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Replacing Special Mechanism of Tractor with Planer Mechanism

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ABSTRACT: This research paper focuses on improving the performance of the four-bar mechanism commonly used on tractors to accommodate plows, with the aim of better meeting the requirements of farmers during field preparation for the upcoming harvest season. Traditionally, tractor manufacturers have employed a spatial four-bar mechanism for this purpose. However, our study proposes the adoption of a planar mechanism as an alternative approach. We investigate the feasibility and effectiveness of this change by conducting experimental analyses and comparative assessments. The results of our research provide valuable insights into the potential advantages and implications of implementing a planar mechanism in place of the traditional spatial four-bar mechanism for tractor operations in agricultural settings.

KEYWORDS: Tractor, agricultural machinery, special mechanisms, planar mechanisms, four-bar mechanism, technology integration, efficiency, reliability, sustainability, farming practices.

I. INTRODUCTION

The agricultural sector plays a vital role in India's economy, providing livelihoods to a significant portion of the population. However, despite advancements in technology across various industries, agricultural practices in India often lag behind in terms of automation and efficiency. This is particularly evident in tractor operations, where the costs associated with plowing and other tasks can be significant for farmers, impacting their overall profitability.

To address this issue, this project aims to explore methods for reducing the operating costs of tractors by developing an economical and efficient mechanism for plowing operations. Over the past few decades, there has been a rapid growth in knowledge and understanding in the field of kinematics and dynamics of machinery. Kinematic synthesis, a branch of mechanical engineering, focuses on designing mechanisms to achieve specific motion characteristics. By leveraging principles from kinematic synthesis, we aim to design a mechanism that not only optimizes performance but also minimizes operating costs.

Central to the design of the mechanism is the four-bar linkage, a fundamental mechanical system widely used in various applications. In the context of tractor operations, the four-bar mechanism plays a critical role in coordinating the motion of different components involved in plowing. One of the key considerations in designing a four-bar mechanism is the transmission angle, which determines the smoothness of operation. Typically, designers aim to maintain equal deviation of the maximum and minimum transmission angles from 90° to ensure optimal performance. However, achieving precise coordination between input and output motions can be challenging due to working inaccuracies in machine parts.

Another important factor in mechanism design is the ability to transmit torque and forces effectively. In agricultural machinery like tractors, where heavy-duty tasks are common, minimizing the force required for operation is essential for efficiency and cost-effectiveness. Four-bar linkages are well-suited for this purpose, given their ability to distribute forces efficiently across different components. By optimizing the design of the four-bar mechanism, we aim to minimize torque ratio variation within the operating range and ensure that the speed ratio between input and output remains within specified limits.

However, designing an efficient mechanism for tractor operations involves more than just optimizing geometric parameters. It also requires an understanding of the unique challenges and requirements of the agricultural sector. In India, where agriculture is a primary source of livelihood for millions of people, any technological advancements must be accessible and affordable to farmers. Therefore, our project focuses not only on improving performance but also on ensuring that the proposed mechanism is economical and easy to implement.

II. LITERATURE REVIEW

Emanuela et al. (2014) conducted a study on LOCOSTRA, a sustainable tractor engineered for agricultural operations and humanitarian demining tasks. They found that LOCOSTRA had superior power-to-mass ratio and reduced mass compared to similar machinery, performing well in both agricultural and demining scenarios.

G.P. Moreda et al. (2016) explored the impact of high voltage electrification in agricultural machinery, particularly tractors. They discovered substantial efficiency improvements, including heightened energy efficiency, reduced fuel consumption, and lower CO2 emissions. Electric drives also enabled enhanced torque and speed control, noise reduction, and increased operator comfort.

Subodh Patil et al. (2019) utilized Finite Element Analysis (FEA) to assess the stress and safety factor values of a tractor's lift arm within a three-point hitch system. They found that using AISI 1040 material and design modifications significantly enhanced the safety factor and reduced deformation, recommending its adoption for improved load-bearing capacity.

R. Abrahám et al. (2014) demonstrated the effectiveness of special wheels with auto-extensible blades in increasing a tractor's drawbar pull, particularly on soil with varying moisture levels. They observed significant improvements in drawbar pull and reduced wheel slip, potentially leading to decreased fuel consumption and minimized soil damage.

Chi Zhang et al. (2017) conducted a comprehensive examination of a multi-robot tractor system designed for agricultural field work. Through simulations, they found that the system exhibited high efficiency, especially in larger fields, by coordinating robot tractors to form spatial patterns and execute turns without collisions.

Md. Israil Hossain et al. (2012) provided evidence of the advantages of strip tillage seeding equipment in dry land farming. They discovered significant increases in effective field capacity, reductions in fuel consumption and planting costs, and improved crop yields compared to conventional seeding systems.

Linenko et al. (2019) evaluated a multi-robot tractor system's performance through simulations, emphasizing its efficiency in different field sizes and formation patterns. They highlighted the system's capability to reduce total work time and enhance efficiency by coordinating robot tractors to form spatial patterns and execute turns safely.

Abdullah et al. (2018) conducted a kinematic analysis of a tractor engine's crank-rod mechanism using MSC ADAMS software, providing insights into mechanical system performance and optimization. They demonstrated the software's accuracy in replicating real-world mechanical phenomena and highlighted the importance of selecting optimal crank-rod dimensions for efficient engine performance.

Zhe Xin et al. (2023) introduced a terrain-adaptive hitch mechanism tailored for hilly tractors, showcasing improved lifting performance and adaptability in complex terrains. Their findings highlighted advancements in agricultural technology for addressing challenges in rugged terrains and optimizing operational performance.

Pessina D. et al. (2015) emphasized the critical role of Roll Over Protective Structures (ROPS) in mitigating injuries during tractor overturns. They advocated for higher load limits for foldable roll-bars on narrow-track tractors to improve ergonomic considerations and tractor safety.

III. DESIGNING A MECHANISM

Brodel and Soni have developed an analytical method of synthesizing the crank-rocker linkage in which the time ratio Q =1

$$\gamma \min = 180° - \gamma \max$$

$$\gamma \min + \gamma \max = 180°$$

Where γ is transmission angle.
As a first attempt it is decided to use the result obtained by Brodel and Soni for a crank-rocker mechanism with optimum transmission angle. The result obtained by them is as under

$$\frac{r_3}{r_1} = \sqrt{\frac{1 - \cos\theta}{2\cos^2 \gamma \min}}$$

$$\frac{r_4}{r_1} = \sqrt{\frac{1 - \left(\frac{r_3}{r_1}\right)2}{1 - \left(\frac{r_3}{r_1}\right)2 \cos^2 \gamma \min}}$$

$$\frac{r_2}{r_1} = \sqrt{\left(\frac{r_3}{r_1}\right)2 + \left(\frac{r_4}{r_1}\right) - 1}$$

A. Selection of Transmission Angle

The transmission angle, denoted as γ, represents the angle between the coupler and the rocker in a four-bar linkage system and serves as a crucial indicator of its effectiveness. When the transmission angle deviates significantly from a right angle, the mechanical advantage decreases. If it diminishes too much, even slight friction can lead to the mechanism locking or jamming. To mitigate this issue, it's generally advised not to operate a four-bar linkage when the transmission angle falls below approximately 45° or 50°. Optimal force transmission in a four-bar linkage is achieved when the transmission angle deviates minimally from 90°. Thus, our design approach aims to develop a mechanism with transmission angles deviating by a maximum of 30° from 90° in either direction.

$$\gamma \max = 90° + 30° = 120°$$

$$\gamma \min = 90° - 30° = 60°$$

B. Deciding the swing of rocker

Farmers prepare fields for the next harvest using ploughs, which dig soil to depths ranging from 3 inches to 30 inches, depending on soil type. These ploughs are attached to a four-bar mechanism mounted on the tractor, allowing for soil penetration. The rocker's swing must enable plough blade penetration to a maximum depth of 30 inches without causing harm to any tractor parts. Additionally, the plough blade must be safely released from the soil and lifted to a height ensuring safety when the tractor is parked. Hence, parameters for the mechanism are determined accordingly.

i) Length of rocker from pivot to the position Where the plough is attached = 28”
ii) Swing of rocker = 34°
iii) Height of fixing pivot of rocker from ground = 8.8”

The ploughs which are manufactured are such that the mechanical arrangement which is provided for fixing it to a tractor mechanism is such that it remains about 12’’ to 15’’ above the ground whereas the blades remain penetrated in the ground. The modified arrangement of rocker is shown in Fig. 1 and 2 the height of fixing pivot of rocker is changed from 8.8’’ to 22’’ and (ii) the length of rocker is also increased from 28’’ to 36’’.

While deciding the length of rocker, due consideration has been given to the swing angle. From geometrical principles we have decided to use the length of rocker AE as 36’’ so that the swing angle is about 48°. With this angle of swing and length of rocker we can lift the plough to a position EI such that the plough tip is just above the ground. If we intend to lift beyond position. EI we have to increase the rocker angle by certain desired amount. However this can also be achieved by rotating the crank beyond the limiting position, and since it is a no-load lifting it does not harm the mechanism in any way. Hence the new constrains on the mechanisms for rocker are,

i) Length of rocker from pivot to the position Where the plough is attached = 36”
ii) Swing of rocker = @ 48°

iii) Height of fixing pivot of rocker from ground = 22”

The swing angle of rocker is decided by considering the difference between fixing arrangement of the upper part of plough blade as 12".

If the difference between fixing arrangement and the upper part of plough-blade is 15" then the bottom most ploughing position would be AE3 and thus the swing angle requirement is further reduced.

The bottom pivot A is located 22" above the ground level and the second pivot D is planned to be positioned at a distance of 20" from the first pivot, on the body of tractor.

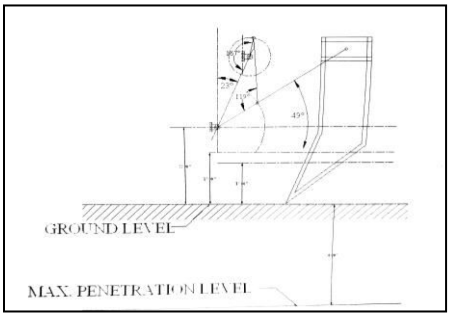

Fig. 1

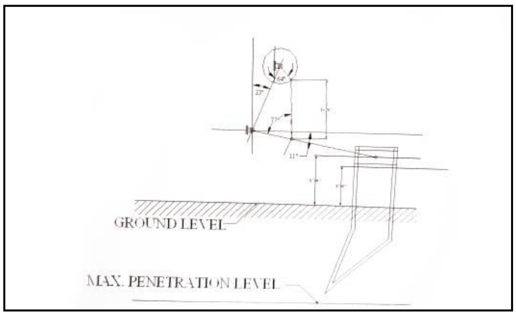

Fig. 2

C. Deciding length of other links

With the considerations as mentioned above we have decided the parameters to be used in Brodel and Soni's equations as under.

i) $\gamma \min = 60°$

ii) Distance between pivot centre i.e $r_1 = 20$’’

iii) $\gamma = 50°$

$$\frac{r_3}{r_1} = \sqrt{\frac{1 - \cos 50°}{2\cos^2 60°}} = 0.8452$$

$$\frac{r_4}{r_1} = \sqrt{\frac{1-(.8452)2}{1-(.8452)2\ c\cos^2 60}}$$

$$\frac{r_4}{r_1} = \sqrt{\frac{.2856}{.8214}}$$

$$\frac{r_4}{r_1} = .5897$$

$$\frac{r_2}{r_1} = \sqrt{.8452^2 + .5897^2 - 1}$$

$$\frac{r_2}{r_1} = .2492$$

Now $r_1 = 20$’’

$$\therefore r_2 = 20 \times 0.2492 = 4.984 = 5.00\text{’’}$$

$$\therefore\ r_3 = 20 \times 0.8452 = 16.904 = 16.90\text{’’}$$

$$\therefore\ r_4 = 20 \times .5897 = 11.794 = 11.80\text{’’}$$

The mechanism with the above link is drawn in Fig. 3, Fig.4 with Scale as 1cm =2’’

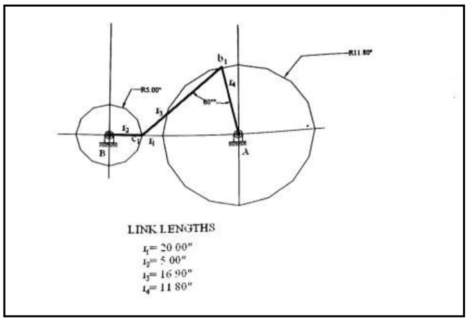

Fig. 3

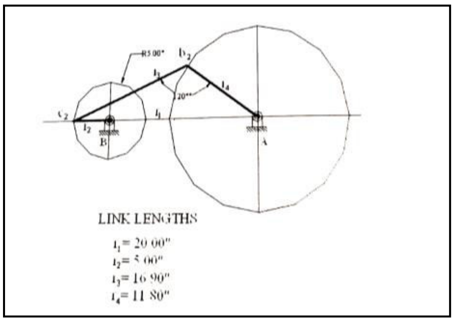

Fig. 4

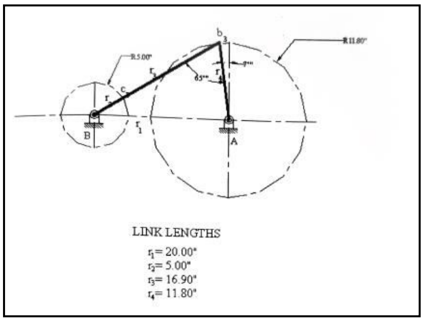

Fig. 5

From the (Fig. 3) it is very clear that the condition of γmax =120° and γmin =60° is perfectly satisfied for the link length obtained by using the Brodel's and Soni's equation. The mechanism is shown in (Fig. 5) toggle position as AB_3C_3D,and the transmission angle in this position is 65.5°.

Now, in the toggle position, the sine of the angle β is zero and the mechanical advantage is infinite. Thus, at such a position, only a small input torque is necessary to overcome a large output torque load. This is the position of a mechanism where the driver i.e. input link r_2 is a line with the coupler i.e. link 3. Now let us explore the possibility of fixing this mechanism on tractor such that when the plough is inserted in soil input the bottom most position of rocker the r_2 must be in line with r_3. The rocker is inclined with vertical line at 7° in this position in C.C.W. direction. However the inclination desired by us is 11° in C.W. direction. Hence it would be better if we mount this mechanism on the tractor in such a manner that the fixed link is inclined with the vertical axis of tractor at an angle of 23° in C.W. direction.

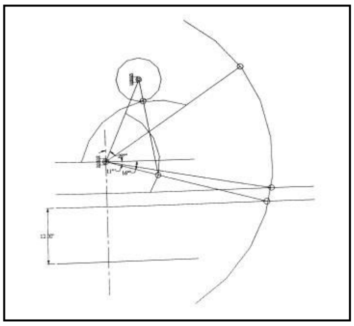

Fig. 6

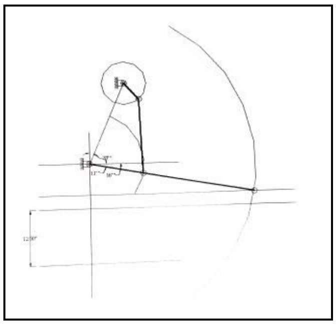

Fig. 7

The designed mechanism in the tilted position is as shown in Fig.6, Fig 7 The only undesirable characteristic of this mechanism is that the operation range of crank angle is about 0 to 148° Such a heavy range of operation will not beneficially permit the use of hydraulic actuator to operate the mechanism. Hence let us try to design a mechanism by using 3 position synthesis methods based on graphical procedure.

D. Designing a Mechanism by using 3 position syntheses

Deciding the required parameters

i) Length of the fixed link r_1 : Let us use it as obtained in the first attempt r_1 =20''

ii) Length of crank r_2 : Let us use the length of crank also the same as obtained in the first attempt i.e. r_2 =5''

iii) Positions of crank & corresponding position of rocker.

Since this mechanism is neither required to generate a path or function. nor it is used for guiding the body along a specified path with desired orientation maintained constant when the body is being moved along the path. Hence we decided to divide the range of rocker motion in 2 equal parts as under to provide three position of rocker

$$\phi_{12} = \phi_{23} = 25^\circ$$

In the chapter on optimal positioning of actuator we decide the crank range as 30° which gives the three positions of crank as under when divided in 2 equal parts.

$$\theta_{12} = \theta_{23} = 15^\circ$$

Graphical Design

The mechanism is designed graphically as shown in Fig. 8. From fig it is clear that the transmission angle obtained is very small & hence the mechanism is not suitable as far as the M.A would be less. The obtained parameters are as under.

$$r_1 = 20''; \quad r_2 = 5''; \quad r_3 = 18''; \quad r_4 = 2.8''; \quad \gamma = @11^\circ$$

Since the obtained parameters are not desirable hence let us attempt to design a mechanism using graphical method of two position synthesis.

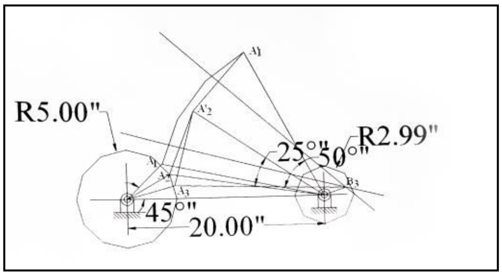

Fig. 8

E. Designing a Mechanism using Two-position Synthesis

The mechanism is designed graphically as shown in Fig. 5.3.2. The results obtained are as under,

$$r_1 = 20''; \quad r_2 = 5''; \quad r_3 = 15.6''; \quad r_4 = 1.8''; \quad \gamma = 72^\circ$$

Here no doubt that we are getting right values of i.e. 78° but the length of r_4 is very small i.e. 1.8'' & hence this design is also not suitable. Hence we decided to use Freundenstein equation for designing the mechanism.

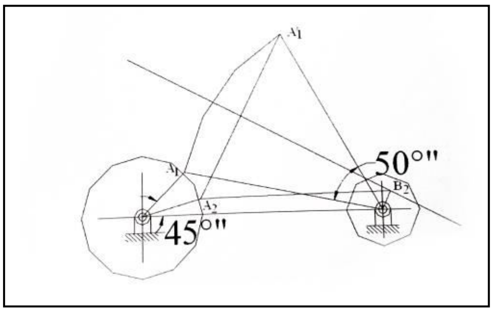

Fig. 9

F. Design based on Freundenstein equation:-

Definition of problem:

The desired input-output angle co-ordination is as shown in Fig.10 As per the mechanism designed on the basis of Brodel's & Soni's equations for optimum transmission angle the link lengths are obtained as

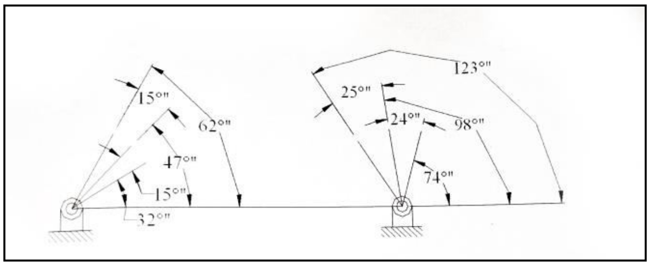

Fig. 10

$$r_1 = 20''; \quad r_2 = 5''; \quad r_3 = 16.9''; \quad r_4 = 11.8''; \quad r_{min} = 60^\circ, \quad \gamma_{max} = 120^\circ$$

When mounted on the tractor, Fig. 11, Fig. 12 is found that fixed link is tilted by 23° to vertical on right side & the crank angle w.r.t. fixed link are:

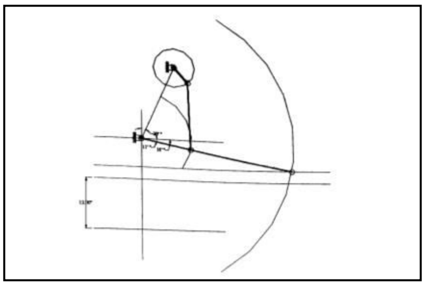

Fig. 11

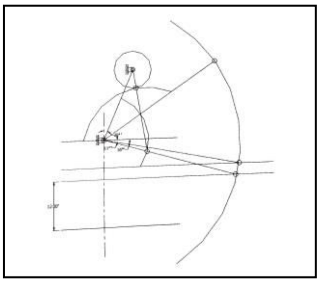

Fig. 12

i) When in toggle position i.e. in the bottom- most position for 20" digging and 12'' dist. Between fixing print & the upper most part of plough-blade

ii) When the dist between fixing print & plough is 15'' and again for 30'' digging.

iii) When is raised position or non-working position of plough

But with hydraulic actuator it is not possible to operate the crawl from 32° to 193° (Range 161°)

Hence we decided to design this mechanism using Freudenstein equation to explore the possibility of reducing range of crawl motion for desired range of 9 to 300 wing of rocker to 49°.

Thus assuming the range of crank as 30° as obtained from the calculation mode for optimal mounting of actuator, the crawl position w.r.L. fixed link are 32°, 47°, 62°. Hence the values of θ_2 to θ_2 be used in Frecedenstein equation are

Position	θ_2	θ_4
1	32°	180+74 = 254°
2	47°	180+98 = 278°
3	62°	180+123 = 303°

The Frendenstein equation is

$$K_1\cos\theta_2 + k_2\cos\theta_4 + k_3 = \cos(\theta_2 - \theta_4)$$

Where,

$$k_1 = \frac{r_1}{r_4}$$

$$k_2 = \frac{r_1}{r_2}$$

$$k_3 = \frac{(r3^2 - r1^2 - r2^2 - r4^2)}{2r_2 r_4}$$

$$k_1 \cos32 + k_2 \cos254 + k_3 = \cos(32 - 254)$$

$$k_1 \cos47 + k_2 \cos278 + k_3 = \cos(47 - 278)$$

$$k_1 \cos62 + k_2 \cos303 + k_3 = \cos(62 - 303)$$

$$0.8480k_1 - 0.2756k_2 + k_3 = -0.7431$$

$$0.3820k_1 - 0.1392k_2 + k_3 = -0.6293$$

$$0.4695k_1 - 0.5446k_2 + k_3 = -0.4848$$

Equation (1) - (2)

$$0.166k_1 - 0.4148k_2 = -0.1138$$

$$K_1 - 2.4988k_2 = -0.6855$$

Equation (1) - (3)

$$0.3785k_1 - 0.8202k_2 = -0.2583$$

$$K_1 - 2.1670k_2 = -0.6824$$

Equation (5) - (7)

$$0 - 0.3318k_2 = -0.0031$$
$$K_2 = 0.0093$$

From (7)

$$K_1 = -0.6842 + 2.1670 \times 0.0093$$
$$K_1 = -0.6622$$

From (1)

$$K_3 = -0.7431 - 0.8480k_1 + 0.2756k_2$$
$$K_3 = -0.7431 - 0.8480 \times (-0.6622) + 0.2756 \times 0.0093$$
$$K_3 = -0.7431 + 0.5615 + 0.0026$$
$$K_3 = -0.179$$

Hence,

$$r_2 = \frac{r_1}{k_2}$$
$$r_2 = \frac{20}{0.0093}$$
$$r_2 = 2150.54$$
$$r_4 = \frac{r_1}{k_1}$$
$$r_4 = \frac{20}{-0.6622}$$
$$r_4 = -30.20$$
$$k_3 = \frac{(r3^2 - r1^2 - r2^2 - r4^2)}{2r_2\ r_4}$$
$$-0.179 = \frac{(r3^2 - r1^2 - r2^2 - r4^2)}{2r_2\ r_4}$$
$$-0.179 = \frac{(r3^2 - 20^2 - 2150.54^2 - 30.20^2)}{2 \times 2150.54 \times (-30.20)}$$
$$23250.778 = r3^2 - 4626134.3$$
$$r3^2 = 4649385.1$$
$$r_3 = 2156.24''$$

Since the length of r_2 and r3 is coming very large we can't mount this mechanism on the tractor. In the other words it is not possible to reduce the range of operation of crank from 32° to 193°.

However we can verify the proposed mechanism by using Frendenstein equation for operating range of crank equal to 161°.

Hence the new values of θ2 & θ4 will be as

Position	θ2	θ4
1	32°	180+74=254°
2	114°	180+98=278°
3	193°	180+123=303°

Hence,

$$k_1 \cos 32 + k_2 \cos 254 + k_3 = \cos(32 - 254)$$
$$k_1 \cos 117 + k_2 \cos 278 + k_3 = \cos(114 - 278)$$
$$k_1 \cos 193 + k_2 \cos 303 + k_3 = \cos(193 - 303)$$
$$0.8480k_1 - 0.2756k_2 + k_3 = -0.7431$$
$$-0.4067k_1 - 0.139k_2 + k_3 = -09613$$
$$0.9744k_1 - 0.5446k_2 + k_3 = -0.3420$$

Equation (1) - (2)

$$1.257k_1 - 0.4148k_2 = -0.2128$$
$$K_1 - 0.3306k_2 = -0.1739$$

Equation (1) - (3)

$$1.822k_1 - 0.8202k_2 = -0.4011$$
$$K_1 - 0.4501k_2 = -0.2201$$

Equation (5) - (7)

$$0\ 0.1195k_2 = 0.394$$
$$K_2 = 3.2971$$

From (7)

$$K_1 = -0.2201 + 0.4501k_2$$
$$K_1 = -0.2201 + 0.4501 \times 3.2971$$
$$K_1 = -0.2201 + 1.4840$$
$$K_1 = 1.2639$$

From (1)

$$K_3 = -0.7431 + 0.2756k_2 - 0.8480k_1$$
$$K_3 = -0.7431 + 0.2756 \times 3.297 - 0.8480 \times 1.2639$$
$$K_3 = -0.9062$$

Hence,

$$r_2 = \frac{r_1}{k_2}$$
$$r_2 = \frac{20}{3.2971}$$
$$r_2 = 6.0659$$
$$r_4 = \frac{r_1}{k_1}$$
$$r_4 = \frac{20}{1.2639}$$
$$r_4 = 15.8240$$
$$k_3 = \frac{(r3^2 - r1^2 - r2^2 - r4^2)}{2r_2\, r_4}$$
$$-0.9062 = \frac{(r3^2 - r1^2 - r2^2 - r4^2)}{2r_2\, r_4}$$
$$-0.179 = \frac{(r3^2 - 20^2 - 6.0659^2 - 15.8240^2)}{2 \times 6.0659 \times 15.8240}$$
$$-173.9665 = r3^2 - 687.1941$$

$$r3^2 = 513.2276$$
$$r_3 = 22.6545''$$

The results obtained are as under

$r_1 = 20''$
$r_2 = 6.07''$
$r_3 = 22.65''$
$r_4 = 15.82''$

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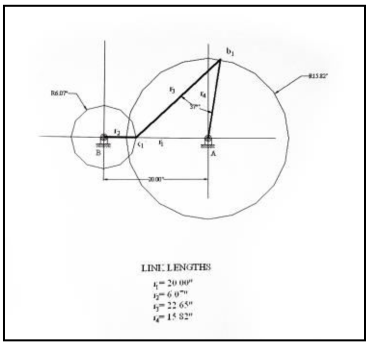

Fig. 13

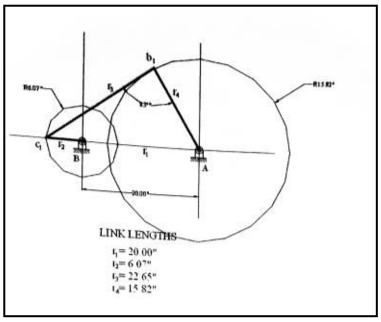

Fig. 14

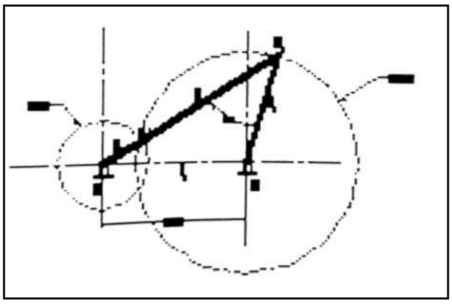

Fig. 15

The mechanism for the result obtained and plotted in Fig. 13, Fig. 14, Fig. 15 above & it satisfies the input output angle co-ordinations desired. However we obtained link length different. The mechanism is almost in toggle position during the working positions & hence gives M.A. as infinite however the γmin is 40° and γmax is 82°.

G. Comparison between existing mechanism and proposed mechanism

Particulars of Mechanism	r_1	r_2	r_3	$^{r_4}/_{r_4}$ extended	γ(Trans. Angley in Lower positio	γin Upper position	Torque (T_2) needed in lower position in K-Nm	Torque (T_2) needed in upper position in K-Nm	Mechanical Advantage in Lower position	Mechanical Advantage in Upper position
Existing	20’’	12’’	21.5’’	14’’/28’’	49°	95°	69		1.24	
Final Design 1	20’’	5.5 "	13’’	16’’/28’’	79°	123°	13.20		6.75	
Final Design 2	20’’	8’’	13’’	16’’/36’’	98°	134°	54.34		2.12	
Final Design 3	20’’	10’’	13’’	16’’/36’’	107°	145°	75.81		1.54	

Crank angle in upper most position. (in deg.)	Crank angle in lowermost position (in deg.)	Range of crank (in deg.)	Rocker angle in lowermost position (in deg.)	Rocker angle in uppermost position (in deg.)	Range of rocker (in deg.)	βupper (in deg.)	βlower (in deg.)
105	51	54	-19	28	9	100	45
178	67	111	-10	23	13	146	25
141	94	47	-8	18	10	127	69
132	97	35	-8	18	10	128	82

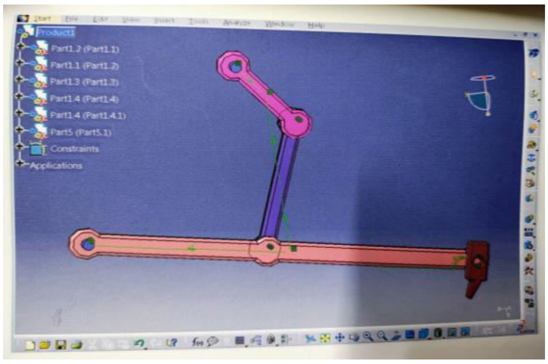

Fig. 16 CATIA Modelling

Fig. 17 Mechanism Installed into Tractor

IV. CONCLUSION

We tried to design a mechanism to satisfy the desired constraints by using various methods such as Brodell and Soni's equations for the desired maximum and minimum transmission angle. However, since Brodell and Soni's equations are meant for a time ratio of one, hence this approach is suitable for such mechanism only.

The range of operation of crank in such mechanism is 180° which is not suitable for the mechanism operated by hydraulic actuators. Hence, we decided to manipulate the link-lengths with an intension to satisfy all the constraints and also to reduce the operating range of crank to 30°.

In all we tried for three different mechanisms and reduced the operating range of crank to 35° .The designed mechanism satisfies all the set constraints also. Hence this mechanism can be safely used on tractors.

While designing the mechanism we have to scarify for the torque requirement to some extent to reduce the operating range of the crank.

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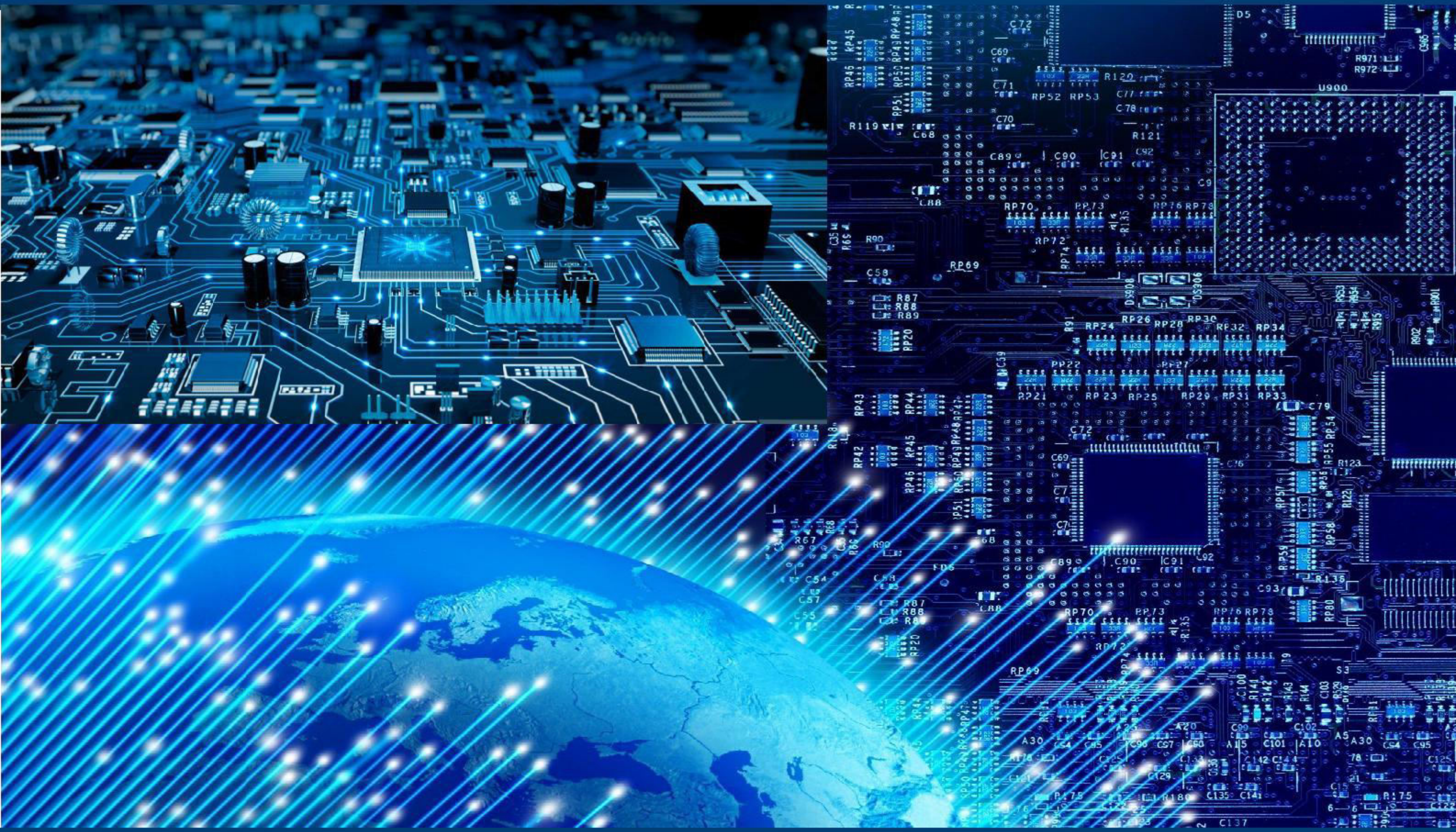

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