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Design and Evaluation of Grapes Crusher De-Stemmer Machine for Small to Medium-Scale Grape Producers

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Abstract

This research presents the design and evaluation of a grape crusher-destemmer machine specifically developed for small to medium-scale grape producers. The objective was to enhance processing efficiency, reduce production costs, and improve the quality of grape products in the local wine industry. The machine was designed and developed at TEMDO and was rigorously tested for key performance metrics, including crushing efficiency, destemming accuracy, throughput capacity, and juice yield. Results indicated an average crushing efficiency of 92%, destemming accuracy of 96% and SD of 1.14, and a throughput of 800 kg of grapes per hour, all meeting or exceeding established targets. The juice yield averaged 70.35 % and a standard deviation of 0.772, demonstrating the machine's effectiveness in maximizing extraction while minimizing waste. Economic analysis revealed that the machine, constructed using locally sourced materials, is a viable investment for grapes processors, with an estimated return on investment within two to three years. The study highlights the potential for the machine to enhance the local grape processing sector, promote sustainable practices, and support economic growth in rural areas. Recommendations for future improvements include automation features, further testing with diverse grape varieties, and establishing user training programs. This work contributes to the ongoing efforts to modernize agricultural practices in Tanzania and supports the development of a competitive local grape industry.

Keywords: Crusher; De-stemmer; Grapes; Juice; Producers.

1. Introduction

Agriculture is vital to Tanzania's economy, contributing significantly to employment, food security, and export revenue. Grape production holds considerable potential among the various cultivated crops, especially in regions such as Dodoma, the country's wine-producing hub [1]. Although grape cultivation has grown in recent years, post-harvest processing machines are not common in grape-growing areas, limiting the growth of the local wine industry and other grape-derived products. Although imported grape crusher-destemmer machines are available, they are often too costly for many small-scale producers in Tanzania. Additionally, the high maintenance requirements, lack of readily available spare parts, and limited compatibility with the specific types of grapes grown in Tanzania make these machines less than ideal for local conditions. Local manufacturers and entrepreneurs can develop a machine that addresses these challenges by customizing the design to the types of grapes grown in Tanzania and the available resources of grape processors.

The most common grape varietals in the region are Chenin Blanc, Syrah, and Cabernet Sauvignon [2]. Dodoma has two harvest seasons annually, starting from February to March and August to September. However, demand for grapes domestically seems to be very high. This is because a significant amount of table grapes and concentrated bulk wine are sometimes imported, mostly during grape off-seasons [3].

The primary step in grape processing is the separation of grapes from their stems and crushing them to extract juice. In modern wineries, this is achieved by using specialized machines known as grape crusher-destemmer [4, 5]. This machine facilitates the efficient crushing and destemming of large volumes of grapes, essential for producing high-quality wine. However, the high cost of imported crusher-destemmer machines and a lack of locally tailored designs present challenges for Tanzania's small and medium-sized grape producers. Developing a cost-effective, efficient, and easy-to-maintain grape crusher-destemmer machine could thus meet local needs and support the growth of Tanzania's grape-processing sector.

The destemming process involves separating grapes from their stems and other herbaceous parts immediately after harvesting [6]. While traditional methods may help ensure the selection of high-quality grapes and are crucial for reducing the presence of bitter tannins during fermentation, these practices often fall short in efficiency. The presence of stems in the must can lead to undesirable bitterness in the final product, which is detrimental to wine quality [7].

While imported grape crusher-destemmer machines are available, they are often too costly for many small-scale producers in Tanzania. Additionally, the high maintenance requirements, lack of readily available spare parts, and limited compatibility with the specific types of grapes grown in Tanzania make these machines less than ideal for local conditions. Local manufacturers and entrepreneurs are thus presented with a unique opportunity to create a machine that addresses these challenges by tailoring the design to the types of grapes grown in Tanzania and the resource constraints of Tanzanian grape processors.

Although Tanzania is the second largest wine producer in Sub-Saharan Africa after South Africa, it is still facing some difficulties in the selling market due to the low technological development of the existing wine industry [1, 8]. Hence, it lacks much international recognition. As a result, farmers are struggling with poverty and lack the courage to increase production.

This research aimed to design, fabricate, and evaluate a locally optimized grape crusher-destemmer machine that is affordable, easy to maintain, and efficient. By doing so, the machine will serve as an accessible solution to small and medium-sized grape processors and enable processors to improve their production, productivity, and competitiveness in the grape and wine industries. This study investigated the design requirements for a grape crusher-destemmer machine that meets the needs of grape farmers and processors. It developed a functional prototype that can effectively crush and destem locally grown grape varieties, assessed the machine’s performance through experimental trials, evaluated parameters such as crushing efficiency, destemming accuracy, throughput, and grape juice yield and determined the cost implications and economic feasibility of local manufacturing and maintenance of the machine.

2. Methodology

2.1. Machine Components

This design comprises several sub-assembly components, all assembled to form a complete machine (Figure 1). Those sub-assembly components assembled to form the machine are: *Hopper*- this is the part where grapes to be de-stemmed are introduced within and emptied from the bottom to the sieve through a rectangular hole using a feeder screw; *Feeder screw*- this is typically a helicoidal conveyor screw necessary to convey material from the hopper to the sieve (inner chamber of the machine); *Destemming blades*- necessary to separate stems from the berries as the grapes reach the sieve to pass through; *Sieve*- this part of the machine is responsible for separating grape berries from their stems. For this machine, one hole of the sieve is given a diameter of 22 mm; *Grape’s outlet chamber*- this is an opening through which de-stemmed grapes come to be discharged; *Sprockets and chain*- to transfer rotating motion from the de-stemming blade to the feeder screw in the hopper. This is essential because the friction of a drive belt would be insufficient to transfer that power; *Belt and pulleys*- responsible for transferring rotating motion from the motor pulley to the conveyor and de-stemming blade; *Bottom cover*- for the temporary storage of de-stemmed grapes before coming out of the machine; *Tyres*- to provide the necessary traction on the surface on which the machine will rest; *Motor*- driving all the machine elements in the machine.

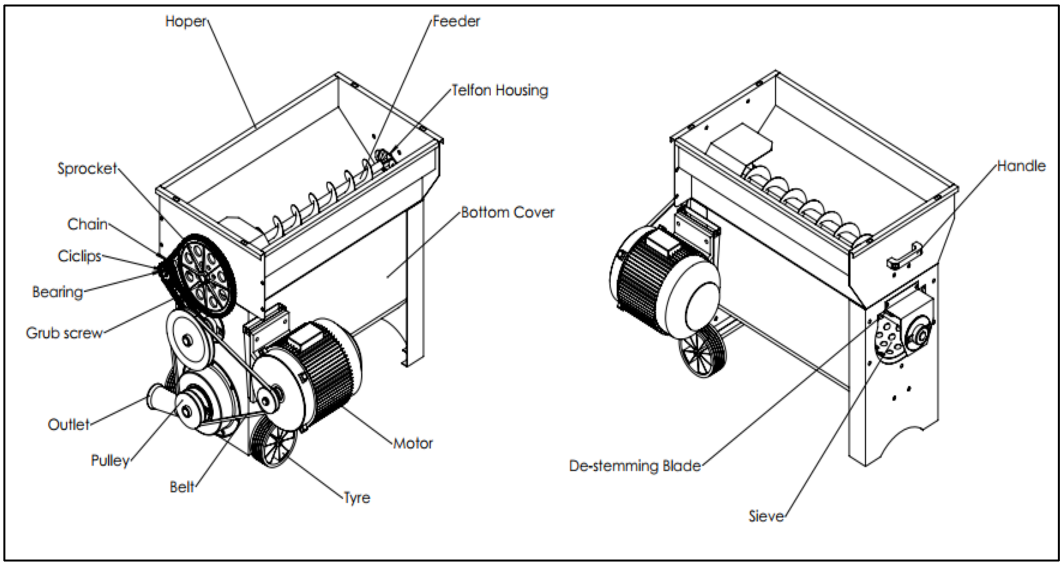

Figure-1. Parts of grapes de-stemmer machine

2.2. Working Principle of the Machine

Due to economic considerations, the machine was designed to load grapes bunches with a total mass of 28.16 kg retained in the hopper with an average size of 72,210 cm^3. A 1.5 kW electric motor powered the machine; in this case, the grapes were fed into the hopper, where they came in contact with the feeder screw, which was in charge of driving grapes bunches to the sieve by passing through the de-stemming blade, which is there to detach the grapes berries from its stem. An adequate size of the sieve hole was considered in the design to avoid de-stemmed grapes unfit to pass through those holes. After passing through the sieve, the grapes then reach the conveyor screw, which is essential importance to discharge the de-stemmed grapes out of the machine for further process of wine-making to take place. In other ways de-stemming blade, feeder, and conveyor screw were all driven by the same motor; hence a V- belt and pulleys were designed accurately to ensure the appropriate operational speed of conveyance.

2.3. Design Considerations

Based on customer needs and literature related to the grapes de-stemmer machine, the following product design specifications were considered to obtain high efficiency, reliability, and quality products from the machine (Table 1).

Table-1. Design specifications

S/n	Specification	Details
1.	Usage	For small-scale industrial use
2.	Rated load capacity (Performance)	Max 28 kg of grapes at once
3.	Materials	Stainless Steel
4.	Operation type	An electrical motor is used
5.	Safety	All moving parts are covered and sharp edges removed
6.	Product Cost	3,500,000/=
7.	Tyres	Solid tyre instead of pneumatic tyre
8.	Installation	No special tools are required during installation

2.4. Design Computation of different Machine Elements

Essential design calculations of different machine elements involved in the machine were done systematically to determine the size and hence be able to select the standard size of those particular components. This was done with the aid of the results and established formulae in the design analysis as follows:

2.5. Belts and Pulleys

In the V-belt drive design, the following parameters were considered: sheave sizes, belt length, belt size, and number of belts, as shown in Figure 2.

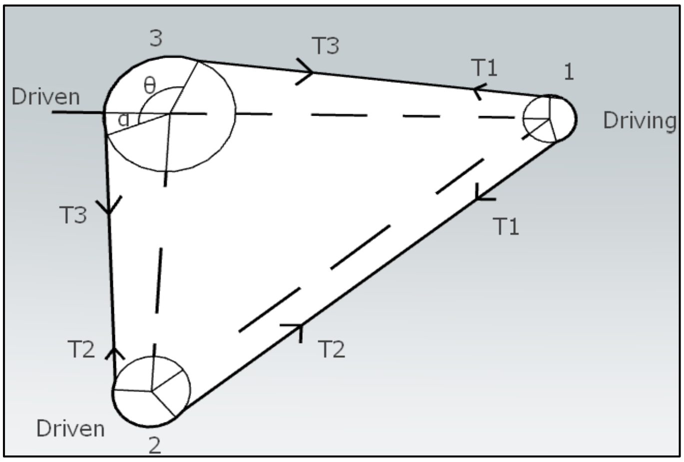

Figure-2. Arrangements of pulleys

The V-belt drive was selected since the three pulleys are close to each other and the system involves light-duty power transmission. However, for a V-belt to overcome slippage during power transmission on the belts and pulleys, the pulley ratio should be 3:1 or less.

Determination of the speed ratio of the pulleys

Taking the diameter of the motor pulley (D_1) = 85mm, Then reduce the speed of the driving pulley (N_1) from 1360 rpm to 1000 rpm the speed of the conveyor pulley (N_2)

Let; D_2 = Diameter of the driven pulley conveyor

D_3 = Diameter of driven destemming pulley

Thus, from the formula

SR = $\frac{N_1}{N_2} = \frac{1360}{1000} = 1.36$ ……………………………………………………………………………(1)

Since 1.36 < 3 Hence, the speed ratio condition is satisfied.

Then; $\frac{D_2}{D_1} = 1.36$

$D_2 = 1.36 D_1 = 1.36 \times 85 = 115.6$

∴ The diameter of a driven conveyor pulley is 115.6 mm

For practical purposes take the diameter of the conveyor pulley = 116 mm

Similarly

Reduce the speed of the driven conveyor (N_2) from 1000 rpm to 565 rpm the speed of destemming blade pulley (N_3)

SR = $\frac{N_2}{N_3} = \frac{1000}{565} = 1.77$

Since 1.77 < 3 Hence speed ratio condition is satisfied

Then; $\frac{D_3}{D_2} = 1.77$

$D_3 = 1.77 D_2 = 1.77 \times 116 = 205.32$

∴ The diameter of the driven destemming blade pulley is 205.32mm

For practical purposes take the diameter of the destemming blade pulley = 205.32 mm

2.5.1. Selection of V-Belts for Power Transmission

The following procedure was used to choose the type of belt section and the number of V-belts needed to drive the machine and transmit the motor power.

(i) Selection of the V-belt section: this was done by determining the drive's design horsepower. The Mechanical Power Transmission book by William J. Patton states that the service factor is 1.2, and the motor's rated horsepower is 2 hp.

Then the *design horsepower = Motor horsepower x service factor* …………………………………(2)

This gives us *2 hp x 1.2= 2.4 hp*

Then, from the graph of the rpm of the small sheave vs. rated horsepower, as illustrated in Figure 4, 5, 6, 7 8, of Mechanical Power Transmission by Patton [9], the intersection of the 2.4 hp and 1360 rpm is in the region of the A-section. Therefore, belt section (class) A was selected.

(ii) Computation of the belt speed: the peripheral velocity of the belt on the driving pulley is given by

$$V_1 = \frac{\pi D_1 N_1}{60} \text{ (m/s)} \quad \text{……………………………………..…………………..(3)}$$

Where: D_1 = diameter of the drive pulley (m)

N_1 = motor speed (rpm),

V_1 = The peripheral velocity of the belt on the drive motor pulley

$$V_2 = \frac{\pi D_2 N_2}{60} \text{ (m/s)} \quad \text{………………………………………………………….(4)}$$

Where D_2 = diameter of the driven conveyor pulley (m)

N_2 = speed of the driven conveyor shaft (rpm)
V_2 = The peripheral velocity of the belt on the driven conveyor pulley

$$V_3 = \frac{\pi D_3 N_3}{60} \text{ (m/s)} \quad \ldots\ldots(5)$$

Where D_3 = diameter of the driven conveyor pulley (m)
N_3 = speed of the driven conveyor shaft (rpm)
V_3= The peripheral velocity of the belt on the driven conveyor pulley
But when there is no slip, $V_1 = V_2 = V_3$
Then the peripheral velocity of the drive pulley of the machine is: $V_1 = \frac{\pi D_1 N_1}{60} = \frac{\pi \times 0.085 \times 1360}{60} = 6.05$ m/s

Therefore $V_1 = V_2 = V_3 = 6.05$ m/s

According to ISO: 2494 – 1974, the following are the parameters of the selected V-Belt class A: top width (b) = 13 mm, thickness (t)= 8 mm and weight per meter length [N/m] = 1.06 N/m ≡ 0.106 kg/m.

(iii) Materials of the selected belt: the material of the selected belt is determined from the relation of mass belt per length as follows;

mass of the belt per meter length (m) = area x density

$$m = b \times t \times \rho \quad \ldots\ldots(6)$$

$0.106 = 0.013 \times 0.008 \times \rho$

Density (ρ) = 1019.2 $^{kg}/_{m^3}$

From a textbook of machine design by Gupta in Table 18.1 of belt materials, the calculated density is around 1000 $^{kg}/_{m^3}$ which is the density of the leather material

(iv) The required belt length: three considerations were taken into account to determine the total length of the belt drive.

First consideration involved the motor drive pulley (d_1) and conveyor pulley (d_2). At this case suppose the value of C = 16.5in

Length of the belt is given by: $L_1 = 1.57(d_1 + d_2) + 2(C) + \frac{(d_2 - d_1)^2}{4C}$ ……………………….(7)

Where: d_1 = diameter of the drive pulley [in]
d_2= diameter of the conveyor pulley [in]
C = pulleys center distance [in]

Thus; $L_1 = 1.57(3.3+4.5) + 2(16.5) + \frac{(4.5-3.3)^2}{4(16.5)} = 45.27$ in

- The second consideration involved the motor drive pulley (d_1) and de-stemming blades pulley (d_3). In this case, suppose the value of C = 13.8in

The length of the belt is given by: $L_2 = 1.57(d_1 + d_3) + 2(C) + \frac{(d_3 - d_1)^2}{4C}$ …………………..(8)

Where: d_1 = diameter of the drive pulley [in]
d_3 = diameter of the de stemming blades pulley [in]
C = pulleys center distance [in]

Thus; $L_2 = 1.57(3.3+8.1) + 2(13.8) + \frac{(8.1-3.3)^2}{4(13.8)} = 45.92$ in

- The third consideration involved the conveyor pulley (d_2) and de-stemming blades pulley (d_3). In this case, suppose the value of C = 9.4in

The length of the belt is given by: $L_3 = 1.57(d_2 + d_3) + 2(C) + \frac{(d_3 - d_2)^2}{4C}$ …………………..(9)

Where: d_2 = diameter of conveyor pulley [in]
d_3 = diameter of the de stemming blades pulley [in]
C = pulleys center distance [in]

Thus; $L_3 = 1.57(4.5+8.1) + 2(9.4) + \frac{(8.1-4.5)^2}{4(9.4)} = 38.93$ in

Now the total length of the belt (L) = $\frac{1}{2}\sum_1^3 L$ ……………………………………………………………(10)

$L = \frac{1}{2}(45.27+45.92+38.93)$ in = 65.06 in

∴From standard size of belt choose the belt length close to the calculated value which is 67in

From this point, the summary of V-Belt selected the references of figures, and the Table was taken from the book *Mechanical Power Transmission by William J. Patton.*

(v) Computation of power rating per belt:

The equivalent diameter (d_e) = Pitch diameter of the small pulley × Correction Factor…………………(11)
Since the Speed ratio is 1.77 the correction factor factor was taken as 1.11
$d_e = 3.3 \times 1.11$
$d_e = 3.66$

The graph of A-belts shows that the equivalent diameter of 3.66 and belt speed of approximately 1800fpm give a maximum horsepower rating per belt of 2.1 hp.

(vi) Number of V-Belts required;

For an open belt drive

$\sin\alpha = \frac{d_3 - d_1}{2x}$ ………………………………………………………………….……..(12)

$\alpha = Sin^{-1}\left(\frac{d_3 - d_1}{2x}\right) = Sin^{-1}\left(\frac{8.1 - 3.3}{2 \times 13.8}\right) = 10.02°$

But the angle of the wrap

$\theta = 90 + 2\,\alpha = 110°$ …………………………………………………….……….(13)

The correction factor for an arc contact of 110° is 0.787

Hence the final corrected horsepower per belt is

2.1 hp × 1.0 × 0.71 = 1.6527 hp

Since 2.4 hp must be transmitted then

Number of v-belts = $\frac{Design\ power\ to\ be\ transmitted}{corrected\ horse\ power/belt} = \frac{2.4}{1.7} = 1.4$ …………………………………..(14)

Hence take the number of belts = 1

2.5.2. Mathematical Analysis of the V-Belt

Let us consider the following transmission system: -

T_1 = tension in the tight side of the belt

T_2 = tension in the slack side of the belt

V = peripheral velocity of the belt (m/s)

m = mass of the belt per meter length

T_c = centrifugal tension

This centrifugal tension is given by:-

$T_c = mV^2$ ………………………………………………………………..(15)

But weight per meter length of the selected belt [N/m] = 1.06 N/m = 0.106 kg/m.

Then $T_c = mv^2 = 0.106 \times (6.05)^2 = 3.8799$ N

The maximum tension in the belt, $T = \sigma\ x\ a$ ……………………………………………………..(16)

Where: σ = allowable tensile stress of the belt materials

a = belt cross-section area (b×t)

Therefore, the belt's cross-section area (a) is 13 mm x 8 mm = 104 mm^2.

The allowable tensile stress of the belt material = 2.5 MPa = 2.5 N/mm^2

Thus T = 2.5 x 104 = 260 N

2.5.3. Sprocket's Design

Due to frictional factors, this mechanical drive must transmit power from one shaft to another when the belt drive is insufficient.

In selecting sprockets for chain drive, the sprockets with 12 teeth were selected simply because their tangential forces are minimal, the chain articulation around the sprocket is smooth with little or no chordal effect, so they are cost-effective.

Let D = Diameter of pitch circle, N = Number of teeth, P = Chain pitch, O.D = Outside diameter, P.C.D = Pitch Circle Diameter, d_1 = Roller diameter

Therefore;

$D = \frac{P}{\sin\frac{180°}{N}}$ & O.D = P.C.D + 0.6…………..0.7d_1 for N = 12……25)….……(17)

O.D = P.C.D + 0.7…………..0.8d_1 for N>25 ……………………...(18)

Using the following data

Driving sprocket revolution (n_1) = 564 rpm

Number of teeth of the small sprocket selected (N_1) = 12

Let; VR = Velocity Ratio

2.5.4. Determination of the Number of Teeth of the Idle Sprocket

$V.R = \frac{n_1}{n_2} = \frac{N_2}{N_1}$ ……………………………………………………………………………….(19)

Where by; Number of teeth of the small sprocket (N_1) = 12

Driving sprocket revolution (n_1) = 564 rpm

Now the driving sprocket revolution was reduced from n_1= 564 rpm to n_2= 520 rpm (the revolution of driven idle sprocket)

Then Mathematically.

$N_2 = \frac{n_1}{n_2} \times N_1$

$N_2 = \frac{564}{520} \times 12 = 13.01 \approx 13$

For practical purposes the number of teeth N_2 is 13

Similarly, the number of teeth of the large sprocket was calculated as;

$V.R = \frac{n_3}{n_2} = \frac{N_2}{N_3}$ ……………………………………………………………………………………(20)

Whereby; Number of teeth of the idle sprocket (N_2) = 13
Driven idler sprocket revolution (n_2) = 520 rpm
The driven idler sprocket revolution was reduced from n_2= 520 rpm to n_3= 106 rpm the revolution of the large sprocket.
Then mathematically,

$N_3 = \frac{n_2}{n_3} \times N_2$

$N_2 = \frac{520}{106} \times 13 = 63.77 \approx 64$

∴For practical purpose the number of teeth N_2 is 64
Determination of sprocket size
Case 1: Small Sprocket

- Pitch Diameter of Small sprocket

$D_1 = \frac{12.7}{\sin\frac{180°}{12}} = 49.07$ mm

Thus, for practical purposes, the pitch diameter of a small sprocket will be 49.1 mm

- Outside diameter

From the formula given by:
Outside diameter = P.C.D + $0.65d_1$
= 49.07 + 0.65(8.51) = 54.6 mm
∴The outside diameter of the small sprocket is 54.6 mm
Case 2: Idle sprocket

- The pitch diameter of the idler sprocket

$D_2 = \frac{12.7}{\sin\frac{180°}{13}} = 53.07$mm

Thus, for practical purposes, the pitch diameter of the idle (medium) sprocket will be 53.1 mm

- Outside diameter

From the formula given by:
Outside diameter = P.C.D + $0.65d_1$
= 53.07 + 0.65(8.51) = 58.6 mm
∴The outside diameter of the idle sprocket is 58.6 mm
Case 3: Large sprocket

- The pitch diameter of a large sprocket

$D_3 = \frac{12.7}{\sin\frac{180°}{64}} = 258.83$mm

Thus, for practical purposes, the pitch diameter of a large sprocket will be 258.8 mm

- Outside diameter

From the formula given by:
Outside diameter = P.C.D + $0.75d_1$
= 258.83 + 0.75(8.51) = 265.2 mm
∴The outside diameter of the larger sprocket is 265.2 mm

2.5.5. Design of Sprocket Teeth for Roller Chains

The cross-section of sprocket teeth is presented in Figure 3.

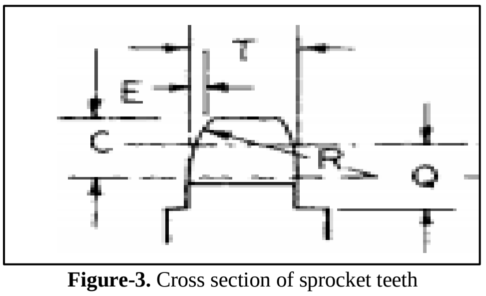

Figure-3. Cross section of sprocket teeth

Since it is a single-strand roller chain
Then; Strand roller chain (T) = 0.9W− 0.006 ………………………………………………………….(21)
But according to the roller chain selected W = 8.51
T = 0.9(8.51) − 0.006 = 7.653mm
For the case of Letter C is given by;
C = 0.5p = 0.5(12.7) = 6.35 mm ………………………………………………………… (22)
Considering Letter E which is given by;
$E = \frac{1}{8}p = \frac{1}{8}(12.7) = 1.5875\ mm$ ……………………………………………………….…….. (23)
Considering Letter $R_{(min)}$ which is given by;
$R_{(min)}$ = 1.063p = 1.063×12.7 = 13.5001 mm ……………………………………………. (24)

Lastly Letter Q was considered which is given by;

$Q = 0.5p = 0.5x(12 \cdot 7) = 6.35\ mm$ .... (25)

2.5.6. Chain Drive

According to the Catalogue of Chain Drives table from the Standard Handbook for Mechanical Engineers [10] the horsepower rating is 1.81, as indicated by the belt calculations, nearly equal to the revolutions of the given shaft.

Design Power = Rated power × Service Factor (Table 2) .... (26)

Table-2. Service factors

For electric load and shock load, (K_1)	1.3
Relatively clean and moderately temperature, (K_2)	1.0
Range of working hours 8-10hrs, (K_3)	1.0

But Service Factors (Ks) = $K_1K_2K_3$ .... (27)

Then Design HP = 1.81×1.3×1.0×1.0

= 2.353 hp

No. of strands = $\frac{Power\ of\ the\ motor}{Design\ Horse\ Power} = \frac{2}{2.353} = 0.85 \approx 1$ .... (28)

∴The power is reduced to accommodate a single-strand chain since the reduction is achieved using sprockets and pulleys.

Selecting the pitch of the chain drive: Use the chart for drive chain selection from the Mechanical Power Transmission by William J. Patton to select chain number 40 (0.5" or 12.7mm pitch), as shown in Table 3.

2.6. Parameters of the Selected Chain Drive

ISO Chain No.40 ≡ 08B

Table-3. Parameters of the chain drive

Pitch (p)	Roller diameter (d_n)	Width between inner plates (b_1)	Transverse pitch (P_1)	Breaking Load (kN)		
				Simple	**Duplex**	**Triplex**
12.7	8.51	7.75	13.92	17.8	31.1	44.5

Now determining the pitch circle diameter and pitch line velocity of sprockets

The pitch circle diameter of a small sprocket

$d_1 = p\text{Cosec}\left(\frac{180}{N_1}\right)$ .... (29)

$d_1 = 12.7\times\text{Cosec}\left(\frac{180}{12}\right) mm = d_1 = 49.06\ mm = 0.049\ m$

The pitch circle diameter of the medium sprocket

$d_2 = p\text{Cosec}\left(\frac{180}{N_2}\right)$ .... (30)

$d_2 = 12.7\times\text{Cosec}\left(\frac{180}{13}\right) mm = d_2 = 53.07\ mm = 0.053\ m$

The pitch circle diameter of a large sprocket

$d_3 = p\text{Cosec}\left(\frac{180}{N_2}\right)$ .... (31)

$d_3 = 12.7\times\text{Cosec}\left(\frac{180}{64}\right) mm = d_3 = 258.83\ mm = 0.2588\ m$

The pitch line velocity of the small sprocket

$v_1 = \frac{\pi\, d_1 n}{60}$ .... (32)

$v_1 = \frac{\pi\times0.049\times564}{60}$

$v_1 = 1.44\ m\ /\ s$

The average velocity of the chain drive is 1.44m/s

Since all three sprockets are connected in the chain then; $v_1 = v_2 = v_3$

2.7. Determination of the Load on the Chain Drive

From the mathematical relation

$w = \frac{Rated\ power}{Pitch\ line\ velocity} = \frac{1.81}{1.44}$ kN $= 1.257$kN ....(33)

Factor of safety $= \frac{w_B}{w} = \frac{31}{1.257} = 25$ .... (34)

∴The condition is satisfied since the safety factor obtained is greater than the value indicated by Khurmi and Gupta [11].

Taking a centre distance of 40 times the pitches

∴Centre distance between the sprockets

= 40p = 40× 12.7 =508 mm

But to accommodate the initial sag in the chain drive, the value of the centre distance is reduced by 2 to 5 mm

Hence, the correct centre distance for both chain drives will be

$$x = 508 - 4 = 504\ mm$$

2.8. Determination of the Length of the First Chain Drive

The number of chain links is given by

$$K = \frac{N_1+N_2}{2} + \frac{2x}{P} + \left[\frac{N_2-N_1}{2\Pi}\right]^2 \frac{p}{x} \quad \text{.......... (35)}$$

$$K = \frac{12+13}{2} + \frac{2\times504}{12.7} + \left[\frac{13-12}{2\Pi}\right]^2 \frac{12.7}{504} = 12.5 + 79.37 + 0.000638 = 91.87$$

∴Length of the chain L = K.p = 92 × 12.7 =1169 mm

2.9. Determination of the Length of the Second Chain Drive

Similarly, the number of chain links is given by

$$K = \frac{N_2+N_3}{2} + \frac{2x}{P} + \left[\frac{N_3-N_2}{2\Pi}\right]^2 \frac{p}{x} \quad \text{.......... (36)}$$

$$K = \frac{13+64}{2} + \frac{2\times504}{12.7} + \left[\frac{64-13}{2\Pi}\right]^2 \frac{12.7}{504} = 38.5 + 79.37 + 1.66 = 119.5$$

∴Length of the chain L = K.p = 120 × 12.7 =1524 mm

2.10. Shaft Design

The following equation was used to obtain the diameter of the shaft subjected to twisting moment only:

$$\frac{T}{J} = \frac{\tau}{r} \quad \text{.......... (37)}$$

Where: T = twisting moment (torque) acting upon the shaft

J = polar moment of inertia of the shaft about the axis of rotation

τ = allowable torsional shear stress

r = distance from the neutral axis to the outermost fibre (r = d/2 where d is the shaft diameter)

$$d = 36.5\left(\frac{P}{N\tau}\right)^{\frac{1}{3}} = 36.5\left(\frac{1.5\ x\ 10^3}{106\ x\ 64.5}\right)^{\frac{1}{3}} \approx 22.07\ mm \quad \text{.......... (38)}$$

For practical purposes, a shaft diameter of 25 mm was selected.

The summary of the design calculations is shown in Table 4.

Table-4. Summary of the design calculations

S/N	Machine element	Parameter	Selected Value
1	Pulley	Diameter D_1 D_2 D_3	85 mm 116 mm 205.32 mm
2	Sprocket	Number of teeth N_1 N_2 N_3	12 13 64
		Pitch circle diameter $(POD)_1$ $(POD)_2$ $(POD)_3$	49.1 mm 53.1 mm 258.83 mm
		Outside diameter OD_1 OD_2 OD_3	54.6 mm 58.6 mm 265.2 mm
3	Chain drive	Number of strands Pitch selected p Centre distance x Length of the chain L_1 L_2	1 12.7 mm 504 mm 1169 mm 1524 mm
4	Belt	Belt class selected Materials of the belt Length of the belt Number of V belts	A Leather 67in 1
5	Shafts	Diameter ∅ Length of feeder shaft Length of destemming blade shaft Length of conveyor shaft	25 mm 920 mm 1060 mm 940 mm
6	Helicoidal Screw	Pitch p Max load capacity	80 mm 276.15N

		Number of revolutions Helix angle	8.5 13°
7	Motor	Power	1.5kW having 1360 rpm

2.11. Testing Protocols

A series of tests were conducted to evaluate the machine's performance, using locally grown grapes in simulated production conditions. The tests focused on measuring key performance indicators, including crushing efficiency, destemming accuracy, throughput, and juice yield.

2.12. Crushing Efficiency Test

The crushing efficiency was tested by running a set quantity of grapes through the machine, then examining the crushed output for intact grapes and seed damage. Crushing efficiency was calculated using the following formula:

$$Crushing\ efficiency = \left(\frac{Number\ of\ crushed\ grapes}{Total\ Number\ of\ grapes\ processed}\right)\ x\ 100\%$$

2.13. Destemming Accuracy Test

Destemming accuracy was assessed by measuring the proportion of stems removed from a batch of grapes processed through the machine. The following formula was used:

$$Destemming\ accuracy = \left(\frac{Number\ of\ detached\ grapes}{Total\ number\ of\ grapes\ processed}\right)\ x\ 100\%$$

2.14. Throughput Test

Throughput was measured by processing a set weight of grapes over a fixed time and recording the quantity processed per hour. The formula for throughput calculation is:

$$Throughput = \left(\frac{Total\ weight\ of\ grapes\ processed}{Total\ processed\ time\ (\ in\ hours)}\right)\ x\ 100\%$$

2.15. Juice Yield Test

Juice yield was assessed by measuring the volume of juice extracted from a set weight of grapes and calculating the yield percentage:

$$Juice\ yield = \left(\frac{Volume\ of\ juice\ collected}{Total\ number\ of\ grapes\ processed}\right)\ x\ 100\%$$

2.16. Data Analysis

The data from these tests were analyzed to evaluate the machine's performance against the design objectives. Descriptive statistics, including mean values and standard deviations, were used to summarize performance metrics. Comparative analysis was conducted between the machine's performance and industry benchmarks to assess its competitiveness.

2.17. Economic Feasibility Analysis

The final component of the methodology involved an economic feasibility analysis. This included calculating the machine's production cost, estimated maintenance expenses and projected savings for users. A cost-benefit analysis was conducted to determine the machine's viability for small and medium-scale grape producers.

3. Results and Discussion

3.1. Machine Performance Evaluation

Each component of the grape crusher-destemmer machine was tested separately and as part of the integrated system. The primary performance metrics assessed were crushing efficiency, destemming accuracy, throughput, and juice yield. The fabricated machine is shown during work in Figure 4.

Figure-4. De stemmer machine (a) after fabrication and (b) during work

3.2. Crushing Efficiency

Figure 5 shows that the machine achieved an average crushing efficiency of 92%, which meets the target efficiency threshold of 90%. This was consistent across multiple tests, suggesting stable performance across different batches. The results indicate that the screw spacing and pressure applied were adequate in breaking the grape skins while preserving the seeds, preventing the release of unwanted bitter compounds into the juice. This is important in maintaining the juice quality, which is essential for producing high-quality wine and grape products. However, slight adjustments to roller spacing improved efficiency by an additional 2%, demonstrating the importance of fine-tuning based on grape size and variety.

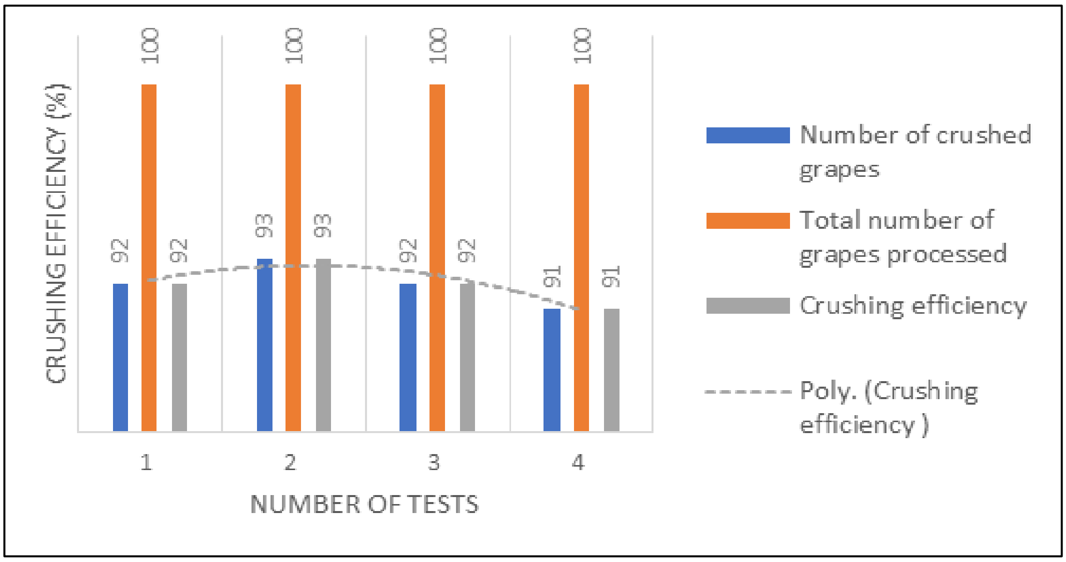

Figure-5. Crushing efficiency of grapes

3.3. Destemming Accuracy

Figure 6 shows the results of destemming, with different rounds giving an accuracy of 95, 94.5, 97.5, and 95.5 %. These results gave an average accuracy of 96%, surpassing the target of 95%. The results indicate the data is relatively consistent, with a slight standard deviation (1.14) showing minimal variation in destemming accuracy. This was achieved by adjusting the drum speed and the angle of the beater paddles within the drum. The high destemming accuracy indicates that the rotating drum and beater configuration effectively detached the grapes from the stems. Proper destemming is crucial to avoid the inclusion of bitter compounds from the stems, which can affect the flavour profile of the end product. The consistency in destemming performance across trials suggests that the design is well-suited for various grape types. Similar results of destemming grapes were obtained by Coetzee and Lombard [12], Pawar, *et al.* [13].

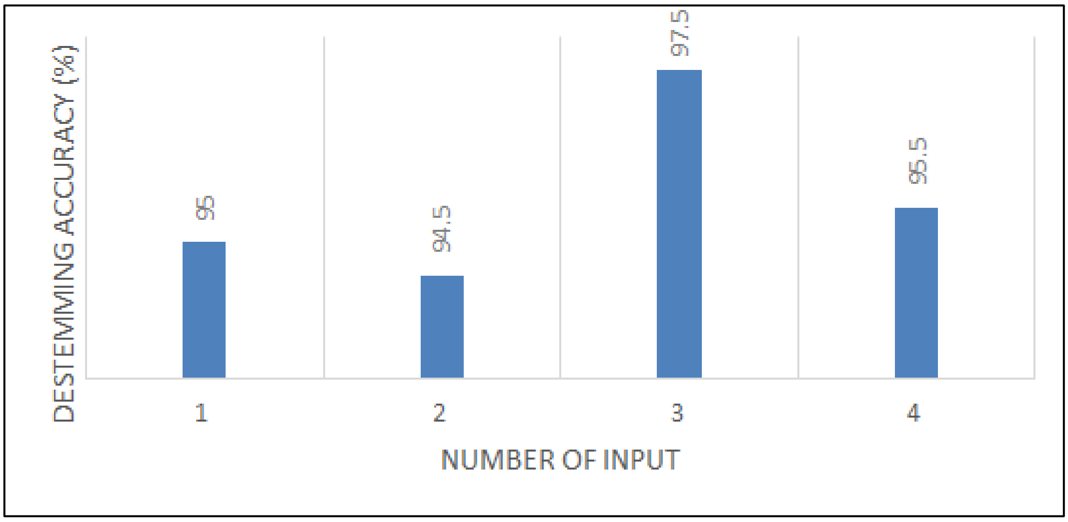

Figure-6. Destemming accuracy

3.4. Throughput

The machine processed 20 kg of grapes for 1 minute and 30 minutes, an average of 800 kg of grapes per hour as shown in Figure 7. The throughput is suitable for small to medium-sized grape producers, allowing them to process large volumes efficiently during harvest. This throughput capability will enable producers to meet demand and minimize processing delays. Additionally, adjusting drum and roller speeds provides flexibility for various processing needs, making the machine adaptable to varying grape sizes and crop loads.

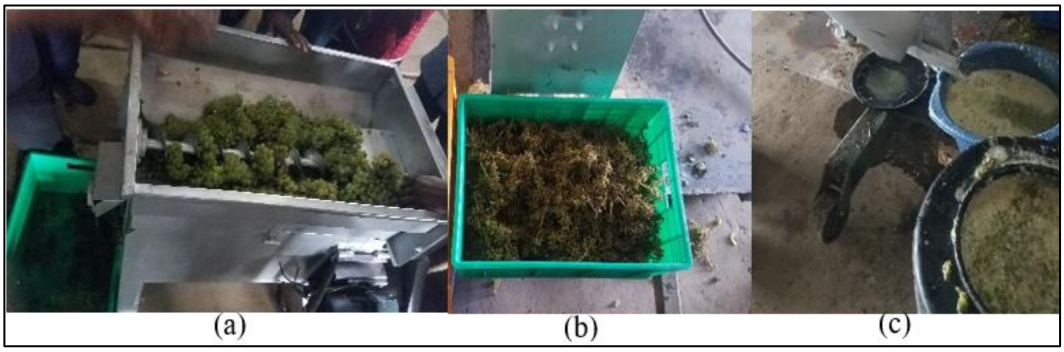

Figure-7. Grape in process (a) input of grapes (b) stem residues (c) extracted juice

3.5. Juice Yield

It was found that the juice collected during testing gave 70.4, 70.0, 71.4 and 69.6 % of the input weight to the juice extracted (Figure 8). This is an average of 70.35 % and a standard deviation of 0.772. These results align with the industry standards for grape processing equipment (Weavers, 2018). This yield is promising, demonstrating the machine's ability to extract a considerable volume of juice without causing excessive grape damage. It indicates that the machine optimizes the extraction process, balancing sufficient crushing pressure and minimal waste. Consistent juice yield across trials supports the machine's reliability and efficiency, making it a valuable asset for producers focusing on volume and quality.

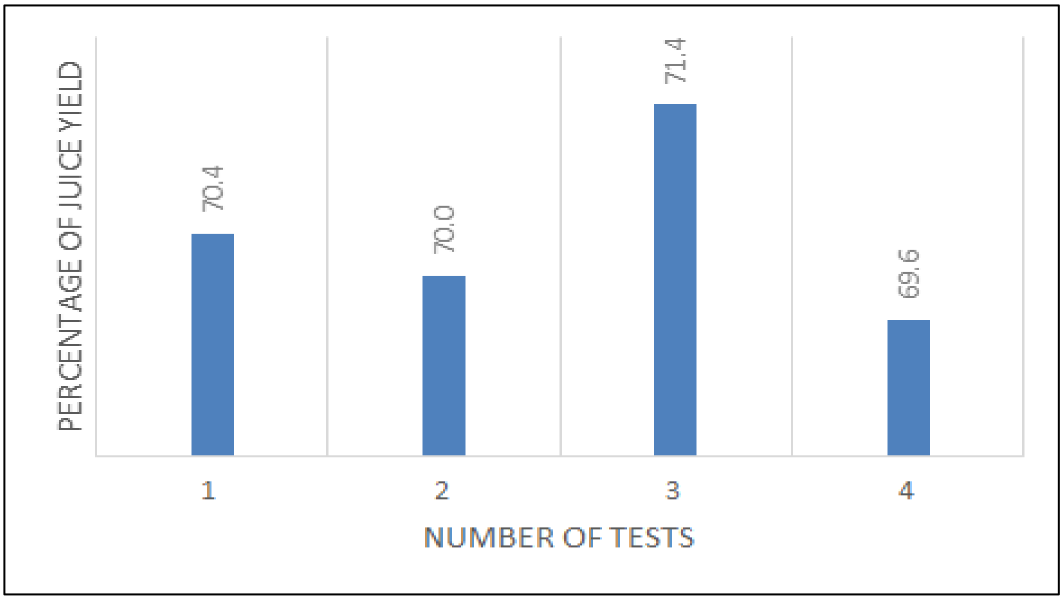

Figure-8. Juice yield

3.6. Qualitative Observations on the Use and Maintenance

Several qualitative aspects of the machine's performance were observed during testing, including ease of use, maintenance, and power requirements. The machine's design allows simple operation, with minimal adjustments between batches. Operators required minimal training to operate the machine effectively, an essential feature for small-scale producers with limited technical skills. The machine's modular design and using locally available materials make it straightforward to maintain. Regular cleaning of the rollers and drum, as well as periodic lubrication of moving parts, were the primary maintenance activities observed. This ease of maintenance supports the machine's longevity and cost-effectiveness. The machine operated efficiently using a small electric motor. Provision for solar power compatibility was evaluated, demonstrating that the machine can be adapted for areas with limited or unreliable electricity.

3.7. Economic Feasibility Analysis

The economic feasibility analysis was conducted to determine if the grape crusher-destemmer machine is a viable investment for Tanzanian grape producers. This analysis considered the machine's production cost, maintenance expenses, and potential cost savings. The machine was constructed with locally sourced materials, reducing production costs by approximately 30% compared to imported options. Given the machine's durable components and simple design, estimated annual maintenance costs were minimal. Based on throughput, juice yield, and labour savings, the machine is expected to offer small to medium-scale producers a return on investment (ROI) within two to three years. This ROI is enhanced by reduced labour costs due to the machine's efficient processing capacity, allowing producers to meet high demand during peak seasons.

Conclusion and Recommendations

This research focused on designing and evaluating a grape crusher-destemmer machine tailored for use by grape producers in Tanzania. The study highlighted the importance of creating locally relevant agricultural machinery that addresses the unique needs and constraints faced by small and medium-scale producers in the region.

The designed machine achieved impressive performance metrics, including crushing efficiency, which averaged 92% above the target threshold of 90%. Destemming Accuracy: It achieved an accuracy rate of 96%, surpassing the target of 95%. Processed an average of 800 kg of grapes per hour, fitting well within the desired range of 500-1,000 kg per hour. Attained a juice yield of approximately 70.35 %, consistent with industry standards.

The economic analysis indicated that the machine is a viable investment, providing an accessible solution for local grape producers. Using locally sourced materials significantly reduced production costs, making the machine more affordable than imported alternatives.

The grape crusher-destemmer machine not only fulfils the processing needs of grape producers but also contributes to the sustainability and growth of the local grape industry. By facilitating efficient grape processing, the machine has the potential to enhance the quality of grape products, expand market access, and improve the livelihoods of local farmers.

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