

Effects of Roller Diameter and Number on Fatigue Lives of Cam Roller Follower Bearings

2011-01-0489

Published
04/12/2011Jaewon Lee and Jwo Pan
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doi:[10.4271/2011-01-0489](https://doi.org/10.4271/2011-01-0489)

ABSTRACT

Effects of roller diameter and number on the contact pressures, subsurface stresses and the fatigue lives of cam roller follower bearings are investigated in this paper. Finite element analyses under plane strain conditions were conducted to identify the effects of the diameter and number of the rolling elements and the thickness of the outer ring. The fatigue life of the inner pin generally increases as the roller diameter increases. But, reducing the number of rollers to accommodate larger rollers does not necessarily increase the fatigue life. The inevitable decrease of the thickness of the outer ring due to the increase of the roller diameter results in the increase of compliance for the outer ring. This increase of compliance leads to excessive deformation of the outer ring and consequently more load must be carried by fewer number of rolling elements. For cam roller follower bearings, a parametric study based on the finite element analyses should be conducted to obtain the optimum geometry which gives the lowest contact pressure and maximum subsurface Mises stress which results in the longest fatigue life.

INTRODUCTION

Cam followers are an essential component of internal combustion engines and are required to control air and fuel flow into the combustion chambers. Previously and even in the current applications, cam followers with sliding contact have been and are still being used due to its simplicity. The fatigue lives and wears of cam followers with sliding contact have been investigated [1, 2, 3]. However, due to the requirements of high performance engines of today [4, 5], engineers have sought design changes which demand

valvetrains with higher speed and acceleration and thus more force applied to the cam follower system. Since the friction force is dictated by the normal contact force, the current trend is to use cam roller followers instead of cam followers with sliding contact to minimize frictional forces. Cam roller followers are expected to soon replace most cam followers with sliding contact and take the majority of the cam follower system [6]. The advantages of using cam roller followers instead of cam followers with sliding contact also includes less wear and energy loss [7]. However, there are also disadvantages due to the increase in contact stress, complex functional requirements and manufacturing processes [6].

Studies have been conducted on the fatigue lives of cam roller followers both experimentally and analytically. Gecim [8] investigated the lubrication characteristics between the cam and cam roller follower, and derived a fatigue life equation. Lee et al. [9] developed a method to calculate the normal force and contact stress between the cam and cam roller follower. Krepulat et al. [10] obtained the contact pressure between the cam and the cam roller follower with extensive use of finite element analyses. Hua et al. [11] conducted an investigation on the contact fatigue of the cam surface using a multi-scale system analysis.

Cheng et al. [12] have identified through experiments that the failure mode of cam roller follower bearings is subsurface crack initiation and propagation. Also, an exhaustive review by Sadeghi et al. [13] on rolling contact fatigue can provide insight on the fatigue life assessment of cam roller follower bearings. However, the preliminary results of this study have shown that due to the unique boundary condition of cam

Table 1. Dimensions for the 18-roller and the 15-roller model (mm).

	18-1	18-2	18-3	15-1	15-2	15-3
Inner pin diameter, (d_i)	9	9	9	9	9	9
Outer ring diameter, (d_o)	12.1	12.4	12.6	13	13.3	13.6
Outer diameter, (D_o)	17.8	17.8	17.8	17.8	17.8	17.8
Rolling element diameter, (d_r)	1.58	1.70	1.82	2	2.15	2.31
Relative radius of curvature, (R)	0.67	0.71	0.75	0.81	0.87	0.92

roller followers, the previous rolling fatigue analyses may not be suitable for applications to cam roller follower bearings.

A simple solution to increase the fatigue lives of bearings is to utilize rollers with larger diameters. A larger roller will increase the contact area and allow the applied load to spread out. Consequently, the contact pressures and the subsurface stresses are reduced. However, for cam roller follower bearings, there exists a trade-off between the increase in the roller diameter and the decrease in the outer ring thickness. A larger roller must be accompanied by a decrease of the outer ring thickness or the inner pin diameter due to the geometric restriction of the bearings. The outer ring thickness directly affects the compliance and the amount of deformation of the outer ring under a given load. The effect of this increase of compliance has not been investigated previously.

In this paper, finite element analyses were conducted to understand the effects of the diameter and number rolling elements on the contact pressure distribution, maximum subsurface Mises stress, and fatigue lives of the cam roller follower bearing. Two groups of cam roller follower bearing models having different diameters and numbers of rolling elements are investigated using two-dimensional plane strain finite element analyses. Based on the results of the finite element analyses, the fatigue lives of the cam roller follower bearings of these bearing models are estimated and compared. Finally, conclusions are made.

FINITE ELEMENT ANALYSES

The major parameters which are investigated in the finite element analyses are the diameter and number of rolling elements and the thickness of the outer ring. Several assumptions have been made to simplify the finite element model. Needle rollers in cam roller follower bearings have a length to diameter ratio about 10. Therefore, the plane strain assumption will give good approximation near the middle portion of the contact area where most fatigue failures are initiated. By using this assumption, the edge effect of the rollers on the contact surface is ignored [10]. Also, a concentrated force will be used to simulate the load applied to

the cam follower by the cam. In fact, a finite contact area exists between the two elastically deformable bodies. Therefore, the cam will apply a distributed force to the cam follower over this contact area. However, for simplicity, the distributed force on this finite contact area will be replaced with a single concentrated force applied to a single node. Another assumption is the use of a constant load. The cam follower is under variable loading conditions during the normal operation of an engine. However, the load applied to the cam roller follower bearing model is assumed to be at a constant maximum load. The resulting contact pressures and subsurface stresses will represent the values that the cam roller follower bearing experiences at the peak loading of the cam and cam follower interface. This assumption can underestimate the fatigue lives of the cam roller follower bearings. However, a comparison between different models can be nonetheless obtained to find the optimal geometry which gives the longest fatigue life for the given conditions.

ROLLER BEARING MODELS

In this investigation, roller bearing models consist of three components: the outer ring, inner pin and rolling elements. Two groups of roller bearing models have been created. The first group has 18 rolling elements (denoted as the 18 roller model group) while the second group has 15 rolling elements (denoted as the 15 roller model group) with comparatively larger diameters of rolling elements than those of the 18 roller models. Each group has three models. The six models have been labeled as 18-1, 18-2, 18-3, 15-1, 15-2, and 15-3 with increasing diameters of rolling elements. The dimensions are summarized in Table 1 and shown schematically in Figs. 1(a) and 1(b). The location of the roller is shown as a function of θ as marked in Figs. 1(a) and 1(b). Note that the outer diameter for the cam follower is not a variable parameter and cannot be changed without a redesign of the cam system. The rolling element diameter is smallest for 18-1 and largest for 15-3 as listed in Table 1. The values of the inner pin diameter, outer ring diameter, outer diameter and the relative radius of curvature (explained later) are also listed in Table 1.

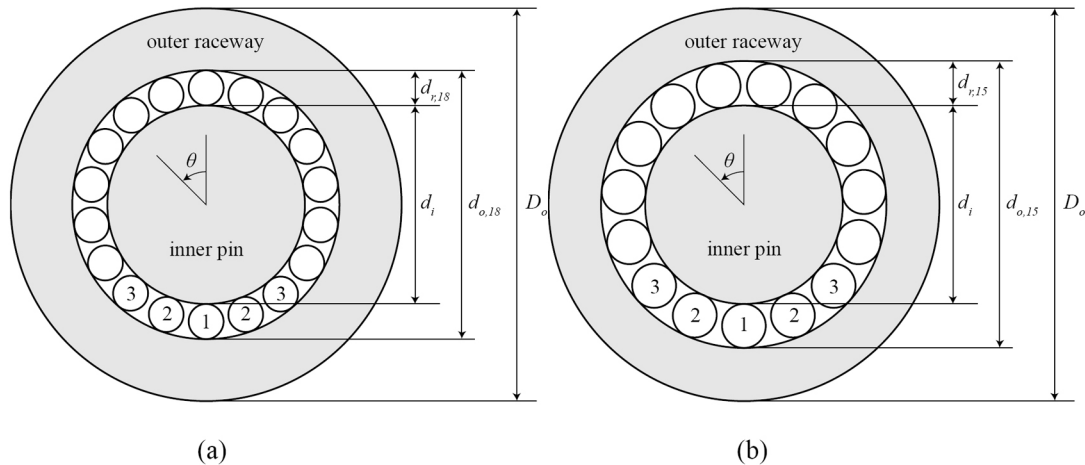


Fig. 1. Schematics of the cam roller follower bearings for (a) 18-1 of the 18 roller model and (b) 15-1 of the 15 roller model.

FINITE ELEMENT MODELS

Finite element analyses of the cam roller follower bearings based on the dimensions given in [Table 1](#) were carried out to obtain the contact pressures as well as the subsurface stresses on the inner surface of the outer ring and the surface of the inner pin. Two-dimensional plane strain finite element models are used to simulate the middle portion of the cam roller follower bearings. Second-order, isoparametric, plane strain, quadrilateral, reduced integration elements (CPE8R) are used in the models. The minimum element size is 1.54×10^{-3} mm. The finite element model is shown in [Figs. 2\(a\)](#) and [2\(b\)](#). The elastic modulus, E , is 200 GPa and the Poisson's ratio, ν , is 0.3 for the outer ring, inner pin, and rolling elements. The steels for bearing components usually have the yield stresses of more than 1200 MPa. For the given designed load, the stress states near all contact surfaces are elastic. Computations were performed using the commercial finite element software Abaqus v6.8 [14].

The outer surface of the cam roller follower bearing is free of contact except for the area of contact with the cam. Therefore, the outer surface of the outer ring is absent of constraints except for the contact region in the finite element analyses. For the contact region, a concentrated force is applied to simulate the distributed load between the cam and the cam follower as discussed in the previous section. The arrows in [Figs. 2\(a\)](#) and [2\(b\)](#) show the directions and locations of the applied forces. The magnitude of the concentrated force is 255 N/mm, which represents a typical maximum load produced between the cam and the cam follower during a normal valvetrain operation. Constraints for all in-plane motions have been applied to the central region of the inner pin, shown as the shaded areas in [Figs. 2\(a\)](#) and [2\(b\)](#). The rolling elements have no constraints except for the spring elements attached to the rolling elements to prohibit the rigid body motion. The spring elements have a relatively

small stiffness and should not affect the computational results.

FINITE ELEMENT ANALYSES RESULTS

[Figs. 3, 4](#) and [5](#) show the contact pressure, percentage of the load carried by the 1st rolling element, and the maximum subsurface Mises stress in the inner pin under the 1st rolling element from the three 18 roller models and the three 15 roller models as a function of the relative radius of curvature. It should be noted that the elastic Hertzian solution [15] give the maximum Mises stress to occur at depth of about 70% of the half width of contact. It has been concluded in a parallel investigation [16] that the subsurface damage of the inner pin occurs prior to the other two components. Therefore, only the results for the inner pin will be discussed in this paper. The relative radius of curvature, R , is defined as

$$\frac{1}{R} = \frac{1}{d_r/2} + \frac{1}{d_i/2} \quad (1)$$

where d_r is the diameter of the rolling elements and d_i is the diameter of the inner pin. Note that a larger relative radius of curvature represents a larger rolling element diameter since the diameter of the inner pin is maintained to be constant.