3D modeling of supply and exhaust ventilation in the concept of passive building in the software package Autodesk 3ds Max

O A Korol, A A Petrov and K A Shreiber
Moscow State University of Civil Engineering, Yaroslavskoe shosse 26, Moscow, 129337, Russia
E-mail: ProfessorKorol@mail.ru, alex08_96@mail.ru, konstantin.shreiber@yandex.ru

Abstract. The main issue of the study is the problem of the possibility and feasibility of 3D modeling, and the use of supply and exhaust systems for heat and moisture recovery in the design of a passive building, using a 2-stage scheme of air heating in climatic and geographical conditions for the city of Kaliningrad. After all, using this mechanical ventilation and air heating scheme in a passive building, it is possible to reduce annual energy costs by 3-4 times. The higher capital costs of a passive building are offset by lower operating costs due to a significant reduction in energy consumption.

1. Introduction
The actual problem is to reduce the cost of maintenance of engineering systems of buildings and structures. Such costs include the transportation of energy resources from source to consumer, the complexity of heat supply systems due to the large number of equipment, and as a consequence, the presence of excessive heat losses. One of the solutions is the construction of buildings that meet modern energy efficiency requirements. [1-7]

This article discusses a set of parameters that contribute to the implementation of the concept of "Passive building". Application of mechanical supply and exhaust ventilation as an alternative production of thermal energy for the heat supply system (fig. 1). The main characteristics for heat and moisture recovery systems are established. It is revealed that the costs of heating and air conditioning will not exceed the set maximum [7-12].
Figure 1. 3D modeling of air ducts supply and exhaust ventilation, in the software package Autodesk 3ds Max.

2. Materials and Methods
The implementation of energy efficiency measures can significantly reduce the amount of energy spent on utilities. Energy-saving effect is achieved due to the total architectural, construction and engineering solutions aimed at saving energy. At the engineering design stage, we achieve energy efficiency through the following recommendations [13-18]:

- Ventilation of premises organized by mechanical supply and exhaust ventilation with stimuli of movement of supply and exhaust air with obligatory use of heat of exhaust air on heating of supply external air (application of installation of utilization allows to reduce heat consumption on heating and ventilation of the passive building to 61,3% in one step); [19]
- For use in Windows of thermally insulated energy-saving triple glazing with two low-emission coatings and filling with inert gas. In such a double-glazed window there are two chambers with inert gas filling, and two low-emission coatings. The heat transfer coefficient is: \( k = 0.5 \text{ W} / \text{m}^2 * ^\circ \text{C} \);
- Increasing the thickness of the insulation in excess of minimum sanitary standards in 4-5 times.

The energy efficiency indicator of the facility is the amount of heat energy consumed [14-16] per square meter (kWh / m²) during the year or during the heating period. In a passive building, the index does not exceed 15 kWh / m² [5,19].

Installation of supply and exhaust ventilation with heat and moisture recovery helps to achieve a comfortable microclimate. This system humidifies the air to the sanitary standard that is needed in an urban setting. [19-24] to significantly reduce the ever-increasing demand for energy, it is necessary to use different approaches that limit its consumption. It is necessary to reduce heat loss. Only a combination of several solutions gives a rational economy. [24-31]

The purpose of this study is the possibility and feasibility of using heat and moisture recovery systems in the design of passive buildings in climatic and geographical conditions for the city in this climatic zone.

To achieve this goal it is necessary to solve the following tasks:
- Determine the optimal location in the space of a given geographical location;
- Identify technical solutions to reduce energy consumption;
• Perform a feasibility study of the selected technical solutions.

To calculate heat losses, the value of the inverse reduced resistance to heat transfer, which is called the heat transfer coefficient, is used. To calculate the amount of heat (kWh) passing through 1 m² of the outer wall, the K value is multiplied by the number of days with an average daily air temperature ≤ 10°C and the average temperature difference during the heating period. Therefore, we obtain the total loss of thermal energy through 1 m² of the outer wall with the value of the heat transfer coefficient equal to

$$K = \frac{1}{2.8} = 0.357 \text{ W/m}^2 \cdot ^\circ\text{C};$$

(1)

Calculate the average heat loss for the heating period per 1 m² in kWh:

$$Q = \frac{K \cdot (t_{in} - t_{hp}) \cdot z_{hp} \cdot 2.4}{1000} = \frac{0.357 \cdot (22 - 2.1) \cdot 213 \cdot 2.4}{1000} = 36 \text{ kWh/m}^2;$$

(2)

Where:
- $t_{in}$ - the internal temperature of the air in the room, taken by the sanitary norms, is equal to 22 °C;
- $t_{hp}$ - average outdoor temperature for the heating period ($t_{on}=21 ^\circ\text{C}$);
- $Z_{hp}$ - duration in days of the period with average daily air temperature ≤ 10°C ($Z_{on} = 213$ days);
- 24 - number of hours per day;
- 1000 - heat transfer conversion factor (from W to kW).

3. Results and Discussion

In the calculated climatic conditions the outside air entering the room during the cold period for this climatic zone has an inflow temperature, ($t_{in}=-19 ^\circ\text{C}$) equal to the outside temperature.

Optimum parameters of exhaust air to be removed:
- temperature $t_{y1} = 23 ^\circ\text{C}$;
- humidity $\varphi_{y1} = 39$ %;
- moisture content $d_{y1} = 7 \frac{g}{kg}$; enthalpy $l_{y1} = 41 \frac{kJ}{kg}$ [14].

To avoid freezing of the condensate falling out at cooling and drying of exhaust air, we accept the recommended temperature of the cooled exhaust air $t_{y2} = 4 ^\circ\text{C}$. The exhaust air serves as a source of low-potential heat for the heat pump evaporator (which is the heat exchanger in the exhaust air: acts as an evaporator in winter and a condenser in summer).

The parameters for the required thermal efficiency air cooled heat exchangers in the cooling and dehumidification of air at the relatively "dry cooling" defined by the construction of I-d – diagram is possible for appropriate heating mode of inlet outdoor air is extracted by heat of exhaust air in the disposal facility with intermediate heat carrier antifreeze (table 1):

| Points on the chart | $t$ °C | $P_n$ | $Y_{dry1}$ | $Y_{dry2}$ | $Y_1$ | $Y_2$ | $f$ |
|---------------------|-------|-------|------------|------------|-------|-------|-----|
| Temperature         |       |       |            |            |       |       |     |
| Humidity            |       |       |            |            |       |       |     |
| Moisture            |       |       |            |            |       |       |     |
| Enthalpy            | $i (kJ)$ | $Y_{f}$ |            |            |       |       |     |
| Temperature         |       |       |            |            |       |       |     |
| Humidity            |       |       |            |            |       |       |     |
| Moisture            |       |       |            |            |       |       |     |
| Enthalpy            | $i (kJ)$ | $Y_{f}$ |            |            |       |       |     |

| Temperature         | $t$ °C | $P_n$ | $Y_{dry1}$ | $Y_{dry2}$ | $Y_1$ | $Y_2$ | $f$ |
|---------------------|-------|-------|------------|------------|-------|-------|-----|
| Temperature         |       |       |            |            |       |       |     |
| Humidity            |       |       |            |            |       |       |     |
| Moisture            |       |       |            |            |       |       |     |
| Enthalpy            | $i (kJ)$ | $Y_{f}$ |            |            |       |       |     |

The order of construction on the i-d diagram:

• Determine the initial parameters of the cooled air:
  - $t_{y1} = 23 ^\circ\text{C}; \varphi_{y1} = 39$ %; $d_{y1} = 7 \frac{g}{kg}$; $l_{y1} = 41 \frac{kJ}{kg}$;

• Determine the final parameters of the cooled air:
  - $t_{y2} = 4 ^\circ\text{C}; \varphi_{y2} = 88$ %; $d_{y2} = 4.7 \frac{g}{kg}$; $l_{y2} = 15.7 \frac{kJ}{kg}$;
The possible relative humidity \( \varphi_{y2} \) should be taken depending on the initial relative humidity of the cooled air \( (\varphi_{y1} = 39\%) \): for \( \varphi_{y1} \) up to 40% it is taken \( \varphi_{y2} = 88\% \).

- Next, it is necessary to connect the straight lines \( p \cdot Y_1 \) and \( Y_2 \), continue the straight line until the intersection with the curve \( \varphi = 100\% \) and at the intersection get \( p \cdot f \).

\( p \cdot f \) corresponds to the average conditional temperature on the surface of the air cooler: \( t_f = 20^\circ C \) (is the dew point temperature of the air);

- From \( p \cdot f \) on the vertical line of constant moisture content \( d_f = 4.4 \) g / kg, one should find the points of intersection with the enthalpies \( l_y1 \) and \( l_y2 \) with temperatures \( t_{y, dry1} \) and \( t_{y, dry2} \).

Therefore, according to the construction on the I-d diagram, we obtained conditionally dry cooling temperatures \( t_{y, dry1} = 29.6^\circ C \) and \( t_{y, dry2} = 5^\circ C \).

As a cooling medium, we use antifreeze with a freezing temperature not higher than the outdoor temperature \( t_{oc} = -19^\circ C \). Accordingly, we take antifreeze with \( t_{af} = -20^\circ C \).

The average heat capacity at freezing temperature \( t_{af} = -20^\circ C \) of an aqueous solution of ethylene glycol - C\(_2\)H\(_4\)(OH)\(_2\) (the main coolant antifreeze) [15]:

\[
c_{af} = 3.51 \frac{kJ}{kg \cdot ^\circ C}
\]

The initial temperature of antifreeze should be taken not lower, than \( t_{af} = -4^\circ C \) for reliable operation of the installation and to prevent freezing of condensate in the exhaust air stream; \( t_{af1} = -3^\circ C \)

3.1. Heat engineering efficiency of the heat exchanger

The regime of "dry" air cooling (with constant moisture content):

\[
\theta_{t, y, dry} = \frac{t_{1} - t_{2}}{t_{1} - t_{w,t}} = \frac{29.6 - 5}{29.6 + 3} \approx 0.75
\]

Where: \( t_{w,t} = -3^\circ C \) - initial temperature of the cooling source.

An indicator of the heat engineering efficiency of a heat-extracting heat exchanger in an exhaust unit at a given antifreeze temperature is necessary for implementing the heat extraction mode.

By plotting on a graph to find the heat engineering efficiency indicator in a heat exchanger, we get the following data:

For \( \theta_{t, y, dry} = 0.75 \) and \( W = 0.3 \) (indicator of the number of heat capacity of the flows) the required indicator \( N_t = 1.62 \) (indicator of the number of units of transfer of apparent heat).

With an increase of \( N_t \), the \( \theta_t \) is significantly reduced. Therefore, for a sufficiently deep extraction of heat from the exhaust air, I accept the recommended value of \( N_t = 1.5 \) for \( W \approx 0.25 \) \( \theta_t = 0.73 \).

Desired antifreeze temperature:

\[
t_{af1} = t_{y, dry1} \frac{t_{y, dry1} - t_{y, dry2}}{\theta_t} \approx 29.6 \cdot \frac{29.6 - 5}{0.73} = -4^\circ C
\]

The resulting antifreeze temperature corresponds to the permissible value.

Desired heat exchanger surface:

\[
F_{hn} = \frac{N_t \cdot \rho \cdot c_p}{k \cdot 3.6} = \frac{1.5 \cdot 1.20 \cdot 1.20}{4600 \cdot 3.6} = 0.06 \ m^2
\]

Where: \( \rho \) - mass density of air \( (\rho = 1.26 \ kg / m^3); \)

\( c_p \) - heat capacity of air \( (c_p = 1 KJ / kg \cdot ^\circ C); \)

\( L \) - air flow through the heat exchanger \( (L = 540 \ m^3 / h); \)

\( k \) - heat transfer coefficient in the heat exchanger \( (k = 4600 \ W / m^2 \cdot ^\circ C) \)

The required air capacity of the supply unit is the minimum inflow required by sanitary standards for the considered two-story individual residential building:

\[
L_{tn} = F_{tot} \cdot 3 = 180 \ m^2 \cdot 3 = 540 \ m^3 / h
\]

At the request of sanitary standards, the supply air in a specific amount of 3 m\(^3\)/h for 1 m\(^2\),

Where: \( F_{tot} \) - the total area for the required influx (1-2 floor).
We accept the productivity: \( L_y = L_{\text{th}} = 540 \, m^3/h \), \( \rho_y = 1,22 \frac{kg}{m^3} \) and
\( \rho_{\text{th}} = 1,36 \frac{kg}{m^3} \).

Supply air heating temperature with utilized heat:
\[
t_{u2} = t_{u1} + \frac{L_y \cdot \rho_y \cdot (t_{y1} - t_{y2})}{t_{\text{in}} \cdot \rho_{\text{th}} \cdot c_p} = -19 + \frac{540 \cdot 1,22 \cdot (41-15,7)}{540 \cdot 1,36 \cdot 1} = + 3,69 \, ^\circ C.
\]

We calculate the required amount of heat to heat the fresh air supply:
\[
t_{u2} = + 3,69 \, ^\circ C \text{ to } t_{u1} = t_{\text{in}} = 20 \, ^\circ C:
q_{\text{r,th}} = L_{\text{th}} \cdot \rho \cdot c_p \cdot \frac{(t_{\text{in}} - t_{u2})}{3,6} = 540 \, m^3/h \cdot 1,26 \frac{kg}{m^3} \cdot 1 \cdot \frac{(20-3,69)}{3,6} = 3049,6 W \cdot h
\]
Where: \( \rho \) – mass density of air (\( \rho = 1,26 \frac{kg}{m^3} \)); \( c_p \) – heat capacity of air (\( c_p = 1 \, kJ/kg \cdot ^\circ C \));

Without the use of a recycling plant, the amount of heat will be required to heat the external air sanitation:
\[
q_{\text{r,th}} = L_{\text{th}} \cdot \rho_{\text{th}} \cdot c_p \cdot \frac{(t_{\text{in}} - t_{u2})}{3,6} = 540 \, m^3/h \cdot 1,36 \frac{kg}{m^3} \cdot 1 \cdot \frac{(20+19)}{3,6} = 7870,84 \, W \cdot h.
\]

The use of a recycling plant reduces the heat consumption for heating and ventilation of a family house:
\[
\frac{7870,84 \, W \cdot h \cdot 3049,6 \, W \cdot h}{7870,84 \, W \cdot h} \cdot 100 = 61,3\%
\]

To find the required surface \( F_{\text{nh}} \) of the heat-transfer heat exchanger in the supply unit at a given indicator \( W_{\text{af}} = 0,25 \), we calculate the required flow rate of the circulating antifreeze:
\[
G_{\text{af}} = \frac{L_y \cdot \rho_y \cdot c_p}{W_{\text{af}} \cdot c_f} = \frac{540 \cdot 1,22 \cdot 1}{0,25 \cdot 3,51} = 742,7 \, kg/h.
\]

The required temperature difference of antifreeze to extract the estimated amount of heat from the exhaust air:
\[
\Delta t_{af} = \frac{L_y \cdot \rho_y \cdot (t_{y1} - t_{y2})}{G_{\text{af}} \cdot c_f} = \frac{540 \cdot 1,22 \cdot (41-15,7)}{742,7 \cdot 3,51} = 6,3 \, ^\circ C.
\]

We calculate the temperature of the heated antifreeze extracted from the exhaust air:
\[
t_{af2} = t_{af1} + \Delta t_{af} = -4 + 6,3 = +2,3 \, ^\circ C.
\]

The required efficiency of the heat-transfer heat exchanger in the supply unit:
\[
\theta_{t,\text{har}} = \frac{t_{u2} - t_{u1}}{t_{w1} - t_{u1}} = \frac{3,69 + 19}{2,3 + 19} \approx 1,065,
\]
Where: initial temperature of the heating source \( t_{w1} = 2,3 \, ^\circ C \).

For \( t_{y,\text{dry}} = 0,75 \, ^\circ C = 0,3 \) (indicator of the number of heat capacity of the flows) the required indicator \( N_t = 1,32 \) (indicator of the number of units of apparent heat transfer).

Desired antifreeze temperature:
\[
t_{af1} = t_{y,\text{dry1}} = \frac{L_y \cdot t_{dry1} - t_{dry2}}{\theta_t} = 29,6 - \frac{29,6 - 5}{0,75} \approx -3 \, ^\circ C.
\]

The resulting antifreeze temperature corresponds to the permissible value:
\[
F_{\text{nh}} = \frac{N_t \cdot L \cdot \rho \cdot c_p}{k \cdot 3,6} = \frac{1,32 \cdot 540 \cdot 1,26 \cdot 1}{4600 \cdot 3,6} \approx 0,05 \, m^2.
\]

Where: \( \rho \) – mass density of air (\( \rho = 1,26 \frac{kg}{m^3} \));

\( c_p \) – heat capacity of air (\( c_p = 1 \, kJ/kg \cdot ^\circ C \));

\( L \) – air flow through the heat exchanger (\( L = 540 \, m^3/h \));

\( k \) – heat transfer coefficient in the heat exchanger (\( k = 4600 \, W/m^2 \cdot ^\circ C \)).

Supply air heating temperature with utilized heat:
\[
t_{u2} = t_{u1} + \frac{L_y \cdot \rho_y \cdot (t_{y1} - t_{y2})}{t_{\text{in}} \cdot \rho_{\text{th}} \cdot c_p} = -19 + \frac{540 \cdot 1,22 \cdot (41-15,5)}{540 \cdot 1,36 \cdot 1} \approx + 3 \, ^\circ C.
\]

Where \( \rho_y = 1,22 \frac{kg}{m^3} \) and \( \rho_{\text{th}} = 1,36 \frac{kg}{m^3} \).

We calculate the required amount of heat to heat the fresh air supply:
natural exhaust ventilation, not compensated by heated supply air, is determined by the formula:

\[ W_{\text{воздуха}} = \left( \frac{t_{\text{вх}} - t_{\text{вх}} + \Delta t_{\text{воздуха}}}{3.6} \right) \cdot \frac{0.3 \cdot \rho \cdot c}{m^3} = 540 m^3/h \cdot 1.26 kg/m^3 \cdot 1 \cdot \left( \frac{22 - 3}{3.6} \right) = 3665.34 W \cdot h \]

Where: \( \rho \) – mass density of air \((\rho = 1.26 kg/m^3); C_p \) – heat capacity of air \((C_p = 1 kJ/kg \cdot °C)\);

Without the use of a recycling plant, the amount of heat will be required to heat the external air sanitation:

\[ W_{\text{св}} = L_{\text{ин}} \cdot \rho \cdot c \cdot (t_{\text{вх}} - t_{\text{вх}} + \Delta t_{\text{воздуха}}) \approx 3665.34 W \cdot h \]

Where: \( \Delta t_{\text{воздуха}} \) – additional heat loss with respect to cardinal points:

\[ \Delta t_{\text{воздуха}} = 5\% \]

The required antifreeze differential for extracting the calculated amount of heat from the exhaust air:

\[ \Delta t_{\text{af}} = L_{\text{ин}} \cdot \rho \cdot c \cdot (t_{\text{вх}} - t_{\text{вх}} + \Delta t_{\text{воздуха}}) = 540 \cdot 1.22 \cdot 1 \cdot \left( \frac{22 + 19}{3.6} \right) = 8274.47 W \cdot h \]

The use of air heat recovery unit (recuperator) can reduce heat consumption:

\[ \frac{8274.47 W \cdot h}{8274.47 W \cdot h} \cdot 100 = 55\% \]

To find the required surface \( F_{\text{ин}} \) of the heat-transfer heat exchanger in the supply unit for a given indicator \( W_{\text{ап}} = 0.3 \), we calculate the required flow rate of the circulating antifreeze:

\[ G_{\text{af}} = \frac{L_{\text{ин}} \cdot \rho \cdot C_p}{W_{\text{ap}} \cdot C_{\text{af}}} = \frac{540 \cdot 1.22 \cdot 1}{0.3 \cdot 3.51} = 618.9 kg/h. \]

The required antifreeze differential for extracting the calculated amount of heat from the exhaust air:

\[ \Delta t_{\text{af}} = L_{\text{ин}} \cdot \rho \cdot c \cdot (t_{\text{вх}} - t_{\text{вх}} + \Delta t_{\text{воздуха}}) = 540 \cdot 1.22 \cdot \left( \frac{22 + 19}{3.6} \right) = 7.3 °C. \]

We calculate the temperature of the heated antifreeze extracted from the exhaust air:

\[ t_{\text{af}2} = t_{\text{af}1} + \Delta t_{\text{af}} = -3 + 7.3 = 4.3 °C. \]

The required efficiency of the heat-transfer heat exchanger in the supply unit:

\[ \theta_{\text{транс}} = \frac{t_{\text{вх}} - t_{\text{вх}} + \Delta t_{\text{воздуха}}}{t_{\text{вх}2} - t_{\text{вх}1}} = \frac{3 + 4.3}{4.3 + 19} \approx 0.94 \]

Where: initial temperature of the heating source \( t_{\text{вх1}} \approx 4.3 °C. \)

**Calculation of heat loss of a building (table 2)**

The thermal power of the heating system is determined for each room by the formula:

\[ Q_{\text{ц.о.}} = Q_{\text{вн}} + Q_{\text{ин}} - Q_{\text{дс}} \]

where: \( Q_{\text{вн}} \) – heat loss through the building envelope, W; \( Q_{\text{ин}} \) – heat loss for heating the infiltrating air entering through windows, gates, slots, W; \( Q_{\text{дс}} \) – heat from domestic sources, W;

We number the rooms with a 3-digit number (the first digit is the floor, the third is the room number) from the upper left further clockwise. The staircase is indicated by the letters – SC.

We introduce the following notation: EW - external wall; WI - a window; CE- ceiling; FL - floor; DD - double door; CR - corner room; OR is an ordinary room.

Heat losses through the building envelope are determined by the following formula:

\[ Q_{\text{вн}} = F n \left( t_a - t_s \right) \left( 1 + \sum \beta \right) \]

where: \( F \) – area of the fence, m²; \( n \) – coefficient taking into account the position of the outer enclosing structure with respect to the outside air; \( k' \) – heat transfer coefficient of the fence (calculated for walls, floor, ceiling, windows).

For other enclosing structures \( k' \) for external double doors \( k' = 2.3 \), \( t_a \) – temperature of internal air, °C; \( t_s \) – internal air temperature, °C; \( \sum \beta \) – additional heat loss.

where: \( \beta_1 \) – additional heat loss with respect to cardinal points: N, E, N-E, N-W = 10% – \( \beta_1 = 0.1 \); W, S-E = 5% – \( \beta_1 = 0.05 \); S, S-W = 0% – \( \beta_1 = 0 \)

The cost of heat for heating the infiltrating air in rooms in residential and public buildings with natural exhaust ventilation, not compensated by heated supply air, is determined by the formula:

\[ Q_{\text{ин}} = 0.28 \cdot L \cdot \rho \cdot c \cdot (t_a - t_s) \cdot k, \]
where \( L \) – volumetric flow rate of the removed air uncompensated by the heated supply air, \( L = 3 \text{m}^3/\text{h} \cdot \text{m}^2 \) for residential premises and kitchens; \( c \) – specific heat of air \((c = 1)\); \( \rho \) – air density in the room, \( \text{kg/m}^3 \), is determined by the formula: \( \rho = \frac{353}{(273 + t_n)} = 1.2 \text{ kg/m}^3 \);

\( k \) – coefficient of accounting for the influence of the oncoming heat flux in the structures, equal to 0.7 for the joints of wall panels and windows with triple bindings.

For residential buildings accounting of heat flow entering the rooms and kitchens in the form of household heat is made in the amount of 15 \( W \) per 1 \( m^2 \) of floor space for a passive building:

\[ Q_{ds} = 15 \cdot F_n \]

where: \( F_n \) – floor area of the room, \( m^2 \).

### Table 2. The heat loss of the building.

| № of the room | \( t_{\text{в}} \), °C | Fence feature | \( a, m \) | \( b, m \) | Square (\( A \)), \( m^2 \) | \( K, W/\text{m}^2\cdot\text{°C} \) | \( n \) | \( Q_{\text{env}}, W \) | Additives | \( 1 + \sum \beta \) | \( Q_{\text{inf}}, W \) | \( Q_{\text{ds}}, W \) | \( \Sigma Q_{\infty} \) |
|--------------|----------------|---------------|----------|---------|---------------------|---------------------------|--------|---------------------|------------|------------------------|-----------------|----------------------|-----------------|
| 105          | 22             | EW1 W         | 5,45     | 2,7     | 12,4               | 0,3                       | 7      | 1                   | 193        | 0.05                   | 0               | 1                    | 10.5            | 394,6              | 620,8           |
|              |                | EW2 S         | 2,5      | 2,7     | 4,5                | 0,3                       | 6      | 1                   | 41         | 66                      | 0               | 1                    | 1               |                     |
|              |                | WI1 W         | 1,5      | 1,5     | 2,25               | 0,5                       |        | 1                   | 41         | 48                      | 0.05            | 1                    | 10.5            | 1                   |
|              |                | WI2 S         | 1,5      | 1,5     | 2,25               | 0,5                       |        | 1                   | 41         | 46                      | 0               | 1                    | 1               |                     |
|              |                | FL            | 2,5      | 5,45    | 13,6               | 0,2                       | 3      | 0.6                 | 24,6       | 77                      | 0               | 1                    | 1               | 204,6              |
| 106          | 22             | EW1 W         | 5,75     | 5       | 2,7                | 13,3                      | 0,3    | 6                  | 193        | 0.05                   | 0               | 1                    | 10.5            | 416,3              | 660             |
|              |                | EW2 N         | 2,5      | 2,7     | 4,5                | 0,3                       | 6      | 1                   | 41         | 73                      | 0.1             | 1                    | 1               |                     |
|              |                | WI1 W         | 1,5      | 1,5     | 2,25               | 0,5                       |        | 1                   | 41         | 48                      | 0.05            | 1                    | 10.5            | 1                   |
|              |                | WI2 N         | 1,5      | 1,5     | 2,25               | 0,5                       |        | 1                   | 41         | 50                      | 0.1             | 1                    | 1               |                     |
|              |                | FL            | 5,75     | 5       | 14,3               | 0,2                       | 9      | 0.6                 | 24,6       | 81                      | 0               | 1                    | 1               | 215,8              |

As a result, the value of heat loss was 61,716 kWh/m², and in a passive building heat loss does not exceed 15 kWh/m² [5], it is necessary to reduce them with the help of heat recovery unit (recuperator).

Heat loss of the building taking into account the use of heat recovery in one stage amounted to 35,795 kWh/m², it is necessary to reduce them with the help of additional installation of heat recovery in two stages (sequential installation of recuperators). It is also necessary to increase the thickness of the insulation of the wall, floor, ceiling [4].

Heat loss of the building, taking into account the 2-stage scheme of air heating (fig. 2) in the ventilation system and additional insulation of the enclosing structures will amount to 13,565 kWh/m², then the energy efficiency index will not exceed 15 kWh/m², therefore the building will be passive.
4. Conclusions
Summing up the above, we came to the conclusion that the use of this supply and exhaust ventilation system, in the design and 3D modeling of passive buildings, in climatic and geographical conditions for the city in this climatic zone, is possible and appropriate. After all, the use of this 2-stage air heating scheme for the city of Kaliningrad allows to reduce annual energy costs from 61,716 kWh/m² * year) to 13,565 kWh/(m² *year).

It is economically advantageous that the costs for the life cycle of a passive building, i.e. the total cost of design, construction, operation for 30 years [16], will not exceed the costs for the life cycle of a traditional residential building. Higher capital costs of a passive building are offset by lower operating costs due to a significant reduction in energy consumption [17].

Summarizing the work done, using a 2-step scheme of air heating in the concept of a passive building, in the city of this climatic zone, it is possible to create conditions for more profitable energy savings.

References
[1] Ehhorn H., Reiss J., Kluttig H., Hellwig R. Efficient building. Analysis of the current state and prospects of development on the basis of implemented projects. AVOC. No.2. 2006.
[2] Tabunshchikov Yu. A., Brodach M. M., Shilkin N. V. Energy Efficient buildings. AVOC. Moscow, 2003
[3] Tabunshchikov Yu. a. Microclimate and energy saving: it's time to understand the priorities. AVOC. No. 5. 2008.
[4] Gagarin V. G. Methods of economic analysis of increasing the level of thermal protection of enclosing structures of buildings. AVOC. No. 1-3, 2009.
[5] Kostenko V.A., Gafiyatullina N.M., Semchuk A.A., Kukolev M.I. Geothermal heat pump in the passive house concept. Magazine of Civil Engineering. 2016. No. 8. P. 18–25
[6] Wang L., Gwilliam J. Case study of zero energy house design in UK. Energy and Buildings. 2009. No. 41. P. 1215–1222
[7] Borkovskaya V.G. Environmental and economic model life cycle of buildings based on the concept of “Green Building”. Applied Mechanics and Materials 467. Materials Science and Mechanical Engineering. Chapter 2: Building Materials and Construction Technologies. P. 287-290. 2013. DOI: 10.4028/www.scientific.net/AMM.467.287
[8] Karlsson J.F., Moshfegh B. Energy demand and indoor climate in a low energy building – changed control strategies and boundary conditions. Energy and Buildings. 2006. No. 38. Pp. 315–326
[9] Chlela F., Husaunndee A., Inard C., Riederer P. A new methodology for the design of low energy buildings. Energy and Buildings. 2009. No. 41. P. 982–990
[10] Parker D.S. Very low energy homes in the United States: perspectives on performance from measured data. Energy and Buildings. 2009. No. 5(41). P. 512–520
[11] Gorshkov A. S., Dergunov D. V., Zavgorodny V. V. Technology and organization of building construction with zero energy consumption. Construction of unique buildings and structures. 2013. No. 3. P. 12-23
[12] Gorshkov A.S., Gladkikh A.A. Measures to improve energy efficiency in construction. Academia. Architecture and Construction. 2010. No. 3. P. 246–250
[13] Borkovskaya V.G. Complex models of active control systems at the modern developing enterprises. Advanced Materials Research V.945-949. Chapter 22: Manufacturing Management and Engineering Management. June 2014. P. 3012-3015. DOI: 10.4028/www.scientific.net/AMR.945-949.3012
[14] Borkovskaya V.G. Environmental and economic model life cycle of buildings based on the concept of “Green Building”. Applied Mechanics and Materials 467. Materials Science and Mechanical Engineering. Chapter 2: Building Materials and Construction Technologies. P. 287-290. 2013. DOI: 10.4028/www.scientific.net/AMM.467.287
[15] Borkovskaya V.G. Post bifurcations of the concept of the sustainable development in construction business and education. Advanced Materials Research V.860-863. Chapter 26: Engineering Education. P. 3009-3012. 2013. DOI: 10.4028/www.scientific.net/AMR.860-863.3009
[16] Borkovskaya V.G. The concept of innovation for sustainable development in the construction business and education. Applied Mechanics and Materials. V.475-476. Chapter 15: Engineering Management. 2013. P. 1703-1706. DOI: 10.4028/www.scientific.net/AMM.475-476.1703
[17] Borkovskaya V.G. Project Management Risks in the Sphere of Housing and Communal Services. MATEC Web of Conferences, Volume 251, 06025 (2018). DOI: https://doi.org/10.1051/matecconf/201825106025
[18] Borkovskaya V.G., Lyapunstova E.V., Nogovitsyn M. Risks and safety in construction by increasing efficiency of investments. E3S Web of Conferences. V.97 .2019. 06036. DOI: https://doi.org/10.1051/e3sconf/20199706036
[19] Polyakova V, Degaev E., El Haddad P. Reduction of Ecological and Economic Risks in
Utilization of Solid Domestic Wastes and Construction Waste. MATEC Web of Conferences Volume 251 2018. DOI: https://doi.org/10.1051/matecconf/201825106017

[20] Degaev E., Orlov A., El Haddad P., Pleshivtsev A.. Ecological and Economic Risks of Fire Protection of Warehouses and Tank Parks. MATEC Web of Conferences Volume 251 2018. DOI: https://doi.org/10.1051/matecconf/201825106013

[21] Degaev E. New classification of foaming agents for fire extinguishing. 2018 MATEC Web of Conferences

[22] Russian State Standard GOST 30494-2011. Residential and public buildings. Microclimate parameters in the premises. Moscow, 2013

[23] Flaga-Maryanczyka A., Schnotalea J., Radonb J., Was K. Experimental measurements and CFD simulation of a ground source heat exchanger operating at a cold climate for a passive house ventilation system. Energy and Buildings. 2014. No. 68. P. 562–570.

[24] Korol, E., Shushunova N. Benefits of a Modular Green Roof Technology. Procedia Engineering, Volume 161, 2016, pp. 1820-1826.

[25] Korol, E., Shushunova N. Research and Development for the International Standardization of Green Roof Systems. Procedia Engineering, Volume 153, 2016, p. 287-291.

[26] Korol, E., Shushunova, N., Rerikh, S. New green roof and green wall systems for implementation in the coverings. E3S Web of Conferences 97, 06023 (2019) https://doi.org/10.1051/e3sconf/20199706023

[27] Korol, O., Shushunova, N., Lopatkin, D., Zanin, A., Shushunova, T., Application of High-tech Solutions in Ecodevelopment. MATEC Web of Conferences 251:06002, 2018, DOI: 16.1051/matecconf/201825106002

[28] Korol E., Shushunova N. Green roofs: standardization and quality control of processes in green construction. MATEC Web of Conferences Cep. "International Science Conference SPbWOSCE-2016 "SMART City"" 2017. C. 06014.

[29] Korol O.A. The impact of energy-corrective measures at the design stage of construction projects. 2018. MATEC Web of Conference.

[30] Pleshivtsev A., Korol O., Barkhi R. Risks on Optimization of Life Cycle of Technology of Installation of Transformed Low-rise Buildings from Sandwich Panels. MATEC Web of Conferences Volume 251 (2018). DOI: https://doi.org/10.1051/matecconf/201825106024

[31] Korol E.A., Korol O.A. Modelling of energy consumption at construction of high-rise buildings. 2018. E3S Web of Conferences.