A STUDY OF PARAMETERS INFLUENCING THE PERFORMANCE OF A PRESSURE WAVE SUPERCHARGER (PWS)

Vladimir Mardarescu*, Marius Hirceaga ,Gheoghe-Alexandru Radu Cristian Leahu

Transilvania University of Brasov

KEYWORDS – Rotor, shock wave, wave speed, boundary conditions, pressure ratio

ABSTRACT – The main objective of the following paper is to identify and elaborate the parameters that have a significant impact on the operational characteristics of the Pressure Wave Supercharger also known as Comprex. This type of supercharger can transfer some of the energy from the exhaust gases in the form of pressure waves, directly to the fresh air entering the internal combustion engine. The device is in fact a pressure exchanger with a greater adiabatic efficiency than other well known steady-flow machine like compressors or turbochargers. Besides other advantages like fast response on acceleration (no turbo lag) and relative high boost pressure, even when the engine is running at partial loads and low speeds, the PWS has also some shortcomings in matching the speed and load requirements for the entire operational range of the engine. The reason for this problem is an inadequate speed of the supercharger rotor which can lead to a poor distribution of the pressure-wave phenomena that can compromise the performance of the engine due to an inappropriate gas exchange at he inlet phase.In order to determine the optimal speed of the Comprex, the authors focused on operating parameters like pressure and temperature that have a fundamental influence on the processes taking place inside the PWS. The study was made with the help of a commercially CFD software, which enabled a precise simulation of the high pressure part of the supercharger.

INTRODUCTION

The Pressure Wave Supercharger, also known by its commercial name, Comprex, is a pressure wave machine whose working principle is based on the fact that when two fluids with different pressure are brought in direct contact, the pressure equalization occurs faster than the mixing process. Claude Seippel from Brown Boveri Company, (later Abea Brown Boveri), produced the first operational pressure wave machine in 1940 (1). He used this device as a high-pressure stage for a gas-turbine locomotive engine. Later, prestigious institutions like NASA, Rolls Royce, Indiana University Purdue University Indianapolis or Michigan State University used the same technique in order to obtain higher overall efficiency from gas-turbine engines. Their research programs include projects like Pulse Detonating Engine, Wave Rotors or Ultra-Micro Gas Turbine. (10, 12)

The Brown Boveri Company also made extensive researches for the implementation of the Comprex as a boosting device for Internal Combustion Engines. The PWS was seen as an alternative to the turbocharger and it was used mainly for truck engines. Only in the late 80s the Comprex was mass-produced by the Mazda Company for the 626 Cappela model in order to equip the 2-liter Diesel engine (9). Until now, it is the only major commercial application of this type of supercharger. The main advantages of the PWS compared with the turbocharger are higher boost pressure and fast response to acceleration (no turbo-lag).

The Comprex is more suitable for small displacement engines, (< 2 dm^3), where it has greater efficiency than other supercharging device. It can be used both for SI engines and Diesel engines, with the mention that, because of the high boost pressure (up to 2.8 bar) knocking can occur for the SI engine if it's not properly controlled. The non-steady nature of the wave phenomena limits the operating range of the PWS , so matching it with the whole range of loads and speed of the ICE engine is challenging. In our days, the company Swissauto Wenko AG produces a modern version of Comprex, called the Hyprex (11). This device is used as a supercharger for small SI engines.

Another application of the pressure wave machines is in refrigeration systems. The main advantage in using such a pressure wave machine in a refrigeration system is that it increases efficiency and reduces the size and cost of the system. In this area the company Power Jets Ltd studied the implementation of wave rotor technology in refrigerating cycles using water as a refrigerant (10).

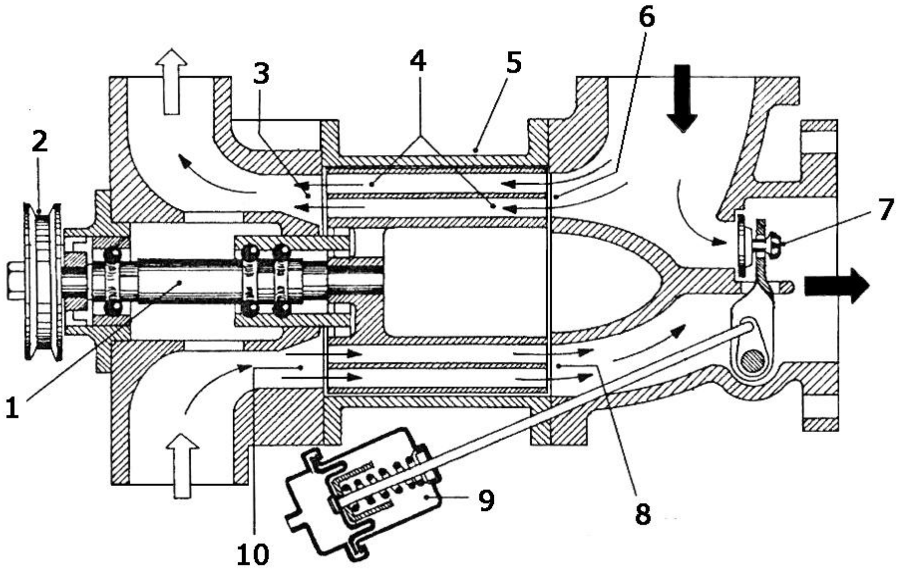

Fig.1. The main components of the PWS (7): 1 - drive shaft; 2 - drive pulley; 3 - compressed air exit port; 4 - rotor channels; 5 - rotor case; 6 - exhaust gas intake port; 7 - wastegate valve; 8 - exhaust gas exit port; 9 - wastegate valve actuator; 10 - air intake port.

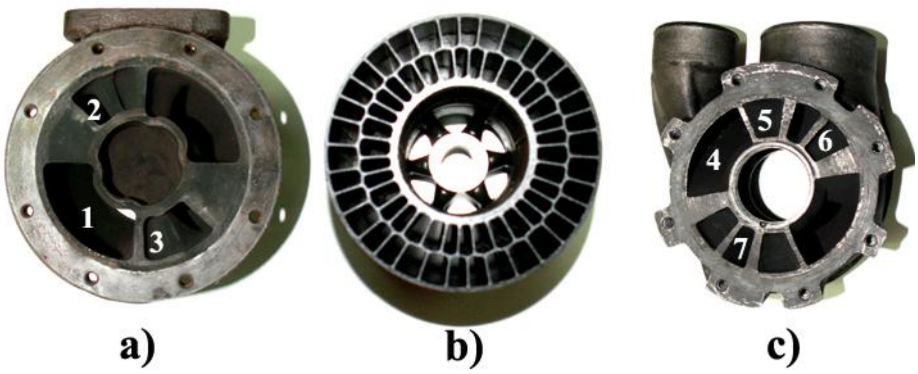

Fig.2. a) "hot" stator; b) wave rotor; c) "cold" stator: 1 - gas exit port; 2 - gas intake port; 3 - gas pocket; 4 - air intake port; 5 - air expansion pocket; 6 - air compression pocket; 7 - air exit port.

In the wave rotor the pressurized exhaust gases compress the fresh air. It has a cylindrical shape with many axial channels stacked within 2 rows. The rows are slightly displaced from one another, in order to reduce through interference, the specific high frequency sound of the compressor. The rotor is made from a nickel-steel alloy by precision casting. The stator of the Comprex has two parts: the “hot” part, made from cast iron, where the exhaust gases enter and exit the wave rotor, and the “cold” part, made form cast aluminum, where the fresh air enters and the compressed air exits the rotor. The waste-gate valve, which controls the exhaust gas flow into the PWS, is situated in the hot part of the stator. Also, the cold part of the stator contains the drive shaft and the bearings of the wave rotor.

In order to reduce the speed and the dimensions of the PWS, the stator has two intake and exit ports for each part, which translates in two complete cycles for every rotation of the wave rotor (3). This solution also helps to reduce the thermal stress in the hot part of the stator. Apart the intake and exits port, there are also so called “pockets” imbedded in the stator, as can be seen in figure 2: the air compression and expansion pockets in the cold part of the stator and the gas pocket in the hot part of the stator. These pockets have the role to minimize the effect of the mismatched waves that occur when the Comprex is driven at a non-optimal speed. The driving of the PWS is needed only for matching the rotor position with the correct opening and closing of the ports. The fresh air is compressed exclusively by exhaust gases and not by mechanical work taken from the engine like in the case of the supercharger.(6).

WORKING PRINCIPLE

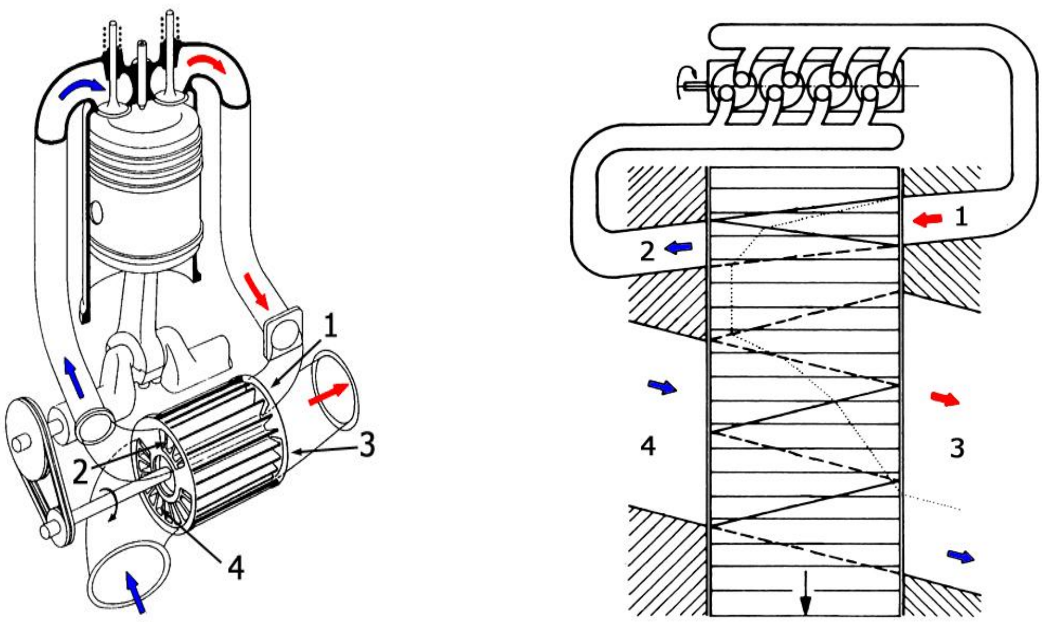

Fig.3. PWS’s working principle (2): 1 - gas intake port (GI); 2 - air exit port (AE); 3 - gas exit port (GE); 4 - air intake port (AI)

The working principle of the PWS is presented in figure 3. In the unwrapped picture of the wave rotor it can be seen how the waves propagation occur in the rotor channels during a full cycle. The lines represent compression waves, the dashed lines, expansion waves and the dotted line represent the air/gas interface. At the beginning of the cycle in the rotor cells there is only fresh air at approximately atmospheric pressure. As the rotor turns, the gas intake port, (GI), starts to open. Because of the pressure difference between the hot exhaust gases and the air inside the rotor channel, a compression wave is triggered.

This primary compression wave travels the length of the rotor and it reflects itself back on the closed wall of the cold part of the stator. The air exit port, (AE), opens up after the compression wave is reflected back. In this way the air inside the channel is compressed twice. The ideal speed of the Comprex is achieved when the reflected compression wave reaches the hot part of the stator and the GI is starting to close. When this happens an expansion wave is triggered which should reach the cold part of the stator at the time when the AE starts to close. After the AE is closed, the air/gas mix inside the rotor channel still has a higher pressure than the ambient pressure so when the gas exit port, (GE), opens, another expansion wave is triggered. The air intake port, (AI), is opened when this expansion wave reaches the cold part of the stator. Now, the fast exiting gas sucks fresh air into the wave rotor. With the closing of the GE the cycle is complete and the process can start all over gain.

During an ideal cycle the air/gas interface does not reach the AE and no exhaust gas is sucked into the engine cylinders. In reality there are a multitude of factors which can have a negative impact on the overall efficiency of the PWS. First of all, the ports do not open and close instantly. Because of this the wave front is not very well defined and the pressure variation is not that steep. Some mixing in the air/gas interface does occur and also the air is heated supplementary because of the direct contact between the exhaust gases and the fresh air. There is also some leakage between the rotor and the stator and when the ICE is idling or functions at very low loads, the pressure difference between the exhaust gases and the fresh air is very small. Because of this some of the exhaust gases will remain inside the rotor, which eventually will enter in the ICE. This could cause a loss of power or in the worse case scenario, it will completely stall the engine. To avoid this problem, the Comprex has a start-up valve which can be closed when the engine starts or is in idle. In this regime, the ICE functions like a normally aspirated engine (4, 5).

GOVERNING PARAMETERS OF THE WAVE PHENOMENA INSIDE THE PWS

The dynamics of the shock waves, which take place inside the wave rotor, can be approximated with the help of the Hogoniot - Rankine equations for a one-dimensional shock wave in a narrow adiabatic tube (8):

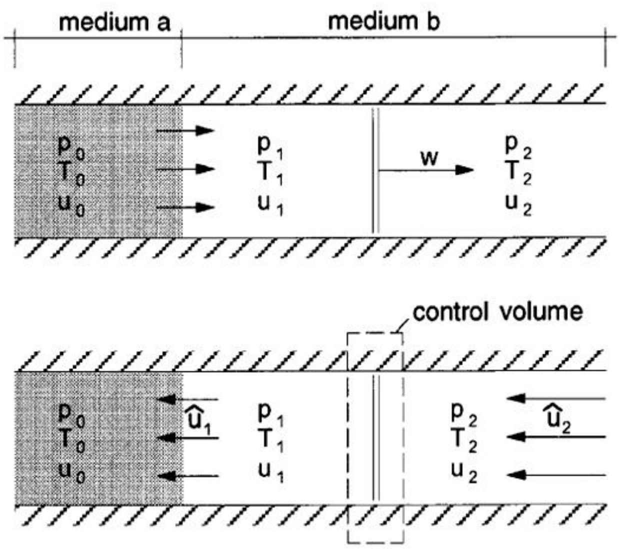

Fig.4. One-dimensional shock wave: Above - one-dimensional shock wave in a narrow, adiabatic tube; Below - stationary shock wave in the control volume.

For the stationary shock wave, we have the following equations:

$$\hat{u}_1 = w - u_1 \quad [1]$$
$$\hat{u}_2 = w \quad [2]$$

Mass balance:

$$\rho_1 \cdot \hat{u}_1 = \rho_2 \cdot \hat{u}_2 \quad [3]$$

Impulse balance:

$$p_1 + \rho_1 \cdot (\hat{u}_1)^2 = p_2 + \rho_2 \cdot (\hat{u}_2)^2 \quad [4]$$

Energy balance:

$$c_p \cdot T_1 + \frac{(\hat{u}_1)^2}{2} = c_p \cdot T_2 + \frac{(\hat{u}_2)^2}{2} \quad [5]$$

Perfect gas behavior:

$$p_1 = \rho_1 \cdot R \cdot T_1 \quad [6]$$

Contact conditions:

$$u_0 = u_1 \quad [7]$$
$$p_0 = p_1 \quad [8]$$

After some mathematical transformations, we can extrapolate the unknown parameters as follows:

Pressure ratio over the shock:

$$\pi_s = \frac{p_0}{p_2} \quad [9]$$

Fluid velocity in state 1:

$$u_1 = \sqrt{2 \cdot R \cdot T_2} \cdot \frac{\pi_s - 1}{\sqrt{\pi_s \cdot (\kappa + 1) + \kappa - 1}} \quad [10]$$

Shock velocity:

$$w = \sqrt{\frac{R \cdot T_2}{2}} \cdot \sqrt{\pi_s \cdot (\kappa + 1) + \kappa - 1} \quad [11]$$

Fluid temperature in state 1:

$$T_1 = T_2 \cdot \pi_s \frac{\pi_s \cdot (\kappa - 1) + \kappa + 1}{\pi_s \cdot (\kappa + 1) + \kappa - 1} \qquad [12]$$

The annotation used in the equations [1-12] are:

$\hat{u}_1$	fluid velocity traveling away from the stationary shock wave;
$\hat{u}_2$	fluid velocity traveling towards the stationary shock wave;
w	shock velocity;
$u_{0,1,2}$	fluid velocities;
$p_{0,1,2}$	fluid pressures;
$T_{0,1,2}$	fluid temperatures;
$\rho_{1,2}$	fluid densities;
R	universal gas constant;
κ	adiabatic exponent;
c_p	specific heat capacity (p = ct).

According to the equations [10-12], the governing parameters of the wave phenomena inside the PWS are:

p_eg	exhaust gas pressure entering the rotor's channel;
T_eg	exhaust gas temperature entering the rotor's channel;
p_air	air pressure in the rotor's channel;
T_air	air temperature in the rotor's channel;
n	polytropic exponent of the air/gas in the rotor's channel.

The air pressure inside the rotor's channel is usually very close to value of the atmospheric pressure and it will be regarded as constant during the simulations. Also the polytropic exponent of the air/gas is parameter that is very hard to estimate and calculate accurately. For this reasons it will be replaced with the constant value of the gases adiabatic exponent.

Using a commercially CFD software, a 3D transient-dynamic model of the PWS has been developed in order to simulate the high pressure part of the supercharger. The main purpose of the simulations was to investigate the impact of the operating parameters of the Comprex on the optimal rotational speed of the rotor. As described earlier , the ideal rotational speed of the rotor is attained when the secondary compression wave reaches the hot end of the stator after the gas inlet port GI is closed. If the secondary compression wave reaches the hot end of the stator before the GI is closed, it will act like an additional resistance in the exhaust system, and this leads to a reduction of the global efficiency of the engine.

The initial boundary condition for the PWS simulations were:

	Parameter	**Value**	**Unit**
p_eg	exhaust gas pressure entering the channel	1,46	bar
p_air	air pressure inside the channel	0,96	bar
T_eg	exhaust gas temperature entering the channel	569	K
T_air	air temperature inside the channel	320	K
w_CX	initial rotational speed of the PWS	7000	rpm

The results of the simulations are summarized in the following graphics:

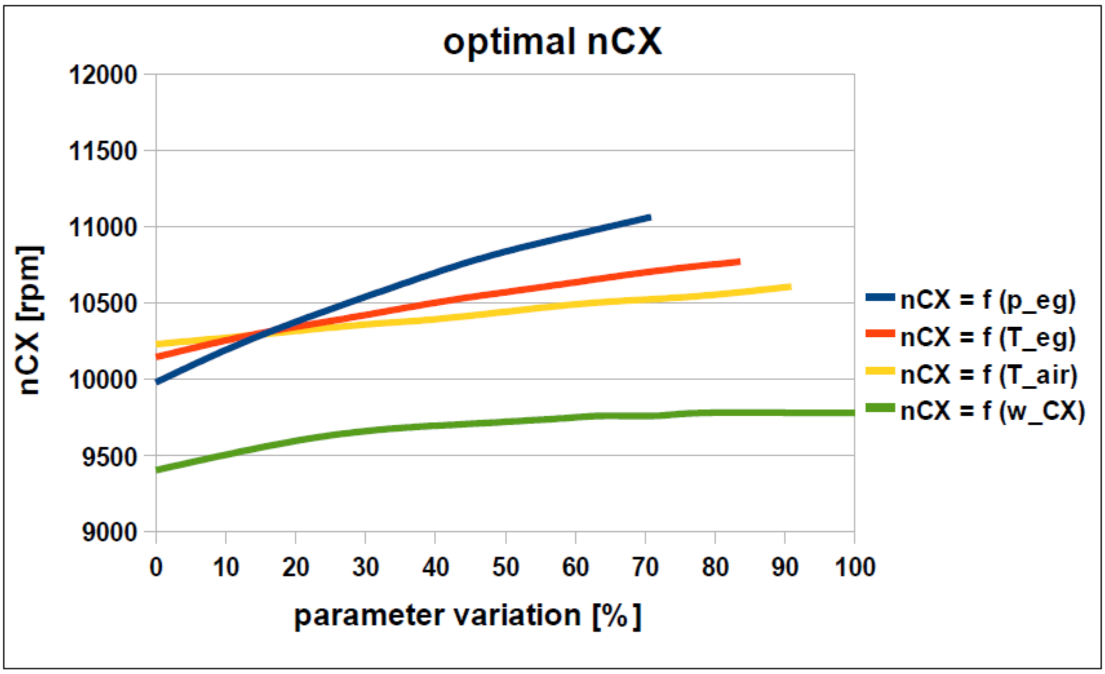

Fig.5. Optimal rotational speed variation as parameter function

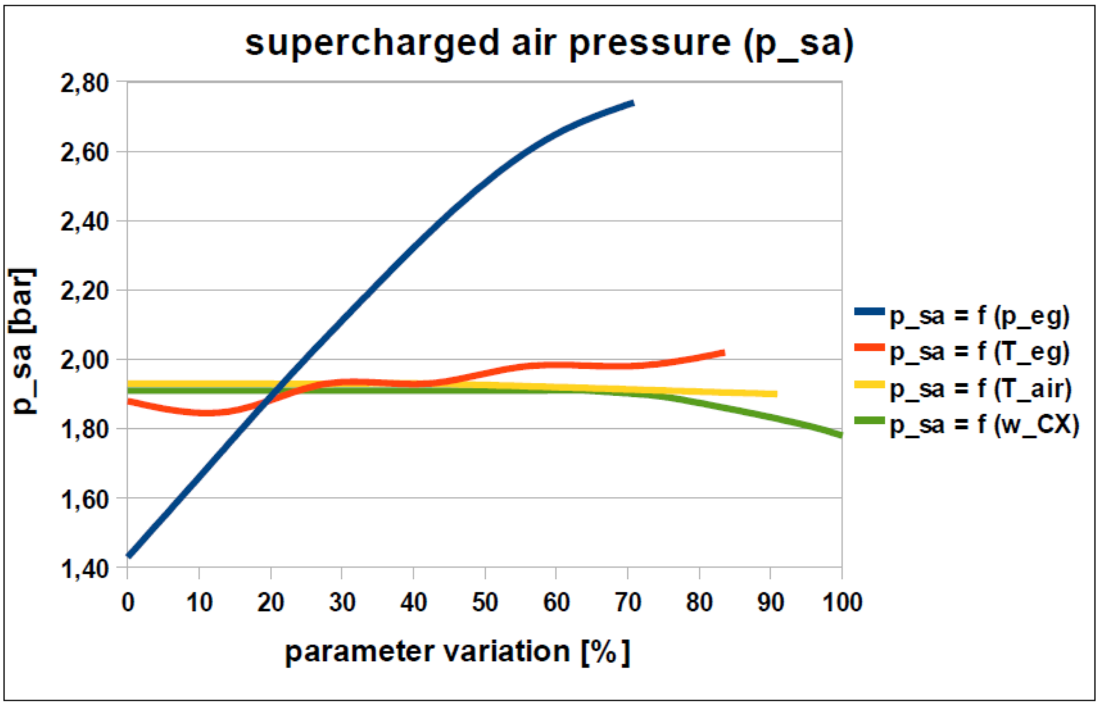

Fig.6. Supercharged air pressure variation as parameter function

CONCLUSIONS

The most significant parameter which has the biggest impact on the optimal rotational speed of the rotor and the supercharged air pressure is the exhaust gas pressure entering the PWS (blue curve in the graphics).

The optimal speed of a PWS is not proportionally with the engine speed. It depends on the load of the ICE, which affects the pressure and the temperature of the exhaust gases. The power consumed in driving the compressor is very small, so an electric engine could drive the Comprex at an optimum speed.

The boost pressure depends mainly on the exhaust gas pressure entering the Comprex. By controlling the exhaust gas flow into the PWS with the help of the waste-gate valve, it is possible to control the power output of the supercharged engine.

The simulation results also show that the rotational speed of the rotor plays an important role for the calculation accuracy, compared with the results obtained using a simple static PWS model.

REFERENCES

(1) Berchtold, M., "The Comprex Diesel Supercharger", SAE Paper 590001,1959;

(2) Doerfler, P.K., "Comprex Supercharging of Vehicle Diesel Engines", SAE Paper 750335, 1975;

(3) Gyarmathy, G., "How Does Comprex Pressure-Wave Supercharger Work?", SAE Paper 830234, 1983;

(4) Schruf G. M., Kollbrunner T. A., "Application and Matching of the Comprex Pressure-Wave Supercharger to Automotive Diesel Engines", SAE Paper 840133, 1984;

(5) Mayer A., El Nashar I., Komauer C., "Characteristics and Matching of the Pressure Wave Supercharger Comprex to a Passenger Car Engine", SAE Paper 845015, 1984;

(6) Zehnder, G. and A. Mayer., "Comprex Pressure-Wave Supercharging for Automotive Diesels - State-of-the-Art", Proceedings of International Congress & Exposition. Detroit, MI, 1984;

(7) Heisler, H., "Advanced Engine Technology", Warrendale, 1995;

(8) Weber, F., "Mean Value Modelling of Wave Pressure Supercharger Including Exhaust Gas Recirculation Effects", Swiss Federal Institute of Technology: Zürich, Switzerland, 2001;

(9) Müller N., Akbari P., "Gas Dynamic Design Analyses of Charging Zone for Reverse-Flow Pressure Wave Superchargers", ASME Paper ICES2003-690, 2003;

(10) Akbari, P., M.R. Nalim, and N. Müller, "A Review of Wave Rotor Technology and its Applications, Proceedings of 2004", ASME International Mechanical Engineering Congress & Exposition, 2004;

(11) http://www.swissauto.com

(12) http://www.grc.nasa.gov/WWW/cdtb/projects/waverotor/index.html