THE ECONOMICS OF HEAT RECOVERY SYSTEMS FOR ANIMAL SHELTERS

B.S. Larkin[1] and J.E. Turnbull[2]

[1] *Division of Mechanical Engineering, National Research Council of Canada, Ottawa, Ontario K1A 0R6; and* [2] *Engineering and Statistical Research Institute, Research Branch, Agriculture Canada, Ottawa, Ont. K1A 0C6. Contribution No. 650[2]*

Received 5 Sept. 1979

Larkin, B.S. and **J.E. Turnbull.** 1979. The economics of heat recovery systems for animal shelters. Can. Agric. Eng. 21: 53-59.

In severe climates, livestock housed in a building over the winter generate moisture, and the building must be ventilated to control moisture. The energy required to heat the incoming air can be extracted from the exhaust air by a heat exchanger. The savings that result have been calculated for growing-finishing pigs and caged laying chickens. The heat exchanger cannot always satisfy design requirements for the building during extremely cold weather; during these short periods the choice is to accept some substandard ventilation or make up the heat deficit by other means. In one of the cases considered the load factor of the system was low and the rate of return on investment was only average. In three cases with a higher load factor the savings were large. A design is proposed to illustrate how the thermosiphon heat exchanger could be fitted into a typical swine growing/finishing barn.

INTRODUCTION

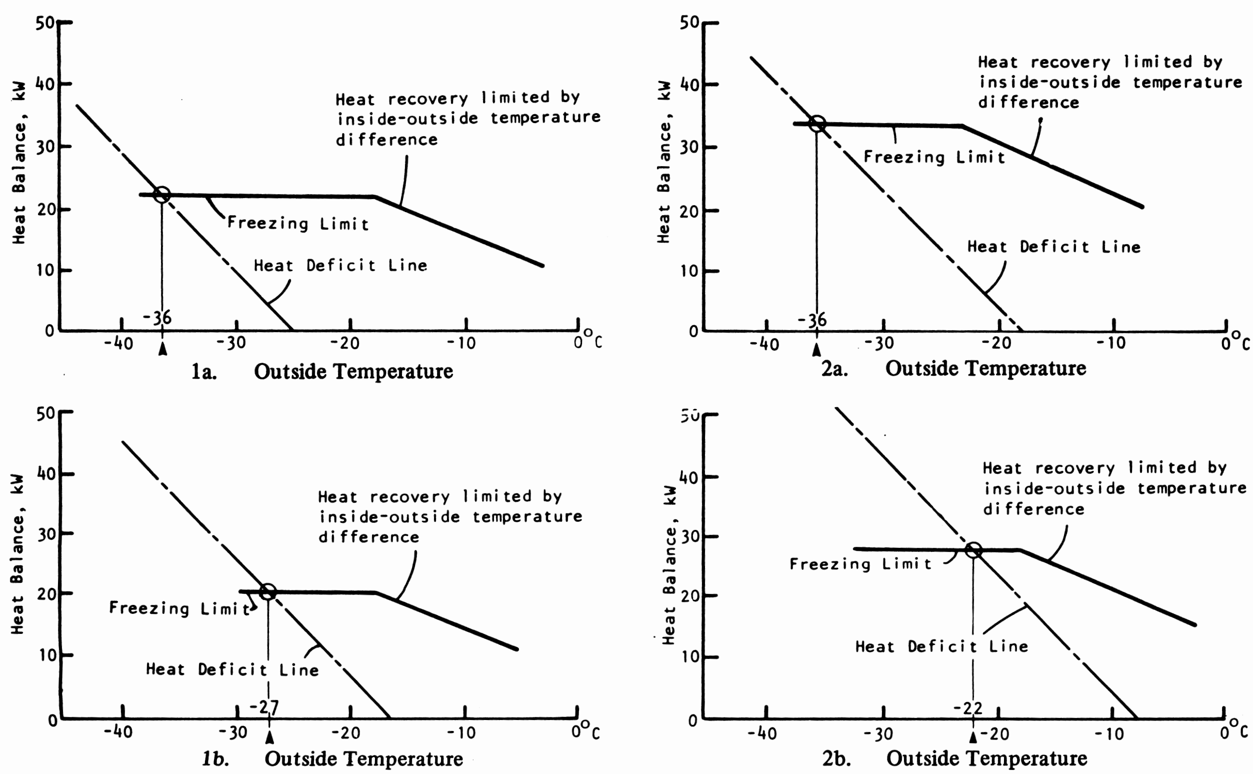
The two components of winter heat loss from an animal shelter are the conduction losses (through the structure) and the ventilation losses (heat required to bring the incoming ventilation air up to the room temperature). In typical modern farm buildings, the ventilation component is the larger of the two; ventilation heat loss depends on the ventilation rate and the outside temperature. The livestock generate considerable quantities of heat, sufficient to maintain the desired room temperature with adequate ventilation when the outside temperature is moderate. But whenever the outside air temperature is too low, supplementary heat is required; otherwise, even the minimum acceptable ventilation rate will cause the room temperature to fall.

Conventional supplementary heat systems use electric heat, oil, propane or natural gas. One alternative is a heat recovery system, using a heat exchanger to extract heat from the exhaust air to preheat the inlet air. Theoretically, it is simple to design and construct a heat exchanger that, when clean, will recover the required heat. In practice, heat recovery systems have not been used very often, partly because heat exchanger performance falls off rapidly due to dust in typical farm conditions.

Using an experimental installation in a poultry house, Larkin et al. (1975) and Larkin and Turnbull (1977) have shown that a system consisting of filters and a thermosiphon heat exchanger is practical. The filters remove most of the fouling and can be cleaned quickly and easily. The heat exchanger too must be cleaned, at longer intervals, and is designed for easy, convenient washing. This paper discusses the economics of such a system for two typical farm applications. Although it is based on the thermosiphon heat exchanger, many of the points raised are relevant for other types of heat exchangers.

CONVENTIONAL SUPPLEMENTARY HEAT SYSTEM

The usual ventilation system in northern U.S.A. and Canada consists of several fans controlled by thermostats set in temperature steps so that the number of fans operating and hence the ventilation rate decrease with outside temperature. The smallest fan is sized to give a minimum ventilation rate that just falls short of



Figures 1 and 2. Heat balance diagrams. 1. Case 1, chickens (a) daytime, (b) nightime. 2. Case 2, chickens (a) daytime, (b) nightime.

controlling relative humidity during the coldest weather anticipated.

If supplementary heat is provided it is switched on and off by thermostat to maintain the desired room temperature while the smallest fan is running. Figure 1 shows how the deficit in the room heat balance increases as the outside ambient decreases, for a typical set of conditions; this is the heat that should be supplied by the supplementary heating system.

HEAT RECOVERY SYSTEM

If a heat recovery system were to be used it would replace the smallest fan, and ideally the supplementary heating system as well. One possible scheme for a 500-pig grower-finisher unit is shown in Fig. 5. Fresh air is ducted (7) into the inlet side of the heat exchanger (8), through the inlet fan (9) and into an insulated duct (10) in the attic from which it is distributed throughout the building. Exhaust air (1) enters the heat exchanger compartment through filters (2), goes through the exhaust fan (3), and the exhaust side of the heat exchanger (4), then to outdoors (5) by way of a wind-resistant weather-hood.

Ideally, a thermostat-controlled supplementary heat system automatically supplies just enough heat to maintain room temperature. A heat recovery system behaves differently; the heat recovered depends upon the inlet conditions of the two flows. Figure 1 shows the variation in heat recovery with outside temperature for a typical set of day and night conditions in a caged poultry unit. During mild winter weather the heat exchanger recovers more heat than is required to maintain the room temperature. In this case the room temperature will rise, and the next larger fan in the ventilation system will start switching to control the temperature in the building.

Freezing

If the exhaust flow is cooled too far then the saturated exhaust will deposit ice in the heat exchanger, beginning at the coldest part of the exhaust section. If this is allowed to continue the heat exchanger will block up entirely. Freezing imposes a natural limit on heat recovery, as shown in Fig. 1. One satisfactory method of control is to install at the outlet from the exhaust side of the heat exchanger a thermostat which switches the inlet fan to low speed just before freezing starts. The cooling effect on the exhaust flow is then reduced. Any ice which has formed is melted and, when the final exhaust air has warmed sufficiently, the thermostat returns the inlet fan to full speed and normal operation resumes. During very cold weather the inlet fan cycles between full and low speed.

The heat recovery may be limited either by the effectiveness of the heat exchanger or by the freezing limit. In Fig. 1 the horizontal part of the "heat recovery" line represents this upper practical *limit.*

At the point where either the freezing line or the heat recovery line (whichever is lower) intersects the heat deficit line, the heat balance is in equilibrium. If the outside temperature falls lower, the heat recovery is insufficient to maintain design conditions in the building. A room thermostat then cycles the exhaust fan between full speed and half speed, reducing the average ventilation rate and allowing the humidity to rise above the design level.

Dust Loading

In the system tested and reported previously by Larkin and Turnbull (1977), the main effect of dust is to increase the flow resistance of the filters. Secondary effects are to increase the flow resistance and decrease the heat transfer coefficients in the heat exchanger. These effects all tend to decrease the heat recovery and some loss of performance is inevitable between cleanings. The cleaning routine must be a satisfactory compromise between the work involved and the heat recovery required.

As reported previously, the dirt accumulating in the filters over a period of 2 days reduced the exhaust flow by 15%. The temperature interval through which the exhaust flow can be cooled is limited by the onset of freezing so that a 15% reduction in flow means that the possible heat recovery is reduced by 15%. The filters can be cleaned in a few minutes with a vacuum cleaner. Over a period of 30 days, dirt in the heat exchanger reduced heat recovery by 6%. The heat exchanger is cleaned by removing the top cover of the exhaust section and washing the tube banks with a water jet, taking at most 30 min. With this cleaning routine, just before cleaning filters and heat exchanger after 30 days, the heat recovery was about 79% of the "clean" heat recovery. However, this situation is not as unfortunate as it may appear at first glance.

During most of the winter the outside air temperature is in the range where 79% of the "clean" heat recovery is sufficient to maintain the design minimum ventilation rate. Whenever the weather is cold enough to require the maximum heat recovery, the system could be cleaned more frequently (e.g. every day). Similarly, during milder winter weather the intervals between cleanings can be extended. To compensate for fouling, in future designs the design flow could be chosen to be (say) 5% higher than the minimum ventilation rate.

ESTIMATING ENERGY REQUIREMENTS

The desirability of a heat recovery system depends on the severity of the winter, the heat and moisture production of the livestock, the shelter construction and insulation, the cost of energy, etc. The procedure used here is to select typical cases, define the basis for comparison and calculate the useful heat recovery for these particular sets of conditions.

Conventional supplementary heating is normally designed to supply enough energy to maintain the minimum ventilation rate down to the design outside temperature. Usually the design temperature is slightly above the expected minimum temperature. Occasional short periods of extreme cold result in substandard ventilation; this is usually acceptable for the sake of reducing capital costs. In this study -34°C has been taken as the design outside temperature.

TABLE I. SASKATOON WINTER TEMPERATURES

	Avg of daily maximums (°C)	Avg of daily minimums (°C)
Oct.	+11	- 1
Nov.	- 1	-11
Dec.	- 9	-19
Jan.	-13	-24
Feb.	- 9	-21
Mar.	- 3	-14
Apr.	+ 9	- 3

Theoretically, a heat recovery system could also be designed to correspond exactly to design conditions by variations of the heat exchanger parameters. For convenience, the authors used the basic design of the experimental system described in previous papers (1975, 1977), varying only the number of banks of finned tubing to give a close approximation to design requirements.

Heat Recovery System Specifications

Filter
- Face velocity 0.5 m/s
- Filter mesh (approx.) 1.8 mm

Thermosiphon heat exchanger
- Effectiveness 0.40
- Face velocity through tube banks 2 m/s
- Pressure drop per tube bank 23 Pa (2.3-mm water guage)
- Fin/tube surface ratio 20.3/liter
- Tubes: outside diam. 25.4 mm; inside diam. 22.9 mm; 4 tubes per fin bank, spaced 76.2 mm oc
- Fins (pressed onto tubes): thickness 0.38 mm, depth 50.8 mm, spacing 4 mm.

CLIMATE

The location selected for this analysis was Saskatoon, Saskatchewan, representative of the coldest climates in which commercial Canadian livestock operations are common. For Saskatoon, the National Building Code of Canada (1977) lists 6077 heating degree-days below 18°C; this compares with 6145 degree-days at Grande Prairie, Alberta, and with 6037 degree-days at Brandon, Manitoba. Table I gives more winter temperature information for Saskatoon.

Saskatoon temperature records were searched to find, for each winter month, an example with average temperature close to the 30-yr average temperature for that month. In this way an "average" winter was assembled and the number of hours at each temperature was counted for use in the analysis.

LAYING CHICKENS IN CAGES

Specifications (Canada Plan Service, plan 5212)

10 000 White Leghorn laying chickens (see Table II), in triple-decked cages

Building exterior length and width, 46.8 × 10.2 m

Floor to ceiling height, 2.95 m

Wall and ceiling insulation 150 mm glass fiber (RSI-3.5).

The Canadian Farm Building Code (1977) recommends a continuous minimum ventilation rate of 0.23 L/s (0.5 ft³/min) per laying hen. However, using removal of vaporized moisture reported by Ota and McNally (1961) as the basis for ventilation design, the daytime minimum rate was calculated to be 0.14 L/s per chicken. This was the figure used in this analysis.

Case 1, Chickens at 16°C

For "day operation," Fig. 1a is the heat balance diagram. The "heat deficit" line shows the supplementary heating rate calculated to maintain the design conditions defined above. It is obtained by subtracting the sensible heat produced by the chickens from the sum of the building heat loss and the heat required to bring the incoming air up to room temperature. The heat recovery line shows the heat recovered by the heat exchanger. The freezing limit shows the restriction on heat recovery due to freezing in the last tube bank of the heat exchanger.

Ideally, a conventional supplementary heat system supplies just the heat required to maintain the minimum ventilation rate. This heat corresponds to the heat deficit line until the design outside temperature of -34°C is reached. If the outside temperature falls further the supplementary heat system runs continuously at full power.

A heat recovery system will recover heat as shown by either heat recovery line or the freezing limit. Until one of these lines crosses the heat deficit line, more heat is recovered than is necessary to achieve design conditions within the building. This results in more ventilation and relative humidity less than 75%; no commercial value was assessed to this improvement, although there may be other real benefits obtained from the increased ventilation and improved air quality made possible by a heat recovery system. Over the range of outside temperature above the equilibrium point, the heat recovery system is given credit only for the heat required to maintain the minimum ventilation rate. At lower temperatures, when the heat recovery is insufficient to make up the heat deficit, the heat recovery system is given credit for all the heat recovered.

For the "night operation," Fig. 1b shows the heat balance conditions. According to Ota and McNally (1961), chickens at night give off less heat and moisture than during the day. Based only on humidity control, the minimum night ventilation rate should be 0.1 L/s per chicken. In practice this variation between day and night is usually ignored. Similarly, in this analysis it will be assumed that the minimum ventilation rate and supplementary heat system designed for daytime conditions will be used unaltered at night.

As the chickens are generating less heat than in the daytime the supplementary heat system can only maintain the ventilation rate of 0.14 L/s down to an outside temperature of -27°C. If the temperature drops further the ventilation rate will fall. The chickens generate less moisture than in the daytime so that the relative humidity will not rise above 75% until the outside temperature is about -40°C. But humidity is not the only criterion by which ventilation should be judged. The ventilation at low temperature may be substandard from the point of view of ammonia and odors. As before, the heat recovery system is given credit only for heat required to maintain the minimum ventilation rate until the equilibrium point is reached, after which it gets credit for all heat recovered. See Table III for the results of this analysis.

Case 2, Chickens at 21°C

The higher room temperature is of interest because it can increase feed conversion efficiency. If the minimum ventilation rate is calculated to give a relative humidity of 75% at design conditions, the ventilation rate is found to be 0.12 L/s per chicken, compared with 0.23 L/s recommended in the Canadian Farm Building Code. It is probably unwise to reduce the ventilation rate too far due to

TABLE II. DAY/NIGHT HEAT AND MOISTURE PRODUCTION OF CAGED WHITE LEGHORN LAYING CHICKENS, AVG. 1.8 kg (Ota and McNally 1961)

	Case 1		Case 2	
Design min. ventilation			0.14 *L*/(s·chicken)	
Room temperature	16°C		21°C	
Room relative humidity	75%		75%	
No. of thermosiphon tube banks	4		5	
	Day	*Night*	*Day*	*Night*
Sensible heat (W/chicken)	7.8	6.2	7.6	5.6
Vaporized moisture (g/(h·chicken))	4.9	3.4	5.1	4.0

TABLE III. ANNUAL ENERGY BUDGET 10 000 HENS AT SASKATOON

	Case 1	Case 2
Design conditions	16°C, 75% RH	21°C, 0.14 *L*/(s·chicken)
Heat recovery (kWh)	8 400	24 780
Energy consumption of heat exchanger fans (kWh)	1 200	2 500
Capacity of equivalent supplementary heat system (kW)	19	30

TABLE IV. HEAT AND MOISTURE PRODUCTION OF GROWING/FINISHING PIGS, AVG. 54 kg (Bond et al 1959)

	Case 3	Case 4
Room temperature (°C)	16	21
Room relative humidity (%)	75	75
Sensible heat production (W/pig)	88	70
Vaporized moisture (g/(pig·h))	84	100
No. of thermosiphon tube banks	6	7

TABLE V. ANNUAL ENERGY BUDGET, 500 GROWING/FINISHING PIGS AT SASKATOON

	Case 3	Case 4
Design conditions	16°C, 75% RH	21°C, 75% RH
Heat recovery (kWh)	30 480	48 970
Energy consumption of heat exchanger fans (kWh)	3 180	5 290
Capacity of equivalent supplementary heat system (kW)	22.8	31.4

odors; 0.14 L/s has been selected for this analysis, giving the heat balance diagrams in Fig. 2. At night, if the outside temperature falls below -22°C the heat recovery is not sufficient to maintain 75% RH. See Table III for details of the energy saving.

GROWING-FINISHING PIGS

It is assumed that cleaning a heat exchanger for pigs will be the same as for chickens.

This has not been justified experimentally. The system will probably foul up less quickly than in a chicken house, but the dirt may be more difficult to remove. A typical population of 500 growing-finishing pigs was used for this analysis, housed as in Table VI and as follows:

Building Specifications (Canada Plan Service, plan 3028)

Exterior length and width, 31.8 × 10.8 m
Floor to ceiling height 2.6 m
Wall insulation 90 mm glass fiber (RSI-2.1)
Ceiling insulation 150 mm glass fiber (RSI-3.5)
Foundation perimeter insulation 50 mm polystyrene board (RSI-1.4).

Case 3, Pigs at 16°C

The winter minimum design flow rate at 16°C is 2.5 L/(s·pig). Separate data were not available for waking and sleeping pigs so that only an average calculation is possible.

Figure 3 is the heat balance diagram. Pigs are considerably less self sufficient than chickens as far as heat is concerned. The equilibrium point is at -25°C outside temperature; at lower temperatures the relative humidity in the building will increase. See Table V for details of energy budget.

Case 4, Pigs at 21°C

As in the case of chickens a higher room temperature is of interest because it can give better feed conversion efficiency. The minimum ventilation rate is 2.0 L/(s·pig).

Figure 4 is the heat balance diagram. The equilibrium point is at an outside temperature of -24°C. The increase in humidity at lower temperatures is shown in the upper part of Fig. 4. See Table V for details of the energy budget.

COST COMPARISON

Capital costs and running costs vary considerably, depending on the energy sources available. Also, any comparisons are very sensitive to future increases in energy costs. Thus an analysis can only be very general and must be interpreted in the light of local conditions. The calculations are simple so that it is easy to substitute local costs in the illustrations.

Electrical energy is widely used and is the most convenient for comparing heating-ventilating costs. Installation of electric heating was estimated to cost \$100/kW, and electric energy was *\$0.02/kWh* with a

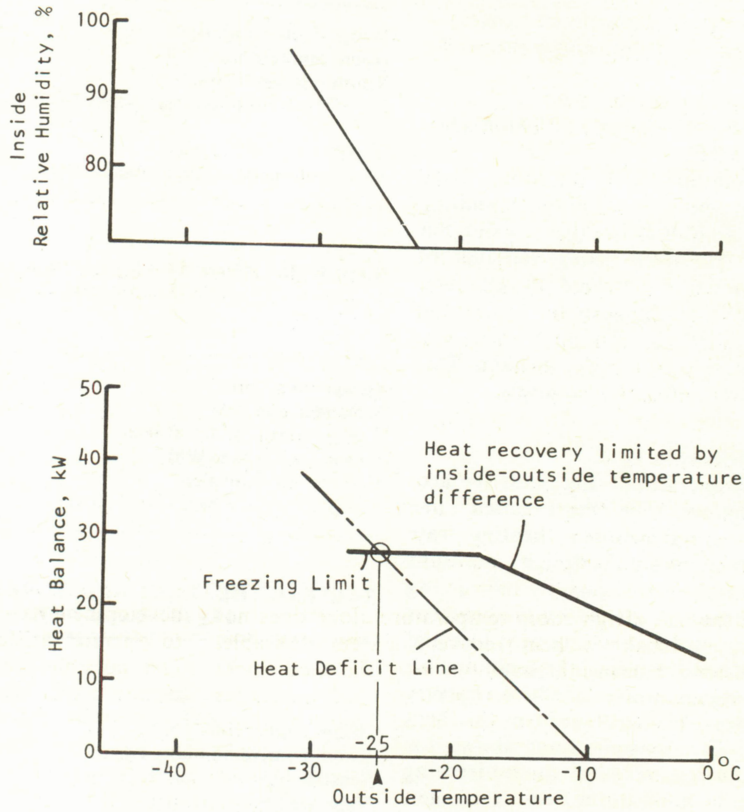
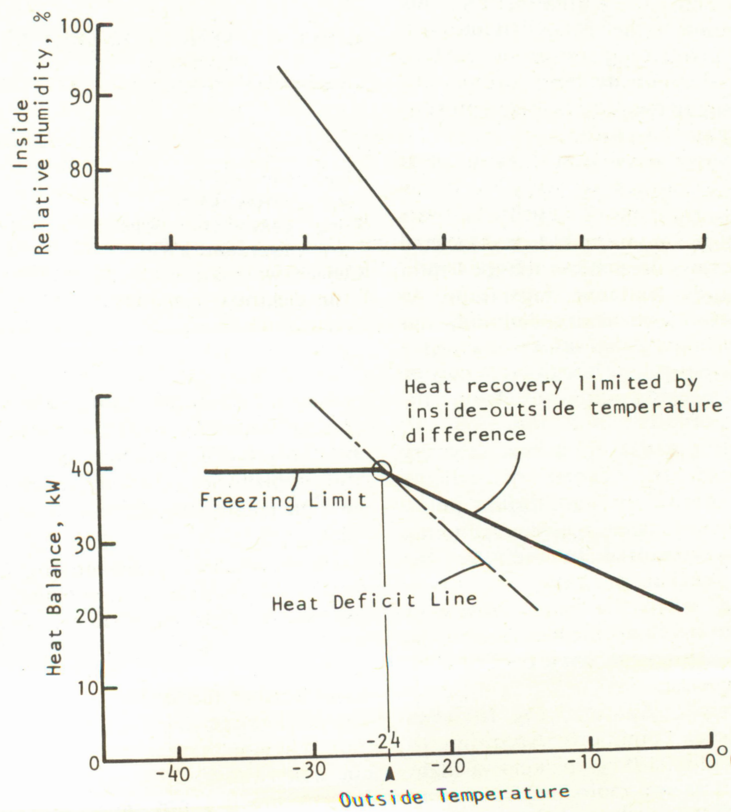


Fig. 3 Heat Balance Diagram, Case 3 Pigs



Figures 3 and 4. Heat balance diagrams. 3. Case 3 pigs. 4. Case 4 pigs.

TABLE VI. ANNUAL COST COMPARISON: ELECTRIC SUPPLEMENTARY HEAT SYSTEM vs. HEAT RECOVERY SYSTEM

		Laying chickens		*Growing/finishing pigs*	
		Case 1	Case 2	Case 3	Case 4
(1)	Room temperature	16°C	21°C	16°C	21°C
(2)	Design standard	75% RH	0.014 *L*/s	75% RH	75% RH
	Capital costs ($)				
(3)	Electric supplementary heat system	1900	3000	2280	3140
(4)	Step 1 fan conventional system	200	200	200	200
(5)	Cost† of heat recovery system	4200	4500	4240	3650
(6)	Additional cost of heat recovery system = (5)-(4+3)	2100	1300	1760	310
	Annual operating costs ($)				
(7)	Energy cost for electric heat	168	495	610	980
(8)	Demand charge	152	240	184	250
(9)	Energy cost for heat exchanger fans	24	50	64	106
(10)	Demand charge for heat exchanger fans	16	16	16	16
(11)	Cost of labor for cleaning heat recovery system @ $5/h	100	150	150	
(12)	Operating cost saving due to heat recovery system = (7) + (8) - (9 + 10 + 11)	180	519	564	

† Estimated from prices of similar commercial equipment, see text.

demand charge of $2/kW, for each of the four winter months.

The cost of the heat exchanger has been estimated by comparison with the cost of a heat exchanger made by Q-Dot Corp., Dallas, Texas, and marketed in Canada by KeepRite Products Ltd., Brantford, Ont. This is basically similar to the experimental NRC unit but with some detail differences.

Labor for cleaning the heat recovery system was charged at $5.00/h.

DISCUSSION

The results show the importance of the load factor of the heat recovery system. The less self-sufficient in heat production the livestock are, the more attractive a heat recovery system becomes. Factors which can increase the economic advantage of a heat recovery system are:

(a) *High room temperature* — Apart from considerations of feed efficiency some livestock require a high room temperature.

(b) *High standard of ventilation* — If, for example, the usual flow rate of 0.23 L/s is necessary for the laying chicken, rather than 0.14 L/s as used here, the economics of the heat recovery system would be much enhanced.

(c) *Poorly insulated buildings* — Heat recovery systems may be attractive for old buildings with inadequate insulation.

(d) *Low winter temperatures* — Saskatoon has a very cold winter. In less severe climates a heat recovery will be less attractive.

(e) *The cost of energy* — There are some areas where electricity already costs significantly more than $0.02/kWh as assumed here. The cost of energy will probably continue to increase; a heat recovery system could be viewed as protection against future high prices.

High room temperature alone does not make a heat recovery system desirable. Farrowing barns for sows and their litters require high temperature but little ventilation. Brooding chickens need only minimum ventilation at the time when the building must be kept at a high temperature. In neither case is a heat recovery system attractive. Similarly a requirement for high ventilation rates does not call for heat recovery if the building temperature is low (cattle loose housing, for example).

In areas where the power supply is unreliable a heat recovery system has the advantage that the standby generator required is an order of magnitude smaller than would be required for an electric heating system.

The analysis reported here is conservative in some respects. No credit has been taken for the heating effect of the two heat exchanger fans. All of the electrical energy supplied to the inlet fan is useful heat input to the building and about 40% of the energy supplied to the exhaust fan is recovered. This amounts to about 1½ kW while the fans are running. The power input to the exhaust fan also raises the freezing limit slightly.

It was assumed the freezing limit occurs when the temperature of the last tube bank reaches 0°C, as calculated in the heat exchanger computer program. Experimental evidence appears to show that ice blockage does not occur until this temperature is lower than 0°C, and there is reason to believe that the freezing limits given here are pessimistic.

One important consideration is the durability of the heat recovery system. The filters, fans and thermostat controls should present no special problems. The thermosiphon heat exchanger is comparatively novel. It is a very simple device; research and development have shown that it can be made to operate satisfactorily for long periods. One possible problem is corrosion due to contaminants such as ammonia in the exhaust air. The cost estimates for the heat exchanger are based on all-aluminum construction. With chickens there was no significant corrosion of the aluminum fins after three winters so that aluminum should be a satisfactory material.

The design of the heat recovery system has been optimized. Changes in face velocity, fin pitch, etc., may improve the economics of the system. Use of a larger filter area would reduce the cleaning work load.

No reduction in performance due to fouling has been assumed. This is equivalent to assuming that the system will be cleaned as frequently as necessary to maintain a close approximation to "clean" performance.

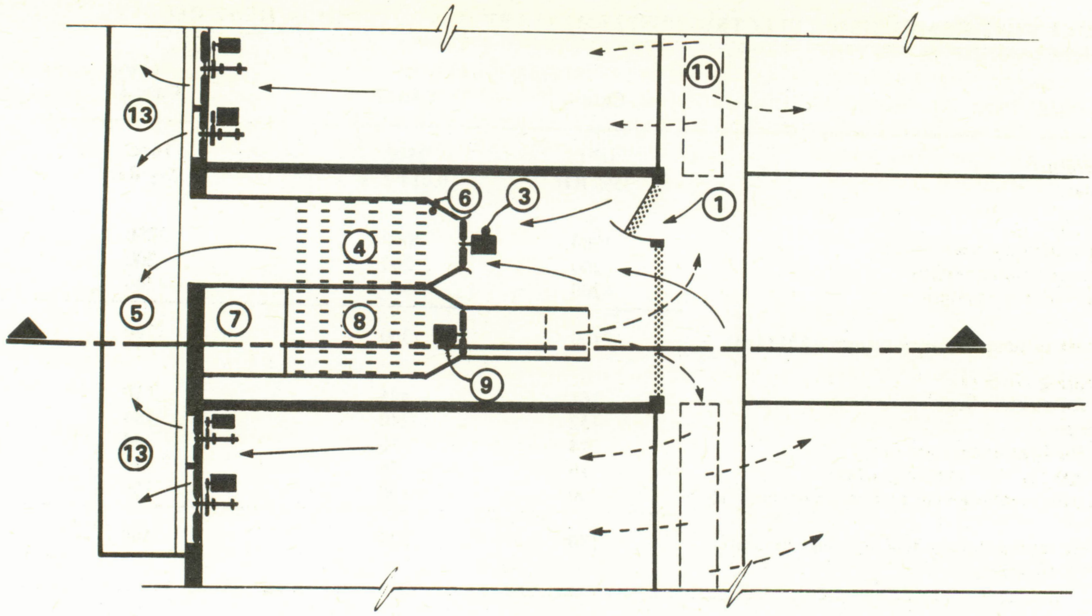
CONCLUSIONS

Chickens, Case 1 (Room temperature 16°C, relative humidity 75%).

The heat recovery system can recover sufficient heat during the day to fulfill design requirements. At night, the daytime ventilation rate cannot be maintained when outside temperatures are below -27°C but the relative humidity does not exceed 75%. The saving in operating costs (with the assumptions used here) does not justify the extra capital cost of a heat recovery system.

Chickens, Case 2 (Room temperature 21°C, minimum ventilation 0.14 L/s).

During the day a heat recovery system can recover sufficient heat to fulfill design requirements. At night the design ventilation rate cannot be maintained when outside temperatures are less than -22°C, although the relative humidity does not



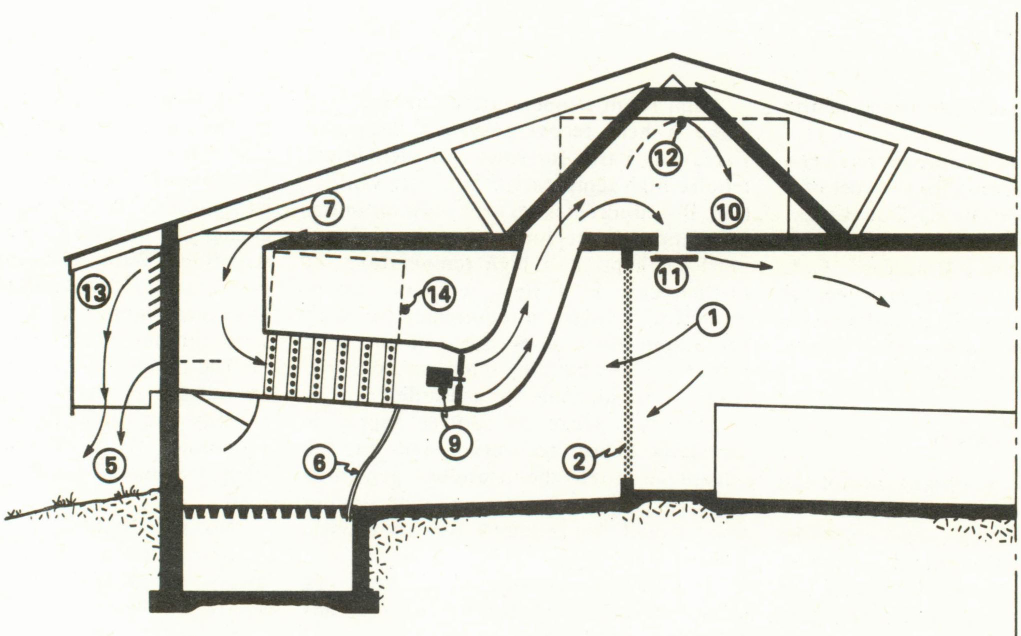


Figure 5. Proposed arrangement of thermosiphon heat recovery system in a swine finishing barn (500 hogs, CPS Plan 3428). 1. Warm room air to dust filter(2). 2. Dust filter 2 × 2 m, includes access door. 3. Exhaust fan to heat exchanger(4), 1200 l/s @ 13 Pa static. 4. Heat exchanger cooling section, 6 tube banks 0.9 × 0.7 m. 5. Cooled exhaust to outdoors. 6 Condensate drains at lower corner, hose to manure trench. 7. Cold air from ventilated attic to (8). 8 Heat exchanger warming section, 6 tube banks 0.9 × 0.7 m. 9. Intake fan, to (10). 10. Insulated attic duct, to (11). 11. Adjustable baffled air inlet slot. 12. Summer air inlet doors 2.4 × 1.2 into (10), both gables. 13. Warm weather exhaust fans. 14. Access covers hinge up for washing (4) and (8).

exceed 75%. The extra capital cost of a heat recovery system is justified by the saving in annual operating cost.

Pigs, Case 3 (Room temperature 16°C, relative humidity 75%) and **Case 4** (Room temperature 21°C, relative humidity 75%).

Below -25°C outside, a heat recovery system cannot recover *sufficient heat* to maintain the design standard of ventilation. In an average Saskatoon winter there are about 200 h below -29°C, giving a significant increase in humidity. A dual system could be used: a heat recovery system to recover most of the heat required and a conventional supplementary heat system which will operate only when the heat recovery is insufficient. The annual energy input of the conventional system would be small but most operators would accept substandard ventilation some of the time rather than go to the complication and expense of a dual system. The large saving in energy cost makes a heat recovery system very attractive for growing/finishing pigs.

SUGGESTIONS FOR A COMMERCIAL SWINE INSTALLATION

Larkin and Turnbull (1977) previously proposed a design for the installation of a thermosiphon heat recovery system in a typical commercial poultry house (Canada Plan Service, plan 5212).

Figure 5 shows a corresponding design proposed for a grower-finisher unit to house

500 pigs, as in Cases 3 and 4 (above). Here it was most convenient to dedicate one pen-space for the heat exchanger, located on one side at mid-length of the barn. Air ductwork is simplified by using the ventilated attic space as a wind-proof winter intake plenum supplying fresh air (7) to the heat exchanger. This offers a further heat gain due to solar heating through the roof on sunny winter days.

To distribute warmed intake air to the full length of the building, the heat exchanger connects to a well-insulated duct (10) built into the roof trusses. This same duct also serves as a summer air supply by opening big doors (12) in both gable ends, by stopping the heat exchanger fans, and by readjusting the ceiling air inlet baffle (11) to handle the greatly increased air flow.

For easier maintenance, exhaust air filters are made large, located in the center service hallway, and protected with a pig-proof grill at the bottom. The equipment room size is determined by accessibility for opening and washing.

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