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Ergonomics of the thermal environment — Evaluation of thermal environments in vehicles —

Part 4: Determination of the equivalent temperature by means of a numerical manikin

Ergonomie des ambiances thermiques — Évaluation des ambiances thermiques dans les véhicules —

Partie 4: Détermination de la température équivalente à l'aide d'un mannequin numérique

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Contents

Page

Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

The procedures used to develop this document and those intended for its further maintenance are described in the ISO/IEC Directives, Part 1. In particular, the different approval criteria needed for the different types of ISO documents should be noted. This document was drafted in accordance with the editorial rules of the ISO/IEC Directives, Part 2 (see www.iso.org/directives).

Attention is drawn to the possibility that some of the elements of this document may be the subject of patent rights. ISO shall not be held responsible for identifying any or all such patent rights. Details of any patent rights identified during the development of the document will be in the Introduction and/or on the ISO list of patent declarations received (see www.iso.org/patents).

Any trade name used in this document is information given for the convenience of users and does not constitute an endorsement.

For an explanation of the voluntary nature of standards, the meaning of ISO specific terms and expressions related to conformity assessment, as well as information about ISO's adherence to the World Trade Organization (WTO) principles in the Technical Barriers to Trade (TBT), see www.iso.org/iso/foreword.html.

This document was prepared by Technical Committee ISO/TC 159, *Ergonomics*, Subcommittee SC 5, *Ergonomics of the physical environment*.

A list of all parts in the ISO 14505 series can be found on the ISO website.

Any feedback or questions on this document should be directed to the user's national standards body. A complete listing of these bodies can be found at www.iso.org/members.html.

Introduction

The interaction of convective, radiant and conductive heat exchange in a vehicle compartment or similar confined space is highly complex. External thermal loads in combination with the air conditioning system in a vehicle compartment create non-uniform thermal environments, which are often the main cause of complaints of thermal discomfort. In vehicles with poor or non-existent air conditioning systems, non-uniform thermal environments can also be created by the interaction between the ambient climatic conditions and vehicle structures. While a subjective evaluation reflects the total sensations of a human body, these often incur great costs while the study phase is being conducted. Physical measurements provide detailed and accurate local information; however, these results must be integrated in some way to predict the thermal effects on humans. Furthermore, since specific climatic factors sometimes play a dominant role in the overall heat exchange of a human body, an evaluation method that accounts for the relative importance of these factors is required.

This document is part of the ISO 14505 series. To meet the above-stated requirements, this document provides calculation methods that utilize numerical simulations to assess the total thermal environment of vehicles. The equivalent temperature, obtained from measurements taken using a thermal manikin, is defined in ISO 14505-2. This document extends the definition of the ISO 14505 series to include numerical evaluation when this document is used in conjunction with the equivalent temperature defined in ISO 14505-2.

As described in ISO 14505-2, an equivalent temperature can be utilized in the assessment of vehicle cabins and other various enclosed spaces with non-uniform environments. As is the case for ISO 14505-2, this document can also be applied to vehicle cabins and other enclosed spaces.

This document supposes that the ISO 14505 series will be applied to various situations, such as:

— in the case of experimental facilities that are not prepared;

— in the case of prototypes that are incomplete;

— in the case of conditions that are difficult to simulate in controlled experimental settings;

— in the case that occupants are extrapolated to unknown or virtual environments.

Ergonomics of the thermal environment — Evaluation of thermal environments in vehicles —

Part 4: Determination of the equivalent temperature by means of a numerical manikin

1 Scope

This document provides guidelines for extending the definition of equivalent temperature to predictive purposes and specifies a standard prediction method for the assessment of thermal comfort in vehicles using numerical calculations. Specifically, this document sets forth a simulated numerical manikin as a viable alternative to the thermal manikin for the purpose of calculating the equivalent temperature.

2 Normative references

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 13731, *Ergonomics of the thermal environment — Vocabulary and symbols*

ISO 14505-2, *Ergonomics of the thermal environment — Evaluation of thermal environments in vehicles — Part 2: Determination of equivalent temperature*

3 Terms and definitions

For the purposes of this document, the terms and definitions given in ISO 13731 and ISO 14505-2 and the following apply.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— ISO Online browsing platform: available at https://www.iso.org/obp

— IEC Electropedia: available at http://www.electropedia.org/

3.1
numerical manikin
virtual thermal manikin recreating a thermal manikin, or a digital model of a thermal manikin used to calculate performance

3.2
physical manikin
real thermal manikin to measure real environment

3.3
computational fluid dynamics
CFD
simulation of a series of calculations based on specific boundary conditions and specific parameters associated with fluid and thermal fields using discrete equations based on the Navier-Stokes/Lattice-Boltzmann equations as well as heat transfer equations that consider convection, radiation and conduction, and generally account for the effects of turbulent flow

4 Symbols

A complete list of symbols used in this document is presented in Table 1.

Table 1 — Symbols and units

Symbol	Term	Unit
α	Solar absorptivity on clothing (skin on unclothed area) surface	–
A	Skin surface area	m^2
C	Convective heat loss from clothing (skin on unclothed area) surface	$W/m^2°C$
f_{cl}	Area factor (ratio of clothed to nude area)	–
h_c	Convective heat transfer coefficient	$W/m^2°C$
h_{cal}	Total heat transfer coefficient in a standard environment	$W/m^2°C$
h_{cs}	Convective heat transfer coefficient in a standard environment	$W/m^2°C$
h_r	Radiant heat transfer coefficient	$W/m^2°C$
h_{rs}	Radiant heat transfer coefficient in a standard environment	$W/m^2°C$
I_{cl}	Thermal insulation of clothing	$m^2°C/W$
Q	Total heat loss from skin surface	W/m^2
Q_{set}	Set value of Q at constant heat flux mode	W/m^2
R	Radiant heat loss from clothing (skin on unclothed area) surface, including effect of solar radiation	W/m^2
R_{cr}	Thermal insulation between core and skin assumed by comfort equation	$m^2°C/W$
R_t	Total thermal resistance between the manikin skin surface and the environment	$m^2°C/W$
S	Mean solar radiation reached on clothing (skin on unclothed area) surface	W/m^2
t_a	Air temperature	°C
t_{aset}	Air temperature at h_{cal} calculation	°C
t_{cl}	Clothing (skin on unclothed area) surface temperature	°C
t_{cr}	Core temperature assumed by the comfort equation	°C
t_{eq}	Equivalent temperature	°C
t_o	Operative temperature including the effects of solar radiation	°C
t_r	Mean radiant temperature	°C
t_{sk}	Skin surface temperature	°C
t_{skset}	Set value of t_{sk} at constant temperature mode	°C
v_a	Air velocity	m/s
n	Suffix: segment number of each body part	–
whole	Suffix: whole body	–
Symbols used in Annex E		
A_b	Body surface area of the manikin	m^2
$A_{e,i}$	Elemental surface area of the element i	m^2
A_n	Segmental surface area of the segment n	m^2
$B_{i,j}$	Absorption factor of radiation from surface elements i to j	–
$F_{i,j}$	View factor of radiation from surface elements i to j	–
$h_{cal,n}$	Total heat transfer coefficient of segment n for calibration	W/m^2K
$h_{cal,whole}$	Total heat transfer coefficient of the entire manikin for calibration	W/m^2K
i,j	Variable body surface element number	–

Table 1 *(continued)*

Symbol	Term	Unit
k	Variable spatial volume element number for calculation of t_a Variable body surface element number in the recurrence equation of $B_{i,j}$	–
m_b	Number of body surface elements	–
$m_{e,n}$	End body surface element number of segment n	–
$m_{s,n}$	Start body surface element number of segment n	–
m_v	Number of spatial volume elements	–
m_w	Number of wall surface elements	–
n	Variable local segment number of the manikin	–
n_{seg}	Number of manikin segments	–
$Q_{e,i}$	Heat flux of element i	W/m^2
Q_n	Averaged heat flux over segment n	W/m^2
Q_{whole}	Averaged heat flux over the entire manikin	W/m^2
R_{cr}	Thermal insulation between core and skin assumed by comfort equation	m^2K/W
$R_{cr,e,i}$	Thermal insulation of element i for the comfort equation mode calculation	m^2K/W
T_a	Averaged air temperature in the standard chamber (in Kelvins)	K
t_a	Averaged air temperature in the standard chamber (in Celsius)	°C
$t_{a,e,k}$	Air temperature of the spatial volume element k	°C
$t_{a,in}$	Air temperature entering the standard chamber	°C
t_{cr}	Core temperature assumed by the comfort equation	°C
$t_{cr,e,i}$	Core temperature of element i for the comfort equation mode calculation	°C
t_o	Operative temperature in the standard chamber	°C
T_r	Mean radiant temperature of the wall of the standard chamber (in Kelvins)	K
t_r	Mean radiant temperature of the wall of the standard chamber (in Celsius)	°C
t_{sk}	Skin surface temperature of the manikin	°C
$t_{sk,n}$	Averaged skin surface temperature of segment n	°C
$t'_{sk,n}$	Estimated average skin surface temperature of the segment n from the comfort equation	°C
$t_{sk,whole}$	Averaged skin surface temperature of the entire manikin	°C
T_w	Wall surface temperature of the standard chamber (in Kelvins)	K
t_w	Wall surface temperature of the standard chamber (in Celsius)	°C
u_a	Air flow velocity in the standard chamber	m/s
$\dot{V}_{a,in}$	Volumetric air flow rate entering the standard chamber	m^3/s
$V_{e,k}$	Volume of the spatial volume element k	m^3
V_0	Volume of the spatial region in the standard chamber	m^3
ΔQ_n	Correction amount for generated heat of segment n	W/m^2
Δt_{ce}	Threshold of difference between $t'_{sk,n}$ and $t_{sk,n}$ for iterative convergence	°C
Δt_o	Threshold of difference between t_a and t_r for iterative convergence	°C
ε_j	Emissivity of the surface element j	–
ε_{sk}	Emissivity of the manikin	–
ε_w	Emissivity of the wall of the standard chamber	–

Table 1 *(continued)*

Symbol	Term	Unit
ξ_m	Conversion factor between the actual wall surface temperature and mean radiant temperatures	–

5 Assessment of thermal environments in vehicles

The method of assessment by equivalent temperature is defined in ISO 14505-2. The assessment procedures in ISO 14505-2 are applicable to numerical evaluations, for which "numerical manikin" is defined in this document. Figure 1 shows the role of this document and its relations with the other parts of the ISO 14505 series as well as different International Standards.

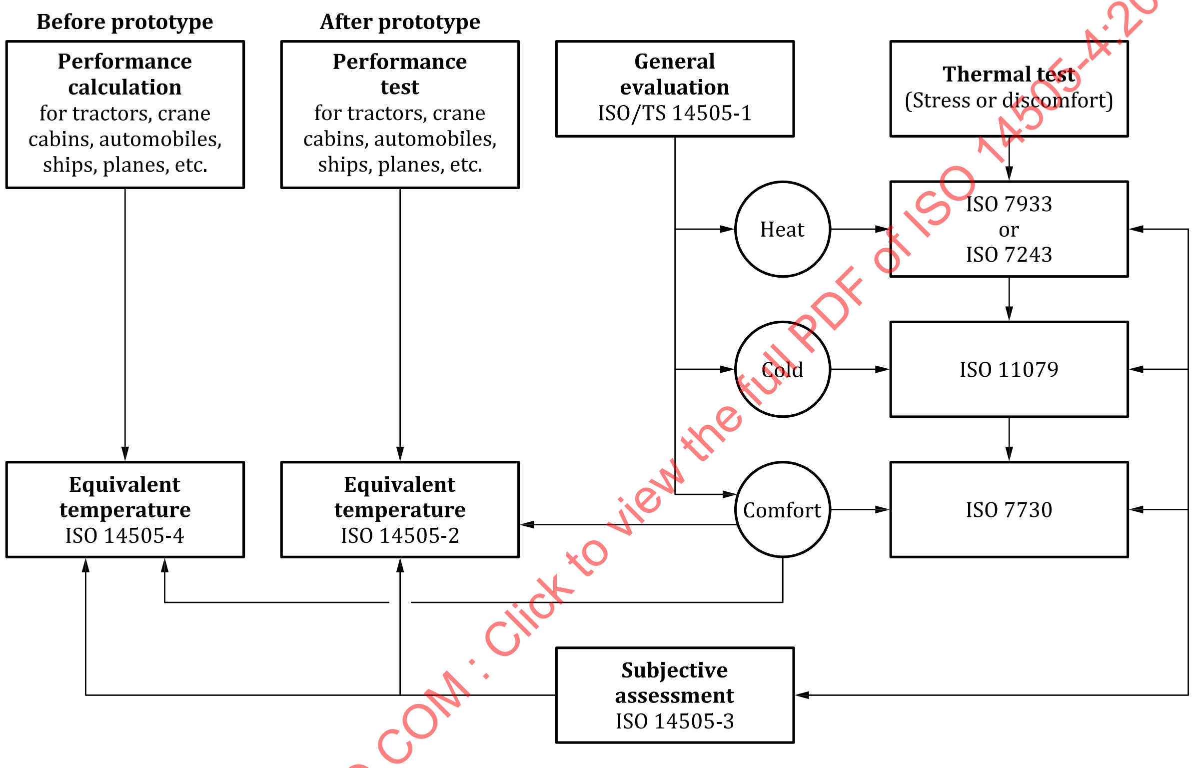


Figure 1 — Role of numerical evaluation among different International Standards

6 Principles of assessment utilizing a numerical manikin

This document presents two methods for calculating the equivalent temperature. One is a calculation method coupled with a numerical manikin, as described in Clause 7. The other is a calculation method using thermal factors, as described in Clause 8. Either method can be used to evaluate the thermal environment in vehicles.

The former calculation method coupled with a numerical manikin is intended for use with a simulation tool, such as computational fluid dynamics (CFD). A numerical manikin imitates a physical manikin to calculate the equivalent temperature. The method of calculation using thermal factors estimates the equivalent temperature by assuming the existence of the imaginary numerical manikin. In this method, the equivalent temperature is calculated using the thermal factors, air temperature, radiant temperature, air velocity and solar radiation. Figure 2 shows a schematic of the two methods.

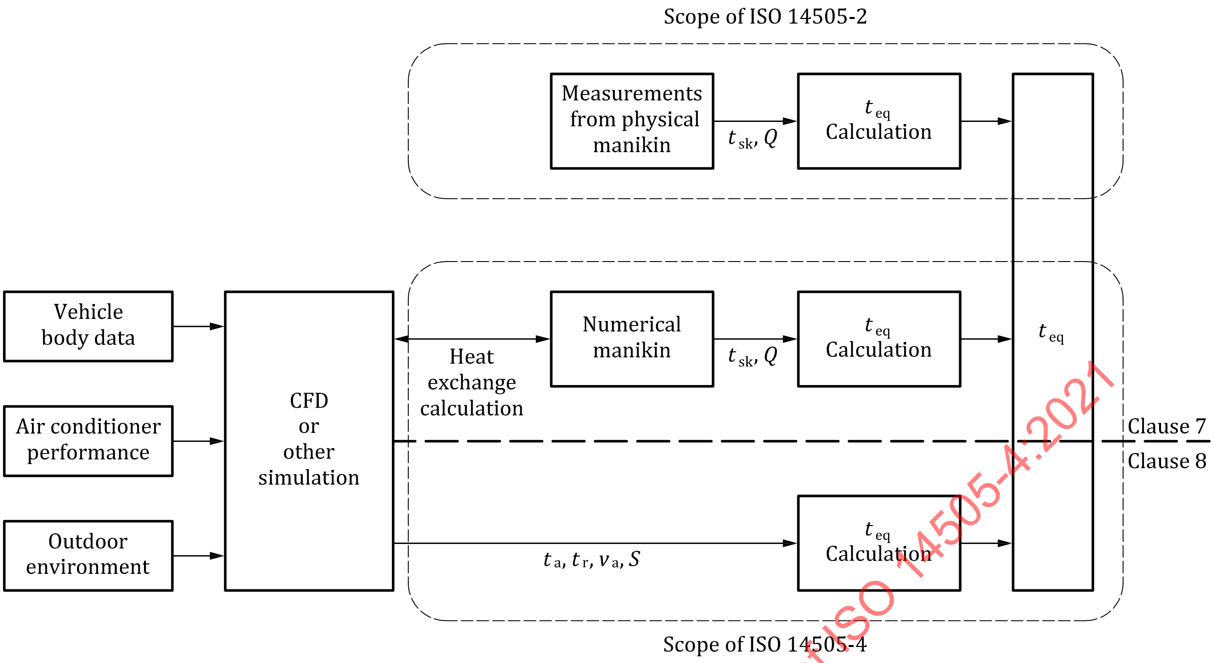


Figure 2 — Two methods for calculating equivalent temperature

7 Calculation method coupled with numerical manikin

7.1 General

This clause describes the framework of the calculation. To evaluate the indoor environment of a vehicle numerically, the following issues should be considered and taken account (see Figure 3):

a) heat flow through the shell structure of the vehicle;

b) flow and thermal field in the cabin;

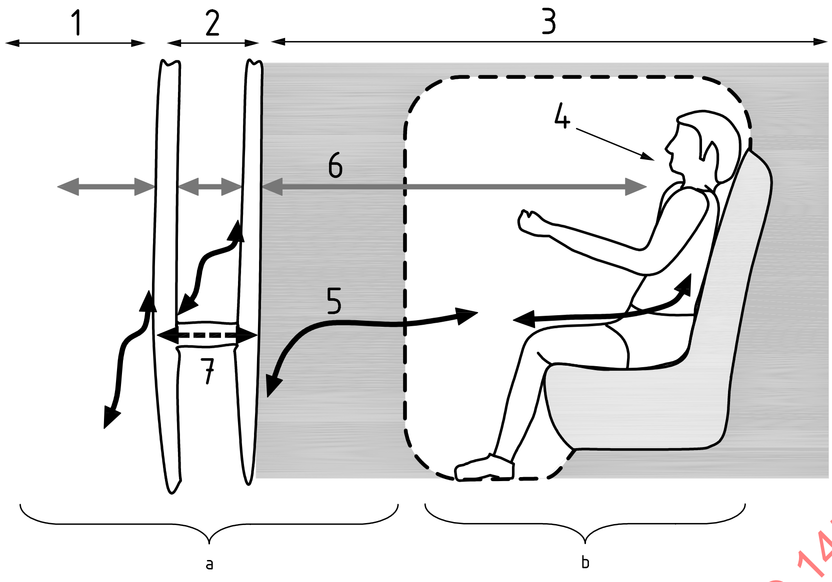
c) radiant field (including solar radiation) in the cabin;

d) conductive heat exchange;

e) heat balance of the thermal manikin model (numerical manikin).

This document is intended to be applied to the region around the manikin, relating to items b) and e). The heat flow through the structure of the vehicle body is defined as suitable. This document defines indispensable ideas concerning the above items, which will enable successful and useful calculations in these situations. However, this document does not define any specific methods for utilization because all methods present both advantages and disadvantages for particular problems.

Once the environmental state concerning thermal comfort in the cabin is calculated, evaluation becomes possible. Items a) to d) give the principal local parameters of air velocity, temperature and radiation, though some simultaneous calculation coupling with the heat balance calculation is required. This calculation will produce a heat transfer value close to that measured using the physical manikin. Therefore, the evaluation method described in ISO 14505-2 is applicable.



Key

1 outside of vehicle

2 shell structure

3 interior of vehicle

4 thermal manikin

5 convection

6 radiation

7 conduction

a Other standards (TC22 related).

b This document.

NOTE Evaluation of the area in contact with the seat is outside the scope of this document.

Figure 3 — Framework of heat transfer system

7.2 Flow and thermal field around manikin

7.2.1 Convective heat

The flow and thermal field in the cabin are estimated via calculations. One practical option for this is CFD. The informative concrete contents of this method are represented in Annex A. The outputs are the air velocity vector and air temperature in a cabin. The heat flux on the wall and surface are also obtained through this calculation.

The primary problem in CFD calculations is the treatment of boundary conditions. This can be overcome in practice by selecting any of the following:

a) Calculate the heat transfer on the surface of the manikin using CFD directly.

b) As a preliminary, calculate the heat transfer coefficient on the surface of the manikin using CFD. Then calculate the flow and thermal field coupling using the heat balance calculation of the manikin.

c) Utilize the heat transfer coefficient obtained by measurement. Then calculate the flow and thermal field coupling with the heat balance calculation of the manikin, as described in b).

d) Estimate the heat transfer coefficient using a predictive formula based on the air velocity or temperature. Then calculate the flow and thermal field coupling using the heat balance calculation for the manikin, as described in b).

7.2.2 Radiant heat

The radiant field is calculated based on the geometric condition in a cabin separated from the flow field calculation. Regarding long-wave radiation, as a preliminary, the view factor relating to all potential combinations between different surface elements of the wall and the manikin or human body should be calculated. Once those factors have been obtained, the radiant heat exchange is calculated when the temperature of a pair of surface elements is given. As stated previously, the radiant heat is involved in the boundary conditions of the heat transfer equation, so that the temperature is calculated iteratively until convergence. For convenience, those factors are converted to the mean radiant heat transfer coefficient.

Short-wave radiation (solar radiation) can be treated as energy flux striking the surface. Here, the transmission loss through the window glass should be taken into consideration. Solar radiation can be regarded as divided into the following components:

a) direct solar radiation;

b) sky solar radiation;

c) reflection on the ground.

The solar radiant heat is also involved in the boundary conditions of the heat transfer equation. In the case of a climate wind tunnel test performed without use of a solar lamp, it is supposed that the effects of solar radiation are neglected. Concrete informative treatments are represented in Annex C.

7.2.3 Conductive heat

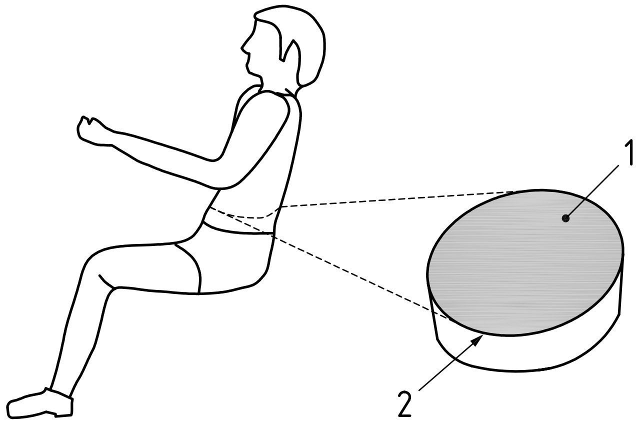
Evaluation of the contacted segment is outside the scope of this document. The conductive heat transfer between the manikin and seat is disregarded in the calculations.

7.3 Calculation of heat exchange on manikin

7.3.1 Structure and control of numerical manikin

Figure 4 shows the theoretical structure of the numerical manikin with regard to a human-shaped one. The centre of each segment is assumed to consist of an adiabatic core. A heat generator is equipped on the surface of the manikin.

The shape of the manikin is defined by calculation grids for CFD, as shown in Figure 5 a). The partition is performed to imitate an actual manikin, as in Figure 5 b). Boundary conditions are given for each segment.

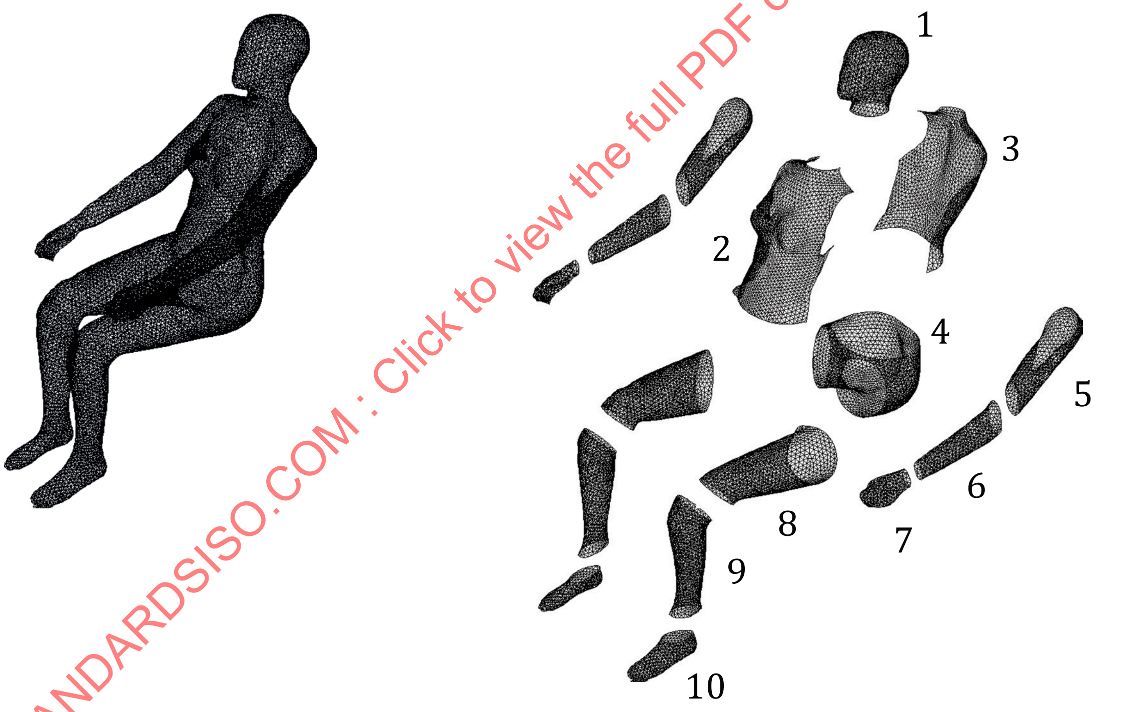


Key

1 adiabatic core

2 heat generator on surface

Figure 4 — Theoretical structure of numerical manikin



a) Full-body model of manikin **b) Partition of surface grids (16-segment case shown)**

Key

1 head

2 chest

3 back

4 pelvis

5 upper arm

6 forearm

7 hand

8 thigh

9 leg

10 foot

Figure 5 — Surface grids for numerical calculation

The control model is intended to imitate a physical manikin and includes the following three operating modes:

a) Controlled surface temperature (constant temperature mode): generally, the surface temperature of all segments is maintained at 34 °C (see ISO 14505-2).

b) Controlled heat generation (constant heat flux mode): this method uses a metabolic rate to represent the heat flux for all segments. An example of the metabolic rate during vehicle driving is shown in ISO/TS 14505-1 and ISO 8996. Note that this method is less commonly used than the other two.

c) Described by comfort equation (comfort equation mode): generally, the parameter values for all segments in this mode are t_{cr} = 36,4 °C and R_{cr} = 0,054 m^2°C/W[3].

7.3.2 Calculation of heat exchange

The primary problem for the heat transfer calculation is determining the appropriate treatment of the boundary condition on the surface of the manikin. Once this has been designated, the following treatments can be used:

a) Flow field, radiant field and heat transfer on the surface of the manikin are solved simultaneously. The temperature distribution is calculated directly by solving for entrainment of heat in the boundary layer; however, the convective heat transfer coefficient is not explicitly calculated. In this case, the grid size near the surface should be small enough to resolve the heat transfer (the boundary layer). Otherwise, a well-tuned wall function developed to calculate the heat transfer near the solid boundary should be adopted.

b) The convective and mean radiant heat transfer coefficients are treated as known values. In this case, the grid structure near the surface can be determined only to calculate the flow field, resulting in coarser grids compared to the prior treatment (a).

Regardless of the calculation method used, flow field calculations should account for buoyancy effects when the air velocity is small (i.e. less than 0,1 m/s). As such, the air motion should be calculated via coupling with the heat transfer equation.

7.4 Calculation of h_{cal}

Three methods of calculating and defining the value of h_{cal} are considered:

a) Apply the CFD calculation to standard conditions to obtain the “calibrated” characteristics of h_{cal}. Informative concrete treatments for this are presented in Annex E.

b) Adopt measured data gleaned using a physical manikin corresponding to the “numerical manikin”. Practical measurement methods for this item are detailed in ISO 14505-2.

c) Estimate it using Formula (10) in 8.3.

7.5 Calculation outputs

The input and output data to or from the numerical manikin is shown in Table 2. The informative concrete treatments used to calculate the equivalent temperature are presented in Annex B.

Table 2 — Input and output to or from the numerical manikin

Control principle	Inputs	Outputs
Constant temperature mode ($T_{sk,n}$ = T_{skset})	T_{skset}, $I_{cl,n}$	Q_n
Constant heat flux mode (Q_n = Q_{set})	Q_{set}, $I_{cl,n}$	$T_{sk,n}$
Comfort control mode (Q_n = (T_{cr}−$T_{sk,n}$)/R_{cr})	T_{cr}, R_{cr}, $I_{cl,n}$	$T_{sk,n}$, Q_n

The equivalent temperature in each body segment is calculated using Formula (1), as defined in ISO 14505-2.

$$t_{eq,n} = t_{sk,n} - \frac{Q_n}{h_{cal,n}} \qquad (1)$$

The whole-body equivalent temperature is calculated using Formulae (2) to (4), as defined in ISO 14505-2.

$$t_{eq,whole} = t_{sk,whole} - \frac{Q_{whole}}{h_{cal,whole}} \qquad (2)$$

$$t_{sk,whole} = \frac{\sum_{n \in n(whole)} (t_{sk,n} \cdot A_n)}{\sum_{n \in n(whole)} A_n} \qquad (3)$$

$$Q_{whole} = \frac{\sum_{n \in n(whole)} (Q_n \cdot A_n)}{\sum_{n \in n(whole)} A_n} \qquad (4)$$

The calculated values differ depending on the control mode even if the environmental conditions are the same (see Clause D.2). As described in ISO 14505-2, it is recommended that either constant temperature mode (at 34 °C) or comfort equation mode (with t_{cr} = 36,4 °C and R_{cr} = 0,054 m²°C/W) be used. The calculated t_{eq} should be presented together with the control mode and its parameter(s).

8 Calculation method using thermal factors

8.1 General

It is also possible to calculate the equivalent temperature without the coupling calculation by assuming the existence of an imaginary numerical manikin. In this method, the equivalent temperature is calculated from the thermal factors, air temperature, radiant temperature, air velocity and solar radiation. It should be noted that the effect of environmental heating by manikin is neglected in this one-way calculation.

This method can be applied to the calculation of the equivalent temperature on either individual manikin segments or the whole body. In this document the above method is intended for use with a numerical calculation tool; however, it can also be applied to calculations using measured thermal factors.

The formulae discussed in this clause are described in more detail in Annex F.

8.2 Flow and thermal field around manikin

8.2.1 Convective heat

The convective heat loss, C, from the surfaces of clothing and unclothed skin is defined by Formula (5):

$$C = f_{cl} \cdot h_c \cdot (t_{cl} - t_a) \qquad (5)$$

8.2.2 Radiant heat

The radiant heat loss, R, from the surfaces of clothing and unclothed skin is defined by Formula (6):

$$R = f_{cl} \{ h_r \cdot (t_{cl} - t_r) - \alpha \cdot S \} \qquad (6)$$

8.2.3 Conductive heat

As described in 7.2.3, the conductive heat transfer between the manikin and seat is neglected.

8.3 Calculation of heat exchange

The total heat loss, Q, from the surfaces of clothing and unclothed skin is expressed using either Formula (7) or Formula (8):

$$Q = \frac{(t_{sk} - t_{cl})}{I_{cl}} \quad (7)$$

$$Q = C + R \quad (8)$$

The total heat loss, Q, is defined by Formula (9), which is derived from Formulae (5), (6), (7) and (8):

$$Q = \frac{t_{sk} - \dfrac{h_c \cdot t_a + h_r \cdot t_r + \alpha \cdot S}{h_c + h_r}}{I_{cl} + \dfrac{1}{f_{cl} \cdot (h_c + h_r)}} \quad (9)$$

8.4 Calculation of h_{cal}

Coefficient h_{cal} represents the total heat transfer coefficient through clothing, i.e. the amount of total heat loss per °C of difference between the skin and the environment, in the standard environment described in ISO 14505-2. This coefficient can be expressed as Formula (10):

$$h_{cal} = \frac{1}{I_{cl} + \dfrac{1}{f_{cl} \cdot (h_{cs} + h_{rs})}} \quad (10)$$

8.5 Calculation outputs

8.5.1 General

The equivalent temperature, t_{eq}, is defined by Formula (11), as shown in ISO 14505-2:

$$t_{eq} = t_{sk} - \frac{Q}{h_{cal}} \quad (11)$$

The practical methods for calculating t_{eq} under each manikin control mode are shown in 8.5.2 to 8.5.4.

The calculated values differ depending on the control mode, even if environmental conditions are the same (see D.2). As described in ISO 14505-2, it is recommended that either constant temperature mode (at 34 °C) or comfort equation mode (with t_{cr} = 36,4 °C and R_{cr} = 0,054 m²°C/W) be used. The calculated t_{eq} should be presented together with the control mode and associated parameter(s).

8.5.2 Constant temperature mode

When the value of t_{sk} in the constant temperature mode is defined as t_{skset}, the equivalent temperature, t_{eq}, is calculated using Formula (12), which is obtained by inserting Formulae (9) and (10) into Formula (11):

$$t_{eq} = t_{skset} - \frac{t_{skset} - \dfrac{h_c \cdot t_a + h_r \cdot t_r + \alpha \cdot S}{h_c + h_r}}{I_{cl} + \dfrac{1}{f_{cl} \cdot (h_c + h_r)}} \cdot \left(I_{cl} + \frac{1}{f_{cl} \cdot (h_{cs} + h_{rs})} \right) \quad (12)$$

8.5.3 Constant heat flux mode

When Q in the constant heat flux mode is set to be Q_{set}, t_{sk} is calculated using Formula (13), which is obtained from Formula (9):

$$t_{sk} = Q_{set} \cdot \left(I_{cl} + \frac{1}{f_{cl} \cdot (h_c + h_r)} \right) + \frac{h_c \cdot t_a + h_r \cdot t_r + \alpha \cdot S}{h_c + h_r} \tag{13}$$

The equivalent temperature, t_{eq}, is calculated using Formula (14), which is obtained by inserting Formulae (10) and (13) into Formula (11):

$$t_{eq} = \frac{Q_{set}}{f_{cl}} \cdot \left(\frac{1}{h_c + h_r} - \frac{1}{h_{cs} + h_{rs}} \right) + \frac{h_c \cdot t_a + h_r \cdot t_r + \alpha \cdot S}{h_c + h_r} \tag{14}$$

8.5.4 Comfort equation mode

When the control equation in the comfort control mode is expressed by Formula (15), Q and t_{sk} are calculated using Formulae (16) and (17), respectively, which are obtained from Formulae (9) and (15).

$$Q = \frac{t_{cr} - t_{sk}}{R_{cr}} \tag{15}$$

$$Q = \frac{t_{cr} - \dfrac{h_c \cdot t_a + h_r \cdot t_r + \alpha \cdot S}{h_c + h_r}}{R_{cr} + I_{cl} + \dfrac{1}{f_{cl} \cdot (h_c + h_r)}} \tag{16}$$

$$t_{sk} = t_{cr} - R_{cr} \cdot \left(\frac{t_{cr} - \dfrac{h_c \cdot t_a + h_r \cdot t_r + \alpha \cdot S}{h_c + h_r}}{R_{cr} + I_{cl} + \dfrac{1}{f_{cl} \cdot (h_c + h_r)}} \right) \tag{17}$$

The equivalent temperature, t_{eq}, is calculated using Formula (18), which is obtained by inserting Formulae (10), (16) and (17) into Formula (11):

$$t_{eq} = t_{cr} - \frac{t_{cr} - \dfrac{h_c \cdot t_a + h_r \cdot t_r + \alpha \cdot S}{h_c + h_r}}{R_{cr} + I_{cl} + \dfrac{1}{f_{cl} \cdot (h_c + h_r)}} \cdot \left(R_{cr} + I_{cl} + \frac{1}{f_{cl} \cdot (h_{cs} + h_{rs})} \right) \tag{18}$$

Annex A
(informative)

Calculation via computational fluid dynamics (CFD) technique

A.1 General method

A.1.1 Guidance for CFD calculations

To ensure the success of CFD calculations, an appropriate method should be selected while considering the purpose and required accuracy. In general, the following factors should be considered:

— geometric model (e.g. shape, mesh size);

— physical model (e.g. calculation model, boundary conditions);

— calculation errors (e.g. round-off error, truncation error).

When calculations are performed, it is essential that a computer with sufficient performance to accommodate the number of meshes and the calculation model be used. If possible, it is recommended that the calculation accuracy be experimentally verified before engaging in detailed calculations.

For the treatment of radiant heat transfer, refer to Annex C.

A.1.2 Geometry model

Depending on the development phase, a variety of models can be used from simplified geometric models useful for approximate evaluation in the early development phase to detailed geometric models just prior to prototyping. The calculation model should be selected based on the intended purpose.

There are two ways to simulate air flow from an outlet:

a) Set an air flow rate at the root of the air duct or in the HVAC system to simulate the air flow in the duct and that from the outlet.

b) Set an air velocity and direction on a surface of an outlet.

To accurately simulate air flow in the vehicle cabin, it is important to specify the positions of drafters or inlets for circulation, together with the corresponding air flow rates.

A.1.3 Calculation of flow field

The flow field inside the cabin is supposed to be solved based on the Reynolds averaged Navier-Stokes equations (RANS).[4] Commonly, the following typical specifications are applied[5]:

a) fluid: incompressible fluid (air);

b) solving method: finite volume method (FVM);

c) solution of velocity and pressure: SIMPLE type method;

d) discretization scheme: high order up-wind scheme for convection term;

e) turbulence model: two-equation model (e.g. k-ω SST, k-ε model);

f) treatment of the boundary layer: standard wall function;

g) grid system: unstructured grid system.

The above menu is not an exclusive one required for successful calculation. It is possible to adopt the finite element method (FEM) instead of using FVM. It is further possible to solve the velocity-pressure field using the coupled method. Higher accuracy can be achieved through exact calculations of the boundary layer, using a method such as the low Reynolds number model. With regards to the grid system, a structured grid system can also be applied.

It should be noted that the accuracy of a calculation depends on the resolution of the grids. While decreasing the grid size generally improves accuracy, this increases the calculation cost. Therefore, it is recommended that the grid size be selected while considering the balance between the calculation cost and accuracy.

A.1.4 Calculation of thermal field

The thermal field in the cabin is calculated using the Reynolds averaged turbulent heat transfer equation coupled with the RANS equations. To simulate the effects of buoyancy, a gravitation term is added to the RANS equations. The turbulent heat flux is estimated using the turbulent Prandtl number based on the similarity in heat transfer. Generally, the following typical specifications will be applied:

a) solution: common to the flow field solution;

b) treatment of the buoyancy effects: Boussinesq approximation;

c) treatment of the boundary layer: standard wall function for thermal boundary layer.

The effects of radiation are included in the boundary condition. The temperature on the surface of the manikin is determined by the heat balance equation, which consists of the terms of heat convection, radiation and heat conduction. Annex C describes the treatment of the radiation in greater detail. Once the surface temperature has been calculated, it is used as a boundary condition in calculating the thermal field. Therefore, calculation of the thermal field should be coupled with the flow field and heat balance calculations if the surface temperature is not given as a boundary condition.

A.2 Treatment of the boundary condition

Adoption of the wall function, based on Prandtl's log-law, is a convenient and basic technique for boundary layer calculations. The non-dimensional parameter y+ should be suppressed to less than an order of several hundred. When y+ becomes less than 10, a higher precision model is recommended because the wall function is no longer applicable to the region. This is commonly seen in environments such as that in close proximity to the surface, called the viscous sub layer. The wall function of the thermal boundary layer is consistent with that of the velocity boundary layer.

The performance of convective heat transfer on the surface is dominated by the precision of calculated turbulence strength very close to the surface. Thus, a higher precision model is required for the calculation of heat transfer relative to the wall function when accurate calculation is desired. Two typical models for this process are:

— low Reynolds number model;

— two-layer model.

It should be noted that the accuracy of these calculations depends on the resolution of the grids very close to the surface, even if the above models are adopted. Furthermore, sufficient computer resources are required for intensive calculations.

There is another way that the coefficient of convective heat transfer is given as the boundary condition. The advantage of this method is that it is possible to obtain the practical and realistic heat transfer performance as long as the flow condition is not changed significantly from when the value of this coefficient was determined. Additionally, the calculation of the flow and thermal fields can be separated completely, although the results from the flow field calculation are required for calculating the thermal field.

A.3 Calculated results

Typical outputs of the CFD calculations are the pressure, velocity and temperature fields. It is also possible to obtain the temperature distribution, heat loss or both on the surface of the manikin, which is then used to calculate the equivalent temperature, t_{eq}. The typical form of this output is shown in Annex B.

Annex B
(informative)

Typical inputs and outputs of calculation with numerical manikin

B.1 Typical inputs and outputs of the calculation

B.1.1 General

As mentioned in ISO 14505-2, the evaluation method utilizing equivalent temperature, t_{eq}, is applied to both clothed and unclothed surfaces. The inputs and outputs of such dually applicable calculations[6][7] are described in this annex.

The treatment of clothing conditions in these calculations are classified in two typical ways: an exact method and an approximate method. When an exact method is used, digital data from the surface configuration of the clothed manikin and detailed calculations for the gaps between these clothes and the skin surface of the manikin are required. This method requires complex, often impractical, calculations. Conversely, the approximate method described in this annex is convenient and practical when the skin surface temperature, t_{sk}, or heat loss, Q, can be calculated using the thermal insulation, I_{cl}, and area factor, f_{cl}, of the clothing. In this method, the surface of the numerical manikin is assumed to be the surface of the clothing if a given segment is covered with clothing. The following three cases are available for such calculations:

— calculate a nude condition with the surface shape of a nude manikin;

— calculate a clothed condition with the surface shape of a nude manikin;

— calculate a clothed condition with the surface shape of a clothed manikin.

The approximate method can be used for all of the above cases. Table B.1 shows the inputs and outputs to or from the numerical manikin as well as the post process used to calculate t_{eq} in each manikin control mode. The concrete calculation procedures are shown in Figure B.1.

— Clothed segments can be calculated using $I_{cl,n}$ and $f_{cl,n}$. The value of $f_{cl,n}$ can be estimated using the following formula: $f_{cl,n} = 1,00 + 1,97\ I_{cl,n}$.

— Unclothed segments can be calculated using the following parameters: $I_{cl,n} = 0$, $f_{cl,n} = 1,00$.

While the heat loss and skin temperature of the numerical manikin are essentially matched to the physical manikin after the post processing is performed, the following issues should be noted when the t_{eq} of a clothed segment is calculated using the surface shape of a nude manikin:

a) Heat transfer through clothing: air ventilation of the clothing and heat transfer between segments is neglected, since clothing is modelled only with $I_{cl,}$ and f_{cl}.

b) Effects of the manikin on the thermal environment: the air flow and other thermal factors surrounding the numerical manikin can be different from the actual environment since the shape of the numerical manikin is different from an actual clothed manikin.

c) Effects of the environment on the manikin: the air flow or radiation received by the numerical manikin can be different from those experienced by an actual clothed manikin, since the shape of the numerical manikin is different from an actual clothed manikin.

d) Total heat generated by the manikin: the heat generated by the numerical manikin is reduced to 1/$f_{cl,n}$, and thus the heating effects of the manikin on the environment are underestimated.

However, when a manikin in a nude condition or unclothed manikin segment is calculated, the above issues are not observed. When a clothed segment is calculated with surface shape of a clothed manikin, issues b), c) and d) can be avoided.

Table B.1 — Inputs and outputs of the calculation

			Constant temperature mode $t_{sk,n} = t_{skset}$	**Constant heat flux mode** $Q_n = Q_{set}$	**Comfort equation mode** $Q_n = (t_{cr} - t_{sk,n})/R_{cr}$
Input and output for heat exchange calculations	Thermal insulation under the manikin surface	m²°C /W	Input: $f_{cl,n} \cdot I_{cl,n}$	No need, even if clothed (0)	Input: $f_{cl,n} \cdot (I_{cl,n} + R_{cr})$
	Temperature under the thermal insulation	°C	Input: t_{skset}	Output: $t_{cl,n}$	Input: t_{cr}
	Heat flux on the surface of the manikin	W/m²	Output: $C_n + R_n$	Input: $Q_{set}/f_{cl,n}$	Output: $C_n + R_n$
Post process to calculate final output, Q_n and $t_{sk,n}$	Heat loss (Q_n)	W/m²	Calculation from output data: $f_{cl,n} \cdot (C_n + R_n)$	Using input data: Q_{set}	Calculation from output data: $f_{cl,n} \cdot (C_n + R_n)$
	Skin temperature ($t_{sk,n}$)	°C	Using input data: t_{skset}	Calculation from output data: $t_{cl,n} + Q_{set} \cdot I_{cl,n}$	Calculation from output data: $t_{cr} - Q_n \cdot R_{cr}$

B.1.2 Constant temperature mode

Manikin control: Constant temperature: $t_{sk,n} = t_{skset}$ (for example, t_{skset} = 34 °C)

Input to manikin: Set the thermal insulation, without thickness, under the manikin surface: $f_{cl,n} \cdot I_{cl,n}$

Fix the temperature under the thermal insulation (surface temperature on an unclothed segment): t_{skset}

Output from manikin: Calculate the heat loss from the surface of the manikin: $C_n + R_n$

Post process: Calculate the heal loss from the skin surface: $Q_n = f_{cl,n} (C_n + R_n)$

Calculate the equivalent temperature from $t_{sk,n} = t_{skset}$ and Q_n: $t_{eq,n} = t_{sk,n} - Q_n / h_{cal,n}$

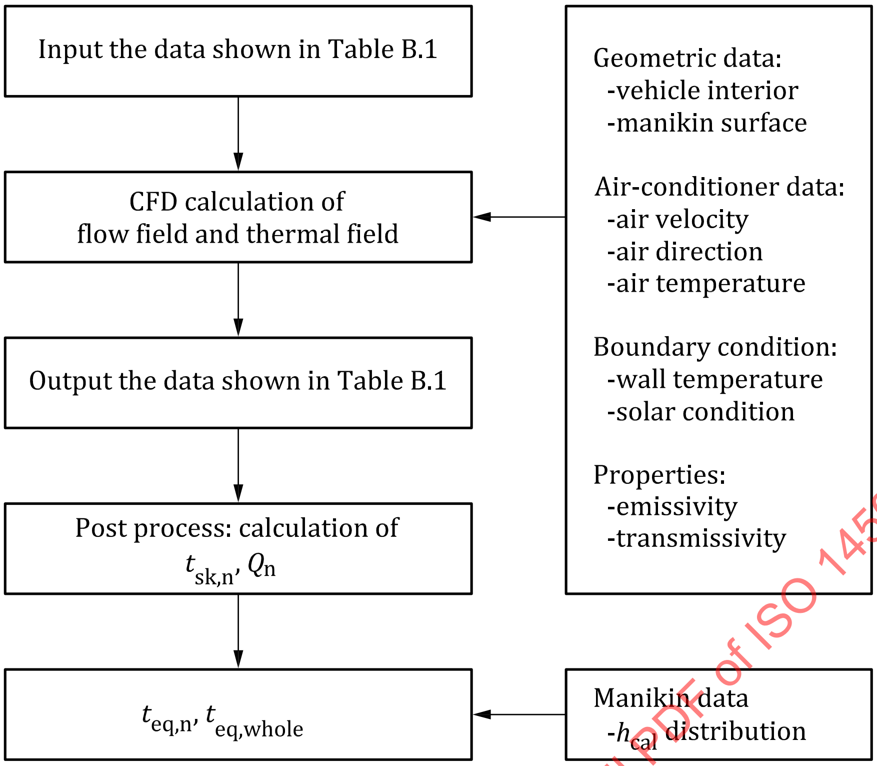


Figure B.1 — Typical calculation procedures

B.1.3 Constant heat flux mode

Manikin control: Constant heat flux: $Q_n = Q_{set}$ (for example $Q_{set} = 60\ W/m^2$)

Input to manikin: Set the constant heat flux on the surface of the manikin: $Q_{set}/f_{cl,\,n}$

Output from manikin: Calculate the surface temperature of the manikin: $t_{cl,n}$

Post process: Calculate the skin temperature: $t_{sk,n} = t_{cl,n} + Q_{set} \cdot I_{cl,n}$

Calculate the equivalent temperature from $t_{sk,n}$ and $Q_n = Q_{set}$: $t_{eq,n} = t_{sk,n} - Q_n / h_{cal,n}$

B.1.4 Comfort equation mode

Manikin control: Comfort equation: $Q_n = (t_{cr} - t_{sk,n})/R_{cr}$ (for example t_{cr} = 36,4 °C, $R_{cr} = 0{,}054\ m^2°C/W$)

Input to manikin: Set the thermal insulation, without thickness, under the surface of the manikin: $f_{cl,\,n}\ (I_{cl,n} + R_{cr})$

Fix the temperature under the thermal insulation (surface temperature on unclothed segment): t_{cr}

Output from manikin: Calculate the heat loss from the surface of the manikin: $C_n + R_n$

Post process: Calculate the heal loss from the skin surface: $Q_n = f_{cl,n} (C_n + R_n)$

Calculate the skin temperature: $t_{sk,n} = t_{cr} + Q_n . R_{cr}$

Calculate the equivalent temperature from $t_{sk,n}$ and Q_n: $t_{eq,n} = t_{sk,n} - Q_n / h_{cal,n}$

B.2 Example of output form

Figure B.2 shows the geometric data of a manikin and the interior surface of a cabin. Figure B.3 shows the stream line of the air conditioning flow in this cabin. Figures B.4 and B.5 present the velocity distribution and air temperature distribution close to the body surfaces. Figure B.6 shows the distribution of t_{eq} over the manikin, which is calculated from the surface temperature distribution and the heat generation of the manikin. Figure B.7 is the t_{eq} distribution for individual body segments. Some more calculation examples can be found in References [7],[8] and[9].

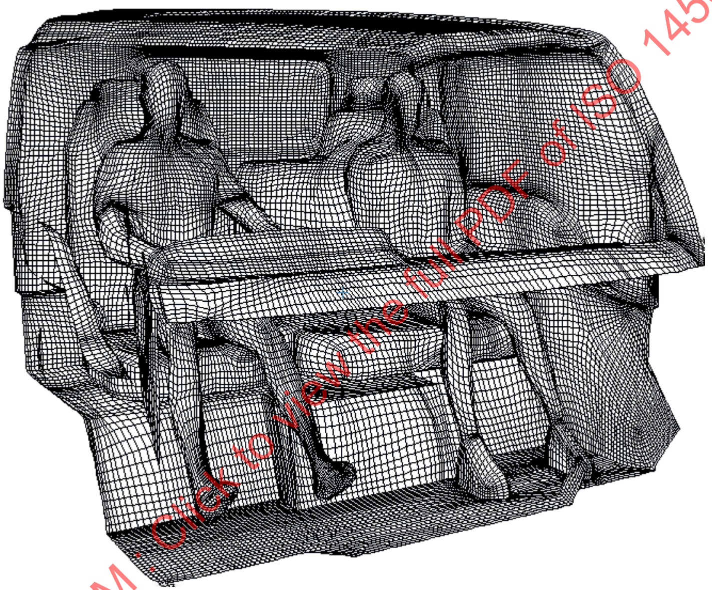


Figure B.2 — Geometric data of manikins and the interior surface of the cabin

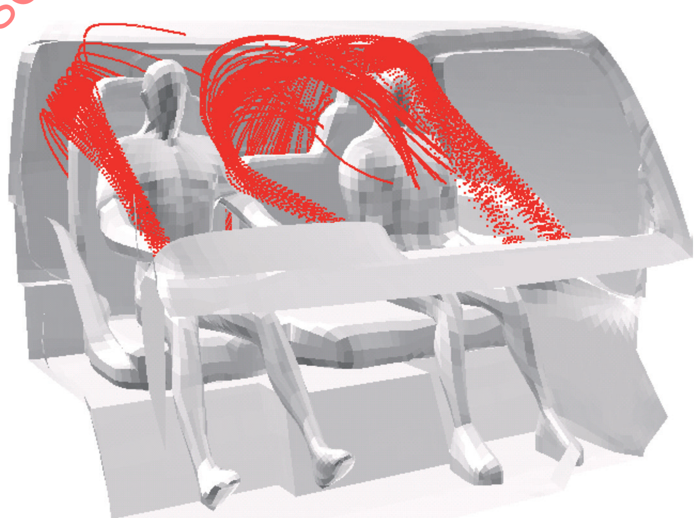


Figure B.3 — Stream line of air conditioning flow in the cabin

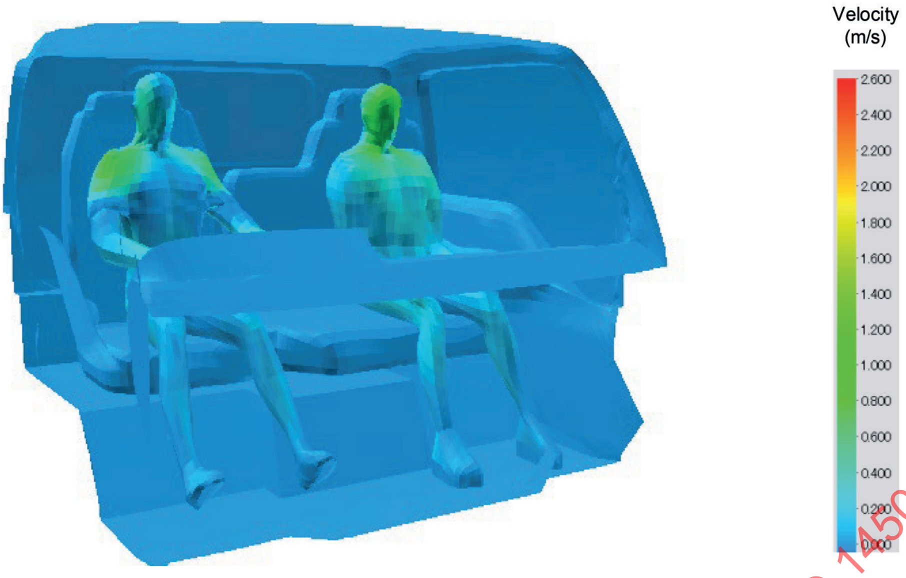


Figure B.4 — Velocity distribution close to body surfaces

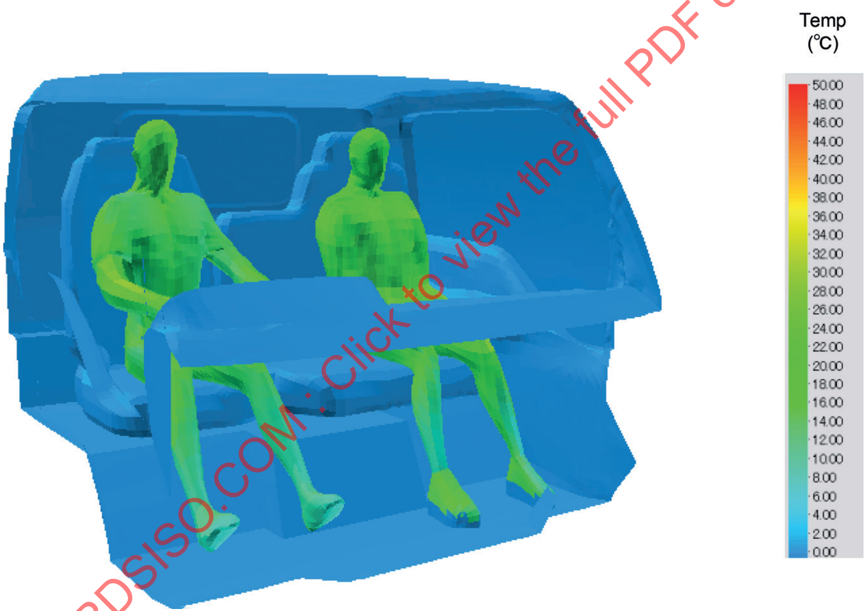


Figure B.5 — Air temperature distribution close to the body surface

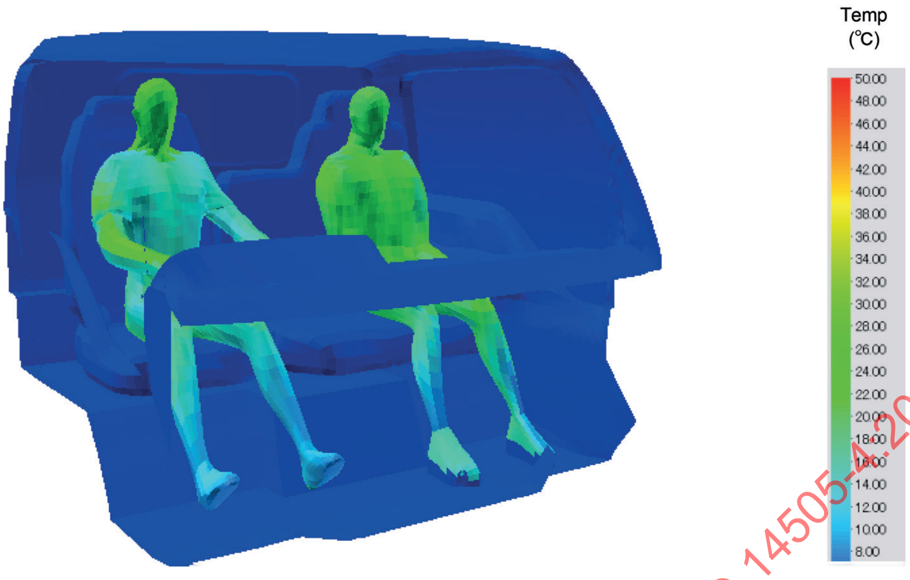
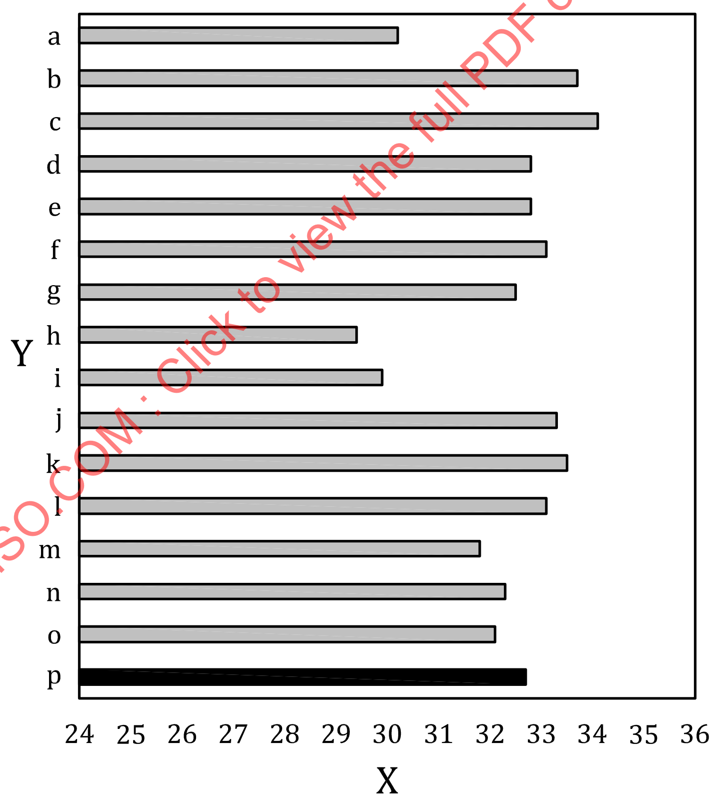


Figure B.6 — t_{eq} distribution close to body surfaces



Key

X equivalent temperature t_{eq} (°C)

Y body segment

a	head	b	chest	c	pelvis	d	left upper arm
e	right upper arm	f	left forearm	g	right forearm	h	left hand
i	right hand	j	left thigh	k	right thigh	l	left leg
m	right leg	n	left foot	o	right foot	p	whole body

Figure B.7 — t_{eq} distribution for body segments (results)

Annex C
(informative)

Treatment of radiant heat transfer

C.1 Wall radiation

C.1.1 Assumption

Wall radiation is generally calculated using the following assumptions:

— the radiant wall is grey and perfect diffusing;

— radiant heat is derived using the Stefan–Boltzmann law;

— radiant heat has no influence on the air in vehicles.

Typical emissivity on walls is represented in Table C.1.

C.1.2 Radiant heat exchange on walls

When calculating the radiant heat exchange, the geometry of the vehicle surface is divided into minute elements in which the physical temperature and properties can be considered to be uniform. The radiant heat exchange between these elements is primarily calculated using the radiant equilibrium equation based on the view factor or emitted ray tracing.[6][7][9] The view factor can be calculated using the Monte-Carlo method,[10] the Hemi-cube method[11] or the numerical integration method[12].

The radiant heat exchange calculation using view factor is commonly used in CFD.[6][7][9] The radiation exchange in enclosed grey-diffuse surfaces is calculated based on the view factors among the surfaces of interior parts and manikin.[13] The assumption of this method is that the air in the enclosed space does not absorb, emit or scatter radiation. When the heat exchange is calculated, it is essential that the radiant and convective calculations are performed simultaneously.

C.1.3 Additional information

— Spectral reflection on elements is available for the calculation of wall radiation.

— It is preferable to check the accuracy of both the reciprocity and summation theorems in the calculated view factors.

— In a vehicle evaluation, calculating the wall radiation is usually more important in winter conditions than in summer conditions. This is because the temperature differences between the numerical manikin and the walls are larger under winter conditions[6][7].

Table C.1 — Example of typical emissivity

Material	Steel (coated)	Fabric (white)	Plastic	Skin	Glass
Emissivity (N.D.)	0,85 to 0,95	0,93	0,85	0,95 to 0,99	0,90 to 0,95

C.2 Solar radiation

C.2.1 Assumption

Solar radiation is generally calculated using the following assumptions:

— solar radiation contains wavelengths ranging from 300 nm to 2 100 nm;

— the averaged solar constant is assumed to be between 1 350 and 1 370 W/m^2;

— transmissivity of the atmosphere is defined by the local weather condition (generally lower than 0,75).

C.2.2 Solar radiation on walls

In calculating the solar radiation, it is assumed that the solar input enters into the cabin through the windows of the vehicle.[14] The solar radiation reaching the calculated elements is divided into the following component classifications:

— direct solar radiation;

— sky solar radiation;

— reflection from the ground or surrounding obstacles.

The following component is additionally considered:

— multiple reflections on elements in the cabin.

The solar radiation absorbed by these elements is calculated by multiplying the total solar radiation reaching the elements by the absorptivity of the walls. The typical absorptivity and transmissivity on walls are listed in Table C.2.

C.2.3 Additional information

— Thermal properties (absorptivity, transmissivity, and reflectivity) generally differ according to the heat source, such as sun or solar lamp. Most notably, the thermal properties of glass are quite different between these two sources (see Table C.2).

— Solar radiation calculations can be separated from those of radiant heat transfer and CFD.

Table C.2 — Example of typical absorptivity and transmissivity

Material	Steel (coated)	Fabric (white)	Plastic	Skin
Absorptivity (N.D.)	0,30 to 0,80	0,37 to 0,47	0,90	0,70
Transmissivity (N.D.)	0,00	0,00	0,00	0,00

Material	Green glass (2 + 2mm)	IR-cut glass (2 + 2mm)	Green glass for IR lamp (2 + 2mm)	IR-cut glass for IR lamp (2 + 2mm)
Absorptivity (N.D.)	0,42	0,54	0,53	0,82
Transmissivity (N.D.)	0,53	0,40	0,42	0,12

Annex D
(informative)

Typical inputs and outputs of calculations using thermal factors

D.1 Typical inputs and outputs of the calculation

An example of a calculation using thermal factors:

Calculation condition: t_a = 25,0 °C, t_r = 30,0 °C, v_a = 0,5 m/s, S = 50 W/m^2, I_{cl} = 0,155 m^2°C/W

Manikin control mode: Constant temperature mode, t_{skset} = 34,0 °C

Other parameters: $f_{cl} = 1{,}00 + 1{,}97 \cdot I_{cl}$,

h_c = 3,1 (v_a < 0,2),[15] $h_c = 8{,}3 \cdot v_a^{0,6}$ (0,2 < v_a < 4,0),[15] h_r = 4,7 [15],

h_{cs} = 3,1, h_{rs} = 4,7, α = 0,7.

The equivalent temperature, t_{eq}, is calculated using Formula (12) as follows.

$$t_{eq} = 34{,}0 - \frac{34{,}0 - \dfrac{5{,}5 \cdot 25{,}0 + 4{,}7 \cdot 30{,}0 + 0{,}7 \cdot 50}{5{,}5 + 4{,}7}}{0{,}155 + \dfrac{1}{1{,}31 \cdot (5{,}5 + 4{,}7)}} \cdot \left(0{,}155 + \frac{1}{1{,}31 \cdot (3{,}1 + 4{,}7)} \right) = 30{,}4\,°\mathrm{C}$$

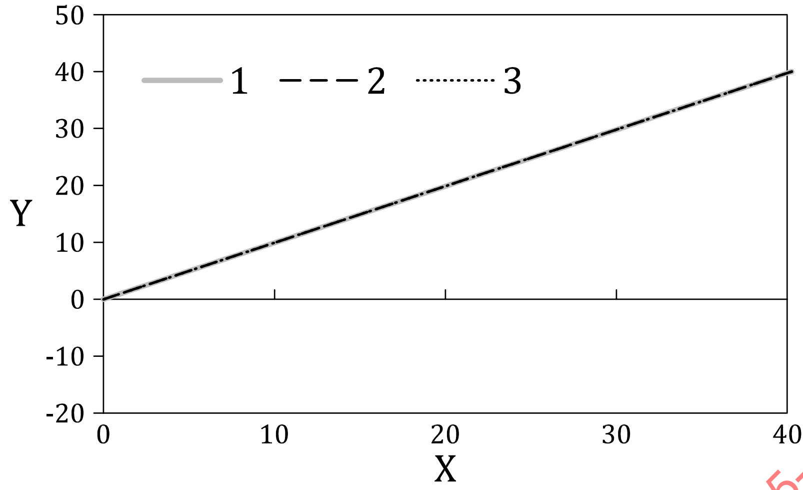
D.2 Comparison of calculation results among control modes

Figures D.1 to D.3 show examples of calculation results with the three following control modes:

- constant temperature mode (t_{skset} = 34 °C);
- constant heat flux mode (Q_{set} = 60 W/m^2);
- comfort equation mode (t_{cr} = 36,4 °C, R_{cr} = 0,054 m^2°C/W).

The calculation results of t_{eq} depend on the control mode theoretically. Therefore, the calculated t_{eq} should be presented together with the specifications of the manikin used, including its control mode and parameters.

When $h_c = h_{cs}$, implying that v_a is very low, no difference will be made by altering the control modes (Figure D.1). The differences are larger when v_a is larger (compare Figure D.1 with Figure D.2). However, the differences are reduced when I_{cl} is larger (compare Figure D.2 with Figure D.3).



Key

X air temperature t_a = radiant temperature t_r (°C)

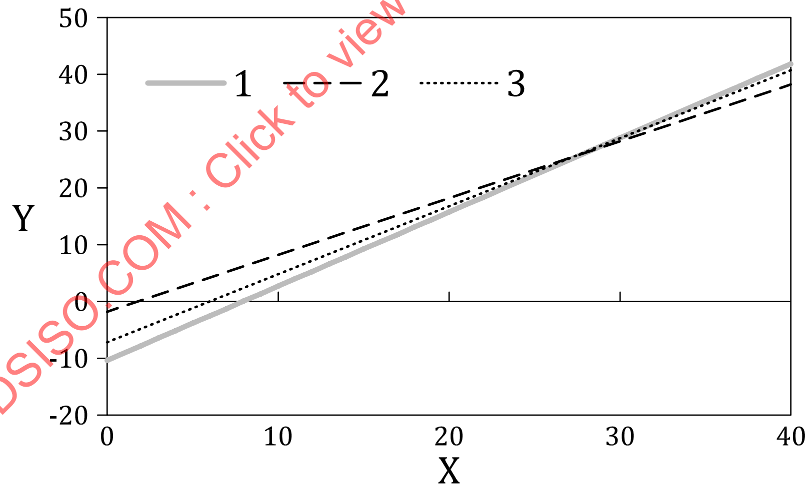
Y equivalent temperature t_{eq} (°C)

1 constant temperature mode

2 constant heat flux mode

3 comfort equation mode

Figure D.1 — Calculation results of t_{eq} with three control modes, calculation conditions: t_a = t_r, v_a = 0 m/s, S = 0 W/m², I_{cl} = 0 m²°C/W



Key

X air temperature t_a = radiant temperature t_r (°C)

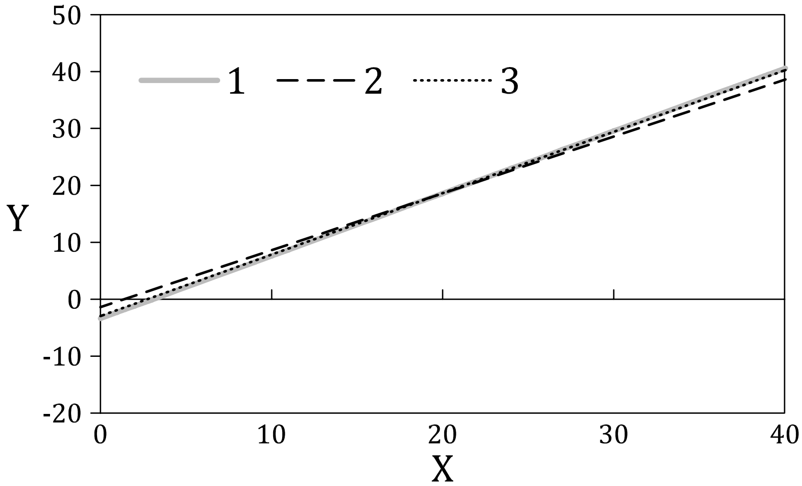
Y equivalent temperature t_{eq} (°C)

1 constant temperature mode

2 constant heat flux mode

3 comfort equation mode

Figure D.2 — Calculation results of t_{eq} with three control modes, calculation conditions: t_a = t_r, v_a = 0,5 m/s, S = 0 W/m², I_{cl} = 0 m²°C/W



Key

X air temperature t_a = radiant temperature t_r (°C)

Y equivalent temperature t_{eq} (°C)

1 constant temperature mode

2 constant heat flux mode

3 comfort equation mode

Figure D.3 — Calculation results of t_{eq} with three control modes, calculation conditions: $t_a = t_r$, v_a = 0,5 m/s, S = 0 W/m², I_{cl} = 0,155 m²°C/W

Annex E
(informative)

Calculation method of h_{cal}

E.1 Basic concept of h_{cal} calculation

E.1.1 Aims and selection of method to obtain h_{cal}

Various methods are considered for obtaining the value of h_{cal} for the t_{eq} calculation. As described in the main body of this document, potential options include experimental, typical or calculated values. The first two can be obtained experimentally, the procedures for which do not need to be described here. The aim of this annex is limited to defining the method and procedure for obtaining the h_{cal} value via CFD calculation.

E.1.2 Flexibility of the calculation method

One issue encountered when calculating h_{cal} is calculating the radiation, including the reflection effects of the surface of the manikin and the wall of the standard chamber. Using the calculation method, it is possible to arbitrarily select between two cases: where the influence of reflection is treated approximately and where it is treated as strictly theoretical. Another issue is creating correspondence between the calculation method and the manikin control algorithm. For this, it is possible to arbitrarily select between three types of control algorithms: the constant temperature mode, the constant heat flux mode and the comfort equation mode.

E.2 Preconditions and assumptions of the method

Using the calculation method, in order to remain consistent with the provision in ISO 14505-2, h_{cal} is calculated based on the following preconditions and assumptions:

a) This method applies only to naked body conditions. For the clothed condition, based on the results calculated using the naked body condition, the influence of the thermal resistance and the expansion effect of the surface area are corrected theoretically and evaluated. After the value of h_{cal} with a naked body ($= h_{cs} + h_{rs}$) has been computed, h_{cal} with clothing can be determined via Formula (10).

b) Inflow temperature $t_{a,in}$ of the air conditioning flow in the standard chamber and the volumetric flow rate $\dot{V}_{a,in}$ are assumed to be given parameters. Here, the flow rate is determined to be the averaged flow velocity of u_a = 0,05 m/s passing through the cross-section of the standard chamber. The inflow temperature should be set as $t_{a,in}$ = 24,0 °C and the maximum and minimum temperature conditions of the environment should be also evaluated. However, when the manikin is operated using the comfort equation mode, only the $t_{a,in}$ = 24,0 °C condition is required.

c) It is assumed that the wall temperature of the standard chamber t_w is uniform.

d) The wall surface temperature t_w is determined such that the averaged air temperature in the environmental chamber t_a and the mean radiant temperature on the wall surface t_r are equal.

e) The surface of the manikin and the wall of the standard chamber are assumed to be a black or grey body and their emissivities (ε_{sk} and ε_w) are given based on the assumption that their surfaces are uniform.

f) The CFD calculation imitates the geometry of the manikin and accounts for the interference of the radiation effects between local parts of the manikin.

g) In the CFD calculation, the heat transfer characteristics are evaluated by combining radiation and convection while considering the influence of buoyancy.

E.3 Calculation method

E.3.1 "Main routine" of the calculation

The "main routine" of the calculation consists of processes a) to m), as follows. The "manikin-environment simulation" described in E.3.3 is implemented in the "main routine" as a sub-process. Figure E.1. shows a flow chart of the "main routine."

a) Set boundaries and calculation conditions.

1) Give values for $t_{\mathrm{a,in}}$, t_{sk}, t_{w}, Δt_{o}, Q_{whole}, $\dot{V}_{\mathrm{a,in}}$, $\varepsilon_{\mathrm{sk}}$ and ε_{w}.

2) Depending on the selected calculation method, t_{sk} or Q_{whole} will be treated as a provisional value or ignored.

b) Calculate the absorption factor $B_{\mathrm{i,j}}$ including choice of reflection effect treatment (see E.3.2).

c) Calculate conversion factor ξ_{m} between the actual and mean radiant temperatures using the following formula:

$$\xi_{\mathrm{m}} = \varepsilon_{\mathrm{sk}} \cdot \sum_{\mathrm{i}=1}^{m_{\mathrm{b}}} \sum_{\mathrm{j}=m_{\mathrm{b}}+1}^{m_{\mathrm{b}}+m_{\mathrm{w}}} \left(B_{\mathrm{i,j}} \cdot \frac{A_{\mathrm{e,i}}}{A_{\mathrm{b}}} \right)$$

d) Apply "manikin-environment simulation" process (see E.3.3).

e) Calculate the averaged air temperature t_{a} in the standard chamber using the following formula:

$$t_{\mathrm{a}} = \frac{1}{V_0} \sum_{\mathrm{k}=1}^{m_{\mathrm{v}}} \left(t_{\mathrm{a,e,k}} \cdot V_{\mathrm{e,k}} \right)$$

f) Estimate the mean radiant temperature T_{r} from the wall temperature T_{w} using the following formula:

$$T_{\mathrm{r}}^4 = \xi_{\mathrm{m}} \cdot T_{\mathrm{w}}^4$$

g) Calculate iteratively until the difference between t_{a} and t_{r} is less than Δt_{o}, implying the convergence state has been reached. Here, the recommended Δt_{o} value is on the order of 0,01.

h) Update the wall temperature t_{w} (T_{w}) for t_{r} (T_{r}) to be t_{a} (T_{a}) using the following formula:

$$T_{\mathrm{w}}^4 = \frac{T_{\mathrm{a}}^4}{\xi_{\mathrm{m}}}$$

i) After the convergence state is reached, t_{a} ($= t_{\mathrm{r}}$) is redefined as the operative temperature t_{o}.

j) Calculate the averaged heat flux Q_{whole} over the entire manikin using the following formula:

$$Q_{\mathrm{whole}} = \sum_{\mathrm{i}=1}^{m_{\mathrm{b}}} \left(Q_{\mathrm{e,i}} \cdot \frac{A_{\mathrm{e,i}}}{A_{\mathrm{b}}} \right)$$

k) Calculate the local heat flux Q_n over the segment n using the following formula:

$$Q_\mathrm{n} = \sum_{\mathrm{i}=m_{\mathrm{s,n}}}^{m_{\mathrm{e,n}}} \left(Q_{\mathrm{e,i}} \cdot \frac{A_{\mathrm{e,i}}}{A_\mathrm{n}} \right) \quad \text{for n} = 1 \sim n_{\mathrm{seg}}$$

l) Calculate overall value of $h_{\mathrm{cal,whole}}$ using the following formula:

$$h_{\mathrm{cal,whole}} = \frac{Q_{\mathrm{whole}}}{A_\mathrm{b} \cdot \left(t_{\mathrm{sk,whole}} - t_\mathrm{o} \right)}$$

m) Calculate the local value of $h_{\mathrm{cal,n}}$ using the following formula:

$$h_{\mathrm{cal,n}} = \frac{Q_\mathrm{n}}{A_\mathrm{n} \cdot \left(t_{\mathrm{sk,n}} - t_\mathrm{o} \right)} \quad \text{for n} = 1 \sim n_{\mathrm{seg}}$$

E.3.2 Calculation of the absorption factor $B_{\mathrm{i,j}}$

E.3.2.1 General

When the radiation is calculated via CFD, the view factor $F_{\mathrm{i,j}}$ is generally also calculated; as such, it is assumed here that the view factor has been obtained previously during prior CFD processing. In process b) of the "main routine" (Figure E.1), it is possible to select one of two methods for determining the absorption factor $B_{\mathrm{i,j}}$: one considering only the absorption effect and the other considering both the absorption and reflection effects. This absorption factor can be determined using the view factor in both cases.

E.3.2.2 Considering the absorption effect but ignoring the reflection effect

This method produces an approximate solution for a convenient calculation. If the emissivity of the manikin surface and the wall surface are both 0,85, the error will theoretically remain below 2 % even if the reflection effect is ignored. If the area of the wall of the standard chamber is much larger than the surface area of the manikin, the error is further reduced. Additionally, if the emissivity of either the manikin or the wall surface is closer to 1,0, the error becomes smaller. Therefore, when an ordinary standard chamber and ordinary manikin are assumed, the results of this approximation are considered sufficiently reliable.

Specifically, in the defining formula (recurrence equation) for the absorption factor $B_{\mathrm{i,j}}$, this approximation is made using the following formula (neglecting the reflection term):

$$B_{\mathrm{i,j}} = F_{\mathrm{i,j}} \cdot \varepsilon_\mathrm{j} \quad \text{for i and j} = 1 \sim m_\mathrm{b} + m_\mathrm{w}$$

where

$\varepsilon_\mathrm{j} = \varepsilon_{\mathrm{sk}}$ for $\mathrm{j} = 1 \sim m_\mathrm{b}$;

$\varepsilon_\mathrm{j} = \varepsilon_w$ for $\mathrm{j} = m_\mathrm{b} + 1 \sim m_\mathrm{b} + m_\mathrm{w}$.

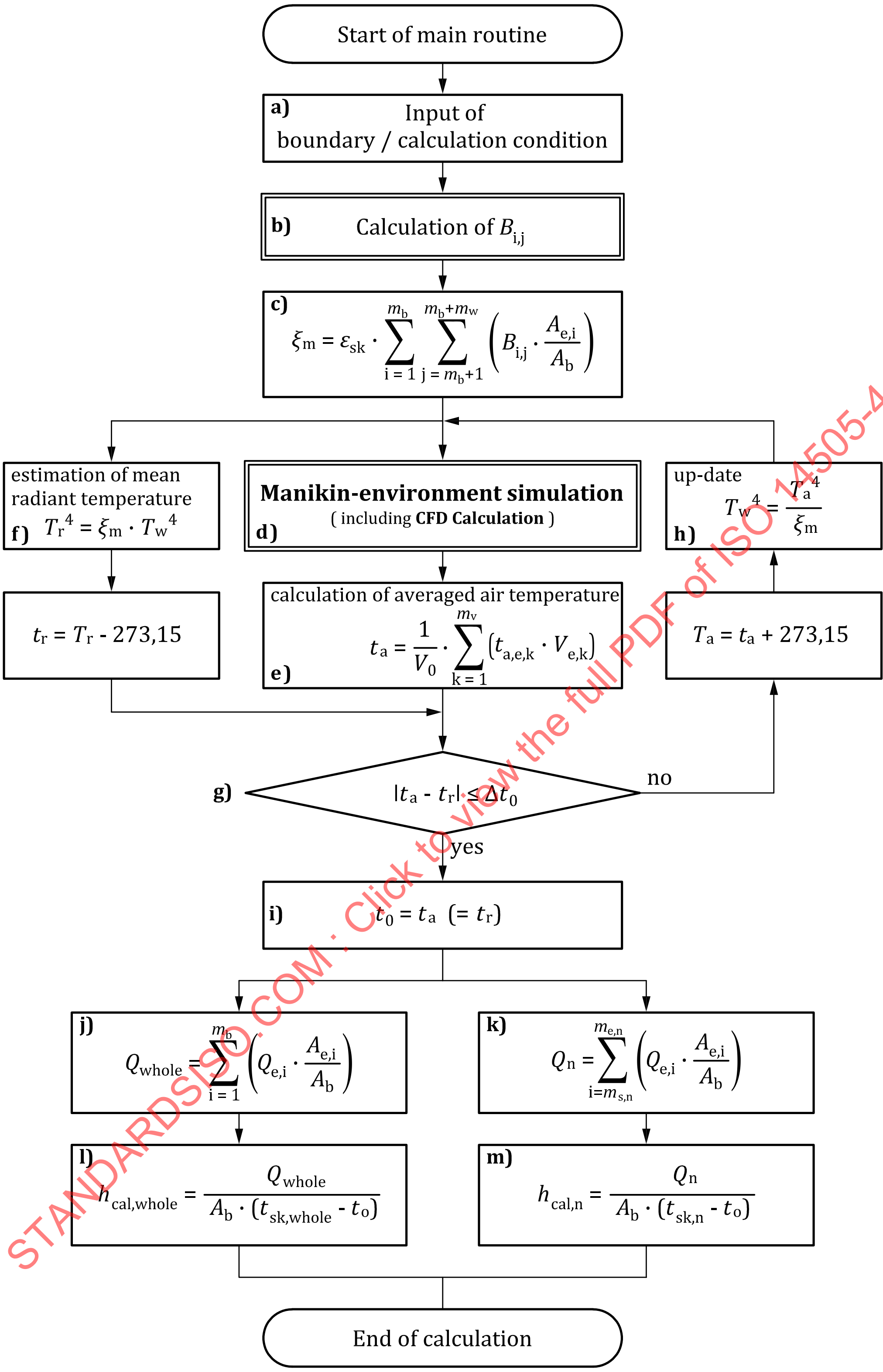


Figure E.1 — Flow chart of the “main outline”

E.3.2.3 Considering both absorption and reflection effects

This method produces an exact solution for a theoretical calculation. This method gives strictly correct theoretical results under any conditions, including cases where the emissivity is small.

Specifically, the absorption factor $B_{i,j}$ is obtained by solving the following recurrence equation:

$$B_{i,j} = F_{i,j} \cdot \varepsilon_j + \sum_{k=1}^{m_b+m_w} \left\{ F_{i,k} \cdot (1-\varepsilon_k) \cdot B_{k,j} \right\} \quad \text{for i and j} = 1 \sim m_b + m_w$$

where

ε_j or $\varepsilon_k = \varepsilon_{sk}$ for j or $k = 1 \sim m_b$;

ε_j or $\varepsilon_k = \varepsilon_w$ for j or $k = m_b + 1 \sim m_b + m_w$.

However, this method requires enormous computational capacity when using surface elements from CFD. Therefore, this method is only practical for use when the number of surface elements is small.

E.3.3 "Manikin-environment simulation" for meeting control algorithms

E.3.3.1 General

Regarding the "manikin-environment simulation" referred to in process d) of the "main routine", three types of methods are considered to correspond to the manikin control algorithms: i) the constant temperature mode; ii) the constant heat flux mode; and iii) the comfort equation mode. According to the selected algorithm, one of following methods should be adopted.

E.3.3.2 Constant temperature mode

This method is adopted when the surface temperature of the manikin is uniformly controlled at t_{sk}. As shown in Figure E.2, it is calculated using the following procedure:

a) Define the thermal boundary condition of the surface elements using the following formula:

   $t_{sk,e,i} = t_{sk}$ for $i = 1 \sim m_b$

b) Calculate the convection and radiation fields simultaneously via CFD, where $t_{sk,e,i}$ is given as the Dirichlet condition.

c) Obtain the calculated results of the heat generation distribution over the surface element $Q_{e,i}$.

d) Return to the "main routine."

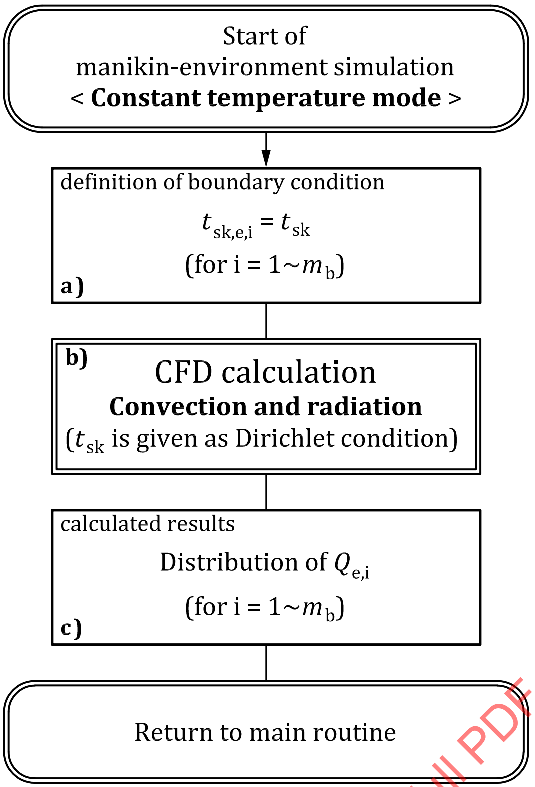


Figure E.2 — Algorithm for the constant temperature mode

E.3.3.3 Constant heat flux mode

This method is adopted when the heat generated by the manikin is uniformly controlled to be Q_{whole}. As shown in Figure E.3, it is calculated using the following procedure:

a) Define the thermal boundary condition on the surface elements using the following formula:

$$Q_{e,i} = Q_{whole} \quad \text{for } i = 1 \sim m_b$$

b) Calculate the convection and radiation fields simultaneously via CFD, where $Q_{e,i}$ is given as the Neumann condition.

c) Obtain the calculated results of the temperature distribution over the surface element $t_{sk,e,i}$.

d) Return to the “main routine”.

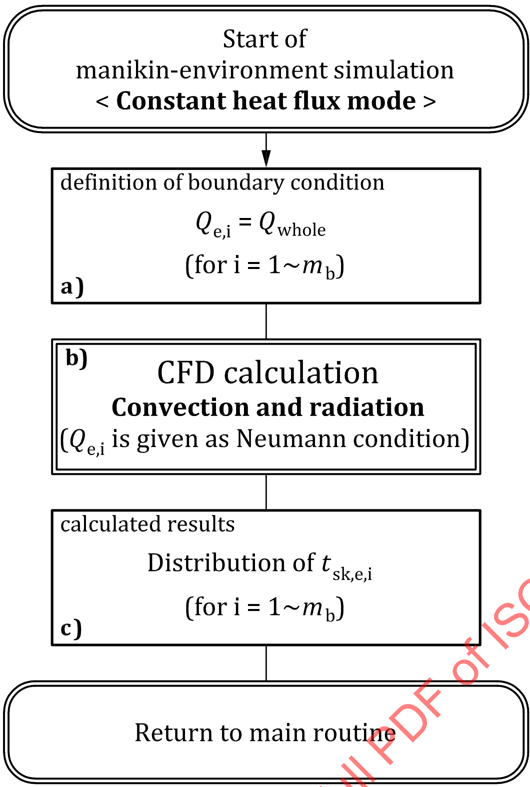


Figure E.3 — Algorithm for the constant heat flux mode

E.3.3.4 Comfort equation mode

This method is adopted when the manikin is controlled by the comfort equation. Generally, R_{cr} = 0,054 and t_{cr} = 36,4 are used for this control. In this case, the distribution of the temperature and the generated heat are determined such that the comfort equation is satisfied independently over each local segment of the manikin. As shown in Figure E.4, this is calculated by the following procedure:

a) As an initial condition, the provisional values of the generated heat $Q_{e,i}$ and Q_n are given as equal for all surface elements and local segments by the following formula, where t_{sk} is also a provisional value input in the “main routine”:

$$Q_{e,i} = Q_n = \frac{36,4 - t_{sk}}{0,054} \quad \text{for } i = 1 \sim m_b, \, n = 1 \sim n_{seg}$$

b) Calculate the convection and radiation fields simultaneously via CFD, where $Q_{e,i}$ is given as the Neumann condition.

c) Obtain the calculated results for the temperature distribution over the surface element $t_{sk,e,k}$.

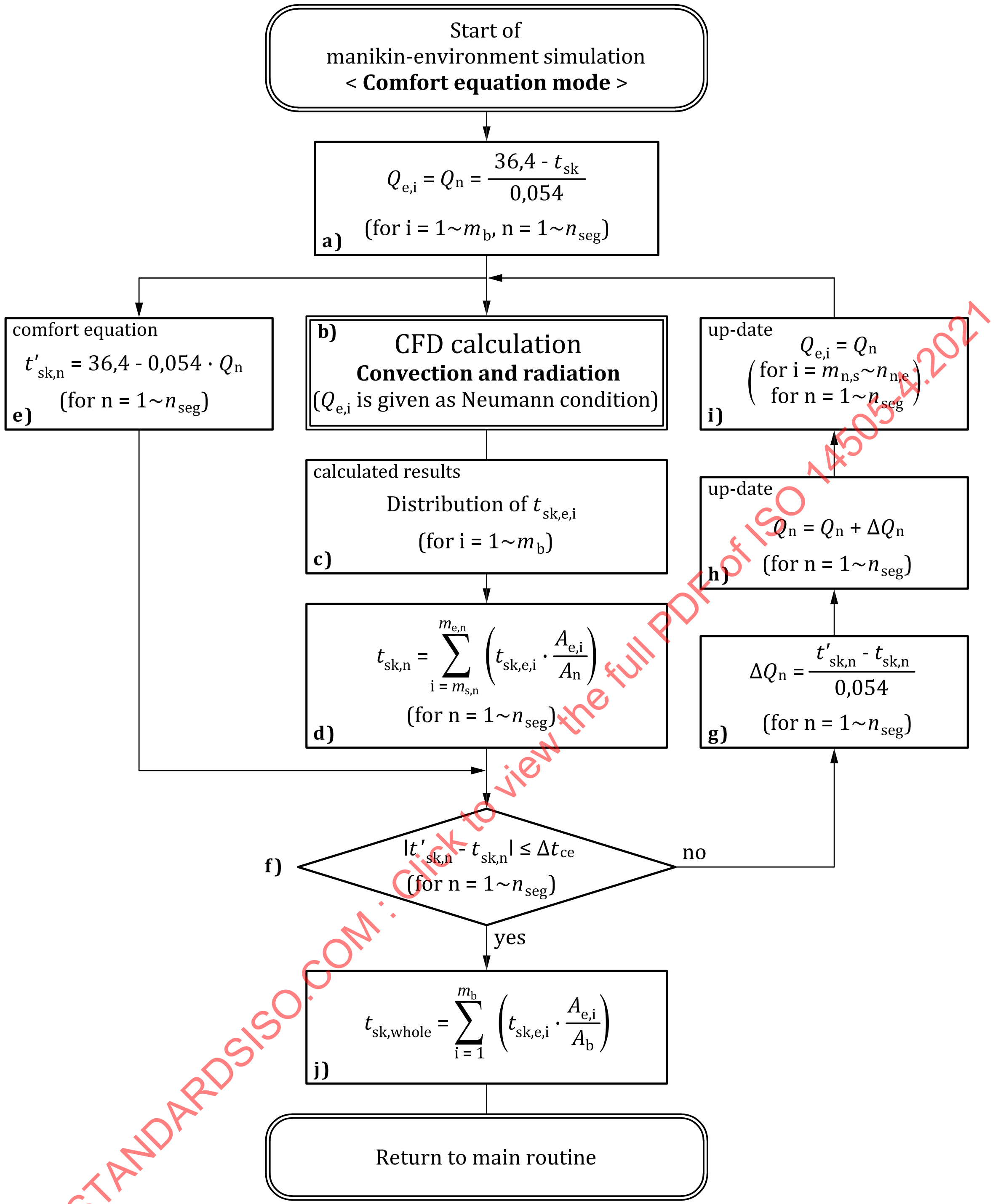


Figure E.4 — Algorithm for the comfort equation mode

d) The averaged surface temperature $t_{sk,n}$ of the local segment n is calculated using the following formula:

$$t_{sk,n} = \sum_{i=m_{s,n}}^{m_{e,n}} \left(t_{sk,e,i} \cdot \frac{A_{e,i}}{A_n} \right) \quad \text{for } n = 1 \sim n_{seg}$$

e) Estimate the averaged surface temperature $t'_{sk,n}$ of the local segment n based on the following comfort equation:

$$t'_{sk,n} = 36{,}4 - 0{,}054 \cdot Q_n \quad \text{for } n = 1 \sim n_{seg}$$

f) Calculate iteratively until the difference between $t'_{sk,n}$ and $t_{sk,n}$ is less than Δt_{ce}, implying the convergence state has been reached. Here, the recommended value of Δt_{ce} is on the order of 0,01.

g) Calculate the correction amount for the generated heat ΔQ_n for each local segment n using the following formula:

$$\Delta Q_n = \frac{t'_{sk,n} - t_{sk,n}}{0{,}054} \quad \text{for } n = 1 \sim n_{seg}$$

h) Update the generated heat Q_n for each local segment n using the following formula:

$$Q_n = Q_n + \Delta Q_n \quad \text{for } n = 1 \sim n_{seg}$$

i) Update the generated heat $Q_{e,i}$ for each surface element i using the following formula:

$$Q_{e,i} = Q_n \quad \text{for } i = m_{n,s} \sim m_{n,e}\text{ , } n = 1 \sim n_{seg}$$

j) After the convergence state is reached, $t_{sk,whole}$ is calculated using the following formula:

$$t_{sk,whole} = \sum_{i=1}^{m_b} \left(t_{sk,e,i} \cdot \frac{A_{e,i}}{A_b} \right)$$

k) Return to the “main routine”.

Among commercial software packages, some CFD solvers are equipped with an option labelled “thermal resistance boundary condition.” This allows the imaginary core temperature $T_{cr,e,i}$ and the thermal resistance $R_{cr,e,i}$ at the surface element i to be specified as a boundary condition and supports iterative calculations using the logic detailed above. If this option is installed, then by employing the process as shown in Figure E.5 instead of Figure E.4 it is possible to utilize “thermal comfort equation mode.” Here, at step a), $T_{cr,e,i}$ = 36,4 °C and $R_{cr,e,i}$ = 0,054 m²K/W are given as Dirichlet conditions.