Scientific Papers. Series E. Land Reclamation, Earth Observation & Surveying, Environmental Engineering. Vol. XI, 2022
Print ISSN 2285-6064, CD-ROM ISSN 2285-6072, Online ISSN 2393-5138, ISSN-L 2285-6064

THEORETICAL RESULTS BY DETERMINING THE POWER OF THE DRIVEVIBRO-OIL PRODUCER

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Abstract

A vibratory oil maker to produce environmentally friendly butter is presented. A constructive scheme and design of a butter maker with a churning mechanism in the form of a flexible vibration drive (a membrane that is also the bottom of the container, which performs periodic oscillatory movements by means of a crank mechanism) has been developed, which reduces the energy intensity of churning and loss of butter due to its sticking to the churning mechanism. A force analysis is presented, in which the forces acting in the oil maker during its operation are considered and a formula is obtained for determining the power to drive the vibratory oil maker, considering the division of masses into rotational and reciprocating masses of the vibratory oil maker knocking down mechanism. The power for the drive of the vibratory oil maker was calculated considering the change in the angle of rotation of the crank φ = 0...360 degrees. The obtained values are presented as a graph of the dependence of the power on the drive on the angular velocity and radius of the crank at given angles of rotation of the crank. The maximum (peak) value of power per drive is determined - 125 W. At the same time, the energy intensity of butter churning of the vibratory oil maker was Es = 3.84 Wh/kg with a productivity of Qm = 11.25 kg/h, and the degree of use of milk fat S = 99.6%, which corresponds to the waste of fat into buttermilk 0.4% and does not exceed the requirements of GOST. The maximum (peak) value of power per drive is determined - 125 W. At the same time, the energy intensity of butter churning of the vibratory oil maker was Es = 3.84 Wh/kg with a productivity of Qm = 11.25 kg/h, and the degree of use of milk fat S = 99.6%, which corresponds to the waste of fat into buttermilk 0.4% and does not exceed the requirements of GOST. The maximum (peak) value of power per drive is determined - 125 W. At the same time, the energy intensity of butter churning of the vibratory oil maker was Es = 3.84 Wh/kg with a productivity of Qm = 11.25 kg/h, and the degree of use of milk fat S = 99.6%, which corresponds to the waste of fat into buttermilk 0.4% and does not exceed the requirements of GOST.

***Key words**: butter, crank mechanism, environmentally friendly, membrane, vibration drive.*

INTRODUCTION

Much attention has been paid to the research of batch oil producers (Melken, 1991; Yashin et al., 2017-2021). This is evidenced by the analysis of literary sources, reviewing various methods for obtaining oil grains, based on which a wide variety of oil producers was obtained. Of greatest interest are the designs of batch oil makers, in which the working bodies are made in the form of a rotating container. Such designs are less perfect, since churning is carried out for a long time from 30 to 120 minutes, because of which the productivity of the oil maker decreases and power costs increase (Shumaev et al., 2020; Shumaenv et al., 2021).

Buttermakers with rotating working bodies eliminate the existing shortcomings of the previously described designs, however, the waste of fat into buttermilk increases and ranges from 1 to 3%.

Relevant and practically significant for the agro-industrial complex is the problem of reducing the energy intensity of churning butter and increasing the degree of fat utilization, which is solved by improving the designs of butter makers.

To eliminate existing shortcomings, such as increased losses of butter due to its sticking to the churning mechanism and increased churning energy intensity, a butter maker design was developed with a churning mechanism, which is a membrane, which is also the bottom of the container. The membrane performs periodic oscillatory movements by means of a crank mechanism and an electric motor (Parfenov, 2014; Patents 2012, 2017).

MATERIALS AND METHODS

The developed design of the vibratory oil maker (Figure 1) works as follows. The cylindrical container 9 is filled with cream through the open filling hole 4. After that, the hole 4 is closed with a shutter 5 and locked with a lock 6. The electric motor 14 is started, because of which the crank 13 and the connecting rod 12 transmit periodic vibrations to the elastic funnel-shaped membrane 10, and from it to the cream itself.

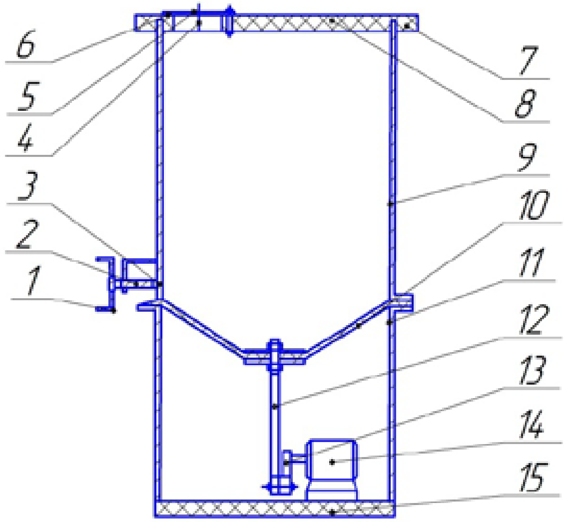

Figure 1. General view of the vibratory oil maker:
1 - handle; 2 - rod; 3 - unloading valve; 4 - loading window; 5 - damper; 6 - retainer; 7 - clamp; 8 - cover;
9 - container; 10 - membrane; 11 - frame; 12 - connecting rod; 13 - crank;
14 – electric drive; 15 - vibration damper

Intensive mixing is carried out under the action of a vibrational impulse, which contributes to the emergence of a turbulent movement of the cream. As a result, the process of forming butter grains is accelerated, the time for churning butter is significantly reduced, productivity increases and energy consumption decreases.

Visualization of the oil grain formation process is carried out by means of a cover 8 made of a transparent material. The unloading of the finished oil layer is carried out by opening the discharge valve 3, which also serves to remove excess moisture from the oil layer. Its opening occurs because of rotation of the handle 1 of the rod 2.

As a result, the motor 14 is turned off. If the oil grain needs to be refined to the desired consistency, then it is necessary to reduce the oscillation frequency of the flexible vibration drive, or use an oil homogenizer, if available.

RESULTS AND DISCUSSIONS

To determine the drive power of a vibratory oil maker, it is necessary to consider the forces acting during its operation (Figure 2).

Let us assume that container 5 is filled with a certain volume of cream with a height H.

The knocking mechanism consists of a membrane 4 with a rigid center 3, a connecting rod 2 and a crank 1.

The pressure force on the knocking mechanism acts vertically downwards, and its modulus is determined by the formulas 1, 2, 4, 9 and 10:

$$F_p = p \cdot \underline{Sef}., \text{H} \tag{1}$$

where:

- p - pressure on the membrane with a rigid center, Pa;
- $S_{ef.}$ - effective area of the membrane with a rigid center, m^2.

Print ISSN 2285-6064, CD-ROM ISSN 2285-6072, Online ISSN 2393-5138, ISSN-L 2285-6064

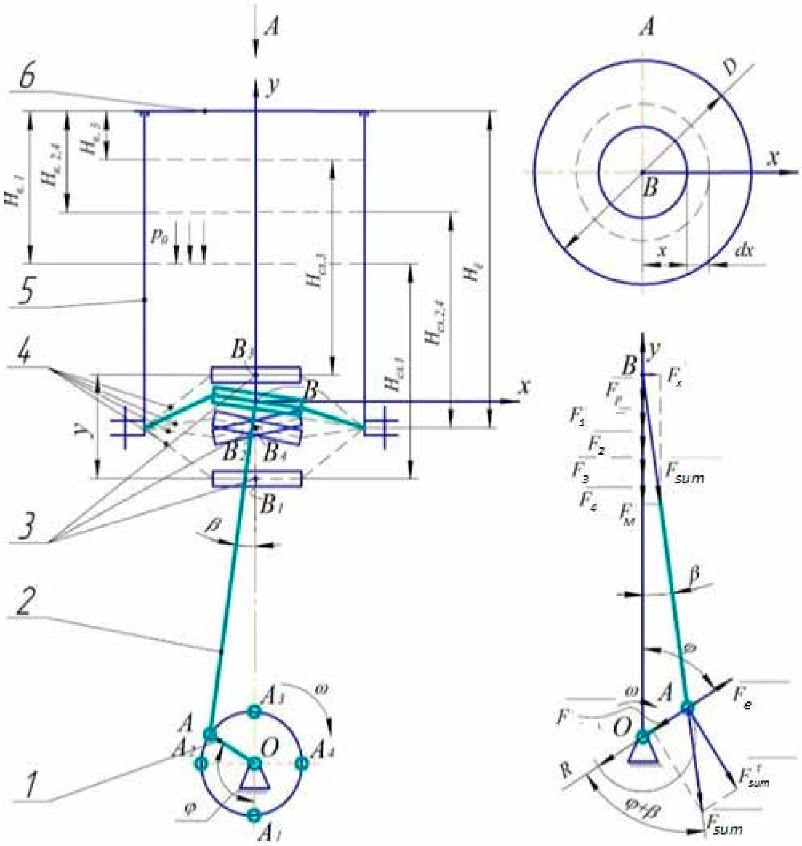

Figure 2. Scheme for determining the required power for the drive of the vibratory oil maker: a) - scheme of the vibratory oil maker for determining the pressure force at point B of the rigid center; b) - view A of the scheme of the vibratory oil maker from above with the cover removed to determine the force of internal friction of the cream; c) - diagram of the forces acting in the vibratory oil maker: 1 - crank (F_1); 2 - connecting rod (F_2); 3 - hard center (F_3); 4 – membrane (F_4); 5 - capacity; 6 - cover

In this case, the effective area of a membrane with a rigid center is the area, the value of which characterizes the ability of the membrane to convert pressure into axial force 1, 2, 4, 9, 10:

$$S_{ef.} = \frac{\pi}{12}(D^2 + Dd + d^2)\text{, m}^2 \qquad (2)$$

where:

- D - membrane (tank) diameter, m;
- d -rigid center diameter, m

In order to determine the magnitude of the forces of inertia that are formed during the movement of the parts of the crank mechanism, it is important to determine the corresponding masses. To this end, we replace the real masses of the moving parts with a system of masses dynamically equivalent to the real system. Since the connecting rod performs a complex plane-parallel motion, we would replace its mass with three masses concentrated at the point of attachment of the connecting rod to the membrane with a rigid center m_{1con}, reciprocating with respect to the crank m_{2con}, performing rotational motion, as well as in the center of mass of the connecting rod m_{3con} performing plane-parallel motion. As a result of this, it becomes clear that the mass of the connecting rod at the center of mass m_{3con} is negligible compared to the indicated masses. To move to a dynamically equivalent system, several restrictions must be observed, such as:

– the sum of all masses is equal to the mass of the connecting rod:

$$m_{con} = m_{1con} + m_{2con}, \text{ kg} \quad (3)$$

where:

- m_{1con} -the mass of the connecting rod concentrated at the place of its attachment to the membrane with a rigid center, kg;
- m_{2con} -the mass of the connecting rod concentrated at the place of its attachment to the crank, kg.

– with the center of gravity of all masses must coincide with the center of gravity of the connecting rod:

$$m_{1con} \cdot l_1 = m_{2con} \cdot (L - l_1) \quad (4)$$

where:

- l_1-distance from the center of mass of the connecting rod to the place of its attachment to the membrane with a rigid center, m;
- L -connectingrodlength, m

– the sum of the moments of inertia of all forces about the axis passing through the center of gravity of the connecting rod:

$$I_{сцm} = m_{1con} \cdot l_1^2 + m_{2con} \cdot (L - l_1)^2, \text{ kgm} \quad (5)$$

– the masses must be located on one straight line passing through the center of gravity of the connecting rod.

Let us determine by the formula the mass of the connecting rod concentrated at the place of its attachment to the membrane with a rigid center m_{1con} and the mass of the connecting rod, concentrated in the place of its attachment to the crank m_{2con}:

$$m_{1con} = m_{con} \cdot \frac{L - l_1}{L}, \text{ kg} \quad (6)$$

$$m_{2con} = m_{con} \cdot \frac{l_1}{L}, \text{ kg} \quad (7)$$

Then the third constraint, formula (5), considering (6) and (7) will take the following form:

$$I_{con} = m_{con} \cdot l_1^2 \cdot \left(1 - \frac{l_1}{L}\right) + m_{con} \cdot \frac{l_1 \cdot (L - l_1)^2}{L}, \text{ kgm} \quad (8)$$

In this case, the restriction is not satisfied, since the mass of the connecting rod is replaced by two masses, i.e. to obtain a dynamically substituting system, it would be necessary to the system of masses m_{1con}, m_{2con} add a negative moment of inertia. Since it has an insignificant impact, it can be excluded from further consideration.

Let us determine the mass of the connecting rod concentrated at the place of its attachment to the membrane with a rigid center m_{1con} and the mass of the connecting rod, concentrated in the place of its attachment to the crank m_{2con} taking into account the formulas (6), (7) and with respect $\frac{l_1}{L} = \frac{2}{3}$ according to the formulas:

$$m_{1con} = \frac{1}{3} \cdot m_{con}, \text{ kg} \quad (8)$$

$$m_{2con} = \frac{2}{3} \cdot m_{con}, \text{ kg} \quad (9)$$

The force of inertia of the reciprocating masses is directed opposite to the movement of the listed elements. It includes the mass of the membrane, the mass of the rigid center, the mass of the connecting rod concentrated at the point of its attachment to the membrane with a rigid center. Let us determine the module of the inertia force of the reciprocating motion of the masses by the formulas 1, 2, 4, 9, 10:

$$F_м = m_м \cdot \ddot{y}, \text{ H} \quad (10)$$

where $m_м$ - mass of reciprocating moving elements, kg;

The mass of reciprocating elements is defined as their sum

$$m_м = m_{мем} + m_c + m_{1con}, \text{ kg} \quad (11)$$

where:

- $m_{мем}$ is the mass of the membrane, kg;
- m_c is the mass of the rigid center, kg;
- m_{1con} - the mass of the connecting rod concentrated at the place of its attachment to the membrane with a rigid center, kg.

Taking into account the formulas (8), (10) And (eleven) in the final form, the inertia force of reciprocating masses will be found by the formula:

$$F_м = (m_{мем} + m_c + \frac{1}{3} \cdot m_{con}) \cdot R \cdot \omega^2 \cdot (\cos\varphi + \lambda \cos 2\varphi), \quad (12)$$

Centrifugal force of inertia of rotating masses. It includes the mass of the axis of the crank and the mass of the connecting rod concentrated at the point of its connection with the crank is directed along the radius of the crank. We find its modulus by the formulas 1, 2, 4, 9, 10:

$$F_e = m_e \cdot a_{c.ac.}, \text{ N} \tag{13}$$

where:

- m_e - mass of elements involved in rotational motion, kg;
- $a_{c.ac.}$ - centrifugal acceleration of rotating elements, m/s^2.

The power to drive the oil maker is the product of the torque, the angular velocity of the crank, and the safety factor:

$$N = \kappa_p \cdot F \cdot R \cdot \omega \cdot \frac{\sin(\varphi + \beta)}{\cos\beta}, \text{ W} \tag{14}$$

where κ_p – power reserve factor, considering power costs for idling.

Considering the equations (13), (14) power to drive the oil maker with a power factor $\kappa_p = 1{,}2$ in final form could be found by the formula 15.

To determine the maximum power for the drive of the oil maker, we equate to zero the right side of the formula (15). Next, you need to determine the derivative of this function with respect to φ. Then you need to calculate the critical points. As a result, the resulting function after differentiation will be equal to zero or will not exist. Therefore, it is necessary to consider the range of real values.

$$\begin{aligned} N = \kappa_p \cdot \Bigg(& \frac{\pi \cdot g}{12} \cdot (D^2 + D \cdot d + d^2) \cdot \left(\rho_в \cdot R \cdot \left((1-\cos\varphi) + \frac{\lambda}{4} \cdot (1 - \cos 2\varphi)\right) - \rho \cdot H\right) + \\ & + \frac{4 \cdot E_{мем} \cdot h_{мем} \cdot R \cdot \left((1-\cos\varphi) + \frac{\lambda}{4} \cdot (1-\cos 2\varphi)\right)}{3 \cdot (1 - \mu_{мем}) \left(\frac{(D^2 - d^2)}{4 \cdot \pi} - \frac{D^2 \cdot \ln^2 \frac{D}{d}}{\pi \cdot \left(\frac{D^2}{d^2} - 1\right)} \right)} + \\ & + \frac{\pi}{12} \cdot \rho \cdot H \cdot R \cdot \omega^2 \cdot (D^2 + D \cdot d + d^2) \cdot (\cos\varphi + \lambda \cdot \cos 2\varphi) + \\ & + (m_{мем} + m_c + \frac{1}{3} \cdot m_{con}) \cdot R\omega^2 \cdot (\cos\varphi + \lambda \cos 2\varphi) \Bigg) \times \\ & \times R \cdot \omega \cdot \left(\sin\varphi + \frac{\lambda \cdot \sin\varphi}{2 \cdot \sqrt{1 - \lambda^2 \cdot \sin^2\varphi}} \right), \text{ N} \end{aligned} \tag{15}$$

In order to identify the sign of the derivative of the function, it is necessary to study the intervals to the right and left of the obtained points. Since we are trying to get the maximum value, we need to investigate only intervals with a positive sign of the derivative of the function or with a value equal to $+\infty$. The found function is difficult to conduct research and determine the extremum.

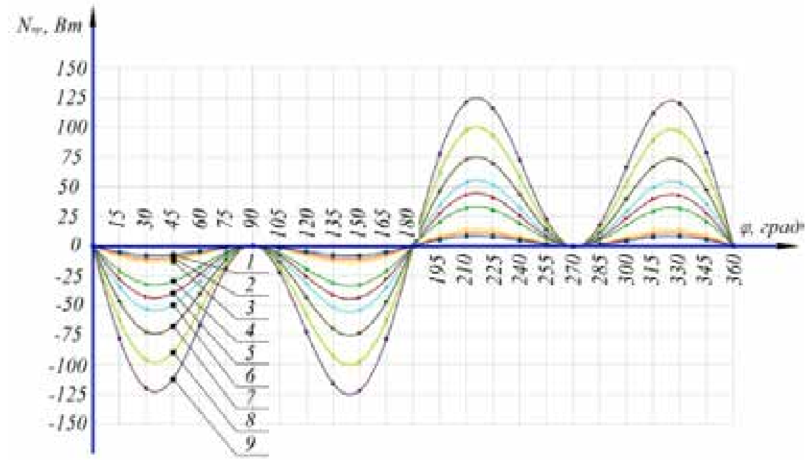

Figure 3. Graph drive power dependencies N from angular velocity ω and crank radius R at the angle of rotation crank φ
1 - at $R = 0{,}003$ m and $\omega = 30$ s–1; 2 - at $R = 0{,}003$ m and $\omega = 40$ s–1;3 - at $R = 0{,}003$ m and $\omega = 50$ s–1;
4 - at $R = 0{,}006$ m and $\omega = 30$ s–1; 5 - at $R = 0{,}006$ m and $\omega = 40$ s–1; 6 - at $R = 0{,}006$ m and $\omega = 50$ s–1;
7 - at $R = 0{,}009$ m and $\omega = 30$ s–1; 8 - at $R = 0{,}009$ m and $\omega = 40$ s–1; 9 - at $R = 0{,}009$ m and $\omega = 50$ s–1

Therefore, we set ourselves constant values and perform the calculation taking into account the change in the angle of rotation of the crank $\varphi = 0...360$ deg. We write the results obtained in the form of a graph $N = f(\varphi)$

(Figure 3). And then, when designing the drive, you can already determine the maximum value of the required power for the drive of the oil maker.

CONCLUSIONS

As a result of theoretical studies, a dependence was revealed to determine the power to drive an oil maker with a flexible vibration drive (15). The performed calculation made it possible to determine the values depending on the change in the design, kinematic and technological parameters. In the considered range of variation of factor values, the maximum (peak) value of power per drive is determined - 125 W.
Considering the results obtained, an AIRE 56V4 electric motor and a SG 62 gearbox were used to drive the oil maker with a flexible vibration drive.

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