Acoustic Evaluation of the Air-Conditioning Unit in the Room

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Abstract. This paper deals with an acoustic simulation, evaluation and measurement of an indoor air-conditioning unit commonly used in hotel rooms. The unit is situated in the plaster board ceiling of a hotel room above the entrance of the hotel room. The first step was to create a model of the real indoor air-conditioning unit from the catalogue data of a manufacturer, then built the model for the simulation of the internal sound propagation in the hotel room. The second step was to mock-up the real hotel room. The goal of the paper is to compare the calculated data from the simulations with the measured data in-situ.

1. Introduction

A current trend in civil engineering projects is to create simulations to provide usable data for case studies and ideas of different solutions. The goal of the environmental simulations is to predict the results of heating, cooling, ventilation and solar systems or acoustical engineering. This paper deals with the acoustical simulation of a hotel which was under reconstruction. The investor had very strict demands for the indoor sound pressure level from the heating, cooling and ventilation system. It was necessary to prepare the acoustical simulation of a typical hotel room with a commonly used air-conditioning unit to predict the noise emitted in the hotel room. After the simulation was completed, the mock-up of the projected hotel room was built, it was possible to validate the model with noise measurements.

The indoor air-conditioning unit is situated in the plaster board ceiling above the corridor part of the hotel room, see fig.1. The air-conditioning unit is a rectangular case (dimension: height 200 mm, width 750 mm, length 620 mm) with a diagonal fan in the inlet side, then there is a heat exchanger and outlet of the unit. The unit is connected to a mixing chamber (dimension: height 125 mm, width 825 mm, length 800 mm) where a pipe with conditioned air from the central ventilation is also connected. The inlet of the conditioned air, to the hotel room, is conducted from the mixing chamber. The walls, bottom and ceiling of the indoor space of the mixing chamber are covered with sound absorbing material with a thickness of only \( \approx 20 \text{ mm} \) (ArmaSound RD240 or Gumex Airpren Pell). The height of the free cross-section of the air outlet is approx. 85 mm due to the sound absorption material on the surfaces of the mixing chamber. A commonly used blade grill is installed as a terminal diffuser of the inlet to the hotel room. The ventilation unit sucks air from the plaster board ceiling. The plaster board ceiling is connected with the hotel room by a 50 mm slit above the door. There is no sound absorbing material in the plaster board ceiling.
The hotel room includes a bedroom (3.5 m x 4.5 m, height 3 m), a corridor space (1.6 m x 2.3 m, height 2.4 m), and a bathroom, where the central ventilation suction is situated. The space of the bathroom is not important for the noise evaluation of the air-conditioning unit because it is separated by a door, however, the corridor space is not separated by a door. So, the bedroom with the corridor space is the fundamental size of the acoustic model, see fig. 2.

2. The models of the sound propagation of the air-conditioning unit

A lot of computational software products exist for acoustic assessment. The differences are mainly in the possibilities of entering more complicated geometric situations, the possibility of entering parameters for the sound characteristics of materials, the directionality of the sources and in the computational methodology. The more sophisticated the software is, of course, the more expensive it is, so it is always necessary to assess each situation and make decisions as to how large the monitored area will be and what details are decisive for the noise assessment. These details may include reflections from surfaces, the sound insulation of the materials, the frequency composition of the sound, the proximity of the point to the source and the size of the sound source, its directivity, and the number of sources to be considered in the model.

It is not possible to use the same approach for the acoustic assessment of, e.g., traffic noise in the city, or to provide a detailed explanation of the noise propagation from the cooling unit to the protected area near the installation. A completely different approach is used for these noise source models such as aerodynamic noise. There must be simulated physical principals. The model simulates noise by solving the fluid mechanics equations, which describe fluctuations of air mass, and subsequently describe sound propagation by wave equations, see [1] and [2]. This detailed assessment, when the sound propagation is described by wave and fluid mechanics equations or detailed measurement around the source [3], is not practical due to the huge calculation time.

A similar approach for the type of calculation, described in this paper, was performed in [4]. However, in [4] it was solved without the sound insulation of the construction, because the dominant noise propagation to the surrounding was the direct field and reflection. In this paper the principal of the calculation is similar to the model of the noise propagation from outside space in front of a façade into the indoor space of some building based on the sound insulation of the facade [5]. In [5] the same model was used, described as ‘Particles model’. This model describes the trajectory of a number of particles which represent the noise propagation. The Particles model could use the CUDA calculation which runs on the graphics card of a computer.

The noise model was performed using CADNA R software (Datakustik GmbH). The 3D model of the calculation was used with the base plane at the floor of the hotel room. The area of the model was

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**Figure 1.** A plan of the air-conditioning system with the air-conditioning unit, the measurement points R1–R5 and C1–C3.

**Figure 2.** A plan of the acoustic model hotel room.
limited to the indoor space of the bedroom and corridor, see fig.2. The boundary areas of the model are the walls, floor and ceiling of the hotel room, see figure 3. The bathroom was not included in the calculation. The detailed view of the model inside the plaster board ceiling with the air-conditioning unit is presented in figure 4.

Figure 3. 3D view of the model of the hotel room.

Figure 4. 3D view of the plaster board ceiling with the air-conditioning unit.

Table 1. The sound absorption index of the surfaces used in the model.

| Type of surface                                      | Octave band [Hz] of the sound absorption index $\alpha$ [-] |
|------------------------------------------------------|----------------------------------------------------------|
|                                                      | 31  | 63  | 125 | 250 | 500 | 1000 | 2000 | 4000 | 8000 |
| Laminate floor                                       | 0.01| 0.01| 0.02| 0.03| 0.03| 0.04 | 0.06 | 0.05 | 0.05 |
| Wall with plaster                                     | 0.02| 0.02| 0.04| 0.04| 0.04| 0.04 | 0.04 | 0.04 | 0.04 |
| Plaster board ceiling                                | 0.04| 0.05| 0.1 | 0.1 | 0.05| 0.02 | 0.02 | 0.02 | 0.02 |
| Wooden surface                                       | 0.05| 0.05| 0.25| 0.1 | 0.11| 0.09 | 0.09 | 0.2  | 0.2  |
| Window and mirror                                    | 0.05| 0.08| 0.28| 0.2 | 0.11| 0.06 | 0.03 | 0.02 | 0.02 |
| Metal surface of the furniture                       | 0.02| 0.02| 0.02| 0.03| 0.03| 0.04 | 0.04 | 0.03 | 0.03 |
| Upholstered furniture                                | 0.03| 0.03| 0.06| 0.27| 0.44| 0.5  | 0.4  | 0.35 | 0.35 |
| The surface of the bed and pillows                   | 0.08| 0.1 | 0.15| 0.4 | 0.5 | 0.6  | 0.5  | 0.45 | 0.45 |
| Sound absorption material in the mixing chamber, thickness 20 mm. | 0.01| 0.02| 0.09| 0.28| 0.77| 0.8  | 0.85 | 0.85 | 0.85 |

Sound is spread from the unit to the hotel room in three ways. The first one is directly from the outlet of the unit through the mixing chamber and through the terminal diffuser. The blade grill was used as an HVAC terminal diffuser so one of the sound sources is an aerodynamic noise from the flow over the blades. The fresh air inlet is also a sound source, which generates noise from the central ventilation system. This type of noise should be reduced by silencers in the pipe, but in this case it was one of the dominant sound sources. Of course it is simple to reduce the central ventilation system noise by the addition of silencers. The model was mainly focused on the sound propagation from the air-conditioning unit, because this device is not only used for cooling but also for heating the hotel room. The second way sound is spread, is from the suction of the unit, see figure 4, the noise spreads through the air gap in the plaster board ceiling. The third way is based on the transmission of sound through the plaster board ceiling. Due to this transmission it was necessary to calculate the model with sound reduction insulation. The computational model used is the ‘Particles model’, see [5], with the total number of particles, after the resulting iterations, of $1.024 \times 10^8$ (room volume is 71.13 m$^3$). The uncertainty of this
The model used different kinds of surfaces, from the laminate floor to the sound absorption material in the mixing chamber, see table 1. The sound reduction insulation of the plaster board ceiling used in the model is in the table 2.

### Table 2. The sound insulation of the surfaces used in the model.

| Type of construction | Octave band of the sound insulation $R'$ [dB] |
|----------------------|---------------------------------------------|
| Plaster board (white board), one board thk 12.5 mm, weight min. 7.1 kg/m² | 3  8  11  15  20  25  28  24  24  24 |

3. **The sound sources used in the model**

The air-conditioning unit case without the fresh air outlet to the mixing chamber is not described in this paper, because the outlet from the central ventilation is easy to reduce, but the problem is with the continuous operation of the air-conditioning unit in the plaster board ceiling. There is no space for silencers, and the unit is used for cooling and heating the hotel room. The air-conditioning unit includes 3 types of sound sources. The volume sound source represents the noise generated by the construction of the device. The second and third types of sound source are the vertical area sound sources, the suction and discharge of the unit, see figure 4.

### Table 3. The comparison of the model and catalogue data of the air-conditioning unit in HIGH mode.

| Sound source | Octave band of the sound power level [dB] (uncertainty ±2 dB) |
|--------------|-------------------------------------------------------------|
|              | 31  63  125  250  500  1000  2000  4000  8000  $L_{WA}$ |
| Volume       | 46.5 47.0 47.5 47.5 45.5 43.0 36.5 30.5 29.5 47.4 |
| Discharge    | 51.0 48.0 37.0 45.0 35.0 44.0 37.0 32.0 24.0 46.0 |
| Suction      | 49.0 47.0 35.0 45.0 33.0 43.5 37.0 31.0 23.0 45.6 |
| The whole model of the unit | 54.0 52.1 48.1 50.8 46.1 48.3 41.6 36.0 31.3 51.2 |
| Catalogue data | 54  52  48  50.5  46  48  41.5  36  30  51 |
| At the distance 1.5 m from the volume, according the MODEL, see fig. 5. | 35.3 34.7 32.4 34.0 30.0 27.8 21.3 14.9 13.8 32.3 |
| At the distance 1.5 m from the volume, according the CATALOGUE, see fig. 5. | 32.0 32.5 33.0 33.0 31.0 28.5 22.0 16.0 15.0 33 |

The acoustic data for the air-conditioning unit is available from the manufacturer’s catalogue. According to the catalogue, the unit, in HIGH mode, has an overall sound power level $L_{WA} = 51$ dB determined by the technical norm ISO 3744. Due to ISO 3744 the noise source is placed in the measuring chamber (sound-absorbing chamber is possible) and the sound pressure level is measured at different points around the source. The sound power level is then calculated from the average, which means that all sources, the volume, suction and discharge, are calculated together. In addition, according to the catalogue, the sound pressure level corrected by filter A is at the distance 1.5 m from the unit $L_{PA – 1.5 m} = 33$ dB in HIGH mode (air flow $480 \text{ m}^3/\text{h}$) and $L_{PA – 1.5 m} = 27$ dB in LOW mode (air flow $384 \text{ m}^3/\text{h}$), see figure 5. The external added pressure drop per unit is 10 Pa. Figure 5 shows the set up for measuring according to the catalogue. The model simulated the same situation according to figure 5. The partial results from the simulation are in the table 3.
The set up measuring according to the catalogue.

The sound pressure level at a distance 1.5 m \( L_{PA} - 1.5 \text{ m} = 27 \text{ dB} \) in LOW mode, corresponds to: Volume \( L_{WA} = 41.4 \text{ dB} \), Discharge \( L_{WA} = 42 \text{ dB} \) and Suction \( L_{WA} = 41.6 \text{ dB} \).

The terminal diffuser (blade grill) was simulated by vertical area sound source. This type of source is generated by aerodynamic noise (airflow over the blades). The data of the acoustic spectrum of this type of sound source was assumed according to experiments [1]. The data of the sound power level for the air flow 480 m³/h is presented in table 4.

Table 4. The sound power level of the terminal diffuser blade grill for air flow 480 m³/h.

| Octave band of the sound power level [dB] | 31  | 63  | 125 | 250 | 500  | 1000 | 2000 | 4000 | 8000 | \( L_{WA} \) |
|-----------------------------------------|-----|-----|-----|-----|------|------|------|------|------|----------|
| Terminal diffuser blade grill           | 22.0| 24.0| 25.0| 27.0| 28.0 | 26.0 | 25.0 | 24.0 | 22.0 | 32.1     |

4. Results of the model

The results of the simulation, in the figure 1 for the HIGH and LOW mode, are shown in table 5. The noise fields from the model are presented in figure 6 – horizontal view, and in figure 7 – vertical view, in HIGH mode. The most problematic place in the room is directly under the air-conditioning unit. This area is most affected by the weak sound insulation of the plaster board ceiling and secondly by the inlet in the plasterboard ceiling, where there are no silencers. Due to the sound absorption material in the mixing chamber, the outlet from the unit is not the dominant sources, see figure 7. Figure 8 shows a 3D view of the area with a fixed level \( L_{PA} = 39 \text{ dB} \). The average value in the living area of the hotel room of the sound pressure level for HIGH mode is \( L_{PA} = 37.4 \text{ dB} \), and for LOW mode is \( L_{PA} = 32.6 \text{ dB} \). The LOW mode is acceptable for an average hotel room, it is possible to sleep comfortably. But for homes this sound level is above the Hygienic limit \( L_{AMAX} = 30 \text{ dB} \).

Table 5. The results of the sound propagation model during the operation of the air-condition unit.

| 1.6 m above floor | Total sound pressure level corrected by filter A, \( L_{PA} \) [dB], (uncertainty ±2 dB) |
|-------------------|-----------------------------------------------|
|                   | High                                         | Low                                          |
| R1                | 37.8                                         | 33.1                                         |
| R2                | 38.2                                         | 33.5                                         |
| R3                | 37.9                                         | 33.0                                         |
| R4                | 36.6                                         | 31.8                                         |
| R5                | 36.3                                         | 31.5                                         |
| C2                | 41.9                                         | 36.8                                         |
Figure 6. The horizontal sound field 1.6 m above the floor, HIGH mode.

Figure 7. The vertical sound field in the plane through the corridor room and the ac unit, HIGH mode.

Figure 8. 3D view of the model with an area of fixed level $L_{PA} = 39$ dB, HIGH mode.
5. Conclusion – validation of the model with the real measurement of the MOCK-UP

The simulations are performed with the fixtures and fittings in the room (the bed, armchair, cupboard, laminate floor, plaster board ceiling for unit, …). The measurements were taken in an empty room which increase the noise in the range of 1.5–2 dB. The measured results are relatively similar to the simulated results, but there are some differences, see table 6. The measured sound pressure level (after correction for the empty room) is lower, especially at the entrance, because the simulation only included one piece of plasterboard (thickness 12.5 mm) in the ceiling for the air-conditioning unit which has a lower sound insulation than the double plasterboard installed in the MOCK-UP.

| Table 6. The results L_{pa} of the measurements in the MOCK-UP and simulations. |
|---------------------------------------------------------------|
| 1.6 m above floor | High | Low |
| Measurement | Simulation | Measurement | Simulation |
| Hotel room  | 38.6 ± 1.5 | 37.4 ± 2 | 33.7 ± 1.5 | 32.6 ± 2 |
| Corridor    | 39.3 ± 1.5 | 41.9 ± 2 | 34.7 ± 1.5 | 36.8 ± 2 |

The generated maximal sound pressure level is under the Hygienic limit for a residential building during the day L_{AMAX} = 40 dB (there was no tonal component). The measured sound pressure level is generated only by the air-conditioning unit in the hotel room, the central ventilation was not considered (the installation has not been completed). The vision of the investor was that the sound pressure level, generated from the technical equipment for heating and cooling, should be under the L_{AMAX} = 30 dB, but with this type of device and solution for ventilation it is almost impossible.

It is possible to reduce the noise generated by the air-conditioning unit, but it would be necessary to change the project and install sound absorption material in the ceiling, where the unit is installed, and remove it the mixing chamber, and replacing it with silencers, and to add silencers to the suction end of the air-conditioning unit. But we cannot expect results lower than L_{AMAX} = 35 dB in the hotel room for HIGH mode (airflow 480 m³/h) of the air-conditioning unit in addition to the central ventilation system. It is necessary to change the conception of the solution. The fan should be installed in the central engine room, because then it is possible to apply the most effective solution to reduce the noise, by silencers, in the central ventilation system.

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