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Design of High Pressure Rotorvane for Quality in Orthodox Tea Processing

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***Abstract* - Orthodox processed tea refers to loose large whole particle tea leaf produced using traditional (or orthodox) methods of tea production, which involve plucking, withering, maceration/rolling, oxidation/fermentation and drying. Current orthodox tea technologies produce teas of very low quality but have large loose-leaf particle teas preferred by Orthodox tea consumers. Conventional black tea process includes CTC (Cutting, Tearing and Curling) which replaces the rolling stage in the Orthodox tea processing. The CTC tea manufacturing process produces high quality teas but is dusty making the tea undesirable to some consumers. From pilot plant laboratory trials, it was found out that the higher the Rotorvane pressure the higher the Orthodox made tea quality up to an optimum at 25 MN/m2, which gives the highest quality on Taste (T), Colour (C) and Mouthfeel (M) scores matching CTC made tea quality. It is possible to fabricate a High-Pressure Factory Rotorvane by redesigning some components of a standard Rotorvane after design calculations and analysis to determine minimum parameters required. This research paper is on design of High Pressure Rotorvane for quality improvement of orthodox large leaf teas to produce Orthodox Large Particle Tea that is able to infuse and taste like normal conventional black CTC teas. Therefore, a process that produces black tea that looks and feels like orthodox large leaf tea but has the liquor characteristics matching CTC made tea quality is required and can be achieved if the proposed design is implemented.**

***Keywords* – Rotorvane; Orthodox Tea; CTC; Black Tea; Maceration**

I. INTRODUCTION

The conventional CTC black tea process involves withering finely plucked leaf from 78% moisture content to 70 ±2% moisture content [1]. The withered leaf is passed through a standard Rotorvane at a pressure of about 5MN/m^2 [1]. The macerated leaf (ex-Rotorvane) is passed through an 8 - 10 tpi triplex CTC, fermented for 2 hours before drying and sorting into various grades [1].

The black tea process flow chart is as shown below:

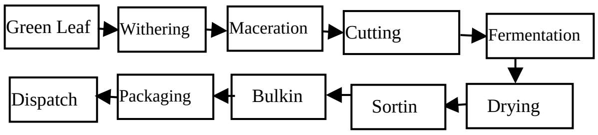

Fig. 1: Black Tea Manufacturing Process [1]

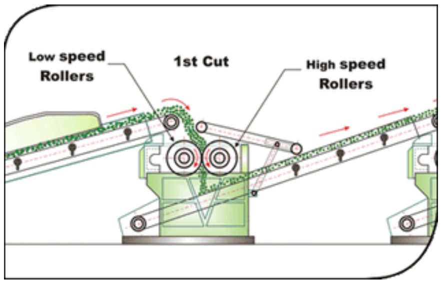

Fig. 2: CTC process [1]

CTC (cutting, tearing and curling), at the cutting stage, in conventional black tea process opens up catechins in plant cells in leaves that later oxidises in the fermentation stage thus producing high quality teas with good attribute of taste (flavour), hue/colour and mouthfeel when infused [2]. CTC teas are dusty making the tea undesirable to Orthodox tea consumers who prefer large whole leaf particle teas [2]. However, current orthodox tea technologies produce teas of low quality because the process releases less catechins in plant cells during the rolling process.

The aim of this research paper is to design a Rotorvane for Orthodox tea processing that will subject the withered leaf to a high pressure above conventional manufacturing pressure of 5 MN/m^2 to rapture leave plant cells so as to release the catechins for the fermentation process. This will lead to elimination of CTC process and production of **orthodox large particle teas** that looks and feels like orthodox large leaf tea and has the liquor characteristics matching CTC tea quality when infused.

Fig. 3: CTC and Orthodox tea images [2]

II. MATERIALS AND METHODS

A. Equipment and Experiments Set up

This research experiments were conducted using 8” pilot plant laboratory scale Rotorvane, design calculations and analysis done to determine minimum parameters for high pressure Rotorvane and teas tasted as per tea tasting categorisation using the Unilever International Tea Categorisation System (ITCS). The research was conducted at Unilever Tea Kenya Limited Kericho in conjunction with Multimedia University of Kenya, Department of Mechanical and Mechatronics Engineering. All the materials were sourced from the local markets.

Rotorvane barrel wall (on the inside) was fitted with a pressure sensor for both the prototype laboratory scale Rotorvane (8" diameter) and the standard Rotorvane (15" diameter). As the Rotorvane vanes press leaf against the resistors (bolted onto the Rotorvane barrel), the sensors picked up the pressure (stress) signal which is converted to a digital electronic form and displayed in the data logger screen every minute. The electronic data logger displayed readings in Pascal (Pa, N/m^2).

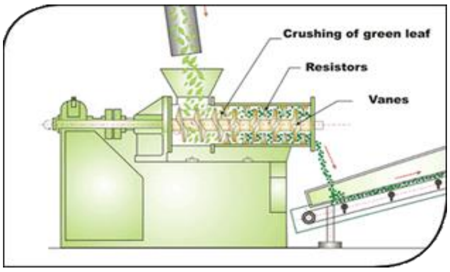

Fig. 4: Rotorvane [1]

Experiments were carried out at the pilot plant using 8” Rotorvane and Standard 15” Rotorvane as control to visualize the design of High Pressure Rotorvane. This involved green leaf of 65% acceptable quality, withered to 70±2% Moisture Content being tracked through the laboratory scale manufacturing process at pilot plant and CTC control in the factory with 5 repeats for each trial so as to overcome the experimental, equipment, process and human errors. The different pilot plant Rotorvane pressure levels was obtained by setting the Rotorvane iris plates opening to different levels (2, 4, 6, 8 and 10 cm). A standard tea processing Rotorvane (15" diameter) has an iris plates opening setting of 10 cm. Samples for Large Leaf (LL) and Medium Leaf (ML) grades (Broken Pekoe1 (BP1) and Pekoe Fannings1 (PF1) for control) were picked and sent to a panel of tea tasters for made tea quality assessment using the Unilever International Tea Categorisation System (ITCS).

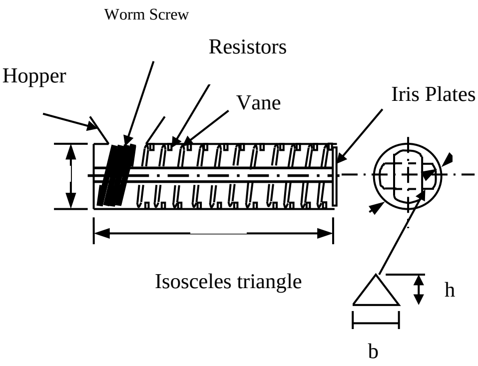

Fig. 5: Rotorvane Sketch [1]

Design calculations and analysis were carried out for High Pressure Rotorvane components (motor, gearbox, wall thickness, iris plates, motor and gearbox pulleys diameters and width, V-belts section, quantity and belt tension, shaft diameter, flange thickness, flange bolt diameters, keys and keyways) to determine minimum design parameters that create high pressure in the Rotorvane barrel.

B. High Pressure Rotorvane Design

i) *High Pressure Rotorvane Power Requirements:* The equations and formulas used in the design were sourced from mechanical engineering design reference books. [3] [4] [5] [6] [7] [8] [9] [10].

Rotorvane maceration behaves like a normal extrusion process. In an extrusion process, the axial force exerted, F; the work done in the process, W; and the power required and determined using the following expressions:

$$F = A_o\, k \ln [A_o/A_e] \quad \ldots\ldots\ldots\ldots\ldots\ldots (1)$$

Where,

A_o = Original cross sectional area

A_e = Extrusion cross sectional area

K = Yield stress constant for the material being extruded (25MN/m^2 for tea in a high pressure Rotorvane).

$$W_d = F \times d \quad \ldots\ldots\ldots\ldots\ldots\ldots (2)$$

Where,

d = distance through which the material is displaced (length of the rotorvane)

$$P = \frac{W_d}{t} \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

Where,

t = residence time

For this design, original cross-sectional area was $0.1141m^2$, extrusion cross sectional area $8.1525m^2$, while the length of the rotorvane was 0.127m. The tea took 60 seconds to move through the rotorvane. Assuming gearbox efficiency to be 80% and using a safety factor of 1.5, the power required was found to be 30 kWatt (40 hp), hence a 30kW motor with 1475 rpm was selected and used, and a step-down gearbox with 20:1 ratio was used.

The high pressure Rotorvane behaves like a pressurised thick-walled cylindrical body at 25 MN/m^2.

For a pressurised cylindrical steel barrel:

$$\sigma_c = \sigma_y = \frac{Pr}{t} \quad \text{.......................(4)}$$

Where,

σ_c = Circumferential stress

σ_y = Yield stress = 400 MN/m^2 for steel

P = Internal pressure

r = Internal radius of the cylinder

t = Rotorvane barrel wall thickness

For the designed Rotorvane, the minimum wall thickness required is 12 mm compared to the current Rotorvane barrel thickness is 13 mm.

ii) *High Pressure Rotorvane Iris Plates Thickness*: The Rotorvane iris plates are made of Brass. The Rotorvane Iris plates behave like a cantilever with a uniformly distributed force along its length, L.

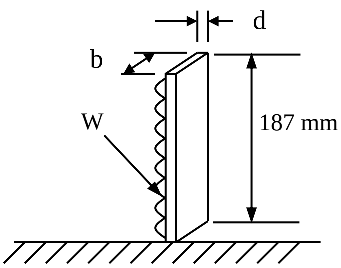

Fig. 6: Iris Plates Dimensions and Pressure Distribution

The internal pressure of 25 MN/m^2 generates the uniformly distributed load, W.

$$W = \text{Pressure, } P \times \text{Area, } A \quad \text{..... (5)}$$

Two iris plates overlap such that the area under pressure is 8.1525×10^{-3} m^2.

$$\frac{M}{I} = \frac{\sigma_{all}}{y} \quad \text{........................ (6)}$$

Where,

$M = \frac{WL^2}{2}$ - Bending moment,

$I = \frac{bd^3}{12}$ - Moment of inertia (b - cross sectional length of the plate and d = thickness of the plate)

σ= bending stress (Allowable bending stress for Brass, σ_{all} = 150 MN/m^2), and

y = Deflection

Maximum deflection for a cantilever with a uniformly distributed force, y_{max}:

$$y_{max} = \frac{WL^4}{8EI} \quad \text{................ (7)}$$

Where,

W = Uniformly distributed force per metre length,

L = Length of cantilever beam,

E = Modulus of elasticity, for Brass, E = 0.98×10^6 kg/cm^2, and

Computing we obtained the minimum plate thickness of 18.5 mm.

iii) *Rotorvane Motor Pulleys and Pulley Belts:*

The pulley material to be used is cast iron and it is to serve for up to ten hours per day. For the selected motor the speed, N_m = 1475 rpm. In practice the best operating rotorvane speed, N_{rv} = 40 rpm, Gearbox ratio, G_r = 20:1. The diameter of the motor pulley is determined

Input speed to gearbox,

$$N_m = N_{rv} \times G_r \quad \text{.......................... (8)}$$

Motor pulley diameter,

$$D_{mp} = \frac{G_r}{N_m D_{gb}} \quad \text{.................. (9)}$$

Where;

D_{gb} - Diameter of gearbox pulley

Pulley Arrangement:

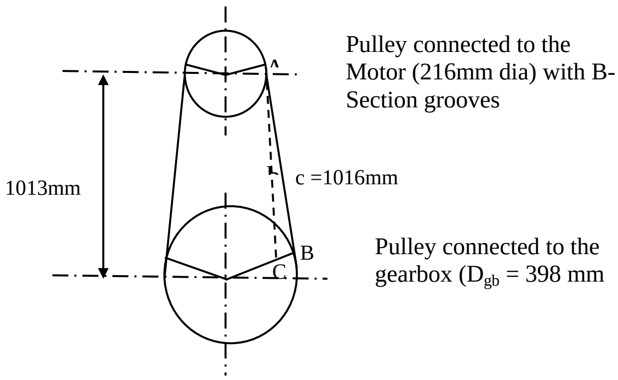

Fig. 7: Pulley arrangement

From cosine rule:

$$c^2 = a^2 + b^2 - 2ab \cos C \text{ .... (10)}$$

Computing the length of V-belt was determined to be 2999 mm (approx. 118 inches), therefore, B118 V-belts was used. For belt length of 3000 mm, the additional frictional/weight power per belt, P_a is 0.04kW.

Number of Pulley Belts, N required is determined from the following expression:

$$N = \frac{P_t F_a}{(P_r + P_a) \times F_c \times F_d} \text{....... (11)}$$

Where,

P_r - Power rating per belt

P_t - Power to be transmitted

P_a - Additional frictional/weight power per belt

F_a =1 - Service factor

F_c = 0.82, F_c - Length correction factor

F_d = Arc of contact correction factor

Given that the yield tensile strength for high tensile rubber belts, σ_y = 7 MN/m^2. Considering a safety factor of 1.5 and cross-sectional area for B-section rubber belt, A_r:

$$F = \sigma_y \times A_r \text{ .... (12)}$$

Power rating per B118 rubber pulley belt, P_r is power that can be transmitted per belt and is calculated as follows:

$$P_r = T \times \omega = F \times r \times \omega = F \times r \times \frac{2\pi N}{60} \text{...(13)}$$

Computing we obtained the force, F = 672 N and power rating per B118 rubber pulley belt, P_r = 11,210 Watts.

Hence, the number B118 belts required, N = 3.25 ≈ 4 belts.

iv) Pulley Width

It is proposed that the belt width, $b_{wb} \leq 125$ mm. For this category of belts, the pulley width, b_{wp} is determined from the following expression.

$$b_{wp} = \sum (b_{wb} + s) \text{ .... (14)}$$

Where,

s = 5 mm - Space between grooves,

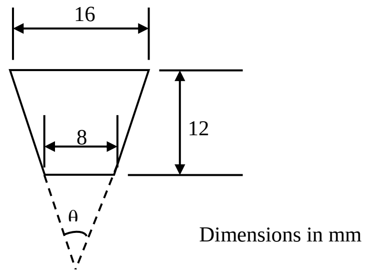

Fig. 8: B-Section V-Belt

Therefore, Pulley width, b_{wp} = 89 mm

v) Maximum Belt Tensions

For the B-Section pulley Rubber Belt Cross Section:

$$\frac{T_1 - MV^2}{T_2 - MV^2} = e^{\mu\alpha/\sin\theta/2} \text{ ... (15)}$$

Where:

T_1 = Tension in side 1

T_2 = Tension in side 2

M = Mass of belt per metre length

μ = Coefficient of friction between pulley and belt

α = Wrap angle

θ = Groove angle

V = Belt circumferential velocity, m/s.

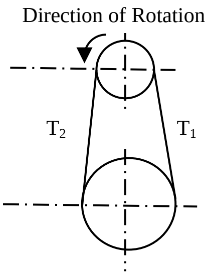

Fig. 9: Pulley arrangement with rotation

The mass per metre length (M) for the V-Belt rubber is calculated using the equation:

$M = A \times \rho_r \times l_r$ ……………….….. (16)

Where:

A = V-belt cross sectional area

ρ_r = Rubber density = 1140 kg/m^3, and

l_r = length of rubber, (l_r = 1 m)

Power transmitted, P:

$P = (T_1 - T_2) V$ ……………………. (17)

Where:

$\mu = 0.3$ (for dry rubber belts)

$\alpha = 178.6^o$ (3.117 rad)

$\theta = 40^o$

$V = \pi DN/60$

Computing we obtained the tensions: T_1 = 170.57 N and T_2 = 1969.13 N, respectively

Maximum belt tension per belt, T_{max} = 492 N < Rated allowable tension of 672 N per belt. Therefore, the belts will not easily fail or slip during normal operation.

vi) Keys and Keyways

Motor and gearbox were supplied with keyway. The following are their dimensions: Key width, B_k =16 mm and key depth, t/2 = 5.5 mm

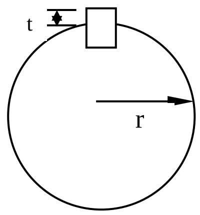

Fig. 10: Keys and Keyways arrangement

To determine the minimum length of the key for no failure:

Rotorvane Pulley Key

Rotorvane pulley key is designed against shear. From the torque equation;

$T = B_k L_s \tau r$ …………………….…. (18)

Where,

T = Torque

B = Key width

τ = Allowable shear stress (= 40 MN/m^2 for steel shafts, Safety factor of 6 used)

r = Shaft radius (27.5 mm for Rotorvane Motor and 31.5 mm for Rotorvane gearbox).

L_s = Key length (Obtained for Rotorvane motor, L_s = 11 mm, against shear)

The power transmitted, $P_r = T \omega = F r \omega$ = 30,000 Watts (High Pressure Rotorvane Power Requirements *(from B. i))*. (F is the Force at the key)

For compression torque, T is:

$T = \sigma_c \times t \times L_c \times \frac{r}{2}$…. ……………(19)

Where:

σ_c = Allowable compressive stress (= 70 MN/m^2 Safety factor of 6 used)

L_c = Key length (Computed for Rotorvane motor, L_c = 18.3 mm, against compression)

Therefore, minimum keyway length for no failure, L_c = 19 mm

Take the larger of L_c and L_s, Therefore Key length will be a minimum of 20 mm

Gearbox Pulley Key:

Similarly, gearbox pulley key is designed against shear. From the torque equation;

For Rotorvane gearbox; N = 800 rpm (83.776 rad/s) and F = 11,368.2 N.

Computing, T = 358.1 NM, and length of the key for gearbox Pulley, L_s = 17.8 mm. A minimum length of the key, L_s of 18 mm was adopted.

For compression torque, T is expressed using torque equation as follows ;

$T = \sigma_{all} \times t \times L_c \times \frac{r}{2}$…………(20)

Where:

σ_{all} = 70 MN/m^2 – Allowable stress

Computing we obtained for gearbox Pulley, L_c = 29.5 mm

Minimum Keyway length for no failure, L_c = 29.5 mm

Take the larger of L_c and L_s, therefore key length will be a minimum of 30 mm

vii) Rotorvane Shaft design on strength basis

Solid shafts are designed on the basis of maximum principal stress theory or maximum shear stress theory. Applying these theories to transmission shafts subjected to combined bending and torsional moments results in the following equations and calculations:

Bending Moment

$M_b = \frac{\pi}{32} * \sigma_b (d_o)^3 (1-k^4)$ …………..…. (21)

Where:

M_b = Bending moment at the point of interest

d_o = Outer diameter of the shaft

k = Ratio of inner to outer diameters of the shaft (for solid shaft, k = 0 because inner diameter is zero and d_o = d).

$\sigma_{b,\ all}$ = allowable bending stress

= 30% of the yield strength (400MN/m^2) but not over 18% of the ultimate strength (420 MN/m^2) in tension for shafts without keyways. These values are to be reduced by 25% for the presence of keyways [7]
= 0.3 x 0.75 x 400= 90MN/m^2

For the existing Rotorvane shaft, d = 63.5mm,

Computing we obtained;
M_b = 2263.280 N-m

Twisting Moment

$M_t = \frac{\pi}{32} * \tau (d_o)^3 (1-k^4)$ ………..…. (22)

Where;
M_t = torsional moment (N-mm)
d_o = shaft outer diameter (mm).

τ = maximum shear stress in shaft = 40 MN/m^2 for steel shafts with keyway. [9]

For solid shaft, k = 0, d_o = d
Computing we obtained;
M_t = 1005.902 N-m

Axial Stress

$\sigma_a = \frac{4\alpha F}{\pi d_o^2}(1 - k^2)$………………(23)

Where,

F = Axial force (tensile or compressive)

α = Column-action factor (= 1.0 for tensile load) [7]
For solid shaft, k = 0, do = d
F = 492 x 4 = 1968 N (Ref: Maximum Belt Tensions design).
Computing d, we get:
d = 5.275mm

Axial stress is usually ignored in solid shaft design due to its insignificant design quantities. [7]

Torsional shear stress

$\tau_{max} = \frac{16 M_t}{\pi d^3}$ ………………………… (24)

Where;

τ_{max} = maximum shear stress in bolt = 40 MN/m^2 for steel shafts with keyway. [9]
M_t = torsional moment (N-mm) (= 1005.902 NM – See twisting moment calculation)
d = shaft diameter (mm).

Computing for d, we get;
d = 50.4mm

Torsional Rigidity:

The angle of twist θ (in degrees) for solid circular shaft on the basis of torsional rigidity is given by,

$\theta = \frac{584 M_t L}{G d^4}$……………………….. (25)

Where;

θ = angle of twist (deg.)
L = length of shaft subjected to twisting moment (mm) = 1,510mm (selected to suit standard Rotorvane).
M_t = torsional moment (N-mm)
G = modulus of rigidity (N/mm^2) (Modulus of rigidity for steel is 79,300 N/mm^2) [6]
d = shaft diameter (mm).
Permissible angle of twist for machine tool applications is 0.25° per metre length [7]

Computing for d, we get;

d = 13.12mm

$\tau_{max} = \frac{16}{\pi d^3}[\sqrt{((k_b M_b)^2 + (k_t M_t)^2)}]$ ………… (26)

$\sigma_b = \frac{32 M_b}{\pi d^3}$ ………………………………… (27)

$\sigma_{b,t\ all} = \frac{16}{\pi d^3}[k_b M_b + \sqrt{[((k_b M_b)^2 + (k_t M_t)^2)}]$ ……(28)

Where;
τ_{max} = maximum shear stress = 40MN/m^2 for steel shafts with keyway…... [9]
$\sigma_{b,\ t\ all}$ = allowable resultant bending and torsional stress
= 30% of the yield strength (400MN/m^2) but not over 18% of the ultimate strength (420 MN/m^2) in tension for shafts without keyways. These values are to be reduced by 25% for the presence of keyways. [7]
= 0.3 x 0.75 x 400= 90MN/m^2
k_b = combined shock and fatigue factor applied to bending moment.
k_t = combined shock and fatigue factor applied to torsional moment
For Steel, k_b 1.5, k_t = 1.0 (for load gradually applied) [7]
Mb - bending moment (N-m)
M_t - torsional moment (N-m)

Computing for d using equation (26), we get;

d = 38.6 mm

Computing for d using equation (27), we get;

d = 63.3 mm

Computing for d using equation (28), and with a safety factor of 1.5, we get;

d = 63.2 mm

From maximum principal stress and maximum shear stress calculations considering transmission shafts subjected to combined bending and torsional moments results, shaft diameter of 63.5mm chosen (same as for existing standard Rotorvane).

viii)3.3.3.10 Shaft Flange Coupling

The torque capacity of the coupling is given as;

$$T = \left(\frac{\pi}{4} d_b^{\,2}\right) . \tau_b{}' . r . n \quad \text{..............................(29)}$$

Where;

d_b = diameter of bolt

τ_b'= maximum shear stress in bolt = 40 MN/m^2 for steel shafts with keyway………. [9]

n = no. of bolts = 4 (Available flange).

r = distance from centre of bolt to centre of coupling = 106.75mm (Available flanges).

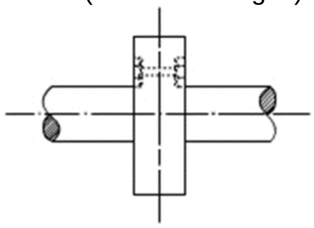

Fig. 11: Flanged Joint

T = 194.22 N-m (Calculated from on Rotorvane Pulley design with a safety factor of 4)

Computing for d_b, we get;

d_b = 15mm (Available 20mm diameter high tensile bolts on flanges were chosen).

III. RESULTS AND DISCUSSIONS

A. Tea Tasting Results and Discussion (Pilot Plant Trials)

The tea tasters tasted made tea samples as per International Tea Categorisation System (ITCS) from each set of pilot plant trials alongside that of standard low pressure Rotorvane. Tasting results for Large Leaf (LL) and Medium Leaf (ML) grades on Taste, Colour, Mouthfeel and Leaf are as summarised graphically below:

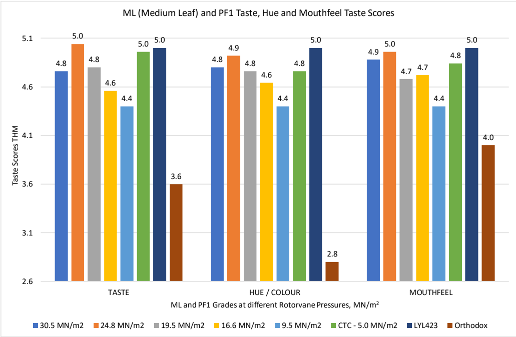

Fig. 12: ML (Medium Leaf) and PF1 Taste, Hue and Mouthfeel Taste Scores

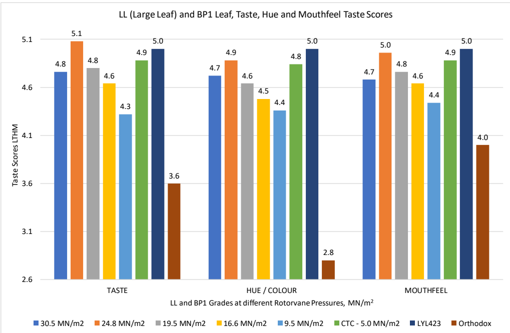

Fig. 13: Large Leaf (LL) and Broken Pekoe1 (BP1) Taste Scores

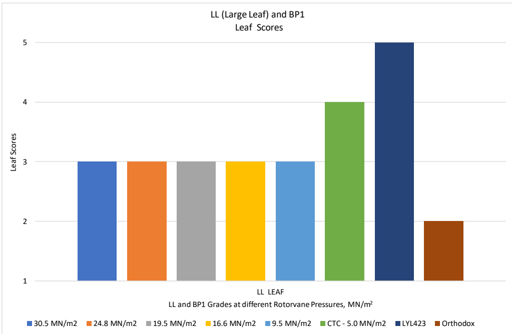

Fig. 14: Large Leaf (LL) and (Broken Pekoe1) BP1 Leaf Scores

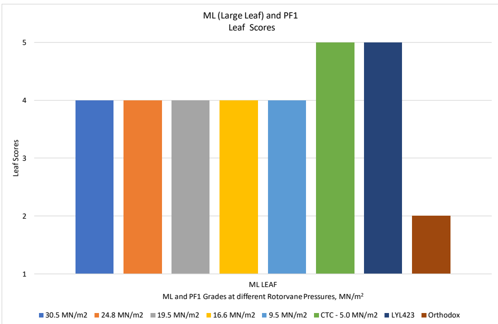

Fig. 15: Medium Leaf (ML) and Pekoe Fannings1 (PF1) Leaf Scores

From the Pilot Plant Trials, Pressure was increased at an average of 5.3 MN/m^2 for each set of trials. From the Pilot Plant Trials, the optimum pressure for High Pressure Rotorvane is 24.8MN/m^2 (approximately 25 MN/m^2) which matches the CTC made tea quality on taste, colour (hue) and mouthfeel with improved made tea leaf appearance compared to Orthodox teas.

B. Summary of High Pressure Rotorvane design Calculations

Rotorvane design calculations and analysis to show capability of the standard Rotorvane (RV) at high pressure were done. A summary of the main calculations and recommendations for modifications on the standard Rotorvane to upgrade it to high pressure are as shown in the table below:

Table 1: High Pressure Rotorvane Design Test Calculations Summary

Part	Material	Standard Rotorvane (Rv)	High Pressure Rotorvane Requirements
Power		15 hp (11 kW)	40 hp (30 kW)
Gearbox	Cast Iron	15 hp (11 kW)	40 hp (30 kW)
Barrel (wall) thickness	Cast Iron	13 mm	12 mm
Iris plate thickness	Brass	15 mm	19 mm
Motor Pulley diameter	Cast Iron	152 mm	216 mm
Gearbox pulley diameter	Cast Iron	398 mm	398 mm
Number of 'V' belts	High tensile Rubber	4	4
Pulley width		89 mm	89 mm
Length of the pulley 'V' belts	High tensile Rubber	B102	B118
Maximum belt tensions		672 N	492 N
Motor Keyway		89 mm	19 mm
Motor pulley Keyway		89 mm	19 mm
Motor pulley key	Mild Steel	90 mm	19 mm
Gearbox pulley keyway		89 mm	30 mm
Gearbox pulley key	Mild Steel	90 mm	30 mm
Rotorvane Shaft diameter	EN24 Steel	63.5mm	63.3mm
Flange bolt diameter	High Tensile Steel	20mm	15mm

In summary, based on design calculations and analysis the standard Rotorvane can be upgraded (design, fabrication and assembly) to high pressure Rotorvane. To do this, a number of modifications need to be made. These include:

i) Purchase and installation of 40 hp (30 kW) motor and gearbox

ii) Fabrication of 19 mm thick new Brass iris plates

iii) Fabrication and installation of 216-mm diameter 4 groove 'B' section Rotorvane motor pulley.

iv) Purchase and installation of 4 pieces of B118 rubber 'V' belts per Rotorvane.

v) All other Rotorvane system parameters were found to be sufficient and were utilised in the design (Standard gearbox pulley, number of V-belts, pulley width, motor key and keyway, gearbox pulley, Rotorvane shaft, flanges, flange bolts, key and keyway).

IV. CONCLUSIONS AND RECOMMENDATIONS

A. Conclusions

i) For Orthodox teas (Large Leaf (LL) and Medium Leaf (ML) grades), the higher the Rotorvane pressure the higher the made tea quality from the innovation process up to an optimum at 25 MN/ m^2, which gives the highest quality on Taste (T), Colour (C) and Mouthfeel (M) scores matching CTC made tea quality with no impact on the rest of the manufacturing processes (withering, drying, sorting, etc.).

ii) It is possible to fabricate a High-Pressure Factory Rotorvane based on design calculations and analysis giving required minimum parameters, by modifying or upgrading some components of a standard Rotorvane.

B. Recommendation for Further Work

i) Since there is a direct correlation between pressure and made tea quality, there is need to develop and/or do further modifications at the pilot plant Rotorvane so that it can handle pressures higher than 25 MN/m^2 with a smooth consistent flow. Since the Rotorvane may be limited on this, it may involve investigation on the usage of other tea maceration machines other than the Rotorvane.

ii) There is need to execute the proposed fabrication, use of the High Pressure Rotorvane to test and commission the design for production of quality Orthodox tea and check potential realisation of commercial and health benefits opportunities.

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