Noise suppression of exhausted air flow at the tube outlet

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Abstract: - This feasibility study deals with use of flow numerical simulation for effectivity prediction of various designs of noise silencer, applied on the air exhaust of large volume flow and complicated by sticky parts contained in exhausted air. Important noise source are here thin metallic walls, excited by pressure pulsations of turbulent flow and a strong free flow with turbulent outer boundary, directed up in the free surroundings. The results demonstrate that several simple modifications decrease the flow kinetic energy at one quarter approx. of the actual value, therefore a specific noise level decreasing can be expected, too.

Key-Words: Turbulent air flow, flow numerical simulation, aerodynamic noise, noise suppression, flow dissipation

1 Introduction

The paper summarizes and extends the results received during own simulations and measuring [1], [2], [3], [4]. In the observed system of exhausted air from the large painting plant, it is flowing very high air volume, over 50 m^3/s approx. The scheme of the geometry is visible in several figures below, for instance Fig. 3 with pressure field inside. Generally, the exhaust fan is installed at the bottom of the cylindrical tube, followed by the rectangular horizontal case (a length of 5 m approx.) and finished by the next cylindrical exhaust up in the surroundings.

Another Fig. 10 with outside free flow shows that the outgoing quick free flow of initial velocity 50 m/s approx. reaches the height some tens of meters, where the turbulent area at the boundary between the flow and surroundings is the source of unpleasant noise.

In addition, the exhausted air contains some sticky parts, which must be periodically cleaned to prevent the obstruction of ducts.

The solution of various complex cases in this feasibility study uses the method of flow numerical simulation. It allows relative quickly and simply to get suitable results which can be used for initial estimation of individual parameters of the flow field, mainly influencing the acoustic field, as flow velocity, flow kinetic energy, turbulent kinetic energy, pressure, etc. Such a solution, which seems to be the best for the intended purpose of noise reduction, can be solved later more precisely, using a complicated and long-time acoustic model.

From the preliminary results of flow numerical simulations and realized field measuring it is clear that the primary noise source is not only the flowing air, but mechanical oscillations of large and thin metallic walls, too, excited by pulsating flow field. Therefore, it is presumable that long-time acoustic simulation in compressible volume of fluid gives not any corresponding results. The available code (state of the year 2014) does not allow the solution of the mutual interaction of the flow with vibrating wall and resulting acoustic pressures in the fluid surroundings.

Due to the large extent of received results, the presented flow fields sometimes do not contain actual scales of parameters of the flow field (velocity, etc.). Actual values are not important here, but important are characteristic isolines of the tested parameters in flow fields, only, their trends, etc. For better understanding - the arrangements of the used scales are standard, the highest value is red and the lowest value is blue, according to the wave lengths in the spectrum of visible light.

2 Possible noise sources

For step by step solution of the problem, it is necessary to classify possible sources of noise from the observed device. Of course, there really exist some combinations of individual kinds of noise sources.

2.1 Forced noise generation

Here, it means pressure fluctuations generated by the rotating blade wheel of the fan at the entry into the system. This case is not the matter of this paper. For illustration, only, the field around the flap rotating at 23 Hz was solved as a simple 2D model. The result of an unsteady solution is presented in the graph in Fig. 1, where the double frequency of 46 Hz is well visible. After the start from zero, the system needs two revolutions to get a standard periodical pressure field.

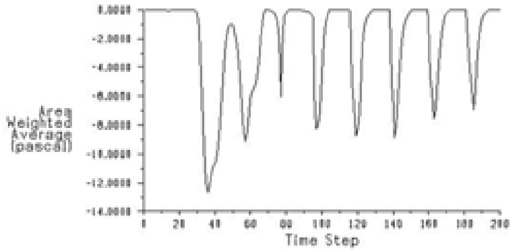

Fig. 1: Pressure fluctuations from rotating flap

Typical field of higher and lower pressure around the rotating flap is presented in Fig. 2. The next serial in Fig. 3 presents the variable pressure field inside the actual system, excited by the rotating flap, installed at the bottom. For the information, only two flap positions of many solved flap positions are presented here.

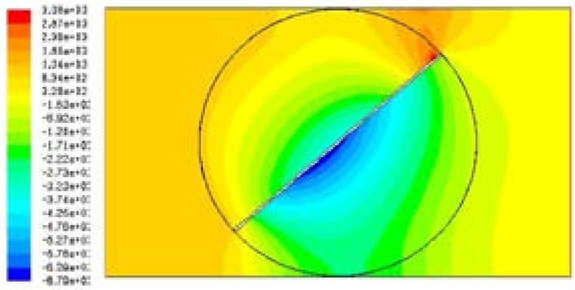

Fig. 2: Pressure field around rotating flap

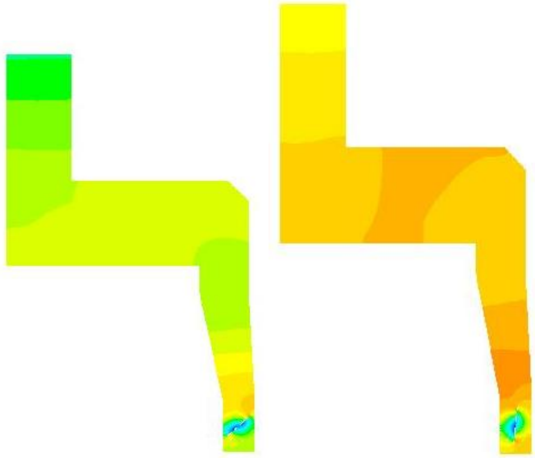

Fig. 3: Pressure field at two different flap positions

2.2 Mechanical vibrations

Mechanical vibrations of thin and flexible metallic walls are actuated by pressure fluctuations in the flow inside. Fig. 4 presents velocity field in vertical symmetry plane with strong flow separation, contraction and backflow in the system of sharp changes of the flow directions. Fig. 5 presents the same as the pressure field.

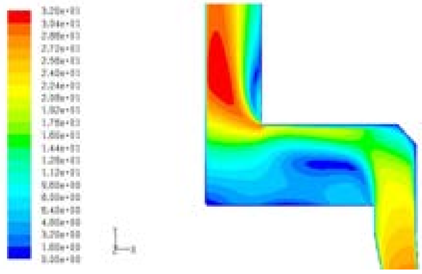

Fig. 4: Velocity field in vertical symmetry plane

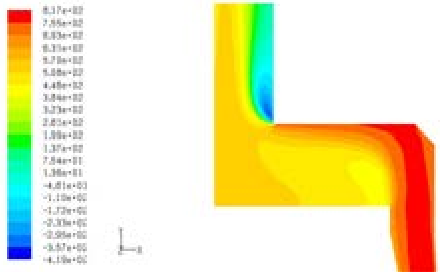

Fig. 5: Pressure field in vertical symmetry plane

Fig. 6 demonstrates deformation of flexible rectangular wall, fixed at two long edges and free on two short edges and loaded simply by constant pressure from the bottom.

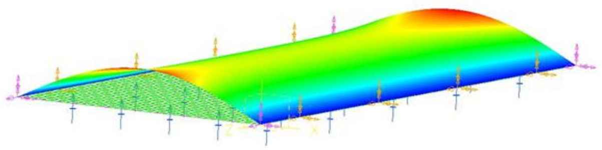

Fig. 6: Deformation of flexible wall loaded by pressure

The real situation is much more complicated. Walls of the horizontal rectangular case are made from thin metallic sheets; therefore, the above mentioned pressure pulsations induce secondary vibrations of walls. From many records of pressure fluctuations in various points of the wall, only one result is presented here.

Fig. 7: Pressure field at the side wall of the case

Fig. 7 presents detailed pressure isolines on the vertical side wall in any random time step of unsteady solution, Fig. 8 presents the record of pressure fluctuations in the middle of this wall during unsteady solution with time step of 2 ms and Fig. 9 presents the frequency analysis of recorded signal in this point.

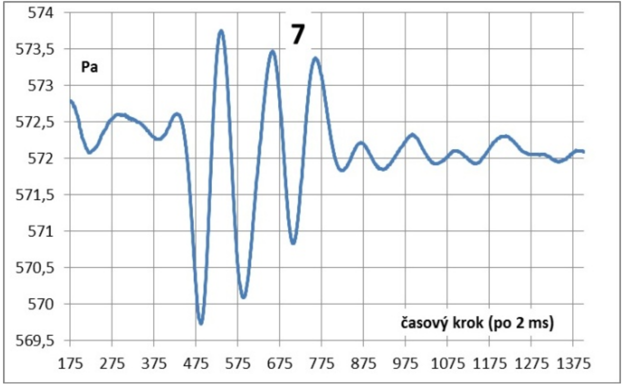

Fig. 8 Recorded pressure fluctuations in the middle of the side wall, time step of 2 ms

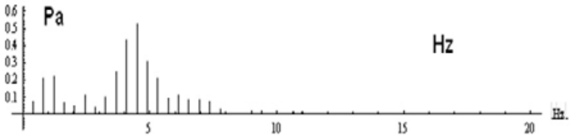

Fig. 9: Frequency analysis of the previous record

The maximum acoustic pressure was detected at the frequency of 4.5 Hz - this value corresponds very well with real field measuring on the actual equipment [2].

The next numerical simulation proved that such pulsations were going to zero after installing the fluent shape of the exhaust tubing [1], [3] because the rectangular horizontal case is no need. For instance, the velocity field in Fig. 10 is not as disturbed as in Fig. 4 above. The reversal pressure field, corresponding to Fig. 5 above, is not presented here again. Finally, the only straight vertical tube was designed because it is not any operational reason for the former S-shaped tubing – see the par. 3 below.

Fig. 10: Velocity field in smooth S-shape

2.3 Aerodynamic noise

The next important part of the total noise level is the aerodynamic noise of the free flow outside.

Illustrative results of the flow numerical simulation are presented in Fig. 11 as velocity field. Added large surroundings (up) represent the very long dissipation of the exhausting flow. Inside the horizontal chamber (former silencer or filter?), the flow is very turbulent, due to the sharp bends of the flow (see also Fig. 4 and Fig. 5). Turbulences, pressure pulsations and turbulent boundary between the free flow and the surroundings are the sources of aerodynamic noise. The walls are made from thin metallic sheets, which are vibrating and the noise from it is spreading into the surroundings, too, see Par. 2.2. Here at, it is well known that for noise damping, the walls should be made massive, not thin and flexible. Sharp direction changes induce the flow turbulences; the result is an uneven velocity profile, high turbulence values and pressure fluctuations.

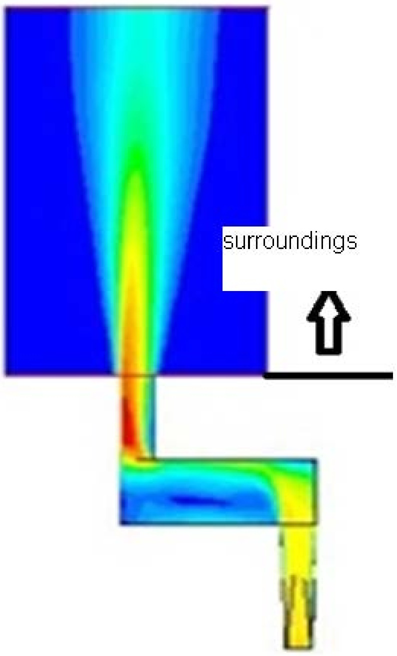

Fig. 11: Velocity field of the total original exhaust

2.4 Sticky parts in exhausted air

A next important problem are sticky parts in exhausted air, which must be cleaned periodically to prevent the piping obstruction. Therefore, the cross sections of the exhaust and silencer must be large enough, but it is not good for the noise attenuation. Maybe, it should be better the cleaning (blasting) of the silencer walls by dry ice [5]. In such a case, in the channel it is inserting the narrow blasting nozzle with an extension, only, so that the channels can be narrow and more effective for noise silencing.

Remark: It is not clear, why in the exhausted air are included sticky parts, when at the exhaust inlet standard separators of sticky parts are installed (should be installed), as water shower wall, sleeve or drum filter etc., which could catch all sticky parts. Probably due any unknown error in the plant operation, the filters are obstructed quickly, therefore they are removed and the remaining empty filter case acts as a noisy drum.

For the suppression of the above mentioned high noise level, the standard labyrinth silencer was designed, containing several shaped channels. The exhaust is oriented down to the roof surface and/or in the horizontal circumferential direction. The total bend angle of the flow is 810°. Due to the noise damping, the silencer is designed as a squared body simply made from thick walls (boards). The design made from thin sheets can be circular, too, but the damping effect of such a system of thin sheets is lower and could be the subject to operational vibrations, too.

Fig. 12 presents the resulting turbulence field in a 1/4 model of such a silencer, when two symmetry planes were used – central inlet from below, outlet in the surroundings in horizontal centrifugal direction. The channels of the labyrinth are designed large enough to allow to enter into and to clean sticky layers on the inner surfaces. So the system must be demountable, for instance with a removable upper part.

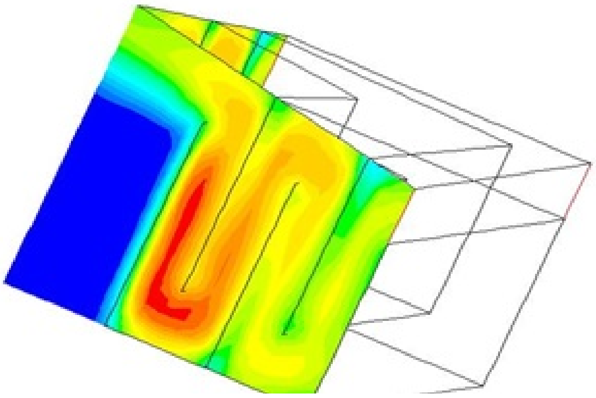

Fig. 12: Turbulence field in 1/4 model of labyrinth silencer

3 Problem Solution

Several solved cases are presented here and discussed their advantages and disadvantages.

3.1 First realization

It is a pity that only a part of the above mentioned labyrinth from Fig. 12 was realized, with a total bend angle of 360°, only, i.e. with an open exhaust up, see Fig. 13.

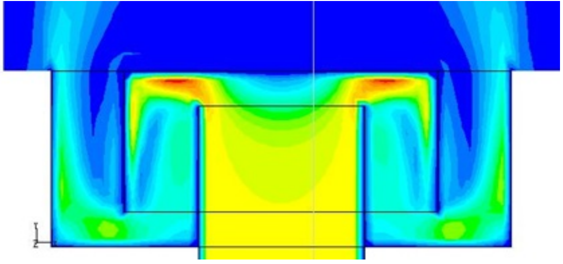

Fig. 13: Velocity field inside - wrong realization

Reputedly, it was realized due to a mass reduction of the equipment on the non-bearing roof and due to misgiving from the possible back suction of the exhausted air, contaminated by volatile parts, into the hall. The resulting velocity field of such a free flow is presented in Fig. 14. For a more expressive display, the reduced scale is used (it is not displayed the area of maximum velocity at the outlet from the silencer).

This design, not created after recommendations from the previous flow numerical simulations, does not bring any improvement because the high column of the free outflow (similar to Fig. 11) remains the source of noise. The additional partial covering of the exhaust orifice after Fig. 15 is operating as a narrowed nozzle, only, i.e. it increases the flow velocity, therefore the velocity field image is very similar as above, the free flow range is long and the decreasing of aerodynamic noise level is none.

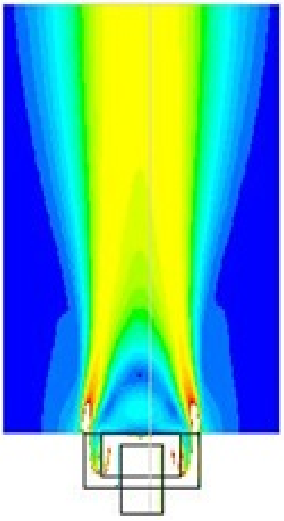

Fig. 14: Velocity field outside (suppressed scale)

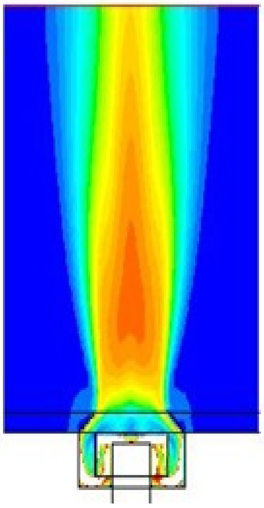

Fig. 15: Partial mouth covering – not any effect

3.2 Outlet turned down

Fig. 16 presents a slightly modified original design of labyrinth from Fig. 12 it fulfils well the requirements for an effective damping of aerodynamic noise. The exiting flow is turned down and is spreading along the flat roof, where the drifted sticky parts, not caught sooner on the inner partitions, are caught on the roof surface. Here is not any intensive „column“ of turbulent free air flow up, as in Fig. 14, etc., which is a significant source of the aerodynamic noise.

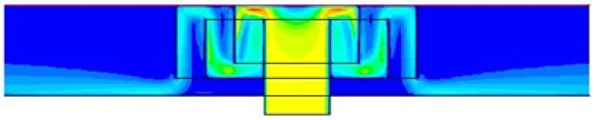

Fig. 16: Velocity field – labyrinth outlet directed down

The dynamic pressure of this outflow on the roof surface after Fig. 17 presents a quick fading out of the exhausted flow.

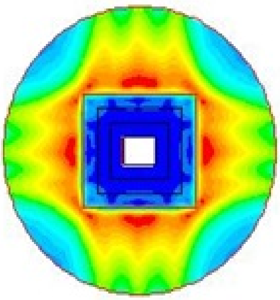

Fig. 17: Dynamic pressure at the roof surface

Further, there are presented two next shape modifications, without any next important influence on the global image of the flow field. In Fig. 18, the outer wall is shortened on one half, to reduce the weight of the equipment a little. The last flow bend in the labyrinth is realized, the main flow down and then along the roof remains. Fig. 19 presents the influence of the next low partition set-off on the roof to prevent eventually the flow spreading along the roof surface in the direction to the near-by air suction into the hall.

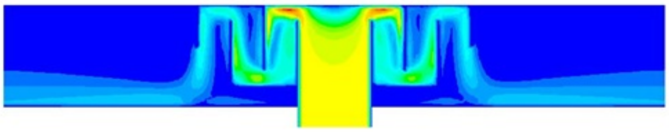

Fig. 18: Velocity field – shorter outer wall

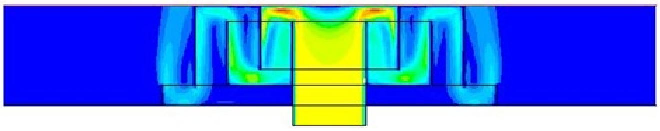

Fig. 19: Velocity field – the added low and set-off partition

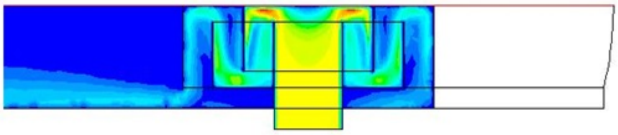

Fig. 20: Velocity field – the silencer just along the skylight

Fig. 20 simulates the possible flow field deformation when the silencer is placed just at the wall of the roof skylight. It has not an important influence on both flow through the silencer and air exhaust, too. In the ground plan, the flow outflow is constricted after Fig. 21 on a certain, not important part of the circumference, only.

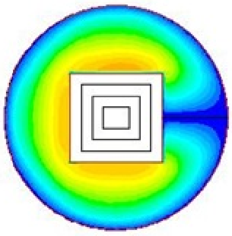

Fig. 21: Velocity field from Fig. 20 in the ground plan

The last Fig. 22 shows the modification with a classical slotted silencer at the outlet. It is effective, well accessible from the roof, simply demountable and cleanable, namely without operation interruption.

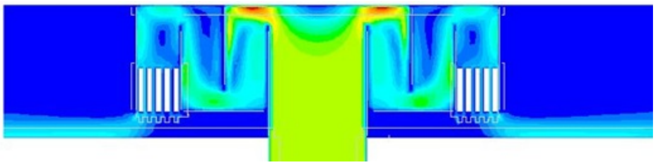

Fig. 22: Added classical slotted silencer

Summary:

The main parameters influencing the level of the aerodynamic noise of exhausted air (velocity, flow kinetic energy, turbulent kinetic energy) are presented in the summary Tab. 1.

Tab. 1: Overview of flow parameters influencing the noise level

exhaust		up	down	down+side
flow	kg/s	100%	100%	100%
outflow velocity	m/s	100%	47%	58%
turbul. kin. energy	m^2/s^2	100%	24%	40%
flow kinet. energy	J	100%	22%	34%

In general, for outlet directed down, the values are of tens of percent lower, compared with former outlet up, all under the condition of the same air flow. It is evident that the realized partial solution after Par. 3.1 is not good.

Remark: Simulating the acoustic field (levels of acoustic pressure etc.) needs much more effort and time of the solution. But in this feasibility study it is not important to evaluate real noise levels before and after design changes, but to show the way to the improvement. Here the presented values of the flow field have sure the influence on the noise level in general, it is possible to judge that using the presented modifications, the noise level is decreasing.

3.3 Influence of surrounding solids

In the previous Par. 3.1 and 3.2, the single exhaust with silencer was solved, without ambient objects. Here is tested and evaluated the influence of the surrounding solids on the air flow exhaust from the silencer body. Fig. 23 presents the actual situation - exhaust from the high chimney and suction under the low shed, with many skylights around [6].

Fig. 23: Situation on the roof – exhaust, suction, skylights

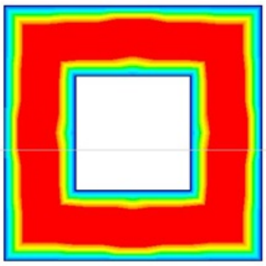

Fig. 24: Simplified (constant) velocity field at the silencer outlet

For this case, the outflow field was simplified after Fig. 24 – here it is not any influence of inner flow in the silencer, see Par. 3.1 and Par. 3.2, i.e. in the whole outlet cross section the velocity is constant.

The next Fig. 25 shows the velocity field in the vicinity of the exhaust in the ground plan 0.1 m over the roof plane – its range is relative short.

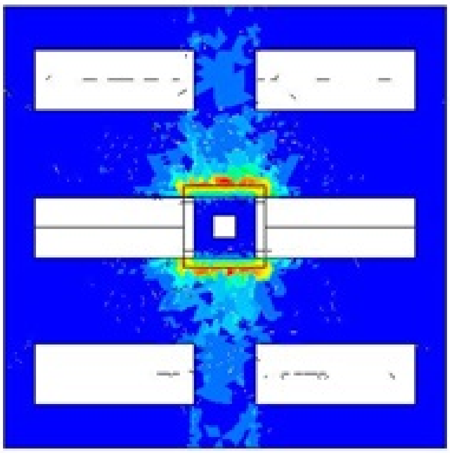

Fig. 25: Velocity field in the outlet vicinity

The checking Fig. 26 and Fig. 27 present velocity field on the roof in vertical cross sections (at different velocity scales!) as information, only. The vertical velocity is very low, maximum of 0.2 m/s, only. The horizontal velocity is higher, the maximum of 10 m/s is in the outlet orifice, but with an increasing circumference of the flow, the value is decreasing quickly.

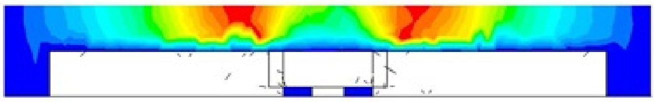

Fig. 26: Vertical velocity (max. 0.2 m/s)

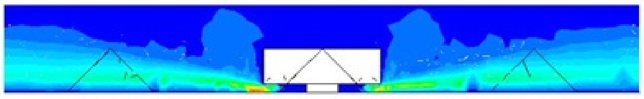

Fig. 27: Horizontal velocity (max. 10 m/s)

4 Contamination in suction

As stated in the Par. 3, the exhausted air, directed down to the roof, is fading out quickly at a short distance from the outlet cross section. Additionally, the system of skylights has two important effects - the quick flow fading out and some noise dispersion, too.

But it remains one important doubt whether the fresh air flowing back into the hall through the suction device nearby is not contaminated by exhausted air. Therefore, the next model of flow of contaminated air was solved here.

The impurity concentration is simply simulated as air volumes of different temperatures – the exhaust temperature is of 50 K higher than the temperature in the surroundings (and in the suction, too), the resulting temperature of the mixture corresponds to the impurities concentration. A typical result is presented in Fig. 28 – the area of maximum temperature (= impurity) is situated around the silencer outlet on the roof and quickly is decreasing along the roof in all horizontal directions.

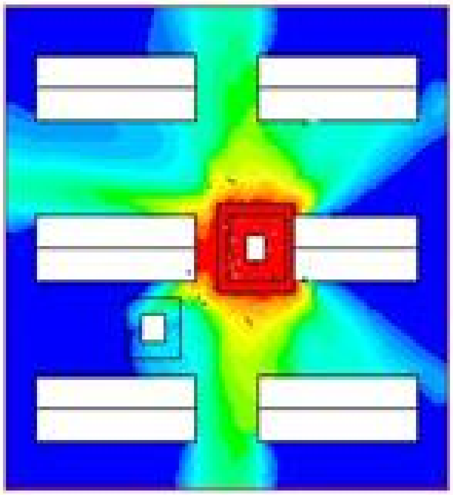

Fig. 28: Air impurity modeled as temperature

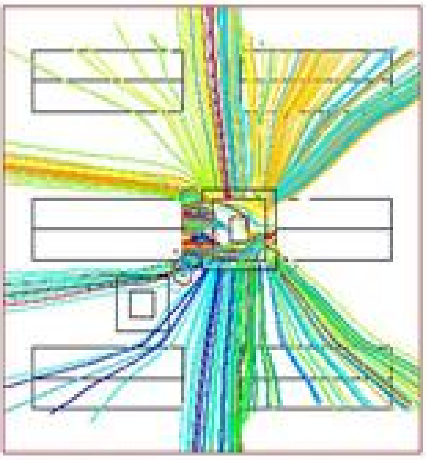

Fig. 29: Streamlines from the exhaust, suction off

Fig. 29 shows streamlines for the operational case when the suction is off - some streamlines from the silencer outlet are flowing around the suction body. In the exhaust, there is some source concentration of impurities (for the next comparison it is defined here as 100%).

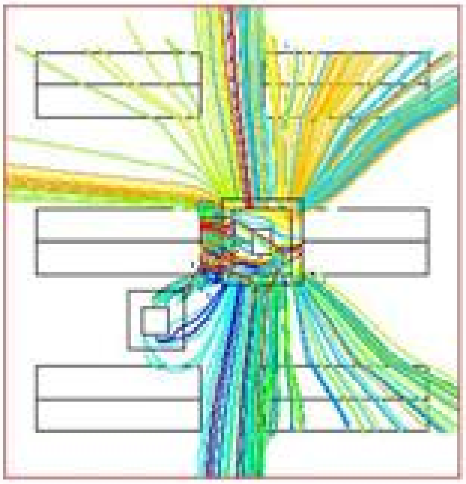

Fig. 30: Streamlines from the exhaust, suction on

In Fig. 30, the suction is on, some streamlines from the silencer exhaust are drawn into the suction, but as calculated from mass and energy balances at outlet/inlet, the concentration of impurities in the suction is 7.5%, only, of the outlet concentration.

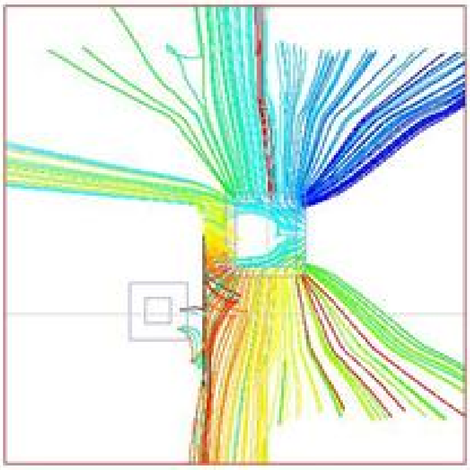

Fig. 31: Streamlines in the ground plan

In Fig. 31, there is inserted a low partition between exhaust and suction, some streamlines are turning up and the direct (short-circuit) flow is reduced; the impurities concentration in the suction is 1%, only, of the concentration in the exhaust. The temperature (= impurities) field of this case is in Fig. 32, for comparison with Fig. 28.

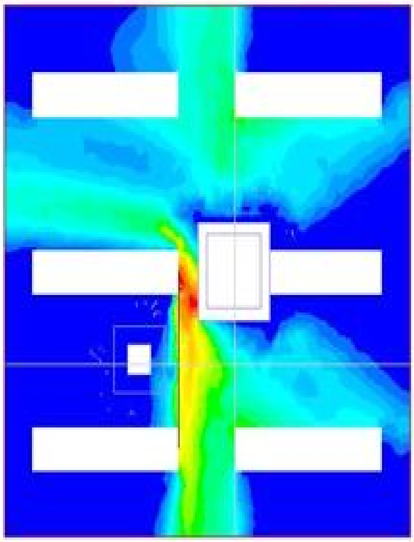

Fig. 32: Impurities blocked by partition

The warmer (= impure) air is flowing up; in the next Fig. 33, there are visible local maxima of temperature at the upper boundary surface of the solved volume.

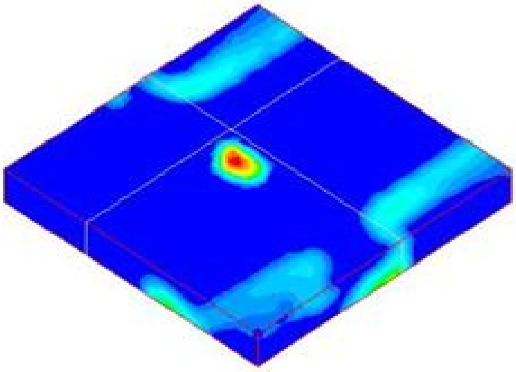

Fig. 32: Local maxima temperature, turned up by the inserted partition

5 Conclusion

After the removing of the thin-walled and bent exhaust tubing, the main source of the noise remains the strong and high turbulent air flow up. From the realized flow numerical simulations result the following recommendations for the next process. Simply said, those recommendations are well known, but oft not used – we believe that the numerical simulation helps in their implementation. And more, numerical simulation can simply verify various hypotheses about the effectivity of noise damping, about possible back suction of exhausted impurities, about the influence of skylights system etc.

1. The orifices in the labyrinth silencer should be as narrow as possible, but accessible for cleaning. A suitable cleaning method is the blasting by dry ice. But the primary should be the perfect capture and separation of sticky parts from the exhausted air before the exhausting fan, using an effective water curtain, suitable filtering elements, etc. It is not clear why in this plant the relative high portion of sticky parts is not caught before the exhausting system.
2. The wall of the silencer should be massive and stiff, with reinforcement against vibrations – the designed partitions are fixed on one side, only!
3. The silencer ceiling is necessary, with the slope to the periphery - in an open design, rain and snow are falling in.
4. The air exhaust directed down is more suitable – the parameters of the flow field in the outlet, influencing the level of aerodynamic noise, are of tens of percent lower, compared with the outlet up.
6. The rugged roof of the hall (skylights) helps further to the aerodynamic noise decreasing of the air flow, blown down and dispersed along the roof surface.
7. The simulated impurity concentration in the suction from the roof back into the hall is 7.5% only of the concentration at the outlet. The inserted partition on the roof between the outlet and the suction, the impurity concentration in the suction is decreasing on 1% only of the concentration in the exhausted air.
8. It is possible to situate the similar simple partitions around the next eventually opened skylights.

Well known standard damping systems are not presented here, they are available at many internet addresses.

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[6] Google maps