PRESSURE WAVE TECHNOLOGY – AN INTERESTING APPROACH IN SUPERCHARGING

IULIANA COSTIUC[1], ANGHEL CHIRU[1], LIVIU COSTIUC[2]

Abstract. The current concerns of mankind on reducing the disastrous effects of global climate warming have led to measures that involve improving the performance of internal combustion engines (ICE). Supercharging is one of the methods of increasing efficiency and thus optimizing ICE's energy management. The pressure wave superchargers (PWS) proved to be an economic solution with a simple geometry and fast operating response. The researchers' efforts to develop the pressure wave devices, encountered a rather slow progress during more than a century since the first pressure exchanger was proposed, mainly caused by the meticulous calculation and simulation of the complex phenomena occurring inside the rotor channels. Currently, progress in computerized programs has facilitated modeling and understanding of these processes. Continuous incentives for achieving energy efficiency together with replacing the older technology with new and innovative one have stimulated lately new interest in wave rotor technology. This paper is reviewing the main features of the PWS, acknowledging the past laborious work of several researchers, which led to the actual performances of the pressure wave technology applications, especially on the vehicle and aircraft propulsion systems. Also, it is presented a system of equations for non-steady, compressible and viscous flow of fluids working within a PWS.

Key words: pressure wave supercharger, turbocharging, un-steady flow, wave rotors, cell wheel, shock waves.

1. INTRODUCTION

Optimizing the energetic and ecological management of internal combustion engines (ICE), together with the improvement of the engine's design, dimensions, performances, and, moreover, emissions are, nowadays, important topics for researchers and engineers, driven by society's dependence on thermal engines for transport, commerce and energy generation. Indeed, without the ICE, the world, as it is today, would be extremely different!

ICE has a more than three centuries history, evolving from its strange rudimentary "relative" – the internal combustion piston engine using the gunpowder as fuel, suggested by Jean de Hautefeuille in 1678 – to the

[1] Transilvania University of Braşov, Department of Automotive and Transport Engineering
[2] Transilvania University of Braşov, Romania, Department of Mechanical Engineering

contemporary computer controlled and highly technologized engines. The first engines, as we know them today, developed in the middle of the 19th century and were initially used for transport; later they made possible the development of the marine and aviation industry or other forms of transport, and also helped in the generation of electricity, proving that the ICE is either indispensable in our activities and lives [23].

One of the world's major concerns is the rapid climate change due to the increasing levels of *greenhouse gas emissions*, the main cause of *global warming* and its long-term *dangerous consequences*: melting of polar ice and rising seas levels, extreme weather, negative impact on agriculture, forestry, energy and tourism, risks for wildlife, risks for human health, heavy costs for the society and economy. The European Union established as one of its *key targets* for 2020 a 20% reduction in greenhouse gas emissions compared with 1990, and a 40% reduction for 2030 [1]. Consequently, the EU has taken legislative action to restrict higher levels of pollution, especially arising from burning the fossil fuels.

The primary *consumer of fossil fuels* are the propulsion systems, being stated [2] that the road transportation sector is one of the main sources of greenhouse gas emissions, e.g. CO_2 *and air pollutants*. Therefore, the ICEs have become the point of interest in terms of reducing emissions, as well as efficient use of energy. The most common measures are: the introduction of alternative propulsion technologies by developing the hybrid or electric cars; eco-innovation; improving the efficiency of conventional engines by implementing advanced technologies, and implementing specific technologies for the exhaust system by using oxidation catalysts, catalysts substances injected into the exhaust gas stream, cellular filters, traps and absorbers, etc.

A well-conceived thermal management of the ICE can contribute in achieving the targets outlined above by improving the overall *efficiency* and performance, by *energy conservation* and *waste heat recovery*, resulting the *lowering of fuel consumption* as well as *gas emissions*.

Thereby, it follows a decreased dependence on fossil fuels, and therefore a conservation of natural resources and a reduction in the impact of propulsion systems on the biodiversity.

A commonly used method of increasing the power output and the engine's overall efficiency is *supercharging*, i.e. producing considerable boost for the inlet fresh mixture or combustion air. This is achieved by using mechanical chargers or turbochargers, interleaved positioned within the intake manifold system.

The idea of supercharging appeared at the end of the 19th century. In 1885 *Gottlieb Daimler* patented a technique of forcing air into an internal combustion engine. In 1896, *Rudolf Diesel* tested the pre-compression of combustion air on the efficiency of the engine. Also, around 1900, almost simultaneously, Sir *Dugald Clark*, *Sulzer* and *Renault* discovered that more power is produced by increasing the volume of air charge entering an engine. Between 1912 and 1915, Dr. *A. Büchi* proposed the first turbocharged diesel engine by developing the first exhaust gas-driven charger [3] – an idea considered only in 1925, when Dr. Büchi achieved 40% increase of the power output and demonstrated the advantages of using the energy contained in the exhaust gases. This marked the beginning of using ICE supercharging by the automotive industry [23], initiated in 1938 by the company *Swiss Machine Works Saurer* that first applied turbocharging on truck engines. The need for transportation during the World War II boosted the interest in the charging technology, especially on war planes engines [3].

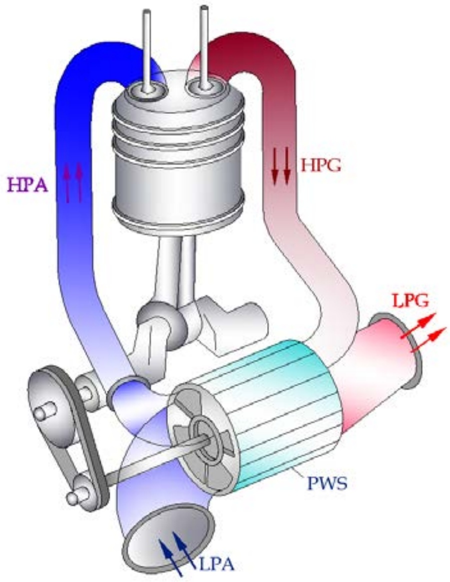

Fig. 1 – BBC Brown Bovery & Co. pressure wave supercharger Comprex® (HPG – high pressure exhaust gas, HPA – high pressure intake air, LPA – fresh air intake, LPG – low pressure exhaust gas)

Turbochargers are the most commonly used solutions for car manufacturers as they have many advantages, e.g. higher thermal efficiency, but studies indicate that turbocharging has also a series of shortcomings that need to be overcome. A particular method of turbocharging is by using the pressure wave device – known as “wave rotor” or “pressure wave supercharger” (PWS) – Fig. 1 – that also transfer the energy of the exhaust gases in order to induce forced air into the admission manifold, but rely on the action of pressure waves inside the rotor narrow channels. The improvement on the engine performance for the whole range of the engine speeds proves that the PWS is a good option for applications on the ICE’s for all types of vehicles.

2. FUNDAMENTALS OF PWS. GENERAL OPERATING PRINCIPLES. DESIGN AND CONSTRUCTION

Pressure wave superchargers exploit the pressure waves to transfer energy from the combustion gases to the air intake. The phenomenon occurs inside the longitudinal narrow channels (cells) of the device. The PWS construction consists of a rotor, also called “cell wheel” in which are machined longitudinal passages (channels), positioned radially on one or two rows (Fig. 1). Inside the channels, PWS transfer energy directly from the exhaust gas towards the fresh air intake by means of shockwaves, through short direct contact between fluids, without using any additional mechanical elements; the interaction between the high pressure hot gases and the low pressure cold air induce boost. In principle, the exhaust gases produce shockwaves that expands within the channels and compress the fresh intake air. The process of equalizing pressures is extremely fast through the channels; therefore the phenomenon of mixing between the two fluids is insignificant.

(a) (b) (c)

Fig. 2 – (a) warm side; (b) rotor, also named cell wheel; (c) cold side.

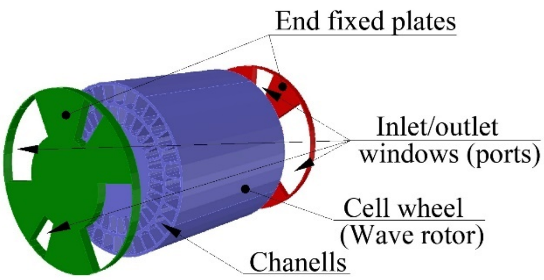

Fig. 3 – PWS construction elements.

The cell wheel rotates inside a cylindrical steel casing, in between two fixed plates, placed at both ends of the rotor (Fig. 3). The plates are provided with passages that allow the air and the gases to flow towards or from the rotor channels. The air passages, connected to the intake manifold is called the “cold stator” or cold side (Fig. 2c) and the exhaust gas passages flow through the passages called the “warm stator” (Fig. 2a). The cell wheel is driven by a separate electrical motor or by a belt driven by the crankshaft.

The longitudinal channels shaped into the cell wheel are opened at the end, forming radial holes arranged in angular staggered rows. It is considered that this configuration substantially reduces – by interference – the whistling which was observed to the original model with only a single row of cells. The rotor is cantilever mounted, supported in its own shaft towards ball bearings; an extension of the shaft after the bearings supports the pulley-belt drive mechanism that rotates the cell wheel with a multiple of the engine speed (3 or 4 times or other) [4].

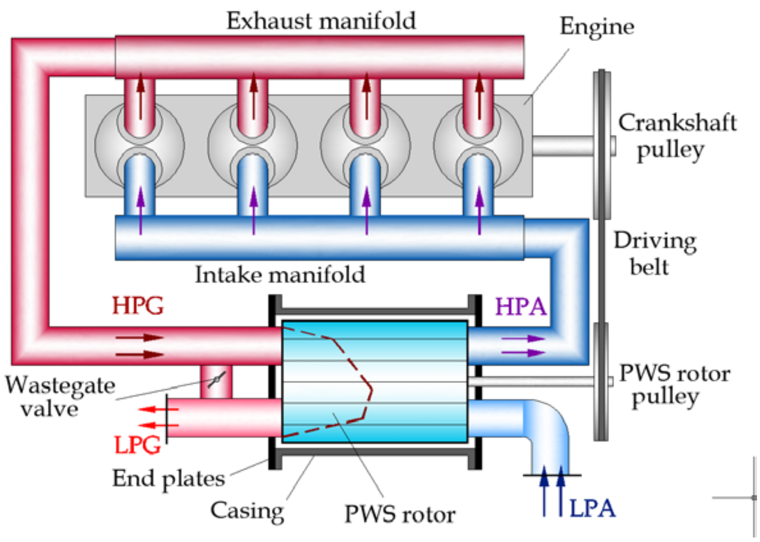

Fig. 4 – Four port PWS system components.

The working inlet fluids for a PWS are: the high pressure (HPG) and the low pressure and low temperature fresh air (LPA) to be delivered into the admission manifold. After the acoustic and thermodynamic processes action through the channels, the outlet fluids leaving the PW supercharger are: the expanded low pressure gases (LPG) evacuated by the exhaust system and the compressed high pressure air (HPA) going towards the cylinders (Fig. 4). The inlet and outlet fluids circulate through well designed “windows”, usually called ports, shaped in the rotor’s end plates. The designed dimensions and form of ports, their number and position are conceived different for each application.

The common configuration with four ports consists on: two ports for the exhaust gases – one port for HPG and one for LPG, the second designed much larger in order to extend the discharge time of the exhaust gases towards the silencer system – as well as two ports for the air: one wide port for LPA and a narrower one for HPA. These passages are connected to the inlet and outlet manifolds, respectively.

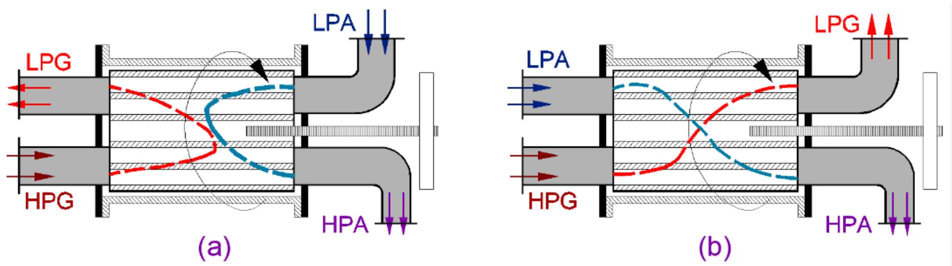

Fig. 5 – Four port PWS configurations: a) The passing – through flow configuration; b) The reverse flow configuration.

PWS can be designed in two configurations: passing-through flow (TF), as shown in Fig. 5a – when the working fluids flow in the same direction and reverse flow (RF) – Fig.5b – when each fluid (gas or air) enters and exits on the same side [11]. The through-flow rotor has both inlet ports (fresh air and exhaust gases) placed on the same end side of rotor, while the outlet ports (compressed air and expanded exhaust gases) are on the other side. The reverse-flow configuration provides the air inlet and outlet ports on one side, named the "air casing", and the gas inlet and outlet ports placed on the "gas casing", allowing for both fluids to flow in and out on the same side. Even though these two configurations ensure similar performance improvement, they are substantially different when investigating their inner processes. In the four ports TF rotor, both fluids – the cold air and the hot gases – travel the rotor longitudinally, maintaining an average temperature for the rotor body, relatively constant over the entire length of the rotor. This allows a self-cooling effect to the rotor, making the TF configuration suitable for gas turbines applications, where gas temperatures are elevated. The RF configuration, where the air casing remains rather cold while the gases casing gets hot, was used mostly in supercharging applications [5].

Essentially, as the cell wheel rotates, the channel openings are exposed alternatively to the inlet or outlet passages, allowing the fluids flow through the ports. The compression and expansion waves are thus initiated inside the rotor channels; the high pressure gases develop pressure waves that evolve inside the channels and compress the intake charge.

Inside the cell wheel, the real energy exchange takes place at the speed of sound, which depends on the exhaust gas temperature, which is not dependent on the engine speed. Therefore, the pressure wave effect is optimal for a single operating designed point. If the rotor is crankshaft belt driven, there is a constant transmission ratio between ICE and PWS. Thus, the wave strength is diminished when the operation conditions are on an off-design point. In order to overcome this disadvantage, there have been designed "pockets" within the rotor housing that allow the PWS to respond rapidly to the changes in engine operating requirements. The pockets ensure good boost pressure curves as well as high efficiency for a wide range of operating conditions [6].

In designing the wave rotor, many challenges had to be assumed and solved: number of channels, shape, diameter, length, number and shape of cell rows, material of rotor and other elements, leakage effects, noise, weight. For instance, the cells usually axial, straight shaped, can be also curved, similar to the turbine blades. More, the number of channels affects the rotor operation - as it increases, the entry flow losses increase. Also, because of the high centrifugal forces, high frequencies shockwaves and cyclic temperature gradients, the rotor material has to be chosen to handle all these stresses. More, by using ceramic materials, the rotor became lighter, suitable for "free-running" (no belt driving necessary), reducing thus the additional load at the crankshaft. In terms of noise, as the frequency lies in the audible zone for the rotational speed range of a common wave rotor, the well-known "whistle" has to be eliminated [7]. One method to reduce the noise was to break the symmetry of the rotor cells or by using multiple rows of channels.

Another important challenge in designing the wave rotor is the leakage that seriously affects the performance. First, the rotor and the casing have to be built of materials with the same expansion characteristics. To minimize the leakages, the gap between the rotor and the end plates has to be minimized, yet preventing the occurrence of contact, regardless of the thermal regime.

Nevertheless, the ability to produce the wave rotor device with low costs, at large scale, has solicited the designers to find solutions for refining the manufacturing process.

3. SHORT HISTORY OF PRESSURE WAVE TECHNOLOGY

The pressure wave technology relies on the energy transfer between two fluids being in direct contact, based on the shockwaves effect transmitted from one fluid to the other. The phenomena that occurs is rather complex, usually involving thermodynamic, acoustic and, sometimes, chemical interactions. These processes are manifesting on Earth since forever, but the scientists learned about them or studied and understood them only few centuries ago. Some phenomena, such as the turbulent movement of particles, the steady or non-steady fluids flow, as well as the oscillatory processes needed laborious and detailed calculations that have hindered progress to be made in this direction for many years. Lately, software development has helped ease the scientists' work, solving thus the complex equations that describe these processes.

During the last decades, a sustained impetus for improving energy efficiency, for the old and new technology by finding innovative technical solutions, has stimulated the interest in using the pressure wave technology. Hence, applications of this technology have been realized such as: shock wave engines with internal combustion, rotary thermal separators, like wave refrigerators, shock tunnels, pulse detonation engines, pressure wave superchargers, etc. [23]. The oldest "relative" was a cell drum used as a pressure exchanger, patented by *Knauff* in 1906 [8]. This device was presented as a rotor with curved blades and with inclined stator nozzles that provided shaft power using a steady-flow energy transfer by mixing compressible or incompressible fluids [9, 23]. The device resembling with the actual pressure wave rotor was patented in 1913 by *Burghard* but was never developed – a rotating drum with axial channels that provided continuously pressurized air [7]. In 1928 another similar device, with a geometry implying long narrow channels, was proposed by *Lebre* [10] to be used for a refrigerating unit. Around the same year, Burghard patented a simpler device [18 in 9] named "dynamic pressure exchanger" to distinguish it from Knauff's previous "static pressure exchangers".

In the 1930s there were presented new ideas for devices using the pressure waves: *M. Kadenacy* (England) tuned the exhaust pipes of an engine by means of the pressure wave effect (the so-called "Kadenacy effect"), *J. Wydler* (USA) used the exhaust gases energy in a rotating device for supercharging a reciprocating engine, and Prof. *A. R. Kantrowitz* (Cornell University) designed a device working with expansion waves [13].

In 1940, during the World War II, *Claude Seippel* from the *Brown Boveri &Co.*, revealed the pressure waves' potential to transfer energy between two gases in direct contact – the expansion of a gas to compress the other. Seippel implemented in Switzerland the first wave rotor device as a high-pressure stage for a locomotive gas-turbine. Seippel named the machine "COMPREX" going from the processes of COMPression and EXpansion that occur inside the rotor channels [23]. This first wave rotor, with 30 longitudinal channels, operated at 6000rpm, at a pressure ratio of 3:1 and reached a global efficiency of 69% [5]. As Seippel described in his patents [14-17], there it was expected an increase for power of 80% and a 25% in the overall performance. Even though the device worked satisfactory, when tested on an engine, it revealed some shortcomings that cancelled any further development [9].

These inventions have outlined the idea of using the pressure wave rotor for charging the engines. Starting with 1949, the first attempt in implementing this new concept was made in Philadelphia, USA, by *ITE Circuit Breaker Co.* and Prof. *Arthur R. Kantrowitz* with the support of the Bureau of Aeronautics. During 1951-1954 the tests came with satisfactory results that proved that PWS can be used for supercharging the diesel engines. In 1955, ITE Circuit Breaker Co. and *Caterpillar Tractor Co.* started a cooperative program, materialized in a small prototype of a pressure wave supercharger. ITE when started the tests on vehicle in 1957 [13], naming the device "COMPREX". Even though this first version of the PWS did not reach sufficient pressure into the admission manifold at very low engine speed [9], the tests proved that PWS is a simple geometric device, it can boost air density over a wide range of speeds and operates under rapid load changes, with no lag or smoke [13,23].

Also in 1955 ITE started another cooperative program with BBC Brown Boveri & Co., continued later only by BBC. As producer of turbochargers, BBC developed PWSs for diesel engines in partnership with the *Swiss Federal Institute of Technology ETH,* Zurich. They succeeded to overcome the deficiencies of the previous version of PWS by making cycle modifications [9].

The British company *Power Jets Ltd*., in the mid 50's, started to investigate multiple wave rotor applications, like ICE supercharging, gas turbines, pressure equalizers, air cycle refrigeration devices. For instance, their engineer, *Jendrassik* investigated the use of the rotor wave as a high pressure topping stage for aircraft engines applications [5]. *D. B. Spalding* from the Imperial College London continued the work of Jendrassik, using his theoretical and experimental results to elaborate the first numerical model for a wave rotor, taking into account the effects of the heat and friction losses [23,26].

In the mid-60s', *Rolls-Royce company* (UK) together with *BBC* and with *Berchtold* (ETH Zurich) and *Spalding* (Imperial College London) as consultants developed a wave rotor for a small helicopter engine [27 in 5]. Their challenges were about overcoming the difficulties that affected the device efficiency like quickly operating changes, start-up, durability of the bearings, leakage and fuel control. Due to financial issues, the program was canceled in 1972 [28].

Only in 1971, the company *Valmet Tractors* from Finland installed the first Comprex® (patented by Brown Bovery & Co.) prototype on a truck engine, and around the same time *Mercedes-Benz* started the tests for using the Comprex for their passenger cars. Starting with 1974, few car manufacturers (*Opel, Ford, Mercedes-Benz, Volvo, Peugeot* and *Ferrari*) started to investigate the use of Comprex for supercharging their diesel cars. The first limited edition release was the 2.3 liter Opel Senator model charged with the Comprex pressure wave supercharger.

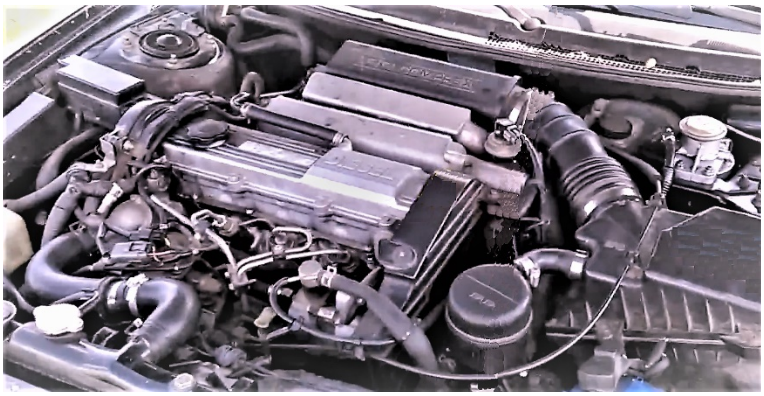

Fig. 6 – Under-hood view of Mazda 626 2.0L Diesel (1991).

In 1987, the new company *ABB* (*Asea Brown Bovery* resulted from the merger of BBC with ASEA) sold the Comprex activity to *Mazda*, that extended the Comprex® application on vehicles, releasing the new model Mazda 626 Capella [29] (Fig. 6) equipped with a 2.0 liter Comprex supercharged engine. More than 150,000 units were sold, this model being considered as the most successful commercial application of the pressure wave technology [23]. The pressure wave supercharger series production was cancelled in 1994, after the company Ford took-over Mazda.

As stated in the sections above, during wave rotor's rather short history modelling and simulating the phenomena occurring within the rotor cells were probably the most difficult steps in investigating the main flow processes and the principal losses, which are usually 2-D or 3-D in nature. Numerical simulation and experimental validations on wave machines were reported starting with the end of 70s' by *Thayer* from *Mathematical Science Northwest Inc*. His efforts were directed on calculations of energy exchanger performance of an experimental wave rotor using *FLOW* software developed for modelling the 1-D unsteady flows. It took into account the wall friction and heat transfer losses, as well as port and channel geometry, the clearance between the rotor and the fixed end walls, viscosity of the working fluids, influence of speed and leakage [5], variations of the air outlet port area and the flow rates through each port. The results showed that leakage and the area of the exhaust gas outlet are key elements for efficient operation of a pressure wave rotor. They also helped to understand the functioning of real energy exchangers and of pressure wave devices [33].

In 1985, *Atul Mathur* from the *Turbopropulsion Laboratory* of the Naval Postgraduate School, CA [31] reported his efforts in modelling the basic flow processes inside pressure exchangers or wave engines by developing an unsteady one-dimensional flow code, using the Random Choice Method to solve the governing Euler equations, based on the Riemann problems solution. His code was useful to ease the calculations of the unsteady process inside the wave rotor and of some initial design issues, but it didn't describe the real flow process [32]. However, based on these results, another software called *ENGINE*, was developed for calculations of jet engine's performance [5].

During the 90's other researchers directed their efforts in describing numerical methods for calculations of processes inside pressure wave machines: *S.Eidelman* (a bi-dimensional code that showed the importance of gradual opening of the passages, of their number and geometry, as well as of the losses occurring due to mixing and wave reflections [35]), *Eldin et al.* of University Wuppertal, Germany, *Piechna et al.* of Warsaw University of Technology, *Oguri et al.* of Sophia University Japan, *Guzzella et al.* at ETH Switzerland (a numerical model for PW devices used for engine supercharging) [5], *Paxson* at *NASA Glenn Research Center* (a quasi-one-dimensional wave rotor simulation code, time-accurate, experimentally validated, capable to describe the gasodynamic processes, to calculate its geometry and to study of their influence upon the on-design and off-design performance [36, 37 in 21] – the code was recognized as a principal tool for wave rotors analysis), etc.

Around 1994, after Ford took-over Mazda, the Comprex rights were sold to Caterpillar that, together with the new company *Comprex AG* (founded by former Brown Boveri Co's employees) continued to produce PWSs for diesel engines [30]. In 1998 Comprex AG was taken-over by the *Swissauto WENKO AG* [23].

The early pressure wave supercharger developed by BBC, the Comprex®, had to be improved before being produced in series. The technical solution challenges that had to be overcome were: noise, unequal heating of the rotor, leakage [34 in 5], as well as configuration new elements like pockets designed to control the reflected waves. The new PWS was a reliable product ready to be used for engines' supercharging. There were manufactured 8 models of Comprex, marked from CX-65 to CX-125 (the digits representing the gross diameter of the rotor) initially designed for supercharging the 0.5…3.5 litter compression ignition engines. Table 1 presents the main features and geometrical dimensions of COMPREX models (except CX-65 used exclusively for small displacement engines, of 0.5-0.8 liter cylinder, and CX-125 used for charging 3-3.5 liter tractor engines), [24] according to the dimensions represented in Fig. 7.

Table 1

COMPREX geometrical dimensions and the effective power of the supercharged engine (see Fig. 7)

COMPREX model	D_0 [mm]	D_{rotor} [mm]	D_{LPA} [mm]	D_{HPA} [mm]	D_{HPG} [mm]	D_{LPG} [mm]	L_{PWS} [mm]	Effective power of ICE [kW]
CX – 71	113	71	49	36	39	50	274	25…40
CX – 78	118	78	55	36	43	56	303	30…48
CX – 85	126	85	62	36	47	61	310	35…57
CX – 93	129	93	64	46	52	67	335	40…70
CX -102	143	102	72	46	57	73	364	50…83
CX - 112	150	112	80	46	62	80	396	60…100

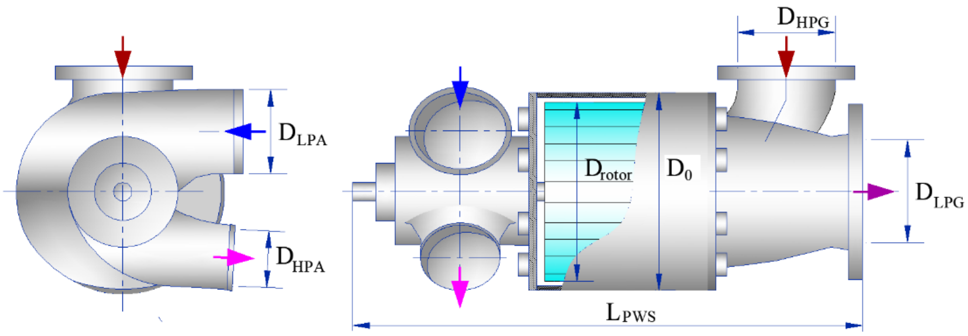

Fig. 7 – Geometrical dimensions of Comprex models.

In the early 90's, *Swissauto WENKO AG* started cooperation with two companies *BRM Design* and *Esoro*, to develop an interesting project for Greenpeace, called *SmILE* (Small, Intelligent, Light and Efficient). SmiLE had as main target: lowering to half the fuel consumption, maintaining the same characteristics of a production car (Renault Twingo): performance, capacity of transport, safety and comfort. SmiLE was the first homologated car that attained a fuel consumption of 2.3-2.5 Liters / 100km in real traffic conditions (3.4 l/100 km for NEDC), validated by the German TüV [42]. A new concept engine [43] called the SAVE engine (Fig.8) of 360cc, PW supercharged, registered emission values under the Euro III standards, with an overall car weight lowered by 150 kg.

Accordingly to the SAVE concept, SmiLE's engine was optimized by providing: a throttle valve to control the engine's load, gas pockets for the PW supercharger to control the operation at high loads, a controlling valve for vary the scavenge air mass flow ejected by the PWS towards the exhaust manifold and a waste-gate valve to adjust the PWS pressure ratio. The SAVE concept novelty consisted mainly in using a PWS to boost pressure, while SAVE was a gasoline engine and PWS was used so far only for diesel engines.

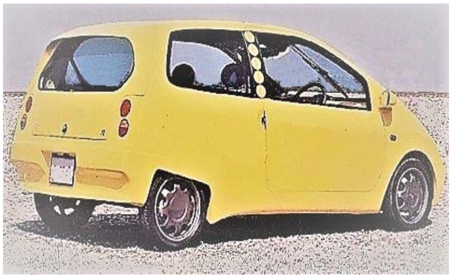

Fig. 8 – Renault Twingo SmiLE car.

The idea of using a PWS came after six years of attempts to use a conventional turbocharger or a mechanical compressor that proved to be unsatisfactory because of noise, comfort or unusual behavior on small displacement (less than 1 liter) engines. Swissauto WENKO AG improved the pressure wave supercharger for the SmiLE project, adjusting the pressure ratio, improving the noise level and increasing the compression efficiency for a wide range of speeds [52].

The Greenpeace Twingo SmiLE car (Fig.8) proved to be the first significant success in applying the technologies of downsizing and PW supercharging, Between 1998 and 2008, Swissauto worked on optimizing the initial concept, releasing in 2008 the VW Golf 5 demonstrator car, with 1.0 liter engine supercharged with the new type of PWS, called *Hyprex*®. Excepting the engine and its new charging system, the demonstrator car kept all the elements of the original Golf 5. Hyprex® brought a notable improvement in the performance, such as 210 Nm torque at 1400 rpm and an output power of 110 kW at 5000 rpm. Also, the acceleration and march-through were refined and the fuel consumption was lowered in comparison with the original Golf 5 [42]. Unlike Comprex®, the new released Hyprex® was driven by an additional electrical motor controlled by the engine's ECU (the Electronic Control Unit). This new pressure wave supercharger had no pockets within the end plates, adjusting the boost pressure and the cooling process with scavenging air by a Gas Pocket Valve, also controlled by engine's ECU. Hyprex® was designed to equip small gasoline engines for which supercharging and EGR significantly influence the engine's overall performance and emissions [19].

Swissauto WENKO AG held all rights from the patent applications regarding PW supercharging of spark-ignition engines [42]. Developing the PW charger for gasoline engines drew up some difficulties, such as the accurate correlation of the supercharger to all the engines' operation states. However, Hyprex® continued to be developed for charging small displacement gasoline engines because of its advantages: simple geometry, fast response, high pressure ratios, reduced specific fuel consumption, lower noise and emissions and reliability [42].

In 2010, the new company *Swissauto Engineering SA* renewed the patent rights on Hyprex® for the following 10 years, being at that time the only known company producing PWSs for small spark-ignition engines. Nevertheless, in 2015 the company was assimilated by *Swissauto Technology AG* that, unfortunately, in January 2017, got into liquidation process.

After 2012, no notable interest was registered for commercial implementation of PW technology. From 2014 to 2017 the small scaled wave-rotors were investigated especially for modelling solutions designed to increase the output power for aircraft engines. *Mataczynski et al.* from Air Force Laboratory, *D. Paxson* from NASA and *Hoke et al.* from Innovative Scientific Solutions Co. started in 2014 tests and flow calculations on a 16 in. length wave rotor for charging a two-stroke ICE, that validated in 2015 the capability of PWS to raise up to 50% the intake mass flow, boosting the inlet air pressure by up to 135% [38,39]. The tests over a large range of operation conditions and optimization of the small PWS continued in 2016 [40,41]. These investigations valued the small PWS performances, and the experimental results proved their potential in increasing the power output for small aircraft propulsion engines.

4. CHARACTERISTICS AND PERFORMANCES OF THE PWS' OPERATION

The wave technology, as shown in the sections above, has sparked since 1906 the interest of researchers and manufacturers. In more than a century, thanks to the development and implementation of new technologies, of new reliable high-strength and high-temperature resistant materials and, nevertheless, of the computational performances, workable wave machineries could be realized and improved. Comparing to other engineering and production fields, the wave technology registered relatively slow progress. Though, considerable results were achieved especially on steady-flow turbo-machineries, despite the well-recognized better efficiency of non-steady flow machines using the shock compression, process more efficient than the isentropic one.

Over the years, one of the wave technology usages, the wave rotor concept called pressure wave supercharger has found its important application in ICE supercharging. As described in Section 2, the underlying principle of wave machines is the exchange of thermal energy in favor of pressure, by direct contact between the fluids and with no means of mechanical components.

The pressure wave supercharger is actually a pressure exchanger – the device that uses the shock waves propagating within a time-dependent gas flow, inside a tube, in order to change the pressure values. The time variation is provided by the rotation of the device, which allows the tubes openings to pass sequentially in front of the inlet and outlet ports [25]. The pressure exchanger exceeds the conventional super- or turbo-chargers shortcomings, having the great potential of changing the pressure values in a very short time and therefore, of tolerating transitory pick pressure and temperature values.

Working as a pressure exchanger, the PWS transmits the energy for the engine’s supercharging from the exhaust gases to the fresh intake air. The energy exchange takes place at the speed of sound, by means of waves propagating inside the narrow PWS’ channels. As the cell wheel is rotating, the openings shaped within the fixed end plates allow the successive alignment of the ends of the channels in front of the fluid's inlet and outlet ports. The PWS’s geometry, dimensions, ports’ opening timing and rotational speed need to be correlated for better results in engine power output and overall performance.

The Comprex PWS operating principles, looking from the wave propagation perspective, was thoroughly described by *P.K. Doerfler* of Brown, Bovery & Co. in [44]. In Fig. 9 is represented the movement of working fluids inside a wave rotor, when operating at optimized speed.

The pressure wave speed depends on the speed of sound, that is function of the fluid’s temperature. Therefore, a PWS will work properly for a specific exhaust gas temperature and at a given rotor speed. Therefore, the PWS can operate in a narrow range of speeds and load performances, because the travel time of the pressure wave need to be correlated to the peripheral speed, in concordance with the engine load and speed. The wave rotor operating range is extended by shaping "pockets" into the fixed rotor ends, as shown in Fig. 9. The pockets are used to prevent the reflection of the waves in a closed cell end, a process which would cause inside the channel a significant change of the flow velocity [22]. The pockets, marked EP and GP in the Fig. 9, allow the fluid to flow from one cell to adjacent ones. They modify the fluid flow so that the additional pressure waves that appear will correct the response of the PWS to changes in engine load or speed.

The expansion pocket EP and the gas pocket GP improve the scavenging effect at all speeds [4]. Therefore, the device can operate with acceptable performance at other loads and speeds because the pockets allow the particle paths to change without major losses [45 in 22].

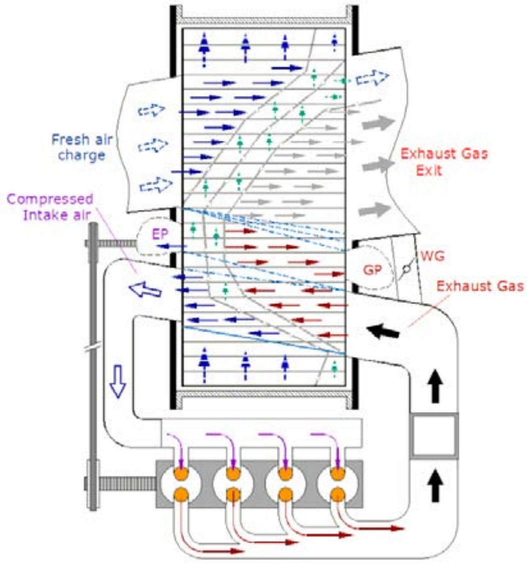

Fig. 9 – Fluid motion within PWS.

Controlling of the pressure ratio can be realized by using a wastegate valve (Fig. 4), working on the principle that the exhaust gases bypass the wave rotor when the charge pressure inside the engine intake manifold has the maximum desired value (Fig. 10). This value is transmitted to the wastegate valve actuator that is gradually opened or closed and thus, a part of the exhaust gases are released into the engine exhaust system instead of entering into the rotor channels.

The pressure of the air to be induced into the engine cylinder is determined by the strength of the compression waves, that is, as described above, function of the engine's exhaust gases temperature. The mass flow entering and leaving the engine are considered equal, the energy contained within the exhaust gases that expand inside the PWS can be sufficient for rising the intake air pressure. The surplus power covers the inefficiencies: losses caused by the leakage into the clearance between the rotor and the stator, losses caused by the incomplete recovery of the kinetic energy of the compressed air inside the intake air manifold and also of the kinetic energy of low-pressure exhaust gases leaving to the engine exhaust system, as well as local pressure losses caused by passing of the fluid over the ports' edges, work dissipation due to flow friction on the channel walls [46]. The compression and expansion processes in a PWS are non-adiabatic, different distinctively from the adiabatic compression and expansion in turbomachines. An overall combined energy efficiency of 74% is declared by Berchtold in [46] taking into account the total of all losses, making the Comprex competitive with turbochargers.

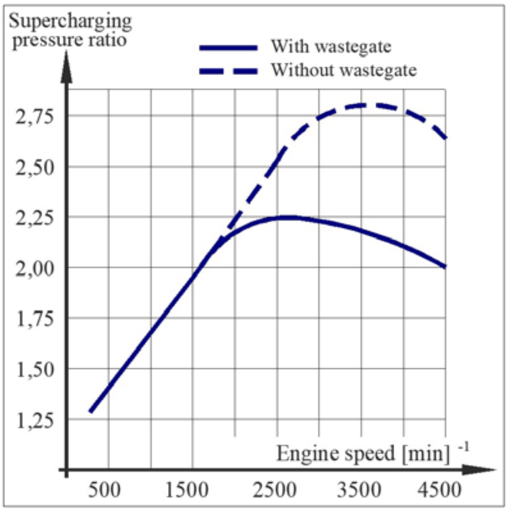

Fig. 10 – The influence of the wastegate valve control on the pressure ratio, function of engine speed variation.

5. PWS MODELING AND CALCULATIONS – COMPLETE FLOW EQUATIONS FOR COMPRESSIBLE, VISCOUS FLUIDS WORKING WITHIN A PWS [54, 55]

As stated above, writing and solving the flow equations for the working fluids inside the PW devices was rather challenging for years. A complete set of equations described by Navier-Stokes form for a Newtonian isotropic fluid with variable properties consists on mass, momentum and energy conservation equations are presented below:

$$\frac{\mathrm{D}\rho}{\mathrm{D}t}+\rho\nabla\cdot\mathbf{V}=0 \tag{1}$$

$$\rho\frac{\mathrm{D}\mathbf{V}}{\mathrm{D}t}=-\nabla p+\rho\mathbf{f}+\nabla\cdot\boldsymbol{\tau} \tag{2}$$

$$\rho\frac{\mathrm{D}}{\mathrm{D}t}\left(e+\frac{1}{2}\mathbf{V}\cdot\mathbf{V}\right)=\rho\dot{q}_{\mathrm{gen}}-\nabla\cdot\dot{\mathbf{q}}_{\mathrm{visc}}-\nabla\cdot(p\mathbf{V})+\rho(\mathbf{f}\cdot\mathbf{V})+\nabla\cdot(\boldsymbol{\tau}\cdot\mathbf{V}) \tag{3}$$

Considering a Cartesian reference system, the mass and energy conservation laws introduce an equation each and the momentum equation introduce three differential equations. The 23 variables involved in the above 5 conservation equations must be described by constitutive equations. These equations are described for:

-viscous stress tensor definition $\boldsymbol{\tau} = \mu\left(\nabla\mathbf{V} + \nabla\mathbf{V}^{\mathrm{T}}\right) + \lambda\left(\nabla \cdot \mathbf{V}\right)\mathbf{I}$, which introduces 6 equations;

-viscous heat equation $\dot{\mathbf{q}}_{\mathrm{visc}} = \dot{\mathbf{q}}_{\mathrm{cond}} = -k\nabla T$ (considering only conduction transfer) which introduces 3 equations;

- heat generation $\dot{q}_{\mathrm{gen}} = 0$ (no heat generation), 1 equation;

- the body force $\mathbf{f}$ ($\mathbf{f} = \mathbf{g}$ if only gravitational field is acting), 3 relations;

- the first viscosity coefficient relation $\mu = \mu(\rho, T)$, 1 equation;

- the second viscosity coefficient relation $\lambda = \lambda(\rho, T)$, 1 equation;

- the thermal conductivity relation $k = k(\rho, T)$, 1 equation;

- the equation of state for real fluid $p = p(\rho, T)$, 1 equation;

- the internal energy relation $e = e(\rho, T)$, 1 equation.

The 23 variables involved in the equation system are: ρ – density, 1 scalar variable [kg/m^3]; $\mathbf{V}$ – velocity vector, 3 variables [m/s]; p – pressure, 1 scalar variable [N/m^2]; e – internal energy, 1 scalar variable [J/kg]; T – temperature, 1 scalar variable [K]; $\boldsymbol{\tau}$ – viscous stress, symmetric tensor, 6 variables [N/m^2]; $\dot{\mathbf{q}}_{\mathrm{cond}}$ – heat flux, vector, 3 variables [W/m^2]; $\dot{q}_{\mathrm{gen}}$ – heat generation, 1 variable [W/m^2]; $\mathbf{f}$ – the body force, vector, 3 variables [m/s^2]; μ – first viscosity coefficient, 1 scalar variable [kg/m·s]; λ – second viscosity coefficient, 1 scalar variable [kg/m·s]; k – thermal conductivity, 1 scalar variable [W/m·K].

Now the system of equations is complete with 23 equations for 23 variables. The material property such as viscosity coefficients, thermal conductivity and equation of state require experimental determination. The model of differential and constitutive equations in order to be physically consistent must respect the second law of thermodynamics.

If the Gibbs equation of entropy is used, $T\,\mathrm{d}s = \mathrm{d}e + p\,\mathrm{d}\left(\frac{1}{\rho}\right)$ than, the second law of thermodynamics becomes:

$$T\frac{\mathrm{D}s}{\mathrm{D}t} = \frac{\mathrm{D}e}{\mathrm{D}t} + p\frac{\mathrm{D}}{\mathrm{D}t}\left(\frac{1}{\rho}\right) = \frac{\mathrm{D}e}{\mathrm{D}t} - \frac{p}{\rho^2}\frac{\mathrm{D}\rho}{\mathrm{D}t} \quad \Rightarrow \quad \rho\frac{\mathrm{D}e}{\mathrm{D}t} = \rho T\frac{\mathrm{D}s}{\mathrm{D}t} + \frac{p}{\rho}\frac{\mathrm{D}\rho}{\mathrm{D}t} \tag{4}$$

Using the energy conservation equation:

$$\rho\frac{\mathrm{D}e}{\mathrm{D}t} = -\nabla \cdot \dot{\mathbf{q}}_{\mathrm{ht}} - p\nabla \cdot \mathbf{V} + (\boldsymbol{\tau} : \nabla\mathbf{V}) \tag{5}$$

and the mass conservation equation:

$$\frac{1}{\rho}\frac{\mathrm{D}\rho}{\mathrm{D}t} = -\nabla \cdot \mathbf{V} \tag{6}$$

in the above equation the resulted expression is:

$$\rho\frac{\mathrm{D}s}{\mathrm{D}t} = -\frac{1}{T}\nabla \cdot \dot{\mathbf{q}}_{\mathrm{ht}} + \frac{1}{T}(\boldsymbol{\tau} : \nabla\mathbf{V}) \tag{7}$$

From this equation, an important conclusion is that the entropy variation is the effect of heat transfer and viscosity work. The second term of the equation can be expressed using the derivation of $\nabla \cdot \left(\frac{\dot{\mathbf{q}}_{\mathrm{ht}}}{T}\right) = \frac{1}{T}\nabla \cdot \dot{\mathbf{q}}_{\mathrm{ht}} - \frac{\dot{\mathbf{q}}_{\mathrm{ht}}}{T^2}\nabla T$ so the equation becomes:

$$\rho\frac{\mathrm{D}s}{\mathrm{D}t} = -\nabla \cdot \left(\frac{\dot{\mathbf{q}}_{\mathrm{ht}}}{T}\right) - \frac{\dot{\mathbf{q}}_{\mathrm{ht}}}{T^2}\nabla T + \frac{1}{T}(\boldsymbol{\tau} : \nabla\mathbf{V}) \tag{8}$$

The second law of thermodynamics expression and the equation obtained above lead to the conclusion that the sum of the last two terms in equation must be always positive, therefore,

$$-\frac{\dot{\mathbf{q}}_{\mathrm{ht}}}{T^2}\nabla T + \frac{1}{T}(\boldsymbol{\tau} : \nabla\mathbf{V}) \geq 0 \tag{9}$$

If only the conductive heat transfer is considered, e.g. $\dot{\mathbf{q}}_{\mathrm{ht}} = -k\nabla T$, the inequality is:

$$k\frac{\nabla T \cdot \nabla T}{T^2} + \frac{1}{T}(\boldsymbol{\tau} : \nabla\mathbf{V}) \geq 0 \tag{10}$$

Considering the above result, the constitutive or material relations, empirical deduced or experimental measured for thermal conductivity $k(\rho, T)$, first viscosity coefficient $\mu(\rho, T)$, second viscosity coefficient $\lambda(\rho, T)$ and after that, the *determination of viscosity stress must to respect the above inequality*.

6. PERFORMANCE COMPARISON BETWEEN TC, PWS, MC AND NA ENGINES

A comparative analysis between performance indicators of naturally aspirated engines (NA) and different types of charged engines (by pressure wave supercharger – PWS, turbocharger – TC or mechanical compressor – MC) show differences in power, torque, efficiency and specific fuel consumption [23]. As represented in Figs. 11-15, supercharging using a PWS is more efficient than other methods in terms of adiabatic efficiency in normal operating conditions, due to the fresh air cooling effect inside the channels, compared to the TC or MC in which the efficiency is lowered by the internal heat increase [50, 51, 52, 53]. For the Comprex PWS, a neglectable power loss is registered by driving the rotor from the crankshaft (~ 0.5% of total power output [49]), but none is lost for compression of fresh intake air. Also, the mechanically charged engine encounter a small variation of torque over the entire speed range, higher but similar in shape with the NA engine [50]. Comparing the torque values for passenger car engines' (PKW) charged by TC or PWS, pique values were achieved close to 2000 rpm, whilst the highest power output was achieved at 3000-3500 rpm, lower for the TC than for the PWS.

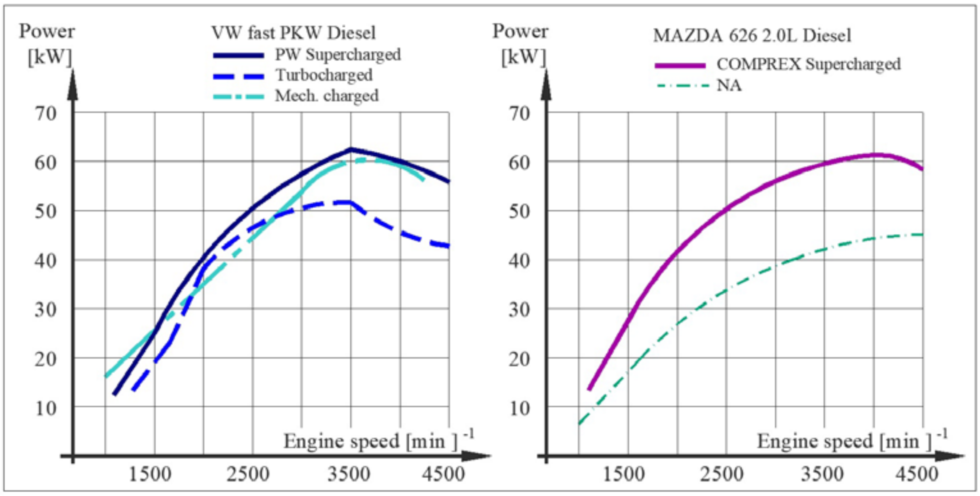

Fig. 11 – Comparative power output for passenger car engines (PKW).

The truck engines (LKW) register rather similar values for power for both TC and PWS, slightly higher for PWS close to the pique values and significantly higher than the NA engines.

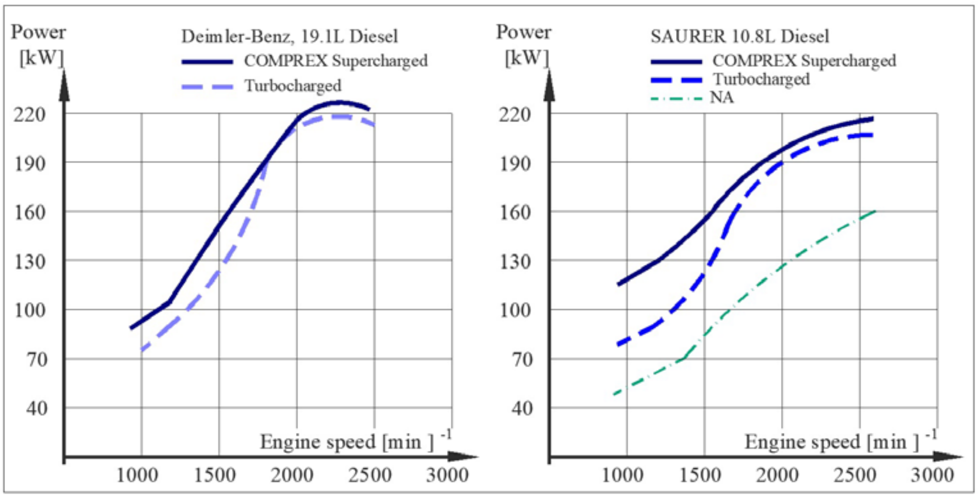

Fig. 12 – Comparative power output for truck engines (LKW).

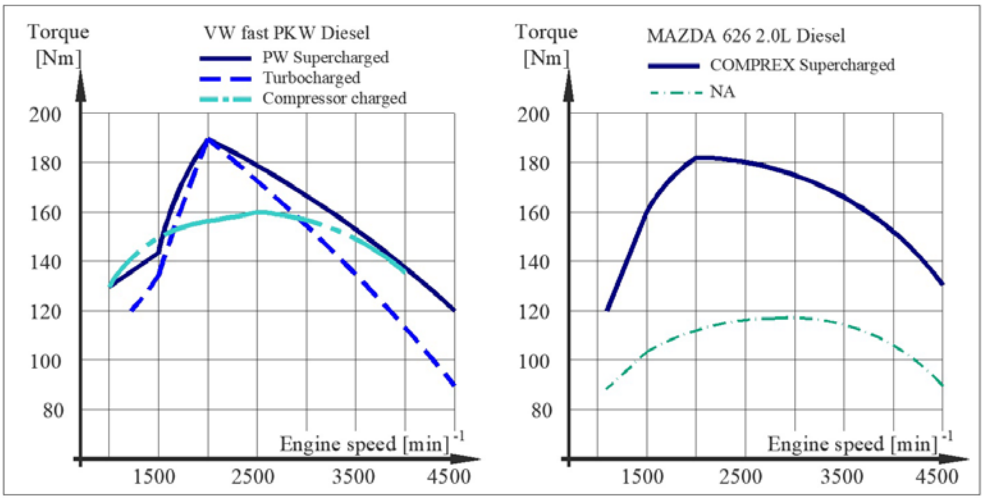

Fig. 13 – Torque variation for passenger car engine.

The fuel specific consumption is lower for the NA engine and for TC at low speeds than the PWS charged similar engine, but less efficient at high speeds. Therefore, as an optimization solution, the engine should operate in a combined mode in order to achieve maximum power with low fuel consumption.

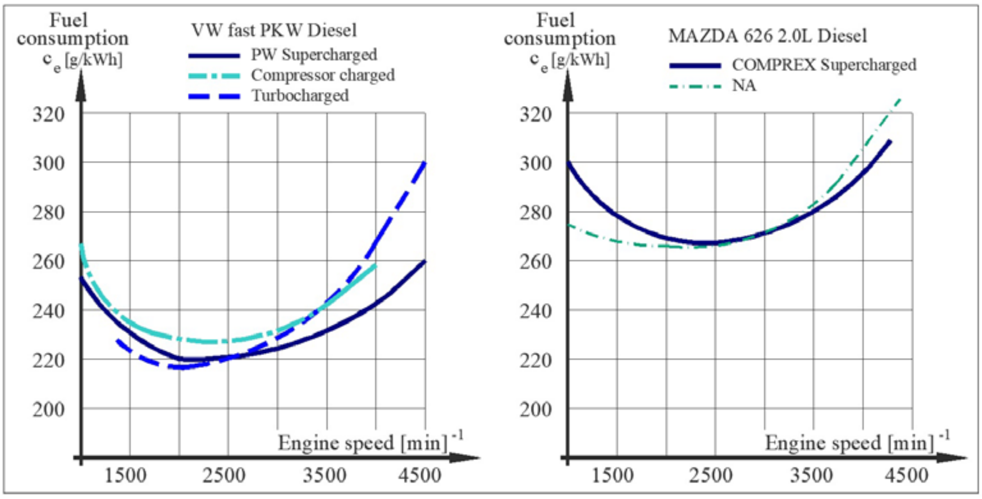

Fig. 14 – Specific fuel consumption for passenger cars' engines.

In conclusion, PWSs showed a general better behavior compared to TC and MC, mainly thanks to the PWS's feature to instantaneously and directly transmit energy from one working fluid to the other. More, PWS has no "turbo lag" in operation, as known for TC. This this leads to better maneuverability of the vehicle equipped by PW supercharged engine, i.e. a faster, more steady and smooth dynamic response to the driver's demands [30, 23].

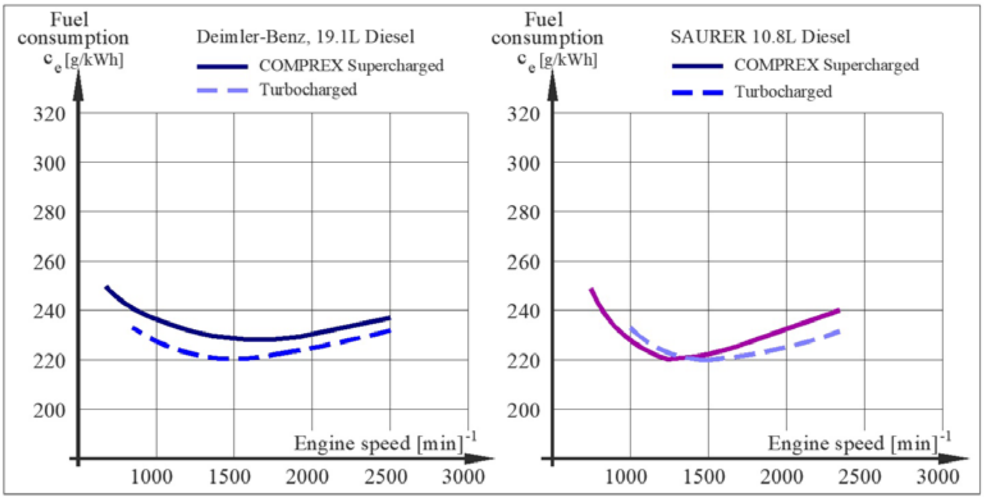

Fig. 15 – Specific fuel consumption for truck engines.

7. CONCLUSIONS - PRESSURE WAVE TECHNOLOGY BENEFITS AND SHORTCOMINGS

When using the pressure wave technology for ICE supercharging, the main advantage of a PWS over the conventional turbocharger lies in its immediate load response, since the transfer of high exhaust gas energy towards the intake air is instantly realized. Also, the PWS shows its operational efficiency, better at high speeds. Furthermore, the PWS has no lag caused by inertia comparing to the turbochargers. Other advantages outlined by ITE Circuit Breaker Company based on road testing with a diesel truck, are: clean exhaust, fewer gearshifts, improved fuel consumption, no danger of over-speed and low sensitivity to unbalance [9].

Compared with the turbomachines, from the performance and construction point of view, the pressure exchanger's *advantages* can be outlined in short:

- Robust and reliable structure;
- More erosion resistant due to easier uptake of particles or droplets contained into the working fluids;
- Better resistance to high temperatures because the exposure to material maximum temperatures is rather short and in some cases a cooling effect is ensured by the cold fresh air flow;
- Higher tolerance to pick transitory pressure values;
- Lower rotational speeds of the rotor than the turbo-machines; allowing a better cell filling or scavenging;
- Faster response to operational transients;
- Important potential to generate significant rise of pressure in a short time;
- No surge limit, as experienced at axial or centrifugal turbo-compressors;
- Lower production costs comparing to some performant turboequipments.

The PW devices *disadvantages* comparing with the turbomachines are:

- Significant noise level, caused by the rotational speed range of the wave rotor that brings the frequency of the noise into the audible zone;
- Low mass flow rate considered on the frontal area, compared with the turbomachines axial flow rate [47];
- Problems induced by fatigue on areas subject to cyclical fluctuations in pressure.

Together with the benefits presented above, the cell wave rotor has some disadvantages that challenged the researchers to overcome by experimental investigations as well as by theoretical studies. In short terms, these are:

- PWS is very sensitive to the changes in the pressure losses inside the intake or exhaust systems (such as the losses caused by the soot filter clogging);
- The limitation of the start-up functioning of the PWS when exhaust gases can escape into the intake manifold and prevent the engine operation;
- Manufacturing difficulties that can raise the PWS costs of production and, therefore, the final price;
- Limited flexibility in montage the PWS system, due to belt drive;
- Significant sensitivity of PWS operation to increased resistance on the low-pressure side;
- High quantities of exhaust gas and scavenging air;
- Noisy operation;
- The difficulty of designing the PWS, that include finding the optimal geometrical configuration, controlling, as well as modelling and understanding of the complex unsteady phenomena that occur inside the rotor cells, sealing and expansion issues, mechanical problems, etc.

The pressure wave supercharger has proved its potential in improving the internal combustion engine’s performance, being in essence a simple device, operating with the fastest response. It can be tuned to the ICE to provide fresh air boost over the full range of engine’s operating speeds. Optimizing solutions for the PWS functioning were presented in the sections above, the promising results achieved by researchers over the years being encouraging for further investigations. Further studies will highlight the advantages offered by the PW technology and point-out on the most appropriate applications.

Received in February 2018

REFERENCES

1. http://ec.europa.eu/clima/citizens/eu/index_ro.htm – European Comision – Climate Action.
2. http://www.eea.europa.eu/ EEA, 2015b, *Monitoring CO2 emissions from new passenger cars and vans in 2014*, EEA Technical report No 16/2015, European Environment Agency.
3. PETITT, J., *Sport compact turbos & blowers*, CarTech Inc, 2004.
4. HEISLER, H., *Advances engine technology*, SAE International, 1995.
5. AKBARI, P., NALIM, R., MUELLER, N., *A review of wave rotor technology and its applications*, Journal of Engineering for Gas Turbines and Power, **128**, *4*, pp. 717–735, 2006.
6. *Automotive Handbook, 3rd Edition*, Robert Bosch GmbH, 1993, SAE.
7. PRANAV, A.S., *Wave rotor test rig design procedure for gas turbine enhancement*, Michigan State University, Department of Mechanical Engineering, 2008.
8. KNAUFF, R., *Converting pressures of liberated gas energy into mechanical work*, British Patent no. 2818, 1906.

9. AZOURY, P.H., *An introduction to the dynamic pressure exchanger*, Proceedings of the Institution of Mechanical Engineers, **180**, *1*, pp. 451–480, 1965.
10. LEBRE, A.F., *British Patent no. 290 669*, 1928.
11. COSTIUC, I., CHIRU, A., *Thermodynamic process modeling in pressure wave superchargers*, International Congress of Automotive and Transport Engineering CONAT, Braşov, Romania, October 2016.
12. WEBER, F., *Mean value modelling of wave pressure supercharger including exhaust gas recirculation effects*, Swiss Federal Institute of Technology, Zürich, Switzerland, 2001.
13. BERCHTOLD, M., *The Comprex diesel supercharger*, SAE Paper 590001, vol. **67**, 1959.
14. SEIPPEL, C., *Swiss Patent no. 225426*, 1940.
15. SEIPPEL, C., *Swiss Patent no. 229280*, 1942.
16. SEIPPEL, C., *"Pressure Exchanger"*, U.S. Patent no. 2399394, 1946.
17. SEIPPEL, C., *"Gas Turbine Installation"*, U.S. Patent no. 2461186, 1949.
18. BURGHARD, H., *German Patent no. 485 386*, 1928.
19. WEATHERSTON, R.C., HERTZBERG, A., *The energy exchanger, a new concept for high-efficiency gas turbine cycles*, ASME Journal of Engineering for Power, **89**, *2*, pp. 217–227, 1967.
20. LEAHU, C.I., ABAITANCEI, H., RADU, S., *Drive with rotative speed independent from the engine, of the pressure wave compressors*, Recent Journal, **14**, *1*, pp. 29–35, 2013.
21. AKBARI, P., NALIM, R., *Review of recent developments in wave rotor combustion technology*, Journal of Propulsion and Power, **25**, *4*, pp. 833–844, 2009.
22. HEYWOOD, J., *Internal combustion engine fundamentals*, McGraw-Hill International Editions, 1988.
23. COSTIUC, I., CHIRU, A., *Evolution of the pressure wave supercharger concept*, IOP Conference Series: Materials Science and Engineering, **252**, p. 012081, 2017.
24. LEAHU, C., *Optimisation of compression ignition engine using supercharging systems*, PhD Thesis, University Transilvania Brasov, 2011.
25. WEBER, H.E., *Shock wave engine design*, John Wiley & Sons Inc., N.Y., 1995.
26. AZOURY, P.H., *Engineering applications of unsteady fluid flow*, John Wiley & Sons, N.Y., 1992.
27. KENTFIELD, J.A.C., *Wave rotors and highlights of their development*, 34th AIAA/ASME/SAE/ASEE Joint Propulsion Conference and Exhibit, p. 3248, 1998.
28. MORITZ, R., *Rolls-Royce study of wave rotors (1965-1970)*, Proceedings ONR/NAVAIR Wave Rotor Research and Technology Workshop, Monterey, CA, Report NPS-67-85-008, pp. 116, 1985.
29. ZEHNDER, G., MAYER, A., MATHEWS, L., *The Free Running Comprex®*, SAE paper no. 890452, 1989.
30. SPRING, P., *Modeling and control of pressure-wave supercharged engine systems*, PhD Thesis, ETH Zurich, 2006.
31. MATHUR, A., *Design and experimental verification of wave rotor cycles*, Proceedings Wave Rotor Research and Technology Workshop, Report NPS-67-85-008, pp. 215–228, 1985.
32. MATHUR, A., *Wave rotor research: a computer code for preliminary design of wave diagrams*, Naval Postgraduate School, Monterey, CA, Report NPS67-85-006CR, 1985.
33. THAYER, W.J., *The MSNW energy exchanger research program*, Proceedings ONR/NAVAIR Wave Rotor Research and Technology Workshop, Report NPS-67-85-008, pp. 85–116, 1985.
34. TAUSSIG, R.T., HERTZBERG, A., *Wave rotors for turbomachinery*, Winter Annual Meeting of the ASME, Machinery for Direct Fluid-Fluid Energy Exchange (ed. J.F. Sladky), pp. 1–7, 1984.
35. EIDELMAN, S., *Gradual opening of rectangular and skewed wave rotor passages*, Proceedings ONR/NAVAIR Wave Rotor Research and Technology Workshop, Naval Postgraduate School, Monterey, CA, Report NPS-67-85-008, pp. 229–249, 1985.

36. PAXSON, D.E., PERKINS, H.D., *Thermal load considerations for detonative combustion-based gas turbine engines*, AIAA paper no. 3396, 2004.
37. PAXSON, D.E., *A performance map for ideal air-breathing pulse detonation engines*, AIAA Journal of Propulsion and Power, AIAA paper no. 3465, 2001.
38. MATACZYNSKI, M.R., POLANKA, M.D., HOKE, J., PAXSON, D.E., *Design and implementation of waverotor technology for small-bore two-stroke internal combustion engines*, 52nd Aerospace Sciences Meeting, AIAA SciTech Forum, AIAA paper no. 0528, 2014.
39. MATACZYNSKI, M.R., HOKE, J., POLANKA, M.D., NEES, J.B., PAXSON, D.E., *Validation and simulation of a small-scale pressure wave supercharger*, 53th Aerospace Sciences Meeting, AIAA paper no. 134, 2015.
40. MATACZYNSKI, M.R., HOKE, J., POLANKA, M.D., PAXSON, D.E., *Experimental performance of a small scale pressure wave supercharger*, 54th Aerospace Sciences Meeting, AIAA paper no. 0768, 2016.
41. MATACZYNSKI, M.R., PAXSON, D.E., HOKE, J., SCHAUER, F., *Design and testing of a small pressure wave supercharger for an industrial diesel engine*, 55th Aerospace Sciences Meeting, AIAA paper no. 1624, 2017.
42. http://www.swissauto.com
43. GUZELLA, L., MARTIN, R., *Das SAVE Motorkonzept*, MTZ Motortechnische Zeitschrift, **59**, *10*, pp. 644–650, 1998.
44. DOERFLER, P., *Comprex supercharging of vehicle diesel engines*, SAE T.P. no. 750335, doi:10.4271/750335, 1975.
45. GYARMATHY, G., *How does the Comprex pressure-wave supercharger work?*, SAE T.P. no. 830234, 1983.
46. BERCHTOLD, M. *Supercharging with Comprex*, Proceedings ONR/NAVAIR Wave Rotor Research and Technology Workshop, Report NPS-67-85-008, pp. 51–74, 1985.
47. KENTFIELD, J.A.C., *The pressure exchanger: an introduction including a review of the work of power jets (R&D) Ltd.*, Proceedings ONR/NAVAIR Wave Rotor Research and Technology Workshop, Report NPS-67-85-008, pp. 9–50, 1985.
48. PISCHINGER, S., *Verbrennungskraftmaschinen II, VK II* (Lecture notes), RWTH Aachen University, 1984.
49. WALZER, P., EMMENTHAL, K.-D., ROTTENKOLBE, P., *Aufladesysteme fur PKW – Antrieb*, Automobil Industrie, 1989.
50. WIEDEMANN, B., RHODE, W., Aufladetechnische Konferenz Lehrstuhl f. Angewandte Thermodynamik, RWTH, LAT–FEV, Aachen, 1984.
51. TATSUTOMI, Y., YOSHIZU, K., KOMAGAMINE, M., *Der Dieselmotor mit Comprex-Aufladung fur den Mazda 626*, Motortechnishe Zeitschrift, **51**, pp. 126, 1990.
52. WUNSCH, A., *Aufladung von Fahrzeugdieselmotoren mit dem Abgasturbolader und mit der Druckwellenmaschine Comprex*, Motortechnishe Zeitschrift, **31**, pp. 19, 1990.
53. HIERETH, H., *Eignungsabschatzung neuer Aufladesystemefur Fahrzeugmotoren,* Aufladetechnische Konferenz Lehrstuhl f. Angewandte Thermodynamik, Aachen, 1984.
54. ANDERSON, J.D. Jr., *Fundamentals of Aerodynamics* (ed. 5), McGraw-Hill, 2011.
55. POWERS, J.M., *Lecture notes on gas dynamics*, University of Notre Dame, USA, 2015.