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THESIS

**EXPERIMENTAL APPROACHES FOR OBTAINING
A TEMPORALLY AND SPATIALLY AVERAGED
REPRESENTATIVE STATIC PRESSURE FOR UNSTEADY
AND NON-UNIFORM COMBUSTOR CONDITIONS**

by

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**EXPERIMENTAL APPROACHES FOR OBTAINING A TEMPORALLY
AND SPATIALLY AVERAGED REPRESENTATIVE STATIC PRESSURE
FOR UNSTEADY AND NON-UNIFORM COMBUSTOR CONDITIONS**

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Submitted in partial fulfillment of the
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ABSTRACT

Detonation-based combustors are an attractive alternative to traditional deflagration-based combustors for airbreathing propulsion systems. Rotating Detonation Engines (RDEs) are a type of detonation-based combustion system that offers some attractive features for the warfighter, namely a reduced combustor volume and the potential for increased thermodynamic efficiency. However, characterizing RDE performance has proven difficult due to the transient nature of detonation conditions. To date, the RDE community has agreed that the equivalent available pressure (EAP) is the standard performance metric, but this value is difficult to implement experimentally. The effective nozzle entrance static pressure, typically measured with a static capillary tube average pressure (CTAP) probe, has shown promise as an adequate surrogate to the instrument-intensive EAP measurement; however, CTAP measurement challenges, such as static pressure azimuthal variability within the RDE, remain. An apparatus was designed and analyzed that could provide a defensible spatially and temporally average static pressure measurement representative of the average static pressure for a given plane within an operating RDE or any unsteady combustor. The design, called a pressure-averaging device (PAD), was characterized via ANSYS Fluent and resulted in a finalized design that was able to both spatially and temporally average static pressure across several different fluidic boundary conditions.

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LIST OF ACRONYMS AND ABBREVIATIONS

CTAP	Capillary Tube Average Pressure
EAP	Equivalent Available Pressure
ITP	Infinite Tube Pressure Transducer
NPS	Naval Postgraduate School
PAD	Pressure Averaging Device
PCB	Fast Piezo-Electric Style Transducer
RPCL	Rocket Propulsion Combustion Laboratory
RDE	Rotating Detonation Engine

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I. INTRODUCTION

A. OVERVIEW

For the past several decades, constant pressure combustion systems have been used in airbreathing aircraft and other tactical systems; however, this mode of combustion has begun to reach a plateau in terms of thermodynamic and volumetric efficiencies [1]. In order to continue to push the bounds of propulsion technologies, new methodologies and thermodynamic cycles must be investigated. One such methodology that could be utilized to increase the capabilities of current propulsion systems is a pressure gain combustion (PGC) system, such as a detonation-based combustor. While modern tactical systems, such as afterburning turbojets, ramjets, and the famous J58 turbo ramjet, all operate using a Brayton Cycle, which in an ideal sense, is a constant pressure combustion cycle [2]. Alternatively, detonation-based combustors operate on a pseudo-Humphrey/Atkinson cycle, and result in PGC [3]. These cycles are thermodynamically more efficient than the Brayton Cycle used by current propulsion systems. In the standard Brayton cycle, work is achieved by compressing air to high pressure which is then mixed with fuel and combusted at a nearly constant pressure where the gas can then expand out of the nozzle [1]. Because of the isobaric heat addition seen in this form of combustion, the maximum amount of work that these systems are capable of is limited. Meanwhile, the detonation cycle is nearly isochoric and both total pressure and total temperature can increase at nearly constant volume, which allows for more available work from the system [1][5]. Figure 1 illustrates the additional work available from the detonation cycle when compared to the typical Brayton cycle.

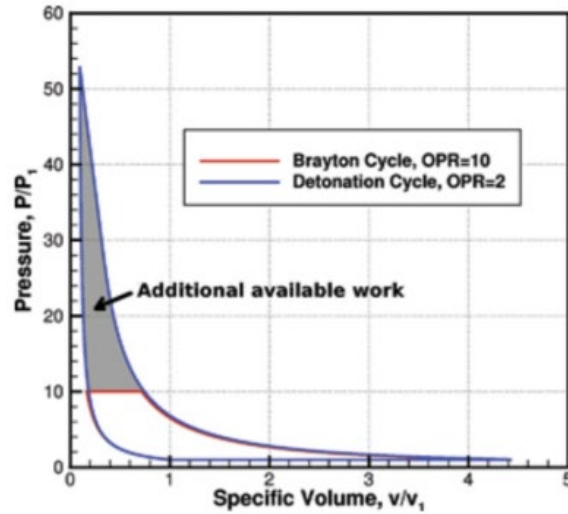


Figure 1. P-V Diagram of Detonation and Brayton Cycles. Source: [5].

Figure 1 shows that the detonation cycle can take advantage of the increase in pressure to increase the available work that the system is able to deliver. This is due to the detonation combustion creating a shockwave that, when coupled with the reaction zone, is propelled forwards by a portion of the heat released during combustion [2]. This results in a pressure rise across the shockwave as it travels through the system which increases the pressure of the working fluid; the reactants then combust over a nearly constant volume, and at the higher-pressure condition. Because the combustion process occurs at a higher pressure, more work can be generated for equal amounts of fuel combusted in the detonation cycle compared to that of the typical Brayton cycle [1].

One such system that takes advantage of the detonation cycle, and subsequently PGC, is the Rotating Detonation Engine (RDE). In RDEs, fuel and air are continuously injected into an annular chamber where they are rapidly mixed, and subsequently detonated [5]. In this way, only an initial detonation is required to sustain a continuous detonation wave that travels circumferentially around the annular combustion chamber so long that fuel and air are continuously injected into the system. This detonation wave travels at speeds equal to or greater than 1 km/s around the combustion chamber [4]. Because RDEs are continuously detonating systems, so long that an appropriate reaction mixture is continuously injected into the annular combustor, high power densities become possible

[1]. If more reactants are added into the combustor faster than a single detonation wave can process, RDEs can naturally bifurcate the detonation and create additional detonation waves to consume the fuel that is being provided. The continuous and self-adjusting nature of the detonation waves within RDEs give way to excellent power density, which in turn allows for higher volumetric efficiencies. This results in RDEs being able to offer more available work for additional thrust or power generation for the same amount of fuel while also being designed considerably smaller than conventional airbreathing combustion engines [5]. This results in reduced combustor volume and weight requirements and is of critical importance for tactical systems, which are typically volume constrained. The reduction of weight and volume costs for these systems could translate into additional volume for alternative systems, or more propellant thereby extending mission range capabilities.

B. MOTIVATION

During the past decade, significant strides have been made in terms of understanding the basic operation of RDEs. This includes quantifying the flow field through an RDE via computational fluid dynamics (CFD) [5–9], understanding what conditions are required to initiate and operate an RDE as well as their modes of operation [11], thermodynamic cycle analysis [3], and general performance metrics [13]–[16]. However, being that RDEs utilize pressure gain combustion, an extremely desirable performance metric is pressure within the combustion chamber to determine if global PGC has been properly achieved [15]. Historically speaking, characterizing the average pressure within an RDE has been extremely challenging. While near instantaneous static pressure can be directly measured with some caveats and limitations, it may not be representative of the true average pressure within an RDE. This is because with an unsteady system such as an RDE, the manner in which the pressure is measured matters [16]. For instance, measuring a mass-weighted average pressure versus area-weighted average pressure. Ideally, the mass-weighted static pressure must be recorded. Additionally, even if high-speed static pressure measurements could provide a mass-weighted averaged static pressure, performing these direct-mounted static pressure measurements are sub-optimal because it can result in the consumption of expensive pressure transducers due to high heat

flux to sensor heads [18]. Additionally, the sensors typically only provide reliable data for very short time durations due to thermal drift of the sensor.

As of now, there are only a couple of defensible techniques that are capable of measuring this pressure. One such technique is known as equivalent available pressure (EAP). The definition of EAP is “the flow stagnation pressure which is representative of a flows ability to do work or provide thrust and thus directly compare to equivalent steady-state systems” [19]. In this manner, the EAP technique is a means of backing out what total pressure conditions must exist within the combustion chamber in order to produce the forces measured by treating the RDE as a control volume. The effective control volume and its representative station identifications are illustrated in Figure 2 [16].

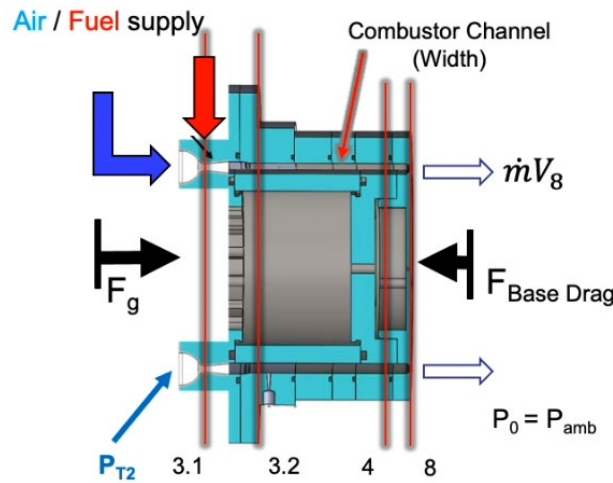


Figure 2. RDE Control Volume, Forces, and Station IDs. Source: [16].

EAP measurements require extensive bookkeeping of momentum, thrust and base drag, which are all required terms in order to back out the total pressure in the combustion chamber that must exist to produce the experimentally measured thrust and base drag described above. This technique inherently includes viscous and thermal losses and produces what is currently believed to be an accurate representation of the total pressure which exists within the detonation chamber [16]. With this, if the acquired EAP value is higher than that of the total pressure at station 2, which represents the total pressure driving

the injector, then global pressure gain combustion has been successfully achieved. EAP is also naturally conservative in its nature, meaning that this technique will not overpredict the performance of an RDE [14]. The problem with EAP lies in its required bookkeeping. Not only is a very sensitive thrust stand required to measure the gross thrust forces, but the RDE must also have a geometry that is compatible with the instrumentation required to measure the base drag via pressure transducers. This level of instrumentation and fixed geometry requirements are not possible throughout the entirety of the RDE community. So, while EAP seems to be a reliable PGC performance metric, a simpler, less instrument intensive approach is desirable [17].

There exists another method which avoids the complicated geometry and instrumentation requirements required for EAP for deriving the total pressure within the combustion chamber. This technique is known colloquially as the “NPS Method” and utilizes capillary tube average pressure (CTAP) measurements to arrive at a static-pressure-derived total pressure based on an assumed average Mach number from local area ratios [16]. Comparison of this method to the EAP technique have revealed a one-to-one experimental correlation between these methods across several RDE geometries, injectors, and flow conditions when measuring total pressure in the combustion chamber. This relationship can be seen in Figure 3.

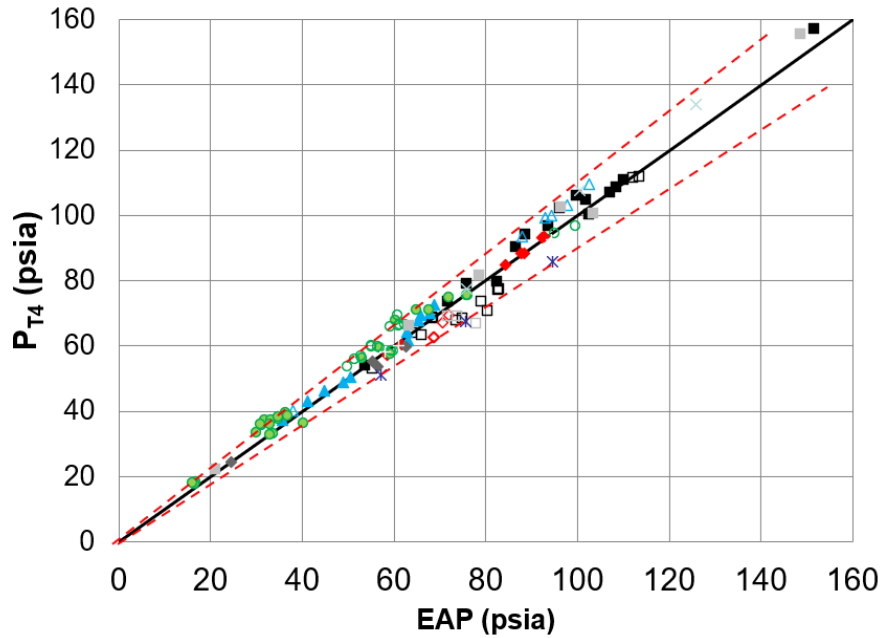


Figure 3. P_{T4} versus EAP. Source: [16].

This would seem to indicate that the CTAP approach is a defensible technique for total combustion measurements; however, a $\pm 10\%$ variation exists within these CTAP measurements when only a single CTAP measurement was utilized. As seen in Figure 4, when this phenomenon was investigated further by means of a CTAP “ring” with 9 azimuthally spaced and axially coplanar CTAPs, a 15% maximum variability from the mean aggregate static pressure was reported, which is considerably higher than the 5%-9% variability that is normally detected. It has been hypothesized that the variability is due to flow asymmetry within the RDE due to additive manufacturing flaws within the fuel injector as well as a failure mode of the injector that caused the fuel injection to not be azimuthally uniform across the RDE combustion chamber [15].

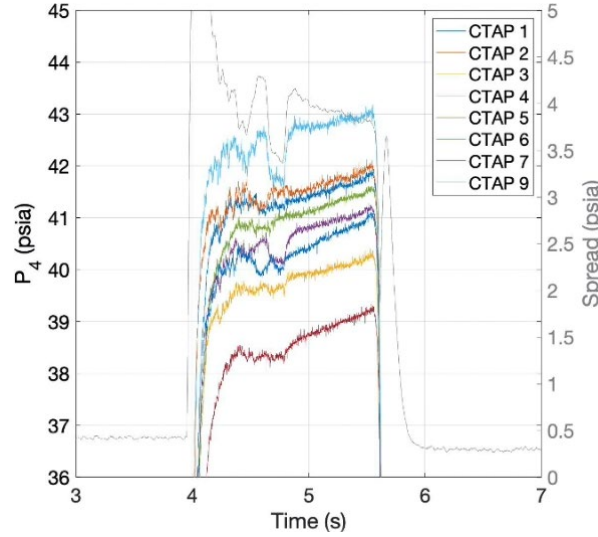


Figure 4. Azimuthal Static Pressure Variability as Reported by CTAP Measurements. Source: [16].

Removing the variability with a spatially- and temporally-averaged static pressure is of critical importance to understanding and implementing a sufficient “NPS Method” measurement to determine the total combustion chamber pressure and thereby characterize RDE performance. Notably, understanding and characterizing the reasons that azimuthal variability in static pressure measurements exists is also of extreme interest.

C. OBJECTIVES

The fundamental goal of this research is to design a method to accurately measure the static pressure within an RDE combustion chamber so that the total pressure can be accurately derived. Currently, only methods of temporally averaging the pressure within an RDE combustion chamber exist, and it is currently unclear what method of averaging delivers the most accurate results in terms of characterizing RDE operating performance. Additionally, it is uncertain how much instrumentation would be required to deliver accurate results for any given engine, as any azimuthal variability is likely to be engine and/or facility dependent. Furthermore, because RDEs are unsteady PGC devices, the mass-weighted average pressure is critical, not necessarily the area-weighted average pressure; therefore, it is unclear if multiple pressure transducer measurements will provide a truly accurate measurement of the pressure within an RDE. Even if this were a good

enough solution, large amounts of instrumentation can be prohibitively expensive and complex in many experimental setups. In this manner, a method of both spatially- and temporally-averaging this pressure is required. The specific objectives of this research are listed as follows:

- Design a device which can spatially-and-temporally-average the pressure within an RDE combustion chamber.
- Evaluate the designed device(s) to confirm their merit.
- Compare the mass-weighted average pressure with an area-weighted average pressure to assess the quality of each method.
- Test and evaluate the hypothesis of azimuthal static pressure variability due to a non-uniform fuel distribution

II. BACKGROUND

A. DETONATION PHYSICS

In order to discuss potential designs for a pressure averaging device (PAD), an understanding of detonation waves and their properties is critical. This section will outline the basic functionality of detonation waves and how they provide thrust in an RDE.

In its most basic form, a detonation wave is simply the coupling of a combustion wave and a shockwave [2]. As the initial shockwave passes by, the temperature and pressure of the working fluid rise via adiabatic compression. This initial shockwave causes most of the reactants in the fluid to dissociate and form a region known as the induction zone. Behind the induction zone, the reactants transform into combustion products and form a combustion front. This process is depicted in Figure 5.

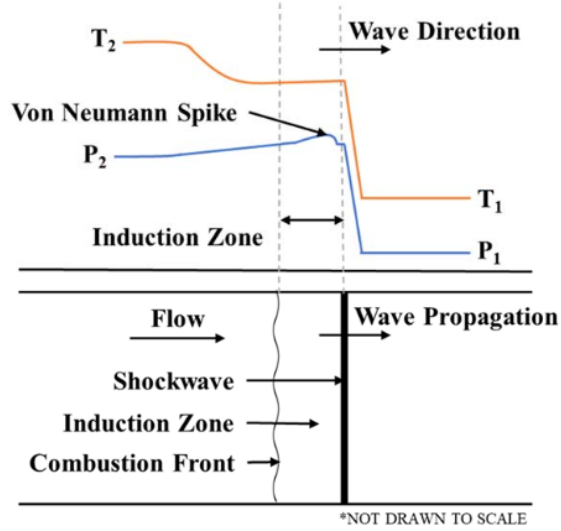


Figure 5. 1-D Detonation Process. Source: [2].

The shockwave results in a rapid increase in pressure and temperature of the reactants. Within the induction zone, the pressure and temperature are relatively constant, with the exception of the von Neumann Spike, which will be discussed later. Followed by

the induction zone, heat release begins to occur within the combustion front, resulting in an increase in temperature, as well as a slight decrease in pressure and density as the flow expands. After the initial shockwave, the speed of the flow is subsonic through the induction zone, and as the reactants recombine into combustion products within the combustion front, acoustic waves are generated and travel upstream towards the shock front which reinforces the leading shockwave. This reinforcement causes an additional increase in pressure within the induction zone near the leading shockwave. This point is known as the previously-mentioned von Neumann Spike, and is depicted in the figure at the point where pressure is maximum. The coupling between the combustion front and incident shockwave is the unique condition which results in detonation – a portion of the chemical energy within the reactants feeds back into the shock front to continue driving the wave forward to sustain detonation [2].

While this basic understanding of detonation waves and their properties is illustrative for understanding the idealized 1-D case and useful for deriving simple quantitative relationships, such as detonation wave speed propagation (Chapman-Jouget wavespeed), detonations are highly three-dimensional processes. The propagation of the detonation waves results in the creation of transverse waves. When the transverse waves intersect the shock front a third shockwave known as the Mach stem forms. The intersection of the three shocks is known as the triple point. At these locations, an ideal environment for a strong combustion front to form is present due to the locally very high values of pressure and temperature, and the bulk of the heat release occurs in these locations [10]. As mentioned previously, a portion of the heat release and energy is transferred to drive the normal shock forward and continue the detonation process. These points and a depiction of detonation wave propagation can be seen in Figure 6.

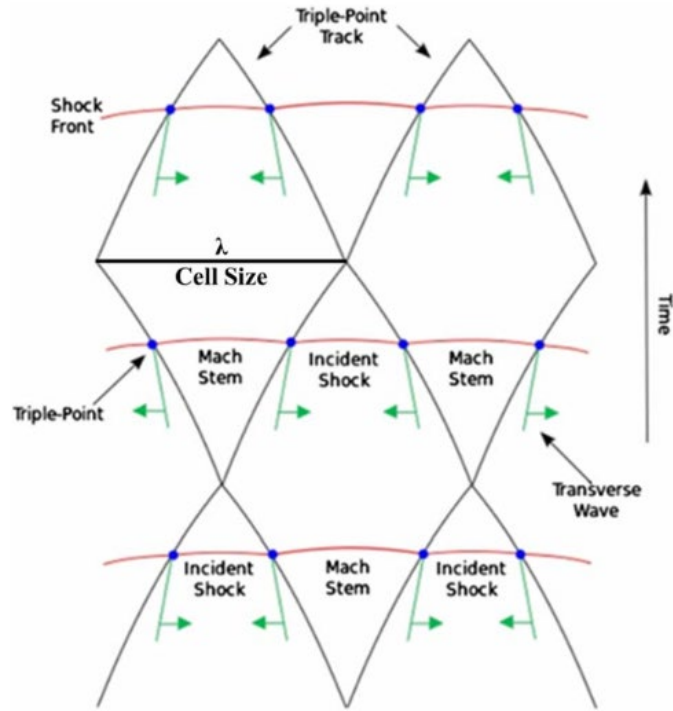


Figure 6. Detonation Wave Propagation and Cell Interactions. Source: [10].

The diamond shaped cells seen in Figure 5 is a time-history of the local intersection of the triple points and represent the detonation cell. The detonation cell size is given by the symbol λ . The cell size is determined by a myriad of factors including the equivalence ratio and type of fuel and is indicative of how “detonable” a reaction mixture is.

In terms of detonability, a smaller cell size is indicative of a larger amount of triple points and therefore more conducive to producing stronger detonation waves and thereby more individual detonation fronts. With a larger cell size, detonation is more difficult to achieve. Figure 7 illustrates how cell size is affected by the fuel type as well as its equivalence ratio. The same general trend can be seen for all fuel species outlined in Figure 7, being that the smallest cell size and therefore the most detonable conditions occur at or near an equivalence ratio of one, followed by a steady increase in cell size as equivalence ratio increases. This trend demonstrates a heavy reliance on the equivalence ratio as a major factor in the determination of how an RDE system may perform. An RDE operating with a small cell size and consequently more detonable conditions will exhibit

higher overall operating efficiencies as well as better performance metrics than an RDE operating with larger cell sizes and consequently less detonable conditions.

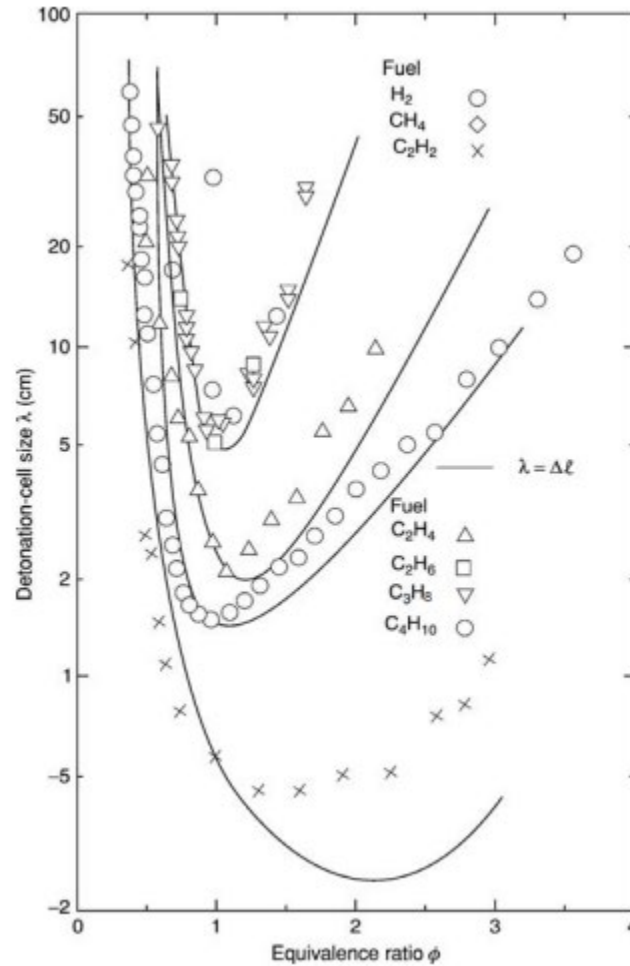


Figure 7. Cell Size in Terms of Fuel Species and Equivalence Ratio. Source: [2].

In situations where the fuel and the oxidizer are pre-mixed, the equivalence ratio of that mixture serves as a good predictor of the global cell size. This is the case because the global equivalence ratio should be approximately uniform at all azimuthal locations of the RDE. However, when RDEs that mix fuel and oxidizer as they enter the annular combustion chamber and the two species are not pre-mixed, there can be local disturbances in equivalence ratio.

B. LOCAL NON-UNIFORMITIES

1. Non-Uniformities by Means of Equivalence Ratio Perturbations

Local non-uniformities in equivalence ratio result in varying cell sizes at different azimuthal locations in an RDE combustion chamber. These non-uniformities can result in varying detonation performance at their respective azimuthal locations. In some instances, locally perturbed equivalence ratios can result in non-reacting regions of the RDE which would drastically affect RDE performance both globally and locally at the perturbed regions [8]. These non-reactive regions have been identified in previous simulations of RDE operation and an example of such can be seen in Figure 8 displays the mass fraction of oxygen for two simulation cases on either side of the detonation front. Both cases show pockets of unreacted oxygen persistent in the flow post-detonation.

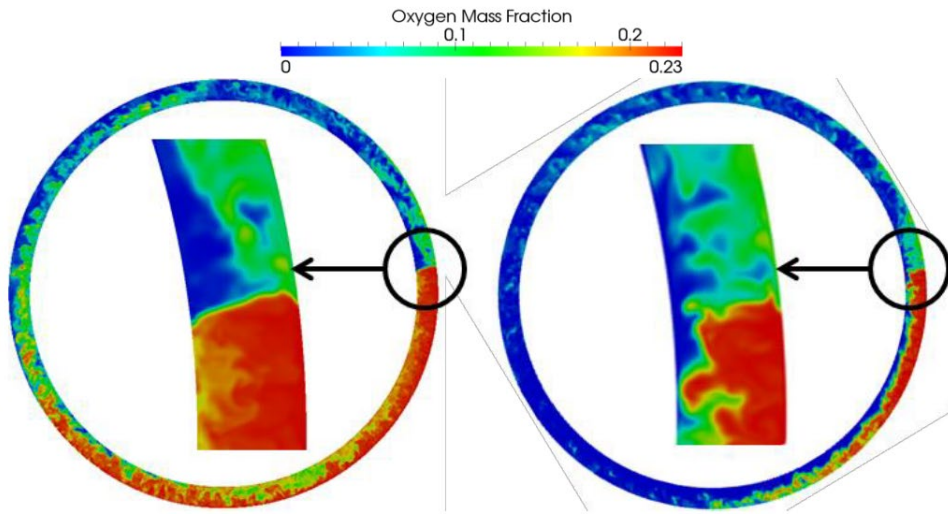


Figure 8. Unreacted Flow Regions of RDE Simulations. Source [8].

In regions where these perturbances of equivalence ratio exist, RDE operation may be locally augmented. Because equivalence ratio plays a major role in the operability of RDEs, regions of locally perturbed equivalence ratio could be expected to exhibit drastically varying detonation characteristics compared to that of the global RDE operability and performance. It has been observed in experiments and computational works

that RDEs have local static pressure oscillations and will be discussed briefly in the next section. While these static pressure oscillations are typically periodic in nature, the cause of such oscillating behavior is not well understood nor characterized to date. Many hypotheses exist, such as a coupling between the combustion chamber and injector dynamics, acoustic wave interaction within the combustor, or local, persistent flow non-uniformities. For instance, if the RDE were to experience a persistent and biased fuel distribution, then the detonation wave may be locally enhanced or attenuated depending on the nature of the non-uniformity. Furthermore, if these local equivalence ratio perturbations persist, it can be reasonably assumed that these perturbations would be reflective in the static pressure jump experienced by the flow across the detonation wave in those regions. This in turn would give way to local non-uniformities in the static pressure within an RDE combustion chamber, which, considering the abridged axial length of these devices, may not adequately “mix-out” prior to exiting the combustion chamber.

2. Oscillatory Static Pressure Behavior

Another factor that leads to local non-uniformities, in the flow characteristics within RDEs, is a low frequency oscillation of static pressure values observed within RDEs [20]. An example of such oscillatory behavior can be seen in Figure 9 where static pressure measurements taken at the same longitudinal plane at three different azimuthal locations can be seen exhibiting a cyclic low-frequency oscillation.

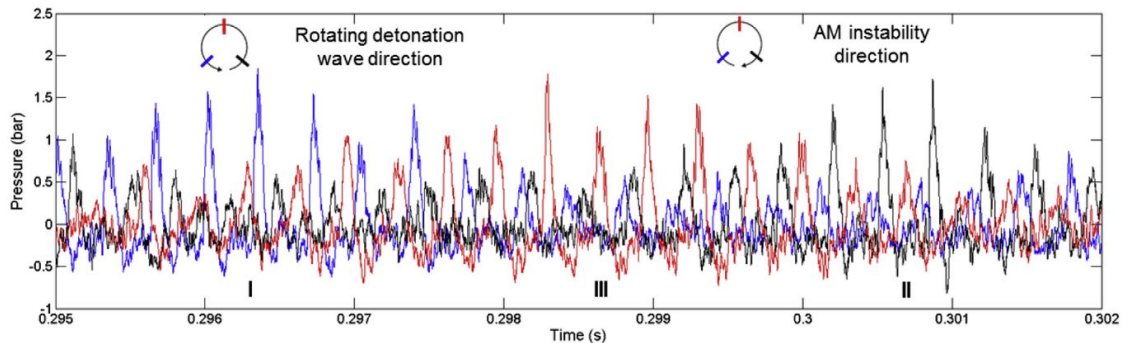


Figure 9. Low Frequency Static Pressure Oscillation in RDE Operation [20].

Similar results have also been detected in simulated RDE operation. Figure 10 illustrates instantaneous static pressure plotted over time for four different azimuthal locations at the same longitudinal plane in a high fidelity RDE simulation. The same general behavior of static pressure oscillations can be seen in both the experimental and simulation results.

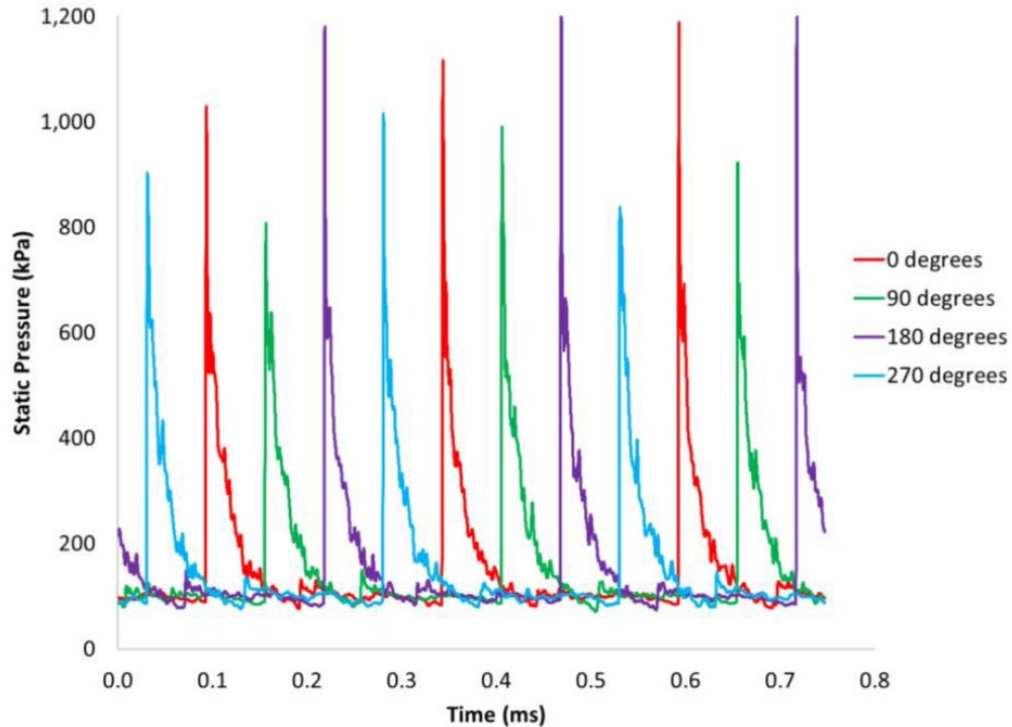


Figure 10. Static Pressure Oscillations in RDE Simulations. Source [8].

The cause for these low frequency static pressure oscillations has remained elusive; however, a few phenomena present within RDE operation could provide an explanation. Acoustic instabilities and interaction within the RDE provide one useful explanation. Reflected waves within the RDE may be creating a transient interference pattern that affects that shock jump conditions of the detonation wave as it travels around the combustion chamber annulus, in effect, locally “priming” the reactants to a higher pre-detonation static pressure [21]. It has also been hypothesized and studied computationally that these acoustic waves and reflections within the RDE combustion chamber have the

ability to affect the bulk flow of the fluid in the air and fuel plenum regions in that they can create a back pressure in these regions that affect the refill qualities and speeds of fuel and oxidizer into the system [7]. This would also influence the mixing of fuel and oxidizer in this region, thus locally perturbing equivalence ratio in an oscillatory fashion if the air and fuel injector experience mis-matched time recovery time scales. These potential causes are extremely difficult to decouple, but nonetheless provide possible explanations of why static pressure oscillations have been detected locally in RDE operation. In an effort to better understand the mechanism causing these local static pressure variations, a brief study and experimental design was conceived to locally augment the fuel injector within the RDE and measure any resulting azimuthal static pressure variability; however, due to lab renovation schedule delays from COVID19, no experimental work was able to be carried out in the timeline of this thesis, nor were the required computational resources available to characterize such a design. The study is detailed in Appendix A.

C. RDE PERFORMANCE CHARACTERIZATION: PRESSURE GAIN COMBUSTION

While it is of great scientific interest to understand the primary factors driving the azimuthal and temporal static pressure variability observed experimentally and computationally, a more practicable consideration is “how can the azimuthal and temporal variability be eliminated for performance characterization?” RDEs are considered a pressure gain combustion (PGC) device. In order to characterize how well an RDE performs with respect to PGC, it is critical to be able to properly measure and report the total pressure within the RDE combustor. Until recently, very few methods for doing this existed [18]. Total pressure measurement devices, such as Kiel probes, are questionable within RDEs due to the extreme fluid flow angles that can exist from shock-induced velocities and rapid expansions post-detonation in addition to shock reflections within the Kiel probe tubes themselves. Instantaneous static pressure measurements are typically difficult or ineffective due to survivability of flush-mounted high speed pressure sensors, the unwanted effect of thermal drift, attenuation affects from standing the pressure transducer off from the combustion zone to protect the delicate sensor, and only report an area-weighted pressure measurement, while it is likely necessary to use a mass-weighted

pressure measurement due to the unsteady nature of RDEs [18]. Examples of these techniques would be high speed pressure measurements using an infinite tube pressure (ITP) measurement setup, where a high-speed pressure transducer is installed tangentially to a small 1/8" tube vented to the atmosphere [22]. Unfortunately, this approach suffers from multiple internal shock reflections, as well as an unknown attenuation of the pressure signal prior to encountering the high-speed transducer sensor. Additionally, a flush mounted fast piezo-electric style transducer (PCB) can be used to measure instantaneous values of static pressure but suffer heavily from sensor consumption and thermal drift effects. The varying degree of “instantaneous pressure measurement” techniques is evident in Figure 11, which shows a flush-mounted PCB signal on the left, compared with various high-speed ITP configurations on the middle and right plots. The overall variability in static pressure magnitude measured by each device is readily evident. The effects of thermal drift on flush-mounted PCBs can also be seen in Figure 12, where the pressure (proportional to the y-axis voltage reading), is constantly decaying throughout the duration of the RDE experiment.

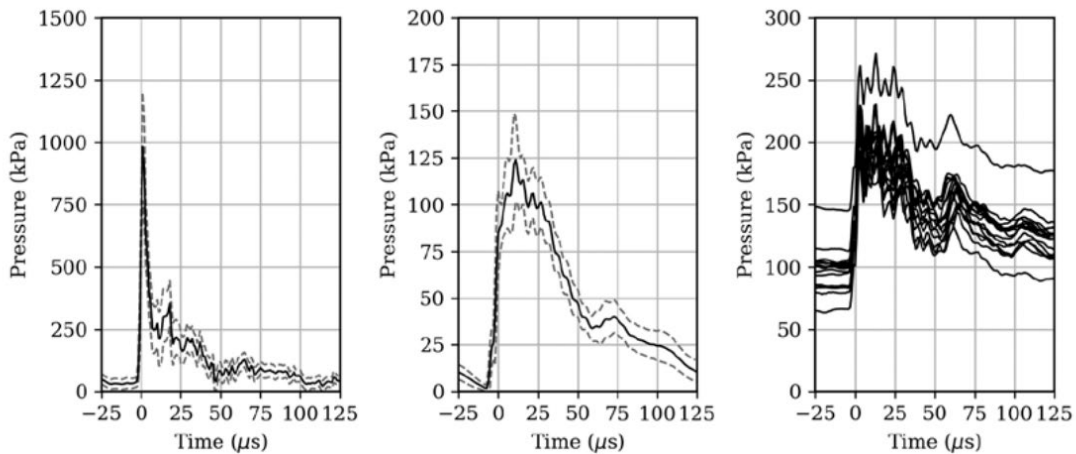


Figure 11. Thermal Drift and Pressure Signal Attenuation within Kiel Probes and ITPs. Source: [22].

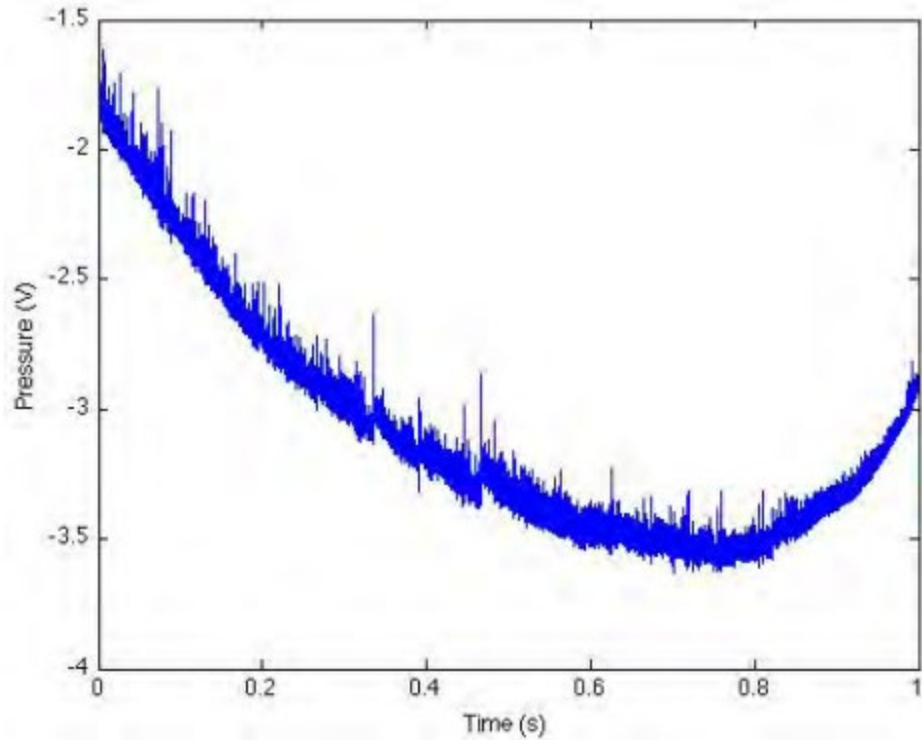


Figure 12. Thermal Drift of High-Speed Pressure Transducers. Source: [23].

In general, the RDE community has agreed that a reliable method for obtaining static pressure measurements with capillary tube average pressure (CTAPs) measurements, where a pressure transducer is installed in a dead-headed configuration at the end of a long, small diameter tube. The high frequency content of the detonation wave is viscously dissipated by the large length-to-diameter (L/D) aspect ratio, and an average pressure within the RDE combustor can be reported [24]. Use of CTAPs to remove frequency content and provide a representative static pressure have been demonstrated in the turbomachinery community for decades [25]. Although, there are still questions about the effects of the detonation forcing function on CTAPs, which can perform work on the fluid and potentially result in an artificially high static pressure measurement value. Regardless, CTAPs are generally perceived to be a reliable method to report an average static pressure within an RDE; however, the question remains: What is an appropriate way to use this static pressure measurement to characterize the total pressure within the RDE and thus the effective PGC?

D. STATIC-TOTAL PRESSURE CORRECTION FOR PERFORMANCE

It is clearly evident that the majority of pressure measurement techniques readily available to characterize the pressure within an RDE are limited, and questionable to some extent. To date, the most defensible method for properly characterizing PGC in an RDE is the EAP method, mentioned briefly in Chapter I and described thoroughly in [26]. Alternatively, it has been shown that the “NPS Method,” appears to produce a reasonable total pressure approximation, when compared to simultaneous EAP measurements [16]. The NPS method utilizes CTAP measurements upstream of the RDE annular exit to determine the total pressure within the combustion chamber, or P_{T4} . This is done by assuming that the Mach number within the combustion chamber can be calculated by steady, one dimensional, isentropic flow relationships [15]. Assuming a combustion Mach number which is based on the area ratio (A_4/A_8), a specific heat ratio γ , and sonic conditions at the exit, the total pressure in the combustion chamber can be calculated by using the static pressure in the combustion chamber measured by a CTAP. This results in a static-pressure-derived total pressure that is represented by Equation 1.

$$P_{T4} = P_4 \left(1 + \frac{\gamma - 1}{2} M_4^2\right)^{\frac{\gamma}{\gamma - 1}} \quad (1)$$

This method of determining total pressure in the combustion chamber has been questioned by the PGC community due to its locally varying Mach numbers in the flow field, as well as potentially unknown effects of applying continuous high-frequency and magnitude forcing functions to the CTAP entrance, which can result in work being performed on the CTAP volume and an augmented resultant pressure. Additionally, P_4 measurements tend to be taken at only a few discrete points within the RDE, and as it was discussed in the previous sections, this may not be representative of the real average static pressure within the RDE due to azimuthal variability within the RDE combustor. Furthermore, to truly characterize total pressure within the RDE, the fluid flow should be appropriately mass-averaged, which is not possible with a single static pressure port if there is a non-uniform momentum/fluid distribution within the RDE combustor. Such a case would require multiple static pressure ports to be implemented; however, doing so with

individual pressure sensors can quickly become cumbersome, prohibitive, and not necessarily representative of the entire RDE flow field, which can remove the advantage of a PGC characterization within an RDE using the NPS Method versus an EAP measurement.

Currently, EAP is the only truly defensible and verified method of determining the total pressure within an RDE combustion chamber [17]. EAP represents a methodology of determining global pressure gain combustion which makes azimuthal variabilities in static pressure unlikely to affect its resulting measurements due to equilibration across the terminal shock at the sonic exit condition of the RDE; however, EAP has proven to be extremely instrument intensive and not attainable for all laboratory setups. This makes methodologies such as the NPS Method more desirable in practice, but the variations in single CTAP measurements are not well understood and produce undesirable uncertainties. Past experimental results have shown a one-to-one relationship between the “NPS Method” and the trusted EAP measurements, which was shown in Figure 3 in Chapter [16]. With this, it is currently believed that a properly taken CTAP measurement applied with the NPS Method results in an accurate representation of total pressure within the combustion chamber.

E. SPATIALLY AND TEMPORALLY AVERAGING STATIC PRESSURE

While the cause(s) of the previously recorded azimuthal static pressure variability is still quantified, it is likely laboratory- and/or RDE-dependent parameters, which makes clarifying a single methodology for correcting these variabilities extremely difficult. In other words, it is likely to be extremely challenging to recommend appropriate standards for how many static pressure measurements should be taken, and how to locate each static pressure measurement, in order to report a single, representative static pressure within an RDE. It therefore becomes desirable to create a method in which the variabilities can be removed from the measurements altogether. CTAPs have proven to be a reliable method of temporally-averaging static pressure; however, laboratory and simulated results have shown a great deal of spatial variability. Methods of spatially-averaging pressure within unsteady combustors have existed since the 1930s via piezometers [27]. An example

piezometer setup can be seen in Figure 13 with a pseudo “ring” placed around a fluid flow which contains small openings to which the fluid flow can enter the ring and become spatially averaged.

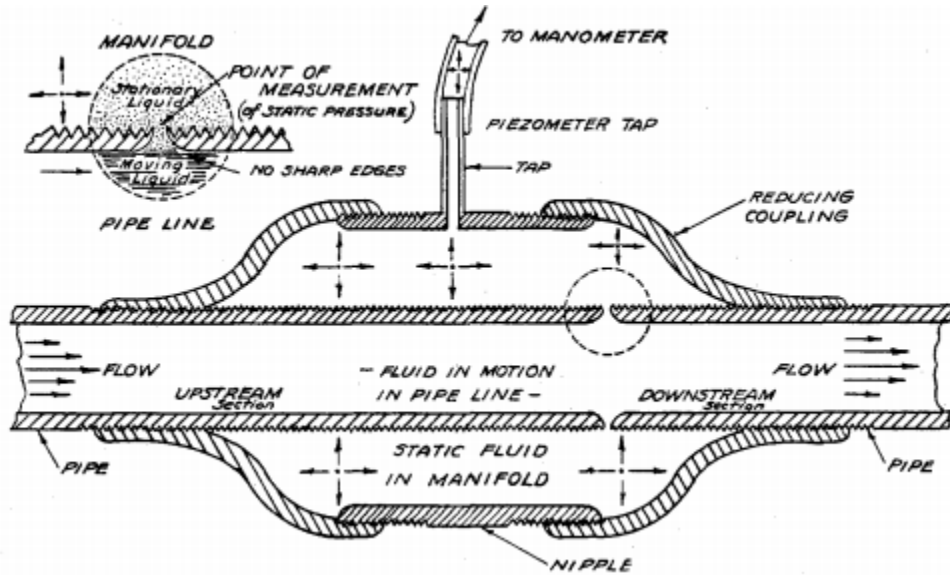


Figure 13. Example Piezometer Setup [27].

This idea of spatially-averaging pressure became the inspiration for creating a device which can both spatially- and temporally-average pressure which is input from an RDE combustion chamber. A device similar to that of a piezometer “ring” would be designed that would spatially-average the input static pressure and thereby remove azimuthal non-uniformities and the effects on static pressure measurements. In this design, CTAP lines and transducers would also be used to remove transient flow effects from the measurement and arrive at a temporally averaged static pressure value which in combination with the spatially-averaging features of the design, would create a device that would both spatially and temporally average the static pressure representative of that in an RDE combustion chamber.

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III. DESIGN OF EXPERIMENT

In order to design a sufficient pressure averaging device (PAD) that can measure a static pressure in unsteady combustors, that is representative of a spatially and temporally equivalent static pressure, the hardware in the laboratory must be conducive to such a design.

Initially, this thesis intended to experimentally quantify whether the previously observed azimuthal static pressure asymmetries were due to a non-uniform fuel distribution, in addition to experimentally quantifying whether an effective spatially- and temporally averaged static pressure could be measured with appropriate apparatus. Although, due to the COVID19 pandemic, lab renovation schedule overruns prevented any experimental tests for this thesis. So, while the intended set of experiments could not be carried out in the current thesis, the overall design and intent of the test campaign is discussed in this chapter.

A. RDE OUTER BODY REDESIGN

In order to support an increase in ease of assembly as well as an increase in both overall instrumentation and ease of instrumentation for future experiments conducted on the RDE in Test Cell #2 at the NPS Rocket Propulsion Combustion Laboratory (RPCL), it was required that the RDE outer body be redesigned. The previous RDE assembly, seen in Figure 14, featured one large outer body piece that connected to the injector system on the upstream side and a final flange piece on the furthest downstream side. The flange held the outer body and air injector in compression between the furthest upstream flange. This design required careful handling of a large, heavy RDE outer body, air injector, and flange simultaneously while the engine was assembled. This assembly process was rather difficult and alone provided enough motivation for a redesign of the RDE outer body. In addition to the issues regarding difficulty of assembly, the current outer body had only eight CTAP ports which were all focused on one quadrant of the RDE combustion chamber. This instrumentation set up is not conducive to measuring and characterizing the azimuthal variability seen in the CTAP measurements as previously discussed, nor is it apparent

whether this distribution would be good enough for providing a spatially-averaged static pressure value. In order to capture more of the variability and to arrive at a true average of the pressure seen in the RDE combustion chamber, more CTAP ports across more azimuthal locations of the RDE are required. With these issues in mind, the outer body was redesigned with the following goals.

- Increase in ease of assembly
- More CTAP ports across more azimuthal locations
- Introduce modularity for future designs and experiments

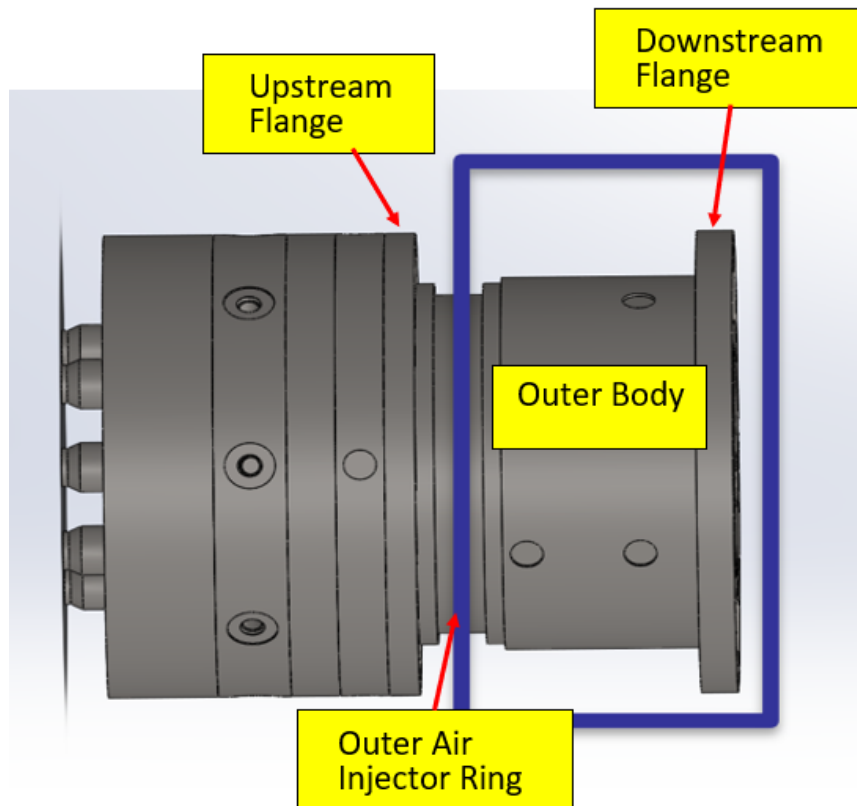


Figure 14. Previous RDE Assembly and Outer Body

1. Increase in Outer Body Ease of Assembly

One of the biggest problems concerning the assembly of the outer body on the rest of the RDE structure was its sheer weight. The current design which featured one large outer body piece, an injector, and a final compression flange, required all three components to be held in place while simultaneously installing the final compression flange. The combination of heavy components, and multiple components, made assembly extremely difficult. When redesigning the outer body stackup, the idea was to break down the single, large, heavy outer body into two smaller, more easily handled outer body pieces with integrated flanges. This would naturally make the individual parts lighter while simultaneously reducing the total number of components that needed to be held in place at any given time during assembly.

Adopting the idea of using multiple outer body pieces rather than one large outer body into the outer body redesign process, it was conceptualized that each new outer body piece should have its own flange in order to couple it to the rest of the RDE assembly. In this manner, each new outer body piece could be locked down to the greater RDE structure without having to be held in place while the next piece is assembled. Additionally, the final flange piece which originally was used to bind the entire outer body to the RDE, was incorporated as the final outer body flange in order to reduce the number of parts required for the RDE. A depiction of this two-piece new outer body design can be seen in Figure 15.

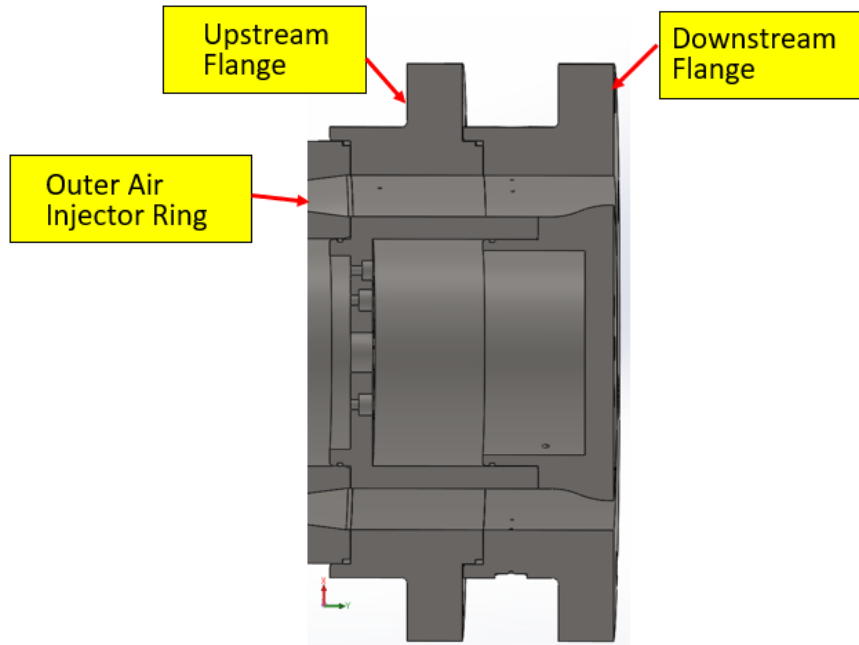


Figure 15. New Outer Body Two Piece Design

Figure 15 illustrates the two new outer body pieces with their own incorporated flanges. The overall length of the two new outer body pieces was designed to perfectly match the combined lengths of the previous singular outer body piece in addition to the final flange piece in order to keep the new design flush with the exit plane of the combustion chamber. This was of critical importance as the new outer body is not designed to change any of the flow properties of the RDE itself. Figure 16 highlights the two flanges of each respective outer body piece and the method in which they interface with the rest of the RDE assembly.

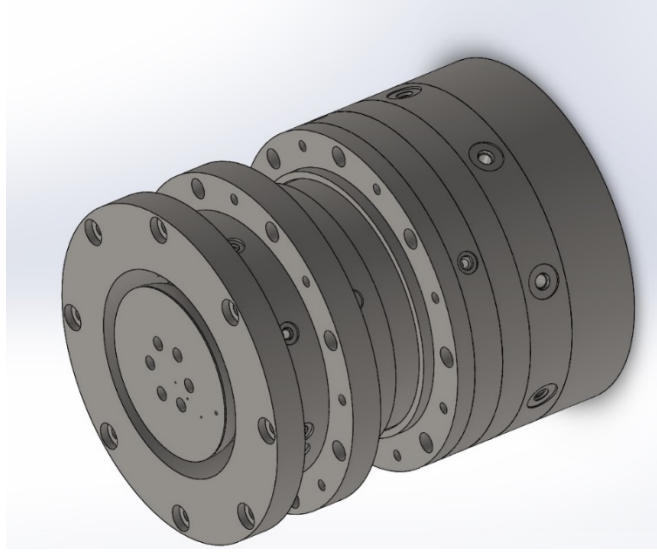


Figure 16. New Outer Body Flanges

The two new outer body pieces connect to each other as well as to the flange immediately upstream of the injector. This is done via a series of inter-digitated counterbored thru-holes, and 3/8-24 threaded holes that will be used to fasten the two new flange pieces together and to the upstream RDE assembly. Each set of counterbored thru-holes align with a set of threaded receiving holes. With these changes, the new outer body will be significantly easier to handle and assemble in future experiments.

2. Instrumentation of New Outer Body Design

As mentioned previously, the current outer body was lacking the instrumentation capabilities required to fully characterize and capture the azimuthal variability previously measured in NPS RDEs. In order to properly characterize the effectiveness of the PAD design, multiple static CTAP ports are required around the entire circumference of the RDE outer body to acquire a sufficient spatial average. Additionally, the capability to capture this variability at more azimuthal locations is critical to future experiments.

One such experiment is designed to test the hypothesis of the CTAP measurement variability being due to an asymmetrical fuel injection. In order to test this hypothesis, a known asymmetry would be induced in one or more quadrants of the fuel injector and multiple, distributed static pressure CTAP measurements would be taken in the combustion

chamber to resolve any induced azimuthal pressure variability correlating to the intentionally biased fuel injection. In order to do this, a grouping of CTAP ports would be required in at least two different azimuthal locations. Each grouping would coincide with a locally perturbed fuel injector quadrant. The recommended grouping of CTAPs is captured in the redesign and shown in Figure 17.

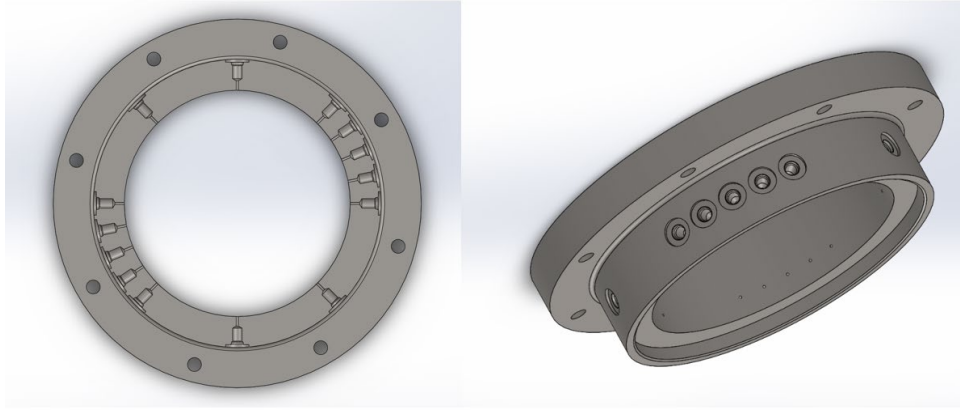


Figure 17. CTAP Port Grouping

Five CTAP ports were grouped tightly together in the redesign within a 45-degree arc. This same CTAP grouping feature is mirrored on the other side of this final outer body piece. This yields two groups of five CTAP ports exactly 180-degrees apart from each other. With this design, the outer body is well-suited to capture any induced static pressure azimuthal variability from an intentionally biased, asymmetric fuel injector.

Another experiment that influenced the instrumentation redesign of the outer body was the one by which this thesis work was originally conceived, being the design of a device, which can spatially- and temporally-average the pressure within an RDE combustion chamber, to provide a sound metric of performance for RDEs while nullifying the azimuthal variability within the RDE. In order to support this experiment, the outer body required more instrumentation ports for CTAP measurements across more azimuthal locations than what was previously available. To fully capture the variability and test if a device proposed in this thesis would be able to correctly nullify the variability seen, measurements would need to be taken from multiple azimuthal locations along the entire

circumference of the RDE combustion chamber. With this in mind, the final flanged component of the new outer body was redesigned to include a total of 14 CTAP ports. This is including the ten from the previous discussion as well as an additional four. Two CTAP ports were placed at even intervals between each grouping of five CTAP ports. These additional CTAP ports can also be seen in Figure 17.

In terms of additional instrumentation for possible future experiments, four additional CTAP ports were included on the upstream outer body piece. It is worth noting that the longitudinal location of the ports correlates with the detonation zone of the combustion chamber. This bears no particular interest to the experiment of this thesis work; however, this may be useful for future students conducting experiments on the same RDE setup, so these additional ports were included in the final outer body redesign.

3. Outer Body Redesign Process and Specifications

The outer body redesign process began with an understanding of the underlying problems with the previous design, namely the difficulties to assemble as well as inadequate instrumentation for currently-planned. From this point, design decisions were completed in an iterative process of Solidworks modeling and fit checking various features of the new design with the previous design and laboratory set up. In this manner, the clearances of mounted CTAP hardware between the flange walls were able to be confirmed as suitable (i.e., no interference would occur during CTAP installation). Additionally, the new design needed to fit within the rest of the RDE system without having to change any other RDE component, such as the axial length of the RDE center body. These requirements determined the size and clearance of each flange, with respect to the CTAP instrument ports, as well as the overlap required for each gland-seal type O-ring between the upstream and downstream outer body components. The final concern was the manner in which the outer body components maintain concentricity, through the aforementioned O-ring gland seals. Given that the outer body pieces would be secured by the overlapping male-female glands, positioned radially away from the rest of the RDE system, a gap of 0.00254 cm (0.001 inches) was designed between the two inner-most outer body pieces to prohibit impingement of the components during installation. These design choices and

specifications can be seen in Figure 18 and machinist drawings of the new outer body design can be found in Appendix B.

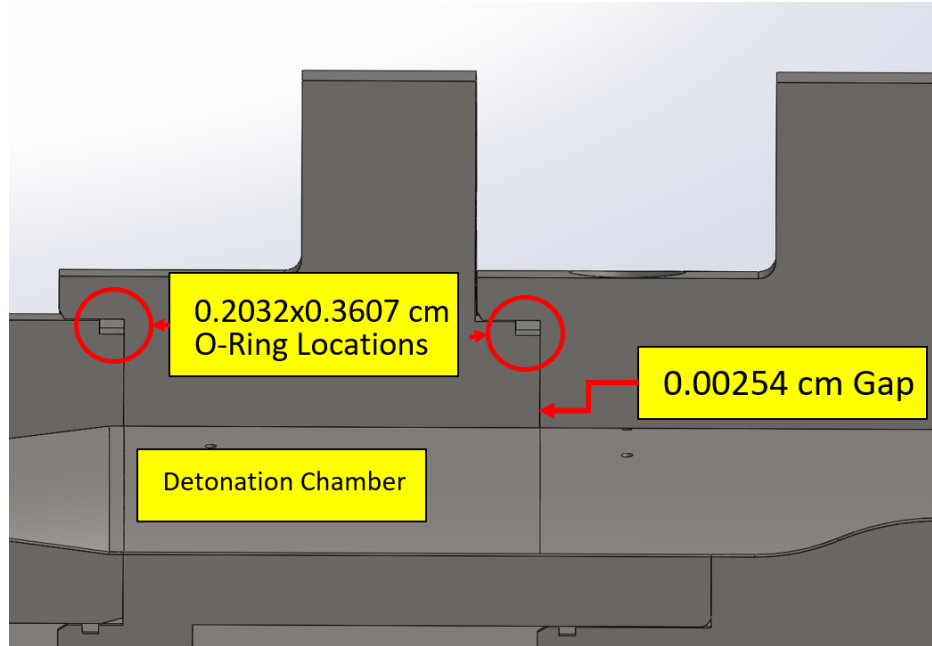


Figure 18. Redesigned Outer Body Specifications

4. Outer Body Design Modularity

Due to the new outer body design being broken into two pieces, it now becomes possible for future students and researchers to design modular pieces for the furthest downstream segment of the outer body. This would be done when the instrumentation capabilities of the proposed new outer body design do not meet the requirements of a future researcher for the purposes of their work. This introduces modularity into the outer body assembly that previously did not exist and would have required a full redesign of the outer body. In this way, future researchers will be able to easily design a new downstream, or upstream, component for the outer body that meets their given test requirements.

B. PRESSURE AVERAGING DEVICE

In designing a pressure averaging device (PAD), a few things needed to be considered that are of critical importance. First, the device needed to be able to “take in”

pressures from various azimuthal locations in order to capture the potential full spread of azimuthal variability seen in previous CTAP measurements. The devices also need a way to confirm that the measured pressure from within the device is properly averaged. Another major consideration was the time required to fully pressurize a proposed PAD design. Most NPS RDE runs last only approximately 1.5 seconds; therefore, any proposed PAD design would need to be fully pressurized well within this time window.

1. PAD Design and Features

The basic premise behind the PAD is to use the temporally averaging characteristics of CTAPs in conjunction with a settling reservoir/plenum to spatially and fluidically average the static pressure values taken from multiple points within the RDE. The PAD itself is an external device which receives multiple CTAP lines from the RDE outer body. This technique makes the pressurization time requirement particularly important because both the PAD and the individual CTAP lines being routed to the PAD will need to be pressurized prior to measuring a steady-state, effective average pressure. This requirement correlates to a very small volume for the PAD pressurization chamber. This small volume would be difficult to machine traditionally, and the optimal volume is difficult to estimate a-priori; therefore, 3-D printing is the best available option for the manufacturing of the designed PAD because the internal reservoir volume can be cheaply and quickly iterated upon by utilizing NPS-owned additive manufacturing resources. However, 3-D printing places geometric constraints on the size of the PAD design. In order to be printed on the DMLS 3-D printer available to the NPS Rocket Lab, the PAD must be within 80x80x80 mm (3.15x3.15x3.15-in) in overall size. The aforementioned scaling parameters are the main constraints on the overall PAD design.

The designed PAD was conceptualized as a two-sided design. One side would contain the pressure inlets for the device, called inlet ports. The other side would contain multiple pressure outlets in the form of CTAP ports, called outlet ports. Multiple outlet ports are needed in this design to confirm that the pressure within PAD is properly averaged in that each independent CTAP measurement from each outlet port should record the same value. CTAP lines from the RDE will terminate at the inlet ports and should provide a

temporally-average pressure signal to the pad; in the reservoir, these temporally-averaged pressures, which are likely to have different static pressure magnitudes due to the RDE azimuthal variability, will be fluidically spatially-averaged. Any residual temporal effects from the cyclical pressurization of the PAD reservoir, could then be eliminated via a CTAP configuration on the outlet port, if required.

Determining the overall shape of PAD was another trade space which required several considerations. Knowing that the volume of the pressurization chamber needs to be small enough to allow for a reasonable pressurization time-scale but also large enough to act as a settling reservoir to properly distribute the fluid and provide a good representation of the spatially-averaged pressure an annular ring design was settled on as the most suitable candidate. This is because the annular ring pressurization would allow for a large amount of CTAP input ports on the centerline, while also offering a high degree of control over the pressurization chamber volume by adjusting the depth and thickness of the reservoir. Given this general geometry, size, and pressurization rise time constraints outlined above, 12 CTAPs were able to be fitted on the inlet side of the PAD design as can be seen in Figure 19.

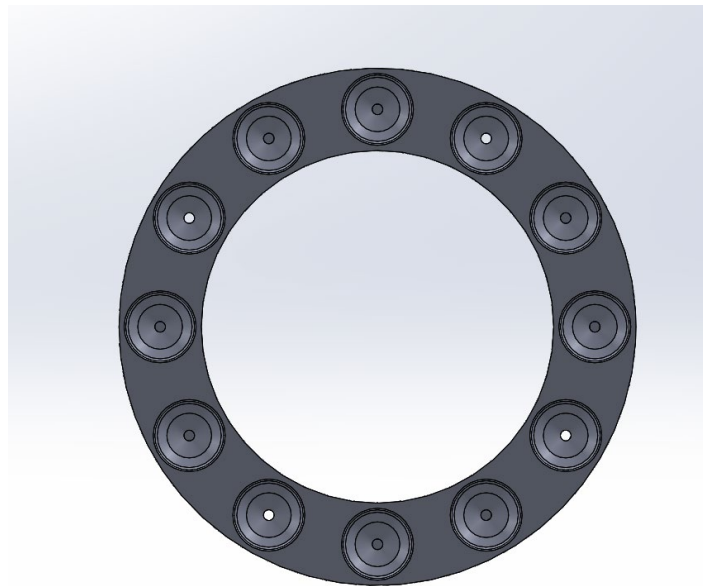


Figure 19. Inlet Side of PAD

As previously stated, the overall pressurization time was another major concern in the design process for the PAD. Figure 20 shows representative rise times for pressurization given different length CTAPs.

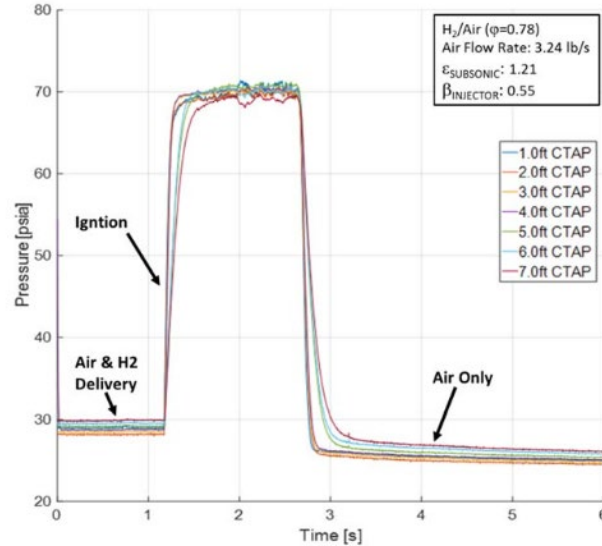


Figure 20. Rise Times for Pressurization of Different Length CTAPs. Source: [7].

Given that even for the longest tested CTAP tube of 2.13-meter (7 feet), the pressurization occurred relatively rapidly and well within the overall RDE run time, the 2.13-meter CTAP volume and overall pressurization time provided a benchmark for the PAD design. The inner diameter of the CTAPs in question is 0.0762 cm (0.030 inches), which results in a total volume of approximately 0.967 cm^3 (0.059 in^3) for a single 2.13-meter CTAP. Knowing this volume gives a good understanding of the relationship between the pressurization volume and the overall rise time associated with that volume, considering an RDE operating at roughly 482 kPa (70 psi). The pressurization should increase for an RDE operating at a lower-pressure condition. In order to keep the pressurization rise time of the PAD design on the order of magnitude similar to that of single CTAPs, the height of the internal pressurization chamber was elected to be just high enough to not impede the inlet of the CTAP port. This height was found to be

approximately 0.165 cm (0.065 in). In order to match the pressurization volume of the 2.13-meter CTAP, the depth of the PAD pressure reservoir, defined as the distance between the inlet and outlet CTAP ports, was determined to be 0.2794 cm (0.11 in). It is important to keep in mind that the pressure inputs into the PAD design will pressurize the PAD volume in parallel in that with more inlet ports being utilized on the PAD design, the faster the PAD will pressurize. With this in mind, the final pressure reservoir volume within the PAD design, in which the static pressure from the RDE combustion chamber becomes spatially averaged was 0.97 cm^3 (0.059 in^3) which is approximately equivalent to that of a single 2.13-meter long CTAP line. The pressure reservoir volume of the designed PAD can be seen in Figure 21.

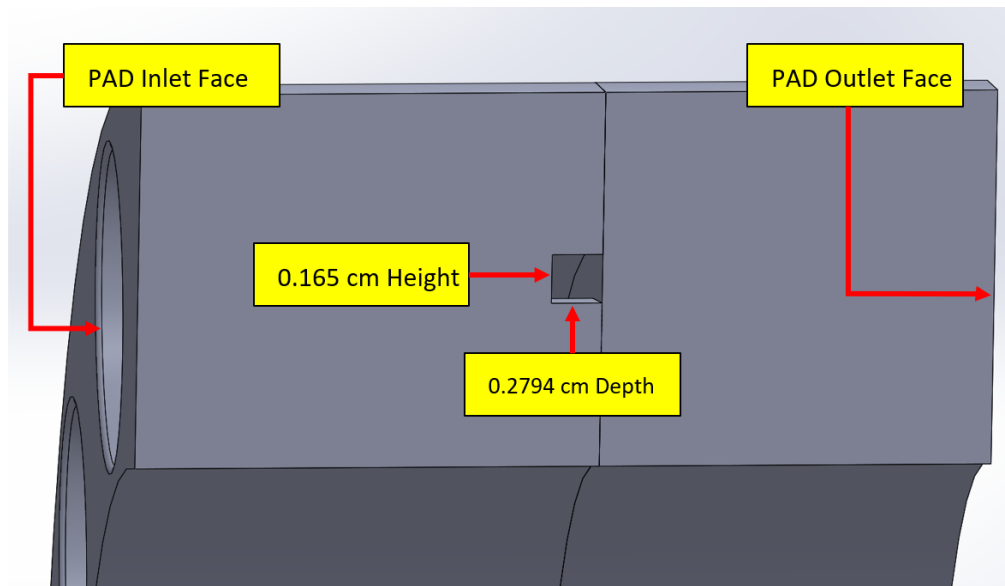


Figure 21. PAD Pressurization Volume

It should also be noted that the designed PAD does not require all 12 ports be used as some of the inlet ports can be plugged. Also, by using a PAD in conjunction with the redesigned outer body, the ability to compare static pressure measurements derived from the designed PAD and conventional CTAP methodologies is possible. This simultaneous comparison can be accomplished by splitting the flow with a T-connection just prior to fluid entering the settling plenum/reservoir through the inlet ports. In this manner, the flow

can go to a dead-headed pressure transducer, similar to a traditional CTAP measurement, as well as to the inlet port feeding the PAD device. This approach would help to provide comparison of the pressure measurements gathered from PAD versus pressure measurements taken straight from the RDE combustion chamber via traditional CTAP measurements and evaluate the difference between the two methodologies. This comparison has the potential to provide further support that the designed PAD nullifies the azimuthal variability seen in CTAP measurements taken from NPS RDEs over various azimuthal locations around the combustion chamber.

The outlet section of PAD consists of four CTAP ports. Four CTAP outlets were included in the design to provide the ability to check multiple outlets simultaneously in order to verify that the pressure within PAD is the same at various azimuthal locations. The outlet side of PAD can be seen below in Figure 22.

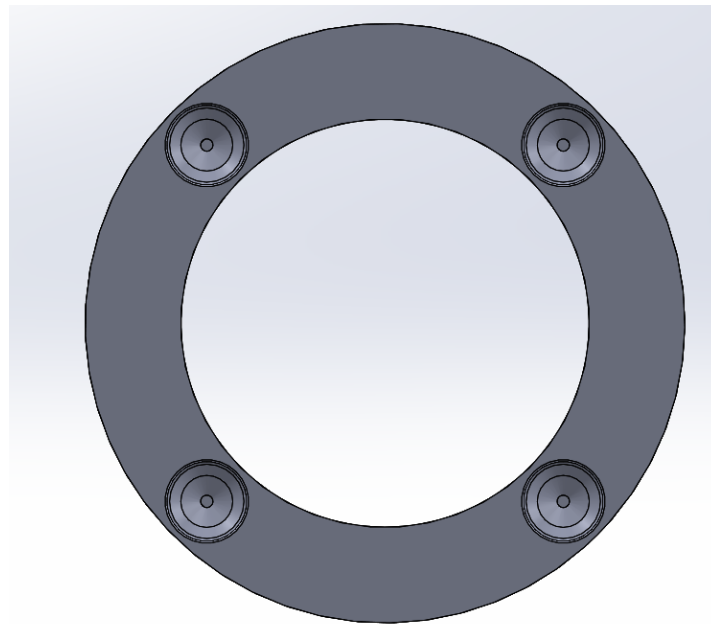


Figure 22. Outlet Side of PAD

With the careful design process outlined previously, the designed PAD will be able to adequately average the pressure present in an RDE combustion chamber in both a spatial

and temporal fashion. The PAD design will also be able to provide verification of proper averaging by both having four outlets and, through use in conjunction with the redesigned outer body, be compared against conventional single CTAP methodologies of measuring static pressure within an RDE combustion chamber. Additionally, the NPS Test Cell #2, where these tests will likely be conducted, has enough expansion capability to measure EAP simultaneously, which would provide a complete dataset comparing the three techniques for performance characterization within an RDE. To that end, it is recommended to perform all three tests simultaneously for, multiple mass flow rates, equivalence ratios, and ideally back-pressurization ($A_8/A_{3.2}$) ratios.

IV. COMPUTATIONAL MODELING

A. GENERAL CFD SETUP AND PREPARATION

Due to challenges in working in the COVID-19 environment, it became impossible, given the timeline of lab renovations and my personal graduation date, to test the PADs in the lab. Instead, suitable CFD simulations were created in ANSYS Fluent to characterize the validity of the designs outlined previously. In order to simulate the validity of each PAD, four CTAP lines joining in a common plenum were modeled in ANSYS.

To characterize the performance of a PAD design it is of critical importance to characterize the manner in which measurements would be taken. In this regard, it was important to model and characterize how CTAPs respond to an unsteady forcing function. This introduced a group of simulations that would perform two functions for the purposes of this thesis. First, they would provide an understanding of the inner mechanics of CTAP pressurization, I.e., the resulting pressure profile given an unsteady forcing function for two CTAP lengths. Second, they would serve as the basis for a mesh refinement study from which the resulting meshing methodology could be applied to the larger simulations, which are naturally more time-consuming to numerically model. The CTAP computational model geometry was designed to match the setup described in Fotia et al. [28], which uses a similar CTAP geometry to the standard NPS geometry. The intent of closely matching the experimental setup described in Fotia et al. is to provide an experimental comparison to the current numerical modeling effort, assuming a similar input forcing function profile can be generated and compared. (The CTAP geometry consists of a 0.762mm (0.030-in) internal diameter tube of various lengths (0.25m and 1.00m depending on the simulation), terminating in a cylindrical reservoir. NPS uses CTAPs of varying length, but for the purposes of this research, 0.25m CTAPs were used for the mesh refinement study in order to reduce the element count of the resulting mesh and therefore reduce the required computational time to run these simulations. A comparison of a 1.00m CTAP response is also modeled. The CTAP tubes terminate in a cylindrical reservoir that is designed to simulate the dead-headed volume present for a real CTAP transducer installation. The aft wall of these cylindrical volumes are representative of the pressure sensor membrane,

where pressure would be physically measured in an experimental setup. resulting in a reservoir volume of 56.67mm^3 (0.003 cubic inches). Figure 23 illustrates the CTAP computational model. The inlet can be seen on the left side and the reservoir can be seen on the right side of the figure. Dimensions for the modeled single CTAP are as follows: CTAP tube diameter (D) = 0.0762 cm (0.030 in), CTAP tube length (L) = 0.25 m, reservoir diameter (R_d) = 0.82 cm, reservoir length (R_L) = 0.11 cm.

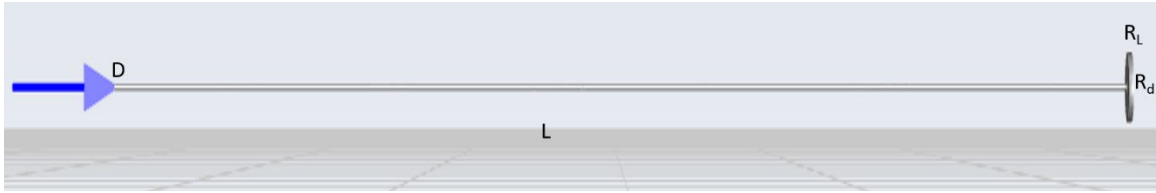


Figure 23. CTAP Model Geometry

B. COMPUTATIONAL PARAMETERS AND BOUNDARY CONDITIONS

All simulations were set up as transient, density-based simulations. The k-omega viscous model was used as it is most appropriate for characterizing flow conditions near a wall as is present in the case of modeling a CTAP as well as PADs. The valid boundary conditions used in this research were arrived at via an iterative process of trial and error. A basic requirement for these simulations is the ability to accurately model the inputs that would be seen in an RDE; however, they also needed to be feasible in terms of computational resources as well as overall computational time. There are many ways to potentially model the pressure input into a CTAP delivered by an RDE, chiefly a simple stepping function, a sinusoidal wave, a sawtooth function, or a delta function with an exponential decay. Each of these options provide their own set of benefits and drawbacks in terms of numerical accuracy, computational complexity, and required resources. All of these options could in theory provide a suitable simulation for fluid flow in a CTAP tube driven by unsteady combustion operation.

1. Pressure Input Modeling

Initially, it was conceived that an initial pressure wave could be modeled in the system via a delta function causing the pressure input boundary condition to suddenly step from the initialized fluid domain pressure to some higher pressure. In this model, the fluid volume is initialized to 101.3 kPa (1 atm), and undergoes an instantaneous pressure rise to 303.9 kPa (3 atm) at the inlet boundary. The thought being that the response function of the CTAP, as well as the pressurization rise-time (time to acquire steady state) could be modeled. This was quickly proven to be ill-suited for the current work for two core reasons. First, the immediate pressure jump proved to be too quick of a pressure rise for the mesh to accurately resolve the resulting flow conditions throughout the flow field, without a prohibitively small time step, even for the larger mesh elements studied herein. Second, the amount of computational time required for the fluid volume to reach a steady state condition when initialized at 1 atm proved to be intangibly long to run the simulations when considering resource limitations. From this, the future simulations needed to be initialized closer to their appropriate estimated steady state value, and a new pressurization function would be required. These implementations would allow for a generally coarser mesh as well as shorter computational times.

Considering these lessons-learned, a sinusoidal pressurization function was implemented at the inlet. This behavior is more closely matched to that of RDEs as it simulates the varying pressure values present at the entrance to the CTAP around a central mean pressure value. In other words, a sinusoidal pressure input would result in both higher and lower pressures at the inlet that would create regions of locally high and low pressures at some known frequency, which are attenuated down the length of the CTAP. In this manner, CTAPs remove the high-frequency pressure fluctuations within unsteady combustion flows and provide a temporally-averaged pressure measurement at a point location. The sinusoidal function utilized was centered at 3 atm with an amplitude of 2 atm and a frequency of 3 KHz which closely matches RDE operating conditions. The basic equation for this sinusoidal pressure input can be seen in Equation 3. This pressure input was found to be suitable in terms of computational complexity and time while also

providing a more accurate representation of the pressure input that would be delivered by RDE operation than a simple stepping function.

$$p(t) = A\sin(2\pi ft) + B \quad (2)$$

2. Transient Simulation Time Step Selection

Another trade space which was examined was the time step required for fully resolving the flow characteristics within the modeled CTAP. Being that these simulations are transient, the selected time step is of crucial importance to the overall accuracy of the simulation. A simulation which has too large of a time step will fail to accurately resolve important flow characteristics within the flow field as well as have the potential to diverge from any solution entirely. Conversely, a simulation with too small of a time step will require more computational time and resources than is necessary. Time step determination is driven by the Courant–Friedrichs–Lewy (CFL) number. The CFL number, or CFL condition, is a number which represents the distance that information travels during a time step, with the distance being in terms of mesh elements and can be seen numerically in Equation 3.

$$C = a \frac{\Delta t}{\Delta x} \quad (3)$$

To resolve a numerical simulation accurately, this number should theoretically always be less than or equal to one. If the CFL number is greater than one, information is traveling further than one element in the mesh per time step which can result in numerical instabilities. In order to ensure that the simulations in this body of research are fully resolved, the adaptive time-stepping feature was utilized in ANSYS Fluent with a target CFL number of one. The adaptive time-stepping feature updates the time step after each successive time step iteration in order to provide a suitable, but non-uniform, time step. With this, all simulations would be fully resolved without using inappropriately small, or large, time steps.

3. Temperature Conditions

In order to accurately simulate RDE input behavior into a CTAP, the temperature conditions of the walls of the CTAP, the pressure inlet, as well as the fluid volume must be considered. It was determined that the fluid volume should be initialized to 300 K and the pressure inlet should be set to 1750 K in order to properly model real-world conditions; however, the wall conditions proved to be more elusive. Initially, the wall conditions were set as adiabatic to reduce potential computational time with the understanding that the physics would not be exactly representative of a CTAP, which effectively has an isothermal wall boundary condition; however, in a trial run of this condition in which both the fluid volume and the pressure inlet of the modeled CTAP were set to a temperature of 1750 K, it was found that adiabatic compression was occurring at the inlet to the CTAP fluid volume. This phenomenon can be seen in Figure 24, and is not thermodynamically accurate to its real-world counterpart in which heat transfer is allowed to take place. Simultaneously, it was realized that the largest factor in determining simulation run times (wall time) was the extremely small time steps being taken by setting the CFL number to 1. Whereas convergence within each time step was achieved extremely rapidly due to the relatively small mesh sizes used. It quickly became evident that employing a more realistic isothermal wall boundary condition would be unlikely to drastically alter the simulation times necessary.

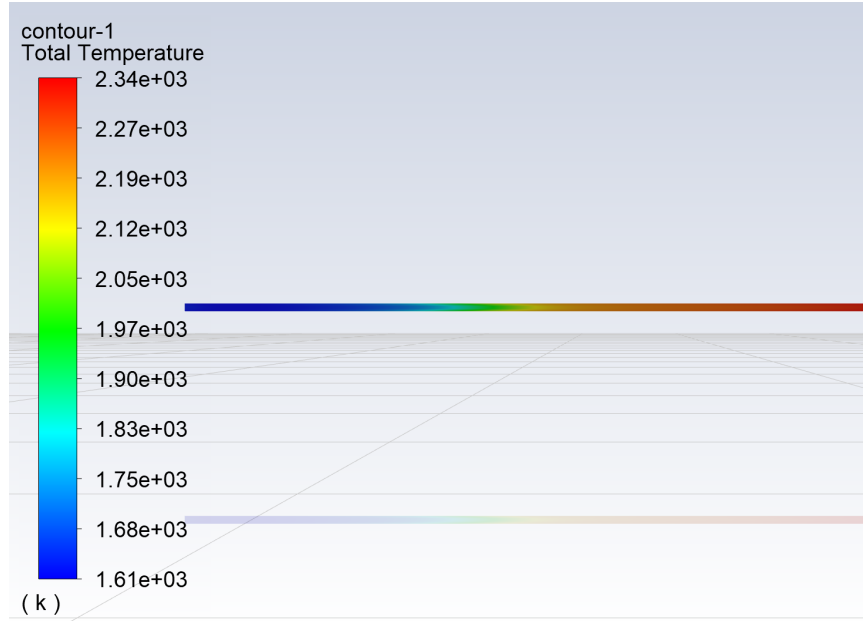


Figure 24. Adiabatic Compression along CTAP Wall

Once this was realized, in order to model the system more accurately, the wall boundary conditions were adjusted in order to allow for heat transfer between the CTAP wall and the fluid volume. In order to accurately model this condition as well as remain consistent with the fluid volume conditions, an isothermal wall boundary condition was used, with the wall temperature held constant at 300 K.

4. Boundary Condition Summary

Due to the iterative trial and error process in which the appropriate boundary conditions were determined, it is important to explicitly outline the boundary conditions used in this research in order to assist in further research as well as provide additional clarity. The pressure inlet was defined by a sinusoidal function centered around 3 atm with an amplitude of 2 atm and a frequency of 3000 KHz in the manner shown by Eqn. 2. The temperature of the pressure inlet was set to 1750 K in order to simulate a representative input temperature from an unsteady combustor. The wall boundary conditions were set as an isothermal wall at 300 K, allowing for heat transfer between the CTAP wall and the fluid volume. The fluid volume was also initialized to 300 K. Additionally, to reduce computational time, the fluid volume was pre-initialized to 2.75 atm. Doing so ignores the

initial pressurization behavior and characterization, but does accurately model the “steady state, temporally-averaged” pressure of interest for the present study.

Using the boundary conditions described above, a mesh refinement study was conducted. The single CTAP model provided the basis for this mesh refinement. Three mesh “cases” were examined. The first case was a mesh with three elements across the diameter of the modeled CTAP inlet. The second case was a mesh with five elements across the inlet diameter and the third case was a mesh with seven elements across the inlet diameter. Figure 25 illustrates the five elements across the inlet diameter mesh with four total elements and two half elements, totaling in five elements.

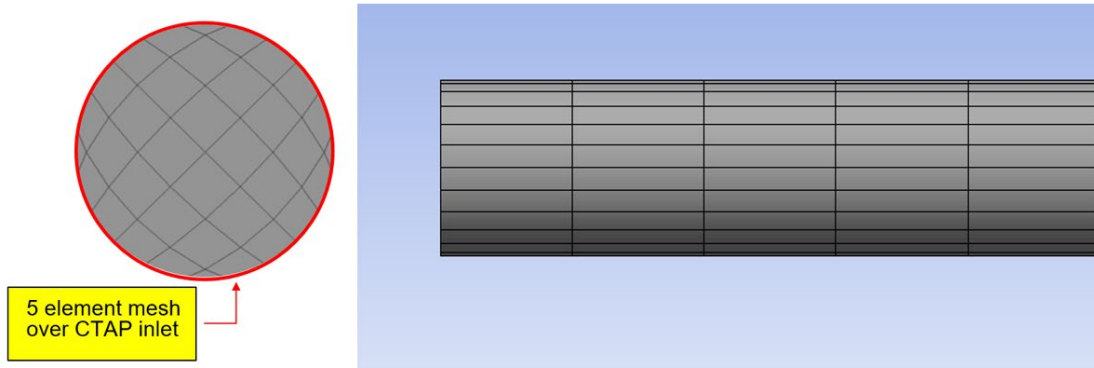


Figure 25. Five Element Mesh

Increasing the number of elements across the inlet diameter resulted in an exponential increase in both element count as well as the time step taken by the adaptive time step feature utilized in these simulations. Recall Equation 3 which illustrates that for a decrease in mesh size, the time step must also be reduced proportionally to maintain the same CFL number. This drastically increased the overall computational time for a simulation with a fine mesh, which is a factor that was considered when selecting a mesh to use for later simulations. The element counts and approximated time steps are shown in Table 1.

Table 1. Mesh Refinement Statistics

Mesh Case	Element Count	Time Step
3 Element Case	999	~100 ns
5 Element Case	6550	~80 ns
7 Element Case	116000	~10 ns

The results of the mesh refinement study can be seen in Figure 20 with the fluidic flow time plotted on the x-axis and the static pressure on the y-axis. The inlet boundary condition pressure are shown by the solid black line with a peak-to-peak amplitude of 101.3 to 506.5 kPa (1 to 5 atm) and the area weighted static average pressure for the transducer wall of the CTAP model, which is shown representatively in Figure 26. The results of the mesh refinement study show that with an increase in mesh refinement (decrease in element size), the time average static pressure of the transducer wall of the simulated CTAP becomes closer to mean input static pressure of 303.9 kPa (3 atm) with the seven-element mesh only being 2.13% different than the mean value. Whereas the five and three element meshes were 3.48% and 4.97% different, respectively. Note, the resulting pressure measurement along the effective pressure sensor head initially over-shoots the mean pressure, and asymptotically approaches the overall mean pressure input value. Unfortunately, due to time constraints, it was not possible to run the fluid simulation long enough to accurately represent the final steady-state pressure value.

Even though the simulations did not run long enough to reach a steady-state convergence to the expected mean pressure value the general oscillating trend remained the same for each mesh case tested in this mesh refinement study. This would indicate that using the most refined mesh would not be necessary to capture the trends and general behavior of the fluid flow in both the single CTAP simulations as well as the PAD model simulations. With this in mind, the five-element mesh was selected as the most suitable mesh for this particular research for its abilities to capture the trends of the fluid flow suitably while avoiding the longer computational times associated with the seven-element mesh.

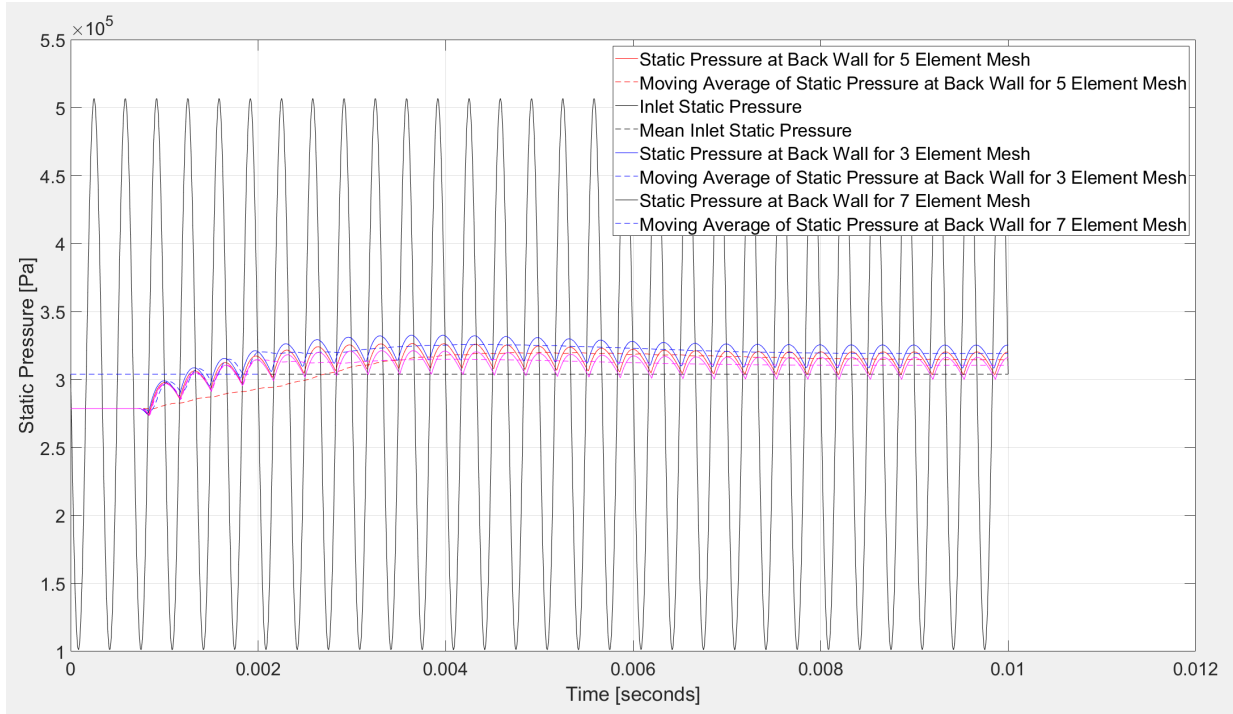


Figure 26. Back Wall Pressure of Each Mesh

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V. COMPUTATIONAL RESULTS AND ANALYSIS

A. SINGLE CTAP RESULTS

Aside from providing the basis for mesh refinement, the simulations of a single CTAP line ending in a simulated pressure transducer provide insights into the flow characteristics within CTAPs. Additionally, these simulations provide a point of comparison to the later PAD computational models. Single CTAP simulations were broken into two categories with the same boundary conditions, but with two different lengths of 0.25 meters and 1 meter, respectively.

1. 0.25 Meter Single CTAP Simulation Results

The 0.25-meter CTAP simulations illustrated an oscillatory behavior of the mean static pressure value. This oscillatory behavior effectively has an AC-component, as well as a mean value similar to a DC-shift. In theory, the CTAP geometry effectively attenuates the AC-component, while preserving the signal of the DC-shift. It makes sense to characterize the oscillatory nature of the measured signal in terms of the variability about the mean. In the 0.25m, 5-element case, the oscillatory variability is 3.42% from its mean value. This variability was calculated by taking the standard deviation of the static pressure oscillations and dividing by the mean value of that oscillation and is represented in Equation 2. This technique is also known as the coefficient of variation.

$$\text{Variation} = \text{SD}/P_{\text{Avg}} \quad (2)$$

This variability is meant to indicate the “tightness” around the mean value measured by a pressure transducer and indicates how significant the oscillations of measured pressure values are from timestep to timestep. This oscillation is illustrated in Figure 27, by the blue line of the 5-element mesh representative instantaneous averaged pressure. Where the flow time is shown on the x-axis and static pressure on the y-axis. This variation has the potential to produce significant effects downstream in a PAD as it could create more local fluctuations and perturbations in the flow that could affect the final value at any given transducer location.

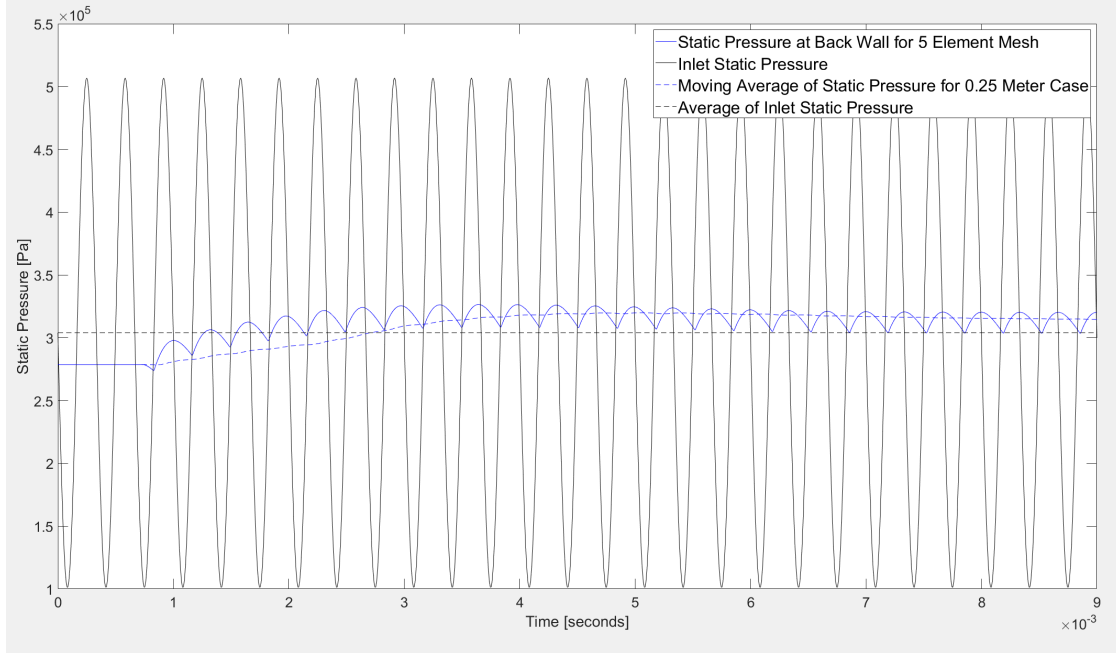


Figure 27. 0.25 Meter CTAP Static Pressure Measurements

Despite this variability, Figure 27 also shows a significant reduction in the pressure wave amplitude by the time it reaches the transducer location. This signifies that the CTAP line itself dampens the sinusoidal pressure input as it travels along its length. However, this begs the question of how a longer CTAP line will affect both the variability of the pressure signal at the transducer location as well as its AC-component attenuation effects.

2. One Meter Single CTAP Results

The one-meter single CTAP simulations were run both to answer the questions outlined previously as well as provide a simulation for a CTAP line closer to the lengths typically used in NPS labs. This one-meter CTAP simulation was set up with the exact same boundary conditions and mesh generation techniques used in the 0.25-meter CTAP simulations. Figure 28 displays both the results of the 0.25-meter CTAP as well as the one-meter CTAP for static pressure at the simulated transducer location. The one-meter case exhibited significantly less variability than the 0.25-meter case at 0.47%. The one-meter case also shows significantly more attenuation of the AC-component of the input pressure signal by the time the signal reaches the transducer face of the simulated CTAP. In

combination, this creates a “smoothing” effect of the input pressure signal. When connected to a PAD, this would result in less local fluctuations and perturbations in regions where a CTAP terminates in the settling reservoir/chamber. However, one drawback of using longer CTAP lines is the required pressurization time. This can also be seen in Figure 28 where the pressure signal requires an additional 2.18 ms to reach the transducer location on the CTAP. This must be considered in a designed experiment as adding pressurization time effects the amount of quality data that is able to be gathered. It should also be stated that the one-meter CTAP simulations have not yet appeared to have begun asymptoting toward a steady-state value. Further simulations were attempted to extend the flow time of the one-meter CTAP simulations; however, technical issues with the Hamming Supercomputer which these simulations were ran on prevented longer duration simulations from being achieved.

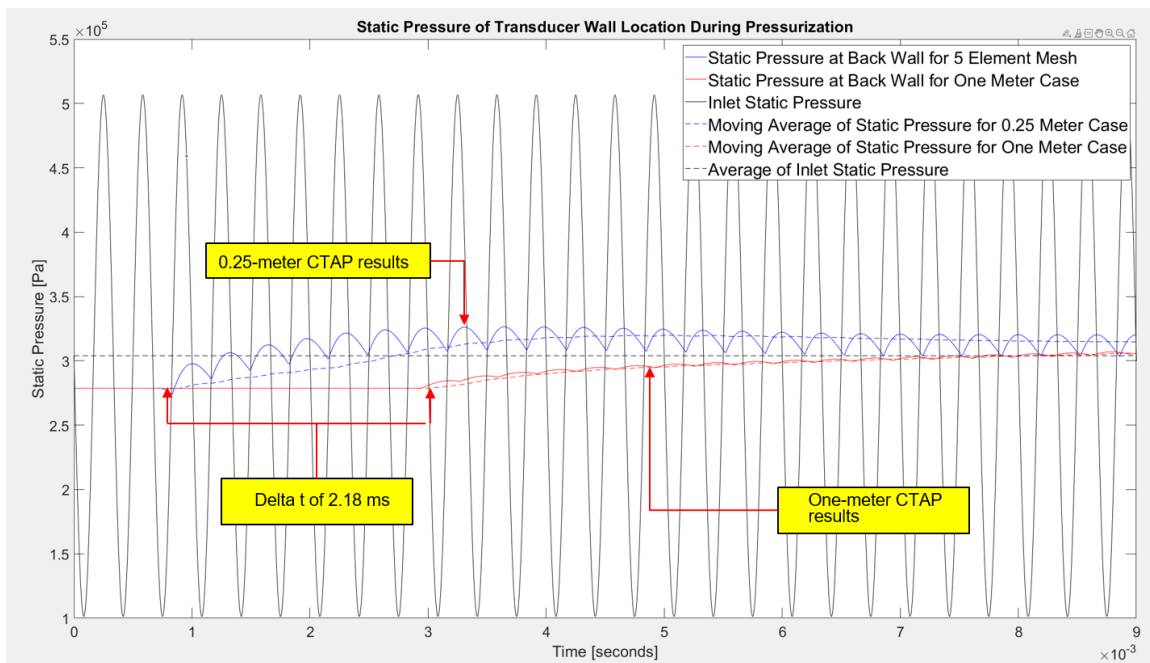


Figure 28. One Meter and 0.25 Meter Single CTAP Static Pressure Measurements

Despite the one-meter CTAP providing many benefits that would be useful in a PAD configuration, it was determined that the 0.25-meter CTAP would be used as the

pressure inlet sources to the simulated PAD. This was done to reduce computation time and complexity as adding more length increases the overall element count of the mesh and as a result increases the computational time required to run a simulation. Additionally, using the 0.25-meter CTAP provided a good test for the quality of the PAD design and its ability to spatially-and temporally-averaged pressure in sub-optimal conditions where the variability of a pressure input is relatively high. As such, if the PAD does an effective job at this condition, it should only be improved with a more realistic CTAP length.

B. SIMULATED PAD RESULTS

In modeling the PAD design in a computational environment, it was important to reduce the computational complexity where possible in order to reduce computational times required for each simulation. As previously stated, this was done by utilizing the 0.25-meter CTAP line as each of the inlets to the modeled PAD. Additionally, the computational complexity of the PAD simulation was reduced by reducing the number of inlets feeding the modeled PAD. As previously stated, the designed PAD has the capability to support 12 CTAP inlets; however, using 12 inlets in a computationally modeled PAD would prove to be too complex and require prohibitively large computation resources which were not available within the time frame of this body of thesis research. With this in mind, 4x CTAP inlet ports, spaced in 90-degree increments, were numerically modeled to feed the PAD settling reservoir. This number and configuration were selected as each CTAP port would be able to provide a varying pressure within each quadrant of the PAD design and was deemed to still provide a suitable proof of concept for the PAD design while keeping the computational complexity and times relatively low. Additionally, these simulation results deemed to be conservative in nature, as the real PAD, with more CTAP inlet ports, should theoretically experience a more uniform pressure distribution/fluidic averaging response. The volume of the pressure reservoir in the modeled PAD design was also reduced in comparison to the real-world PAD design from 0.972 cm³ to 0.639 cm³ to both reduce computational time as well as compensate for only using 4x CTAP inlets vice 12 in the real-world design. This change should also make result in a more conservative modelling result, as a larger settling volume is more likely to achieve a uniform pressure distribution. The computationally modeled PAD design can be seen in Figure 29, which

illustrates 4x CTAP inlets feeding into a modeled PAD serving as a common plenum or reservoir for each of the CTAP inlets.

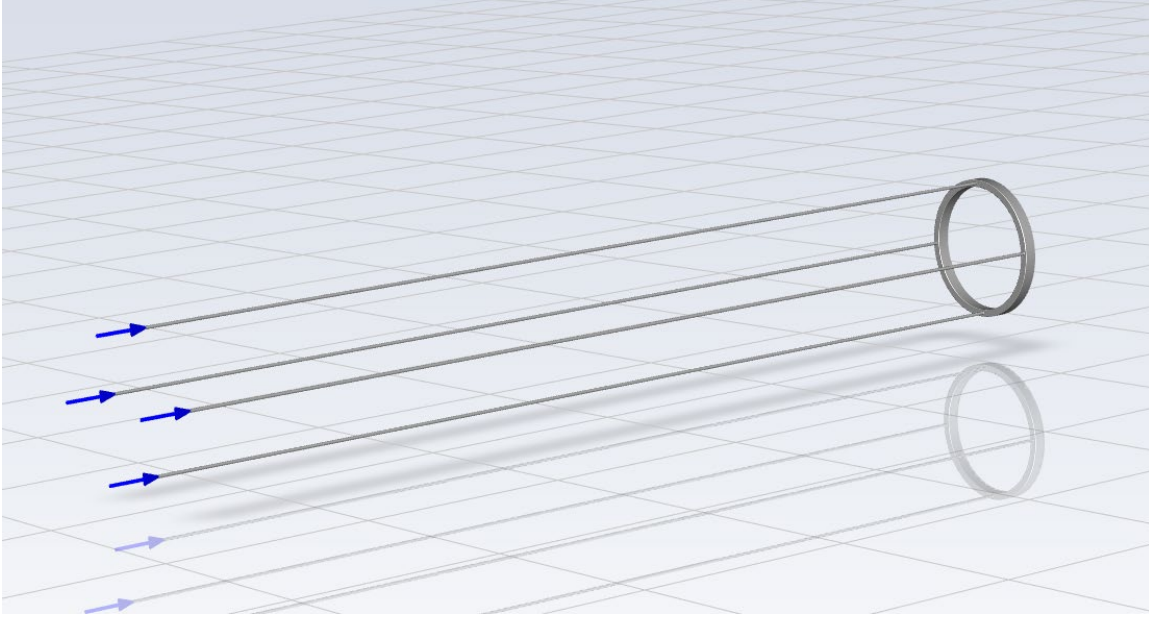


Figure 29. 4 CTAP Geometry and Configuration

The computationally modeled PAD was applied with the same basic set of boundary conditions utilized in the single CTAP case. The boundary conditions being an isothermal wall condition set to 300 K, a sinusoidal pressure input function across each of the four inlets with a temperature of 1750 K, and the fluid volume being initialized to 300 K. An adaptive time step was also used for PAD simulations in order to ensure that the flow characteristics were fully resolved without taking too small of a time step. As per the conducted mesh refinement study, the 5 elements across the inlet diameter mesh case were utilized for the PAD simulations with 5 elements across the inlet diameter for each CTAP inlet feeding into the modeled PAD. The resulting mesh contained 212,014 elements. The sinusoidal pressure input function was adjusted in phase and/or magnitude for each case of the PAD simulations, but the frequency of 3KHz remained the same for each simulation.

Four PAD simulation cases were modeled. The first case being a baseline result, with each sinusoidal input function being identical to one another in both phase and

amplitude to pressurize the PAD reservoir. This represents a case of longitudinal pulsing that can be seen in RDEs but is also a mode of operation seen in other unsteady combustors. The second case adjusted the sinusoidal pressure inputs across each inlet by changing their phase to simulate as if each CTAP inlet was being extracted from 90-degree varying locations about the circumference of the RDE at the same longitudinal plane. In this manner, each CTAP inlet in this case was adjusted with a 90-degree phase shift from one another. The third case utilized the same phase shift seen in case two, but also included a $\pm 15\%$ variation in the amplitude of each sinusoidal pressure input in order to simulate and test PAD performance when the pressure inputs have increased or reduced variability about the mean input forcing function value of 303.9 kPa (3 atm). The fourth case used the same phase shift of the sinusoidal inputs in case two, but also implemented a $\pm 15\%$ variation in the mean static pressure delivered by each sinusoidal pressure input in order to simulate and test PAD performance when the pressure inputs are varying in their mean value but the global average of 3 atm is conserved. A summary of the sinusoidal inputs for each case can be seen in Table 2 as well as a graphical representation of each pressure input condition and case in Figure 30.

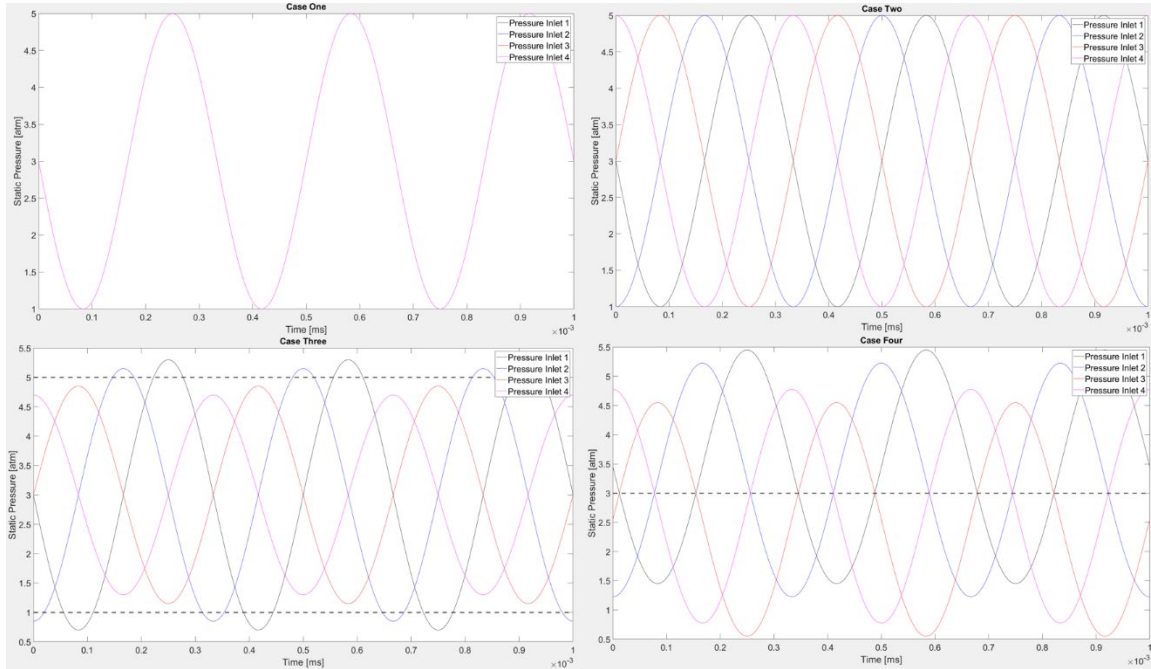


Figure 30. Graphical Representation of Each PAD Simulation Case and Inlet

Table 2. Summary of Sinusoidal Boundary Conditions for Each PAD
Simulation Case and Inlet

Inlet Number	Amplitude	Phase	Frequency	Mean Value
Case 1				
Inlet 1	2 atm	0°	3 KHz	3 atm
Inlet 2	2 atm	0°	3 KHz	3 atm
Inlet 3	2 atm	0°	3 KHz	3 atm
Inlet 4	2 atm	0°	3 KHz	3 atm
Case 2				
Inlet 1	2 atm	0°	3 KHz	3 atm
Inlet 2	2 atm	90°	3 KHz	3 atm
Inlet 3	2 atm	180°	3 KHz	3 atm
Inlet 4	2 atm	270°	3 KHz	3 atm
Case 3				
Inlet 1	2.3 atm	0°	3 KHz	3 atm
Inlet 2	2.15 atm	90°	3 KHz	3 atm
Inlet 3	1.85 atm	180°	3 KHz	3 atm
Inlet 4	1.7 atm	270°	3 KHz	3 atm
Case 4				
Inlet 1	2 atm	0°	3 KHz	3.45 atm
Inlet 2	2 atm	90°	3 KHz	3.225 atm
Inlet 3	2 atm	180°	3 KHz	2.55 atm
Inlet 4	2 atm	270°	3 KHz	2.775 atm

1. Case One PAD Simulation Results

The baseline PAD simulation, while being the most unrealistic simulation for a properly operating RDE, does provide a basis of understanding for a scenario in which each sinusoidal input function is identical. The pressurization of the modeled PAD can be seen in Figure 31 with the sinusoidal pressure waves traveling into the PAD simultaneously with the resulting static pressure distribution on the outlet face of the PAD where pressure transducers would be located.

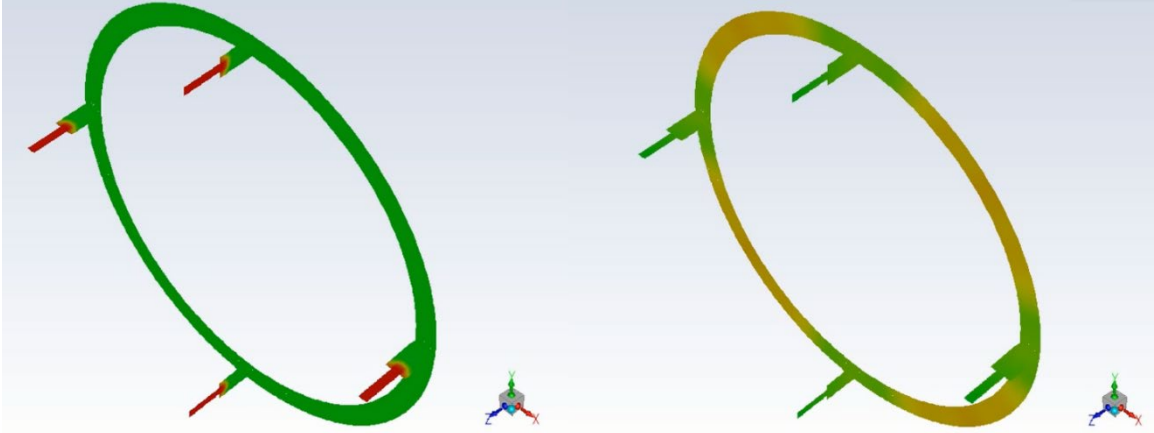


Figure 31. Case 1 PAD Pressurization

Because each sinusoidal pressure wave is in phase as it travels into the PAD, a pseudo-chugging effect is created for the static pressure on the outlet face. This pseudo-chugging effect is illustrated in Figure 32 where the global static pressure for the entire PAD device is averaged at each time step along with the corresponding minimum and maximum values of static pressure at that time step. The chugging effect creates a non-trivial amount of variability in the global static pressure measurement along the outlet face of the PAD; however, this variation is centered about a consistent mean value. This indicates that with proper temporal averaging done by pressure transducers along the outlet face of the PAD, a representative spatially-averaged static pressure value can likely be extracted.

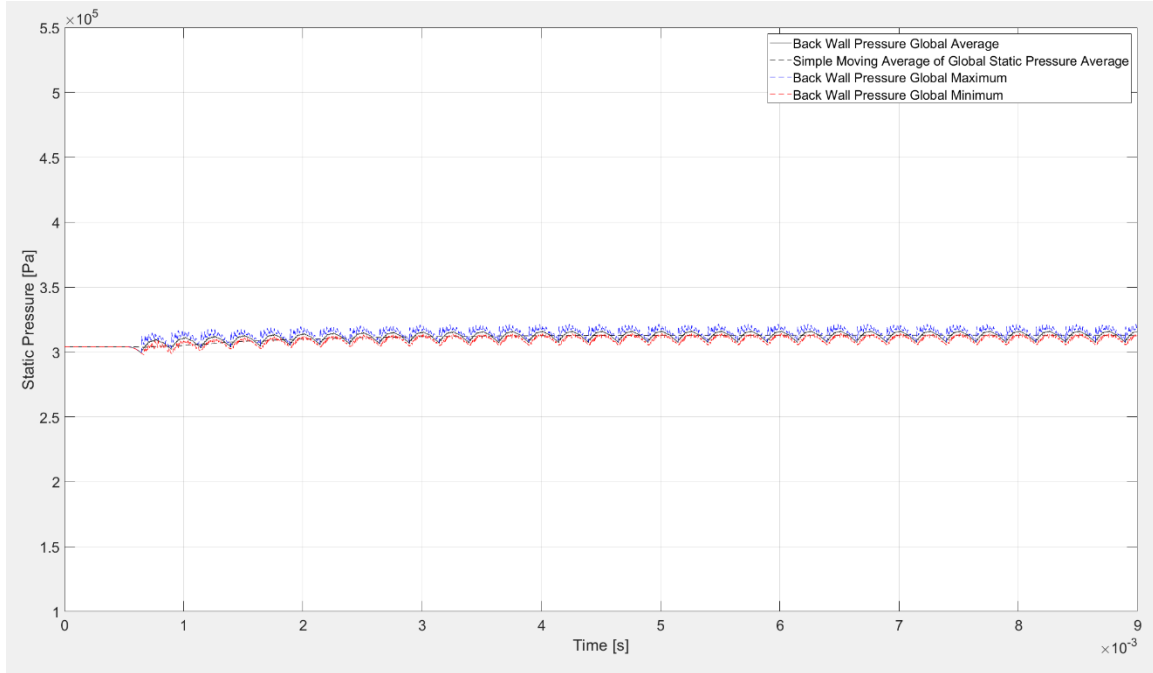


Figure 32. Static Pressure Results for Case 1 PAD Simulation

2. Case 2 PAD Simulation Results

In the case 2 simulations where a phase shift across each pressure inlet was introduced, the chugging effect seen in case 1 was nullified as the sinusoidal pressure inputs were no longer simultaneously pressurizing the PAD in the same manner, but rather their local maximum values reached the PAD 90 degrees out of phase with one another. This created circular movement of the global maximum average as each pressure wave entered the modeled PAD. This behavior can be seen in Figure 33 with each pressure wave moving into the PAD from its corresponding inlet.

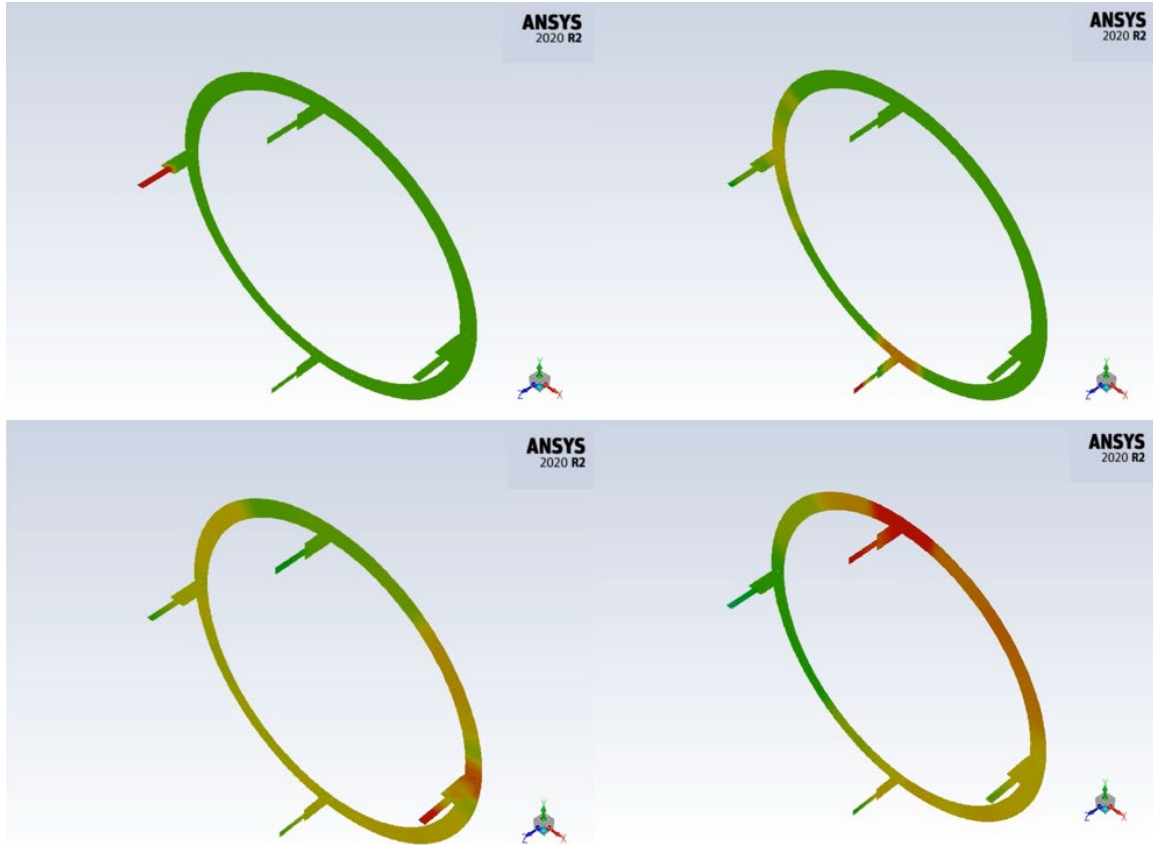


Figure 33. Case 2 PAD Pressurization

Despite local perturbations of minimum and maximum static pressure values, the global average of static pressure for case 2 remained consistent, without the variability seen in case 1. This behavior is illustrated in Figure 34 where despite those local perturbations the global average pressure is still conserved. This too would indicate that the PAD model can achieve a measurement of representative spatially-averaged static pressures when considered globally in the case of pressurizing the PAD with identical pressure waves at 90-degree offsets around the circumference of an RDE. It is worth noting that the mean average pressure value is decreasing by approximately 56.97 Pa per millisecond for the last two milliseconds of flow time for this simulation. This would indicate that the simulation has yet to achieve steady state and that, for a simulation with longer flow times, the fluctuations about the mean static pressure value would be likely to improve. This behavior was seen for all PAD simulations that were a part of this research.

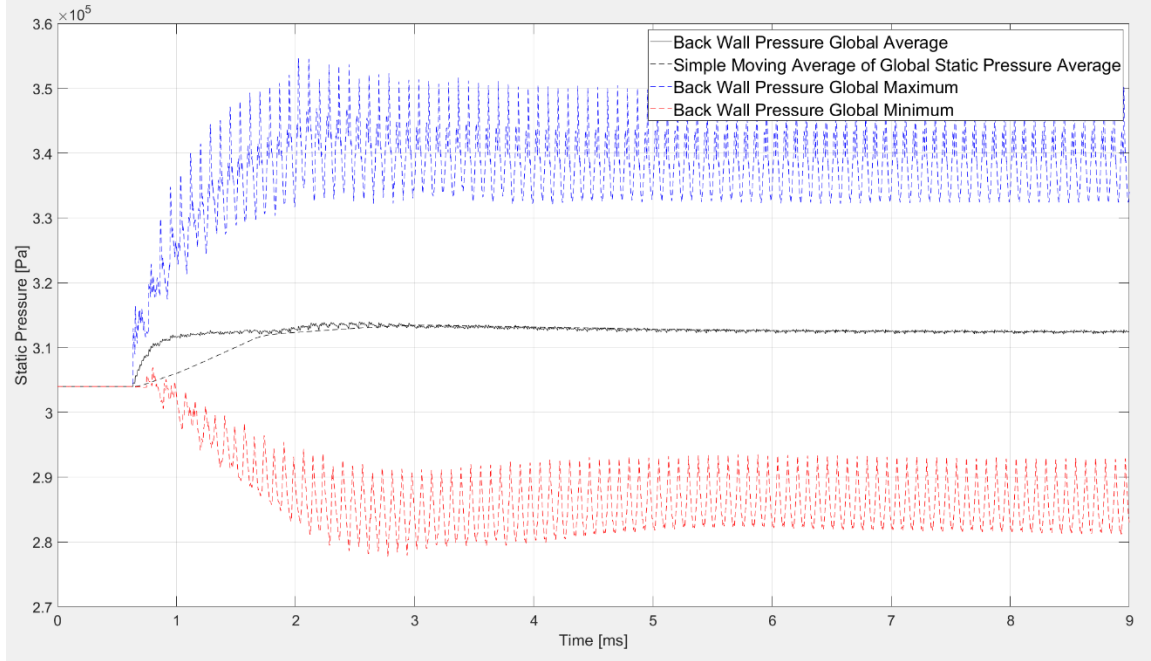


Figure 34. Static Pressure Results for Case 2 PAD Simulation

As previously stated, the global spatial average of static pressure at the outlet face of the modeled PAD has minimal variability about its mean value especially compared to that of Case 1. It can be reasoned that with temporal averaging done by pressure transducers at this outlet face of the PAD (I.e., by installing one or more CTAPs), the variability would be reduced even further, resulting in a clear spatially- and temporally-averaged static pressure value which would be present in the PAD, and representative of the spatially- and temporally-averaged pressure distribution within an unsteady combustor.

3. Case 3 PAD Simulation Results

Case 3 for the PAD simulations maintained the same phase shift as seen in case 2, but also introduced a $\pm 15\%$ variation in the sinusoidal pressure input amplitude. It is important to note that the selection for which inlets would receive what level of amplitude variation was carefully determined to purposefully create a bias. This was done so that the distribution of pressure wave amplitudes would be known and have a clear distinction so that the effects can be more carefully analyzed.

The pressurization sequence for Case 3 looks virtually identical to that of Case 2, with the only difference being globally higher and lower maximum and minimum pressures compared to Case 2. This is due to the amplitude variation that was introduced with Case 3. Despite larger fluctuations in the global maximum and minimum values of static pressure, the average static pressure at the outlet face of the modeled PAD maintained a consistent value of globally average static pressure. This behavior is displayed in Figure 35. Despite more unsteady minimum and maximum global pressure values than what were observed in Case 2, the global average static pressure at the outlet face of the PAD remained relatively constant with low variability about its mean value. When temporally-averaged via a CTAP-type pressure transducer on pressure wall, it would be expected that an appropriate representative static pressure value could be measured from a system with a mode of operation similar to that of Case 3.

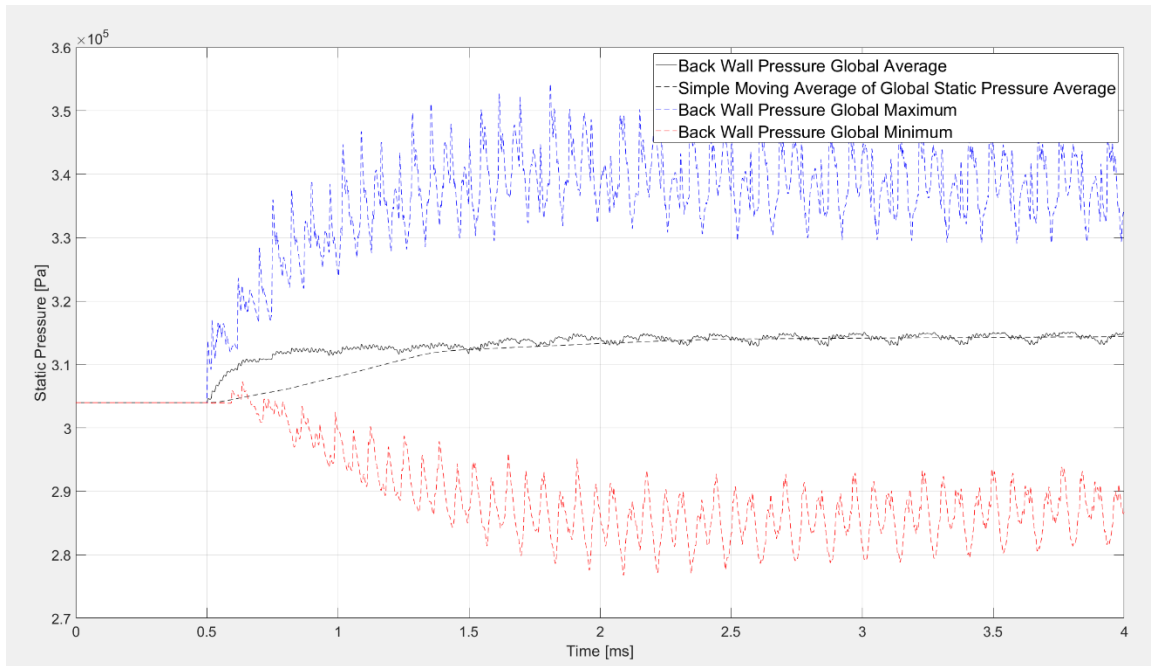


Figure 35. Static Pressure Results for Case 3 PAD Simulation

4. Case 4 PAD Simulation Results

Case 4 of the PAD simulations was meant as a test for PAD performance if the local mean for each sinusoidal pressure input varied by $\pm 15\%$, but the global mean average was conserved at 3 atm. Case 4 was modeled by using Case 2 to effectively pre-condition the flow field to reduce the overall computation time required. Case 2 and Case 4 have the same out-of-phase behavior, but different mean pressure amplitudes on the inlet boundary condition. After Case 2 had been run for 5 ms of flow time, the inlet boundary conditions were changed to match those of Case 4. In this way, computational time could be reduced by means of the modeled PAD being pre-pressurized by Case 2, and the PAD could be examined in the case of changing operational modes of an RDE, which was simulated here by transitioning from Case 2 to Case 4 after 5 ms of fluidic flow time. The resulting global average static pressure of the outlet face of the modeled PAD plot can be seen in Figure 29. The mean shifted perturbations introduced in Case 4 do cause the global static pressure average to exhibit more variability about its mean than seen in Cases 2 and 3, while the average value of the mean pressures at each time step increases steadily. This steady increase is most likely representative of the system not reaching a steady-state by this time. Figure 36 also illustrates the transitional period between two operational modes of an RDE with the lag between Case 2 and Case 4 behavior being consistent with the amount of time required for the corresponding pressure wave to travel down the 0.25-meter CTAP seen previously.

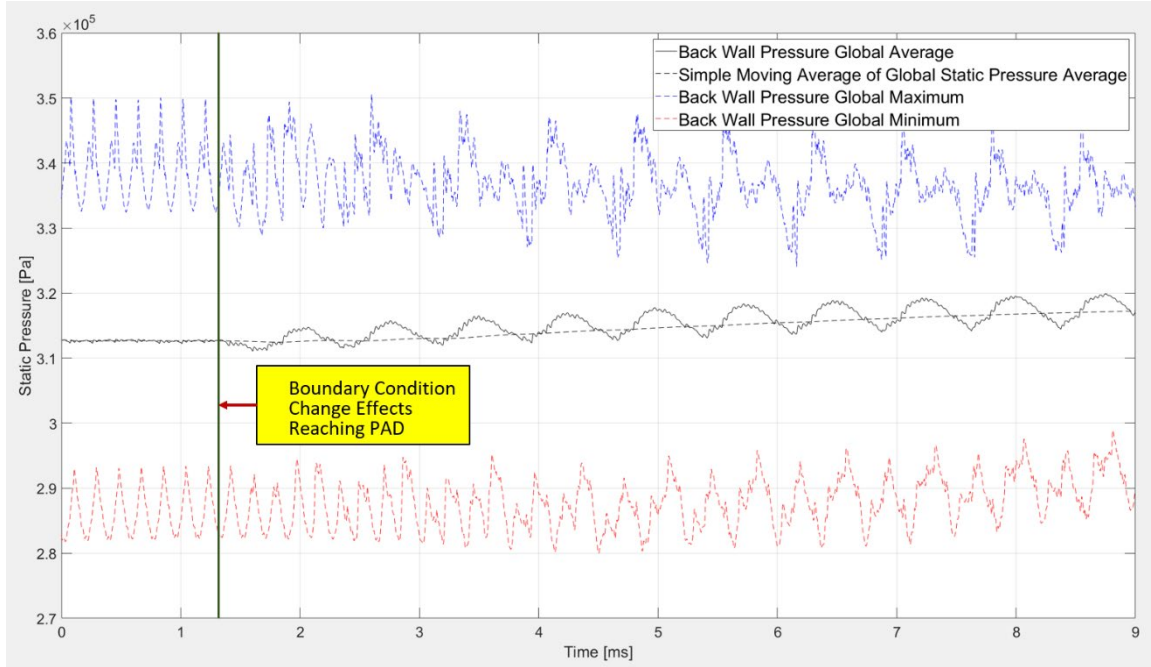


Figure 36. Static Pressure Results for Case 4 PAD Simulations

While these simulations do provide a good indicator that the modeled PAD design will be able to extract a good representative value of static pressure globally, they do not indicate what an individual pressure transducer would measure for any given location along the outlet face of the modeled PAD. The previously shown data for the PAD simulations only outlines global trends and does not isolate individual transducer locations where the PAD spatially-averaged pressure becomes temporally-averaged.

C. MODELING INDIVIDUAL TRANSDUCER LOCATIONS FOR PAD SIMULATIONS

In order to represent the effective ability of the PAD to spatially-average the unsteady pressure inputs, an area-weighted average is measured at multiple azimuthal locations about the pressure wall of the PAD. Initially, three discrete transducer locations were selected as simulated transducers. One being directly in line with a pressure CTAP inlet, a second spaced perfectly between two pressure CTAP inlets, and a third spaced between the two previously described CTAP inlets. The average static pressure within each transducer area, defined as 0.03-inch radius circle were then calculated at each time step

and plotted over time. Figure 37 displays the location of the three simulated transducers. This test was performed for the simulation in which Case 2 transitioned to Case 4 at a flow time of 5 ms as described previously. The resulting plots of average pressure within the discrete transducer locations can be seen in Figure 38.



Figure 37. Simulated Transducer Location

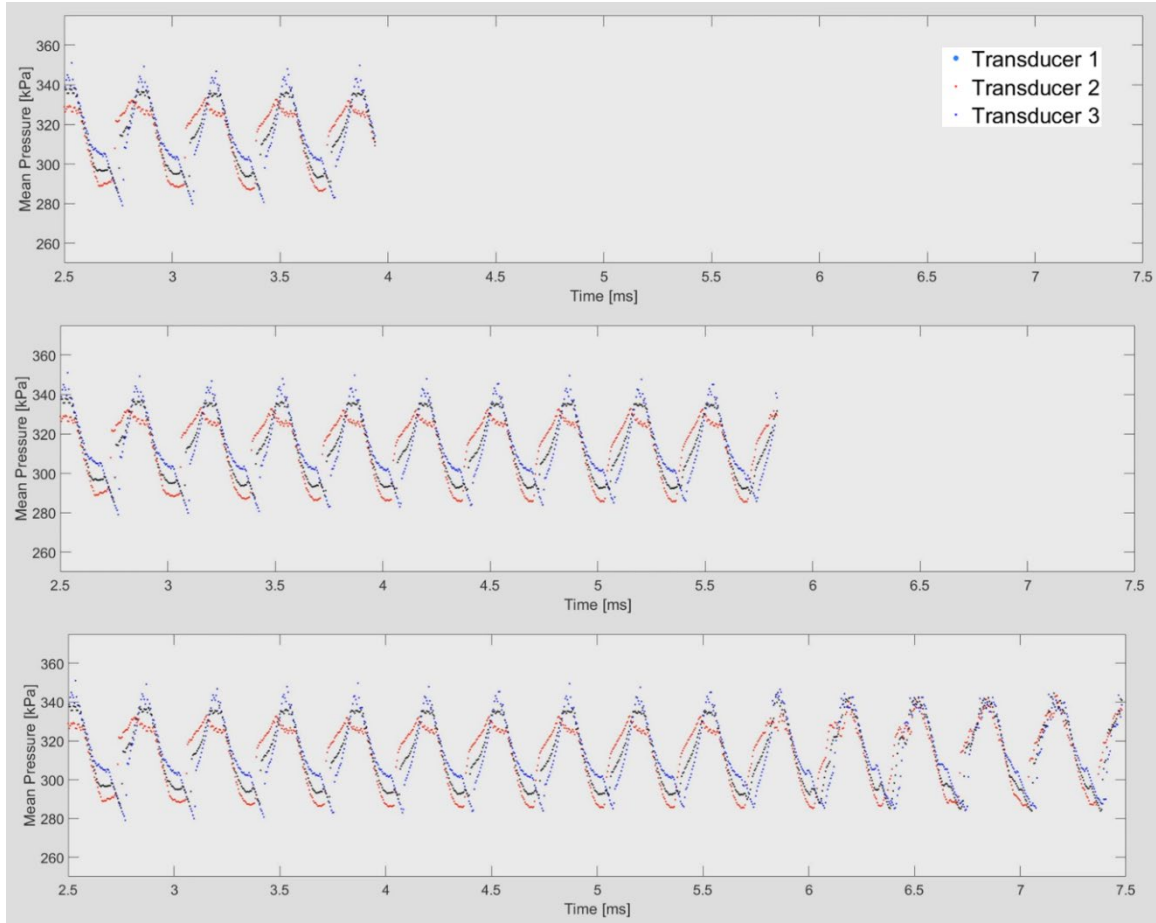


Figure 38. Average Static Pressure for Transducer Locations

Despite the global fluctuations in pressure seen in both Cases 2 and 4, the global average of static pressure is conserved both globally and locally for the three simulated transducer locations. Local maximum and minimum values adjust depending on transducer proximity to a pressure inlet, but nonetheless the mean static pressure of the system is still conserved even through the transition from Case 2 to Case 4. This indicates that transducer location does not affect the spatially and temporally averaged pressure extracted from the system. To investigate this further, twenty random transducer locations were selected, and their time averaged pressure were examined for each of the four test cases. In doing so, a representative spatially-averaged pressure delivered by the modeled PAD would then be temporally averaged at twenty random transducer locations to evaluate the effect of transducer location on the measured value of static pressure.

Twenty random points were selected by a random number generator in MATLAB that corresponded to twenty random transducer locations. The transducer angular locations (theta) as well as the representative spatially- and temporally-averaged value of static pressure were then plotted. Figure 39 illustrates the locations of the randomly selected transducers on the outlet face of the modeled PAD design while Figure 40 illustrates the corresponding theta locations for each simulated transducer and its accompanying time averaged static pressure value through the duration of each simulation case for the modeled PAD.

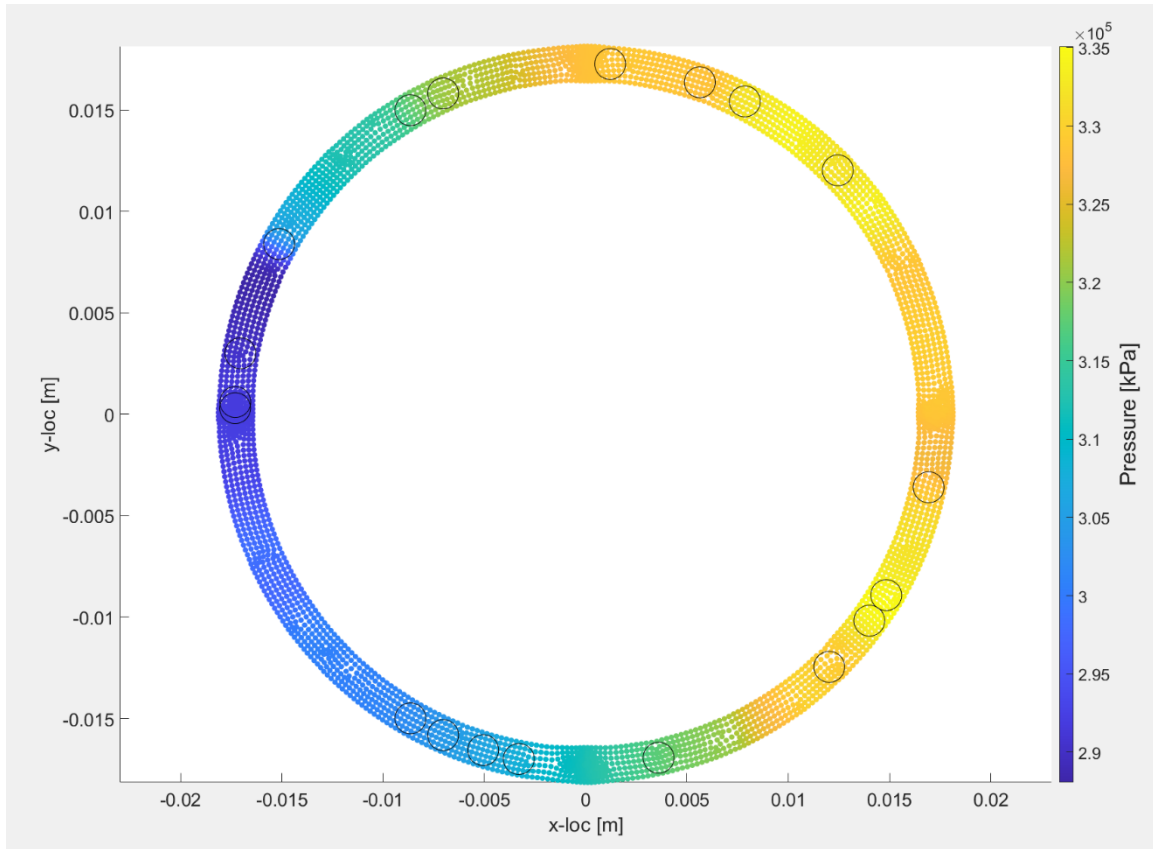


Figure 39. Random Transducer Locations

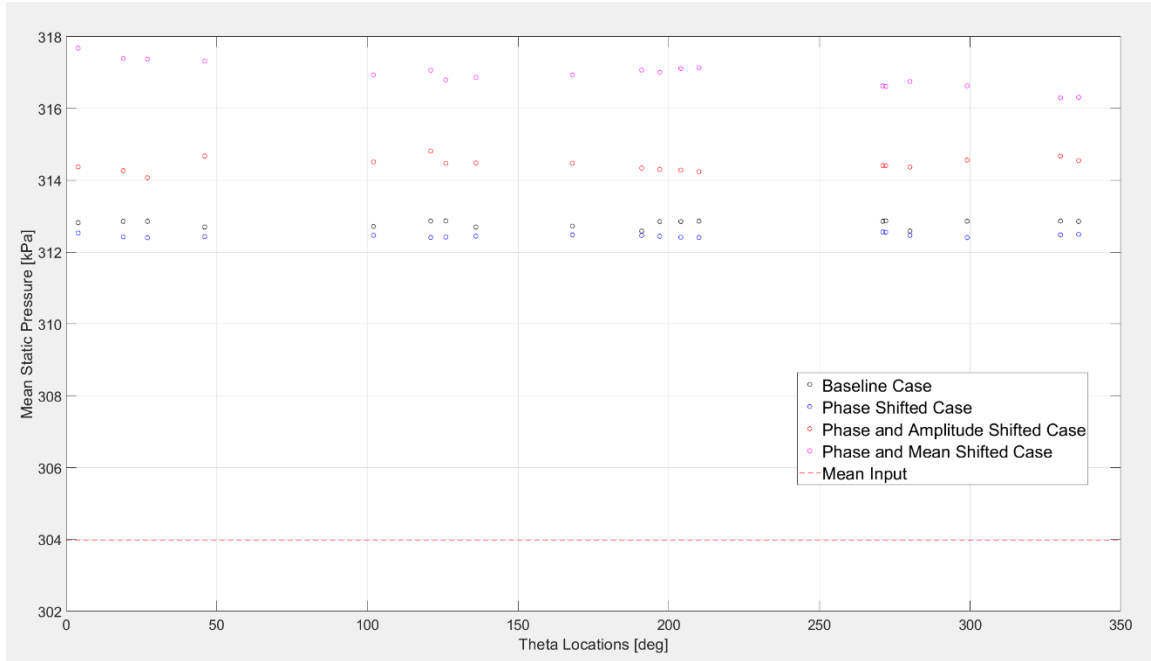


Figure 40. Transducer Locations and Their Accompanying Static Pressure Value for all PAD Simulation Cases

Figure 40 shows that even for a random selection of transducer locations along the outlet face of the modeled PAD the global static pressure is conserved. This is illustrated by the theta locations of each transducer having a minimal effect on its corresponding time averaged pressure value. The maximum variability in time averaged static pressure between two transducer locations for the same case was determined to be $\pm 0.23\%$. This demonstrates that the modeled PAD design is able to spatially average the pressure in such a way that when the static pressure is temporally-averaged by a transducer, the location of that transducer will not affect the final value of that measurement.

Interestingly, the simulation case for the modeled PAD did influence the measured mean static pressure values. All tested cases show static pressure measurements greater than the mean inlet static pressure value of 3 atm. There are three major predictions for this phenomenon. The first being that this system is underdamped. The single CTAP simulation results showed a non-trivial overshoot of the mean inlet static pressure before asymptoting to their final value. It is very possible and highly likely that the same behavior is occurring within the PAD simulations. Another reason for this behavior is the simulation having not

yet achieved steady-state conditions and the final value that would correspond to static pressure along the outlet face of the PAD has not yet been achieved. Lastly, in the mesh refinement study it was observed that the finer meshes tended to asymptote closer to the mean inlet static pressure value and given that the mesh used in this simulation was not the finest mesh achievable, the measured values of static pressure in the modeled PAD would share a similar behavior to that of the single CTAPs where the measured static pressure values were higher than that of the mean inlet static pressures. As for why the simulation case influences the final value of measured static pressure within the PAD, it is noteworthy that the cases that exhibited an increase in mean static pressure were the cases that introduced higher local values of static pressure by either means of amplitude fluctuations or mean static pressure fluctuations. Being that the system has yet to achieve steady state in any simulation, lasting effects of these higher local values of static pressure are likely biasing the system to higher values of static pressure globally.

Regardless of perturbances from the inlet mean static pressure value, the behavior of the PAD simulation across each tested case indicates that the modeled PAD provides a representative static pressure value of the RDE combustion chamber by means of spatially averaging in the PAD reservoir and temporal averaging by means of pressure transducers. Additionally, it has been shown through the results of these simulations that the location of the pressure transducer on the PAD does not appreciably influence the final representative value of average static pressure present in the system.

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VI. CONCLUSIONS

A. RESEARCH ANALYSIS CONCLUSIONS

The results of this research provide succinct conclusions about the designed PAD as well as demonstrates a path forwards for future researchers in the field. The modeled PAD designed indicated that when pressurized by sinusoidal input functions with frequencies representative of RDE operation, the device is capable of providing a spatially- and temporally-averaged pressure which is indicative of the conditions across a particular plane within the RDE combustion chamber. It is noteworthy to state that the variation in mean static pressure delivered by 0.25-meter CTAPs would be drastically reduced by using one-meter CTAPs in their place. This would likely reduce the variability seen within the PAD even further at the cost of requiring longer pressurization times. The modeled PAD demonstrated that not only is the mean static pressure preserved globally despite local perturbations but is conserved locally as well in that varying the location of the azimuthal pressure transducer location will not affect the measured value of static pressure. It was found that for transducer locations spaced between pressure inlets to the pressure reservoir the difference between maximum and minimum recorded values of static was less than that for transducer locations located in line with or very near the pressure inlets. This is a factor to consider when utilizing a PAD as less variability in instantaneous static pressure will only help to deliver a more succinct value of time-averaged static pressure. This is an excellent indicator that the PAD is appropriately spatially-averaging the static pressure being input to the device in such a way that the local perturbations of the individual inlets do not affect the global and local values of time-averaged static pressure within the PAD. Altogether, this indicates that the actual PAD design based on the physics of the modeled PAD, will be able to provide a representative value of spatially and temporally averaged pressure.

The PAD was designed to work in conjunction with the designed outer body to test its capabilities in a variety of conditions. With CTAP ports located along a wide variety of azimuthal locations around the combustion chamber, PAD-averaged CTAP measurements will be able to be compared directly against conventional CTAP measurements to provide

a comparison on the two methods of averaging static pressure within an RDE combustion chamber.

B. FUTURE WORK

The research conducted in the body of this thesis have laid the groundwork for future lab experiments as well as future computational modeling efforts for additional inlet boundary condition forcing functions. A major question raised by this research is the effect of the mesh refinement on the outcome of the static pressure measurements in a CFD environment. This research showed that a more refined mesh tended to bring the time averaged value of static pressure closer to its appropriate mean input value, but due to time and resource restrictions, a lesser-refined mesh built around capturing flow behavior and trends was utilized. A more refined mesh should be used in the future in a similar battery of simulations to appropriately characterize the PAD's tendency to correctly average the input pressure values. Additionally, a mesh which is more refined around the inlet and pressure reservoir sections of the model would be particularly useful for this reason, as well as potential time and resource utilization benefits. It is also desirable to simulate the modeled PAD with more pressure inputs as would be the case in a laboratory experiment. Sinusoidal pressure inputs were utilized in this experiment as an appropriate approximation of the inputs that would be delivered by an RDE; however, other input functions that more closely model RDE operation should be tested such as a delta function with an exponential decay. It was also realized near the conclusion of this work that it would be interesting to record the output of the CTAP tube into both the CTAP reservoir and PAD to investigate whether a single CTAP output can be used as the input to a PAD reservoir, while still maintaining an acceptable representation of the major physics in lieu of modeling an entire CTAP tube feeding a PAD reservoir. If this is possible, then it would vastly reduce the computational requirements to rapidly iterate over PAD designs with various inlet boundary conditions (mean pressure, amplitude, waveform, tube length, etc.).

CFD analysis of the PAD proved to be extremely promising, but the PAD itself needs to be tested in the lab in conjunction with the new outer body to truly verify the results of this research. Additionally, the PAD design could be refined further to be placed

directly on the outer body of the RDE instead of an external volume as is the case with the PAD design proposed in this research. The PAD design should be tested for a variety of flow conditions and RDE operating modes to verify its functionality. Regular single CTAP measurements should be made along with the PAD measurements to provide a point of comparison between PAD delivered results as well as traditional CTAP results done in past experiments.

C. FINAL REMARKS

This research has proven that a PAD design shows great promise in delivering a spatially- and temporally-averaged static pressure value which is representative of the conditions present within an RDE combustion chamber or another unsteady combustor. This will help to more succinctly measure the performance of an RDE and determine if pressure gain combustion has truly been achieved. Additionally, a computational method and process has been developed by this research that future researchers will be able to utilize and refine to test PAD-like devices and make design choices before moving into a laboratory environment. This computational resource has proven to be efficient and work over a variety of boundary conditions that are suitable to simulate RDE operation. The PAD design delivered by this research shows extreme promise in improving the RDE community's ability to measure the performance of their RDE setups by being able to correctly determine a representative spatially and temporally averaged static pressure present within their RDE combustion chambers.

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APPENDIX A. RESEARCH ON PERTURBED FUEL INJECTION EFFECTS

A. LOCAL PRESSURE AUGMENTATION VIA EQUIVALENCE RATIO

Another source of local non-uniformities are augmentations of the local pressure due to local perturbances in equivalence ratio. Equivalence ratio dictates the amount of bulk heat released into the system during a combustion process [12]. Given a non-uniform distribution in equivalence ratio, it can be assumed that there would be a non-uniform distribution of bulk heat release present in the system. This has the potential to drastically augment local performance metrics such as pressure within an RDE. For example, experimental results for percent pressure gain versus equivalence ratio can be seen in Figure 41.

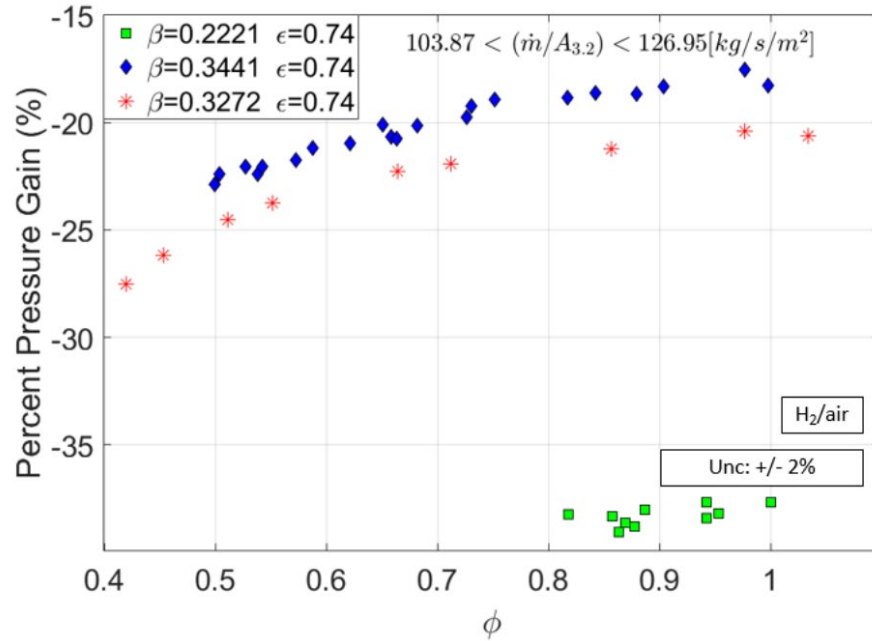


Figure 41. Percent Pressure Gain Versus Equivalence Ratio. Source: [16].

It can be seen that the percent pressure gain increases with an increasing equivalence ratio before asymptoting to a given value. If the experimental dataset were

extended to more fuel-rich equivalence ratios, the performance would be seen to peak, before gradually decreasing at more fuel-rich conditions. The local effects on pressure via perturbations in local equivalence ratio can be seen in Table 3 where P_j represents the local equivalence ratio over a 180-degree swath of a simulated RDE and $P_{a, h}$ and $P_{a, l}$ representing the local pressures of the high- and low-pressure regions, respectively. Table 1 illustrates that local perturbations of equivalence ratio have local effects on pressure creating a non-uniform pressure distribution within the RDE which has lasting effects on the overall performance of the RDE signified by the percent thrust vectored, indicated by T_v .

Table 3. Effects of Local Perturbations of Equivalence Ratio. Source: [9].

Condition	v (m/s)	$m_{a,h}$ ($\text{kg}\cdot\text{s}^{-1}\cdot\text{m}^{-2}$)	$m_{a,l}$ ($\text{kg}\cdot\text{s}^{-1}\cdot\text{m}^{-2}$)	$m_{f,v}$	$p_{a,h}$ (atm)	$p_{a,l}$ (atm)	T_v
Baseline	1891.7	214.5	214.5		2.9371	2.9371	
$P_j=1.5$	1891.6	339.7610	175.2781	93.84%	3.9950	3.1297	27.65%
$P_j=2.0$	1870.3	458.4608	136.3403	236.26%	5.1040	3.3419	52.73%
$P_j=2.5$	1829.1	587.5472	98.4764	496.64%	6.1895	3.5930	72.27%
$P_j=3.0$	1797.9	720.6604	66.4858	983.93%	7.2467	3.9389	83.98%

B. AZIMUTHAL VARIABILITY IN STATIC PRESSURE MEASUREMENTS

NPS RDEs operate in a non-premixed manner, meaning that they can be susceptible to the locally varying cell size and thus locally varying performance perturbations previously discussed. These perturbations were first detected via Capillary Tube Average Pressure (CTAP) measurements, via a process which will be discussed in more detail later. These CTAP measurements showed varying values of static pressure at different azimuthal locations along the RDE combustion chamber [16].

When the variability of static pressure measurements was first detected, there were several potential causes that were subsequently investigated. Some of the potential causes for this variability were poor sensor calibration, poor sensor installation, node/anti-node behavior from unique RDE combustion modes (co-rotating, counter-rotating, slapping, etc.), as well as jetting within the airstream caused by either facility or engine non-uniformities. During multiple experimental studies at NPS using a “CTAP ring,” which

allowed for 9 static pressures at multiple azimuthal locations and a single axial plane, some of the previously mentioned possibilities were ruled out. Namely, careful calibration of the pressure transducers, swapping pressure sensors, re-installation of static pressure ports, repeatability, and operation in multiple RDE modes, eliminated the first three possibilities. While air/fuel fluid jetting was not entirely ruled out, experiments on a new RDE with a new facility connection, which resulted in similar asymmetry observations, indicated that fluid/momentum jetting were unlikely. These potential causes were all systematically investigated in two separate RDEs owned by NPS and found not to be the probable cause for the measured variability [15].

Once these hypotheses disprove, non-uniformities were discovered in the fuel injection holes of the fuel injector. These non-uniformities were caused by additive manufacturing errors and limitations that created drastically different hole shapes and sizes around the circumference of the fuel injector. These non-uniformities can be seen in Figure 42 [16].



Figure 42. Fuel Injector Non-Uniformities and Failure Points. Source: [16].

Additionally, at some point during the test campaign in which these non-uniformities were discovered, the fuel injector encountered a failure mode, which can also be seen in Figure 42. Both the hole non-uniformities as well as the failure mode of the injector itself, can create a large amount of non-uniformity in the distribution of fuel in the combustion chamber. This can lead to some azimuthal regions of the RDE being more favorably fueled than others which in turn creates different local equivalence ratios which

differ from the global equivalence ratio. This affects the strength of the detonation wave and thusly affects the downstream total pressure in the combustion chamber if insufficient axial length exists to effectively “mix-out” the flowfield [2].

Assuming that the fuel injector failure was indeed the cause for the spike in pressure measurement variability and negating this failure, a variability of 5–9% was still measured within these total pressure measurements [16]. This variability is still undesirable as it detracts from a truly accurate representation of the total pressure within the combustion chamber. From this, a designed PAD must be able to virtually extinguish this variability so that the total pressure within the combustion chamber can be measured accurately either with or without any sort of failure within the RDE.

Previous research has noted a similar spatially and temporally varying static pressure though both experimental results and computational fluid dynamic (CFD) simulations. These results and the results demonstrated at NPS indicate that at least one potential cause of this variability is due to a persistent non-uniform fuel distribution, which, in the case of NPS, is due to additive manufacturing errors with the fuel injector as described previously [16]. This leading hypothesis indicates that there is a persistence of locally fuel rich and locally fuel lean equivalence ratios which give rise to the local non-uniformities discussed previously. Knowing this, as well as the general goal of this thesis work, another opportunity presented itself to attempt to characterize this spatial non-uniformity by purposefully introducing a known asymmetry via an asymmetrical fuel injector. This provides two major benefits. The first being the opportunity to better understand the cause of azimuthally asymmetrical static pressure measurements which would lead to a better understanding of the instrumentation requirements for static pressure measurement ports. The second benefit being a unique opportunity to validate the performance of a proposed pressure averaging device by evaluating how well it averages the pressure of an intentionally biased RDE and whether it provides a static pressure that is truly representative of the conditions within the RDE combustion chamber. Additionally, intentionally perturbing the local equivalence ratio of an RDE will provide an opportunity to evaluate the effects of such a perturbation and how it affects overall RDE performance metrics such as thrust.

C. INVESTIGATIONS INTO PRODUCING AN INTENTIONALLY PERTURBED GEOMETRY

The underlying premise of this research is to put forth an experiment that will test the hypothesis that flow asymmetries present in NPS RDEs are created by a non-uniform fuel distribution as well as whether an intentionally created asymmetry could result in thrust vectoring. In order to do this, the experiment must be designed in such a way that an intentionally perturbed fuel injector can be properly compared to an unperturbed, or baseline case, fuel injector. Moreover, the results of the experiment must be readable and discernable from each other using the current CTAP or NPS Method of acquiring the static pressure within the combustion chamber. The operability range of NPS RDEs must also be taken under consideration in the design of the experiment. Three separate staggered orifice ring injectors (SORIs) were designed with these requirements and limitations in mind. One newly designed fuel injector represents the baseline case with no intentionally designed perturbations, while the other two fuel injectors contain intentionally created nonuniformities so that they intentionally create asymmetrical flow through the RDE combustion chamber. Each new fuel injector body will be additively manufactured, while the holes of the injector will be traditionally machined in order to remove the flaws seen in Figure 42. The baseline case was designed to be consistent with previous fuel injector designs in order to provide a good case of comparison between the results of this proposed experiment and previous experimental results. The baseline SORI contains one hundred total fuel injection holes that are positioned perpendicular to the flow of air into the combustion chamber creating a jet-in cross flow. This jet-in cross flow method of fuel injection is favorable in RDEs because of the need to maintain a good air-fuel mixture moving into the combustion chamber in order to maintain detonation [5]. The hole of each fuel injection hole of the SORI is exactly 0.15875 cm (0.0625 in) in diameter. This ensures that there will be an equal mass flow rate of fuel coming out of each hole of the SORI. With this, the fuel should be equally distributed across all azimuthal angles of the combustion chamber. Designing the perturbed case SORI geometry required extensive analysis and consideration of several factors. Some of these factors included ensuring that the selected geometry is capable of producing equivalence ratios within NPS RDE's operational ranges, producing measurable variability in azimuthal static pressure in the

combustion chamber, as well as manufacturing limitations. The first factor to consider was how to keep the global equivalence ratio at its desired value while increasing the hole size across a certain region of the SORI. It was determined at the best way to accomplish this was to decrease the hole size across the rest of the injector in order to maintain the global equivalence ratio even while perturbing the local equivalence ratio across the large hole region and small hole region. The next step in this analysis process was to determine what the hole size needs to be. However, before this could be done, there was another factor which required consideration. Namely, the size of the large hole perturbed region of the SORI needs to be considered. This could not be predetermined, so several cases were designed in which the size of the overall arc sweep of perturbed region was changing. The cases analyzed in this research were a 180-degree, 90-degree, 45-degree, 33.3-degree, and 15-degree arc sweep of large hole perturbed regions of the SORI fuel injector. The next step in the process was to determine the small hole size for the SORI. It was determined that the best way to set the small hole size was to fix the small hole size at specific values. In this way the mass flow rate of fuel going through the small holes could be determined and help to drive the large hole size to a required value to achieve a given global equivalence ratio. With this, several different cases of small hole size were considered in order to examine which small hole size would be most conducive to driving a particular perturbed case geometry that would satisfy the considerations and requirements outlined previously. The hole sizes placed under consideration were 0.127 cm (0.050 in), 0.1397 cm (0.055 in), and 0.1524 cm (0.060 in) in diameter.

D. PERTURBED CASE FUEL INJECTOR GEOMETRY SELECTION

In order to select an appropriate hole geometry, all the requirements must be considered and consequently satisfied. One major requirement is for the resulting local equivalence ratios to be within the operational range of NPS RDEs. This range, through past experiments, has been found to be approximately between 0.6 and 2.0 [11]. Another factor to consider is the difference between the local equivalence ratios in each given case. If the change between the local large and local small equivalence ratio is too large, it is assumed that the RDE will not operate properly. Additionally, a very important

requirement is that the perturbed geometry must be able to generate a measurable change in the pressure downstream at the exit across each different local perturbation.

An investigation into the effects of each case in question was conducted with the results shown in Figure 44 which shows that a good candidate for best hole size across all three arc sweep cases presented lies with the 0.1524 cm (0.06 in) diameter hole size.

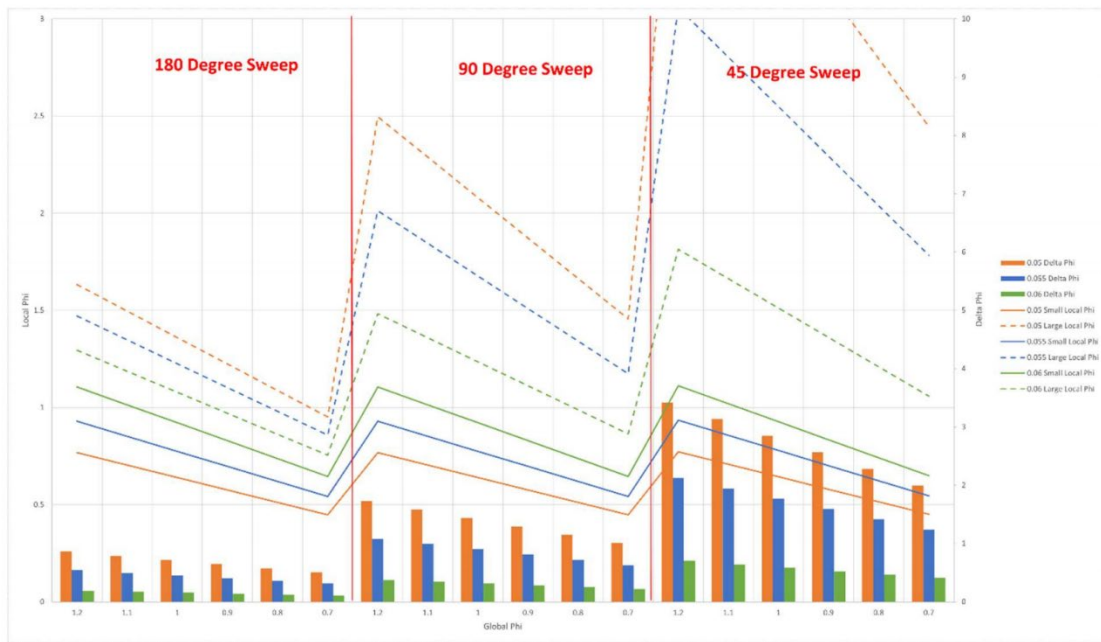


Figure 43. Plot of Local Equivalence Ratios and Their Differences According to Small Hole Size and Arc Sweep

This hole size corresponds to the lowest value local equivalence ratio for the large holes as well as the largest value local equivalence ratio for the small holes, and thusly, smallest delta equivalence ratio. However, the 0.1524 cm (0.06 in) diameter hole size would not be the best selection for all arc sweep cases. If a 180-degree arc sweep were to be used, the difference in equivalence ratio would not lead to a measurable difference in pressure measurement. In previous experiments, the change of equivalence ratio on the range presented by the 0.1524 cm (0.06 in) diameter hole size delta equivalence ratio, does not lead to a large enough change in pressure that would distinguish itself from the variability in pressure measurements already seen in laboratory experiments at NPS. It is

worth noting that the equivalence ratios of the previously referenced past data is the global equivalence ratio. For the purposes of this research, it is assumed that the conditions created by past experiment's global equivalence ratios will be characteristic of the local equivalence ratios that would be present in the experiment put forth by this research.

As for the final selection of the two geometries selected as the perturbed cases, one of them was determined to need to contain a 180-degree arc sweep of large hole perturbations. This was done in order to provide a good comparison metric to the previous research into changing the local equivalence ratio to induce TVC by perturbing a 180-degree arc of their geometry [9]. As for the hole size, a 0.1397 cm (0.055 in) diameter hole was selected as it provided local equivalence ratios within the operational range of RDEs while still producing relatively low deltas in the local equivalence ratio, but large enough deltas that an appreciable change in static pressure downstream could be measured. An arc sweep of 90-degrees with a small hole size of 0.1524 cm (0.06 in) diameter was determined as the second perturbed geometry case. This was done for the same reasons as the first perturbed case in terms of equivalence ratio, as well as for investigative purposes of examining the effects of lowering the arc angle for the large holes and thusly, increasing the size of the large holes within that region. To summarize, the two perturbed geometry cases were identified through careful analysis to be:

- 180-degree large hole arc sweep with 0.1397 cm (0.055 in) diameter small holes
- 90-degree large hole arc sweep with 0.1524 cm (0.06 in) diameter small holes

E. NEXT STEPS FOR RESEARCH

Due to the computational demands of this proposed experiment as well as the inability to access laboratory facilities, the experiment proposed in this body of research was unable to be conducted. However, a future researcher should consider this proposed experiment as well as the utilization of the PAD design presented in this Thesis to provide further insights into the effects of equivalence ratio perturbations and their effects on RDE performance.

SECTION A-A

SECTION B-B

DETAIL C
SCALE 2 : 3

TOLERANCES:
LINEAR:
.X ± .03
.XX ± .01
.XXX ± .005
.XXXX ± .0005

FINISH:

DEBURR AND BREAK SHARP EDGES

DO NOT SCALE DRAWING

REVISION

DRAWN: Alexis Thoeny
DATE: 22 March 2011

CHK'D:
APP'VD:
MFG:
Q.A.

MATERIAL: SS 304

WEIGHT:

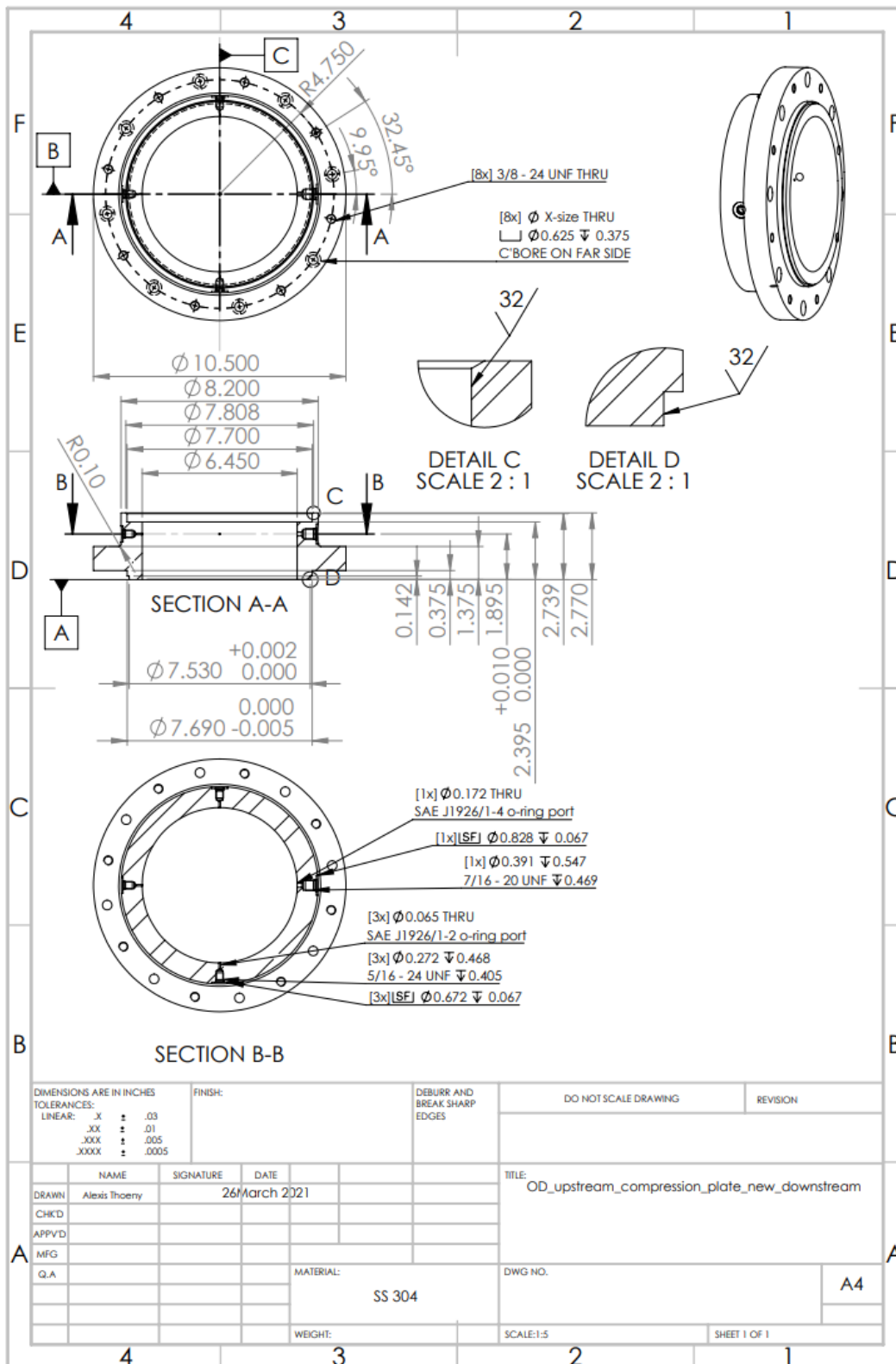
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DWG NO.

A4

SCALE: 1:5

SHEET 1 OF 1



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