Optimization of Scroll Compressor Counterweight Design Through FEA And Experimental Analysis

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Abstract- *This research paper addresses fracture of counterweight ears during staking operation & fatigue fracture of counterweights in operating conditions. This research paper presents on design of a scroll compressor counterweight that balances all the eccentric loads in a high frequency scroll compressor system, FEA simulation of counterweight staking operation by providing the optimum staking force required and further FEA analysis for fatigue of newly proposed counterweight design followed by field testing also ensures that counterweight ears and rotor post don't show fatigue fracture in the operating conditions. As the rotating components of the compressor subjected to changes have different masses with different C.G. location, rotating in different planes in the compressor system and hence dynamic unbalancing principles were applied here. Dynamic unbalance calculations were performed and it is verified that the mass of counterweight is optimum for the given position. New counterweight was designed by analytical and graphical method. This newly designed counterweight balances all the eccentric rotating loads in the scroll compressor system and maintains stability of the system. Vibration testing confirms that the amplitude of the vibrations of scroll compressor obtained with newly designed counterweight are well with allowable limits. Finite element analysis was done in order to perform simulation for staking operation of counterweight and the results were verified with actual staking loads. Along with centrifugal force of rotor rotation, repeated start-stop conditions of rotor apply cyclic loads on counterweight ears and which causes fatigue fracture. Hence focus was maintained on finite element analysis for fatigue conditions and made efforts to estimate stresses on counterweight-rotor post assembly under compressor start-stop conditions. Design changes were made for safe counterweight design and prototype counterweight-rotor post assembly was fatigue tested to verify the fatigue strength of newly designed counterweight-rotor post assembly.*

Keywords- Counterweight, scroll compressor, imbalance, vibration, staking, FEA, fatigue

I. INTRODUCTION

If a new compressor platform with new efficiency improving scroll design and new refrigerant is expected to launch then this new compressor is attempted to incorporate new scroll technology with minimum changes in the existing scroll compressor models. However in order to comply with the new scroll design, it would cause considerable changes in the shape and position of various rotating component in the compressor. Because of these changes, this new refrigeration scroll compressor would demand imbalance study and verification whether there is need of new counterweight for the system or not. The need of new counterweights for a refrigeration scroll compressor was identified. A new counterweight is designed as per the counterweight imbalance study and sent for pilot testing. However, during pilot run, counterweight showed non-conformance. Probable Causes at counterweight staking stage would be sole or accumulated effect of one of the following reasons – manufacturing stack-up issues and staking overload.

II. PROBLEM DEFINITION

It has been observed that reed fractures at the tip location of the reed and it is on the edges locations, these fractures show characteristics of impact fatigue, seen in Figure 2[12]. Fractographic examinations detect primary crack region & besides the performance, a valve should also last till designed life which is indicated by root stress, fatigue strengths and impact velocity. Typically, valve materials have high tensile strengths and endurance limits. Also, in a study conducted by Sandvik (Svenzon, 1976)[10] it was shown that over the period of time, valve surface near the ports shows pitting. This type of damage compromises the structural integrity of the reed valve and leads to severe leakages around the valve. The same study correlates the damage on the valve to its impact acceleration and suggests a limiting value for this acceleration [3].

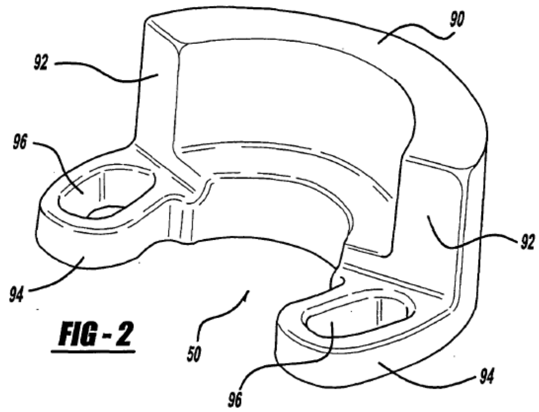


Figure 1. Typical Shape of Counterweight Used In Scroll Compressor [18]

III. OBJECTIVE

The main objectives are highlighted according to the sequential approach to solve the problem statement: calculate the imbalance in the current compressor system of rotating components; Design a counterweight that balances the imbalance in this system; perform FEA simulation to determine the optimum counterweight staking force which conforms to the internal design standards; define a FEA methodology to perform the FEA analysis of this counterweight assembled to the rotor for fatigue with an existing baseline counterweight; validate the proposed counterweight designs through set FEA methodology for best suitable solution among them and understand the fatigue strength of the newly designed counterweight-rotor post assembly.

IV. LITERATURE REVIEW

Jim Lyons in the paper, “Dynamic Balancing-Causes, Corrections and Consequences” explains what the dynamic balancing is, its causes, how to correct it and its consequences in the paper he discusses a level of unbalance that is acceptable at a low speed is completely unacceptable at a higher speed. This is because the unbalance condition produces centrifugal force, which increases as the speed increases. In fact, the force caused by unbalance increases by the square of the speed. It is the force that causes vibration of the bearings and surrounding structure. Prolonged exposure to the vibration results in damage and increased downtime of the machine. [4]

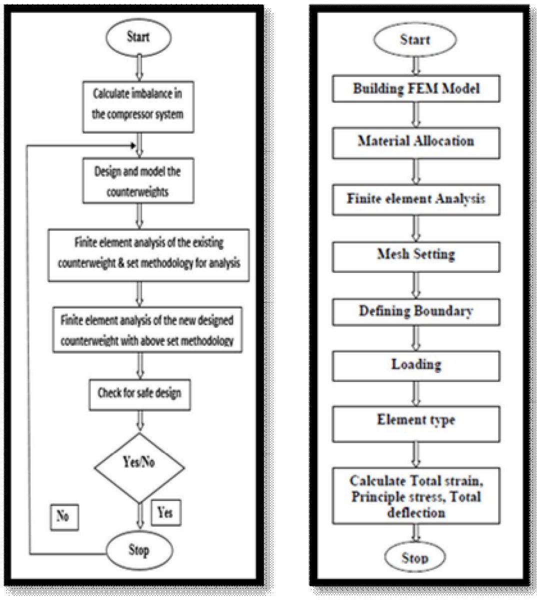
Masato SoWa, Yoshiyuki Nakane, Toshiro Fujii, Tatsuyuki Hoshino, Ryuta KaWaguchi, Takahiro Moroi, Tsutomu Nasud: In the patent “Method of correcting imbalance of scroll compressor”, authors address originated from tolerance of machining and assembling by redressing the imbalance generated in the compressor by means of balancers. The present invention also provides a method of correcting imbalance on a drive shaft of a scroll type compressor. The compressor includes a motor housing, the drive shaft, a stator and a rotor. The method includes assembling the compressor Without the motor housing and the stator and placing the compressor to a dummy housing having another stator, driving the compressor in the dummy housing, measuring the amplitude and the phase of the vibration caused by imbalance in the compressor, and correcting the imbalance on the drive shaft based on the measured amplitude and the measured phase While the compressor is in the dummy housing.

Troy D. Feese, Phillip E. Grazier focus on six case histories, which illustrates the actual problem that were encountered while trying to field balance different types of rotating machinery. Coupling lockup, thermal bows, eccentricity, looseness and structural resonances can prevent successful balancing if not properly identified and corrected. This paper shows how these real world pitfalls can complicate balance procedures and provides some ways of dealing with these issues such as indexing coupling parts, using inference fits, adding structural bracing and acquiring vibration measurements at additional locations. [3]

Earl M. Halfen in his paper “Shop Balancing Tolerances a Practical Guide” he conveys that even when the proper residual unbalance level has been achieved, mechanical changes in the rotor after balancing, such as improper clearances, rotor sag, eccentric bearings, etc. can cast false doubt about the balancing operation. In order to establish and maintain credibility in the balancing operation a proving test should be applied, using a known mass applied to the rotor in a logical sequence. [1]

V. METHODOLOGY

This project focuses on the study test results through the FEA and gets optimum value of staking force based on the FEA result field testing and lab testing has been performed and then compare and validate the results of both.



Analytical Calculations FEM Model

Figure 2. Methodology

Below is the cross section of typical scroll compressor with the counterweight and rotor assembly. Below image shows the different rotating components associated with dynamic balancing of the scroll compressor system.

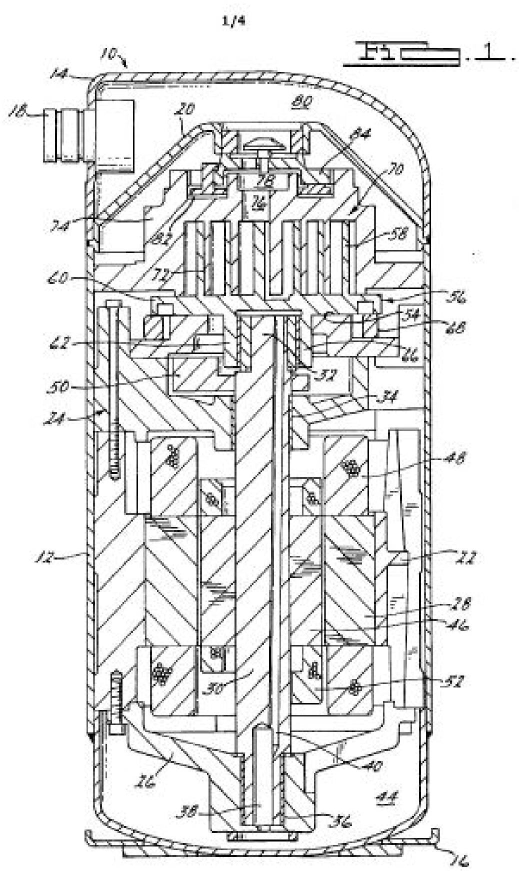


Fig. 3. Scroll Compressor Vertical cross section [12]

Table 1. Bill OF Material (BOM) For Fig 1 [12]

Balloon No.	Part Name	Qty
10	Scroll compressor	1
12	Cylindrical hermetic shell	1
14	Welded cap	1
16	Welded base	1
18	Refrigerant discharge fitting	1
20	Transversely extended partition	1
22	Compressor mounting frame	1
24	Main bearing housing	1
26	Lower bearing housing	1
28	Electrical motor stator	1
30	Crankshaft	1
32	Eccentric crank pin	1
34	Bearing in (24)	1
36	Bearing in (26)	1
38	Central bore in crankshaft	1
40	Eccentric bore in crankshaft	1
44	Oil sump	1
46	Rotor	1
48	Winding	1
50	Upper counterweight	1
52	Lower counterweight	1
54	Flat thrust bearing surface	1
56	Orbiting scroll member	1
58	Spiral vane of 56	1
60	End plate of 56	1
62	Cylindrical hub	1
66	Drive bushing	1
68	Oldham coupling	1
70	Non orbiting scroll member	1
72	Spiral vane of 70	1
74	End plate of 70	1
76	Central discharge passage in 70	1
78	Open recess	1
80	Discharge muffler chamber	1
82	Annular recess	1
84	Floating seal assembly	1

VI. CALCULATION FOR ROTATING ECCENTRIC IMBALANCE PRESENT IN THE SCROLL COMPRESSOR SYSTEM

Balancing by Graphical Method [6]

The procedure followed in balancing is as follows:

i. Take one of the plane say ‘L’ as reference plane (R.P.). The distance of other planes to the right of R.P. are taken as positive and to the left are taken as negative.

ii. Tabulate the centrifugal forces and couples due to centrifugal forces. The planes are tabulated in the same order as they are located from left to right.

iii. Draw the couple polygon, taking some suitable scale. Since couple $m_1r_1l_1$ is negative, draw $– m_1r_1l_1$ radially inward in the reverse direction of Om1. The couple $m_2r_2l_2$ is positive so draw couple $m_2r_2l_2$ radially outward i.e. in the direction of Om_2. Similarly draw couples $m_3r_3l_3$ and $m_4r_4l_4$. Closing vector i.e. d’o represents the couple $m_Mr_Ml_M$ which is called as balancing couple.

iv. Knowing the radius ‘r_M’ the magnitude of balancing mass ‘m_M’ can be obtained as follows:

d’o’ x scale of couple polygon = $m_Mr_Ml_M$
m_M = (d’o’/r_Ml_M) x scale of couple polygon

v. From couple polygon, the angular position of balancing masses ‘m_M’ is obtained.

vi. Another balancing mass ‘m_L’ can be found by drawing the force polygon:

Draw ‘oa’, to represent m_2r_2 parallel to ‘O m_2’.
From a draw ‘ab’, to represent m_3r_3, parallel to ‘Om_3’.
From b draw ‘cd’, to rpresent m_4r_4, parallel to ‘Om_4’.
From d draw ‘de’, to represent m_Mr_M parallel to ‘Om_M’.
Join ‘eo’ to represent the balancing centrifugal force m_Lr_L.

vii. Knowing the radius ‘r_L’ the magnitude of balancing mass ‘mL’ can be obtained as follows:

eo x scale of force polygon = m_Lr_L
m_L = (‘eo’/ r_L) * scale of force polygon

viii. From force polygon, the angular position of balancing masses ‘m_L’ is obtained.

ix. Draw the angular position of balancing masses ‘m_M’ and ‘m_L’ as shown in figure.

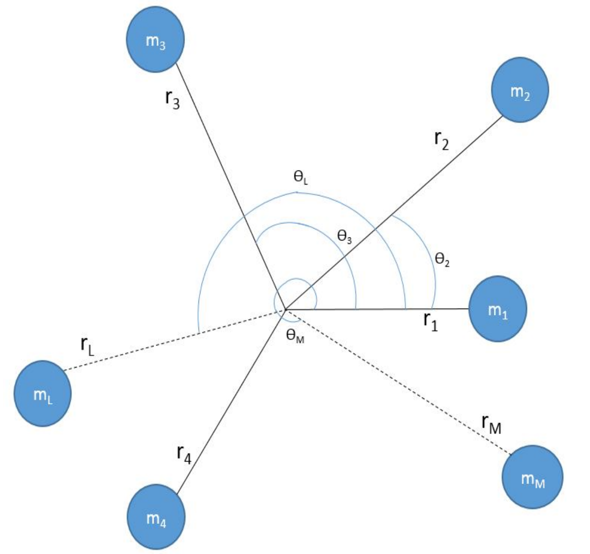


Figure 4. Angular Positions of Masses [6]

Table 2. Centrifugal Force and Couples [6]

Plane	Mass (m) Kg	Radius (r) m	Centrifugal Force / w2 (mr) Kg-m	Distance from R.P. (l) m	Couple/ w2 (mrl), Kg- m2
A	m_1	r_1	m_1 r_1	$-l_1$	$-m_1$ r1 l_1
L (R.P.)	m_L	r_L	m_L r_L	0	0
B	m_2	r_2	m_2 r_2	l_2	m_2 r_2 l_2
C	m_3	r_3	m_3 r_3	l_3	m_3 r_3 l_3
M	m_M	r_M	m_Mr_M	l_M	m_M r_M l_M
D	m_4	r_4	m_4 r_4	l_4	m_4 r_4 l_4

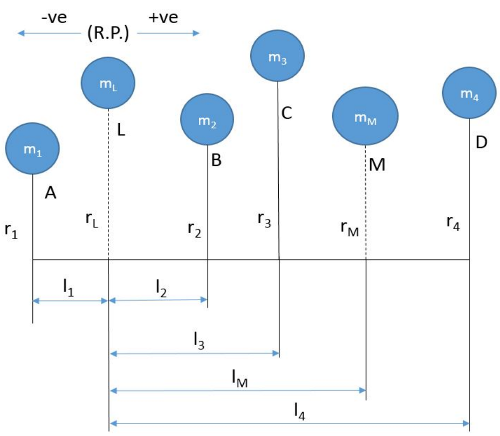


Figure 5. Positions of Planes of Masses [6]

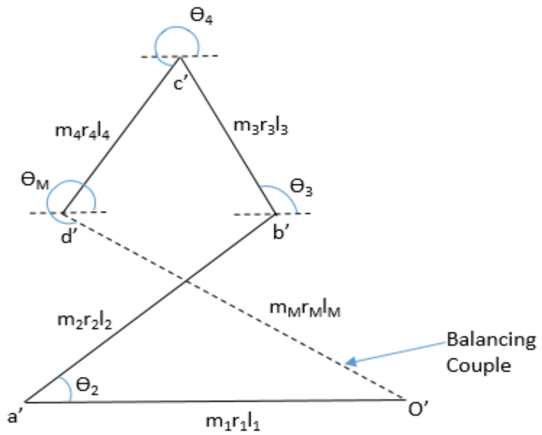


Figure 6. Couple Polygon [6]

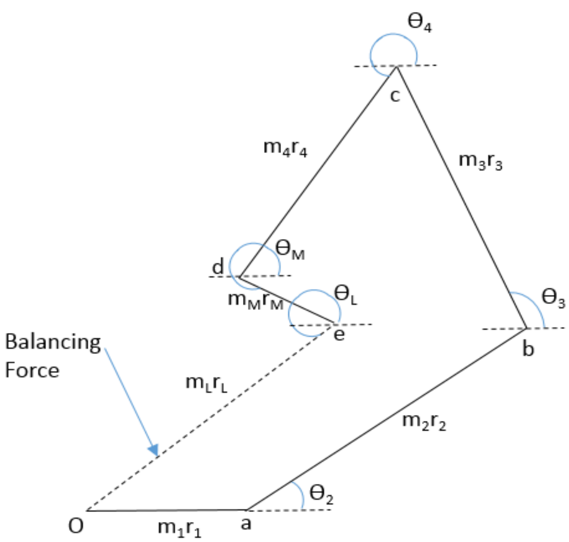


Figure 7. Force Polygon [6]

Note:

Angular directions are measure in anticlockwise direction here. Since w2 for all masses here, the centrifugal forces and centrifugal couples are taken in terms of (mr) and (mrl) respectively.

The two unknown balancing masses can be found by drawing force polygon and couple polygon. Now the question arises which polygon is to be drawn first. That can be decided from table 3.1. It is seen that, in centrifugal force column both balancing masses are unknown while in couple column only single balancing mass is unknown. Hence first draw the couple polygon so that the closing vector of couple polygon will give the magnitude and direction of that single unknown balancing mass. And then draw the force polygon so that the closing vector of force polygon will give the magnitude and direction of that remaining unknown balancing mass.

Balancing By Analytical Method [6]

Table 3. Analytical Calculations [6]

Conditions for complete balancing of system
$\sum$(couples due to centrifugal forces) = 0 or $\sum$(mrl] $\sum$(centrifugal forces) = 0 or $\sum$(mr) = 0
Condition I
$\sum$(couples due to centrifugal forces) = 0 or $\sum$(mrl]
(i) Resolve the couples horizontally and vertically and find their summation:
Horizontal: $\sum(\mathbf{mrl}) = \mathbf{0}$
- m_1 r1 l_1 cos 0 + m_2 r_2 l_2 cos Θ_2 + m_3 r_3 l_3 cos Θ_3 + m_M r_M l_M cosΘ_M + m_4 r_4 $l_4$cos Θ_4 = 0 m_M r_M l_M cosΘ_M = m_1 r1 $l_1$cos 0 - m_2 r_2 l_2 cos Θ_2 - m_3 r_3 $l_3$cos Θ_3 - m_4 r_4 l_4 cos Θ_4 Put, CH = m_1 r1 $l_1$cos 0 - m_2 r_2 l_2 cos Θ_2 - m_3 r_3 l_3 cos Θ_3 - m_4 r_4 $l_4$cos Θ_4 Hence, m_M r_M l_M cosΘ_M = C_H …..(1)
Vertical: $\sum(\mathbf{mrl}) = \mathbf{0}$
- m_1 r1 l_1 sin0 + m_2 r_2 l_2 sin Θ_2 + m_3 r_3 l_3 sin Θ_3 + m_M r_M l_M sin Θ_M + m_4 r_4 $l_4$sin Θ_4 = 0 m_M r_M l_M sin Θ_M = m_1 r1 l_1 sin0 - m_2 r_2 l_2 sin Θ_2 - m_3 r_3 $l_3$sin Θ_3 - m_4 r_4 l_4 sin Θ_4 put, Cv = m_1 r1 l_1 sin0 - m_2 r_2 l_2 sin Θ_2- m_3 r_3 l_3 sin Θ_3 - m_4 r_4 $l_4$sin Θ_4 Hence, m_M r_M l_M sin Θ_M = Cv…..(2)
(ii) Calculate the magnitude of balancing
mass mM: From above equations (1) & (2), (m_M r_M l_M cos Θ_M)2 + (m_M r_M l_M sin Θ_M)2 = (CH)2 + (Cv)2 m_M r_M l_M = $\sqrt{C_H{}^2 + C_V{}^2}$ m_M r_M = $\frac{\sqrt{C_H{}^2+C_V{}^2}}{lM}$ …..(3)
(iii) Calculate angle made by balancing mass m_M with the horizontal line OX:
From above equations (1) & (2), $\frac{mMrMlM \sin \Theta M}{mMrMlM \sin \Theta M} = \frac{C_V}{C_H}$ $\tan \Theta M = \frac{C_V}{C_H}$ $\Theta M = \tan^{-1}(\frac{C_V}{C_H})$ …..(4)
Condition II

$$\sum(\text{centrifugal forces}) = 0 \text{ or } \sum(mr) = 0$$

(i) Resolve the forces horizontally and vertically and find their summation:

Horizontal: $\sum(\mathbf{mr}) = \mathbf{0}$

$m_1 r_1 \cos 0 + m_L r_L \cos\Theta_L + m_2 r_2 \cos \Theta_2 + m_3 r_3 \cos \Theta_3 + m_M r_M \cos\Theta_M + m_4 r_4 \cos \Theta_4 = 0$

$m_L r_L \cos\Theta_L = - [m_1 r_1 \cos 0 + m_2 r_2 \cos \Theta_2 + m_3 r_3 \cos \Theta_3 + m_M r_M \cos\Theta_M + m_4 r_4 \cos \Theta_4]$

Put, $F_H = - [m_1 r_1 \cos 0 + m_2 r_2 \cos \Theta_2 + m_3 r_3 \cos \Theta_3 + m_M r_M \cos\Theta_M + m_4 r_4 \cos \Theta_4]$

Hence,

$m_L r_L \cos\Theta_L = F_H$…..(5)

Vertical: $\sum(\mathbf{mr}) = \mathbf{0}$

$m_1 r_1 \sin 0 + m_L r_L \sin \Theta_L + m_2 r_2 \sin \Theta_2 + m_3 r_3 \sin \Theta_3 + m_M r_M \sin \Theta_M - m_4 r_4 \sin \Theta_4 = 0$

$m_L r_L \sin \Theta_L = - [m_1 r_1 \sin 0 + m_2 r_2 \sin \Theta_2 + m_3 r_3 \sin \Theta_3 + m_M r_M \sin \Theta_M - m_4 r_4 \sin \Theta_4]$

Put, $F_v = - [m_1 r_1 \sin 0 + m_2 r_2 \sin \Theta_2 + m_3 r_3 \sin \Theta_3 + m_M r_M \sin\Theta_M - m_4 r_4 \sin \Theta_4]$

Hence,

$m_L r_L \sin \Theta_L = F_v$ …..(6)

(ii) Calculate the magnitude of balancing mass mL:

From above equations (5) & (6),

$(m_L r_L \cos \Theta_L)2 + (m_L r_L \sin \Theta_L)2 = (F_H)2 + (Fv)2$

$m_L r_L = \sqrt{F_H^2 + F_V^2}$ …..(7)

(iii) Calculate angle made by balancing mass mL with the horizontal line OX:

From above equations (5) & (6),

$\frac{mLrL \sin \theta L}{mLrL \cos \theta L} = \frac{F_V}{F_H}$

$\tan \theta L = \frac{F_V}{F_H}$

$\theta L = \tan^{-1}(\frac{F_V}{F_H})$ …..(8)

Here we get magnitude and direction of remaining unknown balancing mass of the rotating imbalance system.

By using above calculation method we determine the location & mass of two counterweights that are used in the scroll compressor system. The mass of the counterweights is maintained equal to the values determined by expressions (3) & (7) and the C.G location is maintained at the reference planes with angle opposite to that determine by expressions (4) & (8).

VII. RESULTS AND DISCUSSION

3D CAD model is created to define the Functional dimension analysis and Fit function analysis. FEA is performed with the help of different analysis software tools for fatigue and staking simulation.

A. Staking Operation

In rotor-counterweight staking operation, a tool with specific geometry at its bottom is rammed with a force on the rotor post after putting counterweight on rotor and the resultant deformed rotor post fills into as well as on the counterweight ear and locks the assembly.

FEA simulation was done in order to predict the optimum force required for this operation to obtain a unit displacement into the rotor post during staking operation. Force Vs deflection plot was plotted. It should be noted that, in figure 11, beyond 0.25-unit graph is extrapolated as finite elements starts undergoing very high distortions. Adaptive re-meshing techniques were explored but they are not mature as of today, and failed to result in any better answer.

FEA Geometry:

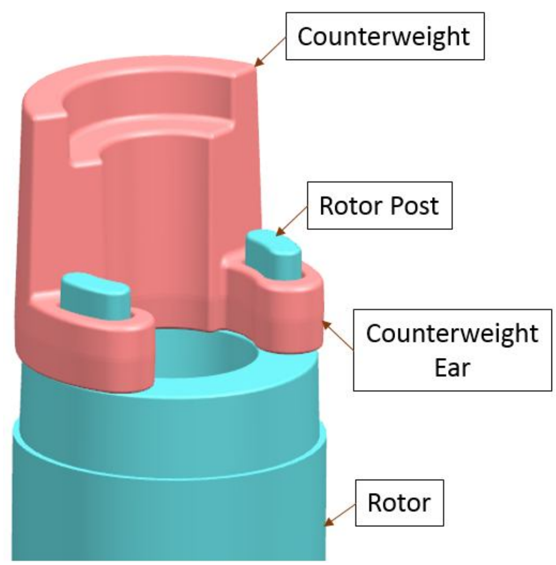


Figure 8. Assembly Geometry

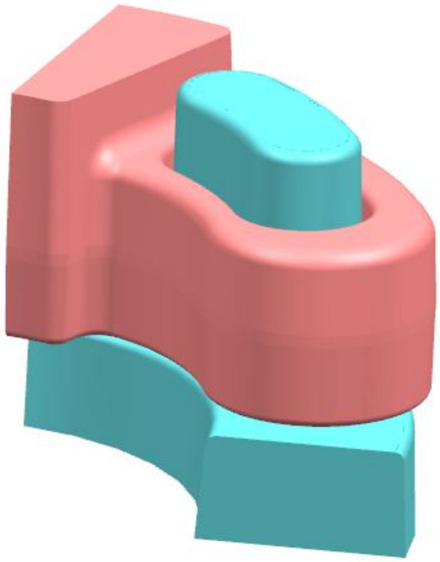


Figure 9. Assembly Geometry-Simplified

Simulation Images:

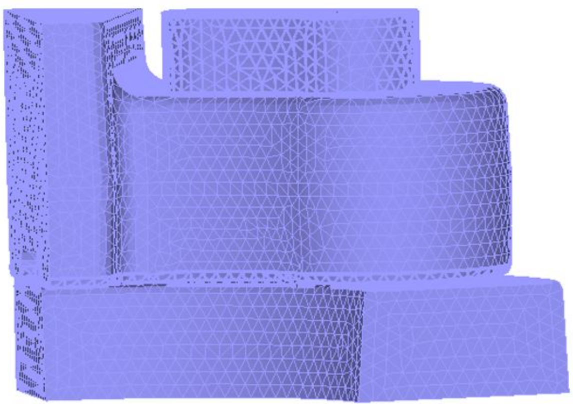


Figure 10. Meshed Geometry

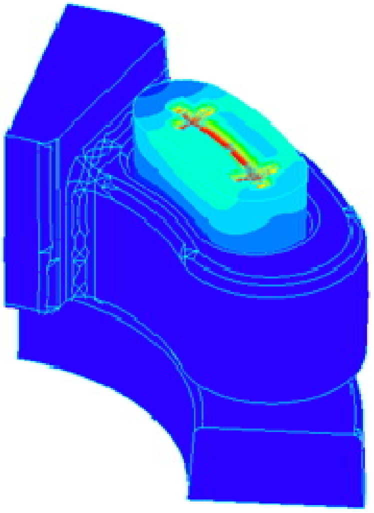


Figure 11. Deformation after the Staking Simulation

Figure 12. Normalized Punch Force Vs Displacement Plot

Comments on Test Results:

During testing, a batch of counterweight-rotor assembly was tested against the different loads and checked if they are filled sufficiently by cutting the cross section of the rotor-counterweight assembly.

Pilot test of staking operation performed on a batch of counterweight-rotor showed that test and FEA results deviation is within 5 %.

B. Counterweight Fatigue FEA results & Testing

Along with centrifugal force of rotation in scroll compressor, repeated start-stop conditions of rotor apply cyclic loads on counterweight ears and which causes fatigue fracture. Hence focus was maintained on finite element analysis for fatigue conditions and made efforts to estimate stresses on counterweight-rotor post assembly under compressor start-stop conditions. Design changes were made for safe counterweight design and prototype counterweight-rotor post assembly was fatigue tested to verify the fatigue strength of newly designed counterweight-rotor post assembly.

Stress Distribution:

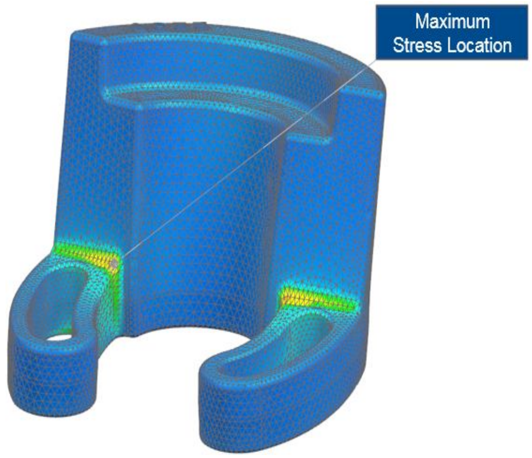


Figure13. Critical Stress Location on Counterweight Ear

Bench Test Set-up:

Laboratory bench test set-up was prepared to simulate the fatigue conditions on the counterweight-rotor assembly. % of the counterweight-rotor assembly passing the test was well within internal standard specifications.
For the ones that were showing non-conformance, it was observed that FEA and test predicted the non-conformance at the same location.

Table 4. FATIGUE BENCH TEST AT GLANCE

Testing Load Type	Application Point	Post-Result Comment
Centrifugal Force	Centre Of Gravity	Design is Safe For fatigue

C. Scroll Compressor Vibration Testing

This test is conducted to make sure if designed counterweights actually balance all the eccentric loads in the scroll compressor during operating conditions as well as to ensure the counterweights are installed correctly. Accelerometers are placed at specific locations on the compressor to measure the peak to peak vibrations and the maximum value of the vibration is taken into consideration for the conclusion comments.

Test Results:

This test is conducted to make sure if designed counterweights balance all the eccentric loads in the scroll compressor during operating conditions as well as to ensure the counterweights are installed correctly

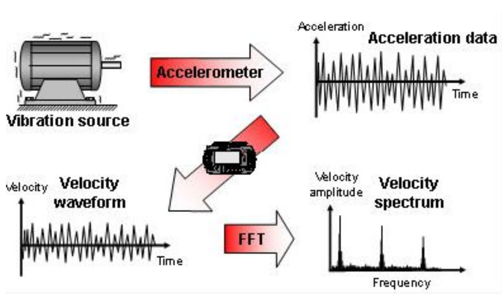


Figure 14. Typical workflow of vibration measurement [15]

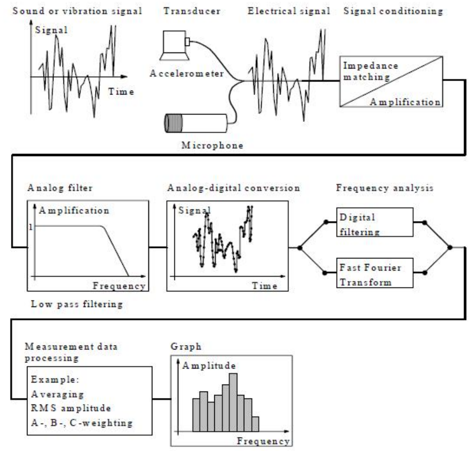


Figure 15. Flow chart of a digital accelerometer measurement system [15]

There are critical target vibration values that are targeted which should not be crossed during testing. Vibration are measured with help of accelerometers as peak to peak amplitude. An accelerometer is a sensor that measures the dynamic acceleration of a physical device as a voltage. Test was conducted at different speeds on the compressor and it was observed that the at all speeds vibrations in the compressor are maintained well below the targeted values. It should be noted that test result of single counterweight-rotor assembly for different speeds has been presented below as an example.

Table 5. Normalized Bench Test Results

Result Number	Vibration Test Results (%)	Target Vibration Value (%)
1	91	100
2	92	100
3	88	100
4	93	100
6	96	100

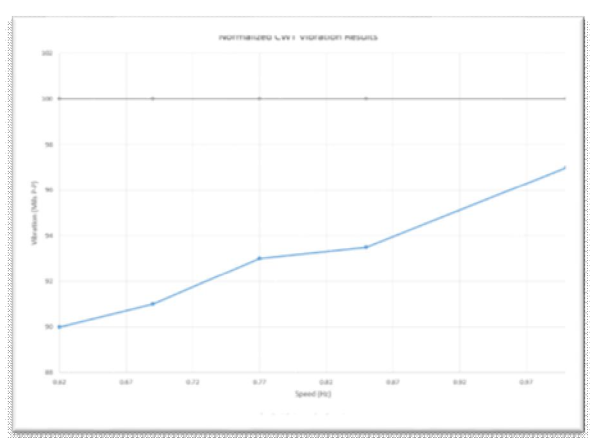


Figure 16. Normalized Vibration Amplitude vs. Speed Graph

VIII. RESULTS AND DISCUSSION

1. Objective of the study to be a good learning experience was achieved as during the study focus was well distributed over balancing calculations of the counterweight, FEA simulation for staking operation & fatigue as well as testing result validation.
2. Good correlation in the FEA simulation results & testing values is achieved that efficiently helped in determining optimum staking force for rotor-counterweight assembly process.
3. New design has also proved its good fatigue strength as it has been analysed for fatigue using FEA tool and verified the same through experimental tests.
4. From the vibration testing of the scroll compressor after installation of new design counterweight, theoretical imbalance calculations are validated. Targeted specific limit of vibrations is achieved. It also ensured safe installation of counterweights in the scroll compressor.

IX. FUTURE SCOPE

There is scope for more refined FEA results for simulation process because beyond 0.25-unit deformation is extrapolated as FEA failed to converge. For the future scope, adaptive meshing techniques can be improved to solve such critical deformation problems.

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