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Thermal Interaction Between a Human Body and a Vehicle Cabin

Dragan Ružić and Ferenc Časnji

Additional information is available at the end of the chapter

1. Introduction

The main functions of a vehicle cabin are to provide a comfortable environment for its occupants and to protect them from vibrations, noise and other adverse influences. Since the conditions inside mobile machinery cabins (agricultural tractors and construction equipment) affect the health, performance and comfort of the operator, this research focuses on these types of cabins. Modern agricultural tractors are complex, highly efficient systems, and the aim of the development of these systems is the reduction of the negative impact they have on the environment. Main goals are achieving higher fuel economy, lower emission and less soil compaction caused by tractor wheels. There is also a tendency to improve the ergonomics of operator enclosures. An operator enclosure can be treated as workspace and the conditions inside a tractor cab have significant impact on the performance of the operator, and in that way on the total efficiency of the *operator-tractor-environment* system as well. From the operator's point of view, tractor cab ergonomics is a key factor in ensuring his optimum working performance, which could easily become the weakest link in the working process. While, on the one hand, the tractor cab offers mechanical protection and the protection from adverse ambient conditions, on the other hand, even under moderate outside conditions, a closed tractor cab acts like a greenhouse and its interior could become unpleasant, unbearable and even dangerous.

The project presented in the report by Bohm et al. (2002), deals with the thermal effects of glazing in cabs with large glass areas. Using the thermal manikin AIMAN, they studied and evaluated the effects of different kinds of glass and design of the windows, as well as the effects of sun protection and insulation glazing. The results showed that neither in severe winter conditions, nor in sunny summer conditions, could acceptable climate be obtained with standard glazing in cabs with large glass areas.

In a comprehensive project which was aimed at the reduction of vehicle auxiliary load, done by National Renewable Energy Laboratory, Golden, USA (Rugh and Farrington, 2008), a

variety of research methods was used to research and develop innovative techniques and technologies that reduce the amount of fuel needed for the air-conditioning system, by lowering thermal loads. They concluded that the reflection of the solar radiation incident on the vehicle glass is the most important factor in making significant reductions in thermal loads. The use of solar-reflective glass reduced the average air temperature and the seat temperature. The use of reflective shades and electrochromic switchable glazing are also effective techniques for the reduction of the solar energy that enters the passenger compartment. They also found that solar-reflective coatings on exterior opaque surfaces and vehicle body insulation can reduce a vehicle's interior temperature, but they can do this to a lesser extent than solar-reflective glazing, shades, and parked-car ventilation.

Both projects aimed at the reduction of thermal loads in vehicle cabins, primarily considering the advanced techniques of heat rejection and insulation. The research described in the paper by Currle and Maue (2000) deals with the numerical study of the influence of air flow and vent geometrical parameters (area and position of the vents) on thermal comfort of passenger car occupants. The results showed significant influence of the vent area, not only on velocity levels in the cabin, but also on overall as well as local thermal comfort.

This chapter deals with the characteristics of the agricultural tractor cabs that have direct or indirect effect on thermal ergonomics of operator enclosures. For this purpose, thermal processes between hot (summer) environment and a tractor cab, as well as between the cab and an operator are analysed. The hot conditions are more complex and more demanding in terms of achieving comfortable thermal conditions in a tractor cab. The complexity of thermal interactions between a human, a tractor cab and the environment is shown in Fig. 1.
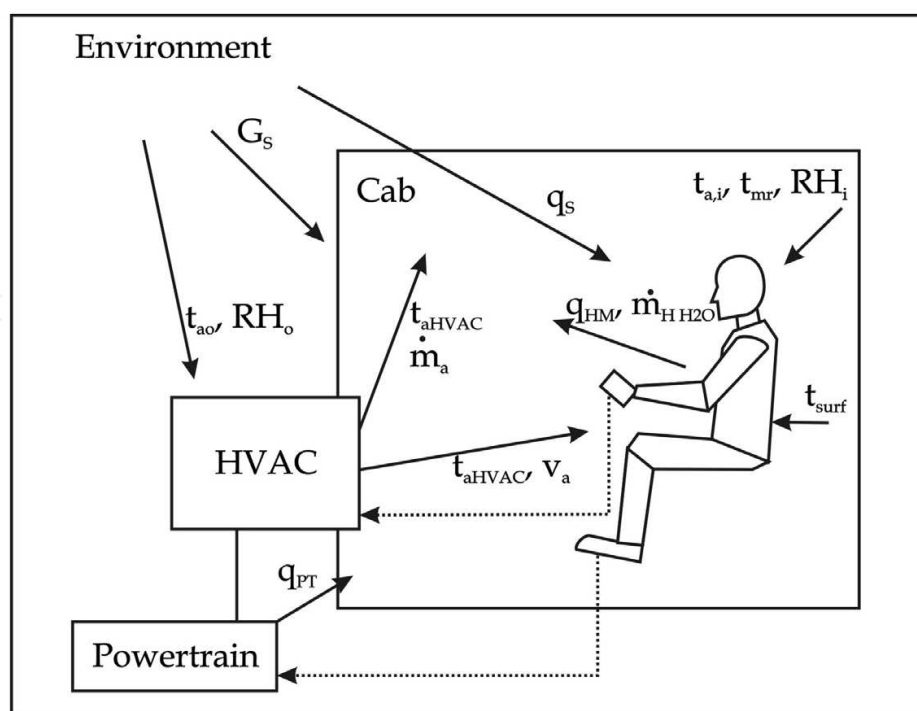

Figure 1. Thermal interaction between a human, a cabin and the environment. Dotted line arrows are control paths

In order to be able to improve thermal ergonomics, it is necessary to have a good knowledge of thermal properties of materials and other design parameters of importance for the cabin thermal processes, as well as the knowledge of the properties of the human body and its thermoregulation system. The main methods for both the improvement of the microclimatic conditions and the reduction of air-conditioning energy consumption in any type of cabin are:

- Passive:
  - prevention of cabin heat gain, using principles of heat rejection and insulation;
  - prevention of heat transfer to the operator's body by reducing the transferred solar radiation, as well as thermal radiation and heat transfer from the cabin interior surfaces.
- Active: cooling the operator's body by the effective use of airflow from the air-conditioning system.

The aims of the research are:

- to identify and evaluate the most important influences on heat transfer processes in the human-cabin-environment system,
- to explain the effects of the ventilation (air distribution) system design and settings on the operator's heat loss and thermal sensation in hot conditions.

2. Main design features of tractor cabs

An overview of the design features related to the thermal processes and HVAC characteristics was done using relevant technical documentation and by analysing the design of several agricultural tractors (Ružić and Časnji, 2011). Orchard and vineyard tractors (narrow track tractors) were not included in the analysis. All tractors from the sample were 4WD wheeled tractors, with power ranging from 40.5 to 155 kW, their weight ranging from 2750 to 8410 kg and their wheelbase ranging from 2.055 to 3.089 m. The outer lengths of their cabs were between 1.40 and 1.77 m, the cab widths were between 1.38 and 1.70 m, and the cab heights were between 1.45 and 1.80 m.

2.1. Materials

The basic materials for tractor cab frames are steel profiles, which primarily must meet the demands for mechanical protection of the operator. Cab frames usually have six pillars, although there are designs with four pillars.

The floor is made of a steel sheet supported by a steel frame. For the purposes of sound and heat insulation, the floor has an interior lining and, in some designs, an outer lining. The powertrain (transmission) of typical tractors is positioned beneath the cab. The powertrain of smaller tractors partly "protrudes" into the cab space, while larger tractors have a flat floor. In the interior, rubber or polymer flooring is used. Some cabs have a foam insulation material between the flooring and the floor. If there is additional heat and sound insulation beneath the floor, composite materials are used. The typical composite heat shield is made of reflective outer aluminium layer bonded on synthetic fibre core.

Cab roofs are made of polymers, and the cab can be equipped with a roof-window. The components of the ventilation system and the air-conditioning are generally placed in the roof.

Cab glazing in modern tractors takes approximately 60% of the total cab surface area. Glass is generally tempered and tinted. In the sample of analysed tractors, the windshield inclination is in the range of 8 – 20° and side windows inclination is in the range of 7 – 10°.

An overview of thermal properties of materials used in tractor cabs is given in Table 1.

	thermal conductivity, k (W/mK)	surface emissivity, ε	solar absorptivity, α_s	solar transmissivity normal to surface, τ
tempered single glass, clear	0.8	0.8 – 0.95	0.08	0.84 – 0.90
tempered single glass, green tinted	0.8	0.8 – 0.95	0.45	0.49
metal, painted white	40 – 45	0.85 – 0.96	0.21 – 0.25	-
polymer, white	0.12 – 0.20	0.90 – 0.97	0.23 – 0.49	-
metal, painted black	40 – 45	0.97	0.80	-
rubber, black	0.10 – 0.13	0.86 – 0.92	-	-
insulating foam	0.037 – 0.050	0.95	-	-
composite heat shield	0.040 – 0.060	0.10 – 0.20	-	-

Table 1. Thermal properties of tractor cab materials (McAdams, 1969; Incropera and DeWitt, 1981; Siegel & Howell, 1992; ASHRAE, 1997c; Saint-Gobain, 2003)

Cabs usually have an air distribution system with air vents (outlets of ventilation duct system) placed on the ceiling. Some of the cabs from the sample also have vents placed on the instrument panel, usually for heating. The air vents on the ceiling can have a symmetrical or an asymmetrical layout, while the vents on the instrument panel always have a symmetrical layout. The vents mostly have a circular cross-section and the operator can change the direction of the air jet.

Vapour compression systems are used for air-conditioning (AC), same as in automotive applications, with R134a as the refrigerant. Installed compressors can take around 4 – 8 kW from the tractor engine, which presents 2.5 – 15% of the rated power in this category of tractors.

3. Heat exchange between a tractor cab and the environment

Thermal processes in a tractor cab are in close relation to outside thermal conditions (air temperature, air velocity, intensity and direction of solar radiation). Thermal processes are more or less independent from tractor's working operations, whereas the noise and vibrations depend on them. Since the decrease in the cab heat gain means the reduction of

the thermal load to the operator as well, the analysis was done for adverse environmental conditions that can be encountered over the summer period:

- maximum outdoor air temperature above 30°C,
- total solar irradiation on the outer surfaces up to 1000 W/m²,
- closed cab so that the operator is protected from noise and air pollution,
- low tractor velocity that does not promote natural tractor cab ventilation.

The sum of heat gain for a closed tractor cab in hot environment, under the steady-state conditions, consists of the following (Fig. 2.):

- heat transfer through the cab envelope due to the temperature difference Q_k,
- heat transfer through the cab roof caused by solar radiation Q_s,
- solar radiation transmission through glazing Q_{Gs},
- heat gain from the powertrain Q_{PT},
- sensible (Q_{HM}) and latent heat ($Q_{H\,H2O}$) released by the operator.

Heat load by infiltration is assumed to be zero due to the pressurisation of the tractor cab. In order to maintain the interior temperature constant, the heat removal by the air-conditioning (Q_{AC}) should be equal to the heat gain. According to the requirements for thermal comfort in summer conditions, interior air temperature should be in the range of 23 to 28°C (ASHRAE, 2003).
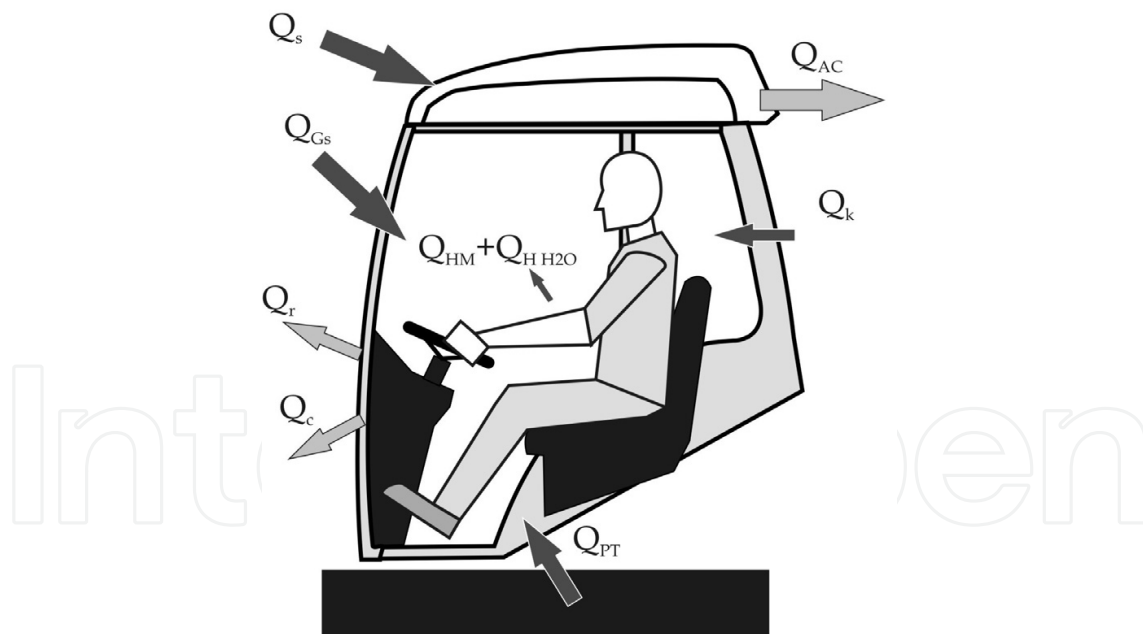

Figure 2. Thermal processes between a tractor cab and hot environment

For an analytical investigation of thermal processes, it was assumed that the heat flow is a one-dimensional steady flow through a plane wall with natural convection on both sides. A problem that comes up in the analytical determination of heat flux is the calculation of the convection heat transfer coefficients, which differ according to various sources (Incropera and DeWitt, 1981; ASHRAE, 1997a; Conceicao, 1999; Grossmann, 2010). Radiant heat

exchanges among inner surfaces were neglected, because of the complexity of surface shapes and relatively small differences in temperature. The thermal radiation inside the cab is important for the thermal load of the operator, and this will be discussed later.

3.1. Heat transfer through cab walls due to temperature differences

Assuming that the air temperature in the vicinity of the walls is uniform, the one-dimensional heat load through this part of cab walls is (Fig. 3):

$$Q_k = A_k \cdot q_k = A_k \cdot U \cdot (t_{a,o} - t_{a,i}), \text{ W} \tag{1}$$

Parameter U is the total heat transfer coefficient, which combines convections from each side of the wall with conduction. General equation for a wall of area A_j with s layers:

$$U_j = \cfrac{1}{\cfrac{1}{h_{c,i}} + \sum \cfrac{\delta_s}{k_s} + \cfrac{1}{h_{c,o}}}, \text{ W / m}^2\text{K} \tag{2}$$

For all the surfaces with the heat transfer due to the difference in air temperature, the total value of U is:

$$U = \frac{1}{A_k} \sum U_j A_j, \text{ W / m}^2\text{K} \tag{3}$$

This value is dependent on the thermal characteristics of the material that walls are made of (thermal conductivity k, W/mK), of the thickness of the layers and the surface area. The surfaces where this equation applies are the surfaces exposed neither to the solar radiation nor to the radiation from the powertrain. For example, these surfaces are windows on the shaded side of the cab.
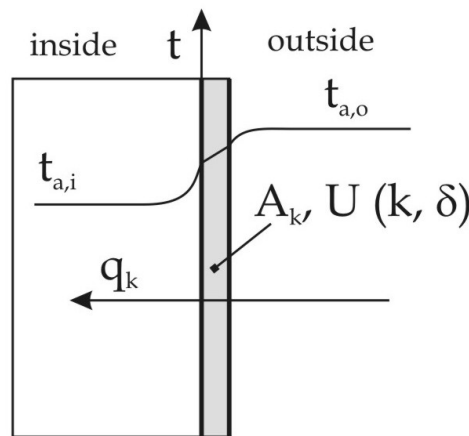

Figure 3. Heat transfer through the cab wall due to air temperature difference

Based on very few available sources, it is assumed here that the cab of a stationary vehicle has U value of around 10 W/m²K (Türler, 2003; Karlsson, 2007; Tavast, 2007; Grossmann,

2010). Consequently, the estimated heat flux through the cab wall (q_k) would be around 40 W/m²for the difference between the outside and the inside air temperature of (30 – 26)°C = 4°.

3.2. Effects of solar radiation

Cab glazing is a semitransparent medium, where solar irradiation can be partially reflected, absorbed and transmitted, Fig. 4.
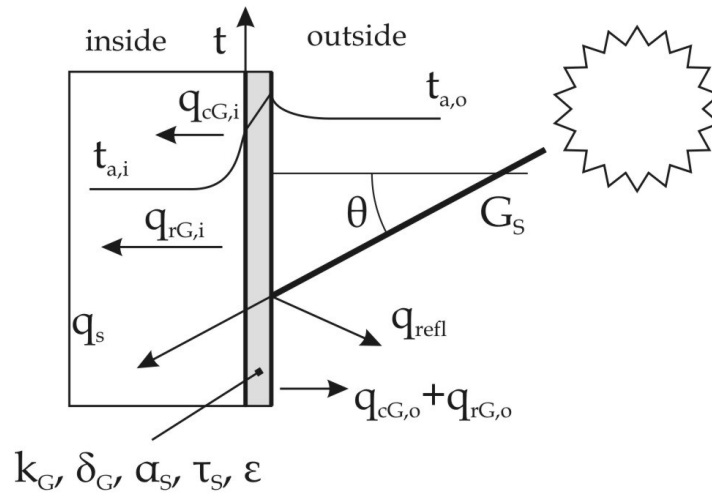

Figure 4. Solar energy transmission through glass

Solar irradiation on a tractor cab surface is variable depending on the position of the sun as well as the orientation of the cab surfaces. An example of maximum ("clear sky") intensity of normal irradiation and its variations in time and the influence of the surface inclination are shown in Fig. 5. The values are given for a vertical surface and for the surface inclined at 20°. The diagram shows that the vertically positioned glass receives around 15% less solar irradiation than the glass inclined at the angle of 20°.

The thermal balance equation for the outer glass surface can be written as:

$$G_s \cdot \alpha_s - \frac{k_G}{\delta_G}\left(t_{G,o} - t_{G,i}\right) - h_{G,o}\left(t_{G,o} - t_{a,o}\right) - \varepsilon \cdot \sigma\left(T_{G,o}^4 - T_{sky}^4\right) = 0 \tag{4}$$

The thermal balance on the inner glass surface is:

$$\frac{k_G}{\delta_G}\left(t_{G,o} - t_{G,i}\right) - h_{G,i}\left(t_{G,i} - t_{a,i}\right) - q_{rG,i} = 0 \tag{5}$$

The total heat flux through the glass exposed to the solar radiation consists of the transmitted part of the solar radiation, the convection on the inner side and longwave radiation (neglected here due to a small temperature difference between interior surfaces):

$$q_s = G_s \cdot \tau_s + h_c\left(t_{G,i} - t_{a,i}\right), \text{ W / m}^2 \tag{6}$$

Using available values of the thermal properties of glass (Table 1) with the solar irradiation on inclined glass equal to 876 W/m², the estimated total heat flux through the glass will be around 740 W/m² for clear glass, and around 530 W/m² for tinted glass (green, with 75% transmittance of visible light; Saint-Gobain Sekurit, 2003). The air temperature difference was the same as above, i.e. 4°C. The drawback of use of the tinted glass is its solar radiation absorptivity, which causes the rise in the temperature of glass. In this example, the calculated glass temperatures will be around 30°C and 50°C, for clear and tinted glass respectively. In comparison to solar absorbing glass, a better but also a more expensive solution is the infra-red reflective glass that rejects almost a half of the solar radiation energy with less obstruction of visible light transmission (Bohm, 2002; Türler, 2003; Saint-Gobain, 2003). This kind of glass is used in automotive applications.
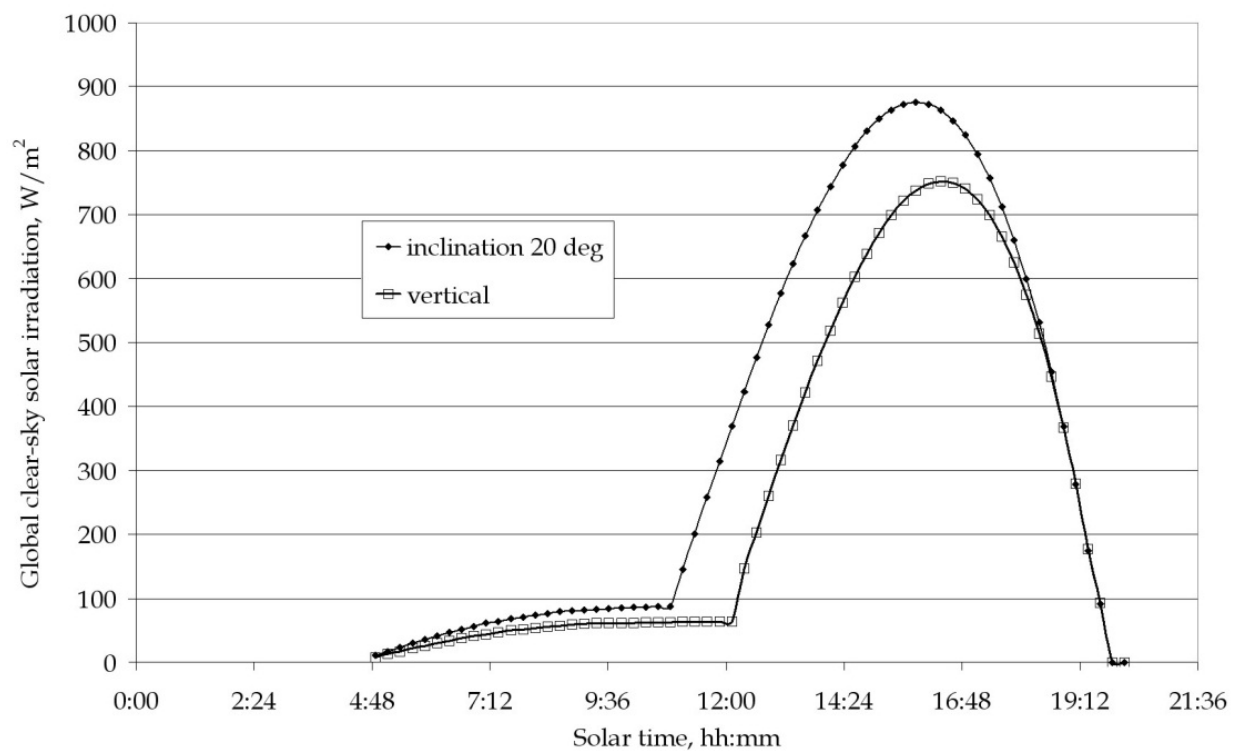

Figure 5. Global solar clear-sky irradiation G_s on a west-faced vertical plane and on a plane inclined at the angle of 20° from vertical, in a central European region on a day in July
(http://re.jrc.ec.europa.eu/pvgis/apps3/pvest.php)

The total amount of the heat transmitted through the glass caused by solar radiation is related to the normal projection of the tractor cab in the direction of radiation. The maximum solar transmissivity of glass is in the region of incident angles that are less than 30°, being the most inconvenient case both in terms of solar radiation transmittance to the cab interior as well as the direct effect on the operator.

The absorbed part of the solar irradiation heats the cab's outer opaque surfaces (the roof, for example) and their temperature rises. The maximum solar irradiation on a horizontal roof surface in the central Europe region on a summer day may exceed 900 W/m². After some exposure to solar radiation, the thermal balance between heat gain and heat release will be

achieved. The heat from the outer surface is released into the environment (mostly to the sky) by longwave radiation, and into the surrounding air by convection. From the inner surface, the heat from the heated wall of the cabin is realeased into the air by convection and it is emitted by longwave radiation. The cab roof is an example of a surface where this mode of heat transfer takes place, Fig. 6.
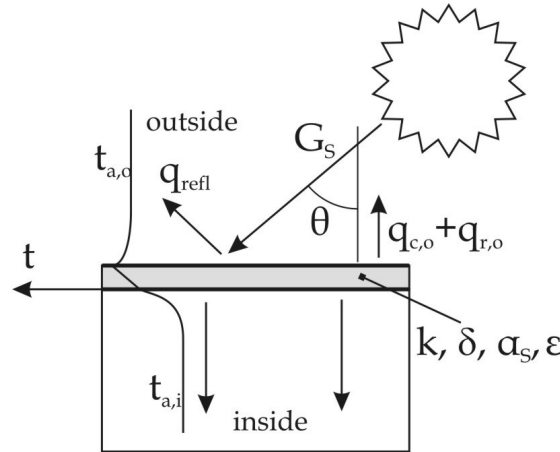

Figure 6. Heat transfer through the cab roof caused by solar radiation.

If there are any components of a ventilation system and an AC system in the roof, the heat transfer from the hot outer roof surface is not direct, but is transferred through the air space. The thermal balance equation for the outer surface of the roof can be written as:

$$G_s \cdot \alpha_s - \varepsilon \cdot \sigma \left(T_{roof,o}^4 - T_{sky}^4 \right) - h_{c,o}(t_{roof,o} - t_{a,o}) - \frac{k_{roof}}{\delta_{roof}}(t_{roof,o} - t_{roof,i}) = 0 \qquad (7)$$

The thermal balance on the inner roof surface is:

$$\frac{k_{roof}}{\delta_{roof}} \left(t_{roof,o} - t_{roof,i} \right) - h_{roof,i} \left(t_{roof,i} - t_{a,i} \right) - q_{r,roof,i} = 0 \qquad (8)$$

Using an example of the heat flux assessment for an opaque roof, treating the roof as a single layer wall, design parameters of the roof will be discussed. The radiation heat exchange between interior surfaces is neglected here, supposing that temperature differences among the surfaces are small. Hence, the equation of heat flux that flows through the roof, caused by solar radiation, is:

$$q_s = \alpha_s \cdot G_s - \varepsilon \cdot \sigma \left(T_{roof,o}^4 - T_{sky}^4 \right) - h_{c,o}(t_{roof,o} - t_{a,o}) =$$
$$= \frac{k_{roof}}{\delta_{roof}}(t_{roof,o} - t_{roof,i}) = h_{c,i}(t_{roof,i} - t_{a,i}) \qquad (9)$$

For example, the normalized data of heat flux and relative surface temperature, for 800 W/m² of solar irradiation on a horizontal surface, are given in Table 2. Four designs are analysed here, the cabs with roofs made of metal or polyester, both with and without a

thermal insulating material on the inner side. Solar absorptivity of the outer surface is 0.3 and the AC keeps the interior air temperature at 26°C.

	Metal	Metal with insulation	Polyester	Polyester with insulation
Heat flux through the roof	100% (~30 W/m²)	70%	90%	60%
Inner surface temperature reduction	0 (surface temperature ~45°C)	6°	2°	7°
Inner surface radiation	100%	94%	99%	94%

Table 2. Evaluated relative values of the heat transfer parameters for different roof materials

As it was in the case with cab glazing, irradiation and solar absorptivity coefficient are dependent on the solar radiation incident angle, θ. The worst case is, of course, at noon, when the radiation is almost normal to the surface.

Solar absorptivity is also dependent on the colour of the surface. In terms of colour, light colours are preferable, and are widely used for tractor roofs. Under the same conditions, but with a dark coloured roof with solar absorptivity of 0.9, the heat transfer rate increases three times in comparison with the roof that has the solar absorptivity of 0.3. Also, the temperature of the roof interior will be 18 – 30°C higher, while the radiant heat flux emitted by the roof's inner surface would in this case rise more than 50%. Despite the high thermal conductivity, one of the most popular materials for the reflection of thermal radiation is aluminium (α_s = 0.09 – 0.15; Incropera and DeWitt, 1981; Siegel & Howell, 1992), but it is not used for these purposes in tractor cabs.

Obviously, the solar absorptivity and longwave emissivity of the roof material, as well as thermal conductivity, are very important design factors. Therefore, the outer roof surface should have a low solar absorptivity coefficient and high emissivity (small α_s/ε ratio is preferable), with low thermal conductivity at the same time. However, in order to reduce thermal radiation towards the operator's body surface, the inner roof surface should have low thermal emissivity. Unfortunately, this is not the case with the kinds of materials usually used for the underside of the roof.

3.3. Heat gain from the powertrain

Heat transfer through the floor is modelled as the sum of the radiation from hot powertrain surfaces and the natural convection from the surrounding air, as in Fig. 7. It is assumed that the air and the outer surfaces of the powertrain have the temperature of 80°C (based on temperature of the transmission oil; data from tractor service manuals). In addition to the radiative properties and the temperature of surfaces, geometry also has some influence on

the heat exchange. The geometry is described by the view factor. Because the distance between the powertrain and the floor is small, it is assumed here that the view factor is equal to unity (Incropera and DeWitt, 1981).
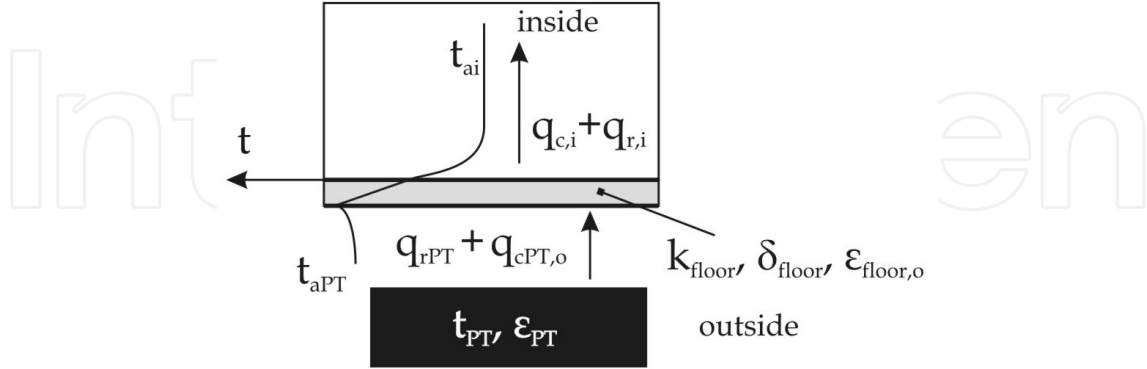

Figure 7. Heat gain from the powertrain

The heat flux that the floor absorbs from the hot powertrain surface and the surrounding air can be written as:

$$q_{PT} = \varepsilon_{PT:floor} \cdot \sigma\left(T_{PT}^4 - T_{floor,o}^4\right) + h_{c,o}\left(t_{a,o} - t_{floor,o}\right), \ W/m^2 \qquad (10)$$

The factor $\varepsilon_{PT:floor}$ takes into account different emissivity of involved surfaces (Mc Adams, 1969):

$$\varepsilon_{PT:floor} = \frac{1}{\dfrac{1}{\varepsilon_{PT}} + \dfrac{1}{\varepsilon_{floor,o}} - 1} \qquad (11)$$

Under the thermal equilibrium, the same heat flux is transferred through the floor and transmitted to the interior air:

$$q_{PT} = \frac{k_{floor}}{\delta_{floor}}\left(t_{floor,o} - t_{floor,i}\right) = h_{c,i}\left(t_{floor,i} - t_{a,i}\right), \ W/m^2 \qquad (12)$$

For a simple floor without a heat shield on the outer side and with rubber flooring only, the estimated value of the heat flux through the cab floor caused by powertrain is around 110 W/m². If there is a heat shield beneath the floor and insulation between the flooring and the metal, the heat gain can be reduced to one third. Consequently, in the first case, the inner temperature of the floor will be above 50°C, while in the second case the temperature will be lower, around 34°C.

3.4. Total thermal load of the tractor cab

The total thermal load of the cab under the chosen summer conditions will depend on the size of the cab and on its orientation to the sun. The worst case would be that the largest side

of the cab is facing the sun and the tractor cab is not equipped with the mentioned means for heat gain reduction. Therefore, for approximately 1/4 of the glazed area exposed to the sun (around 1.5 m²), the resulting solar heat gain would be around 1100 W (in tractors with clear glazing). The rest of the cab surface, (approximately more than 7 m²), would transfer up to 800 W. In total, adding the heat released by the operator, the estimated thermal load of the tractor cab would be around 2 kW. Fig. 8 shows the percentage of different modes of thermal loads and their reduction via the mentioned methods for heat rejection and insulation. The chart in Fig. 8 shows that glazing has the most important role in thermal processes that influence the microclimate conditions inside the tractor cab.
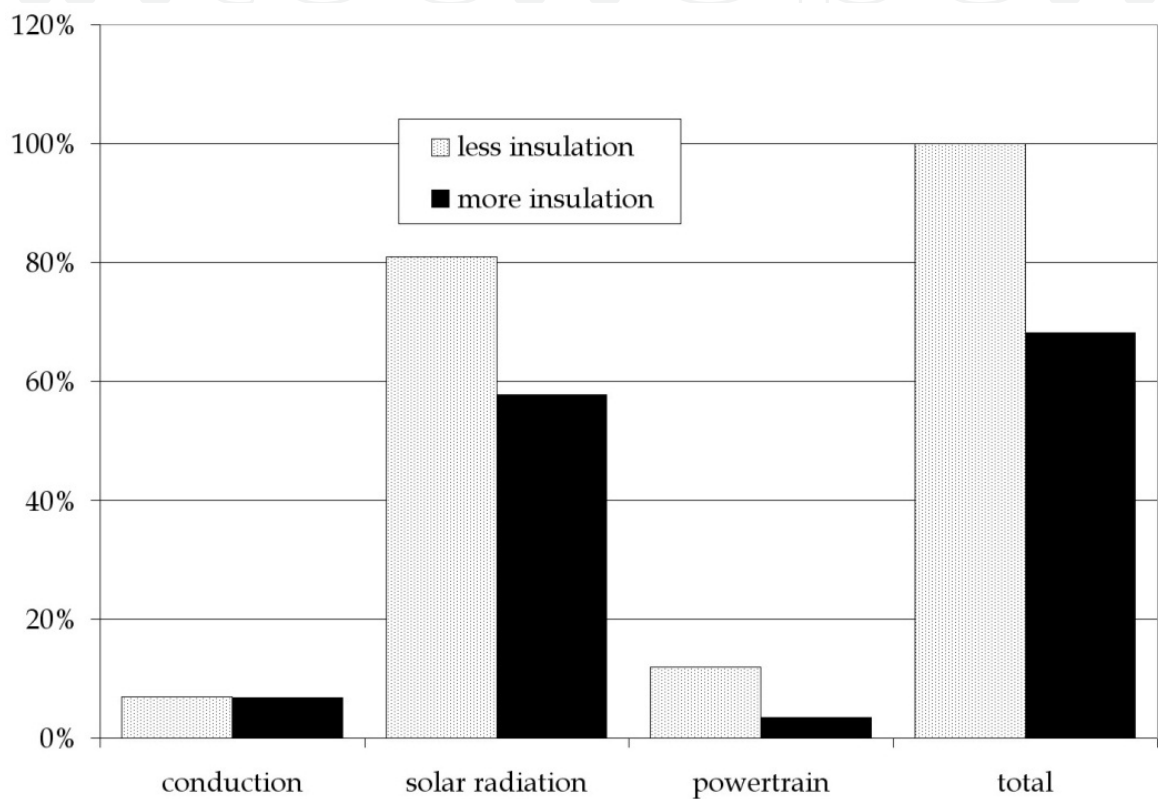

Figure 8. Percentage of heat fluxes of different modes of heat transfer to the tractor cab and the effects of application of the heat rejection and insulation

4. Thermal processes between an operator and a tractor cab interior

The conventional approach for the analysis of thermal processes is based on the balance between the sensible and latent thermal load of the cabin and the heat removal by supplied air. Therefore, quantity and conditions of the air supplied by the AC influences the overall thermal state in the cabin, as shown in Fig. 9. This approach, although technically correct, is not suitable for the assessment and prediction of the operator's thermal sensation. Due to the complex nature of the thermal environment in a tractor cab, it is not possible to accurately describe the thermal conditions inside a tractor cab by a single value, even when the value combines the microclimate factors (for example PMV index; Fanger, 1970). Because of this, other assessment parameters have been developed. One of them is the

equivalent temperature (t_{eq}), defined as the temperature of a homogenous space, with mean radiant temperature equal to air temperature and with zero air velocity, in which a person exchanges the same heat loss by convection and radiation as in the actual conditions. The equivalent temperature can be evaluated for the whole body or for individual body parts.
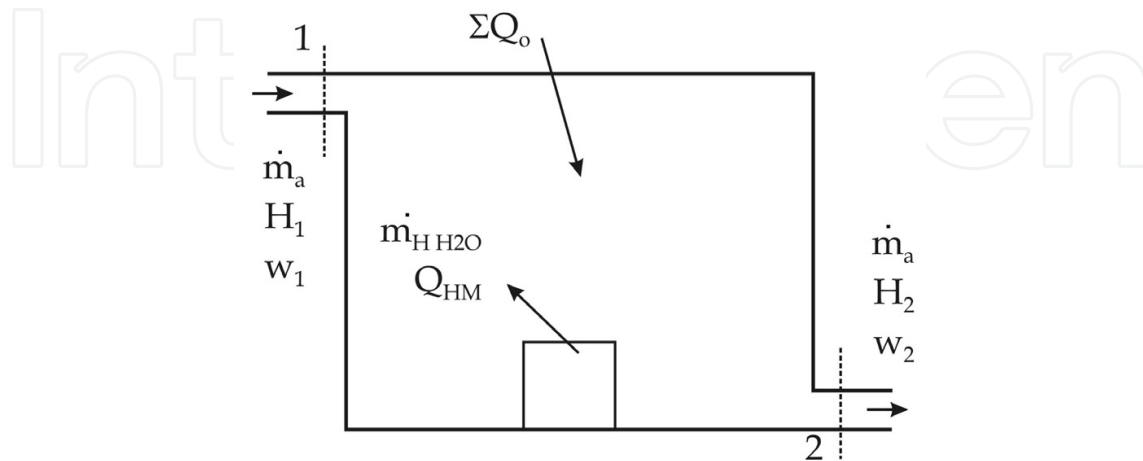

Figure 9. Schematic of the conventional approach to the analysis of air-conditioned space, considering only the air conditions at the inlet (1) and the outlet (2) and the incident rates of energy and moisture gains

In the conventional approach, an operator is treated only as a source of certain amount of sensible and latent heat. The human-based design takes into account the operator's shape, volume, and his thermal sensitivity.

4.1. Human body thermoregulation and thermal sensation

Microclimate conditions and hence human thermal sensation as well, are dependent on air temperature, air velocity, relative humidity and mean radiant temperature. However, individual differences regarding physiological and psychological response, clothing insulation, activity and preferences in terms of air temperature and air movement also have a strong impact on thermal sensation.

In an uncomfortable hot ambient, owing to the thermoregulation system, the human body sets off the process of vasodilatation and sweating, trying to prevent the rise of internal body temperature. Since these processes can cool down the body only to a certain degree, in order to prevent further rise in body temperature and to avoid the risk of hyperthermia, it is necessary to make the space comfortable or to enhance the process of releasing the heat from the body. Both of these methods are used in vehicles, the latter being more suitable for transient (cool-down) conditions.

The heat loss from the body surface in warm conditions relies on convective heat loss and the heat loss by sweat evaporation from the skin surface (Fanger, 1970; Parsons, 2003). The equation for the dry heat flux by convection can be written as:

$$C = f_{cl} \cdot h_c \cdot \left(t_{cl} - t_a\right), \; W / m^2 \tag{13}$$

Factor f_{cl} is the clothing area factor, which describes the ratio between the area of the clothed body and the area of the nude body. The temperature of the clothing surface (t_{cl}) will be equal to the skin temperature (t_{sk}) on body surfaces without clothing: face, forearms, hands and neck.

According to the equation (13), the higher the temperature difference between the air and the body surface, the higher the heat loss from the body. In addition, the coefficient of heat transfer by convection (h_c) is dependent on air velocity and the shape of a body part. The coefficient increases with the increase of air velocity (Fanger, 1970; De Dear et al., 1997; Parsons, 2003). Therefore, the ventilation system is able to influence this part of the heat transfer directly, by changing the airflow rate through the ventilation outlets (air vents). The general form of the equation for forced and natural convective heat transfer coefficient for the whole body is (Fanger, 1970; De Dear et al., 1997):

$$h_c = B_f \cdot v_a^n, \text{ for forced convection, and} \tag{14}$$

$$h_c = B_n \cdot v_a^m, \text{ for natural convection.} \tag{15}$$

The exponents *n* and *m* and the constants B_f and B_n are mostly dependent on body posture as found in various literary sources. The overview of the equations for the calculations of the heat transfer coefficient can be found in Fanger (1970), ASHRAE (1997b), De Dear et al. (1997) and Parsons (2003).

Evaporative heat loss from the skin is dependent on air humidity, skin wetness and evaporative heat transfer coefficient (h_e). The evaporative heat transfer coefficient is calculated using the Lewis relation (Parsons, 2003):

$$h_e = LR \cdot h_c, \; W / m^2 kPa \tag{16}$$

Lewis ratio (LR = 16.5 K/kPa) makes the evaporative heat transfer coefficient dependent on air velocity as well. In addition, air-conditioning lowers the relative humidity of the interior air and consequently improves the latent heat loss by the evaporation of sweat.

The body's overall thermal sensation is affected by the local thermal state of individual body parts, at the same level of cooling, due to different physiological properties (different sensibility for warm versus cool sensations, different sweating rate etc.; Zhang, 2003):

- Back, chest, and pelvis strongly influence overall thermal sensation, which closely follows the local sensation of these parts during local cooling;
- The head region, arms and legs have an intermediate influence on the body's overall thermal sensation;
- Hands and feet have much less impact on the overall sensation.

4.2. Local air velocities around a human body

The working principle of the vehicle air-conditioning and the ventilation system is based on driving the conditioned air through adjustable air vents in the vehicle cabin. This airflow causes changes in local and overall microclimate conditions, consequently also changing the heat loss from the operator's body. This airflow is characterized by spatially distributed local air velocities and air temperatures.

Local velocity of the airflow should have the ability to penetrate the operator's free convection flow. For example, Melikov (2004) recommends local air velocities around 0.3 m/s as the minimum values. ASHRAE 55-2009 standard suggests elevated air velocities up to 1.2 m/s, but only if the air speed is under the control of an exposed person and in the temperature range of 28 - 31°C. These values apply to a lightly clothed sedentary person (0.5 - 0.7 Clo), with the metabolic rate in the range between 1.0 and 1.3 Met. In the case of higher difference between the temperatures of ambient air and (colder) local airflow, the operator will prefer lower air velocities. On the other hand, higher metabolic activity, up to 3.2 Met for the operation of mobile machinery, could allow higher air velocities (ASHRAE, 2003; Melikov, 2004; Arens et al., 2009).

Other very important parameters for thermal sensation and the sensation of air quality are characteristics of the air in the operator's breathing zone. The breathing zone is a semi-spherical space around the mouth and the nose. In terms of perceived air quality, still air in the breathing zone is not acceptable in a warm ambient, even when overall comfort is achieved. The temperature of the localized air in the breathing zone should be equal or 3 - 4°C lower than the interior air temperature. In order to achieve the mentioned requirements, it is recommended to place air vents at distances from 0.4 to 0.6 m from the operator's breathing zone (Melikov, 2004). This should be done in agreement with the geometry of the interior space of the vehicle cabin.

4.3. Effects of thermal radiation and transmitted solar radiation

When a tractor cab is exposed to the sun, the operator's body receives the heat partly by solar radiation transmitted through the glass and partly by longwave thermal radiation from the surrounding surfaces. The amount of solar radiation energy that will be absorbed by the body will depend on the effective projected area and on the solar absorptivity of the body surface. The largest effective projected radiation area A_{eff} of a person in the sitting position is with the azimuth and altitude angles of 30° and 15° respectively, and the total surface is equal to (Fanger, 1970; Parsons, 2003):

$$A_{eff} = 0.72 \cdot 0.33 A_{Du} = 0.238 A_{Du}, \ m^2 \qquad (17)$$

The total surface of the human body (A_{Du}) can be calculated from the body weight and height (after Dubois; Fanger, 1970; Parsons, 2003). The factor 0.72 in equation (17) is the effective radiation area factor, and the factor 0.33 is the projected area factor (Fanger, 1970; Parsons, 2003). Solar absorptivity of the human skin is around 0.62, while solar absorptivity

for clothing depends on the colour (Incropera and DeWitt, 1981; Parsons, 2003). Tractor cab design and the operator's position on the seat do not offer good protection from solar radiation, although large cabs create better shading for the operator. As it can be seen in Fig. 10, the head and the chest are exposed and protected only by solar properties of glass, unless some kind of solar shading devices are used. Shading devices can be placed on the inner side, or as a better solution, on the outer side of the cab. In both cases, the shading devices must not restrict the operator's normal field of vision in the working area of the tractor. For that reason, sun visors or curtains must be easily adjustable. In addition to shading the operator, the outer shading devices prevent exposure of glass and other surfaces to solar radiation. A common solution for these purposes is the use of cab roof overhangs, since aerodynamics is not an issue in agricultural tractors.
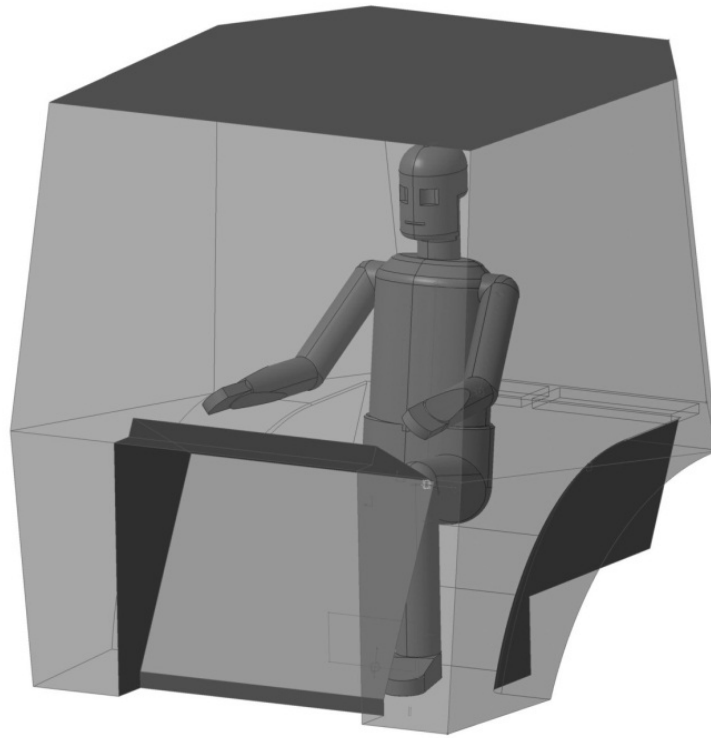

Figure 10. Projection of the operator inside the tractor cab from the direction of the azimuth and altitude angles of 30° and 15°, respectively

When surrounded by surfaces with higher temperatures than that of their skin and clothing, a person receives the heat by thermal radiation. Thermal radiation flux from hot surrounding surfaces depends on the wall surface temperature and the emissivity of the wall. The wall temperature and its emissivity are expressed by mean radiant temperature (t_{mr}) and linear radiative heat transfer coefficient (h_r):

$$q_r = h_r \cdot f_{cl} \cdot (t_{mr} - t_{cl}), \ W/m^2 \tag{18}$$

$$h_r = 2.88 \cdot \varepsilon \cdot \sigma \left(273.2 + \frac{t_{cl} + t_{mr}}{2} \right)^3, \ W/m^2 K \tag{19}$$

The emissivity of non-metallic surfaces is generally high, 0.9 and higher, just like the emissivity of the human skin (0.95; Incropera and DeWitt, 1981; ASHRAE 1997c) and clothing (0.77 for cotton; Siegel & Howell, 1992). For example, under the thermal conditions mentioned above, in the case of a single-layer metal sheet roof, the thermal radiation load to the operator's head would be around 70 W/m². In the case of having the polyester roof with internal thermal insulation, the thermal load would be less than 30 W/m².

The AC system is not able to directly change mean radiant temperature, except by decreasing the temperature of the inner surfaces (although very slowly).

5. Design of the air distribution system

The ventilation system drives and distributes the air inside the tractor cab using the air distribution system. The air distribution system consists of a blower, air ducts and several air vents usually positioned on the cab ceiling. The tasks of the ventilation system are:

- to remove excess heat from the operator's body,
- to supply the breathing zone with fresh air,
- to reduce the temperature inside the cab, and
- to pressurize the cab.

The air supply rate of tractor cab ventilation systems are generally in the range of several hundreds of cubic meters of air per hour, and are sufficient for up to 10 kW of cooling power.

According to the air jet theory, the air velocity along the airflow centreline is in direct proportion with the outlet velocity and in inverse proportion with the distance from the air vent. Consequently, the air jet direction, the distance from the body and the velocity profile on the outlet of the air vent are the main factors that influence the characteristics of the airflow over the body surface. These factors are determined by the interior geometry of the tractor cab, the operator's position in relation to the air vents and by the design of the ventilation ducts and vents (cross-section, number of vents), as well.

Control of the microenvironment is necessary due to individual differences among various operators, and airflow characteristics must be suited to individual human thermal sensation. For example, a powerful air-conditioner that easily decreases the temperature inside the tractor cab could produce an unpleasant stream of air, a draught. On the other hand, despite sufficient cooling power of the AC system in some cases, there is the possibility that a part of the operator's body would not be cooled satisfactorily, especially if exposed to solar radiation.

5.1. Evaluation of the air distribution system's efficiency and thermal sensation

The evaluation of thermal sensation in a tractor cab can be done by using human subjects, by directly measuring the microclimate physical quantities at discrete points in the cab or by using special human-shaped sensors. Complex human-shaped measuring instruments, the so-called thermal manikins, are the most suitable tool for a reliable, repetitive and objective evaluation of thermal conditions in a non-homogenous environment. Numerical methods

for the research of vehicle cabin microclimate are also used. When CFD (Computational Fluid Dynamic) methods for the modelling of the *human-cabin-environment* system are to be used, it is necessary to have a model of the human body that is geometrically and thermally appropriate, the so-called computer simulated person (CSP). Fig. 11.

The analysis of different tractor cab ventilation layouts was described in Ružić et al. (2011) and Ružić (2012). In this research, the heat losses from body parts were observed on a virtual thermal manikin using the CFD simulation. All the simulations were performed without solar radiation. The airflow through the ventilation system was approximately 380 m³/h, with the air temperature of 20°C. In one group of simulations, the conditions were isothermal (Ružić et al., 2011), while in the other (Ružić, 2012) the initial conditions were the same as the hot ambient with the cab temperature of 30°C.
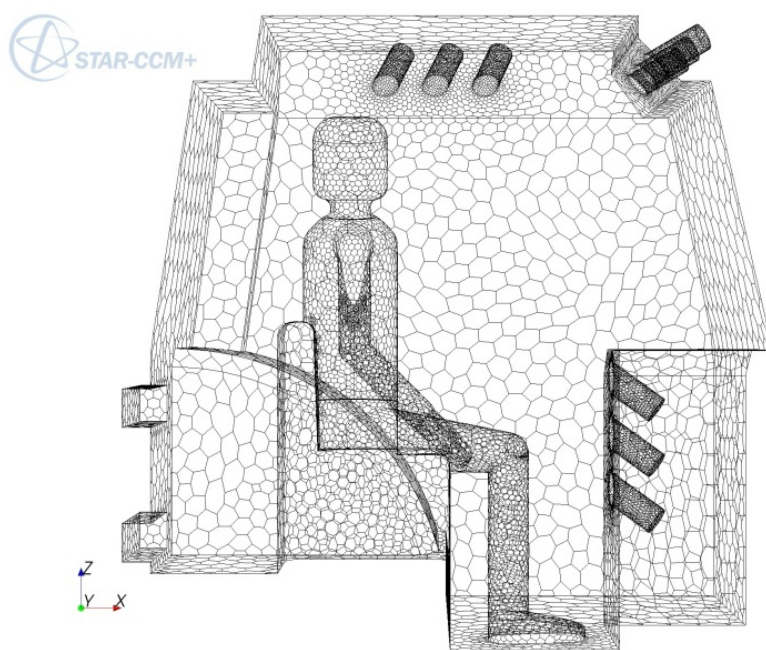

Figure 11. An example of a model of an agricultural tractor cab with a virtual thermal manikin, divided into the volume mesh for the CFD simulation

The results showed that the heat loss from the operator's body can vary significantly depending on the position and the direction of the vents, but all the simulations were done with the same cooling power. Consequently, the operator's thermal sensation will vary accordingly. As it can be seen in Fig. 12, under the same tractor cab thermal load and the same heat removal, but with two different air distribution designs, in the example *A* the total heat loss from the operator is 42% lower than in the example *B*. This indicates that human body characteristics and air distribution systems are two very important factors as well.

In terms of heat loss for the whole body and for thermally most sensitive body segments, the statistical analysis of results for various combinations of vents positioning and airflow direction shows that airflow direction is a more significant parameter than positioning of the vents. Therefore, it is important that the operator be able to change the characteristics of air jet, that is, the air flow direction and the velocity.
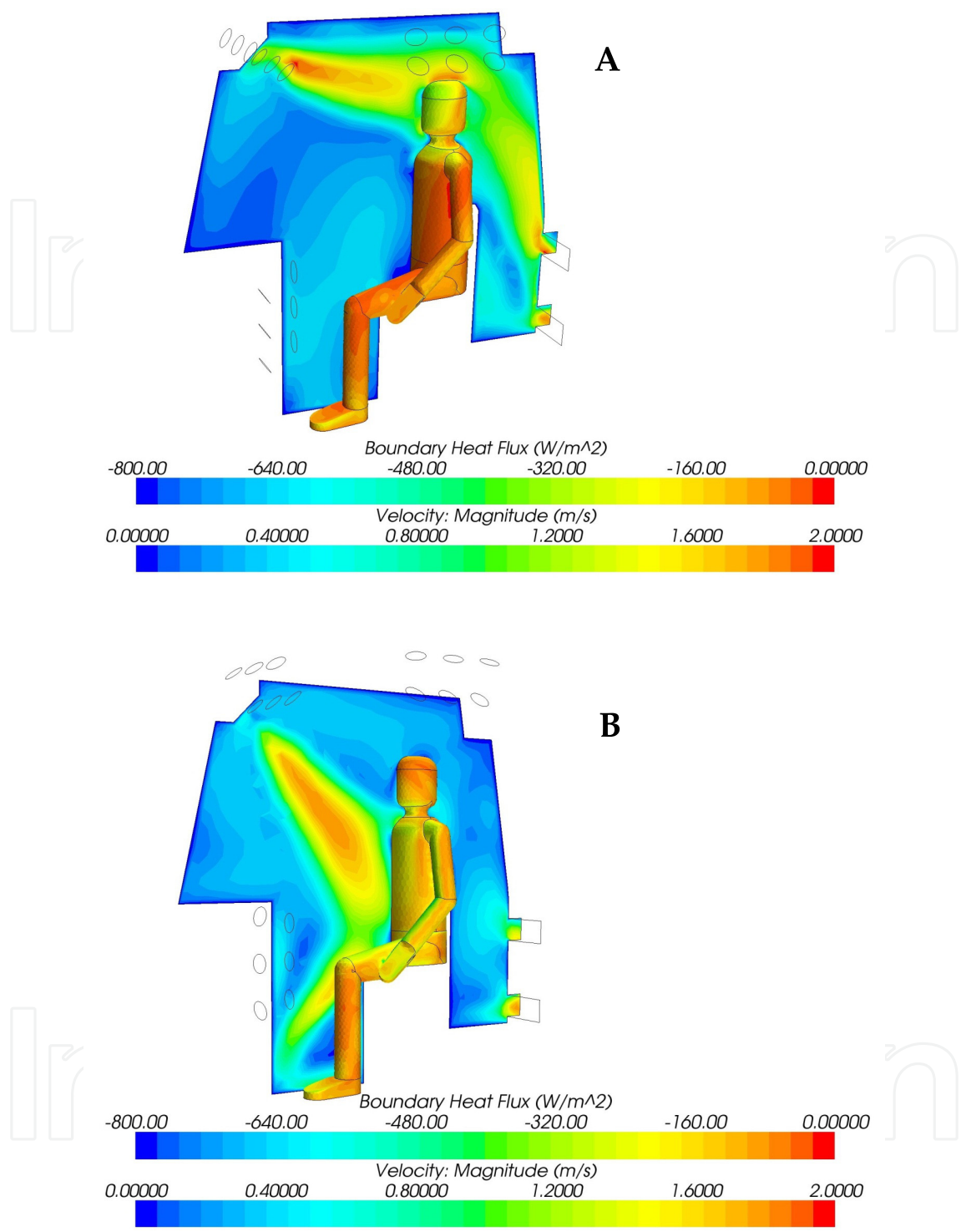
Figure 12. Comparison of air velocity distribution and heat loss from different parts of the operator's body using the CFD technique with virtual thermal manikin (Ružić, 2012)

In the non-isothermal cases (Ružić, 2012), the assessment criterion was the deviation from neutral values of equivalent temperatures. For example, under these conditions, the lowest deviation of the equivalent temperature for the whole body was achieved with the vents placed in the front part of the ceiling, directed at the operator's chest. The highest rate of

heat loss from the operator's body was achieved in the layout with the vents placed in the instrument panel, directed at the operator's pelvis (H-point). This mode of operation would be of interest for cooling down of the operator under the conditions of high thermal load, for example caused by solar radiation. The difference between the best and the worst cases regarding equivalent temperature for the whole body was 10 degrees. For individual body segments, the difference was considerably larger. Consequently, if the operator sets the air distribution system in a wrong way, unwanted thermal conditions can be produced. For that reason, the interface for the AC and the ventilation system control must be designed in such a way that it is easy to use.

More information about the significance of certain factors could be obtained by expanding the range of factor variations. However, there is almost an unlimited number of different settings of just one air distribution design, as well as a variety of ambient conditions. Fortunately, the cabs of modern middle-range agricultural tractors are all similar in design and shape, and most of these cabs have the AC air distribution system with the vents on the ceiling. According to the results of those simulations, it is possible to improve and optimize thermal conditions in tractor cabs by implementing conventional air distribution systems.

6. Conclusions

Although the cabs of modern tractors have many common features, the results show that there are significant differences regarding thermal loads, caused by variations in tractor cab designs. Conclusions can be summarized as follows:

- The highest heat flux that enters the cab is caused by solar radiation through the glass (which is several times higher than the heat transferred by other modes). Paying attention to solar characteristics of glass is a direct way to reduce the operator's thermal load.
- Further improvements in the reduction of the heat load are the use of roof overhangs, adjustable sun visors and less inclined glass. A thermally reflective heat shield placed beneath the cab floor also improves thermal conditions. However, some of the existing technologies for thermal load reduction are not used to the full extent, especially in low-class tractors.
- The ventilation system and the AC system should be designed in such a way that the optimum heat loss from individual body parts can be obtained in most conditions. The direction of the air jet is one of the most crucial factors regarding heat losses from the body. Due to a wide range of boundary conditions and different individual preferences regarding thermal conditions, the air distribution system must be adequately adjustable and easy to use.
- For the analysis and optimization of the air conditioning system, experimentally verified numerical methods (CFD) with computer simulated person or virtual thermal manikin are an inevitable tool.

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Nomenclature

A – area, m^2
f_{cl} – clothing factor, (-)
G_s –solar irradiation, W/m^2
h – heat transfer coefficient, W/m^{2}K
H – enthalpy, J
k – thermal conductivity, W/mK
LR – Lewis ratio (16.5 K/kPa)
$\dot{m}$ – mass flow, kg/s
q – heat flux, W/m^2
Q – heat transfer, W
RH – relative humidity, %
t, T – temperature, °C, K, respectively
U – total heat transfer coefficient, W/m^{2}K
v – air velocity
w – humidity ratio, kg/kg

Greek symbols

α – surface absorptivity, (-)
δ – layer/wall/glass thickness, m
ε – surface emissivity, (-)
σ – Stefan-Boltzmann constant (5.670·10^{-8} W/m^{2}K^4)
θ – incident angle of solar radiation, degrees
τ – normal solar transmissivity, (-)

Subscripts

a – air
AC – air-conditioning
c – convection

cl – clothing
e – evaporation
eff – effective
G – cab glazing
H – human
HVAC – heating, ventilation, air-conditioning
i – inside
k – conduction
M – metabolic activity
mr – mean radiant
o – outside
PT – powertrain
r – thermal radiation
refl - reflection
s – solar
surf – surface

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