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POSTER 10

# SIMULATION OF THERMAL COUPLING BETWEEN A RADIATOR AND A ROOM WITH ZONAL MODELS

**ABSTRACT**
Zonal models are a promising way to predict air movement in a room with respect to comfort conditions and gradient of temperature, because they require extremely low computer time and may be therefore rather easily included in multizone air movement models.
The main objective of this paper is to study the ability of the zonal models to predict the thermal behaviour of air in case of natural convection coupled with a radiator.
First, we present simplified two zone and five zone models. With the support of the IEA Annex 20 (Air flow pattern within buildings) testcase, we compare the results of the models. Furthermore, a comparison is made with the results of Chen obtained with a Low Reynolds number k-E model.
It appears that five zone model give indoor air temperature profiles consistent with Low Reynolds number k-E model. Concerning the convective heat fluxes, except for the two zone model, the values computed by the models are of the same order of magnitude with lower values for the Low Reynolds k-E number model.

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## LIST OF SYMBOLS

| | | |
|---|---|---|
| Axx | :area | ($m^2$) |
| Gau | :enthalpy flow upper zone-central zone | (W/K) |
| Gla | :enthalpy flow central zone-lower zone | (W/K) |
| Gpl1 | :enthalpy flow lower zone-air leaving the radiator | (W/K) |
| Gppl | :enthalpy flow air leaving the radiator-plume | (W/K) |
| Gup | :enthalpy flow plume-upper zone | (W/K) |
| Lra | :length of the radiator | (m) |
| Rxx | :thermal resistance | (K/W) |
| Rxxc | :convective thermal resistance | (K/W) |
| Rxxr | :radiative thermal resistance | (K/W) |
| Txx | :surface temperature | (°C) |
| Tl | :mean air temperature of lower zone | (°C) |
| Tp | :mean air temperature of radiator plume | (°C) |
| Tp1 | :mean air temperature leaving the radiator | (°C) |
| Troa | :mean air temperature of central zone | (°C) |
| Tror | :room radiant temperature | (°C) |
| Tu | :mean air temperature of upper zone | (°C) |
| $z_0$ | :virtual origin of radiator plume | (m) |

Subscripts:

gl :glazing
wa :vertical walls
ce :ceiling
fl :floor
fa :facade
ra :radiator
tr :trail

## 1-INTRODUCTION

The present paper deals with different ways to model the thermal behaviour of a heated room with simplified zonal models. With the support of Annex 20 testcase d (natural convection with a radiator) and according to the monozone simulation procedure presented by Lemaire [1], we developed a two zone and a five zone models.

## 2-SPECIFICATION OF THE TESCASE

Figure 1 shows the thermal network of the testroom with a radiator.

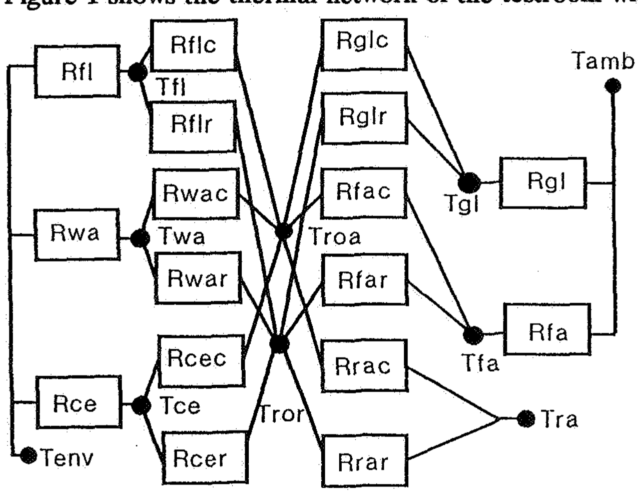


Fig. 1: Thermal network of testroom according to Lemaire [1]

The dimensions of the testroom are 4.2x3.6x2.5 m.
The configuration consists of:

- a window (single glazed) positioned in the front wall (facade) in contact with ambient temperature (Tamb)
- the other walls are in contact with environmental (plenum) temperature (Tenv)
- a radiator (single panel) located below the window in front of the front wall

The dimensions of the window and the radiator are 2.0x1.6m and 2.0x0.3m.
According to Lemaire [1], Table 1 gives the thermal resitances and area of the testroom walls.

| | Axx ($m^2$) | Rxx (K/W) | Rxxc (K/W) | Rxxr (K/W) |
|---|---|---|---|---|
| Window | 3.20 | 0.0150 | 0.0887 | 0.0656 |
| Facade | 5.80 | 0.5225 | 0.0718 | 0.0325 |
| Walls | 30.00 | 0.1010 | 0.0139 | 0.0063 |
| Ceiling | 15.12 | 0.2004 | 0.0275 | 0.0125 |
| Floor | 15.12 | 0.2004 | 0.0275 | 0.0125 |

Table 1: Area and thermal resistances of testroom walls

The radiator convective heat transfer coefficients are calculated using a Nusselt-Rayleigh relation of Kriegel quoted by [1]:

$$Nu = 0.118\ Ra^{1/3}$$

The radiator radiative heat transfer coefficients are computed using simplified Stefan-Boltzmann law for radiation between two surfaces:

$$hrar = \epsilon ra\ \sigma_0\ [(Tra^2 + Tror^2)(Tra + Tror)]$$

with: $\epsilon ra$ radiator LW emissivity
Furthermore, heat transfer coefficients between the radiator rear face and its backwall are computed assuming that the backwall is adiabatic. For more details see Lemaire [2].

Three cases with different radiator temperatures and ambient temperatures have been considered as can be seen in Table 2.

| Case number | Tamb (°C) | Tra (°C) | Rrar (K/W) | Rrac (K/W) |
|---|---|---|---|---|
| d1 | +6.0 | 46.0 | 0.2389 | 0.1424 |
| d2 | -0.9 | 55.0 | 0.2283 | 0.1290 |
| d3 | -8.0 | 65.0 | 0.2170 | 0.1190 |

Table 2: Key parameters of the three testcases (Tenv=20°C)

## 3-THE FIVE ZONE MODEL

### 3-1-Convective network

The indoor air volume is split into five zones coupled each other with air mass flow rates:
- air leaving the radiator (Tp1)
- thermal plume (Tp)
- upper zone (Tu)
- central zone (Troa)
- lower zone (Tl)

The convective network is shown in figure 2.

Comparing to the thermal network of Fig. 1, we add the trail which represents the opaque wall in contact with the thermal plume.

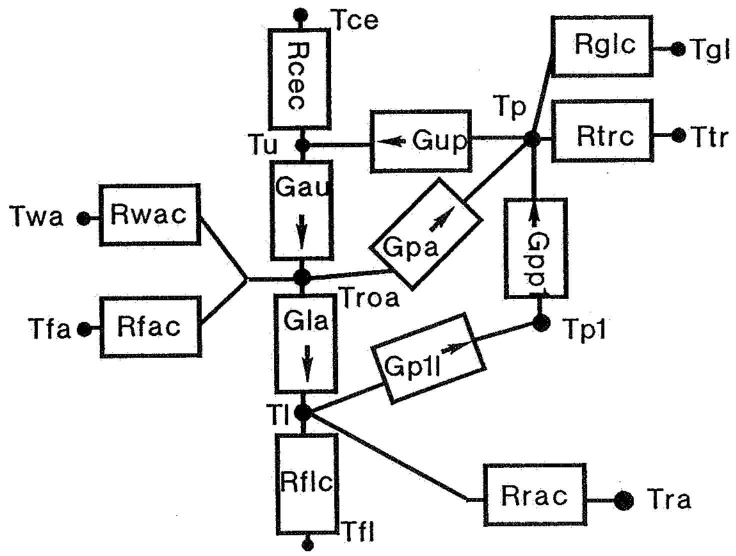


Figure 2: Convective network of testroom with five zones

**3-2-Evaluation of enthalpy flows and air temperatures**

The convective resistance values are the same than those prescribed in Tables 1 and 2. The trail convective heat transfer coefficient value is the same than the window one (3.52 W/m²K).

From the expression of air mass flow rate in the plume of the radiators [3] and assuming that the radiator is located at 0.1m from the floor, we write:

$$Gpp1 = 9.10^{-3} Cp\left[\left(\frac{Tra-Tl)}{Rrac}\right)/Lra\right]^{1/3}(Hra+0.1-z_0)Lra$$

$$Gup = 9.10^{-3}\left[\left(\frac{Tra-Tl}{Rrac}-\frac{Tp-Tgl}{Rglc}-\frac{Tp-Ttr}{Rtrc}\right)/Lra\right]^{1/3}(Height-z_0)Lra$$

Then, mass balance equations give:

Gpa = Gup - Gppl (1)
Gau = Gup (2)
Gla = Gau - Gpa (3)
Gpll = Gla (4)

Zone energy balance equations allow us to calculate air temperatures:

Gpll(Tpl - Tl) = (Tra - Tl)/Rrac (5)
Gppl(Tp - Tpl) + Gpa(Tp - Troa) + (Tp - Ttr)/Rtrc + (Tp - Tgl)/Rglc = 0 (6)
Gup(Tu - Tp) + (Tu - Tce)/Rcec = 0 (7)
Gau(Troa - Tu) + (Troa - Twa)/Rwac + (Troa - Tfa)/Rfac = 0 (8)
Gla(Tl - Troa) + (Tl - Tfl)/Rflc = 0 (9)

Because Gppl and Gup also depend on Troa and Tp, the calculation must be performed iteratively.

At last, using surface and radiant energy balance we compute surface temperatures and room radiant temperature (Tror).

## 4-THE TWO ZONE MODEL

In this model, the convective scheme is based on the studies made by A.T.Howarth [4]. The room is divided into two zones, an upper one and a lower one, separated by a neutral plane, accross which the net vertical air mass flow rate is equal to zero.

The main difference with the five zone model is that the plume behaviour is entirely described by only one equation. This equation is a thermal balance for the upper zone, the central zone and the thermal plume of the five zone model. In fact, the two zone model is a simplification of the five zone model.

### 4-1-Conductive and radiative model

The Howarth's formulation only deals with convective heat fluxes modelling. In order to compare with the model elaborated by Lemaire and with the five zone model, it was necessary to couple this convective formulation with a conductive and a radiative network. For the good homogeneity of this comparison, we choose to adopt the conductive and radiative scheme described by Lemaire.

As in the five zone model, the test chamber is divided into:

- the vertical walls (except the facade)
- the ceiling
- the floor
- the window
- the facade
- the trail

The radiant heat transfers are modelled through heat transfer coefficients calculated with simplified Stefan-Boltzmann law (see Tables 1 and 2).

### 4-2-Convective model

As a result of the splitting of the room air volume, the unknown temperatures are:

- mean air leaving the radiator (Tp1)
- upper zone mean air temperature (Tu)
- lower zone mean air temperature (Tl)
- average air temperature of the room (Troa)

In order to determine these temperatures, there must be four equations to close the system. The first equation is a balance equation between the convective heat output of the radiator and the convective heat losses at the surfaces.

The heat flow $\phi$convwa from the core of the room to the walls and the part of the facade which is not located into the plume is calculated according to the correlation of De Graaf and Van der Held quoted by [5]:

$$\phi convwa = 0.729\ (Twa - Troa)^{1.36} Height^{0.08} \qquad (W/m^2)$$

The heat transfer coefficient at the floor is taken equal to 1 W/m²K, which is consistent with the correlations presented by many authors. According to Table 1, the window convective heat transfer coefficient is taken equal to 3.52 W/m²K and for the ceiling we adopt the value 2.40 W/m²K.

The average air temperature is calculated as a mean value over the height:

$$Troa\ Height = Tl\ zneut + Tu\ (Height - zneut)$$

zneut is the height of the neutral plane. This altitude is calculated assuming the uprising mass flow rate in the plume to be equal to the downward mass flow rate in the boundary cold layers along the wall. The plume mass flow rate is computed by the correlation established by Inard [3] from experiments:

$$Gpp1 = 9.10^{-3} Cp \left[\frac{(Tra-Tl)}{Rrac}/Lra\right]^{1/3} (Hra+0.1-z_0)\ Lra$$

and the mass flow rate mwa in the boundary layers due to natural convection along the walls is expressed in the form [5]:

$$mwa = 0.00145(Troa-Twa)^{0.36}Height^{0.08} \qquad (Kg/ms)$$

The third equation gives the departure temperature of the plume leaving the radiator (eq.5):

$$Gpp1\ (Tp1-Tl) = (Tra-Tl)/Rrac$$

The last equation assumes that the radiator plume is discharged almost immediatly in the upper region. Thus, this zone is in equilibrium at the departing plume temperature modified by heat losses to the glazing, the ceiling and the trail.

We get this equation from five zone model balance equations (eq.1, 6 and 7) and assuming that Tp=Tp1 and Troa=Tu:

$$Gpp1(Tu-Tp1) + \frac{(Tp1-Tgl)}{Rglc} + \frac{(Tu-Tce)}{Rcec} + \frac{(Tp1-Ttr)}{Rtrc} = 0$$

This is the main simplification of the five zone model into a two zone model.

## 5-RESULTS

### 5-1-Air temperature profiles

Figure 3 show air temperature profiles computed with the different models. Concerning the two zone and the five zone models, we have arbitrarily located the upper and lower temperatures at 0.10m from the ceiling and the floor, and the central zone temperature at mid height of the cell. Furthermore, we superimposed on these figures the results computed by Chen [6] with a Low Reynolds k-E model.

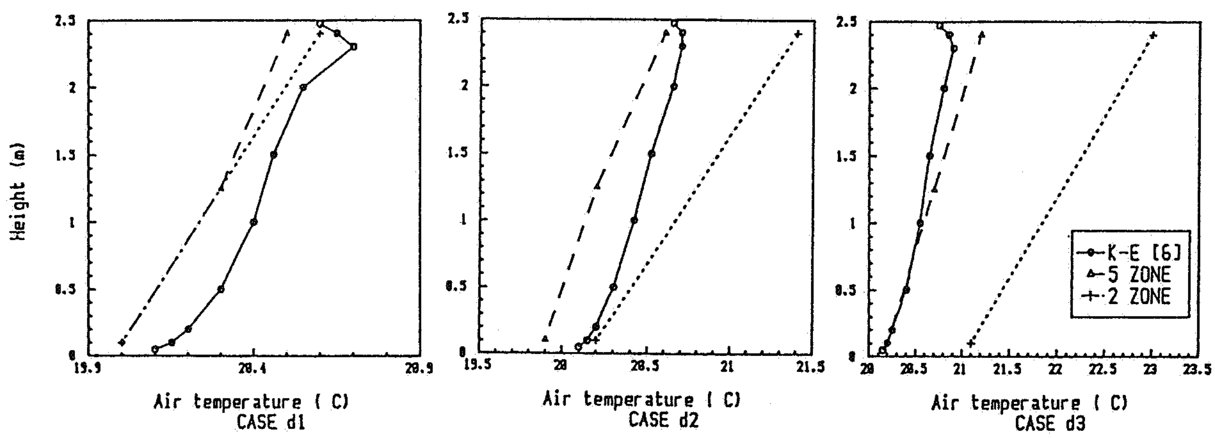


Figure 3: Air temperature profiles

These figures show that air temperature profiles computed with a five zone model and a Low Reynolds k-E number model are very close each other. Concerning the results calculated with the two zone model, we can see that the gap with the others air temperature profiles increase when ambient temperature decrease. In fact, using mean air temperature leaving the radiator instead of plume air temperature for window convective heat flux calculation results in higher values of air temperature leaving the radiator and upper zone air temperature for a given radiator air mass flow.

### 5-2-Convective heat exchanges

Table 3 shows the convective heat fluxes exchanged between the radiator and the walls of the testroom.

The convective heat fluxes calculated by the five zone model and Chen [6] are of the same order of magnitude though those computed with a Low Reynolds k-E number model are lower with a maximum gap for the radiator convective heat power of 24% (case d3). Note that for Chen's simulation the boundary conditions are the inside surface temperatures of the walls.

Concerning the results computed with a two zone model and for the same reason raised in section 5-1, we obtain higher convective heat fluxes values.

| case d1 | two zone model | five zone model | k-E model [6] |
|---|---|---|---|
| window | -209 | -120 | -108 |
| walls | -63 | -63 | -48 |
| radiator | 272 | 183 | 156 |
| case d2 | two zone model | five zone model | k-E model [6] |
| window | -296 | -178 | -162 |
| walls | -111 | -94 | -55 |
| radiator | 407 | 272 | 217 |
| case d3 | two zone model | five zone model | k-E model [6] |
| window | -391 | -241 | -217 |
| walls | -172 | -136 | -71 |
| radiator | 563 | 377 | 288 |

Table 3: Convective heat fluxes (W)

## 6-CONCLUSION

We built up two simplified zonal models in order to describe the thermal behaviour of a room heated with a radiator. With the support of the Annex 20 testcase d, we compared the air temperatures and convective heat fluxes calculated. Furthermore, the results are also compared with those computed with a Low Reynolds number k-E model [6].

The simplified five zone model give results consistent with k-E model and the simplified two zone model give higher air temperatures and convective heat fluxes.

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