An attempt at complete assembly contact analysis of a high precision reduction gear

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Abstract: High precision reduction gears are composed of many cranks and gears which are in high stress contact through bearings. In order to precisely evaluate stresses in critical components, an attempt was made to analyze a complete model of a high precision reduction gear taking into account the contact conditions between the parts. Using the latest contact analysis function of ABAQUS 6.6, a 650,000 element model with 190 contact surfaces was successfully analyzed in under 24 hours. The results revealed the exact stress distribution in each part based on the local deformation of the components. This achievement could be a breakthrough for rational design and satisfy the growing demand for higher durability under higher load within limited space.

Keywords: many contact parts, spring element, contact element,

1. Introduction

A high precision reduction gear is a complex mechanism that achieves a large reduction ratio by means of precession motion driven by plural gears accompanied by rolling and sliding on many pins inside a hub. In recent years, the optimum design for strength and durability has been accompanied by demands for higher loads and further space savings. To achieve this optimum design requires stress analysis that accounts for the deformation of all parts in the assembly. The problems with this kind of analysis are the large number of elements required for a complete model and the treatment of contact between the many bearings, pins and gear teeth. Two kinds of approaches were employed: a simplified model using spring elements at contact surfaces, and a more rigorous full scale contact model.

2. High precision reduction gear principle

The mechanism of a high precision reduction gear can be understood by referring to the cross-section of a typical reduction gear shown in Figure 1.

- Power is fed from the input shaft to the crankshafts via a reduction mechanism such as planet decelerators. (From “input gear” to “spur gear” in Figure 1.)
- These crankshafts, driven by the spur gears, cause an eccentric motion of two epicyclic gears that are offset 180 degrees from each other to provide a balanced load.
- The eccentric motion of the gears causes engagement of the cycloidal shaped gear teeth with cylindrically shaped pins located around the inside edge of the case.
- In the course of one revolution of the crankshafts the teeth of the epicyclic gears move the distance of one pin in the opposite direction of the rotating crankshafts. (Figure 2.) The motion of the gear is such that the teeth always remain in close contact with the pins and many teeth share the load simultaneously.
- The output can be either the shaft or the case. If the case is fixed, the shaft is the output. If the shaft is fixed, the case is the output.

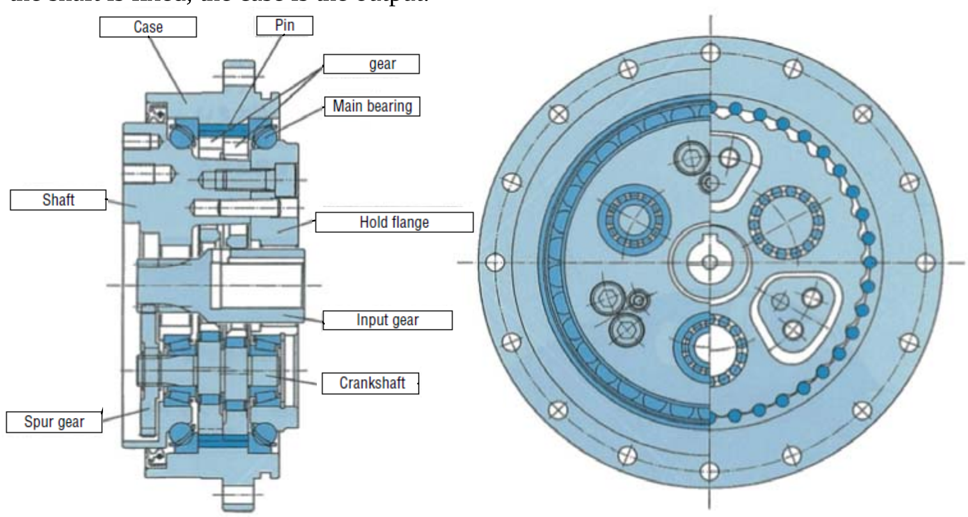


Figure 1. Typical structure of a high precision reduction gear.

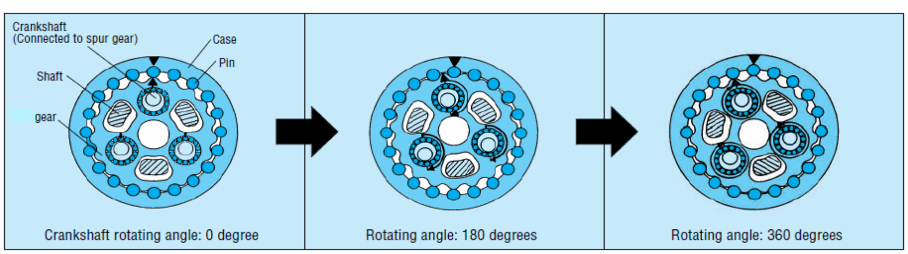


Figure 2. The precessional motion of the epicyclic gears.

3. Analysis

3.1 Models

3.1.1 The spring model.

This model is composed of six kinds of parts and is shown in Figure 3(a).

- Spring elements were used in the bearings parts and in the pins between the gears and hub.
- All six degrees of freedom were restrained in the flange part and rotation was restrained in the cover side face of the Crankshaft. Output torque was set in the output side of crankshaft (Setting the output side torque confirms the existence of input side torque)
- Spring rigidity was calculated from analysis results on an individual part (Figure 4.) With the exception of the springs between the gears and the hub, each spring was set to be effective only in the direction radiating from the central axis of rotation of the part.
- Both hexahedron and triangular pyramid elements were used. (Figure 5.)

3.1.2 The contact model.

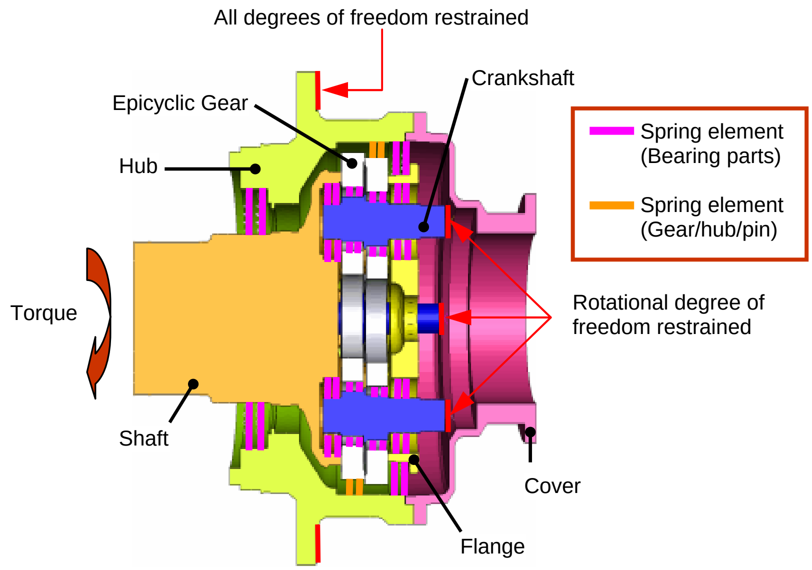
Here, contact elements replaced spring elements in the main bearing parts. (Figure 3(b).)

- Bearing needles were used instead of spring elements between the crankshafts and the gears. In total, 130 contact elements were used between the crankshafts and the needles and between the needles and the gears.
- Pins were used between the epicyclic gears and the hub. Furthermore, the hub mesh was modified to include the actual profile of the teeth. In total, 60 contact elements were used between the gears and the pins and between the pins and the hub. Spring elements were used in all other bearing parts in the same way as in the spring model.
- Both hexahedron and triangular pyramid elements were used. (Figure 5.)

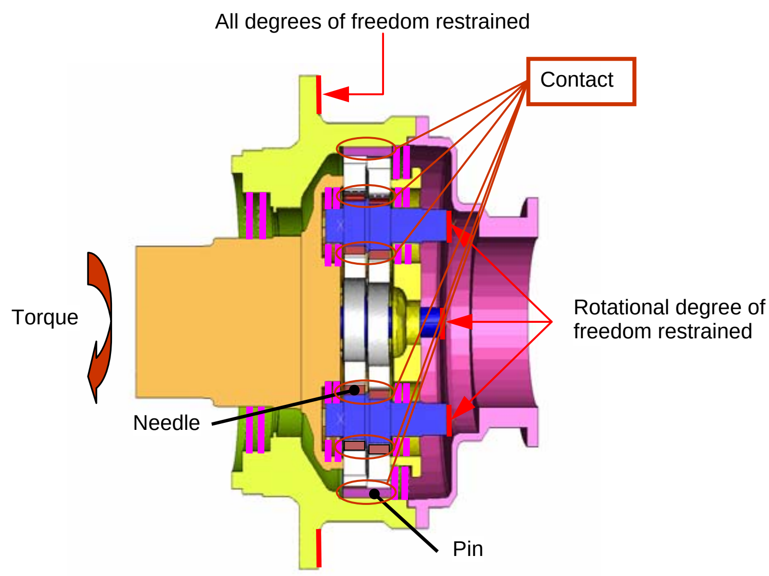
Table 1 shows the approximate number of elements and nodes used. The contact model used almost three times more elements and twice as many nodes as the spring model.

3.2 Software

ABAQUS version 6.6 was used as solver and ABAQUS/CAE version 6.6 was used in both the pre and the post-processing.



(a) Spring model. (Spring elements were used for all contact surfaces.)



(b) Contact model. (Contact elements were used in parts of the spring model.)

Figure 3. Simulation models. (Boundary conditions.)

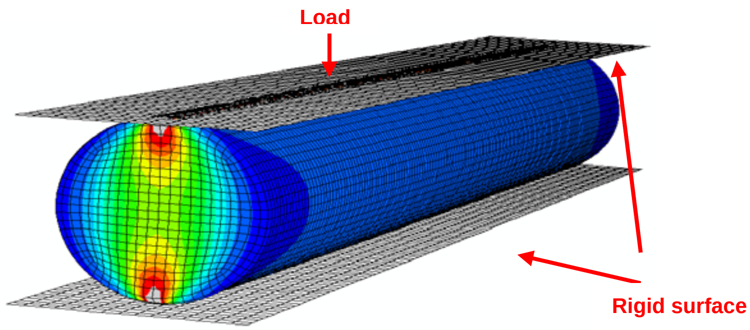


Figure 4. Typical result of the FEA to determine the rigidity (elasticity) of a pin.

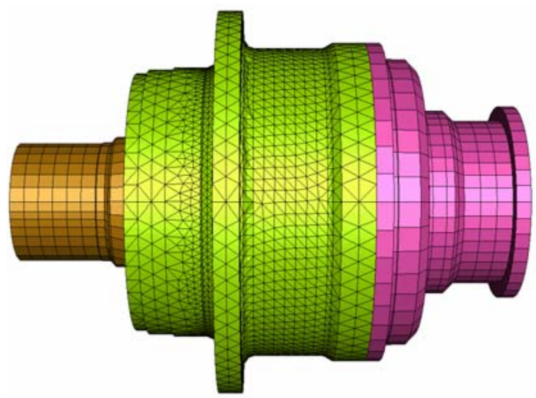


Figure 5. Configuration of element division (Hexahedron and triangular pyramid).

Table 1. Element and node counts for the two models.

	Spring model	Contact model
Number of elements	240,000	650,000
Number of nodes	610,000	1,250,000

4. Results

4.1 Comparison between theoretical calculation and spring model result

Pin load values from a theoretical calculation of an individual pin were compared with those from the FEM spring model analysis. The FEM analysis, which took into account the elasticity of all parts, gave a greater pin load variation around the hub than did the theoretical calculation, which was based on a rigid model. However, the total reaction force (given in Table 2.) shows that, although the distribution of pin load differs markedly between analysis and calculation, the difference in the total pin load is only about 5%.

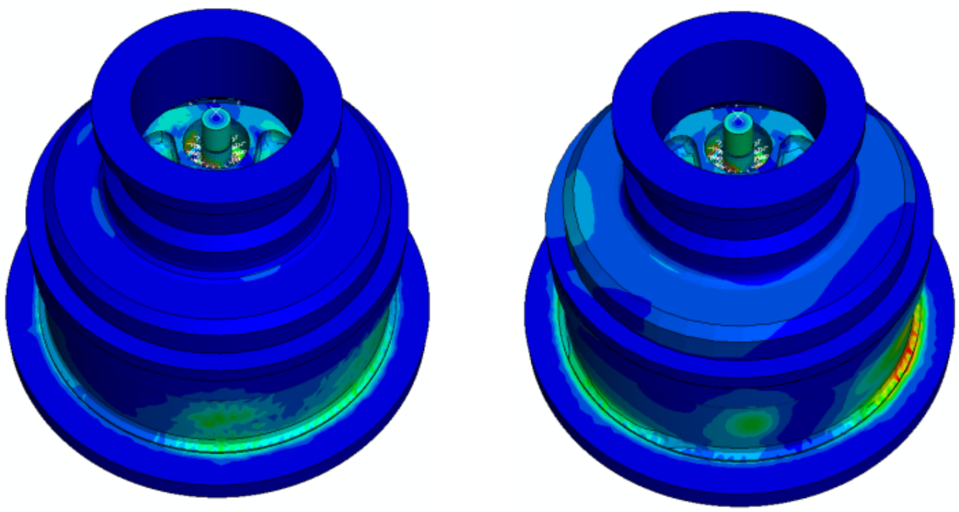
Table 2. Comparison of calculation and simulation results of pin load.

	Theoretical calculation of the rigid model	Simulation (Spring model)
Total pin load ratio	1	0.95

4.2 Comparison between the spring model and the contact model

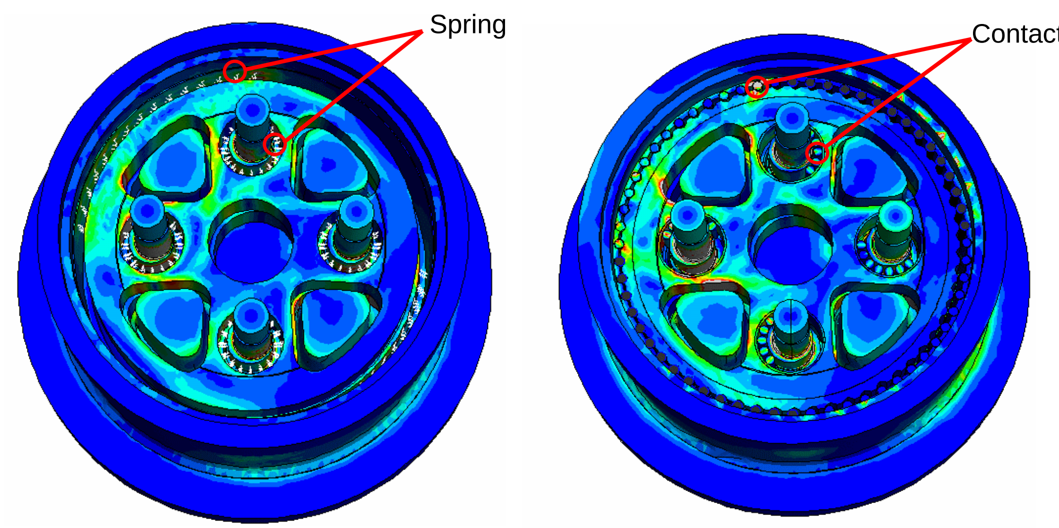
- **Overall appearance**

  The calculation results of stress analysis on the reduction gear assembly are shown in Figure 6, and clearly reveal the load transmitted to the external face of the outer cover. In addition, the results with the cover and the flange removed are shown in Figure 7. In the spring model, the springs are displayed as white in the red circles. In the contact model, springs are replaced by contact elements at the needles and pins and these are shown on the right of Figure 7. Stress can be observed as a result of load transmitted from the crankshafts to the needles and the gears. In the spring model, it occurs through springs. In the contact model, it occurs through contact between each part. These results are as expected and show that the calculations finished successfully. Figure 8 shows an enlarged view around the gear/pin/hub mating part. As a result of the variation from pin to pin mentioned in 4.1, the load transmission at different mating points has a wide distribution.



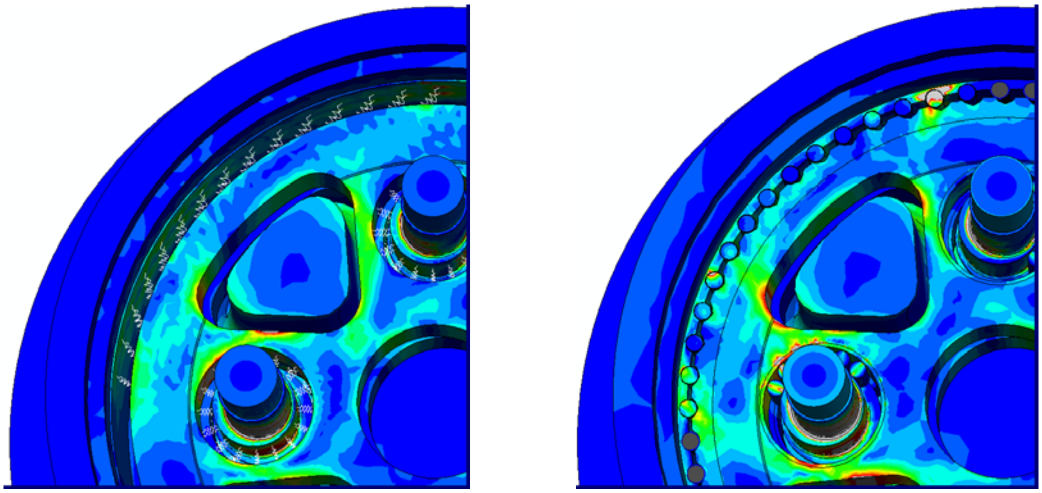
Spring model Contact model

Figure 6. Stress contour plot for each model (Outer cover).



Spring model Contact model

Figure 7. Stress contour plot for each model (with cover and flange removed).



Spring model **Contact model**

Figure 8. Stress contour plot for each model.
(Enlarged around a gear/pin/hub mating area)

- **Contact between gear, pin, and hub**

In the area where there is contact between the gears, pins and the hub, a large local indentation occurs as a result of the reaction force of the parts contacting the pin. The analysis results show the amount of indentation from the contact model is greater than that from the spring model. Consideration of hub deformation reveals that, for the contact model, the stress was concentrated around the inner part of the hub coinciding with the region of local deformation of the gear mentioned above. Analysis results from the contact model also reveal that the contact between the gear and the pin is biased towards the edge of the pin.

- **Contact between gear and needle**

In the region where a crankshaft makes contact with a gear the spring model results show high stress at the connection points of the springs. This follows from the fact that this model uses only two springs to simulate the contact through each needle. In contrast, the contact model results show that contact stress varies widely and continuously, but the peak stress occurs at the edge. This follows from the crankshaft expressing the effect of bending as a result of the reaction force. In the needle itself, the highest stress occurs at the edge of the contact part.

- **Output torque comparison**

As mentioned in 3.1.1, this analysis calculated input torque. The output torque to input torque ratio was then evaluated and used to convert actual output torque from the driving motor to output torque of the reduction gear assembly. This enabled a comparison between output torque derived from a theoretical calculation, and that from both the spring and contact models. The results of this comparison are shown in Table 3 as the ratio of output torque to theoretical result. Compared to the theoretical value, output torque is 10% lower in the spring model and 10% higher in the contact model. (The measured output torque, including friction, is about 10% less than the theoretical value)

Table 3. Comparison of the torque derived from FEA.

	Spring model	Contact model
Output torque ratio	0.90	1.10

- **Calculation time comparison**

The analysis calculation times are shown in Table 4. The contact model took almost six times longer to complete than spring model.

Table 4. Calculation time comparison.

	Spring model	Contact model
Calculation time (hours)	3.9	21.8

Machine Specification: Double Itanium2 1.6 GHz CPUs with 16 GB Memory

5. Discussion

- **Pin load**

The difference between the pin load value obtained from the spring model analysis and that from the theoretical calculation is only about 5%. Despite load transmission occurring through many complex parts a good degree of accuracy is achieved. Furthermore, although the load distribution obtained from analysis does not closely fit that from the theoretical calculation, this is to be expected as the deformation of the hub that supports the pin is not considered in the calculation but is accounted for in the analysis models. In addition, the results of gear and crankshaft deformation reflect the direction of the individual reaction forces in the hub, the pins and the gear

- **Output torque**

As the measured output torque including friction is about 90% of the theoretical value, it follows that the influence of the deformation of parts on torque appears extremely small. This result shows that the torque can easily be evaluated within an error margin of about 10%. The reason for the output torque value from the spring model being less than the theoretical value is the spring element's inability to completely match the actual direction of the load. In the contact model, the course nature of the mesh around the needles and pins might be responsible for an output torque value greater than the theoretical value.

As mentioned above, the deformations and reaction forces between contacting parts can be calculated to a relatively high degree of accuracy with both the spring and contact models. This follows from the fact that the deformation of the crankshafts and the hub that cause the reaction forces can be calculated even by the spring model. Therefore, the spring model can be used for the calculation of stress and deformation in parts such as the pins, crankshafts and gears.

However, the contact condition in the needles and the pins cannot be evaluated in the spring model. The contact model overcomes this limitation, as using this model calculates the precise edge contact condition caused by the bending of a crankshaft and hence enables evaluation of the maximum local pressure on each of the contact parts. This now provides a means for determining the rolling motion fatigue strength in the sliding parts such as the needles and pins. This important value cannot be evaluated by the rigid body model used in standard design calculations. The mesh around the pins and needles is still a little coarse in the contact model, and so to calculate an even more accurate load distribution and maximum local pressure the mesh in the contact areas should be further refined. However, an increase in the number of elements is difficult without excessively increasing analysis time and cost and the problem of finding an optimum distribution of the element density between contact and non contact parts needs to be addressed.

6. Conclusions

i. As a result of this complete assembly analysis of a precision reduction gear, the output torque can be evaluated within an error margin of about 10% and the reaction force in the sliding surfaces through precession movement can be determined to within 5%.

ii. Evaluation of the deformation of complete parts such as the gears, the crankshafts, the sliding pins and the needles is now a possibility. These results confirm the considerable effect that the operating loads and stresses have on such parts.

iii. The evaluation of reaction force, deformation and stress is possible even with the spring model. This means that the fatigue strength of components such as the crankshafts can be evaluated with the more simple spring model.

iv. In the contact model, the local contact conditions that cause deformation of parts such as the pins, the gears and the needles can be calculated. Consequently, the evaluation of rolling motion fatigue in the sliding parts has become possible.

Through this analysis, the multipoint contact analysis (190 places) converged comparatively easily. This result shows that ABAQUS has a high functionality in solving contact problems.

7. Acknowledgements

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