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ANALYSIS OF LEAKAGE INFLUENCE WITHIN CAPILLARY HEAT PIPE RECUPERATIVE HEAT EXCHANGER

Summary

The paper analyses the leakage of air-to-air recovery heat exchanger consisting of 100 capillary thermal tubes 1.5 m long and aligned in ten rows. The operation of the recovery exchanger prototype was tested in a pigsty. The analysis focuses on how the exchanger leakage affects the heat utilisation efficiency of the air ducted from the pigsty, changes in supply/exhaust air flow rate ratio, supply air temperature, thermal output obtained, and exchanger's total thermal and mass balances.

The results obtained in in-process measurements and the mass and thermal balances show that the exchange leakage causes the outside supply air to contain 12.1–37.3% of the pigsty exhaust air. The percentage of the pigsty air increases with the increase of mass flow rates of supply and exhaust air. This leakage resulted particularly in a major change in supply and exhaust air mass flow rates, higher moisture content, temperature and thermal flow of the air conducted in the pigsty. The change in air flow rates resulted in ventilation change so that the overpressure ventilation took place instead of balanced ventilation. This change led to increased thermal losses by air penetrating through leaks in the peripheral building structures. Increased supply air moisture content made it necessary to increase ventilation intensity in order to meet the required pigsty air parameters with respect of animal wellbeing and protection of peripheral building structures against dampness deterioration. The thermal flow increase of the air conducted in the pigsty due to the high enthalpy of the pigsty air mixed resulted in the higher secondary heat utilisation efficiency. The actual efficiency, however, was considerably lower.

The differences of 2.7 – 14.5% between the exchanger's measured and actual thermal efficiency according to relation (16) and those of 18.7 – 55.8% between the exchanger's measured and actual running efficiency (13) increase as the ratio of supply and exhaust air mass flow rates increase.

Key words: exchanger; recuperation; thermal performance; blending efficiency; thermal efficiency; thermal balance; mass balance

INTRODUCTION

A negative result of leaking separation walls or heat-exchange surfaces of air-to-air type recuperation exchangers is blending of the exhaust air with the supply air. This problem becomes significant in stables. Leaking exchanger separation walls can lead to increased concentration of noxious gases, humidity and also undesirable micro-organisms inside the stable.

Untightness of air-to-air recuperation heat exchangers is objectively expressed in terms of the air blending efficiency, introduced by Marquardt (1983):

$$E_m = \frac{x_{e2} - x_{e1}}{x_{i1} - x_{e1}} \; [-] \qquad (1)$$

where in relation (1):

x_{e2} – is moisture content of outdoor air behind the exchanger [$kg.\ kg_{da}^{-1}$];

x_{e1} – is moisture content of outdoor air before the exchanger [$kg.\ kg_{da}^{-1}$];

x_{i1} – is moisture content of stable air before the exchanger [$kg.\ kg_{da}^{-1}$].

The issues of leaking plate exchangers in local ventilation units were studied by Manz, Huber, Helfefinger [2001]. They analysed the effects of leakages and blending of air streams on the efficiency of utilising heat from the exhaust air. Between 5 and 24% of exhaust air was added to the supply air in the test units. This resulted in reduced efficiency of heat utilisation from the exhaust air by as much as 24%, with the minimum drop 3%.

The issues of identifying leaks in central ventilation units were studied by Roulet *et al.* [2001]. To find leaks, they used visualisation gases injected into the air stream. A 7% leakage was found in the exchanger. The whole unit showed a 20% leakage. Due to the leakages, the efficiency dropped on average by 45%.

Adamovský, Hutla, Blažek [1996] studied the effects of leakages and contamination of heat-exchange surfaces on the efficiency of a gravitational heat pipe heat exchanger in a stable used for breeding Broiler chickens. During the first six days of the breeding process, exchanger leakages combined with an intensive fouling of heat-exchange surfaces, resulted in a drop in thermal efficiency by 5.8% a day.

Kára Adamovský [2000] studied the changes in air blending efficiency in a calf breeding stable as a function of operation time. In a new exchanger, blending efficiency E_m = 0.15 – 0.22 was obtained, measured according to relation (1). After 11 months of the exchanger operation in the mode without condensation of air moisture from the exhaust air, the blending efficiency dropped to E_m = 0.01 – 0.02. When running the exchanger with partial condensation of the air moisture, the efficiency dropped after 9 months of operation to only E_m = 0.005 – 0.01.

The objective of our study was to determine leakages in a prototype capillary heat pipe heat exchanger, and analyse the effects of these leakages on the utilisation of heat from the exhaust air, changes in the ratio of the supply and exhaust air, the achieved thermal output and overall thermal and mass balance of the exchanger.

METHODS

1. Measurement method

The prototype recuperation capillary heat pipe exchanger was tested in a stable used for breeding pigs. The exchanger was assembled from 100 aluminium heat tubes arranged in 10 rows on a plane. The effective tube length was 1455 mm. The tubes were filled with ammoniac. The transversal as well as longitudinal spacing of the tubes in rows was 64 mm. The cross-sectional area of the supply as well as exhaust air flow was 0.3194 m^2. The fans were powered by two-speed electric motors.

Schematic diagram of the exchanger measurements is shown in fig.1.

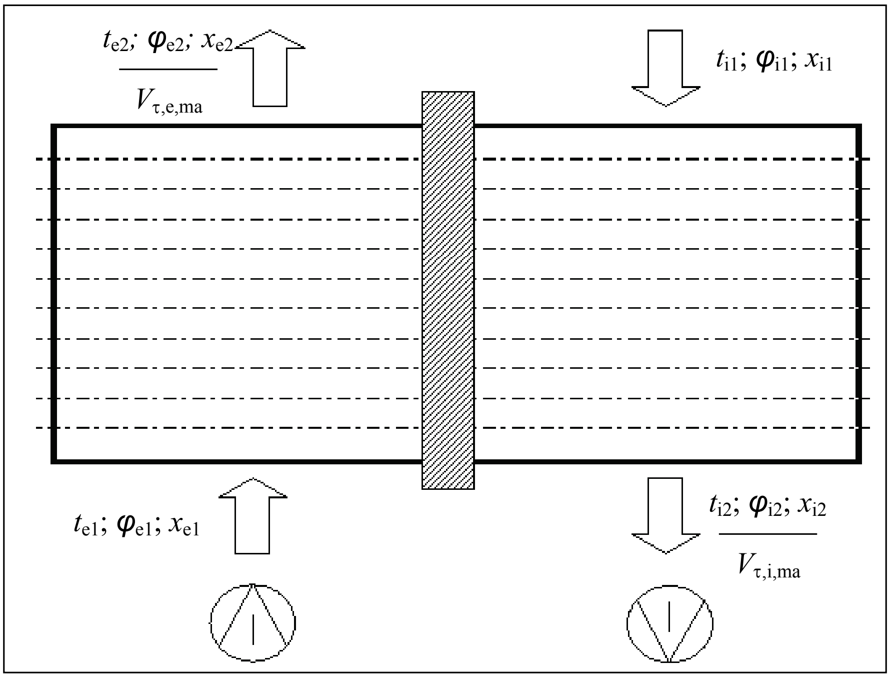

Figure 1. Schematic diagram of capillary heat pipe heat exchanger measurements AGRA 630/2 axial fan; 2. AGRA 465/2 axial fan

Legend to Fig. 1:
t_{i1} – temperature of stable air before the exchanger [$°C$];
φ_{i1} – relative humidity of stable air before the exchanger [-];
x_{i1} – moisture content of stable air before the exchanger [$kg.\ kg_{da}^{-1}$];
t_{i2} – temperature of stable air behind the exchanger [$°C$];
φ_{i2} – relative humidity of stable air behind the exchanger [-];
x_{i2} – moisture content of stable air behind the exchanger [$kg.\ kg_{da}^{-1}$];
t_{e1} – temperature of outdoor air before the exchanger [$°C$];
φ_{e1} – relative humidity of outdoor air before the exchanger [-];
x_{e1} – moisture content of outdoor air before the exchanger [$kg.\ kg_{da}^{-1}$];
t_{e2} – temperature of outdoor air behind the exchanger [$°C$];
φ_{e2} – relative humidity of outdoor air behind the exchanger [-];
x_{e2} – moisture content of outdoor air behind the exchanger [$kg.\ kg_{da}^{-1}$];
$V_{\tau,i,ma}$ – volumetric flow of stable air behind the exchanger [$m^3.s^{-1}$];
$V_{\tau,e,ma}$ – volumetric flow of outdoor air behind the exchanger [$m^3.s^{-1}$].

2. Theoretical analysis

Blending of exhaust stable air and outdoor air occurs when $x_{e2} > x_{e1}$. The volumetric flow of exhaust stable air $\Delta V_{\tau,i,ma}$, which is added to the outdoor supply air can be derived from the exchanger mass equilibrium equation:

$$\Delta m_{\tau,i,da} \cdot x_{i1,2} + \overline{m}_{\tau,e,da} \cdot x_{e1} = m_{\tau,e,da} \cdot x_{e2} \quad [kg.s^{-1}] \tag{2}$$

$$\overline{m}_{\tau,e,da} = m_{\tau,e,da} - \Delta m_{\tau,i,da} \quad [kg_{da}.s^{-1}] \tag{3}$$

where:

$\Delta m_{\tau,i,da}$ – is mass flow of stable air added to supply air, converted to 1 kg of dry air [$kg_{da}.s^{-1}$];

$x_{i1,2} = \dfrac{x_{i1} + x_{i2}}{2}$ – is median moisture content [$kg.\ kg_{da}^{-1}$];

$\overline{m}_{\tau,e,da}$ – is the actual mass flow (without added air) of outdoor air, converted to 1 kg of dry air [$kg_{da}.s^{-1}$];

$m_{\tau,e,da}$ –is measured mass flow of outdoor air, converted to 1 kg of dry air [$kg_{da}.s^{-1}$].

Index:

i – is air removed from the stable;
e – outdoor air supplied to the stable;
da – dry air;
ma – moist air.

For mass flows of dry air $m_{\tau,da}$, the following equation applies:

$$m_{\tau,da} = \frac{m_{\tau,ma}}{(1+x)} = \frac{V_{\tau,ma} \cdot \rho_{ma}}{(1+x)} \; [kg_{da}.s^{-1}] \tag{4}$$

where:

$V_{\tau,ma}$ – is volumetric air flow [$m^3.s^{-1}$];

ρ_{ma} – is density of air [$kg.m^{-3}$];

x – is moisture content of air [$kg.\, kg_{da}^{-1}$.]

By substituting relation (4) into equations (2) and (3), we get for $\Delta V_{\tau,i,ma}$ the following equation:

$$\Delta V_{\tau,i,ma} = \frac{(x_{e2} - x_{e1})}{(x_{i1,2} - x_{e1})} \cdot \frac{(1 + x_{i1,2})}{(1 + x_{e2})} \cdot \frac{\rho_{e2}}{\rho_{i1,2}} V_{\tau,e,ma} \; [m^3.s^{-1}] \tag{5}$$

where:

$\rho_{i1,2}$ – is the median of air densities at temperature $t_{i1,2} = \frac{t_{i1} + t_{i2}}{2}$ [$kg.m^{-3}$].

The actual volumetric air flows $\overline{V}_{\tau,i,ma}$ and $\overline{V}_{\tau,e,ma}$ we calculate from equations:

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

$$\overline{V}_{\tau,e,ma} = V_{\tau,e,ma} - \Delta V_{\tau,i,ma} \; [m^3.s^{-1}] \tag{7}$$

Thermal output $Q_{\tau,R}$, which was obtained by heat transmission through the exchanger wall between stable air and outdoor air is determined by equation:

$$Q_{\tau,R} = m_{\tau,e,sv}(h_{e2} - h_{e1}) \;\; [W] \tag{8}$$

where:

h_{e2} – is specific enthalpy of outdoor air after being heated in the exchanger [$J.kg_{da}^{-1}$];

h_{e1} – is specific enthalpy of outdoor air before being heated in the exchanger [$J.kg_{da}^{-1}$].

Recuperated thermal output $Q_{\tau,R}$ can be calculated from the equilibrium equations of thermal outputs inside the exchanger. On the left side of the equation there is the actual output $\overline{Q}_{\tau,e1}$ supplied by the flow of outdoor air, thermal output $\Delta Q_{\tau,i}$ communicated to the supplied outdoor air by blending it with the stable air, and thermal output $Q_{\tau,R}$. On the right side of the equation there is output $Q_{\tau,e2}$ supplied to the stable.

The following relation applies:

$$\overline{Q}_{\tau,e1} + \Delta Q_{\tau,i} + Q_{\tau,R} = Q_{\tau,e2} \quad [W] \tag{9}$$

In equation (9):

$$\overline{Q}_{\tau,e1} = \frac{\overline{V}_{\tau,e,ma} \cdot \rho_{e1}}{(1 + x_{e1})} \cdot h_{e1} \quad [W] \tag{10}$$

$$\Delta Q_{\tau,i} = \frac{\Delta V_{\tau,i,ma} \cdot \rho_{i1,2}}{(1 + x_{i1,2})} (h_{i1,2} - h_{e1,2}) \ [W] \tag{11}$$

where:

$h_{i1,2}$ – is air enthalpy corresponding to air temperature $\frac{t_{i1} + t_{i2}}{2}$ $[J.kg_{da}^{-1}]$;

$h_{e1,2}$ – is air enthalpy corresponding to air temperature $\frac{t_{e1} + t_{e2}}{2}$ $[J.kg_{da}^{-1}]$.

$$Q_{\tau,e2} = \frac{V_{\tau,e,ma} \cdot \rho_{e2}}{(1 + x_{e2})} h_{e2} \quad [W] \tag{12}$$

The efficiency of utilising heat present in the stable exhaust air, determined by the ratio between the gained thermal output $Q_{\tau,R}$ and the overall performance which can be obtained from the cooled air $Q_{\tau,i1} - Q_{\tau,e1}$, we have defined from the equation [Adamovský *et al.* 2004]:

$$\eta_R = \frac{Q_{\tau,R}}{Q_{\tau,i1} - Q_{\tau,e1}} \quad [-] \tag{13}$$

For thermal flows $Q_{\tau,i}$ and $Q_{\tau,e1}$ the following applies:

$$Q_{\tau,i1} = m_{\tau,i,da} \cdot h_{i1} \quad [W] \tag{14}$$

$$Q_{\tau,e1} = m_{\tau,e,da} \cdot h_{e1} \quad [W] \tag{15}$$

where:

h_{i1} – is specific enthalpy of stable air before the exchanger $[J.kg_{da}^{-1}]$.

Thermal efficiency $\eta_{R,t}$ which eliminates the effect of mass flows, was calculated using the following relation:

$$\eta_{R,t} = \frac{t_{e2} - t_{e1}}{t_{i1} - t_{e1}} \quad [-] \tag{16}$$

To calculate the actual temperature $\overline{t_{e2}}$ achieved by mere heating of the air by mentioned above thermal output $Q_{\tau,R}$, we used the relation [Marquart 1983]:

$$\overline{t_{e2}} = \frac{t_{e2} - E_m \cdot t_{i1}}{1 - E_m} \quad [°C] \tag{17}$$

RESULTS AND DISCUSSION

Table 1 summarises the results of measurements and solving equations (2) through to (17). The values with upper line are based on actual volumetric flows $\overline{V}_{\tau,i,ma}$ and $\overline{V}_{\tau,e,ma}$, which eliminate thermal flows and enthalpies generated by adding exhaust air into the supply outdoor air.

The following arises from the measured values and calculations:

– share of $\Delta V_{\tau,i,ma}$ in $V_{\tau,e,ma}$ for the given exchanger is 12.1 – 37.3%;

– with growing $\overline{V}_{\tau,i,ma} / \overline{V}_{\tau,e,ma}$, $\Delta V_{\tau,i,ma}/V_{\tau,e,ma}$ increases

– increase in $\Delta V_{\tau,i,ma}/V_{\tau,e,ma}$ corresponds to increasing difference $\Delta x_e = x_{e2} - x_{e1}$. With $\Delta V_{\tau,i,ma} \geq 0{,}3\ V_{\tau,e,ma}$, $\Delta x_e > 1\ g \cdot kg_{da}^{-1}$;

– thermal efficiency η_{Rt} decreases with increasing $\Delta V_{\tau,i,ma}/V_{\tau,e,ma}$;

– differences between the actual $\overline{\eta_{Rt}}$ and measured η_{Rt} thermal efficiency (16) are in the range 2.7 – 14.6%, and increase with increasing $\Delta V_{\tau,i,ma}/V_{\tau,e,ma}$;

– ratio $\dfrac{Q_{\tau,R}}{Q_{\tau,e2} - \overline{Q}_{\tau,e1}}$ decreases with increasing $\Delta V_{\tau,i,ma}/V_{\tau,e,ma}$;

– ratio $\dfrac{\Delta Q_{\tau,i}}{Q_{\tau,e2} - \overline{Q}_{\tau,e1}}$ increases with increasing $\Delta V_{\tau,i,ma}/V_{\tau,e,ma}$;

– actual efficiency $\overline{\eta_R}$ is significantly lower than η_R.

Table 1. Results of process measurement and calculation

	Measurement number							
	1	2	3	4	5	6	7	8
	Results of measurement							
t_{i1} [$°C$]	14.0	13.7	15.0	15.3	14.3	17.0	19.0	14.0
φ_{i1} [-]	0.74	0.77	0.78	0.73	0.75	0.77	0.75	0.73
x_{i1} [$g.kg_{da}^{-1}$]	7.36	7.50	8.24	7.95	7.60	9.30	10.30	7.35
t_{i2} [$°C$]	3.7	1.7	4.2	4.2	4.4	4.2	5.4	2.7
φ_{i2} [-]	0.98	0.99	0.98	0.98	0.94	0.98	0.98	0.97
x_{i2} [$g.kg_{da}^{-1}$]	4.73	4.20	4.90	4.90	4.70	5.00	5.40	4.40
$V_{\tau,i,ma}$ [$m^3.s^{-1}$]	0.36	0.26	0.26	0.26	0.14	0.26	0.15	0.42
t_{e1} [$°C$]	0.0	-1.4	1.9	1.9	3.4	2.0	2.2	-2.9
φ_{e1} [-]	0.83	0.84	0.81	0.81	0.80	0.81	0.84	0.81
x_{e1} [$g.kg_{da}^{-1}$]	3.10	2.83	3.49	3.49	3.83	3.52	3.73	2.55
t_{e2} [$°C$]	10.0	6.4	9.3	9.3	11.1	11.4	15.0	8.0
φ_{e2} [-]	0.46	0.64	0.63	0.63	0.53	0.55	0.41	0.45
x_{e2} [$g.kg_{da}^{-1}$]	3.47	3.91	4.58	4.58	4.47	4.61	4.30	2.95
$V_{\tau,e,ma}$ [$m^3.s^{-1}$]	0.672	0.672	0.674	0.674	0.300	0.674	0.310	0.800
$V_{\tau,i,ma}/V_{\tau,e,ma}$ [-]	0.536	0.387	0.386	0.386	0.467	0.386	0.484	0.525
	Results of calculation							
$\Delta V_{\tau,i,ma}$ [$m^3.s^{-1}$]	0.0844	0.2422	0.2395	0.2514	0.0825	0.2026	0.0427	0.0968
$\overline{V}_{\tau,i,ma}$ [$m^3.s^{-1}$]	0.4444	0.5022	0.4995	0.5114	0.2225	0.4626	0.1927	0.5168
$\overline{V}_{\tau,e,ma}$ [$m^3.s^{-1}$]	0.5876	0.4298	0.4345	0.4226	0.2175	0.4714	0.2673	0.7032
$Q_{\tau,e2}$ [kW]	15.601	13.646	17.387	17.387	8.265	19.088	9.758	15.400
$\overline{Q}_{\tau,e1}$ [kW]	5.856	3.147	5.904	5.743	3.593	6.510	3.935	3.154
$\Delta Q_{\tau,i}$ [kW]	1.132	3.461	3.079	3.166	0.727	2.917	0.695	1.649
$Q_{\tau,R}$ [kW]	8.613	7.039	8.402	8.478	3.946	9.662	5.128	10.597
$\overline{t_{e2}}$ [$°C$]	9.62	4.204	7.602	7.359	10.446	10.099	14.62	7.455
$\eta_{R,t}$ [-]	0.714	0.517	0.565	0.552	0.706	0.627	0.762	0.645
$\overline{\eta_{R,t}}$ [-]	0.687	0.371	0.435	0.407	0.646	0.540	0.739	0.613
η_R [-]	0.836	0.629	0.735	0.727	0.865	0.793	0.917	0.712
$\overline{\eta_R}$ [-]	0.587	0.176	0.187	0.169	0.333	0.284	0.641	0.525
$\overline{V}_{\tau,i,ma}/\overline{V}_{\tau,e,ma}$ [-]	0.756	1.168	1.150	1.210	1.023	0.981	0.721	0.735

CONCLUSIONS

The results of testing the prototype recuperation capillary heat pipe exchanger proved leaks in the exchanger separation walls. Those leaks in the separation wall were manifested by increased value of the thermal flow and moisture content of the air supplied to the stable and also by a significant change in the $V_{\tau,i,ma}/V_{\tau,e,ma}$ ratio.

The increased value of thermal flow of the air supplied to the stable, caused by high enthalpy of the added stable air had an impact on the secondary heat utilisation efficiency η_R. This acquired higher values. The actual efficiency $\overline{\eta_R}$ was lower.

Increasing moisture content of the supplied outdoor air induced increased ventilation intensity. The reason for this was to maintain the required parameters of the stable air to provide comfort for the animals and to protect peripheral building structures against damage by moisture.

Change in the $V_{\tau,i,ma}/V_{\tau,e,ma}$ ratio lead to change in the character of ventilation. Instead of the designed equal-pressure ventilation when $\overline{V}_{\tau,i,ma} / \overline{V}_{\tau,e,ma} \approx 1$, overpressure ventilation took place, $V_{\tau,i,ma}/V_{\tau,e,ma} < 1$. This change resulted in increased thermal losses due to uncontrolled infiltration of air through the building's peripheral walls.

No negative consequences of blending the exhaust air with the supply air, which may have lead to increased concentrations of unsuitable or harmful microorganisms in the stable air, have been proven.

We came across the problem of leaking heat-exchange surfaces also when testing plate recuperation exchangers, as well as gravitation heat pipe heat exchangers.

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