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DESIGN OF FOUR BAR CHAINLESS BICYCLE -ANTHROPROMETRIC AND KINEMATIC ANALYSIS

K. Taranikanth[*1] & G. Srinivasa Tarun[2]
[*1]Assistant Professor, Department of Mechanical Engineering, VIIT, Duvvada, Visakhapatnam, India
[2]Undergraduate Student Department of Mechanical Engineering, GITAM University, Visakhapatnam, India

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ABSTRACT

This paper is developed for the users to rotate the back wheel of a two wheeler by replacing traditional sprocket wheel mechanism with new planar four-bar linkages in the transmission system of bicycle, based on the lever. Usually in two wheelers, chain and sprocket method is used to drive the back wheel. This concept is explored for improving of the driving technique of bicycle. The Primary advantages are considerably improved driving efficiency, simplified drive wheel change out, a simple frame design and zero maintenance on any drive component. The improved driving efficiency is mainly due to the replacement of the drive chain, with a Four bar mechanism. Added advantages of this new mechanism is that all components are fully enclosed and thus don't have the potential of staining and jamming clothes as it is the case with conventional, exposed, grease lubricated chain drives. Due to its small motion range, the new mechanism can be applied in both the traditional bicycle, special for the handicapped and in the design of the vehicles.

KEYWORDS: Anthropometric Analysis, Chainless bicycle, Kinematic Design

1. INTRODUCTION

This bicycle is reformed by using traditional bicycle as the base, and adopting flat fourbar linkage to replace chain driven mechanism. Planar four-bar linkages are constructed from four links connected in a loop by four one degree of freedom joints.

With development of industrial technique, E-bicycles and magnetic powered bicycles are emerged, but these products are still powered by energy. Human power drive bicycles are still well received. Therefore, improving its working transformation efficiency would be the key. The limited natural resources makes the energy issue become the hot topic throughout the world. In additional to the efficiency gain, the efficiency of the human leg joint is also improved. Due to the fact that the arc motion of the leg joints requires only one quarter of the horizontal movement for a given vertical power stroke, as compared to the circular motion of the conventional bicycle crank.

2. LITERATURE REVIEW

The Dandy horse, also called Draisienne or Laufmaschine, was the first human means of transport to use only two wheels in tandem and was invented by the German Baron Karl von Drais. It is regarded as the modern bicycle's forerunner. Drais introduced it to the public in Mannheim in summer 1817 and in Paris in 1818. Its rider sat astride a wooden frame supported by two in-line wheels and pushed the vehicle along with his or her feet while steering the front wheel.

Pierre Lallement took bicycle design in a new direction by adding a mechanical crank drive with pedals on an enlarged front wheel. Another French inventor named Douglas Grasso had a failed prototype of Pierre Lallement's bicycle several years earlier. Several inventions followed using rear-wheel drive, the best known being the rod driven velocipede by Scotsman Thomas McCall in 1869. In that same year, bicycle wheels with wire spokes were patented by Eugene Meyer of Paris. Further innovations increased comfort and ushered in a second bicycle craze. Scotsman John Boyd Dunlop introduced the first practical pneumatic tire, which soon became universal. Soon after, the rear freewheel was developed, enabling the rider to coast. Derailleur gears and hand-operated Bowden cable-pull brakes were also developed during these years, but were only slowly adopted by casual riders.

By the turn of the century, cycling clubs flourished on both sides of the Atlantic, and touring and racing became widely popular. The bicycle has undergone continual adaptation and improvement since its inception. These innovations have continued with the advent of modern materials and computer-aided design, allowing for a proliferation of specialized bicycle types.

3. PROPOSED MODEL OF CHAINLESS BICYCLE

The Grashof's condition for a four-bar linkage states: If the sum of the shortest and longest link of a planar quadrilateral linkage is less than or equal to the sum of the remaining two links, then the shortest link can rotate fully with respect to a neighbouring link. In other words, the condition is satisfied if $S+L \leq P+Q$ where S is the shortest link, L is the longest, and P and Q are the other link

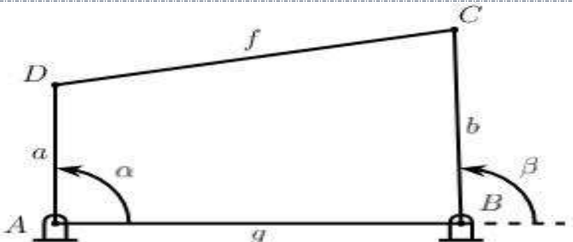
The movement of a quadrilateral linkage can be classified into eight cases based on the dimensions of its four links. Let a, b, g and h denote the lengths of the input crank, the output crank, the ground link and floating link, respectively. Then, we can construct the three terms:

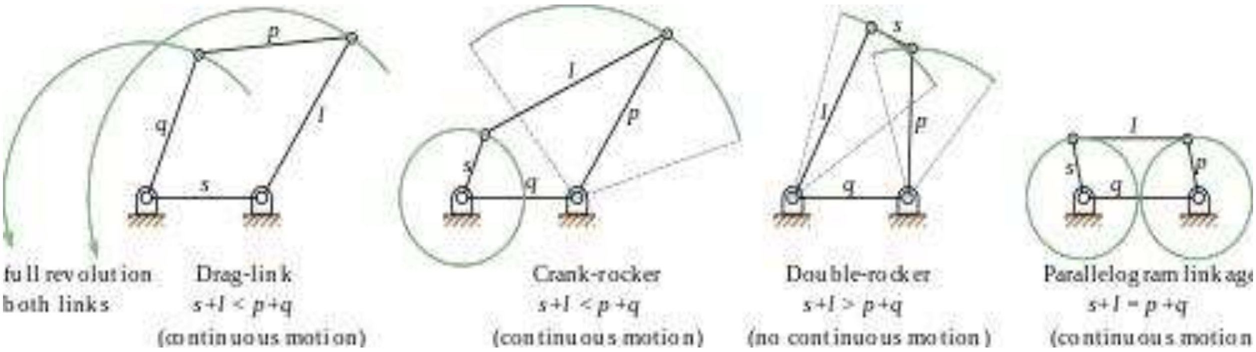
The movement of a quadrilateral linkage can be classified into different types based on the positive and negative values for these three terms, T1, T2, and T3.
$T1=g+f-a-b$, $T2=b+g-a-f$, $T3= b+f-a-g$
A planar four-bar linkage consists of four rigid rods in the plane connected by pin joints. We call the rods/ links:
- **Ground link 'g':** Fixed to anchor pivots A and B.
- **Input link 'b':** Driven by a lever with input angle
- **Output link 'a':** Gives output angle
- **Floating link 'f':** Connects the two moving pins C and D.

Degree of Freedom = 1

Table: 01

T1	T2	T3	Grashof's Condition	Input link	Output link
-	-	+	Grashof	Crank	Crank
+	+	+	Grashof	Crank	Rocker
+	-	-	Grashof	Rocker	Crank
-	+	-	Grashof	Rocker	Rocker





- The entire mechanism is designed using Grashof's law.
- Here the output crank is driven by its neighbouring oscillating arm.
- This arm is connected with lever mechanism with pedal on the other end and hinged at a point to transmit required force.
- When the force is applied on the pedal, the rocker arm starts oscillating which further helps in rotating the wheel.

4. CUSTOMIZATION

In 2016, we truly are on the verge of seeing mass customization arise and provide a viable alternative to the process of homogenized mass production that has been so prevalent. At the same time, technology itself has become more advanced, allowing business to build sophisticated, yet easy-to-use configurations.

Here, he handle and seat used here are customizable depending on the height of the person to make it more personal and comfortable while riding the bicycle.

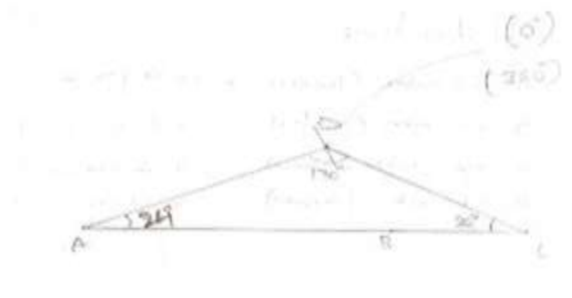
The lower point of handle is hinged, so that it oscillate with respect to Y- axis. This helps in adjusting the handle to possible comfortable position of the rider. Also, the handle can move in X-axis i.e., in horizontal direction to make it more customizable and to provide lot more comfort in riding the bicycle when compared to conventional bicycles

5. ANALYSIS AND CALCULATION

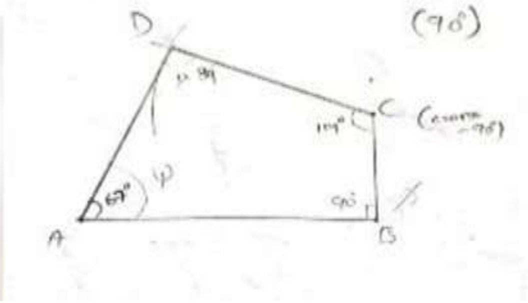
KINEMATIC ANALYSIS:

VALUES OF FOUR BAR LINKS:

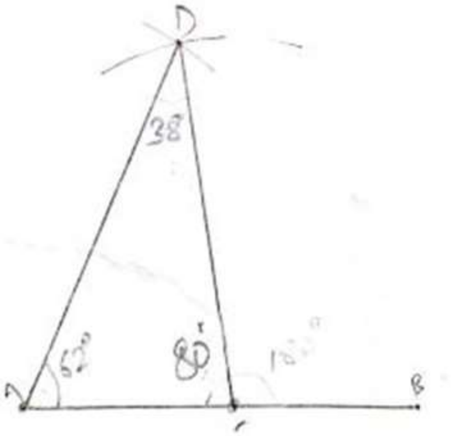
- ROCKER =13.97 cm
- COUPLER =12.7 cm
- CRANK = 7.62 cm
- GROUND =16.51 cm



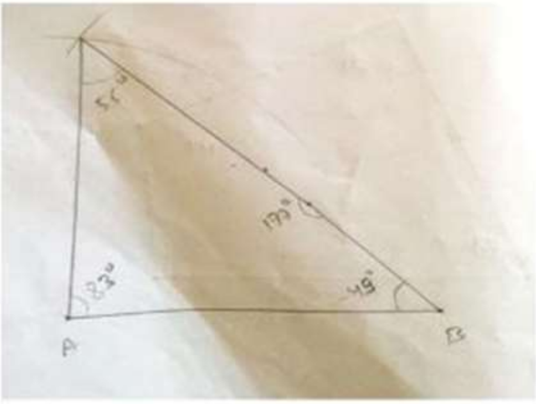
0 (or) 360°



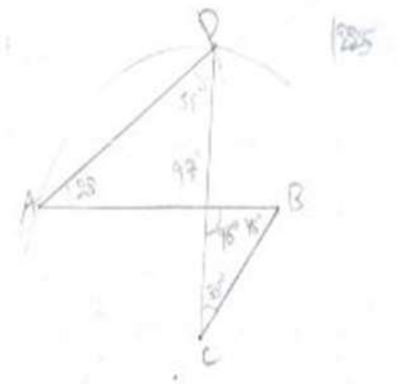
90°



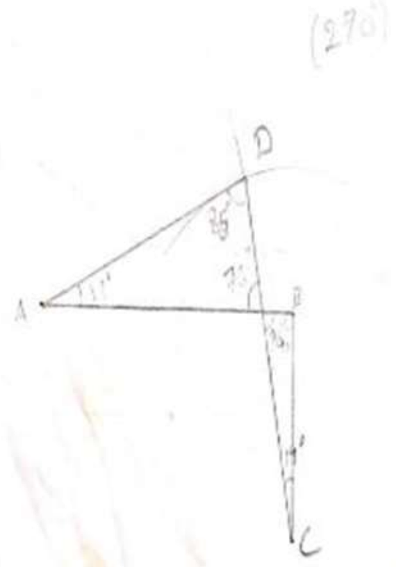
180°



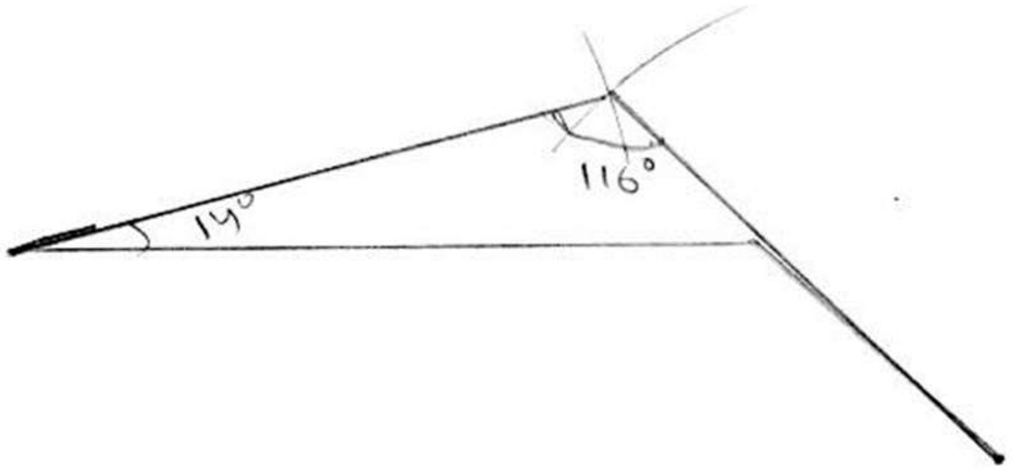
135°



225^0



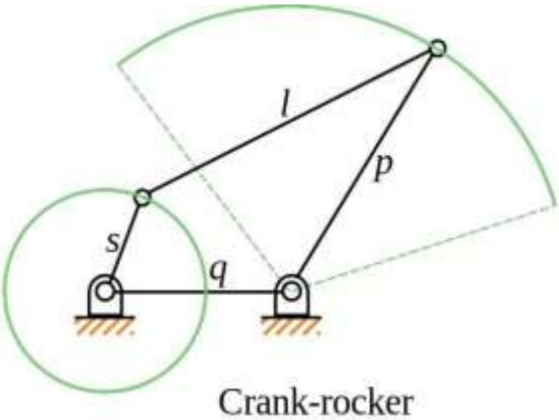
275^0



315^0

6. ROCKER CRANK VERIFICATION

T1= g+f-a-b (q+l-p-s)
T2= b+g-a-f (s+q-p-l)
T3= b+f-a-g (s+l-p-q)



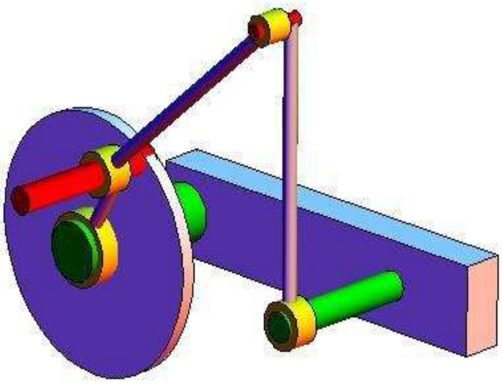
Crank-rocker
$s+l < p+q$
(continuous motion)

a (rocker) =13.97cm
b (crank) =7.62cm
g (ground) =16.57cm
f (coupler) =12.7cm

T1=16.57+12.7-13.97-7.62
T1=7.62 (+) (verified)

T2=7.62+16.57-13.97-12.7
T2=-2.54 (verified)

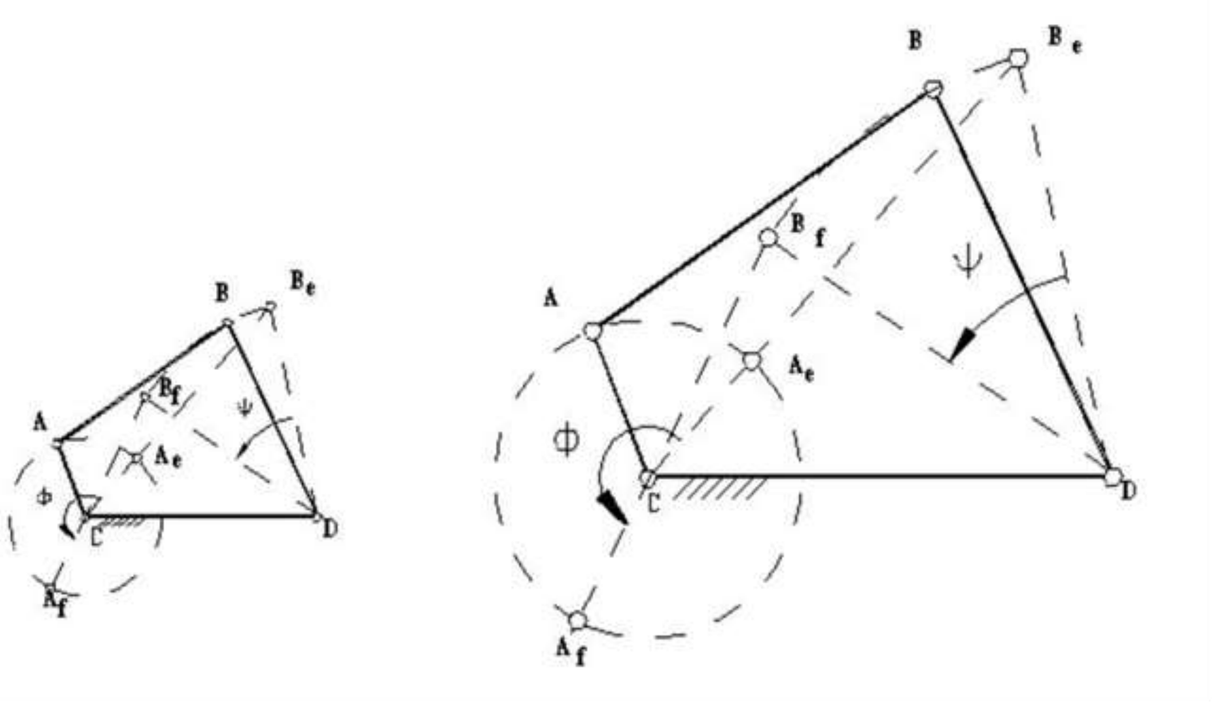
T3=7.62+12.7+-13.97-16.51
T3=-10.16 (verified)



It is rather important to understand how the mechanism will function under loaded conditions in practice while the kinematic characteristics of the mechanism is being considered. By the performance of the mechanism we mean the effective transmission of motion (and force) from the input link to the output link.

This also means that for a constant torque input, in a well performing mechanism we must obtain the maximum torque output that is possible and the bearing forces must be a minimum. Of course, torque and force are not the quantities that has been in the kinematics and whatever kinematic quantity we use to define the performance of the mechanism, this quantity will only approximate the static force characteristics of the mechanism.

The dynamic characteristic, which is a function of mass and moment of inertia of the rigid bodies, may be several times more than the static forces and the behaviour of the mechanism under the dynamic forces cannot be predicted by kinematics. Still, some rule-ofthumb of the behaviour of the mechanism under load is better than none.



$$\cos\beta=\frac{(a2+a3)^2+a1^2-a4^2}{2a1(a3+a2)}$$

$$\cos(\pi-\varphi 1)=\frac{a1^2+a4^2-(a2+a3)^2}{2a1a4}$$

$$\cos(\beta+\emptyset-\pi)=\frac{(a3-a2)^2+a1^2-a4^2}{2a1(a3-a2)}$$

$$\cos(\pi-\varphi 1-\varphi)=\frac{a1^2+a4^2-(a3-a2)^2}{2a1a4}$$

$$\cos\mu(\text{min or max})=\frac{a4^2+a3^2-a1^2-a2^2}{2a4a3}\pm\frac{a1a2}{a3a4}$$

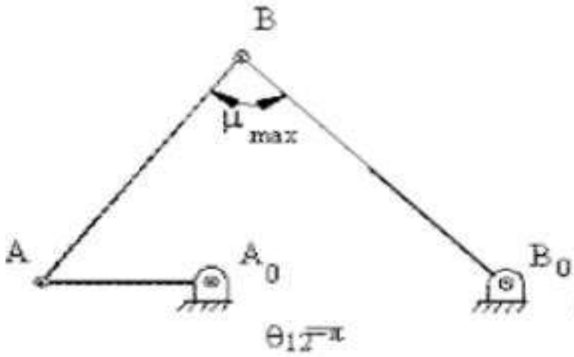
Clearly, the optimum value of the transmission angle is 90^0. Since the angle will be constantly changing during the motion cycle of the mechanism, there will be a position at which the transmission angle will deviate most from 90^0. In practice it has been found out that if the maximum deviation of the transmission angle from 90^0 exceeds 40^0 or 50^0 (depending on the type of application), the mechanism will lock.

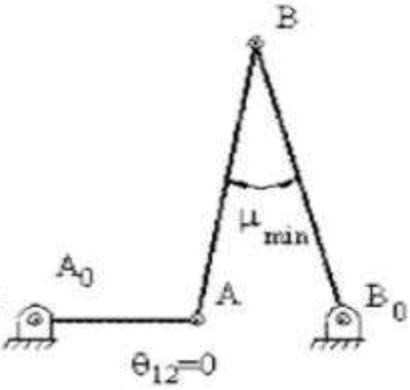
In certain cases this maximum deviation must be kept within 20^0 (e.g. reciprocating pumps) and in certain other applications maximum deviations of up to 70^0 may be permissible (e.g. aircraft landing gears). One must consider the practical application of a mechanism in order to give a limit to this deviation (whenever in doubt, try to keep this deviation to less than 40^0 or 50^0).

$$\Delta_1=|90-\mu\text{min}|$$

$$\Delta_2=|90-\mu\text{max}|$$

$$\Delta_{\text{max}}=\text{max}(\Delta 1,\Delta 2)$$





The critical transmission angle is either min or max, whichever deviates most from 90^0. Sometimes, for the transmission angles greater than 90^0, instead of m (180^0-m) is used for the value of the transmission angle. In such a case, there are two minimum values of the transmission angle (mmin1= mmin, mmin2= 180^0-mmax) The most critical transmission angle is the minimum of mmin1 and mmin2. Note that the deviation of the transmission angle from 90^0 at the two extreme positions will be equal if such four-bar mechanisms are known as *centric four-bar*. In centric four-bar mechanisms the time ratio is unity (the crank rotation between dead-centres is 180^0) and they will have a better force transmission characteristics as compared with the other crank-rocker proportions.

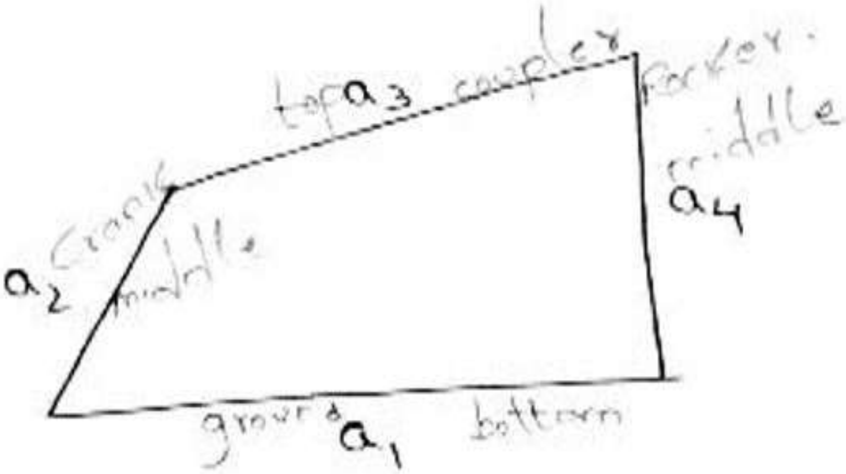
7. TIME RATIO

With four bar mechanisms there are two strokes, the forward and return, which when added together create a cycle. Each stroke may be identical or have different average speeds. The time ratio numerically defines how fast the forward stroke is compared to the quicker return stroke.

$$TR = \frac{\text{time its takes for forward stroke}}{\text{time its take for backward stroke}}$$

$$= \frac{\emptyset}{360^\circ - \emptyset}$$

Calculations



Swing angle = ψ
Crank Rotation = φ
Where,
A_1= 16.51cm
A_2= 7.62cm
A_3=12.7cm
A_4=13.97cm

$$\cos\beta = \frac{(a_2 + a_3)^2 + (a_1)^2 - (a_4)^2}{2 * a_1 * (a_3 + a_2)}$$

$$= \frac{(7.62 + 12.7)^2 + (16.51)^2 - (13.97)^2}{2 * 16.51 * (12.7 + 7.62)}$$

$$= \frac{489.99}{670.9664}$$

$$= 0.7289$$

$$\therefore \beta = 43.20°$$

$$\cos(\beta + \varphi - \pi) = 0.61538$$

$$(\beta + \varphi - \pi) = 52.02013$$

$$\varphi = 188.9401°$$

$$\cos(\pi - \psi 1 - \psi) \quad = \frac{(a_1)^2 + (a_4)^2 - (a_3 - a_2)^2}{2 * a1 * a4}$$

$$= \frac{(16.51)^2 + (13.97)^2 - (12.7 - 7.62)^2}{2 * 16.51 * 13.97}$$

$$= 0.9580$$

$$(\pi - \psi 1 - \psi) = 16.6561$$

From the above calculations we get

$$\psi = 66.5238°$$

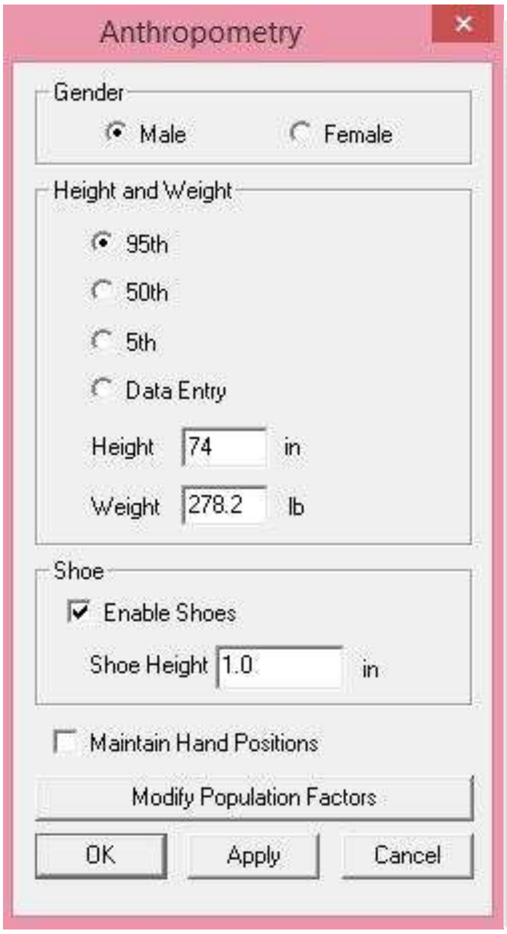
$$\cos\mu_{max}^{min} = \frac{(a_4)^2 + (a_3)^2 - (a_1)^2 - (a_2)^2}{2 * a_4 * a_3} \pm \frac{a_1 * a_2}{a_3 * a_4}$$

$$\therefore \mu_{min} = 38.58°$$

$$\mu_{max} = 50.48°$$

8. ANTHROPOMETRIC ANALYSIS

Anthropometry plays an important role in industrial design, clothing design, ergonomics and architecture where statistical data about the distribution of body dimensions in the population are used to optimise products. Changes in lifestyle, nutrition, and ethnic composition of populations lead to changes in the distribution of body dimensions (e.g. the rise in obesity), and required regular updating of anthropometric data collections.

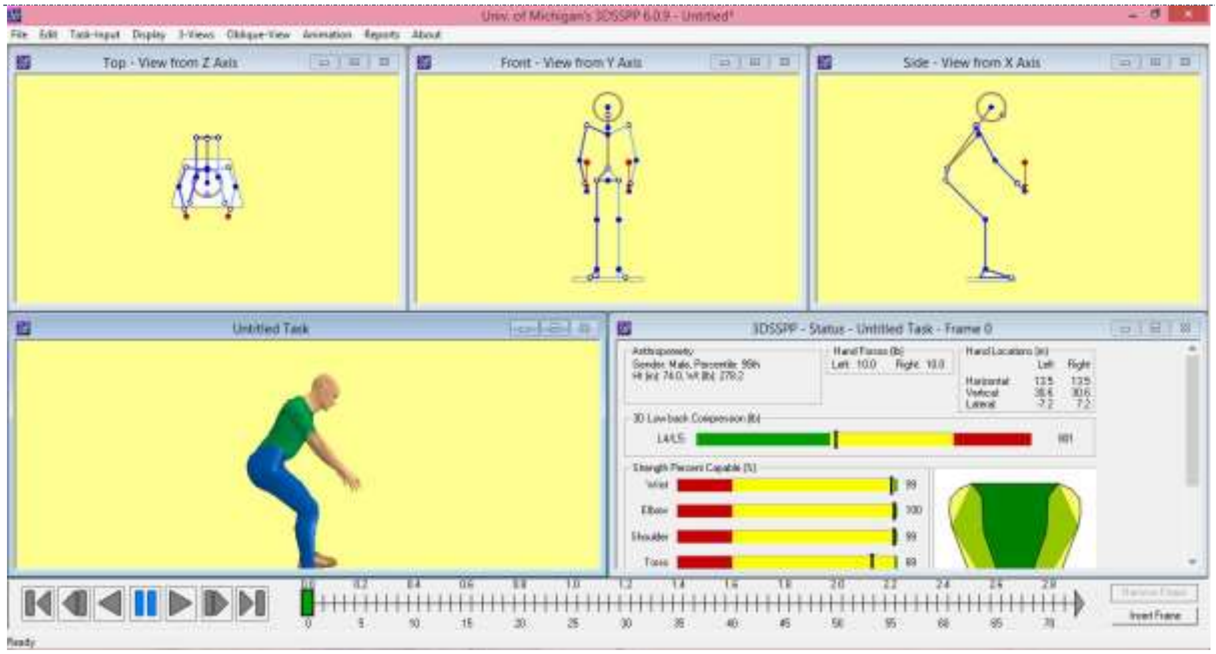


● Male ○ Female
● 95th
○ 50th
○ 5th
○ Data Entry
Height 74 in
Weight 278.2 lb
☑ Enable Shoes
Shoe Height 1.0 in
☐ Maintain Hand Positions

MALE ANTHROPOMETRY DATA

○ Male ● Female
● 95th
○ 50th
○ 5th
○ Data Entry
Height 68 in
Weight 253.5 lb
☑ Enable Shoes
Shoe Height 1.0 in
☐ Maintain Hand Positions

FEMALE ANTHROPOMETRY DATA



Anthropometry
Gender: Male, Percentile: 95th
Ht (in): 74.0, Wt (lb): 278.2

Hand Forces (lb)
Left: 10.0 Right: 10.0

Hand Locations (in)

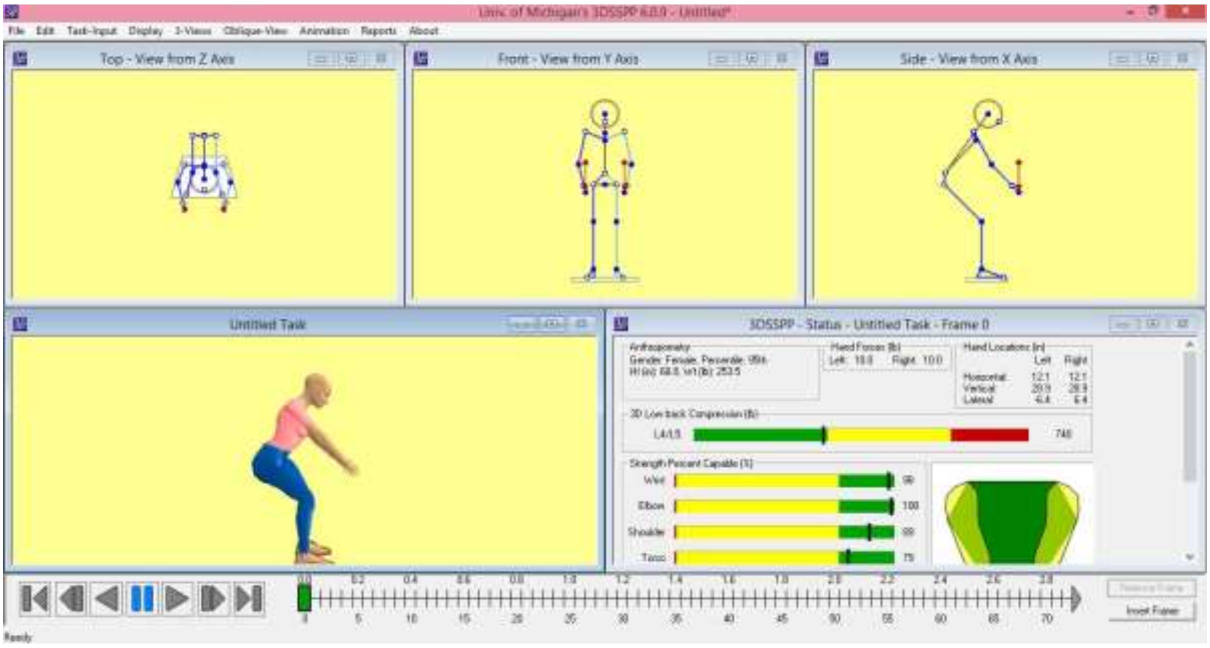
	Left	Right
Horizontal	13.5	13.5
Vertical	30.6	30.6
Lateral	-7.2	7.2

3D Low back Compression (lb)
L4/L5: 801

Strength Percent Capable (%)

Wrist	99
Elbow	100
Shoulder	99
Torso	89

MALE ANTHROPOMETRY ANALYSIS 95th PERCENTILE



Anthropometry
Gender: Female, Percentile: 95th
Ht (in): 68.8, Wt (lb): 253.5

Hand Forces (lb)
Left: 10.0 Right: 10.0

Hand Locations (in)

	Left	Right
Horizontal	12.1	12.1
Vertical	28.9	28.9
Lateral	-6.4	6.4

3D Low back Compression (lb)
L4/L5: 748

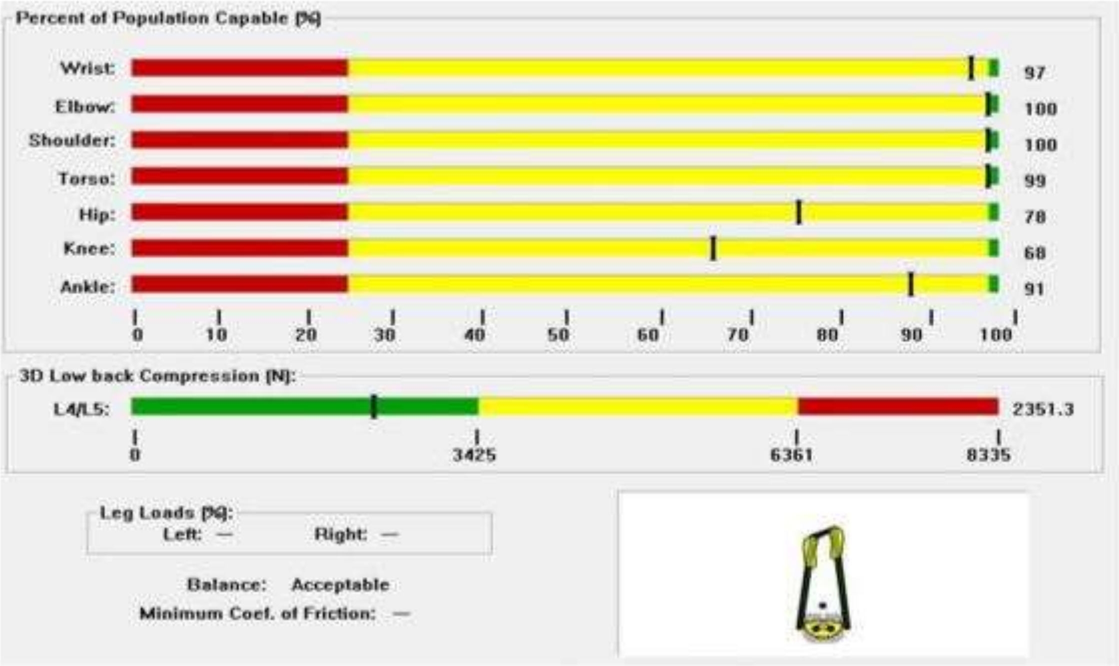
Strength Percent Capable (%)

Wrist	99
Elbow	100
Shoulder	99
Torso	75

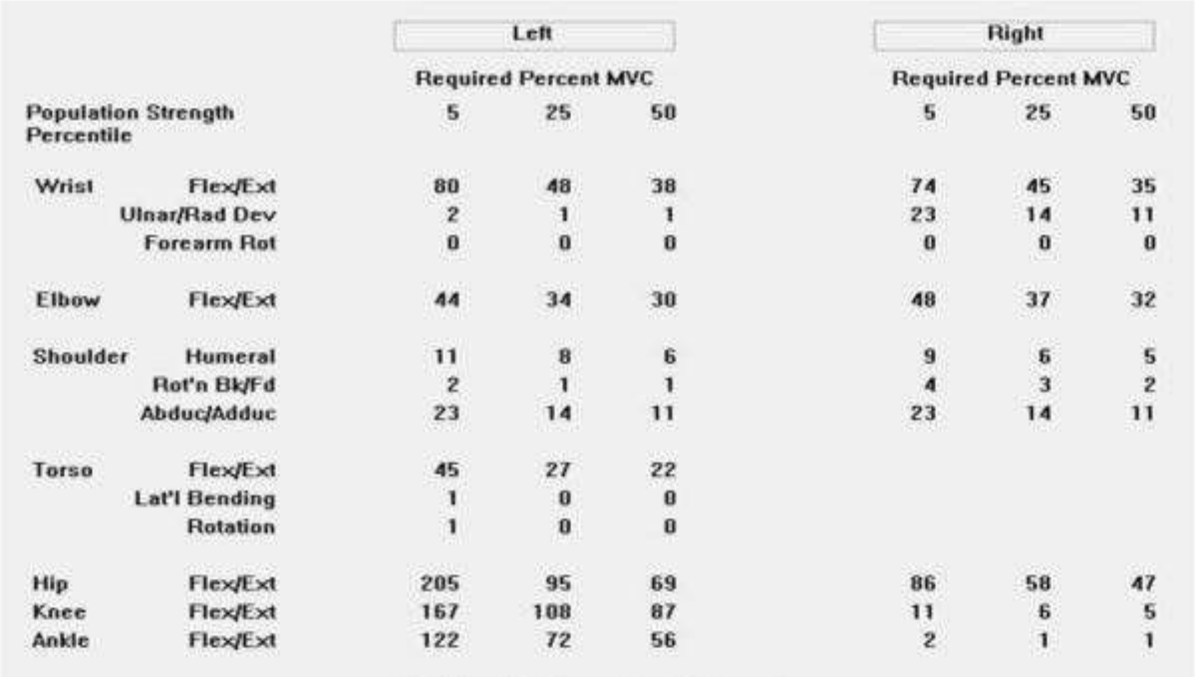
FEMALE ANTHROPOMETRY ANALYSIS 95th PERCENTILE

Analysis on male for 95th percentile:

MALE ANALYSIS SUMMARY ON 95th PERCENTILE:

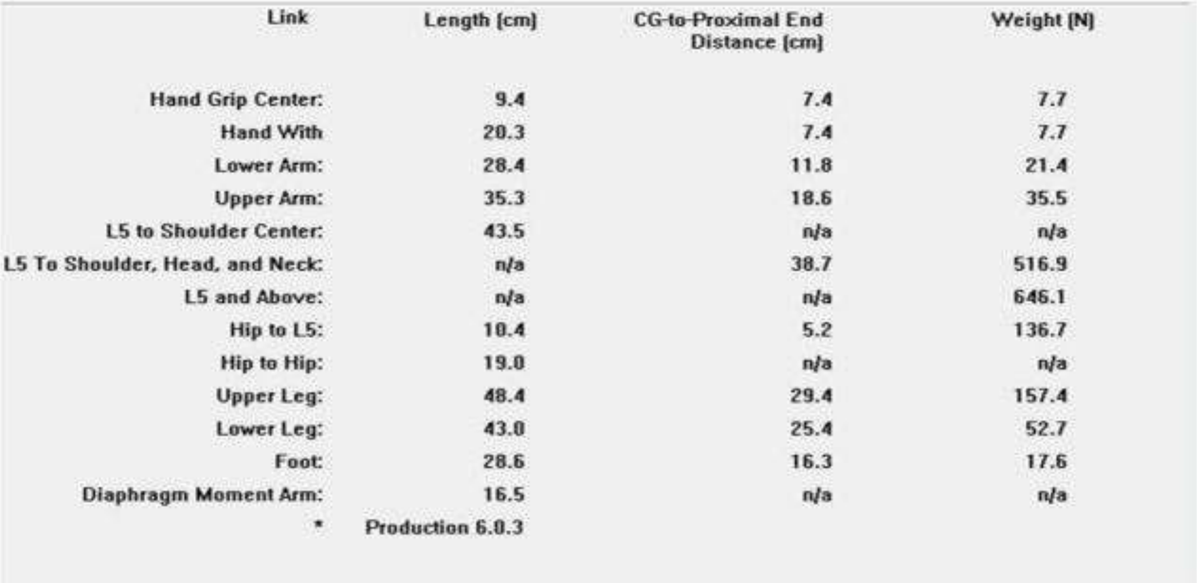


MALE FATIGUE ANALYSIS ON 95th PERCENTILE:



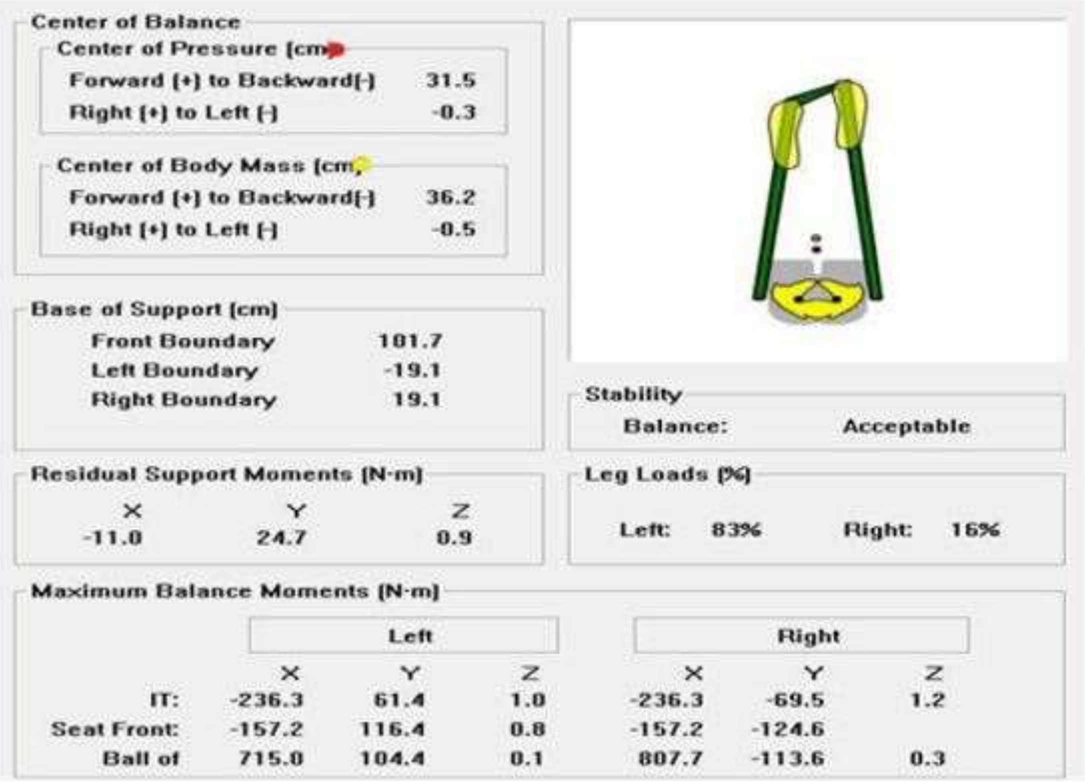
		Left			Right		
		Required Percent MVC			Required Percent MVC		
Population Strength Percentile		5	25	50	5	25	50
Wrist	Flex/Ext	80	48	38	74	45	35
	Ulnar/Rad Dev	2	1	1	23	14	11
	Forearm Rot	0	0	0	0	0	0
Elbow	Flex/Ext	44	34	30	48	37	32
Shoulder	Humeral	11	8	6	9	6	5
	Rot'n Bk/Fd	2	1	1	4	3	2
	Abduc/Adduc	23	14	11	23	14	11
Torso	Flex/Ext	45	27	22			
	Lat'l Bending	1	0	0			
	Rotation	1	0	0			
Hip	Flex/Ext	205	95	69	86	58	47
Knee	Flex/Ext	167	108	87	11	6	5
Ankle	Flex/Ext	122	72	56	2	1	1

MALE ANTHROPOMETRY ANALYSIS ON 95thPERCENTILE:



Link	Length (cm)	CG-to-Proximal End Distance (cm)	Weight (N)
Hand Grip Center:	9.4	7.4	7.7
Hand With	20.3	7.4	7.7
Lower Arm:	28.4	11.8	21.4
Upper Arm:	35.3	18.6	35.5
L5 to Shoulder Center:	43.5	n/a	n/a
L5 To Shoulder, Head, and Neck:	n/a	38.7	516.9
L5 and Above:	n/a	n/a	646.1
Hip to L5:	10.4	5.2	136.7
Hip to Hip:	19.0	n/a	n/a
Upper Leg:	48.4	29.4	157.4
Lower Leg:	43.0	25.4	52.7
Foot:	28.6	16.3	17.6
Diaphragm Moment Arm:	16.5	n/a	n/a
*	Production 6.0.3		

MALE BALANCE REPORT ON 95thPERCENTILE:



Center of Balance

Center of Pressure (cm)

Forward (+) to Backward(-)	31.5
Right (+) to Left (-)	-0.3

Center of Body Mass (cm)

Forward (+) to Backward(-)	36.2
Right (+) to Left (-)	-0.5

Base of Support (cm)

Front Boundary	101.7
Left Boundary	-19.1
Right Boundary	19.1

Stability

Balance: Acceptable

Residual Support Moments (N-m)

X	Y	Z
-11.0	24.7	0.9

Leg Loads (%)

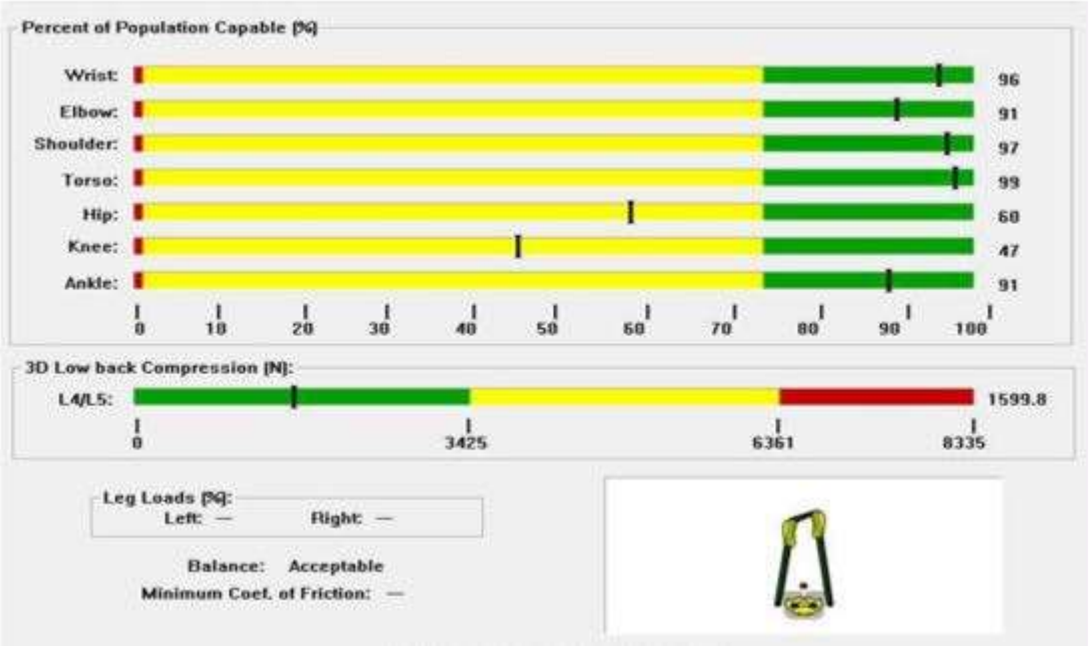
Left: 83% Right: 16%

Maximum Balance Moments (N-m)

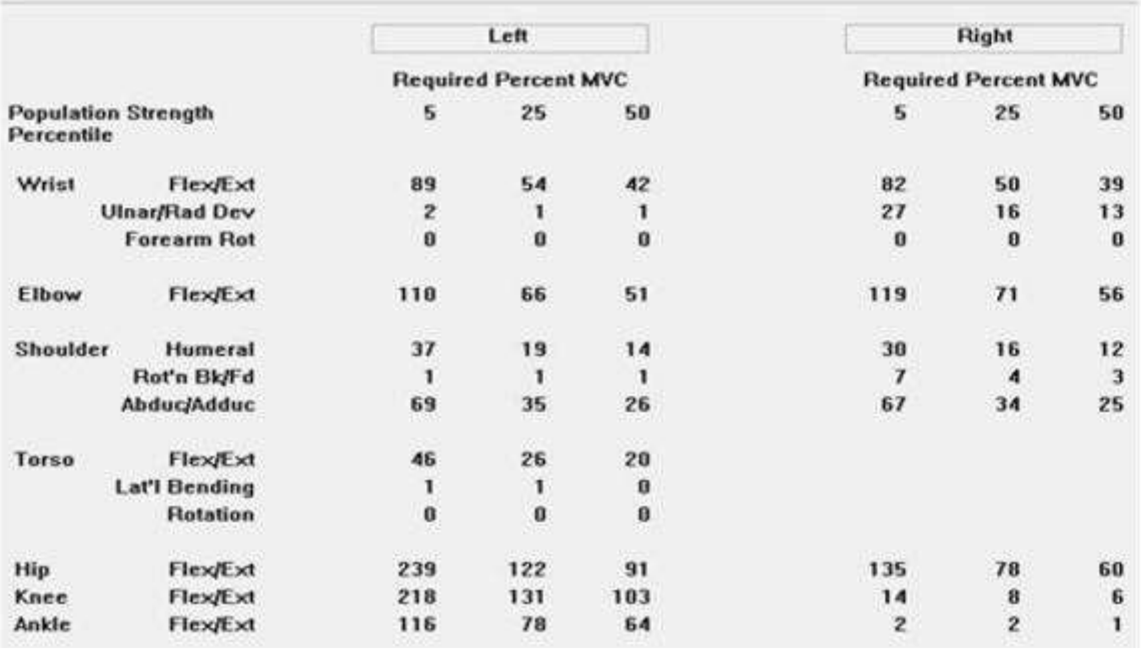
	Left			Right		
	X	Y	Z	X	Y	Z
IT:	-236.3	61.4	1.0	-236.3	-69.5	1.2
Seat Front:	-157.2	116.4	0.8	-157.2	-124.6	
Ball of	715.0	104.4	0.1	807.7	-113.6	0.3

Analysis on female for 95th percentile:

FEMALE ANALYSIS SUMMARY ON 95thPERCENTILE:

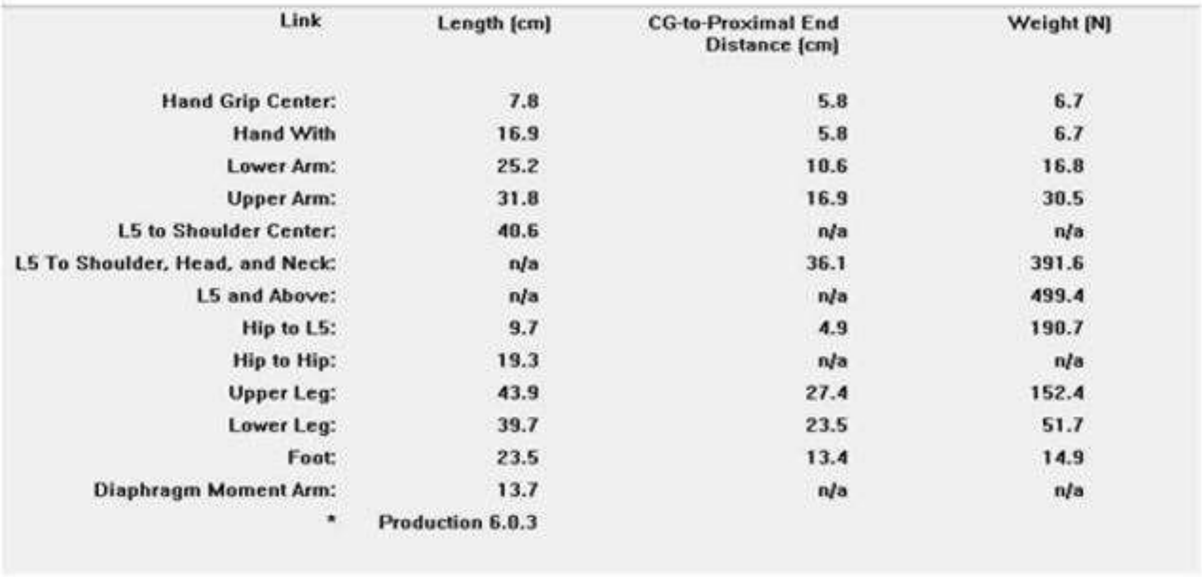


FEMALE FATIGUE ANALYSIS ON 95th PERCENTILE:



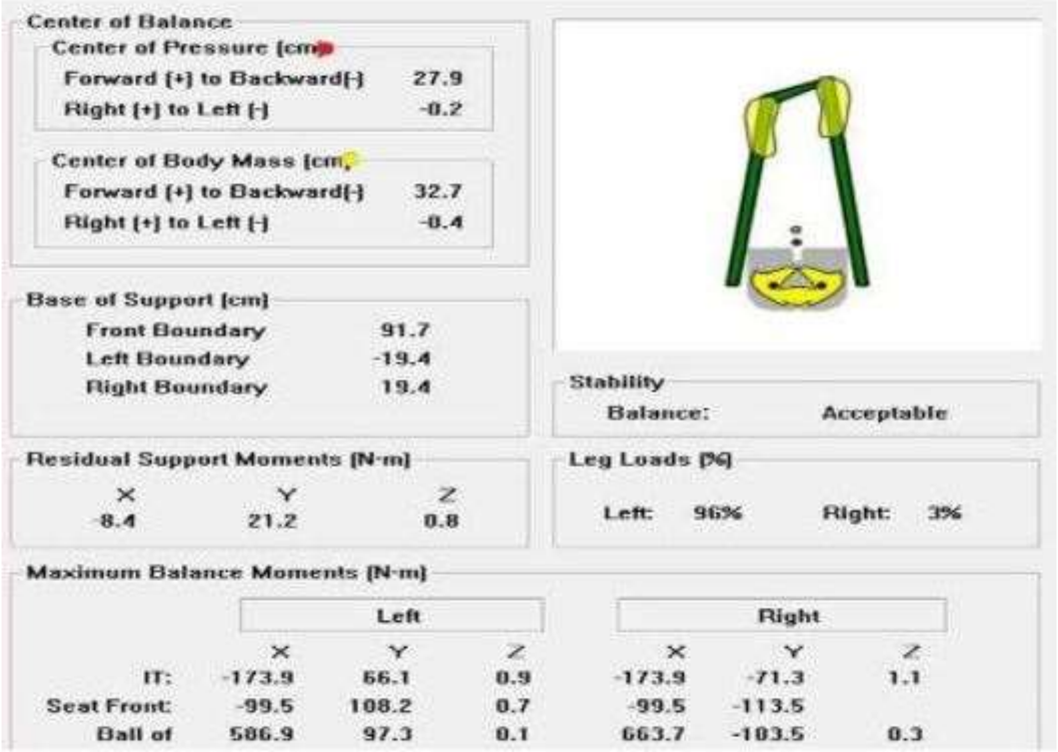
		Left			Right		
		Required Percent MVC			Required Percent MVC		
Population Strength Percentile		5	25	50	5	25	50
Wrist	Flex/Ext	89	54	42	82	50	39
	Ulnar/Rad Dev	2	1	1	27	16	13
	Forearm Rot	0	0	0	0	0	0
Elbow	Flex/Ext	110	66	51	119	71	56
Shoulder	Humeral	37	19	14	30	16	12
	Rot'n Bk/Fd	1	1	1	7	4	3
	Abduc/Adduc	69	35	26	67	34	25
Torso	Flex/Ext	46	26	20			
	Lat'l Bending	1	1	0			
	Rotation	0	0	0			
Hip	Flex/Ext	239	122	91	135	78	60
Knee	Flex/Ext	218	131	103	14	8	6
Ankle	Flex/Ext	116	78	64	2	2	1

FEMALE ANTHROPOMETRY ANALYSIS ON 95th PERCENTILE:



Link	Length (cm)	CG-to-Proximal End Distance (cm)	Weight (N)
Hand Grip Center:	7.8	5.8	6.7
Hand With	16.9	5.8	6.7
Lower Arm:	25.2	10.6	16.8
Upper Arm:	31.8	16.9	30.5
L5 to Shoulder Center:	40.6	n/a	n/a
L5 To Shoulder, Head, and Neck:	n/a	36.1	391.6
L5 and Above:	n/a	n/a	499.4
Hip to L5:	9.7	4.9	190.7
Hip to Hip:	19.3	n/a	n/a
Upper Leg:	43.9	27.4	152.4
Lower Leg:	39.7	23.5	51.7
Foot:	23.5	13.4	14.9
Diaphragm Moment Arm:	13.7	n/a	n/a
*	Production 6.0.3		

FEMALE BALANCE ANALYSIS ON 95thPERCENTILE:



Center of Balance

Center of Pressure (cm)

Forward (+) to Backward(-)	27.9
Right (+) to Left (-)	-0.2

Center of Body Mass (cm)

Forward (+) to Backward(-)	32.7
Right (+) to Left (-)	-0.4

Base of Support (cm)

Front Boundary	91.7
Left Boundary	-19.4
Right Boundary	19.4

Stability

Balance: Acceptable

Residual Support Moments (N-m)

X	Y	Z
-8.4	21.2	0.8

Leg Loads (%)

Left: 96% Right: 3%

Maximum Balance Moments (N-m)

	Left			Right		
	X	Y	Z	X	Y	Z
IT:	-173.9	66.1	0.9	-173.9	-71.3	1.1
Seat Front:	-99.5	108.2	0.7	-99.5	-113.5	
Ball of	586.9	97.3	0.1	663.7	-103.5	0.3

9. CONCLUSION

Based on the Grashof's law and its verifications, it is proven that the designed lengths and shapes of the four bar mechanism links are satisfied and are optimal. This design eliminates the more number of moving parts there by reducing friction. The transmission system introduces new oscillating pedals instead of rotating pedals which increases the overall run time of bicycle by almost two times the normal run time. New design also improves the overall riding comfort and driving efficiency of bicycle. In many ways, this new chainless transmission system is superior to the traditional design.

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