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LEAN-BURN ENGINE – POTENTIAL ANALYSIS

A. KOWALEWICZ*

Technical University of Radom, Al. Chrobrego 45, 26-600 Radom, Poland

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ABSTRACT–Analysis of the thermodynamic cycle of IC engine from the point of view of economy and emissions was carried out. From this analysis potential capability of engine development was derived. This potential capability is lean-burn engine, fuelled with homogeneous mixture with $\lambda \geq 1.4$. Several different modes of fuelling were proposed and tested on one-cylinder test engine from the point of view of extending lean operating limit of the engine, emissions and fuel economy. Among them were: fuelling with evaporated preheated gasoline, with gas (LPG evaporated) and with liquid butane. From these modes, fuelling with liquid butane injected to inlet port was selected and finally tested. This novel system of fuelling offered better than standard engine performances and emissions at lean operating limit. These results were validated on full-scale two-cylinder engine.

KEY WORDS : Lean operating limit, Performances, Emissions, Combustion, LPG

1. INTRODUCTION

Further development of CI engines is enhanced by restrictions imposed upon both emissions and fuel economy. These demands can be fulfilled neither by present SI engine supplied with catalyst nor by CI engines with exhaust gas recirculation and with particulate trap. Controlled combustion and decreasing of toxic components of combustion products *in statu nascendi* seems to be a right way of reducing emission and fuel consumption with simultaneous reducing emission of greenhouse gas as far as SI and CI engines are concerned. Although SI and CI engines preserve some inherent features, their development trends coincide: lean burning and more and more homogeneous mixture are concidered to be potential capabilities of engine development. They may be derived from the following analysis.

2. THERMODYNAMIC CONSIDERATIONS

A potential capability of engine development results from analysis of the thermodynamic cycle of the engine itself (Kowalewicz, 1998). This analysis enable to show energy losses, which may be divided into two groups: energy losses due to differences between actual and theoretical thermodynamic cycle and due to differences between actual working gas and ideal gas assumed in theoretical cenciderations.

**Corresponding author.* e-mail: kow-

As it is well known, unless there is severe limit of maximum available pressure, Otto cycle has the best efficiency. As far as Otto cycle is concerned, the greatest difference between theoretical (Otto cycle) and actual cycle of IC engine depend on time of heat conduction (in actual engine – combustion time) which is infinitely short in Otto cycle, while in actual cycle is equivalent to ca 30 deg CA. In order to increase burning rate of the charge and in that way increase thermal efficiency of the engine, the mixture should be as well prepared for combustion as possible. One should take into consideration that combustion itself may have rather character of thermal explosion (e.g. ATAC concept based on Onishi *et al.* approach: Onishi *et al.*, 1979), than sequential combustion of the mixture layers in flame spreading from the spark as in conventional engine.

The efficiency of Otto cycle depends only on CR and adiabatic exponent k. The higher the CR, the higher the efficiency (for constant k). However, due to knock and limited strength of engine structure, CR is roughly limited. Also, the higher the adiabatic exponent k, the higher the thermal efficiency. Adiabatic exponent is dependent on features of working gas: the more is it diluted by air, the higher the exponent k (for air k = 1,4) and the higher the efficiency. Work of expansion is higher if number of moles of gases increases during combustion, what depends on fuel chemical composition (e.g. for alcohols, hydrocarbon fuels etc. number of moles during combustion increases).

In conclusion of thermodynamic analysis, thermal

efficiency may be increased by means of:
- Increase of CR, which is however limited,
- Increase of burning rate as much as possible.

As far as the problem of emissions is concerned, it is well known, that the higher A/F ratio, the lower the emission of NO_x, CO and, to some extend, HC. The barriers of mixture leaning are (Różycki, 1999):
- Increase of cycle-to-cycle variations,
- Increase of HC emission.

However, leaning of the mixture also results in decrease of engine torque.

Summing up the above analysis, in the case of CR = idem, the increase of efficiency and decrease of emissions are enhanced by:
- Fuelling with lean mixture,
- Rapid combustion,
- Properties of fuel which may increase efficiency and decrease emissions.

Some of the means of increasing burning rate and some concepts of fuelling are presented in this paper.

3. OBJECTIVE OF THE WORK

Main objective of the investigation was to make an attempt to answer for the questions:
- How to lean burn?
- What measures enhancing lean burning may be applied?
- To what extend the mixture may be leaned?

The work was carried out in four stages.

In the first, preliminary stage, the different measures of fuelling were investigated. In the second stage the best mode of fuelling (i.e. with gas butane) was tested on account on ignition timing. In the third stage the gas fuelling was improved: not gaseous butane, but liquid butane was injected to the inlet port of the engine. In the last, fourth stage experiments were carried out on full scale production engine (two-cylinder engine), while in previous stages only on one cylinder engine (the inlet duet of the second cylinder was chocked). The course of experiments is summarised in Table 1.

4. TEST STAND

Experiments were carried out on the test stand shown in Figure 1. As an experimental engine, the engine of Fiat CINQUECENTO 704 was used, of which only one cylinder worked, while the inlet channel of the second cylinder was choked. The engine (1) was connected to the eddy-current dynamometer Vibrometer 3WB15 (2). Air was supplied to the engine through the flowmeter (6A), surge-tank (4), pipe (5) and carburettor (3). Liquid fuel was supplied to the carburettor (3) from the tank (9); the fuel flowrate was measured with the use of volumetric method (11). In the case of fuelling with fuel vapour, the fuel was injected into the vaporiser (19) heated by exhaust gases and evaporated. The fuel vapour was supplied to the engine through the pipe (22). Emissions were measured as follows: NO_x with Beckman analyser Model 951 (16), hydrocarbons with Beckman Model 402, CO, CO_2 and O_2 with AVL Analyser DiGas 465 (which measured also additional hydrocarbons and air excess coefficient, λ (17)). Pressure in the cylinder

Table 1. Stages of experiments.

Stage No	1	2	3	4
Aim of the experiment	Selection of fuelling mode	Determination of the influence of ignition timing on performance and emission for gas fuelling	Testing of both types of fuelling: with liquid butane and gasoline	Final testing of full scale engine equipped with selected fuel system
Condition of experiment	One-cylinder engine. Fuelling modes: – thermal activation of the mixture, – charge turbulization, – fuelling with LPG, – carburettor; – gas mixer. Partially open one throttle (part load), ignition timing as for MBT (48 deg BTDC), n = 3000 rpm	One-cylinder engine. Fuelling with evaporated butane-air mixture. Ignition timing: 35, 30, 25 deg BTDC. Gas mixer. Fully open both throttles (full load), n = 3000 rpm.	One-cylinder engine. Fuelling by injection of liquid butane to inlet port and carburation of gasoline. Fully open both throttles (full load) and additionally only one throttle (part load), n = 3000 rpm.	Full-scale engine. Fuelling with liquid butane injected to inlet port. Fully open both throttles (full load), n = 3000 rpm

was measured with the AVL 8QP500 sensor (12) with the use of the measuring and data acquisition system (15). Engine speed was measured with the Introl encoder of the C.A. (13). The measuring and data acquisition system based on the Kithley Co. A/D card (developed at the Department of IC Engines and Automobiles: Różycki, 1995) was used for computation of:

- mean indicated pressure of each cycle (imep) and mean of 200 cycles,
- maximum pressure of each cycle and mean of 200 cycles,
- coefficient of variation of:
  – maximum pressure,
  – mean indicated pressure (imep).

As far as carburettor-fuelling mode is concerned, changing of air-fuel ratio was carried out with the use of different fuel nozzles with different flow cross section area. For fuelling with fuel vapour, a special system was utilised. A main part of this system was the vaporiser (19), to which fuel was injected with the atomiser (20) controlled by a computer to which signal of the pressure inside the vaporiser, measured with the sensor (21), was sent. The fuel vapour was continuously supplied through a pipe (22) to the inlet valve of the engine (1) under small constant pressure, so the vapour could enter the cylinder mainly during a suction stroke.

Directed air stream was created by the injection of air through a thin pipe with the use of the small compressor (27).

Measured quantities were as follows: Pressure in engine cylinder, torque, fuel consumption, air flow, exhaust gas components, including NO_x, CO, HC and air excess coefficient λ. Air excess coefficient λ was measured directly with DiGas 465 analyser and also computed from measured air flow and fuel flow rates (for control purposes).

As a measure of cycle-to-cycle variations, two coefficients (Ozdor *et al.*, 1994; Heywood, 1988) were used:

- the coefficient of variation in maximum pressure, COV_{Pmax}, defined as follows

$$\overline{COV}_{Pmax} = \frac{\sigma_{Pmax}}{\bar{P}_{max}} \qquad (1)$$

- the coefficient of variation in indicated mean effective pressure

$$\overline{COV}_{imep} = \frac{\sigma_{imep}}{imep} \qquad (2)$$

where σ is the standard deviation of P_{max} and imep in the individual cycles, respectively. For further evaluation of the lean limit of engine operation, COV_{imep} was used, due to its steeper increase at this limit than COV_{Pmax}.

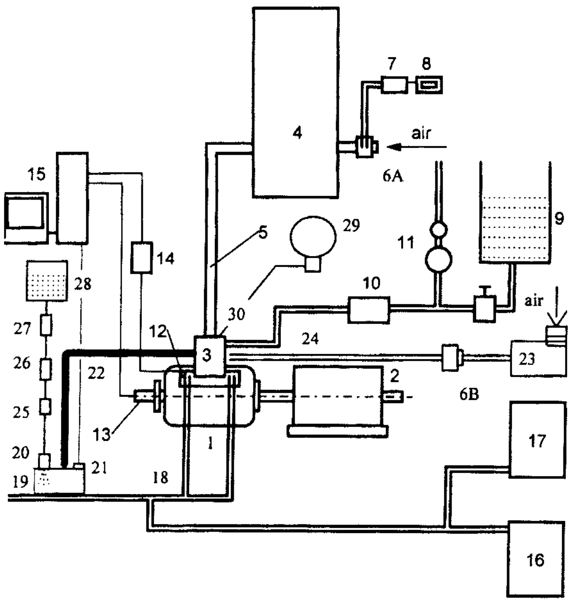

Figure 1. Test stand: 1-engine, 2-dynamometer, 3-carburettor, 4-surge tank, 5-air pipe, 6-laminar flowmeter (A-large, B-small flow-rate), 7-voltage converter, 8-indicator of flow-rate, 9-fuel tank, 10-fuel accumulator, 11-volumeter, 12-pressure sensor, 13-speed sensor INTROL, 14-amplifier, 15-data acquisition system with A/D converter and control system, 16-NO_x analyser Beckman Model 951, 17-analyser AVL Model 465, 18-exhaust manifold, 19-vaporiser, 20-injector of fuel which will be evaporated, 21-pressure sensor of fuel vapour, 22-fuel vapour pipe, 23-compressor of additional air, 24-additional air pipe, 25-pressure regulator, 26-filter, 27-fuel pump, 28-fuel tank, 29-gas propane-butane installation, 30-outlet of gas propane butane.

5. EXPERIMENTS

The aim of these experiments was to investigate the influence of different modes of fuelling and fuel types on combustion, emission and engine performances. The following types of fuelling (fuelling modes) were applied (Kowalewicz *et al.*, 1998; Pawlak, 1999):

(1) Fuelling with the fully evaporated gasoline which was provided to the inlet duct with a thin pipe ahead of the inlet valve. In this case the mixture was chemically activated and turbulised by gasoline vapour jet. The gasoline was evaporated in the vaporiser heated by exhaust gases.

(2) Fuelling with fully evaporated gasoline provided in the same way as in the point No 1, but additionally turbulized by air jet. The air jet (about 10% of total air flow) was provided to the cylinder by additional thin pipe ahead the inlet valve. In this case charge was turbulized both by the vapour jet and by the air jet.

(3) Fuelling with gasoline provided by conventional carburettor and additional air jet (as in the point No 2) was

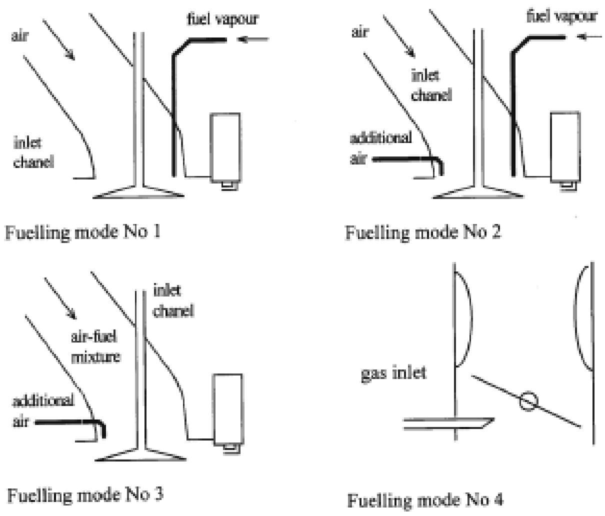

Figure 2. Four modes of unconventional fuelling.

applied for turbulization of the charge.
(4) Fuelling with propane – butane – air mixture created in the mixer, which was installed under Venturi nozzle.
(5) Standard fuelling with carburettor was used (mode 0).
Above mentioned fuelling modes are shown in Figure 2.

5.1. Results of Preliminary Experiments

5.1.1. Lean Operation Limit

Influence of air excess coefficient λ on lean operating

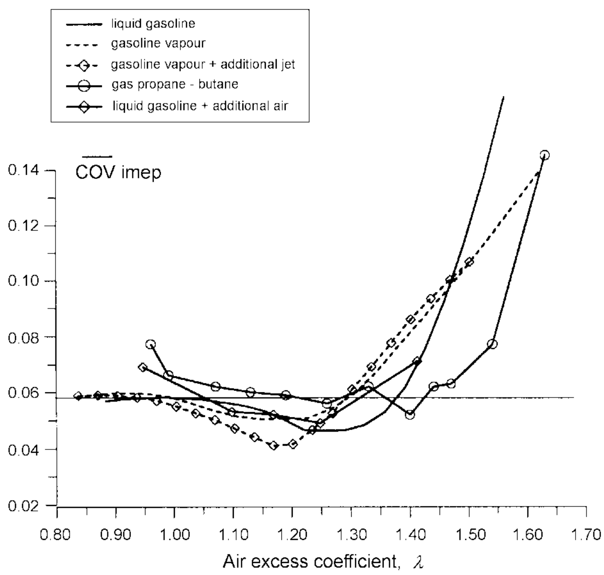

Figure 3. COV_{imep} as a function of air excess coefficient l for unconventional fuelling modes shown in Fig. 2. Ignition timing 48 deg BTDC, WOT, n = 3000 obr/min.

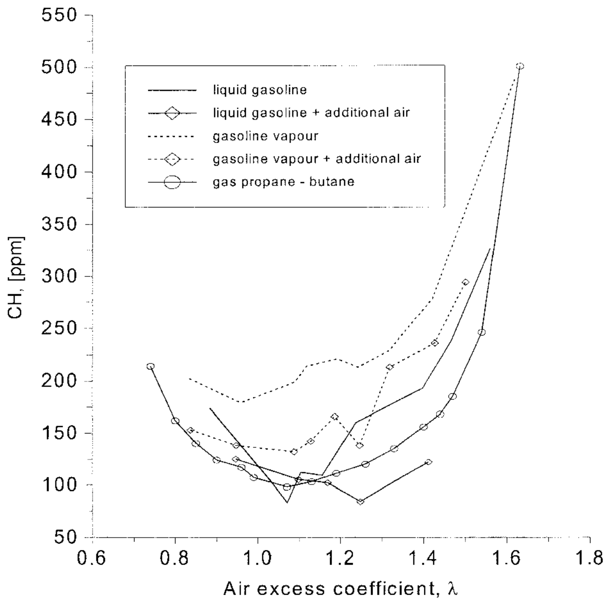

Figure 4. HC emission as a function of air excess coefficient λ for conditions as in Fig. 3.

limit was investigated with the use of fuelling with all above-mentioned modes. The results are shown in Figure 3 and Figure 4.

From the dependence of coefficient of variation of imep COV_{imep} on air excess coefficient λ (Fig. 3) may be seen that:

- Leaning the mixture to some extend results in lower COV_{imep} than for stoichiometric mixture.
- For $\lambda > 1.30$ for all modes of fuelling COV_{imep} increased.
- The leanest mixture was available for gaseous fuelling. For assumption of maximum COV_{imep} at the level of 10%, one gets $\lambda_{max} = 1.6$.

Similar results are obtained from the dependence of HC emission on λ (Fig. 4): The leanest mixture was obtained for gaseous fuel[1].

As far as NO_x emission is concerned, the 80% decrease of emission of that at the stoichiometric level (Fig. 5).

5.1.2. Influence of Air Excess Coefficient on Performances

Influence of air excess coefficient λ on engine torque was also investigated for all fuelling modes mentioned before. Two interesting results were observed (Fig. 6).

For fuelling with liquid gasoline, torque decreases with increase of λ, and for evaporated gasoline torque first increases and than, for $\lambda > 1.1$ begins to decrease with

[1] For evaporated gasoline there was phenomenon of "short-circuiting": the flow from inlet valve directly to exhaust valve, due to that HC emission level was high.

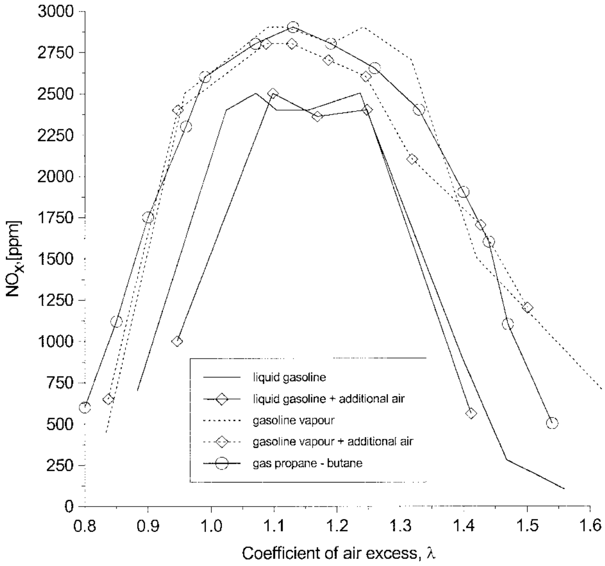

Figure 5. NO_x emission as a function of air excess coefficient λ for conditions as in Fig. 3.

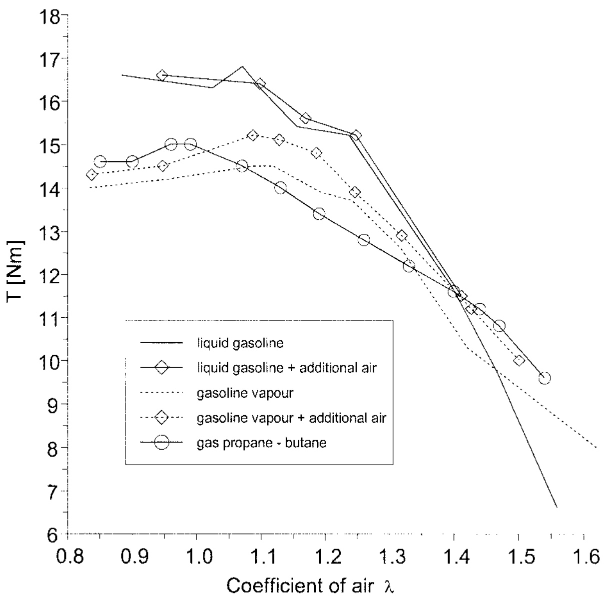

Figure 6. Brake torque as a function of air excess coefficient λ for conditions as in Fig. 3.

mixture leaning. For gas propane-butane, maximum torque was obtained for stoichiometric mixture. General result is as follows: Torque can be controlled not by throttle but by control of A/F ratio.

5.1.3. Influence on Fuel Type on Burning Rate and Combustion Pressure

As it was expected, fuel supplied in gaseous phase burnt quicker than supplied in liquid state (with carburettor fuelling) – Fig. 7. Especially, burning rate of propane-butane was very high.

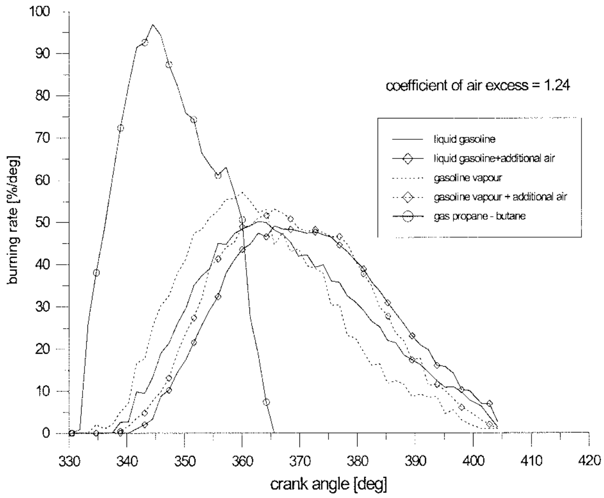

Figure 7. Burning rate as a function of CA for 5 modes of fuelling shown in Fig. 2.

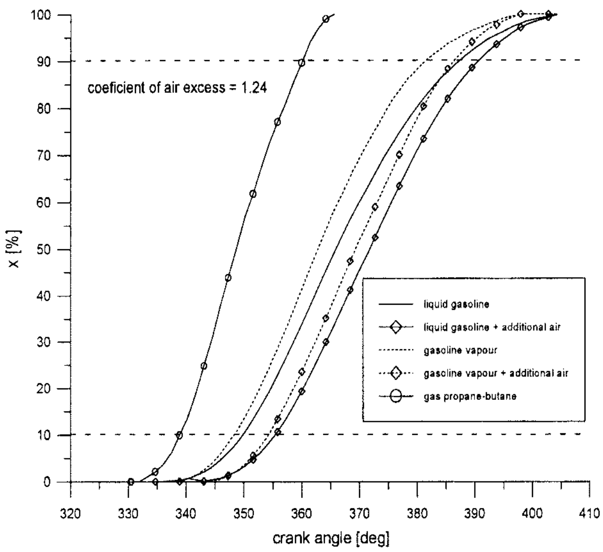

Figure 8. Mass fraction burnt as a function of CA for conditions as in Fig. 3.

As far as combustion pressure is concerned, in the whole range of lean mixture, the highest increase of pressure vs. CA was obtained for gaseous fuels. Maximum pressure was obtained for evaporated gasoline.

5.1.4. Effect of Turbulent Jet

Injection of air jet to inlet port normally enhances combustion, but here had only slight effect on burning rate of liquid and evaporated gasoline Figure 8. This problem is wider discussed in (Pawlak, 1999; 1997).

5.2. Second Stage of Experiments: Gaseous Butane Fuelling

The aim of these experiments was to investigate the influence of ignition timing on performance and emissions for fuelling with gaseous fuel (butane) and comparison of this fuelling with gasoline fuelling (standard for this engine). Instead of LPG used in the preliminary stage of experiments, here butane was used in order to have constant properties of the fuel during emptying the bottle with gas during experiments. The experiments were carried out on the same test stand as preliminary ones, for three different ignition timing: 25, 30 and 36 deg BTDC (Kowalewicz *et al.*, 1999).

As a result of experiments, similar emissions, performances and operating limits were obtained for butane fuelling and gasoline fuelling (Heywood, 1988). Some results are shown in Figures 9-12.

From these figures the following results may be drawn:

- The best results were obtained for ignition timing 36 deg BTDC.
- The lean operating limit is determined by $\lambda = 1.5$ for both fuels.
- There are only slight differences in performances and emissions for fuelling with gasoline and butane.

Because of fuelling with gaseous medium, volumetric efficiency could to be lower (due to volume of the gas fuel), in the next step experiments, liquefied gas was recommended for further experiments.

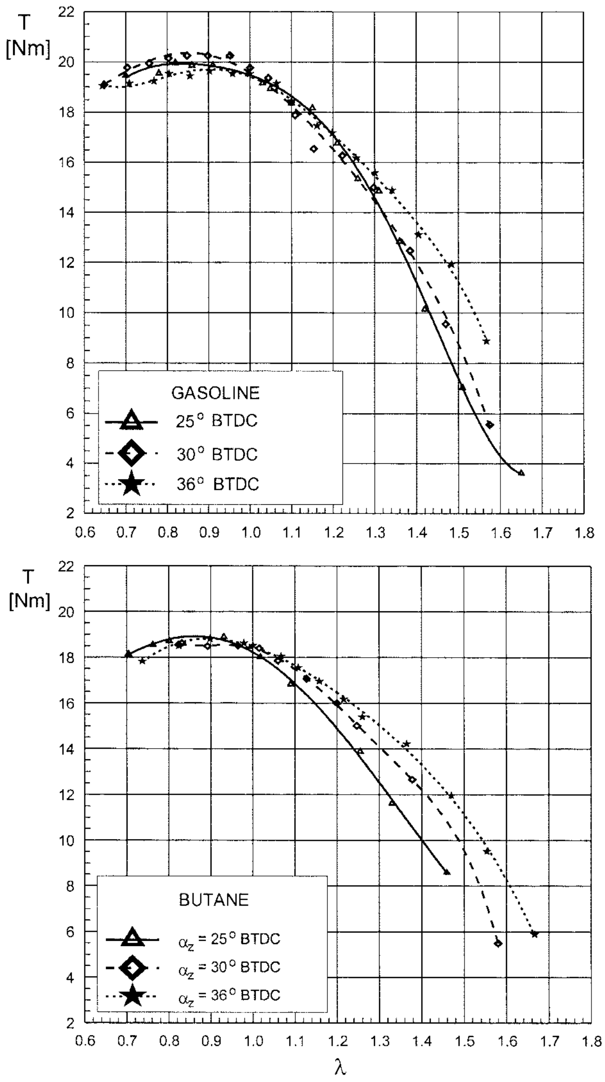

Figure 9. Engine brake torque as a function of air excess coefficient λ for three ignition timings and for fuelling with gasoline and gaseous butane. Fully open both throttles, n = 3000 rpm.

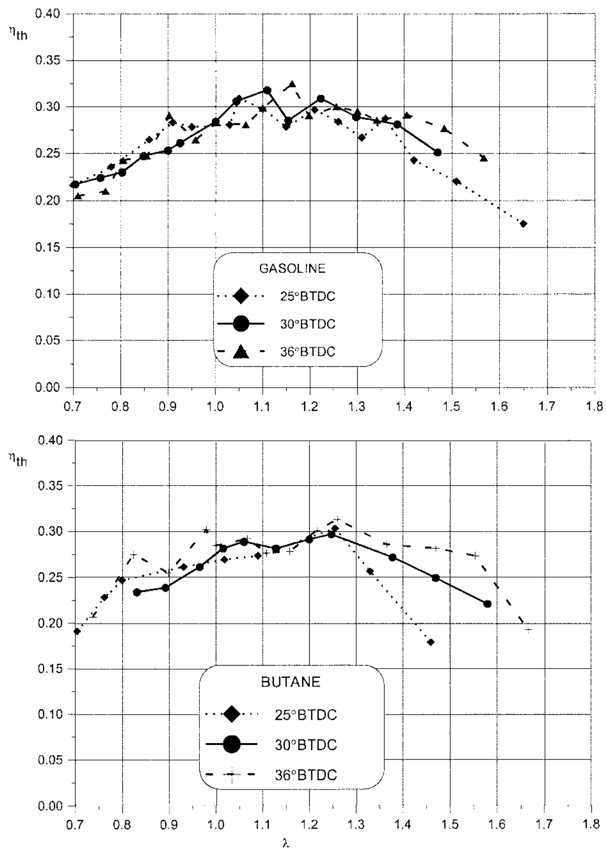

Figure 10. Engine thermal efficiency as a function of air excess coefficient λ for three ignition timings and for fuelling with gasoline and gaseous butane. Conditions as in Fig. 9.

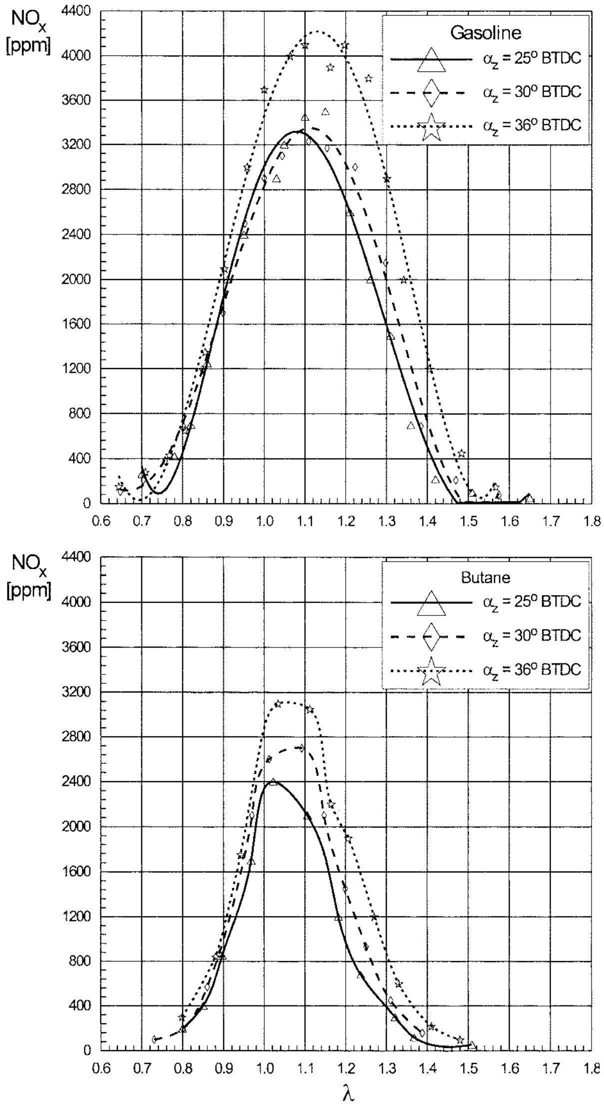

Figure 11. NO$_x$ emission as a function of air excess coefficient λ for three ignition timings and for fuelling with gasoline and gaseous butane. Conditions as in Fig. 9.

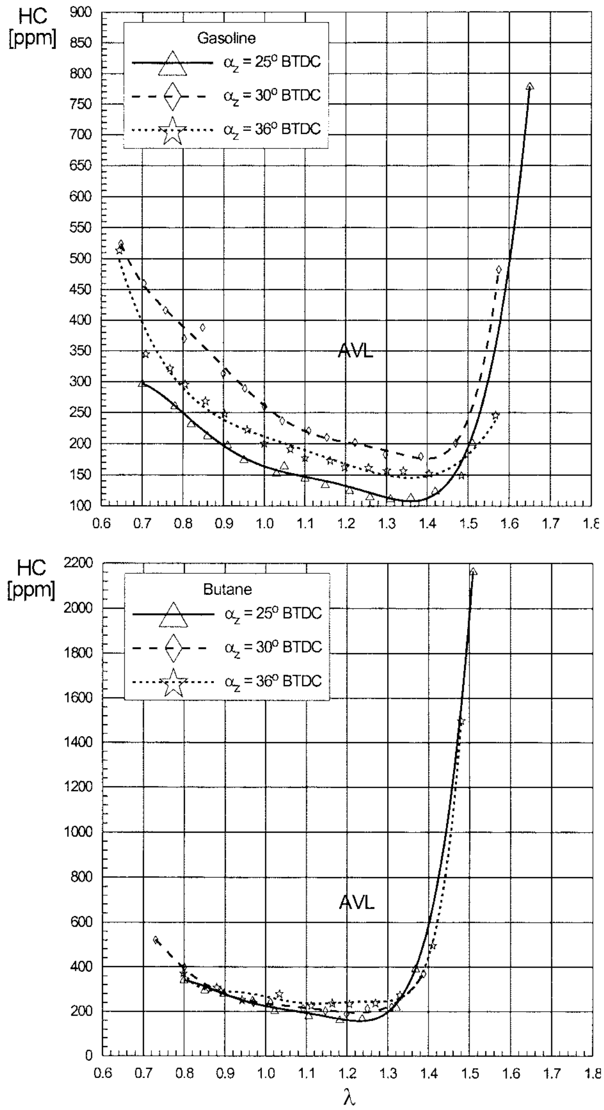

Figure 12. HC emission as a function of air excess coefficient λ for three ignition timings and for fuelling with gasoline and gaseous butane. Conditions as in Fig. 9.

5.3. Third Stage of Experiments: Injection of Liquid Butane to Inlet Port

The aim of this stage of experiments was to improve fuelling with butane. Instead of mixing butane with air with the use of mixer installed in inlet duct, liquid butane was injected to inlet port. Sequential injection was synchronised with valve lift: injection started when valve began to open. In this way coefficient of volumetric efficiency was expected to be improved. The same as in the previous experiments ignition timing was applied: 25, 30 and 36 deg BTDC and optimal selected. Fuelling with gasoline was performed for conventional carburettor system (without heating the inlet manifold) and for comparison with butane fuelling.

The following results were obtained:

- Hydrocarbons begin increase very rapid for $\lambda \geq 1.35$ independently of ignition timing.
- Nitric oxides depend very strongly on ignition timing: the later the ignition, the lower NO$_x$ emission.
- Thermal efficiency depends very strongly on ignition timing: for lean mixtures ignition angle 36 deg BTDC is the best.
- Torque also depends very significantly on ignition timing: for lean mixtures the most convenient is 36 deg BTDC.

Farther experiments depended on comparison of perfor-

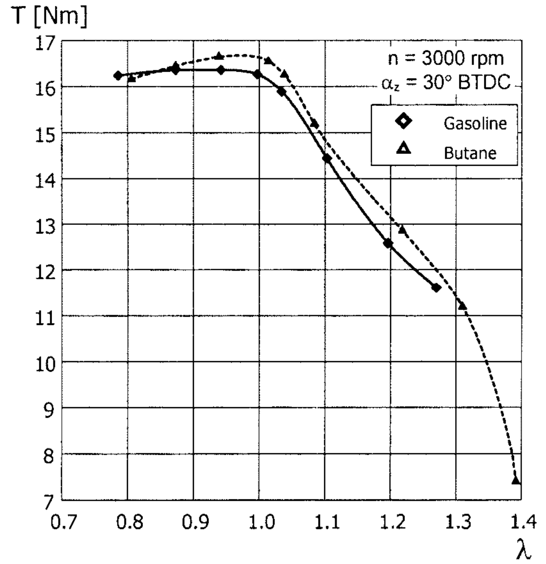

Figure 13. Comparison of engine torque for fuelling with liquid butane and gasoline. Partial load (open only one throttle), n = 3000 rpm.

mances and emissions of one cylinder engine for fuelling with liquid butane and gasoline for partial load operation (open only one throttle). For both fuels ignition angle was 30 deg BTDC. Following results were observed:

- Leaner mixture could be provided for butane fuelling (up to $\lambda = 1.4$).
- Lower HC emission (FID) was obtained for butane fuelling.
- In general lower NO_x emission was obtained for butane fuelling for $\lambda \leq 1.25$, but for $\lambda \geq 1.25$ higher.
- Thermal efficiency and torque were higher for butane fuelling, Figure 13 and Figure 14.

5.4. Fourth Stage of Experiments: Full Scale Engine Fuelled with Liquid Butane Injected Intermittently to Inlet Port

The fuelling of liquid butane developed with the use of one-cylinder engine was adapted to full scale two-cylinder engine. Results of experiments were similar as for one-cylinder engine. Increase of engine efficiency in the range of lean mixture in relation to standard fuelling was obtained. Also decrease of NO_x emission is possible. However increase of efficiency and decrease of NO_x emission demand different ignition timing; early ignition for increase of efficiency and late for decrease of NO_x – Figure 15 and 16.

6. CONCLUSIONS

6.1. Results

During three stages of experiments the novel fuel system of Cinquecento engine was developed. The system characteristics are as follows:

- Intermittent injection of the liquid butane to inlet port.
- Control of torque (and power) accomplished by means

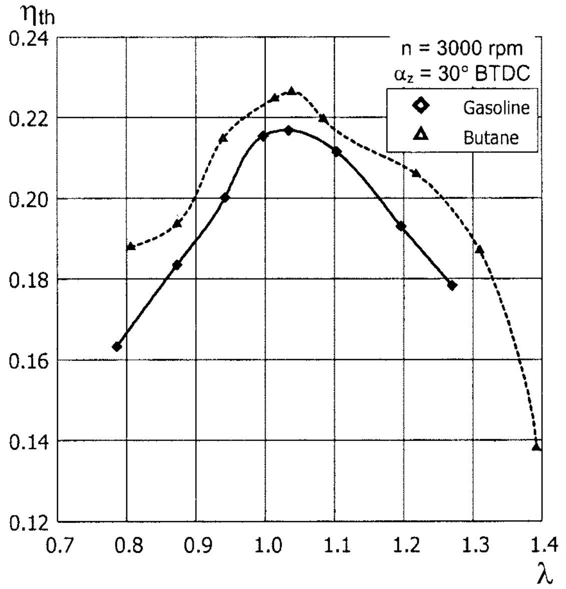

Figure 14. Comparison of engine thermal efficiency for fuelling with liquefied butane and gasoline. Partial load (open only one throttle), n = 3000 rpm.

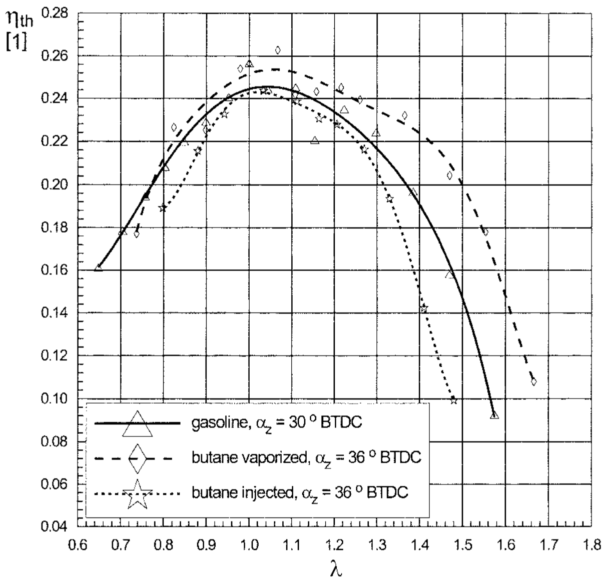

Figure 15. Engine overall efficiency as a function of air excess coefficient λ for 2-cylinder engine. Both throttles fully open, n = 3000 rpm.

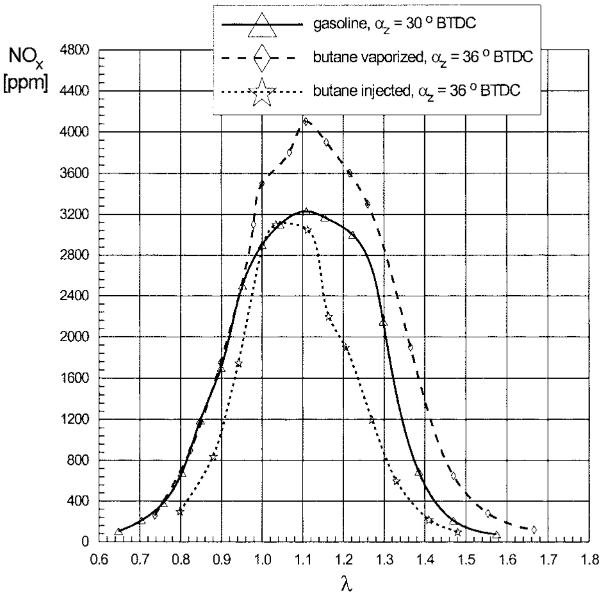

Figure 16. NO_x emission as a function of air excess coefficient λ for 2-cylinder engine. Both throttles fully open, n = 3000 obr/min.

of changing A/F ratio, as in CI engine.

- For partial load engine has higher torque and higher thermal efficiency than conventional engine.
- Lower emission of NO_x or higher efficiency than of conventional engine.

6.2. Discussion

- Operational characteristics of the engine fuelled with liquid butane should be carried out in order to show its superiority over conventional gasoline carburettor standard engine.
- Some measures should be undertaken for example as application of turbulent jet or thermal/chemical activation to increase burning rate.
- Novel fuelling system of sequential injection of liquid butane “transvalve – injection” should be worked out.

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