ENVIRONMENTAL ENGINEERING

The 8th International Conference
May 19–20, 2011, Vilnius, Lithuania
Selected papers

ISSN 2029-7106 print / ISSN 2029-7092 online
ISBN 978-9955-28-828-2 (2 Volume)
ISBN 978-9955-28-827-5 (3 Volumes)
http://enviro.vgtu.lt

TEMPERATURE EFFICIENCY OF ROTARY HEAT RECOVERY UNITS IN VENTILATION AND AIR CONDITIONING

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Abstract. The article focuses on processes in rotary heat recovery ventilation and air conditioning units. Here we offer approximated engineering formula of thermal efficiency based on dimensionless parameters and further analysis of final results within all possible range of values. The results are illustrated graphically and with calculated data.

Keywords: heat recovery, efficiency, ventilation, air conditioning, rotary.

1. Introduction

For heat recovery in exhausting ventilation heat exchangers of different types could be used (Shen C.M.; Worek W.M.; Worsøe-Schmidt P. 1991; 1993; Белова Е. М. 2006), such as cross-current or rotary heat exchangers and systems with intermediate coolant. The thermal efficiency is a main characteristic of any heat exchange unit and its performance and quality. In most cases for heating and heated up agents these parameters differ and could be calculated as follows (Белова Е. М. 2006):

$$\varepsilon_h = \frac{t_{1h} - t_{2h}}{t_{1h} - t_{1c}},\ \varepsilon_c = \frac{t_{2c} - t_{1c}}{t_{1h} - t_{1c}},$$

Here t_{1h} and t_{2h} are inlet and outlet temperatures of the heating agent; t_{1c} and t_{2c} are the same for heated up one.

So ε_h and ε_c are dimensionless parameters. And both of them are the ratio of flow temperature variation along the heat exchange surface to the maximum temperature difference in the heat exchanger ($t_{1h} - t_{1c}$). Those values are good when we make check calculation of heat recover units. In those cases we have to determine real fresh air temperature at the outlet of heater. It is most actual during the variable outside temperature period, for example, in spring or autumn. The best efficiency is able in cases with equal supply and exhaust flow volume. In such a case ε_h and ε_c factors will be equal one another and both could be indicated as ε.

The main feature in rotary heat recovery unit's analysis is non-stationary heat transfer of each plate of the device. The temperature of each plate is changing across the surface due to the washing it air flow. Uniform rotation of heat transfer unit allows us to consider each plate as just one with steady temperature field on its surface because of permanent change of plates by next ones with the same properties. In that case we might consider all the plates of heat exchanger as a mass uniformly distributed in space of heat recovery unit. This assumption allows using heat transfer equation for rotary heat exchanger in the differential form for steady case.

Another way to simplify our mathematical model is to ignore the unbalanced distribution of temperature across the flow and take into account only temperature changes along the device. So in further heat transfer equations we have to use average across the section temperature $\theta_{r.m}$. Besides we don't take into consideration the conductivity heat transfer along the plates and use only convective part of heat exchange on the surface of plates and heat assimilated and carried by air. In addition we fully ignore the reverse air overflow because of its small effect in general conditions (Shaha R.K.; Skiepko T. 1999; Skiepko T. 1993; Белова Е. М. 2006).

2. Method

Using dimensionless extra temperatures of supply air $\theta_{in} = \frac{t_c(x') - t_{1c}}{t_{1h} - t_{1c}}$ and the same for the rotary unit $\theta_r = \frac{t_r(x',\varphi) - t_{1c}}{t_{1h} - t_{1c}}$, the system of equations for out case will be as follows:

$$\frac{d\theta_c}{dx'} = NTU \cdot \left[\theta_{r.m}(x') - \theta_c(x')\right];$$

$$\frac{\partial\theta_r}{\partial\varphi} = \frac{2}{\pi} \cdot NTU_r \cdot \left[\theta_c(x') - \theta_r(x',\varphi)\right]. \quad (1)$$

The same for exhaust part of device, but using dimensionless exhaust air temperature $\theta_h = \frac{t_h(x') - t_{1c}}{t_{1h} - t_{1c}}$. Then $\varepsilon_c = \theta_c$ when x' = 1, and $\varepsilon_h = 1 - \theta_h$ when x' = 0. Here x' = x/l is dimensionless spatial coordinate, equal to ratio of current spacing interval from device inlet (x) to the length of the heat exchanger along the air flow (l), $t_h(x')$ and $t_c(x')$ are temperatures, ^oC, of supply and exhaust air in part of unit with the coordinate x', φ is the angle of inlet fitting measured in radians.

Value $NTU = \frac{3.6 \cdot UA}{Gc}$ shows the number of heat transfer points for air flow (Белова Е. М. 2006). Here U is the heat transfer coefficient of heat exchanger, W/(m^2·K); A is the surface area of heat exchanger, m^2; G is mass flow of heat agent (mass airflow), kg/h; c is its specific heat capacity, kJ/(kg·K). Then value $NTU_r = \frac{Uz}{\delta c_r \rho_r}$ can be called as the number of heat transfer points for the device. Here z is the time of complete turnover, s; δ is the depth of its plates, m; c_r is the specific heat capacity, J/(kg·K); and ρ_r - density, kg/m^3 . Physical meaning of NTU_r is a ratio of heat transfer rate on a surface to full thermal capacity of its material passing along the section of device in a second.

The same one can say about NTU in general, but with ratio to heat capacity of continuous mass flow. The surface of heat exchange does not included in calculations for NTU_r. Besides we should note that only a part of surface of rotating unit is actually interacting with flow in every moment. Thus the real time of a contact makes only z/2, i.e. during half of turnover. And plates washed by air from both sides so the calculation of its surface should be made with A/2, and instead U = α used U = α/2. The set of equations (1) was resolved numerically using the code developed by authors on base of Compaq Fortran-6.6. We used first order approximation accuracy of Euler type. For initial conditions were $\theta_c = 0$ with x' = 0 and $\theta_h = 1$ with x' = 1, and as an edge condition for a rotary unit was used coincidence with initial value of plate temperature after its full turnover and passing through both air flows. Such coincidence was reached by iteration method. The same way was made validation of calculus of temperature for exhaust air, because at a direction of an X axis along the incoming flow the calculation on an exhaust air should be done towards to a flow, starting from unknown value at the outlet of the device. The match point here was the equality to unit of extra temperature of exhaust air at inlet of device. And in case of a divergence its launching value was corrected and calculations started again.

3. Results

The results of calculations were treated the way to reach influencing on temperature efficiency of parameters NTU and NTU_r:

$$\varepsilon = \frac{NTU \cdot f(NTU_r)}{2 + NTU \cdot f(NTU_r)}. \quad (5)$$

This formula is similar for the scheme with intermediate coolant (Самарин О. Д. 2009), differing from it only by factor representing some function from NTU_r. The research of its behavior demonstrates, that at $NTU_r \to 0$ we have $f(NTU_r) \to 1$. It corresponds with other things being equal to limit $z \to 0$, i.e. indefinitely spinning of a rotary unit, at which one the plates will not have time to change temperature. Then, pursuant to (Богословский В. Н. 1983), the regenerative heat exchanger will be truly equivalent to devices with intermediate coolant, as the heat transfer will be carried out continuously.

The function f (NTU_r) is approximated by the following simple formula:

$$f(NTU_r) = \frac{1}{1 + 0.3 \cdot NTU_r^2}. \quad (3)$$

Its error at $NTU_r < 1.6$ (because the higher values are rare) does not exceed 1 %, and at NTU_r from 1.6 up to 2.0 does not exceed 3 %, that with allowance for approximate character of a received numerical solution is possible to consider completely unessential. After permutation in (2) and some transformations final expression for temperature efficiency of the rotary regenerator is received:

$$\varepsilon = \frac{NTU}{2 + NTU + 0.6 \cdot NTU_r^2} \quad (4)$$

It is possible to judge accuracy of a ratio (4) on Fig 1, where as an example the continuous line figures the chart of an approximating function for ε (4) at NTU = 2, and points show outcomes of numerical calculation. Obviously, for other values NTU the picture will be the same.

4. Example

As an example it is considered an efficiency of the heat recovery unit with the rotated heat exchanger for the AHU of a type КЦКП-10 made by "Veza" Ltd. according to data www.veza.ru. We received nominal volume-flow of air L = 10000 m^3/h, then mass flow (G) will be peer approximately 10000·1.2 = 12000 kg/h. Here 1.2 is air density, kg/m^3, at temperature + 20^{O}C (Леонтьев А. И. 1997). The heat exchange surface A for used in unit of КЦКП-10 heat recovery unit of a considered type using data (Белова Е. М. 2006) will be about 850 m^2. It is also possible to define, that at indicated in www.veza.ru the geometrical sizes of the device and arising in this case speeds of airflows the heat transfer coefficient U at its calculation under the data (Леонтьев А. И. 1997) will reach a value of 37 W/(m^2·K). Then the actual level of parameter NTU will appear equal to 3.6·37·850/(12000·1.005) = 9.38.

Time of a turnover of a rotary device z = 10 s, i.e. rotation rate equal to 6 min^{-1}, depth of plates $\delta = 0.0001$ m, density of aluminum $\rho_r = 2700$ kg/m^3 and its specific thermal capacity $c_r = 896$ J/(kg· K) (Леонтьев А. И. 1997). Thus the value NTU_r will make 37·10/(0.0001· 896·2700) = 1.53, that lies within the limits for the formulas (3) and (4).

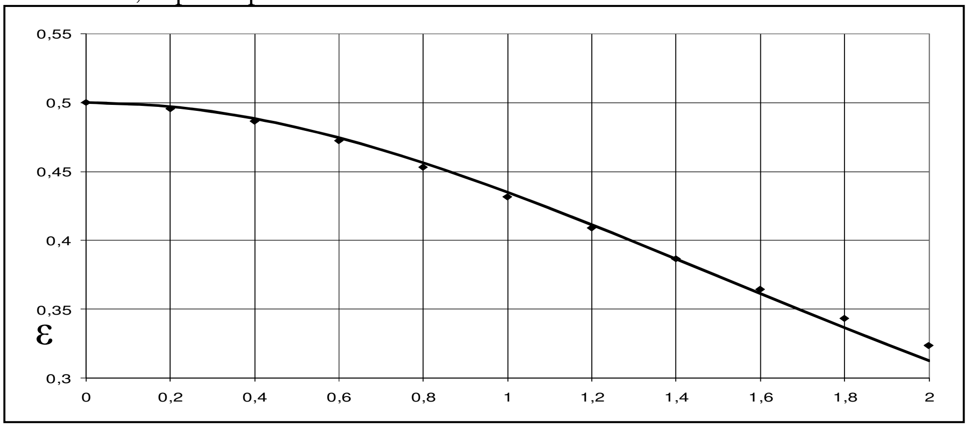

Fig 1. Rate of temperature efficiency of the rotary device from NTUr at NTU = 2: lines – approximating on (4), points – numerical calculation

Therefore, $\varepsilon = 9.38/(2 + 9.38 + 0.6 \cdot 1.53^2) = 0.734$, that gives us a heating of supply air after the heat exchanger approximately up to + 7.2^{O}C, provided that the temperature of a outdoor air for Moscow according to actual data of national standards –28^{O}C, and for exhaust air is + 20^{O}C. It is coincides a mean level of regenerator's efficiency in (Wu Z.; Melnik R. V.N.; Borup F. 2006; Белова Е. М. 2006; Богословский В. Н. 1983). The calculation as in the scheme with intermediate coolant, i.e. disregarding temperature variations fittings crosswise of flow, results under the same conditions in value $\varepsilon = 9.38/(2 + 9.38) = 0.824$, or on 12 % is higher, that is noticeable error, which one is impossible to neglect.

5. General conclusions

We have received simple ratio for temperature efficiency of regenerative heat exchangers with a rotating heat exchanger for the equal flow of supply and exhausting air. These results are suitable for design and check calculation of such devices, and also for research of varying modes of their operation during the cold time, at least, at the stage of preliminary calculus and design.

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