PRESSURE WAVE SUPERCHARGER 0-D MODEL

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SHRNUTÍ

Příspěvek představuje využití a potenciál 0-D algebraického modelu tlakového výměníku. Tlakový výměník je nestacionární zařízení, které využívá tlakových vln ke stlačování tekutin v sérii axiálních kanálků. Ke stlačování dochází přenosem energie z vysokotlakého do nízkotlakého proudu bez oddělení pevnými stěnami. 0-D model vychází z fyzikálního popisu procesů uvnitř tlakového výměníku. Zjednodušený popis procesů je korigován kalibračními koeficienty určenými pomocí 1-D modelu. Kalibrační koeficienty jsou do 0-D modelu přeneseny regresními funkcemi.

KLÍČOVÁ SLOVA: TLAKOVÝ VÝMĚNÍK, PŘEPLŇOVÁNÍ, ALGEBRAICKÉ MODELY

ABSTRACT

This paper is aimed at presenting the role and potential of a 0-D algebraic model of a pressure wave supercharger (PWS – Comprex® or Hyprex®). The PWS is a non-steady flow device that uses shock waves to pressurize fluids by transferring energy from a high-pressure flow to a low-pressure flow without separation by physical walls in a series of axial cells. The 0-D model aims at offering a simple description of processes inside a PWS based on physics, and enables close to real time simulation. The simplified processes are corrected with calibration coefficients determined using a 1-D model and transferred to the 0-D model using regression functions.

KEYWORDS: PRESSURE WAVE SUPERCHARGER, TURBOCHARGING, SUPERCHARGING, ALGEBRAIC MODEL

1. INTRODUCTION

A pressure wave supercharger (PWS – Comprex® or Hyprex®) poses a challenging task for a simplified model. It can be simulated using 1-D methods – cf. [2] but the simulation takes quite a long time due to the need for fine discretization of the many PWS cells. For optimization of engine operation, a faster model was required.

The developed model replaces a standard turbocharger model for engine transient operation introduced in [1]. It is important to repeat the significant features of this model for better understanding of the PWS model function. In the 0-D turbocharger model, the current pressure (both intake and exhaust), temperature in the exhaust manifold and the turbocharger speed is known from the previous time history. The mass flow rate through a turbine is determined from a turbine model, which makes it possible to predict the further pressure and temperature development in the exhaust manifold. The

turbocharger model calculates the turbine power to find the time history of turbocharger speed while comparing torques of both turbine and compressor. Turbocharger speed is used together with the current pressure at the compressor outlet to determine compressor mass flow rate and outlet air temperature.

Accordingly, the mass flow rates of air delivered to the engine intake manifold and that of exhaust gas leaving through the PWS cells can be determined using a PWS model together with the temperature of compressed air. Moreover, EGR peculiar to the PWS has to be predicted; i.e., the exhaust gas content in the intake manifold mixture should also be calculated. The scavenging of the PWS, i.e., the air passed directly to exhaust, and the resulting temperature of exhaust gas leaving the PWS may also be simulated.

The PWS speed is important with respect to efficiency. Hence if necessary, the PWS speed can also be simulated as a dynamic value. However, in this case the link between engine parameters

and the PWS shaft torque is very weak. For the initial model development, optimum PWS speed is assumed at all times. The 0-D model was defined and calibrated after some of its variants were tested using 1-D results.

2. PRESSURE WAVE SUPERCHARGER 0-D MODEL

2.1 PWS MODEL FEATURES

At the beginning of the exhaust process the PWS cell is filled with air of mass m_a , which will be delivered to the engine later (Figure 1). The density of inlet air ρ_{a1} is reduced by inlet volumetric efficiency η_{va} , which takes into account residual gases and charge heating by hot walls. Unlike its well known IC

engine definition, it is here defined with respect to the outlet air mass flow rate, which avoids some unnecessary iterations. The cell content is compressed by exhaust gas to a volume that depends on the exhaust gas pressure. Since the process is (approximately) adiabatic and no external work is done, the temperature (in fact, enthalpy) of the exhaust gas is (approximately) constant. The real density of exhaust gas at the start of compression is corrected by the exhaust volumetric efficiency η_{ve} . The compression of air is assumed to be irreversible and (approximately) adiabatic. The density of air after compression is calculated by isentropic change corrected with respect to irreversibilities and heat transfer using compression efficiency η_a . The mass m_e of exhaust gas delivered to the PWS determines the exhaust mass flow rate at given pressure and temperature in an exhaust manifold upstream of the PWS.

FIGURE 1: Schematic of gas fluxes in a PWS cell
OBRÁZEK 1: Toky plynů v kanálu tlakového výměníku

The air trapped in a cell before compression starts is assumed to be delivered to the engine (a), in the general case with some additional exhaust gas recirculation. The air delivered through an inlet port ($a1$) is nevertheless partially lost to the exhaust port. For this reason, the need for empirical correlation factors in the PWS is higher than in the case of a turbocharger.

2.2 BASIC EQUATIONS OF A PWS 0-D MODEL

The basic equations are written for the internal volume of PWS cells, which yields the PWS volumetric flow rate capacity with inner area of cells A_{PWS}

$$\begin{aligned} V &= A_{PWS} L_{PWS} \\ \dot{V} &= A_{PWS} L_{PWS} \frac{n_{PWS}}{60} \end{aligned} \quad (1)$$

The basic equation for exhaust mass flow rate can be found at the end of compression

$$\begin{aligned} V &= V_e + V_a = \frac{m_e}{\rho_e} + \frac{m_a}{\rho_a} \text{ for unknown } m_e \\ \dot{m}_e &= m_e \frac{n_{PWS}}{60} \\ \rho_e &= \frac{P_e}{r_e T_e} \eta_{ve} \\ \dot{m}_a &= \eta_{va} \rho_{a1} \dot{V} \end{aligned} \quad (2)$$

Equation (2) can be solved if the pressure and density of air at the end of compression are known. If inlet manifold pressure is determined from the engine model, the air temperature remains the last unknown in this process. This will be introduced later. The mass conservation for the whole device (Figure 1) yields the link between 4 variables. If one of them is independent, we need to define 2 other relationships. This is done using scavenging coefficients λ_a , λ_e below. Moreover, we need to define internal flows inside the PWS, causing exhaust gas recirculation – λ_{EGR} is defined for this reason.

$$\dot{m}_{a1} - \dot{m}_a + \dot{m}_e - \dot{m}_{e2} = 0$$

$$\dot{m}_a = \lambda_a \dot{m}_{a1} \quad (3)$$

$$\dot{m}_{e2} = \lambda_e \dot{m}_e$$

$$\Delta \dot{m}_e = \lambda_{EGR} \dot{m}_{a1}$$

All three coefficients have to be determined from either a 1-D model simulation or from measurement on a real engine. The procedure for λ_a , λ_e is clear. The λ_{EGR} can be determined using a slightly tricky method, which will be described later using mass balance of oxygen in a model. The definitions stated above yield some relationships between fluxes usable for the further explanations.

$$(1 - \lambda_a) \dot{m}_{a1} + (1 - \lambda_e) \dot{m}_e = 0$$

$$\frac{1 - \lambda_a}{\lambda_e - 1} = \frac{\dot{m}_e}{\dot{m}_{a1}}$$

$$\eta_{V,a} \lambda_a = \frac{\dot{m}_e}{\dot{V} \rho_{a1}} \frac{\lambda_e - 1}{1 - \lambda_a} \quad (4)$$

$$\dot{m}_{e2} = \dot{m}_e - \Delta \dot{m}_e + \Delta \dot{m}_a$$

$$\Delta \dot{m}_a = \dot{m}_e (\lambda_e - 1) + \Delta \dot{m}_e$$

$$\dot{m}_a = \dot{m}_{a1} + \Delta \dot{m}_e - \Delta \dot{m}_a \quad \lambda_{EGR} \dot{m}_{a1} = \Delta \dot{m}_e$$

$$\Delta \dot{m}_a = \dot{m}_e (\lambda_e - 1) + \lambda_{EGR} \dot{m}_{a1} = (1 - \lambda_a + \lambda_{EGR}) \dot{m}_{a1}$$

If oxygen content in outlet boosted air is taken into account, the relationship between fresh air mass flow at PWS inlet (oxygen mass fraction 23%) and PWS outlet (oxygen mass fraction σ_{Oa}) can be found. The oxygen content in used exhaust gas is σ_{Oe} downstream of the engine.

$$\dot{m}_{Oa} = 0.23 \left(\dot{m}_{a1} - \Delta \dot{m}_a \right) + \Delta \dot{m}_e \sigma_{Oe} = 0.23 \dot{m}_{a1} (\lambda_a - \lambda_{EGR}) + \sigma_{Oe} \lambda_{EGR} \dot{m}_{a1}$$

$$\lambda_{EGR} = \left(1 - \frac{\sigma_{Oa}}{0.23} \right) \frac{\lambda_a}{1 - \frac{\sigma_{Oe}}{0.23}} \quad (5)$$

$$\dot{m}_{Oa} = 0.23 \dot{m}_a \left(1 - \frac{\lambda_{EGR}}{\lambda_a} \left(1 - \frac{\sigma_{Oe}}{0.23} \right) \right) = 0.23 \left(\dot{m}_a - \dot{m}_{EGR} \right) + \sigma_{Oe} \dot{m}_{EGR}$$

$$\dot{m}_{EGR} = \frac{\lambda_{EGR}}{\lambda_a} \dot{m}_a$$

The energy conservation yields relationships between pressures and temperatures. Heat transfer to the cell walls is almost zero during the whole cycle providing short duration fluctuations are neglected. Some heat is of course transferred directly from the exhaust gas to the inlet air.

The energy conservation takes into account slightly different mass-flow rates at PWS interfaces:

$$\begin{aligned}
 & - \dot{Q}_{walls} + \dot{m}_e c_{pe} (T_e - T_{ref}) - \dot{m}_{e2} c_{pe} (T_{e2} - T_{ref}) + \\
 & + \dot{Q}_{walls} - \dot{m}_a c_{pa} (T_a - T_{ref}) + \dot{m}_{a1} c_{pa} (T_{a1} - T_{ref}) = 0 \\
 & c_{pa} = r_a \frac{\kappa}{\kappa - 1} \quad c_{pe} = r_e \frac{\kappa_e}{\kappa_e - 1} \\
 & \dot{m}_e c_{pe} [T_e - \lambda_e T_{e2} - (1 - \lambda_e) T_{ref}] - \dot{m}_{a1} c_{pa} [\lambda_a T_a - T_{a1} - (\lambda_a - 1) T_{ref}] = 0
 \end{aligned} \tag{6}$$

The relationship between pressure ratios and temperatures can be based on the definition of isentropic efficiency, similar to turbomachinery. If this is used for determining temperatures only, not for specific powers, it can be used even for diabatic flows. The same pattern of PWS internal leakages was applied for both ideal (isentropic) process and for the real process

$$\dot{m}_{a1} c_{pa} T_{a1} \left(\lambda_a \left(\frac{p_a}{p_{a1}} \right)^{\frac{\kappa-1}{\kappa}} - 1 - (\lambda_a - 1) \frac{T_{ref}}{T_{a1}} \right) = \dot{m}_{a1} c_{pa} (\lambda_a T_a - T_{a1} - (\lambda_a - 1) T_{ref}) \eta_a \tag{7}$$

The definition of compression efficiency yields

$$\begin{aligned}
 \eta_a &= \frac{\frac{p_a}{p_{a1}}^{\frac{\kappa-1}{\kappa}} - 1 - \frac{\lambda_a - 1}{\lambda_a} \frac{T_{ref}}{T_{a1}}}{\frac{T_a}{T_{a1}} - 1 - \frac{\lambda_a - 1}{\lambda_a} \frac{T_{ref}}{T_{a1}}} = \frac{\frac{p_a}{p_{a1}}^{\frac{\kappa-1}{\kappa}} - K_a}{\frac{T_a}{T_{a1}} - K_a} \\
 K_a &= \frac{1}{\lambda_a} + \frac{\lambda_a - 1}{\lambda_a} \frac{T_{ref}}{T_{a1}}
 \end{aligned} \tag{8}$$

and the temperature after compression can then be calculated using the previous definition

$$T_a = T_{a1} \left[K_a + \frac{\left(\frac{p_a}{p_{a1}} \right)^{\frac{\kappa-1}{\kappa}} - K_a}{\eta_a} \right] \tag{9}$$

In a similar way, for the exhaust side the expansion efficiency is defined

$$\begin{aligned}
 \dot{m}_e c_{pe} T_e \left(E - \lambda_e \left(\frac{p_{e2}}{p_e} \right)^{\frac{\kappa_e-1}{\kappa_e}} \right) \eta_{ex} &= \dot{m}_e c_{pe} [T_e - \lambda_e T_{e2} - (1 - \lambda_e) T_{ref}] \\
 T_{e2} &= \frac{T_e - T_e \left(E - \lambda_e \left(\frac{p_{e2}}{p_e} \right)^{\frac{\kappa_e-1}{\kappa_e}} \right) \eta_{ex} - (1 - \lambda_e) T_{ref}}{\lambda_e}; \quad E = 1 - (1 - \lambda_e) \frac{T_{ref}}{T_e}
 \end{aligned} \tag{10}$$

For the energy balance of the whole PWS, an equation similar to turbocharger power balance, can be found with K_a from (8)

$$\dot{m}_e c_{pe} T_e \left(1 - (1 - \lambda_e) \frac{T_{ref}}{T_e} - \lambda_e \left(\frac{p_{e2}}{p_e} \right)^{\frac{\kappa_e - 1}{\kappa_e}} \right) \eta_{ex} \eta_a = \dot{m}_{a1} \lambda_a c_{pa} T_{a1} \left(\left(\frac{p_a}{p_{a1}} \right)^{\frac{\kappa - 1}{\kappa}} - \frac{1}{\lambda_a} - \frac{\lambda_a - 1}{\lambda_a} \frac{T_{ref}}{T_{a1}} \right)$$

$$\dot{m}_a = \dot{m}_e \frac{c_{pe} T_e}{c_{pa} T_{a1}} J.EX \text{ while } EX = \frac{\eta_{ex} \eta_a}{\left(\left(\frac{p_a}{p_{a1}} \right)^{\frac{\kappa - 1}{\kappa}} - K_a \right)}; J = \left(E - \lambda_e \left(\frac{p_{e2}}{p_e} \right)^{\frac{\kappa_e - 1}{\kappa_e}} \right)$$
(11)

The reason for using $J.EX$ instead of just a defined overall PWS efficiency will be explained later. This last relationship yields another opportunity for determining air mass flow rate through the PWS instead of using (2). The η_{Va} must be rather variable unlike (hopefully) $\eta_a \eta_{ex}$. The η_{Va} can be determined directly from numerical experiments, but it must satisfy the following result of (2), (11) and (14):

$$\eta_{Va} \frac{p_{a1}}{r_a T_{a1}} = \eta_{Ve} \frac{p_e}{r_e T_e} \frac{c_{pe} T_e}{c_{pa} T_{a1}} \left[1 - \frac{\eta_{Va} p_{a1}}{p_a} \left(K_a + \frac{\left(\frac{p_a}{p_{a1}} \right)^{\frac{\kappa - 1}{\kappa}} - K_a}{\eta_a} \right) \right] \left(E - \lambda_e \left(\frac{p_{e2}}{p_e} \right)^{\frac{\kappa_e - 1}{\kappa_e}} \right) \eta_{ex} \eta_a$$
(12)

2.3 PROCESS CHART OF 0-D MODEL PROGRAM

The turbocharger model may be substituted by the following equations for PWS "measured" on a virtual GT Power engine. It is recommended to determine the model coefficients from engine "experiments", e.g., at constant engine speed and load shock changes. The recommended regression takes the following regressors into account:

$$\eta_{Va} = f(n_{PWS}, p_{a1}, p_a, p_e, p_{e2}, T_{a1}, T_e, n_{engine}) = b + a_1 \frac{n_{PWS}}{1000} + a_2 \frac{n_{engine}}{1000} + a_3 \left(\frac{n_{engine}}{1000} \right)^2 + a_4 p_{a1} + a_5 p_a +$$

$$+ a_6 p_a^2 + a_7 p_a^3 + a_8 p_e + a_9 p_e^2 + a_{10} p_e^3 + a_{11} p_{e2} + a_{12} T_e + \frac{a_{13}}{T_e} + a_{14} p_a \frac{n_{engine}}{1000} + a_{15} p_e \frac{n_{engine}}{1000} + a_{16} p_a p_e$$
(13)

and in similar way for η_{Ve} , λ_a , λ_e , λ_{EGR} , η_a , $\eta_a \eta_{ex}$, $J.EX$

The exhaust gas mass-flow rate is

$$\text{if } \eta_{Va} < 1 \quad \dot{m}_e = \dot{V} \eta_{Ve} \frac{p_e}{r_e T_e} \left[1 - \frac{\eta_{Va} p_{a1}}{p_a} \left(K_a + \frac{\left(\frac{p_a}{p_{a1}} \right)^{\frac{\kappa - 1}{\kappa}} - K_a}{\eta_a} \right) \right]$$

$$\text{else} \quad \dot{m}_e = \dot{V} \eta_{Ve} \frac{p_e}{r_e T_e} \left[1 - \frac{p_{a1}}{p_a} \left(K_a + \frac{\left(\frac{p_a}{p_{a1}} \right)^{\frac{\kappa - 1}{\kappa}} - K_a}{\eta_a} \right) \right]$$
(14)

The inlet mass-flow rate is computed using definitions of $J.EX$ and E from (11) and (10) as

$$\begin{aligned} \text{if } |J - 0.5| < 0.4 \text{ then } \dot{m}_a &= \dot{m}_e \frac{c_{pe} T_e}{c_{pa} T_{a1}} \frac{J}{\left(\left(\frac{p_a}{p_{a1}} \right)^{\frac{\kappa-1}{\kappa}} - K_a \right)} \eta_{ex} \eta_a \\ \text{else } \dot{m}_a &= \dot{m}_e \frac{c_{pe} T_e}{c_{pa} T_{a1}} J.EX \\ \text{end} \end{aligned} \quad (15)$$

The reason for using equations (15) instead of (2) for inlet mass-flow rate computation is because of the significant relationship between air flow and exhaust flow and temperature, with lesser impact on the variable product $J.EX$ – cf. Figure 2. On the other hand, the PWS volumetric efficiency η_{Va} is variable. The re-circulated amount of exhaust gas is

$$\dot{m}_{EGR} = \dot{m}_a \frac{\lambda_{EGR}}{\lambda_a} \quad (16)$$

and engine inlet air temperature

$$T_a = T_{a1} \left[K_a + \frac{\left(\frac{p_a}{p_{a1}} \right)^{\frac{\kappa-1}{\kappa}} - K_a}{\eta_a} \right] \quad (17)$$

The expansion efficiency is

$$\eta_{ex} = \frac{\eta_{ex} \eta_a}{\eta_a} \quad (18)$$

and the outlet gas temperature

$$T_{e2} = \frac{T_e - T_e \left(1 - \lambda_e \left(\frac{p_{e2}}{p_e} \right)^{\frac{\kappa_e-1}{\kappa_e}} - (1 - \lambda_e) \frac{T_{ref}}{T_e} \right) \eta_{ex} - (1 - \lambda_e) T_{ref}}{\lambda_e} \quad (19)$$

Substituting the PWS equations into a 0-D turbocharger model has helped to create a 0-D PWS model.

2.4 BEHAVIOR OF THE DEVELOPED MODEL DURING ENGINE LOAD RESPONSE

The results were tested using engine 1-D model outputs for calibration with positive preliminary results. Moreover, the computation time of the engine load response decreased considerably; from several hours in a 1-D code to couple of minutes.

Figure 2 presents patterns of the 0-D model coefficients defined in the above equations as obtained representing the 1-D simulation results. Although the total PWS efficiency can reach a pole (discontinuity) during pressure recovery after a shock transient load change under some extraordinary off-design conditions – cf. Figure 2, the product $J.EX$ (introduced by equation 11) is almost constant except for at the start of the load change at 3.5 s.

FIGURE 2: Traces of PWS coefficients in relation to time after a shock load change at 3.5 s in the case of a PWS operated outside the design conditions; constant and low engine speed.

OBRAZÉK 2: Průběhy koeficientů po skokové změně zatížení motoru v čase 3,5 s v závislosti na čase; tlakový výměník provozován za nenávrhových podmínek; konstantní a nízké otáčky motoru

Just after the change of load, the accumulation capacity of the PWS makes itself felt, which causes a very short deviation from the continuity equation – cf. the negative peak of mass balance $del\ m$ between 3.5 and 3.7 s. This part should be excluded from the calibration range, which does not cause any substantial problems. Due to the extensive scavenging of the PWS, the trend of the coefficient λ_e (visible in Figure 2 after 10 s) declines from part load value of 5.7 to 1.1 at full load.

A significantly worse feature is the pole of total PWS efficiency at 9.5 s. This unstable behavior is typical for off-design operation of an uncontrolled PWS due to PWS speed or size deviations from the optimal design. The stability of the determined $J·EX$ product and the reasonable behavior of exhaust volumetric efficiency yield good results from the simplified model even in this case if regression representations of calibration coefficients are used. The physical reasons for the behavior close to 9.5 s are outside the scope of this paper. It may be stressed here only that the occurrence of such operation points is very important for the engine operation features. As there are no published studies covering these operating areas, the authors plan to further investigate the effect of the pole on engine operation during transient conditions in more detail.

Finding these unstable working areas is therefore another important contribution of the 0-D model for the development of the whole system.

3. CONCLUSION

The 0-D model of a PWS was developed and implemented into an existing close to real time model with standard turbocharger for engine transient simulation. The PWS model determines the mass flow rates of air delivered to the engine and engine exhaust mass flow rate. It also predicts compressed air temperature and internal EGR of the PWS. The 0-D model calibration coefficients, which may be found by experimentation or using higher level simulation, were defined. Using an example of engine transient response, an accumulation capacity of a PWS and unstable behavior of PWS total efficiency were demonstrated.

Surprisingly, the derivation of the 0-D algebraic model for a complex thermo-aerodynamic system of a PWS was more demanding than in the case of a differential model, especially if it uses ready modules (pipe, or flow spilt, etc.), as in the case of 1-D codes. Constructing this apparently simple model calls for a deeper understanding of thermodynamics and wave phenomena. Therefore, the physically transparent description of the processes inside the simulated system is desirable for the successful validation of the model. On the other hand, it has the advantage of rapid computational capability (30 s of transient response are computed within 2 minutes, i.e. 200 times faster than using a 1-D code), which makes it suitable for RT predictive models (see for instance [1]) trained by physical or numerical experiments.

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LIST OF NOTATIONS AND ABBREVIATIONS

A	flow area [m ²]
c_p	specific heat at constant pressure [J/kg.K]
D	diameter [m]
L	length [m]
$\dot{m}$	mass flow rate [kg/s]
n	speed [1/min]
p	pressure [Pa]
$\dot{Q}$	heat transfer [J/s]
r	gas constant [J/kg.K]
T	temperature [K]
t	time [s]
$\dot{V}$	volume flow rate [m ³ /s]
η	efficiency [1]
κ	isentropic exponent for fresh air [1]
κ_e	isentropic exponent for exhaust gas [1]
λ	scavenging coefficient [1]
π	pressure ratio [1]
ρ	density [kg/m ³]
σ	mass fraction [1]

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