University of Manitoba

MECH 4860 - Engineering Design

Phase 3: Final Design Report

Sonic Flow Rig Fixture Design

Prepared for:

StandardAero	Client
Kathryn Atamanchuk	Supervisor
Paul Labossiere	Instructor

Submission Date: Wednesday, December 8^{th}, 2021

Prepared By:

Abdelaziz, Basel	7846463
Birch, Hayden J.	7776583
Porteous, Colton D.	7828029
Sweeney, Riley	7810475

Executive Summary

StandardAero's Helicopter Programs business unit requires fixtures to be designed to allow five guide vanes from two new engine models to be mounted on their proprietary Sonic Flow Rig for testing [1]. This test is intended to find the effective flow area of the guide vane to meet OEM certification requirements [2]. Currently, the guide vanes for these engine models are tested on a machine that uses low-pressure, subsonic flow to determine effective flow area [2]. These flow conditions do not accurately represent the choked-flow conditions experienced during regular engine operation [2]. The Sonic Flow Rig recreates these conditions much more accurately and is already used to test guide vanes of other engine models [2]. The goal of this project is to allow component testing of these vanes on the Sonic Flow Rig, thereby increasing the subsequent first test pass rate of the overall engine and reducing test cell costs [2].

Creating fixture designs for five different guide vanes required the team to perform thorough project definition and concept development phases. The vanes vary in size, geometry, and features, so the team needed to understand how their similarities and differences would influence the fixture designs. The team identified the client's needs for the project, all fixtures must be compatible with the current test setup and tooling, the fixtures hold the vane similarly to in the engine, all airflow is forced across the airfoils, and cost is minimized where possible. These needs were utilized during the concept development phase, both when generating concepts and systematically selecting which concept would become the final design for each vane.

Phase 3 focused on the final refinement and analysis of the final concepts to ensure all client needs have been met. The team also finalized all the deliverables expected by the client, including a CAD model, preliminary engineering drawings, failure analysis, and a bill of materials. Due to time constraints, the last three deliverables mentioned were prepared only for one fixture, but the results of each are expected to be roughly the same.

The report follows these main topics:

- Introduction, including a summary of the progress of the project, the project scope, and the project needs and objectives.
- The detailed refinement process for one of the vane fixtures.
- A detailed design analysis of the selected vane fixture.
- Instructions to assemble the selected vane fixture.
- Design refinement summary of the remaining fixtures following the concept development phase.

Overall, the team was able to satisfy all project needs and objectives, while also undercutting cost expectations for the manufacturing of the fixture.

Contents

List of Figures

List of Tables

1 Introduction

This final design report delineates the entire process of the Sonic Flow Rig fixture design project for StandardAero. Before presenting the details of phase 3, the previous phases will be recapped in this chapter for context. The process of the project definition and concept generation phases will be summarized, followed by the scope changes that have occurred between the beginning of this project and phase 3, which will provide the guideline for the contents of this report in the following chapters.

1.1 Client Summary

StandardAero is a world leader in the maintenance, repair, and overhaul (MRO) of gas turbine engines [1]. They employ over 6,000 people worldwide across 50 major facilities, mainly located in the United States and Canada [3]. Since the company was founded in 1911, they have continued to expand and now include business aviation, commercial aviation, government and military aviation, helicopter programs, and industrial power [1], [3]. This project comes from the helicopter programs division and is introduced below [1].

1.2 Project Summary

An important part of product verification after the MRO process is testing [2]. Organizations such as the Federal Aviation Administration as well as the engine manufacturer have strict performance metrics defined for many parts of a gas turbine engine [4]. In general, repaired engine parts are tested on their own before the engine is reassembled and tested as a unit [2]. The cost of an entire engine test far exceeds that of individual parts, thus individual testing is performed to maximize the probability of a first test pass for the entire engine [2]. Therefore, it is imperative that the procedures for testing individual engine parts recreates the conditions experienced during engine operation as closely as possible [2].

This project focuses on the testing processes of helicopter turbine vanes, which direct air towards turbine blades at the optimal angle to maximize work output [2], [5]. Two examples of helicopter guide vanes are shown below.

(a) Example 1.

(b) Example 2.

Figure 1.1. Example helicopter turbine vanes [2].

StandardAero currently uses a subsonic flow apparatus which directs air through the vane to collect data regarding the effective flow area of the vane, which is an important performance metric [2]. While this device is recommended by the OEM, the airflow conditions do not closely resemble true operating conditions [2]. This sometimes results in guide vanes passing initial test, then failing the final engine test because the flow conditions are different [2]. This results in increased rework and testing costs [2].

To mitigate the amount of rework required for the vanes, StandardAero designed and built the Sonic Flow Rig (SFR) [2]. The SFR produces sonic flow across the vane by having high pressure air released from a storing tank through a discharge pipe and across the airfoils of the vane [1], [2]. This entire process is called the blow-down phase [1], [2]. Sonic flow conditions closely resemble those present during regular engine operation [2]. Figure 1.2 shows a schematic of the SFR.

[redacted]

Figure 1.2. SFR schematic [2]

StandardAero implemented the project completed in this report to expand the SFR capabilities to two additional engine models, named X and Y [1]. Actual names cannot be stated due to a non-disclosure agreement with the client [6]. Compatibility with the SFR encompasses the use of fixturing which holds the vane in place during test [2]. For the fixture to be serviceable, it must mate to a swinging plate, shown in Figure 1.3, which places the vane in the path of the air stream during blow-down.

[redacted]

Figure 1.3. Swinging plate [2].

There are five distinct vanes across the two engine models, each of which may be found in Figures 1.4, 1.5, and 1.6 [2]. The primary objective of this project is to design five fixtures, one for each vane, to provide the client with SFR compatibility and effictively increase first test pass rates for engines X and Y [1], [2]. Table I shows the full name of each of the vanes, and the nicknames given to each by the team for brevity.

Table I. Naming convention for the five vane models.

[redacted]

[redacted]

(a) X1 vane.

[redacted]

(b) X2 vane.

Figure 1.4. Engine Model X vanes.

[redacted]

(a) Y1 vane

[redacted]

(b) Y2 vane

Figure 1.5. Engine Model Y vanes.

[redacted]

(a) Vane M inlet

[redacted]

(b) Vane M exit

Figure 1.6. Inlet and exit views for the master vane M.

The first two phases of this project, project definition and concept development, have already been completed, and are summarized below.

1.2.1 Phase 1: Project Definition

Phase 1 focused on developing a thorough understanding of the problem and expectations of the client, then applying that knowledge to generate a report to outline the project. The team identified project objectives, deliverables, needs & target specifications, constraints, and created a preliminary schedule for the remainder of the project.

Project Objectives

The objective of this project was to design fixturing which allows for the five vane configurations to be mounted to the SFR and withstand the loading conditions during the blow-down phase [1]. Through discussion with the client, the project scope was limited to a detailed design specifically for X1, while the remaining four were to have a working CAD model of a refined design concept provided to the client. The deliverables are summarized below:

1. 5 CAD models corresponding to one fixture per vane configuration [1].
2. Preliminary engineering drawings for the X1 fixture [1].
3. Failure analysis (yielding, buckling and fatigue) using finite element analysis, as well as a failure mode & effects analysis for the X1 fixture [1].
4. Bill of materials characterizing the required materials, their respective quantity, and the cost of each, and an estimated price to manufacture the X1 fixture [1].

Table II shows which deliverables are to be prepared for each vane fixture.

Table II. Expected deliverables for each vane fixture.

	Vane Fixture				
Deliverable	**X1**	**X2**	**Y1**	**Y2**	**M**
CAD Model	✓	✓	✓	✓	✓
Preliminary Engineering Drawing	✓				
Failure Analysis	✓				
Bill of Materials	✓				

Client Needs & Target Specifications

Client needs were ordered from highest (1) to lowest (5) priority and may be found below. Each need (aside from number 5) has corresponding sub-needs which, when satisfied, indicate the overall satisfaction of the primary need.

1. Fixture designs are compatible with the SFR setup and standard hand tools, while also ensuring safety.
   (a) Fixtures fit within the design space [2], [7].

   (b) Fixtures are proven to be safe through ability to withstand loading conditions in a finite element simulation [1].
   (c) Mounting to the SFR is to the front of the swinging plate (denoted S.P. in Table III) [2].
   (d) Specified fasteners are able to be tightened through the use of a torque wrench [8].

2. The vanes are fixed in a manner that represents similar conditions as the engine.
   (a) The constraining face on the vane is the same as that used on the engine [2].
   (b) Vanes are restricted from rotation by the fixture during the blow-down event [2].
3. All airflow is forced across the airfoils.
   (a) The hollow core and mating interfaces are sealed [2], [8].
   (b) Cooling slots within the airfoils of the vanes (if applicable) are sealed [1].
4. Implicit and explicit costs are reduced where possible.
   (a) Readily available, commercial o-rings are used where possible [8].
   (b) Average cost per fixture is below a client approved value [9].
5. Weight is reduced where possible [8].

Target specifications were able to be generated based on the sub-needs. The final target specifications for X1 are shown in Table III. Target specifications for the other vanes may be found in Appendix A.

Table III. Metrics and their associated marginal & target values for X1.

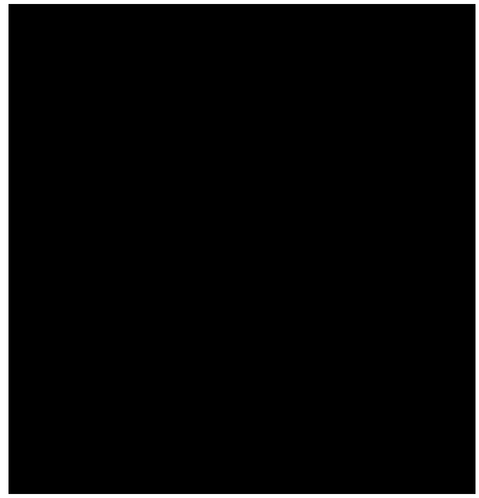
[redacted]

1.2.2 Phase 2: Concept Development

Phase 2 began concept development for four of the five vanes. M was not considered at this point considering other engine tests depend on its use [12]. Through client dialogue, there was agreement to obtain the master vane during phase 3 [12]. Thus, the design of M and its associated CAD model may be found in this report.

The beginning of phase 2 consisted of a site visit to see the SFR and the existing compatible fixtures that had been designed for another engine model [12]. Four scrap vanes were provided to the team by the client, and each team member took one vane to generate concepts for [12]. The site visit gave the team insight as to how the fixtures would be mounted to the swinging plate of the SFR [12]. Also, looking at the existing fixtures for the SFR gave the team a rough idea of how the fixtures should be shaped and assembled [12]. Figure 1.7 shows an assembled fixture and one mounted to the SFR.

(a) An existing vane fixture.



(b) Outer shell of fixture placed on swinging plate.

Figure 1.7. Pictures from the team's site visit.

The next step of phase 2 involved performing an external search for design concepts. The purpose of this was identify if there are any existing patents to ensure no concepts infringe on them. No patents for vane fixtures similar to those of the SFR were found, likely because StandardAero's patent of the SFR only recently expired in 2018 [13].

Various drawings and CAD models were provided by the client, in particular drawings of the SFR, and CAD Models of the X1 and X2 vanes. Each team member took measurements of their vane to either create a CAD model to aid in concept generation or verify that the supplied model was accurate. Additionally, a model of the swinging plate was made in SolidWorks using engineering drawings of the plate [14], [15]. This was intended for use in assemblies to understand how a given concept interacts with the SFR [14]. Figure 1.8 shows the swinging plate in SolidWorks.

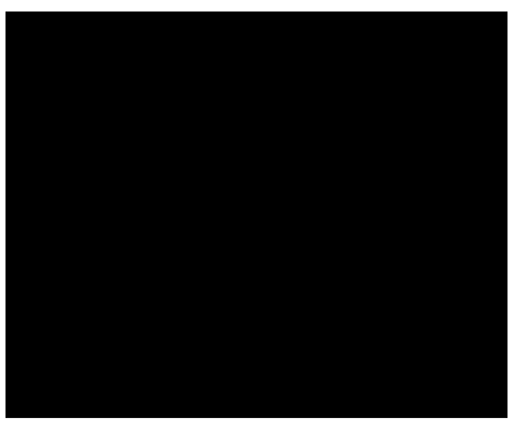


Figure 1.8. Swinging plate in SolidWorks [2].

The team then began concept generation. Weekly design review meetings were held with the client to discuss concepts created and brainstorm new ideas. This process enabled each team member to generate at least three different fixture concepts to compare during concept selection.

Using the technical specifications from phase 1, the team came up with criteria for a weighted decision matrix (WDM) to be used for concept selection. Four criteria were used:

1. [redacted]
2. [redacted]
3. [redacted]
4. [redacted]

Each team member decided on the relative weight of each criterion based on the individual properties of their vane, with the sum equalling 100%. An outline of the WDM used can be found in Appendix A. The fixture with the highest score in the WDM became the final concept design. The four selected fixtures can be found in Figures 1.9 and 1.10. Refinement of these concepts are discussed in this report, beginning with X1 in Chapter 2.

[redacted]

(a) Selected X1 fixture.

[redacted]

(b) Selected X2 fixture.

Figure 1.9. Concept selection results for engine model X.

[redacted]

(a) Selected Y1 fixture.

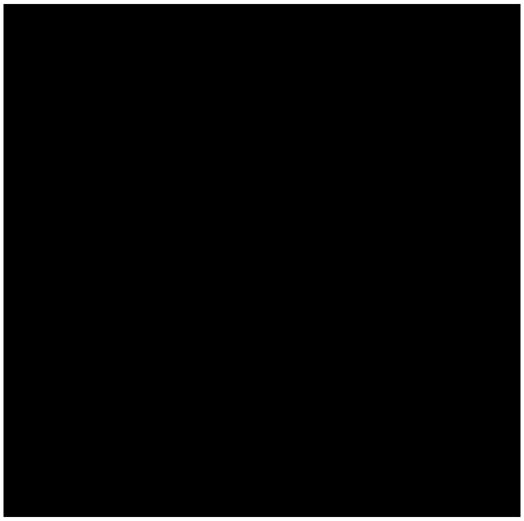
[redacted]

(b) Selected Y2 fixture.

Figure 1.10. Concept selection results for engine model Y.

2 X1 Fixture Detailed Design

This chapter involves the detailed design process of the fixture for the X1 vane. This includes the iterative design refinement performed on the fixture, then the analysis and justification of the final design. Images of a scrap X1 provided by the client are shown below.



(a) Inlet view.

(b) Exhaust view.

Figure 2.1. Inlet and exhaust view of X1.

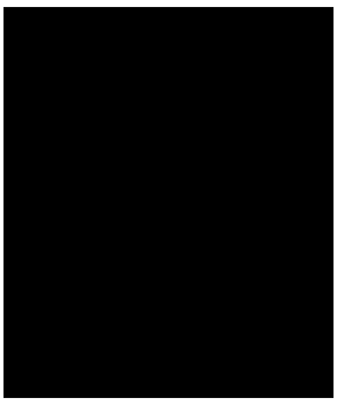
Figure 2.2. Side view of X1.

2.1 Design Refinement

Throughout the refinement process, design reviews with the client were frequently held to facilitate discussion and provide insight as to how the design could be improved. Figure 2.3 illustrates the important features of the fixture concept selected in phase 2 for X1. This concept was used as the foundation for design refinement, which is discussed in this section.

(a) Inlet view.

(b) Cutaway of the sealing plate showing the O-ring slot.



(c) Sealing plate and O-ring.

(d) Exploded view.

Figure 2.3. Concept selected in phase 2 for X1.

Figures 2.3b and 2.3c show the sealing method used, where an O-ring is housed on the back of the sealing plate [16]. This method is currently used by the client on the sub-sonic flow fixture for X1 [16]. Anti-rotation is provided by a plate which matches the geometry of the outer-most vane flange.

Refinement was performed through an iterative process which focused on general fixture features as opposed to dimensions. There were four iterations total, beginning with reducing the quantity of bolt holes, discussed below.

First Iteration - Reducing the Quantity of Bolt Holes

After discussion with the client, there was agreement that the fixture model for X1 shown in Figure 2.3 contained more bolt holes than necessary [17]. Twenty thru-holes of matching size are shown on the sealing, anti-rotation, and connecting plates, with each hole being used to connect the fixture and vane. This is impractical when considering operator use [17]. Assembly time for the technician(s) is directly related to the quantity of fasteners which require tightening [17]. Thus, the fewer fasteners, the better, provided those specified are capable of withstanding any axial load not supported by the swinging plate [17].

As there are twenty evenly spaced thru-holes on the outer-most vane flange, the reduced quantity on the fixture must be a factor of twenty to maintain even spacing. The client noted from experience that four fasteners is sufficient [17]. The changes were then applied to each piece of the fixture. Figure 2.4 shows those changes on the anti-rotation plate.

(a) Inlet view.

(b) Exhaust view.

Figure 2.4. Anti-rotation plate updated to have four thru-holes.

Another concern with the concept selected in phase 2 was that the bolt holes went all the way through the entire fixture [17]. Any mating interface which is exposed to the pressurized air released from the storage tank must prevent air from passing through the microscopic gap between the mating components [17]. This means a form of sealing would be required at the mating interfaces between the fasteners and sealing plate, as well as the fastener and center cap for the hollow core [17].

The client suggested tapping holes from the back side of both the sealing plate and inlet center cap for the hollow core [17]. To illustrate the changes, the sealing plate is used as an example by comparing a cutaway of the updated piece to the original.

(a) Original cross-section.

(b) Updated cross-section.

Figure 2.5. Sealing plate comparison.

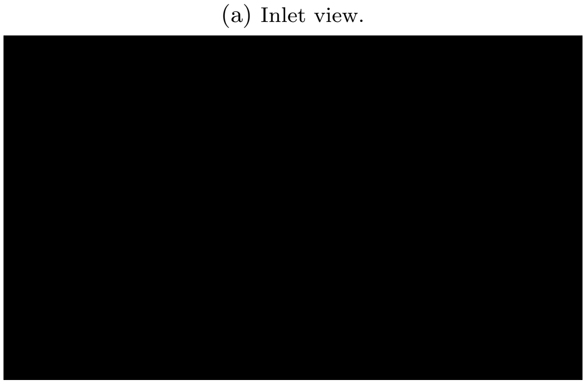
The final adjustment in this iteration was combining the anti-rotation and sealing plates to reduce both manufacturing and assembly time. Included was increasing the diameter of the sealing plate to match that of the anti-rotation plate. Having one plate results in less work for the technician(s) in terms of alignment and setup. Assembly could be accomplished by placing the plate down on a workbench, followed by the vane, and then the plate which connects the fixture to the swinging plate. This is illustrated in the figure below.

Figure 2.6. Fixture assembly shown hypothetically on a workbench.

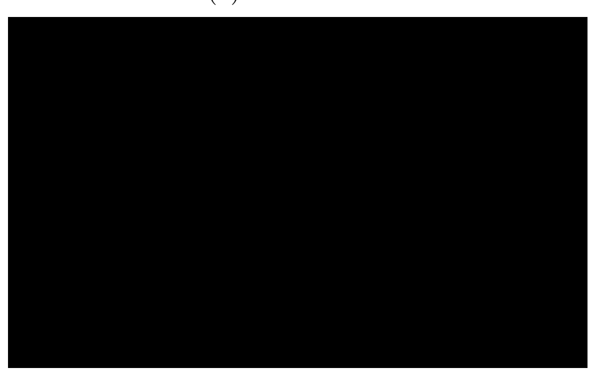
Moving forward, the combination of these two plates is referred to as the "inlet plate". To simplify the name of the plate which connects the fixture and swinging plate, it is referred to as the "rear plate" throughout the remainder of the report. The team now presents a summary of the design after the first iteration. O-rings and fasteners are left out to put focus on the fixture itself.

(a) Inlet view.

(b) Exhaust view.



(c) Exploded view 1.



(d) Exploded view 2.

Figure 2.7. First iteration of X1.

Second Iteration - Simplified Approach To Implementing Anti-Rotation

The method of anti-rotation up to this point has been machined contours on the inlet plate of the fixture which match those of the outer-most vane flange. As the vane is subjected to a torque during blow-down, it cannot rotate as the noted contours on the inlet plate are fixed. Although this is an effective method theoretically, the client noted that it imposes a significant increase in machining time [18]. Whether they manufacture the fixture in-house or elsewhere, increased complexity leads to a similar trend in cost.

To simplify the design, the four thru-holes on the fixture defined in the first iteration were used to house dowel pins instead of fasteners [18]. However, this meant that there was no method of connecting the inlet and rear plates. Thus, there were six thru-holes added to the rear plate. These were spaced evenly on a diameter greater than the absolute maximum of the vane as to avoid interference. Identical holes were tapped on the back side of the inlet plate. These changes are called out in Figures 2.8 and 2.9. The inlet and exhaust center caps did not change and thus, they are left out until the final model for this iteration is shown.

(a) Inlet plate. (b) Rear plate.

Figure 2.8. Adjustments made to the inlet and rear plates.

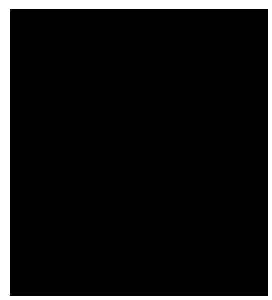
Figure 2.9. Placement of the dowel pins.

While the addition of the six thru-holes enabled connection between the two plates, the vane was not supported from the rear. The inner diameter of the rear plate was then increased to create space for a rear support which mates to the dowel pins, rear plate, and exhaust side of the outer-most vane flange. The support is shown below.

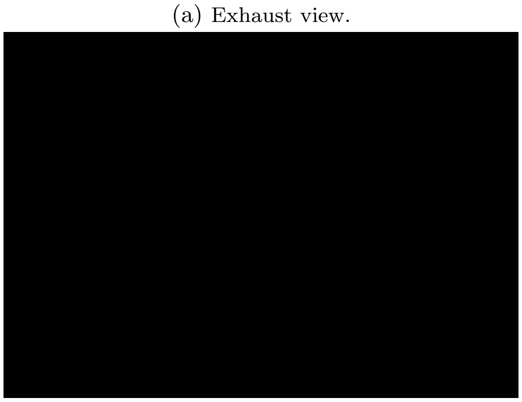
Figure 2.10. Rear support with key features identified.

Applying this support ensures the vane is supported in each direction, while maintaining the ability to prevent rotation using the dowel pins. The final model for the second iteration is presented below; the inlet view is the same as Figure 2.7a.

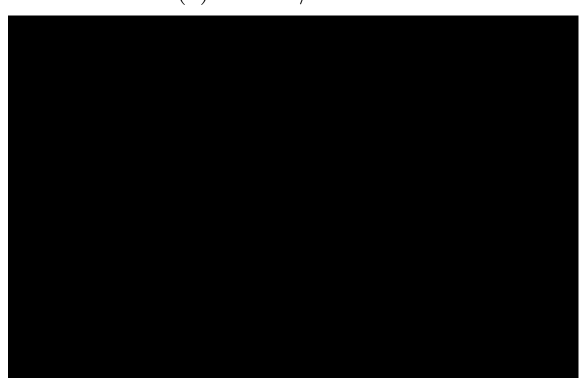
(a) Exhaust view.



(b) Exhaust/Side view.



(c) Exploded view 1.



(d) Exploded view 2.

Figure 2.11. Second iteration of X1.

Third Iteration - Combining The Rear Plate And Support

The positioning of the rear support in the second iteration introduced problems with respect to fastener loading. Feedback from the client was that the fasteners would be loaded in tension during blow-down [19]. This results from the rear face of the support sitting in a plane behind the front of the swinging plate [19], [20]. To illustrate this, the fasteners were added to the model described in Figure 2.11 such that a free body diagram (FBD) could be created.

[redacted]

(a) Fixture cutaway showing the FBD region.

[redacted]

(b) FBD of the reaction forces on the fasteners.

Figure 2.12. Fasteners loaded in tension due to rear support placement.

To ensure compressive loading on the fasteners, the team designed a new rear plate which combined the best aspects of the rear plate and support from the second iteration. The new plate had its rear-most face in the same plane as the front of the swinging plate, while the overall shape was similar to the rear support. Figure 2.13 shows the new rear plate.

[redacted]

Figure 2.13. Rear plate for the third iteration.

The rear plate for this iteration was then used to create an updated model, shown below. Fasteners to connect the inlet and rear plates as well as dowel pins are included in the various views. Again, the inlet view for the third iteration is the same as Figure 2.7a.

[redacted]

(a) Exhaust view.

[redacted]

(b) Front plane of swinging plate in comparison to aft side of fixture.

[redacted]

(c) Exploded view 1.

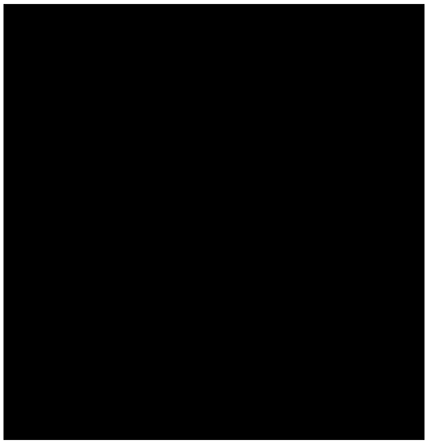
[redacted]

(d) Exploded view 2.

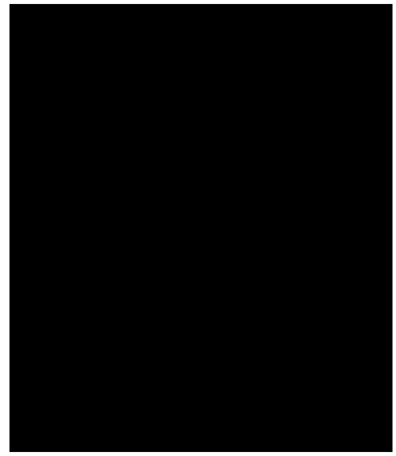
Figure 2.14. Third iteration of X1.

Final Iteration - Additional Sealing

Two additional surfaces required sealing. The first was the mating interface between the inlet and rear plates [21]. Ability to seal was implemented by inserting an O-ring slot on the aft side of the inlet plate and is shown in Figure 2.15.



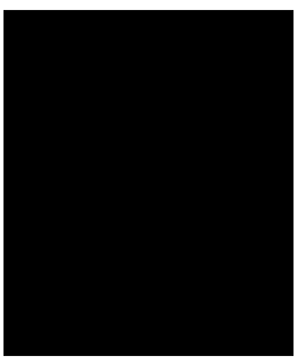
(a) Part view.



(b) Assembly cutaway view.

Figure 2.15. O-ring slot added to seal the interface between the inlet and rear plates.

The second surface was the mating interface between the inlet center cap and inner diameter of the hollow vane core [21]. Since the hollow vane core increases in diameter when moving from inlet to exhaust, there was minimal contact area to implement an O-ring seal [21]. Further, any available contact area is quite sharp which raised concerns that an O-ring seal might fail during blow-down [21]. The client agreed that it was best to design a custom gasket in this case which sits inside the center cap and has an interference fit with the inner diameter of the hollow core [21]. Figure 2.16 shows the gasket alone and fitted in the back of the inlet center cap.



(a) Gasket fitted inside the inlet center cap.

(b) Assembly cutaway view.

Figure 2.16. Custom gasket to seal the interface between the inlet center cap and inner diameter of the hollow core.

The final X1 fixture consists of 6 major components, including gaskets. The specifics of the gaskets will be covered in Section 2.8. Nomenclature was slightly adjusted for the refined product to differentiate from the iterations. Figures 2.17 and 2.18 show all of the components and the captions denote their respective nomenclature. Fastener sizes will be outlined in the next section, but the assembly of the fixture requires seven threaded rods, washers, and nuts, as well as four locating dowel pins.

(a) Inlet center cap. (b) Center gasket. (c) Exhaust center cap.

Figure 2.17. Center parts of the X1 fixture.

(a) Inlet plate. (b) Bell mouth O-ring. (c) Exhaust plate.

Figure 2.18. Outer parts of the X1 fixture.

The remainder of this chapter details material selection, sourcing of hardware such as fasteners and dowel pins, as well as the design analysis to justify the decisions in creating the project deliverables for this fixture.

2.2 Fixture Material & Fabrication Considerations

When choosing the material to be used for fabrication of the fixture there were a few extra considerations that were examined. The first of which is the weight of the material used. Since the guide vane fixture is to be installed manually by a technician, it is preferable to reduce the weight of the fixture to a point where assembling and mounting the fixture is not unreasonably strenuous [8]. The second consideration is the hardness of the material being used. Using a material that is at or near the hardness of the guide vanes could be damaging to the guide vanes that they are securing. This is especially true in locations where the guide vane comes into direct contact with the fixture or its components, which is often the case for the features designed for anti-rotation.

The last two aspects of concern are of cost and availability. To reduce the cost of implementation of the design, it is important to not only consider the cost of stock material, but the time-cost associated with obtaining the material for fabrication as well as the time associated with fabrication itself.

The availability of the material is an indirect cost that arises from the time in which it takes the client to receive it. The consequence of this is a delay in implementation and therefore a delay in the realization of the project's benefits. The main purpose of this project is to increase the first-test pass rate of the guide vanes, in turn reducing the turnaround period for refurbishments and subsequently freeing up time and resources to conduct more repairs for clients [1], [2].

The availability of the material regarding shape is also important for the same reason just mentioned, but also for cost associated with fabricating the part. For example, ideally, the client would order 15-inch diameter bar stock and use a series of turning operations on a lathe to attain the desired geometry, which is generally cheaper. The problem would be that if the only available shape that meets all of the other material constraints is for example, a plate, this would increase the complexity of the machining operation as well as the waste material, further driving up the cost of implementation.

By recommendation from the client and the justification provided in this section, the team decided to proceed with an aluminum alloy (6061-T6) which is softer than the steel alloy used for the vanes [22]. This alloy was chosen as it is also strong, lightweight, and ubiquitous in the aerospace industry [23].

2.3 Threaded Rod, Nuts & Washers

This section discusses the selection process for the threaded rod, nuts, and washers used on the fixture for X1. The client advised the team that they often use McMaster-Carr as a supplier for assembly components such as threaded rod, nuts, and washers [21]. Also, they commonly reference the United States Military Standard (MIL-SPEC) when sourcing components, so the team specified components from McMaster-Carr which adhere to this standard where possible [21].

2.3.1 Threaded Rod

Threaded rod was chosen as opposed to bolts to prevent constraining the fixture length in the axial direction any more than that of the technical specifications. Specifically, that of metrics 2 and 3 which stated that the maximum depth of the fixture axially is 9 inches forward and 7 inches aft, both with respect to the front of the swinging plate [7]. To improve efficiency for the client, the team sought to use a common thread size and length for all fixture designs. Based on the requirements of X2 that are discussed in Chapter 4, the best thread size was found to be a UNF 10-32 which has a major diameter of 0.1900 inches [24], [25]. McMaster-Carr does not have any MIL-SPEC threaded rod, so the team selected the next best option, a grade 8, high-strength steel, 10-32 threaded rod [24]. Since these fixtures are not being subjected to extreme test environments or forces, the effect of the threaded rod not being MIL-SPEC is assumed negligible.

The fixture requires seven threaded rods: one to fasten the inlet and exhaust center caps together, and the remaining six to fasten the inlet and exhaust plates. Maintaining a mindset of efficiency, the team selected the threaded rods to all be of the same length. This meant having to consider a limiting case. As per the Industrial Fasteners Institute [26], there must be at least two thread pitches aft of the nut's rear face. Analysis in Section 2.9 shows that the length from the front of the inlet center cap to the back of the exhaust center cap is larger than that of the inlet and exhaust plates. The length of the former was found to be 2.25 inches. Based on McMaster-Carr's inventory, the shortest rod which would meet the requirements of the Industrial Fasteners Institute is 3 inches with part number 90322A642 [24], [26].

2.3.2 Nuts

Nuts are required to apply a compressive force on the aft side of the fixture considering the use of threaded rod as opposed to bolts [20]. Selection was accomplished by noting the UNF 10-32 thread size and investigating McMaster-Carr's database. As per the client needs, fastener components should be able to be tightened through use of a torque wrench [8]. Specifically, in the case of nuts, six-point hex head is best [8]. The team selected a MIL-SPEC 18-8 stainless steel, six-point hex nut with part number 91240A411 from McMaster-Carr [27]. The width and height of the nut are 0.375 inches and 0.125 inches, respectively [27]. One nut is required for each threaded rod for a total of seven nuts.

2.3.3 Washers

Through discussion with the client, the decision was made to use an aluminum washer to prevent damage to the soft aluminum fixture if it were to be in direct contact with the steel nut [28]. The washer expands the bearing area and reduces the local stress applied to the fixture [23]. Washers are selected based on the thread size and nut width [29]. In this case, the outer diameter of the washer must be greater than 0.375 inches to ensure the bearing area is increased [27]. For a thread size of 10-32, part number 94589A628 from McMaster-Carr was selected which has an outer diameter of 0.438 inches [29]. This washer is also military grade [29]. Seven washers are required for the fixture.

2.4 Dowel Pin Sizing

Dowel pins are being used to transfer the torque applied to the vane through to the swinging plate to prevent rotation [18]. The dowel pin holes on the inlet plate are shown in Figure 2.19. The torque from the airfoils was calculated using computational fluid dynamics (CFD). Flow simulations allowed for analysis of fluid particles travelling across the airfoils of the vane which experience a change in momentum [5]. This momentum change exerts a force on the airfoils [5]. Since the airfoils are not located along the central axis of the vane, a torque is generated [23]. The torque was found to be [redacted]. Detailed analysis as to how the torque was computed may be found in Appendix C.

[redacted]

Figure 2.19. Detail of dowel holes on the inlet plate.

Four dowel pins were specified in the refinement process. However, the shear force was evaluated assuming only one shear pin is engaged to consider a worst-case scenario. Since the dowel pin is connected to both the inlet and rear plates, as well as the vane, it is in double shear [23]. The force in the dowel pins may be approximated by converting the torque to an equivalent force along the pitch circle diameter of the thru-holes on the outer-most vane flange [23]. Based on the torque of [redacted] and an [redacted] pitch circle diameter, the resultant force is [redacted] [23]. The shear force felt by the dowel pin is [redacted] since the load is distributed across two planes [23].

As per the Machinery's Handbook [30], the length of a dowel pin should be two times its diameter. Since the size of the thru-holes on the outer-most vane flange are 0.25 inches, the minimum pin length is 0.5 inches [30]. To uphold a factor of safety of 2, a 4037 alloy steel, 0.25 inch diameter dowel pin of length 2 inches was selected from McMaster-Carr with part number 98381A550 [31]. The pin has a strength of 10,000 lbf in double shear [31]. Four pins are required for the fixture.

2.5 Measurements of the Scrap X1

As was previously discussed, the client was able to supply a 3-D model of the part [32]. However, there was uncertainty with respect to the accuracy of the dimensions in the model [32]. To accurately define the fixture dimensions, the team measured a scrap X1 provided by the client [12].

Measuring the vane was accomplished using a six-inch digital caliper. As all the necessary dimensions are circular or on a circular face, six points were measured along the circle of interest to obtain a range of values, from which a maximum, minimum and average could be defined. Only dimensions which were critical for fixture design were taken, each of which are lettered and called out in Figures 2.20 through 2.22.

Figure 2.20. Inlet view showing dimensions A-D which required measurement on the scrap X1.

(a) Inlet view showing dimension E.

(b) Dimensions F & G.

Figure 2.21. Dimensions E-G which required measurement on the scrap X1.

(a) Side view showing dimensions H & I.

(b) Exhaust view showing dimensions J & K.

Figure 2.22. Dimensions H-K which required measurement on the scrap X1.

Measurements of the radius were performed in the cases of dimensions B, C and D above due to caliper size. Each of those dimensions are greater than six inches, thus they relied on A as a reference datum to determine the mid-point of the fixture. All of the dimensions shown are described in Table I, along with the measured values.

Table I. Description and measurement results of the dimensions necessary for X1 fixture design.

[redacted]

The client advised the team that if any measured value was within 0.030 inches of the 3-D model, including tolerances, the value in the model was acceptable for use in design [28]. Values which were not within that tolerance had to be passed on to the client for further review [28]. Two dimensions had a difference greater than 0.030 inches. The first was dimension G which has a value of 0.335 inches in the model, while the second was dimension J which has a model value of 8.68 inches. In both cases, the client deemed the model values to be most accurate considering the measured part is scrap, meaning it likely has some defects [33].

2.6 Tolerancing

Before defining the dimensions of the fixture, the client informed the team that interacting components between fixture and vane should have clearance to account for any slight dimensional differences from one vane to another [28]. Specifically, interacting components were to have at least a 0.010-0.015 inch clearance applied before the addition of drawing tolerances [28].

Unless otherwise stated, the drawing tolerance used was ±0.005 inches [28]. This value is commonly used by the client and is attainable with the manufacturing processes they are able to perform in-house [28]. In total, this means the total fit tolerance is 0.015 inches.

2.7 O-Ring & Groove Sizing

Through discussion with the client, it was decided to switch the O-ring seal for the bell mouth to a custom gasket. The location of the bell mouth seal is shown below in the cutout of the fixture.

Figure 2.23. Fixture cutaway showing the O-ring seal converted to a custom gasket seal.

This change was made as a result of the difference in pressure between the SFR and sub-sonic flow apparatus [34]. An O-ring would be much more sensitive to tolerances at the higher pressures associated with sonic flow [34]. Since the tolerances are already quite tight in this location, a custom gasket is better equipped to handle this environment [34]. Gasket sizing is discussed in the following section.

This left one O-ring and groove which had to be sized. The location of the required seal is shown in the figure below. Its purpose is to seal the mating interface between the inlet and exhaust plates against the external pressure during blow-down [35].

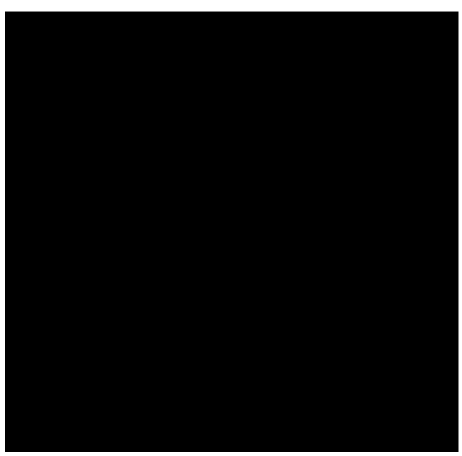


Figure 2.24. Fixture cutaway showing the remaining O-ring and groove to be sized.

As is shown in Figure 2.24, there is limited space in which the groove can be machined without interfering with the outer-most vane flange and thru-holes for threaded rod. The threaded rods are to be placed around a [redacted] diameter. The full detail as to how this value was chosen may be found in Section 2.9. The radius of the outer-most flange (dimension C in Figure 2.20) is [redacted] and was obtained from the 3-D model.

Since the O-ring is subjected to external pressure during blow-down, it must be stretched during assembly [35]. An O-ring was sourced by specifying an inner diameter of [redacted] on McMaster-Carr, as this would place the groove approximately in the middle between the outer-most vane flange and threaded rod.

An inner diameter of [redacted] corresponds to a dash number of [redacted], which is used to represent the cross-sectional diameter of the O-ring [36]. In the case of [redacted], the nominal cross-sectional diameter is [redacted] [36]. The next parameter to choose was material. The client noted that they often use polyurethane for their gaskets, so the team chose this material for the O-ring as well [28]. This allowed for selection of the O-ring with part number 9558K264 from McMaster-Carr [36].

With respect to grooves, they are based on two primary dimensions: groove width and gland depth [30], [35]. Both of these are shown in Figure 2.25.

[redacted]

Figure 2.25. Gland depth and groove width example.

Using the O-ring dimensions and multiple external resources, the groove dimensions were computed. The entire analysis may be found in Appendix B. Table II summarizes the results.

Table II. Summary of the groove dimensions.

[redacted]

As per Global O-ring and Seal [35], a global leader in O-ring manufacturing, the inner and outer edges of the groove are to be machined to a radius of 0.010 inches and 0.005 inches respectively, for a [redacted] O-ring. Figure 2.26 shows the groove relative to its surrounding components along with a callout of the inner and outer edges.

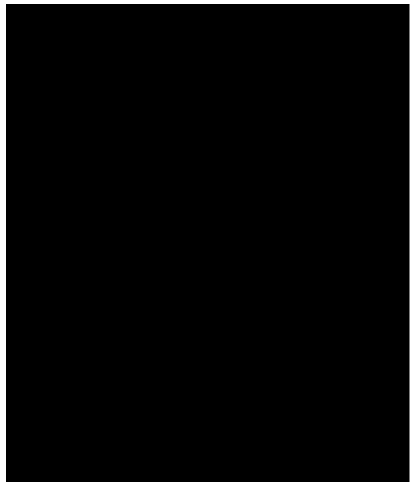


Figure 2.26. Final O-ring groove to be machined into the inlet plate.

2.8 Gasket & Recessed Slot Sizing

Two custom gaskets are required, one for the bell mouth and the other for center cap. Each of these will be placed inside a recessed slot machined into the corresponding component of the fixture [34]. The design of both the gasket and recessed slot may be found in this section, beginning with that of the bell mouth.

2.8.1 Bell Mouth Gasket

The first gasket seals the bell mouth of the vane, as was discussed in Section 2.7. During assembly, the front of the outer-most vane flange will rest against the aft side of the inlet plate. This is illustrated below.

(a) Exhaust plate hidden to highlight the region shown close-up on the right.

(b) Close-up.

Figure 2.27. Reference as to how the vane and inlet plate will interact during assembly.

Using this interface as a datum and the nomenclature of Section 2.5, three dimensions define the location of the bell mouth gasket. Those three are dimensions B, G and I. Figures 2.28 and 2.29 show those dimensions, with their respective value according to the 3-D model.

[redacted]

(a) Dimension B.

[redacted]

(b) Dimension G.

Figure 2.28. Values of dimensions B and G obtained from the 3-D model.

[redacted]

Figure 2.29. Value of dimension I obtained from the 3-D model.

The outer and inner radius of the recessed slot must then contain dimension B. That is, the inner radius must be less than [redacted], while the outer radius must be greater than this value. Considering dimension G, this provides a minimum inner radius of the slot since the inlet plate must not protrude the gas path. Applying the maximum 0.015-inch clearance between interacting components specified in Section 2.6, the inner radius of the slot must be between [redacted] and [redacted]. To allow for material support radially below the slot, the inner radius of the bell mouth gasket was chosen to be [redacted].

The outer radius of the slot was specified by considering the thickness, denoted dimension K in the figure below.

[redacted]

Figure 2.30. Dimension L, the thickness of the recessed gasket slot for the bell mouth seal.

Since the dowel pins are located on an [redacted] pitch circle diameter ([redacted] radius), they require material support radially below the pin hole. The outer radius of the recessed slot was then specified such that it would be in-line with the step-down from the dowel pin. This corresponded to a thickness of [redacted]. The recessed slot for the bell mouth seal is shown in the assembly view below.

[redacted]

Figure 2.31. Final recessed slot for bell mouth seal shown in the assembly cutaway.

The client specified that they would prefer between [redacted] and [redacted] of gasket material extending out of the slot to form an adequate seal with the bell mouth [37]. Polyurethane comes in standard sheet sizes of varying thickness with the minimum thickness purchasable from McMaster-Carr being 0.125 inches [38]. The axial depth of the slot was chosen to be [redacted], as this value ensures there is adequate gasket material to form the seal.

An interference fit between the gasket and slot was specified such that the gasket will remain in place during assembly without the need for additional fasteners [34]. Based on the standard drawing tolerance of

±0.005 inches, the maximum thickness of the recessed slot is [redacted]. The inner and outer diameters of the gasket were then specified as [redacted] and [redacted], respectively. This creates a minimum interference of [redacted] which is expected to be sufficient to hold the vane in place. Images of the bell mouth gasket are shown below.

[redacted]

Figure 2.32. Custom gasket to seal the bell mouth.

Other fixtures for the SFR use a 90A durometer, Diamondback polyurethane with a minimum tensile strength of 6500 psi from a separate manufacturer [39]. The team researched the part number of this material, but were unable to find any meaningful results as it seemed as though the material is no longer made. Recommendation was provided to the client to use this material if at all possible to maintain consistency between fixtures. Should they not be able to obtain the Diamondback polyurethane, the client noted that they would like an alternative specified from McMaster-Carr [37]. Thus, a 12x12-inch, 0.125-inch thick, 80A durometer polyurethane sheet with part number 8716K62 was selected [38]. The tensile strength of the polyurethane is 6700 psi [38]. Both the durometer and tensile strength ratings of the Diamondback could not be matched by McMaster-Carr simultaneously [38]. Thus, the tensile strength was matched to avoid the potential for catastrophic failure during test.

2.8.2 Inlet Center Cap Gasket

The two dimensions considered while designing the inlet center cap gasket were dimensions A and F. Figure 2.41 illustrates these dimensions and the values obtained from the 3-D model.

[redacted]

(a) Dimension A.

[redacted]

(b) Dimension F.

Figure 2.33. Values of dimensions A and F obtained from the 3-D model.

Similar to the analysis of the bell mouth gasket, the inner and outer radius of the recessed slot in the center cap must contain dimension A. Dimension F provides a maximum outer radius for the slot. Applying a 0.015-inch clearance as per client requests which were discussed in section 2.5, the outer radius must be between [redacted] and [redacted] [28]. To allow for room for a lip to contain the gasket from the outside, the outer radius was specified as [redacted].

The inner radius had a maximum possible value of [redacted]. The decision was made to have an inner radius of [redacted] as this provided a fair amount of clearance from the maximum allowable value, while also being a whole number. Slot depth was taken to be [redacted] as this allows for the same polyurethane sheet from McMaster-Carr to be used [38]. Once the bell mouth gasket is cut, the remaining material can be used to manufacture the inlet center cap gasket. Since the inlet cap will be compressed to the inner diameter of the hollow core by the threaded fastener, there is no need to consider material extending from the slot.

An interference fit was specified for this gasket as well to avoid the need for additional fasteners. Ensuring a minimum interference of [redacted] between the gasket and center cap, the inner and outer diameters of the gasket were specified as [redacted] and [redacted], respectively. Figure 2.34 shows the gasket.

[redacted]

Figure 2.34. Custom gasket to seal the inlet center cap.

2.9 Defining the Remaining Aspects of the Fixture Geometry

Much of the fixture geometry has been specified in previous sections. Components of the fixture which still require geometrical definition are discussed in this section. Each component is given its own subsection. In order of occurrence, the parts covered are the inlet plate, exhaust plate, inlet center cap, and exhaust center cap. The team made preliminary engineering drawings of these components for the client as per the project deliverables [1]. The drawings may be found in Appendix D and highlight all the features and dimensions of the parts discussed in this section.

2.9.1 Inlet Plate

[redacted]

Figure 2.35. Inlet plate for the X1 fixture (rear view).

The O-ring groove and recessed slot for the bell mouth gasket are features of the inlet plate which have already been discussed. Those which remain are the outer diameter, axial thickness, the face flush with the outer-most vane flange, and tapped holes.

Outer Diameter

The outer diameter of the inlet plate was selected to match that of the exhaust plate's forward side, which is [redacted]. Full detail as to how the exhaust plate was designed may be found in Section 2.9.2.

Axial Thickness

To determine the inlet plate axial thickness, which is depicted in Figure 2.36, the team had to first compute the minimum length of engagement for the threads of the fasteners, L_e'. Considering the inlet plate is tapped from the back, the thickness had to be made greater than L_e'. The team was able to follow the process outlined in Appendix **??** to determine that [redacted] [30]. Thread depth for the rods was then specified as 0.5 inches to include a small safety factor and round off the value to a standard size. To ensure there is no possibility of tapping all the way through, the axial thickness was defined to be [redacted].

[redacted]

Figure 2.36. Axial thickness of the inlet plate.

Face Flush with the Outer-Most Vane Flange

The face flush with the outer-most vane flange is shown in Figure 2.37. This face already had its inner diameter specified as [redacted] during the recessed slot design for the bell mouth gasket, leaving two dimensions to analyze. First, the depth of the face, also shown in Figure 2.37, is measured from the aft-most plane of the inlet plate. The depth must be at least 0.010 inches greater than the thickness of the outer-most vane flange to adhere to the client's clearance requests on clearance discussed in section 2.5 [28]. From the 3-D model, the outer-most vane flange thickness is [redacted], thus the depth of the face was specified as [redacted].

The second dimension to specify was the outer diameter of this face. This value is dependent on the maximum diameter of the outer-most vane flange, which was found to be [redacted] in the 3-D model. Ensuring the minimum clearance of 0.010 inches is upheld, the outer-diameter was specified as [redacted] [28].

[redacted]

Figure 2.37. Dimensions which define the face flush with the vane.

Tapped Holes

There are two sets of tapped holes, one for the threaded rods and the other for the dowel pins; each are shown in Figure 2.38. Considering the threaded rod holes, these were designed to be extruded to a depth of [redacted] with [redacted] of 10-32 internal thread. This then satisfies the minimum length of engagement requirement, as was previously discussed [30]. The pitch circle diameter was specified as [redacted]. Since the outer diameter is [redacted], the holes for the threaded rod were placed a safe distance radially below the outer diameter to prevent significant stress concentrations [23].

For the inner set of holes, the dowel pins must sit on a [redacted] pitch circle diameter to pass through the holes on the outer-most vane flange. Thus, the same pitch circle diameter was used on the inner set of tapped holes. Although the dowel pin diameter is 0.25 inches, the client noted they preferred if the dowel pin holes were machined to a size slightly greater than the dowel pin [28]. This would allow for some play during assembly [28]. To accommodate this request, the team specified a hole of depth [redacted] using a [redacted] drill bit.

[redacted]

Figure 2.38. Tapped holes on the inlet plate.

2.9.2 Exhaust Plate

[redacted]

Figure 2.39. Exhaust plate for the X1 fixture (front view).

Dimensions of the exhaust plate have not been discussed through this point of the report. This section provides full detail as to how the exhaust plate was designed.

Thru-Holes

The client uses [redacted] UNC socket head cap screws to secure fixtures to the swinging plate [39]. Metric 7 states that there must be 6 thru-holes of diameter [redacted] (metric 8) spaced evenly around a [redacted] pitch circle diameter (metric 9) to allow for use of these screws [2]. Thru-holes of these dimensions were then placed along the noted pitch circle diameter, and can be seen on the outer edge of the part in Figure 2.39. None of these holes are to be threaded [20]. Since there are already threads tapped into the swinging plate of the SFR, the compressive force required for the exhaust plate is provided by the head of the fastening bolt [20].

There are two other sets of thru-holes: one for the size 10-32 threaded rods and the other for the 0.25-inch dowel pins, similar to the design of the inlet plate. The differences for the exhaust plate holes are that both sets are to be fully through the part and those for the rods are not to be threaded [20]. Like what was noted for the swinging plate mounting holes, since the inlet plate is tapped with thread, the required compressive force to secure the exhaust plate is provided by the hex nuts [20]. Each set of thru-holes are shown in the figure below.

[redacted]

Figure 2.40. Rear view of the exhaust plate showing all thru-holes.

Inner Diameter

The inner diameter of the exhaust plate was selected to provide radial support below the dowel pins, which are stationed on a [redacted] pitch circle diameter. Another consideration was the bearing area between the vane and exhaust plate. During blow-down, the vane exerts a force on the exhaust plate along the mating interface between the two [40]. Since stress is inversely proportional to area, the greater the bearing area, the lower the stress [23]. Although, there is a minimum inner diameter to prevent interference during assembly which is defined by the difference between dimensions C and K of the vane. Both are shown below with their respective values having obtained from the 3-D model.

[redacted]

(a) Inlet view showing dimension C.

[redacted]

(b) Exhaust view showing dimension K.

Figure 2.41. Values of dimensions C and K obtained from the 3-D model.

The difference between the two dimensions yields a circle with diameter [redacted], which is the minimum diameter for the exhaust plate. Adhering to the 0.010-inch minimum clearance between fixture and vane, the inner diameter of the exhaust plate was specified as [redacted] [28].

Aft Side Thickness

The exhaust plate features a radial step-down which results in two outer diameters of differing thicknesses, one for the aft side and the other for the forward side. Both thicknesses are illustrated in Figure 2.42. This radial step-down was implemented to ensure the client does not have to purchase additional socket head cap screws for the swinging plate. The thread length of the [redacted] UNC screws they have on hand is [redacted] [39]. Increasing the thickness of the aft side decreases the length of engagement between the internal and external threads of the swinging plate and screw, respectively.

[redacted]

Figure 2.42. Side view of the exhaust plate showing the aft and forward thicknesses.

As per the Nord-Lock Group [41], internal and external threads made from the same material should have a length of engagement of at least one nominal bolt diameter. Thus, the thickness of the aft side was specified as [redacted]. This leaves [redacted] of engaged thread.

Forward Side Thickness

The thickness of the forward side of the plate was designed through consideration of overall exhaust plate stress distribution. Of all the fixture components, the exhaust plate will experience the greatest stress due to the bearing load imparted by the vane during blow-down [40]. Thus, the overall thickness of the exhaust plate was made the largest of any component by specifying a forward side thickness of [redacted]. This creates an overall plate thickness of [redacted]. Stress distribution in the exhaust plate is discussed further in section 2.10.2.

Aft & Forward Side Outer Diameters

The aft and forward side outer diameters are shown in Figure 2.43. Both were designed to have sufficient bearing area for the [redacted] UNC screws [39]. The outer radius of the screw's hex head is [redacted], meaning the aft side outer diameter must be greater than [redacted] [42]. In the case of the forward side, the outer diameter must be less than [redacted]. Considering technical specification number 1, the nominal outer-most diameter of the fixture cannot exceed [redacted], thus the aft side diameter was specified as [redacted] [2]. This maximizes bearing area and matches what the client is using on existing SFR fixtures [39].

[redacted]

Figure 2.43. Side view of the exhaust plate showing the aft and forward outer diameters.

The outer diameter of the forward side was specified as [redacted] inches as this leaves [redacted] radially for a torque wrench [39]. Again, this is the value used by the client on other SFR fixtures which include a radial step-down from the aft side [39].

2.9.3 Inlet Center Cap

[redacted]

Figure 2.44. Inlet center cap for the X1 fixture (rear view).

The inlet center cap was previously analyzed to determine the recessed slot dimensions for the custom gasket to seal the hollow core. This section discusses the remaining dimensions, beginning with the outer radius.

Outer Radius

The outer radius of the cap must be greater than that of the recessed slot to allow for a lip of material radially above the slot. This is illustrated in Figure 2.45. This lip establishes the interference fit with the gasket by supporting the normal force exerted by the gasket, which is trying to return to its natural, undeformed state [23]. From section 2.8.2, the outer radius of the recessed slot is [redacted], while the maximum possible radius of the inlet center cap is [redacted]. The outer radius of the inlet center cap was specified as [redacted], maximizing the lip thickness, and reducing the overall stress in the cap [23].

[redacted]

Figure 2.45. Side view cutaway of the inlet center cap showing the outer radii of the component and recessed slot.

Fillet on Inlet Face

The client noted that the inlet center caps on the fixtures for other engine models make use of a fillet on the edge of the inlet face [12]. This helps guide air which has impacted the front face towards the airfoils, rather then a sudden change of geometry associated with a sharp edge. Fillets of radius [redacted] are specified for all fixture edges to prevent stress singularities, however that of the inlet face for this component was specified as [redacted]. Figure 2.46 shows this fillet.

[redacted]

Figure 2.46. Side view of the inlet center cap showing the fillet applied to the edge of the inlet face.

Axial Thickness

The axial thickness of the inlet center cap was determined by following a similar procedure to that of the inlet plate. The cap is tapped from the back and houses one 10-32 UNC threaded rod. As is discussed in Appendix **??**, the minimum length of engagement between a steel external and aluminum internal thread is 0.3891 inches [30]. The thread depth of the tapped hole was then specified as [redacted] for the same reasons discussed for the inlet plate. Lastly, the axial thickness of the inlet center cap was designed to be [redacted] inches to prevent any possibility of tapping all the way through.

Tapped Hole

Since there is only one threaded rod for the inlet center cap, the corresponding tapped hole was placed at the mid-point of the rear face, which is synonymous with a pitch circle diameter of zero. The tapped hole is shown in Figure 2.47. Specifications for the hole are a total depth of [redacted] with [redacted] of 10-32 internal thread.

[redacted]

Figure 2.47. Side view of the inlet center cap showing the fillet applied to the edge of the inlet face.

2.9.4 Exhaust Center Cap

[redacted]

Figure 2.48. Exhaust center cap for the X1 fixture (front view).

The exhaust center cap is the simplest of the fixture components. Three features completely define the cap: the outer diameter, axial thickness, and thru-hole. Each are discussed in the proceeding subsections.

Outer Diameter

The outer diameter must be great enough to ensure sufficient bearing area between the exhaust center cap and resting face on the vane. This is illustrated in Figure 2.49.

[redacted]

Figure 2.49. Location of the resting face which the exhaust center cap is mated to.

Inner and outer diameters of the resting face were not able to be measured through use of a caliper. However, the client gave permission to use the 3-D model [34]. The locations of the inner and outer diameters are shown in Figure 2.50.

[redacted]

(a) Inner diameter.

[redacted]

(b) Outer diameter.

Figure 2.50. Exhaust view showing the dimensions of the resting face for the exhaust center cap when assembled.

From the 3-D model, the outer and inner diameters of the resting face are [redacted] and [redacted] inches, respectively. The outer diameter of the exhaust center cap was specified as [redacted]. This maximizes bearing area, while applying a clearance of 0.015 inches to account for discrepancies from one vane to another.

Axial Thickness

The axial thickness was determined by noting that the stress applied to the exhaust center cap is quite low in comparison to the other fixture components. Thus, the thickness may also be small in comparison. The final value specified was [redacted].

Thru-Hole

There is one thru-hole on the exhaust center cap for a 10-32 UNC threaded rod to pass through and fasten to the inlet center cap. This thru-hole is not to be threaded since the compressive force required to secure the exhaust center cap in place is provided by the hex nut [20]. Figure 2.51 shows the thru-hole.

[redacted]

Figure 2.51. Thru-hole on the exhaust center cap.

2.10 Failure Analysis

Two methods of failure analysis were performed: failure modes and effects, as well as finite element simulations. Both are discussed in this section.

2.10.1 Failure Modes And Effects Analysis

The team developed a Failure Mode and Effect Analysis (FMEA) to track potential failures for the fixtures. The FMEA summarizes the failure, the potential failure mode, and its potential effect. After finding this information, the team then developed criteria for the severity, rate of occurrence, and the detectability of the failure as shown in Tables III, IV, and V. This information is then used to find the risk priority number (RPN) and the future controls to mitigate the failure risk. The FMEA is then developed and shown in Figure 2.52.

Table III. FMEA severity ratings

Rating	Criteria: A failure could ...
10	Injure an employee
9	Violate safety regulations or codes
8	Render the fixture unfit for use
7	Cause extreme customer dissatisfaction
6	Result in partial malfunction
5	Cause a loss of performance
4	Cause minor performance loss
3	Cause a minor nuisance that can be overcome with no loss
2	Be unnoticed; minor effect on performance
1	Be unnoticed; no effect on performance

Since these fixtures are being used in a high-pressure environment, a major malfunction could have devastating effects [12]. The team considered risks to workplace safety the highest severity, aligning with StandardAero's own emphasis on workplace safety [12]. Below that, the team considers damage to the fixture or vane the next highest severity [1]. This is due to the high explicit costs associated with damaged parts. Moving further down, the criteria includes measurement inaccuracies which may cause need for extra rework on parts and carries a high implicit cost [2]. Finally, near the bottom of the severity list are things that would only have little to no effect on the performance or accuracy of the test. These would be more related to quality of life, such as issues assembling the fixture or installing it on the SFR.

Table IV. FMEA occurrence ratings

Rating	Probability of occurrence
10	$>$30%
9	$\leq$30%
8	$\leq$5%
7	$\leq$1%
6	$\leq$0.3 per 1000
5	$\leq$1 per 10000
4	$\leq$6 per 100000
3	$\leq$6 per million
2	$\leq$3 per 10 million
1	$\leq$2 per billion

It is likely that these fixtures will be used for testing ample times, which adds emphasis to the fixtures' consistency and durability over its lifetime. The highest occurrence ratings correspond to failures occurring multiple times per day, to a few times per week. This kind of performance would be unacceptable and would likely result in the client abandoning that fixture design. Failures occurring one to a few times a month could range from a slightly high-maintenance fixture to once again being shelved for a redesigned fixture. This would depend on the severity of the failure but is still preventable through diligent design. The lowest ratings would be attributed to things like wear and tear and are generally expected from parts such as this.

Table V. FMEA detectability ratings

Rating	**Definition of detectability**
10	Defect caused by failure is not detectable
9	Occasional units are checked for defects
8	Fixtures are systematically sampled and inspected
7	Fixtures are manually inspected
6	Manual inspection with mistake-proofing modifications
5	Process is monitored and manually inspected
4	Process is monitored with an immediate reaction to defects
3	Fixtures are inspected semi-automatically to detect defects
2	Fixtures are inspected automatically to detect defects
1	Defect is highly detectable

Detectability is a major factor in preventing many mechanical failures, regardless of severity or occurrence. Detecting problems that could cause failures is dependent on two factors: the organization's system for inspections, and the nature of the defects themselves. StandardAero is highly organized and systematic with inspections, so the team is less worried about that factor. What is more concerning is if parts of the fixtures have a propensity to accumulate defects that are hard to detect, even with regular inspection. The criteria reflect this.

Figure 2.52. Failure Mode and Effect Analysis.

Failure	Potential Failure Mode	Potential Effect	SEV	OCC	Current Controls	DET	RPN	Future Controls
Air Leakage	Missing/Damaged Fasteners	Air leakage causing unreliable test results	8	3	Fasteners are always designed to be in tension	2	60	Gaskets and fasteners are inspected before performing the test
	Missing/Damaged Gaskets	Air leakage causing unreliable test results	8	3	Fasteners are always designed to be in tension	2	60	
Vane Damaging	Metal on metal contact	Air leakage causing unreliable test results	7	1	Usage of Gaskets when necessary	3	30	Gaskets and fasteners are inspected before performing the test
		Damaging the Vane	6	1			24	
Part not fitting on fixture	Wear and tear	Inability to perform testing	7	3	Increasing the tolerance in critical parts to count for the wear and tear of the vane	2	42	Ensuring the fixture tolerance counts for the size change of the part during test and wear of the vane.
Rotation of the Vane during test	Shearing of anti-rotational pins	Inability to perform testing	7	4	NA	4	112	Inspecting Pins before testing the vane

2.10.2 Finite Element Simulations & Analytical Calculations

The team performed finite element simulations to ensure the fixture can withstand the loading conditions present during the blow-down phase. Through discussion with the client, the decision was made to focus on a worst-case scenario for the simulations [40]. Since the rear plate of the fixture is stationed between the inlet and swinging plates, the client noted that it will experience the greatest compressive force [40].

During blow-down, the vane is constrained in the axial direction by the rear plate, meaning the former is exerting a force on the latter [40]. The force was estimated by assuming the vane acts as a flat plate perpendicular to the oncoming air with a circular cross-section and a diameter equal to that of the outer-most vane flange [40]. This was a conservative approach considering the vane exhausts air through the airfoils rather than acting as a blunt body [40]. The client advised the team that there is a maximum pressure differential of [redacted] across the vane during blow-down which is equivalent to a force of [redacted] acting on the hypothetical flat plate [40]. Detail as to how this force was computed may be found in Appendix **??**.

Force exertion from the flat to rear plate is by way of a bearing surface defined by the outer and inner diameter of the flat and rear plates, respectively [40]. The reaction force from the swinging plate is applied to the mating surface between the rear and swinging plates [40]. Both the bearing and mating surface, as well as their respective simulation conditions are shown in the following figure.

[redacted]

(a) Inlet view showing the bearing load.

[redacted]

(b) Exhaust view showing the fixed geometry.

Figure 2.53. Simulation conditions applied to model the stress distribution in the rear plate.

Yielding Analysis

Yielding occurs when the maximum stress in a material exceeds its yield strength [23]. In most applications, yielding is to be avoided, as is the case in this project [7]. To ensure a factor of safety of at least 2 against yielding, as per metric number 4, multiple simulations were performed [7]. The results from the simulation with the greatest number of elements is shown in Figure 2.54.

[redacted]

Figure 2.54. Finite element analysis results for yielding with 70,038 elements.

The factor of safety against yielding was found to be [redacted], which satisfies metric number 4. Convergence was analyzed in Appendix ?? and the team concluded that although convergence is not achieved, there are no stress singularities in the model [43]. Thus, convergence is anticipated with an increase in mesh refinement; a software with increased computing capabilities would be required to verify this, but such is out of the project scope [43]. The maximum stress of [redacted] shown in Figure 2.54 is used in proceeding calculations.

Finite element simulations must be verified either by engineering justification or analytical calculations, where possible. With respect to the latter, this is usually accomplished by comparing a computed analytical stress value in a given region, to that computed in the simulation. For this project, the normal stress along the bearing surface was computed analytically and compared to that of the model [44]. The two computational methods were within 4 % of one another, which was deemed to be an acceptable error given the high safety factor. Appendix ?? details the entire verification process.

Fatigue Analysis

Cyclic loading conditions effectively reduce the strength of the material as cracks propagate with increased loading cycles [45]. This is defined as fatigue [45]. Fatigue performance of a material is governed by a Stress-Life, or S-N, curve which plots the maximum stress amplitude as a function of the number of cycles (on a log-log scale) [45]. The stress amplitude is based on a completely reversed behaviour, where the material would oscillate between tension and compression of equal stress magnitude [45].

In the case of the exhaust plate, it is subjected to zero-based loading where the material is repeatedly loaded and unloaded [45]. Using the S-N curve for zero-based loading requires a Goodman correction, which converts the zero-based amplitude to an equivalent completely reversed amplitude [45]. The stress amplitude for zero-based loading is half the maximum stress, which was determined to be [redacted] in the yielding analysis [45]. The Goodman correction was applied in Appendix **??** and it was determined that the equivalent stress amplitude is [redacted].

The team anticipates that the designed fixture for X1 will be used approximately 1000 times throughout its service life. As per Shigley's Mechanical Engineering Design [45], the maximum stress amplitude corresponding to 1000 cycles for aluminum is [redacted]. Compared to the actual stress amplitude of [redacted], a safety factor of [redacted] exists against fatigue. This satisfies metric 5 [7].

Buckling Analysis

Buckling is a common phenomena of columns or other slender structures which are subjected to compressive loading from both sides along a given axis [23]. In the case of the simulation conditions presented in Figure 2.53, the exhaust plate is subjected to bending and bearing stresses, neither of which lead to buckling [23]. While the bearing stress exerted from the vane causes a normal stress, there is no constraint on the aft side of the bearing surface, meaning the threat of buckling is minimal [23].

To confirm that buckling will not occur during blow-down, the same simulation conditions were applied to the exhaust plate such that a buckling analysis could be performed in SolidWorks. The results of the simulation are shown in Figure 2.55. Highlighted in the figure is the buckling load factor, which was found to be [redacted]. This value represents the factor in which the applied load must be scaled to result in buckling [46]. Since the area is not changing with an increase in load, the buckling load factor is equivalent to the factor of safety against buckling [46]. Since the safety factor is greater than 2, buckling is not expected, and metric number 6 has been satisfied [7].

[redacted]

Figure 2.55. Finite element analysis results for buckling with 66,951 elements.

2.11 Cost

This section details the costs of producing and implementing the fixture. The scope of these costs includes the procurement of the material and fabrication cost. The scope of the cost does not include internal engineering related costs associated with implementation such as transferring the CAD models between software.

2.11.1 X1 Fixture Bill of Materials

Below is a table of materials required for the fixture.

All the items listed in the table above were sourced from McMaster-Carr and may not necessarily be the same source from which the stock materials are procured [40]. In particular, the 6061-T6 stock material will more than likely be purchased locally, and the cost would be lower [40]. The details of the procurement of the alloy are at the discretion of StandardAero. All the fastening components used in the fixture models such as threaded rods, bolts, nuts, and washers are intended to be purchased from McMaster-Carr. The total estimated material cost of the X1 vane fixture is [redacted].

2.11.2 Fabrication

To fabricate the plates and caps, a set of turning operations will be performed [40]. Using turning operations allows StandardAero to fabricate these parts internally and therefore the cost of fabrication was estimated by the amount of time the parts take to machine and an approximate hourly rate for two machinists at StandardAero [40]. The time to complete sets of roughing and finishing operations including setup and take down was estimated to be around [redacted] by the client [37]. The client provided a combined hourly rate of [redacted] an hour for the two machinists [37]. Thus, the fabrication cost is estimated to be around [redacted]. This fabrication cost is intentionally overestimated due to uncertainty and lack of information available to the team regarding wages and equipment available. The total estimate cost of producing the X1 Fixture is [redacted], immensely undercutting the target cost of [redacted].

3 Assembly Instructions

This chapter aims to aid the client in creating an official work instruction for the X1 vane fixture once it has been manufactured. It is important to explicitly explain how the fixture is intended to be assembled to ensure safety of the operator assembling the fixture, as well as to maximize the performance and lifetime of the assembly. The instructions will be given as a numbered list, with sub-lists to describe detailed steps within a larger step if necessary. Steps will be supplemented with pictures of the 3-D model, to be replaced with pictures of the actual parts once they have been fabricated.

Prior to beginning the instructions for assembly, the assumptions being made are listed below.

- All parts of the fixture are accessible and disassembled.
- All fasteners, sealing elements, and tools are accessible.
- Testing of the master vane has already been performed. These instructions are intended only for the X1 vane fixture.
- Operators shall aim for the mean of a given torque range. For example, if a range is given as [redacted] inch-pounds, they shall aim for [redacted] inch-pounds.
- Installation of the fixture onto the SFR will not be covered.

The major parts of the fixture assembly will each be assigned a number for brevity within the instructions. The parts and there respective numbers can be found in Figures 3.1 and 3.2.

[redacted]

(a) Inlet center cap (1).

[redacted]

(b) Center gasket (2).

[redacted]

(c) Exhaust center cap (3).

Figure 3.1. Center parts of the X1 fixture.

(a) Inlet plate (4).

(b) Bell mouth gasket (5).

(c) Exhaust Plate (6)

Figure 3.2. Outer parts of the X1 fixture.

1. Prior to assembly:
   (a) Inspect all O-rings and gaskets prior to assembly. Replaced damaged parts if necessary.
   (b) Inspect all fasteners for damage or wear and tear. Replace damaged parts if necessary.
   (c) Inspect the vane for any damage or wear and tear. Any damage on the vane should be repaired prior to testing.
2. Place the inlet center cap (1) face down so that the threaded rod is sticking up. Place the center gasket (2) on top of (1) so that (2) fits snugly in the recessed channel.

Figure 3.3. Gasket (2) inside center cap (1).

3. Orient the test vane so that the airfoil leading edges point down. Place the test vane on top of the gasket (2) so that the edge of the center opening of the vane presses into the gasket.

[redacted]

Figure 3.4. Test vane on top of (1) and (2).

4. Place the exhaust center cap (3) through the center threaded rod of (1) and ensure the outer edge of (3) is in contact with the test vane. Secure with a washer and nut. Tighten to [redacted] inch-pounds [39].

[redacted]

Figure 3.5. Exhaust center cap (3) fastened to test vane.

5. Place the inlet plate (4) so that the holes for the threaded rods and locating pins face up. Install 4 dowel pins in the innermost set of holes, and 6 threaded rods in the middle set of holes. Place the bell mouth gasket (5) in the O-ring groove of (4). The O-ring groove is the edge just inside from the locating pins on (4).

[redacted]

Figure 3.6. Inlet plate (4) with bell mouth gasket (5) installed.

6. Place the test vane, airfoil leading edges facing down, onto (4), aligning the holes on the outer flange of the vane with the 4 locating pins on (4).

[redacted]

Figure 3.7. Test vane resting on (4) and (5).

7. Orient the exhaust plate (6) so that the flat side is facing up. Carefully lower (6) onto the back of the test vane, aligning the holes with the locating pins and threaded rods.

[redacted]

Figure 3.8. Exhaust plate (6) resting on the test vane.

8. Secure the 6 remaining threaded rods with a washer and nut. Tighten to [redacted] inch-pounds [39]. The fixture is now assembled.

[redacted]

Figure 3.9. Fully assembled X1 fixture.

4 Design Refinement

Following concept selection in phase 2, each vane fixture design was refined to ensure that each CAD model was complete for submission. Any refinements made were endorsed by the client. This chapter summarizes the refinement changes made to the X2, Y1, Y2, and M fixtures. After the refinements to these CAD models, these four vane fixtures are considered to be completed in terms of project deliverables.

4.1 X2

Refinement for the X2 fixture mainly involved adding features to the 3D model to make it ready for fabrication in the future. This includes adding tapped holes for the 10-32 studs that will hold the fixture together. These holes have been tapped for a Helicoil insert, since there is little clearance between the holes and the inner edge of the fixture. The Helicoil inserts will make repairs easier in this space, since the inserts can be replaced.

Another change made to the X2 fixture was the addition of new holes to hold dowel pins to provide extra anti-rotation during the blow-down phase. The bolts connecting the fixture plates to the vane use the smaller [redacted] holes on the outer flange. Since these bolts will be quite small, adding [redacted] dowel pins to the larger [redacted] holes will ensure the vane holes are not damaged by the bolts during the blow-down phase. The dowel holes are drilled partially through the back of the front plate, and completely through the back plate. The updated holes can be seen below in Figure 4.3.

[redacted]

(a) Rear view of the front plate.

[redacted]

(b) Front view of the back plate.

Figure 4.1. Detail of updated holes on the X2 fixture.

Figure 4.2 below shows the CAD model for the refined X2 fixture.

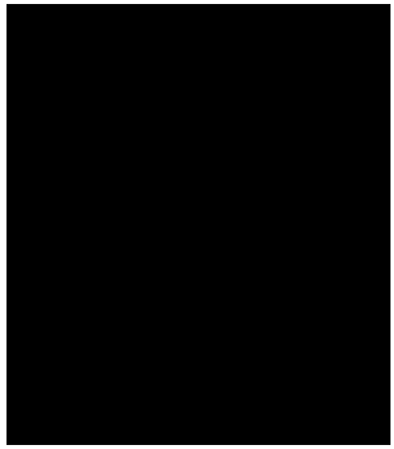
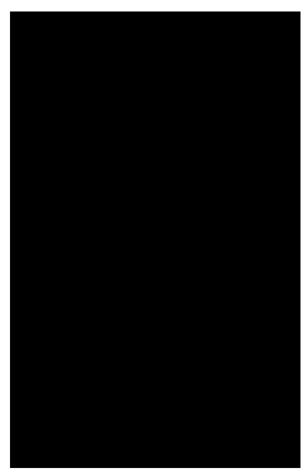


Figure 4.2. Refined X2 model.

4.2 Y1

There were two changes made to finalize the selected Y1 guide vane fixture design. The first was to standardize the bolts and bolt holes used to secure the fixture halves together. The second was a redesign of the center plug fastening feature. The standardized bolts are 10-32 which call for an outer diameter of 3/16-inch threaded holes. The orientation of the plug bolt was reversed, and a threaded female hub was extruded from the rear of the front plug. This was done to remove the hole from the front surface of the plug which would have required additional sealing elements. In Figure 4.3b, the standardized bolts used are pictured in blue and the hub extrusion is highlighted in yellow. Figure 4.3a is an isometric view of the final assembly to provide context for the cut away representation in Figure 4.3b.

(a) Isometric view of the final assembly.



(b) Standard bolts (blue) and plug feature (yellow).

Figure 4.3. Detail of Y1 Final changes.

4.3 Y2

After discussions with the client about the Y2 fixture design, changes were made to improve the fixture design and simplicity. The first change made to the Y2 fixture was done to the inlet and exit caps. The number of holes on the exit cap was reduced from eight to two holes [19]. The two holes being used to connect the anti-rotation bar, exit cap, and the vane to prevent its rotation [19]. The inlet cap holes were all removed to prevent the need of covering the holes to prevent air leakage [19]. Instead, a tap hole is made to the inlet cap, this connects the anti-rotation bar to the inlet cap [19]. The changes made to the exit cap are shown in Figures 4.4 and 4.5. The changes made to the inlet cap are shown in Figures 4.6 and 4.7.

(a) Exit cap before changes.

(b) Exit cap after changes.

Figure 4.4. Detail of changes made to the design of the exit cap.

Figure 4.5. Detail of the anti-rotation bar connection to the exit cap and vane.

[redacted]

(a) Inlet cap before changes.

[redacted]

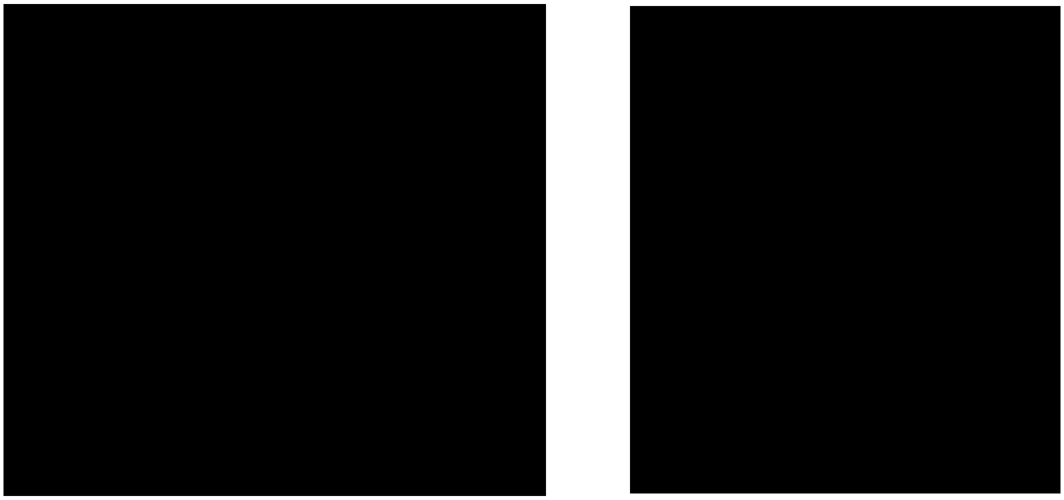
(b) Inlet cap after changes.

Figure 4.6. Detail of changes made to the design of the inlet cap.

[redacted]

Figure 4.7. Detail of anti-rotation bar connection to the inlet cap.

The other improvement made on the fixture was changing the anti-rotation bar to accommodate the new changes made to the inlet and exit caps [19]. The first change was to extend the bar from the vane side to connect the bar directly to both the exit cap and the vane [19]. This change removed the need to have separate holes for the anti-rotation bar to connect it to the exit cap [19]. The second change was to reduce the length of the anti-rotation bar connected to the axial support frame [19]. The changes to the anti-rotation bar are shown in Figure 4.8 below.



(a) Anti-rotation bar before changes. (b) Anti-rotation bar after changes.

Figure 4.8. Detail of changes made to the anti-rotation bar.

4.4 M

The master vane shares some measurements with the Y2 Vane. The center hollow portion of vanes M and Y2 are similar as well as the size of the thru-holes used to prevent the rotation of the vane during the blow-down phase. This enabled the team to use the same inlet and exit caps and the anti-rotation bar for vane M as shown in Figure 4.9.

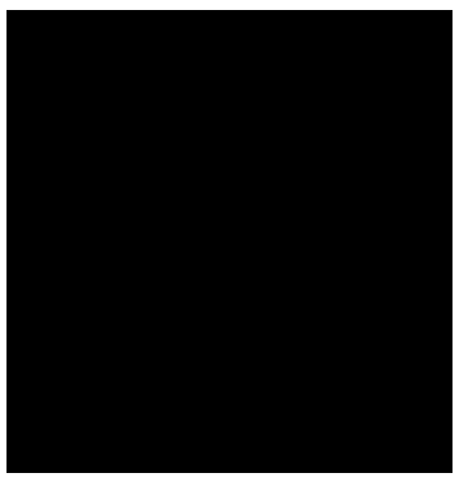
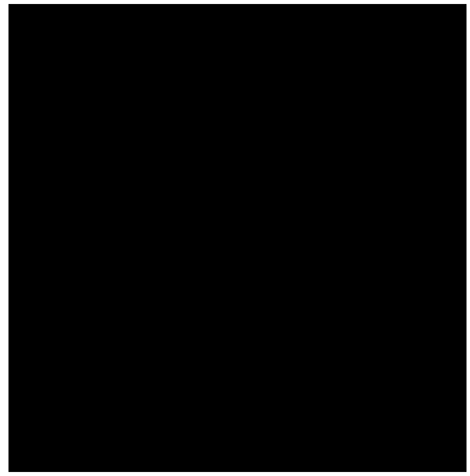


Figure 4.9. Master vane M sharing inlet cap, exit cap, and anti-rotation bar.

Vane M also has the same thickness as the Y2 vane. This allowed the team to use the same set of fasteners used by the Y2 vane. The significant difference between the two vanes is the inlet and exit outer diameters. The different diameters forced the team to design new parts to house vane M. The final assembly for vane M fixture is shown in Figure 4.10 below.

(a) Master vane isometric view (exit).



(b) Master vane isometric view (inlet).

Figure 4.10. Master vane M fixture design.

5 Final Design Summary & Conclusion

To summarize, the team was required to design fixtures to allow five guide vanes from two new engine models to be mounted to the StandardAero's Sonic Flow Rig. The goal of using the Sonic Flow Rig for testing is to increase the first test pass rate of the guide vanes, and the engines housing the vanes. To achieve this, the team took a structured approach defined by the course, which consisted of three phases.

The first phase of the project was project definition. The team met with the client, and learned as much as they could about the problem from the communication and documentation provided by the client. Through this research and communication, the team created explicit objectives and needs for the project in order to provide the deliverables specified by the client and ensure that they would be satisfied with the fixture designs. Once both the team and client were satisfied with these needs and objectives, the project moved to phase 2.

Phase 2 was the concept development phase, composed of concept generation and selection. The team kept in constant communication with the client during the generation phase to get feedback on concepts, and learn from the client's expertise to improve the designs. These concepts were brought to concept selection, where the team systematically scored and selected final design concepts. This was done using a weighted decision matrix. With the final concepts selected, the project moved to phase 3.

Phase 3 focused on finalizing the design for the X1 fixture, and performing a full analysis to justify the decisions made. The CAD models for the other fixtures were also completed, but due to time constraints, the full design analysis was not performed on them. However, the same process performed on the X1 fixture can be copied on the others to make them ready for fabrication. This process is summarized below.

First, multiple design reviews were held with the client to iteratively refine the X1 fixture. This involved inspecting the modelled assembly and parts of the fixture, and applying problem-solving techniques to ensure the design was both functional and feasible. Once the major components had been refined, the team sourced hardware such as fasteners and dowel pins, and attempted to standardize this hardware across all of the fixtures where possible.

With a refined model complete, the process of analysis and justification began. First, a measurement analysis was performed on the X1 vane to establish baseline tolerances for the fixture to ensure that variations from vane to vane will not affect the fixture's performance. Next, an analysis was performed to ensure the fixture would properly seal off airflow to everywhere except through the vane's airfoils. This was essential to increase the first test pass rate of engines through accurate part testing results. The rest of the geometry of each part was then analyzed. The final parts, excluding fasteners, are shown below.

[redacted]

(a) Inlet center cap.

[redacted]

(b) Center gasket.

[redacted]

(c) Exhaust center cap.

Figure 5.1. Center parts of the X1 fixture.

[redacted]

(a) Inlet plate.

[redacted]

(b) Bell mouth gasket.

[redacted]

(c) Exhaust plate.

Figure 5.2. Outer parts of the X1 fixture.

Having resolved the concerns with tolerancing and sealing, the team then sourced an adequate material for the fixture. This allowed for a failure analysis to be performed, which was done two ways. First, the team performed a failure modes and effects analysis to identify potential failure modes and come up with preventative solutions and fault monitoring. The second failure analysis was an FEA simulation on a worst-case scenario for the fixture to confirm that the part can withstand the loading conditions during testing.

The final analysis was a cost analysis, where the team sourced the prices for parts and raw material, and estimated the machining time necessary to fabricate the fixture. This also contained the bill of materials for the X1 fixture.

To aid the client once the fixture has been fabricated, the team included assembly instructions for the X1 fixture. These serve as a guideline for the work instruction that the client will need to produce for the fixture, and include pictures of the 3-D model being assembled for clarity. A render of the final assembly can be seen below.

Figure 5.3. Render of the X1 fixture.

This design satisfies all the needs defined in phase 1 of the project. These include being compatible with the SFR and the tools available, ensuring safety, mounting the vane similar to how it would be mounted in the engine, sealing off airflow to everywhere except the airfoils, and reduced weight and cost.

Finally, the refined models of the remaining fixtures were presented. Again, these fixtures have not been analyzed like the X1 fixture, but the client can follow the same process to ensure they will meet expectations.

[redacted]

(a) X2 fixture.

[redacted]

(b) Y1 fixture.

[redacted]

(c) Y2 fixture.

[redacted]

(d) M fixture.

Figure 5.4. Refined fixture models.

The team was successfully able to produce all deliverables expected by the client. A breakdown of the completed deliverables can be found in Table I.

Table I. Completed deliverables for each vane fixture.

	Vane Fixture				
Deliverable	**X1**	**X2**	**Y1**	**Y2**	**M**
CAD Model	✓	✓	✓	✓	✓
Preliminary Engineering Drawing	✓				
Failure Analysis	✓				
Bill of Materials	✓				

At the beginning of the project, the client provided a baseline cost per fixture between [redacted]. The final estimated cost for the X1 fixture was approximately [redacted]. It is likely that the other fixtures will also fall below the marginal cost as well.

As next steps, the team recommends moving forward with fabrication of the X1 fixture. It is recommended to use the parts sourced from McMaster-Carr since the team has confirmed that they are sufficient for the testing conditions. As for the remaining fixtures, the team recommends following the process performed for the X1 fixture beginning at Section 2.3 to ensure the fixtures will perform adequately.

References

[1] P. Labossiere, "23-StandardAero Sonic Flow Fixture Design," *universityofmanitoba.desire2learn.com*, 2021. [Online]. Available: https://universityofmanitoba.desire2learn.com/d2l/le/content /443426/viewContent/2558636/View?ou=443426. [Accessed: Sept. 10, 2021].

[2] StandardAero, private communication, [Engine Model X & Y] Vane Matching with Sonic Flow [PowerPoint Slides], Sept. 21.

[3] A. Pauch et al, private communication, Sept. 28, 2021.

[4] StandardAero, "About Us," *standardaero.com*, 2017. [Online]. Available: http://www.standardaero.com/AboutUs.aspx. [Accessed: Sept. 28, 2021].

[5] Federal Aviation Administration, "Engines and Propellers - Regulations & Policies," *faa.gov*, Feb. 18, 2020. [Online]. Available: https://www.faa.gov/aircraft/air_cert/design_approvals/engine_prop /engine_prop_regs/. [Accessed: Dec. 5, 2021].

[6] H. Saravanamuttoo et al, *Gas Turbine Theory*, 7th ed. United Kingdom: Pearson, 2017. [E-book] Available: VitalSource Bookshelf.

[7] StandardAero, private communication, Sept. 21, 2021.

[8] P. Kotelko et al. "Answers to Questions Part II" Personal email (Sept. 27, 2021).

[9] P. Kotelko et al. "Answers to Questions - Kickoff Meeting" Personal email (Sept. 21, 2021).

[10] P. Labossiere. "RE: Mech 4860 - PDR" Personal email (Sept. 27, 2021).

[11] Aviation Maintenance Publishers, *Standard Aviation Maintenance Handbook*, 1985.

[12] A. Pauch et al, private communication, Oct. 8, 2021.

[13] E. Evangelista, "Gas flow area measurement," U.S. Patent 6,148,677, 18 Mar., 1999.

[14] Dassault Systemes, *SolidWorks*. [Download]. Paris, France: 2020.

[15] StandardAero, "[Swinging Plate Drawings]", unpublished.

[16] Y. Sanchak. "RE: [Engine Model X] 1st stg NGV - pictures and details" Personal email (Oct. 21, 2021).

[17] A. Pauch et al, private communication, Nov. 2, 2021.

[18] A. Pauch et al, private communication, Nov. 9, 2021.

[19] A. Pauch et al, private communication, Nov. 16, 2021.

[20] P. Kotelko et al. "Fasteners Loaded in Tension" Personal email (Dec. 4, 2021).

[21] A. Pauch et al, private communication, Nov. 23, 2021.

[22] A. Pauch et al, private communication, Oct. 12, 2021.

[23] F. P. Beer, *Mechanics of Materials*. New York, NY: McGraw-Hill, 2015.

[24] McMaster-Carr, "High-Strength Steel Threaded Rod," *mcmaster.com*, Dec. 30, 2020. [Online]. Available: https://www.mcmaster.com/90322A642/. [Accessed: Dec. 5, 2021].

[25] Carbide Depot, "Tap Chart - UNC/UNF Threads," *carbidedepot.com*, N.D. [Online]. Available: http://www.carbidedepot.com/formulas-tap-standard.htm. [Accessed: Dec. 5, 2021].

[26] Industrial Fasteners Institute, "Technical Bulletins - Thread Inspecting," *indfast.org*, 2007. [Online]. Available: https://www.indfast.org/shop/product_detail.asp?cat=24&m=0&id=478. [Accessed: Dec. 3, 2021].

[27] McMaster-Carr, "Mil. Spec. 18-8 Stainless Steel Hex Nut," *mcmaster.com*, Dec. 30, 2020. [Online]. Available: https://www.mcmaster.com/91240A411/. [Accessed: Dec. 5, 2021].

[28] P. Kotelko et al, private communication, Nov. 29, 2021.

[29] McMaster-Carr, "Aluminum Mil. Spec. Washer," *mcmaster.com*, Dec. 30, 2020. [Online]. Available: https://www.mcmaster.com/94589A628/. [Accessed: Dec. 5, 2021].

[30] E. Oberg et al., *Machinery's Handbook*: South Norwalk, CT: Industrial Press, Inc., 2020.

[31] McMaster-Carr, "Dowel Pin," *mcmaster.com*, Dec. 30, 2020. [Online]. Available: https://www.mcmaster.com/98381A550/. [Accessed: Dec. 5, 2021].

[32] A. Pauch. "RE: [Engine Model X] Vane Mounting to the Sonic Flow Rig" Personal email (Oct. 13, 2021).

[33] A. Pauch. "RE: [Engine Model X] Vane Mounting to the Sonic Flow Rig" Personal email (Dec. 1, 2021).

[34] P. Kotelko et al. "RE: [Engine Model X] 1st stg NGV - pictures and details" Personal email (Dec. 3, 2021).

[35] Global O-ring and Seal, "O-Ring Groove Design," *globaloring.com*, 2021. [Online]. Available: https://www.globaloring.com/o-ring-groove-design/. [Accessed: Dec. 3, 2021].

[36] McMaster-Carr, "Oil- and Abrasion-Resistant Polyurethane O-Ring," *mcmaster.com*, Dec. 30, 2020. [Online]. Available: https://www.mcmaster.com/9558K264/. [Accessed: Dec. 3, 2021].

[37] P. Kotelko et al, private communication, Dec. 7, 2021.

[38] McMaster-Carr, "Abrasion-Resistant Polyurethane O-Ring," *mcmaster.com*, Dec. 30, 2020. [Online]. Available: https://www.mcmaster.com/8716K62-8716K35/. [Accessed: Dec. 4, 2021].

[39] StandardAero, "[Fixture Drawings]", unpublished.

[40] P. Kotelko et al, private communication, Dec. 1, 2021.

[41] Nord-Lock Group, "The Experts: Thread engagement in a tapped hole," *nord-lock.com*, 2021. [Online]. Available: https://www.nord-lock.com/insights/bolting-tips/2015/the-experts-thread-engagement-in-a-tapped-hole/. [Accessed: Dec. 3, 2021].

[42] W.W. Grainger, Inc., "Cylindrical, Socket Head Cap Screw, 1/2"-13, Steel, Alloy Steel, Black Oxide, 1 1/4 in Length," *grainger.com*, 2021. [Online]. Available: https://www.grainger.com/product/GRAINGER-APPROVED -Cylindrical-22TR57. [Accessed: Dec. 1, 2021].

[43] P. Labossiere. "RE: Mech 3502 - Project 2" Personal email (Nov. 26, 2020).

[44] P. Labossiere. "RE: Mech 4860 - FEA Simulation" Personal email (Dec. 2, 2021).

[45] R.G. Budynas and J.K. Nisbett, *Shigley's Mechanical Engineering Design, 11th Edition.* Arlington, TX: McGraw-Hill Education, 2020.

[46] Dassault Systemes, "Buckling Load Factor," *help.solidworks.com*, 2021. [Online]. Available: https://help.solidworks.com/2012/english/SolidWorks/cworks/Buckling_Load_Factor.htm. [Accessed: Dec. 1, 2021].

[47] McMaster-Carr, "Multipurpose 6061 Aluminum," *mcmaster.com*, Dec. 30, 2020. [Online]. Available: https://www.mcmaster.com/1610T78/. [Accessed: Dec. 4, 2021].

[48] McMaster-Carr, "Multipurpose 6061 Aluminum," *mcmaster.com*, Dec. 30, 2020. [Online]. Available: https://www.mcmaster.com/8974K98-1610T45/. [Accessed: Dec. 4, 2021].

Appendices

A Supplementary Info From Previous Phases

A.1 Phase 1

The following tables list the technical specifications defined for the three other guide vanes initially supplied to the team.

Table I. Metrics and their associated marginal & target values for X2.

[redacted]

Table II. Metrics and their associated marginal & target values for Y1.

[redacted]

Table III. Metrics and their associated marginal & target values for Y2 & M.

[redacted]

A.2 Phase 2

This table is the outline used for concept selection of each vane fixture during phase 2. Scores were given as a percent value, chosen individually for each vane, and summing to 100%.

Table IV. Weighted decision matrix outline

		Options					
		Concept 1		**Concept 2**		**Concept 3**	
Criteria	**Weighting**	**Score**	**Total**	**Score**	**Total**	**Score**	**Total**
Manufacturability							
Mass							
Practicality of Use							
Ability to Seal							
	TOTAL:						

B Groove Dimension Calculations

Calculations performed in this appendix are in accordance with SAE AS6235 [6], [7].

There are two dimensions which define a groove: the gland depth, AF, and groove width, AG, both of which are called out in the figure below [7], [8].

[redacted]

Figure B.1. Gland depth and groove width called out.

The O-ring is housed in a face seal configuration subjected to external pressure, meaning the groove width and gland depth are dependent on three O-ring properties [7], [8]:

1. Size.
2. Squeeze.
3. Stretch.

Size is based on the o-ring selected and is fixed throughout the calculations. For the fixture, a [redacted] polyurethane o-ring with the following dimensions was selected:

Table I. O-ring dimensions.

[redacted]

The squeeze of an o-ring is defined as the difference between the stretched width of the o-ring when compressed and the gland depth [7]. Often, squeeze is calculated as a percent relative to the stretched width as follows [7]:

$$Sq = \frac{W - AF}{W}(100) \tag{B.1}$$

SAE standards specify that the target squeeze range of an o-ring should be between 10-20 percent for a -1XX O-ring [7]. As there are tolerances involved in the manufacturing of both the groove and o-ring, the maximum and minimum squeeze must be computed. Resources such as [8] have re-defined gland depths based on dash number to meet the squeeze specifications. In the case of [redacted], the gland depth was given as [redacted] [8]. Since this is quite a tight tolerance, the team sought to expand the range without shifting out of the squeeze range. The largest tolerance attainable was found to be [redacted], corresponding to a gland depth of [redacted]. The maximum and minimum squeeze are then,

To determine the groove width, the fill percentage, $Fill$, must be known. $Fill$ is defined as the volume of the o-ring, V_O, relative to that of the groove, V_G [7]. SAE recommends that the fill be no greater than 85 percent thus, using the following equations, the minimum groove width may be solved for:

$$V_{O,max} = \frac{\pi^2}{4}(ID_{max} + W_{max})W_{max}^2 \tag{B.4}$$

$$V_{O,min} = \frac{\pi^2}{4}(ID_{min} + W_{min})W_{min}^2 \tag{B.5}$$

$$V_{G,max} = \frac{\pi}{4}(ODG_{max}^2 - IDG_{min}^2)AF_{max} \tag{B.6}$$

$$V_{G,min} = \frac{\pi}{4}(ODG_{min}^2 - IDG_{max}^2)AF_{min} \tag{B.7}$$

$$Fill_{max} = \frac{V_{O,max}}{V_{G,min}} \tag{B.8}$$

where ODG is the face groove outer diameter and IDG is the face groove inner diameter [7]. As per [8], it is best practice to design IDG to be larger than the o-ring inner diameter, ID, by no greater than 5 %. IDG was then set to have a minimum value of [redacted]. Parker Hannifin states that the tolerance on IDG should be +0.060 inches thus, the maximum IDG was found to be [redacted] [11].

ODG is a function of IDG and AG as per the following equation [7]:

$$ODG_{max} = IDG_{max} + AG_{max} \tag{B.9}$$

$$ODG_{min} = IDG_{min} + AG_{min} \tag{B.10}$$

Using equation B.8, the minimum groove volume may be computed which allows for computation of the minimum ODG. To ensure SAE standards are met, the team set the maximum fill to be 80 %.

The client specified a tolerance of +/- 0.005 inches where possible and thus, AG_{max} was found to be [redacted]. Computing the minimum fill,

With both the squeeze and fill requirements met, the final parameter to check is percent o-ring stretch, *Stretch*, which results from the o-ring inner diameter being less than that of the groove [7], [8]. SAE standards state that the stretch should be no greater than 5 % for a face seal subjected to external pressure. The following equations may be used to compute the maximum and minimum stretch [7]:

$$Stretch_{max} = \frac{IDG_{max} - ASID_{min}}{ASID_{min}}(100) \tag{B.19}$$

$$Stretch_{min} = \frac{IDG_{min} - ASID_{max}}{ASID_{max}}(100) \tag{B.20}$$

where $ASID$ is the squeezed o-ring inner diameter and is equal to [7]:

$$ASID_{max} = ID_{nom} + W_{nom} - L_{min} \tag{B.21}$$

$$ASID_{min} = ID_{nom} + W_{nom} - L_{max} \tag{B.22}$$

where L is the cross-sectional length of the o-ring after being squeezed [7]. L may be found using the following formulae:

$$L_{max} = \frac{\pi}{4AF_{min}}(W_{nom}^2 - AF_{min}^2) + AF_{min} \tag{B.23}$$

$$L_{min} = \frac{\pi}{4AF_{max}}(W_{nom}^2 - AF_{max}^2) + AF_{max} \tag{B.24}$$

Substituting in the appropriate values:

$$L_{max} =$$

$$L_{min} =$$

The maximum and minimum $ASID$ are then,

$$ASID_{max} = \text{[redacted]} \tag{B.27}$$

$$ASID_{min} = \text{[redacted]} \tag{B.28}$$

Finally, calculating the maximum and minimum stretch:

$$Stretch_{max} =$$

$$Stretch_{min} =$$

The groove width and gland width calculated satisfy SAE AS6235 for a face seal subjected to external pressure with a dash number of -1XX [6], [7].

C Computational Fluid Dynamics

To determine the torque exerted by the vane onto the fixture, an internal flow simulation was done in SolidWorks. Two tubes were extrude from the part. One at the bell mouth to serve as the inlet and one at the end of the guide vanes case to serve as the outlet.

[redacted]

Figure C.1. Internal Flow Simulation Domain

Assuming standard air properties, the inlet velocity of the fluid was determined with the gauge pressure of the air in the sonic flow rig pressure tank.

$$V = \sqrt{\frac{2P_{gauge}}{\rho}} \tag{C.1}$$

$$V = \sqrt{\frac{2\text{[redacted]}}{1.225}} = \text{[redacted]} \tag{C.2}$$

The outlet condition was simply atmospheric pressure, taken as 101.3 KPa.

The objective of this simulation was to determine the pressure distribution across the vane as result of the flow velocity around the airfoils inside. Below is an image visualizing what this flow looks like. Blue indicates the lower velocity flow corresponding to higher pressure and green indicates the higher velocity flow which corresponds to lower pressure. A key feature of this visualization is the change in the direction of the flow velocity vectors, indicating a change in momentum of the fluid relative to the flow path. This results in a force that is exerted on the vanes by the fluid and thus a torque that is transmitted through the solid body as the force acts a distance and normal to the guide vane's centre of mass.

Figure C.2. Simulation flow trajectories.

Once the simulation was done, the pressures probed at several locations along the airfoil of the vane. The surface area of each of these sections was known and the force on each of these sections was approximated by treating each section as a panel and taking the product of the area and pressure. The force on each panel was considered to be normal to the tangent plane coincident at each panel's midpoint. The geometry of the airfoil was then used to determine to angles of each of these forces and subsequently the resultant force was calculated from the summation of the forces normal and tangent to the plane representing the front most panel on top.

Figure C.3. Visualization of single guide vane airfoil and panels.

The colours in the above figure correspond to the pressure on the surface of each section. In order of increasing pressure the corresponding colours are blue, green, yellow then red.

Summation of forces on aerodynamic panels tangent the plane which is tangent to the top-front panel.

$$F_t = \sum P_i A_i cos(\alpha_i) \tag{C.3}$$

Summation of forces on aerodynamic panels normal the plane tangent to the top-front panel.

$$F_n = \sum P_i A_i sin(\alpha_i) \tag{C.4}$$

Resultant Force

$$F_r = \sqrt{F_n^2 + F_t^2} \tag{C.5}$$

Direction of resultant force relative to top-front panel.

$$\phi = arctan\left(\frac{F_n}{F_t}\right) \tag{C.6}$$

Knowing the angle between the first panel and the plane perpendicular to the flow allowed the lever force to be calculated and subsequently the torque as a contribution from a single airfoil. This is then multiplied by the number of airfoils in the guide vane to get the total torque as a result of the flow.

$$F_{lever} = F_r \sin(\theta - \phi) \tag{C.7}$$

$$T = F_{lever} \cdot r_{vane} \tag{C.8}$$

$$T_{total} = T(\text{[redacted]}) = \text{[redacted] [N-m]} \tag{C.9}$$

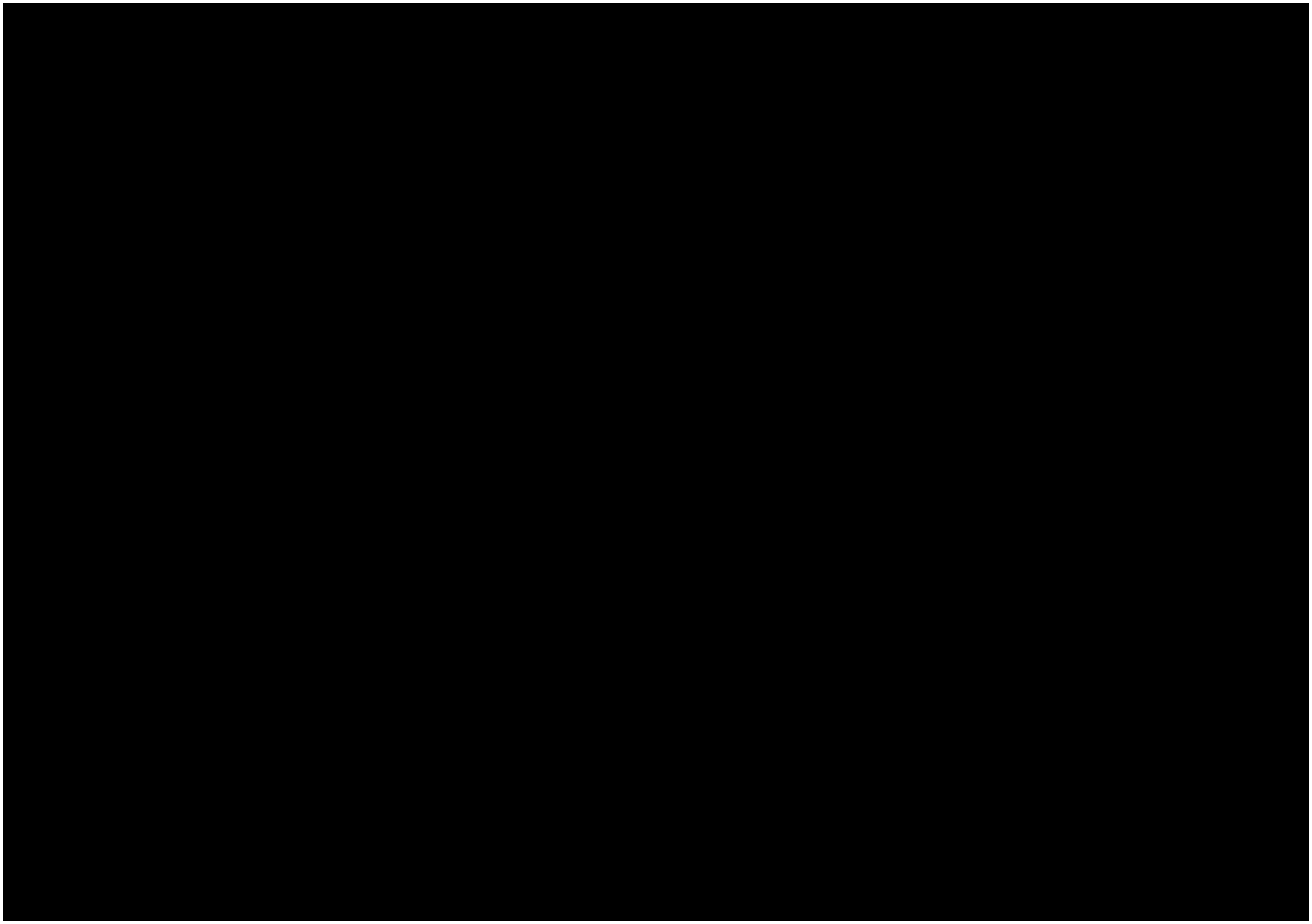
This preliminary CFD analysis served to provide a material requirement for the pins used to restrict any rotation of the vane. In other words, the pins used for anti-rotation must be able to withstand shear force generated from this torque without shearing or otherwise deforming. Failure to do so could result in a catastrophic failure of the vane fixture; posing a significant safety risk.

The boundary conditions of the simulation are listed below.

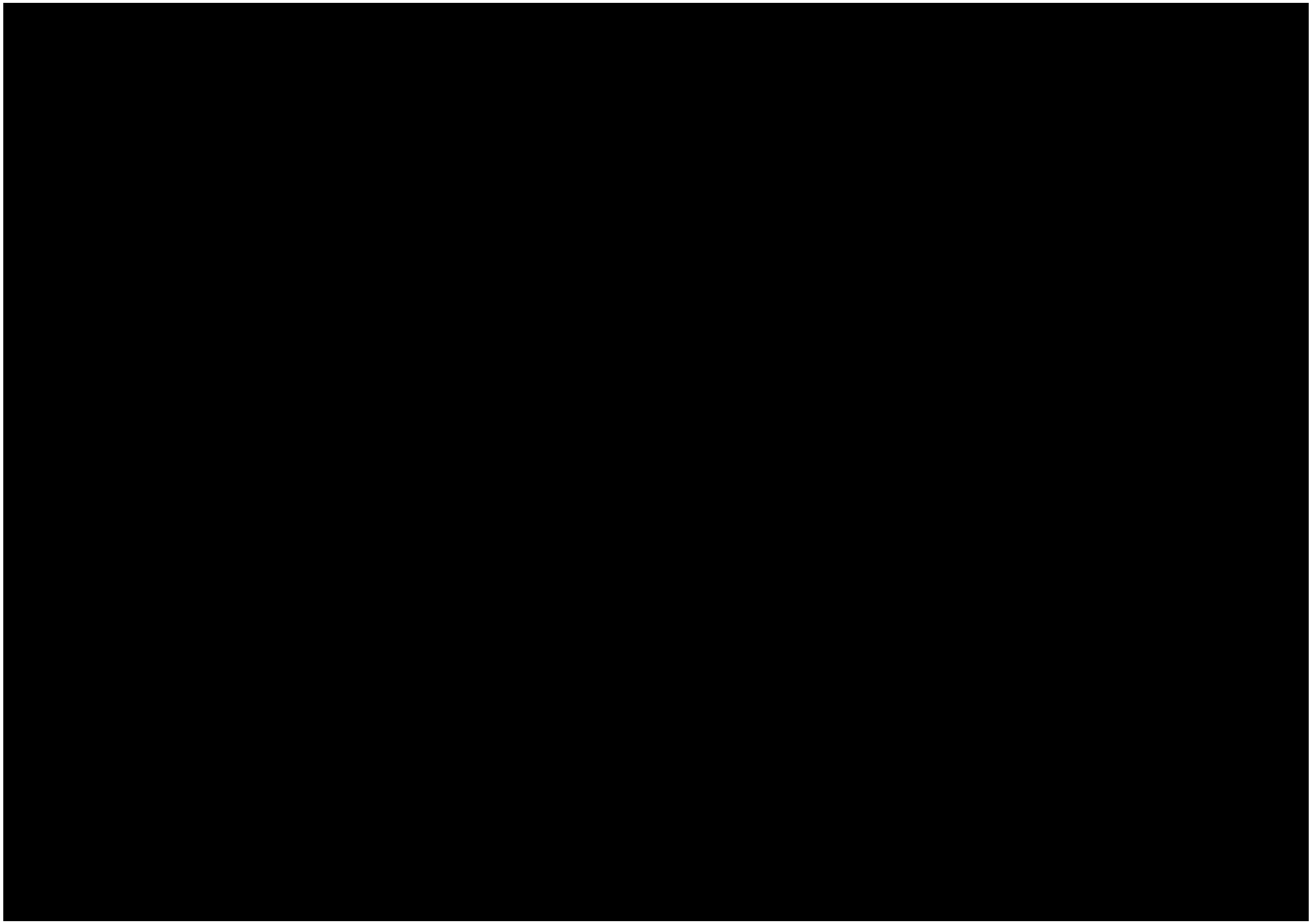
D Engineering Drawings

Please find the engineering drawings prepared for the X1 fixture on the following pages.

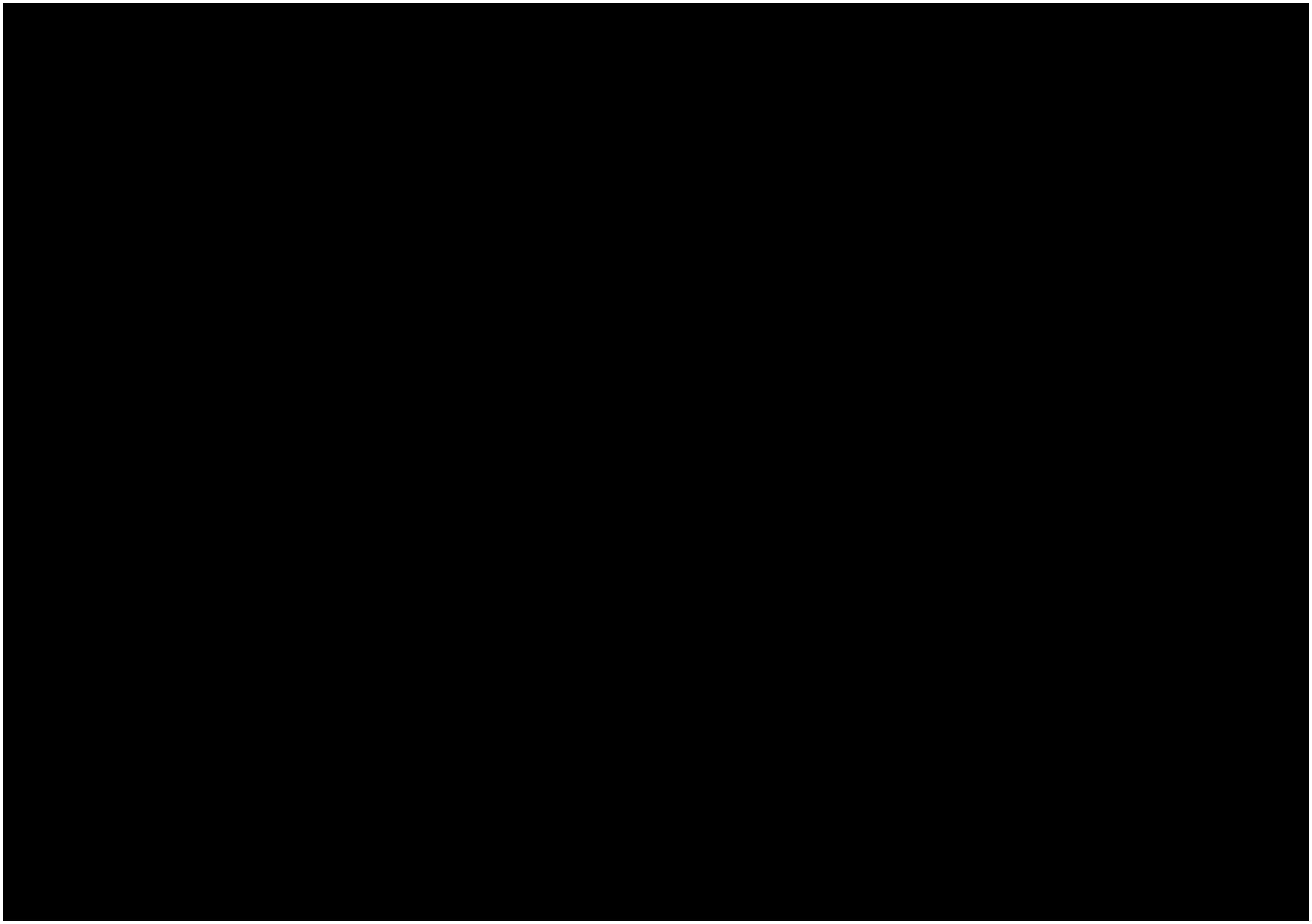
D.1 Inlet Plate



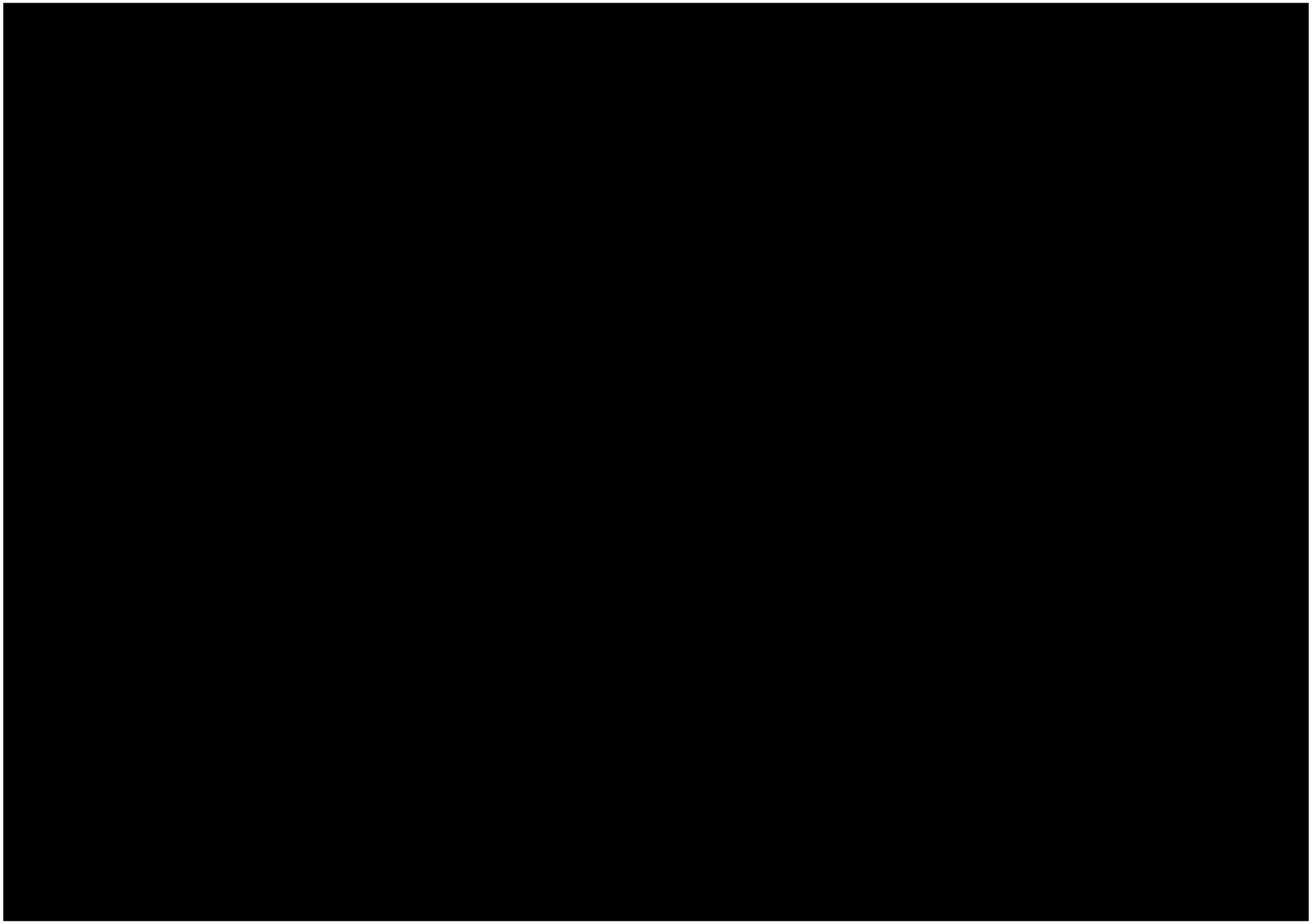
D.2 Exhaust Plate



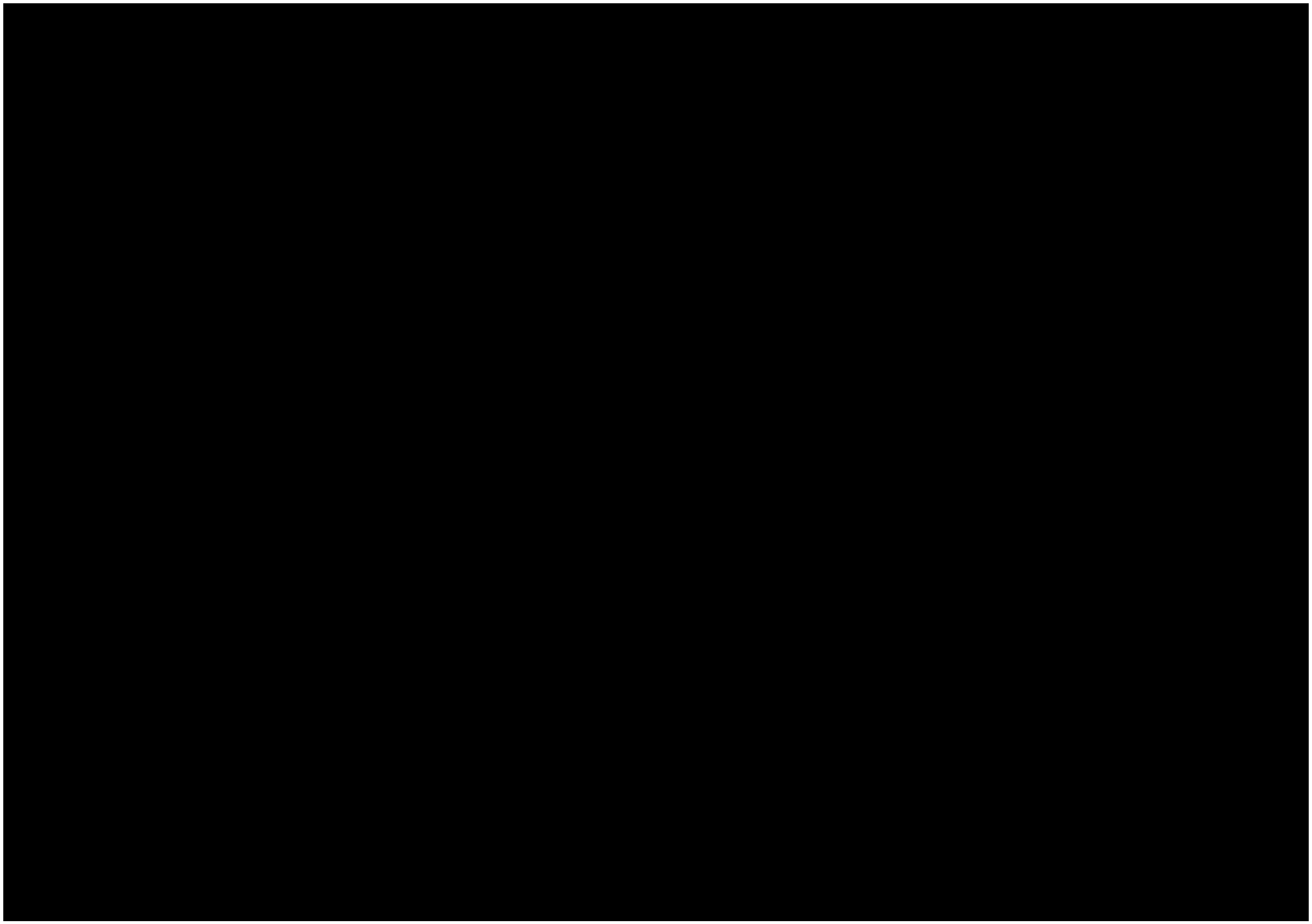
D.3 Inlet Center Cap



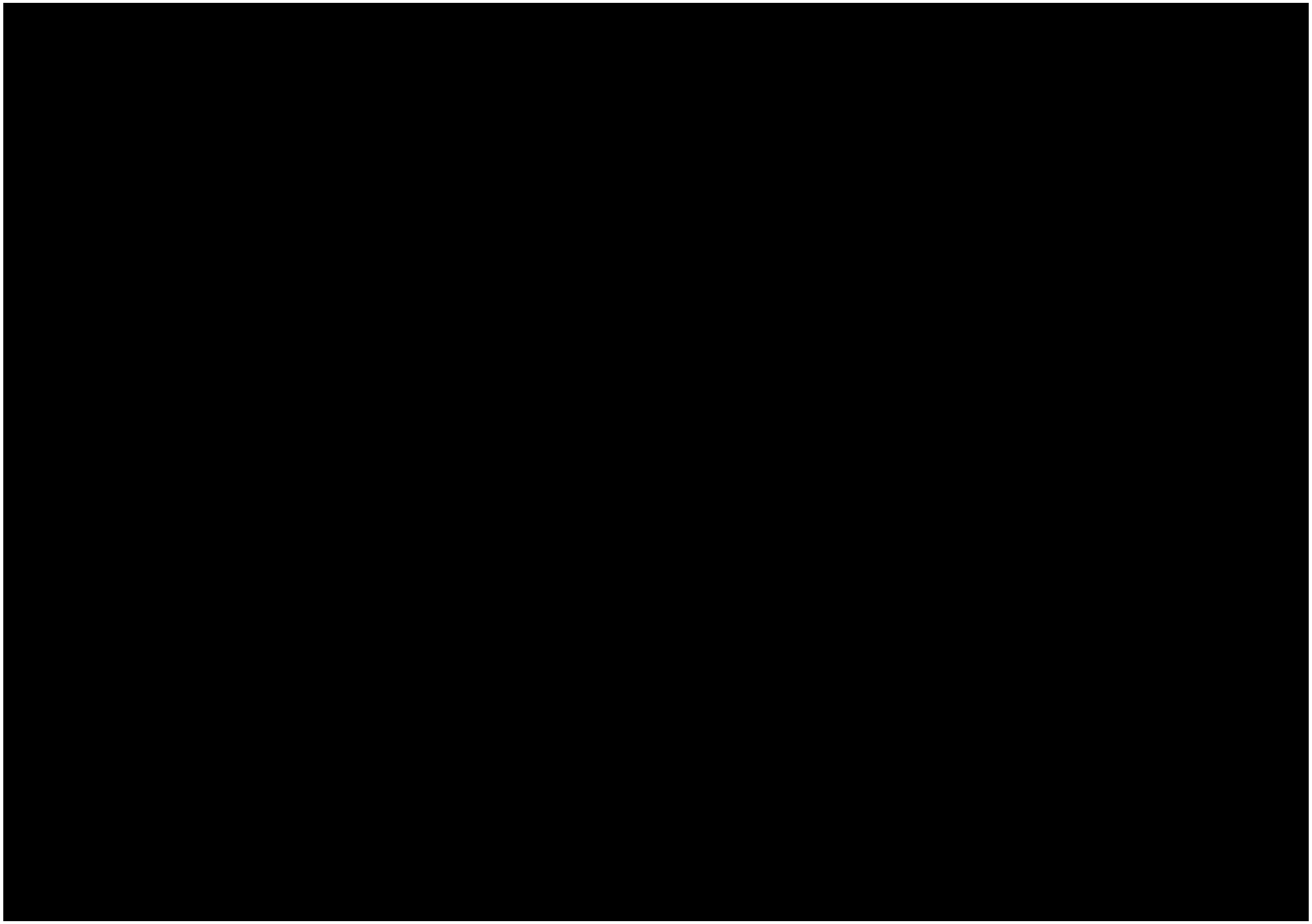
D.4 Exhaust Center Cap



D.5 Bell Mouth Custom Gasket



D.6 Inlet Center Cap Custom Gasket



E Appendix References

References

[1] StandardAero, private communication, [Engine Model X & Y] Vane Matching with Sonic Flow [PowerPoint Slides], Sept. 21.

[2] P. Kotelko et al. "Answers to Questions Part II" Personal email (Sept. 27, 2021).

[3] P. Kotelko et al. "Answers to Questions - Kickoff Meeting" Personal email (Sept. 21, 2021).

[4] P. Labossiere. "RE: Mech 4860 - PDR" Personal email (Sept. 27, 2021).

[5] A. Pauch et al, private communication, Sept. 28, 2021.

[6] P. Kotelko et al, private communication, Dec. 1, 2021.

[7] P. Labossiere. "RE: Mech 3502 - Project 2" Personal email (Nov. 26, 2020).

[8] P. Labossiere. "RE: Mech 4860 - FEA Simulation" Personal email (Dec. 2, 2021).

[9] F. P. Beer, *Mechanics of Materials*. New York, NY: McGraw-Hill, 2015.

[10] F. McQuiston et al., *Heating, Ventilation and Air Conditioning: Analysis and Design.*. Hoboken, NJ: Wiley, 2004.

[11] R.G. Budynas and J.K. Nisbett, *Shigley's Mechanical Engineering Design, 11th Edition*. Arlington, TX: McGraw-Hill Education, 2020.

[12] E. Oberg et al., *Machinery's Handbook*: South Norwalk, CT: Industrial Press, Inc., 2020.

[13] Omnia Manufacturing, "How to Calculate Fastener Minimum Thread Engagement Length," *omniamfg.com*, 2019. [Online]. Available: https://www.omniamfg.com/mechanical/2020/10/3/how-to-calculate-fastener-minimum-thread-engagement-length. [Accessed: Dec. 4, 2021].

[14] McMaster-Carr, "High-Strength Steel Threaded Rod," *mcmaster.com*, Dec. 30, 2020. [Online]. Available: https://www.mcmaster.com/90322A642/. [Accessed: Dec. 5, 2021].

[15] D. Singer, "B7 vs. Grade 8," *aatprod.com*, Jun. 17, 2020. [Online]. Available: https://www.aatprod.com/b7-vs-grade-8/. [Accessed: Dec. 4, 2021].

[16] eFunda, "Unified Screw Threads, Standard Series," *efunda.com*, 2021. [Online]. Available: https://www.efunda.com/DesignStandards/screws/unified.cfm. [Accessed: Dec. 4, 2021].

[17] Carbide Depot, "Tap Chart - UNC/UNF Threads," *carbidedepot.com*, N.D. [Online]. Available: http://www.carbidedepot.com/formulas-tap-standard.htm. [Accessed: Dec. 5, 2021].

[18] Society of Automotive Engineers, "SAE AS6235," *sae.org*, Dec. 19, 2015. [Online]. Available: https://www.sae.org/standards/content/as6235a/. [Accessed: Dec. 3, 2021].

[19] Global O-ring and Seal, "O-Ring Groove Design," *globaloring.com*, 2021. [Online]. Available: https://www.globaloring.com/o-ring-groove-design/. [Accessed: Dec. 3, 2021].

[20] McMaster-Carr, "Oil- and Abrasion-Resistant Polyurethane O-Ring," *mcmaster.com*, Dec. 30, 2020. [Online]. Available: https://www.mcmaster.com/9558K264/. [Accessed: Dec. 3, 2021].

[21] Parker Hannifin, "O-Ring Selector," *divapps.parker.com*, 2021. [Online]. Available: https://divapps.parker.com/divapps/oring/ORingSelector?cntry=us/17563lang=enregion=naelqTrackId=6249D354877 [Accessed: Dec. 3, 2021].

[22] Parker Hannifin, "Parker O-Ring Handbook," *parker.com*, 2021. [Online]. Available: https://www.parker.com/Literature/O-Ring%20Division%20Literature/ORD%205700.pdf. [Accessed: Dec. 3, 2021].

[23] P. Kotelko et al, private communication, Nov. 29, 2021.