Development of Vibration Calculation Code for Engine Valve-Train

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Abstract

A vibration calculation code for an engine valve-train was developed. This code can be used in a similar manner to Mitsubishi Motors Corporation (MMC)'s engine performance simulator[1] with which MMC engineers are very familiar. The code quickly predicts how the valve lift curve obtained by using the engine performance simulator affects the valve-train vibration. The calculation results help engineers to understand the valve-train vibration characteristic and to take appropriate countermeasures.

Valve-train vibration calculation is somewhat difficult because of nonlinear elements such as valve clearance, but this code ensures a stable calculation by employing finite difference calculus with the implicit method. The code also has some useful auxiliary software. For example, the data is input by mouse operation and the spring vibration can be observed as an animation. All operations can be done on Windows' applications.

Key words: *CAE, Vibration, Valve Train*

1. Introduction

Computer progress has lead to our engine performance simulator easily predicting the engine performance once inlet and exhaust systems and valve timing are defined. Although the best valve lift curve for good engine performance can be obtained, the lift curve sometimes can not be used because of the unfavorable valve-train condition. Recently the valve-train mechanism has been changing from a type of shifting intake and exhaust valve timing for bringing out better engine performance to another type of VVT (Variable Valve Timing) system which also shifts a valve lift continuously by using a multi-link system. And this evolution can not be stopped. Considering the trend, a vibration prediction technique is desired in order to obtain maximum valve-train performance.

A retro styled valve-train vibration analysis is one that the amplitude of vibration at each order is calculated with Fourier series expansion and the valve-train natural frequencies which are strongly influenced by the amplitude around high engine speeds are marked for designing. Nowadays MMC is trying a full FEM valve-train vibration calculation with commercially available LS-DYNA™ and a multi-dimensional vibration calculation with also commercially available ADAMS/Engine™[2]. But these codes take many hours to compose their input data, so these are really not design tools which can be used to calculate many cases. Our original valve-train vibration calculation code was therefore developed with the aim of being used like our engine performance simulator which MMC engineers are very familiar with. The overview and calculation examples are introduced in this report.

2. Overview of vibration calculation code for valve-train

2.1 Vibration calculation code for valve-train and auxiliary software

The vibration calculation code for valve-train and auxiliary software are shown in **Fig. 1**. In the operating environment, all calculations can be done on Windows' applications. The input models are made with MSNpre (④), the valve lift curves are made with ValveLift (③) and the calculations are run with MSnetwork (①). When making input data, FnSpring (②) is used to make the multi mass model for the valve spring. The calculation result graphs are quickly drawn in the window by using GraphMaker (⑤) to help analyze the phenomena without loss of time. Also, the spring motion can be observed in the animation by using MSNanime (⑥). This series of software is completely made by MMC.

In addition, MSNpre enables the user to compose a one dimensional vibration system model interactively with mouse and button operations. As an element representing mass, the fixed point or lifted point is selected with a button and allocated on the sheet of the software. A spring element is defined by the mouse drag from the knob of an element to the knob of another one. The input data can be provided for the element with the dialog which is opened by clicking the mouse right button.

ValveLift, a share code with our engine performance simulator, generates a polynomial valve lift curve, a combined sine and parabola, and a combined sine and oval.

MSNanime enables the display of the spring vibration on the screen from the calculation results about

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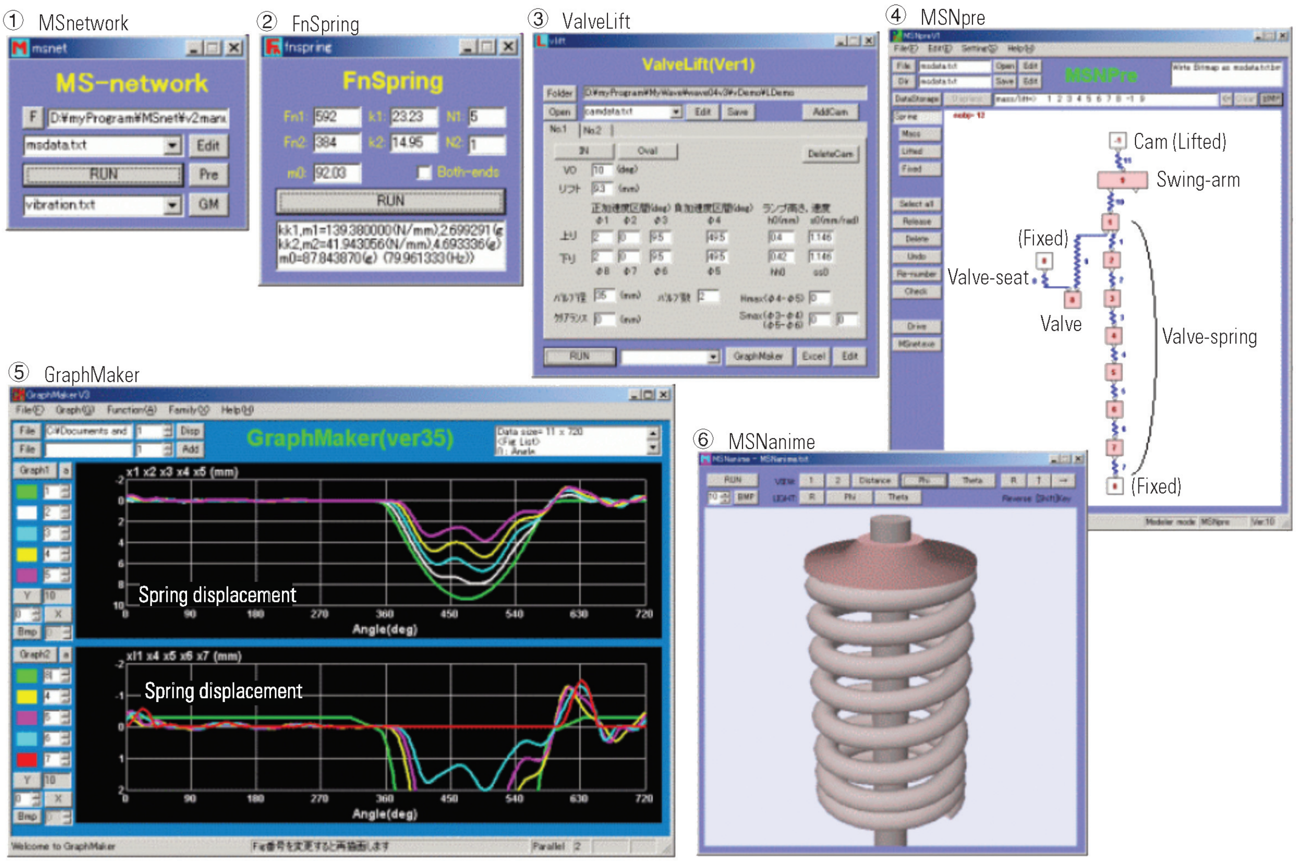


Fig. 1 Valve-train calculation code (MSnetwork) and auxiliary software

mass displacement with only providing the text style input file in which the spring descriptions such as wire diameter, coil diameter, the number of coils and so on are written. Because it has three dimensional display data in it, it can be observed from different view points and angles. In addition, it has a function making a Bitmap file from the picture on the screen at each crank angle previously defined. An animation is also made by compiling these Bitmap files.

2.2 Method of valve-train vibration calculation

2.2.1 One dimensional vibration model

This code handles a one dimensional vibration model. To be specific, the connection of the model is defined by way of describing what boundary element (such as mass, fixed or lifted) is placed at both ends of the spring in the input data. At first glance, a whole vibration model showed in **Fig. 2** looks like a quite simple spring-mass model, but there are some difficult phenomena peculiar to valve-trains from the discontinuities such as valve clearance, valve sitting on the sheet, the jump which is said that the tappet leaves from the cam phase during a valve lift period at high engine speed, and a spring wire collision which is when a portion of the spring wire collides with another portion at higher engine speed. In other words, the sudden spring constant switching happens everywhere in the model.

In addition, almost all valve springs have a portion of a so called close coil, a closer winded portion of the coil, in order to restrain its vibration. This portion is usually closed, but when the vibration becomes larger at the valve sitting timing, the portion turns to be open and switches the spring constant to change the natural frequency of the valve-train. The natural frequency of the valve-train used here means the value defined from the spring constant and the mass including a tappet, a retainer, a valve and some of a spring.

Therefore in valve-train vibration calculations, a suitable algorithm which can be running stably without divergence whenever such a sudden change of the spring constant happens is required to be chosen.

2.2.2 Algorithm of vibration calculation

A complete implicit method was adopted to ensure stability for the vibration calculation. This method uses an iteration loop in which the presumed mass displacement at current time is gradually adjusted until all displacement becomes a suitable value without discrepancy.

Taking a valve-train with two masses in **Fig. 2** for example, the dynamic equations around mass 1 and mass 2 are described as equations (1) and (2) respectively:

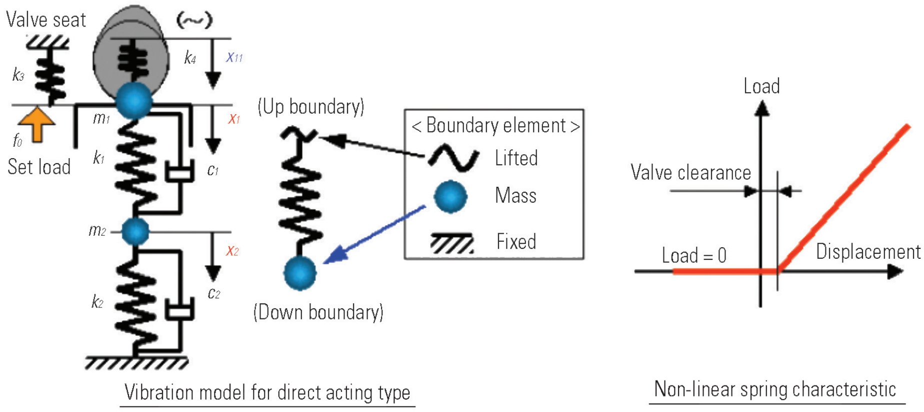


Fig. 2 Details of vibration calculation code for valve-train

$$m_1\frac{d^2x_1}{dt^2}=k_1(x_2-x_1)-k_3x_1+k_4(x_{11}-x_1)+c_1\left(\frac{dx_2}{dt}-\frac{dx_1}{dt}\right)-f_0 \tag{1}$$

$$m_2\frac{d^2x_2}{dt^2}=k_1(x_1-x_2)-k_2x_2+c_1\left(\frac{dx_1}{dt}-\frac{dx_2}{dt}\right)-c_2\frac{dx_2}{dt} \tag{2}$$

m: mass,
x : displacement,
k : spring constant,
c : damping coefficient,
t : time,
f_0 : load,
suffix: 1, 2, 11 show their ID-numbers.

Applying the 2nd order one side difference[3] at the time direction, its acceleration and velocity are expressed as equations (3) and (4) respectively.

$$\frac{d^2x}{dt^2}=\frac{2x^n-5x^{n-1}+4x^{n-2}-x^{n-3}}{dt^2} \tag{3}$$

$$\frac{dx}{dt}=\frac{3x^n-4x^{n-1}+x^{n-2}}{2dt} \tag{4}$$

The superscripts show time steps (n; current, n–1; 1 step before, n–2; 2 step before, n–3; 3 step before).

Substituting these equations (3) and (4) into equations (1) and (2), the m_1 and m_2 displacement can be showed as discrete equations (5) and (6) below.

$$x_1^n=\frac{\frac{m_1}{dt^2}\left(5x_1^{n-1}-4x_1^{n-2}+x_1^{n-3}\right)+k_1x_2^n+k_4x_{11}^n+\frac{3c_1}{2dt}\left(3x_2^n-4x_2^{n-1}+x_2^{n-2}+4x_1^{n-1}-x_1^{n-2}\right)-f_0}{\frac{2m_1}{dt^2}+k_1+k_3+k_4+\frac{3c_1}{2dt}} \tag{5}$$

$$x_2^n=\frac{\frac{m_2}{dt^2}\left(5x_2^{n-1}-4x_2^{n-2}+x_2^{n-3}\right)+k_1x_1^n+\frac{3c_1}{2dt}\left(3x_1^n-4x_1^{n-1}+x_1^{n-2}+4x_2^{n-1}-x_2^{n-2}\right)+\frac{3c_2}{2dt}\left(4x_2^{n-1}+x_2^{n-2}\right)}{\frac{2m_2}{dt^2}+k_1+k_2+\frac{3c_1}{2dt}+\frac{3c_2}{2dt}} \tag{6}$$

Now observing the current notation superscript n, it can be understood that x_1^n is solved with x_2^n and adversely x_2^n is solved with x_1^n. That means an iteration is required to find the solution where the x_1^n and x_2^n are compromised without discrepancy. But all displacement solved here is compromised without contradiction, so the calculation can be run with good stability whenever any spring constant change occurs suddenly.

Another merit is that the calculation time step is able to be fixed to one degree of the crank angle because the constraint of the calculation is not very strict. Although the time interval for actual calculations is extremely different through the engine speed, thanks to the implicit method the calculation can be run without any problems.

2.2.3 Validation of calculation

In a simple spring model with one mass, the mass is offset by certain distances and vibrated, so its damping is observed in order to validate the accuracy of the finite difference algorithm. The calculation results are posted in **Fig. 3 (a)**. The results of the highly accurate 4th order Runge-Kutta method has little damping for 0.1 second, and on the other hand, that of the 2nd order

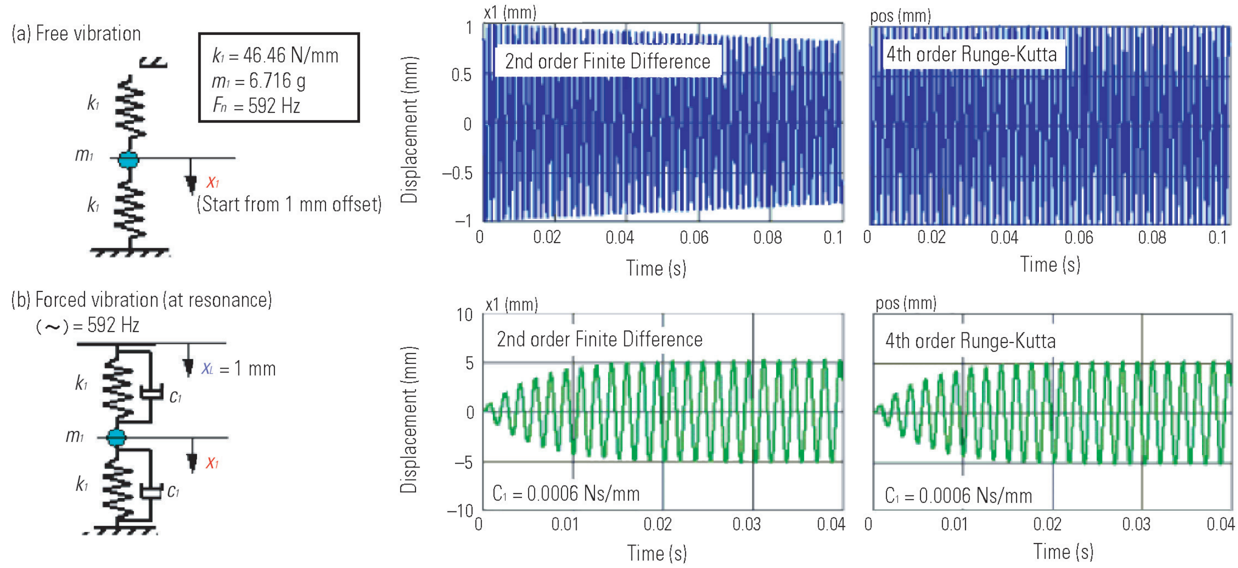


Fig. 3 Precision of calculus of finite difference with second-order accuracy

finite difference method has about 20 % damping of the amplitude during the same period because of the artificial viscosity generated from the residual error.

Fig. 3 (b) shows an example when the mass displacement is observed under a forced oscillation at the resonant frequency on the end of the spring. An adequate damping coefficient is provided here. It is found that the amplitude of the vibration of the 2nd order finite difference method is almost the same as that of the Runge-Kutta method under this condition. Since this level of the damping coefficient is usually provided on the spring element, it is considered that the 2nd order finite difference method will not create any problems in actual use.

The 1st order finite difference method can be also provided for a pre-calculation when the adequate damping coefficient is not sure. Using the 1st order method, the calculation is able to be run without setting the damping coefficient because the artificial viscosity made from the residual error acts adequately.

2.3 Multi-mass modeling for valve spring

Special software (FnSpring) is provided to make a multi-mass model for a valve spring. If only the spring constant, natural frequency and the number of masses are defined, the dispersed masses are solved in this software. And it also supports a spring with some close coils. This software also adjusts the valve-train mass (m_1 at **Fig. 4 (c)**) because the part of the spring mass included in its mass is slightly changed.

The dispersed masses are calculated through this procedure. A flexibility matrix is composed with influence coefficients, and the matrix is solved with the iterative solution method[4]. In case of dividing the spring showed in **Fig. 4 (a)** into some masses, the matrix and the influence coefficients are written bellow.

$$\begin{pmatrix} x_1' \\ x_2' \\ x_3' \end{pmatrix} = \omega^2 \begin{pmatrix} a_{11} & a_{12} & a_{13} \\ a_{21} & a_{22} & a_{23} \\ a_{31} & a_{32} & a_{33} \end{pmatrix} \begin{pmatrix} x_1 \\ x_2 \\ x_3 \end{pmatrix} \tag{7}$$

$$a_{11} = \frac{m_1}{k_1 + 1/\left(1/k_2 + 1/k_3 + 1/k_4\right)} = \frac{3m}{4k} \tag{8}$$

$$a_{21} = a_{11}\frac{1/\left(1/k_2 + 1/k_3 + 1/k_4\right)}{1/\left(1/k_3 + 1/k_4\right)} = \frac{2}{3}a_{11} \tag{9}$$

$$a_{31} = a_{11}\frac{1/\left(1/k_2 + 1/k_3 + 1/k_4\right)}{k_4} = \frac{1}{3}a_{11} \tag{10}$$

$$a_{22} = \frac{m_2}{1/\left(1/k_1 + 1/k_2\right) + 1/\left(1/k_3 + 1/k_4\right)} = \frac{m}{k} \tag{11}$$

$$a_{12} = a_{22}\frac{1/\left(1/k_1 + 1/k_2\right)}{k_1} = \frac{1}{2}a_{22} \tag{12}$$

$$a_{32} = a_{22}\frac{1/\left(1/k_3 + 1/k_4\right)}{k_4} = \frac{1}{2}a_{22} \tag{13}$$

Here the influence coefficient a_{ij} represents the affection of how the displacement of the mass point i (x_i) is influenced by the mass m_j. For instance, a_{21} shows how strong the mass m_1 influences the displace-

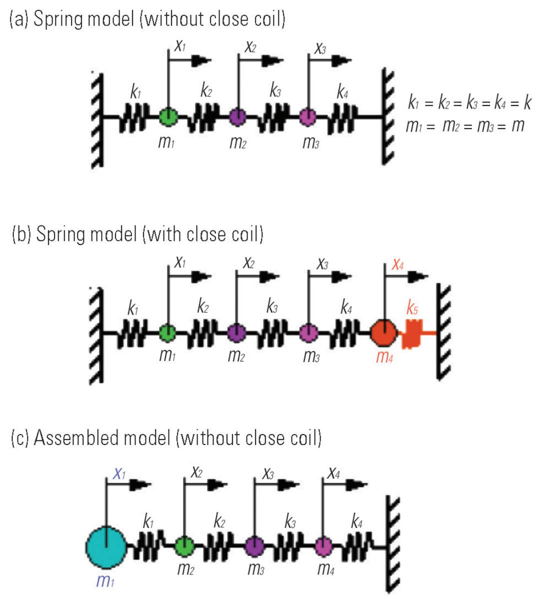


Fig. 4 Multi-mass calculation model for valve spring

ment of the mass point 2 (x_2).

ω is the natural angular frequency. The coefficients $a_{13} - a_{33}$ are omitted because they are the similar equations as $a_{11} - a_{31}$.

The iterative solution method carries out the following procedure. At first the matrix of equation (7) is composed after the influence coefficients are made. Then adequate initial values set in $x_1 - x_3$, and $x_1' - x_3'$ are solved with the matrix. After that, this calculation replacing $x_1' - x_3'$ with $x_1 - x_3$ is repeated until both will be the same value.

In this case, since the targeted natural frequency is sought through modifying mass m, the tentative values placed initially are modified repeatedly through the above procedure until the target natural frequency will be obtained.

The mass (m_4 at **Fig. 4 (b)**) in the close coil and the adjusted valve-train mass(m_1 at **Fig. 4 (c)**) are solved in the same way with the mass solved here.

3. Examples of calculation

3.1 In case of direct acting valve-train

3.1.1 Valve-train vibration at maximum engine speed

A calculation example of a so called direct acting valve-train which directly pushes the tappet by the cam and opens the valve is shown here. This vibration model has a valve spring with six masses including the close coil, and the spring constants of the cam and the valve sheet with the valve clearance between them. The valve spring is also given the set load (f_0).

Fig. 5 shows an example around the limited engine speed (7,500 min^{-1}). A lift with the clearance generating an external force is set at the cam. The valve-train load, the valve acceleration multiplied by the valve-train mass, is designed to have enough margin against the spring static load. But because the spring is excited well at this speed, it is a concern that some periods when the spring force can not sustain the valve-train load will emerge above this engine speed. A close coil (k_7) placed at the bottom of the spring makes the portion contact and non-contact repeatedly during the valve seating, and reduces the spring vibration. Some impact force is observed when the spring coils contact each other. And it is also observed that the spring loads (k_1, k_7) at both ends are vibrating by the opposite phase.

3.1.2 Valve jump simulation

A so called jump happens when the engine is driven over a certain engine speed. This means the phenomenon that a tappet leaves the cam-face at the timing when the spring no longer can sustain the valve-train load.

The valve-train behaviour at the start of jumping is showed in **Fig. 6**. The calculation conditions are the same as those of **Fig. 5**. The distance between the cam lift and the valve lift (displacement for m_1) is uncertain at the top figure in which the both are superimposed. This is enlarged in the figure below. It is seen here that two small jumps happen around the top lift. These jumps occur at the timing when the valve-train load can not be sustained by the spring force because of its vibration. The jump can be clearly simulated because the spring element of the cam has a constraint not to generate force, or to switch the spring constant into zero, at tension. Look at the cam load (k_9). It is observed that some impact loads happen just at the moment that the jumping tappet returns to the cam-face. And it is seen from the enlarged valve lift curve that the bounce, although tiny, happens and it is gradually reducing. The bounce means the phenomenon that the valve sits on the seat with velocity (ramp velocity), and if the impact force can not be absorbed around the seat, the valve rebounds.

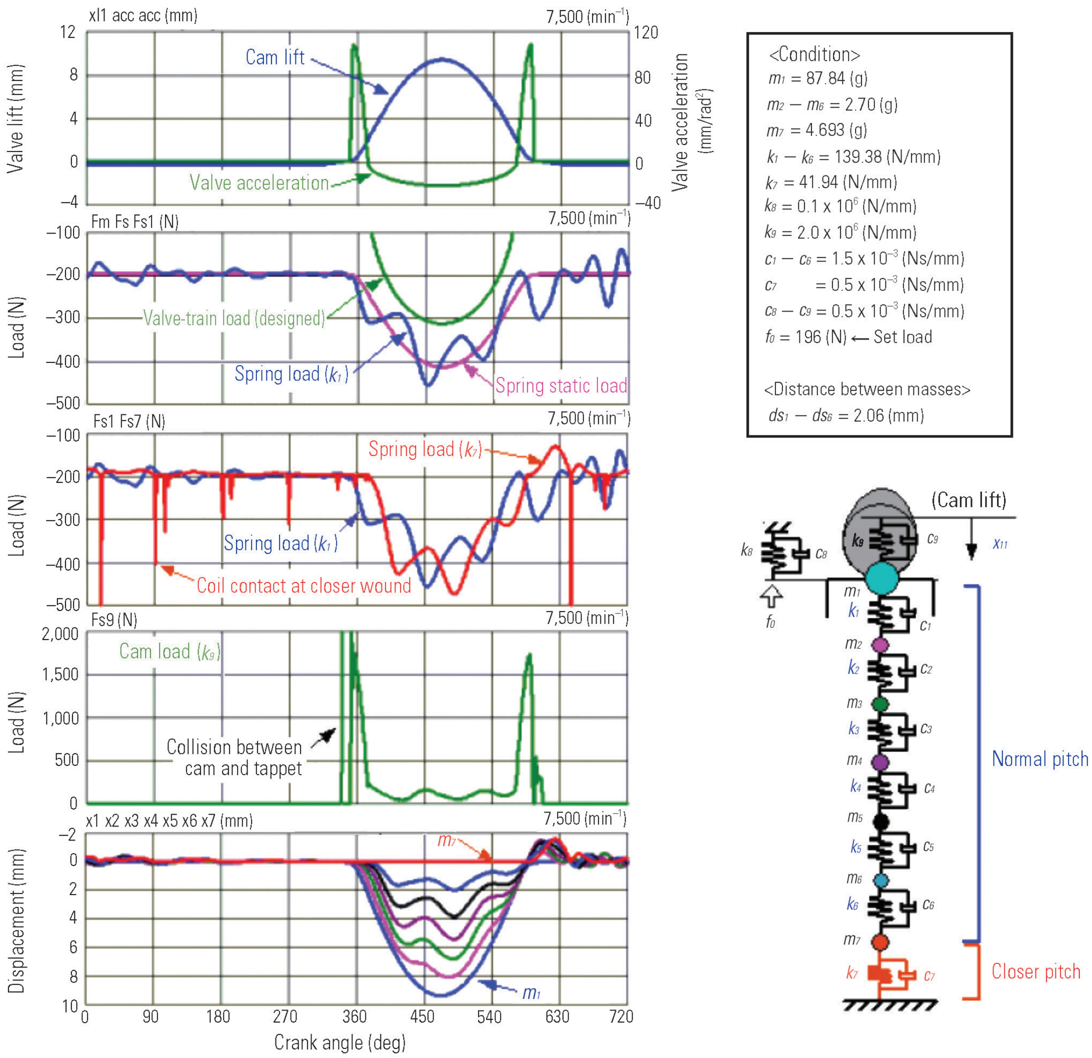


Fig. 5 Example of direct-acting valve-train calculation (7,500 min^{-1})

3.1.3 Spring coil contact simulation

When the engine speed is increasing, the spring force is not able to sustain the valve-train load anymore and a big jump which is much bigger than the cam-lift curve occurs as shown in **Fig. 7**. If it comes, a big bounce also occurs because the valve is not able to move according to the designed valve lift and the valve is sitting on the seat at a much higher speed than the ramp velocity. This bounce is gradually damped but continues during the valve seated period.

If such a big jump happens, the increased valve lift presses the spring more than expected, correlating with the inner spring vibration, and spring coil contact happens, which means some portions of the coil contact with each other everywhere.

Fig. 7 shows that some coil distances reach zero and the impact force occurs by the coil contact.

Switching to the large spring constant is carried out when the distance between the masses is smaller than defined in order to simulate the coil contact.

3.2 In case of valve-train with swing arm

This software has a unique function that handles an element with a rocker ratio like a swing arm. A calculation example of a valve-train with a swing arm whose rocker ratio is 1.4 (constant) is shown in **Fig. 8**. This vibration model includes the valve stem as a spring element and the valve head as a mass element.

This valve lift is rocker ratio times higher than the cam lift in the figure. And the load during positive acceleration at the cam side of the swing arm is also rocker ratio times higher than that at the valve side.

The valve clearance between the cam and the swing arm, and a nonlinear spring constant (k_{11}) between the cam and the swing arm and another nonlinear spring constant (k_{10}) between the swing arm and the valve, are placed to transfer the load at compression. That enables a simulation as the swing arm moves up and down in the clearance when the valve is seated.

The valve stem is affected by the spring set load as tension when the valve is seated, and by the load which the valve acceleration multiplied by the valve head

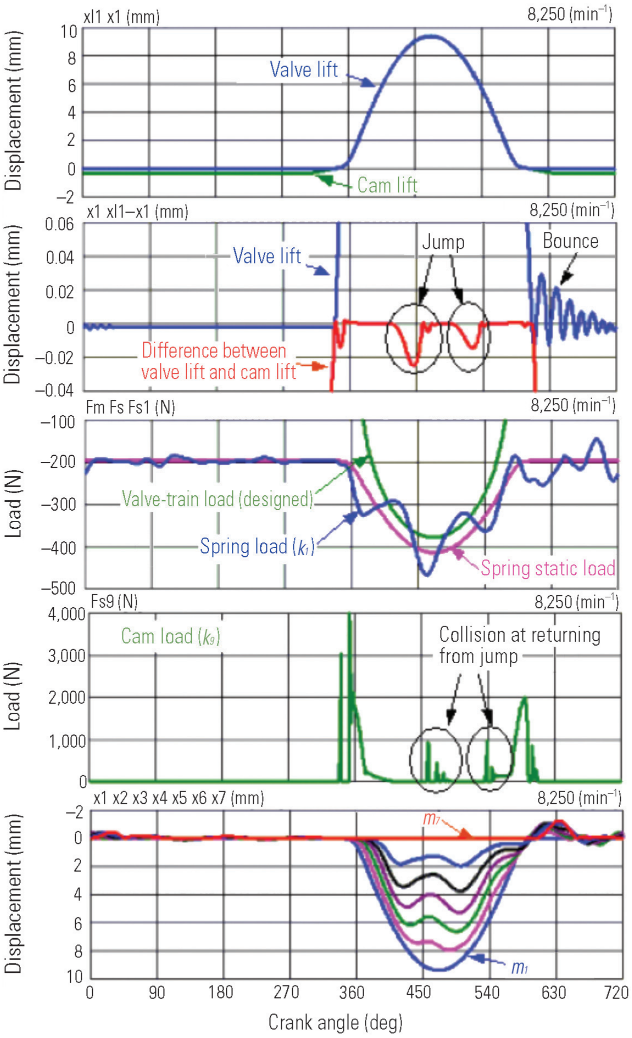


Fig. 6 Example of direct-acting valve-train calculation (8,250 min^{-1})

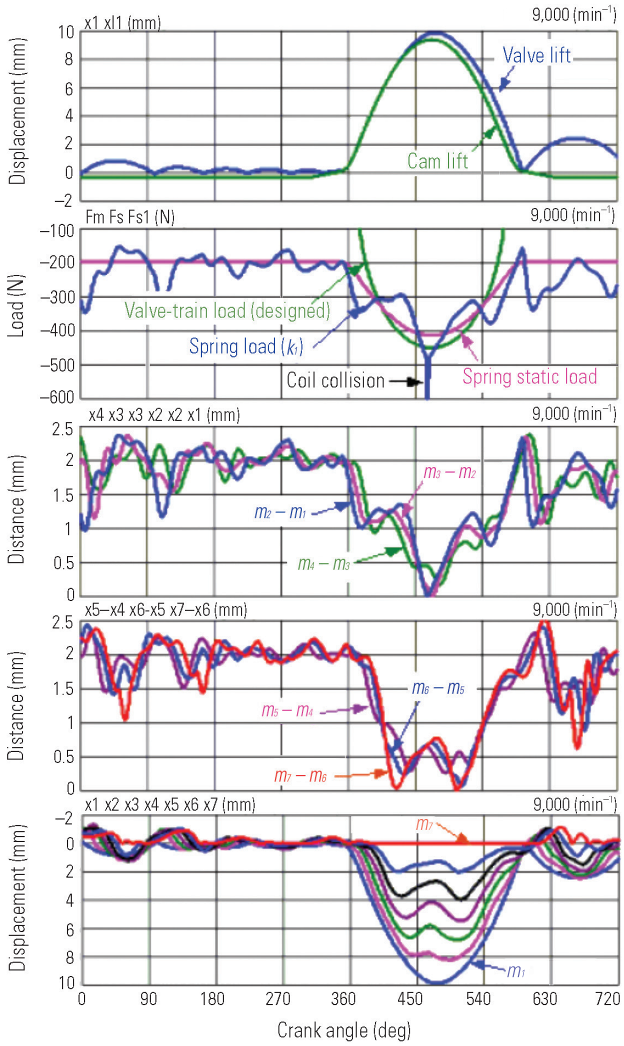


Fig. 7 Example of direct-acting valve-train calculation (9,000 min^{-1})

mass is equal to when the valve is lifted. And at the moment that the valve sits on the seat, it is observed that the spring set load is larger by the impact force.

3.3 Animation display of spring vibration

It is difficult to have an image of how the spring coil is vibrating from a figure drawn of the mass displacement along with the crank angle as the x-axis. Therefore animation software displaying the spring vibration is prepared to make users understand the phenomena visually.

Fig. 9 shows the valve spring motions at 7,500 min^{-1} of a direct acting valve-train in 3.1.1. From the pictures of the spring in this figure, it is observed that the bottom end coil of the spring is compressed earlier when the valve is being lifted and the top end coil of the spring is opened earlier when the valve is being returned. Then it also can be observed that the close coil which is usually closed opens a little after the valve closed timing.

4. Summary

Although many nonlinear elements like valve clearance and so on have to be handled in the valve-train calculations, the implicit algorithm used in this calculation code enables stable calculations. This report shows that this calculation method is able to handle any valve jumps, bounces and spring coil contacts. And this code is arranged to be used in the same sense as our engine performance simulator which users are very familiar with.

Because vibration calculations are not more difficult than fluid dynamics calculations, it is anticipated that if the damping coefficient is chosen well from experimental results, it is possible to develop a valve-train without making a prototype.

As computers progress, whole engine vibration calculations including a valve-train, engine main moving and timing parts has come into sight and this is one of our targets. In addition, since both side pressures of an engine valve calculated by our engine performance simulator are known, the advanced vibration calculation

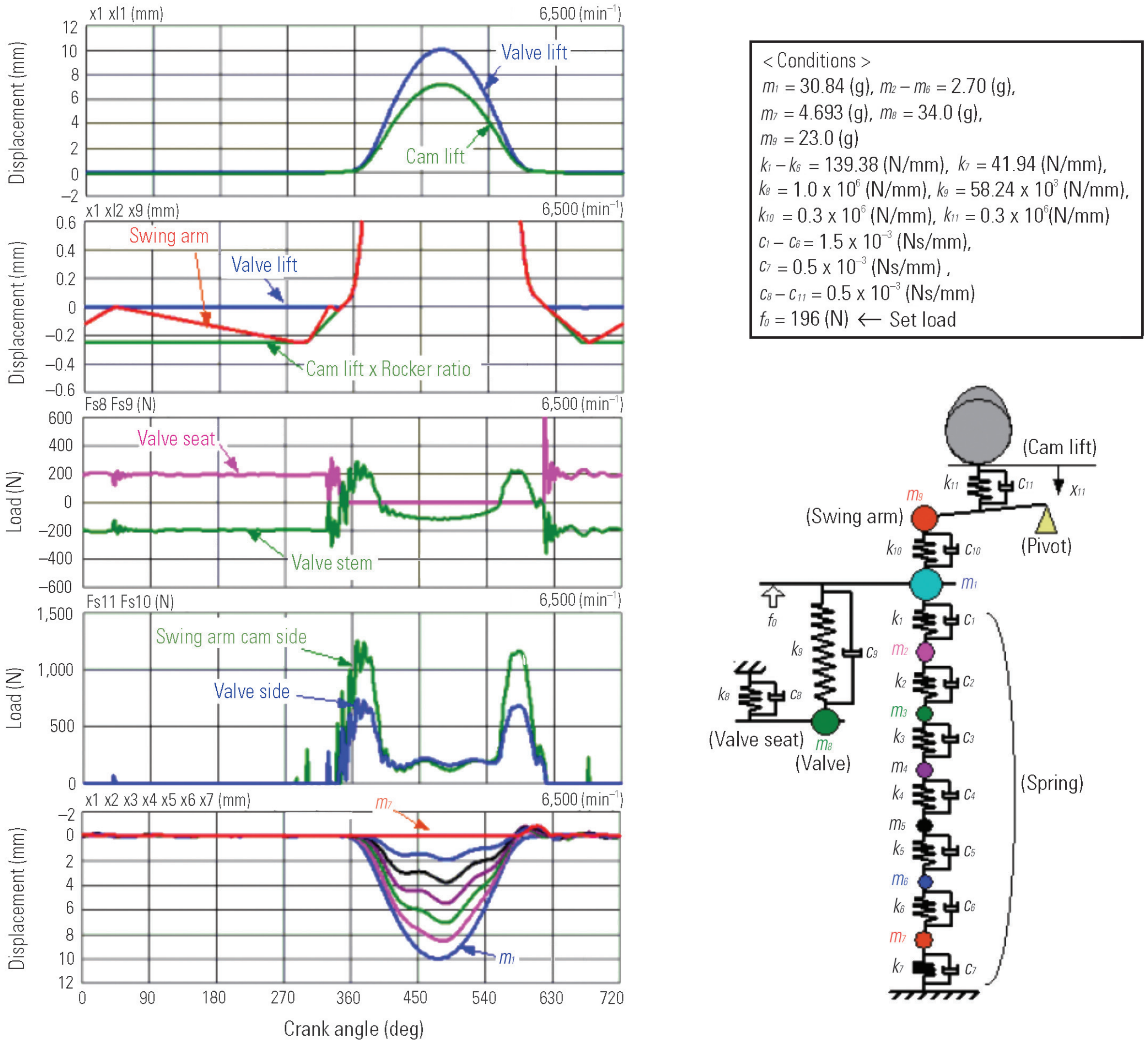


Fig. 8 Example of swing-arm-type valve-train calculation (Rocker ratio 1.4)

including the affect of these pressures is being considered.

5. After word

Since many kinds of commercial software have been available, the CAE job has changed from developing CAE software to applying CAE software to our development process. From this change, sometimes the results are misled because the engineer does not have enough knowledge about dynamic equations of the calculation target or the calculation algorithm itself. To avoid this bad situation, if CAE software can be made in a relatively easy algorithm, we dare to continue developing software ourselves in the future.

Mr. Fujimoto and Mr. Ohsawa in Advanced Powertrain Dept. gave a hand to verify the precision of this software in the developing process.

References

(1) T. Kitada, M. Kuchita,T. Ohashi: Mitsubishi Motors Technical Review No.11, P. 29, 1999

(2) A. Fujimoto, H. Higashi, M. Osawa et al.: Mitsubishi Motors Technical Review No.19, P. 19, 2007

(3) The Japan Society of Mechanical Engineers, Numerical Simulation of Flow, P. 58, Korona-sya, 1988 (Japanese)

(4) K. Kitasato, M. Tamaki: Mechanical Vibration (Basic and application), Kougaku-tosyo , p. 84, 1980 (Japanese)

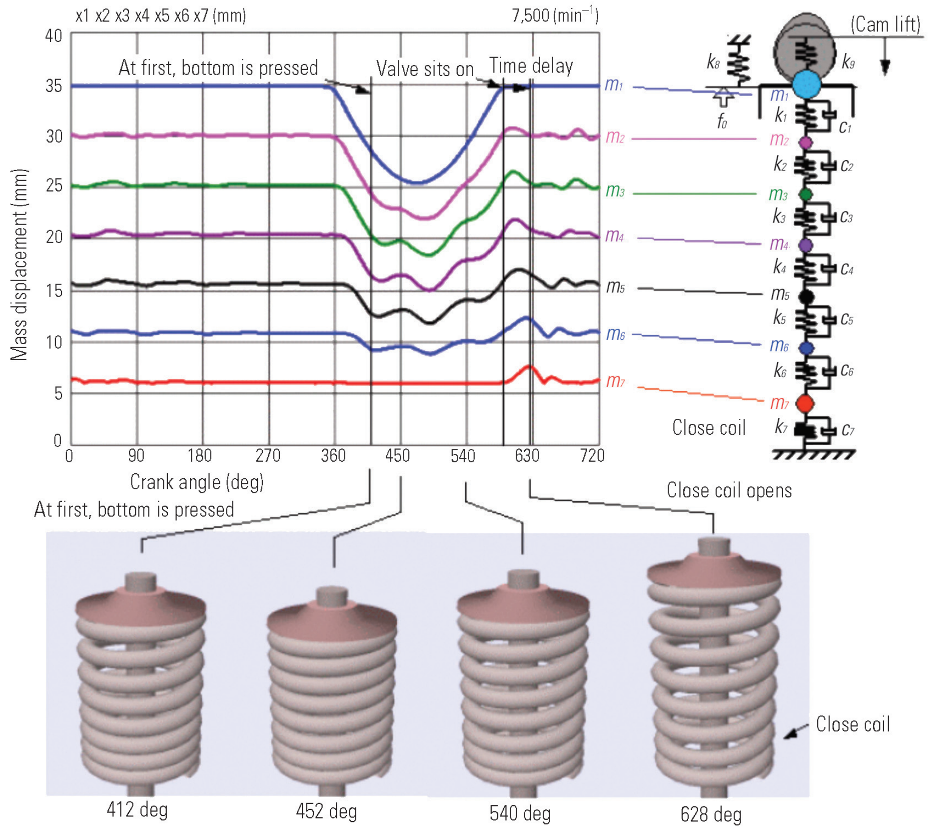
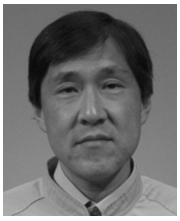


Fig. 9 Spring movement animation (Direct-acting type at 7,500 min^{-1})



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