Optimal design of the tractor-front-mounted sugarcane grab loader

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Abstract: The percent of sugarcane mechanical sugarcane loading (in Egypt) has not been exceeded 10% because of the poor finance-ability of the farmers to by the imported self-propelled loaders. The local workshops have been attempting to fabricate a sugarcane loader attached to the farm tractors as a cheap alternative. Farmers did not accept the tractor-mounted loaders fabricated locally because of the problems concerning poor efficiency and poor balance of the tractor-mounted loader while operation. The main objective was to overcome the problems concerning precise fabrication of the tractor-front-mounted loader and to develop an efficient dynamically balanced machine. Fabricating a sugarcane grab loader for the tractor faces difficulties due to the variation of the farm tractor types and sizes. Reference to the size of the transport vehicles was supposed to be loaded, the medium size tractors 70-90 hp was considered to be equipped with the sugarcane loader. The most important specifications of the prevailing medium size farm tractors found in the sugarcane production area were recorded through field survey. Important tractor parameters related to the loader design were specified. For the prevailing medium size tractors of power range 70-90 hp, wheel base within 2.4 m. Total tractor weight about three tons, distributes as two thirds on the rear axle and one third on the front axle. The tractor operator is supposed to have clear view over the boom lowered for pilling, so that the height of steering wheel is important parameter determined as 1.9 m for the medium size tractors. Important dimensions such as the tractor chassis dimensions and other dimensions related to the sugarcane loader were identified. A chain of measurements and computations were conducted to facilitate optimizing the design of a tractor mounted sugarcane grab loader dynamically balanced with no need for counterbalance weights. Considering the data of the medium size tractor, the dimension of the loader components computed in accordance. A general formula relates the loader size to the tractor on which the loader supposed to be mounted was developed to facilitate fabricating the balanced loader for any size of the farm tractors. The loader was fabricated and tested in the field. The loader proved high dynamical stability and economical operation efficiency. The field test results show operating the loader for loading sugarcane transport vehicles, loading cycle time was about 1.7 min, loading rate up to 14 ton/h and operation efficiency was over 90%.

Keywords: sugarcane grab loader, tractor-front-mounted-loader, loader design, loader balance

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1 Introduction

About 20 million tons is the annual production of vegetative sugarcane in Upper Egypt. Few mechanical sugarcane loaders were available till the activity of Aswan Mechanization Company was established 1984. At that time large number of bell type sugarcane loaders was imported and custom operated that increased the percent of mechanical loading directly to be 10% of the total cane production. Because of the rapid increase of the bell loader price and several other reasons, the company could not buy new loaders to replace old ones. Therefore, the percent of mechanical loading of sugarcane declined and stay below 10% for long years. As a cheap alternative, tractor-mounted grab loaders were fabricated locally as attempts to replace bell loaders. The local fabricated loaders have been suffering from the problems of poor balance, poor stability and low loading rate. Farmers rejected these attempts because operating such machines was boring and represented waste of time. The current research was devoted for sizing the

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tractor-front-mounted sugarcane grab loader in relation to the tractor on which the loader supposed to be mounted. The developed loader should consider farmers' desire represented in; a) the loader should be front-mounted on the prevailing sizes of farm tractors, b) no counterbalance weights needed and c) the loader should be capable to load all types of sugarcane transport vehicles. Necessary measurements, analysis and computations were followed for sizing the loader and achieve the farmers' desires. Abdel-Mawla (2012) reported that, mechanizing sugarcane loading as a major farming/ industrial activity will release labors from one of the most adverse operations, develop the technical level of the people, save hard currency, save more job opportunities and secure higher labor income. Efficiency of sugarcane mechanical loading in Egypt studied by Abdel-Mawla (2010) stated that mechanical loading of sugarcane in the Nile Valley in Upper Egypt is required to substitute for the decreasing availability of laborers for manual loading. Farmers have been striving to increase the number of loaders available. Practical procedures can be developed which facilitate the operation of cane loaders with increased efficiency and productivity. Operational strategies which facilitate profitable operation of cane loaders will lead to a significant increase in the proportion of the crop which is mechanically loaded. Worley and Saponara (2011) reported that the front-end loader is an indispensable machine for the off-road construction equipment industry. It is a classic example of a working machine with complex interactions between its subsystems (hydraulic, mechanical, and electrical). Dynamic models of the full-scale vehicle coupled with event-based operator models are currently used to help quantify the overall system performance, efficiency, and operability. However, these models are complex and not always necessary to characterize the response of individual subsystems. There is great value added to the design process–especially in prototyping of new vehicle platforms–in development of simpler models that can quickly and accurately define first-order measures of system loads and performance. Gawlik and MichaПowski (2009) explained that excavators and loaders present the biggest chance in the increase of efficiency through the recuperation of hydraulic energy. It is a result of repeatability of work cycles and potential energy of lowering excavator's linkage possibility to use. Counterweights are assembled quite often in the construction equipment structures but these elements are normally fixed. The counterweight is connected with boom mechanism through a link mechanism. This solution unloads boom drive in approximate mode and increases the stability of the crane. This is a conception of how excavators could evolve over the next two decades. The active counterweight unloading boom mechanism is only one of many conceptions. Rehnberg (2008) stated that the wheel loader is a type of engineering vehicle used primarily to move crude material over shorter distances. As the vehicle is designed without wheel suspension, wheel loader drivers are exposed to high levels of whole body vibration which influences ride comfort negatively. An analytical model is used to study the effect of front and rear suspension characteristics on the pitching response of the wheel loader, showing that a stiffer rear suspension is favorable for reducing pitching but also that a similar effect is attainable with a stiffer front suspension. Results are compared to multi-body simulations which show the same trend as analytical predictions. Debeleac (2009) repoted that wheel deflections and other states of the system would be kept at a constant value, most of the time, which means that the wheels would absorb or diminish the dynamic loads from any of the road disturbances. Hereby, through an optimal choosing of the loader wheels, these should absorb the most of the dynamic effects due to the road disturbances. Thus, one of the conclusions is that, if the loader will be equipped with an intelligent control system, which have to drive both the electro-mechanical, and the hydraulic systems, it will be assured all the premises for diminishing of transitory states. The bucket is the most sensitive component of the loader to the dynamic disturbances, thus that the angular displacement of this component could be substantial reduced by means of this automation of driving system. Kiliç (2009) developed a dynamic model to perform the hydraulic and mechanical simulation of the loader system of a backhoe-loader. Instead of deriving and programming the hydraulic and mechanical system equations, physical simulation toolboxes inside MATLAB environment are used to model the hydraulic and mechanical systems of the machine. In conclusion, this dynamic machine model, which includes the hydraulic and mechanical systems, can be used in determining the dynamic loads on the joints and attachments of the backhoe-loader. Then, these dynamic loads may be used as an alternative loading condition for the stress analyses of the attachments. In addition to that, this model may be integrated into the design process in order to reduce prototyping time and costs during the design process. Popescu and Sutru (2009) explained that loaders mounted at the front-end of agricultural wheel tractors represent cost effective loading equipment. Computer simulation allows the study of the longitudinal stability of the systems through application for the constructive tractor-loader models.

1) The front loaders mounted at the front-end of agricultural wheel tractors are increasingly employed for the mechanization of material loading and unloading operations into/from transport means or other locations on low and medium agricultural farms.

2) The manipulation of both pallets and box-pallets in agriculture farms can be performed on the relatively short distances (within a hangar or between close locations) and on the vertical up to certain heights, using the forklift equipment mounted on the tractor rear hitch.

3) The constructive and functional parameters of front and rear loaders mounted on agricultural wheel tractors have to satisfy the requirements of the working process and of the dynamic stability and have to correspond to the structures of the tractors they are mounted on.

4) The braking of the tractor equipped with a front and rear loaders during descending a longitudinal slope with the filled bucket or forklift in transport position are in relation to the longitudinal stability of the system the most difficult situation of the traveling process.

5) The dynamics of tractor–front end loader and fork lift loader systems can be analyzed by mathematical modeling of the equivalent dynamic models of the real systems, taking into account the exterior forces to which they are subjected in various working situations.

6) Based on the equivalent dynamical models of tractor–front end loader and fork lift loader systems it can be elaborated the mathematical model describing the dynamical behavior of the forklift truck during the descending on a slope by slowing down (breaking) of the vehicle and acceleration of the fork while lifting the load.

Boast and Meyer (1985) stated that using the trailer-mounted crane system instead of the self-propelled loader results in savings, but its full potential has not yet been realized. Indications are that when the crane's performance equals the loading rate of a small grab loader, with a lighter trailer, the tractor will be able to operate faster and costs will be reduced even further. The present 'crane rig' system seems to be worthwhile only for growers producing less than 7,000 tons of cane annually, but with further developments, it may suit growers producing up to 12,000 tons of cane. Those growers who previously could not justify using a mechanical loader, may now find it feasible to do so. Bartlett (1963) reported that introduction of the self-loading cane trailer into the sugarcane harvesting systems employed in Natal, has achieved several advantages over the former method of hand loading the cane. It is apparent, however, that the task of lifting, carrying and loading or stacking is far more tiring and requires more physical exertion than cutting the cane, especially if the cane has to be carried over long distances and up ladders. From an engineering standpoint, it is far easier and less expensive to mechanize the loading operation than the cutting operation, especially when one has to handle badly lodged, twisted or trashed cane.

As previously mentioned, the problems of expensive prices of the self-propelled loaders delayed the mechanization of sugarcane loading. The major objective of the current study is to develop a tractor-mounted-loader as a cheap alternative machine for sugarcane mechanical loading. The problems of balance forced the fabricators to use extra counterbalance weights to secure the dynamic balance of the tractor-mounted-loaders during loading operation. The use of counterbalance weights makes the machine more heavy, more costly and of less efficiency. The plan of the current study is to conduct the necessary measurements and computations to compromise the loader-steel-weight, loader-grab-load and the loader boom length that secure the tractor balance with no need for extra counterbalance weights.

2 Materials and methods

2.1 Availability and features of the tractors:

Medium size tractors of power range 70-90 hp represent the farm power available in the sugarcane area. Most of those tractors are equipped with hydraulic system of adequate power to operate the proposed sugarcane grab loader. The most important tractor features that determine the design of the tractor mounted sugarcane loader are total weight of the tractor, weight distribution over the front and rear wheels, tractor wheel base and the height of steering wheel. Other tractor details related to the loader base bolting on the tractor chassis as well as chassis dimensions were also important.

2.2 Main components of the loader:

The components of the tractor mounted loader were decided to be: loader bases and supports, loader boom, loader-grab and the loader hydraulic system.

2.2.1 Loader bases and supports

The loader bases are two similar steel constructions to be fixed on both sides of the tractor chassis, which both sides of the loader arm and corresponding lift cylinders supposed to be mounted on. Bases are fabricated as strong parts to support the tractor chassis while the loader operation.

2.2.2 Loader boom

The boom arm that consists of two similar sides each is hinged by single pin to each one of the above described bases. The front half of the boom welded bent to the rear part downward with 45^0 with respect to horizontal. Both sides of the boom were connected to each other with two strong transverse steel supports so that the two sides of the boom became one part. The base of the clamping hydraulic cylinder was welded to the transverse supports at the mid distance between the two sides of the loader arm. Two pin holes were provided to both sides of the loader boom for hinge connection to the top point of the vertical supports of the bases fixed on both sides of the tractor. Other pin holes were provided to the front end of the two sides of the loader arm for hinge connection of the grab. Clamping mechanism base has two holes on which the clamping cylinder pin connected to the rear one and the middle point of the clamping arm connected to the front one. Clamping mechanism consists of the clamping arm and the clamping link. The clamping arm is of two sides connected to each other with a welded thick wall short tube in the middle of the arm in which a pin hinge it to the front end of the clamping mechanism base. The arm has other two holes at its two ends on which the clamping cylinder and clamping arm are connected. The clamping link also has two hinge holes at its ends on at which it connected to the clamping arm and the clamping point at the rear end of the grab.

2.2.3 Loader-grab

The grab designed to be a push pillar sugarcane grab. The lower fork of the grab should provide a straight outer surface to slide over the ground while the push pilling action. The upper fork of the grab should provide an internal curved surface for smooth follow of the cane while pressing and holding. Grabbing force should be provided by pressing both grab forks to each other. Therefore the hydraulic cylinders should be hinged to the lower fork and the piston connected to the upper fork arm. The grab should be hinged to the loader boom and clamped with hydraulic cylinder connected to the clamping mechanism. The grab structure should be strong enough to face stresses due to grabbing and clamping.

2.2.4 Loader assembly

The loader was designed to be a tractor front end mounted sugarcane grab loader. The rear end of the loader arm hinged to the respective hitch point at the top of the vertical support. The hydraulic cylinders of the loader arm connected to their respective hitch points each with two pins at the vertical support and the loader arm. The clamping mechanism assembled to the clamping base and the grab assembled to the front end of the arm. Figure 1. Assembly of the proposed tractor mounted loader components.

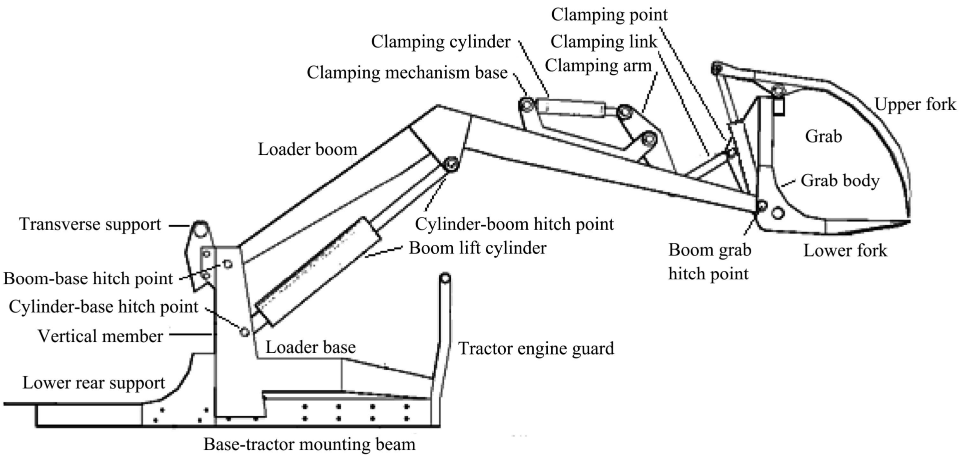

Figure 1 Assembly of the proposed tractor mounted loader

The principle dimensions of the loader components and parts were determined through the following steps:

1) Estimating the malleable cane stalk properties related to bundling and grabbing.

2) Estimating the principle grab dimension based on maximum cane bundle.

3) Estimating the dimensions related to boom-grab assembly and base-boom assembly at pilling situation.

4) Estimating principle dimensions related to loading situation.

5) Sizing loader boom length at non counterbalance weights added.

Verification of loader stability.

Develop general formula relate the loader dimensions to the tractor size.

The loader was fabricated and considering the optimum dimensions estimated through the above mentioned steps and tested in the field. The optimal design means; the loader boom components dimensions, weight of steel parts and maximum grab load that satisfy loader balance without the need for rear counterbalance weights.

2.3 Theoretical foundation approach

Starting from the cane load of the grab, the function of the loader is to pile, grab, lift and load a cane bundle on the transport vehicle. The size of the bundle represents the basic dimension according to which grab dimensions estimated. Either manually or mechanically the harvested cane is pilled in the form of windrow waiting for loading. The loader grab holds cane pile compressing the stalks to each other to minimum spaces among cane stalks only at the point of holding. The maximum load for the tractor mounted sugarcane loader considered 0.5 tons. Table 1 show the principle dimensions of the cane bundle estimated based on a maximum load 500 kg cane.

Table 1 Malleable sugarcane stalks properties related to bundling and grabbing

Item		Value	
		Range	Approximated Results
Malleable stalk diameter, cm		1.5 – 3.2	2.2
Computed cross section area, cm^2		1.3 – 8.0	3.80
Malleable stalk weight, kg		0.3 – 1.5	0.85
Clean topped cane stalk (one year age) length, m		1 – 2	1.6
Spaces binet cane stalks in hydraulically pressed bundle	Erect cane, %	10	10
	Lodged cane, %	Up to 20	20
Maximum grab design load, kg		500	200
Number of stalks in the 500 kg round bundle, stalks		500 ÷ 0.85	600
Cross section area of the tight compressed 600 stalks round bundle (taken from erect cane), cm^2		(600 ×3.8) + (600 ×3.8 ×0.1)	2340
Cross section area of the tight compressed 600 stalks round bundle (taken from lodged cane), cm^2		(600 ×3.8) + (600 ×3.8 ×0.2)	2736
Radius of the cross section of 500 kg round bundle (taken from erect cane crop), cm		$r=\sqrt{2340 \div \pi}$	27.3
Diameter of the cross section of 500 kg round bundle (taken from erect cane crop), cm		27.3 × 2	54.6
radius of the tight compressed 600 stalks round bundle (taken from lodged cane), cm^2		$r=\sqrt{2736 \div \pi}$	29.5
Diameter of the cross section of 500 kg round bundle (taken from lodged cane crop), cm		29.5 × 2 = 59	60
Computed radius of the tight compressed 500 kg bundle of cane crop, cm		27.3 – 29.5	30
Computed diameter of the tight compressed 500 kg bundle of cane crop, cm		54.6 – 59	60

2.3.1 Principle grab dimension based on maximum cane bundle

Figure 2 shows the dimensions at maximum grab open. Table 2 shows the values of these dimensions based on the maximum cane bundle size.

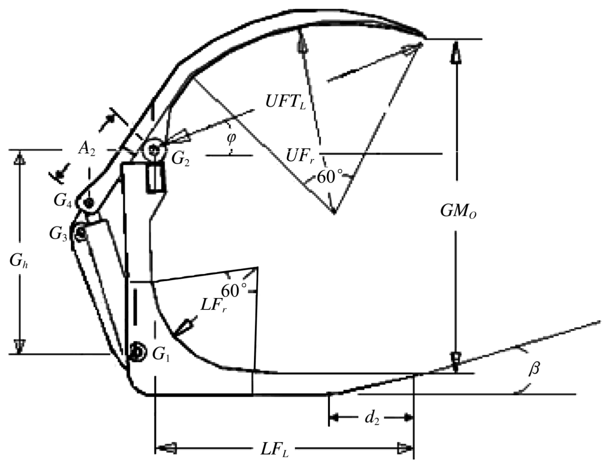

Figure 2 Dimensions estimated for the grab assembly

Table 2 Principle dimensions of the sugarcane grab based on the tight compressed sugarcane bundle.

Item	Symbol	Computation	Approx. dimension /cm
Grab basic dimension, cm	D_B	Diameter of the largest cane bundle	60
Grab height from the center of loader boom suspension to the center of upper fork hinge, cm	G_h	$G_h = 1.25D_B$ $G_h = 1.25 \times 60 = 75$	75
Lower fork length, cm	LF_L	$LF_L = D_B + 1/3D_B$ $LF_L = 60 + (1/3) \times 60 = 80$	80
Radius of the inside curvature of the grab body- lower fork, cm		$LF_r = D_B/2 = 60/2 = 30$	30
Upper fork tin length, cm	UFT_L	$UFT_L = LF_L + 0.25D_B$ $= 80 + 0.25 \times 60$	95
Radius of the inside curvature of the upper fork radius, cm	UF_r	$UF_r = D_B = 60$	60
Clamping arm length, cm	A_3	$A_3 = CA_L = 2/3G_h$ $= (2/3) \times 75 = 50$	50
Length of the trimmed end of the lower fork, cm	d_2	d_2 = lower fork thickness ÷ sinβ = 7 ÷ sine15 = 27	25
Upper fork arm, cm	A_2	$A_2 = 0.25UFT_L$ $= 0.25 \times 95 = 23.75$	25
Total length of the upper fork, cm	UF_L	$UF_L = UFT_L + GA_L$ $= 95 + 25 = 120$	120
Angle of grab max. open over horizontal, degree	φ	$\varphi = \cos^{-1}(LF_L/UFT_L)$ $= \cos^{-1}(80/95) = 32.6$	30°
Grab maximum open, cm	GM_O	$GM_O = G_h + UFT_r \sin\varphi$ $= 75 + 95\sin30 = 122.5$	120
Grab cylinder closed length, cm		$=\sqrt{(25^2+75^2)-2(25\times75\cos 30)}$ = 54.8cm	55

The diameter of the largest cane bundle (supposed to be held and lifted by the grab) will be considered to decide the grab dimensions. The grab is intended to be a push pillar type where the lower fork push and form the cane bundle. Actually the lower fork slides under the cane windrow while the open grab pushes the cane for pilling. The outer surface of the lower fork which is supposed to slide underneath the cane windrow on the soil surface should have straight extension of the front tip for efficient pilling performance. The upper fork of the grab designed to perform the function of catching the pilled cane, grabbing and pressing it into bundle then sliding that bundle inside the grab. Therefore, the upper fork length should be longer than the lower fork to maintain catching the cane pilled over the lower fork and force it inside the grab. The front part of the lower fork should be upward trimmed to reduce the probability of soil penetration. Other dimensions of the grab may be estimated in relation to the correct positions of the grab while pilling and grabbing.

2.3.2 Principle dimensions related to base-boom assembly and boom-grab assembly at pilling situation

Optimum dimensions of the loader boom were estimated based on the ideal situation for pilling a cane bundle. Figure 3 shows the loader assembly at pilling situation. The loader boom and grab dimensions as well as relative positions to the ground surface should maintain the following criteria:

I) Criteria of the ideal situation of boom-grab assembly

1) The grab should take a position where the lower surface of the trimmed end of the lower fork should be parallel to and smoothly slide over soil surface.

2) At this particular situation the transverse support of the grab body should have clearance ***c*** with the ground surface and the outer face of the lower fork makes angle β with respect to horizontal. This may minimize damage to crop roots in case of infield loading and minimize trash and dry leaves as well as dirt collected by the grab. This will also maintain pilling a clean cane bundle with minimum push force.

3) The clamping mechanism. This mechanism consists of a double side arm of a middle pivot point and

a link. The arm has to hitch points at its two ends. The clamping cylinder connected to the upper side and the clamping link connected to the lower point. When the clamping cylinder stretch pushes the upper part of the arm forward and the lower part move rearward that pivot the grab upward against gravity as indicated in the drawing.

a) To facilitate easy operation of the loader with minimum mistakes of the driver, the dimensions of the clamping mechanism parts and their relative position should achieve the condition of the grab be at pilling situation when the clamping cylinder is fully stretched. This will make the operation cycle faster and the pilling operation more efficient.

b) The fabricator should be sure that clamping cylinder closed length + clamping cylinder arm length more than the distance between the two hitch points of the clamping mechanism base for the cylinder do not change the direction of the clamping link motion.

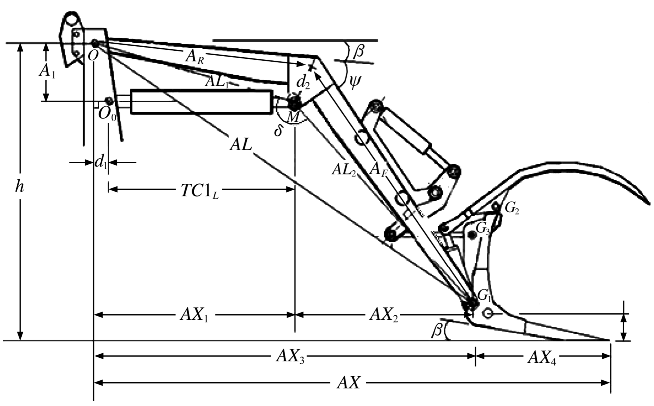

Figure 3 Principal dimensions estimated from ideal push pilling situation

II) Criteria of the ideal situation of the base-boom assembly

1) The loader boom hitch point *O* should not be higher than the level of the steering wheel and should be within reach of the labor standing on the ground for easy attaching and removal of the loader to or from the bases.

2) The point O_0 of the boom lift cylinder hitch to the base is displaced toward the front with suitable distance with respect to the vertical line passes through the boom hitch point on the base.

3) At the ideal pilling situation, the rear part of the loader boom A_R should be horizontal or lowered down with angle β with respect to horizontal. This will maintain clear view over the loader boom from the position of driver seat.

4) The loader boom lift cylinder *C*1 should be completely closed and horizontally oriented. At the situation of pilling.

5) The loader boom fabricated where the front part of the loader *AF* bent down 45° with horizontal (i. e. the angle between *AR* and *AF* = 135°).

6) Considering the hitch points, the distance $OM = AL_1$ and $MG_1 = AL_2$ represent the straight lines between the hitch points and the angle between the lines AL_1 and $AL_2 = \delta = 150°$. The distance $OG_1 = AL$ which is the actual length of the boom.

7) The length of the part of the boom in front of the clamping mechanism base = the longitudinal dimension of the clamping mechanism base.

8) At the situation of pilling, the point G_3 will be on the vertical line pass through the point G_1.

9) The clamping link will take horizontal position at the situation of pilling.

Tables 3 and 4 show the dimensions of base-boom assembly and the dimensions of grab-boom assembly at pilling situation computed for the sugarcane grab loader mounted to the Belarus 90 hp tractor of the project. This type of tractor represents the most prevailing tractor types in sugarcane production area.

Table 3 Dimensions related to boom-base assembly

Item	Symbol	Computation	Approx. dimension
Height of the boom rear end mounting point, cm	h	Height of the tractor steering wheel ≥ h (for prevailing tractors)	190
Vertical distance between the point of boom hitch to the point of boom lift cylinder hitch on the base, cm	A_1	At the situation of pilling $A_1=AL_1 \sin\beta=150\sin15=39$ At this particular situation	40
Horizontal distance between the point of boom hitch and the point of cylinder hitch on the base, cm	d_1	$d \geq 15$ cm	15
Length of the rear part of the boom arm, cm	A_R	For the medium size tractor's	155
Straight distance from the point of boom hitch to the point of *C*1 piston hitch on the boom, cm	AL_1	$AL_1 = A_R \cos\beta$ $AL_1 = 155 \cos 15 = 149.7$	150
Projection of the rear part of the loader boom at the position of pilling, cm	AX_1	$AX_1 = AL_1 \cos\beta$ $AX_1 = 150 \cos 15=144.9$	145
Total lift cylinder closed length including cylinder ends, cm	$C1_L$	$C1_L \geq AX_1 - d_1$ $C1_L \geq 145 - 15$ $C1_L \geq 130$	130

Table 4 Dimensions based on the pilling situation

Item	Symbol	Computation	Approx. dimension cm
Ground clearance under the grab body, cm	c	The back of the grab should have ground clearance within 20 cm	20
Angel of trim of the lower fork front tip, degree	β	$\beta = \sin^{-1}(c/LF_L)$ $\beta = \sin^{-1}(20/80) = 14.48$	15
Bent angel of the front part of the boom with respect to horizontal,	Ψ	$\Psi \leq 45°$	45^0
Angel between A_R and A_F, degree		$180 - \Psi = 180 - 45 = 135°$	135°
Length of the front part of the boom, cm	A_F	$A_F = [h - c)] \div \sin(\beta + \Psi)$ $= (190 - 20) \div \sin(15 + 45)$ $= 170 \div \sin 60 = 196.3$	200
Length of the straight line between the cylinder hitch point and the grab hitch point., cm	AL_2	$AL_2 = [h - (A_1 + c)] \div \sin(\beta + \delta)$ $= [190 - (40+20)] \div \sin(15+30)$ $= 130 \div \sin 45 = 183.9$	184
Angel between AL_1 & AL_2	δ	$\delta = 150^0$	150°
Projection of the front part of the boom at the situation of pilling	AX_2	$= A_F \cos(\beta + \psi)$ $= 200\cos(15 + 45)$ $= 200\cos 60 = 100$	100
Projection of the grab lower fork, cm	AX_4	$AX_3 = (LF_L + \text{thickness}) \cos\beta$ $= (80+8)\cos 15 = 85$	85
Total boom projection at pilling, cm	AX_3	$AX_3 = AX_1 + AX_2 = 145 + 100 =$	245
Total horizontal projection of the loader at pilling situation, cm	AX	$AX = AX_3 + AX_4 = 245+85 = 345$	330
Boom length (straight distance between the base-boom hitch point to the grab-boom hitch point), cm	AL	$A_R \cos 25 + A_F \cos 20$ $= 155 \cos 25 + 200\cos 20 = 327$	327

2.3.3 Principle dimensions related to loading situation

Principle dimensions related to the operational positions at loading a sugarcane bundle over a transport vehicle presented in Figure 4. While loading, the tractor should maneuver to become in certain position with respect to the vehicle. The tractor operator should maneuver to bring the tractor oriented where the longitudinal axe of the tractor perpendicular to the vehicle box longitudinal axe. The relative size of both loader and the vehicle box should maintain reasonable reach of the loaded cane bundle to the far side of the vehicle. The height of the lowest point of the grab at the situation of loading should be equivalent to the height of load expansion over the vehicle box. At the situation of discharge, the grab lower fork should be oriented down with sufficient inclination angle θ and the clamping cylinder should be completely closed. The maximum stretch length of the loader boom cylinder should maintain a loader boom lift angle Φ that conserve minimum distance X_1 of reasonable value between the front end of the tractor and the vehicle body. Minimum reach of the grab over the trailer box X_3 is to have the grab boom articulation point over the vehicle box side and the full length of the grab will be over the vehicle.

Figure 4 shows the operational dimensions of the loader relative to the transport vehicle. Table 5 shows the relative operational dimensions of the loader at the situation of loading estimated for the most prevailing tractors found in the sugarcane area.

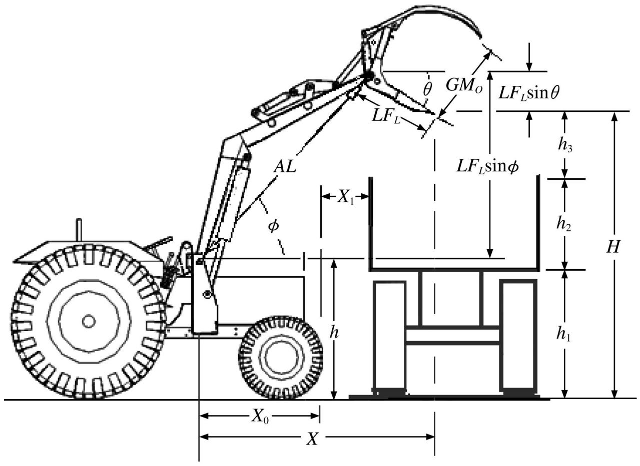

Figure 4 Dimensions related to loading situation

Table 5 Dimensions related to loading situation

Item	Symbol	Computation	Considered dimension
Horizontal distance from the boom base hitch point to the front end of the tractor front wheel	X_0	According to the tractor For prevailing tractors	135
Maximum lift angel made by the line AL	Φ	$\Phi = 75–30 = 45$	60^0
Distance between the front end of the tractor front wheel and the vehicle at loading situation, cm	X_1	$AL\sin 60 - X_0$ $327\cos 60 - 135 = 28.5$	30
Max loading height, cm	H	$H = h + AL \sin\Phi - LF_L \text{Sin}\,\Theta$ $=190+327\sin 60 - 88\sin 45 = 410$	410
Maximum reach of grab inside the trailer	X	$X = (AL\cos 60 + LF_L\cos 45) - (X_0 + X_1) = 226 - 165 = 61$	61

2.3.4 Loader boom length at non counterbalance weights added

Figure 5 and Table 6 show the parameters related to loader balance.

The concept of sufficient weight should be preserved on the rear wheel when mounting and loading the front end loader was set as follow:

- For all the tractors, the weight of the tractor W is distributed (2/3W) over the rear wheel and (1/3W) on the front wheel.

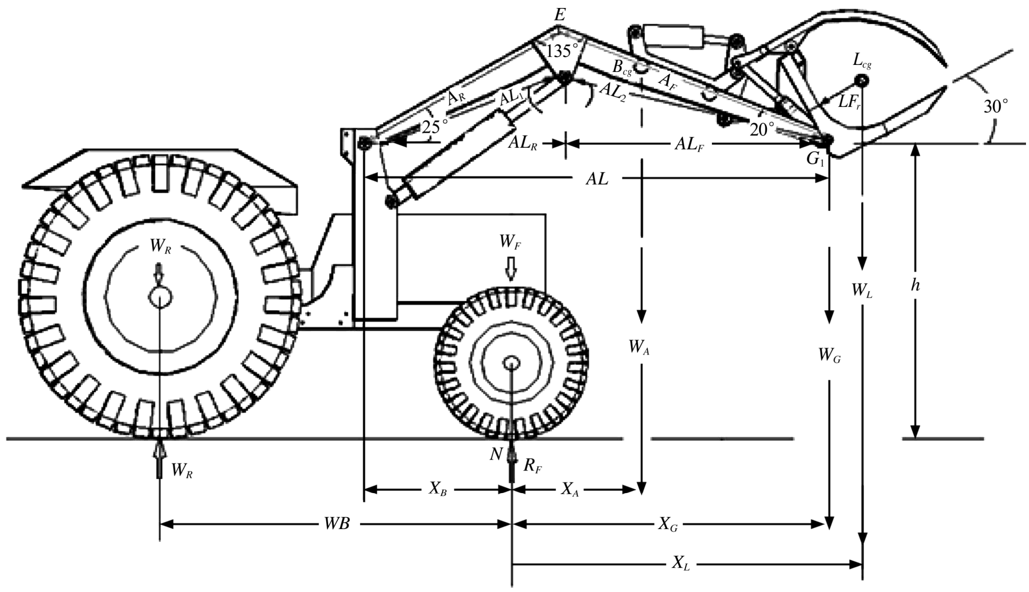

Figure 5 Parameters related to the balance of the e tractor front mounted sugarcane grab loader

Table 6 Values of parameters related to the balance of the sugarcane loader mounted on a Belarus 90 hp tractor

Item		Symbol	Computation	Value
Tractor with 70 kg driver	Tractor		balanced	3170
	Tractor + operator		balanced	3240
	Rear		balanced	2160
	Front		balanced	1080
Tractor with driver and loader 770 kg				
Tractor + loader	Tractor + loader		balanced	3940
	Rear wheels		balanced	1870
	Front wheels		balanced	2070
Weight of the loader boom with clamping cylinder and clamping arm		W_A	205 + 20 + 25 Boom Lift cylinders not included	250
Weight of the loader grab with grabbing cylinders, kg		W_G	105 +(2 × 15) +15	150
Weight of the grab load, kg		W_L	Maximum	500
Wheel base, cm		WB	According to tractor catalog	240
Distance from the point of loader hitch to the tractor front axle, cm		X_B	Measured after assembly	100
Horizontal distance of the boom center of gravity to N, cm		X_A	= {AL – 2(75) cos 25–X_B = (327 – 150 cos 20) – 100 = 86	86
Horizontal distance from the grab center of gravity to N, cm		X_G	= AL – X_B = 327 – 100 = 227	227
Horizontal distance from the load center of gravity to N, cm		X_L	= AL – X_B + LF_r cos 30 = 327 – 100 +30 cos 30	253

- This ratio of weight distribution exists in all 2WD tractors and 4WD tractors (i. e. 1/3W is sufficient to support the front drive wheel of the tractor for its function as a traction wheel).

- Therefore 1/3W is sufficient to support the drive wheel against excessive slip at dynamic conditions.

- Since the tractor is dynamically balanced at this condition, so that the tractor will also be dynamically balanced if the front mounted loader with its maximum load designed to preserve 1/3W over the rear wheel.

- Therefore, the length of the boom should be computed to maintain at least 1/3W over the rear wheel of the tractor when the grab is loaded with its maximum load at the most critical situation while lifting the load.

- From the point of view of balance, the most critical situation of lifting the grab exists when the dimension AL become horizontal while lifting the load.

- The front end loader expected to transfer the weight from the rear wheel to the front wheel. If the weight over the tractor rear wheels (WR) became considerably small because of the long boom and/or overloaded grab, excessive slip of the rear wheel may occur.

- Taking moments around the point *N* shown in Figure 12 we find:

$$WR \times WB + WF \times Zero - W_A \times X_A - W_G \times X_G - W_L \times X_L = 0$$

- At this particular situation (when the weight on the rear wheel become zero), the tractor rear wheel may jump up and the tractor is about to be inversed to the front.

- Considering the above concept keeping 1/3W over the rear wheel to secure balance and efficient traction performance. The above equation should be:

$$W_2 R \times WB + W_2 F \times Zero - W_A \times X_A - W_G \times X_G - W_L \times X_L = 1/3\ W_1 \times WB$$

To facilitate computing the length of the loader boom, the following items will be considered:

- Since the concern is to prove the longitudinal stability of the tractor mounting sugarcane loader, longitudinal dimensions will only be considered.

- The length of rear part of the boom A_R = 155 cm is of size sufficient for the most prevailing medium size tractors for which the sugarcane loader designed and will be considered constant.

- The front part of the boom A_F could be variable according to the size of the tractor. The value of A_F will be computed as a result of the tractor balance equation. Table 6 presents the values of the parameters related to the balance of the tractor on which the front mounted loader is attached.

- Table 7 presents the dimensions of the tractor front mounted balance parameters in terms of A_F.

Table 7 Values the parameters related to balance of the sugarcane grab loader mounted on a medium size tractor in terms of boom length

Parameter		Symbol	Computation	Value
	A_R		Constant	155
	A_F		Variable	200
In triangle OEG_1	Angle OEG_1		$OE\ G_1 = 180 - \psi$	135^0
	Angle EG_1O		$\sin^{-1} E\ G_1O = (OE \sin 135) \div AL$ $= (155 \sin 135) \div 327 = 19.8$	20
	Angel EOG_1		$180 - E - G_1$ $= 180 - 135 - 20 = 25$	25
Dimension in terms of A_F		AL_R	Constant for medium size tractors $A_R \cos 25 = 155 \cos 25 =$	140
		AL_F	Variable $A_F \cos 20$	$0.94A_F$
		AL	$AL = AL_R + AL_F$ $= A_R \cos 25 + A_F \cos 20$ $= 155 \cos 25 + A_F \cos 20$ $= 140 + 0.94\ A_F$	$(140 + 0.94A_F)$
		X_L	$X_L = \mathrm{AL} + LF_r \mathrm{Cos}\ 30 - \mathrm{X_B}$ $= AL + 0.30 \cos 30 - 1$ $= AL - 0.75$ $X_L = 1.4 + 0.94A_F - 0.75$	$(0.94A_F + 0.65)$
		X_G	$X_G = A_L - X_B = AL - 100$ $X_G = (1.4 + 0.94A_F) - 1$	$(0.94A_F + 0.40)$
		X_A	$X_A = AL - (2 \times 0.75A_F \sin 20) - X_B$ $= AL - 0.51A_F - 1$ $= 1.4 + 0.94A_F - 0.51A_F - 1$	$(0.43A_F + 0.40)$

Now Let us go for the following sub-steps to prove the stability of the loader at the boom length computed to satisfy pilling situation and at the maximum grab load. Referring to Figure 4 you will see the loader at the most critical situation while lifting the load. Table 6 presents, values of the loader components (weights and arms) for the sugarcane loader mounted on a 90 hp Belarus tractor.

I Actual WR at the given boom length: For the current tractor (Belarus 90 hp) and considering the dimensions of the current loader computed for optimum pilling and grabbing situation that resulted straight boom length 327 cm, A_R = constant =155, A_F = 200 cm , Ψ = 45°. The computed weight over the rear wheel = Actual WR computed as follow:

2090× 2.4 + 1850 × Zero – 250 × 0.86 – 150 × 2.27 – 500 × 2.53 = 3195 kg.m

Actual WR at maximum grab load = 3195 ÷ 2.4 = 1332 kg

II Computing maximum length of the variable part of the boom A_F (conserving WR= 1/3W):

The items X_A $(0.43A_F + 0.40)$, $X_G = (0.94A_F + 0.40)$, $X_L = (0.94A_F + 0.65)$ are developed based on the terms found in Fig (4) and Tables (6 and 7).

Compensating the values of X_A, X_G, X_L in terms of A_F as follow:

$W_2\ R \times WB - W_A\ (0.43A_F + 0.40) - W_G\ (0.94A_F + 0.40) - W_L\ (0.94A_F + 0.65) = 1/3W_1 \times WB$ 2090×2.4 – 250 $(0.43A_F + 0.40)$ – 150 $(0.94A_F + 0.40)$ – 500 $(0.94A_F + 0.65)$ = 1080 × 2.4

$5016 - 3077 = 719A_F$

$A_F = 2.7$ m

III Computing maximum boom straight length AL conserving WR =1/3 W as follow:

$AL = 140 + 0.94A_F = 140 + 270 \times 0.94$

$AL = 140 + 254$

$AL = 394$ cm

V Computing the loading height for loader of maximum boom straight length conserving WR = (1/3W):

$H = h + AL \sin \Phi - LF_L \cos \theta)$

$H = 190 + 363 \sin 60 - 80 \sin 45 = 475$ cm

Table 8 presents the final value (size) of computed balance parameters of the tractor front mounted sugarcane loader with maximum grab load 500 kg. Table 9 includes the verification of the sugarcane loader balance.

Table 8 Values of principle parameters as a result of sizing the tractor front mounted loader with no counterbalance

Tractor Weight, kg W_1	WB, m	W_1 R Tractor only	Tractor + loader W_2	W_2 R Tractor + loader (AL=3.27)	W_2 F Tractor + loader (AL=3.27)	1/3 W_1	Actual WR	A_F up to	Max. Boom length AL, m	Loading height at Φ= 60^0
3240	2.4	2160	3940	2090	1850	1080	1332	2.7	3.94	4 .75

Table 9 Verification of the balance of the tractor mounted sugarcane loader grab loader

Design Parameter	Balance Range	Actual value	Results
Weight on the Tractor Rear Wheel (WR), kg	≥ 1080	1322	Balanced
Boom Length (AL), m	≤ 394	327	Balanced
Height of Loading, m	≤ 475	410	Balanced

General formula:

This range of computations may cover several types of tractors of power range from 70 to 90 hp, since the tractor weight and the wheel base of these types very close to the values used in the computations. This range mainly covers the loader size mounted to the Belarus MTZ tractor and the Universal UTP tractor that represent 78% of the tractors found in the sugarcane production area. Other tractors of power larger than 90 hp may be of larger weight and wheel base on which longer boom size could be used. A general formula may be developed for sizing the loader for other tractors as follow:

For any tractor:

The items $X_L = (0.94A_F + 0.65)$, $X_G = (0.94A_F + 0.40)$, X_A $(0.43A_F + 0.40)$ are developed based in the terms found in Figure 4 and Tables (6 & 7).

$W_{(T+L)}R \times WB - W_A\,(0.43A_F + 0.40) - W_G\,(0.94\,A_F + 0.40) - W_L\,(0.94A_F + 0.65) = 1/3W_T \times WB$

$W_{(T+L)}R \times WB - 1/3W_T \times WB = (0.43A_F \times W_A + 0.94A_F \times W_G + 0.94A_F \times W_L) + 0.40W_A + 0.4W_G + 0.65W_L$

$WB\,(W_{(T+L)}R - 1/3W_T) = A_F\,(0.43W_A + 0.94W_G + 0.94W_L) + 0.40W_A + 0.4W_G + 0.65W_L$

The rear part of the boom of 155 cm was found to be sufficient for most of the medium tractors. To find the boom length of ant tractor:

Compensating the values of W_A = 250 kg, W_G = 150 kg, W_L = 500 kg

$A_F\,(0.43 \times 250 + 0.94 \times 150 + 0.94 \times 500) = (0.40 \times 250 + 0.4 \times 150 + 0.65 \times 500) - WB\,(W\,R_{(T+L)} - 1/3W_T)$

$719A_F = 485 - WB\,(W\,R_{(T+L)} - 1/3W_T)$

$$A_F = \frac{485 - WB(WR_{T+L} - \frac{1}{3}WR_T)}{719} \quad (1)$$

The values of constants 485 and 719 satisfy all the above assumptions related to maximum load and loader components steel weights. For whoever desires to change these assumptions, a general form may be developed.

Or in more generalized form;

$$A_F = \frac{a - WB(WR_{T+L} - \frac{1}{3}WR_T)}{b} \quad (2)$$

where, A_F: Length of the front part of the boom as a part of variable dimension (depend on the tractor size) that satisfy the proper situation of pilling and satisfy the stability of the tractor on which the loader is loaded at $1/3WR_T$. a, b: Are constants depend on dimensions and weights of the loader component and the maximum design load of the grab. WB: Tractor wheel base represents the distance between the front and rear wheel centers. WR_{T+L}: Weight on the rear wheel of the tractor on which the empty loader mounted. WR_T: Weight on the rear wheel of the tractor before mounting the loader. $1/3WR_T$: Weight should be conserved on the tractor rear wheel that secures stability of the tractor front mounted sugarcane loader at dynamic conditions.

4 Results and Discussion

The loader was fabricated according to the optimization described previously, assembled to the tractor and tested at several locations in the sugarcane production area in the Nile Valley. The tractor-front-mounted-loader was tested for loading the main sugarcane transport vehicles namely; decauvelle wagons, trailers and lorries that transport sugarcane from loading sites to the sugar mill. Figure 6 show the tractor-front-mounted sugarcane grab loader in operation. The field evaluation of the loader included recording the important notes concerning the balance of the tractor while operation. The test also included estimating loading cycle time, loading rate and operation efficiency while loading the prevailing sugarcane transport vehicles.

As shown in Figure 6, the tractor front mounted loader does not provided with rear counterbalance weights. The loader with loaded grab is moving over sloped land with no problem concerning stability. For a full season operation in all possible conditions either inside sugarcane fields or in sugarcane loading sites, the loader proved high stability. No problem concerning the tractor balance recorded during a full season of operation. The figure shows that the loader push-pile and hold the cane bundle inside the grab efficiently. The loader lifts the cane bundle up while maneuvering and travelling for loading with sufficient stability. The loader can also load the cane up to reasonable height over the transport vehicle box level.

Figure 6 The tractor-front-mounted sugarcane loader in operation.

The loader was operated in the fields of sugarcane to load decauvelle wagons that travel over a narrow rail slide lines. The machine was also operated to load other road transport vehicles represented in lorries and trailers equipped for sugarcane transport. The vehicles equipped for sugarcane transport are provided with side columns' with 1.75 m height. The vehicle load of sugarcane may vertically expanded over the side columns'. The vehicles are loaded in the sugarcane loading site where several farmers bring their cane to be loaded and transported to the mill.

Loading cycle time as one of the important parameter affecting the loader performance is shown in Figure 7. For site loading, the variation of cycle time was limited to small range from 1.68 to 1.76 min/cycle. Figure 8 show the loading rate of the loader while loading the main transport vehicles in the sugarcane site. Loading rate of the tractor-front-mounted loader ranged between 12 to 13 ton/h for all the loaded sugarcane vehicles. As the most important evaluation parameter, loading rate depends on loading cycle time and average grab load. Average grab load did not affected by the type of the vehicle and stay almost 400 kg/cycle. Therefore, the limited variation of the loading rate may mainly refer to the variation of the cycle time and efficiency. Figure 9 show loading efficiency of the tractor-front mounted loader while loading the main sugarcane transport vehicles in the site. Efficiency is a parameter that largely affected by the operation conditions. Since the operation conditions of the sugarcane loading sites do not vary so much, it is not expected to find wide variation concerning operation efficiency. Actually, the lower value of loading efficiency was very close to 90% and the higher value was up to 94%.

Data concerning the loader fabrication cost, tractor operation cost loading rate and efficiency were analyzed to estimate the machine operation costs. The cost analysis proved that the tractor front mounted sugarcane loader is a cost effective machine within wide range of operation conditions.

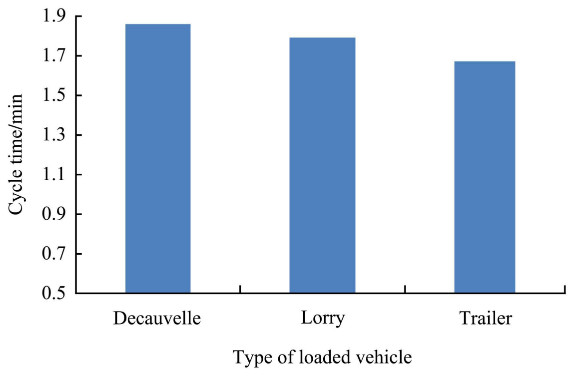

Figure 7 Loading cycle time

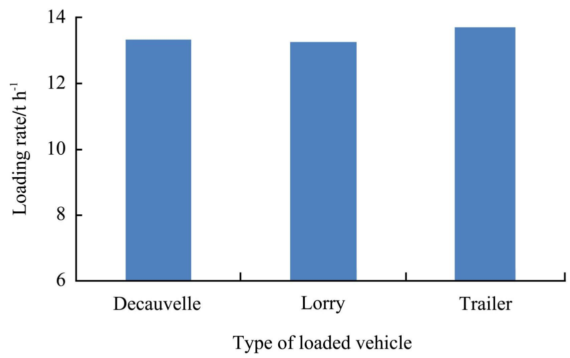

Figure 8 Loading rate

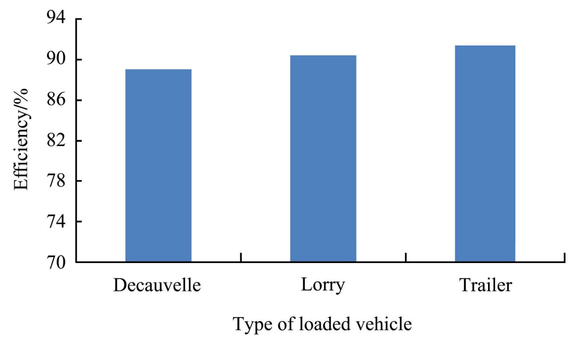

Figure 9 Loading efficiency

5 Conclusions

The dimensions of the tractor-mounted-loader should be relative to the tractor size. The design of the loader bases should be compatible to the type of the tractor on which the loader mounted. Dynamic balance of the tractor-mounted loader while operation is the main condition determines the size of the loader. Optimizing the design of the tractor mounted sugarcane loader facilitates fabricating a balanced tractor front mounted loader that may load the transport vehicles to the maximum load height. Necessary measurements and computations were conducted to facilitate optimizing the design of the tractor front mounted loader without the need for counterbalance weights. The technique was used to compute the principle dimensions and to estimate the tractor longitudinal balance of the loader mounted on the medium size tractor. General formula was also developed to facilitate estimating the loader dimensions for a tractor with any size conserving the condition of balance.

The tractor-front-mounted-loader was fabricated and mounted on a Belarus 90 hp tractor (the most available tractor type in the sugarcane production area in Egypt). The machine was operated for loading all types of sugarcane transport vehicles with no problem recorded from the point of view of balance. The machine evaluation results show that the loading cycle time within 1.7 min. The machine may achieve up to 13 ton/h loading rate and operation efficiency within 90%. Farmers accepted the loader as a cost effective machine performed efficiently at wide range of operation conditions.

Acknowledgment

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