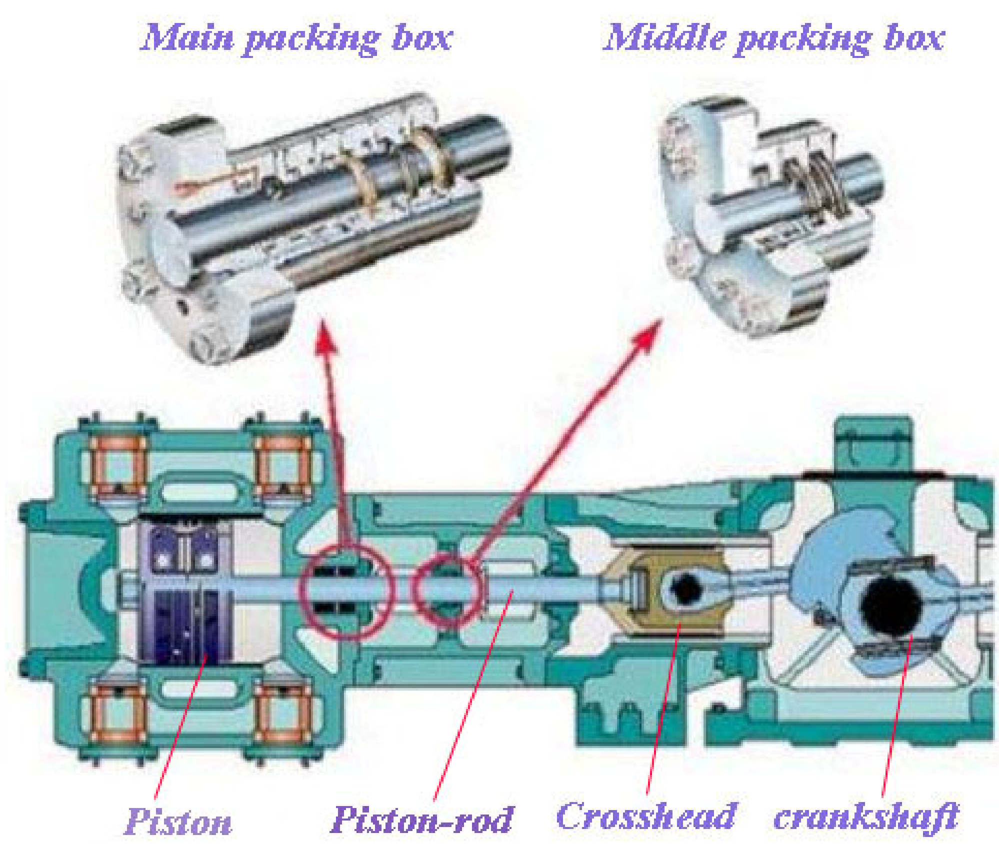
Simulation of Crankshaft Torsional Vibration by Flexible-body Dynamics

JunMing Cheng
Xi’an Jiaotong University, China

Introduction

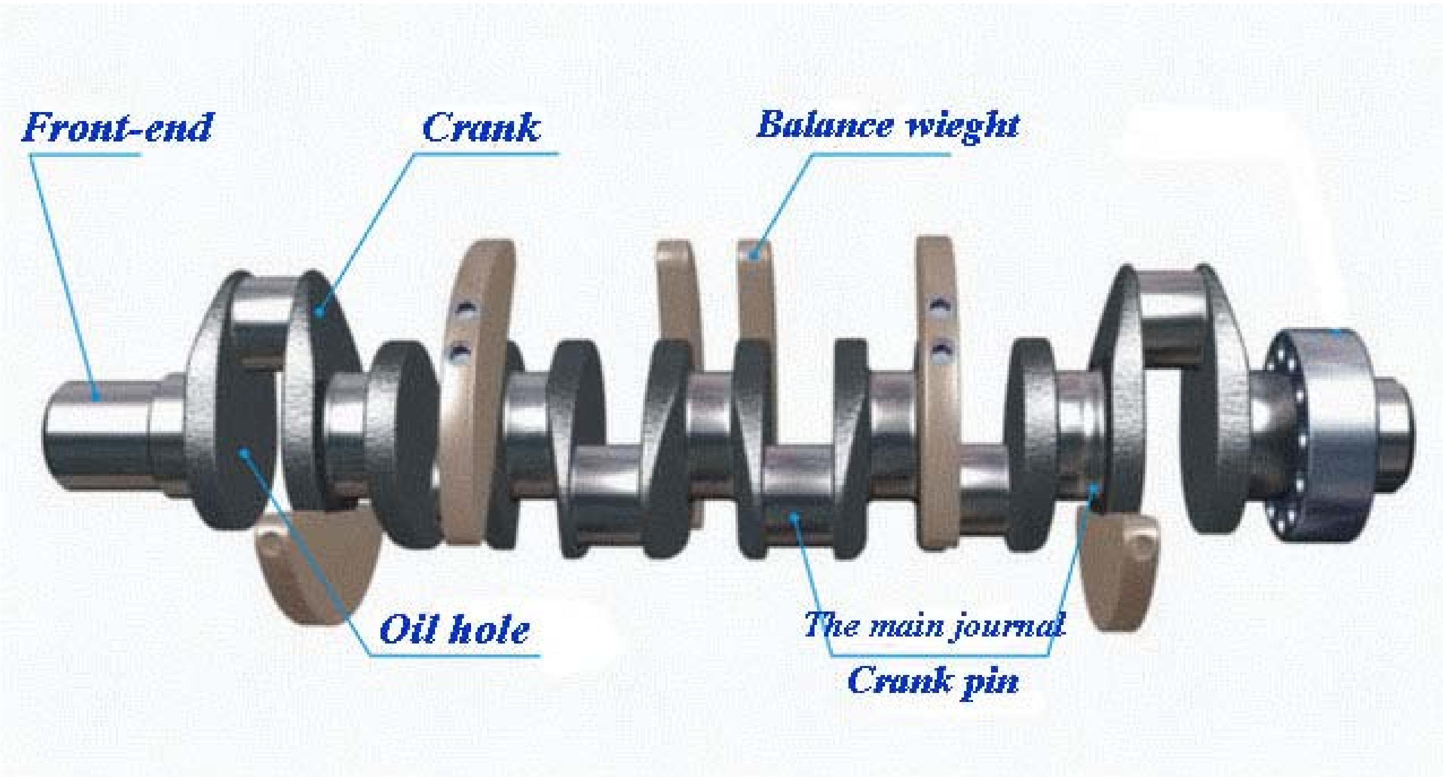
- The reciprocating compressor, as core equipment, plays an increasingly important role in the modern petrochemical industry.
- The large-scale muti-row reciprocating compressor is rapidly expanding



Introduction

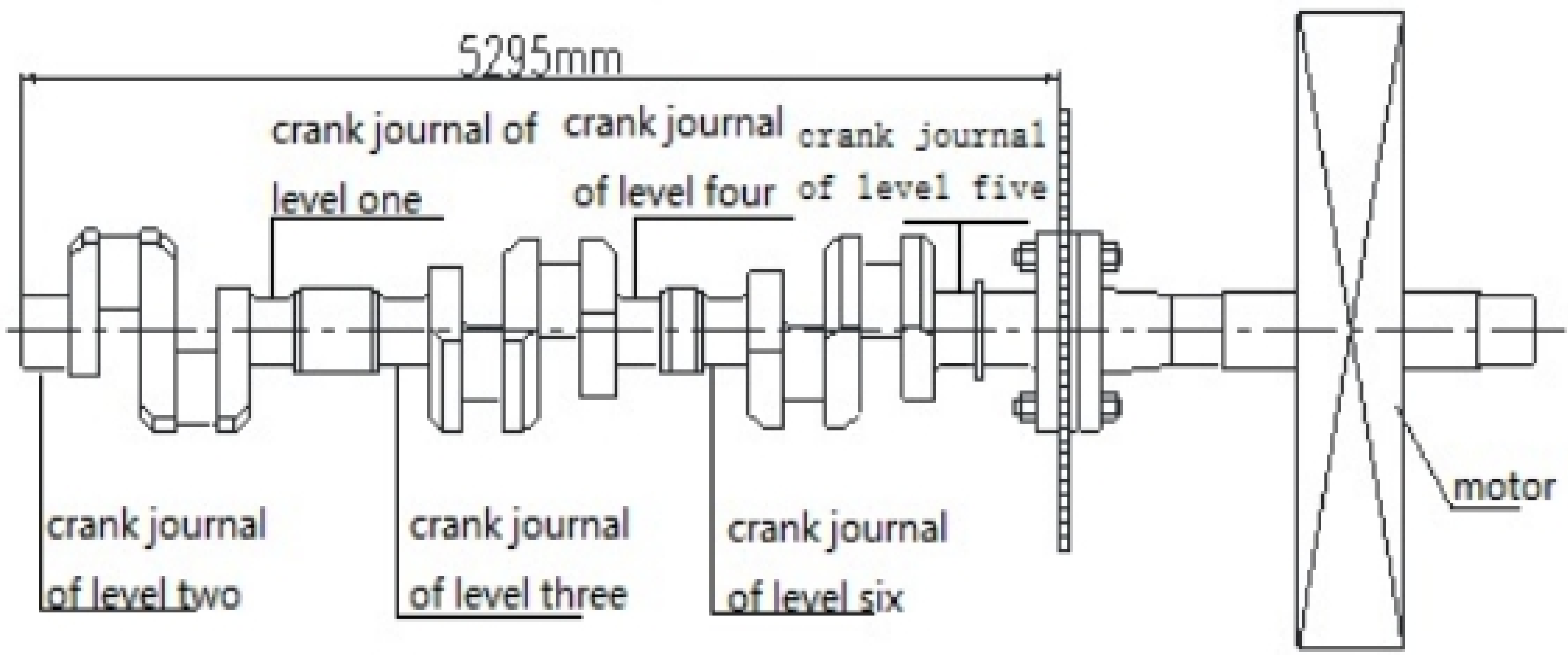
➢The crankshaft is the main loading part of reciprocating compressor

➢the crankshaft torsional vibration has become the bottleneck of development of the large-scale multi-row reciprocating compressor.



The engineering case

- *We choose the 6M50 reciprocating compressor with 6 rows and the maximum piston force of 50 tons as an example to study.*
- *The crankshaft material is 35CrMo, and it has 6 cranks.*
- *Each cylinder was arranged in one row*



The engineering case

we choose the 6M50 reciprocating compressor as an example to study

The main technical parameters of 6M50 compressor as follows

List	Value
The number of rows	6
Suction pressure(*MPa*)	0.1
discharge pressure(*MPa*)	31.4
The stroke of pistons(*m*)	0.45
Rotational speed($r \cdot min^{-1}$)	300
Crank throw(*m*)	0.175
The length of the connecting rod(*m*)	0.95
The maximum gas force(*kN*)	380
The maximum piston force(*kN*)	363
The diameter of the main shaft(*mm*)	300
The cylinder diameter of level one(*mm*)	1390

The engineering case

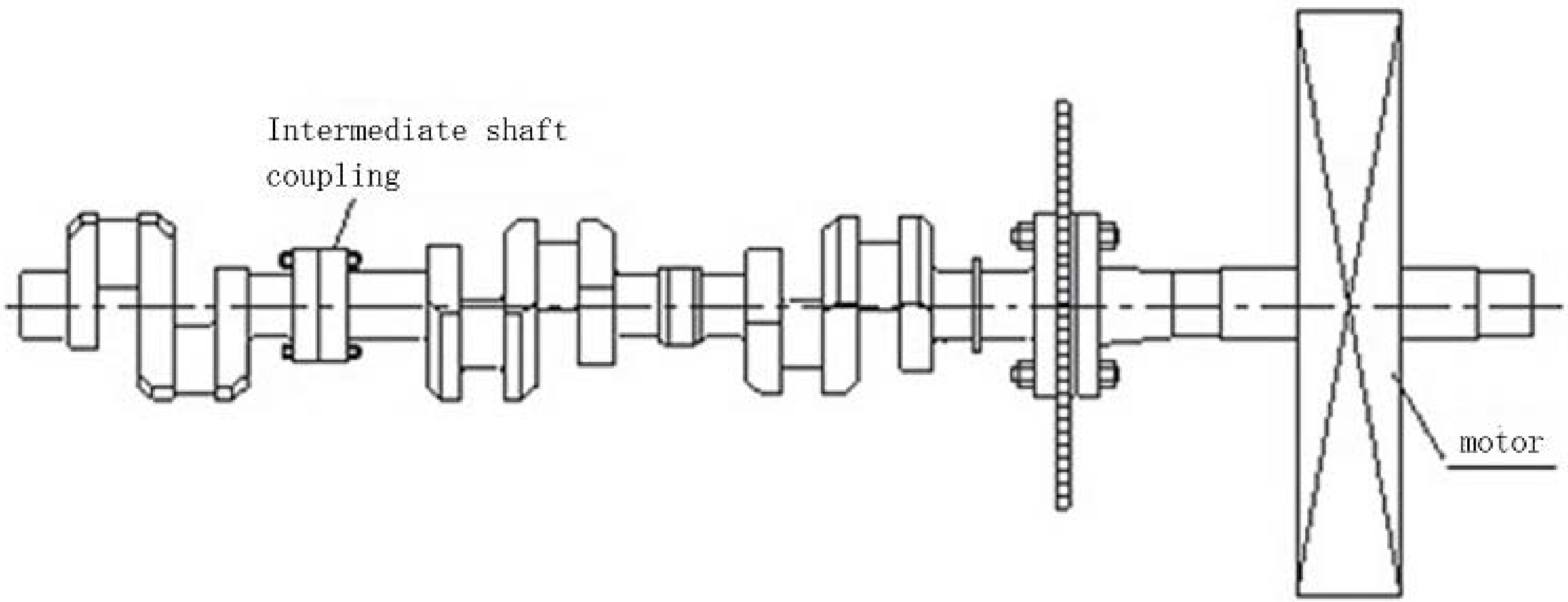
- *The major accidents happened on the crankshaft are the crank pins of the first row and the second column scraping frequently*
- *The crankshafts average life were only 202h and 304h respectively*

Crank pin

Crankshaft

The engineering case

- In order to eliminate the accidents, the company made a new crankshaft from the two waste crankshaft
- the first and two rows were connected with the other four row-by a rigid coupling



Simulation model

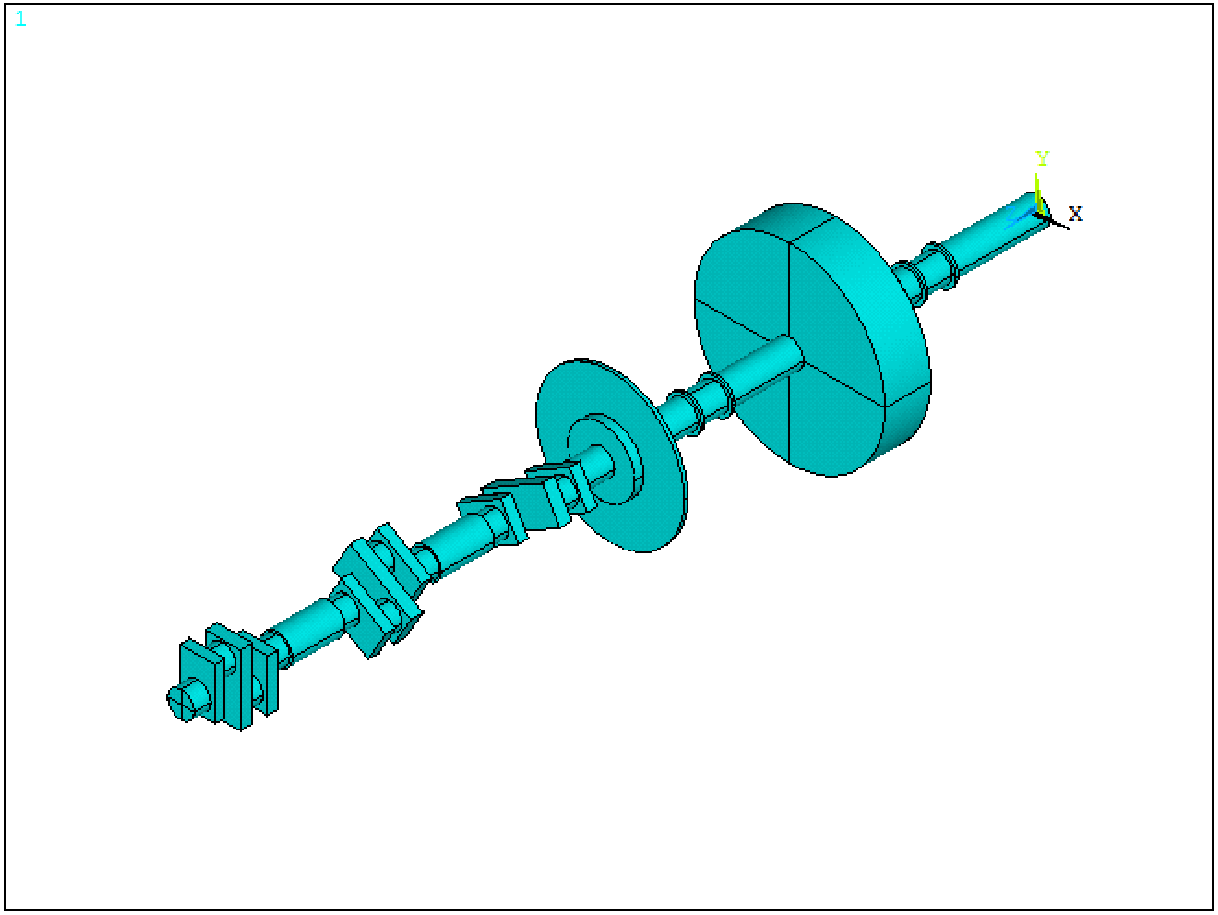
- *To analyze the crankshaft accidents, this paper proposed a new calculation method of the crankshaft torsional vibration based on flexible multi-body dynamics theory.*
- *The new crankshaft and the old one were both simulated by ADAMS respectively, and analyzed the crank pin failure reasons.*

The modeling steps as follows

Simulation model

1) *The crankshaft MNF files*

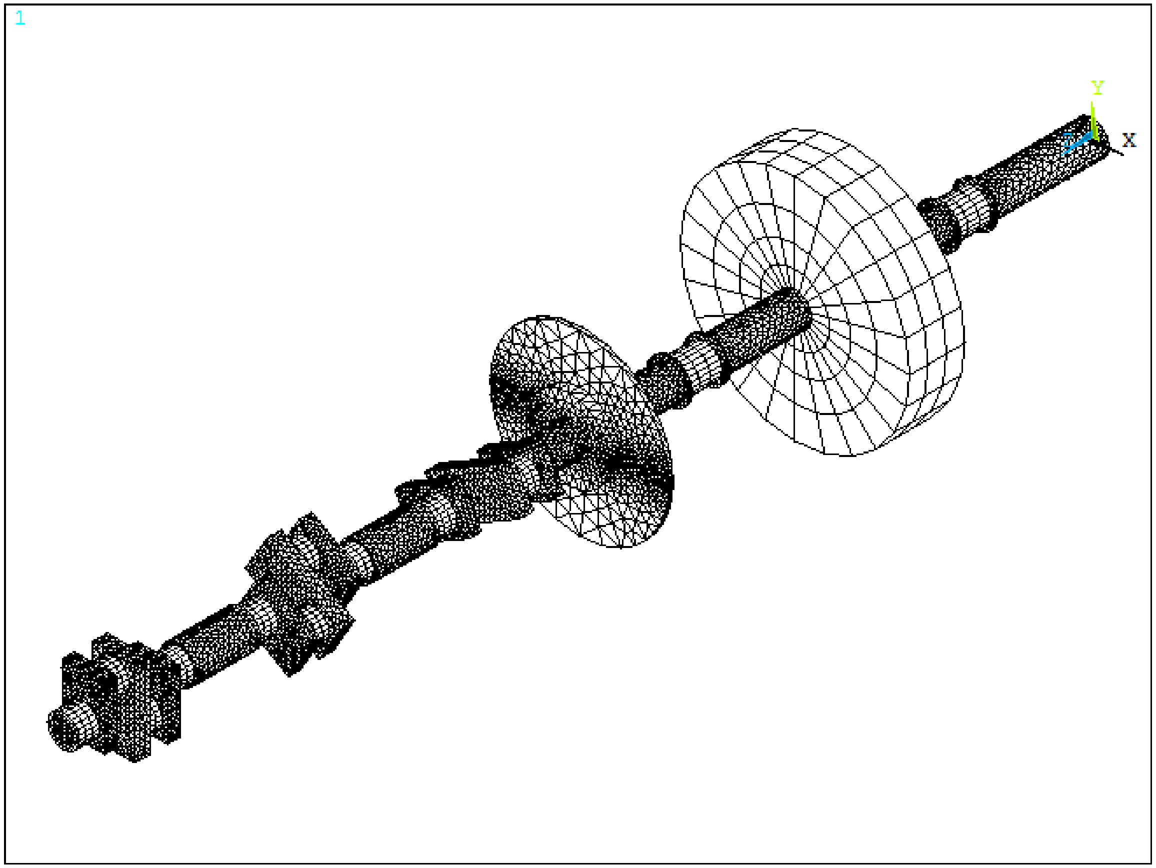
Firstly, the crankshaft 3-D solid model was established in 3-D software



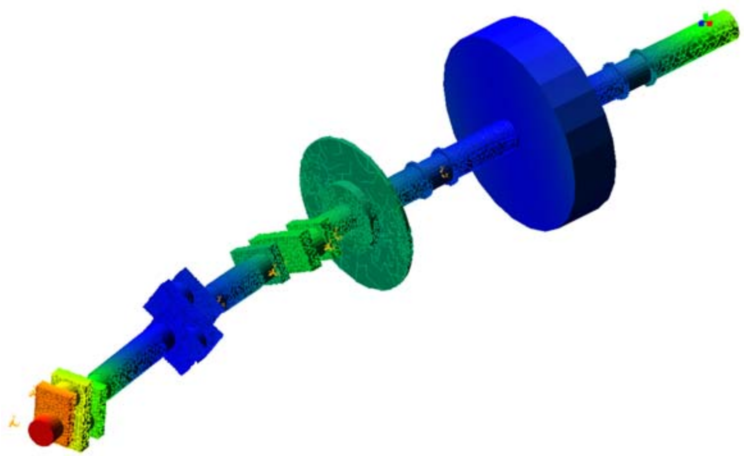
Simulation model

2) the 3-D solid model was imported into ANSYS software, and physical parameters of the crankshaft were input. The crankshaft model is divided into 99519 elements of SOLID73.

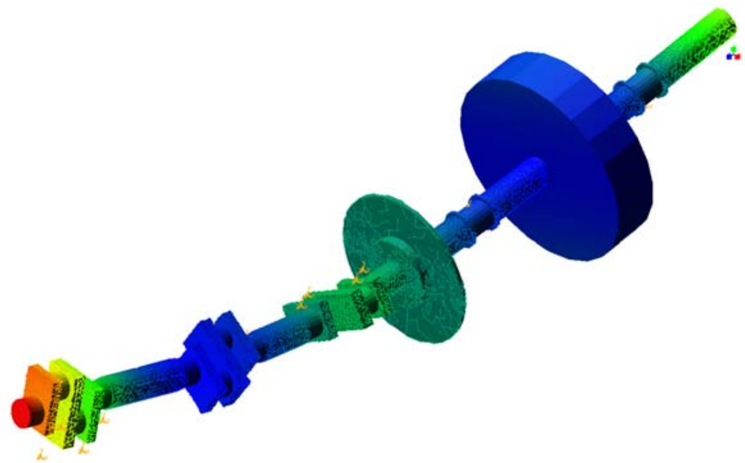
3) At last a MNF file was output to ADAMS for torsional vibration model establishment



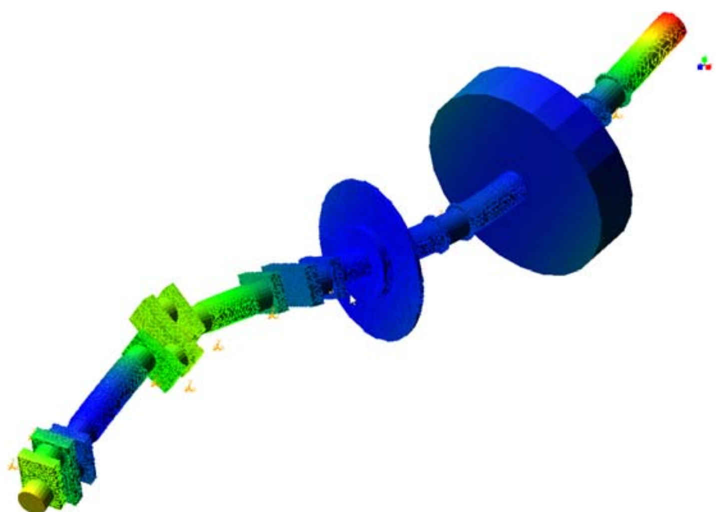
Simulation model



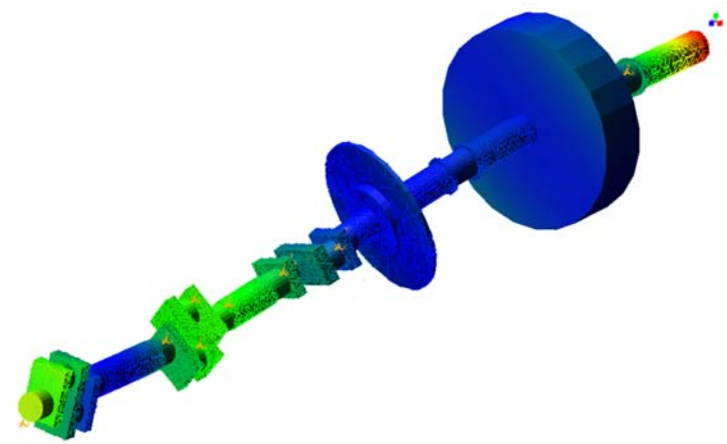
The seven-order



The eight-order



The nine-order



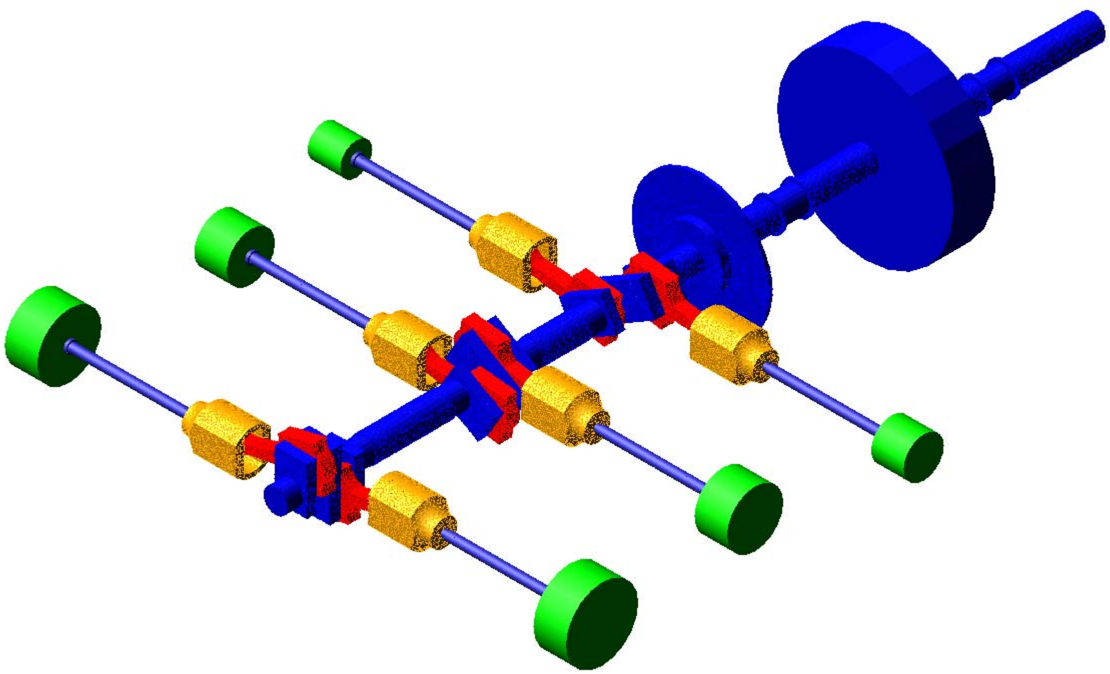
The ten-order

Simulation model

order	*frequency(Hz)*	*mode*
7	7.36	The first-order bending in XOZ plane
8	7.94	The first-order bending in YOZ plane
9	21.54	The second-order bending in XOZ plane
10	22.61	The second-order bending in YOZ plane
11	37.87	The first-order torsion
12	39.24	The third-order bending in XOZ plane

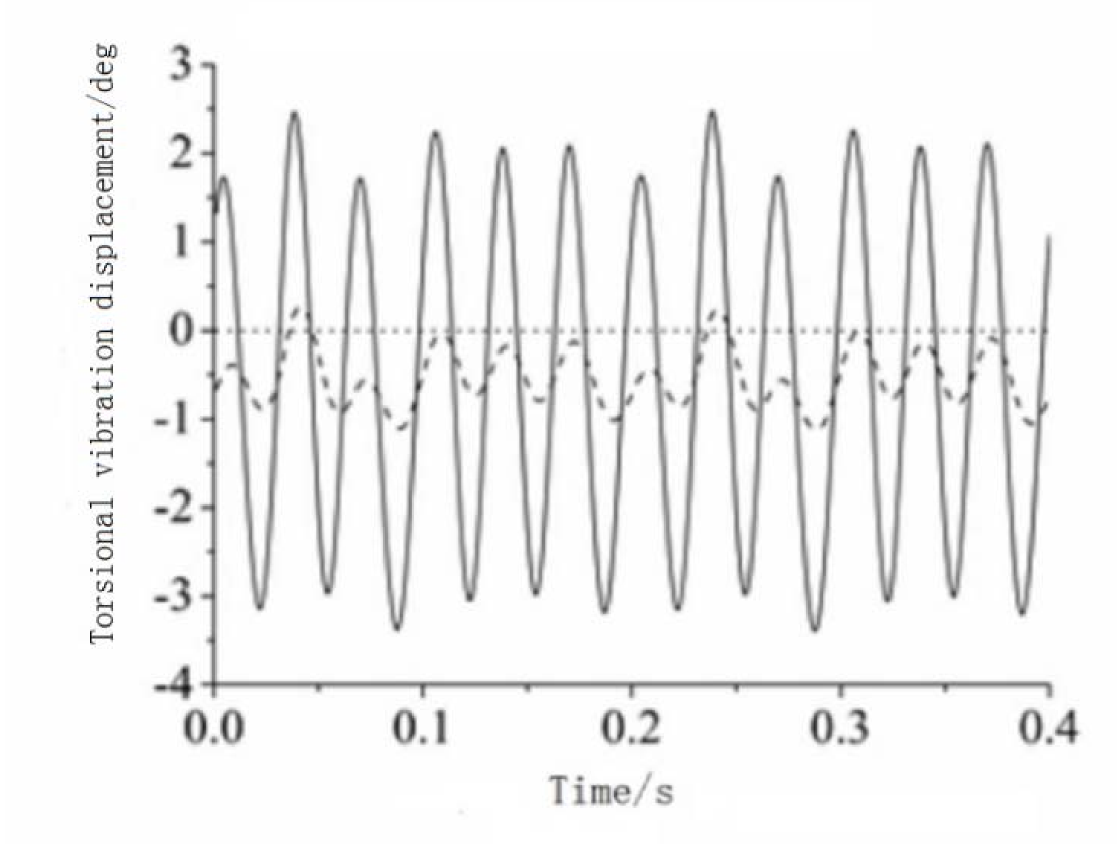
Simulation model

- *In ADAMS the crankshaft is connected with the connecting rod, cross head, piston rod and the piston*
- *Add the movement and boundary conditions to the model*



Analysis of result

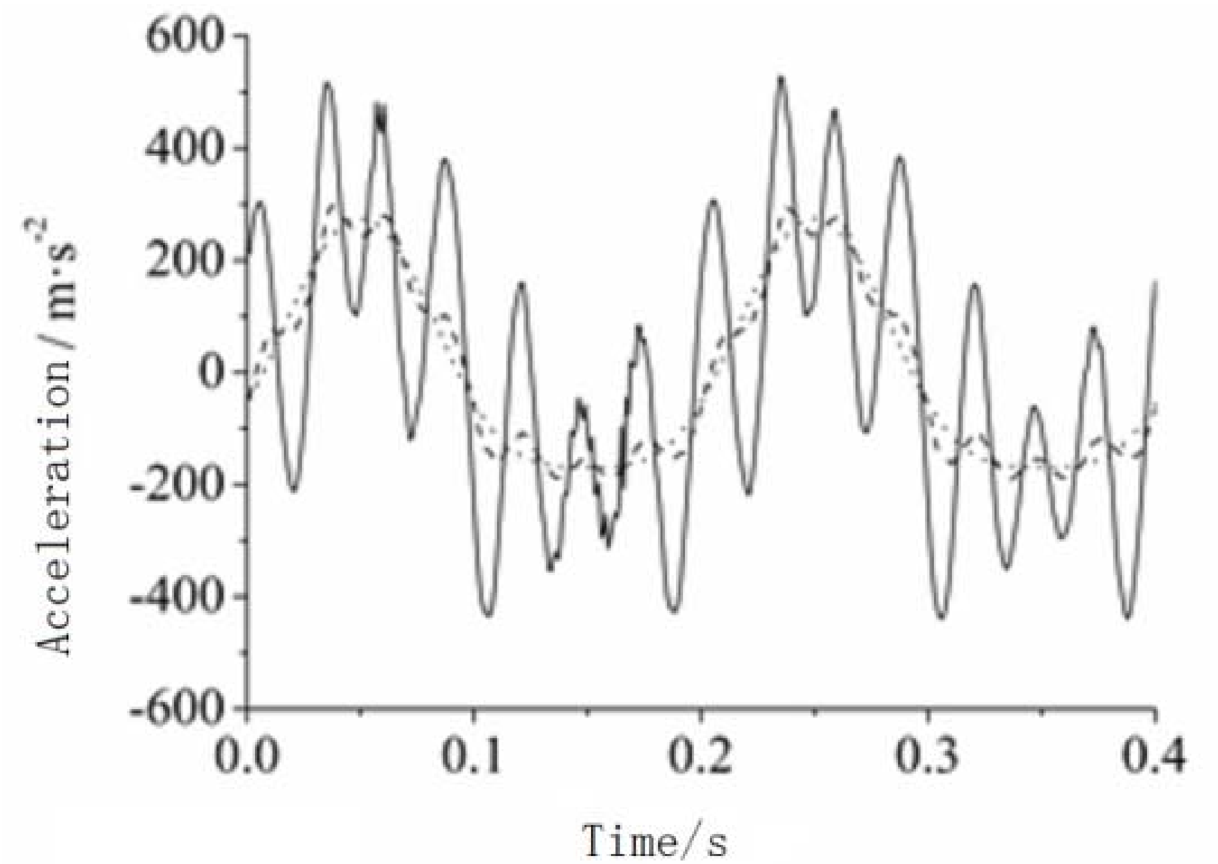
It can be seen from this picture that the crank pin torsional vibration of the new crankshaft decreased greatly. The maximum torsional displacement of the first row crankpin of the old crankshaft is about -3.30deg



—— old crankshaft ---- new crankshaft

Analysis of result

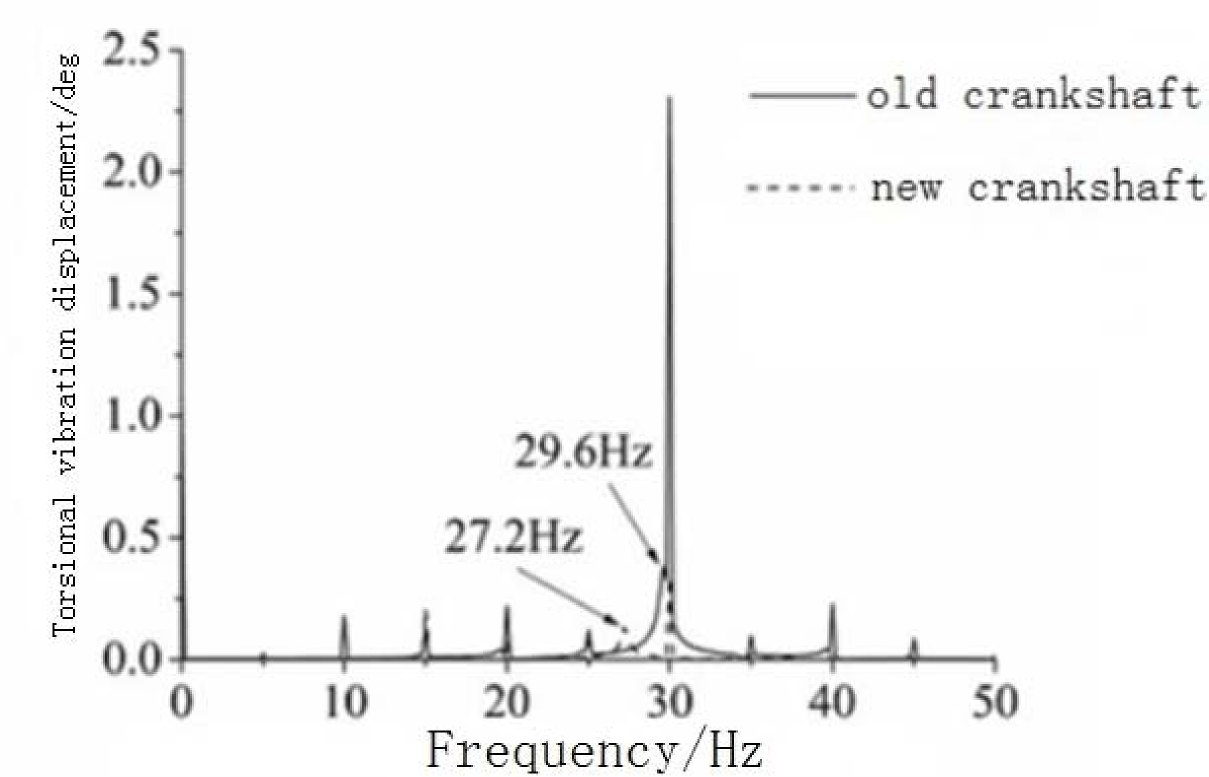
It can be seen that the new crankshaft fluctuation value has greatly reduced, which is almost similar to the rigid body dynamics



—— old crankshaft ------ new crankshaft

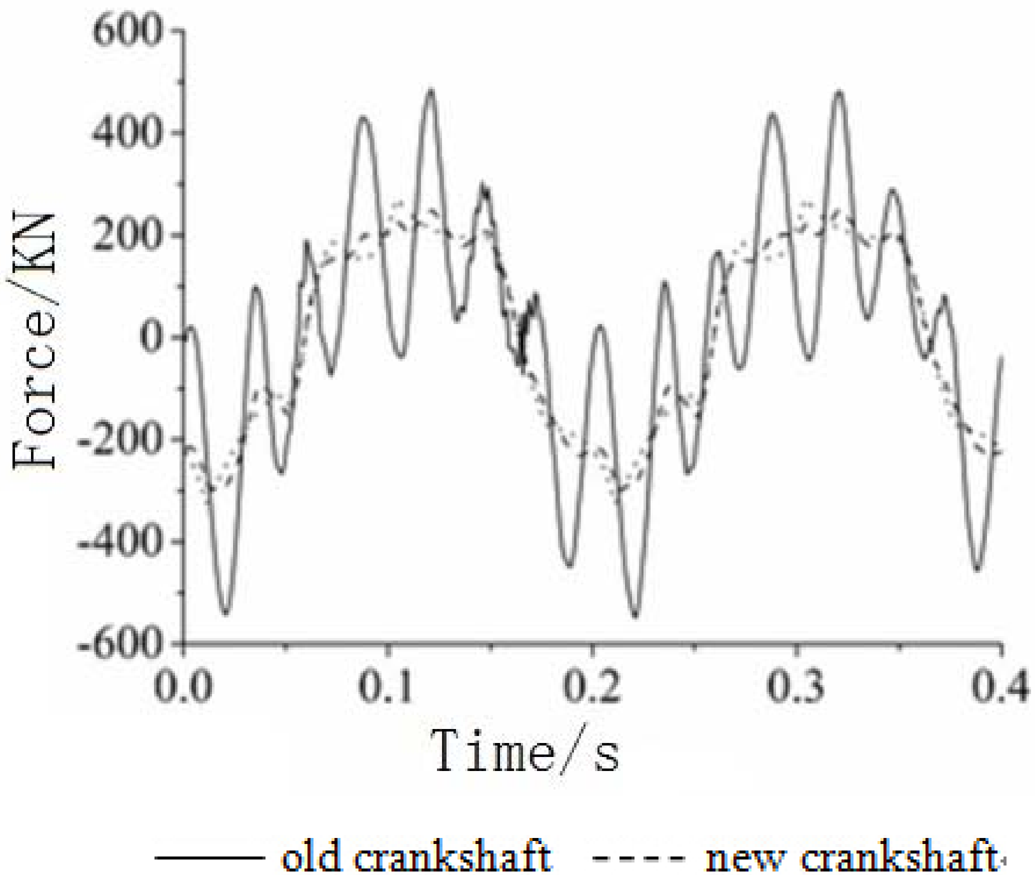
Analysis of result

the crankshaft torsional vibration has a peak in 5 Hz and its integral multiples. These peaks are the comprehensive effect of the each force results.



Failure Analysis of Crankpin Bearing

the piston force of the old crankshaft has a large torsional vibration amplitude. In the compressor, reciprocating mass of the low stage at the first and the second rows are very large, and the piston pin force of these stage could exceed the design value.



Failure Analysis of Crankpin Bearing

	First column	*Second column*	*Third column*	*Fourth column*	*Fifth column*	*Sixth column*
Design value(MPa)	8.3	7.8	6.6	7.9	9.2	10.6
Old crankshaft(MPa)	15.2	14.5	7.9	8.4	10.4	11.1
New crankshaft (MPa)	9.0	8.7	6.8	8.0	9.3	10.7

The maximum crank pin bearing pressure ratio of each row of 6M50 compressors

Failure Analysis of Crankpin Bearing

	First column	*Second column*	*Third column*	*Fourth column*	*Fifth column*	*Sixth column*
Design value	1.7	1.6	1.6	1.6	1.7	1.9
Old crankshaft	2.9	2.8	2.0	1.8	1.9	2.0
New crankshaft	1.8	1.8	1.7	1.7	1.7	1.9

The crank pin bearing impact coefficient of each row of 6M50 compressor

Conclusion

- *This paper proposed a new calculation method based on the flexible multi-body dynamics theory.*
- *According to this method, the torsional vibration characteristic of a 6 row crankshaft were calculated.*
- *The results show that torsional natural frequency of the crankshaft before transformation is close to the excitation frequency, so the crankpins of the first and the second row often cracks because of the torsional vibration.*
- *After transformation, the new crankshaft natural frequency is away from the excitation frequency, and the vibration amplitude of the crankshaft decreases substantially.*

This work is supported by the National Natural Science Foundation of China [NO.51106121].

Thank you !