1.0 Executive Summary

1.1 Summary

Formula One racing is an exceedingly competitive

1.2 Conclusions

1.3 Recommendations

2.0 Introduction

Formula Racing is one of the largest spectating racing sports in the world. As part of this heavy competition, every avenue is explored to decrease each lap time for the driver. The current approach is to increase the horsepower and or rotational speed of the engine. This approach has been hampered as Formula 1 has made several specifications to limit the horsepower available to the drivers, all to increase the safety of the race. The proposed method of decreasing lap-time is therefore to increase the efficiency of the engine, as this would decrease the need to make pit stops for refueling. This can be accomplished by increasing the internal temperature of the engine. The purpose of this report is to outline a method of increasing the efficiency of a Formula 1 engine with the end goal being selecting a new material to be used as a piston as well as a better insulator for the piston sleeve, both of which will increase the internal temperature of the engine. The report will also outline preliminary findings including the calculations and simulation of selected materials. This report contains a SECTION

3.0 Background of Problem

Formula One racing grew out of the European Grand Prix motor racing, getting its official start in 1946. From then on, there was a never ending race to achieve higher and higher horsepower in the name of decreased lap time. This ended in 1994 after a sequence of driver deaths lead the Fédération Internationale de l'Automobile (FIA) to severely restrict the horsepower output of the F1 engines.

These restriction have led to each individual car having the same horsepower output. In the drive to decrease lap times, the next logical step will be to increase the efficiency of the engine. By increasing the efficiency of the engine, each gallon of gasoline will be able to propel the car longer decreasing the need for the driver to refuel. To increase the efficiency, we are required to pick materials that have a higher heat capacity as well as can insulate the individual cylinders.

4.0 Theory of Design

The modern automobile engine is at its core simply a heat engine. Because of this, 100% efficiency can never be obtained. In a perfect engine, called a Carnot engine, the efficiency can be determined from the following equation:

$$\eta_{Th} = 1 - \frac{\dot{Q}_{out}}{\dot{Q}_{in}}$$

Where $\dot{Q}_{in}$ is the heat transfer due to combustion and $\dot{Q}_{out}$ is the heat transfer lost to the atmosphere. From this equation, you can see that in order to make the engine more efficient we would need to decrease the amount of heat loss to the surroundings. One big way to do this would be to reduce the heat leaving the engine through the engine block. In order to accomplish this, we need to find two materials, one for the piston and one to insulate the cylinder wall. Because of strict FIA rules on displacement, the overall geometry of the piston and cylinder sleeve. These dimensions can be seen in Appendix XX. Because of this limitation on the design, we must therefore work within existing parameters to find a material that can withstand the loading as well as provide to be a better insulator to the combustion process.

5.0 Material Selection

5.1 Piston

Javier, I need some stuff for here

5.2 Insulator

Need Some way to put some stuff in here.

6.0 Thermal Analysis

6.1 Theory of Four Stroke Engines

Formula One Racing implements internal combustion engines, which is common among cars, trucks, motorcycles, aircraft, construction machinery and many others. These engines run on a four-stroke cycle. The four strokes refer to intake, compression, combustion (power), and exhaust strokes that occur during two crankshaft rotations per working cycle of the gasoline engine and diesel engine.

The cycle begins at Top Dead Center (TDC), when the piston is farthest away from the axis of the crankshaft. A stroke refers to the full travel of the piston from Top Dead Center (TDC) to Bottom Dead Center (BDC). A description of each stroke follows:

1. INTAKE stroke: On the intake or induction stroke of the piston, the piston descends from the top of the cylinder to the bottom of the cylinder, reducing the pressure inside the cylinder. A mixture of fuel and air is forced by atmospheric (or greater) pressure into the cylinder through the intake port. The intake valve(s) then close.

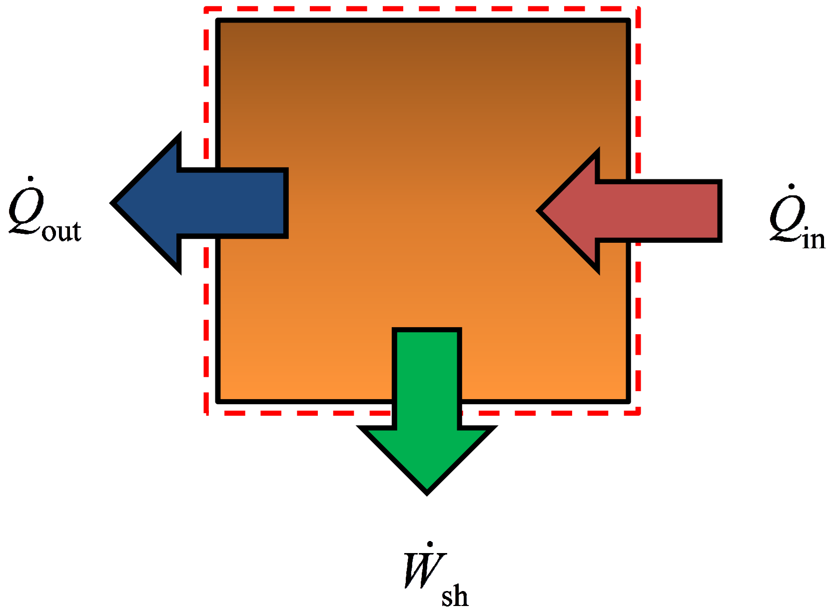
2. COMPRESSION stroke: With both intake and exhaust valves closed, the piston returns to the top of the cylinder compressing the fuel-air mixture. This is known as the compression stroke.

3. POWER stroke: While the piston is close to Top Dead Center, the compressed air–fuel mixture is ignited, usually by a spark plug. The resulting massive pressure from the combustion of the compressed fuel-air mixture drives the piston back down toward bottom dead center with tremendous force. This is known as the power stroke, which is the main source of the engine's torque and power.

4. EXHAUST stroke: During the exhaust stroke, the piston once again returns to top dead center while the exhaust valve is open. This action evacuates the products of combustion from the cylinder by pushing the spent fuel-air mixture through the exhaust valve(s).

6.2 Thermal Analysis

The four stroke process happens so rapidly we can model it as a system with heat being transferred in from combustion, producing shaft power due to rotation of the crankshaft, and heat being transferred out from material heat conduction as well as exhaust. In other words, the system is at steady state. The following diagram depicts the preceding:



Due to the first law of thermodynamics we can algebraically describe the system using the following energy equation:

$$\frac{dE}{dt} = \dot{m}j_1 - \dot{m}j_2 + \dot{Q} - \dot{W}_{ext}$$

Since the system is at steady state we can simplify the first law and show that the net heat transfer produces an equal value to the power output.

$$\dot{Q} = \dot{W}_{ext}$$

In order to incorporate a more efficient engine an attempt will be made to better insulate the cylinder. This will allow for a greater temperature to be stored during the combustion stroke and provide more pressure during the power stroke, ultimately producing more work on the crankshaft. To put it simply, an attempt will be made to reduce the heat being transferred out of the system in order to increase the internal energy inside of the system. This is shown by the following thermal efficiency equation:

where η is the thermal efficiency, $\dot{W}_{\text{ext}}$ is the external work of the system done by the crankshaft, $\dot{Q}_{\text{in}}$ is the average heat being transferred into the system from the spark plug, and $\dot{Q}_{\text{out}}$ is the heat being transferred out of the system through exhaust, material conductivity, and other losses.

6.3 Current Thermodynamics of Formula One

Through researching Formula One engines we found that the average F1 engine runs at roughly 33% thermal efficiency. Meaning that one-third of the heat produced by the combustion stroke generates as work through the piston rotating the crankshaft (which is relatively good considering commercial engines run at about 20% thermal efficiency).The following will demonstrate the power output of the Formula One engine as it runs at 19,250 rpm (maximum power) and provide numerical values for the heat being transferred.

$$\dot{W}_{\text{ext}} = \dot{W}_{\text{sh}} = 2\pi\omega T = 2\pi\left(\frac{19250\,\text{rpm}}{60\,\text{s}}\right)0.279\,\text{kN}\cdot\text{m}$$

$$\dot{W}_{\text{sh}} = 562.4\,\text{kW}$$

Since two-thirds of the heat produced by the engine is lost after the full four stroke cycle we can conclude the following:

$$\dot{Q}_{\text{in}} - \frac{2}{3}\dot{Q}_{\text{in}} = \dot{W}_{\text{sh}} = 562.4\,\text{kW}$$

so

$$\dot{Q}_{\text{in}} = 1{,}687.2\,\text{kW} \qquad \dot{Q}_{\text{out}} = 1{,}124.8\,\text{kW}$$

Let us take a closer look at the energy transferred *out* of the system. We know that we are losing heat through two primary mediums, the exhaust and the cylinder walls. All other losses are negligible compared to these. From our research we found that a common cylinder used in Formula One racing is an aluminum alloy with silicon.

$$\dot{Q}_{\text{out}} = \frac{k\left(T_{\text{inside}} - T_{\text{ambient}}\right)\text{Area}}{\text{thickness}} = \frac{120\,\text{W/m}^{-1}\cdot\text{K}^{-1}\left(500^\circ\,\text{C} - 30^\circ\,\text{C}\right)0.01225\,\text{m}^2}{0.005\,\text{m}}$$

$$\dot{Q}_{\text{out, mat}} = 138\,\text{kW}$$

This will

7.0 Material Analysis

Now that we have found a material to use in both the piston and cylinder sleeve, the question then becomes whether it can handle the complex loading of an internal combustion engine. The goal of this section is to prove whether or not it can handle this through both hand calculations as well as through the Finite Element Analysis capability in SolidWorks. This section contains an Internal Pressure, Thermal Stress, Failure Theory, Cyclic Loading, Crack Length, and Material Conclusion section.

7.1 Internal Pressure

The largest stresses felt on both the piston as well as the piston sleeve will be due to the internal pressure due to the combustion of the gasoline. This combustion will be cyclic in nature, and therefore will cause the internal stresses to vary with time. In order to gain an accurate estimation of the internal pressure, we need to use two methods to find the internal pressure: the pressure as a result of the acceleration as well as the pressure does to the ignition temperature. Both of these pressures should be nearly equal to each other.

As there are a variety of factors that can affect the internal pressure of the cylinder, we have to make several assumptions to gain a model. We will first be modeling it as if the engine is rotating at a constant speed. This speed will correspond to the max torque of the engine and is equal to 19,250 RPM.

7.1.1 Acceleration

One method to approximate the internal pressure of the cylinder would be to find the acceleration of the piston. The max acceleration will happen at either extreme of the piston cycle, with the largest magnitude happening at Top Dead Center. Once we have the acceleration of the piston, we can approximate the internal pressure by replacing this force by a distributed load on the top of the piston. The analysis of this can be found in Appendix 1. The tabulated results can be found below in Table Y. This table represents only the maximum pressure, the actual loading will by cyclic in nature.

Engine Speed	Max Acceleration	Max Force	Max Internal Pressure
19,250 RPM	9843.9g	21.245kN	2.8165MPa

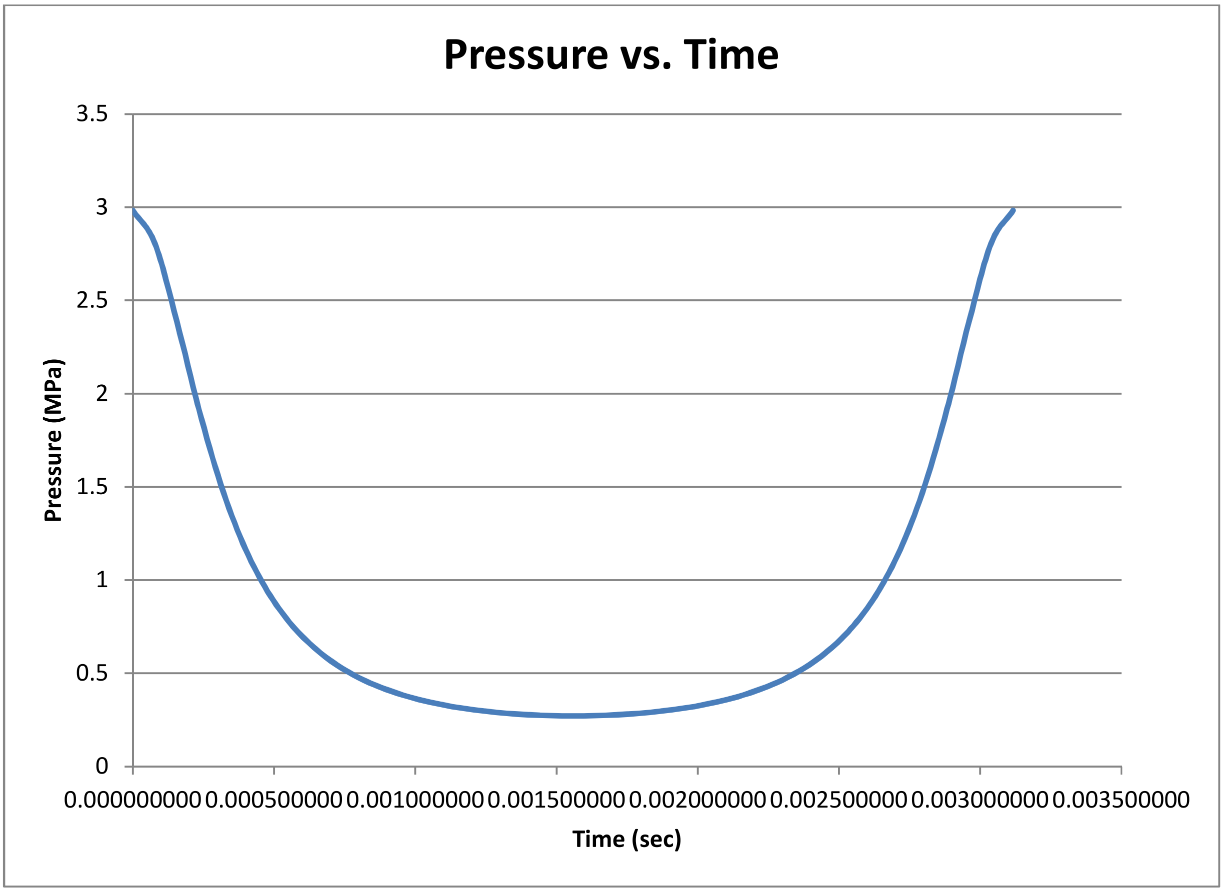
7.1.2 Ignition Temperature

One method to correlate the ignition process with a temperature would be to use the ideal gas law:

$$P = \rho RT$$

This equation will yield the initial pressure just after combustion. This pressure is what is used by the engine to generate the work required to propel the race car. Because of the complexities of the combustion cycle, this is just an approximation for the internal stress. In reality, there would be a discontinuity at the inlet of the gas and the exhaust of the fumes.

The benefit of this model is that, like the acceleration, it can be shown to be shown to vary with time. The graph of the internal pressure vs time can be seen below in GRAPH NUMBER. This pressure is useful for finding the load on the piston as well as the load on the cylinder wall.

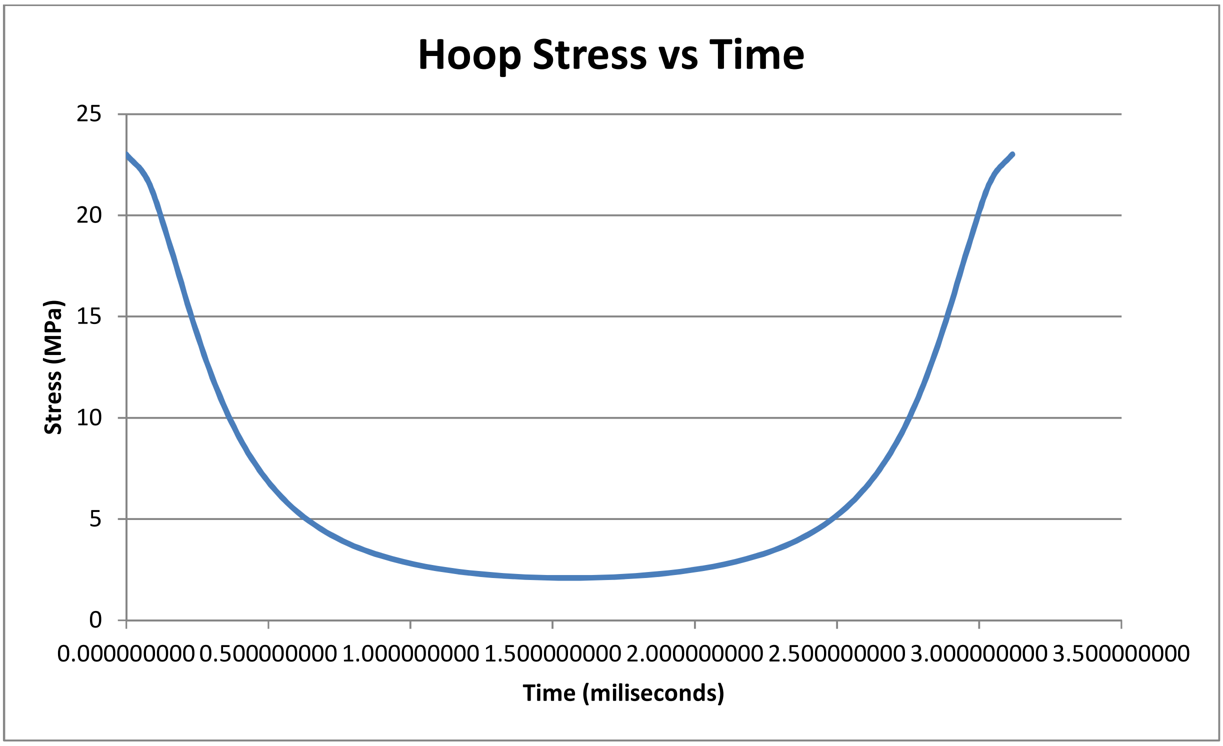


7.1.3 Hoop Stress and Longitudinal Stress

The largest stress that the cylinder wall will be under will be from the so called hoop stress. This is a stress in the radial direction that is a consequence of the internal pressure. Because of the geometry of the cylinder, the shear stress on the material can be neglected. The hoop stress can be determined from the following equation:

$$\sigma = \frac{Pr}{t}$$

This stress, like the internal stress, will be cyclic in nature. The following graph shows how the longitudinal stress varies with time:



This graph implies that the stress felt in the radial direction varies between 23.01MPa and 2.09MPa, implying a change in stress of 20.92MPa. This change in stress will dictate the life of the cylinder sleeve.

Because the cylinder is constrained on the top and bottom, there is also a longitudinal stress in the axial direction. All longitudinal stresses correspond to the same time and position to the hoop stresses. This stress varies between 5.75MPa and .52MPa. The longitudinal stress change is therefore 5.23MPa.

7.2 Thermal Stress

The largest stress that the material will be under will be from the thermal expansion as the sleeve is constrained on the top and bottom. If we use super position, we can derive the following relationship, shown in APPENDIX XX:

$$\sigma_{th} = \alpha \Delta T E$$

Since the average operating temperature is 500°C, we can find that the stress to be 480.8MPa in compression. This large thermal stress is due in part because of the large modulus of elasticity. This strain can also be reduced if a slight gap was allowed to exist between the cylinder sleeve and the engine block. This would allow the material to expand unconstrained, causing no stress to be developed in the longitudinal direction.

Because of the constraint, the thermal strain in the radial direction should be taken into account. However, the way the piston has been designed allows a 1mm gap between the cylinder sleeve and the piston. The Poisson effect will not close this gap. The way this is taken closed is from the piston rings. These rings are normally made of hardened steel and have a radius larger than the piston to close the gap between it and the sleeve. It has an angular gap in it that serves two purposes. The first is to allow the ring to be inserted onto the piston. The second, and more important, purpose is to allow the ring to have a constant radius as it warms to the steady-state temperature. As the ring warms, the gap closes due to thermal expansion, allowing the rings outside radius to remain constant. This design eliminates the need to closely match the thermal strain of both the piston and cylinder sleeve, allowing most designs to use a different material for each part.

7.3 Failure theory

Picking a proper failure theory will be crucial to determining if our selected material can handle the applied stresses. While it is not the most conservative theory, the Von Mises Failure theory is a convenient theory to use to choose as we can verify that our part does not fail with common Finite Element Analysis software that commonly use the Von Mises Criterion.

The Theorem states the following:

$$\frac{\sqrt{2}}{2} * [(\sigma_1 - \sigma_2)^2 + (\sigma_1 - \sigma_3)^2 + (\sigma_2 - \sigma_3)^2]^{\frac{1}{2}} < \sigma_y$$

Using the current loading supplied we can find that:

$$\sigma_1 = -475\text{MPa}$$

$$\sigma_2 = 23\text{MPa}$$

$$\sigma_3 = 0$$

Where the 1-direction is in the longitudinal direction and represents the addition of the thermal stress and the longitudinal stress of a cylinder; the 2-direction is the hoop stress in the radial direction. From the above stresses and the Von Mises failure theory, we can determine that our yield stress needs to be. We find it to be:

$$\sigma_y > 490\text{MPa}$$

This is a very high stress, but Silicon Nitride can handle it if the part was manufactured with the correct properties. The vast majority of this stress is also in compression; like all ceramics, Silicon Nitride handles compressive loads very well and could handle this load fairly easily.

While Silicon Nitride can handle this stress, it is found to be so high because of the thermal stress of the part. If a slight gap was allowed under for the piston sleeve to expand prior to touching the engine block, the stress would be significantly lower.

7.5 Cyclic Loading

As GRAPH ### shows, the piston and cylinder sleeve are going to be under a cyclic load each and every engine cycle. As we have chosen Silicon Nitride for our material of choice, we have to examine the fatigue life of the material.

Because Silicon Nitride is highly used in high temperature applications, most of the data that was readily accessible came from tests performed on much higher temperatures. [REF]

From these numbers we can obtain the following relationship, the derivation of which can be seen in APPENDIX XX.

$$\Delta\sigma N_f^{.0637} = 135.889$$

The graph of which can be seen in FIGURE XX below. Note that under the temperatures we run the piston at, the real graph will be higher, giving us more cycles to failure. Also, as with most ceramics, there is no clear endurance limit; eventually our parts will fail under any cyclic loading.

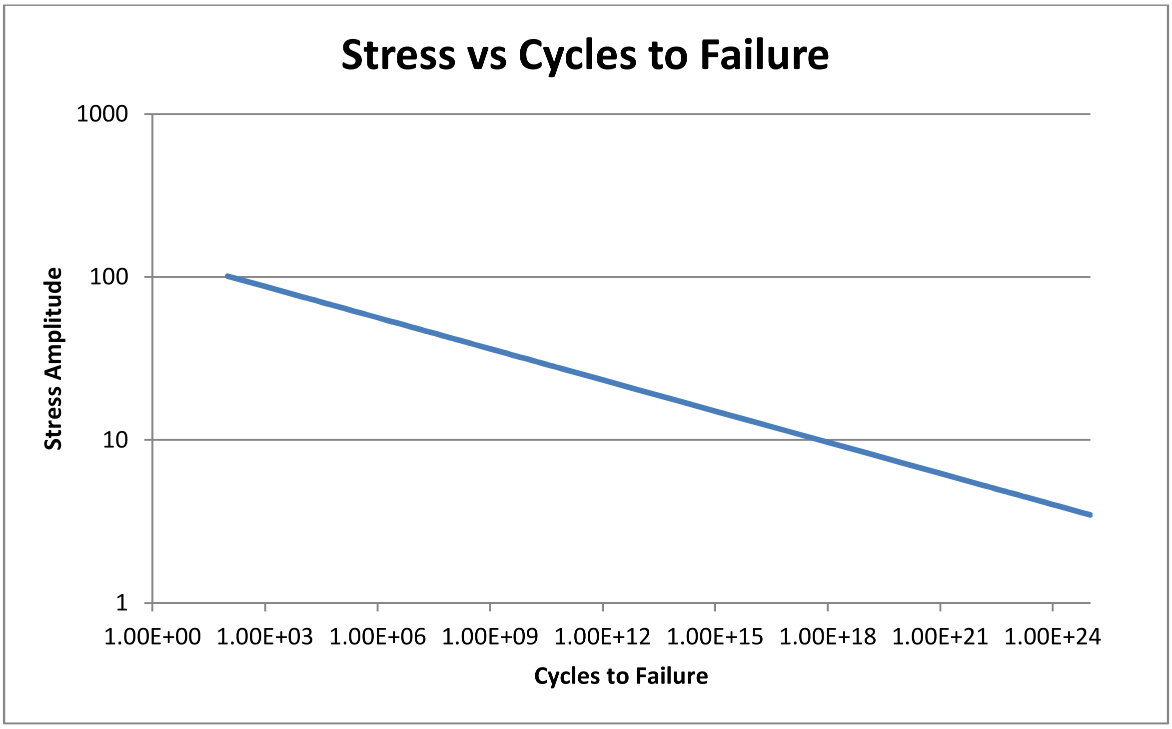


Figure XX. A log-log graph of the Wöhler curve for Silicon Nitride.

From the Wohler curve, we can deduce that the cylinder sleeve can last approximately $5.715*10^{12}$ cycles. In terms of the engine cycles, this we be half as the load happens twice every revolution. This implies that the engine can last $2.85*10^{12}$ revolutions. Assuming the engine could run at 19,250 RPM the entire race, there are $193.25*10^{6}$ revolutions per race. These numbers show that our cylinder sleeve will outlast the life of the engine by several orders of magnitude.

7.6 Crack Length

As with all materials, a large consideration needs to be placed on the critical crack length of the material. This is the max length that a crack can be in a material without failing under the load. The Irwin modification to the Griffith equation, shown below, provides us a way to relate the critical crack length wit the applied strength.

$$\sigma Y\sqrt{\pi a} = K_{IC}$$

Since Silicon Nitride has a low fracture toughness of 6.1 MPa $\sqrt{m}$, the crack length is going to be small. Solving the above for a with our maximum applied stress we obtain that:

$$\sigma Y\sqrt{\pi a} = K_{IC}$$

$$480\text{MPa} * 1 * \sqrt{\pi a} = 6.1\text{MPa}\sqrt{\text{m}}$$

$$\Rightarrow a = .05\text{mm}$$

This means that we can have a center crack length of 2a = .1mm. While this is a relatively short crack length, the manufacture of Silicon Nitride will allow the part to be made with a max crack length less than this value due to the sintering process as will be discussed in Section 8.0.

7.7 Material Conclusion

After analysis through both hand calculations as well as through the Finite Element Analysis it was shown that the material can withstand the internal loading required of it. This adds credibility to selecting it in part because of it thermal properties as well as high strength. The last limiting factor will be the cost associated with the manufacture of the material which will be discussed in Sections 8 and 9.

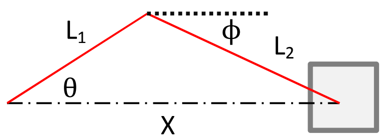
8.0 Material Manufacture

9.0 Cost Analysis

10.0 Future Design Considerations

11.0 Conclusion

Appendix I



In order to solve for the acceleration of the piston, shown in Figure X above, we need to determine the acceleration of X ($\ddot{X}$). In order to find this, we need to first form a closed vector loop which can be seen below in Equation 1:

$$L_1(\cos\theta\vec{\imath} + \sin\theta\vec{\jmath}) + L_2(\cos\phi\vec{\imath} - \sin\phi\vec{\jmath}) + X\vec{\imath} = 0$$

By taking the derivative of (1) once, we can find the velocity, and twice, we can find the acceleration which can be seen below in Equations 1 and 2, respectively.

$$L_1(-\dot{\theta}\sin\theta\vec{\imath} + \dot{\theta}\cos\theta\vec{\jmath}) + L_2(-\dot{\phi}\sin\phi\vec{\imath} - \dot{\phi}\cos\phi\vec{\jmath}) + \dot{X}\vec{\imath} = 0$$

$$\Rightarrow \dot{X} = L_1\dot{\theta}\sin\theta + L_2\dot{\phi}\sin\phi$$

$$\Rightarrow \dot{\phi} = \frac{L_1\dot{\theta}\cos\theta}{L_2\cos\phi}$$

$$L_1\left(\left[-\ddot{\theta}\sin\theta - \dot{\theta}^2\cos\theta\right]\vec{\imath} + \left[\ddot{\theta}\cos\theta - \dot{\theta}^2\sin\theta\right]\vec{\jmath}\right) + L_2\left(\left[-\ddot{\phi}\sin\phi - \dot{\phi}^2\cos\phi\right]\vec{\imath} + \left[-\ddot{\phi}\cos\phi + \dot{\phi}^2\sin\phi\right]\vec{\jmath}\right) + \ddot{X}\vec{\imath} = 0$$

$$\Rightarrow \ddot{X} = L_1(\ddot{\theta}\sin\theta + \dot{\theta}^2\cos\theta) + L_2(\ddot{\phi}\sin\phi + \dot{\phi}^2\cos\phi)$$

$$\Rightarrow \ddot{\phi} = \frac{\ddot{\theta}\cos\theta - \dot{\theta}^2\sin\theta + \dot{\phi}^2\sin\phi}{\cos\phi}$$