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Zhan Jiaying^a, Mohammad Fard^a & Reza Jazar^a

^a School of Aerospace, Mechanical & Manufacturing Engineering, RMIT University, Melbourne, Australia

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NOTES

A Model to Assess the Comfort of Automotive Seat Cushions

Zhan Jiaxing
 Mohammad Fard
 Reza Jazar

School of Aerospace, Mechanical & Manufacturing Engineering, RMIT University, Melbourne, Australia

A large number of independent and interacting factors affect seating comfort such as seat shape, stability, lumbar support and seat height. Although many subjective comfort studies have been conducted, few of them considered seating comfort from its subassembly level. This paper analyzed the automotive seat cushion designed with geared four-bar linkage for the seat height adjustment. The operation torque and lift distance of this mechanism was investigated as 2 major comfort factors. Ten cushions with this kind of design in the market were compared and assessed.

seat cushion geared four-bar linkage comfort

Notations

1	seat track
2	support link
3	bracket
4	geared link
5	driving pinion
6	sector of geared link
$\theta_1, \theta_2,$	
$\theta_3, \theta_4, \theta_5$	horizontal angles of parts with ground
a_1	angle from geared link to down stop position
a_2	angle from design position to down stop position
a_3	angle from design position to up stop position
a_4	adjustable angle of linkage (min = $-a_2$, max = a_3)
S	load point
F	occupant load on cushion
M_c	lift torque on sector of geared link
T	operation torque
Z_T	gear number of driving pinion
Z_{M_c}	gear number of sector

L_1	length of seat track
L_2	length of support link
L_3	length of bracket
L_4	length of geared link
L_5	distance from point S to point C
L_{CSx}	length of the distance from point C to point S (along axis x)

1. INTRODUCTION

A seat is one of the most important components of a vehicle, where drivers spend most of their time [1]. A well-designed automotive seat not only can bring comfortable driving experience, but also keep the driver in a safe driving position [2]. Although the driving comfort and driving safety are determined by many design parameters, a height adjustable seat will bring a lot of benefits to drivers. According to Reed, added height of seat helps to achieve good eye position [3]. This ensures a clear view of the instrument and good all round vision. Higgins concluded that the higher you ride in the car, in some cases, the

Correspondence should be sent to Zhan Jiaxing, School of Aerospace, Mechanical & Manufacturing Engineering, RMIT University, Melbourne 3001, Australia. E-mail: raulzjx@hotmail.com.

more comfortable you feel behind the wheel [4]. Ajayeoba and Adekoya presented in their research the importance of seat height for the ergonomic suitability of the passenger [5].

To design and produce a height adjustable seat structure, seat and car manufacturers did a lot of work to distinguish their products from those of their competitors. Publications based on their design are reviewed in this study. Major efforts can be categorized into five groups: (a) safety, (b) driving comfort, (c) multifunctions, (d) improved operation ability and (e) simple construction and minimum space requirement.

Matsuhashi invented a protection structure when a rearward load is input into the seat back; the structure could suppress a downward displacement during rear collision [6]. Yokota designed a seat height adjustment mechanism [7]. The advantage of this mechanism is to prevent the height adjustment mechanism against deformation in the case of rear-end collision, without being influenced by the degree of the backward inertia. Schumann, Teufel and Mühlberger [8] and Mühlberger and Teufel [9] produced two kinds of new crash locking devices which could lower the manufacture cost and withstand front- and rear-end crashes during car collision.

Nishino improved the conventional height adjusting device adapted to raise or lower the rear portion of a seat cushion [10]. The new invention provided a new seat height adjusting device, which provided a comfortable sitting feeling and was simple in structure. By using a pair of first and second sprocket wheels and an endless chain wound thereabout, Hatta eliminated the rattling or wobbling problem from previous designs [11]. Ito and Nawa designed a new device which could overcome the drawback produced by the engagement backlash between the inner and outer teeth portions, which will bring the occupant's discomfort [12]. Ramaseshadri and Brewer invented a lift structure which permitted movement of a restraint seatbelt with the height adjustment of the vehicle seat, thus reducing the potential for discomfort of the occupant [13].

Pallant and Johndrow utilized a clutch in their design thereby operating front- and rear-end seats simultaneously or separately [14]. Kawade

invented a height adjusting device with a function of moving the front part of the seat up and down to set up a desired inclination of the seat and a function of moving both the front and rear parts of the seat up and down to adjust the level of the whole seat [15]. Maruyama, Oishi, Terada, et al. produced a device for adjusting the front part of seat cushions [16]. It can simultaneously adjust the height and the hardness of a cushion pad in response to the occupant's individual preference.

Nishino invented a height adjuster operation mechanism which avoided the drawbacks of poor operation ability and discomfort to occupants [17]. Saito and Kanai proposed an invention to provide an improved height adjusting device which is easy to effect the locking and unlocking of seat height adjusting mechanism and retain the locked state thereof in a positive manner [18]. Kanai improved the operation force and simplified the operation of a lifter mechanism [19]. Uchimoto, Monmasu, Tomita, et al. simplified the construction of an adjusting device and made it possible to easily operate it with one hand [20]. The device with a handle separated assembly from the elevating mechanism can be easily repaired.

Kazaoka and Hayashi designed a supporting assembly which is simple in construction and very easy to fit within the limited space of the seat assembly [21]. Hoshi, Sugiyama and Yoshida invented a height adjusting device to prevent the side frames and the installation surfaces from being damaged or deformed [22]. Also, with a simplified construction, the seat frame obtains high rigidity which can stably support the seat. Willms and Garrido claimed that their inventions had many advantages, which were durable, easy to assemble and quiet in operation [23]. Hatta [24] and Matsumoto, Ikegaya and Sugimoto [25] focused their designs on simplifying devices thereby giving a wide space in the cabin of the automobile.

The objective of this paper is to present a study of a geared four-bar linkage for the height adjustment of an automotive seat cushion. Operation torque and lift distance will be investigated as major factors affecting the occupant's comfort. Ten automotive seat cushions in the market with

this kind of geared four-bar linkage will be compared and assessed. Finally, the resulting data will be discussed and conclusions will be drawn.

2. DESCRIPTION OF GEARED FOUR-BAR LINKAGE OF AUTOMOTIVE SEAT CUSHION

This section describes the fundamental operating principle of the geared four-bar linkage of automotive seat cushion, which consists of the driving pinion, geared-link, support link, bracket and seat track. Figure 1 shows the three-dimensional layout and partial definitions of the geared-four bar linkage system. The system is based on the seat

track, which is mounted on the vehicle body (Figure 1).

Figure 2 shows a two-dimensional schematic illustrating the operating principle of the geared four-bar linkage. The support link and the geared-link are pivoted to the seat track. The bracket is connected to the geared-link and support link with tube. The driving pinion is mounted on the bracket. When the driving pinion rotates about the axis p , the geared-link has a rocker motion around the center of the axis d . Then, the motion is transmitted to the whole four-bar linkage. Finally, the seat cushion can be moved up and down to satisfy the different requirements of occupants.

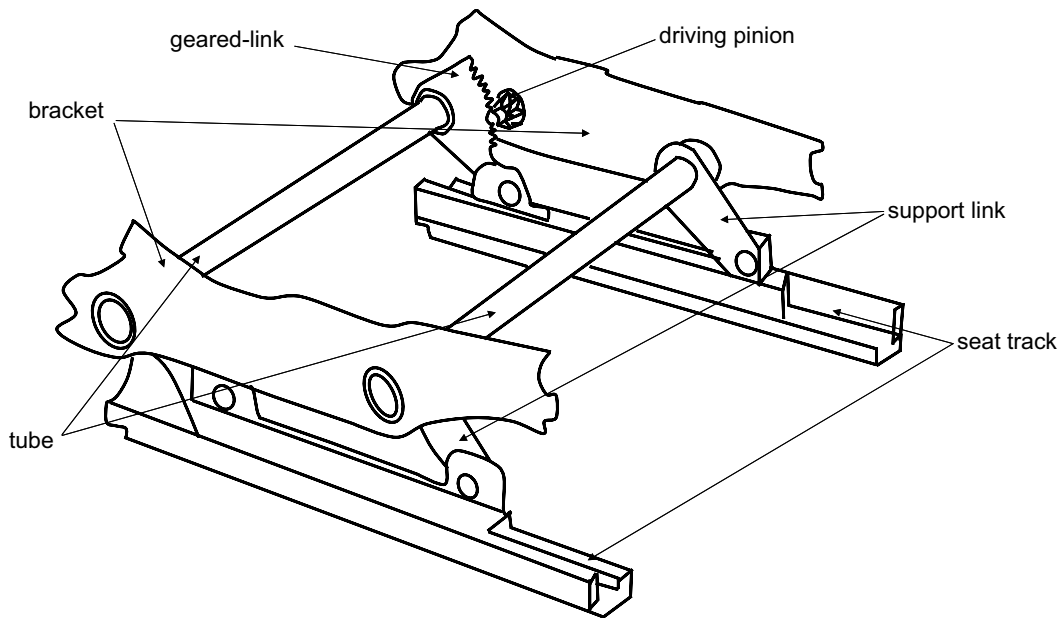


Figure 1. Three-dimensional layout of vehicle seat cushion with geared four-bar linkage.

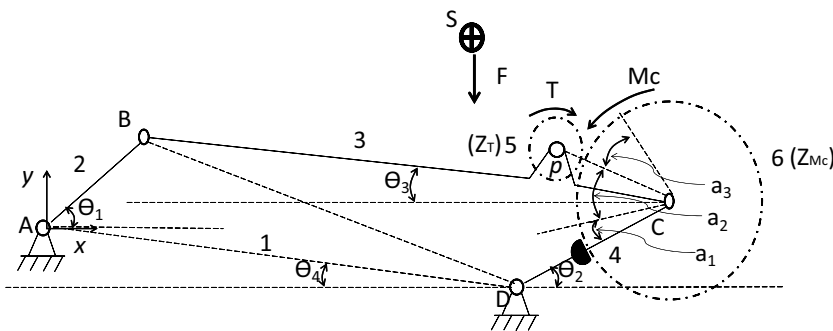


Figure 2. Two-dimensional schematic of seat cushion designed with geared four-bar linkage.
 Notes. A, B, C, D = points; p , x , y = axes; 1 = seat track, 2 = support link, 3 = bracket, 4 = geared link, 5 = driving pinion, 6 = sector of geared link; θ_1 , θ_2 , θ_3 , θ_4 = horizontal angles of parts with ground; F = occupant load on cushion; T = operation torque; Z_T = gear number of driving pinion; Z_{Mc} = gear number of sector; S = load point.

3. FORCE ANALYSIS OF GEARED FOUR-BAR LINKAGE

To analyze force acting on the linkage system, angles between different parts need to be calculated. Considering actual engineering design, L_1 , L_2 , L_3 , L_4 , θ_3 , θ_4 , a_1 , a_2 , a_3 and $\angle PCB$ (original design angle of sector on geared link) will be defined as original design parameters, the angle between bracket and geared-link can be defined with Equation 1:

$$\angle BCD = a_1 + a_2 - (\angle PCB - \theta_3) + a_4. \quad (1)$$

So, the angle between geared-link and ground is

$$\theta_2 = \angle BCD - \theta_3. \quad (2)$$

From the cosine theorem in $\triangle BCD$ and $\triangle BAD$, the angle $\angle BAD$ can be calculated with Equation 3:

$$\angle BAD = \arcsin \left(\frac{L_1^2 + L_2^2 - L_3^2 - L_4^2}{2 \times L_1 \times L_2} + \frac{2 \times L_3 \times L_4 \times \cos(\angle BCD)}{2 \times L_1 \times L_2} \right). \quad (3)$$

The angle between support link and ground can be described with Equation 4:

$$\theta_1 = \angle BAD - \theta_4. \quad (4)$$

Because no torque acts on the support link, the support link is a two-force member. The force can be calculated with Equations 5–7:

$$F_{32y} + G_{12y} = 0 \quad (\sum F_y = 0), \quad (5)$$

$$F_{32x} + G_{12x} = 0 \quad (\sum F_x = 0), \quad (6)$$

$$F_{32y} \times L_2 \times \cos(\theta_1) - F_{32x} \times L_2 \times \sin(\theta_1) = 0 \quad (\sum M_A = 0). \quad (7)$$

After eliminating L_{AB} , Equation 3 can be described with Equation 8:

$$F_{32y} = \tan(\theta_1) \times F_{32x}. \quad (8)$$

According to the four-bar linkage system, the bracket has the same function as a coupler:

$$F_{23y} + F_{43y} - F = 0 \quad (\sum F_y = 0), \quad (9)$$

$$F_{23x} + F_{43x} = 0 \quad (\sum F_x = 0), \quad (10)$$

$$F_{23y} \times L_3 \times \cos(\theta_3) + F_{23x} \times L_3 \times \sin(\theta_3) - F \times L_{CSx} = 0 \quad (\sum M_c = 0). \quad (11)$$

Due to action–reaction (Newton's third law)

$$F_{23y} = -F_{32y}, \quad (12)$$

$$F_{23x} = -F_{32x}, \quad (13)$$

$$F_{34y} = -F_{43y}, \quad (14)$$

$$F_{34x} = -F_{43x}. \quad (15)$$

Then, from Equations 4, 7 and 8, the reaction force from the support link to the bracket can be obtained with Equation 16:

$$F_{23x} = \frac{F \times L_{CSx}}{L_3 \times (\tan(\theta_1) \times \cos(\theta_3) + \sin(\theta_3))}. \quad (16)$$

The geared-link is the key part in this system, because the torque produced by the occupant is applied on this part (Equations 17–19):

$$F_{34y} + G_{14y} = 0 \quad (\sum F_y = 0), \quad (17)$$

$$F_{34x} + G_{14x} = 0 \quad (\sum F_x = 0), \quad (18)$$

$$G_{14x} \times L_4 \times \sin(\theta_2) - G_{14y} \times L_4 \times \cos(\theta_2) + M_c = 0 \quad (\sum M_c = 0). \quad (19)$$

The relationship between Equations 5, 6, 8, 9, 10, 11, 13 and 14 can be described with Equations 20 and 21:

$$G_{14y} = -F_{34y} = F_{43y} = F - F_{23y}, \quad (20)$$

$$G_{14x} = -F_{34x} = F_{43x} = -F_{23x}. \quad (21)$$

By substituting Equations 20 and 21 in Equation 19, the operation torque of the driving pinion can be expressed with Equation 22:

$$T = \frac{Z_T}{Z_{MC}} \times F \times L_4 \times \left(\cos(\theta_2) + \left(\frac{L_{CSx} \times (\sin(\theta_2) - \tan(\theta_1) \times \cos(\theta_2))}{L_3 \times (\tan(\theta_1) \times \cos(\theta_3) + \sin(\theta_3))} \right) \right). \quad (22)$$

4. PATH OF LOAD POINT S

Because the path of the load point S is a simple planar question, the geared four-bar linkage can be simplified as a typical four-bar linkage (Figure 3). The point S is a point on the coupler BC. The basic four links have the following vector relationship:

$$\vec{L}_1 + \vec{L}_4 = \vec{L}_2 + \vec{L}_3. \tag{23}$$

Projecting Equation 23 along axes *x* and *y* changes the equation into

$$\begin{cases} L_1 \times \cos(\theta_4) + L_4 \times \cos(\theta_2) \\ = L_2 \times \cos(\theta_1) + L_3 \times \cos(\theta_3) \end{cases} \tag{24}$$

$$\begin{cases} L_1 \times \sin(\theta_4) + L_2 \times \sin(\theta_1) \\ = L_4 \times \sin(\theta_2) + L_3 \times \sin(\theta_3). \end{cases} \tag{25}$$

Then, the co-ordinates of points A, B, C, and D are determined with Equations 26–29:

$$A_x = 0 \quad A_y = 0, \tag{26}$$

$$B_x = L_2 \times \cos(\theta_1) \quad B_y = L_2 \times \sin(\theta_1), \tag{27}$$

$$\begin{aligned} C_x &= L_2 \times \cos(\theta_1) + L_3 \times \cos(\theta_3), \\ C_y &= L_4 \times \sin(\theta_2) - L_1 \times \sin(\theta_4), \end{aligned} \tag{28}$$

$$D_x = L_1 \times \cos(\theta_4) \quad D_y = L_1 \times \sin(\theta_4). \tag{29}$$

Finally, the position of the point S moving along the desired path is expressed with Equations 30–31:

$$S_x = L_2 \times \cos(\theta_1) + L_3 \times \cos(\theta_3) - L_5 \times \cos(\theta_5), \tag{30}$$

$$S_y = L_4 \times \sin(\theta_2) - L_1 \times \sin(\theta_4) + L_5 \times \sin(\theta_5). \tag{31}$$

5. COMPARISON OF SEAT CUSHIONS

To compare the seat cushion from the aspect of operation torque and lift distance, parameters of 10 seat structures were obtained (Table 1). The evaluated seats were selected from 10 vehicle manufacturers. The cushions (A–J) were all designed with geared four-bar linkage system and they were up-to-date platform productions (manual or power) of different manufacturers. There is no name on seat cushion because no permission was granted by the vehicle manufacturers.

Obtaining the accurate load point and H-point of every seat cushion was impossible in this research. Therefore, to compare the operation torque and lift distance fairly, the 10 seat cushions were analyzed with a similar set-up:

1. The friction in linkage system was not taken into account.
2. The length of *L*₅ was 151.3275 mm and θ_5 was 7.595°.
3. There were 8 teeth in gear *Z*_T.
4. The load on point S was 1000 N (95th percentile male).
5. The load point S is identical with H-point in this study.

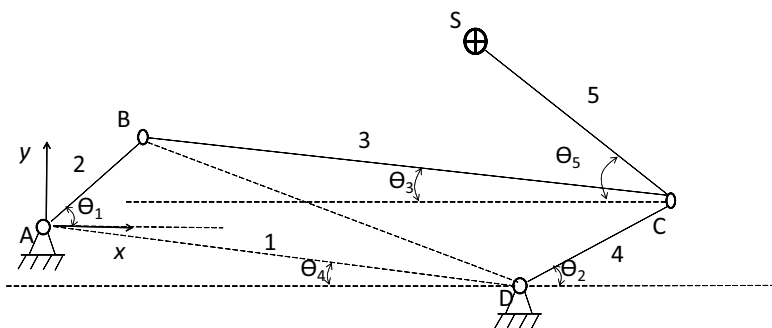


Figure 3. A simplified geometry of geared four-bar linkage. Notes. A, B, C, D = points; *x*, *y* = axes; 1 = seat track, 2 = support link, 3 = bracket, 4 = geared link, 5 = driving pinion; θ_1 , θ_2 , θ_3 , θ_4 , θ_5 = horizontal angles of parts with ground; S = load point.

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TABLE 1. Design Parameters of Geared Four-Bar Linkage

Parameter	Seat Cushion				
	A	B	C	D	E
AD(L1) ¹	291	269.76	284.2	309.72	278
AB(L2) ¹	80	122	85	77	77
BC(L3) ¹	316	306.5	306.5	322.4	292.5
CD(L4) ¹	97	160	100	87	87
a ₁ ²	21	45	26	23	31
a ₂ ²	19	14	21	19	14.5
a ₃ ²	33.5	18.5	19.5	30.5	29.5
θ1 ²	41.846	53.103	43.527	35.002	38.686
θ2 ²	29.8	45.16	33.23	29.578	31.21
θ3 ²	5.994	6.538	6.538	5.09	4.196
θ4 ²	5.501	10.898	6.262	5.094	3.774
∠PCB ²	10.2	13.8	13.8	12.4	14.3
Z _{MC}	66	97	97	64	64

Parameter	Seat Cushion				
	F	G	H	I	J
AD(L1) ¹	261.6	309.8	287.19	291	302.62
AB(L2) ¹	77	77	98.5	80	114
BC(L3) ¹	285	322.4	298	316	303.1
CD(L4) ¹	95	87	103	97	98
a ₁ ²	21	23	14.5	19	13.5
a ₂ ²	18.5	17.5	20.5	17.5	30.5
a ₃ ²	30	26.5	32.5	29.5	34.5
θ1 ²	42.68	32.536	30.963	39.941	43.78
θ2 ²	32.624	27.2	23.617	27.62	33.887
θ3 ²	2.614	6.089	6.648	4.865	5.148
θ4 ²	2.608	6.115	4.994	4	0.606
∠PCB ²	6.9	13.3	11.4	8.9	10.1
Z _{MC}	82	64	67	66	68

Notes. AD = ground link, AB = output link, BC = coupler, CD = input link; L₁ = length of seat track, L₂ = length of support link, L₃ = length of bracket, L₄ = length of geared link; a₁ = angle from geared link to down stop position, a₂ = angle from design position to down stop position, a₃ = angle from design position to up stop position; θ₁, θ₂, θ₃, θ₄ = horizontal angles of parts with ground; ∠PCB = original design angle of sector on geared link; Z_{MC} = gear number of sector; 1 = in millimetres, 2 = in degrees.

6. RESULT AND DISCUSSION

Figure 4 presents the operation torque from lowest to highest position when people adjusted the seat cushion. The lowest torque at lowest position is 6.5 Nm (cushion H) and the highest is 10.5 Nm (cushion J). However, at the highest position, the torque of cushion E decreased from 7.2 Nm to ~3 Nm, which performed the smallest operation torque. It was interesting to note that all operation torque curves had a parabolic trajectory except

cushion J. The curve of cushion J dropped sharply from ~10.5 Nm to ~4.2 Nm.

Figure 5 shows that cushion D provided narrow distance (57.1 mm) for the occupant to adjust while cushion A brought 85.3-mm adjustable range in the vertical direction, which is the widest among these 10 designs. In the horizontal direction, there is 73.7-mm displacement of cushion J, whereas the design of cushion F only gave 44.1-mm space for the adjustment of the customer (Figure 6).

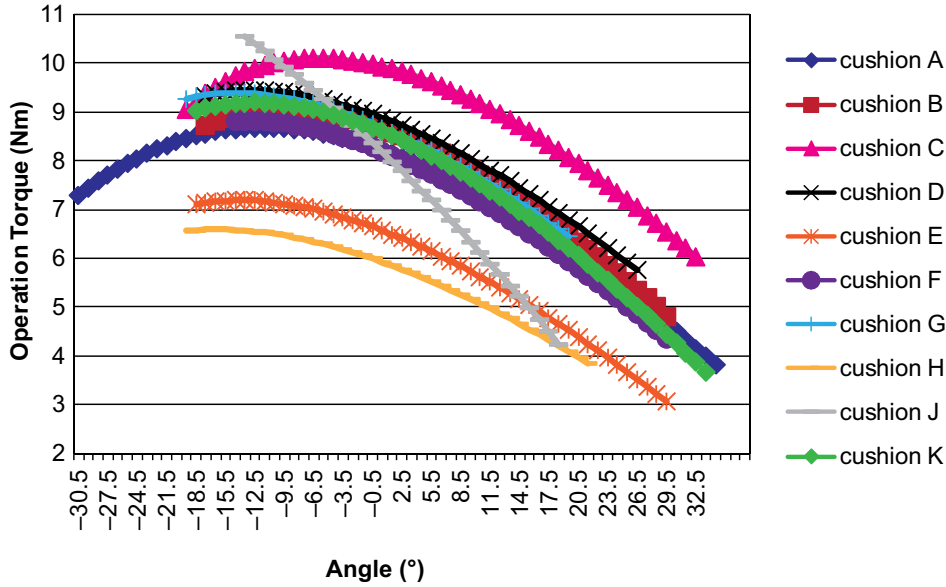


Figure 4. Output torque of 10 cushions with change of lift angle.

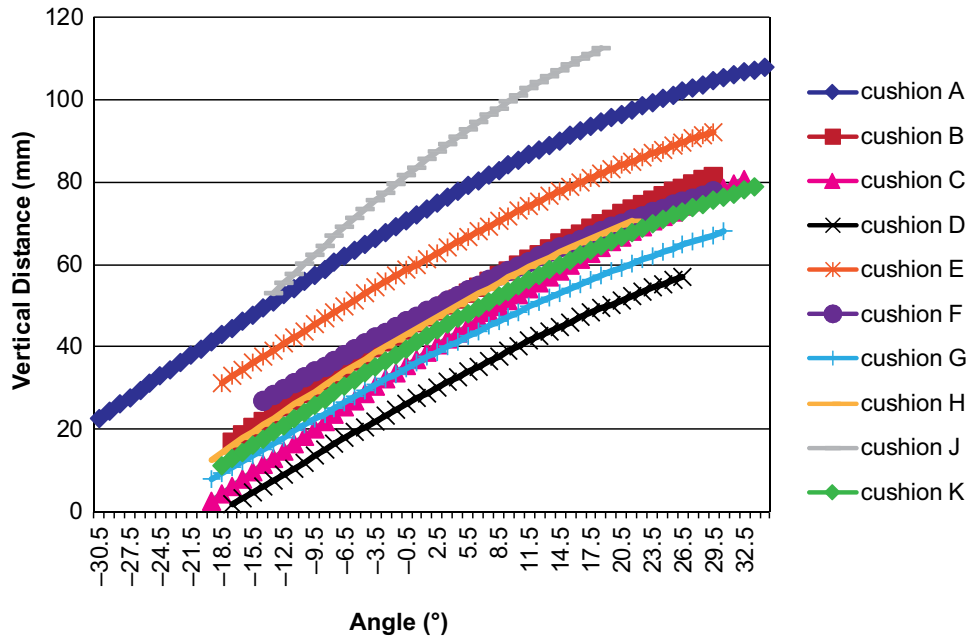


Figure 5. Moving range of seat cushion in horizontal direction with change of lift angle.

Admittedly, the results of this study are theoretical. Comfort aspects were independently assessed from different design outputs. It is, however, plausible that, e.g., cushion J, with a higher load point S, is more comfortable design compared to the others. The interdependence of various design outputs should be investigated as part of future research.

This study suggests that the design of a comfortable automotive seat could be considered at every design level. Considering the increasing comfort requirement of customers, engineers of the automotive seating industry should enhance the concept of design for comfort at every level of their design. This may help accelerate the design circle and lower the final cost of production.

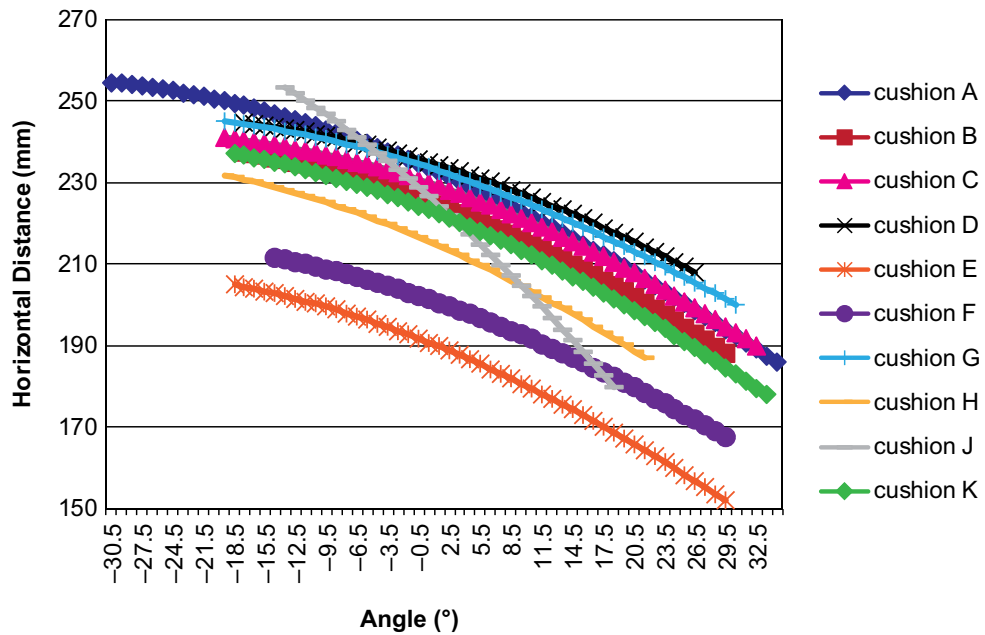


Figure 6. Moving range of seat cushion in vertical direction with change of lift angle.

7. CONCLUSIONS

From the designer's point of view, cushions H and E are preferred in terms of operation torque because both of them have a relatively lower operation torque and a smooth performance curve. Thus, the occupant will find it much easier to adjust the driving pinion. The design of cushion J should be avoided in a seat structure design. Firstly, its operation torque in the lowest position is too high. Secondly, the torque curve with a sharp decrease may bring negative impact to the self-lock ability of the seat structure.

Considering the seat adjustments are supplied to provide for some customization of the interior environment to the preferences of the occupant, cushion A performed the best while cushion F had the poorest design.

The load point S in the vertical direction of cushion J was consistently higher than the other cushions in the curve (Figure 5). Therefore, cushion J will provide good eye vision and will be more comfortable than the other nine cushions.

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