ANALYSIS OF CHANGES OF THERMIC EFFICIENCY DURING OPERATION OF THE TABLE RECUPERATION EXCHANGERS

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The contribution analyses the effect of contamination of table recuperation exchangers after 9 and 11 months of work in calf-barns on milk and plant nutrition. The effects of contamination of heat-exchanging areas of the volume flows, efficiency of realisation of utilisation of secondary heat, thermal power, mixing of air during operation of exchangers with water condensation from air and dry operation were studied. Based on the results of testing, recommended intervals of cleaning of heat-exchanging areas of desk exchangers were determined.

stable; calves; ventilation; secondary heat; table exchanging; heat-exchanging areas; efficiency of exchanger; ratio of air mixing

INTRODUCTION

During operation of exchangers for heat recuperation from ventilating air of stables, the effect of action of technological lines, biological production of minerals as well as high content of dust in ambient medium results in contamination of heat-exchanging surfaces of exchangers. Contamination of heat-exchanging areas causes the decrease of coefficient of heat transfer through the dividing walls. Both they have a negative impact on the effects of recuperation from ventilating air. The aim of the studies was to determine recommended intervals of cleaning of table recuperation exchangers on the basis of analysis of changes in thermic efficiency during work in large-capacity barns.

MATERIAL AND METHODS

Measurement was done in two stables of large-capacity calf-barns of the Co-operative Farm Zálší at Kosořín. In the stable for 615 to 620 calves on milk diet and in the stable for 640 to 700 calves on plant diet table exchangers

ZV-3-022, manufactured by the Co-operative Farm Horní Brusnice, were tested.

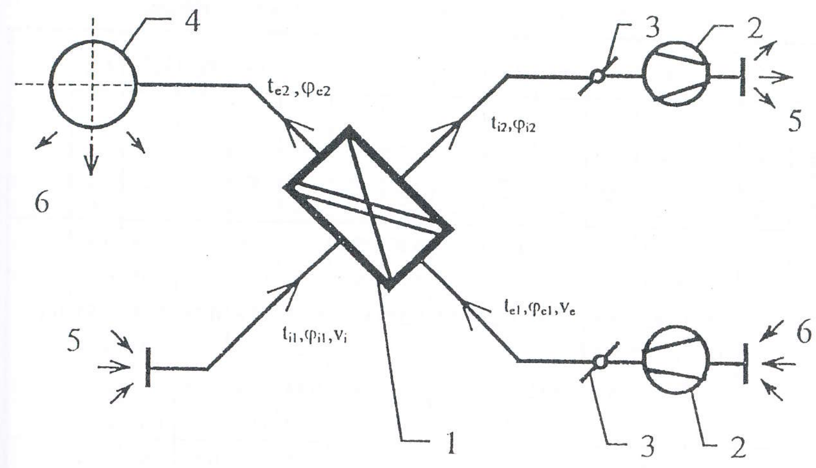
The basic characteristics of the exchanger and types of fans:

- size: 0.98 m in length, 0.57 m in width, 0.65 m in height
- weight: 280 kg
- number of cells: 16
- pitch of air channel among cells: 0.015 m
- heat-exchanging surfaces: flat tables of zinc-coated sheet of 0.55 mm thickness
- volume air flow: 0.4 to 0.7 $m^3.s^{-1}$
- inlet cross section for waste discharged air 0.62 m^2
- inlet cross section for external supplied air: 034 m^2
- fans: 2 fans LAN W 450.4 with electronic control of revolutions ALR 3/21 within 10 to 100 % rated revolutions.

Measurements of parameters of discharged and supplied air according to the scheme in Fig. 1 were done immediately after instalment of exchangers, further in the calf-barn on plant nutrition after nine months of work and in the calf-barn on milk nutrition after 11 months of work. Exchangers were installed in April when practically no heating is needed. Therefore, further measurements were carried out in December and in February of the next year. Exchangers operated the whole time, but the most contamination appears in winter during operation with water condensation from ventilated waste stable air. With respect to the purpose of the trial, exchangers were not cleaned the whole time in order to find the dependencies of decrease of flows and thermic efficiency on the time of operation.

In the barns of light steel construction with aluminium cover and thermal insulation, the calves were housed in letter-free technology on steel grates with elevated standing.

Measurements of parameters of both types of air were carried out at different revolutions of fans in order to study the changes in operating thermic efficiency at different air flows. With respect to the cross-flow exchangers the velocities and temperatures are not fully uniform in inlet and particularly in outlet cross sections, cross sections during measurement were divided into the system n of areas S_n and parameters of air were recorded in different fields. Mean values were determined from measured values of air temperature t, relative air moisture φ and air flow rate v and volume air flows, thermic efficiency of heat recuperation, and ratio of air flows mixing was summed up. Air temperatures were measured by portable measuring switchboard AMR THERM 3400; relative air moisture by digital psychrometer THIES CLIMA was used to measure the rate of air flowing.



1. Diagram of measurement of recuperation exchanger
1 – Table recuperation exchanger ZV 3-022
2 – Axial ventilator LAN W 450.4
3 – Flaps of side channels of exchanger
4 – Perforated foil sleeve
5 – Discharge of internal cooled stable air
6 – Supply of external heated air

RESULTS AND DISCUSSION

Results of measurements are processed in Table I. The group of values, marked A, was measured immediately after instalment of exchangers. The values, marked B, were measured in the calf-barn on milk diet after 11 months of operation of recuperation exchanger. The group of values, marked C, was measured in the calf-barn on plant nutrition after 9 months of operation of recuperation exchanger.

Thermic efficiency of heat recuperation from ventilating air in recuperation exchangers η_{Rt} is defined generally by ratio of recuperation heat power Q_R to the total power Q_i–Q_e, that can be obtained in using the whole temperature gradient between internal and external air:

$$\eta_{Rt} = \frac{Q_R}{Q_i - Q_e} \quad [-] \tag{1}$$

From the measured and calculated values of volume flows, temperatures and relative air moistures according to the following relationship, derived in the study written by A d a m o v s k ý et al. (1996), it can be calculated:

I. Results of testing of table recuperation exchanger ZV 3-022 in the calf-barns

Measurement: Designation	Measurement: number	U (V)	V_e ($m^3.s^{-1}$)	t_{e1} (°C)	φ_{e1} (–)	t_{e2} (°C)	φ_{e2} (–)	V_i ($m^3.s^{-1}$)	t_{i1} (°C)	φ_{i1} (–)	t_{i2} (°C)	φ_{i2} (–)	V_i/V_e (–)	η_{Rt} (–)	E_m (–)
A	1	160	0.29	–6.2	–	10.1	–	0.40	19.6	–	12.6	–	1.38	0.52	–
	2	170	0.34	–5.7	–	8.7	–	0.46	19.8	–	11.7	–	1.35	0.47	–
	3	175	0.42	–6.0	0.70	9.2	0.37	0.58	19.6	0.48	12.8	0.78	1.38	0.50	0.15
	4	210	0.56	–5.0	–	5.9	–	0.73	19.3	–	11.5	–	1.30	0.38	–
	5	215	0.60	–5.0	0.64	6.5	0.41	0.80	18.9	0.44	12.2	0.68	1.33	0.42	0.15
	6	290	0.68	–5.2	0.65	5.7	0.46	0.90	18.7	0.39	11.2	0.63	1.32	0.40	0.22
B	1	228	0.28	3.3	0.85	15.8	0.37	0.54	22.9	0.35	16.3	0.53	1.93	0.34	0.01
	2	215	0.28	2.2	0.80	14.8	0.35	0.54	21.7	0.40	14.8	0.63	1.93	0.35	0.02
	3	215	0.28	1.3	–	13.9	–	0.54	20.5	–	13.9	–	1.93	0.35	–
C	1	150	0.24	–1.7	0.78	9.9	0.35	0.41	15.7	0.65	10.5	0.97	1.7	0.44	0.005
	2	160	0.24	–1.7	0.78	9.9	0.35	0.41	15.7	0.65	10.5	0.97	1.7	0.44	0.005
	3	170	0.29	–1.3	0.75	10.0	0.35	0.49	15.7	0.82	10.8	0.93	1.69	0.44	0.008
	4	170	0.29	–1.3	0.74	10.0	0.35	0.49	15.7	0.82	10.8	0.93	1.69	0.44	0.01
	5	195	0.34	–1.3	–	8.4	–	0.57	14.5	–	9.8	–	1.68	0.40	–
	6	205	0.35	–1.9	–	8.0	–	0.59	14.4	–	9.8	–	1.69	0.40	–
	7	225	0.37	–1.9	0.62	7.5	0.40	0.62	14.5	0.75	9.9	–	1.68	0.38	0.06
	8	230	0.37	–1.9	0.62	7.5	0.40	0.62	14.5	0.75	9.9	–	1.68	0.38	0.06
	9	258	0.38	–1.4	–	7.8	–	0.64	14.8	–	10.0	–	1.68	0.37	–
	10	258	0.38	–1.0	–	7.9	–	0.64	15.0	–	10.2	–	1.68	0.36	–
	11	258	0.38	–1.0	–	8.3	–	0.64	16.5	–	11.1	–	1.68	0.35	–

a) Thermic efficiency of heat recuperation from ventilating air (at $x_{i1} = x_{i2}$). The relationship suits to all practical applications of recuperation exchangers, i.e. measurement on testing tables as well as in the buildings for animal production.

$$\eta_{Rt} = \frac{t_{e2} - t_{e1}}{\dfrac{V_i \,.\, \rho_i \,.\, c_{pi}}{V_e \,.\, \rho_e \,.\, c_{pe}} \,.\, t_{i1} - t_{e1}} \quad [-] \qquad (2)$$

b) Enthalpy efficiency of heat recuperation from ventilating air (at $x_{i1} > x_{i2}$).

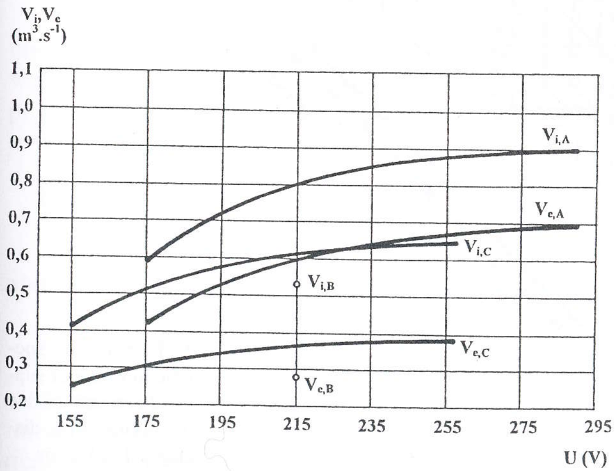
$$\eta_{Rh} = \frac{h_{e2} - h_{e1}}{\dfrac{V_i \,.\, \rho_i}{(1 + x_{i1})} \,.\, \dfrac{(1 + x_{e1})}{V_e \,.\, \rho_e} \,.\, h_{i1} - h_{e1}} \quad [-] \qquad (3)$$

The relationship describes better the state of exchange of heat and mass, particularly in regeneration exchangers. In well designed, manufactured and operated recuperation exchanger, conversion of mass occurs, and the values of thermic and enthalpy efficiency are identical.

c) The ratio of mixing of air flows (calculated according to the study written by Marquardt, 1983) is expressing in new recuperation heat exchangers the quality of production version and older equipment signals bringing of non-sealed segments in operation (the lower exchange of specific moistures between air flows, the tighter the exchanger).

$$E_m = \frac{x_{e2} - x_{e1}}{x_{i1} - x_{e1}} \quad [-] \qquad (4)$$

where: V – volume air flow ($m^3.s^{-1}$)
Q – heat power (W)
t – air temperature (°C)
ρ – specific air weight ($kg.m^{-3}$)
c_p – specific heat air capacity under the constant pressure ($J.kg^{-1}.K^{-1}$)
h – specific air enthalpy ($J.kg^{-1}$)
x – specific air moisture ($kg.kg^{-1}$)
i index – internal barn air



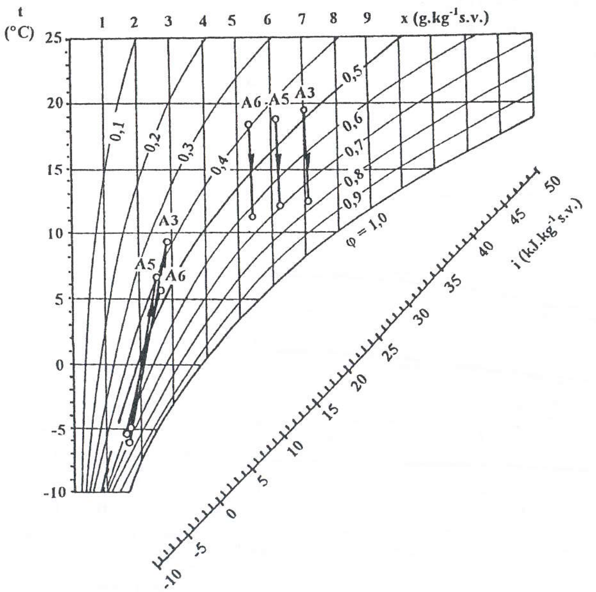
2. The effect of contamination on heat-exchanging surfaces of exchanger on the changes in volume flows of supplied external air V_e and discharged stable air V_i

e index – external outdoor air
R index – recuperation
Rt index – thermic recuperation
Rh index – enthalpy recuperation
index *1* – starting state
index *2* – final state

The results of measurements showed that cooling of discharged air at measurements A and B was done without water condensation from air ($x_{i1} = x_{i2}$) and the measurement C with partial water condensation from air ($x_{i1} > x_{i2}$).

Fig. 2 shows the effect of heat-exchanging surfaces of exchanger on the changes in volume flows of discharging V_i and supplying air V_e. Results show a great contamination of heat-exchanging surfaces of exchanger that is manifested by reduction of volume flow of air at the same regulation tension U of fan engine. For example U = 215 V is according to A5 and B2 in new exchanger V_i = 0.8 m^3.s^{-1} and after 11 months of operation V_i = 0.54 m^3.s^{-1}.

There was an interesting finding that higher contamination of heat-exchanging surfaces was on the side of supplied air. At U = 215 V in new exchanger V_e = 0.60 m^3.s^{-1} and after 11 months of operation under the same regulation tension of fan engine V_e = 0.28 m^3.s^{-1}. On the side of discharging ventilating air volume air flow dropped after 11 months of operation by ΔV_i =



3. Increase of specific moisture of supplied external air due to mixing with discharged stable air

0.26 m^3.s^{-1}, i.e. to 67.5% of an original value. Volume flow of supplied external air fell for the same time by ΔV_e = 0.32 m^3.s^{-1}, i.e. to 46.6% of original value. The reason of surprisingly high contamination of heat-exchanging surfaces on the side of external air was dry windy weather and great dustiness of surrounding roads. However, the main and more distinguished factor was mixing of discharging air into air supplying due to non-tightened surfaces of recuperation exchanger. This hypothesis has been confirmed later decrease of air flows mixing E_m. The ratio of airs mixing was E_m = 0.15 to 0.22 in a new exchanger. After 11 months of operation E_m fell due to silting up of non-tightened segments of exchanger by dust particles to 0.01 to 0.02. It follows from comparison of the values of measurement (C) of exchanger working with partial water condensation from discharged air that E_m decrease is yet more marked, E_m = 0.005 to 0.010 after 9 months of operation.

Decrease of volume flow on the side of discharging ventilating air due to contamination of heat-exchanging surfaces takes place almost at the same time during dry operation as well as with partial water condensation from air. While contamination of heat-exchanging surfaces on the side of external supplying air takes place more intensively during wet operation of exchanger. This fact evidently follows from faster silting up of non-tightened segments by dust particles during water condensation from air, what has been confirmed by faster decrease of efficiency of air mixing E_m.

Fig. 3 shows an increase of specific moisture x_{e2} in Mollier's $i - x$ diagram of supplied air *e* caused by mixing with discharged ventilating air *i*. The diagram shows the results of the measurements A3, A5, A6 on new exchanger working without water condensation from discharged air $x_{i1} = x_{i2}$.

In contamination of heat-exchanging surfaces, except the change in volume flows, decrease of coefficient of transfer of heat *k* through the dividing wall of the exchanger is another significant factor affecting obtained heat power and thermic efficiency of recuperation. Coefficient of heat transfer for new exchanger can be expressed by equation:

$$k = \frac{1}{\frac{1}{\alpha_i} + \frac{s}{\lambda} + \frac{1}{\alpha_e}} \quad [\mathrm{W.m^{-2}.K^{-1}}] \qquad (5)$$

where: α_i – coefficient of heat transfer on the side of discharged ventilating air (W.m^{-2}.K^{-1})
α_e – coefficient of heat transfer on the side of outdoor supplied air (W.m^{-2}.K^{-1})
s – thickness of dividing wall between discharging and supplying air (0.00055 m)
λ – coefficient of thermal conductivity of dividing wall (50 W.m^{-1}.K^{-1})

Coefficient of heat transfer α can be expressed from criterial equation formed on the basis of experimental testing of exchanging of similar geometrical accomplishment (D o b i á š et al., 1984):

$$Nu = \frac{\alpha \,.\, D_h}{\lambda_l} = 0{,}20 \,.\, \mathrm{Re}^{0{,}67} \,.\, Pr^{0{,}33} \quad [-] \tag{6}$$

where: λ_l – coefficient of thermal conductivity of flowing air ($W.m^{-1}.K^{-1}$)
D_h – hydraulic diameter, determined by the length of exchanger table in the direction of flow rate of flowing air (m)
Re – Reynold's number (–)
Pr – Prandtl's number (–)
Nu – Nusselt's number (–)

After substitution of coefficients of heat transfer (α_i = 18.45 $W.m^{-2}.K^{-1}$; α_e = 18.7 $W.m^{-2}.K^{-1}$), calculated from equation (6) and the values s, λ into the equation (5), coefficient of heat transfer through the dividing wall of new exchanger k = 9.29 $W.m^{-2}.K^{-1}$.

Later exchangers, e.g. RVD A 10, manufactured by Šumstav Český Krumlov reach at V_i/V_e = 1.3 of the value of coefficient of i heat transfer k = 12.9 $W.m^{-2}.K^{-1}$ (Š t u l c et al., 1987). Significantly higher value is given by forming of tables of heat-exchanging areas, lower pitch of air channels (0.01) as well as material (Al) of dividing walls of the exchanger.

After 11 months of the operation of new exchanger in the calf-barn on milk nutrition on the side of discharging ventilating air non-homogenous layer on approximate thick-layer s_i = 3 mm was formed. This layer consisted mainly of dust of feeding mixture and hay. On the side of supplying outdoor air was formed non-homogenous layer of soil dust of approximate thickness s_e= 2 mm. The change of the value of coefficient of heat transfer through the dividing wall of exchanger induced by formation of layers on heat resistances

$$R_i = \frac{s_i}{\lambda_i} \quad [m^2.K.W^{-1}] \text{ and } R_e = \frac{s_e}{\lambda_e} \quad [m^2.K.W^{-1}]$$

can be expressed by equation:

$$k = \frac{1}{\dfrac{1}{\alpha_i} + \dfrac{s_i}{\lambda_i} + \dfrac{s}{\lambda} + \dfrac{s_e}{\lambda_e} + \dfrac{1}{\alpha_e}} \quad [W.m^{-2}.K^{-1}] \tag{7}$$

where: λ_i, λ_e – coefficients of thermal conductivity of different layers ($W.m^{-1}.K^{-1}$)

Coefficients of thermal conductivity λ can be determined according to the superposition principle by the following relationship (B l a h o v e c , 1993):

$$\lambda = \left(\frac{v_s}{\lambda_s} + \frac{v_w}{\lambda_w} + \frac{v_l}{\lambda_l}\right)^{-1} \quad [m^2.K.W^{-1}] \tag{8}$$

where: v_s, v_w, v_l – volume concentration of dry matter, water, air (–)
λ_s, λ_w, λ_l – coefficients of thermal conductivity of dry matter, water, air ($W.m^{-1}.K^{-1}$)

λ_i = 0.05 $W.m^{-1}.K^{-1}$ and λ_e = 0.06 $W.m^{-1}.K^{-1}$ follow from the diagram of coefficients of thermal conductivity of a material composed of water, dry matter and air (B l a h o v e c , 1993), calculated according to the superposition principle. By substitution into the relationship (7) the value of the coefficient of heat transfer through the dividing wall of exchanger after 11 months of operation, k = 4.97 $W.m^{-2}.K^{-1}$ is obtained. The calculated value is considered only as informative, because its exactness is influenced by some factors that were not taken into account in the calculation. Above all these are the changes of coefficients of heat transfer α_i, α_e due to the shape of the surfaces of the formed layers and presumption of identical thicknesses of the layers s_i, s_e along the whole surface of dividing walls. The determination of the values λ_i, λ_e, too, is affected by non-homogenousness of layers and by estimate of different volume concentrations of components of dry matter, water and air.

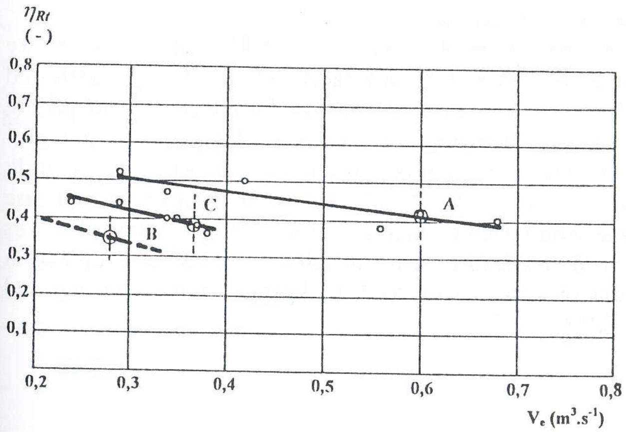
Statistically processed values of thermic efficiency $\eta_{Rt} = f(V_e)$ are presented in Fig. 4. Linear regression functions are in the form:

$$\eta_{Rt.A} = 0.602 - 0.319\, V_e \quad [-] \tag{9}$$

$$\eta_{Rt.B} = 0.390 - 0.155\, V_e \quad [-] \tag{10}$$

$$\eta_{Rt.C} = 0.586 - 0.558\, V_e \quad [-] \tag{11}$$

The circle and vertical line show the changes in the volume flow of outdoor air V_e at the constant revolutions of the fan corresponding to regulation tension U = 215 V after 9 (C) and 11 (B) months of work of recuperation exchanger. The volume flow of a new exchanger (A) was V_e = 0.60 $m^3.s^{-1}$. For measurements A and C through the measured points the lines were plotted through the point from the measurement B the line of identical trend with



4. Changes in practical thermic efficiencies η_{Rt} of the exchanger ZV 3-022 due to contamination of heat-exchanging surfaces

the previous states A and C. In the region of measured values, it was the line dependence statistically found.

Significant decrease of efficiency η_{Rt} is evident from graph in Fig. 4, reaching almost 20% at the constant air flow V_e as well as at the constant regulation tension U or constant revolutions of the fan. The effect of contamination of heat-exchanging areas on the efficiency of exchanger is almost identical during dry operation and during operation with condensation of water from discharging air.

When no regular cleaning of heat-exchanging areas of exchangers is provided, the given decrease of efficiency of heat recuperation from ventilating air in production conditions should be considered. In addition, to reach the required air flows to discharge moisture and harmful substances from the stable and to supply of outdoor air, it is necessary to increase regulation tension of fan engines what results in increased consumption of electric energy for drive of ventilating fans.

Based on the results of testing where the main guidance are falls of air flows (Fig. 2) and decrease of thermic efficiency of exchangers (Fig. 4), after 9 months of work the function of an exchanger can be characterised as insufficient. At almost linear growth of contamination production parameters in about half of this time can be considered as convenient. With respect to these results and experience from operation in large-capacity calf-barn of the farm Kosořín and prophylactory of the large-capacity cow house of the farm Oucmanice (both the farms are a part of the Co-operative Farm Zálší), recommended intervals of cleaning of heat-exchanging surfaces of table exchangers were determined. In prophylactories and calf-barns on milk nutrition interval of cleaning of exchangers of impurifying layer at practically linear growth was recommended to be 6 months. In calves on plant nutrition the cleaning interval is 4 months in the case when during distribution of dusty hay ventilating fans are switched of, and 2.5 months, if fans are continuously working.

Realization of recuperation exchangers in the stables of large-capacity calf-barn of the Co-operative Farm Zálší at Kosořín was positively manifested immediately in the first year of operation, through for the reasons of testing ordinary cleaning of heat-exchanging surfaces of exchangers was not provided purposefully. Increase of the temperature in the barn of calves on milk nutrition in winter season by 4 to 6 °C and decrease of relative air moisture to $\varphi = 0.40$ to 0.55 was manifested by increase of weight increments of calves and by decrease of their death rate. After installation of table recuperation exchangers the weight increment rose by 0.12 kg per animal and day and death rate of calves fell by 3.43%.

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Analýza změn termické účinnosti při provozu deskových rekuperačních výměníků.

Scientia Agric. Bohem., *31*, 2000: 285–297.

Cílem příspěvku je analyzovat vlivy znečištění teplosměnných ploch deskového rekuperačního výměníku na změny objemových toků vzduchu, účinnost využití druhotného tepla a směšování vzduchu při provozu výměníků s parciální kondenzací vody ze vzduchu a při provozu suchém.

Měření se uskutečnilo ve dvou stájích velkokapacitního teletníku ZD Zálší v Kosoříně. Ve stáji pro 615 až 620 telat v mléčné výživě a ve stáji pro 640 až 700 telat v rostlinné výživě byly ověřovány deskové rekuperační výměníky ZV 3-022, vyráběné ZD Horní Brusnice.

Základní rozměry výměníku a typy ventilátorů:

- rozměry: délka 0,98 m, šířka 0,57 m, výška 0,63 m
- hmotnost: 280 kg
- počet článků: 16
- světlost vzduchového kanálu mezi články: 0,015 m
- teplosměnné plochy: ploché desky z pozinkovaného plechu v síle 0,55 mm
- objemový průtok vzduchu: 0,4 až 0,7 $m^3.s^{-1}$
- vstupní průřez pro odpadní odváděný vzduch: 0,62 m^2
- vstupní průřez pro venkovní přiváděný vzduch: 0,34 m^2

– ventilátory: 2 ventilátory LAN W 450.4 s elektronickou regulací otáček ALR 3/21 v rozsahu 10 až 100 % pneumatických otáček.

Schéma měření teplot t, relativních vlhkostí φ a rychlostí proudění vzduchu v je uvedeno na obr. 1. Indexem i jsou označeny hodnoty odváděného stájového vzduchu, indexem e hodnoty přiváděného venkovního vzduchu. Z naměřených rychlostí proudění vzduchu a průtočných průřezů byly stanoveny objemové toky vzduchu V.

Výsledky měření jsou zpracovány v tab. I. Skupina hodnot označených A byla naměřena ihned po instalaci výměníků. Hodnoty označené B byly naměřeny ve stáji pro telata v mléčné výživě po 11 měsících provozu rekuperačního výměníku. Skupina hodnot označených C byla naměřena ve stáji pro telata v rostlinné výživě po 9 měsících provozu rekuperačního výměníku. Z naměřených hodnot byly podle vztahů (2), (4) vypočteny termické účinnosti η_{Rt} a účinnosti mísení toků vzduchu E_m. Výsledky měření ukázaly, že ochlazování odváděného vzduchu při měření A a B probíhalo bez kondenzace vody ze vzduchu ($x_{i1} = x_{i2}$) a měření C s parciální kondenzací vody ze vzduchu ($x_{i1} > x_{i2}$). Hodnota x znamená měrnou vlhkost vzduchu (kg.kg^{-1} s. v.).

Na obr. 2 je znázorněn vliv znečištění teplosměnných ploch výměníku na změny objemových toků odváděného V_i a přiváděného V_e vzduchu. Výsledky ukazují značné znečištění teplosměnných ploch výměníku, které se projevuje snížením objemových toků vzduchu při stejném regulačním napětí U motoru ventilátoru. Z grafu na obr. 2 je patrné, že dochází ke značnému znečištění teplosměnných ploch i na straně venkovního přiváděného vzduchu V_e. Příčinou byla vysoká prašnost okolních komunikací a zejména mísení odváděného vzduchu do vzduchu přiváděného, způsobené netěsností dělicích teplosměnných ploch výměníku.

Na obr. 3 je znázorněno v Mollierově i – x diagramu zvýšení měrné vlhkosti x_{e2}, přiváděného vzduchu e, způsobené mícháním s odváděným větracím vzduchem i. V diagramu jsou zakresleny výsledky měření A3, A5, A6 na novém výměníku.

Znečištění teplosměnných ploch ovlivňuje negativně rovněž hodnotu součinitele prostupu tepla k dělicí stěnou výměníku, a tím i získaný tepelný výkon a účinnost rekuperace η_R. Součinitel prostupu tepla dělicí stěnou nového výměníku, vypočtený podle vztahů (5) a (6), má hodnotu k = 9,29 W.m^{-2}.K^{-1}. Po 11 měsících provozu se na obou stranách teplosměnných ploch vytvořily nehomogenní vrstvy o přibližných tloušťkách s_i = 3 mm a s_e = 2 mm. Na straně odváděného vzduchu tvořil vrstvu s_i prach krmné směsi a sena o součiniteli tepelné vodivosti λ_i = 0,05 W.m^{-1}.K^{-1}. Vrstvu s_e na straně vzduchu přiváděného tvořil prach zeminy o součiniteli tepelné vodivosti λ_e = 0,06 W.m^{-1}.K^{-1}. Výpočtem podle vztahů (7) a (8) jsme získali informativní hodnotu součinitele prostupu tepla dělicí stěnou výměníku po 11 měsících provozu k = 4,97 W.m^{-2}.K^{-1}.

Statisticky zpracované hodnoty účinnosti $\eta_R = f(V_e)$ jsou znázorněny na obr. 4. Lineární regresní funkce mají tvar vztahů (9), (10) a (11).

Z grafu na obr. 4 je zřejmý významný pokles účinnosti η_{Rt} dosahující téměř 20 % při stálém průtoku vzduchu V_e i při stálém regulačním napětí U, resp. stálých otáčkách ventilátoru. Vliv znečištění teplosměnných ploch na účinnost výměníku je téměř stejný při suchém provozu a při provozu s kondenzací vody z odváděného vzduchu.

Na základě výsledků ověřování byly stanoveny doporučené intervaly čištění teplosměnných ploch deskových výměníků. V profylaktoriích a stájích pro telata v mléčné výživě je doporučený interval čištění výměníku 6 měsíců. U telat v rostlinné výživě je interval čištění 4 měsíce v případě, že při rozvozu sena jsou vypínány větrací ventilátory, a 2,5 měsíce, pokud jsou ventilátory neustále v provozu.

Realizace rekuperačních výměníků ve stájích velkokapacitního teletníku ZD Zálší v Kosoříně se projevila pozitivně ihned v prvním roce provozování. Zvýšení teploty ve stáji pro telata v mléčné výživě v zimním období o 4 až 6 °C a snížení relativní vlhkosti vzduchu na φ_i = 0,40 až 0,55 se projevilo zvýšením přírůstku telat a snížením jejich úhynu. Po instalaci deskových rekuperačních výměníků se zvýšil přírůstek o 0,12 kg.d^{-1} a úhyn telat poklesl o 3,43 %.

stáj; telata; větrání; druhotné teplo; deskový výměník; teplosměnná plocha; účinnost výměníku; poměr mísení vzduchu

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