

DYNAMIC BALANCE OF A SINGLE CYLINDER RECIPROCATING ENGINE WITH  
OPTICAL ACCESS

By

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## ABSTRACT

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The balance within reciprocating engines has always been a concern of engine designers. Any unbalance within an engine can result in component fatigue and failure, excessive vibration, and radiated noise (airborne and structural). When an engine is created with optical access, a Bowditch piston is placed on top a standard piston. This results in the reciprocating mass being significantly greater, and it greatly increases the reciprocating force.

The components that affect the balance of the engine were identified and different design aspects within each component that affect the balance were explored. The calculations that were used to calculate static and dynamic balance of the system and how these affect the design of our engine were investigated. Practical techniques were demonstrated to validate the balance of each component after they had been fabricated.

Through testing, the first and second order balance effects were analyzed and the harmonic resonances within the system were identified. The interactions between the harmonic resonances and the first and second order forces were also explored.

## ACKNOWLEDGEMENTS

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## LIST OF SYMBOLS AND ABBREVIATIONS

|          |                                     |
|----------|-------------------------------------|
| $\theta$ | Angle                               |
| A        | Fan angle                           |
| a        | Acceleration                        |
| cos      | Cosine                              |
| F        | Force                               |
| g        | Gravity                             |
| gr       | Gram                                |
| Hz       | Hertz                               |
| Kg       | Kilogram                            |
| kHz      | Kilohertz                           |
| l        | Length                              |
| mm       | Millimeter                          |
| r        | Rotational distance                 |
| R        | Radius of counterweight             |
| T        | Thickness of counterweight          |
| $\omega$ | Angular speed                       |
| 3-D      | Three dimensional                   |
| BDC      | Bottom dead center                  |
| CAS      | Combustion analysis system          |
| CX       | Dynamic balance term in x direction |
| CY       | Dynamic balance term in y direction |
| FFT      | Fast Fourier transform              |

RPM Rotations per minute

MSU Michigan State University

TDC Top dead center

WR Weight

## CHAPTER 1: INTRODUCTION

### 1.1 Motivation

The balance within reciprocating engines has always been a concern of engine designers. Any unbalance within an engine can result in component fatigue and failure, excessive vibration, and radiated noise (airborne and structural). When creating the engine with optical access, a Bowditch piston is placed on top of a standard piston. This makes the reciprocating mass significantly greater and greatly increases the reciprocating force. If this reciprocating force is not addressed, the engine will vibrate excessively. Table 1 depicts the measured differences between the reciprocating forces within a standard single-cylinder engine and those within a single-cylinder optical access engine. Balanced multi-cylinder stock engines typically have a residual imbalance of  $\pm 1439 \text{ gr}\cdot\text{mm}$ . In comparison, the residual imbalance is  $\pm 359 \text{ gr}\cdot\text{mm}$  for street performance and  $\pm 143 \text{ gr}\cdot\text{mm}$  for professional racing engines. Table 1 compares the difference in the reciprocating forces between a standard, single-cylinder engine with a reciprocating mass of 1216 grams and an optical, single-cylinder engine with a reciprocating mass of 4304 grams. If these forces are not counteracted by balance shafts it is noticed that the second-order force of the optical engine is greater than the first-order force of a standard, single-cylinder engine. This is the primary motivation to implement both first and second-order balance shafts within optical, single-cylinder engines.

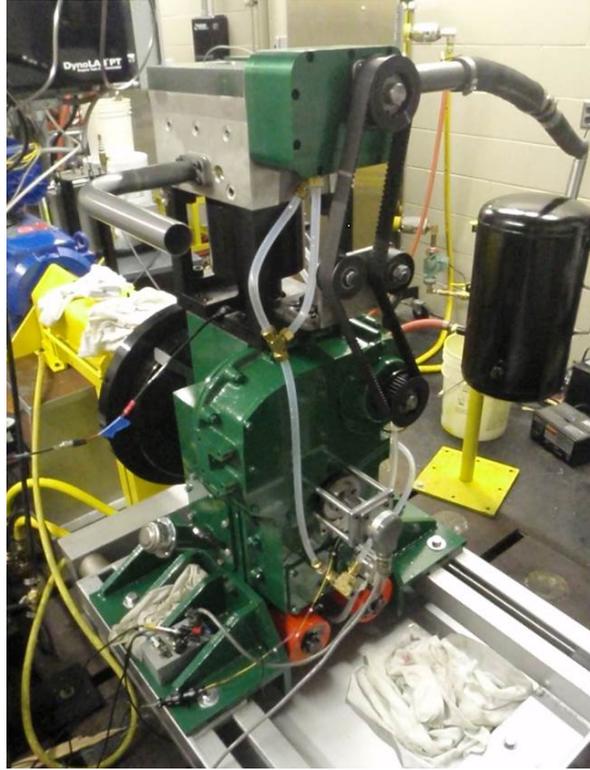
| Term Order   | Formulation                                | Standard Engine                    | Optical Engine                     |
|--------------|--------------------------------------------|------------------------------------|------------------------------------|
| First Order  | $(mr\omega^2/g)\cos(\theta)$               | $54598.4(\omega^2/g)\cos(\theta)$  | $193249.6(\omega^2/g)\cos(\theta)$ |
| Second Order | $(mr\omega^2/g)(r/l)\cos(2\theta)$         | $16128.0(\omega^2/g)\cos(2\theta)$ | $57084.9(\omega^2/g)\cos(2\theta)$ |
| Fourth Order | $(mr\omega^2/g)((r/l)^3/4)\cos(4\theta)$   | $351.8(\omega^2/g)\cos(4\theta)$   | $1245.2(\omega^2/g)\cos(4\theta)$  |
| Sixth Order  | $(mr\omega^2/g)((r/l)^5/128)\cos(6\theta)$ | $8.6(\omega^2/g)\cos(6\theta)$     | $30.5(\omega^2/g)\cos(6\theta)$    |

$r$ =length of crank throw ( $r = 44.9\text{mm}$ )                       $\theta$ =crank angle from TDC  
 $l$ =length of connecting rod ( $l = 152\text{mm}$ )                       $\omega$ =angular speed of crank  
 $m$ =recipricating mass ( $m_{\text{standard}} = 1216\text{g}$ ,  $m_{\text{optical}} = 4304\text{g}$ )

*Table 1: Calculated differences between the reciprocating forces within a standard single cylinder engine and a single cylinder optical engine.*

## 1.2 Previous Work

In previous years, the Energy and Automotive Research Laboratory (EARL) team at Michigan State University (MSU) used a commercial single cylinder engine for a variety of experiments. This engine was designed in a way that allowed for modifications of different internal engine components such as the crankshaft, connecting rod, piston, piston cylinder liner, and cylinder head. The engine was used in two different configurations: optical access and standard. The optical access mode was used for cylinder velocity flow mapping, fuel injector research, fuel distribution research, and combustion research. The optical configuration can also be used to visually and mathematically analyze in-cylinder combustion. The standard configuration was used with prototype heads to obtain power and torque measurements. Figure 1 shows a single cylinder engine with a prototype diesel, direct-injection head installed in the optical-accessible configuration.



*Figure 1: Single Cylinder Engine with a prototype head installed in an optical accessible set-up.*

The engine utilized first-order, counter-rotating balance shafts to cancel the force of the reciprocating components of the engine. Incorporating second-order balance shafts into an engine reduces engine vibration and improves experimental options. This publication will explore the implementation of second-order balance shafts and their effect on the engine operation and will also cover some practical approaches to engine balancing.

### **1.3 Vibration and Force Fundamentals**

#### *1.3.1 Vibration*

“Vibration is defined as the response resulting from any force repeatedly applied to a body. In the case of forces generated within an engine, the magnitudes will be constant at any given speed and load. Since the frequency of

the vibration is determined by the engine speed, it is convenient to define the vibration order as the vibration frequency relative to the shaft speed. A first-order vibration is generated by forces that are applied over a cycle defined by one crankshaft revolution. A second-order vibration occurs twice during every crankshaft revolution and so on (Hoag, 2006, p.57).”

The main forces acting within a combustion engine are the centrifugal force generated from the crankshaft, combustion events, and the reciprocating force generated by the piston moving up and down within the cylinder. For the crankshaft as it spins each element of mass located at a distance away from the crankshaft centerline generates an outward force calculated as

$$F_{Rotational} = \frac{ma}{g} = \frac{mr\omega^2}{g} \quad (1)$$

where  $m$  is mass,  $a$  is the acceleration,  $r$  is the rotational distance from the shaft centerline,  $\omega$  is the angular velocity (in radians per unit time), and  $g$  is the gravitational constant. For a multi-cylinder engine, each cylinder is addressed separately and superposition is used to calculate each crankshaft journal’s portion of the force; the forces are then added together. In this research, single cylinder engines are used; therefore, only one calculation is necessary. For the engines described in this publication, the crankshaft is dynamically balanced, resulting in the rotating mass being counterbalanced within the crankshaft. The rotating mass is identified as the mass of the crankshaft pin and its cheeks, along with the large end of the connecting rod. The allocation of the mass of the connecting rod, between the large end of the rod and the small end of the rod, will be discussed later. To balance the crankshaft, counterweights were added to oppose the force

generated by the mass of the crankshaft pin and the large end of the connecting rod. The reciprocating force follows the same general equation as equation 1, but the acceleration will oscillate from positive to negative as the piston moves from top dead center (TDC) to bottom dead center (BDC). “Its approximate value can be written as

$$a = -\omega^2 r (\cos(\theta) + \left(\frac{r}{l}\right) \cos(2\theta) - \left(\frac{r^3}{4l}\right) \cos(4\theta) + \left(\frac{9r^5}{128l}\right) \cos(6\theta) - \dots \quad (2)$$

where  $a$  is the acceleration,  $r$  is the rotational distance from the shaft centerline,  $\omega$  is the angular velocity (in radians per unit time),  $l$ =length of connecting rod, and  $\theta$ =crank angle from TDC (Sharp, 1907, p.86).” Equation 2 demonstrates the breakdown of the acceleration that gives the first-order through the sixth-order terms of the reciprocating force. The first-order term is:

$$a = -\omega^2 r (\cos(\theta)) \quad (3)$$

And the second-order term is:

$$a = \omega^2 r \left(\frac{r}{l}\right) \cos(2\theta) \quad (4)$$

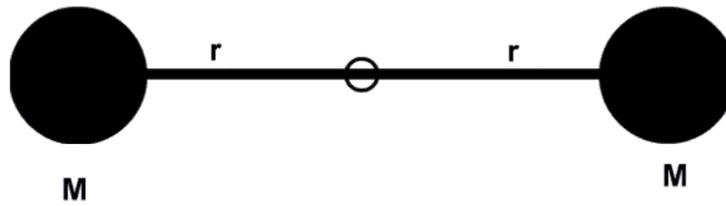
In fact, an infinite number of higher order terms are introduced into the piston acceleration. These terms complicate the balancing of an engine. Fortunately, as the terms increase in order, their magnitude decreases, so they become less significant. In practice, it is normal to only consider the first and second-order terms when performing balance calculations. The reciprocating forces with a frequency equal to the engine RPM are known as primary, or first-order, forces, and the reciprocating forces which cycle at twice the engine speed are known as secondary, or second-order, forces.

### *1.3.2 Rotating Balance*

Any rotating object can produce net rotating unbalanced forces if not properly balanced. Components that should be addressed include the clutch assembly, alternator, flywheel, driveshaft, and crankshaft. These unbalanced forces are due to asymmetrical mass distribution about the rotating axis of the object in question. There are two aspects of rotating balance which need to be considered: static and dynamic. It is possible for an object to be statically balanced while being unbalanced dynamically. The reverse is not true; any object in dynamic balance is also in static balance. All particles within a spinning object produce what is commonly called centrifugal force, and this force acts radially outward from each particle. If the sum of all the centrifugal forces is zero, then the object is balanced. (Foale, 2007, p.1)

### *1.3.3 Static Balance*

The test for static balance is quite simple. If the object is mounted in low friction bearings with the axis of rotation horizontal, then the object will remain stationary regardless of its initial starting position. However, if any static imbalance exists within the object, the object will tend to come to rest in a position with the heaviest section at the bottom as shown. When an object has two identical masses, these masses must simply be placed equal distances from the axis of rotation (see Figure 2) for the object to be in a state of static balance. (Foale, 2007, p.1)



*Figure 2: An object in static balance with two identical masses placed equal distances from the axis of rotation. Reprinted from Some Science of Balance, by T. Foale. Retrieved November, 2014 from <http://www.tonyfoale.com/Articles/EngineBalance/EngineBalance.pdf>. Copyright 2007 by Tony Foale. Reprinted with permission.*

However, often an object possesses components of unequal mass. If an object has two components of differing mass, than those components must be placed a distance from the rotating axis that is proportional to the ratio of the masses. For example, if an object has one component with a mass that is twice that of the other component, then each mass must produce an equal and opposite moment about the axis in order to achieve static balance. Figure 3 shows that the lighter mass must be mounted at twice the distance from the axis as the heavier mass in order to fulfill this condition. If there are multiple components of differing masses, then the components must be placed in such a way that the vector sum of all the moment arms equals zero as shown in Figure 4.

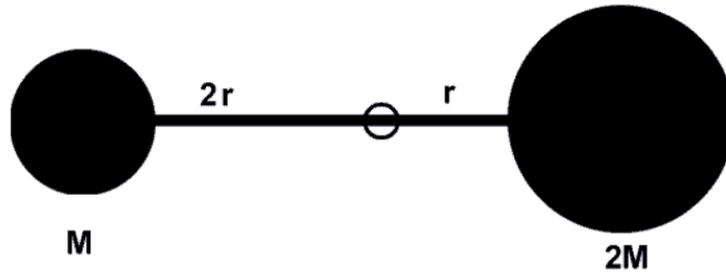


Figure 3: An object in static balance with unequal masses placed a proportional distance from the axis of rotation. Reprinted from Some Science of Balance, by T. Foale. Retrieved November, 2014 from <http://www.tonyfoale.com/Articles/EngineBalance/EngineBalance.pdf>. Copyright 2007 by Tony Foale. Reprinted with permission.

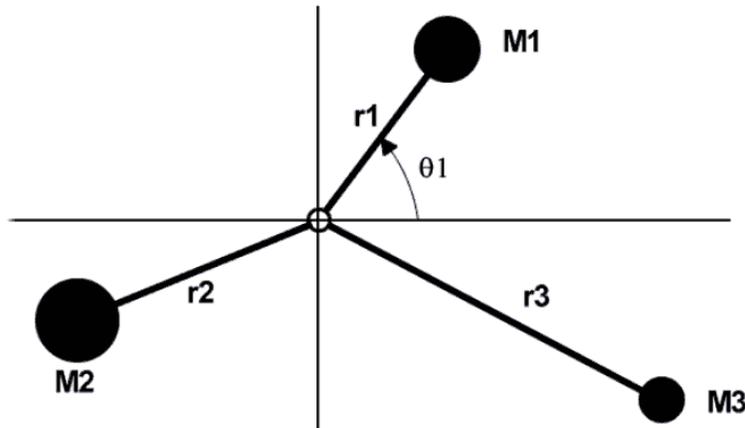
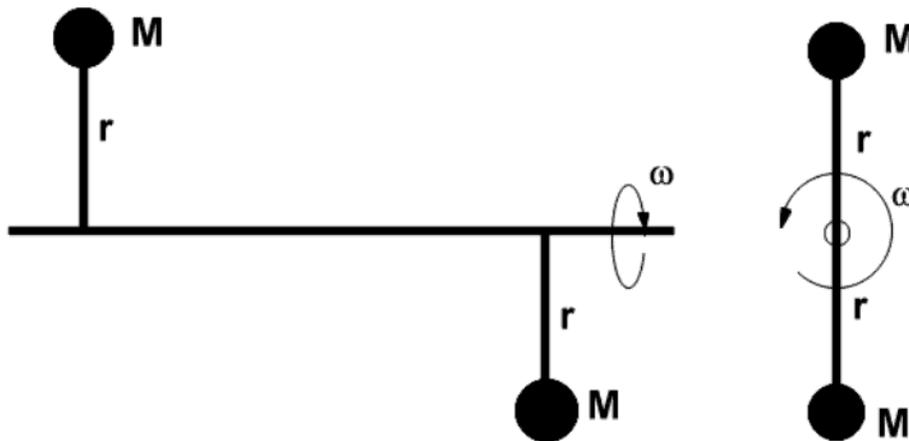


Figure 4: The general condition for balance with numerous masses is that the vector sum of all the moments equates to zero. Reprinted from Some Science of Balance, by T. Foale. Retrieved November, 2014 from <http://www.tonyfoale.com/Articles/EngineBalance/EngineBalance.pdf>. Copyright 2007 by Tony Foale. Reprinted with permission.

### 1.3.4 Dynamic Balance

Dynamic balance is defined as: “The condition which exists in a rotating body when the axis about which it is forced to rotate, or to which reference is made, is parallel with a principal axis of inertia; no products of inertia about the center of gravity of the body exist in relation to the selected rotational axis (Dynamic Balance, 2003).”

The object shown in Figure 5 is in static balance, because the moments of the two masses balance each other about the spin axis. The offset of the two masses along the spin axis gives rise to what is often known as a rocking couple. As such an object rotates, the orientation of this couple also rotates, attempting to move the axis in a conical manner; this results in a state of dynamic imbalance. To achieve dynamic balance, the masses must be rearranged so that the rocking couple disappears (Foale, 2007). This can be achieved by either removing material from the object or by adding material to the object. Figure 6 shows how the addition of two extra masses can achieve dynamic balance, which also guarantees that the object is statically balanced (Foale, 2007). The idea of the possible generation of a rocking couple is vital to the subject of engine balance (Foale, 2007).



*Figure 5: Two masses having centrifugal forces produce a rocking couple due to their separation along the axis of rotation.* Reprinted from *Some Science of Balance*, by T. Foale. Retrieved November, 2014 from <http://www.tonyfoale.com/Articles/EngineBalance/EngineBalance.pdf>. Copyright 2007 by Tony Foale. Reprinted with permission.

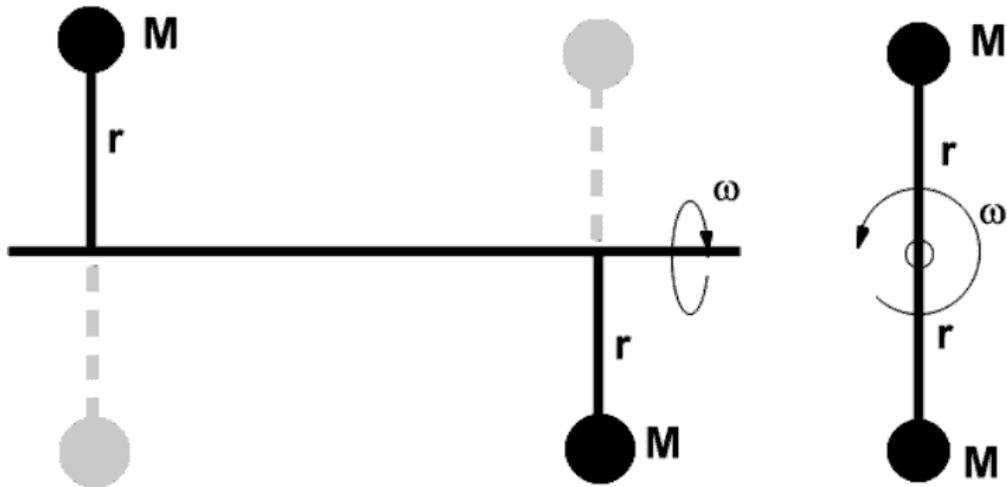


Figure 6: The addition of two opposite, but otherwise identical masses, removes the rocking couple and restores dynamic balance. Reprinted from *Some Science of Balance*, by T. Foale. Retrieved November, 2014 from <http://www.tonyfoale.com/Articles/EngineBalance/EngineBalance.pdf>. Copyright 2007 by Tony Foale. Reprinted with permission.

### 1.3.5 Reciprocating Balance

The piston in an engine moves along a straight line defined by the axis of the cylinder (Foale, 2007). The acceleration of the piston is continually changing throughout a cycle, reaching a maximum acceleration at both TDC and BDC. Oscillating forces must be applied to the piston to cause these alternating accelerations (Foale, 2007). These oscillating forces must pass through the connecting rod, to the crankshaft, on to the main bearings, and to the crankcase (Foale, 2007). If the inertia forces are not balanced internally within an engine, the forces will be transmitted from the crankcase to pass into the frame of the vehicle through the engine mounts.

The motion of the piston is approximately sinusoidal, therefore, so are the acceleration forces (Foale, 2007). If the connecting rod was infinitely long, the motion

would be truly sinusoidal, but most connecting rods are approximately twice the length of the crankshaft stroke (Foale, 2007). This relative shortness of the connecting rod means that, except for the TDC and BDC positions, the rod will not remain in line with the cylinder axis through a working cycle (Foale, 2007). “The angularity of such a short connecting rod throughout a complete crankshaft revolution modifies the piston motion (Foale, 2007, p. 3).” With a very long connecting rod, the maximum velocity of the piston should occur at  $90^\circ$  of rotation from TDC. With a 2:1 connecting rod length to stroke ratio, maximum velocity occurs just past  $77^\circ$  of rotation (Foale, 2007). In fact, an infinite number of higher order harmonics are introduced into the piston acceleration. These harmonics complicate the balancing of an engine. Fortunately, as the harmonics increase in order, their magnitude decreases, and so they become less important (Foale, 2007). In practice, it is normal to only consider the first and second harmonics when performing balance calculations.

The reciprocating forces are due to the change in acceleration of the piston, piston pin and a portion of the connecting rod. At TDC both primary and secondary forces act in the same direction resulting in a 25% increase in the total force, compared to the primary forces alone. At BDC the separate forces oppose and the peak force is reduced by 25%. For a connecting rod length to stroke ratio of 2:1, the peak magnitude of the secondary force is one quarter that of the magnitude of the primary force. Figure 7 illustrates how the primary and secondary reciprocating forces are combined. Although the primary and secondary forces combine to produce a single overall effect, it is both convenient and typical, for analysis purposes, to analyze them separately. The piston is

accelerated only in a straight line along the axis of the cylinder, and so the reciprocating forces only act along the axis of the cylinder. (Foale, 2007, p.3-4)

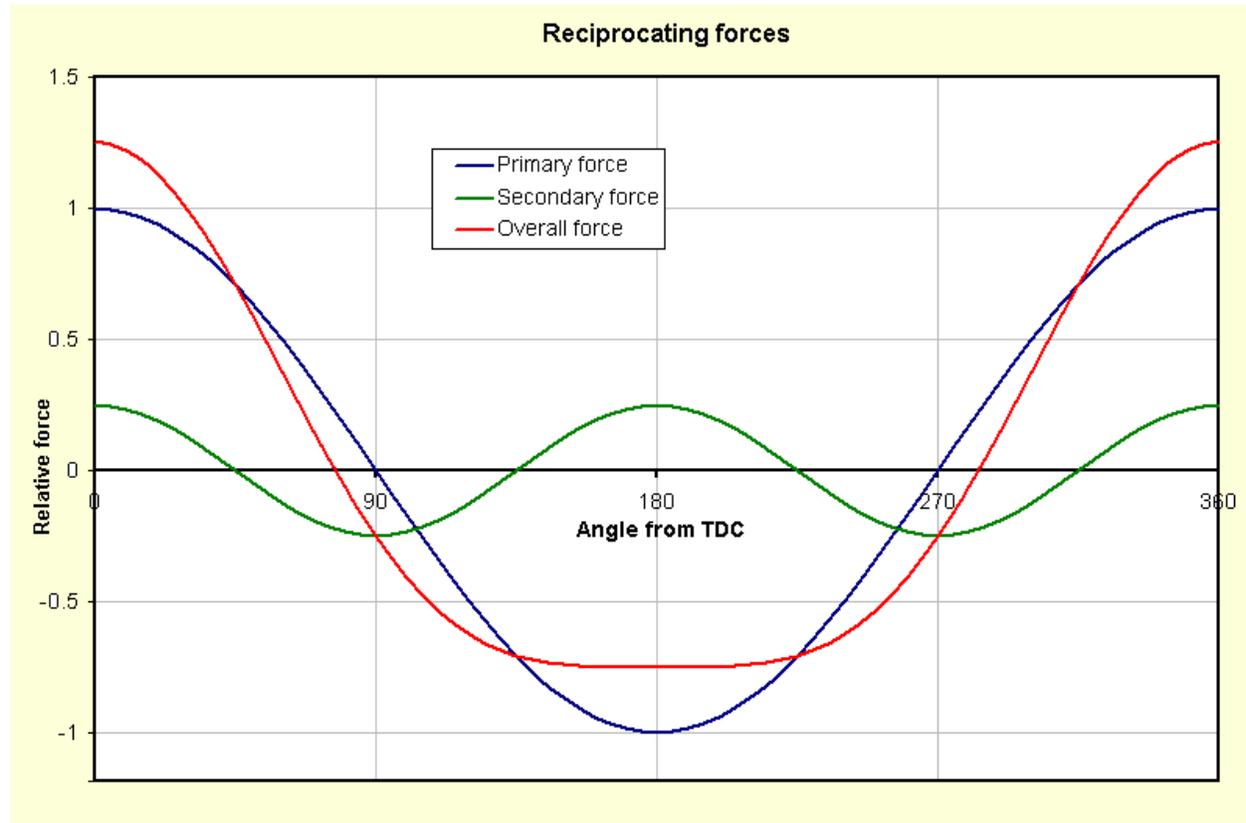
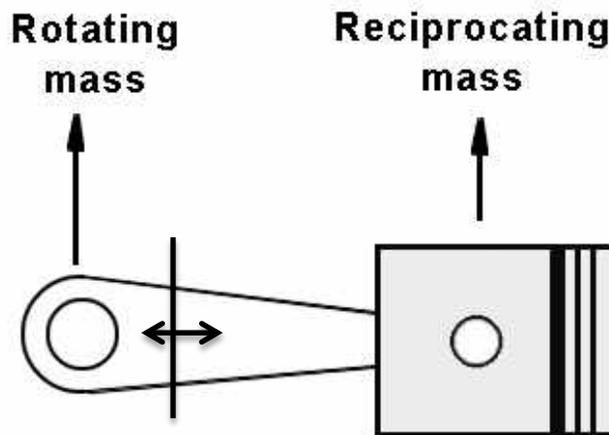


Figure 7: Primary and secondary forces are shown separately and combined. Positive values represent the reciprocating forces pulling on the connecting rod. Reprinted from Some Science of Balance, by T. Foale. Retrieved November, 2014 from <http://www.tonyfoale.com/Articles/EngineBalance/EngineBalance.pdf>. Copyright 2007 by Tony Foale. Reprinted with permission.

The connecting rod is also a factor in the reciprocating balance and is a bit more difficult to analyze. The mass of the small end of the connecting rod reciprocates in exactly the same manner as the piston and so can be directly added to the total reciprocating mass of the piston and piston pin. The big end of the connecting rod rotates with the crankshaft so it can be perfectly balanced by the

addition of a counterweight to the crankshaft on the opposite side, but the middle of the connecting rod will experience a combination of rotational and linear motion. To enable a simplified balance analysis to be made, part of the connecting rod mass is assigned as purely rotational and the remaining mass as purely reciprocating. If the connecting rod and piston assembly are supported at the axes of the big and small ends (Figure 8), then the weight, as measured at those supports, is used to represent the relative contributions to each part. Figure 8, below, demonstrates a simplified approach to the balance analysis that is commonly used. (Foale, 2007, p.5)



*Figure 8: The connecting rod and piston assembly is assigned as strictly rotating or reciprocating divided in the center.* Reprinted from *Some Science of Balance*, by T. Foale. Retrieved November, 2014 from <http://www.tonyfoale.com/Articles/EngineBalance/EngineBalance.pdf>. Copyright 2007 by Tony Foale. Reprinted with permission.

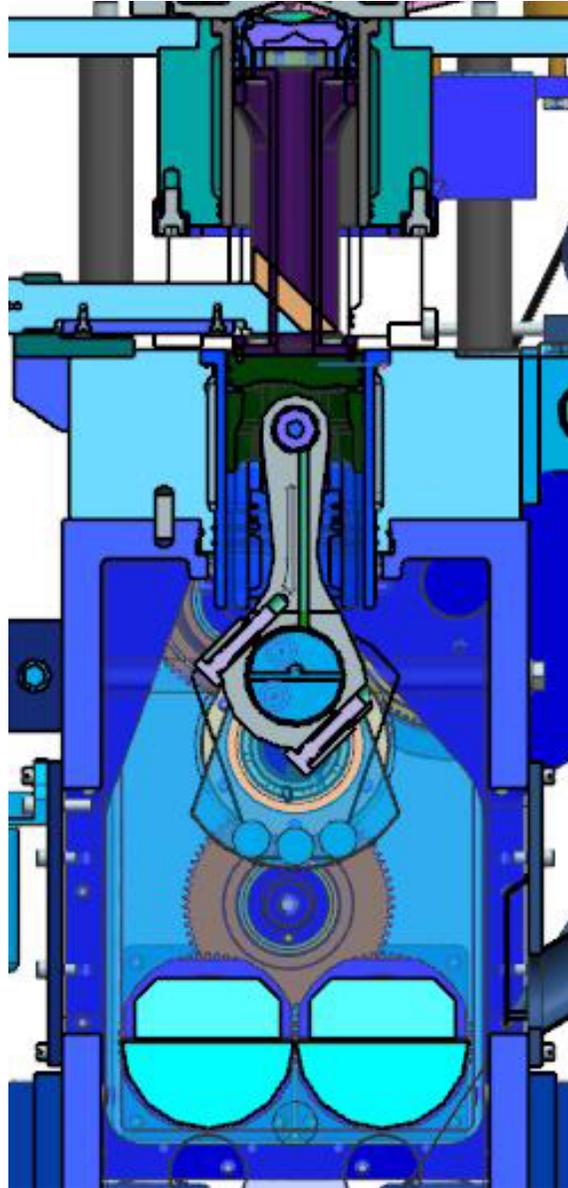
### *1.3.6 Cylinder Pressure-Induced Vibration*

The cylinder pressure varies with time; therefore, the torque produced also varies. This torque variation can be diminished by the use of high inertia rotating parts, such as a massive flywheel. However, excess rotation inertia has several other disadvantages, such as reducing acceleration. This torque variation is ultimately reacted into a pulsating chain force and pulsating loads on the engine mountings. In other words, it is a source of vibration for the engine, although generally of less concern than the forces due to the imbalance discussed before. The cylinder pressure does not create the vibration but the torque that is produced from the pressure creates the vibration. (Foale, 2007, p.19)

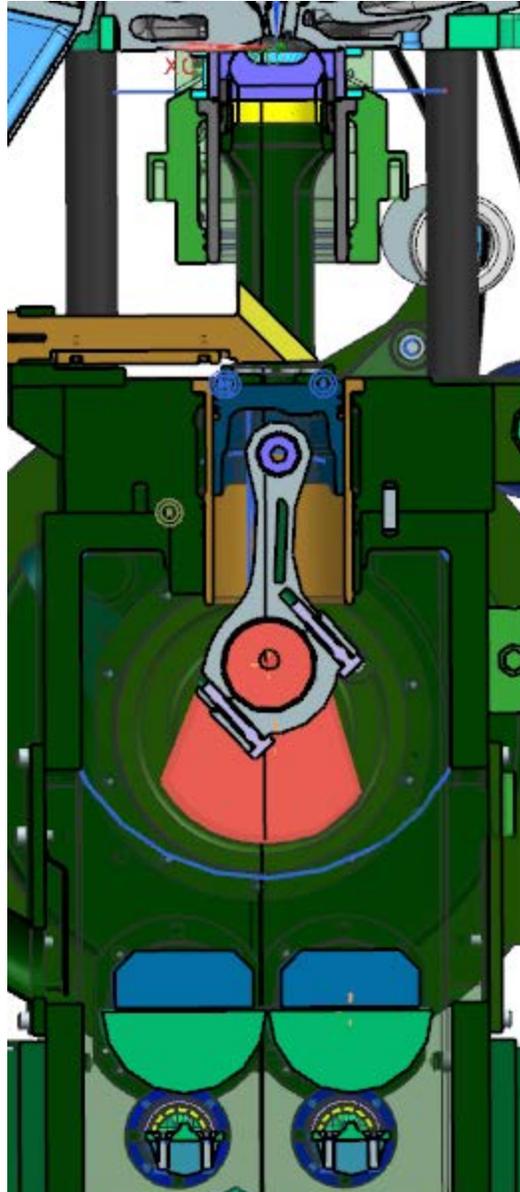
## CHAPTER 2: EXPERIMENTAL EQUIPMENT

### 2.1 Engine Configurations

Two, single-cylinder, test engine configurations were evaluated to confirm the effects of changing the balance from one configuration to the other. The force of the reciprocating parts and the counterweight force from the balance shafts was calculated for each configuration. The actual forces on the engine were measured using accelerometers and the results were compared. In both engine configurations, the crankshaft was one-hundred percent balanced to itself with the large end of the connecting rod. Both engine designs were nearly identical. All of the components were similarly designed with slight variations such as a difference in throw, rod length, piston top geometry, balance shaft configuration, piston offset, and piston pin offset. The first configuration tested was an optical configuration with no piston pin offset and with only first-order balance shafts. A sectioned view of this configuration can be seen in Figure 9. The second configuration was another optical configuration, but this one had a combined 10mm piston and piston pin offset and both first and second-order balance shafts. It also had a 116g unbalanced weight within the reciprocating parts. An aluminum spacer deformed during the initial use of the engine and was replaced with a steel spacer weighing 116g more. The balance shaft weights were not updated due to the time and cost required to manufacture new weights. A sectioned view of this configuration can be seen below in Figure 10.



*Figure 9: Optical engine configuration with no piston pin offset and first-order balance shafts only.*



*Figure 10: Optical engine configuration with first and second-order balance shafts and piston offset.*

## **2.2 Data Acquisition System and Sensors**

### *2.2.1 Data Acquisition System*

For the testing of the engines, an A&D Technologies Combustion Analysis System (CAS) was used to capture the data. It is a high-speed data acquisition module that performs online combustion analyses. It performs real-time, cycle-by-cycle calculations on combustion

data and processed data can be streamed to third party devices for closed loop control, or stored to memory. User defined cycle-by-cycle calculations are supported, allowing the integration of proprietary algorithms. Data can be set to acquire based on either crank-angle position or time. The CAS system stores raw signals in memory and downloads the data to a computer for post-processing using its own software. It has high speed analog inputs, digital inputs, analog outputs, and digital outputs. The data acquisition rate is 2 MHz and the analysis rate without missing an engine cycle is 1.2 MHz. Figure 11, below, shows an A&D CAS system.



*Figure 11: A&D Technologies Combustion Analysis System (CAS)*

### 2.2.2 Sensors

Two types of sensors were used to collect data during engine testing. The first sensor is an accelerometer. The accelerometers that were chosen are top-mount, premium-grade accelerometers from Omega Engineering. They feature a dynamic sensitivity of +/-5%:

100mV/g, acceleration range of 80g peak, and a frequency response of +/-5%. These accelerometers only measure one direction, so three of them were used. The second sensor that was used is an incremental, optical, rotary encoder. An incremental encoder produces a series of square waves as it rotates. The encoder used, produces a wave pattern every degree of rotation and another wave pattern every 360 degrees of rotation, or every full revolution. The encoder is mounted on the crankshaft. This encoder provides information regarding the position and speed of the crankshaft. These readings are used to correlate the data taken with the calculated, unbalanced force.

## CHAPTER 3: EXPERIMENTAL PROCEDURE

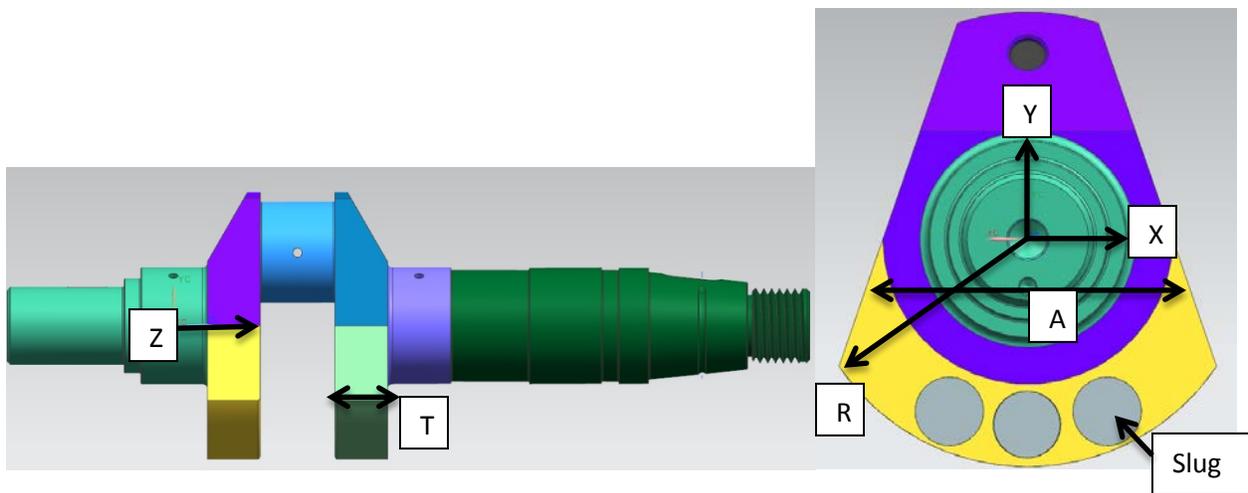
### 3.1 Computer-Aided Analysis and Calculations

#### 3.1.1 Computer-Aided Crankshaft Design and Balance

Most manufactured parts are drawn using a 3-D modeling program at some point in the design process. The 3-D model is used to create blueprints, run stress analysis and used to compute other difficult computations quickly and easily. For balancing the crankshaft, a 3-D modeling program was used to find the weights and centers of gravities within the crankshaft to balance the crankshaft to itself. The crankshaft in an internal combustion engine is a complex component. It requires intricate machining, and is highly stressed. Within the design of a crankshaft there are many factors to consider, including bearing loads, oil film thicknesses, cylinder pressures, rotating weights, and clearances with other parts, just to name a few. This section focuses on some of the design parameters that are considered in just the balancing portion of the crankshaft design.

In both of the research engines tested, the crankshaft is balanced with the heavy end of the connecting rod and it is balance one-hundred percent to itself. When designing a crankshaft, it should be designed in such a way that it is slightly overbalanced. Within the manufacturing process a small amount of material will be removed in the final balancing process to achieve the proper balance. It is always easier to remove material from the counterweights then to add it in later. It is usually desired that the crankshaft should be 1.5 to 2 kg\*mm overbalanced in the design process to accommodate manufacturing variations (Ron Sears, personal interview, February 5, 2014). The crankshaft will go through a final dynamic balance as the final step in the manufacturing process that will remove the overbalance. Some of the parameters which can

greatly affect the balance of the crankshaft are, in order of most significant effect: the radius of the counterweight (R), the thickness of the counterweight (T), the fan angle of the counterweight (A), and an optional tungsten alloy slug added for additional counterbalance. These parameters are depicted below in Figure 12. When calculating the effect of the tungsten slug, the weight added will be the volume of the slug multiplied by the difference in the density between the tungsten alloy and the steel used in the crankshaft.



*Figure 12: Crankshaft design parameters to consider.*

During the design process, the balance of the current computer model can be mathematically calculated. The first step in this process is to divide the crank into sections which consist of the counterweights, the cheeks, and the pins. Figure 12 depicts these divisions as indicated by different colors. After the crank is divided into sections, the material properties are applied within the model. Then, each section is analyzed to obtain the initial input data for the calculations. The input data that is needed is the weight of each section and the center of gravity locations in the X, Y, and Z planes shown in Figure 12. Once this data is obtained a table can be created to organize the procedure. A sample can be seen in Table 2, below, where the areas

highlighted in green are the input data which consist of the weight (WR), the X locations of the center of gravity (X), the Y locations of the center of gravity (Y), and then the Z locations of the center of gravity (Z). The initial starting datum that the measurements are taken from should be placed in the center of the first main bearing. The static balance calculation was then performed by taking the weight (WR) of each part and multiplying it by the X and Y distances for each component (see Equation 5 and 6). This data is recorded in the WR X, WR Y portion of Table 2.

$$WR \times X = \text{static portion in X} \quad (5)$$

$$WR \times Y = \text{static portion in Y} \quad (6)$$

The static portions in X of each component are summed and will, ideally, equal zero. The static portions in Y are also summed and should be equal to zero. If the sums do not equate to zero, the crankshaft should be altered, changing R, T, A, or adding a slug. It should be remembered that the general condition of balance with numerous masses is that the vector sum of all the moments equals zero. Along with being statically balanced, the crank also needs to be dynamically balanced. Equations 7 and 8 can be used to find the dynamic portion of each component by multiplying the static values by the Z distance of its center gravity.

$$WR \times X \times Z = \text{Dynamic portion in X} \quad (7)$$

$$WR \times Y \times Z = \text{Dynamic portion in Y} \quad (8)$$

These values are summed similarly to the static values. Both the static and dynamic calculations should be figured at the same time and both should be equal to zero. To satisfy this condition, the

counterweight that is closest to the initial starting datum will be heavier than the counterweight that is further away. Usually, the final balancing hole that is used to remove the slight overbalance of 1.5-2 kg mm designed into the crank for machining variations is drilled in the counterweight that is furthest from the initial starting datum. The crankshaft should always be dynamically balanced on a crankshaft balancing machine to achieve the final balance of the part. Equations 7 and 8 are used to guide the design of the crankshaft and make sure the crank is just slightly over balanced so the balance machine will have to remove very little from the crank to balance it.

| Component         | Wr<br>kg | X<br>mm | Y<br>mm  | Wr X<br>kg mm | Wr Y<br>kg mm | Z<br>mm  | CX<br>kg mm <sup>2</sup> | CY<br>kg mm <sup>2</sup> |
|-------------------|----------|---------|----------|---------------|---------------|----------|--------------------------|--------------------------|
| Balance Hole 1    | 0.0000   | 0.0000  | 0.0000   | 0.0000        | 0.0000        | 38.3443  | 0.0000                   | 0.0000                   |
| Counterweight 1   | 1.8650   | 0.0348  | -58.2207 | 0.0649        | -108.5823     | 38.3443  | 2.4878                   | -4163.5146               |
| Cheek 1           | 2.7743   | -0.0356 | 6.8137   | -0.0986       | 18.9033       | 39.1488  | -3.8619                  | 740.0420                 |
| Pin 1             | 1.2003   | -0.0003 | 47.1528  | -0.0003       | 56.5958       | 79.6191  | -0.0249                  | 4506.1057                |
| Est. Bob Weight 1 | 1.8820   | 0.0000  | 47.3000  | 0.0000        | 89.0186       | 79.3500  | 0.0000                   | 7063.6222                |
| Oil Weight        | 0.0147   | 1.7614  | 28.2513  | 0.0258        | 0.4140        | 70.9334  | 1.8308                   | 29.3638                  |
| Counterweight 2   | 1.3790   | 0.0000  | -55.2893 | 0.0000        | -76.2440      | 120.3577 | -0.0001                  | -9176.5508               |
| Cheek 2           | 2.7925   | 0.0000  | 7.1265   | 0.0000        | 19.9006       | 118.3353 | 0.0013                   | 2354.9466                |
| Balance Hole 2    | 0.0000   | 0.0000  | 0.0000   | 0.0000        | 0.0000        | 120.3577 | 0.0000                   | 0.0000                   |

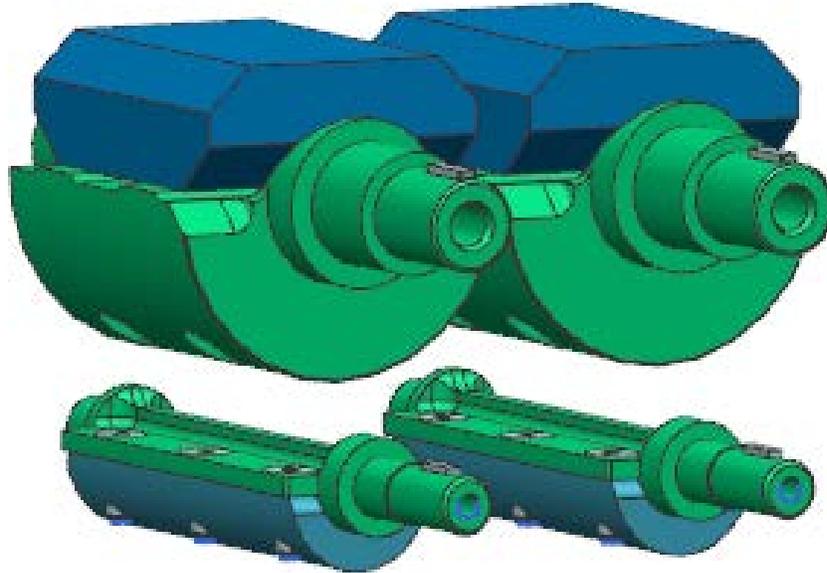
= Static Balance     
  = Dynamic Balance

Table 2: Static and dynamic balance calculations.

### 3.1.2 First and Second-order Balance Shafts

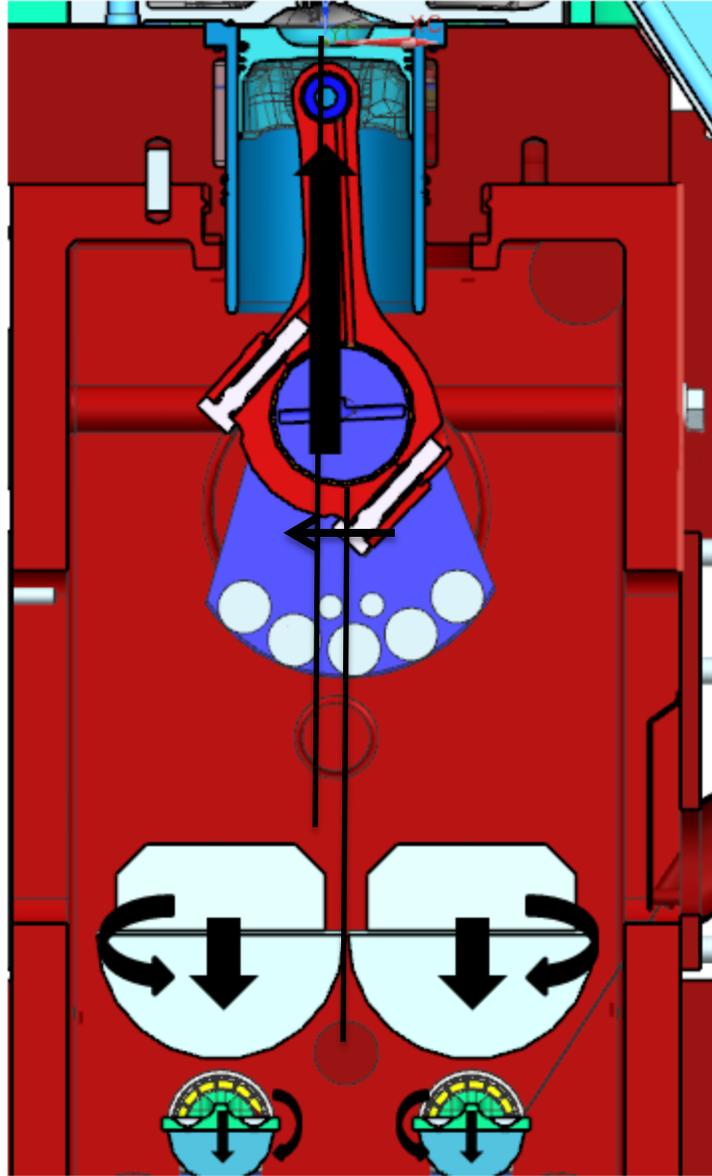
The MSU EARL team has designed and built numerous single cylinder test engines, some with just first-order balance shafts and some with first and second-order balance shafts. The balance shafts within these engines are designed so that the balance of the engine can be changed as the configuration of the engine changes. This is done by designing the shafts with a

bolt-on weight that can be taken on and off and changed as needed. Figure 13, below, represents this design where the balance shafts are green and the removable weights are blue.



*Figure 13: A computer generated model of the first-order (top) and second-order (bottom) balance shafts.*

The purpose of the balance shafts is to generate a force equal to the reciprocating force, but in the opposite direction. This is accomplished by using pairs of counter-rotating shafts: one pair for the first-order force and one pair for the second-order force. The first-order shafts will rotate at the same speed as the crank, and the second-order balance shafts will rotate at twice the speed of the crank. Figure 14 illustrates this configuration and the forces within the engine. It also shows the forces of each of the balance shafts along with the force of the reciprocating parts. It shows the rotation of the balance shafts, the axes that the forces act in, and the resultant moment from the offset of the axes due to the piston and piston pin offset.



*Figure 14: The rotation and forces within the engine at TDC*

For any configuration, whether it be an optical Bowditch configuration or a standard configuration, the weights for the balance shafts can be calculated. In the design process, computer-aided models are analyzed to obtain weights of all of the reciprocating parts and the balance shafts. The weights of the reciprocating parts are summed and then multiplied by the throw to obtain the force of the reciprocating parts as demonstrated in equations 9 and 10.

$$F_{first\ order} = -m\omega^2r(\cos(\theta)) \quad (9)$$

$$F_{second\ order} = \left(\frac{r}{l}\right)m\omega^2r(\cos(2\theta)) \quad (10)$$

In Equations 9 and 10,  $r$  is the rotational distance from the shaft centerline,  $\omega$  is the angular velocity (in radians per unit time),  $l$  is the length of the connecting rod, and  $\theta$  is the crank angle from TDC.

For the balance shafts, the mass of each shaft and its bolts is measured as is the distance of the center of gravity to the axis of rotation. The force that the shaft will exert is equal to its mass multiplied by the distance between the center of gravity and its axis of rotation. Table 3 is used along with Equations 7 and 8, from above, to calculate the portion of the first-order and second-order forces within it and helps organize the numbers. The first-order force of the reciprocating parts is 49822.3 gr\*mm and the first-order balance shaft force is 49814.8 gr\*mm giving a difference of 7.5 gr\*mm. The second order force of the reciprocating parts is 15001.4gr\*mm and the second order balance shaft force is 14993.8gr\*mm giving a difference of 7.6 gr\*mm. The sum of the differences is 15.1 gr\*mm of residual imbalance, which is very low. Within the table, the computer generated values are in blue and the measured values of the manufactured, physical part are in yellow. The computer-generated numbers were used to get preliminary drawings for the purpose of machining, but in the final calculations the measured values of the physical part were used. It is helpful to examine both the computer generated value, and the measured number together to recognize any variations that may indicate flaws or defects within the computer-aided analysis, the part itself, or the fabrication process. The table below

demonstrates that the difference between the measured values and the computer-analyzed values are almost identical, having a maximum difference of only six grams within any part.

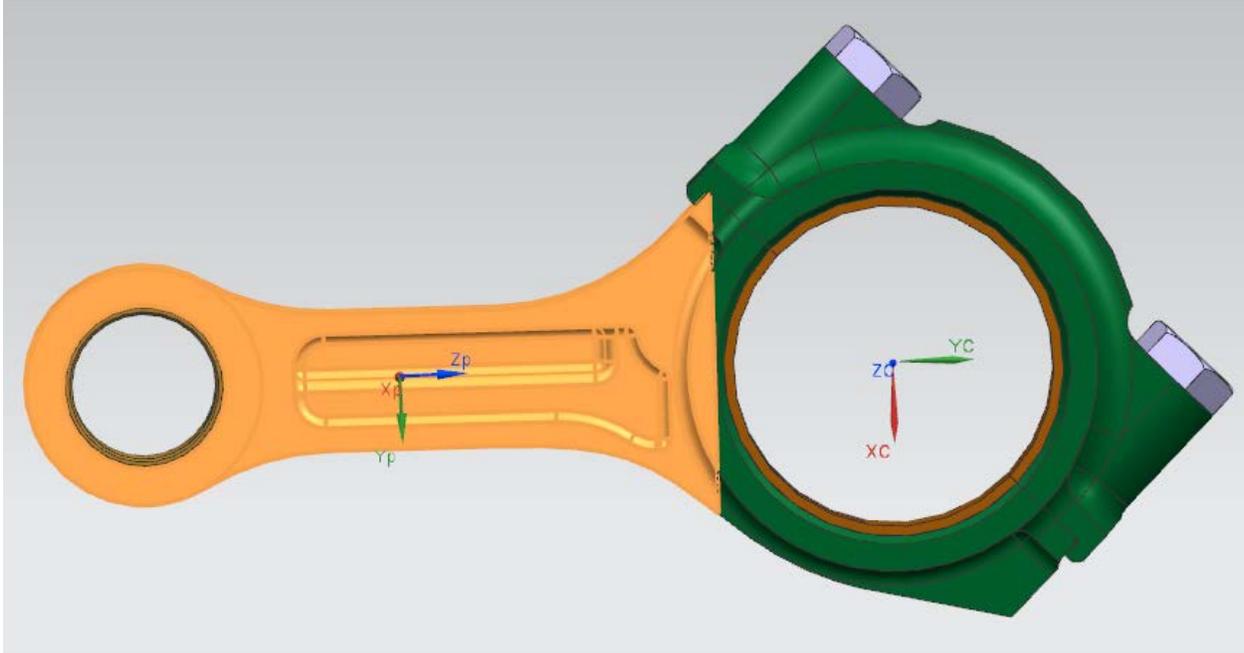
| <b>Balance Shaft Components</b>           | <b>Weight (g)</b> | <b>Weight (g)</b> | <b>Center of Gravity Offset (mm)</b> |
|-------------------------------------------|-------------------|-------------------|--------------------------------------|
| 1st order                                 |                   |                   |                                      |
| 1st order balance shaft mass              | 5751.9            | 5751.6            | 18.7                                 |
| 1st order balance shaft bolts             | 236.1             | 230.9             | 17.7                                 |
| 1st order balance shaft top counterweight | 3516.9            |                   | -24.7                                |
| 2nd order                                 |                   |                   |                                      |
| 2nd order balance shaft                   | 699.4             | 694.8             | -0.4                                 |
| 2nd order balance shaft bolts             | 59.6              | 59.6              | 7.8                                  |
| 2nd order balance shaft counterweight     | 720.9             |                   | 10.1                                 |

 = Computer-Analyzed Data       = Physically Measured Data

*Table 3: Balance shaft component values used in calculating the reciprocating forces.*

### 3.1.3 The Connecting Rod

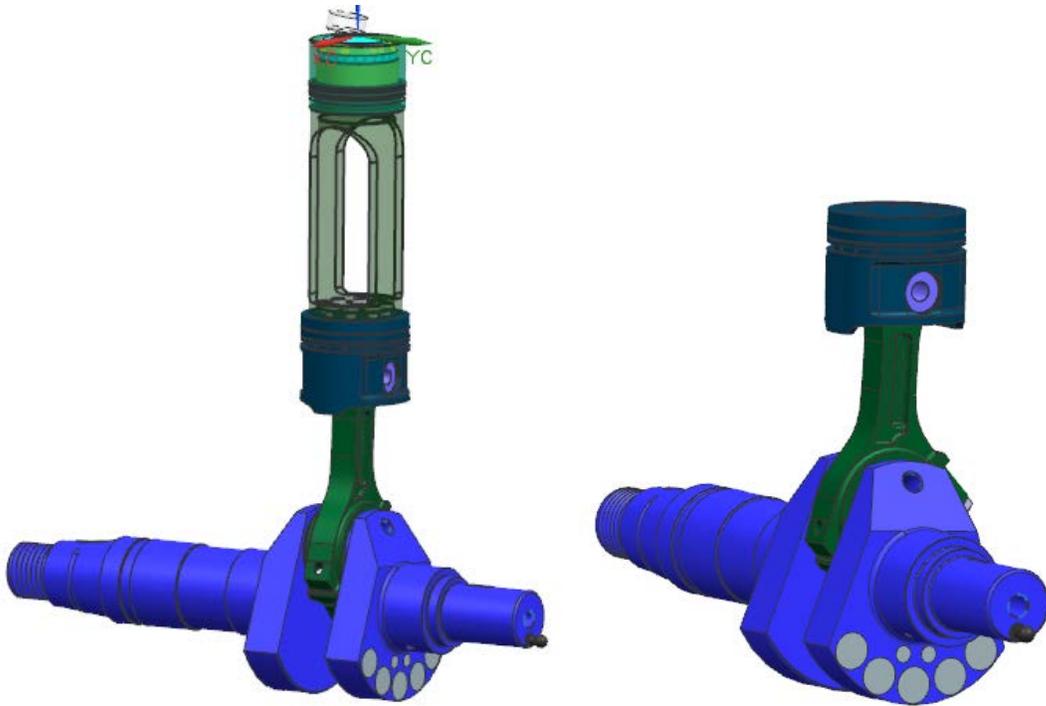
A portion of the weight of the connecting rod was used in both the crankshaft calculations and the balance shaft calculations. For these calculations, the weight of the connecting rod was sectioned into the large and small ends. The large end was used in the crankshaft calculations because it rotates with the crank and the small end is used in the balance shaft calculations because it reciprocates with the piston. The division line of the connecting rod was identified as the center of gravity. All of the bolts and bearings were considered in the calculations. Within the computer model, the center of gravity and weights of each component were easily obtained. Figure 15 shows a model of a sectioned connecting rod.



*Figure 15: An example model of a connecting rod that is sectioned for analysis.*

### *3.1.4 Reciprocating Parts*

All of the reciprocating parts were analyzed within the model for the balance shaft weight calculations. The reciprocating components of the standard configuration consist of the small end of the connecting rod, the piston pin, the piston, and the piston rings. For an optical configuration, the reciprocating components consist of the small end of the rod, the piston pin, base piston, base piston rings, Bowditch piston extension, Bowditch piston bolts, Bowditch piston top, Bowditch piston rings, an optical window, and seals for the optical window. Figure 16 shows a computer-generated model depicting the standard and optical configurations of the reciprocating components.



*Figure 16: The reciprocating components of an optical (left) and a standard (right) configuration.*

### **3.2 Physically Measuring the Balance of Individual Components**

#### *3.2.1 The Flywheel*

The flywheels of the test engines are composed of three components: a hub that mates to the crankshaft, the standard flywheel, and a large additional ring that can be added for extra inertia. The hub and standard flywheel were first dynamically balanced together and then the large ring was added and the assembly was balanced again. All of the parts were balanced on a Balance Engineering Corporation automatic balancing machine (Figure 17). The assembly was loaded onto the machine and the machine spun to the designated rpm. The balancing machine was designed to measure the imbalance and automatically stop the part and rotate it to the

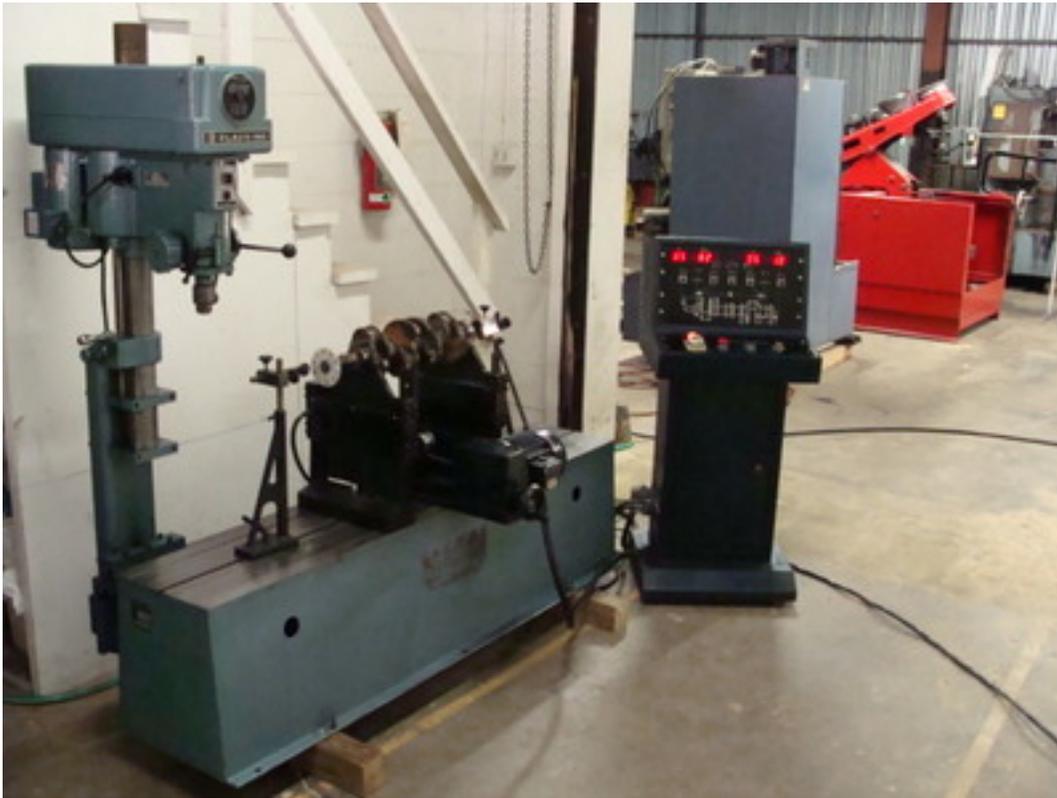
position where a balancing hole needs to be placed then commence to automatically drill the hole, or holes, if more than one is needed.



*Figure 17: Balance Engineering Corporation automatic balance machine.*

### *3.2.2 The Crankshaft*

Each crankshaft is dynamically balanced as the final step in the manufacturing process. To balance the crank, the mass of the large end of the rod must be known and a bob weight of the same mass as the connecting rod must be placed onto the pin of the crank. The crankshaft is mounted into the balancing machine (Figure 18) with the bob weights attached, and the locations of the counterweights are input into the machine. The balancing machine will then spin the crankshaft and measure the balance of the crank. It will then rotate the crank to the location of the counter-balance hole locations and indicate the amount of correction that is needed. Material is then added or removed, as indicated, to achieve proper balance.



*Figure 18: Crankshaft balancing machine.*

As a check, or if the crank, and/or the reciprocating mass, is modified after the manufacturer's dynamic balance process, the crankshaft can be statically balanced using a stand, like that in Figure 19, that is designed to allow for the crankshaft to rotate freely when mounted. The stand usually consists of large wheels mounted on smooth operating bearings. The proper bob weight will be attached to the pin. If the crankshaft is balanced, it will not rotate when it is placed onto the stand. If it does rotate, then the heavier end will rotate to the bottom. If the pin side is heavy, the counterweights will be increased. If the counterweight is heavy, weight will remove from the counterweight in the direction of the vector that is pointing directly downward. This technique can only be used with crankshafts that are one-hundred percent balanced to

themselves. To identify the mass to be removed or added the balance refers to the calculations discussed in section 3.1.1.



*Figure 19: Crankshaft mounted on a static balance stand.*

### **3.3 Parts are Physically Measured to Verify Accuracy**

#### *3.3.1 The Connecting Rod*

There are a several techniques used to check the weight and balance of the connecting rod. It is always good to weigh every part that is going to be rotating or reciprocating and compare it to the weight generated by the computer-aided analysis to ensure that they agree. If there are any discrepancies, the cause of the discrepancies should be investigated and corrected. Since the center of gravity serves as the separation point between the large and small ends, the

center of gravity location must be identified and compared to the computer-analyzed position. To find the center of gravity, the connecting rod can be balanced on a pin or blade. The pin position can then be marked on the connecting rod and it can then be measured to verify that it is in the proper location according to the computer-generated data. Figure 20 shows a picture of a connecting rod mounted on a mill to measure the mark made by balancing it on a pin. The position determined by this method was within 0.02 inches from the position calculated from the 3-D computer solid modeling program.



*Figure 20: A Connecting rod mounted on a mill to measure the mark made by balancing it on a pin.*

Another way of confirming the large end and small end weights of the connecting rod is to place the connecting rod in a fixture that will allow it to be truly horizontal, and have one end of the rod suspended while the other end is placed on a scale and can be measured. This method

is widely used by performance shops because it is an inexpensive and simplistic way to obtain both the large and small end weights. A picture of this setup can be seen below in Figure 21. For one of the connecting rods used in one of the engines, the computer analyzed weight of the large end was 1015.9 grams, the small end was 657.5 grams, and the total was 1673.4 grams. For the same connecting rod, the physically measured weight for the large end was 1262.4 grams, the small end weighed 411.4 grams, and the total weight was 1673.0 grams. Measuring both ends separately and then combining them, produced a total weight of 1673.8 grams. There was a 0.8 gram discrepancy between this value and the value obtained measuring the weight of the connecting rod as a whole, and a 0.4 gram difference from the total weight determined by the computer analysis of the 3-D model. Both methods result in values that agree quite well and the slight discrepancy is negligible. Investigating the weight of the large end of the rod, 1015.9 grams for the computer analyzed weight and 1262.4 grams for the measured weight, there is a large difference between how each method distributes the weight between the large end and small end. For all of the balancing of the other engine components, where either the small end or large end weights are needed, the computer-analyzed weights were used. The computer-analyzed weights were used because, in the design process, these weights are needed for the design of the crankshaft and balance shafts.



*Figure 21: A connecting rod being weighed to find the large and small end weights.*

### *3.3.2 The Balance Shafts and Weights*

For validation, any reciprocating or rotating part was weighed and compared the computer-analyzed weight of the model. All of the components of the balance shafts went through this process. The balance shafts themselves, the balance shaft weights, the bolts, and the keyways are measured with the shafts assembled and then again once they are dismantled. If there are any discrepancies between the physically measured value and the computer-analyzed value, the part is assessed and any necessary modifications are made. A high-precision scale is used to collect the physical measurements of the weight of each component.

## CHAPTER 4: TESTING

### 4.1 Testing Procedure

A dynamometer was connected to the engine to motor the engine for testing. The engine was run from 300 RPM to 2000 RPM in increments of 50 RPM. At each speed, the engine was motored until the set speed had been reached and was stabilized. Data was then captured from three accelerometers mounted to the crankcase in the X, Y, and Z directions of the engine. The Z direction was in the axis of reciprocation. The Y direction was in the axis of crank rotation. The X direction is perpendicular to the engine's side. The connecting rod rotates in the X-Z plane. Data was taken for 15 seconds at a data capture rate of 50 kHz using time-based data acquisition with a 25 kHz cut-off filter. After all of the data points had been captured, the data was then run through Matlab and Fast Fourier Transform (FFT) and plots were obtained. The FFT plots were then assembled into a Campbell diagram for further data analysis. Combustion data was also taken in a similar manner at 1500 RPM and 2000 RPM.

The first configuration test engine was run at 800, 1000, 1500, and 2000 RPM. At each speed, the engine was motored until the set speed had been reached and was stabilized. Data was then captured from three accelerometers mounted to the crank case in the X, Y, and Z directions of the engine. The Z direction was in the axis of reciprocation. The Y direction was in the axis of crank rotation. Data was taken crank-angle based, taking a data point at each degree of rotation of the crankshaft. For each speed the data capture rate is different due to how the data was captured. The data capture rate for 2000 RPM works out to be 12 kHz and would be slower for the slower speeds. The data was processed similarly as that of the second configuration test engine through Matlab to obtain FFT plots.

## 4.2 Fast Fourier Transform (FFT)

Through FFT, time-based data is converted into frequency-based data. The Fourier Transformation takes any complex signal and breaks it down into an infinite number of distinct frequency sine waves having different magnitudes. Below, in Figure 23, both the time-based data and its frequency-based response can be seen. From the frequency response, it can be observed that the time-based data is made up of two sine waves at different frequencies and two different magnitudes. The sum of these two sine waves will produce the complex time-based signal. The FFT allows for the collection of a time-based signal and calculates the frequencies and magnitudes of the sine waves that are apparent in the time-based data.

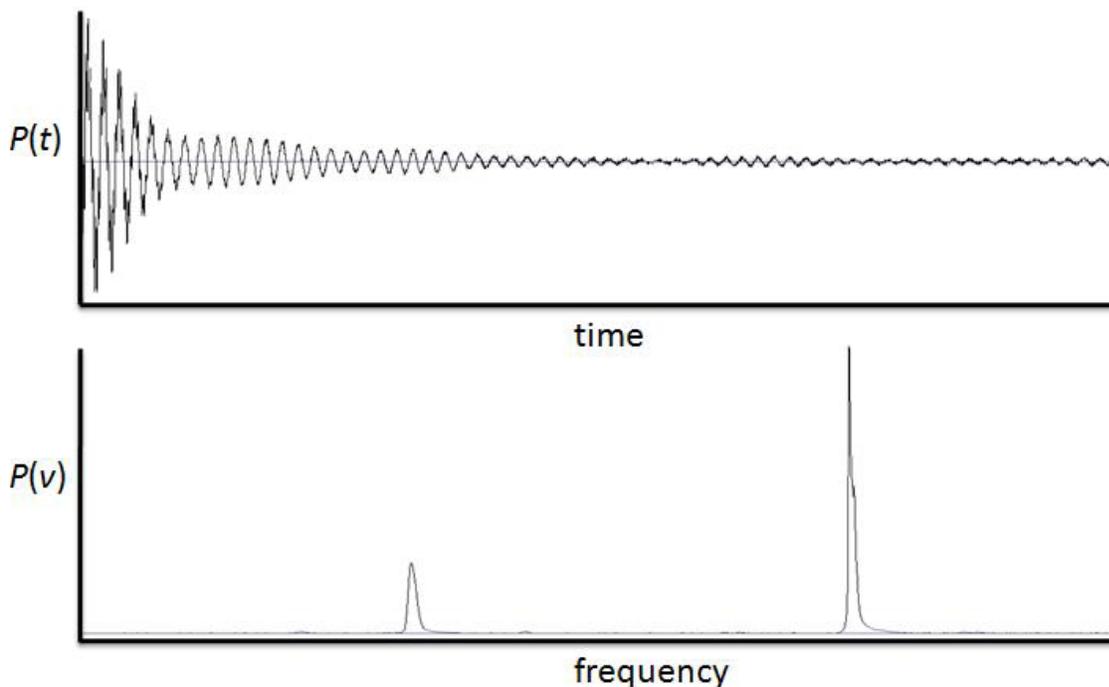


Figure 22: The top graph represents a complex time-based signal, and below it is a FFT graph showing that the complex signal can be broken into two different sine waves of differing frequency and magnitudes. Reprinted from Fast Fourier Transform in Wikipedia (2014), Retrieved November 18<sup>th</sup>, 2014 from [http://en.wikipedia.org/wiki/Fast\\_Fourier\\_transform](http://en.wikipedia.org/wiki/Fast_Fourier_transform).

### 4.3 Campbell Diagram

A Campbell diagram is a 3-D plot that shows the frequency versus speed versus magnitude of the frequency. Campbell diagrams are commonly used in rotational systems to analyze critical frequencies, wearing parts, failure modes, and natural harmonics within a system. For example, within turbines and turbo machinery, the flex and vibration of the turbine blades can be analyzed. Using a Campbell diagram to investigate a gear box reveals gear interactions, gear damage, gear wear, and bearing failure. In all cases, natural harmonic frequencies will also be easily identified with the Campbell diagram. Campbell diagrams show order lines which represent the reaction of the system to some event related to the rotation of the device. As the rotational speed increases, so will the frequency at which the event takes place. Figure 24 depicts an example of a Campbell diagram. The order lines through the graph that increase in frequency as speed increases can be seen as a light line compared to its surroundings. The order of each line is labeled on the right side of the graph. The magnitude of the event is color coded from blue to red; the higher the magnitude of the response, the deeper red it will become. The vertical areas or lines at certain frequencies are the natural frequencies of the system. They occur at 210 Hz, 300 Hz, 510 Hz, 1140 Hz, and 1440 Hz. The interaction of the orders with the natural frequencies can be viewed as the natural frequency lines cross order lines. The orders that will excite the natural frequency more than others can be identified as the higher magnitude areas. The fourth order excites the 300 Hz natural frequency more than the other orders. From this data, the system can be either redesigned to change the factor that corresponds with the fourth order event, the system can be operated in a way to stay below the 300 Hz natural frequency, or the time that the engine operates within that region can be minimalized.

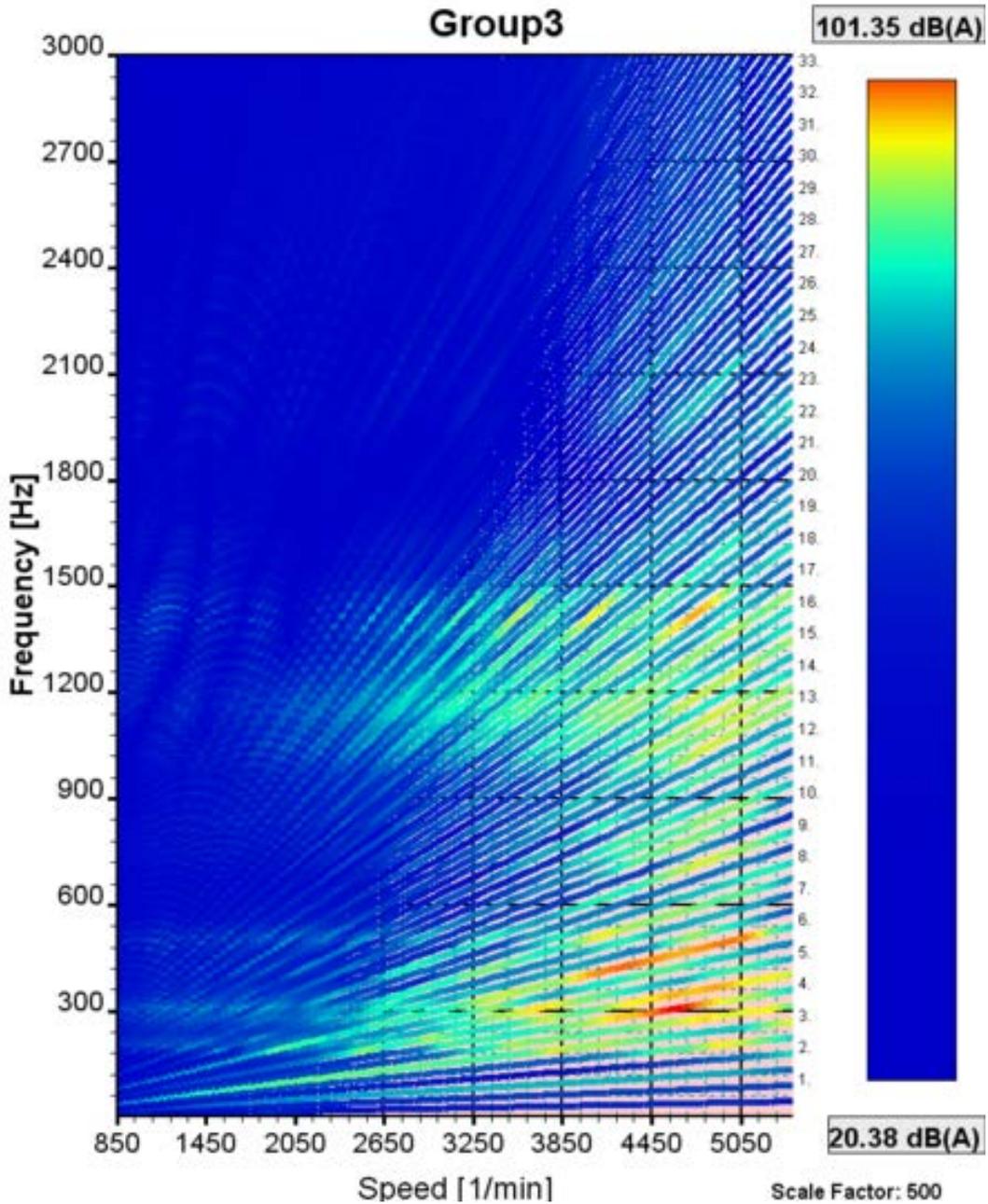


Figure 23: An example of a Campbell diagram showing order lines, natural frequencies, and their interaction. Reprinted from Campbell Diagram, Bing Images (n.d.) Retrieved November, 2014 from <http://www.mnoise-software.com/References.3908.0.html> Copyright 2014 by Engineering Center Steyr. Reprinted with Permission.

## CHAPTER 5: RESULTS AND DISCUSSION

### 5.1 Second Configuration Test Engine Motoring Results

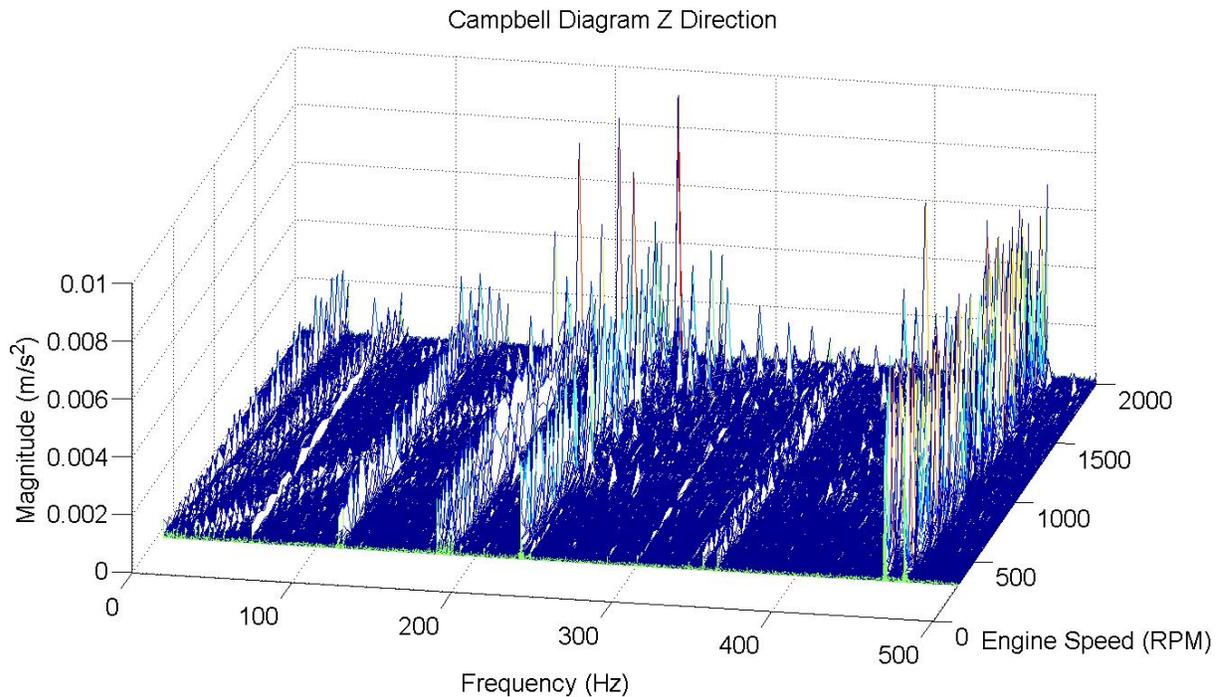
The research relayed in this publication was performed to investigate the reaction of the system to the addition of second order balance shafts into a single-cylinder optical engine. The response of the engine was captured using accelerometers. The accelerometers show the reaction of the system from the point on the engine to which they are mounted. One of the most effective ways to illustrate these results is to display them in a Campbell diagram. Below, in Figure 24, a Campbell diagram of the reciprocating axis, the Z direction, is shown.

It is important to identify the natural frequencies of the system so that these ranges can be avoided in the operation of the engine. These frequencies can be identified easily within the Campbell diagram and are represented by vertical lines running through the diagram. The natural frequencies of the system between 0 and 500 Hz are: 60 Hz, 115 Hz, 172 Hz, 187 Hz, 230 Hz, 350 Hz, 457 Hz, and 470 Hz. The harshness of each natural frequency is shown by the magnitude of the line. The natural frequencies at 457 Hz and 470 Hz are more severe than the other natural frequencies within the system.

The first and second-order effects within the engine are of the most interest. Within the Campbell diagram, the order effects can be seen as lines running at an angle through the diagram. The lower orders start on the left side of the diagram and are nearly vertical at this initial point. As the order increases, the order lines become more horizontal. Three orders can be seen in the motoring results of the second configuration test engine through the entire speed range. These orders are the first-order, second order, and the 5.5 order. At speeds above 1500 RPM, the third order, fourth order, eighth order, and tenth order also begin to show up within the

Campbell diagram. The order lines can be identified by looking at the ending frequency of the line at 2000 RPM. The first order frequency is 33 Hz, second order is 66 Hz, third order is 100 Hz, fourth order is 133 Hz, five and half order is 183 Hz, eighth order is 266 Hz, and the tenth order is 333 Hz.

The first and second order lines indicate that the magnitude of the response increases with speed. This implies the presence of an unbalanced mass within the reciprocating parts. For this configuration there was a 116 g mass within the reciprocating parts that was not balanced by the balance shafts. This response within the Campbell diagram is due to the rotational speed, specifically the  $\omega^2$  term, of the force calculation. For the 5.5 order, the response had the same magnitude over the full speed range from 300 RPM to 2000 RPM. Through the process of investigating all of the rotating parts, such as the balance shafts, crankshaft, flywheel, driveshaft, gears within the geartrain, and bearings, it was discovered that the bearings containing 11 ball-bearings that were used in four locations in the engine, all rotated at half the frequency of the crankshaft rotation giving a 5.5 order effect. Two of these bearings were located in the distributor shaft housing and two more were located in the jack shaft housing. This 5.5 order frequency showed up in all three planes, and this prevalence may be due to the fact that the accelerometers were mounted very near the locations of the bearings. The bearings should be monitored by re-running the same test and comparing the magnitudes of the response for this order. If the magnitude of the 5.5 order increased, this would indicate that one of the bearings is failing and should be replaced. The fourth order excites the harmonic at 115 Hz along with the eighth order exciting the harmonic at 230 Hz at around 1800 RPM. This is represented by the very high peak magnitudes within the Campbell diagram.



*Figure 24: A Campbell diagram generated using the data from the motoring of the second configuration engine.*

Comparing the FFT plot of all three planes (X, Y, and Z) simultaneously can also give insight into the system. Below, in Figure 25, the FFT plots for each plane have been graphed together for the speed of 2000 RPM. Lines have been placed through specific orders where excitation has occurred. The first thing to note is that the scale on the X-axis is twice that of the Y and Z axes. At the first-order frequency of 33 Hz, the X plot does not show any excitation whereas, both the Y and Z directions have noticeable peaks. This indicates that the crank, flywheel, and driveshaft are all balanced correctly. There is, however, a possibility that the system is not agitated by this harmonic in the X direction. If there was any imbalance in these parts, both the X and the Z plot would have shown a spike since the direction of rotation is on the

X-Z plane. Some effect at the first-order frequency from the moment created by the piston and piston pin offset within the X direction would also be expected in the presence of any imbalance. Since there is no noticeable effect in the X direction, it can be assumed that the moment did not have a significant effect on the system. Within the plots, the order harmonics, and also the natural frequency harmonics of the system can be seen. Looking at just the FFT plot without a Campbell diagram, it is harder to distinguish between order harmonics and natural frequency harmonics because both harmonics look the same. Using both the FFT plot and the Campbell diagram together, it is easy to identify the harmonics and to find the values of each peak.

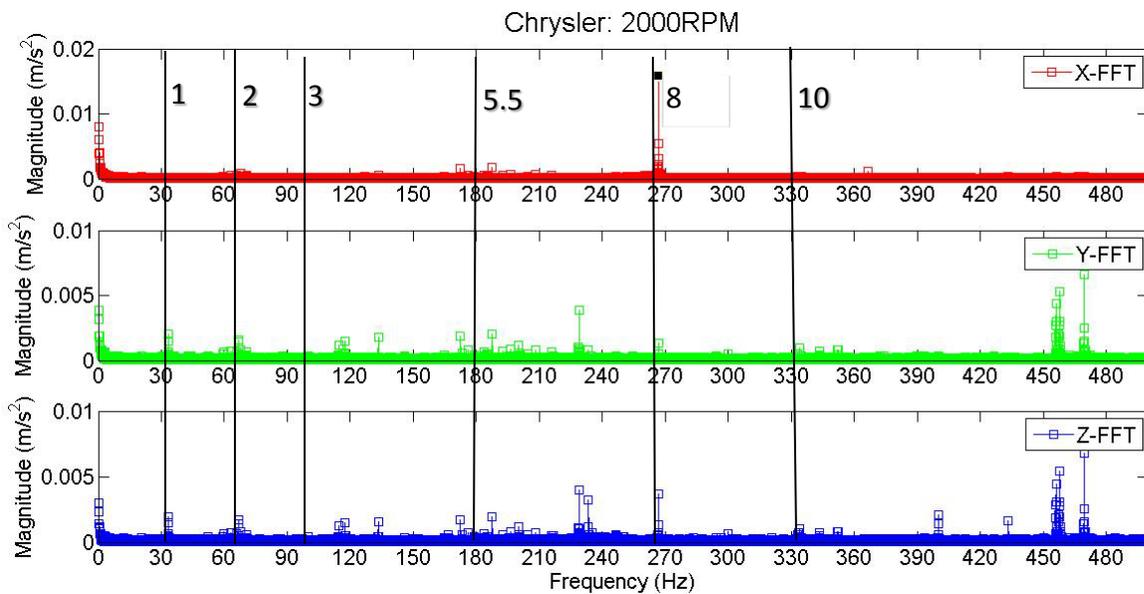


Figure 25: A FFT Plot generated using the data from the motoring of the second configuration test engine at 2000 RPM.

## 5.2 Second Configuration Test Engine Firing Results

The reaction of the system due to the combustion event within the engine was investigated. In Figure 26, FFT plots for each direction have been graphed together for the speed of 2000 RPM at which combustion was occurring. It can be determined, by comparing Figure 25

and Figure 26, that each of the magnitudes in the combustion data increased by an order of magnitude. This excitation in the magnitudes may be due to the torque change caused by the pressure created in the cylinder by combustion. There is also a half-order spike at 16 Hz that represents the combustion event that occurs once every two revolutions.

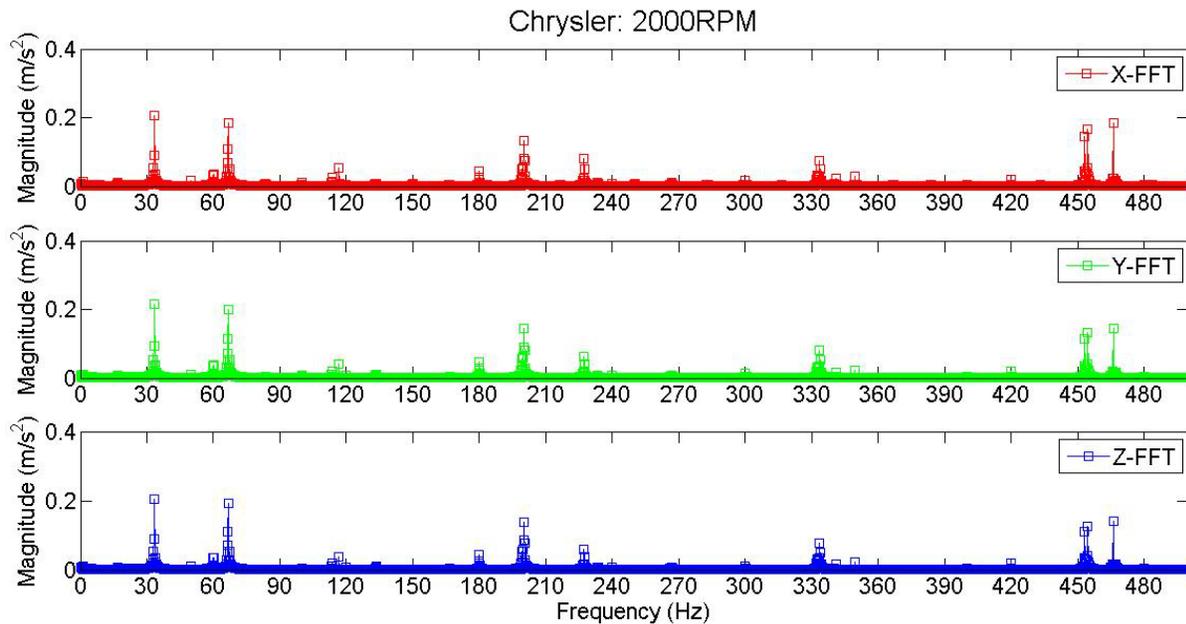


Figure 26: A FFT Plot generated using the data from combustion testing of the second configuration test engine at 2000 RPM.

### 5.3 First Configuration Test Engine Motoring Results

The main differences between the data taken from the first configuration test engine and the second configuration test engine are due to the fact that the first configuration test engine does not have second-order balance shafts. Also, the data was captured in a different way. The data in the first configuration test engine was captured crank-angle based, and the data captured in the second configuration test engine was time-based. The data captured within the second configuration test engine was also taken at a higher data capture rate, which produced results at a

better resolution. A high frequency filter with a cut-off at 25 kHz was also used to analyze the second configuration engine, making the data cleaner with less electrical noise in it. Comparing the results from the first configuration test engine (Figure 27) to those of the second configuration test engine (Figure 25) reveals that the results are similar, though the scales are slightly different. The first-order response of the first configuration test engine was 0.0000846 m/s<sup>2</sup> compared to 0.0000195 m/s<sup>2</sup> for the second configuration test engine. The second-order response of the first configuration test engine was 0.0000648 m/s<sup>2</sup> compared to 0.0000174 m/s<sup>2</sup> for the second configuration test engine. In comparing these results, the intent is to find the response resulting from the addition of the second order balance shafts. It is expected that the response will demonstrate that the addition of the second order balance shafts does actually help the response of engine. The response in the reciprocating axis appears to indicate that the addition of second-order balance shafts improves the engine's response by a factor of four. However, due to the different variations within the two configurations, this direct comparison should not be made. The best way to analyze the effects of the addition of the second order balance shafts is to remove the shafts from the second configuration engine and compare them to the data initially taken from the same second-configuration engine with the balance shafts installed. The test should be run in the same manner as the second configuration test engine was run initially. This approach should have been executed, but due to many factors beyond the control of the lab, the test could not be done in this way.

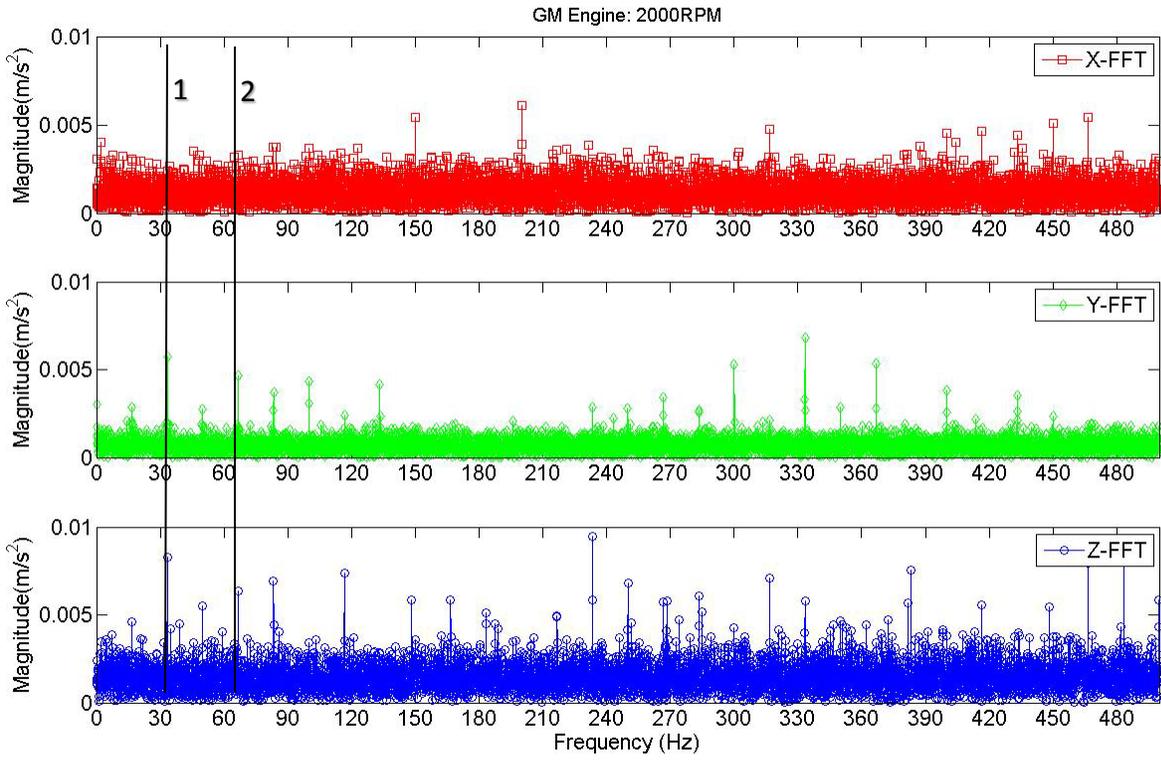


Figure 27: A FFT plot generated using the data from the motoring of the first configuration test engine at 2000 RPM.

## CHAPTER 6: CONCLUSIONS

Within this publication, both analytical and physical processes to balance the components within a single cylinder engine have been identified and discussed. These processes can also be expanded to be used in multi-cylinder engines. Also, a test procedure has been developed to evaluate the dynamic response of a single-cylinder test engine along with data processing techniques that can be used to clearly display the data.

From this research the following conclusions have been reached:

1. The natural frequencies of the second configuration test engine system, between 0 and 500Hz, are: 60 Hz, 115 Hz, 172 Hz, 187 Hz, 230 Hz, 350 Hz, 457 Hz, and 470 Hz.
2. Within the second configuration test engine, the response of the following order harmonics that are present at 2000 RPM between 0 and 500 Hz have been identified to be: first order (33 Hz), second order (66 Hz), third order (100 Hz), fourth order (133 Hz), five and one-half order (183 Hz), eighth order (266 Hz), and tenth order (333 Hz).
3. Noise from the bearings have been identified and could be a sign of initial bearing failure and should be monitored for replacement.
4. The first and second-order responses of the system are low, utilizing the bearing noise response as an indicator. The response of the bearing noise within the system should be very small.
5. The balance of the crankshaft, flywheel, and driveshaft is excellent and the procedure for dividing the connecting rod within a 3-D model is very effective.
6. Combustion within the second configuration test affects the vibrational response of the system dramatically.

7. The response of the system from the moment created by a combined piston and piston pin offset of 10mm, within the second configuration test engine, is minimal.

## CHAPTER 7: RECOMMENDATIONS

There are several other topics that would be interesting to study that are related to this research. One of which would be the effects of the head on the vibration of the engine. The head has several moving parts, including valves and camshafts. In high-performance engines it is becoming popular to also balance the camshafts. This test could be performed by disconnecting the drive belt from the head in either test configuration engine. Another interesting topic that arose during this research was how the connecting rod weight is split between the large end of the rod the small end of the rod. Several engine shops that balance crankshafts were contacted and they primarily use the physically measured values discussed in 3.3.1, but after performing the physical measurements it was discovered that there was a large difference in the weight distribution between the large end and the small end relative to the 3-D analyzed weight distribution. For the experiments in this publication, the 3-D analyzed weight distribution was used and the results were excellent. To complete this experiment, the crankshaft would have to be re-balanced and new balance-shaft weights would have to be manufactured for one of the test engines. Analyzing the response due to the addition of the second order balance shafts within the second configuration by first removing the shafts and running the same initial test would be very beneficial to this research. For any future engine designs that have piston offset or piston pin offset, the balance shafts should be placed directly under the cylinder so that the moment discussed in 3.1.2 is not created.

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## REFERENCES

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