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Economic Comparison of Two Energy-Efficient Design Strategies for Mechanical Ventilation Systems

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Heat exchangers · Design strategy · Energy saving

Abstract
Two systems, one with air re-circulation alone and the other a combination of air re-circulation and energy recovery using plate heat exchangers have been analysed. For these systems, a set of equations was derived for partial or full re-circulation of air with and without plate heat exchangers. Also, as part of the analysis, the reciprocal ratio of investment and running costs was considered. In this analysis, the re-circulation factor provided by ventilation units, a bypass factor and the efficiency of a plate heat exchanger were considered to be variables. For the sake of simplicity, the outdoor temperature was assumed to change uniformly throughout the year from a winter time minimum to a summer time maximum. All the final mathematical expressions derived are non-dimensional. The technical and economic estimations proposed were those which gave the most pessimistic outcome, that is, those for which the outcome gave the lowest guaranteed common profit. The numerical calculations showed that quite different results were obtained depending on specific applications. Several economic indexes were estimated including payback times and capital and running costs for typical examples. The expected payback time for a system with plate heat exchanger is something like 2.0 years for an installation in the climate zone of Saint Petersburg, installed and run in accordance with Russian legislation.

Introduction

The energy consumption of a ventilation system depends essentially on climatic factors in the area where it is installed. So, according to the Air Infiltration and Ventilation Centre in Coventry (UK) [1], the annual energy consumption per unit of fresh air mass at two sites in the US is estimated to be, on average, 22.1 $MJh \cdot kg^{-1}$ for Los Angeles (Calif.) and 102.5 $MJh \cdot kg^{-1}$ for Omaha (Nebr.). In Europe, similarly derived figures range from 45.6 $MJh \cdot kg^{-1}$ up to 102.5 $MJh \cdot kg^{-1}$. No Russian references are available. However, for similar geographical conditions specific energy consumption could be assumed to be commensurable with the above figures.

In Russia, as in Europe, the main usage of energy is for the heating of fresh air, while in the US, energy is used not only for heating, but a large part is used for cooling air using air conditioning units. Some areas of Russia suffer

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extreme climatic conditions. The costs of energy for heating in these parts are, therefore, quite substantial. To give an example: on average, the annual consumption of energy in work places in the North Eastern part of Russia, where outdoor temperatures have been measured as low as −59.3 °C with strong winds adding a wind-chill factor (Verkhoyansk), is about 10,000 $kWh \cdot m^{-2}$ (8.5 $Gcal \cdot m^{-2}$). According to the Russian Integrated Costing Norms for the building trade, about 15–20% of investment costs and about 30% of running costs are accounted for by heating, ventilation and air conditioning. Conventional techniques for energy saving are widely applied and involve the reduction of heat losses through external walls and roofs, and a decrease in air exfiltration by sealing windows, doors, gates and ceilings. At the same time, there are specialised engineering solutions used involving the organisation and technical arrangement of the ventilation and air conditioning systems to ensure an appreciable decrease of energy consumption. These solutions are applied according to the following provisions: (1) optimisation of air distribution inside the ventilated premises; (2) partial or full re-circulation of air; (3) recovery using heat exchangers; (4) regeneration of latent heat from the condensation of excess moisture; (5) use of heat pumps.

According to estimations that have been made [2], these measures could reduce annual energy consumption on average by up to 2,000 $kWh \cdot m^{-2}$ (1.7 $Gcal \cdot m^{-2}$). To make such estimations, we need to analyse those elements in a system that affect the efficiency of re-circulation and recovery and which are most important for energy saving [3]. While it is known that heat pumps are also effective for energy saving [4], it has to be remembered that heat pump installation requires much greater investment costs. As a consequence, heat pumps are not widely used in Russia.

Methods

The following analysis of two cases illustrates the ventilation alternatives that are considered in this paper: (1) partial or full re-circulation of air while assuming that distribution of the air is perfect; (2) the same situation, but with heat recovery using plate heat exchangers.

The second case involves the latent heat regeneration that takes place as a result of water vapour condensation at the relatively cold surfaces of the plate heat exchanger. This phenomenon has an impact upon the efficiency factor of recovery, and consequently a distinction is made between 'dry' and 'wet' recovery efficiency.

The following analysis is overall a general one. However, particular calculations were done for high-level ventilation units of the types described as decentralised heating vortex (DHV; the 1st case) and layered heating weatherproof (LHW; the 2nd case), both of which are manufactured by Hovalwerk AG (Heating Ospelt Vaduz Liechtenstein; Schaan, Liechtenstein). These ventilation systems are considered the best in the world, implementing the most recent engineering designs concerned with energy saving.

Analysis

It is well known that a venture is economically attractive if the gross profit (E) exceeds the capital investments (I):

$$E > I$$

To better analyse this situation, knowledge of the ratio of investment and running costs is important. The initial investment costs are calculated on the basis of the equipment prices and the labour required for transportation, installation and commissioning. The running costs are concerned with the modes of operation, the quantity and costs of consumables required, energy consumption and the labour needed for maintenance, technical services and repairs.

Consider the detailed structure of the economic equation above. The gross profit is a sum of the following items:

$$E = E_W + E_B + E_Z$$

where E_W is the annual saving of energy costs, E_B is the annual saving of running, servicing and maintenance costs, and E_Z is the discount rate due to State privileges, amortisation deductions etc.

Limiting the analysis to the annual saving of energy costs, consider the schematics of two ventilation systems (fig. 1, 2) representing the two cases proposed above.

Using the following notation for the various parameters:

L = the total air flow into a room, $m^3 \cdot h^{-1}$;
Q = the total consumption of heat, $Gcal \cdot h^{-1}$ (kW);
L_1 = air flow produced by units of the type DHV, $m^3 \cdot h^{-1}$;
L_2 = air flow produced by units of either the type DHV with added fresh air (case 1) or type LHW (case 2), $m^3 \cdot h^{-1}$;
Q_1 = heat productivity of units of the type DHV, $Gcal \cdot h^{-1}$ (kW);
Q_2 = heat productivity of units of either the type DHV with added fresh air (case 1) or type LHW (case 2), $Gcal \cdot h^{-1}$ (kW);
n_1 = number of units of the type DHV;
n_2 = number of units of either the type DHV with added fresh air (case 1) or type LHW (case 2);
ε = efficiency of a plate heat exchanger;
γ = bypass factor of a plate heat exchanger.

From this, the following relationships may be written:

$$L = n_1 L_1 + n_2 L_2$$
$$Q = n_1 Q_1 + n_2 Q_2$$

Factors may be defined as ratios of the parameters of the chosen units, such that:

$$K_L = \frac{L_1}{L_2}$$

$$K_Q = \frac{Q_1}{Q_2}$$

Another factor may be defined which is the ratio of the number of each unit:

$$K_n = \frac{n_1}{n_2}$$

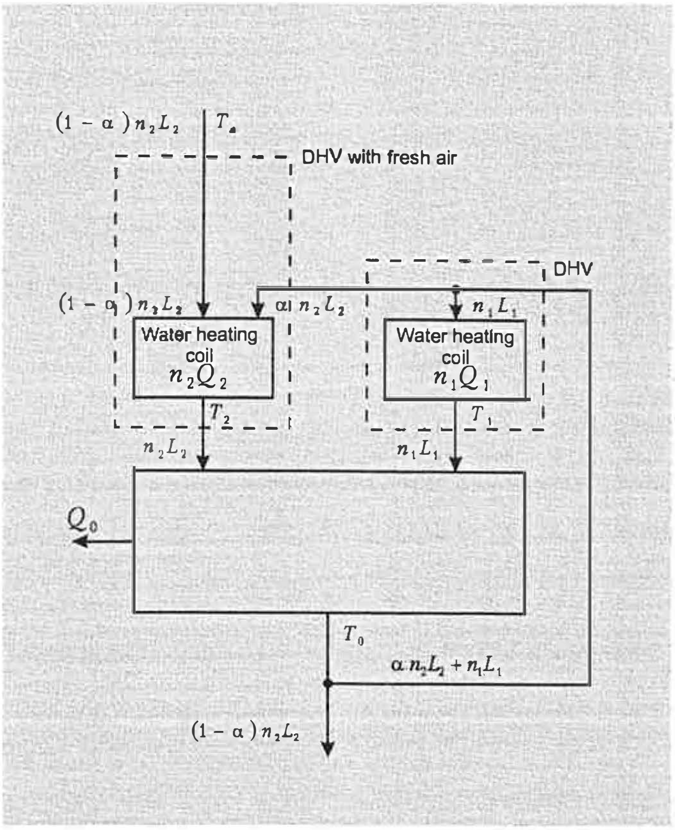

Fig. 1. Case 1 (DHV units with air re-circulation and fresh air supply).

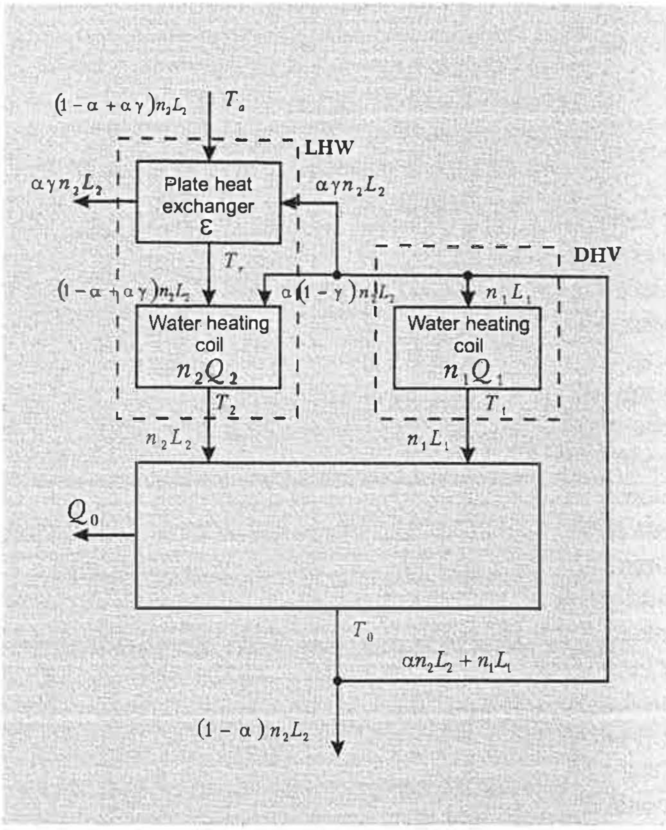

Fig. 2. Case 2 (combined LHW and DHV units with air re-circulation and fresh air supply).

The air flow delivered to premises equipped with either the units DHV or DHV and LHW, according to the schematics (fig. 1, 2) will be:

$$n_1L_1 = L\frac{K_n}{K_n + K_L}$$

$$n_2L_2 = L\frac{K_L}{K_n + K_L}$$

The amount of heat, acting through water-heating coils,

$$n_1Q_1 = Q\frac{K_n}{K_n + K_Q}$$

$$n_2Q_2 = Q\frac{K_n}{K_n + K_Q}$$

If the following parameters are designated:

Q_0 = heat losses through building enclosure components (walls, windows, roof, etc.), Gcal·h^{-1} (kW);
T_0 = indoor air temperature, K;
T_1 = temperature of air supplied by units of the type DHV, K;
T_2 = temperature of air supplied by units of the type DHV with fresh air (LHW), K;
T_a = outdoor air temperature, K;
α = re-circulation factor, provided by units of the type DHV with fresh air (LHW);
c_p = specific heat capacity of air at constant pressure, kcal·kg^{-1}, K;
ρ = air density, kg·m^{-3};

Then the equations of thermal balance for the first case may be written down as follows:

$$Q_1 = 10^{-3}c_p\rho L_1(T_1 - T_0)$$
$$Q_2 = 10^{-3}c_p\rho L_2[(1-\alpha)(T_2 - T_a + \alpha(T_2 - T_0)]$$
$$Q = n_1Q_1 + n_2Q_2 = Q_0 + c_p\rho(1-\alpha)n_2L_2(T_0 - T_a)$$

From here, the specific superfluous quantity of heat spent in this case may be written down as follows:

$$q = \frac{Q - Q_0}{c_p\rho L T_0} = \frac{K_L}{K_n + K_L}(1-\alpha)(1-\theta)$$

where θ is a temperature factor defined as:

$$\theta = \frac{T_a}{T_0}$$

In the second case, in addition, it is necessary to take into account the energy saving, provided by recovery of heat using a plate heat exchanger of a type 'air-air'. Units of a type LHW use exchangers with a dry efficiency not less than 60–65% and a wet efficiency up to

85%. In this case 2, apart from intake air flow, exhaust air flow is provided as well. The relation between intake and exhaust flows is the amount which is under control of the air-regulating dampers provided inside units of the type LHW. If $\gamma = 0$, then the unit operates in a re-circulation mode similar to a DHV. If $0 < \gamma < 1$, then partial air re-circulation occurs. If $\gamma = 1$, then the whole air flow supplied by means of the LHW unit is withdrawn to atmosphere (ventilation mode). Under these conditions, heat recovery takes place.

Hence, a heat balance equation may be written for case 2 as follows:

$$Q_1 = 10^{-3}\, c_p \rho L_1 (T_1 - T_0)$$
$$Q_2 = 10^{-3}\, c_p \rho L_2 [(1 - \alpha + \alpha\gamma)(T_2 - T_r) + \alpha(1 - \gamma)(T_2 - T_0)]$$

$$\varepsilon = \frac{\alpha\gamma(T_0 - T_a)}{(1 - \alpha - \alpha\gamma)(T_r - T_a)}$$

$$Q = n_1 Q_1 + n_2 Q_2 = Q_0 + c_p \rho [1 - \alpha(1 - \gamma)] n_2 L_2 (T_0 - T_a)$$

The specific superfluous quantity of heat spent in case 2 could be written down as follows:

$$q = \frac{Q - Q_0}{c_p \rho L T_0} = \frac{K_L}{K_n + K_L} [1 - \alpha(1 - \gamma)](1 - \theta)$$

Let us consider the ventilation without re-circulation and without recovery ($K_n = 0$, $\gamma = 0$, $\alpha = 0$) as a reference case. In this case, the dimensionless factor $q = 1 - \theta$.

Let us denote the temperature factor corresponding to the minimum outdoor temperature as θ^- and the temperature factor corresponding to the maximum outdoor temperature as θ^+.

Assume that a re-circulation factor α changes regularly over the period from 1 in a cold winter season to 0 in a hot summer season. Then:

$$\alpha = \frac{\theta^+ - \theta}{\theta^+ - \theta^-}$$

Assume that the relation between intake and exhaust flows γ is regulated similarly from 0 in a cold winter season to 1 in a hot summer season. Then:

$$\gamma = \frac{\theta - \theta^-}{\theta^+ - \theta^-}$$

In principle, these assumptions are in accord with the energy-saving possibilities which are provided for in the design of the units discussed above. However, the regulation provided through the use of in-built microprocessors and under the control of the central system DigiNet (Siemens, Landis & Staefa Division in co-operation with Hovalwerk AG) is actually non-proportional, and has more complex features, which provide a much greater efficiency when applied to energy-saving technologies. Thus, by virtue of the assumptions mentioned above, it is prudent to consider that the technical and economic estimations are the most pessimistic, i.e. those for which the outcome gives the lowest guaranteed common profit.

Assuming, as above, that throughout the year the outdoor temperature changes regularly from minimum to maximum values, we could estimate an annual superfluous quantity of heat by the following integral.

$$\overline{E}_W = \int_{\theta^-}^{\theta^+} q \cdot d\theta$$

After necessary substitutions, we have for each case:

reference case

$$\overline{E}_W = \int_{\theta^-}^{\theta^+} (1 - \theta)\, d\theta$$

case 1

$$\overline{E}_W = \frac{K_L}{K_n + K_L} \int_{\theta^-}^{\theta^+} \left(1 - \frac{\theta^+ - \theta}{\theta^+ - \theta^-}\right)(1 - \theta)\, d\theta$$

case 2

$$\overline{E}_W = \frac{K_L}{K_n + K_L} \int_{\theta^-}^{\theta^+} \left[1 - \left(\frac{\theta^+ - \theta}{\theta^+ - \theta^-}\right)^2\right](1 - \theta)\, d\theta$$

This, after integration, gives for each case:

reference case

$$\overline{E}_W = (\theta^+ - \theta^-)\frac{2 - \theta^+ - \theta^-}{2}$$

case 1

$$\overline{E}_W = \frac{K_L}{K_n + K_L}(\theta^+ - \theta^-)\frac{3 - 2\theta^+ - \theta^-}{6}$$

case 2

$$\overline{E}_W = \frac{K_L}{K_n + K_L}(\theta^+ - \theta^-)\frac{8 - 5\theta^+ - 3\theta^-}{12}$$

Relative to the reference case we have

case 1

$$\hat{E}_W = \frac{K_L}{K_n + K_L}\,\frac{3 - 2\theta^+ - \theta^-}{2 - \theta^+ - \theta^-}\,\frac{1}{3}$$

case 2

$$\hat{E}_W = \frac{K_L}{K_n + K_L}\,\frac{8 - 5\theta^+ - 3\theta^-}{2 - \theta^+ - \theta^-}\,\frac{1}{6}$$

Results

Below is a worked example of the calculations for systems in the St. Petersburg region. The ratio of the number of units of type DHV and DHV with fresh air or of type LHW is assumed to be:

$$K_n = \frac{10}{12} = 0.83$$

The ratio of the production of air by the units is:

$$K_L = \frac{8{,}700}{8{,}000} = 1.09$$

The minimum outdoor temperature for St. Petersburg is −26 °C, and the maximum outdoor temperature is

+24.8 °C. According to Russian health legislation, the design indoor temperature for the winter season is +17 °C and for the summer season is +18 °C. Therefore, the temperature factors for summer and winter, respectively, are

$$\theta^{+} = \frac{273 + 24.8}{273 + 18} = 1.023 \text{ and } \theta^{-} = \frac{273 - 26}{273 + 17} = 0.852$$

It follows that $\hat{E}_W = 0.118$ (for case 1) and $\hat{E}_W = 0.19$ (for case 2). Thus, the annual profit for a system set up according to a case 1 design will be at least 11.8% and for a case 2 it will be 19%. Using an annual energy consumption of $10{,}000\,\mathrm{kWh \cdot m^{-2}}$ for the reference case and an energy cost of 100 RUR per Gcalorie for St. Petersburg, we may calculate the annual saving of running costs to be $10{,}000 \times 1{,}000 \times 0.85 \times 10^{-3} \times 100 \times 0.118 = 100{,}300$ RUR per year with a case 1 system for a building with a floor area of 1,000 m^2. This is equal to 4,360 USD per year. The estimated cost of equipment is about 10,000 USD. Thus, the payback time is 2.39 years. Compare this to the saving of running costs in a case 2 system which is $10{,}000 \times 1{,}000 \times 0.85 \times 10^{-3} \times 100 \times 0.19 = 161{,}500$ RUR per year. This is equal to 7,020 USD per year. The estimated cost of the equipment in this case is about 14,000 USD. Thus, the payback time is 2.0 years. Consequently, using a system with a plate heat exchanger gives an energy saving of 60–65%.

Conclusion

A method has been described in detail, whereby an economic comparison between high-level ventilation units with either partial air re-circulation or those incorporating heat recovery using heat exchangers can be made. This type of study will increase in importance as energy saving techniques come into more general use. It is now possible to calculate the overall performance of mechanical ventilation systems at the design stage in order to choose the best solution. Also, to aid costing of the project, the calculated figures include payback time. This method, together with the appropriate algorithm, can constitute the basis for calculating the validity of energy-saving concepts to be implemented for any particular design.

References

1 Colliver DG: Energy Requirements for Conditioning of Ventilating Air, AIVC Technical Note 47, AIRBASE#NO 9104, September 1995.
2 Heat Recovery with Heat Pumps and Dehumidifiers in Swimming Pools, DANTERM 05.91, Technical Guide.
3 Bitcolov NZ: Present Tendencies of Air Quality Control in Ventilation and Air Cleaning Systems, Proceedings of 5th International Conference on Air Distribution in Rooms. 1996, vol 1, pp 369–378.
4 Blok K, Turkenburg WC, Eichhammer W, Farinelli U, Johansson TB: Overview of Energy RD&D Options for a Sustainable Future, Report commissioned by CEC-DG-XII (No. 95043).