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Steering Rod Fatigue Test Bench Cam Loading Analysis

Zhong-xing Yang*,[a], Yun-fei Mai[a]

University of Shanghai for Science and Technology, Shanghai, China

Abstract

Steering rods are subject to variety kinds of loads which will eventually cause failure. Automobile manufacturers and engineers are seeking ways to test the fatigue life of the steering rods. Steering rods under periodic loads over millions of cycles will get fatigued. We here provide a fatigue test bench for steering rods which applies periodic loads to the rod with a cam loading system. Cycloid motion is chosen for the cam follower to achieve the smooth and stable kinematic performance in middle and high speed operation. The accuracy of the follower response is analyzed using dimensionless dynamic equation and the application of Matlab is introduced to solve elastodynamic equation and analyze the prime/residual vibration curve of the follower. The cam loading system is built in dynamic analysis software Adams to simulate its virtual operation performance. By calculating the velocity multiplied by acceleration of the follower over the operation period, the extent to which the inertia load from the follower affects the torque on the cam shaft will be obtained with the help of Adams/postprocessor. The force loads acting on the cam shaft can also be acquired using simulation tools. By virtual prototyping the cam loading system's performance can be foreseen before a real prototype is built. Introducing cam loading method into fatigue test is an innovative application.

Keywords: Steering rod fatigue test; Dimensionless dynamic equation; Prime/residual vibration; Inertia load; Virtual prototyping

1. Introduction

Vehicle steering rod is a rod that allows the driver to steer a vehicle by turning the steering wheel. The torque applied by the driver is taken through the steering rod to the actuator, either with a hydraulic power steering (HPS) or an electric power steering (EPS), which will eventually perform the steering action of the road wheels. Because of impacts caused by terrain conditions or loads from the road and rotation of the rod itself, the steering rod will get fatigued after millions of load cycles which will prevent it from acting swiftly and accurately and finally cause it to fail. It is necessary to test the fatigue life of a steering rod.

In a fatigue test, a rotating rod will be held on both its two ends whilst undergoing a constant periodic load acing on the body of the rod which causes periodic deformation. It is required that the body of the rod bends in opposite directions in turn over the operation time until it gradually fails under constant amplitude loads. A loading force is exerted on the rod which varies from minimum to maximum periodically whilst the rod is rotating.

Hydraulic loads can be applied as the loading force and controlled with high flexibility, however the precise action is hard to achieve because of the delay in response and the oil also has its resilience. Hydraulic loads do have their benefits in flexible programmable loading applications, but the main defect is that hydraulic loads will be affected by surplus force happening when there are deviations or interferences in signals then the actual loading curve differs largely from the desired loading force especially in high speed loading operation. Hydraulic can not apply precise action at high speed. And in some cases we will need to devise ways to eliminate the sharp cusps in its loading curve. We here discuss the use of a cam-follower mechanism in test load application.

Cams are widely used in machining centers and packaging industries where precise action and swift response is required. The loads applied in a fatigue test are standardized and are basic constant amplitude loads [1]. It is both suitable and economic to adopt a cam loading system for the test. Due to the simple structure of the loading system, the mass production cost and the weight of the whole machine are greatly reduced.

High stiffness spring can provide a heavy load which is an alternative way to replace the hydraulic loads. A cam in the loading system can convert the displacement of its follower into a load when a spring is compressed. By choosing the exact stiffness coefficient of the loading spring and the contour of the cam, a force curve against cam revolution angle will be obtained. The frequency of the load peaks is determined by adjusting the revolution speed of the cam shaft and that of the tested rod.

The systematic sketch of the test bench is shown in Figure 1 where a cam applies a load on the rotating steering rod through the loading tip that is pressed against the rod by the compressed spring. Analog signals will be collected by the sensor and sent to the signal processor where analog signals are converted to digital signals and finally the digital signals will be sent to the computer and analyzed. In this paper, we discuss the dynamic performance of the cam-follower loading system.

Because the cam shaft and the tested rod can turn at different speeds, in this case the revolution speed of the cam shaft is twice that of the rod, when the rod completes a turn of 360 degrees it receives the maximum loading

* Corresponding author. e-mail: fivemiles555@163.com

force from the cam loading system twice which act on the opposite cylindrical side of the rod. The loading force curve is a set of pulses with each pulse rising up reaching its peak and then falling, in a same rate of its rising, touching the bottom.

After a physical prototype of test bench is built, when required loads are applied periodically to the rotating rod, sensors mounted on the rod body can measure the deformation of the rod body and physical features of interest can be acquired. With a combination of sensors, analog-digital signal converter and processer, the data will be sent to the computer and analyzed.

We are able to test different rods on the bench and loading force can be magnified by using springs with a higher stiffness. A set of cams with different contours can be designed in preparation for other commonly practiced test curves. It is recommended to apply B-splines method to design the contour of a cam when the loading curve is unique. B-splines method can optimize the follower's motion by minimizing the peak velocity and acceleration and also the jerk by selecting key points which are liable to provide an improved follower motion. Coupled with CAD/CAM and CNC, the cam profile is easy to be designed and manufactured. The way to obtain a cam profile with improved motion is to pick key points in its velocity curve, then calculate the displacement curve by integrating the velocity curve and the acceleration or jerk curve by differentiating. When the basic motion is chosen for the follower, the curve of the cam outline can be interpolated with 3 degree polynomial functions to ensure smoothness of the curve. [2]

By switching cams or springs, we acquire a variety of cam-spring matches that can apply different loads for different test tasks. Another option for applying unique loads is to employ non-linear stiffness springs each with a required load-displacement curve. The smoothness of the follower's motion depends on the contour of the cam which is optimized by picking key points in velocity curve. If we change springs instead of the cam to match different loading curve, this can ensure equal smoothness of the follower motion for different loading curve. However a non-linear spring will greatly increase the development and production cost. Moreover the loading curves for fatigue test are generally basic periodic curves, so it is not necessary to apply special loading curves.

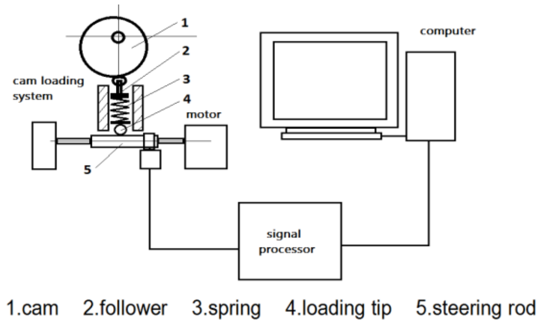

Figure 1: Test bench layout.

For linear spring, the loading force is in proportion to the follower's displacement.

The cam contour is to be designed in consideration of the dynamic performance of the follower to guarantee accurate response and also to avoid follower lift off. Kinematic features such as velocity, acceleration and the product of the two all need to be monitored because in certain applications the inertia load of the follower will have an impact on the cam shaft torque [3]. And the forces acting on the cam shaft also need to be measured for safety operation and service life. Hopefully these calculations can be done conveniently with the help of CAE and Numerical tools.

2. Cam Profile Design

The cam profile is designed according to the constant amplitude loading curve. The profile of the cam should prevent the reciprocating follower from suffering abrupt changes in velocity or acceleration. For high speed application, higher derivative of the displacement will need to be examined to prevent flexible impacts. The cycloid motion is suitable for mediate or high speed application, because it has continuous velocity and acceleration. In resent research, the third derivative of the displacement, known as jerk, is also checked for continuity when cam speed is high. Cycloid curve has many advantages in terms of less noise, vibration and wear, so cycloid curve is always chosen for high speed operation. [2]

The structure of the transmission mechanism should be as compact as possible while ensuring robustness thus saving room for other components.

On an actuate phase, pushed by the outline of the cam, the follower extrudes from its original stationary position while pressing back a spring mounted on it until reaching the farthest position in its motion. The displacement *S* of the follower with respect to the cam's angular displacement is given by Eq. (1).

$$s = \frac{h}{\phi}\varphi - \frac{h}{2\pi}\sin(\frac{2\pi}{\phi}\varphi) \qquad (1)$$

Where φ represents the angular displacement of the cam; ϕ is a constant representing the actuate angle of the cam; h is the displacement made by the follower from its nearest position to the farthest position.

On a return phase, the follower is pushed by a force from the compressed spring until it returns to its original position. Then the spring resumes its original length. The displacement of the follower is given by Eq. (2).

$$s = h - \frac{h}{\phi'}\varphi + \frac{h}{2\pi}\sin(\frac{2\pi}{\phi'}\varphi) \qquad (2)$$

Where ϕ' is a constant representing the return angle of the cam.

Figure 2 shows the follower's reciprocating displacement with respect to cam's angular displacement.

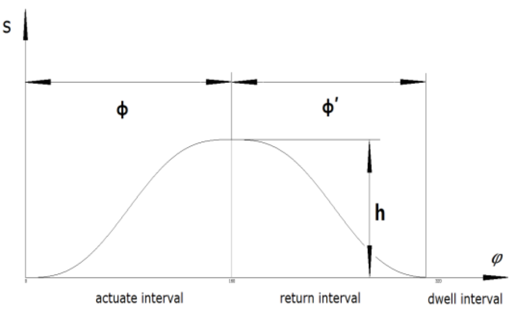

Figure 2: The follower's displacement curve.

By solving the displacement equation for higher derivatives, the follower's dynamic performance at high speed is obtained. B-splines methodology is able to plot the cam outline by polynomial interpolation and also facilitates manufacturing the cam profile with computer numerical control (CNC).

After the actuate length h is determined, the stiffness of the spring can be defined with the maximum load force value divided by h.

3. Elastodynamic Analysis

It is considered that all parts composing the mechanism are resilient and are deflected under loads.

So the follower can be seen as a high stiffness object. If the follower is compressed by high elastic load when accelerated, the deflection produces energy which is stored in the follower, and it will later release substantial energy to cancel out the spring force [4]. This excitation from the load causes transient chaotic motion in the follower and is the reason for the follower's vibration which will last till the end of the actuating period. It is necessary to evaluate the follower's response to the external excitation when the operation speed is high. The dynamic model must be built for analysis.

Within the cam loading model, the follower and the loading tip are respectively defined as objects with masses equal to m_1 and m_2 . The follower receives a displacement excitation s at its upper end where it contacts the surface of the cam with contact stiffness k_1 ; the displacement response of the follower is y_1 measured at the other end of the follower. The follower is connected to the loading tip through a spring with stiffness k_s and preload F_p . The displacement response of the loading tip is y_2 . Steering rod has revolute constraints at both of its ends and is seen as a body with one resolute degree of freedom about its longitude axis. The contact stiffness between the loading tip and the steering rod is k_2 . We obtain the following Eq. (3).

$$\begin{aligned} m_1\ddot{y}_1 &= k_1(s-y_1)-k_s(y_1-y_2)-F_p \\ m_2\ddot{y}_2 &= F_p+k_s(y_1-y_2)-k_2y_2 \end{aligned} \tag{3}$$

Equation (3) is based on the dynamic model (see Figure 3.a) which considers vibration in the loading tip. However the contact stiffness k_2 is much higher than the spring stiffness k_s and there is no external excitation from the steering rod. For simplification, we can remove m_2 from the model and focus mainly on the response of the follower (see Figure 3.b). The modified equation is given as below.

$$m_1\ddot{y}_1 = k_1(s-y_1)-F_p-k_sy_1 \tag{3}$$

Because F_p is only considered for static deformation and we also notice that $k_1 >> k_s$, Equation. (4) is further simplified as below.

$$m_1\ddot{y}_1 + k_1y_1 = k_1s \tag{4}$$

The contact stiffness k_1 is defined as the ratio of contact force to the follower's deformation. As the follower can be made from a variety of materials and designed into different shapes, the contact stiffness is a natural attribute once the cam and the follower are designed. By finite element analysis (FEA), the contact stiffness can be calculated.

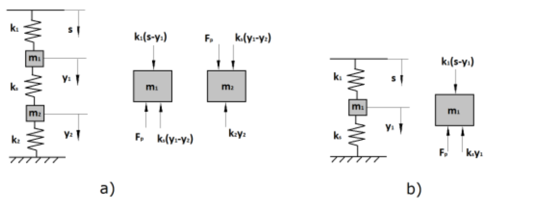

Figure 3: Dynamic model.

On the actuate phase, the follower is excited by the contact force and responses to that force with a displacement in downward direction. This is the prime vibration period of the loading operation and will last until the follower reaches the lowest position of its motion.

A dimensionless equation can be used to compare the follower's dynamic responses with different motion periods regardless of the particular features of a cam loading system [3]. It is assumed that displacement and time are expressed as unities.

With $Y=\frac{y_1}{h}$, $T=\frac{t}{t_h}$, $S=\frac{s}{h}$, $\ddot{y}_1=\frac{d^2y_1}{dt^2}=\frac{t}{{t_h}^2}Y''$, and the periodic features $\omega_n=\sqrt{\frac{k_1}{m_1}}$, $t_0=\frac{2\pi}{\omega_n}$, $\lambda=\frac{t_h}{t_0}$, where ω_n is the natural frequency of the follower and t_h represents the length of time of the actuate phase, we change Eq.(5) into a dimensionless Eq.(6) about the period ratio λ .

$$Y''+(2\pi\lambda)^2Y=(2\pi\lambda)^2S \tag{6}$$

If substituting the actual displacement equation into Eq.(6), the response of the follower for prime vibration period is expressed as below.

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$$Y'' + (2\pi\lambda)^2 Y = (2\pi\lambda)^2 (T - \frac{1}{2\pi}\sin 2\pi T) \qquad (7)$$

When the cam shaft stops rotating, there is no excitation acting on the follower. The residual vibration period is on until the remaining energy in the follower dissipates out for example to be absorbed by the damping or friction. The dynamic equation to describe this period is given by Eq. (8).

$$m_1 \ddot{y}_r + k y_r = 0 \qquad (8)$$

To change it into a dimensionless equation, we obtain Eq. (9).

$$Y_r'' + (2\pi\lambda)^2 Y_r = 0 \qquad (9)$$

The dimensionless displacement Eq. (7) and Eq. (9) are to be solved in conjunction with the initial displacement and velocity conditions. The initial definition of velocity and acceleration need to be described according to practical conditions.

4. Inertia Load Analysis

In high cam speed operation, the follower is moving with high acceleration and velocity. The follower, the reciprocating part of the cam loading system produces inertia load on the cam shaft. Factors like the mass of the follower and the revolution speed of the cam shaft are all responsible for the inertia load. If the follower has a large mass and the operation speed is high, the inertia load will play a significant part in the cam's dynamic load. It is appreciated that the inertia load should be limited to a relatively small value compared to the working load. The magnitude of the inertia load can be deduced from a force diagram of the cam loading system (see Figure 4).

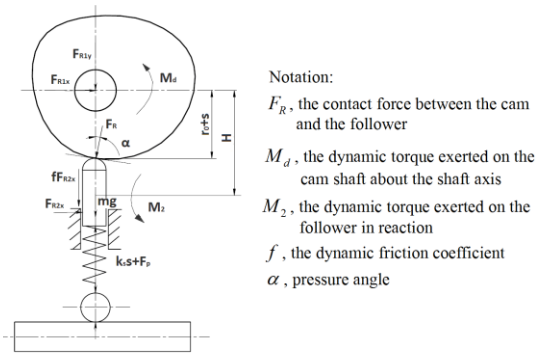

Figure 4: Force diagram.

For the follower, the equilibrium equation is expressed as:

$$\begin{aligned} &-m\ddot{s} + k_s s + F_p - \delta f F_{R2x} - mg - F_R \cos\alpha = 0 \\ &F_{R2x} - F_R \sin\alpha = 0 \\ &F_R (r_0 + s)\sin\alpha - F_{R2x} H - M_2 = 0 \end{aligned} \qquad (10)$$

Where an item with subscript x implies the lateral component of the item and pressure angle is,

$$\alpha = \tan^{-1}\left[\frac{\dot{s}}{(r_0 + s)\omega}\right] \qquad (11)$$

To distinguish the direction of the friction force, it is assumed that If $\dot{s} > 0$, $\delta = -1$; if $\dot{s} < 0$, $\delta = 1$. The first derivative of s indicates the velocity of the follower and positive or negative value of which implies the direction of the follower's motion.

For the cam, the equilibrium equation is:

$$\begin{aligned} &F_{R1y} - F_R \cos\alpha = 0 \\ &F_{R1x} + F_R \sin\alpha = 0 \\ &M_d + F_R (r_0 + s)\sin\alpha = 0 \end{aligned} \qquad (12)$$

Where item with subscript y implies the vertical component of the item. Combining Eq. (11) and Eq. (12), the contact force F_R and the cam shaft dynamic moment load M_d are given,

$$\begin{aligned} &F_R = \frac{k_s s + F_p - mg - m\ddot{s}}{\delta f \sin\alpha + \cos\alpha} \\ &M_d = \frac{\dot{s}}{\omega}\,\frac{mg + m\ddot{s} - F_p - k_s s}{\delta f \tan\alpha + 1} \end{aligned} \qquad (13)$$

By inspection of Eq. (13), the product of the follower's velocity and acceleration partially determines the follower's inertia load on the cam shaft torque. In designing the cam loading system, the mass of the follower, the velocity and acceleration of the follower must be carefully monitored or this will cause a fluctuation in the cam shaft torque load, even more so when the cam revolution speed is high.

5. Example of a Loading Practice

The fatigue life tests are practiced based on the loads data recorded from real road running test. The loading frequency should be higher than 5 Hz or it will have no effect on the steering fatigue life while kept away from the natural frequency of the steering rod to prevent resonant vibration. According to the data collected from the road, only load frequencies higher than 6 Hz will be applied in fatigue test [5]. Here we choose a loading frequency of 10 Hz and loading amplitude of 5000 N. Because the rod natural frequency is estimated at 50 Hz, so the loading frequency will not cause resonant vibration.

With the application of numerical and dynamic software, the loading performance can be tested virtually. As is already explained in section 2, the displacement response Eq. (7) and Eq. (9) indicate the follower's responses during prime vibration phase and residual vibration phase. The differential equations are solved using Matlab tool. Because the response behavior varies according to the period ratio λ, different response curves need to be plotted out for comparison with the theoretical response curve. The initial conditions are specified that the velocity and acceleration are both zeros at the beginning of the practice. A table is shown below indicating the parameters defined for this practice. (See section 2 for the equations of these parameters).

Table 1: Practice parameters

Items	Value
Actuate length h	$0.05m$
Loading spring stiffness k_s	$1.0\times10^5 N/m$
Peak loading force $k_s \times h$	$5.0\times10^3 N$
Actuate angle ϕ	$2.79rad$
Cam shaft speed n	$600RPM$
Contact stiffness k_1	$22,727,300N/m$
Natural frequency ω_n	$3,866.8rad/\sec$
Actuate phase period t_h	$4.4\times10^{-2}\sec$
Natural period t_0	$1.625\times10^{-3}\sec$
Period ratio λ	27

5.1. Matlab solution:

The Eq. (7) and Eq. (9) with their initial conditions are expressed in Matlab respectively as below,

y=dsolve('D2y+…*y=…*(t-sin(…)','y(0)=0','Dy(0)=0','t');

y=dsolve('D2y+…*y=0','y(1)=1','Dy(1)=0','t').

The theoretical response curve is y=t-sin(2*pi*t)/(2*pi).

The displacement curves are plotted as shown in Figure 5 (Figure 5.a represents the prime vibration period and Figure 5.b represents the residual vibration period.). By inspection of figure 5.a, we conclude that the response curve of practice loading (line 2) coincides with the theoretical response curve (line 1). For comparison, when period ratio is equal to 1, the response curve (line 3) is significantly biased from the theoretical one (line 1).

The residual vibration curve has a pattern which resembles a line of leaf shaped patches when the period ratio gets high. (see figure 5.c).

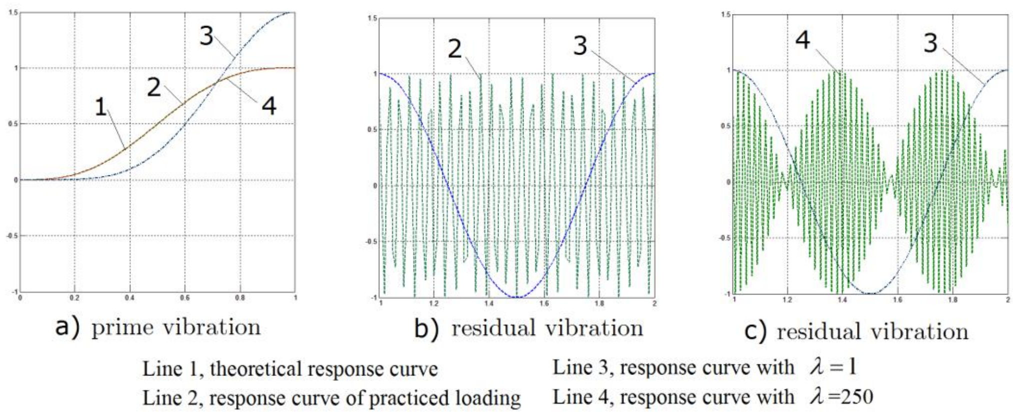

Figure 5: Displacement response curves.

5.2. Adams simulation:

5.2.1. Cam profile modeling with B-splines polynomial interpolation:

Basically, a contour of the cam requires at least 6 key points in either the actuate interval and return interval to manipulate the outline of the cam. The smooth curve to connect the key points is formed using B-splines interpolation. The polynomial interpolation equations have 3 degrees.

An arbitrary point on an interpolated curve which has (k-1=3) degrees and is manipulated with (n+1) key points can be presented as below. [2]

$$P(u) = \sum_{i=0}^{n} N_{i,k}(u)P_i \qquad (14)$$

Where u is the length of interpolation, the range of which is u (0, n-k+2); $N_{i,k}(u)$ is the interpolation function; P_i is a key point. (for more details of Eq. (14) and B-splines interpolation method see reference [2])

Adams models the cam with B-splines interpolation as key points are picked in the path of the outline. This ensures the smoothness of the curve and also facilitates manufacturing with computer aided manufacturing tools. The possible optimization can be done by re-picking the key points.

Table 2 lists the key points which compose the outline of the cam at an interval of 10 degrees starting from 0 to 360 degree. Figure 6 shows the final design of the cam profile.

Table 2: Cam profile key points.

Angular displacement (°)	Radial distance from the cam basic circle (mm)	Angular displacement (°)	Radial distance from the cam basic circle (mm)
0	0	170	49.9203
10	0.0797	180	49.377
20	0.623	190	47.977
30	2.023	200	45.4577
40	4.5423	210	41.727
50	8.273	220	36.877
60	13.123	230	31.1703
70	18.8297	240	25
80	25	250	18.8297
90	31.1703	260	13.123
100	36.877	270	8.273
110	41.727	280	4.5423
120	45.4577	290	2.023
130	47.977	300	0.623
140	49.377	310	0.0797
150	49.9203	320	0
160	50	320-360	0

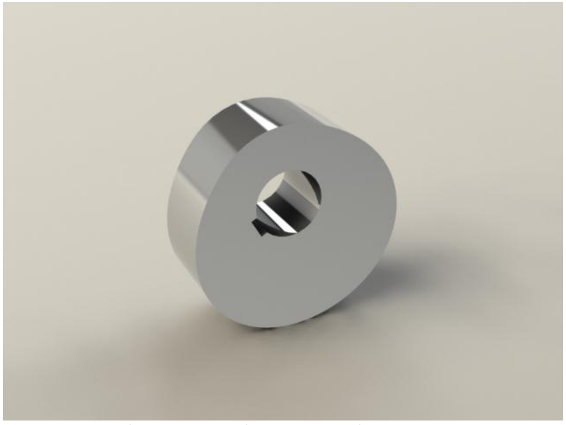

Figure 6: The final design of the cam profile.

5.2.2. Dynamic simulation:

The loading operation can be simulated in Adams to foresee its real performance. The preciseness of the simulation results relies on the correct definition of the behavior and conditions of the loading model.

A dynamic model is built as figure 7 a) shows, where the motions are defined for the cam shaft and steering rod. It is highly recommended to define the contact force as the relationship between the cam's outline and the follower's top end because if the relationship between them is defined as 2D curve-curve constraint type, the lift-off will not be simulated. All the parameters should be set as table 1 shown. A preload of 100N is applied to eliminate the tiny clearance between the cam and the follower.

The extent to which the follower's inertia load can affect the cam shaft torque is obtained by multiplying the acceleration of the follower by its velocity as discussed in Eq. (13). The product of the two characteristics is plotted by using Adams/postprocessor. As shown in figure 8 a), once the simulation is run, we can plot the velocity curve and acceleration curve, and by adding those two plots in one sheet a curve representing the velocity multiplied by acceleration over a same period of time can be plotted using Adams/postprocessor. The time when the maximum point of the product occurs and its specific value can be measured by tracking the plot. Due to the cam profile and the operation speed chosen for the simulation, the maximum value of the product is measured at 220 m^2/sec^3 , however when devided by the revolution speed and multiplied by the mass of the follower the resulting moment value is 5.25 Nm, therefore the cam shaft torque is not affected (see figure 8 b), because the peak torque measured in the loading operation is 100Nm which is much greater than 5.25 Nm caused by inertia load. In figure 8 b), the curve of the product of velocity and acceleration and the curve representing the torque are shown in one sheet. Torque curve implies the magnitude of pressure angle, for when the torque is zero the pressure angle is also zero and when the torque reaches its peak point the pressure angle also reaches its maximum value. Because the inertia torque produced by the follower is partially decided by the pressure angle, by analyzing both the product curve and the torque curve in one sheet can help to focus on the peak value in product curve which coincides with the peak torque value. When two peak values happen at the same time or very near in time, the inertia torque has its greatest effect on the operation torque. In this loading case, the effect is too limited to alter the operation torque.

If in any case the inertia load is large enough to affect the shaft torque, the speed need to be decided again. In loading operation where we have several pairs of cam-spring matches previously prepared, if the operation torque exceeds the designed torque for the shaft, this is the time to switch to a cam with a lower displacement and a matching spring with a higher stiffness. However there is no need adjusting the operation speed or switching to other cam-spring pair when the resulting torque exerted on the shaft is within the safety range. The safety loads for the cam shaft must be checked when designing the shaft. The shaft's mounting diameter and key applied to lock the cam on it should be selected based on the peak loads exerted on the body of the cam shaft. Virtual simulation facilitates the calculation of the critical load values.

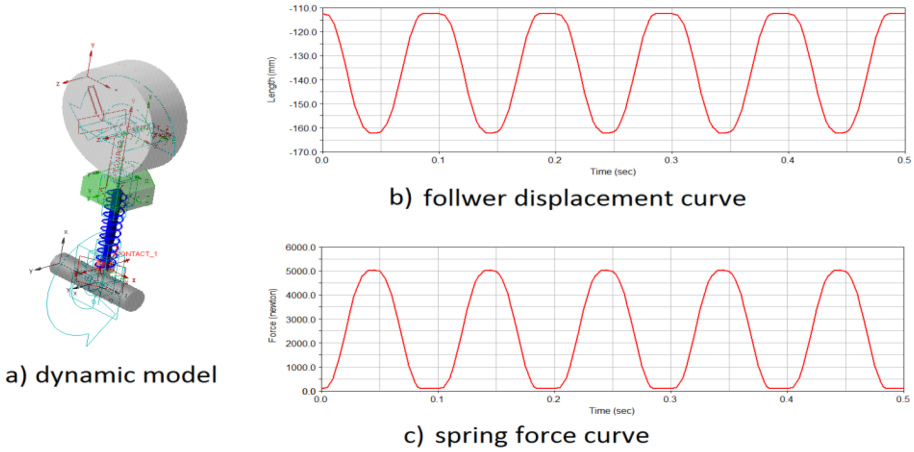

Figure 7: The Adams model of loading system.

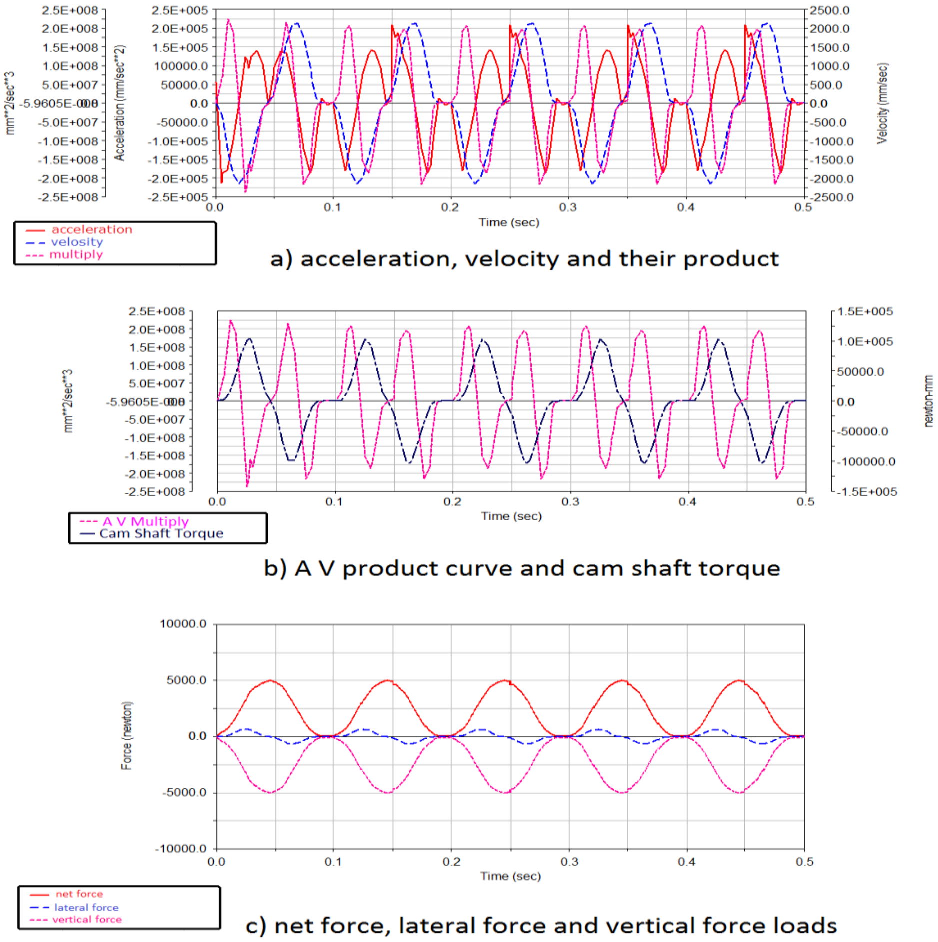

Figure 8: The product of the follower's acceleration and its velocity.

The force loads exerted on the cam shaft are given in figure 8 c). The value of a certain point on the curve can be measured with Adams/postprocessor. From figure 8 c), we can see that the vertical force forms a main part in net force load because the pressure angle (Eq. (11)) is small all the time during the loading practice. The force loads also need to be kept within safety range to prevent failure in cam shaft. Because the cam is locked on the cam shaft with a key, there is no relative rotary movement between the cam and the shaft, so the maximum force load on the shaft reside in a constant area.

6. Conclusion

The cam-spring loading system is introduced to apply periodic loads for fatigue test. With elasto-dynamic analysis and Matlab tools, the responses curves of the follower on its prime and residual vibration periods can be obtained. Based on the calculation of the system's loads in motion and Adams dynamic simulation, the performance of a loading practice can be evaluated. This introduces an alternative way to test the fatigue life of a steering rod by a cam-spring loading set instead of hydraulic means and may also apply to test other rod shaped components when the loading curve has a periodic feature.

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