XVI IMEKO World Congress
Measurement - Supports Science - Improves Technology - Protects Environment ... and Provides Employment - Now and in the Future
Vienna, AUSTRIA, 2000, September 25-28

THE ACCURACY OF PRESSURE MEASUREMENTS

K. Hoffmann, M. Egger and W. Wahler
Institute for Design Engineering and Transport, Handling, and Conveying Systems
Vienna University of Technology, A-1040 Vienna, Austria

Abstract: In a wide range of technical fields it is important to be able to measure contact stresses between elastic bodies. Many methods, including those employing electrical or pressure effects, use measurement sensors in the form of foil inserted between the contacting bodies. However, these foils can have a decisive influence on measurement results. This study investigates the influence of the inserted foil on the accuracy of measurement results. The most significant factors are the thickness and elastic properties of the measuring sheet, and the geometric properties and rigidity of the elastic bodies. Numerical results were obtained by means of non-linear FE calculations and compared with experimental results obtained by two measuring methods:

a) Tekscan systems [1];
b) Fuji prescale film analysed using GODAV [2], [3], [4], [5].

Keywords: contact stresses, pressure measurement, measurement accuracy

1 INTRODUCTION

Measurement of pressure distribution between bodies in contact is a major problem, particularly in mechanical and civil engineering. The most important factor to achieve accurate measurements of contact stresses is the use of a very thin sensor with a stiffness similar to that of the bodies. Many methods of measurement of surface pressure distribution between contacting bodies are derived from two well known loading cases:

Ball on elastic half-space (point contact);

Cylinder on elastic half-space (line contact).

Provided that isotropic, elastic behaviour of the contacting bodies can be assumed, there are analytical solutions for both cases which can be applied for comparison with the measurement results.

2 COMPARISON OF MEASURING METHODS

Two well-proven methods have been used. One is an electrical method, the TEKSCAN system, and the other is an imprint method using the FUJI prescale film. The former is suitable for time-dependent measurement, but the latter only for static loads.

The technology of the TEKSCAN system is based on a conductive gridwork of rows and columns deposited onto thin, flexible film. Each conductive trace exhibits a change in electrical resistance when pressure is applied to its surface. The array is scanned electronically to determine the pressure at each sensing cell. Pressure ranges from 0 up to 170 N/mm^2 can be specified. Using TEKSCAN software the contact stresses can be displayed on a PC screen in various formats, e.g. in 2D colour format or in 3D wire frame formats. The changing of pressure distribution can be displayed in real time up to about 100 Hz.

FUJI prescale film [2] is usually composed of two sheets, A-film and C-film, placed simultaneously between the two bodies in contact. There are types for use with different pressure ranges: Ultra Super Low, Super Low, Low, Medium and High (total pressure range 0.5 – 130 N/mm^2). After applying a load, a red colour is obtained with density depending on the maximum amount of local pressure for the duration of the test.

There are several methods in use with varying degrees of accuracy for evaluating the colour density distribution and determining the pressure level, but only computer-aided evaluation of colour density measured with FUJI prescale film enables quick and accurate interpretation of pressure distribution. The GODAV system developed at Vienna University of Technology [3], [4], [5] is used for these investigations. This system satisfies certain requirements, e.g. low costs, application of a commercially-available PC, multicolour presentation of results, simultaneous digitalisation of measured pressure distribution and calibration spots.

3 BALL IN CONTACT WITH AN ELASTIC HALF-SPACE

3.1 Theoretical solution

The first theoretical problem we investigated is the Hertz problem of a ball contacting an elastic half-space. Initially they are in contact in a single point. Under increasing load, both bodies deform and the projection of the contacting area becomes a circle with radius a.

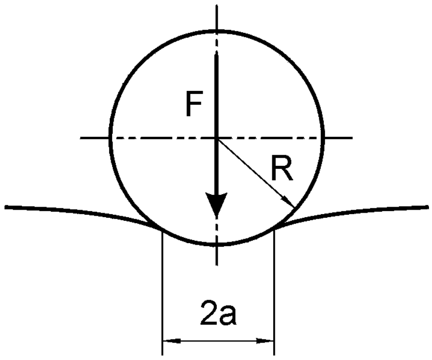

Figure 1. Ball shaped solid on an elastic half-space

The Hertz-theory gives an exact solution for radius a, which can be solved by equation (1):

$$a = \sqrt[3]{1.5\left(1-n^2\right)\frac{F \cdot R}{E}} \qquad (1)$$

If the two solids do not have the same modulus of elasticity, E becomes:

$$E = \frac{2E_1 \cdot E_2}{E_1 + E_2} \qquad (2)$$

The maximum distribution of the hemispherical pressure is given by equation (3):

$$p_{\max} = \frac{1}{p}\sqrt[3]{\frac{1.5 \cdot F \cdot E^2}{R^2 \cdot \left(1-n^2\right)^2}} \qquad (3)$$

The maximum indentation of the two bodies into each other is given by equation (4):

$$w = \sqrt[3]{\frac{2.25 \cdot \left(1-n^2\right)^2 \cdot F^2}{E^2 \cdot R}} \qquad (4)$$

3.2 Numerical Investigation

The measurement device used consists of an elastic solid with finite dimensions and, because of the large radius of a curvature, a spherical segment of a solid ball. We try to evaluate this problem by numerical FE calculation [6] for the actual measuring device (figure 2).

The next section shows that the influence of the inserted pressure sensor depends on the elastic features of the contacting solids compared with those of the sensor. For similar moduli of elasticity, the influence of a thin sensor can be neglected.

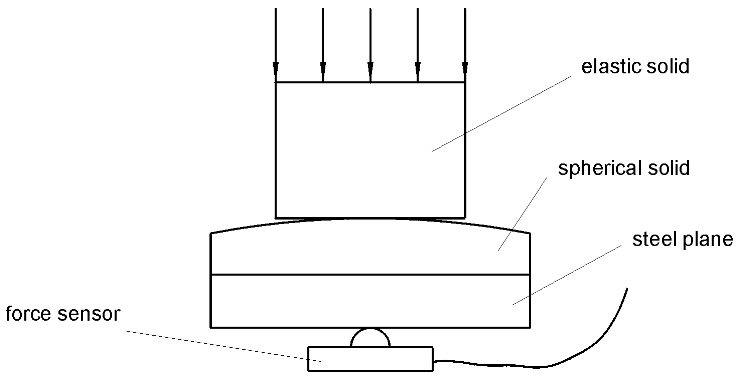

Figure 2. Measuring device for ball/half-space

The following investigations consider an aluminium ball in contact with a steel plane. The plane is loaded by a force of about 1050 N, the radius of the ball is 1000 mm. Figure 3 shows the distribution of the pressure at the surface of contact with and without sensor sheet compared with the theoretical solution. Because of the low E modulus of the used measuring layer (E = 3000 N/mm^2, μ = 0.4) the contact area becomes larger and the maximum pressure decreases.

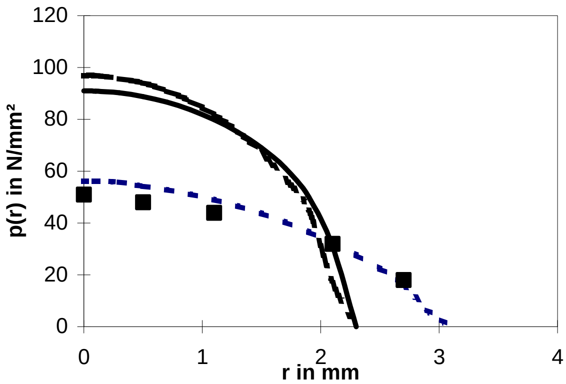

Figure 3. Pressure distribution between an aluminium ball and a steel plane (FE calculation, measurements), *F*=1050N, *R*=1000mm
——— theoretical solution
-·-·-·-·- without sensor (FE calculation)
·········· with sensor (FE calculation)
■ measurements with Fuji/GODAV

The measurements using the device shown in figure 2 were carried out by means of FUJI-prescale-film. Because of the small contact area the TEKSCAN system with its low resolution cannot be used. The dotted line shows results of FUJI/GODAV measurements in the case of a steel cube pressed against an aluminium ball. The measured maximum pressure and the radius of the contact area correlate closely with the numerical calculation. One of the most important influence factors is the tolerance of the geometric shape of the bodies in contact. High surface quality is assumed [7].

4 PROFILED CYLINDER IN CONTACT WITH AN ELASTIC HALF-SPACE

Similar conditions to those depicted in the above diagrams showing the case of ball pressure (3-dimensional) also exist in the case of a smooth cylinder (2-dimensional) pressed onto an elastic half-space. A metrological case deriving from this, which occurs frequently in industrial practice is the measurement of the surface pressure between a profiled cylinder and an elastic half-space over which it is rolling (see figure 4).

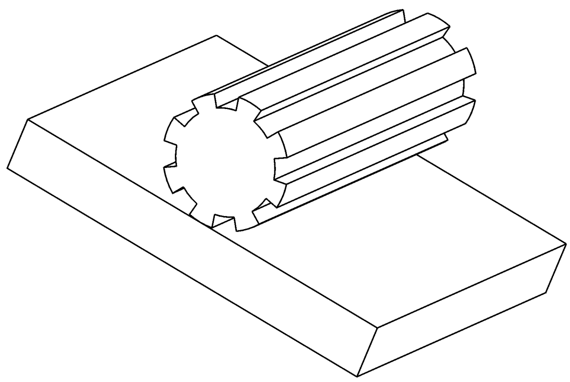

Figure 4. Profiled cylinder pressed against a plane

Such measurement problems are frequently encountered in machinery used in the processing, packaging and printing industries. In these cases, too, the introduction of a measurement sensor of finite thickness alters the surface pressure to be measured, thus naturally falsifying the measurement result. Findings indicate firstly that there is pronounced deviation of the measurement results due to the influence of the sensor thickness, and secondly that no conclusions as to momentary pressure distribution can be drawn from continuous measurement of pressure distribution with Fuji prescale film during the rolling process

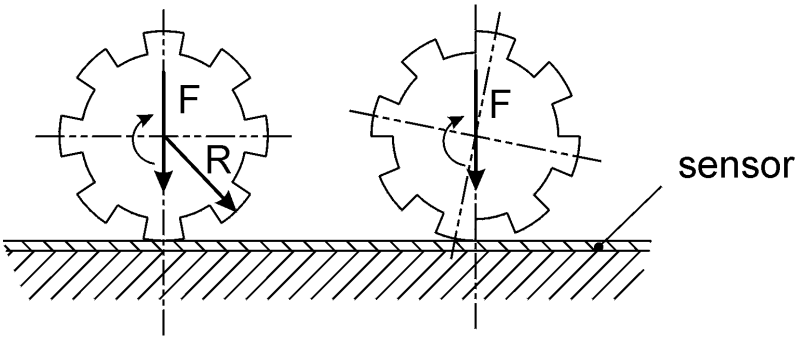

Figure 5. Profiled Cylinder rolling over a plane

The problem studied is the case of an infinitely long profiled cylinder pressed into an elastic half-space over which it is rolling. The solution aspects of most interest are the stresses near the contact area. The first step is the investigation of the stresses produced by a cylinder pressed against an elastic half-space, knowing the exact solution by the Hertz theory. The second case is the influence of the edge while the cylinder is rolling over the plane.

4.1 Theoretical Solution

Initially the cylinder and the plane are in contact in single line. Increasing the load results in deformation of both bodies and the contact area becomes 2-dimensional. They make contact over a long strip of width 2a lying parallel to the former contact line. Hertz considered this case as a limit of an elliptical contact when b was allowed to become large compared to width a [8].

With the nomenclatures of figure 1 the exact solution for a can be determined:

$$a = \sqrt{\frac{8F \cdot R \cdot \left(1 - n^2\right)}{p \cdot E \cdot l}} \tag{5}$$

If the two bodies do not have the same modulus of elasticity, it has to be solved similar to E in equation (2). The maximum value of the hemispherical pressure distribution is given by:

$$p_{\max} = \sqrt{\frac{F \cdot E}{2p \cdot R \cdot l\left(1 - n^2\right)}} \tag{6}$$

4.2 Numerical Investigation

The general aim of contact simulations is to identify the areas on the surfaces that are in contact and calculate the contact pressures. The major problem is to allow forces to be transmitted from one part of the model to another. Another problem is to decide if parts of the surfaces are in contact or not. Especially in ball-plain or cylinder-plain contact simulations there are parts of the surfaces which are not in contact with each other at the beginning, but come into contact when the load increased.

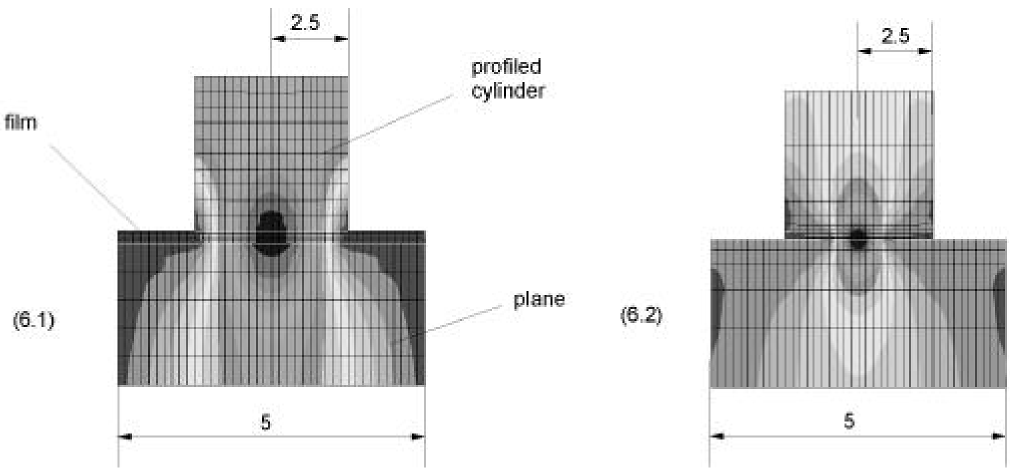

Figure 6. Contour plots of the stress distribution with (6.1) and without film (6.2), units in mm

The analysis must be able to detect when two surfaces are in contact and apply the contact constraints accordingly. To evaluate this problem by numerical FE calculations we use ABAQUS [9], which is able to solve this type of non-linear study.

Figure 6 shows two cases. That on the left is a calculation with a film inserted between the bodies, while that on the right is without film. In both cases, the edge is so far away from the center of the contact area so that there is no contact near the edge.

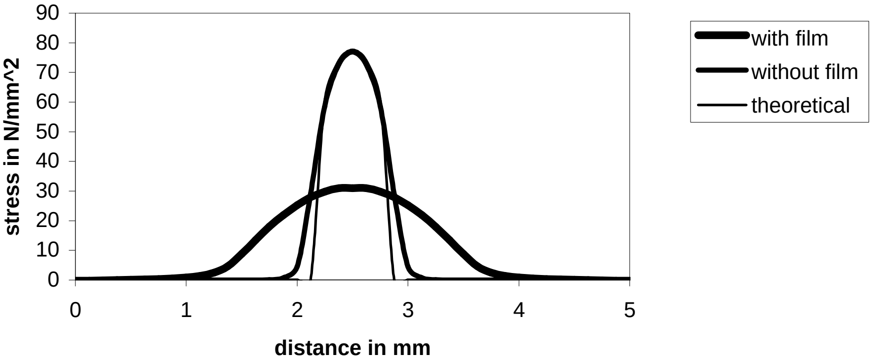

Figure 7. Comparison of the theoretical solution with the FE calculations. F = 50000 N, R = 300 mm.

Figure 7 shows the comparison of the theoretical solution by Hertz, the FE solution with film and without film. The calculated stresses are in vertical direction parallel to the direction of the load in the first node row underneath the contact surface of the elastic half-space. The distance runs from the left side to the right side of the target-body. As shown in figure 7 the maximum of stress is exactly in the middle of the model, the solution without film is lower than with film and the contact area is larger. Theoretical and FE calculation without film accord closely in respect of maximum stress, there is only a difference in the width of the contacting area due to the rough element modeling.

4.3 Rolling-down-process

This section investigates the influence of the edge by rolling down a plane. Two FE models are used, one with film and the other without film.

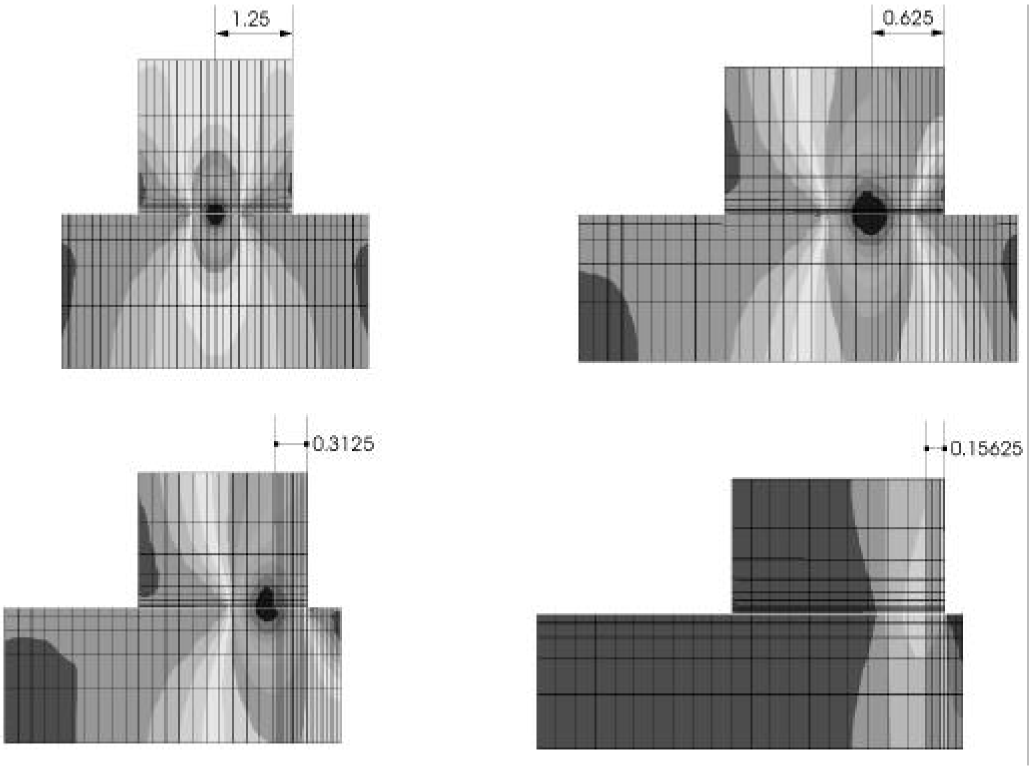

Figure 8. Contour plots of the stress distribution for different distances from the edge in mm without film.

Figure 8 shows 4 FE calculations with different distances of the theoretical center of the contacting area from the edge. The first 3 pictures show contour plots in the case of no contact at the edge. The fourth picture is with edge contact. The same model is used for all investigations, only the distance from the edge being varied.

Figure 9 shows a comparison of the studied cases. The stress direction is the same as in figure 6, as is the load. The distance from the edge is given in the legend of the diagram.

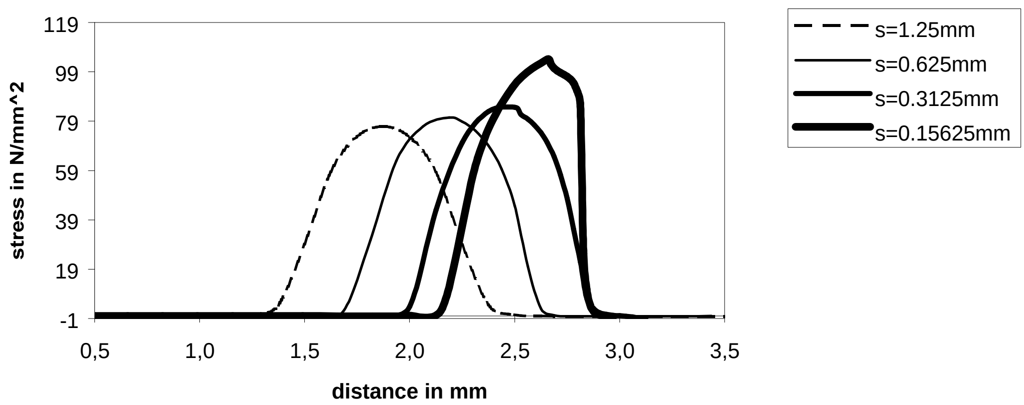

Figure 9. Comparison of 4 FE-calculations for the cases 1 to 4 of figure 8.

This section investigates the same cases as in the above section but under the influence of an inserted film. The film has a thickness of 0.2 mm. Case 1 has no contacting edge, in the subsequent cases there is contact.

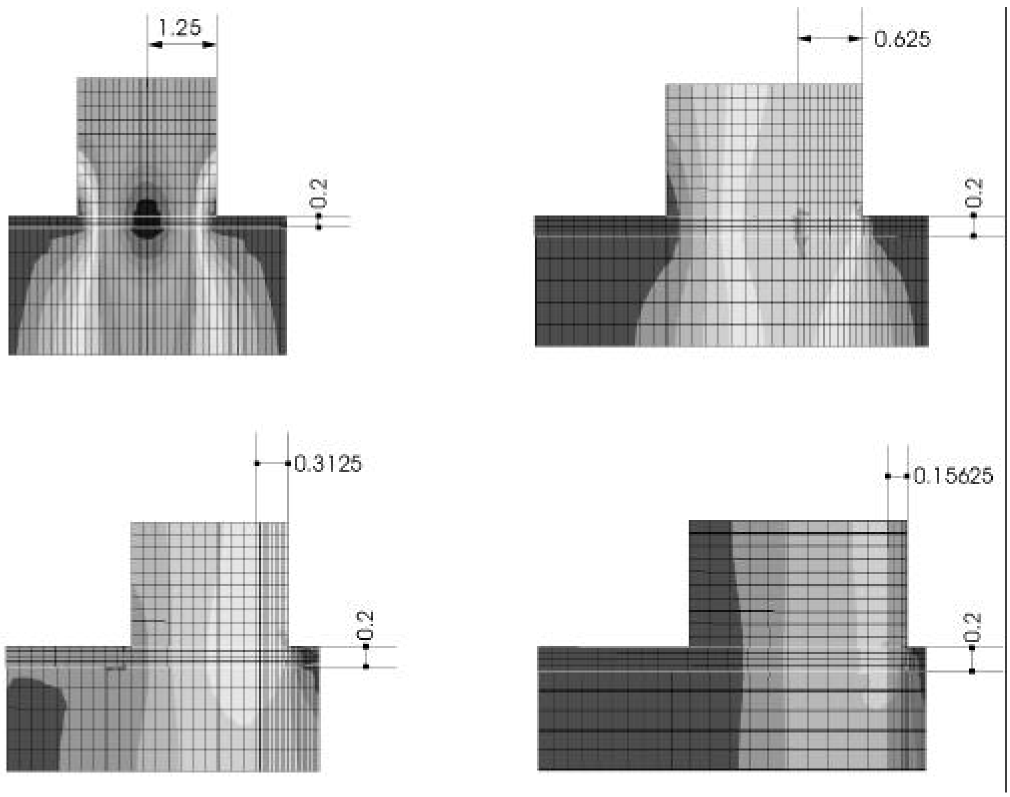

Figure 10. Contour plots of the stress distribution with an inserted film, units in mm.

Figure 11 is a comparison of these cases. The stresses are calculated for the node row in the exact middle of the film. There is an increase of stress depending on the distance of the edge from the theoretical contact-point. They have lower maximum stress and larger contact area compared to the solution without film.

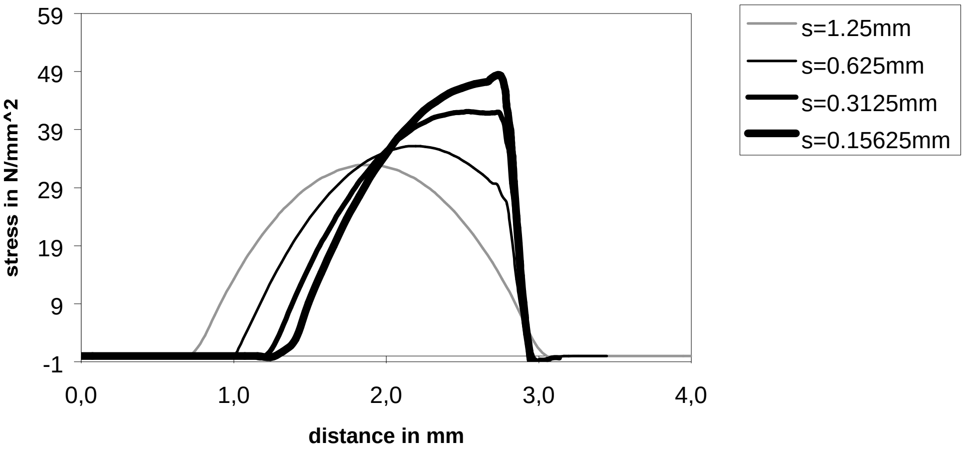

Figure 11. Comparison of stresses in the middle of the film.

Figure 12 shows a comparison of the middle and the first node row underneath the surface of the film. In the first node row there is a peak of increasing stress, when the edge is coming in contact with the film.

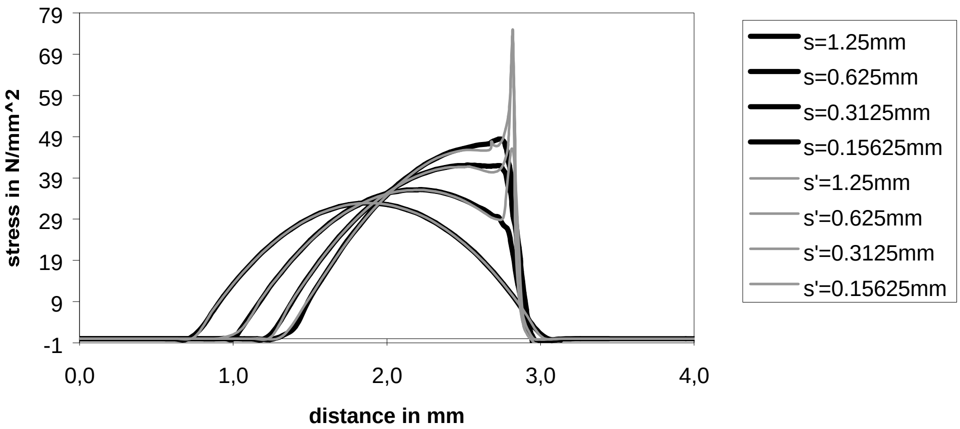

Figure 12. Comparison of the middle and the first node row under the surface of the film.

5 SUMMARY

The problem of surface pressure measurement was illustrated using two classic contact problems, firstly point contact and secondly line contact. It was proven that both the contact area and the pressure distribution are significantly influenced by the insertion of even very thin pressure sensors. The quantitative change in the pressure distribution and thus the distortion of the measurement results can be determined by comparing FE calculations with measurement results. This distortion is particularly pronounced when the two contact bodies have rigidities considerably higher than that of the inserted pressure sensor.

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AUTHORS: K. HOFFMANN, M. EGGER and W. WAHLER, Institute for Design Engineering and Transport-, Handling- and Conveying Systems, Vienna University of Technology, Getreidemarkt 9
A-1060 Vienna, Austria, Phone ++43 1 58801 30746, Fax ++43 1 58801 30799
E-mail: hoffmann@ft.tuwien.ac.at