

IMPROVED METHODS FOR MODELING DYNAMIC STAGE CHARACTERISTICS

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Thesis submitted to the Faculty of the
Virginia Polytechnic Institute and State University
In partial fulfillment of the requirements for the degree of

MASTER OF SCIENCE
in
Mechanical Engineering

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April, 1999
Blacksburg, Virginia

Key Words: Compressor, Dynamic, Modeling, Distortion, Wake, Response

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(ABSTRACT)

An analytical investigation of dynamic compressor characteristics was conducted with the goal to make fundamental improvements in the modeling of dynamic compressor stage characteristics. It was determined that present state-of-the-art in modeling dynamic compressor stage characteristics is the use of steady-state characteristics derived from flow model calculations, with first-order time lag response functions applied to account for dynamic departures from the steady and quasi-steady performance predictions.

This investigation developed a blade frequency response function (FRF) method for describing the dynamic blade response. Once the frequency response function of a blade row has been determined, any time or spatially dependent, non-uniform flow can be applied and the model will predict the dynamic blade response.

The first step of this research was to develop FRFs based on first-order lag equations and to test the method using these simple transfer functions. The next step was to develop FRFs based on a dynamic blade lift model for a simple, idealized compressor blade row model. It was found that chord length has a strong influence on the FRF, which is related to the fluid transport time through the blade passage. The final step was to incorporate experimental data obtained from a study of dynamic wake response of an isolated rotor. It was assumed that the wake response was well correlated with the dynamic lift response of a blade row.

It was found that aerodynamic loading, distortion strength, and span position all influence the frequency response functions, which differ greatly from simple first-order lag equations. It was determined that a number of FRFs are needed to describe the dynamic blade response accurately.

Acknowledgements

There are numerous people that have influenced this work that need to be recognized. I would like to thank the members of this committee for serving in this capacity. I would like to specially thank Dr. Walter F. O'Brien for his guidance and ingenious ideas. By giving me free reign in my investigation, I was able to discover and learn things I would not have if he controlled every step of the thesis. Nevertheless, he was willing to give advice when I needed it. I would like to thank Dr. Peter King for his advice and friendship during my studies. His advice was invaluable to me.

A special thanks goes to Dr. Milt Davis for obtaining funding for this project through Arnold Engineering Development Center. I would also like to thank Captain Michael Sexton (USN Ret.) for spurring my interest in turbomachinery and for introducing me to Dr. O'Brien.

I would like to thank Dr. William Cousins at AlliedSignal for his technical advice and friendship. Dr. Cousins was instrumental in getting me an internship at AlliedSignal before I started graduate school. This internship introduced me to the world of gas turbine engines, and led to future permanent employment with AlliedSignal.

I would like to thank Shaun Boller and Jeff Schwartz for their role in providing experimental data for this work and future numerical studies. I would like to thank Scott Gallimore, Mac Chiu, Dr. Alan Hale, and Maj. Keith Boyer (USAF) for their friendship.

I would especially like to thank my parents for their support during my studies. Because of the values they instilled in me when I was younger, I have the desire to learn and the drive to complete my goals.

As always, there is one person that stands out who has the greatest influence on this work, and that is my wife Cris. Because of her sacrifices, I was able to attend graduate school and complete this thesis. She put her life on hold for two years while I attended graduate school, and during this time she had to endure hours of my frustration as I struggled to finish. I also appreciate her willingness to move to Phoenix, which will move her away from the East Coast and her parents.

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NOMENCLATURE

aoa	angle of attack	Subscripts	
C	chord length	θ	circumferential
FF	Fourier series of forcing function	m	meridional
FRF	frequency response function		
h	specific enthalpy		
I	incidence angle		
L	lift		
M	magnitude per frequency		
$\dot{m}$	mass flow rate		
RF	Fourier series of response function		
U	blade velocity		
V	absolute flow velocity		
W	relative flow velocity		
w_c	compressor work		
α	absolute flow angle		
β	relative flow angle		
β'	blade metal angle		
ρ	density		
ξ	stagger angle		
Φ	phase		
θ'	camber angle		
η	efficiency		

1 Introduction

Current compressor design and performance prediction is based mainly on steady state theory and experiments. In an actual machine, compressor airfoils are subject to many non-uniform and dynamic flows, so the flow response in the passages, and therefore the performance of the airfoils and the machine is not steady. In recent years, there has been increasing interest in the dynamic behavior of compressors related to rapid maneuvers in military aircraft, related non-uniform flows, and active control investigations.

This thesis addresses a new method for representing the dynamic flow response of compressor stages obtained from analysis and experiments. The new method is particularly well suited to dynamic compression system models. As an illustration of the method, a first-order analytical model is applied to represent dynamic pressure response of a stage. First-order response models are widely used in dynamic modeling of compressor systems, but do not accurately predict the response.

Experimental data is desirable for the development of accurate dynamic frequency response functions. Such experimental data is very difficult to obtain, requiring high response transducers and complex experiments. Further, the dynamic response of a stage would be expected to vary with aerodynamic loading, requiring extensive experiments to obtain wide-range data.

A recent experiment obtained dynamic exit flow response data including wake and non-wake regions from an isolated compressor rotor. The results were wake response and not lift response data, but were utilized for this study with the assumption that the lift (pressure rise) response/wake time response functions are similar.

The results are summarized and evaluated to determine the potential of the method, and to assess the available data to provide dynamic response models for compressor stages and compression system models.

2 Literature Review

The ability to predict compressor response to steady and unsteady inlet distortions is a major concern for engine manufactures and their customers. The number of documents in the literature that deal with this topic reflect the relative importance of dynamic compressor response prediction.

A significant amount of work has been done in the area of creating analytical models. To date, these models have been based on empirical data, steady-state characteristics, and first order response functions to account for the dynamic departure. These models must be calibrated to each specific compressor and engine to accurately duplicate experimental results. Work has been done in the area of creating models based on first principles and compressor geometry. This will allow dynamic response predictions while the compressor is in the design stage of development.

One of the most widely used concepts in distortion modeling is the parallel compressor theory. This is a simple model that can be used to predict pressure losses due to circumferential distortions. This theory was developed by Pearson and McKenize (1959). It models the distorted compressor as two separate undistorted compressors operating in parallel. One compressor has inlet conditions corresponding to the undistorted region of the original compressor and the other compressor has inlet conditions similar to the distorted region of the original compressor. Both compressors are assumed to discharge to the same static pressure. By using this method, the response of the original compressor to inlet distortions is assumed to be instantaneous, or quasi-steady, since both the parallel compressors have uniform flow at the inlet. Figure 2.1 shows a visual representation of parallel compressor theory.

PARALLEL COMPRESSOR DISTORTION ANALYSIS

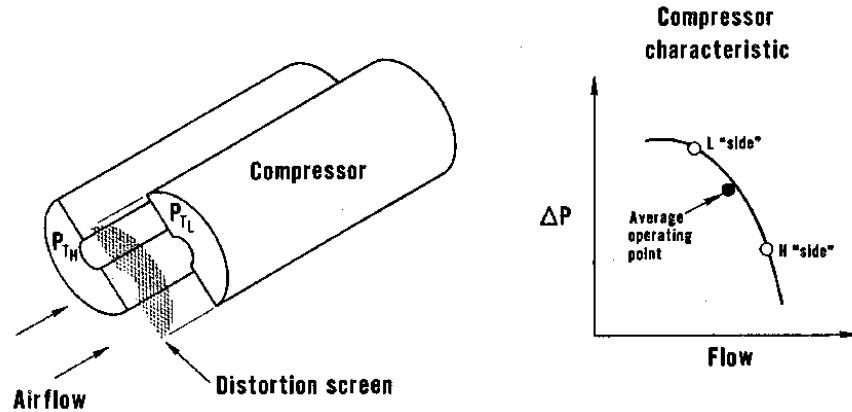


Figure 2.1

Reid (1969) showed that the instantaneous compressor response modeled by parallel compressor code is highly inaccurate for small extent distortions. He developed the idea of a critical angle of distortion extent, which is the smallest circumferential distortion angle at which the quasi-steady assumption is accurate. If the extent of the distortion screen is less than the critical angle the compressor row does not respond to the distortion in a quasi-steady manner. Reid incorporated the critical angle idea into parallel compressor code, which improved its accuracy for small extent distortions. This was the first step in improving dynamic response models.

Roberts, Plourde, and Smakula (1968) took the next step when they investigated the effects of chord length of both the rotor and stator on a low speed, multistage, axial flow compressor. They correctly concluded that chord length was highly correlated to how quickly the blade responds to a circumferential distortion, in this case a 180° distortion screen. They compared experimental data to parallel compressor code results and found that as the blade length was increased the model's inaccuracy increased. They reasoned that the flow along a long chord length takes more time to adjust to the

distortion. They also defined a parameter called reduced frequency. Reduced frequency was defined as a measure of the chord length to the wavelength of the unsteady disturbance, as follows:

$$k = \frac{B \cdot \Omega}{V} \quad \text{Equation 2.1}$$

where

k = reduced frequency

V = average velocity of the air relative to the rotor

B = one half of the rotor chord length

Ω = Frequency of the circumferential disturbance with a window of periodicity of one revolution (rad/sec)

Mikolajczak and Pfeffer (1974) expanded the reduced frequency parameter. This representation showed that the reduced frequency is the ratio of the time a particle spends in the blade passage to the time it takes for the blade to rotate through the distorted region.

$$k = \frac{b}{r} \cdot \frac{360}{\theta} \cdot \frac{U}{V_m} \quad \text{Equation 2.2}$$

where

b = axial projection of the rotor chord

r = compressor radius at the blade section

θ = distortion extent in degrees

U = tangential blade speed

V_m =axial air velocity

Cousins (1997) also discussed reduced frequency and the time an axial or centrifugal compressor takes to respond to an inlet distortion. Cousins examined transonic axial and centrifugal compressors. He noted that much of the diffusion in a transonic axial blade occurs after the first 35% of the meridional distance of the blade is reached. He noted that there is a time response associated with how long it takes a blade to respond to a change in the fluid particle flow. This time response can be calculated by

dividing the distance traveled to the throat of the passage by the relative velocity. If the fluid particle transport time is less than the time it takes for the blade to pass through the distortion, the blade response will be at a maximum. If it is greater, the blade will not fully respond. The response time of a centrifugal rotor is different because of the way the blades create a pressure rise. Cousins stated that once the fluid particle reaches the throat of an axial blade there is a chance the flow could be disrupted causing separation, and the blade could respond more quickly then if the flow did not separate. While in a centrifugal compressor the fluid particle must travel past the throat and further into the rotor before the disruption could affect the radial flow component. Centrifugal compressors have a higher reduced frequency parameter because of longer meridional distances and hence a higher tolerance to inlet distortions.

Mazzawy (1977) developed a non-linear, compressible flow model that uses the parallel compressor method without assuming a constant exit pressure. The model considers individual blade row performance to account for stage matching effects. Therefore, the model requires the determination of blade row characteristics as a function of upstream conditions. Since the dynamic pressure loss and fluid path exit angles differ from the quasi-steady values, Mazzawy incorporates a first order response function to lag the time response. His time constant was based on empirical data that was descriptive of boundary layer response.

Tesch and Steenken (1981) developed a Dynamic Digital Blade Row Compressor System Stability Model for the General Electric Company. The model was a 1-D, pitch line, row by row model using equations of conservation in a volume-averaging form. They used a time marching scheme to model the dynamic response of the compressor row. They also developed an effective incidence angle that was developed by assuming a first order time response. They defined the effective incidence angle as “the incidence angle which is equivalent to the dynamic response of the blade as it passes through the distortion.” The purpose of this effective incidence angle is to characterize the response lag of a blade to an inlet distortion. They used this angle to calculate the dynamic blade characteristic through time instead of the instantaneous incidence angle. Basically, this effective incidence angle is modeling the fluid particle transport time through the blade passage.

Adamczyk (1974) continued to further unsteady dynamic compressor rotor response by developing a new compressor code. His model predicts circumferential distortions with a method similar to parallel compressor theory, but is more advanced because it does not assume constant exit static pressure between regions. This allows for the exit static pressure fluctuations that occur in real machines. It is also more advanced than linear perturbation theory-based models because it is not restricted to small perturbations. Linear perturbation cannot account for degradation in the stability-operating limit because it is restricted to small amplitude distortions. His model divides the flow into two regions: flow in the blade passages, and flow upstream and downstream of the rotor. The flow in the blade passages is based on the time-dependent mechanical energy equation and includes the effects of cascade loss and turning. The flow upstream and downstream is modeled as non-linear, 2-D, time dependent flow fields. Even though the flow field calculations and base assumptions are more advanced, Adamczyk still uses first order lag functions to calculate the blade row dynamic response.

Melick and Simpkin (1972) developed a global dynamic compressor model that related inlet total pressure and temperature distortions to losses in stall margin. They derived an effective angle of attack which was based on a first-order response function. The time constant they used for this equation was proportional to blade chord length. They also arrived at the conclusion that stall and recovery responses are different phenomena and have different response functions. They were able to predict loss in stall margin as a function of the distortion intensity and the reduced frequency parameter. Melick (1973) later improved the accuracy of the model by using a second order response function in deriving the effective angle of attack. He determined the two time constants from experimental test data.

Work has also been done on developing models that will predict the rotating stall characteristic of a compressor. These works have been reviewed by Mokolke (1974). Early models used linearized equations of motion and were therefore only accurate for small perturbations. The compressor stages were modeled as semi-actuator disks where work and rotation are added instantaneously. Nagano and Takata (1970) expanded on these models by using non-linear equations of motion. They also used first order response functions to model the unsteady characteristic.

Sexton (1980) furthered the work of Nagano and Takata in his model. He modeled the blade response using frequency response functions derived from experimental data. He used instrumented blades to measure the dynamic pressure forces. He then derived the frequency response function between the quasi-steady and the dynamic total pressure loss distributions. The quasi-steady total pressure loss was the forcing function while the dynamic total pressure loss was the response function. This use of a frequency response function has higher accuracy than a simple first order response function. This response function was only developed for rotating stall, however, and not the pre-stall region of operation. In addition, this work only dealt with one blade design and chord length.

Cousins and O'Brien (1985) developed a post-stall model using the experimentally determined dynamic loss response function described by Sexton and O'Brien (1980). This new model interactively coupled the upstream and downstream fields with a semi-actuator disk, which represents the compressor rotor. The dynamic loss response is calculated by determining the quasi-steady variations in total pressure loss and multiplying it by a transfer function. This model is capable of defining a compressor characteristic map with pre-stall and post-stall operating lines. This model uses total pressure losses, and the transfer function was developed using on blade experimental measurements.

Conclusions from Literature Review

There is a need to improve the method for modeling dynamic compressor characteristics. Current methods are based on first-order lag equations, which because of their linear nature do not capture the non-linear response of compressor blades. There has been some work in the area of improving the method of dynamic modeling. This work presents the new method in Chapter 4, while Chapter 5 shows results calculated using the new method based on the responses of a simple computer model. Chapter 6 shows results based on experimental data, and discusses which parameters have an effect on the blade response. Chapter 7 presents a summary and conclusions, and Chapter 8 provides recommendations for future work.

3 Steady-State Compressor Analysis

The object of this chapter is to describe steady-state compressor analysis and to define terminology, notation, and sign convention used throughout this thesis. This is the convention used in two-dimensional linear cascade analysis as described by Yocum (1988).

3.1 Two-dimensional Compressor Terminology

Two-dimensional axial compressor analysis involves the use of relative and absolute frames of reference. Figure 3.1 illustrates the absolute and relative flow velocity triangles that are used in steady-state analysis.

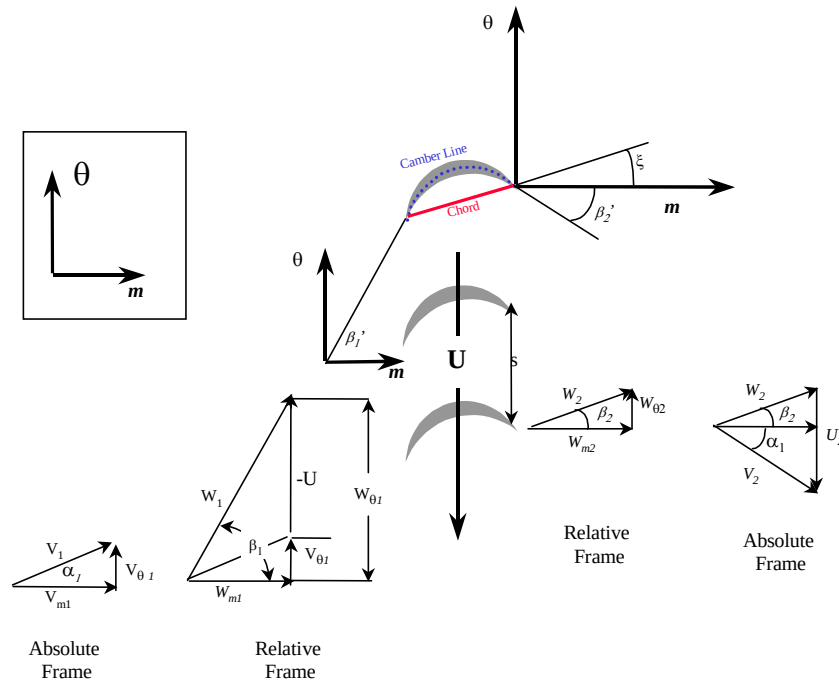


Figure 3.1 Axial-flow Compressor Cascade Geometry and Velocity Triangles

Since the axial blades are moving with a circumferential velocity U in the $-\theta$ direction, it is desired to view the fluid flow from the compressor blade perspective to simplify calculations of compressor blade row performance. As shown in Figure 3.1, the absolute velocities and their corresponding flow angles are expressed as V_i and α_i , respectively, and the relative velocities and their corresponding flow angles are expressed as W_i and β_i , respectively. To transfer from the absolute frame of reference to the relative frame of reference the following vector addition is applied to the velocities.

$$V = U + W \quad \text{Equation 3.1}$$

Additional important terms are the blade metal angles, β_1' and β_2' . These angles are calculated from the meridional axis (m) and the line tangent to the camber line at the leading and trailing edges of the blade. The camber line is the line at the midpoint between the suction and pressure sides along the length of the blade. The stagger angle, ξ , is defined as the angle between the meridional axis and the chord line. From these angles additional important angles can be derived.

$$aoa = \beta_1 - \xi \quad \text{Equation 3.2}$$

$$i = \beta_1 - \beta_1' \quad \text{Equation 3.3}$$

$$\theta' = \beta_2' - \beta_1' \quad \text{Equation 3.4}$$

Equation 3.2 is the equation for angle of attack, which is defined as the angle between the relative velocity W_1 and the chord of the airfoil. An increase in the angle of attack (or an increase in aerodynamic loading) results in more work being done on the fluid. Equation 3.3 is the definition of incidence angle, which is directly related to the angle of attack. Equation 3.4 is the definition of the camber angle, which is associated with the ideal work the blade would transfer to the fluid (Boller).

3.2 Steady-state Compressor Performance Analysis

Pressure rise in compressors is accomplished by imparting work to the fluid flow. In an axial compressor this is accomplished by turning the flow in the tangential direction (θ) in the blade passages. This flow turning decreases the relative velocity in the blade passage causing a pressure rise, which is illustrated in Figure 3.1. The rotor also imparts angular momentum to the fluid causing the absolute velocity in the tangential direction to increase. For a control volume at radius r , the torque, or angular momentum, acting on the fluid in the blade passage can be written as

$$\tau = \dot{m}((rV_{\theta})_2 - (rV_{\theta})_1) \quad \text{Equation 3.1}$$

The work done on the fluid by the rotor can be expressed as

$$w_c = U(V_{\theta 2} - V_{\theta 1}) \quad \text{Equation 3.2}$$

By introducing the first law of thermodynamics, this equation can also be written in terms of the stagnation enthalpy.

$$h_{02} - h_{01} = U(V_{\theta 2} - V_{\theta 1}) \quad \text{Equation 3.3}$$

The entire process for a compressor stage is shown on the enthalpy-entropy diagram in Figure 3.2 (Cousins, 1997).

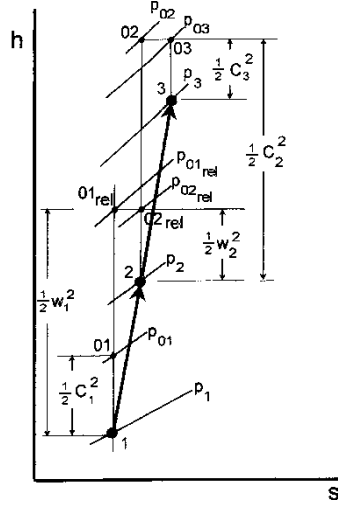


Figure 3.2: Enthalpy-entropy diagram for an axial compressor stage

The total-to-total stage efficiency is defined by

$$\eta_{st} = \frac{h_{03s} - h_{01}}{h_{03} - h_{01}} \quad \text{Equation 3.4}$$

By using this equation and isentropic pressure-temperature relationship the stage pressure ratio may be found by

$$\frac{p_{03}}{p_{01}} = \left[1 + \eta_{st} \frac{\Delta T_0}{T_{01}} \right]^{\gamma/(\gamma-1)} \quad \text{Equation 3.5}$$

As can be seen from the above equations, no time dependent terms are included, and the equations apply only for steady-state analysis. These equations assume that any changes in the inlet properties are carried through the blade instantaneously, and the blade flow response immediately results in the new steady state properties. These are simplified equations, but are very useful for preliminary design.

By looking at the blade loading as a function of distance along the chord, one can see how the blade creates the pressure rise by diffusing the fluid flow. A finite amount of time is required for the fluid particle to traverse the length of the compressor blade. This time is called the fluid transport time, as described by Cousins (1997). Because of this time, the blade will not instantly respond to a change in flow properties upstream of the blade. Figure 3.3 shows the blade loading of a current technology blade.

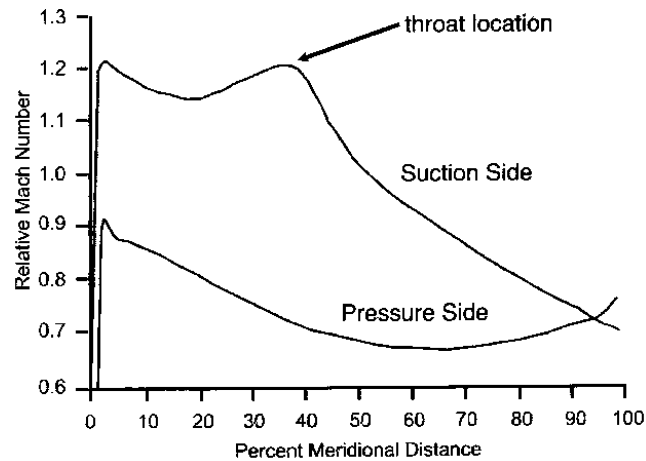


Figure 3.3: Typical Axial Rotor Loading Distribution

A better understanding of how the compressor rotor blade creates a pressure rise allows for the basis of understanding of how a blade will respond dynamically to non-uniform or unsteady flow at the inlet.

4 First-Order Lag Equation and Frequency Response Method

4.1 First Order Lag Function

Currently dynamic compressor response models, such as DYNTECC, which was developed at Arnold Engineering Development Center, use first-order lag equations to model the dynamic departure from the steady-state performance. Hale and Davis (1992) describe the capabilities of the code in detail. The time constants used in these models are based on empirical data or model calibration. Although these model use highly sophisticated equations for calculating steady-state performance and fluid flow field parameters, the simple first-order lag equations limit the model's accuracy. These models apply the first-order lag equations to parameters such as effective incidence angles, dynamic total pressure loss, and lift coefficient.

The first-order differential equation that is used in these models is

$$y' + By = Q(x) \quad \text{Equation 4.1}$$

When the Equation 4.1 is solved, the solution can vary depending on the form of $Q(x)$.

$$Q(x) = 0 \quad y = Ce^{-Bx} \quad \text{Equation 4.2}$$

$$Q(x) = A \quad y = Ce^{-Bx} + A/B \quad \text{Equation 4.3}$$

$$Q(x) = Ax \quad y = Ce^{-Bx} + A/B(x - 1/B) \quad \text{Equation 4.4}$$

The commonly used first-order equation is equation 4.3, where $Q(x)$ is the steady-state solution. This first-order equation can be used with any given flow parameter. An example is given by equation 4.5, which is the equation used in DYNTECC to model the airfoil lift.

$$\text{Equation 4.5}$$

$$\frac{dC_l}{dt} + \frac{C_l}{\tau} = \frac{C_{lss}}{\tau}$$

where C_l is the lift coefficient at any given time, C_{lss} is the steady-state lift coefficient, and τ is the time constant that is derived from experimental data and then the model is calibrated to match experimental results.

The lift coefficient seems to be a very good parameter to base dynamic blade response on since it is directly related to the pressure rise of the fluid flow over a blade. The first order lag function can also be used to find the dynamic loss response, as shown by equation 4.6.

$$\tau \frac{\partial \omega}{\partial t} = \omega_{steady} - \omega \quad \text{Equation 4. 6}$$

where ω = fluid loss in the blade row

This equation is the same first-order function.

The first steps in this thesis were to understand how the first-order equation lagged the blade response and to generate results for later analysis with the new method. Using equation 4.7 with steady state or instantaneous blade lift as the input (forcing function) the following results were calculated and are shown in Figure 4.1.

$$L = L_{\infty} + [L_A - L_{\infty}] \cdot e^{-t/\tau} \quad \text{Equation 4. 7}$$

Where L equals the lift at any time, L_{∞} equals the limiting lift or final steady state lift, and L_A equals the initial lift.

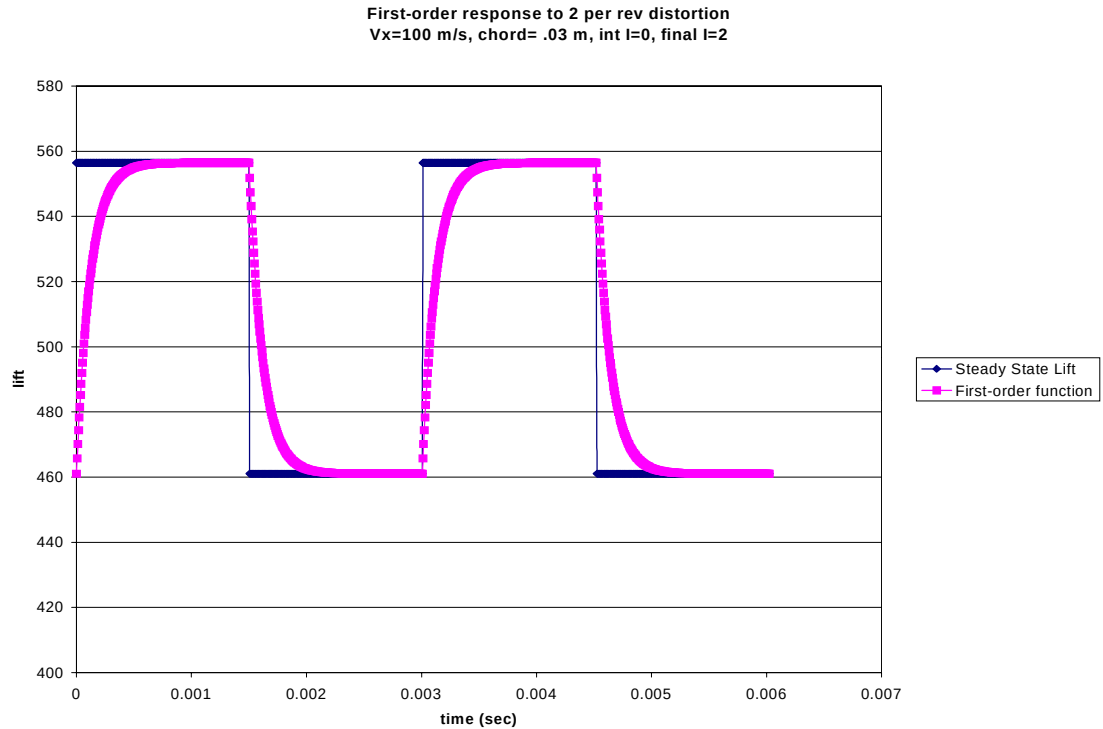


Figure 4.1

Figure 4.1 shows that the dynamic blade response is lagged with respect to the instantaneous solution, which is what will physically happen but not necessarily in accordance with the first-order lag response. As the blade passes into a distortion, certain properties of the new flow field such as pressure information will be communicated downstream at the speed of sound, while other information will propagate downstream at the speed of convection. Mass flow and fluid velocity will move at the speed of convection, the fluid particle transport time. This effect was described by Cousins (1997) as the time it takes a blade to respond to the inlet flow change that has occurred. This time can be calculated by dividing the distance from the blade tip to the throat of the blade passage (the start of diffusion for a supersonic blade) by the relative velocity. For a subsonic blade, the fluid particle transport time will be based on the distance from the tip of the blade to the trailing edge. The fluid transport time can be useful for an estimate of how the blade will respond. If the fluid particle transport time is less than the time a blade spends in a distorted region, the blade will have a maximum response to the distortion. However, if the particle transport time is greater than the time spent in the distorted region the blade response will be reduced. Figure 4.2 shows a visual representation of the fluid particle transport time.

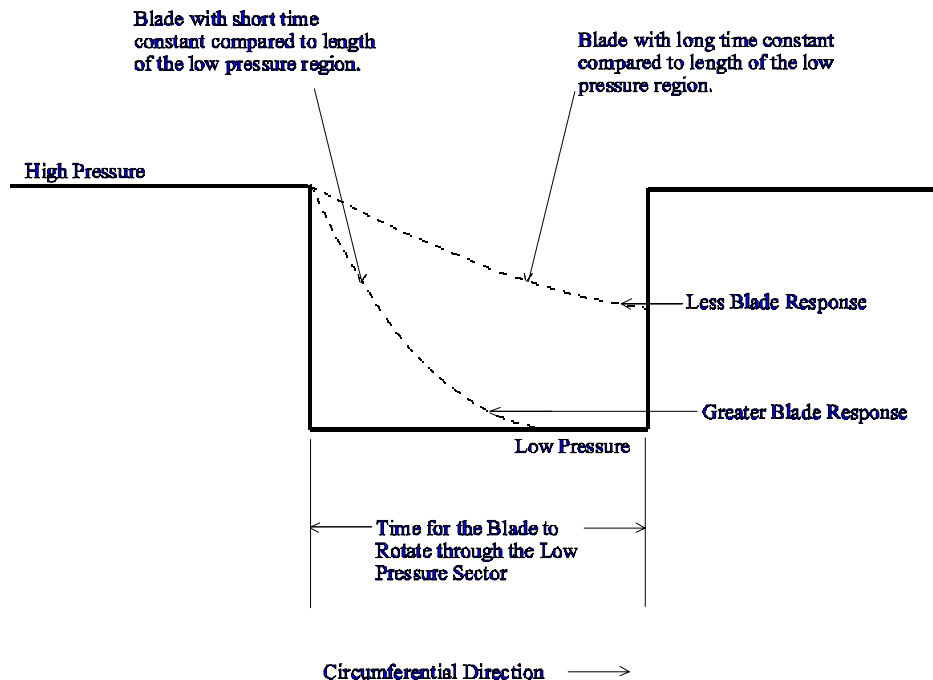


Figure 4.2: Blade Response Based on Fluid Particle Transport Time

Even though the fluid particle transport time can give us a good idea of the time it takes for the blade to fully see the distortion, it does not give the most accurate prediction since it does not include the effect of information propagating at the speed of sound, nor other higher order effects.

Because flow field information is moving at different speeds downstream, and since these properties have different effects on the lift created by the blade, a simple first-order response does not accurately represent what is actually taking place. A higher order response is needed to describe the dynamic departure. Therefore the scope of the next section is the development of a method that will more closely predict the actual blade response (which would include higher order terms) to determine the dynamic response to any number of disturbances.

4.2 Frequency Response Function Method

Since a first-order lag equation and even a second-order lag equation do not describe the dynamic departure with 100% accuracy, it was decided to create a new method for modeling dynamic blade row response. The dynamic blade response was assumed to be linear thus allowing the input (forcing function) to be transformed into a Fourier series which when multiplied by a frequency response function gives the dynamic response. This is an over-simplification because fluid mechanics is non-linear in nature, thus making the dynamic response of a mechanical/fluid system non-linear.

The frequency response function is defined as the Fourier transform of the response function divided by the Fourier transform of the forcing function. Therefore, all that is needed to model the blade frequency response is the time-domain representation of the forcing function and response function. Once the frequency response function, FRF, is identified, any forcing function can then be transformed into the frequency domain, then multiplied by the FRF and inversely transformed back into the time domain as the system response. The FRF will completely define the dynamic characteristic of the blade. The equation 4.8 defines the FRF

$$FRF = \frac{RF}{FF} \quad \text{Equation 4. 8}$$

where RF is the Fourier series of the response function and FF is the Fourier series of the forcing function.

Equation 4.9 shows a continuous Fourier series FRF.

$$FRF = \frac{\int_{-\infty}^{\infty} f(t)_{RF} \cdot e^{-i\omega t} dt}{\int_{-\infty}^{\infty} f(t)_{FF} \cdot e^{-i\omega t} dt} \quad \text{Equation 4. 9}$$

The continuous time series can be discretized into N number of data samples, which allows the Fourier transform to be written as

$$F_m = \sum_{k=0}^{N-1} f_k e^{-i(2\pi mk / N)} \quad m = 0, 1, \dots, \frac{N}{2} \quad \text{Equation 4. 10}$$

Using equation 4.10, equation 4.9 can now be written as

$$FRF = \frac{\sum_{k=0}^{N-1} f_{kRF} e^{-i(2\pi mk / N)}}{\sum_{k=0}^{N-1} f_{kFF} e^{-i(2\pi mk / N)}} \quad m = 0, 1, \dots, \frac{N}{2} \quad \text{Equation 4. 11}$$

Equation 4.10 can be broken down into real and imaginary components:

$$F_m = \sum_{k=0}^{N-1} f_k \cos\left(\frac{2\pi m}{N} k\right) - i \sum_{k=0}^{N-1} f_k \sin\left(\frac{2\pi m}{N} k\right) \quad m = 0, 1, \dots, \frac{N}{2} \quad \text{Equation 4. 12}$$

Equation 4.12 shows that the real and imaginary parts are the sum of all the data samples with a cosine or sine sequence having a period of N/m data samples. This gives us the magnitude of the real and imaginary components at each frequency. The real

component is represented by a_m and the imaginary component will be represented by b_m . Equations 4.13 and 4.14 define a_m and b_m .

$$a_m = \sum_{k=0}^{N-1} f_k \cos\left(\frac{2\pi m}{N} k\right) \quad m = 0, 1, \dots, \frac{N}{2} \quad \text{Equation 4. 13}$$

$$b_m = -i \sum_{k=0}^{N-1} f_k \sin\left(\frac{2\pi m}{N} k\right) \quad m = 0, 1, \dots, \frac{N}{2} \quad \text{Equation 4. 14}$$

Now that the Fourier series is broken down into its real and imaginary components, the FRF can now be calculated using the magnitude and phase of the forcing and response functions. The real and imaginary components at each frequency are transformed from a Cartesian coordinate system into a polar coordinate system and then division is performed.

$$\text{Magnitude} \quad M = \left[(a_m)^2 + (b_m)^2 \right]^{\frac{1}{2}} \quad \text{Equation 4. 15}$$

$$\text{Phase Angle} \quad \Phi = \tan^{-1} \left(\frac{b_m}{a_m} \right) \quad \text{Equation 4. 16}$$

$$FRF = \frac{M_{RF} \cdot \Phi_{RF}}{M_{FF} \cdot \Phi_{FF}} \quad \text{Equation 4. 17}$$

$$M_{FRF} = \frac{M_{RF}}{M_{FF}} \quad \text{Equation 4. 18}$$

$$\Phi_{FRF} = \Phi_{RF} - \Phi_{FF} \quad \text{Equation 4. 19}$$

The Fortran 90 code that calculates the FRF uses Equation 4.10 and Equations 4.15 through 4.19.

The use of a transfer function is only appropriate if the system behavior is linear. Therefore, the frequency response function method can only accurately describe the dynamic response if the blade response is linear or has non-linear dynamic departures that are of small magnitude. The assumption is that the blade response is nearly linear

and that the FRF is generalized enough to be applied to dynamic distortions other than from which it was derived.

The frequency response method was applied to the first-order lag equation to get an idea of the magnitude ratio and phase lag of the FRF. The forcing function was assumed to be a square wave, and was based on the lift created by an ideal blade having a chord of 0.03m, a turning angle of 27 degrees, and a stagger angle of 45 degrees. The blade experiences a two-per-rev change in incidence angle from 0 degrees to 2 degrees (see Figure 4.1). The blade response is lagged with a first-order function. Figure 4.3 shows the magnitude ratio of the FRF and Figure 4.4 shows the phase lag.

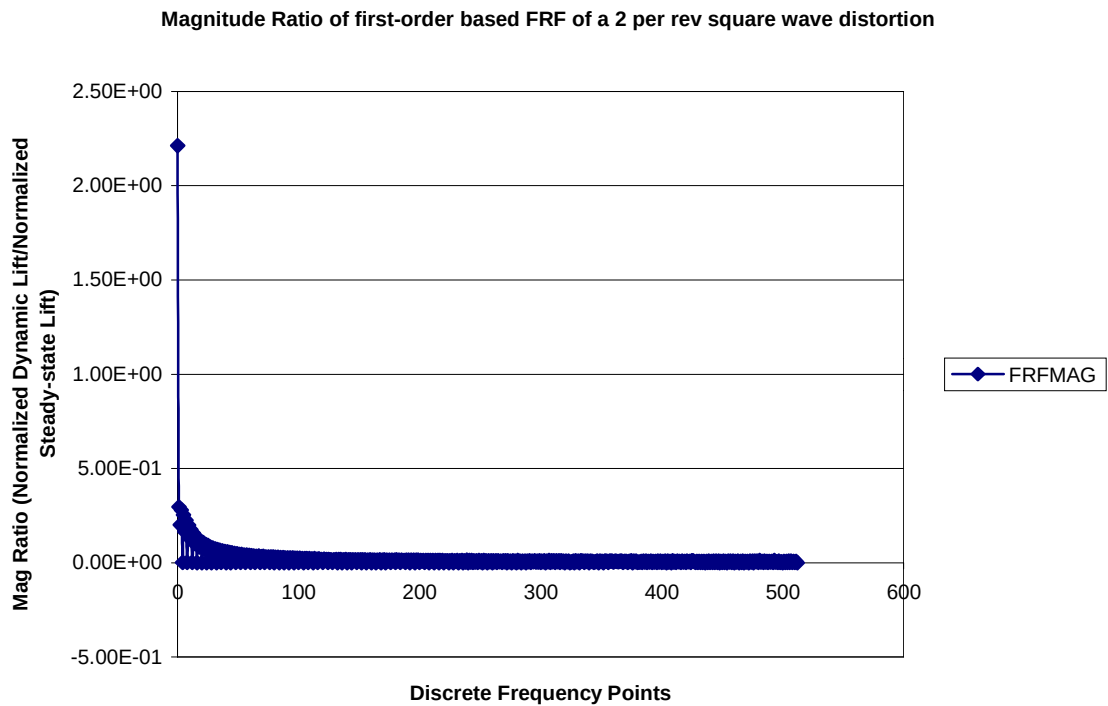


Figure 4.1: First-order FRF Magnitude Ratio

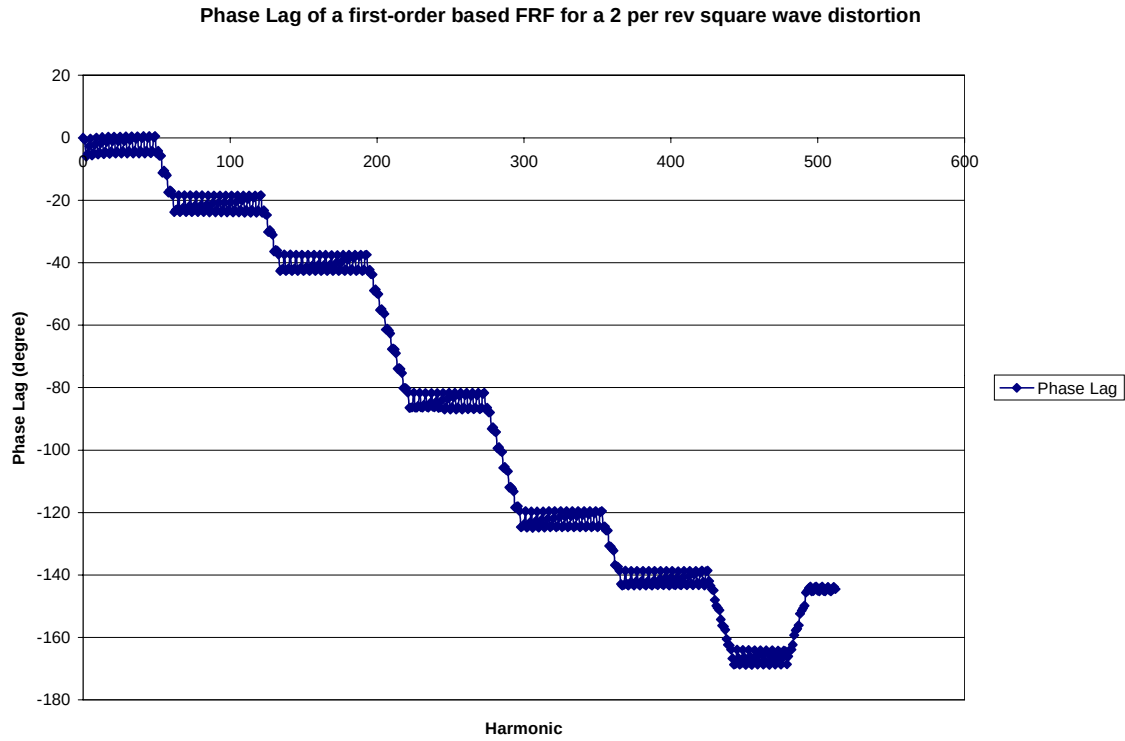


Figure 4.2: First-order FRF Phase Lag (no phase correction to eliminate discontinuities)

From Figure 4.3, it can be seen that at higher frequencies the magnitude ratio between the forcing function and the response function quickly decreases to zero. Only the magnitude ratio for the first few harmonics are important. This shows that the high frequency components of the response contribute very little to the overall blade response. The first several components are significant because of the two-per-rev harmonic of the distortion. The magnitude ratio is larger than two because of the method of normalization by using the maximum value. Since the magnitude ratio at zero frequency is the sum of the data points and because of the method of normalization used, the magnitude ratio at zero frequency is larger than two even though the DC components of the response and the forcing function are the same. If the data had been normalized by its average, then the magnitude ratio would have been one at the zero-frequency point. The phase lag plot shows the response lags the forcing function at higher frequencies according to the affects of a first-order lag function.

The magnitude ratio and phase lag of the first-order based FRF were then multiplied by the forcing function. This created phase and magnitude responses that were then converted into real and imaginary components and inverse Fourier transformed back into the time domain. The FRF calculated response was compared with the original first-order response to check accuracy. Figure 4.5 shows the plot of the square wave forcing function, the original first-order response, and the FRF response. Figure 4.6 shows the percent error between the original first-order response and the FRF-based response.

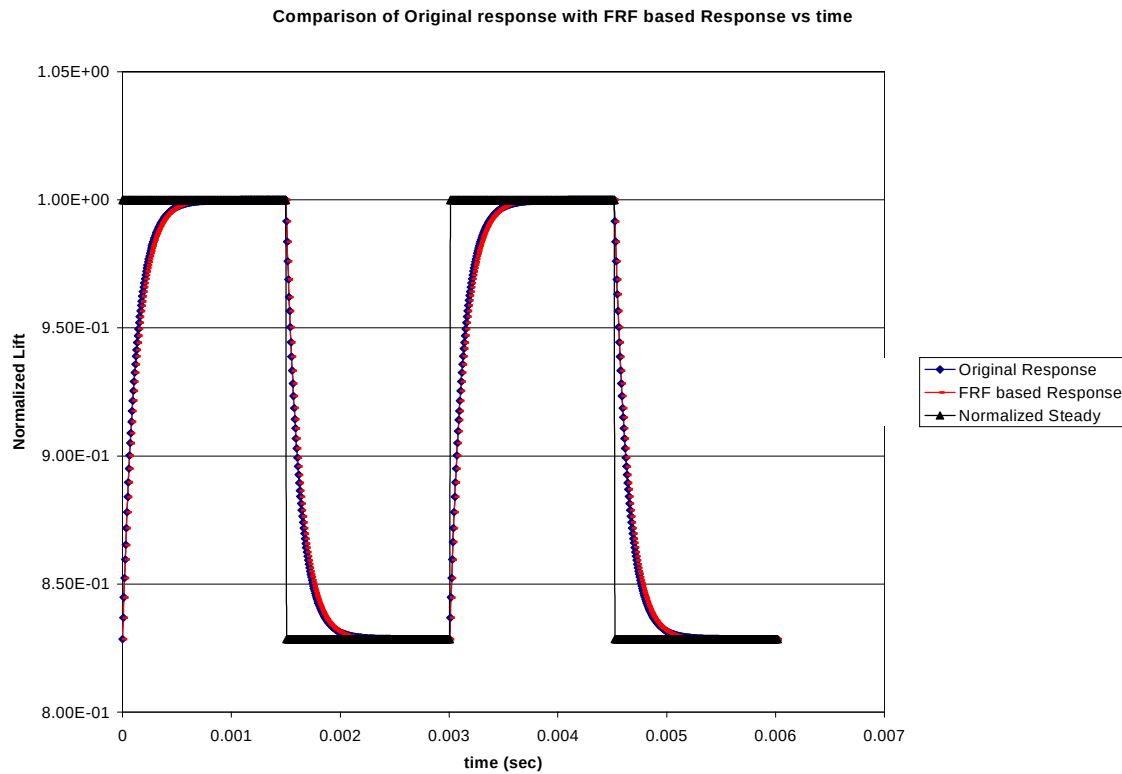


Figure 4.3: Comparison of original and FRF based responses

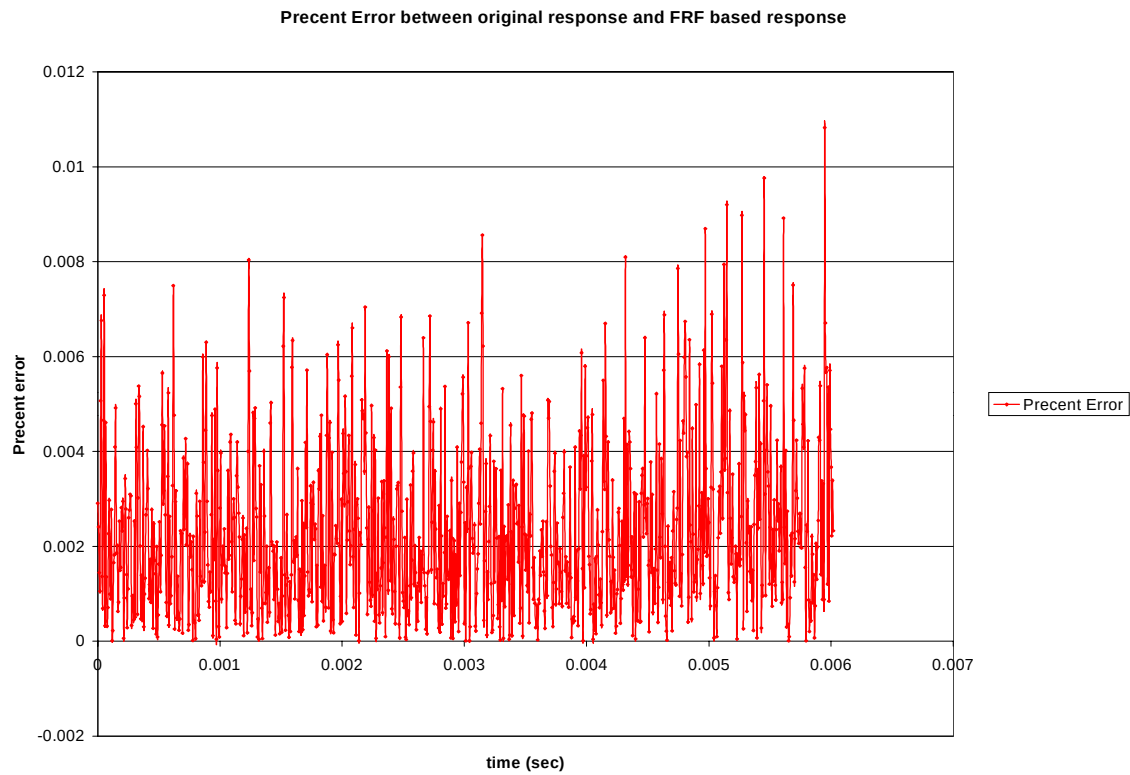


Figure 4.4: Plot of Percent Error between Original and FRF based Responses

This example shows that the FRF method for generating a response can be highly accurate if there are enough data samples in the original input. It will be seen later using experimental data with few data samples that the calculated response decreases in accuracy, but even with its small number of data samples the modeled response is still closely representative of the actual response.

5 FRFs Based on Simple Dynamic Compressor Model

5.1 Simple Dynamic Compressor Blade Response Model

A simple, dynamic, one-dimensional compressor blade response model was created in order to create data sets for use in the development of the frequency response function method. At the outset of this thesis, it was desired to obtain a better understanding of how a blade responds dynamically to a change in inlet conditions. While reviewing multiple papers dealing with dynamic response (Tesch and Steenken, 1981; Melick and Simpkin, 1972), it was noted that most authors used the idea of effective angle of attack to model the dynamic departure. It was felt that the reason the blade response lagged the steady state, or instantaneous response, was because a portion of the new information moved across the blade at the speed of convection. It was decided to create a simple model that used this idea.

Cousins (1998) states that a blade does not react to a change in inlet conditions until the fluid particles carrying the new information had reached the throat of a transonic blade. The present model uses a similar idea. The blade will not fully respond until the fluid particles carrying the new information have traveled the full length of the blade. Each section of the blade will respond as it senses the new flow conditions. The simple model assumes that the dynamic response relies solely on properties that propagate downstream at the speed of convection.

An isolated rotor row is modeled using a cascade with a perturbed flow representing the rotor and the effects of distortion screens. The model assumes ideal flow over the blade with no separation and a two-degree deviation angle to represent a boundary layer over the blade. The flow is assumed incompressible because of the low axial speeds used, and because of the low pressure rise of a single axial compressor rotor. The change in density between the distorted and undistorted regions is included in the model. The distortion in the model is viewed as a change in angle of attack, and therefore a change in the mass flow rate. The model calculates the work done per blade using the Euler turbine equation (Equation 5.1) for the distorted and undistorted regions.

$$w = U \cdot (V_{\theta 2} - V_{\theta 1})$$

Equation 5. 1

Once the work is known, the change in stagnation temperature across the blade can be calculated. The total pressure rise is then calculated using isentropic relationships. The model then finds the static properties at the inlet and exit, which allows for the calculation of the tangential and axial forces acting on the blade. When these forces are known, the distorted and undistorted total blade lift can be calculated.

An idea by Moses (1997) is then used concerning lift. It is assumed that a triangular pressure distribution profile is an approximate representation of the actual pressures on a blade. This approximation allows for the calculation of the blade lift, but does not exactly match the actual pressure side or suction side pressure profiles. The reason for using the triangular pressure distribution idea is to simplify the code, and because an actual blade profile was not used in this program, a true pressure profile could not be calculated. Only the inlet metal angle, exit metal angle, chord length, spacing, and turning angles are used in this model. These geometric properties are based on experimental data for a NACA 65(12)10 blade row with a solidity of 1.0 (Hill, 1992). By using a triangular pressure distribution, it is easier to calculate lift per discrete blade segment. Since the total blade lift differs for the undistorted and distorted regions, the triangular pressure distributions will be different.

The model simulates the dynamic blade response by taking incremental steps in time as the blade moves through the distorted and undistorted regions. At each time step, the total lift of the blade is computed by summing up the lift for each incremental blade segment. The blade is divided into segments based on the relative axial flow velocity times the specified time step. As the fluid flow change sweeps over the blade at the speed of convection, the pressure distribution for that blade segment changes to the new pressure distribution for that segment. This method allows a simply dynamic response to be calculated. The model can calculate the response to multiple distortion waves that sweep over the blade at a given time. Figure 5.1 shows how this scheme works.

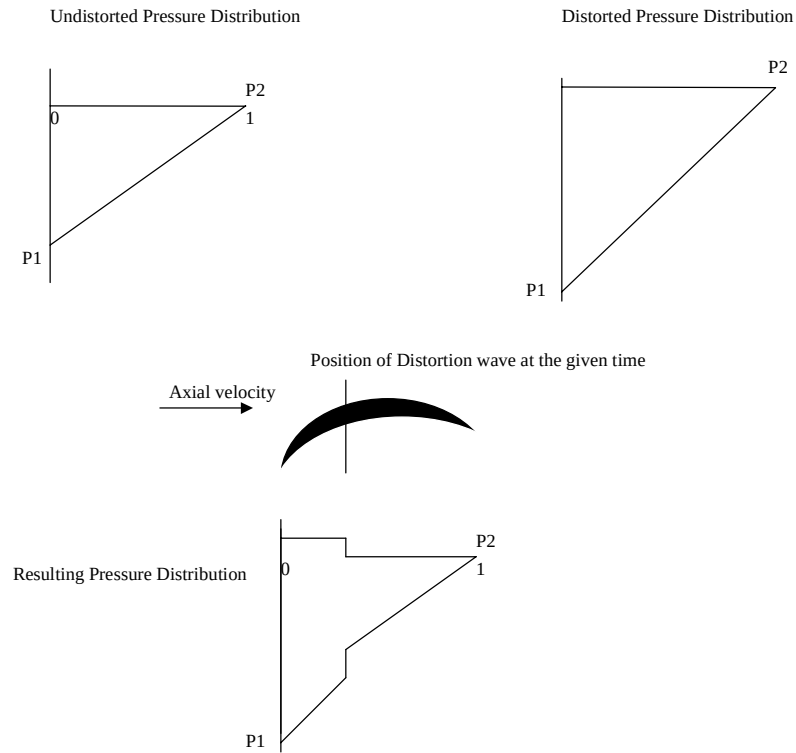


Figure 5.1: Triangular Pressure Distributions for calculating lift and time marching scheme of change in lift for dynamic response

Although this model is relatively simple in its calculation of flow properties, it does have the flexibility to run different blade geometry and distortion patterns. This allows an investigation into the geometric and distortion parameters that will have the largest impact on the dynamic response.

5.2 FRFs Based on the Model Simulations

The first cascade simulated by the model had the following parameters: Chord = .03m; Inlet Total Pressure = 101 kPa; Inlet Total Temperature = 300 K; Undistorted Incidence Angle = 0 deg; Distorted Incidence Angle = 2 deg; Two 90 deg distortion screens each 90 deg apart; Rotor Speed = 10000 RPM.

Figure 5.2 illustrates the steady state, or instantaneous lift, and the dynamic blade response.

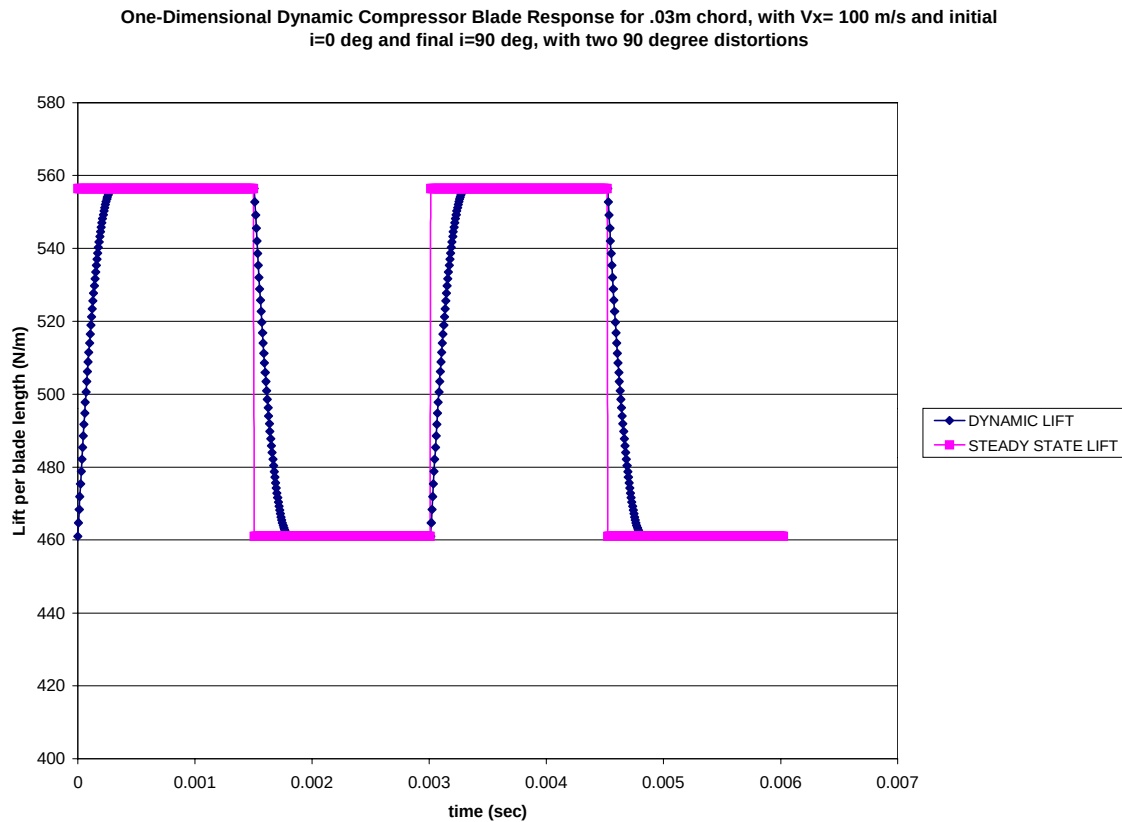


Figure 5.2: Blade Response based on simple Dynamic Compressor Code

The dynamic blade response calculated in the one-dimensional compressor code resembles a first-order lag response, but it is not. Figure 5.3 illustrates the differences between the two responses.

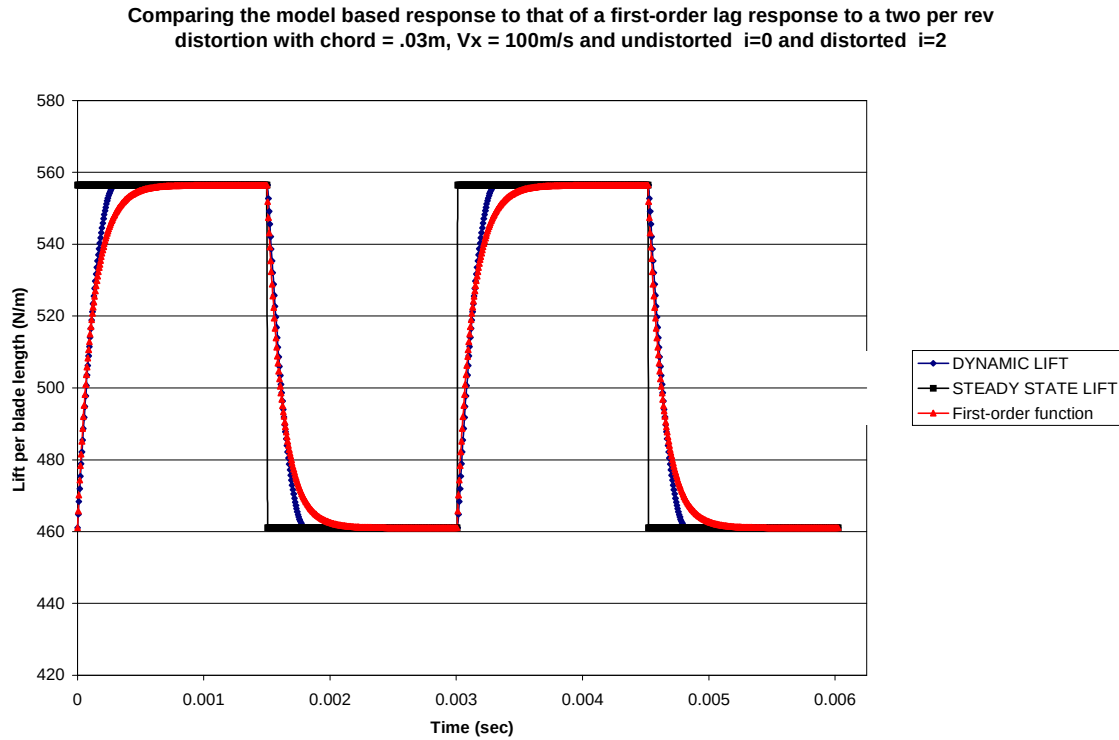


Figure 5.3 clearly shows that the first-order lag response initially has a higher rate of response, but as time continues the first-order response predicts a lower response magnitude than the model-based response to the distortion. Overall, the first-order lag response takes longer to respond to a distortion than the response based on the model. Figure 5.4 shows the magnitude ratio of the FRF based on the model response, and Figure 5.5 shows the phase lag of the FRF. The magnitude ratio and phase lag differ from the first-order lag magnitude ratio and phase lag. The phase lag is especially different. Note that at higher frequencies there is an increase in phase lag, but the increased phase lag acts on a smaller magnitude ratio, which at high frequencies is insignificant (negative phase means lag, positive phase means lead).

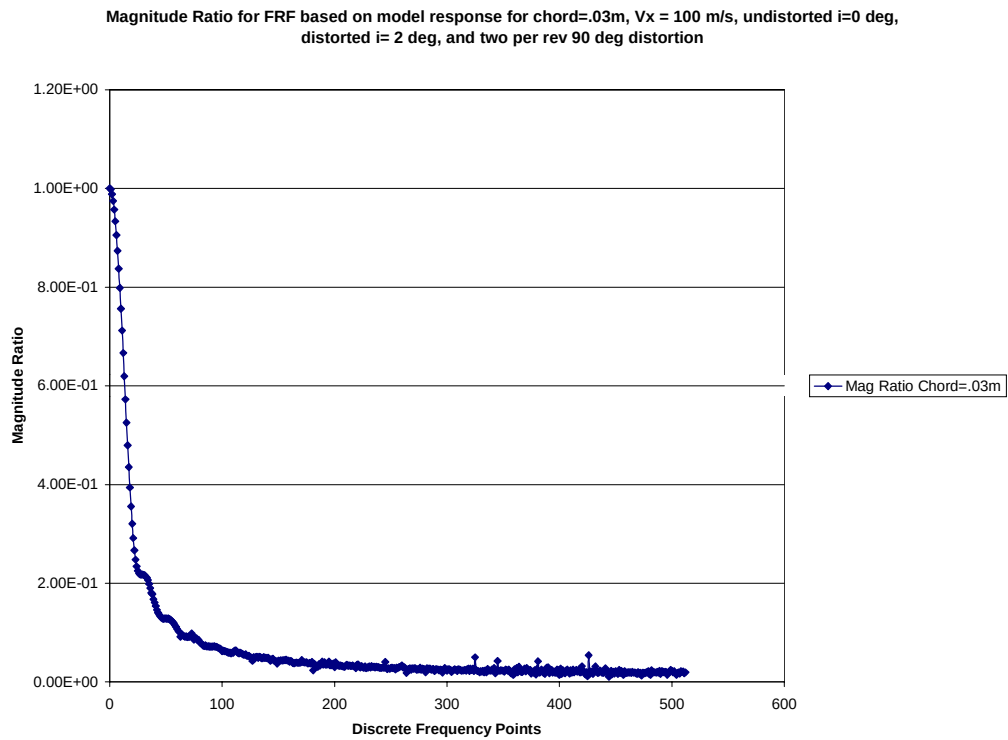


Figure 5.4: Magnitude Ratio for FRF based on model response to a 2 per rev distortion

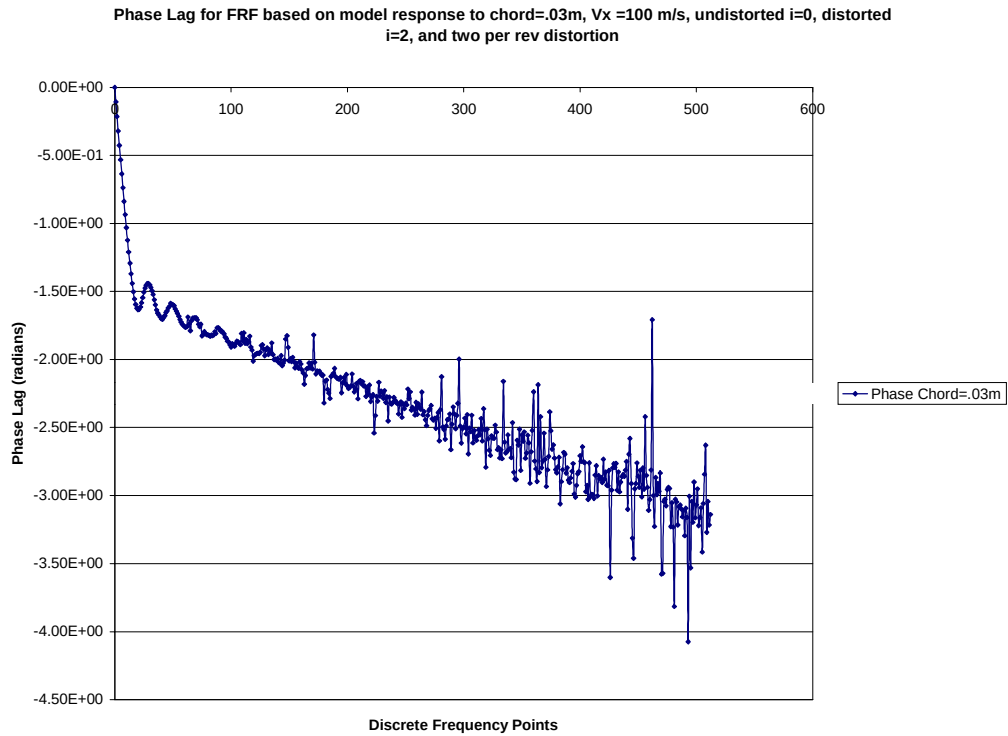


Figure 5.5: Phase Lag for FRF based on model response to a two per rev distortion

Next, the influence of distortion frequency was tested to see what effect it had on the FRF. The FRF generated using a two-per-rev distortion (the previously discussed FRF) was multiplied by a six per rev 30 degree distortion wave with the undistorted incidence angle equal to 0 degrees and the distorted incidence angle equal to 2 degrees. The chord length was also held constant at 0.03m. Figure 5.6 shows the results of the FRF predicted response compared to the model calculated response for the same forcing function. Both the FRF response and the model response are identical. This illustrates that the frequency of the distortion has no effect on the frequency response function of a specific blade. This illustrates the generality of the FRF by showing the ability to calculate the blade response for more than just the combination of inlet and exit conditions from which it was created.

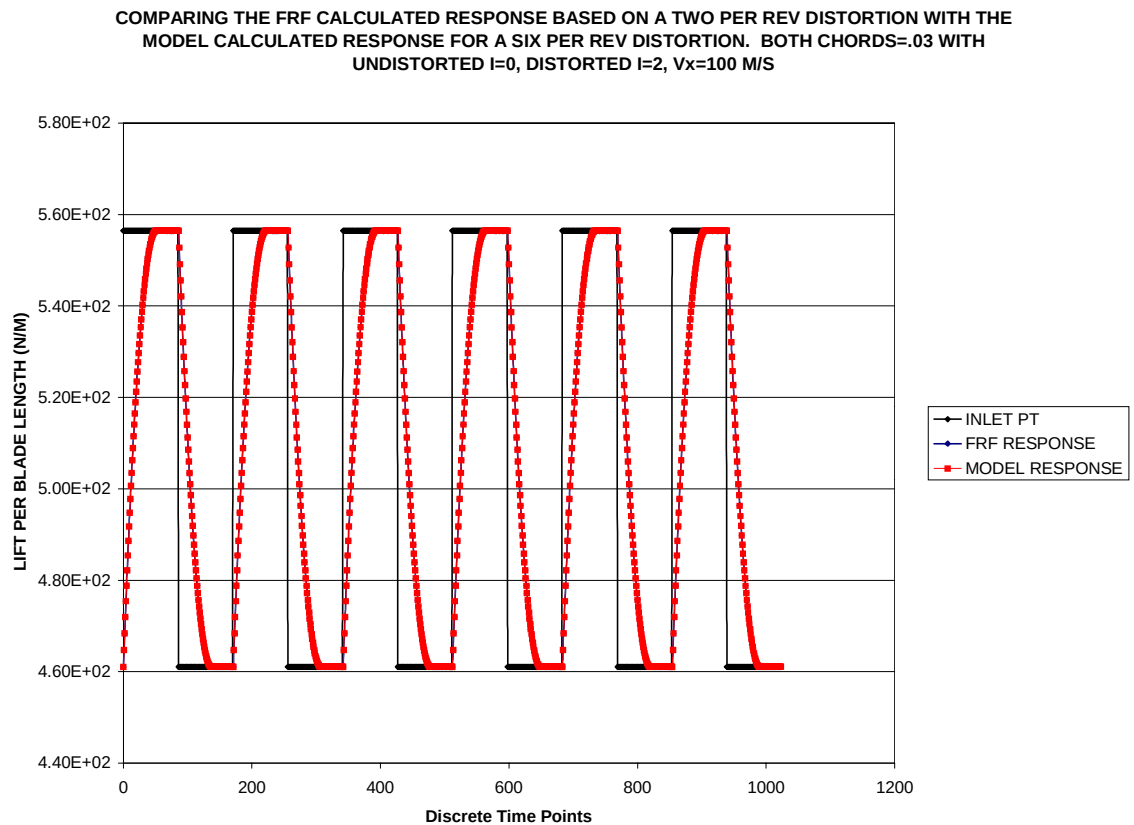


Figure 5.6: Comparing the FRF response that was generated using a two per rev distortion with the model response of a six per rev distortion

For the next data set, the chord was shortened from 0.03m to 0.02m. This new response was then compared with the response of the FRF based on a 0.03m chord length for the same forcing function. The results are illustrated in Figure 5.7. The FRF response is slower than the simple model predicted response. This is reasonable, since the FRF response is based on a fluid particle transport time that is longer than the actual transport time. Increasing the axial flow velocity in this model will also have a similar effect to the blade FRF because the fluid transport time will be shorter for a given blade chord length.

Figure 5.8 compares the model predicted blade responses for each chord length. The shorter blade has a noticeably shorter response time. The response time increased when the chord length was extended to 0.035m. These results also show that chord length has a large influence on the blade response.

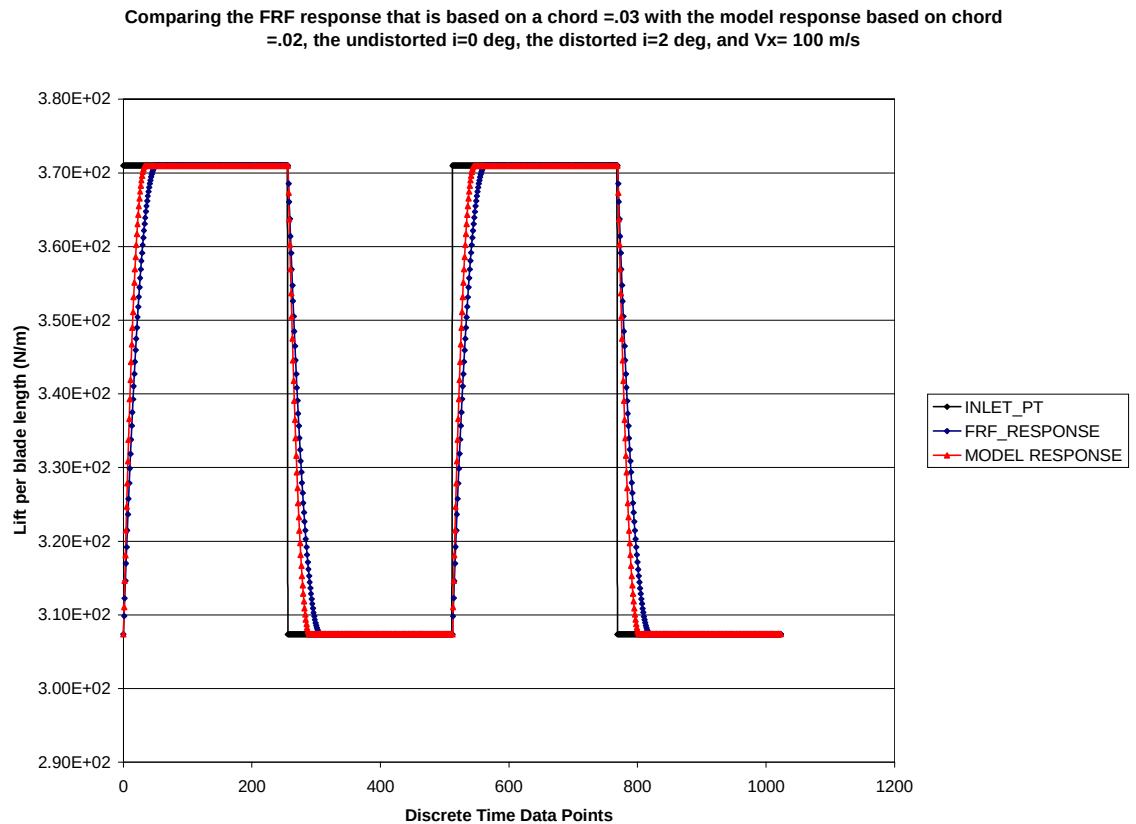


Figure 5.7: Comparing FRF Response based on a chord equal to .03m to the model predicted response for a .02m blade.

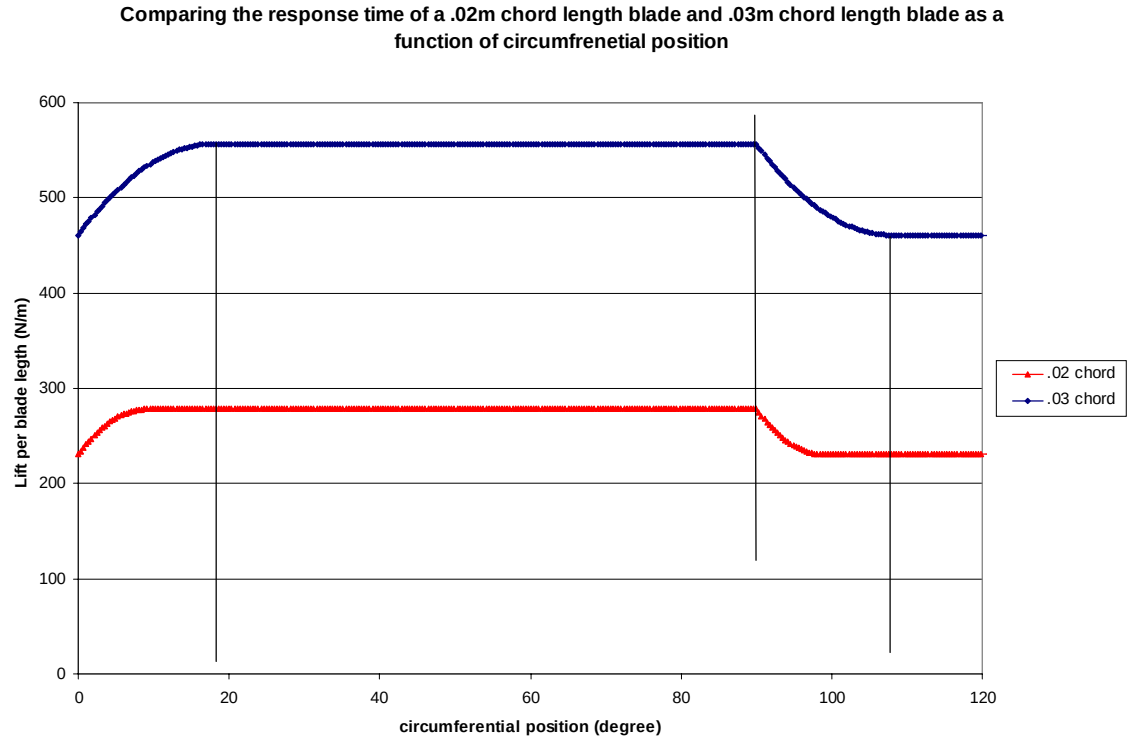


Figure 5.8: Comparing model responses for a chord equal to .03m to a chord equal to .02m

The difference in FRF responses based on either the 0.03m blade or the 0.02m blade is also noticeable. The 0.02m chord magnitude ratio curve is shifted towards the higher frequencies, which is illustrated in Figure 5.9. This causes the magnitude ratio for each harmonic to be higher than the magnitude ratio for the 0.03m chord blade. The shorter blade will have a higher response, which corresponds to higher frequencies. The 0.02m chord phase lag curve is also shifted over to the higher frequencies, which is shown in Figure 5.10. This further enforces that the faster response has larger values at the higher frequencies.

Comparing the effect of chord length on Mag Ratio, with all other parameters held constant

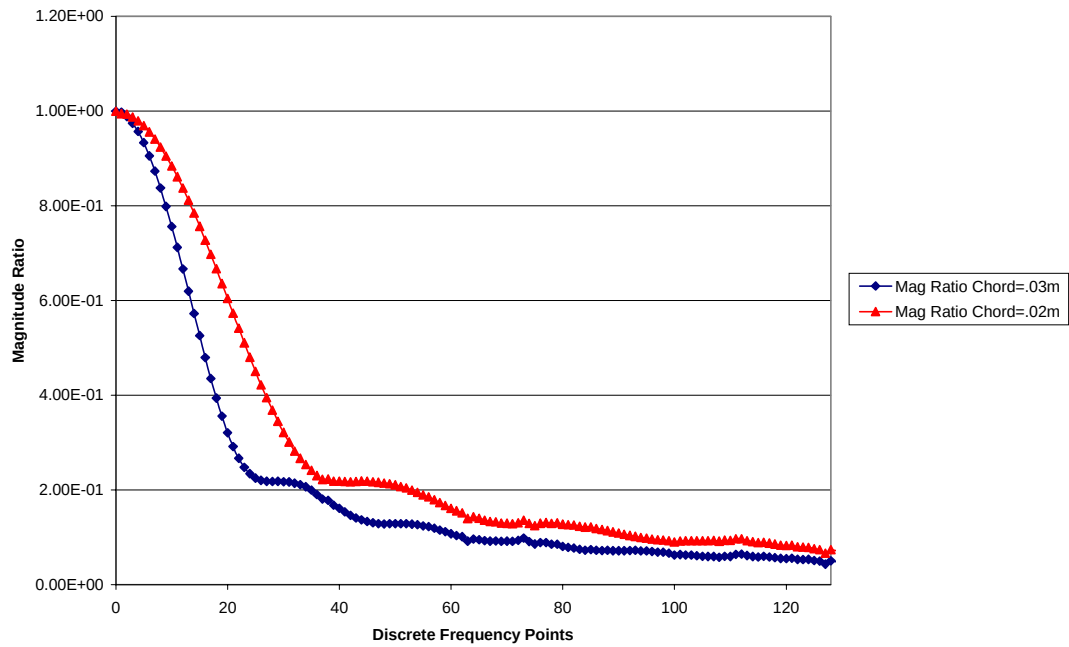


Figure 5.9: Comparing the Effect of Chord Length on the Magnitude Ratio of the FRF

Comparing the phase lag for FRFs based on .02m or .03m chord lengths with $V_x=100$ m/s, and initial $i=0$, final $i=2$

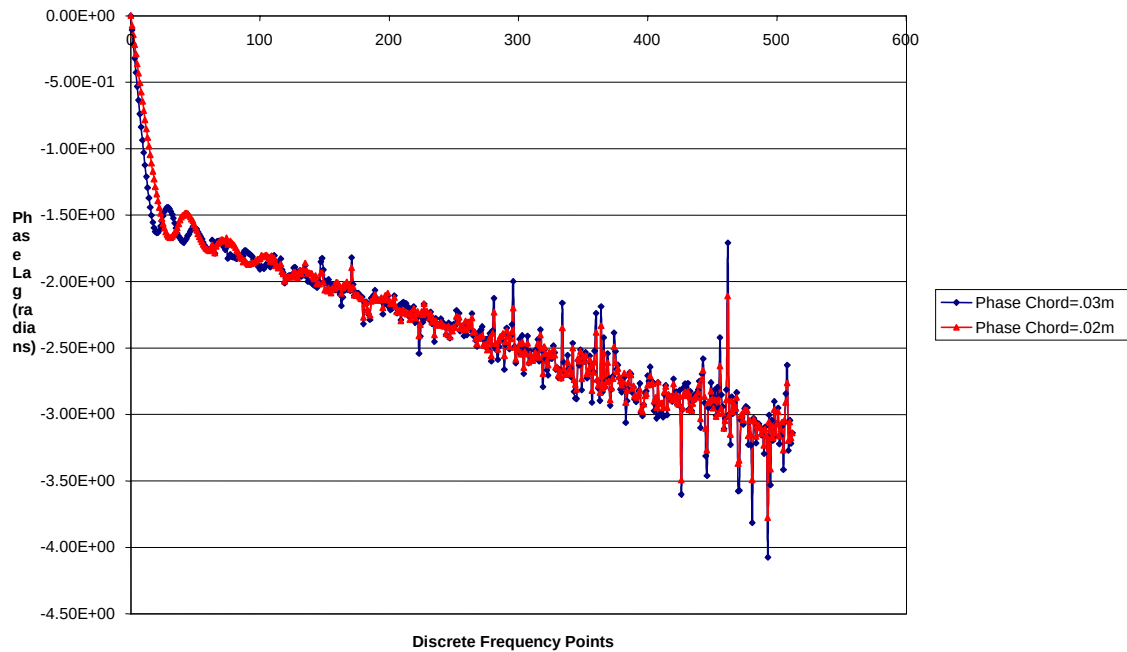


Figure 5.10: Comparing the Effects of Chord Length on the Phase Lag

The effect of distortion intensity on the FRF of the dynamic blade lift was investigated. The geometry of the blade remained the same with chord length equal to 0.03m, and the axial velocity remained the same. The original distortion involved a step change of a two-degree increase in the incidence angle. The new distortion will increase that step change to four degrees, but will still retain the two-per-rev cycle. The model response to the new distortion was compared to the response of the FRF that was based on the original distortion intensity. The result of the FRF predicted response compared to the model generated response is illustrated in Figure 5.11.

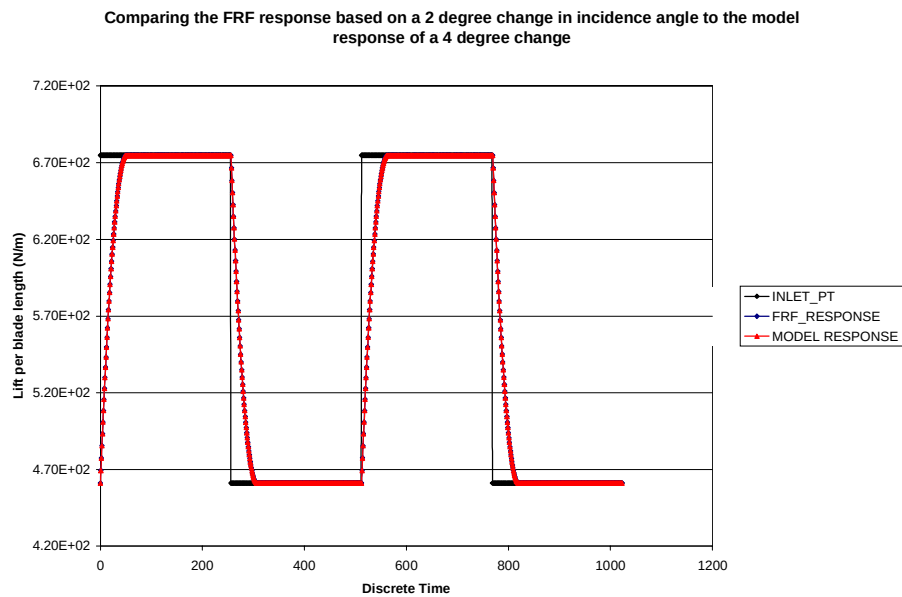


Figure 5.11: Comparing the responses to a 4deg step change distortion of a FRF generated from a two-degree change and that of the model prediction

The distortion intensity has no effect on the ideal lift FRF of a particular blade. For an actual blade this may or may not hold true since generally other parameters other than the angle of attack are changing when the blade passes through a distortion. This result increases the practicality of using an FRF to predict dynamic blade response for other types of distortions.

The effect of the turning angle of the blade, and hence the inlet and exit metal angles, on the FRF was then examined. The original inlet and exit metal angles based on a NACA 65(12)10 blade are 55 degrees and 28 degrees, respectively. The new angles based on the NACA 65(12)10 turning profile are 60 degrees and 30 degrees for inlet and exit metal angles, respectively. The stagger angle remained at 45 degrees. The chord length remained constant at 0.03m and the distortion step change in incidence angle remained at two degrees with $V_x=100$ m/s. The simple model response was then compared to the FRF predicted response, generated using the original turning angle, to the new steady-state lift. Figure 5.12 shows the results.

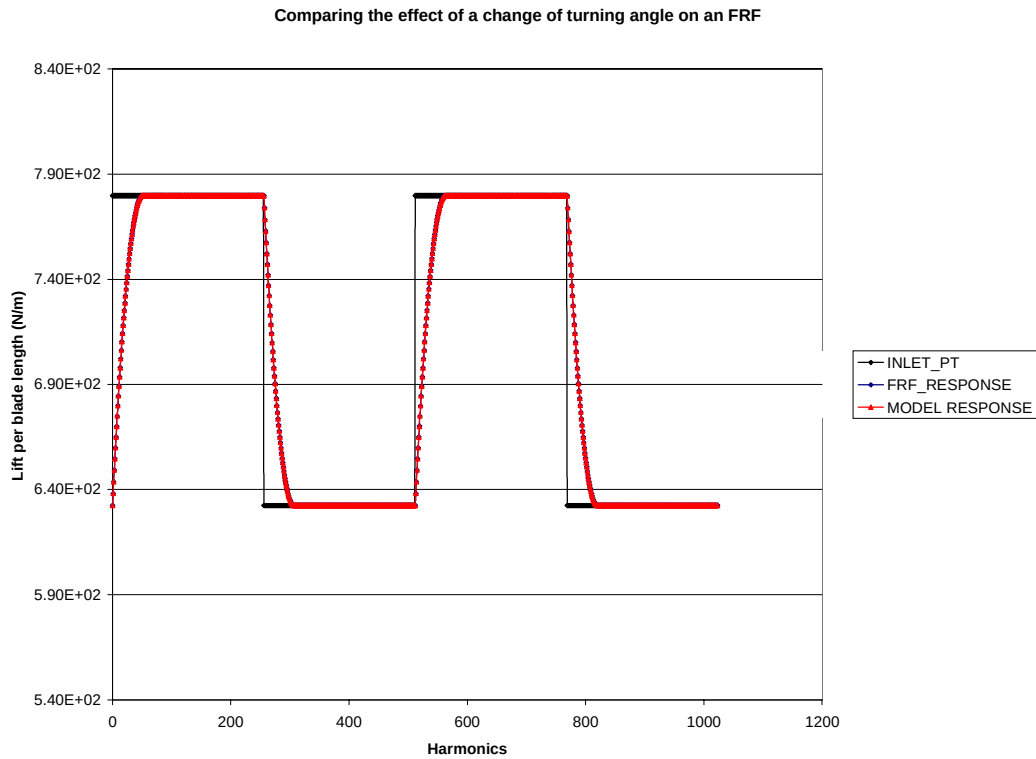


Figure 5.12: Comparing the effect of turning angle on the frequency response function of an ideal blade

The results show that the change in turning angle had no effect on the ideal dynamic blade frequency response function. Because of the simplicity of the model, the change had no effect other than to increase the total blade lift for both the steady-state and dynamic responses. In reality the change in blade geometry could change the blade

response. Each blade design will have a characteristic response associated with it. According to the simple dynamic blade response model, only the change in chord length had an effect on the blade's FRF. This is because the fluid particle transport time increased with increased chord, which changed the blade's response time to a change in flow conditions.

6 Dynamic Wake experiment Data and Resulting FRFs

This chapter is organized in two sections. The first section will cover the dynamic blade wake response experiment performed by Boller (1998). This section will give a brief description of the set up and acquisition of data. The second section will cover the development of frequency response functions generated from the inlet and exit flow data. It will cover which parameters were chosen and why. The FRFs will then be compared and analyzed.

6.1 Dynamic Blade Wake Response Experiment

Boller (1998) completed an extensive investigation into the dynamic blade wake response to determine the relationship between wake response and blade lift response to dynamic inlet changes. His work was the first part in a series of studies to improve dynamic compressor blade modeling. This work is a continuation of that research.

Boller conducted his experiment in a test cell at the Virginia Tech Turbomachinery Laboratory as shown in Figure 6.1. The dimensions of the cell are 25ft x 12.5ft x 15.0ft.

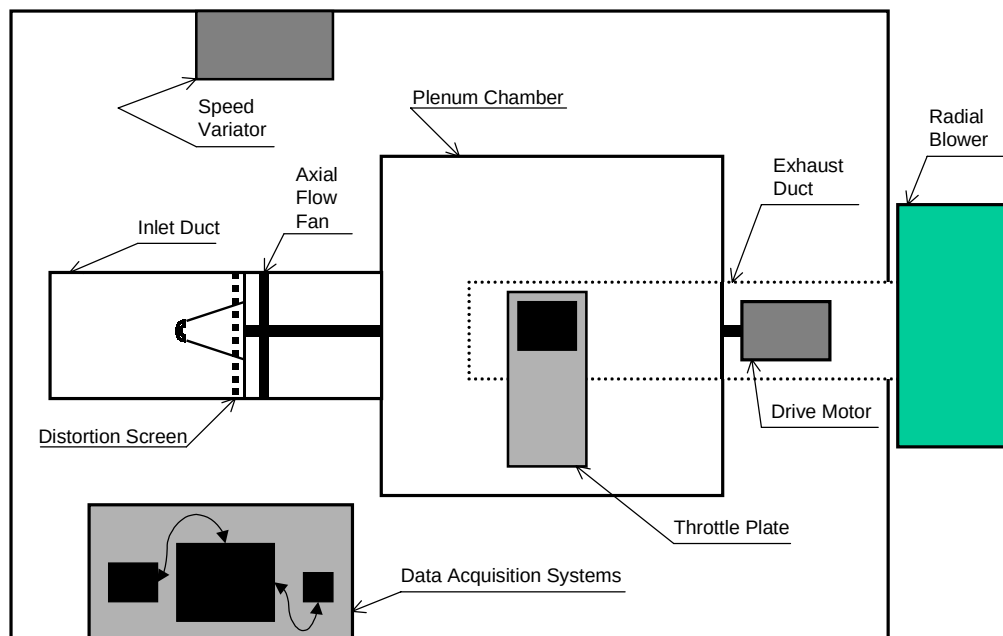


Figure 6.1: Schematic of Test Cell (Top View)

The laboratory test machine consisted of an open-circuit, subsonic General Electric Company Model 5GDY34A1 axial flow compressor. For this experiment, the compressor was operated as an isolated rotor, in order to map the individual blade row wake response without interference from additional upstream and downstream stages. A 5 Hp General Electric motor drives the axial rotor at a rotational velocity of 2100 rpm. The annulus has inner and outer diameters of 12.375 inches and 18.0 inches, respectively. The rotor was bladed with 24 RAF-6 propeller sections whose profile is shown in Figure 6.2. The blades have a span of 2.75 inches and a chord length of 1.67 inches. The blades are set with a stagger angle of 55° at mid span with a twist of 4° . This twist creates a free vortex flow distribution downstream of the rotor, according to a personal conversation with Sexton. The nominal rotor blade casing clearance was 0.05 inches. This arrangement resulted in a solidity of 0.84 at the mean radius, an aspect ratio of 1.68, and a hub to tip ratio of 0.687, which produces a flow area of 0.975 ft^2 . A nose cone with a base diameter of 12.375 inches, axial length of 18.0 inches, and tip radius of 1.5 inches is attached to the front of the rotor.

The inlet duct had a length of 4.0 ft with an attached bellmouth. The flow from the rotor exited into a cubic plenum chamber of 6.0ft x 6.0ft x 4.0ft. A mechanical throttle plate is located at the top of the plenum chamber, and is driven by an electric traverse mechanism. The throttle plate was controlled by a feedback control loop to reduce the unsteadiness of the plenum total pressure during normal unstalled operation.

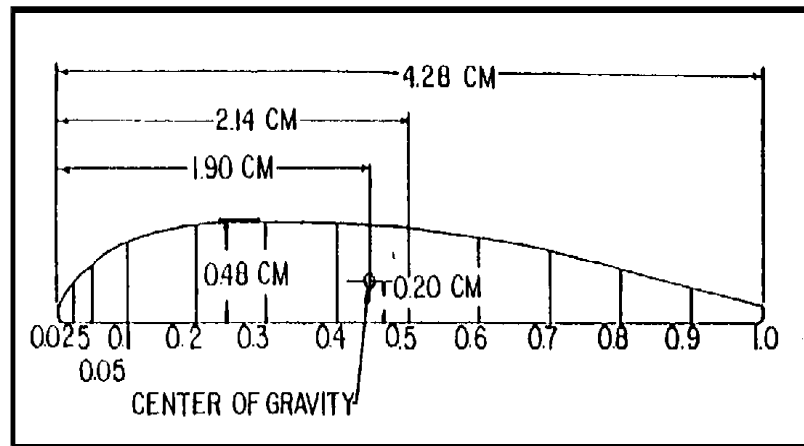


Figure 6.2: Cross Section of RAF-6 Airfoil

The instrumentation used in the experiment was a Pitot probe rake and a Pitot-static probe for measuring the mass flow rate, a five-hole pneumatic pressure probe for the inlet survey, and a piggyback steady/unsteady total pressure probe for the exit survey. A schematic of this set up is illustrated in Figure 6.3.

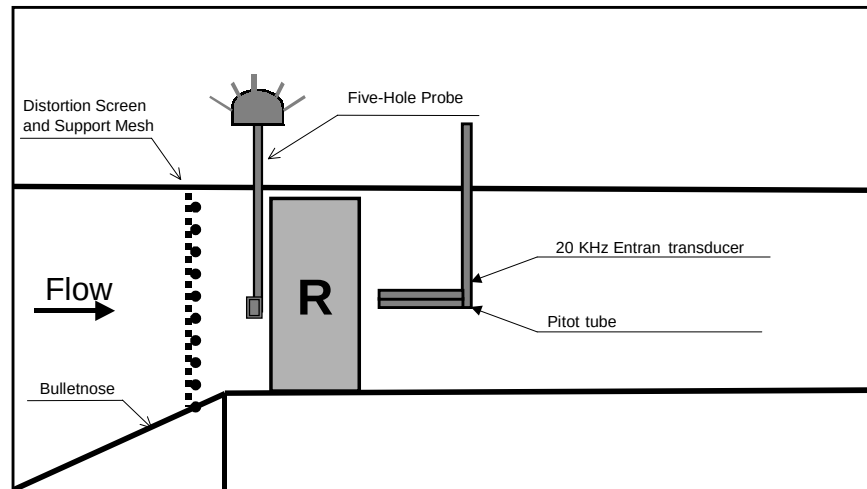


Figure 6.3: Pneumatic Probes Relative to the Rotor in the Meridional Plane at Mid-span

The five-hole probe is 0.25 chords or 0.4175 inches upstream of the rotor. The probe was used in a non-nulling mode with its orientation fixed with respect to the axis of the machine. The five-hole probe measured pressure directly on and in the yaw and pitch plane. This allowed for pressure coefficients to be calculated and for the total pressure, static pressure, and the yaw and pitch angles of the fluid flow at the inlet to be calculated. Figure 6.4 shows an illustration of the five-hole probe.

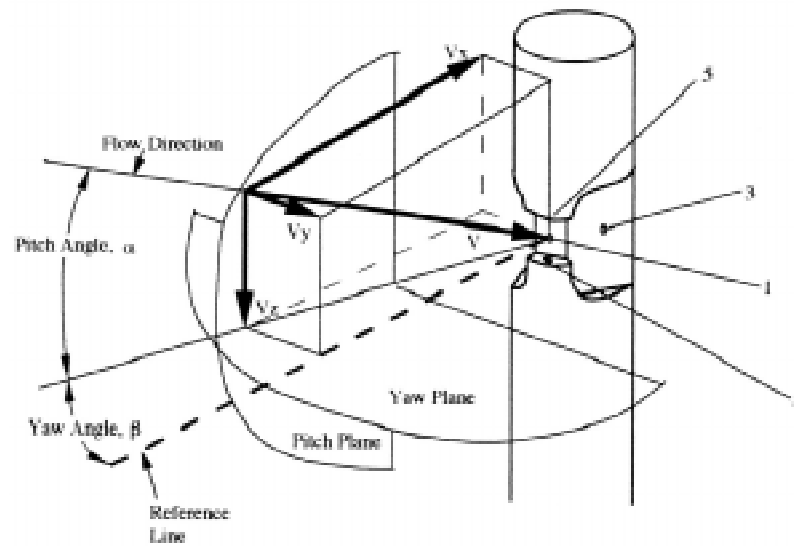


Figure 6.4: Schematic of Five-Hole Pressure Probe Tip and Flow Angles

A piggyback setup of a steady-state Pitot tube with a high-response Entran pressure transducer was used to measure the blade wake. The piggyback probe was placed 0.2 chords behind the rotor blades. The steady and unsteady measurements were recorded at 20 kHz for 0.12 seconds using a LabView virtual instrument. The data was phase locked using a one per rev signal. The data acquisition system for both the inlet and exit surveys was a Gateway P166 laptop with a National Instruments DAQCard-AI-16E-4 I/O card.

The experiment utilized three different strength distortion screens to test the rotor row response to different distortion intensities. The Boller thesis describes the three distortion screens and the one support screen. Boller then created the steady-state characteristic map shown in Figure 6.5.

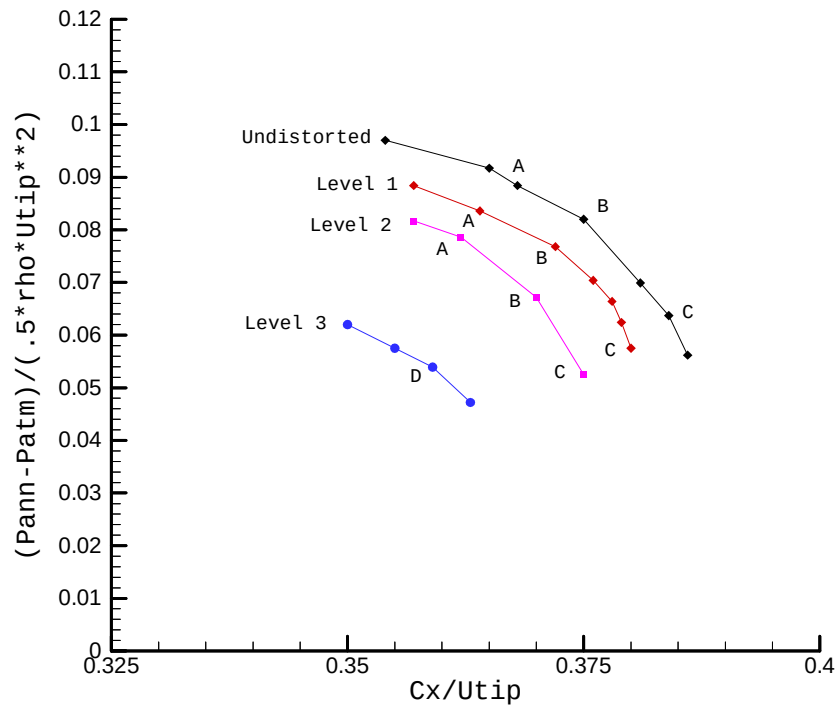


Figure 6.5: Distorted and Undistorted Low-Speed Axial-Flow Compressor Performance Characteristics

Boller then ran tests at each aerodynamic loading point (A, B, C on the map) for each distortion level (1 through 3) at span positions of 1/3 span, midspan, and 2/3 span. This totals 27 different experiments with different distortion intensities, span position, and aerodynamic loading parameters. At the inlet Boller measured the angle of attack, total pressure relative to the ambient, static pressure relative to the ambient, and pitch angle. Equations 6.1 and 6.2 show the non-dimensionalized total and static pressure calculations.

$$P_{total\ Rel} = (P_{total} - P_{amb}) / (\frac{1}{2} \rho \cdot U_{tip}^2) \quad \text{Equation 6. 1}$$

$$P_{static\ Rel} = (P_{static} - P_{amb}) / (\frac{1}{2} \rho \cdot U_{tip}^2) \quad \text{Equation 6. 2}$$

At the exit of the rotor blade, Boller measured the median total exit pressure, suction side semi-wake width, maximum suction side jet magnitude, wake semi-width, and maximum wake defect magnitude, as illustrated in Figure 6.6.

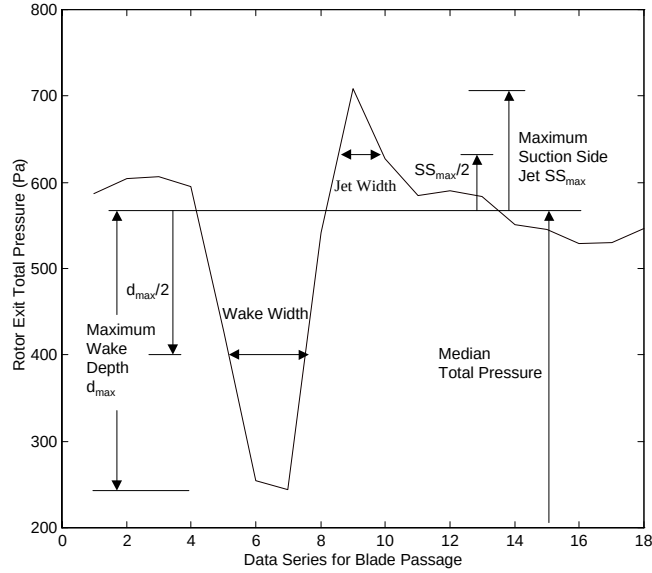


Figure 6.6: Exit Flow Parameters of Sample Blade Passage

The total pressure of the suction side jet and the wake defect were computed relative to the median exit total pressure. These relative pressures were then non-dimensionalized with respect to blade tip speed as presented in Equations 6.3 and 6.4.

$$\overline{SSJetMag} = (SSJetMag - Pt_{median}) / (\frac{1}{2} \rho \cdot U_{tip}^2) \quad \text{Equation 6. 3}$$

$$\overline{WakeDefectMag} = -1 \cdot (Pt_{median} - WakeDefectMag) / (\frac{1}{2} \rho \cdot U_{tip}^2) \quad \text{Equation 6. 4}$$

The suction side jet and wake semi-widths were converted from length to units of time and non-dimensionalized by the time $\Delta t = 1.2$ msec for a blade passage at 2100 rpm as presented in Equations 6.5 and 6.6. The exit median pressure was related to the ambient conditions and then normalized by blade tip speed as defined by Equation 6.7.

$$\text{SS Jet Semi-Width} = (\text{Time for SSjet}/\Delta t) \quad \text{Equation 6. 5}$$

$$\text{Wake Semi-Width} = (\text{Time for Wake width}/\Delta t) \quad \text{Equation 6. 6}$$

$$Pt_{ExitMed} = (Pt_{exit} - P_{amb}) / (\frac{1}{2} \rho \cdot U_{tip}^2) \quad \text{Equation 6. 7}$$

The results of the Boller experiments are graphically represented in Appendix A.

6.2 *Frequency Response Functions Generated from Experimental Data*

This section is organized in three subsections. The first two subsections will cover the two separate sets of data from the Boller experiment that were used to create two separate frequency response functions for the blade at each test condition. These subsections will look at the effect different parameters had on the blade response and will show how the newly-generated FRFs can be used to model the blade response to different forcing functions. The third subsection will discuss a method for graphically showing dynamic compressor response. The reason two data sets were used was to find any trends among the responses and to compare them against each other for similarities. The two data sets that were used were angle of attack as the forcing function with wake semi-width as the response, and inlet total pressure as the forcing function with exit median total pressure as the response. The reason for the choice of parameters was because of their generally high correlation between the input and output parameters. The total pressures had a higher average correlation coefficient for a number of data sets with a correlation greater than 0.9. A correlation coefficient of one means the response is totally dependent on the input with no other influence effecting the output. The angle of attack vs. wake semi-width had lower correlation coefficients, but the coefficients for the most part were higher than 0.7. These coefficients may not be acceptable for linear systems, but since the system is non-linear, it was felt that anything above 0.7 was reasonable. For a linear system a coefficient of 0.95 or higher is considered reasonable.

Since the data set for each run only covered 270° of circumference, the region not measured had to be “padded” for periodicity. This region was directly opposite of the 110° distortion screen, and Boller felt that the flow in this region was steady and did not fluctuate due to the distortion. Because of this the data was assumed to be constant between the 150° data point (last data point) and the -120° data point (first data point). Any discontinuity between the points was smoothed out with a linear curve fit. The discontinuities that were found were all within Boller’s error margin so it was felt that no additional error was introduced into the data with the padding to make the inlet distortion wave periodic. The distortion screen extended from 0° to 110° .

It should be noted that the waveform for the Boller data is different than the waveform in the simple dynamic response code. In the response code, the distortion was

seen as a step change in time by the blade. Because of the fact that the measurements taken by Boller were in the absolute frame of reference and not taken on the blade, the disturbance is actually a spatial wave of an impulse variety. However, because the rotor blade moves through the spatial wave at a certain velocity the blade sees the disturbance as a wave in time and not in space. This allows the inlet spatial wave to be viewed as the forcing function and the exit spatial wave to be viewed as the response function in the relative frame of reference.

After each data set was made periodic, it was normalized by its average. This was done to generate a magnitude ratio of 1.0 for the DC frequency component. The reason this was done was so that the generated frequency response function could then be applied to any inlet and exit parameters with a scaling variable k . This idea is defined by Equation 6.8.

$$Out(\omega) = FRF(\omega) \cdot k \cdot In(\omega) \quad \text{Equation 6. 8}$$

After each data set was normalized it was then inserted into the FRF FORTRAN code and the FRF magnitude ratio and phase lag for each frequency were calculated. Since Boller only measured matching data points every 30° circumferentially the time series only had 13 discrete points which resulted in seven discrete frequency points, based on the Nyquist frequency, including the zero frequency or DC point. This resolution only allowed the first six harmonics to be examined. This poor resolution does not allow for an accurate analysis of the dynamic blade response.

Both the wake semi-width and exit total pressure measurements were taken in the blade wake field. It was reasoned that the dynamic wake response is related to the dynamic lift response. The reason is that the boundary layer dynamic response is related to the blade dynamic lift response. As the incidence angle increases, creating more lift, the boundary layer thickness will grow. The relationship between the boundary layer thickness and the lift allows for the connection between the wake width and lift, since the wake width is an effect of the boundary layer thickness. Measuring the dynamic wake response is simpler than taking on the blade measurements and reduces the time and complexity needed to experimentally investigate the dynamic blade response.

6.2.1 FRFs generated from Angle of Attack and Wake Semi-width

The frequency response functions generated using angle of attack as the forcing function with wake semi-width as the response function will be discussed first. The inlet and exit pressure based FRFs will be discussed after, and then the two will be compared to see if there is any correlation between them. The (wake semi-width / angle of attack) result for Operating point A at midspan and with a level 1 distortion screen is shown in Figure 6.7 and Figure 6.8. The magnitude and phase for both the angle of attack and the wake semi-width are shown along with the magnitude ratio and phase lag of the frequency response function generated from the two parameters. The magnitudes of the parameter's Fourier series are plotted on semi-log graphs. From the plot of FRF magnitude, one can see that there is a higher magnitude ratio at the third and sixth harmonic. The magnitude ratio at the DC frequency is 1, which it should be because of the normalization. The graph of angle of attack compared to the wake semi-width in the time domain shows that the response (wake semi-width) is highly correlated with the forcing function and the response is not leading or lagging the forcing function in time. The graph shows a sudden drop in angle of attack and wake semi-width at the start of the screen, while there is increase in each parameter behind the screen. The cross correlation calculated for this data set gave a correlation coefficient of 0.9766, which means that the response is nearly 100% correlated with the forcing function. Graphs of all the FRFs (ww/aoa) created from each experiment are in Appendix B.

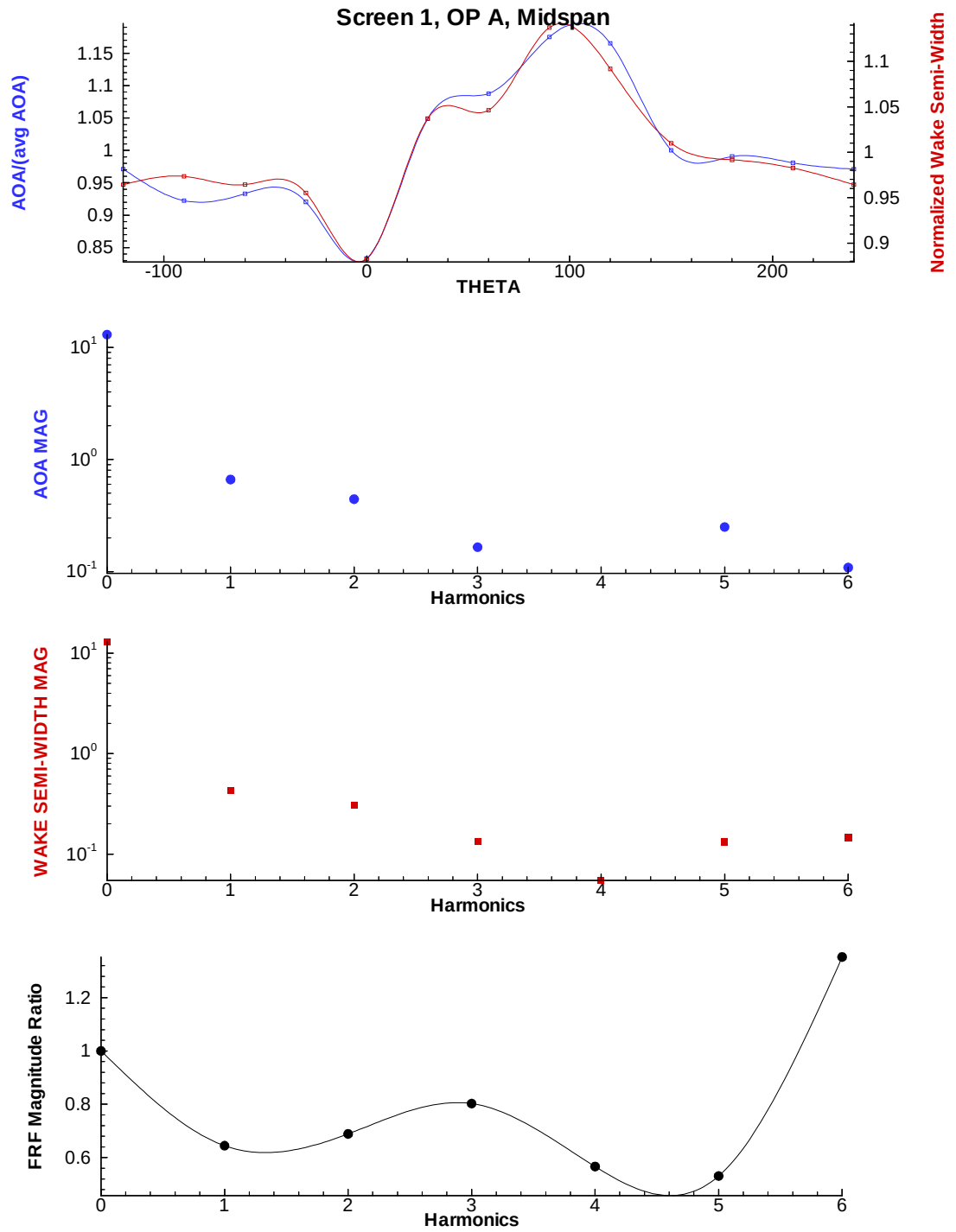


Figure 6.7: Magnitude Results from FRF code for Level 1 Screen at Operating Point A and Midspan

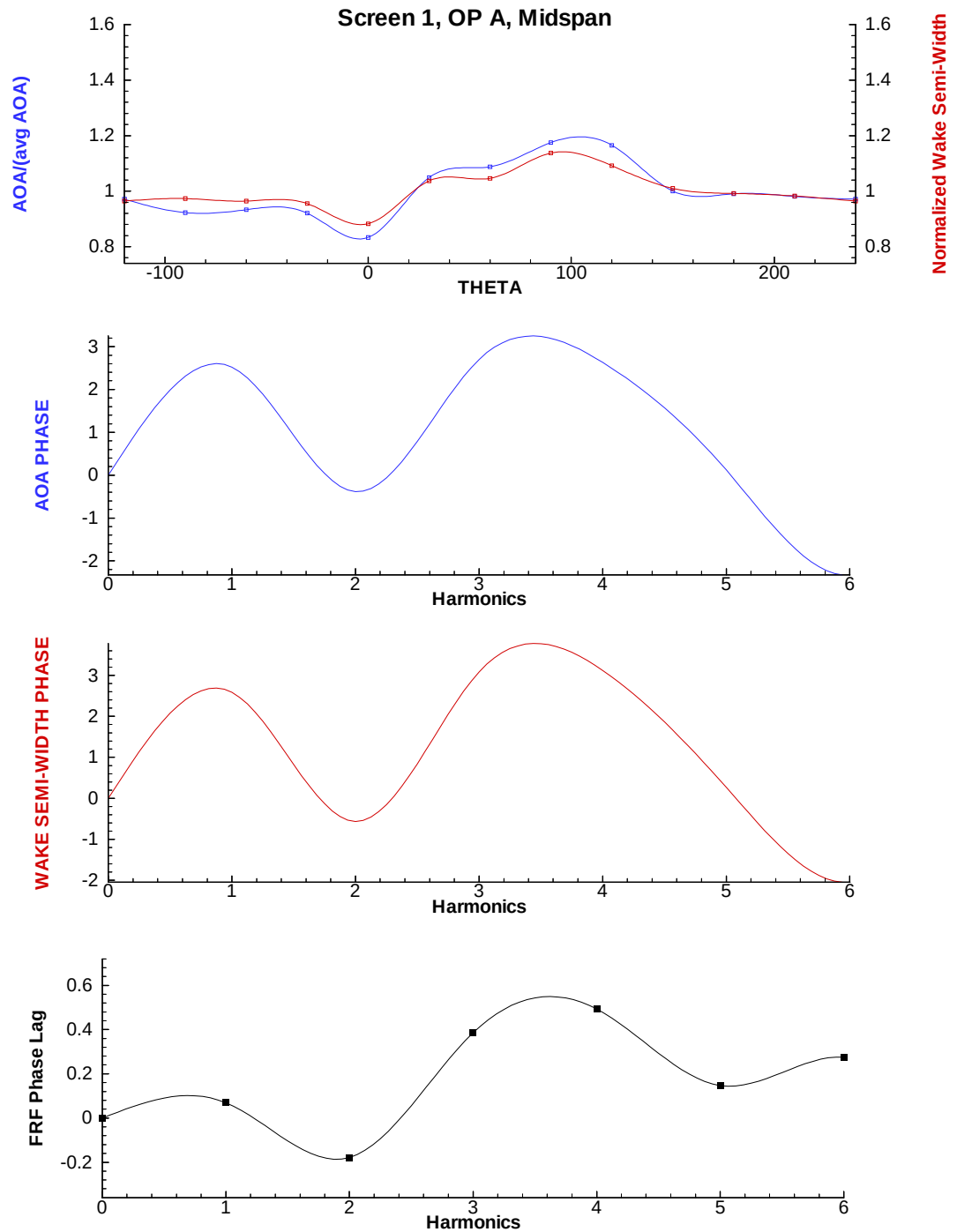


Figure 6.8: Phase Lag for Level 1 Screen at Midspan and Operating Point A

It was found that for most of the data sets the response leads the input; that is, the phase lag is positive. This is contradictory to what is expected. One would expect that the blade response would lag because of the time for the fluid particles to traverse the blade length allowing the entire blade to sense the new flow field information. Even if all of the information about the flow field traveled at the speed of sound, there would still be a miniscule lag in the response. This does hold true when the measurements are taken on the blade, but when the measurements are taken in front of and behind the blade in the flow field, the blade wake response appears to lead the input. The reason for this is as follows. It was discovered that the relative fluid flow travels the length of the blade faster than the amount of time it takes the trailing edge of the blade to reach the same related circumferential position that the inlet properties were measured at. Therefore, a leading response is seen in the absolute frame of reference. This lead can be as much as 15 degrees. In actuality, the blade response is lagged. Figure 6.9 shows this effect of different frames of reference.

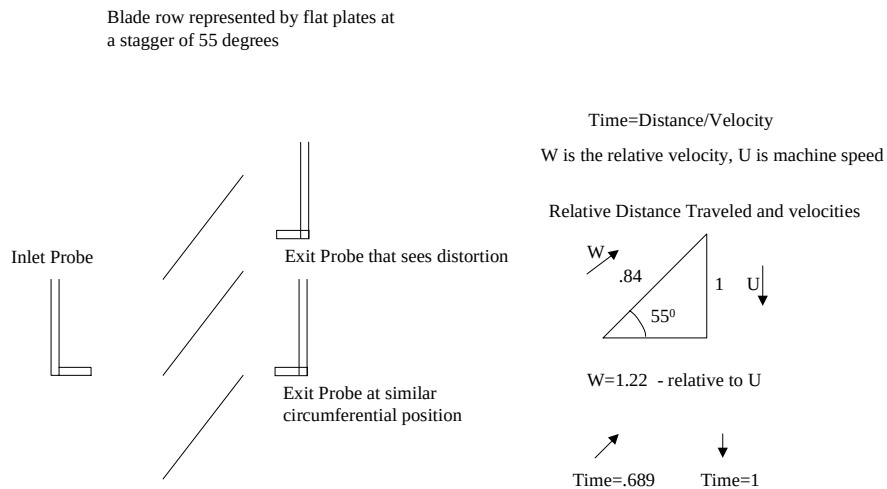


Figure 6.9: Reason for Leading Response in the Absolute Frame

Figure 6.10 shows two data sets that show leading wake response. The first data set is for a level 2 screen at operating point B and at mid-span. The parameters are angle of attack and wake semi-width. The response (wake semi-width) clearly leads before and after the distortion screen. The second data set is from a level 2 screen at operating point A and at 1/3 span. In this case, the response clearly leads the input at the end of the distorted region. The resolution of the 30 degrees between data points also affects how the data appears to respond. A small fluctuation in the response with respect to theta can result in a measured fluctuation of 30 degrees. For example, if the wake response leads by 16 degrees the data set will show a lead of 30 degrees. If instead the response leads by 14 degrees, the measured data would show a lead of 0 degrees.

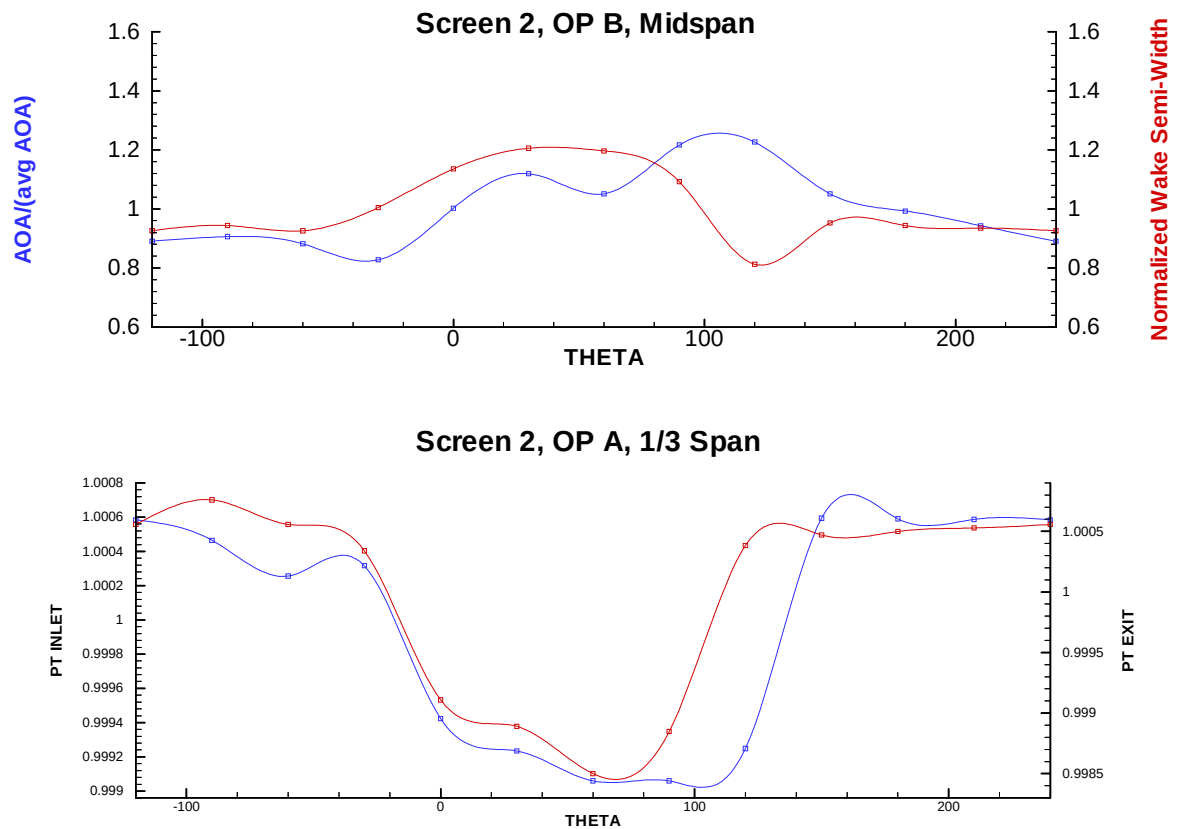


Figure 6.10: Data Sets Showing a Leading Response to the Input Parameter

The effect of distortion intensity on the blade wake response was then analyzed. The magnitude ratio and phase response at each operating point and at mid-span was compared with variations in distortion screen levels. At operating point A and B, only the effects of a level 1 and 2 screen are compared, while operating point C looks at the effects of all three screens. The following is the analysis of operating point A. Figure 6.11 shows the changes in the frequency response magnitude ratio and Figure 6.12 shows the changes in phase lag.

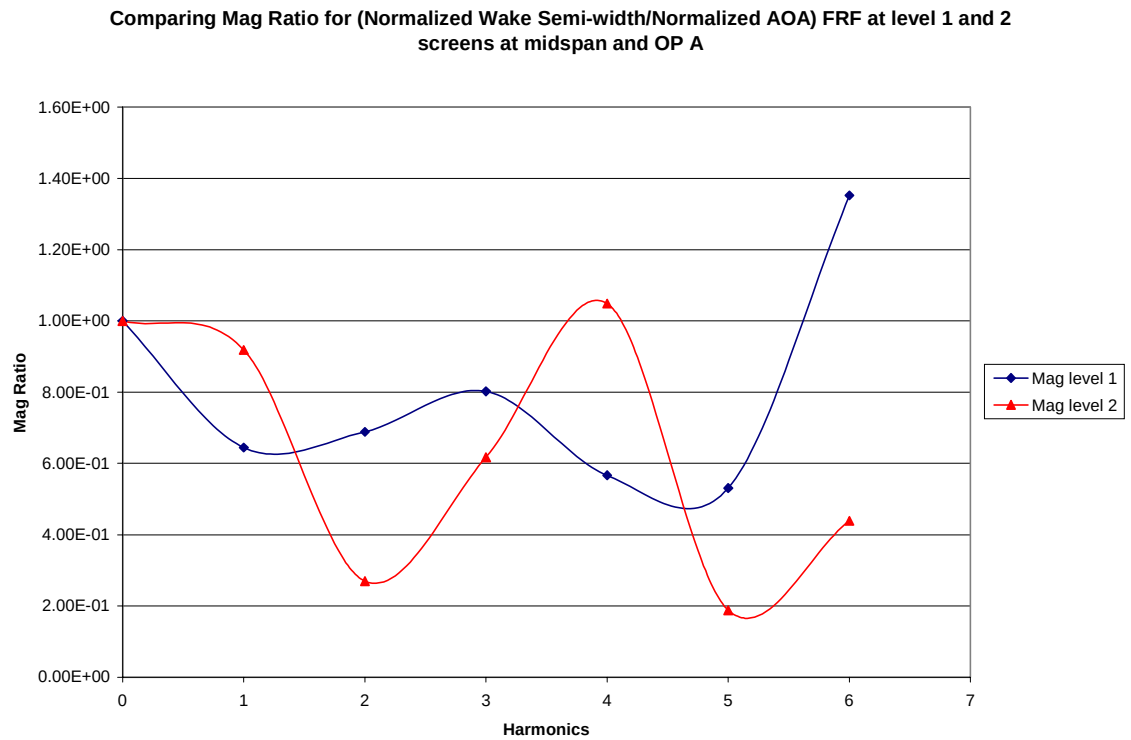


Figure 6.11: Comparing the Effect of Level 1 and 2 Screens on Magnitude Ratio

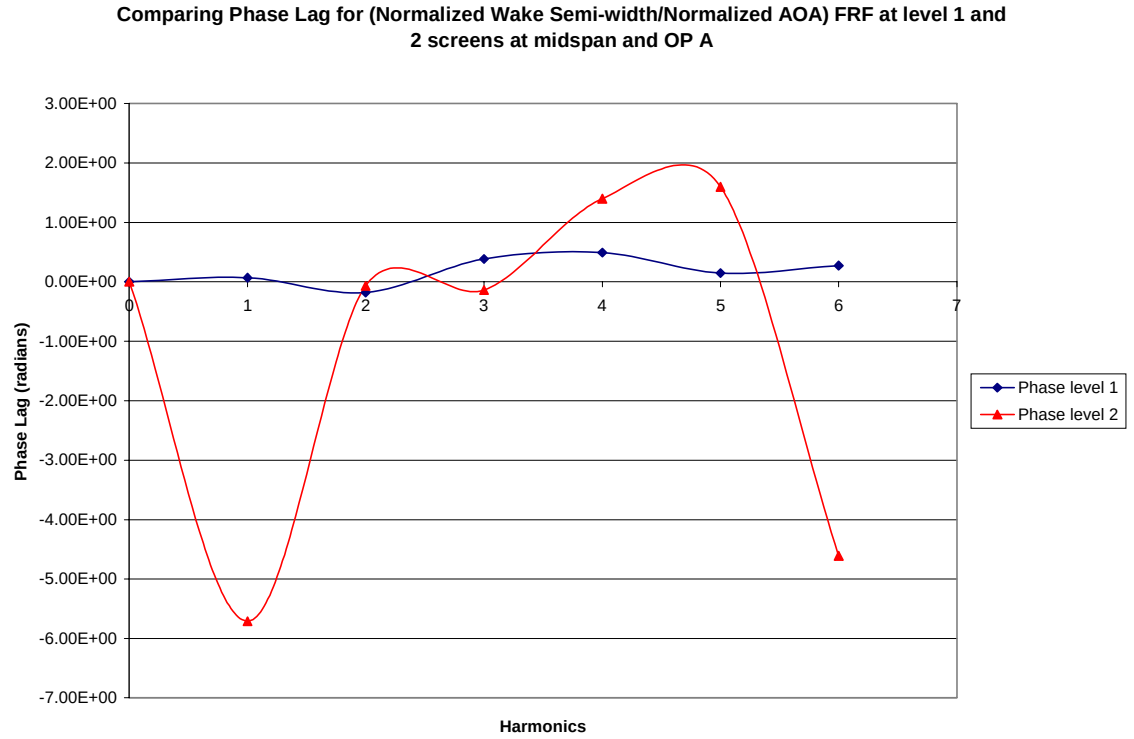


Figure 6.12: Comparing the Effects of Level 1 and 2 distortion screens on the Phase Lag

From Figure 6.11 and 6.12, it can be seen that the level of the distortion does have an effect on the frequency response of the blade wake. The level 2 screen response function has a high magnitude ratio at the fourth harmonic, while the level 1 screen has magnitude ratio peaks at the third and sixth harmonic. That means that the level 2 screen is causing a four-per-rev response while the level 1 screen is creating a three and six-per-rev response. The phase lag for the level 1 screen shows that there is little phase difference between the forcing function and the response function, while the level 2 screen shows that the response lags at the first and sixth harmonic and leads at the fourth and fifth harmonics. The correlation between the input and output may account for some of the difference in the Bode plots. The correlation coefficient for the level 1 screen is 0.9766, but the correlation coefficient for the level 2 screen is .6847. This means that the level 2 screen wake response is not as strongly dependent on the change in angle of attack. The lower correlation coefficient shows that another input parameter can be affecting the output. In this case, this could be noise in one or both the input and output signals, which is might have slightly corrupted the data set. If the noise does not account

for the entire difference, then the magnitude and phase plots of the frequency response function show that at operating point A and mid-span that the distortion intensity does have an effect on how the blade wake responses. It is assumed that the wake and non-wake regions behind the blade characterize the blade lift response. Therefore, the blade lift response will also differ depending on the strength of the distortion.

The level of distortion intensity at operating point B was analyzed to further check for any trends. Figure 6.13 shows the magnitude ratio for level 1 and 2 screens at mid-span and operating point B. Figure 6.14 shows the corresponding phase lags.

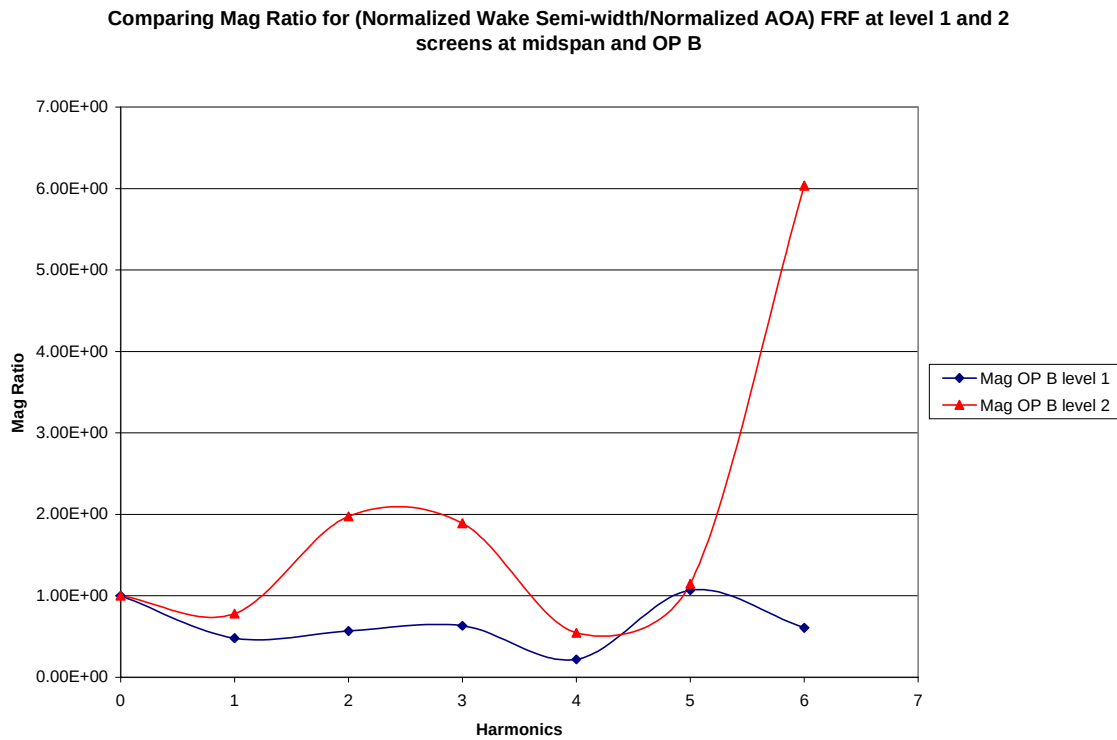


Figure 6.13: Comparing the Effects of Distortion Intensity on Magnitude Ratio at Operating Point B and Mid-span

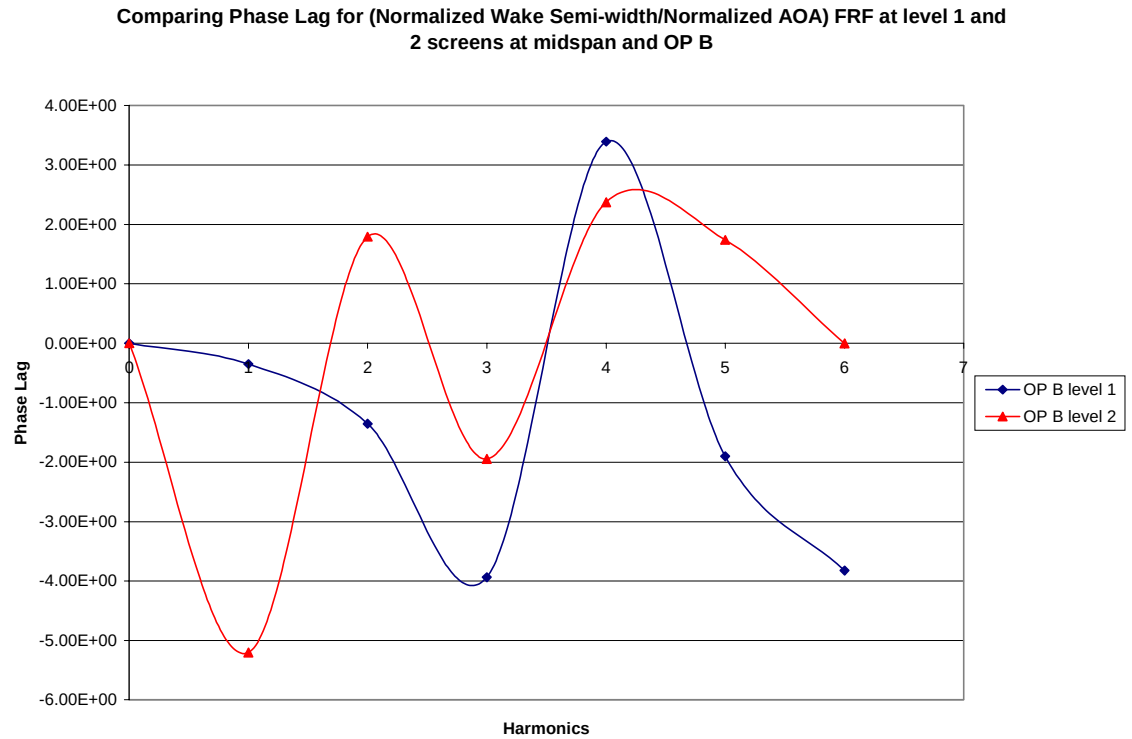


Figure 6.14: Comparing the Effects of Distortion Intensity on Phase Lag at Operating Point B

In the magnitude ratio plot the two curves seem a little more closely related at operating point B than at A. The magnitude ratio for the level 2 screen is higher across the frequencies, which would be expected, since a stronger distortion should lead to a greater wake response. The phase lag plot shows no correlation between the two screen levels, except that both lag at the third harmonic (three per rev) and lead at the fourth harmonic. It is noted that the correlation coefficient for the level 1 screen is 0.6887 and the correlation coefficient for the level 2 screen is 0.2359. The low coefficient for the level 2 screen shows that this data set may not be useful in determining the effects of distortion intensity.

The final case study on the effect of distortion intensity at mid-span was done at operating point C. Figure 6.15 shows the magnitude ratios with level 1,2, and 3 screens. Figure 6.16 shows the corresponding phase lags.

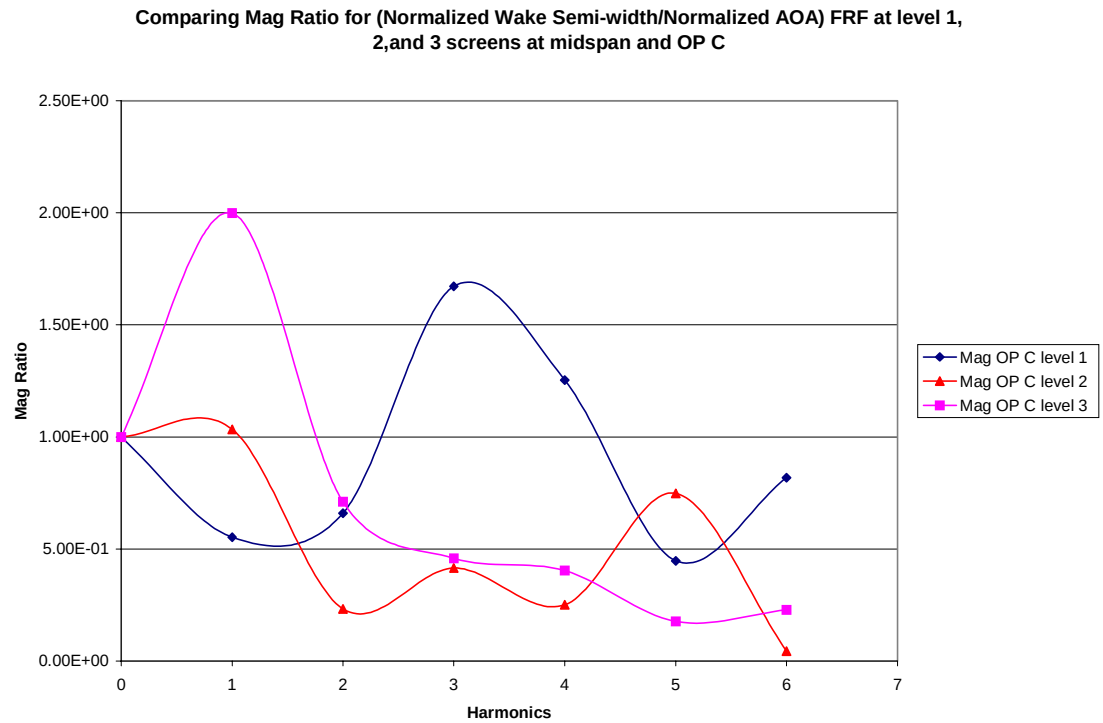


Figure 6.15: Comparing the Effects of Distortion Intensity on Magnitude Ratio for Operating Point C

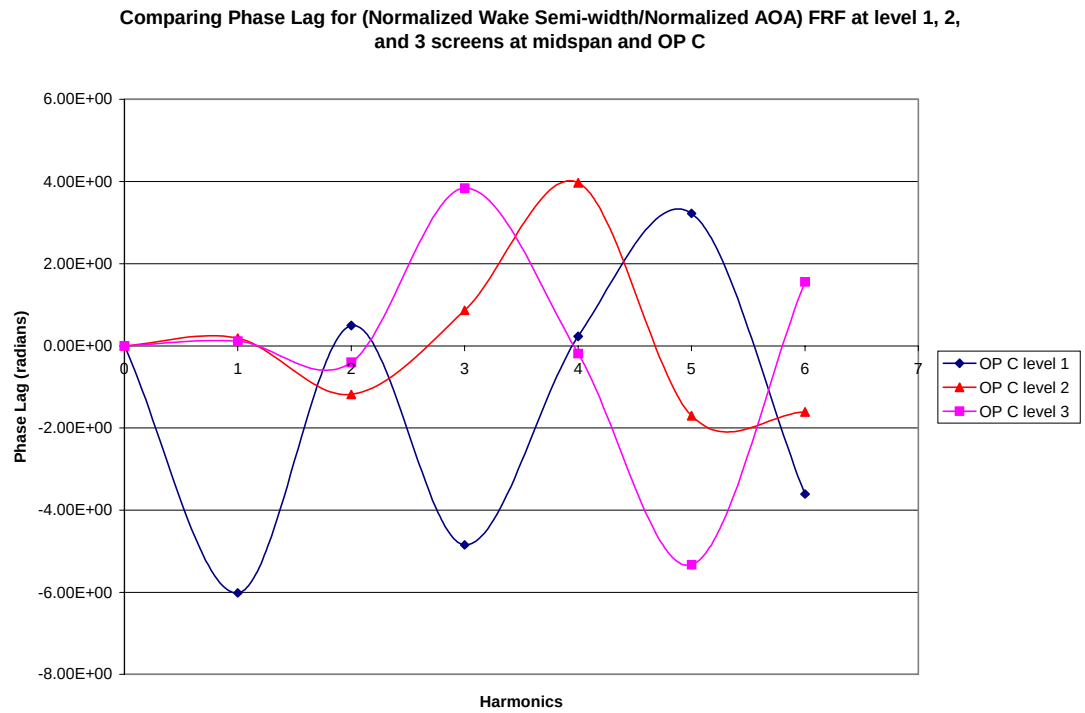


Figure 6.16: Comparing the Effects of Distortion Intensity on Phase Lag for Operating Point C

The magnitude ratios seem to have no common trends. The level 3 screen has a peak magnitude ratio at the first harmonic, showing a strong one per rev influence on the wake response. The level 1 screen has a high magnitude ratio at the third harmonic, suggesting a three per rev influence. The phase lag plot show that the level 2 and level 3 screens seem to have similar curves, with the level two screen curve offset to higher frequencies. These data sets had correlation coefficients of 0.6697 for level 1, 0.6976 for level 2, and 0.7086 for level 3. These plots show that the distortion intensity does seem to have an effect on the dynamic blade wake response.

The frequency response functions generated at 1/3 span and operating point B for level 1 and 2 screens was further looked at because of higher correlation coefficients of .9201 and .7814 for level 1 and level 2, respectively. Figure 6.17 and 6.18 show the magnitude ratios and phase lags, respectively.

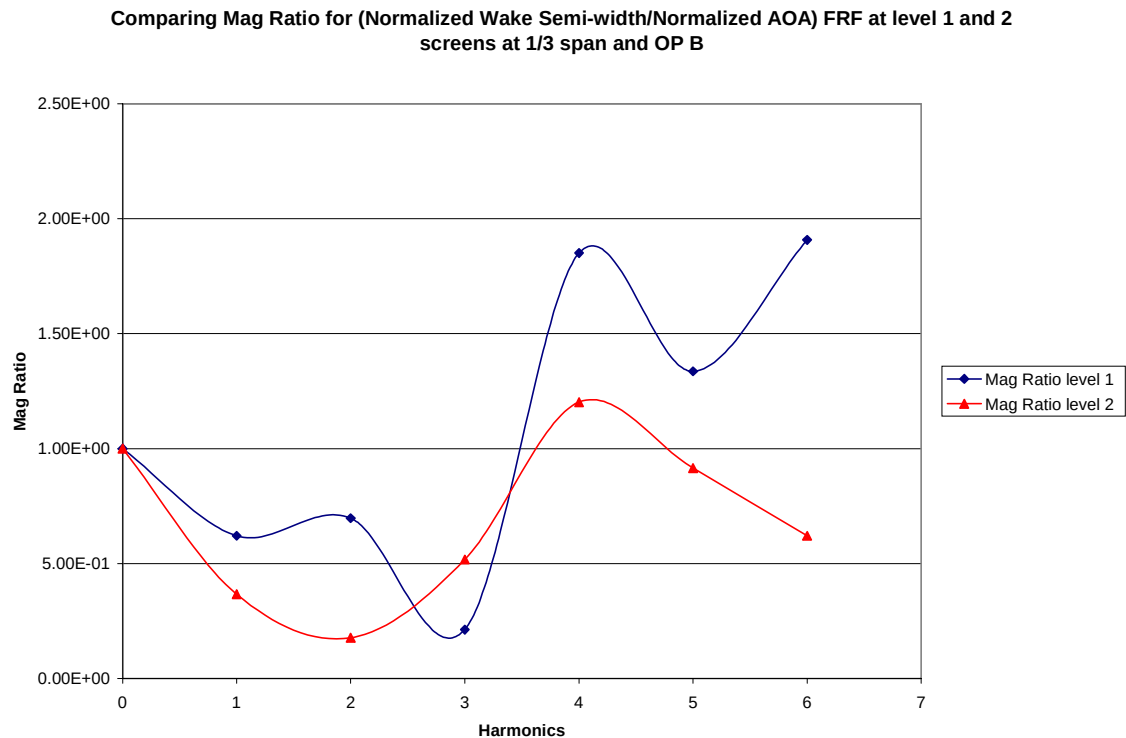


Figure 6.17: Comparing the Effects of Distortion Intensity on Magnitude Ratio at Operating Point B and 1/3 span

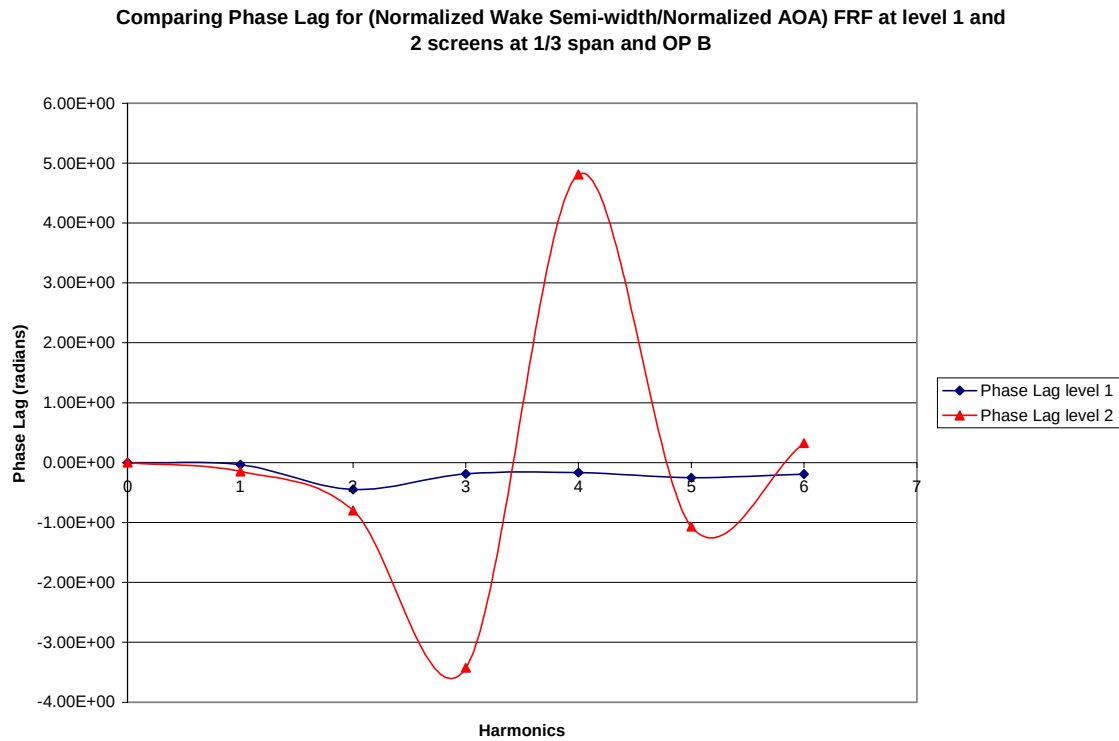


Figure 6.18: Comparing the Effects of Distortion Intensity on Phase Lag for Operating Point B at 1/3 Span

These last two figures further show that the level of distortion does affect the blade wake response, and therefore it affects the blade pressure rise response.

The effect of aerodynamic blade loading was another parameter that was varied to determine the influence it had on dynamic blade wake response. The data sets for operating points A, B, and C at mid-span for a level 1 screen were used to create FRFs that were then compared with one another. Figures 6.19 and 6.20 show the influence aerodynamic loading has on the blade wake response magnitude ratio and phase lag, respectively.

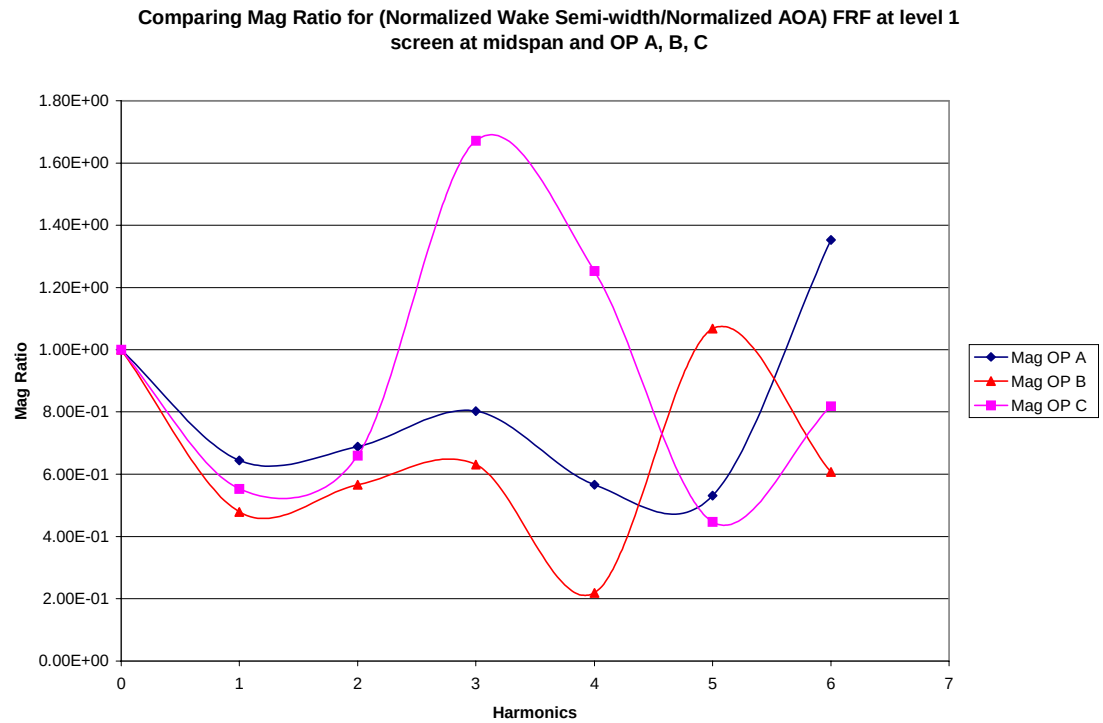


Figure 6.19: Comparing the Effects of Aerodynamic Loading on Magnitude Ratio for Level 1 Screen at midspan

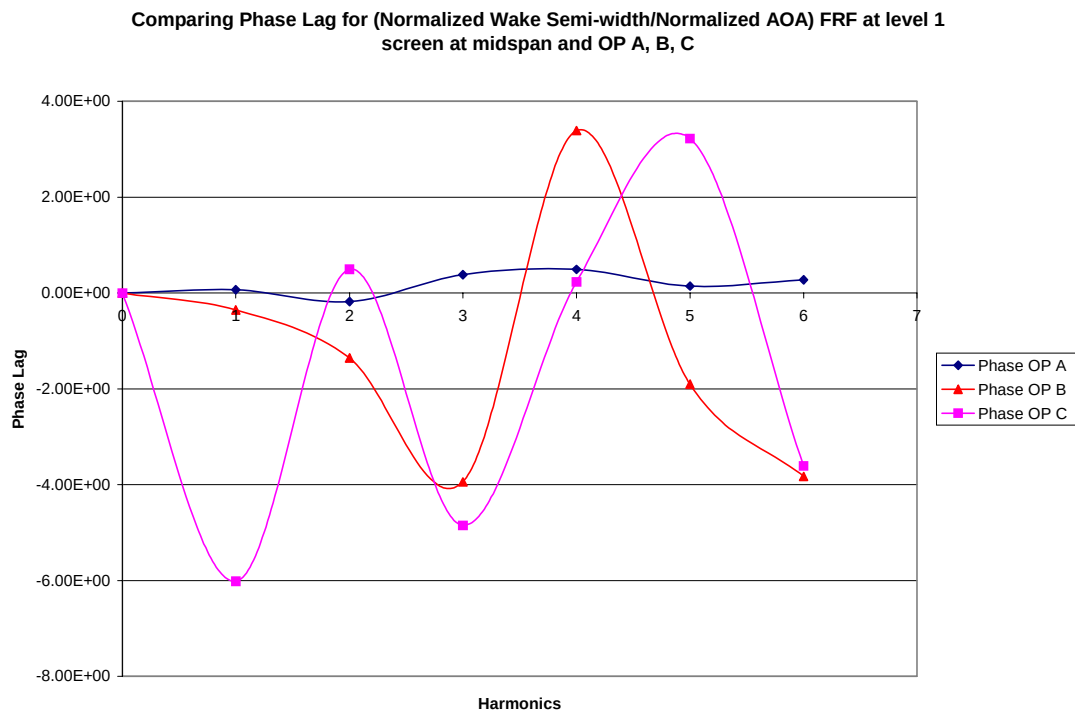


Figure 6.20: Comparing the Effect of Aerodynamic Loading on Phase Lag for Level 1 Screen at midspan

There is considerable change in the magnitude ratio and phase lag with changing aerodynamic loading. With less aerodynamic loading the magnitude ratio and phase lag increase in intensity at each frequency. The harmonic at which the peak magnitude ratio occurred also changed as the aerodynamic loading is decreased. The FRFs created for the level 2 screen and mid-span for various loading levels also showed change in the magnitude ratio and phase lag, but the magnitude ratios did not show the same trend of increasing in intensity as the level 1 screen magnitude ratios. Both showed that the aerodynamic loading does have an effect on the frequency response function of the blade wake, but there seems to be no relationship between loading level and response.

Blade span position effects were also investigated for the effect on the blade wake response. Because of blade twist, end wall boundary layer effects, leakage, and changes in turning angle it is assumed that there is a change in blade wake response at each span position. Figures 6.21 and 6.22 show the magnitude ratio and phase lag, respectively, for level 1 screen at operating point B. The reason for this choice of data sets was because of the higher correlation coefficients shared at each position. The coefficient was 0.6887 for mid-span, 0.9201 for 1/3 span, and 0.6410 for 2/3 span.

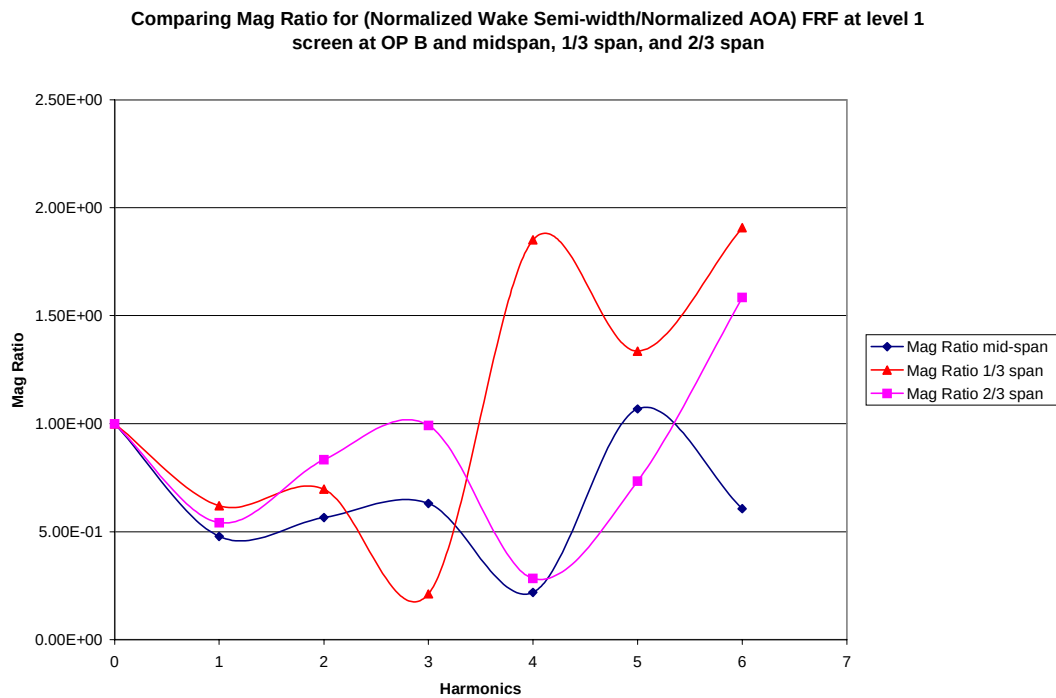


Figure 6.21: Comparing the Effect of Span Position on Magnitude Ratio

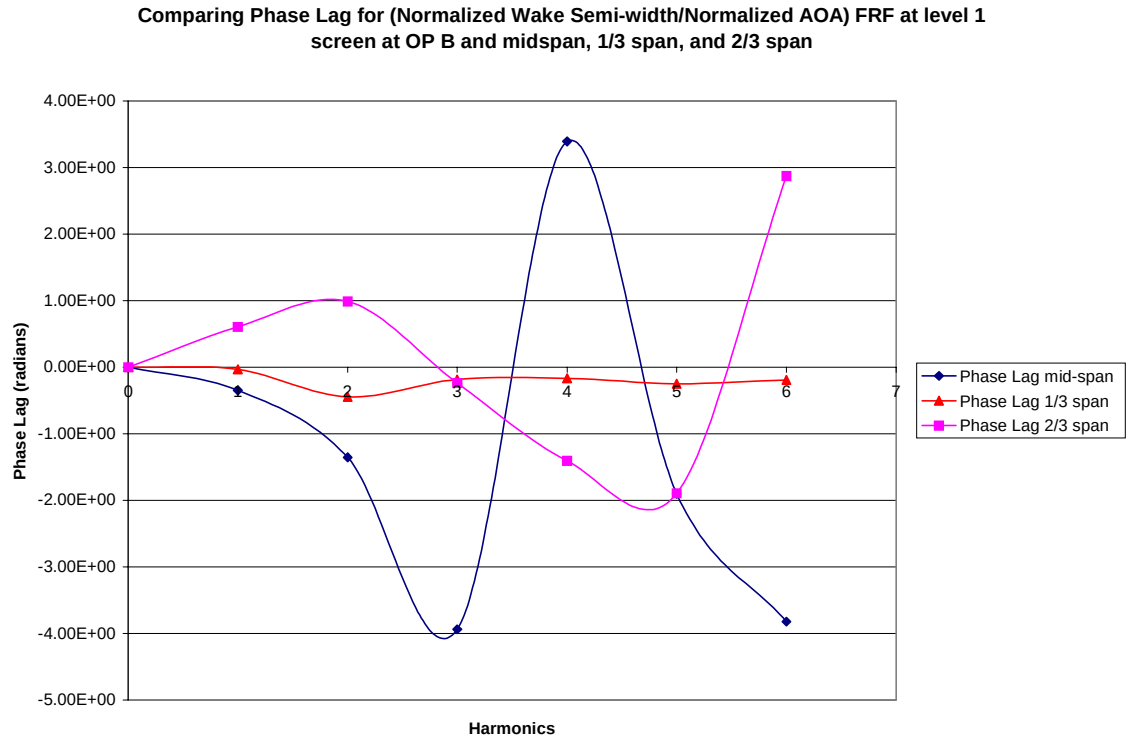


Figure 6.22: Comparing the Effects of Span Position on Phase Lag

In general, it appears that span position does have an effect on the frequency response function. This was expected because of the changes in blade geometry and flow conditions at different span positions along the blade.

Once the frequency response functions for each distortion level, aerodynamic loading level, and span position were calculated, the next step was to test the FRFs ability to calculate reasonable blade responses to different distortion patterns. The distortion that was simulated was that of a perfect distortion screen, which would generate a square wave. Because of the resolution of the test data, the simulated wave had to be sampled at the same circumferential positions. This was done so that the forcing function and response function had the same number of harmonics at the same frequencies. Because of this poor resolution, the input square wave appears to have a linear increase instead of a step increase at the beginning and end of the screen. The actual data set has a step change increase. Figure 6.23 shows the calculated response based on a square wave input for a FRF created using the response at level 1 screen, operating point A, and midspan.

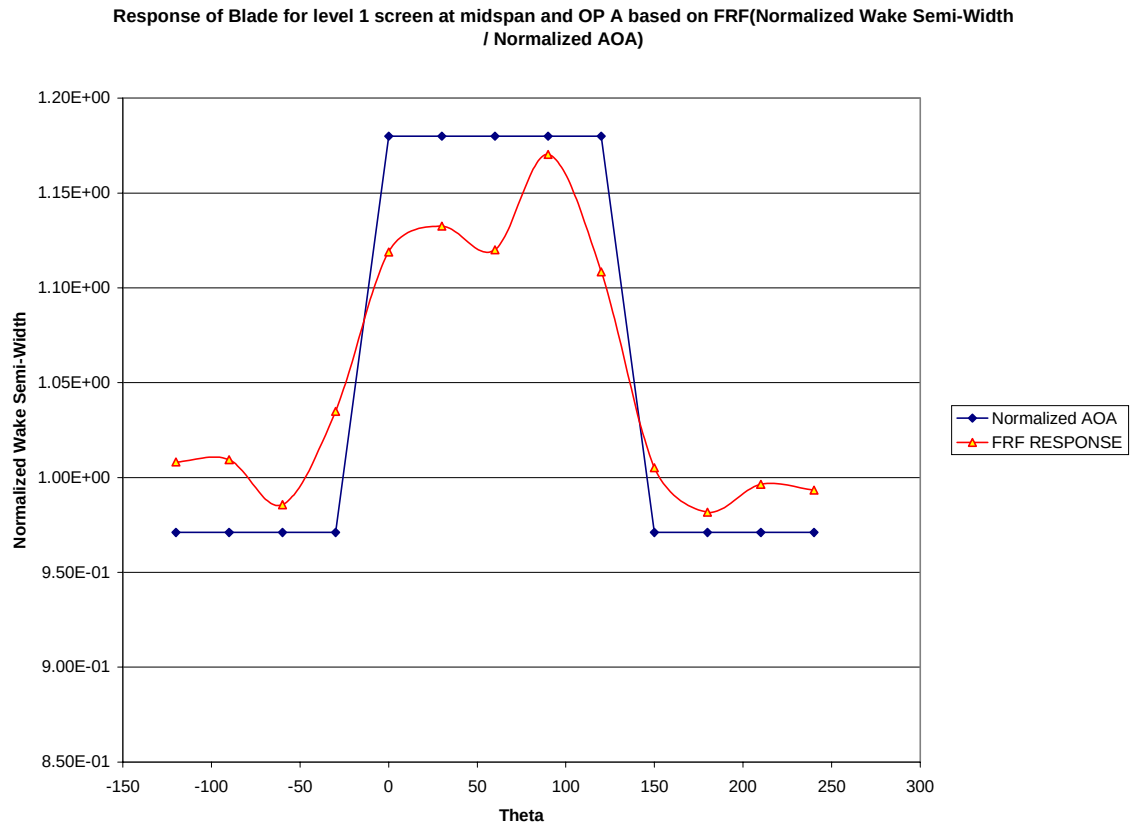


Figure 6.23: Blade Wake Response to a Square Wave

This response seems reasonable within the accuracy of the data. The response is definitely different than that of a first order response, which is generally used in dynamic models. This figure shows that this new method can accurately predict the blade response to any periodic or transient distortion pattern.

6.2.2 FRFs generated from Exit Total Pressure and Inlet Total Pressure

The total pressure parameters were used to generate frequency response functions for each test data set. These FRFs are felt to be more accurate in their representation of the actual blade dynamic lift response to a distortion. The reason for this is the higher correlation coefficients for the total pressure parameters. The normalized exit total absolute pressure is highly correlated with the normalized inlet total absolute pressure for each data set. The correlation coefficients were generally .8 or higher except for a couple of cases. The correlation coefficients for each test run are given in appendix A. This high degree of correlation shows that the exit total pressure is dependent on the inlet total pressure conditions and that there is little noise in the measurements. Since the pressures are so highly dependent, they should allow for a better representation of the dynamic response.

Since these parameters have higher correlation it was decided to reexamine the effects certain loading and flow parameters have on the frequency response functions. The first parameter examined was distortion intensity. The test sets compared were the level 1 and 2 screens at operating point A and midspan. It is noted that for the level 1 screen, the correlation coefficient is .9008, and for the level 2 screen the coefficient is .7269. Figure 6.24 shows the magnitude ratios and Figure 6.25 shows the phase lag of the two FRFs. In Figure 6.24, it is noted that the magnitude ratios for the first harmonic are nearly identical, and that both FRFs have a peak magnitude ratio at the third harmonic. With regard to magnitude ratios for the total pressure FRFs, when operating at near stall conditions the change in distortion intensity seems to have less effect on the dynamic blade wake response. The phase lag plot does show larger differences except at the first and fourth harmonic. Because of these differences in phase lag at each frequency, the FRF for level 1 screen will generate a different response than the FRF for the level 2 screen. Figure 6.25 shows the difference in predicted results of the two FRFs when the input of the level 2 screen data set was used as the forcing function for the FRF based on the level 1 screen. Exit PT is for the prediction of the level 2 screen FRF since the FRF is almost 100% accurate in predicting the actual response. The FRF Response stands for the level 1 screen FRF prediction. The level 1 FRF almost predicts the response at the leading edge of the screen, but misses the response at the trailing edge and

also induces noise during the non-distortion regions. This shows that the effect of distortion intensity still has an effect on dynamic blade wake response, but it is unsure if it has this large an effect on the actual blade lift response.

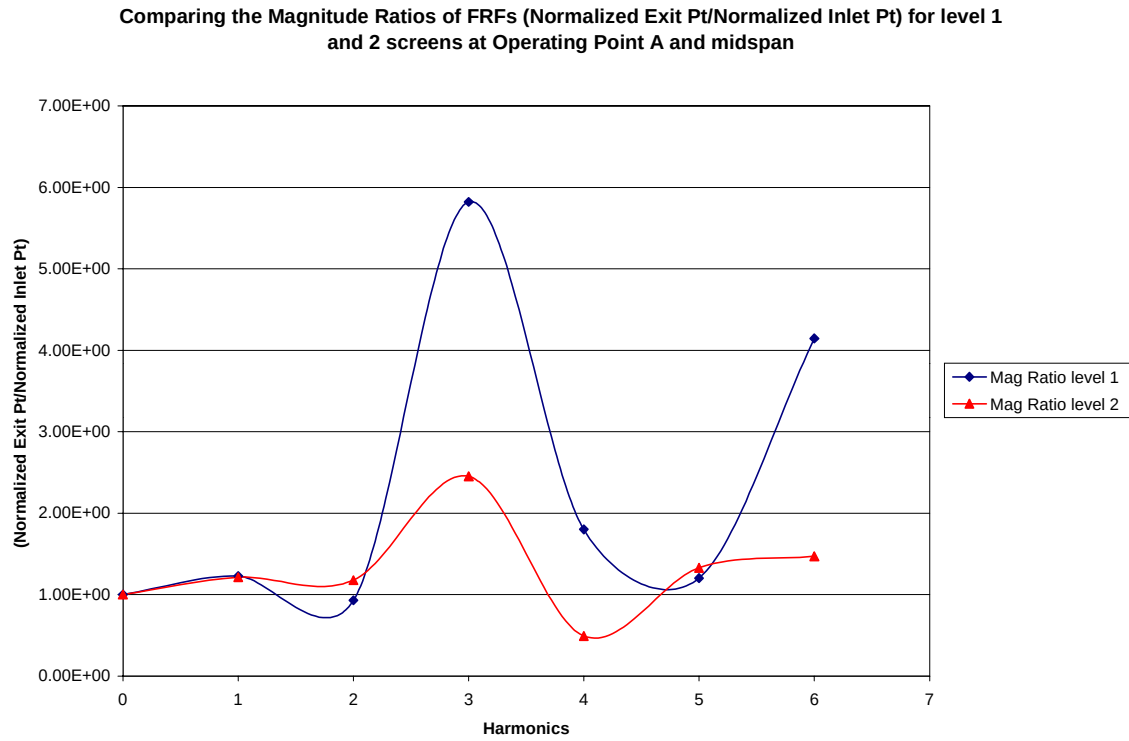


Figure 6.24: Comparing the Effects of Distortion Intensity on Magnitude Ratio for FRFs generated using total pressures

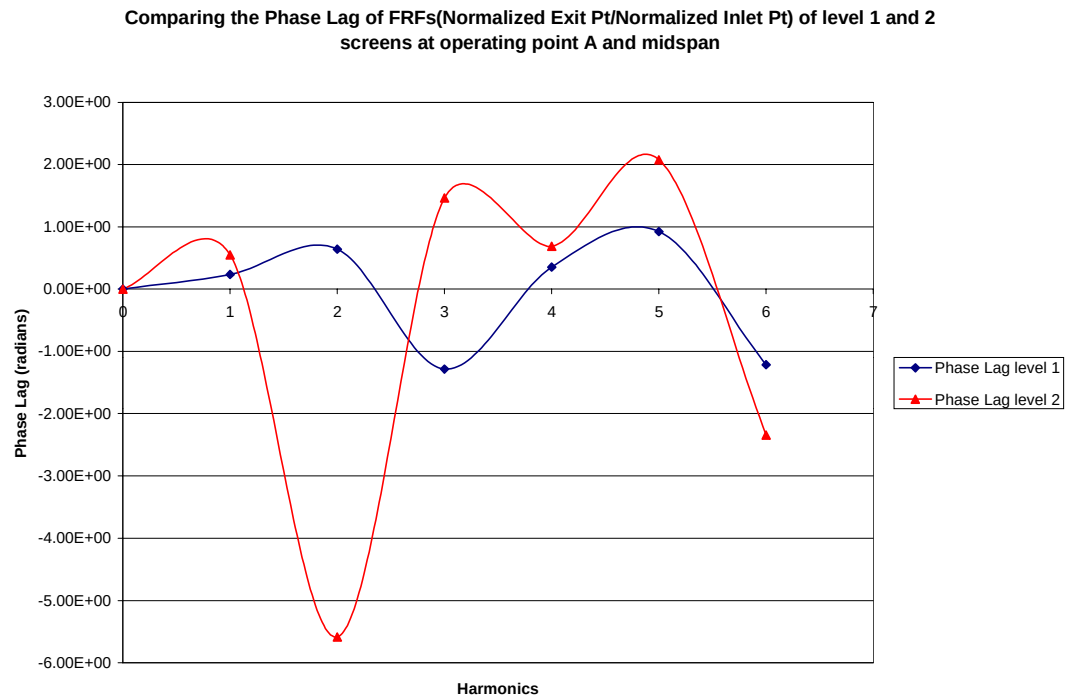


Figure 6.25: Comparing the Effects of Distortion Intensity on Phase lag of FRFs generated using total pressure

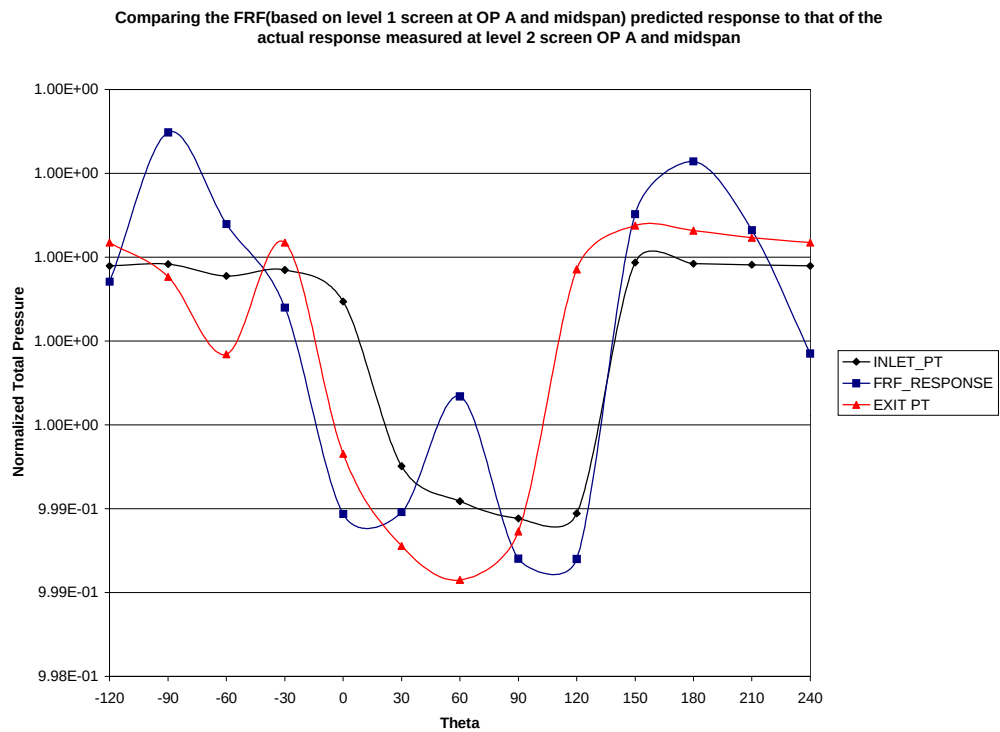


Figure 6.26: Comparing Responses to an Inlet Distortion of two FRFs based on two Different Levels of Distortion

To further check the effect of distortion intensity on the total pressure based FRFs, operating point B level 1 and level 2 screens at midspan were compared to each other. The correlation coefficient for the level 1 screen was .9138, and the level 2 screen had a coefficient of .8084. The results are shown in Figures 6.27 and Figure 6.28, which show the magnitude ratio and phase lag, respectively.

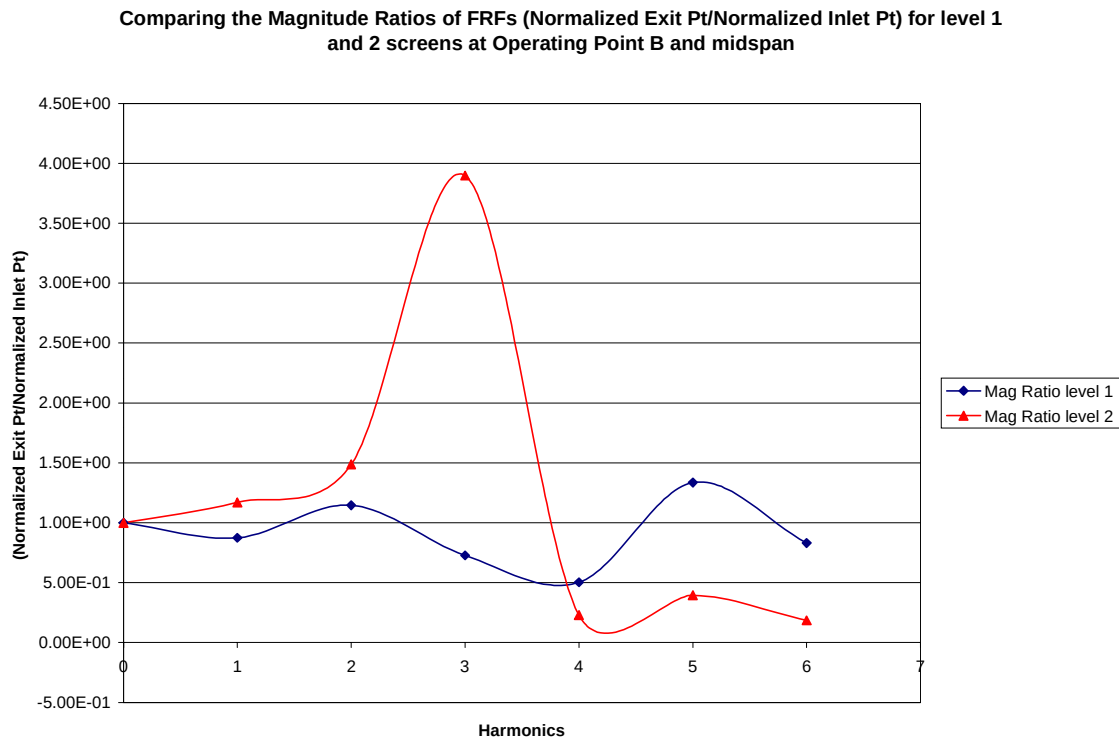


Figure 6.27: Comparing the Effects of Distortion Intensity on Magnitude Ratio for FRFs based on total pressure at Operating Point B

Comparing the Phase Lag of FRFs(Normalized Exit Pt/Normalized Inlet Pt) of level 1 and 2 screens at operating point B and midspan

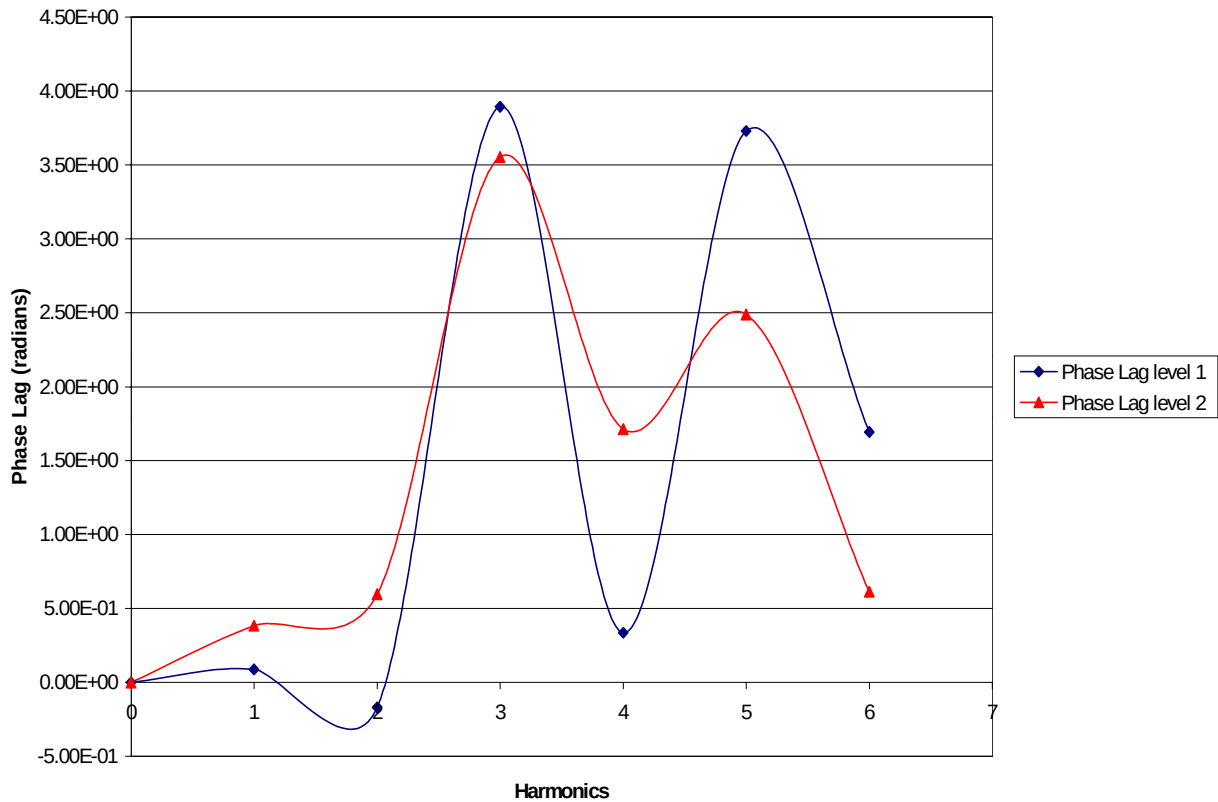


Figure 6.28: Comparing the Effect of Distortion Intensity on Phase Lag for FRFs based on Total Pressure at Operating Point B

There does not seem to be any similarities in the magnitude ratios other than they have close magnitude ratios at the first and second harmonic. At higher frequencies there is a noticeable difference in how the blade responses depending on the distortion intensity. The phase lags of the two FRFs are more alike in their lags at each frequency, but because of the differences in both magnitude and phase, the predicted dynamic wake response will be different for each FRF.

The effect of aerodynamic loading on the frequency response was examined next. The level 2 screen distortion at midspan was studied at three decreasing aerodynamic loading operating points. The correlation coefficients are .7269, .8048, and .7095 for operating point A, B, and C, respectively. Figure 6.29 and 6.30 show the magnitude ratio and phase lag, respectively. In the magnitude plot, it can be seen that at each

aerodynamic loading point there is a strong three per rev influence on the response with similar magnitude increases at the lower frequencies. The phase lag responses at each frequency differ greatly however. This shows that the effect of aerodynamic loading is evident in the frequency response functions. A further test using level 1 screen at midspan at each operating point was also conducted. The results of that comparison showed even greater differences in the magnitude and phase make-up of the FRFs.

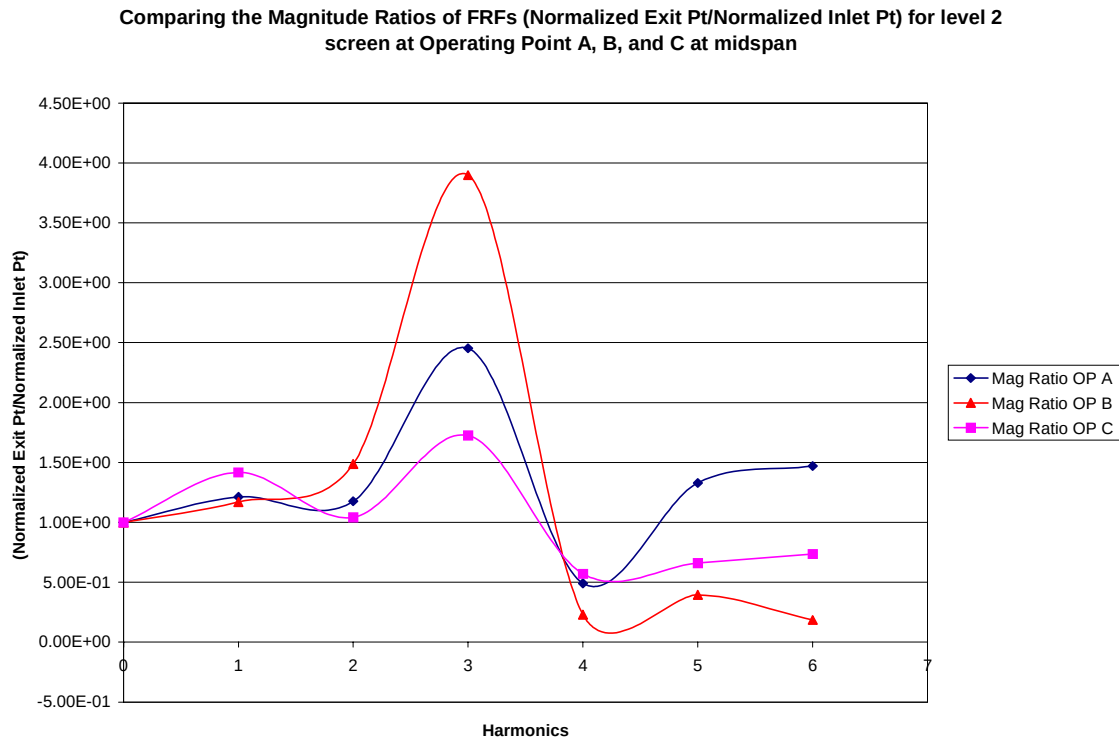


Figure 6.29: Comparing the Effects of Aerodynamic Loading on Magnitude Ratios of FRFs based on Total Pressure

Comparing the Phase Lag of FRFs(Normalized Exit Pt/Normalized Inlet Pt) of level 2 screen at operating point A, B, and C at midspan

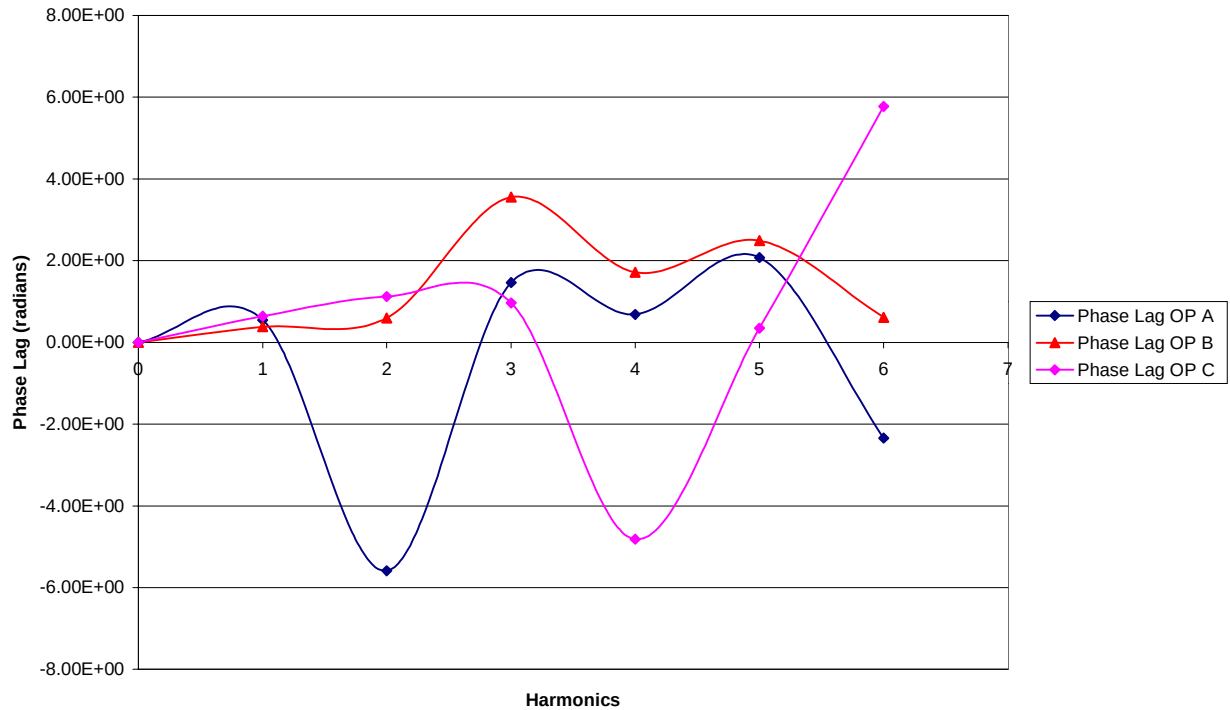


Figure 6.30: Comparing the Effects of Aerodynamic Loading on Phase Lag of FRFs based on Total Pressure

The final test variation that was examined was the effect that span position had on the frequency response of the blade at a certain distortion level and operating point. Figure 6.31 shows the magnitude ratios and Figure 6.32 shows the phase lags, which were examined. Figure 6.31 shows that the effect of span position has an effect on the level and frequency distribution of the magnitude ratio. The midspan position has a peak magnitude at the third harmonic, while the 1/3 span has a peak at the fourth harmonic. There does seem to be a large lag at the third harmonic for each span, but the similarity ends there.

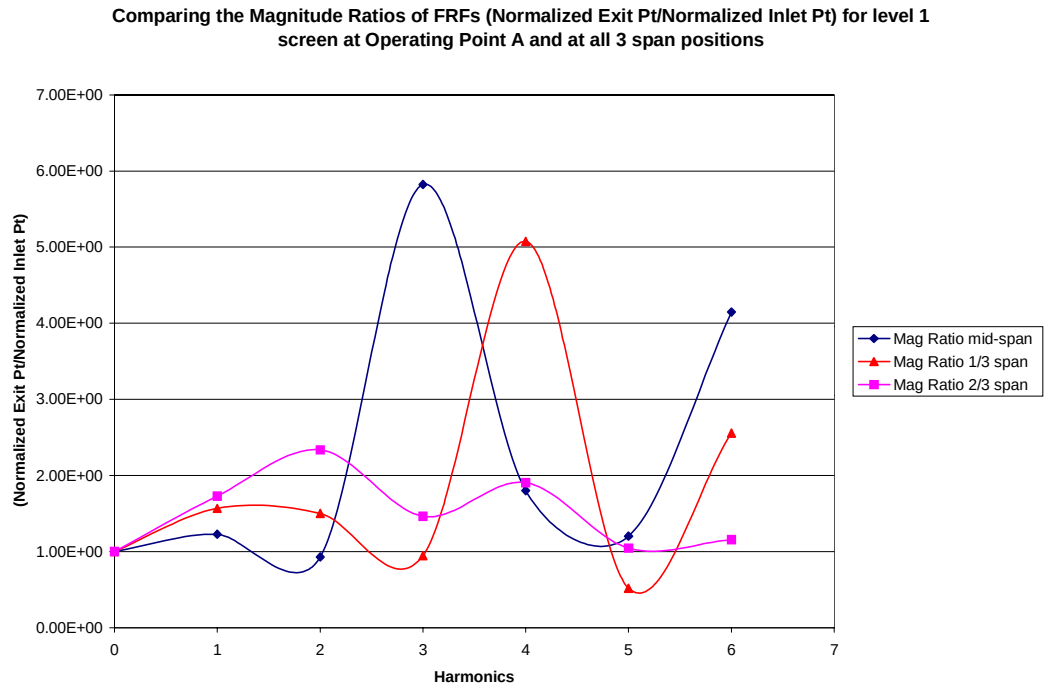


Figure 6.31: Comparing the Effects of Span Position on the Magnitude Ratio of FRFs based on Total Pressure

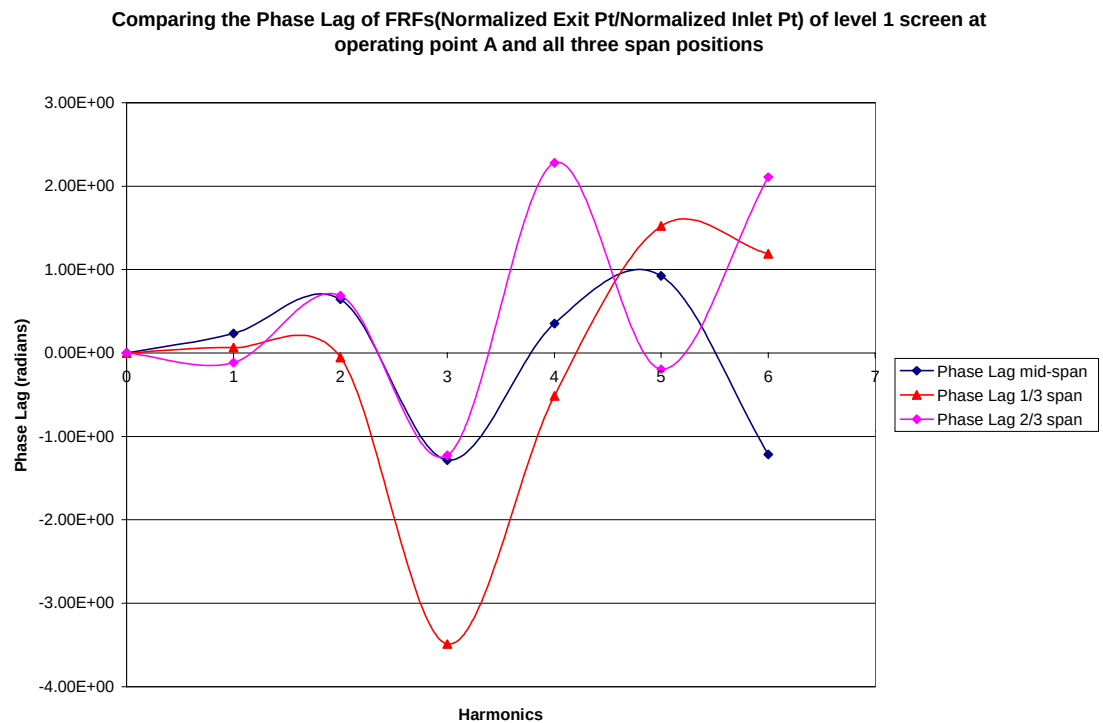


Figure 6.32: Comparing the Effects of Span Position on the Phase Lag of FRFs based on Total Pressure

Generally, the FRFs generated using the total pressures appeared to more accurately describe the dynamic blade response to changes in the inlet. A predicted blade responses to square wave distortions seemed to also be more representative of what the actual response would be to a wave of that nature. Figure 6.33 shows the blade response to a square wave based on the FRF level 2 screen at operating point A and at midspan. The figure clearly shows the response leading the input, which was the effect that Boller measured for this operating condition in the original experiment.

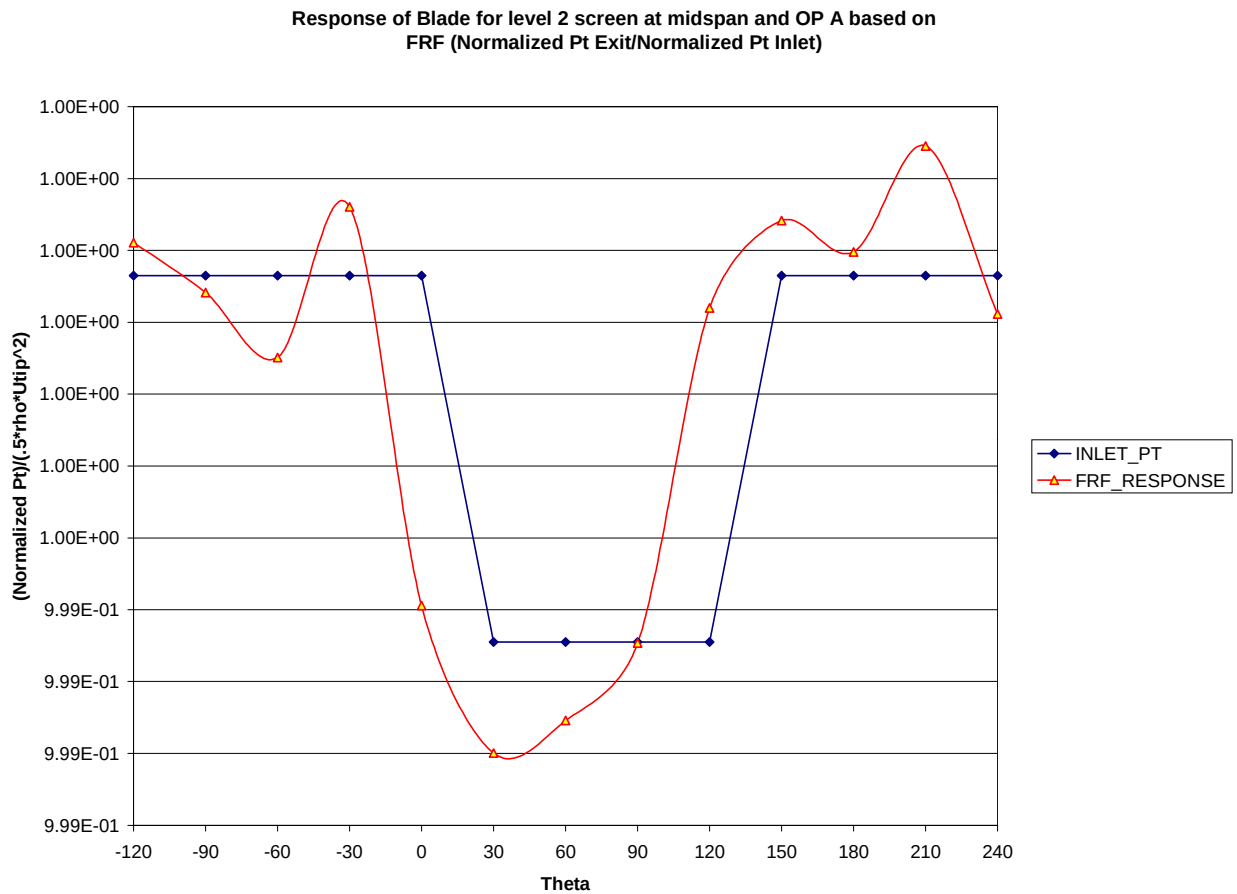


**Figure 6.33: Calculated Response to a square wave based on a level 2 screen, operating point A
midspan FRF**

Figure 6.34 shows another predicted response to the same square wave this time using an FRF based on a level 2 screen at operating point C and at midspan. Again, the predicted response leads the input, and this leading matches the test data set for this condition.

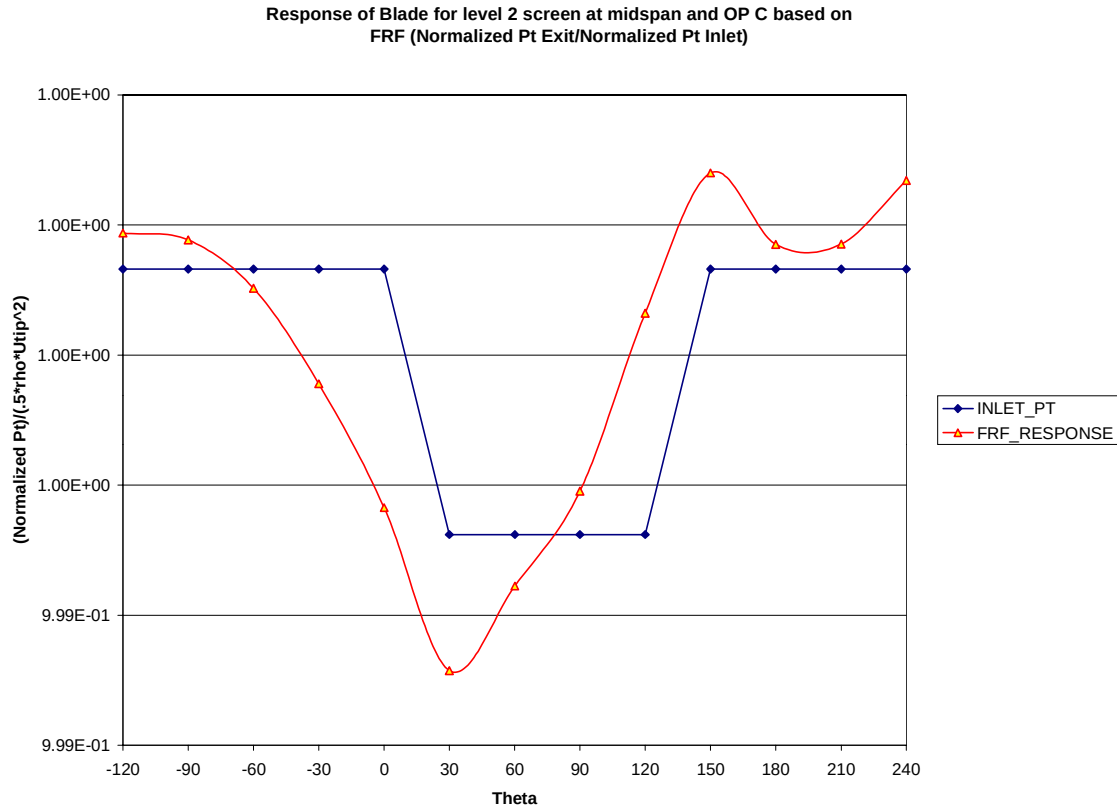


Figure 6.34: Predicted Response to a Square Wave Distortion based on an FRF Generated for a Level 2 Screen at Operating Point C and at Midspan

These plots and the others in Appendix C show that the FRFs generated from test data can predict the dynamic blade wake response to any type of distortion pattern to a certain degree of accuracy at a given operating condition and span position. It should be noted that at each operating condition the FRFs generated using total pressure differed from the ones generated using angle of attack and wake semi-width. This shows that the wake semi-width response is not correlated to the pressure response. It is felt that the FRFs generated using the total pressures are more representative of the blade lift response.

6.2.3 Graphical Representation of Dynamic Blade Response

There is the desire to graphically represent the dynamic response of a blade. This interpretation shows the data in a familiar form. With steady state analysis of compressors, the performance is shown on a compressor map with pressure ratio on the y-axis and mass flow rate on the x-axis, or some variation of these state properties. This map gives a clear picture of what the operating conditions are at a give point along a speed line. The map shows the entire operating regime with efficiency islands overlaid and surge/stall line included. It is desired to graphically show the dynamic response using this compressor map format along with an additional plot of total pressures vs. time. The total pressure is the variable that is changing due to total pressure losses through the distortion screen. The compressor characteristic map will show the path of the dynamic response with respect to pressure ratio and mass flow rate, while the time plot will show the lagged response with respect to time. These two plots will give the data that is needed to interpret the dynamic departure from the steady state.

Figure 6.35 shows these plots for a level 1 screen at operating point A and at midspan. This Figure uses pressure rise coefficient instead of pressure ratio for the compressor map y-axis since this is a low speed machine, and uses flow coefficient instead of mass flow rate on the x-axis. The starting point on the compressor map is labeled 1 on the characteristic map, and this corresponds to the -120° theta position on the time plot. At around point 3 the blade enters the distortion and a sudden drop in the pressure rise coefficient is seen. At point 8 the blade leaves the distorted region and the pressure rise coefficient returns to the non-distortion region on the map. The time plot shows the lag in the response related to the input total pressure.

By combining a compressor map and a pressure vs. time plot, the dynamic performance path with the corresponding lag is represented. Now the dynamic response path can be shown for any distortion allowing for a better understanding of how the compressor reaches its final operating point.

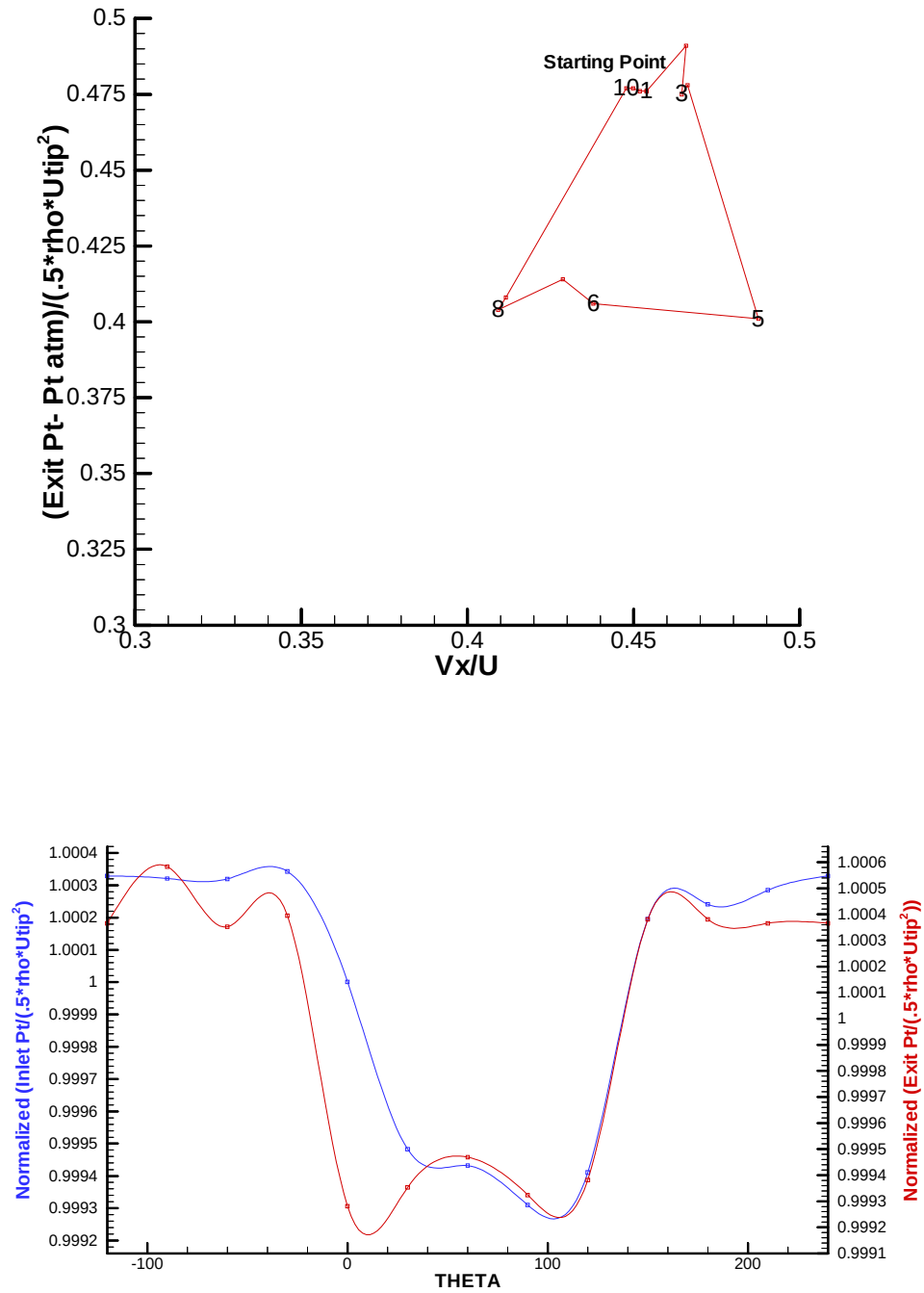


Figure 6.35: Graphical Representation of the Dynamic Response Using a Compressor Map and Time Plot

7 Conclusions

From the results of this investigation the following conclusions may be stated:

- 1) Dynamic flow response models for compressor rotor row flow can be developed using a frequency response function (FRF) method.
- 2) Dynamic blade flow response predictions based on the frequency response method can be more accurate than those based on a first order lag model. Higher order frequency response functions capture more of the dynamic content of the true response. They can be derived from experimental data, and the features of the response functions can be related to physical phenomena. The method developed in this thesis is presently associated with the assumption of linear response of the rotor blade flow to a forcing function, a limitation which should be removed by additional work. It is postulated that the linear method may be satisfactory for limited regions of the rotor flow regime.
- 3) The simple dynamic blade response model presented shows that blade row response should not depend on distortion intensity or frequency, but only on chord length. The results developed from the available experimental data do not confirm this hypothesis, but show large magnitude and phase response variations due to distortion level, average aerodynamic loading, and span position. In certain cases, encouraging similarities in response functions were found. The response of dynamic total pressure was found to be more highly correlated with inlet disturbance than the measured dynamic wake parameters.
- 4) The path of the dynamic compressor rotor row response can be represented on a conventional compressor map using the results from a frequency response function model.

8 Recommendations

The following recommendations are submitted with respect to future work to improve dynamic modeling methods for compressor stage characteristics:

- Perform a similar experiment to the one Boller performed on an isolated rotor except at a higher machine speed and with data sampled at closer spaced circumferential positions. The resolution of the distortion pattern needs to be greatly improved to allow for a more accurate frequency response function. The current FRFs are too coarse and do not contain all the frequency components that make up the distortion wave. The higher machine speed will test for variations in frequency response functions based on speed.
- Perform an experiment involving on blade high response pressure transducers along with a high response pressure transducer down stream of the blade to test the validity of the assumption that the blade wake response is related to the blade lift response.
- Perform an experiment with a more current compressor blade design to create a new set of FRFs based on a more complex blade. By using a new blade design and a higher resolution of the distortion wave, it may be possible to relate the dependence of the magnitude ratio and phase lag at each frequency to a geometric or flow field property. This would make the FRFs even more useful in preliminary design.

Continue to develop the FRF analysis method by investigating non-linear methods.

Find and develop CFD flow models to complement and extend the experiments.

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10 Appendix A

List of Cross Correlation Coefficients of the Boller test data parameters. WSW stands for wake semi-width and AOA stands for angle of attack.

(ex: s1a12 stands for level 1 screen at operating point A and at midspan)

Test Data Set	WSW/AOA	Pt Exit / Pt Inlet
S1A12	.9766	.9008
S1A13	.6749	.9375
S1A23	.3896	.8189
S1B12	.6887	.9138
S1B13	.9201	.7017
S1B23	.6410	.7712
S1C12	.6697	.5413
S1C13	.5405	.9003
S1C23	.8778	.9231
S2A12	.6847	.7269
S2A13	.7122	.8649
S2A23	.6961	.9403
S2B12	.2359	.8048
S2B13	.7814	.9604
S2B23	.1231	.8819
S2C12	.6976	.7095
S2C13	.8427	.9450
S2C23	.3509	.8418
S3C12	.7086	.9839
S3C13	.4840	.8715
S3C23	.5679	.9502

UNDISTORTED DATA

OPERATING POINT A	<i>33 % Span</i>	<i>50 % Span</i>	<i>67 % Span</i>
Angle of Attack	12.6	12.3	12.3
Pitch Angle	20.9	5.3	33.8
Exit Median Total Pressure (non-dim)	0.424	0.486	0.528
Wake Defect Relative Maximum (non-dim)	-0.204	-0.244	-0.241
Wake Semi-Width (non-dim)	0.083	0.089	0.086
Suction Side Jet Relative Maximum (non-dim)	0.114	0.102	0.104
Suction Side Jet Semi-Width (non-dim)	0.095	0.081	0.086
OPERATING POINT B	<i>33 % Span</i>	<i>50 % Span</i>	<i>67 % Span</i>
Angle of Attack	11.1	10.1	11
Pitch Angle	15.3	3.3	25.4
Exit Median Total Pressure (non-dim)	0.418	0.479	0.515
Wake Defect Relative Maximum (non-dim)	-0.191	-0.234	-0.228
Wake Semi-Width (non-dim)	0.063	0.085	0.081
Suction Side Jet Relative Maximum (non-dim)	0.11	0.096	0.105
Suction Side Jet Semi-Width (non-dim)	0.09	0.077	0.08
OPERATING POINT C	<i>33 % Span</i>	<i>50 % Span</i>	<i>67 % Span</i>
Angle of Attack	10.2	8	8.9
Pitch Angle	10.7	2	18
Exit Median Total Pressure (non-dim)	0.414	0.454	0.508
Wake Defect Relative Maximum (non-dim)	-0.185	-0.211	-0.211
Wake Semi-Width (non-dim)	0.054	0.079	0.081
Suction Side Jet Relative Maximum (non-dim)	0.11	0.1	0.088
Suction Side Jet Semi-Width (non-dim)	0.087	0.081	0.074

Figure A.1: Table of Inlet and Exit Flow Characteristics for Undistorted Rotor.

<u>DATA</u>			<u>LOCATION</u>
Distortion Screen	Operating Point	Measurement Span	Page Numbers
Level 1	Operating Point A	50 %, 33 %, 67 %	pp. 81-86
Level 1	Operating Point B	50 %, 33 %, 67 %	pp. 87-92
Level 1	Operating Point C	50 %, 33 %, 67 %	pp. 93-98
Level 2	Operating Point A	50 %, 33 %, 67 %	pp. 99-104
Level 2	Operating Point B	50 %, 33 %, 67 %	pp. 105-110
Level 2	Operating Point C	50 %, 33 %, 67 %	pp. 111-116
Level 3	Operating Point D	50 %, 33 %, 67 %	pp. 117-122

Figure A.2: Table of Locations for Distorted Data.

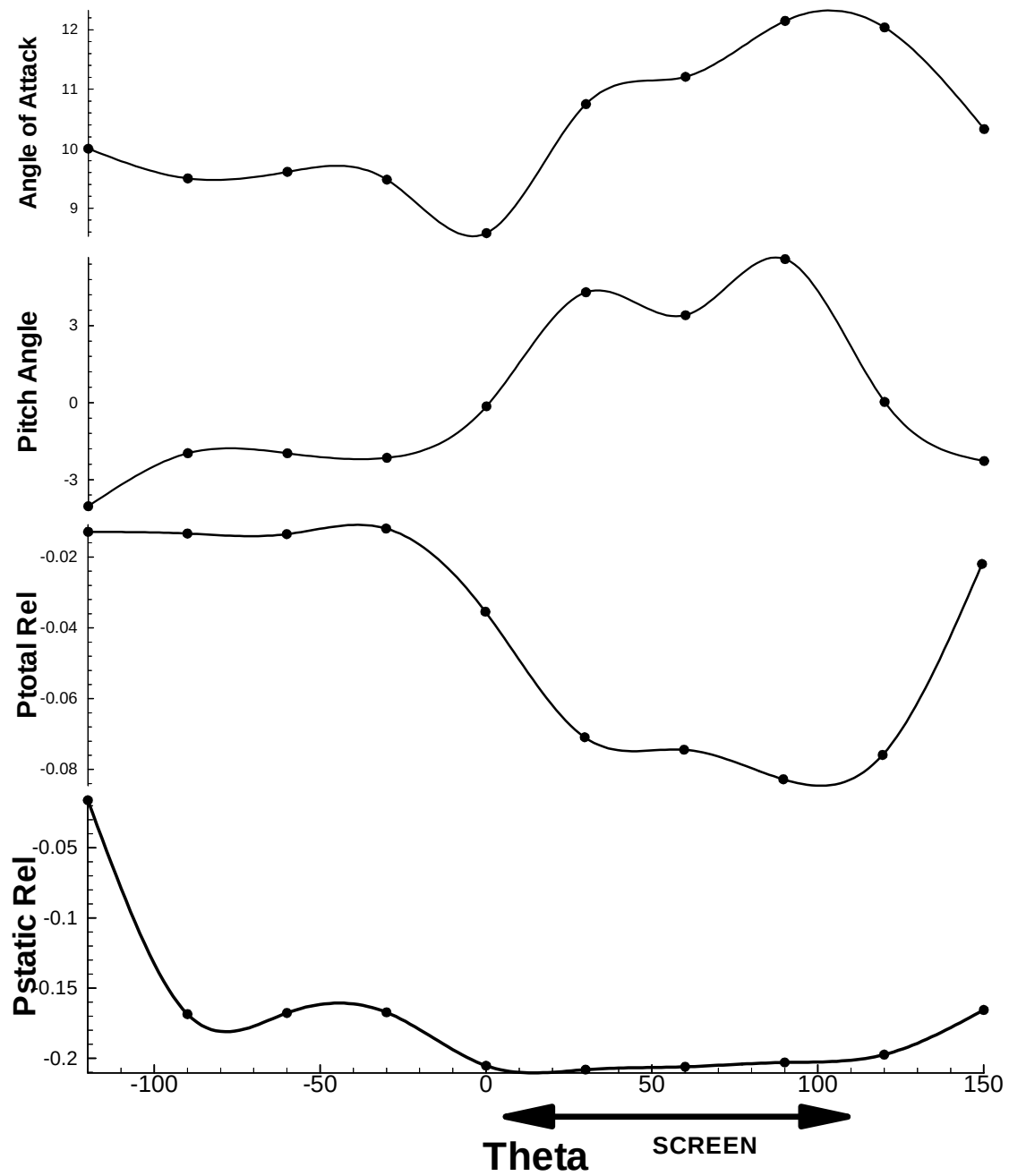


Figure A.3: Inlet Flow Characteristics for Level 1, Operating Point A at 50% Span.

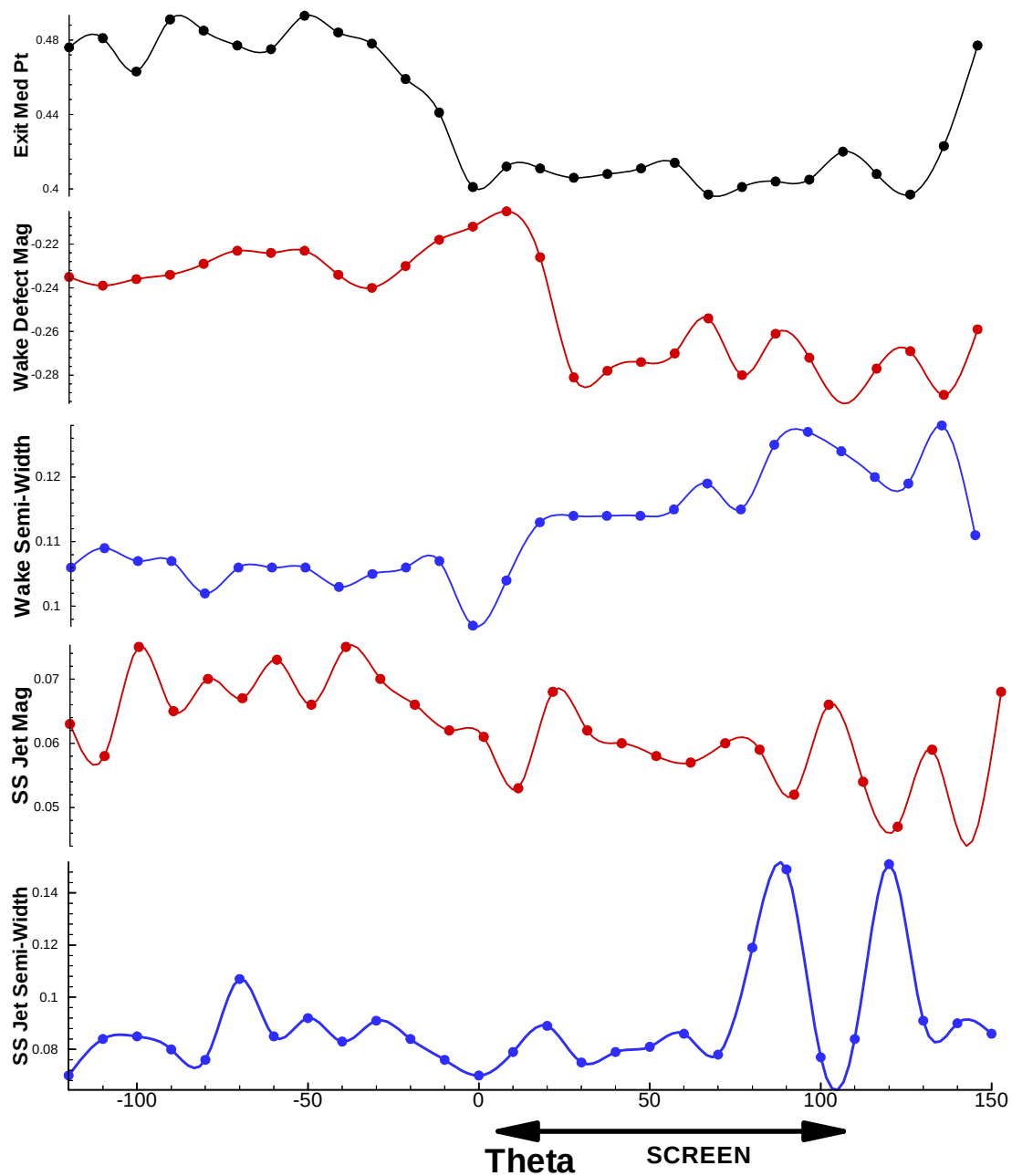


Figure A.4: Exit Flow Characteristics for Level 1, Operating Point A at 50% Span.

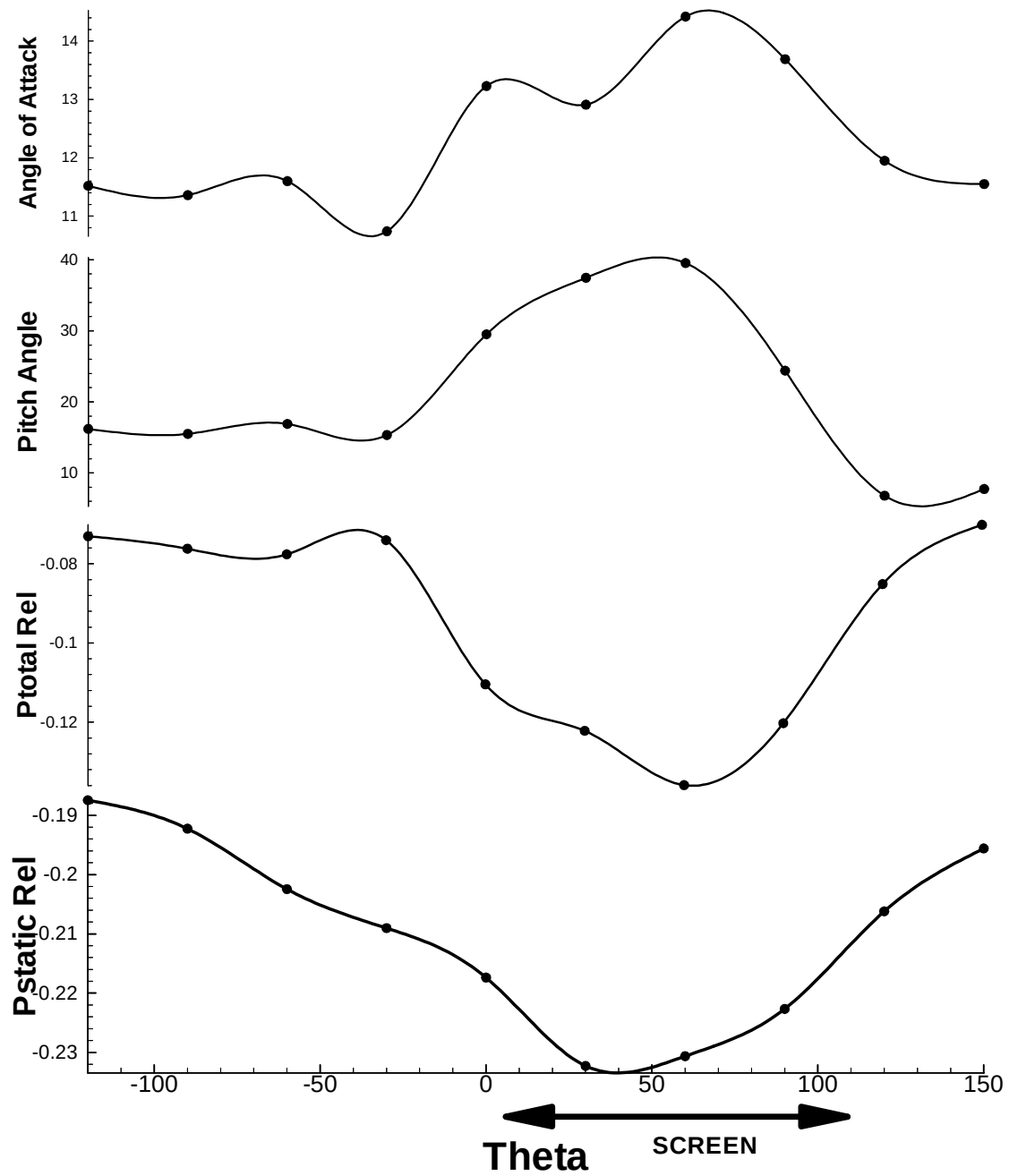


Figure A.5: Inlet Flow Characteristics for Level 1, Operating Point A at 33% Span.

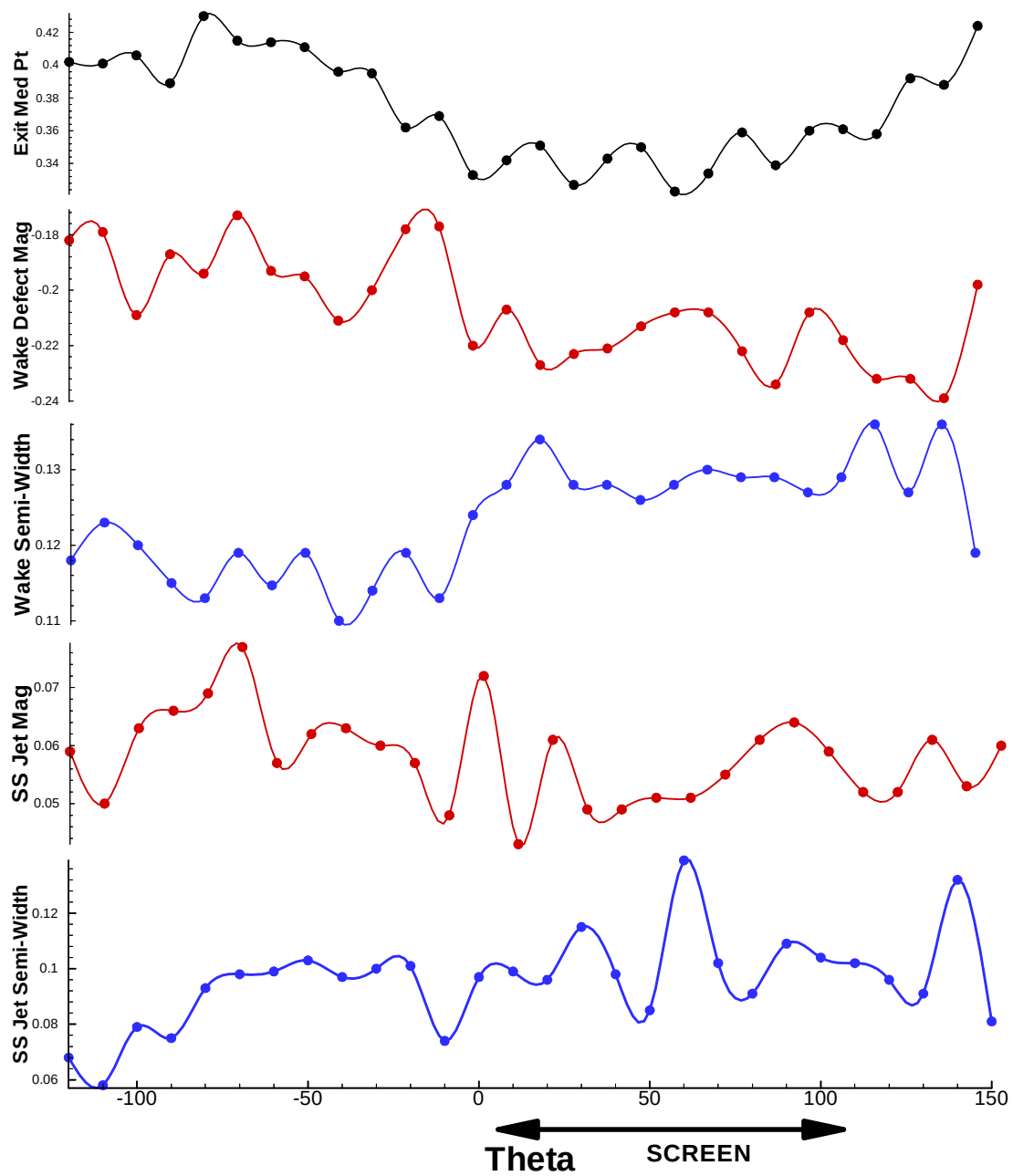


Figure A.6: Exit Flow Characteristics for Level 1, Operating Point A at 33% Span.

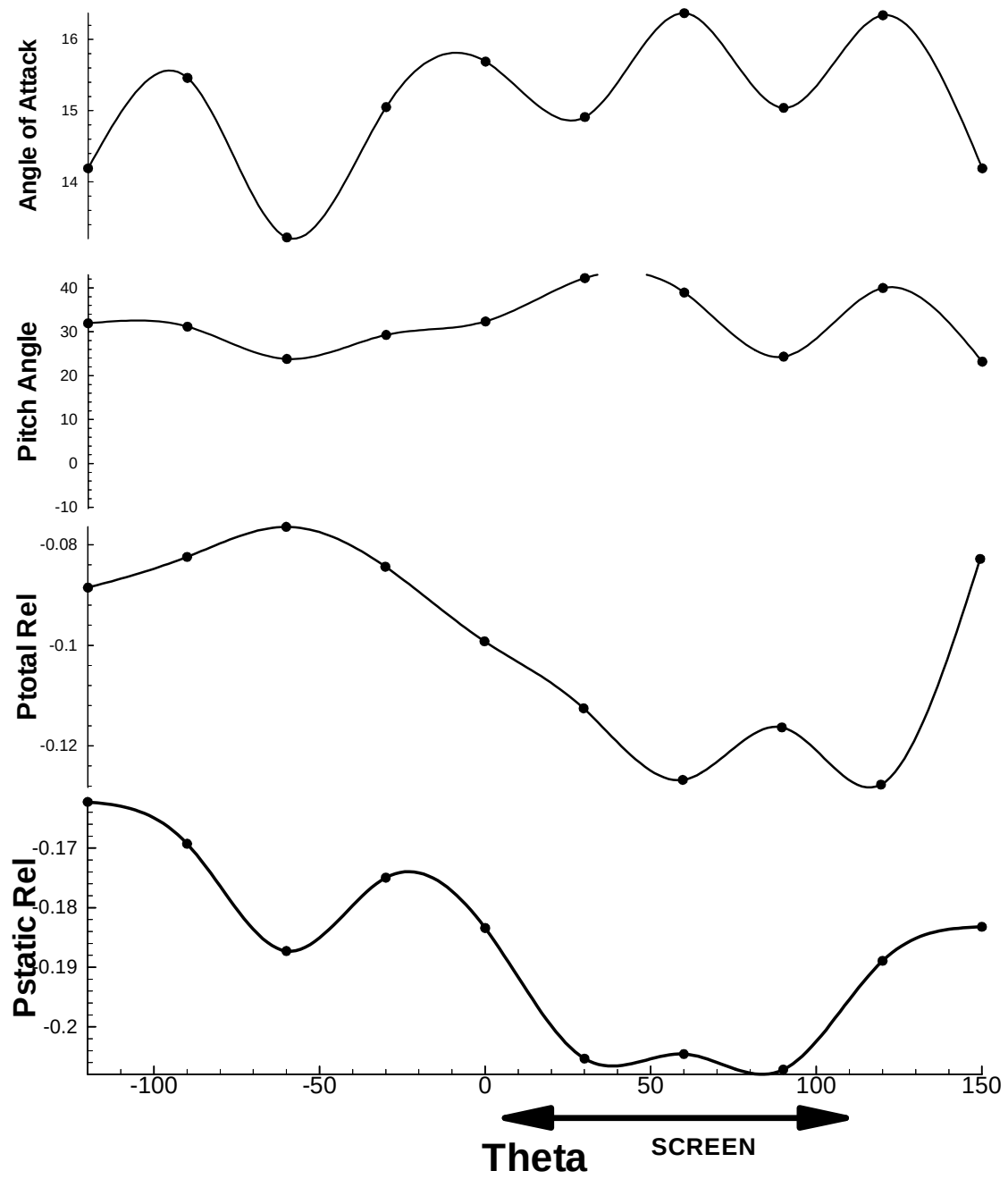


Figure A.7: Inlet Flow Characteristics for Level 1, Operating Point A at 67% Span.

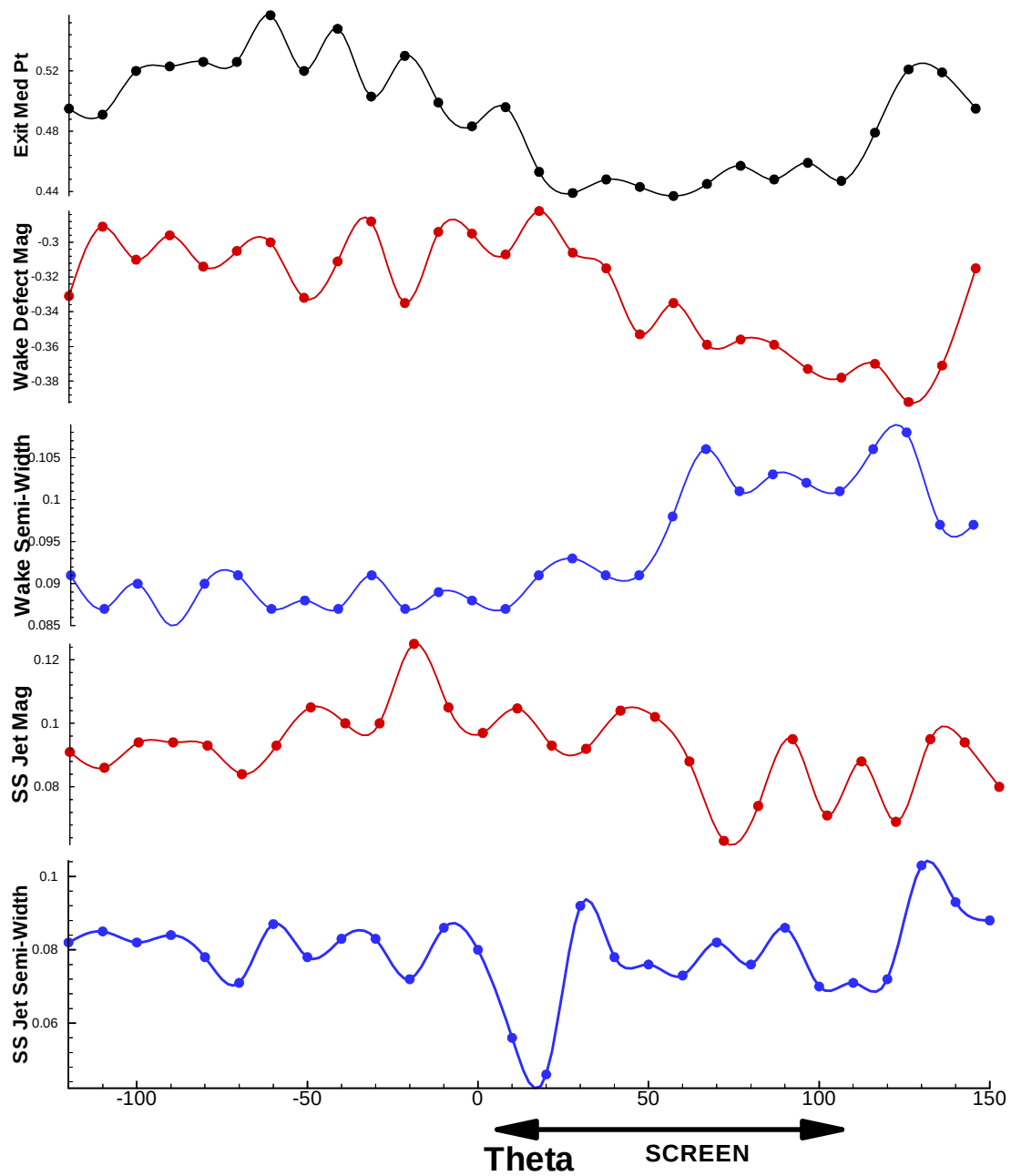


Figure A.8: Exit Flow Characteristics for Level 1, Operating Point A at 67% Span.

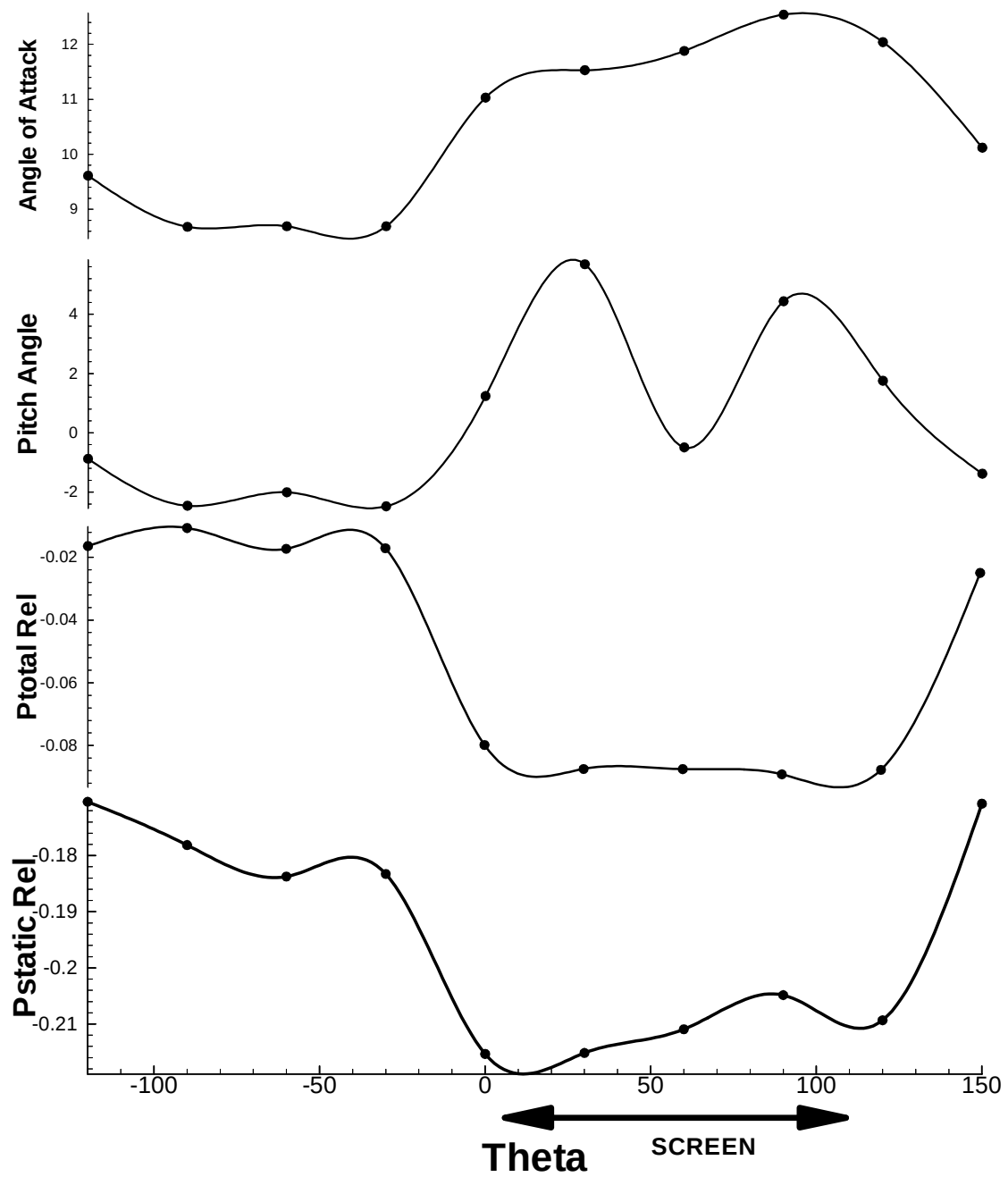


Figure A.9: Inlet Flow Characteristics for Level 1, Operating Point B at 50% Span.

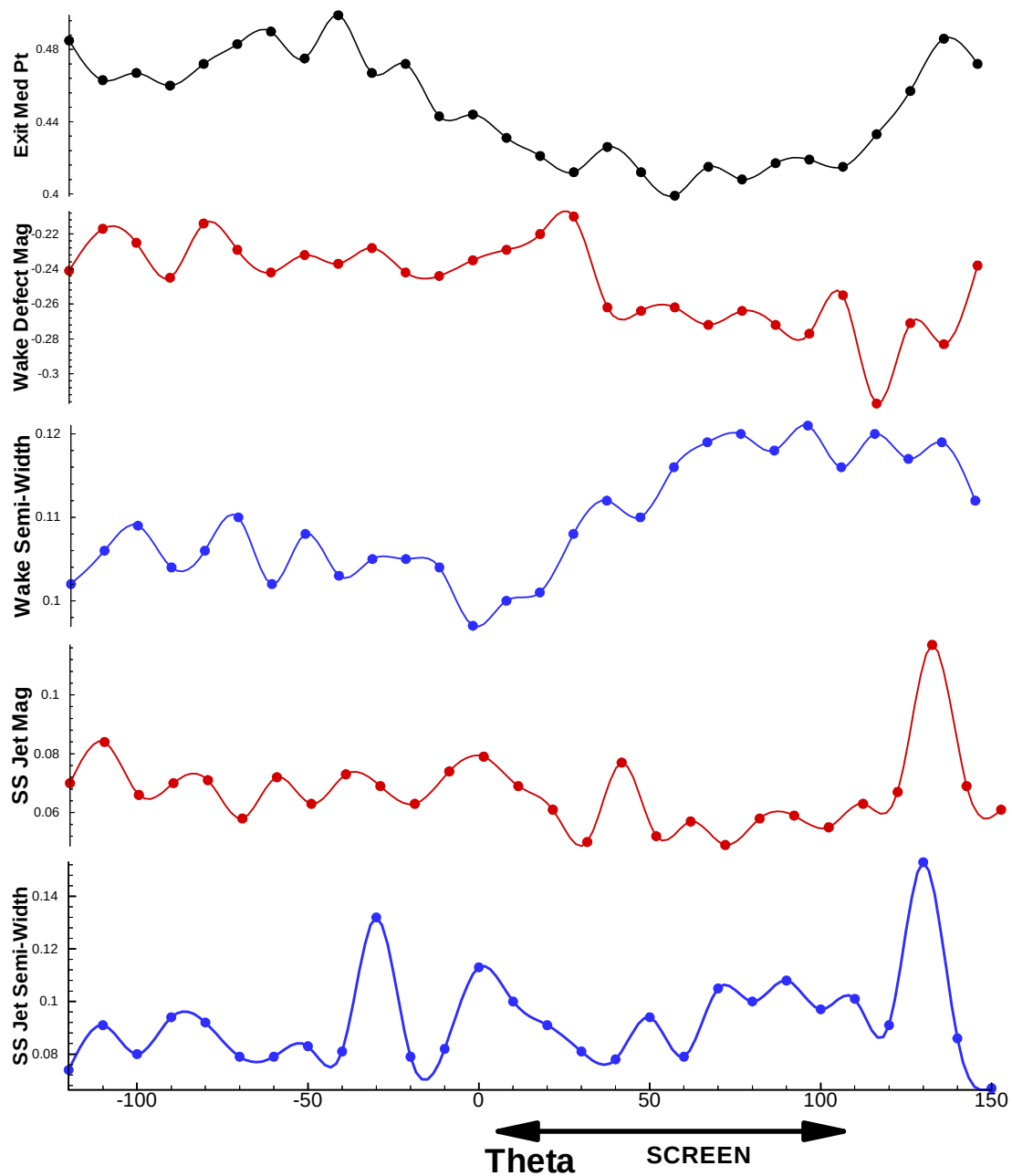


Figure A.10: Exit Flow Characteristics for Level 1, Operating Point B at 50% Span.

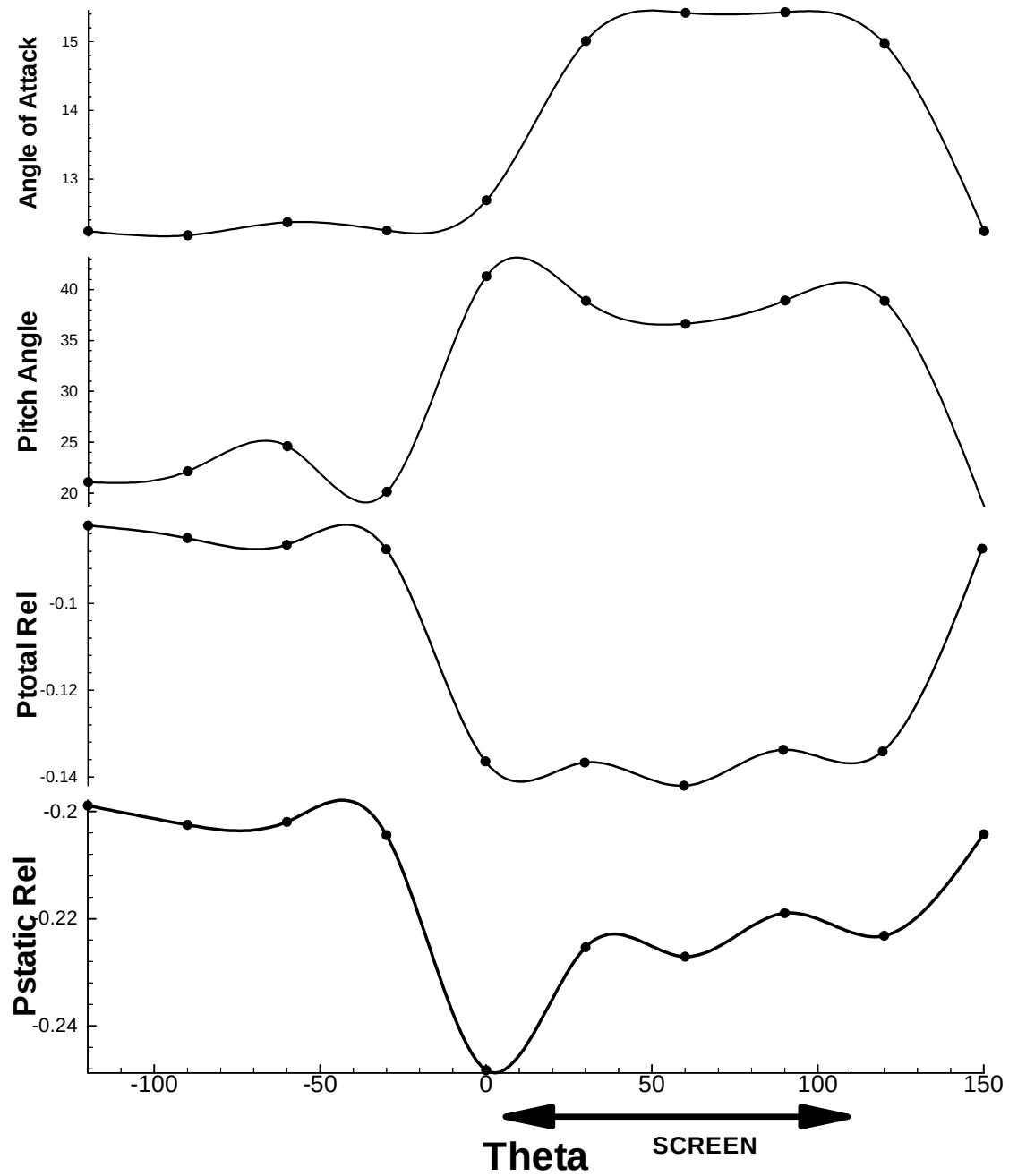


Figure A.11: Inlet Flow Characteristics for Level 1, Operating Point B at 33% Span.

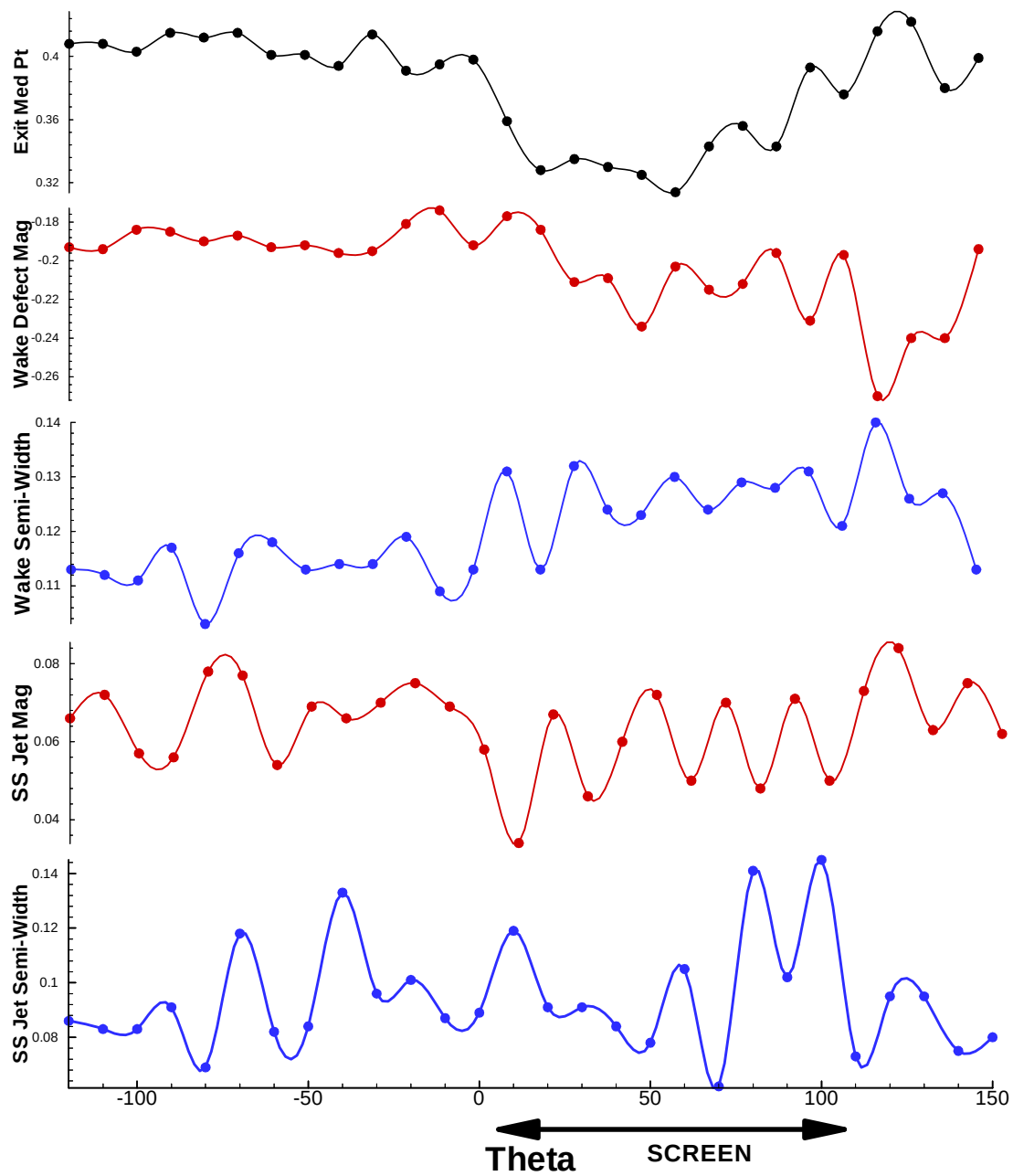


Figure A.12: Exit Flow Characteristics for Level 1, Operating Point B at 33% Span.

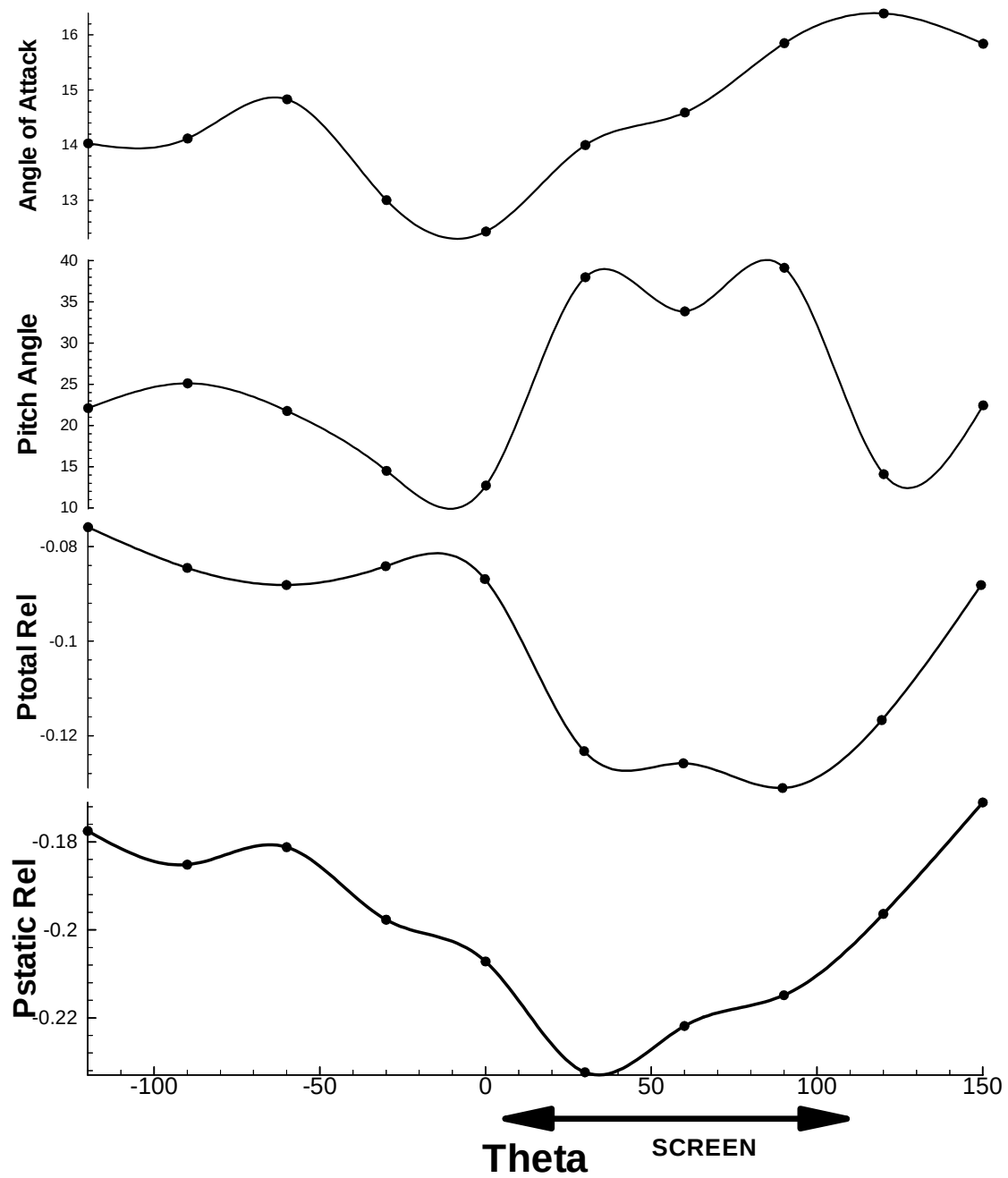


Figure A.13: Inlet Flow Characteristics for Level 1, Operating Point B at 67% Span.

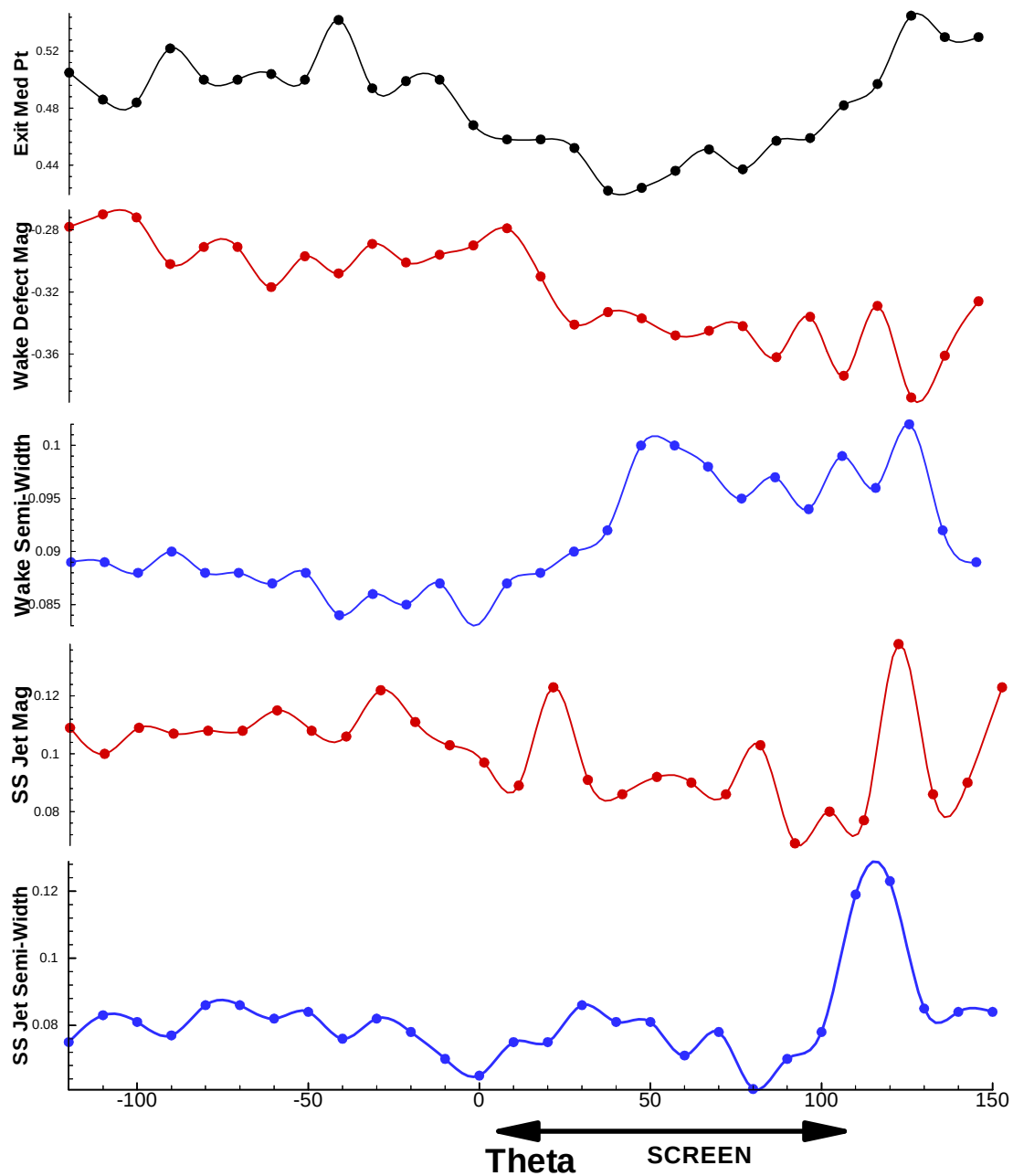


Figure A.14: Exit Flow Characteristics for Level 1, Operating Point B at 67% Span.

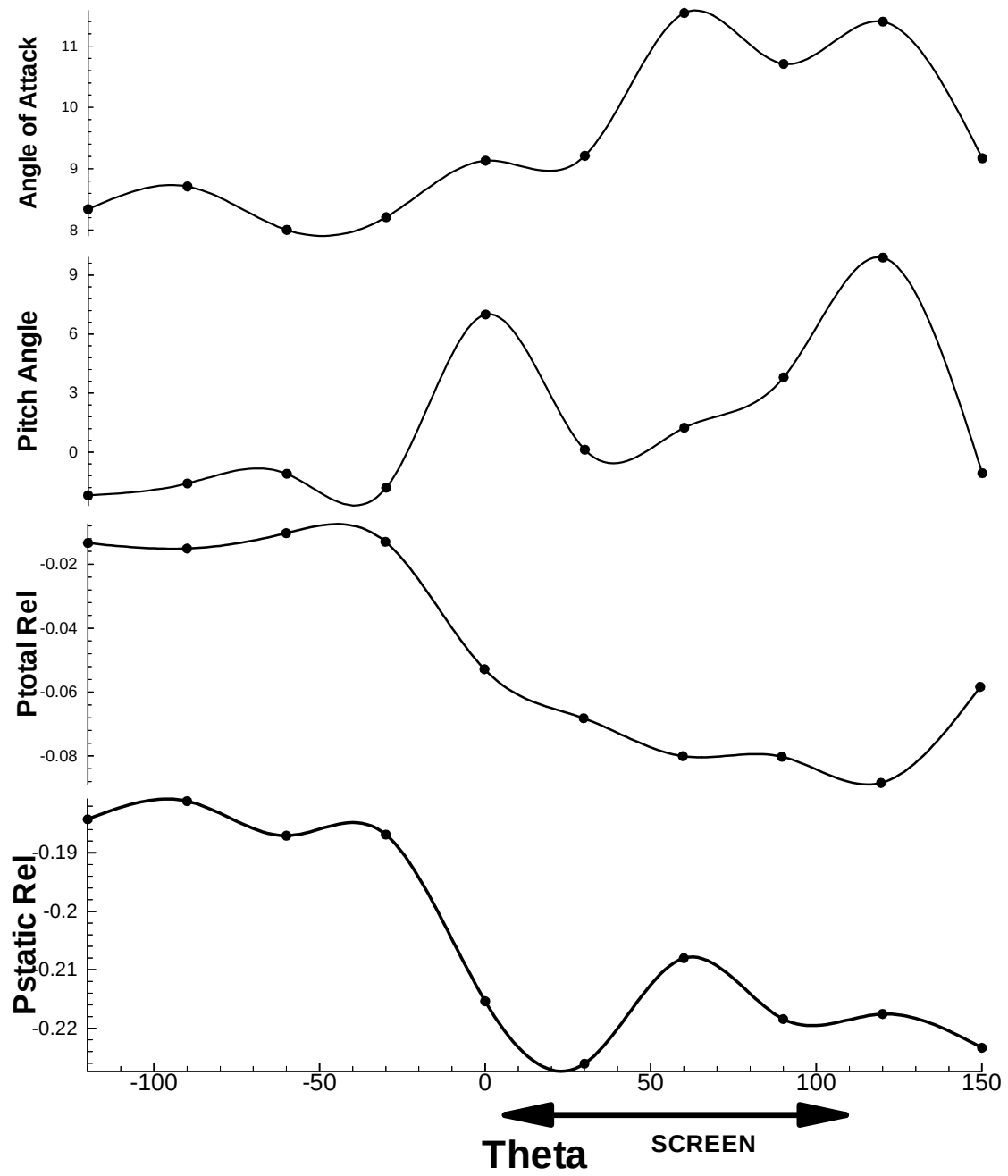


Figure A.15: Inlet Flow Characteristics for Level 1, Operating Point C at 50% Span.

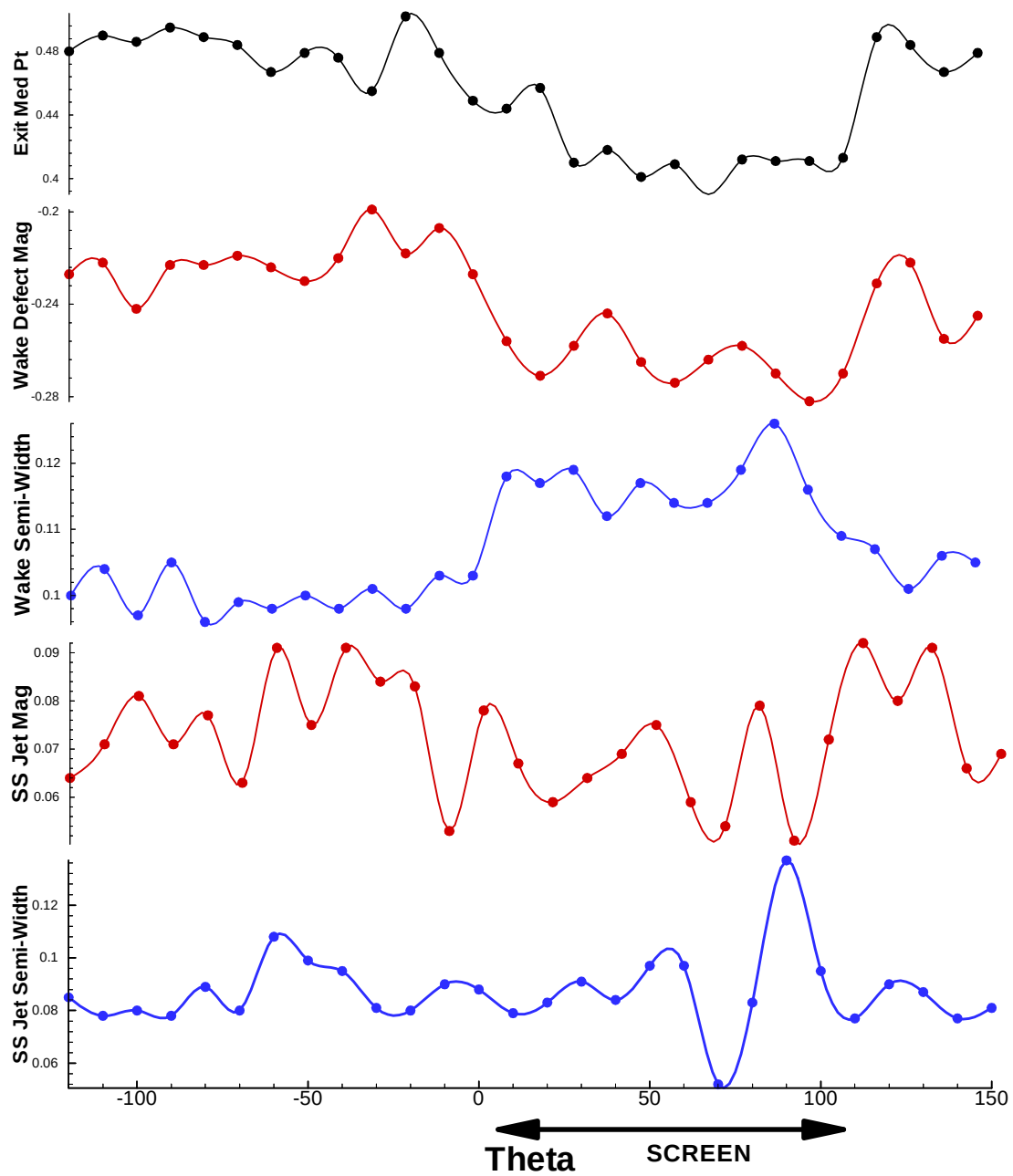


Figure A.16: Exit Flow Characteristics for Level 1, Operating Point C at 50% Span.

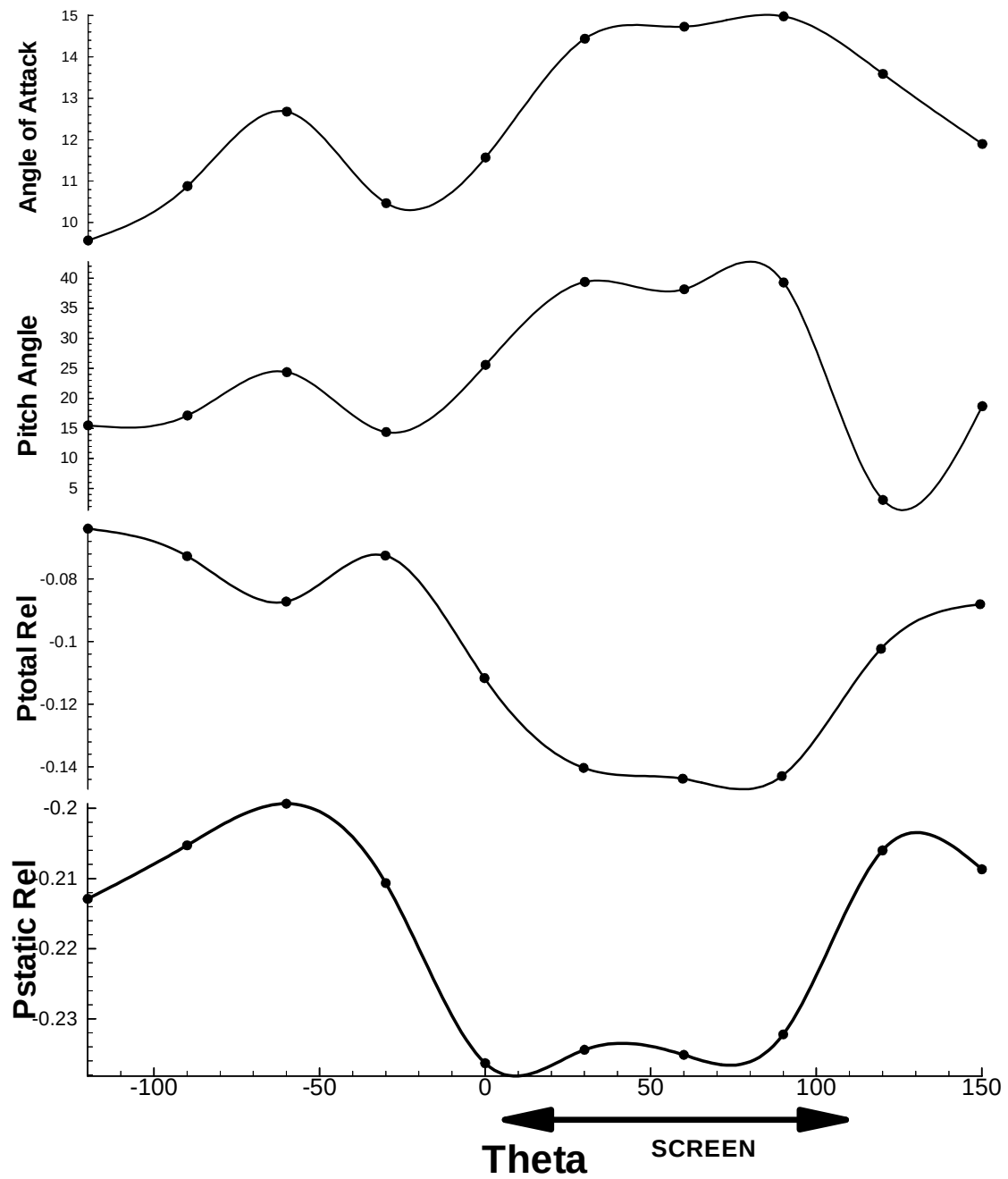


Figure A.17: Inlet Flow Characteristics for Level 1, Operating Point C at 33% Span.

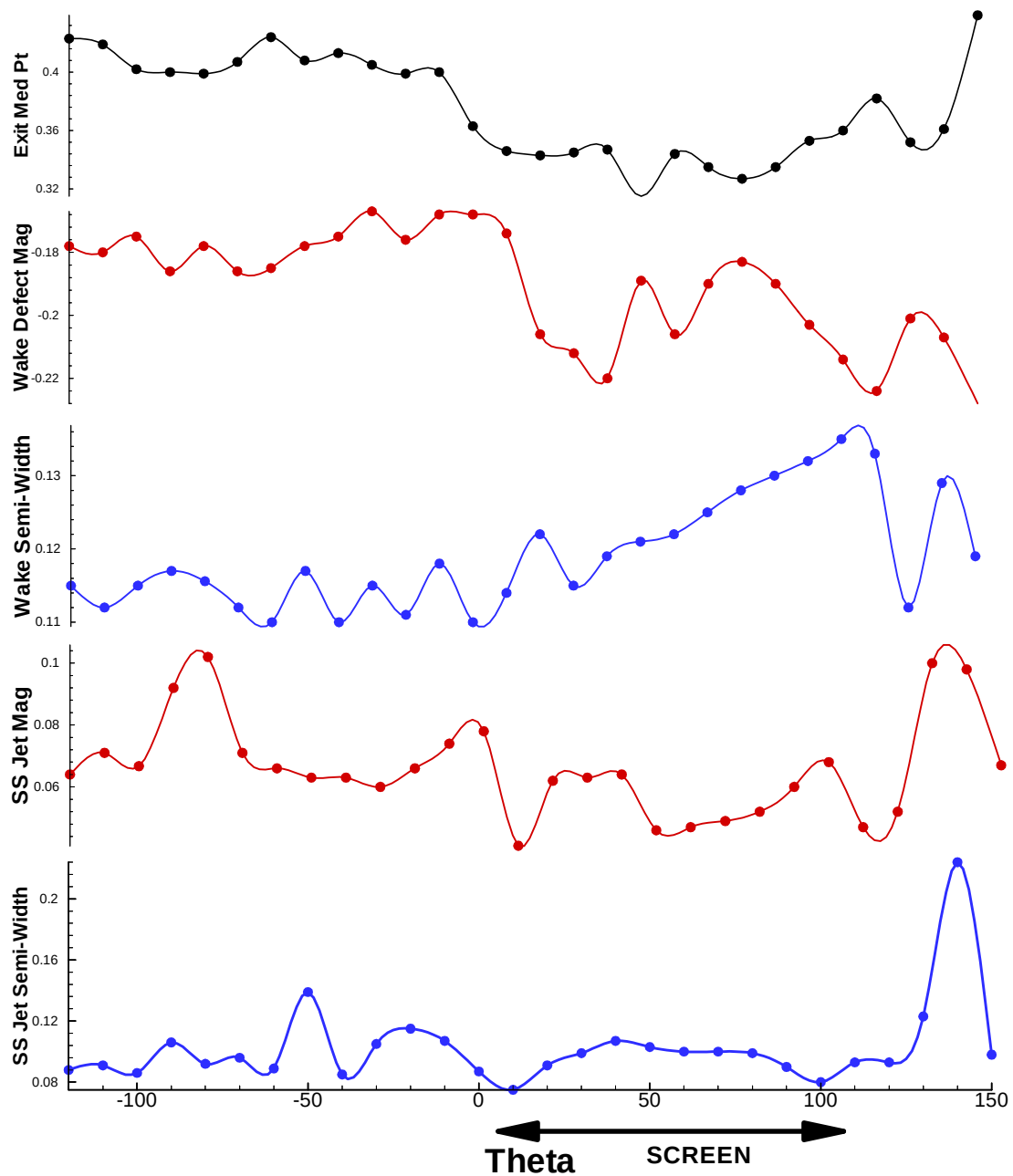


Figure A.18: Exit Flow Characteristics for Level 1, Operating Point C at 33% Span.

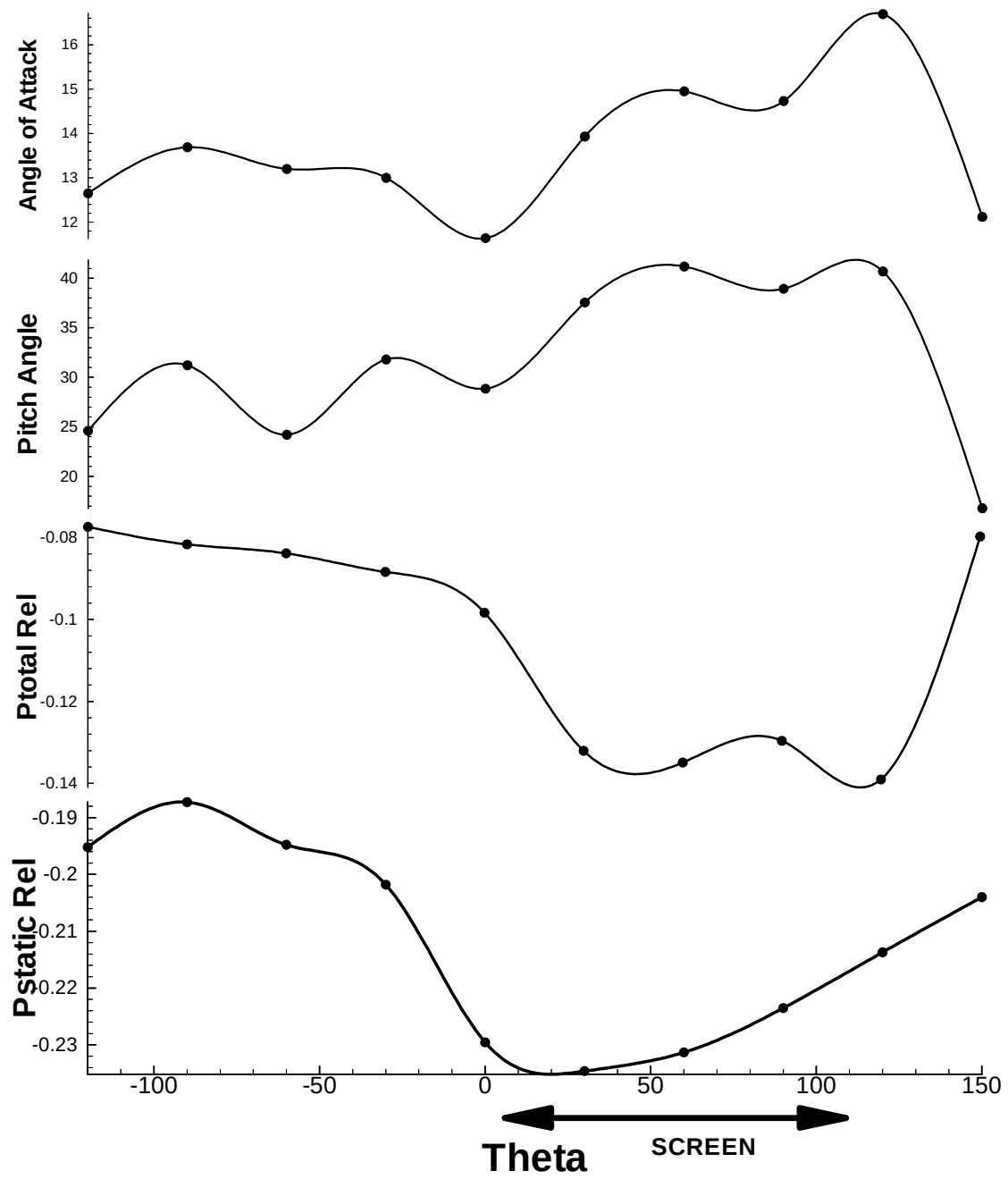


Figure A.19: Inlet Flow Characteristics for Level 1, Operating Point C at 67% Span.

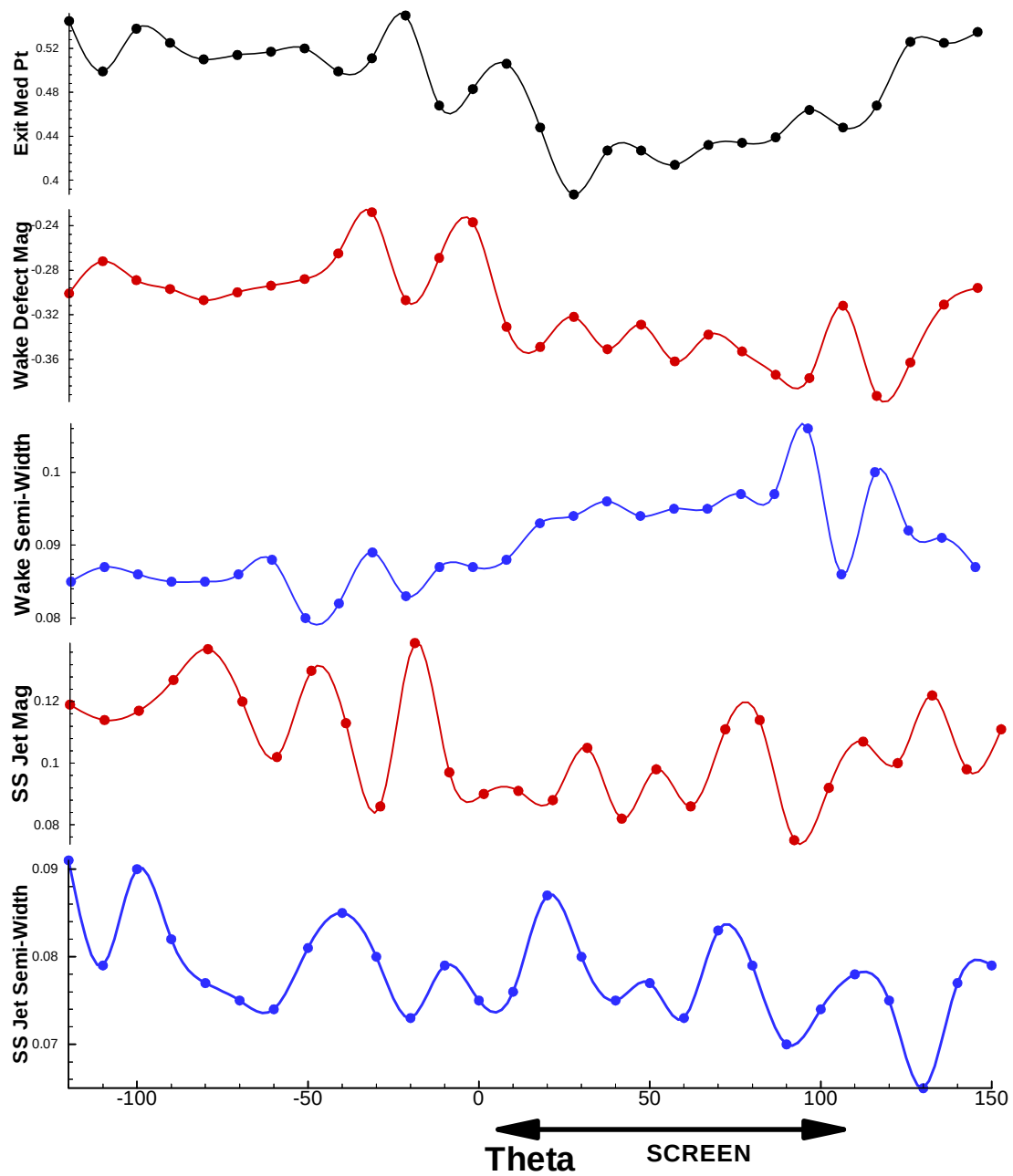


Figure A.20: Exit Flow Characteristics for Level 1, Operating Point C at 67% Span.

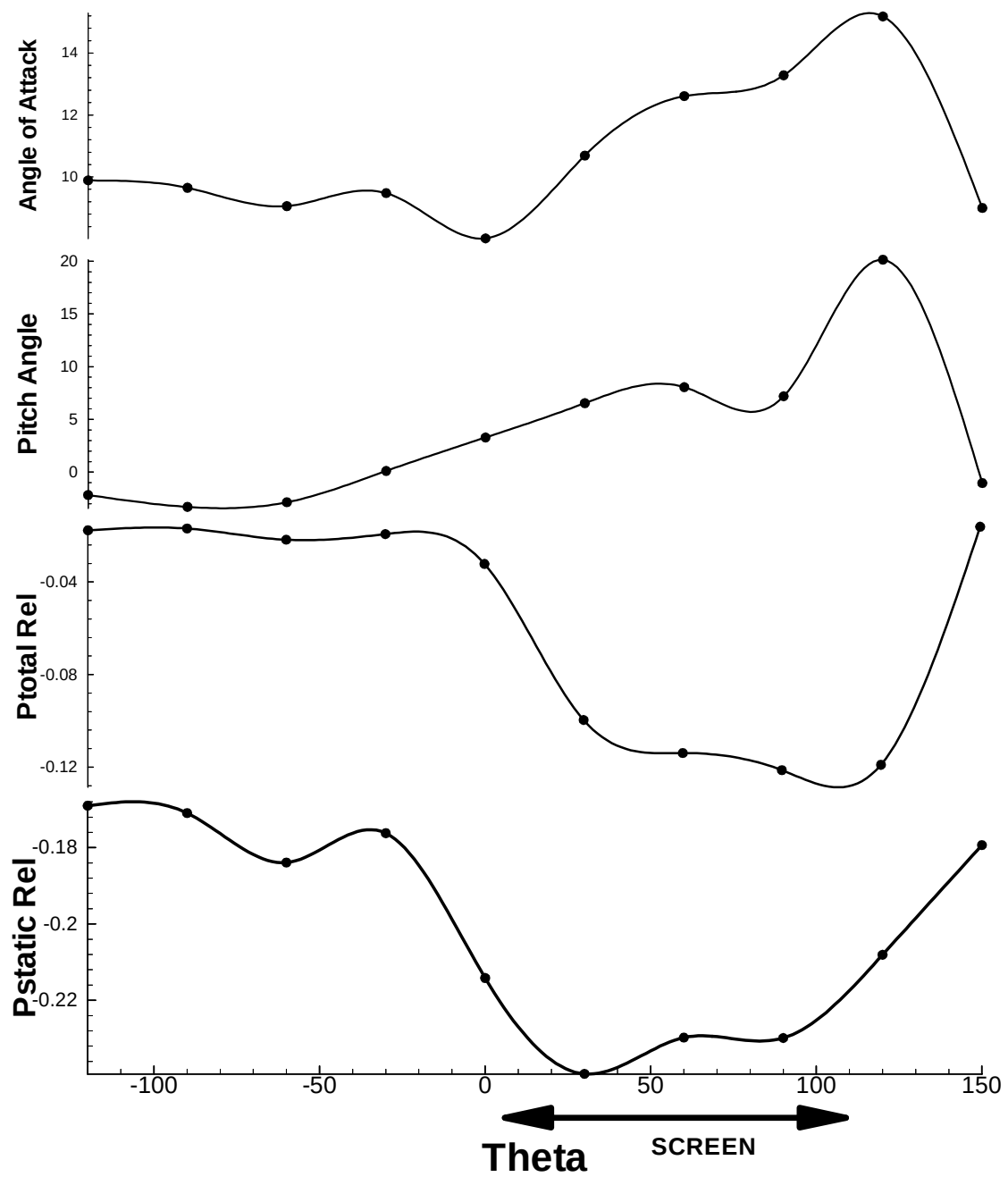


Figure A.21: Inlet Flow Characteristics for Level 2, Operating Point A at 50% Span.

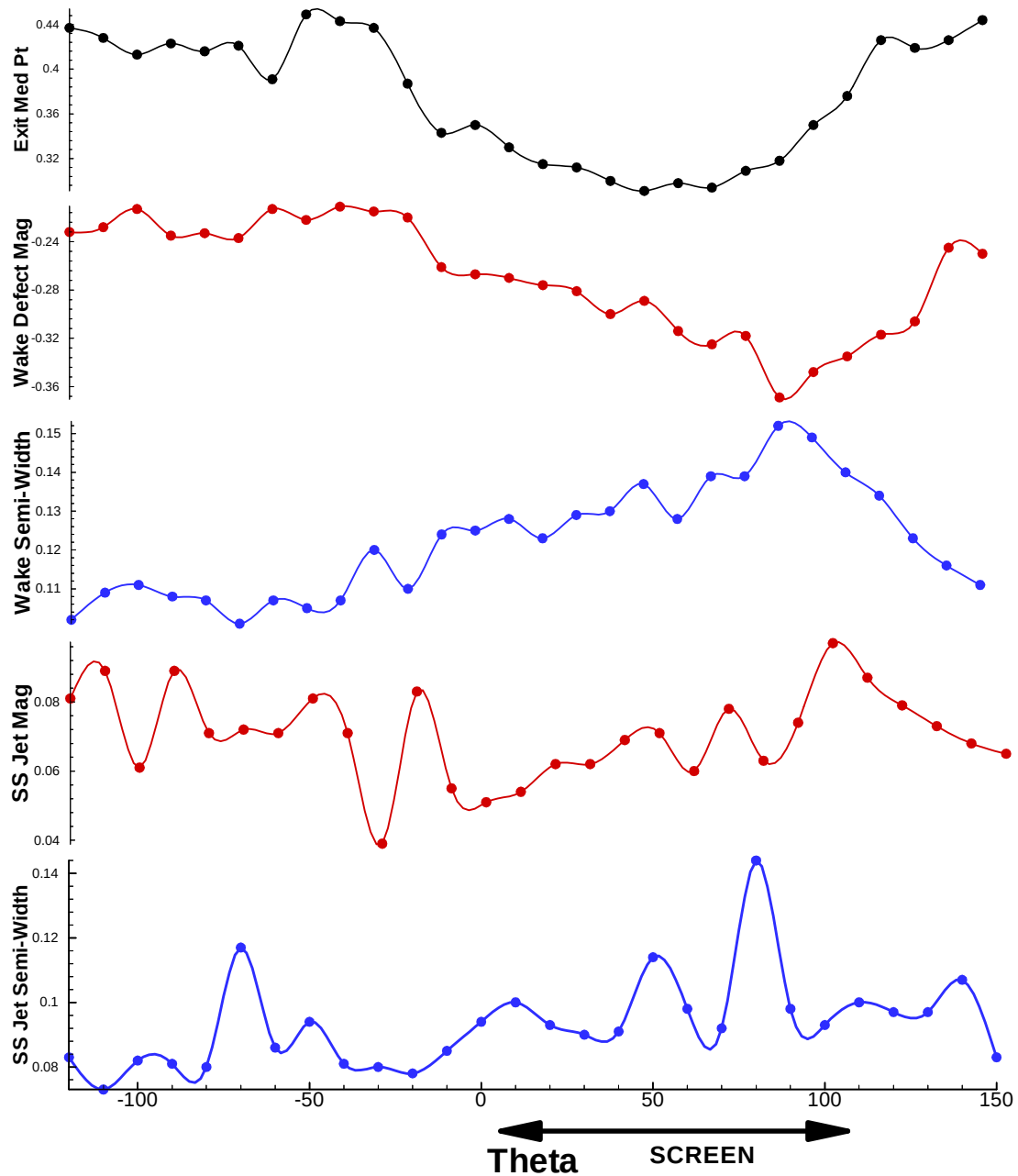


Figure A.22: Exit Flow Characteristics for Level 2, Operating Point A at 50% Span.

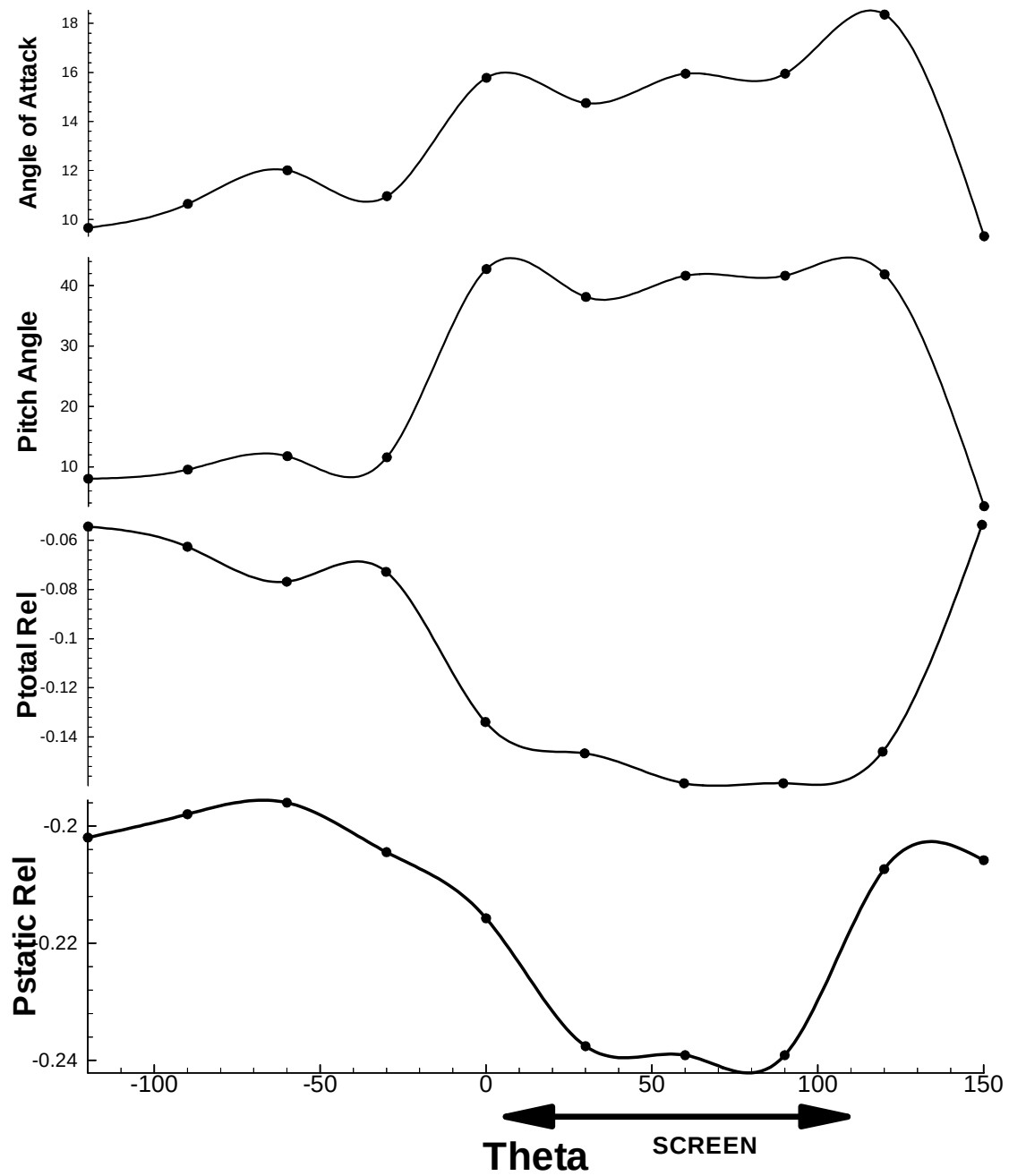


Figure A.23: Inlet Flow Characteristics for Level 2, Operating Point A at 33% Span.

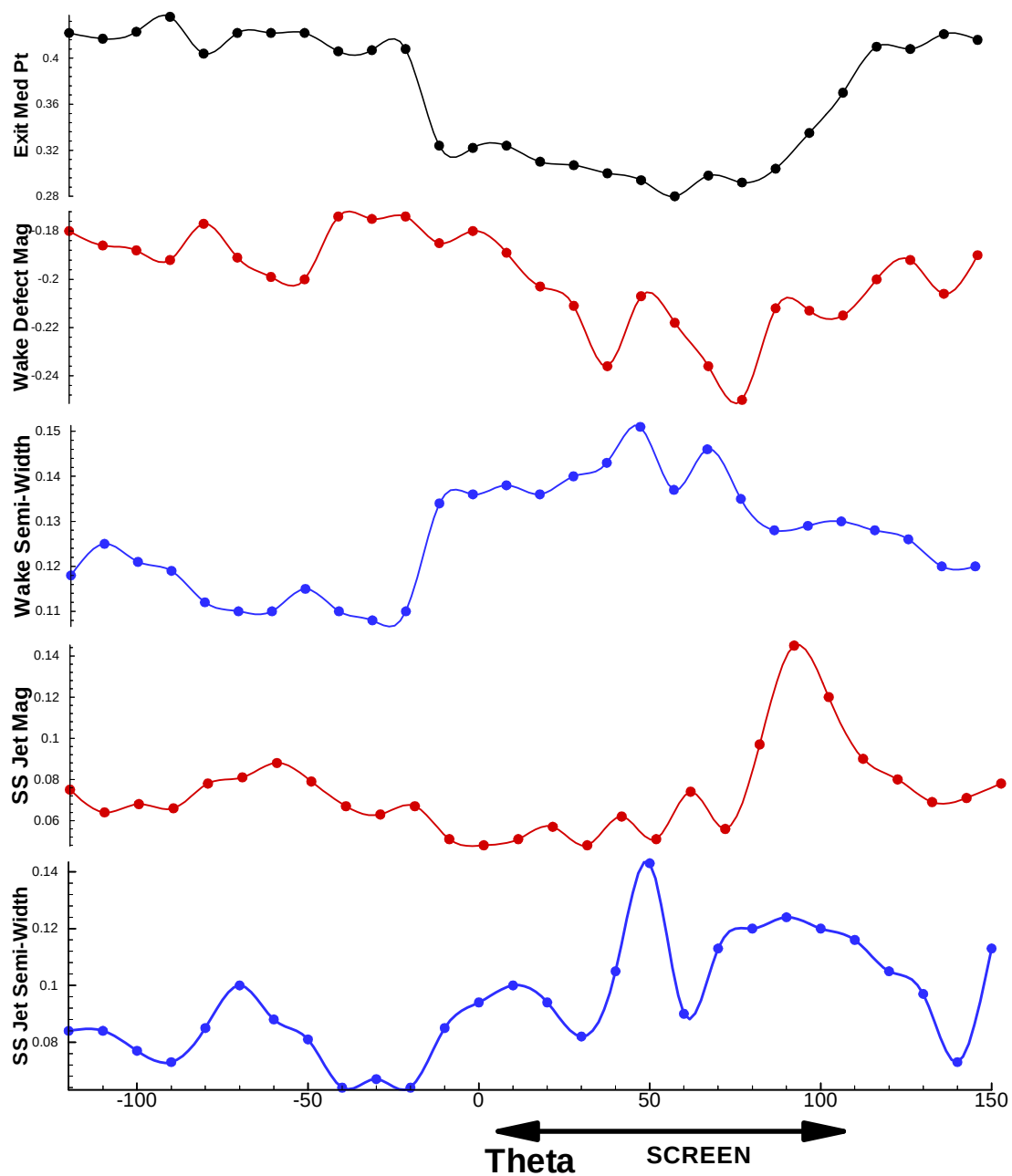


Figure A.24: Exit Flow Characteristics for Level 2, Operating Point A at 33% Span.

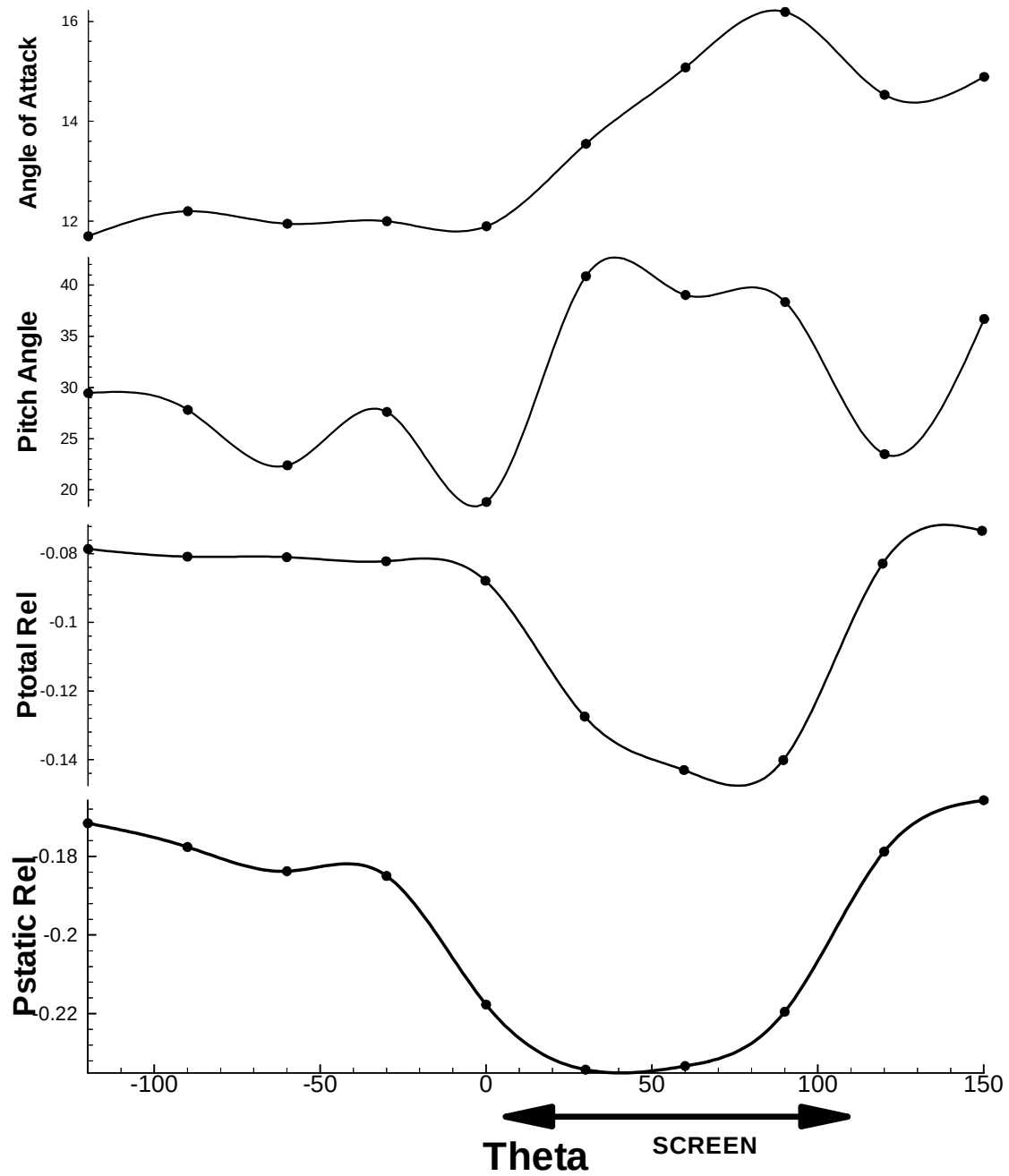


Figure A.25: Inlet Flow Characteristics for Level 2, Operating Point A at 67% Span.

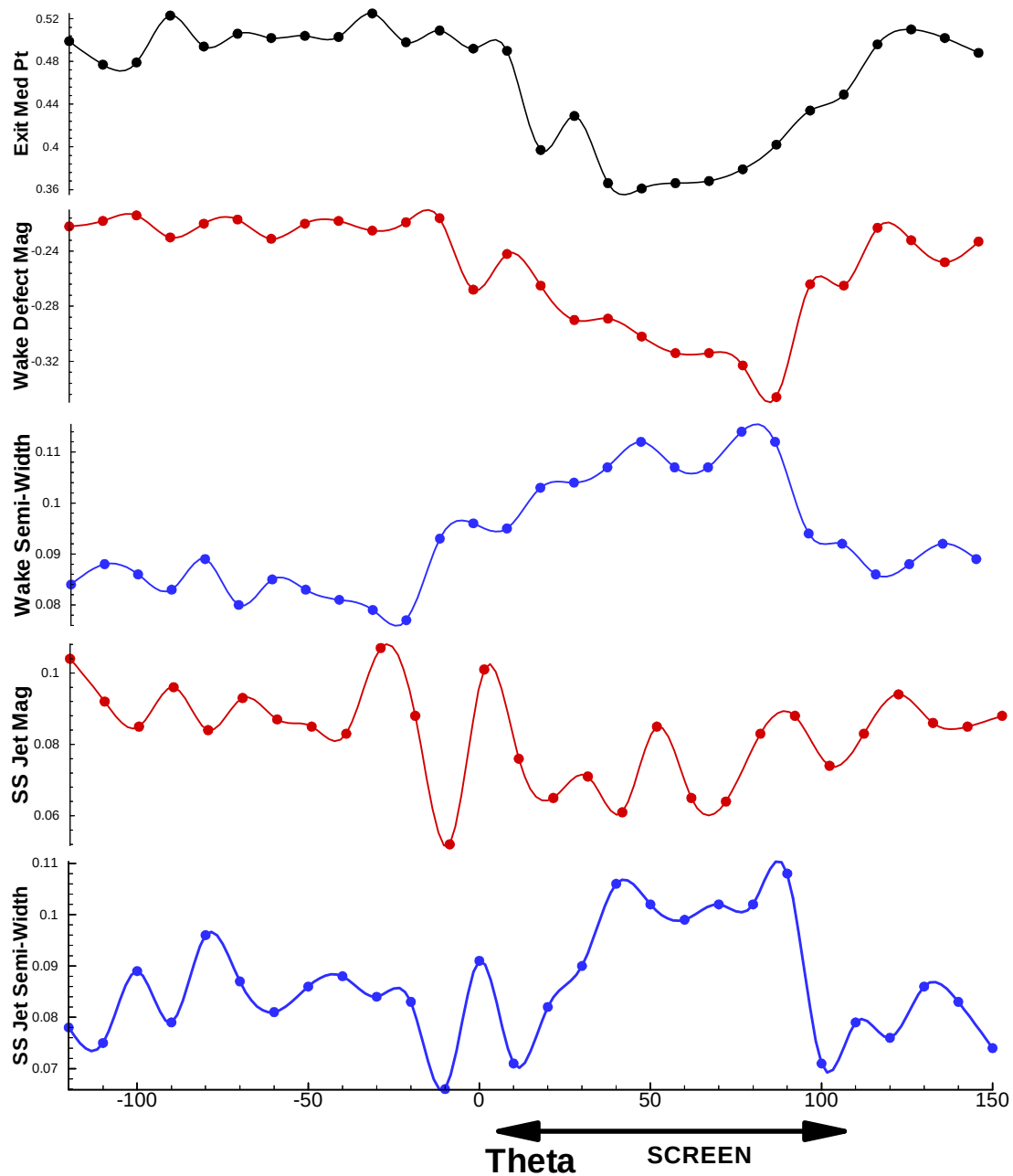


Figure A.26: Exit Flow Characteristics for Level 2, Operating Point A at 67% Span.

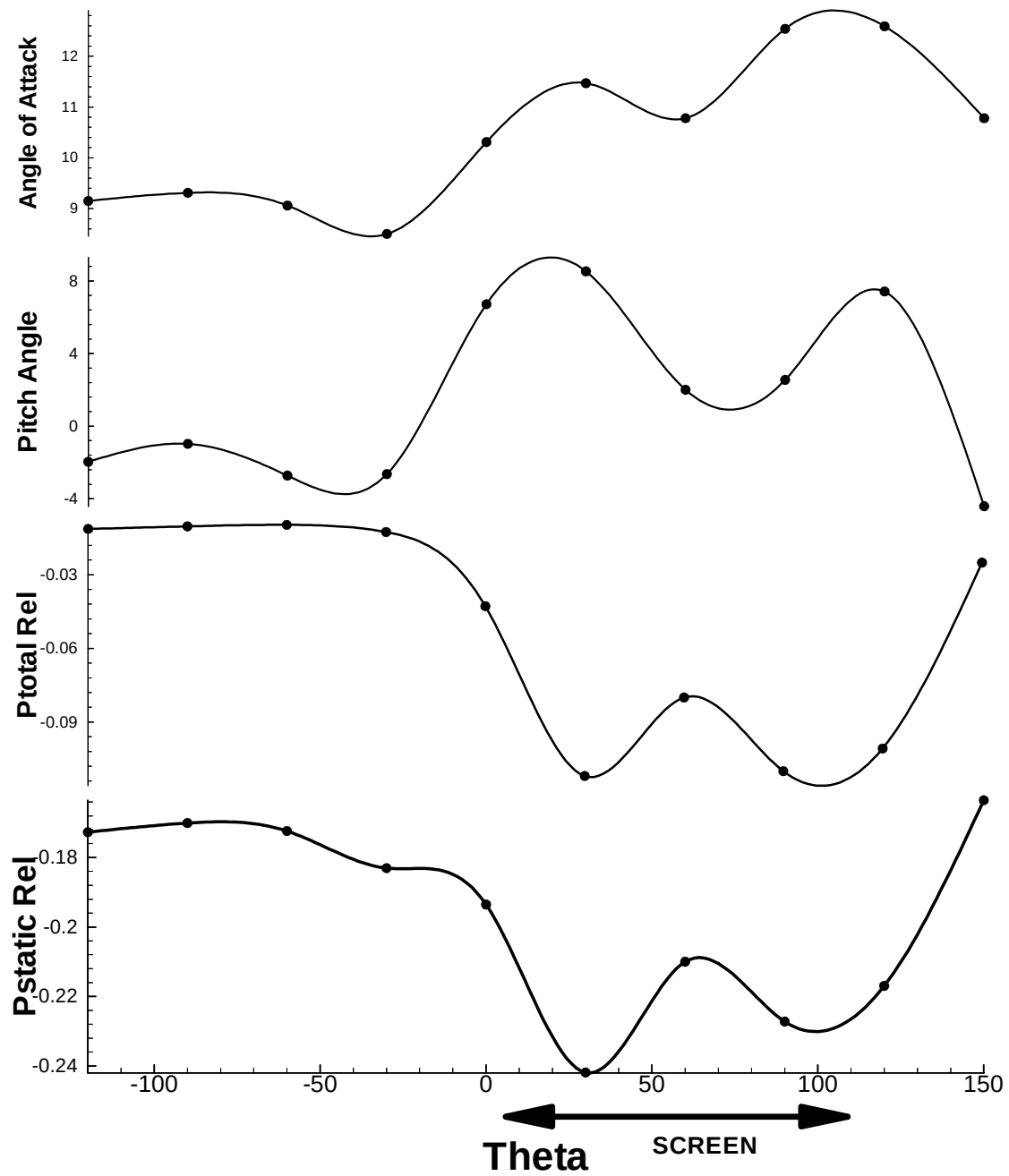


Figure A.27: Inlet Flow Characteristics for Level 2, Operating Point B at 50% Span.

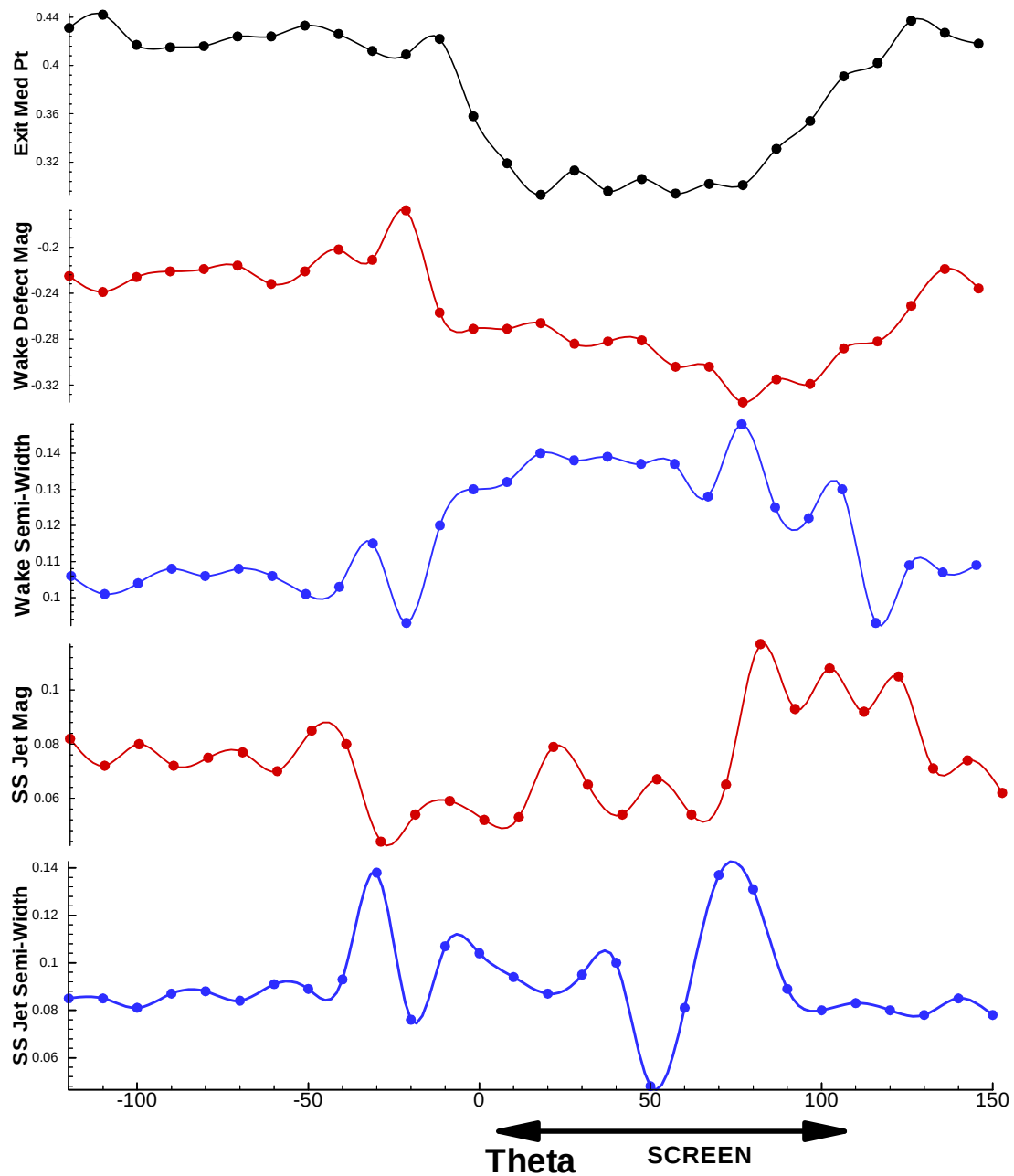


Figure A.28: Exit Flow Characteristics for Level 2, Operating Point B at 50% Span.

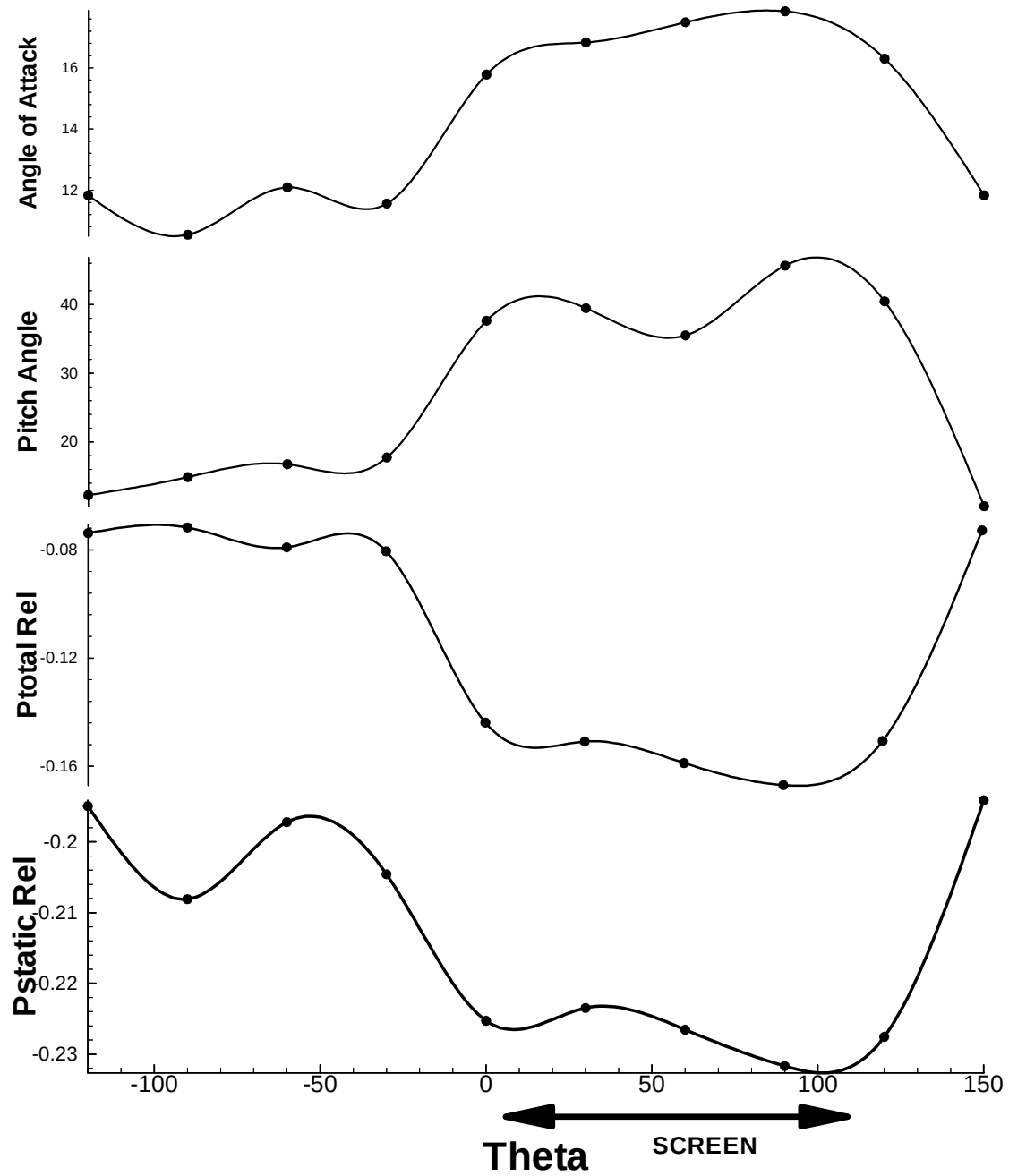


Figure A.29: Inlet Flow Characteristics for Level 2, Operating Point B at 33% Span.

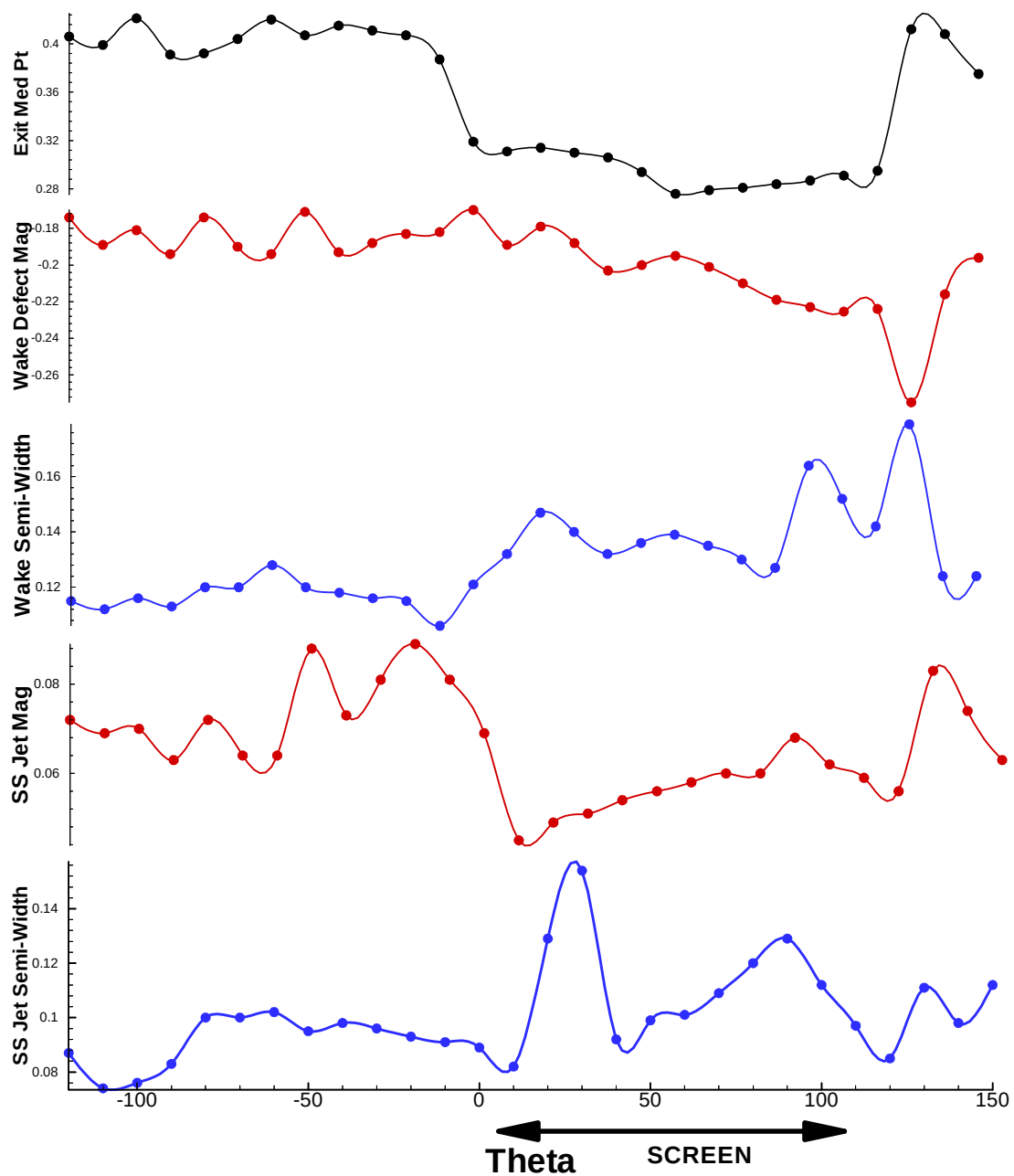


Figure A.30: Exit Flow Characteristics for Level 2, Operating Point B at 33% Span.

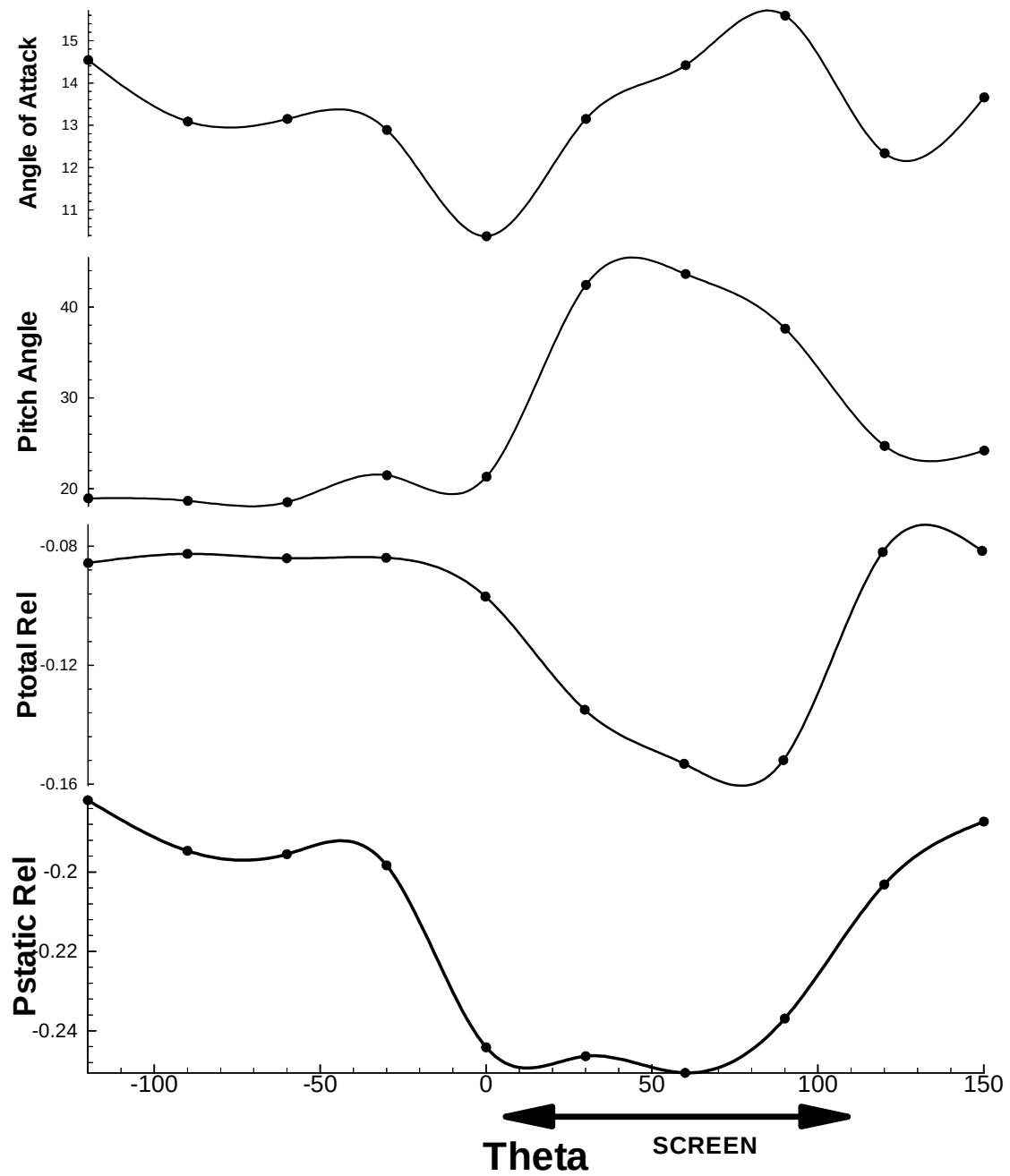


Figure A.31: Inlet Flow Characteristics for Level 2, Operating Point B at 67% Span.

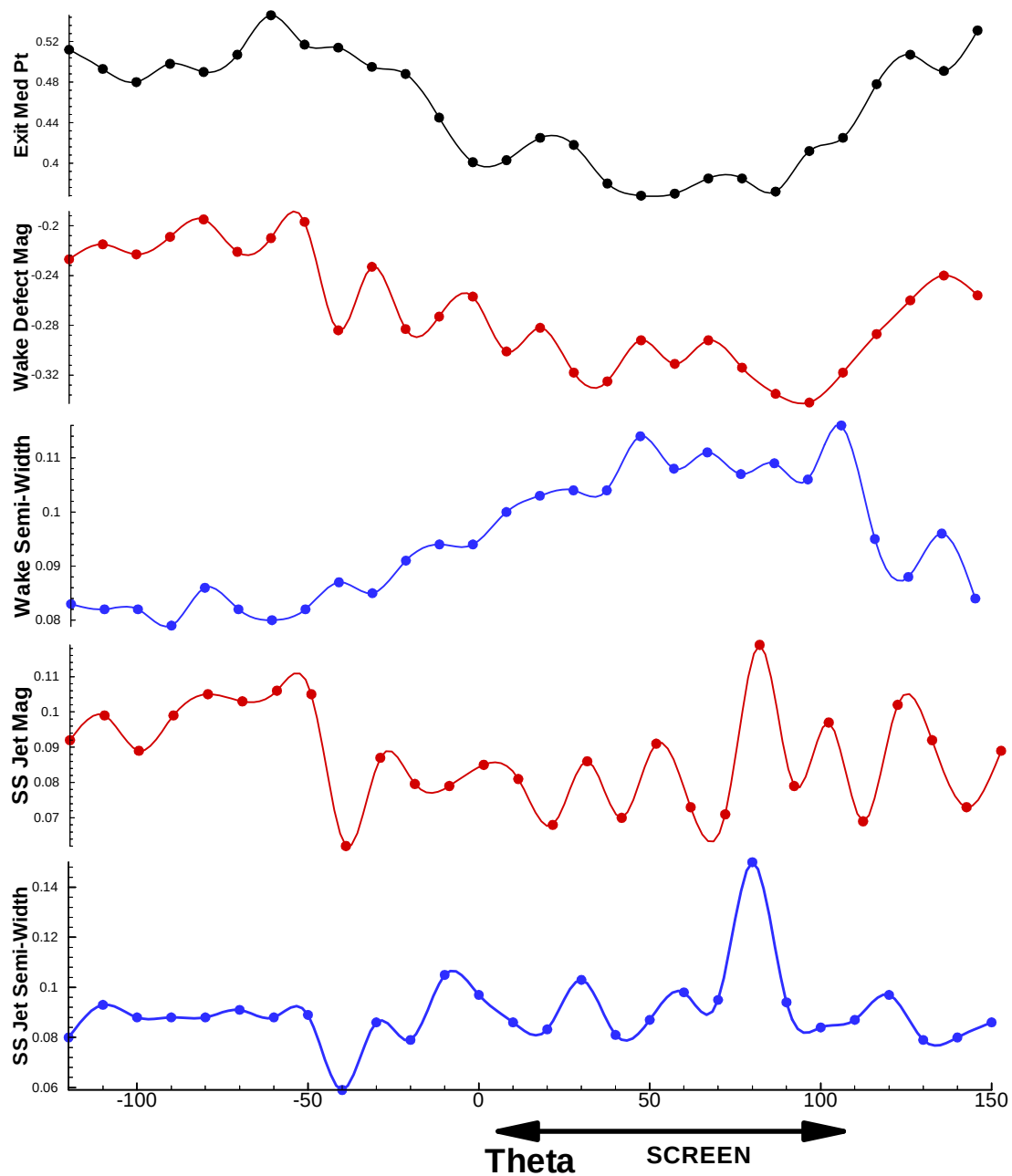


Figure A.32: Exit Flow Characteristics for Level 2, Operating Point B at 67% Span.

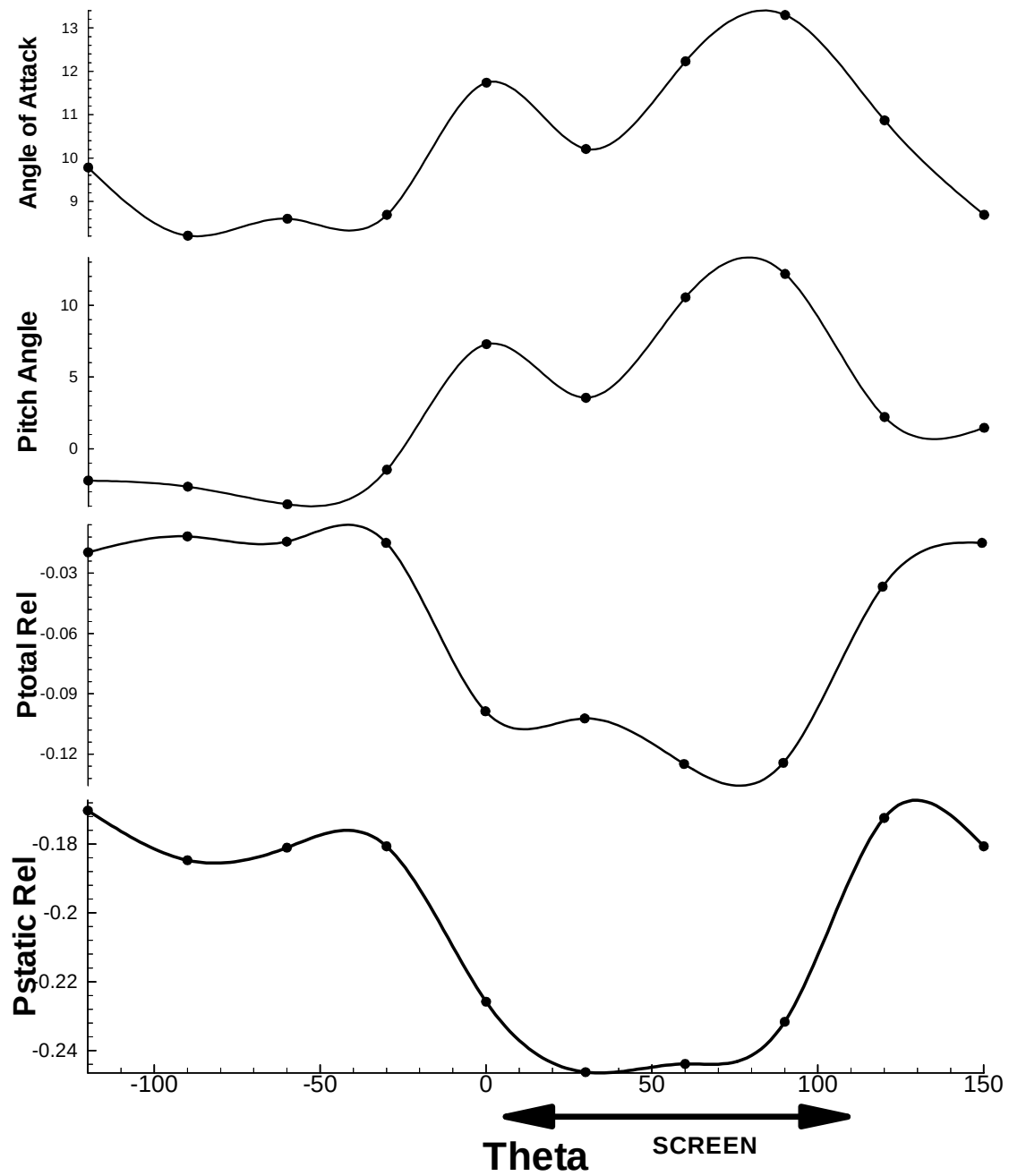


Figure A.33: Inlet Flow Characteristics for Level 2, Operating Point C at 50% Span.

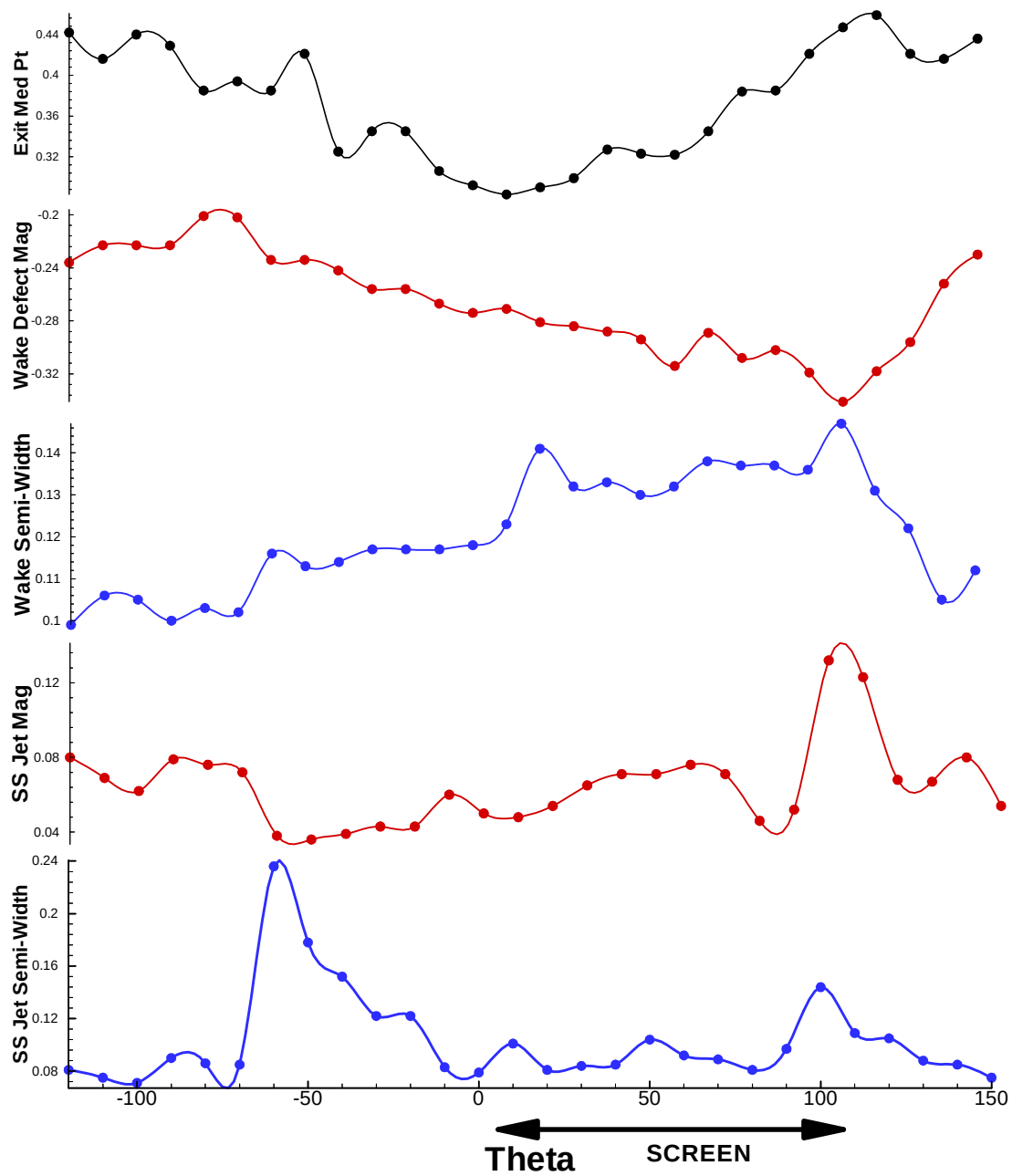


Figure A.34: Exit Flow Characteristics for Level 2, Operating Point C at 50% Span.

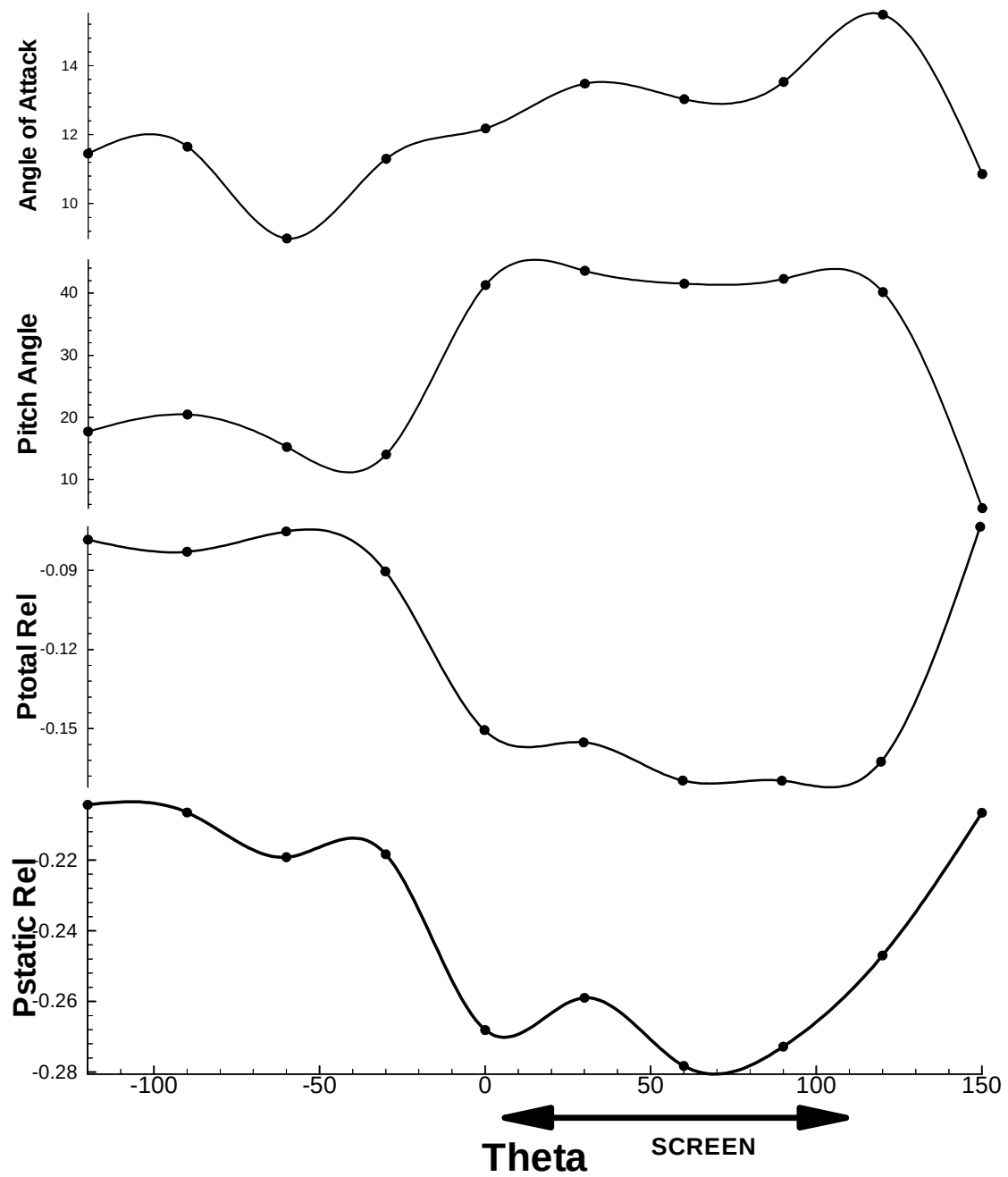


Figure A.35: Inlet Flow Characteristics for Level 2, Operating Point C at 33% Span.

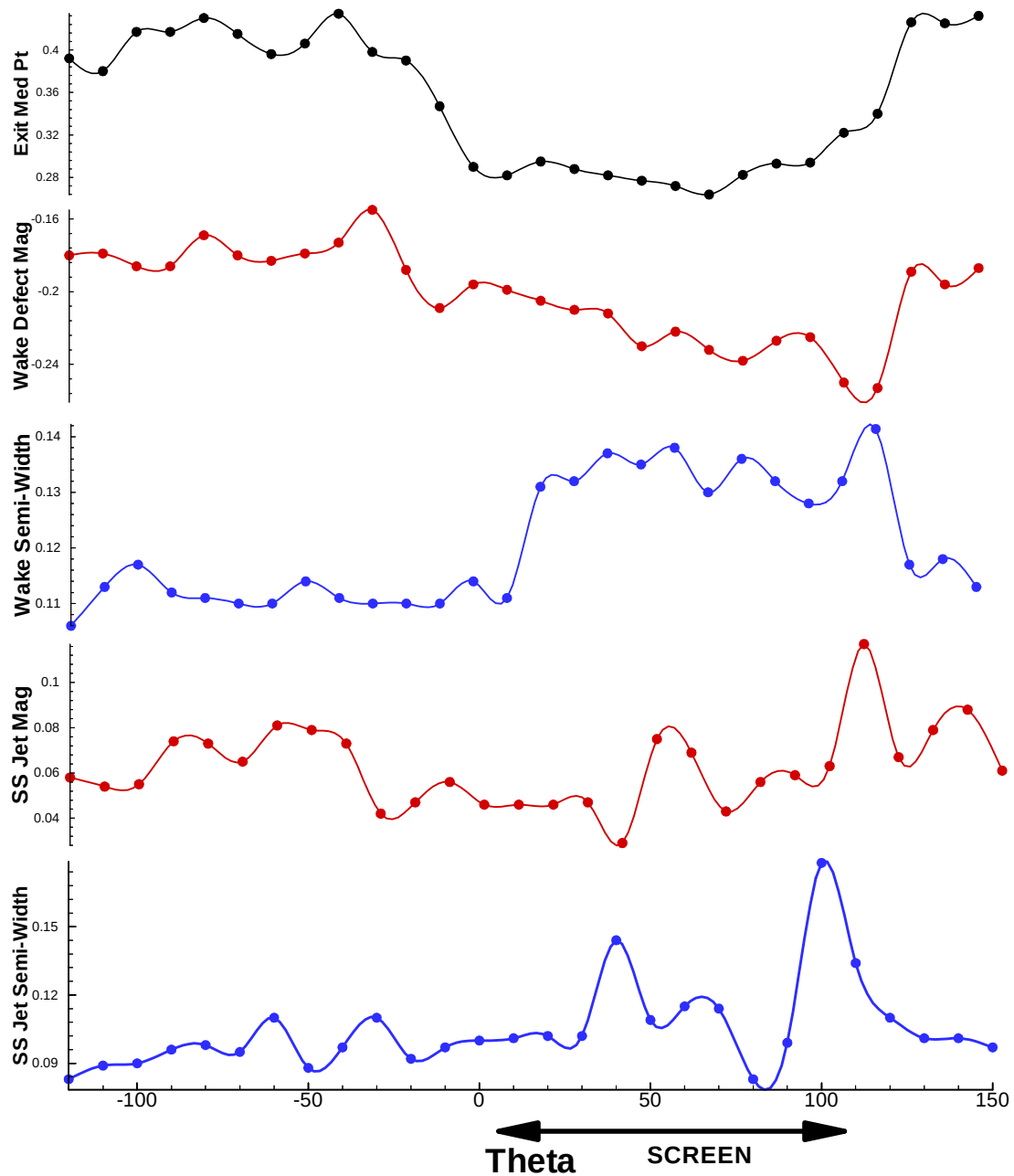


Figure A.36: Exit Flow Characteristics for Level 2, Operating Point C at 33% Span.

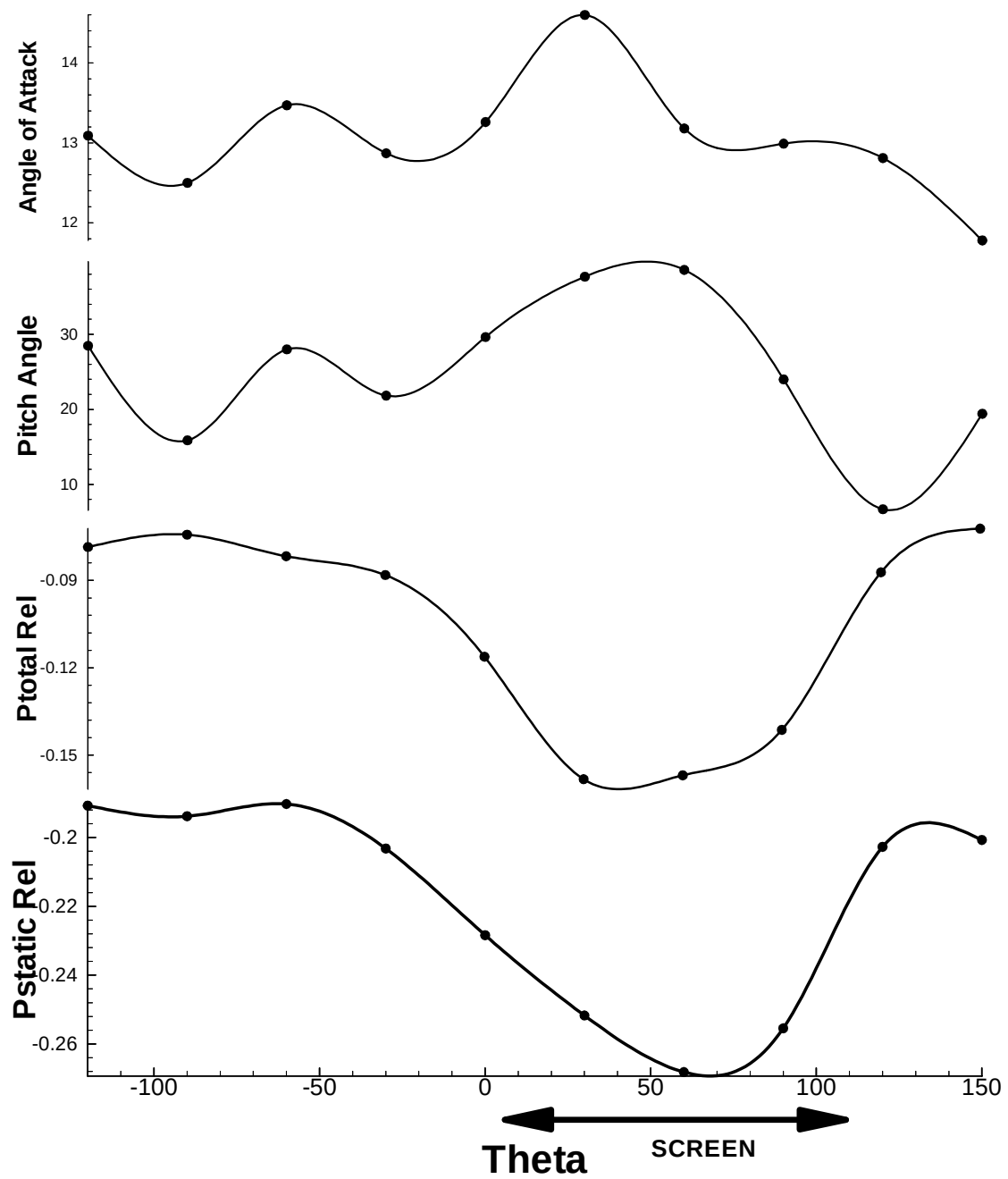


Figure A.37: Inlet Flow Characteristics for Level 2, Operating Point C at 67% Span.

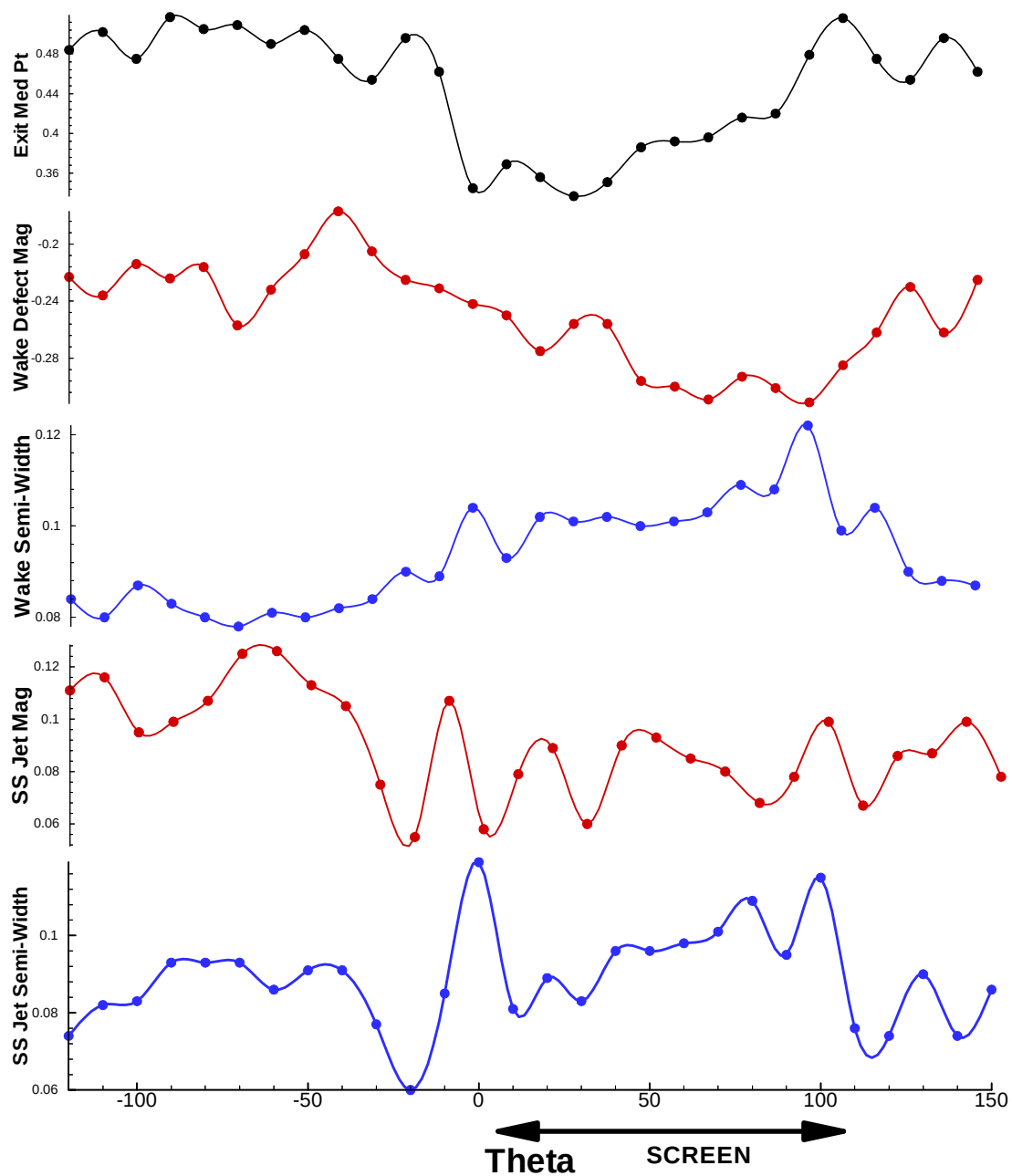


Figure A.38: Exit Flow Characteristics for Level 2, Operating Point C at 67% Span.

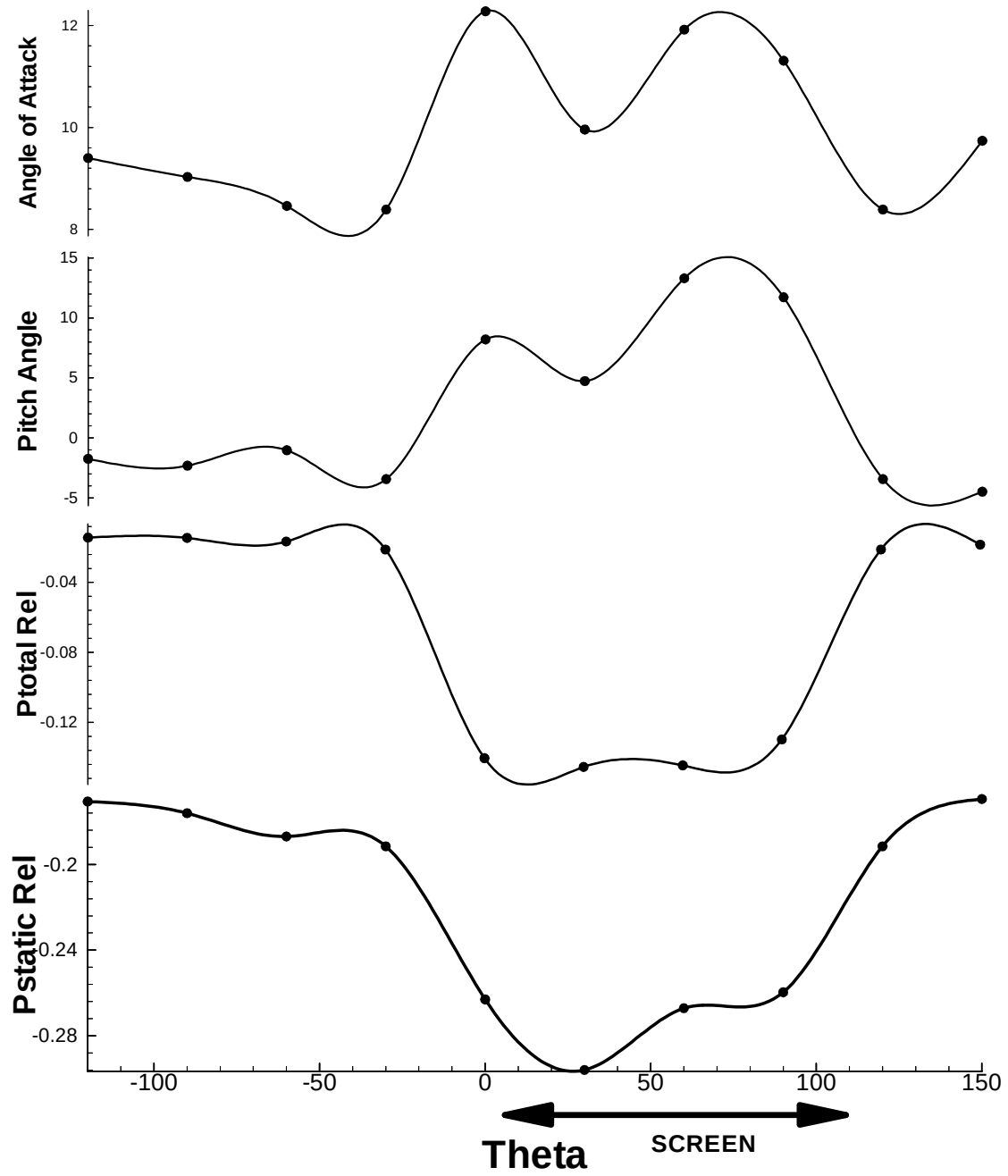


Figure A.39: Inlet Flow Characteristics for Level 3, Operating Point D at 50% Span.

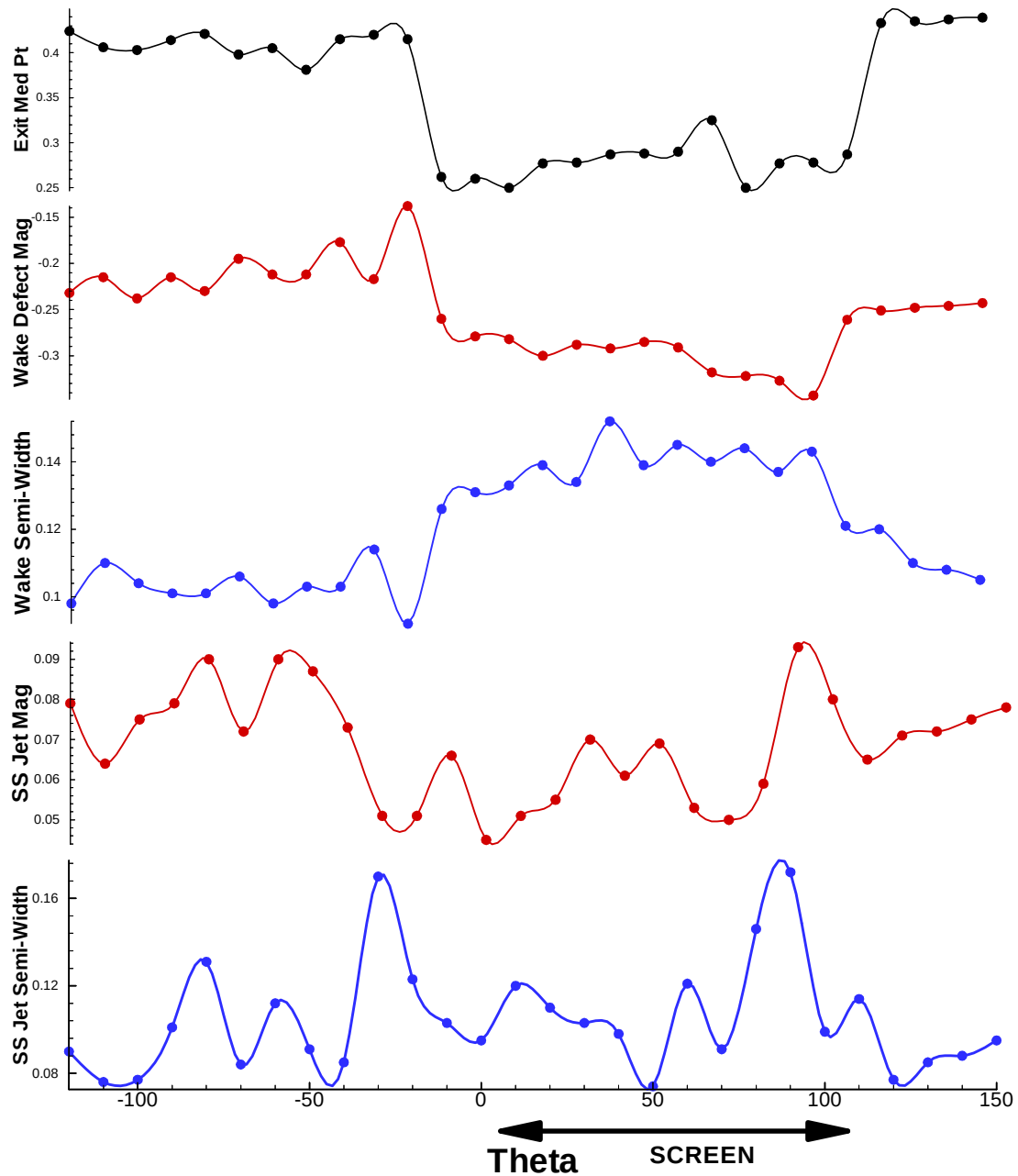


Figure A.40: Exit Flow Characteristics for Level 3, Operating Point D at 50% Span.

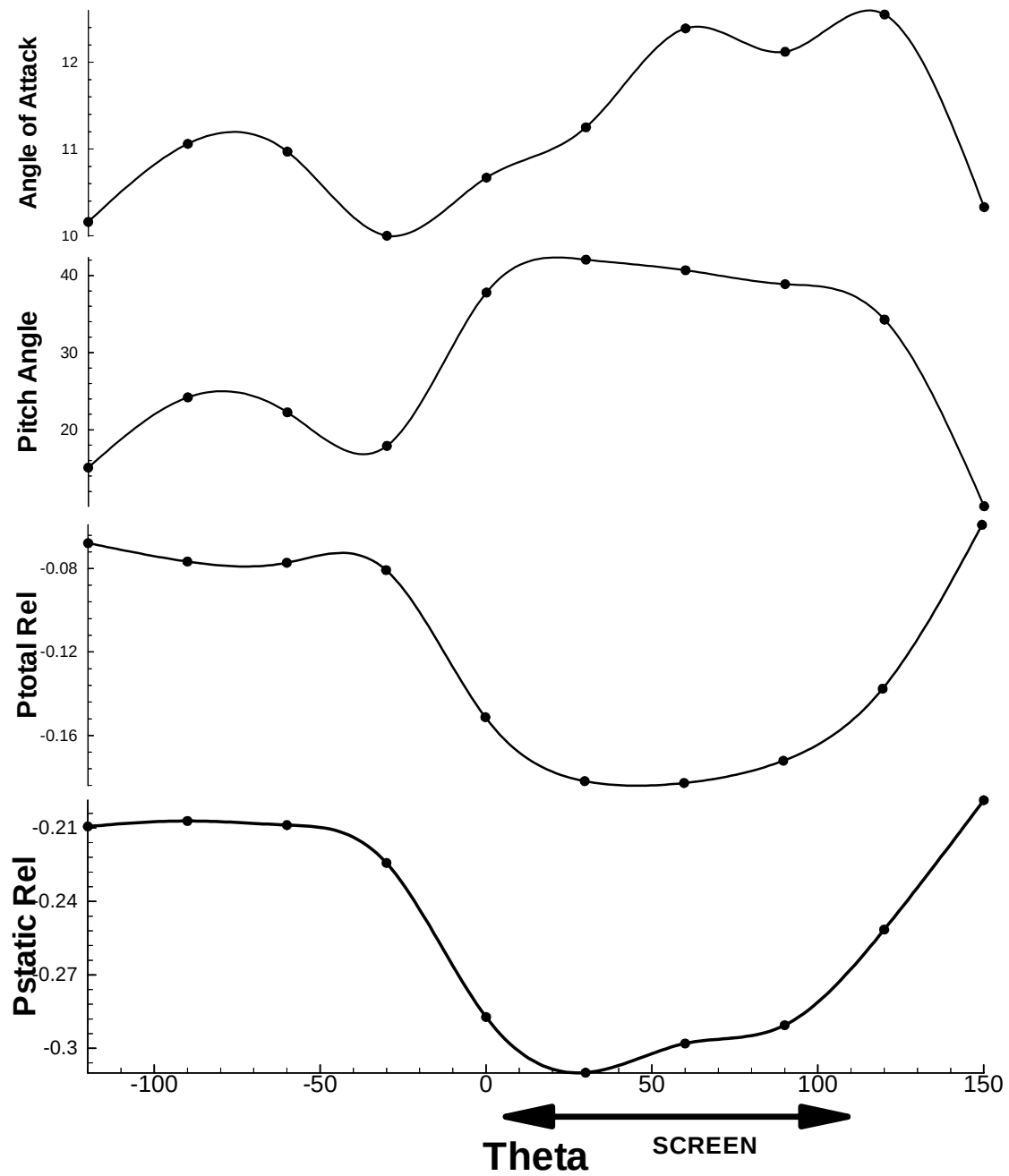


Figure A.41: Inlet Flow Characteristics for Level 3, Operating Point D at 33% Span.

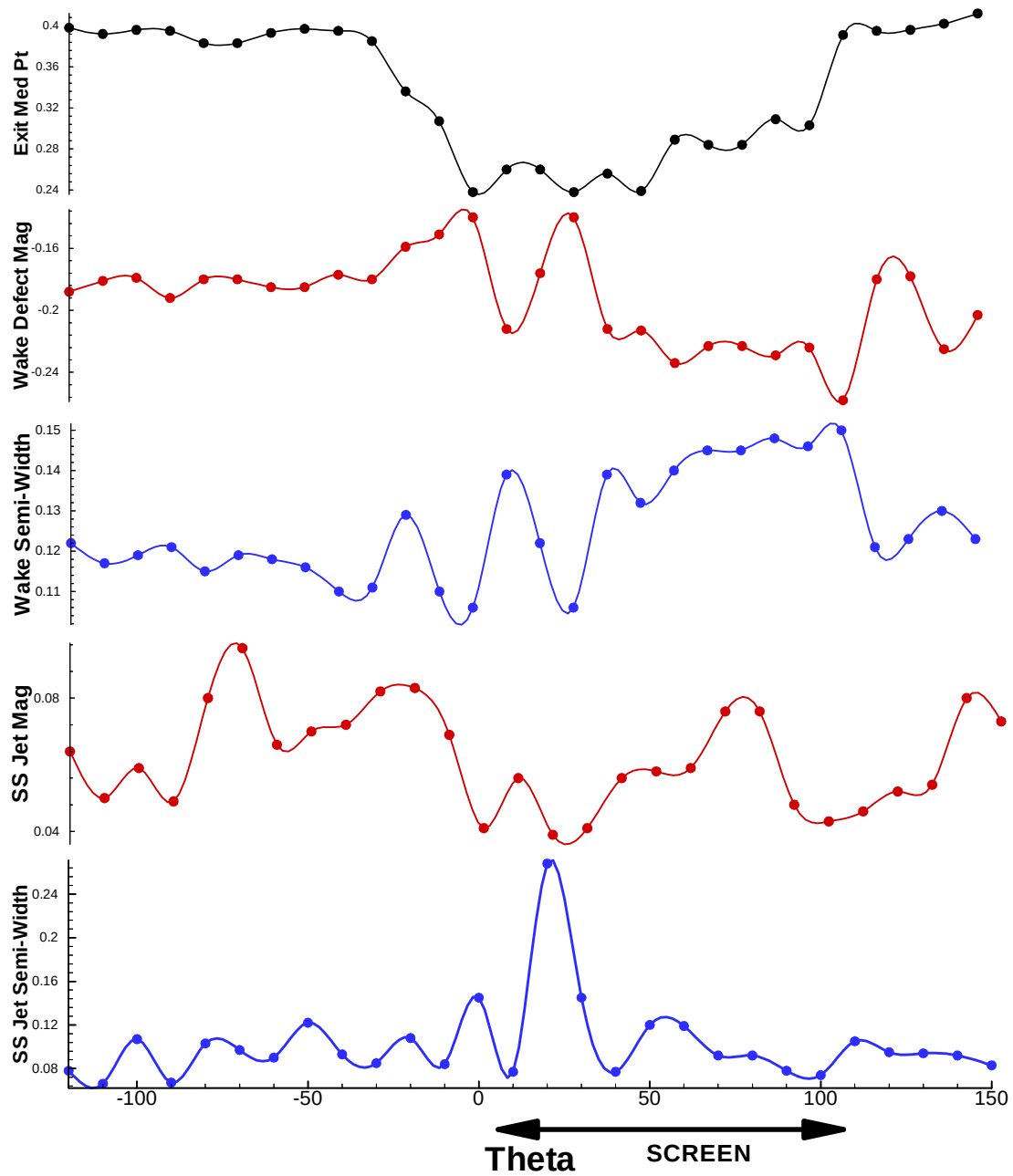


Figure A.42: Exit Flow Characteristics for Level 3, Operating Point D at 33% Span.

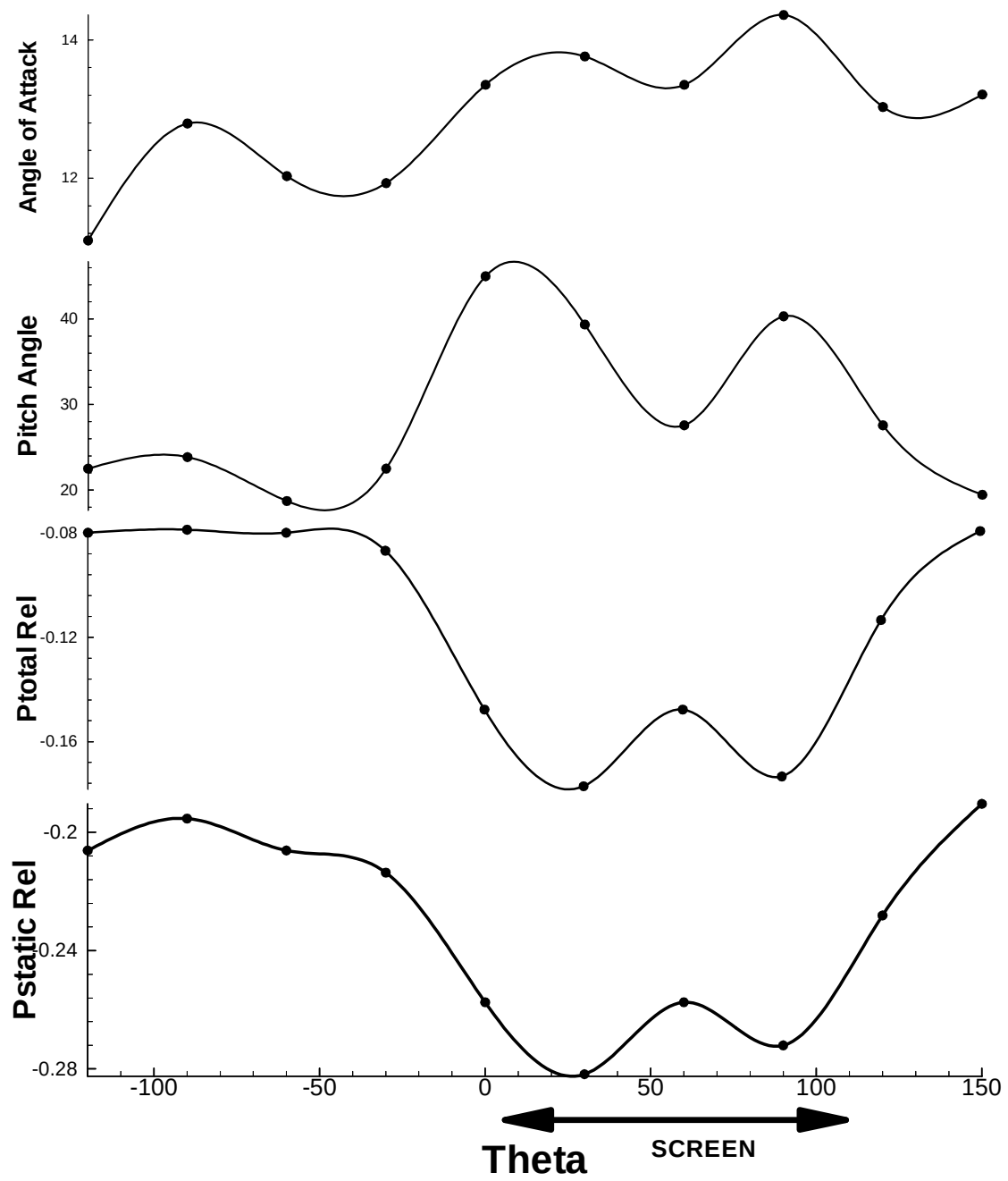


Figure A.43: Inlet Flow Characteristics for Level 3, Operating Point D at 67% Span.

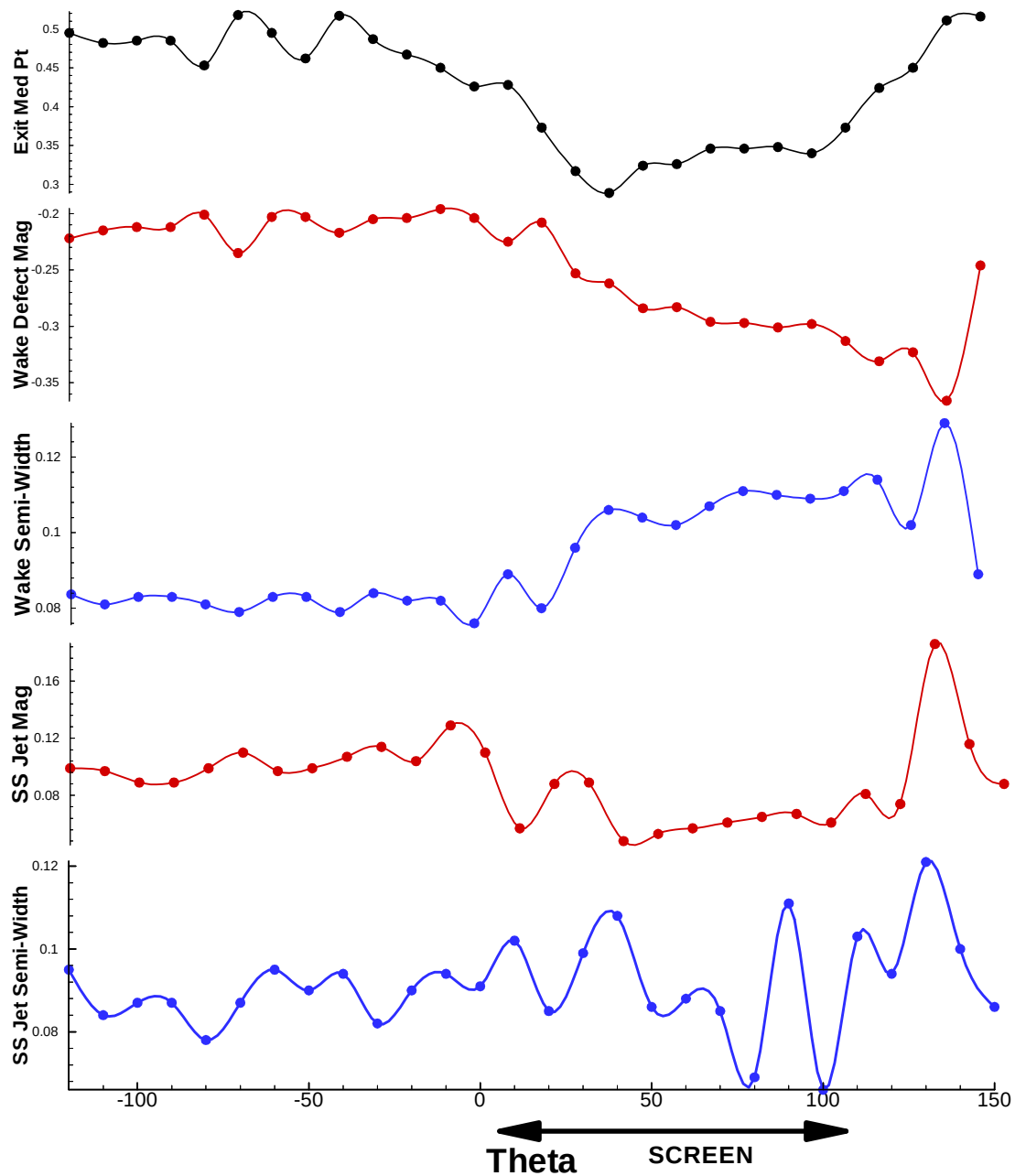


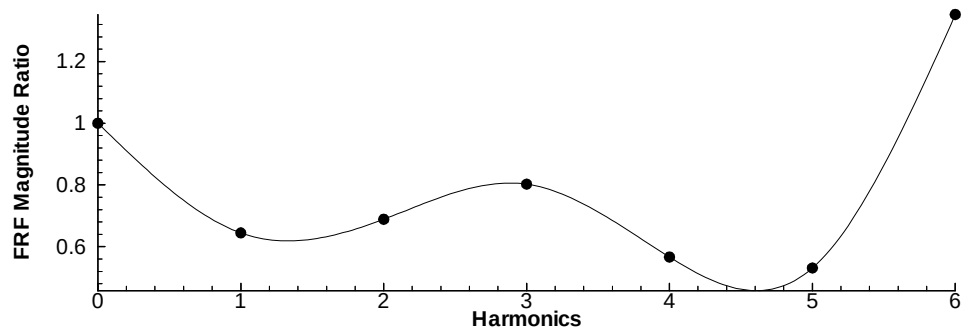
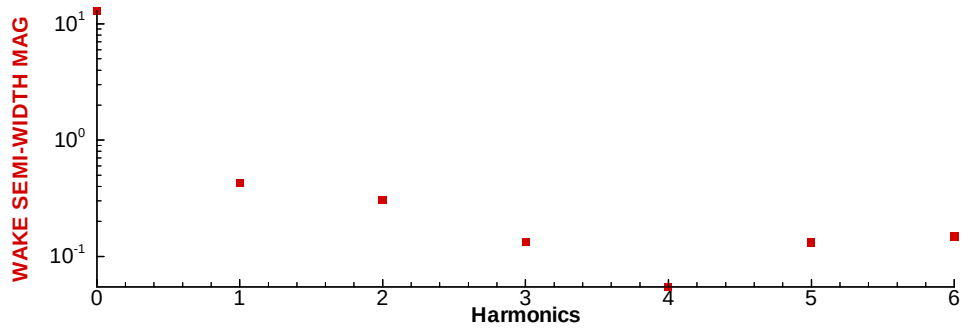
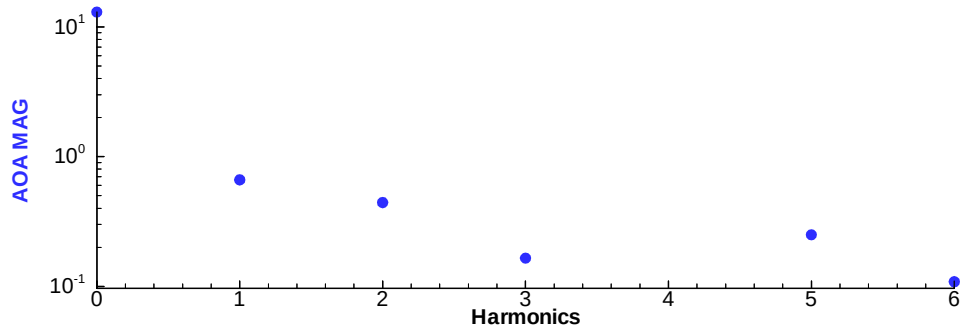
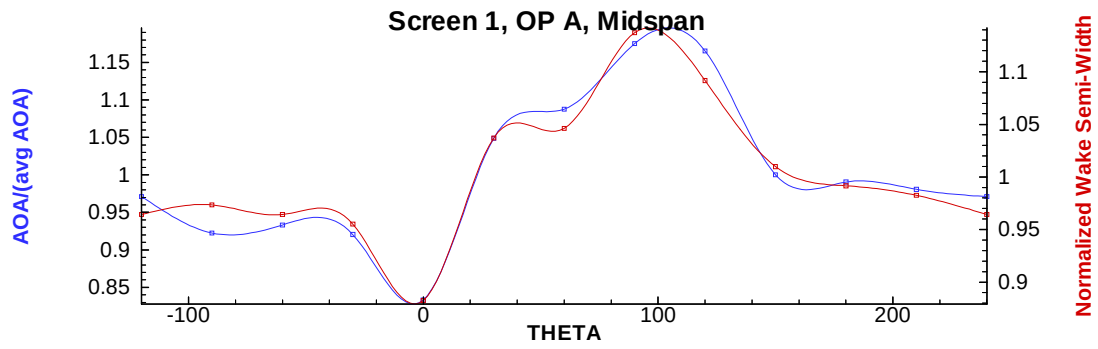
Figure A.44: Exit Flow Characteristics for Level 3, Operating Point D at 67% Span.

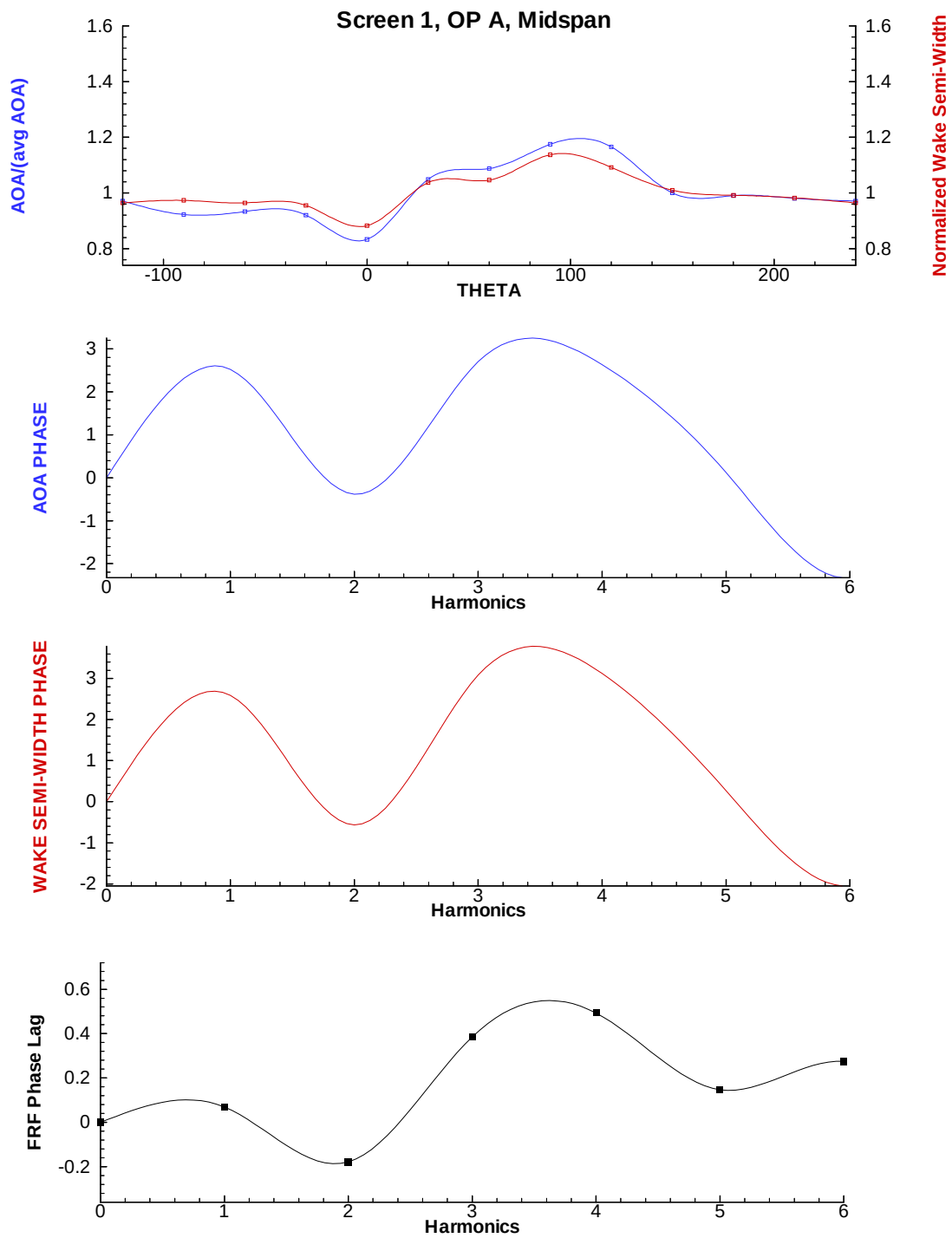
11 Appendix B

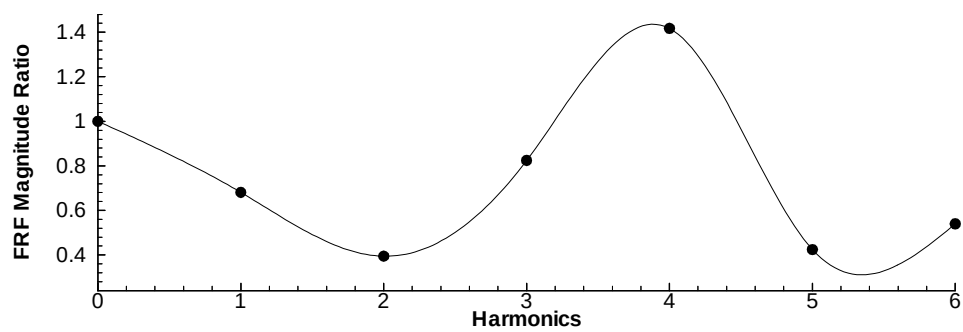
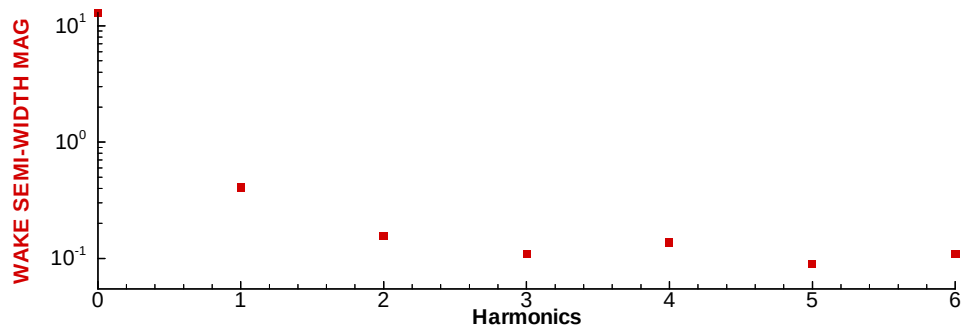
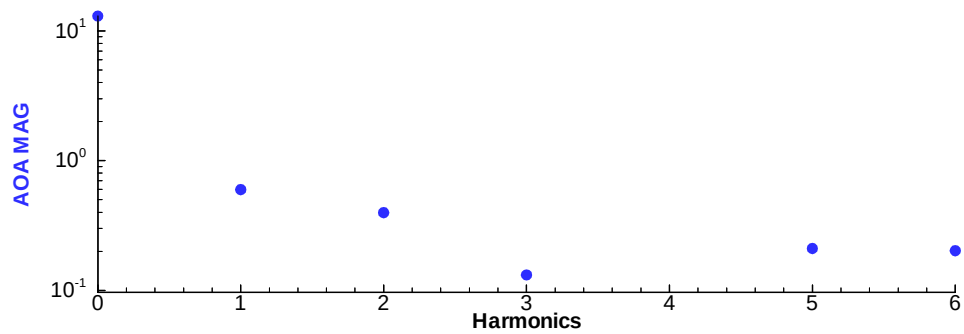
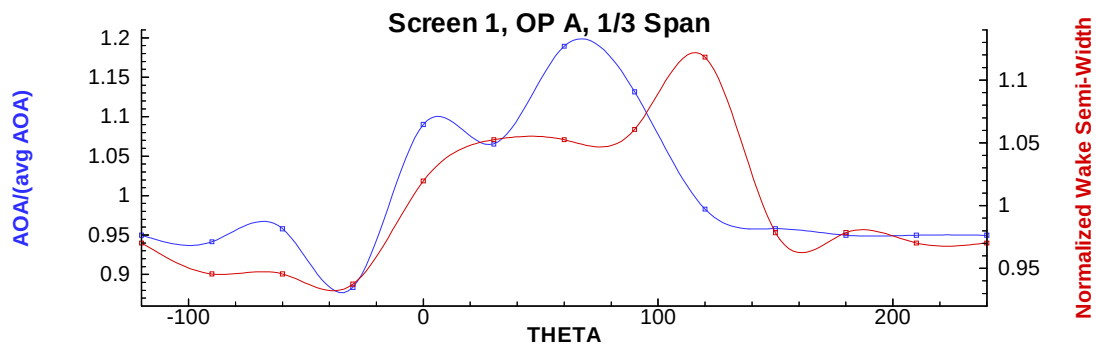
Frequency Response Functions based on Wake Semi-width and Angle of Attack

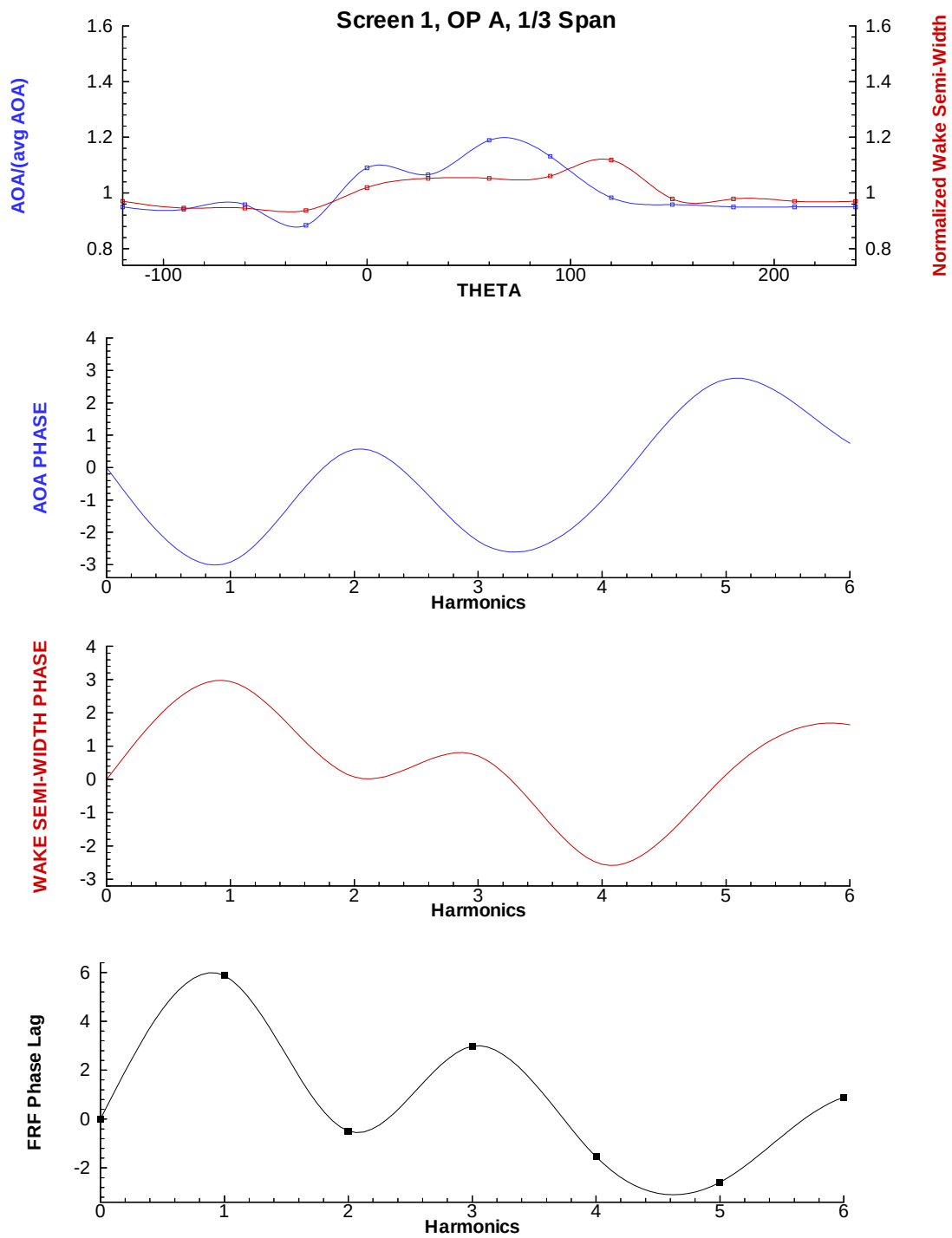
DATA**LOCATION**

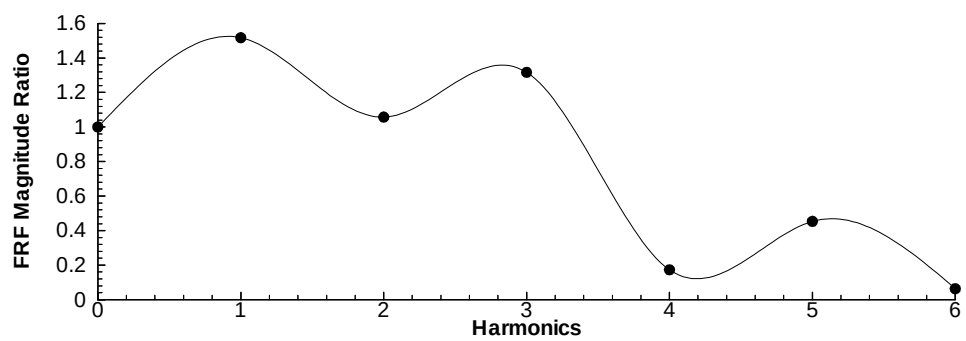
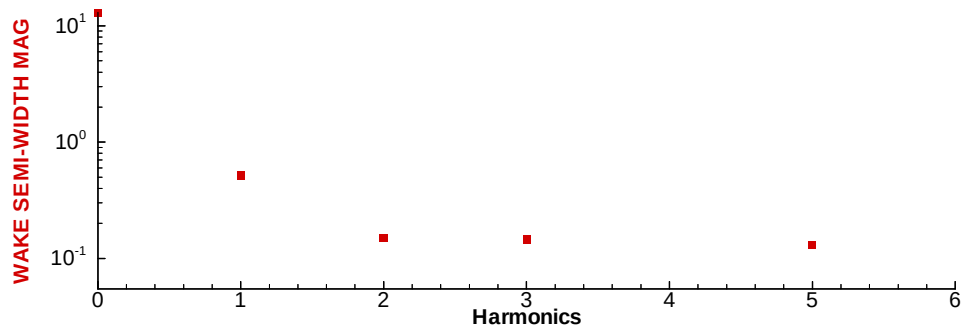
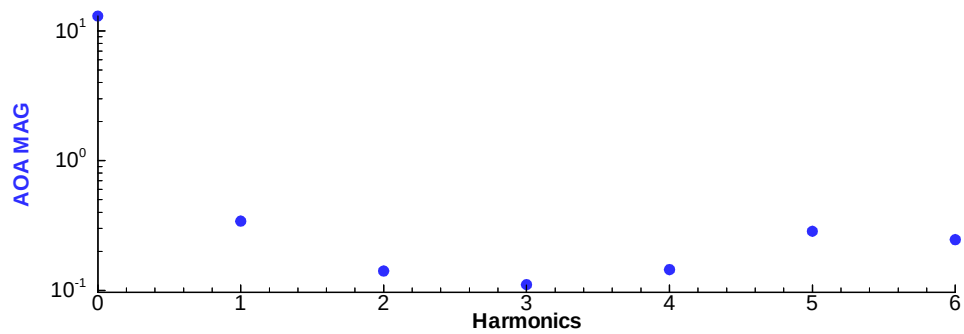
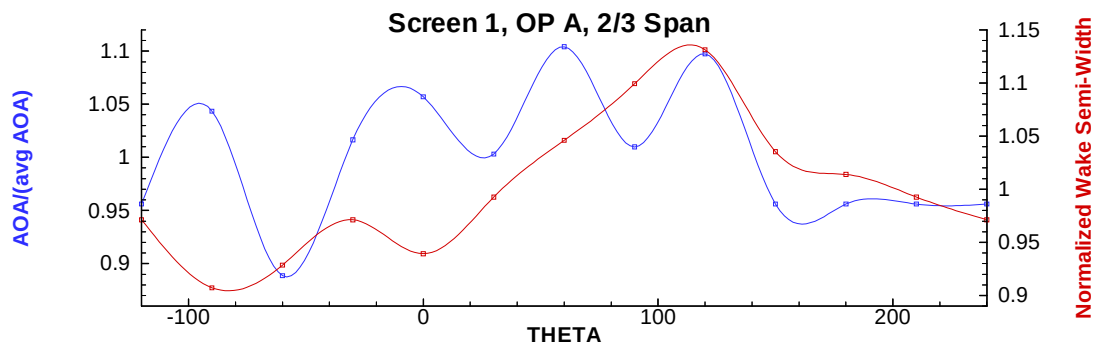
Distortion Screen	Operating Point	Measurement Span	Page Numbers
Level 1	Operating Point A	50 %, 33 %, 67 %	pp. 125-130
Level 1	Operating Point B	50 %, 33 %, 67 %	pp. 131-136
Level 1	Operating Point C	50 %, 33 %, 67 %	pp. 137-142
Level 2	Operating Point A	50 %, 33 %, 67 %	pp. 143-148
Level 2	Operating Point B	50 %, 33 %, 67 %	pp. 149-154
Level 2	Operating Point C	50 %, 33 %, 67 %	pp. 155-160
Level 3	Operating Point D	50 %, 33 %, 67 %	pp. 161-166

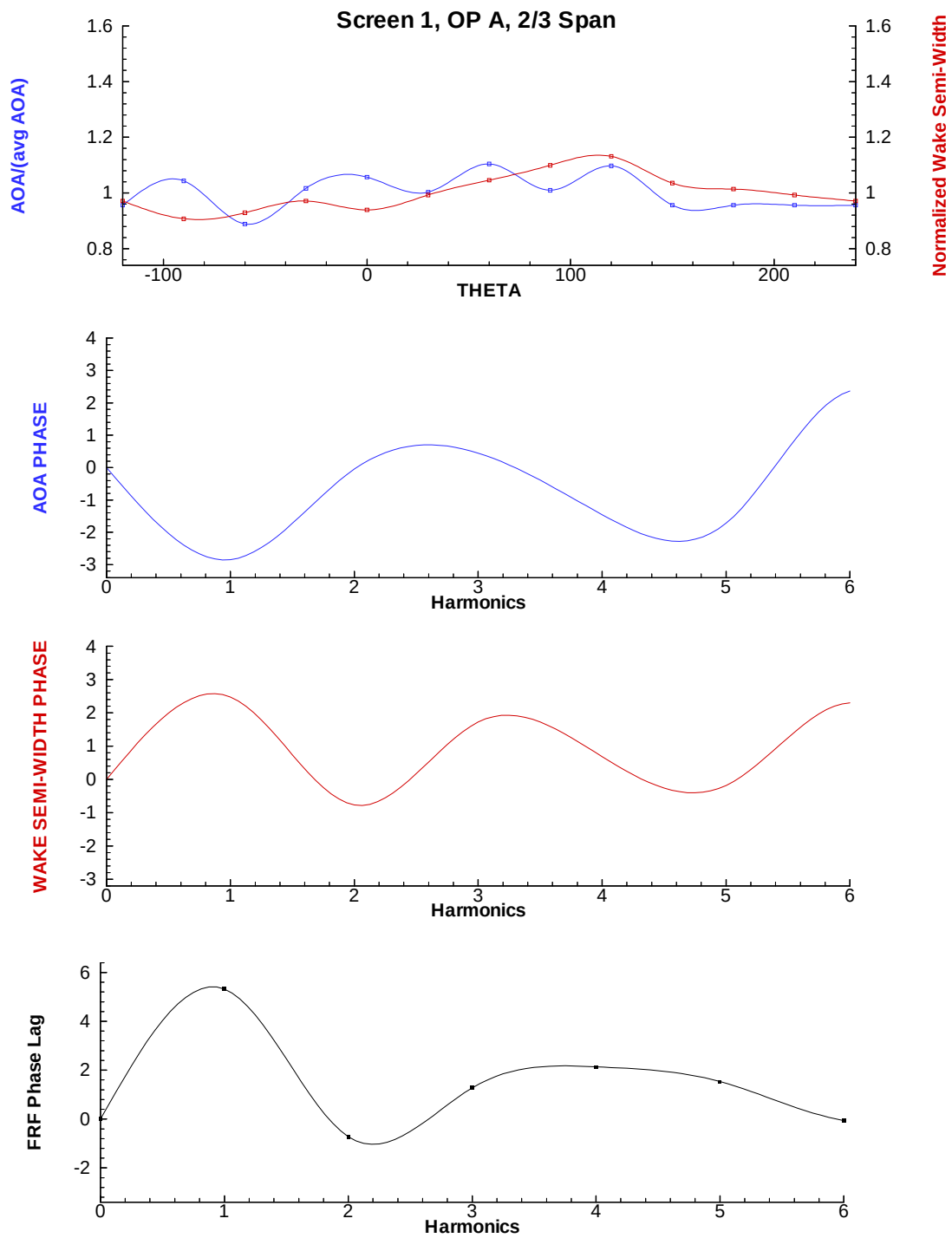


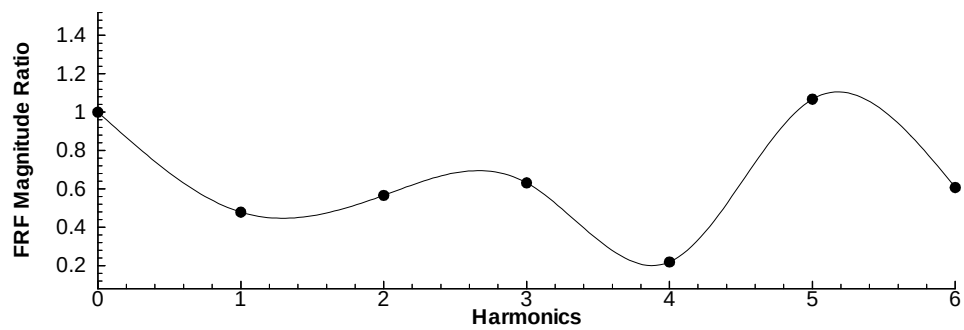
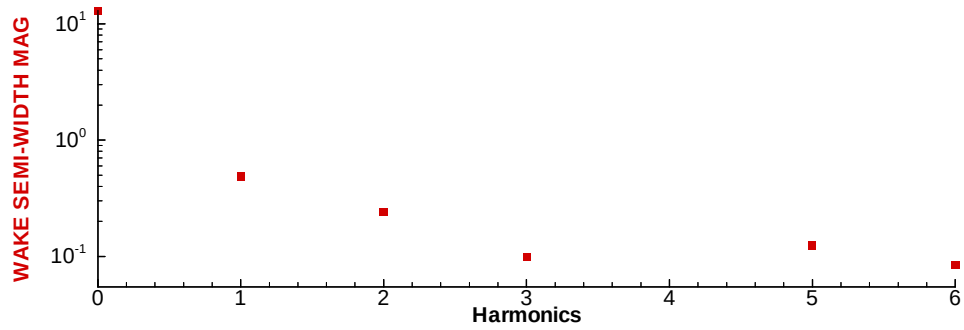
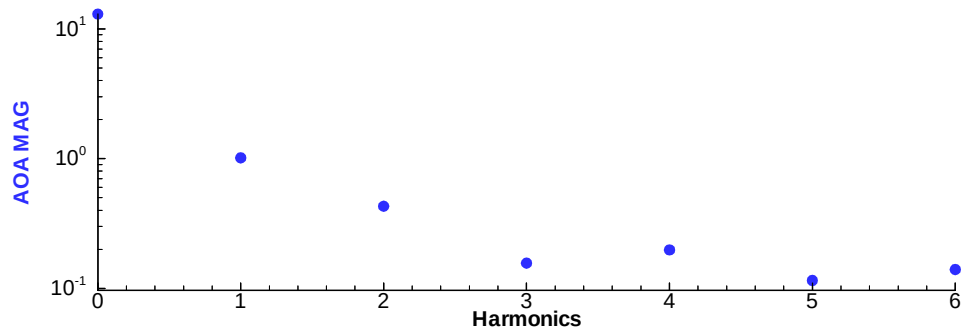
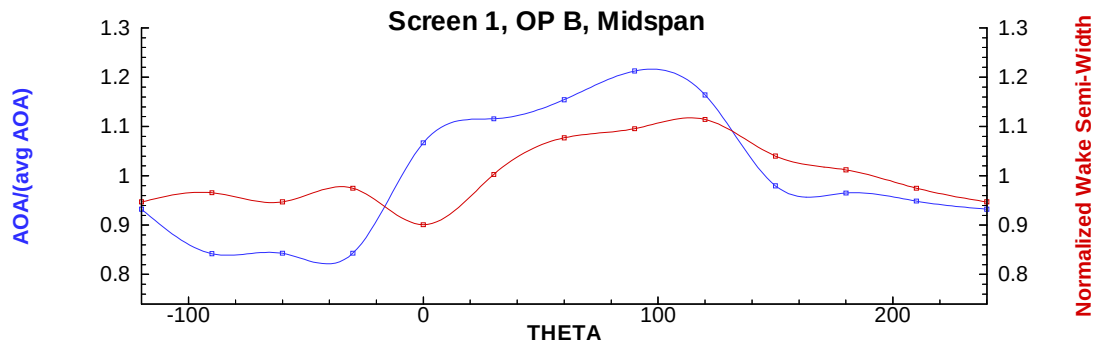


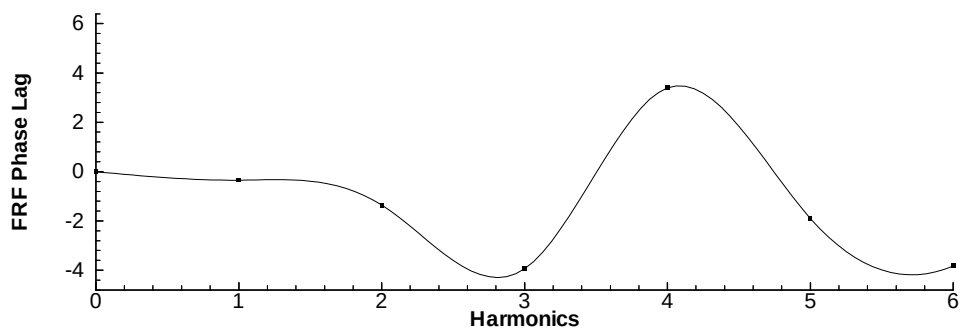
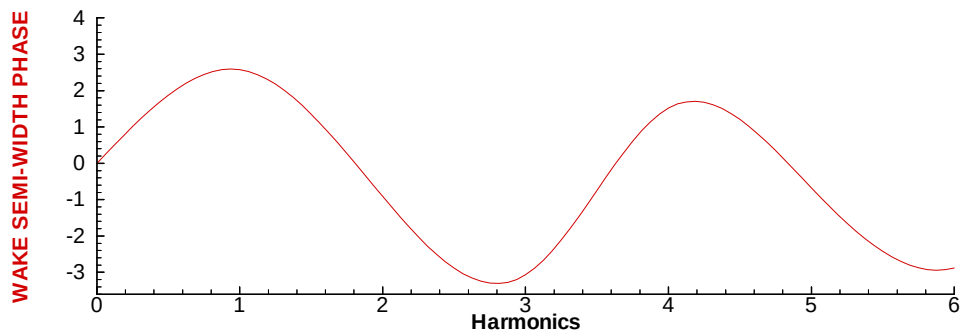
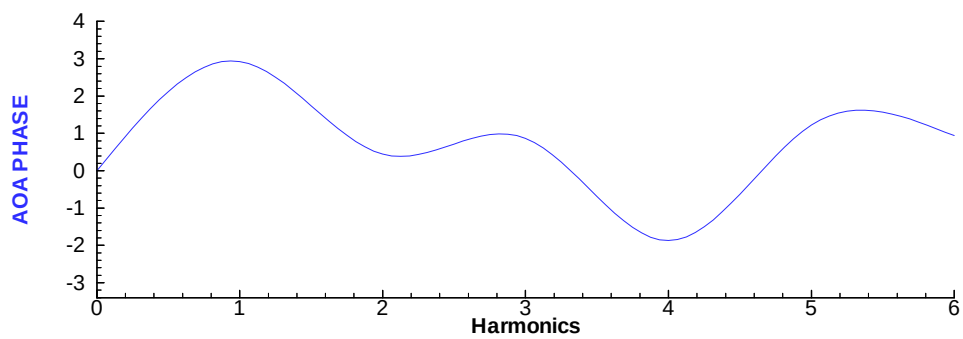
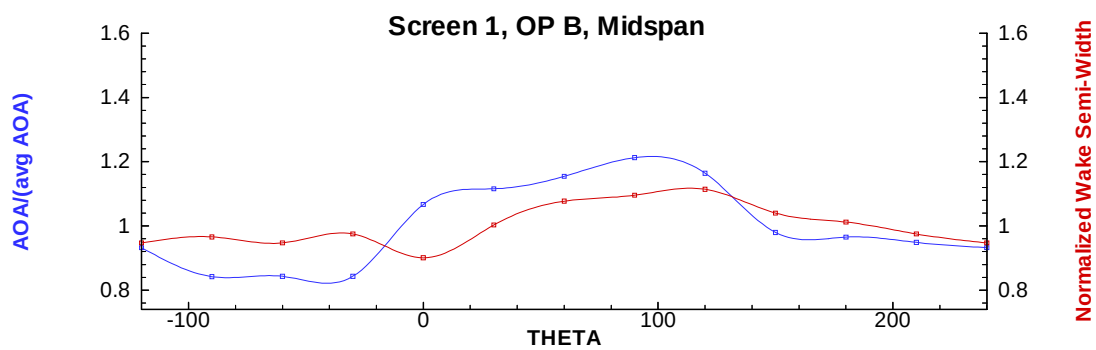


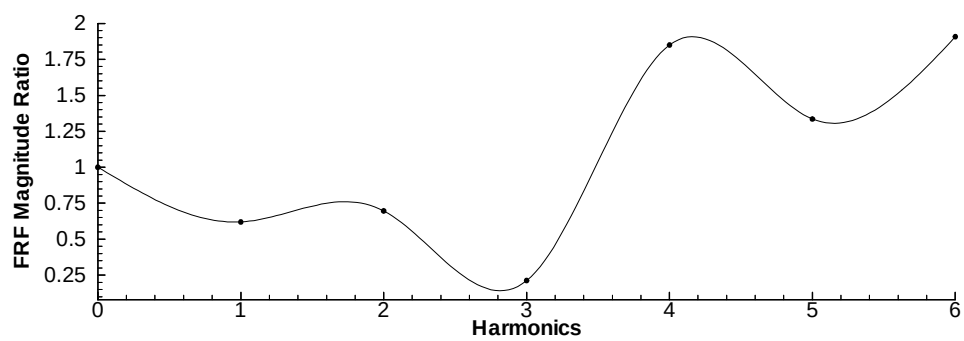
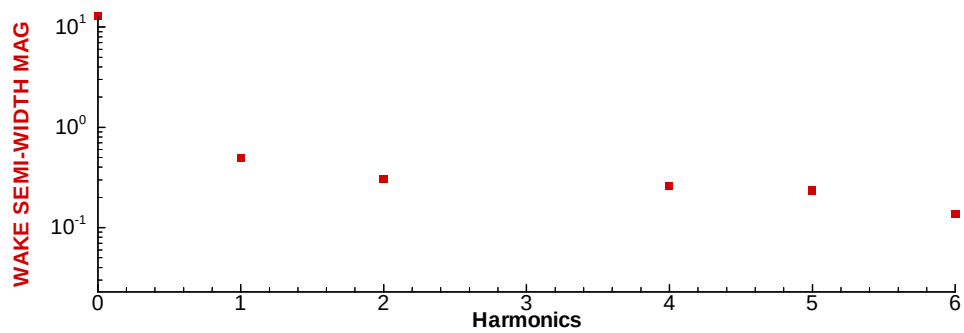
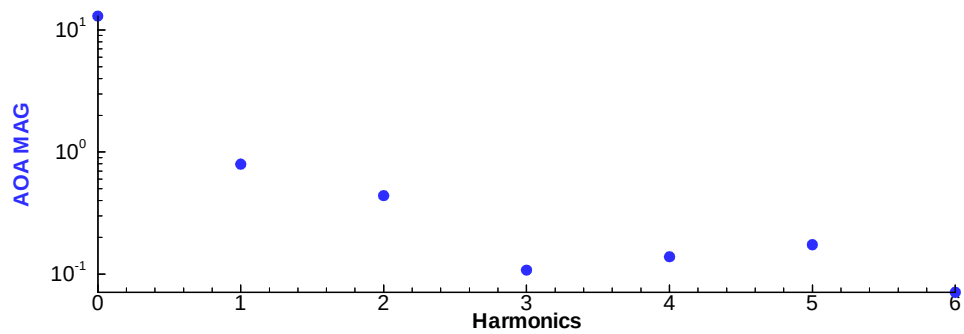
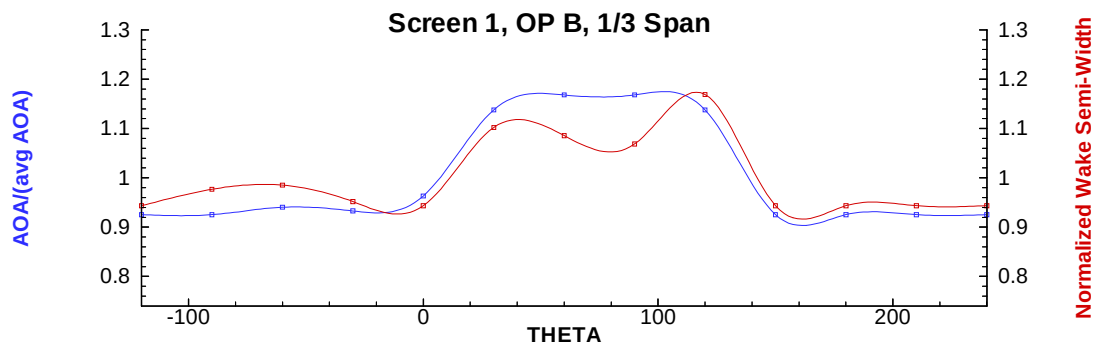


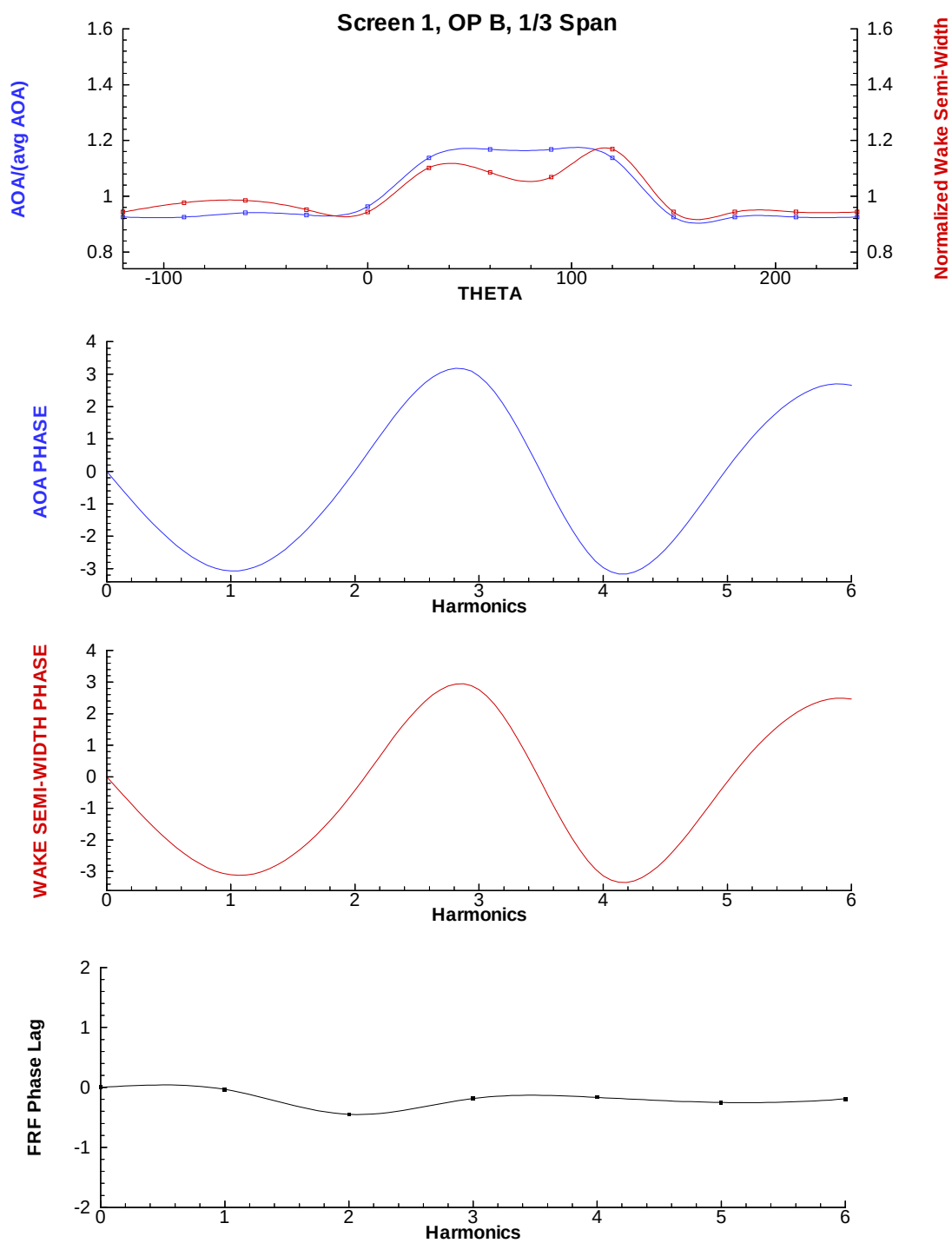


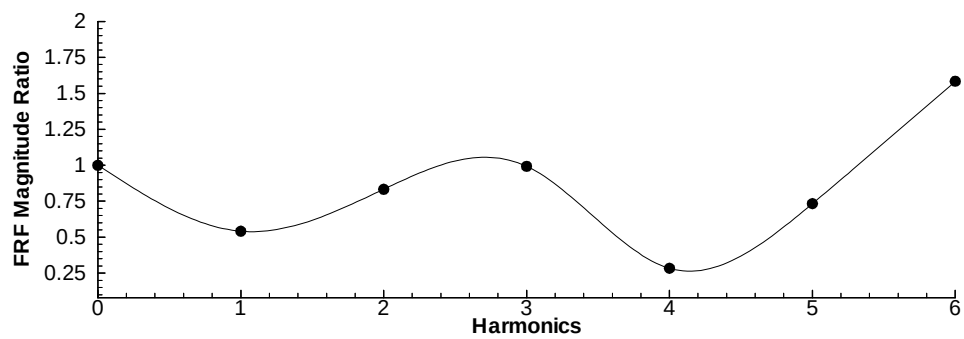
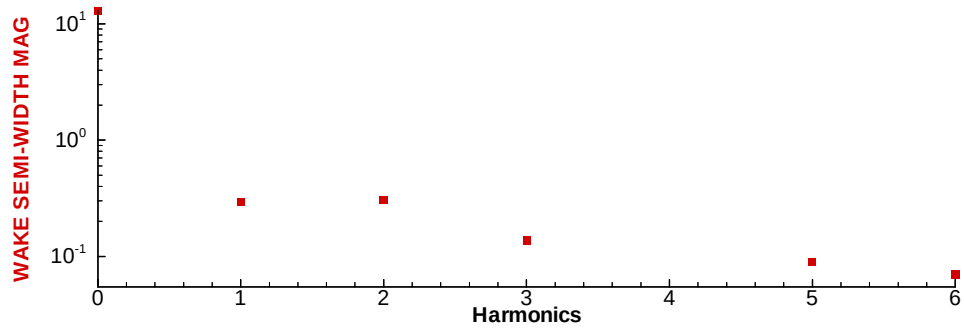
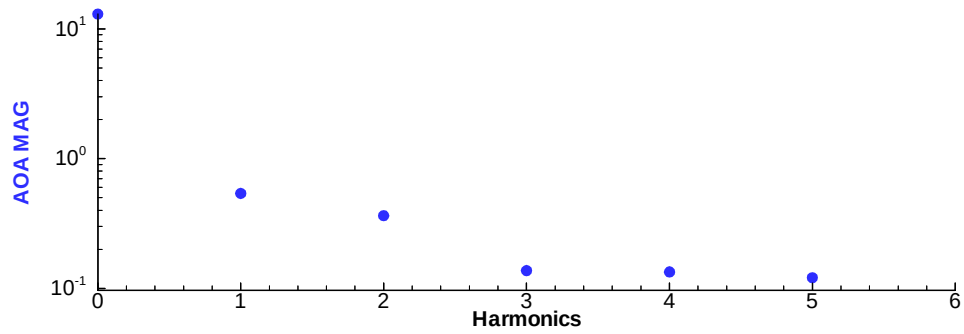
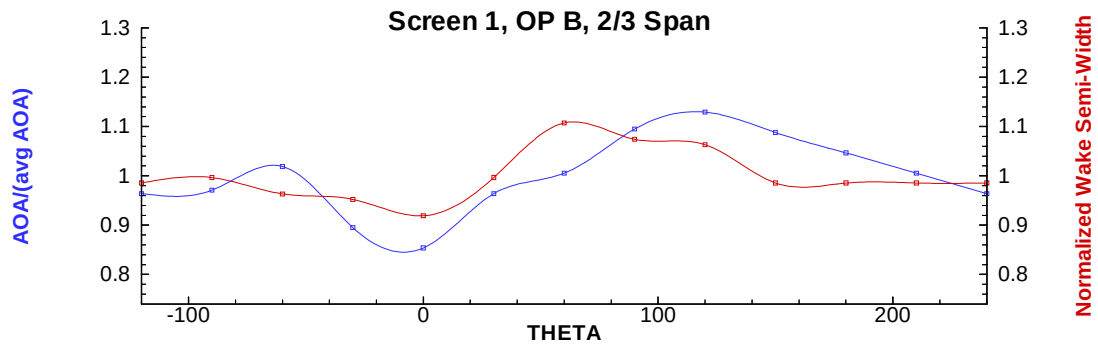


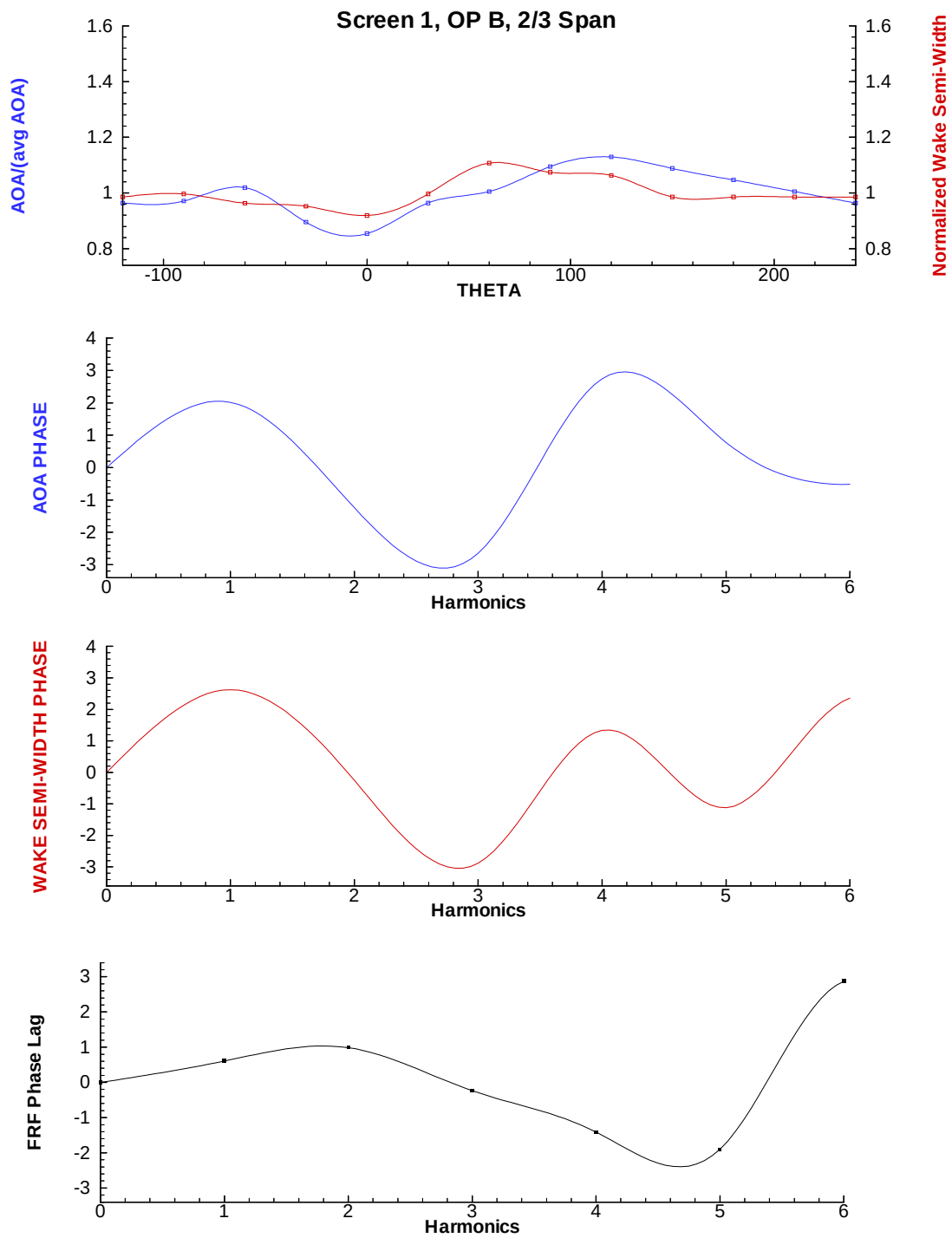


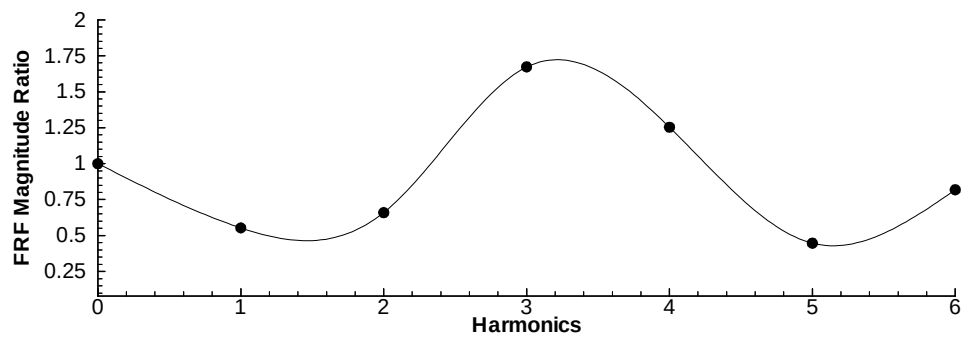
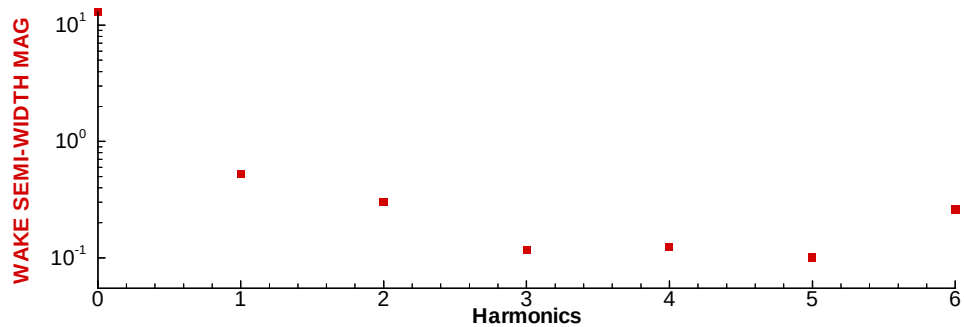
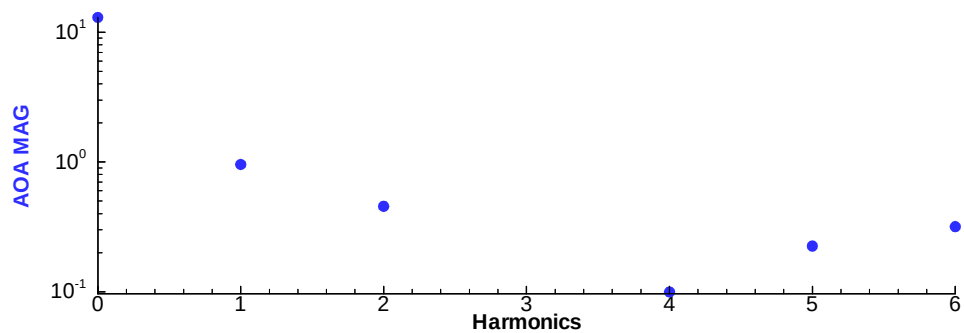
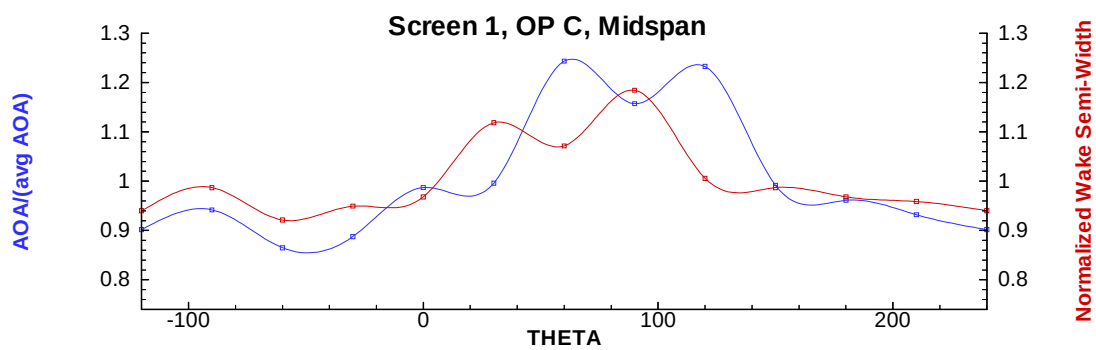


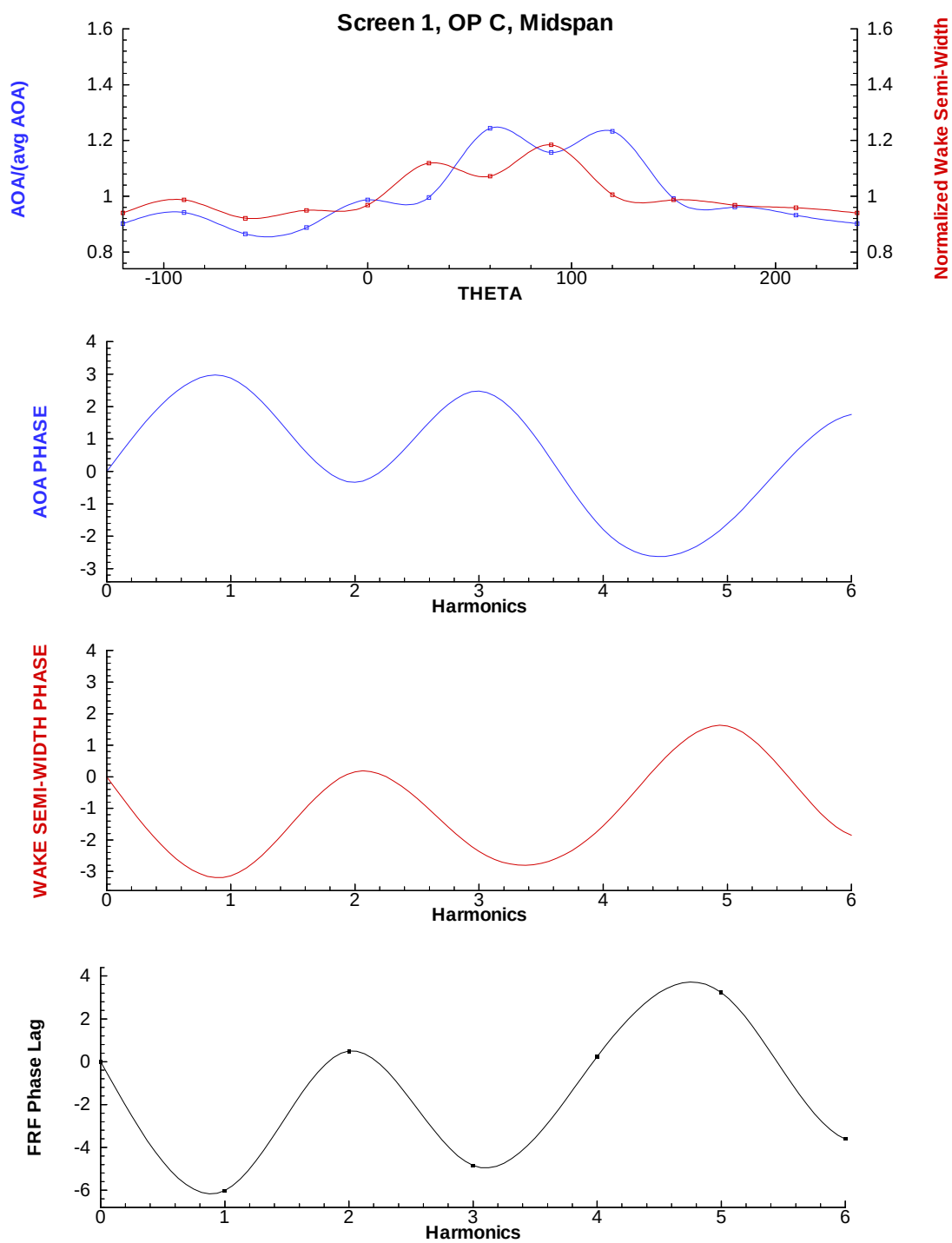


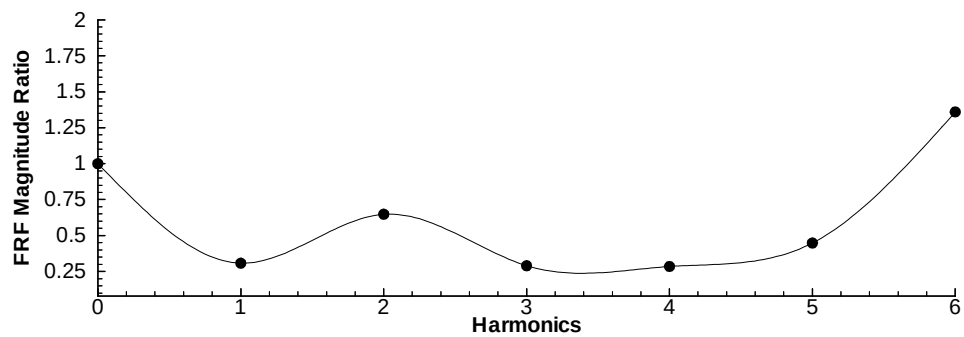
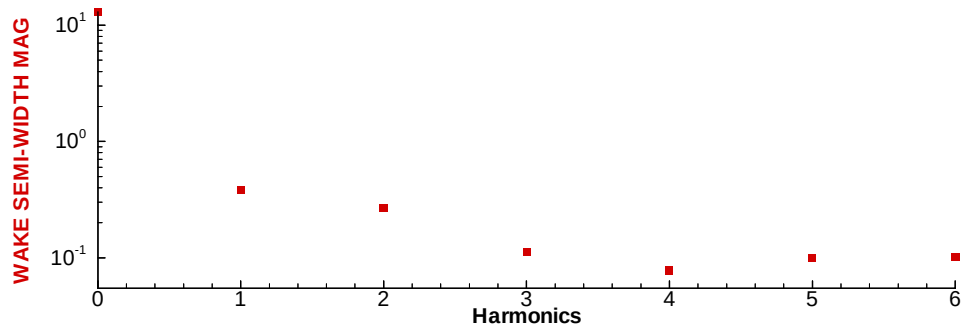
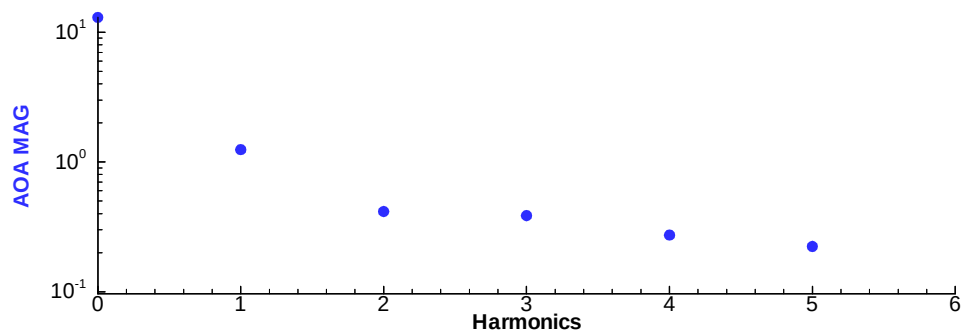
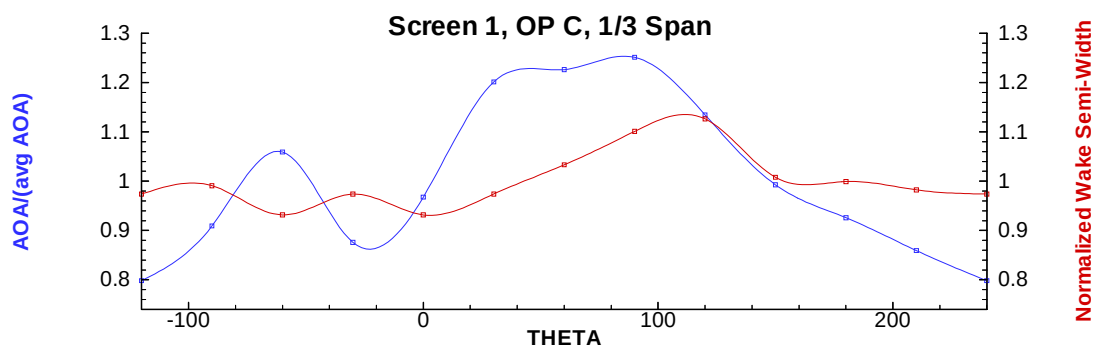


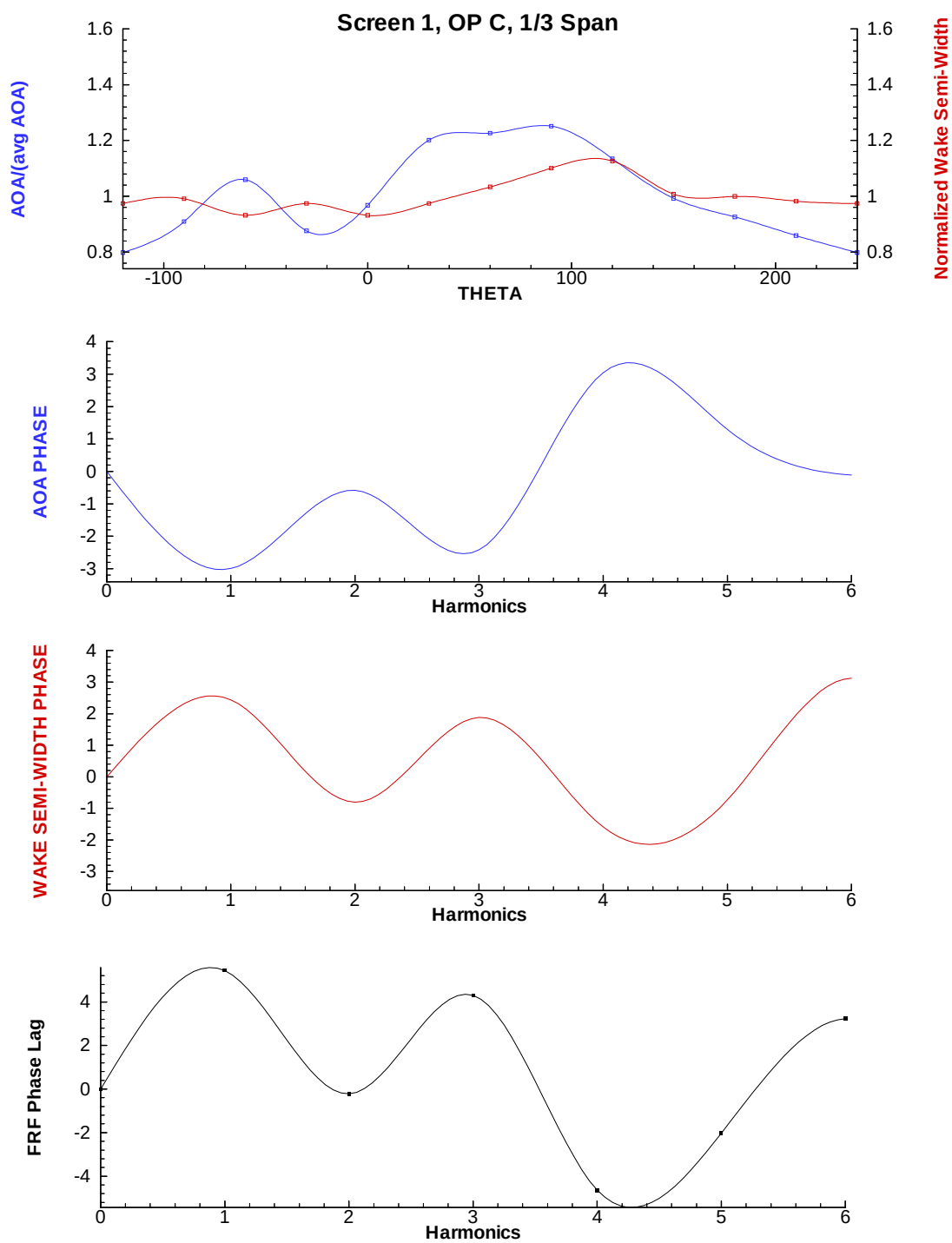


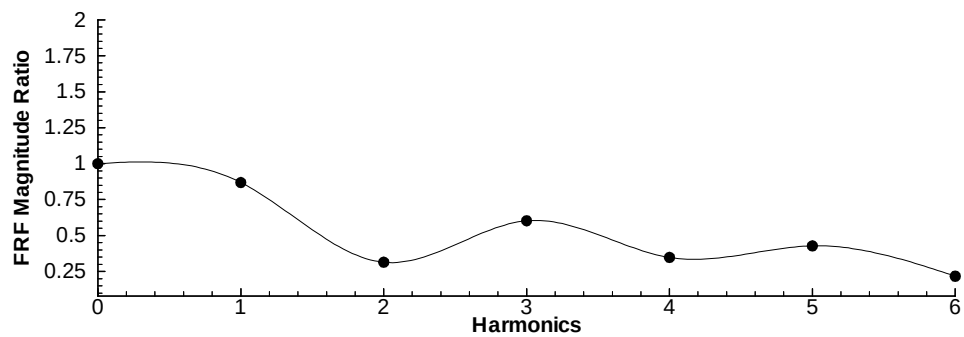
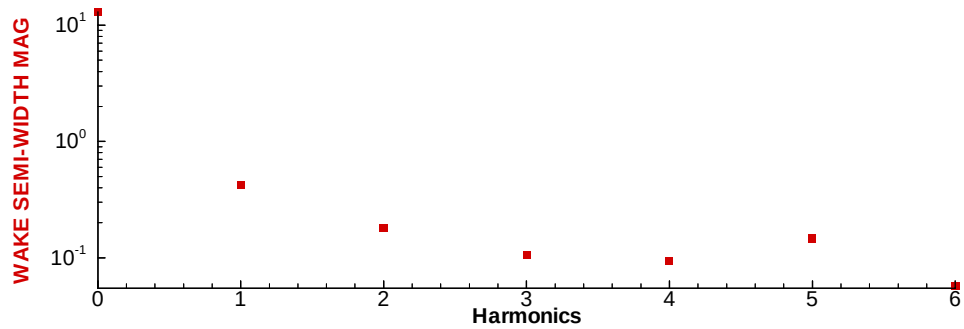
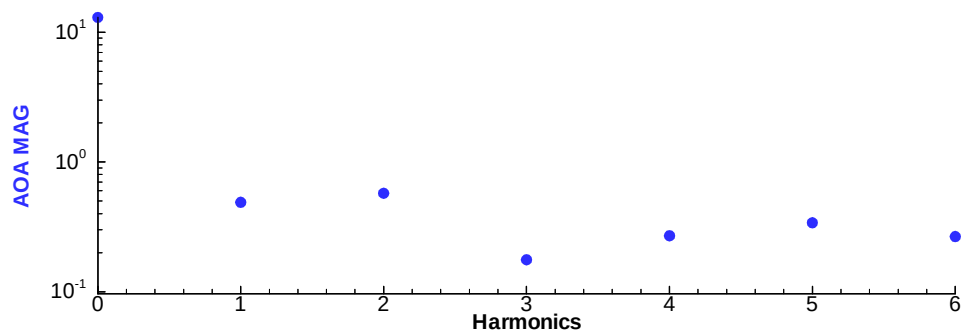
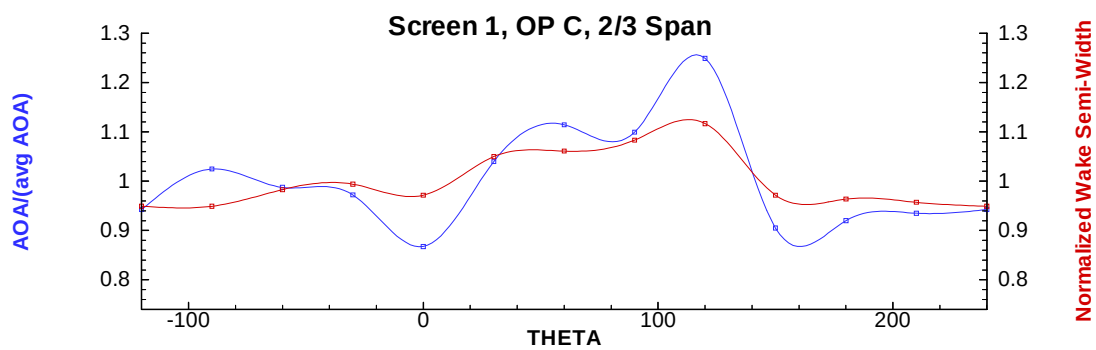


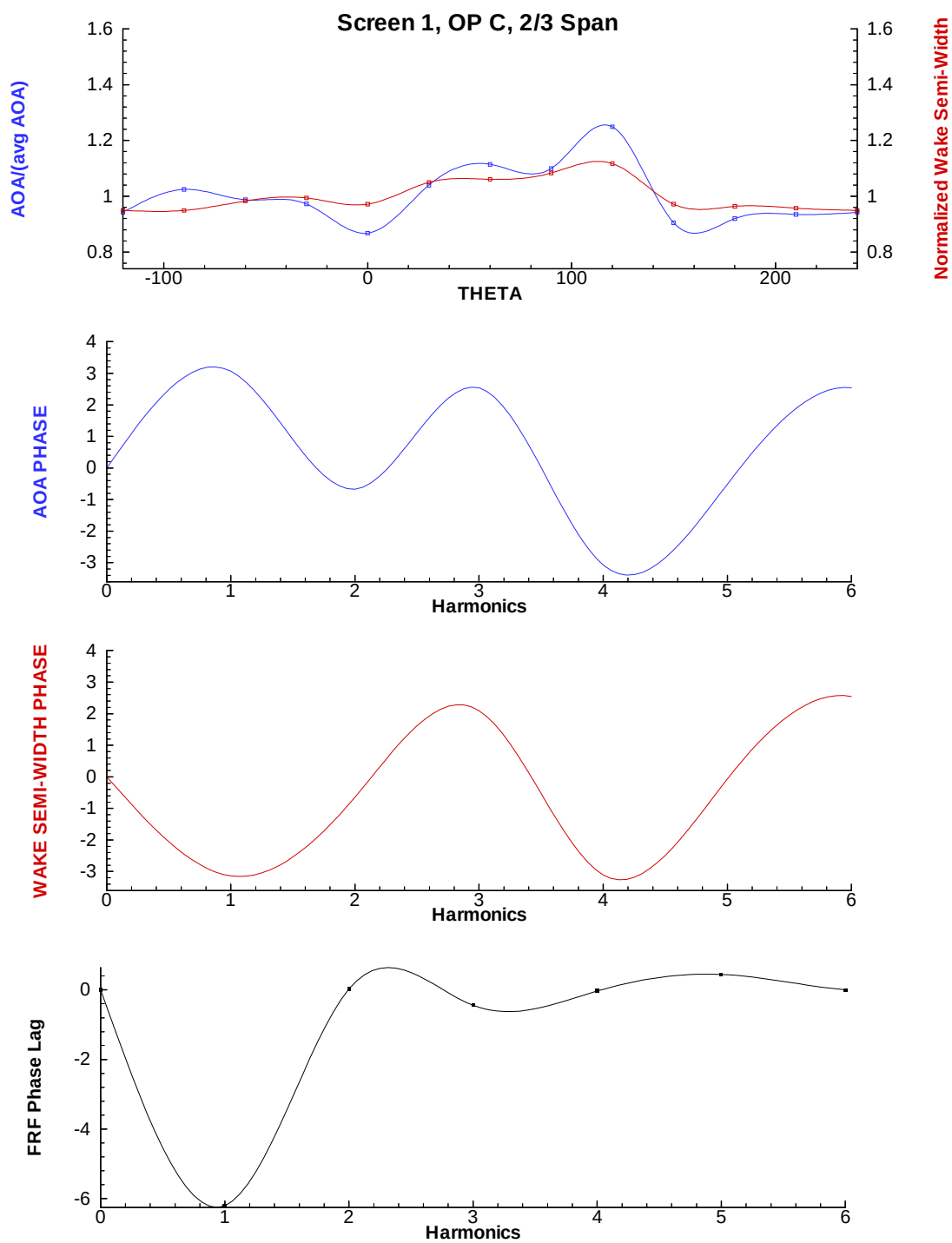


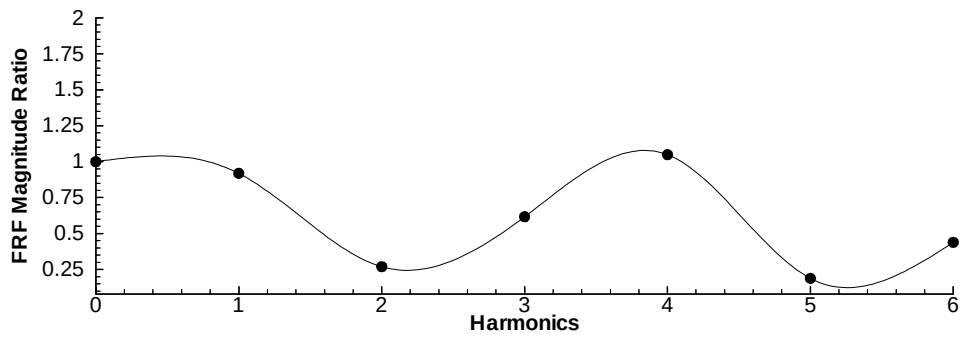
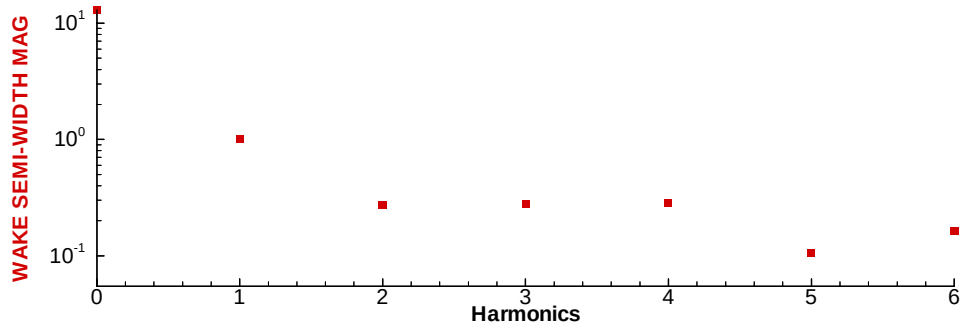
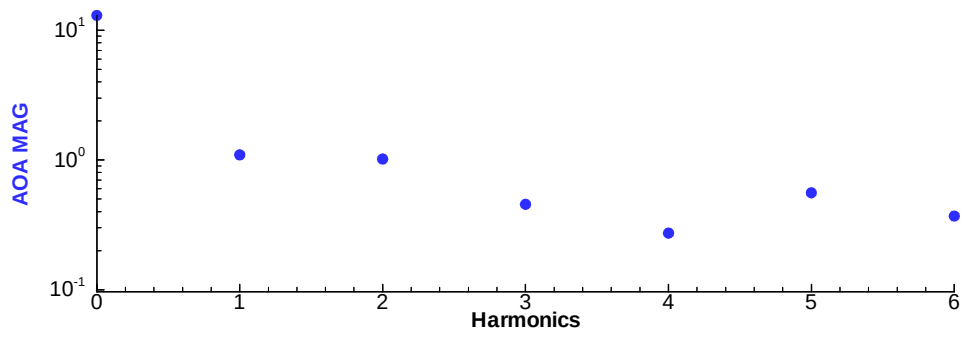
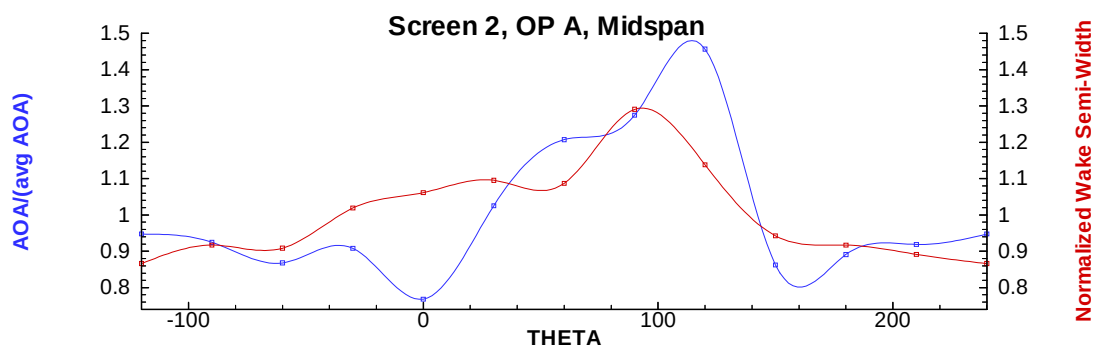


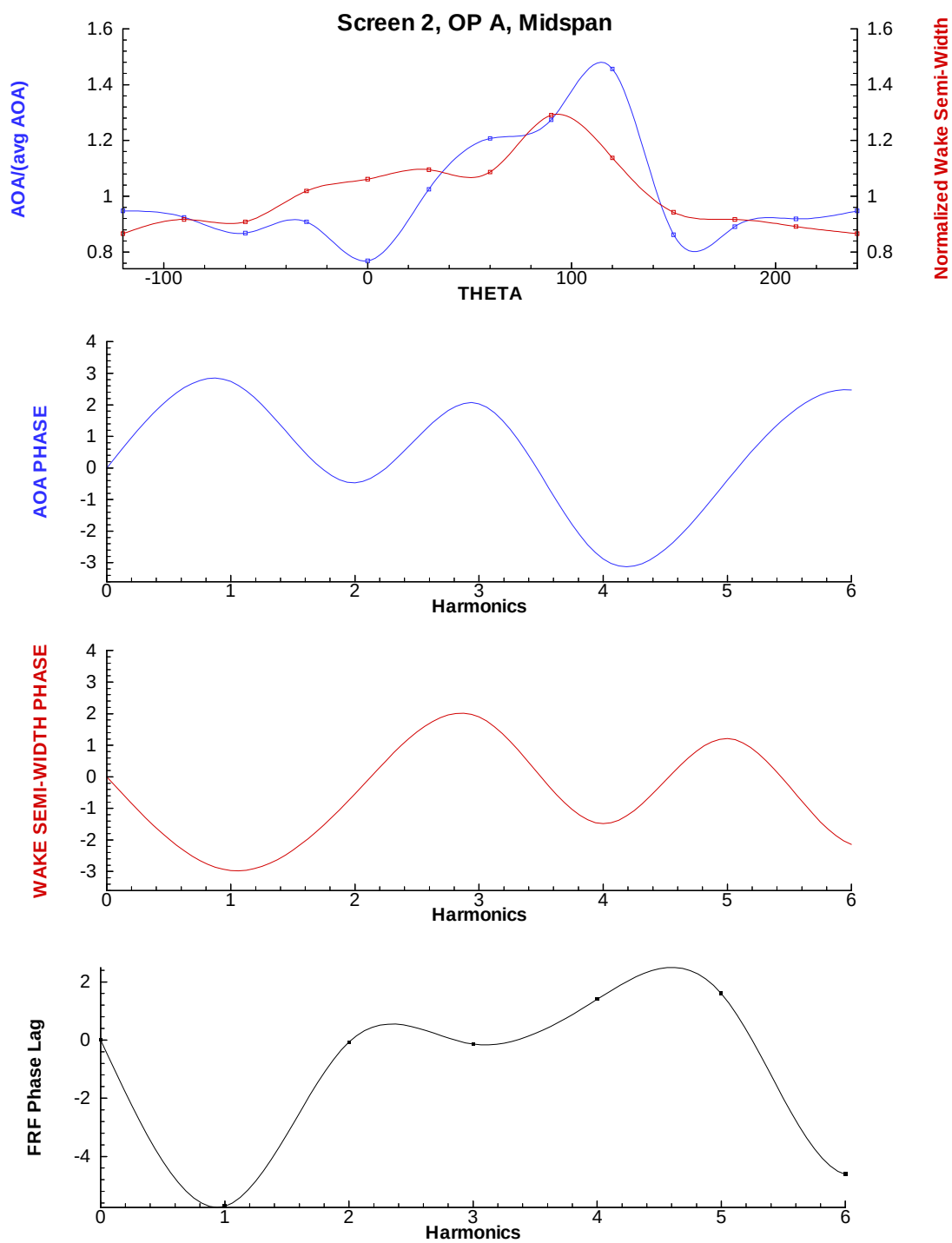


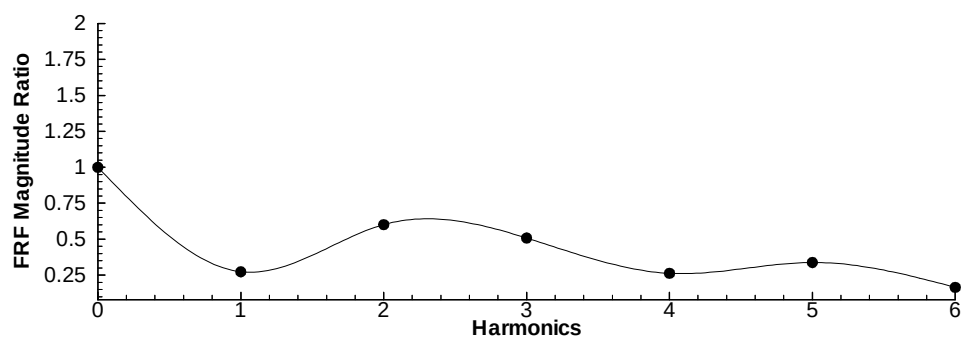
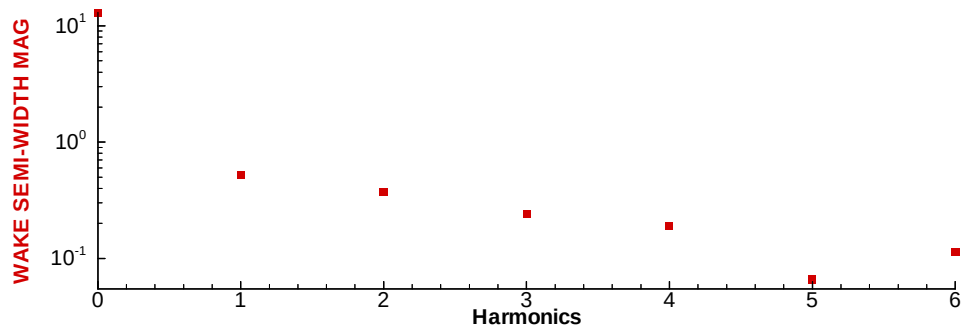
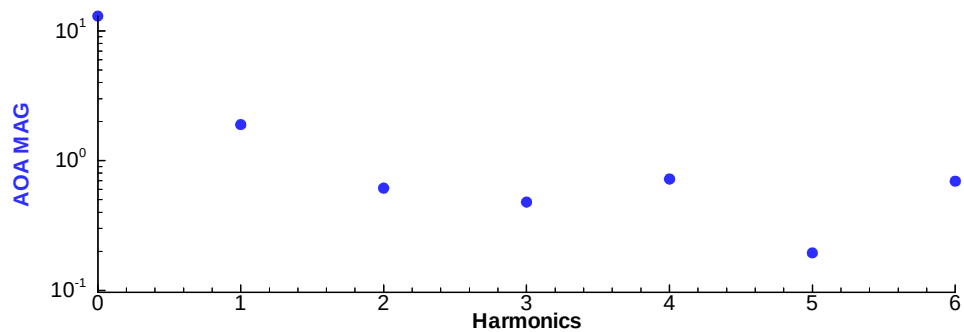
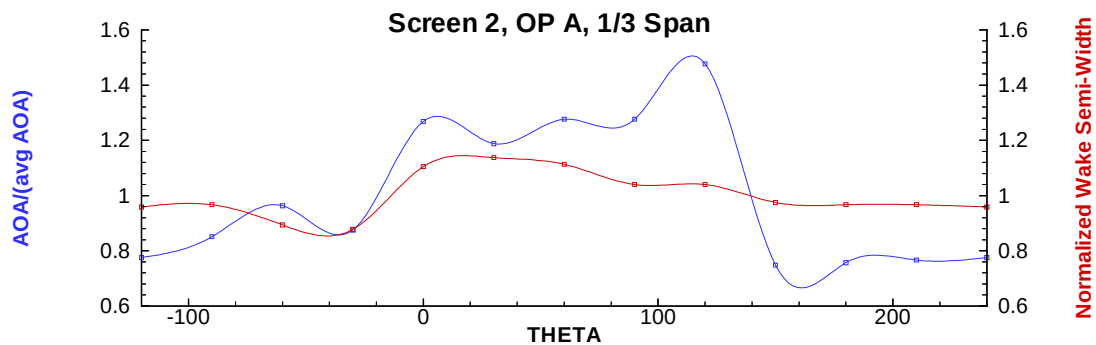


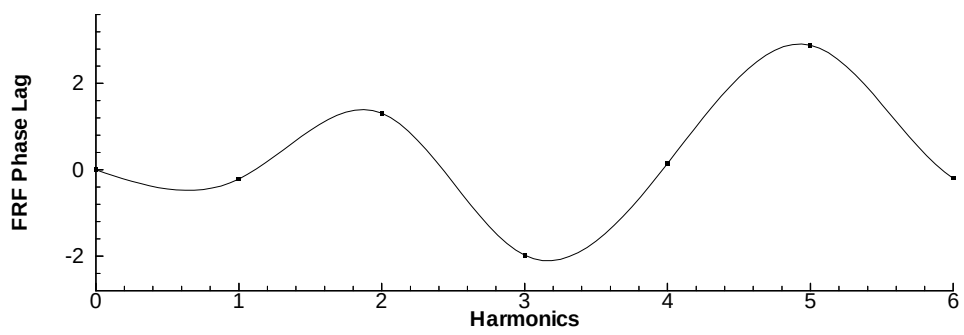
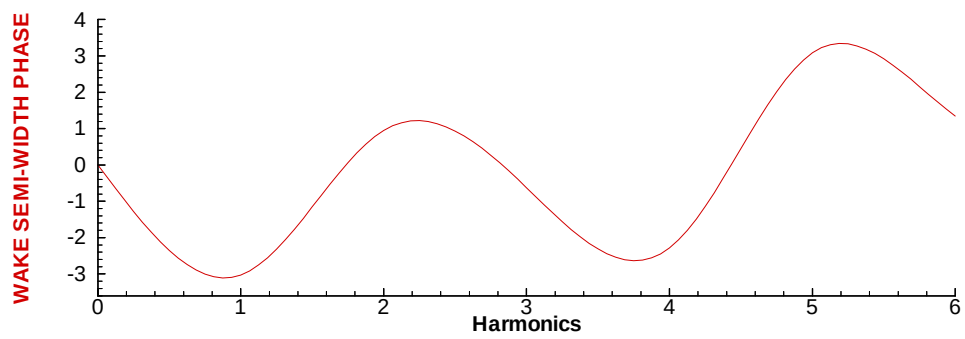
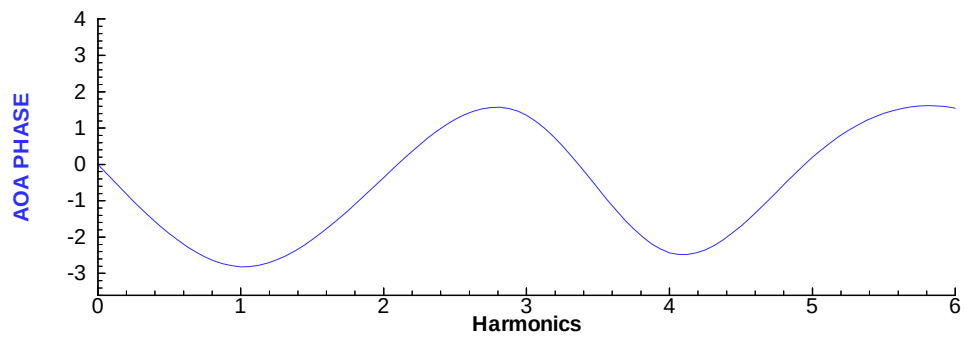
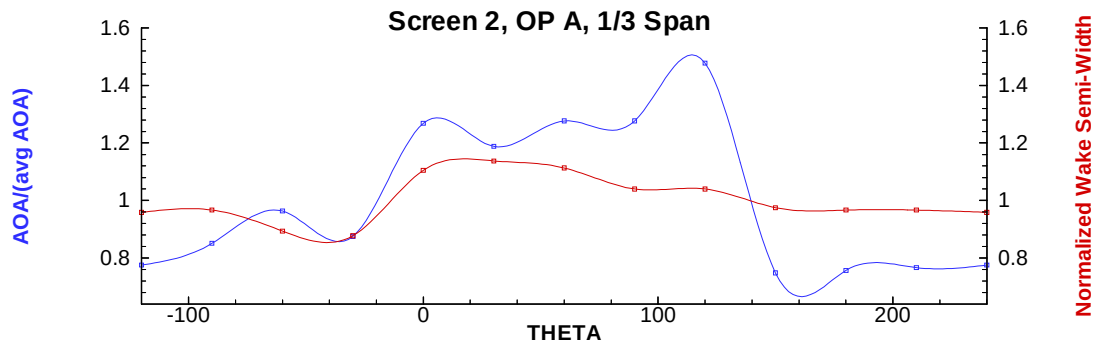


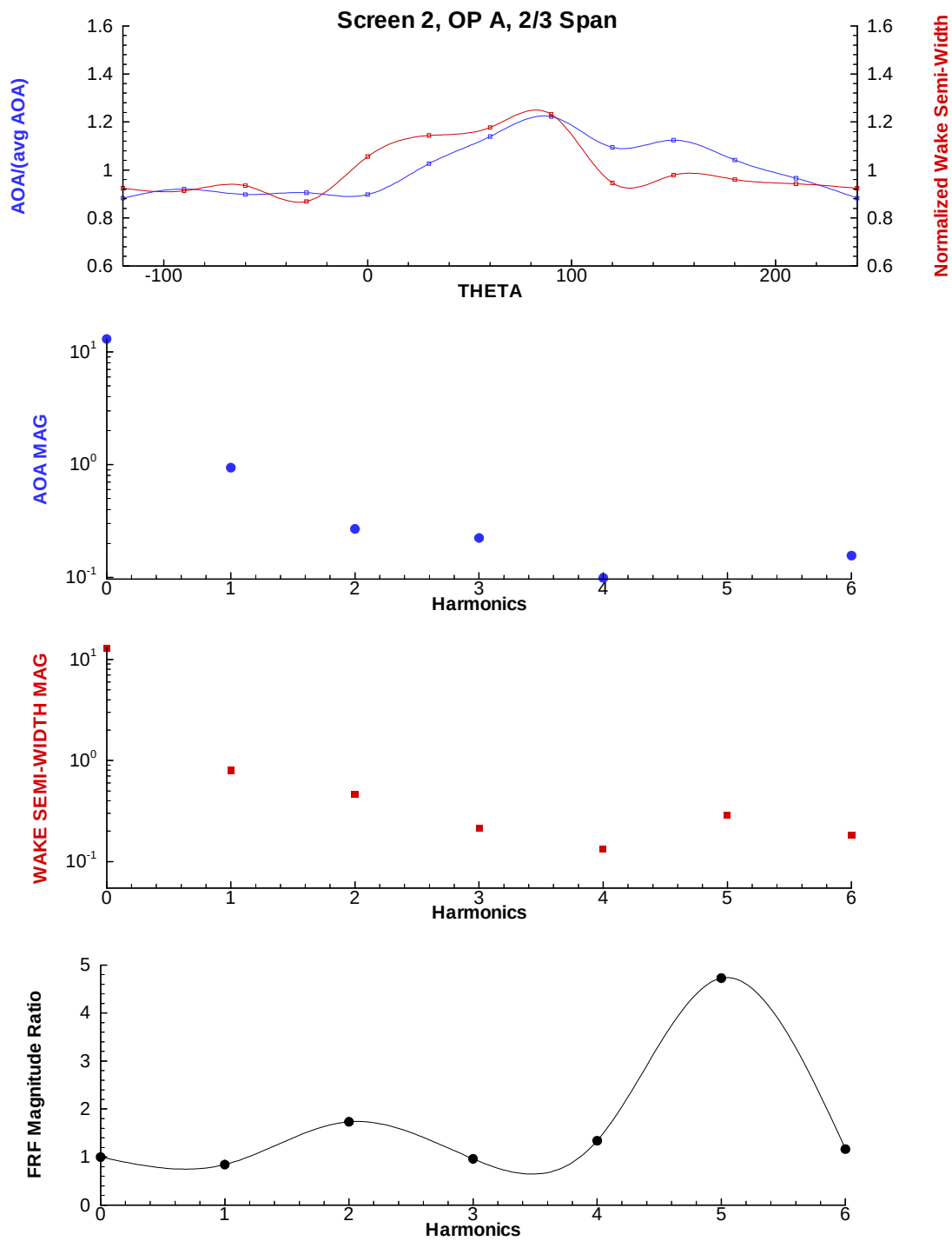


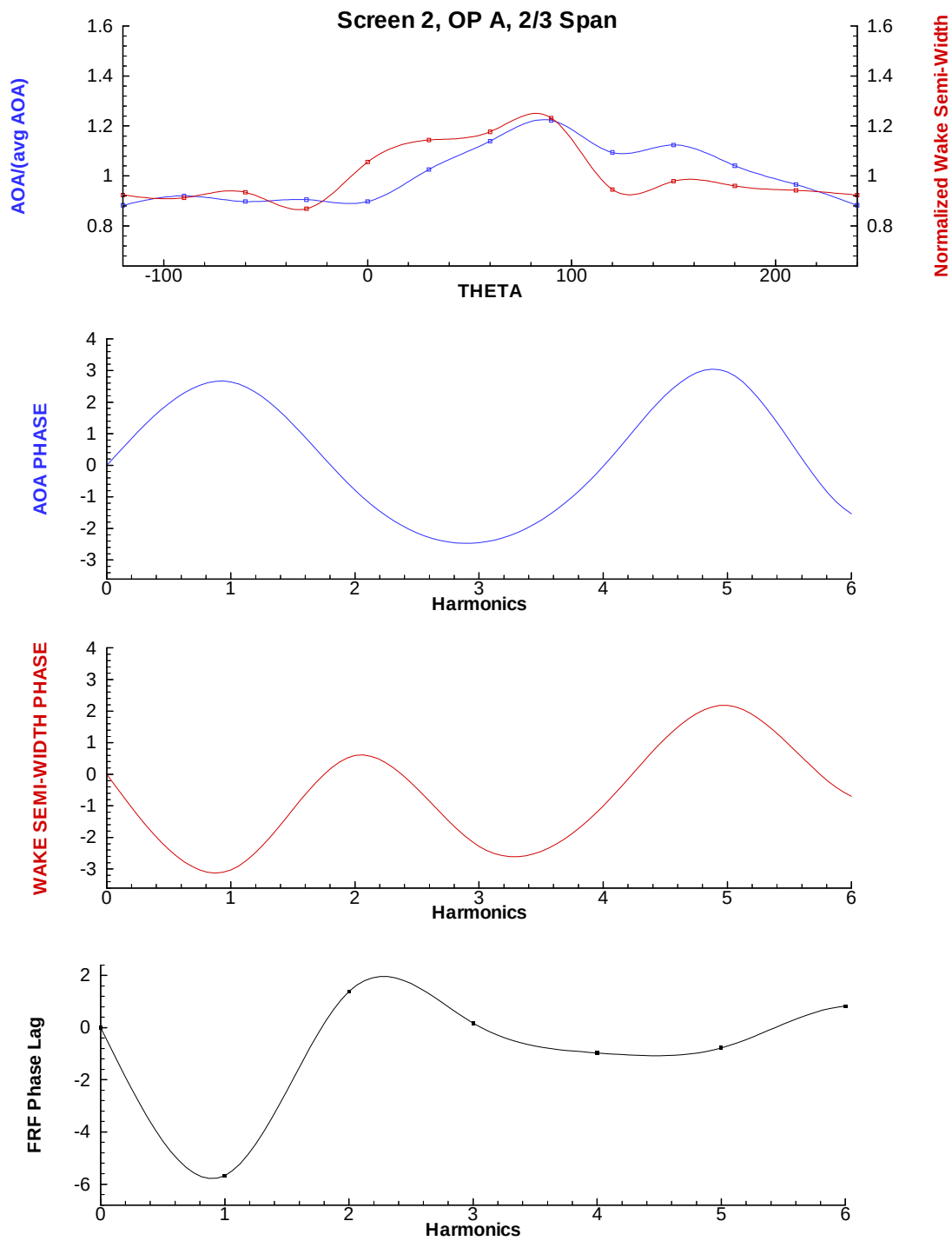


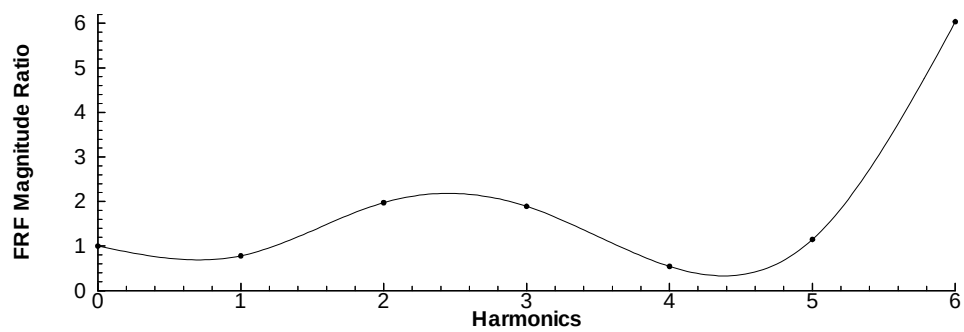
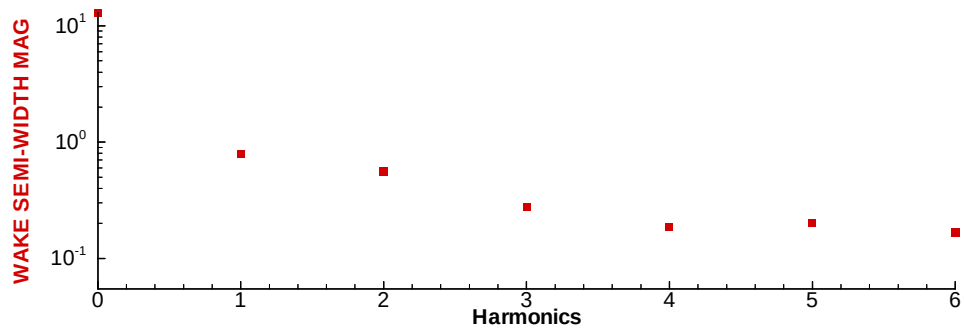
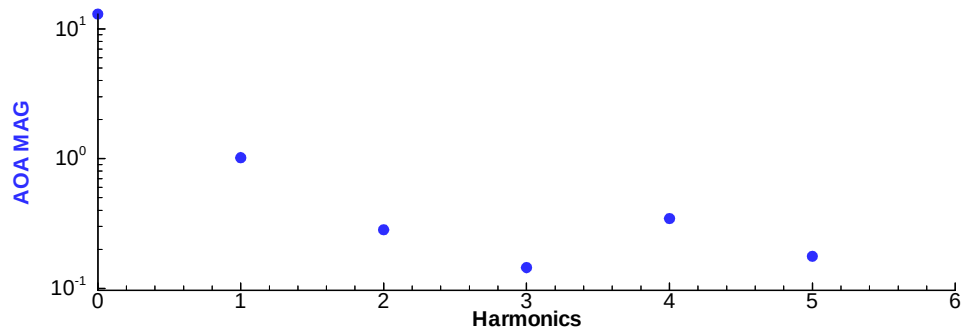
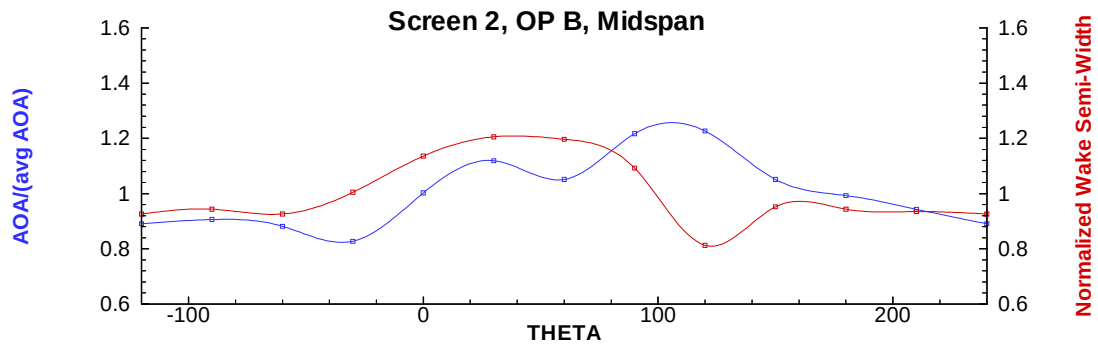


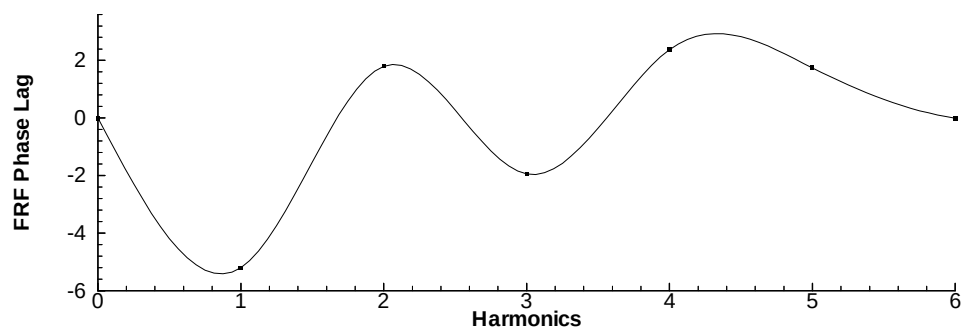
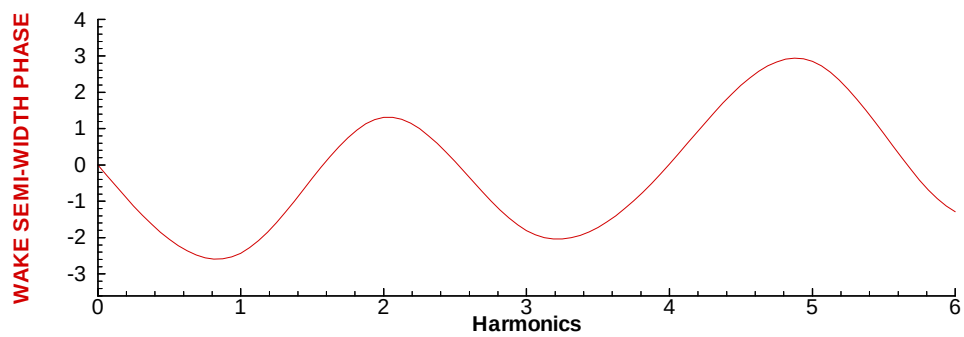
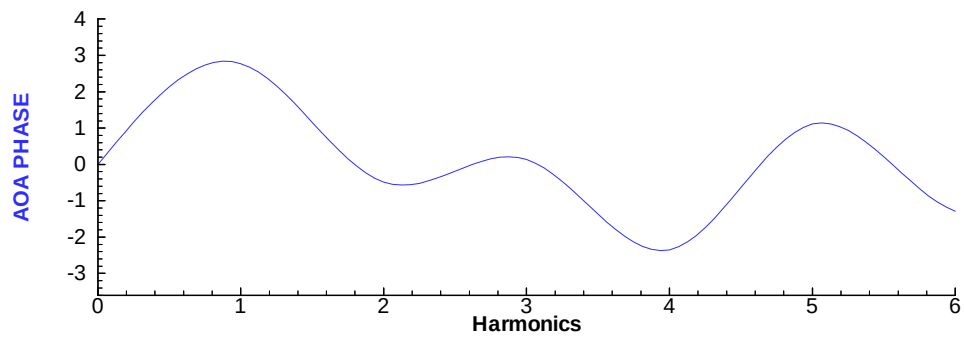
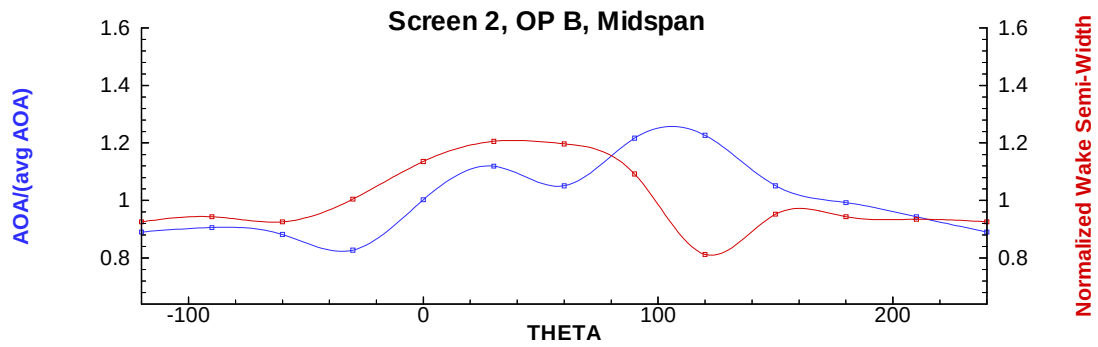


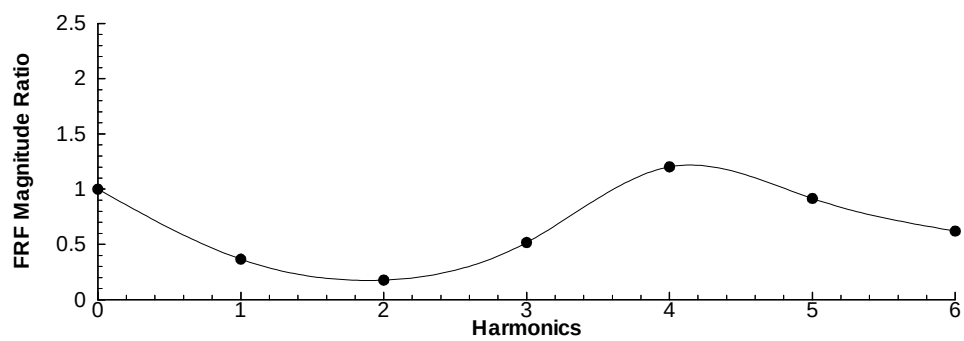
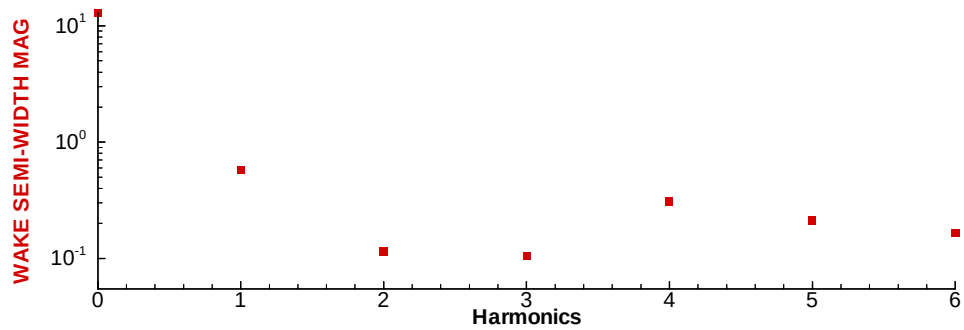
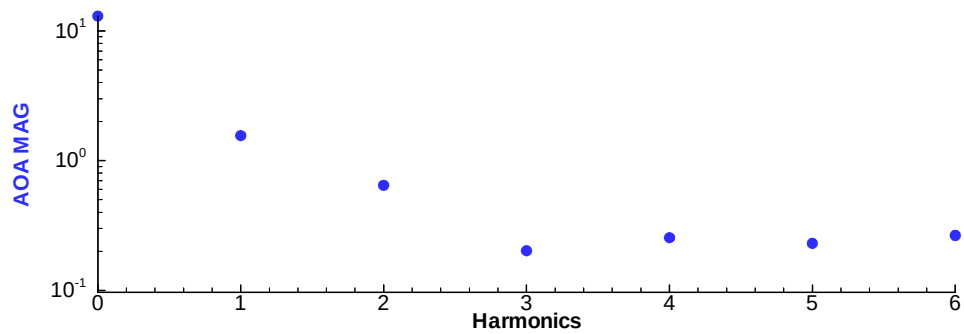
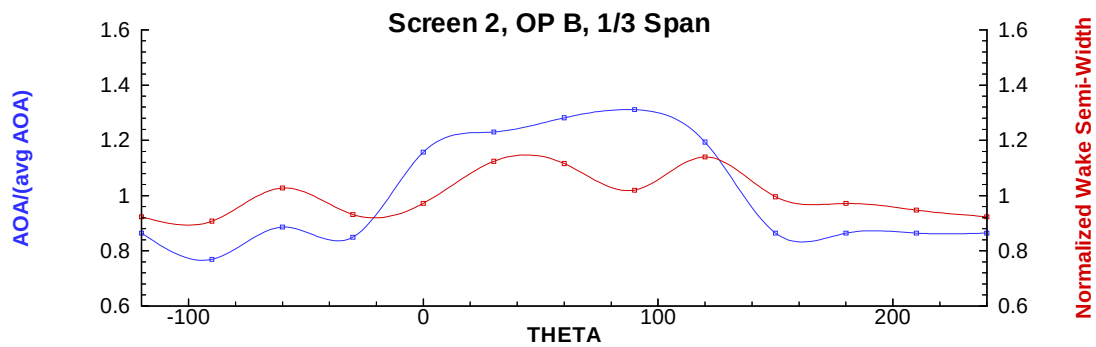


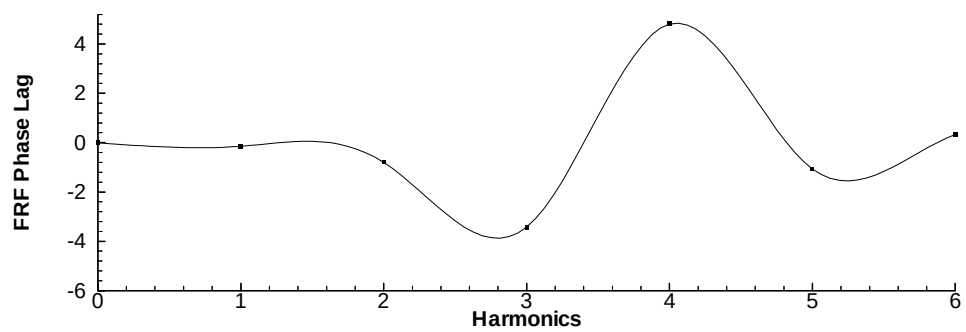
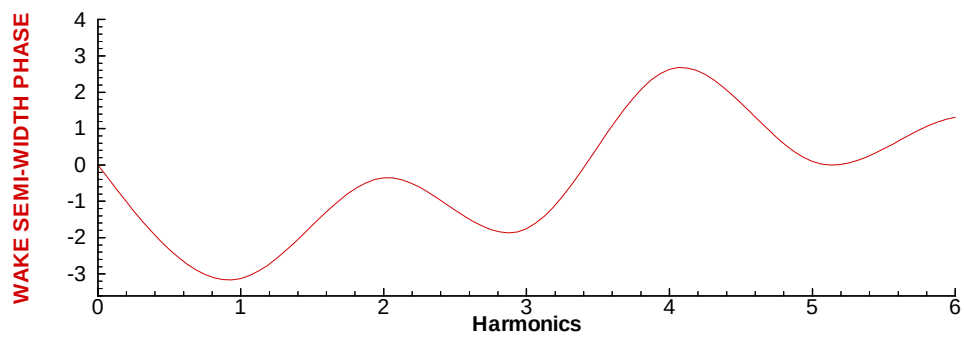
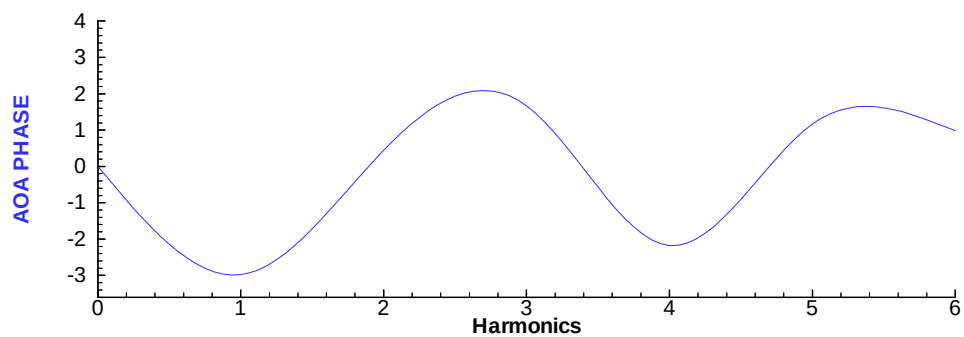
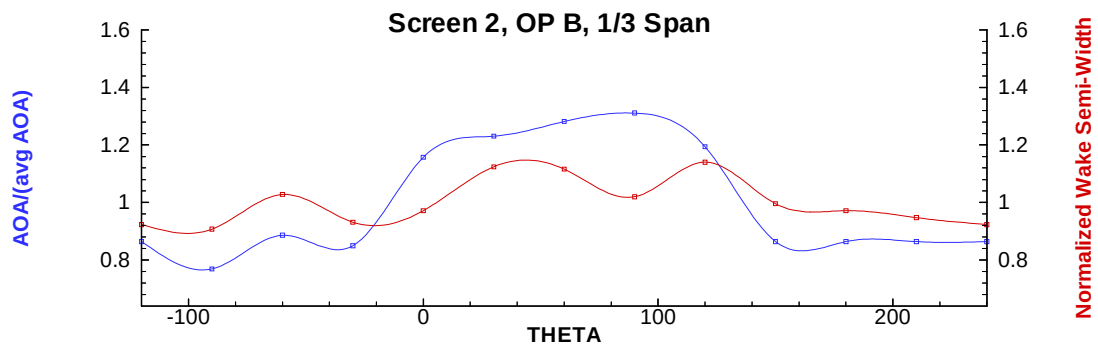


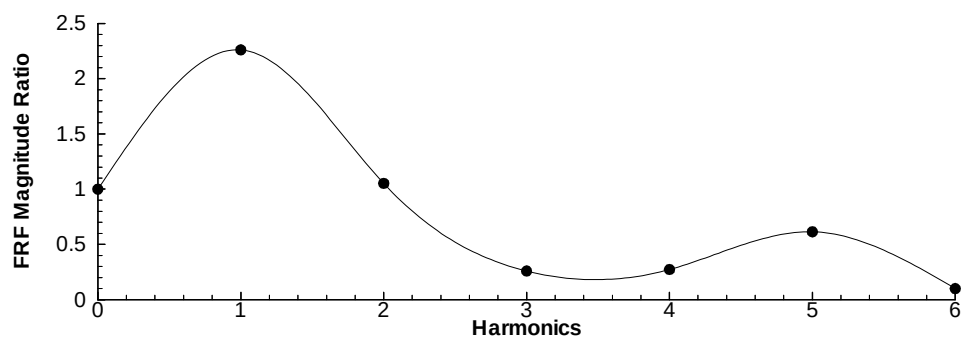
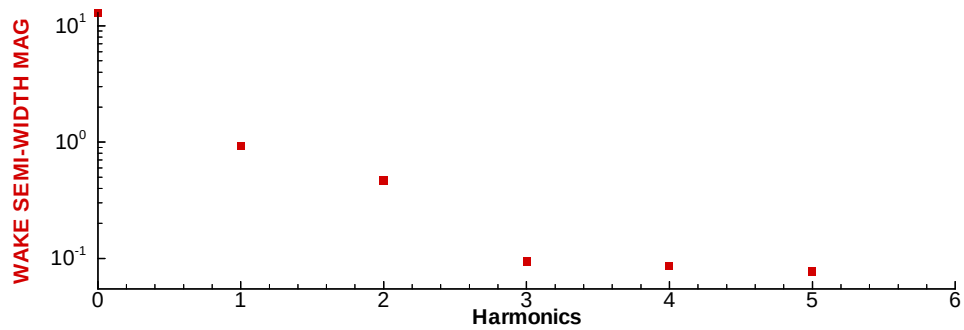
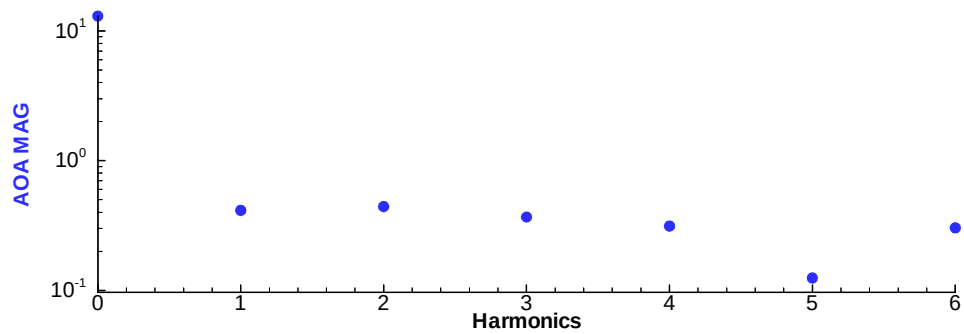
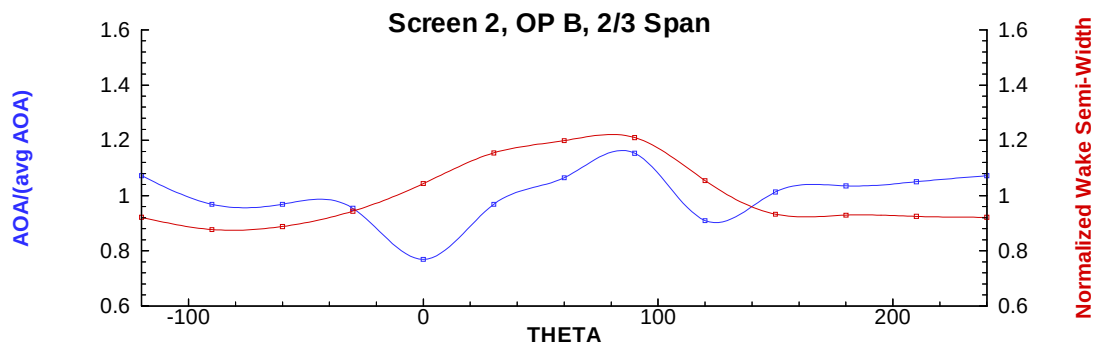


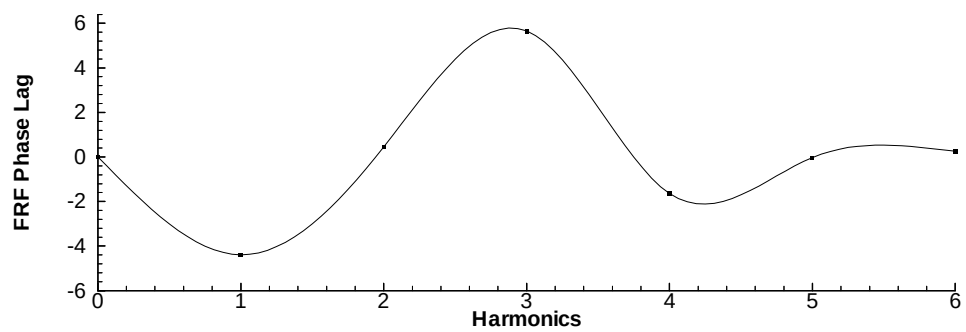
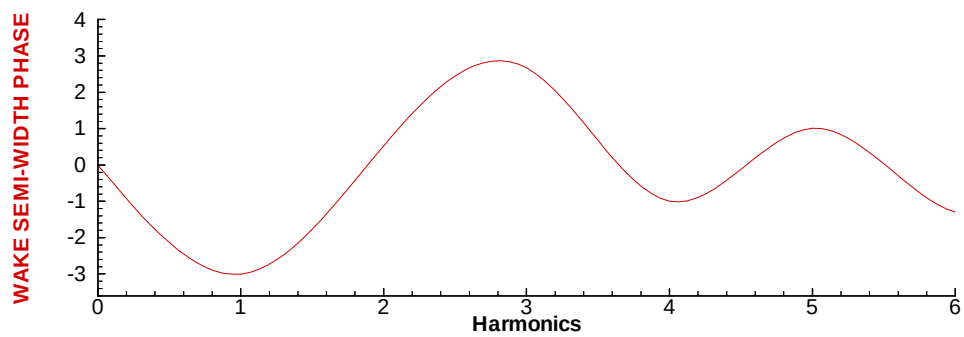
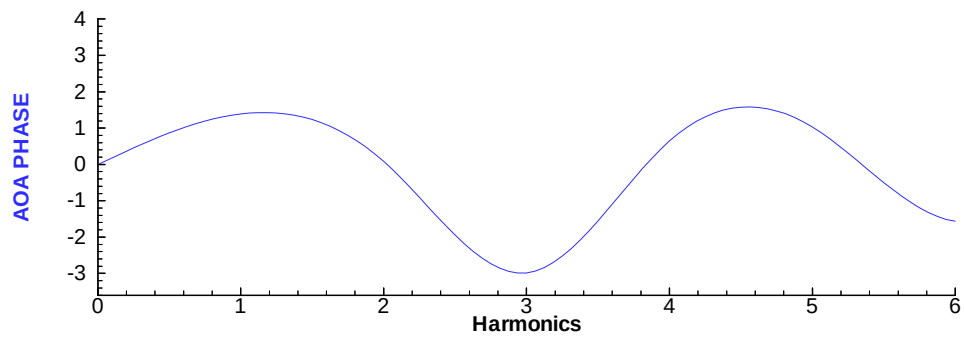
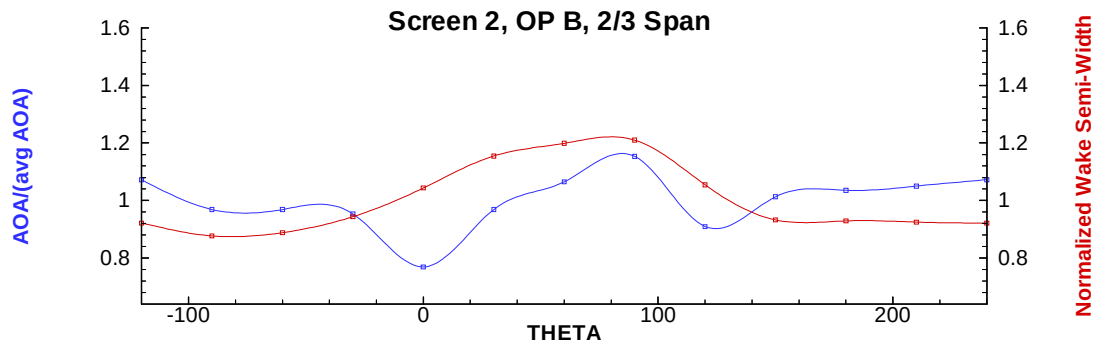


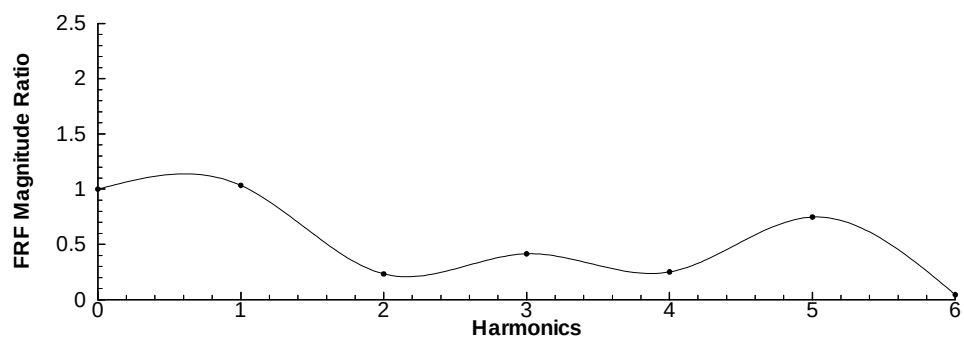
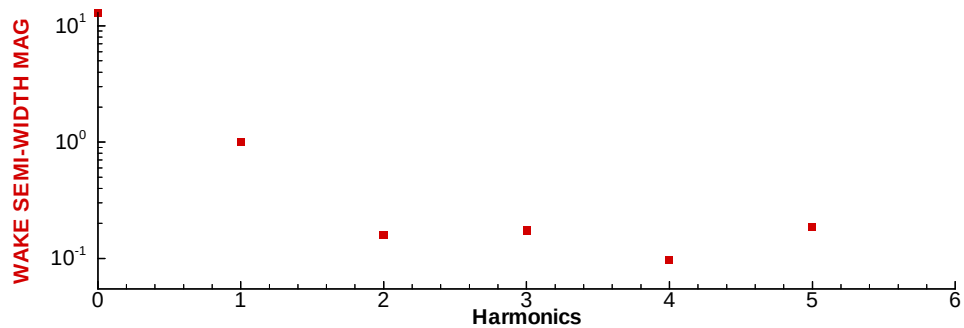
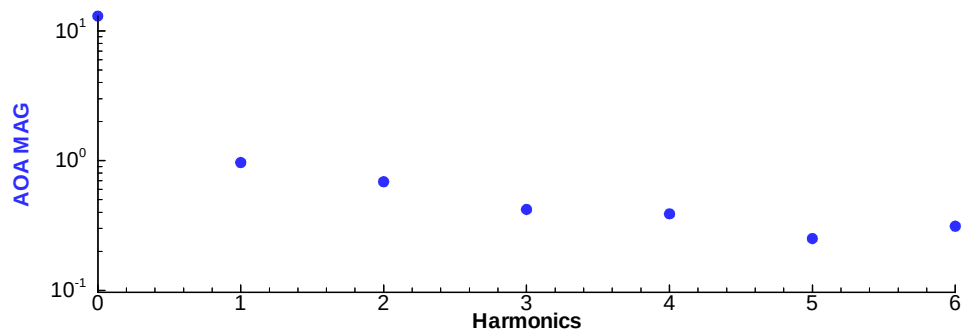
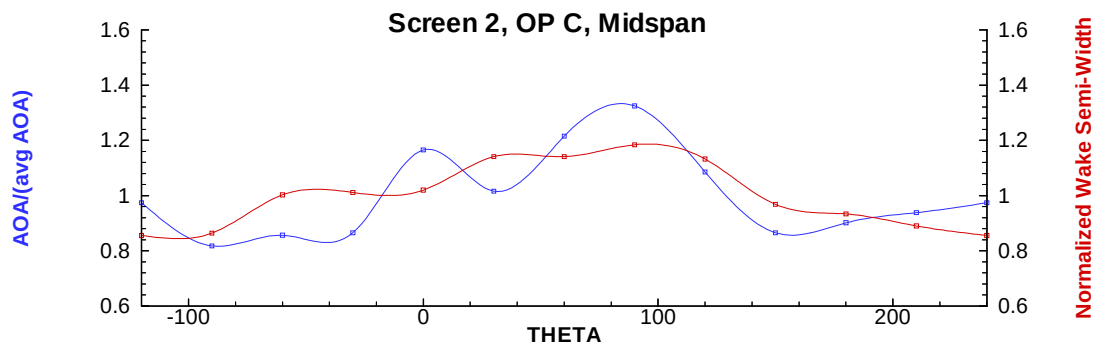


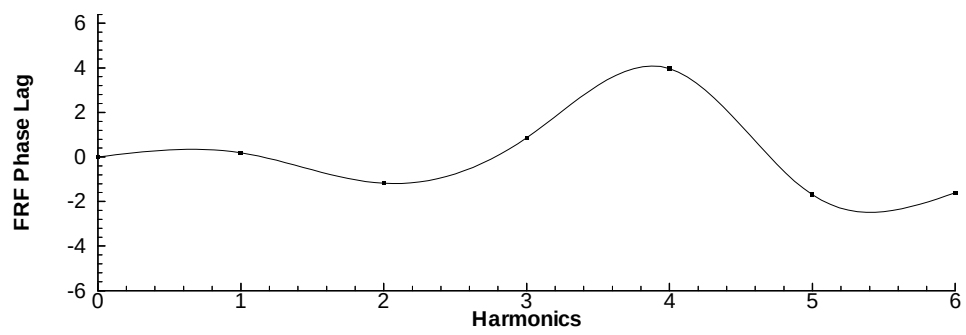
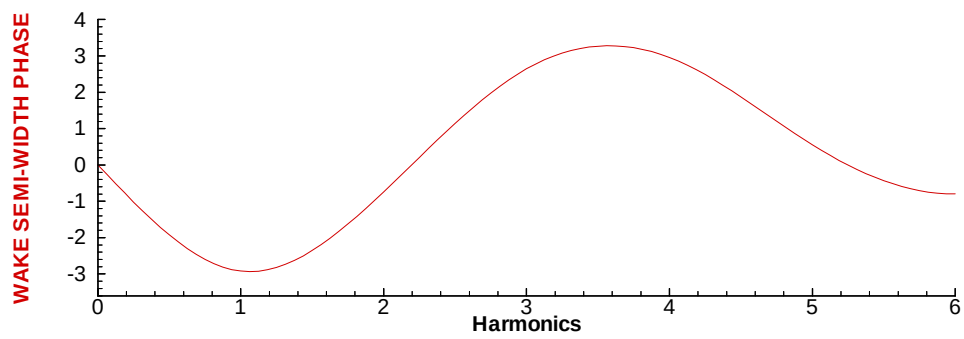
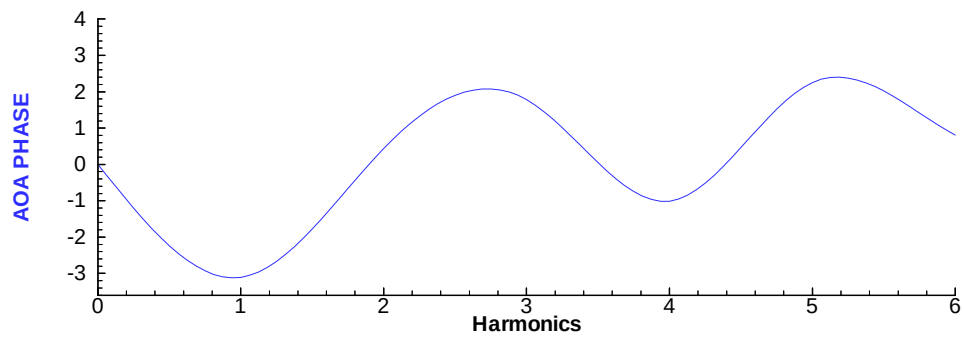
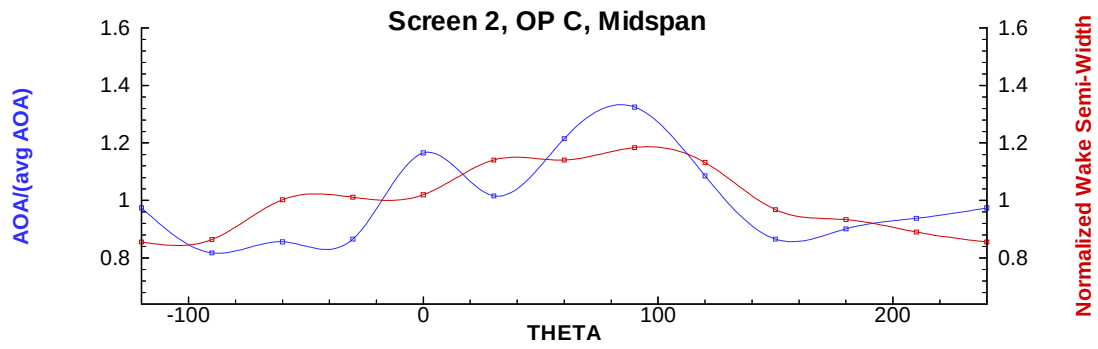


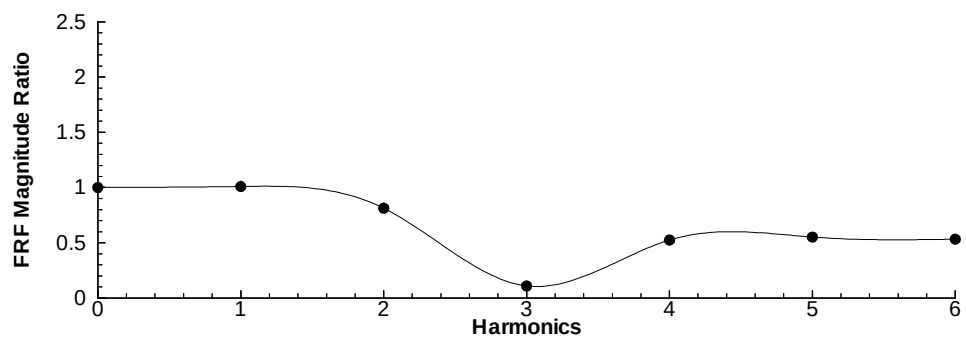
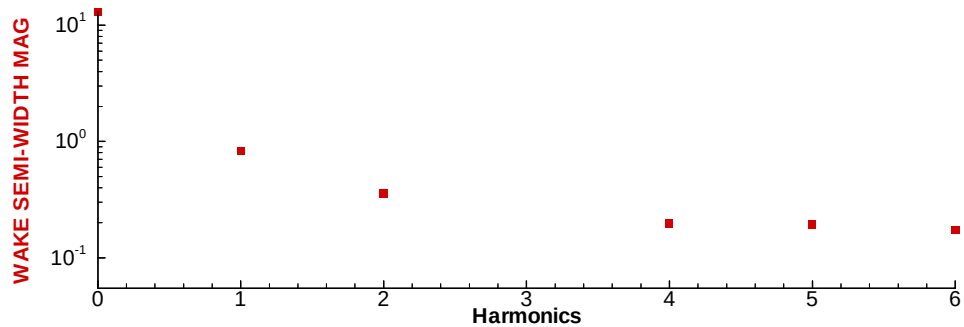
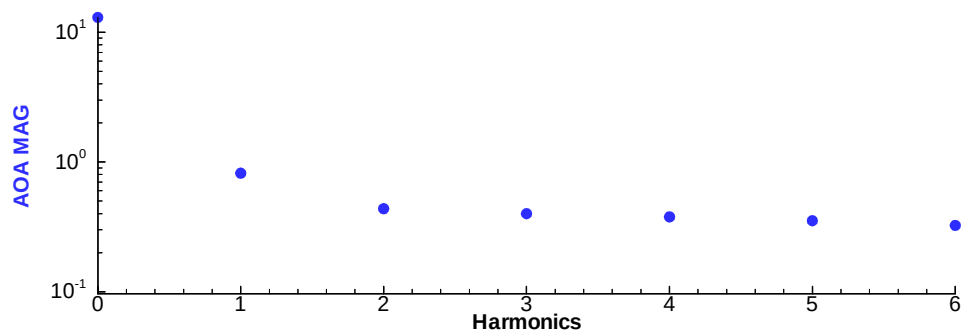
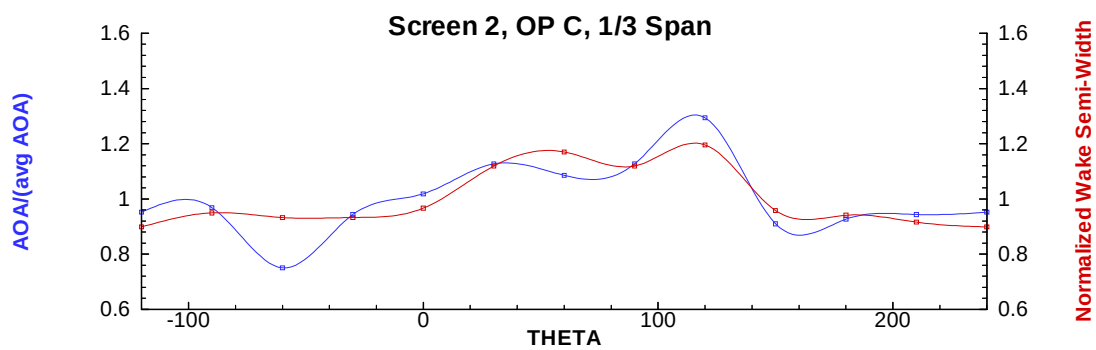


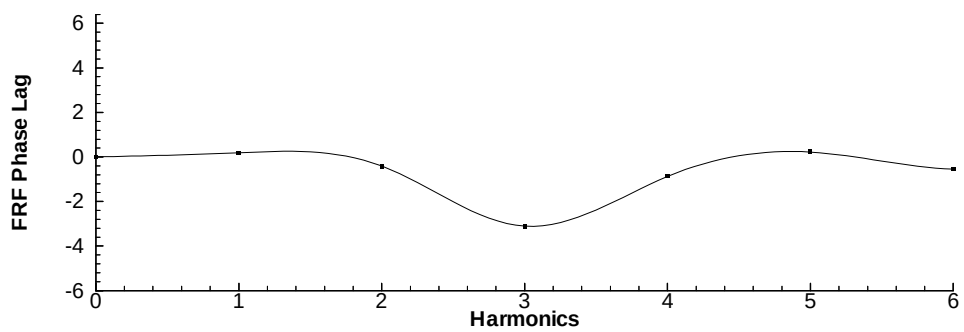
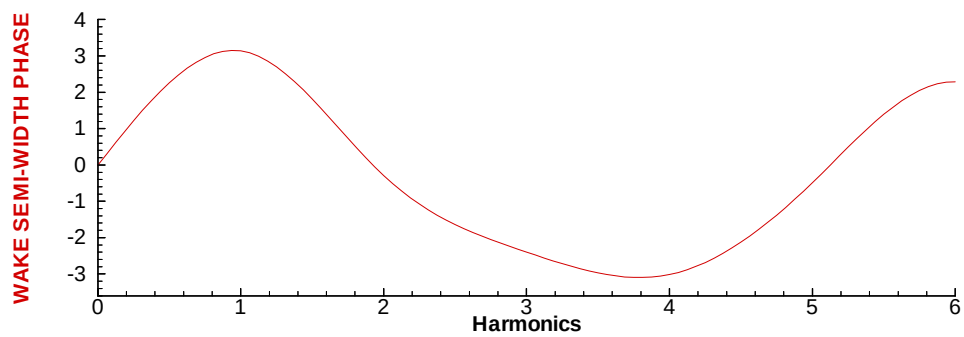
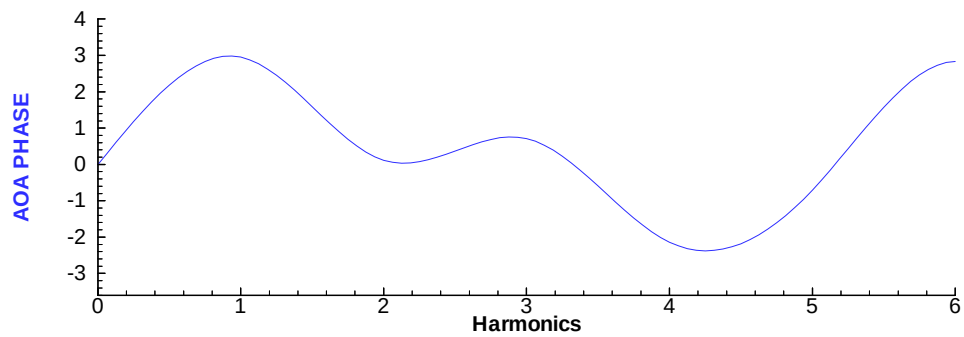
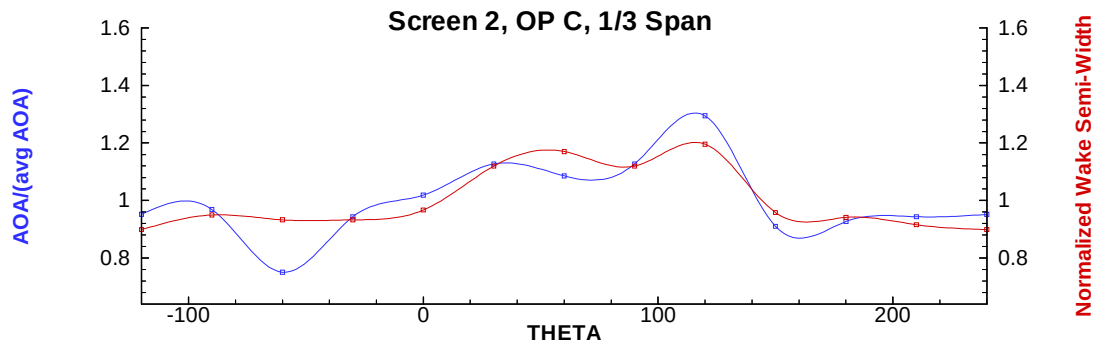


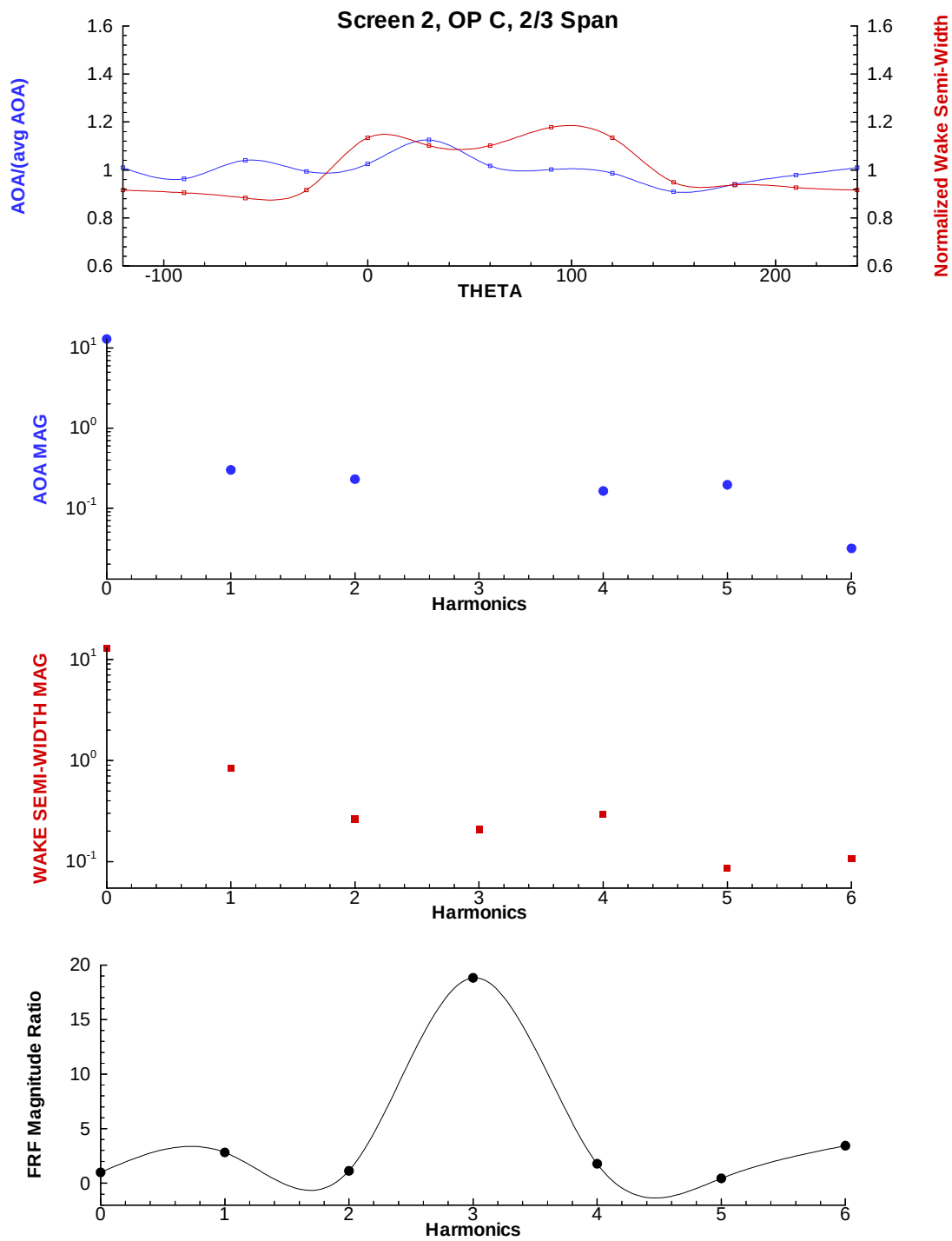


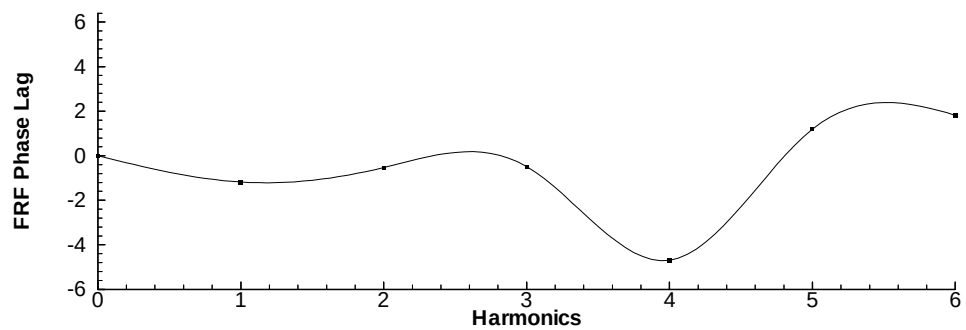
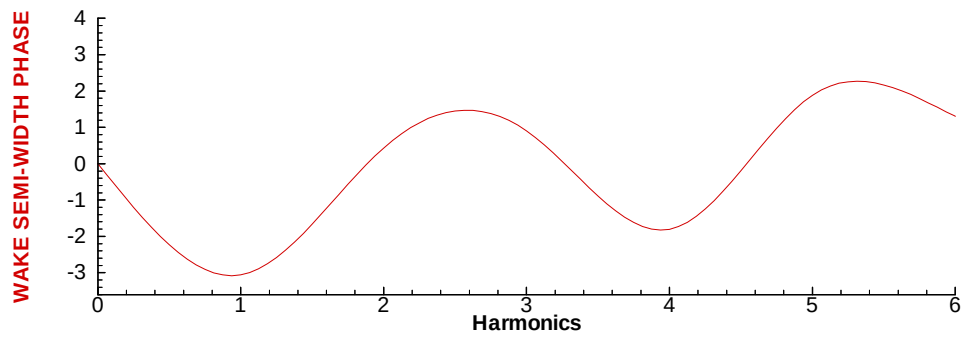
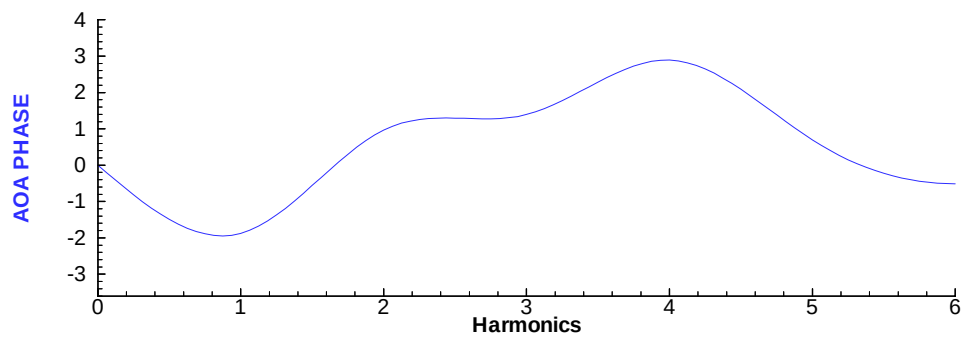
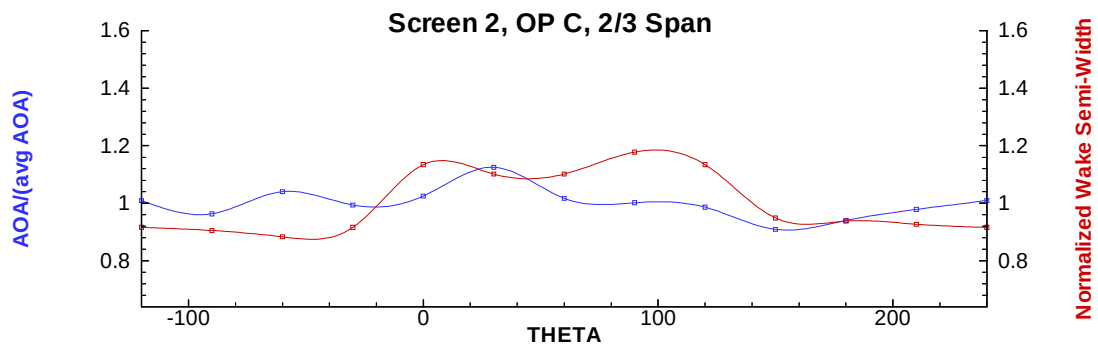


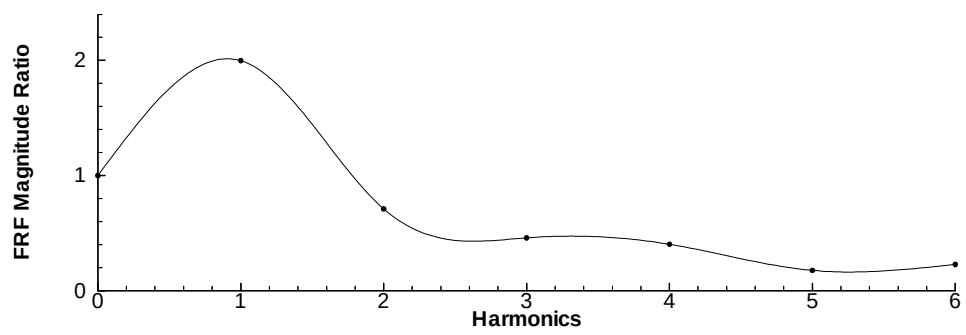
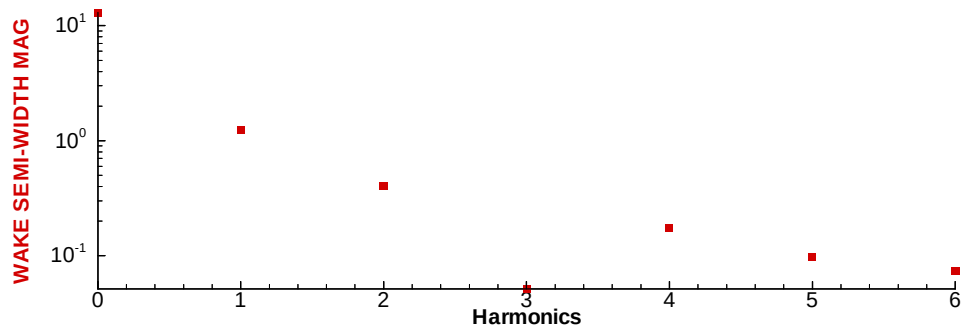
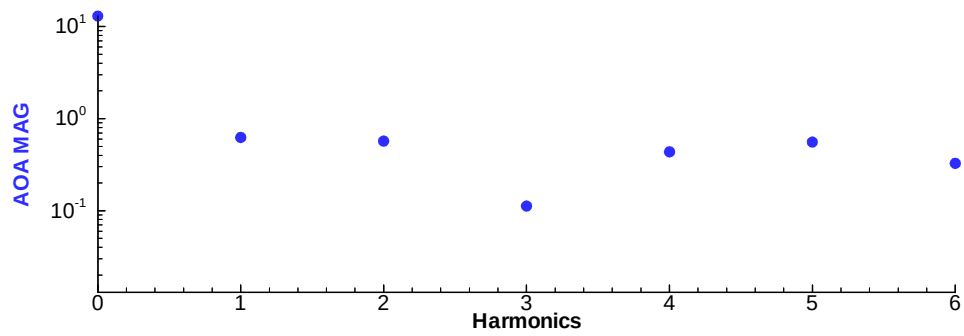
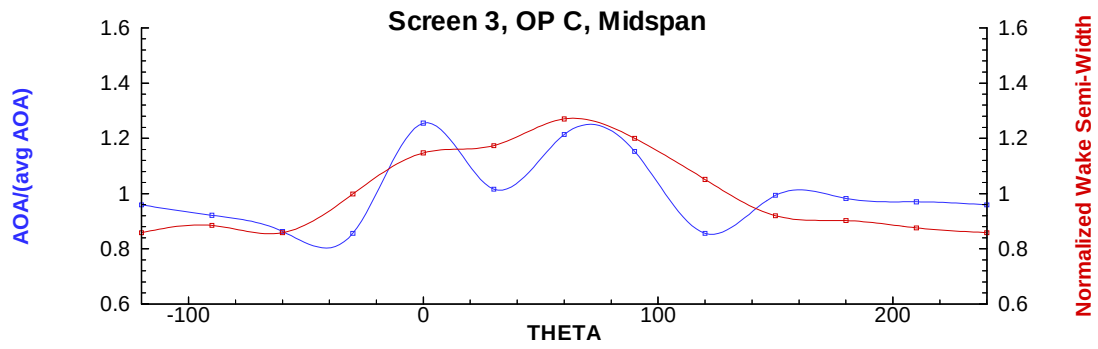


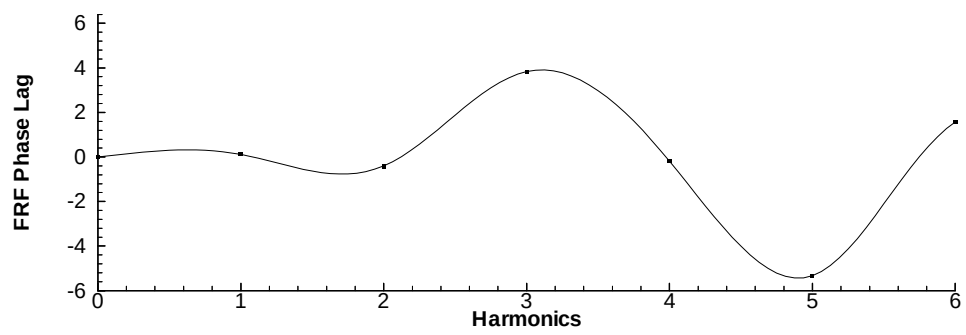
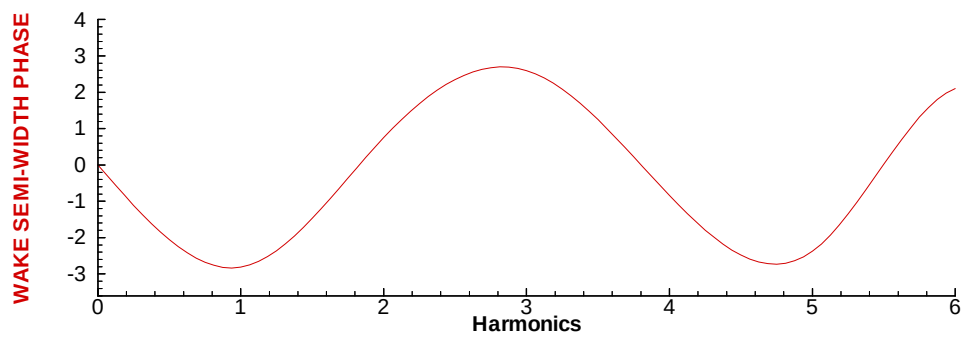
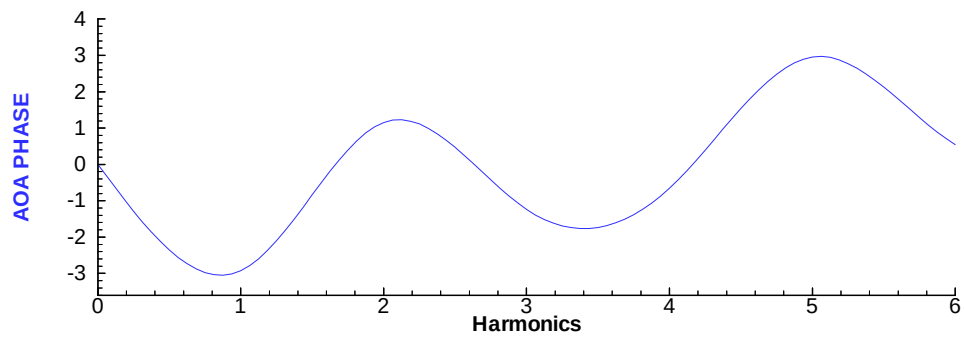
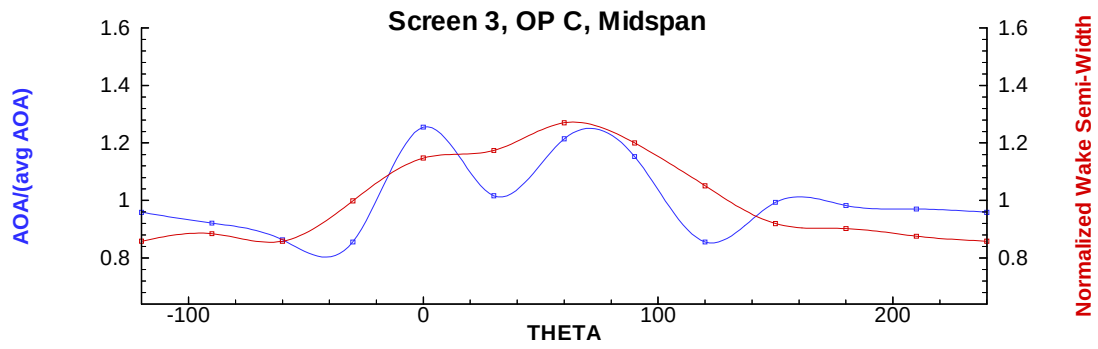


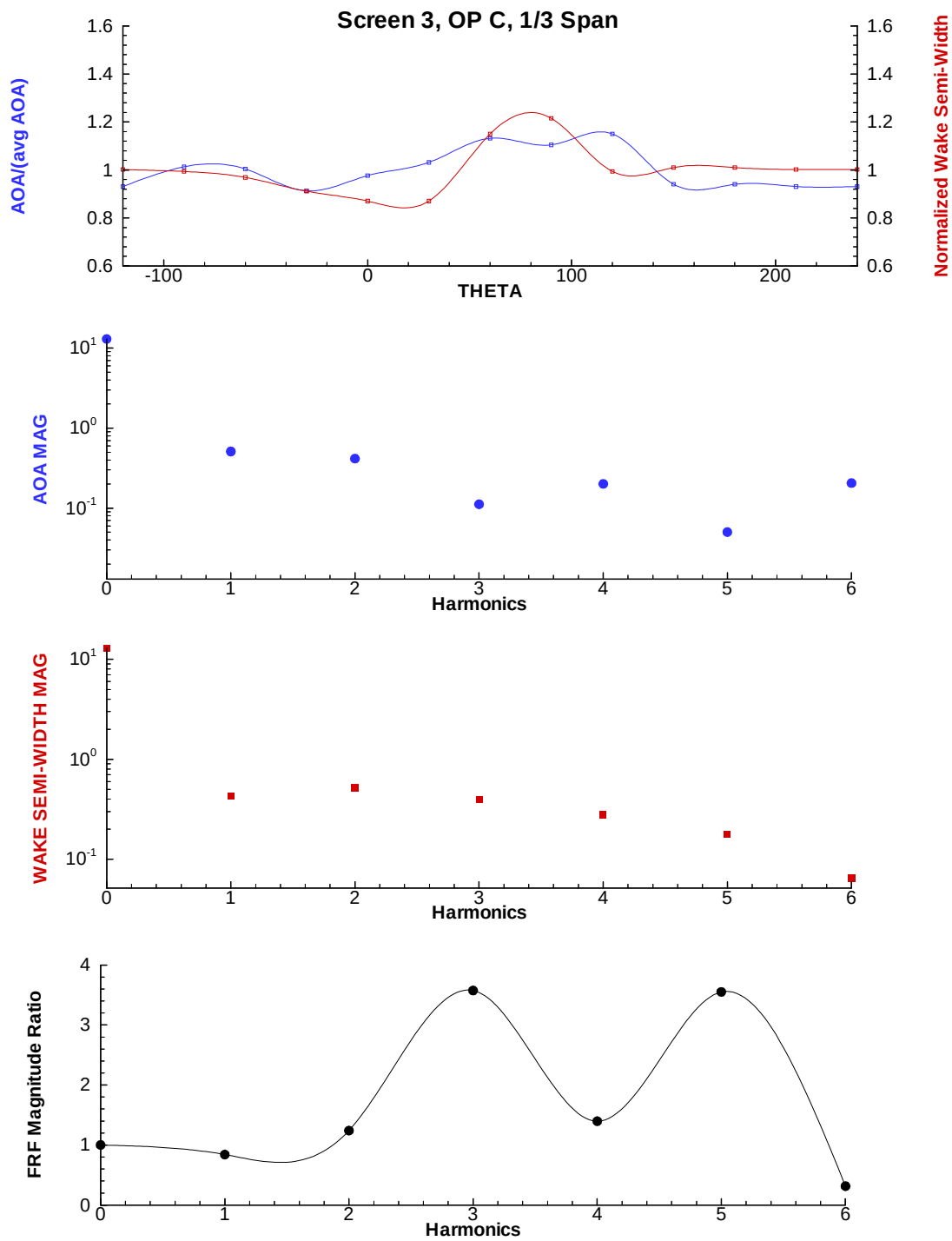


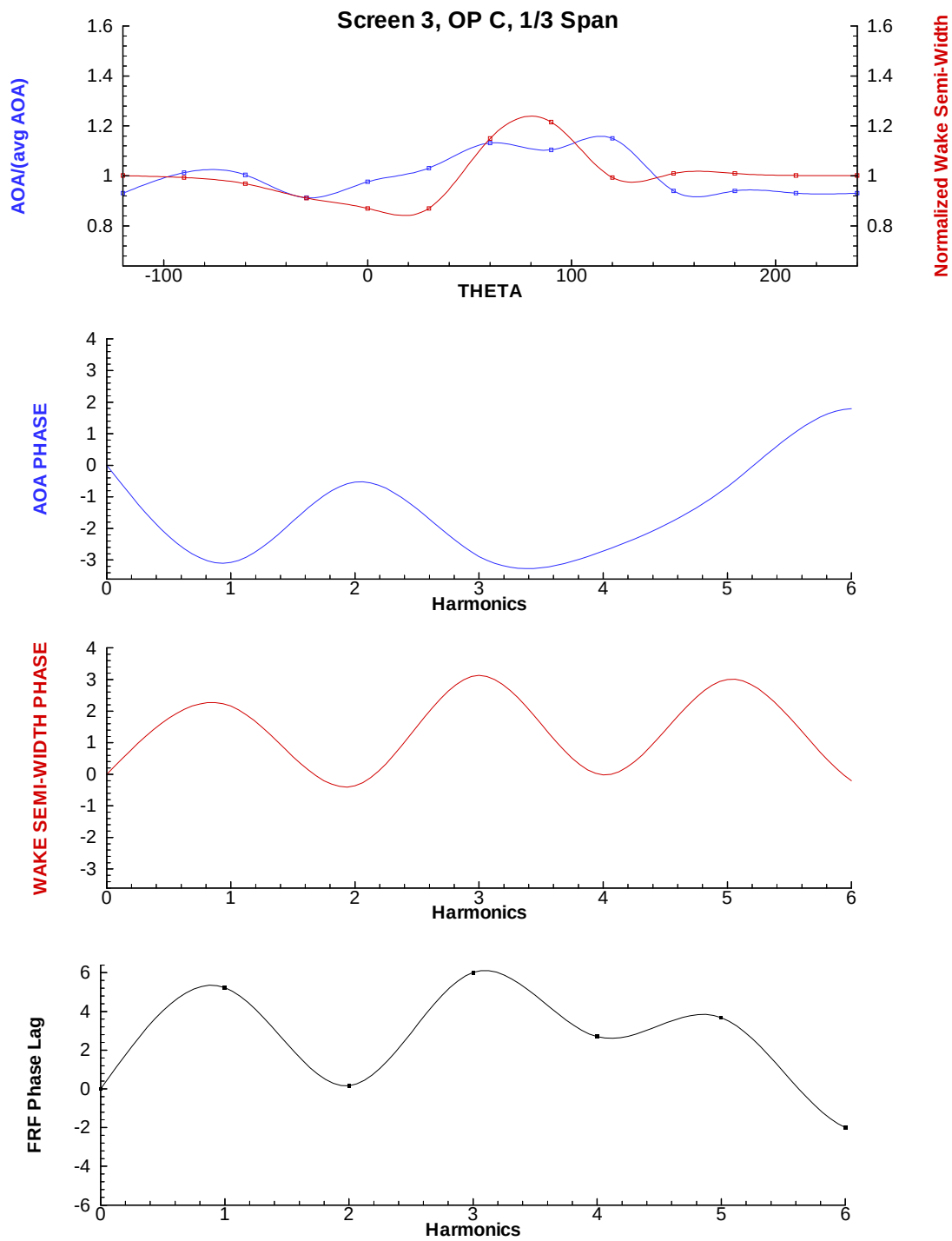


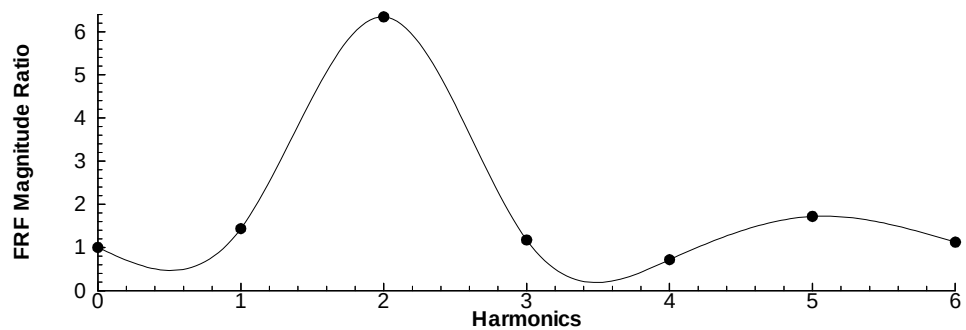
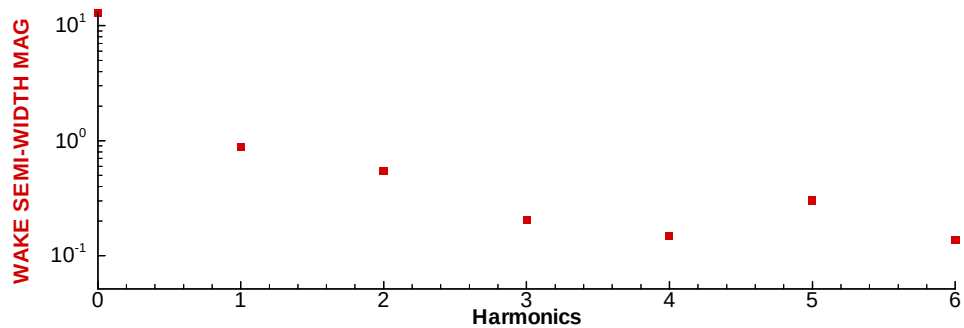
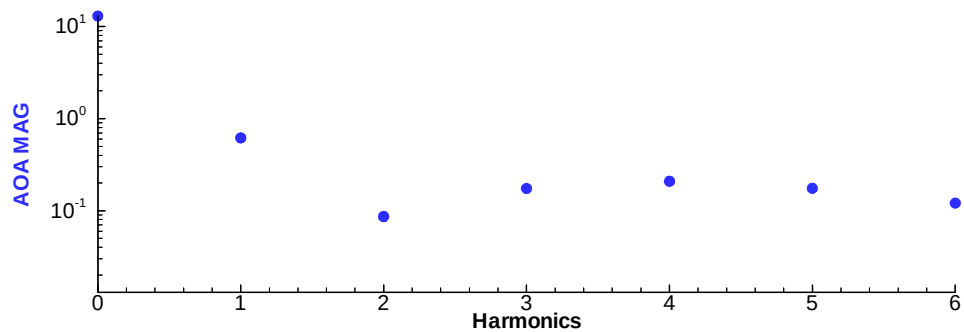
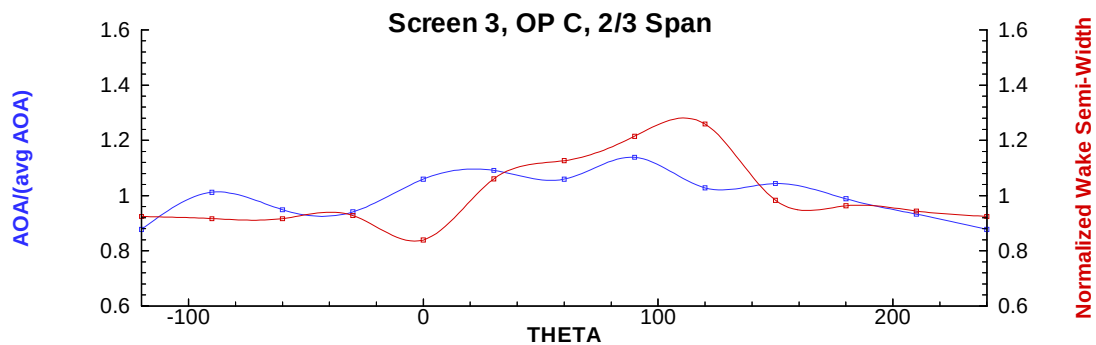


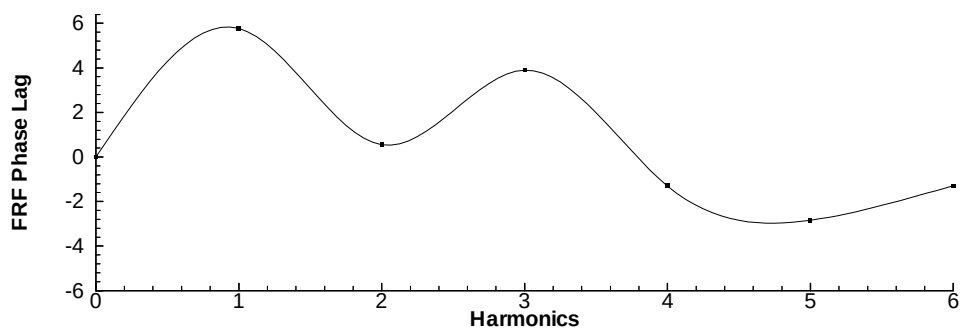
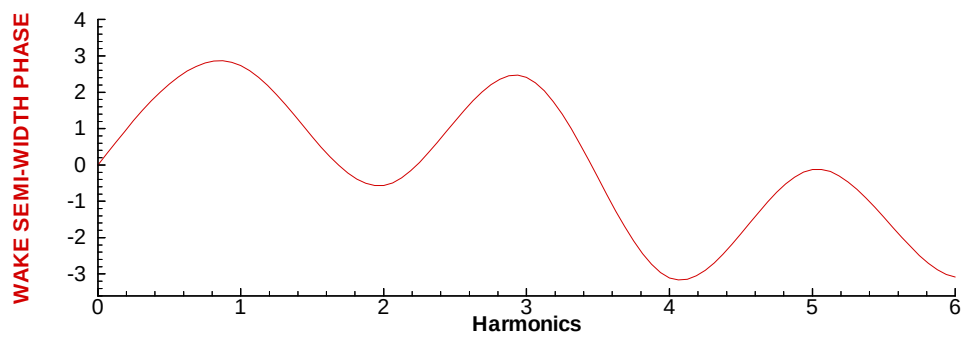
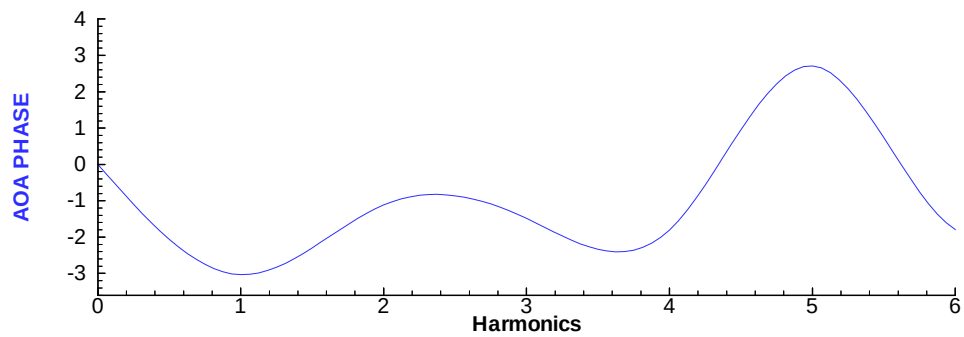
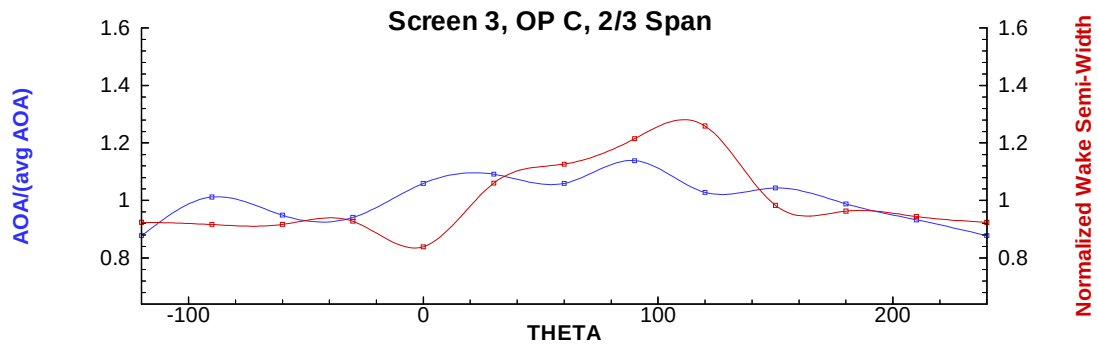




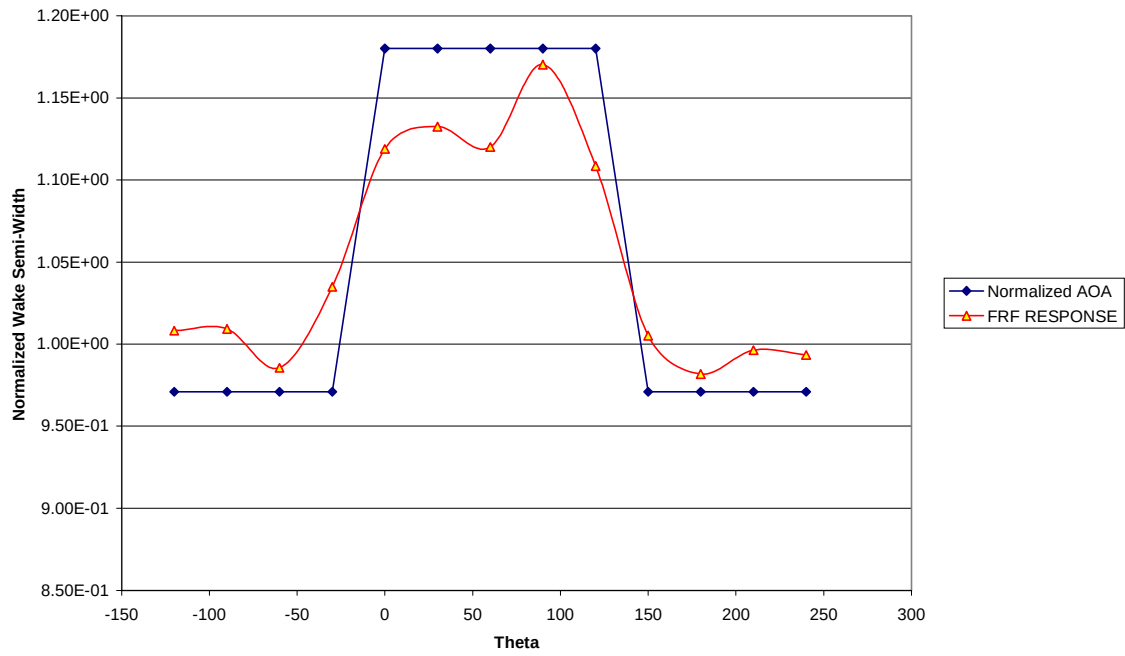




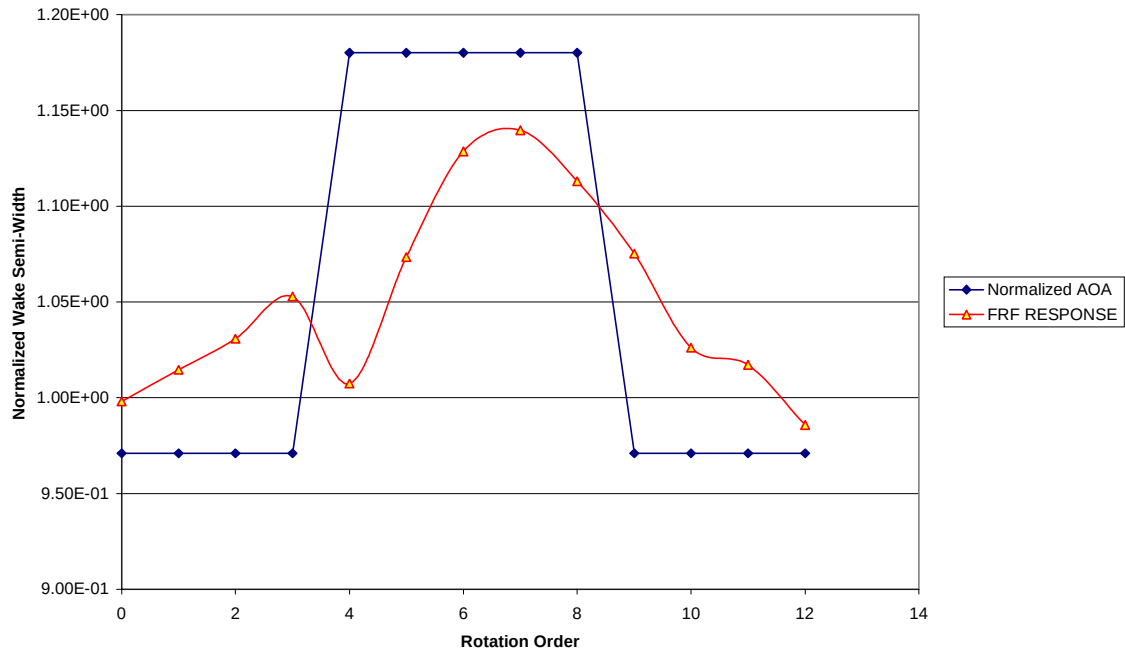




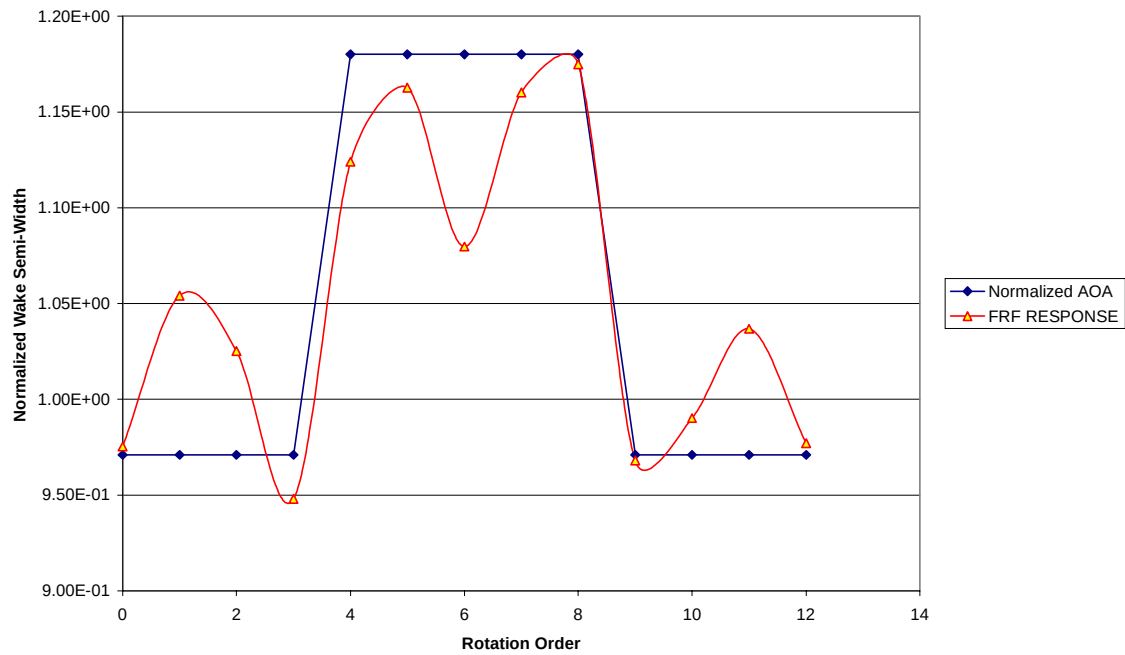
Response of Blade for level 1 screen at midspan and OP A based on FRF(Normalized Wake Semi-Width / Normalized AOA)



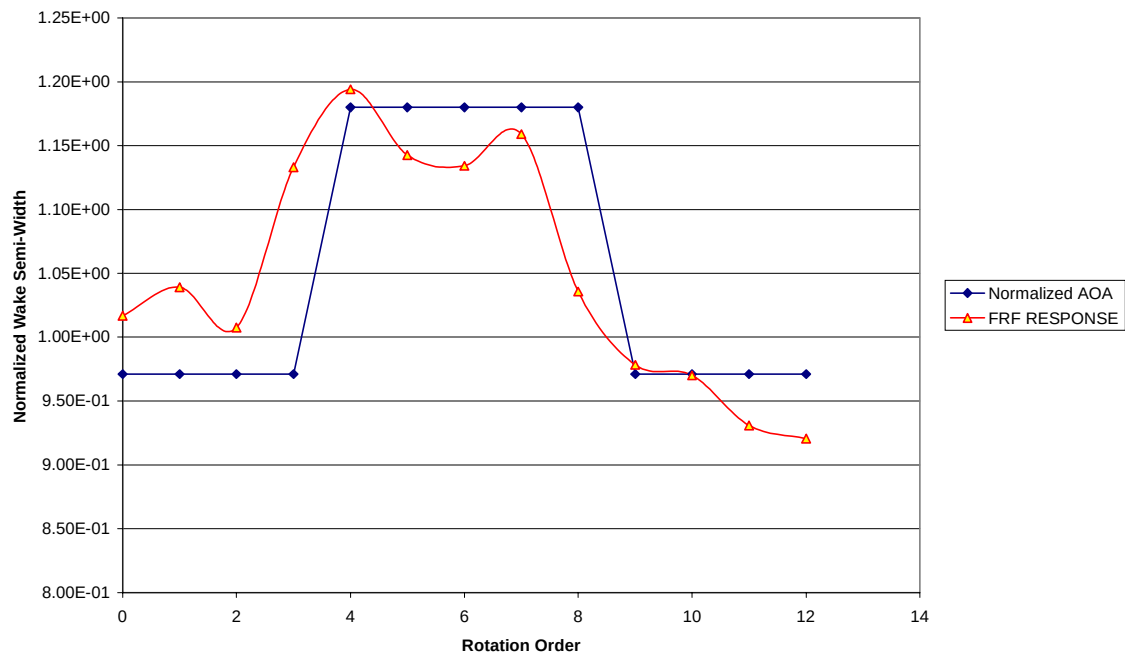
Response of Blade for level 1 screen at midspan and OP B based on FRF(Normalized Wake Semi-Width / Normalized AOA)



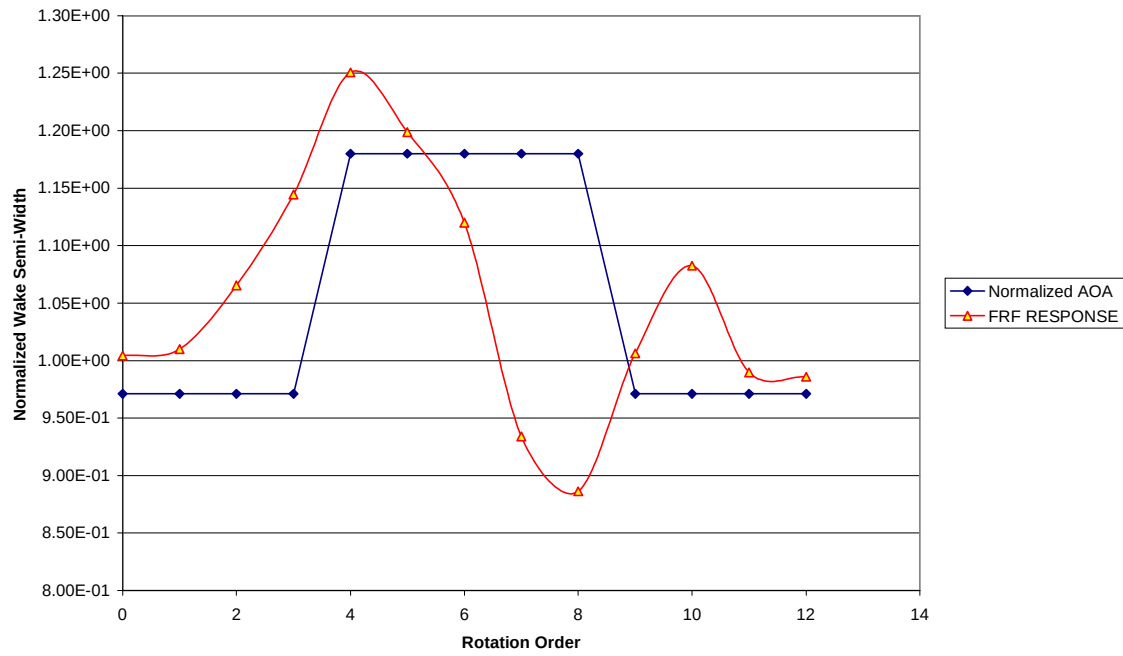
Response of Blade for level 1 screen at 1/3 span and OP B based on FRF(Normalized Wake Semi-Width / Normalized AOA)



Response of Blade for level 2 screen at midspan and OP A based on FRF(Normalized Wake Semi-Width / Normalized AOA)



Response of Blade for level 2 screen at midspan and OP B based on FRF(Normalized Wake Semi-Width / Normalized AOA)



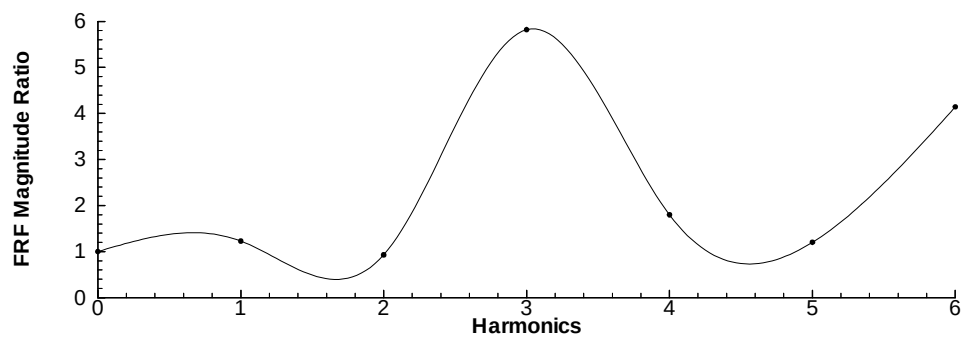
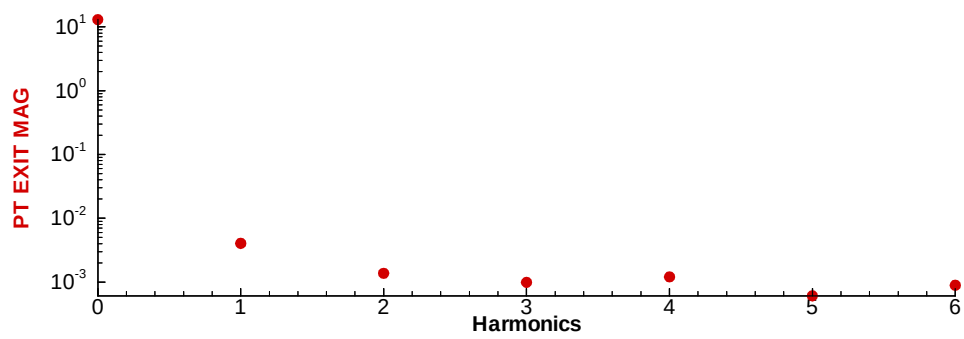
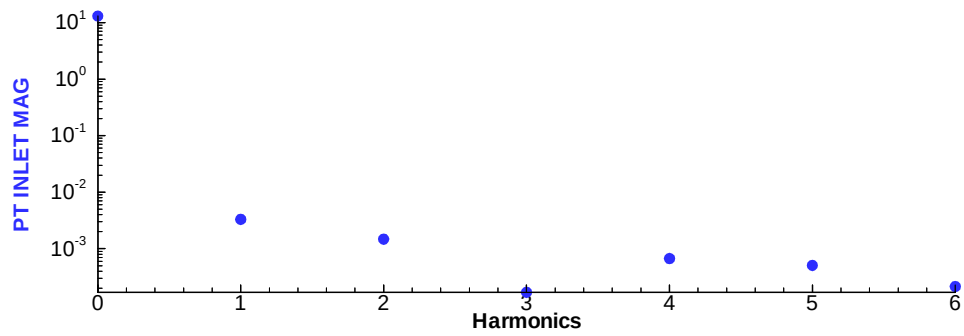
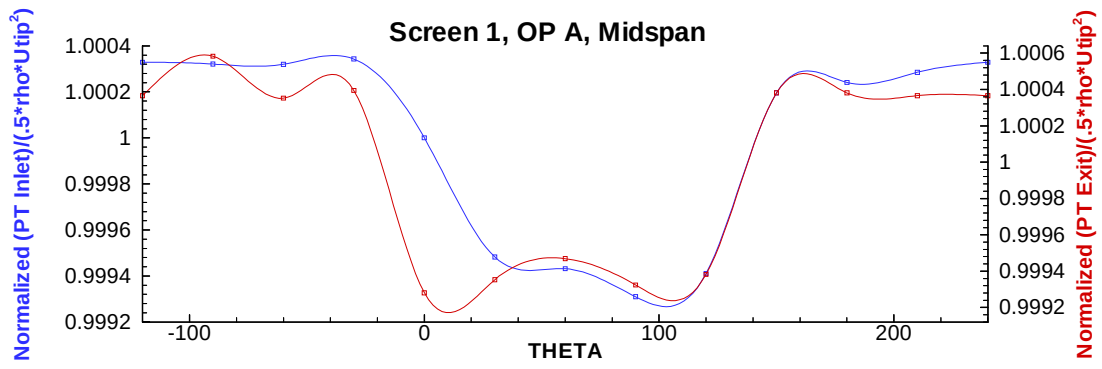
12 Appendix C

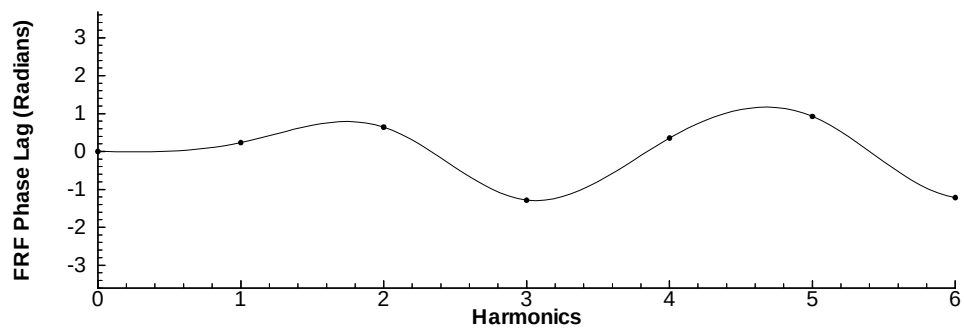
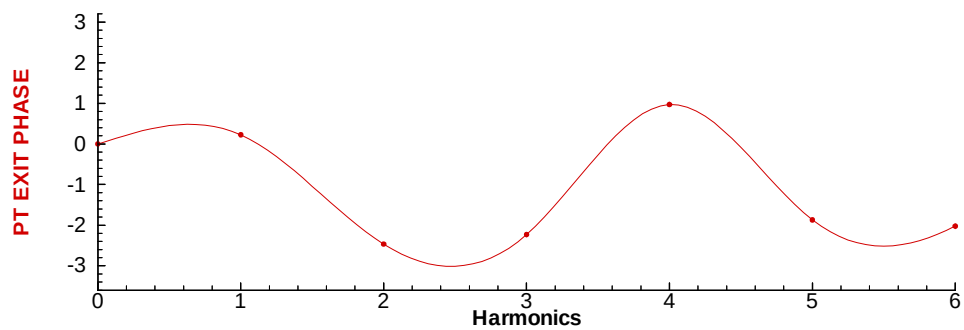
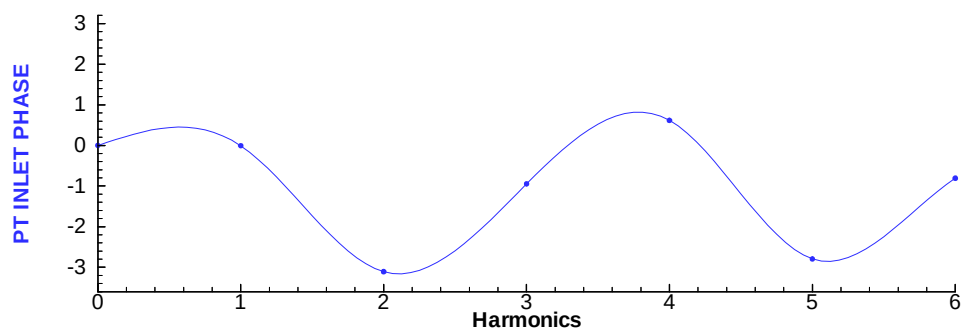
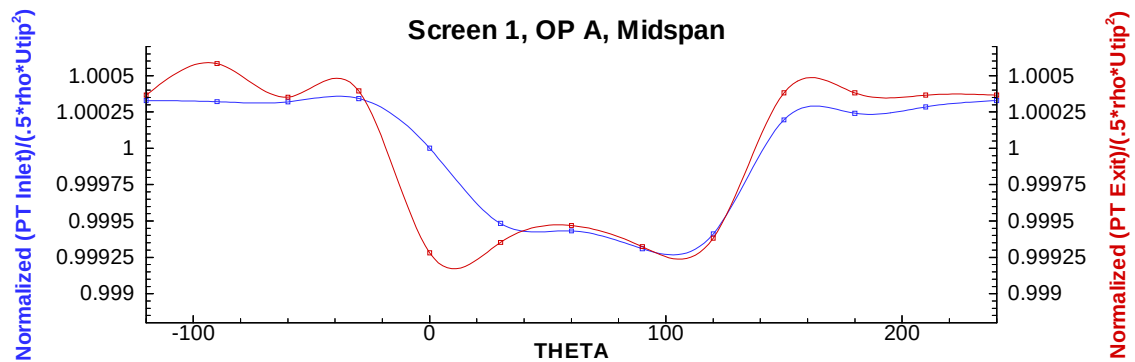
Frequency Response Functions based on Exit Total Pressure and Inlet Total Pressure

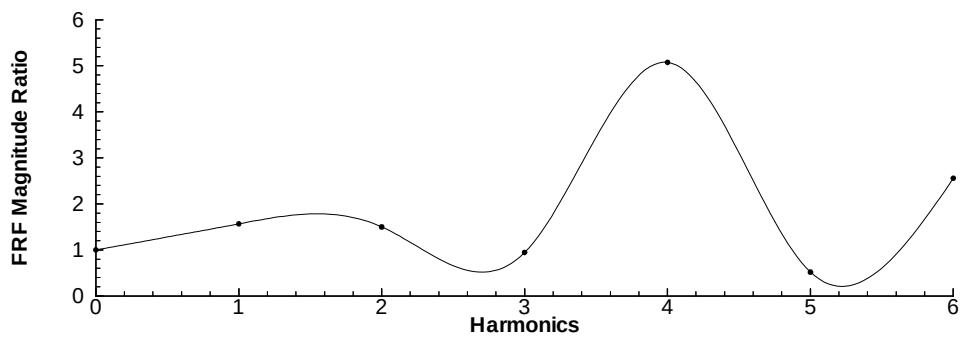
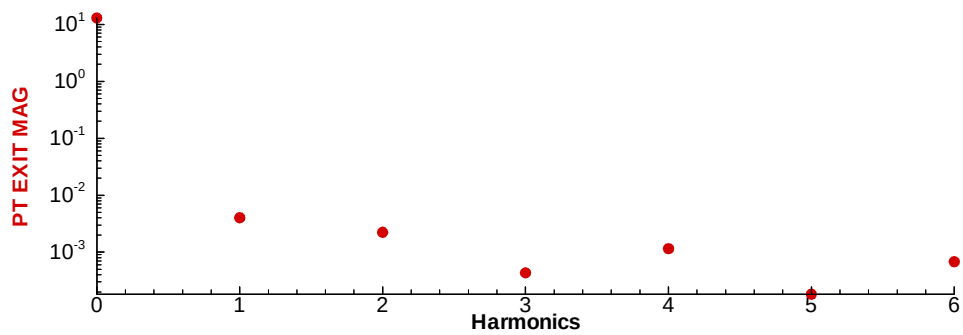
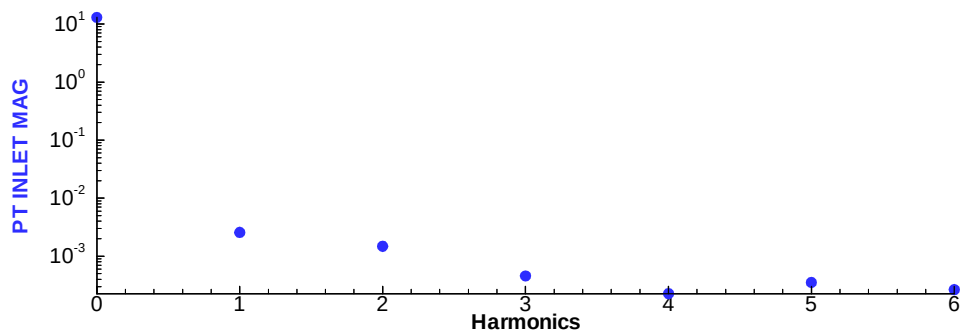
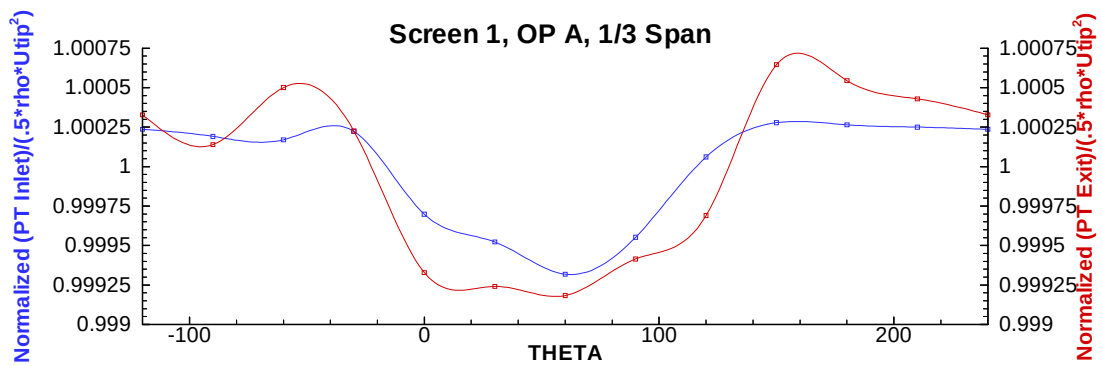
Graphical representation of dynamic response included

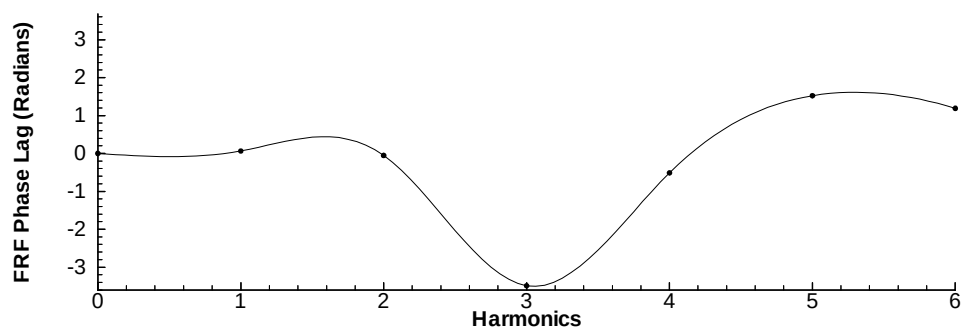
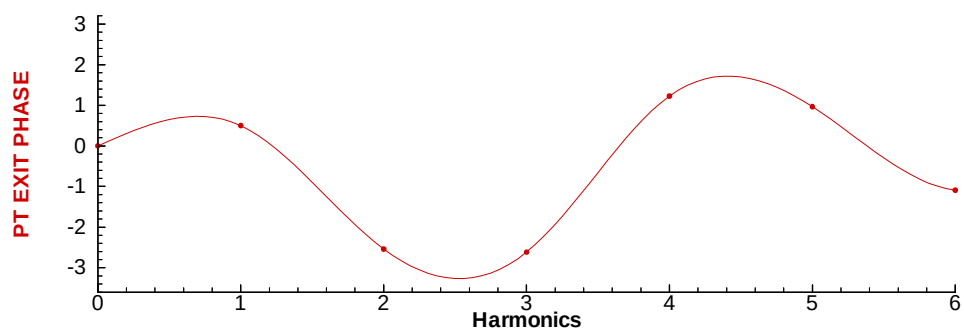
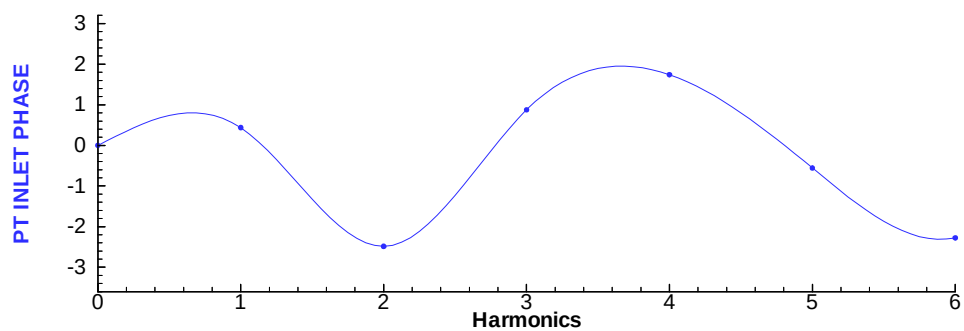
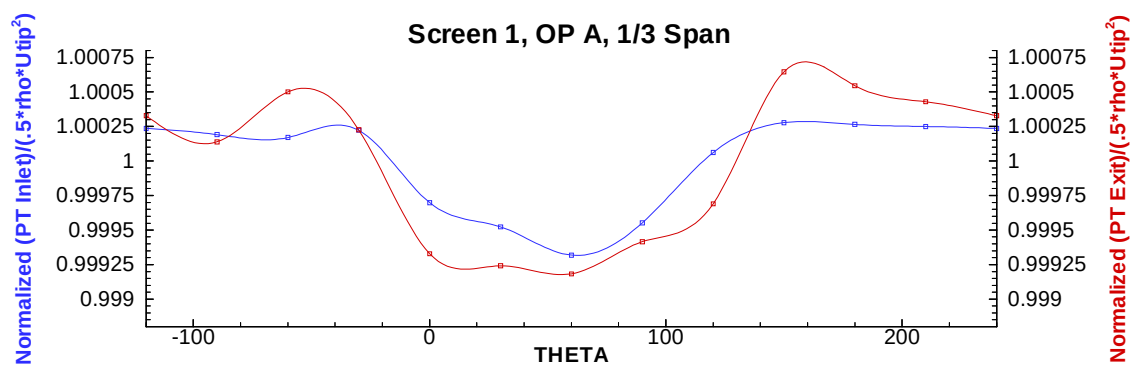
DATA**LOCATION**

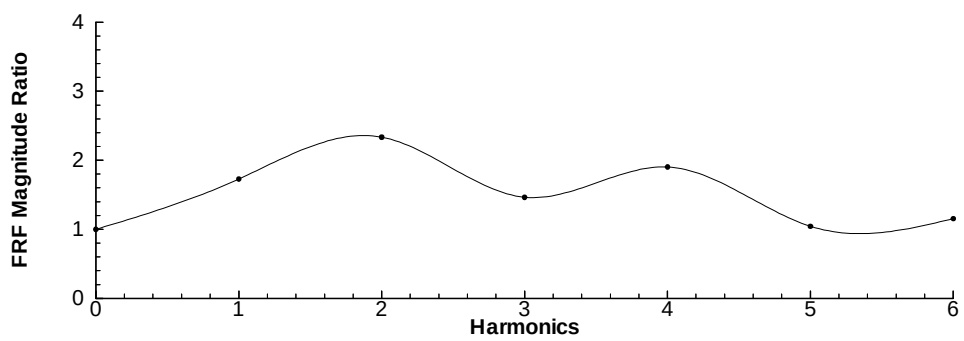
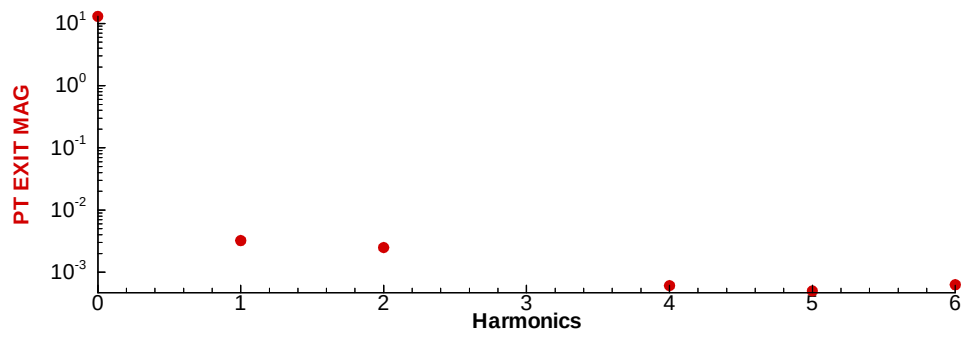
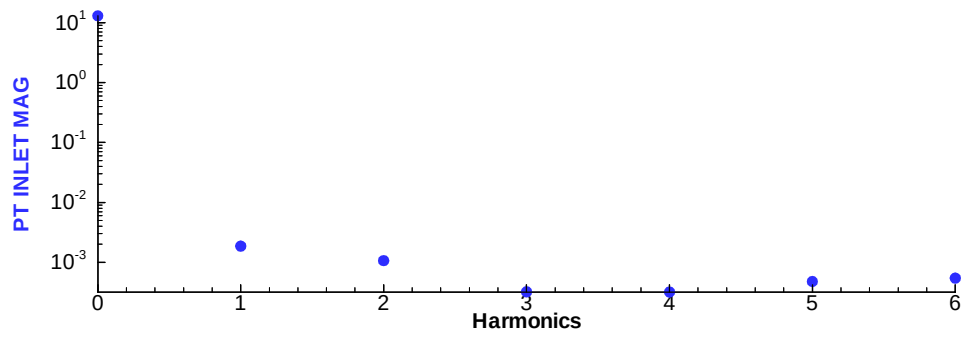
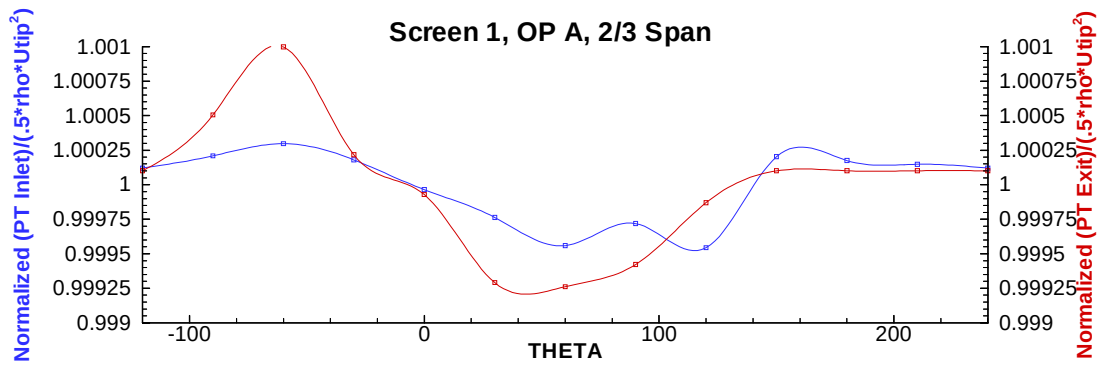
Distortion Screen	Operating Point	Measurement Span	Page Numbers
Level 1	Operating Point A	50 %, 33 %, 67 %	pp. 172-177
Level 1	Operating Point B	50 %, 33 %, 67 %	pp. 178-183
Level 1	Operating Point C	50 %, 33 %, 67 %	pp. 188-192
Level 2	Operating Point A	50 %, 33 %, 67 %	pp. 193-198
Level 2	Operating Point B	50 %, 33 %, 67 %	pp. 199-204
Level 2	Operating Point C	50 %, 33 %, 67 %	pp. 205-210
Level 3	Operating Point D	50 %, 33 %, 67 %	pp. 211-216

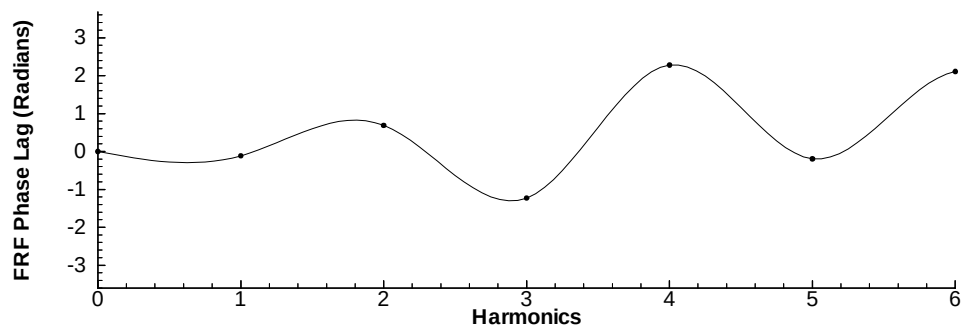
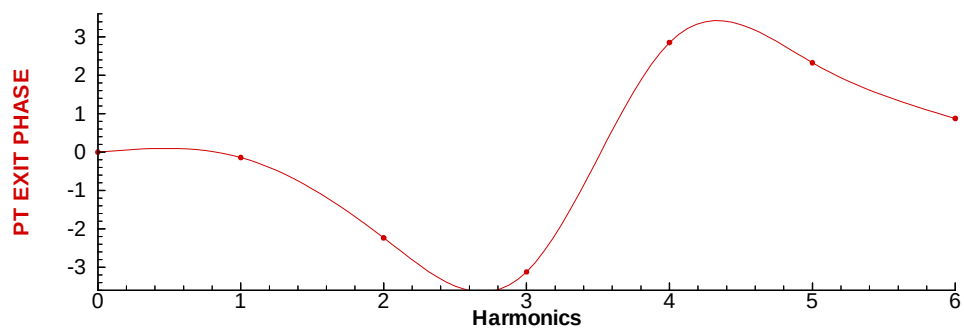
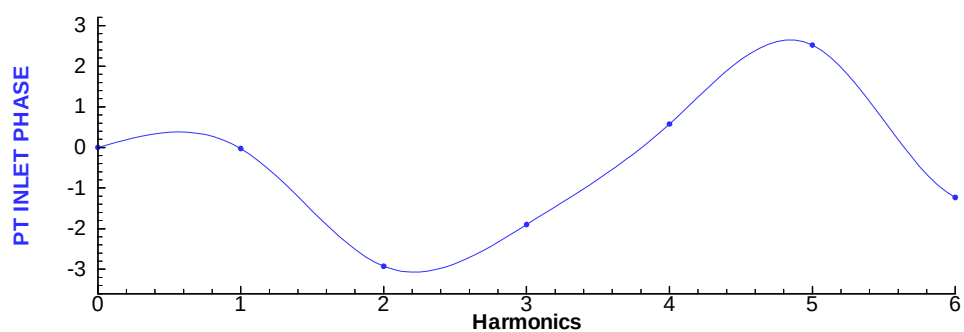
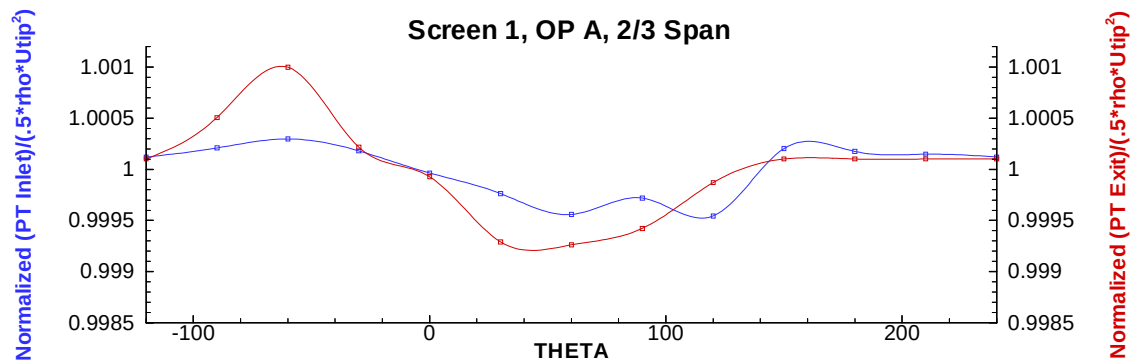


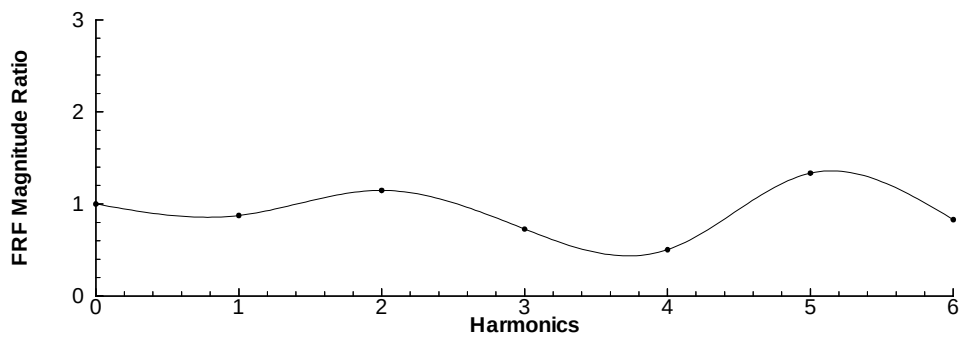
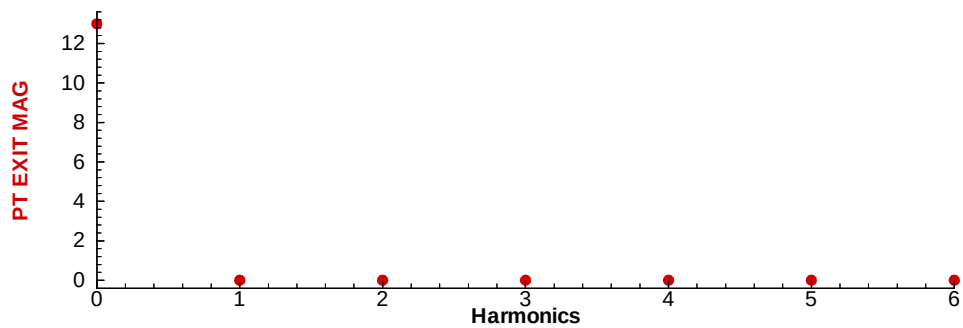
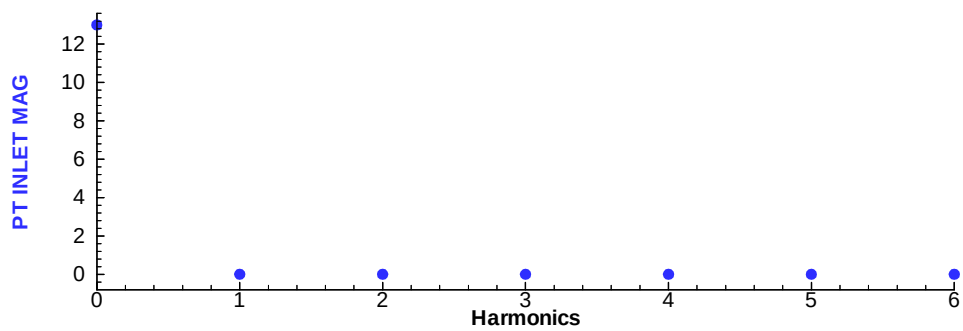
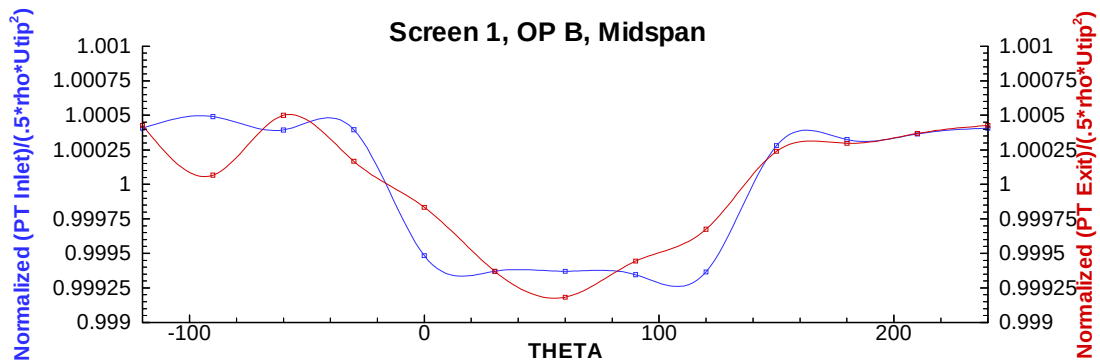


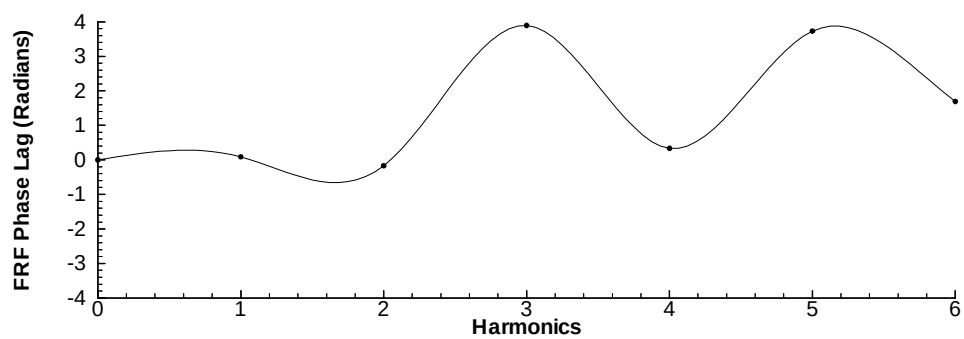
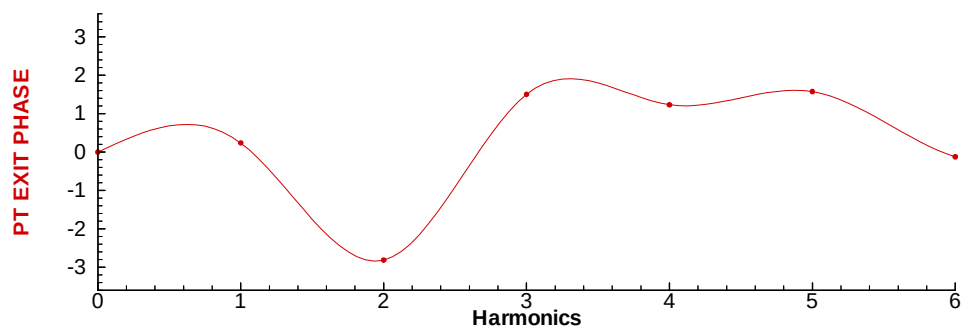
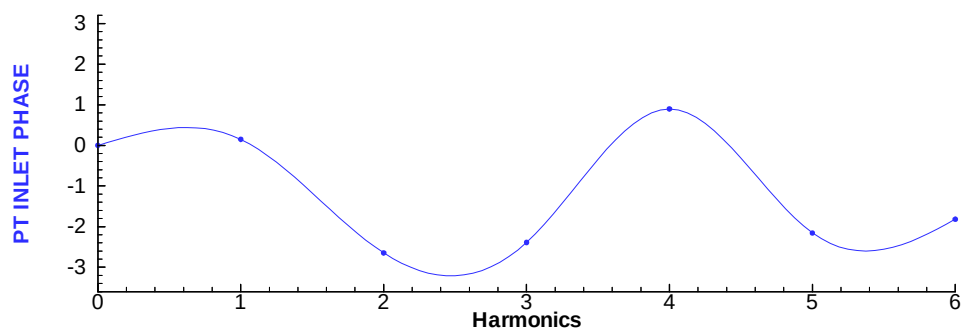
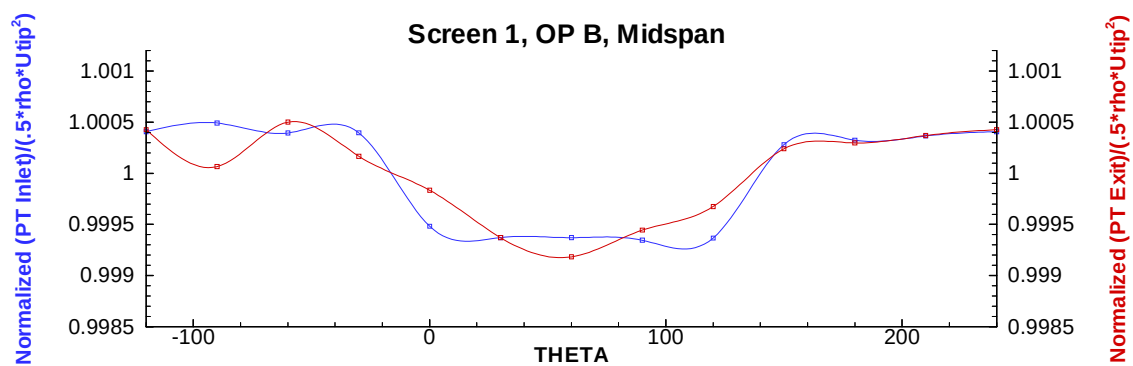


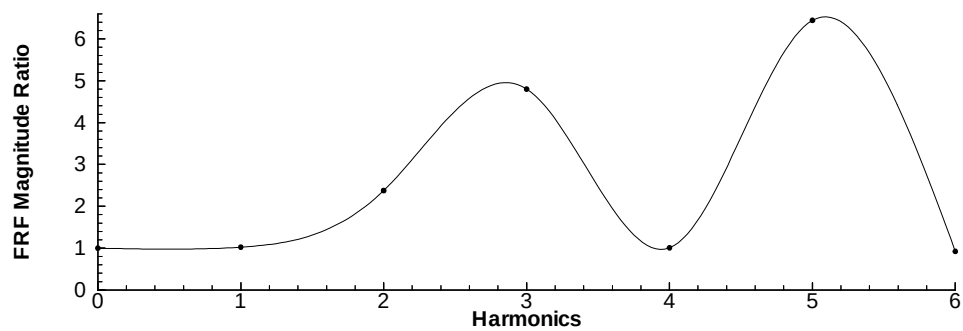
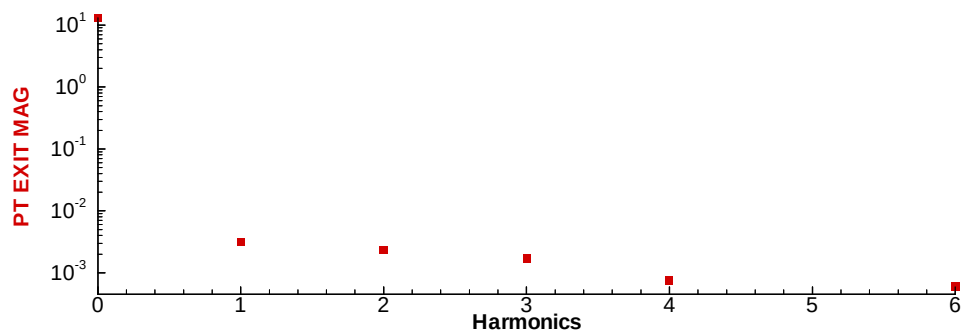
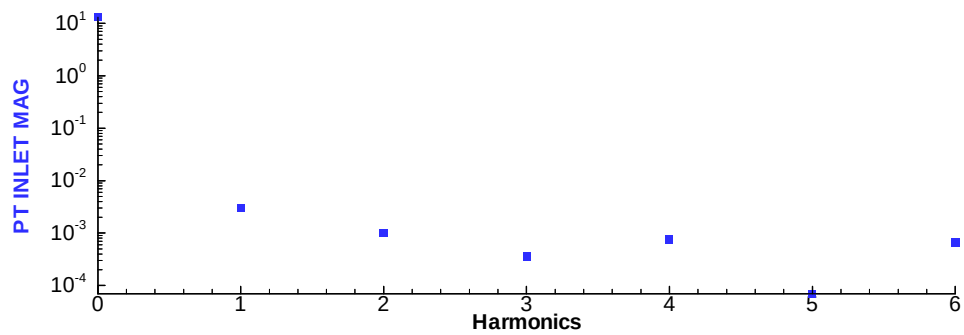
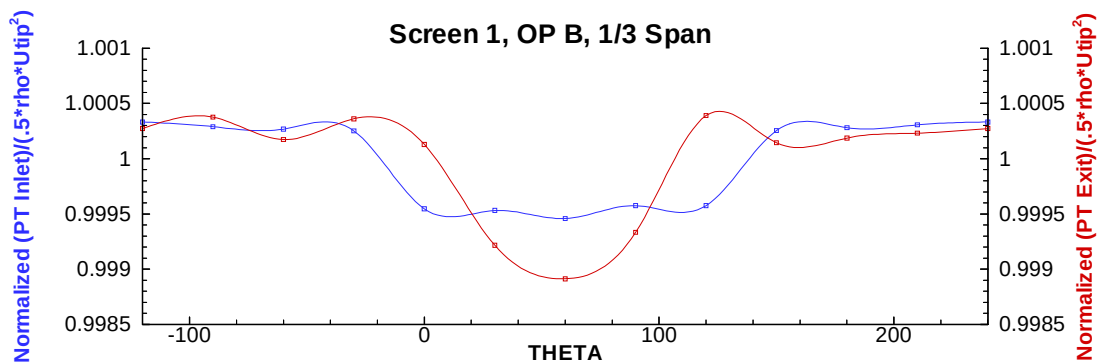


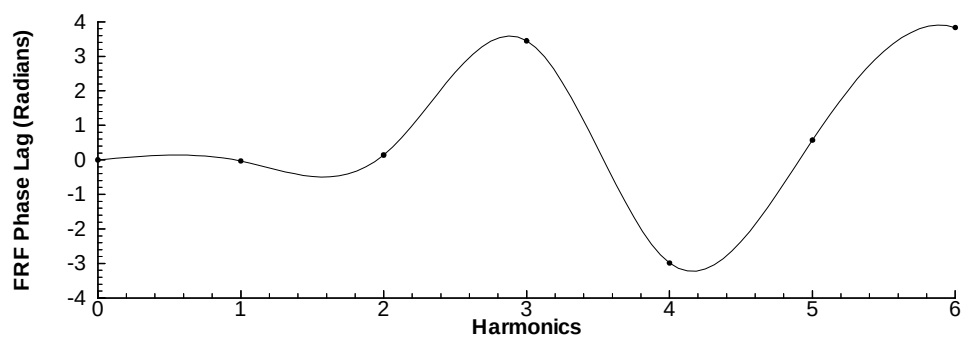
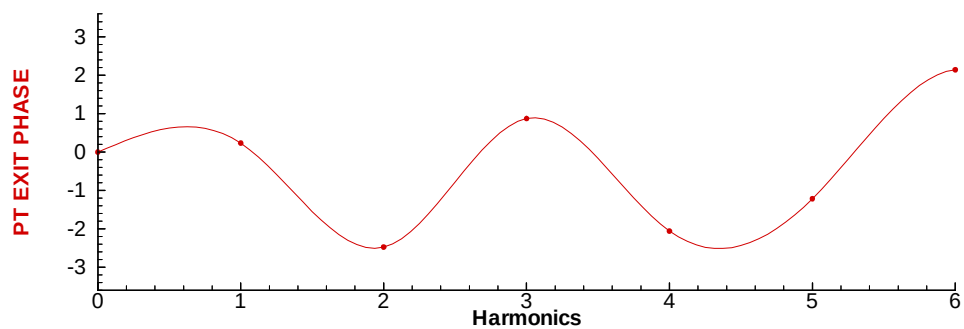
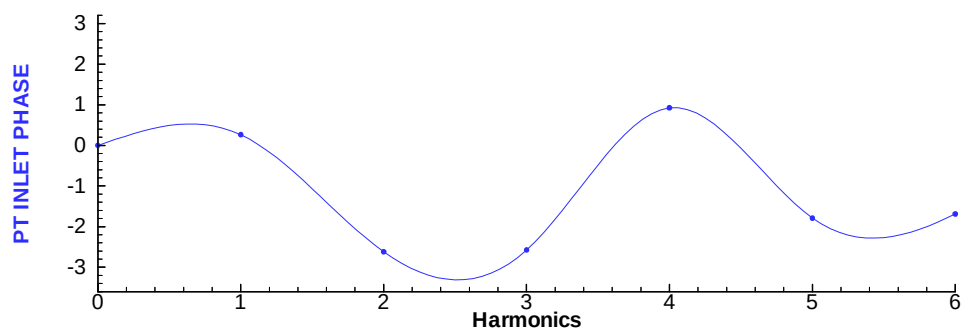
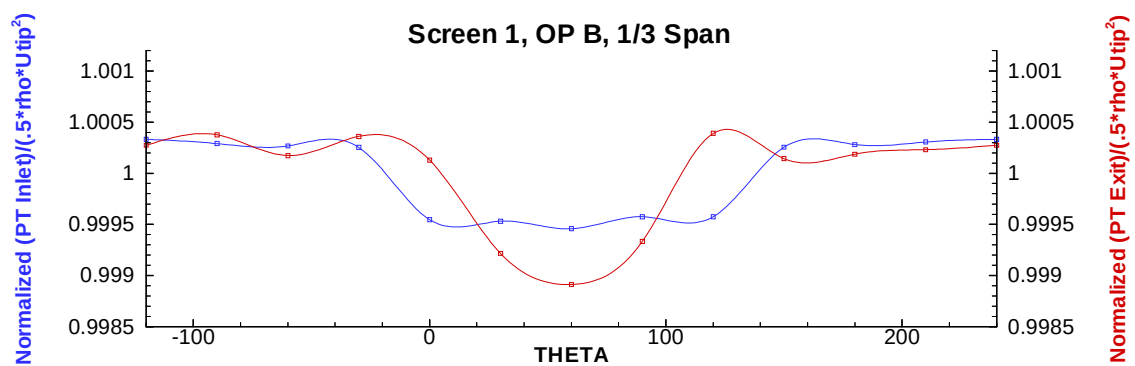


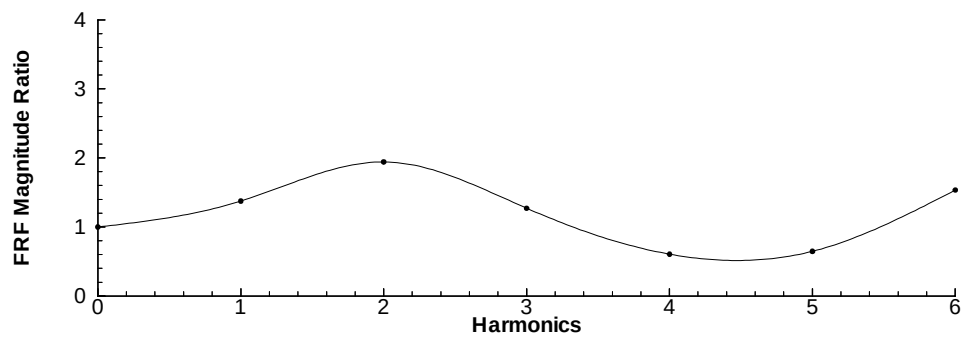
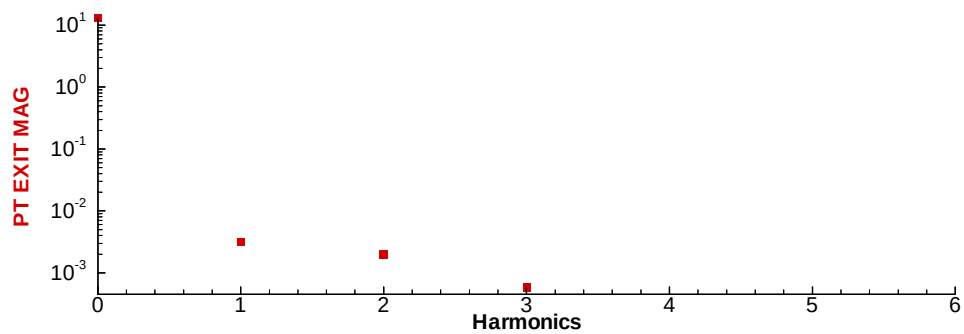
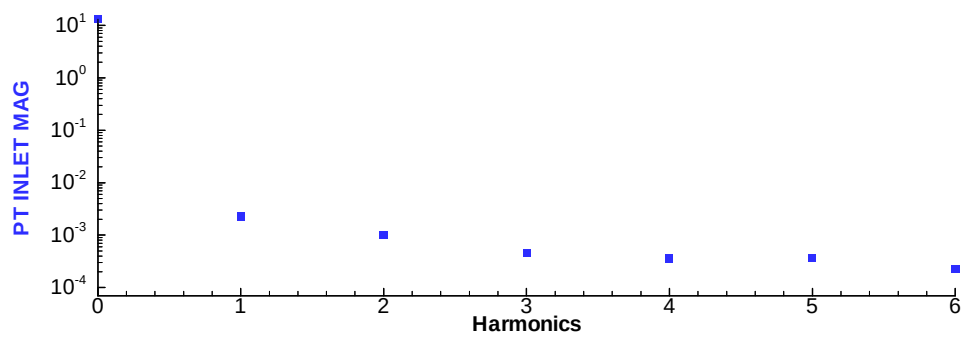
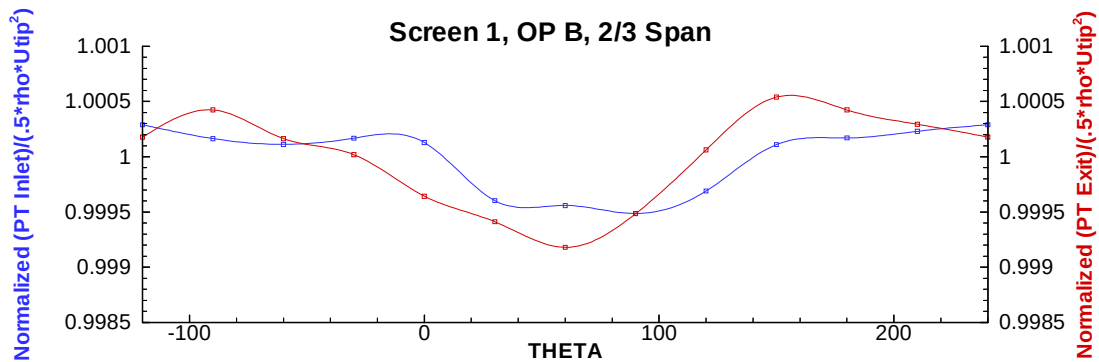


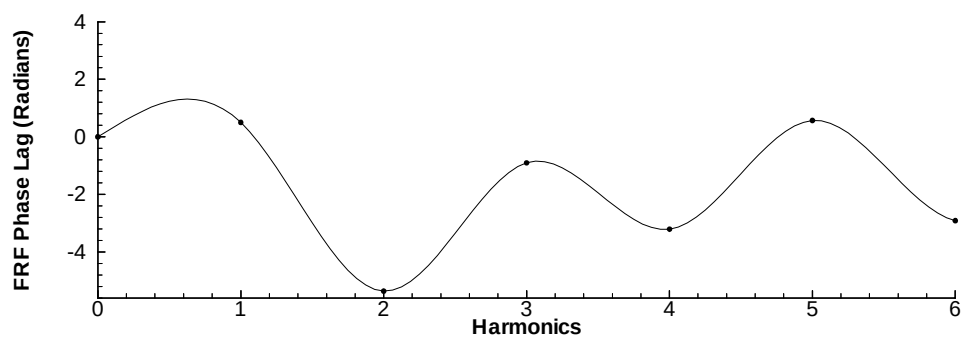
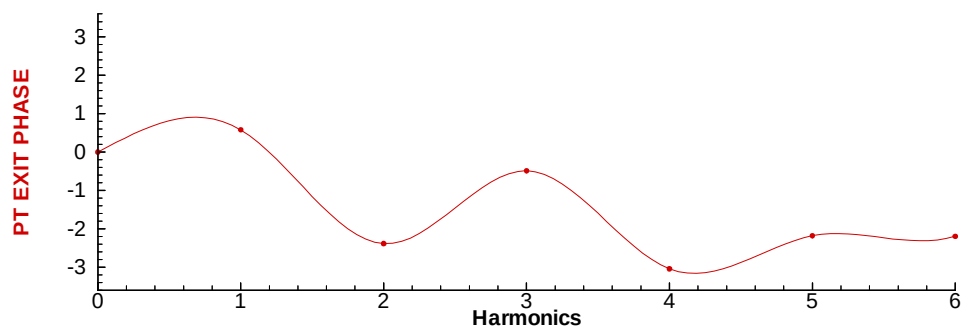
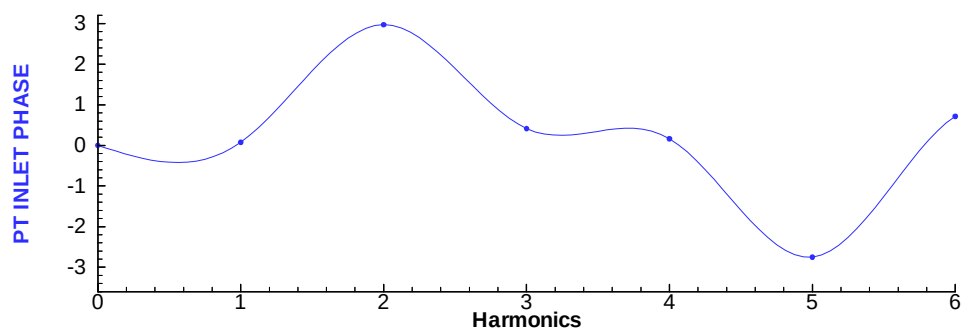
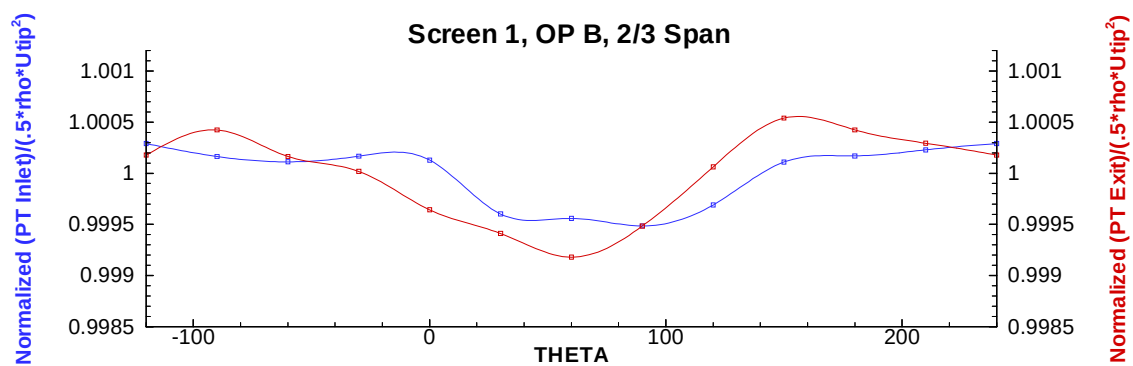


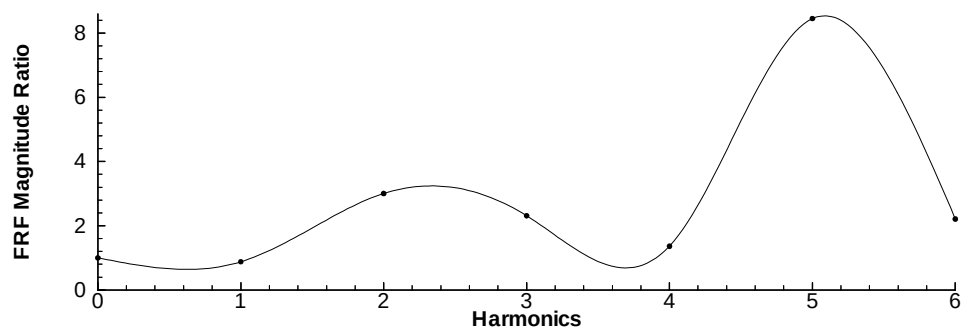
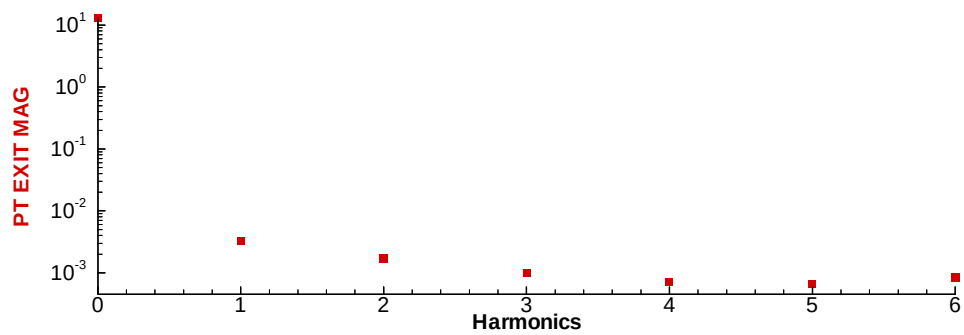
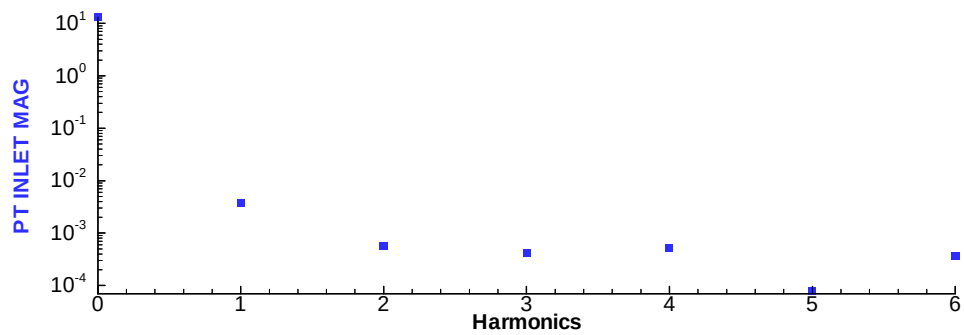
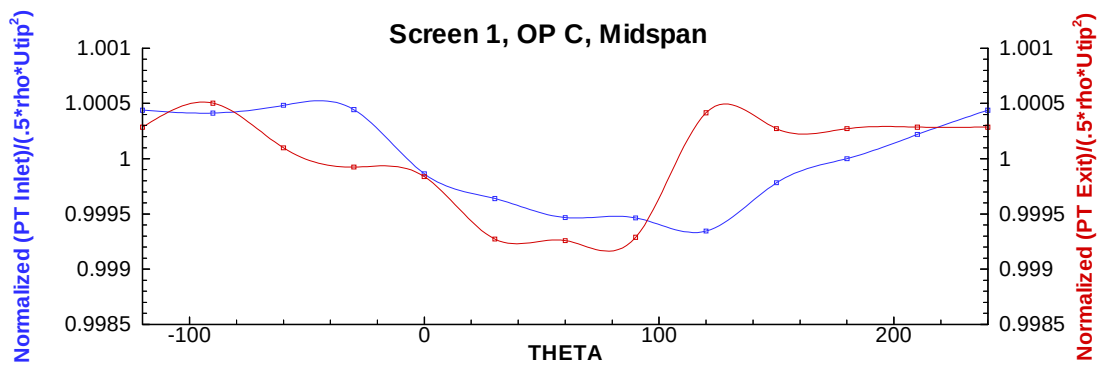


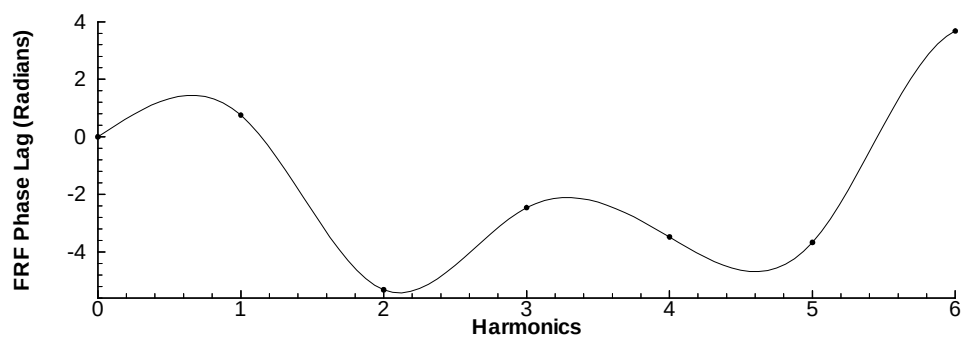
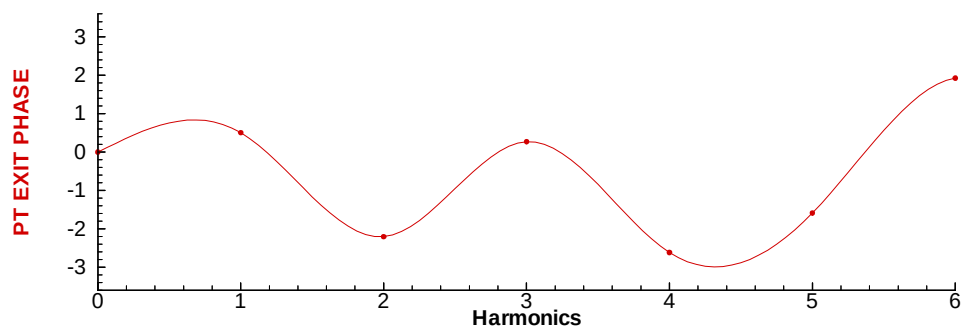
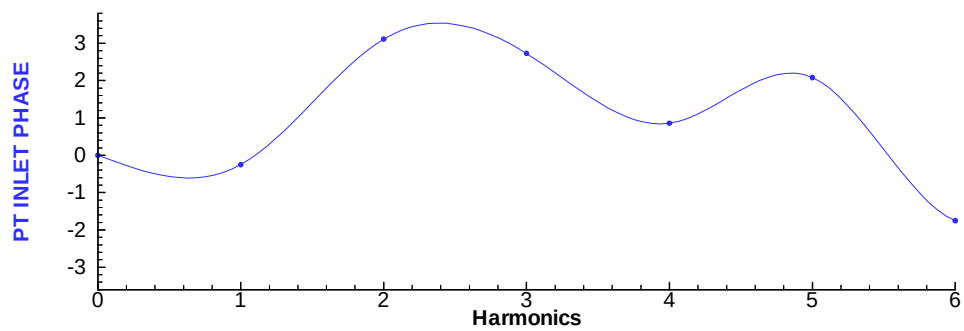
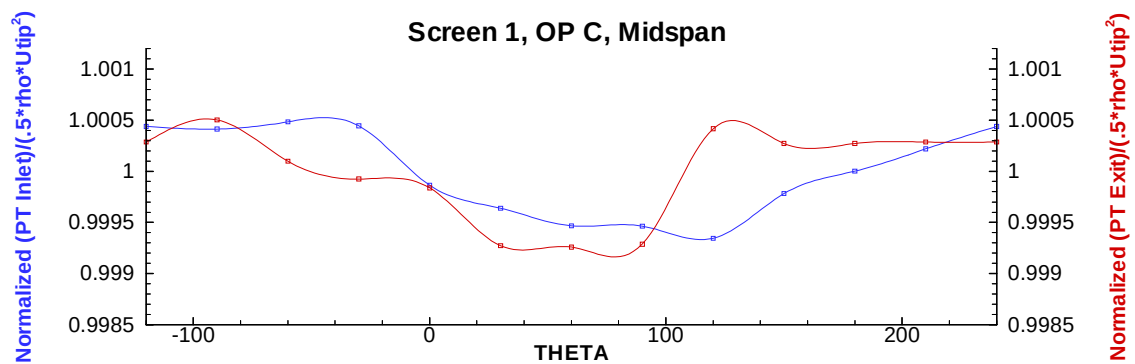


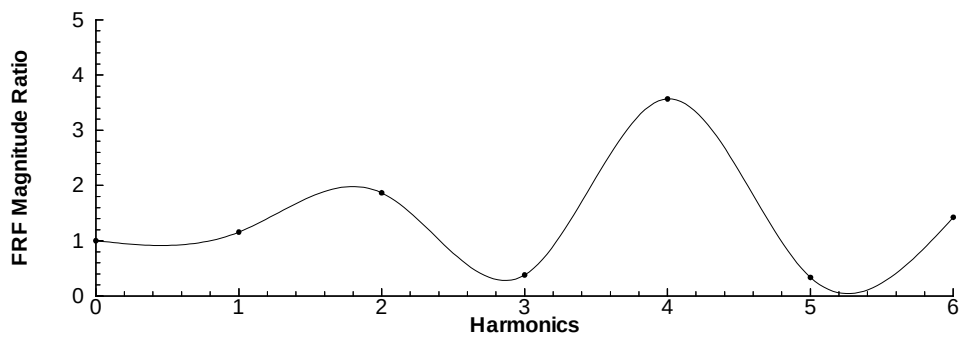
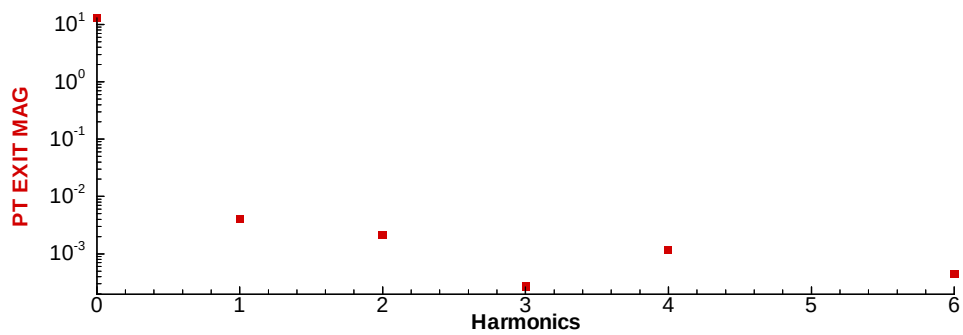
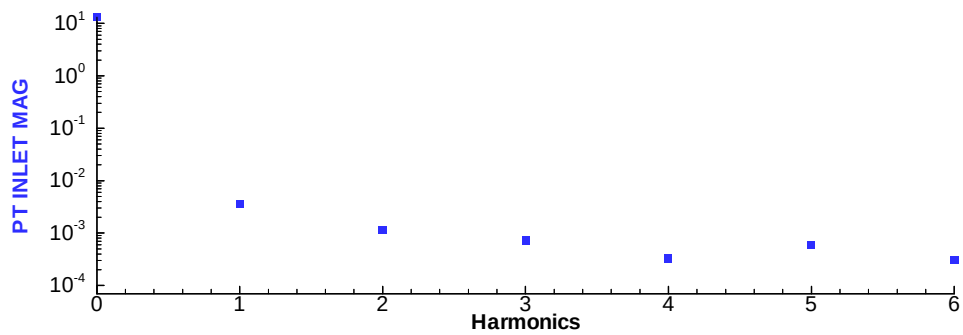
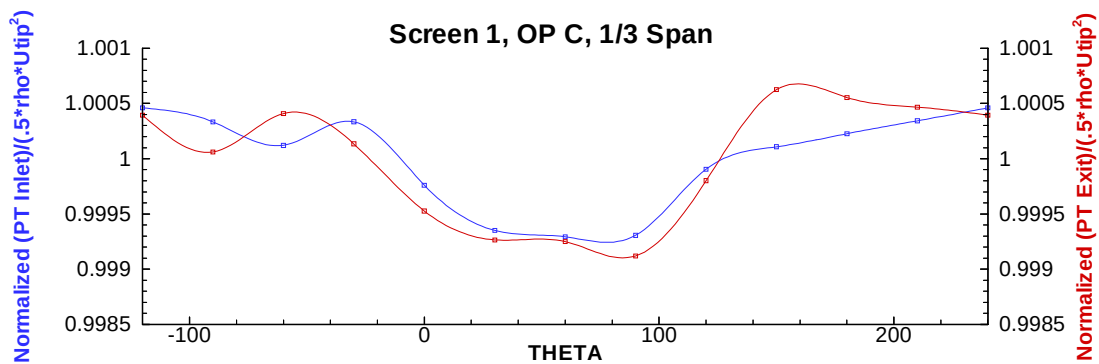


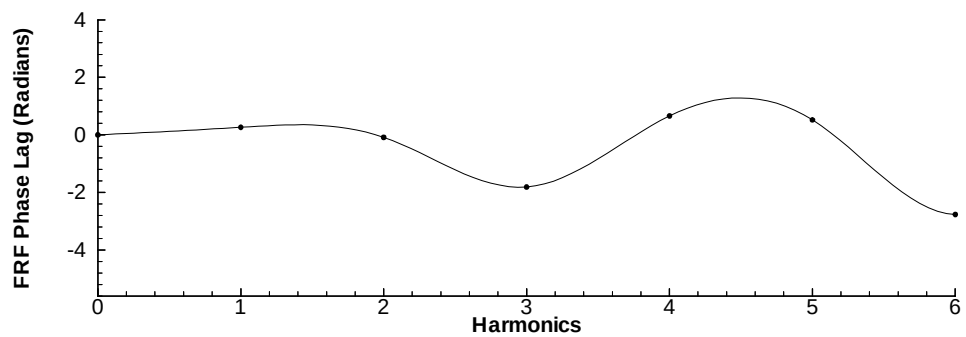
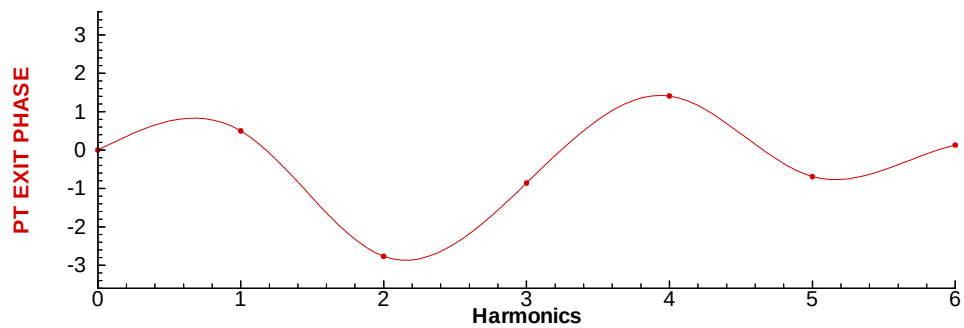
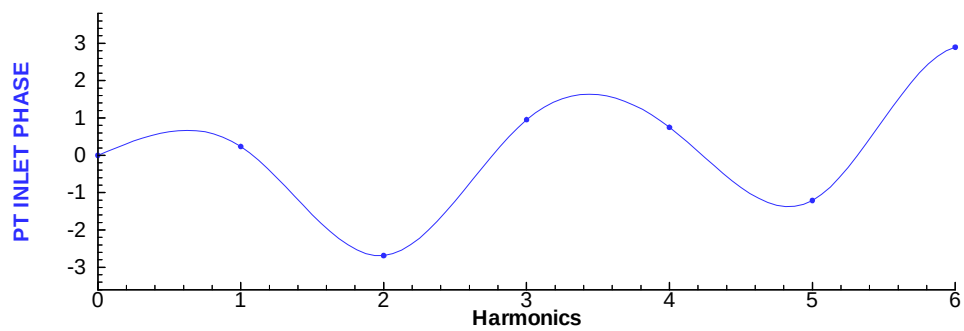
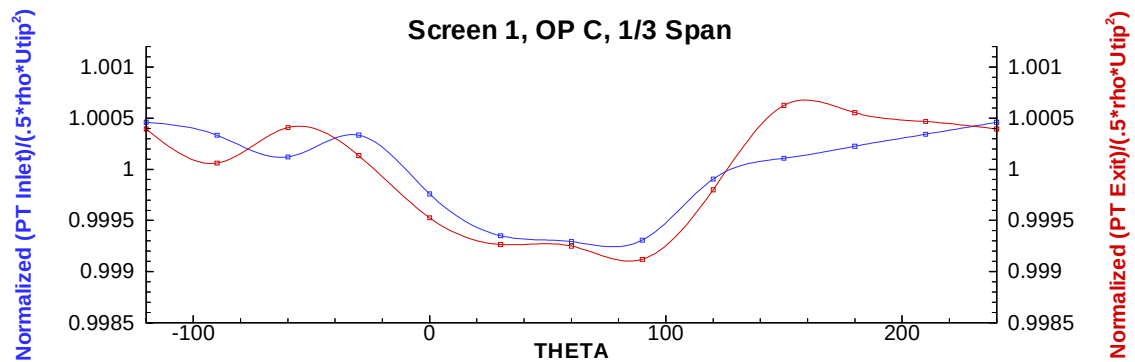


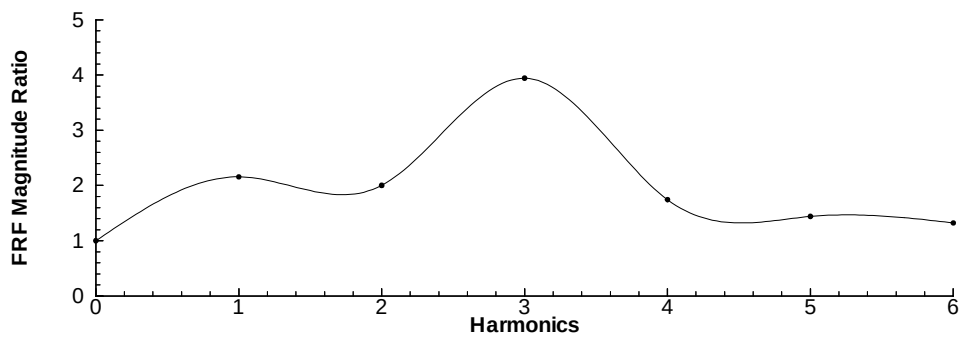
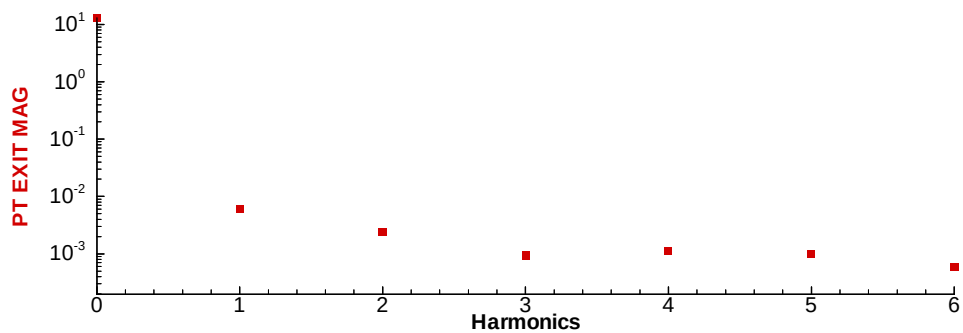
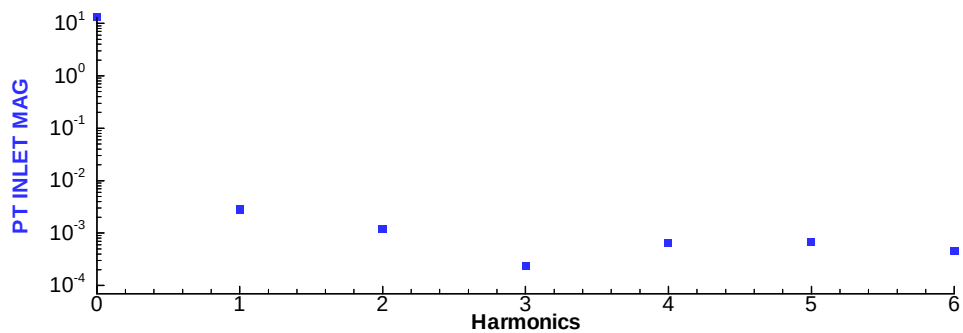
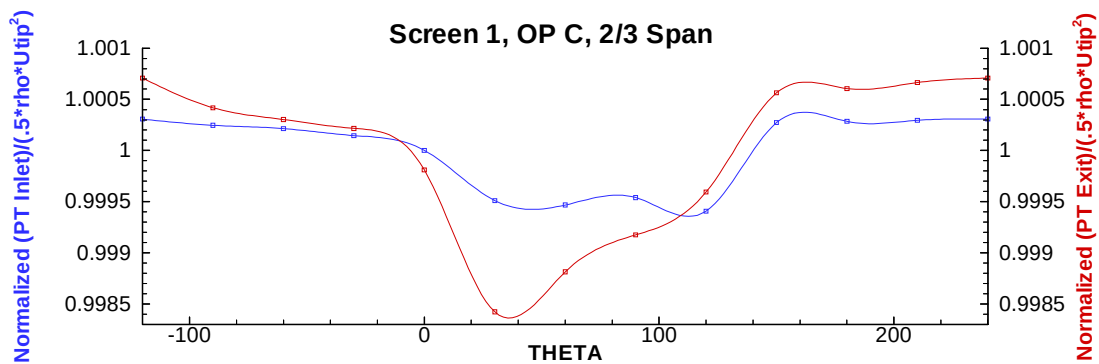


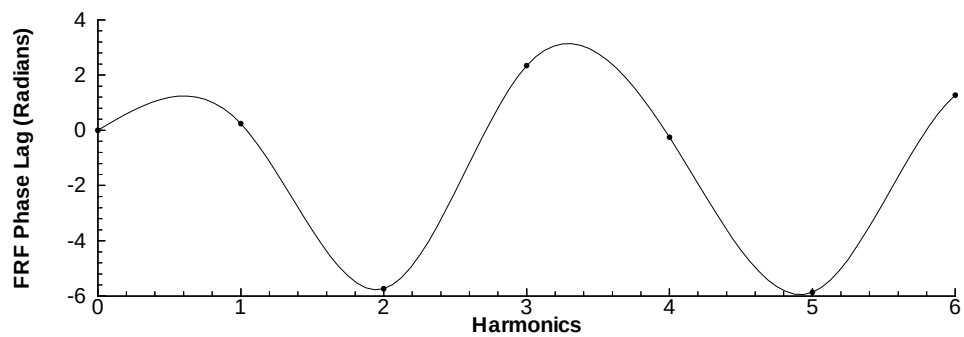
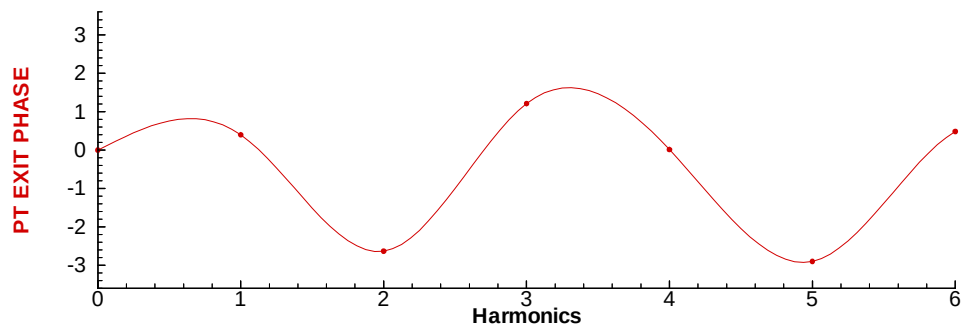
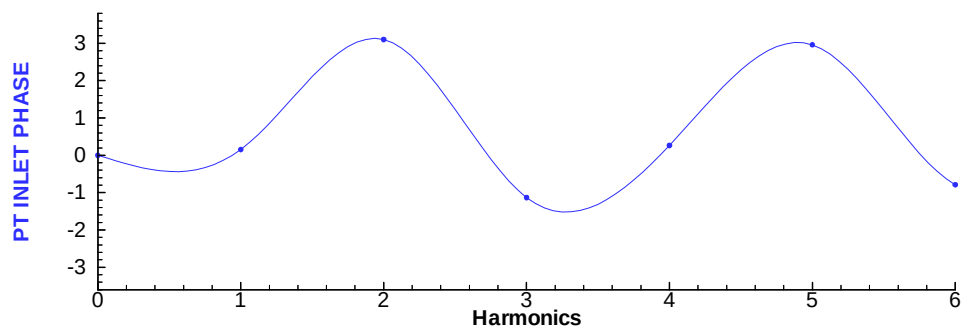
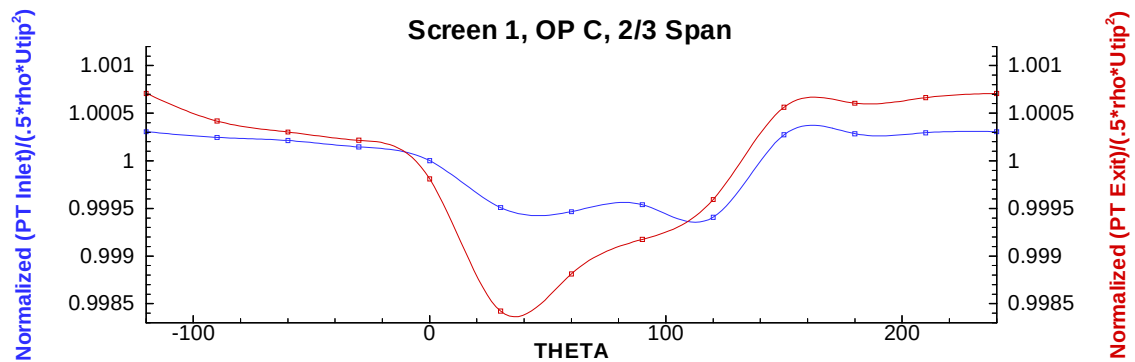


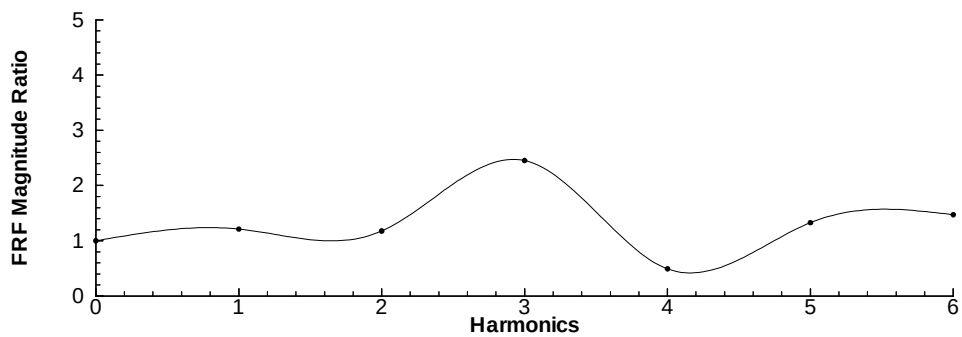
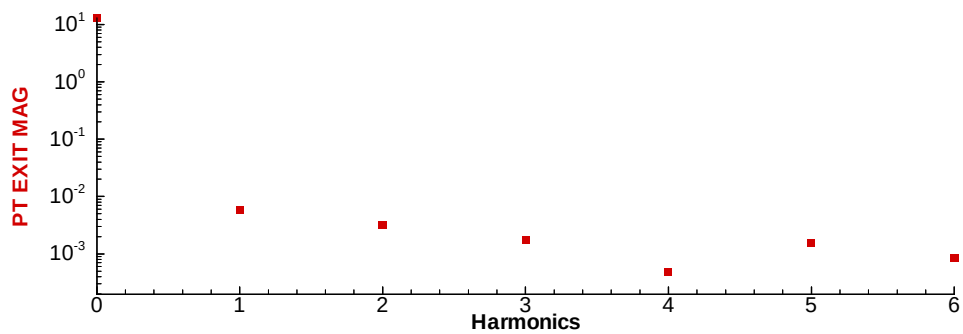
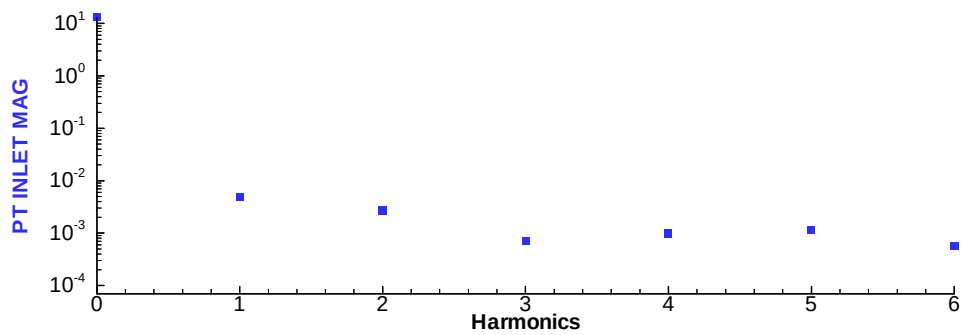
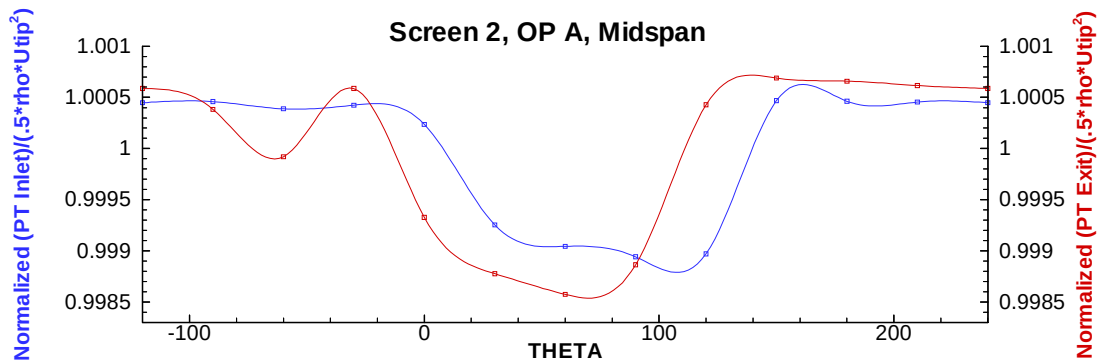


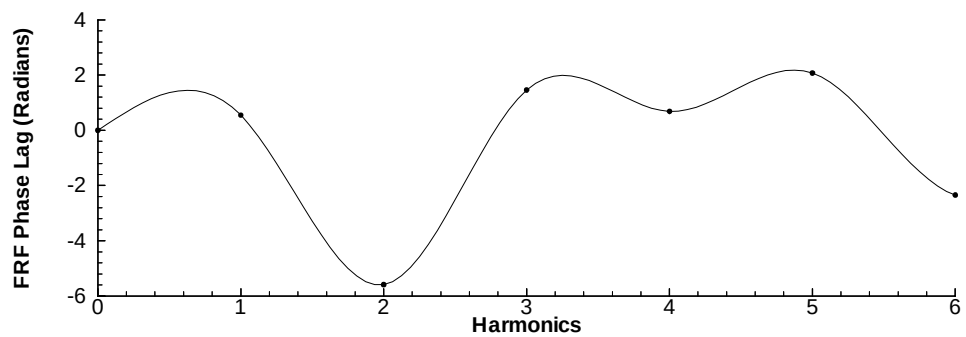
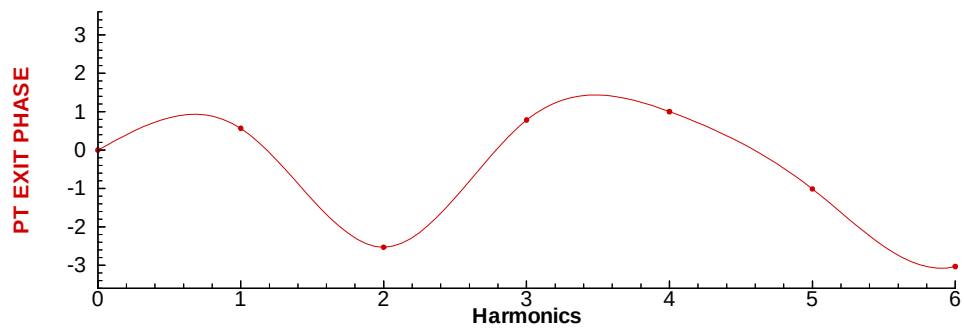
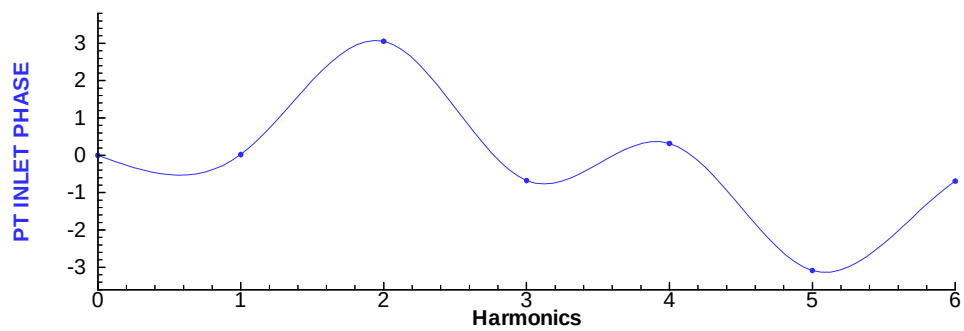
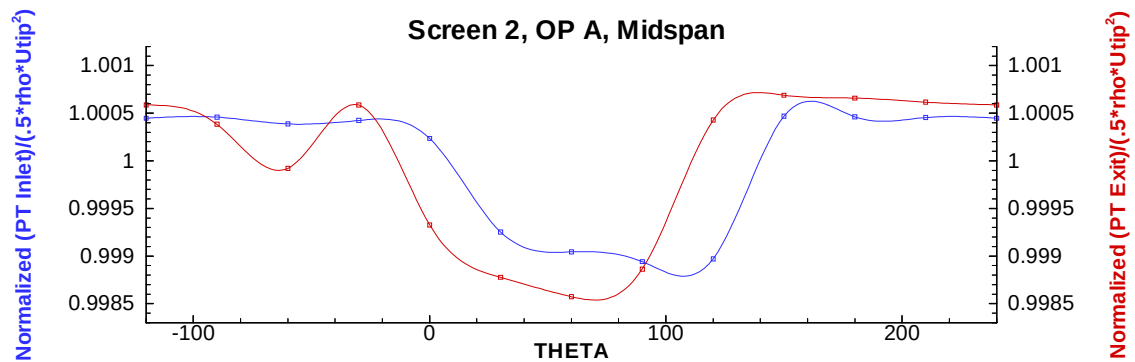


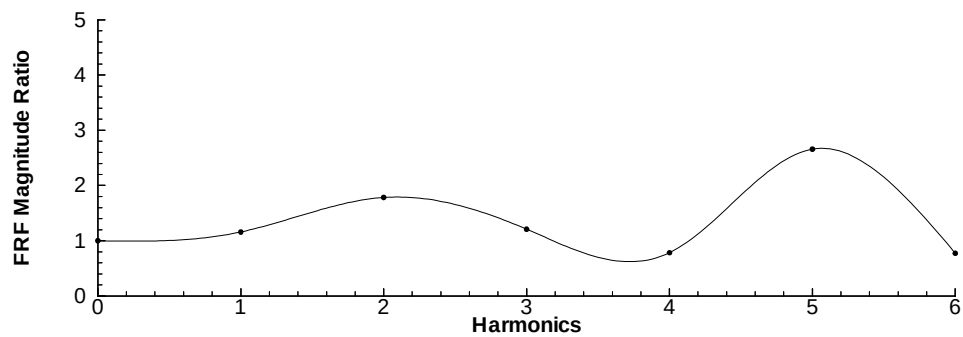
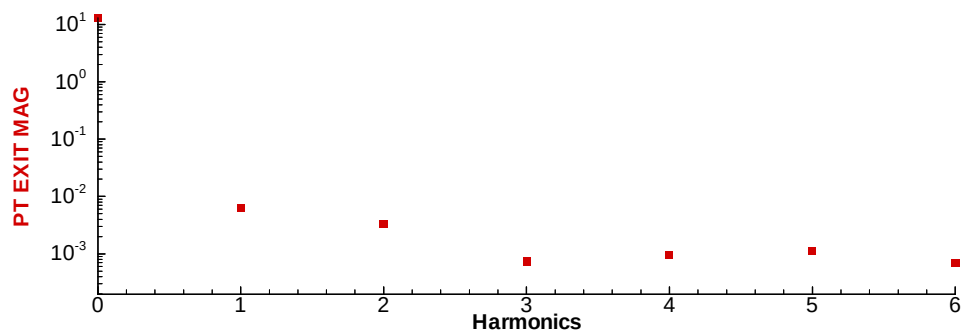
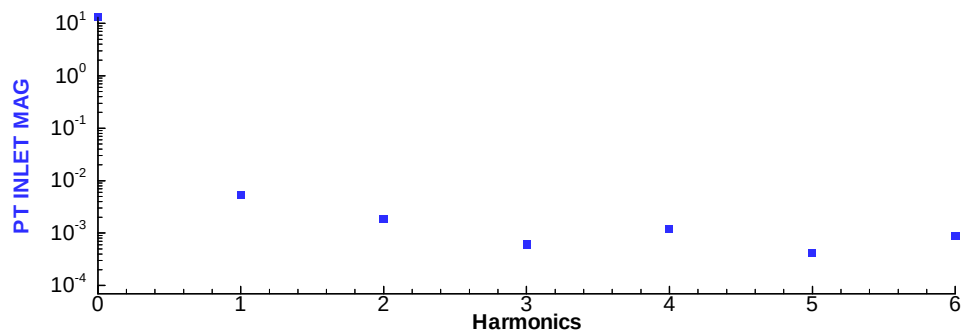
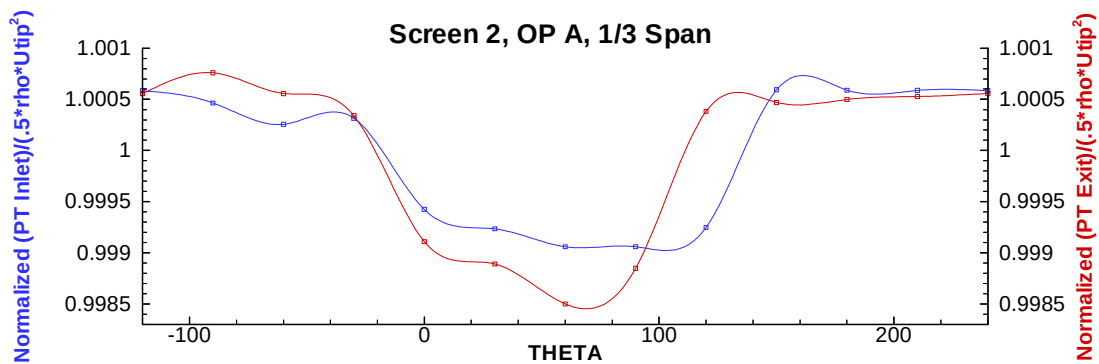


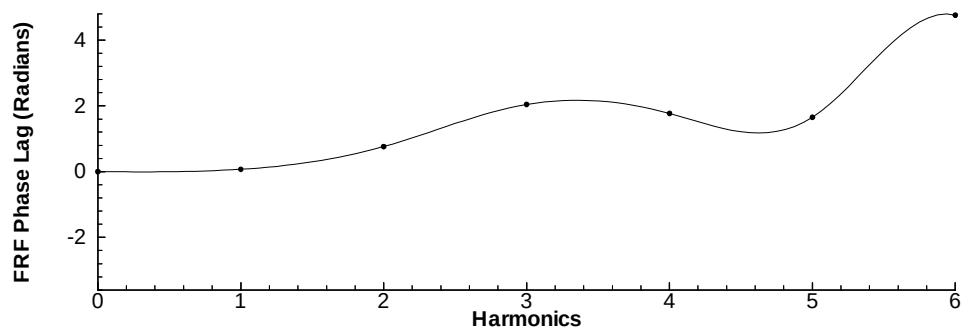
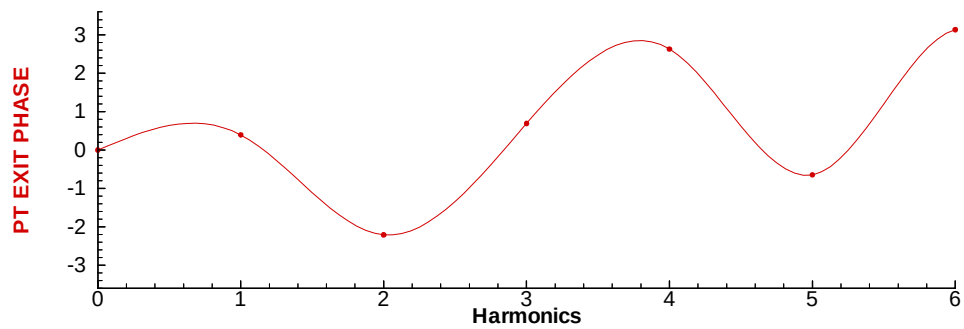
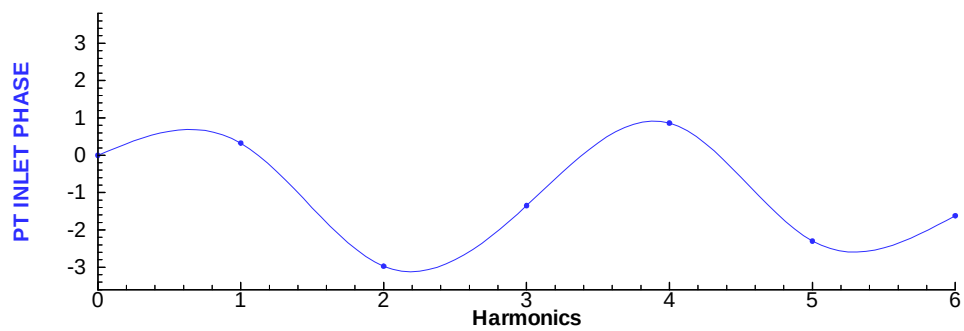
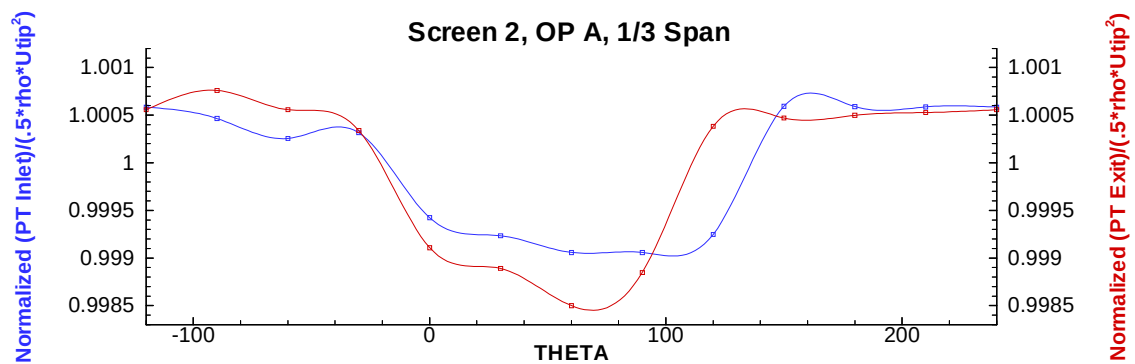


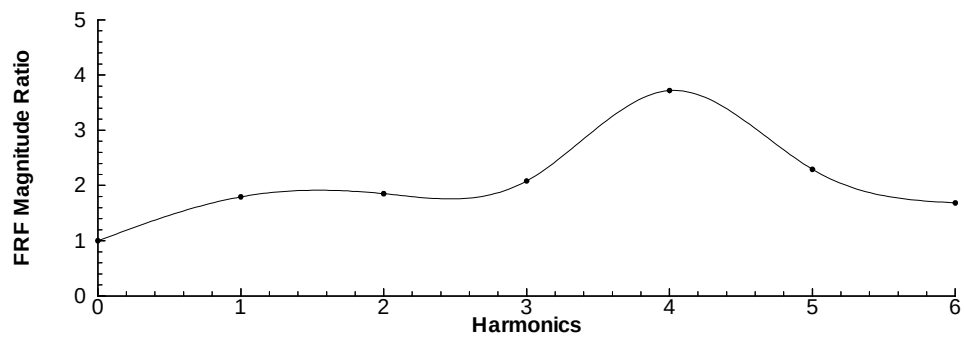
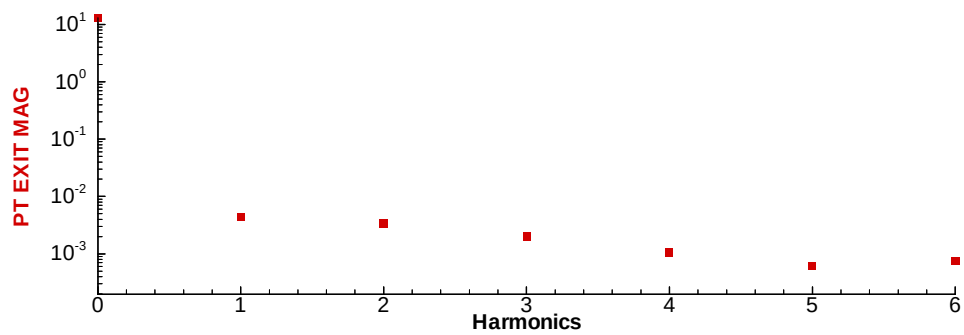
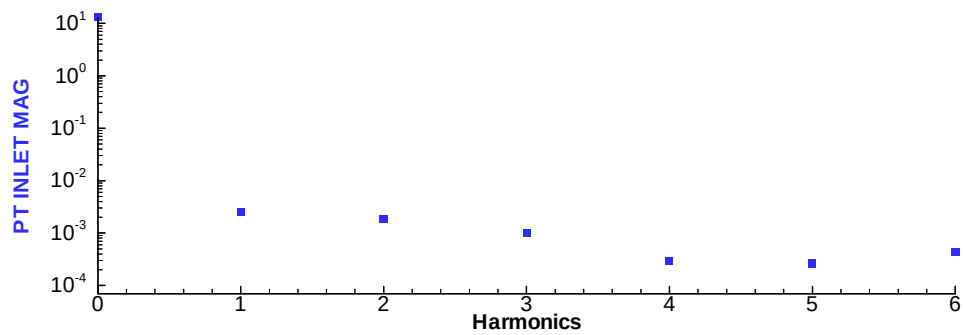
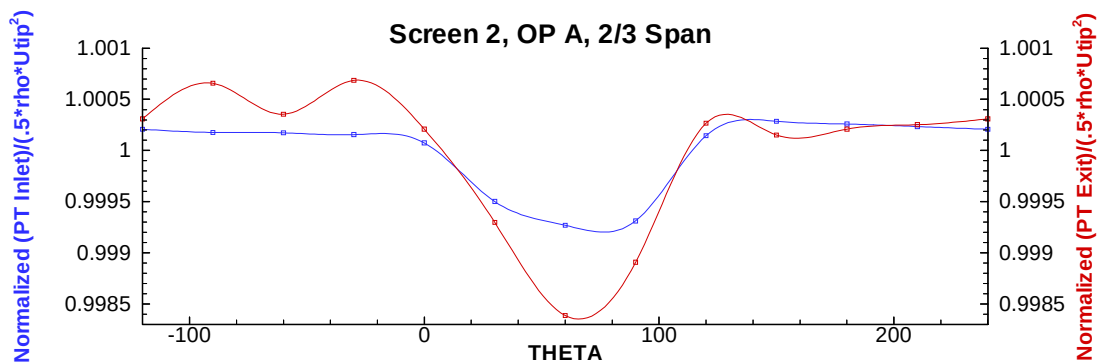


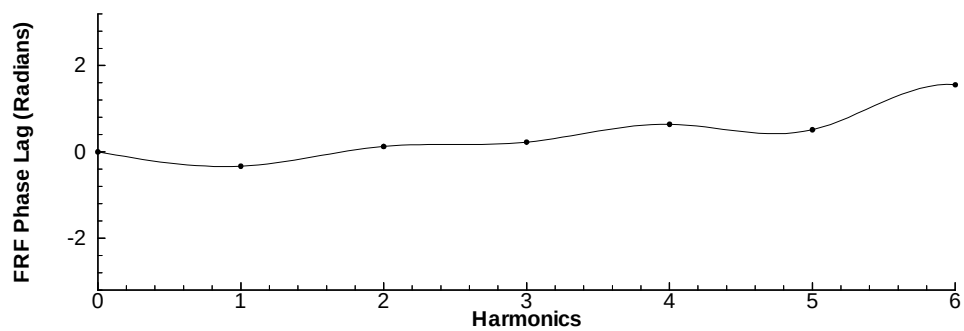
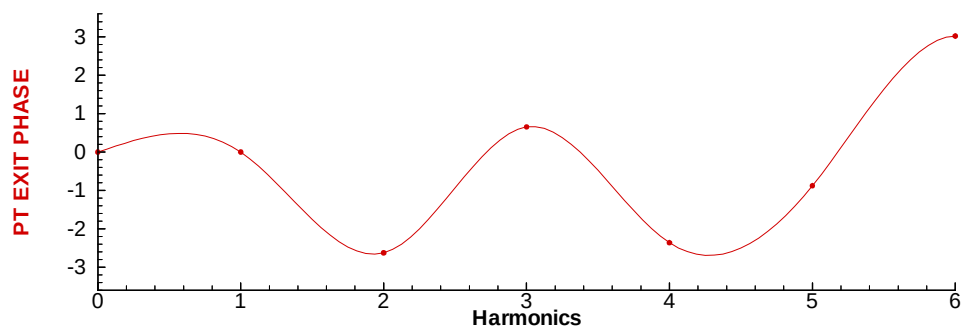
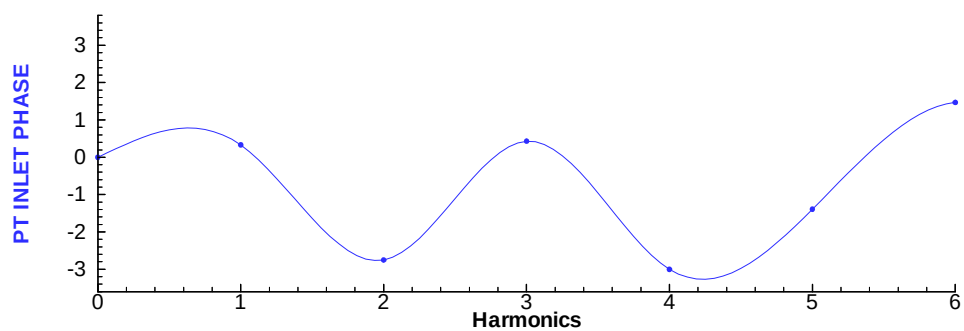
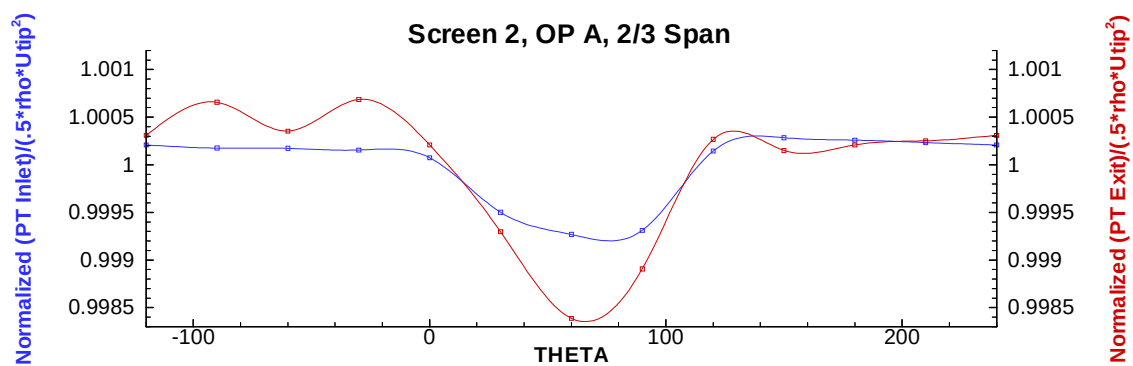


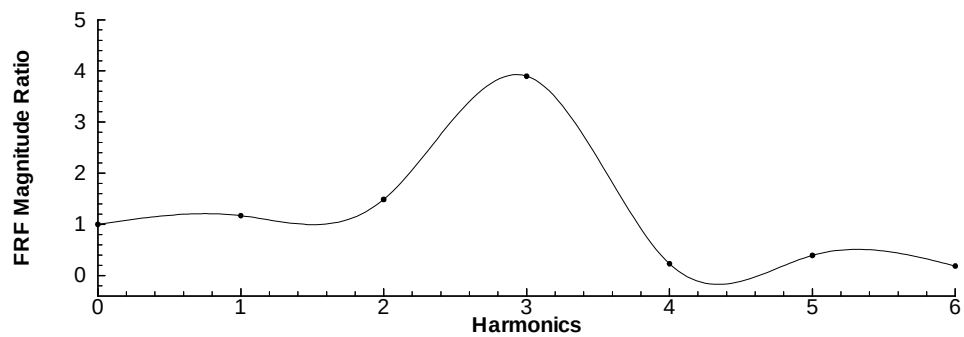
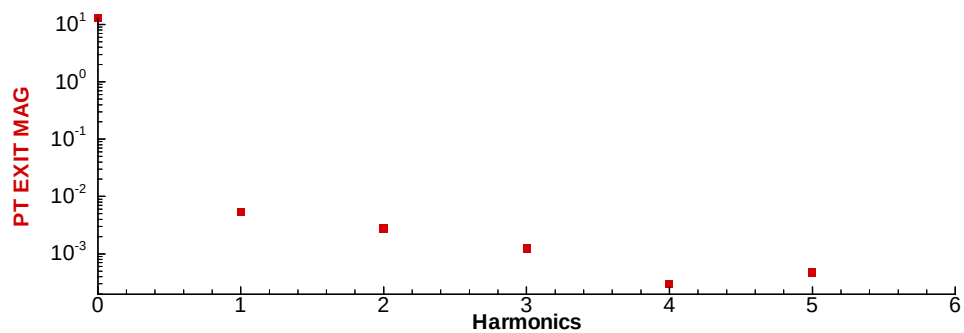
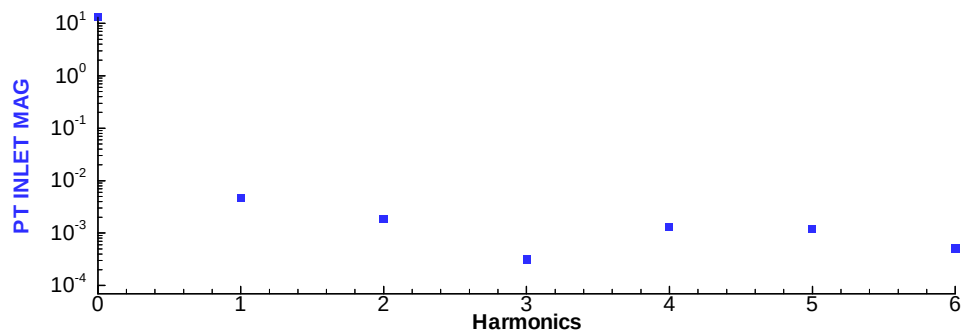
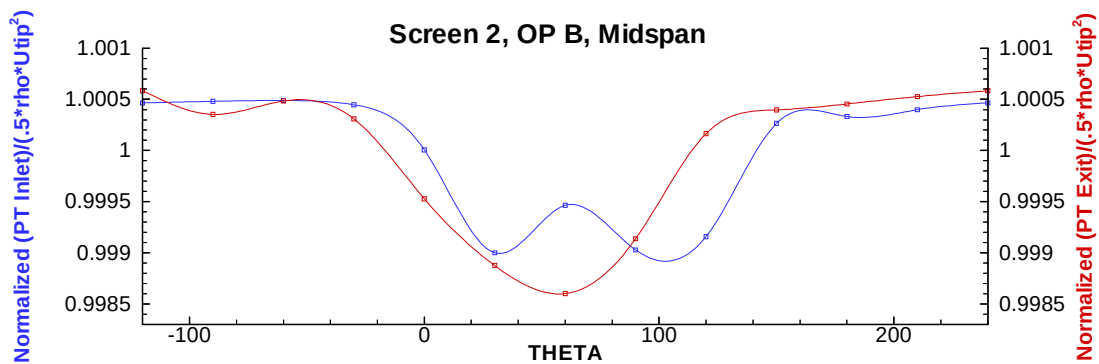


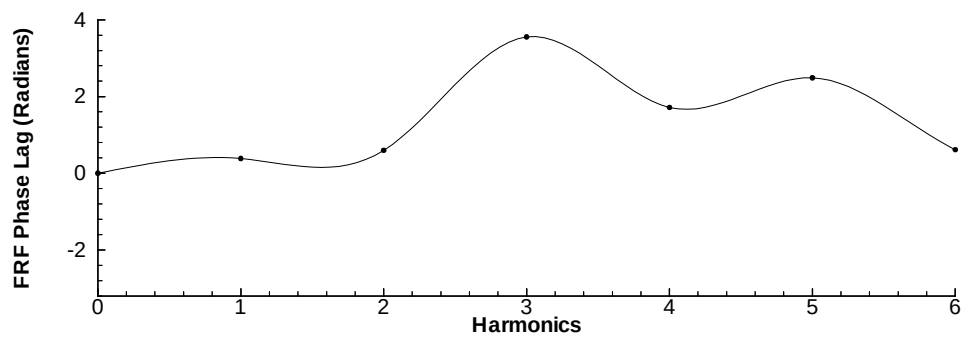
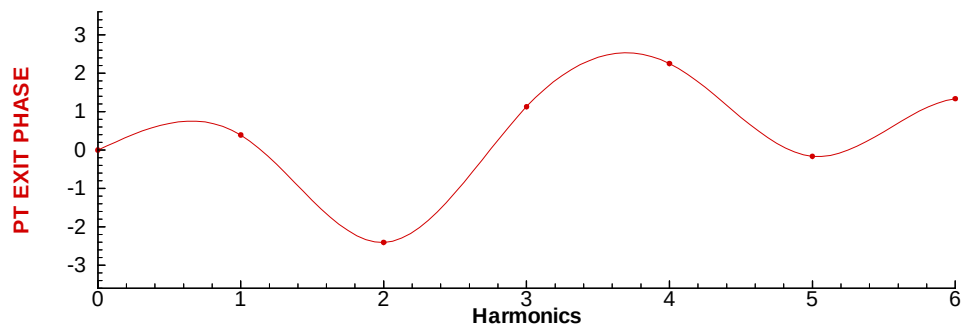
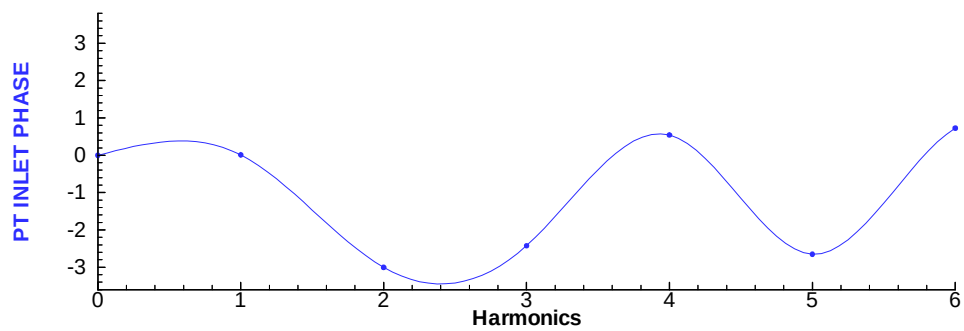
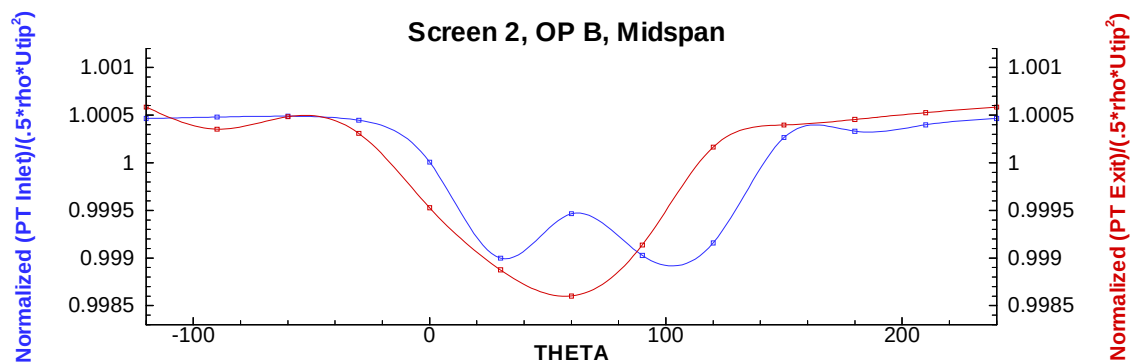


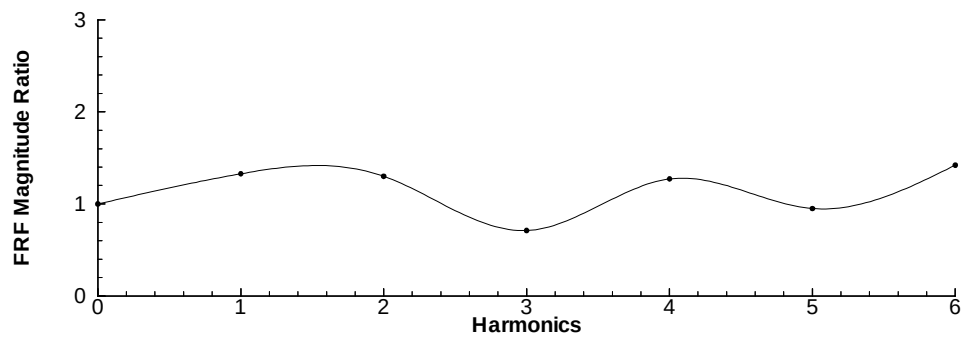
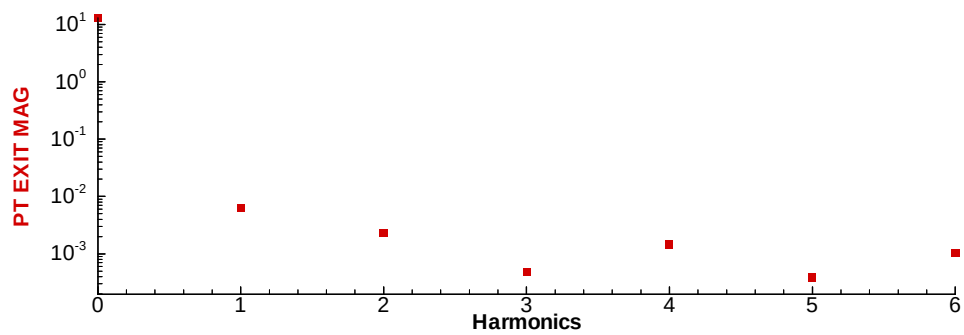
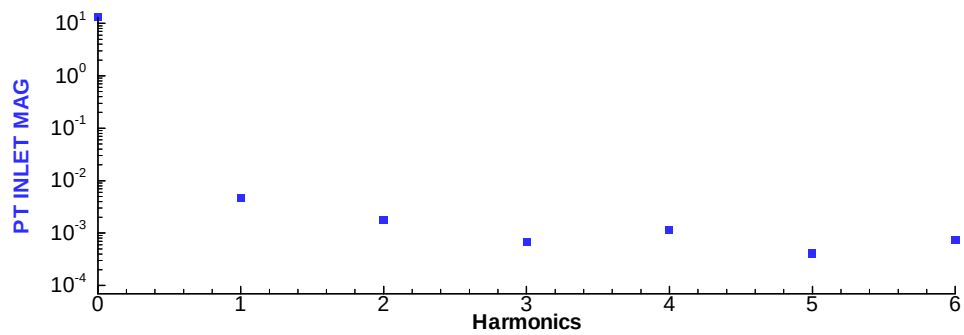
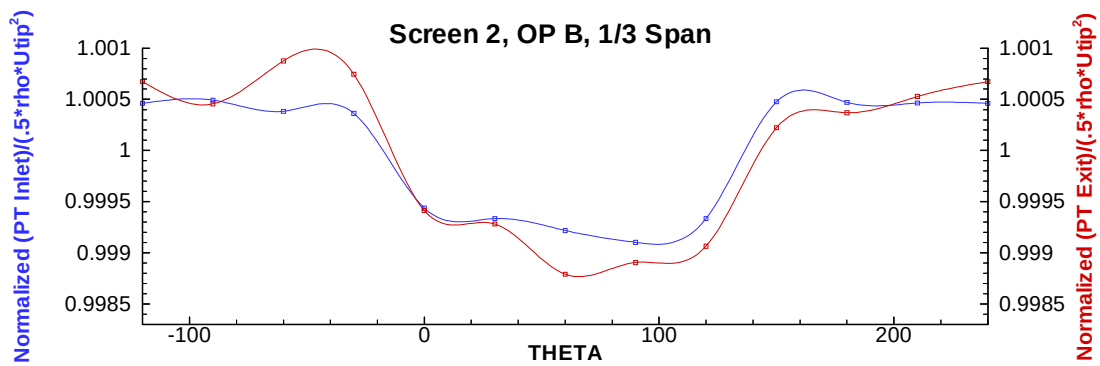


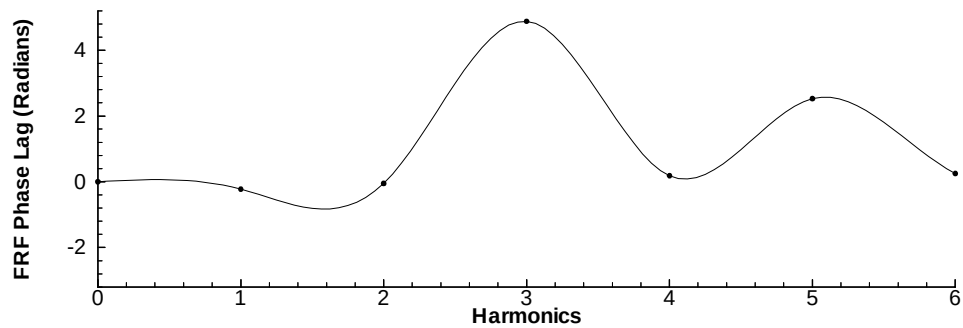
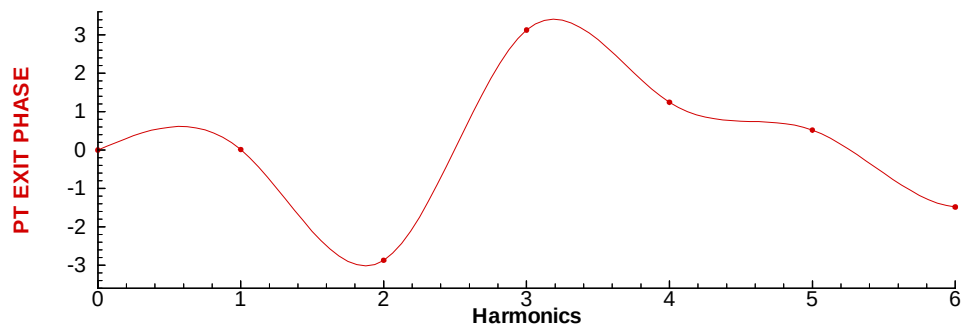
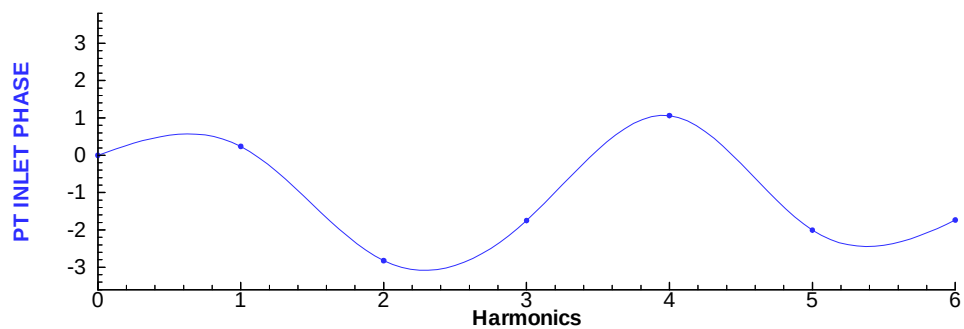
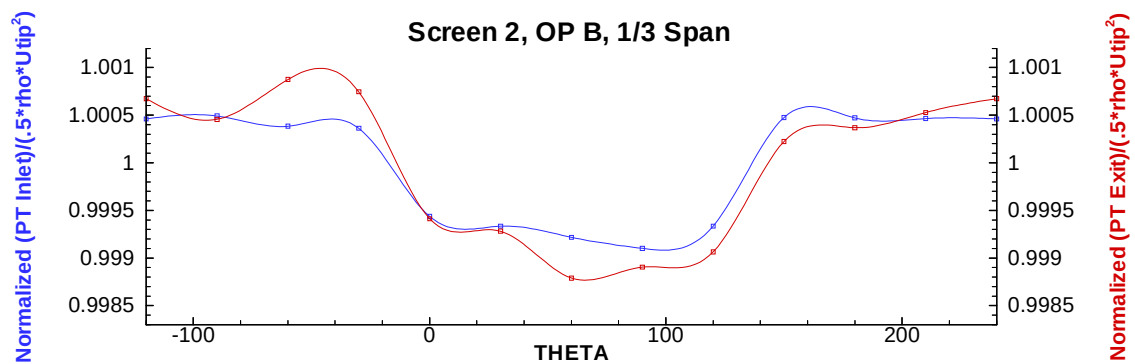


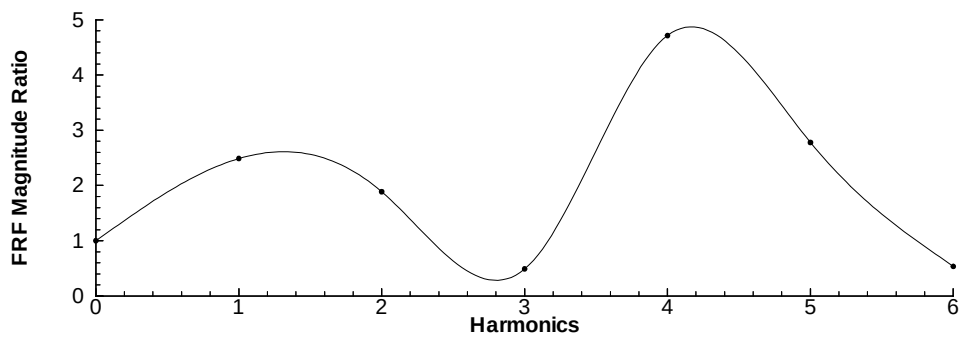
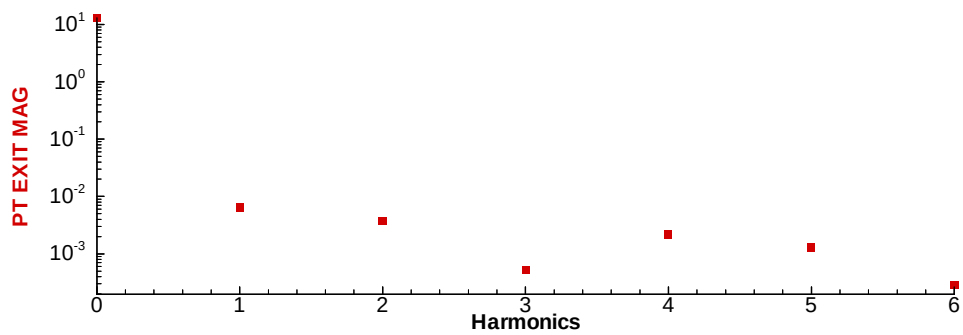
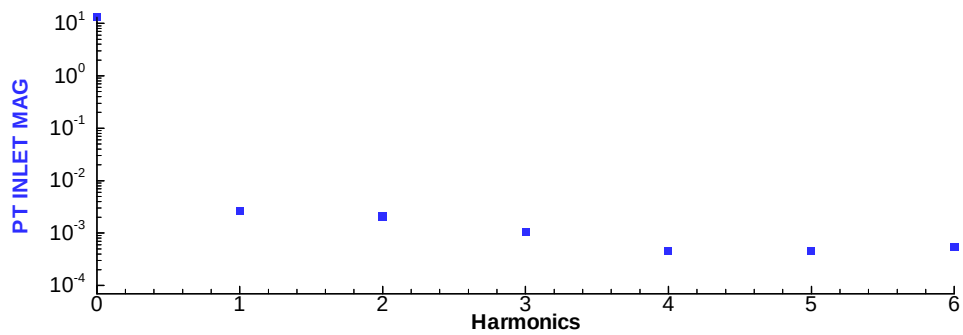
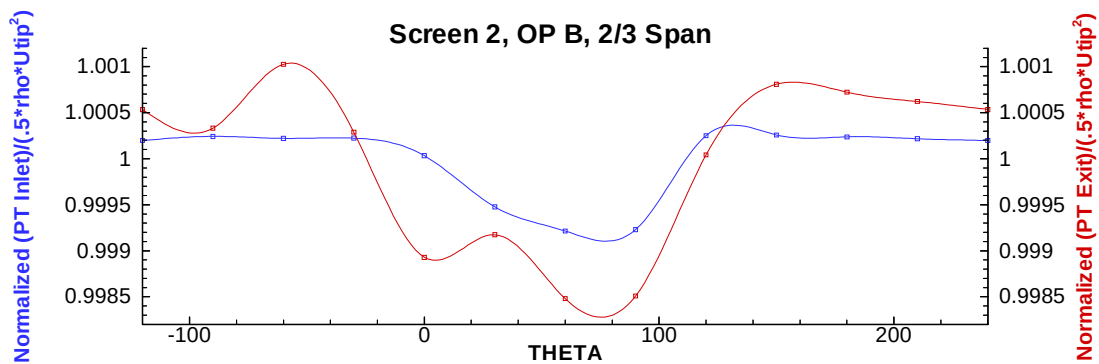


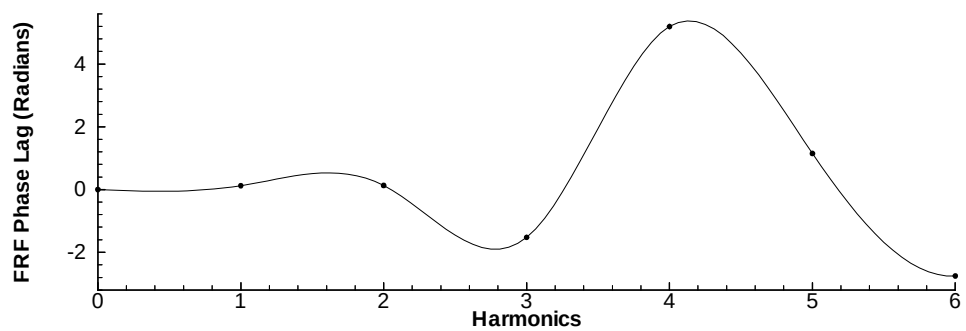
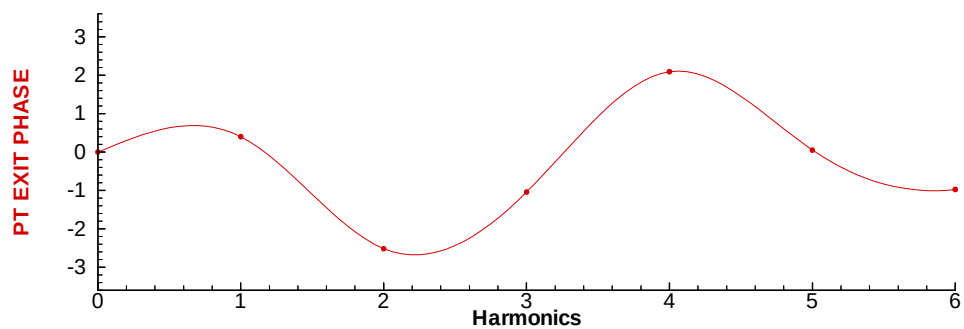
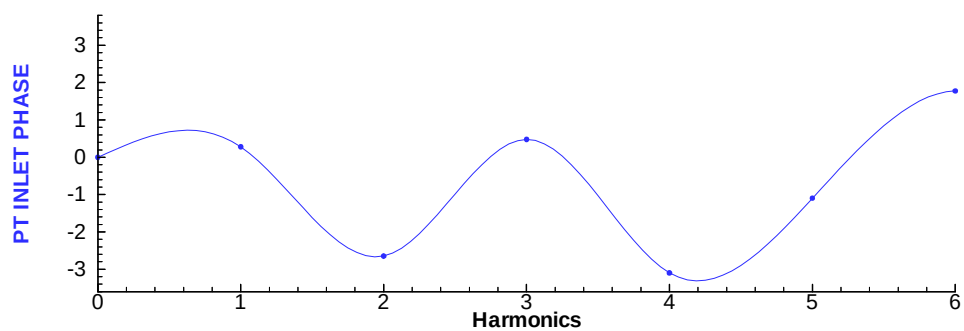
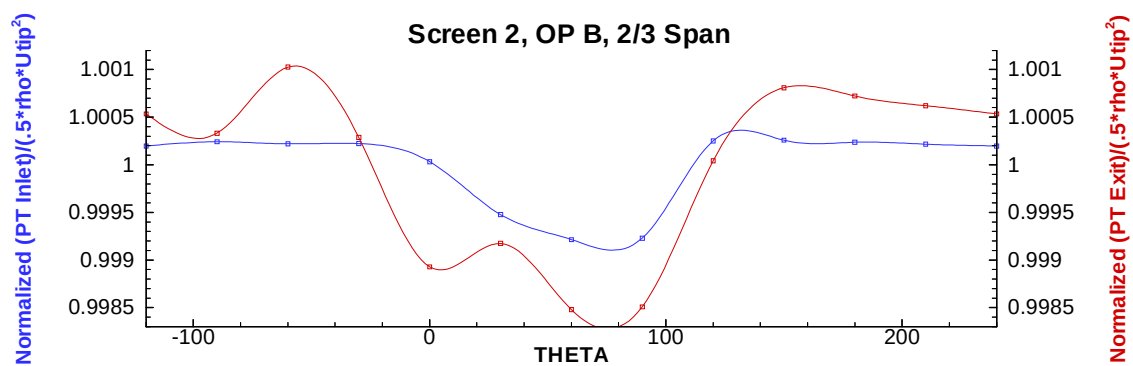


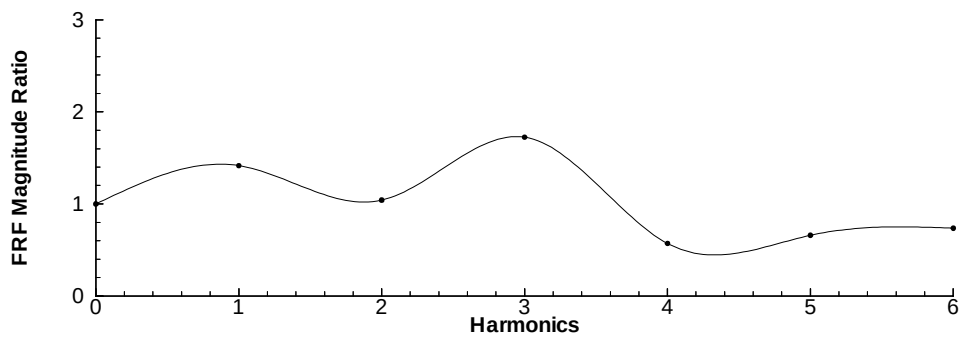
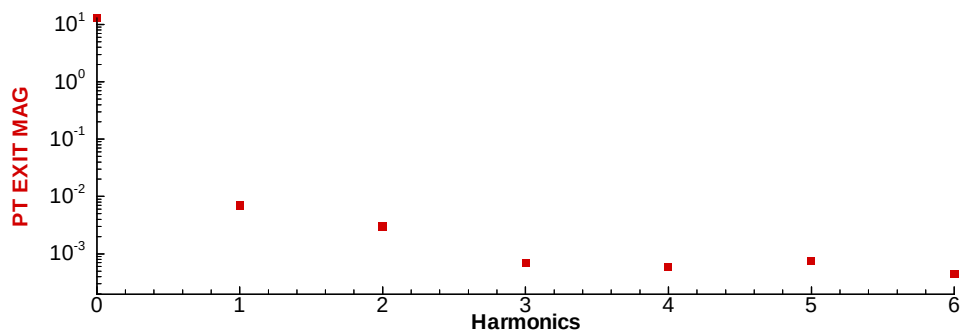
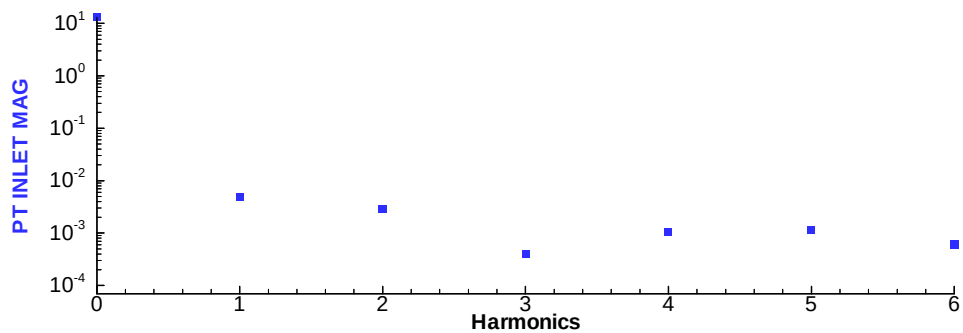
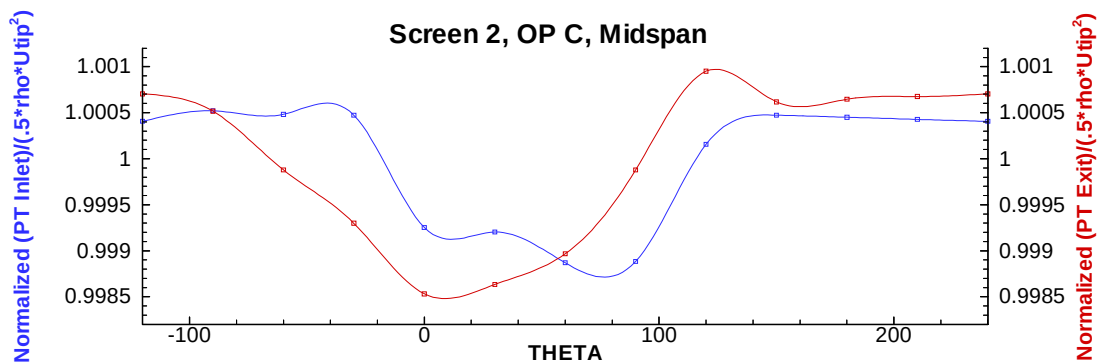


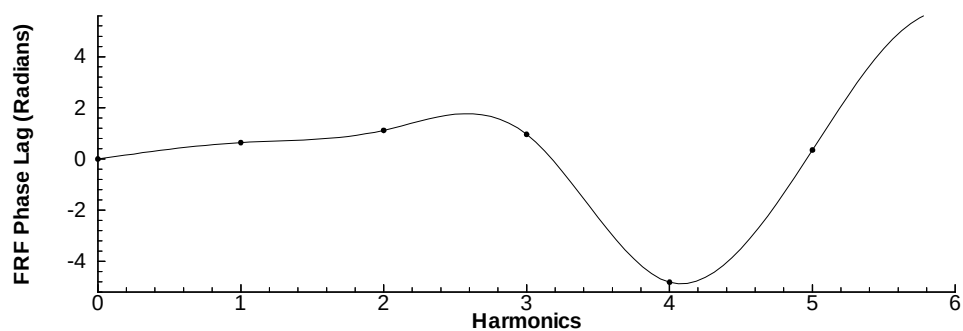
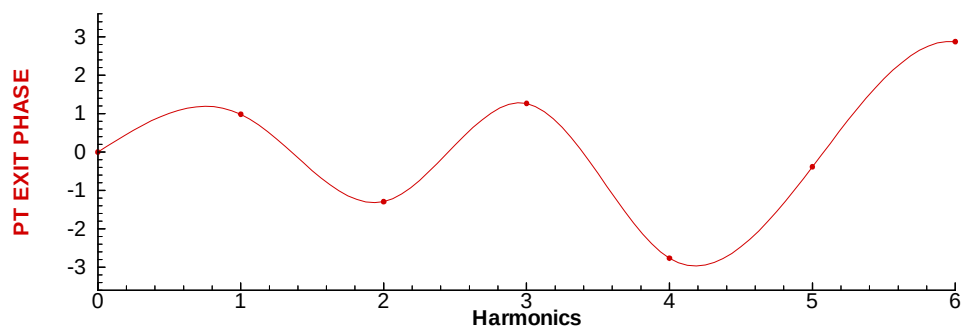
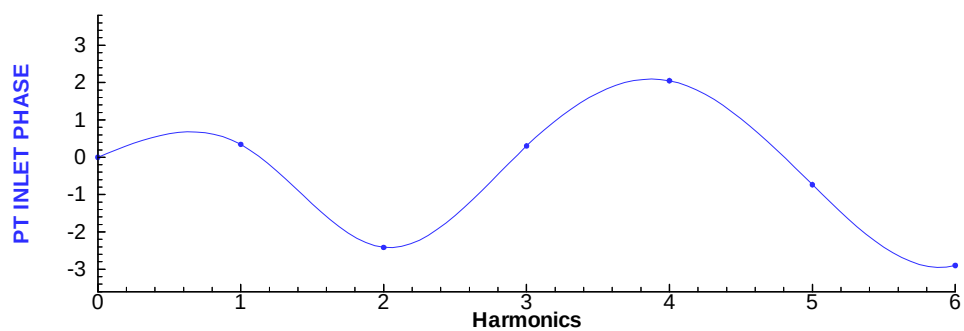
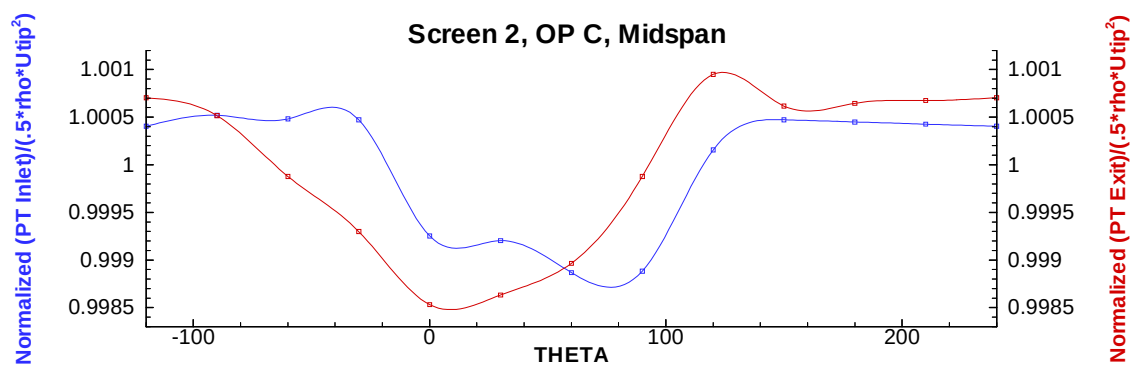


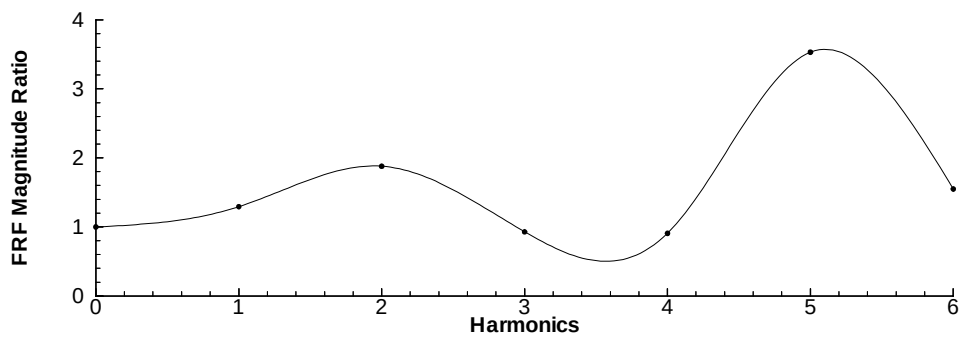
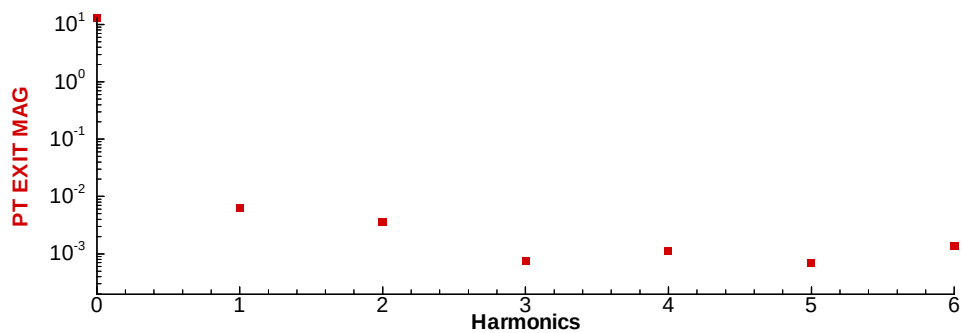
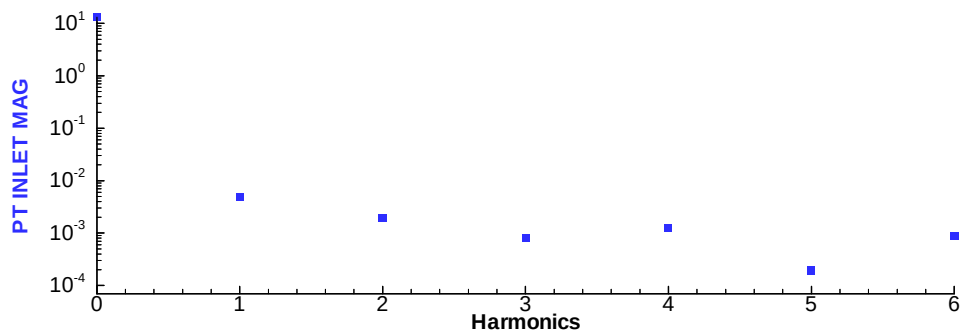
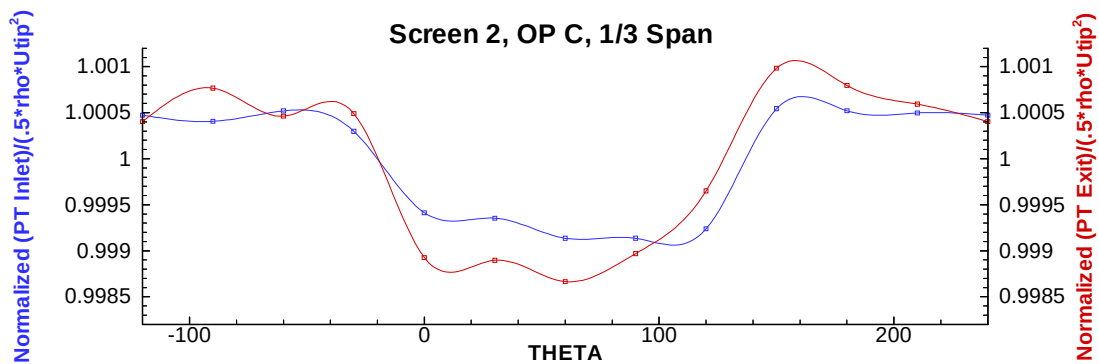


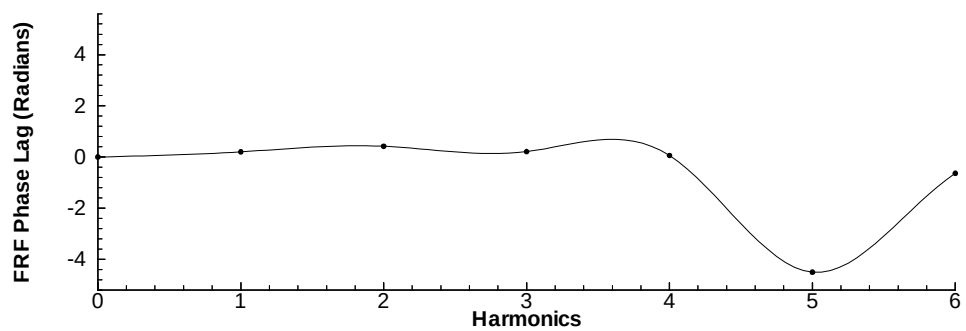
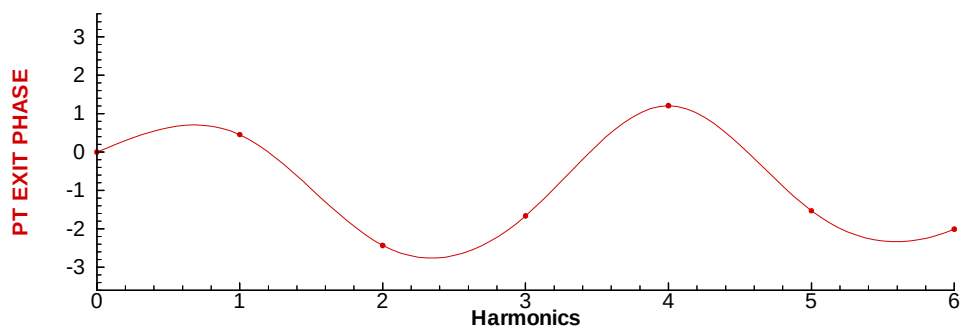
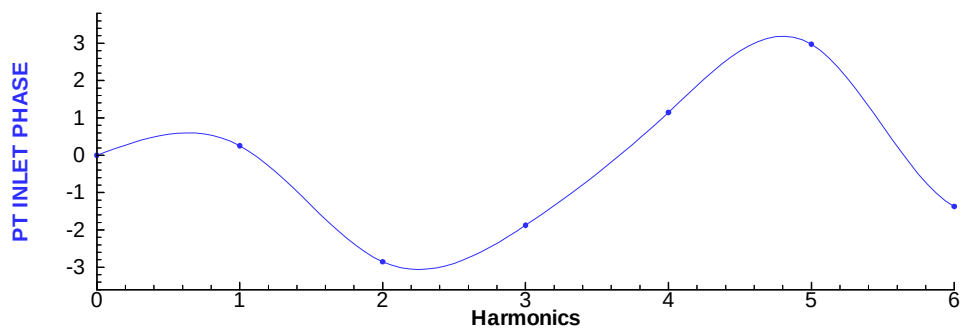
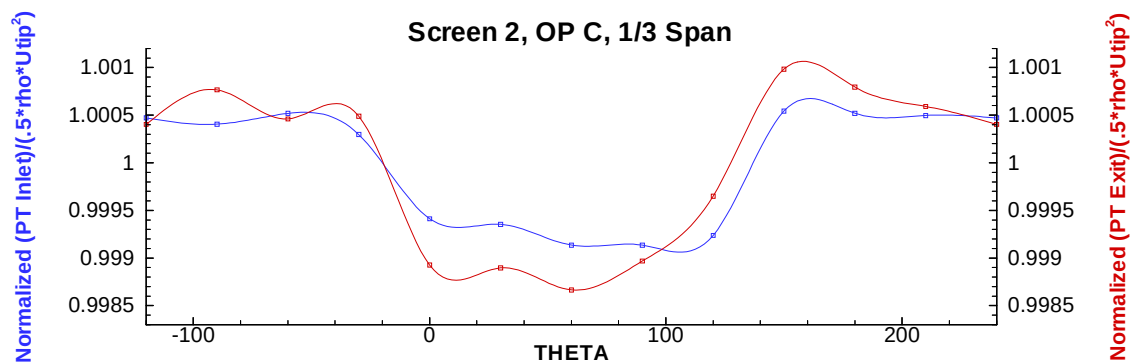


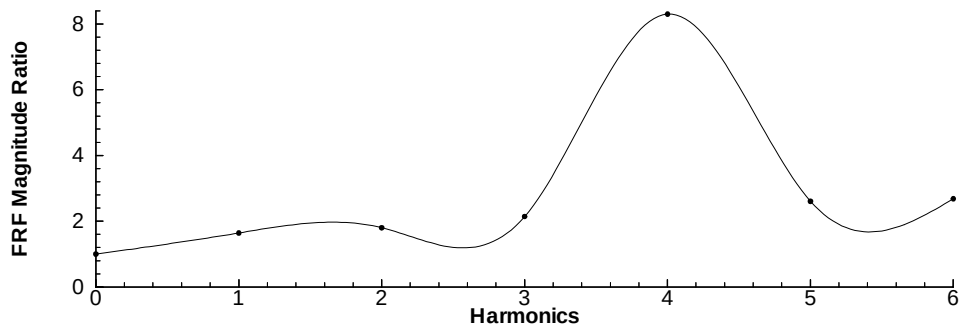
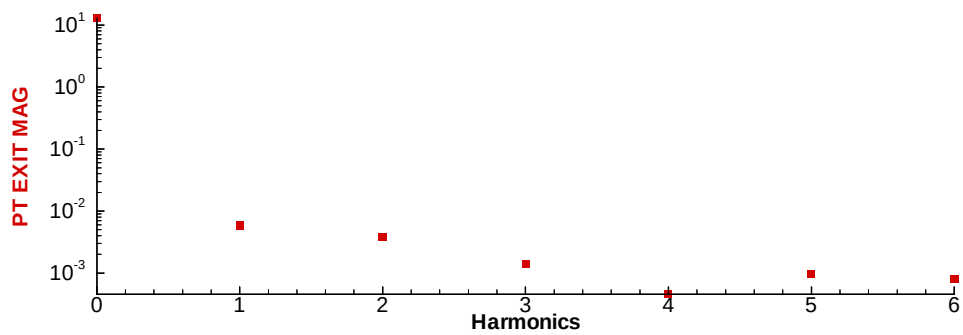
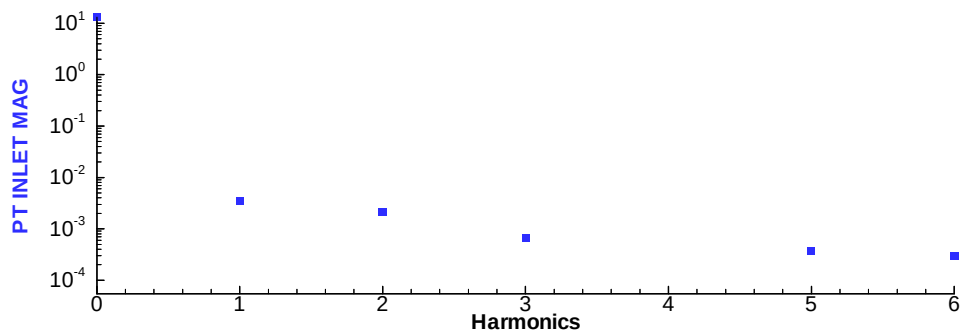
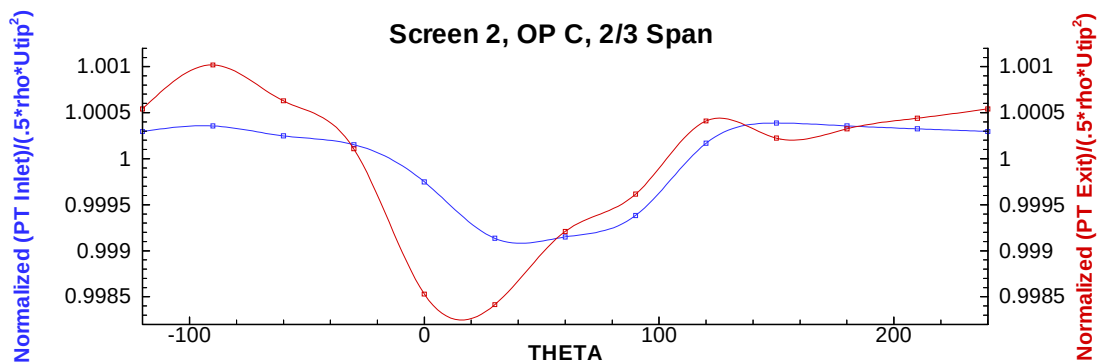


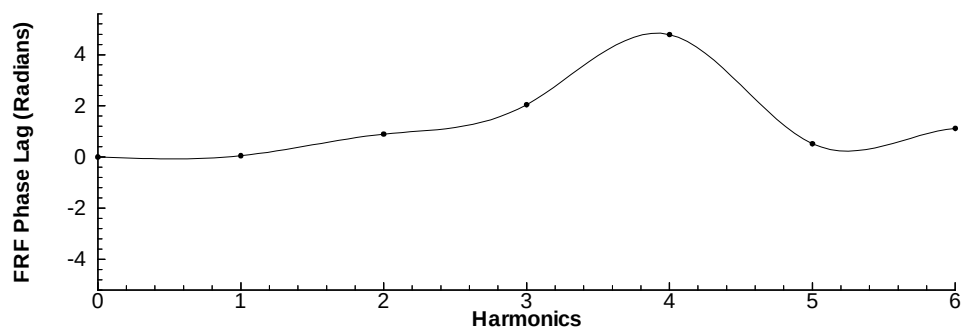
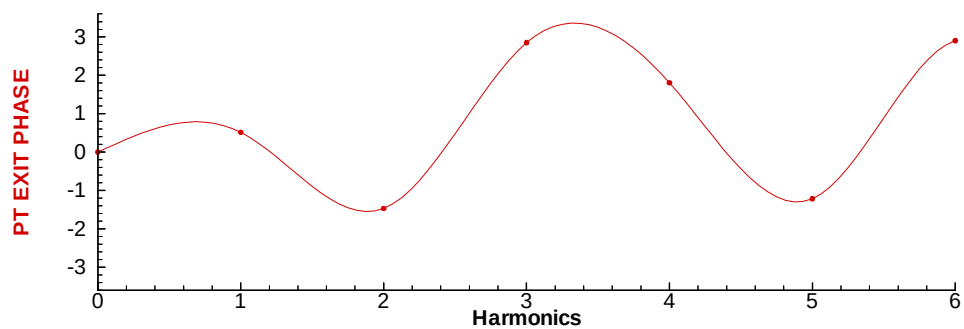
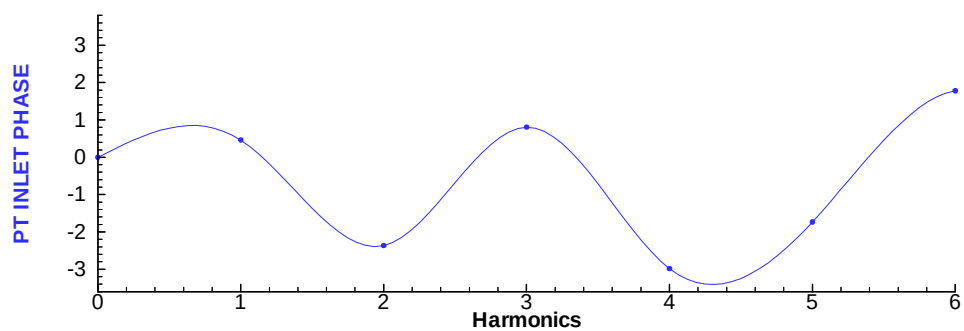
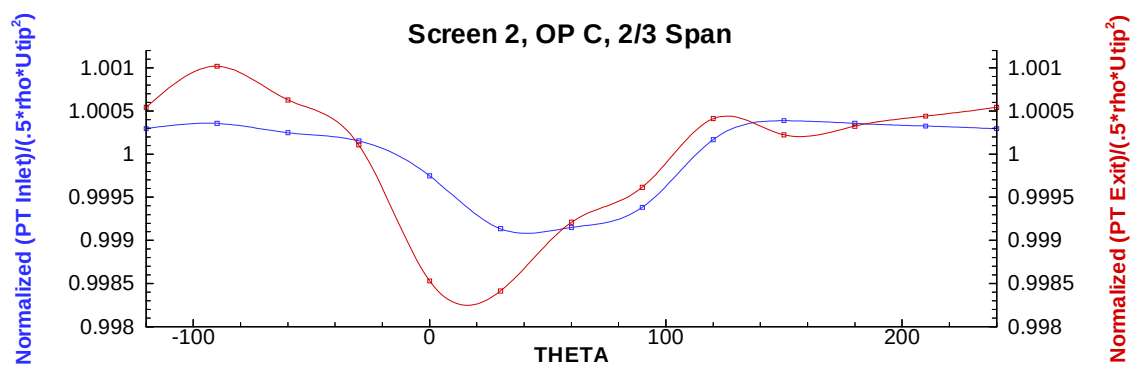


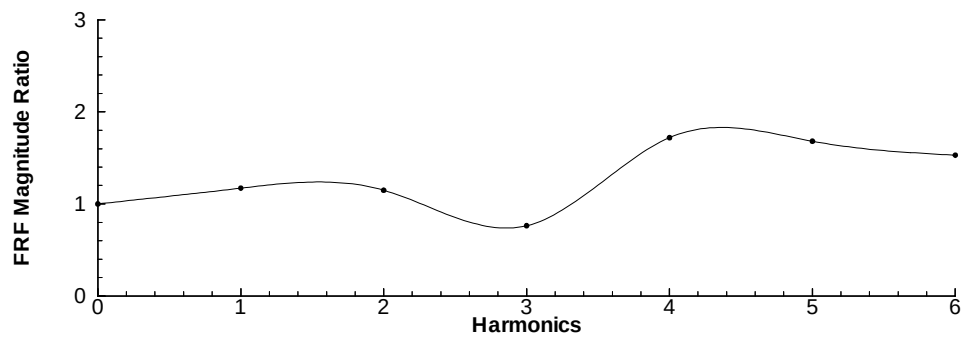
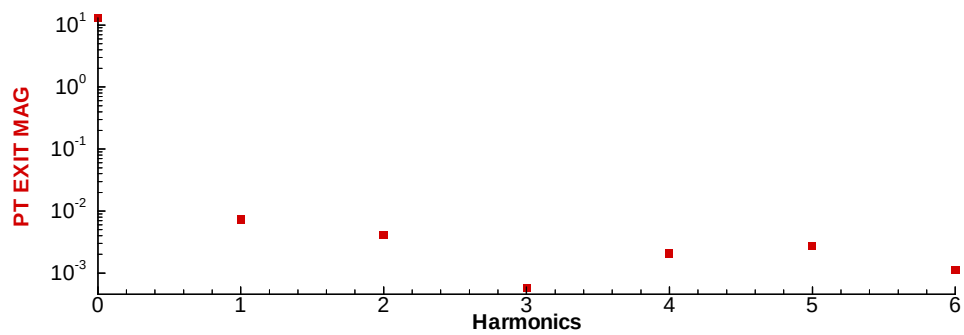
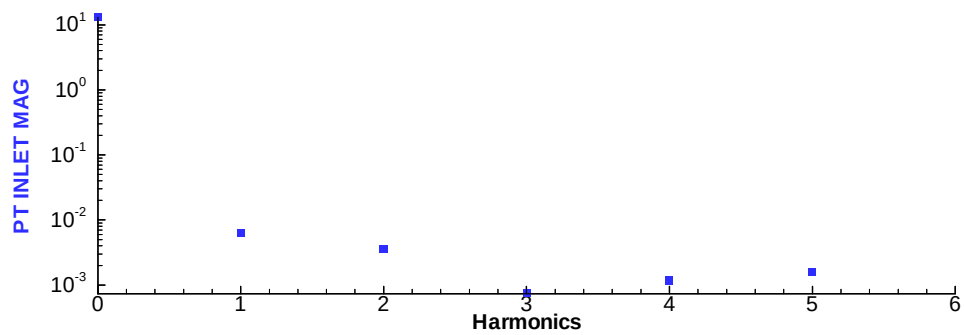
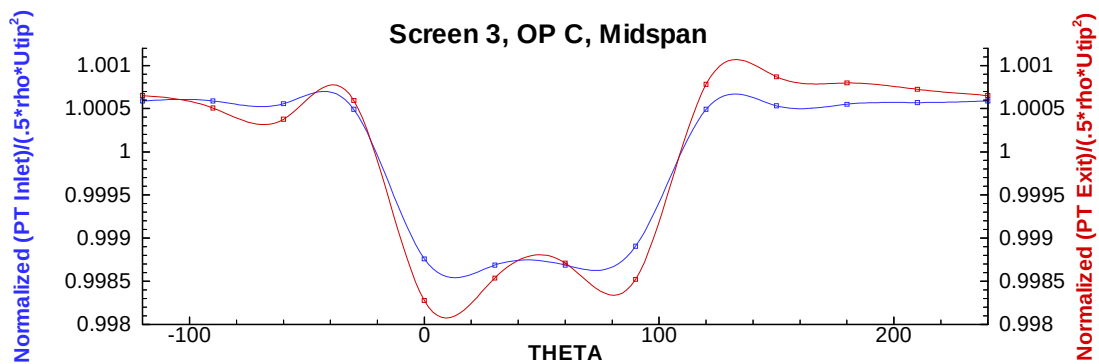


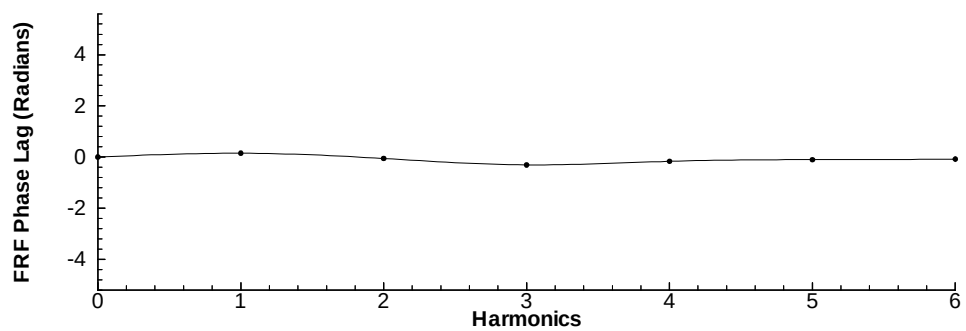
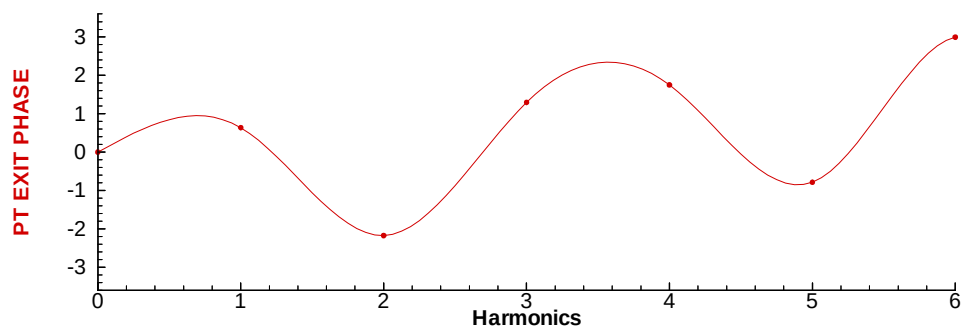
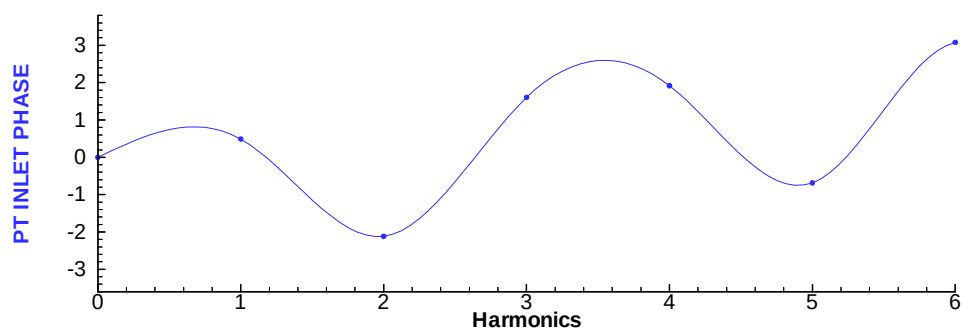
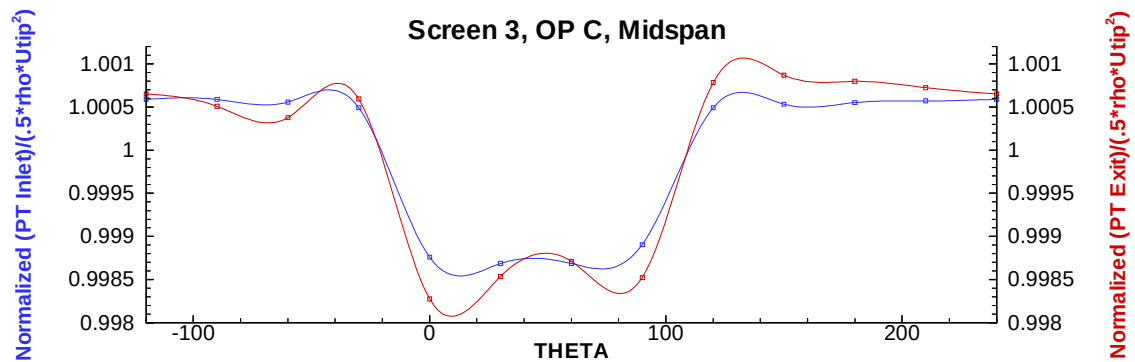


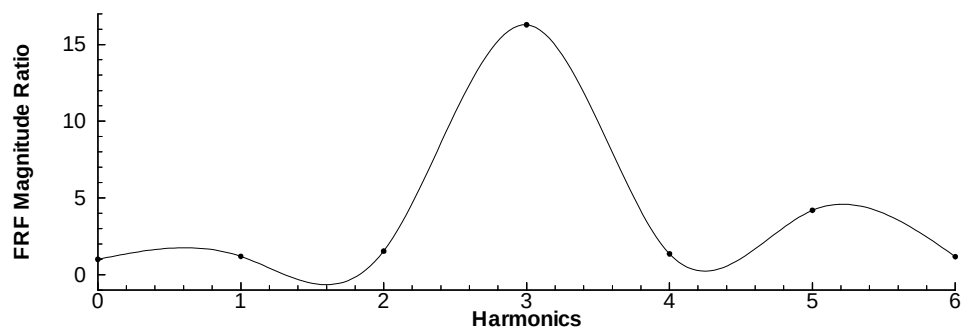
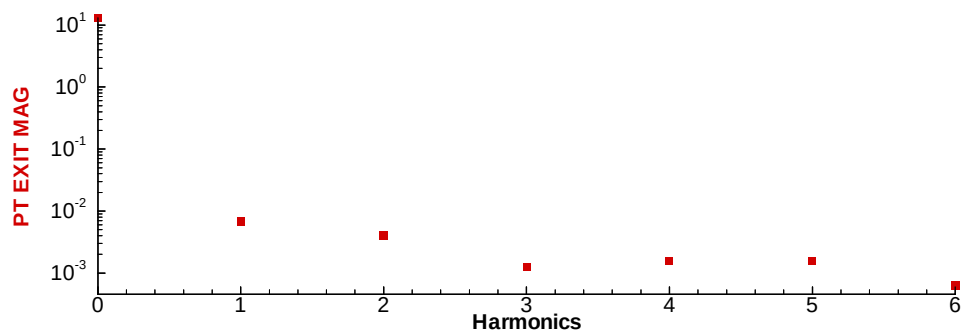
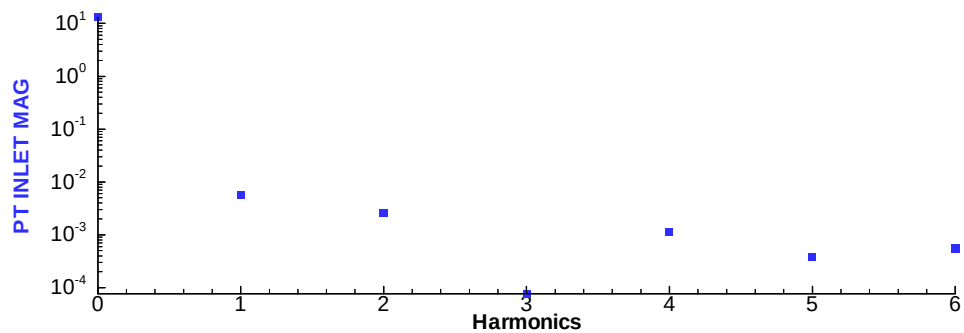
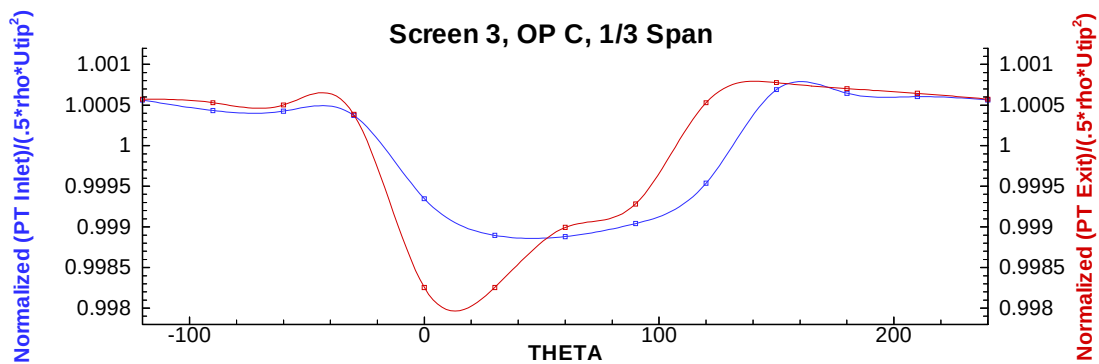


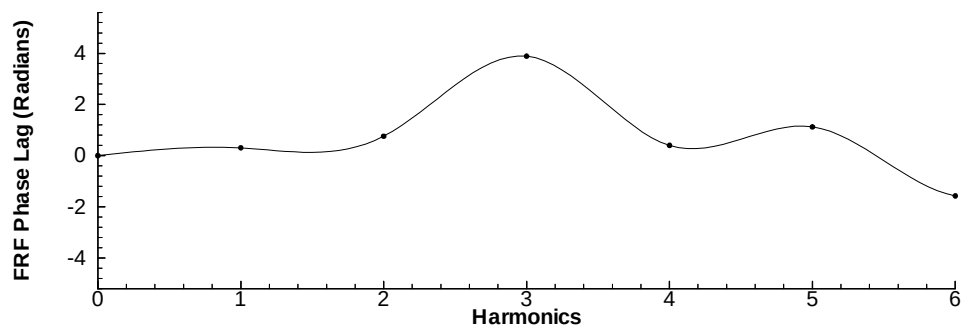
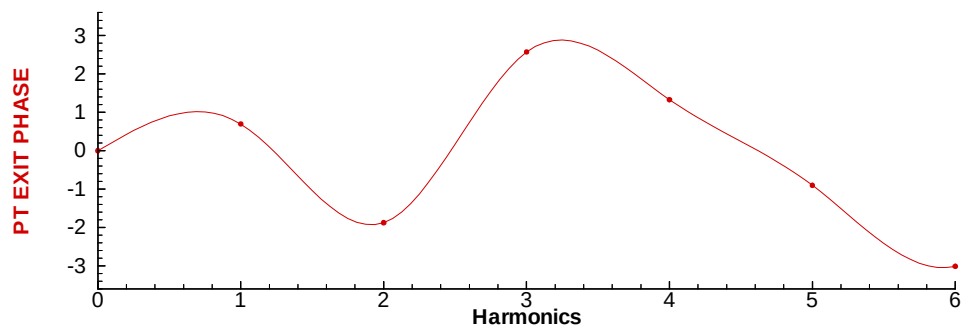
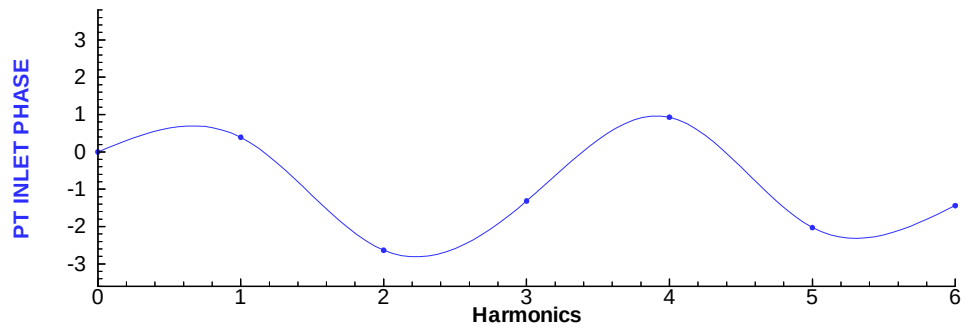
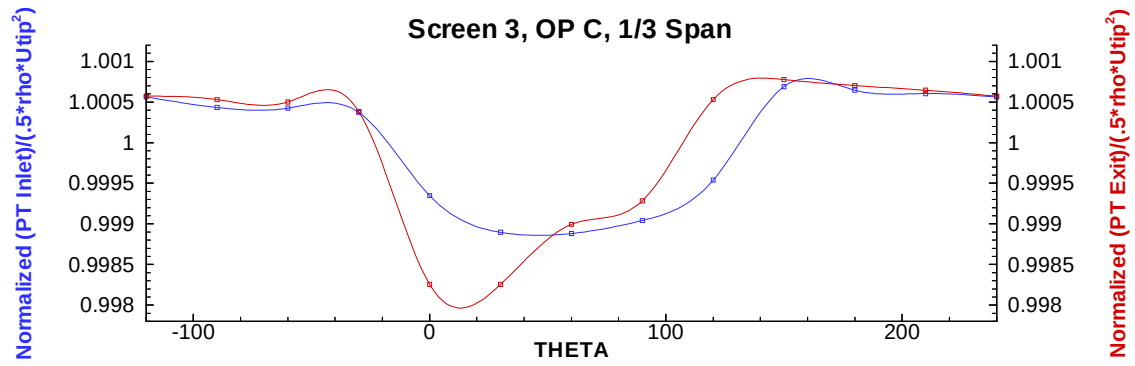


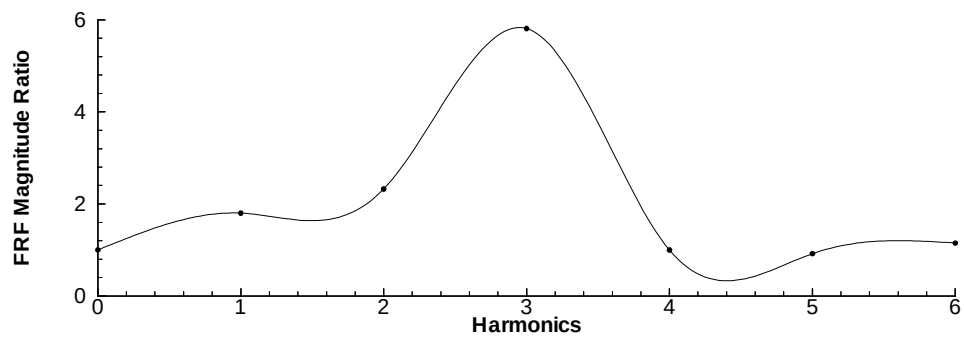
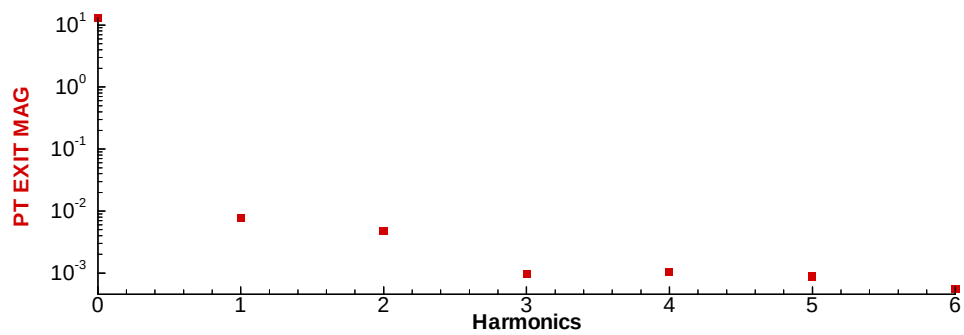
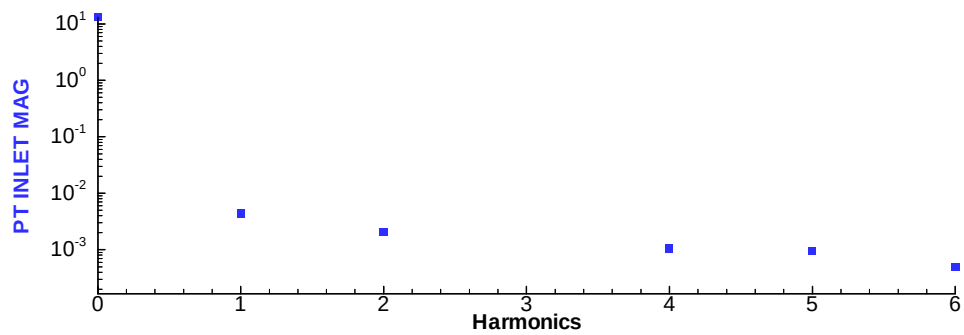
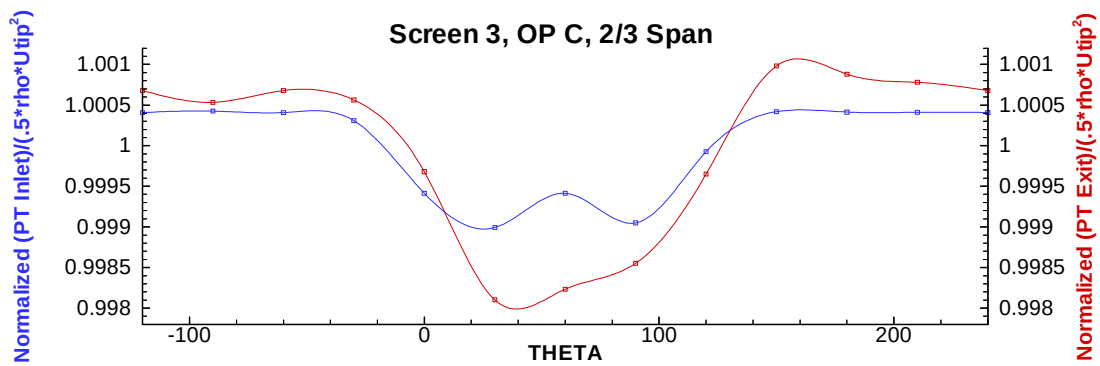


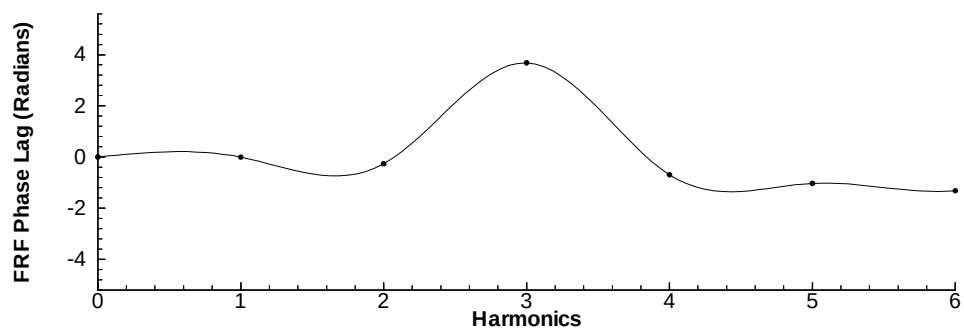
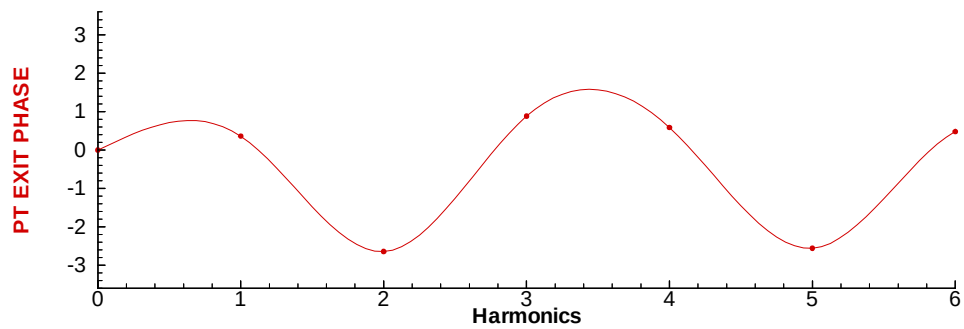
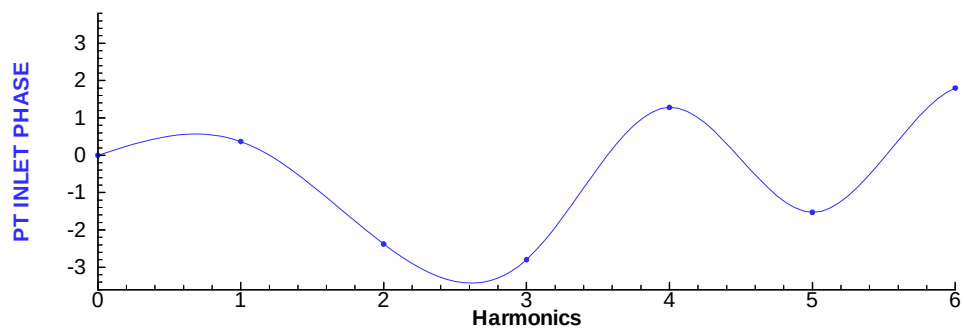
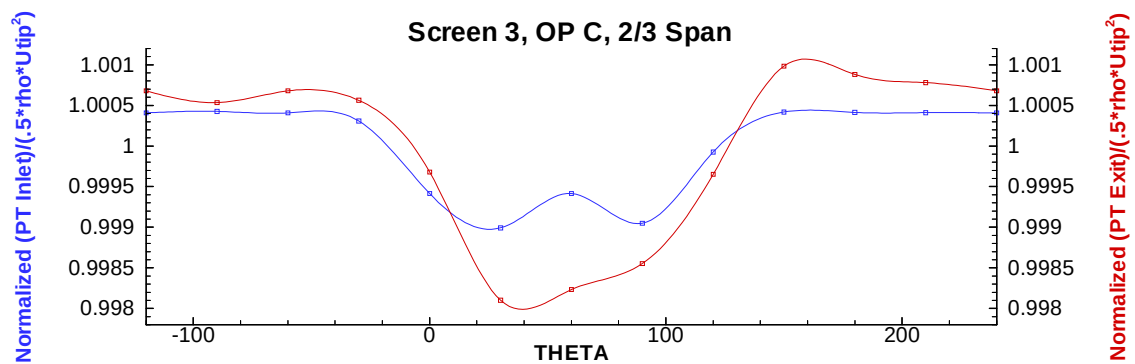




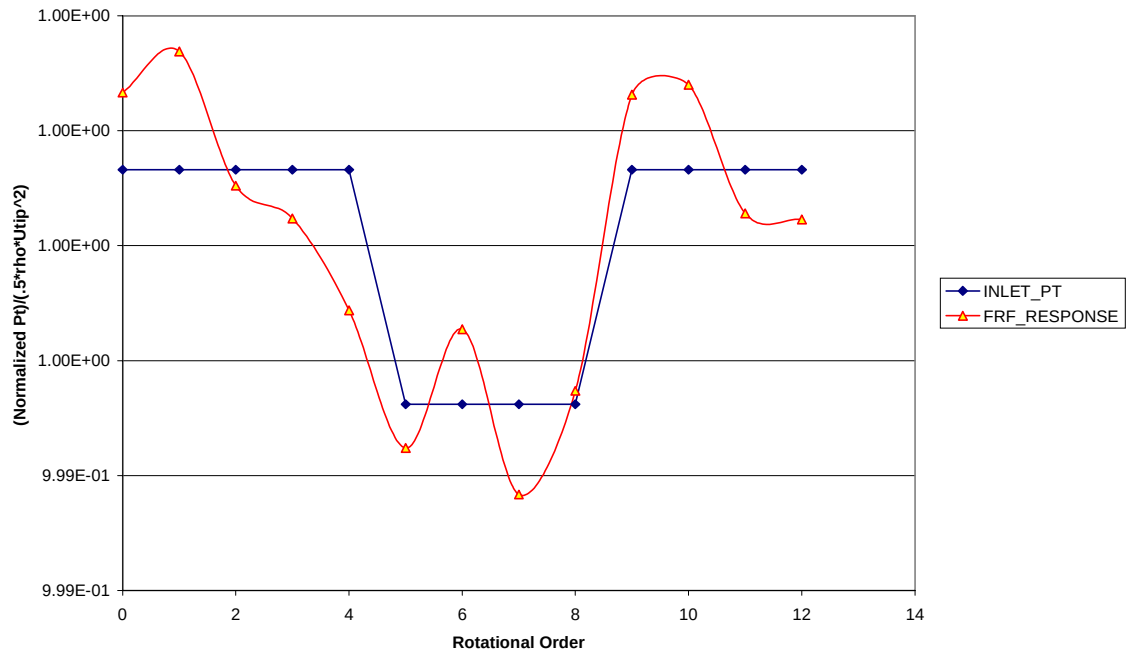




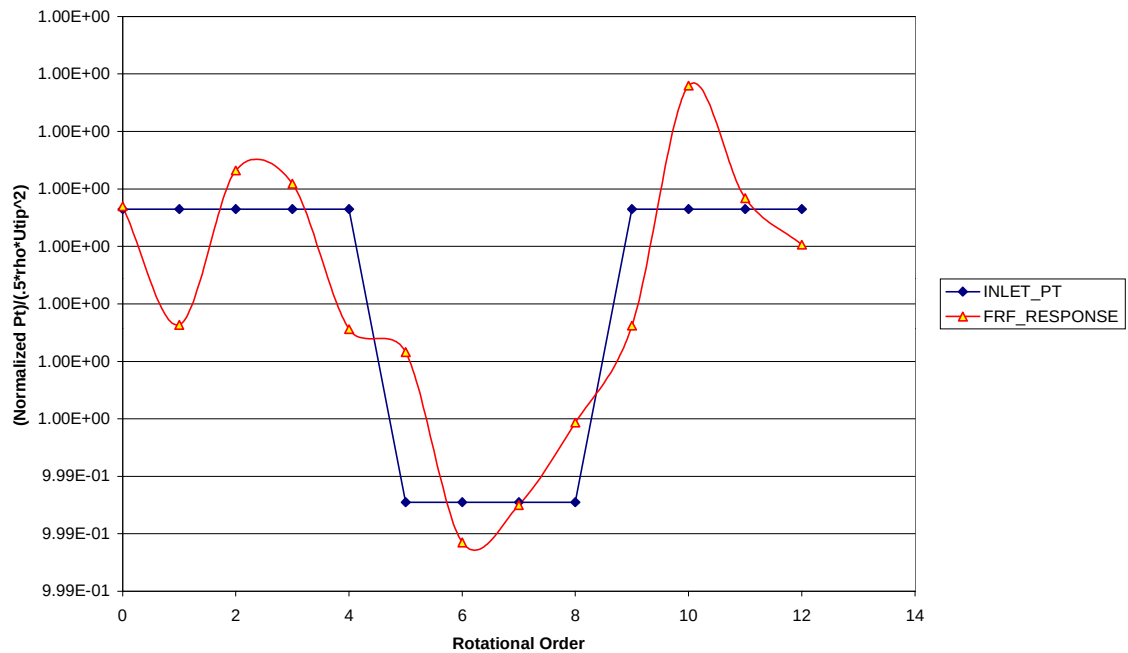




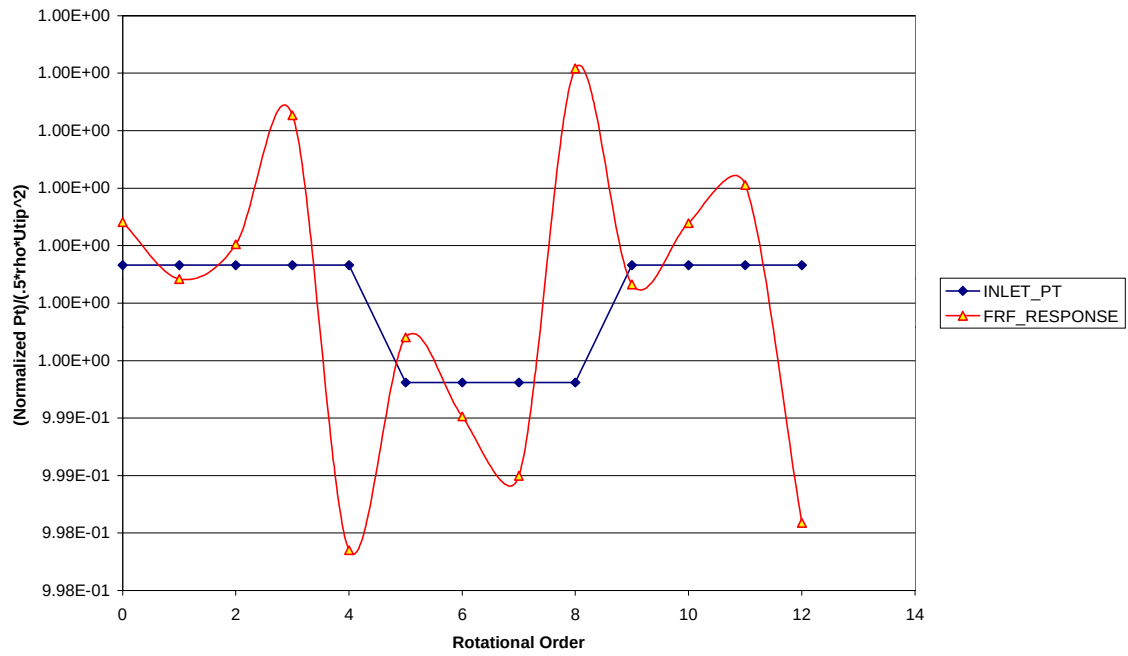
Response of Blade for level 1 screen at midspan and OP A based on FRF (Normalized Pt Exit/Normalized Pt Inlet)



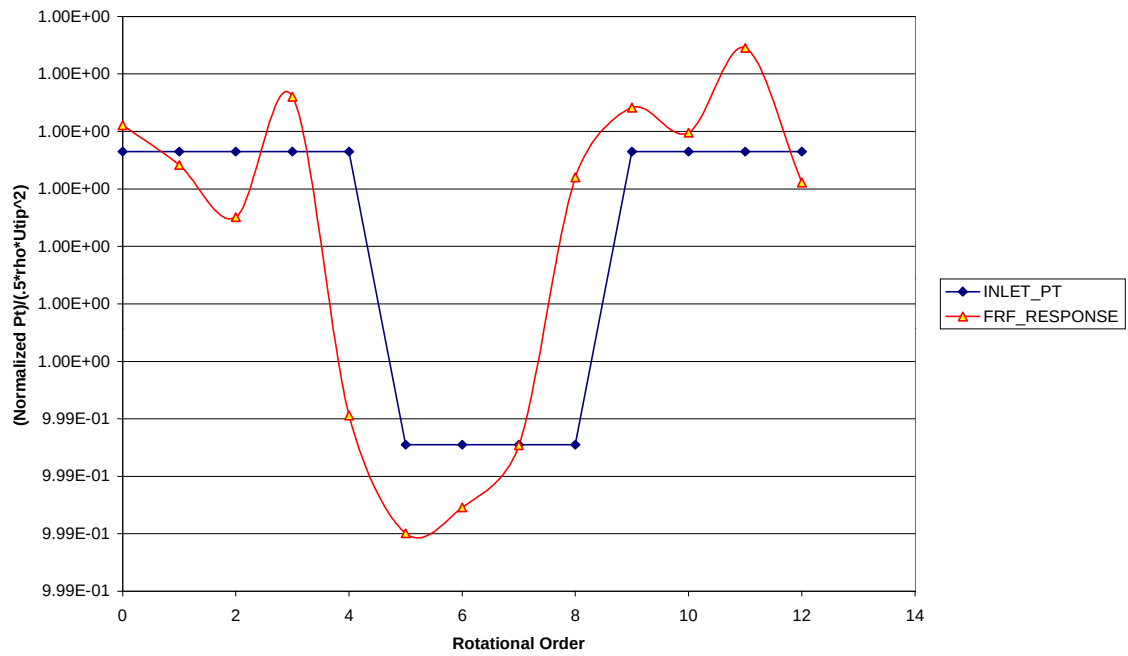
Response of Blade for level 1 screen at midspan and OP B based on FRF (Normalized Pt Exit/Normalized Pt Inlet)



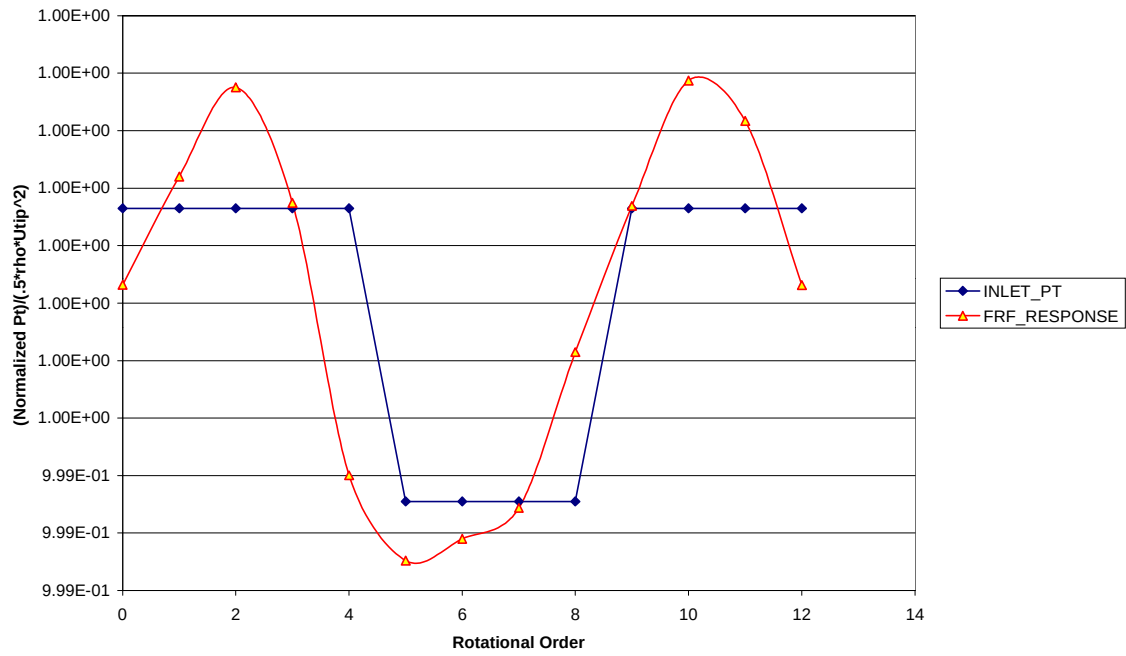
Response of Blade for level 1 screen at midspan and OP C based on FRF (Normalized Pt Exit/Normalized Pt Inlet)



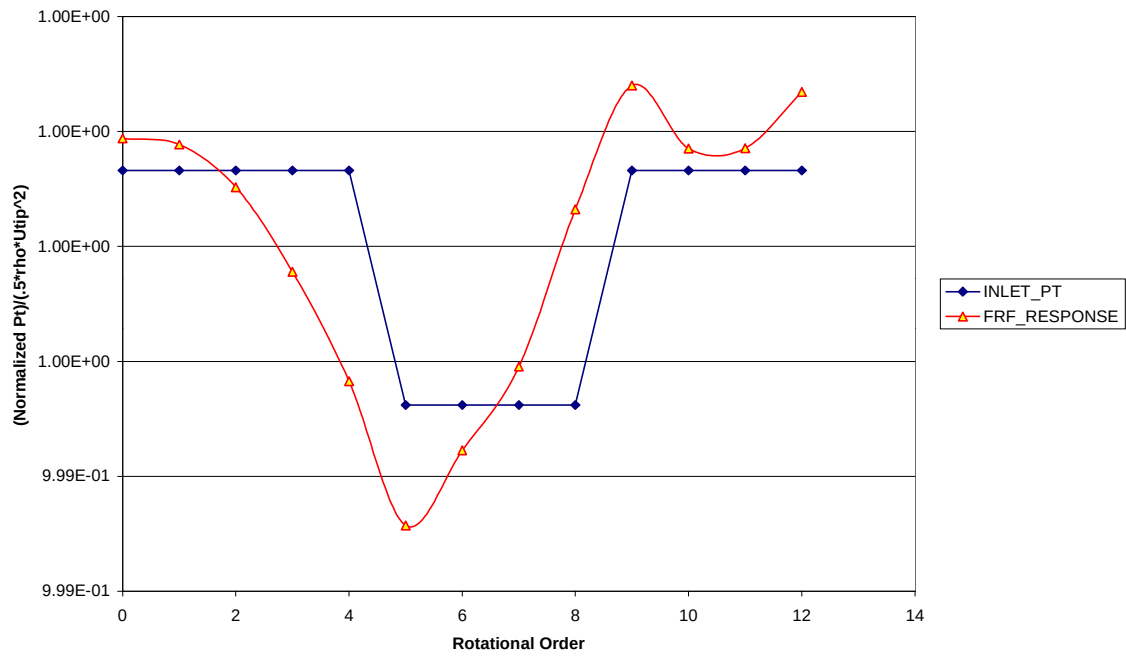
Response of Blade for level 2 screen at midspan and OP A based on FRF (Normalized Pt Exit/Normalized Pt Inlet)



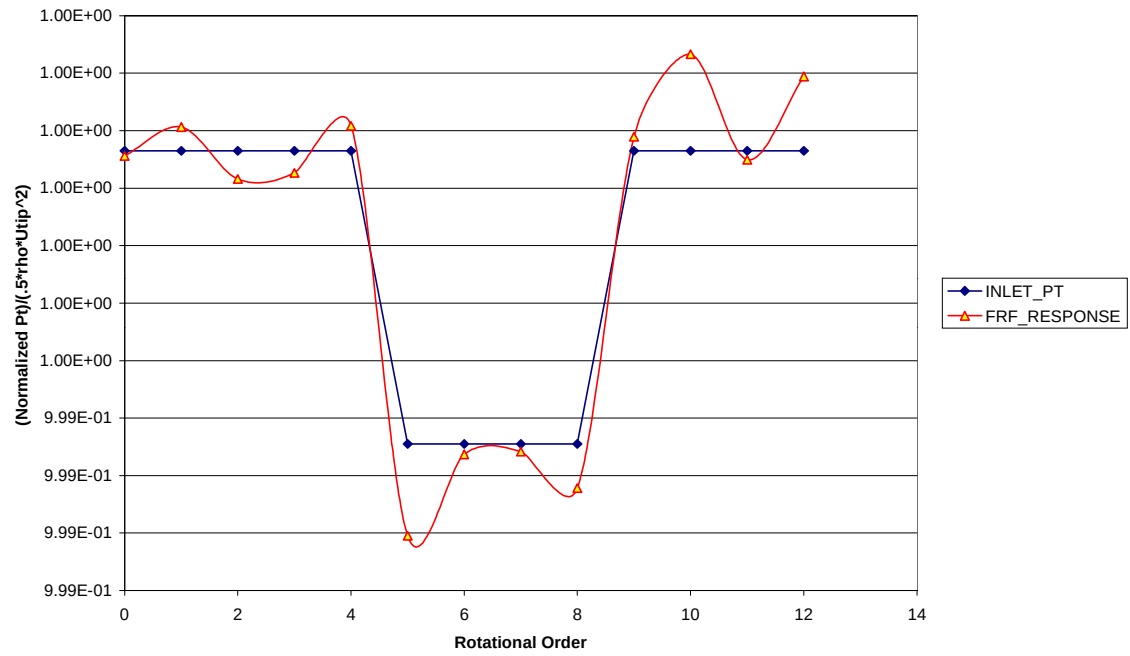
Response of Blade for level 2 screen at midspan and OP B based on FRF (Normalized Pt Exit/Normalized Pt Inlet)

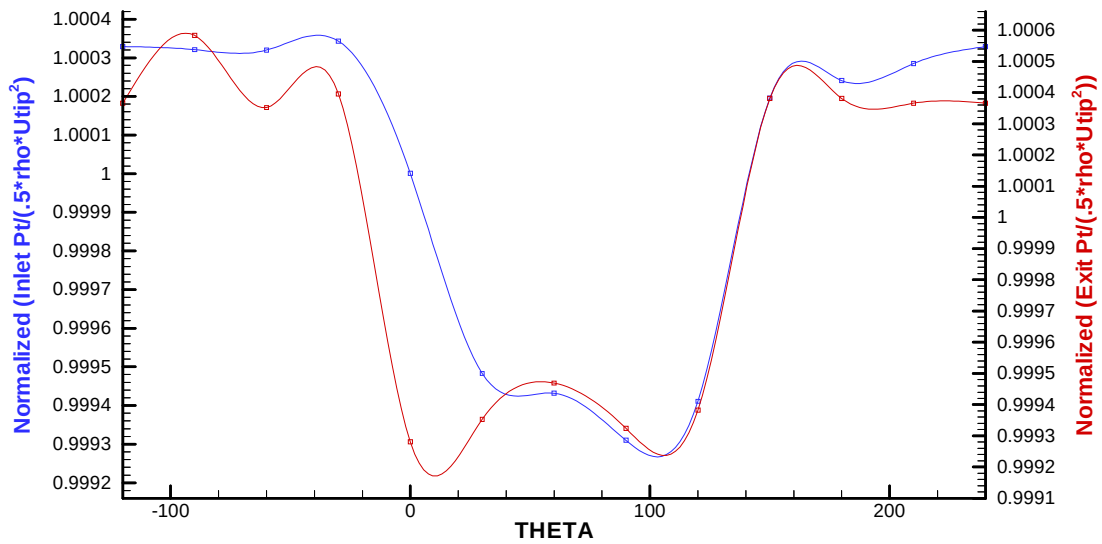
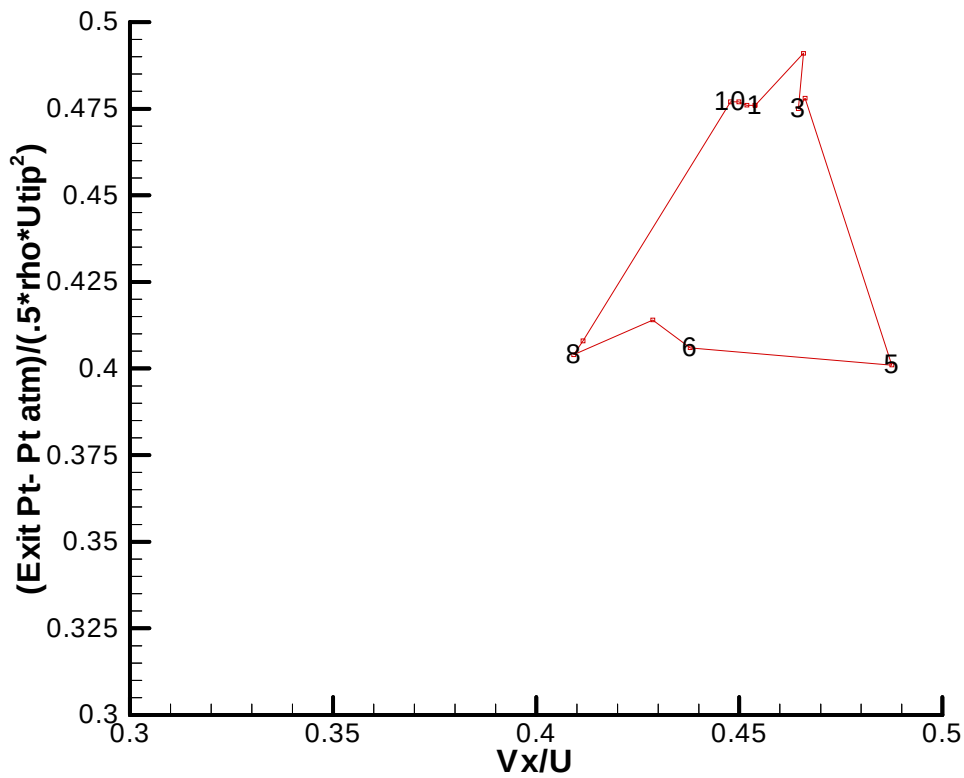


Response of Blade for level 2 screen at midspan and OP C based on FRF (Normalized Pt Exit/Normalized Pt Inlet)



Response of Blade for level 3 screen at midspan and OP C based on
FRF (Normalized Pt Exit/Normalized Pt Inlet)





13 Appendix

Computer Codes

13.1 Simple Dynamic 1-Dimensional Compressor Model

13.2 Frequency Response Code

13.3 Dynamic Compressor Blade Response Code

14 Vita

The author was born in Ft. Lee, Virginia on August 21, 1974. He attended elementary school and secondary school in Los Alamos, New Mexico. In 1993, he entered the Virginia Military Institute, which would change his life. In 1995, he enlisted in the United States Marine Corps Reserves, while he continued his studies at VMI. He completed his Bachelor of Science in Mechanical Engineering in 1997, at which point he began an internship at AlliedSignal Engines in Phoenix, AZ. In the fall of 1997, he began his research assistantship at Virginia Polytechnic Institute & State University. He is a member of ASME, Tau Beta Pi, and is a registered Engineer-in-Training in the State of Virginia. After graduation, he will begin work as an Engineer I at AlliedSignal Engines in the Performance & Operability Department.