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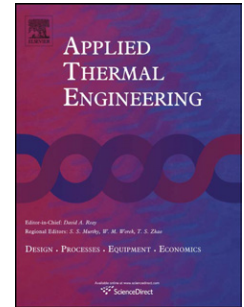


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Analysis of cycle-to-cycle variation and non-uniformity of energy production: tests on individual cylinders of a radial piston engine

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Keywords: radial engine, non-uniform operation, cycle-to-cycle variation, combustion, IMEP

ABSTRACT

The paper presents results of identification tests on radial piston engine non-uniformity and non-repeatability of energy production. The tests are a part of the ASz-62IR engine modernization project aimed at extending the engine's service life and decreasing fuel consumption. This is to be achieved by substituting a carburetor with an electronic injection control system. The tests consisted in measurements of in-cylinder pressure in each cylinder and were conducted under differing operating conditions. The paper focuses on the analysis of the indicated mean effective pressure (IMEP) as a parameter describing the amount of energy produced by the cylinders.

IMEP mean values indicate significant differences in the amount of energy produced by individual cylinders. A further difference can be observed in the cycle-to-cycle variation of individual cylinders; it consists not only in the non-uniform standard deviation of IMEP, but also in the non-uniformity of IMEP, represented by the forgetting factor. The range of cylinder-to-cylinder variations has been evaluated and attributed to the non-uniform supply with the air-fuel mixture – a result of using a carburetor and an ignition system with fixed settings.

1. INTRODUCTION

Piston engines have been used in aviation since its beginning. Despite the rapid development of jet and turbine engines, piston engines still have a considerable share in the aviation market, especially in the segment of small sport, agricultural and transport planes. This is due to their low running costs and relatively easy maintenance.

Previous attempts to improve the economy and uniformity of piston engines' operation consisted in optimizing the construction of crankshafts, pistons and connecting rods, cooling systems and carburetors. However, the main problem consists in the non-uniform operation of cylinders that results in considerable vibrations of the whole structure and, consequently, a shorter service life. Moreover, the non-uniformity of operation increases toxic emissions. As a result, it may be difficult to design an engine to comply with environment protection regulations – if, following the example of automotive industry, such regulations are enforced in aviation.

The non-uniform operation of individual cylinders is a feature shared by all multi-cylinder engines. It can be attributed not only to the cycle-to-cycle variations of each cylinder [1, 2, 3] but also to each cylinder's operating under different conditions. Considering the design of such engines, one can argue that following factors have an impact on the combustion process and performance of each cylinder:

1. Design details: dimensions of a combustion chamber, construction of pistons, crankshafts and connecting rods, spark plug positioning and the manner a piston moves [4].
2. The quality and quantity of the air-fuel mixture: its mass and composition, stratification inside the cylinder, the intensity and scale of turbulence, fuel atomization and vaporization level [5, 6, 7].
3. The quantity of residual combustion gasses [7, 8, 9] affecting mixture distribution in the combustion chamber, especially in the area of the spark plug, and the initiation of a new

combustion process; this is considered to be a “memory” element of the system that comes from the previous combustion process and affects the current one.

4. Combustion initiation, i.e. spark discharge, kernel and flame front development, extinguishing of flame on combustion chamber walls, spark advance etc [5, 10].
5. Temperature of the cylinder.

Any relative differences of those factors between cylinders result in a non-uniform operation [9].

In a radial engine, such relative differences are intensified. The reasons are listed below.

1. Piston and crankshafts in a radial engine are not identical: one cylinder is connected to the crankshaft by a single joint, whereas all remaining cylinders are connected by two joints.
2. Uneven distribution of the mixture among cylinders [12, 13] is inherent in multi-cylinder carburetor engines [11] as the mixture is being formed in a non-uniform manner in the stream of the air flowing through the inlet duct; in the case of radial engines, the cylinders are placed around the crankshaft, and their axes cross at one point. Air is supplied by a radial compressor, powered by the crankshaft through a multiplier. Although inlet pipes connecting the compressor with engine heads have identical shapes, they deliver the fuel and air mixture of a different composition as a result of gravity. This affects the fuel in both drop and film form.

As the mixture is distributed unevenly among the cylinders, different amounts of energy are generated in them during the combustion process. Consequently, each cylinder has a different temperature. Most radial engines are equipped with air cooling systems incapable of leveling these differences. Such conditions favor detonation combustion and, in extreme cases, may lead to damages to the engine. To prevent this, the mixture is being enriched. This decreases the temperature of the head. However, as the upper cylinders are supplied with the mixture of a proper composition, the bottom ones receive an excessively rich mixture.

The fact of non-repeatability of radial engines' operation and its effect on the quality of operation is independent of the engine model. For instance, tests conducted by Radial Engines, Ltd. on the 7-cylinder Jacobs R755B2 (275 KM) engines [14] to assess the non-uniformity of mixture distribution on the basis of the exhaust gas temperature analysis (Figure 1.1), indicate that this phenomenon causes:

- excessive fuel consumption,
- non-uniform power generation – e.g. the cylinder no. 7 (Figure 1.1) is supplied with the fuel and air mixture of the optimal composition, and it is the only cylinder producing maximum power of approximately 42 HP, while cylinder no. 3, supplied with a much richer mixture, produces only 33 HP ; the difference in power production is thus 25% [14],
- formation of carbon deposits in the outlet valves – the exhaust gas temperature is too low to burn them.

Tests presented in [15] confirm these observations. The authors point out that each cylinder generates a different amount of energy, with differences up to 40%.

Therefore a question arises: are the mixture supply differences the cause of cycle-to-cycle variation of individual cylinders? The literature on the subject claims that the change of mixture quality may have an impact on cycle-to-cycle variation [2, 16]. The issue to be settled is how the location of a cylinder influences non-uniformity and to what degree. The analysis of the non-uniformity and cycle-to-cycle operation of individual cylinders would facilitate understanding of the combustion process and operation conditions of individual cylinders [2, 3].

2. THE OBJECT AND SCOPE OF RESEARCH

The research is aimed at determining the non-uniformity of combustion in radial engine cylinders on the basis of indicated mean effective pressure (IMEP). IMEP is the most effective

parameter describing the amount of energy produced by the cylinder in a single cycle. The data obtained in the experiment allowed the author to determine both the cycle-to-cycle variation of individual cylinders and differences between them.

The tests are a part of the project to modernize the ASz-62IR carburetor-supplied radial engine manufactured by WSK PZL Kalisz, Poland, aimed at identifying operation parameters prior to construction changes. The research objectives included specifying the scope of non-uniformity of the air-fuel mixture in each cylinder.

2.1. TEST OBJECT

The object of tests was a radial air-cooled four-stroke nine-cylinder gasoline engine, mechanically charged by a radial compressor powered by the engine crankshaft. The total engine cubic capacity is 29.87 dm^3 , and the compression ratio is 6.4:1. The maximum take-off power is 1000 HP at 2200 rpm. The maximum fuel consumption is 280 kg/h. The engine is supplied with Avgas 100LL aircraft fuel by a four-choke downdraft carburetor, delivering the mixture in the form of air-fuel emulsion.

2.2. MEASUREMENT SYSTEMS

The engine was examined in a stationary test bed. The combustion process was monitored by means of M3.5x0.6 optoelectronic sensors by OPTRAND, placed in the spark plugs. The sensors detected changes of the intensity of light transmitted by two adjacent optic fibers, one connected to a LED, and the other to a photo detector. The change of the intensity of light received by the photodiode was caused by the change of the intensity of light reflected by a steel membrane deformed by the in-cylinder pressure.

The rotational speed and crankshaft position were measured by means of a magnetically induced shaft position sensor. The sensors were located in a purpose-built adapter that enabled determining the TDC position of the first cylinder.

The following signals were measured during the experiment: pressure in all nine cylinders, pressure in the selected inlet manifolds (by means of MPX250 sensors) and the position of the

crankshaft. The measurement of all the signals were conducted by means of a digital to analogue converter DAQ-Pad 6070E (National Instruments) at the frequency of 22kS/s per channel. The data were then processed by means of an original program created by the author.

2.3. THE SCOPE OF TESTS

The tests covered the engine operation in a steady state at various rotational speeds. The measurements were carried out at full load, i.e. at rotational speed n ranging from 1250 to 2200 rpm. The minimal rotational speed considered (1250 rpm) resulted from the operating range of a propeller blade regulator.

The tests were conducted in an engine test house. The temperature was maintained at the level of 30 °C. During the tests the atmospheric pressure was 100.5 kPa.

The excess air ratio in the considered measurement points was $\lambda = 0.90$. Its value was measured in a purpose-built exhaust manifold common for cylinders 2 and 3. The measurement was taken by means of a wide band lambda sensor LSU. The remaining cylinders were equipped with individual, 10 m long exhaust pipes providing a free outflow of exhaust gases.

For each engine operating point, 80 to 120 consecutive operating cycles were carried out. The difference in the number of cycles resulted from the testing method. The method required that the engine worked at a certain operating point for a specific fixed time (3 minutes). It was the shortest possible time for the engine to reach the steady state and then to measure fuel consumption. A longer time of measurement would damage the sensors, as they were exposed to extremely heavy operating conditions of high temperature, vibrations and air blasts.

3. TEST RESULTS

The cylinders were numbered clockwise starting from the top, as observed from the back of engine, i.e. looking from the direction of the supply system – this was also the direction the compressor rotated.

The results shown in Figure 3.1 do not present the full characteristic of cylinders 4, 5 and 8 as the pressure sensors in cylinders 4 and 5 failed during the tests, and temporary interferences occurred in cylinders 8 and 9. It can be observed that the individual cylinders differ from each other in the level of mean indicated pressure as well as the level of non-uniformity represented by a scatter of consecutive measurements.

The analysis focused on the pressure in the combustion chamber. The original method of analyzing the signal was described in detail in [17]. The following parameters were calculated:

- mean value of indicated mean effective pressure IMEP [MPa]:

$$\bar{y} = \frac{\sum_{i=0}^N y(i)}{N} \quad (1)$$

where $y(i)$ is the value of IMEP of the i -th sample, and N is the total number of samples (consecutive operation cycles at a particular operation point).

- standard deviation of indicated mean effective pressure [MPa]:

$$\sigma_y = \sqrt{\frac{1}{N-1} \sum_{i=1}^N [y(i) - \bar{y}]^2} \quad (2)$$

- forgetting factor of indicated mean effective pressure IMEP [MPa] [17]:

$$\xi = 0.8866 \cdot \frac{\sum_{i=1}^N |y(i) - y(i-1)|}{N \cdot \sigma_y} - 0.0029 \quad (3)$$

The constants in Equation (3) were established empirically and rescaled so that the value of the forgetting factor is between 0 and 1. This is to facilitate interpretation of the forgetting factor: a value of 1 represents entirely random conditions (no relationship between consecutive cycles), and 0 stands for direct relationship between consecutive cycles.

Figure 3.2 illustrates a mean value of IMEP – the mean of all the cylinders. It can be observed that this characteristic, in contrast to typical characteristics of car engines at full power, does not have an extreme. Indicated pressure increases together with rotational speed. The lack of extremes (inflection of the characteristic) indicates that the engine may produce more energy but it is limited by the characteristic of propeller resistance.

This is due the fact that the propeller resistance characteristic curve crosses that of the engine torque at full load (Figure 3.3) below the speed corresponding to the maximal working point. In order to reach a higher torque and power, a propeller with lower resistance and with a smaller outer diameter could have been used for the test. However, considering the through-plane velocity of an aircraft powered by such engines, the propeller is optimally matched.

Figure 3.4 shows the mean indicated pressure at different rotational speeds. Even though values for cylinders 4 and 5 are missing, it can be observed that the lowest pressure occurs in cylinder 9 and the highest – in cylinder 3. The shift of the characteristic towards cylinders 2 and 3 occurred at each tested value of rotational speed. Unfortunately, a failure of two sensors prevented determining if the shift of characteristic applied also to cylinders 4 and 5. It can be observed that the characteristic bends distinctly at cylinder 9 at the velocity below 1600 rpm.

The shift of IMEP curve (corresponding to energy production curve) towards the right bottom cylinders (3-5) is illustrated in Figure 3.5. The mean values of energy generated in particular cylinders differ significantly. Cylinder 9 generates only 60% of the energy produced by cylinder 3. Inevitably, these differences contribute to engine vibration.

Figure 3.6 presents the standard deviation of IMEP, a mean of all the cylinders, in the function of engine rotation speed. The mean standard deviation, in contrast to the mean indicated value, has its extreme, the minimum, at the speed of 2000 rpm that corresponds to the full power used during flight. Moreover, the mean standard deviation is relatively small in the range of 1800-2100 rpm, which can be attributed to the optimization of spark advance angle (control parameter) and an engine design (object of control) to the range of speeds used in basic operation of the aircraft.

Standard deviation of IMEP in individual cylinders is shown in Figure 3.7. For low rotational speeds (1200 and 1400 rpm) the highest values occur in cylinders 2 and 6, just as in the case of IMEP (Figure 3.4). However, at higher speeds, the maximum of standard deviation is shifted. At speeds above 1600 rpm, the lowest standard deviation occurs in cylinders 2 and 8. In the case of the rotational speed below 1600 rpm, the standard deviation for cylinders 2 and 8 is 30 % higher than cylinder 9. This tendency changes at higher speeds: standard deviation increases for cylinders 1, 3, 6 and 9 and decreases for cylinders 2, 7 and 8. Therefore, the characteristic of variability at speeds above 1600 rpm turns through an angle of 90° relative to the characteristic at speeds below 1600 rpm. The greatest relative difference of standard deviation occurs at 1900 rpm between cylinders 2 and 9, and equals 50%.

Considering the mean standard deviation for each cylinder (a mean of all operation points of each cylinder), shown in Figure 3.8, it can be observed that there is no evident pattern of differences between the cylinders. The value for cylinder 8 is low because the sensor was damaged at low rotational speed, where standard deviations are the highest. This affected the cylinder 8 mean value. The differences between the remaining cylinders are about 12%.

Figure 3.9 and 3.10 show values of forgetting factors at 1900 rpm and 1400 rpm together with return maps for the individual cylinders. Scatter of points in the return maps differs from cylinder to cylinder - not only in terms of concentration, but also in terms of location of the cloud of points. The latter is related to the difference in mean value of IMEP. Lower concentration of points means higher standard deviation. The shape of scatter depends on the “memory” of the previous combustion cycles in a cylinder.

In the case of cylinders 1 and 6 at 1900 rpm (Figure 3.9), the scatter of points forms a circle, which points to the relationship between IMEP in consecutive combustion cycles being weak with domination of a random element. Therefore, the value of forgetting factor is close to 1. For cylinders 8 and 2, the points form an oval. This indicates lower values of the forgetting factor and higher dependence between the consecutive combustion cycles. Cylinders on the right bottom part of the engine are characterized by forgetting factors closer to 1. It could be expected that the maximum values of forgetting factor occur in cylinders 4 and 5.

Forgetting factor values at a lower speed are shown in Figure 3.10. In general, for all the cylinders, values of the forgetting factor dropped significantly with the decrease of the rotational speed (the scale of the radar chart is different than in Figure 3.9). The shape of clouds of points in the return maps confirms this phenomenon – the points form narrow ellipses, arranged along the axis $y=x$. This is particularly evident in the case of cylinders 2 and 7, whose forgetting factors are below 0.4.

Figure 3.11 presents mean value of forgetting factor (mean of all the cylinders) according to rotational speed and proves that the level of forgetting factor depends the speed. High values occur in the range of 1700 – 2000 rpm, i.e. the basic area of operation of the aircraft. This confirms that the engine has been optimized for this range of speeds, as was concluded in the analysis of standard deviation of IMEP. The maximum of the forgetting factor (0.81) indicates that the combustion process cannot be considered random, and each cycle depends on the combustion process in the previous cycle.

Figure 3.12 presents the forgetting factor values according to cylinder and rotational speed. It can be observed that at higher speeds (1900 and 2100 rpm) there is little difference in forgetting factor values of individual cylinders. At the rotational speed below 1900 rpm, the forgetting factor reaches higher values in cylinders 6 and 3, which suggests the shift of the characteristic towards the right bottom cylinders (4 and 5). This can be clearly seen in the case of a mean value calculated from all the speed values shown in Figure 3.13.

4. DISCUSSION

The mean indicated pressure test demonstrates that power production in the radial carburetor engine varies according to a cylinder. The differences between the cylinders are as high as 40 %. More power is generated by the cylinders at the right side of engine (looking from the back).

A question arises: what these differences may be attributable to. As described in the first chapter, the reason may be the design of the engine a difference in the quality and quantity of the air-fuel mixture. In the case of the analyzed ASz-62IR engine, the differences in construction

of particular cylinders are negligible, as the elements of each cylinder, head, piston, valve, inlet pipes are fully interchangeable. The only difference consists in the mode of movement of the crankshaft-piston assembly: cylinder no. 1 requires one articulation joint, the remaining cylinders – two joints. This causes a change in the characteristics of the piston movement, apparent in the shift of the cylinders' upper dead centers, shown in Figure 4.1. However, the distribution of shifts is different from the distribution of energy production in the cylinders (represented e.g. by differences in mean indicated pressure shown in Figure 3.4).

The ignition system is also to be excluded from the list of potential causes of differences in the cylinders' energy output. The engine is equipped with a magneto ignition system common for all the cylinders, and the energy and ignition angle are the same for all the cylinders.

Therefore, the shift of the power generation characteristic results from differences in composition of the air-fuel mixture delivered to the cylinders. On the basis of the test results, it can be claimed that a richer mixture is forwarded to cylinders 2 and 3, whilst cylinders 6, 7, 8, 9 and 1 receive a leaner one.

This shift results from the design of the fuel supply system. The fuel is delivered to the engine from the carburetor mounted in front of the compressor. The intake of the air-fuel mixture is located in the area of cylinder 1, but above the axis of the engine. This location and the direction the compressor rotates explain why more fuel is delivered into the cylinders first in line (2, 3 and 4) and why cylinder 9 receives the smallest portion of the fuel. This is the case regardless of the engine speed. However, at speeds lower than 1600 rpm, the shift of power generation characteristic is more visible. At the speed of 1250 rpm, cylinder no 9 generates only 30% energy generated by cylinder 3.

The difference in the mixture composition affects also the standard deviation of IMEP. However, on the basis of Figures 3.7 and 4.1 one can observe that the standard deviation depends also on the geometry of the piston-crankshaft assembly. As underlined in the first chapter, the geometry significantly affects the cycle-to-cycle variation. In the case of the tested engine, the difference in the piston's mode of movement changes the rate of pressure build-up during compression, and thus the mixture temperature at the moment of ignition (a change of heat release time at compression). This fact is a cause of the standard deviation being the greatest

for cylinders 3 to 5 (presumably – as a sensor failure prevented full measurement), and the least for the cylinder no. 8

The highest relative cylinder-to-cylinder differences in standard deviation occur at high rotational speeds (here above 1600 rpm), which is typical of all types of internal combustion engines [16]. At lower speeds, cylinders 8 and 9 (and probably 4 and 5) operate smoothly. At higher speeds, lowest standard deviation occurs in cylinder 2 and 8. This difference according to speed range is attributable to the change of combustion process conditioned by ignition optimized for speeds higher than 1600 rpm.

Standard deviation depends greatly on rotational speed. For speeds ranging from 1800 to 2100 rpm., standard deviation of IMEP is low, which indicates low fluctuation of combustion processes in a cylinder, as can be gathered from Figure 3.6. This results from the optimization of the combustion chamber design and ignition settings to fit this speed range: the engine is equipped with a mechanical ignition system with a constant spark advance angle, independent of engine speed and load. This means that spark advance is adjusted only to a narrow range of engine operation (through flow speed of 1900 rpm).

These fixed engine settings cause ignition delay at low speeds. As a result, the combustion process is non-uniform and affected by the previous cycles. This is expressed also by the forgetting factor which drops below 0.5 at low rotational speed.

Values of forgetting factor prove that residual gases have a “positive” impact on the combustion process. The residual gas, which remains in a cylinder from the previous combustion process, supports the mixture preparation and formation in the cylinder and its correct distribution in the area of spark plugs (the engine is equipped with two spark plugs per cylinder). Therefore, a proper combustion in the previous cycle improves the process in the next cycle. However, if combustion does not proceed in a correct way because of random effects, the next process may fail. This is of great importance when the spark advance angle is not optimal [3] which occurs at lower rotational speeds.

5. CONCLUSIONS

The author claims that:

1. There are significant differences in the operation of individual cylinders of a radial engine working at full power, these include mean pressure values (and thus the energy generated by each cylinder and the engine as a whole) as well as cycle-to-cycle variation.
2. Relative differences of IMEP are up to 40%, and distribution of IMEP in individual cylinders indicates that the leanest mixture flows to cylinder 9 and the richest one to cylinders 2 - 6.
3. Non-uniform operation of the individual cylinders, represented by the standard deviation of indicated pressure, depends on the rotational speed. The non-uniformity is the lowest at the speed range of nominal engine work as a result of an ignition system design.
4. Standard deviation of IMEP demonstrates dependence on location of the cylinder. The distribution of the cylinders' standard deviation shifts relative to the rotational speed: a change occurs at speeds higher than 1600 rpm.
5. Another measure of cylinder-to cylinder differences in the cycle-to-cycle variation is the forgetting factor. Distribution of its values indicates that the previous combustion process affects the next one stronger in the upper cylinders than in the bottom ones.
6. All these differences between individual cylinders result from uneven mixture distribution and the engine design.

In conclusion, it is argued that a carburetor supply system is evidently the source of a considerable non-uniformity of fuel supply. This results in a higher level of engine vibration caused by its non-uniform operation (during a single engine rotation) as well as the increased cycle-to-cycle variation of the individual cylinders. For this reason, it is essential to use an individual supply system for each cylinder to decrease the non-uniform operation. This can be achieved only by means of an electronically controlled multi-point injection system. Such a system would deliver an appropriate amount of fuel directly to the inlet manifold of each cylinder, irrespective of the factors related to gravity and a blending process in the compressor system. The implementation of such a system would extend the engine's service life.

On the basis of test results the author claims that a similar non-uniformity of operation is characteristic of any radial engine. This is due to the mode of air flow and fuel supply method. It can be also concluded that the distribution of cycle-to-cycle variation of IMEP as described in the paper is typical for radial piston engines in general and attributable to the piston's mode of movement.

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Fig. 1.1. Temperature of exhaust gasses of the individual cylinders in the JACOBS R755B2 engine (cruising power, $n = 2000$ rpm, $h = 600$ m above the sea level, maximally enriched mixture) [14]

Fig. 2.1. Test stand. ASz-62IR Engine. Front view.

Fig. 3.1. Indicated pressure for the individual cylinders - full load $n=1900$ rpm

Fig. 3.2. Full power characteristic of mean indicated effective pressure from all the cylinders.

Fig. 3.3. Engine torque characteristic at a maximal working point of an engine and a propeller resistance moment

Fig. 3.4. Mean indicated pressure for the cylinders at specific rotational speed.

Fig. 3.5. Mean indicated pressure value for the individual cylinders from all full power characteristics.

Fig. 3.6. Standard deviation of IMEP in the function of engine rotational speed.

Fig. 3.7. Standard deviation of IMEP for the individual cylinders at different rotational speed

Fig. 3.8. Standard deviation of IMEP for the individual cylinders out of all full power characteristics.

Fig. 3.9. Forgetting factor value and return map for the individual cylinders – a full power characteristic $n = 1900$ rpm

Fig. 3.10. Forgetting factor value and return map for the individual cylinders – a full power characteristic $n = 1400$ rpm

Fig. 3.11. Mean forgetting factor value in the function of rotational speed

Fig. 3.12. Forgetting factor for the individual cylinders at different rotational speed

Fig. 3.13. Mean forgetting factor value for the individual cylinders from a full power characteristic.

Fig. 4.1. Angle difference of TDC for the individual cylinders (cylinder 1 = 0)

