Cost and Weight Optimization of a Transport Refrigeration Compressor Crankshaft

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ABSTRACT

The main objective of this paper was to investigate weight and cost reduction opportunities for a Transport Refrigeration Compressor Crankshaft. These Compressors are generally run by a diesel engine operating at non-uniform speeds and unlike Hermetic compressor shafts they tend to act differently based on the engine characteristics. Torsional characteristics of the system and bearing stiffness effects on the shaft stress are to be considered while analyzing these crankshafts.

The objective being to outline the steps and various aspects considered and followed for the development of a cost and weight optimized and reliable crankshaft for transport refrigeration compressor. This study consists of five major areas: (1) dynamic load analysis, (2) static and cyclic stress analysis over complete cycle, (3) Fatigue analysis and verification (4) Torsional analysis of engine- coupling and clutch – compressor system, (5) Dynamic balancing and Balance weights topology optimization.

1. INTRODUCTION

The objective being to outline the steps and various aspects considered and followed for the development of a cost and weight optimized and reliable crankshaft for transport refrigeration compressor.

A benchmark dynamic simulation was conducted and verified on the existing forged production crankshafts and compared against the various design variants. The procedure consisted of determining dynamic operating forces. The pressure-volume diagram and compressor application specifications were used to calculate the load boundary condition in dynamic simulation model. The dynamic analysis was done analytically and was verified by simulations in CAE which resulted in the load spectrum applied to crankpin bearing. This load was then applied to the FE model in the simulation tool, to obtain Stress variation over the compressor cycle. The gas pressure and inertia forces acting on crankshafts generate both bending moments and torsional moments. Because the magnitude and phase relationships between the maximum bending moment and maximum torsional moment vary with speed and torque, analysis of stresses due to maximum bending and maximum torsion was done separately and verified by separate bending and torsion tests by strain gages attached to several locations on the crankshaft and comparing with an equivalent static measurement. Fatigue life analysis is conducted separately for both torsional and bending cyclic loads. Resonance excitation tuning fork method of fatigue testing at the maximum operating speed of the crankshaft is run, for accumulating test cycles at a higher rate than on running compressors. Modal analysis was performed to obtain the vibration signature and critical speeds of the shaft, the modal analysis were conducted for free-free, solid fixation and flexible fixation. Dynamic balancing was performed to optimize the counter balance weights, Topology optimization was used to get the optimum shape of balance weights.

Results achieved from aforementioned analysis were used in optimization of the crankshaft. Geometry, material, and manufacturing processes were optimized considering different constraints, manufacturing feasibility, and cost. The optimization process included geometry changes compatible with the regular established materials and processes, resulting in weight reduction, increased fatigue strength and reduced cost of the crankshaft, without changing any other compressor parts.

2. BACKGROUND

The Crankshaft is a critical component with a complex geometry in the compressor, which converts rotary motion to the reciprocating displacement of the piston with a four link mechanism. Since the crankshaft experiences a large number of load cycles during its service life, fatigue performance and durability has to be considered in the design process. Design developments have always been a challenge to manufacture a less expensive component with the minimum weight possible and proper fatigue strength and other functional requirements.

Crankshaft experiences large forces from gas compression. This force is applied to the top of the piston and is transmitted to the crankshaft through the connecting rod. The magnitude of the force depends on many factors which consist of crank radius, connecting rod dimensions, and weight of the connecting rod, piston, and pin. Gas compression and inertia forces acting on the crankshaft cause two types of loading on the crankshaft structure; torsional load and bending load. Due to the crankshaft geometry and compressor mechanism, the crankshaft fillet experiences a large stress range during its service life. The load from the piston is transmitted to the crankpin, causing a large bending moment on the entire geometry of the crankshaft. At the root of the fillet areas stress concentrations exist and these high stress range locations are the points where cyclic loads could cause fatigue crack initiation, leading to fracture. Therefore, fillet areas are locations that experience the most critical stresses during the service life of the crankshaft. In a compressor, two load sources apply force on the crankshaft. The load applied by gas compression in the cylinder to the piston is transmitted to the crankpin bearing by a four bar slider-crank mechanism. This is the main source of loading in the compressor. The other load source is due to dynamic nature of the mechanism. Since the compressor operates at high speeds, the centrifugal forces are present at different rotating components such as connecting rods. These load sources apply both torsional and bending load on the crankshaft.
The common failure in the crankshafts is mechanical crack nucleation at the fillet radius of journal bearings.

3. DYNAMIC LOAD ANALYSIS

The main objective of this part is to determine the magnitude and direction of the loads that act on the bearing between connecting rod and crankshaft, which was then used in the FEA over an entire cycle. An analytical approach was used on the basis of a single degree of freedom slider crank mechanism. The analytical approach was solved for a general slider crank mechanism which results in equations that could be used for any crank radius, connecting rod geometry, and connecting rod mass, connecting rod inertia, speed, piston diameter, piston and pin mass, pressure inside cylinder, and any other variables of the compressor.

The slider-crank mechanism with a single degree of freedom considered for solving the equations of motion. These equations where used in analytical calculation and provided the values of angular velocity and angular acceleration of the connecting rod, linear acceleration of center of gravity of the connecting rod, and forces at the connecting rod-piston bearing and connecting rod-crankshaft bearing. The pressure versus crank angle for a high compression load was considered. The analytical approach used in this study was verified by 2D dynamic simulation of the crankshaft, connecting rod, and piston assembly. For the purpose of this simulation crankshaft and connecting rod and piston assembly were digitized and the generated geometries were used to obtain the accurate location of the center of gravity of the parts and the magnitude of its inertia. The stick model created to perform transient analyses in ADAMS is shown in Figure 3.1.

These analyses provided the following information as a function of crank angle.

- Pin loads
- Bearing loads
- Angular velocity
- Input torque
- Free forces
- Free moments

The results from ADAMS software are plotted in Figures 3.2 through 3.5 and compared with that of the analytical approach. The outputs indicated agreement of the results from analytical programming with results from ADAMS software.
Figure 3.2 shows the variation crank pin forces over one complete cycle of the compressor.
Figure 3.3 shows the variation of bearing loads over an entire cycle;
Figure 3.4 shows the variation of torque
Figure 3.5 shows the free moments.

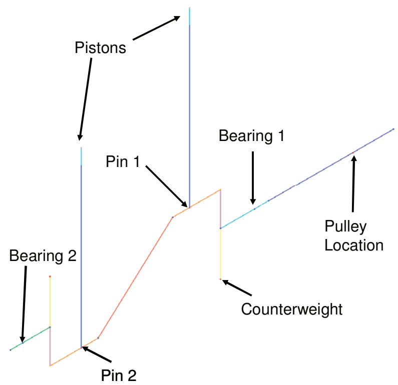


Figure 3.1: Stick model of mechanism

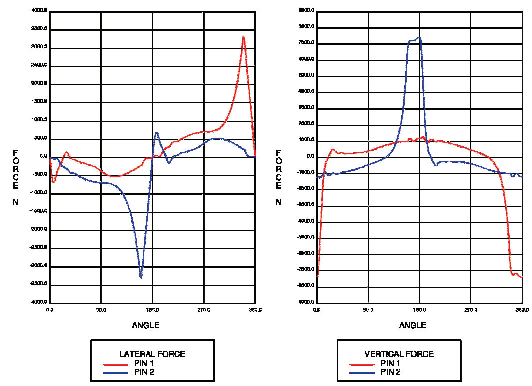


Figure 3.2: Crank Pin Forces

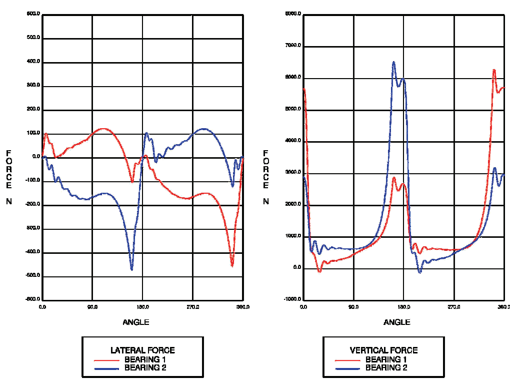


Figure 3.3: Bearing Forces

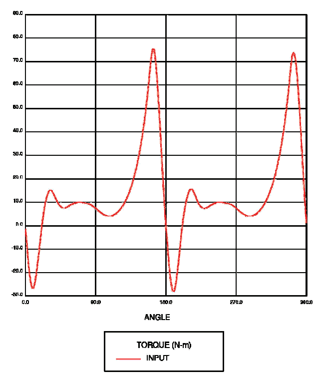


Figure 3.4: Shaft Torque

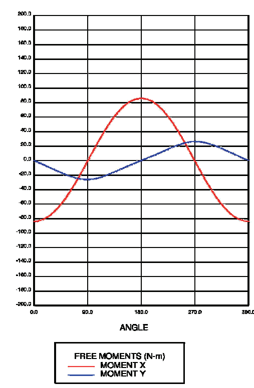


Figure 3.5: Free Moments

4. TORSIONAL SYSTEM ANALYSIS

This part presents the analysis of a torsional system. Diesel engines are used in Transport refrigeration units. When planning a device using a diesel engine as the prime mover, it is important to ensure that no critical speed is present within or in the vicinity of the range of revolution at which the driven machine is driven. To avoid the risk of torsional vibration, it is necessary to either avoid the critical speed by changing the natural frequency or to suppress the amplitude of vibration to a lower level. There are three methods to change the natural frequency: by changing the driven shaft system, by changing the equivalent length of the shaft system by inserting a flexible joint between the engine and the driven machine or by changing the equivalent mass. To change the amplitude of vibration, a damper is used. To estimate the natural and damped frequencies of the Torsional system, Complete Torsional system analysis is used for the Engine- Coupling and compressor. The variation of torque profile is used as input loading condition for anlysing the torsional fatigue strength of the shaft. Various parameters are iteratively changed in the Torsional system program to design the system to avoid the possible risks of torsional vibrations. Torsional system data and output torque charts of the engine have been obtained directly from Engine Manufacturers (Figure 4.1). Pistons tangential torque as well as inertia forces had to be designed approximately to accomplish desired and known output torque. Simulink shape quantity fluctuation description function has been used for the engine (pistons) torque description. From this engine harmonic excitation frequencies are obtained (Table 4.1).

From the compressor system torsional data and load torque data, Harmonic components of compressor shaft torque at that particular speed is obtained (Table 4.2). Moment of inertia of clutch and coupling assembly are calculated from the 3D models. Rubber bushings are used in the coupling/clutch for the engine and the compressor misalignment elimination and also for isolation of vibration. The radial stiffness of bushing has been calculated from the known torsional stiffness of the coupling. The belt stiffness has been measured under different conditions on a test stand.

Matlab program has been written for free vibration calculation to estimate the natural frequencies of the system. Several models have been prepared for simulation of the forced vibration of the systems. The first steps are equations of motion describing the system. Then the Laplace transformation is used to create equations that is transformed to Simulink blocks system.
Table 4.3 shows the natural frequencies of the Torsional system and Figure 4.2 shows the damped torque characteristic of the compressor shaft system.

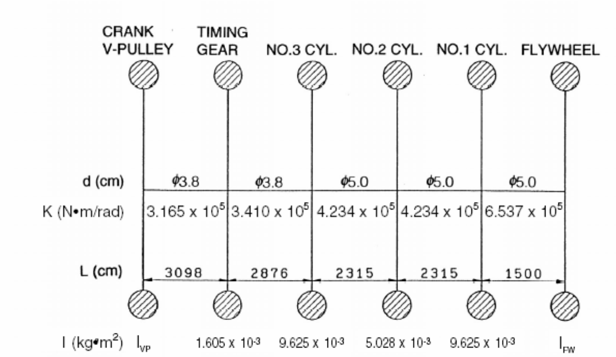


Figure 4.1: Equivalent torsional system of an engine

Table 4.1: Main engine excitation

Harmonic order	Frequency [Hz]
2	73.33
4	146.67
6	293.33
8	366.67
12	440.00
14	513.33

Table 4.2: Harmonic components of compressor shaft torque

Harmonic order	Component frequency[Hz]	Torque harmonic		Harmonic order	Component frequency[Hz]	Torque harmonic	
		Amplitude[lb.ft]	Phase[deg]			Amplitude[lb.ft]	Phase[deg]
4	**146.67**	**18.18**	**-100.9**	**16**	**586.67**	**1.64**	**157.9**
8	**293.33**	**5.97**	**-168.6**	**20**	**733.33**	**0.74**	**123.5**
12	**440.00**	**2.72**	**-168.6**	**24**	**880.0**	**0.34**	**82.1**

Table 4.3: Natural frequencies of system

mode n.	1	2	3	4	5	6	7	8	9
f [Hz]	402.1	455.0	630	1172	1743.1	1759.0	2080.4	2343.3	4806.4

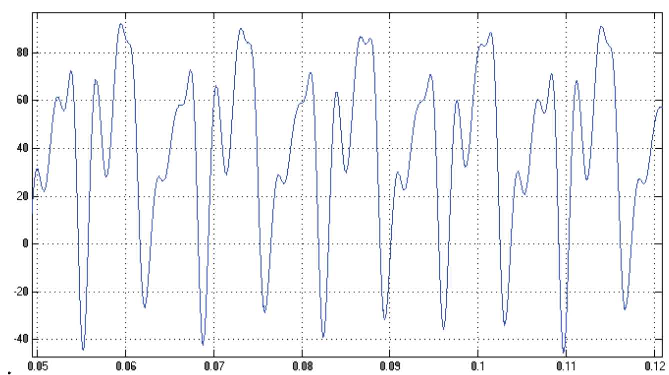


Figure 4.2: Compressor shaft tq. [lb.ft] - time [s]

5. STATIC AND CYCLIC STRESS ANALYSIS

5.1 Approach

There are many different approaches for applying the loads on the crankshaft to obtain the stress time history. One method is to run the FE model at selected times over 360° to define the stress-time history of the component. Finite element modeling of any solid component consists of geometry generation, applying material properties, meshing the component, defining the boundary constraints, and applying the proper load type. These steps will lead to the stresses and displacements in the component. In this study, similar analysis procedures were performed for different material and geometries of crankshafts.

5.2 Solid Modeling

Solid models were generated from the accurately measured dimensions of the crankshafts. Drilled holes on the counter weights and internal threads on oil holes were not included in the models since their presence makes the geometry complicated but they do not affect stresses at critical locations. Another important characteristic is the dynamic balance of the geometry. Examining the location of center of gravity of the digitized models showed a very close distance from the main bearings center line. The weight comparison between the actual crankshaft and digitized model and dynamic balance of geometry indicated the accuracy of the generated models.

5.3 Mesh Generation

Solid tetrahedral elements with free meshing were used to mesh the crankshaft finite element geometry. In this feature, the global mesh size could be defined, while for critical locations free local meshing could be used to increase the number of elements for accurate stresses at locations with high stress gradients. Convergence of stress at different locations was considered as the criterion for mesh size and number of elements selection. Satisfactory results were obtained using 380,000 elements. In order to, verify the accuracy of the FE model, analysis was performed on crankshaft for the simulated test setup conditions. In the simulated test setup crankshaft bearing ends are gripped in rigid fixture and unit load is applied at the journals and the resultant strains are measured at specified locations with strain gages. This measured deformation is compared with the FE simulated results for validation of the FE model.

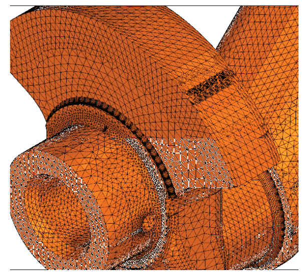


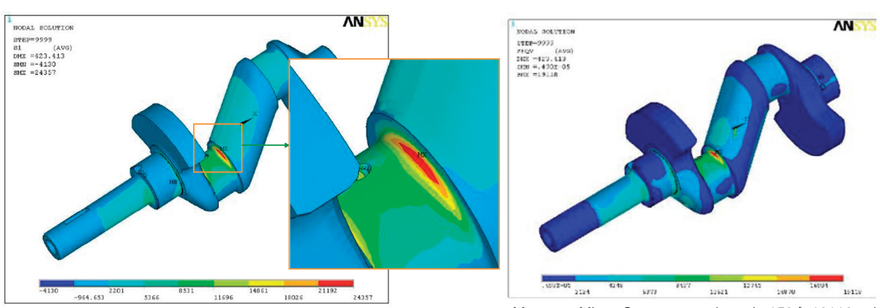
Figure 5.1: Meshing

5.4 Loading and Boundary Conditions

Using proper boundary conditions and loading strongly affect the results of the finite element analysis. The crankshaft is constraint with ball bearings from both the sides. The ball bearing is press fit to the crankshaft and does not allow the crankshaft to have any motion other than rotation about its main central axis; in addition each bearing has specific stiffness characteristics. For this purpose, the bearings are modeled as a set of linear springs and the node in the center is held against translation in all three directions and the outer nodes are attached to the solid model at the surface of the bearing step on the centerline plain of the bearing. The stiffness of the linear springs is set to give an equivalent bearing stiffness. In addition, on the drive side bearing, linear springs are added in the axial direction to stop axial motion of the crank shaft. The distribution of load over the connecting rod bearing is uniform pressure on 120° of contact area.

5.5 Finite Element Analysis

FE analysis was run at 4 points in the cycle (selected according the peaks and valley of the load cycle) along with 30 deg interval over the entire cycle. Stress results of these runs from FE analysis on the forged crankshaft are tabulated in Table 5.1. This information is used to evaluate various optimization options. Investigation of the FE models shows that the fillet areas experience the highest stresses during service life of the crankshaft. Since stress range and mean stress are the main controlling parameters for calculating fatigue life of the component, alternating stresses are also calculated as shown in Figure.5.3. One of the main objectives of performing the dynamic FE modeling was to determine the design loads for optimization of the forged steel crankshaft. The maximum load that is applied to the crankshaft during its service life is the load corresponding to the peak gas pressures, occurring at about 159° & 339° crank angles. Stress results also show that the maximum stress occurs at these crank angles. Therefore, these loading conditions are the most severe case of loading resulting in the maximum magnitude of von Mises stress.



Maximum Principal Stress for Crank angle 159 ° : 24357 psi

Max von- Mises Stress at crank angle 159 °: 19118 psi

Figure 5.2: Stress distribution at 159° & 339° crank angles.

Table 5.1: Stress distribution vs. crank angles.

Crank angle (Degree)	von-Mises stress (psi)	First principal stress (psi)	Factor of Safety
0	18547	21771	2.16
30	9134	11244	4.38
60	9892	11989	4.04
90	8477	9491	4.72
120	7631	8422	5.24
150	11322	12970	3.53
159	19118	24357	2.09
180	18093	21338	2.21
210	7541	8025	5.30
240	7782	7726	5.14
270	7958	9124	5.03
300	7761	8435	5.15
330	11700	12987	3.42
339	19525	24129	2.05

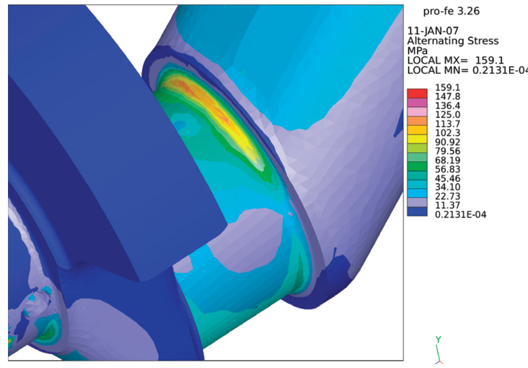


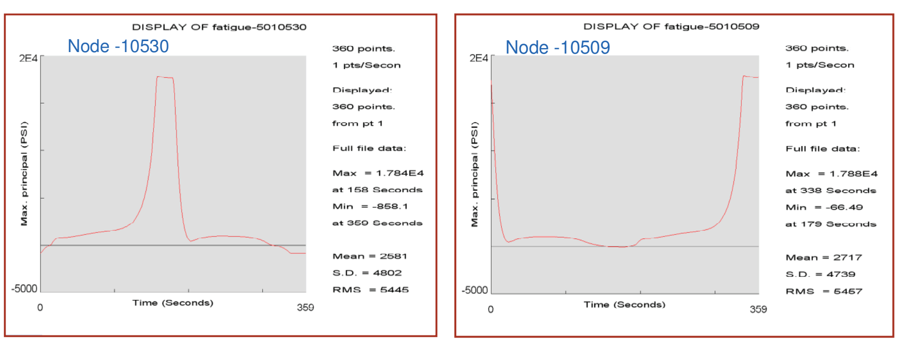
Figure 5.3: Alternating stresses

6. FATIGUE ANALYSIS AND VERIFICATION

Load variation over a cycle results in variation of stress. For calculations of fatigue damage in the component using the rainflow cycle counting method on the critical location, stress-history plot was generated. The stress-time history of a two critical locations at the speed of 2300 rpm is shown in Figure 6.1.

The gas pressure and inertia forces acting on crankshafts generate both bending moments and torsional moments. Because the magnitude and phase relationships between the maximum bending moment and maximum torsional moment vary with speed and torque, it is typical to analyze the stresses due to maximum bending and maximum torsion separately. Validation of separate bending and torsion analyses requires separate bending and torsion tests. Resonance techniques such as the "tuning fork method" are commonly employed to allow performance of high frequency fatigue tests at stress/strain levels that would require higher direct acting loads than can be generated with available electrodynamic shakers. The frequency range of electrodynamic shakers is much higher than the maximum operating speed of the crankshafts. Therefore tests may be run on at the maximum operating speed of the crankshaft or higher, thereby accumulating test cycles at a higher rate than on running compressors. Faster tests allow more samples or more design variations to be tested in the same period of time. In this study both types of fatigue tests are planned.

Bending fatigue is verified by direct load application using staircase method. Test set-up for bending test is shown in Figure 6.3. Crankshaft is supported on fixtures at main journal and pin journal and load applied at other pin journal using servo-hydraulic actuator to simulate the gas pressure and inertia load acting on the crankshafts through connecting rod. As the testing is done for the half throw by cantilever condition instead of completed throw, the test load will be half of that exerted by gas compression. In staircase method the first specimen is tested close to the expected median of the transition range. The next specimen is tested at lower load range if the previous specimen is failed and at a higher load range if the previous specimen is passed.



Alternating stress : 9349 psi
Mean stress : 2581 psi

Alternating stress : 8973 psi
Mean stress : 2717 psi

Figure 6.1: Critical nodes stress time

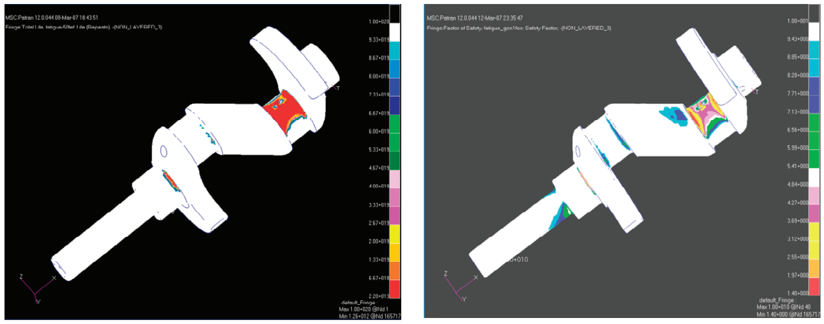


Figure 6.2: Fatigue life plots

In the tuning fork method, the crankshaft or a section of the crankshaft is assembled with two plates that become "tines" of the tuning fork as shown in the figure. Careful selection of the length and mass of the tines results in a first bending natural frequency of the crankshaft / tine assembly of 50 to 150 Hertz or higher. The assembly is easily excited at its natural frequency by an electrodynamic shaker. Finite element analysis is used prior to testing to determine both the optimum mass of the tines and the distribution of stress during testing to assure that the stress / strain distribution during testing is similar to the stress /strain distribution during actual use.. Stress distribution of the first bending mode of the crankshaft / tine assembly is usually very similar to the stress distribution in actual use. Because the crankshaft is tested at the resonant frequency of the assembly, the force applied by the shaker is not easily correlated to equivalent force at the crank-pin. Rather, one or more strain gages applied to the crankshaft are used to calibrate and control the test. The highest strain is usually in a fillet. A strain gage in a fillet with high strain may fail or the bond to the test sample may fail before the crankshaft fails, so additional strain gages placed in flat areas with lower strain than in the fillet are used for control. In addition, an accelerometer is placed on one of the tines and the acceleration level may be correlated to strain. The crank / tine assembly is hung from a supporting structure and shaker is suspended from the same structure. A "stinger" attached to both the shaker and one of the tines is set to excite the tuning fork at its resonant frequency and its amplitude is controlled to provide the required strain in the fillet. Depending on the controller used, a failure or crack will either be evident because the frequency can no longer be controlled, or the frequency necessary to maintain the assembly in resonance will drop.

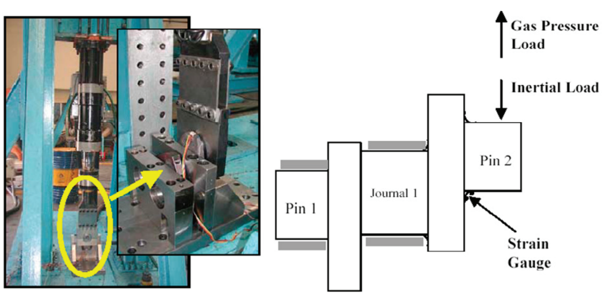


Figure 6.3: Bending Fatigue setup

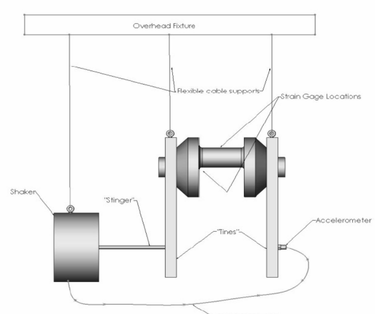


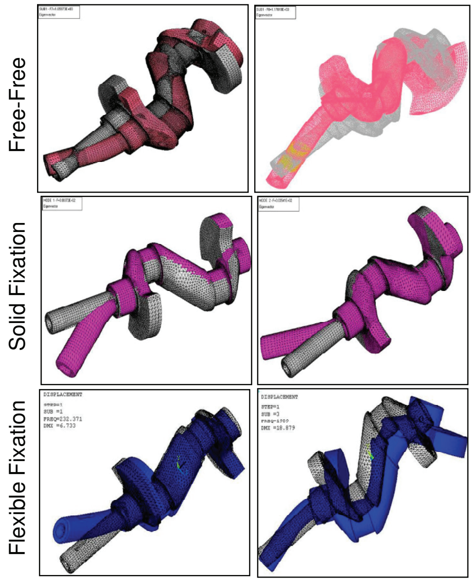
Figure 6.4: Vibration resonance Fatigue setup

7. MODAL ANALYSIS

This part presents the modal analysis of the crankshaft. The objectives are finding the normal modes of crankshaft and estimate the critical speed and the mode shapes. Three types of modal analyses are carried out,

a) Free-Free Analysis: No constraints used.
b) Solid Fixation: The bearing area is constrained in all DOF.
c) Flexible Fixation: The bearing area is connected with flexible springs.

From the table, frequency of first mode for Flexible fixation is 232 Hz and converting into RPM units critical speed 13920 RPM , Considering a +/- 25% tolerance critical speed ranges from 10400 to 17400 RPM . The maximum operating speed of crankshaft is well away from the critical speed.



Mode No.	Free- Free (Hz)	Solid Fixation (Hz)	Flexible Fixation (Hz)
1		300	232
2		304	244
3		2046	1909
4		4335	2246
5		4679	2280
6		5990	3252
7	1060	7560	4228
8	1178	7744	4797
9	2201	7923	5070
10	2558	8192	7027

Figure 7.1: Mode shapes and natural Frequencies

8. DYNAMIC BALANCING AND OPTIMZATION

This part discusses the dynamic balancing and optimization options. The main objective of this analysis was to optimize the weight of the crankshaft. Optimization carried out is not the typical mathematical sense of optimization, so each optimization step was judged through the sense of engineering knowledge. The two main factors which were considered during optimization are stress range under dynamic load and bending stiffness. These factors were verified to be in the permissible limits.

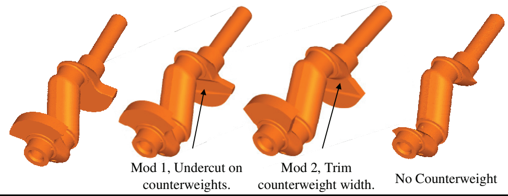
8.1 Optimization Approach

The objective function is minimizing the weight, maximum stress at critical locations, and cost. The bounded constraint is maximum allowable stress of the material. The equality constraint is geometry limitations. And the design variables are upper and lower limit for size and geometry, material alternatives, and manufacturing processes. As discussed earlier, the optimization stages were considered based on judgment using the results of the FEA and dynamic service load. The judgment was based on mass reduction, cost reduction, and improving fatigue performance and bending stiffness using alternative materials and considering manufacturing aspects. Manufacturing process and material alternatives were studied in a trial and error approach, after the geometrical optimization. Since the current crankshaft has proper fatigue performance, optimization was carried out in such a way that the equivalent local stress amplitude at any location of the optimized model did not exceed the equivalent stress amplitude at the critical location of the original model. Since the optimized crankshaft was expected to be interchangeable with the current one, the dimensions of crank radius, location and Geometry of main bearings and thickness and geometry of connecting rod bearing were not changed.

8.2 Dynamic analysis and Balance weight optimization

Investigating the stress contours of the crankshaft FEA model during the cycle showed that locations of the counter weights experience lower stresses. The crankshaft has to be dynamically balanced and counter weights have a key role in balancing it. Therefore, although stresses at these sections are low, they can not be removed, but can only be changed according to other modifications made on the component. The optimization criterion for balance weight is minimizing the mass without affecting the inertia characteristics of the part. For this purpose dynamic analysis is carried out using MSC-Adams to evaluate the unbalanced forces and moments, for the various configurations.
Initially effect of inertia characteristics on different material removal options was studied. From which, the required inertia characteristics were finalized. Then shape optimization was applied to minimize the balance weight mass for the same inertia characteristics. For mass optimization, Rectangular slot dimensions, Outer radius, corner radius and web angle were considered with in the limits. Web width reduction increase was not considered as any increase will

cause the connecting rod interference with the web. FE models of combined features were created and analyzed for strength and dynamic characteristics. It was ensured that the stress ranges at critical location does not change and the stiffness of the optimized crankshaft is within the acceptable limit. The modified crankshaft with the final optimized geometry is shown in Figure 8.3.



	Original	Mod 1	Mod 2	No Counterweight
Volume	316600 mm^3	291500 mm^3	285500 mm^3	241200 mm^3
Weight Reduction %	0.0%	7.9%	9.8%	23.4%
Increase in Free Moments	0.0 N-m	18.7 N-m	22.2 N-m	78.1 N-m

Figure 8.1: Inertia Characteristics of material option

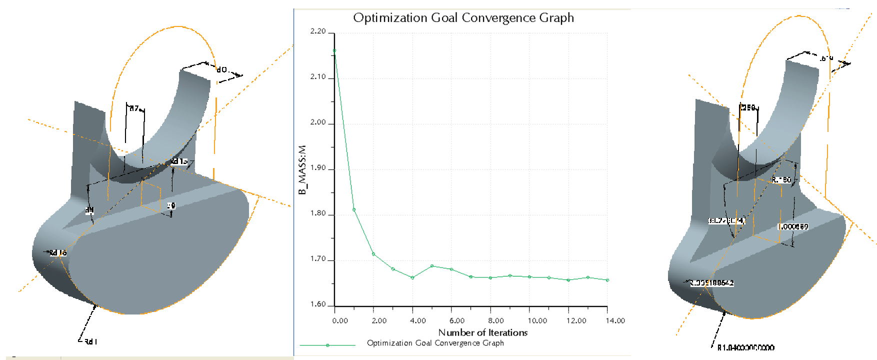


Figure 8.2: Balance weight shape optimization using BMX.

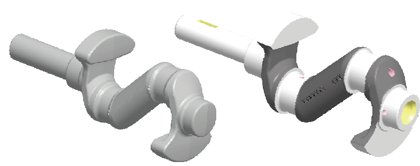


Figure 8.3: Final modified crankshaft.

9. CONCLUSIONS

Overall Weight reduction of 12 % and Cost reduction of 23% was achieved from the optimization process from original crankshaft with out affecting the life or vibration characteristics.

REFERENCES

Abdulla S. Rangwala, Reciprocating Machinery Dynamics, Design and Analysis book, Marcel Dekker, Inc; New York 10016, 2001.

Benson H. Tonque, Principles of Vibration, New Yourk, Oxford University Press, 1996.

Vladimir Chlumsky, Pistove kompresory (Pistons compressors) SNTL, Prague 1958.

Petr Hošek, Torsional analysis of engine - coupling - compressor and engine - clutch – compressor & el. Motor systems in TK refrigeration units Research Institute Of Refrigeration Engineering Prague , Czech Republic.