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DESIGN OF THE MECHANICAL STRUCTURE OF A ROTARY-DISC INCUBATOR AND CALCULATION-BASED JUSTIFICATION OF ITS MAIN COMPONENTS

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Abstract. This article addresses the design of the structural and mechanical system of a rotary-disc incubator intended for the incubation of silkworm eggs and provides a calculation-based justification of its principal components. The geometric dimensions of the working chamber were selected with due consideration of biological and technological requirements, minimization of heat losses, air exchange, and the rational arrangement of egg boxes. The obtained results demonstrate that the proposed design provides compactness, energy efficiency, ease of maintenance, and suitability for operation under humid environmental conditions.

Keywords: incubator, rotary disc, mechanical structure, plywood disc, central shaft, bearing, gear transmission, strength, stiffness, air exchange, energy efficiency.

Annotatsiya. Ushbu maqolada ipak qurti tuxumlarini inkubatsiya qilish uchun mo'ljallangan aylanuvchi diskli inkubatorning konstruktiv-mexanik tizimini loyihalash masalalari ko'rib chiqilgan hamda uning asosiy tarkibiy qismlari hisob-kitoblar asosida asoslab berilgan. Ishchi kameraning geometrik o'lchamlari biologik va texnologik talablar, issiqlik yo'qotishlarini minimallashtirish, havo almashinuvi hamda tuxum qutillarini oqilona joylashtirish imkoniyatlari hisobga olingan holda tanlangan. Olingan natijalar taklif etilgan konstruksiyaning ixchamligi, energiya samaradorligi, texnik xizmat ko'rsatish qulayligi hamda yuqori namlik sharoitida foydalanishga mosligini ko'rsatadi.

Kalit so'zlar: inkubator, aylanuvchi disk, mexanik konstruksiya, fanera disk, markaziy val, podshipnik, tishli uzatma, mustahkamlik, bikrlilik, havo almashinuvi, energiya samaradorligi.

Аннотация. В данной статье рассматриваются вопросы проектирования конструктивно-механической системы ротационно-дискового инкубатора, предназначенного для инкубации яиц тутового шелкопряда, а также приводится расчётное обоснование его основных конструктивных элементов. Геометрические размеры рабочей камеры выбраны с учётом биологических и технологических требований, минимизации тепловых потерь, обеспечения воздухообмена и рационального размещения коробок с яйцами шелкопряда. Полученные результаты показывают, что предложенная конструкция обеспечивает компактность, энергоэффективность, удобство технического обслуживания и пригодность для эксплуатации в условиях повышенной влажности окружающей среды.

Ключевые слова: инкубатор, вращающийся диск, механическая конструкция, фанерный диск, центральный вал, подшипник, зубчатая передача, прочность, жёсткость, воздухообмен, энергоэффективность.

INTRODUCTION

In sericulture, the egg-hatching stage is one of the key technological processes that directly affects the stability of subsequent biological development. In addition to maintaining temperature, relative humidity, and gas composition within specified limits in the incubation chamber, it is essential to ensure that all egg boxes are exposed to a uniform microclimate throughout the chamber volume. Therefore, the mechanical structure of an incubator does not merely serve as an external enclosure; it integrates heat transfer, airflow, load distribution, rotation, and maintenance processes into a single system.

Conventional stationary rack-type designs may exhibit local variations in temperature and humidity, reduced airflow in certain zones, and the need for manual repositioning of boxes by the operator. In a rotary-disc configuration, however, the spatial position of the egg boxes is periodically changed, thereby altering their position relative to the microclimate sources. This approach requires the geometry of the structure, disc stiffness, shaft stability, bearing service life, and the torque-speed characteristics of the drive to be evaluated in an interrelated manner.

The aim of this study is to design the working chamber, disc mechanism, central shaft, bearings, and drive

components of a rotary-disc incubator within a unified calculation model; to determine their principal geometric and strength parameters; and to substantiate the practical feasibility of the proposed structural solution. To achieve this aim, the chamber volume and external surface area, air-renewal time, disc deflection and stress, combined compression-torsion state of the shaft, Euler buckling stability, bearing life, and drive torque were calculated.

LITERATURE REVIEW

The thermal design of incubators is based on heat- and mass-transfer principles described by F.P. Incropera et al. [1] and Y.A. Cengel and A.J. Ghajar [2]. Their works show that chamber geometry, surface area, thermal resistance, and temperature difference directly affect heat losses and energy efficiency.

The mechanical strength of structural components is addressed by J.M. Gere and B.J. Goodno [3], F.P. Beer et al. [4], and S.P. Timoshenko and J.M. Gere [5]. Their methods provide the basis for calculating the frame, rotary disc, shaft, and supporting elements. R.G. Budynas and J.K. Nisbett [7], together with R.L. Norton [12], emphasize the importance of torque, friction, transmission efficiency, and design margin in drive selection.

N.A. French [8] and E. Decuyper and H. Michels [9] investigated the influence of incubation temperature and incubator design on embryonic development. W. Bolton [11] considered mechanical, electrical, and control subsystems as an integrated mechatronic system. These studies provide the theoretical basis for the calculation-based design of a rotary-disc incubator and its main mechanical components.

RESEARCH METHODOLOGY

Scientific and engineering literature on incubator design primarily focuses on stable microclimate control, heat and mass transfer, ventilation, and the reliability of structural components. Heat-transfer theory shows that heat losses from an enclosed chamber depend on its surface area, the thermal resistance of the enclosure layers, and the temperature difference between the chamber and the ambient environment [1, 2]. From this perspective, reducing the surface-area-to-volume ratio among geometries of equal volume is an important design factor for lowering energy consumption.

In mechanical design calculations, strength of materials and machine-element theory provide the fundamental methodological basis. The analysis of beam and plate deflection, second moment of area, normal and shear stresses, strength theories, and Euler critical loads for compressed members enables the safe dimensioning of structural components [3–5]. For rotating components, consideration of the moment of inertia and acceleration torque is also essential when selecting the drive.

Standard calculations for bearing supports and gear transmissions employ dynamic load ratings, equivalent loads, and nominal service-life indicators [6]. Although the required mechanical power in low-speed mechatronic systems is often small, starting torque, friction, transmission efficiency, and an adequate design margin are decisive factors in motor selection [7]. Studies of automated incubation systems have devoted considerable attention to sensors and control algorithms; however, solutions in which mechanical geometry, a rotating platform, and microclimate requirements are jointly substantiated by engineering calculations have received comparatively limited attention [8, 9].

Accordingly, this study considers the mechanical subsystem not as a collection of isolated components, but as a basic structural platform for the thermal, ventilation, and automatic control systems. In this framework, geometric parameters are linked to biological capacity and air exchange, disc and shaft dimensions to structural strength, and the drive system to the required rotational regime and energy efficiency.

The study employed calculation-and-design, analytical, and engineering-comparison methods. The initial design conditions specified the placement of 12 egg boxes on a three-tier rotary mechanism inside the chamber, the maintenance of unobstructed airflow around the boxes, and an overall structure height of approximately 1 m. The side length of the working chamber was set to $a = 0.8$ m. The cubic geometry was selected because it reduces the external surface area for a given volume, limits heat losses, and allows the rotary mechanism to be arranged symmetrically.

$$V = a^3 = 0,8^3 = 0,512 \text{ m}^3 \quad (1)$$

$$S = 6a^2 = 6 \cdot 0,8^2 = 3,84 \text{ m}^2 \quad (2)$$

At a fan capacity of $Q = 0.76 \text{ m}^3/\text{min}$, the theoretical air-renewal time in the chamber was determined as the ratio of the working volume to the airflow rate:

$$t = \frac{V}{Q} = \frac{0,512}{0,76} = 0,67 \text{ min} = 40 \text{ s} \quad (3)$$

The rotary disc diameter was selected as 780 mm, corresponding to a radius of $R = 0.39$ m. Within an 800 mm internal chamber width, this leaves an approximately 10 mm technological clearance on each side. The spacing between the three tiers was set to $H = 0.21$ m, giving an active disc-stack height of approximately 0.42 m. Four egg boxes are placed on each disc.

Disc stiffness was evaluated using a conservative calculation scheme in which a radial sector was modeled as a cantilever beam fixed at the center. For one egg box, a mass of $m_q = 0.04$ kg and a load factor of $k = 1.5$ were adopted, resulting in a design force of $P \approx 0.59$ N. The distance from the rotation axis to the center of the box was $L = 0.20$ m, the effective sector width was $b = 0.15$ m, and the elastic modulus of plywood was $E = 7 \times 10^9$ Pa.

$$I = bh^3/12 \quad (4)$$

$$\delta = PL^3/(3EI) \quad (5)$$

The disc thickness was determined from the allowable-deflection condition $\delta_{\text{allow}} = 0.8$ mm, and the standard value $h = 5$ mm was adopted. In the strength verification, the maximum bending stress at the cantilever section was evaluated. For the central shaft, axial compression, torsion, and equivalent stress according to the distortion-energy criterion were calculated. The adopted parameters were $d = 15$ mm, $L = 0.82$ m, $E = 2 \times 10^{11}$ Pa, and $W = 60$ N.

$$\sigma_c = \frac{W}{A} = \frac{60}{1.77 \cdot 10^{-4}} = 0.34 \cdot 10^6 \text{ Pa} = 0.34 \text{ MPa} \quad (6)$$

$$\tau = \frac{16 T}{\pi d^3} = \frac{16 \cdot 1}{\pi \cdot 0.015^3} = 1.51 \cdot 10^6 \text{ Pa} = 1.51 \text{ MPa} \quad (7)$$

$$\sigma_{ekv} = \sqrt{\sigma_c^2 + 3\tau^2} = \sqrt{0.34^2 + 3 \cdot 1.51^2} = 2.6 \text{ MPa} \quad (8)$$

$$P_{kr} = \frac{\pi^2 EI}{(KL)^2} = \frac{\pi^2 \cdot 2 \cdot 10^{11} \cdot 2.49 \cdot 10^{-9}}{(0.82)^2} = 7.3 \cdot 10^3 \text{ N} = 7.3 \text{ kN} \quad (9)$$

The drive was calculated for the condition that the disc rotates through 180° within 15 s. The operating disc speed was taken as $n_d = 2$ rpm, the angular velocity as $\omega = 0.209$ rad/s, the acceleration time as $t_a \approx 2$ s, and the angular acceleration as $\varepsilon \approx 0.105$ rad/s². The moment of inertia of the rotating system was $J = 0.34$ kg·m². The frictional resistance was conservatively assumed to be $\tau_{\text{load}} \approx 0.18$ N·m. For the gear pair, $z_1 = 20$, $z_2 = 50$, the transmission ratio was $i = 2.5$, and the efficiency was $\eta \approx 0.9$.

$$\tau_{\max} = 0.036 + 0.18 = 0.22 \text{ N} \cdot \text{m} \quad (10)$$

$$T_m = \frac{\tau_{\max}}{i \cdot \eta} = \frac{0.22}{2.5 \cdot 0.9} = 0.09 \text{ N} \cdot \text{m} \quad (11)$$

$$P = \tau_{\max} \cdot \omega = 0.22 \cdot 0.209 = 0.04 \text{ Vt} \quad (12)$$

ANALYSIS AND RESULTS

The calculations enabled the incubator design parameters to be coordinated with one another. The working chamber has a volume of 0.512 m³ and an external surface area of 3.84 m² and accommodates a total of 12 egg boxes on the three-tier disc assembly. A floor area of 0.6 m² was designated as the working surface for the infrared heating element. Based on the fan capacity, the theoretical air-renewal time was approximately 40 s.

The general configuration and principal dimensions of the structure are shown in Figure 1. The symmetrical arrangement of three discs along the central shaft simplifies axial load transfer and contributes to maintaining structural balance during rotation.

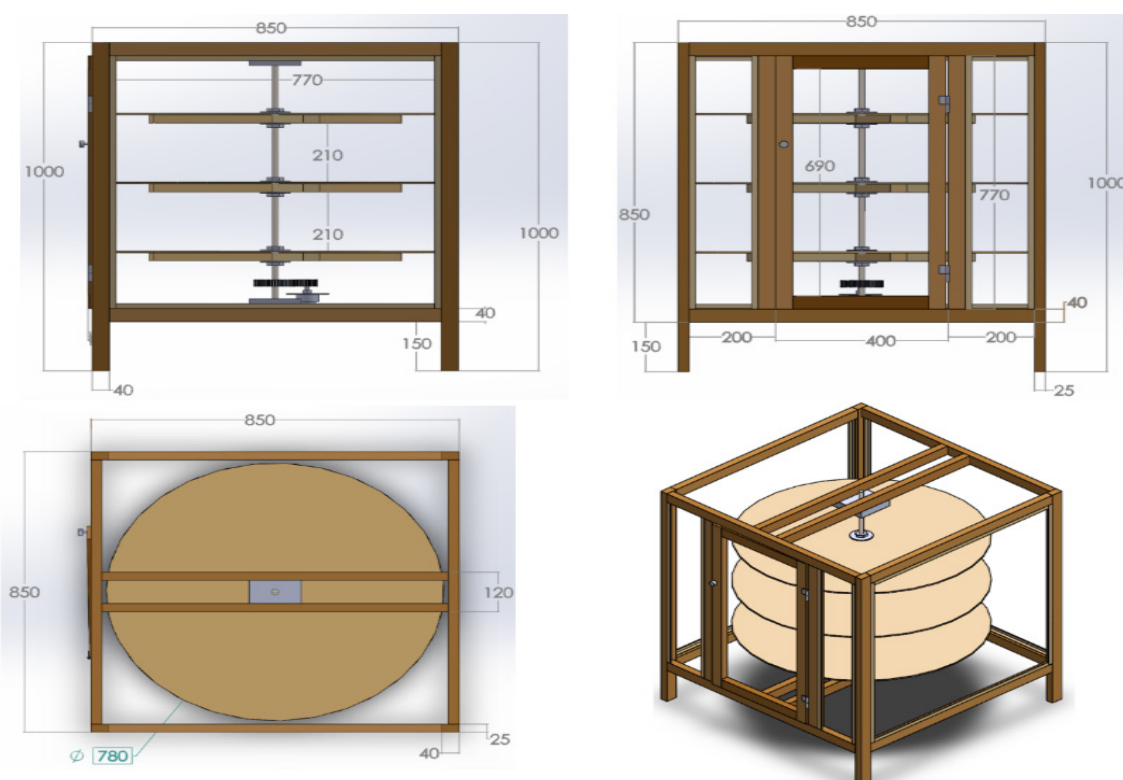


Figure 1. General configuration and principal dimensions of the incubator structure.

Table 1. Main Design Parameters of the Incubator

Parameter	Symbol	Value
Working chamber side length	A	0.8 m
Working volume	V	0.512 m ³
External surface area	S	3.84 m ²
Heater area	S _p	0.6 m ²
Overall height	H	≈1.0 m
Air-renewal time	T	≈40 s

For the disc mechanism, $R = 0.39$ m and $H = 0.21$ m were adopted. The calculated maximum deflection of the 5 mm plywood disc was 0.14 mm, approximately 5.7 times lower than the allowable value of 0.8 mm. The maximum stress was 0.19 MPa, which is far below the adopted allowable stress of 20 MPa for plywood. Thus, the stiffness criterion, rather than the strength criterion, governed the selection of disc thickness.

Table 2. Main Parameters of the Rotary Disc

Parameter	Value	Assessment
Disc radius	0.39 m	Fits inside the chamber with technological clearance
Number of tiers	3	12 boxes in total
Inter-tier spacing	0.21 m	Sufficient for airflow
Disc thickness	5 mm	According to the stiffness criterion
Maximum deflection	0.14 mm	< 0.8 mm
Maximum stress	0.19 MPa	< 20 MPa
Mass of one disc	1.39 kg	Low inertia

The total axial load on the central shaft was assumed to be $W = 60$ N. For a shaft diameter of $d = 15$ mm, the compressive stress was 0.34 MPa, the torsional stress was 1.51 MPa, and the equivalent stress was 2.6

MPa. This value is approximately 37 times lower than the allowable stress of 100 MPa. The Euler critical load was 7.3 kN, approximately 122 times greater than the actual load, indicating a high safety margin against axial buckling.

UCF 202 housed bearings with a $\varnothing 15$ mm bore were selected for the upper and lower shaft supports. Since the applied load of approximately 60 N is very small relative to the dynamic load rating $C \approx 7.65$ kN, bearing life is not limited by load-carrying capacity. The principal reasons for this selection are ease of installation, standardized dimensions, and compatibility with the central shaft.

For the drive system, the transmission ratio $i = 2.5$ enables a disc speed of 2 rpm while increasing the available torque. The calculated disc torque is approximately 0.20-0.22 N·m, whereas the required torque referred to the motor shaft is in the range of 0.09-0.10 N·m. The required mechanical power is approximately 0.04-0.05 W. If the disc rotates by 180° for 15 s once every hour, the drive duty cycle is approximately 0.42%. This operating regime reduces the energy consumption of the mechanical subsystem and alleviates the thermal loading of the motor.

The results indicate that the proposed structure can be implemented without excessive metal consumption or a high-power drive. The cubic working chamber with a side length of 0.8 m maintains a relatively low surface-area-to-volume ratio while providing sufficient radial clearance for the 780 mm disc. From a structural standpoint, this clearance prevents contact between the disc and the chamber walls, compensates for assembly tolerances, and permits airflow through the peripheral zones.

The disc-thickness results lead to an important methodological conclusion: under small loads, the governing factor may be the geometric deformation limit rather than the material failure or bending-stress limit. Although the stress of 0.19 MPa in the 5 mm plywood disc is very low, the deflection criterion was evaluated separately to ensure structural accuracy and level positioning of the egg boxes. Because the actual disc is spatially stiffer than the simplified cantilever-beam model, the adopted calculation scheme is conservative and therefore provides an additional safety margin.

The central-shaft calculation likewise demonstrated a substantial strength margin. Nevertheless, static strength alone is insufficient for practical operation. Uneven loading of the discs may produce eccentricity, thereby increasing radial forces and vibration. Consequently, symmetrical placement of the boxes on the discs, verification of shaft verticality during installation, and adjustment of the center distance of the gear pair are necessary to ensure long-term structural reliability.

The UCF 202 bearings have a load capacity far exceeding the calculated load. Under low-speed operation, their service life is therefore more dependent on the humid environment, lubrication condition, corrosion, and installation accuracy than on load-carrying capacity. Accordingly, the use of sealed bearings and protection of metal surfaces exposed to moisture is advisable.

Material selection was coordinated by considering the low thermal conductivity and affordability of the wooden frame, the low mass of plywood, the transparency of acrylic or polycarbonate, the strength of the steel shaft, and the quiet operation of POM/nylon gears. At a relative humidity of 75-85%, wooden and plywood surfaces must be protected with a moisture-resistant, hygienic coating. This measure not only extends service life but also limits significant changes in the elastic properties assumed in the calculations.

The very low duty cycle of the drive is advantageous in terms of energy efficiency. However, selecting the motor solely on the basis of the calculated mechanical power of 0.04 W would be inappropriate: practical motor selection must include allowances for gearbox friction, starting torque, voltage drop, gear-transmission losses, and the possibility of mechanical jamming. Therefore, the rated torque of the geared DC motor should be substantially higher than the calculated requirement of 0.09-0.10 N·m.

At the next stage, the proposed structure can serve as a mechanical platform for integrating temperature, humidity, and CO_2 sensors, an infrared heating element, a fan, and a microcontroller-based control system. Such sequential integration of the mechanical and control subsystems will enable experimental optimization of the technological parameters of the device.

CONCLUSION AND SUGGESTIONS

The study established that a cubic working chamber measuring $0.8 \times 0.8 \times 0.8$ m, with a volume of 0.512 m^3 and an external surface area of 3.84 m^2 , is structurally appropriate for the proposed rotary-disc incubator. The use of a three-tier disc mechanism with a diameter of 780 mm enables the placement of 12 egg boxes in total, while the 210 mm spacing between tiers provides unobstructed airflow and convenient arrangement of structural components. Strength calculations showed that the 5 mm plywood disc has a maximum deflection of 0.14 mm and a maximum stress of 0.19 MPa, whereas the $\varnothing 15$ mm central shaft has an equivalent stress of 2.6 MPa and an Euler critical load of 7.3 kN, providing adequate strength and stability under the 60 N operating load. The selection of UCF 202 bearings was justified by structural compatibility and ease of installation. In

addition, a gear transmission with $i = 2.5$, a disc rotational speed of 2 rpm, and an approximately 0.2 N·m driving torque provide the required operating regime; performing a 15 s rotation once per hour results in a duty cycle of only 0.42%, thereby contributing to the energy efficiency of the system. The selected materials and assembly configuration support reliable operation in a humid environment, hygienic cleaning, corrosion resistance, and ease of maintenance. Overall, the results provide an adequate mechanical basis for integrating electronic control, microclimate monitoring, and automation systems into a unified rotary-disc incubator design.

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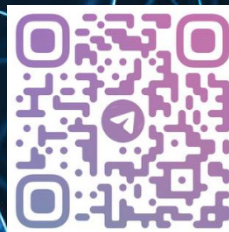
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