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Noise Reduction at the Fan Outlet

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Abstract - The paper deals with numerical flow simulation in the fan outlet of a large painting shop. The received results of pressure fluctuations in numerical models are evaluated using both frequency analysis of pressure fluctuations and measuring and evaluation of a really operating system. From the conclusion there is defined the hypothesis of noise origin and more, it is proposed a more suitable design of the system without creation of pressure fluctuations. The system is ready for implementation.

Keywords— noise reduction; fan outlet; numerical flow simulation;

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1. Introduction

The paper deals with numerical flow simulation in the outlet system of a large painting shop. The increased noise level is spread into the surroundings. Several models were used for the identification of noise sources.

The relatively simple geometry of the large volume of about 100 m^3 consists from rectangular volumes made from thin metallic sheets with a cylindrical outlet, see following Fig. 1, 2, 3. Due to the high and quick volume flow, the walls of the channel are vibrating and thundering, mostly in the central horizontal part of the observed system. The aim of the numerical flow modeling is to survey the pressure / velocity fields in the system and to define the source of the noise.

2. Resolution

2.1. Numerical Model in General

The simple three-dimensional (3D) geometry is evident from the following Fig. 1 to Fig. 3, where the longitudinal plane of symmetry can be used. The rectangular inlet is situated on the right side from below; the cylindrical outlet is situated on the left side upwards. Both parts are connected by a horizontal prismatic part. The defined pressure difference of 1000 Pa creates the inlet velocity, which corresponds well with the real air flow of 58 m^3/s in the real size of inlet cross-section. The k-ε used standard commercial code for incompressible ideal gas and model of turbulence.

2.2 Steady 3D Solution – Results

On both pressure and velocity fields, see Fig. 1 and Fig. 2, it is visible that in the system with the rectangular changes of both cross-sections and flow direction, there are present intensive pressure and velocity gradients, the large areas of flow separation with backflows etc., which could be the reason of the pressure forces, causing the vibrations of the relative thin structure of

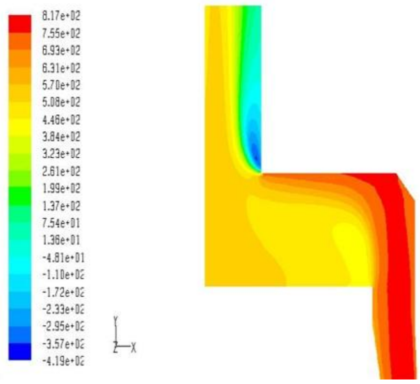


Fig. 1 Pressure field – steady 3D solution

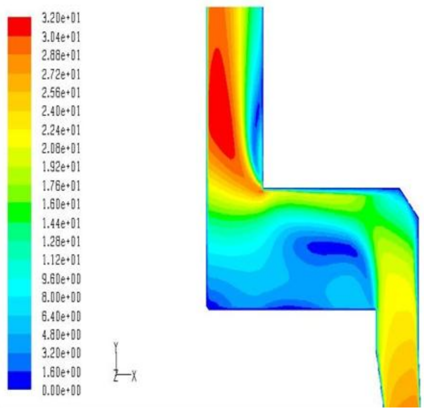


Fig. 2 Velocity field – steady 3D solution

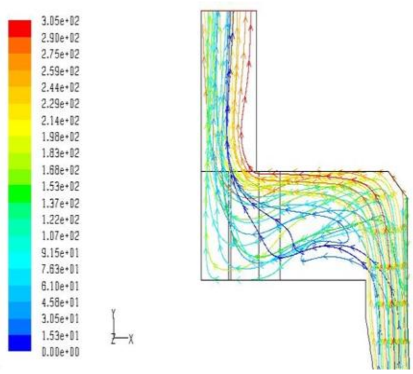


Fig. 3 Streamlines – steady 3D solution

air channels. Another view on the uneven flow field, there are the streamlines in Fig. 3. Generally said, the cross-section of the model volume is not fully filled by air flow.

The idealized evaluation of the results leads to significant force effects – the inlet velocity of 22 m/s represents the momentum of about 1500 N and the impact power of such flow reaches a value of about 30 kW respectively, idealized as full impact effect. Of course, real flow is not fully stopped, so such excessive values are the theoretically possible maximum, only.

2.3 Unsteady Solution – Harmonic Analysis

From the following unsteady simulation, some pressure fluctuations were detected in the selected points. Fig. 4 shows pressure fluctuations, recorded in the centre on the wall of the horizontal part of the system (the geometry see the previous Fig. 1 to 3).

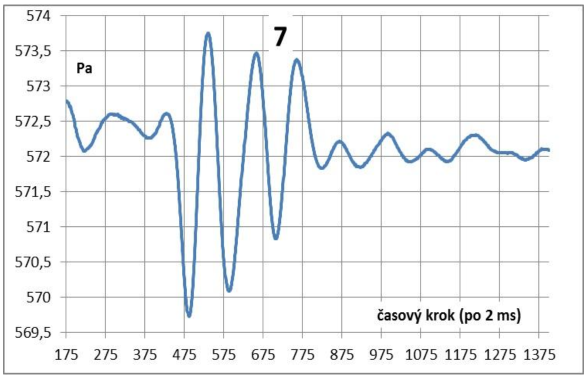


Fig. 4 Detected pressure fluctuations (time step of 2 ms)

The result of the frequency analysis of the recorded pressure fluctuations is presented in Fig. 5. It is visible the highest amplitude of 0,5 Pa approx. at the frequency of about 5 Hz, but it is not any audible frequency. Simply calculated, on the large wall surface of 12 m^2, made from a thin metallic sheet, there acts the total pressure force of 6,5 N approx. Excessively said, it looks like when on this sheet surface a 0,6 kg hammer is falling 5 times per second. It should be a really intense noise! Of course, due to real stiffening by the channel frame, the real force effect would be smaller.

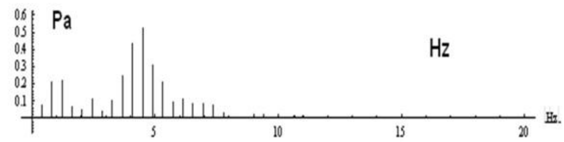


Fig. 5 Frequency analysis of recorded pressure fluctuations

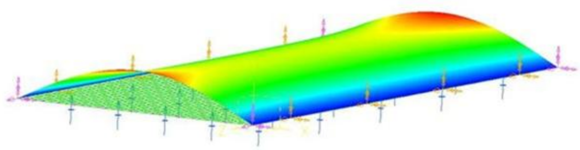


Fig. 6 Deformation of the thin sheet by constant pressure

The possible deformation of such a thin rectangular sheet, loaded by uniform pressure, is shown in Fig. 6 – boundary conditions simplified as two sides free, two sides fixed and at constant pressure value.

From the above presented results, the hypothesis was determined that such primary low frequencies (pressure fluctuations) of the flow, separated from walls of the rectangular channel, could cause any secondary audible vibrations of the thin metallic sheets, which could be further amplified in the following outlet volume as in any trumpet.

2.4 A Planar (2D) Model With Fine Mesh

To be sure that the above observed pressure fluctuations are not caused – maybe – by the relatively coarse mesh, used in this large model, the next model was created in 2D, only, but with the refined mesh and with boundary layers, too. The results are very similar so that in Fig. 7 similar to Fig. 2, it is presented the velocity field, only, as an example. Thus the reason of the detected pressure fluctuations and subsequently of generated sound frequencies, too, could be the flow separation from the walls in the sharp changes of the flow direction.

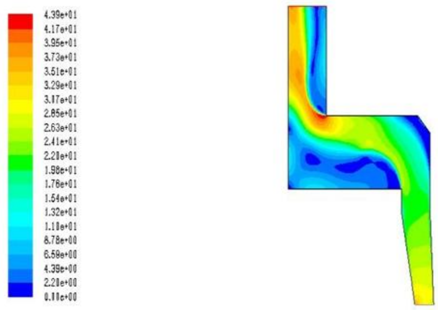


Fig. 7 Velocity field in the 2D model, fine mesh

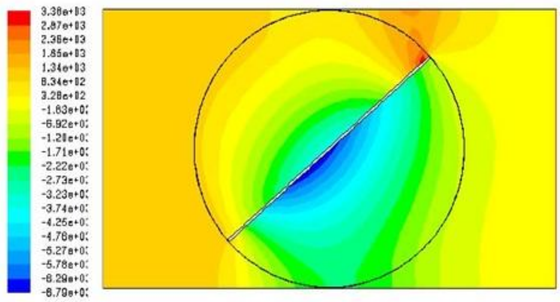


Fig. 8 Pressure field around the rotating flap at the inlet

2.5 Simulation of the Rotating Fan Rotor Influence

The rotating blade wheel of the exhaust fan creates other pressure fluctuations in the observed system. The generation of such fluctuations was simply simulated as a flap, rotating in the inlet cross-section of the previous model. As an illustration, in Fig. 8 there is the pressure field around such rotating flap in any random position.

The recorded pressure fluctuations are presented in Fig. 9 and the relevant result of the frequency analysis in Fig. 10. The maximum amplitude at the frequency of 4,5 Hz approx. remains the same as above (see Fig. 5), the next local amplitude maximum at the frequency of about 43 Hz corresponds with the fourth harmonic frequency of the rotating blade wheel.

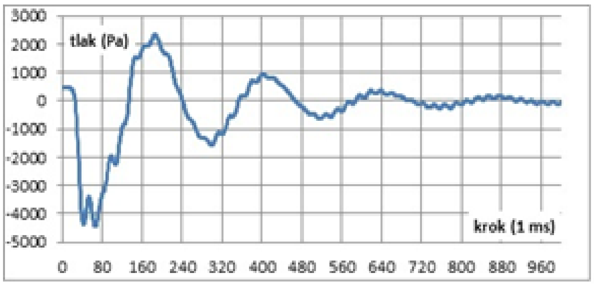


Fig. 9 Pressure fluctuations after the rotating flap (time step of 1 ms)

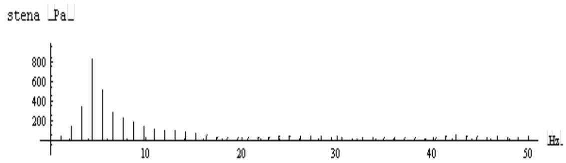


Fig. 10 Frequency analysis of pressure fluctuations after the rotating flap

Of course, the real rotating blade wheel could be simulated, too, but for the representation of periodical inlet excitations of the flow field the elementary rotating flap is sufficient.

2.6 Field Measurements

The results of the field measurements on the site [1] are very similar to the above presented results of numerical flow simulations. Measured in the air flow, the frequencies of 2,6 – 3,5 – 4,5 Hz were detected, depending on the actual position of the measuring point. The measured values are very similar to the above mentioned simulated values. It is hardly to get any better coincidence because the exact fixation of the pressure sensor in the strong air flow is difficult. And more, the next detected frequencies are very expressive harmonic multiples of the basic frequency 11,5 Hz of the rotating blade wheel.

As an illustration, only, Fig. 11 presents the result of harmonic analysis in one position of the pressure detector, where the first amplitude maximum at the frequency of 4,5 Hz corresponds with the basic pressure fluctuations found by numeric simulation and the second amplitude maximum at the frequency of 92,2 Hz is the eighth harmonic of the fan rotation frequency etc. Other local amplitude maximums are situated at frequencies approx. 45 – 66 - 88 Hz, corresponding to other harmonic multiples of the basic rotational frequency of 11 Hz.

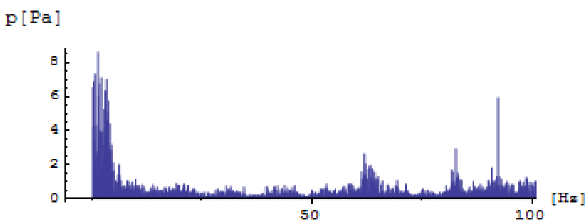


Fig. 11. Field measurement - record of frequency analysis

3. New Design

It is clear that for suppressing the above identified pressure fluctuations it should be to improve the flow field in the system. In other words, it is necessary to design and to use a better shape of the exhaust channel, which complies better with the natural image of the simulated flow field. In the former channel of rectangular both cross-section and changes of the flow direction, the flow fills only a part of the whole cross-section, due to the large areas of the flow separation from the wall just behind each of rectangular bends.

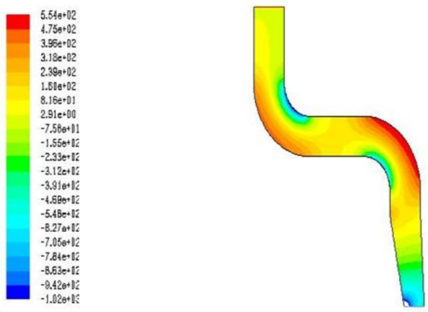


Fig. 12 Pressure field – smooth shape

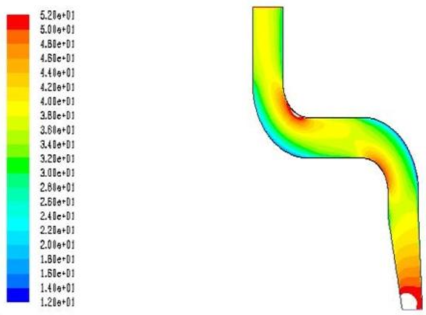
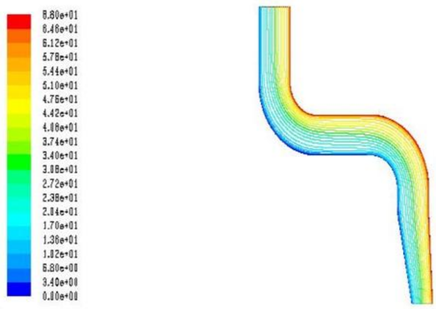


Fig. 13 Velocity field – smooth shape

Fig. 14 Streamlines – smooth shape

Using a smaller and circular cross-section, designed after



results of above presented simulations, the flow field becomes smoother, practically without flow separation – there are not any sharp changes of the flow direction. For comparison with Fig. 1 to Fig. 3, here, there are presented analogous Fig. 12 to Fig. 14 – the pressure field with typical maximum at the outer diameter of the channel bend and minimum at the inner diameter of each bend, the velocity field with typically inverse values after the Bernoulli's equation (maximum value at the minimum radius and minimum value at the maximum radius) and the streamlines, differed by color, well following the channel shape without separation.

After the start-up period of simulation, the observed pressure fluctuations are going practically to zero, see Fig. 15. So it is possible to expect the flow field without pressure fluctuations, given by the above mentioned strong flow separation.

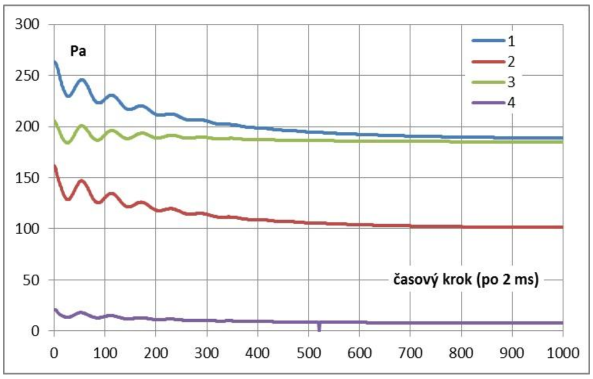


Fig. 15 Pressure fluctuations – smooth shape (time step of 2 ms)

4. Conclusion

The used standard method of numerical flow simulation, here together with verification of results by frequency analysis of recorded pressure fluctuations, gives a suitable guide how to suppress the noise level spreading from the outlet of a fan in the surroundings.

The results of the flow simulation show the possible primary reason of the pressure fluctuations in the air flow, probably subsequently modified into vibrations of thin metallic structure, which are the secondary source of the noise, spread into the surroundings. Pressure fluctuations of the air flow, recorded in numerical models, are verified by real in situ measurements. From the following data analysis it is evident a good coincidence of both methods.

On the basis of the results of numerical flow modeling, there are proposed some arrangements of the geometry – rounded transitions of cross-sections instead of former rectangular ones. The resulting flow field does not show so large pressure fluctuations, which are usually the source of the increased noise level, particularly of the induced noise, generated by the interaction of such disturbed flow with thin metallic walls.

The presented study conserves the actual shape of the system. On the basis of next survey it is clear that significant shape modifications could be made, too, as for instance after Fig. 16. An absolutely straight (vertical) duct, equipped by a noise silencer at the outlet end, is designed as a labyrinth and/or louvers and made as walls resistant to vibrations. Due to the radial (horizontal) flowing in such a silencer, the velocity value is decreasing so that the outlet value is a fraction, only, of the inlet one.

Such solution is not only ecologic – reduced noise level, but economic, too – simple shape. Generally said, an ecologic solution without economic effect is not the right solution.

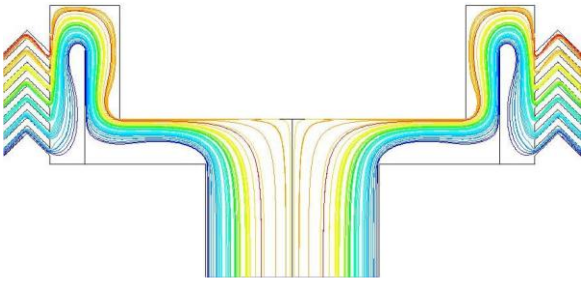


Fig. 16 Labyrinth as a noise silencer

In general, firstly, it should be used a pure technical solution, presented in this paper. The used better shape could suppress or remove the actual reason of the increased noise level. And secondly, only, it should be used any additional noise insulation of the existing system, in principle characterized by some operational defects, leading to increased noise level, as presented above.

References

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