Purdue University
Purdue e-Pubs

International Compressor Engineering Conference | School of Mechanical Engineering

2010

Simple Modeling and Modal Analysis of Reciprocating Compressor Crankshaft System

Binyan Yu
School of Energy and Power

Xiaoling Yu
School of Energy and Power

Quanke Feng
School of Energy and Power

Follow this and additional works at: http://docs.lib.purdue.edu/icec

Yu, Binyan; Yu, Xiaoling; and Feng, Quanke, "Simple Modeling and Modal Analysis of Reciprocating Compressor Crankshaft System" (2010). *International Compressor Engineering Conference.* Paper 1982.
http://docs.lib.purdue.edu/icec/1982

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html

Simple Modeling and Modal Analysis of Reciprocating Compressor Crankshaft System

Bin-yan YU, Xiao-ling YU, Quan-ke FENG*

School of Energy and Power, Xi'an Jiao tong University, Xi'an 710049, China

Tel (Fax): +86-029-82663783 Email: qkfeng@xjtu.edu.cn

ABSTRACT

To simplify the analysis of the three-dimension vibrations of reciprocating compressor crankshaft system under working conditions, a spatial finite element model based on 3-node Timoshenko beam was proposed in this paper. The crankshaft was idealized by a set of jointed structures consisting of simple round rods and simple beam blocks, the main journal bearings were idealized by a set of linear springs and dash-pots, and the flywheel and motor were idealized by a set of masses and moments of inertia. And this model also can be used in dynamic analysis for crankshaft system in subsequent research. In practice, a 6M51 reciprocating compressor crankshaft system was modeled by this method and modal analysis by block Lanzos method which provided by ANSYS package.

1. INTRODUCTION

The design of modern reciprocating compressor is required to make noise and vibration as light as possible. 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 analyzed. 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.

Reciprocating compressor's crankshaft system vibration is a complex three-dimensional coupled vibration under running conditions, including the torsional, longitudinal and lateral vibrations. In earlier research, the equivalent model and the continuous beam model were applied to analyze the torsional or coupled torsional-axial vibration of crankshaft (Tsuda, 1969, Porter, 1968). With the progress of computer simulation technique and emergence of finite element method, spatial finite element models were introduced to estimate the vibration behavior of crankshaft. The generally used finite element models are in two categories: beam elements and solid elements. It is evident that much better results can be obtained by using the models based on solid elements (Kang and Sheen 1998). However, it is computationally expensive to simulate the motion of the operating crankshaft.

Initial, the finite element beam model was proposed by Ruhl and Booker for rotor-bearing systems (1972). Then Nelson and McVaugh (1976), Zorzi and Nelson (1977) utilized finite beam element models to formulate the dynamic equation for a linear rotor system and determine the stability and steady state responses. In order to study the vibrations of the crankshaft, Okmaura.H (1990 and 1995) proposed a finite element model based on the Euler-Bemouli beam for crankshaft, and derived the DSM(Dynamics Stiffness Matrix) in closed form from

TMM(Transfer Matrix Method). And in practice, they have calculated a single crank, three-cylinder in-line, four-cylinder in-line and V-6 crankshafts' bending and axial three-dimensional vibration.

As the Euler-Bemouli beam neglected the shear deformation and rotational inertia effects, a number of Timoshenko beam finite element which considering the shear deformation and rotational inertia effects have been proposed in the literature (Kapur, 1966, Severn, 1970, Davis and Henshell, 1972). And Thomas et *al*. (1973) proposed a new Timshenko element beam which has three degrees of freedom at each of two nodes, and this element has good rate of convergence, and for the analysis of simple structures in which shear and rotary inertia are particularly important, this element will give a better rate of convergence. Vebil Yildirim and Erhan Kiral (2000) compared the calculated results of the Euler-Bemouli beam with Timoshenko beam, the results showed that to determine a more satisfactory result in the analysis of free vibration and forced vibration of the thinner component beam, the shear deformation and rotational inertia must be considered. Smaili (1994) proposed a new finite element model involving a new scheme for modeling the stiffness and damping properties of the journal bearings, and the natural frequencies of the crankshaft of a four in-line cylinder engine are determined.

In this paper, a finite element model was proposed by using a 3-node spatial element based on Timoshenko beam theory which provided by ANSYS package for modal analysis of crankshaft. The crankshaft was idealized by a set of simple round and block beams. The flywheel and motor were idealized by a set of masses and moments of inertia. The main journal bearings were idealized by a set of linear spring and dash-pots. And in practice the natural frequencies and mode shapes were calculated by the finite element method of a 6M51 reciprocating compressor crankshaft system. And this model also can be used in dynamic analysis for crankshaft system in subsequent research.

2. MODELING

2.1 Crankshaft

Crankshaft is considered to be a set of rigidly jointed structures consisting of crankjournals crankpins and crankarms. The crankjournals and crankpins of each throw were idealized by rods of diameters D_j and D_p and lengths L_j and L_p , as shown in Figure 1. Here D_j and D_p were as equal to their original diameters, while L_j and L_p were equal to the length of the original journal together with the thickness of the crankarm H in accordance with Wlisno and Okmauar (1990). The crankarms was idealized by blocks, which dimensions were equal to the original dimensions so as to keep their centers of gravity, masses, moments and stiffness of the same.

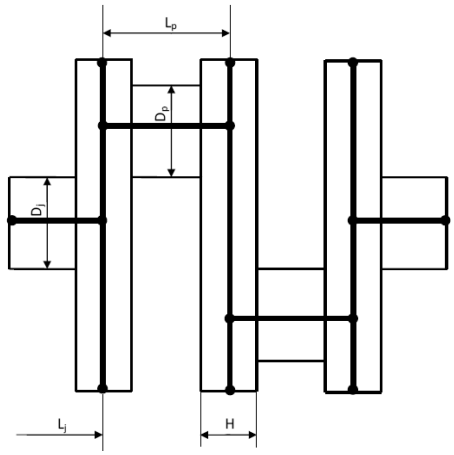


Figure 1: Sketch of the computing modal of crankshaft

2.2 Flywheel and Motor

The flywheel and motor were idealized by a set of masses and moments of inertia about three orthogonal axes attached at their original centers of gravity.

2.3 Crankshaft Main Bearings

To take account of oil film pressure of each main bearing (journal bearings), we idealized a set of linear springs and dashpots in the vertical and horizontal directions attached to the crankjournal axis.

2.4 Reciprocating Masses

The reciprocating masses consist of the masses of the piston, connecting rod, piston rod and crosshead. To take account of the mean kinetic energy that would be in duce by the reciprocating masses under running conditions, one quarter of the total reciprocating masses were attached at the two crankpin ends.

3. ANALYSIS

ANSYS provide element Beam189 is based on quadratic 3-node Timoshenko beam theory which includes shear-deformation effects. It is an element including shear deformation and rotary inertia effects and suitable for analyzing slender to moderately stubby and thick beam structures. The element is a quadratic three-node beam element in 3-D (as in Figure.2).

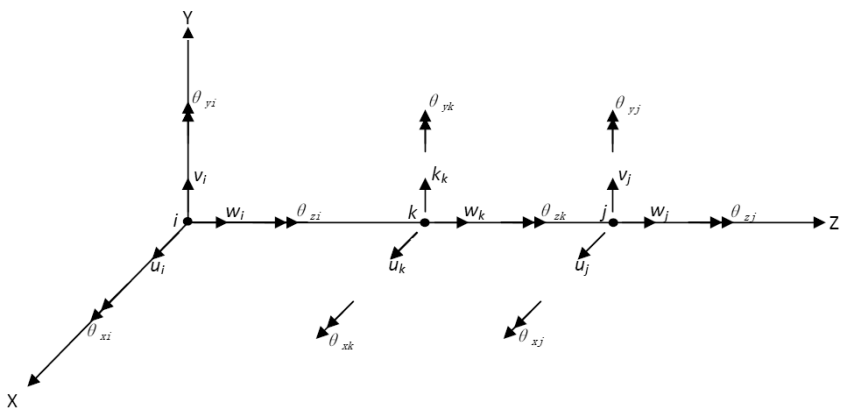


Figure 2: 3-node Timoshenko beam element

With default settings, six degrees of freedom (DOF) occur at each node; these include translations in the x, y, and z directions and rotations about the x, y, and z directions. An optional seventh DOF (warping magnitude) is available, the DOF are defined by

$$\{q\} = [u, v, w, \theta_x, \theta_y, \theta_z, warp]^T \tag{1}$$

The shape function is

$$\mathrm{N_i} = \frac{1}{2}\eta\ (\eta - 1), \mathrm{N_j} = \frac{1}{2}\eta\ (\eta + 1), \mathrm{N_k} = 1 - \eta^2 \tag{2}$$

Where η the nature coordinate ($-1<\eta<1$). Then the displacement of point p is related to the element nodal displacement q_i, q_j and q_k by

$$q_p = N_i \cdot q_i + N_j \cdot q_j + N_k \cdot q_k \tag{3}$$

q_i, q_j and q_k is the vector of displacement at nodes i, j, and k respectively.

The motion equations of crankshaft are first discredited by the standard finite element technique as

$$[M]\{\ddot{q}\} + [C]\{\dot{q}\} + [K]\{q\} = [f] \tag{4}$$

where $\{q\}$ is the vector of displacements, $[f]$ is the vector of externally applied forces, and $[M]$, $[C]$ and $[K]$ are mass, damping and stiffness matrices respectively.

In order to determine natural frequencies or the eigenvalues of the system, omission of the damping and external force terms from the equation of the system leads to

$$[M]\{\ddot{q}\} + [K]\{q\} = \{0\} \tag{5}$$

Assumed the solution can be separated to the amplitude and a time function, we substituted

$$\{q\} = \{Q\}\sin(\omega_n t + \varphi) \tag{6}$$

into Equation (5) , we can derive the flowing equation:

$$([K] - \omega_n^2[M])\{Q\} = \{0\} \tag{7}$$

Since Equation (6) is not the standard form for the eigenvalue and eigenvector problems, we can premultiply $[M]'$ both side of the Equation (6), we can derive the flowing equation:

$$([S] - \lambda[I])\{Q\} = \{0\} \tag{8}$$

where $[S]=[M]'[K]$, called system matrix. In order that $\{Q\}$ has a value not equal to zero, the following condition must be satisfied:

$$\det([S] - \lambda[I]) = 0 \tag{9}$$

Solving this equation for λ, we obtain the eigenvalue $\lambda_n = i\omega_n$ of each mode of vibration of the system. So the vibration problem transform into the eigenvalue problem. In this paper, we utilized block Lanczos method which offered by ANSYS package to determine the eigenvalue of the Equation (9). Block Lanczos method performed with a block of vectors which is used by the block Lanczos method. It is powerful when searching for eigenvector.

The mode shapes can be calculated by substituting the λ_n back into Equation (8) and solving for $\{Q\}$ in the normalized form.

4. APPLICATION

A 6M51 reciprocating compressor crankshaft is modeled by the proposed 3-node Timoshenko beam, in which the crankshaft is meshed into 136 beams element, the journals is meshed into 16 linear springs and dashpots element, the reciprocating masses, motor and flywheel is meshed into 14 masses element (as in Figure 3, dimensions of member elements, mass and moments of inertia shown in Table 1). These models were analyzed by using the block Lanczos method provided by ANSYS package. The calculated results of the first eight orders natural frequencies and the mode of vibration are listed in Table 2, the expression “L” refers to the lateral vibration and

"T" refers to torsional vibration. The first eight orders calculated mode shapes of the crankshaft were shown in Figure 4.

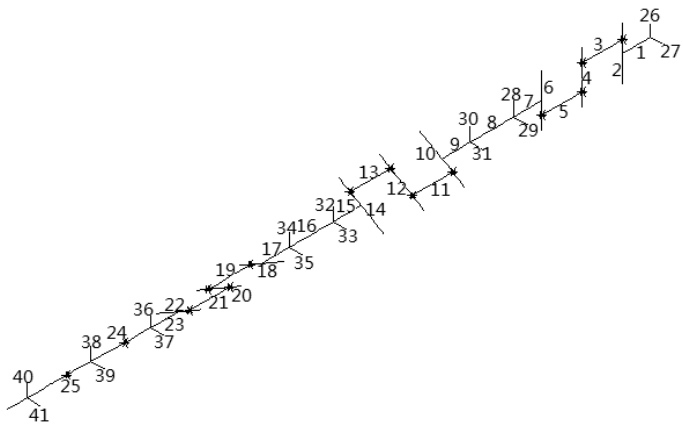


Figure 3: Idealized modeling for 6M51 crankshaft system

Table 1: Dimensions of member elements (a) and
mass and moments of inertia (b) for 6M51 crankshaft system

(a)

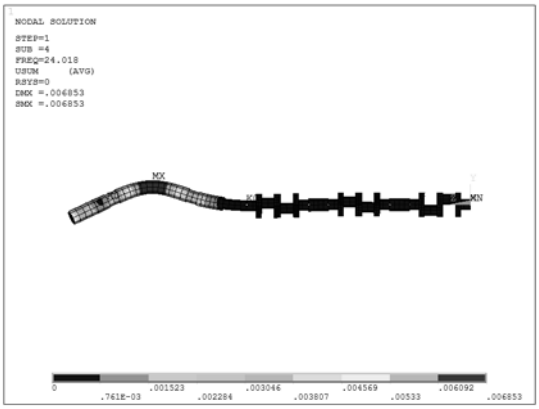
Element				Dimension (mm)
Parts	**Number**	**Type**	**Real Constant**	
Journal	1, 7, 9, 15, 17, 23	Beam 189	DENS=7850 kg / m^3 EX=2.1e11 PRXY=0.3	393×Φ320
Crank pin	3, 5, 11, 13, 19 22			565×Φ320
Crank arm	2, 4, 6, 10, 12, 14, 18, 20, 22			750×410×165
Connection shaft	8, 16			840×Φ398
	24			1515×Φ320
Motor shaft	25			3944×Φ400
Idealized journal	26, 27, 28, 29, 30,31, 32, 33, 34, 35, 36, 37, 38,39, 40, 41	Combin 14	K=1e9 N / m	Null

(b)

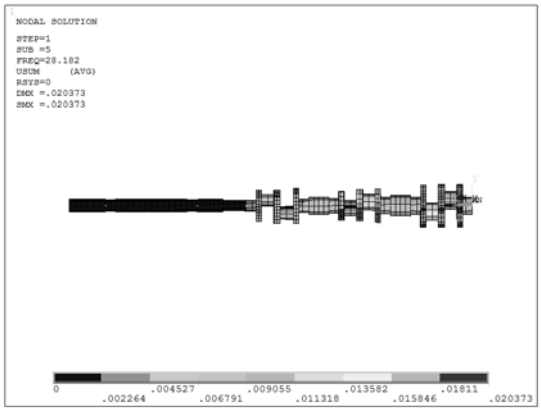
Parts	**Mass (kg)**	**Moment of inertia (kg·m^2)**		
		Ix	**Iy**	**Iz**
Reciprocating masses	592	Null		
Flywheel	1460	355	355	710
Motor	19500	4375	4375	8750

Table 2: First eight calculated natural frequencies of 6M51 crankshaft

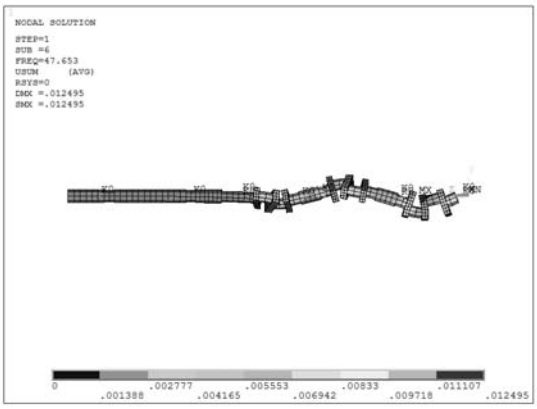
Mode	Calculated Natural Frequency (Hz)	Mode of Vibration
1st	24.014	L
2nd	28.182	T
3rd	47.653	L
4th	50.940	L
5th	51.538	L
6th	54.572	T & L
7th	58.980	L
8th	60.319	L



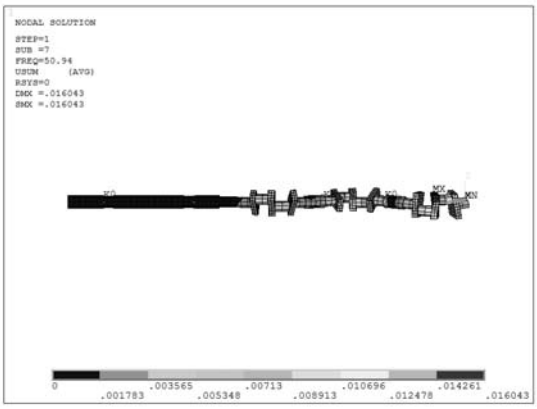
(a) ω=24.014 Hz



(b) ω=28.182 Hz



(c) ω=47.653 Hz

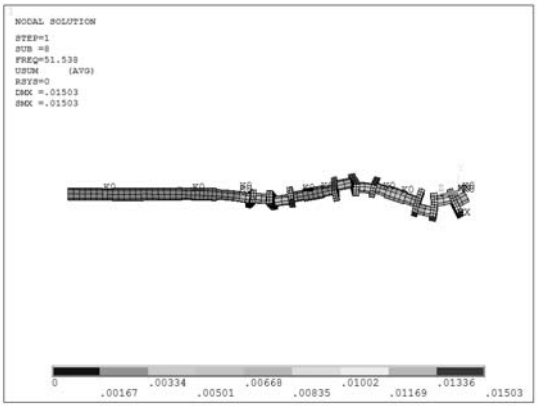


(d) ω=50.940 Hz

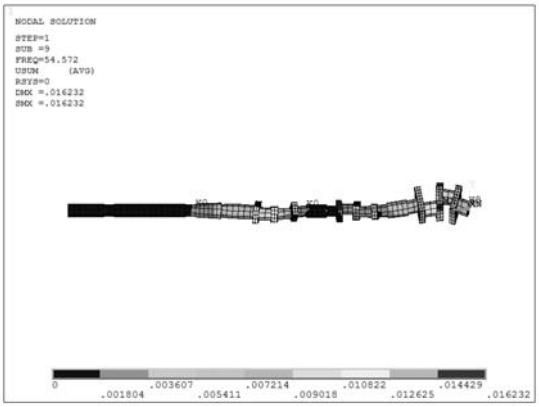
5. Conclusion

A simple model and analysis procedure was proposed in this paper for three-dimensional vibrations of reciprocating compressor crankshaft system, which is very simple and the implementation of simulation is computationally effective. It is competent for the vibration analysis of the crankshaft system of reciprocating compressor. It provides a useful tool for the vibration analysis and the design of crankshaft system. And in

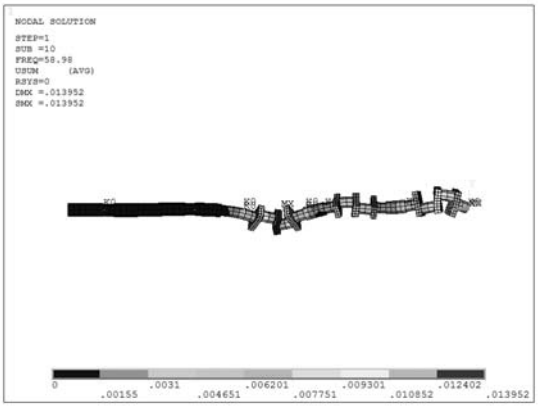
practice, the nature frequencies and mode shapes of a 6M51 reciprocating compressor crankshaft system was calculated by this model and analysis procedure which provide by ANSYS package. The proposed model also can be implemented in dynamic analysis for crankshaft system in subsequent research.



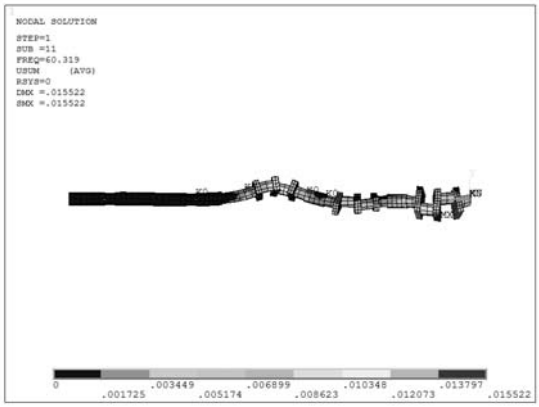
(e) ω=51.538 Hz



(f) ω=54.572 Hz



(g) ω=58.980 Hz



(h) ω=60.319 Hz

Figure 4 (a-h): First eight orders calculated modes of 6M51 crankshaft

References

Davis K. and Henshell R D., 1972, A Timoshenko beam element [J]. *Journal of Sound and Vibration*, vol. 22: p. 475-487.

Kang Y. and Sheen G J., 1998, Modal analyses and experiments for engine crankshaft [J]. *Journal of Sound and Vibration*, vol. 214, no.3: p. 413-430.

Kapur K K., 1966, Vibrations of a Timoshenko beam, using finite element approach [J]. *Journal of the Acoustical Society of America*, vol. 40: p. 1058-1063.

Nelson H D. and Mc Vaugh J M., 1976, The dynamics of rotor- bearing systems using finite elements [J]. *ASME Journal of Engineering for Industry*, vol. 98: p. 593-600.

Okamura H. and Shinno A., 1990,Dynamic stiffness matrix approach to the analysis of three-dimensional vibrations of automobile engine crankshafts. Part 1: Background and application to free vibrations [J]. *ASME Noise Control and Acoustics Division*, vol. 9: p. 47-58.

Okamura H. and Morita T., 1995, Simple modeling and analysis for crankshaft three-dimensional vibration. Part 2: Application to an operating engine crankshaft [J]. *Journal of Vibration and Acoustics, Transactions of the ASME*, vol.117, no.1: p. 80-86.

Ruhl R. and Booker J F., 1972, A finite element model for distributed parameter turbo-rotor systems [J]. *ASME Journal of Engineering for Industry*, vol.94: p. 128-132.

Severn R T., 1970, Inclusion of Shear deformation in the stiffness matrix for a beam element [J]. *Journal of Strain Analysis*, vol. 5: p. 239-241.

Smaili A A. and Khetawat M P., 1994, Dynamic modeling of automotive engine crankshafts [J]. *Mechanism & Machine Theory*, vol. 29, no. 7: p.995-1006.

Thomas D L. and Wilson J M. and Wilson P R., 1973, Timoshenko beam finite elements [J]. *Journal of Sound and Vibration*, vol. 31: p. 315-330.

Tsuda K., 1969, Theoretical analysis of couple torsional—axial undammed vibration of marine diesel engine shaft [J]. *Jap Shipbldg Mar Eng*, vol. 4,no. 1: p. 15-30.

Porter F P., 1968, Crankshaft Stress Analysis and Bearing Load-Carrying Capacity, *ASME Paper*, p.1-12.

Yildirim V. and Kiral E., 2000,Investigation of the rotary inertia and shear deformation effects on the out-of-plane bending and torsional natural frequencies of laminated beams [J]. *Composite Structures,* vol.49: p. 313-320.

Zissimos P. Moutrlatos., 2001, A crankshaft system model for structural dynamic analysis of internal combustion engines. *Computers & Structures*, vol.79: p. 2009-2027.

Zorzi D L. and Nelson H D., 1977, Finite element simulation of rotor-bearing systems with internal damping [J]. *ASME Journal for Power*, vol. 99: p. 71-76.