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MODELING OF GAS FLOW THROUGH VALVE (CRV)

MODELOWANIE PRZEPŁYWU GAZU PRZEZ ZAWÓR (RZR)

Abstract

The aim of this work is to calculate the effective area of a compressor recirculation valve. Analytical solution and numerical modeling for compressible flow through a valve will be presented. Critical parameters of flow will be determined. Numerical modeling will be performed using ANSYS FLUENT. The results of the numerical solution will be compared with the analytical solutions.

Keywords: *compressible flow, CFD, CRV valve, choking*

Streszczenie

Celem artykułu jest obliczenie powierzchni efektywnej zaworu recyrkulacji. Przedstawione zostaną rozwiązywania analityczne oraz modelowanie numeryczne przepływu czynnika ściśliwego przez zawór. Wyznaczone zostaną parametry krytyczne przepływu. Modelowanie numeryczne przeprowadzone będzie za pomocą programu ANSYS FLUENT. Wyniki z rozwiązania numerycznego porównane zostaną z rozwiązaniami analitycznymi.

Słowa kluczowe: *przepływy ściśliwe, modelowanie numeryczne płynów, zawór recyrkulacyjny, zadławienie*

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1. Introduction

In many cases turbo charged gasoline engines will have a recirculation valve across the compressor, this type of solution is shown in Fig. 1. The recirculation valve is operated externally and ways of controlling will not be discussed in this paper. The recirculation valve opens especially but not only in two conditions: first when the clutch on the car engine is pressed and the engine is working without a load, and the second one when the gas pedal is pressed and there is a very high load on the engine caused by increasing acceleration. These two cases generate rapid pressure changes. An open recirculation valve prevents the compressor from going into a surge and avoids large pressure spikes that can occur between the compressor and the throttle [8–9].

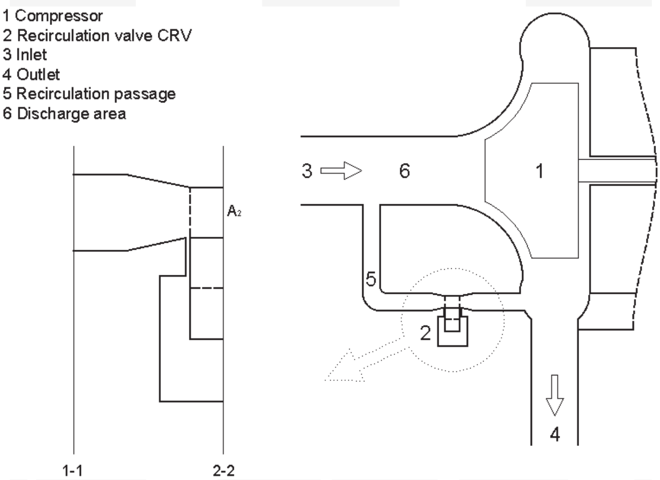

Fig. 1. Compressor with CRV

The compressor operation in the surge region is unstable because the pressure ratio is too large for the small flow rate, causing a reversal flow and pressure drop. The temporary pressure drop allows the flow to reestablish itself but this may lead to compressor longevity issues. Also this causes the turbo speed to drop more rapidly. During the design of CRV its effective area should be calculated.

There are some analytical solutions and one of them will be presented in this article. Simultaneously, CFD modeling will be performed. The aim of this article is to compare these two solutions.

2. Isentropic flow

For reversible adiabatic process [1–6]:

$$p\rho^{-k} = \text{const} \tag{1}$$

the Bernoulli equation of steady flow can be written as:

$$\int \frac{dp}{\rho} + \frac{w^2}{2} = \text{const} \tag{2}$$

The potential energy term is assumed to be negligible. Evaluating the result at two points (1 and 2) presented in Fig. 1 for the isentropic process:

$$\frac{p_2}{\rho_2}\frac{k}{(k-1)}\left[\left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1\right] + \frac{1}{2}\left(w_1^2 - w_2^2\right) = 0 \tag{3}$$

Application of equation (3) to the compressible flow of a body through a fluid. Allow point 1 to be the stagnation point, and take point 2 as an upstream reference point (ahead of the influence region of the body). Because $w_1 = 0$, then:

$$p_1 = p_2\left[1 + \frac{(k-1)}{2k}\frac{\rho_2 w_2^2}{p_2}\right]^{\frac{k}{k-1}} \tag{4}$$

where:

- p – pressure of fluid at the inlet,
- p_2 – pressure of fluid at the outlet,
- w_1 – fluid velocity at the inlet,
- w_2 – fluid velocity at the outlet.

Flows through the valve can be taken as isentropic because flow transit time is so small that any friction is ignored. Noting that the w_1 is small reduces equation (3) to the following expression for velocity w_2:

$$w_2 = \sqrt{\frac{2k}{k-1}\frac{p_1}{\rho_1}\left[1 - \left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]} \tag{5}$$

The condition of the flow in the valve depends on the value of the pressure after the valve in respect to the critical pressure. The critical pressure can be calculated from the following equation:

$$p_{cr} = p_1\left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{6}$$

where p_1 is the absolute pressure at the inlet and it is equal to the sum of static and dynamic pressures:

$$p_1 = p_0 + \frac{1}{2}\rho_1 w_1^2 \tag{7}$$

For $p_2 \geq p_{cr}$ compressible mass flow rate has the form:

$$\dot{m} = w_2 A_2 \rho_2 \tag{8}$$

By applying equation (5) into equation (8) the following equation can be written:

$$\dot{m} = A_2 \sqrt{\frac{2k}{k-1}\frac{p_1}{\rho_1}\rho_2^2\left[1-\left(\frac{p_2}{p_1}\right)^{\frac{k-1}{k}}\right]} = A_2 \sqrt{\frac{2k}{k-1}p_1\rho_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}}-\left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right]\left(\frac{p_1}{p_2}\right)^{\frac{2}{k}}\frac{\rho_2^{\;2}}{\rho_1^{\;2}}} \tag{9}$$

Assuming the equation (1) the relation between pressure at the inlet and outlet takes the form:

$$p_1\rho_1^{\;-k} = p_2\rho_2^{\;-k} \tag{10}$$

The equation for the mass flow (8) can be written as:

$$\dot{m} = A_2 \sqrt{\frac{2k}{k-1}p_1\rho_1\left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}}-\left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right]} \tag{11}$$

When $p_2 = p_{cr}$ the mass flow is equal to:

$$\dot{m} = w_{cr} A_2 \rho_{cr} \tag{12}$$

Applying equation (6) mass flow has the form:

$$\dot{m} = A_2 w_{cr} \rho_1 \left(\frac{2}{k+1}\right)^{\frac{1}{k-1}} \tag{13}$$

After simplification equation (13) takes the form:

$$\dot{m} = A_2 \sqrt{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}} p_1\rho_1} \tag{14}$$

In the design of the recirculation valve (CRV) it is necessary to calculate the minimum size of the valve area A_2. The Aabove formulas (9) and (14) depend on the ratio of the pressures p_1 and p_2. Shapes of valves in case of narrowing the cross-section in the direction of gas flow does not have a significant importance on the speed and mass flow rate of the gas even if taking into account the friction forces on its walls.

3. Calculation of minimum area of valve

The ratio of critical pressure p_{cr} divided by pressure at the inlet p_1 has the form:

$$r_{cr} = \frac{p_{cr}}{p_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \tag{15}$$

assuming air as fluid $k = 1.4$ and $r_{cr} = 0.5283$.

For $r > r_{cr}$ the minimum area of the valve can be calculated from:

$$A_2 = \frac{\dot{m}}{\sqrt{\frac{2k}{k-1} p_1 \rho_1 \left[\left(\frac{p_2}{p_1}\right)^{\frac{2}{k}} - \left(\frac{p_2}{p_1}\right)^{\frac{k+1}{k}}\right]}} \tag{16}$$

For $r \le r_{cr}$:

$$A_2 = \frac{\dot{m}}{\sqrt{k\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}} p_1 \rho_1}} \tag{17}$$

Table 1

Calculated minimum area of valve

$\dot{m}$ [kg/s]	T1 = 293 [K]		T1 = 373 [K]		T1 = 473 [K]	
	p_1/p_0	A_2 [mm^2]	p_1/p_0	A_2 [mm^2]	p_1/p_0	A_2 [mm^2]
0.04	1.33	**144.465**	1.42	**144.936**	1.52	**147.185**
0.05	1.45	**155.286**	1.58	**155.012**	1.68	**161.752**
0.06	1.6	**162.202**	1.75	**164.504**	1.85	**174.572**
0.07	1.75	**170.099**	1.92	**174.231**	2.05	**184.398**
0.08	1.9	**178.318**	2.1	**183.119**	2.35	**187.525**
0.09	2.06	**185.771**	2.35	**187.342**	2.7	**189.848**
0.1	2.255	**190.815**	2.65	**189.88**	3.17	**188.996**
0.11	2.53	**191.606**	3	**191.478**	3.65	**190.297**
0.12	2.85	**191.746**	3.5	**189.14**	4.3	**188.693**

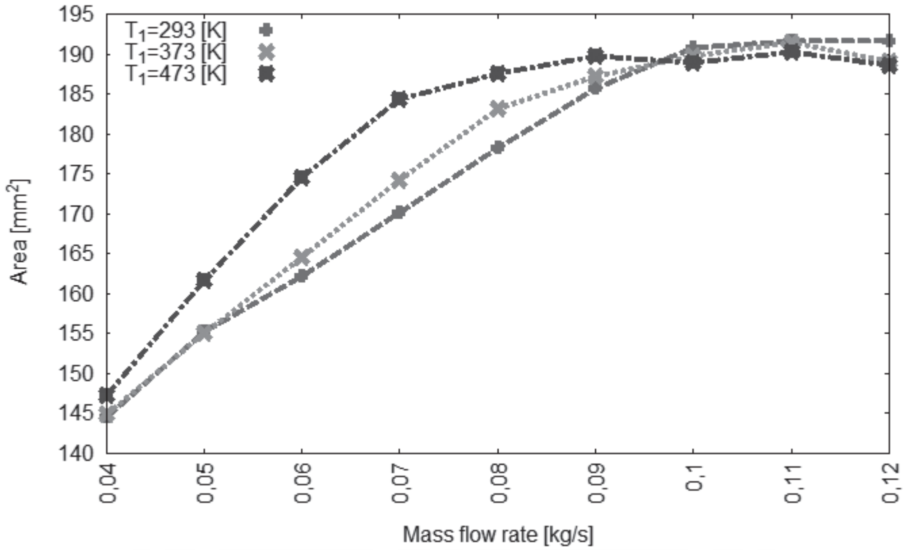

Fig. 2. Minimum area of CRV in the function of mass flow rate

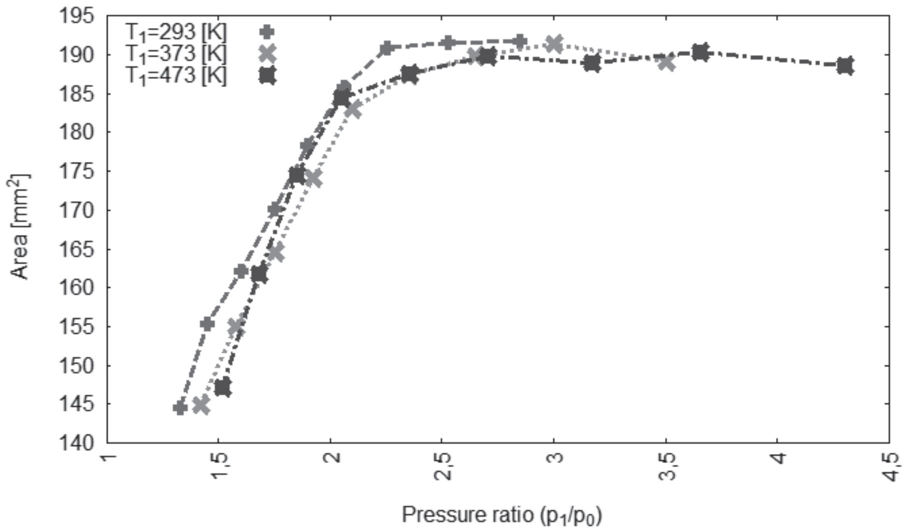

Fig. 3. Minimum area of CRV in the function of pressure ratio

Based on the equation (16) and (17) the minimum area in the function of mass flow rate $\dot{m}$ is presented in Fig. 2 and the minimum area in the function of pressure ratio p_1/p_0 is shown in Fig. 3, where p_0 is the ambient pressure.

Based on Fig. 2 and Fig. 3 the value of minimal area of the CRV valve was calculated $A_2 = 195$ [mm^2].

4. CFD modeling

Numerical modeling of compressible flow through the CRV valve in a steady state has been conducted [10]. A geometrical model was created from a cylinder which narrows in the length of 20 [mm] in the center. The following dimensions were assumed: the length of 260 [mm] diameter of the cylinder in the center 15.8 [mm]. The minimum cross section area was equal to 195 [mm^2]. The model is presented in Fig. 4.

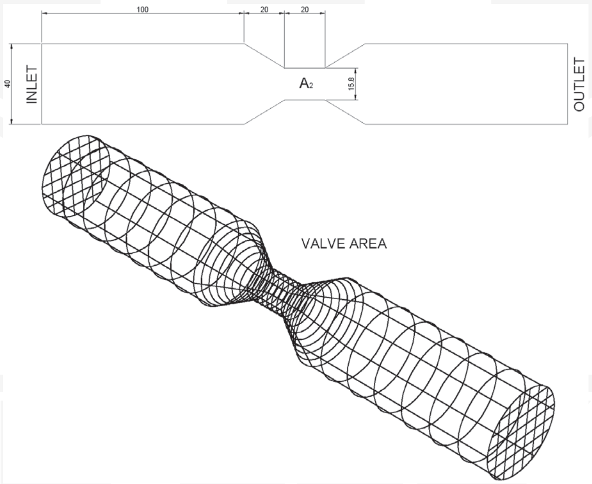

Fig. 4. 2D and 3D model with dimensions and assumed boundary conditions

The model was divided into finite elements tri tetra type of variable size decreasing ten times at the beginning of the narrowing part.

Calculations were carried out based on its density using the equations of mass, momentum and energy conservation. The flow was turbulent and k-ε model with extended walls (enhanced wall fn) was used. For the flowing fluid the air was set at the temperature of 293 [K]; 373 [K]; 473 [K]. The ideal gas law was chosen. Boundary condition for INLET (mass-flow -inlet) was defined by mass flow of 0.04 [kg/s] to 0.12 [kg/s]. At the OUTLET constant operating pressure of 1 [bar] was set. For the adiabatic exponent for air $k = 1.4$ was defined. The pressure p_1 at inlet area was calculated.

The accuracy of the numerical solution is high. The maximum relative error is about 5%. To simplify the problem a two-dimensional model was created with the same sizes in the x and y axis as in the three-dimensional model. The model was divided into finite elements of quad pave type with resizable size decreasing ten times at the beginning of the narrowing area. The computational model and boundary conditions were the same as in the three-dimensional modeling.

Table 2

Comparison of exact values and calculated by CFD

$\dot{m}$ [kg/s]	T1 = 293 [K]			T1 = 373 [K]			T1 = 473 [K]		
	p_{1exact} [Pa]	p_{1CFD} [Pa]	$\sigma = \frac{p_{1exact} - p_{1CFD}}{p_{1exact}} \cdot 100$ [%]	p_{1exact} [Pa]	p_{1CFD} [Pa]	$\sigma = \frac{p_{1exact} - p_{1CFD}}{p_{1exact}} \cdot 100$ [%]	p_{1exact} [Pa]	p_{1CFD} [Pa]	$\sigma = \frac{p_{1exact} - p_{1CFD}}{p_{1exact}} \cdot 100$ [%]
0.04	133000	136025	**2.274436**	142000	141200	**0.56338**	142000	141245	**0.53169**
0.05	145000	149500	**3.103448**	158000	155345	**1.68038**	158000	156690	**0.82911**
0.06	160000	154905	**3.184375**	175000	178245	**1.85429**	175000	174256	**0.42514**
0.07	175000	171900	**1.771429**	192000	194234	**1.16354**	192000	193400	**0.72917**
0.08	190000	185856	**2.181053**	210000	208345	**0.7881**	210000	205789	**2.00524**
0.09	206000	203436	**1.24466**	235000	238543	**1.50766**	235000	228432	**2.79489**
0.1	225500	227478	**0.877162**	265000	266703	**0.64264**	265000	279532	**5.48377**
0.11	253000	248300	**1.857708**	300000	304356	**1.452**	300000	298643	**0.45233**
0.12	285000	291097	**2.139298**	350000	348254	**0.49886**	350000	353575	**1.02143**

5. Conclusions

The method to calculate the effective-minimum area of a compressor recirculation valve (CRV) was presented. Additionally, numerical modeling of compressible flow through the CRV valve in a steady state has been conducted. The created numerical model was presented. The results of the numerical solution were compared with the analytical solutions. The accuracy of the numerical solution is high. The maximum relative error is about 5%.

This type of analysis is applicable in the automotive industry.

References

[1] Anderson J.D., *Modern Compressible Flow with Historical Perspective*, 3rd ed., McGraw-Hill, New York 2003.

[2] Çengel Y.A., Boles M.A., *Thermodynamics: An Engineering Approach*, 4th ed., McGraw-Hill, New York 2002.

[3] Hodge B.K., Koenig K., *Compressible Fluid Dynamics with Personal Computer Applications*, Prentice Hall, Englewood Cliffs, New York 1995.

[4] John J.E.A., *Gas Dynamics*, 2nd ed., Prentice Hall, Englewood Cliffs, New York 1984.

[5] Moran M.J., Shapiro H.N., *Fundamentals of Engineering Thermodynamics*, 4th ed., John Wiley & Sons, New York 2000.

[6] Oosthuizen P.H., Carscallen W.E., *Compressible Fluid Flow*, McGraw-Hill, New York 1997.

[7] Zucrow M.J., Hoffman J.D., *Gas Dynamics, Volumes I and II*, John Wiley & Sons, New York 1976.

[7] Allard A., *Turbocharging and Supercharging*, Cambridge, Patrick Stevens Limited, England 1982.

[8] Gorla Rama, Khan Aijaz, *Turbomachinery Design and Theory*, Marcel Dekker, New York 2003.

[9] ANSYS® Fluent, *15.0, User's manual*, ANSYS, Inc.