Mechanical Design and Theoretical Analysis of a Skip-Cycle Mechanism for an Internal Combustion Engine

Ismail Gerzeli, Cemal Baykara, and Osman Akin Kutlar

Abstract—Skip cycle is a working strategy for spark ignition engines, which allows changing the effective stroke of an engine through skipping some of the four stroke cycles. This study proposes a new mechanism to achieve the desired skip-cycle strategy for internal combustion engines. The air and fuel leakage, which occurs through the gas exchange, negatively affects the efficiency of the engine at high speeds and loads. An absolute sealing is assured by direct use of poppet valves, which are kept in fully closed position during the skipped mode. All the components of the mechanism were designed according to the real dimensions of the Anadolu Motor's gasoline engine and modeled in 3D by means of CAD software. As the mechanism operates in two modes, two dynamically equivalent models are established to obtain the force and strength analysis for critical components.

Keywords—Dynamic Model, Mechanical Design, Skip Cycle System (SCS), Valve Disabling Mechanism

I. INTRODUCTION

INTERNAL combustion engines have a variety of application area owing to the traditional advantages, like providing high power density, having high energy content and the ease of onboard storage of liquid fuels. The concern for the environment and the insufficient fuel resources increases with the growing amount of internal combustion engines. So, the trend in the industry has been increasing engine efficiency which is reducing the fuel consumption and the generation of pollutants. The total efficiency of the conventional spark ignition engines is approximately 30-35% as a result of many factors including internal friction and parasitic losses, rolling resistance and air drag. The IC engines at full load have a limited thermodynamic efficiency. On the contrary, at partial load due to the pumping losses, the efficiency of the engine

dramatically decreases. To solve this problem the variable valve actuation, the reduction of the gas exchange through charge dilution, and for larger engines the cylinder deactivation was developed [1].

The skip-cycle strategy for spark ignition engines, presented and analyzed by Kutlar [2]-[3], allows to change the density of power strokes per revolution of the engine through skipping some of the four stroke cycles. The result proves that the SCS makes it possible to operate the engine at lower speeds and loads with less fuel consumption, without any stability problems. On the other hand, the brake specific fuel consumption increases at high speeds and loads, due to the fuel and air leakage of rotary valves used on research engine. The result shows that the fuel consumption increases the HC and O_2 emissions and decreases CO and CO_2 emissions compared to the normal stroke mode. Kutlar also suggests that the direct use of poppet valves instead of rotary valve for cycle skipping may solve the leakage problem [3].

Some valve disabling mechanisms were represented by the inventors to keep intake and/or exhaust poppet valves in fully closed position during the engine operation. Walsch [4] invented a valve disabling device associated with a rocker arm. Also, a finger follower rocker arm system was represented and patented by Diggs [5] including an actuating mechanism to engage/disengage a pin to limit the movement of the rocker arm. Then, a mechanism was defined by Dyer [6] for controlling the valve operation in an internal combustion engine based on shifting the cam lobes which controls the valve lift at zero to maximum deflection. And finally, another valve actuating mechanism, which offers optional disabling valves, was proposed by Ajiki and Matsuura [7].

On the research engine, rotary valves placed on the inlet port and are used to prevent the gas exchange. Even though, they are in fully closed position, they do not acquire a complete sealing [3]. As a solution, direct use of poppet valves instead of rotary valves is suggested and the mechanism is based on holding valves in fully closed position during the skipped cycle.

II. SUMMARY OF THE MECHANISM AND WORKING PRINCIPLE

The present valve actuating mechanism used on engine consists of a cam lobe, tappet, push rod, rocker arm, poppet valve and its spring. The rocker arm is pivoted on the support shaft which is horizontally placed through the journal of cylinder head. The tappet and pushrod are interposed between

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the cam and the rocker arm. So, one end of the rocker arm is actuated by a cam, and the other end engages the free stem of the intake or exhaust poppet valve. According to the rotation of the cam, the rocker arm undergoes a see saw motion and pushes the valves against the reaction force generated by valve springs.

The new mechanism is designed for Anadolu Motor's water cooled gasoline engine which has limited space and is not suitable to make some constructive changes at particular areas. The most suitable option is the cylinder head assembly to accommodate the mechanism. Although the engine has a single cylinder and one pair of inlet and exhaust valves, this mechanism can be adopted for the multi cylinder engines. As the last step, all the specifications of the engine parts are obtained from the manufacturer and the final assembly was embodied according to the real dimensions of the original engine.

The skip cycle strategy is assured by keeping the poppet valves in the closed position which are operating in two modes, normal or skip mode. In the normal four stroke mode (n), the SCM is disabled, so the valves are activated according to the rocker arm motion. However, during the skip four stroke mode (s), the SCM is enabled and valves remains in the fully closed position during the gas exchange (intake or exhaust).

A. Valve Disabled Mode

The section view of the mechanism during the valve disabled mode is shown in (Fig. 1). In this application, locking plate remains in the retracted position. So, the plunger, which contacts the rocker arm, is free to slide in a close fit inside the spring retainer. The spring, located in the hollow, is used to lift up the plunger to keep always in contact with the rocker arm. Furthermore, the spring retainer is connected to the poppet valve with half collets and clamps the valve spring. The valve spring is stiffer than the plunger spring and assembled preloaded. As the plunger spring has a low spring rate, its reaction force at maximum deflection will not be enough to overcome the initial force of the valve spring. So, the axial force acting on the plunger is not transmitted to the valves, and they remain fully closed position.
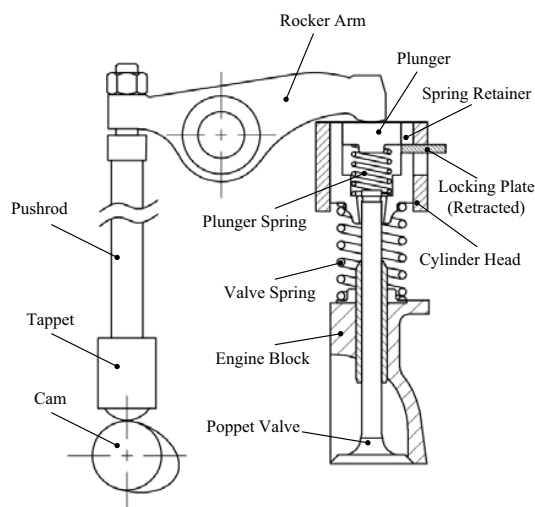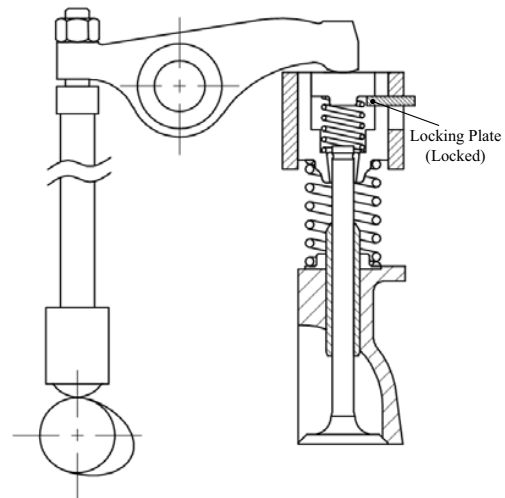

Fig. 2 Valve enabled mode of the skip-cycle mechanism

B. Valve Enabled Mode

The section view of the mechanism during the valve enabled mode is shown in (Fig. 2). In this operating mode, the locking plate slides horizontally and locates under the plunger. So, the movement of the plunger relative to the spring retainer is limited and the mechanism functions as a one single piece. The spring retainer is free to slide in a close fit inside the cylinder head body. Eventually, the oscillating motion of the rocker arm is transmitted to the valves. When the rocker arm pivots in reverse direction, the valve spring force pushes the retainer and the locking plate in upward direction. Simultaneously, it is supposed to pull back the locking plate to the first position.
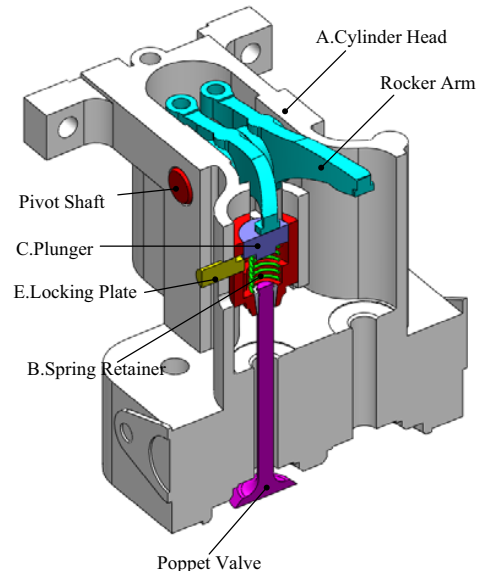

Fig. 3 The 3D model of the skip-cycle mechanism assembly

III. DESIGN OF COMPONENTS

The skip-cycle mechanism assembly (Fig. 3) is located between rocker arm and valve stem, and includes some newly designed and modified components. The design can be

Fig. 1 Valve disabled mode of the skip-cycle mechanism

categorized into five distinct areas, namely; A. Cylinder Head, B. Spring Retainer, C. Plunger, D. Plunger Spring, E. Locking Plate.

A. Cylinder Head

Basically, the cylinder head assembly consists of two parts, the cylinder head body (Fig. 4) and the cover which are located at the top section of the engine. Furthermore, it includes the main parts of the valve actuating mechanism, the rocker arms and the pivot shaft, the valve stems, springs, and spring retainers.

The SCM is interpolated between the rocker arm and the valve stem, which unavoidably increases the dimensions of cylinder head assembly and also changes the axial position of pivot shaft bushing. As the rise of dimension triggers the material consumption and makes manufacturing difficult, it should be preserved at minimum level.
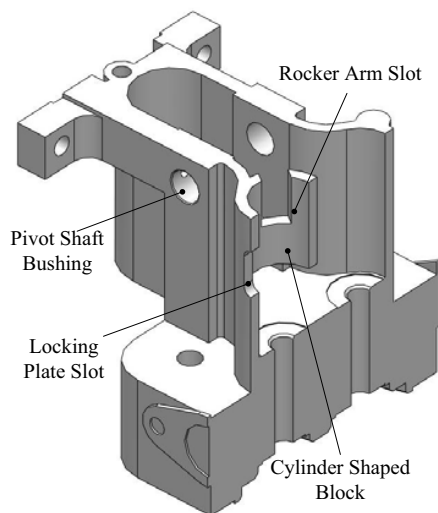

Fig. 4 Cylinder head body

In the valve enabled mode of the mechanism, while the spring retainer and the locking plate have reciprocating motion, both need a close fit to easily slide. For guiding the spring retainer, a cylinder shaped block was designed on the inner front section of cylinder head. However, the walls of housing hinders the see saw motion of rocker arm. So, a slot was drilled to solve this problem. As the last step, for pivoting locking plate a vertical slot was milled to the outer wall of the cylinder head.

B. Spring Retainer

Spring retainers are designed to clamp and securely hold upper portions of the spring and generally used in valve actuating mechanisms. The valve stems are connected to the retainers with half collets. In preliminary steps of embodiment, although the SCM and retainers were decided to be designed as separate parts, to gain space and simplify installation, both of them were formed as a single part (Fig. 5). Hereby, the retainer was rearranged as a main body of skip-cycle mechanism, and includes the plunger and the plunger spring.

To confine the plunger spring, a hole was drilled in the middle of the retainer. The diameter of the hole (18 mm) and the outer diameter (17,6 mm) of the plunger spring are related with each other and optimized in dimensions for best performance. Also, note that the natural diameter changes are considered, when springs are compressed and therefore the hole was drilled in tolerance limits.
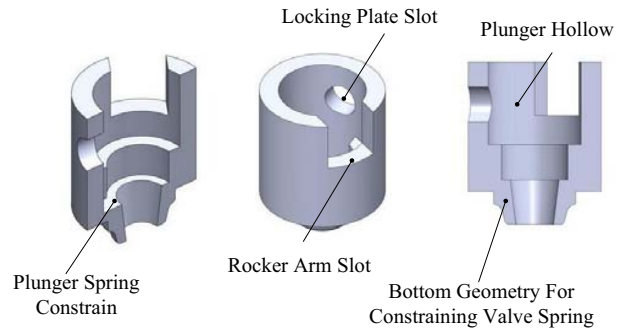

Fig. 5 Spring retainer

To guide the locking plate, a slot is machined on the outer surface of the retainer. The junction point and the bottom geometry of the retainers stayed the same as in the original one.

C. Plunger

The plunger (Fig. 6) is always in contact with the rocker arm. The contact pressure between mating surfaces of the plunger and the rocker arm causes Hertz contact stress distributions. To prevent the surface failure, the maximum surface contact pressure should not exceed the endurance limit.
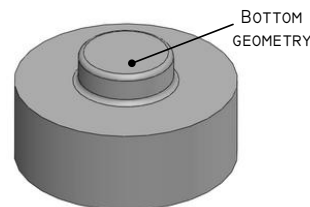

Fig. 6 Plunger

While designing the bottom of the plunger, the original geometry of the spring retainer is used to constrain the upper portion of the plunger spring.

D. Plunger Spring

A constant pitch round-wire helical compression spring is used to raise the plunger during the valve disabled mode. It is important to define the spring geometry for calculation and manufacturing (Table I). The working length of the plunger spring is equal to the stroke distance of the rocker arm. As the limited outside diameter of the spring is known, a suitable spring index C is selected, and the wire diameter is determined, and selected from the preferred diameter table of the material. As a material, the same one with valve spring chrome vanadium, identified A232 in ASTM designations, is

used, because it is suitable for fatigue loading and also good for shock and impact loads [8]. The temperature at the cylinder head is approximately 150°C degrees, which has to be taken into account because the load exerted by the spring will gradually relax with time.

The reaction force of the spring at the maximum deflection should equal to the total weight at the valve side of rocker arm. So, the spring rate is found and subsequently the number of active coils is determined (1).

$$N_a = \frac{d_4 G}{8 D^3 k} \tag{1}$$

where D is mean coil diameter, d is wire diameter, N_a is number of active coils, which is an important parameter for the height of the assembly and G is the shear modulus of material. As the spring has squared-ground ends, number of total coils N_t is equal to:

$$N_t = N_a + 2 \tag{2}$$

So, according to the total number of coils (2), the shut height of the spring is determined. The spring is assembled in the retainer with initial deflection (approx. 20% of the working deflection) and clash allowance is taken (approx. 15% of the working deflection) to avoid reaching the shut height [9].

TABLE I
THE GEOMETRY PARAMETERS OF THE PLUNGER SPRING

d (mm)	D (mm)	C (D/d)	N_a	Min. Working Len. (mm)	Assembly Length (mm)
1,6	16	10	6,5	15,175	25,675

E. Locking Plate

The locking plate (Fig. 7) is formed as cylindrical shape. To increase the contacting area with the plunger the front side of the locking plate is milled. In the preliminary steps of design, the shape and the diameter of the locking plate is not determined. After the dynamic force analysis the final embodiment of this plate is made.
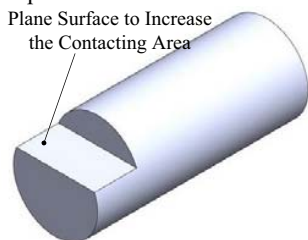

Plane Surface to Increase the Contacting Area

Fig. 7 Locking Plate

IV. EQUIVALENT SYSTEM MODEL OF SKIP-CYCLE MECHANISM

The main purpose of establishing dynamic models is to determine the forces that occur in moving mechanism. Creating lumped parameter model of complicated mechanism greatly simplify the mathematical calculations [10]. All the

mass which moves during the operation lumped together as m_{eff}, the elasticity of all parts as the spring constant k_{eff}. Coulomb friction, resulting between contacting surfaces, and the viscous damping, resulting from the shearing of a lubricant in the gap between the moving parts, are assumed negligible in order to simplify dynamic model and calculations.

The present valve actuating mechanism is modeled as a massless displacement for representing the position of the next component (Fig. 8). In the valve disabled mode, the system displacement represents the position of the plunger, which is coupled through the plunger spring. On the other hand, in the valve enabled mode, the system displacement refers to the position of skip-cycle mechanism which is coupled to the engine block through the valve spring.

All components are treated as lumped masses and all forces are considered to be coaxial and are not generating any moment.

Once all the components reduced to a single degree of freedom equivalent system, free-body diagrams of the locking plate and the plunger are gained and all the acting forces are attached, which are equal to the inertial forces of the components according to d'Alembert principle.

A. Lumped Parameter Dynamic Model

The system is separated individual parts as shown in (Fig. 8) and each of the significant components are represented with a lumped mass element [10]. The elasticity of individual elements is shown as springs that connected together in series. It is assumed that the rocker arm consists of two lumped masses at endings and is connected with a rigid, massless rod together. Furthermore, each mass has own displacement which are symbolized by y_1 to y_5.
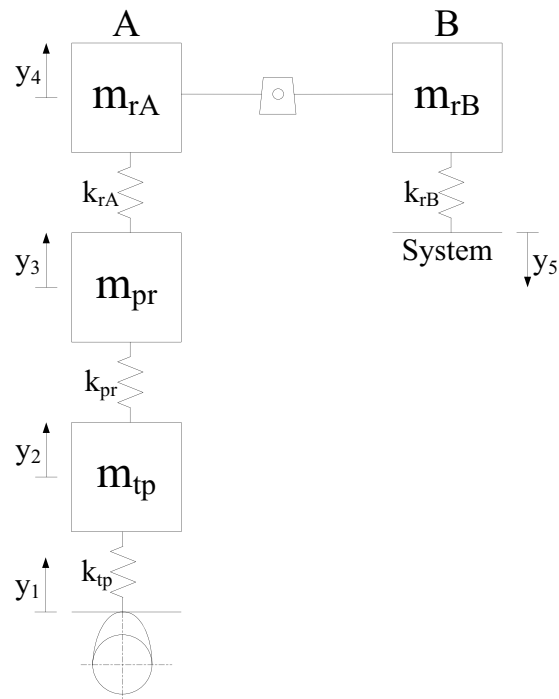

Fig. 8 Five-DOF model of the valve actutating system

B. Determining and Combining Masses and Springs

Spring elements, representing the elasticity of lumped masses, derived from the deformation formula.

Both tappet and pushrod are subjected an axial load, so the equation for the deformation and their spring rate k are;

$$k_{tp} = k_{pr} = \frac{AE}{l} \tag{3}$$

where A is known as cross-sectional area, l is length and E is the modulus of elasticity.

Any applied bending load will causes deflection on rocker arm. As it is lumped with two masses at each side, it can be modeled as it is double cantilever beam fixed at the bearing axis and unsupported at the other end [10]. The bending deflection of a rocker arm is calculated by the formula given:

$$k_{rA} = k_{rB} = \frac{3E\,\mathrm{I}}{l^3} \tag{4}$$

where I is the cross-sectional second moment of area of beam, l is its length, and E is the modulus of elasticity for the material.

Skip-cycle mechanism consists of two helical coil compression springs, valve and plunger spring. As the valve spring specifications were obtained from the manufacturer; its spring rate was directly used in calculations. However, the plunger spring is custom design and its geometry is defined by some parameters which are used for calculation and manufacturing purposes. The rate of a round-wire helical compression spring can be written as:

$$k_s = \frac{d^4 G}{8 D^3 N_a} \tag{5}$$

All the springs, which are connected in series to each other, have own displacement and subjected to the same force. So, the effective spring constant is found respectively for both sides.

$$k_A = \frac{1}{\dfrac{1}{k_{tp}} + \dfrac{1}{k_{pr}} + \dfrac{1}{k_{kA}}}$$

$$k_B = \frac{1}{\dfrac{1}{k_{kB}}} \tag{6}$$

All masses, on the left and right side of rocker arm, are connected in series to each other and have separate displacements. Similar lumped mass model of rocker arm is represented by two masses placed at cam side (A) and valve side (B). Statically equivalent model can be used as a solution, because the mass of the model is assumed equal to the original body and the center of gravity is in the same location as that of the original rocker arm [10]. The rocker arm was modeled as

3D in digital environment and the center of gravity was defined by means of CAD Software (Fig. 9). The moments of lumped masses according to pivot point should be equal, so the static equivalence can be written as:

$$m_{rA} + m_{rB} = m_r$$
$$ml_g + m_{rB}l_b = m_{rA}l_a \tag{7}$$

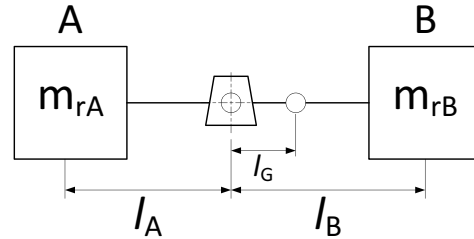Fig. 9 Statically equivalent lumped mass model of the rocker arm

So, the effective masses are found respectively for both sides.

$$m_A = m_{tp} + m_{pr} + m_{rA}$$
$$m_B = m_{rB} \tag{8}$$

All the masses and spring constants should gather at the valve side (B) of the rocker arm to model a single-DOF lumped parameter system for generating dynamic force analysis. The total mass m_A and the total spring constant k_A at cam side (A) are modified by the square of the lever ratio of the rocker arm (9). So, the effective masses and spring constants at point B is derived from the equation given below:

$$m_{eff} = m_B + \left(\frac{a}{b}\right)^2 m_A$$

$$k_{eff} = k_B + \left(\frac{a}{b}\right)^2 k_A \tag{9}$$

C. Dynamically Equivalent System for Valve Enabled and Disabled Mode of the Mechanism

As the locking plate and the plunger are newly designed, the forces acting on, are supposed to be calculated to prevent the mechanical failures.

The dynamically effective system of skip-cycle mechanism for the valve enabled mode is shown in (Fig. 10a), which is a conservative model as it keeps energy constant with no losses. y is the actual lift of equivalent mass and y_c is the rise of cam, which is not equal to y due to the linkage deflection. The free-body diagram (Fig. 10b) shows the acting forces on the SCM, which are the cam force symbolized by F_c, denoted depending on the time, the valve spring force F_{vs}, initial force of valve spring at assembly F_{vsi}, and F_1 is the contact force occurs between the mass of the system and the mechanism. Finally, the force occurred due to the spring constant of the system is represented by F_k.
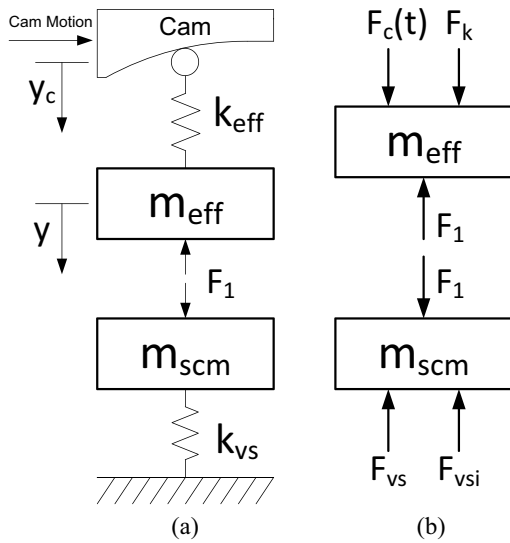
(a)   (b)

Fig. 10 Valve enabled mode of the skip-cycle mechanism: a) Dynamically equivalent model; b) Free-body diagram

Note that, in the valve enabled mode, the mechanism functions as a one single piece, so the acting net force on the locking plate is equal to the inertial force of the SCM.

According to Newton's second law, the equilibrium of vertical forces can be written for the effective mass of the system:

$$F_c(t) + F_k - F_1 = m_{eff}\ddot{y}$$
$$F_k = k_{eff}(y - y_c) \tag{10}$$

And for the locking plate equation can be written as given below:

$$F_1 - F_{vs} - F_{vsi} = m_{scm}\ddot{y}$$
$$F_{vs} = k_{vs}y \tag{11}$$

As both equations (10)-(11) are simultaneously solved the net force yields:

$$F_c(t) + F_k - m_{eff}\ddot{y} - F_{vs} - F_{vsi} = F_{net} = m_{scm}\ddot{y} \tag{12}$$

The mass properties of the newly designed components were estimated by means of CAD software. The mass of skip-cycle mechanism contains plunger, spring retainer, locking plate, half collets and also includes one-third of the actual effective weight of plunger spring.

$$m_{scm} = m_{sr} + m_{lp} + m_{pl} + 1/3\,m_{ps} + 2m_{hc} \tag{13}$$

Some modifications are required during constituting the dynamically equivalent model of the valve disabled mode. The mass of the plunger and the spring rate of the plunger spring, represented relatively as m_{pl} and k_{ps}, are attached to the system displacement of the original model (Fig. 8) and the

dynamically equivalent model for the valve disabled mode (Fig. 11a) and the free-body diagram of the plunger (Fig. 11b) are obtained.
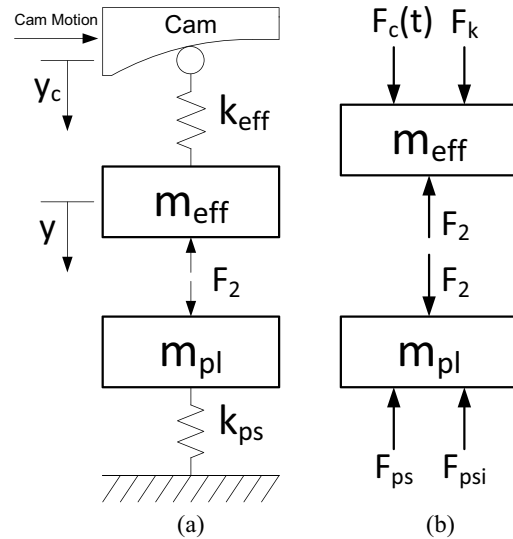

(a)   (b)

Fig. 11 Valve disabled mode of the skip-cycle mechanism: a) Dynamically equivalent model; b) Free-body diagram

The equilibrium of vertical forces can be written for the plunger as given below:

$$F_c(t) + F_k - m_{eff}\ddot{y} - F_{ps} - F_{psi} = F_{net} = m_{pl}\ddot{y} \tag{14}$$

D. Determination of Cam Profile and Actual Cam Displacement

The polydyne cam, which is defined by means of polynomial equation, was chosen as it generally preferred in highly flexible and high-speed mechanisms [11]. As long as manufacturing accuracy is justified, the polynomial curve shows excellent performance, because there is no sudden change in acceleration at the intersection of the dwell periods and the rise curve.

The 3-4-5 polynomial is consistent as the displacement equation. As simultaneously solved, the boundary conditions substituted in the general polynomial equation. So, the equations yield for unit maximum rise in a unit cam angle:

$$y = 1 - 10\theta^3 + 15\theta^4 - 6\theta^6$$
$$\dot{y} = -30\theta^2 + 60\theta^3 - 30\theta^4$$
$$\ddot{y} = -60\theta + 180\theta^2 - 120\theta^3 \tag{15}$$

The displacement formula has units in mm and the acceleration formula has units in millimeter/degree^2 (15). It is convenient to change the independent variable from the cam angle θ to the time t by multiplying the equation with $36s^2$, where s is the speed of the cam (rpm).

Furthermore, it is compulsory substituting the actual rise and cam angle to give the particular equation of skip-cycle mechanism. The conversion is accomplished by multiplying each term of polynomial by the ratio h/β^n, where β is the total

cam angle from initial point to maximum rise, and h is the total rise in cam angle β. The displacement formula is derived for the cam side (A) of the mechanism. Nevertheless, the displacement at the valve side (B) is different due to the lever ratio of the rocker arm (16).

$$y_A = hC_n \left(\frac{\theta}{\beta}\right)^n$$
$$y_B = hC_n \left(\frac{\theta}{\beta}\right)^n \left(\frac{l_B}{l_A}\right) \qquad (16)$$

The polydyne method is applied when designing the actual cam profile, as it cannot be assumed that the follower lift and the cam lift is the same due to clearances and the flexibility of the system. Designing the cam profile to give the desired follower action, polydyne method was derived by Rothbart [11] as given below:

$$y_c = \frac{L + F_{vsi}}{k_{eff}} + \frac{k_{eff} + k_s}{k_{eff}} y + \frac{m}{k_{eff}} \ddot{y} \qquad (17)$$
$$y_c = r_a + k_r y + c\ddot{y}$$

where L is the external load and zero for this case, F_{vsi} is the preload of the valve spring, r_a is the ramp height, which is the sum of the initial static deflection r_s and clearances of backlash in linkage r_k. Initial static deflection is negligible in calculation because it is small when compared with static deflection. Finally, k_r is the spring ratio constant, which is related to the spring constant of the system k_{eff} and the valve spring k_{vs}, and c is the dynamic constant.

V. RESULTS OF THE FORCE AND STRENGTH ANALYSIS

The dynamic force analysis was performed for both mode of the mechanism at a cam speed of 1500 rpm, which is the maximum speed of the Anadolu Motor's gasoline engine. The tappet raises 6.55 mm in with 70 degrees of cam rotation. So, the maximum acceleration at cam side was found approximately 650 m/s^2, however, at the valve side due to lever ratio of the rocker arm it increases to 1000 m/s^2. So, the completely reversed cyclic fluctuating loads are illustrated in (Fig. 12), which was calculated for the locking plate and the plunger.

A. Locking Plate

The plate is subjected to a completely reversed axial force of 150 N. As the actual shearing stress distribution is not uniform, the maximum direct shearing stress at neutral axis for cylindrical section shape is equal to:

$$\tau = \frac{4}{3}\frac{F_a}{A} \qquad (18)$$

where A is the cross-sectional area of the plate and F_a is the alternating load amplitude. Consequently, the shearing stress is found 6.4 MPa (18). As the locking plate is made from the same material of the spring retainer (AISI 1050, normalized at

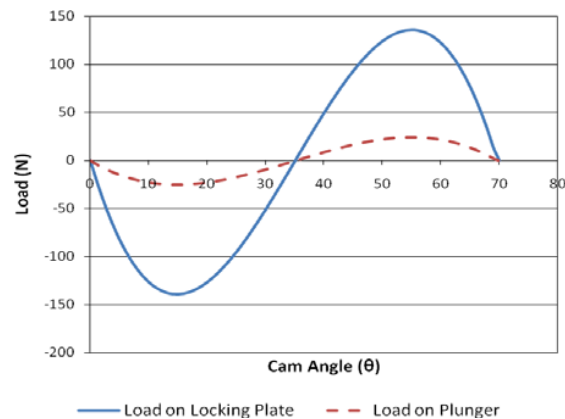900°C), the desired fatigue life is obtained according to the Smith diagram of the material [12].

Fig. 12 The fluctuating load acting on the plunger and the locking plate

B. Plunger Spring

As the spring generates a reaction force only proportional with maximum deflection and the spring rate, the purpose of the spring is not compensate the inertial force of the plunger. However, it is subjected to a fluctuating load and so; the shear stress in the coil is calculated according to the formula given;

$$\tau_a = K_w \frac{8F_a D}{\pi d^3} \qquad (18)$$

where K_w is Wahl's factor, which considers both direct shear stress and the stress concentration due to curvature, and F_a is the alternating load (18.25 N). The alternating shear stress in the coil found 300.5 MPa and consequently, the spring-wire has a torsional endurance limit which for peened springs is approximately 465 MPa (Table II) [13].

TABLE II
THE ENDURANCE LIMIT OF SPRINGS

S_{ew}'	310 MPa	for unpeened springs
S_{ew}'	465 MPa	for peened springs

VI. CONCLUSION & FUTURE WORK

A novel skip cycle mechanism for the Anadolu Motor's gasoline engine was presented. The design was focused on the leakage problem of the rotary valves, which especially affects the efficiency of the skip-cycle strategy at high speeds and loads. As the valves stay in closed position, an absolute sealing is obtained. Furthermore, this proposed mechanism allows to perform different skip cycle scenarios for different engine speeds and loads. Although this solution is proposed for a single cylinder engine, it could be adopted for the multi cylinder engines.

Future work will focus on the actuation of the locking plate because it is not considered as a problem of this study. The geometry of the plate could change depending on the new solutions of actuating mechanism. A linear solenoid, linear motor or a new cam mechanism may bring a satisfactory solution.

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