appropriately placed thus giving a sense of the areas of optimum turbine efficiency with respect to the three independent parameters. In all, a total of 272 individual tests were performed on this turbine, measuring a broad range of both equal and unequal steady-tests. Thus fulfilled one of the main aims of this research: to completely characterize the steady-state operation of a double entry turbine.

Three-dimensional CFD was used to model the response of the turbine to four unequal conditions at a velocity ratio of 0.65. Like the experiments, the boundary condition of one inlet was held constant, while the second inlet was varied over the range of test cases from zero flow (partial admission) to a pressure ratio of 2.6 bar. The partial admission case was approximated using a wall boundary condition on the upper inlet. The mass flow characteristics of the CFD model showed a close match to the experiments over the range of boundary conditions. The simulations also reproduced the interaction between the volute passages that was noted in the experiments. This trend, where the flow rate in one entry is influenced by the pressure in the second, is accurately predicted by the CFD model. This was an important effect to be able to model and therefore allowed the examination of the physical effects that drive this trend. By studying the pressure around the circumference of the nozzle exit, it was found that the static pressure near the two tongues was significantly influenced by the pressure in the second entry.

The double-entry design is unique because each inlet passage feeds a separate section of the nozzle ring. Therefore, when fed unequally, the distribution of pressure and mass flow around the

circumference of the rotor inlet will not be uniform. This leads a situation where each blade passage experiences a variable flow as it rotates between feeding sectors. In order to understand how this will affect the turbine operation, the time required for the flow to pass through the rotor was compared to the time to cover one half of a rotation. If the bulk through-flow time is on the same order as this sector transit time, this will result in a flow field that is never able to fully establish itself in the blade passage. Using the circumferentially-averaged velocity through the rotor from three equal admission cases, it was estimated that the flow requires between 69% and 119% of the $\frac{1}{2}$ wheel rotation to completely travel through the wheel. To test this result, the development of the flow field in the rotor passage was examined in terms of pressure and velocity for the partial admission case at 50% span. The pressure showed significant disturbance in the interspace region prior to the leading edge of the first few blade passages in the flowing region. Furthermore, comparing the contours of relative velocity to the equal admission case suggested that the flow never completely achieves a consistent profile. All this analysis supported the supposition that the flow in the blade passages of a double-entry turbine that is unequally fed will not be able to completely establish thus making it a fully unsteady flow.

The unequal admission efficiency of the turbine predicted by the CFD model was also considered. Like the equal admission, the CFD failed to quantitatively predict the full efficiency loss due to unequal admission but did demonstrate the trend quite well. To gain further insight into where the model predicts loss generation, the difference in the static entropy was calculated across five main areas in the turbine for the partial and equal admission cases. As might be expected, the volute and the exit duct showed little influence due to unequal admission. The nozzle and the vaneless space prior to rotor entry, however, showed a significant increase in loss generation under partial admission conditions. However, the largest increase in the entropy occurred in the rotor. In order to visualize the loss creation on a local basis, the concept of an *entropy generation rate* was used. This led to a series of contour plots highlighting the areas of loss creation as the flow moved through the turbine. In the regions near the end of each sector, the high velocity flow from the lower entry formed a free shear layer with the stagnant flow from the upper entry, thereby resulting in regions of high loss. The largest single area of entropy generation occurred in the interspace volume on the non-flowing sector. Here the rotor blades windmill in a region of largely stationary fluid, resulting in a highly turbulent region of loss generation.

7.3.3 Unsteady, Pulsating Flow

In many ways the unsteady experimental tests are the culmination of the research program presented in this thesis. Three turbine speeds were tested ranging from 50% to 85% speed all maintaining a constant velocity ratio of 0.65. At each speed, three frequencies were tested that corresponded to N/f values of 10, 15, and 20 (turbine speed divided by pulse frequency). At each of the frequencies, the pulses were introduced at the two inlets either in-phase or 180° out-of-phase. Thus, in all, a total of 18 unsteady test cases were considered. In the main, most of the analysis was carried out using the 85% speed tests since this speed produced conditions corresponding to peak, steady-state operation. The differences in passage geometry between the inner and outer entries appeared to produce a differing influence of frequency on the shape of the profile. Also, the out-of-phase pressure profile bore some indications of an interaction between inlets under the conditions of instantaneous unequal admission.

The standard performance plots of mass parameter versus pressure ratio and efficiency versus velocity ratio provided a further insight into the effect of unsteady flow. At 50% speed, the orbit of the unsteady operation showed a similar trend to the steady-state curve thus suggested close to quasi-steady operation. At the 85% speed, however, the orbits show a trend that is not completely in line with the steady-state operation. The turbine was able to sustain a higher mass parameter at the peak of an unsteady pulse than the equivalent steady-state condition. The orbits also demonstrated that as frequency rose; the orbits became more compact - showing less of a variation of inlet conditions at higher frequencies.

The plots of efficiency versus velocity ratio were more difficult to interpret due to difficulties inherent in evaluating the appropriate phase shift between the inlet conditions and the torque measurement. Since the pulse energy at the measuring plain will impact the wheel a moment later, the evaluation of instantaneous efficiency required a phase shift to match these two properties. It was found that the traces of isentropic and output power ceased to match as the frequency was increased making the evaluation of instantaneous efficiency more difficult. Owing to the difficulty in evaluating instantaneous efficiency, the cycle-averaged efficiency was calculated for each unsteady condition. Here, the input power integrated over a pulse cycle was divided by the total shaft power to give a value of efficiency that was independent of pulse phasing. The cycle-averaged efficiency for all 18 test conditions demonstrated a general trend of increased unsteady efficiency as frequency was increased. This analysis also showed that the in-phase and out-of-phase efficiency proved to be very similar in most cases. The former suggested that despite the increase in wave dynamics in the

turbine at the higher frequencies, the performance of the turbine improved under these high frequency conditions. The later was notable considering that unequal flows in a turbine are normally linked to a loss in efficiency compared to equal admission. In order to compare the steady and unsteady performance on the basis of average parameters, the values of unsteady efficiency at a constant frequency of 50Hz were plotted against the value of the speed parameter. These values could then be compared to the values of steady-state efficiency operating at the same average speed parameter and velocity ratio of 0.65. While not a quasi-steady comparison, this did show that a steady-unsteady comparison on the basis of averaged parameters results in a significant discrepancy in the predicted performance.

The operating point of the turbine traced an orbit within a three dimensional space bound by three parameters: two inlet pressure ratios and the velocity ratio. The increase in pulse frequency had two visible effects on the unsteady operating traces. First, the amplitude of variation for both pressure ratio and velocity ratio decreased with increasing frequency, resulting in a more compact unsteady orbit. Second, there appeared to be an increase in the pressure wave unsteadiness resulting from the increasing influence of wave dynamics in the volute passages. Due to the two different passage lengths, the effect of this pressure wave interaction was noted to differ for each entry thus resulting in an increasingly asymmetric operating trace at higher frequency. Thus, these traces showed that the increase in frequency is linked to an increase in the observable wave dynamics but also a reduction in the variation of pressure ratio and velocity ratio over a pulse cycle. This suggests that despite the wave interaction at higher frequency, the turbine is, in fact, less dynamic in terms of the overall variation of instantaneous quantities

The quasi-steady efficiency only matched the unsteady efficiency at the highest frequency (84Hz). As the frequency was reduced (increasing N/f), the unsteady efficiency displayed a much greater decrease than predicted by the quasi-steady performance. The reason for this departure at lower frequency was investigated further by calculating the quotient of unsteady and quasi-steady, time averaged mass flow and output power (I_M , and I_P). These parameters showed that the quasi-steady assumption under-predicted the magnitude of mass flow, especially at lower frequency. Therefore, the energy of the isentropic waveform was greater than the quasi-steady value at lower frequency, but this did not result in additional output power.

For further understanding, the waveforms of quasi-steady and unsteady isentropic input power and mass flow were plotted in time. It was apparent from these plots that the isentropic power measured from the instantaneous inlet quantities was greater than the quasi-steady equivalent at the peak of the pulse. This difference could only be attributed to a difference in mass flow, which was suggested to result from the effect of 'filling and emptying'. In any case, the experimental data

demonstrated that a significant efficiency loss can be attributed to high amplitude, low frequency exhaust pulsations.

The Strouhal numbers calculated for the stator domain suggested that the bulk flow could not be considered quasi-steady due to the importance of the unsteady term. In the rotor, however, the short passage length produced smaller values of the Strouhal number thus suggested a predominantly quasi-steady behaviour albeit only with respect to the pulsating flow. The CFD analysis in Chapter 5 showed that unequal flows in a double entry design leads to an inherently unsteady flow phenomenon in the rotor due to the periodic change in the driving flow conditions as the wheel rotates. This Strouhal number analysis led to an intriguing question. If the Strouhal number is a good measure of ‘unsteadiness’, this would suggest that the higher frequencies pulses should lead to a greater departure from quasi-steady operation. However, with regard to efficiency, the experimental tests have shown the opposite.

The research presented here suggest that despite the increase of pressure wave unsteadiness, the turbine is better able to convert high frequency, lower amplitude pulsations into usable power. It also suggests that while the quasi-steady assumption can reflect some of the rough trends of the unsteady performance; it is not capable of giving quantitative predictions of the turbine efficiency with unsteady admission.

7.3 Future Work

7.3.1 Experimental

The steady-state experimental tests for this particular turbine were very exhaustive - covering a wide range of equal and unequal admission conditions. However, there are specific areas where an even broader range of data could be useful. An asymptotic limit of the mass parameter for this turbine was not reached due to limitations in the power absorption capacity of the dynamometer. There are a number of ways to improve the dynamometer power absorption which would lead to an even broader range steady-state data at higher pressure ratios. First, the cooling capacity of the dynamometer system could be improved to increase heat transfer through the stator plates. It is also possible to change to material of the stator plates to increase the power density of the device.

The other area where the test facility itself could be improved is in providing live data feeds during steady and unsteady testing. Many of the tests completed for this thesis involved maintaining a specific performance parameter such as velocity ratio or pressure ratio at a constant value. This is very important since these constant parameters form the basis of comparison between different tests. However, due to a combination of hardware and software limitations, it was not possible to monitor these performance parameters live on screen. Thus, maintaining constant values was only possible through an involved process of trial and error where the acquired data was processed after each test to see if the right value of the parameter had been achieved. Much of the difficulty is the result of the inability to calculate total values of pressure and temperature. On the hardware side, the capability of simultaneous measurements of mass flow and pressure must be set up so as to provide a live feed to the computer. The software must then be re-written to calculate and display live values of the different performance parameters.

There are a number of interesting avenues to expand on the experimental research presented in this thesis. For example, the effect of different nozzle areas could lead to a profitable investigation. This is easily implemented since all of the equipment setup up for the current research can be used. It has been shown Rajoo [32] that the nozzle opening has a large impact in the unsteady-flow swallowing capacity characteristics. Interestingly, the level of agreement between steady and unsteady behaviour seemed to be heavily influenced by the nozzle geometry. Thus, this investigation could lead to an improved understanding of flow unsteadiness in the turbine.

Lastly, more work needs to be done on how best to define certain unsteady parameters: specifically the instantaneous efficiency and the instantaneous pressure ratio. The instantaneous efficiency is quite a difficult parameter to be able to calculate due the necessity of measuring stage efficiency from properties at the turbine inlet. CFD modelling may be the best way to develop a method to define the instantaneous efficiency. Improvements to the experimental technique may also be found. For example, defining the isentropic power on the basis of the instantaneous pressure ratio across the rotor using pressure transducers before and after the rotor may show a better match to the torque that is produced. Small pressure transducers are available that could be mounted in the interspace between the nozzle and rotor blades.

7.3.2 Computational

The biggest contribution to further the research in this thesis would be to use the three-dimensional CFD model to study the unsteady flow in the double-entry turbine. One of the main

challenges is to appropriately define the fluctuating inlet boundary conditions. Since the pressure and mass flow measured in the laboratory are the result a superposition of entering and reflected waves, this inlet boundary must be quite sophisticated. First, unlike a standard boundary condition, it must allow reflected waves to pass through the boundary unhindered. Secondly, the specified pressure must be representative of the superposition of all the wave dynamics that occur in the experimental facility. Assuming these hindrances are surmountable, the result of such a simulation could greatly increase our understanding of the unsteady flow physics that drive the experimental observations made in this thesis. For example, a great deal of light could be shed on the reasons for the differences between unsteady and quasi-steady performance. A full CFD solution will allow the direct comparison of the entire steady and unsteady flow field.

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Appendix

Error Analysis

Figures A1 to A6 provide the uncertainty associated with the steady-flow performance parameters calculated by propagating the individual measurement uncertainties using equations 4.25 – 4.29. The uncertainties are plotted against the variation of pressure ratio and velocity ratio in the experiment to give some sense of the conditions which produce the greatest uncertainty. The uncertainty in the unequal admission effective area at a velocity ratio of 0.65 was plotted against the ratio of inlet pressures in a similar manner as figure 4.13, again to provide comparison with the experimental conditions. Refer to section 4.5 for a discussion of these plots and their implications on the performance assessment.

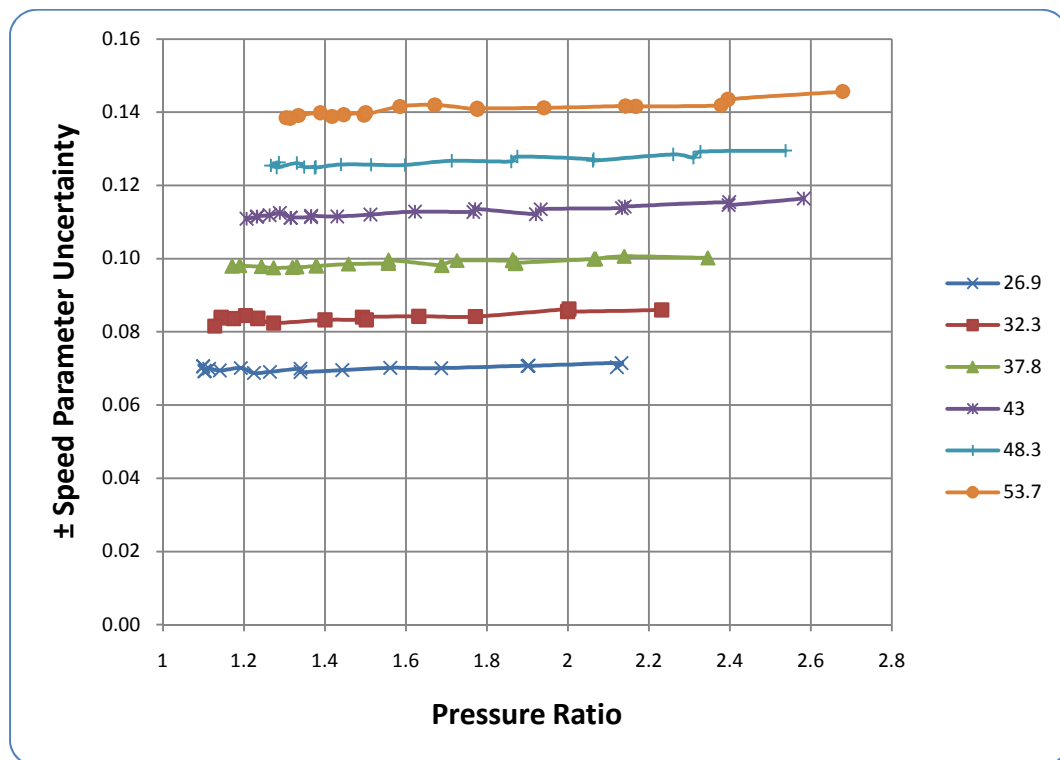


Figure A1: Uncertainty of the steady-state speed parameter against pressure ratio

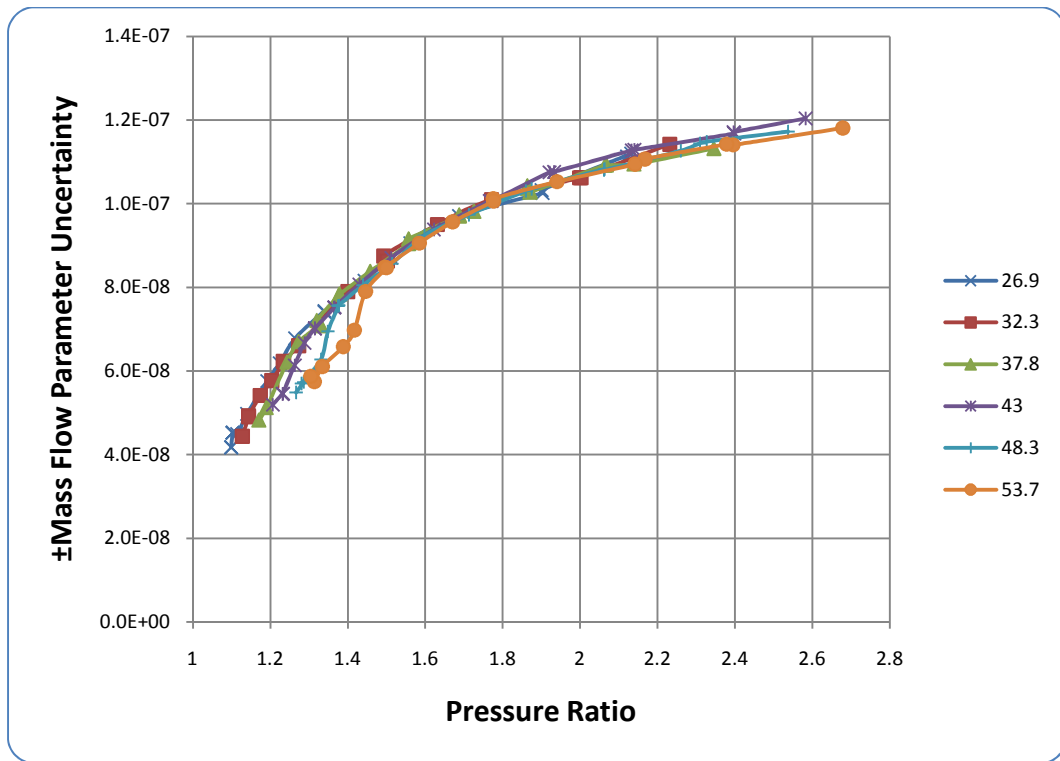


Figure A2: Uncertainty of the steady-state mass flow parameter against pressure ratio

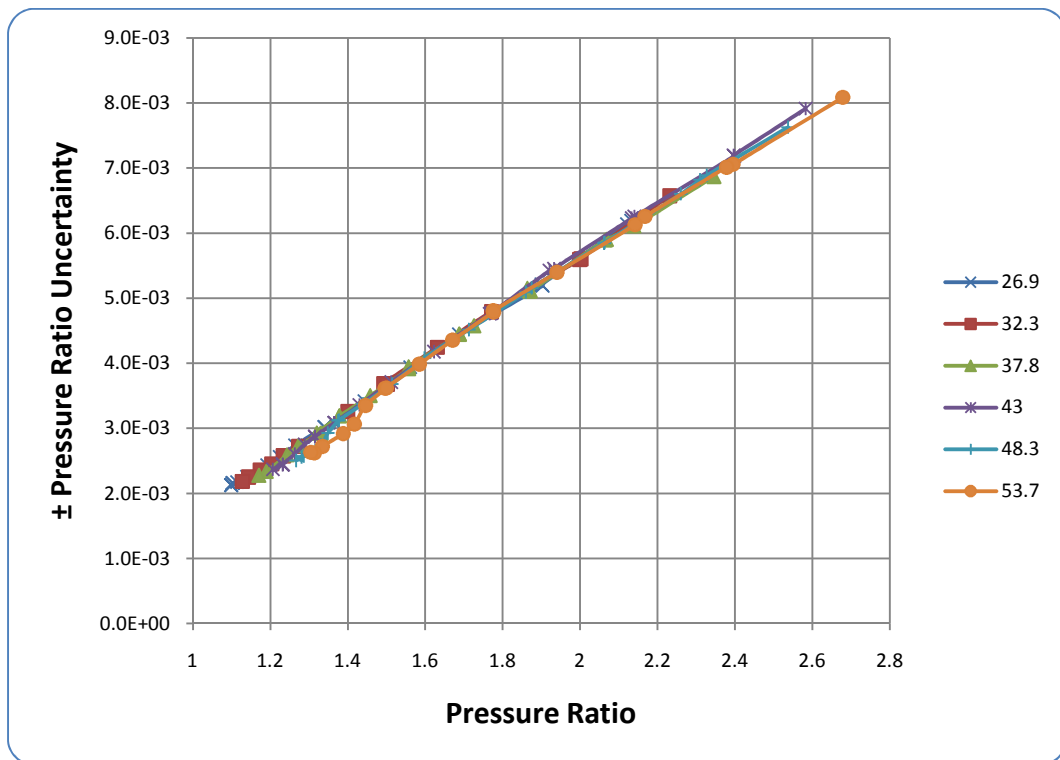


Figure A3: Uncertainty of the steady-state pressure ratio against pressure ratio

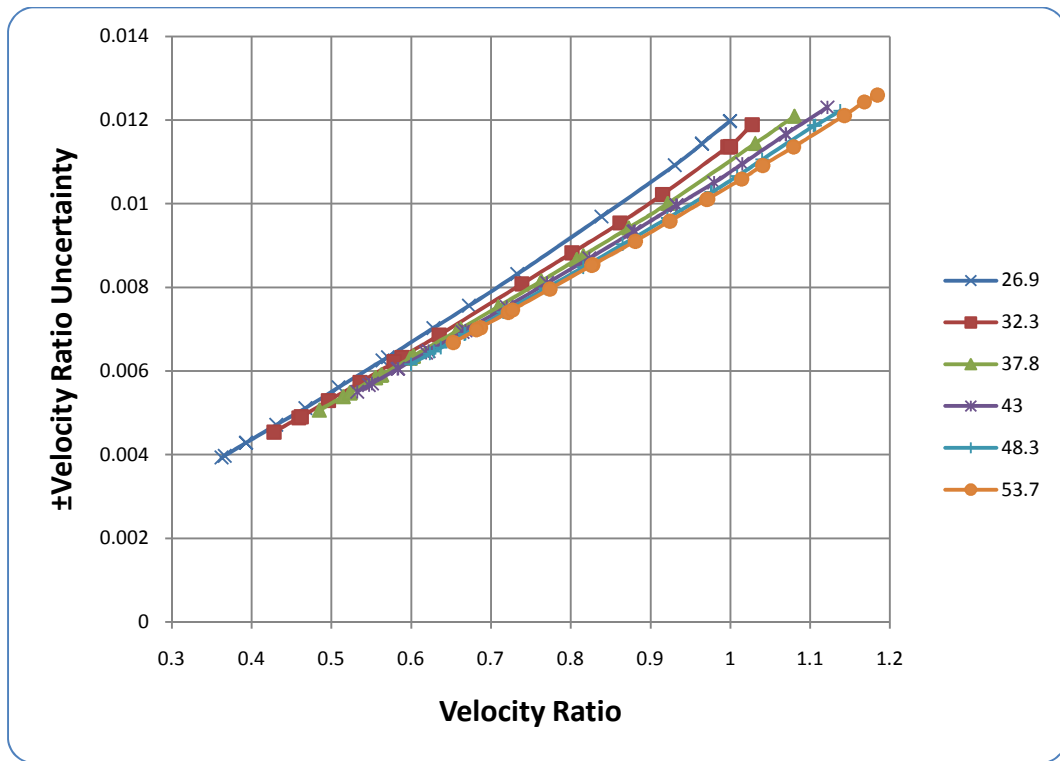


Figure A4: Uncertainty of the steady-state velocity ratio against velocity ratio

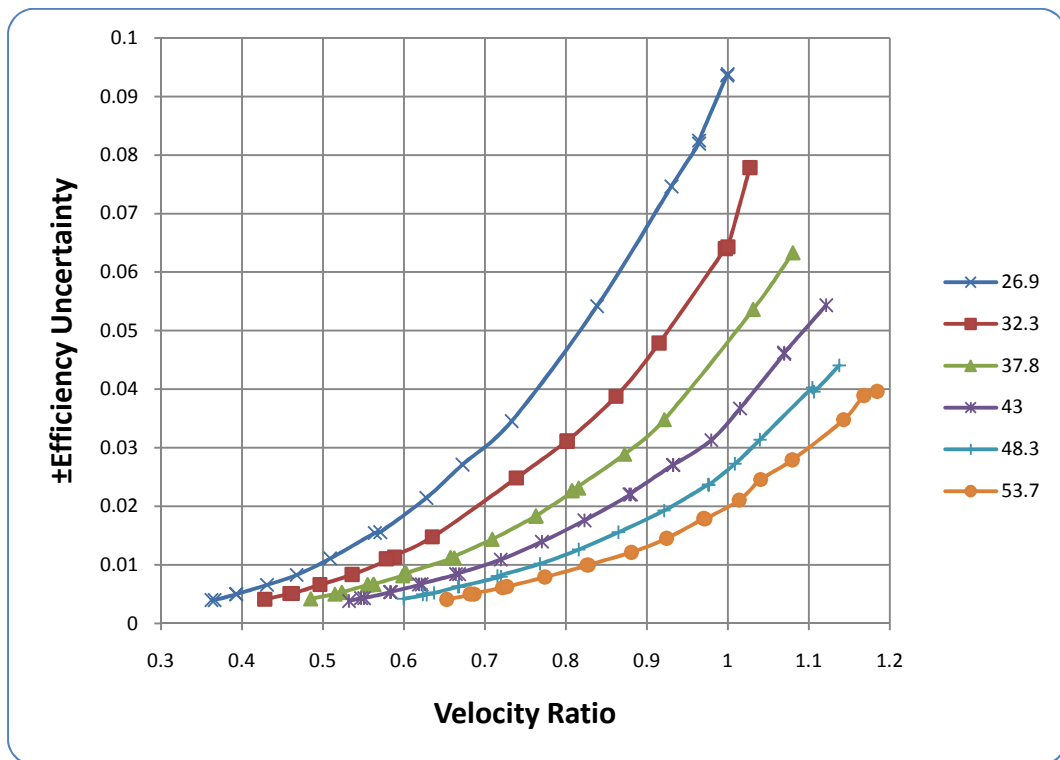


Figure A5: Uncertainty of the steady-state efficiency against velocity ratio

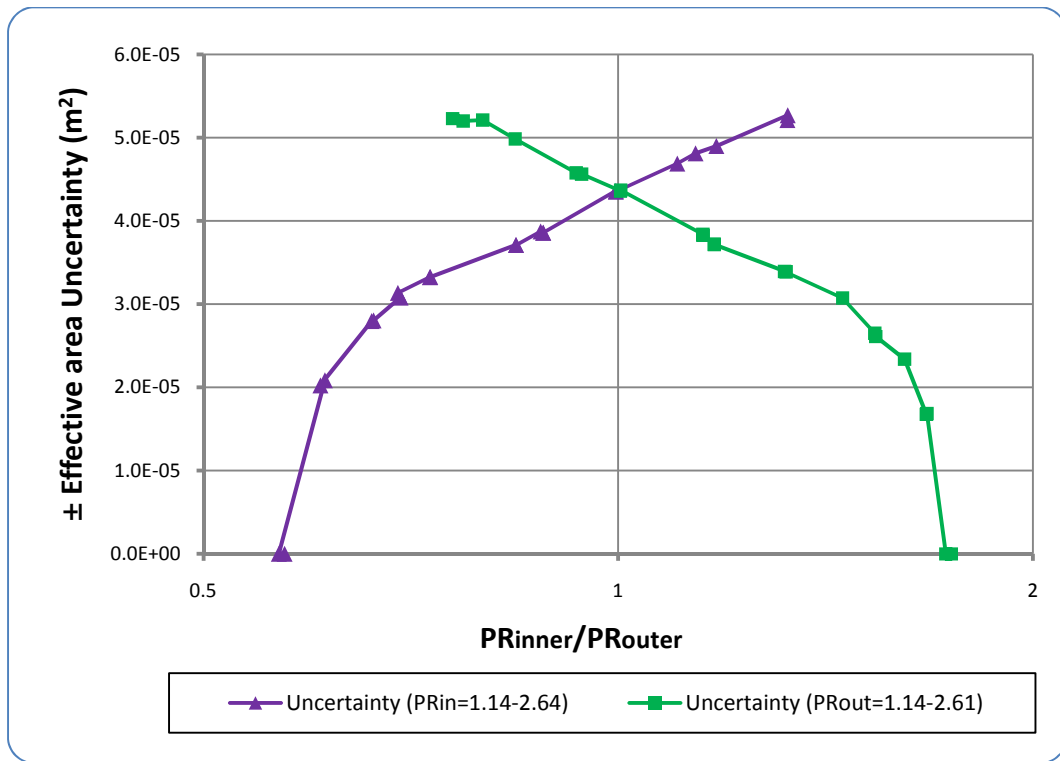


Figure A6: Uncertainty of the unequal admission effective area against the ratio of inlet pressures

Figure A7 to A10 show the individual measurements of mass flow, pressure, rotational speed and torque for 50% and 70% speeds. Both in-phase and out-of-phase data is supplied.

Figures A11 to A14 show the performance parameters for 50% and 70% speeds. Once again, both in-phase and out-of-phase data is supplied.

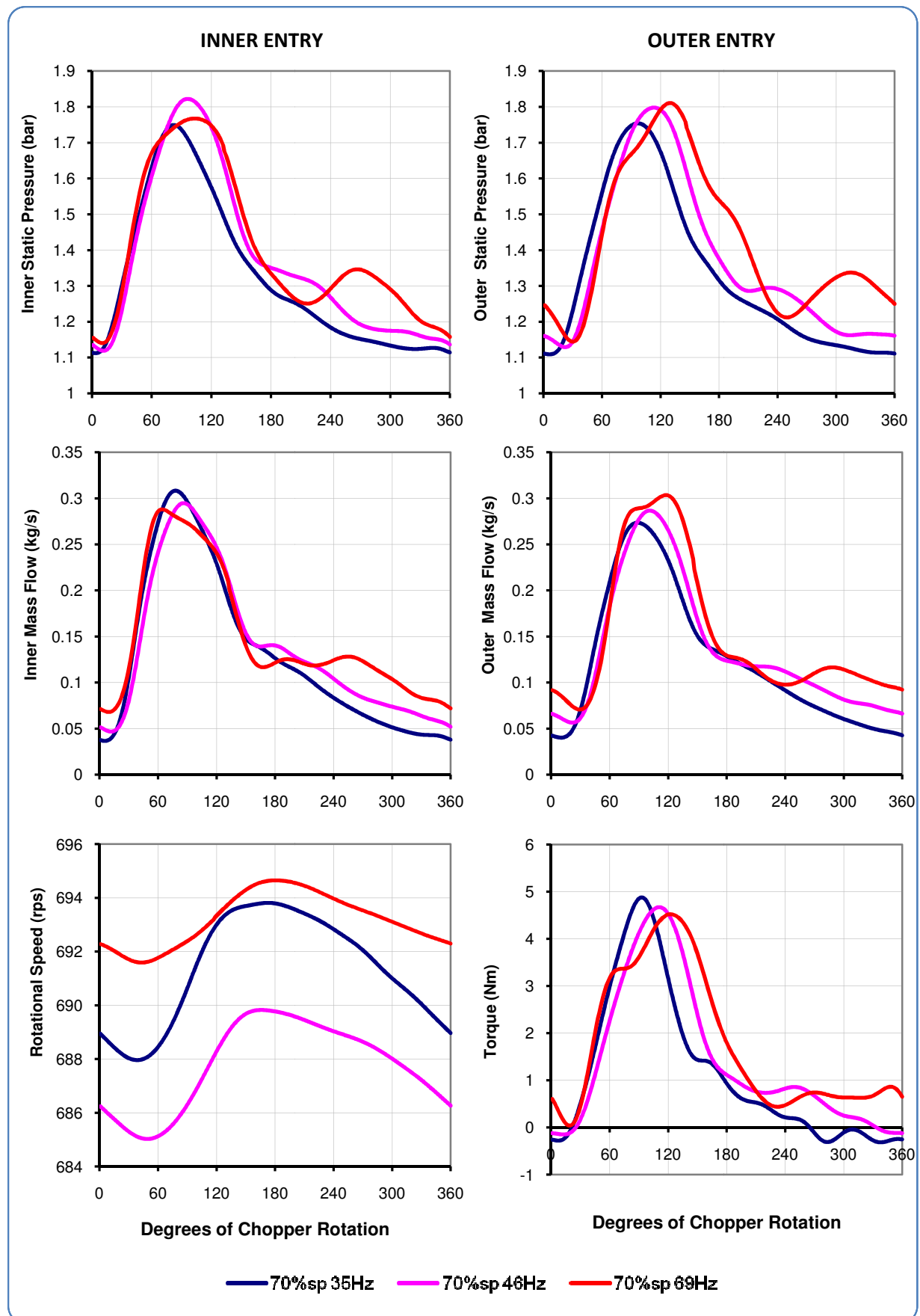


Figure A7: Pressure, Mass Flow, Turbine Speed and Torque (70% speed, in-phase)

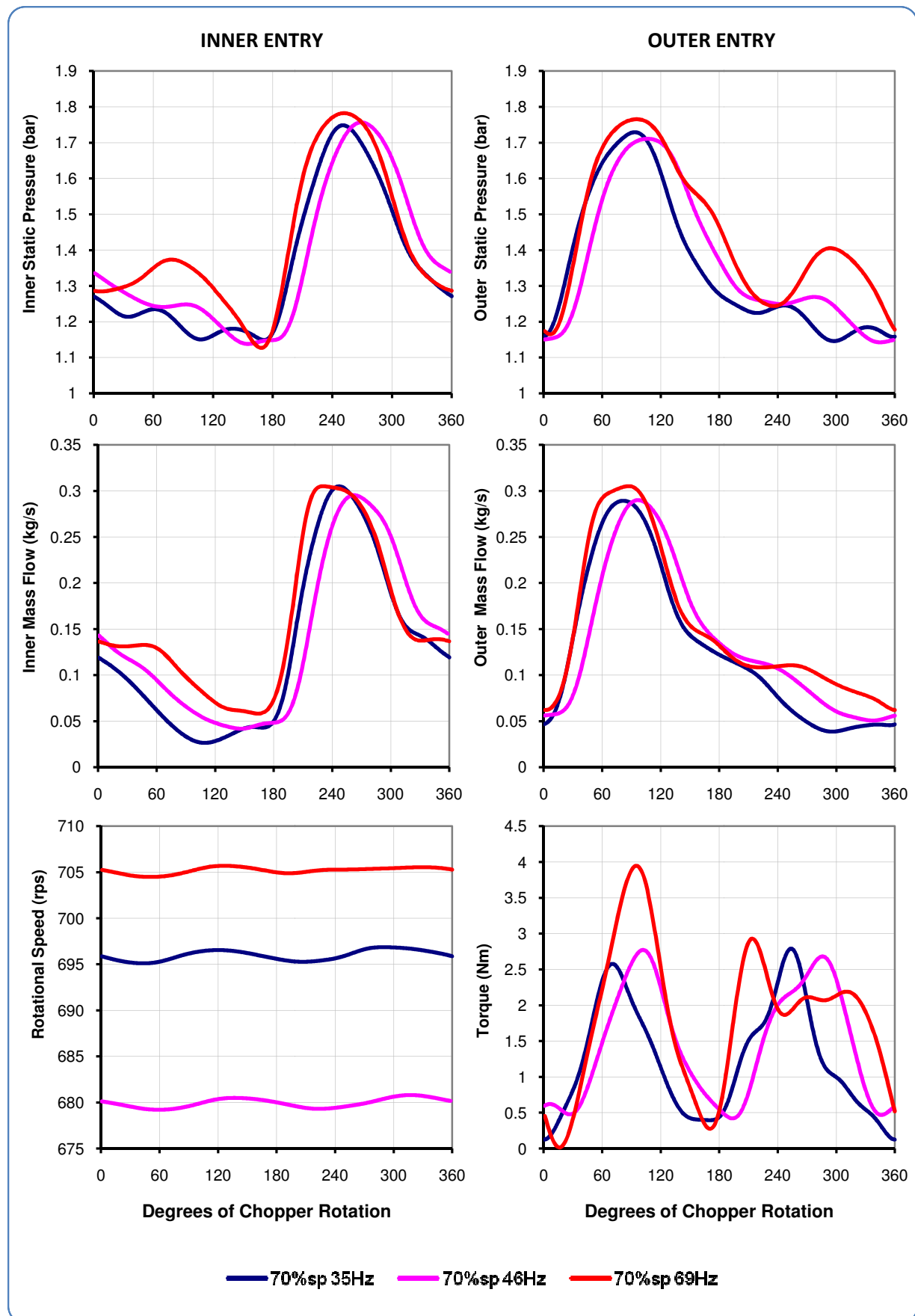


Figure A8: Pressure, Mass Flow, Turbine Speed and Torque (70% speed, out-of-phase)

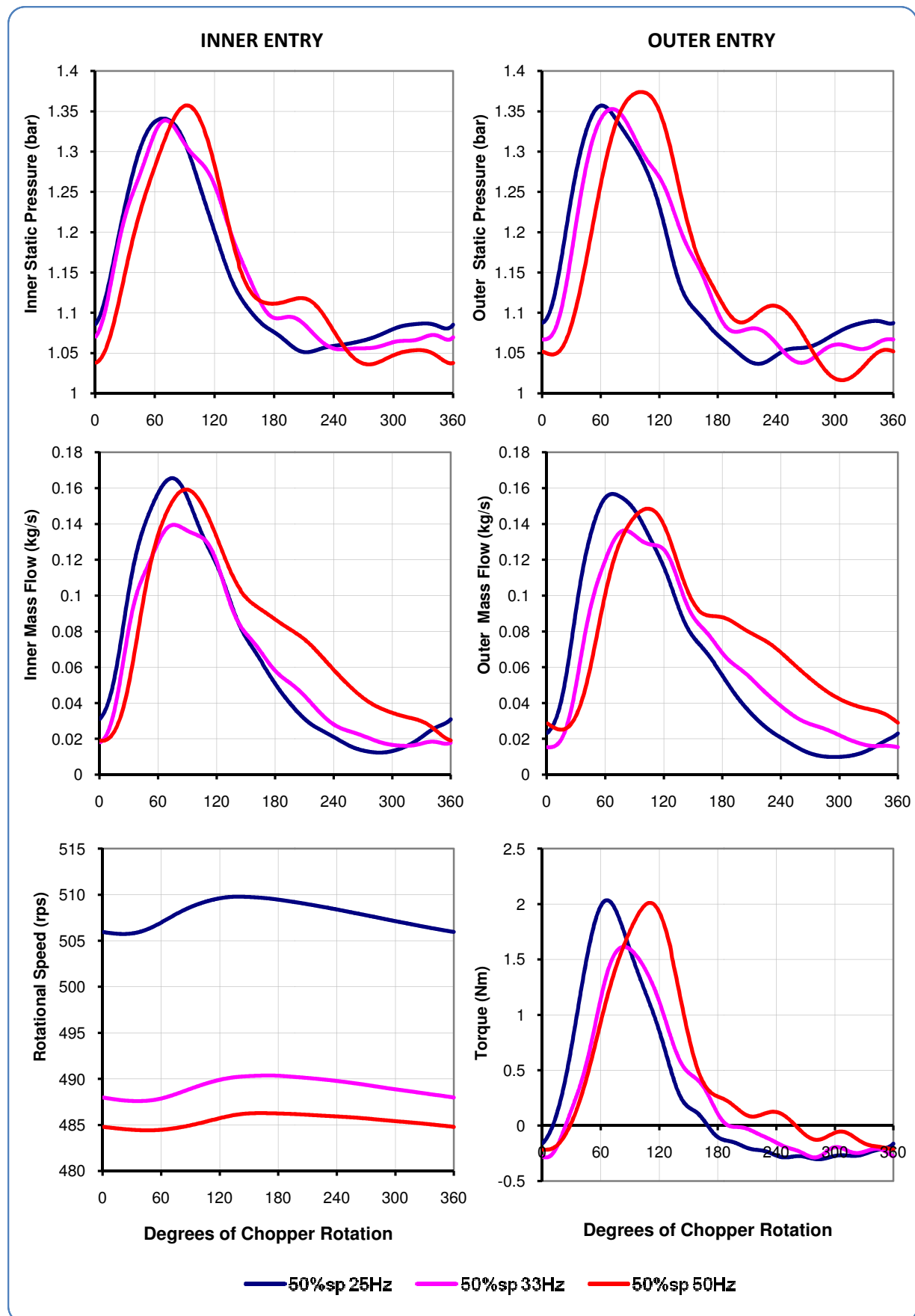


Figure A9: Pressure, Mass Flow, Turbine Speed and Torque (50% speed, in-phase)

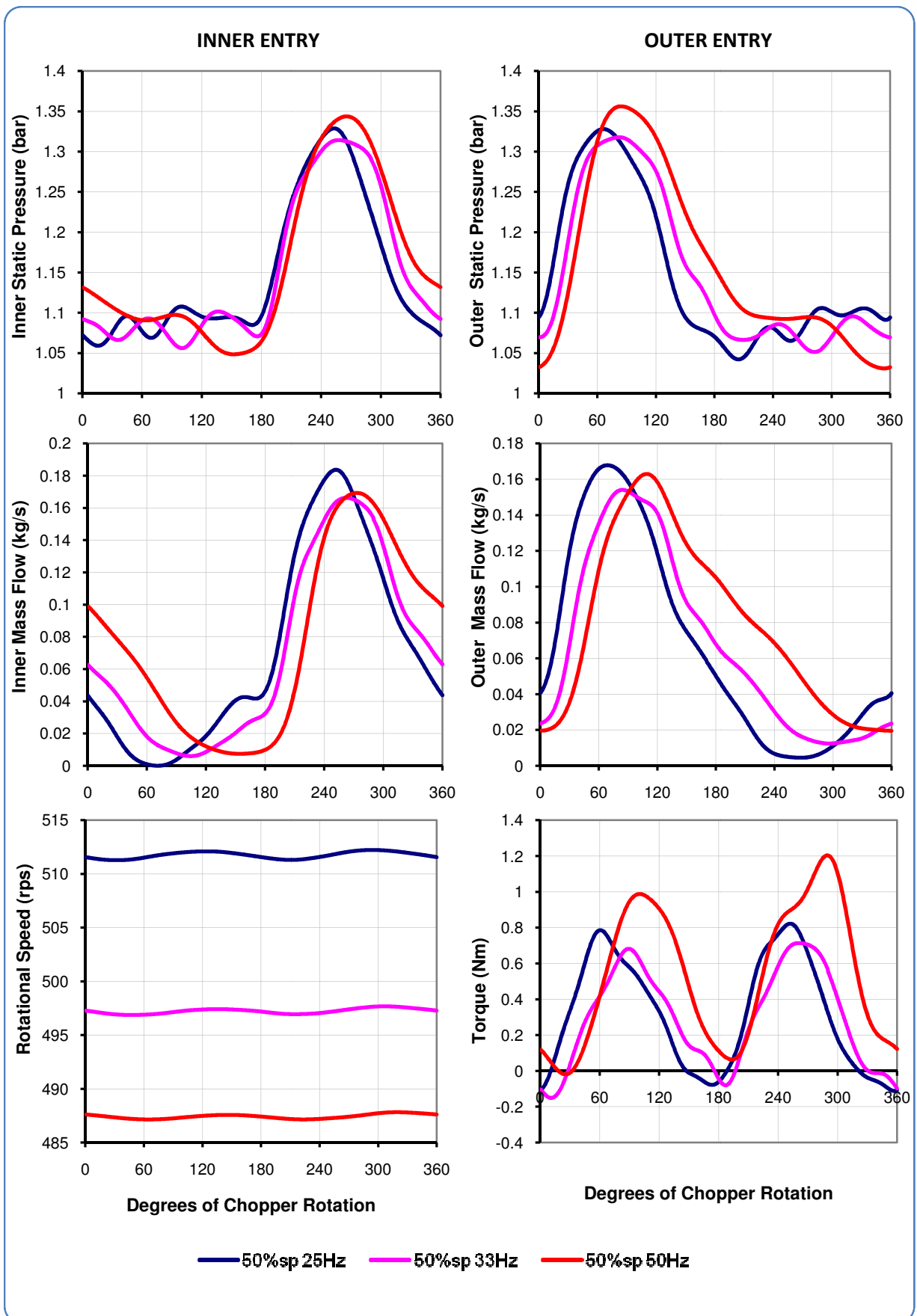


Figure A10: Pressure, Mass Flow, Turbine Speed and Torque (50% speed, out-of-phase)

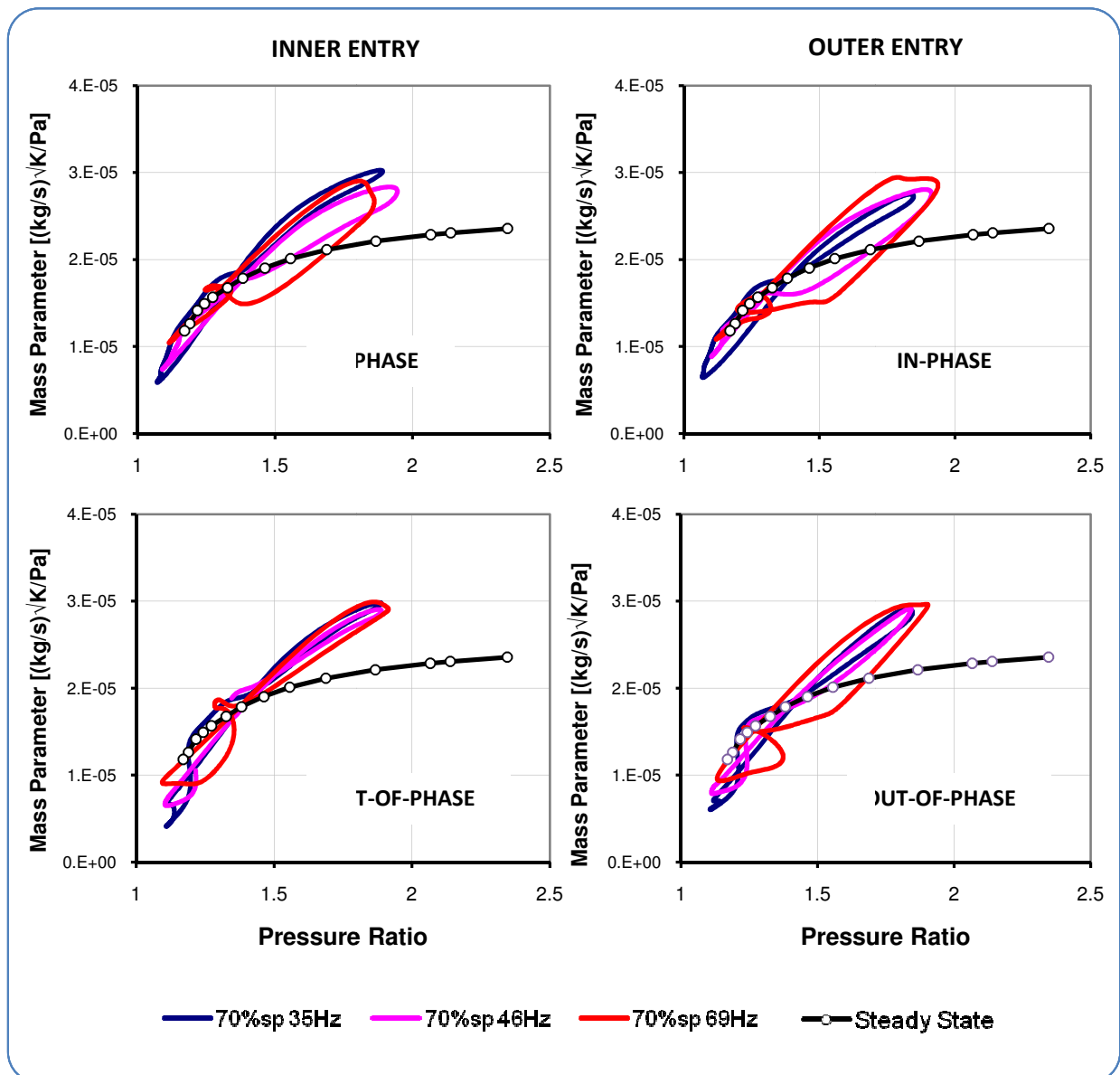


Figure A11: Mass Parameter versus pressure ratio across each inlet passage (70% speed)

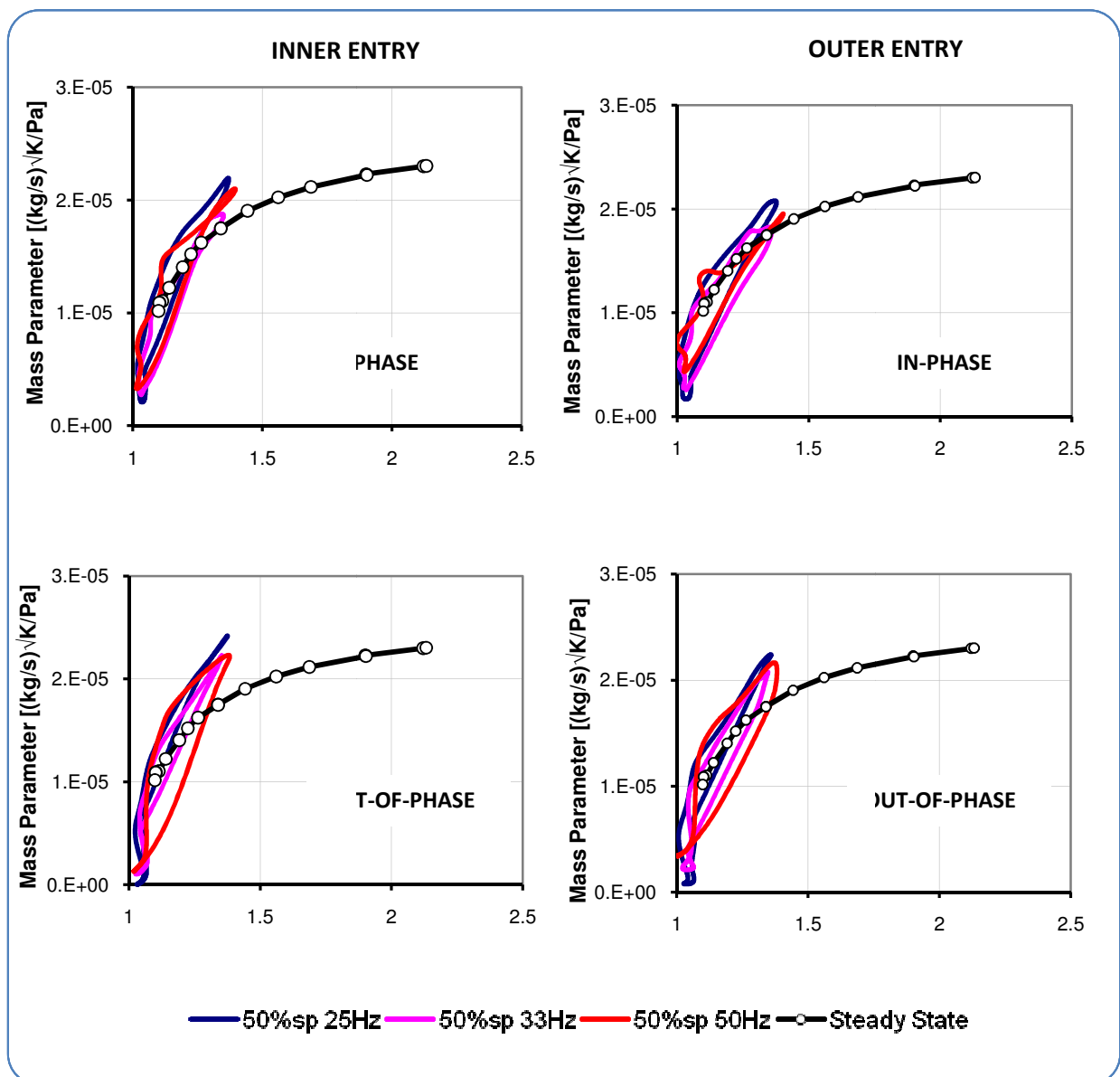


Figure A12: Mass Parameter versus pressure ratio across each inlet passage (50% speed)

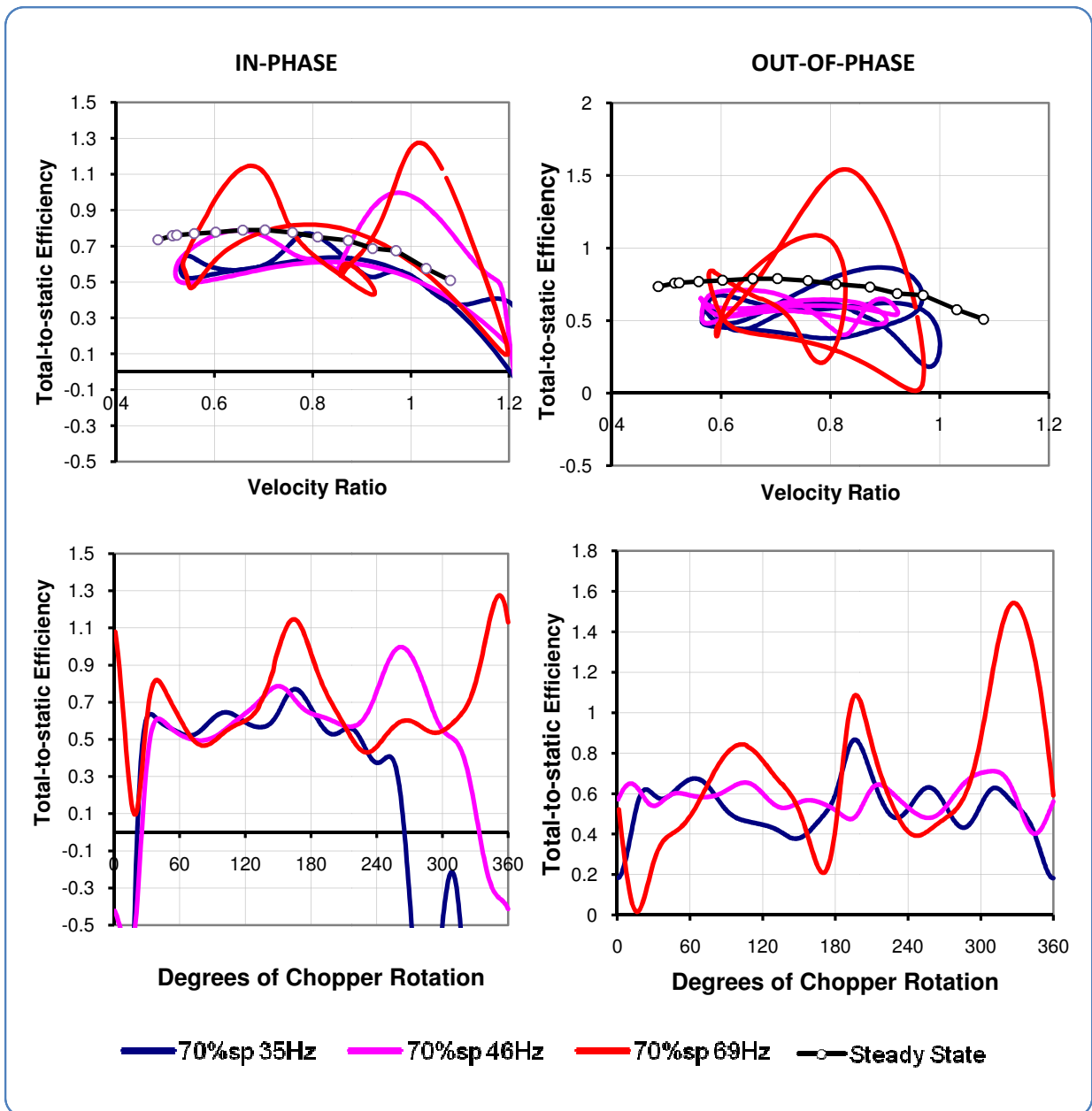


Figure A13: Instantaneous turbine efficiency against velocity ratio (top plots) and chopper rotation (lower plots) (70% speed)

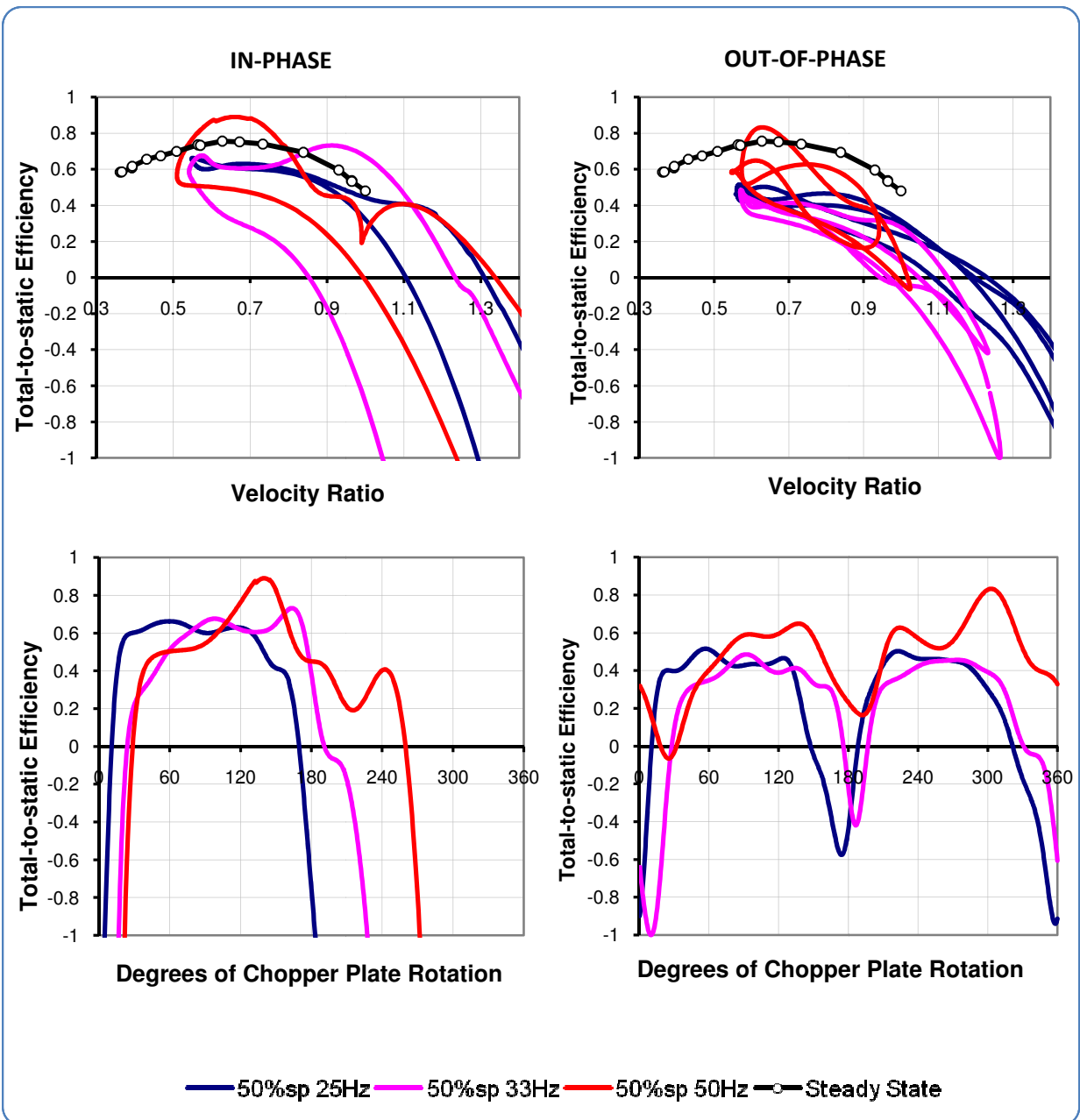


Figure A14: Instantaneous turbine efficiency against velocity ratio (top plots) and chopper rotation (lower plots) (50% speed)

Figure A15 shows the result of a full three dimensional interpolation of the isentropic turbine power evaluated from the full range of steady-state experimental results.

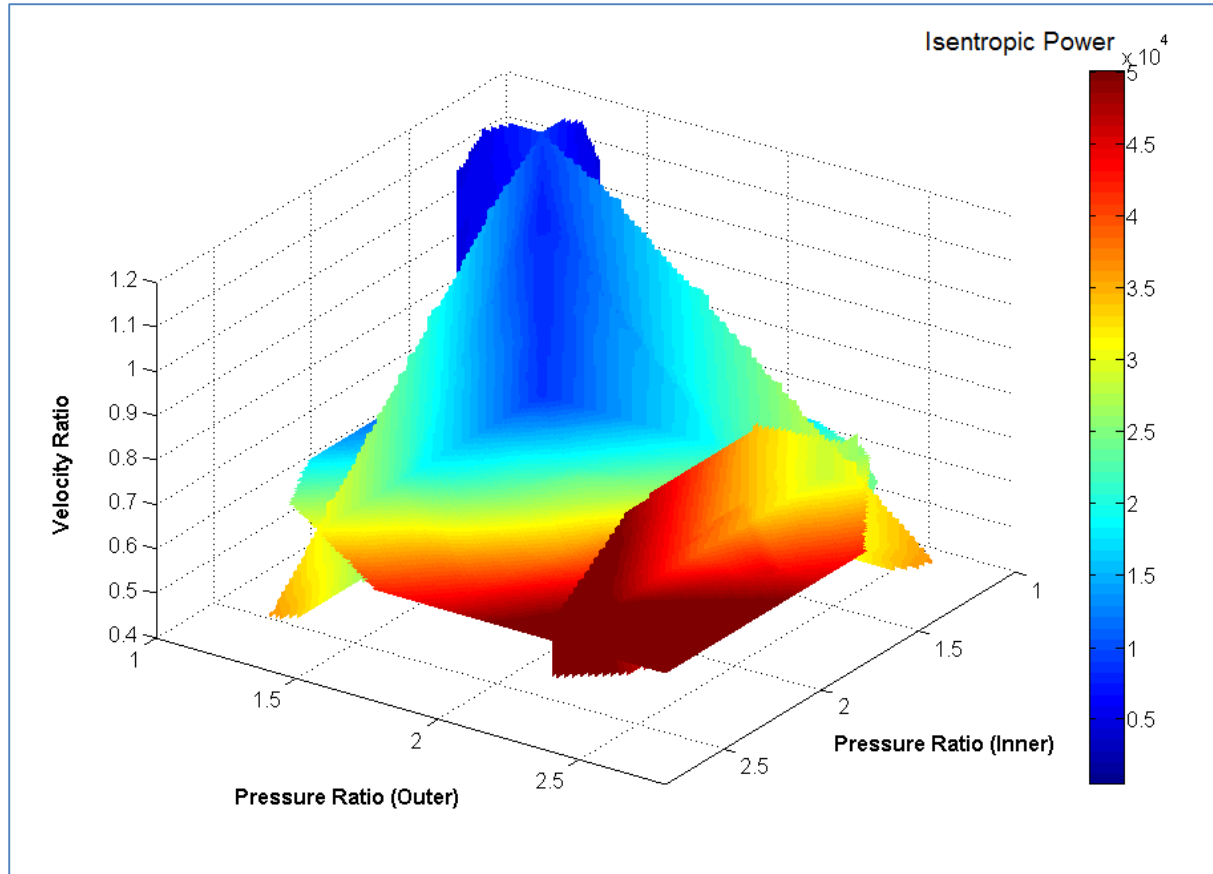


Figure A15: Interpolated steady-state isentropic power as a function of velocity ratio, PR_{inner} and PR_{outer}