Reduction of the Tip Clearance Effect in a Radial Impulse Turbine for Wave Energy Conversion

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Abstract

Radial impulse turbines could be a good option in Oscillating Water Column systems to extract energy from waves. A previous work made by the authors showed that the tip clearance plays a fundamental role in the flow pattern inside the turbine, so seeing the relation between tip clearance's size and turbine performance is a very interesting issue from the point of view of designing this kind of turbomachine.

Evaluate the tip clearance influence on the turbine performance is the purpose of this work. The geometry proposed in (1) with different tip clearance's size and the numerical model described in (2) have been used. The performance comparison shows that changing the tip clearance's size in a 4% of blade span reduces the stationary turbine efficiency up to 9%. Although, the influence of the tip clearance in this geometry is not the same in both flow directions, the efficiency reduction is more pronounced when the flow is centripetal.

From a point of view of manufacturing, to decrease the tip clearance has a limit. Therefore a new turbine design that minimizes the tip clearance effects is proposed. The numerical model is used to evaluate the performance of the new turbine.

Keywords: Fluent, OWC, radial impulse turbine, tip flow.

Nomenclature

A_R	Reference area
b	Rotor blade height
$C_A = \dfrac{\Delta P_o}{\frac{1}{2}\rho\left(v_R^2 + u_R^2\right)}$	Input coefficient.
$C_T = \dfrac{T_O}{\frac{1}{2}\rho\left(v_R^2 + u_R^2\right)A_R r_R}$	Torque coefficient
ΔP_o	Total pressure drop between settling chamber and atmosphere
q	Flow rate
r_R	Mean radius
T_o	Output mechanical torque
T	Period
$u_R=\omega\, r_R$	Circumferential velocity at r_R
$v_R=q/2\pi r_R b$	Mean radial velocity at r_R
η	Stationary efficiency
ρ	Air density
$\varphi= v_R/u_R$	Flow coefficient
ω	Rotational speed

1 Introduction

One of the most widely studied systems to extract energy from the waves is the named Oscillating Water Column (OWC). The devices based in this system extract energy from waves by means a capture chamber and a turbine. The chamber converts the wave motion into pneumatical energy by means the movement of the water free surface inside the chamber, which generates a bi-directional air flow. The chamber is connected to a duct where is placed the turbine. When the free surface in the chamber moves up due to the waves the air inside the chamber is pushed out through the turbine, and, when the free surface moves down the air is sucked into the chamber through the turbine. These processes are called exhalation and inhalation respectively and they are produced cyclically.

Under these special conditions of bi-directional flow is better to use the called auto-rectifying or bi–directional turbines than the classical turbines.

These turbines keep the rotation sense independently of the air flow sense.

There are different types of auto-rectifying turbines. The most used and studied is the Wells turbine, which is the one traditionally installed in OWC systems. It has been studied deeply by many authors (3-4), so its performance is relatively well-known. The alternative to the Wells turbine are the impulse turbines. There are two types: axial one and radial one. The axial impulse turbine has been object of many studies too (5-7), but only a little bibliography can be found about the radial impulse turbine (8-10).

This work is based on the radial impulse turbine, which is less used than the Wells and the axial impulse turbines because of its lower efficiency. However, the radial impulse turbine has several advantages such as no oscillating axial thrust and it is cheaper to manufacture due to its simple geometry.

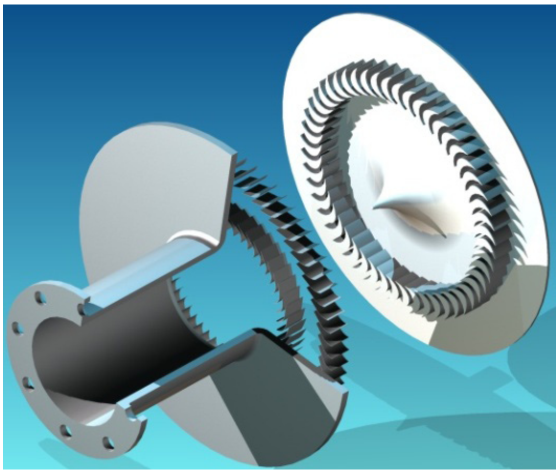

Figure 1. Radial impulse turbine.

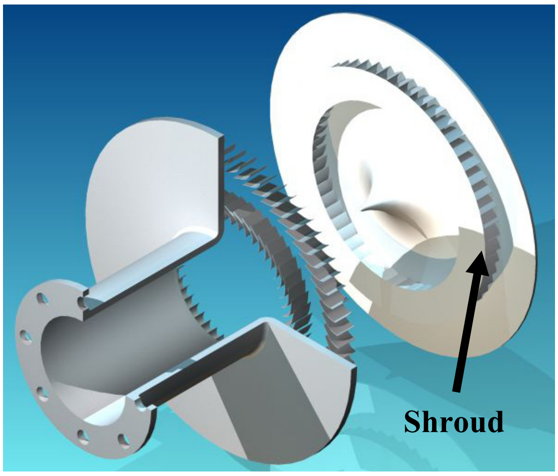

Figure 2. Radial impulse turbine with shroud.

In a previous work of the authors it was showed that the tip clearance size has an important influence on the performance because causes an efficiency reduction (11). There are studies with the same trend for Wells turbines (12) and axial impulse turbines (13), but only in a few of them a solution is proposed to reduce or eliminate the influence of the tip clearance. The same solution is adopted for a Wells turbine (14) and an axial impulse turbine (15). A plate is placed at the tip of each blade to minimize the tip flow and these results show that the maximum stationary efficiency increases around 5%. In this work, a shroud at the blades' tip has been included to close the rotor (Figure 2), so that the tip flow is eliminated. The study has been carried out by a means a numerical model previously validated (2).

2 Numerical model

Hereinafter a brief summary of the numerical model characteristics is shown.

The turbine´s geometry was extracted from (8). This turbine is equipped with two rows of guide vanes, inner (IGV) and outer (OGV), and a rotor, and they are positioned as Fig. 3 shows. The main dimensions and characteristics can be found in Fig. 3 and Table 1. The Reynolds number, based on blade chord, reaches a value of 6×10^4, as is mentioned in (8). This configuration will be called M8Z from now on.

In the initial geometry the blade high is 43 mm, so the tip clearance is 1 mm (8). The inclusion of the shroud at the blades' tip modifies these values because of its own width. The shroud's width is 0.5 mm, the size of the tip clearance and turbine width have been keeping constant in 1mm and 44 mm respectively, so the blade height in the new geometry is 42.5 mm, 0.5 mm shorter. The shroud is extended from radius 190 mm to 251 mm.

The geometry with shroud will be called M8ZR from now on.

	Blade number	Chord length	Solidity	Setting angle	Turbine width
IGV	52	50 mm	2.29	25°	44 mm
Rotor	51	54 mm	2.02	19°/ 36°	44 mm
OGV	73	50 mm	2.28	25°	44 mm

Table 1. M8Z turbine's geometric characteristics. Case 1, (8).

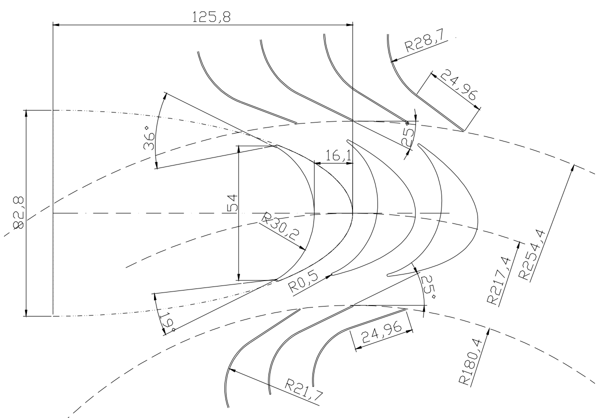

Fig. 3. Blades and vanes profiles of the M8Z geometry

In order to reduce the calculation power needed, simulations were carried out on a periodic domain which is 1/17 of the whole machine, Figure 4.

Changing the vanes number has been necessary to get a periodic domain, so in the numerical model the number of vanes and blades are 51/51/68 in the IGV/Rotor/OGV respectively. A 2-D study made to know the influence of this change show that the difference in the coefficients C_A and C_T are less than 3% in the whole flow coefficient range. Therefore it has been considered this change in the vane's number is possible.

The numerical model geometry was created using INVENTOR®. The mesh was built with GAMBIT® 2.3 and it is composed of 3500k irregular hexaedrical cells. Modeling the tip clearance local flow is one of the main points, so the mesh of the zone close to the tip clearance has a high density (ten cells per millimeter in rotation axis direction).

The real operation of the turbine is unsteady since the magnitude and the direction of the air flow changes in the time. Nevertheless, the time scale of the waves is much greater than the residence time through the turbine, so a quasi-steady behavior of the turbine can be supposed. Therefore, the boundary conditions will remain constant in each simulation (Figure 4), although the model is unsteady since the relative position between the rotor and the guide vanes varies with time. The rotor rotates at 24.52 rd/s and to simulate its movement Sliding Moving Mesh (SSM) technique was applied. There are two interfaces located between IGV–rotor and OGV–rotor.

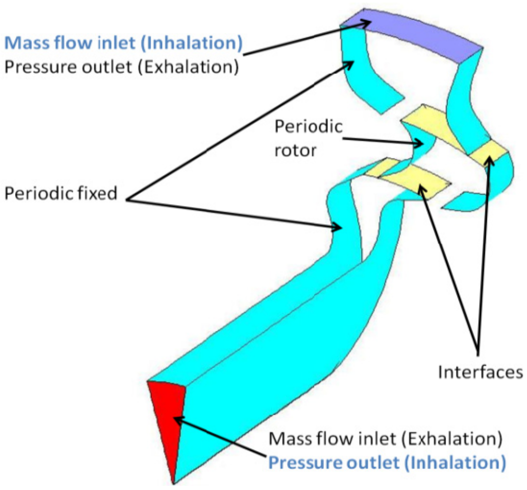

Figure 4. Numerical domain and main boundary conditions

The commercial code Fluent® was used to solve the incompressible fluid conservation equations by using a segregated solver. The turbulence model chosen has been the k-ε realizable. The time dependent term is approximated with a second order implicit scheme. The highest order MUSCL scheme has been used for convective terms´ discretization and the classical central differences approximations for diffusive terms. The pressure-velocity coupling is performed through the SIMPLE algorithm.

The time step chosen to quantify the relative movement of the grid was 10-4s. Hence, one blade of the rotor covers 2560 different steps in a revolution. Residuals were in four orders of magnitude as is usual for turbomachines.

3 Results

The overall performance of the turbine has been evaluated by dimensionless coefficients: flow coefficient φ, torque coefficient C_T, input coefficient C_A and efficiency η.

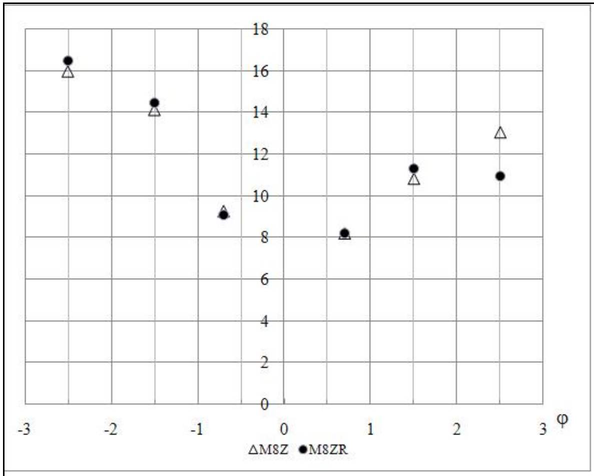

Figure 5. Input coefficient, C_A.

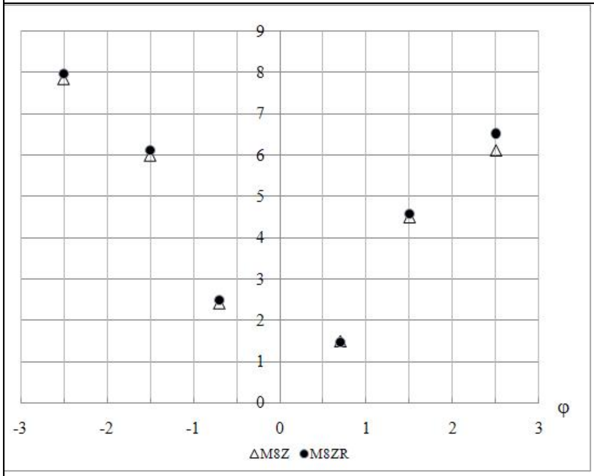

Figure 6. Torque coefficient, C_T.

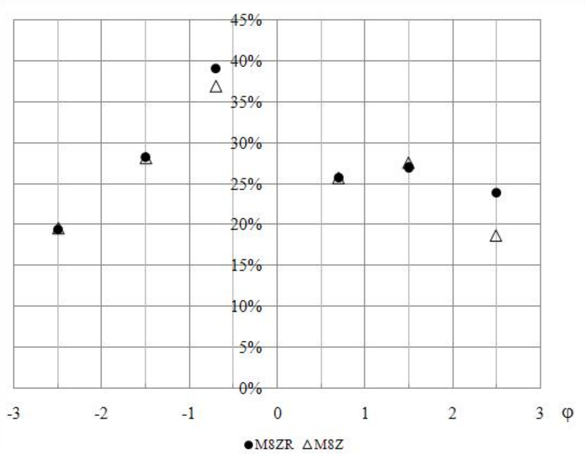

Figure 7. Stationary efficiency.

Figure 5 and Figure 6 represent the coefficients C_A and C_T respectively. In the figures it can be seen that the geometry with shroud, M8ZR, reduces notably the losses at high flow coefficients during exhalation because the turbine needs less energy, C_A, to obtain more torque, C_T. In inhalation the torque produced is higher but the energy used is also higher.

The turbine's efficiency is shown in Figure 7. The new geometry, M8ZR, achieves at least the same efficiency than M8Z, better at high flow coefficients during exhalation and at low flow coefficients in inhalation. The improvement achieved reach a 5% in exhalation at $\varphi = 2.5$, mainly due to the losses reduction.

The difference between inhalation and exhalation is the relative importance of the losses associated to the tip flow respect with friction losses due to the shroud. During exhalation the losses caused by the tip clearance are higher than in inhalation (11), so the improvement achieved with the shroud is higher.

Total pressure contours are used to analyze the improvement achieved due to the shroud. These contours show the presence or not of instabilities inside the rotor, so it can be seen if the shroud prevents the disturbance generation caused by the tip clearance.

The surfaces where the total pressure contours are evaluated are placed as is shown in Figure 8. One of them is in the inner part of the rotor (C1) and the second one in the outer part (C2). These surfaces cut the rotor and define the cross-section of the rotor channel. At the bottom of these sections is placed the hub and at the top is placed the tip clearance and the shroud.

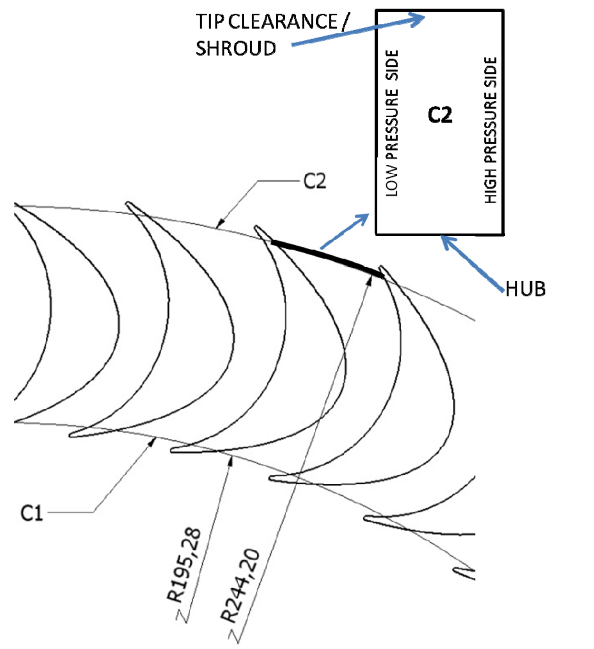

Figure 8. Reference sections C1 and C2.

The disturbances created by the tip clearance are transmitted downstream. Those created during inhalation can be seen on section C1, and those generated during exhalation on section C2. Figure 9 shows the contours on section C2 during exhalation and Figure 10 shows the contours on C1 during inhalation. In both figures there are three different flow coefficients represented for each geometry, M8Z and M8ZR. So it is possible to see the evolution of the improvement achieved with the shroud.

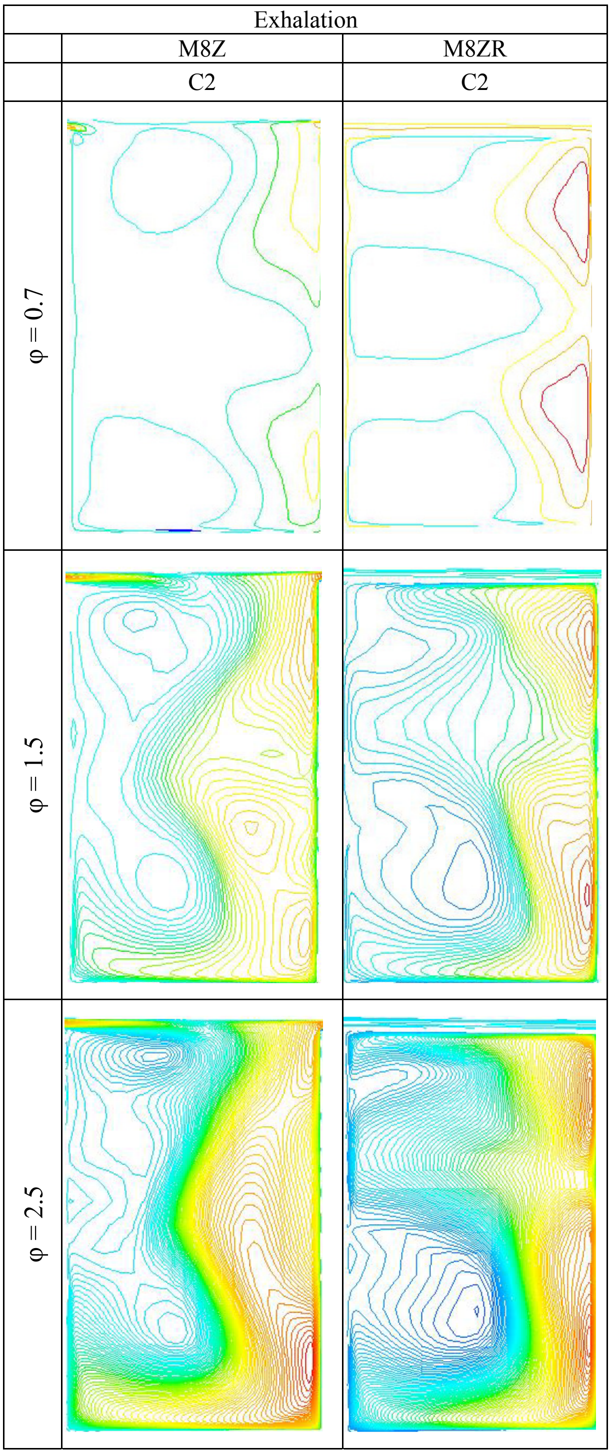

Figure 9. Total pressure contours during exhalation at section C2 (10Pa/isoline).

In Figure 9 the total pressure contours on C2 for M8Z during exhalation are shown on the left. There are two zones clearly separated in the contours. On the right side of the contours there is a high total pressure zone where the flow follows the blade pressure side.

However, there is a low total pressure zone on the left part of the rotor's channel caused by the flow detachment produced on the concave surface of the blade. This flow detachment, caused because the cross–section in exhalation is divergent (11), is strengthen by the instability created by the tip flow which appears in the upper part of these contours. This instability keeps increasing as flow coefficient increases in M8Z. However, due to the shroud, in the M8ZR there is not tip flow and its losses do not exist. In this case, M8ZR, the flow pattern close to the shroud shows that there are losses but they are friction losses which also increase when the flow coefficient rises. From Figure 5 it is possible to conclude that the losses associated to friction in M8ZR are lower than losses associated to the tip flow in M8Z, so the improvement reached due to shroud increases as flow coefficient rises.

At the bottom part of the contours other instability appears. This is created upstream by the vane's tip clearance (11) and is also an important source of losses. The shroud is placed in the opposite side of the machine and it has not effect on this instability.

Figure 10 shows the contours on section C2 during inhalation. Here the effect of the shroud is clearer because it is not hidden by others phenomena as flow detachment or shock losses. In contours of M8Z it can be seen clearly the tip flow through the tip clearance. The instability generated increases as flow coefficient grows, but it is limited to the zone close to the tip clearance. In the rest of the cross–section the flow pattern is much more organized than exhalation Figure 9. This is due to in inhalation the flow detachment on the blade' surface is very small because the cross–section is convergent.

In M8ZR the tip flow disappears and the flow pattern is much more organized than M8Z. So the losses due to the tip flow are zero, however, in Figure 7 the efficiency is the same approximately for the two geometries. This is due to the friction losses caused by the shroud, which counteract the benefit achieved by the elimination of the tip flow effects.

4 Conclusions

Tip clearance is one of the main problems in turbomachinery performance because it is the origin of several instabilities.

This work analyzes the performance improvement achieved in a radial impulse turbine by means of placing a shroud at the blades' tip. The inclusion of this shroud prevents the tip flow in the rotor blades. Therefore the rotor's performance improves especially in exhalation where the torque obtained increases and the losses are lower. Its effect during inhalation is minor because the improvement achieved with the shroud is counteracted by the friction losses caused by it.

The turbine efficiency increases due to the rotor's performance improvement, especially at high flow coefficients during exhalation, up to 5%. During inhalation the improvement achieved with the shroud is only appreciable at low flow coefficients because the friction losses caused by the shroud are not enough high to counteract the positive effect. The efficiency during inhalation improves 3%.

The rotor' shroud provide a low cost solution for the problems arisen from the tip clearance. This solution can be also applied to reduce the losses in the guide vanes. Nowadays the authors are working in a model which incorporates a shroud in the guide vanes. The former results are promising.

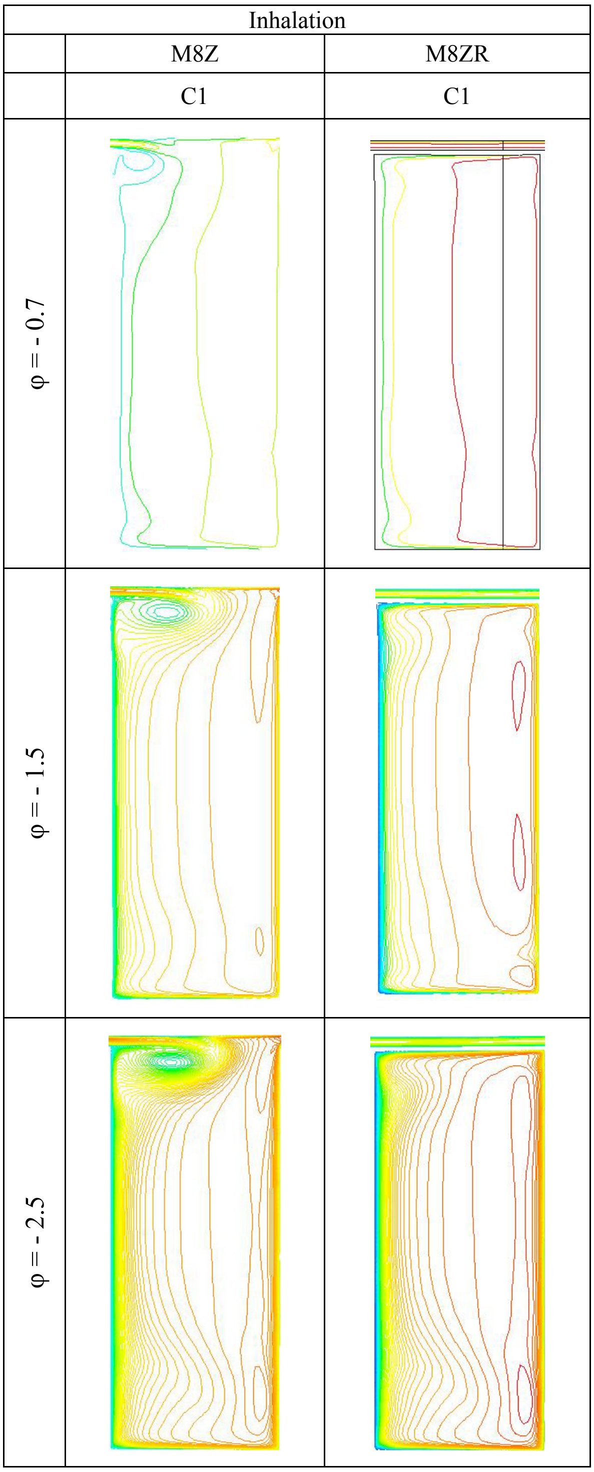

Figure 10. Total pressure contours during inhalation at section C1 (30Pa/isoline).

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