Threesomes Add Possibilities

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Abstract: A common problem in analysis jobs is to learn the loads passing through a point in a model. Such a point, for example, may be the centre of a pin joint or bearing. A long established method to obtain this information is to add a dummy third node into a defined equation at that point. This technique may be expanded to enable introduction of displacements at the point - for example, it represents a simple and efficient means to use one model but with different STEP definitions, to look at the effects of tolerance stack up / misalignment of parts.

This paper revisits the use of such additional third points, and compares their use against a near equivalent capability via the use of connector elements. The basic theory behind the "third node" is explained, and examples are presented from real world applications.

Keywords: Methodology. Force Reaction Multipoint Constraint Equation. Tolerance

1. Introduction

Two different analysis problems are touched upon in this paper, both of which find a ready solution by the use of “third nodes”.

The first problem is that of extracting forces passing through some section of a model, such as a bolted joint or a pin joint in a mechanism. The second problem is how to efficiently model the effects of tolerance on manufactured parts.

The use of an artificial third node, introduced at the joint (whether real joint or not) is shown to provide an easy to understand mechanism for tackling both problem types.

It is emphasised that the techniques presented here are not new – they are explicitly referenced in the Abaqus documentation, Section 34.2.1. However, most analysts tend to resort to the documentation only when things go wrong, and not when planning work. Hence the objective of this paper is to offer some useful hints when planning what may otherwise be somewhat intractable or - worse - tedious problems. It is also noted that all examples presented here could - in theory - be equally well addressed by the use of connector elements: some pros and cons of the two approaches are discussed.

2. Internal Forces

2.1 Example Problem

Consider the statically indeterminate system formed by a loaded shaft (such as an engine camshaft) passing through three bearings, such as illustrated in Figure 1. We need to know the load carried by each bearing – probably throughout a time history of applied loads at the drive point and at cam positions. The bearings will be part of a more complex model of a complete engine cylinder head.

Figure 1. Camshaft support in cylinder head.

For several years now Abaqus/Viewer has provided a tool for integrating the force through a section. If we are lucky, this tool may be able to extract the data we need. However, it is usually not able to do so, due to the impracticality in most cases of finding a suitable section through which all the bearing support load must pass, e.g. as Figure 2. In addition, the method is tedious to use and inaccurate, as the stress integrals are not 100% valid for all elements at all orientations.

An alternate approach might be to use a connector element placed between the housing and the shaft nodes (A and B) in Figure 3. This would enable direct output of the transmitted load via the "CTF" history variable. Previous papers by this author have attempted to illustrate the power of the ABAQUS connector elements for solution of mechanism and other problems. This powerful facility is in regular use within the author's company for loading generation and for building understanding of loading mechanisms. They provide good input to free body diagrams, which should be a regular tool for analysts understanding and checking the behaviour of their models.

However, connector elements prove temperamental in many applications. Selection of valid connector element stiffness is an art rather than a science (e.g. "RIGID" is not rigid!), and unstable solutions are a common occurrence.

A simple method to extract the force history is to introduce an artificial third node into the joint between the housing and shaft nodes, as illustrated in Figure 3. This introduction is via use of a three-part constraint equation in the relevant degrees of freedom. The third node is then fully restrained (or at least, restrained in the degrees of freedom relevant to the joint). Forces passing through the joint are then readily extracted as the reactions at the third node restraint.

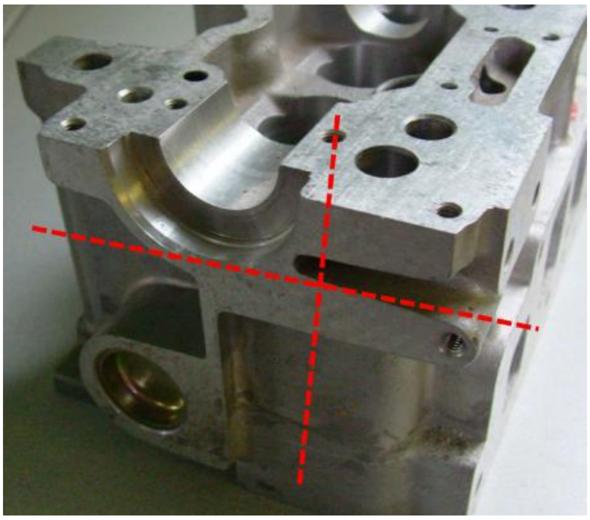
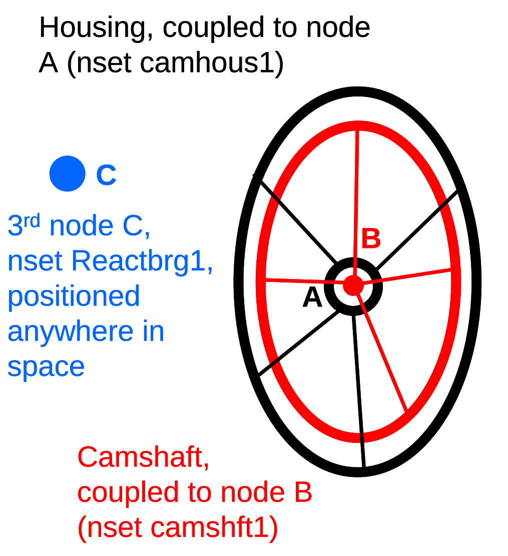


Figure 2. No suitable section for cam bearing load determination.




```
** Constraint: Cambrg1_X
*Equation
3
camshft1, 1, -1.
camhous1, 1, 1.
Reactbrg1, 1, 1.
** Constraint: Cambrg1_Y
*Equation
3
camshft1, 2, -1.
camhous1, 2, 1.
Reactbrg1, 2, 1.
** Constraint: Cambrg1_Z
*Equation
3
camshft1, 3, -1.
camhous1, 3, 1.
Reactbrg1, 3, 1.
```

Figure 3. Specification of 3rd node in joint.

2.2 Local Directions

Unfortunately, most of the local components within a structure tend not to be conveniently aligned with any chosen set of global axes. For example, “Vee” engines will have the camshaft bearings on different banks at different angles.

Fortunately, it is a simple matter to assign suitable local directions to the restrained third node, and hence to extract the resultant forces in the directions we need. Even more simply, we can leave the directions at the third node as global, and make sure that we connect the DOF at the joint, in the local direction we require, to one of the global directions at the (restrained) third node.

2.3 Mechanism Models.

As mentioned above, the author’s company makes regular use of connector elements for modelling such systems as the main running gear of an engine. Guidelines for the stiffness to assign to the connector elements for stable model operation have been established.

The methodology has been extended to include local component flexibility. For example, Figure 4 presents a connector element model of a simple crank / connecting rod / piston assembly, plus that same model with a “real” model of the connecting rod and piston pin included.

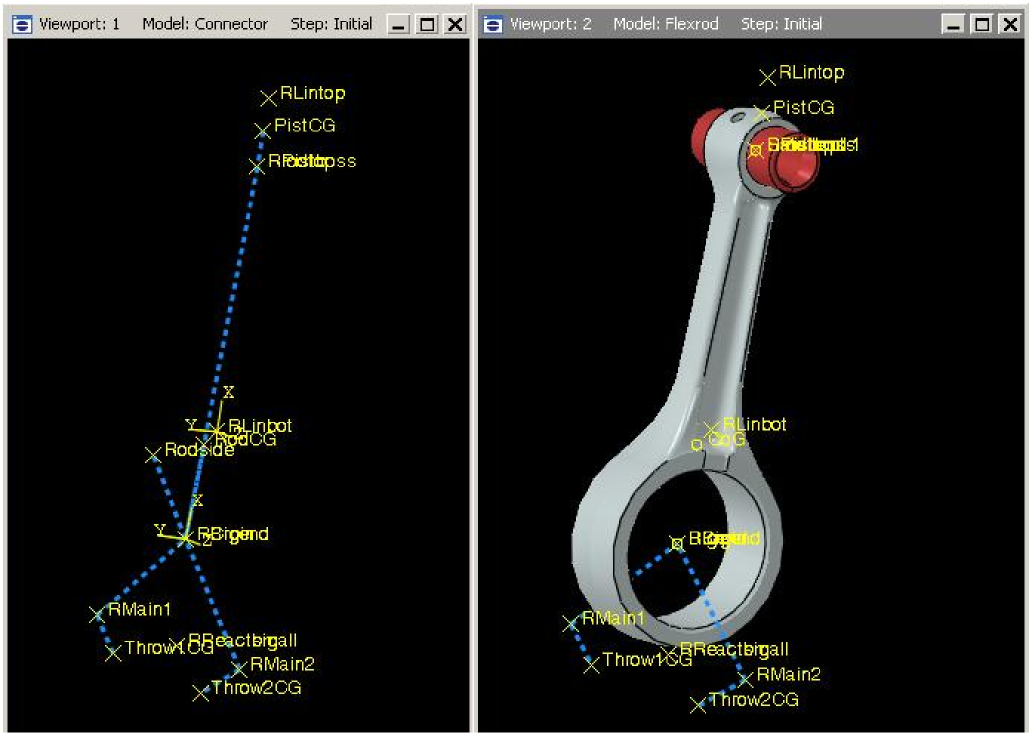


Figure 4. Connector only and mixed element models of crank system.

As a general statement, such mixed (solid elements plus connector element) models are extremely difficult to set up reliably. The inherent flexibility of the explicitly modelled component is generally mismatched with the stiffness of the connector elements, leading to unstable oscillations in the solution. Figure 5 presents the calculated force history at the small end joint, for the connector-only model, and for a version of the model with the connecting rod explicitly modelled. It may be seen that there is an unrealistic oscillation of the force history, rendering the results unusable. Many hours have been spent in attempts to establish guidelines for reliable solution of these mixed models, covering such aspects solution time step, damping, and connector stiffness assignments: no such guidelines have yet been established - indeed, very few successful individual calculations have resulted.

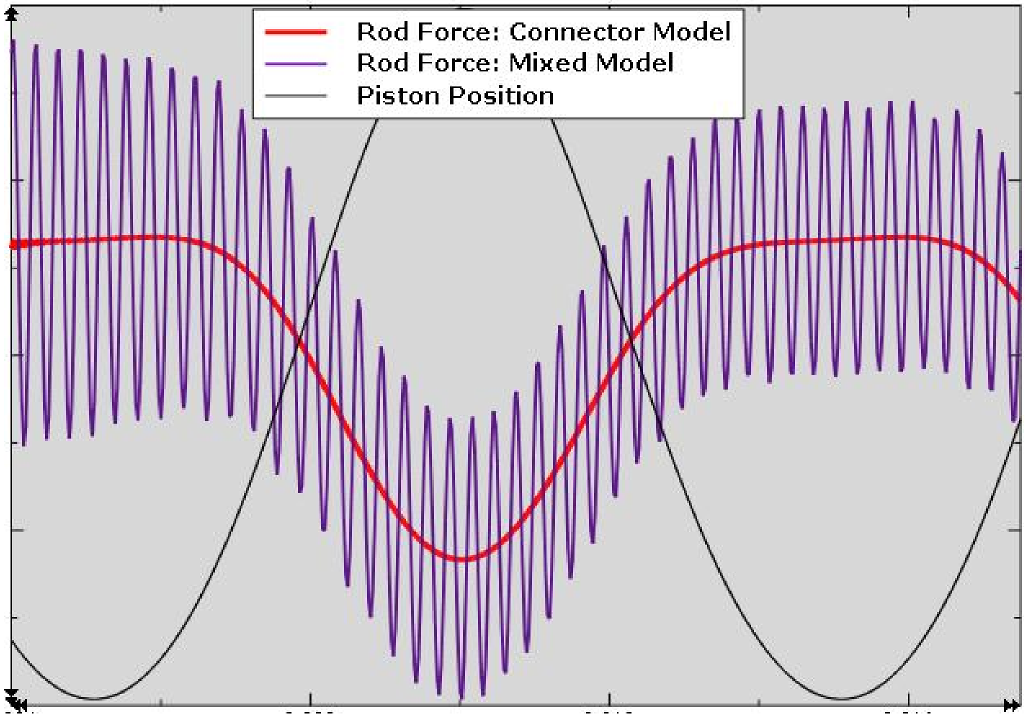


Figure 5. Results from connector only and mixed element models of crank system.

However, dynamic models which are completely built using explicit representation of the components do prove reliable – albeit at some expense in computer run time. The problem that arises is that force histories at the joints cannot be extracted using connector elements, as they introduce the instabilities outlined above.

The third node method completely overcomes this problem. Figure 6 presents such a model at one instant during solution, together with force history extracted via the reaction forces on a third node, introduced at the small end joint. It may be seen that the results are smooth and stable, and agree with those obtained by the connector-only model (and with classical hand calculations).

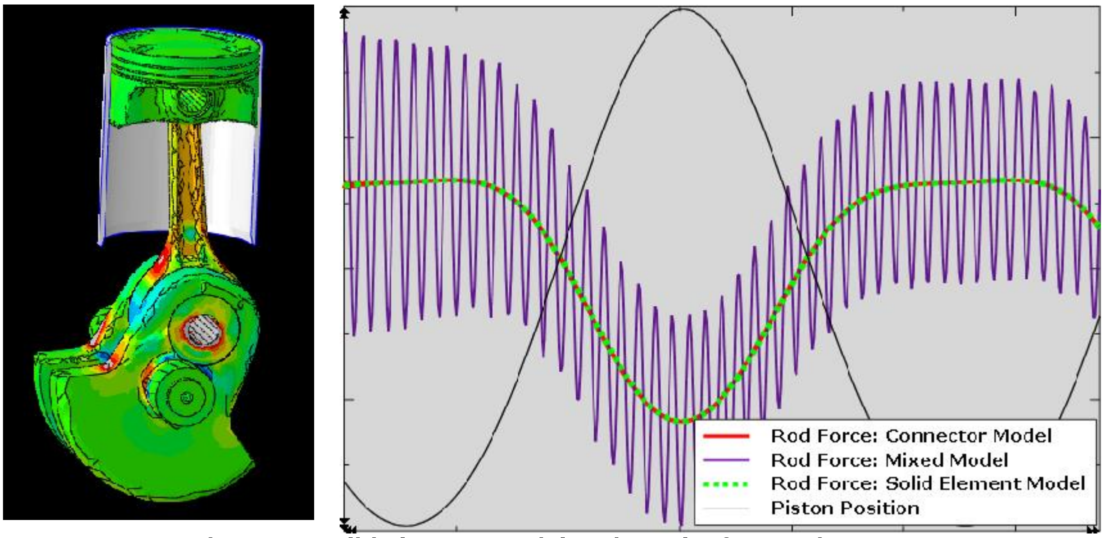


Figure 6. Solid element model and results for crank system.

3. Component Tolerance Effects

Figure 7 presents a much simplified view of all too common poor engineering design practice. Two separate shafts are supported via rolling element bearings in separate housings. The housings

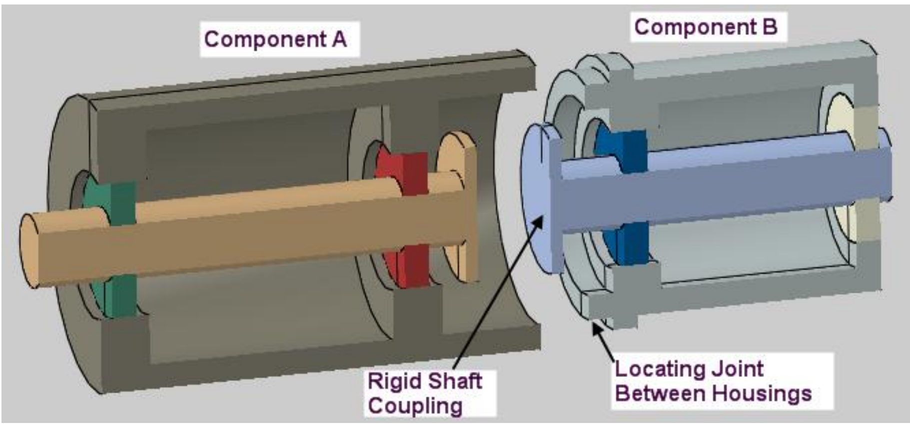


Figure 7. Example of over-stiff assembly design.

are mated together, and the shafts are connected using a nominally solid joint, such as a spline or bolted flange. Such a design leaves no room for manufacturing tolerance: any misalignment of the shafts, such as caused by non-concentric machining of the bearing housings, leads directly to stress in the shaft and housings during assembly.

The analyst is required to determine the effects of a range of tolerance stack up effects on the operating stresses in the assembly. Example effects of deviation from ideal dimensions are shown schematically in Figure 8.

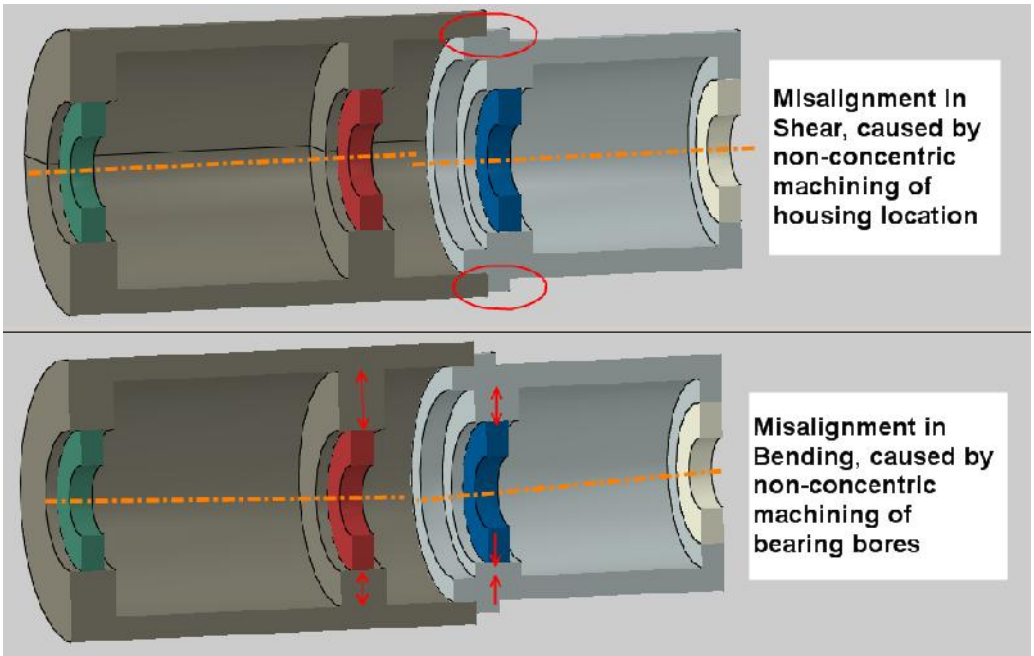


Figure 8. Example of tolerance effects on shaft loading.

The problem faced is how to set up these multiple analysis runs. Any finite element model will have zero stress throughout at the analysis start – and we do not know the stress distribution prior to analysis, so cannot apply it as a pre-determined field.

One approach might be to use an initial analysis step in which the housings are caused to move relative to each other. This is likely to prove difficult, however, as we are interested in the stress in the housing as well as in the shaft: any enforced sliding of the joint between left and right housing parts will inevitably invalidate the housing stresses. Also, misalignment caused by offset of the bearing housing machining of “x” mm will not necessarily cause eventual effective misalignment of the housings by “x”: there will be some deflection in the shaft, some in the housings, according to the relative stiffness of the parts. Solution times would inevitably be long, if (for example), a bolted joint is to be sheared in the first analysis step.

The solution is again the use of third nodes, one introduced at each of the four bearings joints. Rather than restraining these third nodes, they may be given enforced displacements in the assembly step. These displacements correspond to the limits of component tolerance. The effect of the constraint equation is then to force the nominally matching centres of the shafts and bearing housings to displace relative to each other. This does not in any way over constrain the system: for example, if the housing is relatively weak, then the shaft will remain near straight, and the housing will deflect. Conversely, if the shaft is relatively weak, then it will bend within a largely undeformed housing. The correct balance of deflections will result, according to the modelled component stiffnesses.

Subsequent analysis steps may impose operating loads and / or rotations on the shaft. It is a simple exercise to model alternate tolerance stack up effects, by simply changing the specified displacements of the third nodes. This may be done in the first step or in subsequent steps within one analysis. This represents a major saving in terms of time spent building a model at different tolerance settings.

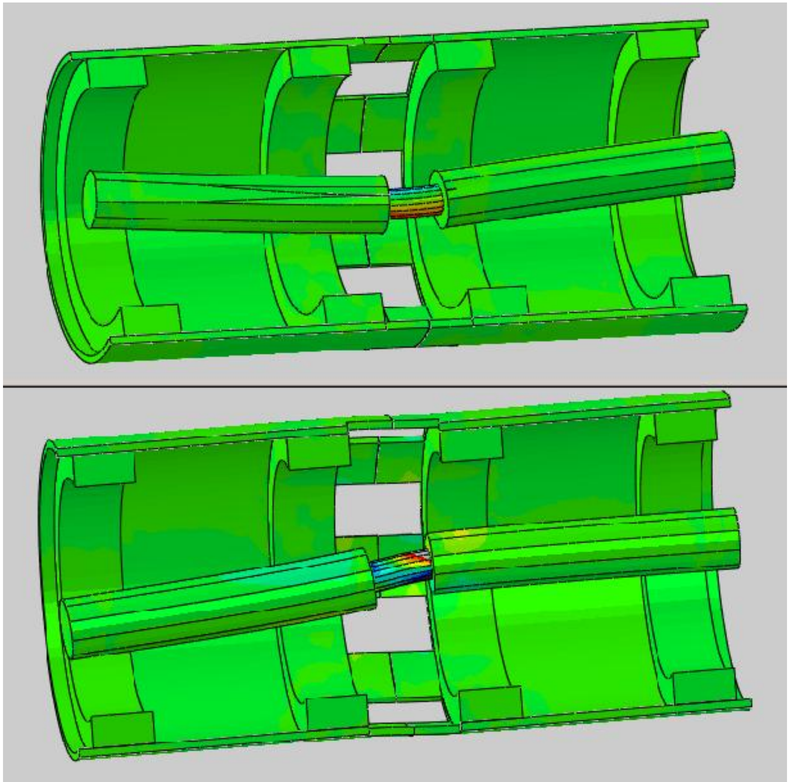


Figure 9. Bending (upper) and shearing tolerance stress results.

Figure 9 presents example results for a shaft subjected to the overall shearing and bending loads illustrated in Figure 8. Note the deformation of both shaft and housings.

4. Other Applications

Variable compression ratio engines offer major benefits over conventional engines in terms of fuel economy across the operating range. One mechanism to achieve variable geometry is to alter the length of the engine connecting rod: a shorter connecting rod means that the piston does not rise as high in the cylinder as with a long rod, hence the compression ratio is lowered.

Preliminary studies into the effects of a variable length connecting rod required that load histories at the bearing be rapidly established, before, after, and during a change in length event.

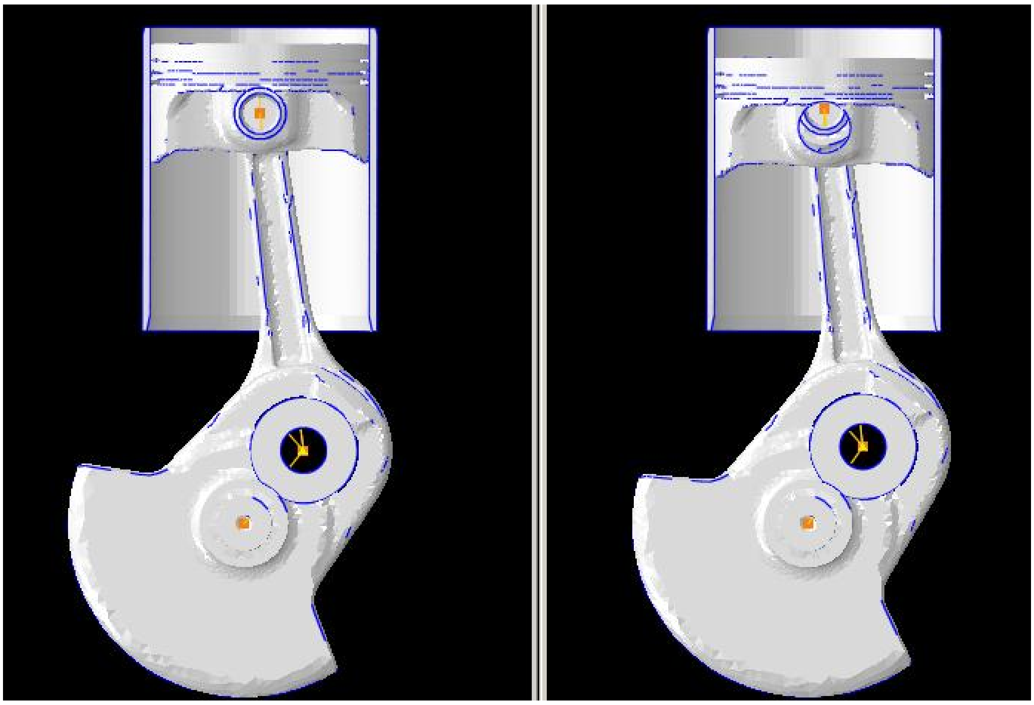


Figure 10. 3rd node offset shifting pin relative to piston.

Figure 10 presents views of the model shown earlier, adapted for these preliminary studies. Thios was a connector only model, but with a "3rd node" joint at the small end. The solid components are modelled as display bodies only. The length-changing mechanism itself was not modelled. Instead, an offset was introduced via the third node at the connecting rod to piston joint. The left view shows the model at 40^o after top dead centre ("ATDC") with the original rod length: the right hand view shows the same model from a later instant in the analysis, at 40^o ATDC with the minimum length connecting rod. Note the height of the piston relative to the cylinder liner in each case, and the mis-alignment of the piston pin in the piston boss caused by the artificial representation of the rod shortening mechanism.

The following listing presents an extract from the input deck for this analysis: it shows how displacement of the third node may be achieved at any point in the analysis sequence. In this particular analysis, the system was modelled in the sequence:

Step 1: 360^{o} rotation at full length connecting rod.
Step 2: 360^{o} rotation, whilst moving the 3^{rd} node to shorten the rod effective length
Step 3: 360^{o} rotation at minimum length connecting rod.

```
*Step, name=Rev1, nlgeom=YES, inc=1000
Max. Rod Length
*Dynamic
<timedeg>, <time1rev>,<mintime>,<timedeg>
**
** Name: Holdreactsmall Type: Displacement/Rotation
*Boundary
Reactpin, 1, 6
** Name: Main1 Type: Displacement/Rotation
*Boundary
Main1, 1, 1
Main1, 2, 2
  ...
rodtop, 5, 5
** Name: Spin Type: Velocity/Angular velocity
*Boundary, type=VELOCITY
Main1, 6, 6, -<angvel>
** OUTPUT REQUESTS
  ...
*End Step
** -------------------------------------------------
*Step, name=Rev2, nlgeom=YES, inc=1000
Shorten Rod
*Dynamic, INITIAL=NO, ALPHA=-0.3
<timedeg>, <time1rev>,<mintime>,<timedeg>
*Boundary, OP=NEW, AMPLITUDE=RAMP1
Reactpin, 1, 1, 6.0
*Boundary, OP=NEW
Reactpin, 2, 2
 ...
*End Step
** --------------------------------------------------
*Step, name=Rev3, nlgeom=YES, inc=1000
Min rod length
*Dynamic, INITIAL=NO
<timedeg>, <time1rev>,<mintime>,<timedeg>
** BOUNDARY CONDITIONS
**
** OUTPUT REQUESTS
*End Step
```

5. Summary

The introduction of third nodes at joints in analysis models is a long established, and perhaps long forgotten, technique. It can provide significant benefits over alternate analysis methods in many cases, and should not be forgotten.