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DESIGN AND ANALYSIS OF RECIPROCATING DOUBLE ACTING TWO STAGE AIR COMPRESSOR CRANKSHAFT

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ABSTRACT:

Air compressor has earned a fair amount of popularity amongst various industries due to diverse uses of compressor air in applications such as driving of air engines(air motors), operation of blast furnace, Bessemer conveyors, supercharging of I.C. engines to name a few. The air compressors are available in various capacities and types. Mainly there are two types of compressor is driven by prime mover such as diesel engines or electric motor or sometimes turbine through crankshaft. In this paper crankshaft was designed by considering torsional and bending moments. After designed model was developed in Creo. Static structural and transient analysis was carried out in ANSYS.

KEYWORDS: Compressor, crankshaft, analysis.

1. INTRODUCTION:

In order to successfully control, the noise and vibration, the vibration of reciprocating compressor crankshaft, which can cause vibration and noise of the compressor, and some even can destroy crankshaft bearing and crankshaft itself must be estimated and analysed. So early in design stage, computations of natural frequencies, mode shapes, and critical speeds of crankshaft system are indispensable. Thus, an accurate model for prediction of the vibration of a crankshaft system is essential for reciprocating compressor. Vibration of the crankshaft system is a complex three-dimensional coupled vibration under running conditions, including the torsional, longitudinal and lateral vibrations.

2. DESIGN:

2.1 CRANKSHAFT DESIGN:

S_e = Allowable fatigue strength

P_r =Tangential force

Table I. Crankshaft Material Properties

Material	SG Iron 600/3
Factor of safety	1.5-2
Ultimate Tensile Strength	600Mpa.
Yield strength	370MPa

Case-I: Position of crankshaft at TDC and it is subjected to maximum bending moment and zero torsional moment.

Table II. Reaction Forces At Different Locations Of Crankshaft

Point	Description	Reaction(kg)		Moment(kg-mm)	
		Horiz	Vert	Horiz	Vert
A	Bearing1	-409.404	-306.54	0	0
B	HP Crank	0	-959.158	-45853.3	34333.08
C	LP Crank	0	2684.885	-98666.5	197608.9
D	HP Crank	0	-959.158	-151479.8	14534.5
E	Bearing2	1371.09	714.02	2154937.5	56769
F	Flywheel	961.6894	254	0	0

Bending moment is highest at point C. So resultant bending moment,

$$M_c = \sqrt{(M_{CH})^2 + (M_{cv})^2} = 220871.855 Kg-mm$$

Torsional moment at c is given by,

$$M_t = F_t . r = 0$$

$$(M_{te}) = \sqrt{(M_c * K_b)^2 + (M_t * K_t)^2} = 3301307.782 Kg-mm$$

Diameter of shaft under shear strength,

$$\tau = \frac{\Pi}{16 * d_c^{\ 3}} M_{te}$$

$$d_c = 42.071 mm$$

Diameter of shaft under bending strength,

$$M_{be} = \frac{1}{2}\left[M_b * K_b + M_{te}\right] = 331307.782 Kg-mm$$

$$d_c = \sqrt[3]{\frac{32 * M_{be}}{\pi * \sigma_b}} = 44.71 mm$$

Diameter of shaft under fatigue strength,

$$d_c = \sqrt[3]{\frac{32 * M_{be}}{\pi * S_e}} = 98.187 mm$$

Design of crankpin,

$$(M_{te}) = \sqrt{(M_b * K_b)^2 + (M_t * K_t)^2} = 79097.653 Kg-mm$$

Diameter on the basis of shear strength,

$$d_c = \frac{16}{\pi * \tau} M_{te} = 26.10 mm$$

Diameter on the bending strength,

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$$M_{be} = \frac{1}{2}\left[M_b * K_b + M_{te}\right] = 75689.9275 Kg - mm$$

$$d_c = \sqrt[3]{\frac{32 * M_{be}}{\pi * \sigma_b}} = 27.33 mm$$

Design of web:

Thickness of web,

$$t = 0.7d = 70mm$$

Width of web

$$w = 1.14 d_c = 114mm$$

Compressive stress due to $(R_1)_v$

$$\sigma_c = \frac{R_1}{w.*t} = 0.1347 Kg / mm^2$$

Bending stress due to $(R_1)_v$

$$\sigma_b = \frac{(b_1 - 0.5l_c - 0.5t)*(0.5t*(R_1)_v)}{\left(\frac{w*t^3}{12}\right)} = 1.5056 Kg / mm^2$$

Total stress,

$$\sigma = \sigma_b + \sigma_c = 1.645 Kg / mm^2$$

Allowable stress,

$$\sigma_c = \frac{38.73}{2} = 19.36 Kg / mm^2$$

Diameter of shaft under flywheel,

$$(M_b)_V = (R_E)_v * c_2 = 159584.080 Kg - mm$$

$$(M_b)_H = (R_E)_v * c_2 = 306439.576 Kg - mm$$

$$(M_b) = \sqrt{(M_b)_V^{\,2} + (M_b)_H^{\,2}} = 345502.955 Kg - mm$$

Diameter of shaft,

$$d_s = 107.55 mm$$

Case-II: On the basis of maximum torsional moment.

TABLE III
Reaction forces at different locations of crankshaft

Point	Description	Reaction(kg)		Moment(kg-mm)	
		Horiz.	Vert.	Horiz.	Vert.
A	Bearing1	-588.8	-118.4	0	0
B	HP Crank	-1740	-1570	-46775.1	13264.8
C	LP Crank	3460	-3120.9	124904.7	173986.1
D	HP Crank	-1740	-1570	-152562	-41364.1
E	Bearing2	352.9	353.3	214937.5	56769.3
F	Flywheel	961.6	254.0	0	0

Bending moment is highest at point D. So resultant bending moment,

$$M_d = \sqrt{(M_{DH})^2 + (M_{DV})^2} = 219.191*10^3 Kg - mm$$

Torsional moment at D is given by,

$$M_t = F_t.r = 130509 Kg - mm$$

$$(M_{te}) = \sqrt{(M_d * K_b)^2 + (M_t * K_t)^2} = 353738.339 Kg - mm$$

Diameter of shaft under shear strength,

$$\tau = \frac{\Pi}{16 * d_c^{\,3}} M_{te}$$

$$d_c = 43.01 mm$$

Diameter of shaft under bending strength,

$$M_{be} = \frac{1}{2}\left[M_b * K_b + M_{te}\right] = 341262.169 Kg - mm$$

$$d_c = \sqrt[3]{\frac{32 * M_{be}}{\pi * \sigma_b}} = 45.15 mm$$

Design of crankpin,

$$(M_{te}) = \sqrt{(M_b * K_b)^2 + (M_t * K_t)^2} = 110485.7 Kg - mm$$

Diameter on the basis of shear strength,

$$d_c = \frac{16}{\pi * \tau} M_{te} = 49.839 mm$$

Diameter on the bending strength,

$$M_{be} = \frac{1}{2}\left[M_b * K_b + M_{te}\right] = 105687.1 Kg - mm$$

$$d_c = \sqrt[3]{\frac{32 * M_{be}}{\pi * \sigma_b}} = 30.5 mm$$

Design of web:

Bending moment due to radial component,

$$(M_b)_r = (R_E)_v * [b_2 - 0.5l_c - 0.5t] = 40097.3 Kg - mm$$

Bending stress due to radial component,

$$(\sigma_b)_r = \frac{(M_b)_r}{\frac{1}{6} * w * t^2} = 2.823 Kg / mm^2$$

Bending moment due to tangential component,

$$(M_b)_t = P_t * \left(r - \frac{d_{s1}}{2}\right) = 44342.5076 Kg - mm$$

Bending stress due to tangential component,

$$(\sigma_b)_t = \frac{(M_b)_t}{\frac{1}{6} * w^2 * t} = 1.92 Kg / mm^2$$

Direct compressive stress due to radial component,

$$(\sigma_c)_d = \frac{p_r}{2 * w * t} = 0.3506 Kg / mm^2$$

Maximum compressive stress,

$$\sigma_c = (\sigma_b)_t + (\sigma_b)_r + (\sigma_c)_d = 5.0936Kg/mm^2$$

Torsional moment

$$(M_t) = (R_2)_h \left[b_2 - \frac{l_c}{2} \right] = 208.053*10^3 Kg-mm$$

$$(M_b)_t = \frac{4.5M_t}{w*t^2} = 11.0068Kg/mm^2$$

Maximum compressive stress

$$(M_b)_t = \frac{\sigma_c}{2} + \frac{1}{2}\sqrt{(\sigma_c)^2 + 4\tau^2} = 12.82Kg/mm^2$$

Diameter of shaft under flywheel,

$$(M_b) = (R_E * c) = 111628.349Kg-mm$$

$$(M_t) = (P_t * r) = 259500Kg-mm$$

Diameter on the basis of shear strength,

$$(d_c)^3 = \frac{16}{\pi * \tau}\sqrt{M_b{}^2 + M_t{}^2} = 39.89mm$$

Diameter on the basis of bending strength,

$$(d_c)^3 = \frac{32 * M_E}{\pi * \sigma_b} = 37.60mm$$

3. ANALYSIS:

3.1 STATIC ANALYSIS OF CRANKSHAFT:

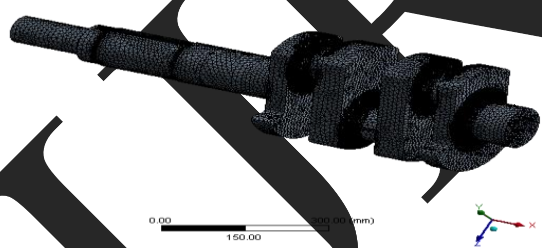

Figure 3.1: Meshing of Crankshaft

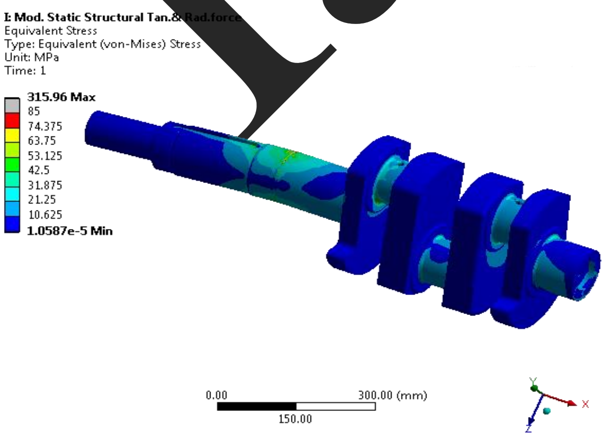

Figure 3.2: Equivalent von-mises stress for Crankshaft

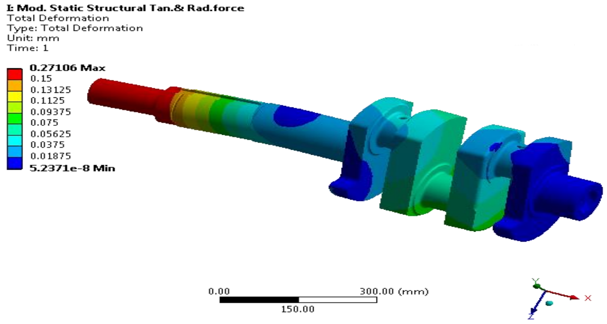

Figure 3.3: Total Deformation of Crankshaft

3.2 TRANSIENT ANALYSIS OF CRANKSHAFT

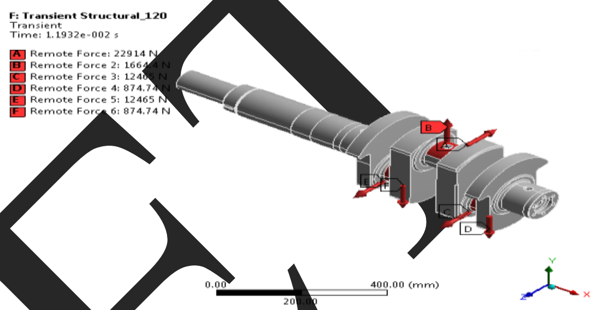

Figure 3.4: Boundary condition for Transient analysis of Crankshaft

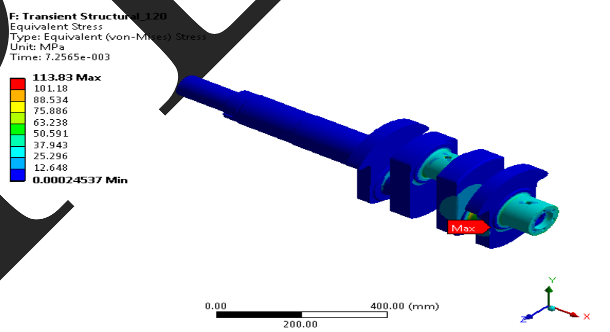

Figure 3.5: Equivalent von-mises stress for Crankshaft in Transient

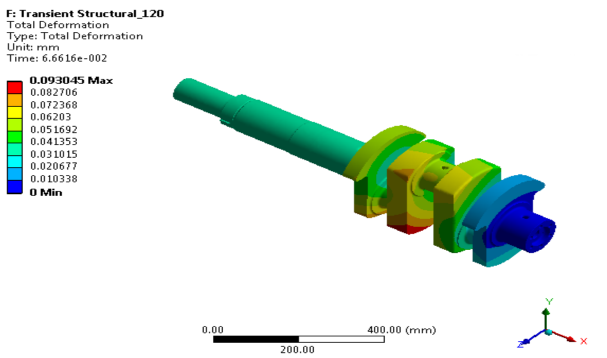

Figure 3.6: Total Deformation of Crankshaft in Transient

4. RESULT AND DISCUSSION:

From FEA static analysis maximum stress on crankshaft is 67.67 MPa and in transient analysis it is 82.05 MPa. Allowable value of stress for crankshaft is 90 MPa. The maximum deformation obtained in static analysis is

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0.27 mm and in transient analysis 0.09mm, which is very small. Thus crankshaft design is safe.

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