Ingineria Automobilului

Registrul Auto Român
SIAR
Society of Automotive Engineers of Romania

DISTRIBUTED WITH AUTOTEST MAGAZINE
No. 19 / JUNE 2011

A Clean Automobile for a cleaner World

- Interview with the Rector of Pitesti University
- Burning mixtures petrol / diesel and the diesel engine n-butanol
- Achievements Student Politehnica Timisoara
- Determination of mechanical gear efficiency
- Cost effective distribution in transport logistics activities

SIAR IS AFFILIATED TO

FISITA
INTERNATIONAL FEDERATION OF AUTOMOTIVE ENGINEERING SOCIETIES

EAEC
EUROPEAN AUTOMOBILE ENGINEERS COOPERATION

European Union (EU) and the Fuels of the Future*

Oil, the main energy source for transport overall, supplying nearly 100% of road transport fuels, is currently expected to reach depletion on the 2050 perspective.

Climate protection and security of energy supply therefore both lead to the requirement of building up an oil-free and largely CO_2-free energy supply to transport on the time horizon of 2050.

Total CO_2 emissions from transport have increased by 24% from 1990 to 2008, representing 19.5% of total European Union (EU) greenhouse gas emissions.

The EU objective is an overall reduction of CO_2 emissions of 80-95% by the year 2050, with respect to the 1990 level. Decarburization of transport and the substitution of oil as transport fuel therefore have both the same time horizon of 2050. Improvement of transport efficiency and management of transport volumes are necessary to support the reduction of CO_2 emissions while fossil fuels still dominate, and to enable finite renewable resources to meet the full energy demand from transport in the long term.

The upcoming White Paper on the European transport policy for the next decade should outline a transport action programme until 2020. It should define the overall framework for EU action over the next ten years in the fields of transport infrastructure, internal market legislation, technology for traffic management and decarburization of transport through clean fuels and vehicles.

The Expert Group on Future Transport Fuels, according to its mandate, should consider the mix of future transport fuels to have the potential for:

- Full supply of the transport energy demand by 2050;
- Low-carbon energy supply to transport by 2050;
- Sustainable and secure energy supply to transport in the longer term, beyond 2050.

Alternative fuels are the ultimate solution to decarbonize transport, by gradually substituting the fossil energy sources, which are responsible for the CO_2 emissions of transport.

Alternative fuel options for substituting oil as energy source for propulsion in transport are:

Electricity/hydrogen, and **biofuels (liquids)** as the main options;
Synthetic fuels as a technology bridge from fossil to biomass based fuels;
Methane (natural gas and biomethane) as complementary fuels;
LPG as supplement.

Single-fuel solutions covering all transport modes would be technically possible with liquid biofuels and synthetic fuels. But feedstock availability and sustainability considerations constrain their supply potential. Thus the expected future energy demand in transport can most likely not be met by one single fuel. Fuel demand and greenhouse gas challenges will require the use of a great variety of primary energies.

The main alternative fuels should be available EU-wide with harmonized standards, to ensure EU-wide free circulation of all vehicles. Incentives for the main alternative fuels and the corresponding vehicles should be harmonized EU-wide to prevent market distortions and to ensure economies of scale supporting rapid and broad market introduction of alternative fuels.

The main alternative fuels considered should be produced from low-carbon, and finally from carbon-free sources. Substitution of oil in transport by these main alternative fuels leads then inherently to a decarburization of transport if the energy system is decarbonized. Decarburization of transport and decarburization of energy should be considered as two complementary strategic lines, closely related, but decoupled and requiring different technical approaches, to be developed in a consistent manner.

The different transport modes require different options of alternative fuels:
Road transport could be powered by electricity for short distances, hydrogen and methane up to medium distance, and biofuels/synthetic fuels, LNG and LPG up to long distance;
Railways should be electrified wherever feasible, otherwise use biofuels;
Aviation should be supplied from biomass derived kerosene;
Waterborne transport could be supplied by biofuels (all vessels), hydrogen (inland waterways and small boats), LPG (short sea shipping), LNG and nuclear (maritime).

(*) Report of the European Expert Group on Future Transport Fuels, January 2011

Mircea Oprean

Summary „Ingineria Automobilului“ No. 19

New series of the Revista Inginerilor de Automobile din România (RIA), 1992-2000
Cod ISSN 1842 - 4074

Interview with Mr. Prof. Univ. Dr. Gheorghe Barbu
Rector of Pitesti University

Ingineria Automobilului (Automotive Engineering): Sir, how would you assess the role of university research and innovation in engineering, its integration into industrial research and the directions that will have to be taken in the future?

The scientific research is, as we all know, the main mechanism generating knowledge within our society. Equally, scientific research is a fundamental axis of higher education so that every university undertaken this mission, too. Unfortunately, lately, in Romania, the scientific research integrated itself slightly within the industrial projects. Why? Which are the causes? We need to discuss a lot here...

Within the automotive engineering, we could be better, taking into account that in Romania, Dacia-Renault works very well since quite a long time ago. The R&D technical centre from Titu (Renault Technologie Roumanie) is already operational, which obviously can favor the integration of university scientific research in the industrial world. About Ford Romania from Craiova, as far as I understood, the things are advancing, so that we can only hope to an improvement of the connections between university scientific researches with the industrial environment.

I rest my case by saying that the university scientific research needs an anchorage in the real, industrial world and I do hope that within the near future this will happen.

A.E: What would be Pitesti University`s contribution to resolve automotive challenges such as car environmental and energy problems, which will have to faced in the future, in the context of foreseeable depletion of oil and natural gas resources?

Indeed, the times we live are very challenging for this particular field. Within the University of Pitesti, over the time there were and are preoccupations on the subject you mentioned. I would like to outline the fact that in Pitesti, the technical higher education within the automotive engineering is a tradition, its birth date (1969) being connected with the creation of Dacia plant at Mioveni.

In order to answer to your question, I enumerate some of the major preoccupations of my colleagues from the Automotive Engineering field, whose purposes is the minimization of the automobile's impact upon the environment: electric/hybrid electric vehicle (we do hope to present a prototype this fall – maybe on the occasion of the CAR2011 International Automotive Congress, organized by University of Pitesti between 2-4 November), variable valve actuation, variable compression ratio (in our labs there are some original engine prototypes of this type), biodiesel fuelling and so on.

Therefore, I can only say that we have the necessary resources (human and material) in order to approach the subject you refer to and to come back to the first question, I believe these resources to develop themselves more and more within the frame of research projects carried out in collaboration with the industrial world.

A.E.: From 2nd to 4th November, this year Piteşti University with SIAR, will organize the 10th edition of the International Automotive Engineering Congress (CAR 2011) , under the patronage of FISITA and EAEC. How would you describe this event and what are your expectations?

Being at its 10th edition, the CAR International Automotive Congress is a scientific event well known in Romania and not only; was its first edition in 1978. Starting with 1997, a strong international component was added, so that it became an important scientific event with proved benefits for the Romanian higher education and for Romania

The CAR International Automotive Congress is part of the scientific events cycle having SIAR and FISITA patronage. RENAULT Romania, FORD, AVL List Austria, HORIBA and ACAROM are amongst the sponsors of our congress.

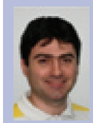
Taking into account the actual context regarding the need for protecting the environment to which we referred to earlier, the general theme which will govern our congress is „Automotive Engineering and the Environment".

Our hope (as organizers) is that the 10th edition, as well, will offer to the participants the frame of some efficient debates and the opportunity of establishing cooperation relations with the university/industrial environment from Romania/abroad.

Ingineria Automobilului (Automotive Engineering): Thank you very much, Mr. Gheorghe Barbu for this interview.

Intake Flow Visualization Using CFD

V. IORGA, A. CLENCI
University of Pitești, Romania; Conservatoire National des Arts et Métiers de Paris, France



Ion TABACU
University of Pitești, Romania

P. PODEVIN, A. DELACROIX
Conservatoire National des Arts et Métiers de Paris, France

This article was produced under the project "Supporting young Ph.D students with frequency by providing doctoral fellowships", co-financed from the EUROPEAN SOCIAL FUND through the Sectorial Operational Program Development of Human Resources

ABSTRACT:
This investigation presents results from a 3D numerical simulation of the air flow within an engine presenting the possibility to continuously vary the intake valve lift during operation. The purpose was

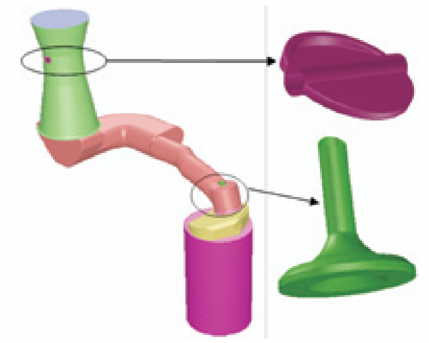
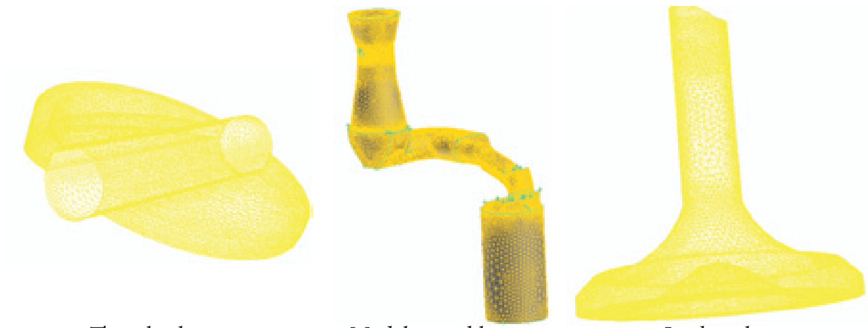


Fig. 1. Geometry model

Table 1. Geometry dimensions

Throttle valve diameter [mm]	63.5
Throttle valve diameter [mm]	29.5
Intake valve diameter [mm]	31.6
Valve seat length [mm]	2.8
Cylinder diameter [mm]	75



Throttle plate   Model assembly   Intake valve

Fig. 2. 3D mesh volume

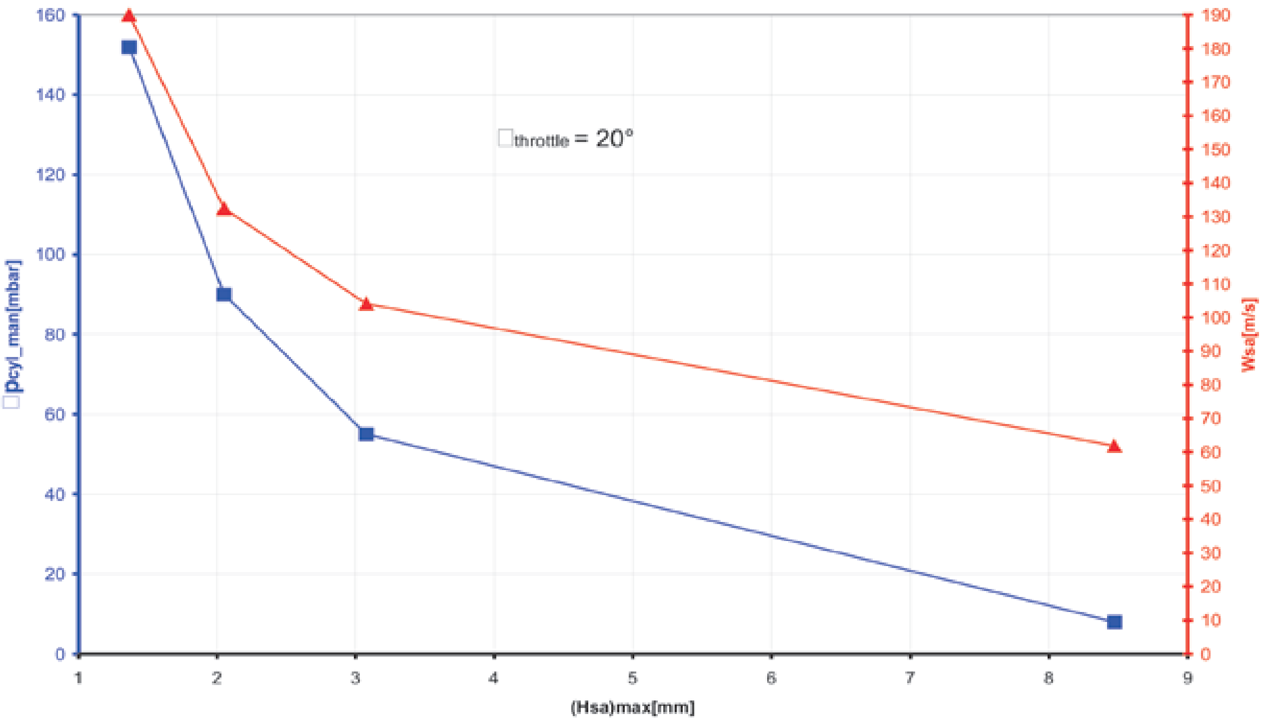
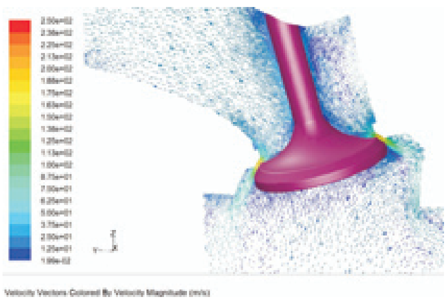


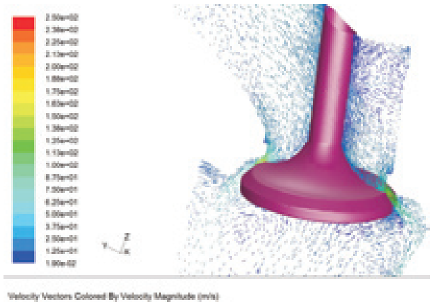
Fig. 3. The flow velocity under valve

Table 2. Throttle valve open φ = 20°. Measured and calculated values

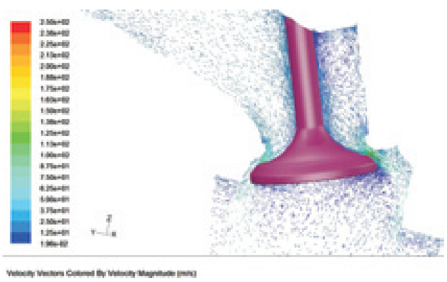
$(H_{sa})_{max}$ [mm]	A_{sa} [mm^2]	p_{cil} [mbar]	p_{atm} [mbar]	p_{col}[mbar] *CFD*	Δp_{cil_col} [mbar]	W_{sa_max} [m/s]
1.37	94.7	446	995	598	152	190.03
2.05	142.2	441		531	90	132.41
3.08	213	474		529	55	104.09
8.47	587.84	460		468	8	61.84



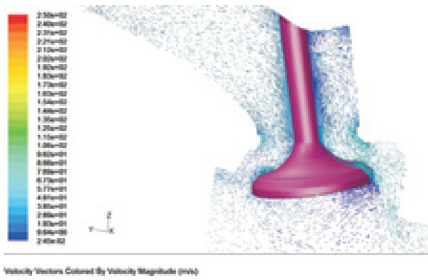
a. H=1.37 mm



b. H=2.05 mm

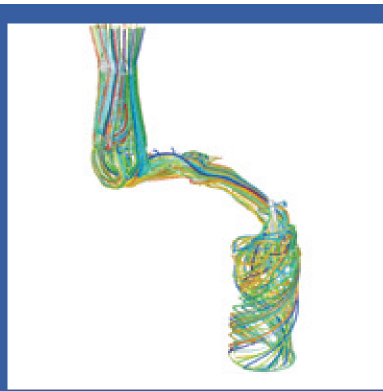


c. H=3.08 mm

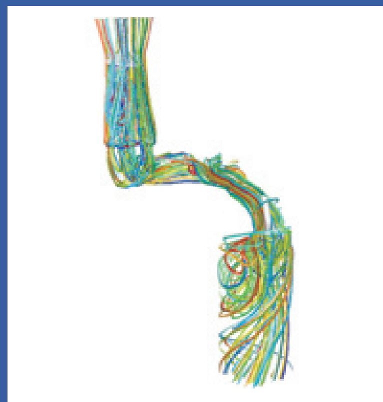


d. H=8.47 mm

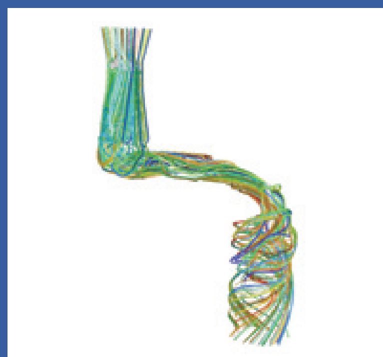
Fig. 4. Velocity vectors



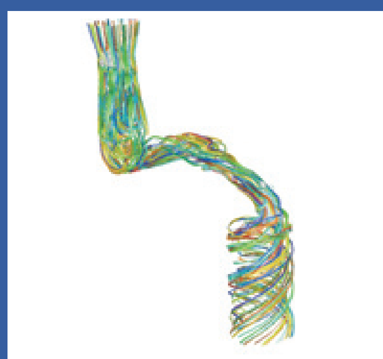
a. H=1.37 mm



b. H=2.05 mm



c. H=3.08 mm



d. H=8.47 mm

Fig. 5. 3D visualization of the flow motion

to obtain results about air velocity, turbulence at the valve gap, for a throttle plate open at 20° and different valve lifts. The 3D CFD simulation was carried out using FLUENT software with k-ω SST turbulence model.

Keywords: CFD, 3D model, steady flow, flow visualization

INTRODUCTION

With the advent of more powerful computers, now mathematical models are increasingly accepted as design and optimization tools for engine development and flow visualization, [1]

Our studies revealed that the ability to control valve lift certainly offers the ability to control intake air mass [3] but also has the added benefit that it improves fuel-air mixing process and controls air motion. This is particularly important at idle and low part loads when low lifts are to be used for achieving the required power, [2].

The CFD investigation was carried out for 20° opening of throttle plate (corresponding to idle operation) and 4 valve lifts: 1.37 mm, 2.05 mm, 3.07 mm, 8.47 mm.

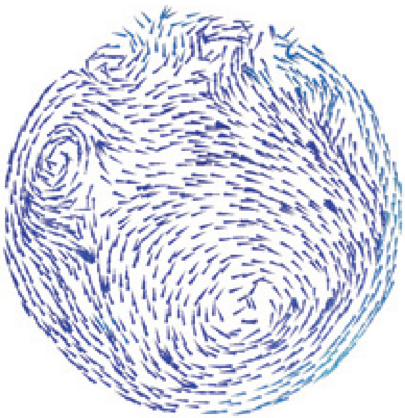
The main goal of this study is to perform a 3D CFD simulation on the effects of valve lifting height on the single cylinder filling process, using FLUENT software, [5]. Will be highlighted the air velocity near the valve and air movements inside the cylinder

3D MODEL GEOMETRY

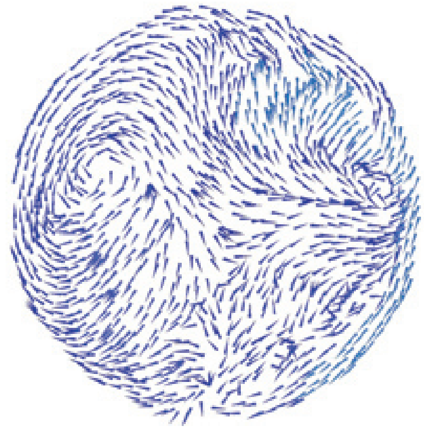
The geometry engine was created using the CATIA V5R19. For this numerical simulation, only one cylinder, throttle body, throttle plate, intake manifold, intake valve and the combustion chamber were kept (figure 1). Basic dimensions of the model are displayed in following table.

MESH GENERATION AND NUMERICAL SUMULATION

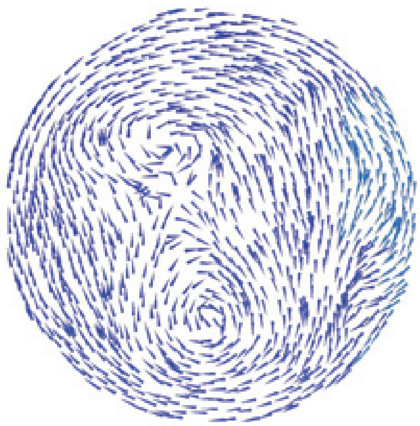
Geometry was transformed in an IGES format. This **.igs* file format, used by several software packages to exchange geometries, is imported in the pre-processor Gambit. First it was necessary to eliminate unneeded planes, edges and vertices and correct some discontinuities and inaccuracies.



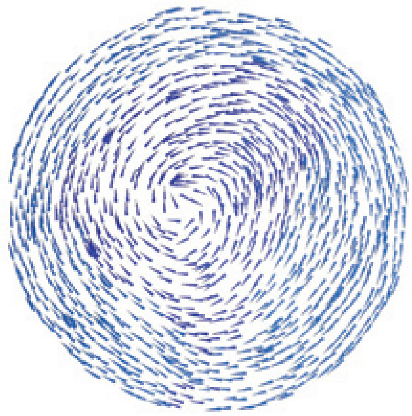
a. H=1.365 mm



b. H=2.05 mm



c. H=3.072 mm



d. H=8.47 mm

Fig. 6. The swirl motion

The geometry grid is composed of 1007208 tetrahedral cells (figure 2). Mesh size varies between 0.2 near the intake valve and throttle plate, growing to 2 mm for the rest of geometry. Pressure inlet is one boundary condition in the throttle plate body. The pressure outlet was chosen as boundary condition at the BDC (bottom dead center) in cylinder.

The first step in Fluent was to import the mesh, to verify and scale it. Next, pressure values needed to be assigned to the inlet and outlet of 3D model. Implicit solver and k-ω SST turbulence model was used to simulate the air flow.

The study was performed to simulate the stationary airflow at an engine speed of 800 rev/min, corresponding to the idle operation. For this particular case, neither the piston nor the intake valve is moving.

In order to set the boundary conditions, data were taken from experiments performed in a steady-state flow within the variable intake valve lift engine fitted with in-cylinder and intake manifold pressure sensors, [4].

RESULTS AND DISCUSSION

In the following, for different maximum valve lifts and one opening of throttle plate, velocity underneath the intake valve is highlighted (see table 2 and figure 3, 4). Also, based on the CFD simulation, the fluid motion inside the cylinder is visualized (e.g. swirl and tumble motion) – figure 5.

The difference in pressure between the intake manifold and the in-cylinder determines the flow velocity underneath the intake valve, *Wsa*, as shown in Figure 3. One could see that the bigger the pressure drop, the bigger the velocity.

For a good visualization of the in cylinder flow, the inside volume was cut with a plane parallel to the cylinder axis (figure 4). The values of flow velocity used in table 2 and figure 3 are actually the maximum ones taken from figure 4.

In figure 5, the path lines are plotted for each case of valve lift height aforementioned. The swirl or rotation of the flow around the cylinder axis is clearly visible. Equally, the smaller the lift the more disorganized is the motion, which favors the fuel-air mixing process.

For a better visualization of the in cylinder flow motion, the inside volume was cut with a second plane perpendicular to the cylinder axis (figure 6). Slices are a common tool used to investigate the properties of the flow inside a volume. Thus, figure 6 reveals the swirl motion on a surface placed in cylinder, at a distance of 26.6 mm from the combustion chamber, parallel with the piston head.

CONCLUSIONS

CFD offers the opportunity to view the progress of flow in the filling process. In general, the optimal flow visualization technique depends on the needs of the user and the nature of vector field. It is possible to emphasize and communicate different visualizing characteristics of the flow using 2D. Visualizing swirl flow using 3D path-lines and vectors is easier than for the case of tumble motion.

Coming back to the effects of intake valve of the intake process, the final goal of the authors is to find a correspondence between the positive effect of improving the airflow velocity and the negative effect of increasing the pumping work when reducing the intake valve lift. The next step of this study is to perform an unstationary numerical simulation by taking into consideration the intake valve and piston motions, as well.

REFERENCES

[1] M.H. Shojaeefard, A.R., Noorpoor – *Flow Simulation in Engine Cylinder with Spring Mesh*, American Journal of Applied Sciences 5 (10): 1336-1343, 2008

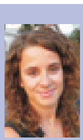
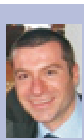
[2] Clenci A., Podevin P., Descombes G., Berquez J. – *Variable intake valve lift on spark ignition engine and its effects on idle operation*, EAEC Congress, Bratislava 2009

[3] Clenci A., Bîzîiac A. – *Distribuția variabilă ca tehnică de control a motorului cu aprindere prin scânteie. Realizări recente la Universitatea din Pitești,* Revista Ingineria Automobilului, aprilie 2006

[4] Iorga-Simăn V., Clenci A., Podevin P., Descombes G. – *On the effects of the valve lifting height upon the filling process*, Scientific bulletin, Faculty of Mechanics and Technology, Automotive series, year XVI, no.20 (1)

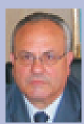
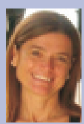
[5] *** Fluent 6.2 *User's guide*. Fluent Inc.

Effect on combustion process of gasoline/diesel and n-butanol/diesel blends in an optical compression ignition engine


Cinzia TORNATORE[1] Luca MARCHITTO Alfredo MAZZEI

Gerardo VALENTINO

Felice E. CORCIONE Simona S. MEROLA

Istituto Motori – CNR, Napoli Italy
E-mail: c.tornatore@im.cnr.it; l.marchitto@im.cnr.it; a.mazzei@im.cnr.it; g.valentino@im.cnr.it; f.corcione@im.cnr.it; s.merola@im.cnr.it
corresponding author Dr Simona Silvia Merola physics MD - chem.eng. phD Researcher Istituto Motori – CNR, Via Marconi, 8 - 80125 Napoli (Italy) www.im.cnr.it s.merola@im.cnr.it tel. +39 081 7177224

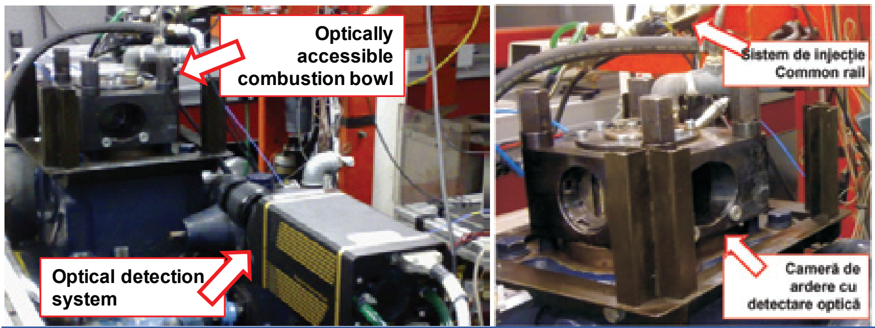


Fig. 1. Optically accessible engine

ABSTRACT

To meet the future stringent emission standards, innovative diesel engine technology, exhaust gas after-treatment, and clean alternative fuels are required. Oxygenated fuels have showed a tendency to decrease internal combustion engine emissions. In the same time, advanced fuel injection modes can promote a further reduction of the pollutants at the exhaust without penalty for the combustion efficiency. One of the more interesting solutions is provided by the premixed low temperature combustion (PLTC) mechanism jointly to lower-cetane, higher-volatility fuels.

To understand the role played by these factors on soot formation, optical techniques and were applied in a high swirl multi-jets compression ignition engine. Combustion tests were carried out using three fuels: commercial diesel, a blend of 80% diesel with 20% gasoline (G20) and a blend of 80% diesel with 20% n-butanol (BU20). The fuels were tested at 70MPa injection pressure and different timings using an open Common Rail injection system. At late injection timing coupled to high EGR rate (50%), the blends increased the ignition delay allowing to operate in partially premixed LTC (PPLTC) regime in which the fuel is completely injected before the start of combustion. Strong reduction of engine out emissions of smoke and NOx were obtained but with a little penalty on engine efficiency. This limitation was overcome operating at earlier injection timing in which a mixing controlled combustion (MCC) LTC regime was realized. In this regime, a good compromise between low engine out emissions and efficiency was achieved.

INTRODUCTION

The trade-off between NOx and particulate matter (PM) emissions from diesel engines is a well-known weakness of conventional diesel combustion. NOx and PM formation are influenced by in-cylinder local equivalence ratio and combustion temperature [1]. Therefore, if the combustion can be shifted outside the regions where the local equivalence ratio, the simultaneous reduction of NOx and PM emissions can be achieved. This is the concept of low temperature combustion (LTC) that can be based on two different mechanism: Homogeneous Charge Compression Ignition (HCCI) and Mixing Controlled Combustion (MCC) [2-8]. HCCI is a premixed combustion mechanism in which the equivalence ratio is less or equal to 1 ($\varphi \leq 1$). It is strongly dependent on kinetics, temperature, combustion timing. HCCI has the following potential limitations: difficult to control the combustion timing; possible knocking and high NOx and PM emissions at high loads; possible liquid fuel impingement on the in-cylinder surfaces; incomplete combustion at light loads (misfire, high UHC and CO). In the Mixing Controlled Combustion (MCC) the equivalence ratio is equal or higher than 1 ($\varphi \geq 1$). It can be realized in conventional diesel engines with appropriate fuel composition, modulation of injection and high level of EGR. MCC presents additional advantages compared to the HCCI, such as low combustion noise and the control of combustion phasing by the injection strategy.

LTC operational range and cetane number are connected.. In a light-duty diesel engine working at high loads, a low-cetane fuel allowed an homogeneous lean mixture with improved NOx and smoke emissions joint to a good thermal efficiency. A blend of diesel and gasoline, termed "dieseline", is beneficial for engine out emissions and combustion stability. [8][9][10][11][12] Nevertheless, the growing energy demand and limited petroleum fuel sources in the world have guided researchers towards the use of clean alternative fuels like alcohols for their better tendency to decrease the engine emissions. Alcohols have less carbon, sulfur content and more oxygen than traditional fossil-based fuels. On the other side, alcohol fuels, generally, produce higher evaporative emissions due to higher vapor pressures while their low energy density causes a drop in engine performance. However, the very low cetane number limits the usage of neat alcohols in diesel engines; they should be blended with diesel fuel without any modifications in the engine fuel system. n-butanol has more advantages than ethanol and methanol. Butanol has a lower auto-ignition temperature than methanol and ethanol. Therefore, it can be ignited easier when used in diesel engines. Besides, butanol is much less evaporative and releases more energy per unit mass than ethanol and methanol. Butanol has also a higher cetane number (12) than ethanol (8) and methanol (3) making it a more appropriate additive for diesel fuel. Butanol is less corrosive than ethanol and methanol and it can be blended with diesel fuel without phase separation. Finally, butanol can be produced by fer-

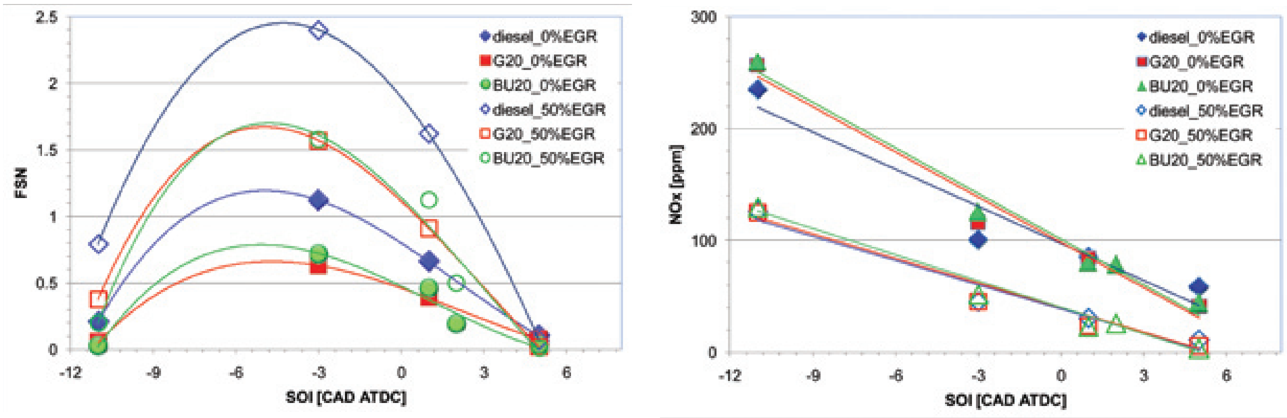


Fig. 2. Engine exhaust emissions of smoke (left) and NOx (right) versus the start of injection (SOI) for all the operative conditions.

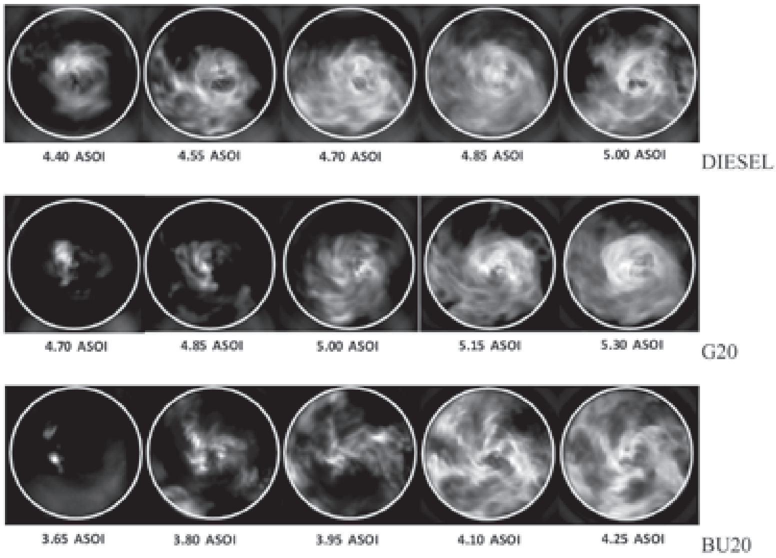


Fig. 3. UV-VIS flame emission for Diesel, G20 and BU20 fuel, SOI=11 CAD BTDC (50%EGR)

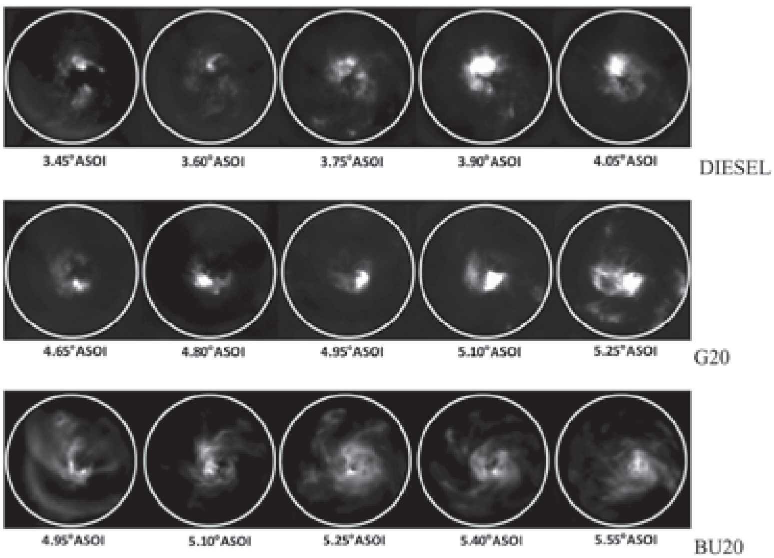


Fig. 4. UV-VIS flame emission for Diesel, G20 and BU20 fuel, SOI=1 CAD ATDC (50%EGR)

mentation of biomass, such as algae, corn, and other plant materials containing cellulose that could not be used for food and would go, otherwise, to waste. [13][14][15][16]

EXPERIMENTAL SET-UP AND RESULTS

The experiments were carried out in an external high swirl optically accessed combustion bowl connected to a single cylinder 2-stroke high pressure common rail compression ignition engine (Figure 1). The external combustion bowl (50 mm in diameter and 30 mm in depth) is suitable to stabilize, at the end of compression stroke, swirl conditions to reproduce the fluid dynamic environment similar to those within a real direct injection diesel engine. The injector was mounted within this swirled chamber with its axis coincident to the chamber axis; in this way the fuel, injected by the nozzle, is mixed up through a typical interaction with the swirling air flow. The combustion process starts and mainly proceeds in the chamber. As soon as the piston moves downward, the flow reverse its motion and the hot gases flow through the tangential duct to the cylinder and finally to the exhaust ports. A common rail injection system was arranged by a solenoid controlled injector with a micro-sac 7 hole, 0.141 mm diameter, 148° spray angle. An external roots blower provided an intake air pressure of 0.217 MPa with a peak pressure within the combustion chamber of 4.9 MPa under motored conditions.

Combustion tests were carried out using three fuels. The baseline fuel was the European low sulphur (10 ppm) commercial diesel with a cetane number of 52. Moreover, two blends were tested; the first was obtained by blending 80% of the baseline diesel and 20% of commercial 98 octane gasoline by volume, denoted as G20 in the following. The second blend was composed by 80% of the baseline diesel and 20% of n-butanol by volume and denoted as BU20. All combustion tests were carried out running the engine at the fixed speed of 500 rpm, injecting a fuel amount of 30mg ±1% at the pressure of 70 MPa. Tests were carried out setting the electronic injection timing (SOI) of 11 CAD BTDC, 3 CAD BTDC, 1 CAD ATDC and 5 CAD ATDC at two EGR rates, 0% and 50%.

One of the main targets of the premixed low temperature combustion is the reduction in engine out emissions without a significant penalty in fuel consumption. The engine efficiency decreased linearly retarding the start of injection at which an increase of the ignition delay is matched. The limit value for engine stability and efficiency was reached at SOI =5 CAD ATDC, which gave an increase in the ignition delay that enhanced the air-fuel mixing before combustion, realizing a Partially Premixed Regime. At fixed SOI the blend BU20 gave a higher engine efficiency, due to the better fuel volatility and mixing rate than diesel fuel (about 5%). The average reduction in thermal efficiency between the tested fuels, comparing the SOI=-11 CAD BTDC and SOI=5 CAD ATDC was lower than 20%, without any influence from EGR rates.

Regarding the exhaust emissions (Figure 2), the blends G20 and BU20 allowed to get the best compromise between NOx and smoke, at acceptable engine efficiency. At retarded SOC, the PPLTC mechanism was dominant: smoke and NOx emissions were close to zero but the engine efficiency decreased. At early injection timing (11 CAD BTDC), the blend BU20 and G20 allowed to operate in mixing controlled combustion (MCC) supplying the highest working area and a good compromise between NOx and smoke emissions. In particular, the blends with EGR=50% guaranteed the best operative condition attaining a reduction in engine out emissions without a significant penalty in engine efficiency.

Optical investigations realized in the combustion chamber allowed to study the effects of the tested fuels on the combustion process. (Figures 3 and 4) The first image of sequence corresponds to the first well resolvable Ultraviolet-visible wavelength signal thus to the first exothermic luminescence reactions. The auto-ignition occurred near the tip of the fuel jets; after this time the flame went up the direction of the spray axis, following the stoichiometric air-fuel ratio path. Due to the strong swirl, the flame, previously induced, spreads in the combustion chamber dragged by the anticlockwise air motion. This had a stronger effect on the longer lift-off jets. For all the conditions, the ignition delay of the G20 and BU20 blends was longer than the diesel fuel, at the same SOI and EGR values. The minimum ignition delay was measured for the SOI=3 CAD BTDC that corresponded to the highest smoke emission, for each condition more retarded SOI, the optical SOC increased about linearly retarding the injection timing, at EGR=50% and with fuels more resistant to auto-ignition (BU20). In particular, the BU20 at EGR=50% showed the longest ignition delay, except for the most advanced SOI, at which G20 with EGR=50% prevailed. In the extreme case SOI=5 CAD ATDC with 50% EGR the whole amount of fuel is injected before the start of combustion, realizing a premixed low temperature regime (PLTC). This occurred for both the blends. It should be noted that no spray evidence was detected also for BU20 at the condition with SOI=5 CAD ATDC and both EGR rates. In this way, the BU20 blend induced a PLTC mechanism in wider range of conditions compared to G20.

CONCLUSIONS

In-cylinder optical investigations, correlated with conventional measurements of engine parameters and exhaust emissions, demonstrated that the blends increased the ignition delay particularly at late injection timing allowing to operate in PPLTC regime in which the fuel is completely injected before the start of combustion. In this regime, strong reduction of engine out emissions of smoke and NOx were obtained. On the other hand this combustion regime reduced the engine efficiency. To overcome this limitation a mixing controlled combustion (MCC) LTC regime was realized by an earlier injection. In this regime, a good compromise between low engine out emissions and a good efficiency was demonstrated. The effect of the fuel quality and injection on the flame lift-off length and soot formation were studied. The increase of lift-off length well matched to a decrease of in-cylinder soot production. The BU20 blend, at 50% of EGR and late injection timing, allowed to operate in LTC regime in which a strong decrease of soot formation joined to reduced engine out emissions were obtained.

REFERENCES

[1] Heywood, J. B., *Internal Combustion Engine Fundamentals*, McGraw-Hill, New York, 1988.

[2] Kook, S.; Bae, C.; Miles, P.; Choi, D.; Pickett, L. (2005) *The Influence of Charge Dilution and Injection Timing on Low-Temperature Diesel Combustion and Emissions*. SAE Tech. Paper n. 2005-01-3837.

[3] Kimura, S., Aoki, O., Kitahara, Y., and Aiyoshizawa, E. (2001) *Ultra-Clean Combustion Technology Combining a Low-Temperature and Premixed Combustion Concept for Meeting Future Emission Standards.* SAE Tech. Paper n. 2001-01-0200

[4] Henein N.A, Bhattacharyya A, Schipper J, Bryzik, W. (2005) *Combustion and Emission Characteristics of a Small Bore HSDI Diesel Engine in the Conventional and LTC Combustion Regimes*, SAE Tech. Paper n. 2005-24-045.

[5] Horibe N., Ishiyama T. (2009) *Relations among NOx, Pressure Rise Rate, HC and CO in LTC Operation of a Diesel Engine.* SAE Tech. Paper n. 2009-01-1443

[6] de Ojeda, W., Zoldak, P., Espinosa, R., and Kumar, R.,(2008) *Development of a Fuel Injection Strategy for Diesel LTC*, SAE Paper n. 2008-01-0057.

[7] Yun, H., Sellnau, M., Milovanovic, N., and Zuelch, S. (2008) *Development of Premixed Low-Temperature Diesel Combustion in a HSDI Diesel Engine*. SAE Paper n. 2008-01-0639

[8] L. Pesant, L. Forti, N. Jeuland (2008) *Effect of Fuel Characteristics on the Performances and Emissions of an Early-injection LTC /Diesel Engine*. SAE Paper n. 2008-01-2408.

[9] Kalghatgi, G., Risberg, P., and Ångström H-E (2006) *Advantages of fuels with high resistance to auto-ignition in late-injection, low-temperature, compression ignition combustion*, SAE Paper n. 2006-01-3385.

[10] Splitter, D.A., Hanson, R., Kokjohn, S., Rein, K., Sanders, S., and Reitz, R.D. (2010) *An Optical Investigation of Ignition Processes in Fuel Reactivity Controlled PCCI Combustion*, SAE paper 2010-01-0345.

[11] Collings, N. and Weall, A.J. (2007) *Investigation into partially premixed combustion in a light-duty multi-cylinder diesel engine fuelled with a mixture of gasoline and diesel*

[12] Han, D., Ickes, A. M., Bohac, S. V., Huang, Z. Assanis, D. N. (2010) *Premixed low-temperature combustion of blends of diesel and gasoline in a high speed compression ignition engine* Energy Fuels, vol. 24 (6), pp 3517–3525.

[13] Lujaji F., Kristof L., Bereczky A., Mbarawa M. 82011) *Experimental investigation of fuel properties, engine performance, combustion and emissions of blends containing croton oil, butanol, and diesel on a CI engine, Fuel*, Vol. 90 (2), pp. 505-510

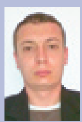
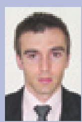
[14] D.C. Rakopoulos, C.D. Rakopoulos, R.G. Papagiannakis, D.C. Kyritsis, (2011) *Combustion heat release analysis of ethanol or n-butanol diesel fuel blends in heavy-duty DI diesel engine*, Fuel, Vol. 90(5), pp. 1855-1867

[15] D.C. Rakopoulos, C.D. Rakopoulos, E.G. Giakoumis, A.M. Dimaratos, D.C. Kyritsis (2010) *Effects of butanol-diesel fuel blends on the performance and emissions of a high-speed DI diesel engine.* Energy Conversion and Management, Vol. 51 (10) pp.1989-1997

[16] Mingfa Yao, Hu Wang, Zunqing Zheng, Yan Yue, (2010) *Experimental study of n-butanol additive and multi-injection on HD diesel engine performance and emissions*, Fuel, Vol. 89 (9), pp. 2191-2201.

Research on the Construction and Performances of the Three Way Catalytic Used in Depolluting the S.I.E


Prof. univ. dr. ing. **Florian IVAN**
Catedra Automobile
Facultatea de Mecanică şi Tehnologie
University of Piteşti
florian.ivan@upit.ro
florianivan2002@yahoo.com



University of Piteşti

drd. ing. **Daniel LIŢĂ**
drd. ing. **Andrei BUŞOI**

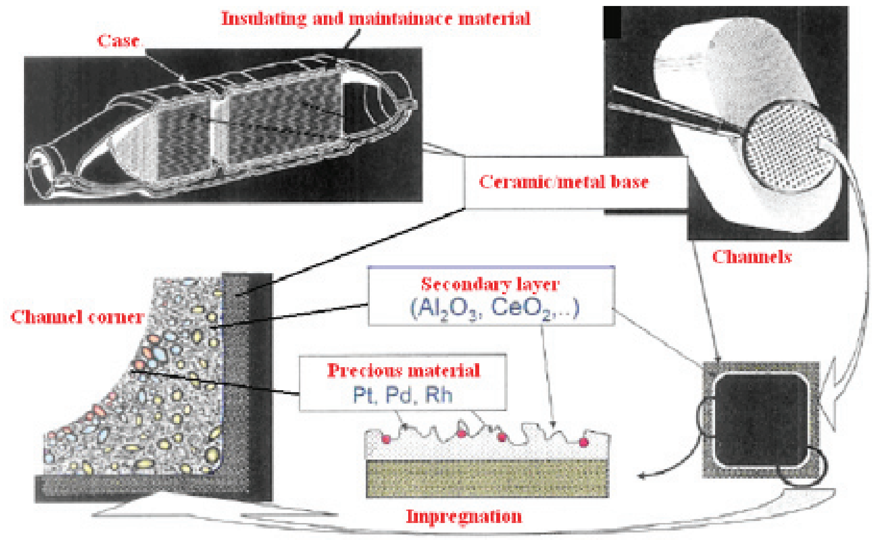


Fig. 1. A three way catalytic construction

ABSTRACT

The paper presents a synthesis of the variants of Three Way Catalytic used in a depolluting system of a S.I.E., highlighting the effects of the constructive and working parameters (running temperature, advanced ingnition) on the conversion efficiency. Then one presents the outcomes of an experimental research on a medium class car on the test stand. One establishes, thus, a correlation between the run-up time of a T.W.C. and the speed of the car, making reference to the low charges and engine rotations cases.

THE NECESSITY OF EXHAUST GASES CATALYTIC TREATMENT OF S.I.E.

Nowadays, one of the major problems the car faces is represented by the environment pollution. Therefore, the more and more severe European standards (table 1) have established the development of complex procedures to ensure the perfectioning of the working processes of spark ignition engine (SIE). Builders have channelled their research on two fundamental directions. A first direction is represented by the genesis depollution by ways which aim at improving the formation and burning of the mixture. The second one presupposes obligatorily the gases catalytic treatment. In this case the pollution standards impose the use of the Three Way Catalytic (T.W.C.) which proved to be an indispensable solution.

THE CATALYST CASE – CONSTRUCTIVE ELEMENTS AND WORKING PERFORMANCES

The catalyst case is a component of the exhaust gases treatment system met both at spark igni-

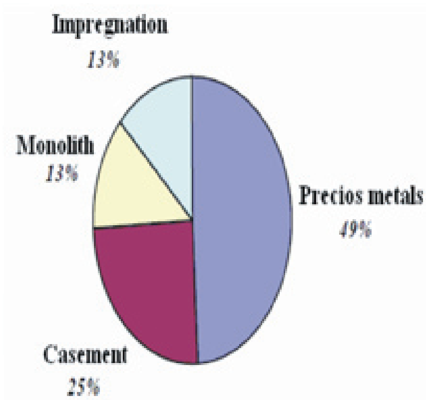


Fig. 2. The making cost structure of a T.W.C.

Tabelul 1. Evolution of the EURO standards for S.I.E.

(g/km)	Euro 1	Euro 2	Euro 3	Euro 4	Euro 5	Euro 5+	Euro 6
CO	2,72	2,2	2,3	1	1	1	1
HC(NMHC)	-	-	0,2	0,1	0,1(0,068)	0,1(0,068)	0,1(0,068)
Nox	-	-	0,15	0,08	0,06	0,06	0,06
HC+NOx	0,97	0,5	-	-	-	-	-
PM (GDI)	-	-	-	-	0,005	0,0045	-0,0045
PN	-	-	-	-	-	-	?

Table 2.The effect of the noble materials on pollutatns

Noble material/ Noxa	HC	CO	Nox
Pt (platinum)	+ + +	+ +	+
Pd (palladium)	+ + +	+ + +	+ +
Rh (rhodium)	+ +	+ +	+ + +

+ + + great efficiency; + + medium efficiency; + low efficiency

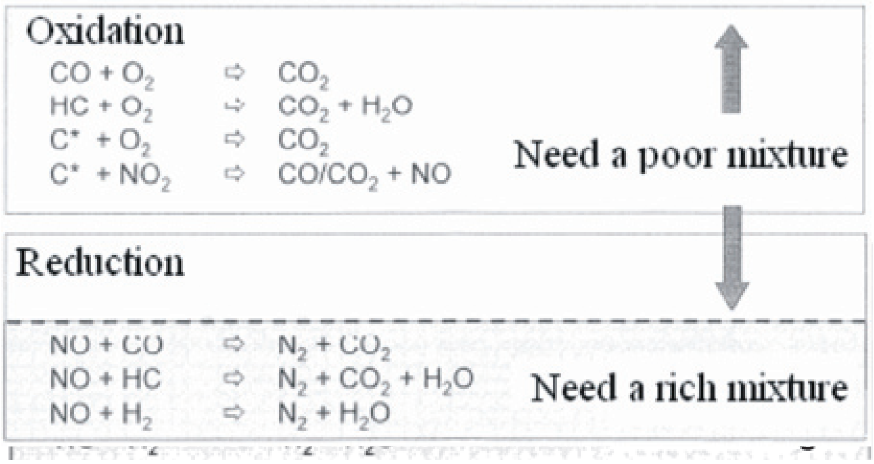


Fig. 3. The catalytic treatment reactions in T.W.C.

tion injection engines in the admission valve, as well as at those with direct gas injection (in the engine cylinder).

Generally, a catalyst case presupposes the following components: metallic case, insulating and maintaining material for the ceramic/metal base, ceramic/metal base (also called monolith) covered with aluminium oxide, surface in which one has impregnated the active material, made up of noble metals (platinum, palladium, rhodium) and Cerium oxide CeO2 which retains oxygen.

Figure shows synthetically the construction of a catalyst case.

Ceramic monoliths have their structure shaped in a honeycomb and they are obtained by pre-synthesis. They can have different shapes: round, oval, rectangular. The cell walls are cca. 0.3 cm thick. From a working point of view, one has in view that the monolith structure should not lead to counterpressures in order to compromise the dynamic and economic performances. The working temperature of the monolith should not be higher than 900°C, having as melting point 1355°C.

For example /1/, a medium class car has the main following features of a catalyst case: total volume: 1.4 [l]; first zone volume: 0.5 [l]; second zone volume: 0.9 [l]; diameter: 125 [mm]; specific geometric surface: 3 [m2 /l]; the material density: 1.7 [g/cm3]; cell number/cm2: 60.

An important working parameter is represented by the necessary heat quantity from 0 to 100°C in the case of the catalytic volume. Experience has shown that in the case of a ceramic monolith this parameter reaches medium values 60-65 kJ/l.

One has to stress the fact that the making costs of such a catalytic are generally important and are shared as follows (figure 2):

The metallic monolith has been used more and more lately, because it has the following advantages: the necessary heat quantity from 0 to 100°C in the case of a one liter monolith is of 31 kJ which reduces significantly the run-up time of the catalyst case, a more improved reliability as to the ceramic one, a smaller volume due to the less thick walls which leads to a lower counterpressure in the exhaust way, a simplified construction (it doesn't need fixing systems), it works at higher temperatures (1100°C).

As to the disadvantages, mention should be made of: the quicker cooling at low charges due to the low mass and to the high thermic conductivity; this disadvantage is partly eliminated by setting near the engine. The active catalytic surface is a laying of noble metals (platinum, palladium, rhodium). In the case of the three way

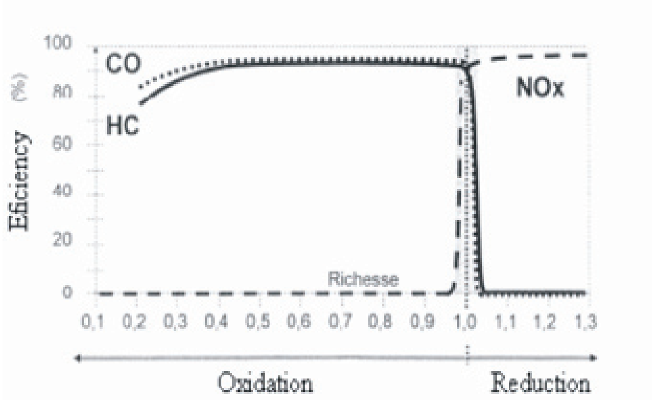


Fig. 4. Working range for T.W.C.

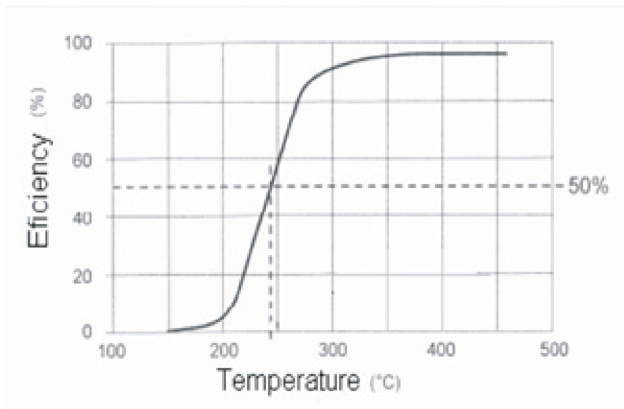


Fig. 5. T.W.C. Eficiency treatment T.W.C.

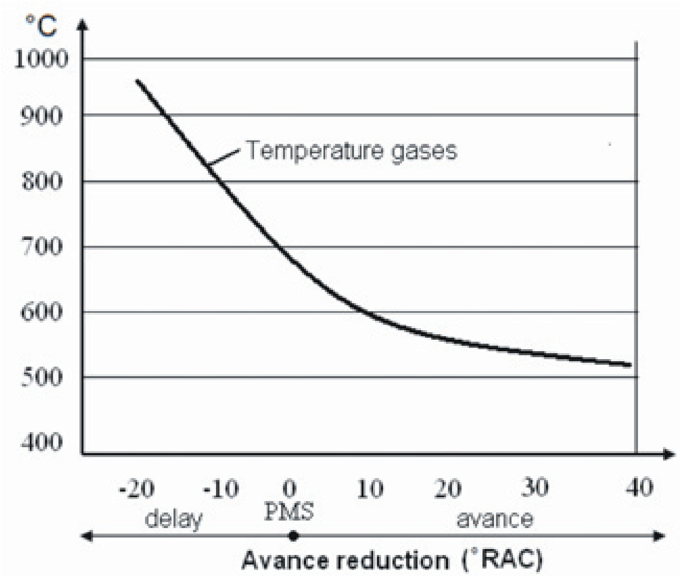


Fig. 6. The influence of the full advance reduction

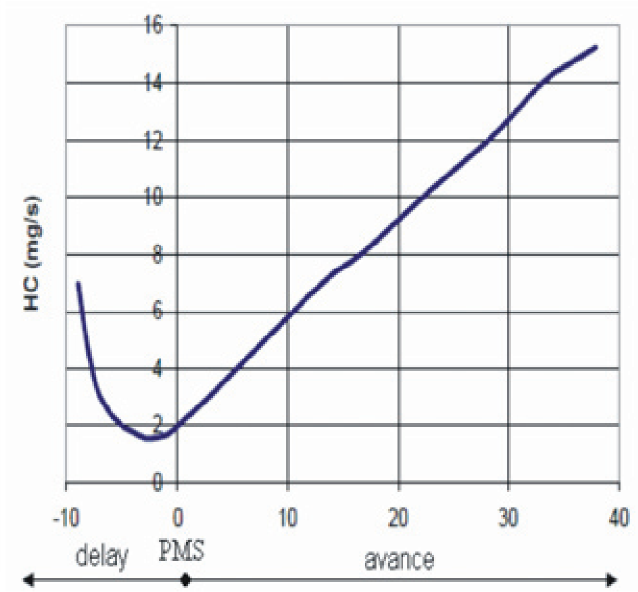


Fig. 7. The influence of the full advenc reduction on H

catalytic one usually uses a mixture of platinum-rhodium. The effect of these noble metals on pollutants is rendered synthetically in table 2.

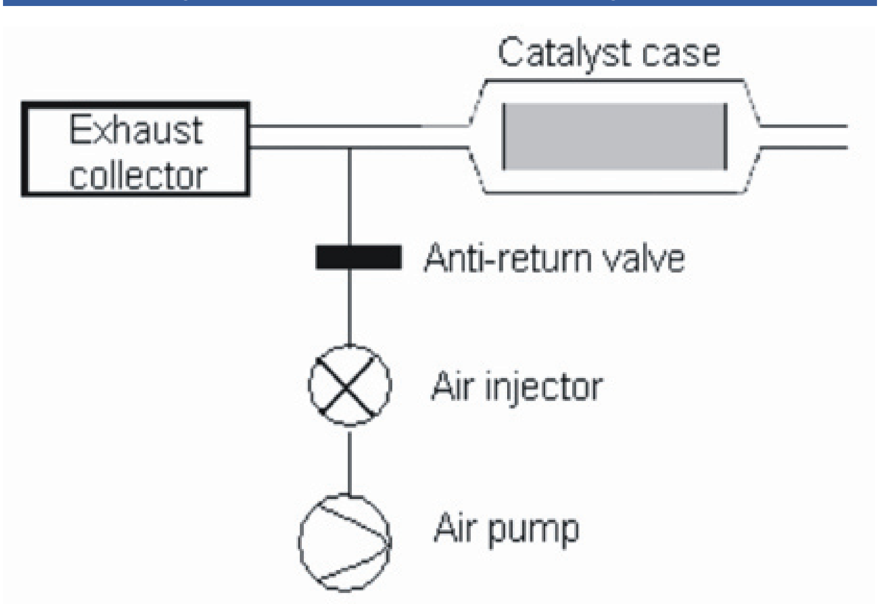
The role of the T.W.C. is to treat three kinds of Noxa which result after the burning of the air-fuel mixture (figure 3). HC (unburnt hidrocarbs), CO (carbon monoxide), NOx (nitrogen oxides), resulting unharmful elements: H2O (water vapours), CO2 (carbon dioxide) and N2 (nitrogen).

The reactions take place in two phases: in the first phase Co and HC are treated by oxidation, the necessary burning oxygen being extraxted as residual oxygen, due to the incomplete burning, and in the second phase the reduction of the nitrogen oxides takes place (Nox). The catalyst case fulfills an extra function, that of stocking oxygen.

This function is ensured by the presence of cerine oxide, CeO2, in the composition of the material covering the active catalytic surface. The cerine oxide acts as regulator of the oxygen concentration in the case of poor mixtures it stocks oxygen, and in the case of high mixtures this one gives oxygen to oxidize CO and HC in CO2.

Experience shows that the conversion degree of T.W.C. is strongly influenced by admitted mixture quality in the engine's cylinders. The mixture quality is characterized by the excess air coefficient, λ, or by the excess fuel coefficient, =1/λ – sometimes also called enriching (richesse).

Fig. 8. The construction of the air injection kicking-off system



As one notices in fugure 4 and figure 5, two fundamental requirements are necessary for the catalyst case to work with maximum efficiency (cca. 98%): the values of λ and be established within the range of 1 0,03 (control window), and the working temperature (run-up) be over 300-350°C.

The first condition is fulfilled by using the lock loop control through the injection computer. The key element that ensures the maintainance of λ in the control window area is the Lambda well – the element measuring the oxygen content in the exhaust gases.

KICKING-OFF METHODS AND THE EFFECT ON DEPOLLUTING PERFORMANCES

Generally, there are two kicking-off methods: the reduction full advance method and the air injection method in the upstream of the catalyst case.

The first method leads to the extension of the burning process to the release motion so that the gases reaching the T.W.C. have a high temperature. Obviously, the method presupposes a sacrifice of dynamic and consumption performances.

Figure 6 presents the evolution of the T.W.C.

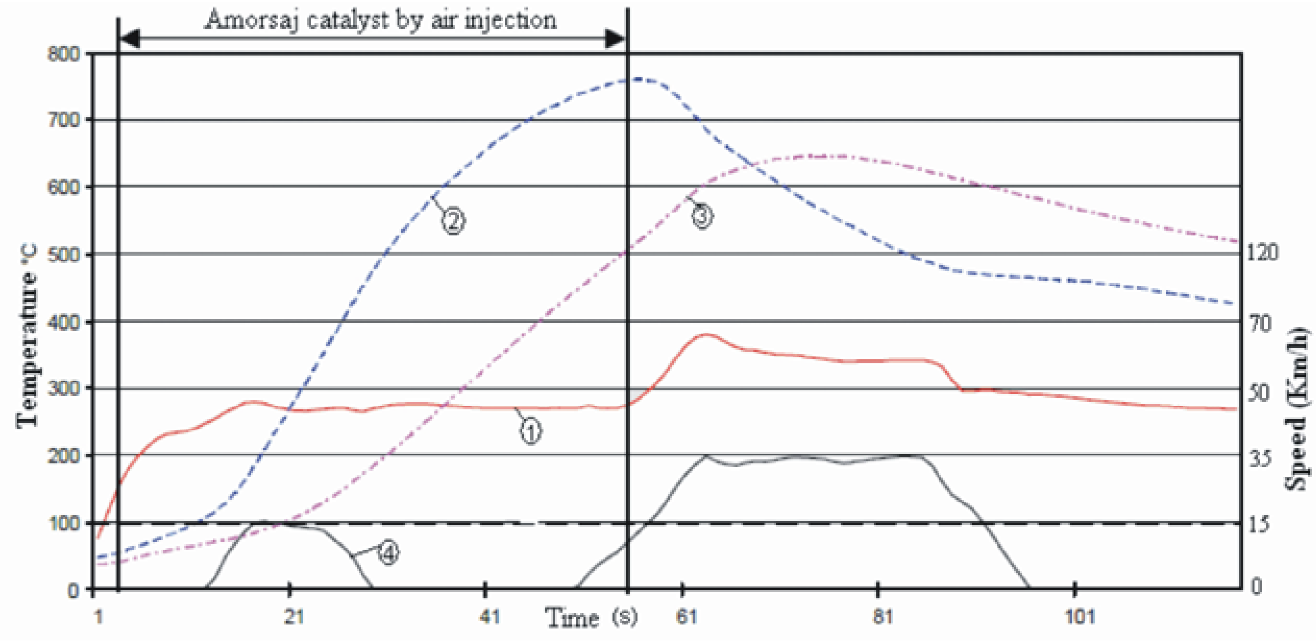


Fig. 9. Graphic representation of the kicking-off time

working temperature depending on the reduction of the preignition, and figure 7 highlights the effect on the HC concentration.

One notices a slight delay in the electric spark start (2…3 0 RAC after the superior dead point – SDP) which leads not only quick start of the catalyst case but also to the minimum of HC concentration. A diminishing of the NOx concentration (effect known in the full advance reduction).This kicking-off way has the advantage of the constrictive simplicity, but also the disadvantage of the fuel consumption increase.

The second method presupposes the injection of some air quantity in the upstream of the T.W.C. using a system made up of the following elements (figure 8): air pump, air injector and antireturn valve. The injected air ensures the necessary oxygen for the post-oxidation reactions for CO and HC so that one obtains a quick temperature increase. In this way the T.W.C. manages to reach easily the running temperature.

Experimental research in the test stand /2/,/6/,/7/ in the case of a middle class car running after the urban cycle (NEDC) have shown that this solution leads to a significant reduction of the run-up time (figure 9). The graphics have the following significance : curve 1 represents the gas temperature entering the catalyst case, curve 2 represents

The temperature of the first catalytic core, curve 3 – the temperature of the second catalytic core, and curve 4 represents the first sequence, respectively the second one of running-speed – in the European cycle (NEDC).

It results that the first catalytic core enters the thermic range (300°C) after almost 20 seconds, and the second core almost 35 seconds, if the car speed isn't higher than 15 km/h. Therefore, the solution allows for the quick kicking-off of the T.W.C. in low charges and rotations of the spark ignition engine conditions, that is the critical conditions regarding Noxa (Co and HC).

CONCLUSIONS

The analysis of the constructive solutions of catalyst cases used to treat the exhaust gases in the depolluting system of the spark ignition engine highlights the fact that obtension of an optimal variant in point of cost and conversion performances require large scale research. Functionally, for a middle class car these research have to aim two objectives: 1. the reduction of the preignition until one reaches a slight delay of the electric spark (2…3 0 RAC after the superior dead point – SDP); 2. the reaching of the working temperature of 300°C between 20 and 30 seconds at the end of the first running sequence after the urban cycle (NEDC), case in which the efficiency of the T.W.C. is of minimum 90%.

* **NOTE**: The theme was done in collaboration with the University of Pitesti, supported by the doctoral scholarships programme POSDRU /88/1.5/S/52826..

REFERENCES:

[1] Heisler H., *Advanced Engine Technology*, SAE, Hardbound, 2005

[2] Plint, J., Martyr, T., *Engine testing , Theory & Practice*, SAE, Casebound,2007

[3] Khair, M., Majewski, A., *Diesel emissions and their control*, SAE, Hardbound,2006

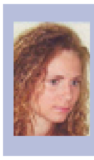
[4] Ivan, F., Liță, D., *Contribution to the improving of the features of a quick cold start Diesel engine*, ESFA București, Volumul 2, 2009

[5] Liță, D., Ivan, F., *Construction, function and influence of exhaust gas recirculation upon pollutants*, Buletin Științific Universitatea din Pitești, Numarul 19, Volumul B, 2009

[6] Liță, D., Ivan, F., *Ways of improving the particle filters' performances used in depolluting the quick Diesel engines*, CONAT Brașov, Volumul 2, 2010

[7] Liță, D., Ivan, F., *Experimental research regarding the depollution of the formation and the improvement of top dynamic performances of cars through the optimization of the final moment of injection*, Buletin Științific Universitatea din Pitești, 2010

Minimizing Distribution Costs – Direct versus Intermediate – Platform Transport



Cristina MANEA
Catholic University of Louvain, Belgium

ABSTRACT

The paper focuses on the key variables to be taken into consideration while choosing the right architecture for the distribution network.

Keywords: distribution network, transport, choice of vehicles, warehouse cost management, transport costs

It begins by presenting the most commonly used distribution architectures discussing their strengths and weaknesses. In the second section, it tackles the issue of the optimal vehicle choice taking as key variable the quantity to be transported and adopting a marginal cost analysis. The choice for the distribution network is discussed next with a special focus on the choice between direct distribution and the use intermediate warehouses. Optimization methods are used based on cost minimization. The last section of the study presents the conclusions of case study pointing out the extent to which the distribution costs might be reduced should the right distribution architecture be used.

INTRODUCTION

Distribution is one of the central pillars of the logistic activity. Today, in a very competitive economy, distribution optimization has to face the challenges of narrowing the delivery time, gaining flexibility and minimizing costs and stocks.

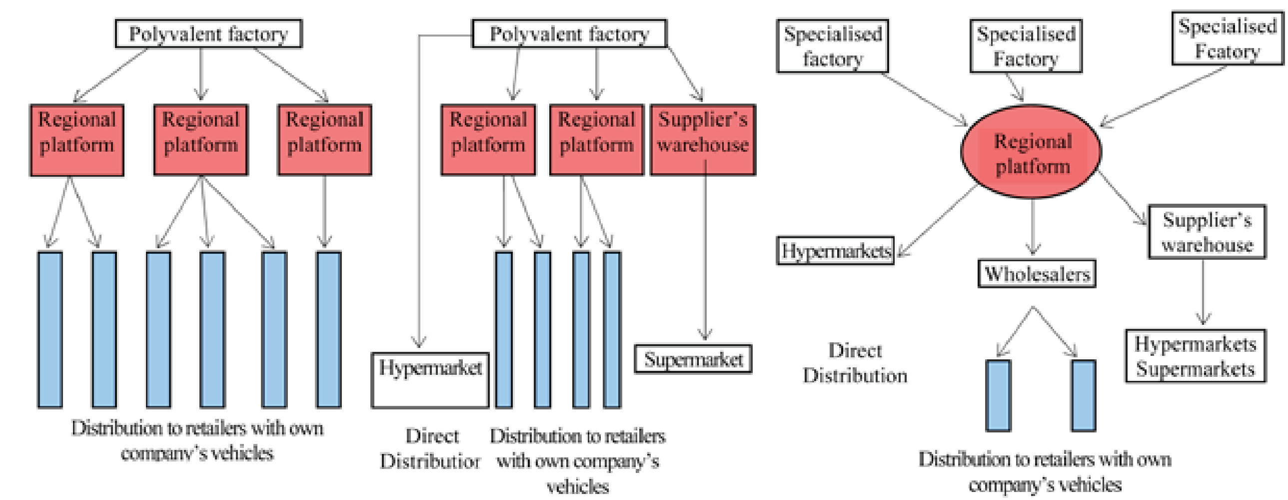
Romanian companies face today the strong foreign competition of the western European markets which are more experimented, have better-organized production flows and integrated logistic chains. Therefore, they have to keep pace by finding solutions in order to optimize as much as possible their flows of resources.

This paper will focus on one key part of the logistic chain of a company, namely distribution of goods, discussing the choice for the right architecture of the distribution network. As road transport is the most commonly used means of transport for freight distribution in Romania, we will narrow our analysis on road distribution.

DISTRIBUTION ARCHITECTURES

Production companies having their production sites in Romania or somewhere else, have to organize their distribution network on the territory of the country in order to optimize their networks. Several objectives such as shorten delivery time, reducing costs, gaining flexibility, proximity to the core clients may be taken into account.

Two decades ago, distribution was made directly from the production plant to the retailers and seldom was the loading rate of a truck taken into account. The pace of distribution was as well not as intensive as it is today, and more time could be waited until the goods were sent to a location. Today, distribution networks are far more complex and the objective is to reduce the intensity of the use of transport by narrowing the number of kilometers traveled by the flows of goods. Logistic solutions today intend grouping the resources and the means of transport. Therefore, a lot of companies build intermediate logistic platforms (We will refer in this paper to a intermediate logistic platform as to a intermediate warehouse and we will ignore all the other function a logistic platform might have.) which allow for grouped transportation operated by big capacity trucks of the merchandise going to different locations up to a location near them (where the



a) Exclusively using intermediate platforms **b) Intermediate platforms and direct distribution** **c) Common intermediate platforms**

Fig.1. Different architectures of the distribution network

Surce (2): Serre, G., L'entrepôt dans la chaîne logistique d'un industriel de grande consommation, présentation au CGPC, 2002 regional warehouse can be found). From here, transport is operated by smaller trucks to each retailer. Therefore, by using big trucks with a low unit cost per kilometer up to a point, the total cost of transport may reduce significantly (and consequently so may the cost of distribution) and by using smaller trucks up to each retailer the company may gain in flexibility.

However, using intermediate platforms is not today a universal solution. It depends primary on the quantities of goods that have to be transported which are depending both on the demand of goods of each retailer and the organization of the production flows (for instance, if they are in *Just in Time* (JIT) or not). Moreover, the cost of distribution doesn't mean only the cost of transport, so the cost of managing a regional platform versus the sum of costs of several small local platforms have as well to be taken into account (often, when using direct distribution and the transport is operated by big capacity trucks, small local warehouses have to build in each city because these types of vehicles are not allowed to enter the urban area).

As we have already pointed out, the great advantage of building intermediate platforms is that the goods can be transported at low unit cost per kilometer up until these warehouses as the total quantity of goods that has to be distributed in the region can be transported together and thus, big capacity trucks with a high loading rate can be used. Intuitively, this implies that there will be a significant advantage in using regional warehouses as long as the quantity that has to be distributed to the retailers is not large enough so as to be directly transported by big capacity trucks.

The quantity to be transported can be a sheer function of the demand of goods, but it can as well be influenced by the organization of the production flows. *Just in Time* (JIT) procedures for instance, require a quasi-continuous production flow as they are aiming at reducing stocks to zero. Consequently, this significantly narrows the quantity to be distributed to each retailer as the distribution has to be operated more often.

When designing the architecture of the distribution network all these aspects have to be taken into account. This is why there are distribution networks which use exclusively direct distribution or exclusively intermediate platforms (fig. 1a) or a mixture of them(fig. 1b).

There are as well companies specialized in distribution which develop strategies of flows coming from different production plants centralization up to a common platform from where the distribution to retailers is going then to be operated (fig. 1c).

This evolution was made possible thanks to the progress of informatics which allowed for an

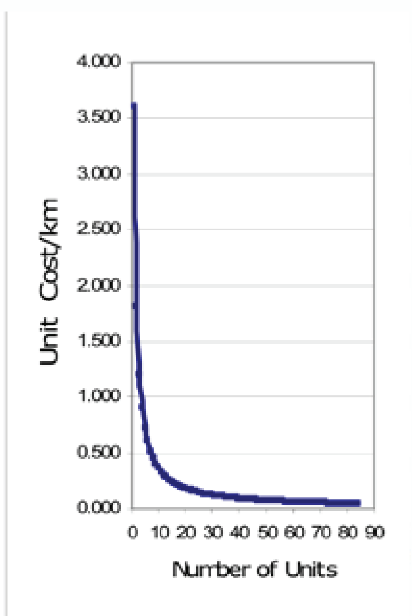


Fig. 2. Unit cost per km and loading rate for a 84m² truck with a trailer attached

Table 1 Usable space [m3] and useful space [units or euroboxes or m3] of the most current used vehicles for goods transport

	Type of vehicle	Dimensions of the loading space			Volume of usable space [m³]	Useful Volume [units of transport or m³]	Ways of loading the units of transport in the usable space		
		Length [m]	Width [m]	Height [m]			In length [units]	In Width [units]	In height [units]
1	Auto chassis with a semitrailer	14,45	2,48	2,91	104	102	17	2	3
2	Auto chassis with a semitrailer	11,9	2,48	2,91	85,9	84	14	2	3
3	Track with a trailer attached	2 x 5,8 = 11,9	2,48	2,91	42,95 x 2= 85,9	84	2 x 7=14	2	3
4	Track without trailer	5,95	2,48	2,91	43	42	7	2	3
5	Van	6,2	1,7	1,94	20,44	20	2	5	2

A „unit of transport" refers to a standard box placed on a standard euro-pallet with a volume of 1m³ (eurobox); The „useful volume" of a vehicle can be easily measured both in number of units of transport (euroboxes) and m³ as the volume of a eurobox is 1m³.

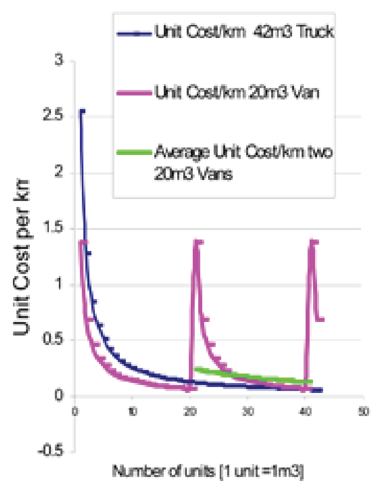
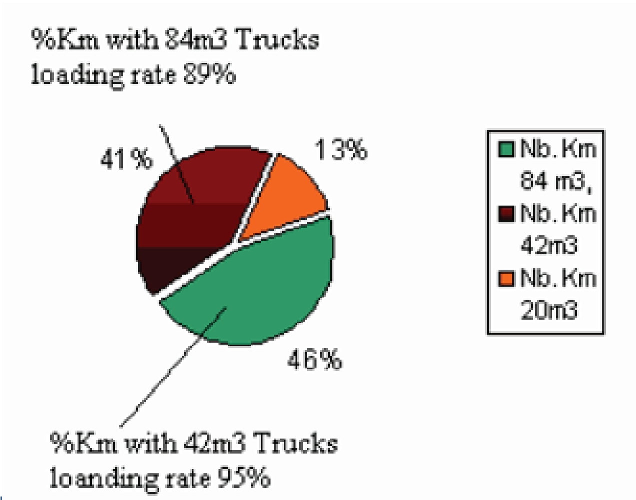


Fig.3. Comparison between unit cost/km for a 42m³ truck and a 20 m³van



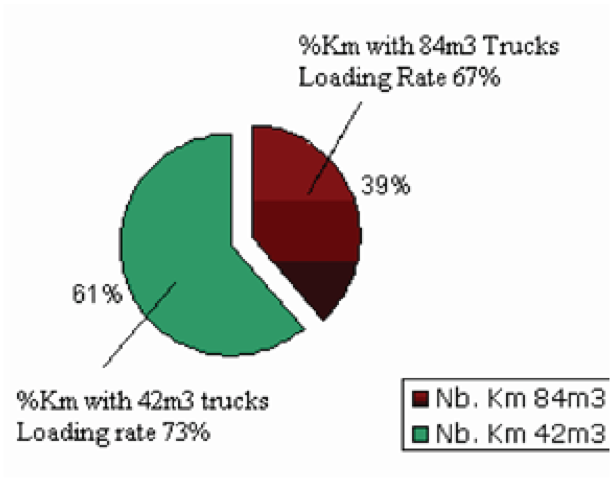


Fig. 4.1. Structure of transport (case with intermediate platforms)

Fig. 4.2. Structure of transport (direct distribution case

integrated management of the production and distribution flows adapting the parameters to a large array of the quantitative and qualitative variables that need to be taken into consideration. «Supply Chain Management» techniques help resource planning and optimization by reducing stocks and costs.

CHOICE OF THE VEHICLES

The transport of goods could be made by several vehicles of different capacities. In the table below (table 1) there are the most commonly used vehicles together with the number of standard boxes on pallets that can be transported with each of them.

While choosing the right vehicle for transport we usually try to have a minimum unit cost by kilometer. In the analysis we are going to suppose that we are working with light loads and therefore we are going to use volumes instead of weights. Moreover, if the loads are light we can attach to a 42m3 truck a 42m3 trailer without biasing its traction and breaking performances. Using this kind of structure, the cost by kilometer will significantly reduce as the price of the auto chassis with a semi-trailer of 84m^3 is higher then the one of a truck of 42m^3 with a 42m^3 trailer attached. Moreover, the taxes in the case of an auto chassis are much higher (almost double in Romania) than in the case of a truck (there is no supplement for the trailer)

However, the unit cost per kilometer depends on the loading rate of the truck. It is true that if we talk about vehicles loaded at their full capacity the unit cost by kilometer will be lower if the capacity of the truck is bigger. However, if the vehicle does not run at its full capacity, the unit cost raises progressively (fig 2).

If the quantity of goods is large enough so as to allow the transport with big vehicles loaded at their full capacity, direct distribution might be the right choice. By contrast, if the quantities to each reseller are not large enough the solution of intermediate platforms might be the optimal one.

Using intermediate platforms, the quantities that have to be transported up to the regional warehouses are larger (as it responds to the demand of all retailers in the sector) and thus, can be transported for a fraction of the distance at lower unit costs per kilometer by large capacity vehicles.

By contrast, the transport from the regional warehouse to each retailer is operated by vehicles of smaller capacity. The choice for these vehicles has to be made depending on the quantity that has to be transported. For instance, a 42m^3 truck can be the right solution, but if it is not loaded at its full capacity the unit cost per kilometer can be higher than the one obtained with a smaller vehicle (a 20 m^{3}van for example) whose loading rate would be much higher. As we can see in figure 3, an attentive analysis has to be drawn.

The issue is to determine the limit (in number of *euroboxes*) starting from which it is better to use vans of 20m^3 instead trucks of 42m^3. In figure 3 we can see the variation of the unit cost per kilometer depending on the loading rate of the vehicle for a 42m^3 truck and a van of 20m^3. As starting with 21 transport unities a second van has to be used, the marginal cost of the 21st unity is very high (a single box bears the whole cost per kilometer of the van).

As we can see in this graphic, for a quantity smaller than 20 unities it is cheaper to use vans of 20m^3 (the unit cost curve depending on the loading rate of a van is laying below the one of a truck of 42m^3). For quantities larger than 20m^3 we need to draw the average unit cost curve as two vans are used. We can see that in this case it is better to use 42m^{3}trucks, even if they are not loaded at their full capacity instead of vans of 20m^3 (the curve of the unit cost depending on the loading rate for a truck of 42 m^3 is laying below the average unit cost of two 20m^3 vans for a quantity equal or larger than 21 m^3).

CHOICE OF THE DISTRIBUTION NETWORK AND COST OPTIMISATION

Distribution costs include both the cost of transport and the cost of warehouses management. Therefore, when choosing the distribution network architecture both two components have to be taken into account.

Direct distribution usually imply that small local warehouses have to be build in each city where the retailers can be found if transport is operated by large trucks as the type of vehicles are not allowed to enter the urban area. By contrast, when choosing to build a regional intermediate platform small local warehouse are useless, as transport from the platform to each reseller is operated by smaller capacity trucks which are allowed to enter the city zone. Therefore, using an intermediate platform might contribute to the reduction of the cost of distribution if the

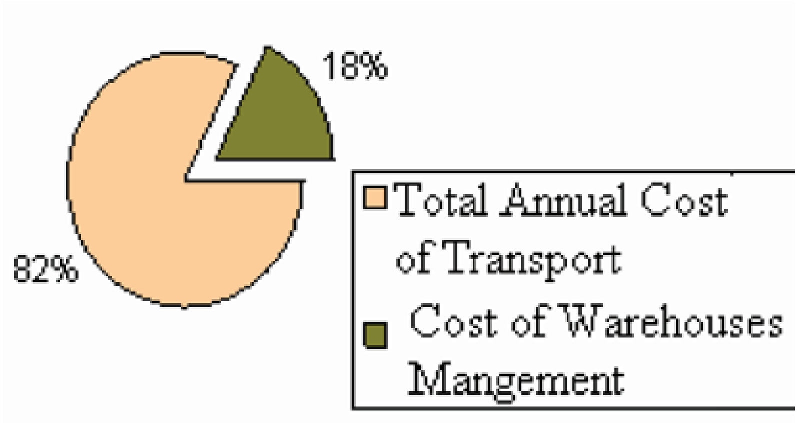


Fig. 5.1. a Comparison between distribution cost repartition between costs of transport and costs of warehouses management- case with intermediate platforms

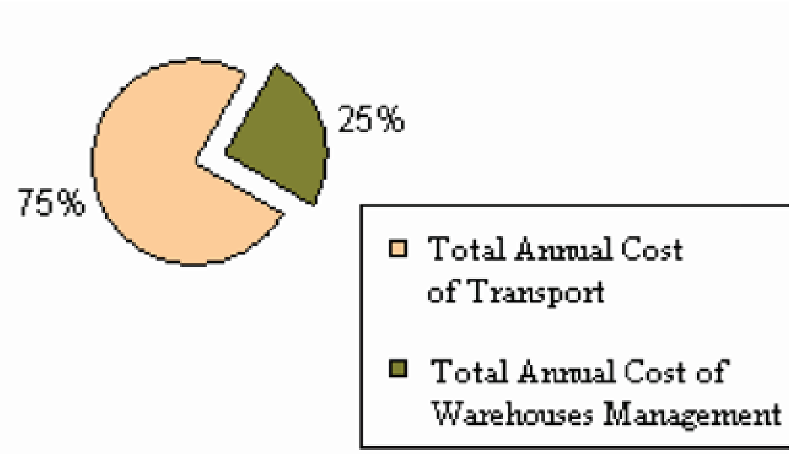


Fig. 5.1. b Comparison between distribution cost repartition between costs of transport and costs of warehouses management- direct distribution case

annual management cost of a regional warehouse is lower than the sum of annual costs of all local warehouses in the region.

A large part of the cost of distribution is represented by the cost of transport and thus, this component has to be as well optimized: depending on quantities that have to be transported and their destination (which depend as well on other characteristics we have already discussed in previous sections) the right architecture of the distribution network has to be chosen.

The two components (transportation cost and warehouse management cost) have to be simultaneously taken into account so as to be able to compare total distribution costs for different types of networks (direct, intermediate-platform or mixed) and to take a decision.

According to a recent study on a Romanian company with a country-wide distribution network and a JIT production structure, its annual distribution cost may be narrowed by 23,98% should it use a distribution network with regional warehouses instead of a direct one. This reduction is the result of a 16% reduction in the cost of transport and 45% reduction in the cost of annual warehouse management costs.

The lower cost of transportation comparatively to direct distribution comes from the fact that the unit cost per kilometer is significantly lower, as a larger fraction of the total distance is operated with larger and more loaded to their full capacity trucks than in the case of direct distribution. If we look at figure 4, we can see that in the case of intermediate platforms 46% of the total distance is operated by 84 m^3 trucks loaded at 89% of their full capacity, while in the case of direct distribution only 39% of the total distance is operated by 84m^3 trucks loaded at only 67% of their full capacity. Moreover, the transport on the residual kilometers are better optimized in the case of intermediate platforms as 46% is operated with 42m^3 at 95% level of loading and the rest of 13% with vans of 20m^3, while in the other case the trucks of 42m^3 are used at 73% of their capacity.

The analysis of costs has been made after transport optimization both in the case of direct and intermediate platform. For the method used as well for all the calculation of costs see reference (1). There are as well a great number of specific software developed for transport optimization which take into consideration distances that have to be traveled and quantities that have to be transported in order to offer the proper distribution architecture. The analysis can be made using management accounting data from the past or future estimates.

As for the other component of the distribution cost, the warehouses management cost (fig5), having one intermediate platform in each region instead of several local ones (one in each city of the region) significantly lowered the annual cost of warehouses management as its proportion in the total distribution cost is only 18% in the case of intermediate platforms versus 25% in the case of direct distribution (moreoever, we have to keep in mind that in the present case study the cost of distribution in the case of intermediate platforms is lower than in the case of direct one which implies the 18% of the distribution cost in the first case is lower than 25% of the distribution cost in the second).

CONCLUSION

As we can see, choosing the right architecture for the distribution network as well as using vehicles with the right capacity can largely contribute to the optimization of the resource use and can help to gain competitiveness on the market.

BIBLIOGRAFIE:

[1] MANEA C., *Variables stratégiques dans l'emplacement des plateformes intermédiaires logistiques*, Bachelor's Degree Thesisi, Economics Studies Bucharest, 2009

[2] SERRE, G., *L'entrepôt dans la chaîne logistique d'un industriel de grande consommation*, CGPC presentation, 2002.

Pollutant Emissions Exhausted by a Spark Ignition Engine Operating with Conventional and Non-conventional Fuels



L. MIHON A. NEGOIȚESCU A.TOKAR

Universitatea Politehnica din Timișoara,
Facultatea de Mecanică, Catedra T.M.T.A.R.
arina.negoitescu@yahoo.com

ABSTRACT

Under the circumstances of an increasing trend in the number of vehicles, as a daily need, the article aim is to evaluate and reduce the pollutant emissions level in urban traffic. Most dangerous effects of pollution produced by internal combustion engines are represented by the emission of harmful gases in the atmosphere. Experimental simulation performed on the roller stand LPS 3000, targeted areas with high congestion of vehicles, both at peak and low traffic times. The analyzed trail has a length of 2km. The CO2, CO, HC and NOx emissions values exhausted by a BMW 535i car, equipped with a spark ignition engine and powered by conventional and non-conventional fuels, were evaluated. The solution of NONEURO vehicles operating with GPL in urban traffic is advantageous from the pollutant emissions point of view.

Keywords: noxious, traffic, pollution, roller stand, gasoline, Liquefied Petroleum Gas

INTRODUCTION

The trend of increasing the number of vehicles, as a daily necessity, follows the evaluation of the pollutant emissions reduction of a vehicle equipped with spark ignition engine [2]. On this purpose were simulated in the laboratory to real driving conditions for the BMW E28 535i vehicle equipped with spark ignition engine, running on LPG and petrol. The studied route creates the vehicles transient mode conditions, which involves idling or unloaded, used to start the engine (engine cold) and waiting situations (traffic lights, congestion), acceleration, deceleration, etc. The tests carried on have allowed a large experimental data recording, plus those obtained by calculation based on measured values [1]. The experimental investigations were performed in the Automotive Engineering Laboratory of the University Politehnica of Timisoara, and in traffic. Experimental researches were focused on areas with vehicles traffic congestion both at peak hours and low period of traffic.


Fig. 1. The Leonardo da Vinci Square, Timisoara


Fig. 2. The experimental model

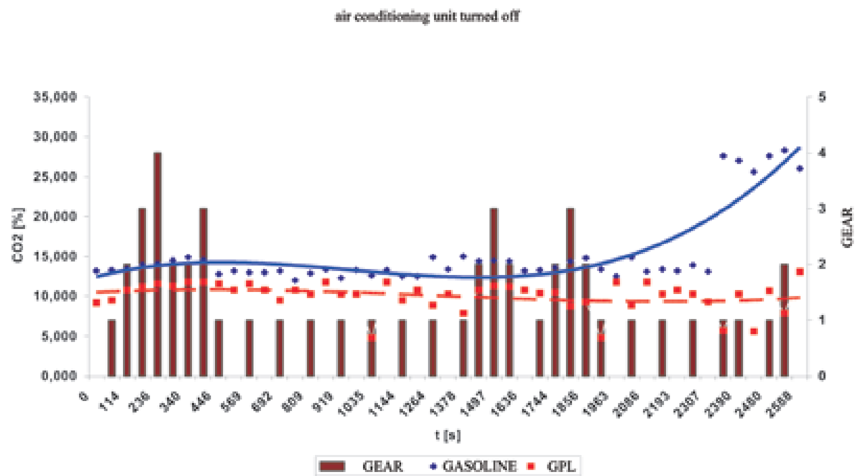


Fig. 3. CO_2 emission versus time

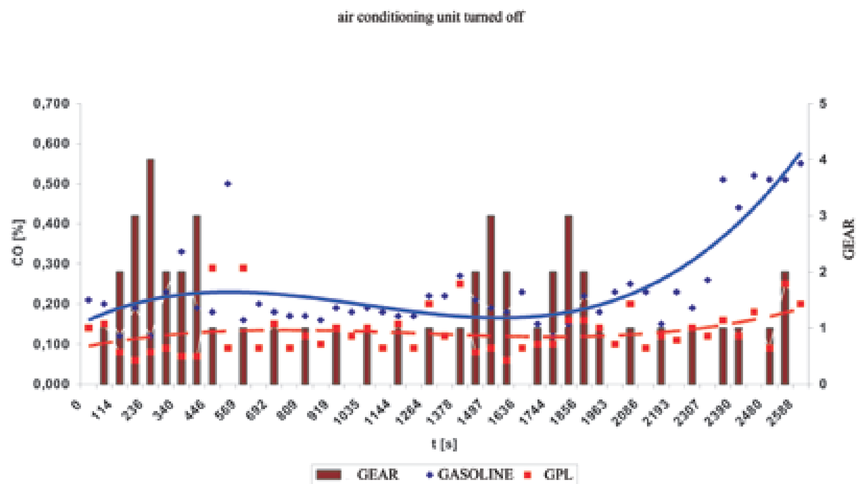


Fig. 4. CO emission versus time

The studied route has a length of 2km, when the movement, at peak hours takes place „bumper to bumper". The four traffic lights crossroads, a roundabout (Fig. 1) and six pedestrian crossings without traffic lights on a relatively short distance create high agglomeration areas of pollution with obvious effects on the environment pollution.

THE INSTALLATION USED FOR TESTINGS

In order to accomplish the experimental model of simulation for running in traffic (Fig. 2) it was used

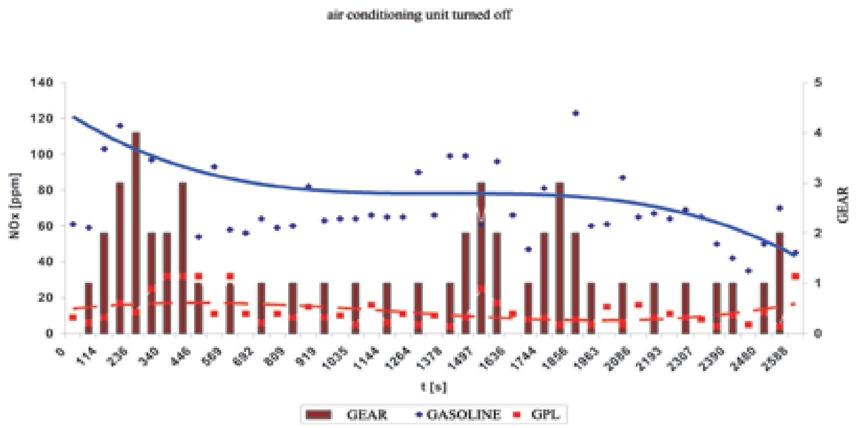


Fig. 5. NOx emission versus time

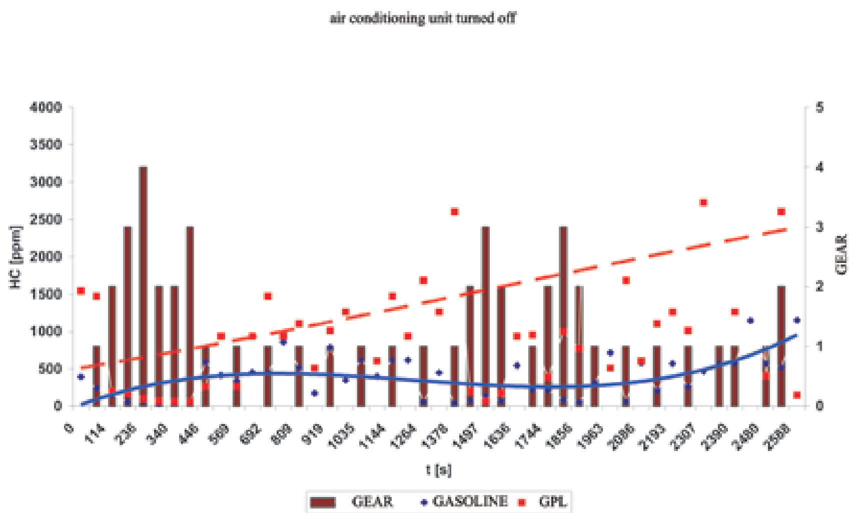


Fig. 6. HC emission versus time

the chassis dynamometer LPS3000 correlated with AVL Dicom 4000 gas analyzer for pollutants measuring [3], [4].

LPS3000 allows the engine performance testing. The simulation on the dynamometer is performed with an eddy current braking system, and can measure the Otto and Diesel engines power. The cooling fan for cooling allows the air resistance simulation. For sampling exhausted emissions AVL Dicom 4000 gas analyzer was used from the laboratory. The infrared measuring is used as the measurement principle for CO, HC, CO2, and electrochemical measurement for NOx.

COMPARATIVE STUDY OF THE EXPERIMENTAL RESULTS OBTAINED UNDER THE SIMULATED WAY CONDITIONS

The evaluation of the possibility to simulate on the test stand was done by following the next steps: A route determination which includes at least one traffic light and one without traffic light crossing (Fig. 1), pedestrian crossings with and without traffic lights; The route passing in order to record the time and traffic conditions; The route simulation on the chassis dynamometer; The creation of climate conditions similar to those of traffic in the laboratory. Evaluations were made on a vehicle equipped with BMW spark ignition engine type (supplied with petrol and LPG), with the passenger area with air-conditioning unit turned off.

When operating on LPG vehicle it can be observed a significant decrease in CO_2 concentration, compared to petrol operating modes. Furthermore, for an increase of the engine thermal modes, the CO_2 concentration reduction in exhaust gases is even more relevant, from 30% to 10%. The measured values are below the calculated ones, the curves variation showing the same of allure (Fig. 3).

When the engines run on LPG, CO emissions have an almost linear trend, falling below 0.2%, being significantly lower than the engine running on gasoline. In this situation the differences between experiment and calculation are very high this time, the calculated values exceeding the measured ones (Fig. 4). The concentration of NOx emissions is about 15ppm, recording a slightly increasing trend along the route and calculated values are considerably higher than the measured ones (Fig. 5). While following the default route, the concentration of HC emission presents an increasing trend from 150ppm to over 1100ppm. It can be observed that the maximum values are recorded in the route zone with several stops and starts, and at higher speed operation these values are much lower. The measured values are above those calculated, recording approaches between those measured and calculated after the First gear operation mode. When LPG is used as fuel, HC concentrations are comparable to those recorded when the engine runs on gasoline (Fig. 6).

CONCLUSIONS

It can be concluded that LPG is an alternative for vehicles that are forced to be within the emission standards which became more stringent, and for those NONEURO (the tested car) is a necessity. The LPG combustion produces up to 15-20% less CO_2 compared to gasoline engines. Modern gasoline engines are excellently suitable for gas conversion. The idling operating mode is in average around 20-30% used, the acceleration mode 20-25%, deceleration around 17-20% and 30-40% the constant speed operating mode, but they largely depend on the vehicle technical characteristics, traffic nature, road quality, fuel, weather, driver physical and mental condition state, etc.

By taking under consideration the large registered NONEURO number of vehicles which are still in traffic, the solution for their operation on LPG under these conditions is advantageous in terms of emissions, even though there are also some disadvantages of using this fuel type (power loss, the piston - rings wear for 50000-60000 km).

BIBLIOGRAFIE:

[1] Tokar A., *Cercetări privind interacțiunea dintre automobilul echipat cu motor cu ardere internă și mediu*, Teza de doctorat, Editura Politehnica Timișoara, 2009.

[2] Negrea V.D., Sandu V., *Combaterea poluării mediului în transporturile rutiere*, Editura Tehnică București, 2000

[3] *** *Chassis Dynamometer LPS3000, Stand Operating Instructions and Users Manual*, MAHA Maschinenbau Haldenwang GmbH&Co.KG, 87490 Haldenwang, Germany, 2003.

[4] *** *AVL DiCom4000, User Manual version 1.01*, AVL LIST GMBH, Graz, Austria, 1998.

Determination of the Mechanical Efficiency of the Gears

F.I. PETRESCU   R.V. PETRESCU

1Polytechnic University of Bucharest, Romania
Petrescu.Florian@gmail.com
Petrescu.Victoria@gmail.com

ABSTRACT

The paper presents an original method to determine the efficiency of the gear. The originality of this method relies on the eliminated friction modulus. The paper is analyzing the influence of a few parameters concerning gear efficiency. These parameters are: z1 - the number of teeth for the primary wheel of gear; z2 - the number of teeth of the secondary wheel of gear; α0 - the normal pressure angle on the divided circle; β - the inclination angle. With the relations presented in this paper, one can make the synthesis of the gear's mechanisms.

Key words: Gear, efficiency, tooth, pressure angle, wheel

INTRODUCTION

In this paper the authors present an original method to calculating the efficiency of the gear.

The originality consists in the way of determination of the gear's efficiency because one hasn't used the friction forces of couple (this new way eliminates the classical method). One eliminates the necessity of determining the friction coefficients by different experimental methods as well. The efficiency determined by the new method is the same like the classical efficiency, namely the mechanical efficiency of the gear.

Some mechanisms work by pulses and are transmitting the movement from an element to another by pulses and not by friction. Gears work practically only by pulses. The component of slip or friction is practically the loss. Because of this the mechanical efficacy becomes practically the mechanical efficiency of gear.

The paper is analyzing the influence of a few parameters concerning gear efficiency. With the relations presented in this paper, one can synthesize the gear's mechanisms. Today, the gears are present every where in the mechanical's worl

DETERMINING THE MOMENTARY MECHANICAL EFFICIENCY

The calculating relations are the next (1-20), (see the fig. 1-2.a):

$$\begin{cases} F_\tau = F_m \cdot \cos\alpha_1 \quad F_\psi = F_m \cdot \sin\alpha_1 \quad \overline{F_m} = \overline{F_\tau} + \overline{F_\psi} \\ v_2 = v_1 \cdot \cos\alpha_1 \quad v_{12} = v_1 \cdot \sin\alpha_1 \quad \overline{v_1} = \overline{v_2} + \overline{v_{12}} \end{cases} \tag{1}$$

With: F_m - the motive force (the driving force); F_t - the transmitted force (the useful force); F_y - the slide force (the lost force); v_1 - the velocity of element 1, or the speed of wheel 1 (the driving wheel); v_2 - the velocity of element 2, or the speed of wheel 2 (the driven wheel); v_{12} - the relative speed of the wheel 1 in relation with the wheel 2 (this is a sliding speed); (see fig. 2.a);

$$P_c \equiv P_m = F_m \cdot v_1 \tag{2}$$

The consumed power (in this case the driving power) takes the form (2). The useful power (the transmitted power from the profile 1 to the profile 2) will be written in the relation (3). The lost power will be written in the form (4).

$$P_u = P_\tau = F_\tau \cdot v_2 = F_m \cdot v_1 \cdot \cos^2\alpha_1 \tag{3}$$

$$P_\psi = F_\psi \cdot v_{12} = F_m \cdot v_1 \cdot \sin^2\alpha_1 \tag{4}$$

The momentary efficiency of couple will be calculated directly with the relation (5).

$$\left\{ \eta_i = \frac{P_u}{P_c} = \frac{P_\tau}{P_m} = \frac{F_m \cdot v_1 \cdot \cos^2\alpha_1}{F_m \cdot v_1} \quad \eta_i = \cos^2\alpha_1 \right. \tag{5}$$

The momentary losing coefficient will be written in the form (6).

$$\begin{cases} \psi_i = \dfrac{P_\psi}{P_m} = \dfrac{F_m \cdot v_1 \cdot \sin^2\alpha_1}{F_m \cdot v_1} = \sin^2\alpha_1 \\ \eta_i + \psi_i = \cos^2\alpha_1 + \sin^2\alpha_1 = 1 \end{cases} \tag{6}$$

One can easily see that the sum of the momentary efficiency and the momentary losing coefficient is 1. It determines now the geometrical elements of the gear. These elements will be used in synthesis of the couple efficiency, η.

THE GEOMETRICAL ELEMENTS OF THE GEAR

And now it can see the next geometrical elements of the external gear (for the right teeth, b=0): The radius of the basic circle of wheel 1 (of the driving wheel) (7); the radius of the outside circle of wheel 1 (8); the maximum pressure angle of the gear (9).

$$r_{b1} = \frac{1}{2} \cdot m \cdot z_1 \cdot \cos\alpha_0 \tag{7}$$

$$r_{a1} = \frac{1}{2} \cdot (m \cdot z_1 + 2 \cdot m) = \frac{m}{2} \cdot (z_1 + 2) \tag{8}$$

$$\cos\alpha_{1M} = \frac{r_{b1}}{r_{a1}} = \frac{\frac{1}{2} \cdot m \cdot z_1 \cdot \cos\alpha_0}{\frac{1}{2} \cdot m \cdot (z_1 + 2)} = \frac{z_1 \cdot \cos\alpha_0}{z_1 + 2} \tag{9}$$

And one determines the same parameters for the wheel 2: the radius of basic circle (10) and the radius of the outside circle (11). We can see now the minimum pressure angle of the external gear (12), and, for the external gear, the minimum (12) and the maximum (9) pressure angle for the right teeth. For the external gear with bended teeth (b¹0) one uses the relations (13, 14 and 15). For the internal gear with bended teeth (b¹0) one uses the relations (13 with 16, 17-A or with 18, 19-B).

$$r_{b2} = \frac{1}{2} \cdot m \cdot z_2 \cdot \cos\alpha_0 \tag{10}$$

$$r_{a2} = \frac{m}{2} \cdot (z_2 + 2) \ (11) \quad tg\alpha_t = \frac{tg\alpha_0}{\cos\beta} \tag{11}$$

$$tg\alpha_{1m} = [(z_1 + z_2) \cdot \sin\alpha_0 - \sqrt{z_2^2 \cdot \sin^2\alpha_0 + 4 \cdot z_2 + 4}]/(z_1 \cdot \cos\alpha_0) \tag{12}$$

$$tg\alpha_t = \frac{tg\alpha_0}{\cos\beta} \tag{13}$$

$$tg\alpha_{1m} = [(z_1 + z_2) \cdot \frac{\sin\alpha_t}{\cos\beta} - \sqrt{z_2^2 \cdot \frac{\sin^2\alpha_t}{\cos^2\beta} + 4 \cdot \frac{z_2}{\cos\beta} + 4}] \cdot \frac{\cos\beta}{z_1 \cdot \cos\alpha_t} \tag{14}$$

$$\cos\alpha_{1M} = \frac{\cos\beta}{\frac{z_1}{\cos\beta} + 2} \tag{15}$$

$$tg\alpha_{1m} = [(z_1 - z_2) \cdot \frac{\sin\alpha_t}{\cos\beta} + \sqrt{z_2^2 \cdot \frac{\sin^2\alpha_t}{\cos^2\beta} - 4 \cdot \frac{z_2}{\cos\beta} + 4}] \cdot \frac{\cos\beta}{z_1 \cdot \cos\alpha_t} \tag{16}$$

$$\cos\alpha_{1M} = \frac{\frac{z_1 \cdot \cos\alpha_t}{\cos\beta}}{\frac{z_1}{\cos\beta} + 2} \tag{17}$$

$$tg\alpha_{1M} = [(z_1 - z_2) \cdot \frac{\sin\alpha_t}{\cos\beta} + \sqrt{z_2^2 \cdot \frac{\sin^2\alpha_t}{\cos^2\beta} + 4 \cdot \frac{z_2}{\cos\beta} + 4}] \cdot \frac{\cos\beta}{z_1 \cdot \cos\alpha_t} \tag{18}$$

$$\cos\alpha_{1m} = \frac{\frac{z_1 \cdot \cos\alpha_t}{\cos\beta}}{\frac{z_1}{\cos\beta} - 2} \tag{19}$$

$$\eta = \frac{1}{\Delta\alpha} \cdot \int_{\alpha_m}^{\alpha_M} \eta_i \cdot d\alpha = \frac{1}{\Delta\alpha} \int_{\alpha_m}^{\alpha_M} \cos^2\alpha \cdot d\alpha =$$

$$= \frac{1}{2\cdot\Delta\alpha}\cdot[\frac{1}{2}\cdot\sin(2\cdot\alpha)+\alpha]_{\alpha_m}^{\alpha_M} =$$
$$= \frac{1}{2\cdot\Delta\alpha}[\frac{\sin(2\alpha_M)-\sin(2\alpha_m)}{2}+\Delta\alpha] =$$
$$= \frac{\sin(2\cdot\alpha_M)-\sin(2\cdot\alpha_m)}{4\cdot(\alpha_M-\alpha_m)}+0.5 \qquad (20)$$

DETERMINATION OF THE EFFICIENCY

The efficiency of the gear will be calculated through the integration of momentary efficiency on all sections of gearing movement, namely from the minimum pressure angle to the maximum pressure angle; the relation (20).

DETERMINING OF GEARING EFFICIENCY IN FUNCTION OF THE CONTACT RATIO

One calculates the efficiency of a geared transmission, having in view the fact that at one moment there are several couples of teeth in contact, and not just one. The start model has got four pairs of teeth in contact (4 couples) concomitantly. The first couple of teeth in contact has the contact point i, defined by the ray r_{i1}, and the pressure angle a_{i1}; the forces which act at this point are: the motor force F_{mi}, perpendicular to the position vector r_{i1} at i and the force transmitted from the wheel 1 to the wheel 2 through the point i, F_{ti}, parallel to the path of action and with the sense from the wheel 1 to the wheel 2, the transmitted force being practically the projection of the motor force on the path of action; the defined velocities are similar to the forces (having in view the original kinematics, or the precise kinematics adopted); the same parameters will be defined for the next three points of contact, j, k, l (Fig. 2.b-4). For starting we write the relations between the velocities (21). From relations (21), one obtains the equality of the tangential velocities (22), and makes explicit the motor velocities (23).

$$\begin{aligned} v_{ti} &= v_{mi}\cdot\cos\alpha_i = r_i\cdot\omega_1\cdot\cos\alpha_i = r_{b1}\cdot\omega_1 \\ v_{tj} &= v_{mj}\cdot\cos\alpha_j = r_j\cdot\omega_1\cdot\cos\alpha_j = r_{b1}\cdot\omega_1 \\ v_{tk} &= v_{mk}\cdot\cos\alpha_k = r_k\cdot\omega_1\cdot\cos\alpha_k = r_{b1}\cdot\omega_1 \\ v_{tl} &= v_{ml}\cdot\cos\alpha_l = r_l\cdot\omega_1\cdot\cos\alpha_l = r_{b1}\cdot\omega_1 \end{aligned} \qquad (21)$$

$$v_{ti} = v_{tj} = v_{tk} = v_{tl} = r_{b1}\cdot\omega_1 \qquad (22)$$

$$v_{mi} = \frac{r_{b1}\cdot\omega_1}{\cos\alpha_i}; v_{mj} = \frac{r_{b1}\cdot\omega_1}{\cos\alpha_j}; v_{mk} = \frac{r_{b1}\cdot\omega_1}{\cos\alpha_k}; v_{ml} = \frac{r_{b1}\cdot\omega_1}{\cos\alpha_l} \qquad (23)$$

The forces transmitted concomitantly at the four points must be the same (24). The motor forces are (25). The momentary efficiency can be written in the form (26).

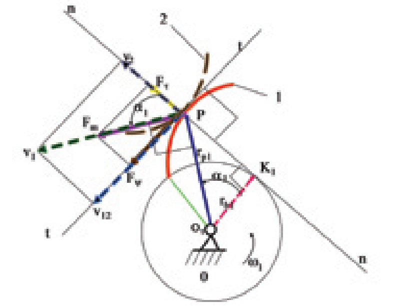


Fig. 2. a.

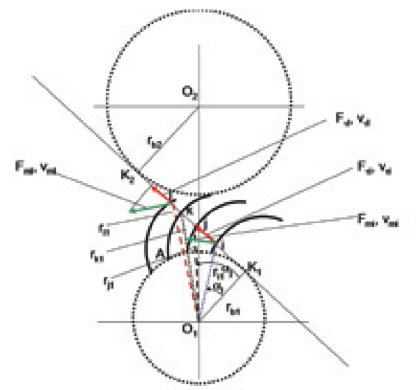


Fig. 2. b.

$$F_{ti} = F_{tj} = F_{tk} = F_{tl} = F_\tau \qquad (24)$$

$$F_{mi} = \frac{F_\tau}{\cos\alpha_i}; F_{mj} = \frac{F_\tau}{\cos\alpha_j}; F_{mk} = \frac{F_\tau}{\cos\alpha_k}; F_{ml} = \frac{F_\tau}{\cos\alpha_l} \qquad (25)$$

$$\eta_i = \frac{P_u}{P_c} = \frac{P_\tau}{P_m} = \frac{F_{ti}\cdot v_{ti}+F_{tj}\cdot v_{tj}+F_{tk}\cdot v_{tk}+F_{tl}\cdot v_{tl}}{F_{mi}\cdot v_{mi}+F_{mj}\cdot v_{mj}+F_{mk}\cdot v_{mk}+F_{ml}\cdot v_{ml}} =$$
$$= \frac{4\cdot F_\tau\cdot r_{b1}\cdot\omega_1}{\frac{F_\tau\cdot r_{b1}\cdot\omega_1}{\cos^2\alpha_i}+\frac{F_\tau\cdot r_{b1}\cdot\omega_1}{\cos^2\alpha_j}+\frac{F_\tau\cdot r_{b1}\cdot\omega_1}{\cos^2\alpha_k}+\frac{F_\tau\cdot r_{b1}\cdot\omega_1}{\cos^2\alpha_l}} =$$
$$= \frac{4}{\frac{1}{\cos^2\alpha_i}+\frac{1}{\cos^2\alpha_j}+\frac{1}{\cos^2\alpha_k}+\frac{1}{\cos^2\alpha_l}} =$$
$$= \frac{4}{4+tg^2\alpha_i+tg^2\alpha_j+tg^2\alpha_k+tg^2\alpha_l} \qquad (26)$$

Relations (27) and (28) are auxiliary.

One keeps relations (28), with the sign plus (+) for the gearing where the drive wheel 1 has external teeth (at the external or internal gearing) and with the sign (-) for the gearing where the drive wheel 1, has internal teeth (the drive wheel is a ring, only for the internal gearing). The relation of the momentary efficiency (26) uses the auxiliary relations (28) and takes the form (29).

$$K_1 i = r_{b1}\cdot tg\alpha_i; K_1 j = r_{b1}\cdot tg\alpha_j; K_1 k = r_{b1}\cdot tg\alpha_k; K_1 l = r_{b1}\cdot tg\alpha_l \qquad (27)$$
$$K_1 j - K_1 i = r_{b1}\cdot(tg\alpha_j - tg\alpha_i); K_1 j - K_1 i = r_{b1}\cdot\frac{2\cdot\pi}{z_1} \Rightarrow tg\alpha_j = tg\alpha_i + \frac{2\cdot\pi}{z_1}$$
$$K_1 k - K_1 i = r_{b1}\cdot(tg\alpha_k - tg\alpha_i); K_1 k - K_1 i = r_{b1}\cdot 2\cdot\frac{2\cdot\pi}{z_1} \Rightarrow tg\alpha_k = tg\alpha_i + 2\cdot\frac{2\cdot\pi}{z_1}$$
$$K_1 l - K_1 i = r_{b1}\cdot(tg\alpha_l - tg\alpha_i); K_1 l - K_1 i = r_{b1}\cdot 3\cdot\frac{2\cdot\pi}{z_1} \Rightarrow tg\alpha_l = tg\alpha_i + 3\cdot\frac{2\cdot\pi}{z_1}$$

$$tg\alpha_j = tg\alpha_i \pm \frac{2\cdot\pi}{z_1}; tg\alpha_k = tg\alpha_i \pm 2\cdot\frac{2\cdot\pi}{z_1};$$
$$tg\alpha_l = tg\alpha_i \pm 3\cdot\frac{2\cdot\pi}{z_1} \qquad (28)$$

$$\eta_i = \frac{4}{4+tg^2\alpha_i+tg^2\alpha_j+tg^2\alpha_k+tg^2\alpha_l} = \qquad (29)$$
$$= \frac{4}{4+tg^2\alpha_i+(tg\alpha_i\pm\frac{2\pi}{z_1})^2+(tg\alpha_i\pm 2\cdot\frac{2\pi}{z_1})^2+(tg\alpha_i\pm 3\cdot\frac{2\pi}{z_1})^2} =$$
$$= \frac{4}{4+4\cdot tg^2\alpha_i+\frac{4\pi^2}{z_1^2}\cdot(0^2+1^2+2^2+3^2)\pm 2\cdot tg\alpha_i\cdot\frac{2\pi}{z_1}\cdot(0+1+2+3)} =$$
$$= \frac{1}{1+tg^2\alpha_i+\frac{4\pi^2}{E\cdot z_1^2}\cdot\sum_{i=1}^{E}(i-1)^2\pm 2\cdot tg\alpha_i\cdot\frac{2\pi}{E\cdot z_1}\cdot\sum_{i=1}^{E}(i-1)} =$$
$$= \frac{1}{1+tg^2\alpha_i+\frac{4\pi^2}{E\cdot z_1^2}\cdot\frac{E\cdot(E-1)\cdot(2\cdot E-1)}{6}\pm\frac{4\pi\cdot tg\alpha_i}{E\cdot z_1}\cdot\frac{E\cdot(E-1)}{2}} =$$
$$= \frac{1}{1+tg^2\alpha_i+\frac{2\pi^2\cdot(E-1)\cdot(2E-1)}{3\cdot z_1^2}\pm\frac{2\pi\cdot tg\alpha_i\cdot(E-1)}{z_1}} =$$
$$= \frac{1}{1+tg^2\alpha_i+\frac{2\pi^2}{3\cdot z_1^2}\cdot(\varepsilon_{12}-1)\cdot(2\cdot\varepsilon_{12}-1)\pm\frac{2\pi\cdot tg\alpha_i}{z_1}\cdot(\varepsilon_{12}-1)}$$

In expression (29) one starts with relation (26) where four pairs are in contact concomitantly, but then one generalizes the expression, replacing the 4 (four pairs from fig. 2.b) with E couples; replacing figure 4 with the E variable, which represents the whole number of the contact ratio +1, and after restricting the sums expressions, we replace the variable E with the contact ratio e12, as well.

The mechanical efficiency offers more advantages than the momentary efficiency, and will be calculated approximately, by replacing in relation (29) the pressure angle α1, with the normal pressure angle α0 the relation taking the form (30); where e12 represents the contact ratio of the gearing, and it will be calculated with expression (31) for the external gearing, and with relation (32) for the internal gearing.

$$\eta_m = \frac{1}{1+tg^2\alpha_0+\frac{2\pi^2}{3\cdot z_1^2}\cdot(\varepsilon_{12}-1)\cdot(2\cdot\varepsilon_{12}-1)\pm\frac{2\pi\cdot tg\alpha_0}{z_1}\cdot(\varepsilon_{12}-1)} \qquad (30)$$

$$\varepsilon_{12}^{a.e.} = \frac{\sqrt{z_1^2\cdot\sin^2\alpha_0+4\cdot z_1+4}+\sqrt{z_2^2\cdot\sin^2\alpha_0+4\cdot z_2+4}-(z_1+z_2)\cdot\sin\alpha_0}{2\cdot\pi\cdot\cos\alpha_0} \qquad (31)$$

$$\varepsilon_{12}^{a.i.} = \frac{\sqrt{z_1^2\cdot\sin^2\alpha_0+4\cdot z_1+4}-\sqrt{z_2^2\cdot\sin^2\alpha_0-4\cdot z_2+4}+(z_2-z_1)\cdot\sin\alpha_0}{2\cdot\pi\cdot\cos\alpha_0} \qquad (32)$$

CONCLUZII

The best efficiency can be obtained with the internal gearing when the drive wheel 1 is the ring; the minimum efficiency will be obtained when the drive wheel 1 of the internal gearing has external teeth. For the external gearing, the best efficiency is obtained when the bigger wheel is the drive wheel; *when one decreases the normal angle a_0, the contact ratio increases and the efficiency increases as well.*

BIBLIOGRAFIE: Petrescu, R.V., Petrescu, F.I., Popescu, N., *Determining Gear Efficiency*. Gear Solutions magazine, 19-28, March (2007)

Laboratory for Internal Combustion Engines
Road Vehicles Department
University of Craiova, Faculty of Mechanics

Lecturer
Ilie DUMITRU
Facultatea de Mecanică din Craiova
Director Departament Autovehicule Rutiere

In order to adapt the educational process to modern society's requests, all the high educational entities are forced to correlate the theoretical knowledge to day-to-day practice, therefore enabling future graduates to be qualified according to the economical and social demands.

The internal combustion engines laboratory form the Road Vehicle Department of the Faculty of Mechanics from Craiova has benefited from an increased attention from the faculty members so that it has received modern equipments with funds obtain due to research activities but also from an excellent cooperation with various vehicle manufacturers (and especially FORD).

Nowadays, the activity here concerns:
• structural and functional analysis of thermal engines
• the study of thermal processes that occurs inside the engines
• determination of various functional and structural parameters
• the analysis of engines based on their performances

In the case of the determination and analysis of functional parameters, it can be done very accurately using a chassis dynamometer.

This equipment allows research studies concerning:
• thermal efficiency determination
• various dependencies: torque – engine speed, power – engine speed, specific consumption – engine speed;
• hour fuel consumption, specific fuel consumption;
• thermal working range
• various studies based on imposed conditions

Fig. 1. A chassis dynamometer used for determination of maximum force, speed and torque

It also allows:
• graphical determination of power and torque
• visual reading of speed, engine speed, oil temperature
• power engine determination according to EEC 80/1269, ISO 1585, DIN 70020;
• live data
• possible connection to gas analysis equipments
• possible connection to equipments for fuel consumption determination
• OBD module

There are also three test stands made by the former "Uzina de Automobile din Craiova", that are also used for various engine determination, but only in the case of spark engines. Beside these, there are also present several mono and multi cylinder engines, both spark and compression types, stands which were donated by Ford and Dacia, stand for testing injection pumps, for testing injectors etc.

There is also the possibility for the students to use modern equipments in order to perform various analyses of the engines (Gutmann tester, Bosch KTS tester and Bosch EsiTronic database etc)

Fig. 2. Spark engine test stand

Fig. 3. Equipments for research and educational activities

University Research

Theoretical and experimental research regarding high speed rotor dynamics with turbochargers application; ways to reduce vibrations and noise

Author: PhD student Eng. BORICEAN Cosmin C-tin

This research project is sustained by the Sectorial Operational Human Resources Development 2007-2013. The main objective of this research is to try to implement in the development of turbochargers new way to sustain turbochargers shafts with additional damping, which reach considerable rotational speeds reaching 200.000 rev/min. For the first stage of the project there were studied dynamical phenomena of turbochargers rotors sustained on rolling bearings with ceramic rolling elements and also the methods to reduce vibrations of these turbochargers rotors using additional damping devices. Also there were made comparative tests using specialized software solutions, between the classical turbochargers rotors and the new rotors which were adapted for rolling bearings. Using FEM methods there were accomplished series of tests which include thermal behavior, modal analysis and also spectral analysis, of the two turbocharger rotors. The tests included also laboratory tests accomplished on specialized platforms in order to identify the natural frequencies of the two turbochargers rotors, mentioned above. In the following it will be made mathematical models which will offer an image of the dynamic behavior, over the classical turbocharger rotors and also of the turbocharger with rolling bearings, concluding with a rigorous analysis of the two turbocharger solutions and also testing the solutions. It will be presented a mathematical model which will be capable to analyze the dynamic behavior by modifying functioning and construction parameters and also it will be presented some methods to reduce and to control vibrations that could appear in the functioning of high speed turbochargers.

Contact person: BORICEAN Cosmin C-tin; University Transilvania Brasov, cosmin.boricean@unitbv.ro

Research on development of a method for improvement of cold starting performance of biodiesel fuelled engines for special vehicles. Project: ID_705, no. 696/2009-2011

The world energy demand is increasing rapidly due to excessive use of the fuels but because of limited reservoirs; therefore, the researchers are looking f or alternative fuels. Biodiesel can be one of the best alternatives. It is made from the oils of various types of oilseed crops like sunflower, palm, cottonseed, rapeseed, soybean and peanut. Cold start diesel engine is one of its sensitive points. Due to physical and chemical properties of biodiesel (T10, viscosity, Cloud point), cold starting of the engine fuelled with pure biodiesel or diesel-biodiesel blends is different and more difficult.

The project aims to develop a method to improve the quality of cold start of engines fuelled with biodiesel for vehicles with special purposes: transportation in areas with environmental restrictions (historic urban centers, natural reserves, etc.), underground (e.g. hydropower, mining), indoors (warehouses), transport-tation of dangerous substances (flammable), etc. The proposed method consists of injecting diethyl ether in diesel engine intake.

Preliminary results:

- Numerical simulation of injected combustible substance processes suffered in the intake manifold (fluent cfd)
- Experimental research to determine performance on cold start (-20° C) of a diesel engine fueled with different proportions of biodiesel and diesel; they were conducted with the support of specialists from Renault Technologie Roumanie; the team provided the know-how and the equipment.

Contact: Rodica NICULESCU, Adrian CLENCI, University of Pitesti, Automotive Department, Tel: +40.348 453 166, Fax: +40.248 216 448, email: rodica.niculescu@upit.ro, adi.clenci@upit.ro

Traffic impact studies on road modernization

Coordinator Assoc. Ph.D., Ilie DUMITRU Department of Automotive Engineering of the Faculty of Mechanical Engineering, University of Craiova.

The research team is composed of professors from the Faculties of Mechanical Engineering and Automation and specialists in road traffic of the Urcosys Company in Craiova. The studies are aimed at quantifying the interdependencies specific to cases of trafficking in modern roads. The application of these studies was issued by regional authorities and covers a large number of modernized roads. The research team, based on experience and its own logistics has developed evaluation algorithms using dedicated software packages have evaluated through systems output types the following groups of indicators: Traffic safety, minimized duration of transport, speed, transport Regularity; riding comfort, qualitative and quantitative integrity of goods, reducing costs, reducing environmental pollution. Achieving these studies will be carried out in 2011 for various modernized roads, once again stresses the complexity of transport and traffic engineering field.

Contact person Prof. Eng Ing. DUMITRU
Email idumitru@mecanica.ucv.ro

Creativity and enthusiasm for student projects

Conf. dr. ing.
Liviu MIHON
Timisoara branch
responsible for SIAR

Every year future graduates have passed a final exam, more demanding, more serious and certainly as most "convincing" for the examination: the diploma exam. No other complex laboratory work, no project of the year, in any discipline, involves so much creative energy, ideas and solutions, looking for a practical realization in a position to prove the entire stock of knowledge accumulated during faculty. Ideas and solutions come from solid documentation, intercolegiale discussions and / or in consultation with teachers and are applied with great confidence in the project. The goal is one: the desire of the final outcome assessment, most often resulted in a prototype, with a practical application, with an experimental stand. At the "Politechnica" University of Timisoara there are numerous examples of projects, with practical implementation, resulting in the construction of vehicles, from original ideas or from existing cars and partially or completely rebuilt. Hours of study, conception and design, fabrication, assembly and testing go unnoticed and are not quantified and not settled on any payroll, but does not compare with any other material rewards that could come into question if such "services". The time devoted to preparation and completion of degree, all these young enthusiasts "forget" for other concerns, form work teams, continuously extended working hours and shorter breaks to see the amazing finished piece, component, subassembly or "end" to comparable to year colleges or group who, in turn, is working on another project or another achievement worthy of consideration. One such car was executed by a team of four enthusiastic students starting from our classic Dacia 1300, from which have retained only the engine and transmission, and other assemblies, such as suspension, steering and braking system have been adapted and modified accordingly. During the six months required for finish this project have made the "jalopy" purchased at a minor cost resulting in a splendid terrain vehicle. Young people have worked together in a student's campus area from Timisoara. All have made a great effort and have contributed ideas and soul to this project, as well as one student said. They changed the main geometry and wheelbase and track, with the whole arrangement of the engine drive and control system, fully adapted to the transmission and rear traction with a completely redesigned and changed steering system. Removing body structure of the original car gave the biggest headaches for the design and implementation of a completely original frame structure, the new solution to powertrain placement and organization of the cockpit were those requiring more detailed studies and more complex analysis. The braking system has been redesigned and more easily adapted to a vehicle, being improved in terms of efficiency and response time. The seater driver protection was another important element in mind, but whose realization is satisfied everyone who saw this vehicle. And finally, because the project needed to have a name, they found an inspired and funny title, i.e. "Mergedes," to secure a long term working time and without compensation. No less commendable is the initiative of another group of students who wanted to create a small vehicle, and easily adapted to an urban traffic system. Since the vehicle was designed for three people places (two in front and one rear, side) and "some" space for luggage, placement of the equipment create a serious problem in the power train solution "all in front" and minimal comfort respectively. The manufacturing of the vehicle started from a multilayer polymeric resins and fiber glass monocoque structure, which was donated by a local business company. The engine was chosen, in the first instance, on purely economic considerations, being bought a two-stroke engine with a displacement of less than 600 cm3, but clearly enough for such a structure and poor performance for which it was designed. Rolling elements, contact with soil, suspension, steering and braking are required to adapt ideas and original solutions. Small size, nice and easy handling operation, more popular and loved color of the University (purple) made this vehicle in a short time to become "the mascot" of the Faculty of Mechanical Engineering and its students. What that all these examples want to prove? The fact, extermely joyful, that the spirit and desire to work among young people has not ceased to exist, and especially the desire to own something with an effort to characterize the most personal way during the student period. There is however a necessary minimum elements that must be provided: a minimum of space required and equipped for such activities, a good collaboration with teachers and qualified staff that are always willing to give a helping hand to carry out the projects. With great joy and confidence we welcome the initiative of the outbreak in an institutional framework, to prepare a student engineering competition, which will motivate and bring to the fore the true creativity of those who have embraced with such confidence, and not only, the field of Automotive Engineering. We hope to mobilize adequate and support the concrete and substantial as the major production companies of the main suppliers of automotive products and services. Clearly, as I said quite often, personal and in various other occasions, Polytechnic University of Timisoara through the Thermodynamics, Thermal Machines and Road Vehicles Department wants and supports any initiative in this direction.

REGISTRUL AUTO ROMÂN

RESERVATIONS

021/9672

INFO

021/318.17.30

ADRESS

Calea Griviţei 391A
sector 1, Bucureşti

www.rarom.ro