HORSE BARN VENTILATION PROBLEM AND SOLUTION

BY

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Report submitted as partial requirement
for the course 336-490N

NOVEMBER, 1985

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ABSTRACT

This paper is a study of ventilation problem in the horse barn at 19212 Gouin St. and a recommended solution.

The relative humidity readings which were taken by using a sling psychrometer were quite high (80 % or over) . The temperature drop across the inside layer of air was substantial, the surface temperature of the ceiling was likely to fall below the dew-point temperature (5°C) . Condensation would then occur on the ceiling surface.

Because it is an existing old building and because of the layout, natural ventilation is not paractical.

The Enercon fresh air, model 400 or the Del-Air heat exchanger, model A-800 (series 1000) for their easy-clean, fast defrost features, simplicity of installation, and costs are recommended.

ACKNOWLEDGMENT

The author would like to acknowledge with appreciation the assistance of Professor G.S.V. Raghavan and M. Gilles Bolduc as the adviser of this project and for his guidance and support throughout this study.

Special thanks to Nick Micopolus for his help.

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I. INTRODUCTION

Environmental modification is a basic reason agricultural structures are built and the level of importance in environmental conditions has a significant impact upon many phases of agricultural production.

Proper housing of animals is essential to the efficient operation and profitable management of a farm. Proper housing means, first of all, providing shelter from the elements- rain, snow, and wind. However, of equal or greater importance is the fact that a farm building must also provide, in addition to more protection, optimum atmospheric condition for animals.

The building should protect animals from extremes of heat and cold, from sudden temperature changes, and from drafts.

Ventilation-insulation is one of the most neglected, but one of the most important aspects of stabling. In livestock buildings, proper ventilation and insulation can make the difference between profit and loss. These buildings cost more, but because good ventilation improves feed conversion and sanitation, its cost is soon offset by greater profits.

An efficient farm ventilating system is necessary, first, for regulating inside temperatures and second, for removing excessive quantities of moisture and odors given off by livestock.

Many livestock buildings are hot and oppressive in spring and fall, and damp and cold in winter. Heat is supplied primarily by the animals. It could be categorized as sensible and latent heat. The body heat given off by convection and radiation that affects the temperature of the air and surrounding is sensible heat. The latent heat of vaporization is released during condensation of the moisture that has been respired by the animals in vapor form. It refers to the energy involved in a change of state and can not be measured with a thermometer, evaporation of water or respired moisture from the lung are examples.

Animals give off moisture during normal respiration, and the higher the temperature, the greater the moisture. This moisture should be removed from buildings. The amount of air required to remove the moisture produced determines the ventilation rate. Lack of removing extra water vapor from barn cause the vapor finds its way into the building materials, and deterioration happens. Dampness in a barn also contributes to wet litter, higher disease rate, and rapid corrosion of metal parts.

Because of high humidity, the air inside the barn becomes stuffy and contaminated with stale odors such as ammonia, production drops, and lastly, condensation on interior surfaces occurs.

The problem of moisture condensation as it relates to high value of relative humidity and poor insulating material, involves the change from the gaseous state of water, known as water vapor,to the liquid state (water) or the solid state (frost or ice). There are many examples of moisture condensation in a barn such as sweating and frosting of windows, condensation within a wall and condensation on the ceiling surface.

The ceiling of such livestock usually are more vulnerable than any other places. Furthermore, water dripping from the ceiling may cause much more damage than water running down the wall or window.

The condensation on the ceiling surface was the severest problem of the barn and its solution is the object of this paper.

In winter, ventilation is a critical factor. Enough air must pass through the structure to keep it dry, yet it must be kept to a minimum to prevent heat loss. More air movement is required in spring and fall when the temperature differential is less.

Ventilation systems can use natural or forced air. Forced air is easier to control but will incur the expense of purchasing, maintaining and operating fans. It was not possible to use natural ventilation because of low inside temperature. Therefore, forced air is the only solution to this problem.

Since energy conservation is a concern, researchers have been exploring the possibilities of minimizing the cost of ventilation. Heat exchangers have revealed a great potential to lower the energy cost of ventilation.

A heat exchanger is a device which provides for transfer of internal thermal energy between two or more fluids at differing temperatures. Heat transfer between the fluids take place through a separating wall. Since the fluids are separated by a heat transfer surface, they do not mix. Common examples of such heat exchangers are the shell-and-tube exchangers, automobile radiators, condensers, evaporators, air preheaters, and "dry" cooling towers. If no phase change occurs in any of the fluids in the exchanger, it is sometimes referred to as a sensible heat exchanger. There are no internal thermal energy sources in a heat exchanger, ruling out fired heaters, electric heaters, and nuclear fuel elements. If the fluids are immiscible, the separating wall may be eliminated, and the interface between the fluids serves as a heat transfer surface as in a direct contact heat exchanger.

A heat exchanger consists of the active heat exchanging elements such as a core or a matrix containing the heat transfer surface, and passive fluid distribution elements such as headers, manifolds, tanks, inlet and outlet nozzles or pipes, or seals. Usually there are no moving parts in a heat exchanger; however, there are exceptions such as a rotary regenerative exchanger, in which the matrix is mechanically driven to rotate at some design speed.

The heat transfer surface is the surface of exchanger core which is in direct contact with fluids and through which heat is transferred by conduction. That portion of the surface which also separates the fluids is referred to as primary or direct surface. To increase heat transfer area, appendages known as fins may be intimately connected to the primary surface to provide extended, secondary, or indirect surface.

Heat exchangers are used in the process, power, automotive, air conditioning, refrigeration, cryogenics, heat recovery, alternate fuels, and manufacturing industries, as well as key components of many products available in the marketplace. These heat exchangers may be classified according to the transfer processes, degree of surface compactness, construction features, flow arrangements, number of fluids, and fluid phase changes or process function.

In various construction modifications, undoubtedly, shell-and-tube exchangers are the most commonly used basic heat exchanger configuration in the process industries. The shell-and-tube heat exchanger provides fairly large ratios of heat transfer area to volume and weight. It provides this surface in a form that is relatively easy to construct in a wide range of sizes and that is mechanically rugged enough to withstand normal shop fabrication stresses, shipping and field erection, and external stresses encountered under normal operating conditions. The shell-and-tube exchanger can be cleaned reasonably easily, and those components most subject to failure-gaskets and tubes-can be easily replaced. Special construction features allow this type to meet almost any conceivable service, including extremely low and high temperatures and pressures, large temperature differentials, vaporizing and condensing services, and severely fouling and corrosive fluids.

Del-Air is another kind of heat exchanger for the winter heating and ventilation problem in agricultural applications. Del-Air Exchanger also use the energy in the waste exhaust air to preheat the incoming fresh air. This minimizes the heat loss and therefore greatly reduces and often eliminates supplimental heat. The Del-Air Exchanger will pre-heat outdoor air to within 65-70 % of room temperature.

II. OBJECTIVES

The objectives of this paper are to:

(a) Find the cause of water vapor condensation on the ceiling.

(b) Recommend a proper system to overcome this problem.

III. REVIEW OF LITERATURE

Experimental evidence on the direct effect of climate on domestic livestock has been obtained from two sources : direct observations in the field, and observations on animals kept in controlled - temperature laboratories or psychrometric chambers. The disadvantage of direct observations is that it is difficult to set up adequately controlled experiments in the field, while the disadvantage of observations in a psychrometric chamber is that even in the largest chamber only a small number of animals can be studied, while it is known that there is a profound difference in the individual response of animals of the same breed to climatic stress (Payne and Hancock, 1957).

All domestic livestock are homeotherms. That is, they attempt to maintain their body temperatures within the range most suitable for optimal biological activity.In order to do this they must preserve a thermal balance between their heat production or gain from the environment and their heat loss to the environment.

Metabolic heat production depends on (a) basal heat production for maintaining essential body processes such as body temperature, (b) digestive heat production depending on the type of digestive system of the animal and on the quantity and quality of the food that it ingests, (c) muscular heat production depending on the amount that the animal moves about in foraging, and (d) increased metabolism due to productive processes such as growth and reproduction. Of the methods of heat loss, evaporative loss is potentially the most important. This subject is discussed in detail by Findlay and Beakley (1954).

Evaporative heat loss depends upon the ambient air temperature, the amount of available moisture, the area of evaporating surface, the absolute humidity and the degree of air movement.

It has been exeprimented in horse that below 35 ^{0}C there is a rise in vaporisation per unit surface area with increasing air temperature and that this rise is less in tropical than in temperature type horse.

The amount of available moisture depends partly on the quantity of sweat produced by the animal. Although both temperature and tropical breed of horse possess sweat glands, those of the latter are larger and more numerous, according to Australian work (Nay and Hayman, 1956)

The move to a fully intensive system of husbandry took place gradually during the 'fifties'. This change in housing was accelerated by developments and it was then fully 'controlled environments'soon came into being, and their success encouraged producers to do likewise, but it has to be realized that the total cost of controlling temperature and humidity may be as high as a considerable percent of the total fixed capital invested (Ffiske 1966).

Insulation helps to conserve heat so that room temperatures remain within the optimum range required ; similarly it helps to exclude solar radiation in the summer which can raise environmental temperatures well above the level required. Heat to be conserved comes from stock as well as any supplied artificially, and the less of the latter the better, because of cost (W.P. Blount 1969).

As hot air rises heat losses through the roof prove greatest which is why this structure must be insulated as completely as possible, particularly as its surface is considerably greater than that of the walls. The latter loss more heat proportional to the area of glass windows present, unless these have been double glazed. Some sets of data, (a) satisfactory range of temperatures, (b) heat input and (c) sources of heat loss has been indicated by (W.P.Blount 1969).

The overall heat transfer coefficient is the rate at which heat will flow from one side to another of any given material or combination of materials (U $w/m^2{}^\circ C$). If building materials get compressed or wet they lose much of their insulation properties. Very careful watch is necessary to see that these materials receive careful attention during building construction.

The main requirements to calculate U are (i) 'k' value of insulating materials, and (ii) certain factors, e.g. internal, external, surface-to-air and cavity. Some of U, internal surface resistances and external surface resistances were indicated by W.P. Blount (1969).

For many years medical men have recognized that a considerable number of diseases are produced by impurities in air, and Dr. Parkes, in "Manual of Hygiene" gives an extensive list of such diseases 1894.

It was unnecessary then to argue at length the importance of a study of ventilation.

Whilst fresh air is necessary to provide oxygen, it also functions as a dilutant to bacteria, including P.P.L.O, virusess, etc., as well as to noxious gases such as CO_2, CO, H_2S, ammonia, methane, etc. (Selyanski 1966).

The ability to maintain desired environmental conditions in livestock buildings is dependent on the design and performance of the ventilation systems. These systems must provide the correct quantities of air flow in the proper distribution patterns to meet the needs of each application. Selection of a correct ventilation system requires an understanding of the principles involving air flow and physiological responses of livestock (Hellickson Walker 1983).

Ventilation systems for livestock structures are of two types, natural or mechanical.

Natural ventilation is the movement of air through specied building openings by the use of the natural forces produced by wind and temperature difference. The ventilation rate depends on the wind speed and direction, design and location of outlet and inlet openings.

Natural ventilation is the oldest form of ventilation and has been used for as long as housing has been provided for animals. Simplicity, low initial cost and low energy cost are primary factors that make it the most common type of ventilation. However, ventilation that is dependent on natural forces is inherently variable and consequently has numerous limitations (Hellickson Walker 1983).

Why is natural ventilation not used more widely ? A major reason is that a natural ventilation system lacks control of the airflow through the building. Yet, with the advent of microprocessors for continuous monitoring and control , and sensors that are accurate yet inexpensive, control of the ventilation rate in a natural ventilation system may no longer be a problem (B.L.Brockett and L.D.Albright 1984).

Currently, automatically controlled natural ventilation systems are being used in animal housing. One example is a setup developed by the Scottish Farm Buildings Investigation

Unit. This system is for a swine building with eave openings, a dampered chimney, and adjustable sidewall vents are opened or closed to achieve a desirable air temperature inside the building (Bruce, 1979).

The basic idea behind the control program is to compare the average inside temperature with the pre-determined optimum temperature. If the temperature is within the acceptable range, the microprocessor continues monitoring the sensors. If the inside temperature dose not fall within the acceptable rang, the program beings by reading the sensors and calculating a new ventilation rate based on the steady- state energy balance. The revised ventilation rate is compared to the minimum acceptable air flow for controlling moisture and contamination levels in the building. The largest of the flows is taken as the new ventilation rate.

The program adjusts all positive (airflow into the building) vents as a unit and all negative (airflow out of the building) vents as a unit to achieve the required ventilation rate. Each separate vent adjustment is then modified to achieve approximately equal airflows per meter length through the negative vents and the positive vents. This based on the premise that equally distributed flows will promote complete air mixing in the building.

When the building reaches an acceptable average inside temperature, the control system continues to monitor the inside temperature. When the inside temperature is no longer within the specified range, the whole process begins again by reading the wind pressure difference, temperature, and solar sensorrs.

The computer program is written in Pascal and was implemented on an IBM PC-XT using a turbo-pascal compiler. The flow chart (figure 1) illustrates program logic (B. L. Brockett and L. D. Albright, 1984).

Mechanical ventilation systems are of three types, based on the means of achieving air movement:(a) Exhaust, (b) Pressure and (c) Neutral. Exhaust systems use fans that draw air out of the structure, thereby reducing inside pressure and creating air flow through ventilation inlets. Pressure systems mechanically force air into the building and discharge air through outlets. Neutral pressure ventilation systems employ dual-acting fans that simultaneously draw air out of and force air into the building, creating a pressure difference in the ventilation system but not in the livestock building (Hellickson Walker, 1983)

Since fans are used more and more their construction, installation and running are very important factors to be appreciated. The subject was discussed in detail by Prosser (1966).

Microprocessors allow increased sophistication in environmental control of agricultural structures. This enhanced level of control is possible since microprocessors or microcomputer control systems allow logic and decision making to be added to the functions provided by the control system. A microprocessor based, ventilation control system, typically consists of the microprocessor or microcomputer, an analog to digital conversion unit, control circuitry, environmental sensors and some means of displaying or recording information, e.g. a digital display, a printer or magnetic tape. Normally, input data are received from the environmental sensors, e.g. dry bulb temperature, dew point temperature, outside temperature, solar radiation, static pressure, etc., converted to digital signals and are forward to the microprocessor where these values are compared with reference values programed into the system. The microprocessor system then sends out signals to the control circuit, which activates the necessary environmental control equipment, fans, heater, evaporate cooler, etc., to bring the environmental condition into the desired ranges. Programing changes can be incorporated into the control system to allow wide ranges of environmental control. Microprocessor based control systems offer tremendous potential for improving environmental control in agricultural structures and are clearly the system for the future. Currently they are primarily experimental and are quite expensive, but system costs are expected to come down with increased development and use (Vosper, F.C., L.H.Soderholm.J.F.Andrew and D.s.Bundy. 1982).

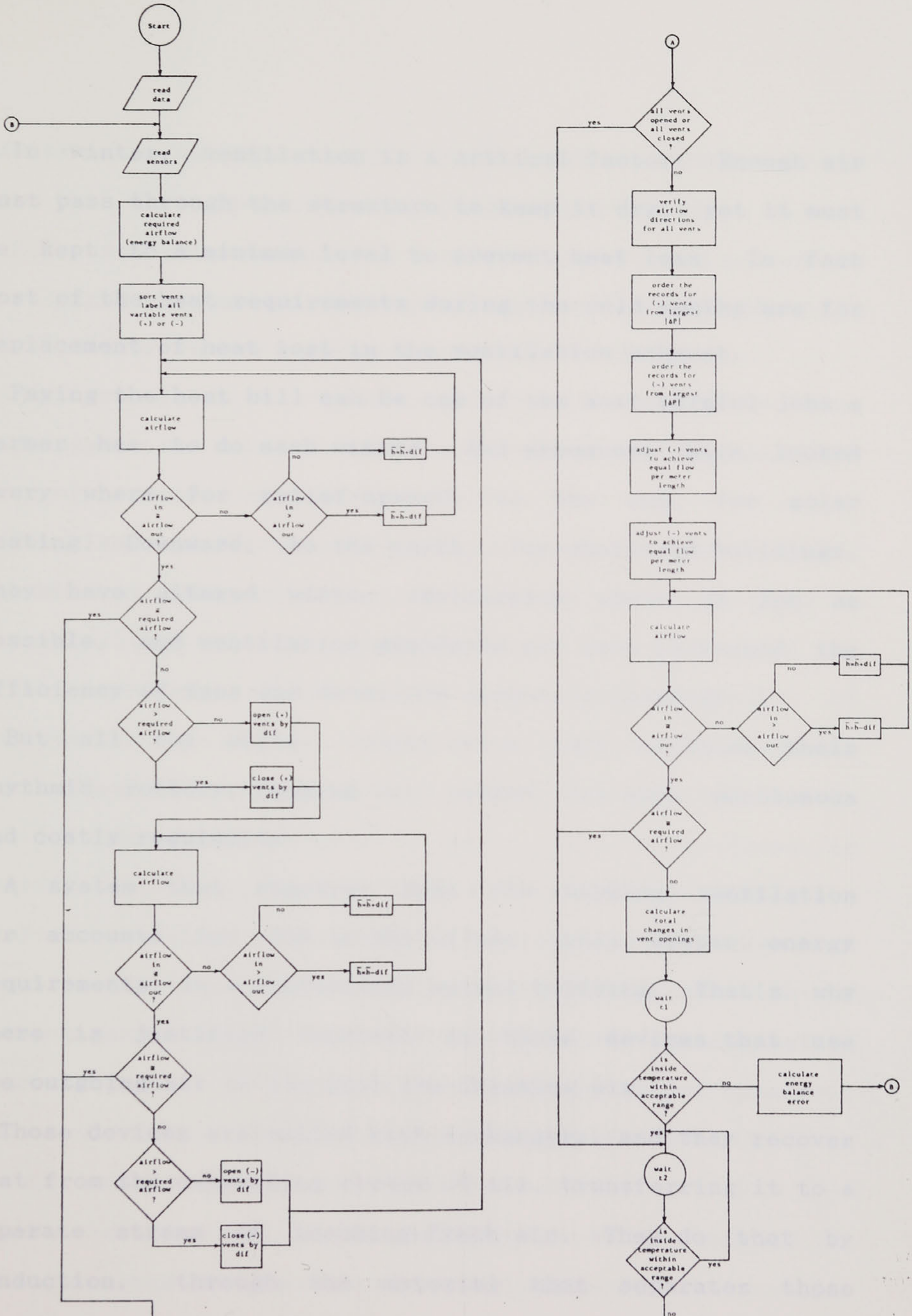


Figure 1. Flow chart.

In winter, ventilation is a critical factor. Enough air must pass through the structure to keep it dry, yet it must be kept to a minimum level to prevent heat loss. In fact most of the heat requirements during the cold months are for replacement of heat lost in the ventilation exhaust.

Paying the heat bill can be one of the most painful jobs a farmer has to do each winter. And producers have looked every where for relief-upward, to the sun, for solar heating. Downward, to the earth, for sheltered buildings. They have altered winter ventilation rates as low as possible, and ventilation manufacturers have increased the efficiency of fans and developed versatile controls.

But all the while, ventilation fans continue their rhythmic robbery, pumping out heated air with monotonous and costly regularity.

A system that requires heat for warming ventilation air accounts for 70% to 90% of the total winter energy requirements in a confinement animal building. That's why there is justified interest in those devices that use the outgoing air to its heat the incoming air.

Those devices are called heat exchangers, and they recover heat from the exhausting stream of air, transferring it to a separate stream of incoming fresh air. They do that by conduction, through the material that separates those airstreams, perhaps aided by fins or plates which extend into both airstreams (Meredith Corporation 1981).

Many new requirements for heat exchangers have arisen in the rapid evolution of technology since World War II, particularly in the aerospace and nuclear fields.

The procedures of thermal and hydraulic design of heat exchangers are based on the laws of thermodynamics and the laws of transport phenomena.

The laws of thermodynamics apply a priori and are well established. The equations of transport phenomena apply a posteriori and are less established.

The importance of these equations in the field of engineering was pointed out by Wilhelm Nusselt as early as 1923. Nusselt was a mechanical engineer, and most of his publications were devoted to the solution of problems in this field. Since 1923 the field of chemical engineering has evolved, raising an awareness of a vast number of new heat transfer problems. Some examples are heat transfer with and without phase change, heat and mass transfer in separation processes, heat transfer in tubular reactors for catalytic reactions, heat and mass transfer in porous media and in drying, and so on.

Heat transfer is also important to those working in nonengineering fields. Examples of some of these areas include geology, meteorology, astrophysics, medicine, and biology. The same statement may be made about mass transfer and momentum transfer.

Heat exchangers include all types of equipment in which the transfer of heat is important or even the rate-controlling process (Schlunder 1983).

The first criterion that a heat exchanger should satisfy is the fulfillment of the process requirements: to accomplish the thermal change on the streames within the allowable pressure drops, and to retain the capability to do this in the presence of fouling until the next scheduled maintenance period. However, it must be recognized that the design process is fraught with uncertainties: The physical properties are seldom known to a high degree of percision, the design methods incorporate basic correlations of various degrees of experimental scatter, the exchangers are constructed only within certain dimensional limits, the actual operating conditions and process stream characteristics vary from day to day, and the fouling characteristics are little more than guesses and vary with time in any case. Therefore, meeting process requirements is at best only a matter of statistical probability. At this point, too little is known about either the statistics of individual units or their interaction in heat exchanger trains and with other process plant components to allow for a quantitative treatment. Hence, the designer must assure himself of a reasonable probability of success by judicious overdesign and by taking advantage of the operational flexibilities and reserve capacity inherent in the rest of the plant.

The second criterion is that the heat exchanger must withstand the service conditions of the plant environment. The immediate consideration here is the mechanical stresses, not only in normal operation but in shipping, installation, startup, shutdown, and off-specification operation caused by plant upsets and conceivable accidents. There are external mechanical stresses imposed by the piping on the exchanger by both steady state and transient flow and temperature variations of the streams. The exchanger must resist corrosion by the service and process streams and by the environment; this is mostly a matter of proper materials selection, but mechanical design does have some effect.

Third, the exchanger must be maintainable, which usually means choosing a configuration that permits cleaning as required and replacement of tubes, gaskets, and any other components that are especially vulnerable to corrosion, erosion, vibration, or aging. This requirement may also place limitations on positioning the exchanger and providing clear space around it.

Fourth, and directly consequent upon the second and third criteria, the designer should consider the advantages of a multishell arrangement with flexible piping and valving provided to allow one unit to be taken out of service for maintenance without severely upsetting the rest of the plant.

Fifth, the exchanger should cost as little as possible, consistent with the above criteria being satisfied.

Finally, there may be limitations on exchanger diameter, length, weight, and/or tube specifications due to site requirements, lifting and servicing capabilities, and maintaining an inventory of replacement tubes and gaskets (Kenneth J. Bell 1983).

It is of utmost importance that the effects and selection of constructional elements in shell-and-tube exchanger must be taken into account such as: Shell type, Tube bundle type, Tube diameter, Tube length, Tube layout pattern and pitch, Baffle type, spacing and cut.

Shell type; The basic shell type is TEMA E, with entry and outlet nozzles at the opposite ends in single shell pass.

Tube bundle type; The tube bundle type determines mainly- as far as the thermohydraulic calculation method is concerned-the possible bundle-to-shell bypass. In mechanical design considerations, it is selected mainly on account of compatibility with thermal expansion considerations such as fixed tubesheet, various types of floating head, and U-tube bundle, just to name the most important types. Not all combinations of baffle type, shell type, and tubepasses are possible.

Tube diameter; Thermohydraulic considerations favor small tube diameter. Also, greater surface density within a given shell is possible with small diameter tubes. Tube cleaning practices limit tube diameters to a minimum of approximately 20 mm OD. For reboilers and condensers, other tube diameter selection considerations will apply.

Tube length; In general, the longer the tube, the lower the cost of the exchanger for a given surface. This is due to the resulting smaller shell diameter, thinner tubesheet and flanges, fewer pieces to handle, and fewer holes to drill. The limitations are accommodating shell-side flow areas with reasonable baffle spacing, and practical design considerations. Usual length-to-shell diameter ratios range from about 5 to 10 for best performance.

Tube layout pattern and pitch; A good practice for tube layout calls for minimum pitch of 1.25 times tube diameter and/or a minimum webb thickness between tubes of approximately 3.2 mm to assure sufficient strength for tube rolling. Generally, the smallest pitch in a triangular 30°

layout is preferred for turbulent and laminar flow in clean service and 90° or 45° layout with 6.4 mm clearance for cases where machanical cleaning is required.

Baffle type, spacing, and cut; The function of the cross baffle is to direct the flow across the tubefield as well as to mechanically support the tubes against sagging and possible vibration. The most common type is the segmental baffle, with a baffle cut resulting in a baffle window (Taborek 1983).

The Enercon Heat Exchanger is made in Canada and it has an exciting track record.

Enercon is easy to install, light weight and compact. It has simple electrical installation-you plug it into a 110 volt grounded outlet. It is self-cleaning. The fans are easy to clean or replace, and the plastic is corrosion resistant. A manual variable speed fan control provides up to 350 cfm and allows flexibility in air volume to meet minimum ventilation requirements. The dual fan system provides the maximum efficiency from the exchanger.

And perhaps most importantly the unit can be easily adapted to utilize the heat exchanger in buildings with pit ventilation (Meredith Corporation 1981).

Del-Air heat exchangers were designed by producers for producers. The developement and testing the exchanger has taken place in both the lab and the field under extreme climatic conditions for a number of years. The experience gained by Del-Air Systems Ltd. has made the unit one of the best in the field. Only Del-Air exchanger have an automatic rapid defrost, a definite requirement in any cold weather heat recovery system. By reversing the fresh air intake fan, defrosting is achieved rapidly and positively. This allows the Del-Air exchanger to operate at a constant, high efficiency.

Del-Air exchangers unique intake nozzle design provides positive air distribution throughout the room, eliminating the need for ductwork. The novel, in-wall design allows installation in rooms and passage-way without occupying limited head room. Moisture from the exchanger is exhausted outside the building, eliminating the need for in-barn drains or over-the-pit mounting. All units are pre-wired to be plugged into a standard 110 volt outlet (EDEN ENERGY EQUIPMENT LTD. 1982).

IV. MATERIALS AND METHODS

A. To make the insulation of the ceiling heavier

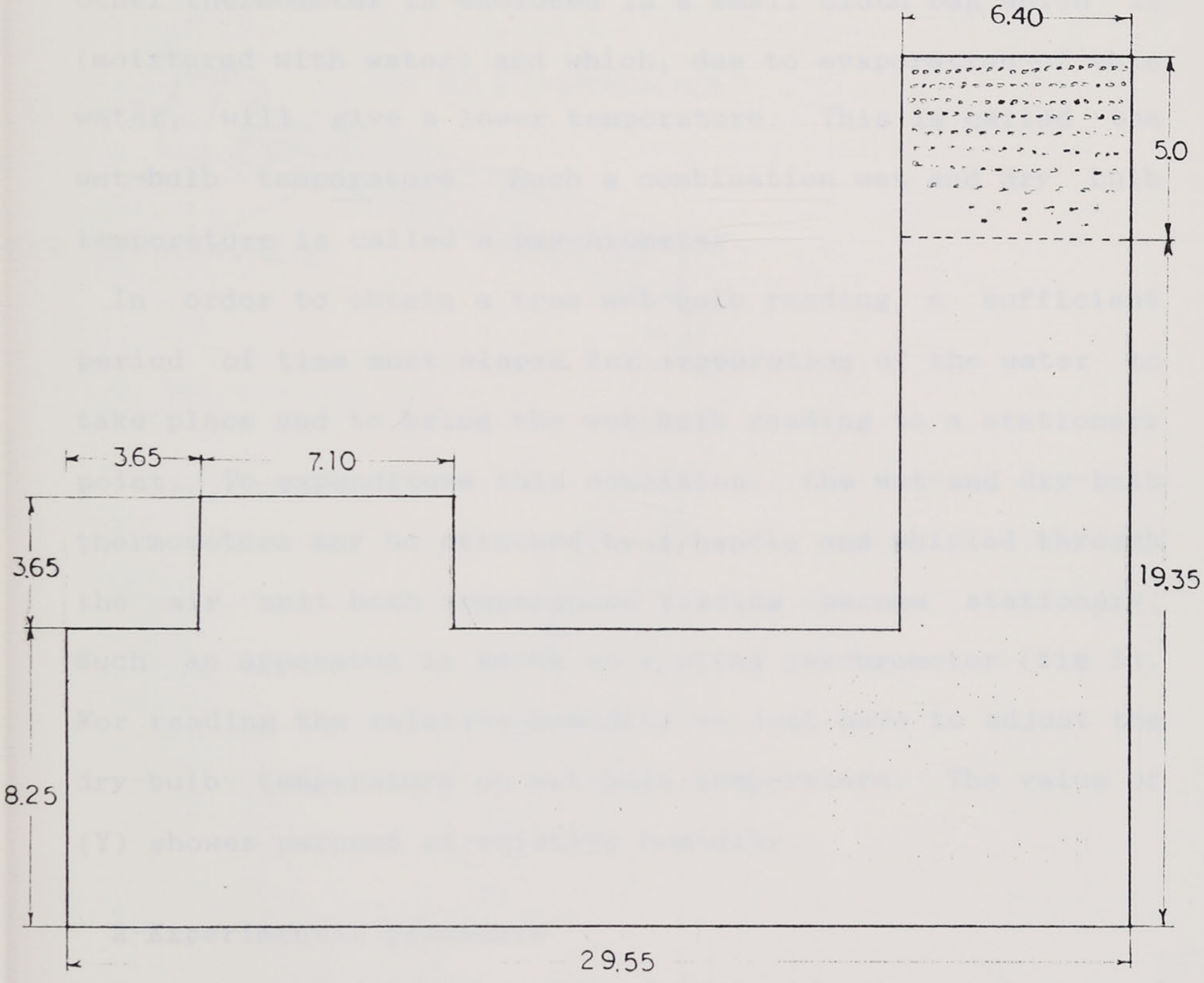
By looking at the barn, it has been thought that the water dripping from ceiling, might be from poor insulation of the ceiling. When the cold air comes into contact with the ceiling, its surface reaches to dew-point temperature, condensation then occurs. In fact, dripping, near the exit door was more than any other places (Figure 2). Therefore, wood chips (which were available on the roof figure 2) were spread around the cold side of the ceiling, near exit door (30 cm) to see if there would be any changes in vapor condensation. The following week we saw that there was no significant difference of water dripping from the ceiling. Then the decision were made to measure the relative humidity inside the barn.

B. To find the rate of ventilation required

1-Principle of operation of a sling psycrometer.

Relative humidity is the ratio of the actual water vapor pressure in the air to the saturated water vapor pressure at a given air temperature and is usually measured by two ordinary thermometers, secured to a common base. The bulb of one of the thermometers is exposed and the temperature reading taken with this thermometer is the same as that taken with any other ordinary mercury thermometer. This is

GURE 2. LAYOUT OF THE BARN



WOOD CHIPS

WATER DROP

SCALE: $\frac{1}{200}$

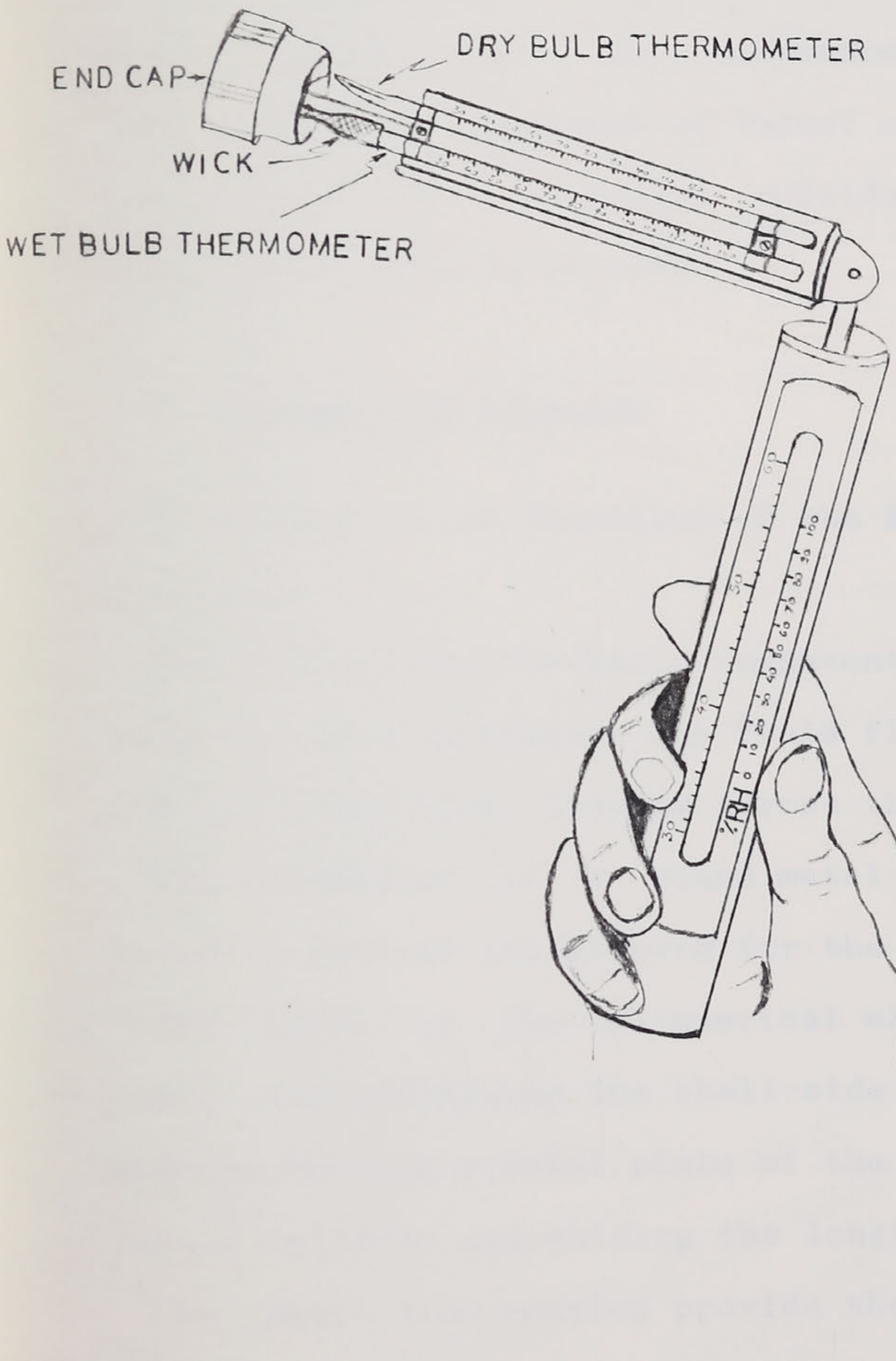
called the dry-bulb temperature to distinguish it from the reading taken with the other thermometer. The bulb of the other thermometer is enclosed in a small cloth bag which is (moistured with water) and which, due to evaporation of this water, will give a lower temperature. This is called the wet-bulb temperature. Such a combination wet and dry bulb temperature is called a psychrometer.

In order to obtain a true wet-bulb reading a sufficient period of time must elapse for evaporation of the water to take place and to bring the wet-bulb reading to a stationary point. To expenditure this condition, the wet-and dry-bulb thermometers may be attached to a handle and whirled through the air unit both temperature reading become stationary. Such an apparatus is known as a sling psychrometer (fig 3). For reading the relative humidity we just have to adjust the dry-bulb temperature on wet-bulb temperature. The value of (Y) showes percent of relative humidity.

2-Experimental procedure

Two sets of data were taken from inside temperature and relative humidity by use of sling psychrometer. One reading was taken on Fourth of February (outside temp.$-13^{\circ}C$) and the other one Fourth of March 1985 (outside temp.0 $^{\circ}C$). There was no need to take any data for summer and fall season, because natural ventilation were being used, even in 9th of

FIG. 3. A SLING PSYCHROMETER



November (outside temp. 3°C).

It was measured the walls segments, windows sizes, heights, doors, the wall thickness and its insulation, the number of average Animal Unit housed and their weights. By using all these information, the exposure factor, sensible and latent heat produced by animal and lastly the rate of ventilation for different outside and uniform inside temperature were calculated.

C. Recommended Solution

1-Principles of Operation of the ENERCON'S FRESH AIR Heat Exchanger.

The tubes are the basic component, providing the heat transfer surface between one fluid flowing inside the tubes and the other fluid flowing across the tubes (fig. 4a2).

The tubesheet is a round metal plate that has been suitably drilled and grooved for the tubes.

The shell is the cylinderical element surrounding the tubes and containing the shell-side fluid. It is commonly made by rolling a metal plate of the appropriate dimensions into a cylinder and welding the longitudinal joint.

The shell side nozzles provide the inlet and outlet ports for the shell-side fluid.

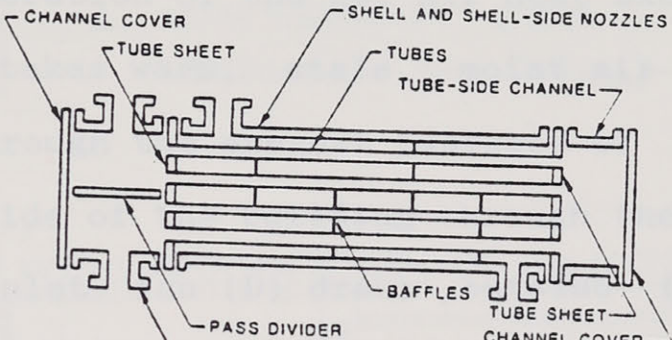
Tube-side channels and nozzles guide the tube-side fluid into and out of the tubes.

The last major component for is the baffle array. The baffles serve two purposees: First, and most important, the baffles support the tubes against bending and vibration; second, they guide the flow back and forth across the tube field to improve heat transfer rate.

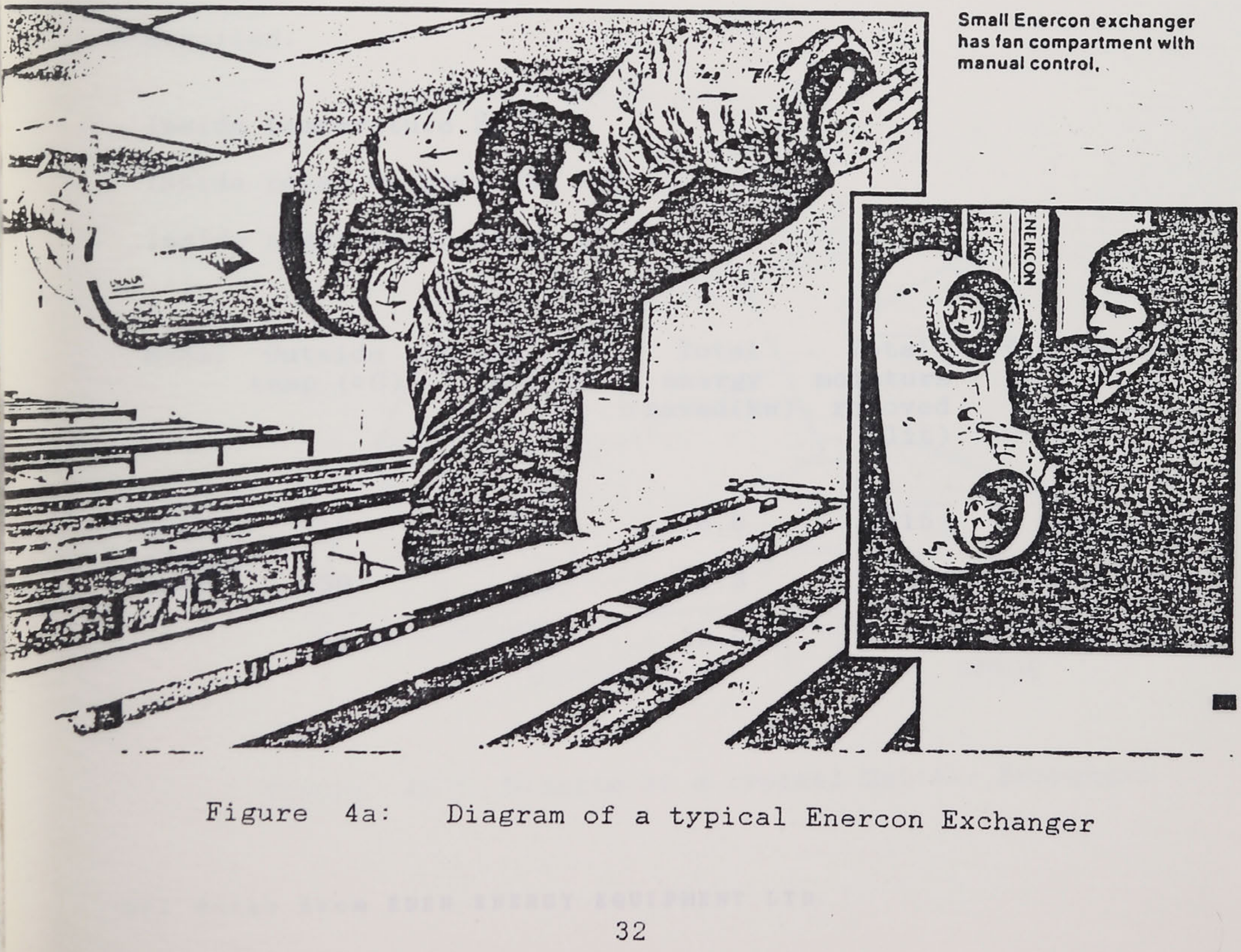
The Enercon unit is a molded plastic drum 27 in. in diameter, 66 in. long, with molded plastic transfer plates inside. The self-contained fan unit also is in a molded plastic housing containing two 8-in. axial fans with the control incorporated in the housing (fig. 4a).

The exchanger has two 8-in. ports at each end for connection to air ducts. The fan assembly attaches to the intake end of the unit and pushes the warm room air through the unit and out the 8-in. tube to the outside of the building. The intake fan pulls air through the unit from the 8-in. intake tube. After this incoming air passes through the exchanger, it is expelled into the room.

The Canadian tests showed Enercon's unit recovers between 59 and 79% of escaping heat.



2 Diagram of a typical (fixed-tubesheet) shell-and-tube heat exchanger.



Small Enercon exchanger has fan compartment with manual control.

Figure 4a: Diagram of a typical Enercon Exchanger

2-Principles of operation of the Del-Air Heat Exchanger

The top fan (A) takes warm, stale, moist air from the room, forces it through the exhaust passages of the core (B), and then outside of the building through the exhaust outlet (C). The inlet fan (D) draws outside fresh air through the fresh air inlet (E), through the alternate (inlet) of the core and into the inlet nozzle (F) (fig. 4b).

Tests carried out under controlled conditions of temperature and relative humidity produced the results shown below. During the tests the following variables were controlled:

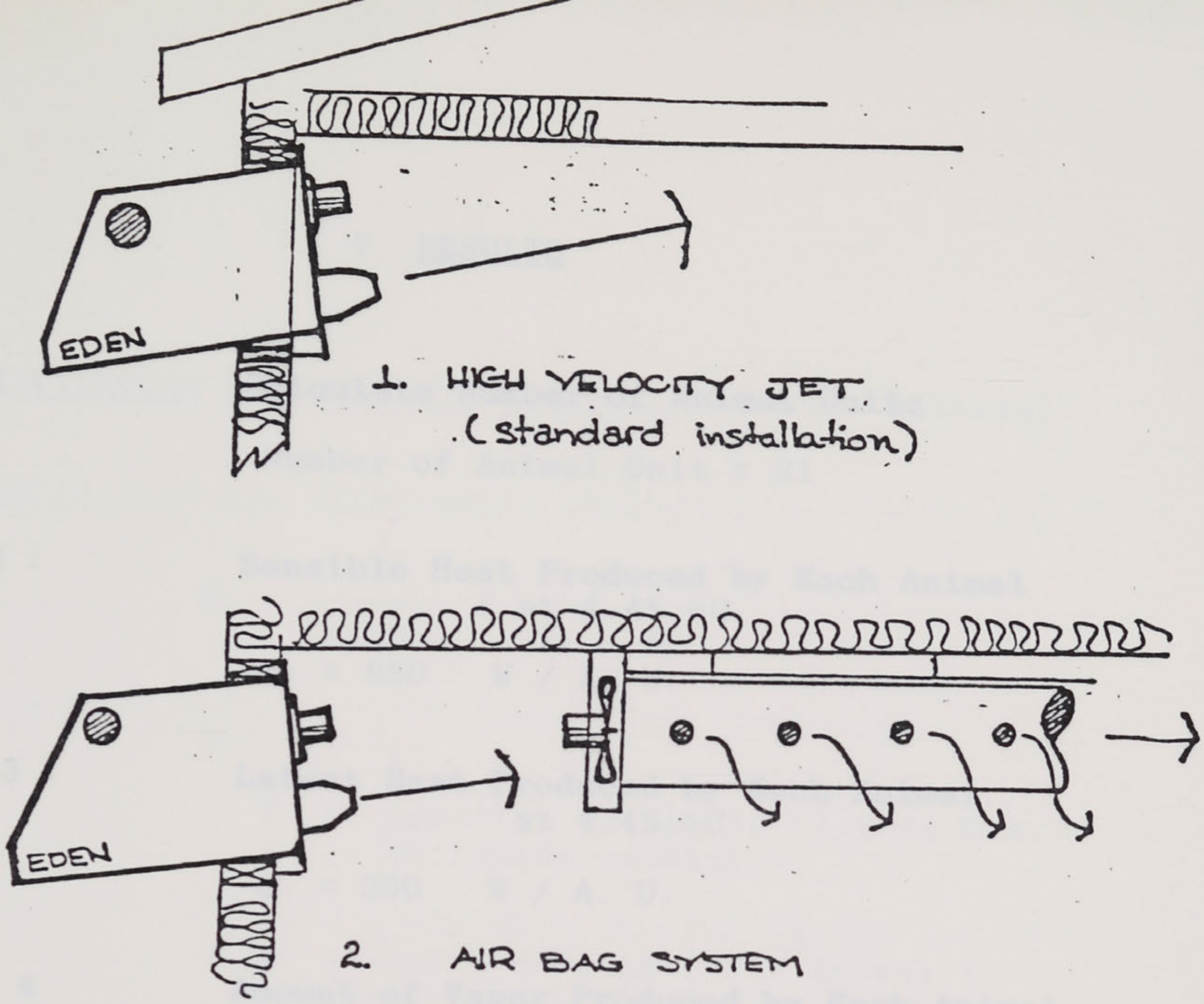
Inside temperature 20°C

Inside relative humidity 70%

Inside negative pressure 25 Pa

Model	Outside temp.(°C)	Fresh air in (°C)	Total energy saved(kw)	Total moisture removed (lit)	Efficiency
A-800 Series 1000	-18	6.4	6.8	415	64%
	-30	4.5	10.3	442	69%
	-40	2.0	12.6	450	70%

-all datas from EDEN ENERGY EQUIPMENT LTD.



TYPICAL INSTALLATIONS

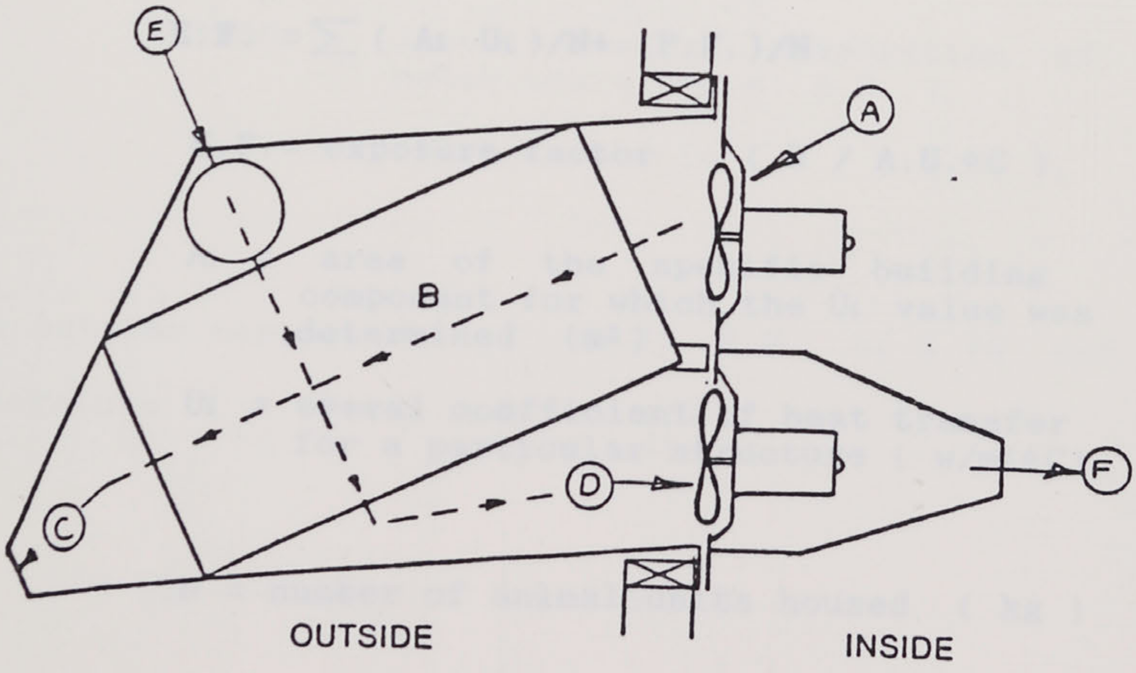


Figure 4b : Diagram of a typical Dei-Air Exchanger

V. RESULTS

Step 1 : **Calculate Number of Animal Units**

Number of Animal Unit = 21

Step 2 : **Sensible Heat Produced by Each Animal** at 4.45 °C

q_s = 850 W / A. U.

Step 3 : **Latent Heat Produced by Each Animal** at 4.45 °C

q_l = 200 W / A. U.

Step : 4 **Amount of Vapor Produced by Each Animal** at 4.45 °C

W_p = .1250 gr / S-A.U.

Step : 5 **Finding The Exposure Factor**

$$E.F. = \sum (A_i U_i)/N + (P.F.)/N$$

E.F.= exposure factor (W / A.U.°C)

A_i = area of the specific building component for which the U_i value was determined (m^2)

U_i = overal coefficient of heat transfer for a particular structure ($w/m^2 °C$)

N = number of animal units housed (kg)

<u>E.F. = 13.04</u> W / A.U.°C

VENTILATION FOR HUMIDITY CONTROL (Moisture Balance)

$$Q_l = \frac{V}{2460} \times \frac{q_l}{W_i - W_o}$$

Where :

V = specific volume in (l / kg D.A.) at inside condition.

q_l = latent heat in (W / Animal).

W_i = water vapor in (gr H_2O / kg D.A.) at inside condition.

W_o = water vapor in (gr H_2O / kg D.A.) at outside condition.

2460 = latent heat of evaporation of water in (l / s- A.U.).

Q_l = ventilation rate in (l / S-A.D.).

For the outside temperature of -13°C, R.H. of % 75 and inside temperature of 4.45 °C, R.H. of % 70.

Q_l = 21.41 l / S-A.U.

Total Q_l = 449.61 l / S-A.U.

VENTILATION FOR TEMPERATURE CONTROL (Heat Balance)

$$Q_s = \frac{V}{1005\ (t_i - t_o)} \times \left[q_s - E.F.\ (t_i - t_o) \right]$$

Where :

E.F. = exposure factor (W / A.U.°C)

V = specific volume in (l / kg D.A.) at inside temperature.

q_s = sensible heat (W / Animal)

1005 = specific heat of air (W.S /kg°C)

t_i = inside temperature °C

t_o = outside temperature°C

For the outside temperature of -13°C, % 75 of R.H. and inside temperature of 4.45°C, % 70 of R.H..

Q_s = 26.70 L / S-A.U.

Total Q_s = 560.70 L / S-A.U.

SUPPIEMENTAL HEAT REQUIRED

$$Q_{sp} = \left[\left(\frac{1005\ W_p}{W_i - W_o} \right) + E.F. \right] \triangle T - q_s$$

Where :

Q_{sp} = supplimental heat (W / A.U.)

1005 = specific heat of air (W.S / kg°C)

W_p = water vapor production (grH_2O/S-A.U.)

E.F. = exposure factor (W / A.U.°C)

T = temperature difference °C

W_i = inside water vapor (gr H_2O / kg air)

W_o = outside water vapor(gr H_2O / kg air)

q_s = sensible heat (W / Animal)

For the outside temperature of -13°C, % 75 of R.H. and inside temperature of 4.45°C, % 70 of R.H..

Q_{sp} = 138.44 W / A.U.

Total Q_{sp} = 2.91 K W / A.US.

Table 1 : <u>VENTILATION RATE FOR HUMIDITY CONTROL</u>

T (°C)	W_o (gr H_2O/kg D.A.)	Q_1 (L/S D.A.)	Q_t (L/Sec)
0	2.5	42.81	899.01
-5	1.6	26.76	561.96
-10	1.2	22.93	481.53
-15	0.85	20.39	428.19
-20	0.45	18.09	379.89
-25	0.25	17.13	359.73
-30	0.15	16.68	350.28

W_i = 4 gr H_2O/kg D.A.

T_i = 4.45 °C

V = 790 L/kg D.A.

Table 2 : <u>VENTILATION RATE FOR TEMPERATURE CONTROL</u>

T (°C)	Q_s (L/S D.A.)	Q_t (L/Sec)
0	123.38	2,590.98
-5	56.56	1,187.76
-10	34.29	720.09
-15	23.16	486.36
-20	16.48	346.08
-25	12.02	252.42
-30	8.84	185.64

V = 790 L/kg D.A.

T_i = 4.45 °C

E.F. = 13.04 W/A.U. °C

Table 3 : <u>AMOUNT OF SUPPLEMENTAL HEAT REQUIRED</u>

T ($^\circ C$)	W_o (grH_2O/kg D.A.)	Q_{sp} (W/A.U.)	Q_t (KW)
-10	1.2	18.56	0.39
-15	0.85	208.38	4.38
-20	0.45	360.65	7.57
-25	0.25	546.16	11.47
-30	0.15	748.40	15.72

W_i = 4 (grH_2O/kg D.A.)

W_p =0.1250 (grH_2O/S A.U.)

E.F. = 13.04 (W/A.U.)

T_i = 4.45 $^\circ C$

Figure 5. Required Ventilation Rate for Humidity & Temperature Control

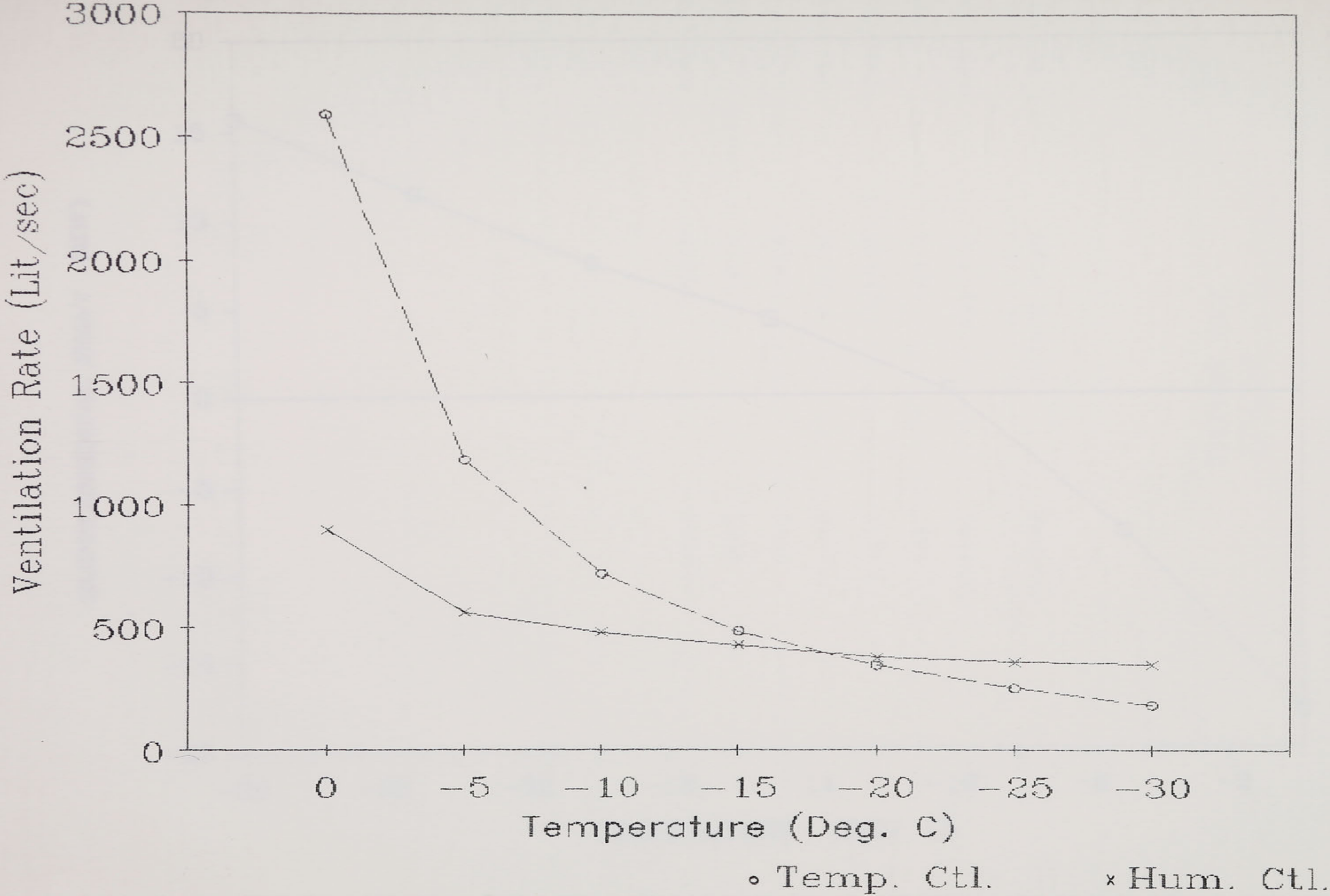
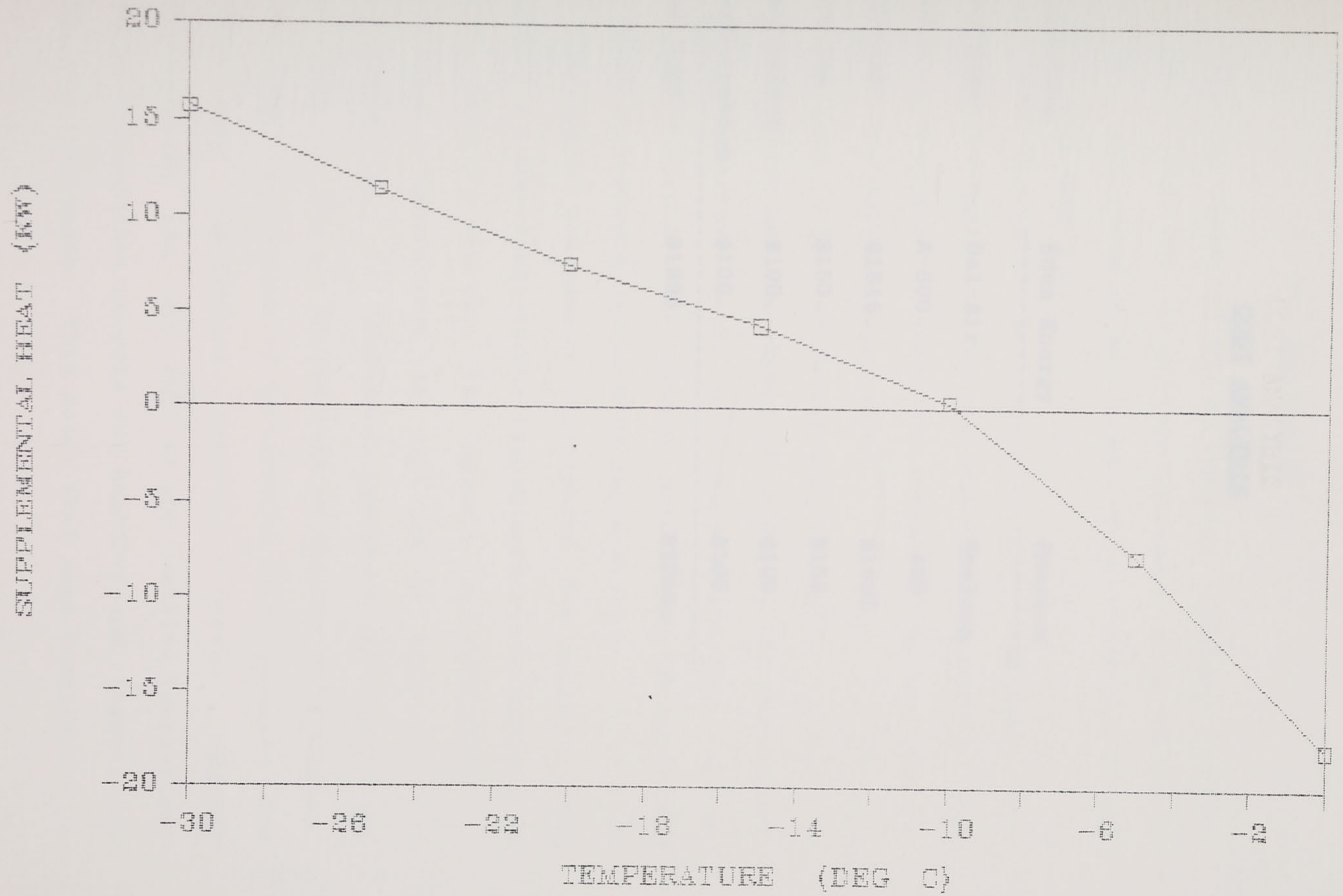


Figure 6 SUPPLEMENTAL HEAT REQUIRED



COST ANALYSIS

Manufacturs	Eden Energy	Enercon
Brand name	Del-Air	Enercon
Model	A-800	400
Unit cost	$1645.	$1695.
Sales tax	$150.	$155.
Installation	$100.	$100.
Administration	$100.	$100.
Total Cost	$1995.	$2050.

VI. DISCUSSION

By using the data and the information obtained from an employee (who was a diploma graduate from Macdonald College), and assuming an Animal Unit of 500 kg, I found the following result .

The rate of sensible heat given by animal was high. This is reasonable because of increased metabolic activity and greater conversion of feed - to - heat energy are used to counteract low ambient temperature.

The exposure factor was relatively high also, because of poor insulation and the number of windows. As the formula for exposure factor shows, by decreasing resistance value, the value of overall heat transfer coefficient increases and the value of exposure factor then increases.

To calculate the rate of Ventilation for humidity and temperature control,the inside temperature was maintained at 4.45°C. This is because the animals are affected by fluctuating temperatures as well as by high or low temperatures. It is therefore important to keep the temperature of the air in the barn as uniform as possible. By plotting the values of ventilation rate required for humidity and temperature control, versus outside temperature, we can see that as the outside temperature decreases the ventilation rate for humidity and temperature control also decreases. This means that less ventilation is

needed in cold weather than hot weather (fig.5). That is why most barns similar to this barn are hot and oppressive in summer and spring and damp and cold in fall and winter.
If we look at the (table 3), up to -8 °C we do not need any supplemental heat, but from -9°C the supplemental heat is required. By plotting the amount of supplemental heat required versus outside temperature(fig.6), unlike the other graph as the outside temperature decreases the supplementary heat increases, which is as expected.

The efficiencies calculated for the heat exchangers by the companies shows that these units are quite good to be used in this case. Especially two Enercons units has been installed in new pigary building and they have been working very well for past 3 years. The only problem was that every almost 6 months they have to open the shell and clean it which is reasonable (because of so much dust and odors).

VII. CONCLUSION

Based on the results, we can conclude that animals give off moisture during normal respiration, and the higher the temperature, the greater the moisture content. This moisture should be removed from buildings through the ventilation.

Lack of heat makes moisture removal more difficult and then, cause condensation on ceiling. As the data shows the relative humidity in the barn was high % 80 whereas the accepted relative humidity is 50 - 75 percent (Ministere de l'Agriculture) and the temperature was 4.45 °C whereas the accepted temperature is 7 - 24 °C . Therefore as soon as ceiling reaches to dew-point temperature condensation occures.

It also can be concluded that as the temperature decrease, less ventilation is needed for moisture removal.

Based on the datas from the companies and their; durable, corrosion-proof construction, easy cleaning, simple installation and costs either of this units could be used for ventilation purposes.

VIII. RECOMMENDATION

The installation of a ventilation system, Enercon's Fresh Air Heat Exchanger Model 400 or the Del-Air Heat Exchanger Model A-800 (series 1000) is recommended as well as an improvement in the insulation of the barn.

When outside temperature was -13 °C, inside temperature was 4.45 °C. Of course the inside temperature goes lower than that by decreasing the outside temperature more.

The ceiling vapor barriers on the warm side of ceiling are strongly recommended in this situation.

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X. APPENDIX

CALCULATIONS

Step 1 : **To Calculate Number of Animal Units**

1 A.U. = 500 kg

Average rate of occupancy = 21 horses

Weight of each horse = 500 kg

$$\textbf{Number of Animal Units} = \frac{21 \times 500 \text{ kg}}{500 \text{ kg}} = 21$$

Step 2 : **To Calculate Sensible Heat At 4.45 °C (q_s)**

q_s for horse at 4.45 °C is 1.7 W / kg

21 X 500 kg X 1.7 W/kg = 17,850.00 W

q_s per A.U. = 17,850 / 21 = 850.00 W / A.U.

q_s = 850.00 W / A.U.

Step 3 : **To Calculate Latent Heat At 4.45 °C (q_l)**

q_l loss of horse at 4.45 °C is .4 W/kg

21 X 500 kg X .4 W/kg = 4,200.00 W

q_l = 4,200 / 21 =200.00 W / A.U.

Step 4 : **To calculate Vapor Produced At 4.45 °C (W_p)**

W_p For horse at 4.45 is .9 gr H_2O/kg.h

21 X 500 X .9 = 9450.00 g H_2O / h

or 2.625 g H_2O / sec.

W_p = 2.625 / 21 = .125 g H_2O / Sec. A.U.

CALCULATION OF THE EXPOSURE FACTOR

$$E.F. = \frac{\sum A_i U_i + P.F.}{N}$$

	Area (m^2)	R ($m^2 \, {}^oC/w$)	U ($w/m^2 \, {}^oC$)	AU ($w/{}^oC$)
Wall	252.40	3.96	.2525	63.73
Window	27	.4	2.5	67.5
Doors	8.41	1.0	1.0	8.41
Ceiling	372.74	5.74	.174	64.93

The total A.U. =195.48 $w/{}^oC$

The value of R/found from table

Exposed Perimeter 100.5 m X .78 $w/m\text{-}{}^oC$ (F from table)

E.P. = 78.39 $w/{}^oC$

$$\textbf{E.F.} = \frac{195.48+78.39}{21} = \textbf{13.04} \quad w/{}^oC \text{ A.U.}$$

VENTILATION FOR HUMIDITY CONTROL

$$Q_1 = \frac{V}{2460} \times \frac{q_1}{W_i - W_o}$$

W_i for 4.45 °C = 0.0040 (kg H_2O /kg D.A.)
from psychrometric chart

W_o for -13 °C = 0.0010 (kg H_2O /kg D.A.)
from psychrometric chart

V = for 4.45 °C = 785 (L /kg D.A.)
from psychrometric chart

$$Q_1 = \frac{785}{2460} \times \frac{200}{(4.0-1.0)} = 21.27 \text{ L /s-A.U.}$$

Total Q_1 = 446.67 L / Sec

VENTILATION FOR TEMPERATURE CONTROL

$$Q_s = \frac{785}{1005\ (t_i - t_o)} \times \left[q_s - E.F.\ (t_i - t_o) \right]$$

V = 785 L/kg

t_i = 4.45 °C

t_o = -13 °C

q_s = 850 w/A.U.

E.F. = 13.04 w/°C A.U.

$$Q_s = \frac{785}{1005\ (5+13)} \times \left[850 - 13.04\ (5+13) \right]$$

Q_s = 26.70 L/Sec A.U.

Total Q_s = 560.70 L/Sec

SUPPLEMENTAL HEAT REQUIRED

$$Q_{sp} = \left[\frac{1005\ W_p}{W_i - W_o} + E.F. \right] \triangle T - q_s$$

W_p = 0.1250 gr H^2O/S A.U. from table

q_s = 850 W/A.U.

T = 18 °C

$$Q_{sp} = \left[\frac{1005 \text{ X } .1250}{(4.0-1.0)} + 13.04 \right] (18) - 850$$

Q_{sp} = **138.47** W/A.U.

Total Q_{sp} = 2.91 KW

DATA

Fourth of February 1985

Inside the Barn

	Wet Balb (oF)	Dry Balb (oF)
	39	42
	38	40
	37	40
	36	38
Average :	37.5	40

On the Roof

	Wet Balb oF	Dry Balb oF
	27	30
	27	30
	27	30
	27	29
Average :	27	30

Outside Temperature	= -13	oC
Outside R. H.	= 72	Percent
Inside R. H.	= 82	Percent

Fourth of March 1985

Inside the Barn

	Wet Balb °F	Dry Balb °
	41	44
	41	44
	41	45
Average :	41	44

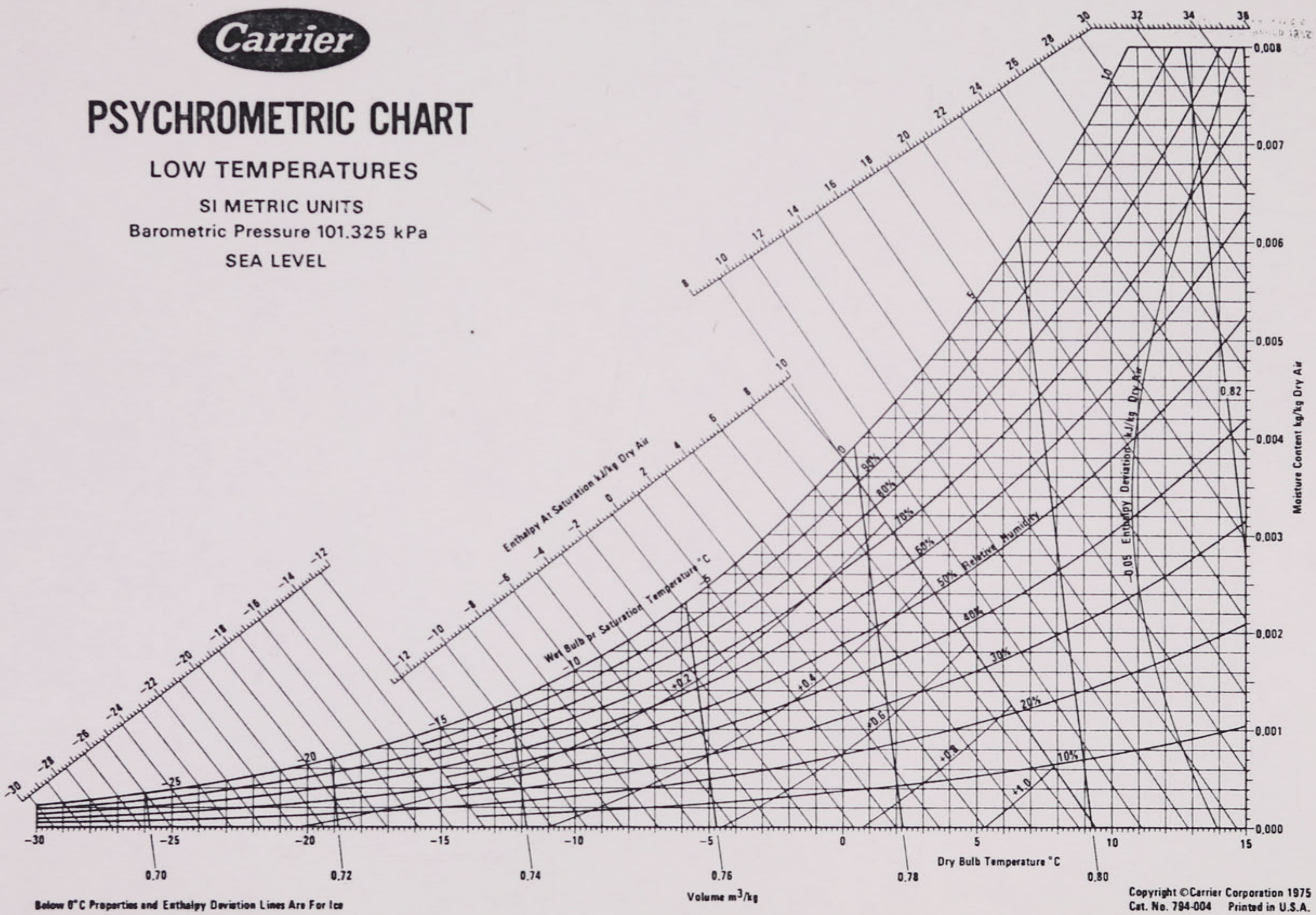
On the Roof

	Wet Balb °F	Dry Balb °F
	45	48
	46	49
Average :	45.5	47

Outside Temperature	= 0	°C
Outside R. H.	= 75	Percent
Inside R. H.	= 80	Percent
R. H. on the roof	= 90	Percent

Average Number of Horses Housed	= 21
Average Weight of Each Horse	= 500 kg
Average Height of the Barn	= 2.65 m
Number of Windows	= 18
Size of Each Window	= 1.2 X 1.2 m^2
Number of the Doors	= 2
Size of the Doors	= 2.05 X 2.05 m^2
Wall Thickness	= 30 cm

Wall Components
- steel
- wood
- fiber glass
- breez block
- cement



Low temperature psychrometric chart.