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Theoretical Investigation of Separator Units in Saw-gin Machines Part II – Dynamic Behaviour

Abstract
This theoretical study aims to investigate the dynamic behaviour of saw-gin ginning machines with special reference to separator units. For this purpose, the relationship between the driving unit and separator is outlined; optimum running conditions were obtained by introducing non-dimensional parameters defining the dynamic behaviour of the separator. Here, the cotton flow-rate estimation results and the dynamic behaviour of the separator are discussed.

Key words: *ginning, saw-gin, separator units, dynamic behaviour, flow rate.*

Introduction

Ginning processes are important in determining cotton fibre quality for the production of cotton yarn; in general, saw-gin and roller-gin machines are used for ginning seed cotton in industrial applications. The working principle of these machines is different from each other. The difference is basically as foll-ows; cotton fibre is separated by the saw in saw-gin machines, while it is separated by a roller in roller-gin machines. The processed cotton is named after these machines, saw-gin cotton or roller-gin cotton. In general, the structure of the roller-gin machine is rather simple. However, its labour expenses are much greater than those of saw-gin machines. Saw-gin machines are more complex than roller-gin machines [1].

Harmancıoğlu's research demonstrated that saw-gin cotton was more suitable for spinning coarse yarn, and roller-gin cotton was suitable for finer yarns. These two ginning processes have little effect on the strength of the yarn [1]. The roller gin is a slow working machine and can process 60 to 80 kg of seed cotton per hour. However, saw gins can process 600-800 kg per hour. Roller gin machines have minimum pre-cleaning equipment, and the lint contains leaf and trash. Saw gin machines, on the contrary, need a great deal of pre-cleaning equipment for seed cotton and lint cleaners after cleaning [2].

A series of metal ribs is used; the saws perform between them with a narrow gap. The clearance should permit seeds to pass through and admit cotton fibres to be circulated around. A seed-roll box, which is a cylindrical hopper, is placed above the saws. Seed cotton is fed into the seed roll, which holds the continuous supply of cotton while ginning, and the saw rotates opposite to the cotton feeding direction [3]. The units working in a selected saw-gin mill are illustrated in Figure 1.

Unit 1 is the separator unit that carries the seed cotton to the saw-gin mill by a vacuum (air suction) from the store. The other sections or elements of the mill handled are given in sequence on the figure. The seed cotton is transferred to the mill by vacuum created through an aspirator found in the store. Since it is necessary to separate the air-cotton blends, the cotton, which is carried by vacuuming, firstly is transferred to the separator. Thus, the air and dust are separated from the seed cotton. Separated dust is carried to the cyclone, and the seed of the cotton is carried to the seed-cleaners by a vacuum under the separator [4].

Figure 2 represents the two units of the separator. The first unit is the first effective section for cleaning the impurities and dust from seed cotton in saw-gin machines. The second unit compensates the flow of seed cotton between the first unit of the separator and the pre-cleaning equipment. The first unit consists of a curved screen and rotating vanes fitted onto a rotor that has a certain running speed.

Extended definitions and the literature reviewed regarding saw-gin ginning machines and the ginning process have been given in our previous paper, and the effect of geometrical parameters on flow rate and the appropriate design suggestions were also investigated in the previous part of this research [5]. In this paper, the dynamic behaviour of the first unit of the separator is investigated. For this purpose, the effects of radius, angular acceleration and velocity of vanes, and the mass moment inertia of the separator on the mass of seed cotton carried in the first unit of the separator are examined by developing a dynamic model. Also, the flow rate estimation regarding the thickness of vanes and operating conditions was carried out in order to predict the relationship between the dimensions and output flow rate of the second unit of the separator.

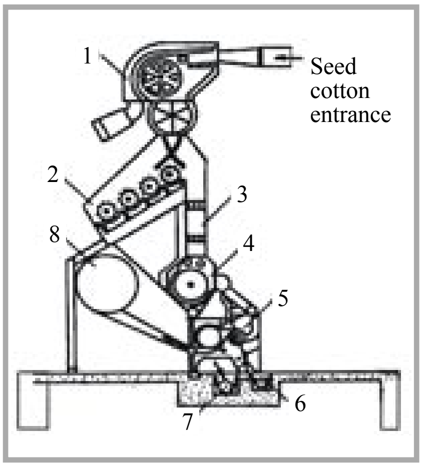

Figure 1. *A typical saw-gin mill and main units [4]; 1 - separator units, 2 - inclined pre-cleaner, 3 - transferring pre-cleaned seed cotton, 4 - feeding to the saw-gin, 5 - saw-gin ginning machine, 6 - helical carrierof burrs, 7 - seed carrying helical unit, 8 - cleaning unit of ginned cotton.*

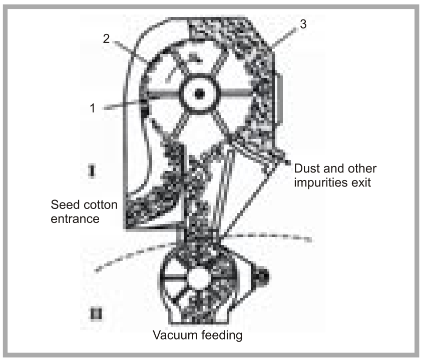

Figure 2. *Units of a separator [5]; 1 - revolving wiper rods, 2 - seed cotton, 3 - perforated curved screen.*

Dynamic behaviour of the separator

Cotton flow rate estimation

The cotton flow rate in the separator units is an important parameter for determining the optimum working conditions. In the first paper, a theoretical model on the separator units in the saw-gin machine has been developed by introducing the non-dimensional parameters defining the mechanical operating conditions, and a design procedure whereby the cotton flow rate and loss of cotton was outlined [5]. In this study, the volumetric flow rate in terms of the non-dimensional parameters regarding the thickness of vanes has been obtained. The volumetric flow rate of the second unit is designated as Q'_2 in this approach, while it is defined as Q_2 in the former part of this research [5].

Figure 3 represents a simple theoretical model for the second unit of the separator (II in Figure 2). In practice, the vane is rectangular-shaped. However, in this theoretical model, the vane shape is considered as a conical or angular segment, in order to maintain the satisfactory approach seen in the figure. The second unit of the separator consists of *z* pieces of vanes, and the inner and outer radius of this unit are defined as r_{i_2} and r_{o_2} respectively; b_2 is the width of a vane; β is the segment angle corresponding to a vane. The volume of the vanes forms the unused volume for carrying seed cotton.

The volume of a vane fitted on the rotating member of the unit can be calculated as follows:

$$V_{vane} = \pi(r_{o_2}^2 - r_{i_2}^2)\frac{\beta}{2\pi}b_2 \qquad (1)$$

By taking into account the vane number *z*, the total volume of seed cotton carried through the gaps between the successive vanes is as follows:

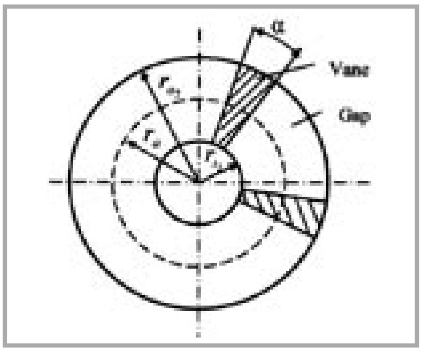

Figure 3. *Theoretical model for second unit of separator.*

$$V_2 = \pi(r_{o_2}^2 - r_{i_2}^2)(1-\frac{\beta}{2\pi}z)b_2 \qquad (2)$$

The flow rate can be determined by producing the effective circumference of the carrying area and the volume carried as given in equation (2). The seed cotton flow rate in the second unit for different running speed or angular velocity may be calculated as follows:

$$Q'_2 = 30\omega\, k_2 b_2\left(r_{o_2}^2 - r_{i_2}^2\right)\left(1-\frac{\beta}{2\pi}z\right) \qquad (3)$$

while k_2 is the coefficient matching with the effective portion of the seed-carrying circumference around the body of the unit taken as about 60-75%, and ω is the angular velocity of the second unit of the separator. Because the design provides a wide range of application facilities and flexibility, it is convenient to change equation (3) into a non-dimensional form. This may be done by defining the non-dimensional parameters in terms of the outer radius of the body as $\bar{b}_2 = b_2 / r_{o_2}$ and $\bar{r}_{i_2} = r_{i_2} / r_{o_2}$ and hence,

$$\overline{Q'}_2 = 30\, k_2 \bar{b}_2\left(1-\bar{r}_{i_2}^2\right)\left(1-\frac{\beta\, z}{2\pi}\right) \qquad (4)$$

where the non-dimensional flow rate is

$$\overline{Q'}_2 = Q'_2 / \omega r_{o_2}^3 \qquad (5)$$

The flow rate can be predicted by considering the non-dimensional radius and the total product of segment angle of the vanes by equation (4). Here, the volume of a carrying unit should be positive, i.e. $\left(1-\frac{\beta}{2\pi}z\right) > 0$. Defining the average thickness on the average radius of the vane as $t = \beta(r_{o_2} + r_{i_2})/2$ and expressing 'β' in terms of '*t*' in the inequality above yields $1 - tz/(\pi(r_{o_2} + r_{i_2})) > 0$. From this, in order to run the separator unit, the relationship between the number of vane and the geometrical parameters of such a vane can be expressed as follows:

$$\frac{\pi}{z} > \frac{t}{r_{o_2} + r_{i_2}} \qquad (6)$$

In the limit, for a given inner and outer radius of vanes, the average thickness of a vane in such a separator unit should meet the following condition:

$$t = \frac{\pi}{z}(r_{o_2} + r_{i_2}) \qquad (7)$$

Another approach mentioned in the first part of this study [5] gives the non-dimensional flow rate for the second unit of the separator as below:

$$\overline{Q}_2 = 30\, k_2 \bar{b}_2\left(1-\bar{r}_{i_2}^2\right) - \pi z \bar{t}\left(1-\bar{r}_{i_2}\right) \qquad (8)$$

Since the flow rate equations given in (4) and (8) should give the same flow rate, and considering these equations together and simplifying the results, we obtain the following:

$$\bar{r}_{i_2} = \frac{2\pi^2 \bar{t}}{\beta} - \frac{4\pi}{\beta z} \qquad (9)$$

or

$$\beta = \frac{2\pi^2 \bar{t}}{\bar{r}_{i_2}} - \frac{4\pi}{\bar{r}_{i_2} z} \qquad (10)$$

Considering the second unit of the separator being full of vanes and no seed cotton being carried, in other words if $\frac{z\beta}{2\pi} = 1$ or $\beta = \frac{2\pi}{z}$ then we obtain the following:

$$\bar{r}_{i_2} = \pi \bar{t} z - 2 \qquad (11)$$

where the non-dimensional thickness, radius and the product of total thickness of the revolving reel with the number of vanes are expressed as $\bar{t} = t / r_{o_2}$, $\bar{r}_{i_2} = r_{i_2} / r_{o_2}$ and $z\bar{t}$, respectively.

Theoretical model for dynamic behaviour

The motion of the mechanical parts of a machine is firstly determined by kinematic analyses in an engineering design. In this analysis, determining the forces and moments causing the unpredicted motions of the machine elements may be necessary, and these forces and moments are determined by considering the dynamic operating parameters. For this purpose, the mass moment of inertia, linear and angular acceleration and the mass of all mechanical parts of the machine should be considered.

Here, the first unit of the separator (shown in Figure 4a) is examined with regard to its dynamic behaviour. The mass of seed cotton should be taken from the entrance and transferred to the second unit by air suction. This mass is transferred by the tip of the vanes as indicated in the figure. The power unit of the separator should resist and overcome the weight of the seed cotton and wiper rods; the proposed theoretical model for the driving unit of the first unit is given schematically in Figure 4b.

The first unit is driven by an electrical motor in saw-gin ginning machine; the weight of seed cotton and the vanes should be resisted by this electrical motor. The total load of the first unit should be resisted by driving the unit to obtain the desired motion. In saw-gin separator units, the belt-pulley is used to reduce the speed of the motor to that of the separator shaft. The degree of reduction is denoted as i and it is called the speed reduction ratio. As expected, the belt-pulley runs as a speed reductor when the speed reduction ratio i is denoted as 2:1, 3:1, … to 10:1. However, the belt-pulley operates to increase the speed when i is denoted as 1:2, 1:3, …, to 1:10.

In this theoretical approach, the standard flat belts, because they are flexible mechanical elements, are suggested for use in the transmission of power, since they are elastic and usually long, and they are capable of absorbing shock loads and damping out the effects of vibrations that might occur during the operation. These elements are mainly considered to be frictional power transmission units, and substantial amounts of power are transmitted by static friction as opposed to sliding friction. The mechanical behaviour of the belt-pulley system has already been analysed theoretically and experimentally. It appears that because of the slip and elastic creep during the operation, the angular velocity ratio between the driving and driven shaft is neither constant nor exactly equal to the ratio of the pulley diameters [7]. A change in belt

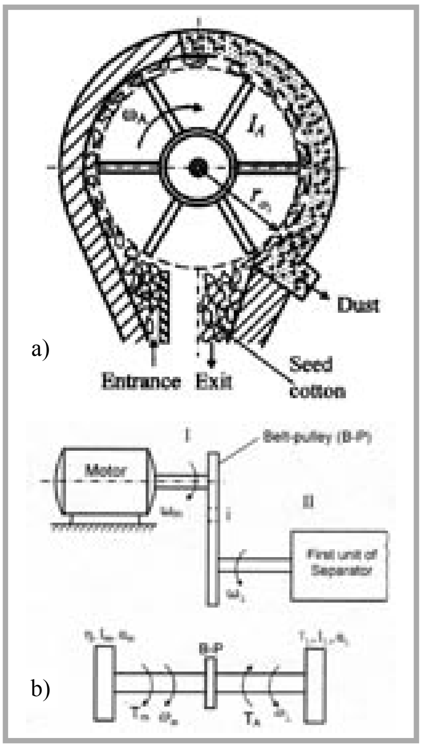

Figure 4. *Theoretical model for dynamic analysis, a) first unit of separator [5], b) theoretical model.*

tension due to friction forces between the belt and pulley will cause the belt to elongate or contract and move relative to the surface of the pulley. In other words, the tensile stress on the tight side of the belt decreases from the maximum σ_1 to the minimum of σ_2 on the driving pulley. Hence, the deformation on the belt is changed from $\varepsilon_1 = \sigma_1 / E$ to $\varepsilon_2 = \sigma_2 / E$ on the same belt moving on the driving pulley. This results in the contraction of the belt length. $\sigma_u = \sigma_1 - \sigma_2$ represents the tensile stress difference, which is denoted as the useful stress maintaining the pulley rotation at the same speed with the belt.

In a typical flat belt-pulley mechanism, due to the existence of elastic elongation or creep depending on the magnitude of $\sigma_1 - \sigma_2$, the elastic slip becomes inevitable and the slip coefficient is approximately defined as follows:

$$\Psi = \varepsilon_1 - \varepsilon_2 = \frac{\sigma_u}{E} \qquad (12)$$

where E is the dynamic elasticity modulus of the belt used, and σ_u is the useful tensile stress for power transmission. Because of the slip, the real speed reduction ratio defined as $i_r = \omega_1 / \omega_2$ is different from the theoretical speed reduction ratio defined as $i_t = d_2 / d_1$. Considering the slip coefficient and re-defining it as

$$\Psi = 1 - \frac{\omega_2 d_2}{\omega_1 d_1} \qquad (13)$$

the relationship between i_r and i_t can be obtained as

$$i_r = \frac{\omega_1}{\omega_2} = i_t\left(\frac{1}{1-\sigma_u / E}\right) \qquad (14)$$

or

$$i_r = \frac{\omega_1}{\omega_2} = \frac{d_2}{d_1(1-\Psi)} \qquad (15)$$

It may be seen that the real speed reduction ratio is slightly greater than the theoretical speed reduction ratio. However the elastic slip (σ_u / E) falls within the range of 0.5 to 1.5% in practical applications [7 - 9]. Therefore, in this theoretical study, the theoretical speed reduction ratio is taken as equal to the real speed reduction ratio, i.e. $i = \frac{\omega_1}{\omega_2} = \frac{\omega_m}{\omega_{out}}$. Hence, the elasticity of the belt or elastic slip or creep is neglected.

The speed reduction ratio i can be defined as follows:

$$i = \frac{\omega_m}{\omega_{out}} = \frac{\omega_m}{\omega_A} = \frac{T_L}{T_m} \qquad (16)$$

where ω_m and ω_A are the angular velocities of the motor and separator respectively. T_L is the total moment of separator or load, and T_m is the driving motor moment or torque.

In this analysis the following characteristic parameters defined have also been used, i.e. I_m is the mass moment of inertia of the electrical motor, I_A is the mass moment of inertia of the wiper rods, I_L is the total mass moment of the separator unit, m is the mass of the wiped seed cotton, r_{o_1} is the outer radius of the wiper rods, and α_m and α_L are the angular accelerations of the motor and separator unit respectively.

The torque required from electrical motor can be estimated as

$$T_m = \alpha_m(I_m + I_L/i^2) \qquad (17)$$

and the torque of the separator unit is

$$T_L = (I_A + mr_{o1}^2)\alpha_L \qquad (18)$$

The mass moment of the inertias of the pulley and shaft are neglected as they are very small in comparison to the separator and electrical motor's inertia. In this case, the equation of motion in the system can be obtained as

$$\eta T_m - T_L = I_e \alpha_m \qquad (19)$$

where η is the efficiency of the motor and I_e is the equivalent inertia of the system [6]. Here, the equivalent inertia of system yields the following:

$$I_e = I_m + \frac{I_A + mr_{o_1}^2}{i^2} \qquad (20)$$

Considering equations (19) and (20) and rearranging the results gives the dimensional mass of seed cotton carried in the system as

$$m = \frac{\frac{\eta T_m}{I_A \alpha_m} - \frac{I_m}{I_A} - \left(\frac{\alpha_L}{\alpha_m} + \frac{1}{i^2}\right)}{\left(\frac{\alpha_L}{\alpha_m} + \frac{1}{i^2}\right)\frac{r_{o_1}^2}{I_A}} \qquad (21)$$

and the motor torque can be determined by rearranging this equation as Equation (22).

$$T_m = \frac{1}{\eta}\left(\alpha_L I_A + \frac{\alpha_m I_A}{i^2}\right)\left(\frac{mr_{o_1}^2}{I_A} + 1\right) + \frac{I_m \alpha_m}{\eta} \qquad (22)$$

The non-dimensional mass and torque of the system can be found in sequence as follows:

$$\bar{m} = \frac{\eta \bar{T}_m - \frac{I_m}{I_A} - \left(\frac{\alpha_L}{\alpha_m} + \frac{1}{i^2}\right)}{\left(\frac{\alpha_L}{\alpha_m} + \frac{1}{i^2}\right)} \qquad (23)$$

$$\bar{T}_m = \frac{1}{\eta}\left(\frac{\alpha_L}{\alpha_m} + \frac{1}{i^2}\right)\left(\bar{m}+1\right) + \frac{1}{\eta}\frac{I_m}{I_A} \quad (24)$$

where the non-dimensional mass of seed cotton is defined as

$$\bar{m} = m r_{i_1}^2 / I_A \quad (25)$$

and the torque of the motor is expressed as

$$\bar{T}_m = \frac{T_m}{\alpha_m I_A} \quad (26)$$

Defining the electrical motor power supplied as N_m, the power required from the electrical motor with 100% efficiency (nominal power) can be calculated by $N_m = T_m \omega_m$. Considering equation (22), motor power is obtained as equation (27), and the non-dimensional form of this power is written as

Equation 27.

$$N_m = \frac{1}{\eta}\omega_m I_A\left(\alpha_L + \frac{\alpha_m}{i^2}\right)\left(\frac{m r_0^2}{I_A}+1\right) + \frac{\omega_m \alpha_m I_m}{\eta}$$

$$\bar{N}_m = \frac{I_A}{I_m}\frac{1}{\eta}\left(\frac{\alpha_L}{\alpha_m} + \frac{1}{i^2}\right)\left(\bar{m}+1\right) + \frac{1}{\eta} \quad (28)$$

where the non-dimensional power is defined as

$$\bar{N}_m = \frac{N_m}{\omega_m \alpha_m I_m} \quad (29)$$

Theoretical results and discussions

With the equations given above regarding cotton flow rate estimation and the dynamic behaviour of the separator, the design and running conditions of the first and second units of the separator are outlined. The cotton flow rate, the cotton mass presented in the separator, ad the torque and power required in the system predicted with the dynamic model we developed are examined.

Flow rate estimation for the 2nd unit

The variation of non-dimensional flow rate ($\bar{Q'}_2$) with a non-dimensional inner radius of a second separator unit ($\bar{r}_{i_2}$) for different βz values is given in Figure 5. Although it may have different values in some applications, k_2 is maintained at 0.7 and $\bar{b}_2$ is selected at 1.0; the non-dimensional radius was varied from 0 to 1, and βz was changed from $\pi/6$ to $3\pi/2$. It may be seen that increasing the radius decreases the flow rate in all βz values, as expected. When a smaller βz is introduced, the variation tends to be slightly different than in the case of the higher βz case. For high βz values, increasing the non-dimensional radius up to 0.3 does not cause much difference in the cotton carrying capacity. In addition, for all the total volume of vanes (βz), the non-dimensional flow rate tends to decrease more rapidly with radius ratio ($\bar{r}_{i_2}$) being greater than approximately 0.3. As expected, it appears that at $\bar{r}_{i_2}$=0 the maximum flow rates are obtained, although this is not useful in designing and operating the separator. No seed cotton carrying takes place at $\bar{r}_{i_2}$=1 due to the disappearance of volume in the unit. In other words, the volume of the vanes existing in the system should be as small as possible, so as to carry more cotton in the separator, and to be as great as possible in order to resist the weight of the cotton in the system. In industrial applications, $\bar{r}_{i_2}$ is selected at a value between 0.3 and 0.5. The outer radius of the second unit (r_{o_2}) can be determined between 0.2 and 0.4 m. For a given inner radius ratio $\bar{r}_{i_2}$=0.4 and $\beta z=\pi/4$; taking k_2=0.7 and $\bar{b}_2 = 1$, the corresponding non-dimensional flow rate is obtained as $\bar{Q'}_2$=15.435, which in turn corresponds to a cotton flow of Q'_2=2.78 m^3/s when the running speed is $\omega=2r/s$ and r_{o_2}=0.3 m. It is found that the separator can carry 15.213 tonnes of seed cotton per hour, considering the density of cotton as 1.52 g/cm^3.

The variation of the non-dimensional flow rate ($\bar{Q'}_2$) with the total volume of vanes (βz) for different non-dimensional radius ($\bar{r}_{i_2}$) can also be seen in Figure 6. As in Figure 5, k_2 is maintained at 0.7 and $\bar{b}_2$ is taken as 1.0, whereas the total volume of the vane (βz) was varied from 0 to 2π, and $\bar{r}_{i_2}$ was changed between 0.3 to 0.5. As expected, the variation is linear, and increasing βz causes to a linear decrease in $\bar{Q'}_2$ for all radius values. Since the separator's carrying volume or capacity of seed cotton decreases by decreasing the non-dimensional radius of the vanes, the flow rate values to be obtained are smaller than those of the greater $\bar{r}_{i_2}$ for the same βz value. It seems that at βz = 0, the maximum flow rates are obtained, although in this case this means no vane is present in the system, resulting in the disappearance of the second unit. Hence, such a system is of no use. Again, seed cotton is not carried at $\bar{r}_{i_2}$ = 1 due to the disappearance of the carrying volume of seed cotton in the unit. In industrial applications, the total product of the number of vanes and the segment angle corresponding to a vane can be selected between $\pi/4$ to $\pi/3$.

βz indicates the restriction of the total vane volume in the second unit. In this approach, vane thickness is related to β, and there should be a limitation on the number of vanes and the segment angle corresponding to each vane. Figure 7 shows the variation of β with z by taking the radius ratio ($\bar{r}_{i_2}$)fixed at 0.4 and varying $\bar{t}$ in the range 0.1 … 0.25. It can be seen that β is negative for some z values. This is not logical; β should be positive in order to design a vane. Hence, the region of the negative β values obtained should not be considered in design. β increases by increasing z for each curve; in addi-

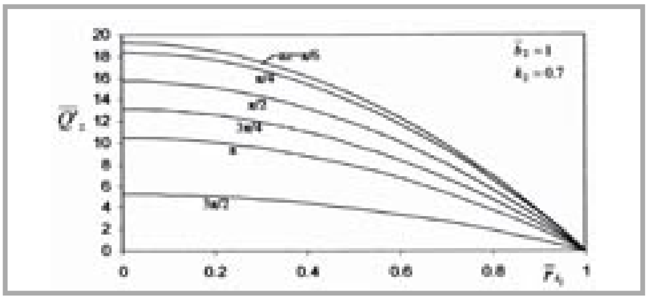

Figure 5. *The variation of $\bar{Q'}_2$ with $\bar{r}_{i_2}$.*

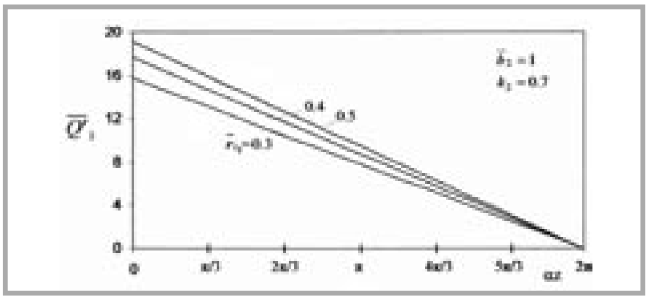

Figure 6. *The variation of $\bar{Q'}_2$ with αz.*

tion, the variation of β is more marked up to the point $z = 10$. However, after this point, the rate at which β increases becomes reduced by further increasing the number of vanes (z). Because of the interrelationship between β and t, increasing $\bar{t}$ increases the β value obtained for a given z value, as expected. It is seen that the selection of the thickness ratio of vane is important for determining the number of vanes. Taking $\bar{t} = 0.15$, the vane number that makes the vane segment angle zero is 4. For $z = 10$, the corresponding β value is 4.25°. The product of these values yields $\beta z = 42.5°$, which is approximately within the range suggested above.

Dynamic behaviour of the first unit

The relationship between the vane radius ($\bar{r}_{0_1}$) of the first separator unit and the carried seed cotton mass (m) for different inertia ratios (I_m/I_A) can be seen in Figure 8. Although the working parameters may take various values, the angular acceleration of the load (α_L) and the motor (α_m) are selected as 2 and 1 r/s^2 respectively. Furthermore, the other working parameters, such as the torque obtained from the motor ($T_m = 50$ Nm), the efficiency of the motor ($\eta = 0.95$), the gear-rate of the belt pulley ($i = 3$) and the inertia of the wipers (vanes) ($I_A = 1$ kg.m^2) are kept constant. These fixed operating values are common values in ginning machines. The inertia ratio (I_m/I_A) is varied between 0.25 and 20, and the carried seed cotton mass is obtained by varying the vane radius between 0 and 2 m. In this theoretical study, the suggested vane radius ($\bar{r}_{0_1}$) is in the range of 0.4 to 0.8 m, and the mass of cotton in the system (m) should be between 50 and 100 kg. It may be seen that the value of the mass present in the system (m) is proportional to the inverse square power of the radius ($\bar{r}_{0_1}$). Hence, the carried seed cotton mass decreases rapidly by increasing the vane radius up to a value of approximately $\bar{r}_{0_1} = 1$ m for all inertia ratios. Further increase in the vane radius results in a nearly constant and slowly decreasing carried mass value up to $\bar{r}_{0_1} = 2$ m. The high value of the radius of the vane ($\bar{r}_{0_1}$) restricts the volume of the seed cotton carrying capacity in the system. Hence, the volume of the gap, in which the seed cotton is circulated by the air vacuum, decreases. The reduced volume of the gap (high radius values) in turn decreases both the moment of inertia required from the electrical motor and the carried seed cotton mass. For instance, upon selecting $I_m/I_A = 5$, 120 kg of seed cotton is carried at the point $\bar{r}_{0_1} = 0.4$ m, which decreases down to 53.14 kg at $\bar{r}_{0_1} = 0.6$ m on the same curve.

The variation of the carried seed cotton mass (m) with the belt pulley gear-rate (i) for fixed running conditions is also analysed for different inertia ratios (I_m/I_A), as shown in Figure 9. In this variation, the specific parameters are taken as constant, as indicated on the figure. The belt-pulley gear-rate (i) is varied between 0 and 5, and the corresponding mass of seed cotton processed in the system is obtained. The carried seed cotton mass (m) increases rapidly with the increase of i to a certain value, and tends to remain constant for $i > 2$. Increasing I_m/I_A decreases the obtained mass of carried seed cotton for the fixed gear-rate value. In Figure 9, small gear-rate values ($i < 0.2$) yield negative values of carried seed cotton mass, which is not acceptable for running the separator in satisfactory conditions. That is to say, it may be observed that with a certain value of i (less than 0.2), the theory yields a negative mass or flow rate, which means that the cotton delivery to the unit exhibits a backflow from the outlet, and this is not logical.

It may be seen that there are three different modes of operation in cotton mass variation determining the design and operation limits of the system. These are marked as region I, II and III in the figure. In the first mode (region I) there is no seed cotton carriage which is not preferred in the design. In the second mode (region II) the seed cotton mass increases to certain maximum points for different I_m/I_A and i values, as shown with a dashed line. The processed cotton mass in the separator should be as great as possible within the range suggested above; this region may be considered as the transitional region. In region III, the model gives maximum seed cotton mass values, and the curves tend to be nearly constant; increasing the belt-pulley speed ratio (i) seems to cause small increases of mass values for all I_m/I_A ratios. In addition, since the system is more stable in region III, the right-hand side of the dashed line and the peaks of the curves can be considered as being the design region. This is the most important conclusion for satisfactory operation of the system which arises from the present theoretical approach. Since the inertia of the motor is greater than that of the separator, by increasing I_m/I_A, the mass present in the system for a given speed reduction value (i) decreases. In the figure, fixing $I_m/I_A=3$ and selecting i=0.5, 1, 3 and 5, the carried seed cotton mass for these four different running conditions gives values of 17.8 kg, 38.4 kg, 55.8 kg and 57.8 kg respectively. In some industrial applications, i takes the values between 3 and 10. Increasing i increases the presence of seed cotton between the wiper bars and the separator body. Thus, the carried seed cotton mass increases, as expected.

Since the seed cotton mass depends mainly on the geometrical and operating parameters of the first unit as explained before, the theoretical analysis was extended, and the results given in Figure 10a and b have been produced

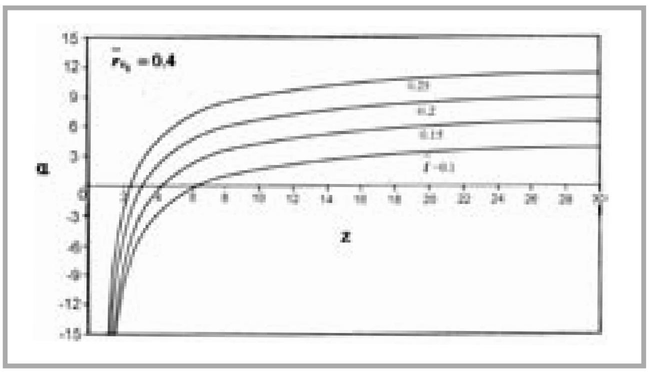

Figure 7. The variation of α *with z.*

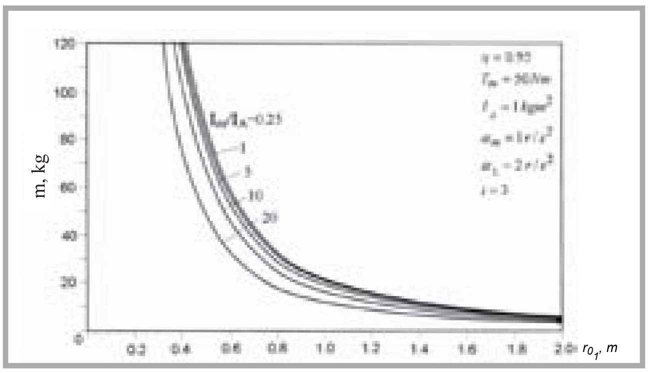

Figure 8. The variation of carried seed cotton mass with vane radius.

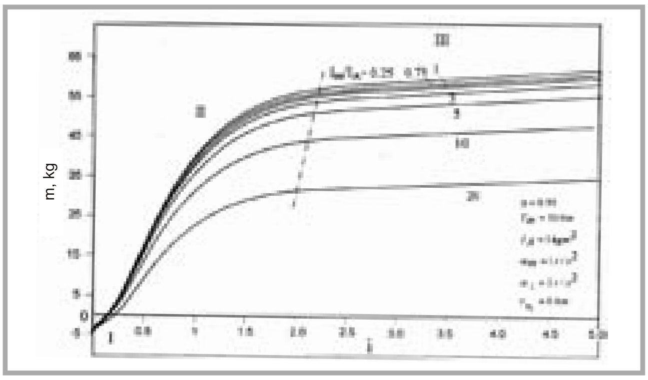

Figure 9. The variation of seed cotton mass (m) with belt pulley gear-rate.

for this purpose. The figures represent the variation of the non-dimensional seed cotton mass ($\bar{m}$) present in the system with the belt-pulley gear-rate (i) for different inertia ratios (I_m/I_A) varied between 0.25 and 20 by selecting the specific parameters indicated on the figures. The belt-pulley gear-rate is varied between 0 and 5 in Figure 10a, while it is varied between 0 and 1 in Figure 10b. Here, $\bar{m}$ increases rapidly to a certain belt pulley gear rate value, and remains approximately constant by increasing the speed reduction value (i). It is noted that an increase in I_m/I_A causes a reduction in the mass present in the system. There are three regions seen on the figure to display the running characteristics of the system. It appears from the analyses of the figures that in Region I the obtained negative mass is neither explicable nor desirable, and so this region should not be considered in design. Region II can be denoted as a transitional region for the mass present in the system, and a rapid increase in the mass is obtained in this region. In Region III, the system characteristics are more stable, and the maximum cotton mass present in the system is achieved. If the gear-rate (i) is adopted to be higher than 1 (speed reducing power transmission system), it can be suggested that the design and operating conditions should be selected from region III in order to maintain satisfactory production in the cotton carriage. Regions I and II may be seen more clearly by considering the belt-pulley gear-rate as a speed increaser ($i < 1$) as shown in Figure 10b. Since the negative and small values of $\bar{m}$ are obtained in these regions, the design parameters are not preferred in region I or in the partial section of region II ($i < 1$ section, compared with the full section of II in Figure 10a) in a separator unit of a ginning machine examined theoretically. The higher i value corresponds to a higher $\bar{m}$ value, for instance, on the curve $I_m/I_A = 3$, selecting $i = 0.5$ for the non-dimensional mass yields $\bar{m} = 3.25$, and the corresponding mass is found as $m = 9.02$ kg with $I_A = 1$ kg·m^2 and $r_{a_1} = 0.6$ m. Since this mass indicates the presence of cotton mass between the wiper rods (vanes) and the body of the machine, which is very small, this value is meaningless and not practical for a saw-gin separator unit. Again on the curve for $I_m/I_A = 3$, selecting $i = 3$, the non-dimensional seed cotton ($\bar{m}$) takes the value of 11.07 and the corresponding dimensional mass value is evaluated as $m = 61.5$ kg by considering $I_A = 2$ kg·m^2 and $r_{a_1} = 0.5$ m.

In order to maintain the continuous feeding and processing of cotton in the separator unit, the electrical motor should transmit the required torque to the separator shaft by a belt pulley mechanism. Here, the required torque can be evaluated by mainly considering the belt-pulley gear-rate and the mass of cotton carried in the system. Figure 11 demonstrates the non-dimensional motor torque variation with i for different non-dimensional mass ($\bar{m}$) levels and for constant parameters as indicated on the figure. Here, I_m/I_A is selected as 0.25, indicating that the inertia of separator is higher than that of the electrical motor. Small torques of electrical motor are desired in a motor-separator system considering it from the point of view if low energy consumption. High $\bar{T}_m$ values are obtained for small i values because the speed of the power transmission system increases ($0 < i < 1$). $\bar{T}_m$ decreases rapidly by increasing the gear rate for all non-dimensional mass levels, and seems to be constant for higher values of i (higher than 2). There are two regions detected on the figure for $\bar{T}_m$-i variation. Since relatively high torques are obtained and the gear-rate works to increase the speed in region I, this region should not be taken as a design region. Motor torque should be as small as possible regarding energy consumption, and the gear rate should meet the condition of $i > 1$. On the curve $\bar{m} = 10$, the non-dimensional torque $\bar{T}_m$ yields 69.7, 35 and 24.7 for the belt pulley gear rate values of 0.5, 1 and 3 respectively. The corresponding dimensional torques obtained from the motor are 139.4 Nm, 70 Nm, and 49.4 Nm respectively by selecting $I_A = 2$ kg·m^2 and $\alpha_m = 1$ r/s^2. The torque obtained from the electrical motor should be suggested within the range of 30 to 75 Nm for satisfactory operation of the system.

Motor power is an important parameter to determine the electrical motor characteristics for satisfactory operation of the system. Figure 12 represents the variation of non-dimensional motor power

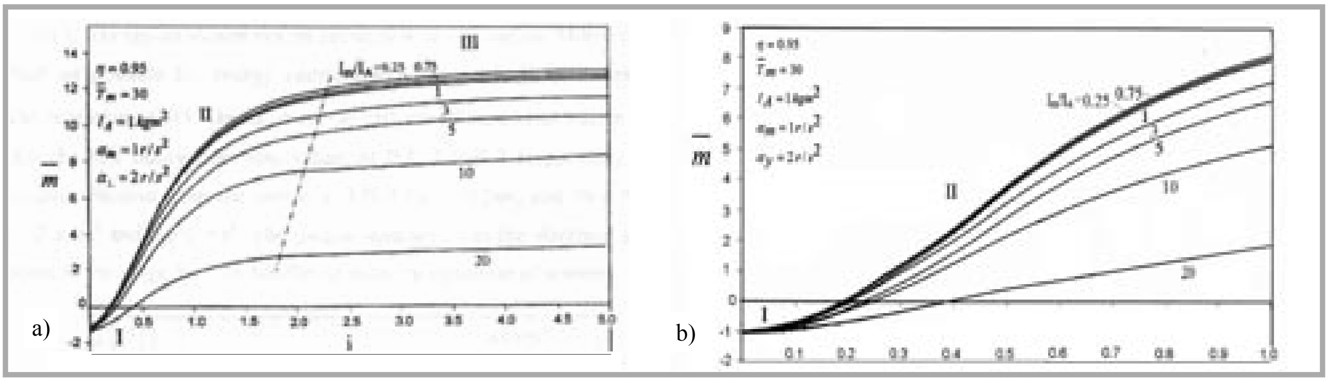

Figure 10. Non-dimensional seed cotton mass versus belt-pulley gear rate'i'.

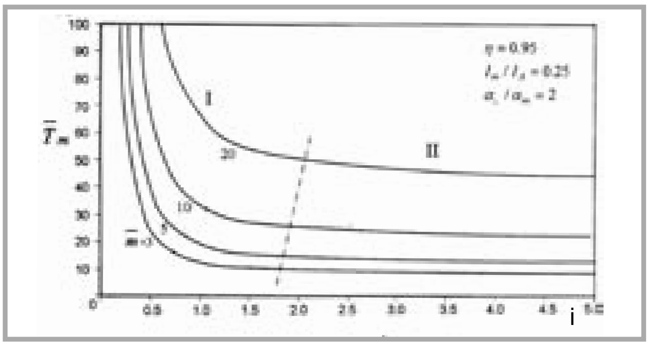

Figure 11. *The var. of non-dimensional torque with belt-pulley gear-rate.*

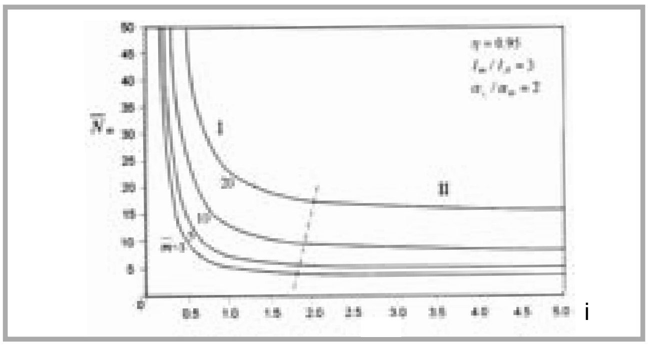

Figure 12. *The var. of non-dimensional power with belt-pulley gear-rate.*

versus belt-pulley gear-rate by considering the specific constant parameters indicated on the figure and varying the non-dimensional mass between 3 and 20; I_m/I_A is selected as 3. The motor power should be as great as possible to drive the system, whereas it should be small as possible regarding energy consumption. $\overline{N}_m$ decreases rapidly by increasing the gear rate for all non-dimensional mass levels, and remains constant for $i > 2$. High $\overline{N}_m$ values are obtained for small i values. The figure is similar to that of Figure 11 and the assumptions are likely to be the same. There are two regions on the figure described as region I and II. High motor power values are obtained in region I, whereas they are nearly constant and small in region II; hence, region II can be selected as the design region. On the curve $\overline{m} = 5$, the non-dimensional torque $\overline{N}_m$ yields 13.65, 7.35, and 5.48 for the belt pulley gear rate values of 0.5, 1 and 3 respectively. Corresponding dimensional powers obtained from the motor are 40.2 kW, 21.78 kW and 16.15 kW by selecting $I_m = 3$ kg·m^2 $\omega_m = 100$ r/s, and $\alpha_m = 1$ r/s^2 respectively. It is clear that increasing the belt pulley gear rate decreases the required motor power. It can be concluded that the motor power should be in the range of 10 to 25 kW for satisfactory running or operation of the system analysed.

Conclusions

As a result of our theoretical investigation carried out into the flow-rate estimation of the second unit and the dynamic behaviour of the first unit of the separator of the saw-gin ginning machine, the following conclusions can be drawn:

1. From this analysis, it is possible to estimate the non-dimensional flow by considering the vane thickness. It is found that the total vane volume (βz) in the second unit should be in the range $\pi/4$ to $\pi/6$. In addition, the number of vanes (z) may be varied between 5 and 10, and the corresponding angle of a vane (β) can be estimated as 4.5° to 12°.
2. It has been shown that a suitable vane radius in the first unit of the separator is important to determine the present mass of seed cotton between the wiper rods and the body of the separator. The theory indicates that r_{a} should be in the range 0.4 to 0.8 m, and also the present cotton mass (m) in the first unit of separator should be taken between 50 and 100 kg for satisfactory operation.
3. Because of the elastic slip or creep, the angular velocity ratio of the driven and driver shafts are not equal to the ratio of the pulley diameters. However, the elastic slip is very small in practical applications. Therefore, in this theoretical study, the elasticity of the belt or elastic slip or creep is not considered.
4. The theory also predicts that there will be a lower limit to the belt-pulley gear-rate at which successful operation can be guaranteed. Depending on the I_m/I_A values, the limiting value of the gear-rate is $i > 2$. The greater the gear-rate, the higher the mass present in the system obtained. It can be concluded that the suggested gear-rate value would be in the range of $3 < i < 10$.
5. From this analysis, the torque obtained from the electrical motor can be estimated by defining the non-dimensional parameters. It is demonstrated that the motor torque should be within the range of 30 to 75 Nm for driving the separator.
6. The required motor power to drive the mechanism is also predicted by the theory developed. High motor power is required by selecting $i < 1$ because of the increasing speed of the power transmission system. In the nominal mode of operation, i takes the value greater than 1. The motor power should be in the range of 10 to 25 kW for running the system satisfactorily.
7. Using the non-dimensional parameters defined, it is always easy and useful to calculate the dimensional parameters by taking into account the different geometrical and operating parameters.
8. Considering the remarks outlined in this paper regarding the flow rate and dynamic behaviour of the separator units, this theoretical approach would be considered as a design tool for the saw- ginning machinery designers with special reference to separator design. However, further experimental substantiation of the results predicted from the model ought to be carried out.

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