Experimental Results for Ventilation System with Local Recirculation Diffusers

Dmitry Vladimirovich Kapko,
LTD “NPO THERMEC”, 46-2, Dmitrovskoe shosse, Moscow 127238, Russia.

Vyacheslav Erikovich Shkarpet, Kristina Vladimirovna Kochariantc,
LTD “Arktos”, 4, Predportoviy proezd 6, Saint-Petersburg, Russia.

Iurii Andreevich Tabunshchikov, Marianna Mihailovna Brodach,
Moscow Architectural Institute (State Academy), 11, Rozhdestvenka street, Moscow 107031, Russia.

Abstract
The paper presents experimental results for a ventilation system with local recirculation diffusers designed by the authors for buildings with high heat emissions (> 25 W/m²), such as office buildings. A 6,000 m³ office building in Moscow has been demonstrated to have indoor excess heat occurring at the outdoor air temperature over -9 °C. A principal diagram of a ventilation system with the designed local recirculation diffusers (LRD) is given.

Tests were carried out on aerodynamic and acoustic benches and on a bench demonstrating the use of the designed system in an office. The tests have confirmed that comfort air temperature and acoustic parameters can be ensured in a room serviced with a ventilation system with local recirculation diffusers when the outdoor air temperature supplied to the LRD is +6 °C and above. It has been confirmed that a local recirculation diffuser can automatically maintain comfort air temperature parameters (air temperature range of 19 to 21 °C, air flow velocity of 0.2 m/s max.) according to the outdoor air temperature (outdoor temperature range of 6 to 18 °C) and the room heat load (recirculation air temperature range of 18 to 25 °C), with the sound pressure level generated by the diffuser being less than 40 dB(A) for a distance of 2-2.5 m.

Keywords: Ventilation systems, recirculation, local recirculation diffuser, diffuser, energy efficiency, experiment, acoustic test, aerodynamic test, full-scale test.

INTRODUCTION
At present, one of the most essential tasks for a designer of ventilation systems is to achieve comfort air temperature conditions at minimum energy consumption. A more and more increasing number of theoretical and experimental studies have been dedicated to upgrading the existing systems and designing new schemes and solutions for ventilation systems of buildings [1-9]. This article provides experimental results for a ventilation system with local recirculation diffusers designed by the authors. These systems are designed for buildings with high heat emissions (> 25 W/m²), such as office buildings. The main feature of these buildings is the occurrence of indoor excess heat during a transitional period between seasons and even during a heating period of a year. According to the analysis conducted by the authors (Figure 1), the indoor excess heat in a 6,000 m³ office building in Moscow occurs when the outdoor air temperature is above -9 °C (mean indoor heat emissions in Russian office buildings are 27.46 W/m², including 5.4 W/m² from people, 12.8 W/m² from lighting, and 9.26 W/m² from office equipment).

Figure 1. Comparison of the indoor heat emissions and the heat losses during cold and transitional periods for a 6,000 m³ office building in Moscow
Reduction of the supply air temperature from the air supply unit to values, at which the indoor excess heat can be assimilated, thus reducing the heat consumption of a ventilation system for heating the outdoor air, can be a rational solution for these buildings. However, this reduction is restricted by regulatory documents (in the Russian Federation, this value is specified in a relevant Rulebook [10]) so that comfort parameters of the supply air stream at the edge of the working area in the room be ensured.

To increase the potential of indoor heat excess assimilation by cool supply air the authors have designed a ventilation system with local recirculation diffusers (Figure 2). This system can supply lower temperature air (from +6 °C) into the room to a local recirculation diffuser which mixes the incoming outdoor air and the recirculation air from the room, thus producing a supply air stream with comfort parameters (temperature and air flow velocity) at the edge of the working area in the room.

The operation of a recirculation diffuser (Figure 3) is described below. The low temperature outdoor air (≥ +6 °C) flows from the air supply unit (not shown) into a pipe (4) and then is supplied into the room via a diffuser (5). The diffuser (5) generates a fan-shaped air stream which adheres to the ceiling. The outdoor air flow rate remains constant for as long as the system is in operation. A fan (7) feeds recirculation air (the air from the serviced room) through a recirculation air pipe (6) into the static pressure chamber (1); the air is cleaned as it passes through a filter (8). Then, the clean air is directed into the room by rows of eddying cells (3) on the face panel (2) of the diffuser. The eddying cells (3) generate a fan-shaped stream of recirculation air which adheres to the ceiling and mixes with the outdoor air stream from the diffuser (5) thus producing a supply air stream. A controller and a frequency drive (not shown) control the speed of the fan (7) and the recirculation air flow rate to maintain the supply air set temperature.

The prospects for designing and using local recirculation diffusers for energy-efficient ventilation systems are substantiated in the work [11]. Computational modeling research into the efficiency of the designed local recirculation diffusers is described in the work [12]. This article considers test results obtained for an experimental installation of a local recirculation diffuser.

Figure 2. The principal diagram of a ventilation system with local recirculation diffusers
METHODS
The experimental installation of a local recirculation diffuser (LRD) (characteristics of the installation are given in Table 1) was subjected to the following tests:
- aerodynamic tests of LRD;
- acoustic tests of LRD;
- tests of LRD in an office room.

Aerodynamic tests of a local recirculation diffuser were carried out on an aerodynamic test bench at the aerodynamics and acoustics research laboratory of LTD “Arktos”, an industrial partner [13]. Figure 4 shows a schematic view of an aerodynamic test bench for aerodynamic tests of LRD.

Table 1: Main parameters and dimensions of the experimental installation of LRD

<table>
<thead>
<tr>
<th>Main parameters and dimensions</th>
<th>Values and units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air purity class of the recirculation air filter</td>
<td>F5</td>
</tr>
<tr>
<td>Dimensions of the recirculation air diffuser panel</td>
<td>450 × 450 mm</td>
</tr>
<tr>
<td>Dimensions of the static pressure chamber (H × W × D)</td>
<td>300 × 424 × 424 mm</td>
</tr>
<tr>
<td>Outdoor air pipe diameter</td>
<td>125 mm</td>
</tr>
<tr>
<td>Recirculation air pipe diameter</td>
<td>250 mm</td>
</tr>
</tbody>
</table>
The objectives of aerodynamic tests were as follows:
- To determine the air drag of LRD in the outdoor air and recirculation air circuits in order to choose the fan to meet the characteristics of LRD;
- To determine the stream hydrodynamic coefficient $m$;
- To determine the maximum ratio of the recirculation air flow rate and the outdoor air flow rate (the air output capacity) in order to verify the conformity of the chosen fan to the stated characteristics.

Aerodynamic tests were carried out in isothermal conditions in accordance with national standards [14-15]. The tests involved the measurement of dynamic and total pressure at measurement cross sections of the test sections along the bench with a Kimo MP 200 thermo-anemometer (France) fitted with a pressure probe (a Pitot tube).

The following parameters were calculated based on the measurement results:

The air flow velocity in the test air ducts $u$, m/s, and the flow rate $L$, m$^3$/h, given by the equations:

$$u = \sqrt{\frac{2P_{\text{dyn}}}{\rho}},$$  

$$L = 3600F_{\text{sect}} \times u,$$

where: $P_{\text{dyn}}$ is the measured dynamic pressure, Pa; 
$\rho$ is the air density, kg/m$^3$; 
$F_{\text{sect}}$ is the cross section area of the test section in the bench, m$^2$.

- The pressure loss coefficient for the diffuser was calculated from:

$$\zeta = \frac{P_t}{P_{\text{dyn}}}.$$  

\textbf{Figure 4.} A schematic view of an aerodynamic test bench for aerodynamic tests of LRD.
Where $P_t$ is the total pressure; $P_{dy}$ is the dynamic pressure defined based on the velocity in the air duct cross section: for the recirculation air section $F_{sect} = 0.049$ m$^2$; for the outdoor air section of the diffuser $F_{sect} = 0.0123$ m$^2$.

Characteristics of the adhering stream (velocity coefficient $m$) were determined by the measurement of velocity fields at cross sections of the generated supply air stream for different distances $Y$ from the axis of the air diffuser in the plane of adherence. The air flow velocity measurements were taken with a TTM-2 thermoanemometer system (EKSIS) and were used to define the maximum air flow velocities, $v_{y}^{max}$, m/s, at each cross section, and the velocity coefficient $m$. The latter was calculated from:

$$ m = \frac{v_{y}^{max}}{v_{0}} \cdot \frac{y}{\sqrt{F_{0}}} \quad (4) $$

where $F_{0} = 0.303$ m$^2$ (the cross section area of the static pressure chamber).

As the result of the tests, the pressure loss coefficient for the outdoor air circuit $\zeta = 7$, the pressure loss coefficient for the recirculation circuit (a non-constant value as the system contains a filter, see Figure 5), and the hydrodynamic coefficient $m = 4.3$ have been obtained. These results make it possible to calculate pressure losses and stream characteristics for any required air flow rate. The data obtained for the recirculation air section allowed to select an optimum fan which will ensure the required fan pressure at low energy consumption.

![Figure 5](image-url)  

**Figure 5.** A diagram of the pressure loss coefficient change for the recirculation air circuit based on the air flow velocity in the recirculation air pipe

The results of the local recirculation diffuser aerodynamic tests, as well as of the calculation of the characteristics for two conditions, are set forth in Table 2.
Table 2. Results of aerodynamic tests and LRD characteristics calculation

<table>
<thead>
<tr>
<th>Item No.</th>
<th>Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Dynamic pressure at the measurement cross section of the recirculation air pipe test section, Pa</td>
<td>4.4</td>
</tr>
<tr>
<td>2</td>
<td>Total pressure at the measurement cross section of the recirculation air pipe test section, Pa</td>
<td>53</td>
</tr>
<tr>
<td>3</td>
<td>Actual air flow rate through the recirculation air pipe, m³/h</td>
<td>480</td>
</tr>
<tr>
<td>4</td>
<td>Pressure loss coefficient for recirculation air, ζ (F₀ = Fₚ = 0.149 m²)</td>
<td>11.9, 24.8</td>
</tr>
<tr>
<td>5</td>
<td>Dynamic pressure at the measurement cross section of the outdoor air pipe test section, Pa</td>
<td>4.9</td>
</tr>
<tr>
<td>6</td>
<td>Total pressure at the measurement cross section of the outdoor air pipe test section, Pa</td>
<td>34, 33</td>
</tr>
<tr>
<td>7</td>
<td>Actual air flow rate through the outdoor air pipe, m³/h</td>
<td>127, 125</td>
</tr>
<tr>
<td>8</td>
<td>Pressure loss coefficient for outdoor air, ζ (F₀ = Fₚ = 0.0123 m²)</td>
<td>7.0, 7.0</td>
</tr>
</tbody>
</table>

Acoustic tests of the local recirculation air diffuser were performed on an acoustic bench of the aerodynamics and acoustics research laboratory of LTD “Arktos”, an industrial partner. The acoustic tests performed on an acoustic bench were aimed at determining the sound power levels emitted by the local recirculation air diffuser when operated in the conditions of simultaneous supply of outdoor and recirculation air at the maximum capacity (at the recirculation air flow rate of 480 m³/h).

In the process of testing, the universal multi-channel multifunctional system “PULSE 3560-B-030”, equipped with 5 microphone units, which contain 4189 type measuring microphone and 2969L type pre-amplifier, was used. During the measurements, UA 0459 type windbreaks were mounted on the microphones. An end-to-end calibration of the test vein with the help of 4231 type acoustic calibrator was carried out before the commencement of measurements.

During the tests, the tested experimental installation of local recirculation diffuser was installed via an adapter onto the air supply unit of the process ventilation unit with a continuously adjustable air flow rate in the center of the anechoic room with a reverberating floor (Figure 6). The measurement points were located on 2 m radius semi-sphere with the measuring microphones’ scanning along five circular paths in accordance with the requirements of national standards [16-17]. The sound pressure levels were averaged per each circular path. The parameters of the circular paths are shown in Figure 7.

Besides, when LRD and the adapter distanced themselves at each circular path, the levels of acoustic noise (noise background) were determined for each air flow rate value. Upon necessity, the measurement results were added with a correction for the level of acoustic noises.

The levels of the sound power Lw emitted by the local recirculation air diffuser were calculated using the following equation:

$$ L_w = L_{pf} + 10 \lg (S/S_0) + C_1 + C_2, $$

where $L_{pf}$ is the mean level of sound pressure or the sound level on the measurement surface:

$$ L_{pf} = 10 \lg \left( \frac{1}{n} \cdot \sum_{i=1}^{n} 10^{0.1L_i} \right), $$

where $L_i$ is the time-averaged sound level for i-th circular path;

- $n$ is the number of circular paths ($n=5$);
- $S = 2\pi r^2$ is the measurement surface area, m²;
- $S_0 = 1$ m²;
- $C_1, C_2$ are the corrections for meteorological conditions calculated in accordance with national standard [16].
During the tests, the sound power levels \((L_w)\) in dB, in 63, 125, 250, 500, 1000, 2000, 4000 and 8000 Hz octave bands, and the corrected levels of sound power \((L_{wA})\) in dBA, emitted by the face panel of LRD, when operated in the air supply conditions at the maximum value of recirculation air flow rate \((480 \text{ m}^3/\text{h})\), were determined.

The results of acoustic tests are summarized in Table 3.

### Table 3. Results of acoustic tests

<table>
<thead>
<tr>
<th>Item No.</th>
<th>Parameter</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Actual air flow rate through the recirculation air pipe, m(^3)/h</td>
<td>480</td>
</tr>
<tr>
<td>2</td>
<td>Actual air flow rate through the outdoor air pipe, m(^3)/h</td>
<td>120</td>
</tr>
<tr>
<td>3</td>
<td>Time-averaged sound level for the i-th circular path, dB</td>
<td></td>
</tr>
<tr>
<td></td>
<td>at point 1</td>
<td>47</td>
</tr>
<tr>
<td></td>
<td>at point 2</td>
<td>49</td>
</tr>
<tr>
<td></td>
<td>at point 3</td>
<td>50</td>
</tr>
<tr>
<td></td>
<td>at point 4</td>
<td>51</td>
</tr>
<tr>
<td></td>
<td>at point 5</td>
<td>48</td>
</tr>
<tr>
<td>4</td>
<td>Mean sound pressure level emitted by the local recirculation air diffuser, dB</td>
<td>49</td>
</tr>
<tr>
<td>5</td>
<td>Total noise level generated by LRD, dB</td>
<td>54</td>
</tr>
</tbody>
</table>

The office room tests of a local recirculation air diffuser, approximated to the truth, were performed on a specially designed and mounted bench approximating office rooms intended for four employees. The principal diagram of the testing bench for an experimental installation of a recirculation diffuser in office rooms is shown in Figures 8, 9.

The objective of these tests was the determination of the dynamic characteristics of the control of recirculation air diffusers, as well as the determination of microclimate parameters (air temperature and velocity) in the working area of the room with the aim to confirm the following:

- the volume of admixed recirculation air, based on the outdoor air temperature (upon its change from 6 to 18 °C) and on the heat load in the room (recirculation air temperature upon its change from 18 to 25 °C), for maintaining optimum comfort parameters of the supply stream at the edge of the working area in the room, is controlled automatically;
- the compliance with service area microclimate parameters regulated by the national standards of the Russian Federation [18] is ensured.

During the tests, a constant outdoor air flow rate in the volume of 240 m\(^3\)/h \((120 \text{ m}^3/\text{h} \text{ per each LRD})\) was supplied from the supply unit. Due to the installed electric heater, the supply (outdoor) air temperature was maintained at the level not lower than +6 °C.

The supplied (outdoor) temperature and the recirculation air temperature were determined by sensors. When the supply air temperature lowered below +18 °C, the controller calculated the recirculation air volume which had to be supplied to LRD for heating the supply air. Based on the
dependence of the air flow rate on the voltage, supplied to the fan and determined at the aerodynamic bench, the controller initiated a signal to the fan control. A wattmeter was used to register the value of the electric power consumed by the fan of the local recirculation air diffuser.

In addition to air temperature and velocity, the Bruel&Kjaer spectrum analyzer sound meter was used for measuring sound levels in the working area of the room. The measurement of microclimate parameters was carried out in accordance with the methods described in the national standard [18], for the presence of people in the room, mostly in a sitting position, at the height of 0.1; 0.6 and 1.7 m from the floor level.

According to the values set forth in Table 3 of the national standard [18] for Category 2 rooms (rooms intended for people engaged in intellectual pursuits or studies), the optimum air temperature in cold season shall fall within the range of 19-21 °C, while the optimum air flow velocity shall not exceed 0.2 m/s.

Table 4. Results of the tests performed in the office room

<table>
<thead>
<tr>
<th>Item No.</th>
<th>Parameter</th>
<th>Conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Actual air flow rate through the outdoor air pipe, m³/h</td>
<td>127 125 126</td>
</tr>
<tr>
<td>2</td>
<td>Outdoor air temperature, °C</td>
<td>+6 +14.7 +20</td>
</tr>
<tr>
<td>3</td>
<td>Actual air flow rate through the recirculation air pipe, m³/h</td>
<td>480 130 -</td>
</tr>
<tr>
<td>4</td>
<td>Recirculation air temperature, °C</td>
<td>+20.4 +21 -</td>
</tr>
<tr>
<td>5</td>
<td>Specific energy consumption of LRD fan, W/(m³/h)</td>
<td>0.13 0.11 -</td>
</tr>
<tr>
<td>6</td>
<td>Air temperature in the working area of the room, °C</td>
<td>19.2-19.5 20.0-20.3 21.0-21.4</td>
</tr>
<tr>
<td>7</td>
<td>Air flow velocity in the working area of the room, m/s</td>
<td>&lt;0.19 &lt;0.1 &lt;0.1</td>
</tr>
<tr>
<td>8</td>
<td>Level of noise generated by the recirculation air diffuser, dBA</td>
<td>43.9-46.4 39.9-40.4 &lt;35</td>
</tr>
</tbody>
</table>

DISCUSSION

Based on the performed tests of an experimental installation of a local recirculation air diffuser it was concluded that:

1) The designed LRD system ensures the recirculation air and outdoor air ratio (4:1) (the necessity of maintaining the stated ratio value is substantiated in work [11]).

2) The air drag at the outdoor and recirculation air circuits has a low value, which ensures low demand for electric power for supplying outdoor air to the working area of the room;

3) The review of the obtained acoustic test results shows that, at the distance of 3.5 m, within the area of the sound direct impact, the sound pressure level generated by the local recirculation air diffuser will come to 45 dB(A) which complies with the requirements of Russian standard [19] for any types of workplaces of administration rooms; and at the distance of 5 m, within the area of direct sound impact, the sound pressure level generated by the local recirculation air diffuser will come to 40 dB(A).

Supplementary research has shown that the main contribution to the total noise exposure, created by the local recirculation air diffuser, is made by the fan, and that the total sound level in dBA is determined by the acoustic radiation at the fan blade passage frequency. This fact has determined the replacement of the fan built into the local recirculation air diffuser, which made it
possible to ensure the sound pressure level generated by the air diffuser of below 40dB(A) at the distance of 2-2.5 m.

4) The tests performed in the office rooms have confirmed automatic attainment of comfort microclimate parameters in cold and transitional seasons in the working area of the room (air temperature and velocity, sound pressure) regulated by the national standard [18], in case of using ventilation systems with local recirculation air diffuser in the conditions of the changing temperature of outdoor air supplied to LRD, as well as the changing recirculation air temperature (due to the change of the heat load in the room).

CONCLUSION
The performed tests have confirmed the assurance of comfortable air-heat parameters in the room serviced by the ventilation system with local recirculation air diffusers. For further investigation and use of the designed ventilation system with a local recirculation air diffuser, a pilot project of the considered system for office rooms shall be developed. This project shall be afterwards implemented and the installed ventilation system with local recirculation air diffusers shall be audited with the aim to identify the following:
- actual energy efficiency of the system in real conditions;
- actual employees’ satisfaction with the air-heat and acoustic comfort maintained by the system in the rooms.

ACKNOWLEDGEMENTS
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