International Journal on Advanced Science Engineering Information Technology

Vol.9 (2019) No. 3
ISSN: 2088-5334

Effects of Frequency and Mass of Eccentric Balls on Picking Force of The Coffee Fruit for The As-Fabricated Harvesting Machines

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***Abstract*—Currently, Vietnam ranks second about the coffee export in the world. To match that position, the use of coffee harvesting tools/machines according to the automatic trend is essential. However, the most common forms of coffee harvesting in Vietnam are manual, improved manual-coffee picking machines that are imported from foreign countries. The above harvesting forms have low productivity and have some disadvantages such as labor cost, labor hiring, high labor cost, and long harvesting time, low harvesting, and post-harvesting quality. Studies of scientists around the world have applied the principle of vibration to produce the picking force for coffee fruits, and the picking force is known to be different from many factors in every region of the world growing coffee. The paper presents the method of design and manufacturing an automatic coffee harvesting machine based on the evaluation of picking force for coffee in Vietnam. The influences of eccentric ball mass and vibrating frequency on the magnitude of the picking force are carefully calculated. On that basis, the experimental programming is applied to find the optimal working point of the picking machine for coffee. The results showed that the eccentric ball mass of 8.5 kg, the rotation speed of the eccentric ball from 480 to 574 rpm would produce the picking force by the coffee fruit of Vietnam.**

***Keywords*—coffee harvesting machine; picking force; eccentric balls; frequency.**

I. Introduction

Vietnam is currently the world's second largest coffee exporter. The popularization of automatic coffee harvesters is necessary if the country is to maintain that position. Despite numerous potential benefits, such an ambition would pose serious challenges to the mechanical engineering industry in terms of how it may fulfill the needs of the nation's coffee exporting industry. In Vietnam, coffee cherries are often either hand-picked or mechanically harvested – a process known as mechanical stripping, in which pickers will be assisted by a tool called "derricadeiras" [1]. In general, these approaches result in submaximal yields and have some inherent drawbacks such as the intense nature of manual labor, high cost and short supply of labour, the protraction of harvest seasons, reduced intra-harvest and post-harvest yield quality. Domestic coffee harvesting techniques are inferior to those abroad, with the majority involving some form of manual labor, resulting in suboptimal yields. Therefore, the industrialization of these methods is critical in reducing manufacturing and processing costs and boosting yield quantity and quality, meaning an upgrade to better and modern equipment, including automatic coffee harvesters is imperative. These machines have the ability to adjust stripping forces, resulting in more uniform batches of beans and less post-harvest care.

D.O.Mbuge [2] tested the possibility of using a power take-off (PTO) driven mechanical shaker for coffee harvesting. It was found that a crank throw of 0.04m produced the highest proportion of ripe cherries. Also, 1.2N and 0.9N force was needed to knock unripe and ripe cherries loose, separately. The force required to shake a coffee tree was 12.8N. These results show that the crank-slider mechanism is suitable for use in coffee harvesting and that multiple coffee trees can be harvested concurrently for higher productivity. Coffee harvesting machines designed and manufactured by Goto et al. [3] offer low yields. Portable coffee harvesting machines developed by Victor M. Alexandrinon [4] include motor-driven rotating swindles with helicoidally threads. The swindles rotate for engaging the cherries or the branch as the operator moves the machine along a branch. The threads are designed to detach the target cherries from a branch. A net will catch the fallen cherries on the ground. Roy Scudder developed a machine for harvesting crops on plants which have stalks growing from the ground. The machine can be advanced in the direction along a row of stalks, and on its frame, there are laterally spaced side conveyors comprised of a rotatable central shaft which straddle the crops. Fábio Lúcio Santos et al. [5] examined the coffee harvesting process with the help of an electromagnetic shaker. The authors run various teststo determine the effect of the amplitude and frequency of vibration on the efficiency of coffee cherries harvesting. The tests were done in a laboratory using an electromagnetic shaker, with amplitudes ranging from 3.75 to 7.50 mm, and frequencies ranging from 13.33 to 26.67 Hz. The results indicate that harvesting efficiency is directly linked to the acceleration reached by the cherries during the harvesting process. The frequencies of 23.33 Hz, 26.67 Hz and amplitudes 6.25 to 7.50 mm offered the greatest harvesting efficiency of ripe coffee cherries for both varieties tested.

The use of coffee harvesting machines on an industrial scale that ensures both harvesting efficiency and high post-harvest cherries quality has not gained wide acceptance in Vietnam. No research has been carried out to determine the suitable stripping force for cherries grown in Vietnam and the province of Dak-Lak in particular. Inadequate stripping force may lead to reduced post-harvest quality. Characteristics of soil and soil landscapes may vary greatly among regions, and these differences have an impact on the pedicel toughness of coffee berries. It is therefore essential to determine the adequate stripping force by region. A research study on contributory factors to stripping force is imperative in that a range of sufficient stripping forces can be established for coffee varieties grown in the province of Dak-Lak. The outcome of such a study may be adopted as recommendations and guidelines for other regions.

II. Material and Method

A. Determine the stripping force for coffee cherries

To ascertain the forces required to separate coffee pedicels from the stem, coffee branches collected in the province of Dak-Lak were transported to the laboratory with minimal delay to ensure minimal leaf and pedicel shrinkage and dehydration, which may discredit the outcomes of the study.

Fig.1. Coffee branches collected in the province of Dak-Lak

Stripping forces for pedicels and petioles were determined with the help of an Instron 5544 Testing Machine. Averages values were calculated using the machine's software. A coffee branch was secured on the fixed crosshead. A hook attached to the moving crosshead was hooked onto the cherries. The moving crosshead was jogged up so that the cherries would be separated from their pedicels. As for the petioles, a similar configuration was employed; however, the hook was removed and replaced by grips. The test was repeated for 100 specimens of each type.

Fig. 2. Instro 5544 Testing Machine

TABLE I
AVERAGE STRIPPING FORCES FOR PEDICELS AND PETIOLES

Type	Color	Ø peticles/ petioles (mm)	Ø cherries (mm)	Average stripping forces (N)	Number of specimens
Peticels	Red	1 to 2	11 to 12	5	100
	Green	1 to 2	11 to 12	5.7	100
Petioles	-	2	-	14	50

In this study, the vibrating shaft and vibrating bars are fabricated by steel CT45. After welding vibrating bars with vibrating shaft, they are treated by the heat in order to increase the strength. The heat treatment for the welds are demonstrated in some published works [6][7].

B. Design of a coffee harvesting machine

The center of the eccentric balls [8]:

$$\boldsymbol{y_c} = \frac{\left(\boldsymbol{H} + \frac{\boldsymbol{4R}}{\boldsymbol{3\pi}}\right).\frac{\boldsymbol{\pi R^2}}{\boldsymbol{2}} + \frac{\boldsymbol{H}}{\boldsymbol{2}}.\boldsymbol{BH} - \frac{\boldsymbol{H}}{\boldsymbol{2}}.\boldsymbol{\pi r^2}}{\frac{\boldsymbol{\pi R^2}}{\boldsymbol{2}} + \boldsymbol{BH} - \boldsymbol{\pi r^2}} = \boldsymbol{f(H, B, R, r)} = \boldsymbol{l_c} + \frac{\boldsymbol{H}}{\boldsymbol{2}} \quad (1)$$

The mass of the eccentric balls (m, kg) [9]:

$$\boldsymbol{m} = \boldsymbol{\rho.t.}\left(\frac{\boldsymbol{\pi R^2}}{\boldsymbol{2}} + \boldsymbol{BH} - \boldsymbol{\pi r^2}\right) \quad (2)$$

Where:
R- Eccentric radius, m
r- Axis of the eccentric balls, m
H- Thickness of cylinder, m
B- Length of rectangular section, m

While rotating, two eccentric balls produce the couple-torque applied to the z-axis which contains the vibrating axis. Because the thickness of the eccentric balls is small, when it rotates around the z1 axis, it should be considered as dynamic reaction affecting the center of support axis.

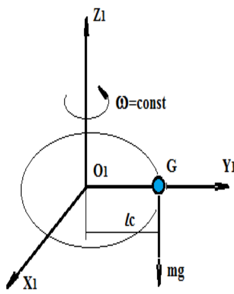

Fig. 3. Force analysis for the eccentric balls

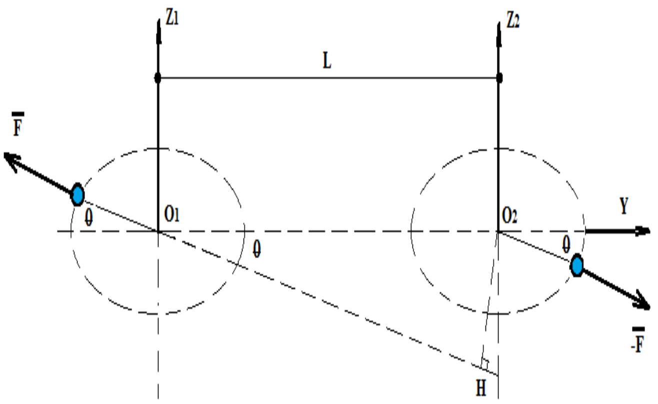

Fig. 4. Analysis for the force of the vibrating structure

Considering 2 eccentric balls and θ = ωt that is considered as rotation angle of the eccentric balls. The couple-torque produced by $(\bar{F}\,, -\bar{F})$ with $F = m\omega^2 l_c$ is [9]:

$$\boldsymbol{M = m\omega^2 l_c . L \sin\theta = m\omega^2 l_c . L \sin(\omega t)} \qquad (3)$$

Where:
M- the couple-torque, Nm
m- the mass of the eccentric balls,kg
ω- the angle velocity of the eccentric balls, rad/s
Lc- eccentric radius, m
ωt – the rotating angle of the eccentric balls, rad

This couple-torque acts on the lower vibration bars. Assuming that the volume inertial-torque of the vibration shaft is $I_z^{(m)}$. The angular acceleration of this axis $\varepsilon_z$that is caused by this torque is calculated by followed equation [10]:

$$\boldsymbol{\varepsilon_z = \frac{m\omega^2 l_c . L}{I_z^{(m)}} \sin(\omega t)} \qquad (4)$$

$$\boldsymbol{\omega_z = \int_0^t \varepsilon_z . dt = -\frac{m l_c . L\omega}{I_z^{(m)}} \cos(\omega t)} \qquad (5)$$

$$\boldsymbol{\varphi_z = \int_0^t \omega_z . dt = \frac{m l_c . L}{I_z^{(m)}} \sin(\omega t) = -\varphi_0 . \sin(\omega t)} \quad (6)$$

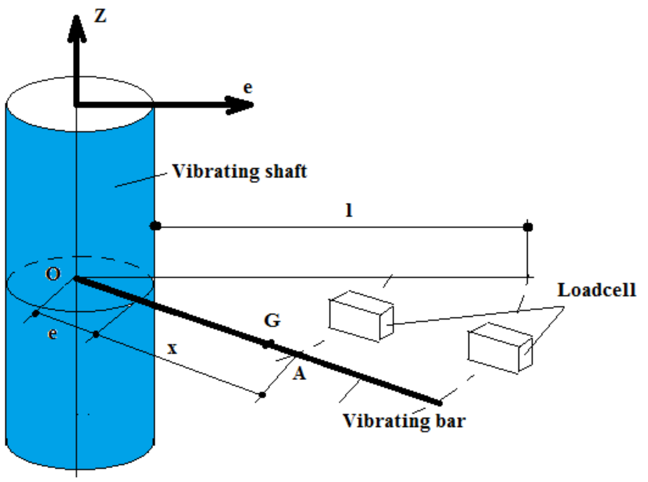

Fig. 5. Analysis for the force of the vibrating shaft

Thus the axis containing the vibrating bars will rotate around its axis with the rule of angle vibration [11]:

$$I_z^{(m)} = \frac{m_t . l^2}{12}(7) \qquad (7)$$

$$\varphi_z = \varphi_0 . \sin(\omega t) \qquad (8)$$

When measuring the impact force at position A with the distance from O center of the axial of (e + x), maximum impact distance (amplitude) is:

$$\Delta_x^0 = \varphi_0(e + x) \qquad (9)$$

When measuring the impact force at position compared with the amplitude $\Delta \lessdot \Delta_x^0$, it is calculated as followed [12]:

$$\varphi = \frac{\Delta}{(e+x)} = \varphi_0 . \sin(\omega t) \qquad (10)$$

$$\omega = arcsin\left[\frac{\Delta}{\varphi_0(e+x)}\right] \qquad (11)$$

Thus, the angular velocity:

$$\omega_z^{\varphi} = \frac{m l_c . L . \omega}{I_z^{(m)}} . \cos\left[arcsin\left(\frac{\Delta}{\varphi_0(e+x)}\right)\right] \qquad (12)$$

The angular velocity of G center when the vibrating bar is at the impact position [13]:

$$_{G}^{\varphi} = \omega_z^{\varphi} = \left(e + \frac{l}{2}\right) \qquad (13)$$

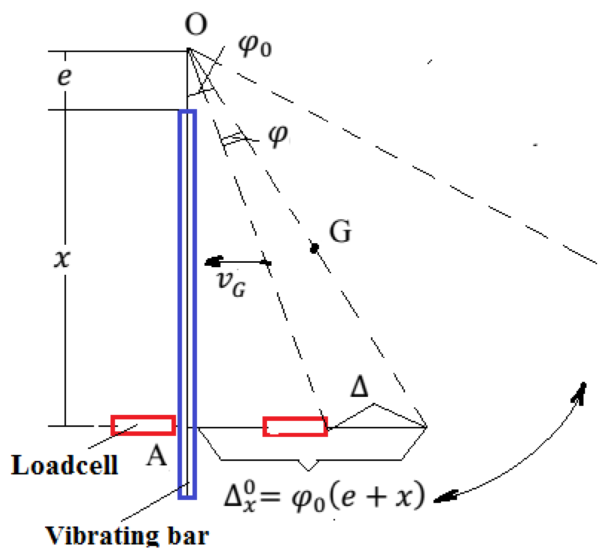

Fig. 5. Position analysis producing the force on the vibrating bar

During the very small impact of the vibration bar with the load cell $\Delta t <<<\approx 0.1\,(s)$, the impact force is depicted [14]:

$$F^{\varphi} . \Delta t . (e + x) = m_t . v_G^{\varphi} . \left(e + \frac{l}{2}\right) + I_z^G . \omega_z^{\varphi} \qquad (14)$$

$$\to F^{\varphi} = \frac{\left[m_t.\left(e+\frac{l}{2}\right)^2+I_z^G\right].\omega_z^{(\varphi)}}{\Delta t.(e+x)} \tag{15}$$

Thus, total force is:

$$F_{max} = \frac{\left[m_t.\left(e+\frac{l}{2}\right)^2+I_z^G\right].\frac{ml_C.L\omega}{I_z^{(m)}}}{\Delta t.(e+l)} \tag{16}$$

C. Design and manufacture coffee picking machines

Since the position of two eccentric balls is symmetrical and rotated in the same direction, in a rotation of the oscillating shaft, two forces of the same magnitude will be generated, in opposite directions. This oscillation travels to the vibrating rods mounted on the vibrating shaft causing them to oscillate, creating a thrust force to pick up the coffee berries, the shaft vibrates freely. Thus, it is necessary to determine the necessary driving force of the appropriate vibrating bars so that the fruit can be knocked out of the branch with the highest rate of loss, without damaging the coffee tree such as broken branches and many deciduous leaves, peeling bark. According to the structure of the vibrating box, position 2 eccentric balls mounted symmetrically through the center of the vibration shaft and 2 eccentric balls rotate in the same direction (via the chain transmission). The rotation of the motor decelerates through the belt transmission into the intermediate shaft. From the intermediate shaft will drive the eccentric balls mounting shaft to create 2 centrifugal forces (F). The centrifugal force produced by the two eccentric balls when lying on a straight line connecting the two centers and passing through the horizontal center of the vibrating axis will be directed apart and suppressed. When these two forces are positioned perpendicular to the seam through the center of the two eccentric balls, they will create a torque at the center of the vibrating shaft.

TABLE II
SPECIFICATIONS OF COFFEE PICKING MODEL

1	Engine	n=1440 rpm Speed reducer 1:5.
2	Belt actuator (i_d=2)	Trapezoidal belttype C, quantity 3, Diameter of driven belt, d1 = 90mm, Diameter of large belt, d2 = 180mm, Axis distance, a = 500mm, Belt length, L = 1400 mm
3	Chain actuator (i_x=1)	Axis distance, a = 450mm. The number of cogs in proactive and passive gear, Z = 27 The number of links, x =108 Diameter of chain wheel, d=164mm Chain pitch, P_c= 12,7mm
4	Distance between two eccentric balls	L = 900mm
5	Mass of the eccentric balls	M = 8.5kg
6	Vibrating bar	Length, l=800mm Diameter, D=12mm Material, steel C45
7	Oscillating bar	Length,l=1.800mm Diameter, D=80mm Material, steel C45

To measure parameters affecting coffee picking force, the coffee picking model has a built-in picking unit with specifications similar to the picking part on the industrial machines used, the model used electric motor. Three-phase power is processed for the current through the control circuit, the inverter is used to change the current frequency before entering the main motor. The rotary motor makes the structure of the picking bar oscillate back and forth creating the force of picking the fruit of the coffee when it affects the branches and fruits of the coffee. Changing the frequency of the current into the motor will cause the speed of the motor to change, resulting in the vibration of the vibrating bars changing, from which the picking force of the vibrating bars changes. On the vibrating box, there are two identical eccentric fruits, matching symmetrically and rotating in the same direction. According to this structure, it will create forced vibration for the vibrating shaft, resulting in vibration bars mounted on the vibration axis. In fact the vibrating bars when vibrating force will force the fruit to fall into the coffee, if the force is large enough, the coffee fruit will fall. In the experiment, the magnitude of this force will be measured by applying the impact bars to a force sensor mounted on the arm of the measuring device. These forces are measured horizontally on the bars, and are taken to the amplifier, into the receiver and signal converter and displayed on the laptop screen.

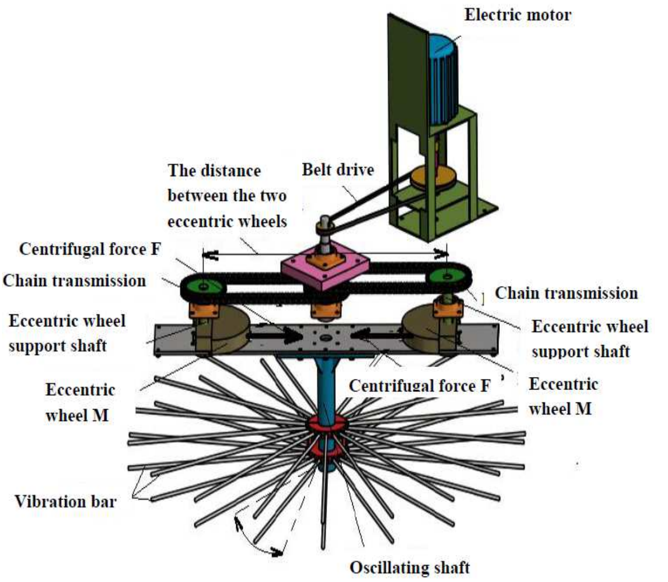

Fig. 6. Picking machine for coffee cherries after design and fabrication

III. RESULTS AND DISCUSSION

A. The relationship between frequency and picking force

The volume of the eccentric balls in the oscillating box has 3 types including 4.5kg, 6kg, and 8.5kg to change in turn during the experiment. At the same time changing the rotation speed of eccentric balls by changing the motor speed through the inverter, the selected frequencies including 30Hz, 32Hz, 33Hz, 34Hz, 35Hz, 40Hz, 42Hz, 44Hz, 45Hz and 47Hz. Selecting each volume in turn, each volume will rotate with 10 speeds of the motor corresponding to the 10 frequencies mentioned above. Each experiment will do 25 times, taking the average of 10 peaks in 1 experiment. Then plan the average value for each type of combination of eccentric mass and frequency.

TABLE III
EXPERIMENTAL RESULTS OF THE RELATIONSHIP BETWEEN THE FORCE WITH THE NUMBER OF VIBRATING SHAFT REVOLUTIONS, THE MASS OF ECCENTRIC BALLS

No	Rotation of vibrating bar (rpm)	Mass of eccentric balls (kg)	Force (N)
1	410	4.5	4.16
2	410	6	4.61
3	410	8.5	5.17
4	445	4.5	4.93
5	445	6	5.37
6	445	8.5	6.66
7	455	4.5	5.42
8	455	6	5.78
9	455	8.5	7.74
10	465	4.5	5.86
11	465	6	6.36
12	465	8.5	7.35
13	480	4.5	6.27
14	480	6	6.8
15	480	8.5	8.13
16	531	4.5	6.98
17	531	6	8.4
18	531	8.5	9.24
19	559	4.5	7.27
20	559	6	9.84
21	559	8.5	9.66
22	569	4.5	7.6
23	569	6	11.39
24	569	8.5	9.9
25	574	4.5	7.61
26	574	6	12.35
27	574	8.5	10.27
28	585	4.5	12.01
29	585	6	12.9
30	585	8.5	13.53

The selected eccentric-mass is 6 Kg, change frequency from 30 Hz to 47Hz in turn to establish conditions for simple regression between force and vibration axis speed is conducted, dependent variable Y: FORCE (Newton), independent variable X: ROTATION OF VIBRATING (rpm), linear model is thus indicated: Y = a+b*X.

Dependent variable: FORCE (N)

Independent variable: ROTATION OF VIBRATING BAR (rpm)

Linear model: Y = a + b*X

TABLE IV
RELATIONSHIP BETWEEN FORCE AND ROTATION OF VIBRATING BAR

	Least Squares	Standard	T	
Parameter	Estimate	Error	Statistic	P-Value
Intercept	-15.72321	1.820495	-8.636778	0.0000
Slope	0.0476348	0.003573184	13.33119	0.0000

TABLE V
ANALYSIS OF VARIANCE IN RELATIONSHIP BETWEEN FORCE AND ROTATION OF VIBRATING BAR

Source	Sum of Squares	Df	Mean Square	F-Ratio	P-Value
Model	80.37516	1	80.37516	177.72	0.0000
Residual	3.618044	8	0.4522555		
Total (Corr.)	83.9932	9			

Correlation Coefficient = 0.9782252
R-squared = 95.69246 percent
R-squared (adjusted for d.f.) = 95.15401 percent
Standard Error of Est. = 0.6724994
Mean absolute error = 0.4789804
Durbin-Watson statistic = 0.7931939 (P=0.0031)
Lag 1 residual autocorrelation = 0.4139605

In order to express the relationship between force and vibration axis speed, we have the regression equation as follows:

$$Y = -15.72321 + 0.0476348X \tag{17}$$

The P value in the table ANOVA <0.05 so the relationship between FORCE and ROTATION OF VIBRATING is statistically significant at the 95.0% confidence level. R-Squared statistics show the model matches 95.69246% of the Y variable.

TABLE VI
PREDICTED VALUES IN RELATIONSHIP BETWEEN FORCE AND ROTATION OF VIBRATING BAR

		95.00%		95.00%	
	Predicted	Prediction	Limits	Confidence	Limits
X	Y	Lower	Upper	Lower	Upper
410.0	3.807059	1.998424	5.615693	2.876356	4.737762
583.0	12.04788	10.30203	13.79373	11.24598	12.84978

TABLE VII
COMPARED REPLACEMENT MODELS

Model	Correlation	R-Squared
Double reciprocal	0.9965	99.30%
Reciprocal-Y logarithmic-X	-0.9954	99.08%
Logarithmic-Y squared-X	0.9949	98.99%
Reciprocal-Y square root-X	-0.9941	98.82%
Exponential	0.9940	98.81%
Logarithmic-Y square root-X	0.9929	98.59%
Reciprocal-Y	-0.9924	98.48%
Square root-Y squared-X	0.9915	98.30%
Multiplicative	0.9913	98.27%
Square root-Y	0.9880	97.62%
Reciprocal-Y squared-X	-0.9878	97.57%
S-curve model	-0.9865	97.31%
Double square root	0.9856	97.14%
Squared-X	0.9840	96.83%
Square root-Y logarithmic-X	0.9827	96.56%
Linear	0.9782	95.69%

The results are consistent with some curve models related to data. Among the suitable models, the linear model that gives R-Squared value is 95.69%. With this suitable model, the graph is shown in Fig. 7.

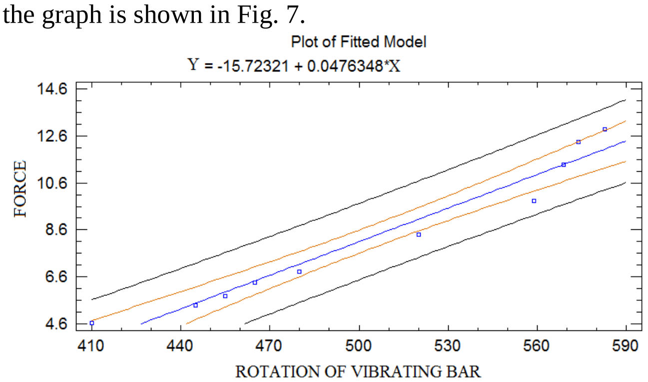

Fig. 7. The graph of the relationship between force and frequency

B. The relations between the eccentric balls mass and picking force

After selecting a fixed frequency of 34Hz, changing the mass of eccentric balls of 4.5 Kg, 6Kg and 8.5 Kg respectively, the achieved results are shown in Table 8.

TABLE VIII
EXPERIMENTAL RESULTS OF THE RELATIONSHIP BETWEEN FORCE AND THE MASS OF ECCENTRIC BALLS

No	Rotation of vibrating bar (rpm)	Mass of eccentric balls (kg)	Force (N)
1	465	4.5	5.86
2	465	6	6.36
3	465	8.5	7.35

Establishing conditions for simple regression between force and vibration axis speed is conducted, dependent variable Y: FORCE (N), independent variable X: ECCENTRICITYMASS (kg), linear model is thus indicated: Y = a+b*X.

Dependent variable: LUC (N)

Independent variable: ECCENTRICITYMASS (kg)

Linear model: Y = a + b*X

TABLE IX
RELATIONSHIP BETWEEN FORCE AND ECCENTRICITY MASS

	Least Squares	Standard	T	
Parameter	Estimate	Error	Statistic	P-Value
Intercept	4.14898	0.108731	38.15821	0.0167
Slope	0.374898	0.01661355	22.5658	0.0282

TABLE X
ANALYSIS OF VARIANCE IN RELATIONSHIP BETWEEN FORCE AND ECCENTRICITY MASS

Source	Sum of Squares	Df	Mean Square	F-Ratio	P-Value
Model	1.147813	1	1.147813	509.22	0.0282
Residual	0.002254082	1	0.002254082		
Total (Corr.)	1.150067	2			

Correlation Coefficient = 0.9990195

R-squared = 99.804 percent

R-squared (adjusted for d.f.) = 99.60801 percent

Standard Error of Est. = 0.04747717

Mean absolute error = 0.02557823

Durbin-Watson statistic = 2.959184 (P)

Lag 1 residual autocorrelation = -0.6530612

In order to express the relationship between force and mass of the eccentric balls, the regression equation is presented as follows:

$$Y = 4.14898 + 0.374898X \qquad (18)$$

The P value in the table of ANOVA <0.05, the relationship between Y and X is statistically significant at the confidence level of 95.0%. R-Squared statistics show that the model matches 99.80% variable Y.

TABLE XI
ANALYSIS OF VARIANCE WITH LACK-OF-FIT

Source	Sum of Squares	Df	Mean Square	F-Ratio	P-Value
Model	1.147813	1	1.147813	509.22	0.0282
Residual	0.002254082	1	0.002254082		
Lack-of-Fit	0.002254082	1	0.002254082		
Pure Error	0.0	0			
Total (Corr.)	1.150067	2			

TABLE XII
PREDICTED VALUES IN RELATIONSHIP BETWEEN FORCE AND ECCENTRICITY MASS

		95.00%		95.00%	
	Predicted	Prediction	Limits	Confidence	Limits
X	Y	Lower	Upper	Lower	Upper
4.5	5.83602	5.039154	6.632887	5.315367	6.356674
8.5	7.335612	6.502299	8.168926	6.760725	7.910499

TABLE XIII
COMPARED REPLACEMENT MODELS

Model	Correlation	R-Squared
Double squared	1.0000	99.99%
Reciprocal-Y	-0.9999	99.98%
Exponential	0.9999	99.98%
Square root-Y	0.9996	99.91%
Squared-X	0.9993	99.87%
Linear	**0.9990**	**99.80%**
Square root-Y squared-X	0.9987	99.75%

Table of compared replacement models shows consistent results of some curved models to the data. Among the suitable models, the linear model brings R-Squared value to 99.80%. With the appropriate model, this model has a graph, as shown in Fig. 8.

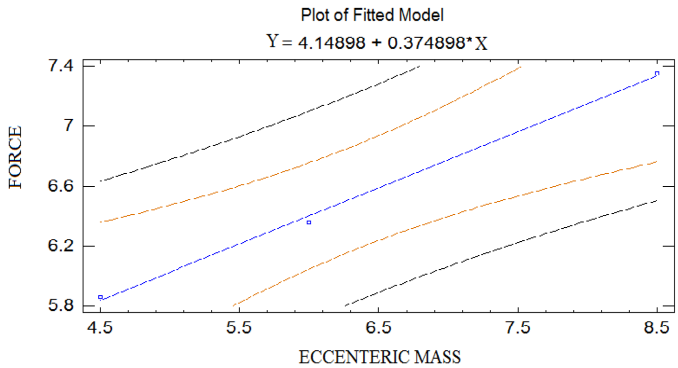

Fig. 8. The graph of the relationship between force and eccentric mass

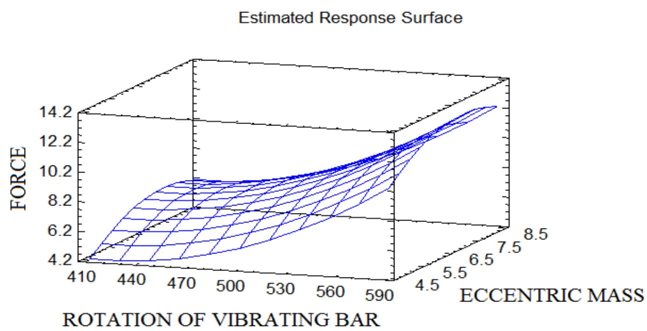

Fig. 9. The graph of the relationship between force and eccentric mass, rotation of vibrating bar

The results of the multicast regression model are suitable to describe the relationship between (Y: force) and 2 independent variables (X1: eccentric balls mass and X2: vibrating rotation) with equations of the form:

$$Y = 24.9336 - 0.1709803X_1 + 4.829996X_2 + 0.000203129X_1^2 + 0.0002776231X_1X_2 - 0.3432381X_2^2 \quad (19)$$

Based on the values in Table 4, the experimental design (DOE-Design of Experiment) is obtained as shown in Fig. 9.

IV. CONCLUSIONS

The relationship between factors such as the frequency of the picking bars, the mass of eccentric balls with the generated force of the pickers is proportional to each other. When increasing the frequency for the motor, or increasing the mass of the eccentric balls, the force on the picking bars also increases. This suggests that it can adjust the picking force in accordance with coffee in different regions, or for coffee gardens with different ripening rates. The result of measuring the average breaking force of the red coffee is 5 Newtons, the green fruit is 6 Newtons and to pull off the coffee leaf stalks is 14 Newtons with the experiment determining the force of picking coffee fruit on the coffee picking model. It is possible to select the rotational speed of eccentric balls from 480 to 574 rpm to generate a force of 7 to 10 Newtons with an eccentric mass of 8.5 kg. In each region of the world growing coffee, the force of picking will be different by many factors. This study contributes to the industrial-oriented coffee harvest for the future in DakLak province, which can then be replicated in other regions in Vietnam.

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