EFFECTS OF POULTRY DUST ON PERFORMANCE OF A THERMOSIPHON HEAT RECOVERY SYSTEM[1]

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A previous paper reported the performance and suitability of the thermosiphon-type heat exchanger for improving the winter heat balance in a caged layer chicken house. Continuation of the work with better instrumentation and different air filtration demonstrated that surface-loading washable air filters were more effective and easier to maintain than the deep-loading type. A maintenance routine consisting of vacuuming the filter inlet surface every 2 days, and washing the heat exchanger and filters every 30 days maintained satisfactory performance.

A design is proposed to illustrate how the thermosiphon heat recovery system could be built into a modern commercial-sized caged chicken house.

INTRODUCTION

This paper describes a continuation of work reported in a previous paper (Larkin et al. 1975) to which the reader is referred for a general description of the problem, the thermosiphon heat exchanger and the test arrangement.

The preliminary work demonstrated that the thermosiphon heat exchanger can improve the winter heat balance in a chicken house by transferring heat from the warm exhaust air to the cold intake air. Maximum heat available was limited by the cooling of the saturated exhaust air down to near freezing, and a simple thermostat control solved the freezing problem. The filter systems used were not satisfactory, and measurements were not accurate enough to properly assess the effects of filter and heat exchanger fouling on rates on heat transfer. This paper reports further experience with improved instrumentation and more suitable air filters.

IMPROVEMENTS IN INSTRUMENTATION

In the previous tests, the air flows were measured with a rotating vane anemometer. In these tests, flow was measured by venturi sections in the inlet and exhaust ducts. As the ducts do not provide standard entry and exit lengths, the barn ducting was duplicated in a laboratory and the venturis were calibrated "as installed" against a standard orifice. Pressur tappings were installed to measure the pressure drops across the heat exchanger and the filters, read by means of an inclined tube manometer which was calibrated against a precision manometer.

The original mercury-in-glass thermometers were replaced with copper-constantan thermocouples. These were calibrated in place by immersion in an ice-bath. Two thermocouples in each duct were sufficient to measure the air temperatures immediately prior to entering the heat exchanger, because the temperature is uniform across the flow. For air streams leaving the heat exchanger where temperature might not be uniform, six thermocouples were spaced across each duct.

TEST PROCEDURE

During the test there were 1,030 egg-strain laying chickens in room 2 and 570 in room 1, all in cages. In both rooms the temperatures were maintained at about 60 F (15 C) throughout the tests. Five tube banks were used in the heat exchanger.

Whenever possible, complete sets of readings were taken before and after cleaning the filters. On some occasions when the outside ambient temperature was low and the exhaust flow was reduced due to fouling of the filters, the heat exchanger exhaust outlet temperature was low enough to cause the inlet fan to cycle. In these cases no readings could be taken until the filters had been cleaned, restoring the exhaust flow to its normal value and raising the exhaust outlet temperature so that the exhaust thermostat no longer caused cycling of the inlet fan.

11 December 1974 (day 1) to 21 January 1974 (day 41)

The heat exchanger fans were set to give a flow of 600 scfm (17 m³/min) with heat exchanger and filters clean. This is equivalent to about 0.35 scfm (.01 m³/min) per hen and represents the minimum average day-night winter ventilation rate for satisfactory control of moisture in caged-hen housing. The filter consisted of one Farr type 44 filter (Farr Co. Ltd., 21 Kern Road, Don Mills, Ontario) in room 1 and one Farr type F/S filter in room 2, both located in ceiling exhaust openings. Both filter types were washable heavy-duty units 2 inches (5 cm) deep made from crimped and stacked galvanized steel mesh, with heavy galvanized steel frames. The Farr type 44 was a galvanized high-velocity, deep-loading, impingement-type filter, whereas the Farr type F/S was a cheaper surface-loading design (costing about $16 for a 16 X 25-inch (40.6 X 63.5-cm) face area) and especially suited to cleaning air containing a high percentage of lint. Exposed filter area from each room was adjusted to give a filter velocity of about 155 ft/min (47 m/min) based on net filter area.

The filters were removed from their ceiling frames and washed under a tap at intervals of 1 - 5 days. On day 41 the heat exchanger was cleaned in place with a hose nozzle connected to the sink tap.

21 January 1975 (day 41) to 21 March 1975 (day 100)

For this period of the test, the heat exchanger fans were reset to give flows of about 800 scfm (23 m³/min), or about 0.5 scfm (0.014 m³/min), which is a generally accepted winter ventilation rate for caged layers.

The filter system was changed for two reasons. Firstly it was found that the fouling collected on the inlet surface of the filters. the Farr type F/S being a "surface loading" filter was found to be easier to clean than the Farr type 44. Also, there was considerable fouling in the heat exchanger during the first part of the test, indicating a need for more effective filtering. Again, the Farr F/S has a finer structure than the Farr 44 (0.070-inch mesh as opposed to 0.125-inch mesh). For the remainder of the test, one Farr F/S was used in room 1 and two similar filters were used in room 2. The increased filter area was provided to reduce the filter pressure drop, with filter velocity reduced to about 107 ft/min (33 m/min).

The filter cleaning procedure was also changed. It was found that it was much easier, and almost as effective, to clean the

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surface-loading filters in position with a vacuum cleaner as it was to remove them and wash them under a tap. After 21 days of regular maintenance with only a vacuum cleaner, the filter flow resistance immediately after cleaning had increased slightly, from 0.01 to 0.03 inch (.025 to .076 mm) water gauge, compared with a total system pressure drop of 0.4 inch (1.0 mm) water gauge. The filters were removed and washed and the resistance returned to the original "clean" value.

On 4 March 1975 (day 83), the heat exchanger exhaust section and the filters were washed again. The test ended on 21 March 1975 (day 100).

RESULTS

Fouling can affect the performance of the system in two ways. Firstly, the exhaust flow is reduced by fouling in both the heat exchanger and the filters. To assess these effects separately, friction factors have been calculated for these two parts of the system. By analogy with the expression for a plain duct, flow through a resistance can be represented by the equation:

$$f = \frac{2g\, \Delta p\, \rho\, A^3}{Q^2\, l\, P}$$

where

f = friction factor (dimensionless)
g = gravitational acceleration (ft/sec^2)
Δp = pressure drop across resistance (lb/ft^2)
ρ = fluid density (lb/ft^3)
A = flow cross-section area (ft^2)
Q = air flow (lb/sec)
l = length of flow passage (ft)
P = wetted perimeter of duct (ft)

When Q, Δp and ρ have been measured, f can be calculated. Figure 1 shows the variation in f over the test period for the exhaust side of the heat exchanger. Figure 2 shows the average filter friction factor plotted against days since cleaning.

The second possible effect of fouling is on the heat transfer between the exhaust air and the heat exchanger surface. As the exhaust flow and the system temperatures vary from day to day, a simple statement of the heat transferred is no indicaion of the degree of fouling. An effectiveness factor has been **defined as the ratio of the heat transfer** measured experimentally to the heat transfer predicted for the heat exchanger for the same inlet temperatures and flow if the heat exchanger were clean. This dimensionless ratio is a measure of the net effect of fouling on the heat transfer by the heat exchanger.

The significance of this parameter lies in its variation during the test. Whether the value is greater or less than unity depends of whether the design correlations are conservative or optimistic and also on the errors in the experimental measurements.

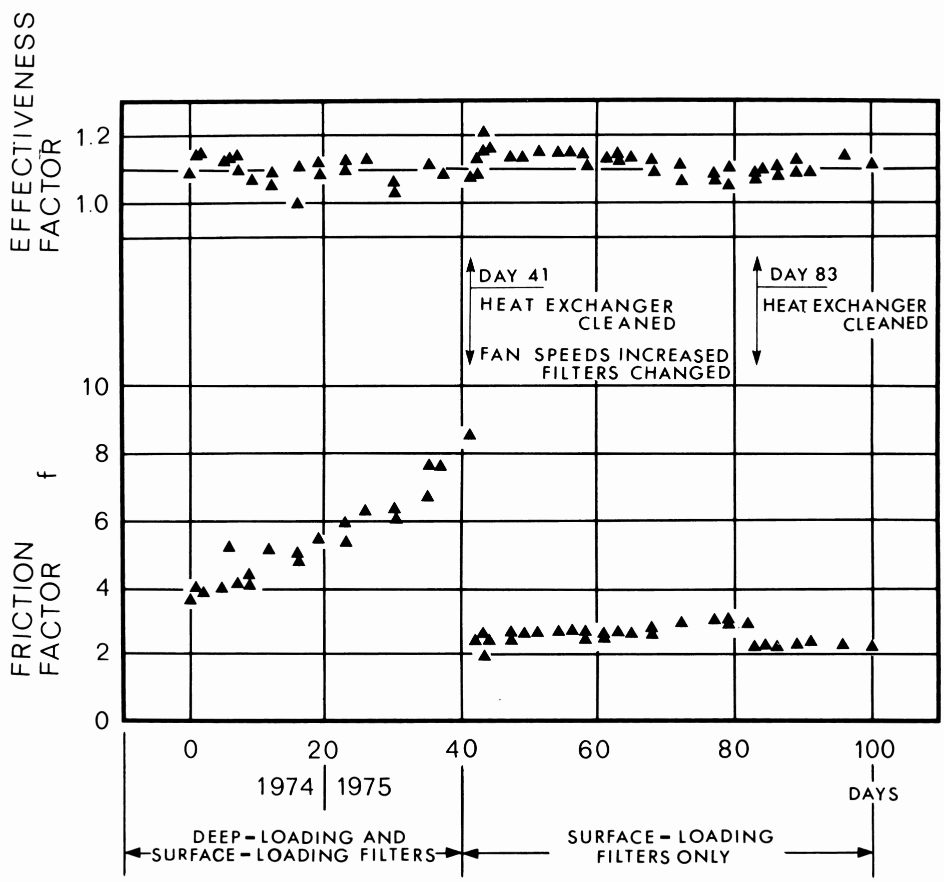

Figure 1. Effects of fouling on heat exchanger effectiveness factor and friction factor.

Figure 1 shows the effect of fouling on heat exchanger effectiveness factor and exhaust side friction factor over the duration of the test. There is some scatter in the experimental results but it is clear that the cleaning procedured are effective and the heat exchanger performance can be maintained indefinitely.

The values of f immediately after the first cleaning on day 41 show a significant decrease from the "clean" values at the beginning of the test. This can be explained by two factors: some of the fins in the heat exchanger were found to be out of alignment and were straightened at this time, and there is a tendency for f to decrease as Q increases. For results with the same fan speed, Q does not vary greatly and this effect has been ignored. The change in fan speed on day 41 caused a considerable change in flow (nearly 50%) which would make a significant contribution to the decrease in f.

During the period up to day 41, f increased by 2.7% per day. With the improved filter system from day 41 to day 83, f increased by only 0.5% per day. It is more difficult to put a value on the change in effectiveness factor but it appears to be a decrease of about 0.2% per day.

The inlet side of the heat exchanger showed no signs of fouling during the test and received no maintenance.

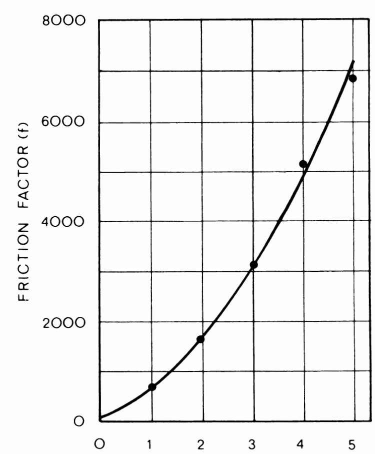

Figure 2. Change in filter friction factor with time since cleaning.

Throughout the test, the thermostat at the outlet from the exhaust side of the heat exchanger was set at 35 F. On several occasions the thermostat caused the inlet fan to cycle on and off but there was no indication of ice formation in the tube banks. This method of preventing freezing in the heat exchanger is simple and appears to be satisfactory.