

THESIS

IDENTIFICATION OF DYNAMIC STRAIN MONITORING LOCATIONS FOR THE MAIN
BEARINGS OF AN INTEGRAL ENGINE-COMPRESSOR USING FINITE ELEMENT
MODELING

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Alexander Tomsick

Department of Mechanical Engineering

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Colorado State University

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Master's Committee:

Advisor: Daniel Olsen

Co-Advisor: Mitchell Stansloski

Paul Heyliger

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ABSTRACT

IDENTIFICATION OF DYNAMIC STRAIN MONITORING LOCATIONS FOR THE MAIN BEARINGS OF AN INTEGRAL ENGINE-COMPRESSOR USING FINITE ELEMENT MODELING

The aging fleet of integral engine-compressors remains the backbone of the North American gas transmission industry. Replacement of these integral engine-compressors is currently not cost-effective due to the extreme operational availability requirements. Given this, it is more important now to utilize a preventative maintenance program that uses condition-based data, which measures performance and health metrics to guide maintenance actions. The health of most components of an integral engine-compressor can be understood through the collection of routine performance analysis data. However, the typical technologies utilized during these performance analyses do not adequately offer insight into the condition of the main bearings.

This research aims to evaluate the measurement of dynamic strain on the main bearing saddles of a large integral engine-compressor to measure their condition during operation. To determine the most effective locations for measuring dynamic strain on the main bearings of a reciprocating machine, a time-dependent finite element model of the rotating assembly of an Ingersoll Rand KVSR 412 was created in ANSYS Mechanical. The loads on the main bearings, which were output from this model, included the effect of both the inertial loads caused by the rotating crankshaft and the reciprocating components, and the gas loads caused by changes in power and compressor cylinder pressures, which varied with time. To ensure accuracy, gas loads applied to the model were calculated using cylinder pressures, which were measured from a KVSR

412 operating at a natural gas compressor station located in Northeastern Utah. The results of the finite element modeling showed that the most effective location for measuring dynamic strain varies between main bearings. This location is dependent on the distance of the nearest compressor cylinder and the number of power cylinders that neighbor each main bearing.

In addition to the finite element modeling, dynamic strain sensors were installed on the main bearings of the same KVSr 412 from which the cylinder pressure data was measured. These sensors were installed for the collection of main bearing strain data during the operation of the integral engine-compressor. This data collection is planned as part of the future work of this research.

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DEDICATION

To my wife, Katelyn, for her constant support.

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LIST OF SYMBOLS

Symbol	Description
a	Acceleration
C	Damping
D_{Comp}	Compressor Cylinder Bore
D_{Pwr}	Power Cylinder Bore
D_{Rod}	Compressor Piston Rod Diameter
e	Eccentricity Ratio
E	Modulus of Elasticity
F	Force
F_{Comp}	Combined Compressor Cylinder Gas Load
F_{Pwr}	Power Cylinder Gas Load
G	Shear Modulus or Balance Quality Grade
Hz	Hertz
K	Stiffness
L_{avg}	Magnitude of the TWAL Vector
L_x	X-Direction Component of Bearing Load
L_y	Y-Direction Component of Bearing Load
m	Mass
N	Crankshaft Rotational Speed
n	Number of Samples
P_{CE}	Crank-End Compressor Cylinder Pressure
P_{HE}	Head-End Compressor Cylinder Pressure
P_{Pwr}	Power Cylinder Pressure
t	Time
U_{per}	Permissible Residual Unbalance
U_{res}	Residual Unbalance
u	Displacement
V	Volt
W	Rotor Weight
x	Time Domain Amplitude
γ	Attitude Angle or Shear Strain
ε	Normal Strain
θ_{Crank}	Crankshaft Angular Position
$\mu\varepsilon$	Microstrain
ν	Poisson's Ratio
σ	Normal Stress
τ	Shear Stress

CHAPTER 1 – INTRODUCTION

1.1 Justification of Project

1.1.1 Background

In 2004, the gas transmission industry operated approximately 4,000 integral engine-compressors, with a median age at that time of 50 years and a median size of 2,000 horsepower [1]. The aging infrastructure of integral engine-compressors still serves as the backbone of the U.S. natural gas transmission industry, moving natural gas through 305,000 miles of transmission pipelines [2]. Despite their age, the associated costs of replacement, as well as logistics such as operational availability, inhibit their replacement [1]. Given this, it is more important than ever to utilize a preventative maintenance program based on condition-based data, which evaluates the performance and health of these reciprocating machines [3]. This approach aims to reduce unexpected downtime and maintenance costs while being able to identify potential failure conditions that are present in a machine before a catastrophic failure can occur [3]. The failure of industrial equipment leads to losses in production and increased maintenance costs, with maintenance costs accounting for between 15% and 60% of the manufacturing costs of final products [4]. Operational performance analysis of reciprocating engines and compressors can provide key insights into the health of reciprocating equipment. Data that are collected in a typical performance analysis study includes the analysis of vibration, indicated horsepower, temperature, and pressure data [5]. Using these data, the mechanical integrity and performance of the power and compressor cylinders and their valves can be measured directly. However, the health of the engine or compressor main bearings is more difficult to diagnose using typical monitoring technologies.

1.1.2 Existing Monitoring Technologies and Their Limitations

Typical main bearing health monitoring technologies for large bore separable engines and compressors, as well as integral engine-compressors, include the measurement of main bearing temperature and the measurement of vibration acceleration on the engine/compressor frame near the main bearing locations. Most large reciprocating engines and compressors utilize hydrodynamic bearings to support the crankshaft. The load-carrying capacity of these hydrodynamic main bearings depends greatly on the minimum oil film thickness of the bearing, as well as the bearing babbitt temperature [6]. Temperature monitored from the bearing embedded temperature transducers, either Resistance Temperature Detectors (RTDs) or thermocouples, provides the most basic data measurement, which can be trended over the engine-compressor's operation and alert the equipment operator to changes in bearing health [6]. To ensure an accurate measurement, the embedded temperature transducers are installed in the main bearing at the circumferential location where peak oil film pressure and minimum oil film thickness are expected [6]. Increases in main bearing temperatures can occur due to many different conditions, which may include insufficient bearing clearance, excessive load, and inadequate lubrication [6]. While bearing temperature measurements are a useful tool to bring attention to a potential fault, they offer little insight into the potential causes that have led to high bearing temperatures and are primarily useful as a trigger for equipment shutdown instead of diagnostics.

For further diagnostic analysis, engine and compressor frame vibration measurements are taken near each main bearing location. These measurements are taken with an accelerometer, which is a vibration transducer that generates an electrical output proportional to the frame's acceleration [3], [7]. The engine-compressor frame's acceleration is the result of the time-dependent forces generated during the machine's operation [8]. The time-dependent loads created

by the reciprocating inertia, rotating unbalance, and the significant gas loads due to combustion act on the main bearings and are transmitted into the integral engine-compressor's frame, leading to oscillatory motion of the frame at the periodic frequencies at which the forces occur. The basic operation of the accelerometer relies on Newton's second law of motion, which states that the force on a body (F) is proportional to the body's mass (m) multiplied by its acceleration (a) [3], which is shown in Equation 1.

$$F = m a \quad (\text{Eq 1})$$

The most used industrial accelerometer is a piezoelectric accelerometer. To measure the acceleration of the integral engine-compressor's frame, the accelerometer is fixed to the frame and oriented in the direction in which the acceleration is desired. When accelerated, a seismic mass internal to the accelerometer, which is attached to the piezoelectric sensing element, causes stress in the piezoelectric sensor element, resulting in an electrical output that is proportional to the measured acceleration [3], [7]. Figure 1 shows a schematic of a typical piezoelectric accelerometer, which may be used in an industrial application such as the one being discussed.

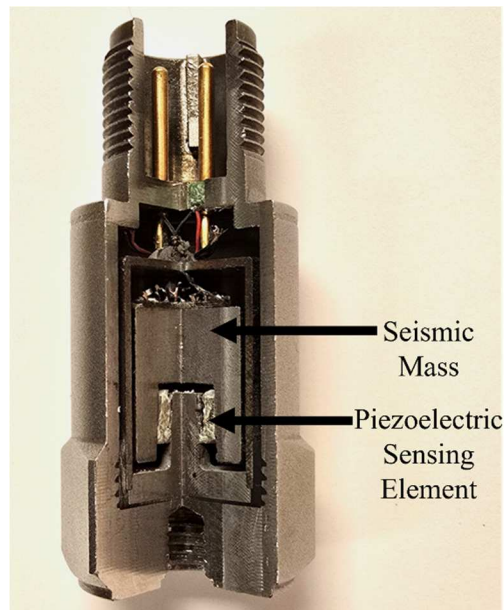


Figure 1: Schematic of a typical piezoelectric accelerometer.

The measurement of frame acceleration provides more diagnostic and analysis capabilities for the health of the main bearings compared to bearing temperature measurements alone. Since the frame acceleration is measured dynamically over time, the acceleration amplitude measured by the accelerometer can be plotted against time or the crankshaft angular position, which is often utilized to identify the cause of events with high acceleration amplitudes throughout the engine or compressor cycle.

However, there are several limitations to the performance of accelerometer measurements collected on the exterior of the frame to monitor the health of the main bearings. There are three main limitations of this measurement methodology that were identified. First, the distance between the location at which the acceleration is measured at the exterior of the frame and the main bearing is significant on a large engine or compressor. This leads to poor transmission of the vibratory response at the main bearings to the exterior of the frame. Second, acceleration data collected on the side of the frame is generally only feasible with the accelerometer installed in a horizontal orientation pointing towards the crankshaft. In this orientation, this measurement axis is parallel with the compressor cylinder in a typical integral engine-compressor, but there is no location available at which the accelerometer can be mounted that measures on the axis of the power cylinders while still being close to the main bearings. Finally, integral engine-compressors have many sources of vibration, which may be mechanically or acoustically driven. These sources include the rotating and reciprocating motion of internal components, the opening and closure of valves, and the pulsation of compressed gas as it exits the compressor cylinder [9]. These forces affect the frame vibration, making it difficult to identify which components of the measured acceleration are related to the main bearings directly and what components can be attributed to a secondary source [9].

For this reason, a monitoring technology that can measure directly at the main bearings would be more beneficial to monitor the health of the main bearings. Similar to large reciprocating equipment, many styles of rotating equipment, including large motors, pumps, fans, turbines, and compressors, utilize hydrodynamic bearings to support rotating components. In the same way as reciprocating equipment, it is critical to measure the health and condition of these hydrodynamic bearings. To this end, the transducer that is most often utilized for this purpose is the proximity probe. A proximity probe is a non-contacting probe, most commonly an eddy current probe, that measures the displacement of an observed surface relative to the probe mounting location [8]. For hydrodynamic bearing monitoring, perpendicular and coplanar proximity probes are typically used to monitor the shaft position in the available bearing clearance [8].

Figure 2 presents the typical installed proximity probe arrangement used for health monitoring of hydrodynamic bearings.

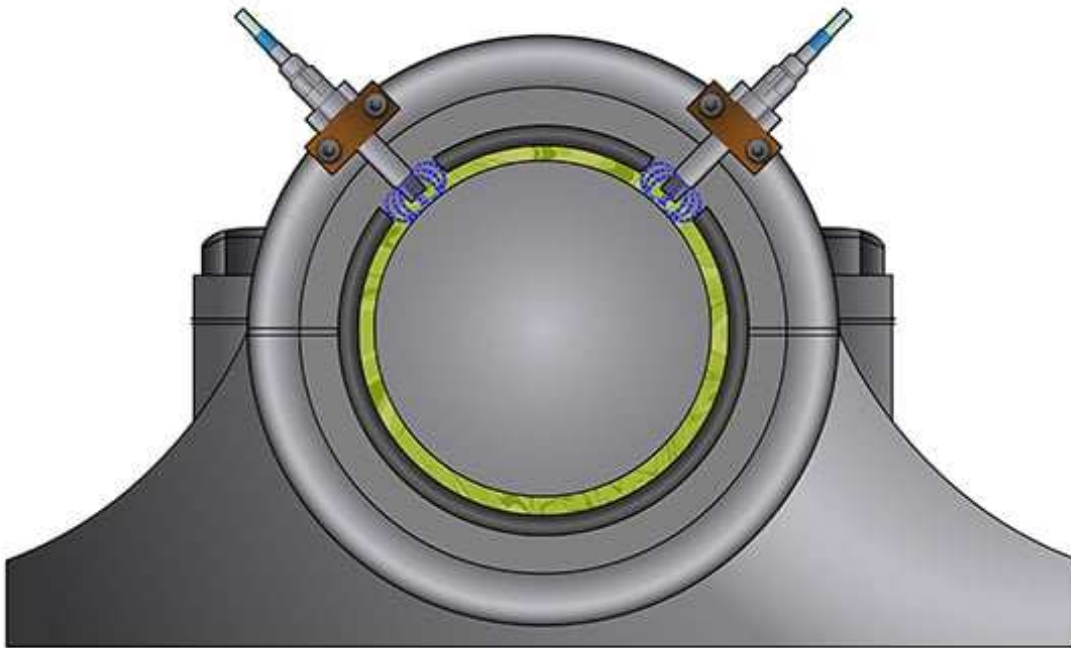


Figure 2: Diagram depicting through-mounted and orthogonally mounted proximity probes [10].

While proximity probes could offer a more direct measurement of vibration at the main bearings, they have several limitations that make them difficult to utilize in reciprocating equipment. The first major limitation is the requirement to retrofit proximity probes into a bearing. If the probes are to be through-mounted, through holes must be drilled in the bearing cap and the bearing insert, and the bearing insert must be counterbored to ensure the material surrounding the probe tip does not affect the resulting data [10]. In bearings that are not prepared by the manufacturer for proximity probe installation, a through-mounted installation requires labor-intensive machine work. The probes can also be mounted externally using brackets, as shown in Figure 3.

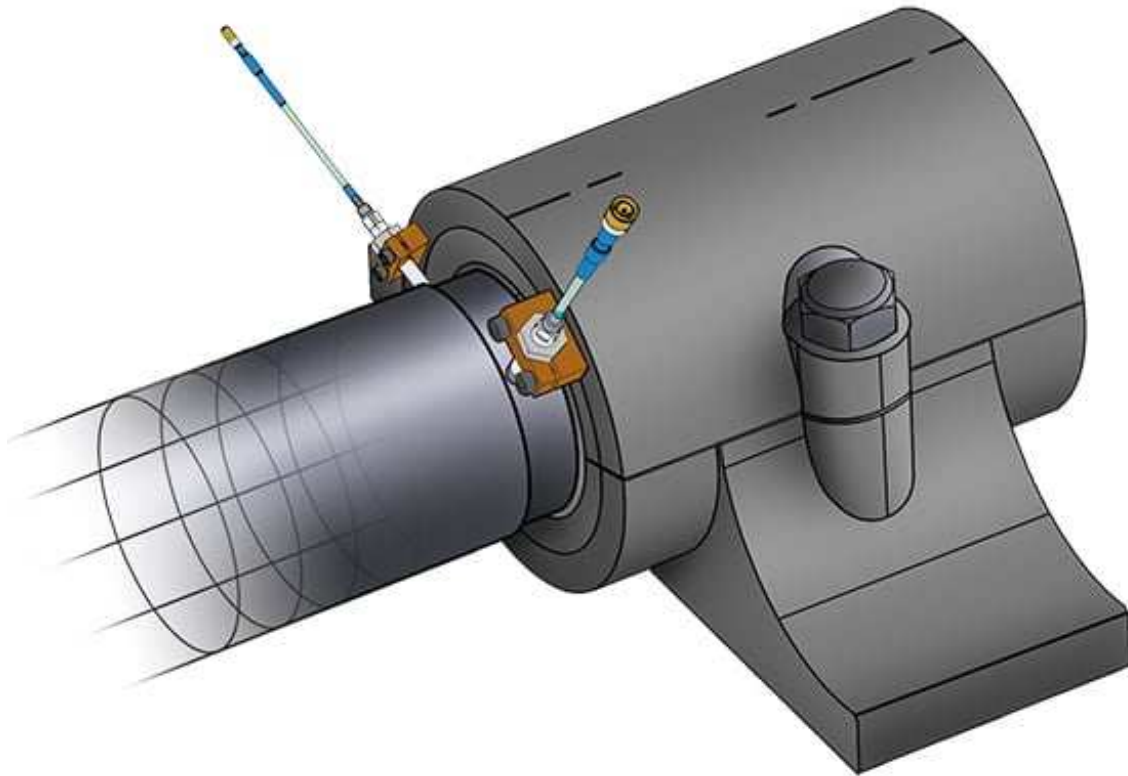


Figure 3: Diagram depicting two orthogonal proximity probes externally mounted on a bearing using brackets [10].

External mounting requires less machine work and does not require the bearing insert to be counterbored for the probe. However, the clearance between the webs of the crankshaft throws and the face of the main bearings is very tight, even in large reciprocating engines and compressors, making this mounting not feasible in many cases. The other main limitation of proximity probes is the effect of shaft runout on the quality of the data measured by the probes. Runout, as it pertains to proximity probes, can be caused by mechanical runout or electrical runout [8]. Mechanical runout can occur due to poor surface preparation, surface scratches or damage, or inconsistent journal geometry [8]. Electrical runout can occur when the electrical conductivity or magnetic permeability varies around the circumference of the shaft [8]. These sources of runout affect the measured shaft displacement in a manner that is not representative of the actual shaft displacement. Many data collection systems, which are designed to measure data from proximity probes, can compensate for the measured displacement associated with the shaft runout. However, this is not true for every monitoring system, and adds more complexity to the use of proximity probes in the monitoring of large reciprocating engine and compressor main bearings.

1.2 Literature Review

1.2.1 Vibration Analysis

Vibration is the oscillatory motion of a physical system that repeats itself over an interval of time [4], [8], [11]. Vibration, which affects all physical systems to some extent, has become a point of interest not just in the design of machines, foundations, and structures, but also in the operation and maintenance strategy of heavy industry [4], [11]. Forces generated by the operation of centrifugal machines, such as motors, turbines, or fans, as well as those generated by the reciprocating machines, such as engines or compressors, lead to vibration. The vibration of machines and structures can lead to material failure from fatigue due to cyclic variation in induced

stress, and can also lead to premature wear in machine components, leading to failure and excess noise [8], [11].

The breakdown and failure of industrial equipment is managed by the maintenance strategies used by the equipment operators. The maintenance techniques used by equipment operators can be separated into three main types, which include breakdown maintenance, preventative maintenance, and predictive maintenance [12]. Since the 1930's, these maintenance strategies can be tracked through three main generations [13]. Before World War II, equipment operators followed a breakdown maintenance strategy, in which equipment was repaired when a breakdown or failure occurred [13]. During and following World War II, maintenance strategies shifted to preventative maintenance, where equipment was maintained through overhauls at fixed intervals [13]. Since the mid-1970's, maintenance strategies in the industrial environment have shifted to predictive maintenance, which focuses on planning equipment maintenance efforts around the condition of the equipment based on collected data and inspections instead of fixed maintenance intervals [4], [13].

Vibration analysis is one of many technologies that may be used in a predictive maintenance strategy for monitoring equipment health, along with others such as the measurement of acoustic emissions, temperature monitoring, and oil analysis [4], [13]. Vibration analysis entails the diagnosis of equipment health based on collected machine vibration data. The machine vibration can be recorded and cataloged using various vibration transducers. A vibration transducer is a device that converts mechanical motion into an electrical signal [8], [10]. Two vibration transducers, the accelerometer and the proximity probe, were discussed in the previous section. Independent of the kind of vibration transducer utilized, the simplest representation of a machine's vibration is a plot that shows the time domain representation of how the measured vibration

amplitude varies through time; this plot is referred to as a time waveform [4], [11]. The electrical signal from this transducer can be acquired by a data acquisition system and rescaled to achieve a representative value of the measured motion over time. Figure 4 shows an example time waveform, which was collected on the frame of an integral engine-compressor near a main bearing location that was measured with an accelerometer.

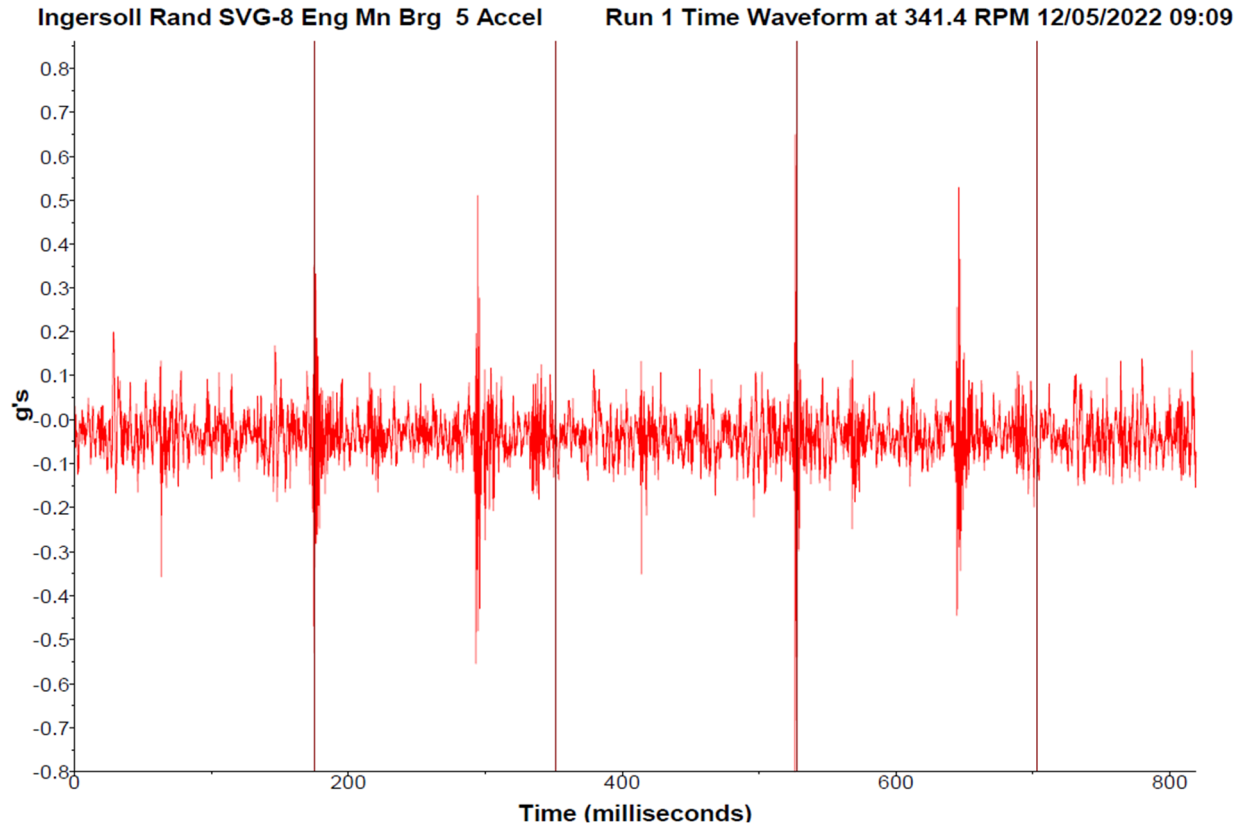


Figure 4: Main bearing acceleration plotted against time, collected on the frame of an integral engine-compressor.

While the time waveform is the most basic representation of the measured vibration, there are limitations in its usefulness. Machinery vibration comprises many different vibration frequencies and their corresponding amplitudes. The amplitude of the vibration time waveform is the magnitude of the vibration at an instance in time and represents the severity of the vibration. While the cumulative amplitude of all vibration frequencies can be inferred from the time waveform, the

amplitude of each discrete frequency is more difficult to determine when many frequencies are present in the vibration data. Additionally, in a vibration waveform in which many vibration frequencies are present, it is difficult to determine each present frequency. For this reason, vibration data is often viewed with a frequency spectrum, which is a representation of the vibration amplitudes of the various frequencies present in the vibration time waveform across a discretized frequency domain [11].

The vibration spectrum is derived by using the Fast Fourier Transform (FFT), which converts the vibration data from the time domain into the frequency domain [4], [11]. This conversion of the time domain data to the frequency domain is completed by a spectral (FFT) analyzer. Figure 5 shows the integrated vibration spectrum calculated from the time waveform data that was presented in Figure 4.

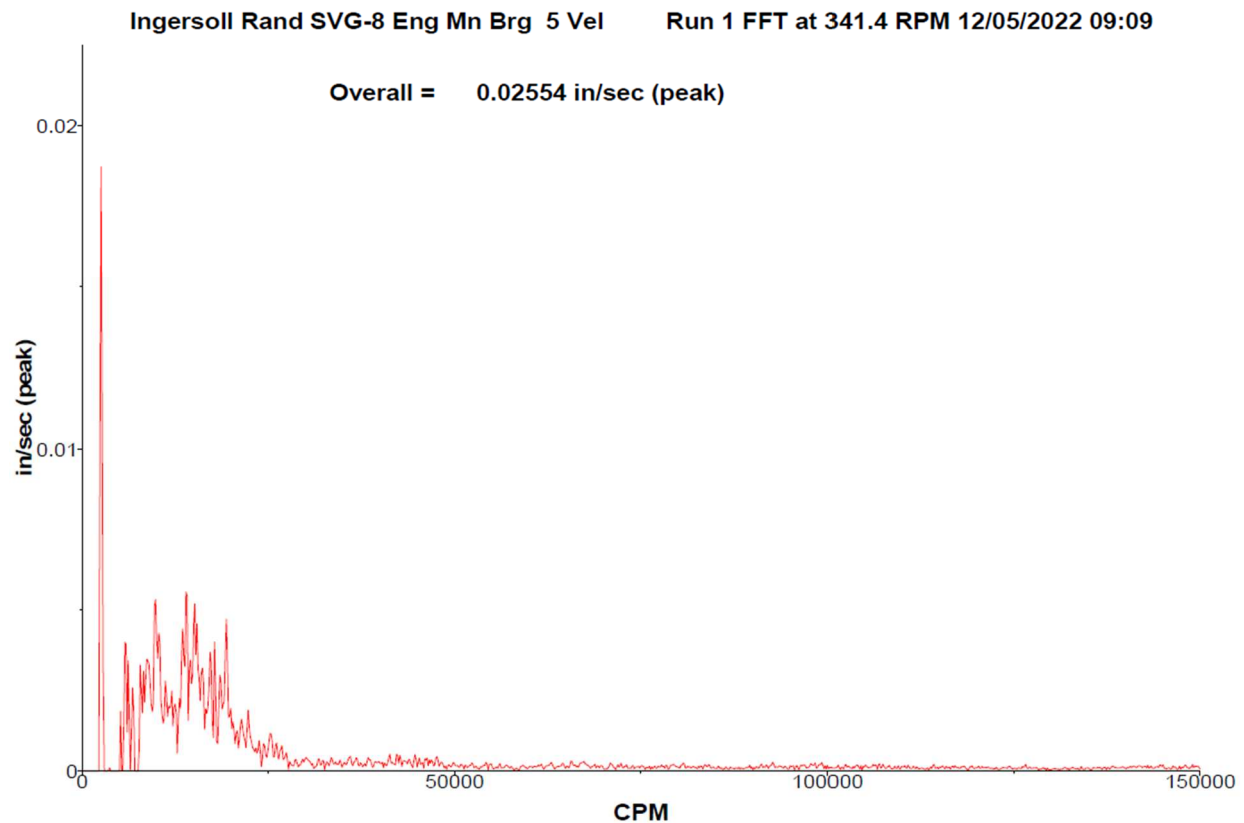


Figure 5: Velocity spectrum calculated from integral engine-compressor frame time domain data.

The vibration spectrum allows the vibration analyst to easily distinguish the main offending frequencies of the measured vibration signal. Since each component of an industrial machine contributes a specific frequency to the vibration signal, the vibration spectrum can be used to detect faults present in a specific component [4].

In addition to the time domain (time waveform) and the frequency domain (spectral) representations of the vibration signal, statistical parameters are often utilized for long-term health monitoring of industrial equipment. The statistical parameter most utilized in vibration analysis is the measure of overall vibration amplitude, which includes peak, peak-to-peak, and Root Mean Square (RMS) signal detection methods [4], [8]. Given a vector of time domain vibration amplitudes, $x(t)$, recorded at discrete values of time, t_i , where $x_i = x(t_i)$. If a total number of samples N is recorded, the overall vibration amplitude using each signal detection method is given by Equations 2 through 4 [4], [8].

$$\text{Peak} = \max(|x(t)|) \quad (\text{Eq 2})$$

$$\text{Peak} - \text{to} - \text{Peak} = \max(x(t)) - \min(x(t)) \quad (\text{Eq 3})$$

$$\text{RMS} = \sqrt{\frac{1}{N} \sum_{i=1}^N (x_i)^2} \quad (\text{Eq 4})$$

In vibration analysis, each signal detection method has benefits depending on the unit of vibration measured. For example, Peak and RMS signal detection methods will often be utilized when measuring the acceleration or velocity of a machine, while a peak-to-peak signal detection method is often reserved for shaft displacement measurements collected using a proximity probe. Regardless of the signal detection method used, the measure of overall vibration amplitude is useful in monitoring for a long-term increase in machinery vibration due to a fault that develops slowly, or alert the analyst of a sudden increase in vibration amplitudes due to a fault that has developed suddenly [4].

1.2.2 Reciprocating Engine and Compressor Introduction

The reciprocating compressor is one of many styles of compressors used in industrial processes. The reciprocating compressor is a positive displacement machine that uses a reciprocating piston to increase a gas stream's pressure [14]. Reciprocating compressors used in natural gas transportation include separable multicylinder reciprocating compressors and integral engine-compressors. These two styles of compressors, while relying on the same compressor cylinder components, differ in the type of equipment that powers the compressor cylinders. Separable reciprocating compressors rely on a separate reciprocating engine or electric motor to drive the compressor crankshaft, which is housed in a separate frame from the driver. An integral engine-compressor utilizes combustion cylinders (power cylinders) that drive a common crankshaft with the compressor cylinders. In an integral engine-compressor, the power and compressor cylinders are mounted to a common frame. The following section provides an overview of the integral engine-compressor operating theory and a description of the equipment components.

1.2.2.1 Integral Engine-Compressor Operating Theory

Figure 6 shows a schematic of the KVSR 412 integral engine-compressor. The power cylinders of integral engine-compressors can operate on a two-stroke or four-stroke cycle. The Ingersoll Rand KVSR 412 integral engine-compressor, the focus of this research, operates on a four-stroke cycle. This means that combustion in each power cylinder occurs only once every two revolutions of the crankshaft. The combustion of the air-fuel mixture contained in the power cylinders forces the piston down and causes the crankshaft to rotate. The crankshaft, which is supported by the main bearings, converts the linear motion of the power pistons into rotational motion. This rotational motion of the crankshaft is converted into the linear motion of the compressor pistons.

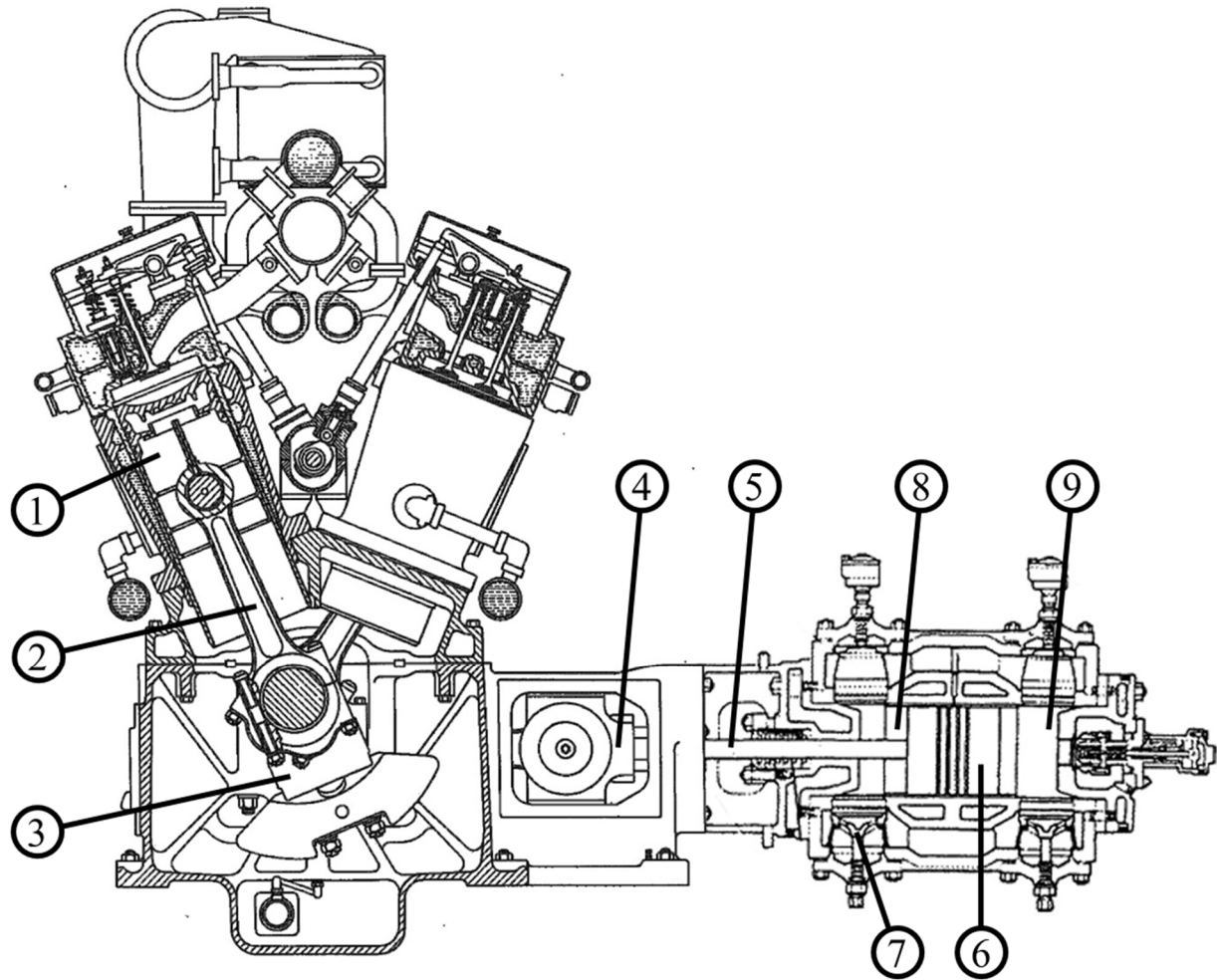


Figure 6: Section view of an integral engine-compressor (Adapted from [15], [16]). (1) Power cylinder piston, (2) connecting rod, (3) crankshaft, (4) compressor crosshead, (5) piston rod, (6) compressor cylinder piston, (7) compressor valve, (8) crank-end compressor cylinder, (9) head-end compressor cylinder.

The power cylinder pistons and the compressor cylinder crossheads connect to the crankshaft journals using connecting rods [14]. A piston rod is used in reciprocating compressors to connect the compressor cylinder piston to the crosshead. The compressor cylinder piston may be single acting, which only compresses gas as the piston travels away from the frame, towards the head-end of the cylinder; but many large reciprocating compressors are dual acting, where gas is compressed on the head-end and crank-end side of the piston [14]. The use of a piston rod allows for flexible packing material to seal around the piston rod, allowing gas to be compressed on both

sides of the piston [17]. The use of double-acting cylinders provides a higher gas flow rate from the compressor cylinders. While the power cylinders may operate on a four-stroke cycle, the compressor cylinders operate on a two-stroke cycle in which gas is discharged from either side of the compressor cylinder once per crankshaft revolution. Gas flow in a reciprocating compressor is controlled using spring-loaded suction and discharge valves, which open when the cylinder pressure exceeds a set point [14].

1.2.2.2 Ingersoll Rand KVSR 412

The following provides information on the configuration of the Ingersoll Rand KVSR 412:

- Four-stroke power cylinder cycle.
- 2,000 BHP
- 330 RPM
- 12 power cylinders (15 ¼" bore X 18" stroke)
- 4 compression cylinders (various bores X 15" stroke)
- 9 main bearings

Figure 7 shows the numbering convention and approximate locations of the KVSR 412 power cylinders, compressor cylinders, and main bearings.

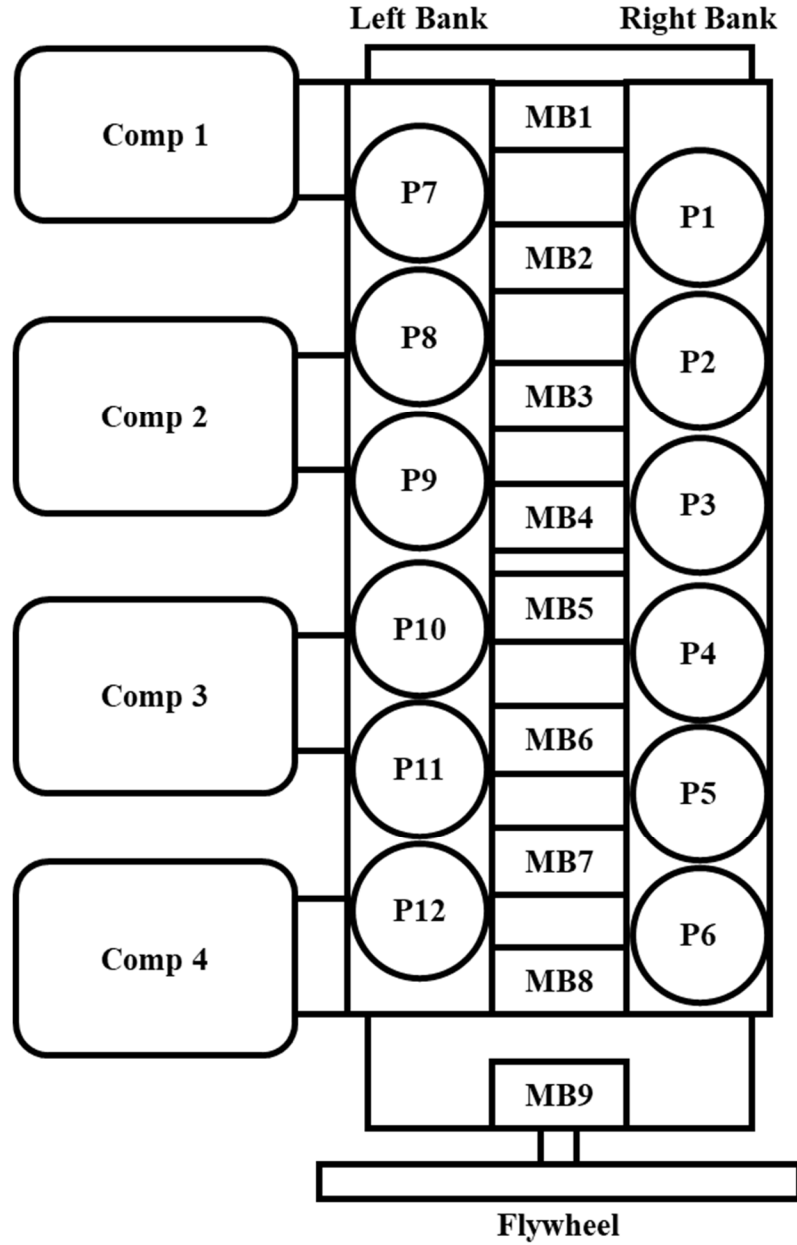


Figure 7: General schematic of the KVSR 412 detailing the number convention of power cylinders, compressor cylinders, and main bearings.

1.2.3 Hydrodynamic Bearing Operating Theory

A fluid film bearing is a bearing that operates in the fully hydrodynamic lubrication regime, lifting the rotating shaft off the bearing surface such that there is no contact between the shaft and bearing [8], [18]. The bearing lubrication regime is dependent on the load per unit length on the

bearing (P), the relative rotational velocity between the rotating shaft and the bearing (N), as well as the oil viscosity (Z). The lubrication regimes of a hydrodynamic bearing during machine startup include dry friction, boundary friction, and fluid friction [18]. Dry friction occurs when the machine rotor is stationary, and asperities on the shaft and bearing surface are in contact. Boundary friction occurs when the developed oil film partially separates surface asperities. Fluid friction occurs when the oil film thickness increases and fully separates surface asperities. The Stribeck curve, presented in Figure 8, relates the bearing coefficient of friction to the bearing parameter (ZN/P), with the three lubrication regimes overlayed over the curve.

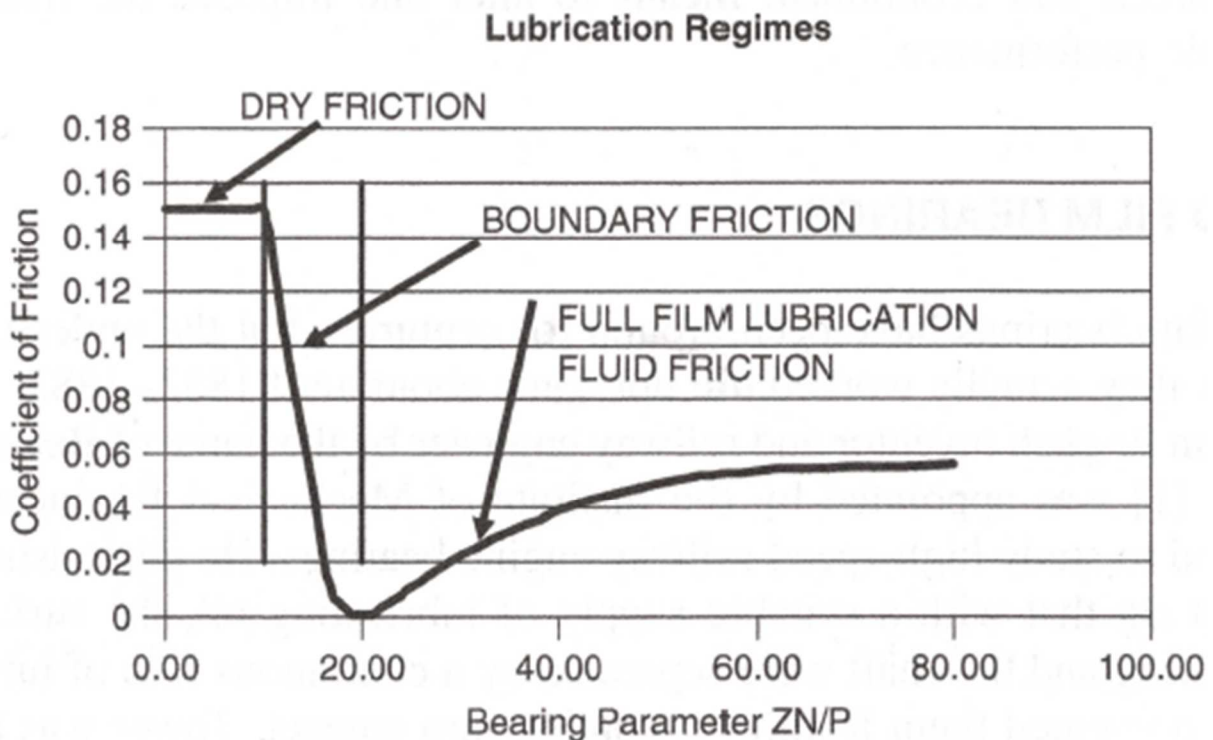


Figure 8: Stribeck curve, which relates the bearing coefficient of friction to the ZN/P bearing parameter (Adapted from [18]).

The region of high oil pressure in a hydrodynamic bearing is developed by the weak pumping action of oil circumferentially around the bearing. As oil clings to the rotating shaft surface, it is drawn into the converging clearance between the shaft and bearing surface. This reduction in

clearance leads to the development of a localized region of high oil pressure, which supports the radial bearing load [8], [18]. The full separation of the rotating shaft and the bearing surface offers hydrodynamic bearings exceptionally low friction and high damping compared to rolling element bearings, while still offering a very high load carrying capacity.

While there are many types of hydrodynamic bearings used in industrial equipment, the simplest hydrodynamic bearing design, which will be discussed in this section, is the plain cylindrical journal bearing, also known as a sleeve bearing [19]. Figure 9 shows the expected oil pressure distribution of a plain cylindrical bearing given a radial force acting downward on the shaft.

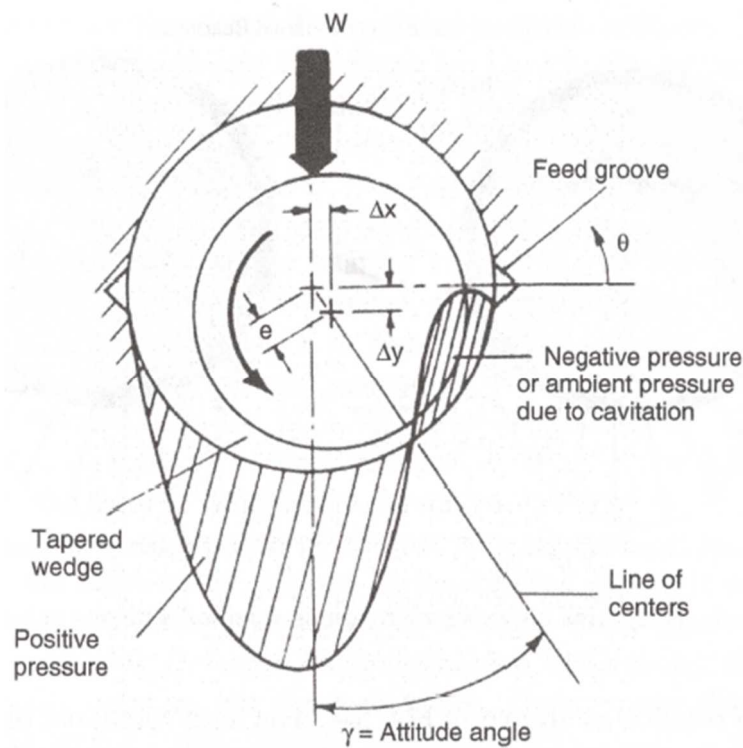


Figure 9: Expected plain cylindrical bearing oil pressure distribution due to a downward load on the rotor [18].

As shown in the pressure distribution, the maximum oil film pressure develops at the location of the converging oil clearance before decreasing in the diverging clearance following the

circumferential location of the minimum oil film thickness. This axisymmetric oil pressure distribution causes the shaft to displace to an equilibrium position, which is offset with components in the load direction and orthogonal to the load direction in the direction of shaft rotation [8], [18]. The displaced equilibrium position of the shaft in the bearing can be defined by two bearing parameters. The bearing eccentricity ratio (e) is the ratio of the distance between the shaft center and the bearing center divided by the bearing radial clearance. The bearing attitude angle (γ) is the angle between the line between the shaft and bearing centers and the load direction. It is important to note that the values of the eccentricity ratio and the bearing attitude angle, and thus the equilibrium position and peak oil film pressure, depend on the magnitude of the load, oil viscosity, and rotational speed of the rotating shaft [20]. However, in plain cylindrical journal bearings, the location of the highest circumferential oil pressure depends on the direction of the radial load acting on the shaft.

1.2.4 Stress-Strain Relationship

When a system of loads is applied to a solid body, the applied loads are transferred to supports, which maintain equilibrium [21]. Under this state of equilibrium, deformation of the body occurs, and distributed forces on internal and external surfaces of the solid body, known as stresses, are generated to limit this deformation [22], [23]. Given complex geometry or complex loading conditions, deformation will not be consistent throughout a solid [22]. In engineering mechanics, the quantity of stress serves as a measure of the intensity of internal forces developed in a body, while strain provides a measure of the intensity of the deformation of the body [22]. Two types of stress can exist in a solid, both of which have dimensions of force per length squared. Normal stress, which is typically denoted by the Greek letter sigma (σ), measures the intensity of the internal forces in a solid that act on a plane perpendicular to the internal force [22], [23]. Shear

stress, which is typically denoted by the Greek letter tau (τ), measures the intensity of internal forces that act on a plane parallel to the internal force [22], [23]. The magnitude of the stresses of a body depends on the magnitude and direction of the applied load, as well as the geometry of the solid. The state of stress at any point in a solid can be described by the nine components of stress (three components of normal stress and six components of shear stress), given by [23]:

$$[\sigma] = \begin{bmatrix} \sigma_{xx} & \tau_{xy} & \tau_{xz} \\ \tau_{yx} & \sigma_{yy} & \tau_{yz} \\ \tau_{zx} & \tau_{zy} & \sigma_{zz} \end{bmatrix} \quad (\text{Eq 5})$$

To maintain equilibrium, adjacent shear stresses must be equal ($\tau_{xz} = \tau_{zx}$, $\tau_{yz} = \tau_{zy}$, $\tau_{xy} = \tau_{yx}$) [22], [23]. Given this, the matrix describing the state of stress at a point in a solid, $[\sigma]$, can be reduced to only include six independent components of stress as follows:

$$[\sigma] = \begin{bmatrix} \sigma_{xx} & \tau_{xy} & \tau_{xz} \\ \tau_{xy} & \sigma_{yy} & \tau_{yz} \\ \tau_{xz} & \tau_{yz} & \sigma_{zz} \end{bmatrix} \quad (\text{Eq 6})$$

Hooke's law for a linear, homogeneous, isotropic material provides the relationship between the stress and strain at a point in a solid body [23]. Just as with stress, there are two types of strain that can exist in a solid. Normal strain, denoted by the Greek letter epsilon (ϵ), is a dimensionless quantity that provides a measure of the elongation or contraction of a solid due to deformation [22], [23]. Shear strain, denoted by the Greek letter gamma (γ), measures the angular distortion of a solid [22], [23]. The relationship between stress and strain is presented in Equation 7, and depends on the material properties of the body, which include the material's modulus of elasticity (E), Poisson's ratio (ν), and the shear modulus (G).

$$\begin{Bmatrix} \varepsilon_{xx} \\ \varepsilon_{yy} \\ \varepsilon_{zz} \\ \gamma_{xz} \\ \gamma_{yz} \\ \gamma_{xy} \end{Bmatrix} = \begin{bmatrix} \frac{1}{E} & -\frac{\nu}{E} & -\frac{\nu}{E} & 0 & 0 & 0 \\ -\frac{\nu}{E} & \frac{1}{E} & -\frac{\nu}{E} & 0 & 0 & 0 \\ \frac{1}{E} & -\frac{\nu}{E} & -\frac{\nu}{E} & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{1}{G} & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1}{G} & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{1}{G} \end{bmatrix} \begin{Bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{zz} \\ \tau_{xz} \\ \tau_{yz} \\ \tau_{xy} \end{Bmatrix} \quad (\text{Eq 7})$$

1.2.5 The Finite Element Method

The finite element method is a numerical approach that is used to approximate the solution to partial differential equations that can describe physical phenomena that could not otherwise be solved through analytical methods [24]. In the finite element method, a problem domain is discretized into many components known as finite elements, with the points at which these elements connect referred to as nodes [21]. The combined collection of these elements and nodes is often referred to as a mesh. Figure 10 shows the discretized two-dimensional mesh of a simplified connecting rod.

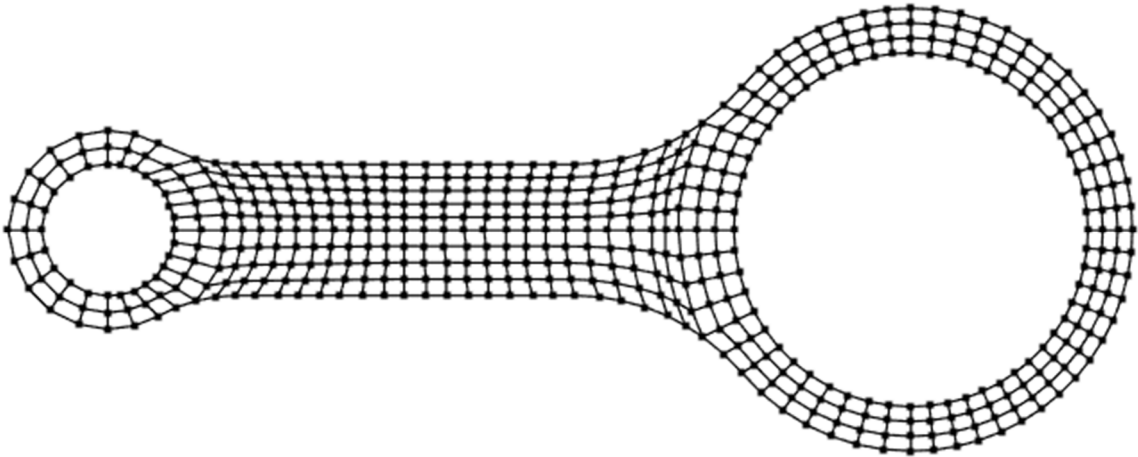


Figure 10: Discretized two-dimensional mesh of a simplified connecting rod.

The finite element method is used widely as it is capable of solving complex problems from various engineering disciplines, including stress analysis, thermal analysis, electromagnetic analysis, crash analysis, and many others [21], [24].

The discretization of the problem domain utilized in the finite element method creates many algebraic equations that are solved simultaneously. This system of algebraic equations in a structural mechanics problem is described by Equation 8, where $\{F\}$ is the nodal force vector, $[K]$ is the global stiffness matrix, and $\{U\}$ is the nodal displacement vector.

$$\{F\} = [K]\{U\} \quad (\text{Eq 8})$$

This global system of equations is formulated through the combination of element equations, nodal compatibility, and boundary conditions [21]. This global system of equations, once assembled, can be solved for the unknown nodal displacements and reaction forces at support locations [21], [24]. To solve for the nodal displacements, the global system of equations is partitioned to separate fixed and free degrees of freedom. This partitioning of the global system of equations can be expressed by Equation 9 [24].

$$\begin{Bmatrix} r_E \\ f_F \end{Bmatrix} = \begin{bmatrix} K_E & K_{EF} \\ K_{EF}^T & K_F \end{bmatrix} \begin{Bmatrix} d_E \\ d_F \end{Bmatrix} \quad (\text{Eq 9})$$

Where K_E relates the reaction forces on fixed nodes to the displacement of those nodes, K_F relates the forces on free nodes to the displacement of these nodes, and K_{EF} relates the reaction forces on fixed nodes to the displacements at free nodes. Using these various matrices and vectors, the displacements of free nodes (d_F) and reaction forces at fixed nodes (r_E) can be solved for using Equations 10 and 11, respectively [24].

$$d_F = K_F^{-1}(f_F - K_{EF}^T d_E) \quad (\text{Eq 10})$$

$$r_E = K_{EF} d_F + K_E d_E \quad (\text{Eq 11})$$

Once the nodal displacements have been determined, other internal forces, stresses, or deformations at any location can be evaluated [21].

Since the finite element method can only provide an approximate solution to the modeled problem, the user must have a method to assess the quality of the solution [24]. Figure 10 shows a flow chart that describes the process of finite element analysis.

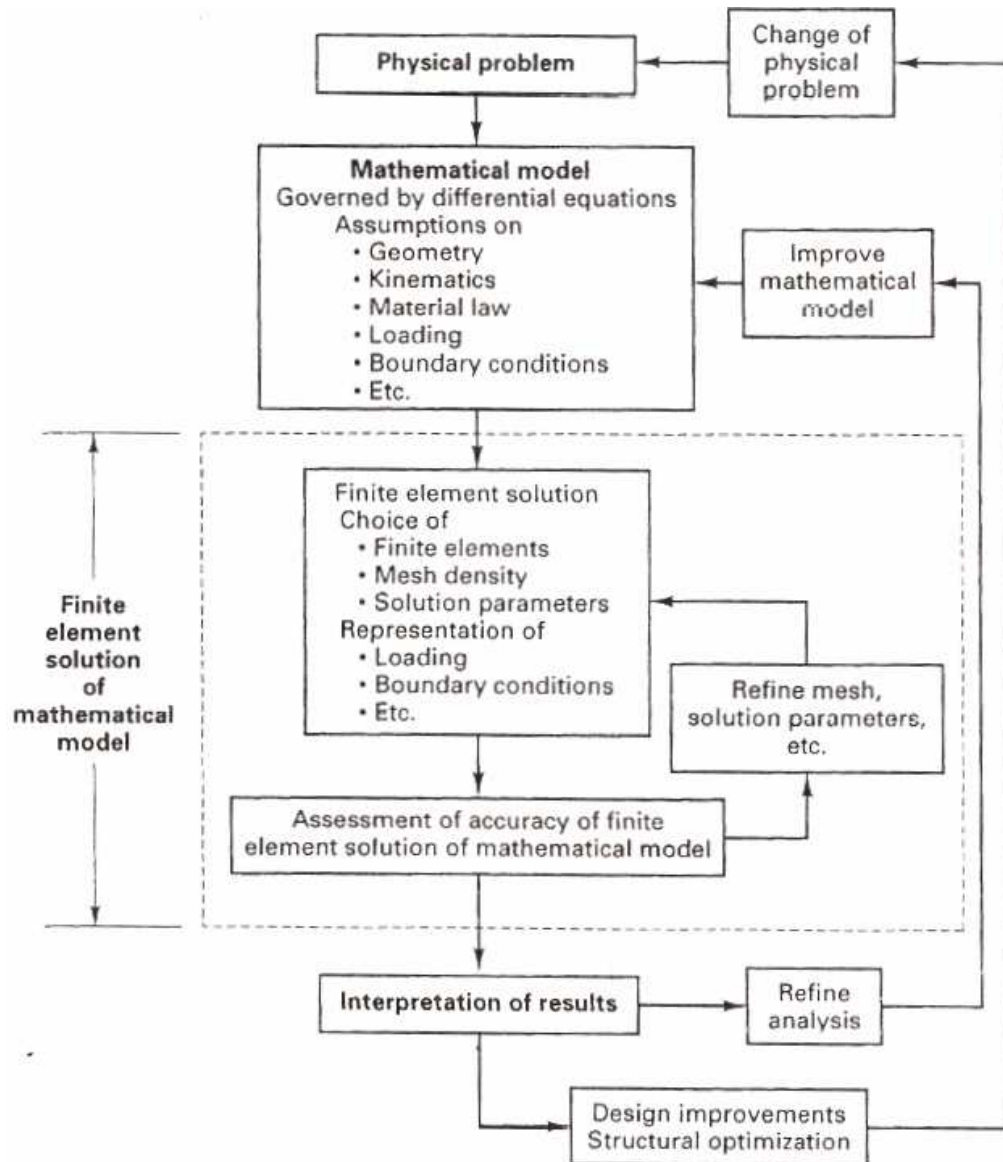


Figure 11: Flow chart describing the finite element process [25].

The discretization of the domain during the development of the mesh plays a large role in the solution accuracy. Given this, if the model's solution is found to be inaccurate, the solution must be repeated with refined parameters until sufficient accuracy is reached [25]. If model assumptions or simplifications are not believed to be the cause of inaccuracies, refinement of the mesh is usually performed to meet the accuracy requirements. In some cases, model accuracy can be evaluated by comparing results to experimental or analytical solutions. However, for cases in which an exact solution is unknown, refinement of the mesh and monitoring of how a result of interest varies with refinement can serve as a measure of accuracy. If, after refinement, the change in the result of interest is large, then the original mesh is inadequate, and further refinement may be necessary [24]. This iterative refinement procedure is repeated until the change in the result of interest meets a defined tolerance. While increasing the number of meshed elements improves the model accuracy, the higher element count also increases the computational requirements of the model.

1.3 Thesis Overview

The goal of this research is to evaluate the measurement of dynamic strain on the main bearing saddles of a large integral engine-compressor for monitoring the condition of the main bearings during operation. This evaluation consists of two main parts, including the evaluation of expected main bearing loads and the determination of optimal main bearing monitoring locations, which utilize an Ingersoll Rand KVSR 412 as the basis of each part of the evaluation. The optimal location to monitor strain along the circumference of each main bearing must be determined. The asymmetric cylinder arrangement utilized on integral engine-compressors leads to more complex bearing loading conditions than in separable engines and compressors, which are often symmetric. Given this, the main bearings of the KVSR 412 do not all carry the same load, as some of the main bearings support more of the load from the compressor cylinders than others. This asymmetric

loading means that the locations for measuring strain data must be evaluated separately for each main bearing. The recommended strain measurement locations will be evaluated based on the time-dependent loads at each bearing throughout a complete working cycle of the KVSR 412. The time-dependent loads at each main bearing will be found through a transient analysis of a finite element model of the KVSR 412.

1.4 Approach

ANSYS Mechanical will be used to model the moving assembly of the KVSR 412. The modeled integral engine-compressor will be created based on information gathered from a KVSR 412 operating at a compressor station in Northeastern Utah. The components that were modeled as part of the moving assembly include the crankshaft, flywheel, power cylinder connecting rods, and compressor cylinder connecting rods. Reciprocating components such as the pistons and crossheads were included in the model as lumped masses. The loads applied to the model due to the pressure developed in the cylinders were calculated based on measured pressure data collected from the power and compressor cylinders of the operating KVSR 412. The collected pressure data were sampled at each degree of crankshaft rotation. The frequency of the collected pressure data will allow the load on each bearing to be determined with the same angular resolution.

CHAPTER 2 – METHODS AND MATERIALS

2.1 Overview of Monitoring Equipment

2.1.1 Engine/Compressor Performance Data Measurement

A Windrock 6400 portable engine and compressor analyzer was used to collect the power and compressor cylinder pressure data, which were later utilized to calculate the applied gas loads that act on each connecting rod in the finite element model. The portable engine and compressor analyzer is utilized widely to measure fundamental performance and condition data on reciprocating engines and compressors. Using the Windrock 6400, various pressure sensors were selected to dynamically sample the pressure in each cylinder of the KVSr 412 from indicator ports, which are threaded into each cylinder. Figure 12 shows an example of the pressure transducers that were used to measure the cylinder pressure throughout the engine working cycle.



Figure 12: Example of a pressure transducer, which was used to measure the pressure in each cylinder of the operating KVSr 412.

Using an encoder that was installed on the engine flywheel during the collection, the pressure data from each cylinder can be resolved to the angular position of the crankshaft on a per-degree basis and referenced to the crankshaft position when the number one power cylinder piston is at top dead center.

2.1.2 Dynamic Strain Sensor

As part of the long-term scope of this research, dynamic strain sensors were installed at each main bearing during the overhaul of a KVSr 412. These sensors are to be utilized in the evaluation of dynamic strain, which is the oscillatory portion of the normal strain of the bearing saddle, for main bearing health monitoring during operational testing. The sensor outputs a scaled output value of microstrain ($\mu\epsilon$), which is equivalent to $1 \times 10^{-6} \epsilon$. The dynamic strain sensors, which were installed on the KVSr 412 main bearing saddles, are shown in Figure 13.



Figure 13: FDS 102B dynamic strain sensor, which was installed on the main bearings of a KVSr 412 [26].

Unlike strain gauges, which require the use of a balanced Wheatstone bridge and are susceptible to shifts in static strain due to the thermal expansion of the base material, the dynamic strain sensor is an Integrated Electronics Piezo-Electric (IEPE) transducer. IEPE sensors rely on the piezoelectric effect, which causes a piezoelectric element to create a voltage that is proportional to an applied stress on the element [27]. To reduce signal noise, an integrated amplifier is utilized in the sensor to increase the voltage developed by the piezoelectric sensing element. Unlike the IEPE accelerometer, which uses a seismic sensor mass to load the piezoelectric element in either compression or shear, in the dynamic strain sensor, the piezoelectric element is strained directly

so that the voltage developed by the element is proportional to the applied normal strain. Table 1 provides an overview of the dynamic strain sensor's technical specifications.

Table 1: FDS 102B dynamic strain sensor technical specifications [26].

Parameter	Value
Sensitivity (mV/ $\mu\epsilon$)	67
Measurement Range ($\mu\epsilon$)	+/-100
Frequency Range (Hz)	0.5 - 10,000

2.2 Finite Element Modeling

2.2.1 Software Overview

To determine expected main bearing loads, the moving components of the KVSR 412 were modeled in ANSYS Mechanical. ANSYS Mechanical is one of many finite element modeling products offered in the ANSYS software suite, which can be used to solve structural analysis problems. In many structural analysis problems, the loads applied to the modeled system are independent of time. When the loading conditions do not change over time, the system can be modeled using static structural analysis. A static structural analysis in ANSYS Mechanical allows for the determination of displacements, stresses, strains, and forces in the modeled system due to the applied loads, which do not induce significant inertia or damping effects. [28].

However, the finite element model of the KVSR 412 could not be analyzed using a static structural analysis as the external loads from the gas loading on the cylinders varied with time. Additionally, the movement of the geometry and the inertial effects of the reciprocating masses could not be modeled using a static structural analysis, as in this analysis, steady loading and response conditions are assumed [28]. For this reason, the finite element model of the KVSR 412

had to be analyzed using a transient structural analysis. The transient structural analysis in ANSYS Mechanical allows for the determination of the time-varying displacements, strains, stresses, and forces in a structure as it responds to any transient load [28]. The use of a transient structural analysis also includes the inertia and damping effects of a system, which would have been ignored in a static structural analysis.

The transient structural analysis in ANSYS Mechanical uses the Newmark time-integration method to solve for the dynamic response of the modeled system due to any general time-dependent loads at discrete time points [28]. This methodology uses the basic equation of motion shown in Equation 12, which can be used to solve for the system's displacement $\{u\}$ due to an array of time-dependent loads $\{F(t)\}$, given the system's mass $[M]$, damping $[C]$, and stiffness $[K]$ matrices [28].

$$\{F(t)\} = [M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} \quad (\text{Eq. 12})$$

This system of equations is used to solve for the static equilibrium at each time step. To evaluate the expected main bearing loads of the KVSr 412 based on the measured pressure data from the operational integral engine-compressor, the time step for the analysis was defined so that each time step corresponds to one degree of crankshaft rotation. Given the rotational speed of the KVSr 412 is 330 revolutions per minute, the time integration step size for the analysis was set at 5.05×10^{-4} seconds.

2.2.2 Geometry

To create the finite element model, the simplified solid geometry of the crankshaft, flywheel, and the power cylinder and compressor cylinder connecting rods were created in SolidWorks. Engineering drawings or solid geometry were not available for any of these components. Given the limited information available, the dimensions of the crankshaft throws and webs were measured in the field on a crankshaft that had been removed from a KVSr 412 during a scheduled

overhaul. The diameters and lengths of the crankshaft's bearing journals were modeled based on the reconditioning report for a replacement crankshaft for the KVSR 412 being overhauled. General information, such as the stroke of the power cylinders and compressor cylinders, and the angle between the crankshaft throws were available in the manual provided to the engine operator from Ingersoll Rand when the KVSR 412 was first commissioned. This information was used to define the other dimensions that were required to model the crankshaft. The geometry of the flywheel was simplified so that the rotational inertia and mass of the flywheel could be included in the finite element model. The diameter and thickness of the modeled flywheel were measured from the flywheel of a KVSR 412 while on-site.

The power cylinder and compressor cylinder connecting rods were not available for field measurements. Instead, general geometry for each of these components had to be created to allow them to be included in the finite element model. The only dimensions that were available were the connecting rod length, which is the distance between the centers of the big-end and small-end bearings, and the big-end and small-end bearing diameters. As the connecting rods had to be included in the model to transfer the loads that act at the small-end bearing of each connecting rod to the crankshaft, the available dimensions were used to create simplified connecting rod geometry.

The power cylinder pistons, compressor cylinder pistons, piston rods, and crossheads were not modeled with solid geometry. Instead, the inertia of each of these components was modeled using lumped masses located at the center of the small-end bearing of each connecting rod. The mass of each of the components was found in past rebuild reports for the KVSR 412, from which the cylinder pressure data were measured. Given that the compressor cylinder piston, piston rod, and the cross head all reciprocate, their masses were combined into a single lumped mass at the small-end bearing of each compressor cylinder connecting rod. The lumped mass modeled at each small-

end bearing of the power cylinder connecting rods was 420 lbm. This mass was based on documentation that was available for the power cylinder piston, but does not include the mass of the piston rings, wrist pin, or the small-end bearing [29]. The lumped mass modeled at the small-end bearing of each compressor cylinder connecting rod was 1,540 lbm. This total mass was based on the measured weights of the piston, piston rod, and crosshead of each cylinder and was provided by the equipment operator for the purposes of the modeling. It is not clear if these weights included the weights of other components, such as the piston rings, rider bands, and jam nuts.

Figure 14 shows the geometry for each component of the KVSr 412, which was included in the finite element model.

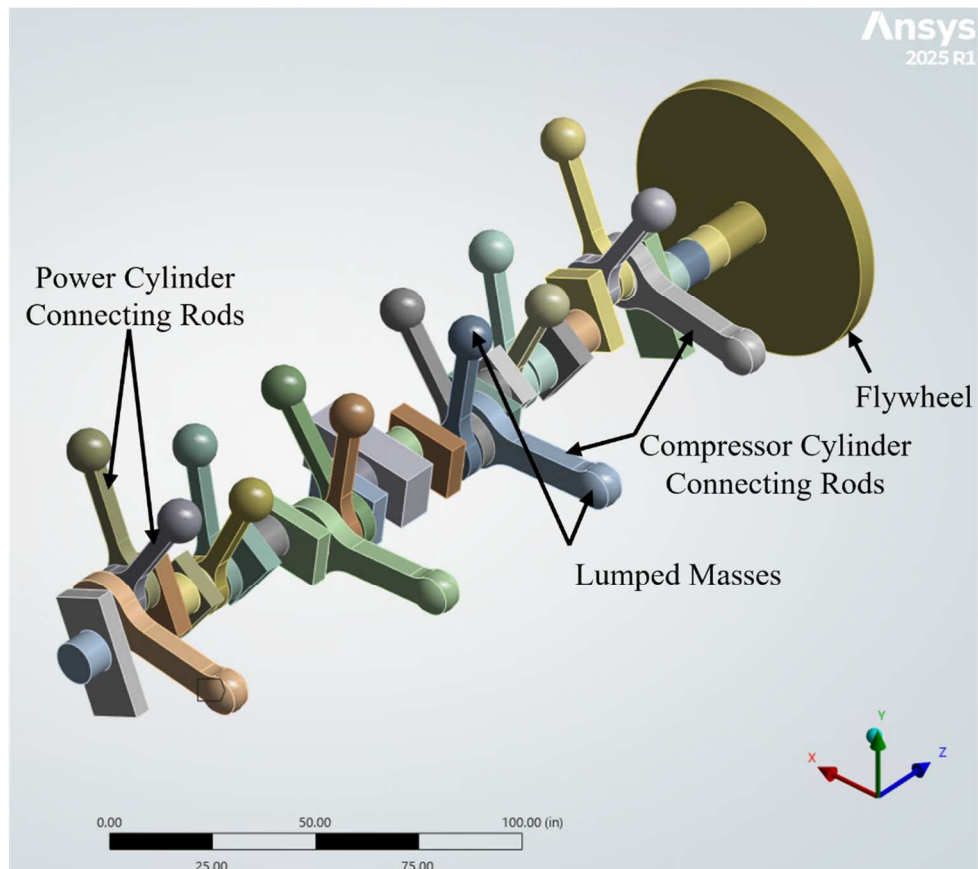


Figure 14: KVSr 412 component geometry imported into ANSYS Mechanical.

2.2.3 Meshing

Once imported into ANSYS Mechanical, the solid geometry was meshed. The number of elements in a finite element model has a significant effect on the time required to compute results. Additionally, the computation of a transient analysis is significant as the system of equations describing the modeled system must be solved at each time step. For this reason, it was important to limit the finite element model's element count to reduce computation time while still providing a fine enough mesh to ensure the results of the model were accurate. In ANSYS Mechanical, the stiffness behavior of the modeled geometry can be set as rigid. Setting the stiffness behavior of a body to rigid does not allow for deformation of the body, but represents the body as a single mass element, including the body's mass and moment of inertia [28]. The ability to represent certain components in an assembly as rigid bodies greatly reduces the required elements in the finite element model, which leads to a reduction in computational needs. To reduce the number of elements required to model the KVSr 412, the flywheel, power cylinder connecting rods, and compressor cylinder connecting rods were modeled as rigid.

Given this, only the crankshaft was meshed for the purposes of this analysis. Since the main bearing reaction forces are the only outputs of interest from this model and not the crankshaft stresses, the crankshaft was originally modeled as rigid. However, it was found that modeling the crankshaft as a rigid body does not allow the separate reaction force at each main bearing to be determined. This meant only the total combined reaction force across all bearings could be evaluated if the crankshaft were a rigid body. To evaluate the reaction force at each main bearing journal separately, the crankshaft had to be modeled as a flexible body and was meshed. Figure 15 shows the geometry with the mesh visible to differentiate the geometries that were modeled as rigid bodies.

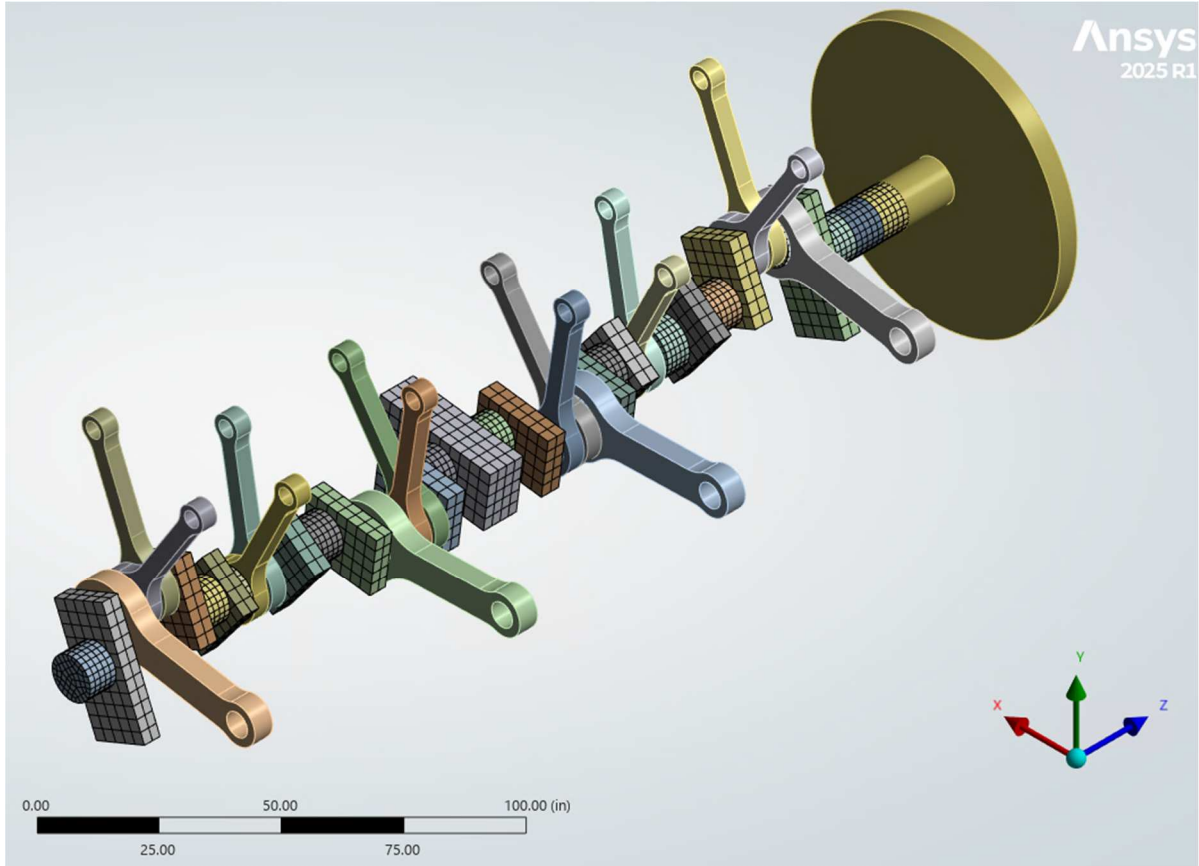


Figure 15: Modeled geometry of the KVSR 412 with the crankshaft mesh visible.

Though the crankshaft geometry is made up of simple geometric shapes, meshing the crankshaft as a single solid geometry would be difficult. To simplify the meshing of the crankshaft, the crankshaft was split at each face, which connected different geometries. This allowed every single component of the crankshaft geometry (e.g. main bearing journal, web, counterweight, crankpin journal) to be meshed separately. The various meshed components of the crankshaft were then joined at common faces using a bonded surface contact. Figure 16 shows the crankshaft mesh with the rigid connecting rods hidden.

The crankshaft geometry was meshed with 8,237 quadratic solid hexahedron elements with a total of 46,334 nodes. No details were available on the specific material properties of the

crankshaft. However, crankshafts in engines of various sizes are commonly forged from carbon steel [30]. Given this, the material properties used in meshing the crankshaft are found in Table 2.

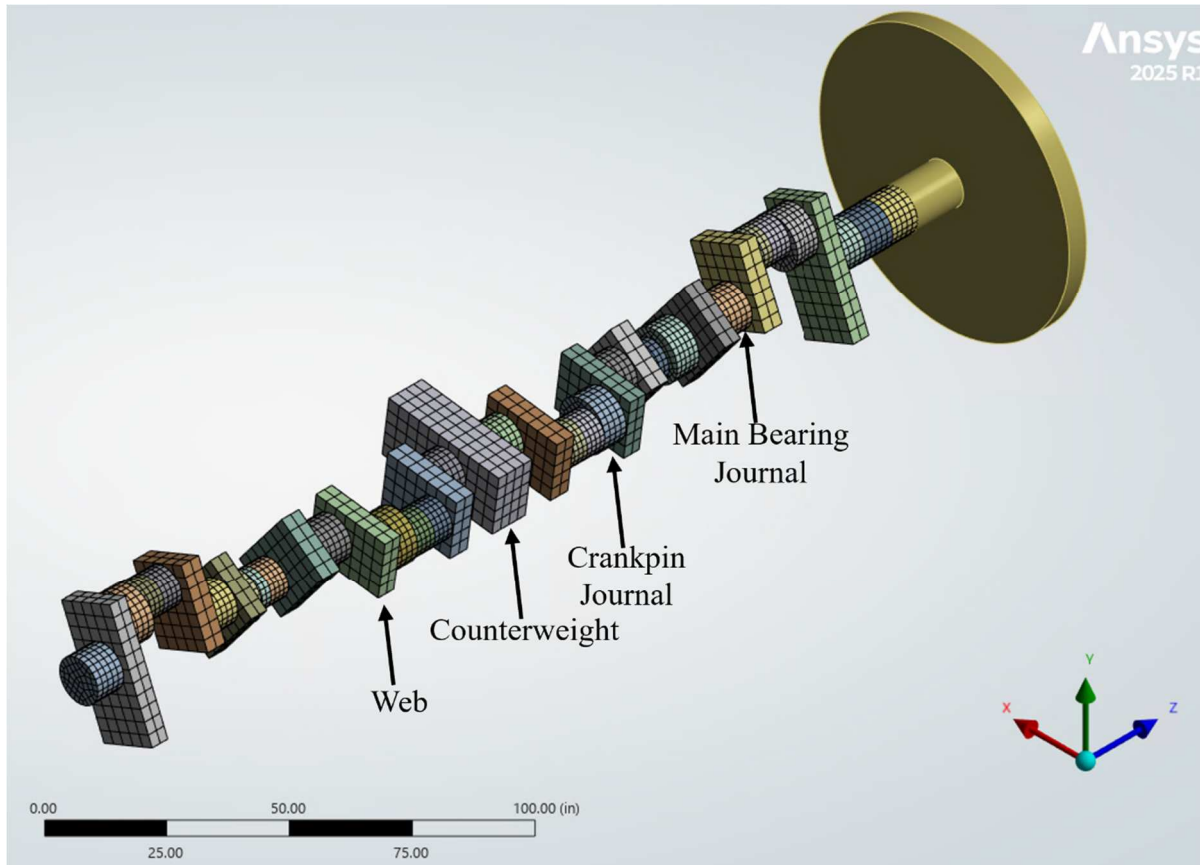


Figure 16: KVSr 412 crankshaft mesh with the rigid connecting rod geometry hidden.

Table 2: Material properties for forged carbon steel assigned to the crankshaft [30], [31].

Material Property	Value
Density (lbm/in ³)	0.283
Poisson's Ratio	0.30
Modulus of Elasticity (ksi)	30,458

Since the main bearing reaction forces are the main interest of the analysis, the coarseness of the mesh should not affect the model results since the constraint reactions are solved to satisfy the global equilibrium of the model. If the crankshaft stresses were of interest, the mesh would need

to be refined further to ensure the results were accurate. To show that mesh refinement does not affect the reaction force at the main bearing journals, a mesh refinement study was performed. This mesh refinement study was performed on the small crankshaft model with a single throw that is shown in Figure 17.

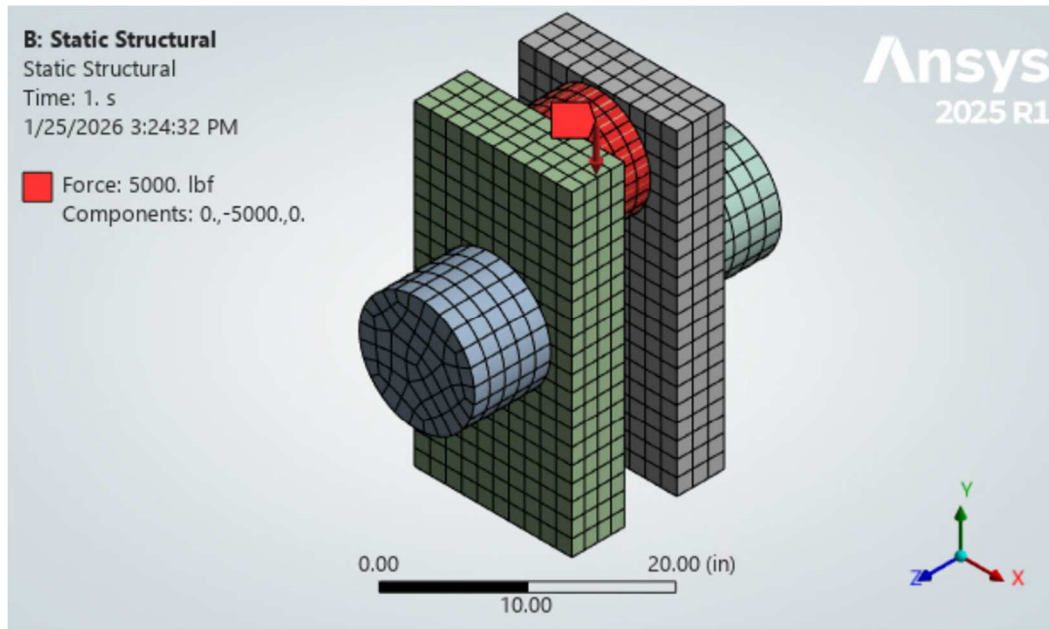


Figure 17: Small single-throw crankshaft model used for mesh refinement analysis.

This small crankshaft model was analyzed using a static structural analysis in ANSYS Mechanical. A load of 5,000 lbf was applied to the crankpin journal in the negative Y-direction. A static analysis of the small crankshaft model was selected to reduce the computation time for the mesh refinement study. The element size of the meshed solid geometry was varied from 3 inches to 0.1875 inches using the parameter set function in ANSYS Mechanical, with the element size halved at each new design point compared to the previous design point. At each design point, the total combined reaction force at the main bearing journals was evaluated based on each tested element size. Figure 18 shows the results of the mesh refinement study performed on the small crankshaft model.

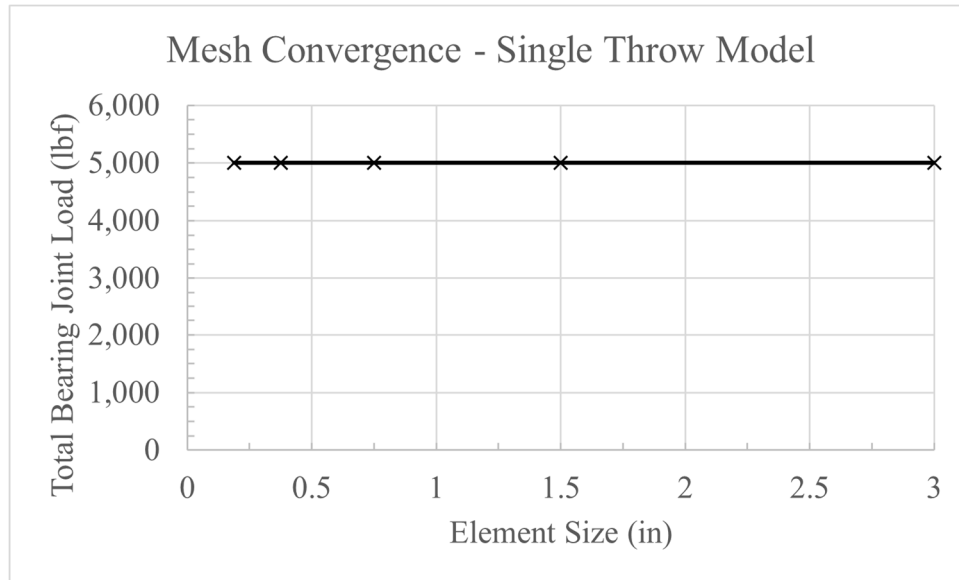


Figure 18: Results from the mesh refinement study comparing total constraint force at main bearing journals to the configured element size.

Based on the mesh refinement study, it was found that the element size does not have a direct effect on the constraint reaction at the main bearing journals. Given this, the mesh of the KVSr 412 crankshaft was left relatively coarse to reduce the computation time for the transient analysis.

2.2.4 Constraints and Boundary Conditions

Various joints were used to constrain the assembly of the crankshaft, power cylinder connecting rods, and compressor cylinder connecting rods while allowing for translation or rotation of the appropriate degrees of freedom. Figure 19 shows the main bearing journal locations. A cylindrical joint was modeled between each main bearing journal and ground, constraining the translational and rotational degrees of freedom in the joints' X- and Y-axes. The joints applied at the main bearing journal location allowed the crankshaft to rotate.

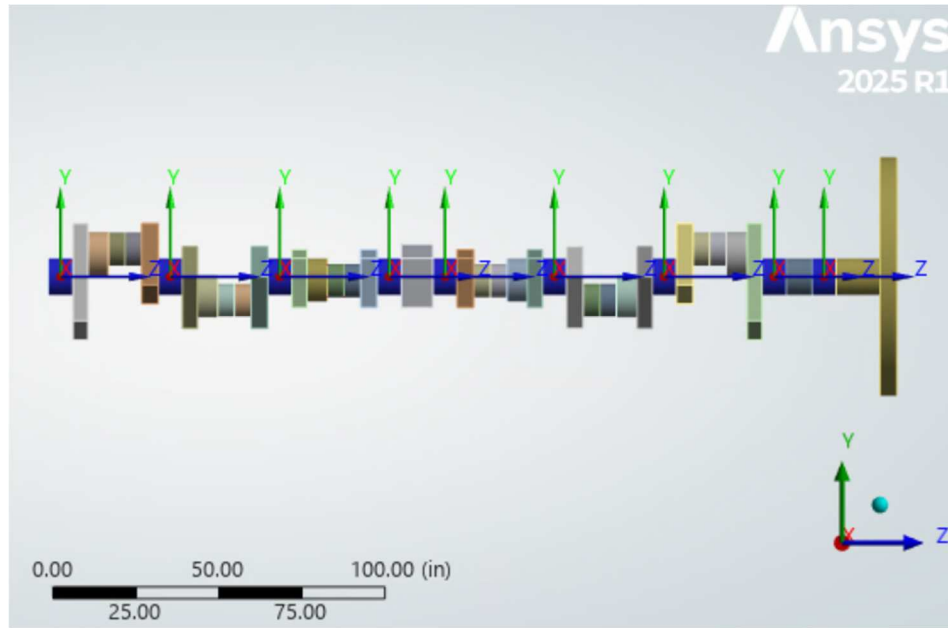


Figure 19: Main bearing journal locations where cylindrical joints were applied.

Revolute joints were used to constrain the big-end bearings of the power cylinder and compressor cylinder connecting rods to allow the connecting rods to rotate about the crankpin journals. Figure 20 shows the joint that constrains the connecting rod for the #1 compressor cylinder.

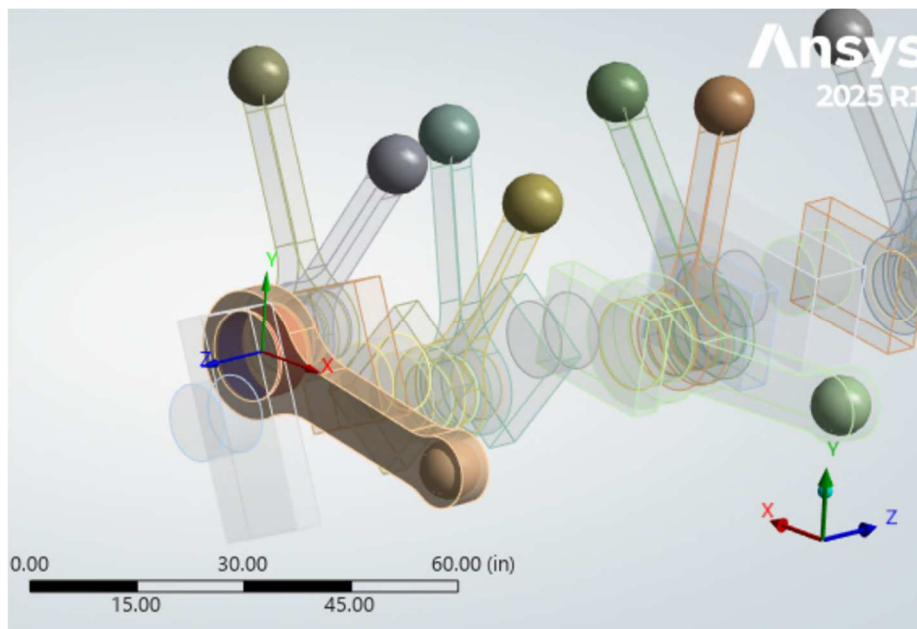


Figure 20: Revolute joint which constrains the connecting rod of the #1 compressor cylinder to the crankshaft.

The revolute joints only allow for the rotation of the connecting rods about the joints' Z-axis, while the translation in all three axes and the rotation about the joints' X- and Y-axes are constrained. General joints were applied at the small-end bearing of each connecting rod to constrain the degrees of freedom that would typically be constrained by the piston. Figure 21 shows the joint coordinate systems at the small-end bearings of the right-bank power cylinder connecting rods, the left-bank power cylinder connecting rods, and the compressor cylinder connecting rods.

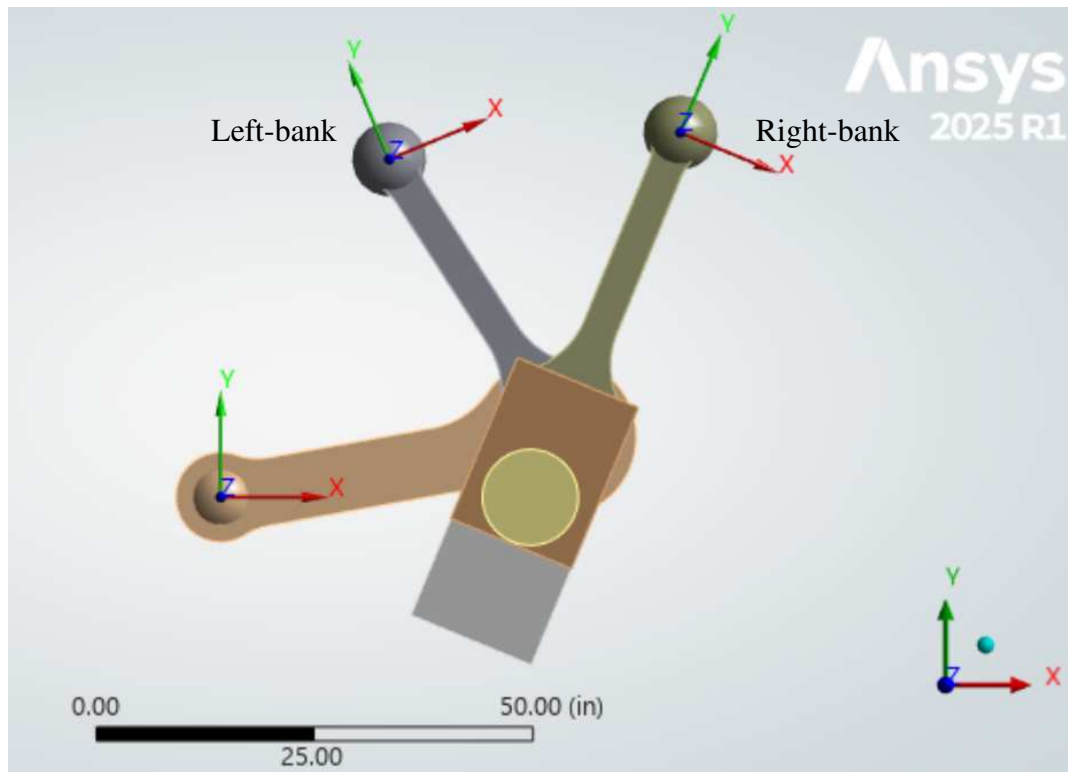


Figure 21: Joint coordinate system at the small-end bearing of each connecting rod.

Based on the displayed joint coordinate systems, the translation of small-end bearings of all power cylinder connecting rods was constrained in the X- and Z-axes, while the rotation at these bearings was constrained about the X- and Y-axes. The general joint applied at the small-end bearing of the compressor cylinder connecting rods only allowed the small-end bearings to translate along the X-axis and rotate about the Z-axis. All joints applied at the small-end bearings were relative to the joints' reference coordinate system. The inclusion of the cylindrical joints at

the main bearings, the revolute joints at the big-end connecting rod bearings, and the general joints at the small-end connecting rod bearings allowed the components in the finite element model to dynamically move like the components in a real KVSR 412. The masses and dimensions of various components are provided in Appendix A.

2.2.5 Calculated Gas Loads

The provided cylinder pressure data, measured at each degree of crank angle rotation, was used to calculate the appropriate gas load that acts on each power cylinder and compressor cylinder piston. Figure 22 shows the crank angle resolved pressure cascade measured from the twelve power cylinders from the sample KVSR 412. The displayed pressure data was averaged over 9 working cycles of the KVSR 412.

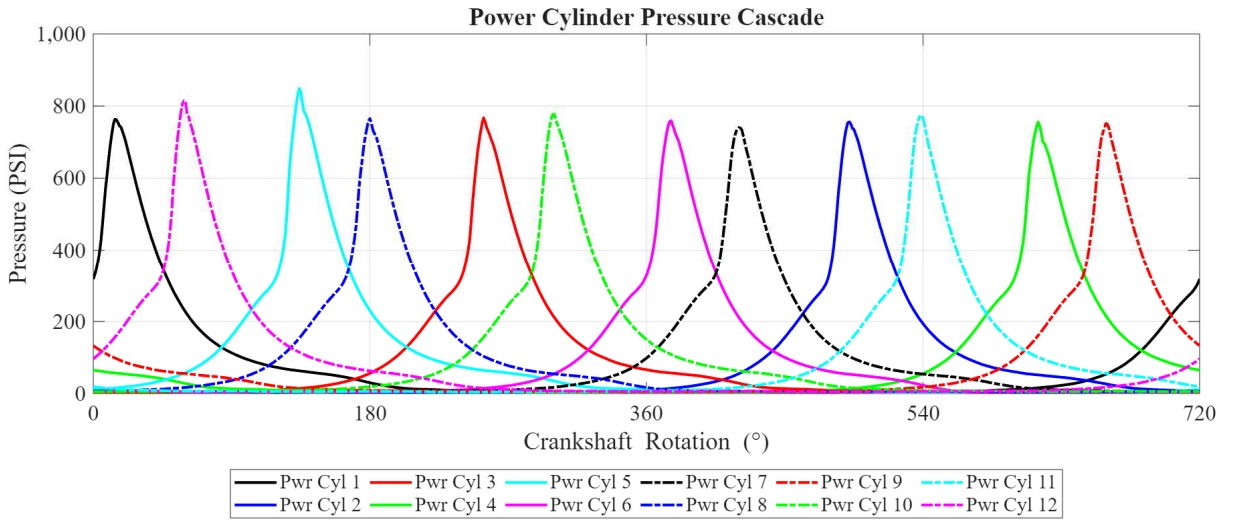


Figure 22: Power cylinder pressure cascade measured across a complete working cycle of a KVSR 412.

Given the pressure measured from each power cylinder (P_{Pwr}) and the bore of the power cylinders (D_{Pwr}), the load that acts on each power cylinder piston was calculated at each angle of crankshaft rotation in the engine operating cycle using Equation 13.

$$F_{Pwr} = P_{Pwr} \left(\frac{\pi}{4} D_{Pwr}^2 \right) \quad (\text{Eq 13})$$

Figure 23 shows the crank angle resolved pressure cascade measured from the head-end and crank-end of the four compressor cylinders from the KVSr 412 operating in the field.

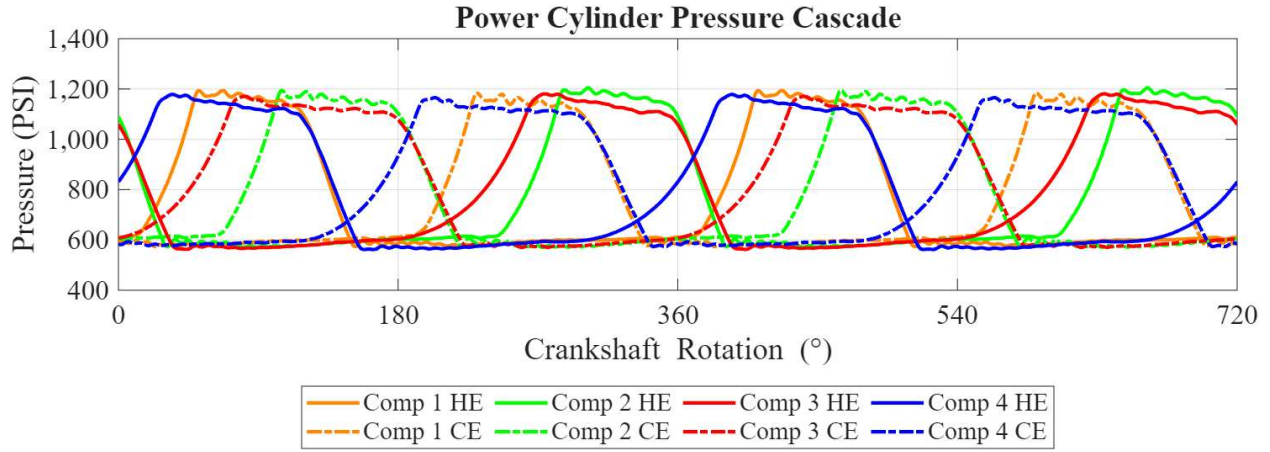


Figure 23: Compressor cylinder pressure cascade measured across a complete working cycle of a KVSr 412.

In addition to calculating the gas load that acts on the power cylinder pistons throughout the KVSr 412 working cycle, the gas load that acts on the compressor cylinder pistons was also calculated using the pressure data measured on both ends of each compressor cylinder. Based on the pressure measured from the head-end (P_{HE}) and the crank-end (P_{CE}) of a given compressor cylinder, as well as the compressor cylinder bore (D_{Comp}), and the piston rod diameter (D_{Rod}), the total gas load which acts on each compressor piston was calculated using Equation 14.

$$F_{Comp} = \left[P_{HE} \left(\frac{\pi}{4} \right) (D_{Comp})^2 \right] - \left[P_{CE} \left(\frac{\pi}{4} \right) (D_{Comp} - D_{Rod})^2 \right] \quad (\text{Eq 14})$$

Using Equations 12 and 13, the total gas loads acting on the power and compressor cylinder pistons were calculated for each degree of crankshaft rotation at which pressure data were measured. Given that the measured pressure data were resolved to crank angle, an array of time steps for each angle of crankshaft rotation had to be calculated so that the gas load on each piston could be plotted against time. Given the crankshaft angular position (θ_{Crank}) expressed in degrees and the rotational

speed of the crankshaft (N_{Crank}) expressed in revolutions per minute, the time step of each degree of crankshaft rotation (0° to 720°) can be calculated using Equation 15.

$$t = \frac{\theta_{Crank}}{6N_{Crank}} \quad (\text{Eq. 15})$$

Figures 24 and 25 show the time-dependent gas loads that act on the power cylinder pistons and compressor cylinder pistons, respectively. A negative sign represents that the load is pointing toward the crankshaft.

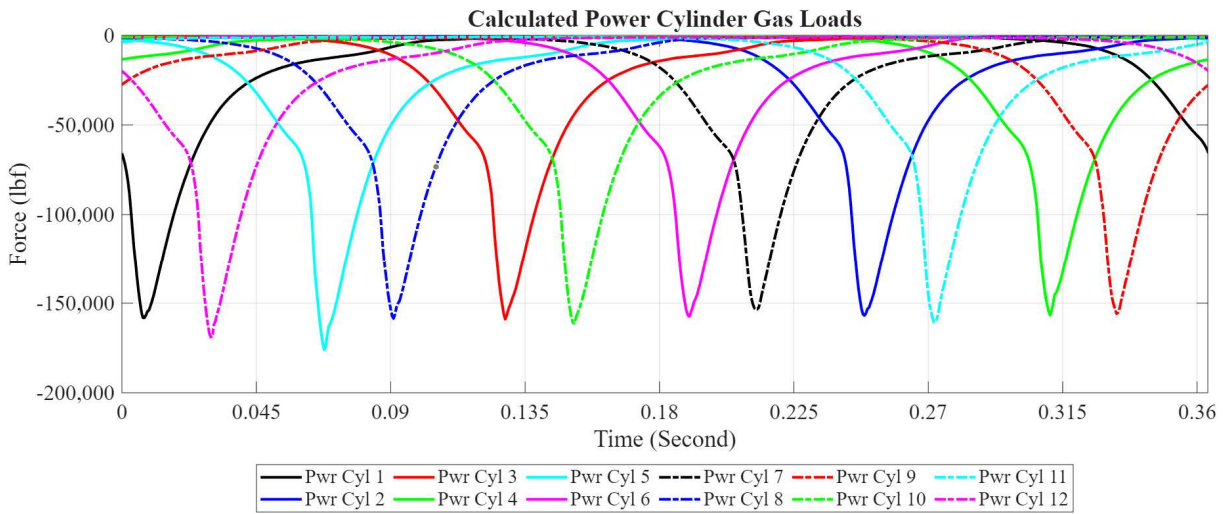


Figure 24: Calculated gas load which acts on each power cylinder piston over the KVSr 412 working cycle.

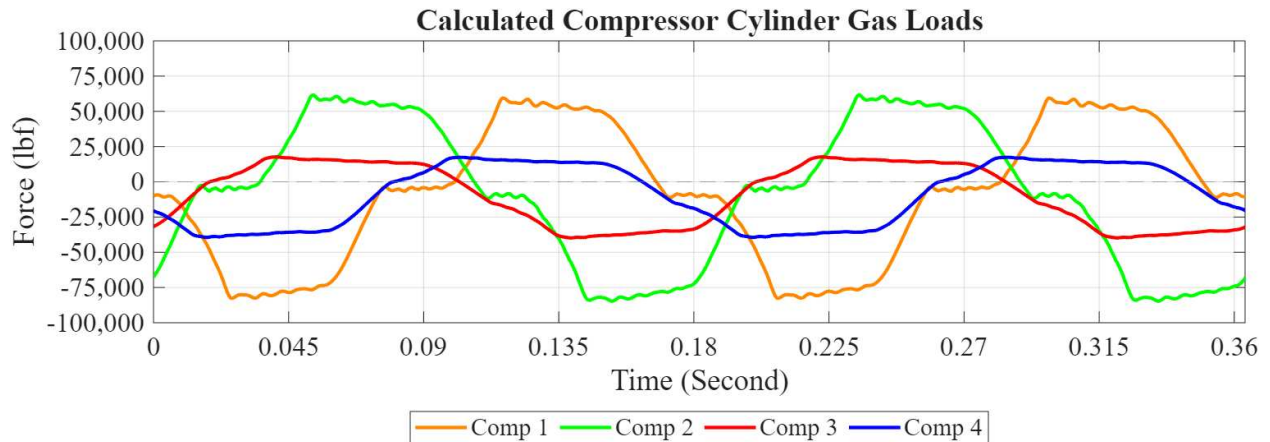


Figure 25: Calculated gas load which acts on each compressor cylinder piston over the KVSr 412 working cycle.

2.2.6 Application of Loads

The reaction force at each main bearing at any given time step is in response to various loads that occur in an integral engine-compressor. The force due to the pressurized gas contained within a cylinder acts on the face of the piston. This load due to the pressurized gas is referred to as the cylinder's gas load. Since only the mass of the pistons was included in the finite element model, the gas loads calculated from the pressure measured on each cylinder were instead applied to the small-end bearing of each connecting rod. The gas loads applied to each connecting rod were modeled using remote forces that act along the axis of each cylinder. Figure 26 shows the remote forces applied to the small-end bearings of the number one and seven power cylinder connecting rods and the number one compressor cylinder connecting rod.

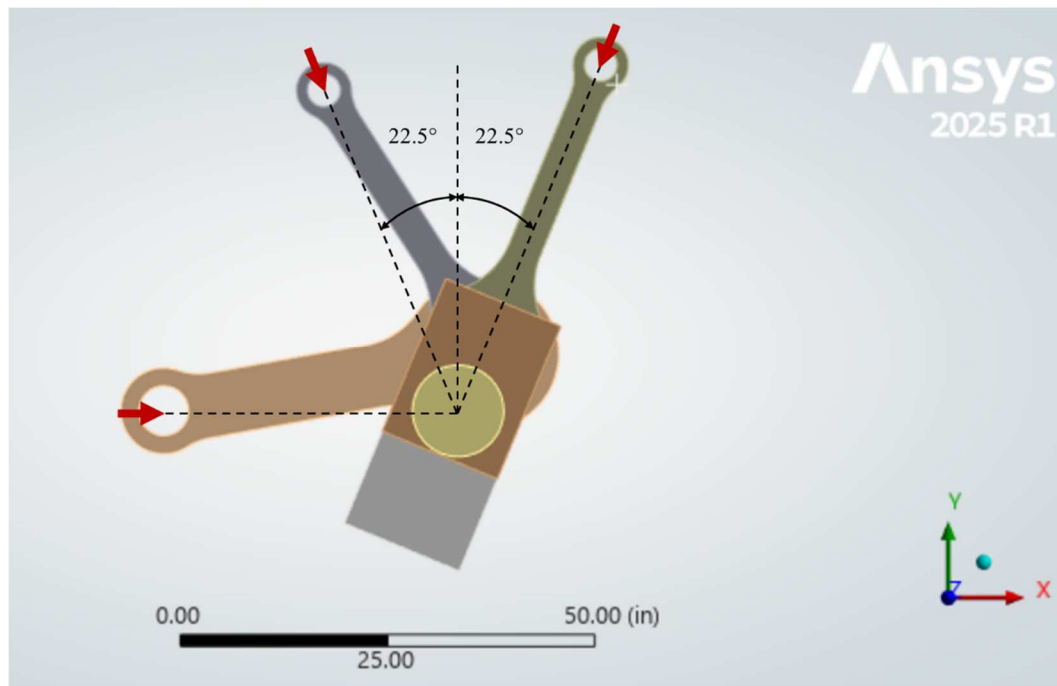


Figure 26: Visual representation of gas load vectors applied to three KVSr 412 connecting rods.

The weight of all components was included in the finite element model by applying gravitational acceleration to all geometry, which acted in the negative Y-direction. At any time step, the gas load on a given cylinder corresponds to a specific angle of the connecting rod relative

to the cylinder's axis. For this reason, the rotational position of the crankshaft had to be controlled to ensure the connecting rods were at the correct orientation to each cylinder's axis for each time step at which the gas loads were applied. Additionally, the rotational speed of the crankshaft had to be enforced so that the effect of the unbalance of the rotating inertia of the assembly, as well as the forces due to the inertia of the reciprocating components, could be included in the model. To control the rotational speed and position of the crankshaft, joint loads that enforced the rotational speed at each main bearing journal were applied to the model. Figure 27 shows the applied joint loads.

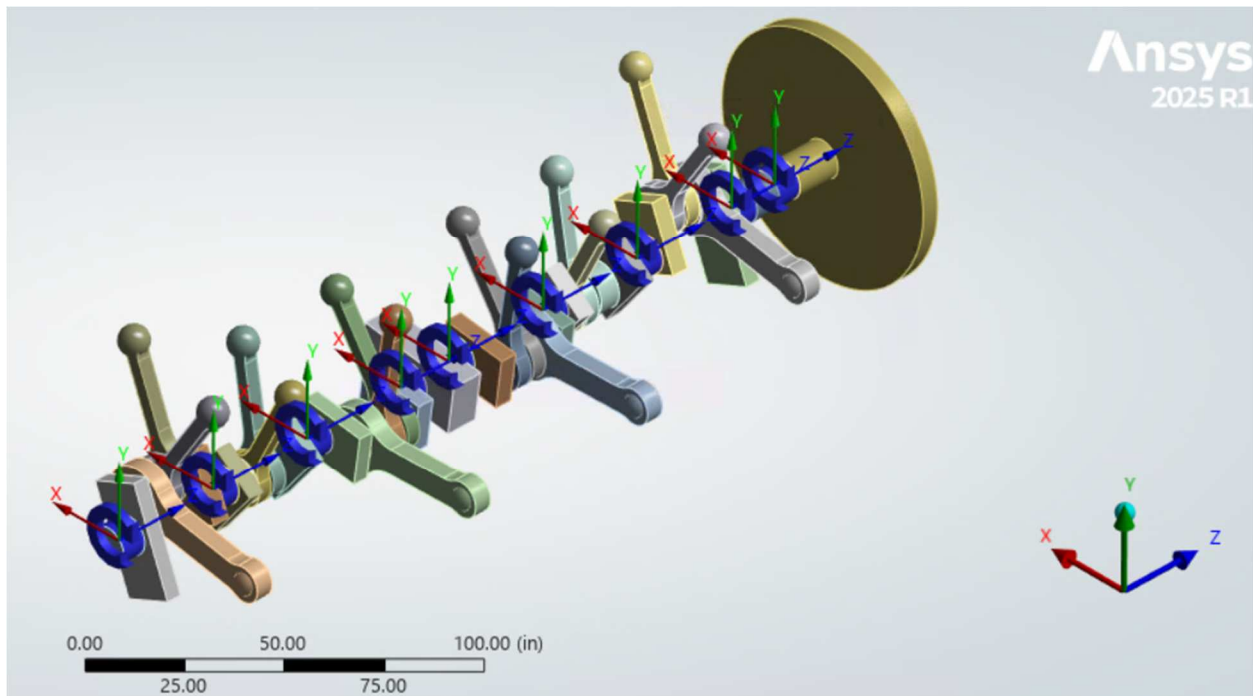


Figure 27: Joint loads applied to each main bearing journal to control the rotational velocity of the crankshaft.

To limit the sudden shock loading that would occur if the large cylinder gas loads were applied at their full magnitudes at the start of the modeled time span, joint loads that constrain the crankshaft's rotational speed were defined so that the crankshaft accelerated from rest to the full

operating speed of 330 RPM over the first revolution of the crankshaft. Figure 28 shows the resulting modeled crankshaft's relative rotation and rotational speed over the modeled time span.

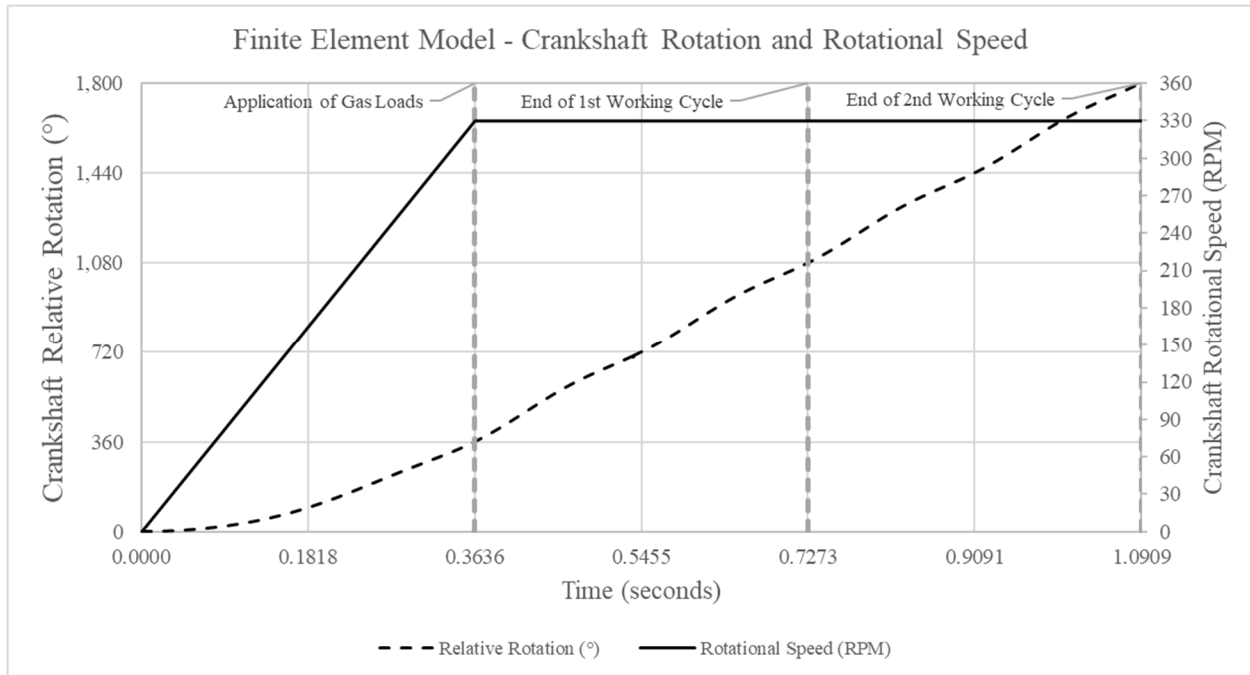


Figure 28: Relative rotation and rotational speed of the modeled KVSR 412 crankshaft over the total modeled time span.

Over the initial revolution of the crankshaft in which the crankshaft's rotational speed is ramped, the gas loads that act at the small-end bearing of each connecting rod were also ramped. Over the first revolution, the gas loads applied to each connecting rod began at a value of 0 and were increased at a fixed rate to the initial value of each cylinder's gas load at the start of the first working cycle based on the cylinder pressures measured on the operating KVSR 412. Two complete working cycles corresponding to 720° of crankshaft rotation were modeled so that the repeatability of the model would be verified.

2.3 Post Processing

2.3.1 Bearing Reactions Versus Bearing Loads

The results of interest from the finite element modeling of the KVSr 412 are the magnitude and direction of the load on each main bearing from the crankshaft. However, the load acting upon each main bearing from the crankshaft is not a direct result of the finite element model. Only the reaction force from each main bearing to the crankshaft journals is a direct output of the finite element model and is obtained by solving for the equilibrium of the nodes whose translational degrees of freedom are constrained by the cylindrical joints applied at the main bearing journals. From the finite element model, the reaction force for each main bearing journal can be exported over the modeled time span in terms of the reaction force components at each time step in the X- and Y-directions. Figure 29 shows an example of the reaction force components provided by the finite element model for a single cylinder engine with a cylinder axis oriented at an angle of 22.5° from the vertical.

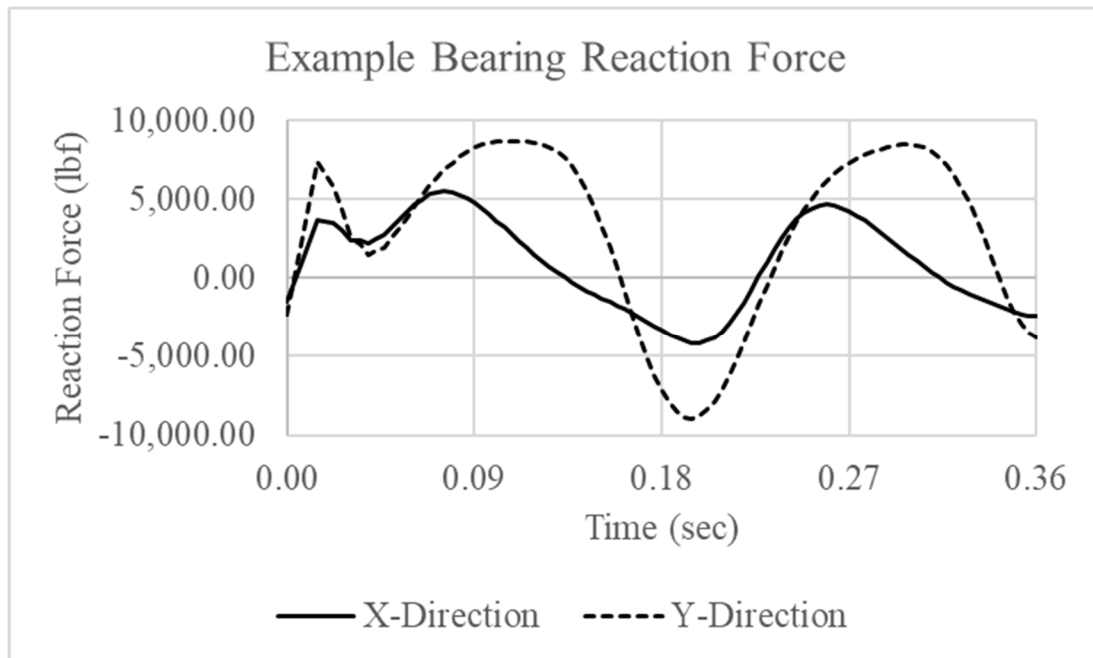


Figure 29: Example bearing reaction force result from the finite element model over a single working cycle.

Figure 30 shows the simplified single cylinder model from which the example bearing reaction force components were determined.

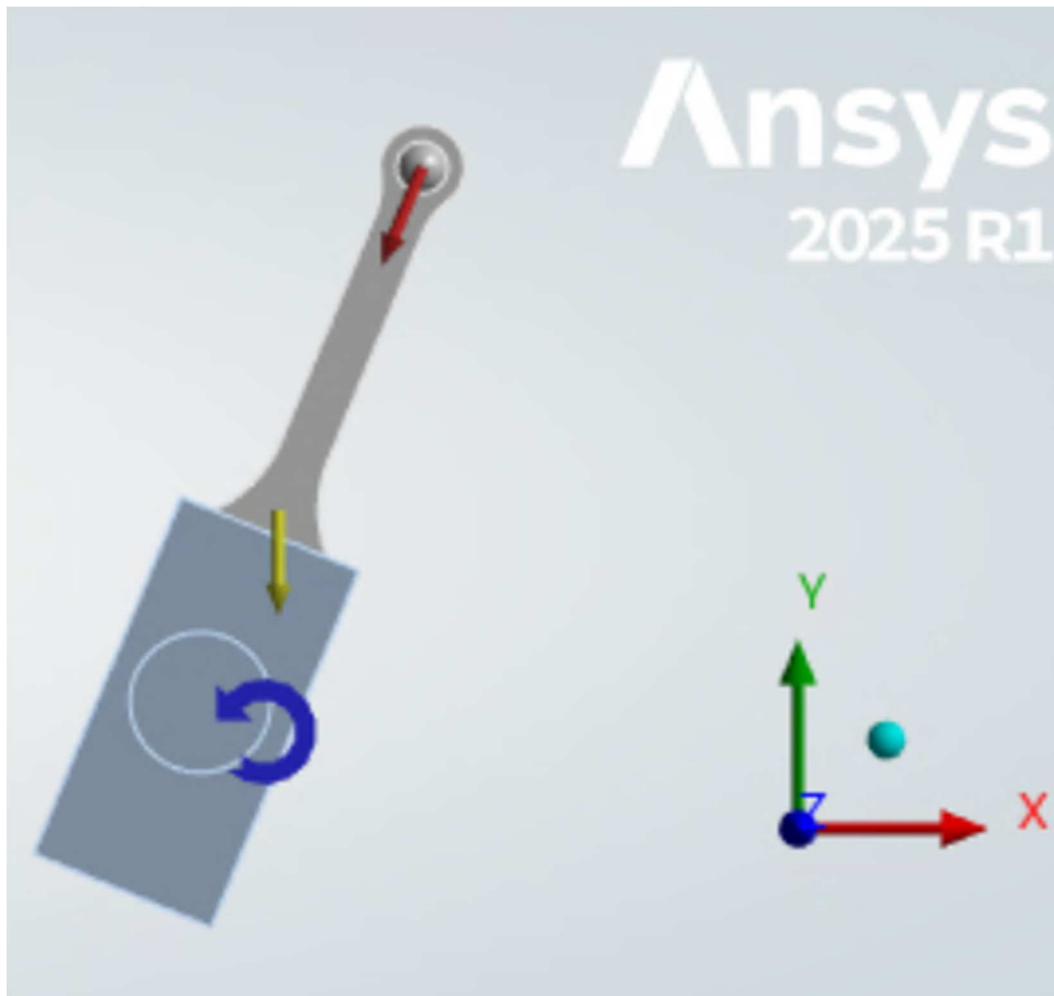


Figure 30: Single-cylinder power cylinder model used to determine the example bearing reaction force data.

To determine the load applied to each main bearing throughout the modeled time span, the components of the main bearing reaction force can be inverted. Since the modeled time step was defined as the time between crankshaft rotation degrees, the inverted reaction force provides the load on each main bearing at each degree of crankshaft rotation throughout the KVS R 412's working cycle. Figure 31 represents the relationship between the reaction force from the finite

element model and the resulting main bearing load. Additionally, this figure provides the relationship between the cylinder orientation and the output XY coordinate system.

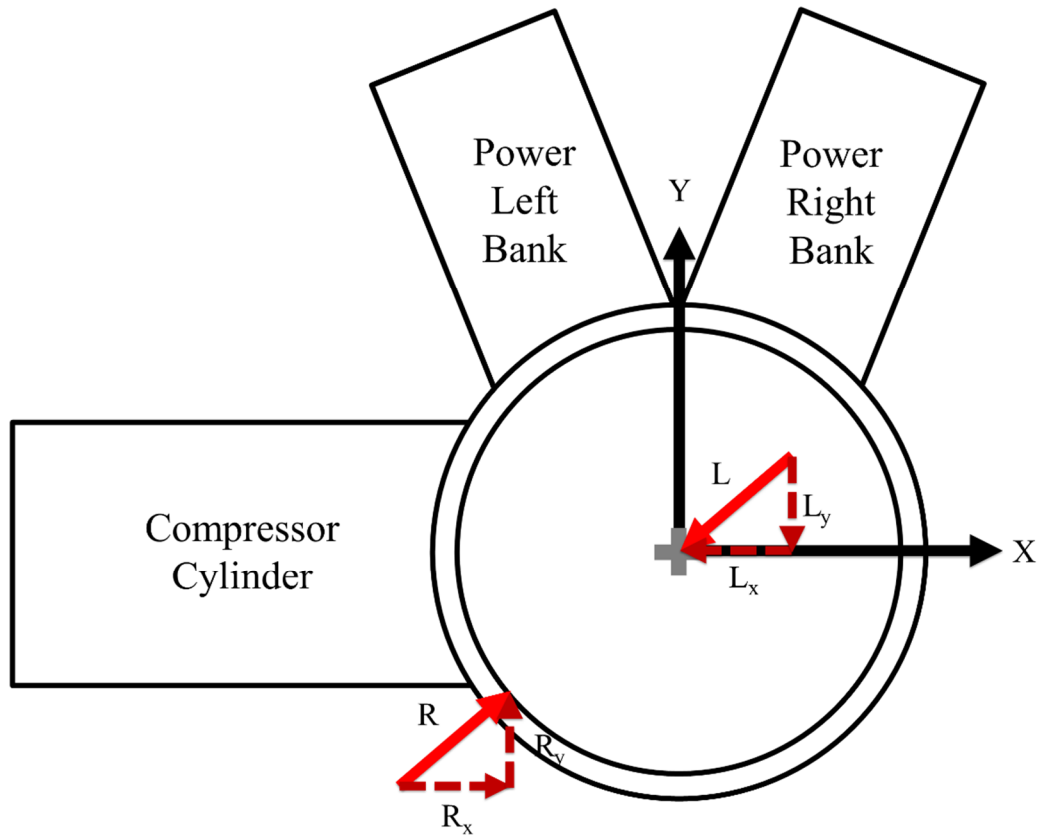


Figure 31: Inversion of bearing reaction forces from the finite element model to calculate loads on main bearings.

2.3.2 Estimation of Model Unbalance

Given that the geometry of the connecting rod was not known, and simplified geometry was created to be used in the finite element model, steps had to be taken to ensure the mass of the simplified connecting rod geometry did not cause a significant rotating unbalance. An unbalance of a rotating assembly occurs when the mass of a rotating body is unevenly distributed about the axis of rotation. A rotating unbalance will cause a radial force that rotates with the unbalanced mass at the speed of the rotating assembly. This radial force will lead to high bearing loads and elevated mechanical vibration. Since the results of interest from the finite element model were the

main bearing reaction forces throughout the modeled time span, a significant rotating unbalance due to incorrect connecting rod geometry could be a significant cause of error in the main bearing reactions. To quantify whether a significant rotating unbalance of the modeled assembly existed, the transient structural analysis of the finite element model was conducted with the remote forces used to model the cylinder gas loads suppressed. By suppressing the remote forces that model the cylinder gas loads, only the inertial loads due to the unbalance of the rotating assembly and the loads due to the reciprocating components affect the reaction forces.

From the finite element model, the X- and Y-components of the reaction forces were extracted over the modeled time span with the gas loads suppressed. To determine the total reaction force on the crankshaft, the components of the reaction forces at each bearing were summed in their respective plane at each time step. The summed reaction force components in the X- and Y-directions were used to calculate the magnitude of the combined bearing reaction force on the crankshaft at each model time step. The combined bearing reaction force on the crankshaft over the modeled time span was repeated several times to create the time waveform shown in Figure 32.

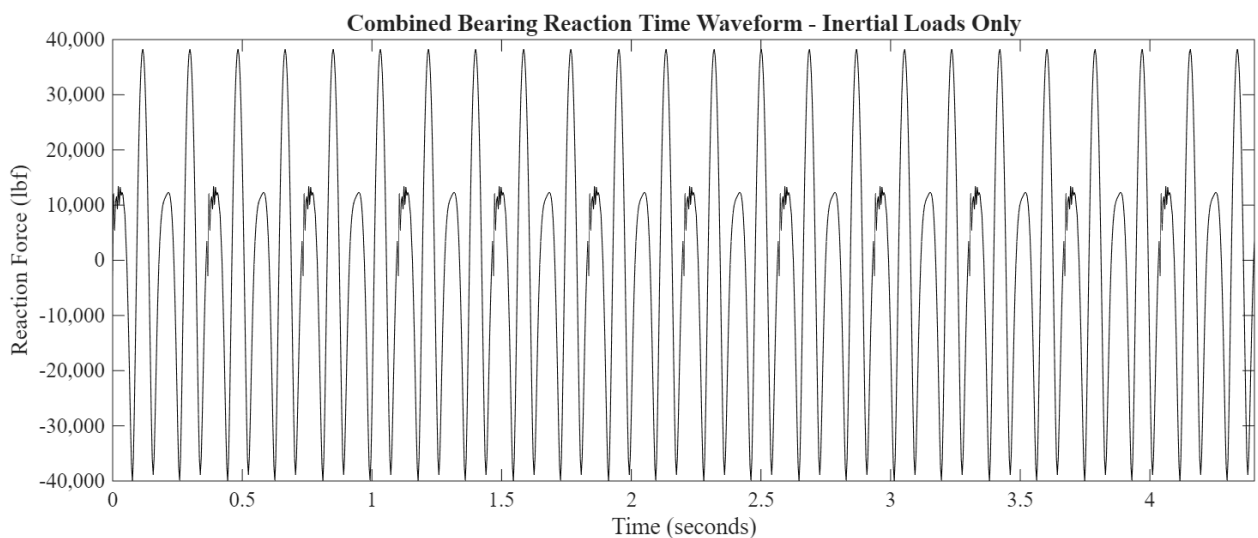


Figure 32: Total combined bearing reaction force time waveform.

The International Organization for Standardization (ISO) provides balance quality grades, which permit a classification of balance quality requirements for various machines [32]. Based on the machine of interest, the balance quality grade of the rotating assembly is used to determine the permissible residual unbalance, which is expressed in units of specific weight (oz-in). Equation 16 represents the total permissible residual unbalance, where G is the specified balance quality grade, W is the rotor weight (lbf), and N is the rotational speed of the rotating assembly (RPM) [32], [33].

$$U_{\text{per}} = \frac{G \times 6.015 \times W}{N} \quad (\text{Eq. 16})$$

There are many different acceptable balance quality grades for crankshaft/drives or complete engine assemblies, which are dependent on the engine configuration and size. While no specific balance quality grades are provided by ISO for integral engine-compressors, a balance quality grade of a G100 is specified for complete engine assemblies for cars, trucks, and locomotives [32]. This balance quality is likely conservative for a KVSR 412. However, a balance quality grade of a G100 could serve as a good reference for whether the force due to a rotating unbalance is too severe in the finite element model. Equation 17 represents the residual unbalance of a rotor, U_{res} (oz-in), where F is the force due to unbalance (lbf), and N is the rotational speed of the rotating assembly (RPM) [33].

$$U_{\text{res}} = \frac{F}{1.77 \times \left(\frac{N}{1000}\right)^2} \quad (\text{Eq. 17})$$

To find the residual unbalance of the finite element model, a spectrum was calculated using the total combined bearing reaction force time waveform from the finite element model with the gas loading suppressed. The calculated spectra allowed the component of the combined bearing reaction force that occurs at the modeled rotational speed of 330 RPM. Figure 30 shows the calculated spectrum.

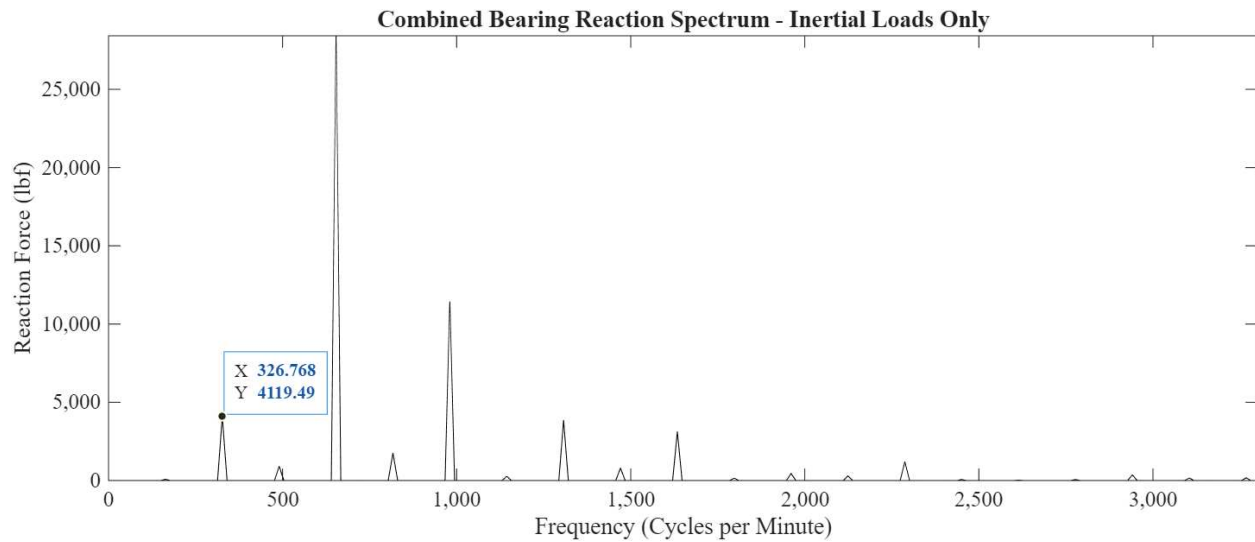


Figure 33: Spectrum calculated from the combined bearing reaction forces with the modeled gas loading suppressed.

From the calculated spectrum, it was found that the component of the total combined bearing reaction force that occurs at the crankshaft operational speed was 4,120 lbf_{pk}. By assuming this component of the total combined bearing reaction force is caused by unbalance, the residual unbalance of the modeled rotating assembly was calculated to be 21,372 oz-in using Equation 17. The calculated residual unbalance was substituted for the permissible residual unbalance in Equation 16. Following this substitution, Equation 16 was used to solve for the corresponding balance quality grade of the rotating assembly. The weight of the rotating assembly, W , used to solve for the corresponding balance quality was 19,170 lbf. This estimated weight included the weight of the crankshaft, flywheel, and one-third of the weight of each connecting rod. From this, the estimated balance quality of the modeled rotating assembly is a G61.2. Given this balance quality grade is below the recommended balance quality grade of a G100, this supports that a significant error in the reaction force at the main bearings of the modeled KVSR 412 will not occur due to an excessive rotating unbalance.

2.3.3 Bearing Load Vector Plots

To determine the optimum location around the circumference of the main bearings where the strain of the bearing saddle should be measured, it is imperative to understand at what angle the primary direction of load on the main bearing acts. Since the main bearings on the KVSr 412 are plain cylindrical journal bearings, the location of the highest oil film pressure and thus the location of the highest bearing saddle strain occurs in the direction of the primary bearing load. To visualize the direction of the main bearing loads throughout the KVSr 412's working cycle, it is useful to view the main bearing loads in an XY vector plot. Figure 34 shows an example of the vector plot based on the loads calculated from the example bearing reaction force values, which were presented in Figure 29.

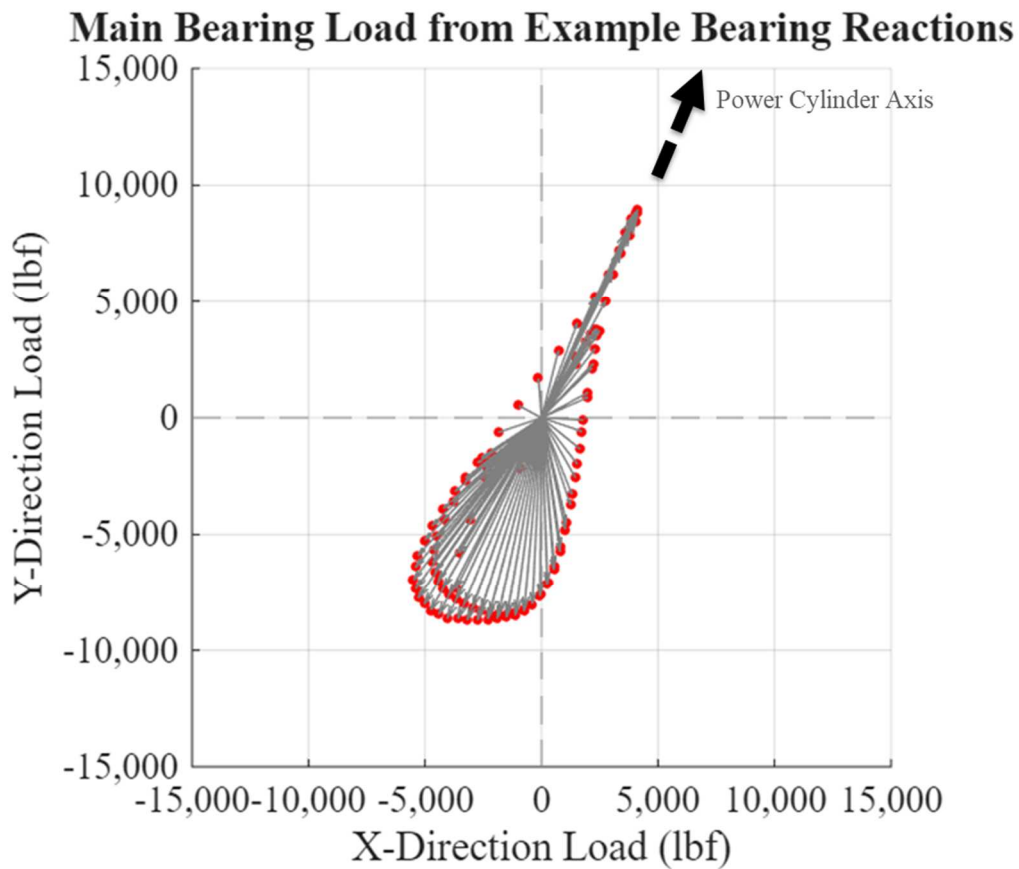


Figure 34: Example vector plot calculated from the example bearing reaction forces.

2.3.4 Calculation of Average Bearing Load Over a Working Cycle

While the vector plot of the bearing load at each modeled time step provides a visual representation of the bearing load throughout the total working cycle, a method to determine the appropriate angle at which the dynamic strain sensor should be located was needed. The bearing load vector plot for the single cylinder example model has a distinct primary direction of bearing load, which follows the angle of the single power cylinder's axis. In the full finite element model of the KVSr 412, the loads are much more complex, and a single primary direction of load is not visually obvious. For this reason, it was determined that the bearing strain should be measured at the circumferential bearing location, which corresponds to the direction of the average bearing load across the complete working cycle of the KVSr 412. This value will be referred to as the *time-weighted average load* (TWAL) and provides a reference for the direction in which the main bearing is loaded most consistently throughout the working cycle. If a bearing that supports a horizontal shaft is symmetrically loaded in the X-direction, the TWAL vector should point downward with an angle of 270°. Equations 21 and 22 were used to determine the magnitude and angle of the TWAL vector, respectively. Where L_{Avg} is the magnitude of the time-weighted average bearing load, θ_L is the angle at which the time-weighted average bearing load acts, n is the number of modeled time steps, and L_{x_i} and L_{y_i} are the values of the X and Y components of the bearing load at each respective time step.

$$L_{Avg} = \sqrt{\left(\frac{\sum_{i=1}^n L_{x_i}}{n}\right)^2 + \left(\frac{\sum_{i=1}^n L_{y_i}}{n}\right)^2} \quad (\text{Eq. 21})$$

$$\theta_L = \tan^{-1} \left[\frac{\left(\frac{\sum_{i=1}^n L_{y_i}}{n}\right)}{\left(\frac{\sum_{i=1}^n L_{x_i}}{n}\right)} \right] \quad (\text{Eq. 22})$$

Figure 35 shows the vector that represents the magnitude of the time-weighted average bearing load and the angle at which it acts, given the example reaction force components for the single cylinder model.

Average Main Bearing Load Direction Over Working Cycle:Single Cylinder Model

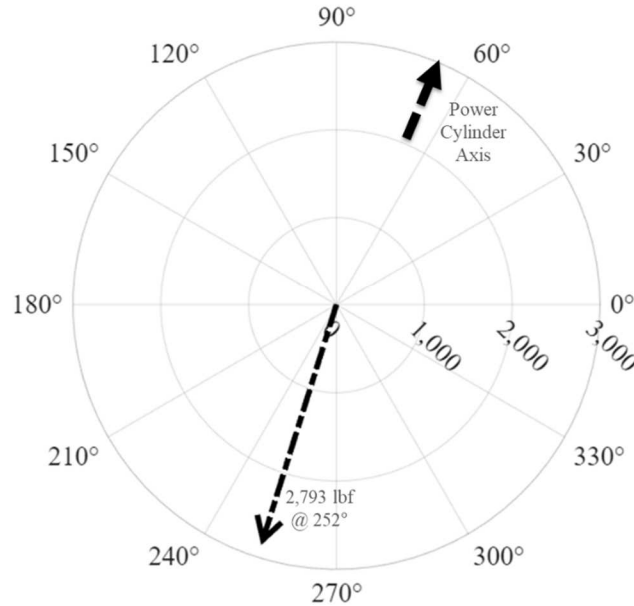


Figure 35: Time-weighted average bearing load of the single cylinder model over the working cycle.

The TWAL vector marks the location along the circumference of the main bearing at which the strain sensor should be placed. Based on the single cylinder example data, it was found that the bearing saddle strain should be measured on the opposite side of the bearing's diameter from the power cylinder axis.

CHAPTER 3 – RESULTS

The finite element analysis results for three different modeled configurations of the KVSR 412 are presented in this chapter. The configurations consist of the complete operational model described in Chapter 2, a modified version that excludes cylinder gas loading, and a simplified model that includes only the power cylinder components and their associated gas loads. For each modeled case, the X- and Y-direction components of the bearing loads, the XY vector plot of the magnitude and direction of the bearing loads, and the magnitude and angle of the TWAL are provided for each main bearing. Figure 36 provides a bearing schematic used to define the coordinate system and the angular positions around the bearing's circumference for the model results. The coordinate system is defined from the perspective of viewing the KVSR 412 from main bearing #9 toward main bearing #1.

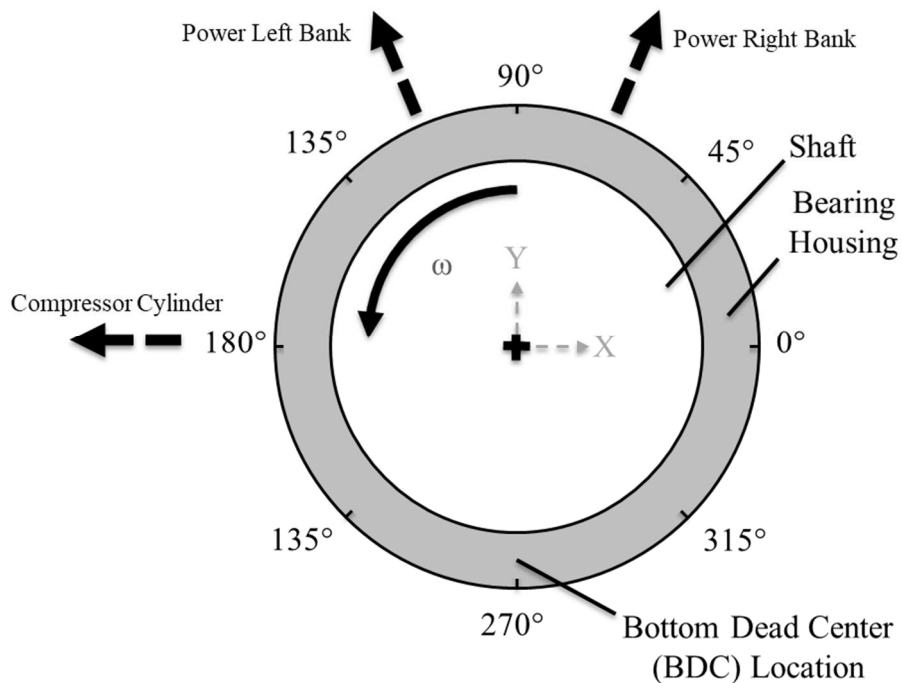


Figure 36: Bearing schematic showing the reference coordinate system and bearing angles for the results.

3.1 Complete Operational Model Results

3.1.1 Bearing Load Vector Plots

The results for the operational model include the effect of all modeled components and the associated gas loads acting on each cylinder. The plots that present the X- and Y-direction components of the load that acts on each main bearing throughout one working cycle from the complete operational model are provided in Appendix B. In general, the load due to the power cylinders and compressor cylinders on a given crankshaft throw is shared between the main bearings on each side of the throw. The relative significance of the load acting on each main bearing from a given cylinder depends on the distance from the cylinder to each main bearing. If a cylinder is closer to a main bearing, that main bearing will support a greater percentage of the cylinder's load.

Significant loads on the main bearings are caused by the loading associated with the power and compressor cylinders, and the rotating and reciprocating inertia of the moving components. The peak firing pressure of the various power cylinders causes the most significant magnitudes of the main bearing loads, but these events have a short duration. However, when the loads from the power and compressor cylinders act in the same direction at a specific crank angle, their combined effect results in increased main bearing loads due to the summation of both forces. In the same way, when loads from a power cylinder and a neighboring compressor cylinder act in opposing directions at the same crank angle, the combination of their forces leads to lower bearing loads. Figure 37 shows the magnitude and direction of the load on the #1 main bearing at each degree of crankshaft rotation throughout the KVSr 412's working cycle. The blue dot, which is overlaid on the vector plots presented in this chapter, represents the tip of the TWAL vector for each main

bearing. The crank angles at which significant events that cause high main bearing loads are noted on the vector plots for each main bearing.

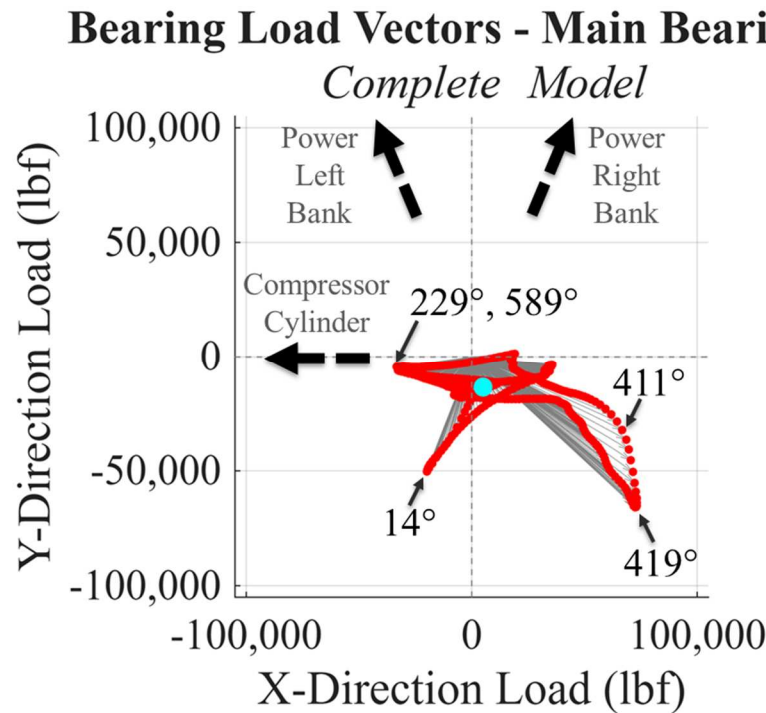


Figure 37: Main bearing #1 load vectors from the complete KVSr 412 finite element model.

The maximum magnitude of the load acting on main bearing #1 occurs at 419° of crankshaft rotation. This crank angle corresponds to the peak firing pressure of the #7 power cylinder. Additionally, an event can be seen at 14° of crankshaft rotation. This crank angle corresponds to the peak firing pressure of the #1 power cylinder. Other events are noted at 229°, 411°, and 589°, which correspond to the crank angles when the total combined load on the crosshead pin of compressor cylinder #1 is high. Figure 38 shows a plot of the gas loads on the #1 and #7 power cylinders, and the combined load at the crosshead pin of the #1 compressor cylinder. The total combined load on the crosshead pin, which is the summation of the gas load of the compressor cylinder and the load caused by the inertia of the cylinder's reciprocating components, was exported from the database software, which is used to store data from the Windrock 6400 [34].

A negative value of the total load on the crosshead pin of a compressor cylinder denotes that the compressor cylinder connecting rod is in compression, and the load on the crosshead pin is acting in the positive X-direction.

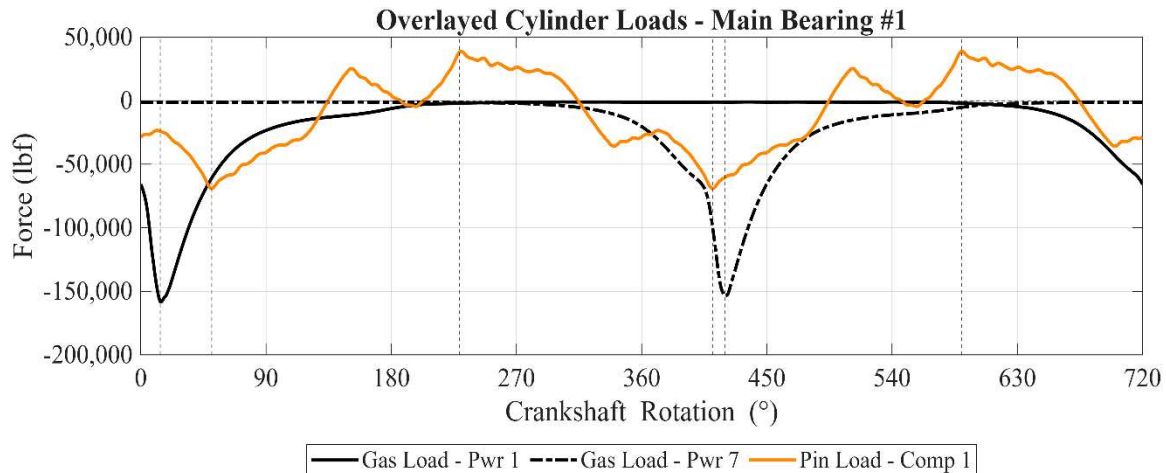


Figure 38: Overlaid plot of cylinder loads on main bearing #1. The vertical hashed lines mark the angles of the significant events marked in Figure 37.

By comparing Figures 37 and 38, the load on main bearing #1 is much higher when the #7 power cylinder reaches peak firing pressure than when the #1 power cylinder does. Part of the difference in bearing load at each event is related to the position of each power cylinder relative to main bearing #1, with the #7 power cylinder located closer to the bearing than the #1 power cylinder. However, the magnitude of the bearing load is also influenced by the direction and relative magnitude of the compressor cylinder load, as well as the timing of the loads from both the compressor and power cylinders. The load from compressor cylinder #1 reaches a maximum in the positive X-direction only 8° before the #7 power cylinder reaches peak firing pressure. Given that the #7 power cylinder is on the left bank of the KVSr 412, the load from this cylinder will have a component that acts in the positive X-direction. The combination of loads from the two cylinders leads to an increase in the bearing load at the crank angles associated with the peak firing pressure of the #7 power cylinder. In contrast, the load from compressor cylinder #1 also reaches

a maximum in the positive X-direction 36° after peak firing pressure on the #1 power cylinder. The greater difference in crank angles between these two events makes the effect the compressor cylinder has on the load of main bearing #1 less pronounced throughout the combustion event of the #1 power cylinder. The same is true for the events that occur at 229° and 589° of crankshaft rotation. These events correspond to the load from the #1 compressor cylinder reaching maximums in the negative X-direction. As no significant power cylinder events occur at these crank angles, the main bearing load acts primarily in the negative X-direction. Since the load due to the #1 compressor cylinder caused higher loading during the combustion event of the #7 power cylinder, the angle of the TWAL vector of main bearing #1 increased from the bottom dead center location at 270° .

Figure 39 shows the magnitude and direction of the bearing load that acts on main bearing #2. Elevated bearing loads are observed at 14° , 180° , and 419° of crankshaft rotation and correspond to peak firing pressures in power cylinders #1, #8, and #7, respectively. Of these events, the maximum bearing load magnitude occurs at the peak firing pressure of power cylinder #1. The #1 power cylinder is closer to the #2 main bearing than it is to the #1 main bearing. This leads to higher loading on the #2 main bearing during the combustion event of the #1 power cylinder. In addition, when the #1 power cylinder is at peak firing pressure, the combined load from the #1 compressor cylinder is low. Since the load from the #1 compressor cylinder does not apply a significant load along the X-axis at this crank angle, the bearing load when the #1 power cylinder is at peak firing pressure follows the approximate bank angle of the power cylinder. The #8 power cylinder is further from the #2 main bearing than the #3 main bearing, as there is an unused connecting rod journal on the second crankshaft throw. Given this, the load on the #2 main bearing related to the combustion event of the #8 power cylinder is relatively low. Since the load from the

first crankshaft throw is shared between main bearings #1 and #2, the bearing load associated with the #1 compressor cylinder and the combustion event of the #7 power cylinder also acts on main bearing #2.

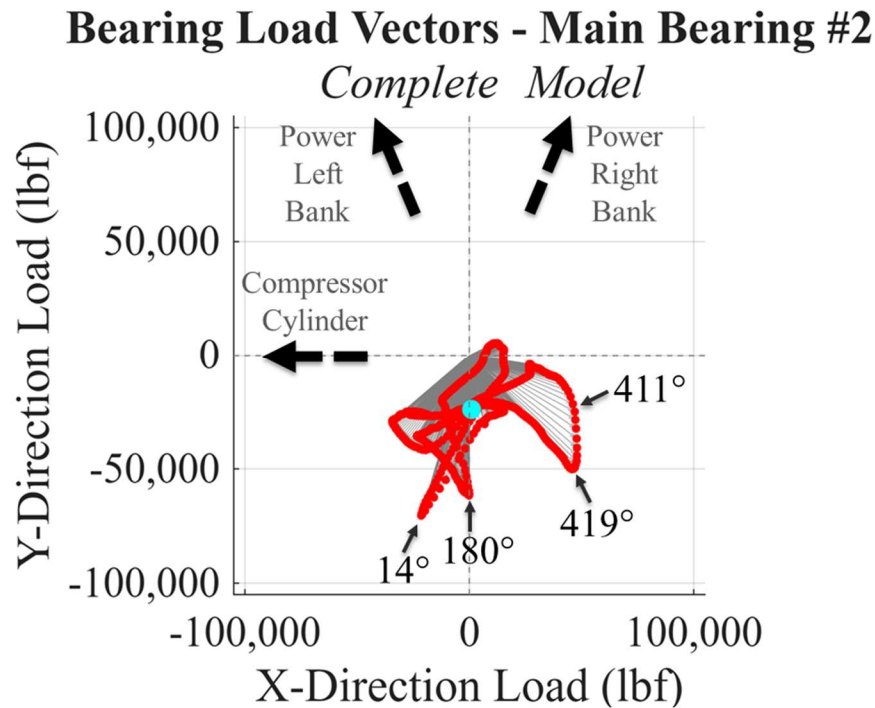


Figure 39: Main bearing #2 load vectors from the complete KVSr 412 finite element model.

The #1 power cylinder is closer to the #2 main bearing than it is to the #1 main bearing. This leads to higher loading on the #2 main bearing during the combustion event of the #1 power cylinder. In addition, when the #1 power cylinder is at peak firing pressure, the combined load from the #1 compressor cylinder is low. Since the load from the #1 compressor cylinder does not apply a significant load along the X-axis at this crank angle, the bearing load when the #1 power cylinder is at peak firing pressure follows the approximate bank angle of the power cylinder. The #8 power cylinder is further from the #2 main bearing than the #3 main bearing, as there is an unused connecting rod journal on the second crankshaft throw. Given this, the load on the #2 main bearing related to the combustion event of the #8 power cylinder is relatively low. Since the load from the first crankshaft throw is shared between main bearings #1 and #2, the bearing load

associated with the #1 compressor cylinder and the combustion event of the #7 power cylinder also acts on main bearing #2. However, since main bearing #2 is farther from the #7 power cylinder and the #1 compressor cylinder, the load is much lower than what acts on main bearing #1. Given the lower load on the bearing associated with the #1 compressor cylinder, the TWAL vector for main bearing #2 increased less than the vector of the #1 main bearing. Figure 40 shows the magnitude and direction of the loads on the #3 main bearing throughout the working cycle of the KVSr 412.

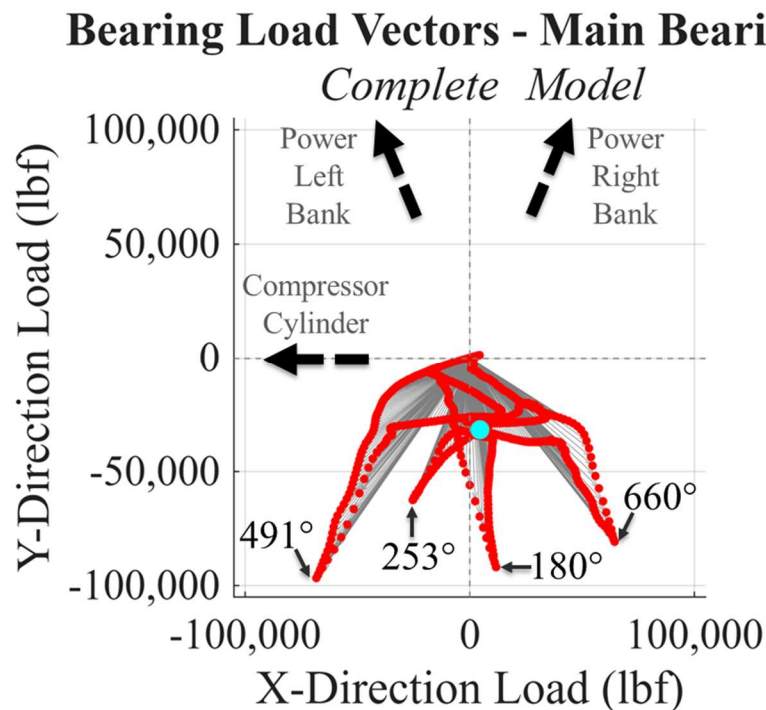


Figure 40: Main bearing #3 load vectors from the complete KVSr 412 finite element model.

Main bearing #3 has the most significant magnitude of load of all the KVSr 412 main bearings. Significant loads which act on main bearing #3 occur at 180°, 253°, 491°, and 660° of crankshaft rotation, which correspond to the peak firing pressure of power cylinders #8, #3, #2, and #9, respectively. The load due to the peak firing pressure of the #8 power cylinder, which is shared between main bearings #2 and #3, is higher on main bearing #3. This is due to power

cylinder #8 being closer to main bearing #3. Figure 41 shows the relative timing of the loads from the power cylinders, which neighbor main bearing #3, as well as the combined load from the #2 compressor cylinder.

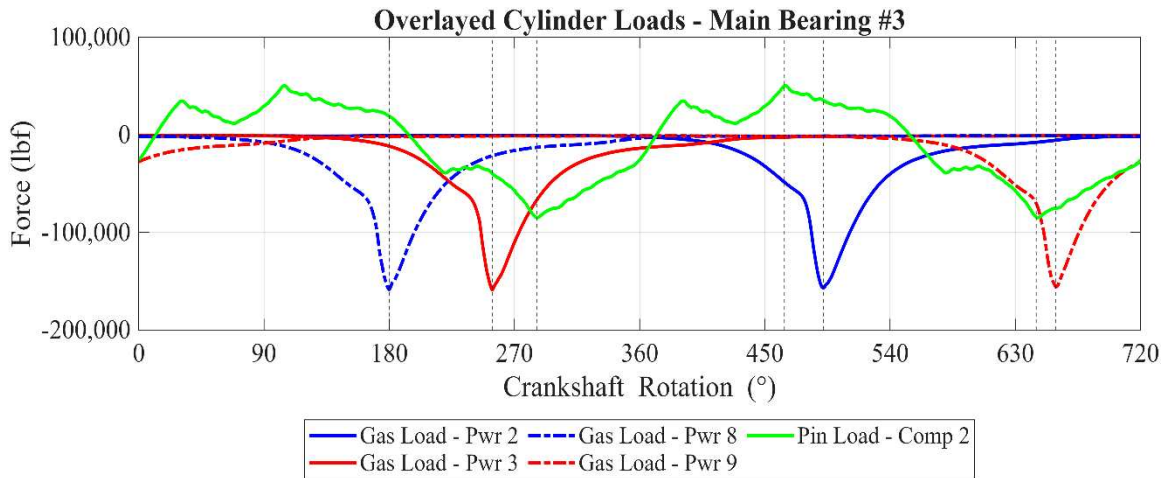


Figure 41: Overlaid plot of cylinder loads on main bearing #3.

While the magnitudes of the gas load from the #2, #3, #8, and #9 power cylinders are all similar at peak firing pressure, the bearing load for each cylinder's combustion event varies in magnitude and direction. The bearing load, which occurs at the peak firing pressure of power cylinder #3, has the lowest magnitude since power cylinder #3 is the farthest cylinder from main bearing #3. Only 14° before the #9 power cylinder is at peak firing pressure, the combined load due to the #2 compressor cylinder is at a maximum in the positive X-direction. This loading acts in the same direction as the X-direction component of the load from the #9 power cylinder. This leads to a higher magnitude of bearing load during this event and causes the bearing load to act at a higher angle than the axis of the #9 power cylinder.

A similar loading occurs since the load due to the #2 compressor cylinder acts in the negative X-direction when the #2 power cylinder is at peak firing pressure. This causes a higher bearing load during the #2 power cylinder combustion event and causes the bearing load to act at a higher angle than the axis of the #2 power cylinder. As the #2 power cylinder is the closest cylinder to the

#3 main bearing, the total bearing load during the combustion event of this cylinder is greater than the bearing load during the combustion event of the #9 power cylinder. Given that main bearing #3 shares the load from four power cylinders and only one compressor cylinder, the load from the compressor cylinder causes only a small increase in the angle of the TWAL vector. Figure 42 shows the magnitude and direction of the loads on the #4 main bearing throughout the working cycle of the KVSr 412.

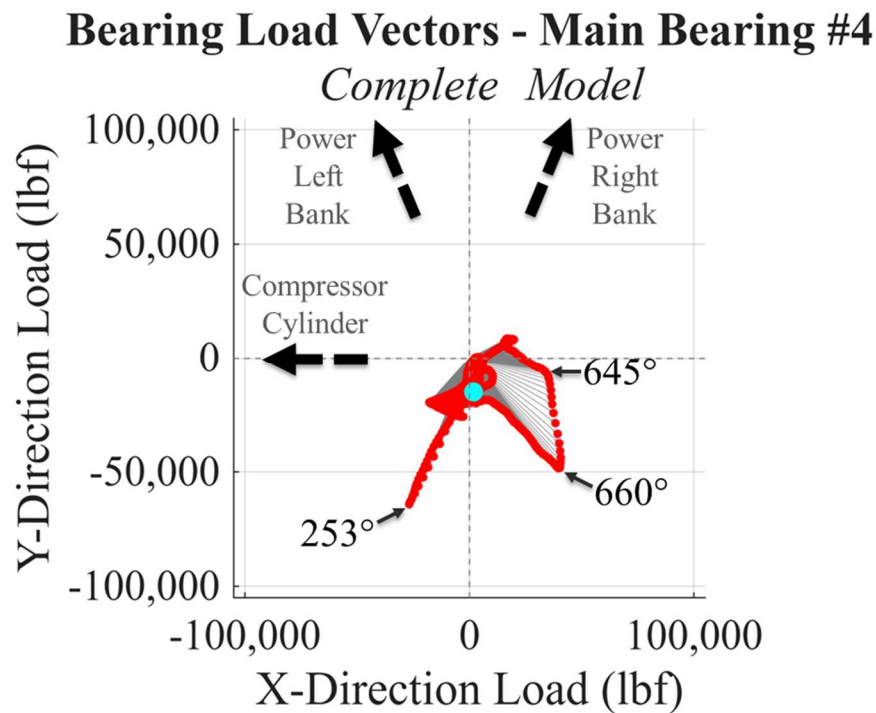


Figure 42: Main bearing #4 load vectors from the complete KVSr 412 finite element model.

The only cylinders that neighbor the #4 main bearing are the #3 and #9 power cylinders and the #2 compressor cylinder. The loads related to the peak firing pressure of the #3 and #9 power cylinders are shared between main bearings #3 and #4. However, the magnitude of the bearing load at peak firing pressure of power cylinder #3 is higher on the #4 bearing than the load related to peak firing pressure of the #9 power cylinder. This is because the #3 power cylinder is closer to main bearing #4. The effect of the load from the #2 compressor cylinder can be seen in the event

at 645° , which leads to the higher loading in the positive X-direction prior to the #9 power cylinder reaching peak firing pressure. The higher loading in the X-direction, which is related to load from the #2 compressor cylinder, causes the angle of the TWAL vector to increase slightly. However, unlike the #1 main bearing, which was located directly next to the #1 compressor cylinder, main bearing #4 is located on the opposite side of the second crankshaft throw from the #2 compressor cylinder. Given this, the load from the #2 compressor cylinder is less significant to main bearing #4, which causes a less pronounced increase in the angle of the TWAL vector than was seen on main bearing #1. Figure 43 shows the magnitude and direction of the load on main bearing #5 throughout the working cycle of the KVSr 412.

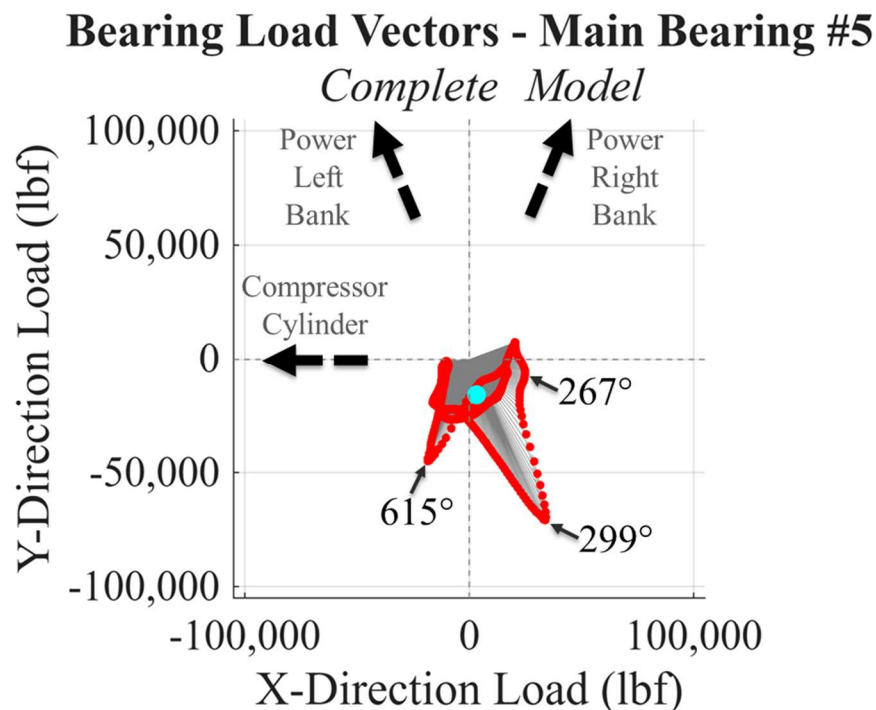


Figure 43: Main bearing #5 load vectors from the complete KVSr 412 finite element model.

The events at 299° and 615° of crankshaft rotation correspond to the crank angles at which the peak firing pressure of the #10 and #4 power cylinders occur, respectively. The event at the crank angle of 267° is related to the load due to the #3 compressor cylinder. Figure 44 provides an

overlaid plot of the loads from the #4 and #10 power cylinders, as well as the #3 compressor cylinder.

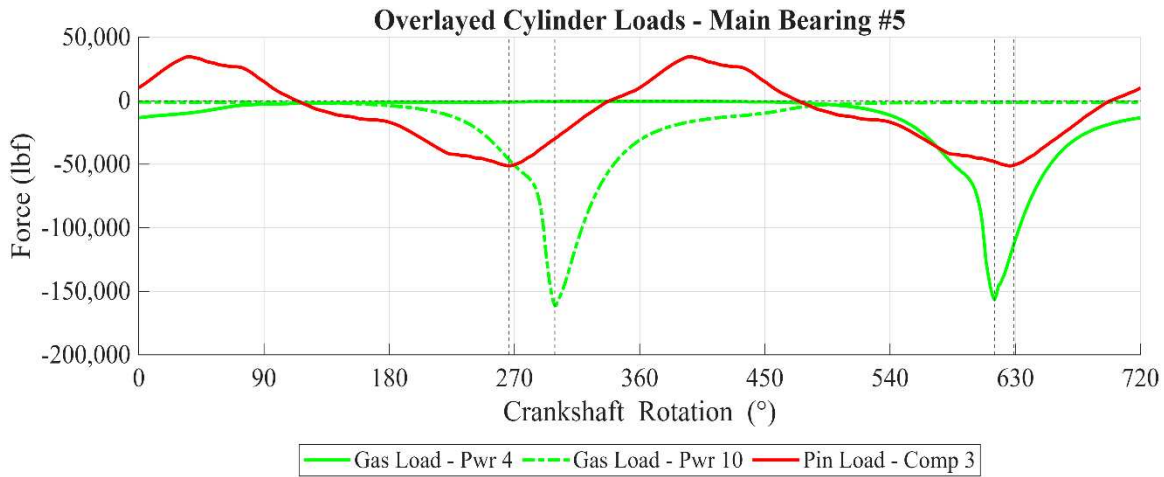


Figure 44: Overlaid plot of the cylinder loads that act on main bearing #5.

The load on main bearing #5 is lower when the #4 power cylinder is at peak firing pressure compared to the load when the #10 power cylinder is at peak firing pressure. This is due to the #4 power cylinder being located further from the #5 main bearing. Additionally, when the #4 power cylinder is at peak firing pressure, the magnitude of the load due to the #3 compressor cylinder is nearly at a maximum in the positive X-direction. This loading in the positive X-direction resists some of the force from the #4 power cylinder, which also leads to lower bearing load during the cylinder's combustion event. In contrast, the load due to the #3 compressor cylinder reaches a maximum in the positive X-direction 33° prior to the #10 power cylinder reaching peak firing pressure. The load due to the compressor cylinder causes a more significant X-direction main bearing load throughout the combustion event of the #10 power cylinder. The higher load in the positive X-direction due to the load from the #3 compressor cylinder causes an increase in the angle of the TWAL vector of main bearing #5.

Figure 45 shows the magnitude and direction of the load on the #6 main bearing throughout the KVSr 412 working cycle. The vector plot of the #6 main bearing shows similar loading to the

#3 main bearing, but the four listed events don't show a significant effect on the bearing load related to the compressor cylinder. This is related to the lower load of the #3 compressor cylinder, as compressor cylinders #3 and #4 had a similar measured pressure to the other compressor cylinders, but also have a smaller piston diameter. The significant loads which act on the #6 main bearing at 134°, 299°, 538°, and 615° of crankshaft rotation correspond to peak firing pressure of the #5, #10, #11, and #4 power cylinders, respectively.

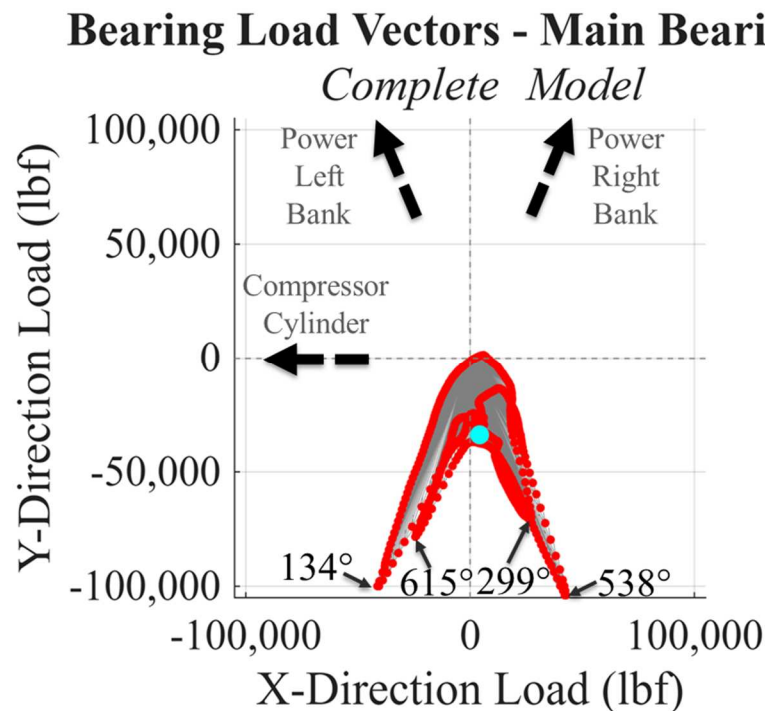


Figure 45: Main bearing #6 load vectors from the complete KVSr 412 finite element model.

The loads acting on the #6 main bearing due to the peak firing pressure of #4 and #10 power cylinders are closer in magnitude than they were on main bearing #5, since the #4 power cylinder is closer to main bearing #6. The events related to the peak firing pressure of the #5 and #11 power cylinders are only minorly effected by the load due to the #3 compressor cylinder, as the load from the compressor cylinder is low at these crank angles. As the load on the #6 main bearing is

primarily controlled by the combustion events of the four power cylinders, the angle of the TWAL vector of main bearing #6 only increased slightly due to the load from the compressor cylinder.

Figure 46 shows the magnitude and the direction of the load on the #7 main bearing throughout the working cycle of the KVSR 412. The highest magnitude of the load on the #7 main bearing occurs at 59° , which corresponds to the peak firing pressure of the #12 power cylinder. The #12 power cylinder is the closest cylinder to the #7 main bearing, which leads to the event at 59° of crankshaft rotation to have a higher magnitude. In addition, the load due to the #4 compressor cylinder affects the bearing load due to the combustion event of both power cylinders.

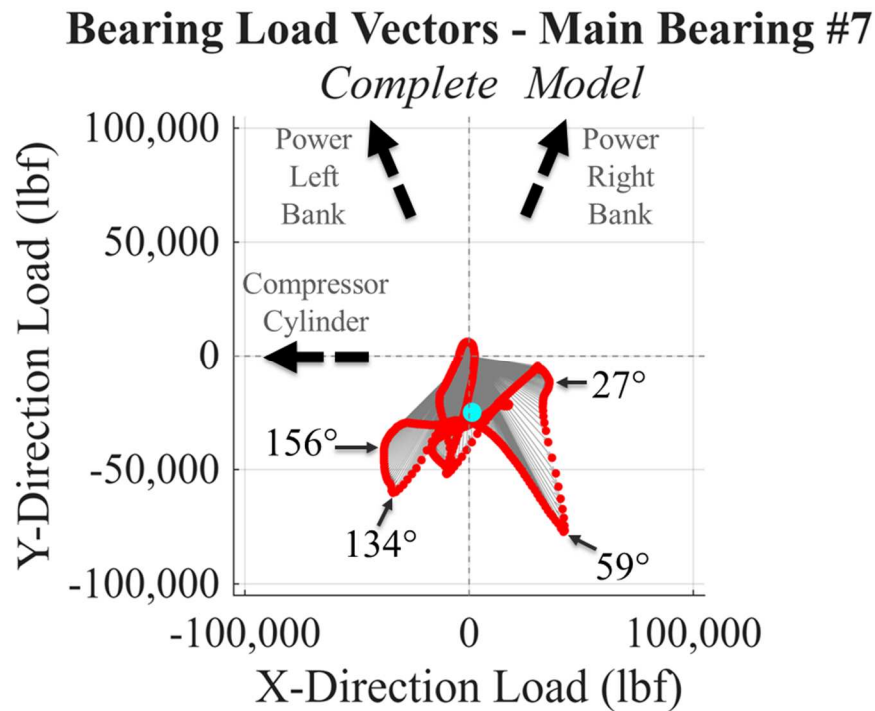


Figure 46: Main bearing #7 load vectors from the complete KVSR 412 finite element model.

Figure 47 provides an overlaid plot of the loads from the #5 and #12 power cylinders, as well as the #4 compressor cylinder. By comparing both figures, it can be seen that the load due to the #4 compressor cylinder reaches a maximum in the positive X-direction 32° prior to the peak firing

pressure of the #12 power cylinder. This leads to a shift of the main bearing load in the positive X-direction, which can be seen between 27° and 59° of crankshaft rotation.

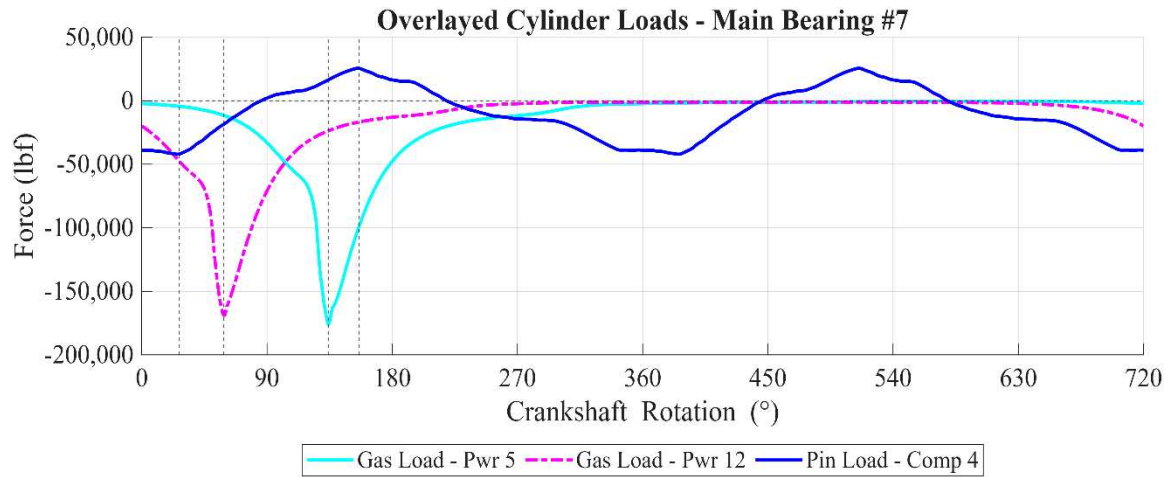


Figure 47: Overlaid plot of the cylinder loads which act on main bearing #7.

Similarly, the load due to the #4 compressor cylinder reaches a maximum in the negative X-direction 22° after the peak firing pressure of the #5 power cylinder. This causes the shift of the bearing load in the negative X-direction, which is seen between 134° and 156° of crankshaft rotation. Given that the load due to the #3 compressor cylinder causes high loading in the positive and negative X-direction, the angle of the TWAL vector of the #7 main bearing does not increase significantly from the bottom dead center location. Figure 48 shows the magnitude and the direction of the load on the #8 main bearing throughout the working cycle of the KVSr 412. The significant events that cause high load on the #8 main bearing to occur at 59° and 375° of crankshaft rotation correspond to peak firing pressure of the #12 and the #6 power cylinder, respectively. The load due to the #4 compressor cylinder near the peak firing pressure of the #12 power cylinder is more significant on the #8 main bearing than it was on the #7 main bearing. This is due to the #4 compressor cylinder being located next to the #8 main bearing.

Figure 49 provides an overlaid plot of the loads from the #6 and #12 power cylinders, as well as the #4 compressor cylinder. The load due to the #4 compressor cylinder reaches a maximum

in the positive X-direction only 10° of crankshaft rotation after the peak firing pressure of the #6 power cylinder. Given the close timing of these two events, the load due to the #4 compressor cylinder does not cause significant X-direction loading on the #8 main bearing. Instead, the event at 375° of crankshaft rotation is narrower. Since the #4 compressor cylinder is located directly next to the #8 main bearing, the load from the compressor cylinder caused a more significant increase in the angle of the TWAL vector for this bearing.

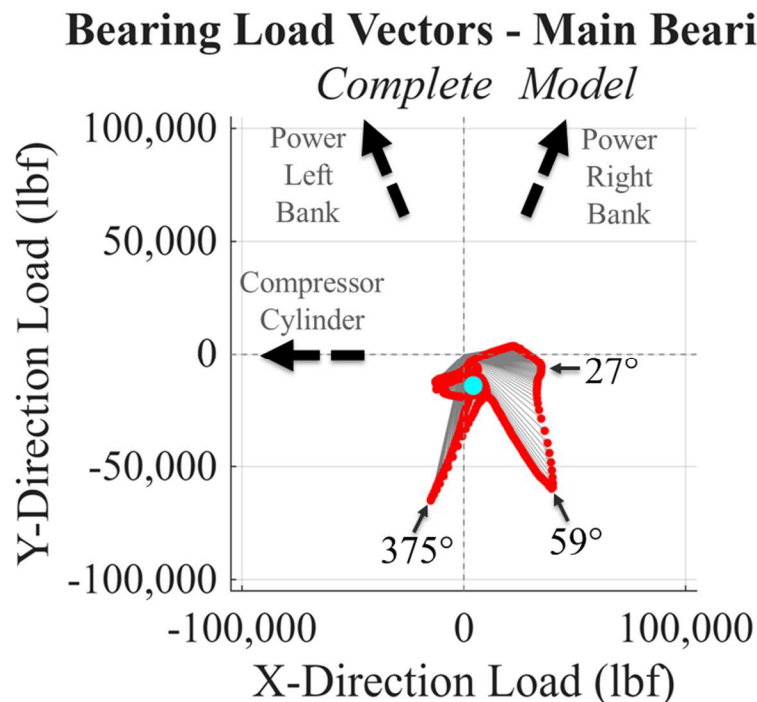


Figure 48: Main bearing #8 load vectors from the complete KVSr 412 finite element model. The load due to the #4 compressor cylinder reaches a maximum in the positive X-direction only 10° of crankshaft rotation after the peak firing pressure of the #6 power cylinder. Given the close timing of these two events, the load due to the #4 compressor cylinder does not cause significant X-direction loading on the #8 main bearing. Instead, the event at 375° of crankshaft rotation is narrower. Since the #4 compressor cylinder is located directly next to the #8 main bearing, the load from the compressor cylinder caused a more significant increase in the angle of the TWAL vector for this bearing.

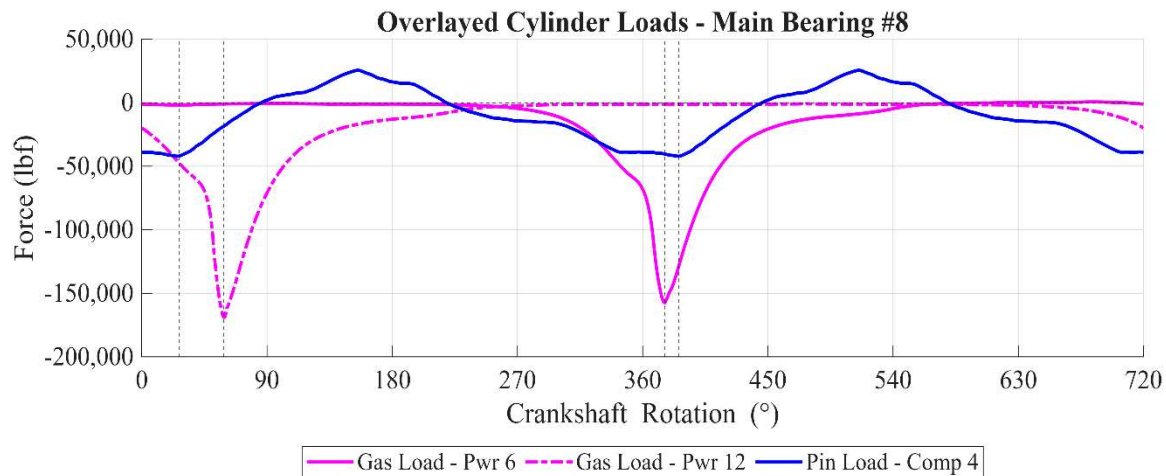


Figure 49: Overlaid plot of the cylinder loads which act on main bearing #8.

Figure 50 shows the direction and magnitude of the load on the #9 main bearing throughout the working cycle of the KVSR 412. However, the #9 main bearing primarily supports the static load of the overhung flywheel. For this reason, the vector points are obscured by the blue marker, which represents the tip of the TWAL vector. Since main bearing #9 supports the weight of the flywheel, the TWAL vector points downward.

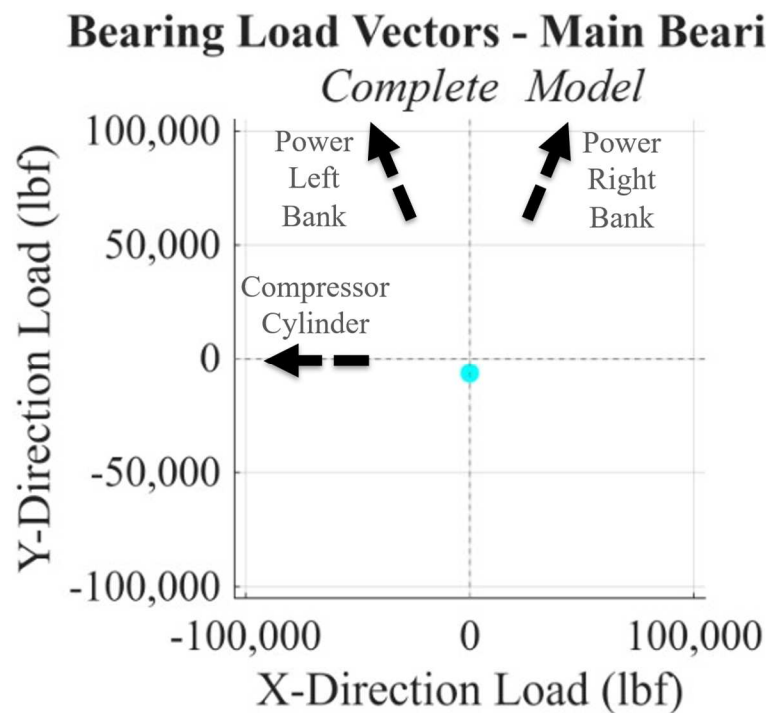


Figure 50: Main bearing #9 load vectors from the complete KVSR 412 finite element model.

3.1.2 Time-Weighted Average Load Vector

The magnitude and angle of the TWAL vector for each main bearing, which were calculated from the bearing load vector plots, are provided in Table 3.

Table 3: Magnitudes and angles of the time-weighted average load vectors calculated for the complete operational model.

Main Bearing	Magnitude (lbf)	Angle (°)
1	14,441	290
2	24,072	272
3	32,043	278
4	15,162	277
5	16,263	282
6	34,112	277
7	25,034	273
8	14,860	287
9	6,487	270

The magnitude of the TWAL vector provides an average value of the load supported by each main bearing throughout the working cycle under the modeled conditions. From these magnitudes, it can be seen that bearings which are equidistant from the large counterweight at the center of the crankshaft share a similar magnitude of load (e.g. main bearing #3 is similarly loaded to main bearing #6). Additionally, the angle of the TWAL vector is similar for bearings that are equidistant from the large counterweight in the middle of the crankshaft.

The angles of the TWAL vectors for the #1 and #8 main bearings increased the most due to the load of the compressor cylinders. The #1 and #8 main bearings directly neighbor compressor cylinders and only neighbor two power cylinders. This causes the loads due to the compressor cylinders to have a more significant effect on the TWAL for these bearings and causes the greatest increase in the angle of each bearing's TWAL vector. While the #3 and #6 main bearings also neighbor compressor cylinders, these bearings support load from four nearby power cylinders.

Given that the load from the four nearby power cylinders is far more significant than the compressor cylinders, the load due to the #2 and #3 compressor cylinders has far less influence on the TWAL of these bearings. This causes less of a shift in the angle of the TWAL vectors for main bearings #3 and #6.

The #4 and #5 main bearings each only support the load of two power cylinders and one compressor cylinder, similar to main bearings #1 and #8. However, the #4 and #5 main bearings are located on the opposite side of the crankshaft throws from the #2 and #3 compressor cylinders. Given this, the loads due to the compressor cylinders only cause a slight increase in the angle of the TWAL vectors of these bearings. Finally, the #2 and #7 main bearings support the load from four nearby power cylinders and are the bearings that are furthest separated from any compressor cylinders. This causes the load due to the #1 and #4 compressor cylinders to have little effect on the angle of the TWAL vectors for these bearings, leading to the angle of the TWAL vector to remain near 270° . This supports that the #2 and #7 main bearings are the most symmetrically loaded. The plots that present the TWAL vectors for each main bearing are provided in Appendix C.

3.2 Simplified Model Without Gas Loading

3.2.1 Bearing Load Vector Plots

This section presents the results for a simplified KVSr 412 model, which includes all modeled components but excludes the gas loads associated with the power and compressor cylinders. The inertial model of the KVSr 412 was created to quantify the portion of the main bearing loads that are related to the inertia of rotating and reciprocating components. Additionally, the results from this model were utilized to quantify the assembled balance quality of the KVSr 412 model as discussed in Chapter 2. The plots that present the X- and Y-direction components of the load that

acts on each main bearing throughout one working cycle for the inertial model are provided in Appendix D. Figure 51 shows the direction and magnitude of the load on the #1 and #8 main bearings throughout the working cycle of the KVSr 412, which was output from the inertial model. Due to the symmetry of the geometry about the large counterweight in the center of the crankshaft's length, the loads throughout the working cycle of the KVSr 412 are similar for bearings that are equidistant from the large counterweight. Given this, the vector plot representing the loads that act on main bearing #1 is similar to the plot representing the loads at main bearing #8. For a rotating unbalance, the XY vector plot of bearing loads at each crank angle of the working cycle would draw a circle. However, the loads associated with the inertia of the reciprocating components of the compressor cylinders, which neighbor main bearings #1 and #8, cause the oval shape.

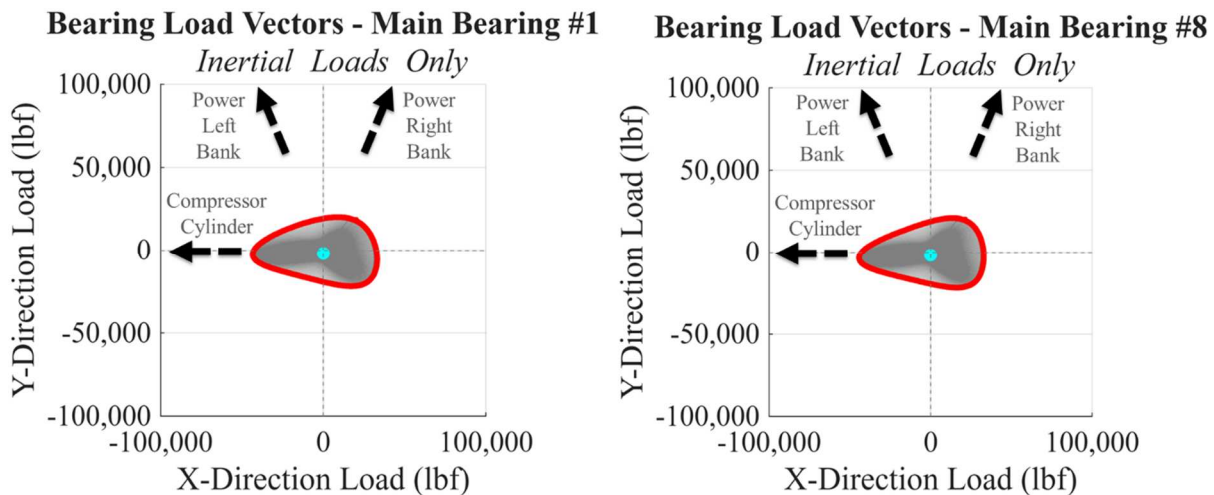


Figure 51: Load vectors for main bearings #1 and #8 from the inertial KVSr 412 finite element model.

Figure 52 shows the direction and magnitude of the load on the #2 and #7 main bearings throughout the working cycle of the KVSr 412, which was output from the inertial model. Main bearings #2 and #7 are the same distance from the large counterweight in the center of the crankshaft, and therefore, both bearings support a similar load in the inertial model.

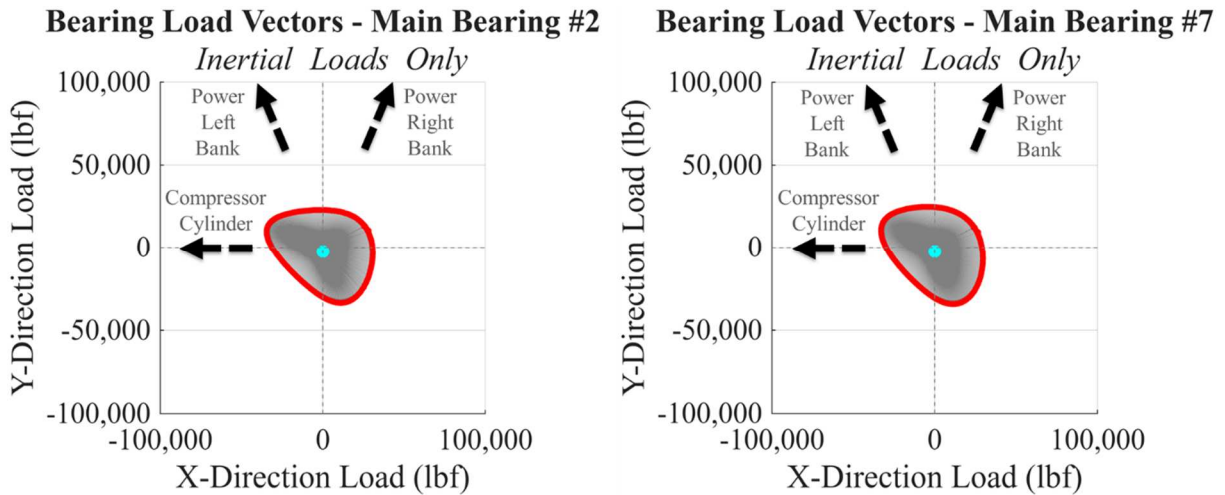


Figure 52: Load vectors for main bearings #2 and #7 from the inertial KVSr 412 finite element model.

As discussed in the complete operational model, the #2 and #7 main bearings are located the furthest from any of the compressor cylinders. Given this, the inertial forces from the reciprocating compressor cylinder components have very little effect on the #2 and #7 main bearings, making the shape of the vector plots for each of these bearings more circular.

Figure 53 shows the direction and magnitude of the load on the #3 and #6 main bearings throughout the working cycle of the KVSr 412, which was output from the inertial model. Main bearings #3 and #6 are equidistant from the large counterweight at the center of the crankshaft and experience a similar load. Both main bearings neighbor a compressor cylinder and support the inertial load from four power cylinders. Since the #3 and #6 main bearings neighbor the most cylinders, they support the largest amount of load due to rotating and reciprocating inertia.

Figure 54 shows the direction and magnitude of the load on the #4 and #5 main bearings throughout the working cycle of the KVSr 412, which was output from the inertial model. Main bearings #4 and #5 are located directly next to the large counterweight in the center of the crankshaft. Since compressor cylinders #2 and #3 are located on the opposite side of the crankshaft throws from these main bearings, the bearings support little inertial load from the reciprocating components of

the compressor cylinders. Instead, main bearings #4 and #5 support a large rotating force due to unbalance related to the large central counterweight. The force due to unbalance from the central counterweight causes the load vector plots for these bearings to be the most circular. Since the #9 main bearing primarily supports the static load of the overhung flywheel, the XY vector plot for this bearing is identical to the results shown in Figure 50.

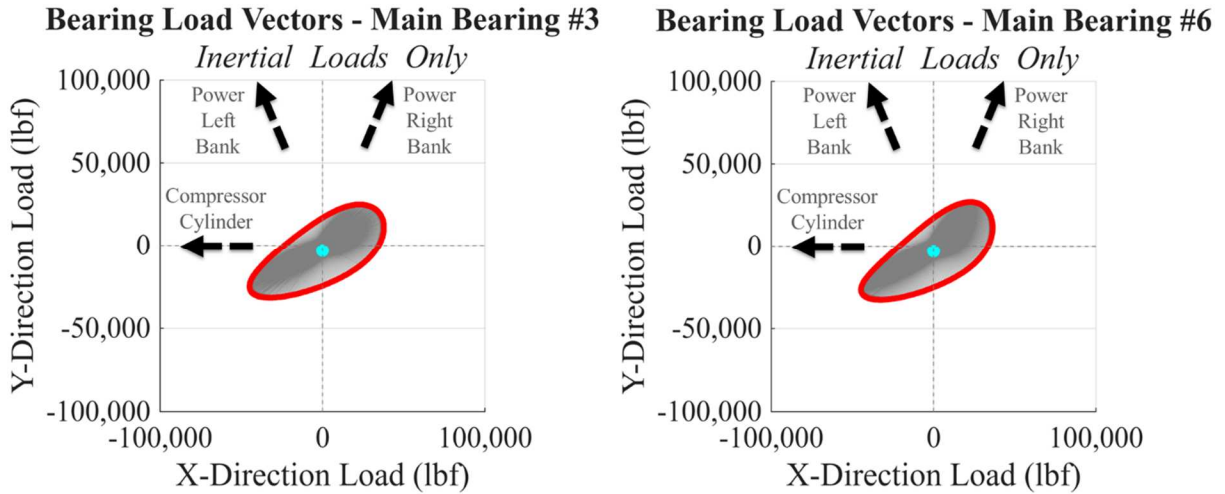


Figure 53: Load vectors for main bearings #3 and #6 from the inertial KVSr 412 finite element model.

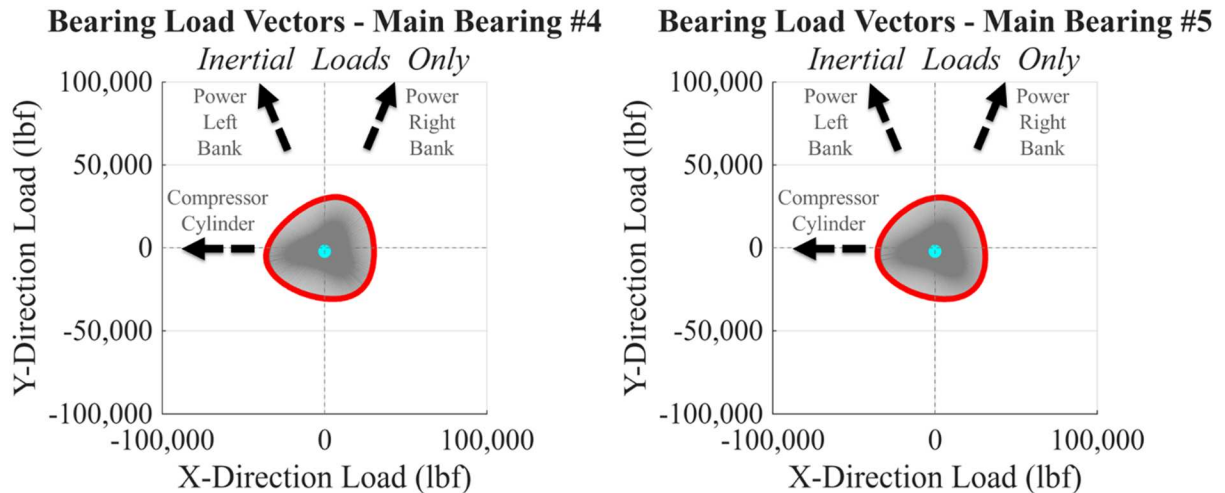


Figure 54: Load vectors for main bearings #4 and #5 from the inertial KVSr 412 finite element model.

3.2.2 Time-Weighted Average Load Vector

Table 4 presents the magnitudes and angles of the TWAL vectors for each main bearing based on only the inertial loads of the KVSR 412. The inclusion of this simplified model, which only includes the inertial loads associated with components of the KVSR 412, provides key insight into the effect the gas loading has on the main bearing loads. By comparing the magnitudes of the TWAL vectors in Tables 3 and 4, it can be seen that the gas loading from the power and compressor cylinders is the primary cause of the high main bearing loads.

Table 4: Magnitudes and angles of the time-weighted average load vectors calculated for the finite element model, which includes only inertial loads.

Main Bearing	Magnitude (lbf)	Angle (°)
1	1,859	270
2	2,486	272
3	3,110	269
4	2,138	270
5	2,141	270
6	3,117	269
7	2,484	272
8	1,800	270
9	6,566	270

The relationship between a higher magnitude of the TWAL vector at bearings that neighbor more power and compressor cylinders was seen in both the complete operational model and the inertial model. In addition to the gas loading being the primary cause of higher TWAL on each main bearing, it is also apparent that the gas loads due to the compressor cylinders lead to the increase in the angles of the TWAL vectors of the main bearings, which support the load of the compressor cylinders. Since the piston area of the crank-end of each compressor is smaller than the head-end due to the area taken by the piston rod, the gas load on each compressor cylinder averaged across the working cycle acts in the positive X-direction. This accounts for the increase

in the angle of the TWAL vector at certain bearings seen in the complete operational model. The plots that present the TWAL vectors for each main bearing are provided in Appendix E.

3.3 Simplified Model with Only Power Cylinders

3.4.1 Bearing Load Vector Plots

This section presents the results for a simplified KVSR 412 finite element model, which includes the modeled power cylinder components and the gas loads associated with each power cylinder, but excludes the geometry and gas loads associated with the compressor cylinders. This model was created to determine the recommended dynamic strain measurement locations for a symmetrically loaded engine. Figure 55 shows the simplified finite element model including only the power cylinders.

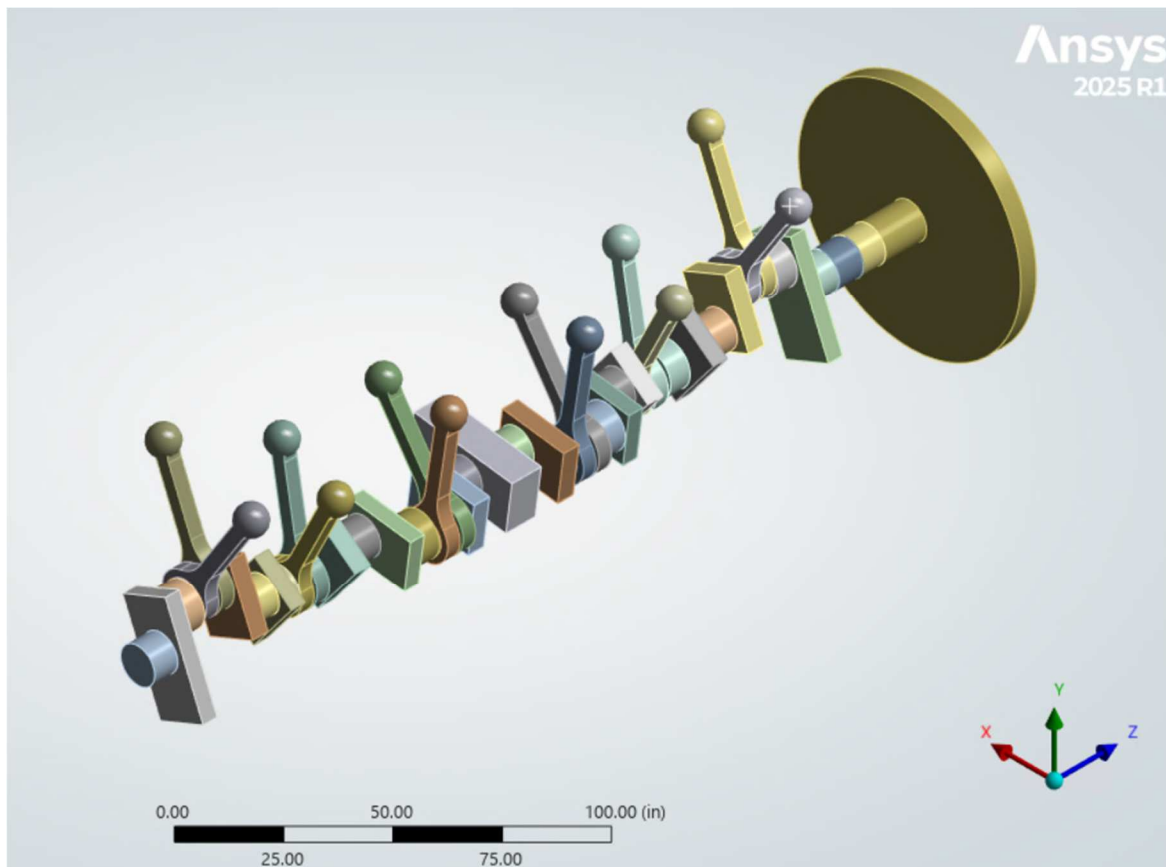


Figure 55: Finite element model of the KVSR 412 with all compressor cylinder components removed.

Since the crankshaft geometry was designed to be balanced with the inclusion of the compressor cylinder components, this simplified model may have higher bearing loads associated with a rotating unbalance. The plots that present the X- and Y-direction components of the load that acts on each main bearing throughout two working cycles from the complete operational model are provided in Appendix F.

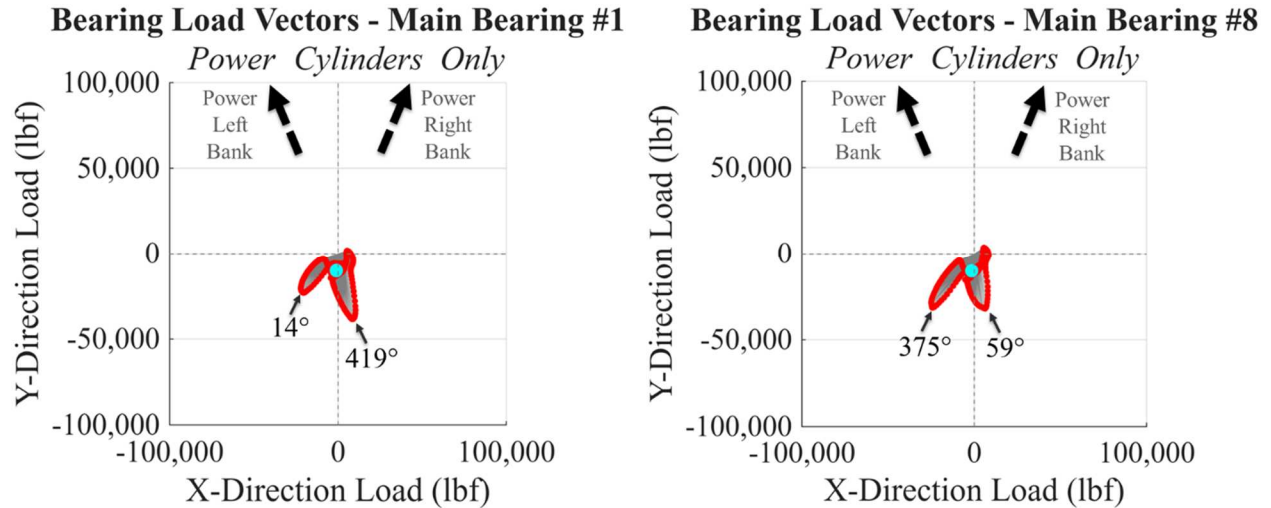


Figure 56: Load vectors for main bearings #1 and #8 from the KVSR 412 finite element model, which only includes the power cylinders.

As was seen in the complete operational model, the load on the two main bearings that neighbor a given cylinder depends on how close the cylinder is to each bearing. Since the #7 power cylinder is closer to main bearing #1 than the #1 power cylinder is, the load related to the peak firing pressure of the #7 power cylinder is higher on main bearing #1. The same relationship is why the load from the #6 power cylinder is highest on main bearing #8.

Figure 57 shows the magnitude and direction of the loads on the #2 and #7 main bearings. Four significant events are noted on the plots for each main bearing. These four events correspond to the peak firing pressure of the four power cylinders, which neighbor each main bearing. High load occurs on main bearing #2 due to the peak firing pressure of the #1 and #7 power cylinders, as well as events at 180° and 491°, which correspond to the peak firing pressure of the #8 and #2

power cylinders. High load also occurs at main bearing #7 when the #6 and #12 power cylinders are at peak firing pressure, as well as events at 134° and 538°, which correspond to the peak firing pressure of the #5 and #11 power cylinders.

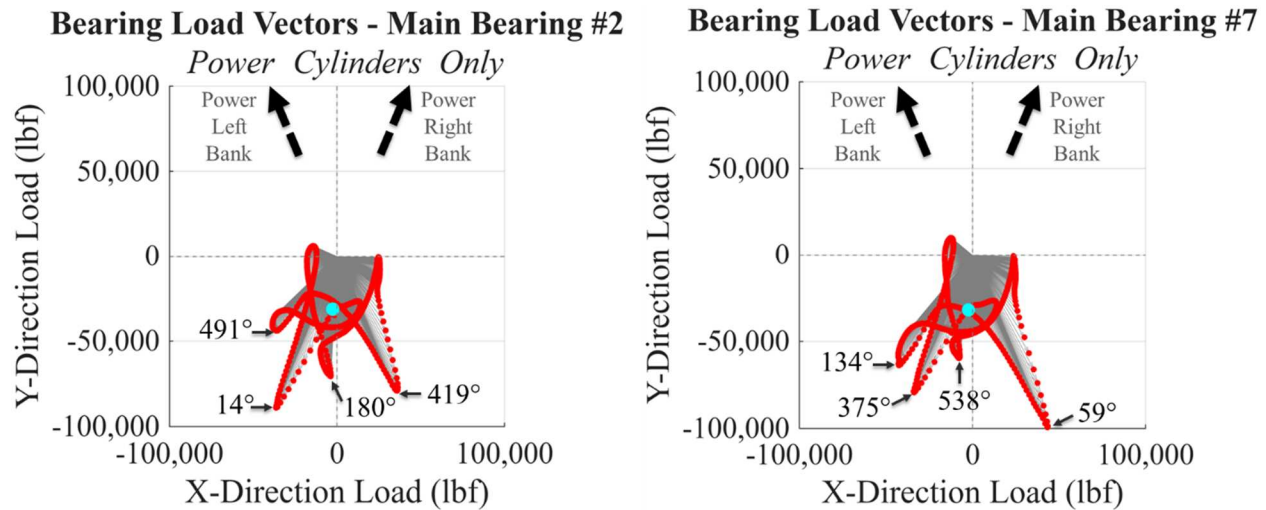


Figure 57: Load vectors for main bearings #2 and #7 from the KVSR 412 finite element model, which only includes the power cylinders.

In the vector plots for the #2 and #7 main bearings, the load from only two of the power cylinders that neighbor each bearing is significant and acts along the axis of their specific cylinder, while the load from the other neighboring cylinders has a much lower magnitude and acts at an angle off of the cylinder axes. While the magnitude of the bearing load due to the peak firing pressure of each power cylinder depends on which cylinder is closest to the bearing, the direction of the bearing load was expected to follow each power cylinder's axis. It was theorized that the events seen in the vector plots that act off of the cylinder axes occur as a result of the inertial load direction at the specific crank angles where each event takes place.

To determine how the inertial loading affects the bearing loads for main bearing #2 and #7, the simplified power cylinder model was analyzed without the gas loading included. Figure 58 shows the vector plots showing the magnitude and direction of the load on main bearing #2 and #7 due to only the inertia of the power cylinder model.

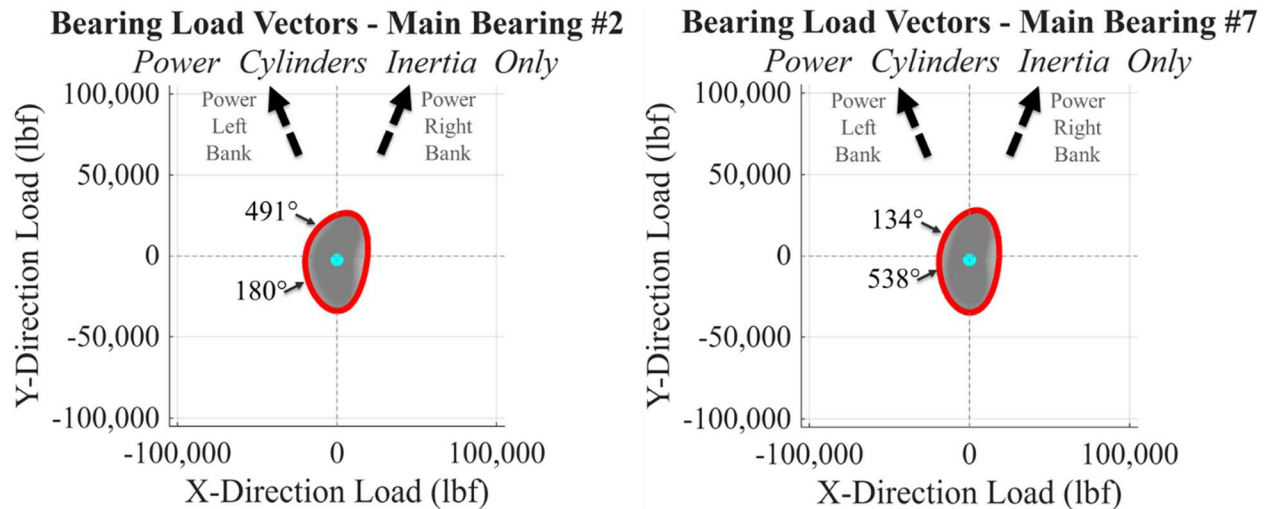


Figure 58: Load vectors for main bearings #2 and #7 from the KVSr 412 finite element model, which only includes the inertia of the simplified power cylinder model.

From the vector plots of the inertial load at these bearings, it can be seen that when the #8 and #11 power cylinders are at peak firing pressure, the inertial load at each bearing acts in the lower-left quadrant of the vector plot. When the #2 and #5 power cylinders are at peak firing pressure, the inertial load at main bearings #2 and #7 acts in the upper-left quadrant. This supports that the inertial load is causing the clockwise shift in the load vectors associated with each of the power cylinders' combustion events relative to the direction of their associated cylinder axes. To better understand this, the X- and Y-direction components of the inertial load at these bearings were subtracted from the results provided in Figure 57. Figure 59 shows the magnitude and direction of the load on main bearings #2 and #7 after subtracting the inertial loads at these bearings. By subtracting the inertial loads at the #2 and #7 main bearings, all events that correspond to the peak firing pressure of each cylinder act along their respective cylinder axis. This shows that even if the engine or cylinder configuration is symmetric, the effect of bearing loading due to the rotating and reciprocating inertia can lead to the main bearing load being unsymmetric over the working cycle.

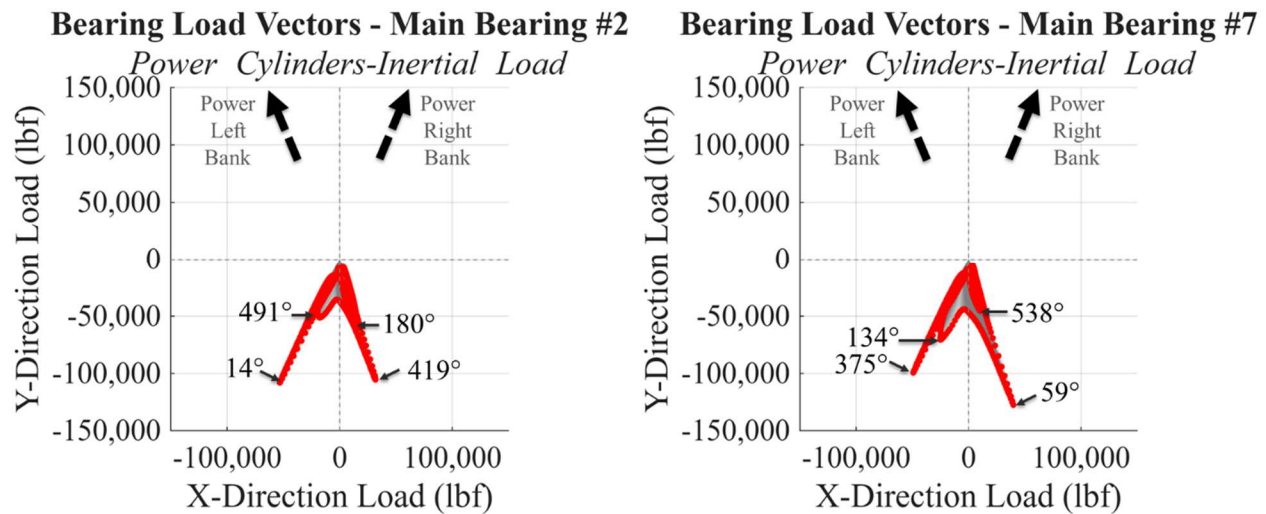


Figure 59: Load vectors for main bearings #2 and #7 from the KVSR 412 finite element model, which only includes the power cylinder and excludes the inertial loads.

A similar clockwise shift in the events associated with the load vectors of two of the four power cylinders, which neighbor the #3 and #6 main bearings are seen. Figure 60 shows the magnitude and direction of the loads on the #3 and #6 main bearings.

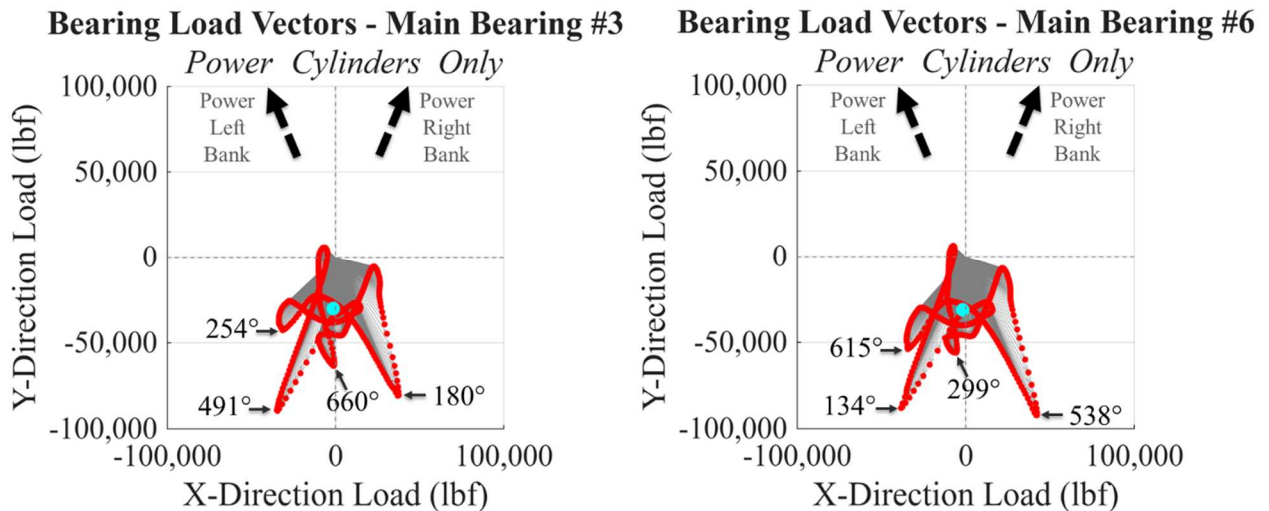


Figure 60: Load vectors for main bearings #3 and #6 from the KVSR 412 finite element model, which only includes the power cylinders.

Like the #2 and #7 main bearings, the events corresponding to the peak firing pressures of the #2 and #8 power cylinders are high on main bearing #3, while the events corresponding to the peak firing pressures of the #5 and #11 power cylinders are high on main bearing 6. Again, the loads

associated with the other neighboring throws to these bearings are lower in magnitude and act off the cylinder axes. The events at 254° and 660° , which act on the #3 main bearing, correspond to the peak firing pressure of the #3 and #9 power cylinders. The events at 299° and 615° , which act on the #6 main bearing, correspond to the peak firing pressure of the #10 and #4 power cylinders. The clockwise shift in the bearing loads associated with the #3, #4, #9, and #10 power cylinders was also found to be related to the inertial loading, which occurs at the same crank angles. Figure 61 shows the magnitude and direction of the #3 and #6 main bearing loads with the inertial load at each main bearing excluded.

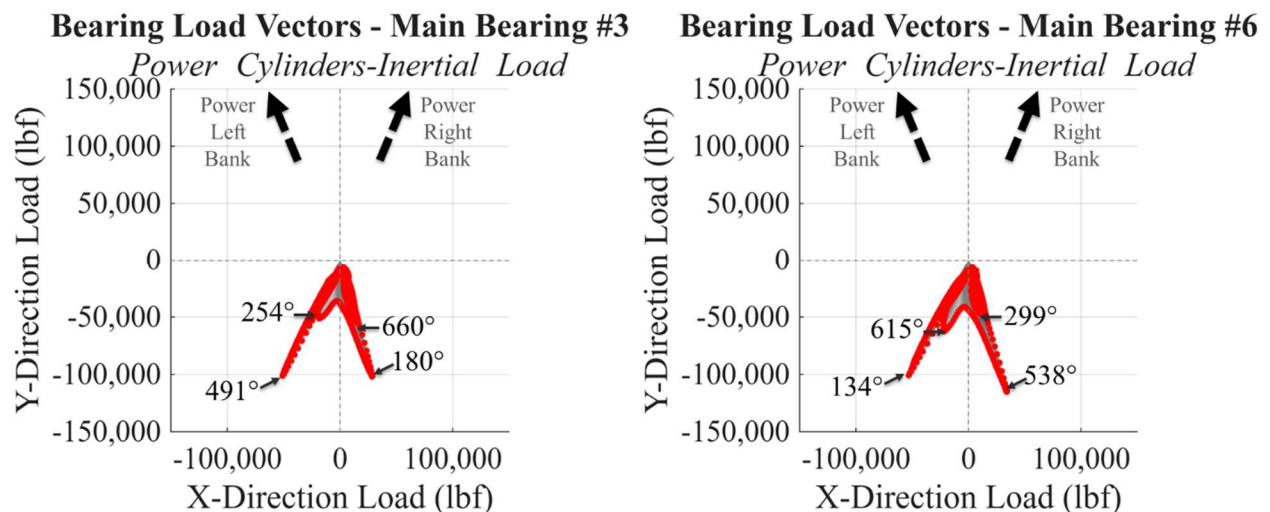


Figure 61: Load vectors for main bearings #3 and #6 from the KVSR 412 finite element model, which only includes the power cylinder and excludes the inertial loads.

After eliminating the effect of the inertial loads at the #3 and #6 main bearings, the events associated with the combustion of the power cylinders that neighbor the main bearings all act along their respective cylinder axes. Just as was shown at the #2 and #7 main bearings, this change in the directionality of the bearing loads after removing the inertial load at each bearing supports that the inertial loads are causing the clockwise shift, which was seen in some of the events. Figure 62 shows the magnitude and direction of the loads that act on the #4 and #5 main bearings.

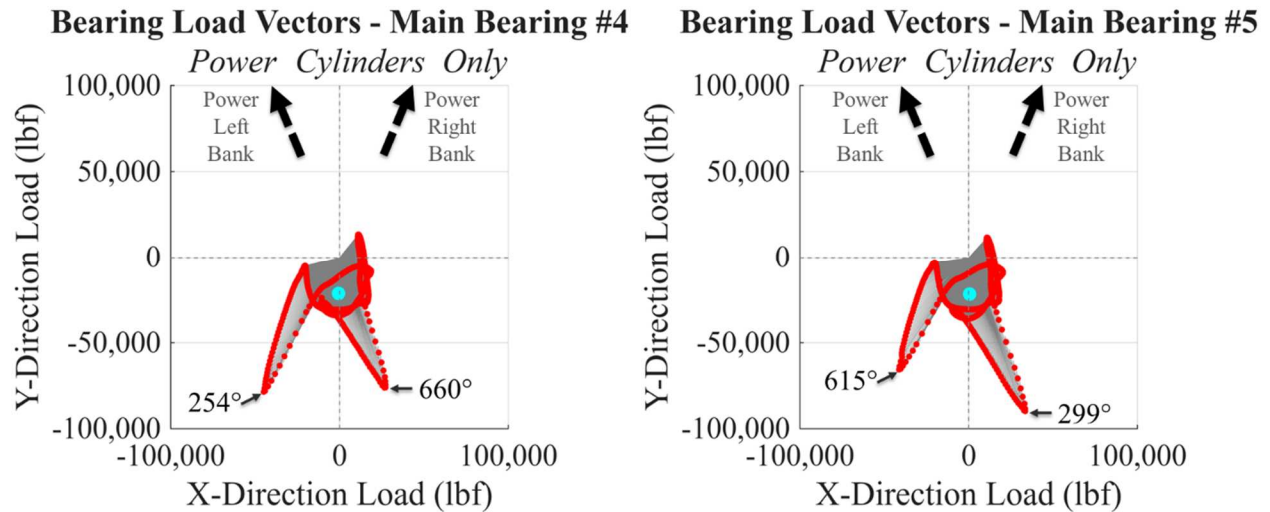


Figure 62: Load vectors for main bearings #4 and #5 from the KVSR 412 finite element model, which only includes the power cylinders.

The significant events that cause high load on the #4 main bearing occur at the crank angles associated with the peak firing pressure of the #3 and #9 power cylinders. The significant events that cause high load on the #5 main bearing occur at the crank angles associated with the peak firing pressure of the #4 and #10 power cylinders. As was seen with the #1 and #8 main bearings, the differences in the events that cause elevated load on the #4 and #5 main bearings are primarily related to which power cylinder is closest to each main bearing. The highest load on main bearing #4 occurs when the #3 power cylinder is at peak firing pressure, since the #3 power cylinder is closer to the bearing than the #9 power cylinder. The load on the #5 main bearing is highest when the #10 power cylinder is at peak firing pressure, since this cylinder is closest to the bearing compared to the #4 power cylinder. As was seen in all other modeled cases, the #9 main bearing supports static load due to the weight of the overhung flywheel and experiences no significant dynamic loading.

3.4.2 Time-Weighted Average Load Vector

Table 5 presents the magnitudes and angles of the TWAL vectors for each main bearing based on the finite element model of the KVSR 412, which only includes the power cylinder

components and their associated gas loads. This simplified model was created to identify differences in the main bearing loading of a separable engine with a symmetrical cylinder arrangement compared to the complete KVSR 412 finite element model.

Table 5: Magnitudes and angles of the time-weighted average load vectors calculated for the model, which only includes the power cylinders.

Main Bearing	Magnitude (lbf)	Angle (°)
1	10,260	265
2	31,410	265
3	30,162	267
4	21,198	269
5	21,658	272
6	31,276	267
7	32,123	266
8	10,211	261
9	6,456	270

A similar symmetry of both the magnitude and angle of the TWAL vectors for bearings that are equidistant from the large counterweight in the center of the crankshaft was also seen in this modeled configuration. However, the load on the bearings is affected by a more significant rotating unbalance in this modeled configuration. This is due to the crankshaft counterweights being sized for the additional mass of the compressor cylinder components. To determine the severity of the rotating unbalance caused by the removal of the compressor cylinder components, the same steps used to determine the balance quality of the complete operational model were repeated. An analysis was completed, which included only the power cylinder components but excluded their associated gas loads. The total combined bearing load based on the inertia of only the power cylinder components was calculated and was used to create the time waveform shown in Figure 63.

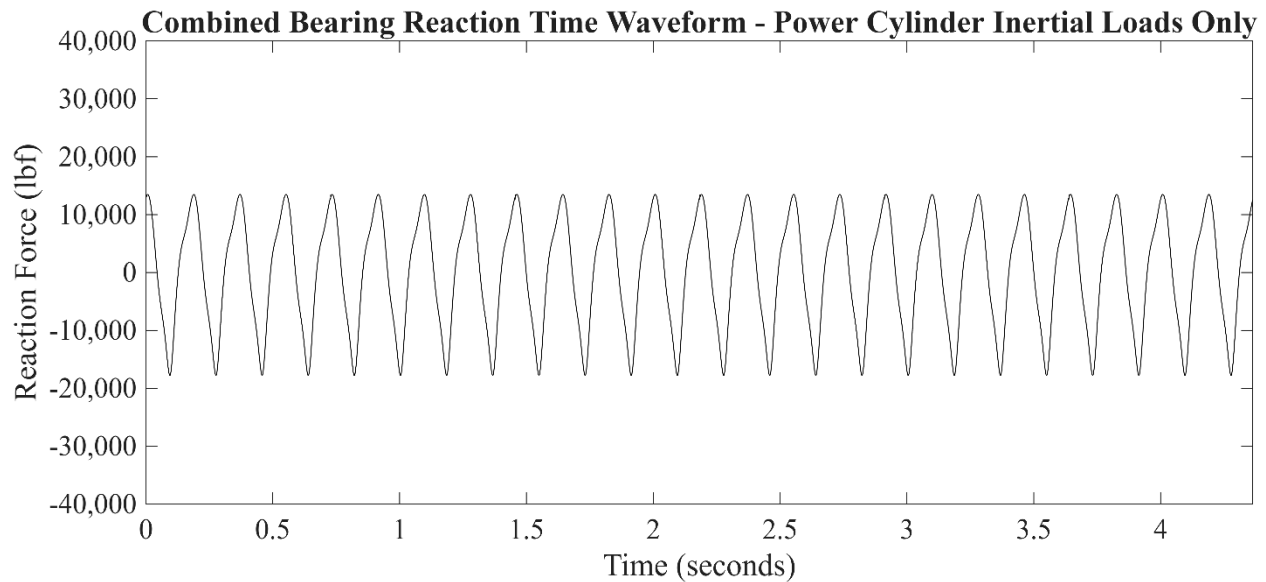


Figure 63: Time waveform of the combined bearing reaction force due to the inertial loads of the KVSr 412 power cylinders.

This time waveform was used to calculate a spectrum to determine the amount of reaction force that occurs at the rotational speed of the crankshaft. Figure 64 shows the spectrum with the total force, which occurs at the rotational speed of the crankshaft, as annotated. From the spectrum, the magnitude of the total combined reaction force occurring at the crankshaft rotational speed was found to be 13,729 lbf_{pk}. This is much higher than the 4,120 lbf_{pk} found from the inertial model, which included the compressor cylinders. The balance quality was calculated based on the magnitude of 13,729 lbf_{pk} and a reduced weight of the rotating components of 18,340 lbf. Using this information, the balance quality of the model, including only the power cylinders, was found to be a G213. This is much higher than the recommended balance quality for a complete engine assembly of a G100 [32]. Given the effect that the inertial loads have on the total main bearing loads and the significant rotating unbalance that exists in this modeled configuration, the TWAL vector magnitudes and angles are likely to be incorrect compared to a V- engine with 12 cylinders, which is properly balanced. While the results provided in this section may not be accurate for

selecting main bearing strain measurement locations, it does provide a reference of the significant effect the compressor cylinder gas loading has on the main bearing loads.

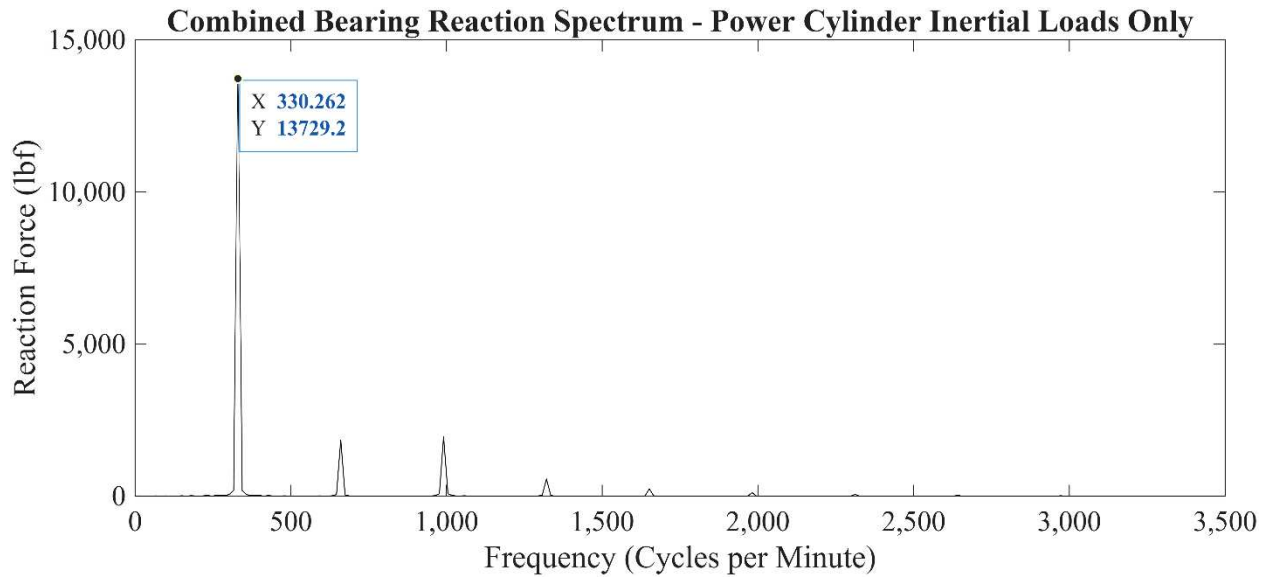


Figure 64: Spectrum calculated from the time waveform in Figure 63.

3.5 KVSr 412 Recommended Measurement Locations

Based on the results from the various configurations of the KVSr 412, the inertial and gas loading due to the compressor cylinders were found to increase the X-direction components of the loads that act upon the main bearings. How significantly the compressor cylinders affect the load on each main bearing depends on how close the compressor cylinder is to the main bearing; the closer the compressor cylinder is to a main bearing, the more it affects the main bearing load. In addition, the timing of events caused by the load due to power cylinders, compressor cylinders, and the inertia of moving components can increase or decrease the load at each main bearing dependent on their directionality. Given the significant dynamic loads that act on the main bearings, the TWAL vector was selected to determine the optimal location about the circumference of the main bearing where the bearing normal strain should be measured. Table 6 shows the recommended measurement location for each main bearing of the KVSr 412 based on the finite

element modeling. Figure 65 shows an example schematic of the measurement locations for all main bearings.

Table 6: Recommended monitoring locations at which dynamic strain should be measured on the KVSR 412 main bearings.

Main Bearing	Recommended Measurement Location (°)
1	290
2	270
3	280
4	280
5	280
6	280
7	270
8	290
9	270

These measurement locations were selected based on the TWAL vectors from the complete operational finite element model. Given the symmetry of the loads that act on bearings which are equidistant from the large counterweight, the angles of the TWAL vectors for like bearings were averaged. To simplify the installation of the dynamic strain sensors, the averaged values were rounded to the nearest 5° increment. These locations were chosen so that the normal strain measured at each main bearing would yield representative data for all loads over a complete operating cycle, while requiring only a single dynamic strain sensor per bearing. However, since these selected measurement locations were developed from the cylinder gas pressures measured from an operating KVSR 412, any deficiencies of the compressor cylinders, such as valve leakage, or any deficiencies of the power cylinders, such as early or late combustion timing or variability

in peak firing pressure between cylinders, could negatively affect these results. The plots that present the TWAL vectors for each main bearing are provided in Appendix G.

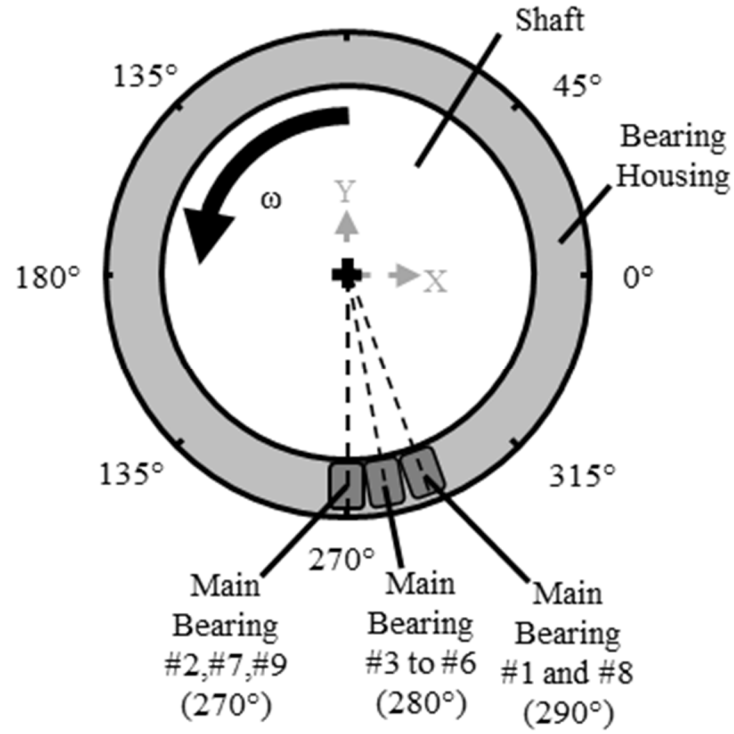


Figure 65: General schematic of the dynamic strain measurement locations for all main bearings.

CHAPTER 4 – FIELD INSTALLATION

4.1 Facility Description

In addition to the finite element modeling of the KVSr 412 to determine expected main bearing loads, experimental data collection of main bearing dynamic strain on an operating KVSr 412 was also set to be included in the research scope. On December 3, 2024, a total of eleven dynamic strain sensors were installed on the main bearing saddles of the same KVSr 412 from which the pressure data used to calculate the modeled gas loads were collected. This installation was performed during a scheduled overhaul of the unit, which operates at Williams Energy's Kastler natural gas compressor station in Northeastern Utah. The unit on which the dynamic strain sensors were installed was one of the five KVSr 412s that operate at this station. Figure 66 shows the KVSr 412 on which the dynamic strain sensors were installed.

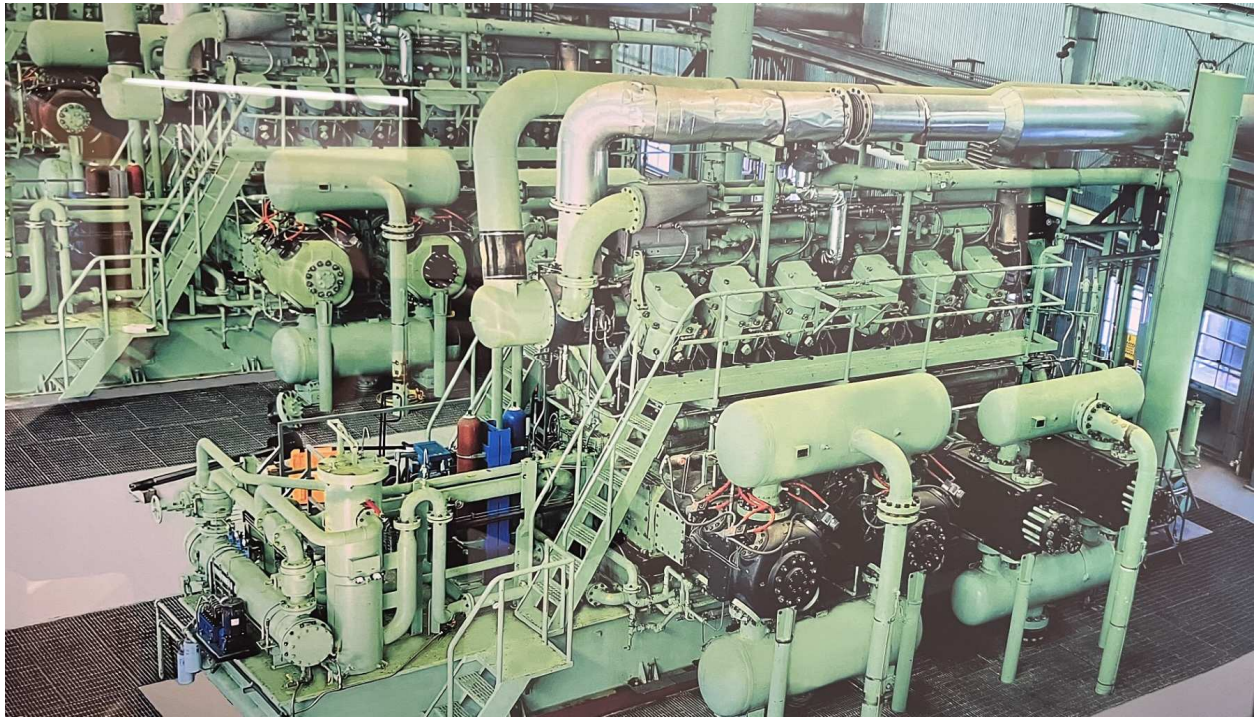


Figure 66: Image of the KVSr 412, which is located at the Kastler station in Northeastern Utah.
(Courtesy of Williams Energy)

The integral engine-compressors, which compress natural gas at the Kastler Station, serve two roles. The first role is the transmission of natural gas along the MountainWest pipeline system, which offers approximately 2,000 miles of pipeline and serves markets in Colorado, Wyoming, and Utah [35]. Additionally, the integral engine-compressors at the Kastler Station are used for natural gas injection into the Clay Basin storage site, which offers 7.8 Bcf of natural gas storage capacity within local underground salt caverns [36]. Figure 67 provides a cross-section of the Clay Basin to provide an overview of the Kastler Station's operation.

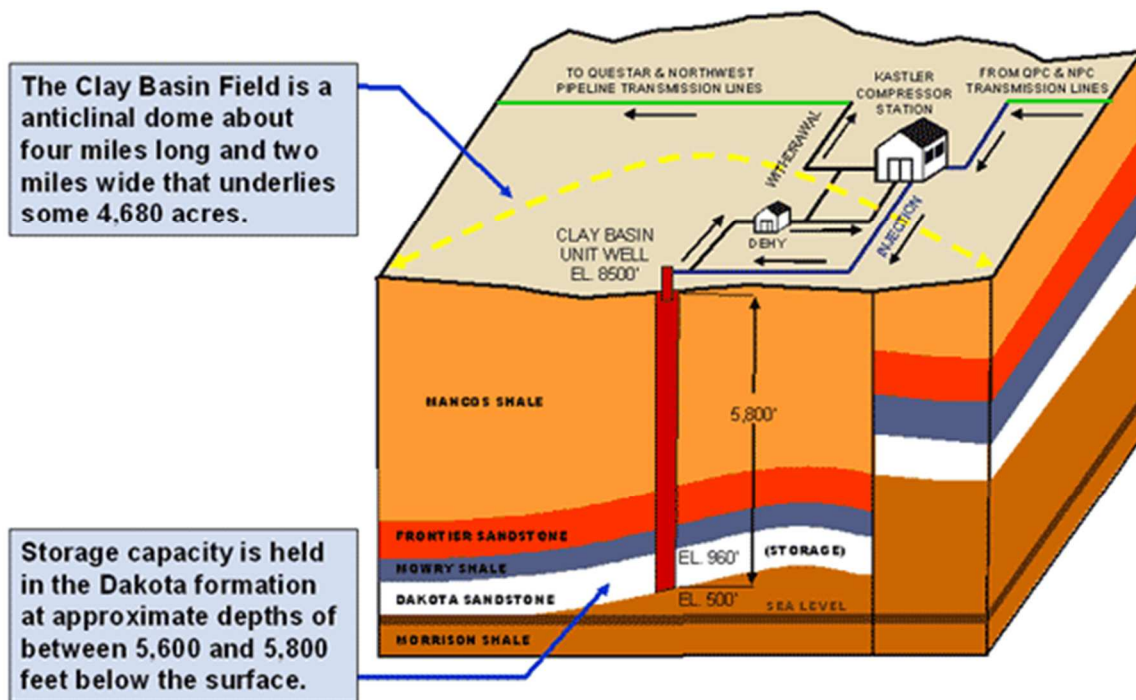


Figure 67: Cross-section of the Clay Basin Storage project [37].

4.2 Sensor Installation Overview

4.2.1 Determination of Mounting Locations

The dynamic strain sensors were installed on the main bearing saddles of the KVSR 412 at the Kastler station before the completion of the finite element modeling of the integral engine-

compressor. For this reason, the locations at which the dynamic strain sensors should be installed had yet to be determined. For this reason, the installation locations on the main bearing saddles were selected based on the recommendations provided in the installation manual for the dynamic strain sensors that were installed. For installation on a machine with plain cylindrical journal bearings like those used on the KVSr 412 at the Kastler station, it was recommended to install a minimum of one sensor directly below the shaft, while a second sensor may be installed 45° from the first sensor following the direction of rotation. Figure 68 shows the potential sensor installation locations based on the recommendations from the sensor manufacturer.

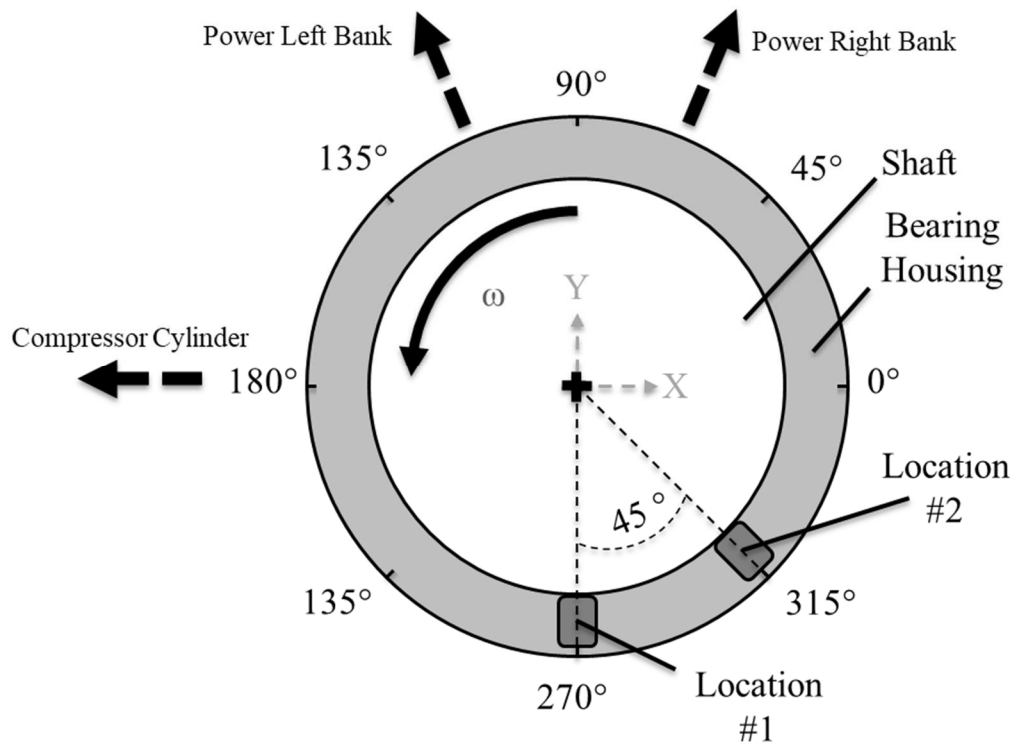


Figure 68: Recommended dynamic strain sensor installation locations.

All nine main bearings were instrumented with at least one dynamic strain sensor. If only one dynamic strain sensor were to be installed on a given bearing, the sensor would be installed below the shaft at Location #1. Only the #2 and #7 main bearings were instrumented with a second sensor

at Location #2. These two bearings were selected based on historical failures that had occurred on each.

4.2.2 Installation Procedure

To directly measure the oscillatory strain on the main bearing saddles, the bottom surface of the dynamic strain sensors was bonded to the main bearing saddles at the selected installation locations using an adhesive. Permabond 820, which is a cyanoacrylate possessing high temperature and chemical resistance, was the adhesive recommended by the sensor manufacturer. Table 7 provides general information about Permabond 820.

Table 7: General material properties for Permabond 820 [38].

Property	Value
Shear Strength (PSI)	2800 - 3800
Minimum Rated Temperature (°F)	-58
Maximum Rated Temperature (°F)	392

Given that the dynamic strain sensors were mounted using adhesive, the surface had to be adequately prepared at the mounting locations, as the strength of the adhesive varies with the level of surface preparation and gap [38]. To prepare the surface, all existing paint was removed with a rotary tool fitted with an abrasive bit. Special care was taken not to compromise the surface flatness, which could reduce the adhesive strength. Once the paint was removed, the surface was thoroughly cleaned using acetone and isopropyl alcohol to remove all surface contaminants. Once the surface was prepared, a small amount of adhesive was applied to the sensor mounting surface. A magnetic fixture tool purchased from the sensor manufacturer was used to apply the appropriate pressure to the sensor to ensure the adhesive formed a thin film while also holding the sensor in place while the adhesive cured.

Figure 69 shows the dynamic strain sensor, which was installed on the #3 main bearing before the cable strain was relieved and routed in the crankcase. This task was performed by the unit operator at a later date.



Figure 69: Dynamic strain sensor installed at Location #1 on the #3 main bearing.

4.3 Monitoring Equipment

For the data acquisition of the measured strain on the main bearing saddles, two different monitoring methodologies were selected. To provide the ability to acquire and log overall amplitudes, each dynamic strain sensor was connected to a separate PCB 682C03 vibration transmitter. The vibration transmitter cards provided continuous sensor excitation power to the dynamic strain sensors, as well as provided a 4-20 mA output, which was scaled to the overall amplitude from each dynamic strain sensor. The overall amplitude from each sensor, which was

transmitted via the 4-20 mA scaled output, was measured using a peak-to-peak signal detection method. The 4-20 mA scaled outputs from the signal conditioner cards were terminated into the Distributed Control System (DCS) for the Kastler Station. This allowed for the long-term logging of the overall amplitudes measured from each dynamic strain sensor.

In addition to the 4-20 mA scaled output, the signal conditioner cards offer a raw voltage output, which can be utilized to measure the raw time domain signal from each sensor. The raw voltage output allows the analyst, who monitors the condition of the KVSr 412 during quarterly performance analyses, to measure the time domain signal from each dynamic strain sensor using a Windrock 6400. All data collected during the quarterly performance analysis can be referenced to the rotational position of the crankshaft using an encoder. This will allow the measured time domain strain data to be resolved to the crankshaft rotation and compared with other acquired data, such as the pressure in each cylinder over the measured working cycle.

4.4 Goal of Operational Data

The goal of the operational data collected from the installed strain sensors is to develop expected strain amplitudes and time domain data to characterize the typical operation of the KVSr 412. However, given that the Kastler Station is utilized for the injection of natural gas into the Clay Basin as well as the transmission of natural gas along the MountainWest pipeline, the integral engine-compressors at this site have various operational load steps based on current needs. The long-term acquisition of overall amplitudes from the installed dynamic strain sensors will provide a quantitative measure for how the load on each main bearing changes under the various load steps. By trending the overall amplitudes and cataloging the typical overall strain amplitude for each specified load step, the overall amplitudes can be used as an alert of a fault in the machine, as well as to better understand the health of the main bearings. Additionally, the ability to measure the time

domain signal from each sensor allows for further diagnostics of a potential problem. The ability to compare the collected strain data to the measured pressure in each cylinder will provide key insights into how faults in a power cylinder or compressor cylinder, such as leakage, affect the load on the main bearings.

CHAPTER 5 – CONCLUSION

Reciprocating equipment, including separable engines and compressors, as well as integral engine-compressors serve a critical role in moving natural gas through the over 305,000 miles of natural gas transmission pipelines in North America. Although the fleet of integral engine-compressors used throughout the gas transmission pipeline is aging, these machines are the backbone of the industry and are not feasible to replace en masse. Given this, it is more important than ever to ensure that any potential equipment malfunctions, if left undetected, do not escalate into major failures that could result in unplanned downtime, substantial maintenance expenses, and safety risks. This is best accomplished through a preventive maintenance program that relies on condition-based data to guide decision-making. While condition-based data is often collected on reciprocating equipment, current monitoring technologies, including the measurement of bearing temperature, shaft displacement using proximity probes, and frame acceleration, all have limited effectiveness in monitoring the health of the main bearings of reciprocating equipment.

This research was the first step to evaluate a new condition monitoring technology, which aims to monitor the health of the main bearings through the measurement of dynamic strain on the main bearing saddles. To determine the most effective locations for measuring dynamic strain on the main bearings of a reciprocating machine, a finite element model of the rotating assembly of an Ingersoll Rand KVSR 412 was created in ANSYS Mechanical. The gas loads that were applied to the small-end bearing of each connecting rod were calculated from cylinder pressure data acquired from a KVSR 412 operating at the Kastler compressor station in Northeastern Utah. Three different model configurations were analyzed to determine the most effective monitoring locations based on the inertial and gas loads and their effect on the main bearing loads.

5.1 Key Findings

5.1.1 Results of Modeled Configurations

The modeled test cases for the KVSr 412 showed significant variability in the magnitude of the load on each main bearing, which was related to the number of power and compressor cylinders adjacent to each main bearing. For example, main bearings #1 and #8 support much lower load than main bearings #3 and #6 when averaged over the working cycle. This is due to main bearings #3 and #6 neighboring more power cylinders than main bearings #1 and #8. Since the main bearings of reciprocating equipment support primarily dynamic loading with significant magnitudes, the TWAL load vector was used to quantify the average magnitude and direction of the main bearing load over the working cycle of 720° of crankshaft rotation.

By comparing the TWAL vectors for each main bearing, it was found that the main bearings that are close to a compressor cylinder and support the load from the fewest power cylinders had the highest shift in the angle of the TWAL vector in the direction of rotation from the bottom dead center of the bearing. The results of the complete operational model and inertial model supported that the increase in the angle of the TWAL vector seen at bearings that neighbor a compressor cylinder was caused by the gas loading of the compressor cylinder. This was found to be related to the compressor cylinder gas load acting in the direction of the crankshaft more consistently throughout each working cycle due to the piston area taken up by the piston rod on the crank-end side of the compressor cylinders. As only some main bearings neighbor these compressor cylinders, the locations at which dynamic strain is measured must be adjusted at each main bearing, to ensure the strain is measured at the best location for each bearing's given loading.

5.1.2 Recommended Strain Monitoring Locations

Based on the TWAL vectors calculated from the X- and Y-direction components of the bearing reactions from the complete operational model, the recommended locations at which dynamic strain data should be measured for the health monitoring of the KVSr 412 main bearings were selected. Given the similarities between the magnitudes and angles of the TWAL vectors for the main bearings that are equidistant from the counterweight that is located between main bearings #4 and #5, the recommended locations were consistent with similar bearings. The monitoring location for each given bearing was calculated by taking the average of the angle of the TWAL vectors for main bearings, which are equidistant from the central counterweight. This value was then rounded to the nearest 5° increment to simplify the installation in the field. This angle represents the location about the circumference of each bearing at which the dynamic strain sensor should be installed. From the finite element modeling of the complete KVSr 412 rotating assembly, it is recommended to install the dynamic strain sensors at:

- 270° on the #2, #7, and #9 main bearings.
- 280° on the #3, #4, #5, and #6 main bearings.
- 290° on the #1 and #8 main bearings.

5.2 Future Work

5.2.1 Rotor Dynamic Modeling

The finite element modeling performed in this research was only concerned with determining the magnitude and the direction of the load on each main bearing and how it varies throughout the working cycle. As such, results from the finite element model of the KVSr 412 exclude the rotor dynamics associated with the plain cylindrical journal bearings that support the rotating crankshaft. The determination of the locations at which the strain of the main bearing saddles should be

measured assumes that the high-pressure oil wedge, which supports the crankshaft, will be located at the circumferential location where high bearing load exists. However, given the large variability in the magnitude and direction of the main bearing loads seen throughout the working cycle, further modeling, which includes the rotor dynamics of the plain cylindrical journal bearings and their effect on the strain of the main bearing saddle, is recommended.

5.2.2 Long-Term Evaluation of Field-Installed Dynamic Strain Sensor

The original scope of this research included the long-term evaluation of the dynamic strain sensors that were installed on the main bearings of a KVSR 412 located in Northeastern Utah. These sensors were installed during a scheduled overhaul of the integral engine-compressor. However, due to unexpected delays during this overhaul, the unit was not operational during the duration of this research. The operational data collection, which was planned, included the long-term collection of overall strain amplitudes from the sensors installed on the main bearings of the KVSR 412. The collection of overall strain amplitudes at each bearing would allow changes in the main bearing strain to be monitored and catalogued in the DCS for the Kastler compressor station. Additionally, the hardware that was used for this installation would also allow for the collection of strain time waveforms at each main bearing. These time waveforms could also be resolved to the crankshaft rotation, which would provide further diagnostics by comparing the strain data to the cylinder pressure data collected during the performance analyses completed on this unit. This operational data will be invaluable for evaluating the use of dynamic strain measurement to monitor the long-term health of the main bearings of reciprocating machinery. Given this, the operational data collection and evaluation are recommended to supplement the results from this research.

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APPENDIX A

Table 8 in this appendix provides the masses and dimensions of various components of the KVSr 412. These masses and dimensions were used to create the finite element model of the KVSr 412 as described in Section 2.2.4.

Table 8: Masses and sizes of various components.

Component	Value
Angle Between Power Cylinders (°)	45.00
Power Cylinder Bore (inch)	16.25
Power Cylinder Connecting Rod Length (inch)	36.25
Power Cylinder Stroke (inch)	18.00
Power Cylinder Crank Pin Diameter (inch)	9.99
Power Cylinder Crank Pin Length (inch)	10.00
Compressor Cylinder 1&2 Bore (inch)	12.54
Compressor Cylinder 3&4 Bore (inch)	8.26
Compressor Piston Rod Diameter (inch)	4.00
Compressor Cylinder Stroke (inch)	15.00
Compressor Cylinder Connecting Rod Length (inch)	39.00
Compressor Crank Pin Diameter (inch)	13.00
Compressor Crank Pin Length (inch)	6.25
Main Bearing Diameter (inch)	10.99
Main Bearing Length (inch)	7.25
Crankshaft Mass (lbm)	22,577.00
Compressor Piston and Piston Rod Combined Mass (lbm)	860.00
Compressor Crosshead Mass (lbm)	680.00
Power Piston Mass (lbm)	420.00

APPENDIX B

The following appendix provides images of the X- and Y-direction components of the load on each main bearing from the complete operational model. The plots are presented over one working cycle of the KVSr 412. The X- and Y-direction components of the load on each main bearing were used to develop the vector plots presented in Section 3.1.1.

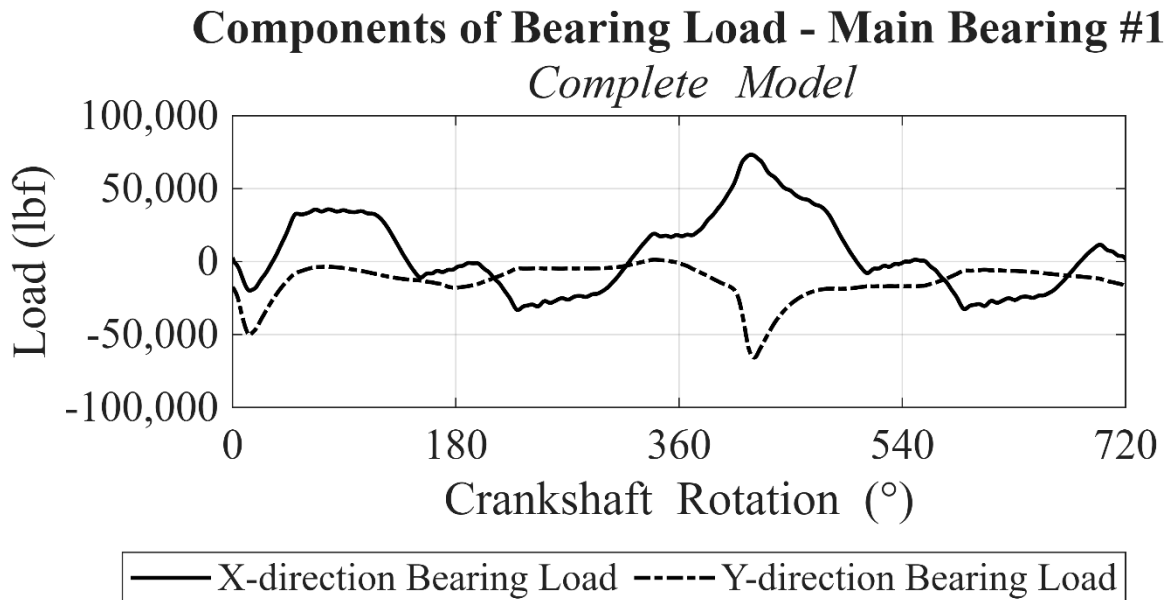


Figure 70: Components of the load on main bearing #1 for the complete operational model.

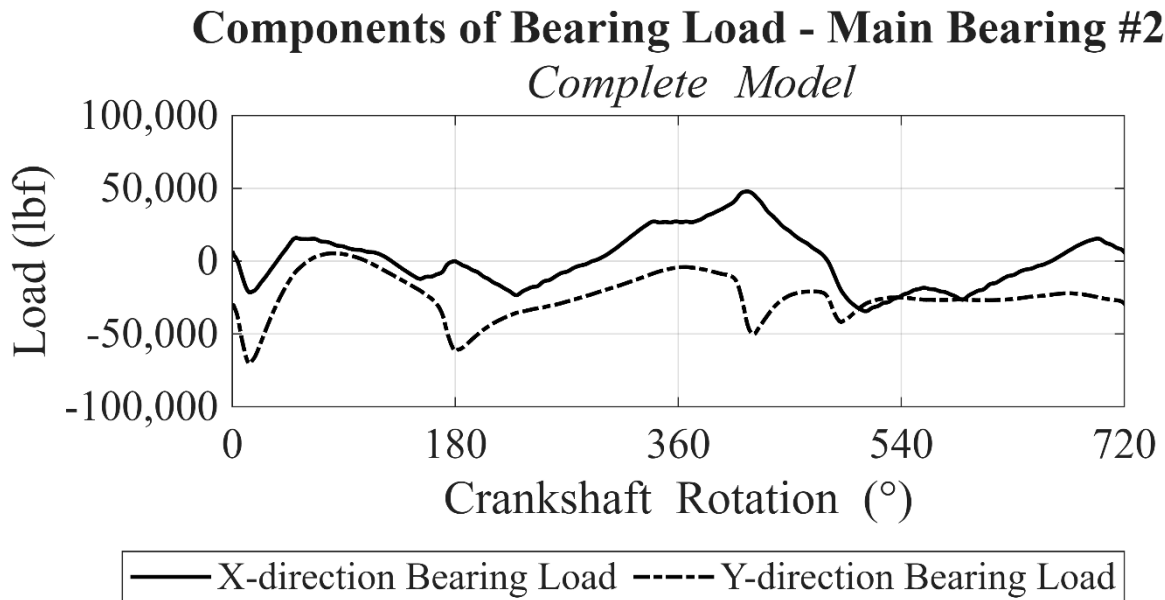
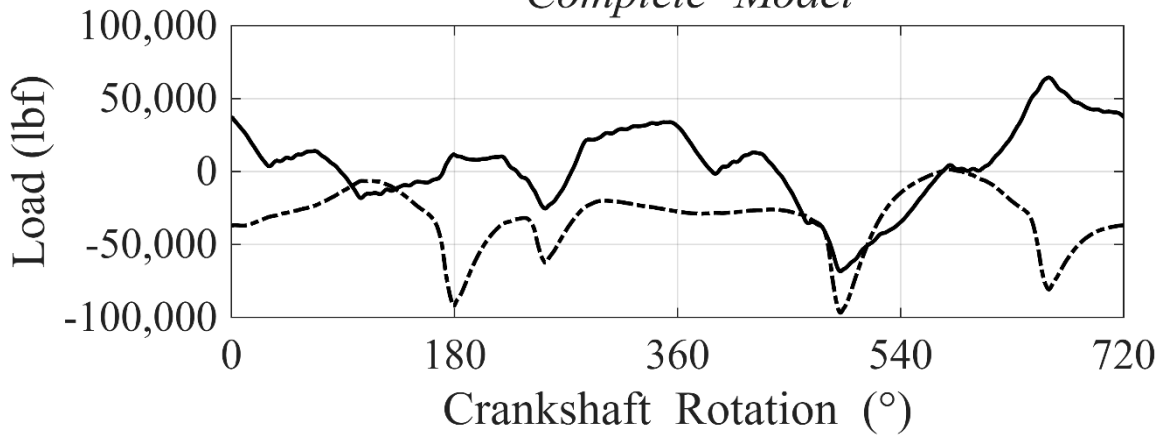


Figure 71: Components of the load on main bearing #2 for the complete operational model.

Components of Bearing Load - Main Bearing #3

Complete Model

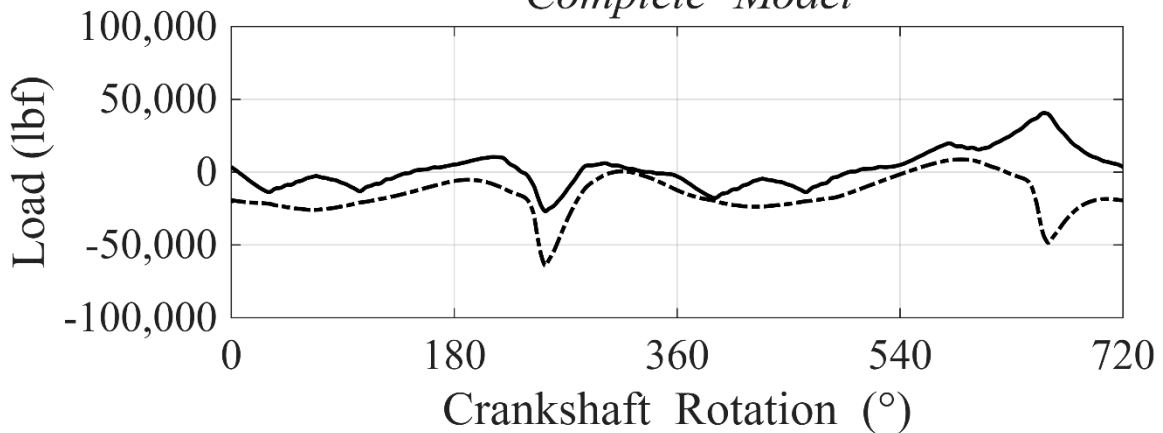


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 72: Components of the load on main bearing #3 for the complete operational model.

Components of Bearing Load - Main Bearing #4

Complete Model

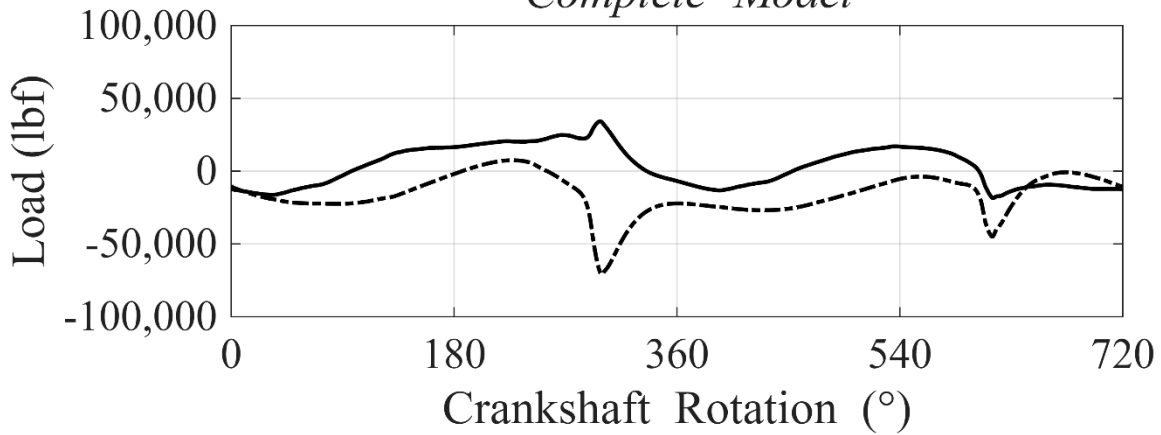


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 73: Components of the load on main bearing #4 for the complete operational model.

Components of Bearing Load - Main Bearing #5

Complete Model

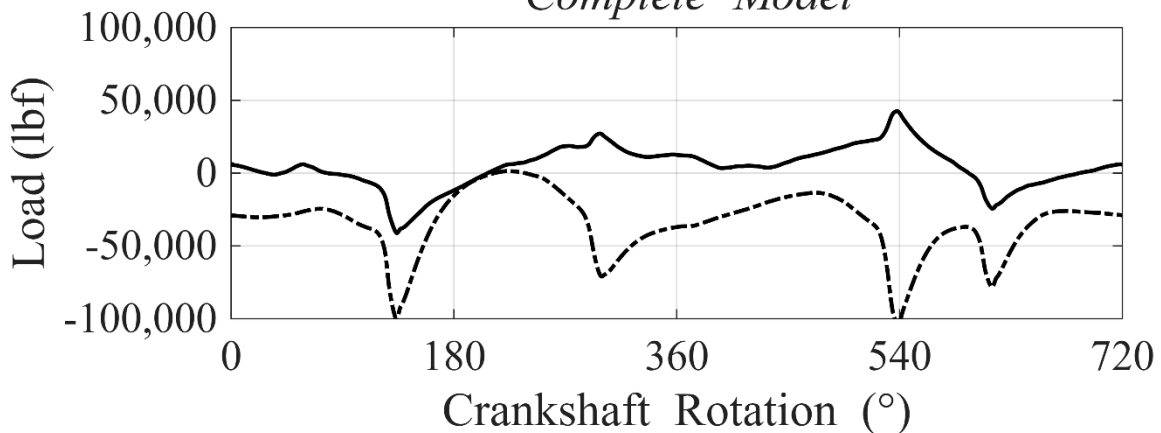


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 74: Components of the load on main bearing #5 for the complete operational model.

Components of Bearing Load - Main Bearing #6

Complete Model

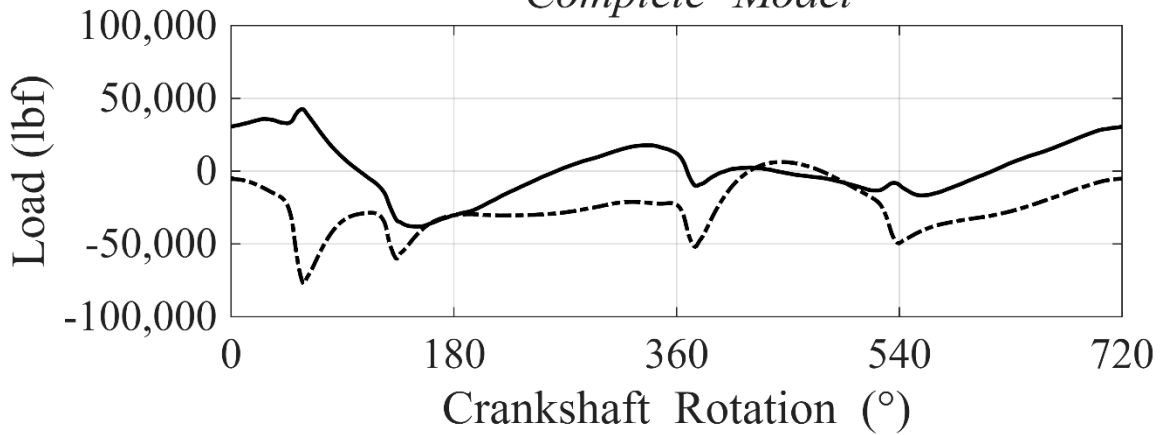


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 75: Components of the load on main bearing #6 for the complete operational model.

Components of Bearing Load - Main Bearing #7

Complete Model

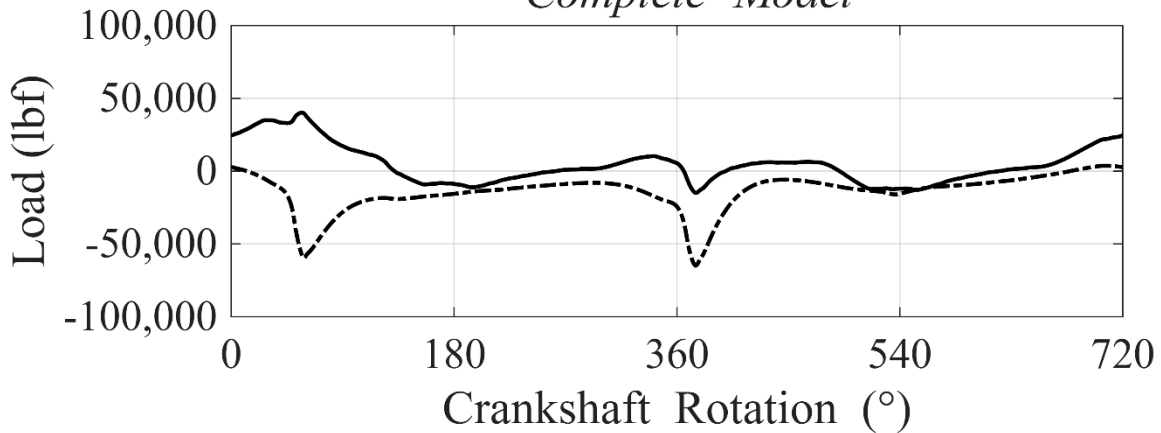


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 76: Components of the load on main bearing #7 for the complete operational model.

Components of Bearing Load - Main Bearing #8

Complete Model



— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 77: Components of the load on main bearing #8 for the complete operational model.

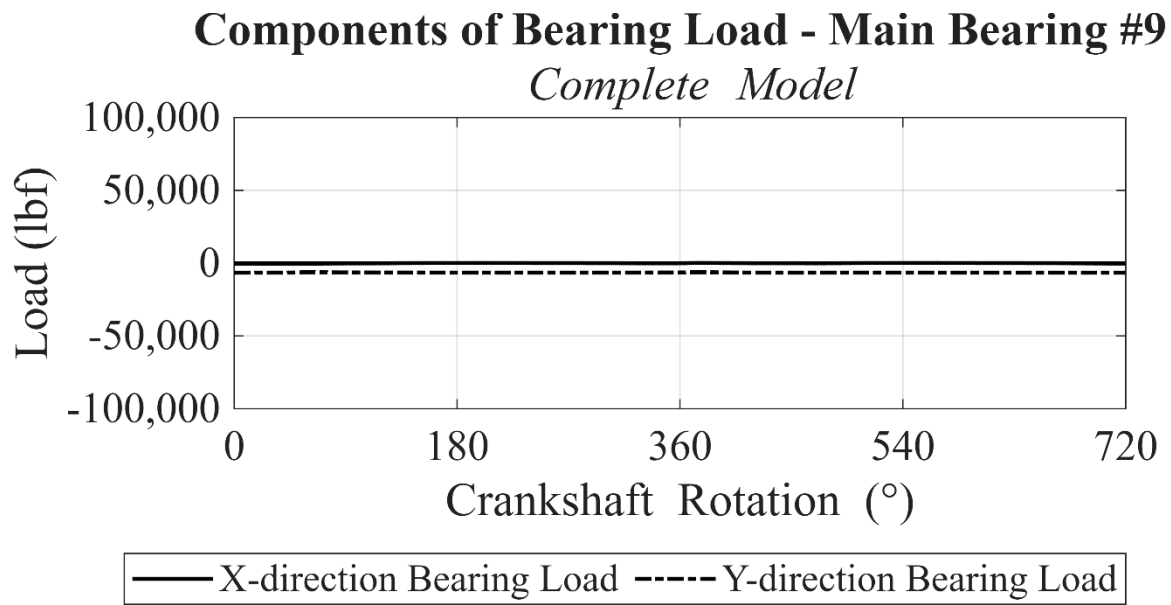


Figure 78: Components of the load on main bearing #9 for the complete operational model.

APPENDIX C

The following appendix provides images of the TWAL vectors for each main bearing for the complete operational model. The magnitude and angle associated with each main bearing's TWAL vector were originally presented in Section 3.1.2.

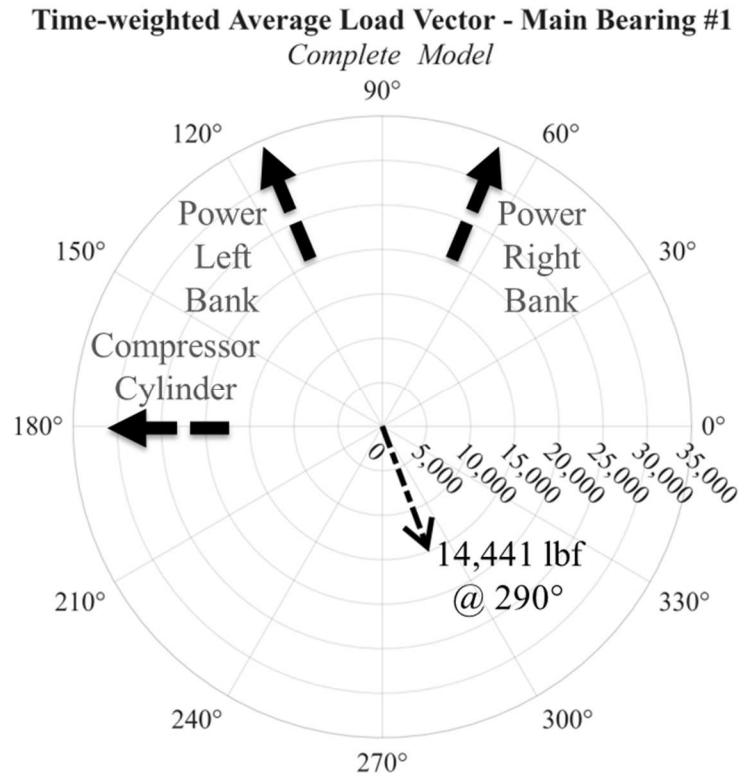


Figure 79: TWAL vector for main bearing #1 for the complete operational model.

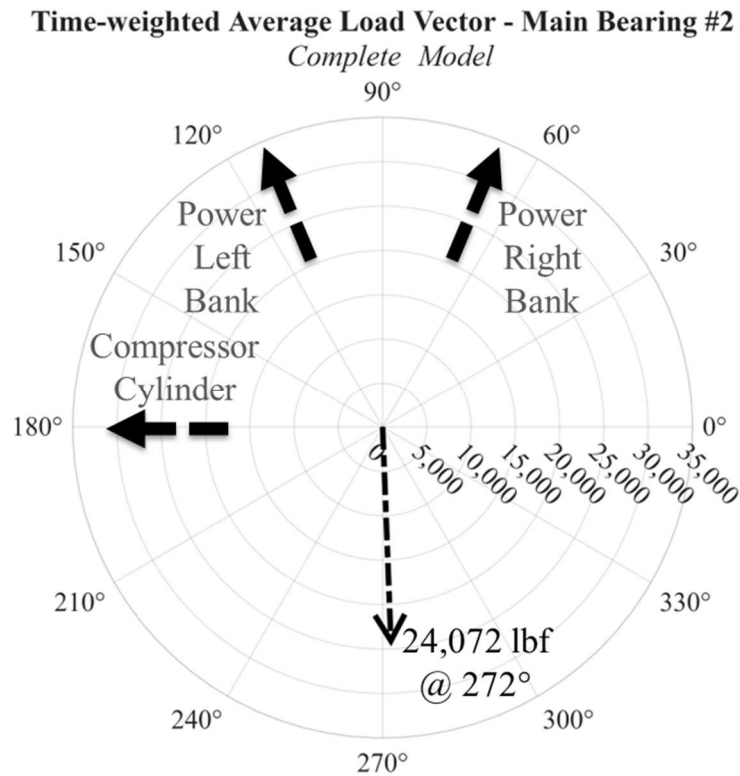


Figure 80: TWAL vector for main bearing #2 for the complete operational model.

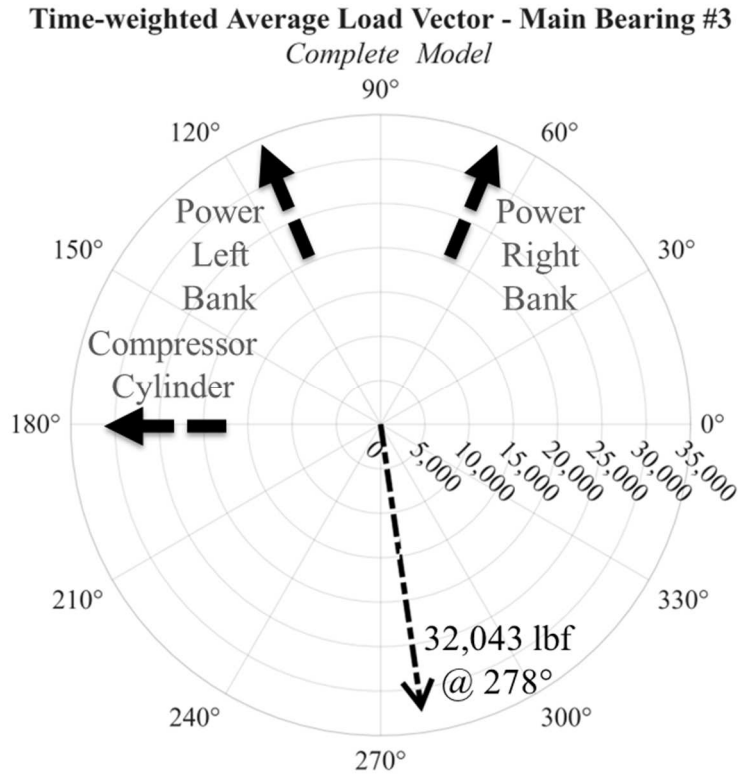


Figure 81: TWAL vector for main bearing #3 for the complete operational model.

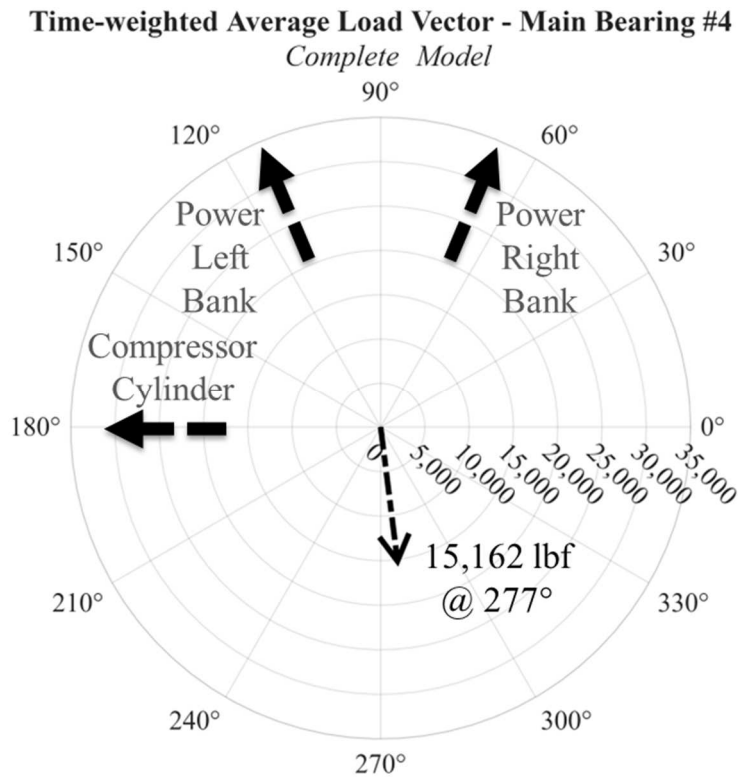


Figure 82: TWAL vector for main bearing #4 for the complete operational model.

Time-weighted Average Load Vector - Main Bearing #5

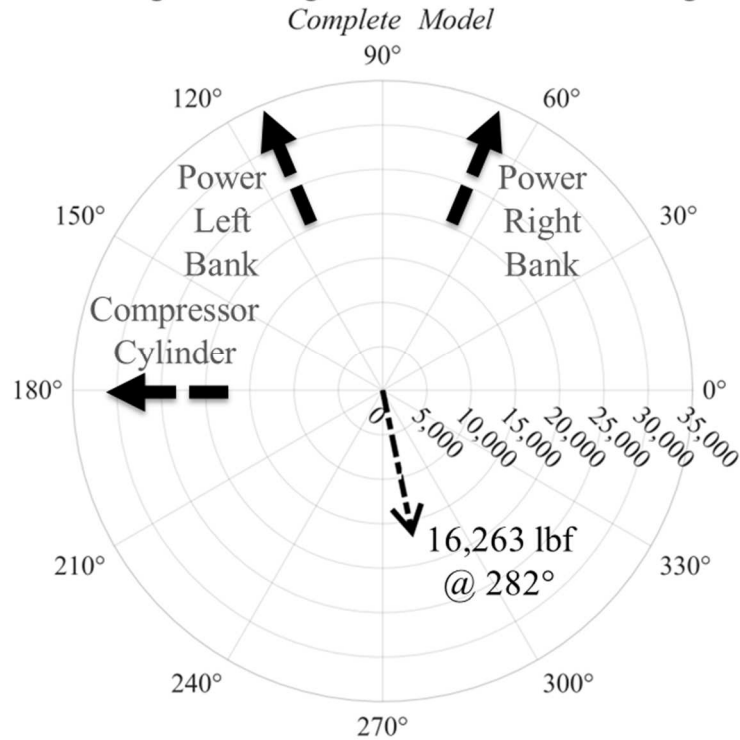


Figure 83: TWAL vector for main bearing #5 for the complete operational model.

Time-weighted Average Load Vector - Main Bearing #6

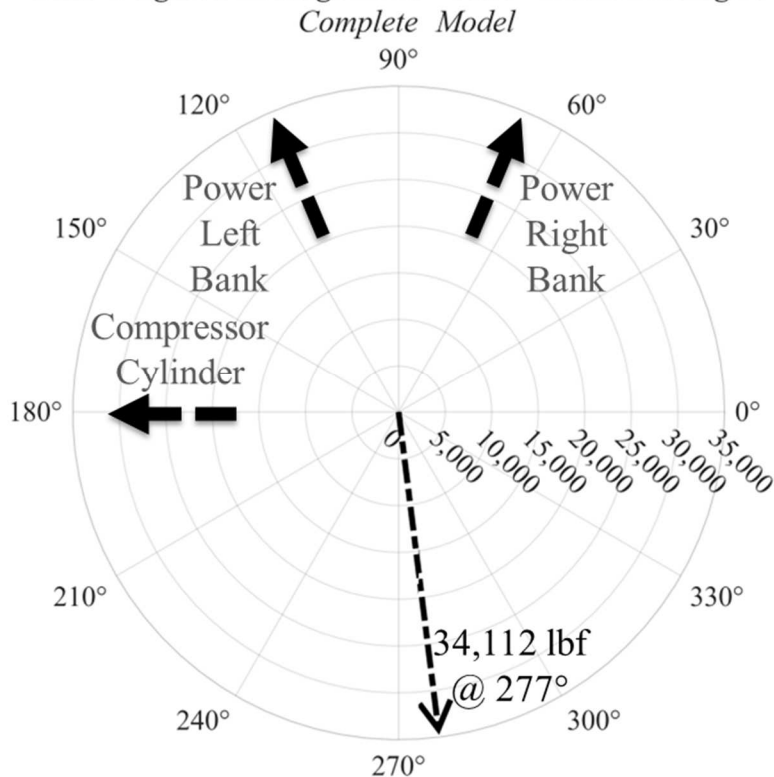


Figure 84: TWAL vector for main bearing #6 for the complete operational model.

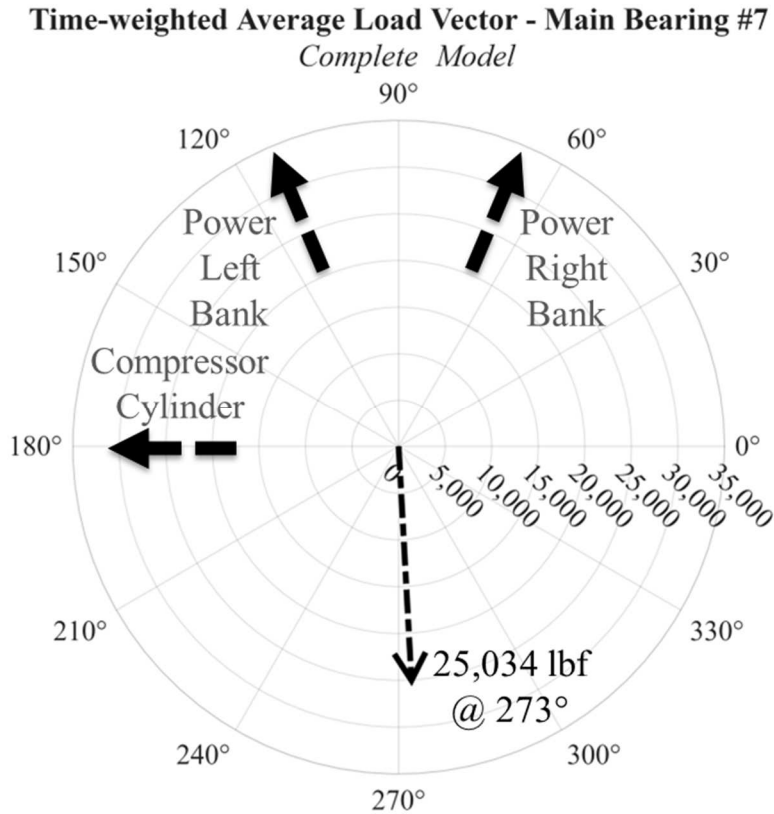


Figure 85: TWAL vector for main bearing #7 for the complete operational model.

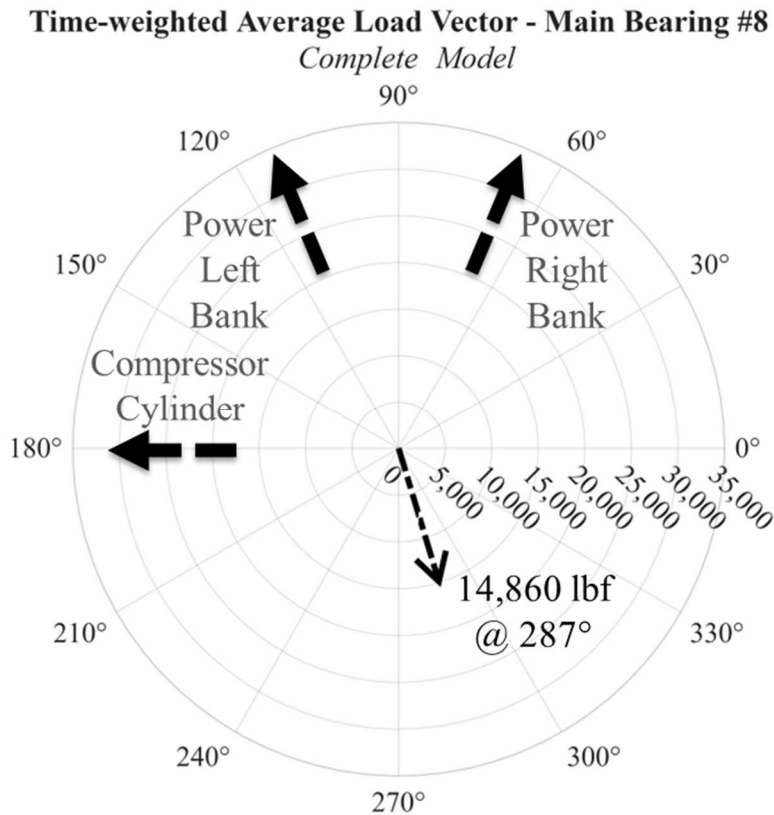


Figure 86: TWAL vector for main bearing #8 for the complete operational model.

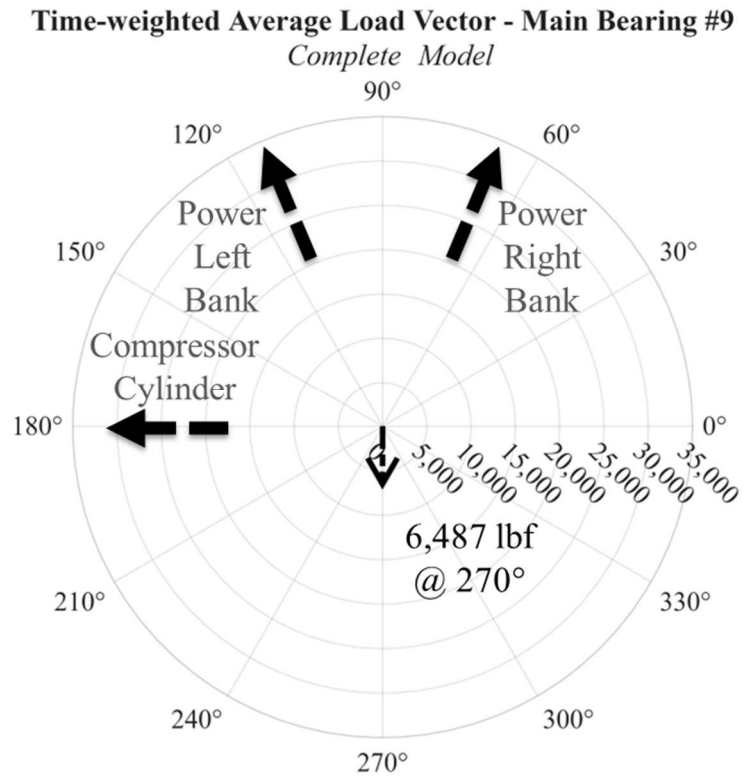


Figure 87: TWAL vector for main bearing #9 for the complete operational model.

APPENDIX D

The following appendix provides images of the X- and Y-direction components of the load on each main bearing for the inertial model. The plots are presented over one working cycle of the KVSr 412. The X- and Y-direction components of the load on each main bearing were used to develop the vector plots presented in Section 3.2.1.

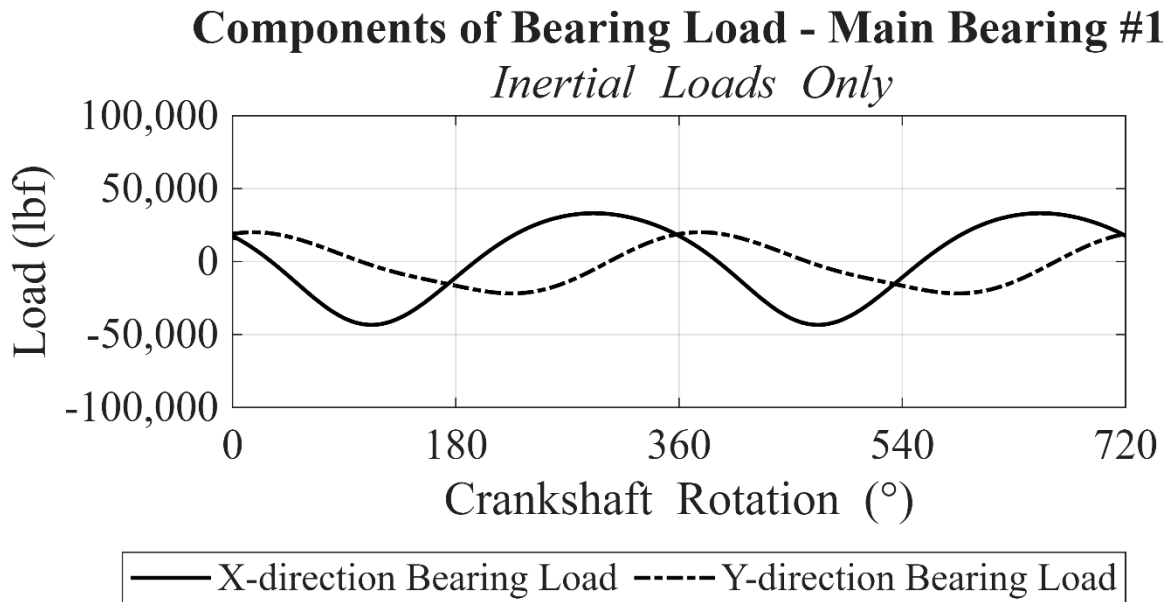


Figure 88: Components of the load on main bearing #1 due to the complete assembly inertia.

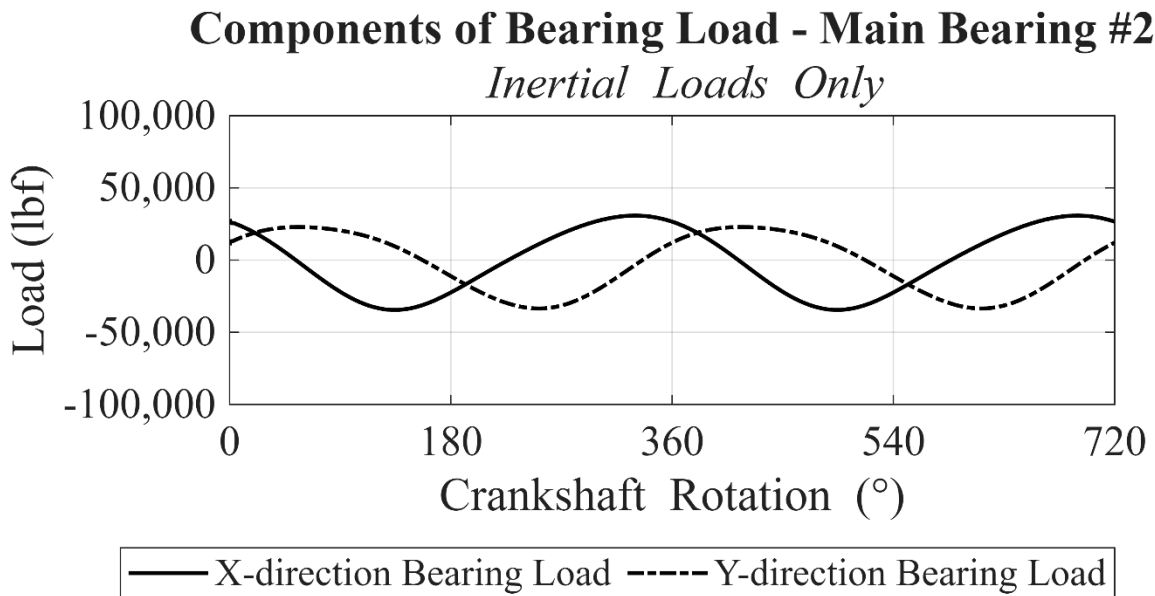
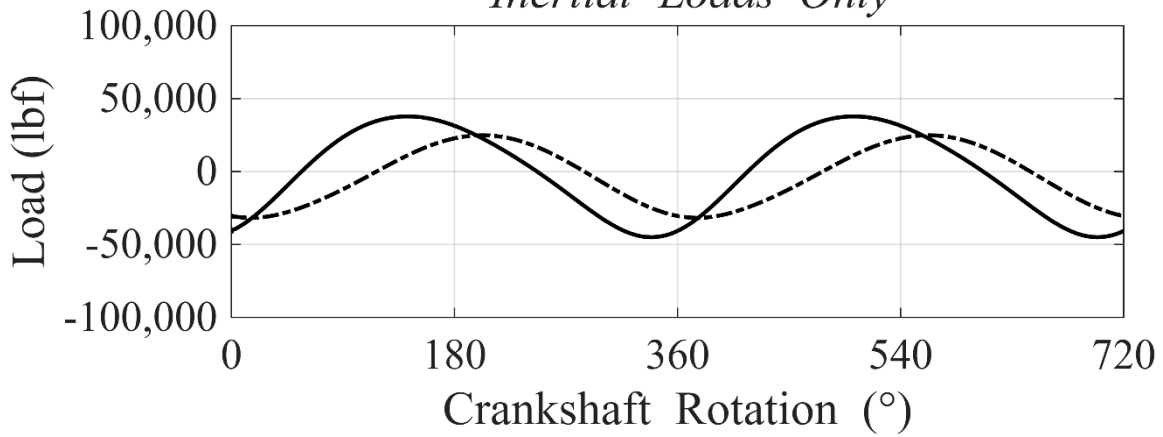


Figure 89: Components of the load on main bearing #2 due to the complete assembly inertia.

Components of Bearing Load - Main Bearing #3

Inertial Loads Only

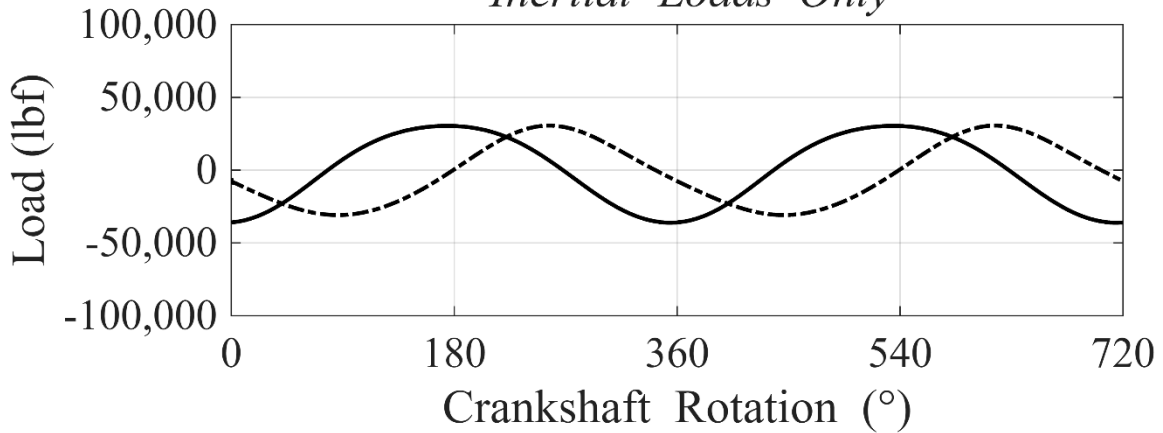


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 90: Components of the load on main bearing #2 due to the complete assembly inertia.

Components of Bearing Load - Main Bearing #4

Inertial Loads Only

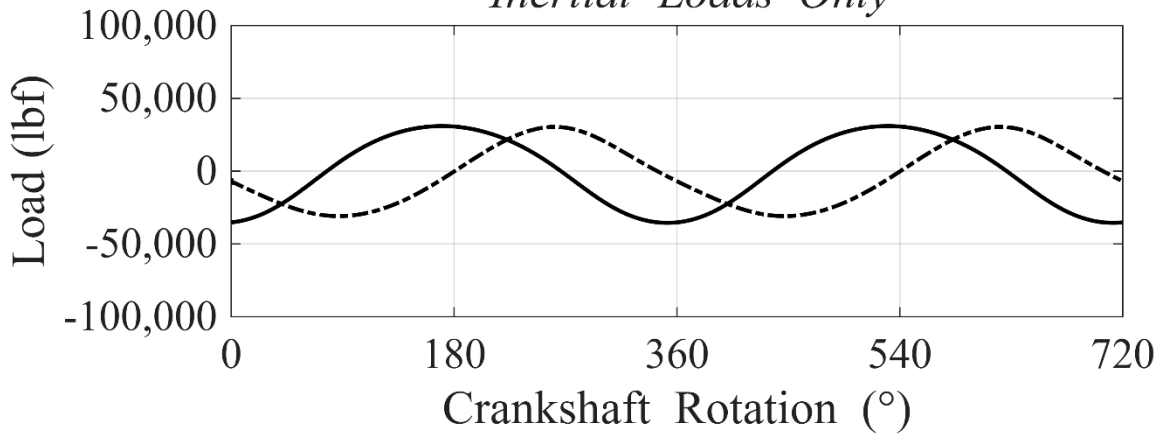


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 91: Components of the load on main bearing #3 due to the complete assembly inertia.

Components of Bearing Load - Main Bearing #5

Inertial Loads Only

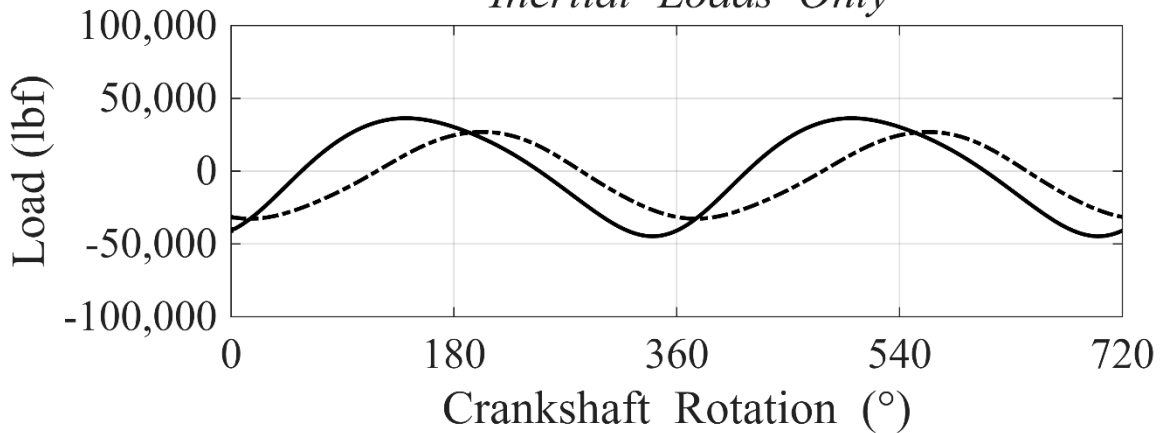


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 92: Components of the load on main bearing #5 due to the complete assembly inertia.

Components of Bearing Load - Main Bearing #6

Inertial Loads Only

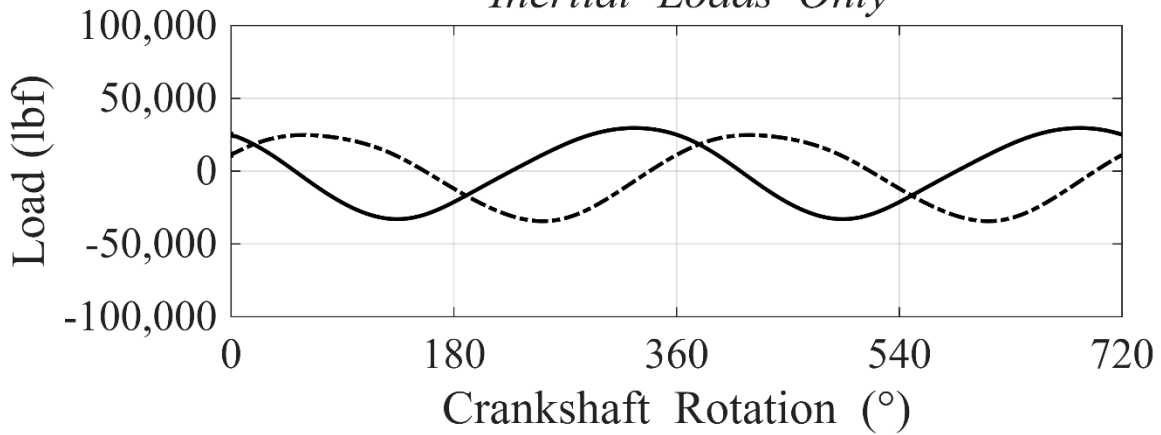


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 93: Components of the load on main bearing #6 due to the complete assembly inertia.

Components of Bearing Load - Main Bearing #7

Inertial Loads Only

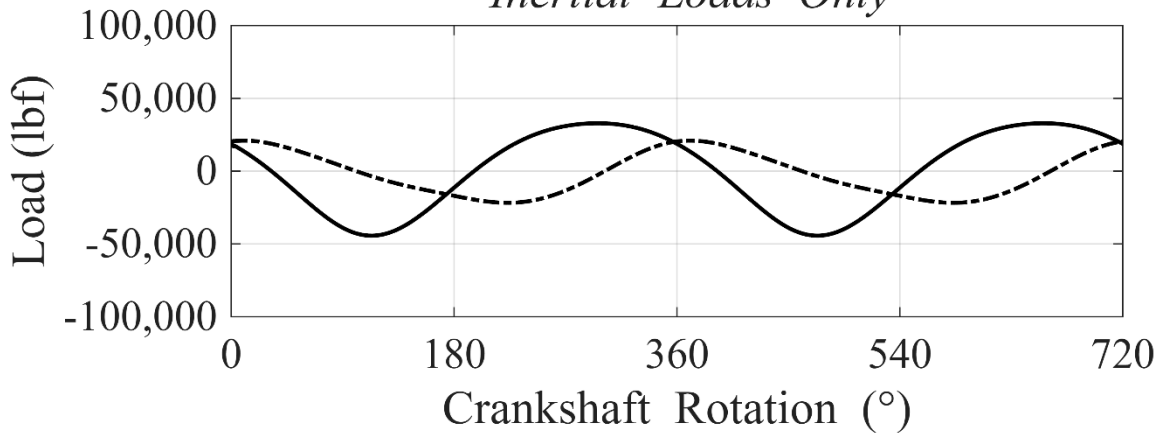


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 94: Components of the load on main bearing #7 due to the complete assembly inertia.

Components of Bearing Load - Main Bearing #8

Inertial Loads Only



— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 95: Components of the load on main bearing #8 due to the complete assembly inertia.

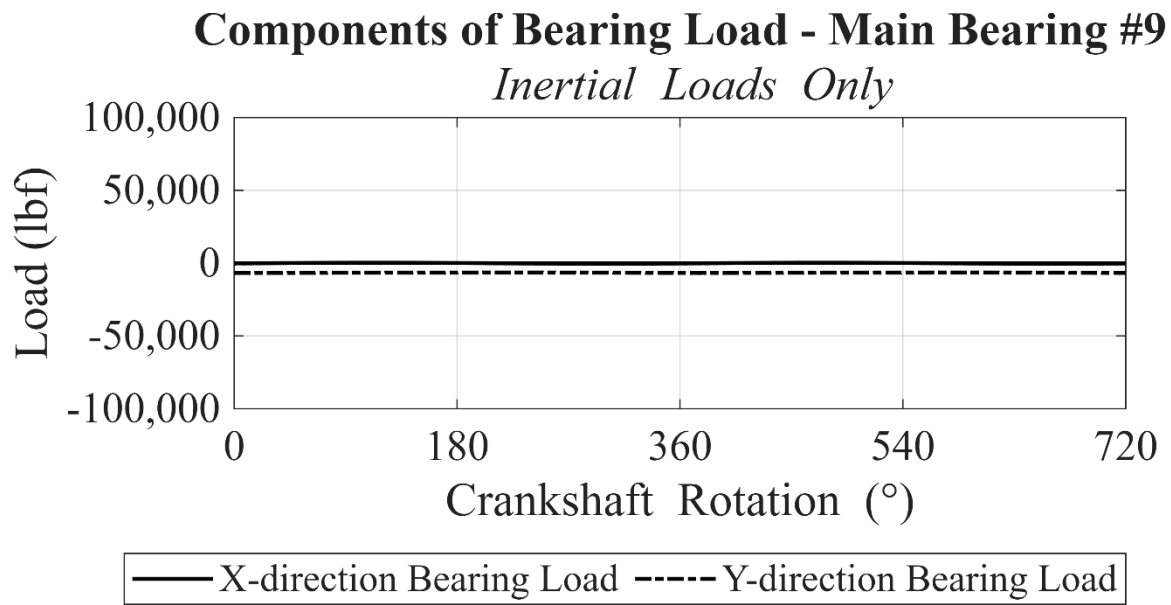


Figure 96: Components of the load on main bearing #9 due to the complete assembly inertia.

APPENDIX E

The following appendix provides images of the TWAL vectors for each main bearing for the inertial model. The magnitude and angle associated with each main bearing's TWAL vector were originally presented in Section 3.2.2.

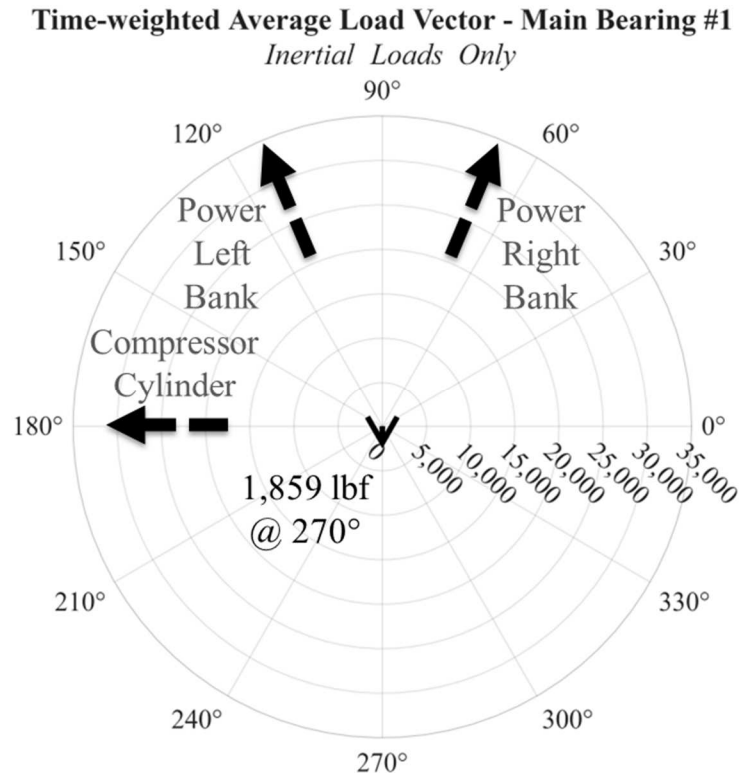


Figure 97: TWAL vector for main bearing #1 due to the inertia of the complete model.

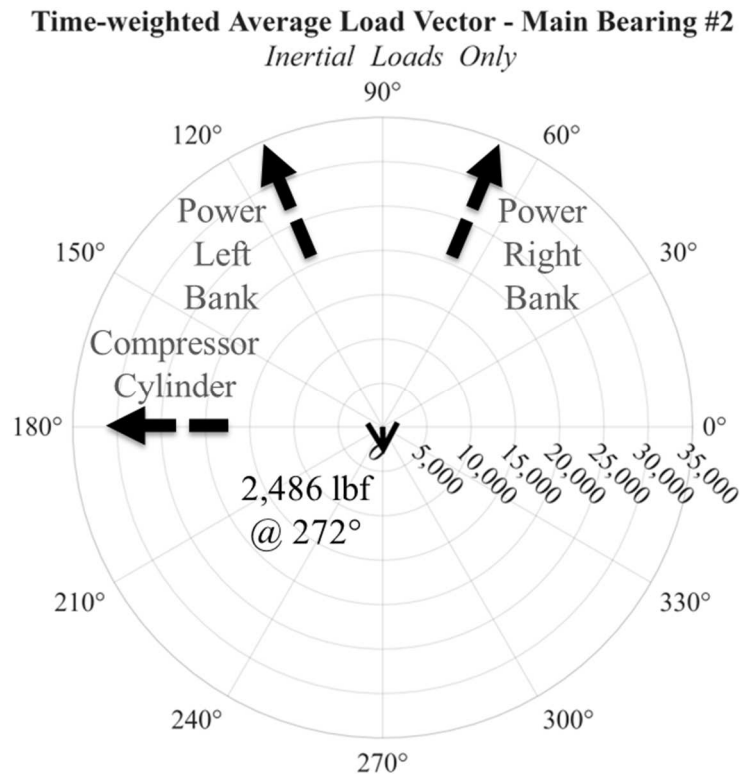


Figure 98: TWAL vector for main bearing #2 due to the inertia of the complete model.

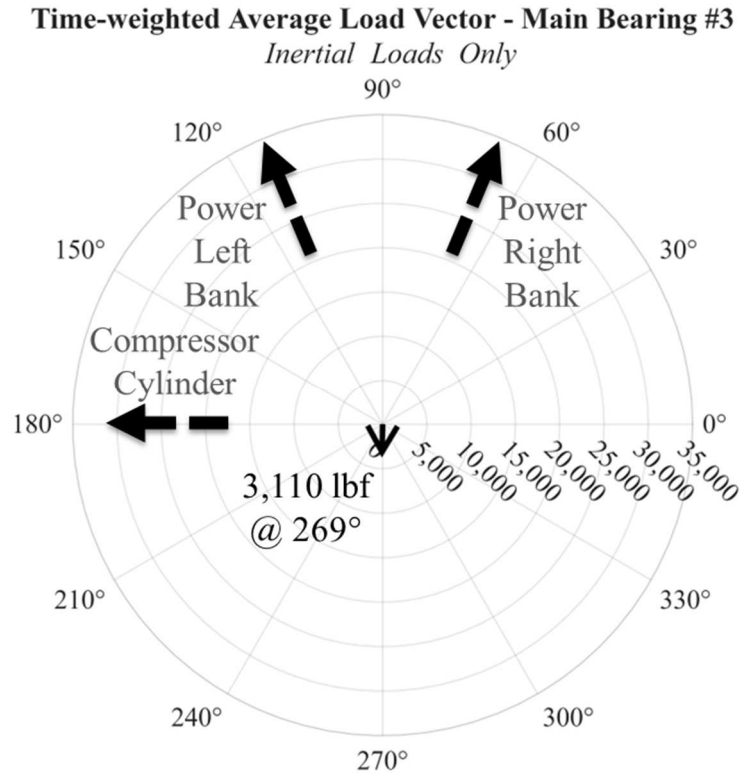


Figure 99: TWAL vector for main bearing #3 due to the inertia of the complete model.

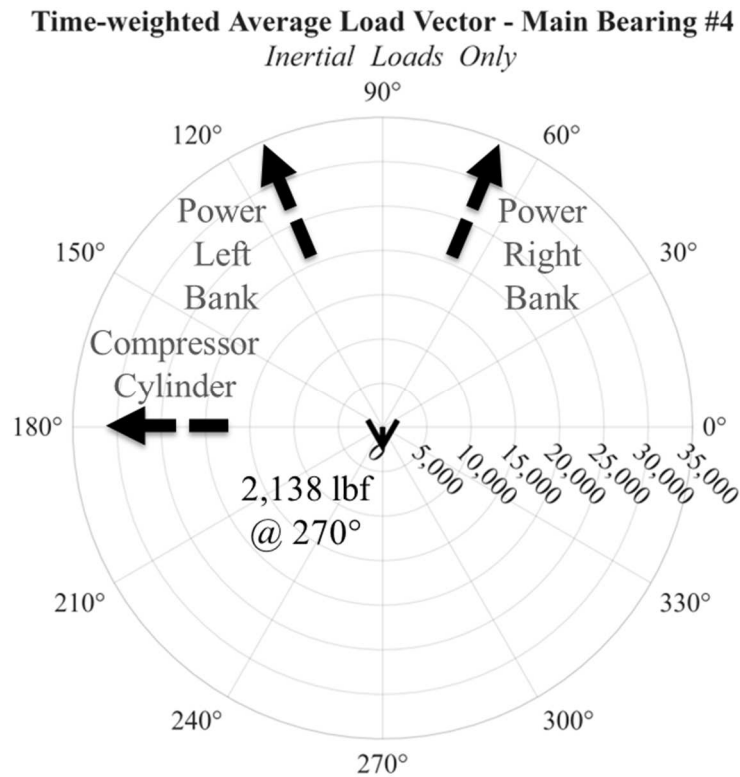


Figure 100: TWAL vector for main bearing #4 due to the inertia of the complete model.

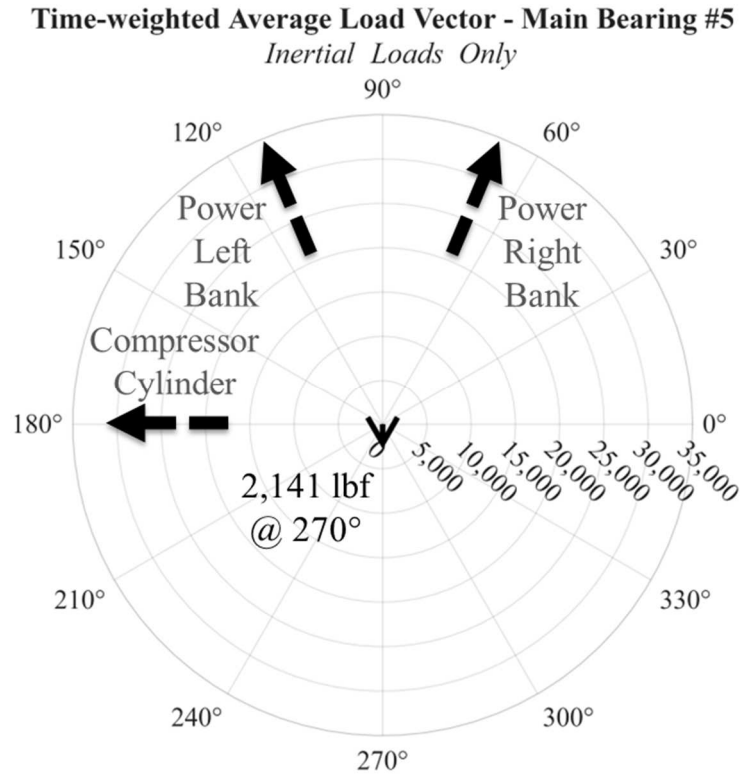


Figure 101: TWAL vector for main bearing #5 due to the inertia of the complete model.

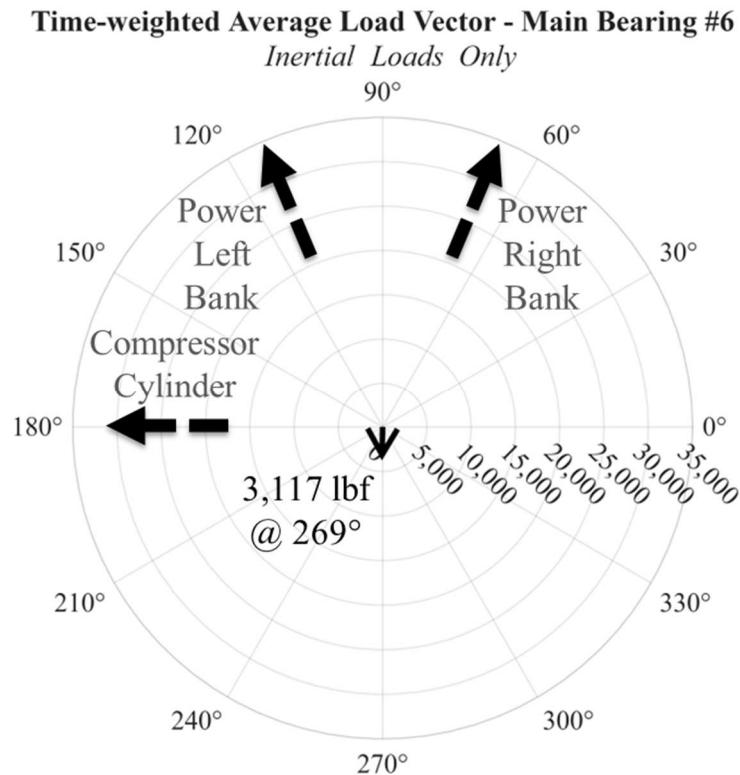


Figure 102: TWAL vector for main bearing #6 due to the inertia of the complete model.

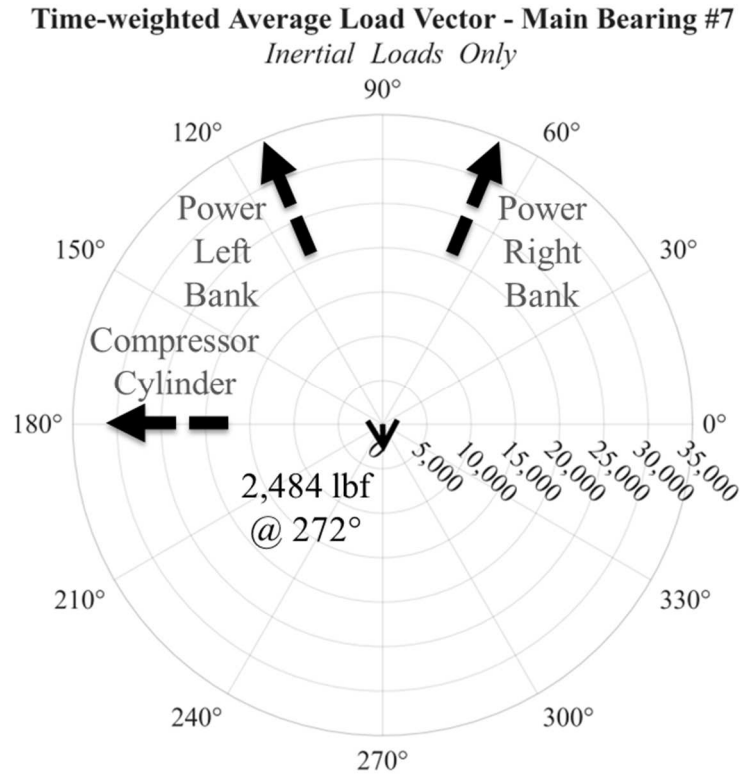


Figure 103: TWAL vector for main bearing #7 due to the inertia of the complete model.

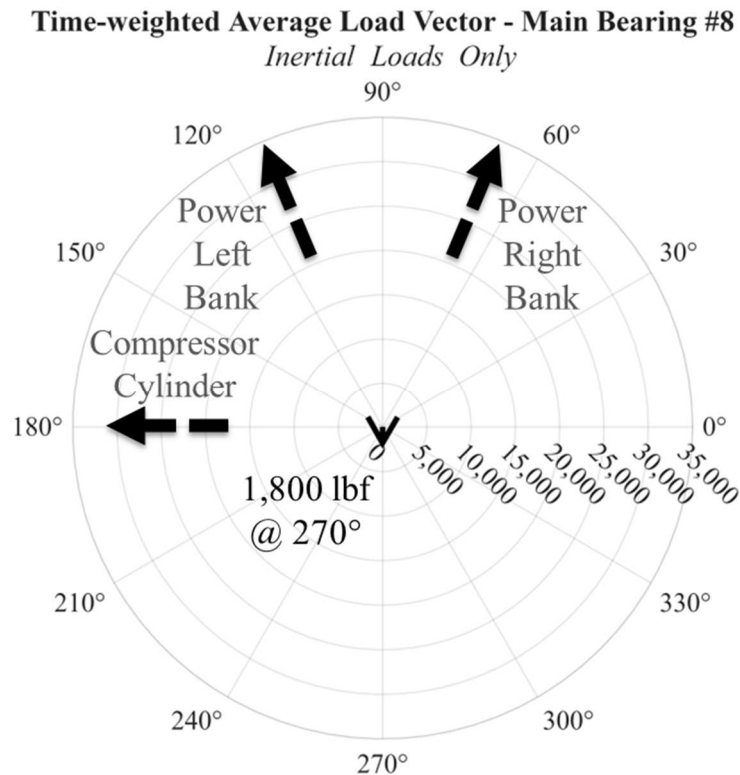


Figure 104: TWAL vector for main bearing #8 due to the inertia of the complete model.

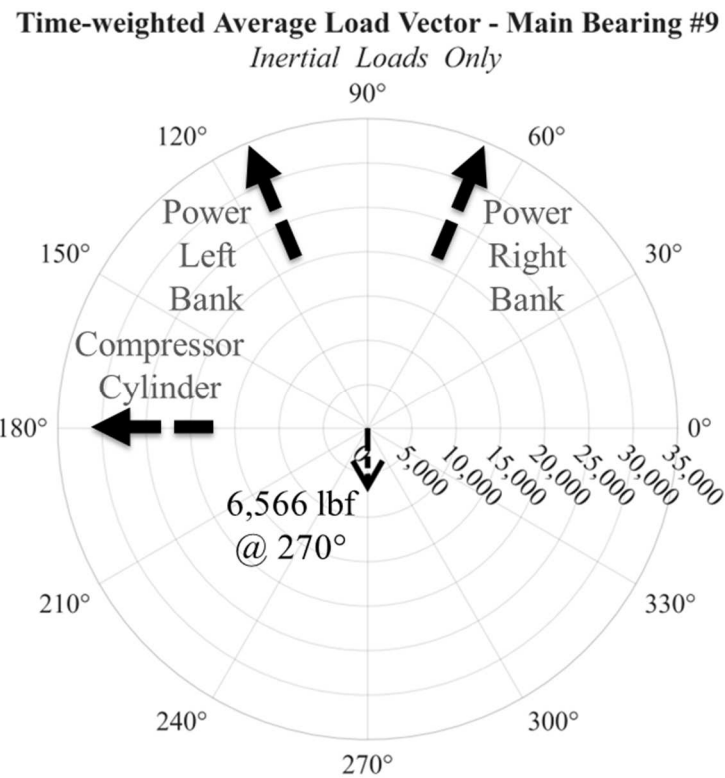


Figure 105: TWAL vector for main bearing #9 due to the inertia of the complete model.

APPENDIX F

The following appendix provides images of the X- and Y-direction components of the load on each main bearing for the finite element model, which only included the power cylinder components. The plots are presented over one working cycle of the KVSr 412. The X- and Y-direction components of the load on each main bearing were used to develop the vector plots presented in Section 3.3.1.

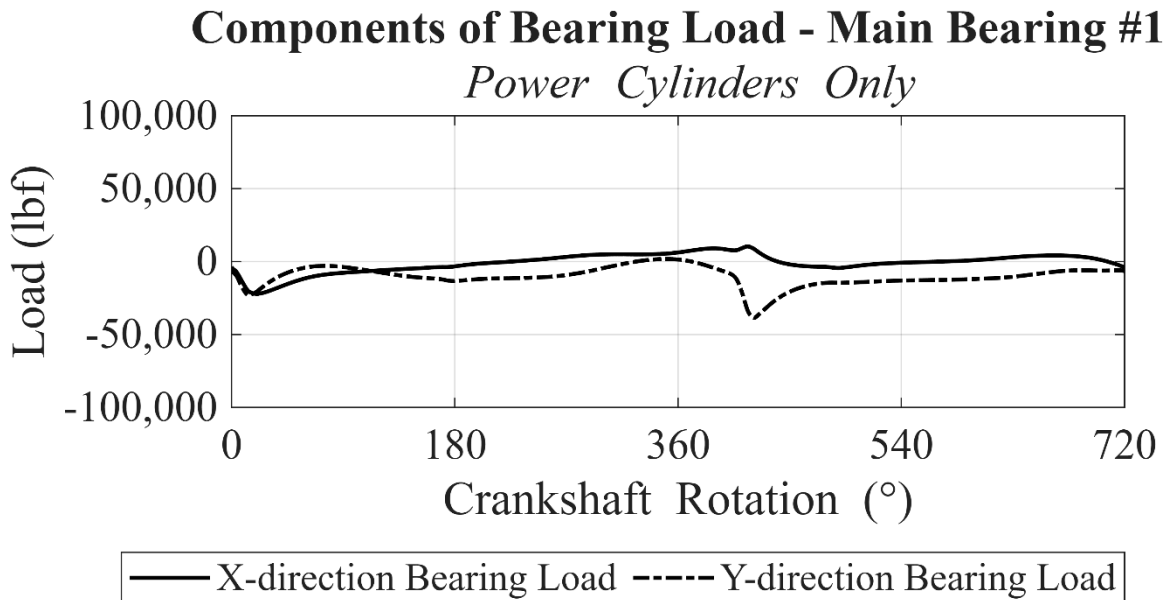


Figure 106: Components of the load on main bearing #1 due to the power cylinders.

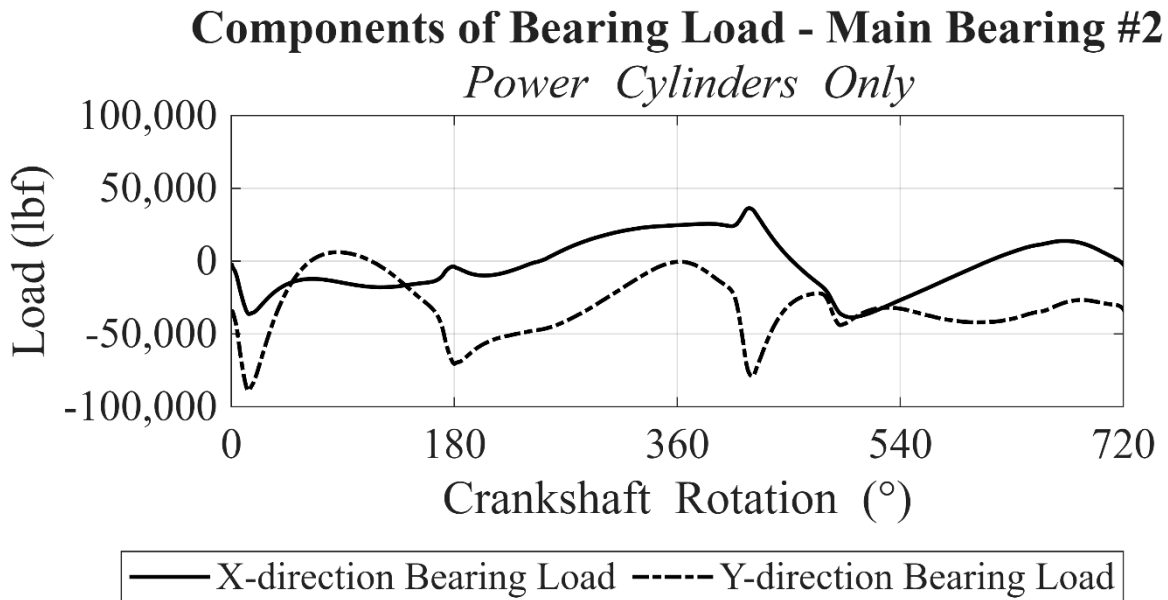
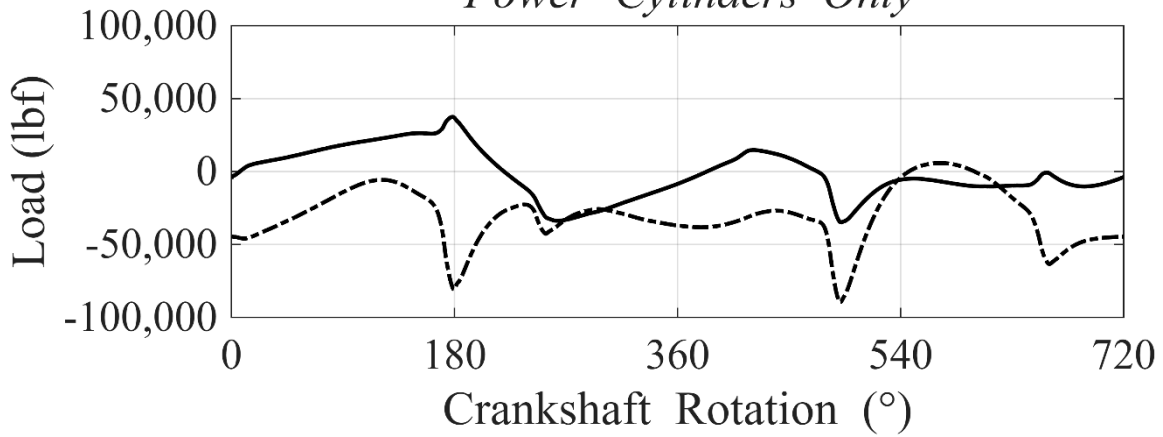


Figure 107: Components of the load on main bearing #2 due to the power cylinders.

Components of Bearing Load - Main Bearing #3

Power Cylinders Only

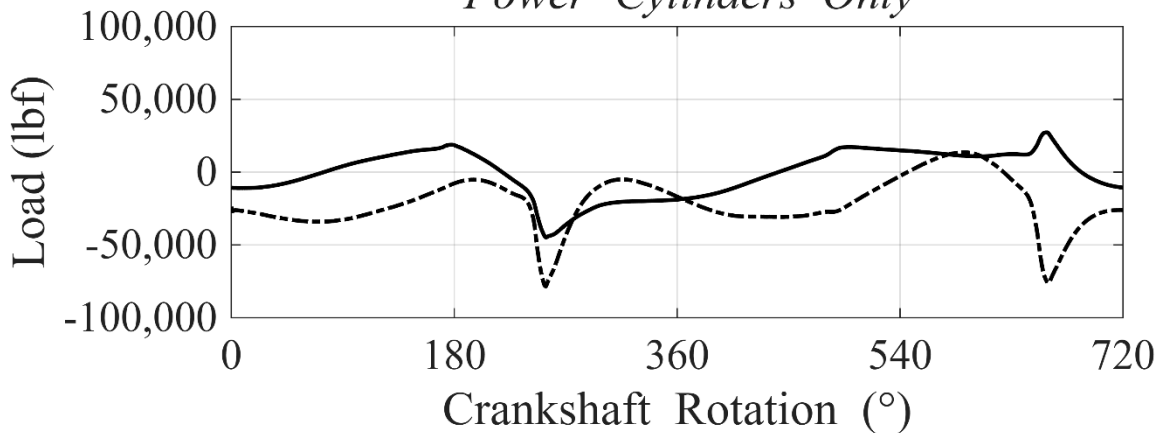


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 108: Components of the load on main bearing #3 due to the power cylinders.

Components of Bearing Load - Main Bearing #4

Power Cylinders Only

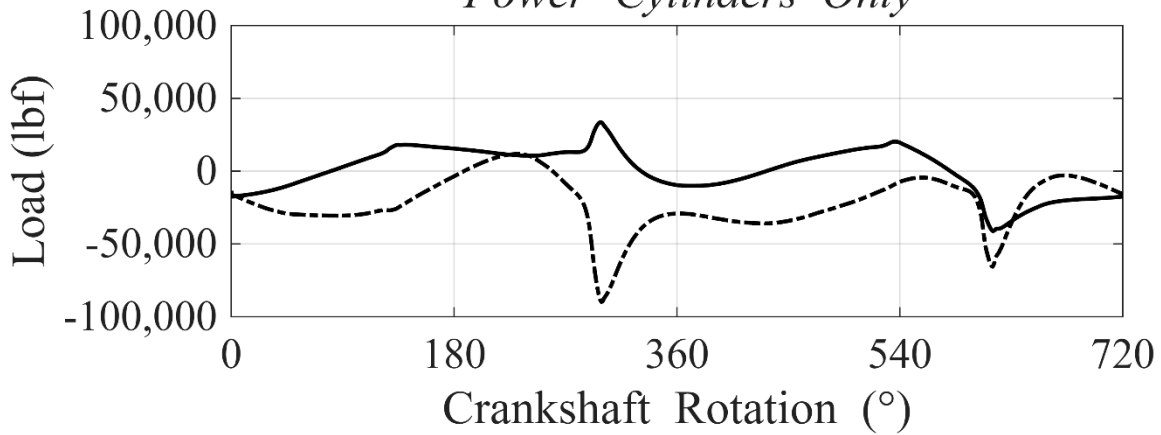


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 109: Components of the load on main bearing #4 due to the power cylinders.

Components of Bearing Load - Main Bearing #5

Power Cylinders Only

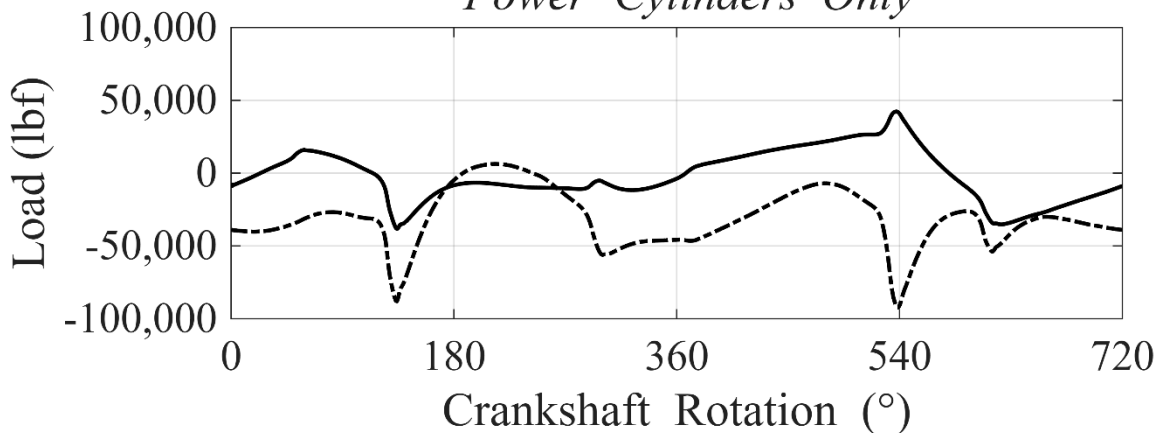


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 110: Components of the load on main bearing #5 due to the power cylinders.

Components of Bearing Load - Main Bearing #6

Power Cylinders Only

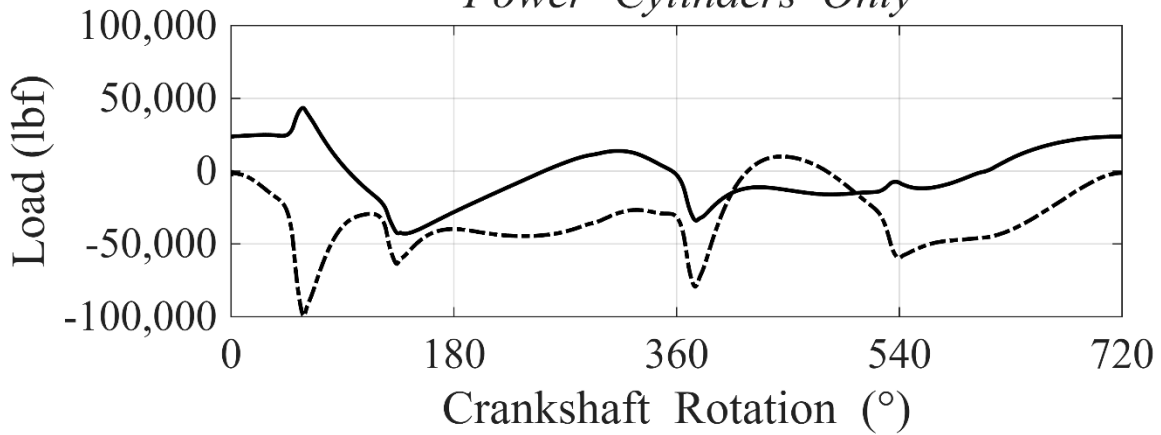


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 111: Components of the load on main bearing #6 due to the power cylinders.

Components of Bearing Load - Main Bearing #7

Power Cylinders Only

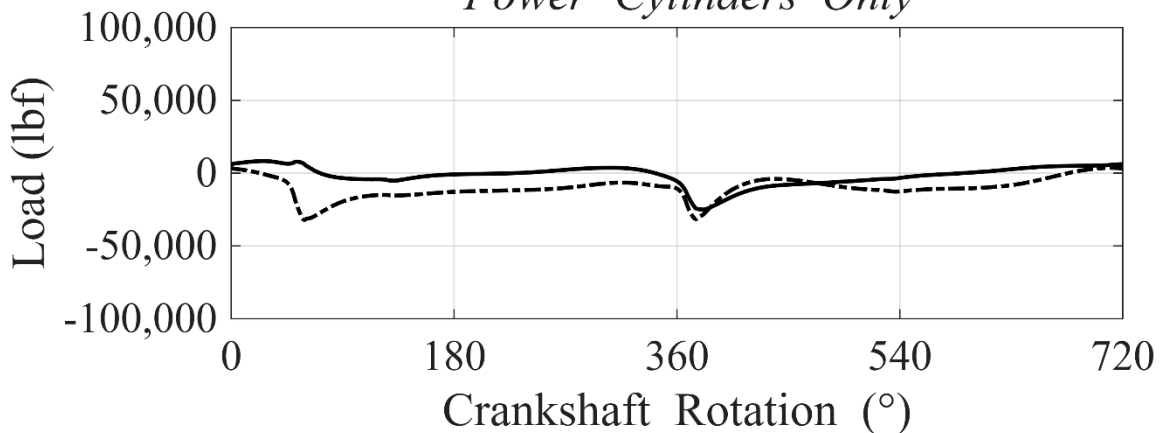


— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 112: Components of the load on main bearing #7 due to the power cylinders.

Components of Bearing Load - Main Bearing #8

Power Cylinders Only



— X-direction Bearing Load - - - - Y-direction Bearing Load

Figure 113: Components of the load on main bearing #8 due to the power cylinders.

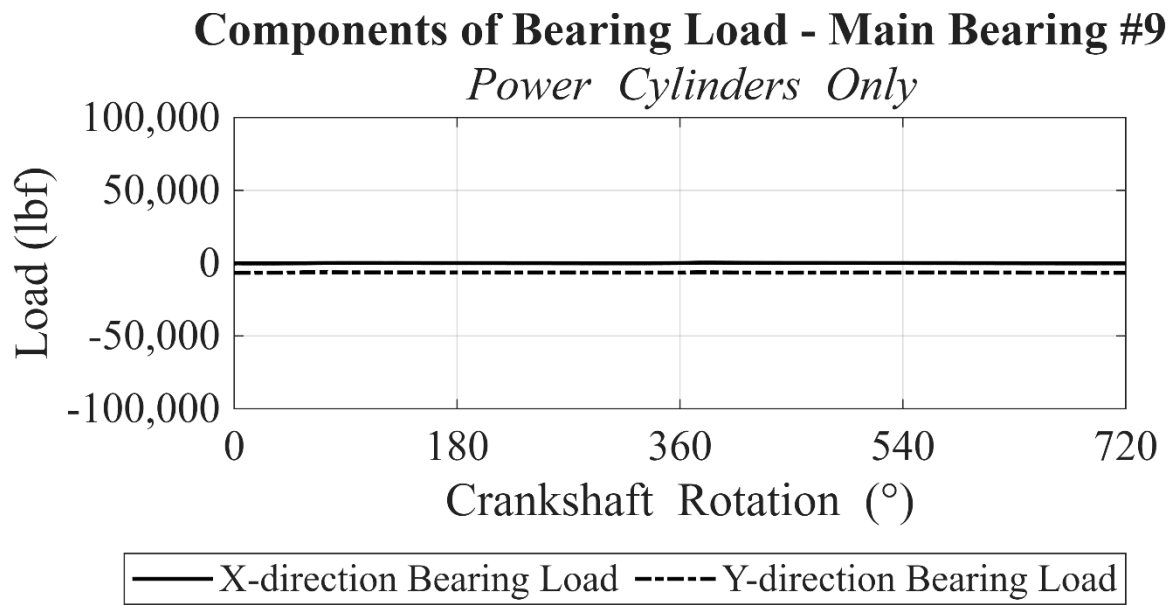


Figure 114: Components of the load on main bearing #9 due to the power cylinders.

APPENDIX G

The following appendix provides images of the TWAL vectors for each main bearing for the finite element model, which only includes the power cylinder components. The magnitude and angle associated with each main bearing's TWAL vector were originally presented in Section 3.3.2.

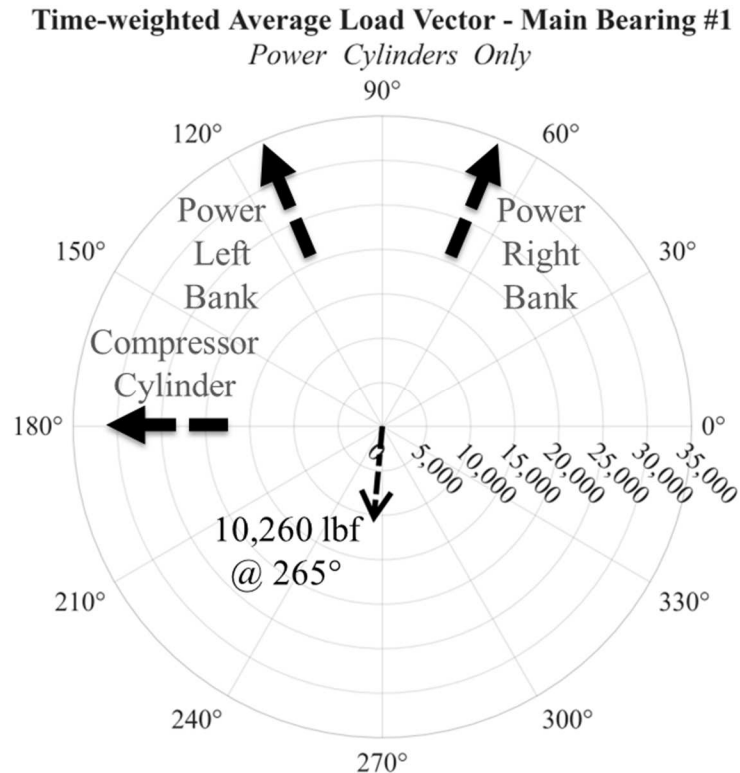


Figure 115: TWAL vector for main bearing #1 for the model with only the power cylinders.

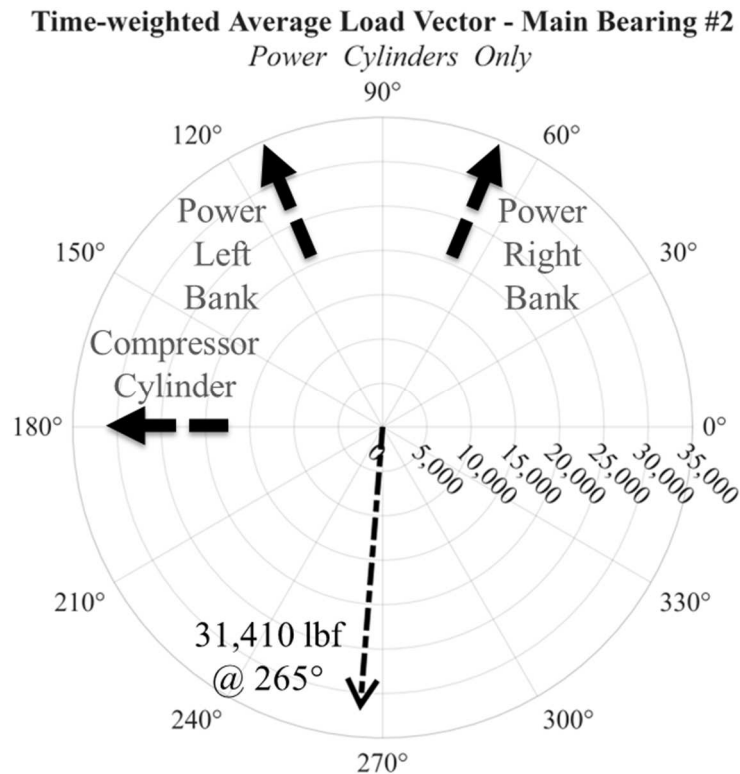


Figure 116: TWAL vector for main bearing #2 for the model with only the power cylinders.

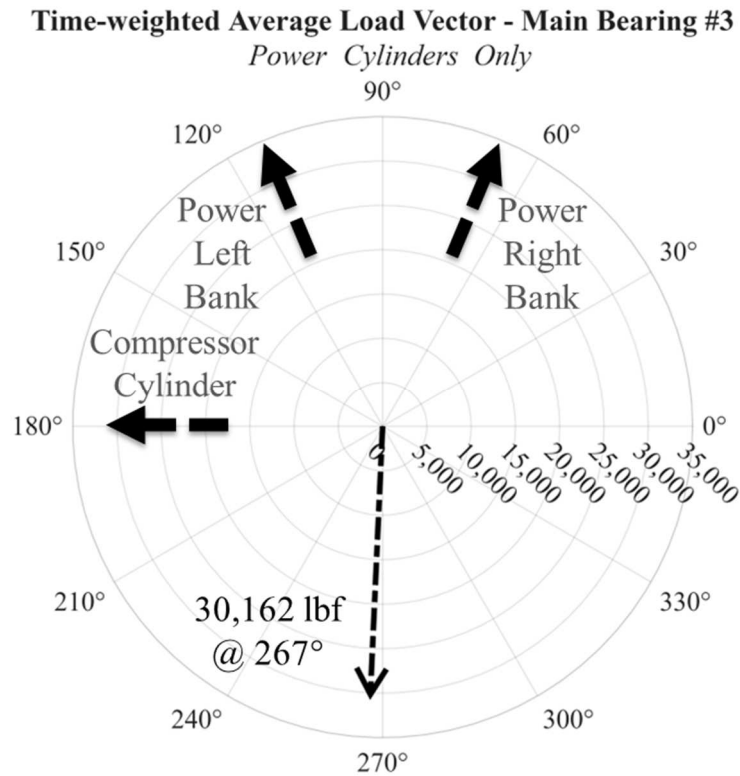


Figure 117: TWAL vector for main bearing #3 for the model with only the power cylinders.

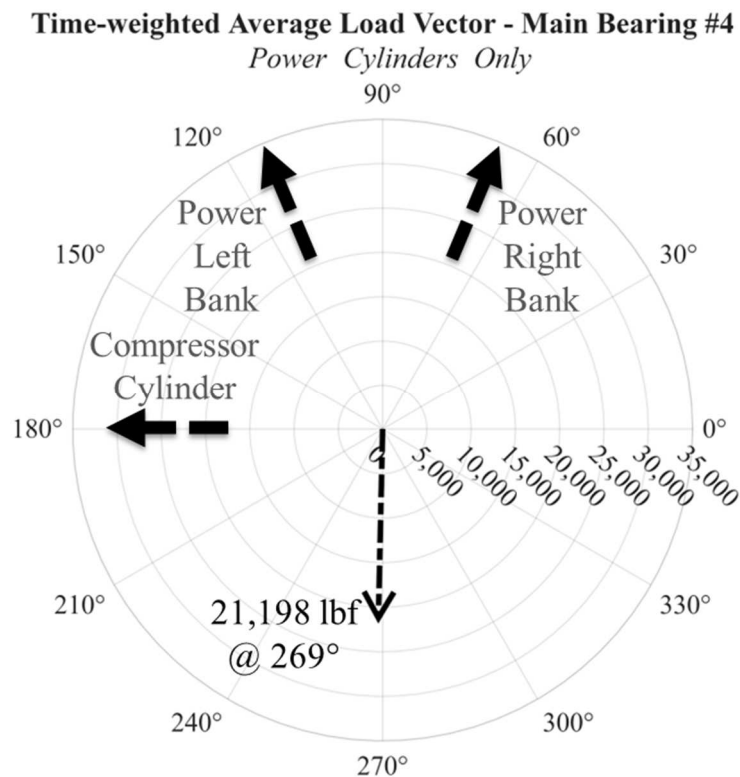


Figure 118: TWAL vector for main bearing #4 for the model with only the power cylinders.

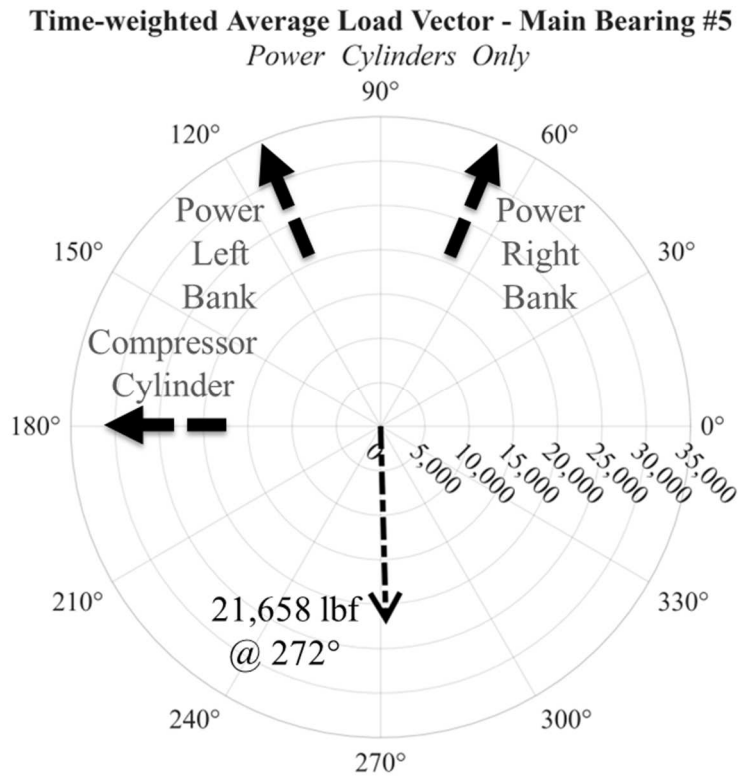


Figure 119: TWAL vector for main bearing #5 for the model with only the power cylinders.

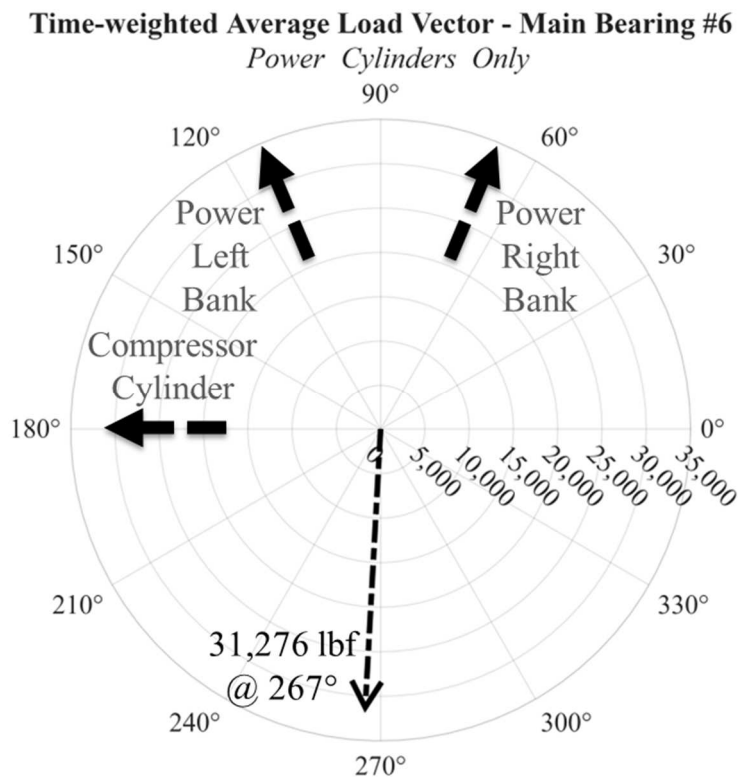


Figure 120: TWAL vector for main bearing #6 for the model with only the power cylinders.

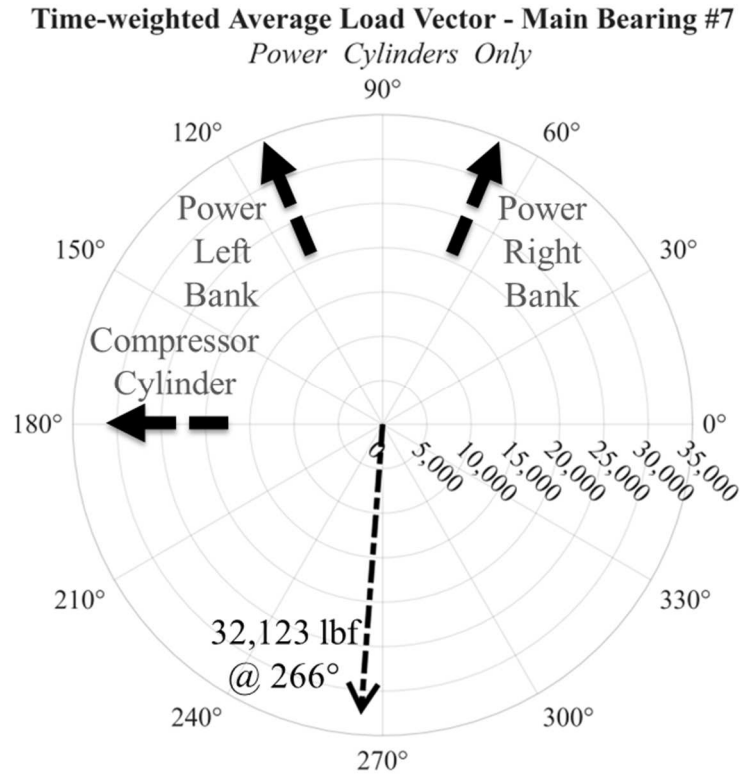


Figure 121: TWAL vector for main bearing #7 for the model with only the power cylinders.

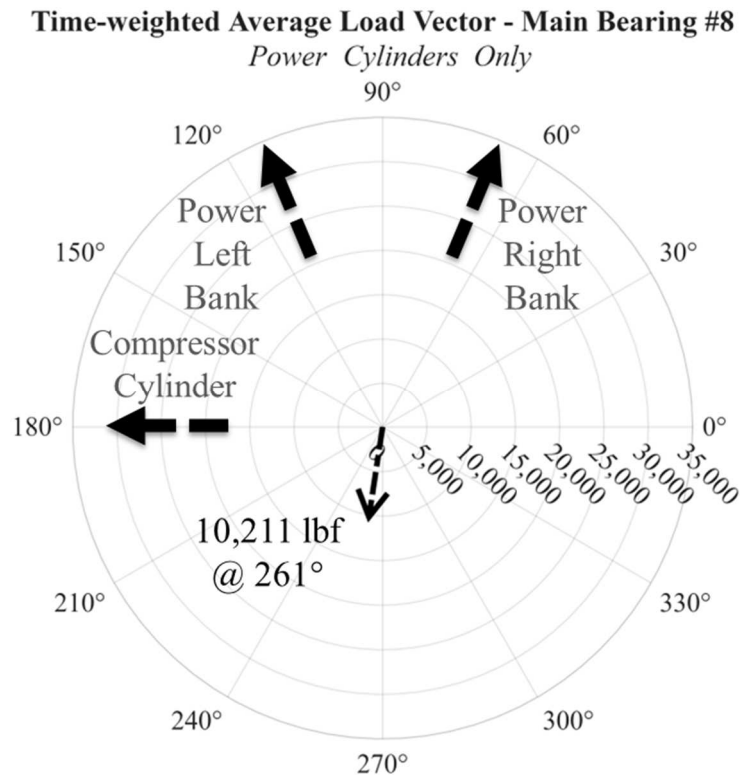


Figure 122: TWAL vector for main bearing #8 for the model with only the power cylinders.

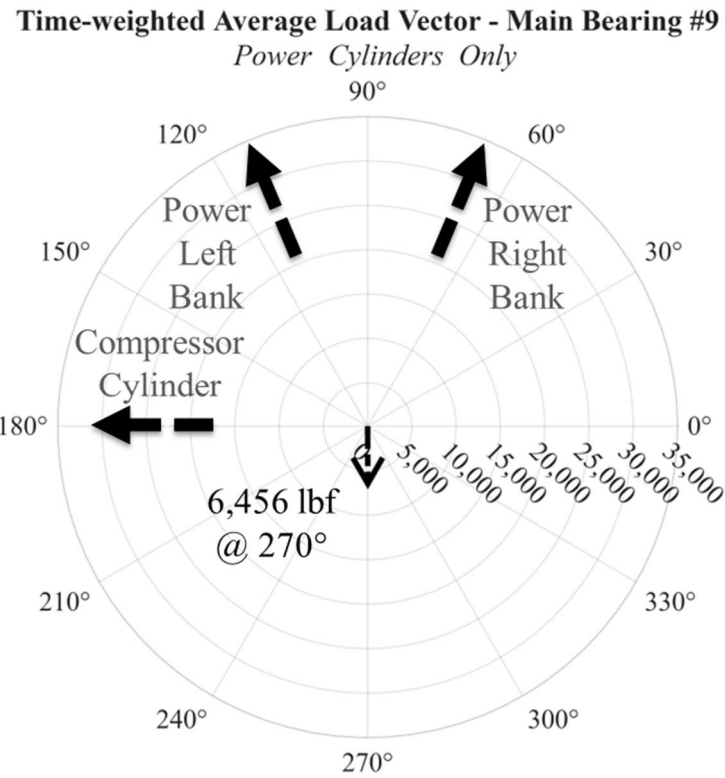


Figure 123: TWAL vector for main bearing #9 for the model with only the power cylinders.

LIST OF ABBREVIATIONS

Abbreviation	Term
BDC	Bottom Dead Center
BHP	Brake Horsepower
CE	Crank-End
FFT	Fast Fourier Transform
HE	Head-End
IEPE	Integrated Electronics Piezo-Electric
RMS	Root Mean Square
RPM	Revolutions Per Minute
RTD	Resistance Temperature Detector
TWAL	Time-Weighted Average Load