Modelling & Optimisation of Crew Module-Service Module Umbilical Separation System

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Abstract

This paper aims on preliminary design of separation configuration for Umbilical boom assembly connecting Crew module and Service module for pneumatic and electrical line communications. Discussion has been made for evaluating minimum force required graphically in-line with the constraints imposed by mission, launch vehicle and geometry of vehicle and payload fairing. This paper does not cover structural analysis, but presents insight of dynamics involved and preliminary inputs for design of umbilical separation system.

Keywords:*Umbilical System, Crew Module, Pneumatic Actuator, Optimisation, Boom Assembly.*

1. Introduction

Indian Space Research Organisation has initiated its first manned space programme intended to put crew in a LEO for a duration of 5-7 days. Mission is planned with ingenious Crew module and Service module to avail habitat and life supporting medium to crew in orbit till there return safely to pre-determined location on earth surface. Crew module (CM) serves as a habitable volume for crew and service module (SM) consist of mainly life supporting system and propulsion module for the mission. [3] CM-SM umbilical system acts as a communication conduit between service module and crew module for transferring liquids, gases and also electrical power supply and data cables.

CM and SM are mechanically coupled to each other generally by explosive bolts, which are fired at the time of separation. Utility connections between CM and SM are facilitated by umbilical system, which provides mechanical coupling to umbilical plate on CM (viz. CM half) in connected position (Fig 1) and is supported on SM structure. Umbilical system remains in mated position throughout the launch and orbital duration. System needs to be separated at the time of re-entry, when CM starts deorbiting. In addition to this, abort conditions also require de-mating of CM from SM. This requires some additional features in design of umbilical and its separation system. The proposed umbilical system consists of Umbilical plate sub-assembly, Boom and Actuator. Umbilical plate sub-assembly which houses fluid/electric connectors, consists of two plates in mated condition during operation. One plate is mounted on the fore-end of deployable boom and other one is fixed to CM. All lines are routed from SM to CM through boom. The boom is a rigid arm with rotational degree of freedom about a fixed point located on SM structure.

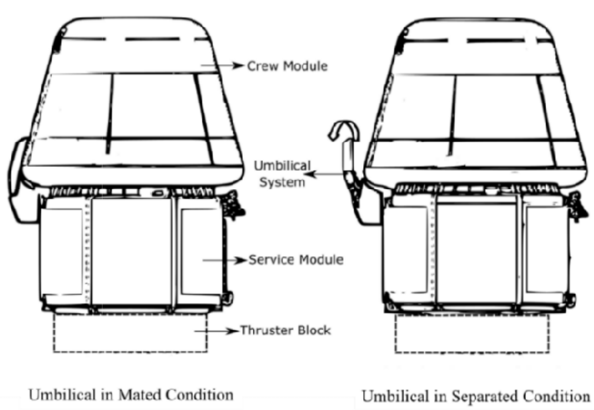

Fig 1 Overall configuration of CM-SM Umbilical

This rotational motion is operated by linear actuators supported on SM structure (Fig 1), which exert a force to rotate the boom and assembly up to required displacement, when required. Umbilical system can be actuated back in multiple ways like linear actuator, rotational actuator etc. Scope of using innovative mechanism can also not be ignored, which may be used as per the mission requirements. Irrespective of type of actuator, some minimum/optimum force requirement will always be there, optimisation of which is discussed in this paper.

2. Nomenclature

m_b	Total mass of boom assembly
m_s	Mass of fore-bent part of boom assembly
m_a	Mass of rear-bent part of boom assembly
l_s	Length of fore-bent part of boom assembly
l_a	Length of rear-bent part of boom assembly
l_b	Total length of boom assembly
ϕ	Inclination of actuator from horizontal
θ	Inclination of boom hinge part from horizontal
β	Slant angle of conical surface of CM
γ	Angle between line joining CG of boom assembly to boom hinge and horizontal
α	Angular acceleration of boom assembly required
δ	Angular movement required at fore-end of boom assembly
t	Time Variable
I_s	Moment of inertia of fore-bent part of boom assembly
I_a	Moment of inertia of rear-bent part of boom assembly
I	Total Inertia of boom assembly
F	Force required to operate boom assembly
g	Acceleration due to gravity
H	Horizontal offset of boom hinge from SM
D	Vertical offset of boom hinge from SM
R	Maximum radial length of boom assembly
h	Horizontal offset of point P from SM
d	Vertical offset of actuator hinge from SM top surface
d_1	Vertical offset of point P from SM top surface
V	Vertical offset of umbilical plate on CM from SM top surface
X_{cg}, Y_{cg}	Coordinates of CoG of boom assembly relative to boom hinge point
r_{cg}	Distance of CoG of boom assembly from boom hinge point
$x_{s,cg}$, $y_{s,cg}$	Coordinates of CoG of fore-bent part of boom assembly relative to boom hinge point
$X_{a,cg}$, $y_{a,cg}$	Coordinates of CoG of rear-bent part boom assembly relative to boom hinge point

3. Development of Mathematical Model

A simplified free body diagram of the configured actuator-boom assembly is shown in Fig. 2(a). Boom is assumed to be of bent type, perfectly rigid and uniform density structure. There are possibilities of wide range of shape of boom which can be finalized with. However, for the present analysis, a simple bent type shape is considered, as it is closest approximation to the optimum shape in almost all the cases. Umbilical de-mates with CM in two stages (refer figure 2) [4]. In mated condition, the boom is supposed to hug CM surface which is very close to a conical shape due to aerodynamic considerations. Thus, a straight portion is always expected in boom which has to flush with CM surface. Length of this portion is 'l_s' as shown in Fig. 2(a) and inclined at angle 'β', which suits the half conical angle of CM structures. Bottom of this fore-end portion, is the hinged portion of boom, which may have different inclination from vertical. Length of this hinge portion is shown as l_a (refer Fig. 2(a)). The mass distribution of boom assembly may be assumed to be uniform throughout the length. This assumption may be justified with the fact that fore-end of boom usually had some concentrated mass of sealing devices, umbilical plate, connectors etc. But electrical and fluid harnesses laid along the boom which terminates at the top and

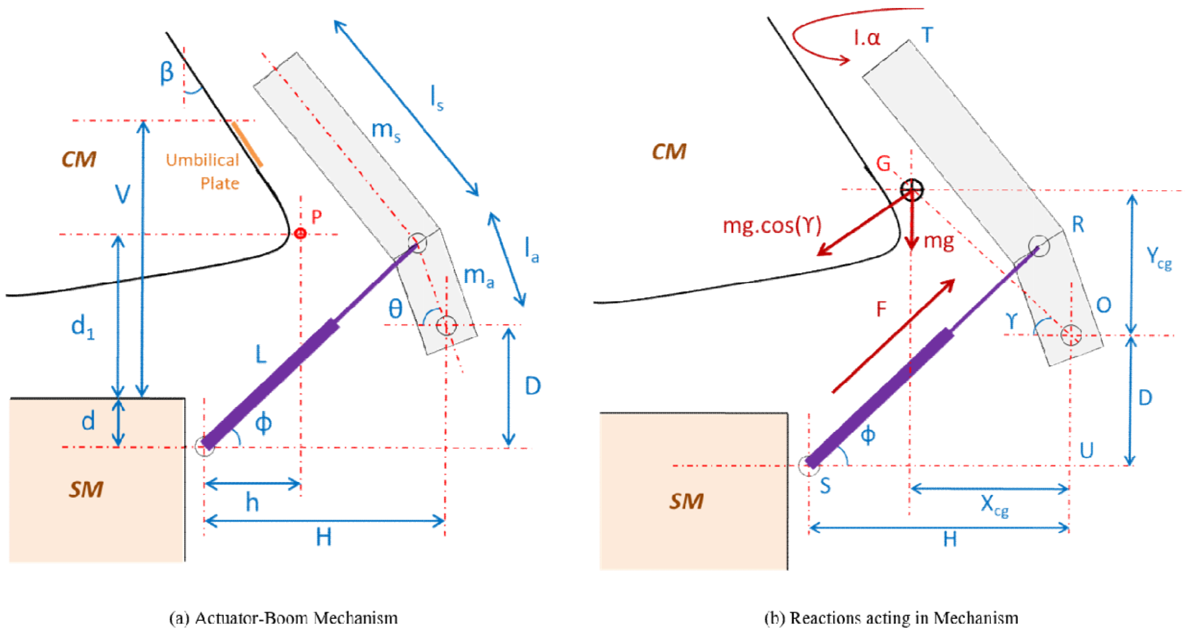

Fig. 2 Simplified Actuator-Boom Assembly figure

remain supported at intermediate length of boom had effect of lowering down the centre of gravity toward hinge. Ribs and supports are also located at the bottom end of boom assembly to provide additional stiffness and rigidity to structure near to hinge. Width of the boom also reduces toward for-end to earn mass & uniform strength benefits. All these facts tend to lower the centre of mass of boom assembly; hence, author has assumed mass to be evenly distributed throughout the length of boom assembly. This consideration holds good to simplify preliminary study and design process.

For detailed analysis, boom structure can be considered as comprising of two elements. First, hinged portion of length l_a and mass m_a. Second, straight portion to flush with CM of length l_s and m_s (Fig. 2(a)). So physical properties of boom can be determined as effective sum of these two parts.

Total mass of Boom assy: $m_b = m_s + m_a$, Total length of Boom assy: $l_b = l_s + l_a$.

3.1 Centre of Gravity of Boom Assembly

Centre of Gravity of boom assembly can be calculated by taking weighted average of CoG individual boom elements with respect to mass.Considering point 'O' as origin (Fig. 2(b)), horizontal and vertical component of CoG can be formulated as:

$$X_{cg} = \frac{x_{s,cg} \cdot m_s + x_{a,cg} \cdot m_a}{m_s + m_a} = \frac{1}{l_b}.[l_s.\{l_a.\cos\theta + 0.5.l_s.sin\beta\} + l_a.\{0.5.l_a.cos\theta\}]$$

$$Y_{cg} = \frac{1}{l_b}.[l_s.\{l_a.\sin\theta + 0.5.l_s.cos\beta\} + l_a.\{0.5.l_a.sin\theta\}]$$

$$OG = r_{cg} = \sqrt{X_{cg}^{\ 2} + Y_{cg}^{\ 2}} \qquad \text{... Eq.(1)}$$

$$\gamma = tan^{-1}\left(\frac{Y_{cg}}{X_{cg}}\right) \qquad \text{... Eq.(2)}$$

Following can be inferred from above expressions: $X_{cg} = X_{cg}(l_s, l_a, \theta, \beta)$, $Y_{cg} = Y_{cg}(l_s, l_a, \theta, \beta)$

3.2 Inertia of Boom Assembly

Inertia of assembly about point 'O' can be expressed as sum of inertia of both elements.

$$I = I_s + I_a \qquad \text{... Eq.(3)}$$

where, $I_a = \frac{m_a.l_a^{\ 2}}{3}$; $\quad I_s = m_s.\left[l_a^{\ 2} + \frac{l_s^{\ 2}}{3} + l_a.l_s.\cos(\theta - \beta)\right]$

Following can be inferred from above expressions:

$$I_a = I_a(l_s, l_a, \theta, \beta), \qquad I_s = I_s(l_s, l_a, \theta, \beta)$$

3.3 Rigid body dynamics

Dynamics equation is formulated using torque balance principle about pivot point 'O' of the boom (Fig. 2(b)). The additional inertial forces, due to acceleration of the vehicle, also plays a role in the dynamics of boom and shall be incorporated in parameter 'g'. As there is no loss in generality, the present study does not account for such inertial force. Considering rotational form of Newton's second law of motion [2], the torque (τ) acting on the boom is expressed about origin 'O'.

$$\Sigma\tau = I.\alpha,\ F_a.l_a.\sin(\phi + \theta) - m.\ g.\ r_{cg}.\cos\gamma = I.\alpha$$

Rearranging above equation will give formulation for actuator force required:

$$F_a = \frac{I.\alpha + m.g.r_{cg}.\cos\gamma}{l_a.\sin(\phi + \theta)} = F_a(l_s, l_a, \theta, \phi, \beta)$$

3.4 Kinetics

Actuation time requirement of boom is very critical, especially in case of abort. Along with time, angular displacement is also important, to clear off boom assembly from separating CM. These requirements impose kinematic constraint on the system in term of angular acceleration.

Acceleration can be formulated in terms of time and arc length required to clear CM during separation. From second equation of rotational motion:

$$\theta = \omega_o.t + {}^1\!/_2.\alpha\,t^2;\quad R = \sqrt{l_s^{\ 2} + l_a^{\ 2} + 2.l_s.l_a.\sin(\theta + \beta)}$$

Solving above, for angular acceleration; $\alpha = \frac{\delta}{\sqrt{l_s^{\ 2} + l_a^{\ 2} + 2.l_s.l_a.sin(\theta + \beta)}}.\frac{2}{t^2}$ …Eq.(4)

Above equation gives value of angular acceleration required on boom, to operate within t time interval and clear δ distance at fore-end of boom to clear CM and avoid any collision between CM and umbilical system. This expression is used in conjunction with force requirement expression.

3.5 Geometric co-relation of Boom and Actuator

Fig. 2 shows the position of boom and actuator in mated condition. Actuator is connected to the boom at point 'R', and is pivoted at 'S' to SM structure. Boom is swayable about its hinge point 'O'. To model the location of boom and actuator mathematically, actuator and boom is assumed a straight-line (TR) withpoint 'S' as origin as given in Fig. 2(b)). Point 'P' is the outermost point on CM and is critical as the straight part of the boom has to flush CM surface and thus, should be passed from this point, having same inclination as that of CM surface (angle β). Thus, equation of boom will be written for straight portion, passing through point 'P'.Inclination of actuator can be expressed as ϕ at any instant passing through point 'S' (origin). Equation of straight line passing through origin and inclined at ϕ angle with positive horizontal is, $y = x.\tan\phi$

Similarly, equation can be written for Boom assembly inclined at angle of (90°+β), and passing through point $(h, (d_1 + d))$ as $y = (x - h).\tan(90° + \beta) + (d_1 + d)$.

Solving above two equations; $$L = \frac{(d + d_1) + h.\cot\beta}{\cos\phi.(\tan\phi + \cot\beta)} \quad \text{... Eq.(5)}$$

Equation 5 ensure boom to mate with CM surface, without hitting. But actuator and boom in this case may interfere with CM surface. Thus, a constraint must be applied on point 'R' to avoid being interfere to CM surface. Following expression serves the purpose.

$$L.\cos\phi > h \quad \text{... Eq.(6)}$$

3.6 Moment Arm Formulation

Moment arm is important for optimising force requirement for actuator. Increasing moment arm will reduce this force but will occupy more space outside SM. Hence this parameter also has to be optimised.

From trapezium ROUS (Fig. 2(b)); $H = L.\cos\phi + l_a.\cos\theta$

$$l_a = \frac{H - L.\cos\phi}{\cos\theta} \quad \text{... Eq.(7)}$$

3.7 Model for Boom-hinge position

Height of hinge with respect to SM can also be defined in terms of length of actuator and moment arm. Thus, serves additional constraint to the model.

Consider trapezium ROUS (Fig. 2(b)).

$$D = L.\sin\phi - l_a.\sin\theta \quad \text{... Eq.(8a)}$$

Boom hinge position is subjected to an additional constraint, which is distance from CM umbilical plate to boom-hinge position. The extreme of boom assembly should at-least be clearing top of umbilical plate fixed on CM structure. This constraint can be formulated by taking vertical height of umbilical plate from SM and equating it to vertical component of boom assembly length.

$$l_s.\cos\beta + l_a.\sin\theta + D - d \geq V \quad \text{... Eq.(9b)}$$

In addition to the above models derived from space constraint & operational requirements, other constrains may also exist depending on mission requirement.

4.0 Problem Formulation

Optimisation problem is formulated for minimum actuator force requirement meeting other space, design and operational constrains. Thus, below equation has to be minimised:

$$F_a = \frac{I.\alpha + m.g.r_{cg}.\cos\gamma}{l_a.\sin(\phi+\theta)} \quad \text{… Eq.(10)}$$

which is an objective function of 5 variables l_s, l_a, θ, ϕ & β. Terms r_{cg}, γ, I, α are represented by equation 1, 2, 3 & 4 respectively. All these terms are function of l_s, l_a, θ & β. From equation 9 it can be observed that F_a is inversely proportional to $\sin(\phi + \theta)$. This seems obvious from Fig. 2(a) also, as angle $(\phi + \theta)$ is the inclination of actuator to boom. To maximize utilization of available arm length l_a, line of action should pass perpendicular to moment arm.

This indicates $(\phi + \theta) = 90°$, for minimum force requirement. …Eq.(11)

However, it should be noted that we are pursuing optimisation for minimum actuator force requirement. There might be a case where sensitivity of F_a in equation 9 on angle $\phi \left(or\ \frac{\partial F_a}{\partial \phi}\right)$, is comparatively more as compared to the case when $\sin(\phi + \theta) = 1$ in equation 9. In such case, optimum solution may not come up with above assumption. However, for the current study, author has checked this dependency and validated the applicability of equation 9 to this problem. Optimisation problem can be stated in mathematical form now. List of constraints against objective function (equation 9) are listed below:

i. $g_1: L = \frac{(d + d_1) + h.\cot\beta}{\tan\phi + \cot\beta}$; *ii.* $g_2: L.\cos\phi \geqslant h$,

iii. $g_3: l_a = \frac{H - L.\cos\phi}{\cos\theta}$; *iv.* $g_4: D = L.\sin\phi - l_a.\sin\theta$;

v. $g_5: (\phi + \theta) = 90°$; *vi.* $g_6: l_s.\cos\beta + l_a.\sin\theta + D - d \geq V$ …Eq.(12)

It should be noted that till now we have considered all parameters as variable only, and have not assigned any value to those parameters which are not variable. Such parameters will be called 'Preassigned variables'[1].Thus, number of variables can be reduced by identifying these values. It can be observed that objective function (equation 9) and constraints (equation 11) consist number of other variables also, which can to be segregated among 'Decision variables', 'Preassigned variables' and 'Derived Variables'[1]. This allows us to proceed with the optimisation.

Decision Variables: ϕ, *Preassigned Variables:* H, d_1, d, h, β, m_b, l_b, V

Derived Variable: L, θ, l_s, l_a, D, m_s, m_a, X_{cg}, Y_{cg}, r_{cg}, γ, α

Constraints listed in equation 11can be breakdown in terms of function of variables as:
From g_1: $L = L(\phi)$; *From g_5:* $\theta = \theta(\phi)$; *From g_3, g_1 & g_5:* $l_a = l_a(\phi)$; *From g_4, g_3& g_5:* $D = D(\phi)$. l_b has been considered as preassigned value (with some arbitrary value) just to facilitate first iteration process which can be refined as the iteration continues. Iterations are required to find optimum values of total length of boom assembly (l_b) and actuator inclination (ϕ). Force requirement decreases with decreasing l_b, and angle ϕ (refer section 5.0). But subjected to constraint g_6, optimum force requirement corresponds to higher value of angle ϕ at lower values of l_b, which in turn increases the force requirement. So iteration allows us to get optimum combination of boom length and actuator inclination.With the inferences made, all the derived parameters can be expressed in terms of single variable ϕ at some constant l_b.

From section 3.1, equation 1: $r_{cg} = r_{cg}(\phi)$, *From equation 2:* $\gamma = \gamma(\phi)$, *From equation 4:* $\alpha = \alpha(\phi)$. Table 1 presents the values of preassigned variables, for the problem in current scope.

Table 1: Values of Preassigned variables

H	300mm	*Maximum radial gap outside between SM and heatshield*
d_1	440mm	*From geometry of CM*
d	50mm	*SM Structural constraint*
δ	330 mm	*Max. angular displacement of boom assembly*
h	120mm	*From geometry of CM*
β	14°	*From CM structure design*
m_b	100kg	*Expected mass of boom, hoses,*
V	700 mm	*Position of fixed CM umbilical plate*
l_b	1000mm	*First iteration value.Approximate expected length.*

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5.0 Optimisation

Values from Table 1 can be used with constraints formulated in equation 11. As inter-dependency of constraints lead to single decision variable, all the design critical variables can be plotted on a 2D graph with decision variable (angle ϕ) on abscissa axis, and other parameters on accordingly scaled ordinate axis. For the first iteration with l_b = 1000 mm, derived parameters are plotted on *MS excel 2016*. Graph is shown in Fig. 3.Force values (in kN) are shown in secondary axis on right of graph.

Variation of actuator force is parabolic. Plot of angular acceleration is not shown in Fig. 3, as the v value is almost constant near 16.5rad.s^{-1}, for feasible values of ϕ. For better visibility variable V hasbeen plotted with 0.5 scale factor. Plot shown can be analysed to find the optimum range of acceptable parameters. Value of ϕ can be identified through the plot shown in Fig. 3, by opting favourable values of other parameters. Optimisation can be initiated by equation 6 which allows a particular range of angle ϕ to have feasible solution. Value L.cosϕ is monotonously decreasing with increasing ϕ. L.cos ϕ is equal to 120 mm at ϕ=76.2°.

Thus, all values greater than ϕ = 76.2° are not feasible as per the constraint in equation 6.Force requirement tends to 3.9 kN, which is increasing with reduction of ϕ till ϕ=52°, where force required achieve maximum requirement value of 4.9 kN. This infers the requirement of reduction of ϕ further at-least to 28.8° which shows same force requirement of 3.92 kN as of ϕ = 76.2°. Reducing ϕ below 28.8°, provides benefit of reducing force requirement, which is the sole purpose of this study. However, lower limit is also imposed by constraint g_6 in equation 11. Value of V should be greater than 700 mm (=350 in reduced scale of graph in Fig. 3), which corresponds to the value of ϕ = 14.5°. These two constraints present an acceptable limit of angle ϕ as 14.5° ≤ ϕ ≤ 28.8°. Force requirement reduces continuously with reduction of value of ϕ in this range, but at the same time, moment arm length tends to increase asymptotically at values closer to ϕ=14.5°. Also, vertical offset of boom hinge point from SM also increases asymptotically in negative direction. Negative direction denotes the hinge point to lower down from SM level, as we have assumed it to be above the SM at the first place. It shall be noted that large negative value of variable D is also undesirable, as it asks for longer boom assembly. This may result in heavier umbilical system with high force requirements and may need even more iterations to converge to any optimum solution. These all limits lowering value of ϕ close to 14.5°. Value of l_a and D changes asymptotically in the vicinity of ϕ=14.5°. This provides an insight of optimum value of angle ϕ and force requirement in range at around ϕ = 16°.From above discussion, minimum feasible value of angle ϕ i.e. 16° can be selected for further analysis. Consequent values can be obtained.

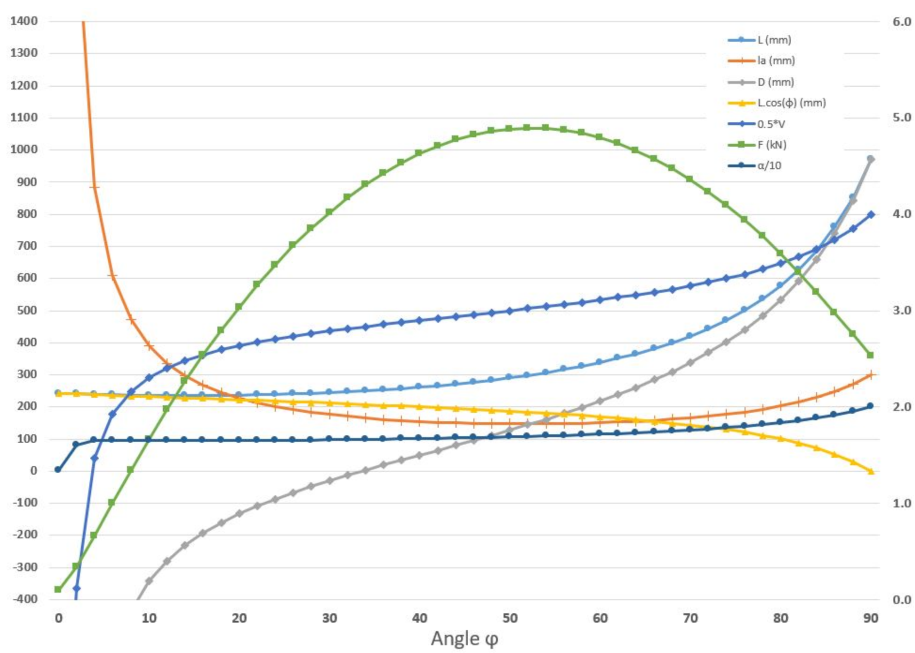


Fig. 3 Plots of derived variables for First Iteration

Table 2: Values derived from constraints after first Iteration

ϕ	16°
L	235.4mm
l_a	267.5mm
D	192.2mm
L.cos ϕ	226.3mm
α	16.5 rad.s^{-1}
V	726 mm
F	2.538kN

Next iteration may be performed with a different value of l_b. Objective of next iteration is to analyse the effect of increasing boom length and corresponding change of actuator inclination (ϕ) on optimum force requirement. Repeating all the steps again, and value of pre-assigned variable l_b= 1050 mm, figure 5 is obtained.

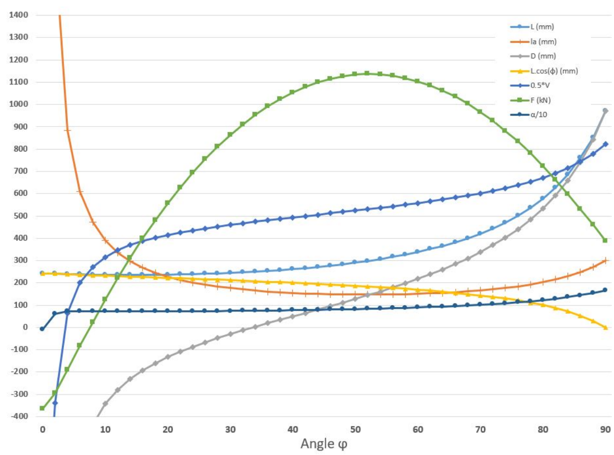


Fig.4 Plots of derived variables for l_b=1050mm, Second Iteration

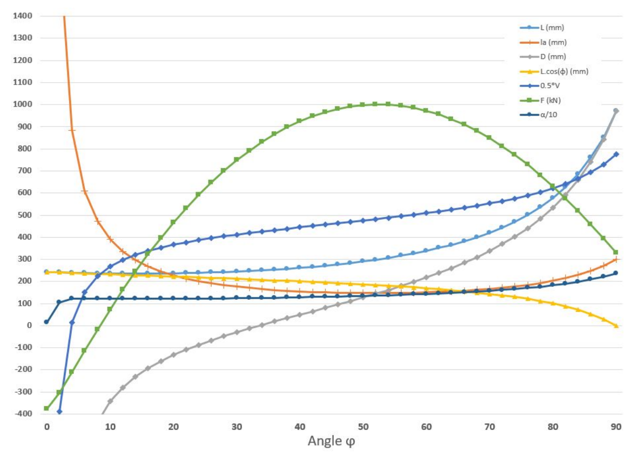
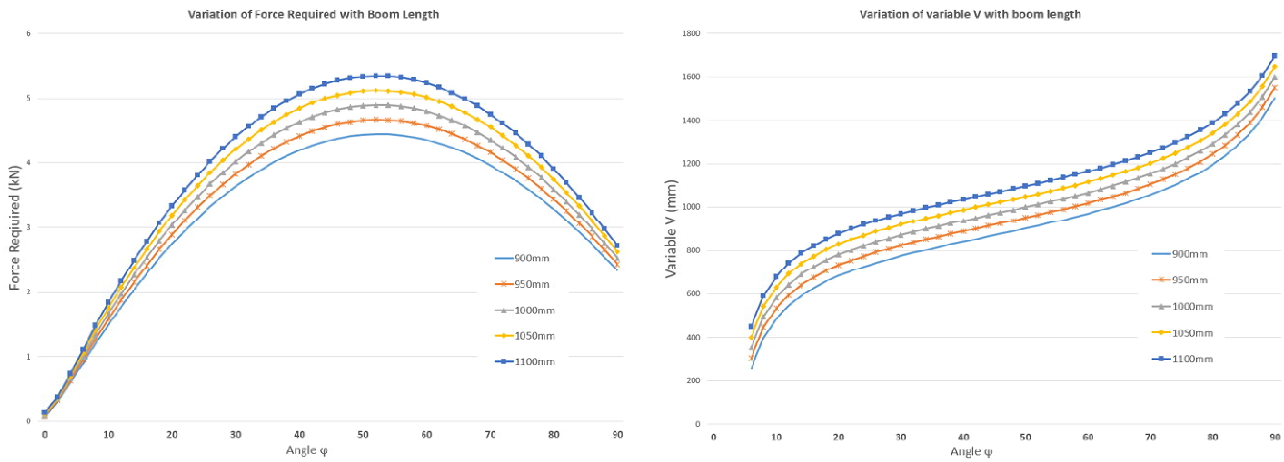


Fig 5: Plots of derived variables for l_b=950mm, Third Iteration

Figure 5 shows the variation of same parameters, as shown in Fig. 3. It should be noted that the change of value of l_b has its effect only on variable F, α and V as these variables are dependent on l_s, which in-turn depend on l_b. Due to this, limit on lower permissible value of angle ϕ, due to asymptotic nature of l_a and D will remain same at ϕ=16°. Now, and for the constraint g_6, value of V should be at-least 700 (= 350 on reduced scale in figure 5) is achieved at ϕ=12.3°. It indicates that increase in total length of boom requires lower value of ϕ for clearing CM umbilical plate in vertical direction. So minimum permissible value of actuator inclination will remain ϕ=16°. Force required in this iteration increases to 2.67 kN, which is higher than previous value corresponds at l_b=1000 mm. Next iteration can be performed with a lower value of l_b= 950 mm. Corresponding graph is shown in figure 6.

This case force curve F and variable curve V reduces with l_b and shifted to lower values. This leads to increase of minimum feasible value of ϕ. This curve projects ϕ=17.5° for V=700 mm, while limitimposed by variables l_a and D remain same at ϕ = 16°. This time feasible range of ϕ gets changed to 17.5° ≤ ϕ ≤ 28.8°. Value of F is still monotonously decreasing in this range as it was in first iteration. Looking for minimum value of F corresponding to 17.5° ≤ ϕ ≤ 28.8°, we get value of F=2.600 kN, which is still greater than the first iteration.

A detailed analysis can be carried out by performing multiple iterations around length l_b = 1000 mm, in smaller steps to refine the minimum force to further extent. Fig. 4(a) shows the variation of plot of force requirement with a given set of preassigned values and varying l_b against angle ϕ. While Fig. 4(b) shows variation of variable V as calculated from equation 8b with l_b against angle ϕ.



(a) F vs ϕ, with varying l_b (b) V vs ϕ, with varying l_b

Fig. 4 Variation of F and V with lb

Continuingthe iterations, minimum F is obtained $l_b$close to 1000 mm, which was the initial guess in present study. This concludes the optimisation process with the optimum values listed in Table 3.

Table 3: Optimum values of Derived Variables after optimisation.

ϕ	16°
L, D, V	235.4 mm, 192.2 mm, 726 mm
l_a, L.cos ϕ	267.5 mm,226.3 mm
α, F	**16.5 rad.s^{-1}, 2.538 kN**
θ, γ	74°, 75.1°
l_s	732.5 mm
m_s, m_a	73.3 kg, 26.7 kg
X_{cg}, Y_{cg}, r_{cg}	128.8 mm, 483.0 mm, 499.8 mm

6.0 Conclusion

A graphical analysis has been performed to optimise the independent parameter which was derived up to be inclination of actuator and total length of boom assembly. Geometric constraints and co-relation have been formulated for present study along with objective function of force requirement. Equations are graphically analysed and optimum values have been approximately selected among the feasible range of values. Iterations are performed to check for optimum set of values actuator inclination and boom length. Solution of system of equations have been solved assuming actuator force acting perpendicular only at the initial phase of de-mating boom assembly. This assumption is problem specific and system of equations may produce other feasible solutions also without considering this condition depending on initial state of study.Though graphical analysis has been performed, solution becomes sensitive to initial values, quantity of constraints and steps of iterations performed. Present study does not consider kinetics and dynamics of vehicle or ambient conditions.

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