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EXPERIMENTAL VALIDATION OF THEORETICAL CALCULATION AND NUMERICAL SIMULATION FOR OPTIMIZATION OF DIVERS'S BREATHING APPARATUS

Tamara STANCIU *
tamara.stanciu@navy.ro

Andrei SCUPI **
andrei.scupi@gmail.com

Dumitru DINU **
dumitru.dinu@gmail.com

** Diving Center, Constanţa, Romania*
*** Constanţa Maritime University, Romania*

ABSTRACT

In order to optimize the breathing apparatus in the open circuit for divers, theoretical calculus and numerical simulation of resistances specific to the potential flow of gas through the studied circuit were made. Respiratory gas flow simulation through three constructive versions of the second stage pressure reducer intake mechanism was done after modeling the respiratory air circuit through the two main restrictors: the first variable (between the seat and the piston) and the second fixed (the hole in the cylindrical piston). The results regarding the theoretical calculation and numerical simulation have been validated by experimental testing of two of the studied models. Experimental measurements were made on a tester at the Diving Center of Constanta's Hyperbaric Laboratory. The volume flow rate of supplied respiratory gas was recorded, together with the inspire depression that opens the mechanism, until the maximum flow rate for each constructive version. After validating the results of the theoretical calculation and numerical simulation on the two models, the conclusion is the same: the resistance decreases if the geometry of the cylindrical hole in the piston (the second fixed restrictor) changes in a conical hole.

KEYWORDS:

Breathing apparatus, theoretical calculation, numerical simulation, experimental validation

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1. Introduction

In a previous paper we checked the variation in the volume flow of gas in the pressure regulator circuit for divers in three constructive versions of the gas intake mechanism by Computational Fluid Dynamics. We made the geometric modeling of the three versions. After meshing the obtained fluid models, the required flow conditions were set. The mass flow, the density at the outlet of the pressure reduction mechanism and the fluid velocities were calculated. For the same flow conditions and the same inspire depression, we determined the external resistances in the three geometric versions of the gas intake mechanism. We have concluded that the best shape of the inlet ports in the medium pressure hose in the variable restrictor is the original one. For the piston, the recommended airflow direction port is the conical section.

The results obtained from theoretical calculations and numerical simulation were experimentally tested on a professional test bench for two design models of the gas direction port (the second fixed restrictor) in the cylindrical pressure reducer piston: 1st Version and 2nd Version. We propose to compare the variance of external resistances to inspire for 1st Version and 2nd Version, resistances determined by theoretical calculation and numerical simulation (Scupi, 2015) with the same variation resulting from the experimental measurements.

2. Theoretical calculation

In the studied pneumatic circuit the points of interest are at the variable restrictor 1 due to the x-displacement of the cylinder and the fixed restrictor 2, the hole in the cylinder. The two restrictors are stuck. The medium pressure hose gas is propagated by the expansion passes through the variable surface restrictor 1 between the seat and the clamp and through the A-section thin wall cylinder. Flowing through the two restrictors at this moment is done with critical flow. The pressure is reduced by restrictors to the value of the outside pressure. In restrictor 1, which is a Laval nozzle, the flow is turbulent for a short time and then becomes stationary. The orifice in the thin-walled cylinder is a nozzle (restrictor 2) (Stanciu, 2018).

The ratio of downstream and upstream pressures is lower than the critical air ratio and in this case the theoretical mass flow is the critical one and it is constant on the pneumatic circuit after the restrictor (Carafoli, 1984):

$$\dot{m}_{t1} = \alpha_1 \left(\frac{2}{k+1}\right)^{\frac{1}{2}\left(\frac{k+1}{k-1}\right)} \sqrt{k} A \frac{p_m}{\sqrt{RT_m}} \quad (1)$$

$\alpha_1 = 1$ (for the theoretical flow) – the flow coefficient (Petcu, 1970)

K = 1.4 – air adiabatic coefficient

X = 0.7 [mm] – opening of the restrictor 1

$R = 287[j/kgK]$ – gas constant

A – normal cross-sectional area of the hole with the cylinder (Stanciu, 2018).

p_m = 8.5 [bar] – midle pressure

T_m = 293 [K] – constant stagnation temperature (20 ºC)

For both versions there is the same mass flow rate:

$$\dot{m}_{t1} = 1\left(\frac{2}{1.4+1}\right)^{\frac{1}{2}\left(\frac{1.4+1}{1.4-1}\right)} \sqrt{287} \cdot 10.9 \cdot \quad (2)$$

$$\cdot 10^{-6} \frac{8.5 \cdot 10^5}{\sqrt{287 \cdot 293}} = 0.022[\mathrm{kg/s}]$$

For the inspire depression $\Delta p = 5[cmH_2O]$ we have:

$$R_{E1} = \frac{\Delta p_1}{\dot{V}_1} = \frac{5}{4.64} = 1.08[\frac{cmH_2O}{l/s}] \quad (3)$$

$$R_{E2} = \frac{\Delta p_2}{\dot{V}_2} = \frac{5}{5} = 1[\frac{cmH_2O}{l/s}] \quad (4)$$

Table no. 1

The external inspire resistances calculated theoretically for the two geometric versions of piston inlet mechanism

Model of the mechanism	$Q_m\left[\frac{kg}{s}\right]$	$\rho\left[kg/m^3\right]$ exit of the cylinder	$\dot{V}\left[l/s\right]$ exit of the fixed restrictor	$\Delta p = 5[cmH_2O]$	$R_E\left[\frac{cmH_2O}{l/s}\right]$
1st Version	0.013	2.8	4.64	5	1.08
2nd Version	0.011	2.2	5	5	1

$$\frac{R_{E2}}{R_{E1}} = \frac{1}{1.08} = 0.92 \qquad (5)$$

In case of 2nd Version with conical hole, the strength decreases by 8%.

3. Numerical simulation

The pneumatic circuit for the second stage regulator is composed of compressed air that is reduced to first stage at an average pressure of 8-9 [bar] above the outside pressure (in this case $p_e = p_N = 101300[Pa] \approx 1[bar]$).

The numerical simulation described in a previous paper was made on three constructive versions, of which we chose for experimental validation 1st Version and 2nd Version. The 3rd Version induces a high external resistance, we gave up the model.

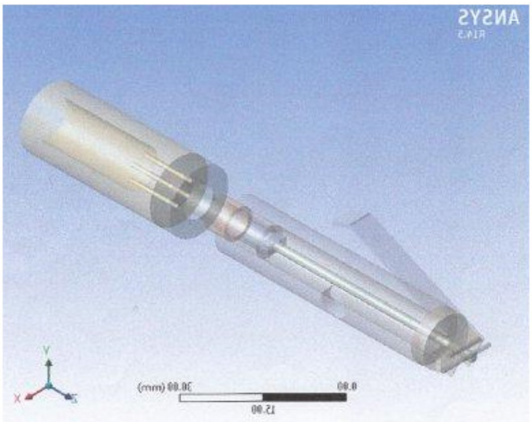

Figure no. 1. 1st Version of pneumatic mechanism with cylindrical hole
(Stanciu, 2017)

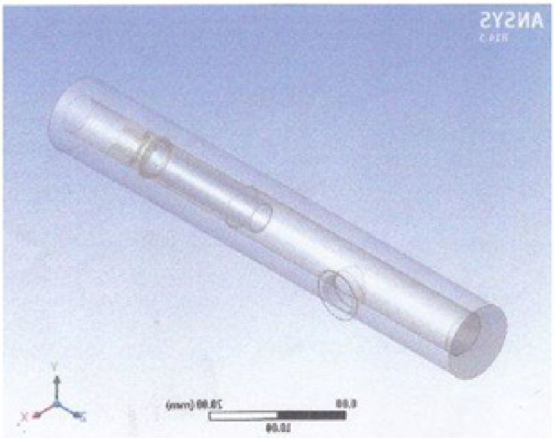

Figure no. 2. 2nd Version of pneumatic mechanism with conical hole
(Stanciu, 2017)

External resistances were calculated with the simplest formula:

$$R_E = \frac{\Delta p}{\dot{V}}\left[\frac{cmH_2O}{l/s}\right] \qquad (6)$$

The inspire depression was imposed: to $\Delta p = 5[cmH_2O]$

The mass flow at the exit from the second restrictor results from numerical simulation.

Density for the two versions is also the result of numerical simulation.

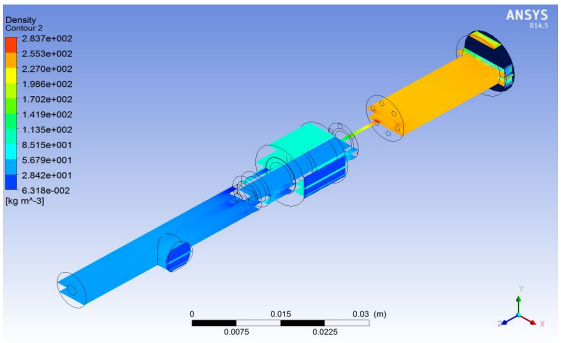

Figure no. 3. Density distribution in two pl parallel to pl xz for 1st Version (Stanciu, 2017)

The volume flow rate at the exit of the cylinder was calculated:

$$\dot{V} = Q_m / \rho [m^3 / s] \quad (7)$$

The external resistances obtained with Ansys Fluent CFD are in Table no. 2.

Referring to the two resistances for 1st Version (cylindrical orifice) and 2nd Version (conical hole) to $\Delta p = 5[cmH_2O]$, we determined the percentage of decrease of inspire resistance in 2nd Version.

Table no. 2

The numerical calculated inspire resistances for the two geometric versions of piston intake mechanism

Model of the mechanism	$Q_m\left[\frac{kg}{s}\right]$	$\rho\left[kg/m^3\right]$ exit of the cylinder	$\dot{V}[l/s]$ exit of the fixed restrictor	$\Delta p = 5[cmH_2O]$	$R_E\left[\frac{cmH_2O}{l/s}\right]$
1st Version	15,225*10⁻³	2,842	5,357	5	0,933
2nd Version	13,433*10⁻³	2,118	6,342	5	0,788

$$\frac{R_{E2}}{R_{E1}} = \frac{0.788}{0.933} = 0.84 \quad (8)$$

In the case of 2nd Version with the conical hole, the resistance decreases by 16 %.

4. Experimental procedures

Experimental determinations were conducted in the Diving Center of the Hyperbaric Laboratory using a Scuba Tools tester. The stand is a professional unit, as shown in Figure no. 4.

At the unit stand was coupled the high pressure supply from a first stage assembled to a pressurized cylinder at 150 [bar]. Another cylinder with first stage supply flow rate (7-10 bar) for vacuum, the Venturi pumps from the top of the stand. The tester simultaneously measures the opening pressure on a Magnehelic gauge, Venturi depression [cmH2O], airflow rate [l/min] on the flow meter, middle pressure [bar] on the gauge prior to the second stage and low pressure [bar] for the Venturi pump at the appropriate gauge. The second stage pressure regulator has been coupled to the flow meter. The bottle was opened and four sets of simultaneous numerical values were read on the measuring instruments until the maximum flow rate delivered by the apparatus stabilized. We modified the piston hole from cylindrical (Figure no. 5) to conical (Figure no. 6) 2nd Version and the measurements resumed.

Figure no. 4. Testing the pressure regulator with Scuba Tools unit stand (Stanciu, 2018)

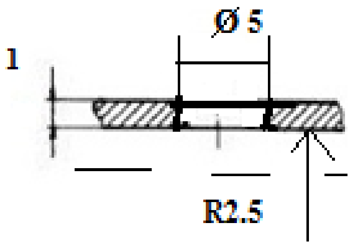

Figure no. 5. Cilindrical hole 1st Version (Stanciu, 2018)

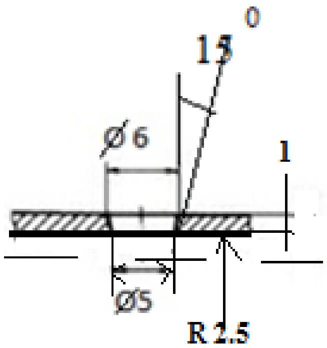

Figure no. 6. Conical hole 2nd Version (Stanciu, 2018)

5. Experimental results

The results were recorded simultaneously, following the European scale.

Figure no. 7. Recording with flowmeter, Magnehelic gauge and medium pressure gauge

Read values are found in Table no. 3.

Table no. 3

The experimental results obtained by the measurements made on the tester for the two geometric versions of piston inlet mechanism (Stanciu, 2018)

Nr.	Model of the mechanism	Supply pressure [bar]	Medium pressure [bar]	Volumic rate flow		Inspire depressure [cmH_2O]
				[l/min]	[l/s]	
1	1st Version	150	8.8	320	5.333	5
			8.5	350	5.833	5
			8.5	400	6.667	6.5
			8.5	300	5	4.5
2	2nd Version	150	8	340	5.667	0.1
			8	420	7	3
			8.5	440	7.333	5
			8.5	480	8	6.5

We calculated the inspire resistances for each version for the depression that generated the maximum constant flow.

1st Version

a. For $\Delta p = 6.5[cmH_2O]$ maximum volumic rate flow delivery of the device is 6.7 [l/s].

$$R_{E1}=\frac{\Delta p_1}{\dot{V}_1}=\frac{6.5}{6.7}=0.97[\frac{cmH_2O}{l/s}] \quad (9)$$

b. For $\Delta p = 5[cmH_2O]$ maximum volumic rate flow deliveried of the device is 5.33 [l/s].

$$R_{E1}=\frac{\Delta p_1}{\dot{V}_1}=\frac{5}{5.833}=0.86[\frac{cmH_2O}{l/s}] \quad (10)$$

2nd Version

a. For $\Delta p = 6.5[cmH_2O]$ maximum volumic rate flow deliveried of the device is 8[l/s].

$$R_{E2}=\frac{\Delta p_2}{\dot{V}_2}=\frac{6.5}{8}=0.81[\frac{cmH_2O}{l/s}] \quad (11)$$

b. For $\Delta p = 5[cmH_2O]$ maximum volumic rate flow deliveried of the device is 7.67 [l/s].

$$R_{E2}=\frac{\Delta p_2}{\dot{V}_2}=\frac{5}{7.333}=0.68[\frac{cmH_2O}{l/s}] \quad (12)$$

We determine the variation in inspire resistance, in changing the geometry of the nozzle for the two depression values.

a. By referencing the two resistances for 1st Version (cylindrical orifice) and 2nd Version (conical hole) at $\Delta p = 6.5[cmH_2O]$, we determine the percentage of decrease of the resistance to inspire in 2nd Version.

$$\frac{R_{E2}}{R_{E1}}=\frac{0.81}{0.97}=0.83 \quad (13)$$

In case of 2nd Version with the conical hole, the resistance decreases by 17 %.

b. By referencing the two resistances for 1st Version (cylindrical orifice) and 2nd Version (conical hole) at $\Delta p = 5[cmH_2O]$, we determine the percentage of decrease of the resistance to inspire in 2nd Version.

$$\frac{R_{E2}}{R_{E1}}=\frac{0.68}{0.86}=0.79 \quad (14)$$

In case of 2nd Version with the conical hole, the resistance decreases by 21 %.

6. Conclusions

We have experimentally validated two methods (theoretical calculation and numerical simulation) for determination the external resistances at the compressible air flow through the pneumatic circuit of the second stage pressure reducer in 1st Version (cylindrical orifice) and 2nd Version (conical hole). We imposed the same flowing conditions:

- Average supply pressure 8.5 [bar]
- Output pressure 1 [bar]
- Opening depression $\Delta p = 5[cmH_2O]$
- Opening of the restrictor 1, x = 0.7 [mm]

Table no. 4

Variation of external resistances determined by the three methods

Method	Theoretical calculation	Numerical simulation	Experimental Procedures
% decrease of external resistance for the conical opening towards the cylindrical orifice	8 %	16 %	21 %

In the case of experimental determinations, the maximum constant flow rate is obtained at $\Delta p = 6.5[cmH_2O]$. In this case, the percentage of reduction of the expiratory resistance is 17 %, close to what results from CFD Ansys Fluent.

In the theoretical calculation the reduction is smaller, but in this case the approximations made have influenced the determined percentage.

Actual mass flow is lower, is directly proportional to $\alpha_1 \neq 1$.

The calculated resistance is inversely proportional to the flow, so higher.

Actual temperature is not constant.

In all cases, the shape of the fixed restrictor (hole) influences the change of inspire resistance.

We can conclude that by changing the nozzle from the cylindrical in conical, to the dimensions in Figure no. 6, resistance decreases by 16-17 %.

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