International Journal of Engineering Studies and Technical Approach

ISSN No. 2395-0900

Modelling, Design and Analysis of Cam and Follower-

A Review Paper

Mr. N. S. Patel

ME Scholar, Department of Mechanical Engineering, North Maharashtra University, Jalgaon

ABSTRACT

The objective of this paper is to modelling design and analysis of a cam and follower. This analysis is an important step for fixing an optimum size of a cam and follower. The model is created by the basic needs of an engine with the available background such as forces acting over the cam by means of valve while running at maximum speed. Here the approach becomes fully CAE based. CAE based approach enriches the Research and limits the time duration. Most of the IC engines used in the market have roller cam and follower mechanisms, having a line contact between the cam and the roller follower. The software (Catia and Ansys) tool has mainly been developed to enhance student learning, but it can readily be used to design modelling and analysis of cam and follower mechanisms for industrial applications. The software generates detailed information about the stress, strain, displacement, velocity etc of the follower. It also provides animation of the cam and follower mechanism.

Keywords - Cam and Follower, CAE, stress, ANSYS, CATIA V5

I. INTRODUCTION

Cam mechanism is preferred over a wide variety of machines because the cam is possible to obtain an unlimited variety of motions. The cam has a very important function in the operation of many classes of machines, especially those of the automatic type, such as printing presses, shoe machinery, textile machinery, gear-cutting machines and screw machines. The cam may be defined as a machine element having a curved outline or a curved groove, which, by its oscillation or rotation motion, gives a predetermined specified motion to another element called the follower [1]. In other word, cam mechanism transforms a rotational or oscillating motion to a translating or linear motion. In fact, cam can be used to obtain unusual or irregular motion that would be difficult to obtain from other linkage. The variety of different types of cam and follower systems that one can choose from is quite broad which depends on the shape of contacting surface of the cam and the profile of the follower. In this work an attempt is made to study the static and dynamic analysis of cam at low speed. In static analysis to study the deflection of cam and follower with respect to angular velocity and in dynamic analysis to calculate natural frequency with respect to given loading condition. The modeling of Cam and follower is done ion CATIA V5 Software and analysis of Cam and Follower is done by using ANSYS 11.0 Software. [1][2]

ISSN No. 2395-0900
IJESTA International Journal of Engineering Studies and Technical Approach

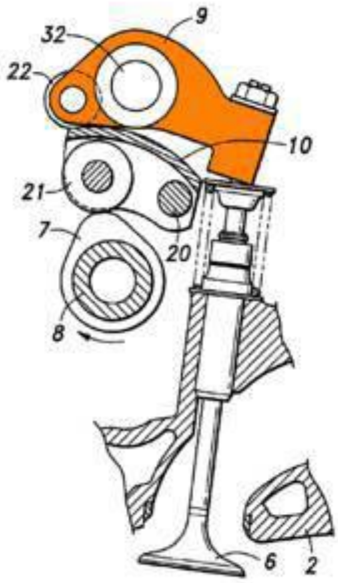

Figure 1: Rocker Arm Mechanism

Cams fail like any other machine parts and cams failure has been defined and documented. Cams failure can be categorized into three different types of failure. Those different types of failure in cam follower systems are pitting, scuffing, and polish wear [2, 3, 4]. The mechanism, which defines a particular type of failure, is complex [5]. Failure by pitting is caused by the repeated cyclic stresses that caused by time varying loads. The time varying loads in cams are due to inertia force that varies over the time, radius of curvature and the spring force. The software provides design tips in the form of warning and error messages whenever the users attempt to enter invalid values of input parameters, and suggests fault-recovery steps that help the users optimize their designs. In the Figure 1, the drive cam (7) is driven by the camshaft (8). This press on the rocker arm (10) up and down about the trunnion pin (20). Friction is reduced at the point of contact by a roller cam follower (21). A similar arrangement transfers the motion via another roller cam follower (22) to a second rocker arm (9). This rotates regarding the rocker shaft (32), and transfers the motion via a tappet to the poppet valve. In this case this opens the intake valve (6) to the cylinder head (2).

II. LITERATURE SURVEY

According to A. Rivola, M. Troncossi, G. Dalpiaz and A. Carlini, in "Elastodynamic analysis of the desmodromic valve train of a racing motorbike engine by means of a combined lumped/finite element model" if a rocker has a ratio of 1.5:1, it should open the valve 1.5 times the amount of the cam lift. Almost all factory type rockers fall short of their claims. Chevy claims a 1.5:1 rocker ratio on small-blocks, I found that most are 1.44:1 and under. In a healthy street motor .020" less valve lift could mean a 10 to 15 hp power loss. So make sure that the rockers that you choose are from a reputable company. I've had good luck with Crane Cams, Iskenderian, and Competition Cams.[5] Also in review shown in www.grapeaperacing.com/tech/ Valvetrain & according to A. Rivola, M. Troncossi, G. Dalpiaz and A. Carlini, in "Elastodynamic analysis of the desmodromic valve train of a racing motorbike engine, the centre pivot point is not on the same plane as the valve contact point. The rocker cannot push on the exact center of the valve tip throughout the entire range of motion; it must travel in an arc. By moving the valve tip contact point higher, most of the side motion occurs at low valve lifts, when spring force is at its lowest. Proper geometry is very important to limit guide wear. Another plus of making rockers with the pivot point on a lower plane than the pushrod cup and valve tip contact point is that the maximum rocker ratio will not be the same point as minimum movement. That will allow us to get some more power. Spring force is commonly referred to as spring pressure, but this is actually the wrong way to look at it. A pressure is measured in force over an area such as pounds per square inch (psi). Valve springs are measured in the pounds of force applied to hold the valve against the seat or rocker arm so it follows the intended motion. The two common measurements are seat (or installed height) force, and over the nose (peak lift) force. These two measurements are very important if the valve is to follow the intended motion of the cam profile. If the seat force is too little, the valves can bounce off the seat when closing. If the over the nose force is too little, the lifters can continue to rise after the cam lobe stops pushing it (it just launches off it) until the spring can slow it down and push it back against the cam. This is known as valve float and is very damaging to an engine. [5]

According to Khin Maung Chin, in"Design and Kinematics Analysis of Cam-Follower System", he said high accelerations are needed to give rapid opening and closing, too rapid a change in acceleration - the 'jerk' or 'jerk rate' - will give rough operation due to the sudden changes in forces. For this reason cam profiles are designed not to give very rapid changes in accelerations. It may also be noted that as higher forces can more easily be provided by the cam than by the valve springs, it is common to us higher accelerations when starting the opening of the valves and when slowing their closing at the end of the closing phase. These aspects are controlled by the cam, whereas the slowing of the valve at the end of the opening phase and the acceleration of the valve at the start of the closing phase are controlled by the valve springs. [1] According to Prof. H.D.Desai Prof. V.K.Patel, in "Computer Aided Kinematic and Dynamic Analysis of Cam and Follower", the analysis of anything other than a simple configuration can be quite complex. The analysis will depend upon the type of follower and the detailed geometry. Because of these difficulties with the analysis it was common for accelerations to be determined graphically.[2] According to Khin Maung Chin, in" Design and Kinematics Analysis of Cam-Follower System" & Dr. David J Grieve, "Forces in the Valve Train of an Internal Combustion Engine", they will make some simplifying assumptions that a knife edge follower is being used. This will not be very accurate, but will give some idea of values. The most simple assumption for analysis is to assume that the opening and closing is simple harmonic motion (SHM). Assume that the engine speed is 4000 revs/minute, this gives a cam shaft rotation speed of 2000 revs/minute (in a 4 stroke engine the cam turns at half the crankshaft speed) so the time taken for 1 revolution of the cam shaft is 0.03 seconds.[1,3] Also according to Yuan L. Lai, Jui P. Hung, and Jian H. Chen, in “Roller Guide Design and Manufacturing for Spatial Cylindrical Cams", he assumes that the cam is opening and closing the valve for 120o of its rotation. Hence the complete valve cycle is completed in 1/3 camshaft revolution, or 0.01 sec. The equation describing SHM is: displacement = amplitude x cos (angular velocity x time) .Where t=time and omega is the 'angular velocity' of the system in rad/s, and is equal to (2 x 3.14159)/ (the time for 1 cycle)

III. STRESSES AND FORCE ANALYSIS IN CAMS

The contact problem between two elastic bodies has had the attention of many researchers in the last century. In 1881, Hertz [9] investigated the contact of elastic bodies under normal loading for the first time. Hertz conducted experiments and computed the load distribution over the contact area and then solved for stresses in the body. In 1913, S. Fuchs [10] performed a laborious arithmetical integration to obtain the stresses. Morton, W. B. and Close, L. J. [11] used zonal harmonics to calculate the stresses in a half space on which a spherical ball is pressed by a normal load. Coker and Ahmed [12] conducted experimental analysis and analytical work to study the plane - stress problem when the part of the boundary of a half plane was loaded with normal pressure. Any time there is a radius in contact with another radius or flat, contact stresses will occur. In the case of two spheres contacting each other, the entire force will be imparted into a theoretical point. Due to elastic properties of the materials this point will deform to a contact area. The deformation that occurs will produce high tensile and compressive stresses in the materials.

Thomas, H. R. and Hoersch, V. A. [13], in 1930, discovered that the shearing stress on the axis of symmetry is a maximum at a distance somewhere beneath the surface under the center of the contact area. Thomas and Hoersch calculations for the stresses agreed with the experimental results. In 1936, Foeppl, L. [14] calculated the stress for the case of a cylinder and a spherical ball pressed on a flat plate. Foeppl verified his findings experimentally by a photo - elastic technique. Lundberg, G. [14], in 1939, was the first one who considered the effect of the tangential load on the stresses. Lundberg developed a general theory of elastic contact between two semi - infinite bodies where he introduced three potential functions that correspond to three components of the load along three axes of a Cartesian coordinate. Lundberg considered the components of the load tangent to the contact area as the frictional forces between the contact surfaces. Lundberg didn't calculate the stresses due to the frictional forces in addition to the normal loads. In 1953, Smith and Liu [10] studied the contact between elastic bodies with and without creep. Their model could be applied only to rectangular contact areas. They deducted the equations to calculate the normal and shear stresses analytically, both in the contact surface and inside the bodies. They used the maximum value for the tangential force, as described by Coulomb's Law. Their studies represented a big step in contact stress evaluation because they calculated the stresses under the surface, instead of only on it, like Hertz did. In 1949, Mindlin, R. D [15], investigated the stress distribution due to the tangential load when one elastic body slides over the other across the contact area. Mindlin found that the stress on the bounding curve of the contact area due to the tangential load is infinite and consequently a state of impending slipping prevails. Corresponding to this condition the intensity of tangential force at a point in the contact area cannot usually exceed the product of the coefficient of friction between the sliding surfaces and the normal pressure at the same point. In this paper, a value of 1/3 is used for the coefficient of friction, in order to show the stress distributions due to the tangential load superimposed on the normal load acting over the same contact area. Poritsky, H. [16], in December 1949, investigated the same problem by two different methods and his results agreed for both methods. Poritsky used a coefficient of friction of 0.3 in his study. In this paper, Poritsky extended the results of the solution to include an interpretation of the significance of these stresses in causing failure by inelastic yielding and by fatigue. In 1949, M'Ewen [17] evaluated the contact between two cylinders and calculated the stress field, taking into account the tangential load due to the friction on the contact area. His study was the first to include the friction force on the model.

There are forces associated with the motion of the cam and the follower. Different types of forces act on cams during their rotational movement. In design process, the magnitude, direction and line of forces action should be defined and evaluated so that the stress distribution can be defined and the dimensions of the cam can chosen to provide sufficient strength and rigidity, given the appropriate choice of materials. Generally cams and follower system are subjected to different types of forces. The type of forces that act on the cam depends on the cam application, which means that not every cam should be subjected to the same type of forces. In general, the forces acting on the cam and follower system are the inertia forces, spring forces, vibratory forces, frictional forces between the cam and the follower and other external forces.

IV. MODAL ANALYSIS

Modal analysis of Cam and follower is performed by Ansys software to determine the static stress. Static analysis check out the deflection and stresses on cam and follower mechanism. The modeling of Cam and follower is done on CATIA V5 Software and analysis of Cam and Follower is done by using ANSYS 11.0 Software. Cam follower first modeled in CATIA V5 which is excellent CAD software, which makes modeling so easy and user friendly. The model is then transferred in IGES format and exported into the Analysis software ANSYS 11.0. The Cam and Follower is analyzed in ANSYS in three steps. First is preprocessing which involves modelling, geometric clean up, element property definition and meshing. Next step includes solution of problem, which involves imposing boundary conditions on the model and then solution runs. Next in sequence is post processing, which involves analyzing the results plotting different parameters like stress.

V. STEPS TO ACHIEVE THE OBJECTIVES

The current mechanism employs a flat follower. We have to change the flat face of follower to a curved face follower, thus achieving the required point contact. We have to keep the frequency of vibration and stress in the mechanism constant after modification of follower.

1. Create a 3D geometry in CAD software from data getting from company.

2. Conduct a static stress analysis using FEA software.

3. Conduct a physical test using single axis strain gauge to validate FEA static stress analysis results with physical tests

4. Modify the CAD geometry to a curve of the follower.

5. Conduct a static stress analysis of the modified shape of the follower.

6. Forces which are induced on rocker arm pivot will be used for static stress analysis of modified follower.

7. Compare results of both static and modal analysis between existing flat follower and modified curved follower.

8. Conduct iterations if necessary.

I. CONCLUSION

ANSYS can carry out advanced engineering analyses quickly, safely and practically by its variety of contact algorithms, time based loading features and nonlinear material models. ANSYS Workbench is a platform which integrates simulation technologies and parametric CAD systems with unique automation and performance. The power of ANSYS Workbench comes from ANSYS solver algorithms with years of experience. Furthermore, the object of ANSYS Workbench is verification and improving of the product in virtual environment. The stress analysis of original roller and modified roller can be done by using CEA software. Computer-Aided Engineering (CAE) software is the designer's best friend, a tool proven to enhance the productivity of the user. It has liberated engineers from the most tedious aspects of their job, shortened project turnaround times and improved the quality and accuracy of their work. In this work finite element approach is used to optimize the shape of flat face of existing follower into a curved face of modified follower, so that the required point contact can be achieved.

VII. REFERENCES

i Khin Maung Chin,"Design and Kinematics Analysis of Cam-Follower System", GMSARN International Conference on Sustainable Development: Issues and Prospects for the GMS, Nov. 2008, pp. 12-14.

ii H. D. Desai and V. K. Patel, "Computer Aided Kinematic and Dynamic Analysis of Cam and Follower," Proceed-ings of the World Congress on Engineering, Vol. 2, 30 June-2 July 2010, London, pp.117-127

iii Dr. David J Grieve, "Forces in the Valve Train of an Internal Combustion Engine", 23rd February 2004.

iv Yuan L. Lai, Jui P. Hung, and Jian H. Chen," Roller Guide Design and Manufacturing for Spatial Cylindrical Cams", World Academy of Science, Engineering and Technology,vol. 38, 2008,pp.109-118.

v A. Rivola, M. Troncossi, G. Dalpiaz and A. Carlini, "Elastodynamic analysis of the desmodromic valve train of a racing motorbike engine by means of a combined lumped/finite element model",DIEM—Department of Mechanical Engineering, University of Bologna, 21,2007, pp.735–760.

vi H. Heisler, "Advanced Engine Technology," Butterworth-Heinmann, Oxford, 2002,2nd Edition,pp 101-108.

vii M. Husselman, "Modelling and Verification of Valve Train Dynamics in Engines," M.Sc. Thesis, Stellenbosch University, Stellenbosch, 2005, pp212-223.

International Journal of Engineering Studies and Technical Approach
ISSN No. 2395-0900

viii R. L. Norton and R. G. Mosier, “Cam Design and Manu-facturing Handbook,” Industrial Press, Inc., New York, 2002.pp 98-112.

ix J. W. David, C. Y. Cheng, T. D. Choi, C.T. Kelley and J. Gablonsky, “Optimal Design of High Speed Mechanical Systems,” North Carolina State University, Raleigh, Tech.Rep. CRSC-TR97-18, 1997,pp 84-97.

x Tounsi M., Chaari F., Walha L., Fakhfakh T., Haddar M, “Dynamic Behavior Of A Valve Train System In Presence Of Camshaft Errors,” Wseas Transactions On Applied And Theoretical Mechanics, Vol.No. 6, 2011, pp. 17-26.

xi A. Cardona, E. Lens and N. Nigro, “Optimal Design of Cams,” Multibody System Dynamics, Vol. 7, No. 3, 2002, pp. 285-305.

xii W. J. Kim, H. S. Jeon and Y. S. Park, “Contact Force Prediction and Experimental Verification on an OHC Fin-ger-follower type Cam valve System,” Experimental Me-chanics, Vol. 31, No. 2, 1991, pp. 150-156.

xiii H.S Jeon, K. J. Park and Y.-S. Park, “An Optimal Cam Profile Design Considering Dynamic Characteristics of a Cam valve System,” Experimental Mechanics, Vol. 29, No. 4, 1989, pp. 357-363.

xiv M. Teodorescu, M. Kushwaha, H. Rahnejat and D. Taraza, “Elastodynamic transient analysis of a four-cylinder valve train system with camshaft flexibility,” Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-Body Dynamics, Vol. 219, No. 1, 2005, pp. 13-25.

xv M. R. M. Rejab, M. M. Rahman, Z. Hamedon, M. S. M. Sani, M. M. Noor and K. Kadirgama, “An Evaluation of Profiles for Disk Cams with In-line Roller Followers,” Proceedings of MSTC08, Kuala Lumpur City Centre, Malaysia, 16-17 December 2008, p.p.56-68.

xvi Faculty of Mechanical Engineering, “Design Data Book Of Engineers", Kalaikathir Achchagam, 2011, 7.110-7.116.

xvii R. L. Norton and R. G. Mosier, “Cam Design and Manufacturing Handbook,” Industrial Press, Inc., New York, 2002.

xviii W. J. Kim, H. S. Jeon and Y. S. Park, “Contact Force Prediction and Experimental Verification on an OHC Finger-follower type Cam valve System,” Experimental Mechanics, Vol. 31, No. 2, 1991, pp. 150-156.