Heat Insulation Performance Analysis on the Vacuum Insulation Composite Compartment of Refrigerated Trucks

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Abstract: The influence of the composite insulation compartment on indoor thermal stability is presented by two introduced factors: thermal comfort degree-hour and response factor. The results show that: when the vacuum insulation material is arranged on the top of the refrigerator compartment, the heat insulation performance of the refrigerated compartment is the best, and the thermal comfort degree-hour and response factors were 2.687,15.554; vacuum separator material disposed in the carriage body insulating material intermediate position, followed by refrigerated compartment insulation properties, thermal comfort degree-hour and response factor were 2.752,15.931; vacuum separator material is disposed against the carriage body in the innermost layer of insulating material, the worst performance refrigerated compartment insulation, thermal comfort degree-hour and compartment response factor were 2.813,16.284.

Introduction

Refrigerated compartment insulation material is an important factor[1] to determine the temperature and the heat stability of the compartment, and usually adopts thermal resistance of the envelope structure or attenuation factor, and the delay time value to evaluate the performance of the envelope structure. However, most of the current vacuum insulated refrigerated compartment envelope is multi - layer composite structure, and various combinations of positions insulating layer, even though the thermal resistance, total thermal inertia and other indicators of the same, not the same as the effect of air inside the compartment. Through information inquiry found that different combinations of building envelope has a certain influence on the indoor air heat stability, even under the same circumstances the total thermal resistance, thermal comfort degree-hour and compartment response factor will be different[2-3], but no special arrangement for the different forms of refrigerated trucks vacuum heat insulating material multilayer composite envelope, the relevant literature on intracellular compartment temperature of the thermal stability of the report. For this reason, thermal comfort degree-hour and compartment response factor index I is introduced into the evaluation of the thermal insulation properties of the refrigerated truck, which can provide a theoretical basis for the quantitative evaluation of the structure of multi-layer composite structure.

Compartment outer temperature determination

Compartment outer wall temperature determination

Comprehensive car outside temperature, namely external thermal environment, including the sky scattering radiation, direct sun radiation, ground reflected radiation and outdoor air temperature four parameters, the outer wall of the compartment temperature is the temperature above four parameters equivalent heat effect compartment wall structure of the resulting integrated into a car outside wall temperature[1], namely

$$T_W = T_Z + \frac{pI}{a_w} - \frac{V\Delta R}{a_w} \tag{1}$$

Where: T_Z-- exterior temperature by time; p -- absorption coefficient, compartment is surrounded by the wall surface and the top surface of the glass steel, p = 0.2; I -- the hourly solar radiation intensity[1]; a_W -- carriage body surface convective heat transfer coefficient; ζ-- outer body longwave radiation; *ΔR* - car body surface and longwave radiation exchange volume between the sky and the surrounding objects.

Since the size of aw, z, and R is related to location of the vehicle, time, outside temperature conditions and other factors, the actual relationship quite is complex, so the actual approximate calculation is generally treated as follows:

$$a_w = 9 + 3.5v^{0.66} \tag{2}$$

Where: v-- vehicle speed.

For the vertical, $\frac{VDR}{a_W}$ is 0 ℃; namely that the long-wave radiation longwave radiation received from the vertical plane of the outer body near the ground is equal to the loss; the horizontal plane, $\frac{VDR}{a_W}$ is 3.5 ~ 4 ℃, namely that the level of the received longwave radiation is slightly larger than the longwave radiation scattered outside, this value is generally measured by the experience.

Compartment outer wall temperature amplitude function

Using the Fourier series expansion, can take different compartment walls' temperature as the disturbance, expressed as a set of simple periodic function, namely:

$$T_W(t) = T_{W,p} + \sum_{n=1}^{\infty} A_n \sin(nwt + j_n) \tag{3}$$

Where: $T_W(\tau)$ - compartment wall temperature by time when in vitro; $T_{W,p}$ -- compartment wall temperature in vitro; n-- order harmonics, n = 1,2, ...,; A_n -- order sinusoidal external disturbance amplitude; τ-- time variable; φ_n -- order sinusoidal external disturbance initial phase; $n\omega$-- order sinusoidal external disturbance frequency.

Simply change the formula (3) and let $\frac{a_o}{2} = T_{W,p}$, $a_n = A_n \sin j_n$, $b_n = A_n \cos j_n$, can be obtained:

$$T_W(t) = \frac{a_o}{2} + \sum_{n=1}^{\infty} [a_n \cos(nwt) + b_n \sin(nwt)] \tag{4}$$

among them: $A_n = \sqrt{a_n^2 + b_n^2}$

$$j_n = \begin{cases} \arctan(\frac{a_n}{b_n}) & a_n > 0, b_n > 0 \\ \arctan(\frac{a_n}{b_n}) + 2p & a_n < 0, b_n > 0 \\ \arctan(\frac{a_n}{b_n}) + p & b_n < 0 \\ \frac{p}{2} & a_n = A_n, b_n = 0 \\ \frac{3p}{2} & a_n = -A_n, b_n = 0 \end{cases}$$

Equation (4) is TW (τ) function of Fourier expansion.

Reuse a series of discrete points spaced approximated limited to items Fourier series and harmonic analysis of the function. The fundamental period [0, T] into N aliquots, in this interval of N discrete point value $T_{Wj}=T_W(j\Delta\tau)$ (j = 0,1,2 ......, N-1), where $\Delta\tau = T / N$. For n-order harmonic amplitude and the initial phase, the use of alternative methods rectangle superimposed summation integral method to calculate, namely:

$$\begin{cases} \frac{a_0}{2} = T_{W,p} = \frac{1}{T}\sum_{j=0}^{N-1} T_{Wj}\,\Delta t \\ a_n = \frac{2}{T}\sum_{j=0}^{N-1} T_{Wj}\cos(nwj\Delta t)\Delta t \\ b_n = \frac{2}{T}\sum_{j=0}^{N-1} T_{Wj}\sin(nwj\Delta t)\Delta t \end{cases} \quad (5)$$

Since the carriage body to the high frequency disturbance attenuation effects, therefore, take the order harmonics of 2 to obtain a limited term harmonic Fourier series which can be expressed as:

$$T_W(j\Delta t) = T_{W,p} + \sum_{n=1}^{2} A_n \sin(nwj\Delta t + j_n) \quad (6)$$

Compartments inside wall temperature

Compartment wall temperature of the delay and attenuation

Compartment wall temperature wave transmitted to the inner wall of the carriage attenuation ratio and delay time calculation, formulas are as follows:

$$V_{e,n} = 0.9e^{\sum_{i=1}^{n} R_i S_i/\sqrt{2}} \left(\frac{S_1+a_{in}}{S_1+Y_1}\right)\left(\frac{S_2+Y_1}{S_2+Y_2}\right)\mathbf{L}\left(\frac{S_n+Y_{n-1}}{S_n+Y_n}\right)\left(\frac{Y_n+a_w}{a_w}\right) \quad (7)$$

$$e_{e,n} = \tfrac{1}{15}\left(40.5\sum_{i=1}^{n} R_i S_i - \arctan\frac{a_{in}}{a_{in}+\sqrt{2}Y_1} + \arctan\frac{Y_n}{Y_n+\sqrt{2}a_w}\right) \quad (8)$$

$$S_i = \sqrt{\frac{2p\,l_i c_i r_i}{24}} \quad (9)$$

Where: $V_{e,n}$-- compartment outer compartment temperature wave transmitted attenuation factor of the inner wall surface; R_i - thermal resistance of the compartment wall layers of material; S_i -- heat storage coefficient of compartment wall layers, the following table for the cabin wall i the layers number; product R and S is the thermal inertia index D; a_{in}—-- car interior wall surface heat transfer coefficient; coefficient Y_i -- outer surface of the heat storage compartment wall layers of material; $\varepsilon_{e,n}$ -- outer compartment temperature wave propagation to delay compartment inner wall surface; λ_i -- thermal conductivity of the layer material; c_i -- layers of the specific heat capacity; ρ_i -- density layers of material.

The value of heat transfer coefficient inside the compartment is determined by the movement of air around the wall of the compartment, and the calculation to vehicle air-conditioning is when air speed is set to 3 m/s, average wind speed is set to 1.5 m/s inside the compartment, when to laminar Flow Reynolds number $Re<5\times10^5$, $a_{in} = 3.86\sqrt{v_n/l_h}$;when the turbulent motion, $Re>5\times10^5$, $a_{in} = 5.94v_n^{0.6}l_h^{-0.2}$ .Where in l_h size is amorphous, as the height of the vertical wall of the vehicle body, as the horizontal width of the body wall[1]; v_n is the vehicle wind speed. For the first layer of the surface of the heat storage coefficient Y1: when $R_1S_1\geq1$, $Y_1=S_1$; when $R_1S_1\leq1$, $Y_1 = \frac{R_1S_1^2+a_{in}}{1+a_{in}R_1}$; for layer 2 to the last layer (n layer) of the surface of the heat storage coefficient Y2, Y3, ..., Yn : When $R_nS_n \geq 1$, $Y_n=S_n$; when $R_nS_n \leq 1$, $Y_n = \frac{R_n S_n^2 + Y_{n-1}}{1+Y_{n-1}R_n}$.

Compartments inside wall temperature determination:

Because the temperature of the outer wall of the vehicle is changed by the attenuation and delay of the wall, the amplitude and phase angle of the original temperature wave function are changed, so the new temperature and the average temperature of the inner wall of the compartment are formed as follow:

$$T_n(t) = T_{nb,p} + \frac{A_n}{V_{e,n}} \sin(nwt + j_n - e_{e,n}) \quad (10)$$

$$\begin{cases} (T_{WZ,p} - T_{W,p})a_w = (T_{W,p} - T_{nb,p})/R \\ (T_{WZ,p} - T_{W,p})a_w = (T_{nb,p} - T_{in})a_{in} \end{cases} \quad (11)$$

Where: $T_{WZ,p}$ - Integrated average temperature outside the car; $T_{nb,p}$ -- average temperature of the inner wall surface; R-- cabin wall overall heat transfer resistance; T_{in}--the air temperature inside the compartment.

Thermal comfort degree-hour and response factor of the compartment structure

Thermal comfort degree-hour of the compartment structure

The thermal comfort degree-hour of the compartment structure:

$$DH = \sum \left[\left| T_{nb,p} - T_n(t) \right| \bullet Dt \right] \quad (12)$$

DH is a vehicle body structure of thermal comfort degree-hour within a period (24h); $T_{nb,p}$ is the average temperature (k) of the inner wall surface envelope structure in a period (24 h); $T_n(t)$ is the temperature (k) at each time of the inner wall surface envelope structure in a period (24 h); Δt is the calculation interval, which takes 1 h.

The response factor of the compartment structure

The response factor of the compartment structure:

$$BER = DH \sum (Q_{hc}^n Dt) \quad (13)$$

$$Q_{hc}^n = \frac{T_w - T_{in}}{R_0} \quad (14)$$

$$R_0 = R_1 + R_2 + \mathbf{L} + R_n \quad (15)$$

Where: $\sum (Q_{hc}^n Dt)$ is a calculation cycle (24h) ,Air conditioning load caused by heat transfer envelope accumulated value; R_0 is the envelope of the total thermal resistance ($(m^2 \cdot k)/W$).

Case analysis

Given the length and width of the refrigerator compartment structure are much greater than its thickness, heat transfer is assumed that the vehicle is a one-dimensional heat transfer; Since the carriage is composed of a variety of composite materials, the calculation can be simplified to a multilayer material in the thickness direction of the superposition, and assuming that the interior material is dried, and inside, the outer surfaces are the two isothermal surface, and the vehicle interior the air temperature is kept constant. Compartment structure is shown in Table 1, which in turn made from outside to inside 3mm glass steel, 40mm polyurethane, 20mm vacuum insulation panel, 40mm polyurethane, 3mm glass steel. In addition to the vehicle body wall vacuum insulation materials, thermal performance parameters are related to the actual test values prevail, vacuum thermal insulation material-related parameters obtained from the literature [4], as shown in Table 2.

Table1. Vacuum insulation composite envelope structure type

	Type 1	Type 2	Type 3
Layer 1	0.003m FRP fiberglass	0.003m FRP fiberglass	0.003m FRP fiberglass
Layer 2	0.04m polyurethane insulating material	0.08m polyurethane insulating material	0.02m vacuum insulation material
Layer 3	0.02m vacuum insulation material	0.02m vacuum insulation material	0.08m polyurethane insulating material
Layer4	0.04m polyurethane insulating material	0.003m FRP fiberglass	0.003m FRP fiberglass
Layer 5	0.003m FRP fiberglass		

Table2. Materials and thermal parameters of the vehicle body

material / Unit	δ /m	R $/(m^2 \cdot k) \cdot W^{-1}$	D	λ $/W \cdot (m^2 \cdot k)^{-1}$	ρ $/kg \cdot m^{-3}$	c $/kJ \cdot (kg \cdot k)^{-1}$
Polyurethane insulation materials	0.1	3.03	1.1	0.033	30	1.38
Vacuum insulation material	0.02	2.86	0.6	0.007	35	0.67
FRP	0.003	0.0058	0.054	0.52	1800	1.26

Note: The actual calculation of the specific heat of units should be taken w·h/(kg·k), and therefore should be multiplied by the conversion factor to calculate the value 0.2778.

The calculation is based on the vehicle (length × width × height were 4.1m × 1.75m × 1.17m)developed by Guangzhou University, and based on the assumption ,inside the compartment air temperature was kept at 0 ℃, the vehicle starting from 00:00 midnight with the speed of 80*km/h* upwind traveling from north to south, and 23:00 end of the trip. According to the conditions given, by querying the Guangzhou summer solstice hourly outdoor air temperature and other parameters, to the top of the compartment parameters, for example, substitute it into formula (1) ~ (2), which can obtain different range compartment at the top of the wall surface's temperature at the summer solstice hourly in Guangzhou area ; Then from (3) to (6) Fourier series expression of the temperature of the outer surface of the wave can be obtained , as shown in Figure 1. Then in the formula (7) to (11) is substituted into the cabin roof outside surface temperature data, as well as related data in Table 2, you can get multiple attenuation and delay time combinations of different insulating materials on top of the vehicle body integrated temperature outside the vehicle, as well as surface temperature within the cabin roof; Finally, by the formula (12) to (15), the thermal comfort degree-hour and response factor of three different compartments of the top of the combination form can be obtained, shown in Figure 1.

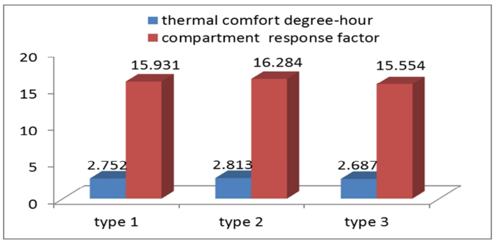

Fig.1 Thermal comfort degree-hour and compartment response factor of different compartment structure types

The thermal comfort degree-hour responses to the influence of carriage structure of vehicle heat stability, the thermal comfort degree-hour smaller, thermal stability of carriage structure of vehicle the better. Compartment of the reaction coefficient reflects the influence of carriage structure of vehicle heat stability and energy consumption of air conditioning system, the higher the reaction coefficient, the worse the thermal stability inside the vehicle, the higher the cooling load caused by the heat transferred for carriage structure of the refrigeration system of the thermal stability. By comparing Figure 1, the thermal comfort degree-hour of the different structural types of refrigerated trucks vacuum insulation composite envelope at the top, Type 3 (2.687) <Type 1 (2.752) <Type 2 (2.813). Similarly, for carriage the reaction coefficient top of the structure, type 3 (15.554) <type 1 (15.931) <type 2 (16.284), indicates that when under the same thermal conditions, vacuum heat insulation material on the thermal stability of the outermost layer of the composite structure is the best, the refrigerated truck refrigeration system energy consumption is the smallest, the vacuum heat insulation material is on the middle level, and on the innermost layer of the thermal stability is the worst.

Conclusions

The heat insulation performance and energy consumption of refrigerated truck is not only related with the thermal insulation material's thermal resistance, thermal conductivity and other parameters, but also with the mounting arrangement in the form of layers of insulating material for multi-layer material combination cold car. For the top of the refrigerated trucks vacuum insulation composite envelope, when vacuum compartment material is disposed in the outer layer by compartment, thermal comfort degree-hour and response factor were 2.687, 15.554; when insulating material is disposed in the compartment at the middle position, thermal comfort degree-hour and response factor were 2.752 and 15.931; when disposed in the innermost layer by compartment, thermal comfort degree-hour and response factor were 2.813 and 16.284. Reasonable layout the combinations of refrigerated compartment insulation material, not only can improve the thermal stability of the environment within a refrigerated compartment, but also can reduce the energy consumption of the refrigeration system refrigerated truck.

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References

[1] Li Jin, Xie Ruhe, Liu Guanghai, et al. Research on the thermal stability of typical refrigerated trucks compartment in the outdoor sol-air temperature conditions[J]. Transactions of the Chinese Society for Agricultural Machinery, 2012,43(8): 159-166.

[2] BAI Gui-ping , JI Zhao-liang. Evaluation of the envelope heat insulation performance by using thermalcomfort degree-hour and envelope response factor[J]. Journal of Guangzhou University (Natural Science Edition), 2005,4(1):87-90.

[3] BAI Gui-ping , JI Zhao-liang. The effects of thermal insulation form on thermal stability and the cooling load of air conditioning[J]. Building energy＆ environment, 2005,24(1):69-72.

[4] Wang Wen-jun, Kan An-kang , Han Hou-de , et al. The study of the best position about the vacuum insulation panel used in mobile house located in cold regions[J].New Building Materials, 2010(2):34-40.

[5] Asksaka H. Development of expanded AMeDAS weather data for building energy calculation in Japan[J]. ASHRAE Trans, 2000, 106(2):455-465.

[6] Huang J. The impact of different weather data on simulated Residential heating and cooling loads[J]. ASHRAE Trans,1998, 104(2):516-527.