Inclusion of Oil Film Pressure Distribution in Contact Analyses

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SUMMARY

This paper covers the research carried out on a technique to simulate the pressure distribution of a typical plain bearing oil film when using the finite element analysis technique. Using an inverse parabolic shaped model to simulate the oil layer, it is possible (via a combination of extra beam elements and sliding multi-point constraints) to produce a model, which simulates the oil layer and is able to rotate without significant loss of accuracy.

1 INTRODUCTION

When modelling a plain bearing (e.g. the big end of a con rod), in the past we either:

– Knew the force and modelled one side of the bearing using a specified pressure distribution, which is not the easiest method. A major problem with advanced analyses, e.g. ones using centrifugal or gravitational loadings, is that we are unable to pre-determine the magnitude of the total contact force. This option is thus unacceptable.

OR

– Model both sides of the bearing, and use a contact. This approach is better for jobs where we are not quite sure of the load position or magnitude. However, contacts tend to give unrealistically high stresses at the outer edges, exactly opposite to the true condition where the centre of the bearing will carry most load (as the oil is at higher pressure due to hydrostatic forces etc).

Historically, it's been shown that some form of parabolic distribution will model the oil film load best. Therefore, the aim of this research was to make a model with a contact, which will also distribute the applied load, typically varying from maximum load at the centre out to near zero load at the periphery. At present, ABAQUS does not feature any specific capability for the modelling of such oil films.

Existing alternate approaches considered:

- Use of DCOUP3D elements, with different weightings for each nodal connection from the bearing surface to the connection central node. Unfortunately these are difficult to set up, especially as each node on the bearing surface will 'see' a different area (consider a central one vs. an outside corner one).
- Use SOFT contacts. At the outset, this approach seemed promising, but tests showed we could not obtain good results. We still end up with high loads at the outer edges of the bearing.
- Model the initial geometry deformed slightly. This could work, but only if we know the load magnitude before we start, so that we can get the relative curvature of the faces correct. It would not work, for example, if it was set up for a load (F), then also run for load (F/1000), as for the latter, only the centre would touch and there would be zero load at the periphery.
- Model a thin layer of 'oil' elements in the bearing, with each element assigned different stiffness properties, getting weaker as we go out. This does work satisfactorily as we can see from Figure 1. (which shows the contact pressure and the stresses acting on the surface of the thin oil layer), but is not an easy method to set up.

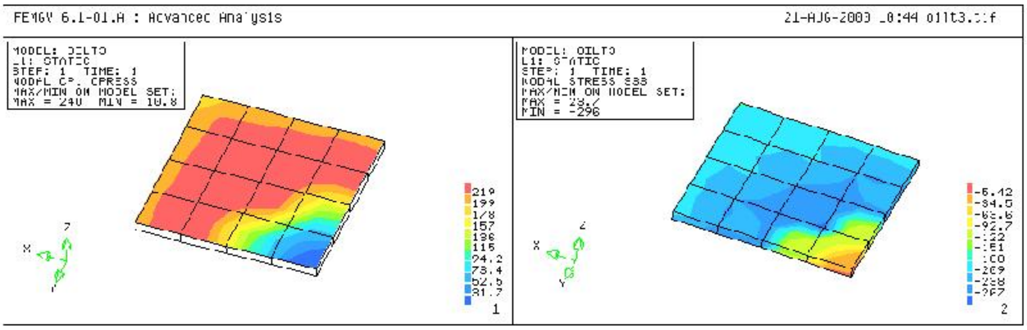

Figure 1. Pressure distributions using a thin layer 'oil' element

As none of the above methods are ideal for everyday use in solving the pressure distribution problem we need to readdress the problem. Overall, we want the force, hence stress, in the centre of the bearing to be greater than that at the edges. We would like to maintain real metallic geometry, and we would like something relatively easy to set up and valid for a range of applied loads.

2 PROPOSED METHOD

2.1 Initial Tests

We model the oil as a film of material of constant material properties. At the simplest, assume that the metallic parts either side of the bearing are infinitely stiff. Then, the relative motion of one face of the bearing to the other will be the same across the bearing face. Now,

	$E = \sigma/\varepsilon$	Where:	E = the Young's Modulus	L = Oil thickness
I.e.			σ = stress	ΔL = Change in Oil thickness
	$\varepsilon \propto \sigma$		ε = strain	

So, for the ± to get bigger in the centre, £ must also get bigger (given constant material properties). Remember that £ = ΔL / L. We've assumed above (stiff metallic parts) that ΔL will be constant across the bearing width. Therefore, we're left with the option of varying the thickness of the oil film, L, as modelled. To achieve this, consider a (flattened) 2D section through a bearing (Figure 2.):

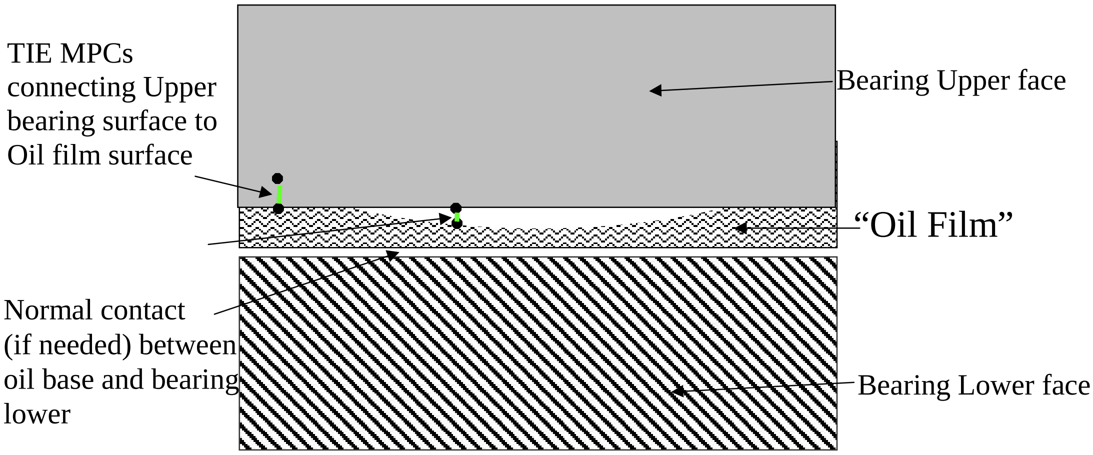

Figure 2. Principal of method

The main difficulty in applying this method involves the generation of the upper shape for the oil film. This is done at AAL using MS Excel to generate the inverse parabolic shape, and to then read in the points into the mesh generation software as points on which to generate a spline line. The Excel method as developed will create suitable curves for both planar (which rarely occur!) and cylindrical bearings. In the latter case, the oil film thickness is generated normal to the bearing local surface.

The advantage of this method is that, being graphical, one can see when the shape of the oil is 'about right'. Figure 3. Shows the shape of the elements used to model the oil film (this figure shows only a quarter of the oil film).

Once the model is built, everything else is straight forward – in most cases the models run with perturbation steps, provided that there are no major geometric non-linearities, such as when the bearing rotates. It is necessary to calculate an effective Young's modulus value for the oil film 'E' to make sure the highly loaded central elements won't go inside out, based on peak expected loads plus minimum element thickness.

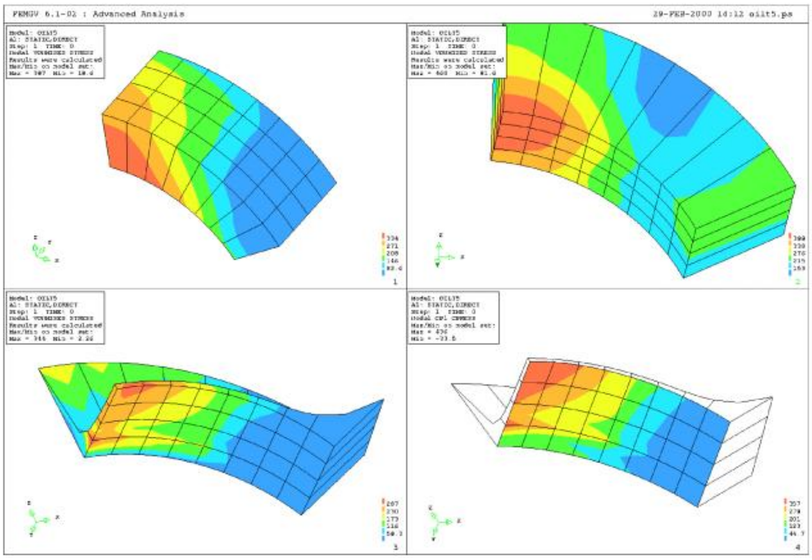

Figure3. Stress and contact pressure results from initial model

The above picture *(Figure 3.)* shows results for the initial method tests, which shows the method works well on a curved bearing. For this, the parabola was set up to give a ratio between minimum and maximum stresses (forces) of 20 – as you can't get the outer edge all the way down to zero, as that would need an infinite-thickness oil element.

Points to note:

- The ABAQUS result is NOT perfect: note the differences above for the loads as seen either side of the bearing and in the oil film. More refining with element sizes should make this better.

- This fundamental method will work for any (known or assumed) pressure distribution – parabolas are generally accepted as correct for plain bearings, but other shapes may occur.

- Trying to achieve the TIE connection using a simple contact definition, with an ADJUST, appears as though it *should* work, but it has problems of deformed elements. This may have something to do with the 21st node that ABAQUS puts into the centre of the face. Interestingly, from test work done ABAQUS seems to generate its stiffness matrix from the modelled dimensions NOT the adjusted ones. This IS what we want here, BUT it may be dangerous in a general case.

- The load is MUCH better than using a contact alone.

2.2 Final Method

Further work on this method started with the modelling of a full model of the bearing. The inner and outer bearing components were built up to 360°, whilst the oil film was built up to a typical subtended arc of 120°. The diagram below *(Figure 4.)* shows the full model.

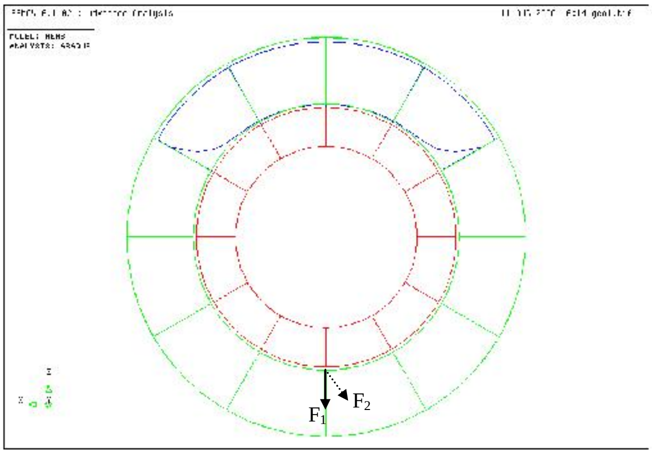

Figure 4. Front view diagram of the full model

The inner surface of the inner bearing was completely constrained, whilst the outer bearing was constrained along its centre line to stop it moving off in the y-axis. Since the constraints that were originally used for symmetry purposes had been removed, it was now necessary to use spring elements to prevent the outer bearing spinning around the inner during the beginning of the analysis.

Initially, a single point load was applied to the bottom of the inner surface of the outer bearing. The point load was applied straight down so as to maximise its effect on the oil film (load 'F_1' above). The results from this analysis *(Figure 5.)* are shown below.

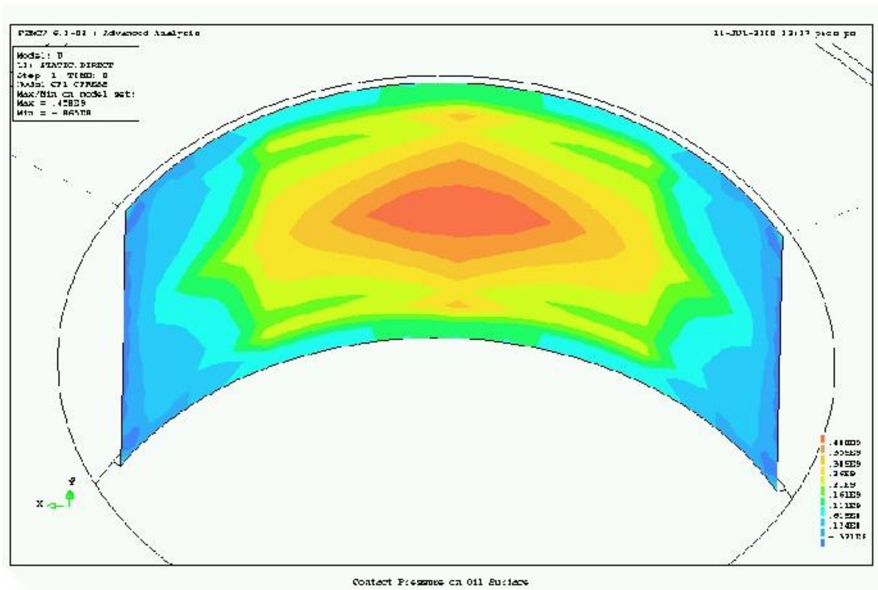

Figure 5. Contact pressure distribution for a static load downward

The aim now was to modify this method so that the oil layer could rotate with the outer bearing when an offset load is applied. The oil film needs to be able to move so that it is exactly opposite the load, and the shape of the oil needs to remain radial to the centre. By doing this the right distribution of forces are exerted on to the oil film hence making it work properly.

The main problem encountered was with the TIE MPC. The TIE works by applying the same changes in the degrees of freedom for two separate nodes. If one node moves in a TIE then the other node will move in the same global direction by the same amount. This constraint serves its purpose well in a small displacement situation, however when a movement such as a rotation is applied, the TIE constraint fails to work the way we need it to. Due to differences in radial dimensions between the oil film and the outer bearing, the circumferential displacements in global terms are not consistent when both parts are meant to rotate the same number of degrees.

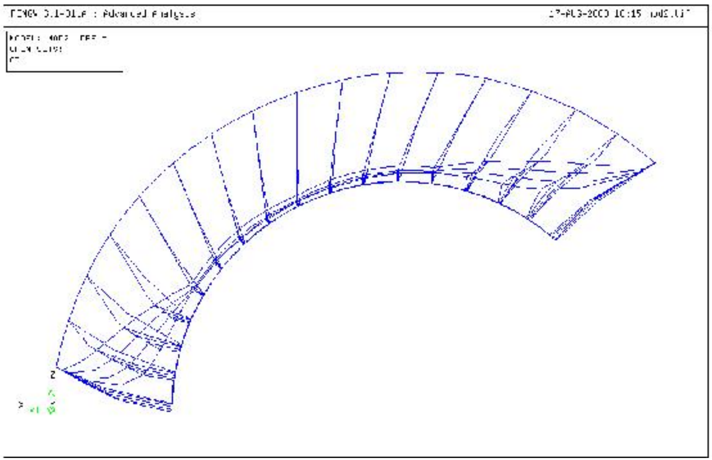

Figure 6. Distorted oil film due to TIE constraint

The effect of this shortcoming is that the oil film distorts out of shape as it rotates - e.g. see Figure 6. Hence, in order to solve the problem a new method of constraining the oil film had to be found. This new method would have to enable the oil film to rotate around the inner bearing, whilst keeping the oil film's structure radial to the centre. The constraining method also had to transmit the load applied from the outer bearing, through to the oil film.

Various methods were looked at including various combinations of LINK and SLIDER MPC's, 3D coupling elements, equation constraints and local cylindrical co-ordinate systems. The problem of using LINK and SLIDER MPC's and equation constraints at the same time is that they all take away the same degrees of freedom. Hence, a conflict occurs when trying to run the analysis. When trying to rotate the model using a local cylindrical co-ordinate system, large deformations occur due to the method ABAQUS uses to work out the displacements. ABAQUS does not assign 'true' cylindrical co-ordinates, but rather works with a local rectangular set, aligned as cylindrical ones would be at the zero-load condition.

The method that worked was the use of stiff beam elements and SLIDER MPCs between the outer bearing and the oil film. The beam elements do the same job as the LINK MPCs but without using up the degree's of freedom, which are now available to the SLIDER MPCs.

To begin with, a model that contained the SLIDER MPCs and the beam elements was run with the load pointing straight down as in the original run. The contact pressure distribution that resulted was the same as that using the TIE MPCs, as we wanted it to be.

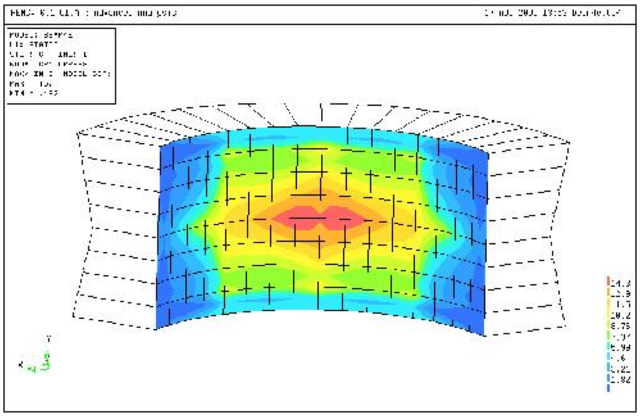

Figure 7. Contact distribution for initial load case

Another run of the model was carried out, this time with the load offset to induce a rotation. The results obtained were again, as we wanted. As you can see from the displacement plot below *(Figure 8.)*, the outer bearing and oil have rotated around with out any serious deformation or surface intersections. The contact plot *(Figure 9.)* shows that the pressure had been correctly transmitted from the outer bearing through to the oil.

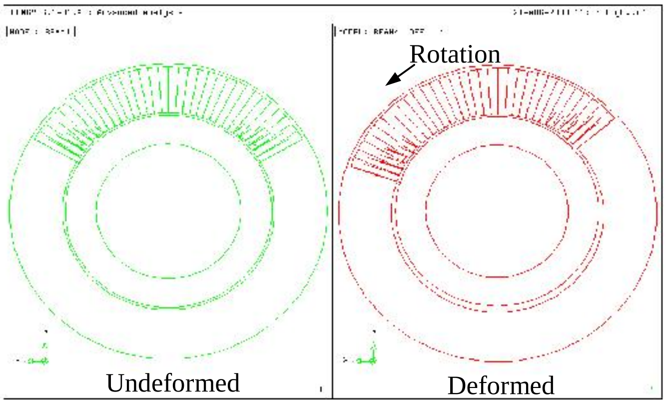

Figure 8. Rotational displacement experienced using the new method

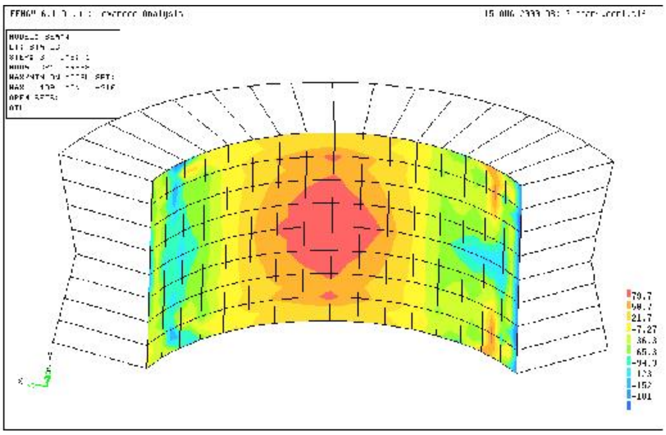

Figure 9. Contact pressure predicted using the new method

3 CONCLUSION

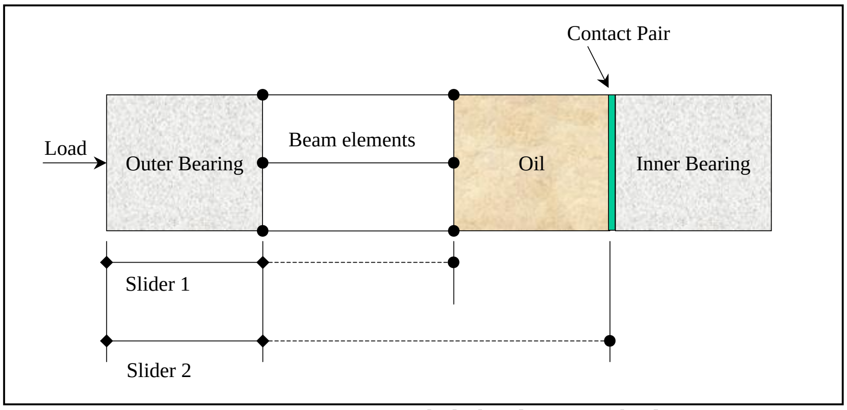

Figure 10. Principle behind new method

The results from this test indicate that the above method satisfies our objective of producing a 'smart' oil film. The method works because of the combination of SLIDERs and beam elements. The diagram above *(Figure 10.)* shows the principle of the method and in what way the beam elements and slider constraints are used. The two sets of SLIDER constraints ensure that the radial lines (created by the nodes on the top and bottom surfaces of the outer bearing) are maintained by the top and bottom surfaces of the oil film, as the outer bearing rotates. The beam elements then ensure that the effect of the load applied to the outer bearing is transmitted through to the outer surface of the oil in a radial manner. This method avoids both the problem of non-consistent circumferential displacements, and the limited availability of degrees of freedom when using MPCs such as the SLIDER and LINK constraints.

Further work is required to refine this method. Improvements could be made in the actual contact distribution generated once the oil film has moved from its original position. In addition, research could be made into finding the optimum stiffness properties for the oil material when used with various bearing materials such as steel or brass. Ideally, this method could be developed into, or superseded by, a new type of 'oil' element. For example, ABAQUS could automatically set up the required geometric shape and constraints to simulate the oil film. By doing this, very little effort or time would be required by the user, hence enabling us to produce models that are more accurate in less time.