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LABORATORY MOCK-UP OF THE SVET-3 SPACE GREENHOUSE: EARLY EVALUATION OF THE HEAT FLUXES THROUGH THE WALLS OF THE CLOSED PLANT GROWTH CHAMBER

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Abstract: *A Laboratory Mock-up of the Svet-3 Space Greenhouse (SG) - a small greenhouse of enclosed and environmentally controlled type is being developed. As a first stage a thermoelectric module (TEM) based air-conditioning system is being built. To select TEM with adequate heat pumping capacity all thermal loads must be evaluated. The study proposed describes the mathematical instrument and calculations of the heat fluxes through the walls of a commercially available closed thermally insulated box. The total thermal load for heating regime at steady state temperatures – chamber temperature* T_{ch} *= 35ºC and ambient temperature* T_{amb} *= 15ºC and in the absence of additional heat sources is calculated. An equivalent electrical circuit is created to model the thermal processes in the chamber using the simulating program OrCAD PSPICE. The total thermal load* $Q_{overall}$ *= 5.128 W, as well as the heat source (Al plate) temperature* T_{pl} *= 54.17°C are assessed.*

ЛАБОРАТОРЕН МАКЕТ НА КОСМИЧЕСКА ОРАНЖЕРИЯ СВЕТ-3: ПРЕДВАРИТЕЛНА ОЦЕНКА НА ТОПЛИННИТЕ ПОТОЦИ ПРЕЗ СТЕНИТЕ НА ЗАТВОРЕНА КАМЕРА ЗА ОТГЛЕЖДАНЕ НА РАСТЕНИЯ

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Ключови думи*: лабораторен макет, КО Свет-3, камера, топлинни потоци, загрята алуминиева плоча, термични съпротивления.*

Резюме: *Разработва се лабораторен модел на космическа оранжерия Свет-3 - малка оранжерия от затворен тип с управляема среда. Като първи етап се изгражда система за кондициониране на въздуха в камерата за отглеждане на растения на базата на термоелектричен модул (ТЕМ). За да се избере ТЕМ с подходящ топлинен капацитет, трябва да се оценят всички топлинни товари. Предложеното изследване описва математическия апарат и изчисленията на топлинните потоци през стените на една комерсиална топлинно-изолирана камера. Изчислен е общият термичен товар за режим на нагряване, температура на въздуха в камерата* T_{ch} *= 35ºC и на външната среда* T_{amb} *= 15ºC в стационарен режим, при отсъствие на допълнителни източници на топлина. За моделиране на термичните процеси в камерата с помощта на симулационната програма OrCAD PSPICE е създадена еквивалентна електрическа схема. Определени са общият термичен товар* $Q_{overall}$ *= 5.128 W и температурата на източника на топлина (алуминиева плоча)* T_{pl} *= 54.17°C.*

Introduction

Background

The Bulgarian Svet SG, flied onboard the Mir Orbital Station in the period 1990-2000 and used to accommodate a series of successful long-term plant experiments [1], had a plant chamber, open type, allowing contact with the station cabin atmosphere [2]. In order to isolate the variable of study (microgravity), flight experiments must be conducted in a controlled environment. Closed chamber plant growth facilities with controlled environment have been developed and flown onboard the U.S. Space Shuttles and the International Space Station (ISS) [3, 4, 5]. As a first stage of the Svet-3 project, a Laboratory Mock-up (LM) of a small greenhouse of closed and environmentally controlled type is being built. The development work at this stage emphasizes on control of plant leaf environment – air conditioning and gas composition control. Our first task is to develop a TEM based system for active thermal control of the chamber environment (from 15ºC to 35ºC) under the conditions of varying ambient temperatures (from 15ºC to 35ºC).

Problem statement

To select TEM with adequate heat pumping capacity the amount of heat to be absorbed or released from the TEM unit must be evaluated considering all thermal loads [6]. In a common case, the thermal load may have passive (radiation, convection, or conduction) and active components. The total amount of heat that must be absorbed or released in order to maintain steady state chamber temperature for any particular operating regime of TEM (cooling or heating) depends on the required temperature difference, the insulating properties of the chamber walls, the availability of radiation sources and electronic components in the chamber volume, etc. [7].

The objective of this research is to select mathematical machinery and to calculate the heat fluxes released from an Al plate (used as a heat source), and transferred through the walls of a commercially available closed heat-insulated box, to be served for a plant growth chamber. The calculations are made for the heating regime, the worst-case temperature difference ΔT = 20°C and in the absence of other heat loads. The hot Al plate temperature (T_{pl}) is calculated in order to check if it is below the maximum admissible operating temperature of about 65ºC for the standard TEM units.

Nomenclature

i	***First bottom index (number of the chamber area)***	
	i = 1 (Left wall), i = 2 (Right wall), i = 3 (Bottom wall), i = 4 (Upper Plexiglas window), i = 5 (Upper ABS area), i = 6 (Front door), i = 7 (Back wall thermal convection area), i = 8 (Al plate-back wall thermal conduction area).	
j	***Second bottom index (number of the wall layer)***	
	j = 1 (Internal air layer for convection), j = 2 (Internal insulating layer), j = 3 (Middle insulating layer), j = 4 (External insulating layer), j = 5 (External air layer for convection).	
h	***Heat transfer coefficient***	
	$h_{i,1}$	Inside heat transfer coefficient for the i^{th} element,
	$h_{i,5}$	Outside heat transfer coefficient for the i^{th} element,
	h_{pl}	Al plate inside heat transfer coefficient.
k	***Thermal conductivity***	
	$k_{i,2}$, $k_{i,3}$, $k_{i,4}$	Thermal conductivity of the i^{th} element – j^{th} layer (j = 2, 3, 4).
Q	***Heat flux***	
	Q_i	Heat flux through the i^{th} chamber area,
	Q_{pl_ch}	Convective heat flux from the plate to the chamber air,
	Q_{pl_wall}	Heat flux from the plate through the back wall to the ambient,
	$Q_{overall}$	The total heat flux relesed from the source (Al plate).
Θ, (R)	***Thermal resistance, (electrical resistance)***	
	$\Theta_{i,j}$, ($R_{i,j}$)	Thermal resistance of the i^{th} element, j^{th} layer,
	$\Theta_{pl\text{-}ch}$, ($R_{pl\text{-}ch}$)	Convective thermal resistance of the transition plate – chamber.

Materials and Methods

Plant Chamber Structure

The experimental equipment consists of a sealed heat insulated box with overall dimensions L=263.5 mm, W=222 mm, H=382 mm and internal dimensions L=212 mm, W=152 mm, H=327.5 mm.

Four of the walls (excluding the top and the front one) are three-layered walls, consisting of an external layer of Acrylonitrile Butadiene Styrene (ABS) material, 2.4 mm in thickness, in contact with the ambient temperature, followed by polyurethane foam insulation and another internal ABS layer, 2 mm thick, in contact with the chamber environment. An Aluminum (Al) plate, 190x305x2 mm in size, mounted inside the back wall in the place of the internal ABS layer and in thermal contact with the other two layers, serves as a heat source.

The front door consists of a double walls made of ABS material, 3 mm in thickness, with an insulating air gap between them.

There is a clear top window, 205x124 mm in size, in the top wall. It consists of two 3 mm thick Plexiglas layers with an air gap between them. The rest area of the wall is of the same three-layered structure (ABS/polyurethane/ABS) as the four walls.

A fan, mounted in the chamber volume, at the centre of the Al plate, provides airflow of 0.1 m/sec velocity for forced convection of the chamber air. Another external fan of the same flow rate (0.1 m/s) blows the chamber outside imitating the forced convection of the air typically existing in the spacecraft cabin.

Theoretical Performance

The objective of this study is to quantify the overall power of a heat source (the Al plate), required to maintain the chamber temperature (T_{ch}) at 35°C under the conditions of 15°C ambient temperature (T_{amb}) outside. In the absence of additional heat sources the overall thermal load will be the heat flux released from the source, and transferred through the chamber air and walls to the ambient by convection and conduction due to the temperature difference $\Delta T = T_{ch} - T_{amb}$ [8].

The overall thermal load ($Q_{overall}$) is the sum of the convective heat flux transferred from the plate to the chamber air ($Q_{pl\text{-}ch}$) and the heat flux transferred from the plate through the back wall to the ambient ($Q_{pl\text{-}wall}$) as indicated in Eq. (1):

(1) $$Q_{overall} = Q_{pl\text{-}ch} + Q_{pl\text{-}wall}$$

The convective heat transfer [9] from the heat source (Al plate) to the chamber can be expressed as:

(2) $$Q_{pl-ch} = (T_{pl} - T_{ch})h_{pl}A_{pl}$$

where A_{pl} – the Al plate area; h_{pl} – the heat transfer coefficient between the Al plate and the chamber air.

Further, the flux $Q_{pl\text{-}ch}$ is transferred to the chamber air by convection, through the walls by conduction and to the ambient by convection.

(3) $$Q_{pl-ch} = \sum_{i=1}^{7} Q_i$$

where Q_i ($i = 1\div7$) – the heat fluxes through the corresponding chamber wall areas described as:

(4) $$Q_{i=(1,\dots,7)} = \frac{T_{ch} - T_{amb}}{\frac{1}{h_{i,1}A_{i,1}} + \frac{D_{i,2}}{k_{i,2}A_{i,2}} + \frac{D_{i,3}}{k_{i,3}A_{i,3}} + \frac{D_{i,4}}{k_{i,4}A_{i,4}} + \frac{1}{h_{i,5}A_{i,5}}}$$

where:

(4.1) $$\frac{1}{h_{i,1(5)}A_{i,1(5)}} = \Theta_{i,1(5)}$$

are the convective thermal resistances of the transitions chamber air - chamber wall ($\Theta_{i,1}$) and chamber wall - ambient air ($\Theta_{i,5}$) respectively;

(4.2) $$\frac{D_{i,2(3,4)}}{k_{i,2(3,4)}A_{i,2(3,4)}} = \Theta_{i,2(3,4)}$$

are the conductive thermal resistances of the corresponding layers of the three-layered walls. $D_{i,j}$ and $A_{i,j}$ are the thickness and area of the i^{th} element, j^{th} layer.

Once the heat fluxes through the chamber walls are determined, the heat flux from the source entered the chamber by convection, can be calculated using Eq. (3). The heat flux calculated by Eqs. (2) and (3) give a possibility to determine the heat source (Al plate) temperature (T_{pl}):

(5) $$T_{pl} = T_{ch} + \frac{Q_{pl-ch}}{h_{pl}A_{pl}}$$

Using the so determined plate temperature, the heat flux $Q_{pl\text{-}wall}$ transferred through the back wall by conduction and to the ambient by convection can be calculated:

(6) $$Q_{pl-wall} = \frac{T_{pl} - T_{amb}}{\frac{D_{8,3}}{k_{8,3}A_{8,3}} + \frac{D_{8,4}}{k_{8,4}A_{8,4}} + \frac{1}{h_{8,5}A_{8,5}}}$$

where:

(6.1) $$\frac{D_{8,3(4)}}{k_{8,3(4)}A_{8,3(4)}} = \Theta_{8,3(4)}$$

are the conductive thermal resistances of the corresponding back wall layers for the conductive Al plate – back wall area;

(6.2)
$$\frac{1}{h_{8,5}A_{8,5}} = \Theta_{8,5}$$

is the convective thermal resistance of the transition back wall - ambient air for the $Q_{pl\text{-}wall}$ flux; and T_{pl} is the Al plate temperature calculated by Eq. (5).

The thermal resistance of the transition Al plate - chamber air $\Theta_{pl\text{-}ch}$ is calculated as:

(6.3)
$$\Theta_{pl-ch} = \frac{1}{h_{pl}A_{pl}}$$

Substituting the values of $Q_{pl\text{-}ch}$ and $Q_{pl\text{-}wall}$ calculated by Eqs. (3) and (6) in (1) the overall heat power of the source can be assessed.

The following calculation succession is used to determine the convective heat transfer coefficients over the flat plate (walls) surfaces $h_{i,j}$ used in Eqs. (2) and (4) to (6).

As a result of the air viscosity a boundary layer exists near the surface where the fluid velocity changes from the value in the free stream (far from the wall) to zero at the wall. The boundary layer is considered laminar for lower Reynolds numbers ($Re < 3.10^5$) and turbulent for higher ones ($Re > 3.10^6$). To determine if the airflow is laminar or turbulent the Reynolds number is calculated as:

(7)
$$Re = \frac{Lv\rho}{\mu}$$

where: L – characteristic length; v – air flow velocity; ρ – air density; μ – dynamic air viscosity.

The Prandtl number is calculated using:

(8)
$$Pr = \frac{c\mu}{k}$$

where c – specific heat of the air; k – thermal conductivity of the air.

For forced convection, the Nusselt number is generally a function of the Reynolds number and the Prandtl number. In case of a flat plate, uniformly heated over its entirety (isothermal plate), and blown by a laminar flow that is parallel to the axis, the average Nusselt number is calculated as:

(9)
$$Nu = 0{,}664(Pr)^{\frac{1}{3}}(Re)^{\frac{1}{2}}$$

The heat transfer coefficients for the corresponding boundary layers are calculated:

(10)
$$h = \frac{k}{L}Nu$$

Since air properties can vary significantly with temperature, a recommended approach is to use the film temperature T_f of the boundary layer, which is the average of the wall and free-stream temperatures. All air properties are determined for the specific temperature of the air boundary layer.

Calculations

Calculation of the heat transfer coefficients:

The Reynolds, Prandtl and Nusselt numbers and heat transfer coefficients were calculated using Eqs. (7) to (10) where L is the characteristic length for the corresponding wall area, v = 0.1 m/s - the air flow velocity and the air properties ρ, μ, c, and k are determined for the specific film temperatures (T_f) of the air boundary layers as follows:

T_f = 16°C (289 K) for the boundary layers outside the chamber walls;

T_f = 34°C (307 K) for the boundary layers inside the chamber walls;

T_f = 43.5°C (316.5 K) for the boundary layer near the plate.

The calculated Reynolds, Prandtl and Nusselt numbers and heat transfer coefficients for every transition are given in Table 1. Since $Re < 3.10^5$ for every transition the airflow is considered laminar.

Table 1.

Re	Pr	Nu	h [W/m^2.K]
943.69	0.697	18.09	$h_{1,1}$, $h_{2,1}$, $h_{3,1}$,$h_{4,1}$, $h_{5,1}$ = 3.19
1051.31	0.697	19.09	$h_{6,1}$ = 3.02
1018.65	0.697	18.79	$h_{7,1}$ = 3.07
948.67	0.697	18.13	h_{pl} = 3.18
1249.48	0.701	20.85	$h_{1,5}$, $h_{2,5}$, $h_{3,5}$, $h_{4,5}$, $h_{5,5}$ = 2.93
2239.02	0.701	27.91	$h_{6,5}$ = 2.19
1826.57	0.701	25.20	$h_{7,5}$ = 2.43

Calculation of the heat fluxes through the chamber walls

The following heat fluxes Q_i (i = 1÷7) through the chamber walls were calculated using Eq. (4), considering the chamber sizes, insulating layers thickness, the calculated heat transfer coefficients given in Table 1, and for T_{ch} = 35°C; T_{amb} = 15°C: Q_1 = 0.6675 W, Q_2 = 0.6675 W, Q_3 = 0.3614 W, Q_4 = 0.3913 W, Q_5 = 0.0939 W, Q_6 = 1.0689 W, Q_7 = 0.2806 W.

The total heat flow from the source (Al plate) entered the chamber air by convection was calculated using Eq. (3): $Q_{pl\text{-}ch}$ = 3.5311 W.

Substituting this value and the data: T_{ch} = 35°C; A_{pl} = 579.5 cm^2; h_{pl} = 3.18 $W/m^2.K$ in Eq. (5) the plate temperature T_{pl} was calculated: T_{pl} = 54.17°C.

Using this value the heat flux $Q_{pl\text{-}wall}$ transferred from the plate through the back wall to the ambient is calculated using Eq. (6): $Q_{pl\text{-}wall}$ = 1.5969 W.

The overall heat power of the source is calculated by Eq. (1): $Q_{overall}$ = 5.128 W.

PSPICE model

A commonly accepted approach to describe the thermal parameters of the experimental chamber by an equivalent electrical circuit [10, 11] is applied. All non-electrical processes are described in terms of their electrical analogies as shown in Table 2.

Table 2.

Thermal parameters	Units	Analogous electrical parameters	Units
Heat, Q	W	Current, I	A
Temperature, T	K	Voltage, V	V
Thermal resistance, θ	K/W	Electrical resistance, R	Ω
Absolute zero temperature	0 K	Ground	0 V

An equivalent circuit is used to model the thermal processes going in the chamber in case of 20°C temperature difference between the chamber and ambient by means of the simulating program OrCAD PSPICE (free edition). The PSPICE produced model is shown on Fig 1.

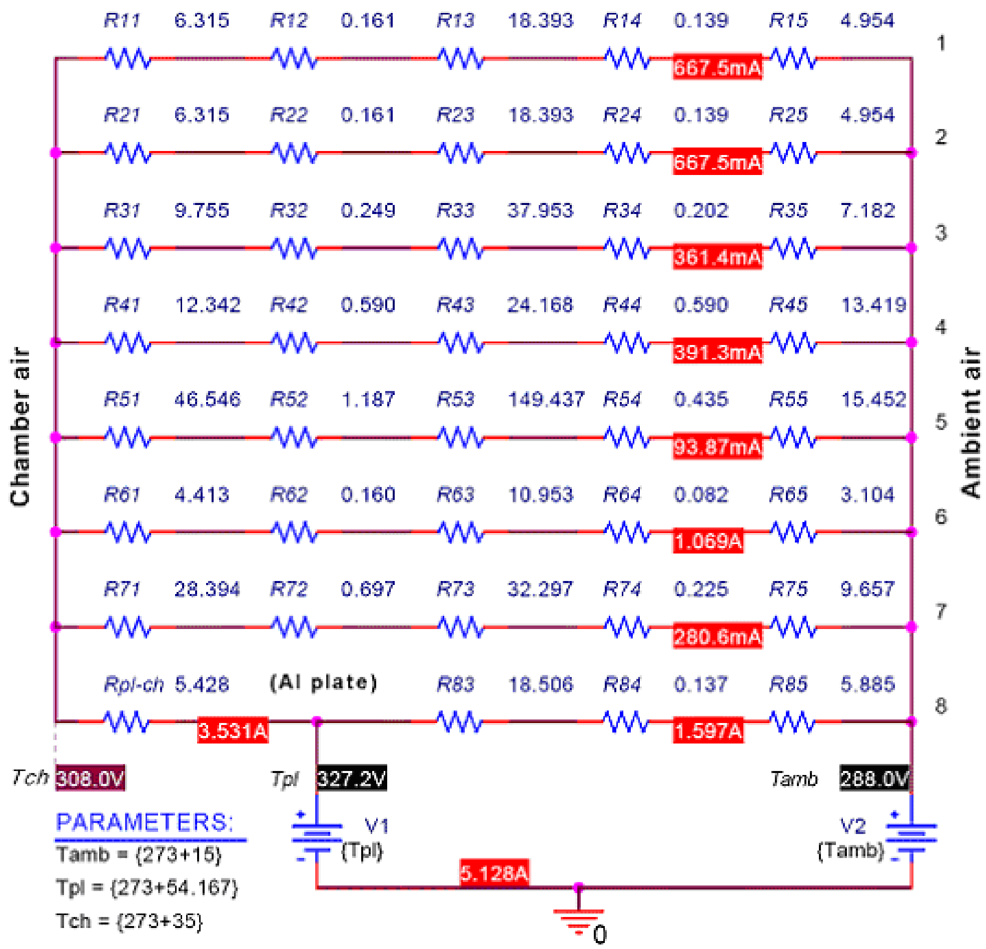

Fig. 1. Equivalent circuit of the plant chamber thermal parameters for steady state

The thermal resistances of all the chamber elements calculated using Eqs. (4.1), (4.2), (6.1), (6.2) and (6.3) are entered as electrical resistances and the heat fluxes and the steady state temperatures at different points are assessed.

At a next stage, this program is expected to be used for testing the time dependent behavior of the chamber. For this purpose, the thermal capacities of all the chamber elements will be calculated and entering these data the simulating program will give a possibility for a direct assessment of the chamber time constant by applying a voltage step across and measuring the system transient response time.

Results and Discussions

The total amount of heat that has to be removed to maintain steady state chamber temperature in the worst case, T_{ch} = 35°C, T_{amb} = 15°C, and in the absence of other heat loads, was assessed: $Q_{overall}$ = 5.128 W. The Al-plate temperature for this case is T_{pl} = 54.17°C.

A standard TEM module will be used to maintain the chamber temperature within the required limits. A practical limitation (of about 65°C) of the maximum admissible hot side temperature exists for all modules of such a type. The calculated hot plate temperature value is below this admissible level.

The upper wall is expected to be a potentially unstable component of the total thermal resistance of the chamber. At a next stage, a powered Light Unit will be placed over the top window. This unit would heat additionally the air in the space between the lamps and the external Plexiglas layer of the top window what would change considerably the heat transfer coefficient and the thermal resistance of the top wall respectively. This would require additional cooling in the lamp area. Calculating the radiated heat by the light source at this stage would require determination of the lamp operating temperature and air temperature in the lamp zone and recalculation of the heat transfer coefficients $h_{4,5}$ and $h_{5,5}$.

Conclusions

The final goal of the presented work is to make assessment the heat fluxes through the chamber walls in heating regime necessary to determine the type and number of Peltier modules at a next stage. Our calculations were based on the physical properties of the materials used in the commercially available insulated box to be served for a plant growth chamber. The presented work describes one of the operating temperature regimes in the plant chamber for the worst case – T_{amb} = 15ºC, T_{ch} = 35ºC, and without additional heat sources.

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