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Skip cycle system for spark ignition engines: An experimental investigation of a new type working strategy

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Abstract

A new type working strategy for spark ignition engine, named skip cycle, is examined. The main idea is to reduce the effective stroke volume of an engine by cutting off fuel injection and spark ignition in some of the classical four stroke cycles. When the cycle is skipped, additionally, a rotary valve is used in the intake to reduce pumping losses in part load conditions. The effect of this strategy is similar to that of variable displacement engines. Alternative power stroke fractions in one cycle and applicability in single cylinder engines are specific advantageous properties of the proposed system. A thermodynamic model, besides experimental results, is used to explain the skip cycle strategy in more detail. This theoretical investigation shows considerable potential to increase the efficiency at part load conditions. Experimental results obtained with this novel strategy show that the throttle valve of the engine opens wider and the minimum spark advance for maximum brake torque decreases in comparison to those of the classical operation system. The brake specific fuel consumption decreases at very low speed and load, while it increases at higher speed and load due to the increased fuel loss within the skipped cycles. In this working mode, the engine operates at lower idle speed without any stability problem; and moreover with less fuel consumption.

Keywords: Spark ignition (SI) engine; Skip cycle system (SCS); Part load; Power frequency; Pumping loss; Fuel consumption

1. Introduction

The importance of environmental impact and preserving the limited fuel resources increases with the growing number of internal combustion engines as power sources in transportation and different power generation units. One of the ways to preserve fuel resources is to increase engine efficiency, which means reducing fuel consumption, hence CO_2 emissions. The maximum effective efficiency of today's spark ignition (SI) engine is about (30–35)%. At full load, the improvement of the mentioned efficiency of the four stroke SI engine is thermodynamically limited. On the other hand, in part load conditions for the same engine, the pumping loss increases simultaneously with the decrease of compression/expansion work. This reduces the indicated efficiency significantly. The main reason for the pumping loss increase at part load is the restriction of the cross section area of the intake due to the more nearly closed throttle valve. Thus, increasing the efficiency in the full operation range of the engine currently is a challenging issue to be solved by researchers and engine manufacturers. Some methods to increase the efficiency at part load are variable valve timing (VVT) and lift, variable compression ratio, stratified charge lean burn engine, supercharging, variable displacement (cylinder disablement or cut-off) and/or use of combinations of the listed systems [1].

The system examined in this study is a new type of working strategy for SI engines, called the skip cycle [2,3]. The effect of the skip cycle strategy (SCS) is similar to that of variable displacement systems; however, the SCS is additionally applicable to single cylinder engines. There are some applications developed by other researchers, which have some similarities with the examined system in this study. Federenko et al. described and examined a fuel injection strategy

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in multi-cylinder engines that shuts off some of the injectors according to the load conditions [4]. A similar system is described and patented for six cylinder engines by Forster et al. [5]. A system with the addition of a hydraulic valve actuator was described and patented by Schechter [6]. Huang et al. developed a two stroke engine with a skip injection system to improve the irregular combustion at idle and light load [7]. Basshuysen proposed a hydraulic valve actuator system to control the engine load with expanding the four stroke strategy to eight and twelve stroke operation [8]. The same proposal is also made by Salber et al. with an electromagnetic valve actuating system [9]. In these eight and twelve stroke operations, one power cycle is followed by one skip cycle and one power cycle is followed by two skip cycles, respectively. A skip cycle is a four stroke frame without combustion and work production. On the other hand, eight and twelve stroke cycle terms are insufficient to describe the more complex SCS applications.

The SCS gives an opportunity to change the density of power strokes per revolution of the engine, which is applicable also even for changing the effective stroke volume of single cylinder engines. The proposed system is experimentally realized by controlling the injection, ignition and a rotary valve that is mounted behind and close to the inlet poppet valve [3]. This study covers the definition of some new terms, a thermodynamic model and experimental results. The objective is to show the potential of the SCS and some problems encountered in practice.

2. Extended cycle for spark ignition engine

The skip cycle strategy (SCS) is a new type working system for the SI engine. The aim is (1) to reduce the effective stroke volume of an engine at part load, skipping some of the four stroke cycles by cutting off fuel injection and (2) to decrease pumping losses, when a cycle is skipped by using rotary valves in the inlet and exhaust ports. As a result, the engine can be operated with wider opened throttle valve in lower power regimes. The SCS, besides having a similar effect as shutting off some cylinders in a variable displacement system, can be used in single cylinder engines and expands the multi cylinder engine control capability. For decreasing the active stroke volume in this system, only some cycles are shut off based on a given strategy, instead of absolutely shutting off some of the cylinders.

A short description of the skip cycle application in this study is as follows: Inlet and exhaust valves are driven with a classical camshaft mechanism. There are additional rotary valves placed on the inlet and exhaust channels with positions as close as possible to the poppet valves. These rotary valves are driven by solenoid actuators (in this experimental set up, the rotary valve is present only behind the inlet poppet valve). The throttle valve in the inlet side is used for precise control of the load (Fig. 1).

The rotary valves have only two positions (entirely open or closed) depending on whether the cycle is in normal (n) or skip (s) mode. The opening or closing actions of the rotary valves are accomplished while the poppet valves are in the closed position.

In the normal four stroke mode (n), the actuators hold the rotary valves in the fully open position. When the skip four stroke mode (s) is used, the actuators hold the rotary valves in the fully closed position during the gas exchange (intake or exhaust). Also, during this mode of operation, fuel injection is canceled.

One should know the valve events exactly in order to command the rotary valves. To achieve this, a detection system should be available. This can be done with an angle

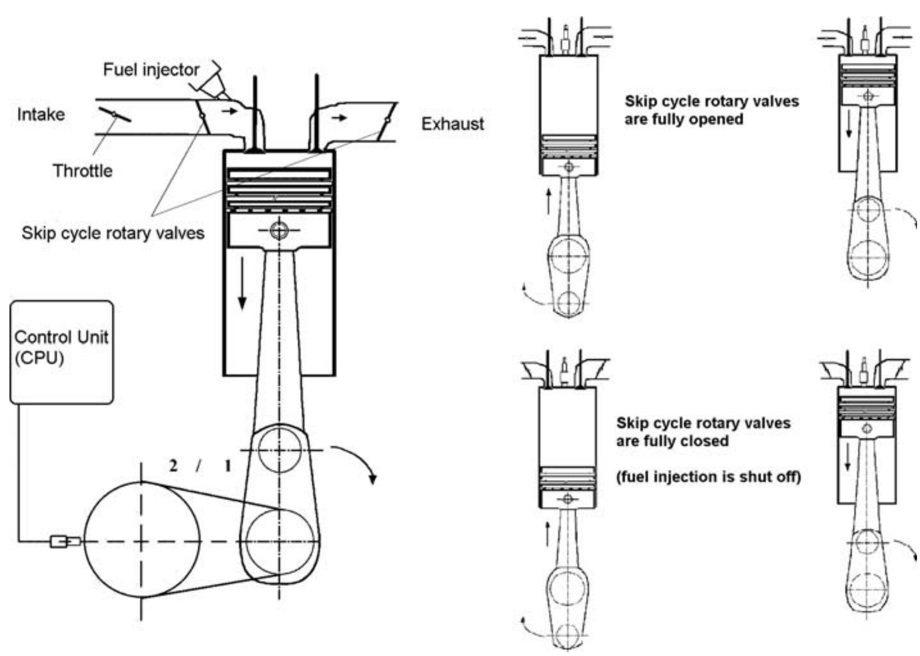

Fig. 1. Schematic view of skip cycle control system.

position sensor placed on a shaft that rotates at half the speed of the crankshaft or on the original camshaft. Rotary valve actuators, fuel supply and ignition systems are governed by an Electronic Control Unit (ECU) after determining the valve events.

When the required power is small, load control is done by skipping some of the four stroke cycles among the normal ones, and thus, the effective stroke volume is reduced. The overall torque and speed remain constant while the net indicated mean effective pressure (IMEP) increases. In the case of using the SCS, the throttle valve is opened wider than in the classical load control in order to achieve the same output.

Fig. 2 illustrates an engine cylinder pressure versus crank angle (p–α) representative diagram for a normal four stroke cycle and following a skipped four stroke cycle, i.e. for one period of the engine's cycle. The first part of the diagram represents the normal four stroke cycle (n). In this section, the rotary valves on the channels are fully open. In the second part of the diagram, the rotary valves are fully closed, and the four stroke cycle is skipped (s). The upper part of Fig. 3 illustrates another mode of a SCS, i.e. two normal four stroke cycles (nn) followed by one skipped cycle (s). The lower part of Fig. 3 shows one normal four stroke cycle (n) followed by two skipped cycles (ss).

Eight or twelve stroke terms are not adequate for describing the more expanded skip cycle strategies (Fig. 3). Therefore, it will be useful to present more descriptive parameters for the SCS.

The indicated power per cylinder is

$$P_{c,i} = W_{c,i} \cdot f_{\text{power}} \tag{1}$$

where $W_{c,i}$ is the net indicated work per cycle in one cylinder

$$W_{c,i} = \oint p\,\mathrm{d}V = \mathrm{IMEP} \cdot V_h \tag{2}$$

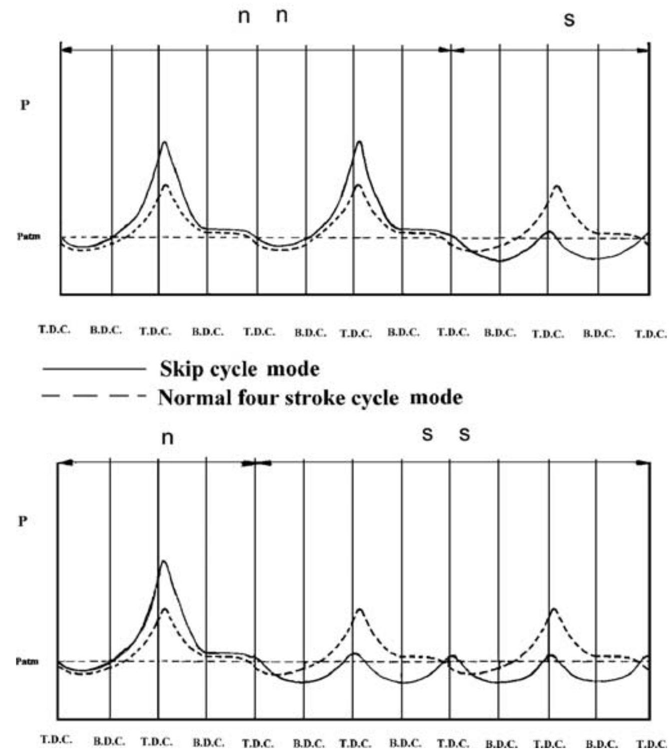

Fig. 3. Representative pressure-crank angle (p-α) diagrams according to nns and nss modes.

and the power stroke frequency f_{power}, Hz (s^{-1}), is defined by

$$f_{\text{power}} \equiv \frac{k \cdot n_e}{30} \tag{3}$$

IMEP is the net indicated mean effective pressure, MPa, V_h is the stroke volume of one cylinder L (dm^3), n_e is the crankshaft rotational speed, rpm (min^{-1}), and k is the work

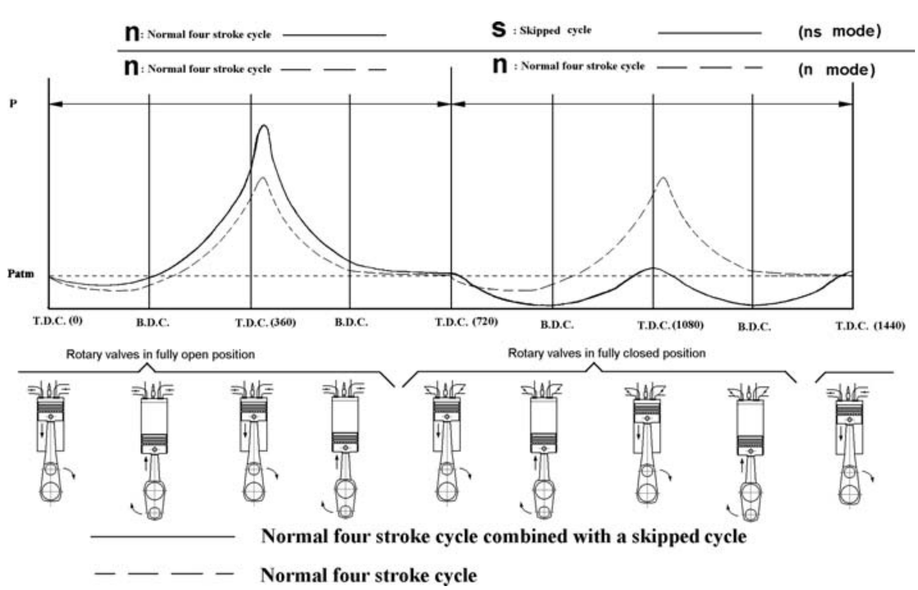

Fig. 2. Representative pressure–crank angle (p–α) diagrams for n and ns modes.

stroke fraction (work strokes divided by total strokes in one period of the engine's cycle).

$$k = \frac{\text{Work strokes in one period of the engine's cycle}}{\text{Total strokes in one period of the engine's cycle}} \quad (4)$$

For example, in a 2 stroke engine, $k = 1/2$; in a 4 stroke engine, $k = 1/4$; in a *ns* mode, $k = 1/8$ (Fig. 2); in a *nns* mode, $k = 2/12 = 1/6$; and in a *nss* mode, $k = 1/12$ (Fig. 3).

Thus, Eq. (1) for the indicated power per cylinder $P_{c,i}$, kW, with the above given variable units takes the form

$$P_{c,i} = W_{c,i} \cdot f_{power} = \frac{W_{c,i} \cdot k \cdot n_e}{30} = \frac{\text{IMEP} \cdot V_h \cdot k \cdot n_e}{30} \quad (5)$$

For a given stroke volume of an engine at constant speed and power (constant load), i.e. at constant IMEP per stroke (similarly to mean work per stroke)

$$\frac{30 \cdot P_{c,i}}{V_h \cdot n_e} = k \cdot \ \text{IMEP} = \text{constant} \quad (6)$$

Eq. (6) shows that IMEP and the k parameter are inversely proportional for a given stroke volume at constant speed and power. According to this equation, the relations between IMEP, the k parameter and the effective stroke volume for two stroke, four stroke and three types of SCS are given in Table 1.

Additionally, a thermodynamic model calculation is made in order to show the theoretical potential of the SCS [10,11]. This model represents pre-mixed homogeneous combustion in classical gasoline engines. All calculations are made using the first law of thermodynamics and the ideal gas equation.

Some assumptions used in the model are

(1) As in ideal cycles, the heat addition is at constant volume and the time factor is not included, but the beginning and end compositions are taken into account.
(2) The cylinder working medium is a mixture of ideal gases, the specific heats of which depend on the temperature.
(3) Heat losses in the intake and expansion processes are included by statistically determined polytropic exponents and heat losses (heat transfer and dissociation of combustion product) during combustion process by a heat utilization factor.

Table 1
The relation between IMEP, work stroke fraction and effective stroke volume for different working modes at constant speed and power of SI engine

Working mode	k	IMEP	Effective stroke volume
Two stroke	1/2	IMEP	Vh
Four stroke	1/4	2 IMEP	1/2 Vh
nns Mode	2/12	3 IMEP	1/3 Vh
ns Mode	1/8	4 IMEP	1/4 Vh
nss Mode	1/12	6 IMEP	1/6 Vh

n: normal four stroke cycle; *s*: skipped four stroke cycle.

(4) Heat addition (combustion process) is dependent on the working mixture and the molecular quantity changes during the combustion process.
The working conditions in the model are
(5) Modeled engine (Table 2).
(6) Relative air–fuel ratio is stoichiometric ($\lambda = 1$).
(7) Engine speed is constant ($n_e = 2000$ rpm).
(8) Load parameter is the volumetric efficiency and $k \cdot$ IMEP is (25–75) MPa.

The working cycle of the modeled engine is evaluated in terms of gross, pumping and net IMEP; and the indicated efficiency η_i. IMEP_{gross} and $\text{IMEP}_{pumping}$ represent production (compression/expansion) and consumption (exhaust/intake) parts of the cycle, respectively; and $\text{IMEP} = \text{IMEP}_{gross} - \text{IMEP}_{pumping}$. Fig. 4 compares the indicated efficiency and pumping fraction of the *n*, *ns* and *nss* modes of operation according to the load.

According to this model, the indicated efficiencies in the *ns* and *nss* working modes increase 34% and 52% at $k \cdot$ IMEP = 25 kPa; and 16% and 23% at $k \cdot$ IMEP = 75 kPa in relation to that in a normal four stroke cycle (*n*), respectively. These results are theoretical limits and in the case of brake mean effective pressure (BMEP), they will be lower due to the additional friction losses in the skipped cycles.

It is possible to control engine power in a wide range by extending the k values, i.e. with various SCS modes. These modes of SCS give an opportunity to minimize sudden changes in torque. Instead of using cylinder cut-off systems, it is possible to apply the SCS to each cylinder independently and to derive new strategies for multi-cylinder engines.

Table 2
Basic properties of the water cooled single cylinder four stroke research engine

Basic engine	Single cylinder 4 stroke water cooled
Number of valves (inlet + exhaust)	2
Displacement	454 cm^3
Bore × Stroke	(85 × 80) mm
Compression ratio	9

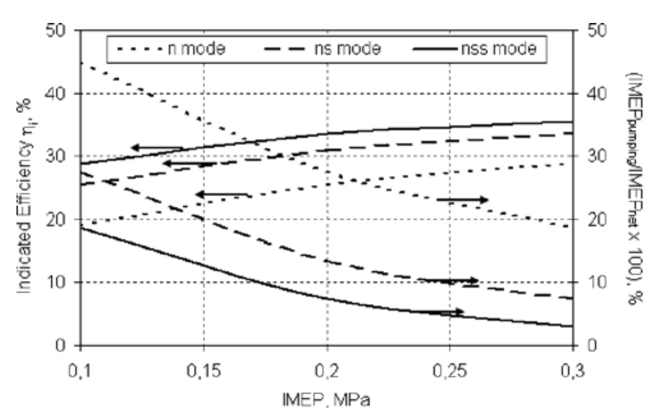

Fig. 4. Calculated indicated efficiency and pumping fraction at *n*, *ns* and *nss* modes of operation at part load.

3. Experimental facility

The experimental set up consists of a research engine, eddy current engine dynamometer, external engine cooling, PC based control and data acquisition and intake air flow, fuel consumption and exhaust emission measurement systems (Fig. 5). The main part of the experimental set up is the research engine. After a detailed investigation, a suitable commercial engine is found that can be used for developing the required research engine. The main properties of this engine are summarized in Table 2. Besides other constructive advantages, being water-cooled is the main reason for choosing this engine.

First, some constructive changes are made on the original engine (e.g. piston geometry, compression ratio, etc.). Although the engine's own internal lubricating system is preserved, an external water cooling system is designed to get reliable and accurate temperature control (Fig. 5).

Second, the fuel injection and ignition parts of a commercial gasoline automobile engine are adapted to this engine. Among the several intake pipe configurations that were tested, the one that directs the fuel injection cone

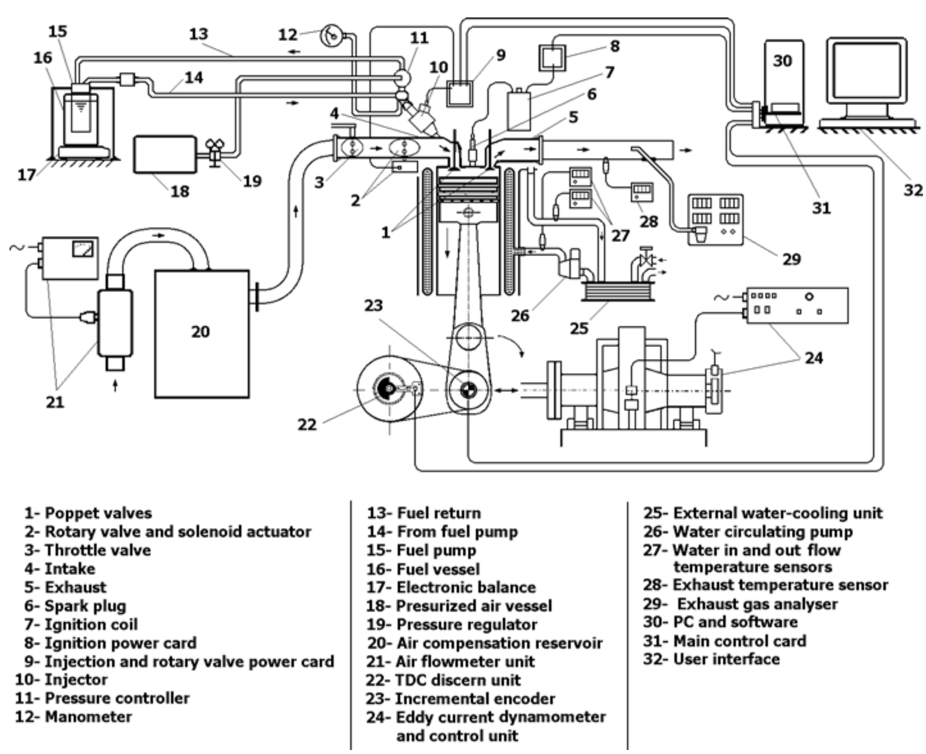

Fig. 5. Schematic view of control and measurement systems on the research engine.

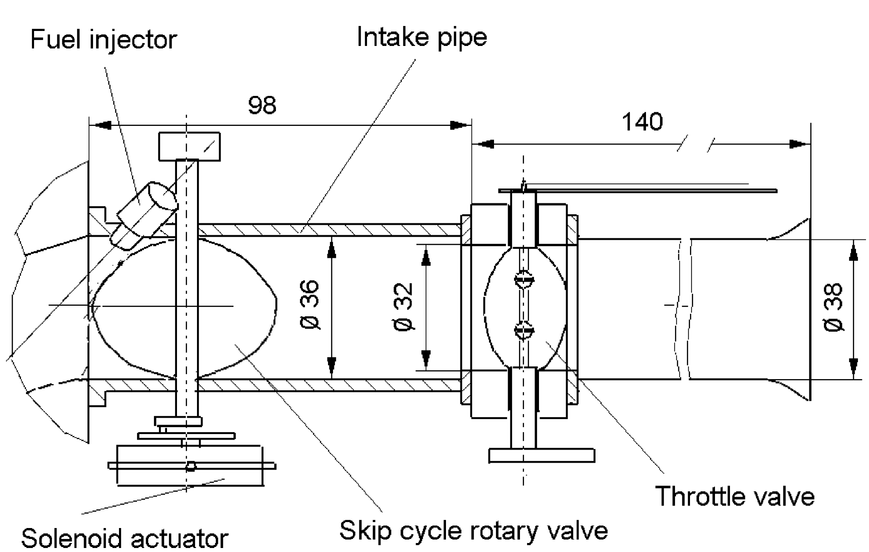

Fig. 6. Intake pipe of the research engine with additional rotary valve.

onto the back face of the intake valve is chosen (Fig. 6). Then, a rotary valve with a solenoid actuator is located in the intake pipe as close as possible to the inlet poppet valve. After that, an incremental encoder is mounted to the end of the crankshaft, and a photocell TDC observation system is mounted on the eddy current dynamometer with a ratio of 1:2. The signals from these two sensors (incremental encoder and TDC observation system) are necessary for the electronic control unit (ECU) to determine the exact cycle position of the piston in order to control the rotary valve actuator, fuel injection and ignition systems (Fig. 5).

Finally, the necessary hardware (base electrical circuit, main control and data acquisition card, ignition control card, injection and SCS card and transmission lines) and software (a user interface with C programming language) are designed for the control and data acquisition tasks, adapted to the research engine and integrated with a standard PC [12]. This PC based controller sets the timing and pulse width of the injection, the timing and dwell of the ignition and the timing and pulse width of the skip cycle rotary valve. During the experiments, (a) mass fuel consumption is measured by a sensitive electronic balance, (b) intake air flux is measured by a flow meter and (c) an exhaust gas analyzer is used for evaluation of the exhaust emissions for defining the air–fuel ratio and the lost of fuel from the previous injection and the air leakage from the rotary valve to the skipping cycles.

4. Experimental results

4.1. Rotary valve behavior and characteristics

The objective is to find the effectiveness of the rotary valve in eliminating the air flow into the cylinder at its closed position. Effectiveness means leakage prevention ability at the closed position of the rotary valve into the cylinder compared with the air charge amount of a normal four stroke cycle at the same throttle valve position and engine speed. The research engine was motored with a DC engine starter at about 420–460 rpm (without fuel injection). Airflow measurement is made with a flow meter mounted to a compensation reservoir that is connected to the intake pipe (Fig. 5). The air flow meter is the AVL EM 4031 type with a measuring range 20–150 l/min and an accuracy of ±5% of full scale. Experiments are done in several throttle valve opening positions with the *n* and *ns* modes. Measured and calculated data are listed in Table 3. The results show that air leakage at the skipping cycle was between (20–30)% of the air charge at a normal four stroke condition at the given motored engine speed, depending on the throttle valve position.

4.2. Engine performance at normal four stroke cycle mode

The working conditions for the whole engine map are at stoichiometric air/fuel ratio ($\lambda = 1$) and minimum spark advance for maximum brake torque (MBT). Because of our interest in part load, the engine tests and related map were performed for the low and middle range of BMEP and speed. BMEP was taken 0.1–0.6 MPa and at engine speeds of 1200, 1500 and 2000 rpm. The brake specific fuel consumption (BSFC) map of the engine related to the BMEP and engine speed is given in Fig. 7. The engine map is generally similar to the characteristics of today's engine. The results of these experiments are used to compare the *n* mode with the SCS.

The measurement devices used and their accuracy and the uncertainty data of the main measured or calculated parameters are given in Table 4.

The total uncertainties are calculated using the root sum squares method (RSS) with a confidence at the 95% probability level estimation (typical value for engineering calculations).

4.3. Engine performance with SCS at low load conditions

In this stage of the experiments, three types of skip cycle modes are tested. These modes are *nns*, *ns* and *nss*. The preliminary tests have showed that the skip cycle work conditions in the *nss* mode give significant results and noticeable potential. Engine tests in the *nss* mode are conducted at constant speed and power ($k \cdot \mathrm{BMEP} = 25$ kPa) and at three different engine speeds of 1200, 1500 and 2000 rpm. The results of the *nss* mode are compared with those of the *n* mode operation.

Table 3
Rotary valve characteristics at different throttle positions

Motored engine speed rpm	Throttle valve position (°)	Operation mode	Air flux l/min	Air charge per *n* mode l/(*n* mode)	Calculated leak air per skipped cycle l/(*s* mode)
460	60	*n*	92.5	0.402	–
438	10	*n*	76.0	0.347	–
435	5	*n*	58.0	0.269	–
418	3	*n*	43.0	0.205	–
				Air charge per *ns* mode l/(*ns* mode)	
464	60	*ns*	61.0	0.525	0.123
452	10	*ns*	52.5	0.464	0.117
450	5	*ns*	37.5	0.333	0.064
455	3	*ns*	32.5	0.285	0.080

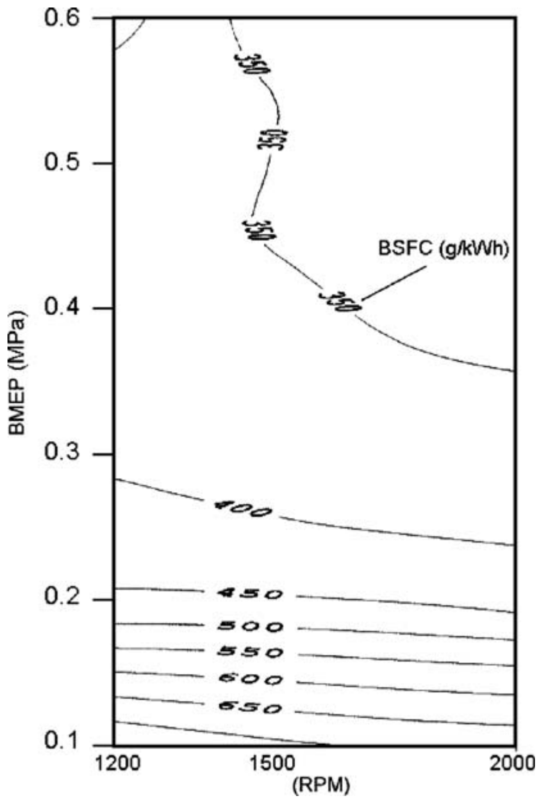

Fig. 7. BSFC map in n mode operation.

Fig. 8 compares the throttle valve opening positions between the n and nss modes at $k \cdot \text{BMEP} = 25$ kPa and 1200, 1500 and 2000 rpm engine speeds. In all of the experimental points, the throttle opening positions for the nss mode are much wider than those for the n mode. This means, for the nss mode, that the intake pipe pressure is higher, causing more air/fuel charge for the working cycles compared with that of the n mode, and thus, the load of working cycles at the nss mode is higher.

Fig. 9 compares the MBT spark advance between the n and nss modes at $k \cdot \text{BMEP} = 25$ kPa and 1200, 1500 and 2000 rpm engine speeds. In all of the experimental points, the MBT spark advance for the nss mode decreases significantly compared with that for the n mode. The decrease of the spark advance means that the combustion is completed in a shorter time, i.e. the flame speed is increased. The reason for this speed increase is that more air/fuel charge is inducted into the cylinder for the working cycle of the nss mode compared with that of the n mode. With the SCS, the cycle approaches the ideal Otto cycle in which the heat addition occurs at constant volume. This gives an opportunity to increase the indicated efficiency of the cycle.

Fig. 10 compares the BSFC between the n and nss modes at $k \cdot \text{BMEP} = 25$ kPa and 1200, 1500 and 2000 rpm engine speeds. The BSFC in the nss mode is improved at one working condition ($k \cdot \text{BMEP} = 25$ kPa; 1200 rpm; nss mode), which is approximately 11% better than that in the n mode operation. In the nss mode, with increasing engine speed, the BSFC increases in comparison to that in the n mode operation. The increase in BSFC at higher speed can be explained by the leakage occurring at the closed rotary valve during skipping cycles and by the problems in the mixture formation process. This can also be observed from the exhaust emissions.

By using the SCS, the HC and O_2 emissions increase while the CO and CO_2 emissions decrease significantly, compared to those of the n mode operation. Additionally, the exhaust temperature decreases significantly in the s mode compared to that in the n mode. These events are caused by two factors. First is the air leakage through the rotary valve at the closed position during the skipped cycles. Second is the insufficient fuel-air mixing and evaporation at the higher pressure conditions that causes a significant wall film ratio (deposition of fuel at intake pipe wall) of the injected fuel in the intake pipe [13,14]. The coupled effect of these two factors causes a loss of a fraction of the injected fuel from the working cycles to the following skipped cycles with the leaked air through the rotary valve. This leaked air–fuel mixture is exhausted without combustion in the following skipped cycle. The negative effects of these factors can be seen in the higher speed and load conditions, which require relative long injection times. The

Table 4
Experimental accuracy of main direct measurements and the uncertainty of the calculated experimental parameters

Measured or calculated parameter	Measurement device	Source of accuracy or uncertainty	Accuracy or uncertainty
TDC	Micrometer and goniometer	Reading error	±0.2°
Spark advance	Incremental encoder	Fitting uncertainty	±0.5°
Throttle position	Goniometer	Reading error	±0.1°
Engine torque	Eddy current dynamometer	Calibration error	±0.5%
	(Schenk W 70)	Reading error	±2.5%
		Total uncertainty	±2.6%
Engine speed	Incremental encoder (Leine & Linde 632006011)	Engine speed fluctuation	±1%
Fuel consumption	Chronometer	Reading error	±1%
	Electronic balance	Reading error	±1%
	(Shinko Denshi PF3000)	Total uncertainty	±1.4%
BMEP (RSS of Engine torque and Engine speed)		Total uncertainty	±2.8%
BSFC (RSS of Engine torque, Engine speed and Fuel consumption)		Total uncertainty	±3.1%

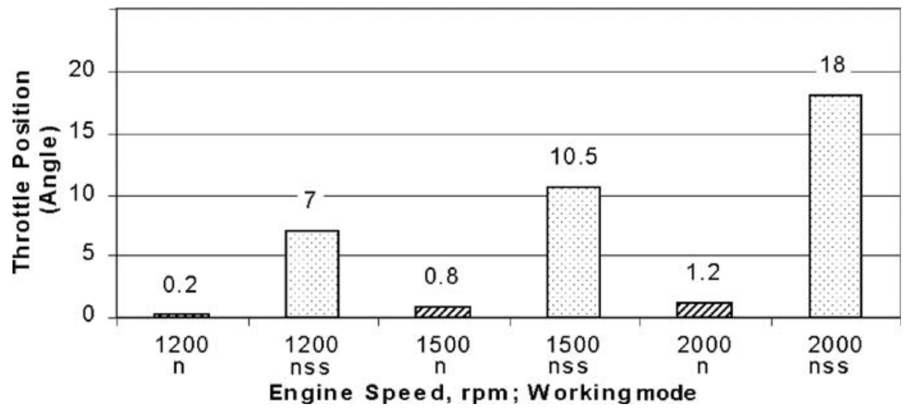

Fig. 8. Throttle positions (opening angle) in *n* and *nss* modes at 1200, 1500 and 2000 rpm and $k \cdot \mathrm{BMEP} = 25$ kPa.

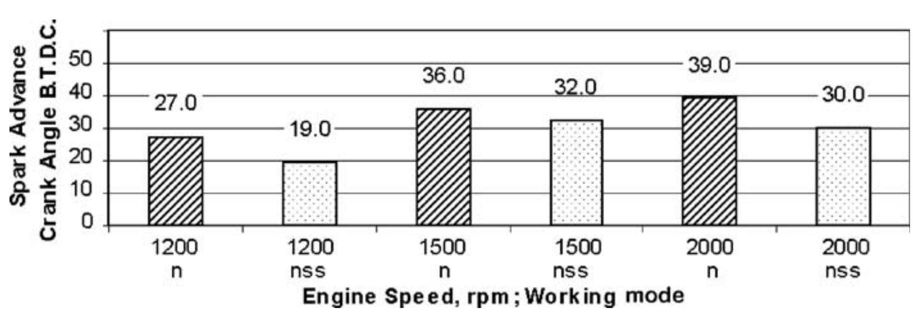

Fig. 9. MBT spark advance in *n* and *nss* modes at 1200, 1500 and 2000 rpm and $k \cdot \mathrm{BMEP} = 25$ kPa.

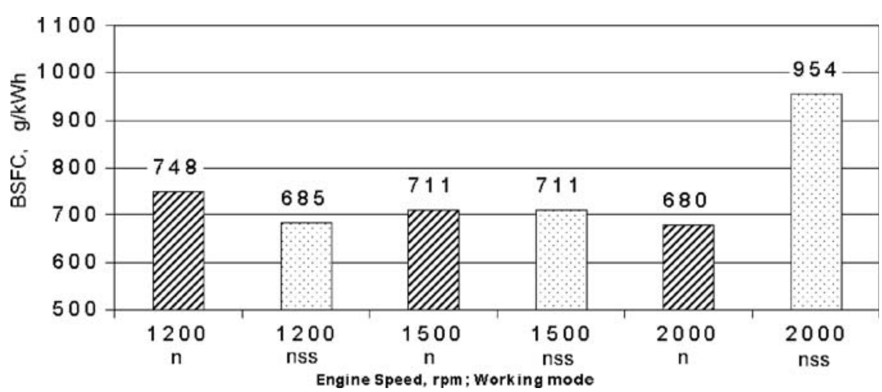

Fig. 10. BSFC in *n* and *nss* modes at 1200, 1500 and 2000 rpm and $k \cdot \mathrm{BMEP} = 25$ kPa.

efforts of developing the injection system with using injectors from a 2*L* four cylinder engine instead of a 1.6*L* four cylinder engine and increasing the injection pressure from 0.3 to 0.4 MPa was not enough to eliminate these negative effects. These are the disadvantages of the skip cycle rotary valve and injection technologies used in the experimental set up. The *nss* mode used in the tests gives an opportunity for dilution and lowering of fuel leakage by using two consecutive skipped cycles with the fuel injection cut-off. This is only a secondary solution to lower the lost fuel, which was also not enough to increase the efficiency at higher loads and speeds. These results show that the skip cycle control system (rotary valves and fuel injection system) play a primary role in the performance of the SCS.

4.4. Idle speed with and without skip cycle strategy

The result is that the SCS allows the engine to work at lower idle speeds without any stability problems. The simple stability criteria are to run an engine without any interference to the experimental system at maximum speed fluctuation of ± 50 rpm and at specified conditions for a minimum of 10 min. Ignition timing is set at a constant 10° before top dead center (BTDC) for all idle operation conditions. Fig. 11 indicates the advantage of the *nss* mode compared to the *n* mode to run at a significantly lower engine speed (44%), hence with lower fuel consumption (33%). It should be pointed out for idle speed that the leakage problems of skip cycle operation are the same as those

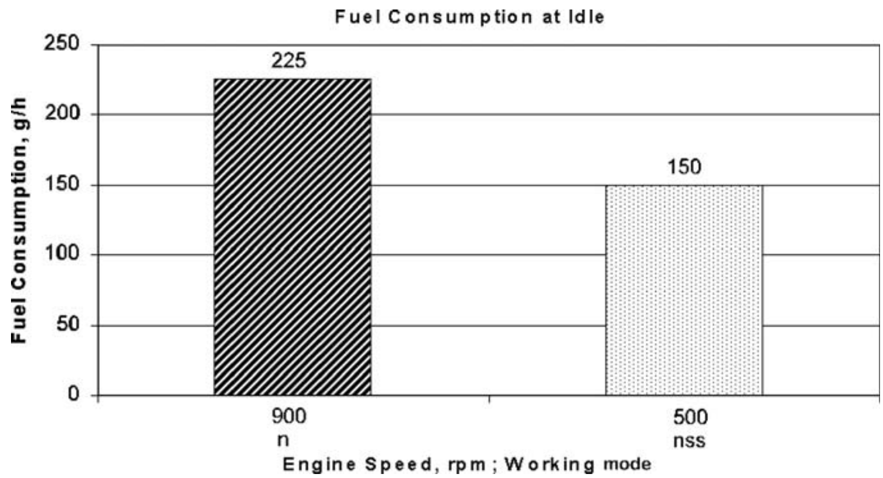

Fig. 11. Fuel consumptions in *n* and *nss* modes at idle speed.

at low load conditions. In spite of these leakage problems, the possibility of running at lower speeds in idle due to the more stable combustion conditions in the cylinder is superior for decreasing the fuel consumption compared to the *n* mode operation. Results show that, to run the engine at idle speed with the SCS gives an opportunity to reduce the cyclic variability of the combustion in the cylinder due to the higher amount of air/fuel charge taken into the cylinder in a working part of the *nss* mode compared to that in the *n* mode [15,16].

5. Conclusion and recommendation for future research

A new method named SCS is presented, modeled and tested in a single cylinder research engine.

The main research findings of this study are as follows:

1. Descriptive parameters for the SCS are defined: power stroke frequency f_{power} and work stroke fraction k.
2. A new thermodynamic model is developed, according to which indicated efficiency in the *ns* and *nss* working modes increases 34% and 52% at $k \cdot \text{IMEP} = 25$ kPa; and 16% and 23% at $k \cdot \text{IMEP} = 75$ kPa in relation to that in the normal four stroke cycle (*n*), respectively.
3. From the experimental results, in comparison to the *n* mode operation:
   (a) The throttle valve is opened wider in the *nss* mode. This indicates that the intake pipe pressure is higher; hence the fuel-air charge amount is more in the working part of the *nss* mode.
   (b) The spark advance is decreased in the *nss* mode. This is the result of the increase in combustion flame speed for the *nss* mode. High flame speed means a closer approach to the ideal Otto cycle, i.e. a better indicated efficiency.
   (c) The BSFC is decreased about 11% for the *nss* mode at very low speed and loads (1200 rpm, $k \cdot \text{BMEP} =$ 25 kPa).
   (d) The BSFC is increased for the *nss* mode at higher speeds and loads. This is due to leakage through the rotary valve.
   (e) The SCS allows the engine to work at lower idle speed (45%) without any stability problems, i.e. with 33% less fuel consumption.
4. The skip cycle control system (rotary valves and fuel injection system) plays a primary role in the performance of the SCS.

These results show that if the fuel and air leakage could be eliminated, it is possible to decrease the fuel consumption significantly at part load conditions of the spark ignition engine with SCS.

Future work recommendations:

(1) Place the rotary valves in the intake and exhaust ports as close as possible to the poppet valves and keep leakages to a minimum when the rotary valves are closed.
(2) Direct use of the poppet valves (instead of rotary valves) as the skip cycle control system for absolute solution of the fuel and air leakage problems [8,9].
(3) Construct a better fuel injection system with better fuel-air mixture formation (high pressure air assisted injection into the inlet pipe or direct injection into the cylinder).

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References

[1] Kutlar OA, Arslan H, Calik AT. Methods to improve efficiency of four stroke spark ignition engines at part load. Energy Convers Manage 2005;46:3202–20.

[2] Kutlar OA. A new method to decrease the fuel consumption at part load conditions of four stroke Otto cycle (Rochas) engine—skip cycle engine. PhD Thesis. Institute of Science and Technology, Istanbul Technical University, Istanbul, 1999 [in Turkish].

[3] Kutlar OA. Four stroke internal combustion engine with skip cycle system. World Intellectual Property Organization (WIPO), Patent application WO9963214, 1998.

[4] Federenko Y, Korzhov M, Fillippov A, Atamanenko N. Method and system for power control of internal combustion engines using individual cycle cut-off. Society of Automotive Engineers 1996; 960462.

[5] Forster HJ, Lubbig BE, Letsche U. Process and apparatus for intermittent control of a cyclically operating internal combustion engine. United States Patent, 4.509.488, 1985.

[6] Schechter MM. Skip cycle strategies for four cycle engine. United States Patent, 5.377.631, 1995.

[7] Huang HH, Jeng MH, Chang NT, Peng YY, Wang JH, Chiang WL. Improvement of exhaust emissions from a two stroke engine by direct injection system. Society of Automotive Engineers 1993; 930497.

[8] Basshuysen RV. Cylinder deactivation and fading out of individual cycles to the fuel economy and exhaust emissions. MTZ Motortechnische Zeitschrift 1993;54(5) [in German].

[9] Salber W, Kemper H, Staany F, Esch T. The electromechanical valve actuation system for future propulsion concepts—part 2. MTZ Motortechnische Zeitschrift 2001;62(1). in German.

[10] Arslan H. Minimum fuel consumption of Otto engine at part loads. PhD Thesis. Institute of Science and Technology, Istanbul Technical University, Istanbul, 2003 [in Turkish].

[11] Arslan H, Mehdiyev R, Kutlar OA, Çalık AT. A new calculation model for spark ignition engines in relation to different combustion mechanisms. In: 8th international combustion symposium, Ankara, Turkey, 2004.

[12] Calik AT, Kutlar OA, Arslan H, Akin A. Design and development of a low cost light duty single cylinder research engine with native technology. In: International scientific-technical conference on internal combustion engines, automobile techniques and transport MOTAUTO'01, Varna, Bulgaria, 2001.

[13] Nagaishi H, Miwa H, Kawamura Y, Saitoh M. An analysis of wall flow and behavior of fuel in induction systems of gasoline engines. Society of Automotive Engineers 1989; 890837.

[14] Servati HB, Yuen WW. Deposition of fuel droplets in horizontal intake manifolds and the behavior of fuel film flow on its walls. Society of Automotive Engineers 1984; 840239.

[15] Sztenderowicz ML, Heywood JB. Cycle to cycle IMEP fluctuations in a stoichiometrically fueled S.I. engine at low speed and load. Society of Automotive Engineers 1990; 902143.

[16] Ozdor N, Dulger M, Sher E. Cyclic variability in spark ignition engines—a literature survey. Society of Automotive Engineers 1994; 940987.