

Effect of a Deformable Roller on Residual Stress Distribution for Elastic-Plastic Flat Plate Rolling under Plane Strain Conditions

M. Y. Ali and J. Pan
 Univ. of Michigan-Ann Arbor

ABSTRACT

In this paper, the differences of the residual stresses due to rolling in a finite elastic-plastic plate by rigid and elastic deformable rollers at very high rolling loads are investigated by two-dimensional plane strain finite element analyses using ABAQUS. In the finite element analyses, the rollers are modeled both as rigid and linear elastic, and have frictionless contact with the elastic-plastic finite plate. The plate material is modeled as an elastic-plastic power-law strain hardening material with a non-linear kinematic hardening rule for loading and unloading. Two new numerical schemes are developed to represent the elastic roller to model the indentation and rolling. The results of the contact pressure and subsurface stress distributions from the two numerical schemes are almost identical. For both roller models, the computational results show that the contact pressure and subsurface stress distributions in the elastic-plastic plate are similar for both indentation and rolling at high rolling loads with extensive plastic deformation. The computational results also indicate that the residual stresses after rolling are nearly the same for both roller models. The computational results suggest that the rigid roller model can be used to obtain the residual stresses in rolling simulations at very high rolling loads without sacrificing the accuracy.

CITATION: Ali, M. and Pan, J., "Effect of a Deformable Roller on Residual Stress Distribution for Elastic-Plastic Flat Plate Rolling under Plane Strain Conditions," *SAE Int. J. Mater. Manf.* 5(1):2012, doi:10.4271/2012-01-0190.

INTRODUCTION

For automotive crankshafts, fatigue damages have been observed near the fillets due to stress concentration. The fillet rolling process has been used for years to introduce compressive residual stresses near crankshaft fillets and, consequently, to improve fatigue strengths of crankshafts [1,2,3]. The residual stresses can significantly affect the fatigue lives of the crankshafts under high cycle fatigue loading conditions. Previously, researchers estimated the residual stresses near crankshaft fillets either using two-dimensional or three-dimensional finite element analyses [2,3,4,5,6,7,8]. Numerous studies were conducted to understand the stress distributions and deformation patterns near rolling contact surfaces [9,10,11,12,13,14,15,16,17,18,19,20,21,22]. A brief discussion of these works can be found in [23].

Most researchers used the moving Hertzian pressure distribution to represent the contact pressure for rolling with p_0/k ratios ranging from 5 to 9, where p_0 is the maximum Hertzian pressure and k is the shear yield stress of the material. For crankshaft rolling, the p_0/k ratio can be as high

as 22 as indicated in [23]. Ali and Pan [23] investigated the differences of the residual stresses distributions due to single indentation and rolling by a rigid roller on a finite plate with a very large contact pressure that is relevant to the crankshaft rolling process with a high p_0/k ratio of 22 based on two-dimensional plane strain finite element analyses. The computational results for the rolling case show a significantly higher longitudinal compressive residual stress and a lower out-of-plane compressive residual stress along the contact surface when compared to those for the single indentation case. Their investigation is further extended in this work to understand the effect of an elastic deformable roller on the residual stresses due to single indentation and rolling with a high p_0/k ratio of 22. A high p_0/k ratio of 22 referred to in this paper is based on the results of a rigid roller indenting an elastic body. For a similar contact configuration, if a rigid roller is replaced by an elastic roller, the p_0/k ratio becomes about 16 due to the elastic deformation of the roller to lower the value of p_0 .

This investigation is aimed to quantify the differences between the residual stress distributions in an elastic-plastic

plate due to single indentation and rolling by rigid and elastic rollers with a very large contact pressure. As shown later in this paper, the maximum contact pressure and contact width for an elastic roller indenting an elastic plate are significantly lower and higher than those for a rigid roller on elastic plate, respectively. More importantly, the advantages of using a rigid roller instead of an elastic roller give the convenience of applying the roller load with a smaller model size. Both conditions of indentation and rolling are selected for this study to identify whether a rigid roller model can be used to replace an elastic roller model for crankshaft rolling simulations with a sufficient accuracy to obtain the residual stresses distribution for further fatigue strength analyses. While a three-dimensional finite element analysis is more appropriate to calculate the residual stresses due to single indentation and rolling, a two-dimensional model is chosen here to save computational effort.

In this paper, a crankshaft fillet rolling along the circumference of the main journal is idealized to a two-dimensional plane strain flat plate rolling with appropriate boundary conditions. The roller is assumed to be both rigid and linear elastic and the contact between the roller and the plate is frictionless. The roller load per unit width for the two-dimensional model is obtained from the corresponding three-dimensional finite element analysis of single indentation on the crankshaft fillet [23]. Here, the elastic Hertzian solution for contact of two elastic cylinders is first reviewed. Then, the results of elastic plane strain finite element analyses of single indentation on a flat plate are presented for both roller models in order to examine the differences of the elastic contact pressure distributions obtained from both roller models. For this case, the plate is modeled as linear elastic. Next, the results of the elastic-plastic plane strain finite element analyses of single indentation and rolling on a flat plate are presented and compared at a high rolling load with extensive plastic deformation. For this case, the plate material is modeled as an elastic-plastic power-law strain hardening material with the non-linear kinematic hardening rule of ABAQUS [24] for loading and unloading. For the rolling case, only a single pass rolling is simulated. The subsurface stress and residual stress distributions under the roller center induced by the rolling are then compared for the rigid and elastic roller cases. Finally, some conclusions will be made based on the results of this investigation.

MODELING PROCEDURES

Figure 1 shows a schematic of a two-dimensional model of a flat plate rolling process. The cylindrical roller is assumed to be both rigid and linear elastic, and the contact between the roller and the plate is frictionless. Figure 1 shows the phases of the rolling simulation. Initially, the roller is sitting on the starting position of the flat plate. Then the roller is moved down with a given load. The roller is then rolled horizontally to the right. After traveling the full rolling length, the roller is moved up and the rolling is completed.

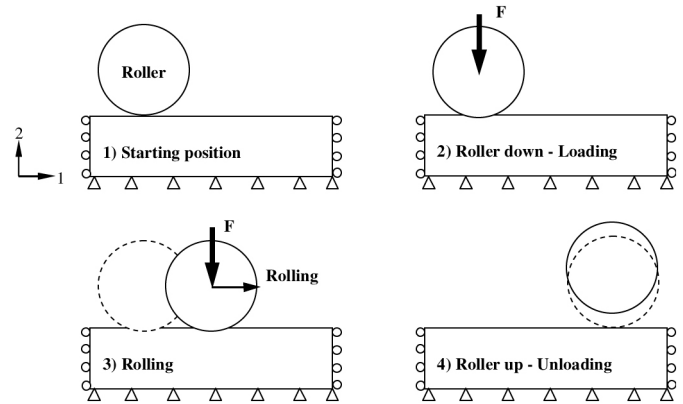


Figure 1. A schematic of a two-dimensional model of a flat plate rolling process. The roller is assumed to be both rigid and linear elastic, and the contact between the roller and the plate is frictionless.

The finite element analyses were performed using the commercial finite element code ABAQUS. The numerical scheme of a two-dimensional finite element analysis of a rigid cylindrical roller rolling on a finite flat plate was presented in Ali and Pan [23] and will not be repeated here. Figure 2(a) shows a two-dimensional finite element model of an elastic cylindrical roller rolling on a finite flat plate. The Cartesian coordinate system is also shown in the figure (throughout this paper the Cartesian coordinate notations x , y , z and 1, 2, 3 will be used interchangeably for convenience). The length and the thickness of the plate are 300 mm and 24.552 mm, respectively. Two-dimensional plane strain 4-noded CPE4R elements of size 0.4 mm by 0.4 mm are used in this model. The selection of the sizes of the finite elements to simulate single indentation and rolling follows the guidelines that were established in Ali and Pan [25] and will not be reported here. The diameter of the roller is 14 mm. The bottom of the plate is fixed, and the displacements on both end surfaces AB and CD of the plate in the x direction are constrained. The rolling start point on the upper surface is located at $x = 62$ mm and the rolling end point is located at $x = 216.264$ mm to eliminate possible end effects.

The loading and boundary conditions on the elastic roller may be applied in different ways to conduct the plane strain finite element simulations of rolling. Two simulation schemes are presented here: BC-1 and BC-2. For BC-1, a detailed view of the elastic roller loading and the boundary conditions are shown in Figure 2(b). An analytical rigid surface or rigid body of ABAQUS is defined horizontally. The rigid surface is in contact with the upper part of the elastic roller. This top rigid surface is used to apply the rolling load to the elastic roller in the downward direction. The rotational degree of freedom of the rigid surface reference node is constrained but it can move vertically with the applied load and horizontally with an applied motion through a rigid body reference node.

It should be noted that the roller mesh must be sufficiently refined at the contact location between the rigid surface and

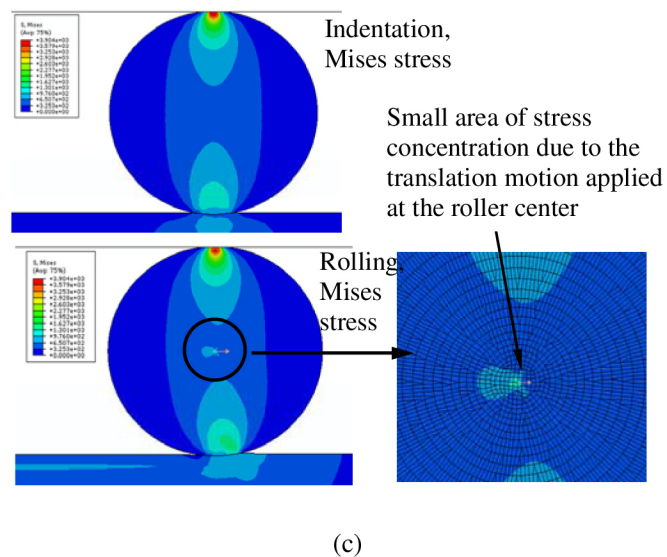
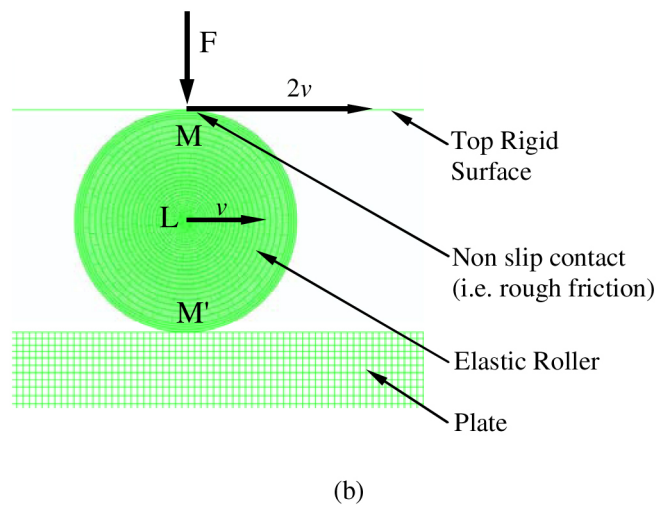
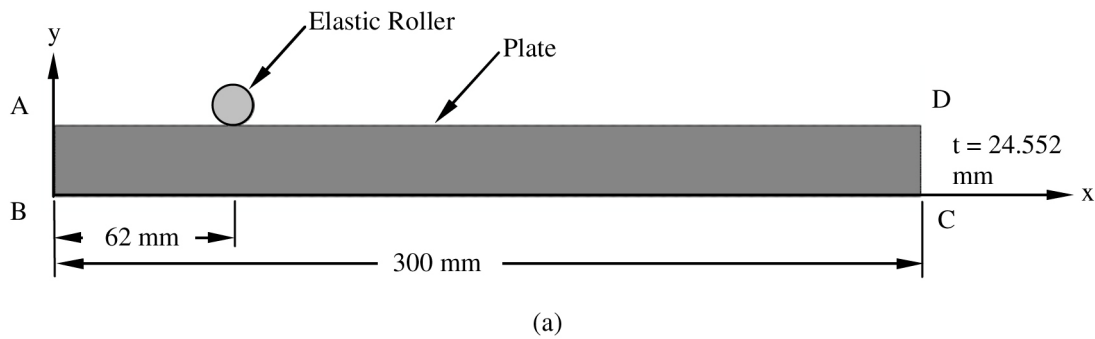


Figure 2. (a) A two-dimensional finite element model of an elastic cylinder rolling on a finite flat plate, (b) a detailed view of the loading and boundary conditions (BC-I) of the elastic roller, and (c) the Mises stress distribution of the roller for single indentation and rolling to examine the effect of the boundary conditions applied at the roller center L and the contact point M.

the roller to avoid numerical convergence issues. The roller is meshed predominantly by quadrilateral elements. In this study, 4-noded quadrilateral elements of size 0.1 mm by 0.16 mm are used for the contact layer of the roller with the 0.1 mm side on the contact side for the given load. The element sizes were selected based on the results of a mesh sensitivity study of the elastic finite element analysis benchmarked with the elastic Hertzian solution as discussed in [25]. The contact between the top rigid surface and the roller is modeled as non-slip (i.e. rough friction) to assist the rigid body rotation of the roller. The arrangement for the rotation of the roller provides the necessary numerical convergence although it is not necessary for the frictionless contact between the roller and the plate as investigated here.

As shown in Figure 2(b), the roller center is denoted by L. At any instant, M and M' represent the contact points on the top and bottom surface of the roller, respectively. The distance between M and M' before loading represents the diameter of the elastic roller. A velocity boundary condition v is applied at the roller center L in the x direction to translate the roller in the rolling direction and to overcome the resistance of the indented surface due to the applied high rolling load. The roller center L is free to move in the vertical direction. The velocity boundary condition of the top rigid surface is chosen to be $2v$ in the x direction based on the rigid body dynamics. With the non-slip condition, the relative velocity of point M is v in the positive x direction with respect to L. On the other hand, the relative velocity of point M' is v in the negative x direction with respect to L. The top rigid surface velocity is chosen here such that there is no relative motion between the roller and the plate at the contact point (near M') according to the rigid body dynamics. This is verified by another finite element analysis that at high rolling load, the roller would be standing still at the rolling start position with only rigid body rotation from the contact at M without the translational motion at the roller center due to frictionless contact between the roller and the plate. This is because the shear traction at point M due to non-slip condition is much smaller than the force necessary to overcome the resistance due to the indented surface at high rolling loads.

It should be noted that the deformed length of LM and LM' would be slightly different under the given rolling load since the elastic roller is contacting the top rigid surface and the lower elastic-plastic plate. The results of the finite element analyses for indentation and rolling indicate that the difference of the deformed lengths of LM and LM' is less than 0.2% of the undeformed roller radius. Therefore, the velocity boundary conditions given at M and L based on the rigid body dynamics appears to be quite acceptable. Figure 2(c) shows the Mises stress distribution of the roller for single indentation and rolling to examine the effect of the boundary conditions applied at the roller center L and the contact point M. As shown in the figure, the Mises stress is localized near the roller center L where the velocity boundary condition is applied and it does not appear to have impact on the stress

distributions of the roller near the contact locations. Thus applying the translation motion at the roller center may be acceptable for the current investigation.

Another set of elastic roller loading and boundary conditions (BC-2) is also used for verification purpose and is schematically shown in Figure 3. For this set of boundary conditions, there is no top rigid surface and the rolling load is directly applied on the upper side of the roller (near M). The rolling load is evenly divided among a few nodes on the top surface of the roller to avoid numerical problems in the finite element analysis. The selected nodes are the corner nodes of the elements on the roller circumference where the total length of the elements is equivalent to the contact width $2a$ (0.76615 mm for the current case) based on the Hertzian solution for the elastic roller contacting a rigid surface under the given rolling load. The effects of these point loads on the roller are much localized and should not affect the stress and strain distributions near the contact surface between the roller and the plate far away from the applied load based on the Saint-Venant's principle. A velocity boundary condition v is still applied at the roller center L in the x direction to translate the roller in the rolling direction and to overcome the resistance due to the indented surface under the high rolling load. Here, the rigid body rotation of the roller is not necessary since the contact between the roller and the plate is frictionless and this differentiates BC-2 from BC-1 as shown in Figure 2(b). The results of the stress distributions in the plate for single indentation and rolling with the BC-2 are found in very good agreement with those of the BC-1. The results of the comparisons are not presented here for brevity. However, for the elastic roller case, the results based on the BC-1 loading and boundary conditions are presented in this paper, since the BC-1 can also be applied to the cases where the friction between the elastic roller and the plate is considered.

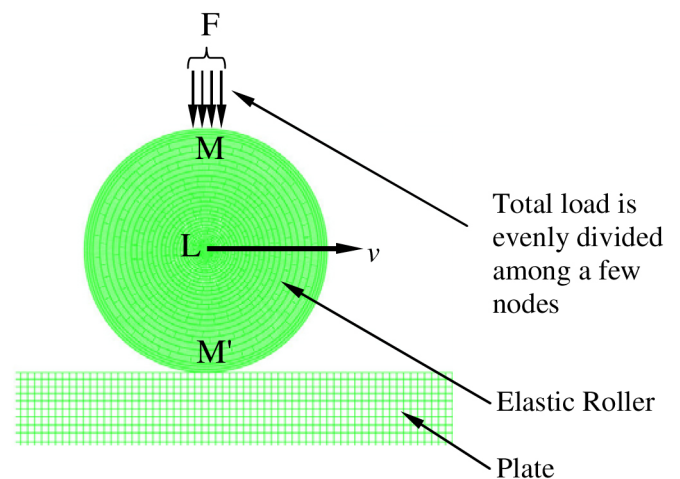


Figure 3. A detailed view of the loading and boundary conditions (BC-2) of the elastic roller.

It should be mentioned again that the numerical scheme of a two-dimensional finite element analysis of a rigid cylinder

rolling on a finite flat plate was developed, and the details of the loading and boundary conditions for the rigid roller were presented in [23]. One goal of this investigation is to examine the residual stresses in crankshafts after fillet rolling by a rigid and an elastic roller models. The total rolling length 154.264 mm and the plate thickness 24.552 mm are corresponding to the circumferential length and the radius of the main journal of a crankshaft of interest, respectively. The details to determine the rolling load can be found in [23].

Figure 4 shows the tensile stress-strain curve of the plate material. The nonlinear kinematic hardening rule of ABAQUS is assumed based on the tensile stress-strain curve shown in Figure 4. The nonlinear kinematic hardening rule in ABAQUS is based on the Mises yield function. For the loading/unloading/reloading process, the yield function can be expressed as

$$f(\sigma_{ij} - \alpha_{ij}, \sigma^0) = \left[\frac{3}{2} (\sigma'_{ij} - \alpha'_{ij}) (\sigma'_{ij} - \alpha'_{ij}) \right]^{\frac{1}{2}} - \sigma^0 = 0 \quad (1)$$

where σ_{ij} is the stress tensor, σ'_{ij} is the deviatoric stress tensor, α_{ij} represents the center of the yield surface, α'_{ij} is the deviatoric part of α_{ij} , and σ^0 represents the size of the yield surface. Here, σ'_{ij} and α'_{ij} are defined, respectively, as

$$\sigma'_{ij} = \sigma_{ij} - \frac{1}{3} \sigma_{kk} \delta_{ij} \quad (1a)$$

$$\alpha'_{ij} = \alpha_{ij} - \frac{1}{3} \alpha_{kk} \delta_{ij} \quad (1b)$$

where δ_{ij} is the kronecker delta.

The associated plastic flow is expressed as

$$\dot{\epsilon}_{ij}^p = \dot{\bar{\epsilon}}^p \frac{\partial f(\sigma_{ij} - \alpha_{ij}, \sigma^0)}{\partial \sigma_{ij}} \quad (2)$$

where $\dot{\epsilon}_{ij}^p$ represents the plastic strain rate tensor and $\dot{\bar{\epsilon}}^p$ is the equivalent plastic strain rate defined as

$$\dot{\bar{\epsilon}}^p = \left(\frac{2}{3} \dot{\epsilon}_{ij}^p \dot{\epsilon}_{ij}^p \right)^{\frac{1}{2}} \quad (2a)$$

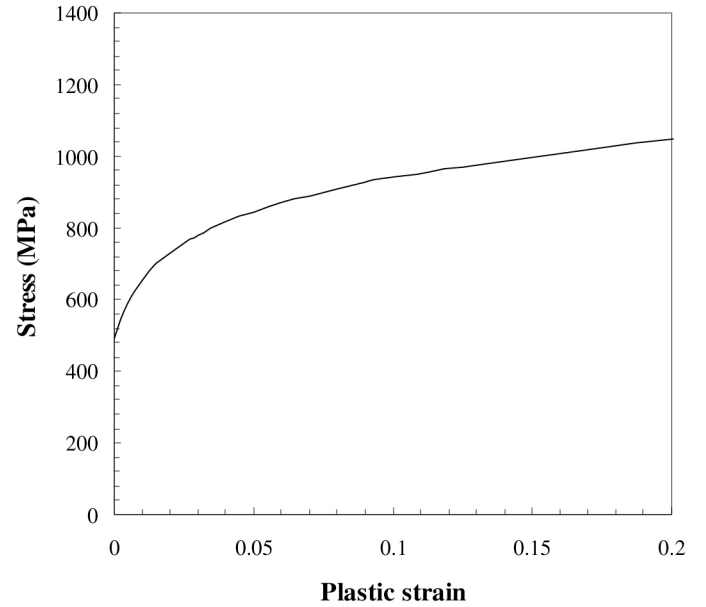


Figure 4. The tensile stress-strain curve of the plate material used in the finite element analyses.

In ABAQUS, the nonlinear kinematic hardening rule is defined to be an additive combination of a purely kinematic term of the Ziegler linear hardening law and a relaxation term which introduces the nonlinearity. The evolution of α_{ij} is expressed as

$$\dot{\alpha}_{ij} = \frac{C}{\sigma_0} \dot{\bar{\epsilon}}^p (\sigma_{ij} - \alpha_{ij}) - \gamma \dot{\bar{\epsilon}}^p \alpha_{ij} \quad (3)$$

where σ_0 is constant and represents the size of the initial yield surface, α_{ij} represents the center of the yield surface, $\dot{\bar{\epsilon}}^p$ is again the equivalent plastic strain rate, and C and γ are material parameters that must be calibrated from cyclic test data. Here, C is the initial kinematic hardening modulus and γ determines the rate at which the kinematic hardening modulus decreases with increasing plastic deformation. When C and γ are zero, the model reduces to the isotropic hardening rule. When γ is zero, the Ziegler linear hardening law is recovered. Note that, for the nonlinear kinematic hardening rule of ABAQUS, the option of “HALF CYCLE” for data type was adopted in order to use the cyclic stress-strain curve of the material as the input stress-strain data [24].