

Effect of Design Features on Dynamic Characteristics of VCC Piston for I. C. Engine

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ABSTRACT

VCC (variable combustion chamber) piston technology would be one of the best substitutes for VCR (variable compression ratio) technology due to its extremely fast dynamic response with simple structure. One of the key issues of VCC piston technology is its dynamic response, which means **fast, timely and enough** variation of the combustion chamber volume during combustion in each working cycle while excess in-cylinder pressure occurs.

Two key design features, ball-profiles and a reset cam, are introduced in order to ensure the availability of transient dynamic characteristics of VCC piston. Several formulas are derived from the structural analysis on VCC piston to describe the relationship between VCC piston design and its dynamic characteristics.

By means of ADAMS dynamic simulation, the affecting factors on the dynamic characteristics of VCC piston, including stiffness, friction, damping, contacting conditions and engine working conditions, are investigated, and the effects of pre-tightening force and the design features on the dynamic response are also investigated.

The research indicates that (1) the dynamic response of VCC piston is very fast due to the design features of both ball-profiles and reset cam, and (2) by means of VCC displacement (the relative movement between the piston crown and the piston skirt), excessive in-cylinder pressure would be limited effectively in each working cycle while a spark-ignition engine running with higher CR (compression ratio), and (3) the dynamic response of VCC piston can be improved by reducing the pre-tightening force of the disc springs properly.

KEYWORDS: variable combustion chamber (VCC), variable compression ratio (VCR), dynamic characteristics, dynamic response, internal combustion engine

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INTRODUCTION

VCC (variable combustion chamber) piston technology would be one of the best substitutes for VCR (variable compression ratio) technology due to its transient dynamic response and simple structure with extremely low cost^[1]. There have been over 100 VCR patents and lots of the patent applications in recent years, but none of these VCR designs has been commercialized yet^[2].

It has been generally believed that the complicated control mechanism and structure is the main obstacle to the fulfillment of VCR technology^[2] ^[3]. Many VCR designs, such as MCE-5 VCR engine^[4], and the VCR engine of FEV company^[5] ^[6] and SVCR engine of SAAB company^[7], constructed their CR control systems by means of electronic-controlled hydraulic mechanisms. NISSAN announced its

multiple-link VCR engine^[8] ^[9], and Honda also reported a design of VCR engine with dual piston mechanism in 2009^[10]. The complicated VCR designs above would result in the huge modification or reconstruction on engine structures with higher cost and lower reliabilities.

Another almost insurmountable obstacle to VCR approach, however, is that VCR modes are far from meeting the requirement for the vehicle engines at transient driving conditions. The reasons include that (1) it is obviously too slow for VCR modes to control CR (compression ratio) at driving conditions due to the inertial motion of the electronic-controlled hydraulic mechanisms, and (2) it is hard for VCR engine to foresee the variation of CR with randomly driving conditions, and (3) the delayed response of VCR not only affects the fuel economy and power output of the engines, but also affects the operating safety of the engines.

Authors also had focused on VCR technology for years in their early researches^{[13] [14] [15]}, and have turned to VCC approach since 2012^[1].

By means of VCC piston technology, excess in-cylinder pressure can be limited during combustion in every working cycle rather than in several hundred cycles of VCR engines. The duration of combustion is a very short time especially in high revolution speed. For instance, the duration is less than 1.0 millisecond at the speed of 6000 rpm. Therefore, it is a big challenge for VCC piston technology to meet the demand for such a transient dynamic response.

There are certainly many problems for this new VCC piston technology, such as the effect on the engine behaviors, the combustion characteristics, the control strategies of in-cylinder pressure, the interrelationship between the performance of VCC engine and the dynamic characteristics of VCC piston, and the reliability of VCC engine and so on.

The object of the paper is to investigate the effect of design features on dynamic characteristics of VCC piston. Two of the design features, including the ball-profile and the reset cam, are discussed in detail. Several formulas are derived to find some theoretical supports for the fast dynamic response of VCC piston. Also, some of the affecting factors on the dynamic characteristics of VCC piston are investigated by means of 3D dynamic simulation.

BALL-PROFILE DESIGN FOR DYNAMIC RESPONSE

Figure 1 shows the schematic of one of VCC piston designs^[12]. Similar to the design of VCC piston in reference [1], between piston crown and piston skirt there is a VCC mechanism, which consists of several disc springs, some balls, a support disk and a limit disk. The support disk is bolted to piston skirt. The balls are spaced in circumference among the disc springs, the support disk and the limit disk. Besides, there is a reset cam fixed at the wrist-pin end of connecting rod.

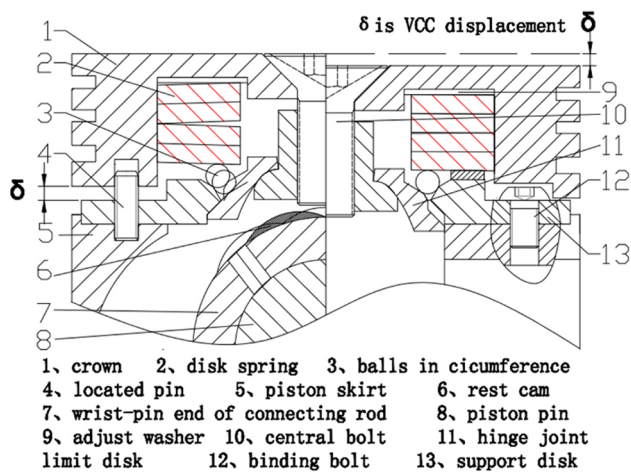


Figure 1. Schematic of VCC Piston

Different with the design of VCC piston in reference [1], the limit disk here is a kind of joint, which can spin slightly around the centre bolt. By means of this design, some unbalanced situations of the balls can be self-adjusted to some extent.

The pre-tightening force of VCC piston is prescribed during assembling. When in-cylinder pressure is lower than the pre-tightening force, there is no relative movement between the piston crown and the piston skirt (called **VCC displacement** in the paper).

When in-cylinder pressure exceeds the pre-tightening force during combustion, a downward VCC displacement occurs. VCC displacement includes not only the deformation of the disc springs, but also the axial displacement from the balls' moving. VCC displacement means a rapid variation of combustion chamber volume near TDC, which will result in a rapid decrease of in-cylinder pressure.

While in-cylinder pressure is rapidly decreased during the expansion stroke, an upward VCC displacement occurs under the action of the spring force. The reset cam spins towards the cylinder centerline, which is helpful to push the limit disk upwards during the process of expansion/exhaust stroke, ensuring the piston crown to be reset in every circle.

Spring designs are common in applications such as valve-train systems, but the design of ball-profile provides VCC piston with much faster dynamic response than traditional spring designs. Figure 2 shows the schematic of an example of the ball-profile. For the sake of analysis, the bottom disk spring is simplified as a horizontal flat plate, and the highlight lines (blue and red lines) show the slope angles of support disk and limit disk. Both surface profiles of the support disk and the limit disk consist of but not limited to two curves (lines) with slope angles of α_1 , α_2 and β_1 , β_2 respectively. The intersection of two lines α_1 and α_2 or two lines β_1 and β_2 is called as inflection point.

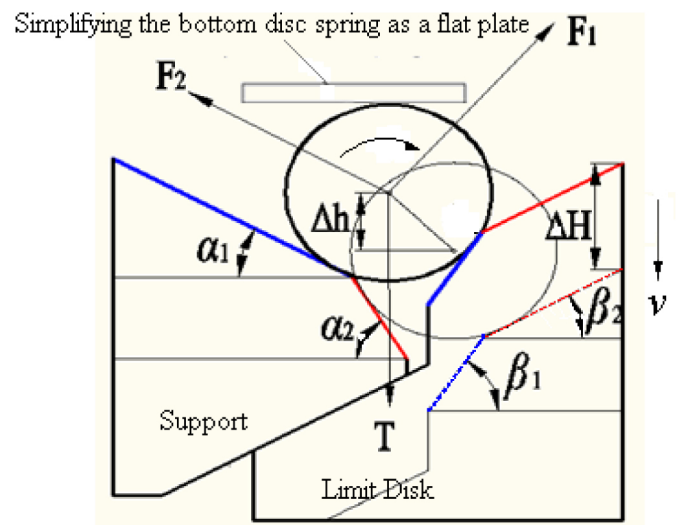


Figure 2. Schematic of an Example of the ball-profile

VCC displacement ΔH includes both the deformation of disc spring Δs and the axial displacement Δh from balls' moving, which can be defined as,

$$\Delta H = \Delta h + \Delta s \quad (1)$$

In general, Δh is much greater than Δs . In order to express the effect of the ball-profile structure on VCC displacement, let $k = \Delta h / \Delta s$. k is called the amplification coefficient of displacement, which can be defined as,

$$k = \frac{\Delta h}{\Delta s} = \frac{\Delta h}{\Delta H - \Delta h} = \frac{\tan \alpha}{\tan \beta} \quad (2)$$

While the friction is not taken into account, the static force balance of ball-profile can be simply defined as,

$$\begin{cases} F_1 \sin \alpha = F_2 \sin \beta \\ F_1 \cos \alpha + F_2 \cos \beta = T \end{cases} \quad (3), (4)$$

$$\begin{cases} F_1 = \frac{\sin \beta}{\sin \beta \cos \alpha + \sin \alpha \cos \beta} T \\ F_2 = \frac{\sin \alpha}{\sin \beta \cos \alpha + \sin \alpha \cos \beta} T \end{cases} \quad (5), (6)$$

$$\frac{F_2 \cos \beta}{F_1 \cos \alpha} = \frac{\sin \alpha \cos \beta}{\sin \beta \cos \alpha} = \frac{\tan \alpha}{\tan \beta} \quad (7)$$

where, F_1 is the sustaining force from the support disk on all balls; F_2 is the sustaining force from the limit disk on all balls; T is the pressing force from disc springs on all balls; the radial component force, $F_1 \sin \alpha$ and $F_2 \sin \alpha$ are mutually balanced. $F_1 \cos \alpha$ is the axial component force of F_1 , and $F_2 \cos \alpha$ is the axial component force of F_2 . Shown in

formula (7), the bigger the force ratio $\frac{F_2 \cos \beta}{F_1 \cos \alpha}$ is, the faster the response of VCC displacement is. The relationship among F_1 , F_2 and T is totally dependent on the slope angles of α and β . The formula (5) and the formula (6) express the ratio relationship among F_1 , F_2 and T .

Table 1. Example of the design of ball-profile

The location of balls	α	$k = \frac{\Delta h}{\Delta s}$	$\frac{F_2 \cos \beta}{F_1 \cos \alpha}$
Before inf. point	$\alpha_1 = 30^\circ$	$\sqrt{3}/3$	$\sqrt{3}/3$
After inf. point	$\alpha_2 = 60^\circ$	$\sqrt{3}$	$\sqrt{3}$
$\beta_1 = \beta_2 = 45^\circ$			

There is an inflection point at the intersection of two lines (in application, two lines are connected with a section of arc). The balls are located before the inflection point with slope

angles of α_1 and β_1 while there is no or less VCC displacement.

Table 1 shows the variation of amplification coefficient of displacement k and the force ratio $\frac{F_2 \cos \beta}{F_1 \cos \alpha}$ with the slot angles α and β . According to formula (2) and formula (7), when $\alpha_1 = 30^\circ$, $\alpha_2 = 60^\circ$, $\beta_1 = \beta_2 = 45^\circ$, the results indicates clearly the reasonable design of ball-profile enables VCC displacement and the axial component force $F_2 \cos \alpha$ to be increased rapidly. Both coefficient k and the ratio of the axial component force $\frac{F_2 \cos \beta}{F_1 \cos \alpha}$ are increased 3 times after the inflection point, which is obviously helpful to improve the dynamic response of VCC piston.

The formula (1), (2), (3), (4), (5), (6), (7) have clear meaning in physics and simple forms in mathematics, and are very valuable for the explanation of VCC piston concept. By means of the formula (1), (2), (3), (4), (5), (6), (7), the basic theory is built to explain the fast and enough variation of combustion chamber volume during combustion.

RESET CAM DESIGN FOR PISTON CROWN RESETTING IN EVERY CIRCLE

Another design feature of VCC piston is the reset cam, and the purpose of the reset cam design is to ensure the piston crown to reset in every working cycle.

As discussed in the previous section, the fast, timely and enough downward VCC displacement might be easy to be guaranteed when in-cylinder pressure is excessive to pre-tightening force by mean of the design of ball-profile with the variation of slope angles α and β .

The situation of upward VCC displacement is totally different when in-cylinder pressure is rapidly decreased. Shown as in figure 3 (paying attention to the directions of the piston crown's movement and the directions of friction forces), while the friction is taken into account and the bottom disk spring is still simplified as a horizontal flat plate, the resetting conditions of the piston crown can be defined as,

$$T > F_g + F_2 \cos \beta + f_2 \sin \beta \quad (8)$$

Where T is the pressing force from the disc springs on the limit disk, F_g is the in-cylinder pressure on the piston crown, F_2 is the force from all balls on the limit disk, f_2 is the friction force between the balls and the limit disk, β is the slope angle of the limit disk.

While letting $F_g \approx 0$, $\beta = \beta_1 = \beta_2 = 45^\circ$, $f_1 = \mu F_1$, $f_2 = \mu F_2$, $f_t = \mu T$ (μ is friction coefficient), based on the static force balance of the ball, F_2 can be expressed as,

$$F_2 = \frac{1}{1+\mu^2} \frac{\sin \alpha + 2\mu \cos \alpha - \mu^2 \sin \alpha}{\sin \alpha + \cos \alpha} \sqrt{2} T$$