2ᵈ edition

Commission Internationale du Génie Rural

2nd Report of
WORKING GROUP
on

CLIMATIZATION OF ANIMAL HOUSES

1992

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INTRODUCTION

STATUS OF THE GROUP

The working group was initiated by the past chairman of the CIGR Section IV (Farm Buildings), Professor Rolf HENDRIKKSON, Sweden, who appointed M. RIST, Switzerland to be chairman of the group. The first meeting was held in Zurich, Switzerland in February 1977. K. SÄLLVIK, Sweden, has been the secretary of the working group.

In 1985 the working group elected K. SÄLLVIK as its chairman and J. CHRISTIAENS as its secretary. For three days once a year meetings were held in Horsens (Denmark), Gumpenstein (Austria), Gent (Belgium) and Sjusøen (Norway).

Each country represented in the group is a member of the CIGR. However the representative for a country has no official authority to approve or disapprove the proposed CIGR Standard. In each country the responsible boards or authorities are free to adopt or adapt these CIGR Standards.

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FOREWORD

Encouraged by the success of the first report of the "Working group of climatization and energy use in animal houses" of the "Commission Internationale du Génie Rural" (CIGR) 1984, we aimed for a second report in order to cover the more practical aspects of climatization. It was finished in 1989.

This second report focuses on the design of climatization systems in animal houses. Since no completeness can be achieved in this matter, the working group confined itself to what has been considered as the basic information needs of advisors in public service and in private companies.

This also implied that no major designing aspect could be left out, so the working group agreed on eight chapters. For each of them one of the members was appointed responsible. Although the content of the chapters and the overall content of the report was discussed within the group at several occasions, the personal opinions of the responsible member were respected.

Obviously the work of the group has an output which goes far beyond the report itself. The international character of the group and the sincerity among the members turned each meeting into a mind expanding experience towards a better understanding of climatization of animal houses.

The working group expresses the hope that this report will contribute to the welfare of men and animals, which is the ultimate aim of every designer.

ir. J.P.A. CHRISTIAENS
Secretary

Prof. K. SÄLLVIK
Chairman

... ABOUT THE SECOND EDITION

This second edition is not a copy of the first one. The authors have taken the opportunity to revise the text and a chapter on mechanical ventilation systems was added. So I want to thank again all the authors for their kind cooperation.

ACKNOWLEDGMENT

As secretary and editor of this report I want to express my profound gratitude to :

- The Institute for the Encouragement of the Scientific Research in Industry and Agriculture (IWONL - IRSIA) , De Crayerstraat 6, 1050 BRUSSELS, who granted my request to accept this responsible task, and who provided the financial means.

- Prof. Dr. ir. M. DEBRUYCKERE, Head of the department of farm buildings of the "Faculty of Agricultural Sciences" of the "State University" of Ghent. Without his moral, scientific and practical support, this work could not be carried to a successful conclusion.

- And last but not least to Mr. Martin SOMMER of the "Centre for Rural Building", Craibstone, Bucksburn, Aberdeen AB2 9TR, Great Britain. He checked the English text for errors and grammar, which was very much appreciated by all the authors.

ir. J.P.A. CHRISTIAENS,
Secretary, editor

Responsible member

VAN OUWERKERK E.N.J.

Chapter 1

MODELLING THE HEAT BALANCE OF PIGS AT ANIMAL AND HOUSING LEVELS

INTRODUCTION

When designing a pig house it is essential to know the heat balance, so that the correct compartment size, floor type, thermal insulation and equipment for climate control can be chosen. Two computer models have been developed to model the heat balance.

The first model, BEZOVA (STERRENBURG and VAN OUWERKERK, 1986), describes the heat balance of pigs at animal level. It is based on reseach done by BRUCE and CLARK (1979). Influences of animal parameters and food intake on the heat production side as well as influences of the building, housing system and ventilation on the heat loss side, define the thermoneutral and comfort zones of the pig. The model has been validated against 635 respiration experiments found in the literature and has an average mean square prediction error of 7.2 per cent.

The second model, STALKL, describes the relationship of heat production and heat losses at housing level.

1.1. Heat production

Total heat production is constant within the thermoneutral zone (area between lower critical temperature LCT and upper critical temperature UCT-tnz in figure 1).

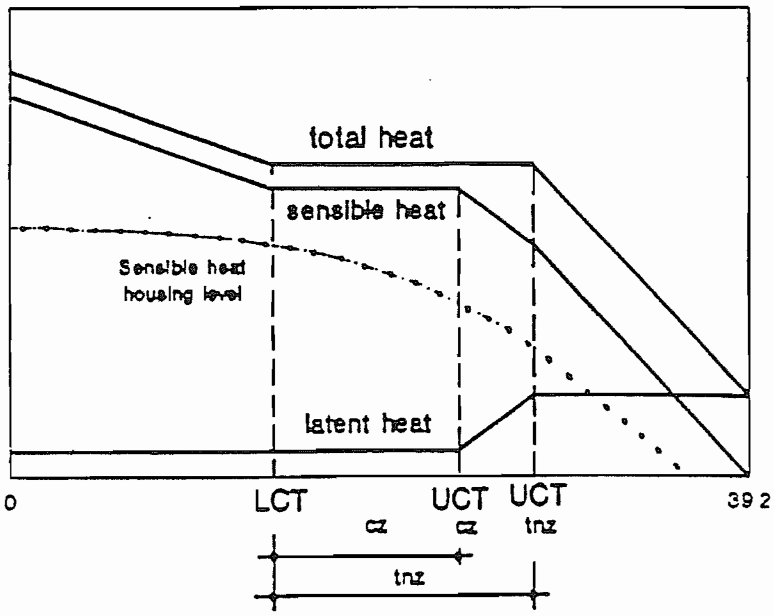

Fig. 1 : Definition of heat production of pigs, used in the BEZOVA and STALKL models

In the BEZOVA model four different equations for metabolizable energy used for maintenance (ME_m) have been tested :

$ME_m = 641\ m^{0.66}$ (FOWLER, 1980),
$ME_m = 440\ m^{0.75}$ (BRUCE and CLARK, 1979),
$ME_m = 719\ m^{0.63}$ (CLOSE and FOWLER, 1983) and
$ME_m = 458\ m^{0.75}$ (CLOSE and FOWLER, 1983).

The total heat production (Q) of a pig is :

$$Q = ME_m + (1 - k) . (ME - ME_m) \quad (kJ/d)$$

where
ME_m = metabolizable energy used for maintenance (kJ/d) ;
m = animal weight (kg)
k = food efficiency for production 0.7
ME = metabolizable energy intake (kJ/d)

For pregnant sows this equation must be corrected for the fœtus.

The best fit for ME_m was :

for piglets ($m < 20$ kg) : $ME_m = 440\ m^{0.75}$
for fattening pigs ($m = 20 - 120$ kg) : $ME_m = 458\ m^{0.75}$
for sows ($m > 120$ kg) : $ME_m = 420\ m^{0.75}$

Figure 2 shows the total result of validating the BEZOVA model against the results of 310 respiration tests carried out by VERSTEGEN (1971) and others. All measured values of heat production calculated with the computed model only deviate from measured heat production by 1.4 % and the mean square prediction error is 6.6 %.

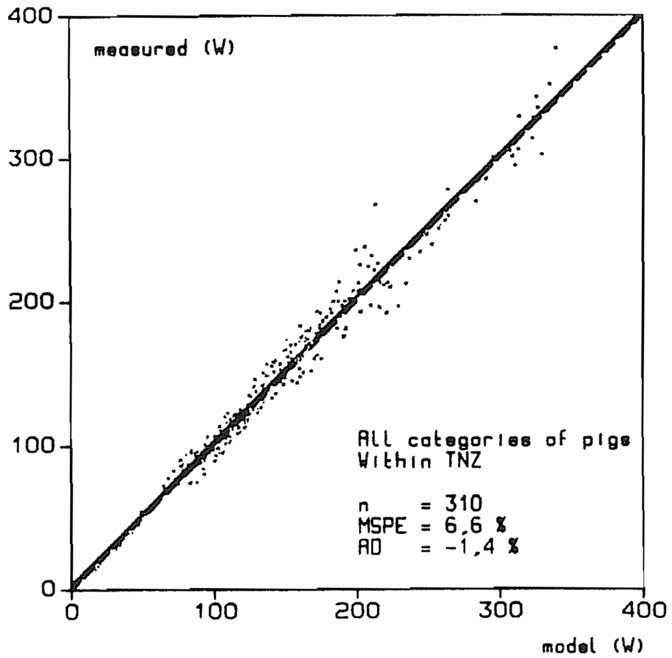

Fig. 2 : Result of validation of the BEZOVA model against a group of respiration experiments where ambient temperature was within the thermoneutral zone

1.2. Heat losses at animal level

Outside the thermal neutral zone heat production must also be equal to heat losses. Heat is lost through different pathways (radiation, convection, conduction and evaporation), all of which involve the animal's body surface area (A). Three different equations for body surface (A) have been tested in the model :

$A = 0.0734\ m^{0.656}$ (KELLEY et al., 1973),
$A = 0.0974\ m^{0.633}$ (BRODY, 1945) and
$A = 0.0900\ m^{0.67}$ (BRUCE and CLARK, 1979).

A combination of equations for body surface area best fitted the validation of the model :

$$A = 0.09\ m^{0.66} \quad (m^2)$$

The body surface area (A) is decreased if pigs huddle together when kept in groups (BOON, 1981). This reduction (r) is $r = 12 - 0.05\ m$ (%) regardless of whether the animals are standing or lying. The BEZOVA model calculates two different situations, firstly for a standing animal (heat losses only by convection and radiation) and secondly for the recumbent animal (heat losses by convection, radiation and conduction to floors).

Contact area between two recumbent animals (A_c) has been estimated at :

$$A_c = A\ .\ (0.15 - 0.01\ \ln\ (m)) \quad (m^2)$$

An individual pig can also reduce its surface area by up to 5 % in extremely cold conditions, by curling up.

For lying pigs the model incorporates a reduction factor of 0.2 (for hot conditions) to 0.1 (for cold conditions) for A as the area in contact with the floor.

The model incorporates the average time that the pig stands or lies. The standing time has been estimated as :

$$b = 25 - 0.1\ m \quad (\%)$$

where $b > 15$

When defining critical temperatures one strikes a time dependent mean for recumbent and standing animals.

The skin temperature of the pig is influenced by deep body temperature, tissue resistance and heat transfer resistances for convection and radiation. Extreme values of tissue resistance when vasodilatation or vasoconstriction occurs are known from the literature (BRUCE and CLARK, 1979).

The maximum tissue resistance of the pig is :

$$r_{tmax} = 0.02\ m^{0.33} \quad (W/m^2K)$$

The minimum tissue resistance is about 25 to 35 per cent of the maximum value (MOUNT, BLAXTER).

Convective heat losses to the surrounding air are most influenced by air velocity (v). The convective heat loss (k_c) is (BRUCE and CLARK) :

$$k_c = (15.7\ v^{0.6}) / m^{0.14} \quad (W/m^2K) \text{ for a dry skin and}$$

$$k_c = (91.3\ v^{0.6}) / m^{0.14} \quad (W/m^2K) \text{ for a wet skin.}$$

where v : air velocity (m/s)

The BEZOVA model contains a number of estimated reduction factors for air velocity around the recumbent pig on various floor types. These factors are :

f_{av} = 0.90 on straw bedding
f_{av} = 0.92 on solid concrete floor
f_{av} = 0.95 on slatted concrete floor
f_{av} = 1.00 on wire-mesh

Radiative heat losses are influenced by surface temperatures, radiating areas and emission factors. The radiative heat losses (q_r) are :

$$q_r = 0.9\ C\ (T_1^4 - T_2^4) \quad (W/m^2)$$

where

c = Stefan-Boltzmann constant $5.67 \cdot 10^{-8}$ (W/m^2K^4)
T_1 = absolute skin temperature (K)
T_2 = absolute radiation reference temperature of the environment (K)

Extreme values of evaporative heat losses are known from the literature. When vasocontriction is at maximum (low temperatures) the evaporative heat loss (Q_{emin}) is (BRUCE, 1981) :

$$Q_{emin} = 0.09\ (8 + 0.07\ m)\ .\ m^{0.66} \quad (W)$$

When vasodilatation is at maximum (high temperatures) the evaporative heat loss (Q_{emax}) is :

$$Q_{emax} = 37\ A \quad (W)$$

Conductive heat losses depend on the conductivity of floor structures and on contact area. These are related to a standard floor resistance factor for a recumbent pig of 45 kg liveweight r_{f145} (BRUCE, 1977). The floor resistance (r_{fl}) is :

$$r_{fl} = r_{f145}\ .\ (m/45)^{0.33}\ .\ n^d \quad (m^2K/W)$$

where

n = number of recumbent pigs in the group

d = empirically found value for the effect of horizontal heat loss through floors

The value

d = 0.5 for solid concrete floor

d = 0.4 for slatted concrete floor and

d = 0.2 for wire-mesh floor

Heat losses by warming up drinking water to body temperature involve the following amount of heat :

$$Q_w = 0.048\ V_{dw}\ (T_b - T_{dw}) \quad (W)$$

where

Q_w = heat required for warming up drinking water (W)

V_{dw} = drinking water intake (l/d)

T_b = body temperature (° C)

T_{dw} = drinking water temperature (° C)

Figure 3 shows the total result of validating the BEZOVA model against 325 respiration tests carried out by VERSTEGEN (1971) and others. All measured values of heat production calculated with the computed model deviate from the measured heat production by only 0.2 % and the mean square prediction error is 7.7 %.

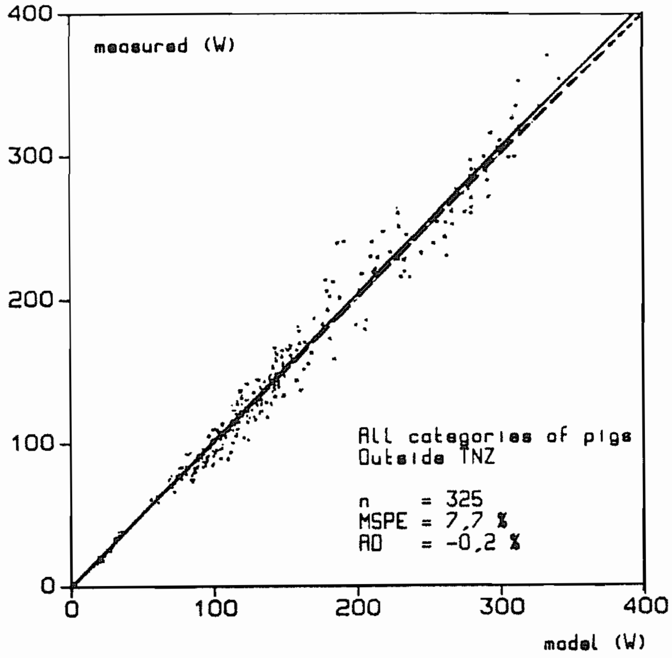

Fig. 3 : Result of validation of the BEZOVA model against a group of respiration experiments where ambient temperature was outside the thermoneutral zone

1.3. Critical temperatures

The computed model calculates critical air temperatures for both dry and wet body surfaces.

The lower critical temperature LCT is found with thermoneutral heat production equal to heat losses and evaporative heat loss at minimum level, vasoconstriction and tissue resistance at maximum level, and huddling for pigs kept in groups.

The upper critical temperature is found with thermoneutral heat production equal to heat losses, when vasodilation is at maximum level, tissue resistance is at minimum level and individual pigs have no contact with other pigs.

The upper critical temperature of the thermoneutral zone UCT-cz is determined for pigs that are breathing normally (evaporative heat loss is at maximum level).

The upper critical temperature of the thermoneutral zone UCT-tnz can be determined for panting pigs (evaporative heat loss is at maximum level).

1.4. Heat losses at housing level

The second model (STALKL) describes the heat balance of an animal house. The heat production of pigs is determined according to the BEZOVA model.

Animal heat is the only source continuously present in the house. It can be divided into sensible heat and latent heat, according to CIGR (the dotted line in figure 1).

The heat losses at housing level are :

1. heat losses by transmission through walls, roof and floors.
2. ventilation.

Ventilation is especially important in regulating the indoor climate.

1.5. Winter problems

The recommended ambient temperature in houses for fattening pigs in winter is the lower critical temperature (figure 4).

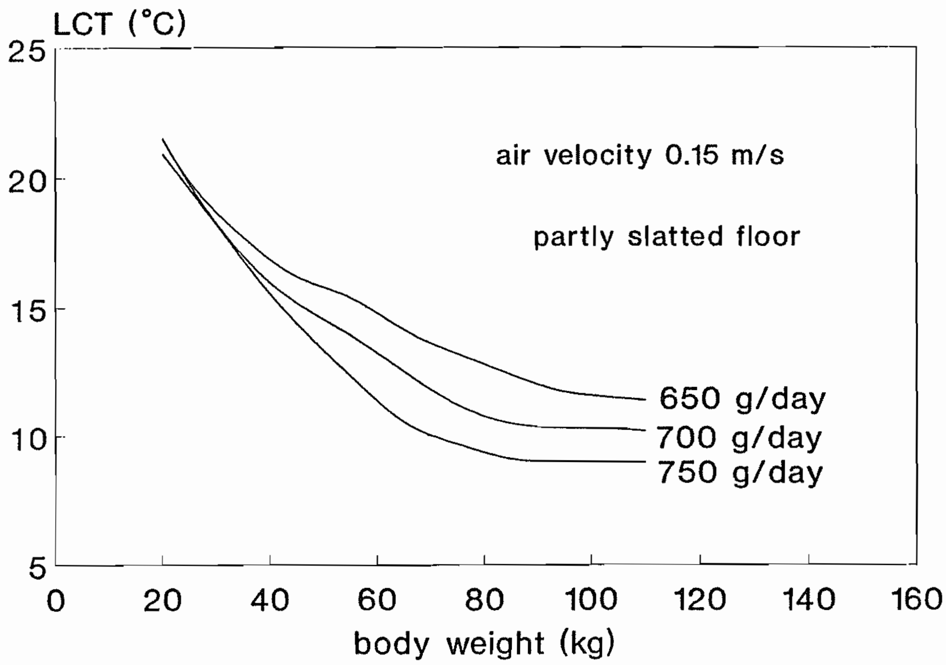

Fig. 4 : Lower critical temperature for pigs for three levels of weight gain

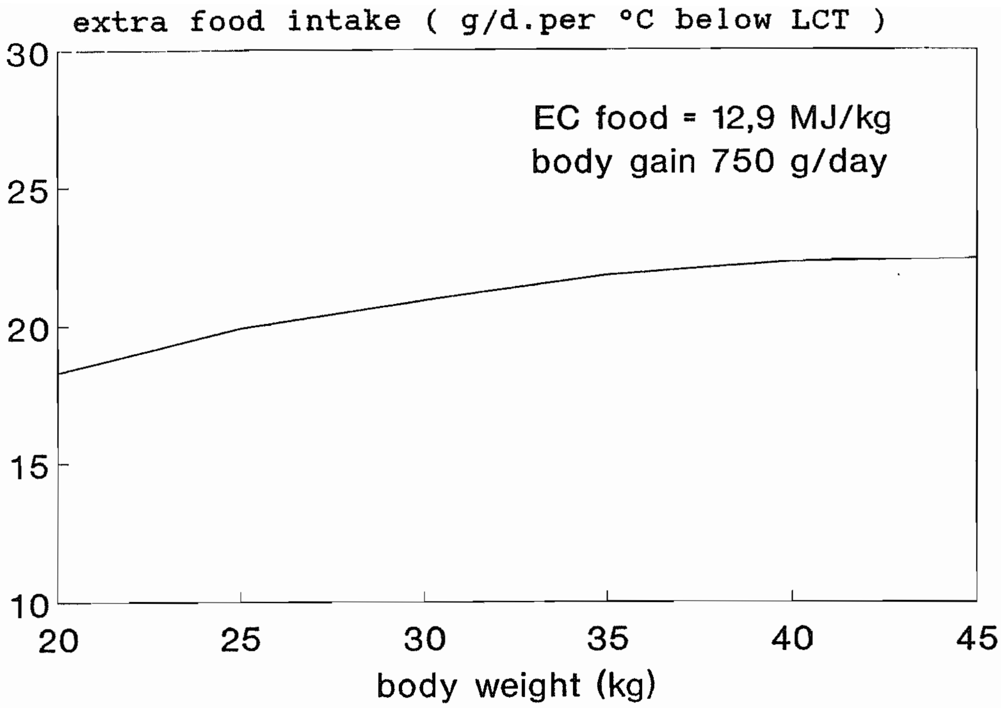

Fig. 5 : Extra food intake when ambient temperature is below LCT

When ambient temperature is below the lower critical temperature the pig needs more food to be in heat balance (figure 5).

At housing level minimum ventilation is required to maintain an acceptable level of CO_2 concentration. In the Netherlands the level of 0.2 % of CO_2 in houses is accepted. Figure 6 shows the minimum winter ventilation for fattening pigs on three levels of daily gain. The current Dutch standard for minimum ventilation results in CO_2 concentrations exceeding the acceptable level.

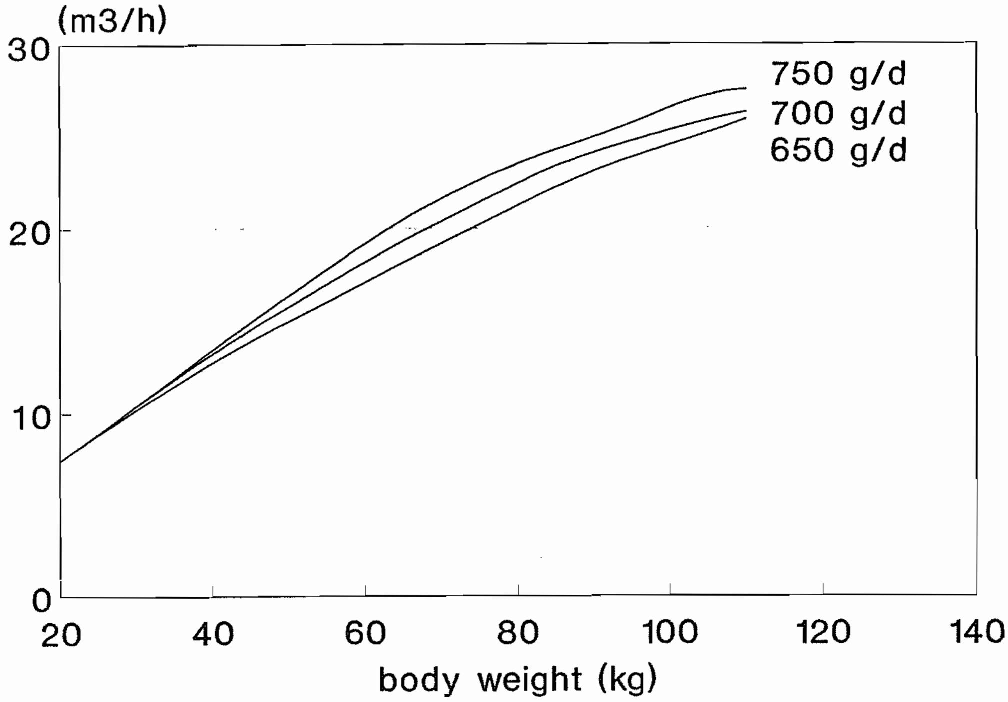

Fig. 6 : Minimum ventilation per fattening pig for three levels of daily gain and an accepted CO_2-concentration of 0.2 vol%.

In winter the ambient temperature sometimes falls below LCT. A number of human interventions are possible to influence heat losses. If the pigs are given better insulated bedding, e.g. straw, LCT falls by 4 to 5 °C (figure 7). If a pig of 30 kg liveweight is on a heated floor the LCT drops about 3 °C when the floor is warmed up 15 degrees (figure 8).

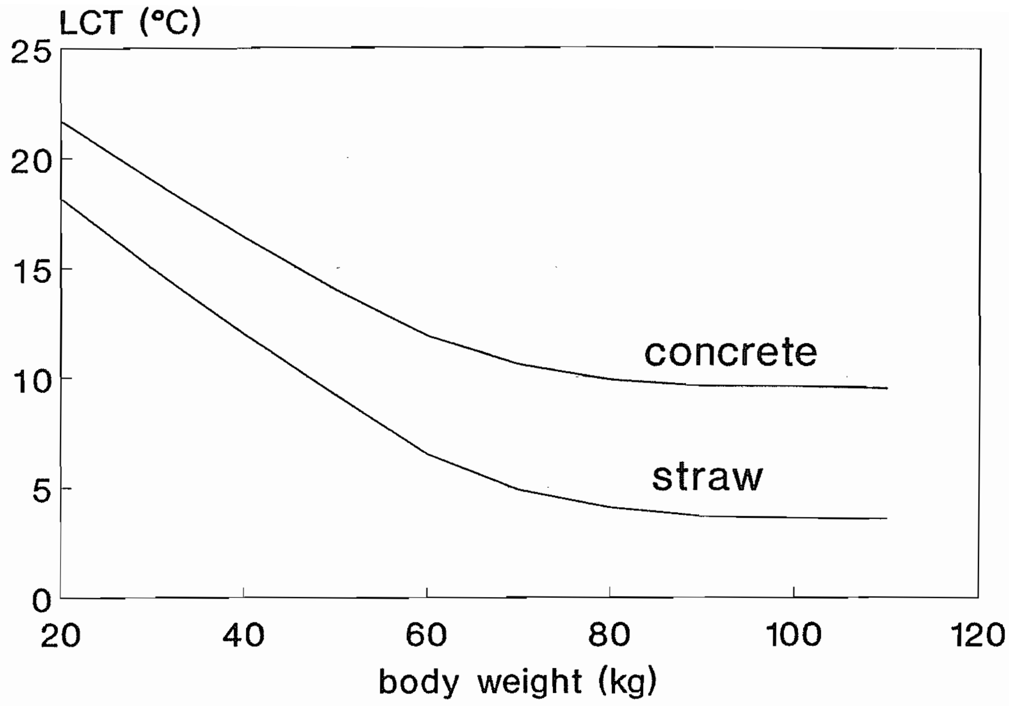

Fig. 7 : Lower critical temperatures on different floors. Average gain 750 g/day

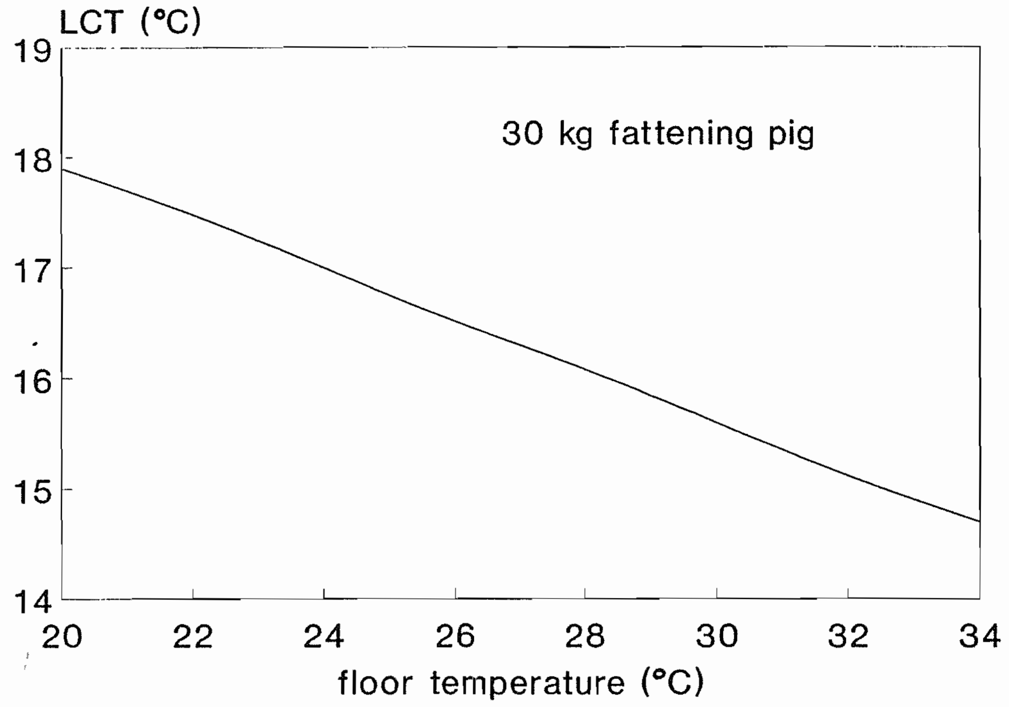

Fig. 8 : Lower critical temperatures as a function of floor temperature effected by heating

Figure 9 shows the effects on LCT of minimizing the air velocity. Warming up the drinking water can also help to decrease LCT (figure 10).

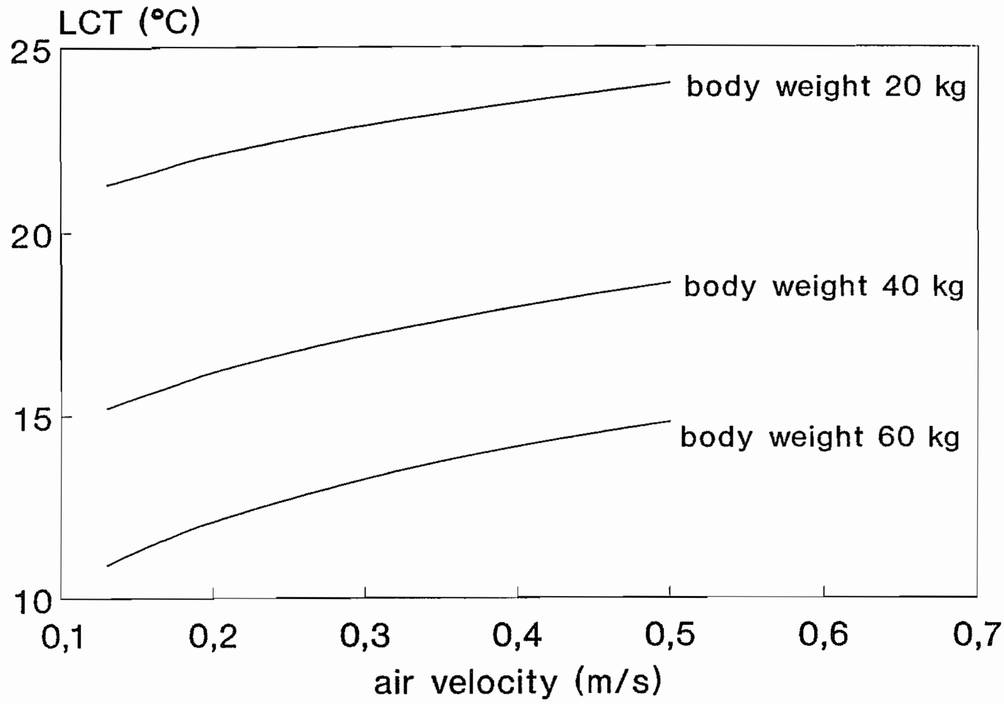

Fig. 9 : Lower critical temperatures of pigs as a function of air velocity

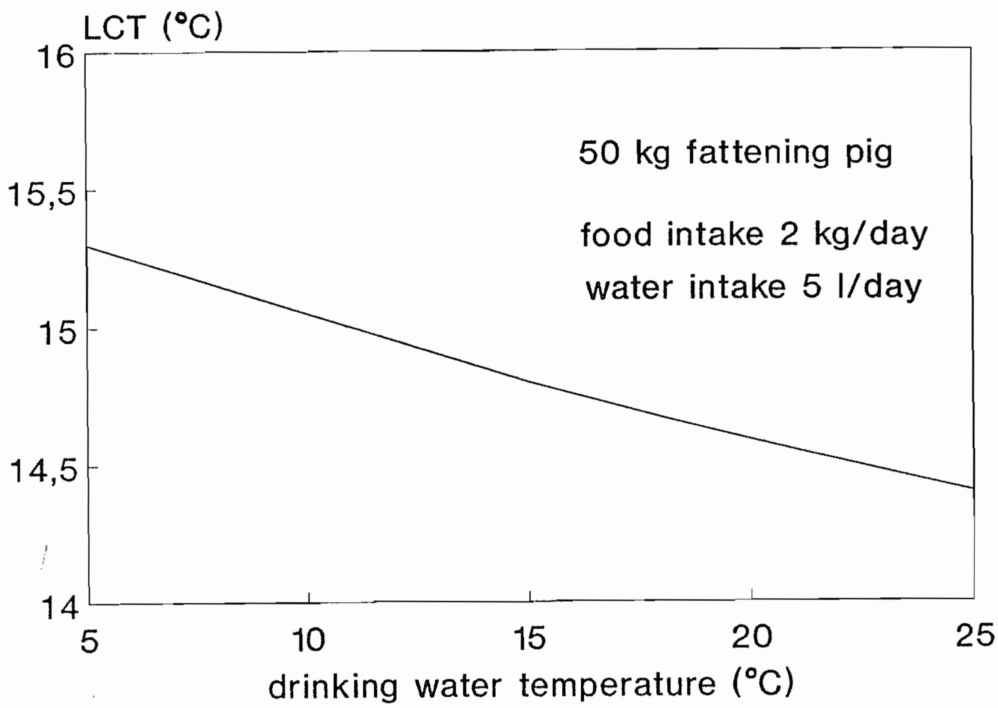

Fig. 10 : Lower critical temperature as a function of drinking water temperature

1.6. Summer problems

In summer, ventilation is required to prevent the ambient temperature of the pigs rising above the UCT of the thermoneutral zone. Summer ventilation rate in pig housing should rise to the values shown in Figure 11. An outside air temperature of 24.1 degrees Celsius is not exceeded more than 1 % of the year in the Netherlands.

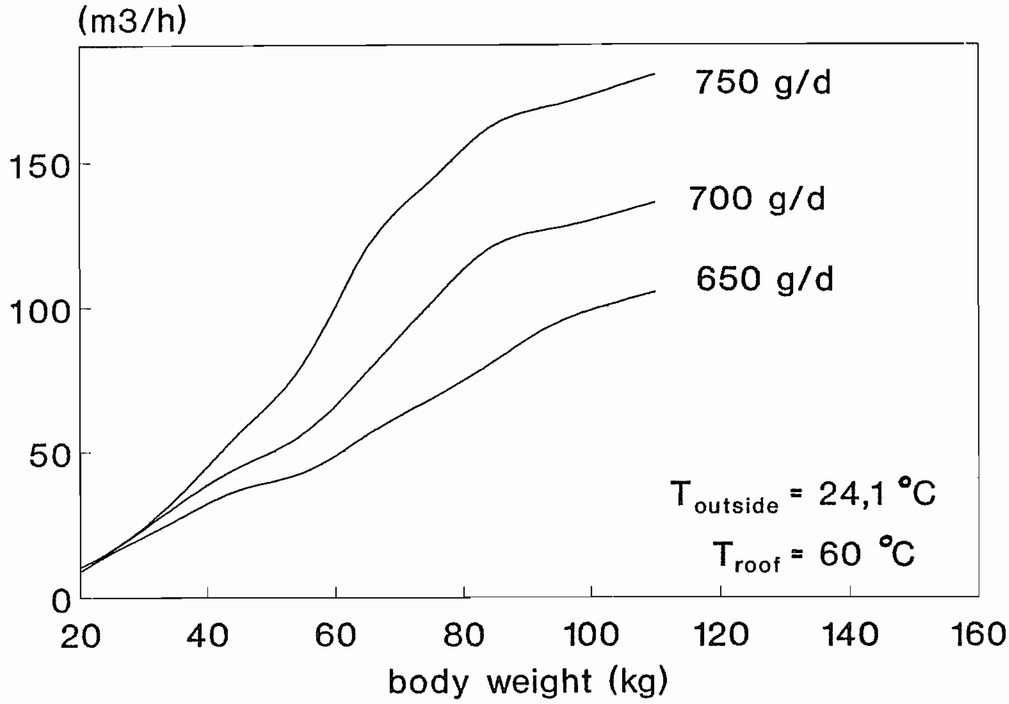

Fig. 11 : Summer ventilation requirement for fattening pigs

In summer the ambient temperature sometimes rises above the UCT of the thermoneutral zone. Figures 12 to 15 show the effects of factors under human control which can help prevent heat stress.

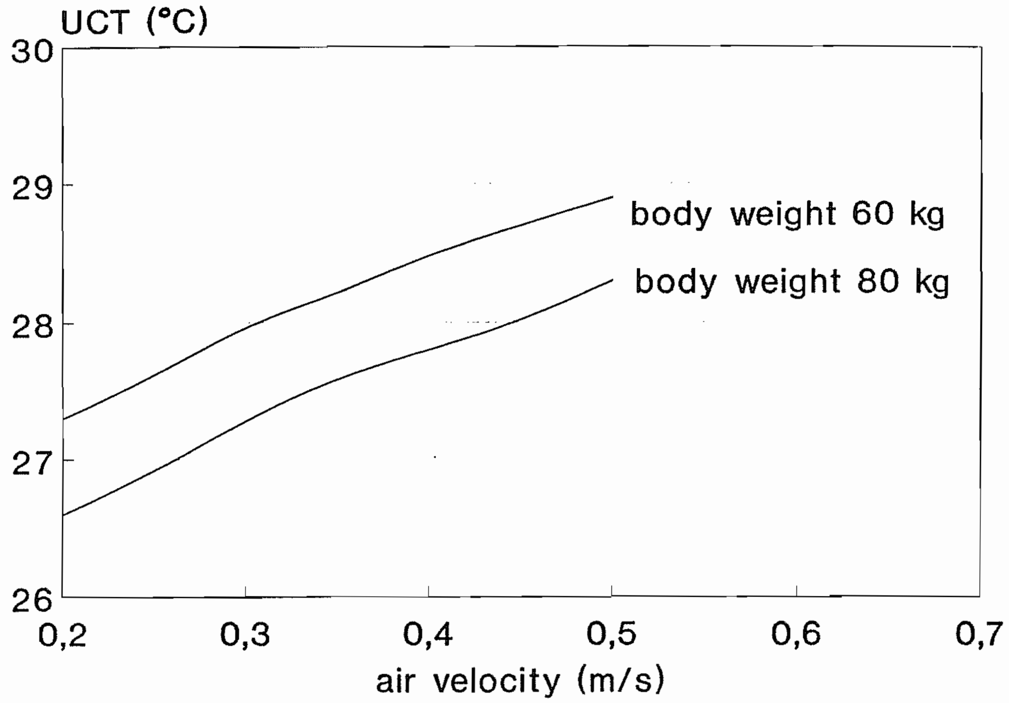

Fig. 12 : Upper critical temperature of pigs as a function of air velocity.

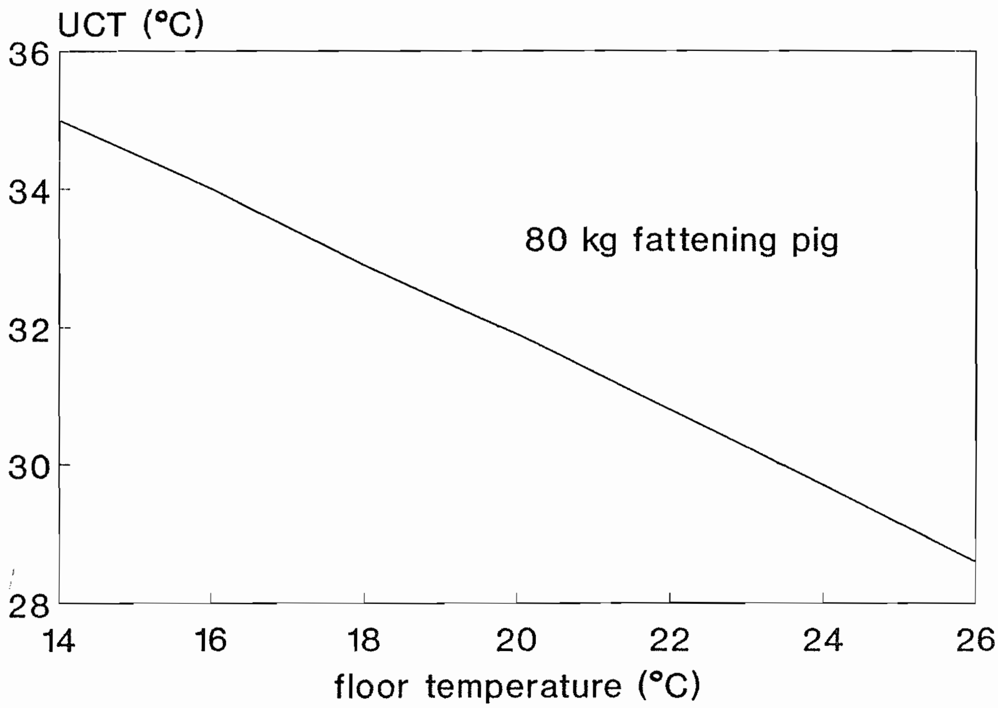

Fig. 13 : Upper critical temperature for a pig of 80 kg weight as a function of floor temperature effected by cooling

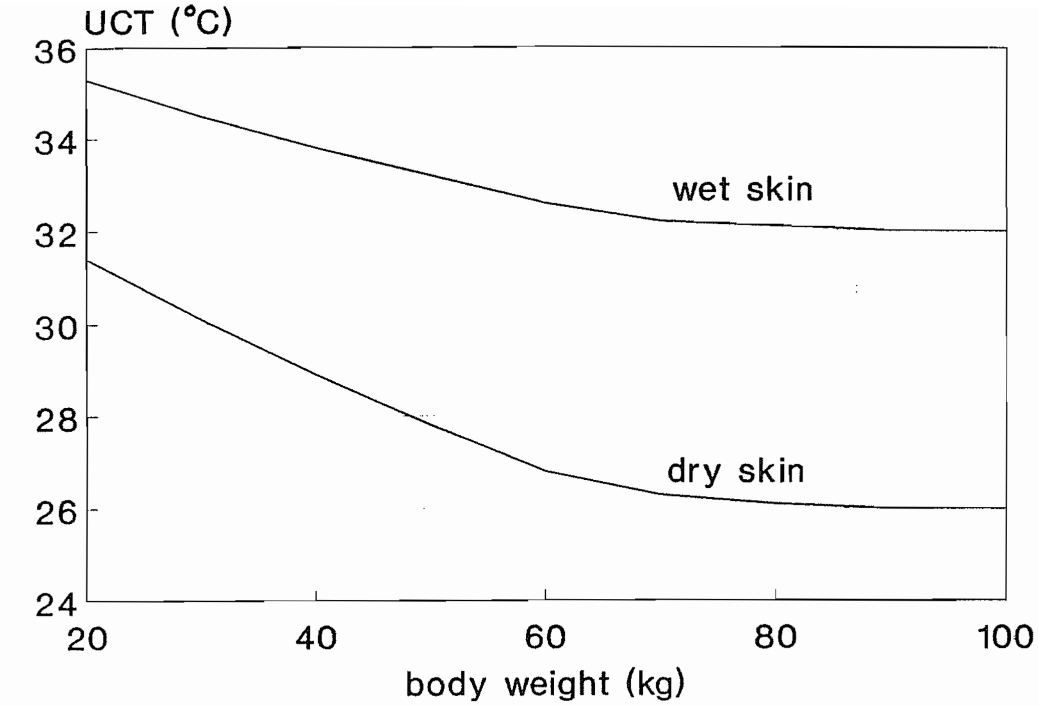

Fig. 14 : Upper critical temperature for pigs effected by sprinkling. Gain 750 g/day.

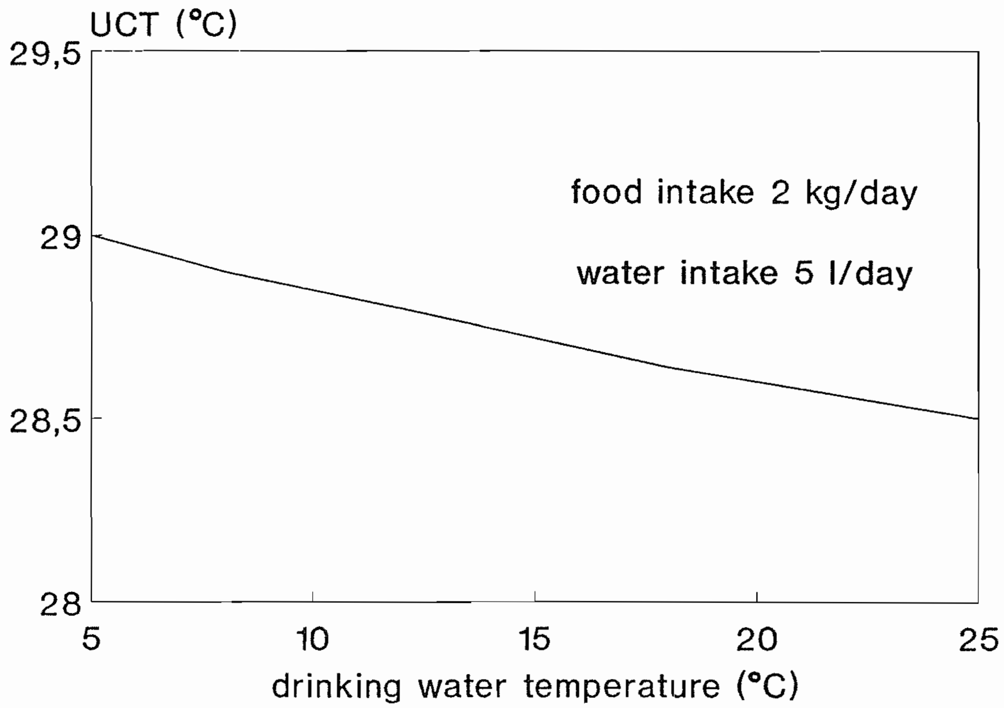

Fig. 15 : Upper critical temperature for a 50 kg fattening pig as a function of drinking water temperature

1.7. Conclusions

The climate of pigs at animal level can be improved by correct human intervention at housing level.

A better climate can lead to better wellbeing of the pig and can also lead to a better result for the farmer.

Models are very useful for finding effective solutions.

Both the BEZOVA and STALKL computed models are being used by extension workers in the Netherlands to give collective or individual advice to farmers.

The computed STALKL model is being used to draw up a standard for the ventilation of pig houses in the Netherlands.

1.8. References

BLAXTER, K.L. The energy metabolism of ruminants. Charles C. Thomas, Springfield, Illinois, 1967.

BOON, C.R. The effect of departures from lower critical temperatures on group postural behavior of pigs. Anim. Prod. 33, 1981.

BRODY, J.M. Bioenergetics and Growth. Rheinhold Corp. New York, N.Y., 1945.

BRUCE, J.M. Conductive heat loss from the recumbent animal. Farm Building R and D Studies 8, 1977.

BRUCE, J.M. and J.J. CLARK. Models of heat production and critical temperature for growing pigs. Anim. Prod. 28, 1979.

BRUCE, J.M. Ventilation and temperature control criteria for pigs. In "Environmental aspects of housing for animal production", ed. J.A. Clark. London, 1981.

CIGR. Report of the working group on Climatization of animal houses, SFBIU, Aberdeen, 1984.

CLOSE, W.H. and V.R. FOWLER. Energy requirements of pigs. In "Recent advances in animal nutrition", ed. Haresign, W. Butterworth, London, 1983.

FOWLER, V.R., M.F. FULLER, W.H. CLOSE and C.R. WHITTEMORE. Energy requirements for the growing pig. "Energy metabolism", ed. L.E. Mount. Butterworths, London, 1983.

KELLEY, K.W., S.E. CURTIS, G.T. MARZAN, H.M. KARARA and C.R. ANDERSON. Body surface area of female swine. Journal of Anim. Sci. 36, 1973.

MOUNT, L.E. The climatic physiology of the pig. Edward Arnold, London, 1968.

STERRENBURG, P. en E.N.J. VAN OUWERKERK. Rekenmodel voor de bepaling van de thermische behaaglijkheidszone van varkens (BEZOVA). Rapport 78, Instituut voor Mechanisatie, Arbeid en Gebouwen IMAG, Wageningen, 1986.

VERSTEGEN, M.W.A. Influence of environmental temperature on energy metabolism of growing pigs housed individually and in groups. Thesis, Agricultural University, Wageningen, 1971.

Responsible member PEDERSEN S.	*Chapter 2* **MOISTURE PRODUCTION AND CORRECTION FACTORS FOR SENSIBLE HEAT**

Chapter 2

MOISTURE PRODUCTION AND CORRECTION FACTORS FOR SENSIBLE HEAT

INTRODUCTION

One of the main goals for the work, presented in the CIGR report on Climatization of Animal Houses, 1984, was to stipulate the heat production for different kinds of animals. In addition to the heat production from the animals it is necessary to take into account the necessary heat for evaporation of water from wet floor surfaces, manure, feeding, bedding, etc. This theme is discussed in the following paragraphs.

2.1. Heat production

The heat production for animals is expressed in hpu (heat producing unit), where 1 hpu is defined as a group of animals with a total heat production of 1 000 W at 20 °C (STRØM, 1978). At lower temperatures the total heat production will be higher than 1 000 W, and at temperatures above 20 °C it will be lower. The total heat production at different temperatures can be corrected by the following formula :

$$F = 4 \times 10^{-5} (20 - t)^3 + 1,$$

where

F = correction factor for total heat,

t = Indoor temperature, °C.

When the total heat is calculated for a certain temperature, the sensible part can be calculated by the following formulae :

$$f_s = 0.8 - 1.85 \times 10^{-7} (t + 10)^4,$$

and the sensible heat by :

$$\Phi_s = \Phi_{at} \times f_s$$

where

f_s = sensible part of total heat
Φ_{at} = total heat, W
Φ_s = sensible heat, W

Figure 1 shows how the total heat is distributed as latent and sensible heat from the animals themselves, originally based on data for cattle, pigs and poultry.

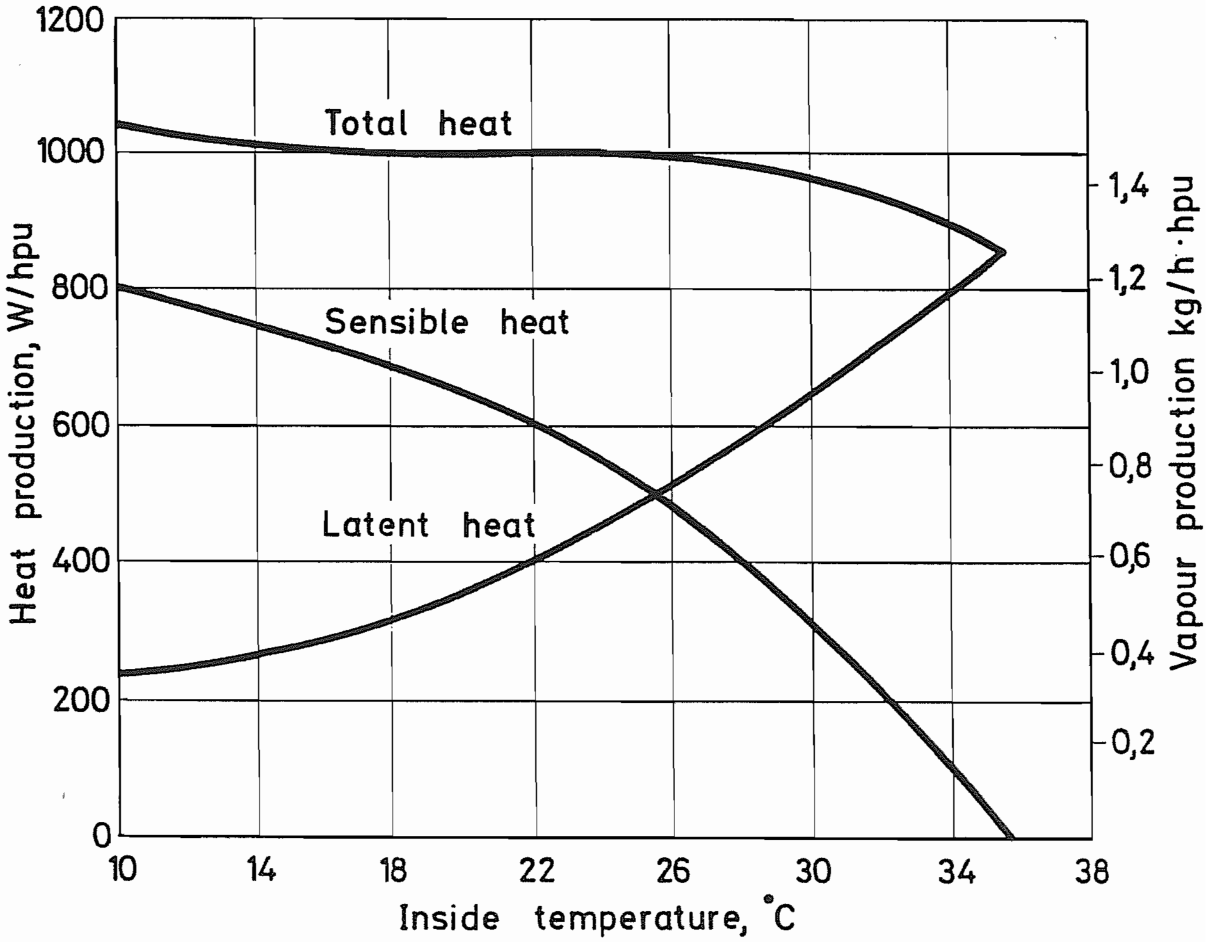

Fig. 1 : Distribution of total heat on sensible and latent heat for a heat producing unit, hpu

2.2. Correction factors for calculating the actual sensible heat

The hpu defined above is based on the information available more than ten years ago, and it is obvious that the distribution of total heat as latent and sensible heat has to be examined further, as well as the influence of evaporation of water from feed and surfaces, which ought to be taken into account for calculating the real sensible heat.

The distribution in Figure 1 was proposed as a "unit of measurement". By using an individual correction factor, k_s, it will be possible to calculate the corrected sensible heat under certain conditions.

The correction factor k_s will mainly depend on :

- Type of animal
- Production levels (feed intake)
- Housing condition (wet or dry)

A correction factor, k_s, of 1 means no correction of Figure 1. If the factor is lower than 1, the sensible heat is lower, and the latent heat correspondingly higher, and vice versa.

If the correction factor k_s is calculated, it is easy also to calculate the correction factor k_l (latent) by using the following formulae :

$$k_l = \frac{1 - f_s \times k_s}{1 - f_s}$$

$$k_s = \frac{1 - (1 - f_s)\, k_l}{f_s}$$

Table 1 shows correction factors for different types of animal, and table 2 shows some provisional recommendations for the correction factor k_s, both presented earlier in the 1984 CIGR Report.

Table 1 : Some correction factors k_s for calculating the actual sensible heat

Description	Indoor temperature (°C)	k_s	Source
Fattening pigs			
on partly slatted floor	13	0.91	1)
	18	0.91	1)
on concrete	13	0.91	1)
	23	0.93	1)
Calves	8	0.91	1)
	13	0.85	1)
Heifers			
silage slatted floor	8	0.78	1)
	13	0.72	1)
Beef cattle			
slatted floor	11 *	0.75	2)
Milking cows	12 *	0.90	3)

* assumed indoor temperature

1) PEDERSEN, S.
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Table 2 : Provisional recommendations for the correction factor for sensible heat, k_S

General conditions	Very dry	Dry	Wet
	Dry feed * Dry floor	Dry feed * Average floor	Wet feed + Wet floor
Pigs	1.0	0.95	0.9
Cattle	0.9	0.85	0.8

* dry feed = hay, straw, grain

+ wet feed = silage DM < 30 %, liquid feed

Since 1984 additional experiments have been carried out. Table 3 shows some Danish results with weaned pigs.

It is assumed that the high k_S at low ventilation rate is due mostly to condensation on the building structure etc., because of high relative humidity.

Table 3 : Calculated correction factors for weaner houses, based on experiments at SjF, winter 1985-1986.

Minimum ventilation rate (m^3/h.pig)	Weight range (kg)	Outside temp. (°C)	Indoor temp. (°C)	Relative humidity %	Correction factor k_S
2	10 - 15	0.5	20.4	76	1.30
	15 - 20	-0.5	21.0	75	1.19
	20 - 30	1.5	20.6	76	1.04
3	10 - 15	4.0	19.8	66	1.26
	15 - 20	-2.0	18.5	69	1.18
	20 - 30	- 4.5	17.7	69	1.02
4	10 - 15	-5.0	18.9	57	1.05
	15 - 20	-6.0	19.5	63	1.06
	20 - 30	-7.5	19.3	68	1.03

At higher ventilation rates the k_S is close to 1. The correction factors, shown in table 3, indicate that the distribution in Figure 1 can be used without any correction factors, but because of insufficient data it is not possible, yet, to verify the provisional data for pigs shown in Table 2.

Figure 2 shows how the correction factor k_S for milking cows (wet condition) is dependent on the indoor temperature. At low indoor temperatures the correction factor is close to 1, but at high indoor temperatures the correction factor is much lower than 1. This means that the sensible heat is less than indicated in Figure 1. The reason could be that cattle produce a greater part of the heat as latent heat than indicated in Figure 1. Another reason could be evaporation of water from wet surfaces. Evaporation of 1 kg water requires an energy of 2 450 kJ, which can only be taken from the sensible heat.

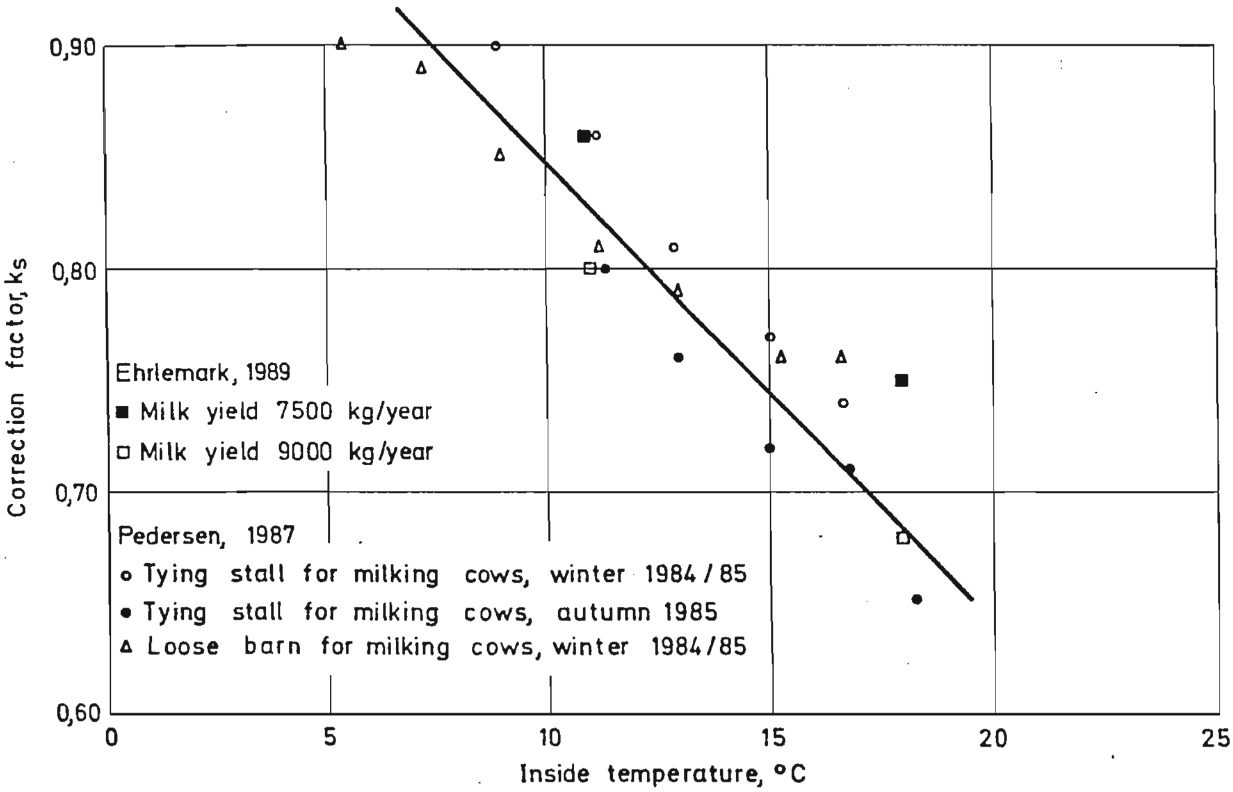


Fig. 2 : Correction factors for milking cows

In Sweden some work has been done on the distribution of total heat as sensible heat and latent heat for cattle at different production levels. EHRLEMARK (1988 and 1989) stated that the amount of sensible heat is limited to the surface area of the animals. This means that the animals with high production levels (high feed intake) will produce a relatively higher part as latent heat, as compared to animals with low production levels. Table 4 shows some calculated values by EHRLEMARK for a herd with continuous calving, at three different yield levels.

Table 4 : Calculated correction factors k_s for milking cows

Yield level (kg/year)	Body weight (kg)	Indoor temperature 12 °C	18 °C	24 °C
5 000	550	0.99	0.84	0.78
7 500	575	0.86	0.75	0.68
9 000	600	0.80	0.68	0.63

Values at a yield of 7 500 and 9 000 kg/year at 12 °C and 18 °C are also presented in fig. 2. The results are in good agreement with the Danish figures, and show that there is a lower correction factor, k_S, at a milk yield of 9 000 kg/year than at a yield of 7 500 kg/year. The results in Figure 2 indicate that the provisional data in table 2 need to be related to the indoor temperature, production level, etc., but more results are needed, before the table can be adjusted.

2.3. Conclusions

In the future more information on heat production will be available, and it should be possible to adjust Figure 1. However, until then the hpu, as described in this chapter, seems to be a good common "unit of measurement", and deviation can be stated by the correction factor k_S. It is likely that the relation between total heat and indoor temperature in Figure 1 is close to the truth and will be changed only slightly in the future, but the distribution of the latent heat and sensible heat will depend on the kind of animal, housing conditions, production level, etc. It will probably be necessary to make individual figures like Figure 1 for different combinations of animals, housing conditions, etc.

2.4. Literature

Climatization of Animal Houses. CIGR-Report, 1984. Published by the Scottish Farm Building Investigation Unit, Craibstone, Bucksburn, Aberdeen.

EHRLEMARK, A., 1988. Calculation of Sensible Heat and Moisture Loss from Housed Beef Cattle, Using a Heat Balance Model. Swedish University of Agricultural Sciences, Department for Farm Buildings, Report no. 60.

EHRLEMARK, A. 1989. Personal communication. Swedish University of Agricultural Sciences.

STRØM, J. 1978. Heat Loss from Cattle, Swine and Poultry as Basis for Design of Environmental Control Systems in Livestock Buildings. SBI-landbrugsbyggeri 55. Statens Byggeforskningsinstitut, Denmark.

Responsible member

SÄLLVIK K.
BARTUSSEK H.
(3.7. to 3.7.7.)

Chapter 3

DESIGN OF VENTILATION SYSTEMS

3.1. Design specifications for a ventilation system

A ventilation system shall, for a given building, interior, lay-out and type of production, provide a climatic environment which maintains efficient production, healthy animals, good working conditions and a long working life of equipment and building.

Designing a ventilation system for a specific building requires specifications for the system performance to be achieved. The design specifications can be divided into four main aspects :

- biological
- technical
- climatic
- economic

3.1.1. Biological design parameters

Specifications needed for the interior environment are :

1) Design temperature, maximum RH and air velocity for **winter** season. Supplementary information on animal response when design temperature and/or velocity is not maintained, i.e. cold stress.
2) Design temperature and range of RH and air velocity for **summer** season. Supplementary information on diurnal cycling temperature or other ventilation methods to relieve animals from undesired heat stress.

3) Maximum levels of air contaminants such as CO_2, NH_3, H_2S, dust, micro-organisms and also sound level.

4) Comfort and management limits of the herdsman.

Basic data can be compiled from the extensive research done on climatic impact on animal production, e.g. feed efficiency, gain, yield, reproduction, health and behaviour (PEDERSEN, S., KRISTENSEN, B. 1977 ; HAHN, G.L., SÄLLVIK, K. 1979 ; HAHN, G.L., NYGAARD, A., SIMENSEN, E. 1983 ; SÄLLVIK, K.,WALBERG, K. 1984 ; STERRENBURG, P., VAN OUWERKERK, E. 1986).

3.1.2. Technical design parameters

The ventilation system must be an integrated part of the total building. In order to achieve this the following specifications are needed :

1) Type of building construction.
2) Size, material and thermal resistance of all surrounding surfaces.
3) Kind, number, production and weight of animals.
4) Distribution of animals in the building.
5) Design, lay-out, including details of fittings.
6) Manure handling system.
7) Obstacles to incoming air jets and other air movements.
8) Fire protection measures.
9) Odour, ammonia and noise emission to the surroundings.
10) Environmental (RH, NH_3, H_2S and dust) effect on the equipment.
11) Energy supply and price.
12) Management and maintenance of the ventilation system.
13) Job routines in the barn.

3.1.3. Climatic conditions

For a specific geographical location specifications needed are probability of temperature and RH to be used for :

1) Technical considerations : over- or underpressure ventilation system, insulation, designing maximum air flow of the ventilation system, heating and cooling

2) Economic considerations : probabilities of production losses due to thermal stress, annual energy consumption.

Such information is available in SÄLLVIK, K., NILSSON, C. and NIMMERMARK, S., 1978 ; HAHN and SÄLLVIK, 1979 and CIGR, 1984.

3.1.4. Economic considerations

Specifications needed are :

1) Probability of output from the animal production caused by environmental factors which can be affected by ventilation.

2) Value of output lost by animals at non-optimum conditions.

3) Future development of prices including energy and labour.

4) Annual investment and running costs.

5) Amount of money available for investment in the ventilation system.

3.2. Ventilation principles - Mechanical ventilation

A ventilation system can be characterized according to :

1) Pressure difference between in- and outside temperature of the building :

* underpressure, negative pressure, exhaust system
* overpressure, positive pressure
* neutral pressure, equal pressure, balanced system

2) Mechanical or natural

In this CIGR report we have chosen to distinguish between mechanical and natural ventilation.

3.2.1. Choosing a ventilation system

Each ventilation system has advantages and drawbacks. See Table 1. Advantages for different systems could be coordinated into a new system avoiding earlier drawbacks.

Table 1 : Comparison between different ventilation systems

System	+	-
underpressure (Δp 5 - 20 Pa)	cheap simple exhausting of manure gases	leakages varying air movements limited air velocity in the air inlet
overpressure (Δp 5 - 30 Pa)	constant air movements	risk of condensation due to air currents into the building construction. Water vapour diffusion and condensation
neutral pressure ($\Delta p \pm 5$ Pa)	constant air movements exhausting of manure gases	some risk of condensation high energy consumption for running fans

The conclusion of the comparison in table 1 is that :

during winter either an **under** or **equal** pressure system is best

during summer an equal or **over** pressure system should be preferred if higher velocities are wanted to improve animal comfort and/or to guarantee an ensured air distribution when windows and doors are opened.

3.3. Incoming air characteristics

Of importance is to quantitatively predict how temperature, velocity and humidity in the air jet entering the barn will be affected by inlet parameters and what secondary effects the entering air jet has on temperature, velocity, humidity, gas-distribution and air movements in the air surrounding the animals.

A lot of work has been done in the field of describing air velocity profiles at different distances from inlets of various configuration and location (DEBRUYCKERE, M., 1967 ; WILSON, J.D., ESMAY, M.L. & PERSSON, S., 1968 ; WHITE, G.M., WALKER, I.M. & BLACK, T.C. 1969 ; SÄLLVIK, K., 1971 ; ALBRIGHT, L.D., 1972 ; PEDERSEN, S., 1976 ; SÄLLVIK, K., 1978).

The major part of these reports is based on isothermal conditions or for vertical jets, which makes the application to barns less useful. This is true especially for winter conditions when simultaneous values of temperature and velocity are creating the thermal characteristic of the jet which is related to animal comfort.

Characteristics of the following types of inlet are to be described :

- slotted inlet for outside air
- air disc (air distributors) with mixing of barn air
- porous ceiling
- inflatable duct of air penetrable fabric

3.3.1. Air velocity in the inlet

As a consequence of the assumptions for exhaust ventilation, air velocity in the inlet is a function of Δt and inlet characteristics including adjustment of inlet openings, assuming no recirculation of inside air. Since buildings are not air tight outside air will also enter through openings other than inlets i.e. infiltration ; equation 1. The proportion of infiltrated air will increase by increased air flow and inlet velocity i.e. the underpressure in the barn (SÄLLVIK, K., 1971).

$$\text{Infiltration} = 1 - \frac{\text{inlet airflow}}{\text{outlet airflow}} \tag{1}$$

3.3.2. Throw in a horizontal jet

The definition used here is how far from the inlet the horizontal jet will travel before it has descended 0.5 m. The influence of inlet parameters on the throw has been quantified by DEBRUYCKERE, M., 1967 and BECHER, 1952. The equation for a flat jet from an inlet close to the ceiling, i.e. the inlet opening closer to the ceiling than 5 times the opening, is presented by DEBRUYCKERE as :

$$x = \frac{w}{\delta}\left[\sqrt[2.5]{\frac{\delta^2 \cdot y \cdot v_t^2 \cdot T_i \cdot \sqrt{\frac{T_i}{T_t}}}{g \cdot w^2 \cdot \Delta t \cdot 0.226}} - 0.205\right] \tag{2}$$

x = throw, m

δ = turbulence coefficient 0.09 - 0.20 for a flat jet according to BATURIN 1959. Used value $\delta = 0.12$

v_t = air velocity in the inlet, m/s
w = width of inlet opening, m
y = vertical distance from the jet to the inlet, m
T_i = absolute temperature, K of room air
T_t = absolute temperature, K of jet in the inlet
g = 9.81 m/s^2
Δt = $T_i - T_t$

The theoretical throw is generally longer than the actual. The equation used can predict the relative influence of Δt and velocity on the throw. When Δt is below 12 °C, a very rapid increase of the throw occurs (Fig. 1). This phenomena has been observed by many farmers and extension people, describing the different air movements in a barn as winter pattern and summer patterns respectively. SÄLLVIK, K. (1979) suggests equation (3) for v_t ranging between 0.5 to 4.5 m/s and for Δt over 12 °C.

$$\text{Throw} = 0.09 + 0.55\, v_t \quad (\text{m}) \tag{3}$$

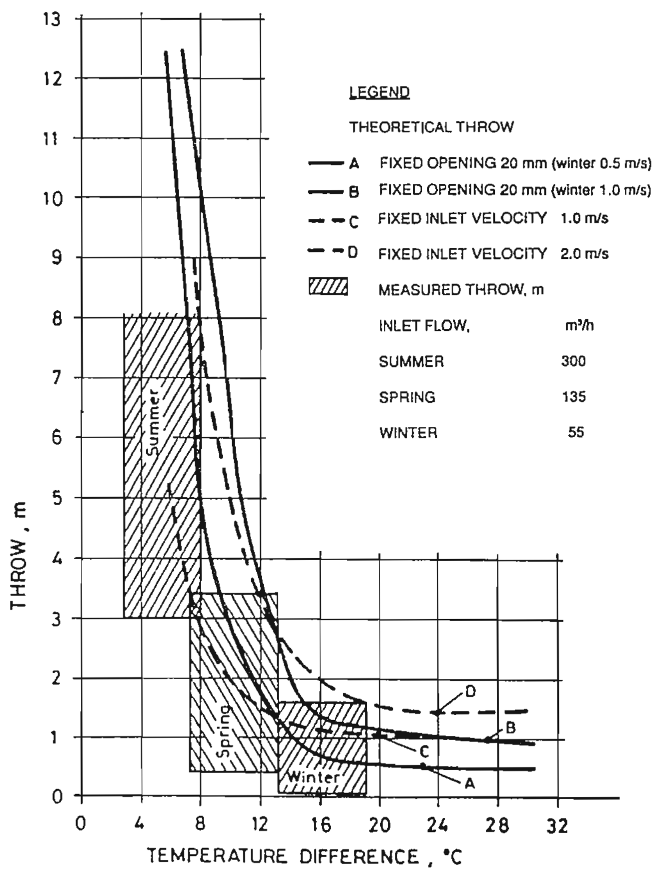


Fig. 1 : Theoretical and measured throw from a baffled slotted inlet

3.3.3. Velocity profile in a vertical jet

The air velocity in a vertical jet up to 2 m from the inlet can be estimated according to equation 4 (SÄLLVIK, K., 1979).

$$v_x = v_t \, . \, 2.3 \, \frac{\sqrt{A}}{x} \qquad (4)$$

v_t = inlet velocity m/s
v_x = air velocity in a vertical jet x m from the inlet m/s
A = inlet opening, m^2

3.3.4. Velocities at the animal level as a function of inlet characteristics

PEDERSEN (1976) always found an increasing relation between the impulse (flow times velocity, kgm/s^2) in the inlet creating a horizontal jet above animals and the average velocity in the dwelling zone of the animals, 0.1 - 1.5 m above floor level (fig. 2). This study was carried out with no animals and no obstacles to air flow. PEDERSEN established the following relationship :

$$v = K \, (\delta \, . \, V \, . \, v_t)^{0.5} \qquad (5)$$

v = average velocity at animal level, m/s
V = volume flow, m^3/s
v_t = inlet velocity, m/s
δ = density of air, kg/m^3 (δ (20 °C) = 1.2 δ (-20 °C) = 1.4)
K = factor depending on width of barn
K = 0.43 for 10 m wide barn
K = 0.41 for 15 m wide barn
K = 0.38 for 20 m wide barn

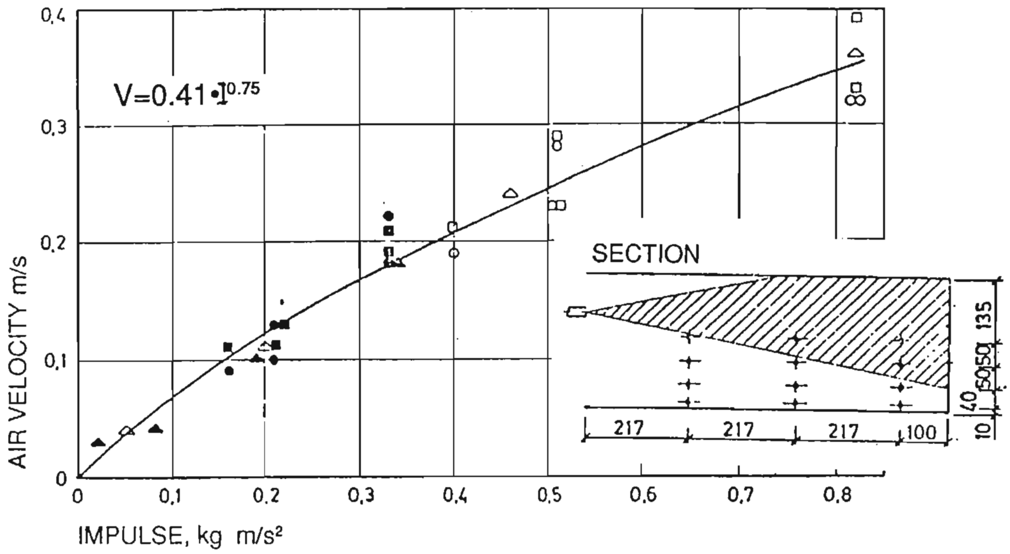


Fig. 2 : Air velocity (V,m/s) in the animal dwelling zone as a function of the impulse (I) of the entering jet. Isothermal conditions. Width of the house 15 m. PEDERSEN, 1976.

For inlets located in the ceiling at the outer wall and directing the incoming air along the ceiling SÄLLVIK, K. (1979) got the opposite relation between impulse and air velocity over the gutter, 1.5 m from the outer wall, cows facing to the center (Fig. 3) so that a higher impulse results in a lower velocity. This is a result of the inlet having no direct effect at high impulse on the air movement over the gutter. At lower impulse the incoming air will fall down over the alley which increases the air velocity over the gutter. Trying to get a low air velocity over the gutter you must remember that the impulse is a function of outside temperature when there is no recirculation of inside air, which restricts the possibilities of obtaining low inlet velocities during the winter.

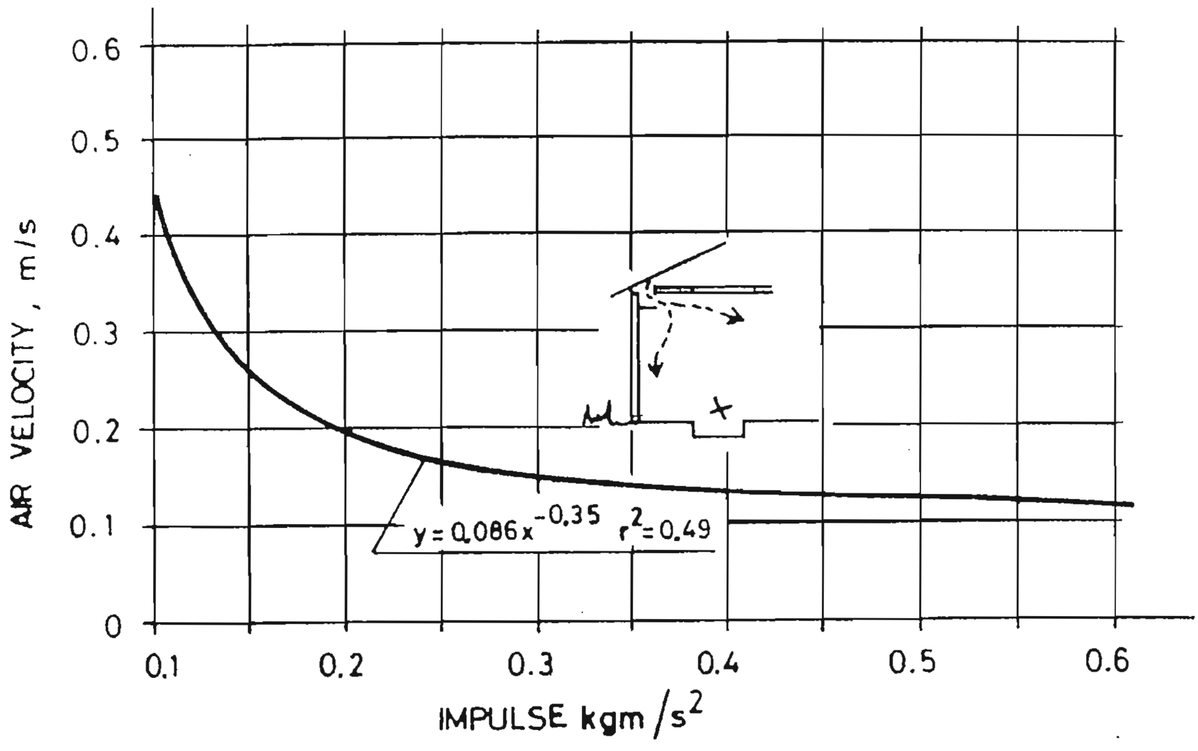


Fig. 3 : Air velocity over the gutter as a function of the impulse in a baffled inlet at the outer wall in a barn for stanchioned cattle. SÄLLVIK, 1979

3.3.5. Temperature profile and mixing ratio in a vertical jet

Free vertical jets originate from inlets located in the ceiling, which are at least 1 m from the outer wall.

Mixing of incoming air and inside air can be expressed in temperatures :

$$\theta = \frac{t_x - t_t}{t_i - t_t} \tag{6}$$

where :

t_x = temperature at distance x m from the inlet, °C

t_t = temperature of the inlet, °C

t_i = temperature in the room, °C

θ = 1 means that the jet has reached room temperature e.g. complete mixing.

SÄLLVIK (1979) proposed equation (7) also illustrated in Fig. 4.

$$\theta = 1 - 0.75\, e^{-(0.08.x/w)} \tag{7}$$

where :

x = distance from inlet, m

w = width of slot, m

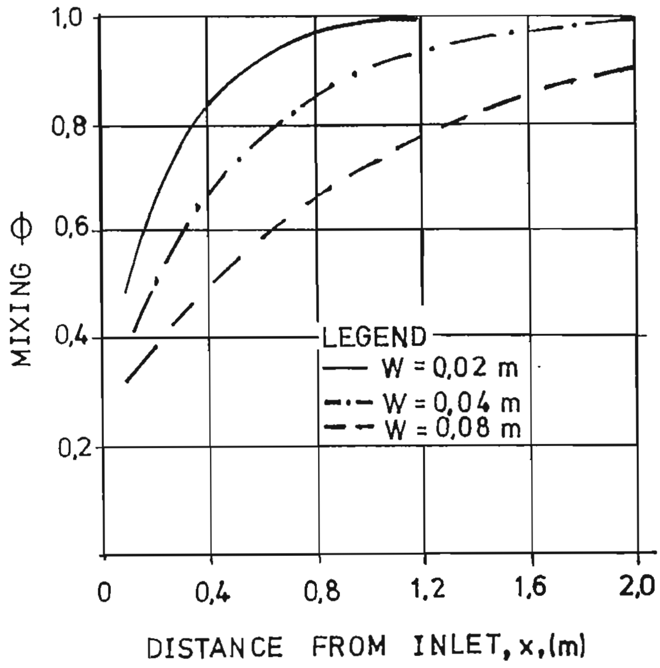


Fig. 4 : Mixing in a vertical free jet as function of width of inlet slot (W) and distance. SÄLLVIK, 1979

3.3.6. Chill factor

The cooling effect of an air jet is due to heat loss from the animal body surface by convection, φ_c (W/m^2)

$$\varphi_c = \alpha_c \, (t_b - t_x) \tag{8}$$

t_b = temperature of animal surface, °C
t_x = air temperature at animal surface, °C

The expression for the convective heat transfer coefficient is

$$\alpha_c = \frac{k_c \, . \, N_u}{d} \tag{9}$$

where :
k_c = thermal conductivity of air (W/mk)
N_u = Nusselt number
d = characteristic dimension of the object (m)

WIERSMA & NELSON (1966) suggested $\alpha_c = 9 \sqrt{v}$ (W/m^2 K) for cattle, BRUCE and CLARK (1979) $\alpha_c = \frac{15.7 \cdot v^{0.6}}{m^{0.132}}$ for pigs.

The convection heat loss caused by a free jet can be found in combining the expressions for mixing and air velocity and using some of the expressions for convective heat transfer coefficient. However it must be remembered that the temperature of animal surface t_b is a function of animal heat balance and can vary considerably, see Table 2.

Table 2 : Surface temperature of cattle, pigs and poultry at different environment temperature

Environment	Cattle		Pigs		Poultry	
temperature, °C	-20	30	0	30	5	30
Surface temperature °C	10	33 - 37	24 - 32	33 - 37	10	33

3.3.7. Flow - pressure drop relation

To design a ventilation system you must know the relation between the air flow and static pressure drop to get a proper selection of fans or design of natural exhaust openings and to avoid too high underpressure in the barn (Fig. 5).

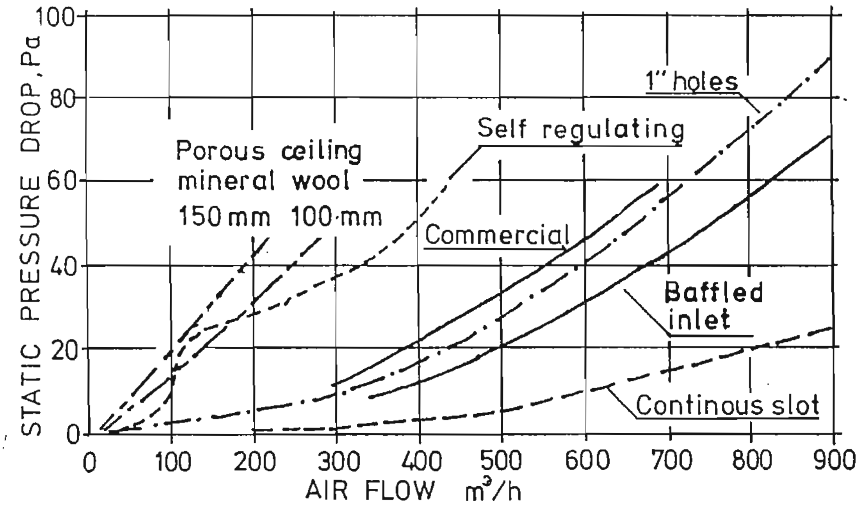


Fig. 5 : Air flow and pressure drop for different types of inlets

For each inlet and a certain opening, the air resistance value, φ, can be measured according to equation (10).

For tested commercial inlets φ-value has varied from 5 to 1.7 for full openings and from 1.6 to 0.7 for 25 % opening.

$$\Delta p = \varphi \,.\, 0.6 \,.\, v^2 \tag{10}$$

Δp = pressure drop, Pa
φ = air resistance value
v = air velocity, m/s

3.4. Inlet facilities

Before discussing inlet facilities it must be pointed out that in a house all openings to the surrounding air can act as inlets due to differences in air pressure. Consequently the building must be as airtight as possible if you want to control the air exchange within the building and/or prevent uncontrolled infiltration of outside air. This demand is important and valid for all types of ventilation system.

Objectives for the inlets are to :

1) Distribute incoming air into the barn according to the animals' need or heat production in the barn

2) Create desired air movements, considering thermal comfort around animals and herdsmen and avoiding release of contaminants into the air.

In this report we have chosen to characterize air inlets according to two different criteria :

* any type of recirculation of inside air with the outside air before air is entering the barn. This will influence temperature, velocity and air flow in the inlet.

* air velocity in the inlet is below 0.2 m/s or above 0.5 m/s. When inlet velocity is below 0.2 m/s there will be no influence on air velocities or air movements in the barn.

The following types of inlets are to be characterized and discussed :

* inlets for outside air
* inlets with recirculated inside air

* air discs (air distributors) with circulation of inside air
* porous ceiling

3.4.1. Inlet for outside air

Using only outside air means that the air flow through the inlet is negatively correlated to the temperature difference between inside and outside (Δt). Furthermore you will get restrictions on air velocity and air flow in the inlet during the cold season. For horizontally baffled inlets the air flow pattern will then vary considerably between winter and summer, even with adjustable openings.

* Inlet velocity will vary between 0.5 - 4.0 m/s in an underpressure ventilation system and between 0.5 - 10.0 m/s in an overpressure ventilation system

* Air flow in an inlet expressed as air velocity is between 2 - 3 m/s and 3 - 4.5 m/s at 10 and 20 Pa static pressure difference respectively for tested inlets. Fig. 6 shows a typical baffled inlet located in the ceiling.

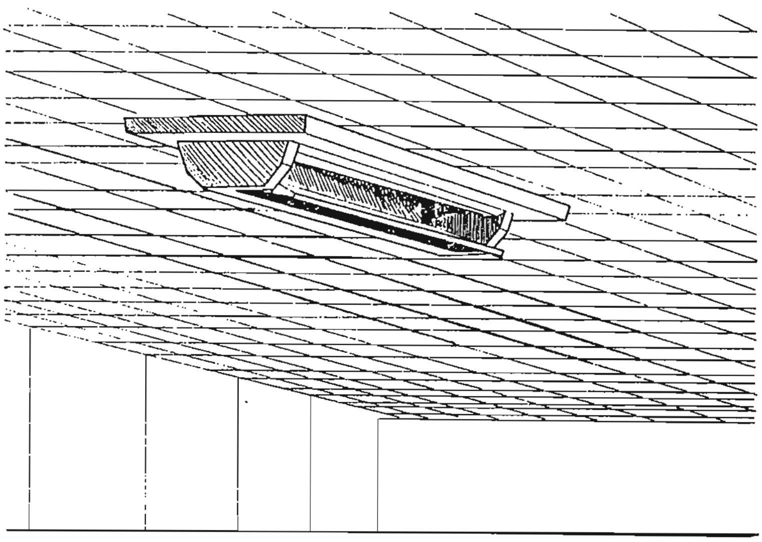


Fig. 6 : Adjustable slotted inlet located in the ceiling. Underpressure system.

* At outside temperatures below -0 °C condensation can occur ; at temperatures below - 10 °C freezing can occur. Inlet design must consider and solve these problems.

* In connection with ducts there should be guide vanes (grilles) in the inlet to direct the air in a right angle out of the duct (Fig. 7).

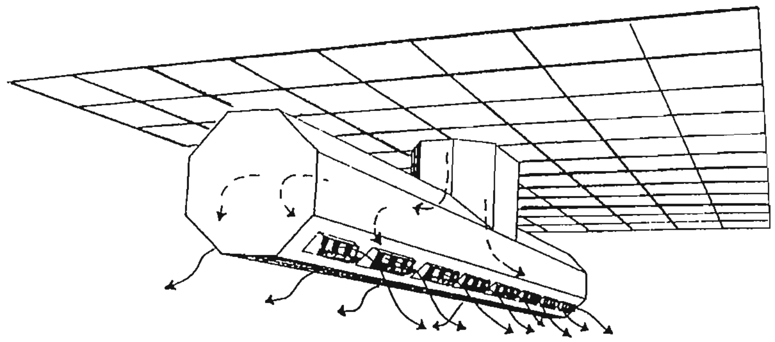


Fig. 7 : Adjustable slotted inlets integrated with guide vanes in an insulated air duct. Over-, equal- and underpressure systems.

3.4.2. Air discs with mixing of barn air

The maximum air flow from one disc is 4 000 m³/h with underpressure and 11 000 m³/h with overpressure.

The diameter of the disc is between 900 - 1 400 mm. It is very important that the air distribution can be adjusted both horizontally and vertically. Fig. 8.

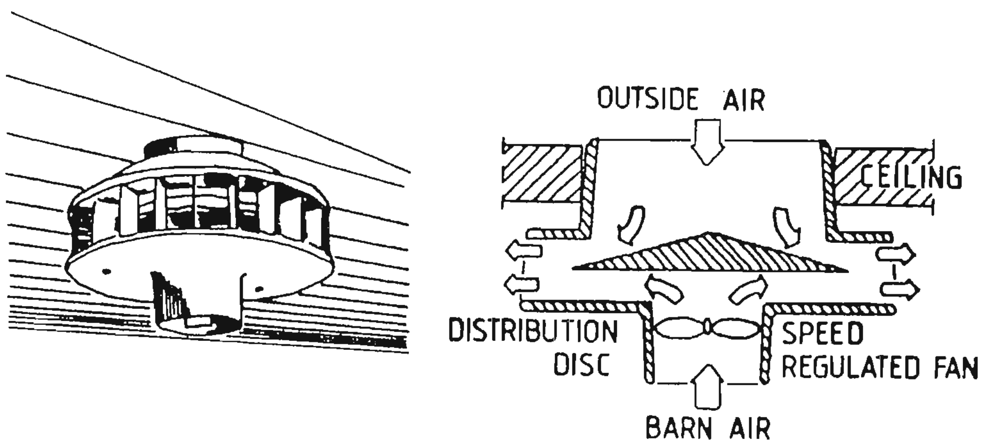


Fig. 8 : Example of an air disc mixing outside air with barn air

The mixing of inside air is used to increase the temperature of the incoming air and thereby get the same air pattern independent of outside temperature. Air velocity in the inlet varies between 4 - 8 m/s. Because of the high impulse in the incoming jet the coordination between the air flow pattern layout and fittings must be carefully done.

3.4.3. Porous ceiling

Instead of using conventional types of high speed air inlets (holes, slots, recirculating inlets, etc.) it is possible to use a porous ceiling as air inlet. As the air enters at an ideal location and with an ideal distribution we get a more even room air temperature distribution than in rooms with other ventilation systems. A breathing ceiling picks up transmission heat loss and preheats inlet air close to room air already at 100 mm underneath ceiling surface.

The basic principle is to have a ceiling which is uniform for air-penetration. Technically the ceiling could be constructed with mineral wool on netting, self supported mineral wool, coconut fibre mats on netting, or wood-wool slab alone as support for mineral wool. It is also possible to use spaced wood slats covered with hay or straw as a porous ceiling. Fig. 9 gives an example.

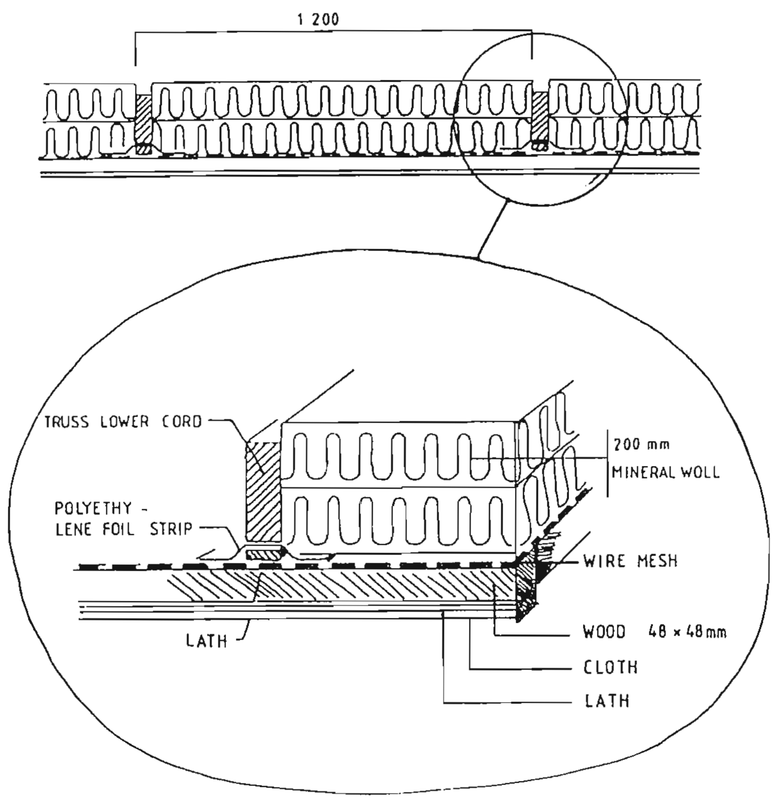


Fig. 9 : Breathing ceiling cross-section with details. GRAEE, 1988.

It is fundamental that the construction shall prevent diffusion of water vapour (and/or leakage of barn air). Since there is no vapour barrier in a porous ceiling, careful design is required.

Advantages :

* No transmission heat losses through the ceiling. The porous ceiling is acting as a simple heat exchanger combined with the function as air inlet. The incoming air recovers transmission heat losses when passing through the porous ceiling.
* No chill effect on the animals by forced convection heat losses. Incoming air is preheated and enters the barn at 0.001 - 0.01 m/s compared to 2 - 5 m/s for conventional inlets.
* Very even air distribution due to the large surface of inlet.
* Simplified ventilation control system, since no adjustments of inlet openings are needed.
* Lower investment costs. No inlets. No cladding sheets.
* Filtering of incoming air, reducing entrance of airborne disease germs.
* Reduction of noise level.
* Daylight trap.

Drawbacks or questionmarks :

* The design guidelines must be very carefully fulfilled and the construction very carefully done.
* Excessive solar heating of attic space in warm weather. Can be solved in different ways :
  . Light roof colours, high roof pitch, 27 ° or more, well ventilated attic space preferably by open ridge.
  . Double roof
* The life time regarding clogging by atmospheric dust of a breathing ceiling ; using mineral wool and an average air flow of 80 m^3/m^2 surface it could be estimated to exceed 20 years.

Design rules

The following guidelines will ensure a proper function of porous ceiling inlet systems :

1. Determine the minimum air velocity, v_{min}, through the ceiling to avoid vapour diffusion. The lowest safe velocity is 3 m/h at -10 °C and 5 m/h at -20 °C.

2. Calculate the maximum acceptable porous area according equation 11.

$$A_{max} = \frac{q_{min}}{v_{min}} \tag{11}$$

A_{max} = maximum acceptable area of the porous celing, m^2

q_{min} = minimum continuous ventilation rate, m^3/h

v_{min} = minimum design air velocity to avoid condensation or backward flow, m/h

3. The minimum acceptable area depends on the highest acceptable air pressure drop through the porous ceiling at maximum ventilation rate i.e. underpressure in the barn.

$$A_{min} = \frac{q_{max} \cdot d}{\Delta P_{max} \cdot l} \tag{12}$$

A_{min} = maximum acceptable area of the porous ceiling, m^2

q_{max} = maximum ventilation rate, m^3/h

ΔP_{max} = maximum acceptable pressure drop through the porous ceiling, usually 10 - 20 Pa

d = thickness of the porous ceiling material, m

l = air penetrability of the material, m^2 (Pa.h) (see table 3 below)

Table 3 : Design values for the air penetrability of mineral wool of different densities

Density of the material, kg/m^3	Air penetrability, l, $m^2/(Pa.h)$
15	0.70
20	0.60
25	0.45
40	0.30

Much information on porous ceilings is given in "PORENLÜFTUNG, 1988".

3.4.4. Inflatable duct of air-permeable fabric

By making the envelope of an air duct of permeable fabric and using the whole area of the duct as an air distributor you will get very low air velocities, 0.01 - 0.1 m/s, and also avoid condensation on the outer surface during winter since the moist barn air will never reach the duct (Fig. 10).

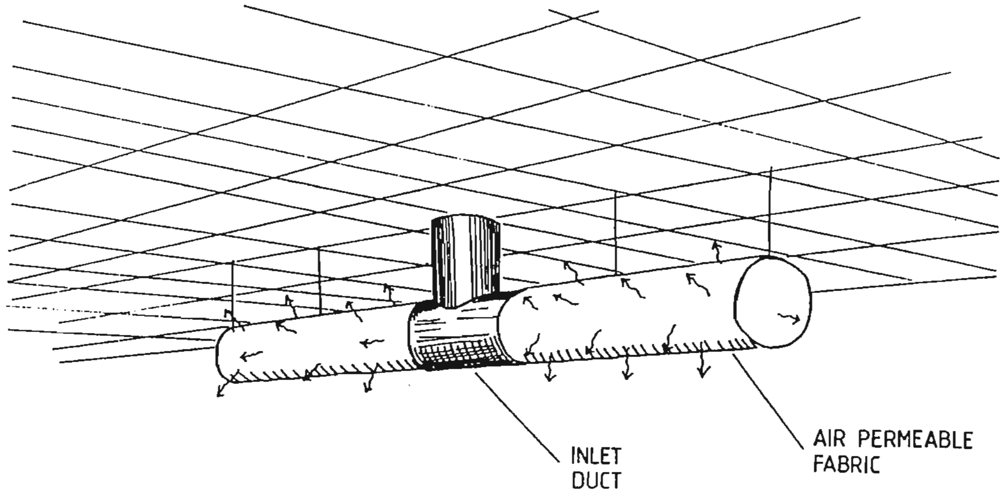


Fig. 10 : Duct of hose of air-permeable fabric gives an even air distribution. Under- or overpressure-system.

Capacity of an air-permeable fabric for use in barns is about 75 $m^3/m^2.h$ at 10 Pa, 150 $m^3/m^2.h$ at 20 Pa and 290 $m^3/m^2.h$ at 40 Pa (only possible with overpressure system) static pressure difference. Ducts usually have diameters between 0.4 - 0.6 m.

To avoid condensation on the ceiling above the duct due to cooling by radiation, which then drips on the duct, there must be at least 25 cm distance between duct and ceiling. In all systems using some kind of ducts for air distribution you can heat or cool the incoming air. Such treatment of air will of course influence the characteristics of the incoming air.

3.5. Outlet facilities

The **objective** for an outlet is to let the moist and/or warm air out from the building.

For **underpressure systems** the outlet should also create the necessary underpressure in the building in order to suck outside air into the building through the inlets (or other openings).

Secondary objectives are to exhaust manure gases at certain locations and to maintain a desired air flow direction in the manure handling system.

The outlet has in most cases a negligible effect on air distribution and air movements compared to the influence of the inlets, walls, temperature differences and animals.

In a mechanical underpressure system two types of fans are used :

* axial flow fans
* centrifugal fans

For ventilation in animal houses the propeller fan (axial flow) is the most used type for economic reasons (investment and running costs) and technical reasons (low demand on total pressure performance and air flow direction). See Fig. 11.

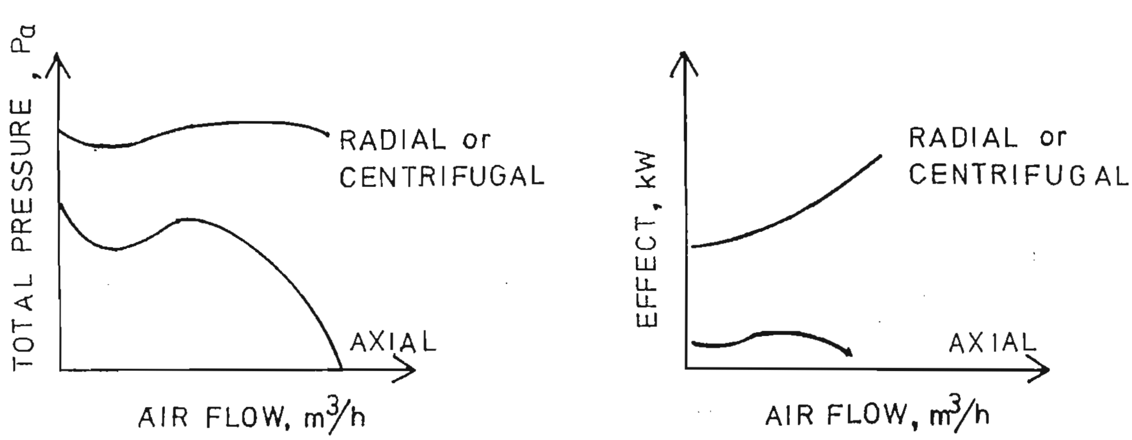


Fig. 11 : Characteristics for radial/centrifugal and axial fans respectively

3.5.1. Choice of fans

For a certain room there should be calculated both minimum and maximum air flows. Type of fan(s) (size, diameter, speed, rpm) should then be chosen to meet these demands also considering ability to regulate the air flow of the ventilation system between the extremes. **Capacity** of tested fans varies between 2 000 - 15000 m^3/h at 20 Pa static pressure difference depending on the size (400 to 630 mm diameter) and the speed (900 - 1 400 rpm).

For exhaust systems fan performance should be tested including all equipment necessary (duct, louver rain/wind shield, etc.). The system air flow performance is then found by assuming the pressure drop for the inlets (Fig. 12).

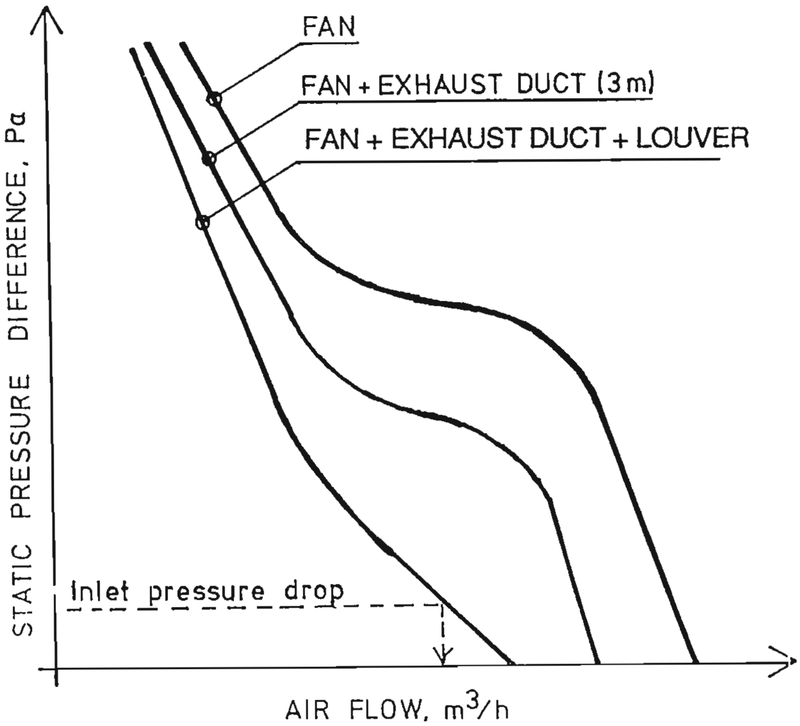


Fig. 12 : Fan performance with different equipment and total system performance with given inlet pressure drop.

Efficiency, i.e. air flow per electrical energy input, m^3/kWh, is also reported in the test reports. Older fans give about 10 000 m^3/kWh, fans tested 1988 have obtained 1000 000 m^3/kWh.

Noise from the fan should either be below 65 dBA (tested fans vary from 63 to 87 dBA) or a silencer should be provided. A simple silencer below the fan reduces noise by 6 - 9 dBA. A silencer integrated in a duct reduces noise 15 - 20 dBA (NILSSON, 1977).

3.5.2. Location of outlet

Generally the location of the outlet has no significant effect on temperature, humidity and air movements.

3.5.2.1. Influence on air or movements

Calculations of the suck effect e.g. air velocity towards the outlet at different distances shows that a very limited space in a barn would be affected when considering natural air movements of 0.25 m/s and air velocity in the outlet between 5 - 10 m/s. Placing the outlet in a corner gives the outlet a better opportunity to affect air movements.

The suck effect from an outlet can be coordinated with other air movements in the barn and thus create a steady air flow pattern at a longer distance. This possibility should be used to prevent manure gases reaching the animals.

3.5.2.2. Influence on temperature and RH distribution

Extreme differences of outlet location can affect temperature distribution. SÄLLVIK (1979) found a 4 - 5 % temperature difference between inside and outside with the outlet located in the respective gables in a 60 m long gestation barn.

PEDERSEN (1974) could not find any difference in temperature between cross sections in a growing finishing barn when comparing a section under the exhausting fan with a section 5 m away.

Regarding the moisture removal and the heat balance during the winter, LILLENG (1961) found no differences between outlets in the ceiling and outlets in the manure tank under the floor.

3.5.2.3. Influence on CO_2 distribution

Even though CO_2 is 52 % heavier than air there has not been found any difference in CO_2 concentrations between floor and ceiling in ventilated barns. Outlet location under the slatted floor or in the gutter has not changed the CO_2 distribution compared to outlets in the ceiling (SÄLLVIK, 1973).

There is no increase of CO_2 towards the outlet. The relative distribution of CO_2 is equal all along a barn when inlets and animals are evenly distributed.

3.5.2.4. Influence on liquid manure gases distribution

Animals and humans are severely hurt by hydrogen sulphide (H_2S) and ammonia (NH_3). Both gases are produced by microbiological processes in the manure. H_2S only under anaerobic conditions. H_2S is released only when liquid manure is agitated in some way. NH_3 is released even from open surfaces with manure.

To avoid manure gases reaching animals or humans the manure system ought to be kept at underpressure compared to the surroundings. To create such an underpressure, the total area of air of penetrable floor (mostly slatted floor) and other openings between the manure system and the surroundings may be restricted in relation to available air flow.

The exhaust system should therefore be designed to create an underpressure in the manure system. Several techniques could be used depending on the conditions (type of floor, liquid or solid manure handling, new or existing buildings, etc.).

* Exhausting of at least the minimum ventilation via the dunging-out channel, provided that there is a gas trap between the barn and the manure storage and/or provided that there are means for air actually to leave the barn (Fig. 13).

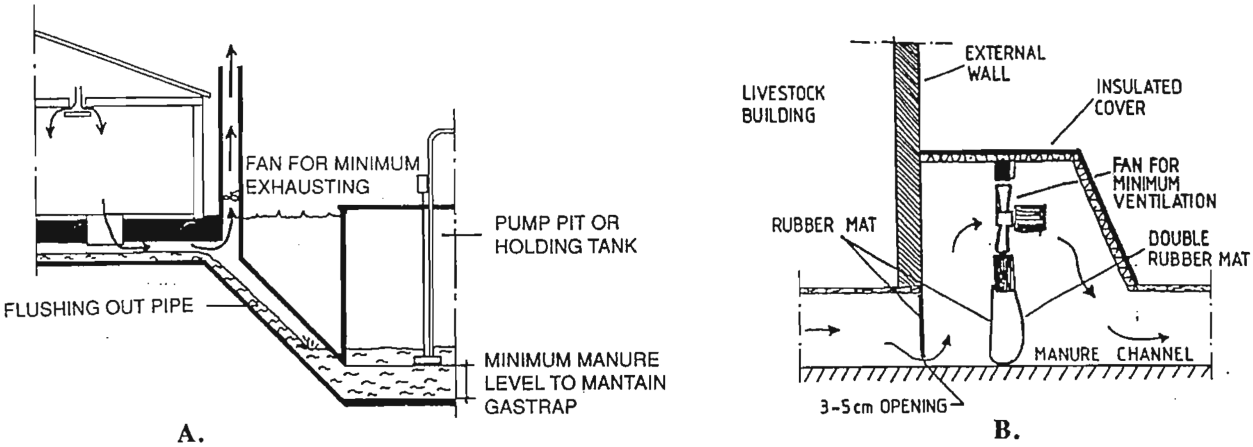


Fig. 13 : Examples of how the problem of exhausting via the manure system to avoid manure gases can be solved. A. Liquid manure. B. Solid manure, mechanical scrapers

* Locating the bottom at least 0.6 m below the slatted floor to reduce effect of air movements in the slots on gas release from the bottom of the channel. The bigger the free distance underneath the slatted floor the longer the distance you get an ensured downward air stream caused by exhausting from the end of the channel (Fig. 14).

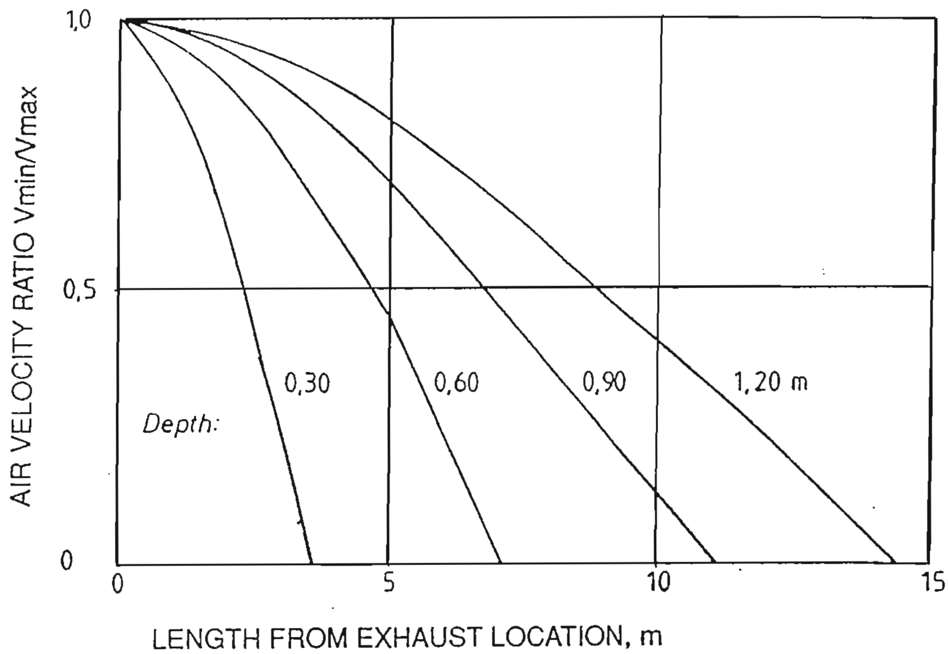


Fig. 14 : Calculated variation in air velocity through a slatted floor (20 % open area) when air is exhausted through the manure channel. GUSTAFSSON, 1988.

* Placing the cross channel with exhausting equipment in the centre of the building to cooperate with the natural air movements in the manure channels which are from the gables towards the center.

* Locating outlets beneath the slatted floor in order to increase the efficiency of manure gas exhausting (Fig. 15). As a rule of thumb a theoretical velocity of 0.2 m/s through the slats is the minimum velocity for creating a steady air flow downward into the slats.

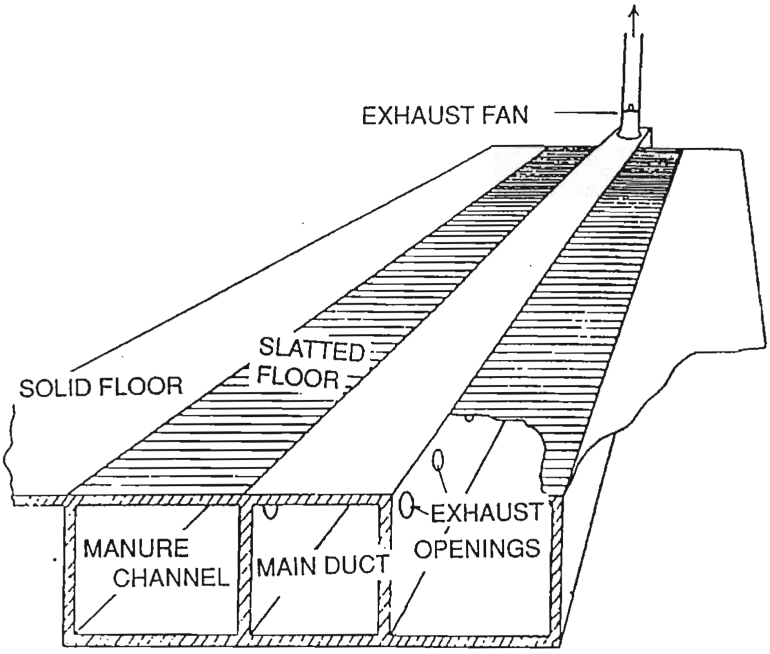


Fig. 15 : Principle of design of a floor exhausting system

The ratio between total area of exhaust openings (A_s) and the cross section area (A_k) influences how evenly air will be exhausted along the channel (Fig. 16). The $(A_s)/(A_k)$ ratio is normally recommended to be 0.8. The distribution of the outlets shall be in accordance to the locations where manure gases entering the animal environment. These locations are found by using cold smoke examination of the air currents in the manure system.

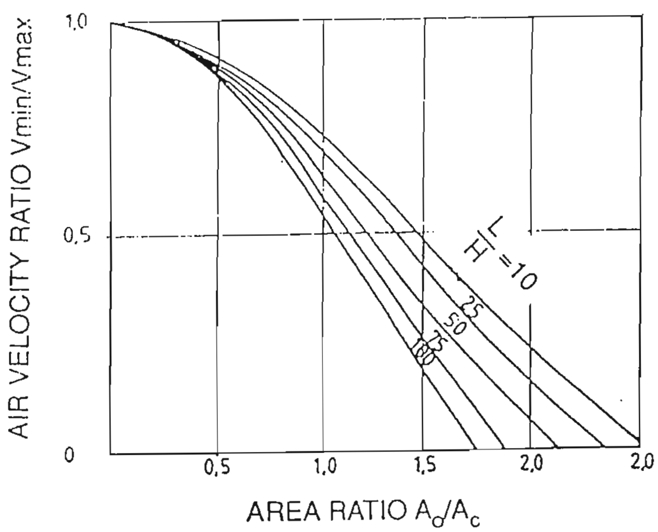


Fig. 16 : Calculated variation in air velocity in the exhaust openings of an exhaust air channel according to area ratio A_o/A_c and length-height ratio L/H. Sum of exhaust openings and cross section. GUSTAFSSON, 1988.

* Preventing manure gases below a slatted floor reaching the animals can be done by especially designed exhausting hoods, installed as a part of the slatted floor (Fig. 17). The location of such a hood must be based on investigations of the natural air currents beneath the slatted floor since these currents must be stopped by the rubber mat and then directed to the exhaust opening at the top of the hood. Since air velocities have been measured to 0.2 ± 0.05 m/s, the exhaust capacity of the local manure gas exhauster must be designed to meet the resulting air flow, 300 - 600 m^3/h.

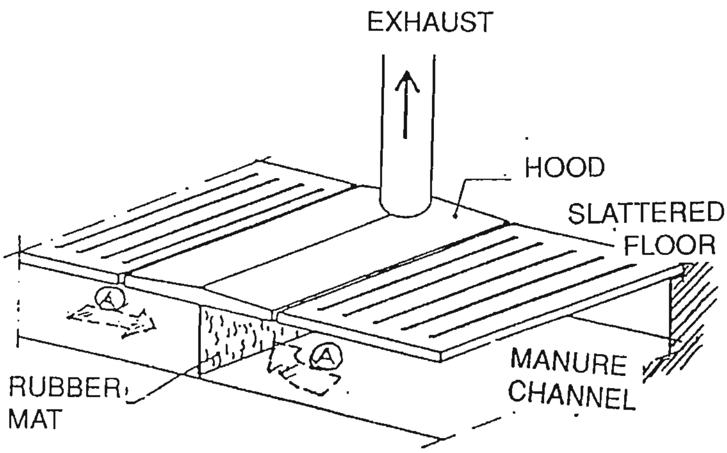


Fig. 17 : Example of equipment, "manure gas exhauster", designed for gutters covered with slatted floor. A = natural air currents. Rubber mat acts as an obstacle to the air currents and a guide towards the exhaust.

3.6. References

ALBRIGHT, L.D. 1972. The low speed non-isothermal wall jet with application to ventilation. Dissertation, Cornell NY.

BECHER, P. 1952. Varme og ventilation 3. Ventilation og tabeller. Köpenhamn.

DEBRUYCKERE, M. 1967. De actuele problematiek omtrent de klimaatregeling van leghennenhokken. Rijksfaculteit der Landbouwwetenschappen. Dissertation, Gent.

GRAEE, T. 1989. Breathing ceiling constructions : properties, experiences and possibilities in Norway. Porenlüftung, 1988. 1. Internationales Symposium über Porenlüftung. Österreichische Kuratorium für Landtechnik, Landtechnisches Schriftenreihe no. 156, Wien.

GUSTAFSSON, G. 1988. Air- and heat balances in animal houses. Swedish University of Agricultural Sciences, Department of Farm Buildings, Report no. 59, Lund.

HAHN, G.L., NYGAARD, A., SIMENSEN, E. 1983. Towards establishing rational criteria for selection and design of livestock environments, ASAE-paper 83 - 4517.

HAHN, G.L., SÄLLVIK, K. 1979. Livestock compensatory performance influences on shelter criteria ; a discussion of concepts. ASAE-paper 79 - 4016, St-Joseph, Michigan.

NILSSON, C. 1984. Buller i stallar. (Noise in animal houses). Swedish University of Agricultural Sciences, Department of Farm Buildings, Aktuellt no. 284, Uppsala.

PEDERSEN, S. 1976. Inblaesningsåbninger og luftordelning i staldrum. Statens Byggeforskningsinstitut, Notat nr. 66 Köpenhamn.

PORENFLÜFTUNG, 1988. Internationales Symposium über Porenlüftung. Wien. Österr. Kuratorium für Landtechnik. 1989.

STERRENBURG, P. en VAN OUWERKERK, E.N.J. 1986. Rekenmodel voor de bepaling van de thermische behaaglijkheidszone van varkens (BEZOVA). Rapport 78, Instituut voor Mechanisatie, Arbeid en Gebouwen IMAG, Wageningen.

SÄLLVIK, K. 1971. Tilluftsdon - inflytande på temperatur och lufthastighet kring uppbundna kreatur. Landtbrukshögskolan, Inst. för lantbrukets byggnadsteknik, Specialmeddelande nr. 5. Lund (Air inlets - influence on temperature and velocity around to thered cattle).

SÄLLVIK, K., NILSSON, C. & NIMMERMARK, S. 1978. Economic heat insulation and ventilation in animal houses. ASAE, Paper no. 4508. St. Joseph, Michigan.

SÄLLVIK, K. 1979. Principles for mechanical exhaust ventilation systems in animal houses. Swedish University of Agricultural Sciences, Department of Farm Buildings, Report no. 8, Lund.

SÄLLVIK, K., WALLBERG, K. 1984. The effects of air velocity and temperature on the behaviour and growth of pigs. Journal of Agricultural Engineering Res. pp. 305 - 312, Vol. 30.

WHITE, G.M. WALKER, I.N. & BLACK, T.C. 1969. Air distribution from a horizontal slot in vertical triangular ducts. ASAE, Paper no. 67-944. St. Joseph, Michigan.

3.7. Calculation of natural ventilation

INTRODUCTION

Natural ventilation offers a valuable alternative to mechanical ventilation in many cases. Obviously the natural ventilation system should be well designed. In order to support the engineer, this report provides a summary of the basic equations of natural ventilation.

3.7.1. By stack effect (winter - see figure 1)

Pressure produced by thermal buoyancy (p_v) can be balanced to the sum of dynamic pressure of flowing air and pressure drops by airflow resistance (p_r).

$$p_v = p_r \quad [\mathrm{Pa}] \tag{1}$$

p_v = produced pressure

p_r = dynamic pressure plus pressure losses by resistance

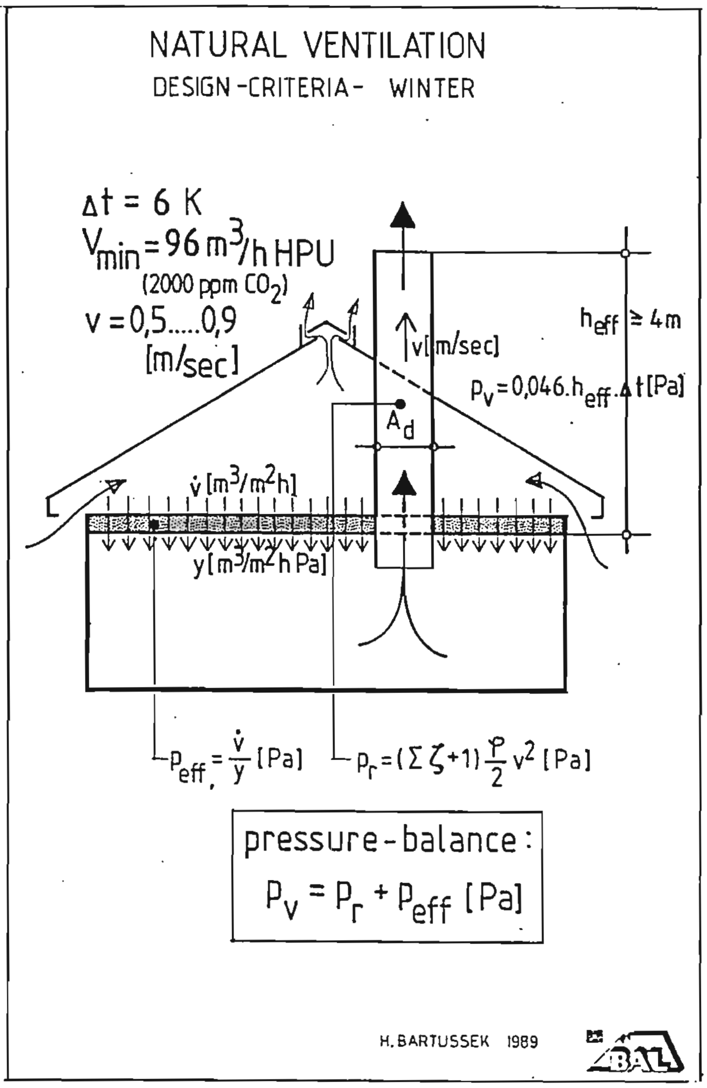


Fig. 1 : Natural ventilation - Design criteria - winter

$$p_v = 9.81\ (\rho_o - \rho_i)\ h_{eff} \quad [Pa] \tag{2}$$

ρ = specific weight of air [kg/m^3]

o = outside

i = inside

h_{eff} = effective height [m]

for practical purposes $9.81\ (\rho_o - \rho_i)$ can be estimated as $0.046 * \Delta t$ (Δt temperature difference inside - outside [K])

with this assumption (2) becomes :

$$p_v \approx 0.046 * h_{eff} * \Delta t \quad [Pa] \tag{3}$$

$$p_r = \Sigma_{in}^{og} \left[(0.03 \frac{1}{d} + \Sigma\xi + 1) \frac{\rho_{in.og}}{2} v^2 \right] \quad [Pa] \tag{4}$$

1	=	length of duct or chimney [m]
d	=	diameter of duct [m]
ξ	=	resistance factor of obstacles [-]
v	=	air velocity of flowing air [m/sec]
in	=	incoming air
og	=	outgoing air

With (3) and (4) combined one can calculate v for a certain design situation and with v and d the ventilation rate V [m^3/h] according (5) :

$$V = \frac{d^2}{4} * \pi * v * 3600 \quad [m^3/h] \tag{5}$$

3.7.2. By wind effect (summer - see fig. 2)

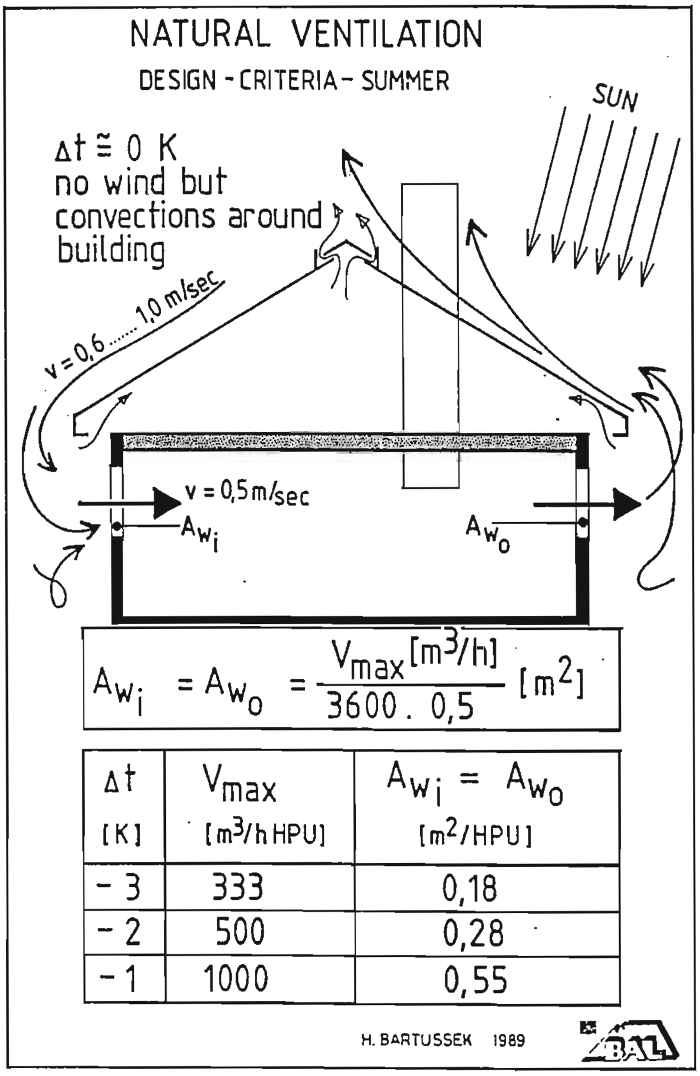


Fig. 2 : Natural ventilation - Design criteria - summer

$$A = \frac{V_{max}}{3600 * v_w} \qquad (6)$$

with

A = inlet area = outlet area (m^2)

V_{max} = summer ventilation rate (m^3/h)

v_w = assumed windspeed (m/s)

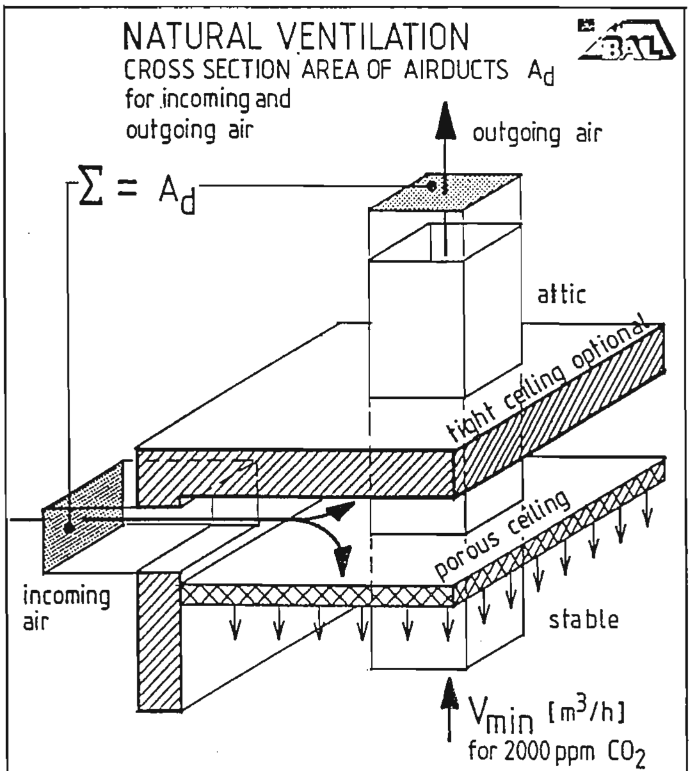
Remark : v = 0.5 m/s will cover most cases 100 % of the time

3.7.3. Practical recommendations

- Vertical chimneys should be as high as possible. In any case they should end above the roof (0.5 m above ridge ; 2 m above eaves). They must be insulated (k-value = 0.5 W/m^2K).

- All air ducts for incoming and outgoing air should be as straight as possible. Obstacles which cause air flow resistance should be avoided or minimized (grids, sharp edged corners or inlets, covers against rain, etc.).

- The sum of cross section area (A_d) of air ducts for natural ventilation by thermal buoyancy, that is all air ducts, respectively to and from the room to be ventilated, should be at least that indicated in figure 3 :



DESIGN CRITERIA		hight of chimney h_{eff} [m]		
		5	10	15
air velocity within duct v [m/sec]		0,5	0,7	0,9
cross section area of airducts A_d	A_d [m^2]	$\frac{V_{min}}{1800}$	$\frac{V_{min}}{2500}$	$\frac{V_{min}}{3200}$
	A_{dHPU} [m^2/HPU]	0,053	0,038	0,030

H. BARTUSSEK 1989

Fig. 3 : Natural ventilation - Cross section area of airducts for incoming and outgoing air

Remark : A_{dHPU} = A_d per HPU (Heat Producing Unit = 1.000 W total heat at 20 °C) (m^2)

- If the cross section area in figure 3 is reduced more than 30 % in cattle houses or more than 15 % in pig houses, or if effective chimney height is less than 4 m, sufficient function of natural ventilation is no longer assured. Mechanical support of air flow should be considered. But at the coldest time of the year, or with wind sucking on the head of chimneys, natural ventilation will still work.

3.7.4. Breathing ceilings as air inlets with thermal buoyancy (see fig.1)

A_c = area of breathing ceiling (m^2)

y = air penetrability coefficient (m^3/m^2 h Pa)

v = specific volume flow through the ceiling (m^3/m^2 h)

$$v = \frac{V}{A_c} \tag{7}$$

V = ventilation volume flow of the whole room (m^3/h)

$$v = y * \Delta p_{eff} \tag{8}$$

Δp_{eff} = effective pressure difference at the surfaces of the ceiling (Pa)

$$\Delta p_{eff} = \frac{V}{A_c * y} \tag{9}$$

Volume of flow through the ceiling is directly proportional to pressure difference, because one can assume laminar current within the porous material of ceiling.

With breathing ceilings equation (4) has to be adapted :

$$p_r = \Sigma_{in}^{og} \left[(0.03 \frac{1}{d} + \Sigma \xi + 1) \frac{\rho_{in.og}}{2} v^2 \right] + \frac{V}{A_c * y} \tag{10}$$

in, og as explained in equation (4)

Combined with (1) and (2) gives (in m^3/m^2 h Pa) :

$$y = \frac{V}{A_c \left\{ 9.81 \, (\rho_o - \rho_i) \, h_{eff} \, \Sigma_{in}^{og} \left[(0.03 \frac{1}{d} + \Sigma \xi + 1) \frac{\rho_{in.og}}{2} v^2 \right] \right\}} \tag{11}$$

3.7.4.1. Assumptions and criteria for design

V = ventilation volume flow = V_{min} for $m_k = 2\ l/m^3$; this gives v according to A_d from Figure 3. $\rho_o - \rho_i$ according $\Delta t = 6$ K (or less) ; A_c, l, d, ξ according to room and air duct design. It is also possible to choose y (practically between 10 and 80 m^3/m^2 h Pa) and calculate A_c.

- Flaps in chimneys for thermal buoyancy must never close completely. With all in - all out management of fattening animals, the rest area should be 2 % of A_d (empirical value to compensate the multiple effect of maximum temperature difference plus sucking of wind at head of chimney). If live mass in the stable is always about the same, rest area should be 5 % of A_d.

- With an inside temperature of design temperature (lower critical temp.) plus 5 °C throttle flaps should open completely. At temperatures in between this band throttle opening is proportional to $t_i + 5$ K.

- At upper critical temperature minus 5 °C window flaps for summer ventilation start opening. They should open completely within a proportional band of 5 K.

3.7.5. Calculation example for natural ventilation

Calculation for practical purposes in stables with breathing ceilings under a freely ventilated attic (open ridge and eaves, no air flow resistance for incoming air except ceiling) using the recommendation of Figure 3 is done as follows :

Example

Stable for 200 fattening pigs (all in all out) ; floor area = ceiling area = 200 m^2 ; chimney height = 5 m >>> v = 0.5 m/sec.acc. table : in figure 3. V_{min} is calculated for pigs at the end of fattening period (100 kg), producing 42 l CO_2/h per pig :

$$V_{min} = \frac{200 * 42}{2.0 - 0.3} = 4941\ m^3/h\ ;\ 0.3 \approx 0.35\ \frac{\rho_i}{\rho_o}\quad (0.35 = m_k \text{ outside})$$

$$A_d = \frac{4\ 941}{3600 * 0.5} = 2.75\ m^2\ ;\ \text{chosen 4 chimneys with } A_d' = \frac{A_d}{4} = 0.688\ m^2$$

>>> d = 0.93 m

$l = 5$ m ; $\Sigma\xi$	:	inlet into chimney :	$\xi \approx 0.5$
		outlet (roof)	$\xi \approx 0.5$
		throttle open	$\xi \approx 0.5$
			$\Sigma\xi \approx 1.5$

p_v according to (3) : $p_v \approx 0.046 * 5 * 6 = 1.38$ Pa ; ($\Delta t = 6$ K)

$A_c = 200\ m^2$; $\frac{\rho_i}{2} \approx 0.6$; all these assumptions in equation (11) gives :

$$y = \frac{4941}{200\left[1.38 - (0.03 * \frac{5}{0.93} + 1.5 + 1) * 0.6 * 0.5^2\right]} = 25\ m^3/m^2\ hPa$$

Window flaps for summer ventilation on two opposite wall sides according to equation (6) ; V_{max} for 200 pigs (80 kg) : $V_{max} = 22\,074\ m^3/h = 6.13\ m^3/sec$;

$$Aw_i = Aw_o = \frac{6.13}{0.5} = 12.3\ m^2$$

3.7.6. Experience (see Figure 4)

Figure 4 shows the result of an investigation in 30 naturally ventilated animal houses in Austria. The specific cross section area per HPU of the chimneys found is shown in relation to the effective chimney height. The line represents this paper's recommendation for design. The mean of all values of A_d ($\overline{X}$) lies close to the line, with large deviations though. 94 % of the farmers evaluated the climate in their animal houses subjectively as very good or good.

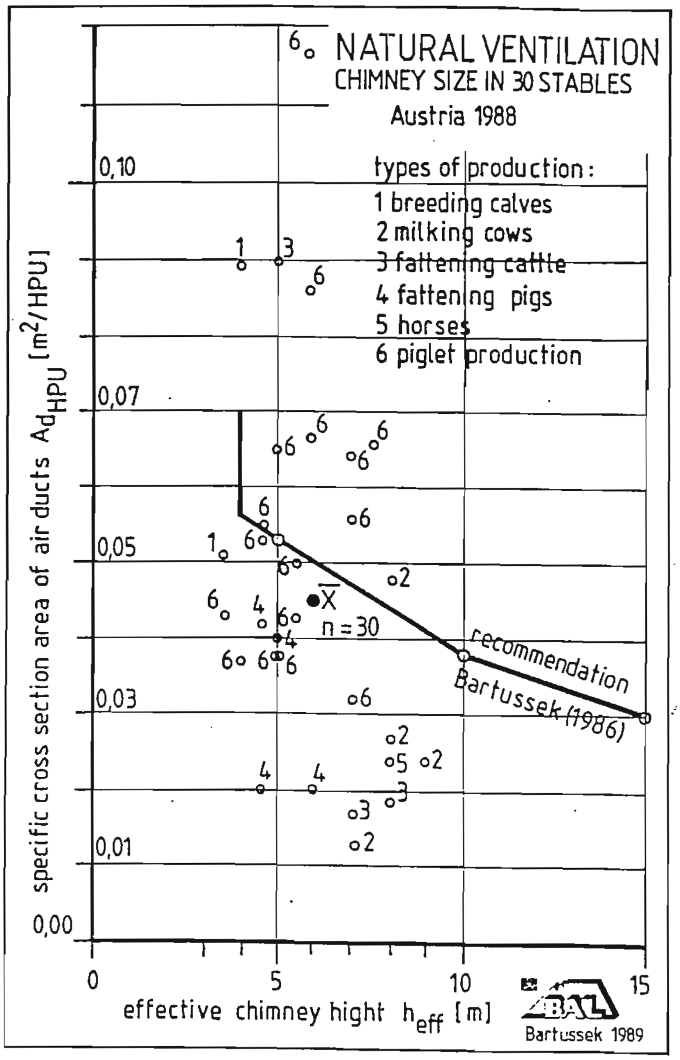


Fig. 4 : Natural ventilation - Chimney size in 30 stables

3.7.7. References

BARTUSSEK, H. Stallklima und Lüftung - Wichtige Umweltfaktoren in der Rinder- und Schweineproduktion. Beratungsservice Landtechnik und Bauwesen, Folge 10, Der Förderungsdienst, 34, 1986, Heft 11, Beilage.

BARTUSSEK, H. Einige Ergebnisse einer Fragebogenerhebung über die Porenlüftung in Österreich. Porenlüftung 1988, Landtechnische Schriftenreihe Heft 156, ÖKL, Wien, 1989, S. 30 - 42.

BARTUSSEK, H. Dimensionierung von Schwerkraft-Porenlüftungssystemen einschließlich Sommerlüftung : Anleitung für die Praxis. Porenlüftung 1988, Landtechnische Schriftenreihe Heft 156, ÖKL, Wien, 1989, S. 127 - 137.

BARTUSSEK, H. Natural ventilation by thermal buoyancy and by outside convections : practical application of natural ventilation systems with chimneys and breathing ceilings. In : DODD, A. and GRACE, P.M. (Ed.) : Agricultural Engineering, Volume 2 : Agricultural Buildings, A.A. BALKEMA, Rotterdam, Brookfield, 1989, p. 1299 - 1304.

BRUCE, J. Automatically controlled natural ventilation (ACNV). Farm Building Progress, 1979, 58, p. 1 - 2.

Responsible member LILLENG H.	*Chapter 4* **MINIMUM VENTILATION PROBLEMS IN CONFINED ANIMAL ROOMS IN COLD AREAS**

4.1. Approach

The needs for minimum ventilation depend on outside climate and required air temperature, relative humidity and limitation of gas concentrations in the animal room. The minimum ventilation can be calculated according to the recommendation given in the CIGR-report "Climatization of animal houses" (1984) (1).

In practice many farmers have problems managing the minimum ventilation at the right level. The problems can be due to insufficient insulation, untight constructions, useless ventilation equipment, wrong use of the equipment or a combination of many factors. A well-insulated and airtight building is of vital importance for a good climate.

This paper deals with the climate problems which can arise in untight buildings, and how to solve it.

4.2. How airtight are our animal houses ?

Few measurements are carried out to bring to light the airtightness of different types of animal rooms.

Obviously the variations are very large, and normally rooms for pigs and chickens are tighter than cow barns.

Different methods can be used to figure out the amount of uncontrolled fresh air entering cracks in the building. The infiltration can be determined by use of a tracer gas or by directly measurement with the pressurization method, figure 1.

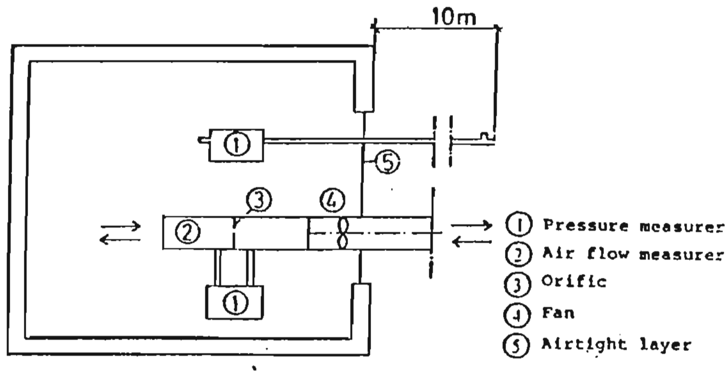


Fig. 1 : Pressurization method for air infiltration measurement

The underpressure method with underpressure was used to determine the leakages in 8 Norwegian cow barns.

Figure 2 shows that the infiltration was 2 - 4 air changes/h at 10 - 15 Pa negative pressure. Examination indicated that doors and windows are especially untight. Table 1 gives measurements for 3 doors, normal type.

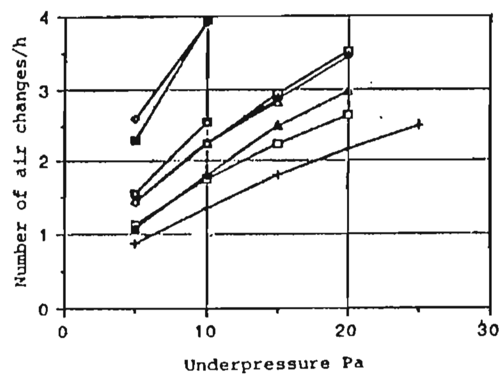


Fig. 2 : Air infiltration in the cow stables on 8 farms

Table 1 : Infiltration caused by untight doors

Door no.	Size m^2	Infiltration, $m^3/h.m^2$ underpressure			
		5 Pa	10 Pa	20 Pa	40 Pa
1	5.7	18.3	25.6	36.0	50.0
2	2.0	20.7	30.1	44.0	63.0
3	2.0	15.7	24.3	35.5	52.0

If 10 - 15 Pa negative pressure the infiltration through door no. 1 will be about 150 m^3/h. In cold weather this is enough to supplies 3 cows or 30 pigs 20 kg or 3 000 day-old chickens with fresh air.

4.3. The factors causing infiltration

Fresh air can be forced through the cracks, by wind, buoyancy, or a negative pressure in the room caused by the ventilating fans.

4.3.1. Infiltration caused by wind

On a free field the wind pressure maximum against high buildings can be

$$\Delta p = \frac{\rho \cdot v^2}{2}$$

where

Δp = dynamic pressure, Pa
ρ = air density, kg/m^3
v = air velocity, m/s

At 0 °C the formula gives

v = 1 m/s, Δp = 0.64 Pa
v = 5 m/s, Δp = 16.10 Pa
v = 10 m/s, Δp = 64.50 Pa

Normally the buildings are not very high, and other buildings and trees damp the wind. Therefore the wind pressure is somewhat lower than expressed by the formula.

Moreover, the wind direction against the walls is important for the pressure. Fig. 3 shows variations along buildings' facades in model tests.

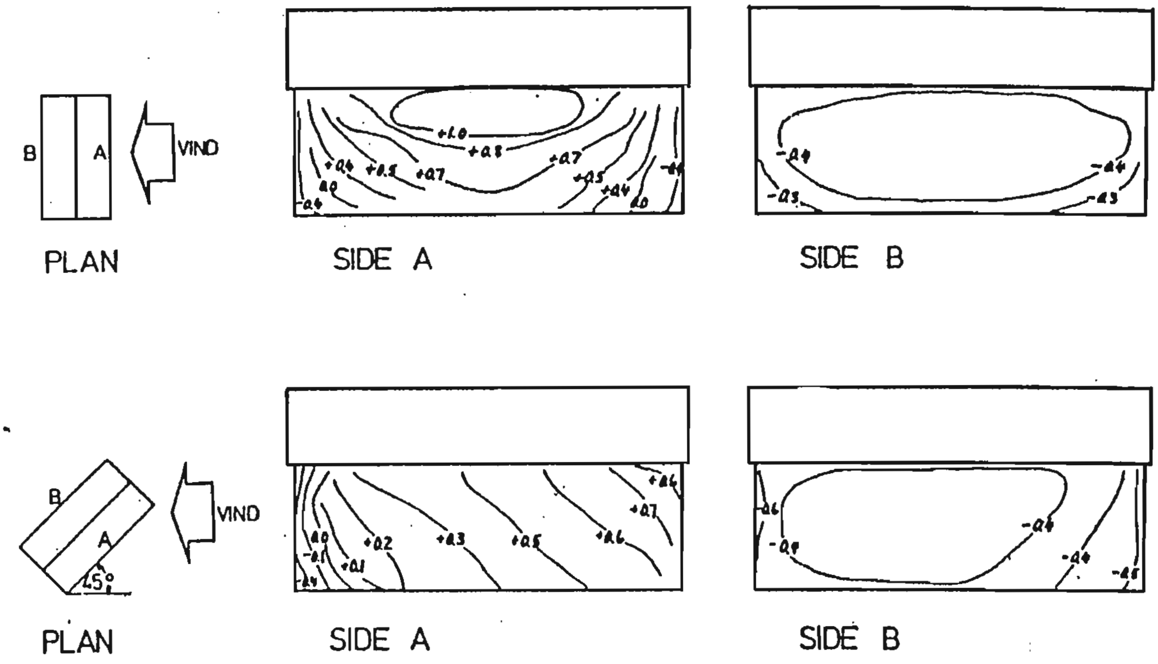


Fig. 3 : Wind pressure against buildings, relative values

The wind forces fresh air into the room through all cracks in places where the outside pressure is higher than the inside.

4.3.2. Infiltration caused by thermal buoyancy

Big temperature differences between inside and outside the building give pressure differences and air infiltration if construction is not airtight. At one point between the floor and the ceiling in an unventilated room, the inside and outside pressures are equal (the neutral zone). The level of the neutral zone depends on the location of any cracks. The cold outside air is heavier than the inside air. Therefore, inside the room we have a higher pressure above and a lower pressure underneath the neutral zone than outside.

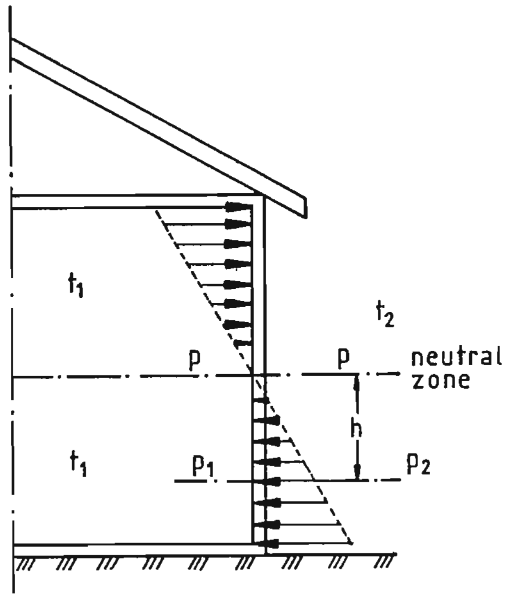


Fig. 4 : Air pressure in an unventilated room, $t_1 > t_2$

The negative pressure in Pa h m underneath the neutral zone will be

$$\Delta p = gh\,(\rho_2 - \rho_1)\ \text{Pa}$$

ρ = air density, kg/m^3

If, as an example, the neutral zone is located 1.5 m above the floor level and inside and outside temperature are 15 °C and -20 °C, the negative pressure near the floor will be $\Delta p = 1.5 * 9.81\ (1.395 - 1.218) = 2.6$ Pa. If the room is higher or there is a cellar underneath a slatted floor, the pressure differences can be higher. Temperature differences cause pressure differences all over the building, and fresh air will be forced into the room through the cracks located below the neutral zone. Very often the largest leakages are located near the floor, underneath the door blades.

4.3.3. Infiltration caused by the outlet fans

Fans blowing used air out of the rooms cause a negative pressure all over the room, if the construction is not very leaky. The negative pressure causes fresh air flow into the room both through the cracks and the planned inlets.

4.4. The disadvantages of leaky buildings

4.4.1. Too high a minimum ventilation rate

Large cracks in the construction may cause a higher ventilation rate than required in cold and windy weather. The consequence is too low temperature in rooms without a heating system. In heated rooms the energy consumption and costs will turn too high and the relative humidity too low if the temperature is maintained on required level.

4.4.2. Draught on the animals

Animal located near big cracks in the construction will be exposed to draught. Draughts from leaky doors are very common. Openings between the door blade and the floor often give cold air streams along the floor. This gives a negative effect on the heat balance for young and small animals.

In untight rooms the negative pressure normally is very low. Therefore the velocity of the fresh air entering the inlets also drops, and the cold air falls quickly downwards and may hit the animals.

4.4.3. Temperature variations

In untight rooms both horizontal and vertical temperature variations are common. Fig. 5 shows temperature observations along the floor inside an untight door.

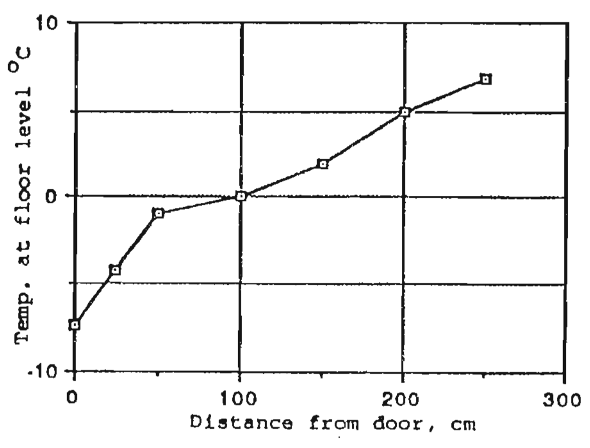


Fig. 5 : The air temperature at floor level at different distances from an untight door at an outside temperature of -9 °C

Table 2 shows observations of vertical temperature variations in a special untight cow barn with openings in the floor. There was a dung cellar underneath.

Table 2 : The temperature conditions in a leaky cow barn

Outside temp. °C	Dung cellar temp. °C	Place in barn	Behind the cows, °C	On the stands °C	In the young °C
		1. Underneath the ceiling	15.8	17.3	16.5
-9	-4	2. 1 m above level	14.2	16.7	15.7
		3. Near the floor level	6.3	9.3	11.3
		Difference 1 - 3	9.5	8.0	5.2

4.4.4. Condensation and ice

Where cold air enters the cracks, the construction will be cooled and condensation and ice may arise on the inside surface. The mixing of cold and warm, moist air near the inside surfaces may give the same effect.

4.5. The way to reduce or eliminate the problems

Infiltration can be reduced or eliminated by sealing the cracks. As mentioned, doors and windows normally are the most leaky constructions.

Leaky door blades can be sealed by means of plastic liner fixed to the door inside. Waterproof plywood can also be used.

Openings between the door blade and the frame can be sealed by means of rubber weather strips, and by using a door lock to force the door blade against the frame.

Openings between the door blade and the floor can be sealed by means of a sweeping weather strips.

Windows also can be sealed, using weather strips and good locks.

If you have a well insulated and airtight animal room, your climatization equipment gets a chance to work properly.

4.6. References

CIGR, 1984. Report of working group on climatization in animal houses.

COLLIVER, D.G., WALTON, L.R. & PARKER, B.F. (1982). Wind speed on residential energy usage. ASAE Winter Meeting, Chicago, Ill.

COSTELLO, Th.A., GARCIA, A. & MEADOR, N.F. (1987). Simplified air infiltration measurement. ASAE Winter Meeting, Chicago, Ill.

FENTON, F.C. & OTIS, C.K. (1941). The design of barns to withstand wind loads. Kansas State College Bulletin.

LILLENG, H., LØKEN, K.A. & SØRBY, K.M. (1987). Luftlekkasjer og temperaturforhold i husdyrrom. IBT-rapport nr. 206, N-1432, Ås-NLH.

LØKEN, K.A. (1986). Lufttetthet av dører i husdyrrom. IBT-rapport nr. 207, N-1432, Ås-NLH.

LØKEN, K.A. (1987). Metoder til måling av luftlekkasjer gjennom dører, vinduer og andre bygningskonstruckjoner. IBT-rapport nr. 208, N-1432, Ås-NLH.

MOUSLEY, L.J., FRYER, J.T. & PIKE, K.S. (1975). Infiltration of a buildings by wind. NIAE Sect Note SN/FB/9/3020.

MOUSLEY, L.J., FRYER, J.T. (1985). Air leakage of broiler houses. Farm Build. & Engineering. 2 (2).

Responsible member RIST M.	***Chapter 5*** **ENERGY RECUPERATION IN ANIMAL HOUSES**

INTRODUCTION

The energy content of feed is partly transformed into heat by the animals' metabolism, so a primary heat source is available in every animal house. Most of the time, the heat production of the animals allows of an optimal temperature control in the house. With low outside temperature, however, the animals' heat production won't meet the heat losses through the structure and by ventilation, so artificial heating will be needed. The necessary heat capacity can be limited by using heat-exchangers or heat-pumps. Both recovery systems can also be used to heat the farmer's house or to heat the washing water.

5.1. Heat exchangers in animal houses

5.1.1. Type of heat exchangers

Different types of air-air heat exchangers can be used in animal houses. The need for air ducts will depend on the way the heat exchanger system is implemented within the whole of the building. In this regard, a control position of the heat exchanger offers an advantage.

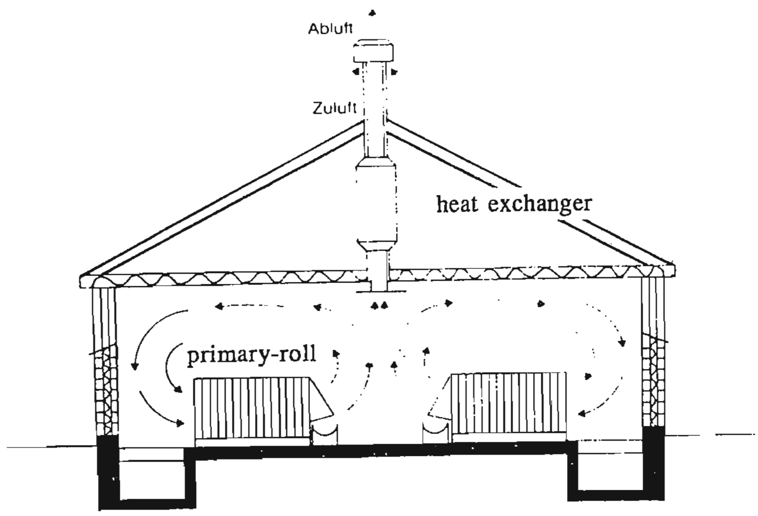


Fig. 1 : Central heat exchanger

Distinction is drawn between

- folded heat exchangers made of chromium or aluminium with plastic corrugation cover (see Fig. 2)
- foil heat exchanger made of plastic sheets
- plate heat exchangers made of glass or aluminium
- pipe heat exchangers made of glass or plastic (see Fig. 3)

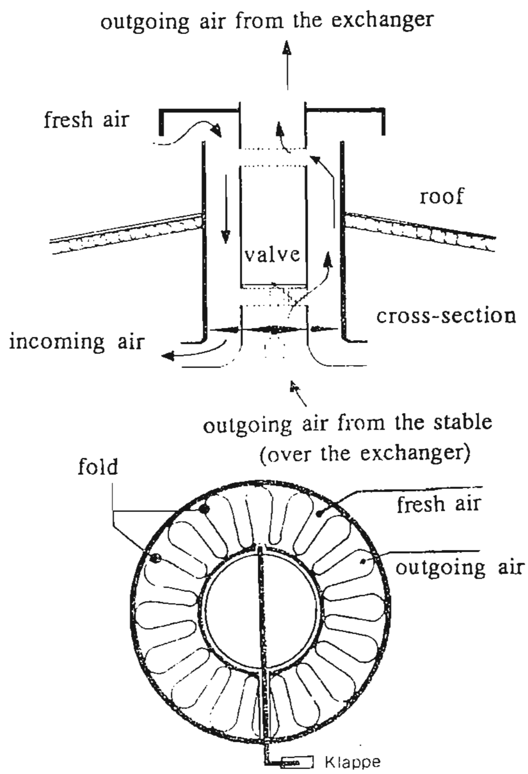


Fig. 2 : Folded heat exchanger

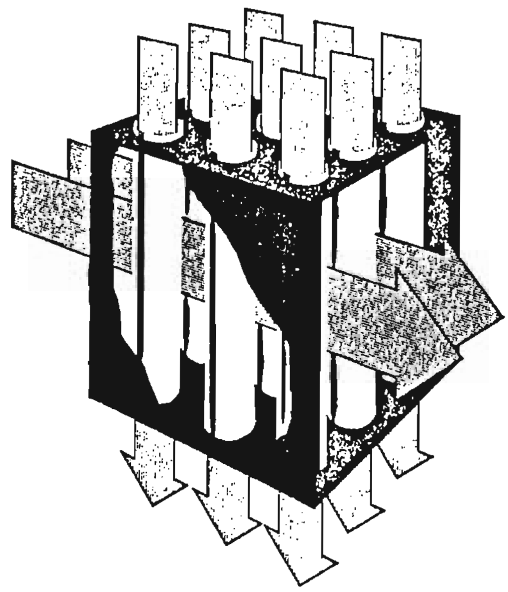


Fig. 3 : Scheme of pipe exchanger

5.1.2. Formulae for air - air heat exchangers

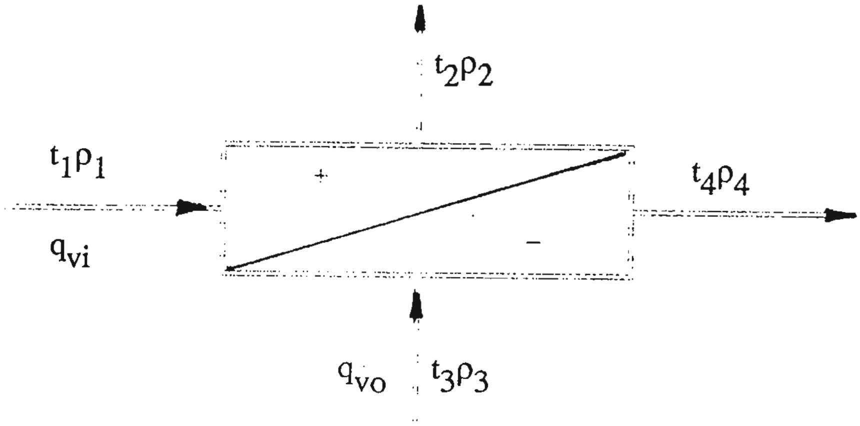


Fig. 4 : Schematic representation of heat exchanger

Efficiency

- with condensation

$$\eta_w = \frac{q_{vi}\,(h_2 - h_1)}{q_{vo}\,(h_3 - h_{1s})}$$

- when no condensation

$$\eta_d = \frac{q_{vi}\,(t_2 - t_1)}{q_{vo}\,(t_3 - t_1)}$$

with

η_d = dry efficiency (2)

q_{vi} = mass flow of incoming air (kg air/s)

q_{vo} = mass flow of outgoing air (kg air/s)

t_1 = temperature of incoming air before the exchanger (°C)

t_2 = temperature of incoming air after the exchanger (°C)

t_3 = temperature of the outgoing air before the exchanger (°C)

t_4 = temperature of the incoming air after the exchanger (°C)

h_w = wet efficiency (-)

h_1 = enthalpy of the incoming air before the exchanger (kJ/kg air)

h_2 = enthalpy of the incoming air after the exchanger (kJ/kg air)

h_3 = enthalpy of the outgoing air before the exchanger (kJ/kg air)

h_{1s} = enthalpy of air at temperature t1 and at saturation (kJ/kg air)

Temperature effectiveness in %

$$\eta_w = \frac{t_2 - t_1}{t_3 - t_1}$$

Heating capacity of the exchanger

$$\Phi_{ex} = (q_{mv})_{min} \,.\, 1010 \,.\, \eta_e \,.\, (t_3 - t_1)$$

with

F_{ex} = heating capacity of the exchanger (W)

$(q_{mv})_{min}$ = required minimum ventilation rate (kg air/s)

η_e = temperature effectiveness

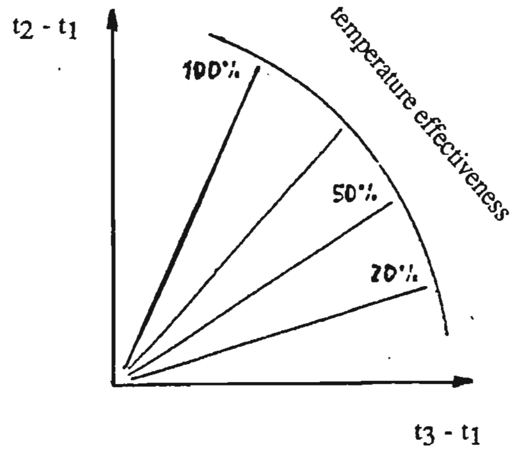


Fig. 5 : Determination of temperature effectiveness and energy-efficiency in practice

5.1.3. Technical view of heat-exchanger

The temperature effectiveness and energy efficiency are determined by the temperature and enthalpy of the incoming and outgoing air (Fig. 5). The actual values will depend on the relative surfaces of the heat exchanger compared to the air flow (which is about 1 $m^2/100(m^3/h)$), on the distance between the separating plates (about 1 - 2 cm), and on the purity of the plates (no dust disposal). In order to keep the plates clean, (automatic) sprinkling at regular intervals (once every 12 h) is necessary.

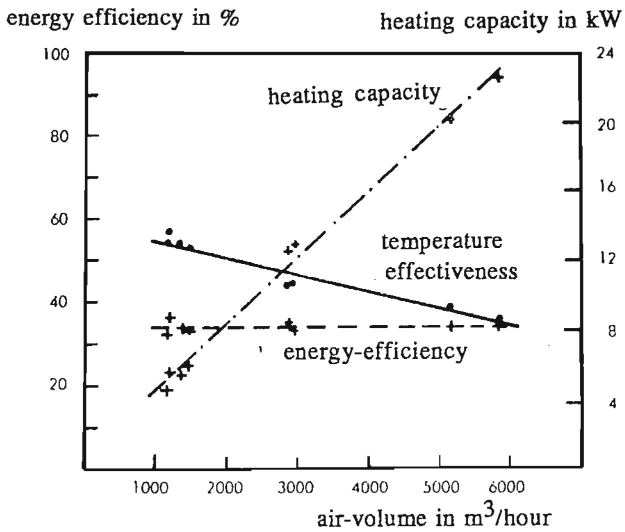


Fig. 6 : Thermal features of a folded heat exchanger with an exchanger surface of 40 m^2, at 15 K temperature difference between incoming and outgoing air.

The energy efficiency of the exchanger remains almost constant at a 34 % level, while the heating capacity increases to 23.5 kW at maximum air flow. The temperature effectiveness decreases at the same time from 55 % to 35 %. The initial temperature difference was 15 K. As this temperature difference between inside and outside air increases, a higher heating capacity will be reached (see Fig. 7).

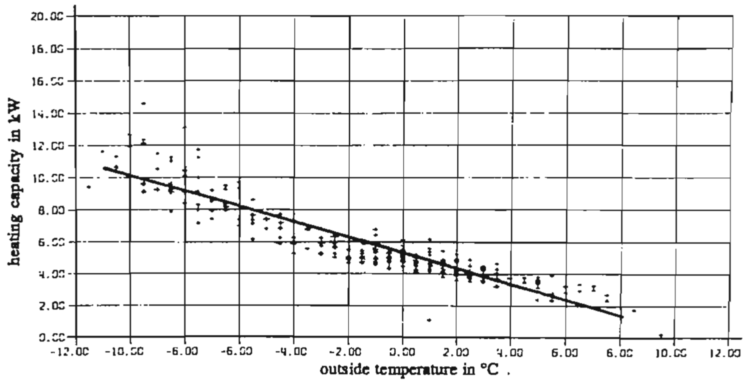


Fig. 7 : Increase in heating capacity with increasing temperature difference

Wherever the inside temperature in the house drops below 0 °C, the condensation within the exchanger may freeze. At the temperature effectiveness of 50 % and a house temperature of 10 °C, freezing will occur as soon as the outside temperature is below -10 °C. In that case the air flow should be stopped, until the ice is melted.

5.2. Heat pump

5.2.1. Formulae

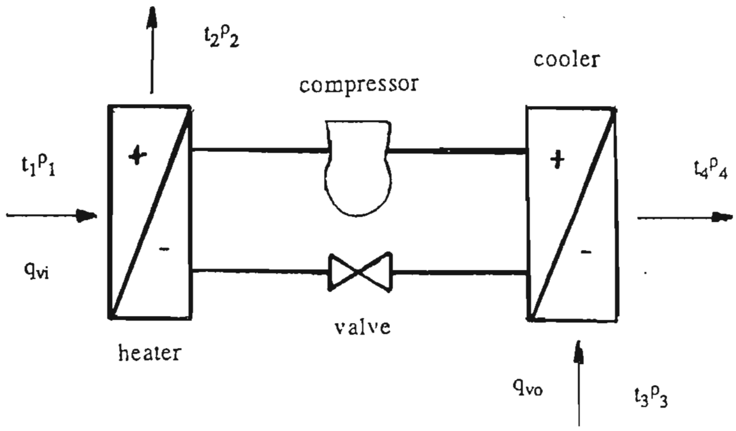


Fig. 8 : Schematic representation of a heat pump

Coefficient of performance

$$COP = \frac{\Phi_{hp}}{\Phi_{el}}$$

$$COP = \eta_{ex} \frac{T_1 + \Delta T}{(T_1 + \Delta T) - (T_2 + \Delta T)}$$

with

COP = coefficient of performance (-)
Φ_p = heat production of the heat pump
Φ_{el} = electricity consumption of the heat pump
η_{ex} = energy efficiency (-)
0.4 for small heat pumps
0.6 for large heat pumps
T1 = absolute temperature of incoming fluid (K)
T2 = absolute temperature (K)
ΔT = temperature difference between heated or cooled fluid and the heat exchanger (K)
$\Delta T = 5$ for water-water heatpump
$\Delta T = 10$ for air-water heatpump

Remark : in practice COP = 2 to 2.5 for air - air heat pump
COP = 3 to 3.5 water - water heat pump

5.2.2. The use of heat pumps in agriculture

Many heat pump application in agriculture use the outgoing air of the animal house as a heat source. By cooling this air a considerable amount of energy can be recovered. However, where this cooling takes place within the house only the heat surplus of the house and the latent heat from dehumidification can be used. In order to heat the farmer's house and the washing water an air - water heat pump is used. In general, three types of heat pumps can be distinguished :

- air - water heat pumps (as mentioned before, see Fig. 9) in a compact design.

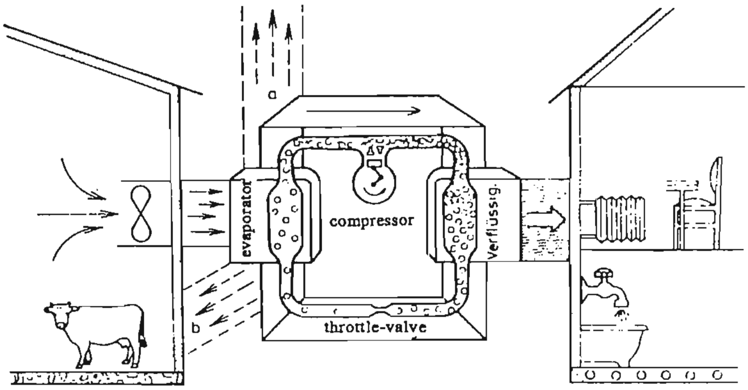
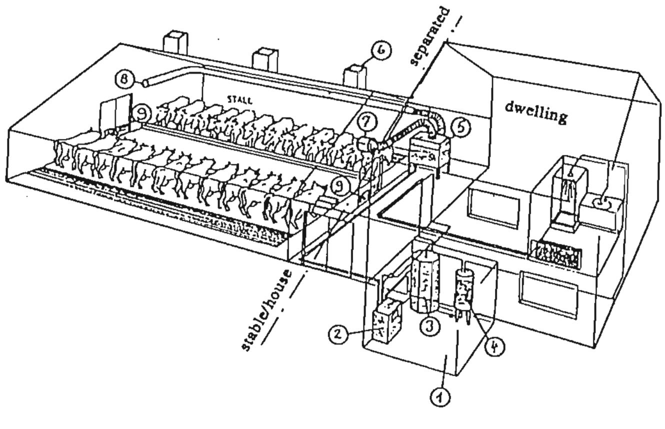


Fig. 9 : Air - water heat pump
a = outgoing air system
b = recirculation air system

- air - air heat pumps with a remote condensor installed in the farmer's house. The absorbers and the compressor part are in the animal house (Fig. 10).



1. heating system
2. heatpump
3. hot water reservoir
4. warm water reservoir
5. evaporator
6. chimney
7. air inlet with filter (warm air)
8. air inlet (cool air)
9. air inlet in the door (fresh air)

Fig. 10 : Air - water heat pump with remote condensor. Air ducts for incoming and outgoing air.

- water - water heat pumps. In this case the condensor and the compressor are placed in the farmer's house, and the absorber in the animals house (Fig. 11).

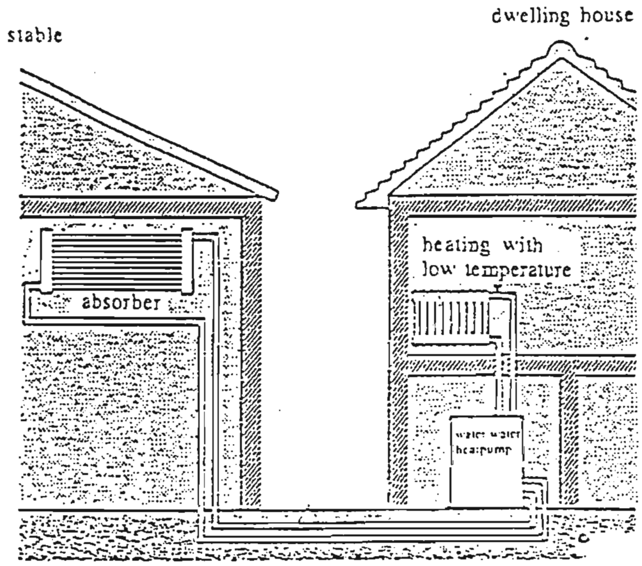
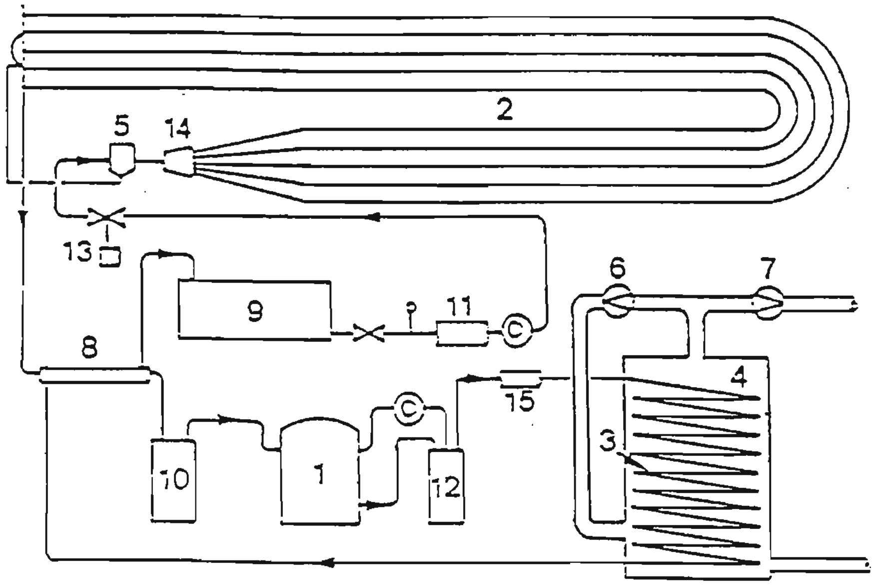


Fig. 11 : Water - water heat pump with absorber in the animal house

Cooling inside the animal house requires a great deal of care with regard to the ventilation pattern in the house. Cooling the outgoing air does not show this inconvenience.

A purely agricultural application of heat pumps is seen with floor heating (particularly for piglets). In this case the heat source can be the manure cellar (Fig. 12).



1. compressor
2. evaporator
3. condensor
4. buffer
5. expansion valve
6. pump (condensor circuit)
7. pump (heating circuit)
8. heat exchanger
9. refrigerant reservoir
10. liquid seperator
11. filter-dryer
12. oil separator
13. magnetic valve
14. distributor
15. silencer

Fig. 12 : Scheme of heat pump application for floor heating for piglets, with evaporation in the manure cellar

5.2.3. Design criteria

The heat demand varies considerably with the season. The maximum heating capacity will be needed only in deep winter. In summertime the heating need will be entirely determined by the heat demand for warming up the washing water, so heat pumps should be designed in view of a maximal output and an optimal COP in winter conditions. Fig. 13 shows the weekly progress of the heat pump performance throughout a winter period.

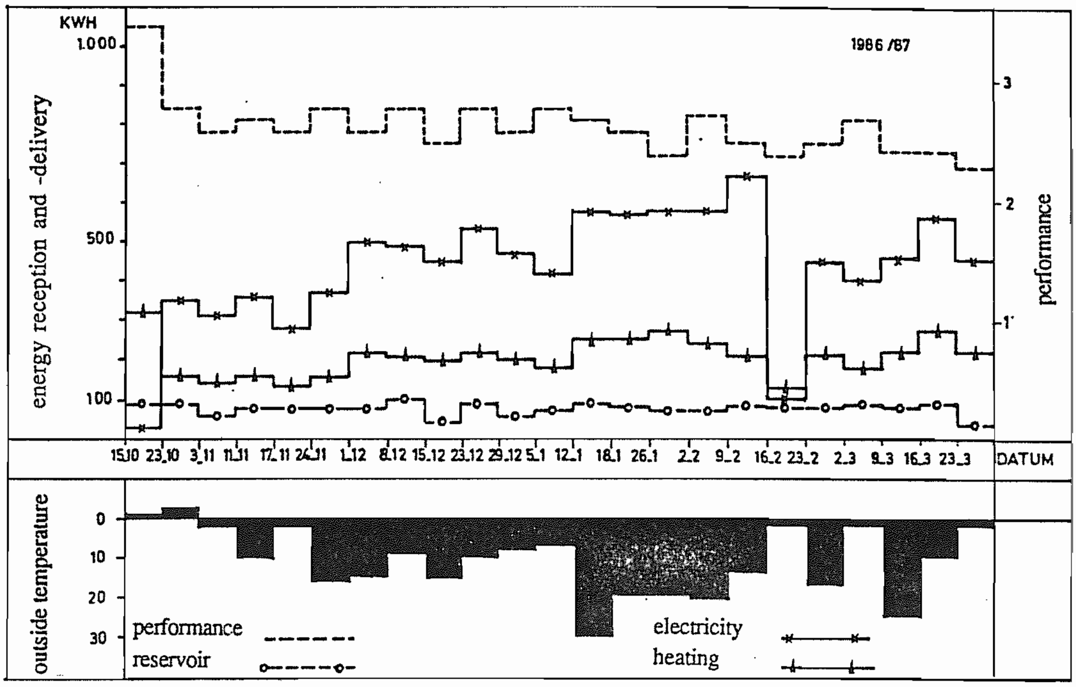


Fig. 13 : Evolution of the heat pump performance throughout a winter period

Responsible member

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(6.4.1.1. and 6.4.1.2.)

Chapter 6

COOLING IN ANIMAL HOUSES

INTRODUCTION

Homeothermic animals can sustain a very small increase in internal body temperature (0.5 to 1 K) for only a restricted period of time (a few hours). In order to avoid heat stress (and in the end death) these animals initiate body heat losses as soon as the environment is getting too warm. Since the animal's potential to do so is limited, the climatization of the house should be designed to guard against the worst conditions. This means that cooling should be applied when necessary.

6.1. The animal's potential to increase heat losses

Animals exposed to a warm environment show three types of reactions. They are simultaneously activated by the homeostatic system of the animal.

- increase in skin temperature (increasing the sensible heat losses) by peripheral vasodilation. Obviously this measure is only efficient when room temperature is lower than internal body temperature.

- increase in evaporation (increasing the latent heat losses) through the skin and in the respiratory tracts. Unless the animal has sweat glands, the evaporation through the skin can not be physiologically controlled and depends entirely on the water vapour pressure of the air. Evaporation through the respiratory tracts can be increased by accelerating respiratory rhythm (up to 10 - 15 times normal frequency, in certain species) and at the same time the depth of respiration is reduced in order to match the oxygen demand of the animal. Obviously, heat loss through evaporation is encouraged by low R.H. Thus hot but dry climates are better tolerated than hot and humid ones. Animals which rely almost exclusively on respiratory evaporation are less sensitive to high R.H. than those animals which dissipate latent heat by sweating.

- the reduction of feed intake, in order to restrict metabolic heat production. This is the most negative reaction from an economic point of view, as production is reduced.

6.2. The upper critical temperature

Physiologists define the upper critical temperature as the temperature above which the animals' heat production is increased by the muscular exertion of the cardiopulmonary system. However the temperature above which the animals reduce feed intake (and consequently show decreased production) is of greater economic importance. For mating stock there might be a different critical temperature indicated by a drop in reproductive performances.

This "economic critical temperature" is not easy to determine as it depends on many factors. The factors relative to the animal are :

- the genetic characteristics of the animal
- acclimatization ; animals which have the opportunity to get used to high temperatures gradually, show an increase ability to disperse sensible heat through greater peripheral vaso-dilation.
- feed intake capacity. Animals which consume large amounts of food produce much metabolic heat and have lower upper critical temperature. So fast growers are more sensitive to heat stress.

The environmental factors which influence the economic critical temperature are :

- duration of the increased temperature. High temperatures are better tolerated if they are of brief duration.
- air speed. When the air is at a lower temperature than the skin temperature, its speed enhances the sensible heat loss.
- the R.H. of the air. At low R.H. values the animal tolerates higher temperatures.

Last but not least the economic critical temperature will be determined by the investment and learning costs of the climatization system.

6.3. The animal's requirements

Bearing in mind the difficulty of an exact determination of the upper critical temperature, a rough indication for different types of animals can be attempted.

- For dairy cows with a good level of production the quantity of milk produced begins to drop considerably at temperatures of 25 °C and over at 50 % R.H. Various indices have been suggested. One of these is the THI (Temperature Humidity Index) proposed by BENG, SHANKLIN and JOHNSON (similar to the Discomfort Index, DI, proposed by CARGILL and STEWARD) is calculated as :

$$THI = 0.72\ (t_d + t_w) + 40.6$$

where

t_d = dry bulb temperature and
t_w = wet bulb temperature

THI values lower or equal to 75 would permit maximum milk production ; values of 80 would provoke a 20 % reduction in production and values of 90 would give rise to 40 - 45 % reduction. Analogous values could also be valid for beef cattle.

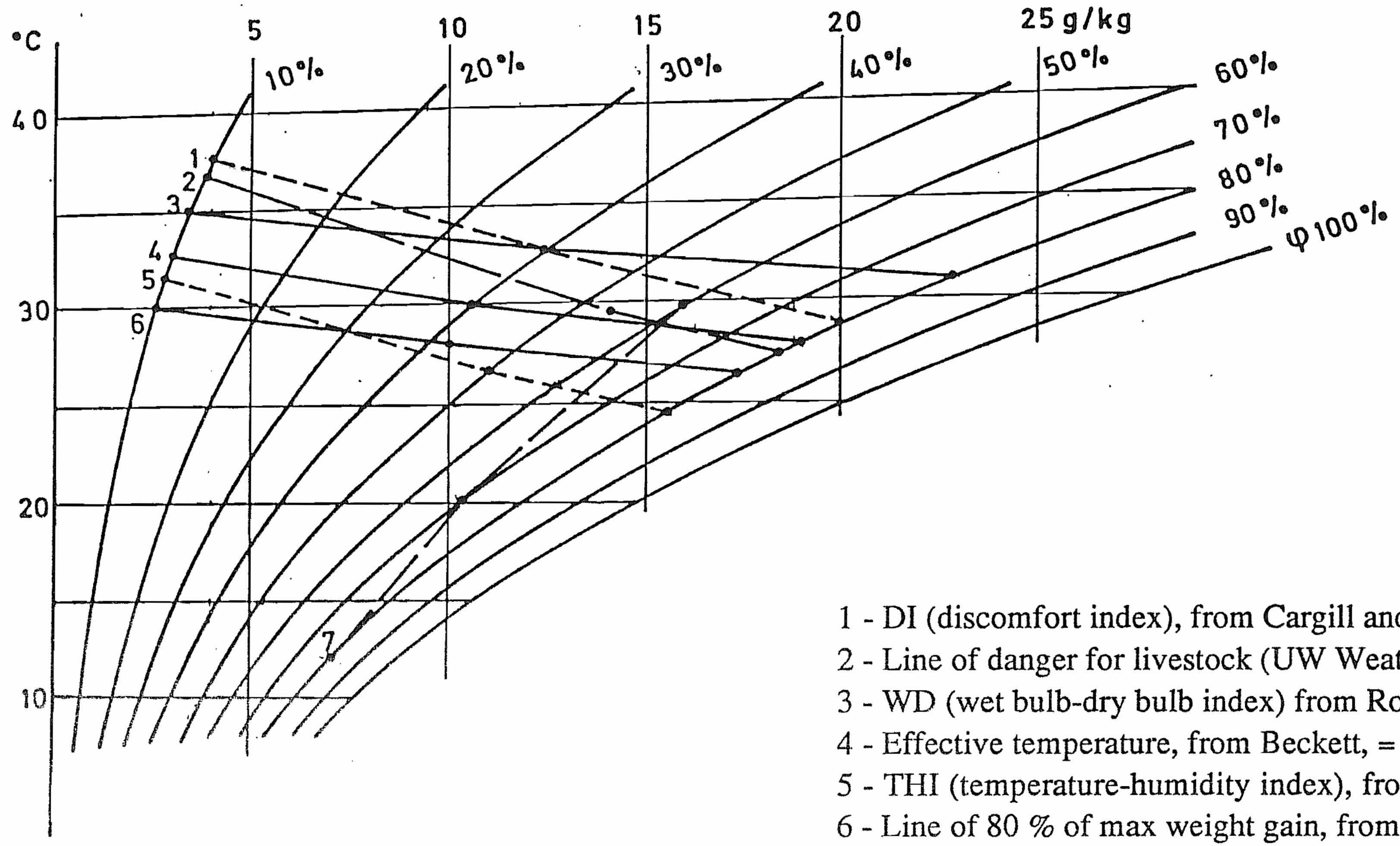
- The dry bulb temperature is particularly important for pigs, as they don't sweat. The formula of BRUCE gives the upper critical temperature according to body weight, feed intake and breeding conditions (whether the animals are single or in groups). The critical temperature is calculated as about 26 - 27 °C for adult fattening pigs.

- Sheep can tolerate even hot temperatures rather well, but if they are well-fed, temperatures should not exceed 30 °C.

- Adult chickens or chickens over 5 weeks old can tolerate temperatures up to 27 °C without problems, independent of the R.H. level. At higher temperatures (up to 32 °C), damage is modest only when R.H. values are low.

- Rabbits are particularly sensitive to high temperatures. At temperatures over 22 °C they begin to reduce food intake ; at above 25 °C they show signs of suffering and at temperatures above 30 °C, pregnant female are subject to increased miscarriages.

A summary of the literature review is given by means of a Mollier-diagram (Fig. 1).



1 - DI (discomfort index), from Cargill and Steward = 80
2 - Line of danger for livestock (UW Weather Bureau)
3 - WD (wet bulb-dry bulb index) from Roller and Goldman
4 - Effective temperature, from Beckett, = 32.2 °C
5 - THI (temperature-humidity index), from Barrada
6 - Line of 80 % of max weight gain, from Morrison
7 - CIGR recommandation

————	hogs
- - - - - - - -	cows
— — — —	livestock, in general

Fig. 1 : Critical temperatures for livestock : summary of the literature.

6.4. Means of cooling in animal houses

Various means of temperature control in the house can be considered :

a) Free cooling or ventilation

This is the easiest and most widely-used technique. It can be calculated as indicated in Chapter 2.6.5. of the 1st report of the CIGR Working Group. It does not permit real cooling, but, at best, it limits the increase of the inside temperature to a couple of degrees above the outside temperature.

b) Mechanical cooling (refrigeration or heat pump)

Using this method, inside temperature can be considerably lower than outside temperature. In this case, ventilation is kept at (winter) minimum level. It is generally considered too costly for use in animal housing.

c) Adiabatic cooling through water evaporation inside the house or (more frequently) in the incoming air flow.

This is an economical method, permitting a temperature reduction of 8 to 10 K, but at the expense of a considerable increase in R.H.

d) Cooling the ventilation air by means of a heat exchanger in a colder environment. Usually the ground is chosen as the cooling medium, making use of underground pipes. In this case the R.H. of the incoming air increases due to the effect of the temperature reduction.

e) Direct cooling of the animals using showers.

6.4.1. Evaporative cooling

6.4.1.1. The design criterion

If the heat exchange between animal and environment depended on the total heat content of the air (i.e. on wet bulb temperature), evaporative cooling would be completely ineffective. Indeed only latent heat dispersion is proportional to wet bulb temperature, whereas sensible heat exchange depends on dry bulb temperature. For this reason adiabatic moisturizing of the air can, at most, allow the animals to eliminate total heat. The problem consists in establishing the best combination of dry bulb temperature and R.H., along the isoenthalpic line. There is little agreement among researchers on this as shown in Fig. 1.

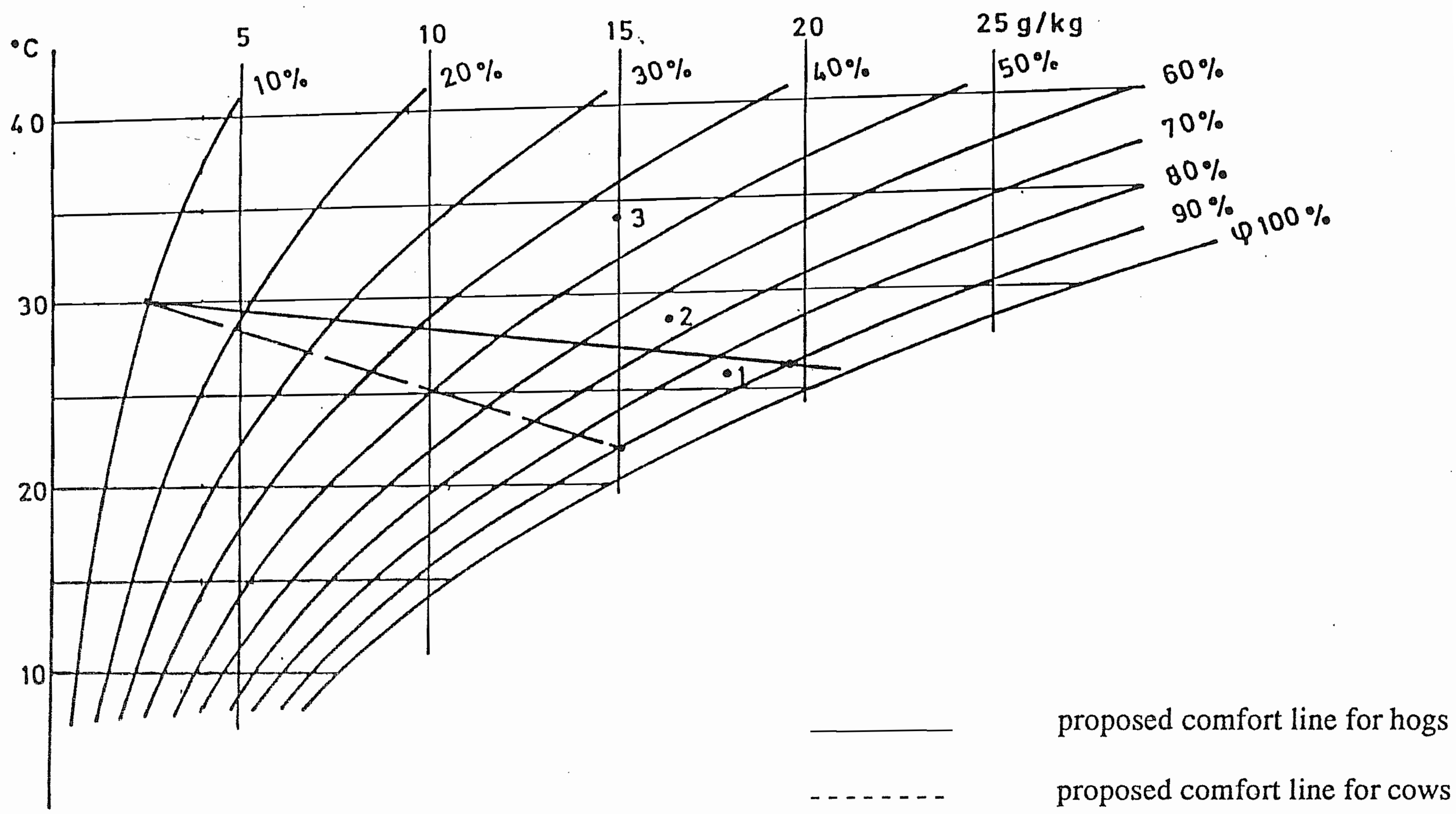


Fig. 2 : Isoenthalpic criteria for livestock : working group proposals.

Based on this review the working group agreed on two lines (one for pigs and one for cows) as a first approximation (Fig. 2).

These lines which establish a linear relationship between temperature and absolute humidity involve that the maximum allowable enthapy can be expressed as a second order function of the inside temperature. Since the coefficient of t_i^2 is small, a good approximation can be made by

$$(h_i)_{max} = \alpha + \beta\, t_i$$

with

$(h_i)_{max}$ = max. tolerable energy of the inside air (kJ/kg)
t_i = inside temperature (°C)
α, β = constants for which we propose

	α	β
pigs	321	-9.44
cows	124	-2.90

6.4.1.2. Calculation method

The second problem is to determine the air flow rate when evaporative cooling is used.

From the heat balance the following equation can be derived :

$$t_i = t_x + \frac{\Phi_s}{\Phi_{mv}}$$

The basic equations for moist air allow one to write :

$$m_{mx} = \varphi_x\, m_{ms}$$
$$h_o = 1.01\, t_o + 2\,500\, m_{mo}$$
$$t_x = (h_o - 2\,500\, m_{mx}) / 1.01$$

with

φ_x = relative humidity of the incoming air after the evaporative cooler (-)

h_o = enthalpy of the outside air (kJ/kg dry air)

Φ_s = sensible heat production in the house (kW)

q_{mv} = ventilation rate (kg dry air/s)

t_o = outside temperature (°C)

m_{mo} = outside absolute humidity (kg H_2O/kg dry air)

m_{mx} = absolute humidity of saturated air with an enthalpy of h_o according to following graph (kg H_2O/kg dry air)

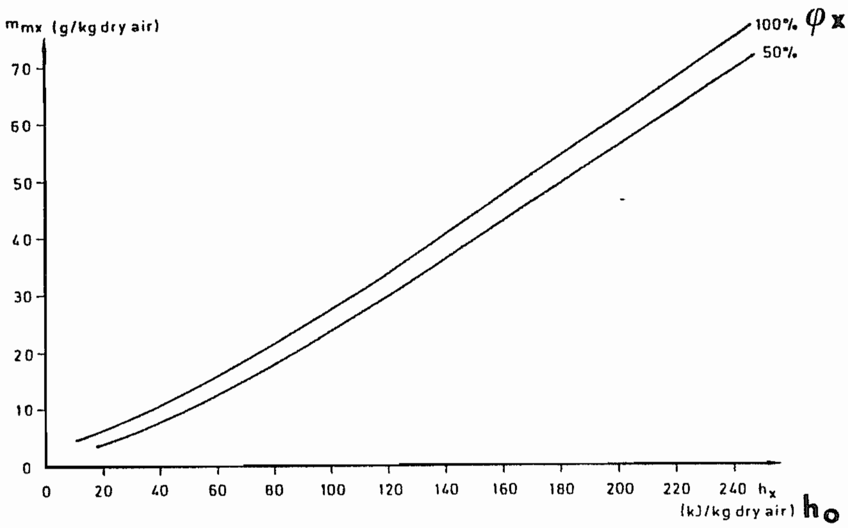


Fig. 3 : Absolute humidity of the air leaving the evaporative cooler as a function of the outside enthalpy

This figure can be established since the process within the evaporative cooler is assumed to be adiabatic. So the maximum tolerable enthalpy to aim for is known as soon as the outside temperature conditions are given, or in mathematical terms :

$$(h_i)_{max} = \alpha + \beta \left[\frac{h_o - 2\,500\, m_{mx}}{1.01} + \frac{\Phi_s}{q_{mv}} \right]$$

On the other hand, the enthalpy of the house can be mathematically expressed as

$$(h_i)_{max} = (h_i)_x \frac{\Phi_s + 2\,500\, q_{mp}}{q_{mv}}$$

Because $(h_i)_x = h_o$, both expressions can be combined in order to obtain the ventilation rate. So

$$q_{mv} = \frac{(1 - \alpha)\,\Phi_s \;+\; 2\,500\, q_{mp}}{\beta + \alpha \left[\frac{h_o - 2\,500\, m_{mx}}{1.01} \right] - h_o}$$

Example : Pig house with 100 dry sows

$t_o = 32\ °C$ $\varphi = 40\ \%$ $m_{mo} = 12.1 \frac{g\ H_2O}{kg\ dry\ air}$ $h_o = 63 \frac{kJ}{kg\ dry\ air}$

$\Phi_s = 10.7$ kW $q_{mp} = 7.2$ g H_2O/s

From the given graph with $\varphi_x = 90\ \%$ one can find $m_{mx} = 18.5$ g H_2O/kg dry air.

So

$$q_{mv} = \frac{(1 + 9.44)\ 10.7 + 2.5 * 7.2}{321 - 9.44 \left[\frac{63 - 2.5 * 15.8}{1.01} \right] - 63}$$

$$q_{mv} = 3.38 \text{ kg dry air/s} = 10\ 150\ m^3/h$$

It should be clear that in the expression for the ventilation rate whenever, the nominator is zero or negative there is no solution to the problem. In other words the maximum allowable enthalpy can't be achieved by means of evaporative cooling. In that case a compromise must be found between the dimensions of the evaporative cooling system and the amount $(h_i)_{max}$ is exceeded. This will also be necessary when the nominator is very small. The alternative to all this is looking for another cooling system.

6.4.1.3. Evaporative cooling can be obtained in various ways

a) Cooling pads (nowadays made of cellulose or plastic). These are designed to be bulky because air speed through the pad should not exceed 1.25 - 1.75 m/s (in the pig shed of the example a pad of little less than 3 m^2 would be needed). The R.H. of the air leaving the pad can reach 90 %. Problems can occur if the water is particularly rich in salts.

b) Low pressure fogger nozzles. These work at 5 - 8 bar pressure and can deliver 8 - 10 l/h. They are placed just below the ventilator. They can get blocked up if the water contains impurities or salt in excess.

c) Spinning disc. These generate a spray of small droplets which are imposed upon an air flow. The spinning discs on the market can evaporate up to 60 l/h of water. In any case it is important that the cooled air travels the shortest possible distance inside the building, to avoid the formation of high temperature and high humidity zones.

6.4.2. Cooling air in underground pipes

In the summer months ground temperature is lower than atmospheric temperature and so passing air through underground pipes can be an effective mean of cooling. A mathematical model for gauging plant dimensions has still not been made, due to the enormous complexity of the problem. Indeed, a vast number of variables come into play and not all of these are known. Today, it is only possible to give indications of an empirical nature.

The first, very important variable is pipe depth ; obviously the results are best at the deepest levels, but installation costs are also greater. For this reason the pipes are normally placed at a depth between 1.50 to 2.0 meters. Pipe diameter influences air volume, since air flow speed can not be increased over a certain limit because of pressure loss due to friction which is proportional to the square of the speed, given by the formula :

$$\Delta p = \lambda \cdot \frac{1}{D} \cdot \frac{\rho}{2} \cdot v^2$$

where

p = pressure loss (Pa)
λ = the friction coefficient (-)
l = tube length (m)
D = tube diameter (m)
ρ = air density (kg/m^3)
v = air speed (m/s)

In practice a speed of 4 m/s is proposed, with a maximum limit of 5 - 6 m/s. For example a plastic pipe of 20 cm in diameter with an air speed of 4 m . sec^{-1}, provokes a pressure reduction of 11.13 Pa every 10 m of length, if air density is 1.16 kg/m.

Besides depending on pipe length, cooling efficiency also depends on the mass of earth involved in the heat exchanges. Since it has been shown by experiments that the layer of earth involved is within 30 - 35 cm of the pipe's surface, a doubling of the diameter from 20 to 40 cm would only increase the earth mass involved about 40 %, while the air flow rate would increase by 400 %, retaining the same speed. It is, then, not economically advantageous to increase pipe diameter over a certain limit. For this reason pipes of 15 - 25 cm in diameter are normally used. Pipe length also has a decisive influence on the earth mass involved ; data obtained from plants in use indicate a length of 4 to 7 m every 100 $m^3 . h^{-1}$ of volume flow for 20 cm pipes. Naturally the highest values should be adopted where hot periods last longer.

Using the dimensions indicated, in Mediterreanean countries we can expect air temperature reductions reaching 12 - 15 K in the hottest hours of the hottest days, but of about 8 - 10 K in the hottest hours of an average summer day. It should be noted that in the coldest hours of the night, the air is actually heated as it passes through the tube. Basically the main effect is the elimination of temperature extremes ; the air leaving the pipes retains a temperature which is very close to the monthly average temperature with very limited deviations (circa $\pm$ 2 K).

Once the length and volume of the single pipe has been defined, the number of pipes should be established. This is based on the total ventilation volume of the building and is calculated according to the method indicated in Chapt. 6.2.5.1. of the "blue book". Considering, however, that in this case incoming air is much fresher than atmospheric air, we can insert a value of Δt also equal to 3 K (or at the most 4 K) into the formula.

Table 1 : Summary of proposed characteristics

max. air speed in the tubes	4 m/s (max. 6 m/s)
diameter of the tubes	15 - 25 cm
length of the tubes	4 - 7 m / (100 m^3/h)
depth of the tubes	1.5 - 2 m
expected temperature effectiveness	50 %

6.4.3. Cooling animals by showers

High temperatures can be tackled by cooling the animals directly. This can be done by periodically subjecting them to water showers. The shower takes heat away from the animal in two ways ; one is by conduction (the temperature of the water being lower than that of the animals' skin), the other is by the evaporation of the water on the animals' skin. Heat loss through conduction only lasts the length of the shower and requires a large amount of water in order to be consistent. Heat loss through evaporation lasts for as long as it takes the animal to dry off (about ten minutes) and requires a modest amount of water. It is preferable to rely on heat loss by evaporation only, in order to restrict water consumption.

The species of animals which benefit most from showers are those which have a low capacity for sweating (basically, pigs). It has been shown that the benefit pigs derive from showers lasts little less than two hours, so the interval between two consecutive showers could be set at between 90 and 120 minutes. The shower need only be short (20 seconds are enough).

Nozzle diameter is much discussed ; if it is too large, water consumption is excessive ; if, on the other hand, it is too small it can produce fog, instead of drops. In this case it may cause evaporative cooling of the air rather than act as a shower. For an efficient cooling action, about 4 - 5 l water per shower should be consumed.

When animals are reared in very dense groups, showers regulated by a timer and covering the whole area of the pen can be installed. If, however, the animals have plenty of space (as in the case of sows in the mating and pregnancy phase) a general system would cause excessive water consumption and it is therefore sensible to install showers which can be regulated directly by the animals themselves.

Table 2 : Summary of proposed characteristics

animal operated installation	
operating interval	90 - 120 minutes
operating time	20 s
water consumption	4 - 5 l/shower

6.5. References

ARMSTRONG, D. 1986. An update on cow cooling methods in the west. Paper, American Society of Agricultural Engineers no. 86-4034.

BARTH, C.L. 1982. State-of-the-art for summer cooling for dairy cows. Proceedings of the Second International Livestock Environment Symposium, Ames, Iowa, 52 - 61.

BENHAM, Jr., WIERSMA, D.S. 1974. Design criteria of evaporative cooling. Paper, American Society of Agricultural Engineers no. 74-4527.

BOTTCHER, R.W., KESLER, D.J., BAUGHMAN, G.R. 1988. Improving evaporative cooling effectiveness by reducing ventilation rate. Paper, American Society of Agricultural Engineers no. 88-4069.

BUFFINGTON, D.E. 1983. Performance characteristics of evaporative cooling systems - progress report. Paper, American Society of Agricultural Engineers no. 83-4023.

CHIAPPINI, U., BARBARI, M. 1985. Raffrescamento dei suini mediante docce. Seminario II Sezione CIGR, Catania, Italy.

CHIAPPINI, U., BARBARI, M. 1984. Raffrescamento dell'aria di ventilazione mediante tubi interrati. Genio Rurale 12, 39 - 52.

CHIAPPINI, U., BARBARI, M. 1985. Tecniche di raffrescamento evaporativo negli edifici zootecnici. Genio Rurale 2, 15 - 34.

CRAMER, C.O., KAMMEL, D.W. 1980. Underground air inlets for farrowing house cooling - a case study. Paper, American Society of Agricultural Engineers no. 80-4556.

DULOSY, E., GLAWISCHING, E. 1980. Fresh air supply for sties through underground storage ducts : technical design. Medizinische Universität Wien.

FEHR, R.L., PRIDDY, K.T., McNEILL, S.G., OVERHULTS, D.G. 1982. Limiting swine stress with evaporative cooling. Proceedings of the Second International Livestock Environment Symposium, Ames, Iowa : 577 - 583.

FLAMENBAUM, I., WOLFENSON, D., MAMEN, M., BERMAN, A. 1986. Cooling dairy cattle by a combination of sprinkling and forced ventilation and its implementation in the shelter system. Journal of Dairy Science (12) 69 : 3140 - 3147.

GARNER, J.C., BUCKLIN, R.A., KUNCKLE, W.E., NORDSTEDT, R.A. 1989. Sprinkled water and fans to reduce heat stress of beef cattle. Appl. Eng. Agr. Vol. 5, 1 : 99 - 101.

GATES, R.S., TIMMONS, M.B. 1988. Stochastic and deterministic analysis of evaporative cooling benefits for laying hens. Transactions of the ASAE, Vol. 31,3 : 904 - 909.

GOETSCH, W.D., MUEHLING, A.J. 1983. Evaluation of earth-tube heat exchangers as air tempering systems for swine farrow-nursery housing. Paper, American Society of Agricultural Engineers no. 83-4073.

GÖTZ, M., RIST, M. 1985. Avoid heat stress in pigs by sprinkling. Seminario II Sezione CIGR, Catania, Italy.

HELLICKSON, M.L., CHEN, C.F. 1987. Simulated and tested performance of a rockbed heat sink for cooling summer ventilation air. Transactions of the ASAE Vol. 30,1 : 207 - 214.

HUHNKE, R.L., LEAL, P.A.M., BRUSEWITZ, G.H. 1983. Evaporative cooling of livestock and poultry - state of the art. Paper, American Society of Agricultural Engineers no. 83-4022.

JUNGBLUTH, T. 1987. Underground air cooling for animal houses. Proceedings of the 3rd international symposium on mechanization and energy in agriculture, Izmir, Turkey : 456 - 469.

MATTESON, D., SCHWARZ, M., MURPHY, J.P. 1981. An underground earth tube ventilation system for swine. Paper, American Society of Agricultural Engineers no. 81-205.

MORRISON, S.R., HEITMAN, H., GIVENS, R.L., BOND, T.E. 1970. Sprinkler use for swine cooling. Paper, American Society of Agricultural Engineers no. 70-913.

MORRISON, S.R., PROKOP, M., LOFGREEN, G.P. 1981. Sprinkling cattle for heat stress relief : Activation temperature, duration of sprinkling and pen area sprinkled. Transactions of the ASAE : 1299.

NEUKERMANS, G., DE SCHRIJVERE, K., DEBRUYCKERE, M., VAN DER BIEST, W., BALEMANS, L. 1989. Conditionnement de l'air de ventilation des porcheries d'élevage par l'échangeur thermique enterré dans le sol. Proceedings of the 11th International Congress on Agricultural Engineering, Dublin, Ireland : 1385 - 1392.

NICHOLS, D.A., AMES, D.R., HINES, R.H. 1982. Evaporative cooling systems for swine. Proceedings of the Second International Livestock Environment Symposium, Ames, Iowa : 204 - 207.

PURI, V.M. 1987. Earth tube heat exchanger performance correlation using boundary element method. Transactions of the ASAE Vol. 30,2 : 514 - 520.

SCHAUBERGER, G., KECK, G. 1984. Der Bodenspeicher als energiesparendes Luftaufbereitungssystem. Institut für Medizinische Physik, Veterinärmedizinische Universität Wien.

SCHAUBERGER, G., AXMANN, H., KECK, G. 1980. Fresh air supply for studies through underground storage, physical properties. Institut für Medizinische Physik, Veterinärmedizinische Universität Wien.

SPENGLER, R.W., STOMBAUGH, D.P. 1982. Design and optimization of a subterranean heat exchanger for swine ventilation. Proceedings of the Second International Livestock Environment Symposium, Ames, Iowa : 184 - 192.

STRICKLAND, J.T., BUCKLIN, R.A., NORDSTEDT, R.A., BEEDE, D.K., BRAY, D.R. 1988. Sprinkling and fan evaporative cooling for dairy cattle in Florida. Paper, American Society of Agricultural Engineers no. 88-4042.

STRICKLAND, J.T., BUCKLIN, R.A., NORDSTEDT, R.A., BEEDE, D.K., BRAY, D.R. 1989. Sprinkling and fan cooling systems for dairy cows in hot, humid climates. Appl. Eng. Agr. Vol. 5,2 : 231 - 236.

TAYLOR, S.E., BUFFINGTON, D.E., COLLIER, R.J., DELORENZO, M.A. 1986. Evaporative cooling for dairy cows in Florida. Paper, American Society of Agricultural Engineers no. 86-4022.

TIEDEMANN, H. 1985. Erdwärmetauscher für Schweineställe. Institut für Landwirtschaftliche Verfahrenstechnik Universität Kiel : 644 - 679.

WEISBECKER, T.L., JACOBSON, L.D. 1980. Analysis of cooling air in a buried pipe. Paper, American Society of Agricultural Engineers no. 80-4555.

WIERSMA, F., ARMSTRONG, D. 1983. Cooling dairy cattle in the holding pen. Paper, American Society of Agricultural Engineers no. 83-4507.

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Chapter 7

VENTILATION AND HEATING CONTROL IN ANIMAL HOUSES

INTRODUCTION

The right ventilation and heating capacity constitutes a major criterion in designing climatization of animal houses. However, the actual inside condition in the house will largely depend on the way the heating and ventilating equipment is put to use. That is why a well designed control system is essential.

As animal production and energy consumption are in direct relation with the output of the central system for 24 hours a day, considerable economical consequences are bound up with the efficiency of the control system.

7.1. Theoretical considerations

7.1.1. The house and control characteristics

An increase in ventilation causes a decrease in inside temperature. The graph (see Fig. 1a) shows this basic relationship which is called the "house characteristic". Depending on the outside conditions and the building characteristics on the one hand and the heat production of the animals and the artificial heating inside the house on the other hand, the house characteristic will be different.

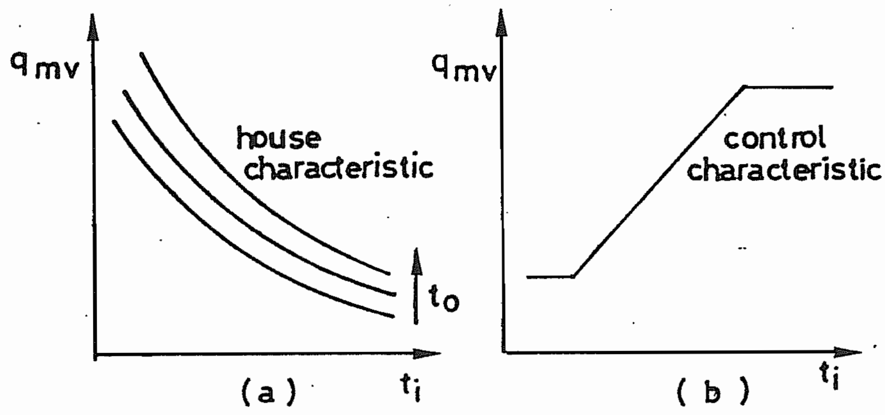

Fig. 1 : The house and control characteristic

Considering the controller, an increase in ventilation should be correlated with an increasing inside temperature. The graph which shows this relation is called the "control characteristic". Fig. 1b shows the control characteristic in the case of a proportional control. Some controllers have a discontinuous characteristic e.g. step control.

The actual temperature that will occur in the house can easily be read from those graphs when they are put together (Fig. 2).

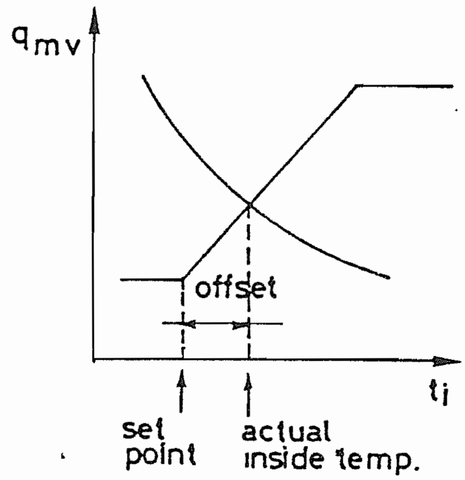

Fig. 2 : Actual inside temperature for a given house and control characteristic

7.1.2. Characteristics of proportional control of ventilation

For many years now proportional control of ventilation has been applied in animal houses. However, in practice there is no proportionality between the temperature difference (actual temperature - set point) and the amount of ventilation. The proportionality exists between the temperature difference and the electric power applied to the fan. Therefore, the power applied to the fan cannot be proportional to the amount of ventilation, unless an airflow measurement is included in the system.

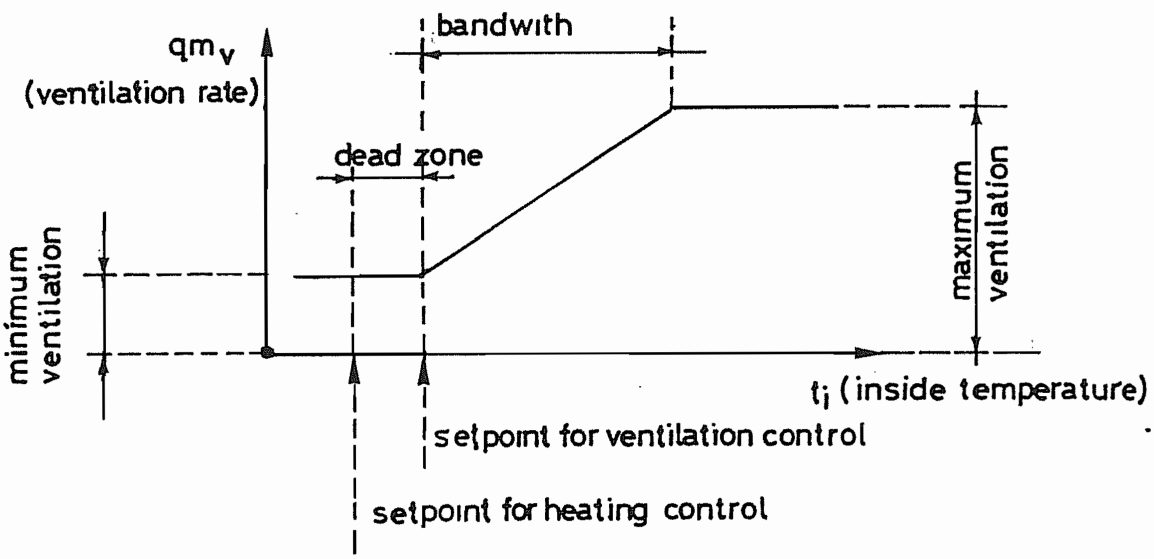

Fig. 3 : Characteristics of the proportional control algorithm

Where natural ventilation with a proportional control is applied, the proportionality exists between the temperature difference and the position of the baffles. This means that Fig. 3 should be considered as the theoretical representation of the proportional control algorithm for ventilation control in animal houses. Redefining the y-axis in terms of electric power or baffle position, leads to the control characteristics as they are actually practised.

Obviously the maximum ventilation rate is determined by the ventilation capacity installed. The minimum ventilation, however, is to be adjusted by the farmer. Although the amount of minimum ventilation can be calculated (see chapter 8) very few guidelines can be given to the farmer as to how he should use his equipment. The professional skill of the farmer in judging the inside conditions is decisive. Nevertheless, it should be obvious that the level of minimum ventilation is determined by the same factors that influence moisture and gas production in the house (see chapter 8).

In order to avoid energy losses a dead zone is necessary between the setpoint for ventilation and the setpoint for heating. As the heating is often controlled by an on-off controller, overshoots in temperature are likely to occur. The dead zone prevents the ventilation from increasing, so the generated heat is not drained away. The difference between both setpoints is usually taken as 1 K or 2 K. Controllers which have this dead zone hardware installed are called "controllers with heating switch". Sometimes the dead zone can be externally adjusted, according to outside conditions. As the artificial heating need increases the need for a large dead zone decreases.

A typical feature of proportional control is the "offset" (see Fig. 2).

This is no real problem from the energy point of view since the situation occurs only when there is a heat surplus in the house. On the other hand, animal performance (feed conversion in particular) can decline. For those reasons one might decrease the difference between the temperature at which the controller imposes minimum ventilation, and the temperature at which the controller imposes maximum ventilation. This difference is usually called the "bandwidth" (see Fig. 3). A direct consequence of a decrease in bandwidth would be a faster response when outside temperature is low. However, a fast response must be restricted since animals cannot sustain fast variations. That is why the bandwidth is to be calculated according to outside temperature so that the largest bandwidth is associated with the coldest outside conditions. Another argument for a rather large bandwidth is related to the occurrence of overshoots. As explained, some heating equipment, when shut off, will continue to dissipate heat which increases inside temperature, even above the set-point of the ventilation controller. With a small bandwidth the ventilation system would react firmly and quickly drain away the generated heat, while an unchanged ventilation level would bring the inside temperature back to the set-point value. Obviously a small bandwidth would increase the energy consumption in those cases, when the dead zone is not large enough.

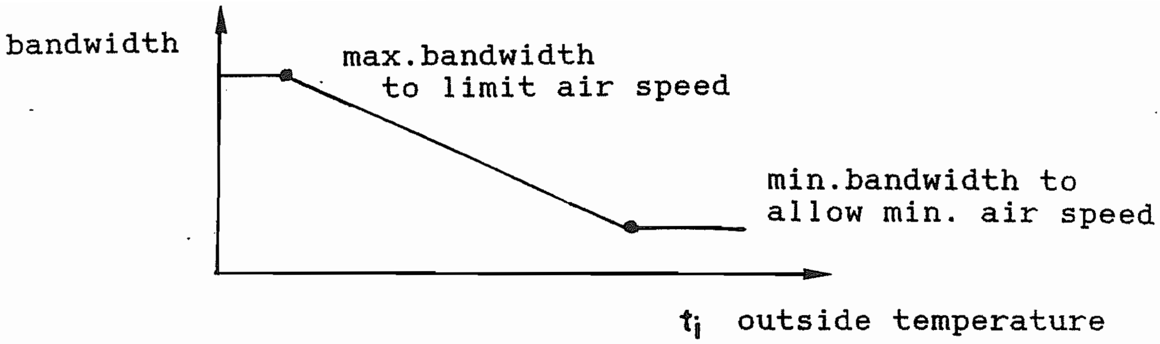

Fig. 4 : Principal relation between bandwidth and outside temperature

These problems can be overcome by making the bandwidth dependent on the outside temperature, so that the largest bandwidth is associated with the coldest outside conditions. When the bandwidth cannot be adjusted, a zone of 4 K is usually taken. The smallest bandwidth should not be less than 2 K. Minimum and maximum bandwidth can be deduced theoretically from the desired air speed in the house. Bearing in mind that an increase in air speed adds to the animal's comfort at high temperature, the smallest bandwidth can be defined as the bandwidth that allows of a minimum air speed.

7.1.3. Characteristics of "three point control" of ventilation

As mentioned before, practical consideration make it unfeasible to install a proportional control according to a straight relation between temperature difference and ventilation rate, unless the latter is measured. This measurement is unfeasible as well, when natural ventilation is considered. Unlike mechanical ventilation, for which a rather reliable relationship can be established between the ventilation rate and the electric "power", natural ventilation shows a much poorer relationship between the ventilation rate and the position of the baffles. This is due to the important influence of the outside conditions (temperature, windspeed). These parameters are beyond control. That is why in many cases it might be more appropriate to rely on the fact that in any case a decrease/increase in ventilation rate can be performed by closing/opening the baffles. In practice, this means that a "three point" or "floating control" will be used, imposing the opening of the baffles when the inside temperature is too high and vice versa.

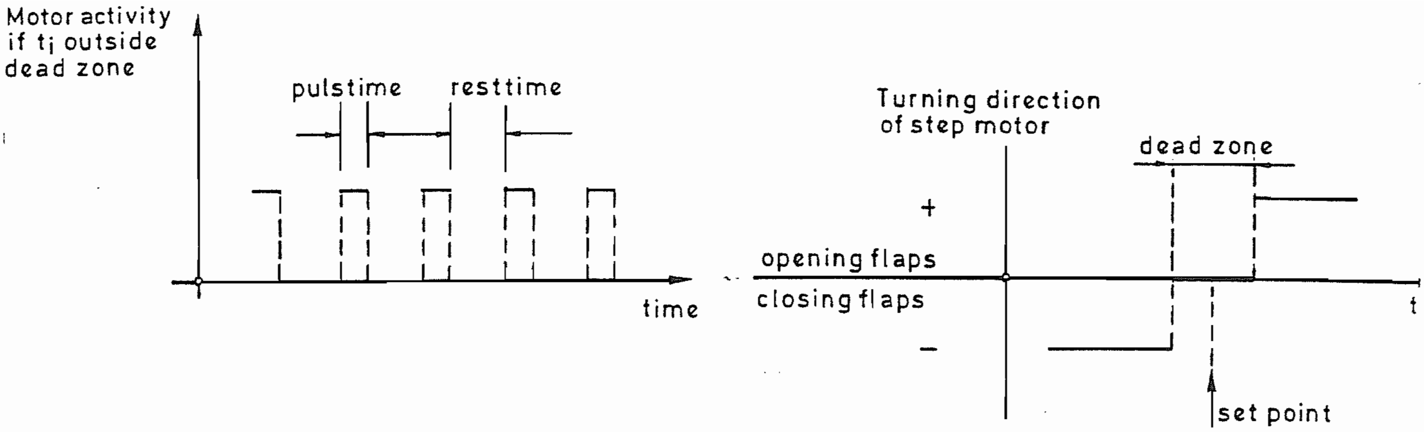

Fig. 5 : Characteristics of a three point control

The actions of opening and closing are performed during a predetermined time (= "action time" or "pulse time") at regular intervals. The time in-between is called the "rest time" or "interval time".

The action time and the interval time are usually chosen as a few seconds and a few minutes respectively. However, a more precise control requires that the action time should be calculated according to the difference between inside temperature and the set point (the larger this difference the larger the change in inlet area should be) and the difference between inside temperature and outside temperature (the larger this difference the smaller the change in inlet area should be). Until now no scientific research has been done to optimize this algorithm. Nevertheless, it should be obvious the action time can be greater in big and well installed houses. In some cases the action time is chosen to be smaller at opening and larger at closing. This should be considered as an improvement because too low a temperature is more harmful than too high a temperature.

The position of the temperature sensor is really crucial regarding stability and accuracy. The problem arises because any change in climate due to a change of the position of the baffles should be sensed. On the other hand, any change in climate due to other influences should be neglected (e.g. fast variation in wind). In practice a compromise is realised by positioning the sensor in the incoming air stream about 1 m from the air inlet, and by having a separate control system for each side of the house (leeward and windward sides).

The second practical elements are the end switches. Since floating control acts relative to the actual position of the baffles, there is a need for detecting when the baffles are entirely closed or entirely opened. This can be done by installing end switches at both ends of the opening range. By choosing the right position of these switches provision can be made for minimum ventilation.

A third implementation requirement is related to the outlet area : when the outlet is closed no control can be achieved ; when the outlet is entirely open, each change in inlet area will have a maximum effect.

This offers the theoretical possibility of controlling the sensitivity of the system. However, from a practical point of view it is more advisable to leave the outlet at maximum opening, so the inlet area can be kept as small as possible in order to minimize draught risks. Nevertheless the outlet should be closed when the inside temperature drops below the critical temperature. That's why in practice the outlet area is changed with the season.

Three point control can also be used to control baffles when mechanical ventilation is applied. In that case the baffle control is intended merely to prevent cold draughts at all ventilation rates. The fan controller takes care of the optimal temperature.

7.1.4. Heating control

Proportional control is also used with heating equipment. Nevertheless in many cases an "on - off" control is preferred with regard to investment costs. This is particularly true when hot water radiators are used. A proportional control in this case would require a three-way valve with a step-motor, which is rather expensive. Thermostatic valves, often used with radiators, have a very small bandwidth, and for that reason can be considered as "on-off"control. A disadvantage of "on - off" control on this type of equipment is the risk of "overshoots" (as already mentioned). In order to overcome this problem the water temperature can be controlled. Since the heat demand increases linearly with the outside temperature and the heating capacity of the warm water heaters is linear with the water temperature, a linear relationship between outside temperature and water temperature is the obvious approach. This linear function is determined at one end by the outside determining temperature and the maximum water temperature, and at the other end by the minimum outside temperature which requires no heating and the water temperature which reduces the heat production of the heater to zero. Obviously this occurs when the water temperature equals the house temperature. Although the last pair of coordinates is determined by the animal heat production in the house, in most cases this heat production will be neglected within this calculation. As a consequence the lowest water temperature and the corresponding outside temperature are both set equal to the house temperature. This method leaves enough heating capacity to perform a sufficiently quick response whenever needed.

When more than one room is connected on the same heating circuit a somewhat different approach will be necessary. This approach can be based on the water temperature at departure and return. Most central heating systems are based on a twenty degrees difference at maximum capacity. Obviously there is no need for any difference when there is no heat demand. Between both situations a linear relationship can be established. This method has no built-in guarantee that each room will get the minimum heating capacity as needed. That's why it should be applied with some care. Nevertheless some reserve can also be introduced here by using the inside temperature of the house instead of the minimum outside temperature which requires no heating.

The mathematical expression for both methods can be given by :

$$t_{hw} = t_i + \frac{(t_wh)_{max} - t_i}{t_i - (t_o)_{min}} (t_i - t_o) \qquad (9)$$

$$\Delta t_{hw} = \frac{(\Delta t_{hw})_{max}}{t_i - (t_o)_{min}} (t_i - t_o) \qquad (10)$$

with :

t_{hw}	= water temperature at departure (°C)
$(t_{hw})_{max}$	= water temperature at maximum heating capacity (°C)
t_i	= inside temperature (°C)
t_o	= actual outside temperature (°C)
$(t_o)_{min}$	= outside minimum design temperature (°C)
Δt_{hw}	= temperature difference between departure and return (K)
$(\Delta t_{hw})_{max}$	= maximum temperature difference between departure and return (K).

As previously mentioned, proportional control instead of "on-off" control is quite possible without expensive means in some cases, e.g. when gas heaters and electrical heaters are considered. Gas burners can be governed by a gas-flow control and electrical heaters can be governed by a voltage control. However in the latter case a large thermal capacity can inhibit effective proportional control when rather fast changes are needed.

According to agricultural application, attention should be paid to floor heating. Whether electricity or warm water is used to heat the floor, the control problem is somewhat the same.

The determining element in floor heating control is the animals'heat loss to the floor. Obviously the floor temperature has to be the controlled parameter depending on the animals' age and the air temperatures. Both factors show a very slow variation since animals do not grow all that fast, and the air temperature is controlled to be constant. Neither is the floor temperature disturbed by fast-fluctuating influences. As a consequence, floor heating can be as well controlled by hand. Only slight modifications involving no labour burden (turning the knob of a dimmer for electrical systems or setting a timeswitch when warm water heating is used with an on - off valve) will suffice. The criterion the farmer can use is the way the animals huddle together.

When automatic control is preferred, two different approaches are possible, depending on whether the floor temperature is measured, or calculated from other measurements. In the former case, attention should be paid to where the sensor is installed. This must be a place whose temperature can serve as a good reference for the whole floor. At the same time this place must not be influenced by animals lying down or by faeces. These are requirements which are difficult to meet.

A sensible expression in terms of an indirect approach could be :

$$t_f = \frac{1}{c} \cdot t_{ab} + \left(1 - \frac{1}{c}\right) t_o \qquad (11)$$

with

t_f = floor temperature (°C)

t_{ab} = deep body temperature of the animal (°C)

t_o = optimal air temperature (°C)

c = coefficient (-) see equation 12

The expression can be derived from a mathematical expression stating the equality between the heat losses of a recumbent animal and the heat losses of an animal standing up at optimal temperature (t_o). In order to be compatible with common practice the value of the coefficient should be calculated as :

$$c = 0.19 * m_a + 3 \qquad (12)$$

with

m_a = animal weight (kg)

Another way of heating, particularly for very young pigs, is achieved by radiant heat. Infrared lamps or dark radiant panels can be controlled in the same way as other electrical heaters. Such systems offer the possibility of control, unrelated to energy consumption, by varying the height the radiator is suspended. Although this method can be justified by the requirements of growing animals (more space, less warmth), in practice little use is made of it.

7.2. Practical considerations

7.2.1. Traditional analogue control equipment

In principle the traditional analogue equipment can be described as an apparatus with one input (related to one sensor) and with one output whose value depends on the sensed value and a set-point. So analogue controllers can be categorised according to the type of input (temperature, humidity, ...), to the type of output (power line, on-off line, ...), or to the kind of control algorithm which is hardware implemented in this type of equipment.The implemented control algorithm is very much the most important factor when energy and climate are considered. Nevertheless a number of practical aspects with which the farmer is confronted should be mentioned first.

The setting of the apparatus is mostly done by means of dialling knobs. Turning this knob moves the mark on it along a scale. If, and only if, everything is well gauged, the right set-points can be introduced. Unfortunately with time and wear (the mark on the knob disappears, the knob itself is not very well fixed anymore, the scale is somewhat moved ...) there is no guarantee that the exact value can be read from the front-panel.

Constant checking of the temperature by means of a (good) thermometer is a must. By checking the temperature on a regular basis there is a good chance that an eventual malfunction of the controller will soon be discovered. Nevertheless checking the controller for correct operation requires at least a minimum-maximum temperature. It should be obvious to anyone who is concerned about climate and energy use in animal houses, that there can be no optimization if regular checking is left out. This is even more true when analogue apparatus is used.

When combining several analogue apparatus within one control system the situation becomes rather complex. First of all, several sensors can perform in a different way even when the same parameter is sensed. These differences do not need to be very large if the sensors concerned are positioned in the same spot. Nevertheless any difference can lead to a substantial effect when heating and ventilation controls are not properly linked.One example of this link, referred to as a "heater-switch" should prevent the fans from increasing the ventilation above minimum level as long as the heater is in operation. This rather simple but very important "heater-switch" makes it impossible that the heating and the ventilation are both working at maximum (or high) capacity without meeting the climatic needs in the house. This situation can occur (when there is no heater-switch) due to an erroneous setting of one of the controllers, to wit when the set-point of the heat controller is higher than the set-point of the ventilation controller. This situation can occur even without the farmer being aware of it, because the scales on the controllers are not always the same, nor are they fully reliable.

7.2.2. Digital and computer oriented equipment

Obviously the specific possibilities of digital technology should be used in order to avoid the problems mentioned which analogue equipment gives cause for. Moreover, new applications should be made possible, avoiding at the same time the creation of new problems. This last consideration requires that the contact the farmer has with the climate in the house should be preserved, so the controller must be placed in the house itself (or very near to it). At the same time management (as discussed later) requires centralisation. In that respect a modular built local network should be seen as the most feasible solution.

A basic module has, or can have, several inputs from the same number of sensors. When more than one sensor is used for the same parameter, an average can be made, and furthermore, alarm can be given when the parameter values differ too much from each other. It will be clear that the relationship between the inputs and the outputs (several outputs are possible) can be very complex, without the necessity of each output having its own input. In this, digital equipment does not show the inconveniences of the traditional apparatus, while on the other hand putting the emphasis on the relationship between inputs and outputs where energy will be saved or lost. This relationship is implemented by a program which can be stored in EPROM making it easy to change.

The digital display is most useful when introducing the setpoints with great accuracy. In the meantime, it displays the actual temperature in a more readable way, improving control by the farmer. A preceding condition to this is a properly operating sensor. In order to check this, a good alcohol or mercury thermometer remains indispensable. A difference between the thermometer reading and the value shown by a display can indicate a hardware failure (as in traditional equipment) or can be due to a mismatch of sensor and software when the sensor signal has to be converted to temperature by calculation.

Not only the basic control module, but also the communal module offers energy saving possibilities by taking into account the parameters which are common for all the houses or compartments (outside temperature, outside humidity, water temperature of the central heating ...). Obviously the emphasis here will also lie on the program within the communal module. But there is more, since the communal module can also be equipped in order to serve as an "energy-monitoring-device". To do so, kWh-meters and calorimeters must be connected to the basic module as well as flow-meters indicating the oil or gas consumption when appropriate. Connecting these measuring-devices is only possible when they have an output which can be made compatible with the requirements of the communal module input.

The third element within the network is the (personal) computer which can be used for other purposes too. In this way the animal-related information (e.g. number and age of animals occupying a particular compartment) stored in the computer for administrative reasons can be used within the climate-control-network (e.g. calculation of temperature and minimum ventilation in that particular compartment).

There is no doubt that this possibility will serve a double goal : optimizing the animals' performance by preventing temperatures which are too low while minimizing the energy demand by preventing temperatures which are too high. This kind of information is sent under a master-slave-protocol to the communal module. In the same way, information from the communal module can also be transmitted to the (personal) computer. This possibility allows the user of a computer-oriented climate-controller to built a true management information system with regard to climate and energy use.

7.2.3. Sensors

- time constant is not critical
- location is most important

  rule : sensed temperature must be in direct relation to temperature the animals experience

  hints : avoid - radiation on sensor
  - outside air streams on sensor

keep sensor away from doors, windows, etc.
the use of more than one sensor is advisable

7.2.4. Alarm units

- an alarm unit is always indispensable
- control parameters would be : minimum temperature
  maximum temperature
  power supply

Responsible member

PEDERSEN S.

Chapter 8

DUST AND GASES

Dust

Airborne dust in animal houses is a serious problem, especially in pig houses, because it causes problems for both animals and humans. During the last decade measurements of dust in animal houses have been carried out in many places over the world. The measurements were carried out in different ways. Soon it has become obvious that it is difficult to compare results obtained by different methods, because the collected part of the total dust in the air depends on the measuring methods. As an example the inlet velocity, the inlet orientation, the location of equipment within an animal house and the period of measuring are of importance for the results obtained. In the following, different methods for measuring dust are discussed.

8.1. Methods of measuring dust

8.1.1. Settlement method

The simplest way is to weigh the amount of settled dust on horizontal plates placed in different places in the room. The settlement is due to gravity and will also be influenced by the air movement, especially for small dust particles.

8.1.2. Photocell method

Figure 1 shows the principle of an instrument for continuous measurements of dust. TAKAI et al. (1985).

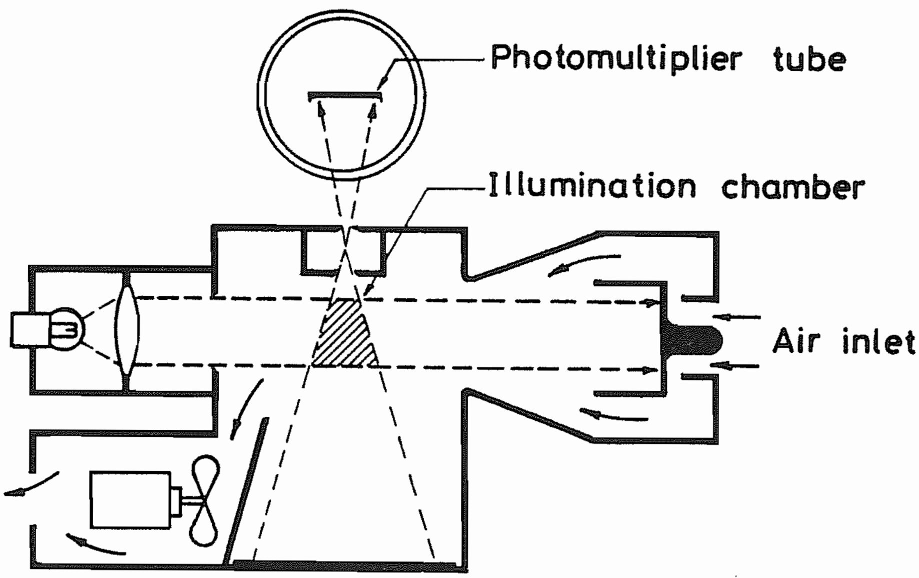

Fig. 1 : Principle of photocell method

Measurements by the photocell method give continuous values of dust at the point where the equipment is placed. Figure 2 gives an example, obtained in a house for finishing pigs.

Advantages

- fast method
- shows continuously the changes in dust concentration

Disadvantages

- gives relative values for the dust concentration, and it is difficult to convert the values e.g. to mg dust per cubic metre of air
- expensive

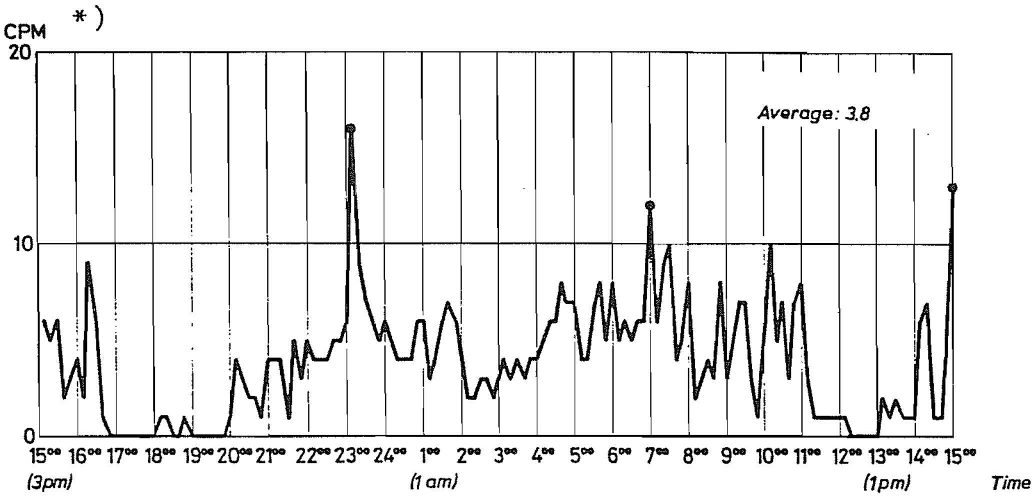

Fig. 2 : Examples of measurements in house for finishing pigs. The pigs are automatically fed at 7.00 a.m., 3.00 p.m. and 11.00 p.m. TAKAI et al. (1985).

8.1.3. Laser method

Figure 3 shows the principle in laser equipment for measuring dust.

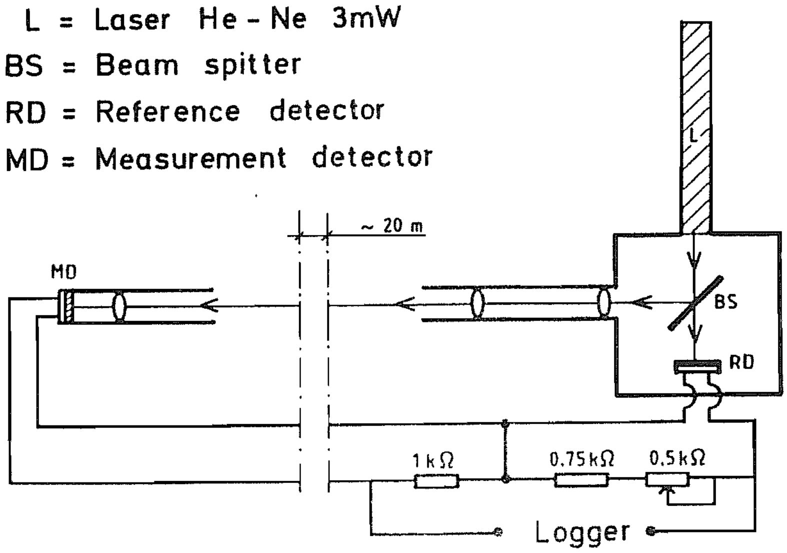

Fig. 3 : Principle of using laser equipment for measurement of dust concentration. NILSSON, C. (1982)

Measurements by the laser method give instantaneous values.

Advantages

- fast method
- shows continuously the changes in dust level
- is able to take in account the amount of dust along the whole length of the animal house.

Disadvantages

- gives only relative values, and it is difficult to convert the values e.g. to mg dust per cubic metre of air.
- expensive
- cannot distinguish between dust particles and drops of water.

8.1.4. Filter method

Figure 4 shows the equipment for measuring dust by means of the filter method. TAKAI et al. (1985).

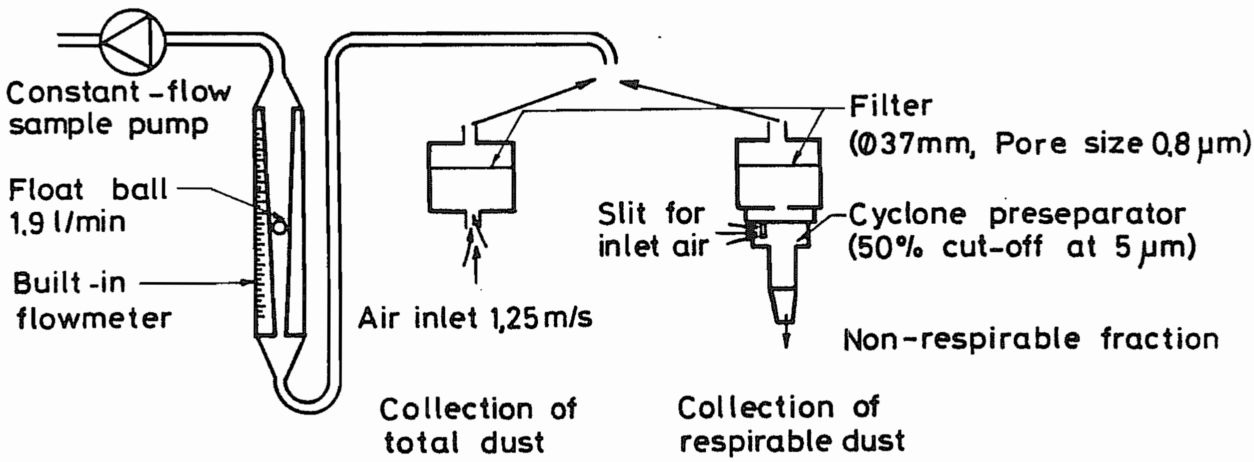

Fig. 4 : Equipment for measuring dust by the filter method

The equipment consists of an air pump with a constant flow of 1.9 l/min and a filter housing with a filter. For measuring respirable dust a cyclone is mounted on the filter housing.

Advantages

- exact values of dust in mg per cubic metre of air
- the inlet velocity is standardized to 1.25 m/s. Can be considered an international standard

Disadvantages

- labour consuming method
- time delay between the measuring time and obtained results due to subsequent laboratory analyses
- time related average concentrations. Cannot be used for instantaneous measurements.

8.1.5. Other methods for evaluation of dust in animal houses

Attempts have been made to express the total dust production in an animal house. GUSTAVSSON (1987) has set up the following formula for pigs :

$$q = 3.8 \times m \times n$$

where :

q = Dust production, mg/h

m = average weight of pigs, kg

n = number of pigs

One part of the dust produced in an animal house will be removed by means of the ventilation plant, and the other part will sink, due to gravity, and lie on the floor, the crates, etc. It must be emphasized that the formula does not take in account the level of pig activity in different pig production systems.

The sinking velocity of spherical airborne particle can be calculated by the following formula (SCHNEIDER, 1986) for a dust particle in still air :

$$v_{dust} = \frac{d^2}{18} \frac{\rho_{dust} \times C(D)}{\mu} \times g$$

where

v_{dust} = sinking velocity, m/s

d = diameter of particle, m

ρ_{dust} = density of dust, kg/m^3 (approx. 1.000)

u = viscosity of air, Ns/m^2

$C(D)$ = Cunningham's slip correction factor
g = 9.81 m/s^2

As an example, the viscosity of air is 170.8 micropoises ($1.708 * 10^{-5}$ Ns/m^2) at 0 °C and 182,7 micropoises ($1.827 * 10^{-5}$ Ns/m^2) at 18 °C. The slip correction factor decreases from 1.1642 at d = 1 μm to 1.0082 at d = 20 μm. For a density of spherical dust of 1.000 kg/m^3 the velocity due to gravitation, in still air, will be as shown in fig. 5.

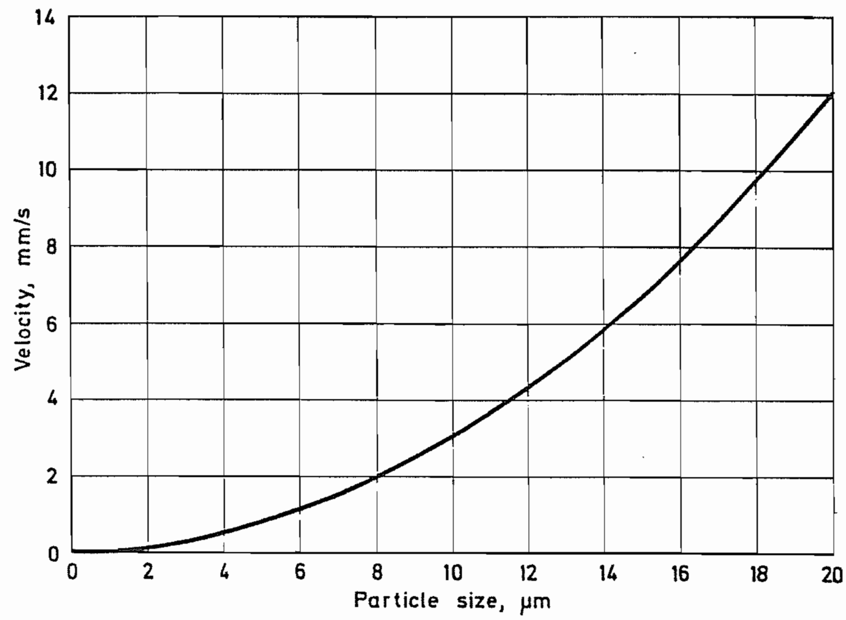

Fig. 5 : Velocity due to gravitation of spherical dust particles of different sizes. The density of dust is 1.000 kg/m^3, and the air temperature 15 °C

As previously mentioned the sedimented dust can simply be measured on horizontal plates and expressed as (e.g.) mg dust per square metre and hour.

When the sedimented dust, the dust concentration and the ventilation flow for an animal house is known, the dust production can be calculated by adding dust measured in the outlet air and dust sedimented in the house.

Because small particles pass on to the lungs, it is also appropriate to know the particle-size distribution.

Particle distribution of dust can be determined by, for example,a computer-aided microscope analyzer.

Other analyses of interest are knowledge of contents of bacteria, fungus, hair, scurf, etc.

8.2. Dust levels in different countries

As mentioned in section 1, dust can be measured in different ways, and it is difficult to compare results, measured by different methods. Great care needs to be taken not to oversimplify a complex problem. The dust concentration changes from time to time, and also inside the animal house there can be differences in dust concentration from one place to another. In the following tables some results from different countries are given.

8.2.1. Scotland

Results obtained by field studies in North-East Scotland show that the majority of pig pens have significantly different concentrations between the front and the back of a pen. Concentrations of total dust from pig houses, monitored by the Centre for Rural Building, Scotland, range from less than 1 mg/m^3 to above 40 mg/m^3, based on an average, time-weighed over a 24 hours period, within each pen (ROBERTSON, 1977), see table 1.

Table 1 : Dust levels in Scottish pig houses

Housing stage Environmental variable	All-farm mean	Range of individual farm means
Farrowing house :		
settled dust	1.5	0.5 - 2.4 $g/m^2/24$ hours
gravimetric dust	2.0	0.5 - 3.0 mg/m^3
total dust	1.8	0.4 - 5.2 mg/m^3
respirable dust	0.17	0.02 - 0.40 mg/m^3
Weaner house, step 1 :		
settled dust	8.0	1.5 - 39.0 $g/m^2/24$ hours
gravimetric dust	8.2	1.6 - 50.6 mg/m^3
total dust	7.1	0.6 - 46.9 mg/m^3
respirable dust	0.32	< 0.01 - 1.16 mg/m^3

(continued)

Housing stage Environmental variable	All-farm mean	Range of individual farm means
Weaner house, step 2 :		
settled dust	8.8	1.5 - 19.3 $g/m^2/24$ hours
gravimetric dust	7.5	2.1 - 14.1 mg/m^3
total dust	3.8	1.0 - 9.5 mg/m^3
respirable dust	0.21	0.04 - 0.67 mg/m^3
Finishing house :		
settled dust	7.3	0.5 - 22.6 $g/m^2/24$ hours
gravimetric dust	8.2	1.1 - 18.6 mg/m^3
total dust	4.4	0.6 - 12.3 mg/m^3
respirable dust	0.29	0.01 - 0.73 mg/m^3

8.2.2. Switzerland

Table 2 shows results from measurements of dust by GRÜTER, 0. (1975) in Switzerland. For measurement, an SM 13400.090 filter from Sartorius,with a diameter of 77 mm and a pressure drop of approx. 5 kPa is used.

Table 2 : Average dust concentrations in the air in animal houses, GRÜTER, O., Switzerland

Type of house	Litter	Fodder	Air change m^3/h x animal	Air change per hour	Dust mg/m^3
Finishing house	none	pellets	55.8	12.7	0.8
Finishing house	none	pellets	56.3	14.5	1.3
Finishing house	none	powder dispenser	39.2	16.7	7.7
Finishing house	none	powder	30.2	15.4	3.9
Finishing house	none	wet	51.9	21.0	2.8
Finishing house	none	wet	17.7	10.7	2.2

(continued)

Type of house	Litter	Fodder	Air change m^3/h x animal	Air change per hour	Dust mg/m^3
House for laying hens	battery, no litter	powder	5.0	36.7	1.5
House for laying hens	battery, no litter	powder	3.0	18.8	3.6
House for laying hens	floor keeping, cut straw	powder	2.5	6.7	6.2
House for laying hens	floor keeping	powder	3.7	21.9	24.3
House for breeding hens (10 % cocks)	floor keeping	powder	2.3	6.1	40.1
House for breeding hens (10 % cocks)	floor keeping	powder	1.9	4.8	32.8
House for breeding hens (10 % cocks)	floor keeping	powder	4.6	11.4	28.5
House for breeding hens (10 % cocks)	floor keeping	powder	3.1	7.4	17.4

8.2.3. Holland

Table 3 shows the results from Dutch experiments with measurements of dust in 4 finishing houses by ATTWOOD, P. (1986).

Table 3 : Dust concentrations in 4 Dutch finishing houses

Size fraction	Mean mg/m^3	Range mg/m^3
$D_{50} \leq 3.5$ µm	0.10	0.06 - 0,25
$D_{50} \leq 8.5$ µm	0.37	0.69 - 0.54
Total dust 100 % < 51 µm	1.32	0,69 - 2.29

The measurements are made by sampling units, using glass micro-fibre filters GF/A (WHATMAN 4.7 cm). D_{50} corresponds to 50 % cut-off effectiveness of 3.5 µm at a flow rate of 50 litres/minute and 8.5 µm at a flow rate of 15 litres/minute.

8.2.4. Denmark

Danish experiments in 42 houses for finishing pigs and 10 houses for farrowing sows have included measuring dust over a 24 hours period by the filter method (TAKAI et al., 1986). The results for farrowing houses are shown in table 4.

Table 4 : Dust level in Danish pig houses, obtained by the filter method, shown in figure 4.

Type of house	Feeding system	Bedding system	Other remarks	Total dust mg/m^3	Respirable dust mg/m^3
Farrowing house	dry	straw	10 houses/24 hours	1.4	0.2

Table 5 shows the average concentration of total dust in different types of houses for finishing pigs (TAKAI et al., 1986).

Table 5 : Average concentration of dust in different types of houses for finishing pigs. Example from Denmark

	Number of houses	Total dust mg/m^3
Dry feeding	19	1.8
Wet feeding	20	1.4
Equal or positive pressure ventilation	17	1.7
Negative pressure ventilation	22	1.4
With bedding	26	1.7
Without bedding	13	1.3
Dry feeding, equal or positive pressure ventilation and with bedding	8	1.9
Wet feeding, negative pressure ventilation without bedding	6	1.4

Table 6 shows examples of exposure of the pigman to dust during the whole day, when feeding and distributing straw respectively (TAKAI et al., 1985).

Table 6 : Total dust, mg/m^3, measured in the breathing zone of herdsman

	mg/m^3
Average for a full day of work (8 hours)	3.2
Feeding only (approx. 1 hour daily)	5.9
Distribution of straw only (approx. 0.5 hours daily)	5.3

8.3. Reduction of dust in the indoor air

8.3.1. Fogging

By means of special nozzles water can be spread in the animal houses. After a period of fogging the air is easy to breathe (NILSSON, 1982 and TAKAI et al., 1986).

Spraying just before the herdsman enters the animal house seems one of the ways to make a good indoor climate for him. For the pigs, which will have to stay in the house over a 24 hour period, there seems to be no, or only a small positive, effect.

8.3.2. Vacuum cleaning

Regular cleaning, e.g. once a week, can be done by means of a vacuum cleaner.

Cleaning will have a positive effect on reducing the dust, but a significantly better climate is not yet achieved.

8.3.3. Showering of passages and fittings

By showering the passages etc. it is possible to moisten the dust to a level where the dust will not rise again. Some positive effect can be expected, but significant results are not yet found.

8.3.4. Ionization of the air

When negative ions are discharged from ionization heads installed in an animal house, the dust particles will be ionized and attracted to earth-connected surfaces.

Since 1985 experiments have been carried out at SjF (Danish Agricultural Engineering Institute). In the beginning, only a little effect on the dust concentration in the air was obtained. During 1988 the experiments were continued with a greater number of ionization heads in a weaner house. With 1 ionization head per 1.6 square metre of floor the following results are obtained (Table 7).

Table 7 : Results from use of ionization in weaner house

	Ionization		Reduction
	off	on	%
Total dust, mg/m^3	3.1	2.8	10
Respirable dust	0.36	0.25	25

Further investigations in Denmark on the effect of ionization, not yet published, show about 30 % reduction both in total dust and respirable dust.

8.3.5. Electrofilter

The principle of an electrofilter is shown in Figure 6. The voltage for the ionizer wire is about 14 000 V. The dust particles which are positively charged, will be attracted by the negatively charged collector plates.

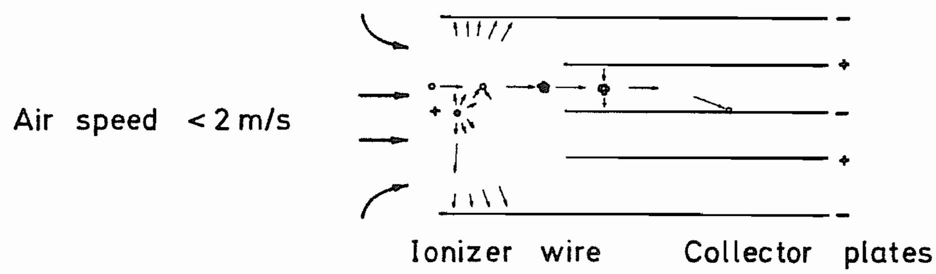

Fig. 6 : Principle of the electrofilter

Experiments with electrofiltration in animal houses showed, for instance, that 81 % of the dust was kept back by the collectors at an air velocity of 1.5 m/s through the electrofilter, but only 33 % at 3.1 m/s.

The air flow through the electrofilter must be the same as, or greater than, the ventilation flow, before a reasonable effect can be expected. This means that it is necessary to use big electrofilters to obtain a sufficient effect. The above mentioned air speed of 1.5 m/s corresponds, for instance, to an air flow of only 5.400 m^3/h per square metre cross section of the electrofilter.

8.3.6. Filtration by means of dust filter

Cleaning of indoor air by recirculation of air through a dust filter section is a possibility, but normally it will be an expensive way to go, because, as mentioned above, the amount of recirculated air must be high compared to the ventilation flow, otherwise the reduction of dust in the air will be low.

8.3.7. Oil spraying

As already mentioned, fogging by means of water has a short reducing effect on the amount of dust in the indoor air. A more permanent effect can be obtained when using oil, because the evaporation of oil is slow.

At SjF (TAKAI, H. and LARSEN, E. (1988)) experiments are carried out concerning spraying a mixture of rape oil, soap and water. Figure 7 shows the principle of the spraying equipment.

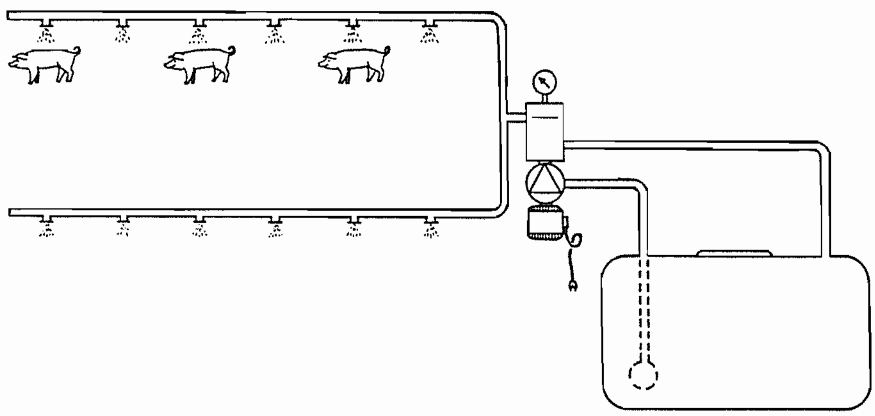

Fig. 7 : Principle of oil spraying system

Some results are shown in Figure 8. As seen, the reduction of dust concentration is high. A main conclusion of several tests with oil spraying since 1985 is that the dust concentration in animal houses can be reduced to 10 - 50 %, depending on the amount of oil sprayed per m^2 and the dust level. Up to now oil spraying seems to be one of the most attractive ways to go, when the dust concentration has to be reduced.

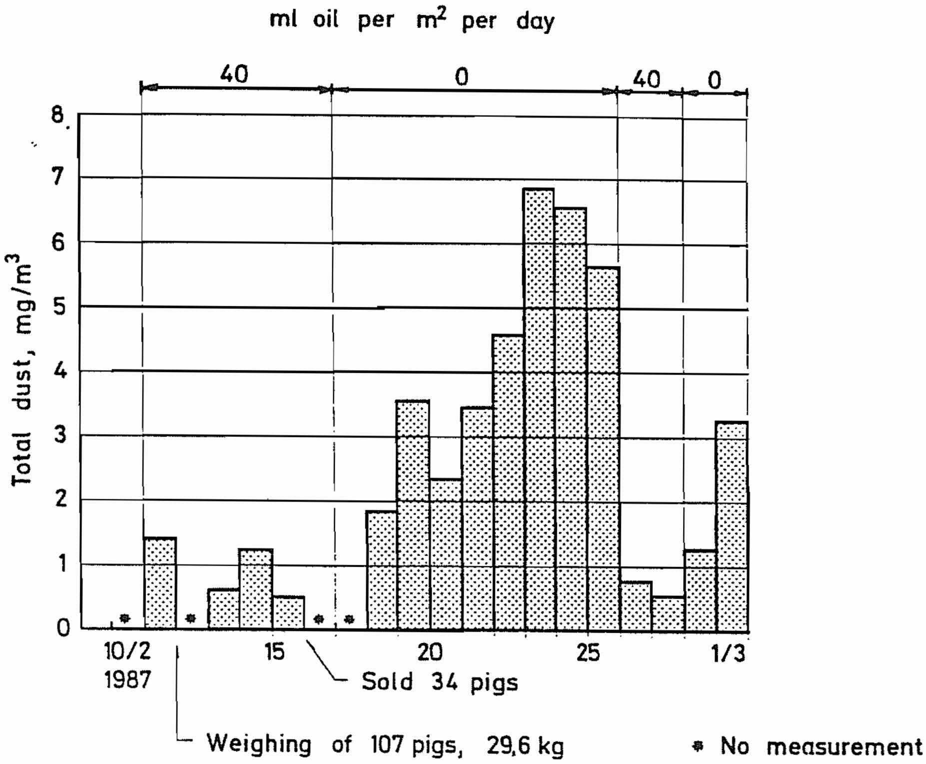

Fig. 8 : Dust concentration in weaner house with and without spraying a mixture of oil, soap and water

8.4. Influence of dust on health

Animals are normally exposed to dust in animal houses 24 hours a day, and the animal keeper is exposed up to 8 hours a day.

It is likely that dust, together with gases such as ammonia, has a great influence on the health of both animals and humans. The biggest dust particles will be retained in the nose and the throat, but the smallest will continue to the lungs, as shown in Figure 9.

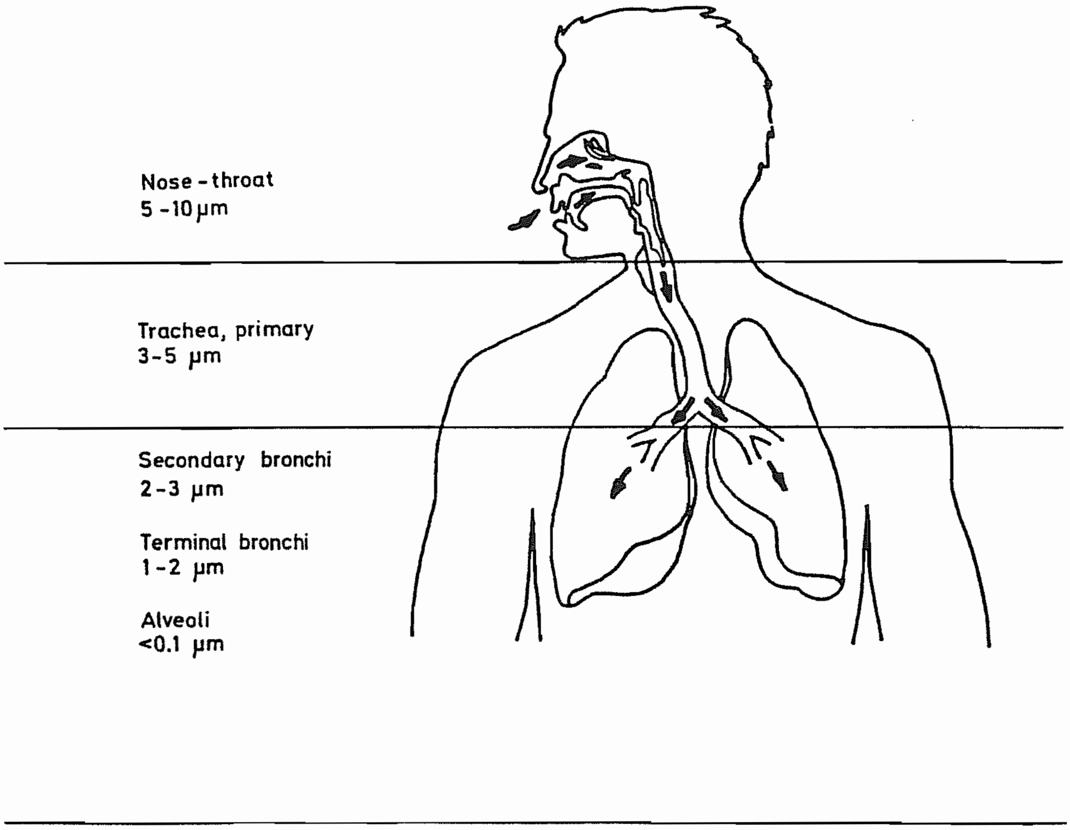

Fig. 9 : Respiratory passage with respect to dust invasion

From Danish investigations the distribution in the air in a finishing house is shown in Table 8.

Table 8 : Frequency of dust particles in the air in finishing houses

Particle size µm	Frequency of particles %
0.5 - 1.5	22
1.5 - 3.0	28
3.0 - 5.0	20
5.0 - 7.0	16
7.0 - 10.0	8
10.0 - 15.0	4
15.0 - 20.0	1
over 20.0	1

IVERSEN, M. et al. (1988) has evaluated the prevalence of asthma and chronic bronchitis on the basis of a questionnaire, sent out to 1685 Danish farmers. The results are shown in Table 9.

Table 9 : Prevalence of asthma and chronic bronchitis among Danish farmers

	Dairy farmers	Pig farmers
Asthma, %	5.5	10.9
Chronic bronchitis	17.5	32.0

The table shows that the incidence of asthma and chronic bronchitis occurs twice as often for pig farmers as for dairy farmers.

The influence of dust on the health of animals is not well known. It is likely that dust together with gases like ammonia has an influence on the incidence of carcass damage noted at slaughtering, which is well known in many countries.

It is not yet possible to state exactly how much dust can be allowed in animal houses. In Denmark, and other countries too, up to 5 mg of total dust (organic) is allowed for an 8 hour working-day. Otherwise, it is necessary to use personal protection. As shown in Table 6, the dust concentration can exceed 5 mg/m^3. The animals are exposed to dust over a 24 hour period which means that the mean dust concentration should not exceed $5 \times (8/24) = 1.7$ mg/m^3. The figures, previously shown in this chapter, often shows dust concentrations above 1.7 mg/m^3 which may suggest that dust is a serious problem for the health of the animals.

Investigations on the health of animals exposed to rape oil, as described above, are in progress at SjF.

Part of the organic dust consists of bacteria and fungus (dead or alive).

Normally, about 10^6 cfu/m^3 of bacteria and 10^4 cfu/m^3 of fungi, where cfu = colony forming units, will be found.

About 1 % of the micro-organisms are alive.

In some animal houses a considerable quantity of endotoxins is found, which is of great interest in relation to the health of animals and humans. More research has to be done in this area, before guidelines can be stated.

8.5. Methods of measuring gases

The most important gases in animal houses are ammonia, hydrogen sulphide and carbon dioxide, but other gases such as carbon monoxide can also be of significance in animal houses with gas burners.

For quick measuring of gases, detector tubes are normally used, e.g. from the firm Dräger AG Lübeck, as shown in Figure 10.

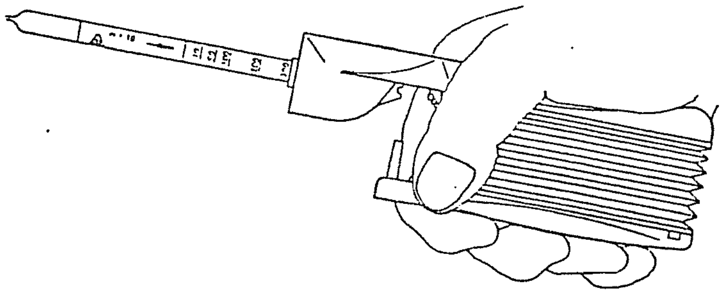

Fig. 10 : Equipment for measuring gases, consisting of a hand pump and a test tube

Measurements with detector tubes are easy to carry out, but give only immediate values. By means of special tubes and an electrically operated pump it is possible to obtain average concentrations over a period of several hours. Figure 11 shows an example of such an equipment.

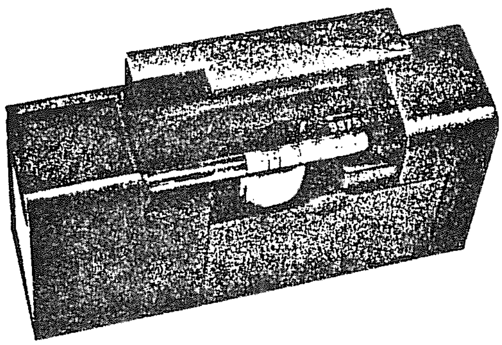

Fig. 11 : Equipment for measurements of average concentration of gases by means of detector tubes

Another method for measuring the average concentration of gases, e.g. over a 24 hour period, is to collect air continuously to a diffusion-proof plastic bag by means of an air pump, and then measure the concentration of gases with detector tubes. Fig. 12 shows the principle of such a method.

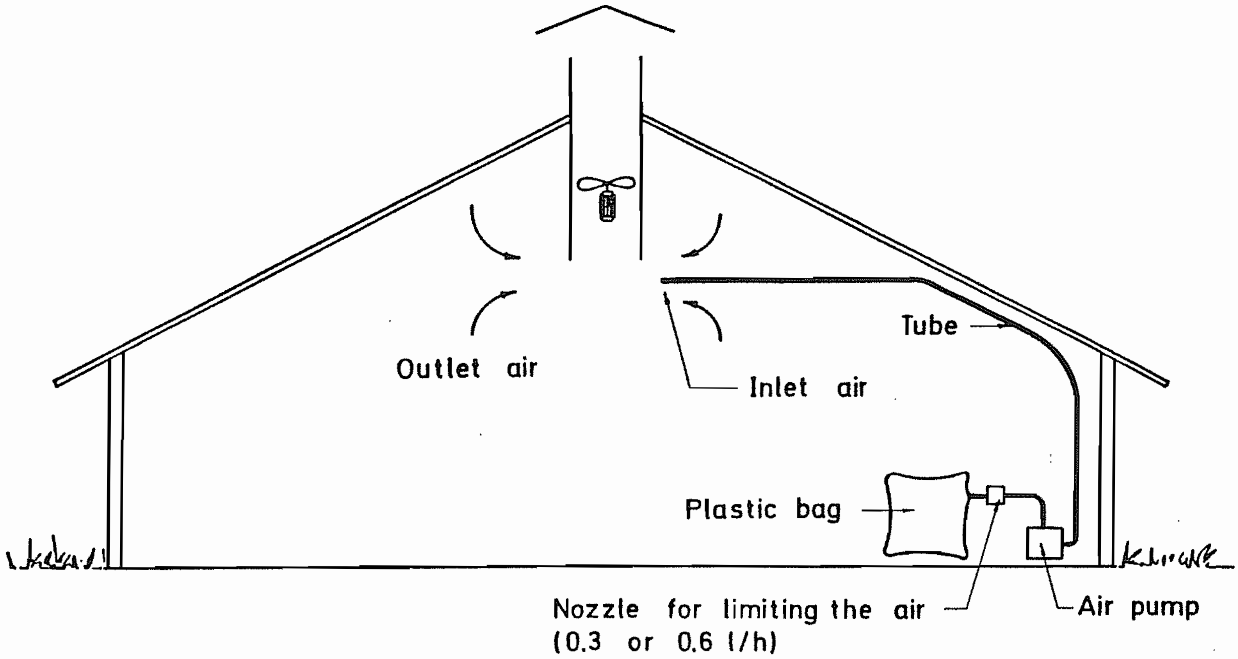

Fig. 12 : Principle of measuring gases over a longer period, e.g. 24 hours

Continuous measurements of gases can be made according to different principles. Figures 13 and 14 show examples of different equipment.

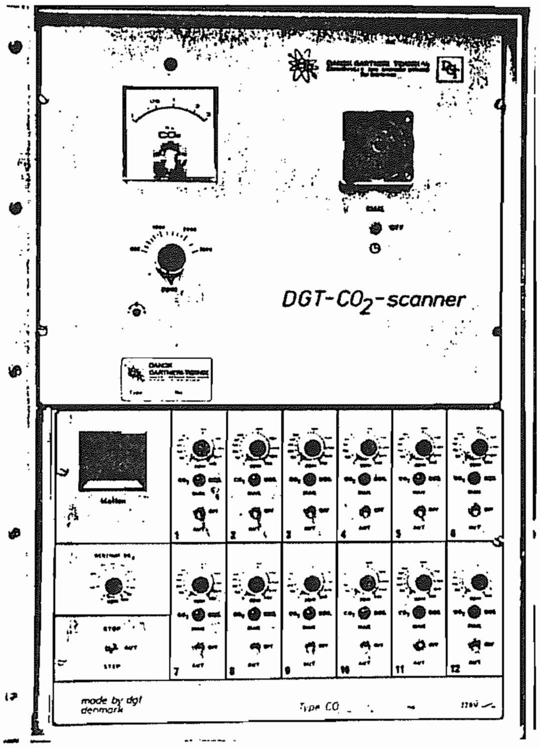

Fig. 13 : CO_2-scanner, 12 channels, based on a gas chromatograph, an air pump, and a series of magnet valves

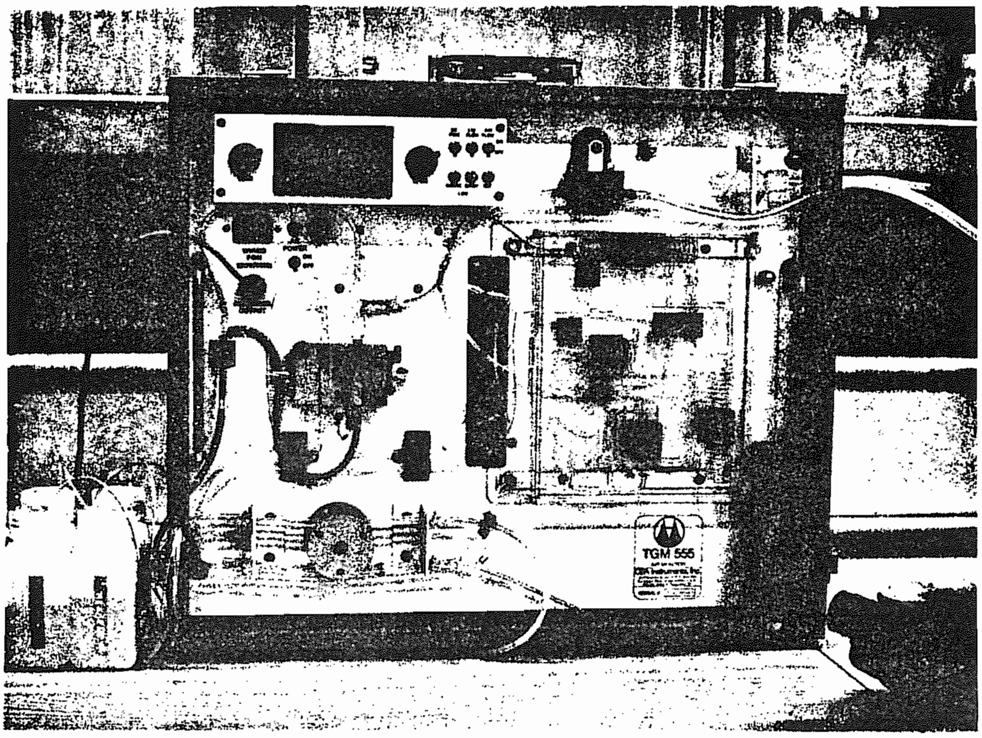

Fig. 14 : Portable gas monitor with a signal output to a data-logger. By changing analytical module and chemistries the monitor can be used to measure different gases, e.g. NH_3, H_2S

8.6. Ammonia concentration in different countries

The concentration of ammonia in animal houses depends on type of animal, organization of the house, season, ventilation principles, hygiene, etc. This means that the concentration will differ from one house to another and from time to time.

The following tables show typical values of ammonia in animal houses in different countries.

8.6.1. Norway

Investigations of ammonia in houses for cattle, pigs and laying hens were carried out in the Seventies, TJERNSHAUGEN, O. (1975).

In 50 houses for dairy cows (stanchion barns) the ammonia concentration was measured in different places. In Table 10 some figures are given.

Table 10 : Ammonia concentration in houses for cattle

	Number of measurements	Mean ppm	Range
0.3 m above manure channel	94	14,7	3 - 44
0.3 m above feeding passage	49	7.3	0 - 25
Close to air outlets	48	12.0	2 - 35

In 81 houses for pigs the results shown in Table 11 were obtained.

Table 11 : Ammonia concentrations in 81 Norwegian pig houses

Type of house	Place of measuring	Number of measurements	Mean ppm	Variation ppm
Farrowing houses	Floor level in rest area	95	11.7	0 - 40
	0.3 m above floor level	97	11.8	2 - 40
Houses for pregnant cows	Floor level in rest area	73	11.8	1 - 26
	0.3 m above floor level	70	12.9	2 - 40
Finishing houses	Floor level in rest area	147	12.2	2 - 50
	0.3 m above floor level	147	12.2	2 - 34

In houses for hens the results shown in Table 12 were obtained.

Table 12 : Ammonia concentrations in houses for laying hens

System	Number of measurements	Mean ppm	Range
Deep litter	260	15.2	0 - 100
Slatted floor	170	11.3	2 - 30
Battery cages	932	6.2	0 - 34
Stepped cages	240	8.3	1 - 50
Stepped cages (open to cellar)	120	7.9	0 - 30

8.6.2. Italy

During the period 1980 - 88 the ammonia concentration was measured in some animal houses, CHIAPPINI, U. (1980-88), and the results are shown in Table 13.

Table 13 : Ammonia concentrations in Italian animal houses

Type of house	Number of houses	Ammonia ppm	Variation ppm
Calves	4	7	2 - 12
Fattening cattle	1	2	
Dairy cows	1	4	
Farrowing sows	7	6	2 - 18
Pregnant sows	4	6	2 - 10
Boars	1	2	
Weaners	3	6	3 - 10
Finishing pigs	1	5	
Hens	1	5 - 20	

8.6.3. Scotland

At the Centre for Rural Building measurements (ROBERTSON, 1977) were carried out for ammonia, and the results are shown in Table 14.

Table 14 : Ammonia concentration in Scottish investigations

Housing type	Farm mean	Range of individual farm means
Farrowing houses	7.9	1.3 - 22.2
Weaner, step 1	5.1	1.4 - 14.8
Weaner, step 2	9.9	3.4 - 30.8
Finishing house	10.6	1.3 - 21.5

8.6.4. Sweden

Investigations on ammonia were done by GUSTAFSSON, G. (1988) in 29 pig houses in the northern part of Sweden (latitude 63) during the wintertime. The results are shown in Table 15. The concentration of ammonia in these experiments is much higher than shown in the Tables 10 to 14, which can be explained by very cold weather together with air leakage between the scraper channel under the slatted floor and outside.

Table 15 : Distribution of ammonia concentrations in 29 pig houses in the northern part of Sweden

Range ppm	Finishing houses (numbers)	Farrowing houses (numbers)
0 - 5	1	1
5 - 10	2	3
10 - 15	4	2
15 - 20	3	3
20 - 25	3	2
25 - 30	3	
30 - 35	1	
35 - 40	1	

Further measurements in southern Sweden were carried out by DONHAM, K. (1986) in autumn 1984 at 30 swine farms, as shown in Table 16.

Table 16 : Ammonia concentrations in pig houses in the southern part of Sweden

	Exhaust air		1.2 m above floor level		In the boxes	
	no. of measurements	Mean ppm	no. of measurements	Mean ppm	no. of measurements	Mean ppm
Finishing house	4	4.0	31	9.4	6	23.2
Farrowing house	8	7.1	10	8.3	9	8.7

8.6.5. Denmark

Measurements of ammonia have been included in many Danish investigations on climatization of animal houses. The measurements can be single, weekly or monthly. In Table 17 the main results since 1978 at SjF (Danish Agricultural Engineering Institute) are shown.

Table 17 : Measurements of ammonia in animal houses in Denmark 1979 - 1987

Housing type	Number of houses	Mean ammonia concentration ppm	Variation ppm	Publication
Weaners	16	2.4	1 - 4	SjF-Orientering no. 37, 1985
Weaners	16	2.6	1 - 4	SjF-Orientering no. 50, 1987
Finishing pigs	100	5.3		SjF-Beretning no. 2, 1979
Finishing houses	14	7.0	2 - 20	SjF-Beretning no. 7, 1981
Finishing houses	5	8.5		SjF-Beretning no. 20, 1985
Calves	54	7.0	1 - 30	SjF-Beretning no. 6, 1981
Calves	4	7.0	9 - 18	SjF-Orientering no. 16, 1983
Calves	3	4.3		SjF-Beretning no. 21, 1984
Calves	8	4.4	1 - 20	SjF-Orientering no. 25, 1984

In addition, in 1985 - 86 measurements were carried out over 24 hours in 54 animal houses, the results being shown in Table 18.

Table 18 : 24 hours measurements in Danish animal houses, 1985 - 86

Housing type	Winter		Summer	
	Total	mean ppm	Total	mean ppm
Dairy cows, mechanical ventilation	8	3.4	6	1.5
Dairy cows, natural ventilation	8	2.4	8	1.8
Young cattle	4	4.1		
Pregnant sows	4	5.5		
Farrowing sows	4	5.5		
Weaners	3	4.7		
Finishing pigs	12	4.1	11	2.1
Laying hens	3	2.5 - 20.0		
Broilers	3	2.0 - 10.0		

8.6.6. Diurnal and yearly variation of ammonia concentration

Figure 15 shows an example of the diurnal variation in ammonia in a finishing house, 50 mm over the slatted floor in the dunging passage (SÄLLVIK, K. et al., 1973).

"High exhausting" means outlets in the roof, and "low exhausting" means that the outlet air is sucked under the slats.

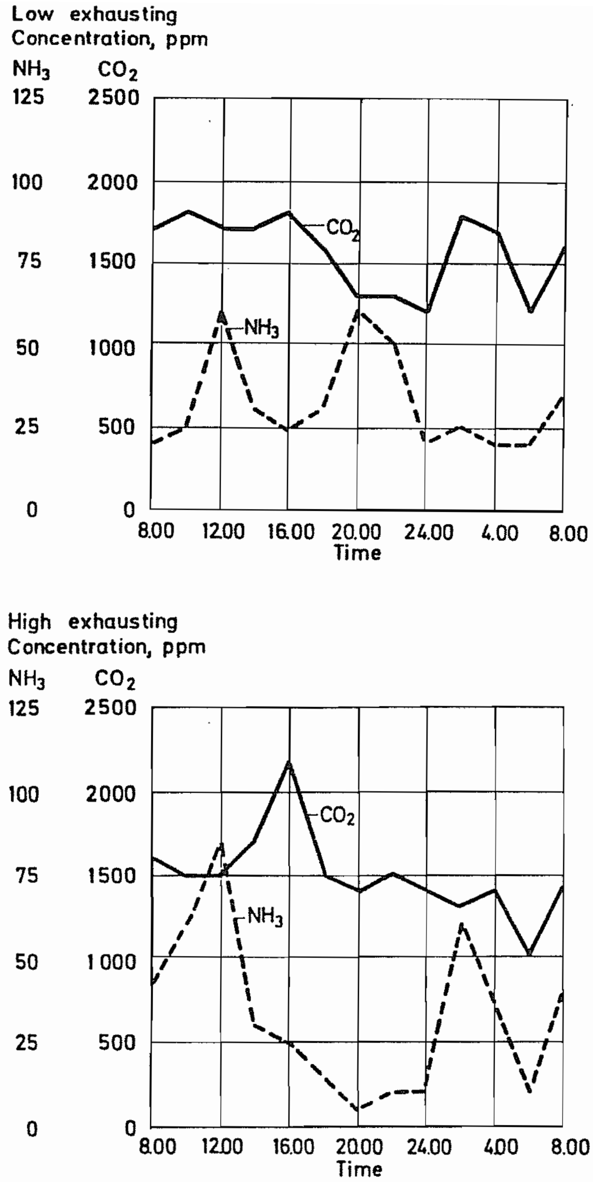

Fig. 15 : Diurnal variation in ammonia concentration

In Danish experiments in progress, measurements of ammonia, carbon dioxide, etc. are carried out in finishing houses. In Figure 16 some results of yearly measurements are shown. In the wintertime, where the ventilation flow is low, the concentrations of ammonia and carbon dioxide are higher than in the summertime.

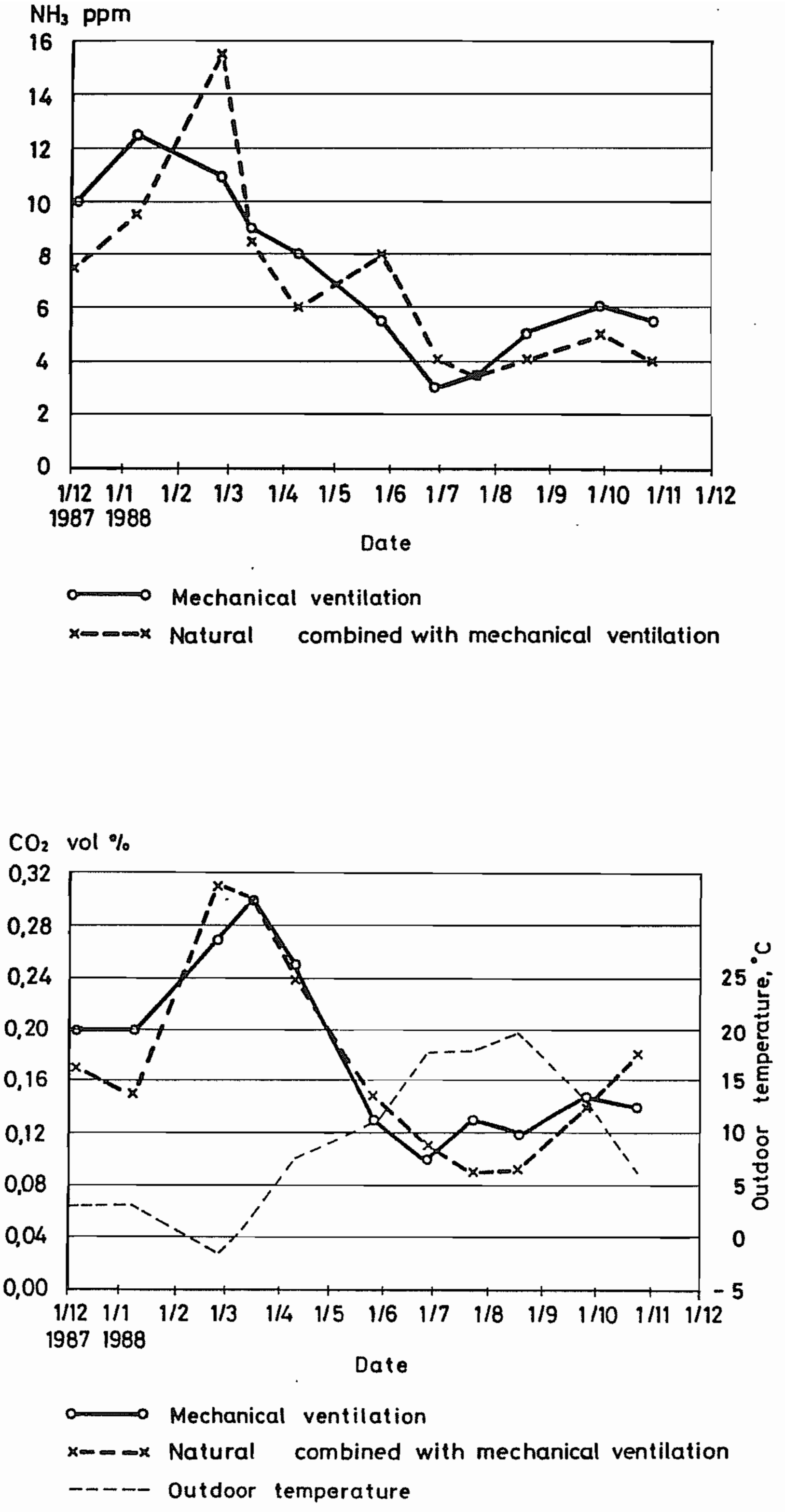

Fig. 16 : Ammonia and carbon dioxide concentrations in a finishing house

8.7. Other gases

8.7.1. Hydrogen sulphide

Hydrogen sulphide normally cannot be measured in animal houses. Only in animal houses with partly or fully slatted floors can H_2S be measured when handling the slurry. Fig. 17 shows an example of what happens when slurry is released from manure cellars.

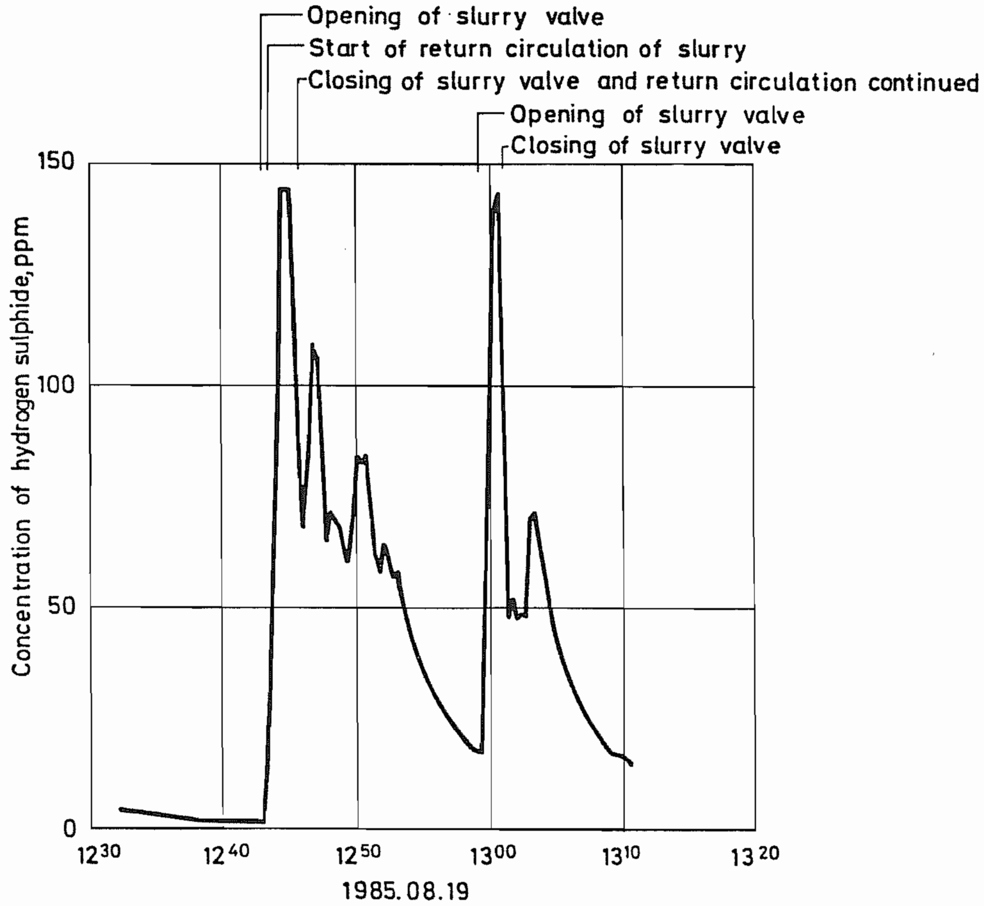

Fig. 17 : Examples of hydrogen sulphide concentration in house for fattening pigs

The concentration can rise from zero to several hundred ppm within one minute.

8.7.2. Carbon dioxide and oxygen

Normally there will be no risk at all concerning too high a concentration of carbon dioxide or too low a concentration of oxygen. Only in the case of ventilation failure in air-tight houses can the concentrations be dangerous.

Figure 18 shows changes in carbon dioxide and oxygen concentration in a weaner house with forced failure of the ventilation (PEDERSON, 1986).

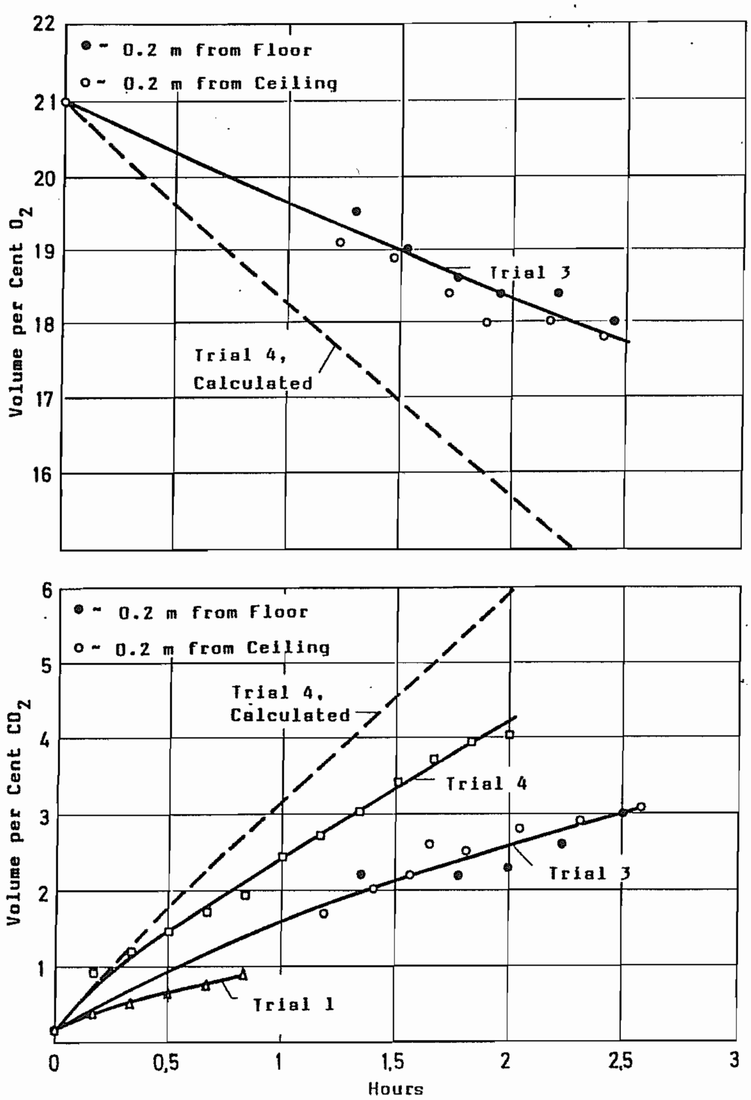

Fig. 18 : Changes in content of oxygen and carbon dioxide in case of ventilation failure

The concentration of animals was 120 pigs of about 12.5 kg, corresponding to about 2 kW of total heat per m^2.

8.8. Influences of gases on health

Abnormal concentrations of gases have influences on health, e.g. excessive concentrations of ammonia, hydrogen sulphide, carbon dioxide or too low a concentration of oxygen.

The following Tables 19, 20, 21 and 22 show how human beings and animal react to the gases ammonia, hydrogen sulphide, oxygen and carbon dioxide.

Table 19 : Reaction to ammonia

Concentration ppm	Symptoms
10	Some negative effects at long term exposure
15	Smell threshold for human beings *
20	Eye irritation for broilers
25 - 35	Stockmen feel uncomfortable
50	Disturbance of productive capacity Water flows from the stockman's eyes
70	Reduced daily gain and poor feed conversion
5 000	Deadly within a few minutes

NIMMERMARK (1984)
* Often ammonia is found also at lower concentrations, down to 5 ppm

Table 20 : Reaction to hydrogen sulphide

Concentration (ppm)	Effect
0.01 - 0.7	Least detectable odour
3 - 5	Offensive odour
10	Eye irritation
20	Irritation to mucous membranes and lungs
50 - 100	Irritation of eyes and respiratory tract (1 hr.)
150	Olfactory-nerve paralysis, fatal in 8 - 48 hr.
200	Headaches, dizziness (1 hr), nervous system depression
500 - 600	Nausea, excitement, insomnia, unconsciousness, possible death (30 min.)
700 - 2 000	Rapidly fatal

NORDSTRØM and McQUITTY (1976)

Table 21 : Reaction to oxygen
Response of man to the inhalation of atmospheres deficient in oxygen.

Stage	Oxygen vol. %	Symptoms of Phenomena
1	12 - 16	Breathing and pulse rate increased, muscular coordination slightly disturbed
2	10 - 14	Consciousness continues, emotional upsets, abnormal fatigue in case of exertion, disturbed respiration
3	6 - 10	Nausea and vomiting, inability to move freely, loss of consciousness may occur, may collapse, and although aware of circumstances may be unable to move or cry out
4	Below 6	Convulsive movements, gasping respiration, respiration stops, and a few minutes later heart action ceases

BERD R.R. (1982)

Table 22 : Reaction to carbon dioxide

Carbon dioxide %	Increase of respiratory capacity %	Symptoms
1	14	
2	29	Disturbance of colour vision
3	57	At light working level, 1/4 of men have some pain in the chest musculature, caused by increased respiration At heavy working level the pulse rate is higher than at normal carbon dioxide concentration
4	100	At moderate working level 3/4 have a mild headache. Reduced staying power at heavy working level.
5	270	At fast run for 6 minutes 1/3 collapsed. Confusion, disturbance of sight and heavy headaches. After a 15 minutes exposure 1/2 have breathing problems, 1/3 headaches, 1/4 trouble with motor activities. Feelings of faintness, headaches and intoxication, pulse rate and blood pressure raised.
17		Unconsciousness after 1/2 minute and convulsion beginning
Above 17		Danger to life

NIELSEN (1984)

The composition by volume of dry, outdoor air is approximately 78.09 % nitrogen, 20.95 % oxygen, 0.93 % argon and 0.03 % carbon dioxide with slight traces of inert gases. Owing to the many processes going on in livestock buildings, the composition of the air can become a complex assortment of chemical compounds. Some important properties of NH_3, H_2S and CH_4, prevalent gaseous products of manure decomposition, are summarized in Table 23.

Table 23 : Relevant properties of gases (At 25 °C, 760 mm Hg, unless specified otherwise)

Gas	Formula	s.g. * (air = 1)	Odour	Colour	Affinity for water
Ammonia	NH_3	0.597	sharp, pungent	colourless	highly soluble (90 g/100 ml H_2O at 0 °C)
Hydrogen Sulphide	H_2S	1.189	offensive (rotten egg)	colourless	moderately soluble (0.6 g/100 ml H_2O at 0 °C)
Carbon Dioxide	CO_2	1.53	odourless	colourless	highly soluble
Methane	CH_4	0.554	odourless	colourless	slightly soluble

* specific gravity

Maximum acceptable concentrations of noxious gases have been discussed for many years, and different guidelines are used. In the previous report "Climatization of Animal Houses", 1984, the Working Group agreed on the guidelines in Table 24.

Table 24 : CIGR recommendations for maximum gas concentrations, based on a Council of Europe proposal

Gas	Maximum concentration	Remarks
Carbon dioxide, CO_2	3.000	
Ammonia, NH_3	20	Measured as a mean in the dwelling zone of animals
Hydrogen sulphide, H_2S	0.5	Intermittently, when dunging out, 5 ppm
Carbon monoxide, CO	10	Considered only when fossil fuel burners are used for additional heating

8.8.1. Comments on the CIGR Recommendations

Carbon dioxide (CO_2) is produced by the animals' metabolism and exhaled. The CO_2 concentration is a measure of general contamination of the inside air. The CO_2 concentration inside the building should be as low as possible. The assumed calculation value for minimum ventilation requirement is 3 000 ppm, but temporarily fluctuations may occur. Outside air contains 300 ppm CO_2, and exhaled air from the lungs contains 40 000 - 60 000 ppm.

Ammonia (NH_3) originates from manure and is therefore primarily a manure management problem.

Hydrogen sulphide (H_2S) originates from the anaerobic fermentation of manure. It is released from the manure when it is agitated. Normally no H_2S should be measurable. The lethal concentration is about 350 ppm, but adverse effects are noticeable at 50 ppm or less.

Carbon monoxide (CO) originates from fossil fuel burning heaters. If more than traces are detected by normal chemical techniques, measures must be taken to adjust the burners.

8.9. Summary

- All measurements in pig houses indicate that there is plenty of dust. Measured as total dust, according to the Johannesburg Convention, there is about 2 mg/m^3, but up to 20 mg/m^3 or more has been measured.

- About 10 % of the amount of total dust is respirable dust.

- Ionization and the use of electrofilters have some reducing effect.

- The most effective method of getting rid of the dust is, up to now, distribution of rape oil by means of a spraying system.

- The concentration of ammonia in animal houses is normally about 5 - 10 ppm and of the same level in different countries, except in areas with extremely cold weather.

- In animal houses with slatted floors, the concentration can go up to about 50 ppm above the dung channels, if there is a leakage to the outside or to other animal houses.

8.10. Literature

ATTWOOD, P. et al. Assessment of Dust and Endotoxin Levels in the Working Environment of Dutch Pig Farmers. A preliminary study, Ann. Occup. Hyg. Vol. 30, no. 2, 201 - 208, 1986.

BERD, R.R. Patty's Industrial Hygiene and Toxicology. Inorganic Compounds of Oxygen, Nitrogen and Carbon. John Wiley and Sons, 1982.

CHIAPPINI, U. Ammonia in Animal Houses in Italy 1980-88. Personal communication.

Climatization of Animal Houses. Report of a CIGR working group. Published by the Scottish Farm Buildings Investigation Unit, Aberdeen, 1984.

DONHAM, K. Studies on Environmental Exposures. Swine health and engineering design in swine confinement buildings in Southern Sweden. Swedish Work Environmental Fund, Report no. 4, 1986.

GRÜTER, O. Ermittlung des Staubgehaltes der Luft in Schweinemast - und Hüehnerställen in Abhängigkeit von Aufstallungs- Futterungs- und Lüftungssystem.

Institut für Physiologie und Hygiene Landwirtschaftlicher Nutztiere der ETH, Zürich, 1975.

GUSTAFSSON, G. Air and Heat Balance in Animal Houses. Swedish University of Agricultural Scienes. Report no. 59, 1988.

IVERSEN, M. et al. Respiratory Symptoms in Danish Farmers : an epidemiological study of risk factors. Thorax, 43 : 872 - 877, 1988.

NIELSEN, G.D. Beskrivelse af kvælende og irriterende gassers toxikologi. (Description of the Toxicology of Suffocating and Irritating Gases). Farmaceutisk Tidende (Danish) nr. 38, 1984.

NILSSON, C. Dust Investigations in Pig Houses.
Sveriges Lantbruksuniversitet, Swedish University of Agricultural Sciences, Report No. 25, 1982.

NILSSON, C. and GUSTAVSSON, G. Damm i slaktvinsstallar (Dust in Fattening Pig Houses). Swedish University of Agricultural Sciences, Special Report no. 140, 1987.

NIMMERMARK, S. Fukt- og värmeproblem i varphöns- och slaktkycklingsstallar. (Moisture and Heat Production in Animal Houses for Hens and Broilers). Swedish University of Agricultural Sciences, 1984.

NORDSTRØM, G.A. and MCQUITTY, J.B. Manure Gases in the Animal Environment. Research Bulletin 76 - 1. Department of Agricultural Engineering, University of Alberta, 1976.

PEDERSEN, S., TAKAI, H. and HANSEN, K. Ammoniakfordampning fra stalde og lagre (Evaporation of Ammonia from Animal Houses and Stores for Animal Waste). Arbejdsrapport no. 4, Miljøstyrelsen, 1987.

PEDERSEN, S. and TAKAI, H. Ammoniakfordampning fra stalde. Statens jordbrugstekniske Forsøg. Danish Agricultural Engineering Institute, Orientering no. 49, 1987.

PEDERSEN, S. Klimaændringer ved ventilationssvigt i smågrisestalde (Change in Indoor Climate in Case of a Total Stoppage of the Ventilation). Ugeskrift for Jordbrug no. 3, 1986.

ROBERTSON, J.F. The Influence of Aerial Contaminants on the Severity of Pig Respiratory Disease. In "Aerosols, their generation, behaviour and applications". Proc. 2nd Conf. Aerosol Society, 1988.

SÄLLVIK, K., WIGSTRÖM, P. and STEINECK, S. Flytgödsel - ventilation. Royal Agricultural College of Sweden. Report no. 193, 1973.

SCHNEIDER, T. Basisbog i teknisk arbejdshygiejne. Arbejdstilsynet, København, 1986.

TAKAI, H., LARSEN, E. and PEDERSEN, S. Støv og støvmåling i svinestalde (Dust and Measurements of Dust in Pig Houses). Statens jordbrugstekniske Forsøg, Danish Agricultural Engineering Institute, Orientering no. 39,1985.

TAKAI, H. and LARSEN, E. and PEDERSEN, S. Støv og støvmåling i svinestalde II (Dust and Measurements of Dust in Pig Houses II). Statens jordbrugstekniske Forsøg, Danish Agricultural Engineering Institute, Orientering no. 43, 1986.

TAKAI, H. and LARSEN, E. Støv og støvmåling i svinestalde III (Dust and Measurements of Dust in Pig Houses III). Statens jordbrugstekniske Forsøg, Danish Agricultural Engineering Institute, Orientering no. 54, 1988.

TJERNSHAUGEN, O. Gas Concentrations in Housing for Domestic Animals. Housing for Milk Production. Agricultural University of Norway. Report no. 82, 1975.

TJERNSHAUGEN, O. Gas Concentrations in Housing for Domestic Animals, Part II. Housing for Pigs. Agricultural University of Norway. Report no. 83, 1975.

TJERNSHAUGEN, O. Gas Concentrations in Housing for Domestic Animals, Part III. Housing for Laying Hens. Agricultural University of Norway. Report no. 84, 1975.