Airflow and thermal management in farm tractor cab

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Abstract. In order to provide fresh air for the operator and proper pressurization against ingress of contaminants and dust, farm tractor cab must have ventilation system that forces filtered outside air into the cab. In the hot conditions, outside air is passed through the evaporator where the heat and moisture must be removed, which can be between 40 and 70% of total heat load. Consequently, there are several contradictory demands on the tractor cab’s air-conditioning system. It is necessary to obtain good air quality and comfort environment keeping the power consumption as low as possible. In the same time, the air-conditioning system should be neither too complex nor too large. In this paper, qualitative and quantitative analysis of tractor cab’s air-conditioning system were carried out. The results are compared in terms of power consumption and air quality and guidelines for efficient cab air-conditioning system design are given.

1. Introduction

An air-conditioner (AC) of motor vehicle cabin is the largest power consumer, excluding the vehicle propulsion system [1-3]. The same applies to farm tractors and mobile machinery. The power consumption of the AC in small tractors can reach 10-100% of the actual engine power and the highest values occur during the engine idle [3]. Measures that can be taken in order to reduce power consumption are increase of the refrigeration system efficiency, reduction of the cab heat gain and efficient use of the air-conditioned air [1], [3, 4].

Apart from experimental researches of air-conditioner performances, many numerical and analytical methods are applicable. The purpose of the researches is manly to investigate influences of design parameters and operating modes on performance of air-conditioning system. The goal is to optimize energy consumption and to obtain thermal comfort for occupants. The air quality is also important factor in assessment of conditions inside the cabin. Amount of fresh air that is entering the cabin influences both the air quality and energy consumption. In the experimental work of Grady at al. [5] an option of partial air recirculation in passenger car is analysed. The results showed that with more than 85% of recirculation, acceptable concentration of cabin CO₂ could be obtained for two passengers at various driving conditions and fan speeds.

The air-conditioned vehicle cabin can be analyzed as an integrated system (an air-conditioner and a cabin with or without occupants), or each subsystem can be modelled separately. In analysis of possibilities of reducing vehicle auxiliary loads, in the project carried out by NREL, equation-based modelling was also used for modelling of thermal processes in air-conditioned vehicle cabin [2]. All the components from the HVAC system and the thermal envelope are broken down into submodels. Each submodel is described by one or more equations. In combination with experimental data, fast and
reliable simulation model was obtained. More detailed generic model of entire system is developed in work of Arici et al. [6], using the principles of thermodynamics. The methodology was validated by comparing the simulation results with the experimental results. The model allows testing the effects of design, component, and structural changes on the thermal conditions of the passenger compartment.

Fayazbakhsh and Bahrami developed the heat balance method for estimating the heating and cooling loads encountered in a vehicle cabin [7]. Using a lumped-body approach for the cabin, their model is capable to estimate the thermal loads for the simulation period in real-time. In the paper of Kamar et al. [8], results of a parametric study performed on a passenger car air-conditioning system in a semi-empirical computer simulation program CARSIM are presented. The effects of the supply air volumetric flow rate, number of occupants, vehicle speed, and the fractional ventilation air intake, on the dry-bulb temperature and relative humidity of the air inside the cabin, and on the evaporator cooling load were assessed.

Modelling of an air-conditioner using general freeware simulation tool CoolPack is presented in work of Ružić and Sovilj [9]. The results obtained in the simulation tool have good matching with empirical values, making this software suitable for fast theoretical analysis of automotive air-conditioning system as well as for educational purpose.

This paper deals with qualitative and quantitative analysis of different air-conditioner operating modes in cab of farm tractor. The results are compared in terms of AC power consumption and air quality inside the cab. The method used in this research is analytical method of modelling thermodynamic processes in an air conditioned cab. The problem is treated as a lumped system model. The reason is the fact that although temperature and velocity fields in the cab volume are not homogenous, strong mixing of the air caused by ventilation creates very uniform distribution of air temperature field, which is evident during the cool-down process (Figure 1). Equations of heat and mass transfer and perfect gas relationships for moist air are solved in MS Excel in the form of some kind of calculator. The results are moist air states at various points of the air-conditioned cab model as well as other system parameters, depending on chosen input variables. The calculator offers a fast way for analysis of a heat exchange at the evaporator in dependency of air mass flow, cab characteristics, ambient and interior conditions and recirculation rate.

![Figure 1](attachment:image.png)

**Figure 1.** Typical changes of air temperatures inside the air-conditioned cabin of passenger car during the cool down. The hot soak was performed by exposure to the solar radiation at ambient air temperature of 28°C [10].
2. Operation modes of an air-conditioner

A typical farm tractor cab is equipped with an air distribution system whose air vents (outlets of ventilation system) are placed on the cabin roof. An operator can change the air velocity, the air temperature and direction of the air jet. Vapour compression system is used for AC, the same as in automotive applications.

The operation modes of AC that were considered and analysed in this research are listed in the Table 1. The table contains qualitative estimation of thermal conditions inside the cab, air quality and pressurization. It is known that 100% recirculation is the most efficient setting in terms of thermal conditions and particle concentration, but this causes elevated CO\textsubscript{2} level in the cab and no pressurization.

In this research, an option of air recirculation is added in the system, although the air recirculation is not usually an option for small cabs. Air recirculation is typical solution for reduction of energy consumption in cabin of passenger vehicles, as well as for prevention of ingress of outside contaminants. However, the pressurization of tractor cab is demanded according to the regulation ASABE/ISO 14269 [11]. The cab should be capable of maintaining a minimum pressurization of 50 Pa and maximum pressurization shall not exceed 200 Pa [11]. The pressurization is possible only by ventilation with outside air and it is required that under all conditions of air-conditioning, heating or ventilation, a minimum of 43 m\textsuperscript{3}/h of filtered fresh air shall be provided [11]. Tractor cab ventilation systems generally have higher capacities, around 300 m\textsuperscript{3}/h and higher [9].

### Table 1. The AC modes of operation and estimated influence on air quality.

| Airflow mode | AC | Thermal conditions | Air quality | Pressurization | Comment |
|--------------|----|--------------------|-------------|----------------|---------|
| recirculation | REC = 100% | no heat gain and thermal conditions outside comfort and safety limits | ingress of contaminants low, interior contaminants, moisture and CO\textsubscript{2} high | 0 | not applicable |
| recirculation | REC = 100% | yes possible to achieve thermal comfort | ingress of contaminants low, interior contaminants, and CO\textsubscript{2} high | 0 | allowed for short time, option with filtration of recirculated air |
| outside air | REC = 0% | no possible to achieve thermal comfort under some conditions, occurrence of high local air velocities | ingress of contaminants possible, interior contaminants, moisture and CO\textsubscript{2} low | normal | conventional system, filtration of outside air is mandatory |
| outside air | REC = 0% | yes possible to achieve thermal comfort | ingress of contaminants possible, interior contaminants, moisture and CO\textsubscript{2} low | normal | conventional system, filtration of outside air is mandatory |

3. Model of the air-conditioned tractor cab

The model corresponds to the model of air-conditioned room with external heat gain and an internal heat and moisture source, i.e. the operator (Figure 2). The air inside the cab is assumed to be well mixed thanks to high air velocities produced by ventilation system, allowing the modelling of the cab as a lumped body. The governing equations are conventional equations of heat and mass transfer applied on an air-conditioned cab, as well as processes in moist air.

Boundary conditions in a tractor cab are in close relation to the outside thermal conditions (air temperature, air velocity, intensity and direction of solar radiation), but more or less independent from tractor's working operations. The sum of heat gains for a closed tractor cab in a hot environment, under the steady-state conditions consist of (Figure 2) [3],[12]:

- heat transfer between the environment and interior air due to the temperature difference $Q_{\text{amb}}$,
- heat gain caused by solar radiation $Q_{\text{sol}}$.
heat introduced with outside air $Q_{\text{air}}$,
sensible and latent heat released by the operator $Q_{\text{H}}$.

Since only quasi steady-state processes are being analyzed, thermal mass of the cab materials is
neglected in the model. However, it is possible to extend the model and changes in interior air
temperature could be calculated.

Figure 2. Schematic presentation of air-conditioned cab with an operator.

3.1. Heat transfer through the cab walls
The heat transfer between the ambient and the interior air due to the air temperature difference can be
calculated using:

$$Q_{\text{amb}} = U \cdot A \left( t_{\text{out}} - t_{\text{ocab}} \right) \quad (1)$$

where
- $U$ - total heat transfer coefficient of cab walls, W/m$^2$K
- $A$ - surface area of cab walls, m$^2$
- $t_{\text{out}}$ - outside air temperature, °C
- $t_{\text{ocab}}$ - inside air temperature, °C

3.2. Heat gain due to air infiltration
Due to difference between the enthalpy of outside and inside air, there is a (positive or negative) heat
transfer caused by the air infiltration:

$$Q_{\text{air}} = m_{\text{out}} \left( h_{\text{out}} - h_{\text{ocab}} \right) \quad (2)$$

where
- $m_{\text{out}}$ - outside air mass flow, kg/s
- $h_{\text{out}}$ - outside air enthalpy, kJ/kg
- $h_{\text{ocab}}$ - inside air enthalpy, kJ/kg

General expression for moist air enthalpy in dependence of air temperature $t$, °C, and humidity
ratio $w$, kg/kg, is [16]:

$$h = 1.006 t + w \left( 2501 + 1.805 t \right) \quad (3)$$

3.3. Heat gain caused by solar radiation
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. The total amount of the heat transmitted through the glass caused by
solar radiation depends on glass solar properties and it is related to the normal projection of the tractor
cab in the direction of radiation.

The absorbed part of the solar irradiation heats the cab’s outer opaque surfaces (the roof, for
example). The maximum solar irradiation on a horizontal outer surface in the central Europe region on
a summer day may exceed 900 W/m$^2$ [3].
3.4. The heat release from the operator

The heat release from the operator is calculated using the equations based on human thermal balance and human physiology [13-15]:

\[ Q_H = Q_M + Q_{WH} \]  

where

\[ Q_M \] - sensible heat release:

\[ Q_M = M \cdot A_{Du} \]  

where

\( M \) - the specific metabolic activity, for the tractor driver it is in range from 85 to 110 W/m² [15]

\( A_{Du} \) - outer surface area of the human body:

\[ A_{Du} = 0.202 \cdot m_b^{0.425} \cdot h_b^{0.225} \]  

Variables \( m_b \) and \( h_b \) are body mass in kg and body height in m, respectively.

\( Q_{WH} \) - latent heat release, kJ/kg, depends on skin temperature \( t_{sk} \), metabolic activity level \( M \) and partial vapour pressure \( p_w \) [13]:

\[ Q_{WH} = A_{Du} \cdot (3.05 \cdot 10^3 \cdot 256 \cdot t_{sk} - 3373 - p_w) + 0.42 \cdot (M - 58.15) \]  

Average skin temperature \( t_{sk} \) under the comfort conditions depends upon activity level [13]:

\[ t_{sk} = 35.7 - 0.0275 \cdot M \]  

Moisture released by the operator can be calculated using:

\[ m_{wH} = \frac{Q_{WH} \cdot A_{Du}}{h_{wH}} \]  

where

\( h_{wH} \) - enthalpy of saturated water vapour at skin temperature, kJ/kg.

3.5. Model of the AC system

The problem of the cab air conditioning is determined by the quantity of air to be supplied and the air condition necessary to remove given amounts of energy from the cab at a specified condition of interior air. The equations of heat and mass balance in the cab are:

\[ m_a \cdot h_2 + Q_{ot} = m_a \cdot h_{cab} \]  

\[ m_a \cdot w_2 + m_{wH} = m_a \cdot w_{cab} \]  

where

\( m_a \) - total air mass flow through the evaporator, kg/s, equation (15),

\( Q_{ot} \) - sum of heat gains, kW:

\[ Q_{ot} = Q_{amb} + Q_{sol} + Q_H \]  

\( h_2 \) - enthalpy of the moist air leaving the evaporator, kJ/kg,

\( w_2 \) - humidity ratio of the air leaving the evaporator, kg/kg,

\( w_{cab} \) - humidity ratio of the inside air, kg/kg.

In order to achieve steady-state thermal conditions with specified condition of interior air at given state of ambient, total air mass flow \( m_a \) and evaporator cooling load \( Q_{evap} \) can be calculated:

\[ m_a \cdot h_1 = m_a \cdot h_2 + Q_{evap} + m_w \cdot h_{w2} \]  

\[ m_a \cdot w_1 = m_a \cdot w_2 + m_w \]  

where

\( m_w \) - moisture removal at the evaporator, kg/s.

If an air recirculation takes place with air mass flow \( m_{rec} \), resulting properties of the mixture of the outside air and the interior air can be calculated using following relations:

\[ m_a = m_{sat} + m_{rec} \]
\[ m_{\text{out}} \cdot h_{\text{out}} + m_{\text{rec}} \cdot h_{\text{cab}} = m_{a} \cdot h_{1} \]  \hspace{1cm} (16) \\
\[ m_{\text{out}} \cdot w_{\text{out}} + m_{\text{rec}} \cdot w_{\text{cab}} = m_{a} \cdot w_{1} \]  \hspace{1cm} (17)

A recirculation rate \( REC \) can be in range from 0 to 100%:

\[ REC = \frac{m_{\text{rec}}}{m_{a}} \cdot \% \]  \hspace{1cm} (18)

4. Calculation of air-conditioner operating parameters

In order to solve the equations listed in the previous sections, equations of heat and mass transfer and perfect gas relationships for moist air are written in MS Excel. The worksheets present some kind of calculator able to determine moist air states at various points of the air-conditioned cab model and other system parameters. The calculator offers a fast way for analysis of a heat exchange at the evaporator in dependency of air mass flow, cab characteristics, outer and interior conditions and recirculation rate. Other relations are also possible, depending on known input parameters.

The boundary conditions and input values for the calculation are based on a procedure for testing of AC performance according to standard ASABE/ISO [11]:

- the ambient air temperature \( t_{\text{aout}} = 32°C \), relative humidity \( RH_{\text{out}} = 54% \) (wet bulb temperature: 25°C);
- solar heating is estimated: \( Q_{\text{sol}} = 0.50 \) kW [3];
- the cab properties: \( U = 10 \) W/m²K, \( A = 12 \) m² [2], [3];
- sensible heat release from the operator \( Q_{H} = 200 \) W (metabolic activity \( M = 100 \) W/m², outer body surface area \( A_{Du} = 2.0 \) m²);
- set interior steady-state conditions: air temperature \( t_{\text{acab}} = 27°C \), \( RH_{\text{cab}} = 50% \) [11].

The limitation of the system are maximum cooling power \( Q_{\text{evap}} = 5 \) kW, maximum total air volume flow rate \( m_{a} = 320 \) m³/h, and minimum air temperature after the evaporator: \( t_{a2} = 6°C \), as values that correspond to typical small farm tractor AC [8].

Using the highest values of AC cooling power and air mass flow, the potentials of cooling down the cab are estimated (Table 2, case 1). The cooling potential is expressed through difference between the available cooling power \( Q_{\text{evap}} \) and total heat gain \( Q_{\text{tot}} \). The higher the difference, the faster will the air temperature drop, under the same other conditions.

The initial interior air temperature at the beginning of the cooling down the heated cab is calculated on the basis of thermal balance between the heat gain and heat loss of the cab heated by the solar radiation (i.e. \( Q_{\text{tot}} = 0 \)), with the AC turned off:

\[ Q_{\text{amb}} = Q_{\text{sol}} + Q_{H} + Q_{\text{air}} \]  \hspace{1cm} (19)

Initial interior air temperature will be:

\[ t_{\text{acab}} = t_{\text{aout}} + \frac{(Q_{\text{sol}} + Q_{H} + Q_{\text{air}}) \cdot (U \cdot A)}{(U \cdot A)} \]  \hspace{1cm} (20)

Iterative calculation must be performed in order to determine the heat release from the operator's body as well as heat exchange by infiltration, both dependent of the interior air state. Humidity ratio \( w_{\text{cab}} \) is assumed to be the equal to the humidity ratio of outside air \( w_{\text{out}} \).

The next case is condition of final state inside the cab when the set temperature and relative humidity are achieved. Under the steady-state thermal conditions, the cooling power \( Q_{\text{evap}} \) is equal to the total heat gain \( (Q_{\text{evap}} = Q_{\text{tot}}) \). Under the given conditions, the cooling power in this model is controlled by the air mass flow through the evaporator \( m_{a} \). A required air mass flow through the evaporator is calculated as:

\[ m_{a} = m_{\text{out}} + m_{\text{rec}} = \frac{Q_{\text{tot}}}{(h_{1} - h_{2} - h_{w2} (w_{1} - w_{2}))} \]  \hspace{1cm} (21)

Again, \( Q_{\text{tot}}, h_{1} \) and \( h_{2} \) are dependent on recirculation ratio and iterative calculation is necessary.

The cases with different values of \( REC \) are considered (cases 3 to 5). Case 5 is the condition with minimum required airflow rate of 43 m³/h of filtered fresh air.
Table 2. Characteristic thermal states of the cab and AC operating modes.

| Case | \( t_{\text{acab}} \) \(^\circ\text{C} \) | RH\( _{\text{cab}} \) | REC | \( m_{\text{air}} \) kg/s (m\(^3\)/h) | \( m_{\text{cap}} \) kg/s (m\(^3\)/h) | \( Q_{\text{evap}} \) kW | \( Q_{\text{air}} \) kW |
|------|--------|-----|-----|-----------------|-----------------|-----------------|-----------------|
| 1    | 37.7   | 39  | 0   | 0.1000 (320)    | 0.1000 (320)    | 5.053           | -0.070          |
| 2    | 27     | 50  | 0   | 0.0396 (127)    | 0.0396 (127)    | 2.268           | 0.902           |
| 3    | 27     | 50  | 50  | 0.0396 (126)    | 0.0198 (63)     | 1.816           | 0.450           |
| 4    | 27     | 50  | 100 | 0.0395 (126)    | 0.00 (0)        | 1.366           | 0.00            |
| 5    | 27     | 50  | 66  | 0.0396 (126)    | 0.0135 (43)     | 1.672           | 0.306           |

4.1. Discussion of results

In the case 1, at the beginning of the cooling-down process, air infiltration contributes to heat removal (negative value of \( Q_{\text{air}} \)). It is recommended to keep the outside air mass flow at maximum but without too much draft and eyes irritation. Recommended maximum velocity in eyes zone is between 0.05 and 0.30 m/s with turbulence intensity up to 70\% [11], [14]. As soon as the enthalpy of interior air becomes lower than enthalpy of outside air, partial recirculation can be activated in order to decrease power consumption.

Partial or full recirculation results with reduction of required power under the same boundary conditions, as expected. However, there are limitation in recirculation ratio due to required pressurization and air quality (see Table 1). Since the operator’s breathing zone is within the reach of airflow from the vents, the ventilation system design should enable that only fresh cooled air is directed to the breathing zone. In this case, if maximum cooling power is used, heat gain caused by outside air would present around 10\% of total cooling power and proper local air quality could be obtained. The other part of total airflow, the recirculated air, has to be used for cooling the operator’s body and the cab, especially the windows and windshield. Cooling the glass surfaces will reduce thermal radiation to the operator. Under the steady-state conditions, cooling power is 20-25\% lower in the cases with partial recirculation (cases 3, 5) than in the case when only outside air is used (case 2).

In conventional automotive air-conditioning and ventilation system with recirculation ability, the outside air and recirculated air are mixed before the evaporator. In order to achieve above mentioned requirements of thermal comfort and air quality using the partial recirculation, the airflow must not be mixed before evaporator and it has to be separated after it and individually controlled. However, this will make the system more complex and consequently increased cost and available space must be considered.

5. Conclusions

The analytical model of air-conditioned farm tractor cab is presented in this paper. The problem is treated as a lumped system model due to assumed homogeneity of an air temperature field inside the cab. The result showed that a partial air recirculation with proper filtration would contribute to reduction of power consumption, keeping in the same time the breathing zone supplied with fresh cool air. In order to better utilize maximum cooling power of the AC, it is recommended to direct recirculated air to the cab’s interior surfaces and smaller proportion of fresh air should be directed to the operator’s breathing zone. In this example recommended fresh air mass flow is around one third of total air mass flow. However, the conventional design of ventilation system can not satisfy those requirements, because of mixing of the outside air and recirculated air before the evaporator. Proposal of AC system redesign must be evaluated in terms of complexity and costs as well.

The calculator presented in this paper is intended for use on steady-state model. Thermal mass of the cab and interior materials is not negligible and must be take into account if transient analyse should be carried out. Structure of the model made in Excel worksheet is suitable for upgrade to transient model. The calculator can be combined with refrigeration cycle model and a whole body thermal comfort model too.
It can be concluded that despite of many advanced (and expensive) software for simulation of 3D or 4D fluid dynamics and thermal processes, 1D and 0D models still have important role in research of air-conditioning processes. Equation-based modelling is suitable for fast solution and should present a first step before more detailed but time consuming finite element and finite volume methods.

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References
[1] Leighton D and Ruth J 2016 Increasing EDV Range through Intelligent Cabin Air Handling Strategies, Annual Progress Report, National Renewable Energy Laboratory USA
[2] Rugh J and Farrington R 2008 Vehicle Ancillary Load Reduction Project Close-Out Report, Technical Report NREL/TP-540-42454, National Renewable Energy Laboratory USA
[3] Ružić D and Časnji F 2012 Thermal Interaction between a Human Body and a Vehicle Cabin, In: Kazi S (Ed.) Heat Transfer Phenomena and Applications, pp. 295-318, InTech
[4] Ružić D and Časnji F 2011 Personalized Ventilation Concept in Mobile Machinery Cab, MVM – International Journal for Vehicle Mechanics, Engines and Transportation Systems 37(1) 9-22
[5] Grady M, Jung H, Kim Y C, Park J K and Lee B C, 2013 Vehicle Cabin Air Quality with Fractional Air Recirculation, SAE Paper 2013-01-1494
[6] Arici Ö, Yang S-L, Huang D and Öker E 1999 Computer Model for Automobile Climate Control System Simulation and Application, Applied Thermodynamics 2(2) 59-68
[7] Fayazbakhsh M A and Bahrami M 2013 Comprehensive Modeling of Vehicle Air Conditioning Loads using Heat Balance Method, SAE Paper 2013-01-1507
[8] Kamar H M, Kamsah N and Senawi M Y 2013 Computerized Simulation of Automotive Air-Conditioning System: A Parametric Study, International Journal of Computer Science Issues 10(1) 787-794
[9] Ružić D and Sovilj V 2017 Application of Refrigeration Simulation Tools in Study of Automotive Air-Conditioner, 18th International Symposium on Thermal Science and Engineering of Serbia SIMTERM, Sokobanja, Serbia, October 17-20, pp. 998-1004
[10] Ružić D 2006 Influence of Air-Conditioning on Thermal Comfort in an Passenger Car, University of Novi Sad, Faculty of Technical Sciences, MSc Thesis
[11] ASABE/ISO 14269-2, 1997, Tractors and Self-Propelled Machines for Agriculture and Forestry – Operator Enclosure Environment – Part 2: Heating, Ventilation and Air-conditioning Test Method and Performance
[12] Grossmann H 2010 Pkw-klimatisierung, Springer
[13] Parsons K 2003 Human Thermal Environments: The Effects of Hot, Moderate and Cold Environments on Human Health, Comfort and Performance, Taylor & Francis
[14] ISO 7730, 1994, Moderate Thermal Environment – Determination of the PMV and PPD indices and Specification of the Conditions for Thermal Comfort
[15] ISO 8996, 2004, Ergonomics of the thermal environment – Determination of Metabolic Rate
[16] ASHRAE, 1997, Fundamentals Handbook – Chapter 6: Psychometrics, American Society of Heating, Refrigerating and Air Conditioning Engineers