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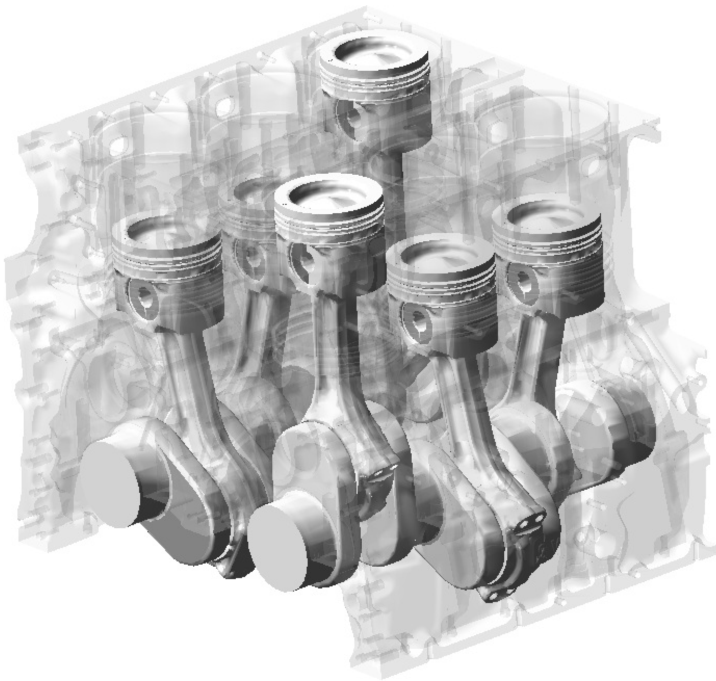

Balancing of Novel Engine Designs

Master's Thesis in Automotive Engineering

JIMMY KLING

RAMADAN FEIM SALIF

Department of Applied Mechanics
Division of Combustion
CHALMERS UNIVERSITY OF TECHNOLOGY
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Cover:
A principal sketch of a Parallel3+3 engine block with its rotating assembly.

Chalmers Reproservice
Göteborg, Sweden 2014

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ABSTRACT

Many manufacturers of heavy duty vehicles are continually developing their engines in order to meet regulations and customer demands. Up until recently the primary focus has been on development of the design of the engines to meet government emission regulations and is now shifting more towards the improvement of the overall engine efficiency. These engines might demand new engine design architectures and balancing of these can however become an issue for such novel engine designs. Truck engines develop high inertial forces which make it more essential that the engine is well balanced. As the new concept designs deviate more from the conventional designs, the existing methods of balancing become unusable due to software limitations.

This study was conducted with the Tribology and Mechanics research group, at the Advanced Technology and Research department at Volvo Group Trucks Technology to investigate a new generic approach to analyse vibrations and to develop balancing solutions for novel engine designs using the multi body simulation software Adams/View. The approach has been validated to give a preface towards understanding of vibrations of new complex engine design and their ability to be balanced. It facilitates the understanding of those new designs as the forces and acting moments due to various motions can be distinguished and also derived from the respective vibrational orders.

The balancing approach has also been utilized to investigate different crankshaft configurations in order to find the optimal design for the cause of balancing. The analysis investigated different amounts of counter weights on a six cylinder engine and showed that twelve counter weights are beneficial from a balancing standpoint. The different concepts that were studied have been proven to be in balance, although with different amount of additional balance shafts for some designs.

Key words: Balancing, Adams/View, Engine design, Volvo GTT, Volvo ATR

Balansering av Nya Motorkoncept
Examensarbete inom Automotive Engineering
JIMMY KLING
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Institutionen för tillämpad mekanik
Avdelningen för Förbränning
Chalmers tekniska högskola

SAMMANFATTNING

Många tillverkare av tunga fordon utvecklar ständigt sina motorer för att uppfylla regler och kundkrav. Fram tills nyligen har huvudfokus legat på utveckling av utformningen av motorer för att möta myndigheternas regler för utsläpp, nu skiftas det mer mot att förbättra den totala motoreffektiviteten. Dessa motorer kan kräva nya motorkonstruktionsarkitekturer och balansering av dessa kan dock bli ett problem. Lastbilsmotorer utvecklar större tröghetskrafter än en passagerarbilmotor som gör det ännu viktigare att motorn är välbalanserad. Eftersom de nya motorkoncepten avviker mer från de befintliga motorerna innebär det även att de konventionella tillvägagångssätten för att analysera balanseringen blir oanvändbara.

I denna studie, som genomförts med forskningsgruppen inom Tribologi och Mekanik på Advanced Technology and Research avdelningen under Volvo Group Trucks Technology, undersöks en ny generisk metod för att analysera vibrationer och balanseringslösningar för nya komplexa motorkoncept. Analyserna görs i mjukvaran Adams/View som är oberoende av konventionella motordelsmallar vilket gör tillvägagångssättet mindre begränsat för nya motorkonstruktioner. Metoden har validerats för att ge god inledande förståelse för vibrationer och obalanser som uppstår i en viss motordesign och dess potential att balanseras ut i ett tidigt stadie. Metodiken underlättar förståelsen av härkomsten av krafter och moment för olika delsystem och rörelser samt med vilken ordning de uppstår.

Balanseringstillvägagångssätt har också använts för att undersöka olika vevaxelkonfigurationer för att hitta den optimala balanseringsdesignen. Analysen undersökte olika mängder av motvikter på en sexcylindrig motor och visade att tolv motvikter är att föredra ur en balanseringssynpunkt. De olika koncept som studerats har visat sig kunna balanseras, dock med olika mängd ytterligare balansaxlar för vissa av konstruktionerna.

Nyckelord: Balansering, Adams/View, Motordesign, Volvo GTT, Volvo ATR

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Preface

In this study, a generic approach of balancing novel engine designs has been examined with emphasis on optimizing the system. The thesis work has been carried out from January 2014 to June 2014 under the supervision of Arne Andersson at Volvo Technology and Sven Andersson at Chalmers University of Technology. This work should enable Volvo GTT to evaluate new engine concepts and their potential for further development. The project is carried out at Volvo Group Truck Technology/Advanced Technology Research.

We would like to thank our supervisor Arne Andersson for initiating and giving us this thesis, and also for being very supportive and helpful during this work. We extend our gratitude to Professor Sven Andersson for his support in knowledge. We would also like to thank Björn Pålsson in Applied Mechanics department at Chalmers University of Technology for his support with Adams/View.

Göteborg June 2014

Jimmy Kling and Ramadan Feim Salif

List of Abbreviations and Notations

Abbreviations

CAD	Computer Aided Design
CoG	Centre of Gravity
DOE	Design of Experiments
FEM	Finite Element Method
RMS	Root Mean Square
Conrod	Connecting rod
MBD	Multi Body Dynamics

Notations

a_{Crank}	Crank radius [m]
F	Force [N]
L	Length [m]
m	Mass [kg]
r	Radius [m]
ω	Angular speed [rad/s]
Θ	Angle [rad]

1 Introduction

In this section the background and the aim of the project is defined and clarified. Further on the limitations of the project are presented.

1.1 Background

The ever increasing demand for cargo and people transportation has increased the need to improve the engine and vehicle efficiency in Heavy Duty vehicles, while meeting increasingly stringent exhaust emissions standards. Fuel efficiency is one of the most important competitive factors in the development and sale of trucks. Fuel costs account for approximately 30% of the total operating cost of ownership of a truck (DFF International, 2013).

A possibility to increase the fuel conversion efficiency is the usage of new engine design architectures, balancing of these can however become an issue for such novel engine designs. Truck engines develop high inertial forces which make it more essential that the engine is well balanced. As the new concept designs deviate more from the conventional designs, the existing methods of balancing becomes unusable due to software limitations.

There are a several unconventional new concepts that have been proven to improve the thermo dynamic fuel efficiency. Although most of these concepts feature asymmetric mechanic systems of the inertial moving parts which causes unbalance forces high enough to exceed the vibrational acceptance, even for a small passenger vehicle engine. The importance of a well-balanced engine is a matter of sustainable construction which is essential for heavy duty vehicles.

Weight reduction is also of interest as it both can reduce the fuel consumption and increase the product value for the customer since the loading capacity increases on the truck. Making the engine smaller will also save space for more efficient packaging as well as potentially reduce the manufacturing costs. Although weight saving is desirable the engine becomes more sensitive to vibrations as the lighter the engine becomes for same forces, the higher acceleration is exerted on the engine block. Also the power outcome has to meet existing conventional engines thereby demanding higher requirement on the balancing system.

1.2 Scope

The aim of the project is to develop a generic approach using the Adams/View MBD simulation tool developed by MSC Software to balance different engine designs, without the limitations that come with the usage of other tools such as GT-Suite. This approach will be used to evaluate several different engine concepts and developing/optimizing appropriate balancing systems.

1.3 Problem Definition

- How can balancing of novel engine designs be accomplished using a generic approach as demands on conventional software's abilities exceed their limitations?
- How can a new concept be balanced without adding to much complexity to the system?
- How do the internal forces and moments affect the main bearing forces when using flexible crankshaft analysis compared to a rigid one?
- How do the gas forces influence the unbalance of the engine?

1.4 Limitations

The balancing will be done in an early phase of the design process, which limits the level of detail in the engine to the most critical moving parts. The project will also be limited in the amount of evaluated designs, i.e. one conventional design and a few new concepts.

The project is focused on the balancing of different novel engine designs and therefore it does not provide details of how synchronisations of shafts and clutches should be developed. Coaxial balance shafts will be used as a solution for balancing, but the actual engineering of the shaft or its mechanical construction will not be covered in the project.

The different new concepts analysed in the project will be limited to only rigid simulations and will not involve a flexible analysis, due to the focus on only the overall balancing and not the internal forces on the components.

2 Theory

2.1 Theory of Implementation of the Centripetal Force

A spinning mass which is not centred in its rotational axis will create an inward directed force with a magnitude according to equation (2.1). Two counter rotating shafts are needed if the force needs to be limited to only one plane instead of two. An example of the basic principle for this is shown in Figure 2.1. Two equally distant masses on each balance shaft are used in order to enable the system to operate without an additional moment unbalance.

$$F = mr\omega^2 \tag{2.1}$$

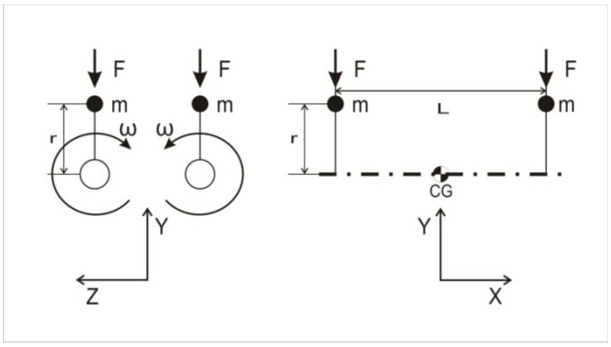

Figure 2.1 Conceptual view of the creating of an oscillating force in only one plane.

Two spinning masses that have a phasing of 180 degrees between them will create a moment unbalance that oscillates in two planes. This is obtained while cancelling out the forces. By adding a second shaft with the same masses that rotates in the opposite direction the moment can be focused to only one plane, this concept is shown in Figure 2.2. The magnitude of the moment can be calculated using equation (2.1) and (2.2).

$$M = F * L \tag{2.2}$$

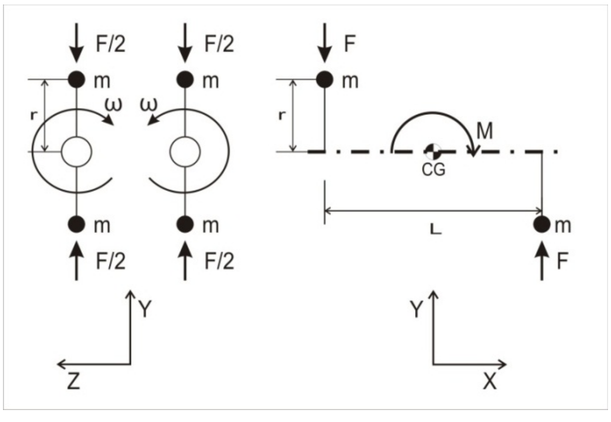

Figure 2.2 Conceptual view of the creating of an oscillating moment in only one plane.

2.2 Design of Experiments

DOE (design of experiments) is a function where all possible combinations of chosen variables are evaluated towards a specific measurement within a given range. This method is used instead of an optimization tool as the different variable combinations give a lot of local minima which makes the optimization tool inefficient for this purpose. The DOE function is used by giving the different variables their boundaries and a total number of levels that each variable range will be divided into. Using different angle and mass variables for individual counterweights for the same force or torque unbalance gives unnecessary cases as the total number of cases in the DOE will be the chosen level powered of the amount of variables. It is desirable to use high number of levels as it gives higher accuracy but in the other hand the computational power demand increases rapidly. Defining the different variables with care and the choice of range can thereby decrease the total DOE-process time efficiently. For instance, balancing a force in only z-direction means that the angle of one counterweight will be the same but in opposite sign than the other counter weight which gives one less variable to compute. However some inaccuracy in the parts provided by the CAD model causes some invalid unbalanced forces. As Adams analyses the dynamics of the system, the DOE can be used to precisely locate the most optimal position for a counter weight in order to balance the forces or torques, even if the model in reality does not have the same asymmetries in the parts and would be perfectly balanced by using more trivial positions for the counter weights.

2.3 Engine Specific Theory

Vibrations in an engine are partly caused by the unbalanced forces induced by the inertia of masses in motion and partly caused by the deformations of the internal components. First and second order unbalanced forces are the most significant forces and they appear from the rotating and reciprocating components in the engine, which can usually be balanced by using a symmetric engine configuration or by adding balance shafts. The cylinder gas pressure produces a force upon the piston which is comprised several orders, seen in Figure 2.3 is a Fourier transformation which shows the orders up to the fifteenth order. Different engine configurations will cancel out different orders from the force but some orders of vibration will remain. The higher order vibrations from the gas forces, engine speed variations and deformations in the material can be dampened with a vibration damper mounted to the crankshaft.

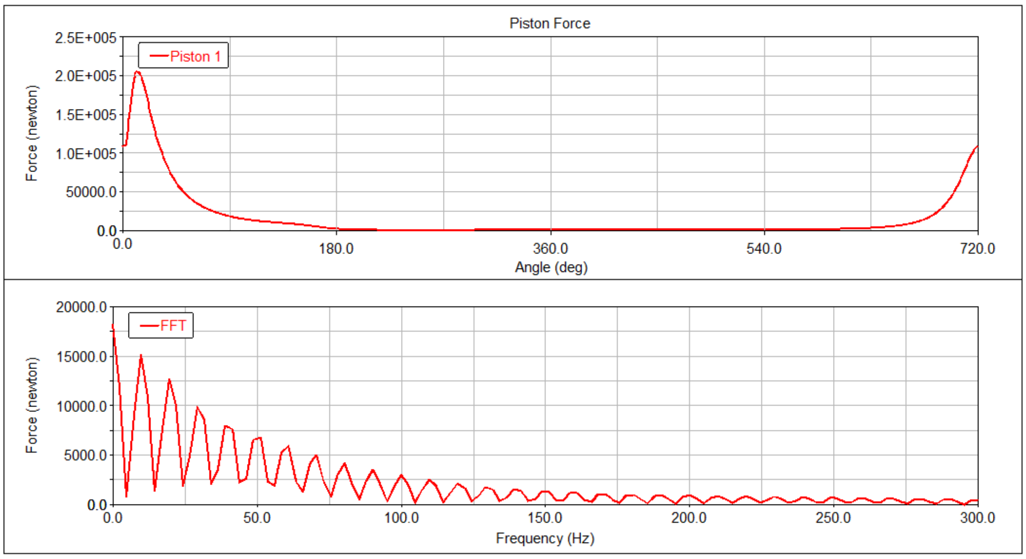

Figure 2.3 Piston force and the subsequent Fourier transform.

The unbalanced forces for a conventional engine can be calculated using equation (2.3) – (2.7) (Garrett, 2000) (Heisler, 1995):

Total reciprocating forces:

$$F_{OSC} = -m_{OSC} * a * \omega^2 (\cos\Theta + \frac{1}{R}\cos 2\Theta - \frac{1}{4R^3}\cos 4\Theta) \tag{2.3}$$

First order:

$$F_{OSC,I} = -m_{OSC} * a * \omega^2 \cos\Theta \tag{2.4}$$

Second order:

$$F_{OSC,II} = -m_{OSC} * a * \omega^2 \frac{1}{R}\cos 2\Theta \tag{2.5}$$

Where:

$$R = \frac{L_{Conrod}}{a_{Crank}} \tag{2.6}$$

The rotating force:

$$F_{rot} = m_{rot} * a * \omega^2 \tag{2.7}$$

As the connecting rod is both reciprocating and rotating the mass is usually divided as 1/3 reciprocating mass and 2/3 rotating mass. The rotating unbalance forces are generally balanced out by adding counter weights on the crankshaft, which also can minimize some of the reciprocating first order unbalance forces in a conventional one cylinder engine but not completely. To eliminate those completely two additional balance shafts needs to be added to the system and these need to rotate in opposite direction. The second order forces will need another pair of balancing shafts operating at twice the speed of the crankshaft. The balancing of the second order forces are usually not considered for smaller engines as the amplitude of those forces are often lower than an acceptable level of vibration. In a multi cylinder engine the firing order is determined with the balancing forces in consideration. A six cylinder engine is considered to be fully balanced for first and second order without suffering from rocking moments. (Taylor, 1985)

Figure 2.4 shows the reciprocating forces in first and second order for a single cylinder engine with and without a counter weight, which has been calculated in Matlab using equation (2.4) and (2.5). Three different counter weight configurations have been calculated and they are 0% (only crank throw and big end of the connecting rod are balanced), 50% balancing factor and 100% (when all the first order unbalance is balanced). The effects of the different balancing configurations in reciprocating and lateral forces are shown in Figure 2.4 and Figure 2.5. (Taylor, 1985)

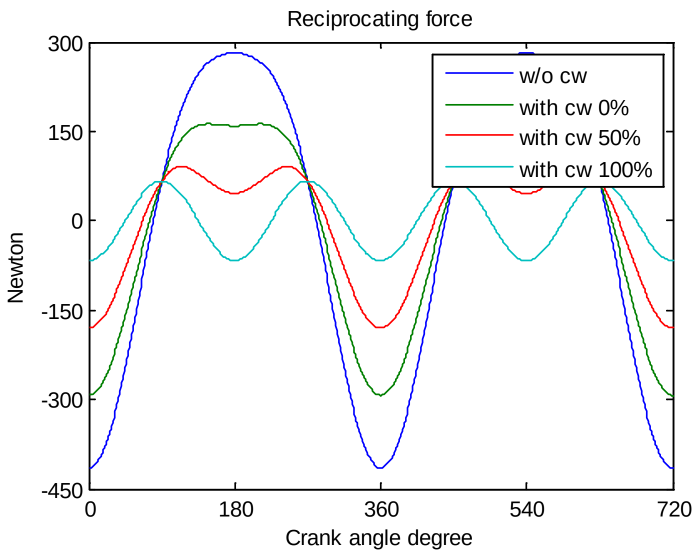

Figure 2.4 Reciprocating forces in a single cylinder engine with different balancing factors.

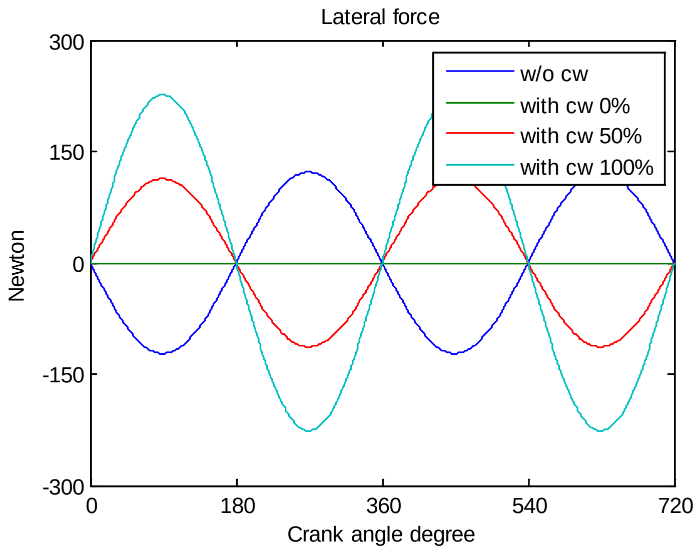

Figure 2.5 Lateral forces in a single cylinder engine with different balancing factors.

Calculation of the reciprocating and lateral forces in a six cylinder engine is shown in Figure 2.6 and Figure 2.7, which makes it clear that a completely rigid engine would be perfectly balanced. Each line in Figure 2.6 and Figure 2.7 is representing a cylinder pair, as an inline six cylinder has three pairs, it can be seen that the sum of the total forces are equal to zero, thereby fully balanced.

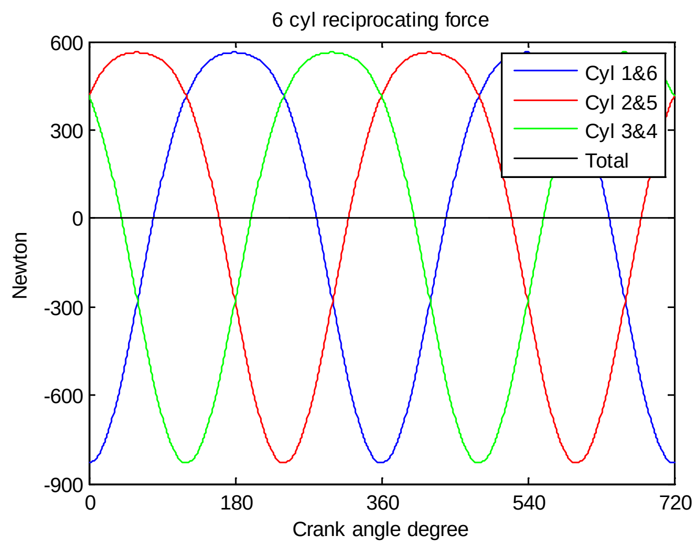

Figure 2.6 Reciprocating forces for a six cylinder engine.

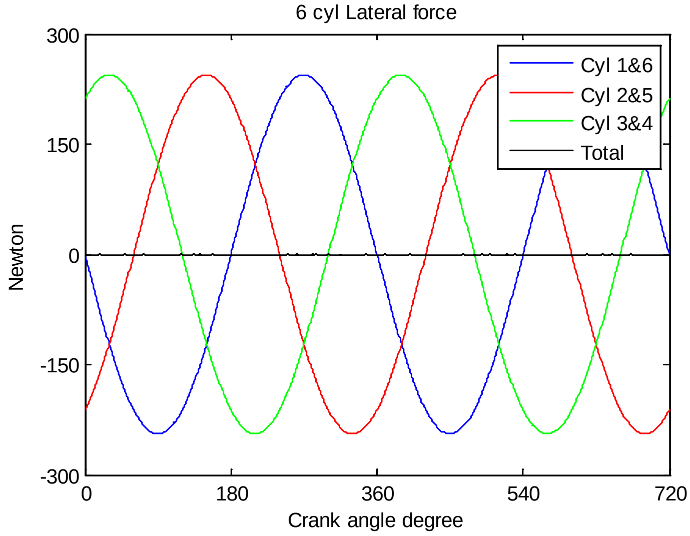

Figure 2.7 Lateral forces for a six cylinder engine.

In multi cylinder engines the crankshaft is usually designed to balance the rotating and reciprocating inertia forces by positioning the crank throws opposite to each other. This will however give "rocking forces" as they appear in different planes. This means that a three cylinder four stroke engine will suffer from these rocking forces where as a six cylinder engine will not as the opposite crank throws are designed to operate in pairs and neutralize the rocking forces. (Taylor, 1985)

3 Methodology

3.1 Analysis and verification of the balancing approach

The reference engine model will be verified by comparison to the conventional approach, both in mathematics and by usage of GT-Suite. This validated balancing approach will be to evaluate the new engine concepts. Also the results from the reference engine can be used as a benchmark to achieve for the new concepts with optimized designs. The analysis will be based on the total vertical and horizontal forces and moments on the engine block. The measuring point is positioned in the centre of the engine block. The results will be analysed by usage of a Fourier transformation tool in Adams/PostProcessor in order to distinguish the different orders of the forces and moments. This enables the opportunity to design the balancing systems especially designed for a specific order as well as allowing deeper understanding of the complete system. The engine speed for the analysis is chosen to be 1200 rpm which means that the first order occur at a frequency at 20 Hz, the second at 40 Hz and the third at 60 Hz. The chosen coordinate system is shown in Figure 3.1, the x-axis is through the crankshaft axis, the y-axis is horizontal and the z-axis is vertical.

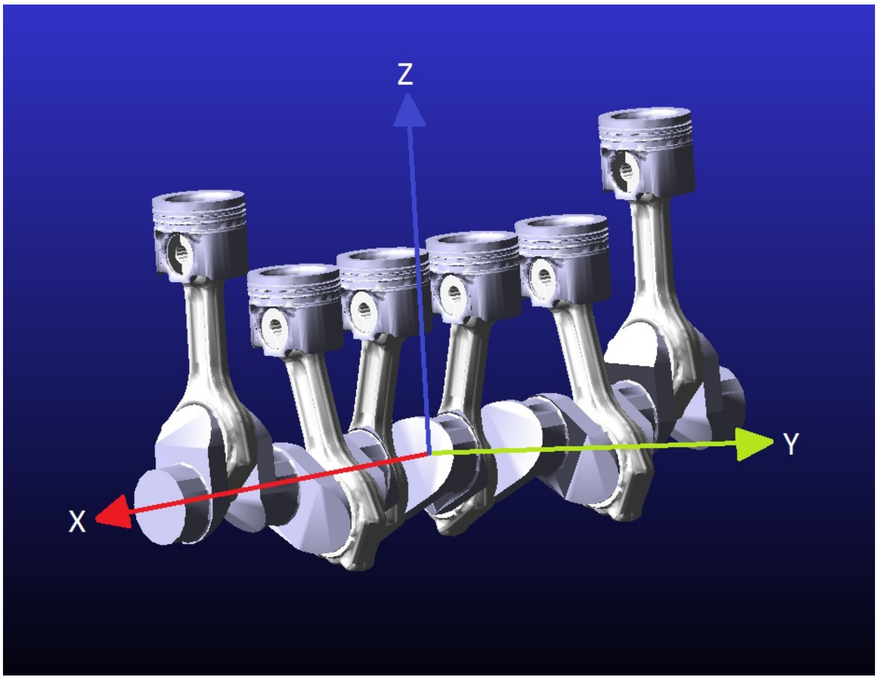

Figure 3.1 Coordinate system used throughout the report.

3.2 Reference model

The six cylinder engine Volvo MD13 will be used for validation of the method. A complete CAD model will be imported into Adams/View as a Parasolid file format in order to transfer all the material properties in to the model. It is preferable to distinguishing rigid analysis with flexible to be able to analyse how the different orders of vibrations appears. The higher order vibrations can be analysed by adding flexible body properties on the inertial parts and adding gas forces in order to design a vibration damper. To validate the optimizing method the crankshaft of the reference engine will be modified to not have any counterweights. Instead there will be point masses around the crankshaft where the location and mass can be varied by Adams/View “Design by Experiment” function in order to optimize the balancing.

3.2.1 Analysis of a flexible crankshaft on the reference model

Running an analysis on a system that utilizes a rigid crankshaft hinders the ability to obtain accurate forces on individual bearings. Studying the forces in the main bearings gives the possibility to determine the balancing strategy that offers the lowest bearing forces. The different strategies utilizes different masses of counter weights, whereby targeting different sources of the forces in the main bearings. Specific information of the strategies can be found in Table 3.1. Cylinder pressures were applied on the pistons as a separate analysis in order to determine the effect of gas forces in a vibrational point of view on the engine.

All analyses were done by introducing a flexible crank in a file format called MNF. Adams/View although has the ability to apply finite elements to an object, but the ability is limited to simple bodies. The crankshaft of the reference model could not be modelled using the internal FEM solver in Adams/View and therefore the MNF-file was prepared using a CAD program. Ansys 15.0 was used for structural analysis in order to import the file into Adams/View.

Table 3.1 Balancing strategies for the flexible analysis

Strategy	Amount of counter weights	The source of the bearing forces
1	Zero	No effect
2	Four	Internal moment
4	Eight	Internal moment
5	Twelve	Internal forces, due to rotating masses

Figure 3.2 shows the possible positions of the counter weights with their subsequent numbers. When running a setup with four counter weights the placement of the counter weights will be in positions 1, 6, 7 and 12.Going from four to eight counter weights adds four more counter weights on webs 2, 5, 8 and 11. The twelve counter weight setup uses all of the twelve placements shown in Figure 3.2.

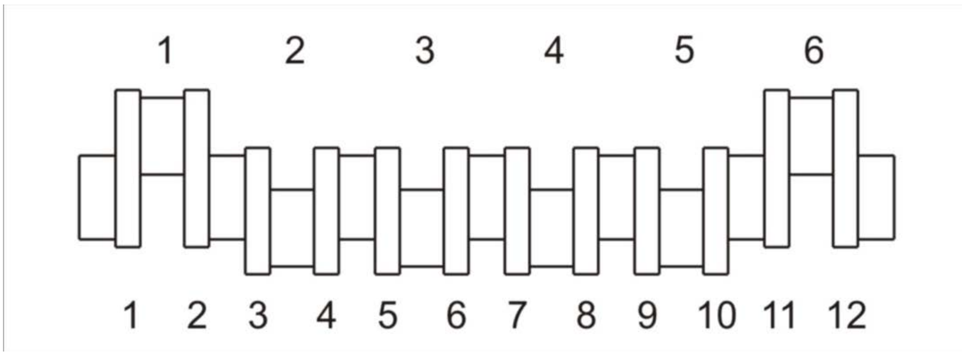

Figure 3.2 Six cylinder crankshaft with possible counter weight placements

3.3 Concept Inline3+3

The Inline3+3 engine concept is an inline six cylinder engine where the crankshaft is split in to two. The two halves of the engine are connected by a clutch located on a countershaft when desired, e.g. at high power demand. The motive is to deactivate half the engine including the crankshaft and the valve train in order to reduce friction losses leading to reduced fuel consumption where high power demand is not necessary. When half of the engine is deactivated the balancing becomes an issue. The inline six cylinder engines are in general well balanced, however switching to a three cylinder engine will introduce rocking moments. It is a necessity to find an optimal compromise working well for both modes without adding to much complexity or friction to the system. This concept has the opportunity to add balancing weights on two shafts (crankshaft and countershaft) to balance out the first order unbalances. By using the Adams/View software it was possible to analyse the existant unbalanced moment in order to design a good solution and optimize the locations and weights of the counter weights.

The angles of the counter weights in relation to the crankshaft position in top dead centre will be given according to Figure 3.3. The counter weights on the crankshafts and on the counter shaft are positioned 91.5 mm (referred from the centre of gravity position of the counterweights on the MD13 crankshaft) from the rotational axis and 50 mm for the counter weights attached to the coaxial balance shaft. A conceptual sketch of the Inline3+3 is shown in Figure 3.4 where the shafts to the left of the crankshaft are the secondary shafts and the shafts on the right side are the coaxial balance shafts. When the engine is running on six cylinders the torque from the front half of the engine will be transferred via gears to the secondary shaft then through a clutch to the rear secondary shaft and then through gears to the rear crankshaft. The concept also utilizes two coaxial balance shafts connected to their respective crankshaft, although the engine half which can be disabled would not need a coaxial balance shaft if the other balance shaft would have a clutch that disengages when the engine is running on six cylinders.

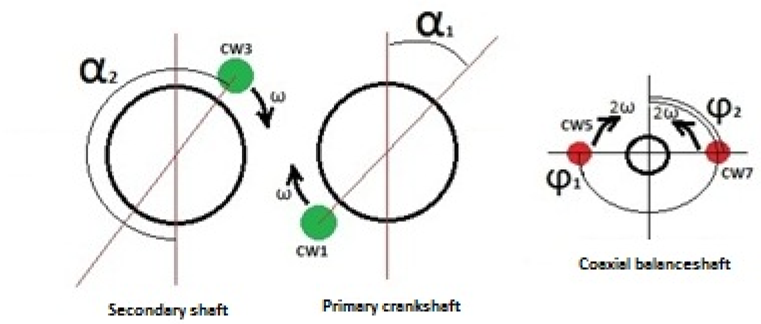

Figure 3.3 Conceptual sketch of the positions, angles and the different counter weights for balancing of the Inline3+3.

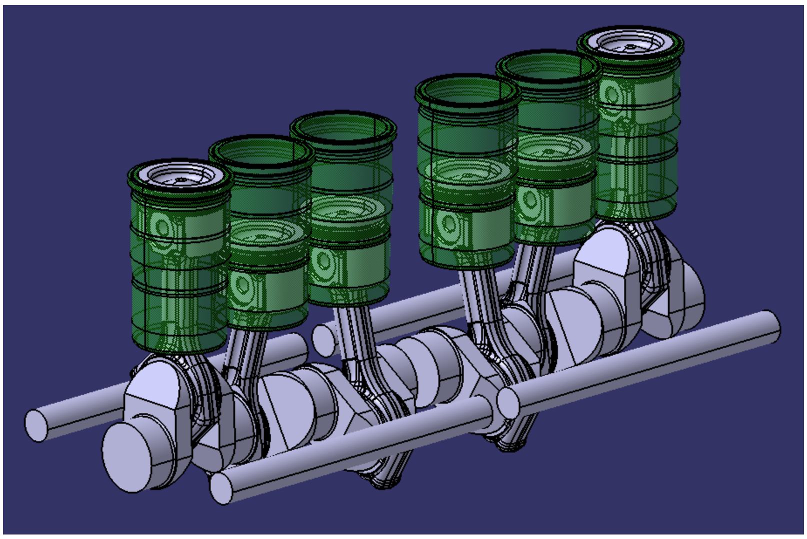

Figure 3.4 Conceptual model of the Inline3+3 engine.

3.4 Concept Parallel3+3

Parallel3+3 engine concept is two inline three cylinder engines located parallel to each other with dual crankshafts, spinning in opposite directions and connected by gears. An illustration of the concept is shown in Figure 3.5. The idea is to neutralize the unbalanced rocking moments using symmetric configuration. The motive with this concept is the same as the previous concept, which is to deactivate the complete half of the engine to decrease fuel consumption by reducing friction losses and the engine will also operate in a more efficient operating point due to the increased load. The design also allows the engine to be very compact which is beneficial for packaging. The shorter crankshafts are an advantage as the weight of the crank shafts can be reduced because the torsional deflection will be less than that of an inline six cylinder engine. The more compact engine design will also result in a more robust construction which could reduce noise and vibration propagation.

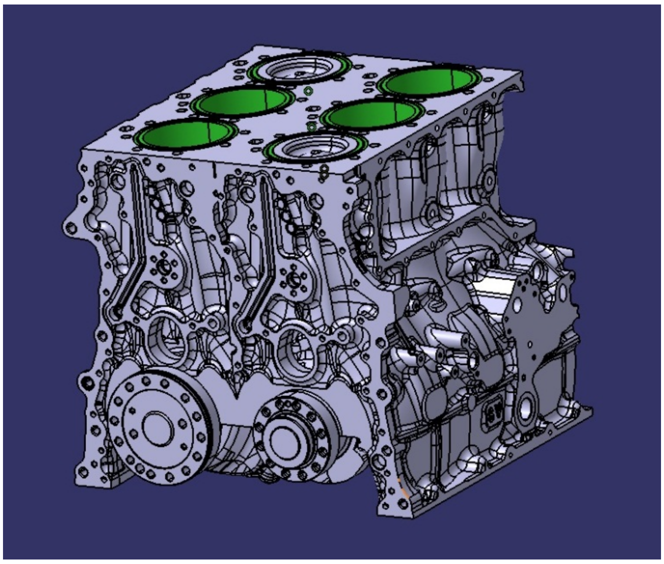

Figure 3.5 Conceptual model of the Parallel3+3 engine.

The rigid body analysis in Adams/View will allow the user to find the most optimal balancing design utilizing the DOE tool. The optimization will be done considering a compromise between the operating modes, dual crankshaft or cylinder deactivated mode. The counter weights will be modelled as point masses in the system attached with a mechanical link constraint to the crankshaft where the mass and position can be varied according to find the most favourable combination.

In order to broaden the analysis of the Parallel3+3 design the concept will be investigated in two different engine configurations to find the most convenient model for the cause. One with the firing order 1-5-3-6-2-4 Configuration 1 and another one with the firing order 1-5-3-4-2-6 Configuration 2. The cylinder numbers are shown in Figure 3.6.

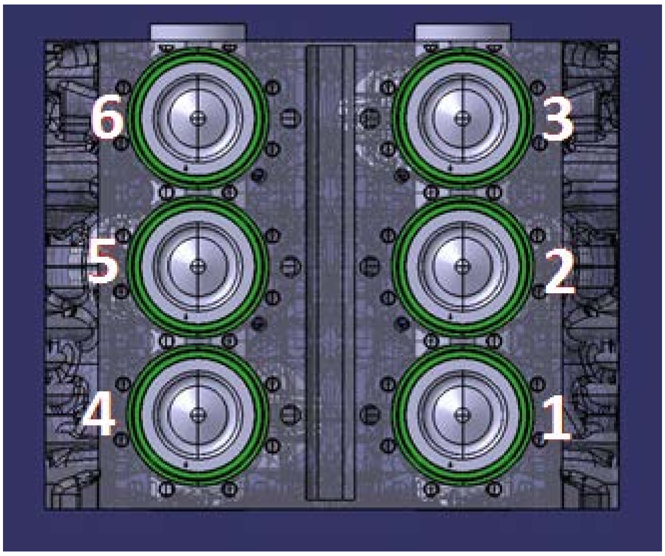

Figure 3.6 The cylinder numbers for concept Parallel3+3.

Configuration 1 will be operating in a more symmetric point of view in two planes considering balancing the moments induced by each inline three cylinder engine shown in Figure 3.7. The distance between the crankshafts is 300 mm ensuring that each crankshaft can rotate freely even if one of them is completely deactivated.

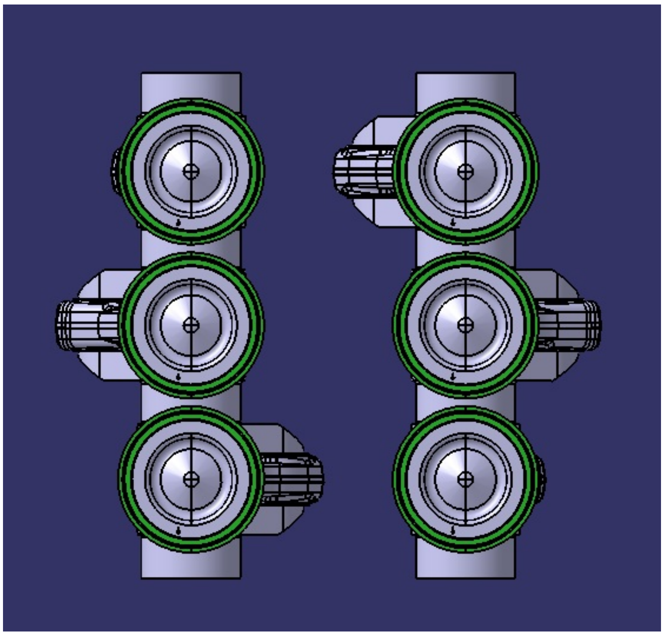

Figure 3.7 Conceptual model on Parallel3+3 crankshaft configuration 1.

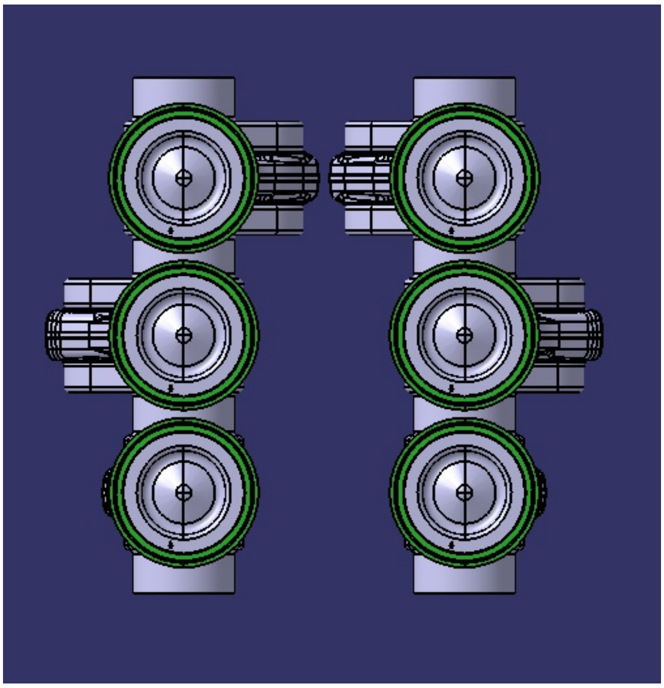

Figure 3.8 Conceptual model on Parallel3+3 crankshaft configuration 2.

The firing order in Configuration 2 allows only a symmetric setup in one plane unlike the previous configuration, shown in Figure 3.8.

The approach will be to optimize the mass and the location of the counter weights as both crankshafts of the engine are operating. Once the half of the engine becomes deactivated there will be two additional coaxial balance shafts operating to balance the moments in several orders for both a conventional inline three cylinder engine and the overcompensated moments induced by the designed counter weights. The coaxial balance shafts are needed because two shafts are required to control the correct imbalance of the system for each order. The first order part of the coaxial shafts will operate in the engine speed as the second order part will operate twice the speed of the crankshaft. The coaxial shafts will only operate if the engine is in cylinder deactivated mode demanding a synchronizing unit as well as for the clutch coupling the crankshafts.

The position of the counter weights will be given in angles according to Figure 3.9, where the counter weights are positioned with the same dimensions as specified in Section 3.3. Figure 3.10 shows a CAD model of the concept Parallel3+3 configuration 1 to enlighten the coaxial balance shafts design.

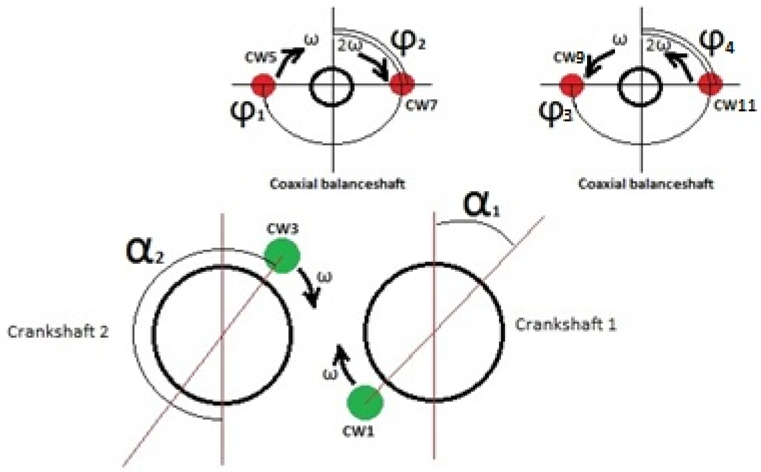

Figure 3.9 Conceptual sketch of the positions, angles and the different counter weights for balancing of the Parallel3+3.

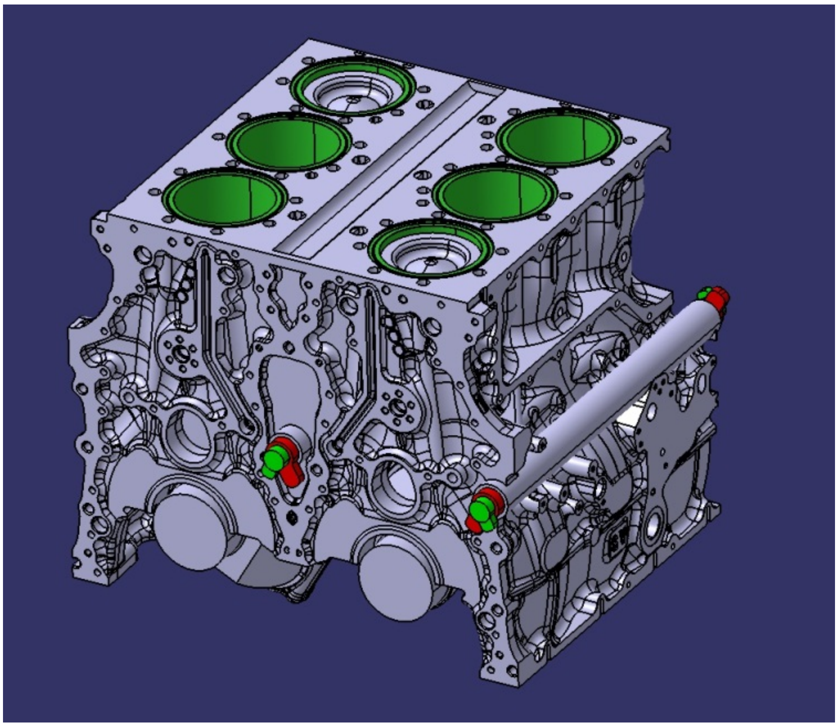

Figure 3.10 CAD model on concept Parallel3+3 configuration 1 with coaxial balance shafts.

4 Results

4.1 Validation of the reference model MD13

It has been discovered that small asymmetries have a significant impact on the results by utilizing the rigid body analysis as the parts in motion have high mass resulting in large error possibilities. The crankshaft of the MD13 model used in Adams/View, which is based on a CAD model assembly, was detected to have its centre of gravity displace from the theoretical rotational centre. This caused inaccuracy in the final results and was proven to be the source of this specific error, which was corrected by modifying the crankshaft in order to improve its location of centre of gravity. Figure 4.1 is showing the resulting forces by rotating the original MD13 crankshaft at a speed of 1200 rpm in comparison with Figure 4.2 showing the modified MD13 crankshaft where the results are as expected, consequently. The CAD model of the crankshaft is purposely made to have some more mass on its counterweights in order to be able to reduce some mass for fine tuning after manufacturing.

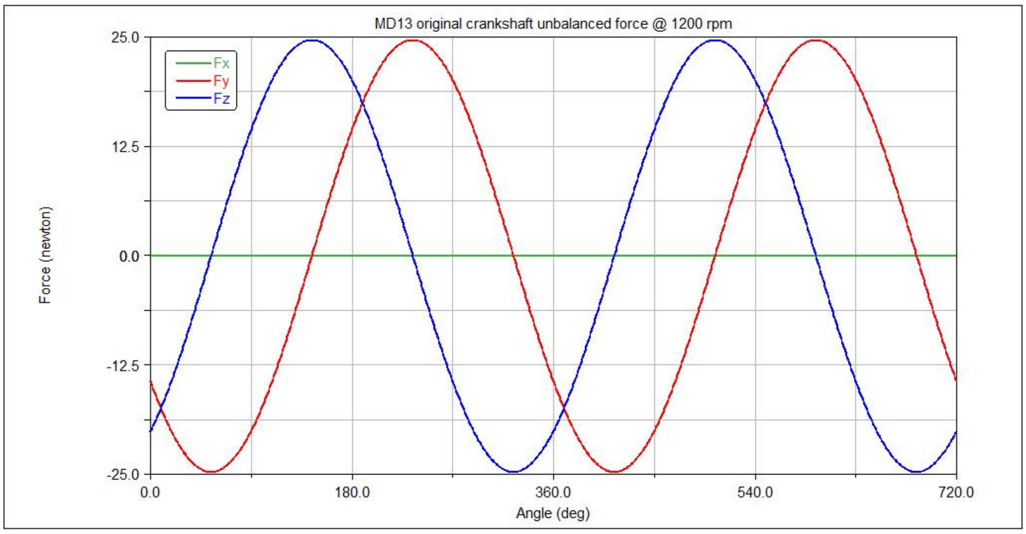

Figure 4.1 Unbalanced forces as a result from the original MD13 crankshaft based on the original CAD model.

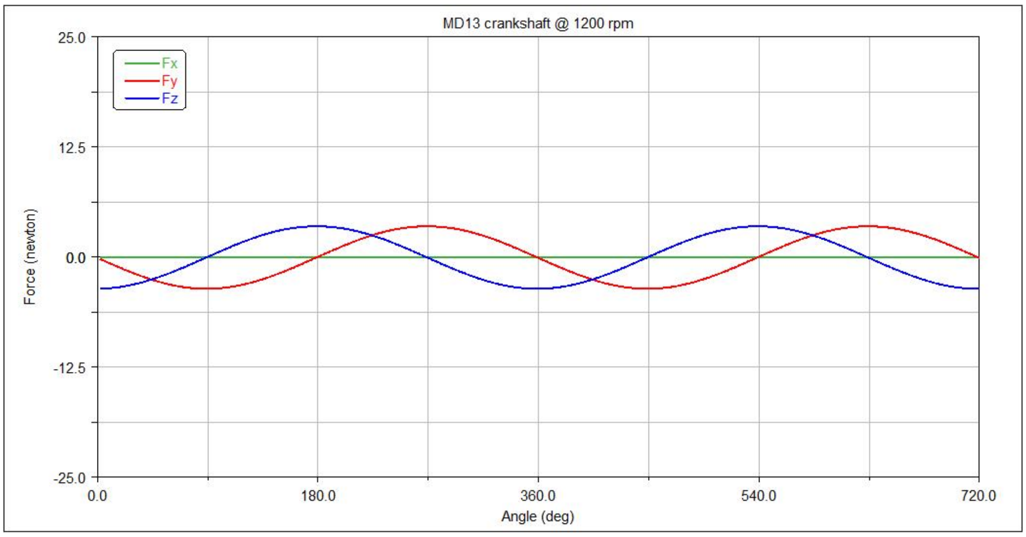

Figure 4.2 Unbalanced forces as a result from the modified MD13 crankshaft to be symmetrically balanced.

It has been shown in Section 2.3 that a six cylinder inline engine is rather well balanced, in pure mathematics considering a completely rigid engine. An additional rigid model has been analysed using GT-Suite to provide higher accuracy for of the validation, Figure 4.3 shows the resulting unbalanced forces. The results from Adams match the results from GT-Suite, which can be observed by comparing Figure 4.3 and Figure 4.4. Therefor the methodology and the MD13 model in Adams/View are considered to be valid.

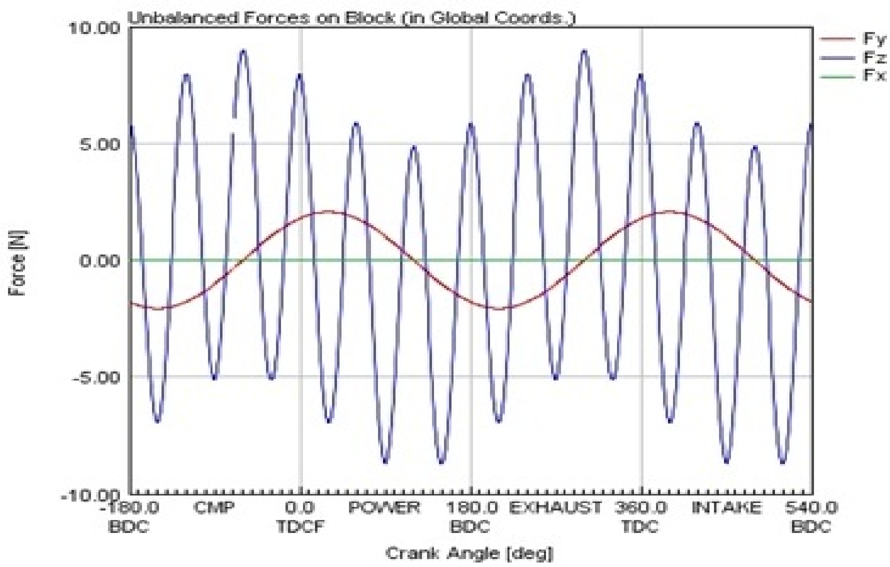

Figure 4.3 Unbalanced forces as a result for MD13 from GT-power.

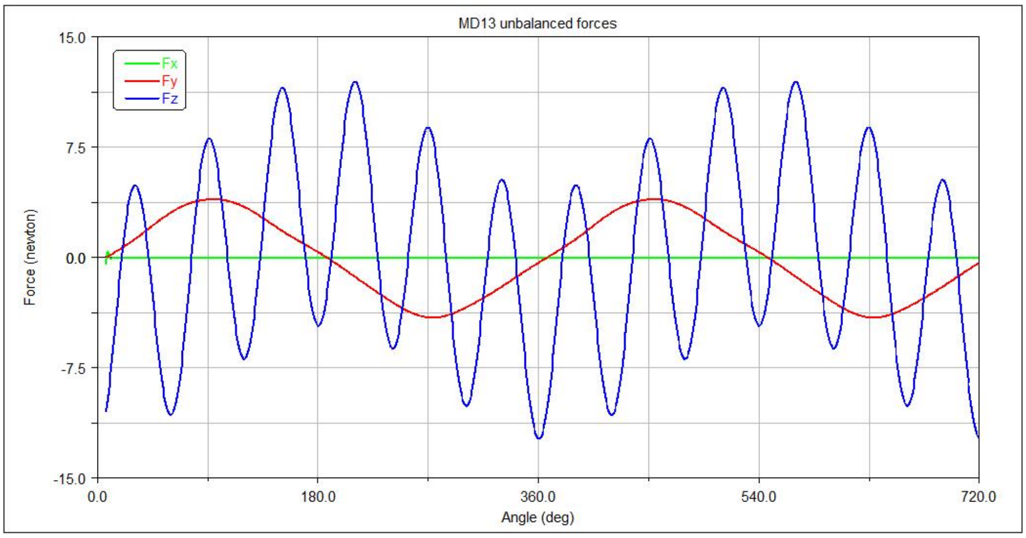

Figure 4.4 Unbalanced forces as a result for MD13 from Adams/View.

4.2 MD13 with flexible crankshaft and cylinder pressure

A study with a flexible crank was conducted in order to obtain the individual forces in the main bearings. It was done using the strategies shown in Section 3.2.1 and the results from that study are shown in Figure 4.5, Figure 4.6 and Figure 4.7. The first figure shows the main bearing forces without any counter weights, when comparing with Figure 4.6 it can be seen that the magnitude of the forces is lowered. Figure 4.7 shows the strategy for eight counter weights, which compared to the four counter weights setup has a lower force magnitude. The best results were obtained by a twelve weight configuration, shown in Figure 4.8.

The counter weights are placed with a counter clockwise offset from crank throw one, except for the twelve counter weight setup that has all of the weights placed 180 degrees from their adjacent crank throw. The final angle and masses of the different counter weights are shown in Table 4.1.

Table 4.1 Weights and angles for the different balancing strategies.

Strategy	Amount of counter weights	Weights (kg)	Angles (deg)
1	Zero	-	-
2	Four	4 * 7.167 kg	124.8° and 304.8°
4	Eight	8 * 4.24 kg	136.7° and 316.7°
5	Twelve	12 * 5.408 kg	180°, 60° and 300°

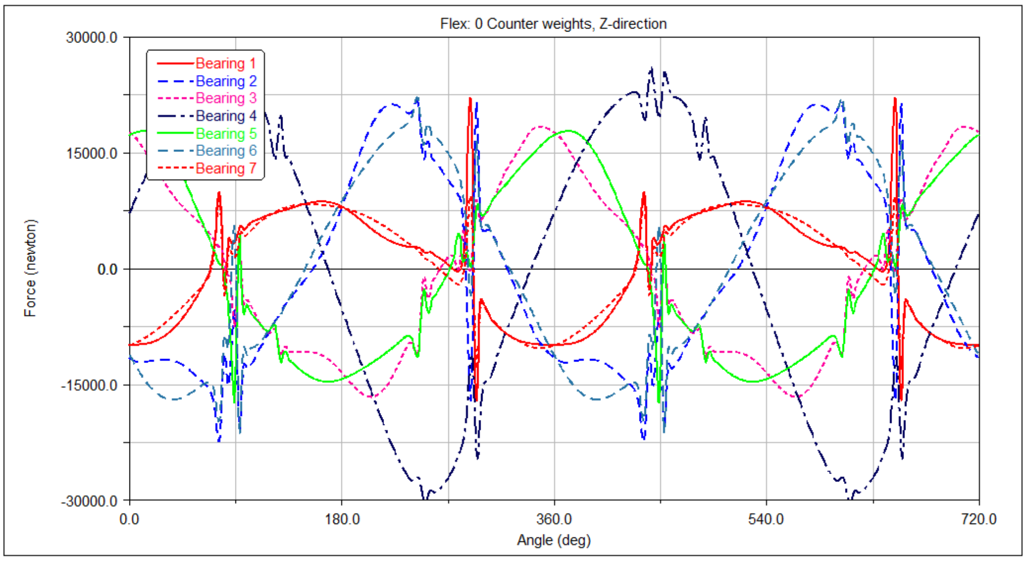

Figure 4.5 Main bearing forces in Z-directions with 0 counter weights.

The figures show all the forces in the main bearings in Z-direction and a comparison is done by comparing the peak levels of each bearing force to the corresponding force of the other counter weight strategies. Introducing a flexible crankshaft makes the model complex which increases the likelihood of numerical errors, which could be the explanation to the sharp peaks seen in Figure 4.5 to Figure 4.8.

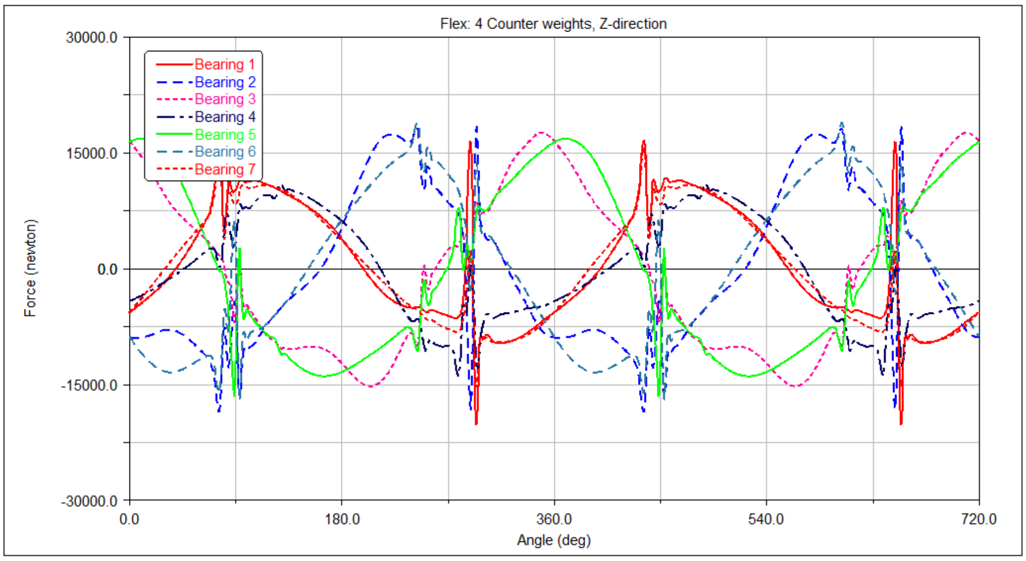

Figure 4.6 Main bearing forces in Z-directions with 4 counter weights.

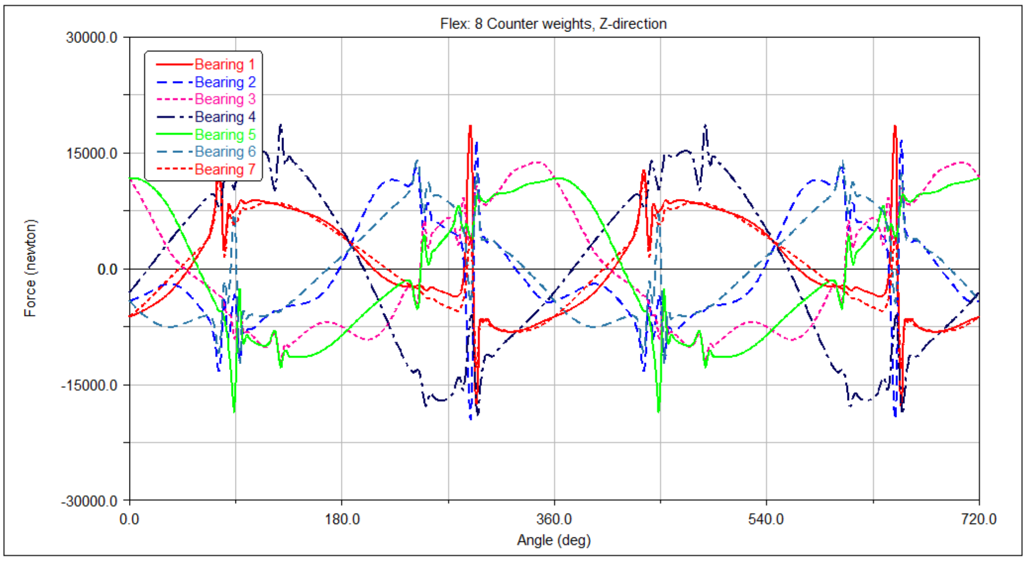

Figure 4.7 Main bearing forces in Z-directions with 8 counter weights.

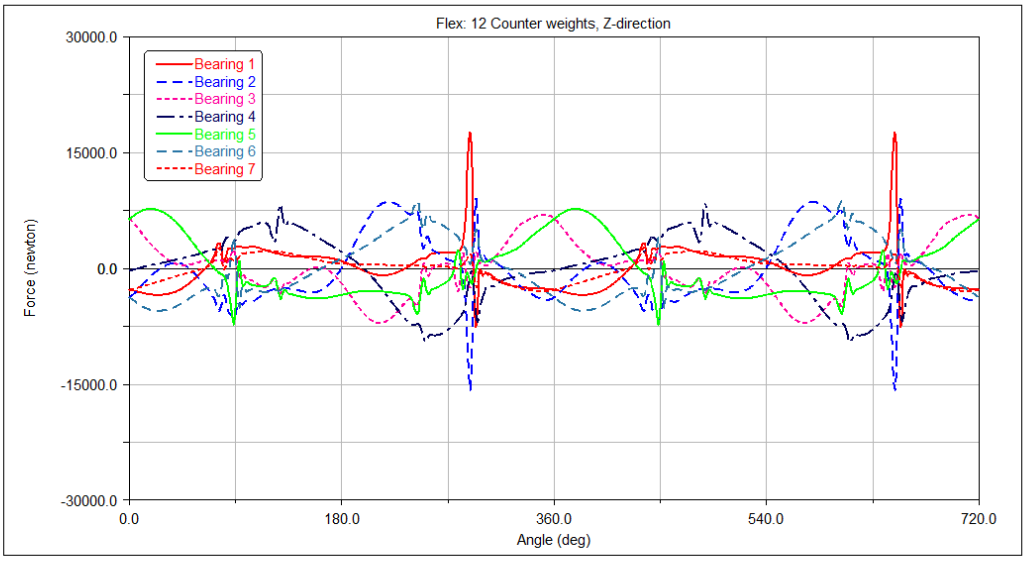

Figure 4.8 Main bearing forces in Z-directions with 12 counter weights.

As seen in Section 2.3 the individual cylinder gas pressure can be seen as containing progressively decreasing magnitudes as the order frequency increases. In Figure 4.9 the effects of running a six cylinder engine is shown. Here it can be seen that at 60, 120, 180 and 240 Hz the force from the individual cylinder remains. These peaks correspond to the third, sixth, ninth and twelfth order which shows that those orders of vibrations on a six cylinder engine originate from the gas forces.

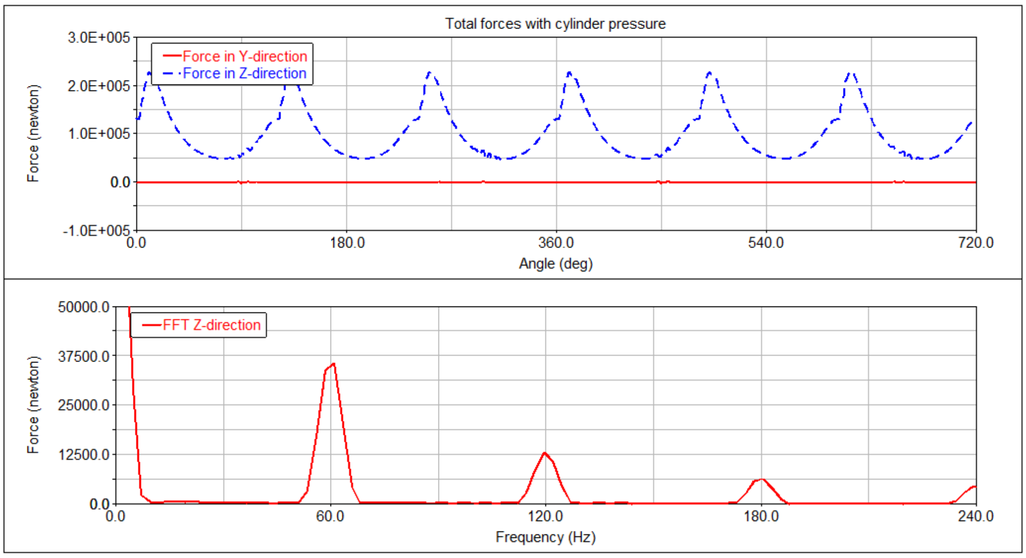

Figure 4.9 Total force for the system with flexible crank and cylinder gas pressure.

4.3 Inline3+3

When running all the six cylinders of the Inline3+3 the engine operates like a conventional six cylinder engine, which means that it is balanced in both total force and total moment. Problems occur when deactivating one half of the engine, which means running it as a three cylinder engine. As seen in Figure 4.10 the three cylinder engine will still be balanced in total force, but as shown in Figure 4.11 the acting moment will not be balanced. To be able to totally balance the acting moment both the crankshafts and the secondary shaft needs to be used, but these can only cancel first order moments. The system needs an additional coaxial balance shaft in order for it to be fully balanced. The coaxial balance shafts utilize two counter rotating shafts, as described in Section 3.4. The masses and angles of the positions of the counter weight are tabulated in Table 4.2.

Table 4.2 Counter weights needed per half of the engine.

Cylinder deactivated mode				
CW No.	**Notation**	**Angular position**	**Opposed unbalance**	**Mass**
1 and 2	α_1	30.0° and 210.0°	1st order moment	8.169 kg
3 and 4	α_2	330.2° and 150.2°	1st order moment	1.717 kg
5 and 6	φ_1	150.3° and 330.3°	2nd order moment	0.2379 kg
7 and 8	φ_2	-150.3° and -330.3°	2nd order moment	0.2379 kg

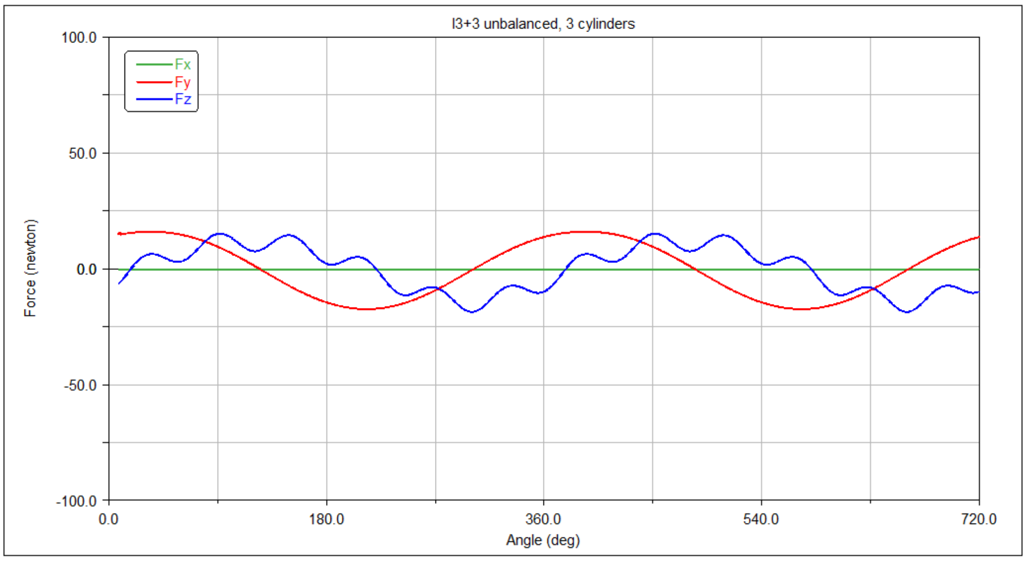

Figure 4.10 Total forces for an engine running three cylinders.

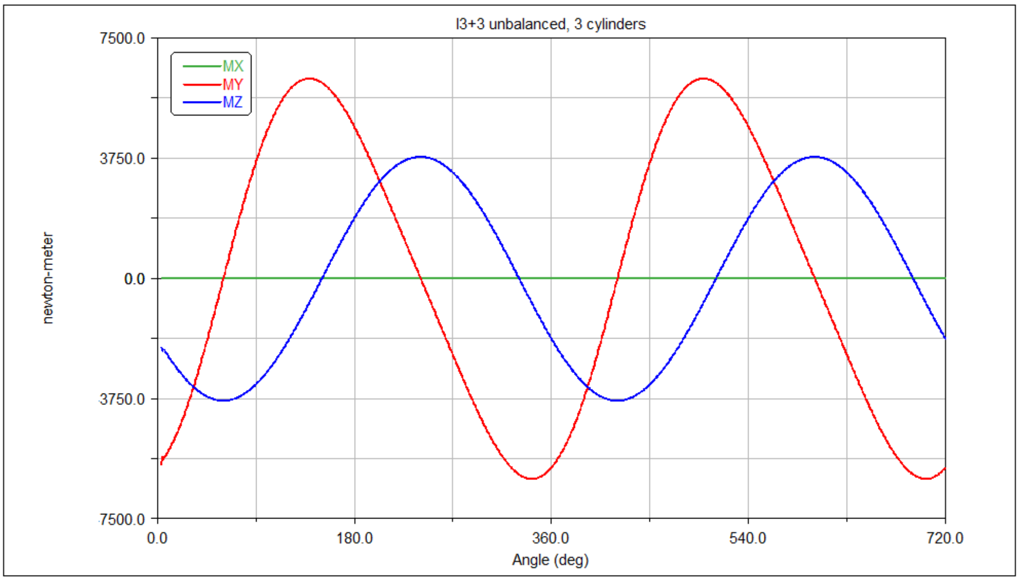

Figure 4.11 Acting moment on an engine running three cylinders.

The moment unbalance for the deactivated engine with the optimized counter weights is as shown in Figure 4.12. When activating the entire engine the acting moment remains in an acceptable level, which can be seen in Figure 4.13.

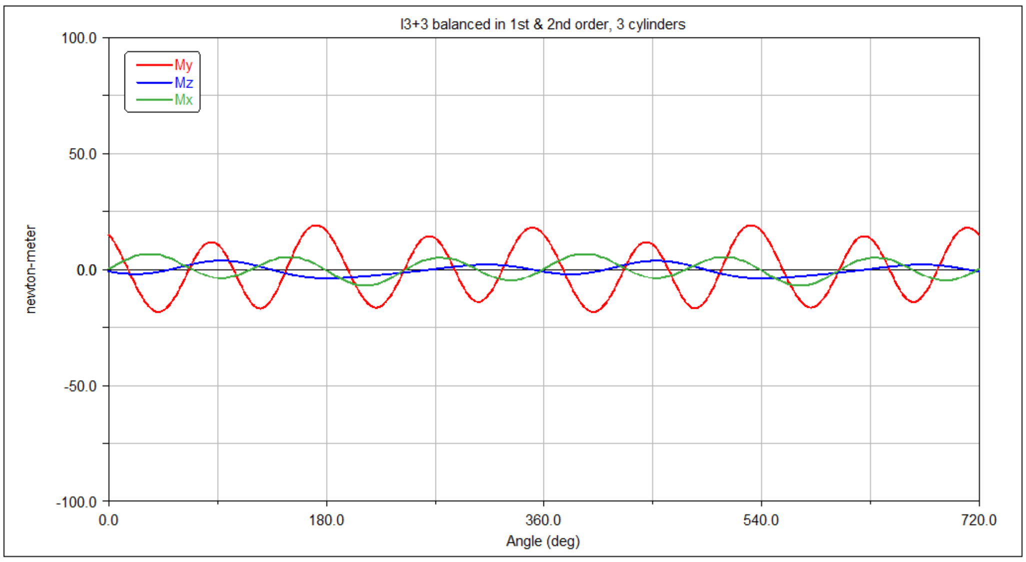

Figure 4.12 Acting moment on the balanced engine when running three cylinders.

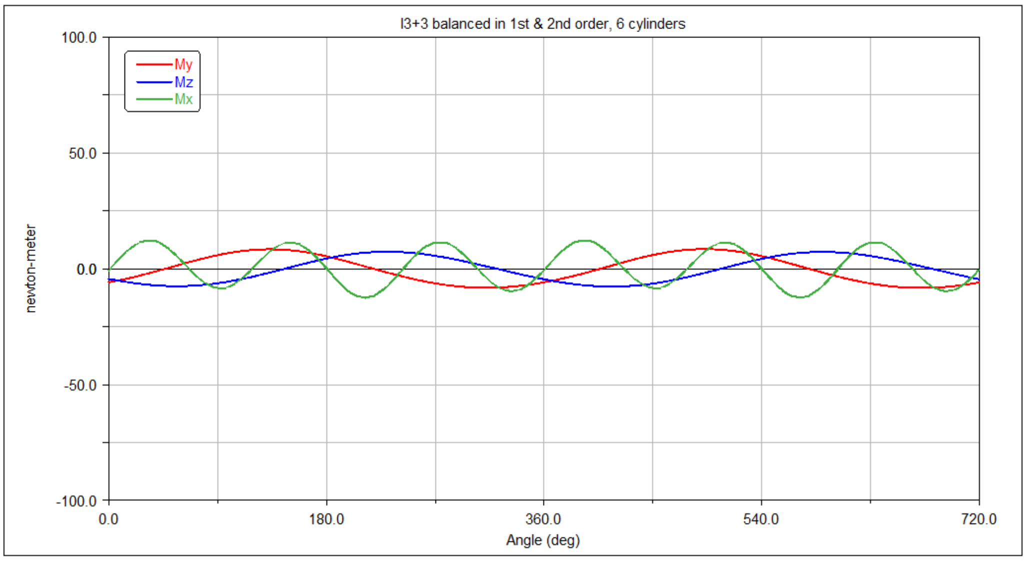

Figure 4.13 Acting moment on the balanced engine when running six cylinders.

4.4 Parallel3+3 configuration 1

The concept Parallel3+3, explained in Section 3.4, is considered to be well balanced considering the total forces as it behaves like two well-balanced three cylinder engines, shown in Figure 4.14. However the balancing issue regards the acting moment that is developed due to each three cylinder side, as explained for the previous concept in Section 4.3.

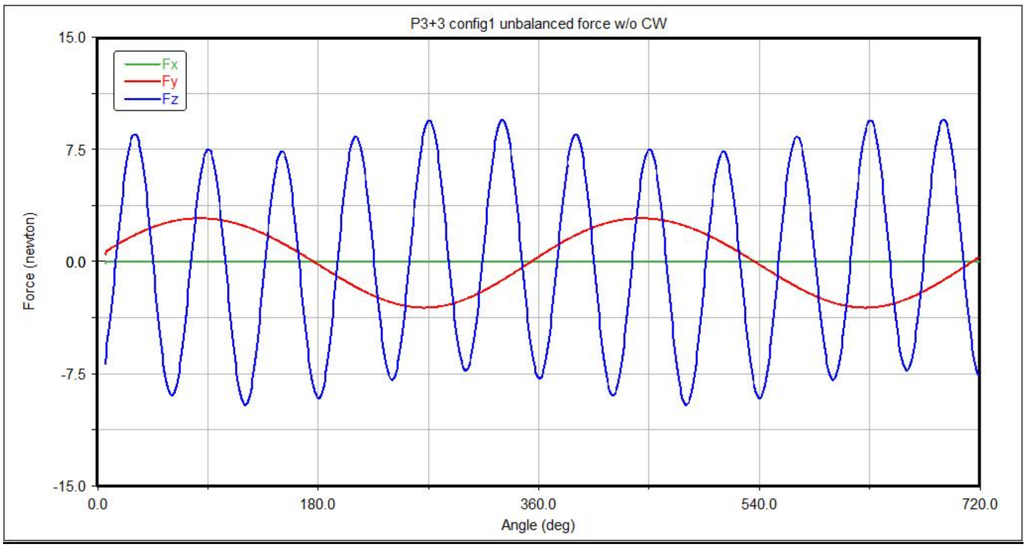

Figure 4.14 Unbalanced forces as a result for Parallel3+3.

Using the Design of Experiment tool in Adams/View the optimized parameters of the balancing system for the dual crankshaft mode were derived, presented in Table 4.3. The counterweights are shown to be optimal with a mass of nearly 7 kg which is considered to be within reasonable limit. They are also proven to be working optimally with an offset of nearly 180 degrees to each other. The reason for the angles and masses to not having round values or exactly 180 degrees offset is due to the minor irregularities in the different parts and their mass properties induced by the different conversions before importing to Adams/View. The parameters defines the position, according to Figure 3.9 in section 3.4, and the mass of the counter weights. The balancing system of the Parallel3+3 concepts is only considering the moment imbalance for both configurations as the total force imbalance for the rigid analysis is considered to be well-balanced.

Table 4.3 Optimized parameters for Parallel3+3 configuration 1 operating in dual crankshaft mode.

Dual crankshaft mode				
CW No.	**Notation**	**Angular position**	**Opposed unbalance**	**Mass**
1 and 2	α_1	210.1° and 30.1°	1st order moment	7.051 kg
3 and 4	α_2	149.8° and 329.8°	1st order moment	7.051 kg

Using the optimized parameters the moment imbalance for the dual crankshaft mode is reduced to be fairly accepted to be balanced, shown in Figure 4.15. As the whole engine is operating there is only a desire to balance out first order imbalance, which is favourable as the crankshafts speed is in the same frequency where the counter weights will be positioned. The advantage of having a symmetric configuration is shown to balance out the higher order imbalances.

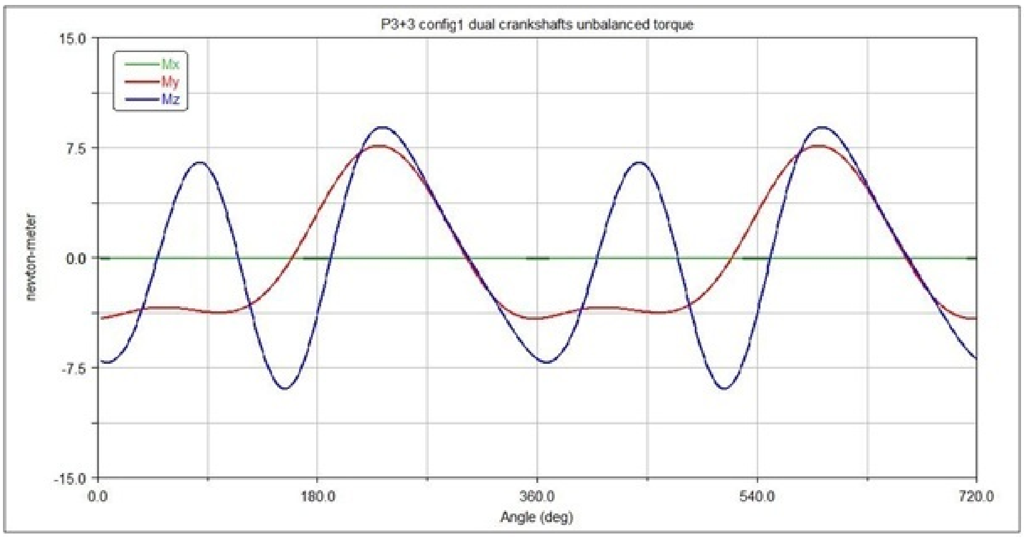

Figure 4.15 Unbalanced moments for Parallel3+3 configuration 1 operating in dual crankshaft mode using optimized counterweights.

In the cylinder deactivated mode the two coaxial balance shafts will be engaged to balance out the imbalances induced by the three cylinder moments and the predesigned counterweights. The optimized parameters are shown in Table 4.4. As only one half of the engine is operating the symmetric advantage cannot be utilized in this case, resulting in an additional demand on second order balancing system. This will be managed by one part of both coaxial shafts. By inserting the optimized parameters the results are proven to be acceptable shown in Figure 4.16, however there will be a torque acting around the x-axis as the measuring point is in-between the two crankshafts. On the contrary the torque around the x-axis is considered to be more convenient to manage as there are torque irregularities on the crankshaft originating from the compression/expansion pulses of the cycles.

Table 4.4 Optimized parameters for Parallel3+3 configuration 1 operating in cylinder deactivated mode.

Cylinder deactivated mode				
CW No.	**Notation**	**Angular position**	**Opposed unbalance**	**Mass**
5 and 6	φ_1	-211.0° and -31.0°	1st order moment	2.505 kg
7 and 8	φ_2	-152.0° and -332.0°	2nd order moment	0.2077 kg
9 and 10	φ_3	211.0° and 31.0°	1st order moment	2.505 kg
11 and 12	φ_4	152.0° and 332.0°	2nd order moment	0.2077 kg

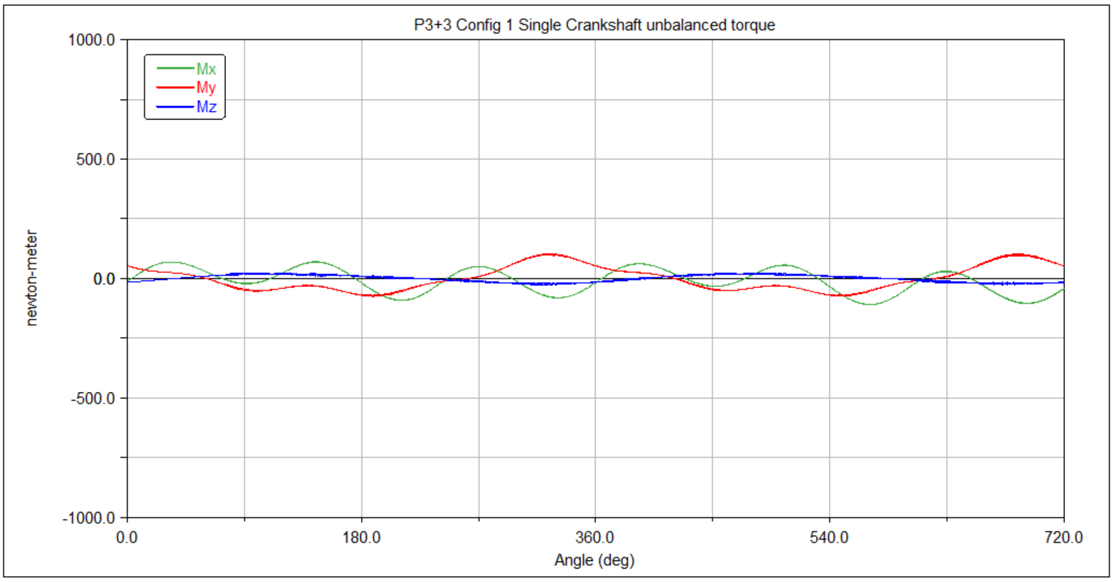

Figure 4.16 Unbalanced moments for Parallel3+3 configuration 1 operating in cylinder deactivated mode using optimized counterweights

4.5 Parallel3+3 configuration 2

Although the concept Configuration 2 has been investigated and optimized which it turns out to be a less preferable alternative than Configuration 1. To clarify Figure 4.17 is showing the acting moments without any counter weights. It can be seen that the moment around the y-axis is not acting as a pure first order moment which means that the system demands a second order balance shaft even for the dual crankshaft operation mode despite Configuration 1 works well without them.

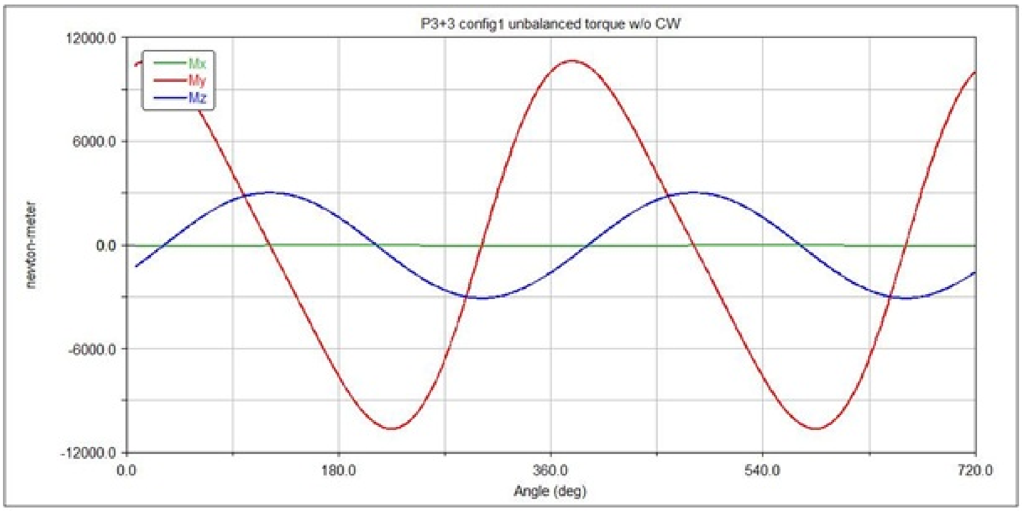

Figure 4.17 Unbalanced moments for Parallel3+3 configuration 2 without any counter weights.

After further optimization using the DOE tool there have been some improvements using the parameters presented in Table 4.5. The result can be seen in Figure 4.18 that there is a pure second order moment around the y-axis confirming the theory from the previous figure. Also the remaining moments are unacceptably high even in this stage where the counter weights are optimized to be over 8 kg. Compared to the configuration 1 counterweights which are 7 kg. Once the cylinder deactivation mode is engaged one can realize need for more balancing shafts or a more complex balance system. Based on the fact and motivation to reduce the friction losses and the complexity level, regarding the weight and also the wear issues, the conclusion is made that Configuration 1 is the preferable and more reasonable option to choose.

Table 4.5 Optimized parameters for Parallel3+3 configuration 2.

Cylinder deactivated mode				
CW No.	**Notation**	**Angular position**	**Opposed unbalance**	**Mass**
1 and 2	α_1	30.0° and 210.0°	1st order moment	8.363 kg
3 and 4	α_2	330.0°and 150.0°	1st order moment	8.363 kg

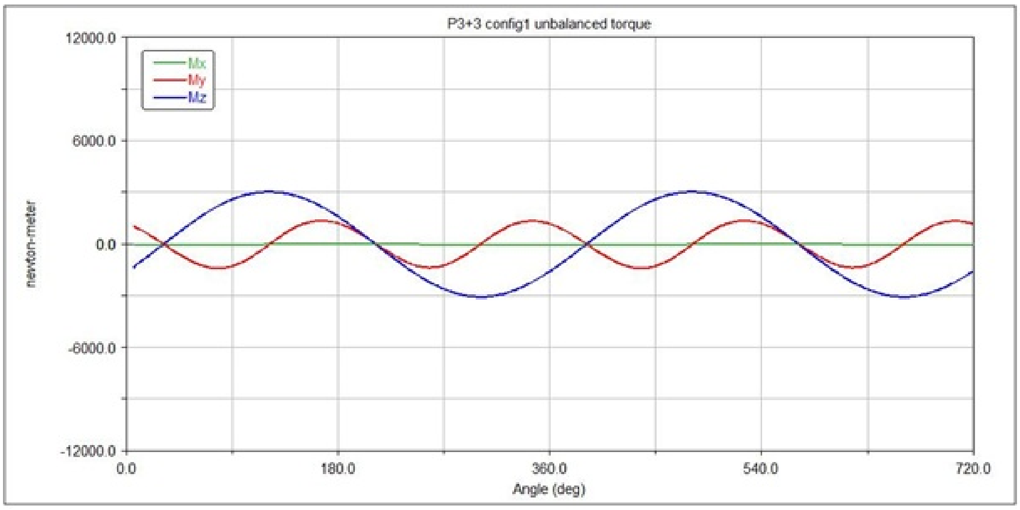

Figure 4.18 Unbalanced moments for Parallel3+3 configuration 2 with partly optimized counter weights.

5 Discussion

In this chapter the usage and performance of the chosen software is discussed. The modelling process were proven to be more tedious and sensitive to inputs than expected. Also the optimization process is not as convenient as desired in most of the cases.

5.1 Modelling in Adams/View

The inconvenience to build a model directly in Adams/View, as it has an inferior modelling ability in comparison with any CAD software, the choice of approach were to import a final CAD model of the engine design into Adams/View. Somehow the import in step file format did not work properly as some mass properties were lost in the transformation process. However using Parasolid as file format for importing models turned to be working properly, most of the times, although the need to resave a step file created in Catia using ProE Creo elements as Catia demands another license to be able to save a Parasolid format outright. Once a successful file import was made to Adams/View all mass properties and constraints has to be implemented which appeared to be very time consuming. At this point any minor change requires to be made in Catia and the whole process must be repeated. Once the model CAD is implemented, Adams/View gets the job done analysing the unbalanced forces and moments in its very generic approach. However there are some more setbacks, complex geometries and extensive models challenges the performance capability of the software making it very slow, giving numerical inaccuracies on the results and very often make the program to shut down. It were discovered that in some cases it were more convenient to change some parts of a model to Adams/View default template parts and manipulating the mass properties to simulate the desired condition.

Another major source of error found in the process was the asymmetry in a part used in rotational motion where the smallest deviation between CoG of the part and the rotational centre caused a major disturbance on the results. For avoidance of this issue all rotating parts were double checked to be symmetric or redesigned if not, such as the crankshafts for instance.

5.2 Design of Experiments in Adams/View

The new concept designs i.e. Inline3+3 and Parallel3+3 contain several heavy parts in motion with relatively high speed the force and torque imbalance were produced in a larger scale, thereby making higher demands on the optimization process. There can be a discussion of whether the DOE process is the most optimal and efficient one or not, although by using the "zoom in" technique setting the parameter ranges and number of levels by intuition and in a progressively iterative way the process could be explained to be acceptably well. However the sensitivity of choosing the error tolerance, which was forced to be changed in some cases with severe driven models, the sensitivity of the choice of the step sizes and other parts under the construction makes a considerable impact on the end results and also gives some numerical errors. This consequently forced the process to be even more extensive and time consuming to end up with reasonable reliable results. For instance, due to numerical error before the simulation reaches steady state, the simulation time was chosen to be longer in order to reduce the influence of the errors as the results are converted into a RMS measure for the DOE process to find optimal variables.

5.3 Flexible versus rigid analysis

Running an analysis with a flexible crankshaft it is preferable that the analysis needs to go into detail on different forces acting on the individual component in the crank assembly. If the analysis is only focused on the overall balance of the system a rigid crankshaft is preferred in order to decrease the demand on the computational performance. The main reason for choosing a rigid crankshaft is the simplification of the model and simulation speed benefits it has over the flexible analysis. Therefore a rigid crankshaft is beneficial for simulations which are focused on the total forces instead of the internal forces.

6 Conclusion

This work presents a new generic approach to analyse vibrations and balancing solutions of novel engine designs in the software Adams/View independent of templates making the approach less limited in comparison to other conventional software tools for the same cause. The approach has been validated to give a good preface understanding of vibrations of new complex engine designs and their potential to be balanced.

The flexible analysis was simulated to be able to obtain the correct forces in the individual main bearings. The analysis showed that the usage of twelve counter weights instead of four would lower the bearing forces, although it would increase the weight of the crankshaft.

The cylinder deactivation concepts have the potential to decrease the friction on low through medium load, but due to added friction from the secondary shaft and the balance shafts the engine will have a decreased efficiency on medium through full load. The Inline3+3 concept also increases the length and width of the engine, which can be aspects that are a hindrance to the concept. Whereas the Parallel3+3 has a decreased overall length while increasing the width even more than the Inline3+3 does.

It has been discovered that the Parallel3+3 concept can be well balanced operating in both dual crankshaft mode and in cylinder deactivated mode by engagement of two additional coaxial balance shafts.

The balancing approach has shown its potential to understand the need of additional complexity for the cause of a well-balanced system for new complex concept design. It certainly facilitates the understanding of those new designs as the forces and acting moments can be distinguished from different motions and also derived from their order. This approach has also been utilized to investigate different crankshaft configurations in order to find the optimized design for balancing.

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Appendix A

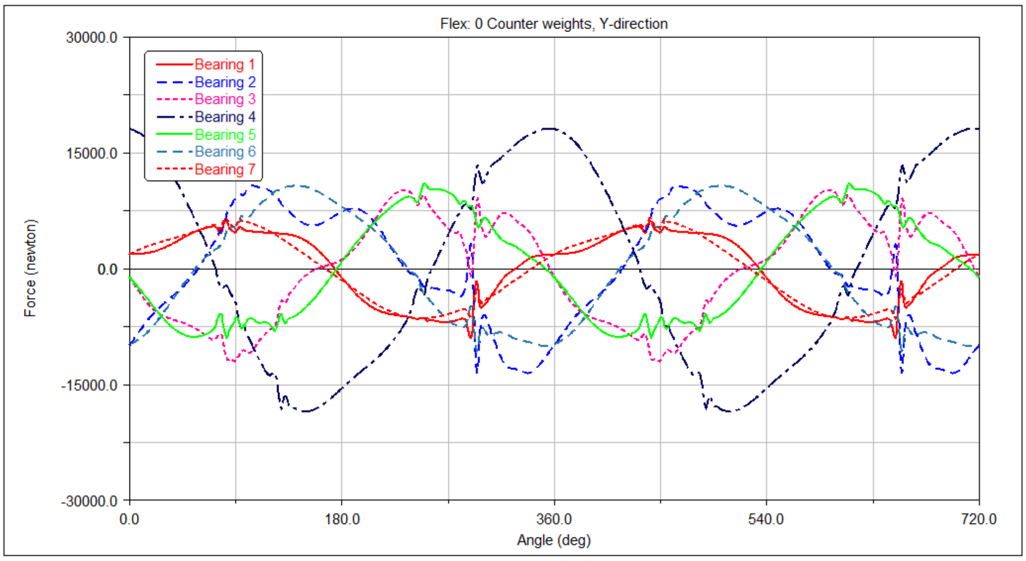

Figure A.1 Main bearing forces in Y-directions with 0 counter weights.

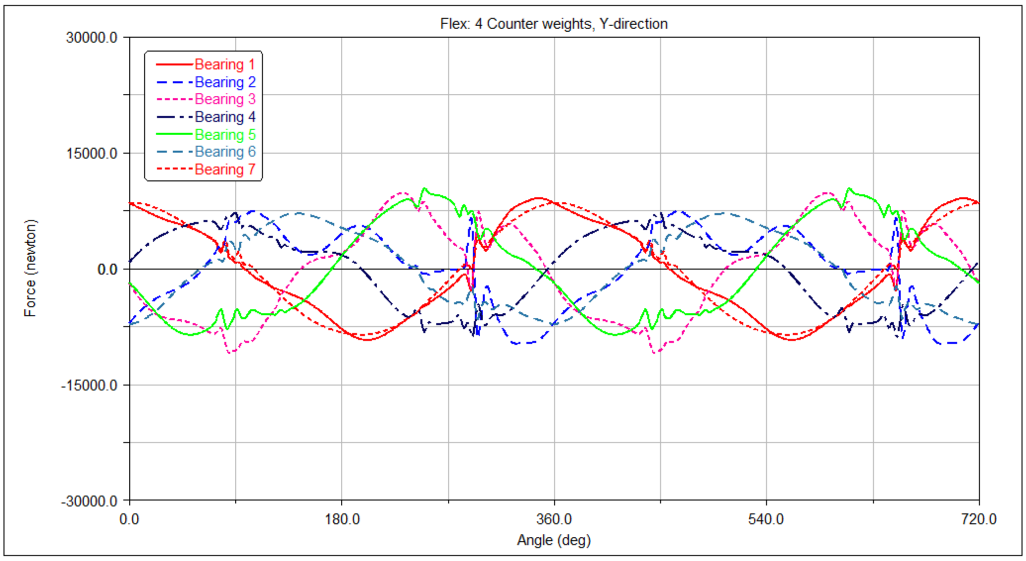

Figure A.2 Main bearing forces in Y-directions with 4 counter weights.

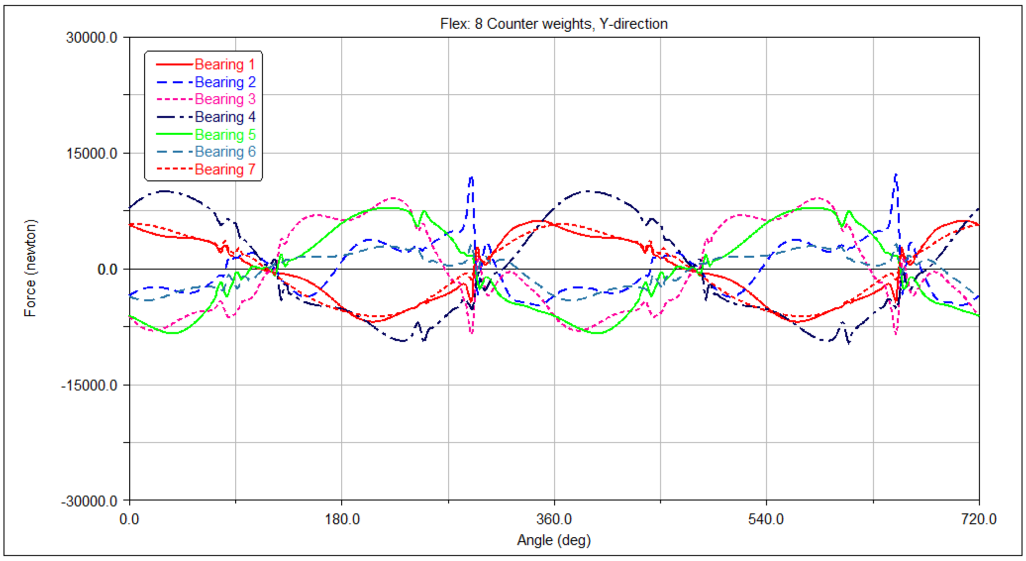

Figure A.3 Main bearing forces in Y-directions with 8 counter weights.

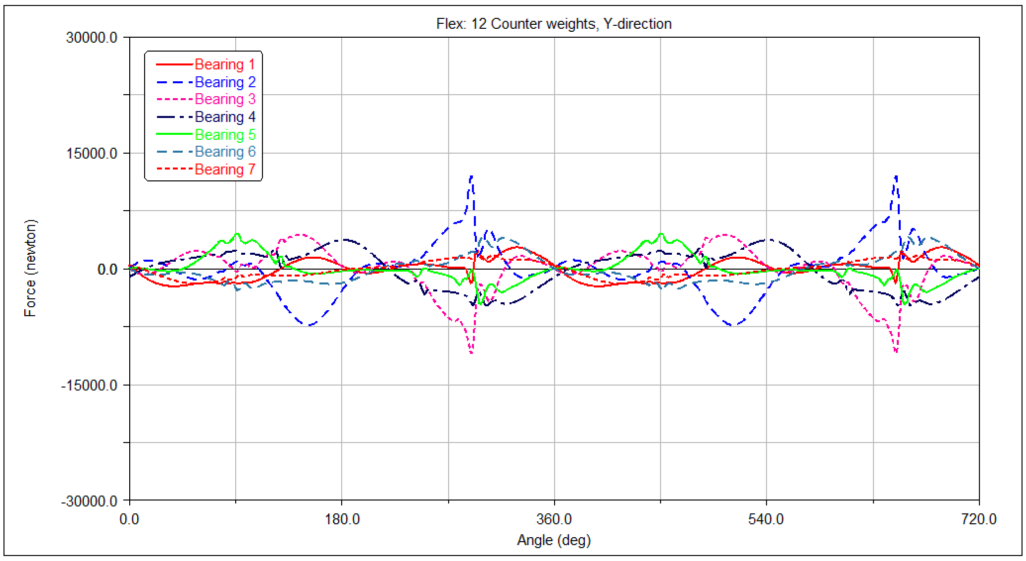

Figure A.4 Main bearing forces in Y-directions with 12 counter weights.