ANALYSIS OF IMPACT OF PLATE AIR-TO-AIR RECOVERY UNTIGHTNESS ON ITS THERMAL AND MASS BALANCE*

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The article deals with the issue of determination of leaks from a recovery heat exchanger using secondary heat contained in ventilation air and an analysis of impact of leak upon the nature of ventilation system, shared thermal flows, thermal and operational efficiency of the exchanger. The verification of the hypothesis based on results of a theoretical analysis of the issue was accomplished at an operational metering of a prototype of ventilation unit provided with a plate exchanger in calf breeding house after the installation of the unit and, furthermore, after 9 months of the operation of the unit. Results of the verification confirmed the conclusions of the theoretical analysis of the issue. They demonstrated that the new unit added 21.1 to 30.8% of extracted air to the atmospheric air supplied to the breeding house. Originally designed negative pressure system became almost a balance ventilation system due to the leak. As a result of mixing the temperature of air supplied to stable increased for 2.7–5.3 K and the air moisture content increased for 0.9–1.1 g.kg^{-1} da. Heat output exchanged during process of fresh air contamination varied between 5.15 to 11.42 kW. Due to admixture of extracted air, the thermal efficiency of the unit grew by 10.5–22.1%. After 9 months of operation in the breeding house, the extracted air volume flow in the supplied air dropped after the leaking points were sealed to 2.1–2.2%. The temperature, specific humidity and efficiency differences caused by air mixing were almost negligible.

ventilation of stable; crossflow heat exchanger; air contamination; heat balance; mass balance

INTRODUCTION

Results of research and operating testing completed both in the Czech Republic and abroad (B a s t r o n et al., 1999; K e n n e d y et al., 1991; L o r d et al., 1989; A d a m o v s k ý et al., 2002; A d a m o v s k ý, A d a m o v s k ý, 2004) showed that energy systems utilizing secondary heat of ventilation air by means of recuperative air-to-air heat exchangers significantly decrease heating energy consumption of stable facilities, improve the stable environment quality, and eliminate the risk of air humidity condensation on interior walls of structures.

However, the ventilation units feature a specific problem resting in their untightness. That untightness causes mixing of supplied air with exhausted air. The stable environment quality is most significantly affected by the exhausted air mixed into the supplied fresh air. As a result of this type of air flow mixing, concentration of harmful substances or undesirable microorganisms in the supplied may increase and its specific humidity may change as well. It is most frequently increasing. Both of these phenomena may negatively affect the stable environment quality and they require more intense ventilation, which results in heat losses caused by forced ventilation.

M a n z et al. (2001) deal with the problems of leaking ventilation units with their air-to-air recuperative heat exchangers. They analyze the impact of untightness and mixed air flows on efficiency of utilization of heat obtained from exhausted air and efficiency of local unit ventilation. They divide the air flows into external and internal ones. The external air flows apply to the whole unit and the internal ones apply to the heat exchanger itself. They monitored shared air flows from the exhausted into the supplied air and vice versa. In three ventilation units the air flow associated with untightness reached from 5 to 24% of the whole airflow volume. As a result of air mixing, the efficiency of recuperation of heat obtained from the exhausted air decreased by up to 24%. The minimum efficiency decrease was 3%. R o u l e t et al. (2001) deal with the problem of untightness of central ventilation units in their publication. Visualization gas injected into the air flow was used for untightness identification. That gas was alternatively injected into the unit's air inlets. Gas concentration was measured at seven locations of the unit and around the unit. This method was used for identification of untightness of the central ventilation unit as a whole, including the recuperative heat exchanger's body itself. Untightness of the heat exchanger reached 7%. The whole unit featured untightness of 20%. As a result of mixing, efficiency of utilization of heat recuperated from exhausted air decreased by up to 12–45% and 43% on average. For example A d a m o v s k ý et al. (2007), solved with the problem of untightness of stable recuperative heat exchangers in the Czech Republic. The article analyzes untightness of a prototype heat exchanger comprising capillary heat pipes, which was tested at a big breeding stable.

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Based on the heat exchanger's weight and heat balances, its untightness impacts on the change of the ratio of air volume flows and efficiency of heat recuperation are determined. Operating test results showed that 12.1–37.3% of exhausted inside air is mixed into the supplied outside air, efficiency of recuperation of heat from the exhausted air decreased by 18.7–55.8%, and the temperature of the supplied air increased due to mixing with the exhausted air by 0.4–2.2 °C. The impact of untightness and contamination of heat transfer surfaces on efficiency of the heat exchanger comprising gravitational heat pipes was monitored at a broiler chicken breeding stable (Adamovský et al., 1996). During the first six days of chick breeding, the heat exchanger's untightness and intense contamination of its heat transfer surfaces caused a decrease of efficiency of utilization of the heat obtained from exhausted air by 5.8% in 24 hours. The velocity of the air flowing in the heat exchanger was 0.51 m.s^{-1}.

The objective of this paper is a theoretical analysis of the issue of determination of leaks from the air/air recovery exchanger and an analysis of the impacts of leaks upon the nature of the ventilation system, shared thermal flows and thermal and operational efficiency of the exchanger. Moreover it is focused on operational verification of results of the theoretical solution and findings obtainment concerning changes of the mixing intensity under operational conditions.

METHODS

Theoretical analysis

While dealing with the issue of supplied air contaminated by exhausted air, then, based on Fig. 1, the following scenarios may occur:

1. $x_{i1} = x_{e1} \Rightarrow \overline{x}_{e2} = x_{e1}$

Mixing of two types of air will not affect the supplied air specific humidity. It will only change the air volume flows $\overline{V}_{\tau,i,ma}$ and $\overline{V}_{\tau,e,ma}$.

2. $x_{i1} > x_{e1} \Rightarrow \overline{x}_{e2} > x_{e1}$

Mixing of two types of air will increase the supplied air specific humidity and it will change the air volume flows $\overline{V}_{\tau,i,ma}$ and $\overline{V}_{\tau,e,ma}$. This scenario prevails at stable structures.

3. $x_{i1} < x_{e1} \Rightarrow \overline{x}_{e2} < x_{e1}$

Mixing of two types of air will decrease the supplied air specific humidity and it will change the air volume flows $\overline{V}_{\tau,i,ma}$ and $\overline{V}_{\tau,e,ma}$.

During solving of the issue of leaking stable units the $x_{i1} > x_{e1}$ case occurs. Following the scheme in Fig. 1, the weight balance of air flows in the heat exchanger may be expressed through the following equations:

$$m_{\tau,e,da}.x_{e1} + \Delta m_{\tau,i,da}.x_{i1} = \overline{m_{\tau,e,da}}.\overline{x_{e2}} \quad [\text{kg.s}^{-1}] \tag{1}$$

$$\overline{m}_{\tau,e,da} = m_{\tau,e,da} + \Delta m_{\tau,i,da} \quad [\text{kg}_{\text{da}}.\text{s}^{-1}] \tag{2}$$

Where:

$m_{\tau,e,da}$ – outside air mass flow before the exchanger, adjusted to 1 kg of dry air [kg$_{da}$.s^{-1}];
$\Delta m_{\tau,i,da}$ – stable air mass flow mixed into the supplied outside air, adjusted to 1 kg of dry air [kg$_{da}$.s^{-1}];
$\overline{m}_{\tau,e,da}$ – outside air mass flow behind the exchanger, adjusted to 1 kg of dry air [kg$_{da}$.s^{-1}].

In the specified equation and other equations the following applies:

Indexes: i – air exhausted from the stable,
e – outside air, supplied to the stable,
1 – values before the exchanger,
2 – values behind the exchanger,
da – dry air,
ma – moist air.

The values with the bar are values affected by mixing of the supplied air with the exhausted air.

Dry air mass flows $m_{\tau,da}$ are subject to the following equation:

$$m_{\tau,da} = \frac{m_{\tau,ma}}{(1+x)} = \frac{V_{\tau,ma} \cdot \rho_{ma}}{(1+x)} \quad [\text{kg}_{\text{da}}.\text{s}^{-1}] \tag{3}$$

Where:

$V_{\tau,ma}$ – air volume flow [m^3.s^{-1}]
ρ_{ma} – air density [kg.m^{-3}]

Upon integration of equations (3) and (2) into equation (1), we may calculate the exhausted stable air volume flow $\Delta V_{\tau,i,ma}$, which is mixed into the supplied outside air by means of the following equation:

$$\Delta V_{\tau,i,ma} = \frac{(\overline{x}_{e2} - x_{e1})}{(x_{i1} - \overline{x}_{e2})} \cdot \frac{(1 + x_{i1})}{(1 + x_{e1})} \cdot \frac{\rho_{e1.ma}}{\rho_{i1,ma}} . V_{\tau,e,ma} \quad [\text{m}^3.\text{s}^{-1}] \tag{4}$$

Equation (4) has no solution for $x_{i1} = x_{e1} = \overline{x}_{e2}$.

Actual operating volume flows in the heat exchanger $\overline{V}_{\tau,i,ma}$ and $\overline{V}_{\tau,e,ma}$ are then calculated via the following equations:

$$\overline{V}_{\tau,i,ma} = V_{\tau,i,ma} - \Delta V_{\tau,i,ma} \quad [\text{m}^3.\text{s}^{-1}] \tag{5}$$

$$\overline{V}_{\tau,e,ma} = V_{\tau,e,ma} + \Delta V_{\tau,i,ma} \quad [\text{m}^3.\text{s}^{-1}] \tag{6}$$

The heat output $Q_{\tau,R}$ shared between the exhausted and the supplied air flows through heat passing through the heat exchanger walls and not through mixing of air flows may be expressed through the heat output balance equation. The left side of this equation is formed by the actual heat output $Q_{\tau,e1}$ brought through the outside air flow, the heat output $\Delta Q_{\tau,i}$ provided to the supplied outside air during mixing with the exhausted stable air, and the heat output $Q_{\tau,R}$. The right side of the equation features the actual heat output $\overline{Q}_{\tau,e2}$ delivered to the stable. The following applies:

$$Q_{\tau,e1} + \Delta Q_{\tau,i} + Q_{\tau,R} = \overline{Q}_{\tau,e2} \quad [\text{W}] \tag{7}$$

In equation (7) is:

$$Q_{\tau,e1} = \frac{\text{V}_{\tau,\text{e,ma}} \cdot \rho_{e1,ma}}{(1 + x_{e1})} . h_{e1} \quad [\text{W}] \tag{8}$$

$$\Delta Q_{\tau,i} = \frac{\Delta \text{V}_{\tau,\text{i,ma}} \cdot \rho_{i1,ma}}{(1 + x_{i1})} . h_{i1} \quad [\text{W}] \tag{9}$$

$$\overline{Q}_{\tau,e2} = \frac{\overline{V}_{\tau,e,ma}.\overline{\rho}_{e2,ma}}{\left(1+\overline{x}_{e2}\right)}.\overline{h}_{e2} \ [\text{W}] \tag{10}$$

Where: h – air enthalpy [J.kg$_{da}^{-1}$]

We express the heat exchanger untightness value through an equation introduced by Marquardt (1983):

$$E_m = \frac{\overline{x}_{e2} - x_{e1}}{x_{i1} - x_{e1}} \ [–] \tag{11}$$

We calculated temperature t_{e2} achieved only through heating of air through shared heat output $Q_{\tau,R}$ by means of the following equation (Marquardt, 1983):

$$t_{e2} = \frac{\overline{t}_{e2} - E_m.t_{i1}}{1 - E_m} \ [^{\circ}\text{C}] \tag{12}$$

Efficiency of utilization of heat contained in the exhausted stable air was calculated through the following equations:

$$\overline{\eta}_R = \frac{\overline{Q}_{\tau,e2} - Q_{\tau,e1}}{Q_{\tau,i1} - Q_{\tau,e1}} \ [–] \tag{13}$$

$$\eta_R = \frac{Q_{\tau,e2} - Q_{\tau,e1}}{Q_{\tau,i1} - Q_{\tau,e1}} = \frac{Q_{\tau,R}}{Q_{\tau,i1} - Q_{\tau,e1}} \ [–] \tag{14}$$

Where:

$$Q_{\tau,i1} = m_{\tau,i,da}.h_{i1} \ [\text{W}] \tag{15}$$

We calculated the heat exchanger thermal efficiency eliminating the volume flow impacts by means of the following equations:

$$\overline{\eta}_{R,t} = \frac{\overline{t}_{e2} - t_{e1}}{t_{i1} - t_{e1}} \ [–] \tag{16}$$

$$\eta_{R,t} = \frac{t_{e2} - t_{e1}}{t_{i1} - t_{e1}} \ [–] \tag{17}$$

We determined the values of the moist air state and other thermodynamic values used in calculations by means of the Vitásek (2006) publication.

Measuring methodology

Ventilation units ZV 3-22 with their plate recuperative heat exchangers were tested at the calf breeding stable of the co-operative farm in Kosořín. The exchanger basic dimensions:

– Length 0.98 m, width 0.57 m, and height 0.63 m;
– Number of air ducts – 16 for exhausted air, and 16 for supplied air;
– Air duct width 15 mm;
– Heat transfer surfaces of air ducts – flat plates of galvanized sheet, thickness 0.55 mm;
– Exhausted air inlet cross-section – 0.62 m^2;
– Supplied air inlet cross-section – 0.34 m^2.

LAN W 450.4 fans with the ALR 3/21 electronic speed regulation were used for air supply and exhaust.

The stable with a light steel structure, aluminum coating, and thermal insulation contained 640–700 calves weighing 80–90 kg on steel rids with no bedding and elevated stalls.

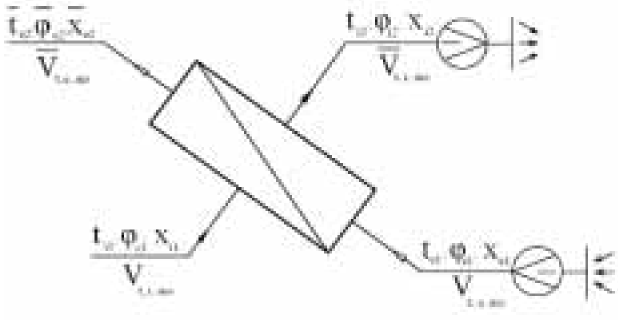

Fig. 1. Scheme of measuring of plate recuperative heat exchanger ZV 3-022

Explanation of the values depicted in Fig. 1:

t_{i1} – Stable air temperature before the exchanger [$^{\circ}$C];
φ_{i1} – Stable air relative humidity before the exchanger [–];
x_{i1} – Stable air specific humidity before the exchanger [kg.kg$_{da}^{-1}$];
t_{i2} – Stable air temperature behind the exchanger [$^{\circ}$C];
φ_{i2} – Stable air relative humidity behind the exchanger [–];
x_{i2} – Stable air specific humidity behind the exchanger [kg.kg$_{da}^{-1}$];
t_{e1} – Outside air temperature before the exchanger [$^{\circ}$C];
φ_{e1} – Outside air relative humidity before the exchanger [–];
x_{e1} – Outside air specific humidity before the exchanger [kg.kg$_{da}^{-1}$];
$\overline{t}_{e2}$ – Outside air temperature behind the exchanger [$^{\circ}$C];
$\overline{\varphi}_{e2}$ – Outside air relative humidity behind the exchanger [–];
$\overline{x}_{e2}$ – Outside air specific humidity behind the exchanger [kg.kg$_{da}^{-1}$];
$V_{\tau,i,ma}$ – Stable air volume flow before the exchanger [m^3.s^{-1}];
$V_{\tau,e,ma}$ – Outside air volume flow before the exchanger [m^3.s^{-1}].

Parameters of stable exhausted air i and supplied outside air e were measured in accordance with the scheme in Fig. 1 immediately after installation of the heat exchanger and also after 9 months of its operation.

The air temperatures were measured by means of the AMR THERM 3400 central measuring station and the AMR THERM 2246 relative humidity digital psychrometer. Air flow velocity was measured by the THIES CLIMA propeller anemometer.

The placement of breeding house air extraction duct and supply of atmospheric air eliminated mixing of both types of air before the exchanger.

What was verified simultaneously with the measuring of leaks was the impact of the clogging of the heat exchanger surfaces upon changes of volume flows of air and the efficiency of the exchanger (Kára, Adamovský, 2001).

RESULTS AND DISCUSSION

Selected results of the operational measuring are summarised in the table 1. Other six operational measuring cycles showed values with similar trends, respectively almost identical values. The values marked A were measured immediately after installation of the heat exchanger in the stable. The values marked B were measured after 9 months of the heat exchanger's operation. The value markings and indexes follow the scheme in Fig. 1 and the value specification available in the Methodologies chapter (Table 1).

Table 1. Results of measuring of plate heat exchanger ZV 3-022

	Measurements				
	A			*B*	
t_{e1} [°C]	–6.0	–5.0	–5.2	–1.7	–1.3
φ_{e1} [–]	0.70	0.64	0.65	0.78	0.75
x_{e1} [g.kg$_{da}^{-1}$]	1.6	1.6	1.6	2.6	2.6
$\bar{t}_{e2}$ [°C]	9.2	6.5	5.7	9.9	10.0
$\bar{\varphi}_{e2}$ [–]	0.37	0.41	0.46	0.35	0.35
$\bar{x}_{e2}$ [g.kg$_{da}^{-1}$]	2.7	2.5	2.7	2.7	2.7
$V_{\tau,e,ma}$ [m^3.s^{-1}]	0.42	0.60	0.68	0.24	0.29
t_{i1} [°C]	19.6	18.9	18.7	15.7	15.7
φ_{i1} [–]	0.48	0.44	0.39	0.65	0.67
x_{i1} [g.kg$_{da}^{-1}$]	7.0	6.2	5.4	7.4	7.7
t_{i2} [°C]	12.8	12.2	11.2	10.5	10.8
φ_{i2} [–]	0.74	0.68	0.63	0.91	0.93
x_{i2} [g.kg$_{da}^{-1}$]	7.0	6.2	5.4	7.4	7.7
$V_{\tau,i,ma}$ [m^3.s^{-1}]	0.58	0.80	0.90	0.41	0.49
$V_{\tau,i,ma}.V^{-1}_{\tau,e,ma}$ [–]	1.38	1.33	1.32	1.71	1.69

Table 2. Calculated values

	Measurements				
	A			*B*	
E_m [–]	0.204	0.196	0.289	0.0208	0.0196
$\Delta V_{\tau,i,ma}$ [m^3.s^{-1}]	0.119	0.160	0.302	0.0055	0.0062
$\bar{V}_{\tau,i,ma}$ [m^3.s^{-1}]	0.461	0.640	0.599	0.405	0.484
$\bar{V}_{\tau,e,ma}$ [m^3.s^{-1}]	0.539	0.760	0.982	0.246	0.296
$\bar{Q}_{\tau,e2}$ [kW]	10.54	11.94	14.93	4.94	6.03
$Q_{\tau,e1}$ [kW]	–1.07	–0.76	–1.03	1.48	1.88
$\Delta Q_{\tau,i}$ [kW]	5.15	6.42	11.40	0.22	0.26
$Q_{\tau,R}$ [kW]	6.46	6.28	4.56	3.24	3.89
t_{e2} [°C]	6.5	3.5	0.4	9.8	9.9
$\bar{\eta}_{R,t}$ [–]	0.594	0.481	0.456	0.667	0.665
$\eta_{R,t}$ [–]	0.489	0.355	0.235	0.660	0.658
$\bar{\eta}_R$ [–]	0.443	0.388	0.455	0.228	0.226
η_R [–]	0.248	0.192	0.136	0.213	0.210
$V_{\tau,i,ma}.\bar{V}^{-1}_{\tau,e,ma}$ [–]	1.076	1.053	0.916	1.667	1.655

Table 2 shows a calculations summary of equations (1) through (17) as shown in the chapter 1.

Values affected by extracted and supplied air mixing are identified by a strip in both tables.

Our process measurements and calculations showed as follows:

1. In all the process measurements $\bar{x}_{e2} > x_{e1}$. It means that the exhausted stable air mixes with the supplied outside air inside of the ventilation unit. Increased specific humidity $\bar{x}_{e2}$ must be followed by higher intensity of stable ventilation, which will result in increased heat output consumption (i.e. heat losses due to forced ventilation).
2. The longer a heat exchanger is operated, the lower the intensity of air mixing inside the unit due to its contamination (clogging). The *B* measurements showed the $\bar{x}_{e2} > x_{e1}$ difference, which is significantly smaller than in case of the *A* measurements, even if the $V_{\tau,i,ma}.V^{-1}_{\tau,e,ma}$ ratio is bigger. This trend confirms even the significant decrease of the E_m relative untightness value.
3. It applies to all the measurements that $x_{i2} = x_{i1}$. Therefore, during cooling of the exhausted stable air there was no partial condensation of water vapor from the air.
4. The share of $\Delta V_{\tau,i,ma}$ in $\bar{V}_{\tau,e,ma}$ reaches, in case of the installed heat exchanger (*A* measurements) 21.1–30.8%, and 2.1–2.2% after 9 months of the heat exchanger's operation (*B* measurements). The intensity of air mixing decreases due to untightness affected by clogging again.
5. The originally designed negative pressure ventilation $V_{\tau,i,ma}.V^{-1}_{\tau,e,ma} > 1$ became an almost balanced ventilation system $V_{\tau,i,ma}.\bar{V}^{-1}_{\tau,e,ma} \approx 1$ due to leaks.
6. Temperature of the air supplied to the stable increased during the *A* measurements due to its mixing with the exhausted stable air by 2.7–5.3 K ($\bar{t}_{e2} - t_{e2}$). A quite negligible difference was identified during the *B* measurements.
7. During the *A* measurements, the temperature efficiency difference $\bar{\eta}_{R,t} - \eta_{R,t}$ was 10.5–22.1%. It was quite negligible during the *B* measurements.
8. During the *A* measurements the heat exchanger efficiency difference $\bar{\eta}_R - \eta_R$ was 19.5–31.9%. The bigger efficiency difference, compared to the temperature efficiency, derives from the increased air enthalpy $\bar{h}_{e2}$ due to the shared humidity $\Delta x = \bar{x}_{e2} - x_{e1}$. The *B* measurements showed a quite negligible difference.
9. Decrease of ratio $\bar{V}_{\tau,i,ma}.\bar{V}^{-1}_{\tau,e,ma}$ respond to increase of heat output $\Delta Q_{\tau,i}$ exchanged to supply fresh air during mixing.
10. Increase of heat output $\Delta Q_{\tau,i}$ causes drop of heat output $\Delta Q_{\tau,R}$ transferred through heat exchanging surface.

Measured and calculated values of air flows transferred through leakages as well as decreased thermal efficiencies of air handling unit confirms values published by Manz et al. (2001) and Roulet et al. (2001).

Air handling unit tested within very severe and dusty operational conditions introduced new findings. During nine operational months when fouling of heat exchanging surfaces proceeded caused almost full blockage of leakages. Foulness of heat exchanging surfaces however causes decrease of air flows. The operational testing results (Kára, Adamovský, 2000) showed that if the same fan motor regulation voltage was used, after 9 months of the heat exchanger operation in the calf breeding stable, heat transfer surface contamination decreased the exhausted stable air volume to 75% of its original value and to 59.7% in case of the supplied outside air. Likewise the specification of unit leakages and theirs influence to ventilation character and heat balance of heat exchanger brings new and practically utilizable findings.

Following work will be verification of obtained facts for other types of recuperative heat exchangers e.g. heat pipes.

The aims presented in the article introduction are in accordance with our opinion accomplished.

CONCLUSION

Objectives presented in the introduction of the paper were met. The results of the operational verification confirmed the conclusions of the theoretical analysis of the problem. The heat exchanger's untightness increased specific humidity, temperature, and their associated values of air heat flow supplied to the stable. These facts had a significant impact on the balance of mass and heat flows and the heat exchanger's operating efficiency. We also identified a significant change of the ratio between the volume flows of the exhausted air and the supplied air. It completely changed the character of the originally designed ventilation system. We verified that contamination of heat transfer surfaces and the associated clogging of untight areas lead to a decrease and later on to a complete elimination of mixing. However, it strongly limits air volume flows.

The requirements of the tightness of heat exchanger surfaces and their protection against pollution for practical applications of energy saving ventilation air secondary heat recovery systems installed in cattle breeding houses and other operational premises imply from solution of the problem.

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Analýza vlivu netěsnosti deskového rekuperačního výměníku vzduch-vzduch na jeho tepelnou a hmotnostní bilanci.

Scientia Agric. Bohem., *39*, 2008: 289–294.

Cílem tohoto článku je teoretický rozbor problematiky stanovení netěsnosti rekuperačního výměníku typu vzduch–vzduch a analýza vlivu netěsnosti na charakter systému větrání, sdílené tepelné toky, teplotní a provozní účinnost výměníku. Dále pak provozní ověření výsledků teoretického řešení a získaní poznatků o změnách intenzity směšování v provozních podmínkách. Měření se uskutečnilo ve stáji pro chov telat ihned po instalaci jednotky a dále pak po devíti měsících provozu jednotky. Ve stáji z lehké ocelové konstrukce s hliníkovým pláštěm a tepelnou izolací bylo ustájeno 640 až 700 telat v bezstelivové technologii na ocelových roštech s vyvýšeným stáním. Měření parametrů odváděného stájového (index *i*) a přiváděného venkovního (index *e*) vzduchu se uskutečnilo podle schématu na obr. 1.

Teoretický rozbor problému kontaminace přiváděného vzduchu vzduchem odváděným vychází z hmotnostní a tepelné bilance výměníku. Pro vyjádření hodnoty poměrné netěsnosti výměníku jsme použili vztah (11). Objemový tok odváděného stájového vzduchu $\Delta V_{\tau,i,ma}$, který je přimícháván do přiváděného venkovního vzduchu, lze vypočítat ze vztahu (4). Tepelný výkon $\Delta Q_{\tau,R}$, sdělený mezi odváděným a přiváděným tokem vzduchu prostupem tepla stěnou výměníku, nikoliv smíšením toků vzduchu můžeme vyjádřit z rovnice (7). Levou stranu rovnice tvoří skutečný tepelný výkon $Q_{\tau,e1}$ přivedený tokem venkovního vzduchu, tepelný výkon $\Delta Q_{\tau,i}$ sdělený přiváděnému venkovnímu vzduchu smícháním s odváděným stájovým vzduchem a tepelný výkon $Q_{\tau,R}$. Pravá strana rovnice je dána skutečným tepelným výkonem $\overline{Q}_{\tau,e2}$ přiváděným do stáje. Účinnosti využití tepla obsaženého v odváděném stájovém vzduchu jsme vypočetli z rovnic (13) až (17).

Výsledky provozního ověřování potvrdily závěry teoretického rozboru problému. Ukázaly, že v nové jednotce se do přiváděného venkovního vzduchu přimíchávalo 21,1–30,8 % vzduchu odváděného ze stáje. Z původně projektovaného podtlakového systému větrání se vlivem netěsnosti stal téměř systém rovnotlaký. V důsledku směšování se zvýšila teplota vzduchu přiváděného do stáje $\bar{t}_{e2}$ o 2,7–5,3 K a měrná vlhkost vzduchu $\bar{x}_{e2}$ o 0,9–1,1 [$g.kg_{da}^{-1}$]. Tepelný výkon sdělený při kontaminaci přiváděného vzduchu vzduchem odváděným $\Delta Q_{\tau,i}$ se pohyboval v rozmezí 5,15–11,42 kW. Při nízkém poměru objemových toků odváděného a přiváděného vzduchu $\overline{V}_{\tau,i,ma} . \overline{V}_{\tau,e,ma}^{-1}$ byl tepelný výkon $\Delta Q_{\tau,i}$ vyšší než tepelný výkon $Q_{\tau,R}$ sdělený prostupem tepla stěnou výměníku. Teplotní účinnost jednotky $\overline{\eta}_{R,t}$ se zvýšila o 10,5–22,1 % a provozní účinnost $\overline{\eta}_R$ o 19,5–31,9 %. Výsledky teoretického rozboru i měření větrací jednotky s deskovým výměníkem se shodují s výsledky měření netěsnosti větrací jednotky uváděnými v publikaci autorů M a n z et al. (2001) i s výsledky měření centrální větrací jednotky (R o u l e t et al., 2001)

Vzduch odváděný ze stáje obsahuje velké množství drobných pevných částí prachu, kůže zvířat, krmiva atd. Vlivem těchto částí a možné parciální kondenzace vody obsažené ve vzduchu se po devíti měsících provozu ve stáji objemový tok odváděného vzduchu ve vzduchu přiváděném snížil vlivem ucpání netěsností na 2,1–2,2 %. Rozdíly teplot, měrných vlhkostí i účinností byly pak téměř zanedbatelné. Je otázkou, zda lze snížení netěsnosti a souvisejících eliminací negativních jevů, zejména zvyšování koncentrace škodlivin, resp. nevhodných mikroorganismů očekávat v méně prašných provozech.

větrání stájí; deskový výměník; teplotní účinnost; kontaminace vzduchu; tepelná bilance; hmotnostní bilance

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