

AIR MOVEMENT AND HEAT DISTRIBUTION
IN HEATED FACTORY SPACES

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1. Introduction

Traditionally in the UK factory spaces were not generally heated throughout, but rather, heating systems were used to provide an 'umbrella' coverage of heat over a certain occupied area. The building fabric was usually poorly insulated and 'leaky' and so it did little to contain the delivered heat within the space, nor did it contribute a great deal to the thermal environment other than to provide basic shelter from the elements. After heat had been delivered to the prescribed area of coverage it was considered more or less lost, i.e. if one was outside this umbrella of heat cover one expected to be cold. In this way the thermal environment was considered to be 'emitter dominated' with the heat emitter being either predominantly radiative or convective in its delivery of heat to the space.

For modern low energy buildings in general the thermal environment is not emitter dominated. Once the building has been 'warmed up' the heating system should play a secondary role to the fabric in maintaining thermal comfort conditions. Modern factories are now designed to be better insulated, at least to the UK Building Regulations recommended levels (1), and are often better sealed in terms of air infiltration rates (2). The fabric can now contain the heat after it has been delivered to the space. However the method of heating system design has not changed generally in concept from that of providing heat coverage, or 'emitter dominated' areas. Many systems, both radiative and convective are not designed to operate, within the comfort region, with the necessary degree of modulation required in 'low energy' factories. Therefore although modern factory heating systems claim to be efficient in converting fuel to heat, they often prove inefficient in the ultimate delivery of heat to the space. This can result in poor environmental conditions and energy waste.

This paper describes the air movement and heat distribution measured in low energy factories which had installed a range of heating systems currently popular in use in the UK, and discusses the results in terms of thermal performance and thermal comfort.

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2. Monitoring Heating System Performance

Over the past three years a number of low-energy factories have been monitored in a field measurement project the aim of which was to demonstrate the energy saving benefits resulting from improved levels of thermal insulation and lower air infiltration rates (2,3). The main aim of the project was to assess the fabric performance, however, the opportunity was taken to carry out additional experiments in order to assess the heating system performance in relation to the 'low-energy' fabric design. The factories, which were predominantly naturally ventilated, and in the size range 180 to 1800 sq.m. floor area, had heating systems which were considered to be representative of current 'state of the art' design for the UK. They are presented in table 1. All were fueled by gas.

Type of system	Control	Floor area (sq.m.)
Unit Warm Air Heater	on/off*	180
Radiant (Black) Tubes	on/off	900
Radiant MPHW Panels	modulating	1800
Radiant Plaques	on/off	180

* Later changed to modulating control

Table 1. Monitored heating systems.

For all the heating systems in table 1, measurements were carried out of temperature gradients with time, during the heating season. Additional measurements were carried out on the unit warm air heater and the radiant tubes. These including gridded space air velocity and temperature measurements (using a Disa Multi-Channel Air Flow Analyser and smoke visualisation), and boundary condition measurements, which for the unit air heater included delivery air temperatures and velocities, and for the radiant tubes, tube surface temperatures.

3. Heat Requirements of Modern Factories

In modern factories in the UK the fabric heat loss has been reduced considerably over the past decade. Typical seasonal energy targets are 0.35 GJ/sqm/annum, and typical design fabric heat loss targets are 60 W/sqm (for a DT of 20 degC) (3). A typical daily space heating energy demand profile is presented in figure 1 for a factory heated by a warm air heater with a modulating control (see also section 4.1). This case serves to illustrate the requirements placed on the design of industrial

heating systems for modern factories. Besides having the capacity to warm the building up 'from cold', a system should have adequate control to deal with variations in demand on both a daily and seasonal basis. In addition, it should deliver and distribute the heat in such a way as to provide comfort in the occupied regions in an efficient manner.

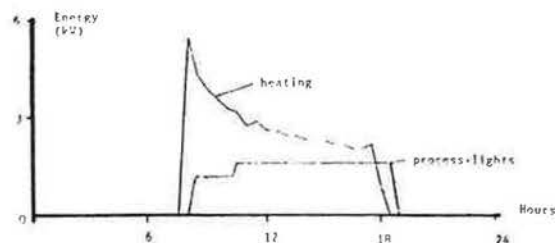


Fig.1 Typical daily heat demand profile and process heat gain.

4. Case Studies

The case studies for the various heating system options (table 1) are presented in terms of the ability of each system to control the heat supply to the space and also of the way in which heat is delivered and distributed to where it is needed.

4.1 Unit Warm Air

The unit warm air heater option was first operated with an on/off thermostat air temperature control situated on the wall just below the heater unit, which was wall mounted. This mode of operation resulted in wide variations of temperature with time, with floor to ceiling temperature gradients peaking as high as 20 degC, as shown in the first four days of figure 2. The space was effectively being heated 'from the ceiling down' and there were complaints from occupants of low temperatures at floor level in relation to the high temperatures at upper levels.

In order to improve matters a modulating control was installed which modulated the gas pressure (down to a minimum of 30% full load) based on return air temperature. The improvement in air temperature profiles with time can be seen in the last three days of figure 2. The ability to modulate to match the variation in load conditions, eg. with the occurrence of process heat gains, was demonstrated on a daily basis in figure 1, which presents typical daily energy profiles for a day after the installation of the modulating control. The improved profiles were related to the reduction in buoyancy forces resulting from the reduced heater leaving air temperatures. This was demonstrated in the air flow vector measurements presented in figures 3 (a) to (c) for three heat output levels.

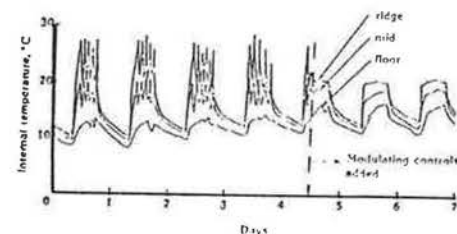


Fig.2 Floor (0.3m) to ridge (5.0m) temperature gradients for a 7 day period during which the controls were changed from on/off to modulating.

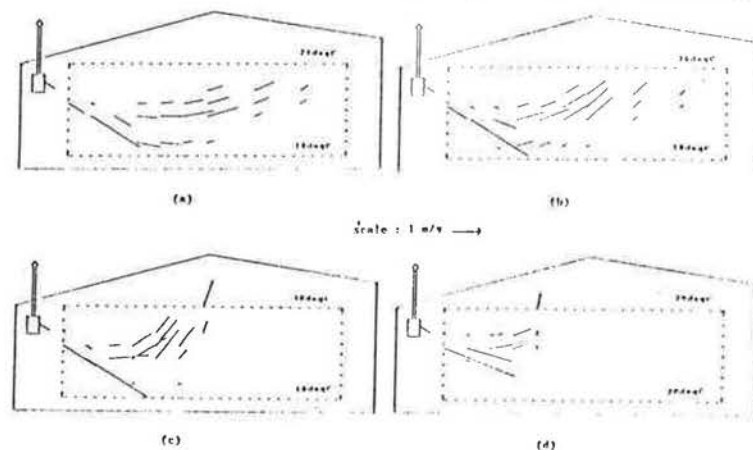


Fig.3 Air flow vectors for three heater temperature settings, namely (a) 25kW, (b) 20kW and (c) 15kW, and (d) for a reduced leaving speed. (maximum temperature gradients are included for the dotted region)

Because the heater operates for much of the time at reduced load the buoyancy forces were weaker and the temperature profiles were more 'stable'. However it did mean that the air movement in the space was often high relative to the air temperature on low 'turn down' and there were complaints of draughts. The system was designed on the concept of supplying a particular area of heat coverage, and therefore, on this basis required a jet velocity sufficient to reach the far side of the factory. In fact the system was oversized in terms of its capacity, in order to achieve the required jet 'throw'. When the heater was modulated on leaving velocity (reduced by about half) as well as temperature, as shown in figure 3(d), the space temperatures were still found to be satisfactory but now the air velocities were reduced. The space became less dominated by the system, and potentially more comfortable.

4.2 Radiant (Black) Tubes

The radiant tube option had an on/off thermostat air temperature control and, as presented in figure 4, it showed a similar air temperature profile over time as for the warm air system in the on/off control mode of operation. In the case of the radiant tubes the floor to ridge air temperature gradient was about 8 degC, being steeper in the upper region (between 3m and 6m).

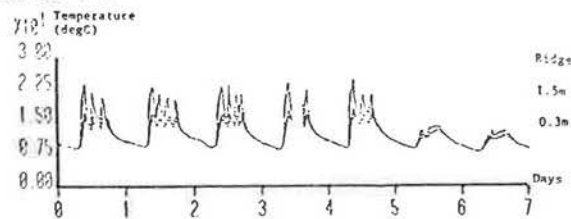


Fig.4 Air temperature profiles for the radiant tube heating system, for a 7 day period, with measurement points at 0.3m, 1.5m, 3.0m and 6m (ie. ridge height)

This implies that there was a significant convective output from the tubes. Measurements of the tube surface temperature along its length were carried out and from these measurements the radiant input to the space was estimated to be 55 % of the total heat input, which is not as high as usually assumed for this type of system.

Measurements of air velocities and air temperatures were carried out at gridded points throughout the space. Velocities were found to be generally less than 0.1 m/s, the pattern of air movement being generated by the relatively weak natural buoyancy forces. Floor to ridge temperature gradients were typically about 8 degC. Air movement in this factory was therefore considered not to be a source of discomfort. The only source of possible discomfort were the 'excessive' variations of temperature with time resulting from the on/off control.

4.3 Radiant MPHW Panels

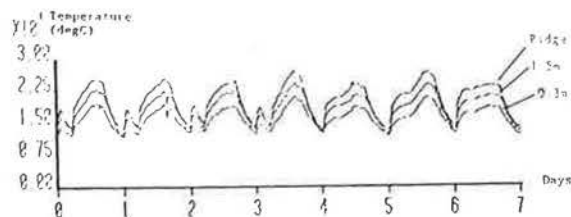


Fig.5 Air temperature profiles for mphw radiant panel heating system.

These were installed in the largest factory investigated, 1800 sqm floor area. The radiant mphw panels option had a modulating control fitted as standard and, as the profiles of figure 5 show, the air temperatures were more stable with time compared to the previous radiant tube option. The radiant efficiency was not very high with the radiant component contributing about 40 % of the total heat delivered to the space. However the factory was able to contain the heat once it had been delivered and perhaps a high radiant efficiency is not necessary. This, together with the modulating control means that the system tends towards a non-emitter dominant mode of operation, and is generally considered to be able to provide a comfortable thermal environment.

4.4 Radiant Plaques

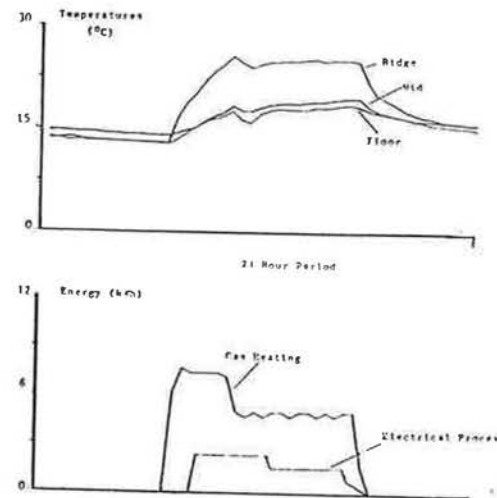


Fig.6 Daily profiles for radiant plaques

The radiant plaque option was chosen for the smaller factories for which there were difficulties at the time in obtaining heating systems to match the (low) heating requirements. Radiant plaques can be distributed about the space to provide a traditional type of heat coverage concentrated around work positions and in sufficient number to match the fabric heat loss. The plaques were controlled by on/off temperature thermostats. However, during occupancy they were usually controlled manually, with operatives turning individual plaques on or off as required. A typical daily profile is presented in figure 6. What was of interest in the use of the plaques was that the operatives tended to turn off the plaques over the work positions, and used the plaques away from the work positions to provide heat to the space. This is contrary to traditional concept of providing heater coverage and again favours the non-emitter dominant mode of operation.

5. DISCUSSION

From the above case studies a number of points arise for discussion related to the heating of modern low energy factories and the suitability of traditional heating system design.

5.1 Control

The least expensive control option is the on/off thermostat, and therefore it is often specified for many system options. In fact a number of currently available systems will only operate in an on/off mode. This dictates a period of maximum heat input followed by a period of no heat. The temperature gradients during this cycling depend on the nature of heat delivery in relation to the buoyancy forces. Because the heat input to the space is always at the maximum level buoyancy forces will be always at their highest which will result in high temperature gradients. This has been shown to be the case for radiant systems as well as for convective systems. In low-energy factories where the heating system is still sized on design heat loss although it operates for the most time at part load the adverse effects of on/off control tend to be greater, as a greater component of the heat requirements are met by incidental heat gains, eg. from process heat.

The modulating control matches the heating system output closer to the (varying) heat requirements of the space and therefore provides a more stable and potentially more comfortable environment and tends to reduce temperature gradients.

5.2 Distribution

The case studies have indicated that for modern low-energy factories the direct distribution of heat by the heating system in terms of providing an area of heat coverage is not as important as delivering the right amount of heat to match the heating needs at a particular time. The fabric of the building is able to contribute to the thermal environment and the heat contained by the building fabric will be distributed by 'natural' forces. Radiant efficiency and jet throw are not considered to be such important factors in heating system design in modern factories, indeed, a non-emitter mode of heat delivery is considered to be potentially more comfortable.

Although temperature gradients can be reduced by using a modulating control they will still occur even with radiant systems. There are destratification options available which claim to reduce and even to eliminate temperature gradients, though they have not been tested during these studies.

5.3 Thermal Comfort

Comfort is affected by heating system control (excessive variations of temperatures with time) and the method of heat delivery and distribution

(in terms of draughts from warm air systems or hot spots from radiant systems). Comfort conditions are potentially easier to achieve when the environment is not dominated by the heating system.

6. CONCLUSIONS

The success of increased levels of insulation in reducing space heating requirements for modern factories has resulted in low rates of heat demand being required from the heating system for much of the occupied period.

Modulation is therefore preferred to on/off control, in order to match the reduced heat requirements and to avoid excessive temperature variations with time.

The 'traditional rules' concerning the provision of areas of heat coverage for both warm air and radiant systems are not directly applicable for low energy factories. Heat need not be directed at people, in an emitter dominant manner. In fact, best comfort conditions seem to occur when the heat is delivered indirectly to the occupied zone in a non emitter dominant mode of operation.

There are still energy inefficiencies due to temperature gradients for both radiant and warm air systems, though they should be reduced by good design.

In the short term, current heating system options can be used in low energy factories with reasonable success provided they have modulating control and they are non-emitter dominant. However, in the longer term there is a need for new options which are better matched to the requirements of modern factories.

7. REFERENCES

1. UK Building Regulations, Approved Document L1, HMSO, 1986.
2. Etheridge D.W., Jones P.J., and O'Sullivan P.E., Ventilation of Factories, Proc. 6th AIC, 11.1-11.19, Netherlands, 1985.
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MESURES IN SITU DU COEFFICIENT MOYEN
DES ECHANGES SUPERFICIELS INTERIEURS*

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1. Introduction

Au cours des années 1984-1986, nous avons réalisé un programme d'essais sur 6 bâtiments situés dans l'Italie du Nord (Alba, Parma, Schio, Tortona) et dans l'Italie du Sud (Campobasso, Lucera).

On a utilisé 38 logements, pour un total de 150 locaux (1,2,3), et on a mesuré: les hétérogénéités des températures superficielles des parois de façade, à l'aide de la thermographie infrarouge; les résistances thermiques des parois les plus significatives; les paramètres microclimatiques des locaux (température, humidité relative et vitesse de l'air; température rayonnante moyenne des parois); les facteurs de lumière du jour des locaux significatifs; l'isolement acoustique normalisé des parois et des éléments de façade, l'isolement normalisé des planchers aux bruits aériens et aux bruits de chocs, l'isolement des bruits des équipements. Les mesures acoustiques ont été effectuées par l'Istituto di Fisica Tecnica della Facoltà di Ingegneria di Bologna.

Nous avons saisi l'occasion pour vérifier la fiabilité des valeurs des coefficients d'échanges superficiels, ainsi que des méthodes pour les calculer.

Cette communication concerne le coefficient d'échanges superficiels intérieurs.

* Conventions de recherche Dipartimento di Energetica del Politecnico di Milano/Ispreidil S.p.A. di Roma 03.02.1984 et 31.10.1984, financées par l'Administration (Comitato Edilizia Residenziale (CER), art. 2, lettre F, Loi 457/78).

2. Position du problème

2.1. Les conditions aux limites de l'équation indéfinie de la conduction de la chaleur sont généralement linéarisées, où coexistent des échanges thermiques par convection et par rayonnement, avec des relations telles que (4):

$$h \cdot (t_f - t_s) = -k \cdot (\partial t / \partial n)_s \quad (1)$$

avec n normale à la surface, orientée vers l'intérieur du corps, et

$$h = h_c + h_r \quad (2)$$

avec

$$h_c = 0,56 \cdot (k/l) \cdot (Gr \cdot Pr)^{1/4} \quad \text{pour } (Gr \cdot Pr) = 10^4 \div 10^8 \quad (3.a)$$

$$h_c = 0,13 \cdot (k/l) \cdot (Gr \cdot Pr)^{1/3} \quad \text{pour } (Gr \cdot Pr) = 10^9 \div 10^{12} \quad (3.b)$$

pour la convection naturelle et pour les surfaces planes verticales (5,6,7);

$$h_r = \sigma \cdot f(e_s, e_p, S_s, S_p, F_{sp}) \cdot (T_p^4 - T_s^4) / (t_f - t_s) \quad (4)$$

avec e_s, e_p les émissivités des surfaces S_s et S_p supposées grises, F_{sp} le facteur de forme.

2.2. La fonction $f(e_s, e_p, S_s, S_p, F_{sp})$ est exprimable par:

$$f(e_s, e_p, S_s, S_p, F_{sp}) = 1 / (1/e_s + (1/e_p - 1) S_s/S_p) \quad (5)$$

où les surfaces soient supposées à température uniforme et le facteur de forme F_{sp} est égal à 1 pour la géométrie de la figure 1.

2.3. Il s'ensuit que le flux thermique qui traverse la paroi suivant la normale en régime stationnaire ainsi que la densité du flux thermique sont donnés par:

$$\Phi = h \cdot (t_f - t_s) \cdot S_s \quad (6)$$

et

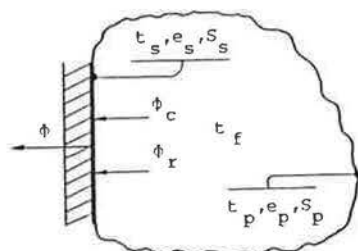


Fig. 1. Schéma des flux thermiques à la surface intérieure de la parois

$$\varphi = h \cdot (t_f - t_s) \quad (7)$$

3. Mesures in situ

3.1. La (7) a été utilisée pour déterminer les coefficients d'échanges superficiels par la mesure directe in situ de la densité de flux thermique et des températures t_f et t_s ; le schéma du système de mesure est représenté par la figure 2. Le même schéma a été utilisé pour mesurer la résistance thermique des parois.

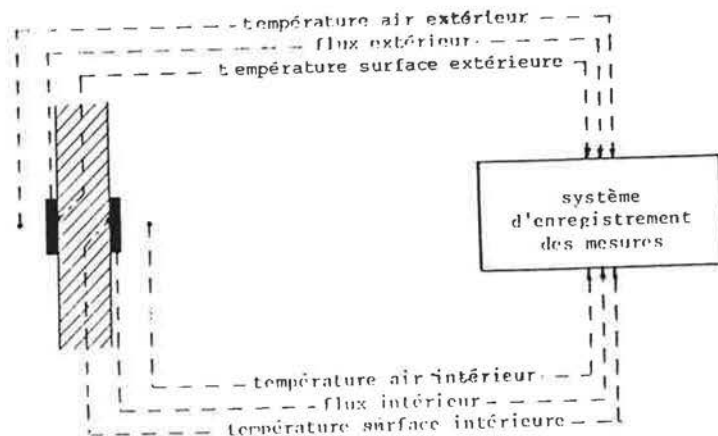


Fig. 2. Chaîne de mesure

3.2. La stationnarité du flux thermique a été vérifiée par des enregistrements de longue durée des températures de l'air intérieure et extérieure, des températures surfaciques intérieures et extérieures et du flux thermique même. Les mesures ont été effectuées au cours des séquences froides les plus longues et au moins pour trois jours consécutifs.

Les zones de mesure à flux unidirectionnel, qui ont été choisies sur parois exposées pour la plupart au nord, ont été localisées avec la thermographie infrarouge associée à des mesures ponctuelles des températures surfaciques. La thermographie infrarouge a été utilisée encore, les fluxmètres installés, pour vérifier la conservation de la unidirectionnalité du flux thermique.

3.3. Le projet du système de mesure a été optimisé par l'analyse des erreurs; l'approximation poursuivie pour la mesure de h a été de $\pm 10\%$.

L'expression de l'erreur probable

$$\Delta h/h = \pm \sqrt{(\Delta\varphi/\varphi)^2 + (\Delta(t_f - t_s)/(t_f - t_s))^2} \quad (8)$$

a suggéré, pour les mesures in situ, des approximations de $\pm 4\%$ pour le flux thermique et des approximations de $\pm 8\%$ pour la différence de température entre l'air et la paroi.

L'excellent accord entre les résistances thermiques des parois, qui ont été mesurées simultanément au coefficient h , et les résistances calculées, représente une vérification indirecte du niveau de fiabilité des mesures des susdits coefficients d'échange superficiels.

4. Résultats

Les résultats sont résumés dans les tableaux 1, 2, 3 et 4.

4.1. Le tableau 1 concerne les valeurs du coefficient de convection h_c calculées pour les températures mesurées; la hauteur de la paroi est la même pour tous les bâtiments: $L = 2,7$ m.

Le tableau 2 concerne les valeurs du coefficient de rayonnement h_r , calculées pour les températures T_f et T_s mesurées, pour les géométries de chaque local et sous les hypothèses suivantes:

- $T_p = T_f$, sauf pour le bâtiment A où la valeur de T_p a été mesurée car il est chauffé par rayonnement (1/3 au plafond, 2/3 au plancher);
- $e_p = e_s = 0,9$.

Tableau 1. Valeurs de h_c calculées pour les températures mesurées et pour $L = 2,7$ m

bât	mes	t_f	t_s	t	T_m	$k \cdot 10^2$	$v \cdot 10^6$	Pr	Gr · Pr · 10 ⁻⁹	h_c
A	1	24,7	23,1	1,6	297,1	2,628	15,50	0,702	3,037	1,832
B	2	29,0	27,5	1,5	301,4	2,662	15,91	0,701	2,660	1,776
B	3	28,4	26,0	2,4	300,4	2,657	15,85	0,701	4,303	2,081
C	4	23,5	19,9	3,6	294,6	2,618	15,39	0,702	6,984	2,409
D	5	20,9	18,4	2,5	292,8	2,597	15,14	0,703	5,055	2,146
D	6	16,6	14,7	1,9	288,8	2,563	14,75	0,704	4,109	1,977
D	7	20,5	17,7	2,8	292,3	2,594	15,11	0,703	5,693	2,230
E	8	19,4	17,5	1,9	291,6	2,585	15,01	0,703	3,924	1,963
F	9	21,9	20,4	1,5	294,3	2,605	15,24	0,703	2,978	1,805
F	10	18,8	17,6	1,2	291,4	2,580	14,95	0,703	2,500	1,686

Tableau 2. Valeurs de h_r calculées pour les températures mesurées T_f et T_s ainsi que pour les géométries de chaque local.

bât	mes	T_f	T_s	T_p	$c_p = c_s$	$\frac{S_s}{S_p}$	h_r
A	1	297,9	296,3	299,2	0,9	0,200	9,574
B	2	302,2	300,7	302,2	0,9	0,386	5,385
B	3	301,6	299,2	301,6	0,9	0,187	5,431
C	4	296,7	293,1	296,7	0,9	0,175	5,145
D	5	294,1	291,6	294,1	0,9	0,142	5,054
D	6	289,8	287,9	289,8	0,9	0,142	4,850
D	7	293,7	290,9	293,7	0,9	0,142	5,026
E	8	292,6	290,7	292,6	0,9	0,205	4,962
F	9	295,1	293,6	295,1	0,9	0,327	5,041
F	10	292,0	290,8	292,0	0,9	0,145	4,978

4.2. Le tableau 3 concerne les valeurs calculées du coefficient d'échanges superficiels, les valeurs mesurées ainsi que les écarts en pourcentage.

Les écarts sont modestes, en moyenne moins du 7% environ, pour les mesures 1, 2, 4, 8, 9; ils sont remarquables et incompatibles avec l'hypothèse $T_p = T_f$ pour les mesures 3, 5, 6 et 7.

La mesure des paramètres microclimatiques, effectuée simultanément aux mesures de résistance thermique, avait signalé pour les bâtiments objet des mesures 5, 6 et 7, des humidités relatives particulièrement élevées

et des températures radiantes moyennes particulièrement basses par rapport aux autres bâtiments; pour le bâtiment objet de la mesure 3, la sus-dite mesure avait signalé une température radiante moyenne anormalement basse.

Pour ces bâtiments la chute des échanges thermiques par rayonnement était, par conséquent, prévisible, ainsi que la surestimation des échanges calculés suivant l'hypothèse $T_p = T_f$ (tableau 2).

Pour les bâtiments objet des mesures 5, 6 et 7 il nous a paru raisonnable attribuer la diminution de la température T_p à l'évaporation des eaux de construction et par conséquent de supposer T_p égale à la température humide de l'air, évidemment pour les seules surfaces poreuses. Les valeurs des coefficients d'échanges superficiels ainsi recalculés confirment cette hypothèse (tableau 4). Dans le cas des mesures 5 et 7 toutes les surfaces sont poreuses sauf celle du plancher; dans le cas de la mesure 6 seulement le plafond est poreux.

Tableau 3. Valeurs de h calculées, valeurs de h mesurées, écarts.

bât	A	B	B	C	D	D	D	E	F	F
mes	1	2	3	4	5	6	7	8	9	10
h_{cal}	11,41	7,16	7,51	7,55	7,20	6,83	7,26	6,93	6,85	6,66
h_{mes}	10,35	7,76	5,02	6,87	3,63	4,92	3,86	6,27	6,79	6,44
$\frac{h_{cal} - h_{mes}}{h_{mes}}$ (Z)	10,2	-7,7	50,4	9,9	98,3	38,8	88,1	10,5	0,9	3,4

Pour le local objet de la mesure 3, la basse température radiante moyenne est due à la contiguïté de certaines parois avec les parties non chauffées du bâtiment. Evidemment aucune hypothèse généralisable est possible sur cette température.

5. Conclusions

5.1. Les valeurs calculées et celles mesurées du coefficient d'échanges superficiels intérieurs se recourent de façon satisfaisante pour des conditions d'essai cohérentes avec les hypothèses de calcul et dans le cas des géométries les plus courantes des locaux des logements.

La moyenne des valeurs calculées, égale à 7,03 $w/m^2 \cdot K$, est en effet en bon accord avec celle des valeurs mesurées qui vau 6,81 $w/m^2 \cdot K$.

Tableau 4. Calcul du coefficient h_r supposant qu'une partie de la surface S_p soit à la température humide de l'air; comparaison entre les valeurs calculées et les valeurs mesurées de h .

mes	5	6	7
T_f	294,1	289,8	293,7
$T_{f, \text{hum}}$	291,6	285,9	291,2
T_s	291,6	287,9	290,9
T_s^p	292,2	288,9	291,8
e^p	0,9	0,9	0,9
e^s	0,94	0,91	0,94
S^p/S	0,142	0,142	0,142
h^s_p	1,210	2,432	1,612
h^r	3,36	4,41	3,84
h^{cal}	3,63	4,92	3,86
h_{mes}			
$\frac{h^{\text{cal}} - h_{\text{mes}}}{h_{\text{mes}}} (\%)$	-7,4	-10,4	-0,5

Ces valeurs sont toutefois un peu moindre de celles les plus répendues car les conditions d'essai sont différentes des conditions réelles des logements habités.

5.2. Les échanges thermiques par convection sont nettement moins importants de celles par rayonnement; leurs variations avec les températures observées pendant les essais, influencent peu la valeur du coefficient d'échanges superficiels.

Le rayonnement vau presque 70% de ce coefficient. Par conséquent la réduction de ces échanges peut contribuer sensiblement à limiter les déperditions thermiques des parois. Le mécanisme consiste dans la réflexion de la chaleur radiante à la surface des parois, avant d'en réduire la conduction avec des isolants thermiques.

6. Symboles et unités de mesure

6.1. Nomenclature

A	bâtiment de Alba
B	bâtiment de Campobasso
C	bâtiment de Lucera

c_p	(J/kg K)	chaleur spécifique à pression constante
D		bâtiment de Parma
E		bâtiment de Schio
e	(-)	émissivité
F		bâtiment de Tortona
F_{sp}	(-)	facteur de forme
$Gr = (L^3 g \Delta t) / (T_m \nu^2)$	(-)	numéro de Grashof
$g = 9,80665$	(m/s ²)	accélération de gravité
h	(W/m ² K)	coefficient d'échanges superficiels
k	(W/m K)	conductivité thermique
$L = 2,70$	(m)	hauteur de la paroi
$Pr = (\mu c_p) / k$	(-)	numéro de Prandtl
S	(m ²)	superficie
T	(K)	température absolue
$T_m = (T_p + T_s) / 2$	(K)	température moyenne de l'air à la surface
t	(°C)	température Celsius
$\Delta t = t_f - t_s$	(°C)	différence de température air-surface

6.2. Lettres grèques

Λ		différence
ν	(N s/m ²)	viscosité
μ	(m ² /s)	viscosité cinématique
$\sigma = 5,66961 \cdot 10^{-8}$	(W/m ² K ⁴)	constante de Stefan-Boltzman
ϕ	(W)	flux thermique
ρ	(W/m ²)	densité de flux thermique

6.3. Indices

c	convection
f	fluide
p	parois intérieures
r	rayonnement
s	surface intérieure de la paroi extérieure

6.4. Abrégés

bât	bâtiment
cal	calculé
hum	humide
mes	numéro de la mesure; mesuré

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FIELD MEASUREMENT AND ENERGY AUDITING

Field measurement

Audit methods and results

Field measurement

AN APPLICATION OF KALMAN FILTERING TO ENERGY MONITORING OF BUILDINGS

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1. Introduction

Analysis of monitoring data from buildings using simple physical models, e.g., energy signatures (ES), is a wellknown method. For a given energy balance model, the ES denotes a set of parameters describing the energy performance of a building. The values of the parameters are estimated from a statistical analysis of the building's performance. In most cases the method of least squares is used. The number of parameters depends on the energy balance model used and may vary from one to eight or more. The models can be either static (1), or dynamic (2).

A basic assumption in most ES models is that a building's performance can be described as a constant-parameter linear system. Consequently, the models are normally identified off-line with a method giving equal weight to data from each point of time, i.e., a complete set of data is analyzed, assuming that the parameters have the same values over the whole measurement period.

The assumption of constant parameters may be violated because of variation in climate variables, not included in the analysis, e.g., a strong gale may cause increased air infiltration and thus increase the energy loss factor. Changes in user habits or malfunctions in heating and ventilation systems may also result in changes of parameter values. It would be useful to detect such changes in energy monitoring of buildings. An inspection of detected changes in parameter values may give a warning that some action should be taken.

Recursive on-line algorithms (3) with a forgetting factor or a rectangular window included, may be used for estimation of the ES of buildings. This makes it possible to continuously track changes in the values of ES-parameters.

2. Theoretical model

The building is modelled by a number of nodal points connected by thermal resistances and capacitances, giving a lumped parameter model. Two simple, but for the present purposes sufficient, models are given in Figure 1.

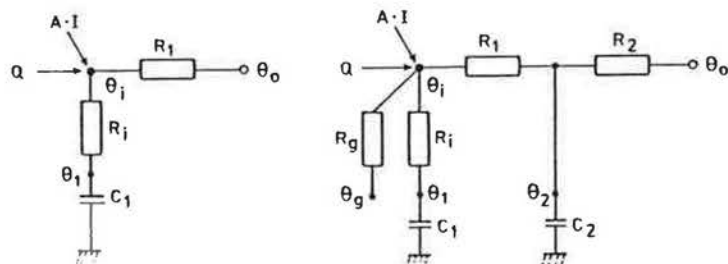


Fig. 1. Thermal networks of a 4-parameter and a 7-parameter model for a building.

The ventilation losses are modelled together with the total conduction losses in one thermal resistance in the 4-parameter model and with two in the 7-parameter model. The thermal capacitance C_1 is transferring its energy to the internal air node through a resistance R_i . In the 7-parameter model, the thermal capacitance C_2 transfers its energy to the internal air and outdoor air nodes. The solar aperture A is equal to an effective aperture area without any losses. These models and variants of them have been discussed by several researchers (2, 4, 5, 6, 7).

We will here discuss the 4-parameter model. Analysis with the 7-parameter model may be carried out in an analogous way (7). The governing differential equations for the 4-parameter model is

$$Q + AI = \frac{1}{R_i} (\theta_i - \theta_1) + \frac{1}{R_1} (\theta_i - \theta_o) \quad (1)$$

$$C_1 \frac{d\theta_1}{dt} = \frac{1}{R_i} (\theta_i - \theta_1) \quad (2)$$

in the five variables

- θ_i = Indoor temperature (measured) [$^{\circ}\text{C}$]
- θ_1 = Temperature of the capacitance (unmeasured) [$^{\circ}\text{C}$]
- θ_o = Outdoor temperature (measured) [$^{\circ}\text{C}$]
- Q = Heat from auxiliary heating (measured) [W]
- I = Solar radiation on the south wall (measured) [W/m^2]

and in the four parameters R_i [$^{\circ}\text{C}/\text{W}$], R_1 [$^{\circ}\text{C}/\text{W}$], C_1 [$\text{kWh}/^{\circ}\text{C}$] and A [m^2].

3. State space formulation

A state space model consists of a measurement equation

$$y_t = z_t' \alpha_t + \varepsilon_t \quad t = 1, 2, \dots, T \quad (4)$$

and a transition equation

$$\alpha_t = T_t \alpha_{t-1} \quad t = 1, 2, \dots, T \quad (5)$$

where α is a $m \times 1$ state vector containing the model parameters and the latent (unmeasured) variables, z_t is a $m \times 1$ vector containing the measured variables and lagged latent variables, T_t is a matrix of order $m \times m$, ε_t is a normally and independently distributed disturbance term with variance σ^2 .

Consider the equation system (1)-(2) above. With a time index on the variables, and using a first order difference approximation, we have

$$\frac{1}{R} (\theta_{it} - \theta_{ot}) = \frac{1}{R_i} (\theta_{1t} - \theta_{ot}) + Q_t + AI_t \quad (6)$$

$$\theta_{1t} = \theta_{1t-1} + \frac{\Delta t}{R_i C_1} (\theta_{it-1} - \theta_{1t-1}) \quad (7)$$

where

$$\frac{1}{R} = \frac{1}{R_i} + \frac{1}{R_1} \quad (8)$$

Inserting equation (7) into equation (6) gives

$$\frac{1}{R} (\theta_{it} - \theta_{ot}) = \frac{1}{R_i} (\theta_{1t-1} - \theta_{ot}) + \frac{\Delta t}{R_i C_1} (\theta_{it-1} - \theta_{1t-1}) + Q_t + AI_t \quad (9)$$

which may be used as the measurement equation (4), where y_t is the indoor-outdoor temperature difference $\theta_{it} - \theta_{ot}$, and where

$$z = \begin{bmatrix} z_{1t} \\ z_{2t} \\ z_{3t} \\ z_{4t} \\ z_{5t} \end{bmatrix} = \begin{bmatrix} \theta_{1t-1} & -\theta_{ot} \\ \theta_{it-1} & -\theta_{1t-1} \\ Q_t \\ I_t \\ 0 \end{bmatrix}; \quad \alpha = \begin{bmatrix} \alpha_{1t} \\ \alpha_{2t} \\ \alpha_{3t} \\ \alpha_{4t} \\ \alpha_{5t} \end{bmatrix} = \begin{bmatrix} R/R_i \\ R \Delta t / (R_i^2 C_1) \\ R \\ RA \\ \theta_{1t} \end{bmatrix} \quad (10)$$

If we assume that the parameters have constant values, the matrix T_t of the transition equation (5) becomes equal to the identity matrix except at the fifth row and second column, where the element takes the value $z_{2t}' \alpha_{1t-1}$.

4. A Kalman filter

The Kalman filter works sequentially. It consists of a sequence of predictions of the state vector. These predictions are up-dated as additional measurements arrive. The prediction equations of the ordinary Kalman filter may be written as

$$\hat{a}_t = T_t a_{t-1} \quad (11)$$

$$\hat{p}_t = T_t' p_{t-1} T_t \quad (12)$$

where a_t is the predictor of the state vector with covariance matrix $\sigma^2 \hat{p}_t$ of the prediction errors, and where a_{t-1} is the estimator of the state at time $t-1$ with covariance matrix $\sigma^2 p_{t-1}$.

When measurements at time t are available the predictions (11) and (12) are updated using

$$a_t = \hat{a}_t + \hat{p}_t z_t' (y_t - z_t' \hat{a}_t) / r_t \quad (13)$$

$$p_t = \frac{1}{\lambda} (\hat{p}_t - \hat{p}_t z_t' z_t' \hat{p}_t) / r_t \quad (14)$$

$$\text{where } r_t = z_t' \hat{p}_t z_t + \lambda \quad (15)$$

and where λ is a forgetting factor. If this factor is put equal to 1, all measurements will have the same weight. A λ -value less than one gives an exponential damping of older measurements.

Another way to discount old data is to put λ equal to 1 but let current estimates be based on a fixed number of the most recent measurements. When updating at time t , the impact of the measurements at time $t-n$ are disregarded. Updating at time t is thus based on recursions starting at time $t-n+1$. Such an algorithm is known as a rectangular window.

We can not use the above algorithm exactly since our model is nonlinear in the parameters. The transition matrix T has elements which are functions of the α 's. To overcome this nonlinearity we first linearize the product $T(\alpha) * \alpha$ in (5) around α . This linear system is then used to obtain a covariance matrix of a_t replacing (12). An algorithm of this type is known as an extended Kalman filter. This reasoning holds when the forgetting factor λ is put equal to 1. With λ -values less than 1, the sampling properties of the parameter estimates a_t is difficult to evaluate.

In order to improve the estimates for the first measurements, calculated thermal parameters have been used to obtain starting values a_0 .

5. Application to solar test cell data

A generally applicable method for on-line testing of passive solar components is currently being developed at the Joint Research Center of the European Community in Ispra, Italy. The solar aperture of different windows installed in standardized test cells has been evaluated with the Kalman filtering technique (7).

We consider different configurations of the south wall component of one of the cells. The glazed area of the component is 4.8 m^2 . Global radiation on the south wall, auxiliary heating, indoor and outdoor temperatures were measured and mean values stored for each 15 minutes. The selected measurement period starts at December 4th and ends at December 30th 1985. During this period the following experimental set-ups were used:

Set-up 1. Partly unshielded window. Aluminium sheets are attached to the bottom part of the window.

Set-up 2. Unshielded window. On December the 11th the aluminium sheets were removed.

Set-up 3. Partly insulated window. On December the 16th the lower part of the window was insulated by 30 centimeter polyurethane.

We estimate the parameters in the 7-parameter model shown in Figure 1. First, separate estimations were carried out for each experimental set-up. The solar aperture increases from 1.8 to 2.4 m^2 when the aluminium sheets were removed. The solar aperture then decreased to 1.6 m^2 when part of the window was insulated. These results are consistent with the calculated values. The solar aperture appears however to be somewhat underestimated.

Secondly, we use the data for a study of the technique described for detection of time varying parameters. Suppose we do not know the changes in the solar aperture resulting from the changes in the window area. To see whether these changes may be detected we use the Kalman filter with a rectangular window of four days (384 data points). For comparison we also apply the Kalman filter without a rectangular window.

The results are shown in Figure 2. The rectangular window makes the changes more pronounced. The figure also shows that the Kalman filter with a window gives estimates which are closer to those obtained for the three experimental set-ups separately.

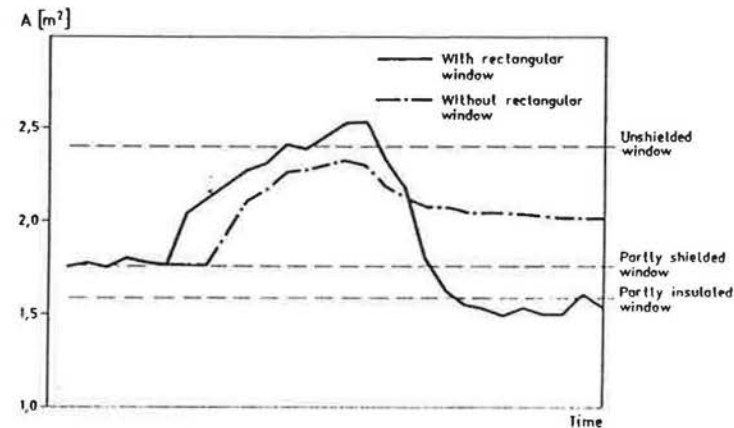


Fig. 2. Recursive estimates of the solar aperture. The horizontal dashed lines indicate the final estimates in the three cases.

6. Application to peak load analysis in occupied houses

Nine single-family houses situated in different parts of Sweden are currently being monitored by The Swedish Building Research Institute in Gävle. The objective is to develop methods for peak-load predictions, based on short-term monitoring of houses. Energy for heating and domestic electricity, indoor and outdoor temperatures are measured and mean values stored for each 30 minutes. Pyranometers have not yet been installed so we must here resort to a 3-parameter model without a solar aperture parameter. This is not a serious shortcoming, since the analyzed data are collected during midwinter when the solar gains are very small.

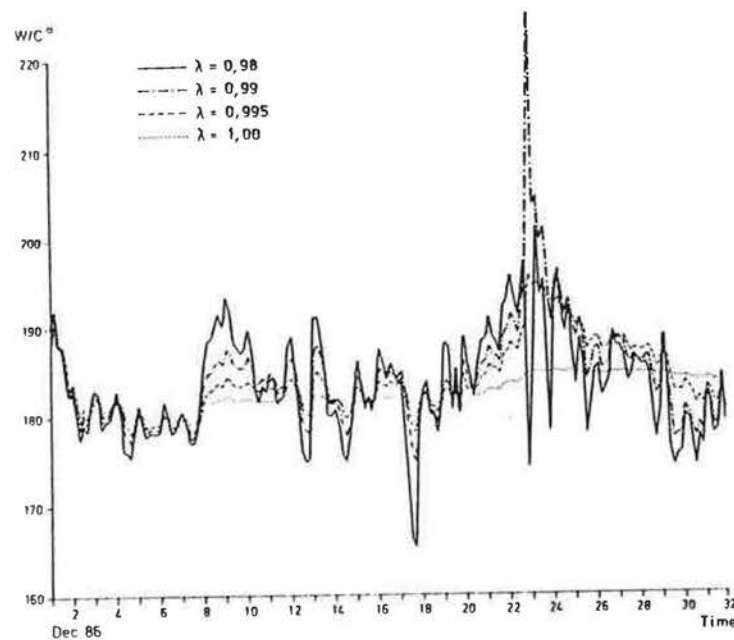


Fig. 3. Heat loss factor estimated with a Kalman filter using different forgetting factors.

Kalman filtering has been applied with the purpose of studying model performance with different forgetting factors ($\lambda = 0.98, 0.99, 0.995$ and 1.00). The total energy loss factor ($1/R$) is shown in Figure 4 for one of the houses with data from December 1986. The house is situated in northern Sweden (Lat 65°) and the outdoor temperature varied considerably during the measurement period.

Christmas is an extraordinary event in Sweden with a lot of extra cooking and cleaning in the houses. This is also true for the house considered. Figure 3 shows that the estimated heat loss factor is noticeably higher on

the days preceding Christmas. If the forgetting factor is put equal to 1, the increase in U-value before Christmas will not be detectable in the results. Between the 8th and 10th of December the weather was stormy, explaining the increased losses during this period.

Also for the other eight houses similar effects of the influence of users on building performance were obtained. For one house, the estimator becomes unstable with a forgetting factor of 0.99. This household use much domestic electricity and hot water which is difficult to model.

7. Conclusions

An on-line identification method for energy monitoring of buildings has been described in this report. The method gives recursive estimates of thermal parameters describing a building's performance.

The algorithm has been used to study time varying parameters by discounting old measurements as new measurements arrive. The two applications presented, one using a rectangular window and the other using a forgetting factor, indicate that this is a feasible way to detect changes in building performance.

The price we pay for the possibility to track time varying parameters is a loss of accuracy in the parameter estimates. Under time invariant conditions, a rectangular window or a forgetting factor yields estimates having the property that the accuracy of parameter estimates never increases with time whatever the length of the measurement period.

Of course, one can employ two recursive identifications in parallel: One identification with and the other without a rectangular window or a forgetting factor. The former identification may be used for control purposes while estimates from the latter identification are the ones used to describe the building's performance.

The proposed identification method or similar methods can easily be incorporated into existing energy management systems. The ability to track changing parameter values makes the method a powerful monitoring tool. Current research at the Joint Research Centre (Ispra, Italy) and at the Swedish Building Research Institute (Gävle, Sweden) aim at developing further on-line identification techniques in the context of energy monitoring.

Acknowledgement

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ETUDE DES PERFORMANCES DES COMPTEURS DE CALORIES UTILISES
SUR DES SYSTEMES THERMIQUES FONCTIONNANT EN REGIME VARIABLE

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1. Introduction

Dans de nombreuses applications, l'utilisation du compteur de calories pour l'enregistrement des consommations énergétiques sur de longues périodes est une solution fiable, précise et peu coûteuse. Néanmoins, ces appareils sont généralement conçus, testés et étalonnés pour des régimes de fonctionnement quasi permanents. Dans le cadre de ces travaux, que nous menons en collaboration avec l'AFME, nous avons entrepris de tester différents compteurs de calories afin de mieux comprendre leur fonctionnement et de mieux connaître leurs performances en régime variable. Ces essais, dont les résultats sont brièvement rappelés ci-après, nous ont permis par la suite de simuler un grand nombre de situations expérimentalement puis numériquement, afin de dégager un certain nombre de recommandations d'installation et d'utilisation de ces appareils.

2. Généralités

2.1. Principe de fonctionnement

Le compteur d'énergie thermique est un instrument permettant de mesurer la quantité de chaleur cédée ou absorbée par le fluide caloporteur entre deux points d'un circuit de chaleur. Quand le fluide est un liquide, l'influence de la pression est négligeable. On définit alors le coefficient calorifique :

$$k = \rho(T_2) \cdot \frac{h(T_1) - h(T_2)}{T_1 - T_2}$$

où $\rho(T_2)$ est la masse volumique du fluide à la température T_2 à laquelle est mesuré le débit volumique du liquide caloporteur Q_2 . L'indice 2 représente donc la partie "froide" de l'installation.

On définit CE la quantité d'énergie mesurée par le compteur de calories entre deux instants T_1 et T_2

$$CE = \int_{t_1}^{t_2} k Q_2 (T_1 - T_2) dt$$

3. Essais réalisés

Au cours de cette étude, nous avons réalisé l'essai de trois appareils du commerce. Les résultats traduits ici concernent essentiellement les capteurs de mesure (sondes de température et débitmètre) ainsi que le compteur de calories complet.

3.1. Résultats des essais sur les compteurs de calories

Nous avons simulé sur banc le fonctionnement d'une installation fournissant de l'eau chaude sanitaire et celui d'une installation de chauffage classique. Les résultats des mesures obtenues avec les compteurs ont été comparés à un compteur de référence préalablement étalonné.

3.1.1. Résultats des mesures effectuées sur l'eau chaude sanitaire

Les essais ont porté sur une installation simulant un chauffe-eau à gaz instantané et sur une installation comportant un cumulus de 150l. Les résultats obtenus sont repris dans les tableaux suivants :

Tableau 1
Simulation chauffe-eau gaz

Type de compteur	Energie compteur kWh	Energie réelle kWh	Ecart énergie %	Ecart volume %
1	16,5	17,38	+5,04	+1,02
2	33	30	-9,08	-0,80
3	25	23,79	-5,09	-1,70

Tableau 2
Simulation ballon "cumulus"

Type de compteur	Energie compteur kWh	Energie réelle kWh	Ecart énergie %	Ecart volume %
1	9,9	10,759	+7,98	+2,77
2	18	19,66	+8,46	+2,35
3	23	23,167	+0,72	+0,16

3.1.2. Essais réalisés sur des installations de chauffage

Nous avons simulé 3 cas de figure :

- Cas 1 : chaudière à faible inertie fonctionnant sans vanne trois voies (débit constant dans l'installation),
- Cas 2 : chaudière à faible inertie munie d'une vanne trois voies en sortie,
- Cas 3 : chaudière avec stock et vanne trois voies en sortie.

Les résultats obtenus sont repris dans les tableaux 3 à 5.

Cas 1. Chaudières à faible inertie sans vanne 3 voies

Tableau 3

Type de compteur	Energie compt. kWh	Energie réelle kWh	Ecart % énergie	Ecart % volume	Ecart ER - ES
2	12	12,69	+5,44	-2,45	VI=251 +2,90
1	11,6	11,606	+0,05	-0,5	VI=11 0,05%
3	11	10,76	-2,25	-2,44	VI=2,51+1,53

Cas 2. Chaudières avec vanne 3 voies sans stockage

Tableau 4

Type de compteur	Energie compt. kWh	Energie réelle kWh	Ecart % énergie	Ecart % volume	Ecart ER - ES
2	12	11,792	-1,76	-5,58	VI=251 12,0
1	12	12,423	+3,40	-0,35	VI=11 -0,28
3	12	12,82	+6,42	-2,15	VI=2,51 0,86

Cas 3. Chaudières avec vanne 3 voies et stockage

Tableau 5

Type de compteur	Energie compt. kWh	Energie réelle kWh	Ecart % énergie	Ecart % volume	Ecart ER - ES
2	10	10,90	+8,27	-0,26	VI=251 +15
1	10	10,762	+7,08	+0,80	VI=11 +0,24
3	12	13,26	+9,5	-3,53	VI=2,51 +0,41

Remarque : Le compteur 2 a été testé avec un émetteur d'impulsion à 2,5 l au lieu de 25 l de façon à raccourcir la durée des manipulations.

3.1.3. Conclusion sur les essais réalisés

Dans la plupart des cas les compteurs de calories ont tendance à sous estimer la valeur de l'énergie réellement mesurée. En ce qui concerne l'eau chaude sanitaire l'erreur enregistrée peut atteindre $\pm 10\%$ et $+6\%$ pour les réseaux de chauffage. L'influence de l'inertie du système semble très importante en regard du mode d'utilisation. Ce point a été à nouveau vérifié lors des simulations informatiques que nous avons menées.

4. Simulations

Afin de compléter les résultats obtenus expérimentalement nous avons entrepris de simuler le comportement d'un compteur de calories installé sur un capteur solaire et sur un ballon d'eau chaude sanitaire. Nous avons utilisé un modèle dynamique à deux températures : /1/

$$\frac{\delta u}{\delta t} + \left[\frac{m}{M} \frac{\delta u}{\delta x} \right] = \frac{h \Sigma}{M \cdot C_p} \cdot (v - u)$$

$$\frac{\delta v}{\delta t} = \frac{h \Sigma}{M' \cdot C'p} \cdot u + \frac{K \cdot A}{M' \cdot C'p} \cdot f - \left[\frac{h \Sigma}{M' \cdot C'p} + \frac{K \cdot A}{M' \cdot C'p} \right] \cdot v$$

4.1. Simulation d'une installation équipée de capteurs solaires

Trois paramètres ont été étudiés :

- l'épaisseur des canalisations et donc l'inertie des tubes
- la longueur de ces canalisations
- la taille du champ de capteurs.

En ce qui concerne l'épaisseur des canalisations, l'erreur augmente lorsque l'inertie propre du tube diminue. Ceci s'explique par la présence de fronts thermiques plus raides et par une augmentation des pertes thermiques.

Les tableaux 6 et 7 reprennent les résultats obtenus lorsque l'on augmente la longueur des tubes et la taille du champ de capteurs. Dans ce dernier cas, nous avons maintenu un rapport $Mf.C / K.ACA$ constant. Les erreurs constatées augmentent toutes dans le même sens que la taille des composants. Ceci est lié à la présence de déphasages importants entre les fronts thermiques qui ne sont pas pris en compte par les compteurs de calories actuels.

On note :

$$E = \frac{\text{Energie Réelle} - \text{Affichage compteur}}{100 \times \text{Energie Réelle}}$$

Tableau 6

LT = 5m ; ACA = 2m²
DCONS = 2,77.10⁻⁵ m³/s

LT (m)	E %
10	7,1
15	9,2
20	23,2

Tableau 7

LT = 10m ; Mf.C / K.ACA = Cte

ACA m ²	DCONS 10 ⁵ m ³ /s	E %
4	5,55	0,37
8	11,11	1,21
16	22,22	1,46
32	44,44	2,81
64	88,88	5,88

VI = 25l ; $\tau_0 = 30s$; $\tau_1 = 35s$;

Influence du temps de réponse des sondes et du volume V

ACA = 2m² ; LT = 10m ; DCONS = 2,77.10⁻⁵ m³/s

Tableau 8

τ_0 s	τ_1 s	E (%) VI = 25l	E (%) VI = 2,5l
30	35	7,10	0,36 %
10	15	12,09	0,17 %
5	7	14,01	0,5 %

On voit ici clairement que le temps de réponse des capteurs de température n'a que peu d'influence. Par contre, le volume VI qui détermine la fréquence à laquelle le compteur effectue le produit $Vix.T$ a une influence très importante.

4.2. Simulation d'une installation sur l'eau chaude sanitaire

Le compteur de calories est placé aux extrémités d'une installation d'eau chaude sanitaire. La sonde froide est montée sur le réseau froid avant le ballon de stockage, la sonde chaude est installée en sortie du ballon sur le réseau utilisateur. Un profil de soutirage type a été établi d'après des données recueillies en /3/.

4.2.1. Influence du profil de soutirage

Des simulations ont été effectuées à partir d'un profil normal, d'un profil pour lequel la durée des impulsions a été réduite de 50%, et des profils comportant une succession de soutirages à 133l/h et 450 l/h d'une durée variant entre 20 et 120 s. L'énergie utile est calculée en sortie du réseau utilisateur.

Le tableau n° 9 reprend ces résultats.

4.3. Influence de la position des sondes

Les sondes sont repérées par une distance en m, 0 étant l'entrée du réseau froid d'une longueur totale de 10 m ou du réseau chaud d'une longueur de 30 m. Le Tableau 10 reprend les résultats obtenus.

Tableau 9

Profil	E mes kWh	EMR %	EMU %
normal	30,332	-3,18	+ 1
réduit	24,705	-1,85	+ 3
4501/h (120s)	77,970	-1,96	+ 0,17
(54s)	33,179	-1,97	+ 2,9
(29s)	16,849	-2,8	+ 6,2
1331/h (60s)	8,134	-2,7	+13,9
(20s)	1,218	-3,5	+54,6

Tableau 10

Position sondes m fr. ch.		E mes en kWh	EMU %
0	0	33,866	+12,8
10	0	33,724	+12,3
0	30	30,176	+0,52
10	30	30,021	0
5	15	32,004	+6,6

EMR : Ecart énergie mesurée / énergie réelle
EMU : Ecart énergie mesurée / énergie utile

4.4. Influence du nombre d'impulsions/litre délivré par le compteur d'eau

On peut constater que l'influence de ce paramètre pour un compteur réel est très faible dans les circonstances de cette simulation. Par contre, il est à noter que la durée de la simulation doit être plus grande dans le cas où VI = 101 que dans le cas où VI = 11, ceci afin de limiter le bruit généré par l'incrémentation discrète de l'énergie mesurée. (cf Tableau 11).

VI	E mes kWh	E réelle kWh	E utile kWh	Ecart % mes/réelle	Ecart % mes/utile
1	52,037	52,299	50,009	+ 0,5	- 4,1
10	33,724	33,723	30,021	+ 0,81	- 3,7

Tableau 11 : Influence du volume d'impulsions sur un compteur réel
Séquence de 5 jours

4.5. Influence de l'inertie des sondes de mesure

Le compteur d'eau étant considéré sans erreur, le tableau reprend les résultats obtenus pour plusieurs temps de réponse des sondes de mesure. On constate que l'écart enregistré passe de 1 à 4% lorsque le temps de réponse à la montée de la sonde passe de 10 à 60s, le temps de réponse à la descente étant pratiquement sans influence comme cela était prévisible pour ce type d'utilisation.

τ montée (s)	τ des- cente (s)	Energie mesurée kWh	Energie réelle kWh	Energie réelle utile kWh	Ecart mes/ réelle %	Ecart mes/ utile %
10	10	31,160	31,389	30,021	+ 0,7	- 3,8
60	60	30,017	31,389	30,021	+ 4,3	0
60	10	30,133	31,389	30,021	+ 4,0	- 0,37
10	60	31,043	31,389	30,021	+ 1,1	- 3,4
Capteur réel		51,442	52,299	50,009	+ 1,64	- 2,86

Tableau 12 : Influence de l'inertie des sondes
(Débitmètre parfait)

4.6. Influence des caractéristiques du compteur d'eau

Nous avons vu précédemment que l'on pouvait caractériser l'erreur de mesure commise par un compteur d'eau par deux coefficients a et b et M le nombre d'impulsions par litre d'eau écoulé.

$$\epsilon = aM + b (\%)$$

Dans ces conditions, nous avons tenté de déterminer l'influence de a et de b.

a	b	E mes kWh	E réelle kWh	E réelle utile	Ecart % mes/ Réelle	Ecart % mes/ Eutile
2	0	31,527	31,389	30,021	- 0,44	- 5,02
8	0	31,778	31,389	30,021	- 1,24	- 5,85
8	4	32,937	31,389	30,021	- 4,93	- 9,71
Capt. réel	VI=101	52,789	52,299	50,009	- 0,94	- 5,6

Tableau 13 : Influence de l'erreur du mesureur de volume
(Sondes parfaites)

L'erreur dynamique est liée à a et l'erreur statique à b. Il apparaît que l'influence de b est nettement prépondérante par rapport à a, une bonne précision du compteur en régime permanent et surtout à faible débit reste indispensable même pour ce type de fonctionnement.