MODELING AND ANALYSIS OF DESMODROMIC VALVE TRAIN

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"I admitted that I have read this paper and in my opinion this paper is enough in terms of scope and quality for granting Bachelor of Mechanical Engineering (Automotive)"

Signature : ....

Supervisor : ....

Date : ....

"I admitted that this report is my own work except summary and extract that each of them I explained it source"

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ABSTRACT

This research reveals the performance prediction of Desmodromic valve train system using multi-body system software, called MSC. ADAMS. The benchmark used was a conventional spring valve system. The research started by selecting a design of several Desmodromic valve systems. Then, a multi-body system model was developed using two stages approaches. At the first stage, the cam profile was developed using the reverse engineering method. Then a constraint was assigned between the cam and the rocker arm as a connection at stage 2. The dynamic analysis was performed to measure the acting force between rocker arm and valve seat and evaluate the performance of Desmodromic valve train over the conventional spring valve system.

ABSTRAK

Kajian ini melibatkan ramalan prestasi sistem deretan injap Desmodromic menggunakan perisian sistem *multibody*, yang dinamakan MSC ADAMS. *Benchmark* yang digunakan adalah satu sistem injap konvensional. Kajian dimulakan dengan pemilihan satu rekaan daripada beberapa sistem injap Desmodromic. Kemudian, satu model sistem *multibody* telah di buat menggunakan dua kaedah. Pada peringkat pertama, profil sesondol telah dibangunkan menggunakan kaedah *reverse engineering*. Kemudian satu kekangan telah ditetapkan di antara sesondol dan lengan jumpelang sebagai satu sambungan di peringkat 2. Analisis dinamik telah dilakukan untuk mengukur daya yang bertindak di antara lengan jumpelang dengan injap dan menilai prestasi terhadap sistem deretan injap Desmodromic untuk melihat perbandingan dengan sistem konvensional.

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SYMBOL	DEFINITION
F	Force, N
D	Drag, N
L	Length, mm
ρ	Density, kg/m^3
V	Velocity, m/s
k_p	The radius of gyration of the rocker about the rocker shaft, m^2
m	Mass, kg
I	Moment of inertia, kg m^2
P	Momentum, kgm/s
a	acceleration, ms^{-2}
ω	Angular speed, rad/s
τ	Torque, Nm
r	Radius, m
KE	kinetic energy, kgm^2

$\mathbf{q}$	Generalized coordinates
$\mathbf{M}(\mathbf{q})$	The mass matrix
$\mathbf{C}$	The constraint conditions
λ	Constraint forces
$\mathbf{u}$	Translational
$\mathbf{\Psi}$	Rotations
$\mathbf{Q}_v$	Quadratic velocity vector
λ_i	The Lagrange multiplier

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CHAPTER 1

INTRODUCTION

1.1 Overview

The Desmodromic is the valve train system operating mechanism where the camshaft controls both the opening and closing of the valve. This paper deals with modeling using multibody system method and dynamic analysis of both systems Desmodromic and conventional spring valve train to investigate the force contact and compare between each system. Basicly, the conventional system use the spring to close the valve. The rpm of the engine for high performance vehicle is greater. So, the system need the high spring stiffness to make sure that the spring can produce the force where are used to return the valve in closing valve condition. The principle of inertia is one of the fundamental laws of classical physics which are used to describe the motion of matter and how it is affected by applied forces. Inertia is the property of an object to remain constant in velocity unless acted upon by an outside force. Inertia is dependent upon the mass and shape of the object. The desmodromic train is a mechanism with positive-drive cams and – in comparison with the widely-used trains having a closing spring presents different dynamic behaviour.

The benefit of Desmodromic system is that when the motor is over-revved, the valves are still controlled, whereas when they are returned by springs the valves can "float" and hit the piston. Another is that the manufacturer can use wilder cam grinds for better performance.

Ducati and Mercedes are two major companies who have used the desmodromic system in racing engines. Ducati has the most experience of any manufacturer in the world at successfully applying desmodromic valve control to production machines. "The specific purpose of the desmodromic system is to force the valves to comply with the timing diagram as consistently as possible. In this way, any lost energy is negligible, the performance curves are more uniform and dependability is better". (Fabio Taglioni)

Two disadvantages on the system valve spring were detected that is:

i. Heavy spring burden, mean work more do by engine (and cause loss of power
ii. "Valve bounce" at high RPM.

1.2 Objektif

To develop a multibody system model and evaluate the performance of Desmodromic valve train over the conventional system. A multi-body system model was developed using two stages approaches. The cam profile was developed using the reverse engineering method. Then a constraint was assigned between the cam and the rocker arm as a connection. The dynamic analysis was performed to measure the acting force between rocker arm and valve seat and evaluate the performance of Desmodromic valve train over the conventional spring valve system.

1.3 Valve train Model

Thus it is possible to identify two parts of the mechanism, each made up of one of the cam discs and the related rocker: they give valve acceleration respectively in positive and negative directions, where the positive direction is considered to be the one of the opening valve. Here the terms 'positive' and 'negative' cam disc/rocker are used; however these links are commonly, but improperly, called 'opening' and 'closing' cam disc/rocker, respectively. It is noteworthy that there is also a small helical spring mounted around the negative rocker pin and properly preloaded; its main function takes place during the dwell phase, when there is separation of the cam discs from the rockers: this spring applies to the rocker the force required for getting the proper contact force between valve-head and seat.

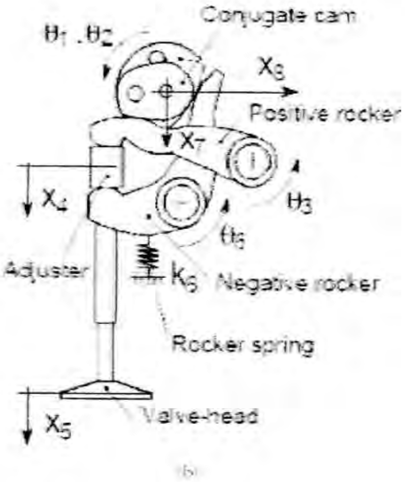
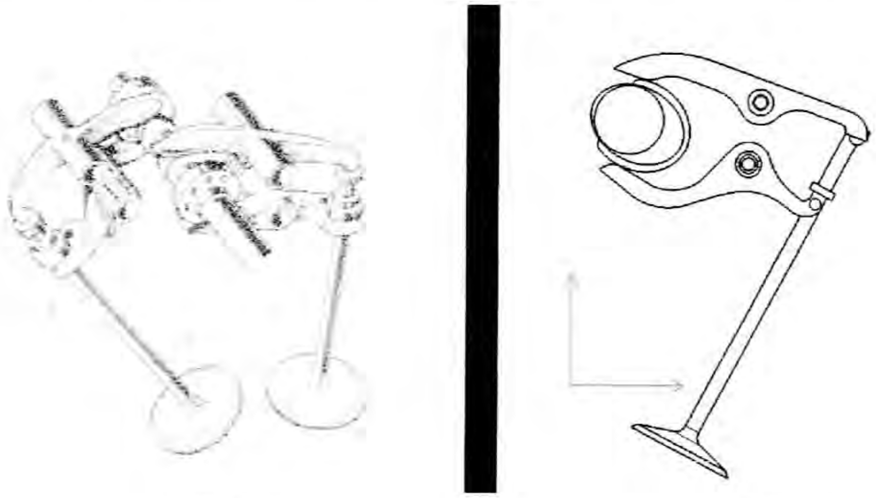


Figure 1.1: schematic of the cam mechanism driving a single valve.



(a) (b)

Figure 1.2: (a) General view of the desmodromic valve train of Ducati motorbike engines with 2 valves percylinder with single camshaft operates open-close of the system. (b) Drawing by Catia.

1. Opening rocker arm (upper)
2. Upper rocker arm adjuster (shim)
3. Hale rings
4. Closing rocker arm adjuster
5. Lower rocker arm return spring
6. Closing rocker arm (lower)
7. Camshaft
8. Valve

With respect to the more widely-used trains with closing spring, the desmodromic trains make it possible to give very high valve accelerations, preventing the follower from jumping off the cam, without employing a very stiff closing spring; on the other hand, the mechanical complexity of the desmodromic system is justified only in high-speed engines with single cylinder heads, as Ducati engines.

1.4 Problem Statement

This study focused to the effect elimination the spring that usually used as close valve. In fact of spring have high compression force, large torque needed by engine to overcome the compression force that will cause power waste on the engine. There is also high frictional force among cam and arm rocker could cause both this part ware faster than Desmodromic valve train.

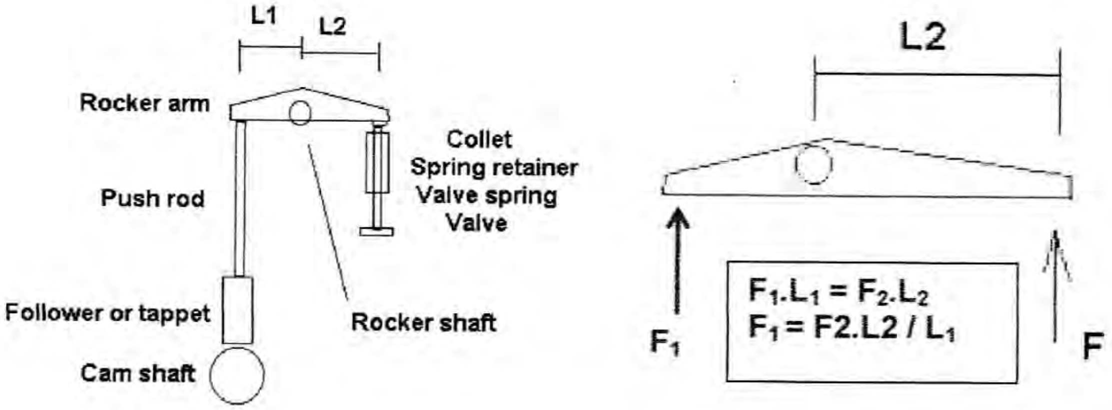


Figure 1.3 – contact force between rocker arm and cam

Valve float is a condition which occurs when the valves do not stay in contact with the camshaft lobe. Stiffer valve springs can help prevent this but only at the expense of increased friction losses. Valve bounce is a related condition where the valve does not stay seated, due to the combined effects of the valve's inertia and resonance effects of metallic valve springs that effectively reduce the closing force, and allow the valve to re-open partially. Various techniques have been used to offset the effect of stiffer springs, such as dual-spring and progressive-sprung valves, and pneumatic valves.

The motorcycle manufacturer Ducati uses the desmodromic (springless) valve system to counter this problem and allow for higher engine speeds. The system consists of a mechanical lifter mechanism that uses a second rocker arm to push the valve closed. Formula 1 engine manufacturers use a pneumatic system to close the valves to allow for very high engine speed without valve float. On very high performance valve spring engines, the spring does not always have time to return to its pre-compressed position, causing the camshaft to recompress the spring and valve prematurely. This is called "valve float". The Desomodromic system also eliminates the extra "work" spent by the motor to open spring actuated valves. Therefore giving more actual power at the wheel rather than using it to work against the seat pressure on the spring.

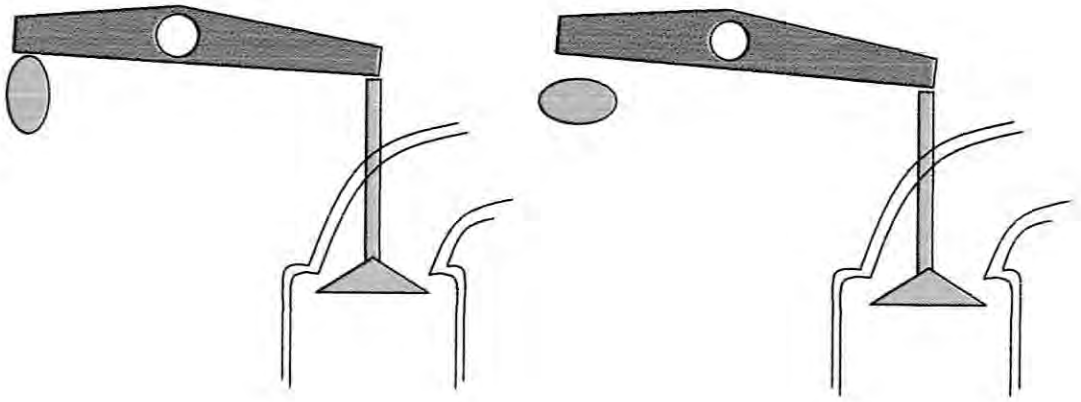


Figure1.4 – valve float situation

1.5 Scope of Project

The scope of this project is to model the Desmodromic valve train and spring valve system using MSC ADAMS software to make the dynamic analysis of both systems. By this analysis, it can investigate the inertia required to perform better performance.

i. Develop multibody system model for desmodromic and conventional valve train system
ii. Carry out dynamic analysis on the both system
iii. Evaluate the performance of desmodromic valve train system
iv. Come out with design recommendation for improvement

1.5.1 Determination of the Forces in the Spring Valve Train in an Internal Combustion Engine

To allow the maximum amount of mixture (or air for a diesel) to be drawn into the cylinders it is desirable that the inlet valve(s) open as quickly as possible, with as high a lift as possible and remain open for as long as possible followed by rapid closing. To achieve these aims the components in the valve train are subjected to high accelerations (and hence forces) as the inertia of the components has to be overcome.

During the valve opening phase the cam provides the necessary force and during the closing phase the valve spring(s) provides the forces (except in an engine with desmodromic valve operation - where a second cam provides the closing forces. Currently only Ducati motorcycles use this system). To simplify the analysis it will be assumed that the cam profile is symmetrical, ie, where the closing phase is the 'mirror image' of the opening phase. In practice this may not be the case.

1.5.2 Practical Details

Although 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.

1.5.3 Analysis - Acceleration of Cam 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.

i) 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 . Also assume that the cam is opening and closing the valve for 120° of its rotation. Hence the complete valve cycle is completed in 1/3 camshaft revolution, or 0.01 seconds. The equation describing SHM is:

displacement = amplitude x cos(omega x t)
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)
So the omega value here is 2 x 3.14159/0.01 = 628.3 rad/sec.

Differentiating the expression for displacement gives:

velocity = - omega x amplitude x sin(omega t), then differentiating again:

$\text{acceleration} = -\,(\text{omega})^2 \text{ x amplitude x } \cos(\text{omega } t)$

The maximum acceleration occurs when the term cos (omega t) has the maximum value of 1, this occurs at the extremes of the motion. If the cam has a lift of 20 mm, the amplitude of the motion is 10 mm, and the maximum acceleration is given by:

$\text{acceleration}_{max} = (628.3)^2 \text{ x } 0.01 = 3948 \text{ m/s}^2$

ii) An alternative assumption is to assume that that the peak of the cam is finished with specific constant radius with a centre at a distance 'e' from the centre of the cam. The term 'e' may be called the eccentricity. The peak radius is often blended to the the base circle by a large radius curve.

In this case the acceleration of a 'mushroom' - flat foot - follower in the vicinity of the maximum cam lift is given by: (cam shaft angular velocity)^2 x eccentricity. If the cam base circle is blended to the nose radius by a large radius, the maximum acceleration occurs when the follower is on this larger radius and is equal to the large blend radius x (cam shaft angular velocity)^2

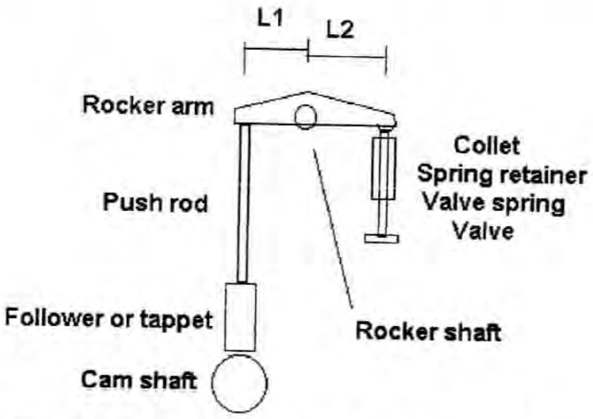
1.5.4 Analysis - Valve Train Forces.

If the cam shaft is a direct acting overhead unit then all the components in the (short) valve train will undergo the same acceleration. The exception to this is the spring, for which an allowance may be made by using 1/3 of its mass as an effective mass. Maximum force will then be:

$F = \text{max. acceleration x (masses of tappet + retainer + collets + valve + } 1/3 \text{ of spring)}$

At least this much force must be generated by the valve spring at its maximum compression. Use the spring design applet with measurements and / or data from the workshop manual to check this.

The analysis becomes more complex when the engine does not have an overhead cam shaft, or when there is an overhead cam shaft that operates the valves by rockers. When this is the case the intertia of the rocker needs to be considered and the 'lever effect' of the rocker needs to be considered - as the two side arms of the rocker are frequently unequal. As an example consider the layout shown below:



General arrangement of valve gear in an engine with the cam in the side of the cylinder block

Figure 1.5: the schematic diagram of spring valve system with pushrod

Let us assume the following values and dimensions:

i. Tappet 200 gm
ii. push rod 250 gm
iii. Valve, collet, retainer and 1/3 spring, total 300gm
iv. Rocker arm 200 gm
v. L1 20 mm
vi. L2 40 mm

The easiest way to carry out the calculations is to determine the equivalent mass of the valve train at the valve side, then use:

$$F_{valve\ side} = m_{total\ equiv.\ on\ valve\ side}\, x''_{valve\ side}.$$

Components in group (3) above are the items on the valve side with a mass of 300 gm. The rocker now needs to be represented as an equivalent mass at the valve side. First the moment of inertia needs to be estimated. Depending upon the design of the rocker it may be possible to perform a 'compound pendulum' experiment to determine this.