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Hybrid Adiabatic-Thermoelectric Cooling of Localized Workplaces

Relevance. Ensuring regulatory microclimate parameters at localized workplaces, in particular at technological equipment control posts (crane cabins, operator cabins), under conditions of extreme thermal loads typical of metallurgical production, is a critically important scientific and technical task. Traditional compressor air conditioning systems operating in an environment with temperatures up to 60 °C and intense infrared radiation demonstrate a significant drop in the coefficient of performance (COP) and high operating costs, which justifies the need to develop alternative, more energy-efficient solutions. The aim of the work is to increase the energy efficiency of local air conditioning systems for mobile facilities by developing, theoretically justifying, and determining the optimal parameters of a new two-level hybrid system that combines the principles of adiabatic and thermoelectric cooling. The research is based on the method of mathematical modeling of complex heat and mass transfer processes. The proposed system consists of two circuits: the external one, which creates an active thermal shield in the form of a blown shell with adiabatic cooling of the supply air, and the internal one, which provides final cooling of the working area using thermoelectric Peltier modules. A complex mathematical model was developed for the analysis, on the basis of which the multifactor parametric optimization problem was formulated. The objective function was chosen to minimize the total energy consumption of the system while observing the limiting conditions regarding the microclimate parameters in the cabin and the physical limitations of the equipment operation. The scientific novelty lies in the development of a new concept of a hybrid system, where the intermediate ventilated space performs a dual function: an active heat shield to intercept up to 80–90 % of external radiative and convective heat input, and an integrated cooling medium to remove heat from the hot side of thermoelectric modules. A mathematical model has been further developed that comprehensively describes the synergistic relationship between the circuits. The developed optimization methodology, in contrast to existing approaches that consider components separately, allows finding optimal design (shell thickness, number of thermoelements) and operating (ventilation efficiency, current strength) parameters of the entire system to achieve a global minimum of energy consumption. The developed approach is a scientifically sound basis for designing a new class of energy-efficient microclimate maintenance systems for mobile and

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stationary facilities (operator cabins, hardware compartments, electrical panels) operated in extreme industrial environments. The results of the work can be used to create specific engineering solutions that provide a significant reduction in operating costs and increased occupational safety.

Keywords: hybrid cooling system, adiabatic cooling, thermoelectric cooling, parametric optimization, energy efficiency, thermal radiation, COP.

Introduction

Problem statement. Ensuring regulatory microclimate parameters at localized workplaces in extreme industrial environments is one of the most complex scientific and technical tasks of modern life support systems engineering. Operation of technological equipment, in particular operator cabins in open-hearth and foundry workshops, occurs at ambient air temperatures reaching 50–60 °C and under the influence of intense thermal radiation from hot surfaces, the power of which can be several kW/m². Such conditions, according to SDS 3.3.6.042-99, are classified as harmful and dangerous, creating significant heat gain, which not only reduces labor productivity and operator concentration, but also poses a direct threat to his health due to the risk of heat stroke [12].

Analysis of existing solutions. Traditional air conditioning systems based on the vapor compression cycle (VCC), which are widely used in civil and commercial buildings, face a number of fundamental problems when operating in such conditions. Their energy efficiency, characterized by the coefficient of performance (COP), drops significantly due to the high condensation temperature of the refrigerant, and reliability is reduced due to the operation of the compressor at the limit of permissible loads [1, 7]. Alternative methods, such as direct adiabatic (evaporative) cooling, although demonstrating high energy efficiency, are thermodynamically limited by the wet bulb temperature and lead to an uncontrolled increase in relative humidity in the working area, which does not meet sanitary standards [2]. Solid-state thermoelectric coolers based on the Peltier effect, which offer high reliability and compactness, are inherently inefficient because their COP critically depends on the efficiency of heat removal from the hot side [3, 4], which is a challenging task in high-temperature environments. A review of the scientific literature shows that although there are studies of hybrid systems [9, 10, 11], there is a lack of comprehensive models for the design and optimization of systems designed to solve the specific problem of local air conditioning under conditions of dominant radiation heat transfer.

Formulation of the purpose of the article. The purpose of this study is to increase the energy efficiency of local air conditioning systems for mobile facilities by developing, theoretically justifying and analyzing a mathematical model for parametric optimization. [13] of a new two-level hybrid adiabatic-thermoelectric system in order to minimize the total energy consumption while maintaining the regulatory and comfortable microclimate parameters in the working area. To achieve this goal, it is necessary to solve the following tasks: develop a mathematical model of the system that takes into account the synergistic relationship between the circuits; formulate a multifactor optimization problem; conduct numerical modeling to determine the optimal design and operating parameters.

2. System description and mathematical model

2.1. Concept and architecture of a hybrid system

The proposed hybrid air conditioning system is based on the principle of two-level functional separation of tasks, which allows achieving high overall energy efficiency in extreme operating conditions. The system architecture (Fig. 1) includes two interconnected circuits: the external one, which performs the function of active thermal shielding, and the internal one, which provides local cooling of the operator's cabin working area. At the same time, forced circulation and the necessary air exchange (10 times/hour are assumed in the model) are provided to compensate for residual heat input (Q_{out}) and internal heat output (Q_{in}).

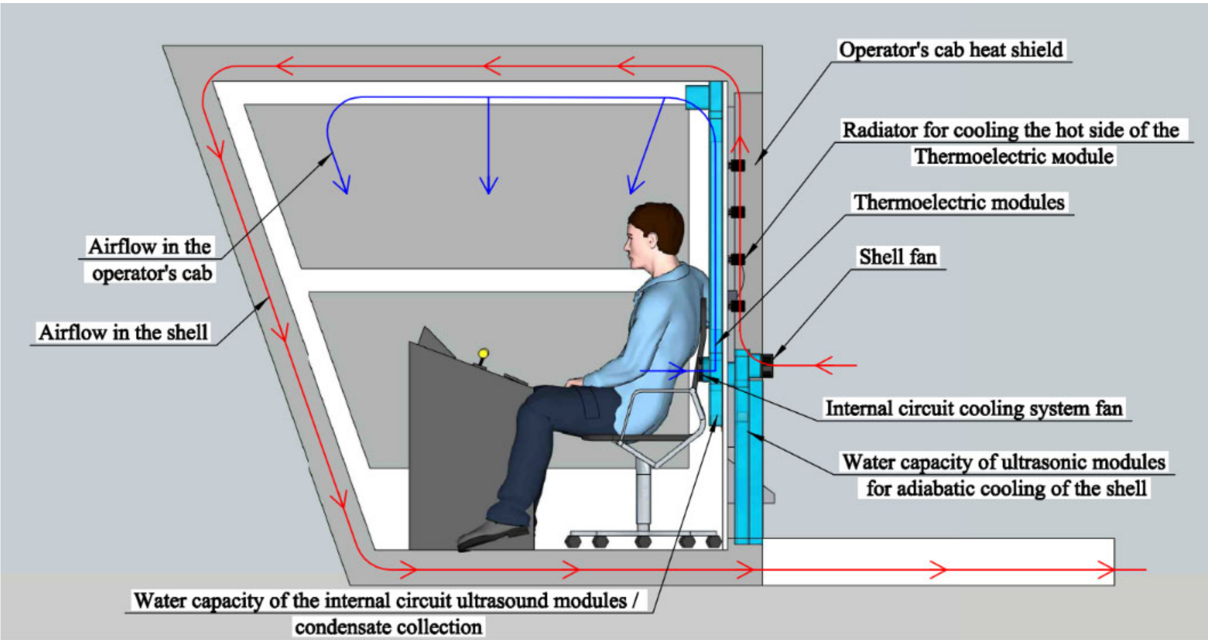

Fig. 1. Schematic diagram of a hybrid adiabatic-thermoelectric system

The external circuit (active heat shield) is designed to intercept and remove the majority of the external heat load. Its main element is a ventilated shell with a calculated thickness d_{sh}, which creates an intermediate space around the operator's cabin. Unlike passive thermal insulation, this shell is an active heat exchanger. A fan with calculated performance V_{sh} takes in external air, which passes through an adiabatic cooling unit, where, due to the evaporation of finely dispersed water, its temperature decreases to a level close to the wet bulb temperature. This flow of cooled air is forcibly circulated in the shell space, actively "washing away" the heat entering its outer surface and is emitted into the atmosphere or workshop premises.

The internal circuit (local cooling) is designed to compensate for residual heat input into the cabin and bring the microclimate parameters to the regulatory values. It is based on a block of thermoelectric modules (TEM) based on the Peltier effect. The cold sides of the modules, equipped with radiators and a fan, take heat from the internal volume of the cabin.

The synergistic relationship between the circuits is a key feature of the developed architecture. The hot sides of the thermoelectric modules are cooled not by high-temperature external air, but by a flow of pre-cooled air circulating inside the shell. This allows the temperature of the hot side of the TEM to be maintained at a significantly lower level, which radically increases their coefficient of performance (COP) and reduces energy consumption. Thus, the external circuit performs a dual function: (1) a heat shield for the operator's cabin and (2) an efficient cooling system for the internal circuit.

2.2. Mathematical model of the system

For quantitative analysis of the system's efficiency, a complex mathematical model has been developed that describes the heat and mass transfer processes in each of the circuits.

2.2.1. Mathematical description of the external contour

The thermal load on the shell (Q_{sh}) is the sum of the convective and radiative components:

$$Q_{sh} = Q_{conv} + Q_{rad} \tag{1}$$

where:

Convective load (Q_{conv}), W:

$$Q_{conv} = h_{out} \cdot A \cdot (T_{out} - T_{sur}) \tag{2}$$

Radiation load (Q_{rad}), W:

$$Q_{rad} = \varepsilon \cdot \sigma \cdot (T_{rad}^4 - T_{sur}^4) \tag{3}$$

where:

- h_{out}– convective heat transfer coefficient, W/(m²·K);
- A– shell surface area, m²;
- T_{out}– outdoor air temperature, °C;
- T_{sur}– shell surface temperature, °C;
- ε– surface emissivity coefficient;
- σ– Stefan-Boltzmann constant, W/(m²·K⁴);
- T_{rad}– effective temperature of radiation sources, °C.

To remove this heat, an air flow cooled in an adiabatic block to a temperature of:

$$T_{ad} = T_{out} - \eta_{ad} \cdot (T_{out} - T_{wb}) \tag{4}$$

where:

- T_{ad}– air temperature after the adiabatic block, °C;
- η_{ad}– adiabatic cooling efficiency (taken 0.85);
- T_{wb}– wet bulb temperature, °C.

The power that can be removed by the air flow is determined by the equation:

$$Q_d = m_{air} \cdot c_p \Delta T_{air} \tag{5}$$

In stationary mode $Q_d = Q_{sh}$.

3.2.2. Mathematical description of the internal contour

The total heat load on the cabin (Q_{cab}) is the sum of heat input through the walls and internal heat output:

$$Q_{cab} = Q_{out} + Q_{in} \tag{6}$$

where:

Heat input through the walls (Q_{out}), W:

$$Q_{out} = U_{sh} \cdot S_{sh} \cdot (T_{sh} - T_{in}) \tag{7}$$

where:

- U_{sh}– heat transfer coefficient of the cabin walls, W/(m²·K);
- S_{sh}– cabin surface area, m²;
- T_{sh}– average air temperature in the shell, °C;
- T_{in}– target temperature inside the cabin (accepted ≤ 25 °C).

The total electrical power consumed by thermoelectric modules (P_{pl}) is calculated through the required cooling capacity and their COP:

$$P_{pl} = n_{pl} \cdot \frac{Q_{cpl}}{COP_{pl}(I_{pl}, \Delta T_{pl})} \tag{8}$$

where:

- n_{pl}– number of thermoelectric modules;
- Q_{cpl}– required cooling capacity of one TEM, W;
- COP_{pl}– productivity coefficient of one TEM;
- I_{pl}– current strength at each TEM, A;
- ΔT_{pl}– operating temperature difference at the TEM, °C.

3.2.3. Synergistic relationship model

The key element that unifies the model is the average air temperature in the shell (T_{sh}). It is the output parameter for the first loop and simultaneously the input for the second.

$$T_{sh} = T_{ad} + \frac{Q_{sh}}{2 \cdot m_{sh} \cdot c_p} \tag{9}$$

- m_{sh}– mass air flow rate in the shell, kg/s;
- c_p– specific heat capacity of air (taken to be 1005 J/(kg·K)).

Temperature T_{sh} directly affects both the cabin load (Q_{sh}) and the efficiency of TEM (COP_{pl}), since it determines the temperature of their hot sides.

3. Parametric optimization methodology

The transition from the analysis of individual operating modes to the design of an energy-efficient system requires the use of mathematical optimization methods. A simple analytical calculation is impossible due to the complex nonlinear dependence and mutual influence of thermophysical processes in the two circuits. Therefore, to find the optimal design and operating parameters of the hybrid system, a methodology was developed based on the formulation and solution of the multifactor parametric optimization problem.

3.1. Formulation of the optimization problem

Formally, the optimization problem is to find a vector of controlled parameters X that minimizes the selected objective function $F(X)$while observing a system of functional and parametric constraints.

Objective function: Minimize the total electrical power (P_{all}) consumed by all system components:

$$P_{all} = P_{ven}(X) + P_{ad}(X) + P_{pl}(X) \rightarrow min$$

Vector of controlled variables:

$$X = \{V_{sh}, n_{pl}, I_{pl}\}$$

where V_{sh}– is the performance of the external circuit fan; n_{pl}– is the number of thermoelectric modules; I_{pl}– is the current strength at each TEM.

System of restrictive conditions:

1. Comfort condition: $T_{in}(X) \leq T_{goal}(X)$.
2. Physical limitations of TEM: $\Delta T_{pl}(X) \leq T_{maxpl}(X), I_{pl} \leq I_{maxpl}$
3. Engineering constraints: The minimum envelope ventilation performance ($V_{sh\,min}$) is established, necessary to ensure turbulent flow regime and effective heat removal.

3.2. Solution method

The optimization problem is a complex nonlinear problem. Numerical methods are used to solve it, in particular, the sequential quadratic programming (SQP) algorithm, implemented in the *fmincon function of the Optimization Toolbox* package of *the MATLAB* environment. This method has proven its effectiveness for solving smooth nonlinear problems with constraints.

4. Modeling results and their discussion

Based on the developed methodology, numerical optimization was carried out to determine the optimal operating parameters of the hybrid system.

4.1. Conditions for carrying out optimization calculations

The calculation was performed for the most complex but realistic scenario simulating the operating conditions near an open-hearth furnace. The following key input parameters were adopted:

- External conditions: $T_{out} = 50°\text{C}$, $\varphi_{out} = 15\%$, $T_{rad} = 80°\text{C}$, Construction: Shell area $S_{sh} = 10\ \text{m}^2$, shell thickness (fixed) $d_{sh} = 0.1$ m, surface emissivity $\varepsilon = 0.05$ (aluminized fabric).
- Target parameters: $T_{goal} \leq 25°\text{C}$
- Internal heat input (from the operator and equipment) was assumed to be Q_{in} = 300 W.

- MT 2.6-0.8-263 T 1 S was selected with the following passport characteristics (at the temperature of the hot side of the thermocouples T_{hot}=50° C): Maximum current (I_{pl}): 15.0A. Maximum cooling capacity (Q_{cab}( ): 319 W. Maximum temperature difference (ΔT_{plmax}): 77° C .

4.2. Results of multivariate optimization

Based on the results of the optimization algorithm *fmincon,* a vector of mode parameters was found that ensures the minimum total energy consumption of the system while observing all restrictive conditions. The summarized results are presented in Table 1.

Table 1

Optimal parameters and system performance for $T_{out} = 50°C$, $\varphi_{out} = 15\%$, $T_{rad} = 80°C$, *modules MT2,6-0,8-263T1S*

Optimal parameter	Value	System indicator	Value
Fan performance	V_{sh}	505.4	m³/h
Thermal load on the shell	Q_{sh}	770.4	Tue
Number of TEMs	n_{pl}	16	pcs
Average T° in the shell	T_{sh}	32.9	°C
Current on TEM	I_{pl}	2.13	AND
Thermal load on the cabin	Q_{cab}	458.2	Tue
Total energy consumption	P_{all}	216.3	Tue
Operating ΔT for TEM	ΔT_{pl}	22.9	°C
COP of one TEM	COP_{pl}	2.50	
Systemic COP	COP_{cab}	2.12	
Total COP	COP_{all}	5.68	
Water consumption for the shell		1.13	l/h

4.3. Discussion and analysis of results

The results obtained are extremely indicative and confirm the high potential of the proposed hybrid architecture.

5.3. 1. The optimal strategy is found. The algorithm has found a balanced engineering solution. The ventilation capacity of 505 m³/h is sufficient to maintain a low average temperature in the shell (only ~33 °C), which creates favorable conditions for the operation of the second circuit. At the same time, the flow is not excessive, which allows keeping the fan power at a low level (~ 18 W).

The key is the operating mode of the thermoelectric circuit. The optimizer determined that to discharge a load of 458 W, it is much more energy efficient to use a large number (16) of modules operating at a very low current (2.13 A). It is worth paying special attention to the fact that the found optimal operating current (2.13 A) is significantly lower than the maximum rated current of the module (15.0 A). The optimizer proved that it is much more energy efficient to use a large number of modules (16 pcs.), but to make them operate in the maximum COP mode (2.50), which is achieved precisely at low currents. This allows them to operate in the zone of maximum efficiency with a COP of each module of 2.50, which is an excellent indicator. The total power of the thermoelectric block was only 183 W.

It is also important to clearly separate the heat load given in Table 1: Q_{sh} = 770.4 W is the total external heat load (radiative and convective), which is effectively "intercepted" and removed by the external adiabatic circuit. Instead Q_{cab} = 458.2 W is the much smaller, residual heat load (which passed through the shell + internal 300 W), and which already compensates for the internal thermoelectric circuit.

5.3. 2. Energy efficiency analysis and comparison. The most indicative is the final system COP.

- The system COP across the cabin (COP_{cab}) is 2.12. This means that for every watt of electricity consumed, the system removes 2.12 watts of heat directly from the cabin. This value is completely competitive and even exceeds the expected COP for most industrial compressor plants under similar extreme conditions (where COP drops to 1.5–2.0).
- The overall thermodynamic COP (COP_{all}) reaches 5.68. This indicator demonstrates the enormous efficiency of the "active heat shield" concept, which, due to insignificant energy consumption (~ 33 W per fan and humidification), neutralizes more than 770 W of external heat load.

Compared to a compressor unit that would have to compensate for the full load on the cabin (~1.2 kW, taking into account the load on the shell if it were not there) and would consume more than 600–700 W, the developed optimized system is 3 times more energy efficient in terms of total consumption (~216 W).

3. Model limitations. It is worth noting that this stationary model does not take into account the dynamics of changes in external conditions and thermal inertia of the system. Also, the model of hydraulic resistance and internal convection is simplified. Further studies may include CFD modeling to refine these parameters.

Conclusions

1. A comprehensive mathematical model and parametric optimization methodology for designing hybrid adiabatic-thermoelectric air conditioning systems have been developed and verified, allowing for finding optimal design and operating parameters to minimize energy consumption.
2. The results of numerical optimization for extreme conditions ($T_{out} = 50°\mathrm{C}$, $\varphi_{out} = 15\%$, $T_{rad} = 80°\mathrm{C}$,) showed that the optimal system configuration is able to provide regulatory conditions in the operator's cabin with a total power consumption of only 216 W.
3. It was found that the optimal strategy for the system operation is to use the first adiabatic circuit to create an effective thermal shield and favorable conditions for the operation of the second thermoelectric circuit in the maximum COP mode (for individual modules ~ 2.5).

4. The calculated system coefficient of performance ($COP_{cab} = 2.12$) is competitive and exceeds the performance of most traditional compressor systems under similar operating conditions. This, combined with higher reliability and lower starting currents, proves the high potential of the proposed system for industrial applications.

Authors' information

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Гібридне адіабатично-термоелектричне охолодження локалізованих робочих місць

Актуальність. Забезпечення нормативних параметрів мікроклімату на локалізованих робочих місцях, зокрема на постах керування технологічним обладнанням (кабіни кранів, операторські), в умовах екстремальних теплових навантажень, характерних для металургійних виробництв, є критично важливою науково-технічною задачею. Традиційні компресорні системи кондиціонування, що працюють в середовищі з температурою до 60°C та інтенсивним інфрачервоним випромінюванням, демонструють значне падіння коефіцієнта продуктивності (COP) та високі експлуатаційні витрати, що обґрунтовує необхідність розробки альтернативних, більш енергоефективних рішень. Метою роботи є підвищення енергоефективності систем локального кондиціонування для мобільних об'єктів шляхом розробки, теоретичного обґрунтування та визначення оптимальних параметрів нової дворівневої гібридної системи, що поєднує принципи адіабатичного та термоелектричного охолодження. В основі дослідження лежить метод математичного моделювання складних тепло- та масообмінних процесів. Запропонована система складається з двох контурів: зовнішнього, що створює активний тепловий екран у вигляді продуваємої оболонки з адіабатичним охолодженням припливного повітря, та внутрішнього, що забезпечує фінішне доохолодження робочої зони за допомогою термоелектричних модулів Пельтьє. Для аналізу розроблено комплексну математичну модель, на основі якої сформульовано задачу багатофакторної параметричної оптимізації. Цільовою функцією обрано мінімізацію сукупного енергоспоживання системи при дотриманні обмежувальних умов щодо параметрів мікроклімату в кабіні та фізичних обмежень роботи обладнання. Наукова новизна полягає у розробці нової концепції гібридної системи, де проміжний вентильований простір виконує подвійну функцію: активного теплового екрана для перехоплення до 80-90% зовнішніх радіаційних та конвективних теплонадходжень, та інтегрованого охолоджувального середовища для відведення тепла від гарячої сторони термоелектричних модулів. Набула подальшого розвитку математична модель, що комплексно описує синергетичний зв'язок між контурами. Розроблена методика оптимізації, на відміну від існуючих підходів, що розглядають компоненти окремо, дозволяє знаходити оптимальні конструктивні (товщина оболонки, кількість термоелементів) та режимні (продуктивність вентиляції, сила струму) параметри всієї системи для досягнення глобального мінімуму енергоспоживання. Розроблений підхід є науково обґрунтованою основою для проектування нового класу енергоефективних систем підтримання мікроклімату для мобільних та стаціонарних об'єктів (кабін операторів, апаратних відсіків, електрощитових), що експлуатуються в *екстремальних промислових середовищах. Результати роботи можуть бути використані для створення конкретних інженерних рішень, що забезпечують значне зниження операційних витрат та підвищення безпеки праці.*

Ключові слова: гібридна система охолодження, адіабатичне охолодження, термоелектричне охолодження, параметрична оптимізація, енергоефективність, теплове випромінювання, COP.

Надійшла до редакції 11.07.25

Appendix 1. MATLAB Optimization Toolbox script

```
% ==
% MAIN SCRIPT FOR OPTIMIZATION OF HYBRID COOLING SYSTEM
% VERSION 9.0: Final..
% ==
% Author: Myronenko Oleksandr Petrovych (model developed based on
% of scientific work)
% Environment: MATLAB R2023a or later
% Requires: Optimization Toolbox (for the 'fmincon' function)
% -----------------------------------------------------------------------

clear; clc; close all ;

%% 1. DETERMINATION OF INPUT PARAMETERS AND CONSTANTS

% --- External conditions ---
p.T_zovn = 50; % [°C] Outdoor air temperature
p.phi_zovn = 15; % [%] Relative humidity of the outside air
p.T_vypr = 80; % [°C] Effective temperature of radiation sources
p.P_atm = 101.325; % [kPa] Atmospheric pressure

% --- Cabin and shell parameters ---
p.A_kab = 10; % [m²] Surface area of the cabin (and shell)
p.V_kab = 3.5; % [m³] Internal volume of the cabin
p.d_obol = 0.1; % [m] Shell thickness (100 mm)
p.W_obol_kanal = 1.5; % [m] Total width of the shell purge channel
p.L_obol_kanal = 4.0; % [m] **NEW INPUT PARAMETER:** Air path length in the shell
p.U_kab = 2.0; % [W/(m²·K)] Heat transfer coefficient of the cabin walls
p.Q_vn = 300; % [W] Internal heat input (operator + equipment)

% --- Target parameters and constraints ---
p.T_kab_tsil = 25; % [°C] Target temperature in the cabin
p.N_kab_vent = 10; % [h⁻¹] Target air exchange rate in the cabin
p.T_vyh_obol_max = 45; % [°C] Max. air temperature at the exit from the shell

% --- Passport characteristics of the used TEM (MT2.6-0.8-263T1S) ---
p.TEC_Imax = 15.0; % [A] Maximum module current
p.TEC_Vmax = 34.0; % [V] Maximum voltage
p.TEC_Qc_max = 319; % [W] Maximum cooling capacity
p.TEC_dT_max = 77; % [°C] Maximum temperature difference
p.TEC_Th_rated = 50; % [°C] Hot side temperature for which Vmax and dT_max are specified

% --- Thermophysical constants and efficiency ---
p.k_pov = 0.028; p.h_zovn = 10; p.epsilon = 0.05;
p.sigma = 5.67e-8; p.rho_pov = 1.15; p.cp_pov = 1005;
p.Lv = 2.45e6; p.nu_pov = 1.8e-5; p.eta_ad = 0.85;
p.eta_vent = 0.5; p.eta_dv = 0.8; p.eta_zvol = 0.8;
p.dT_radiator = 7; p.dT_holodna = 8;

%% 2. SETTING UP THE FMINCON OPTIMIZER
% Vector of variables x: x(1)=V_obol, x(2)=n_Pel, x(3)=I_Pel
nvars = 3;
x0 = [0.25; 8; 6]; % Starting point for search

% Bounds for variables [lower; upper]
lb = [0.1; 1; 2];
```

```
ub = [0.6; 20; p.TEC_Imax];

options = optimoptions( 'fmincon' , 'Display' , 'iter' , 'Algorithm' , 'sqp' , 'PlotFcn' ,
{@optimplotfval});
objectiveFun = @(x) objectiveFunction(x, p);
constraintFun = @(x) nonlinearConstraints(x, p);

%% 3. STARTING OPTIMIZATION
disp( '--- Running the FMINCON optimizer to find optimal parameters ---' );
[x_opt, P_min] = fmincon(objectiveFun, x0, [], [], [], [], lb, ub, constraintFun, options);

%% 4. DISPLAYING RESULTS
disp( '--- Optimization completed ---' );
disp( '=========================================================================' );
fprintf( 'For a shell with a thickness of %.2f m and a channel length of %.2f m:\n' , p.d_obol,
p.L_obol_kanal);
fprintf( 'Optimal solution found with minimum power consumption: %.2f W\n' , P_min);
disp( '=========================================================================' );
disp( 'Optimal system operating parameters:' );
fprintf( ' - Shell ventilation capacity (V_obol): %.2f m³/h\n' , x_opt(1) * 3600);
fprintf( ' - Number of thermocouples (n_Pel): %d pcs.\n' , round(x_opt(2)));
fprintf( ' - Current strength at each TEM (I_Pel): %.2f A\n' , x_opt(3));
disp( '----------------------------------------------------------------' );
% --- Final calculation with optimal parameters for detailing ---
[~, final_params] = objectiveFunction(x_opt, p);
disp( 'Detailed system performance at the optimal point:' );
fprintf( ' - Thermal load on the shell (Q_obol): %.2f W\n' , final_params.Q_obol);
fprintf( ' - Average temperature in the shell (T_ser_obol): %.2f °C\n' , final_params.T_ser_obol);
fprintf( ' - Thermal load on the cabin (Q_zag_kab): %.2f W\n' , final_params.Q_zag_kab);
fprintf( ' - Operating deltaT for TEM: %.2f °C\n' , final_params.delta_T_pel);
fprintf( ' - COP of one TEM (COP_pel): %.2f\n' , final_params.COP_pel);
disp( '----------------------------------------------------------------' );
disp( 'Power consumption distribution:' );
fprintf( ' - Shell fan power: %.2f W\n' , final_params.P_vent);
fprintf( ' - Humidification system power: %.2f W\n' , final_params.P_zvol);
fprintf( ' - Thermocouple power: %.2f W\n' , final_params.P_peltier);
disp( '----------------------------------------------------------------' );

% --- CALCULATION OF OUTPUT PARAMETERS ---
V_vent_kab = p.V_kab * p.N_kab_vent; % Cabin fan performance
m_vody_h = final_params.Q_obol / p.Lv * 3600; % Water consumption
fprintf( 'COLUMN COSTS AND PRODUCTIVITY:\n' );
fprintf( ' - Water consumption for the shell: %.2f l/h\n' , m_vody_h);
fprintf( ' - Cabin fan capacity: %.2f m³/h\n' , V_vent_kab);
disp( '----------------------------------------------------------------' );
fprintf( 'SYSTEM SOR (by cabin): %.2f\n' , final_params.Q_zag_kab / P_min);
fprintf( 'TOTAL THERMODYNAMICAL EFFECT: %.2f\n' , (final_params.Q_zag_kab + final_params.Q_obol) /
P_min);
disp( '=========================================================================' );

%% ==
% LOCAL FUNCTIONS (OBJECTIVE AND CONSTRAINTS)
% ==
function [P_zag, params] = objectiveFunction(x, p)
    % Objective function: calculates the total energy consumption
V_obol = x(1); n_Pel = x(2); I_Pel = x(3);
    if n_Pel < 0.1, n_Pel = 0.1; end
T_mt = Stull_Twb(p.T_zovn, p.phi_zovn);
T_ohol = p.T_zovn - p.eta_ad * (p.T_zovn - T_mt);
Q_obol = calculate_Q_obol_iterative(p, V_obol, T_ohol);
m_dot_obol = V_obol * p.rho_pov;
    if m_dot_obol < 1e-3, m_dot_obol = 1e-3; end % Division by zero protection
T_ser_obol = T_ohol + Q_obol / (2 * m_dot_obol * p.cp_pov);
Q_zag_kab = p.U_kab * p.A_kab * (T_ser_obol - p.T_kab_tsil) + p.Q_vn;
Qc_one = Q_zag_kab / n_Pel;
T_h_pel = T_ser_obol + p.dT_radiator;
T_c_pel = p.T_kab_tsil - p.dT_holodna;
delta_T_pel = T_h_pel - T_c_pel;
[~, V_pel, COP_pel] = Peltier_Performance_Interpolated(I_Pel, delta_T_pel);
P_peltier = n_Pel * (V_pel * I_Pel);
    % Accurate fan power model
d_obol = p.d_obol;
F_kanal = p.W_obol_kanal * d_obol;
```

```
hydraulic_diam = 2 * d_obol;
velocity = V_obol / F_channel;
Re = (p.rho_pov * velocity * hydraulic_diam) / p.nu_pov;
    if Re < 2300, lambda = 64/Re; else , lambda = 0.3164 / (Re^0.25); end % We take into account the
laminar mode
delta_P_obol = lambda * (p.L_obol_kanal / hydraulic_diam) * (p.rho_pov * velocity^2 / 2);
P_vent = (V_obol * (delta_P_obol + 50)) / (p.eta_vent * p.eta_dv);
m_vody = Q_obol / p.Lv;
P_zvol = (m_vody * 1000 * 40) / p.eta_zvol;
P_zag = P_peltier + P_vent + P_zvol;
    if nargout > 1
params.Q_obol = Q_obol; params.T_ser_obol = T_ser_obol; params.Q_zag_kab = Q_zag_kab;
params.delta_T_pel = delta_T_pel; params.Qc_one = Qc_one; params.COP_pel = COP_pel;
params.P_vent = P_vent; params.P_zvol = P_zvol; params.P_peltier = P_peltier;
    end
end

function [c, ceq] = nonlinearConstraints(x, p)
    % Nonlinear constraint function
V_obol = x(1); n_Pel = x(2); I_Pel = x(3);
    if n_Pel < 0.1, n_Pel = 0.1; end
T_mt = Stull_Twb(p.T_zovn, p.phi_zovn);
T_ohol = p.T_zovn - p.eta_ad * (p.T_zovn - T_mt);
Q_obol = calculate_Q_obol_iterative(p, V_obol, T_ohol);
m_dot_obol = V_obol * p.rho_pov;
    if m_dot_obol < 1e-3, m_dot_obol = 1e-3; end
T_ser_obol = T_ohol + Q_obol / (2 * m_dot_obol * p.cp_pov);
Q_zag_kab = p.U_kab * p.A_kab * (T_ser_obol - p.T_kab_tsil) + p.Q_vn;
Qc_one_req = Q_zag_kab / n_Pel;
T_h_pel = T_ser_obol + p.dT_radiator;
T_c_pel = p.T_kab_tsil - p.dT_holodna;
delta_T_pel = T_h_pel - T_c_pel;
[Qc_possible, ~, ~] = Peltier_Performance_Interpolated(I_Pel, delta_T_pel);
    % Limitation on outlet air temperature
dT_progriv = Q_obol / (m_dot_obol * p.cp_pov);
T_out_heat = T_heat + dT_progriv;

c = [];
c(1) = Qc_one_req - Qc_possible;
c(2) = delta_T_pel - p.TEC_dT_max;
c(3) = T_vyh_obol - p.T_vyh_obol_max;
ceq = [];
end

%% ==
% AUXILIARY FUNCTIONS (PHYSICAL MODELS)
% ==
function [Qc, V, COP] = Peltier_Performance_Interpolated(I, deltaT)
    % EXACT MODEL OF THERMO ELEMENT MT2,6-0,8-263T1S (Th = 50°C)
    persistent Qc_interpolant V_interpolant
    if isempty(Qc_interpolant)
I_grid = [0, 2, 4, 6, 8, 10, 12, 14, 15];
dT_grid = [0, 10, 20, 30, 40, 50, 60, 70, 77];
Qc_data = [0 71 133 186 230 264 288 303 319; 0 52 110 160 201 232 254 267 275; 0 32 87 134 172 201 220
230 235; 0 11 63 107 142 168 185 195 200; 0 0 38 79 112 138 155 165 170; 0 0 0 50 80 105 122 130 135; 0
0 0 0 46 68 85 95 100; 0 0 0 0 0 30 45 55 60; 0 0 0 0 0 0 0 0 0];
V_data = [0 4.5 8.8 13.0 17.0 20.8 24.5 28.0 30.4; 0 4.8 9.4 13.8 17.9 21.8 25.6 29.2 31.7; 0 5.1 10.0
14.7 18.9 23.0 26.8 30.5 33.0; 0 5.4 10.6 15.5 19.9 24.1 28.0 31.8 34.0; 0 5.7 11.2 16.4 21.0 25.3 29.0
32.5 34.5; 0 6.0 11.8 17.2 22.0 26.4 30.1 33.5 35.0; 0 6.3 12.4 18.1 23.0 27.5 31.2 34.5 35.8; 0 6.6
13.0 19.0 24.0 28.5 32.0 35.0 36.5; 0 6.8 13.5 19.5 25.0 29.5 33.0 36.0 37.5];
[X, Y] = meshgrid(I_grid, dT_grid);
Qc_interpolant = griddedInterpolant(X', Y', Qc_data', 'linear' , 'none' );
V_interpolant = griddedInterpolant(X', Y', V_data', 'linear' , 'none' );
    end
Qc = Qc_interpolant(I, deltaT); V = V_interpolant(I, deltaT);
    if isnan(Qc) || Qc < 0, Qc = -1; end
    if isnan(V), V = 100; end
Pel = V * I; if Pel <= 0, Pel = 1e-6; end
COP = Qc / Pel; if COP < 0, COP = 0; end
end

function Twb = Stull_Twb(T, RH)
    % Calculation of wet bulb temperature using Stall's formula
```

```
Twb = T * atan(0.151977 * (RH + 8.313659)^0.5) + ...
atan(T + RH) - atan(RH - 1.676331) + ...
0.00391838 * RH^1.5 * atan(0.023101 * RH) - 4.686035;
end

function Q_obol = calculate_Q_obol_iterative(p, V_obol, T_ohol)
    % Iterative calculation of thermal load on the shell
d_obol = p.d_obol;
T_pov = p.T_zovn - 5;
    for i = 1:10
Q_rad = p.epsilon * p.sigma * p.A_kab * ((p.T_vypr + 273.15)^4 - (T_pov + 273.15)^4);
Q_conv_in = p.h_zovn * p.A_kab * (p.T_zovn - T_pov);
Q_in_total = Q_rad + Q_conv_in;
        % Improved internal convection model
F_kanal = p.W_obol_kanal * d_obol;
hydraulic_diam = 2 * d_obol;
velocity = V_obol / F_channel;
Re = (p.rho_pov * velocity * hydraulic_diam) / p.nu_pov;
        if Re < 2300, Re = 2300; end
Nu = 0.023 * Re^0.8 * 0.71^0.4;
h_vnut_obol = (Nu * p.k_pov) / hydraulic_diam;
R_obol = 1/h_vnut_obol;
Q_out_total = p.A_kab * (T_pov - T_ohol) / R_obol;
error = Q_in_total - Q_out_total;
        if abs(error) < 1, break ; end
T_pov = T_pov + error / (p.h_zovn * p.A_kab + 4*p.epsilon*p.sigma*p.A_kab*(T_pov+273.15)^3 +
p.A_kab/R_obol);
    end
Q_obol = (Q_in_total + Q_out_total) / 2;
    if Q_obol < 0, Q_obol = 0; end
end
```