Performance evaluation of first stage support system of launcher

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The First Stage Support (FSS) system of a launcher based on slider-crank mechanism is presented. The unit has to withstand high magnitude of load and operates at high speed. Study includes dynamic analysis of FSS theoretically to examine effects of different parameters at design stage and experimentation to establish the desired reliability. Unit has been designed, fabricated and its performance has been studied for final installation on launcher.

Keywords: Dynamics, FSS, Launcher, Mechanism, Numerical integration

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Introduction

The First Stage Support System (FSS) is an important subsystem of a launcher to support the vehicle to be launched during erection and launching. It consists of a beam, which is hinged on the base structure and support units, viz cup support, FSS and inter-stage supports, all installed on the beam of launcher at various locations (Fig. 1). In horizontal condition of the beam, all the three support units carry weight of the vehicle. During erection of the vehicle, beam rotates about the hinge at its base. At the vertical configuration, vehicle rests on the base support and the FSS supports vehicle against any horizontal force. The other two support units become inactive and are retracted. As the vehicle moves upwards, the arms of FSS are released from vehicle and have to swing sufficiently within a very short time to provide obstruction free path of the vehicle.

While the FSS has to withstand the load on it, because of space restriction, its weight needs to be optimized to achieve the desired performance criterion. Although most of the launchers are provided with supporting units, the operations of the mechanisms may vary. The present FSS is based primarily on slider-crank mechanism, operated by pneumatic piston-cylinder unit. The basic methodology of analysis of such mechanism is based on standard procedure¹,². Slider-crank mechanisms are widely used for various applications³–⁶.

The paper presents results of theoretical and experimental investigations on the performance of FSS mechanism. For theoretical study, the governing equations of motions have been developed from kinematic and force relations and solved as initial value problem by Runga-Kutte method⁷. Primary parameters (length, mass and moment of inertia of different components) have been estimated from the solid model of the unit. The performance of the fabricated unit has been examined on a simulated test rig, developed for the purpose. The other relevant design parameters have been evaluated from the theoretical model at the design stage.

First Stage Support System

FSS (Fig. 2) consists of two arms, at one end of which there are two half pins and the other ends are connected to two turnbuckles. The arms can swing about two hinges. For extra support to the arms, two stay braces are also attached to the arms, which can swing about the same axes of the main hinges. The other ends of the turnbuckles are connected to the piston rod of a pneumatic cylinder through a crosshead. The unit is installed on the beam at the main hinges.

The arms of FSS are closed, for connecting to the vehicle, so that the half pins form one full cylindrical pin. The pins of FSS are attached to the vehicle in its horizontal position. Under this configuration, FSS remains vertical and has to support part of weight of
the vehicle. Air pressure in the pneumatic cylinder is then increased so that the turnbuckles tend to swing the arms. The pressure is set to the desired value. The beam along with the vehicle is subsequently erected to a vertical configuration when the FSS becomes horizontal. At this position, FSS provides the necessary support to the vehicle against horizontal forces and moments due to wind, eccentricity of C.G. of the vehicle with the central axis and deviation of angle of vertical thrust of lifting. They are estimated accordingly and used at the design stage.

During launching, once the vehicle moves by a certain distance, it detaches itself from the FSS. The arms immediately swing sufficiently so that they do not obstruct the lower portion of the vehicle, which is wider, during movement of the vehicle. At the end of its swing, arms are locked to avoid any possibility of retraction (Fig. 2). The time taken by the arms to swing by specified angle (Opening Angle), which is sufficient to provide obstruction-free path of the vehicle, is defined as ‘Opening time’. This value is estimated from the time required by the wider portion of the vehicle to reach the location of FSS during its flight. It is essential that this time is less than a certain specified value.

**Theoretical Formulation**

The analysis has been carried out when the arms are set into motion and up to the end of their travel.
Considering symmetry of the unit, one side of the mechanism of FSS is closed configuration along with geometrical parameters (Fig. 3). AOB is the arm assembly hinged at O, BC as the turnbuckle, the end of which is connected to a slider attached to the piston rod. The arm assembly consists of the arm, stay brace and other attachments. OB of the arm assembly is the crank with hinge at O. The turnbuckle acts like the connecting rod. The unit is operated by pneumatic pressure through the piston CD, which slides parallel to X-axis. The arm OA of the FSS is an integral part of the crank. The geometric parameters of the members are length, angle between various members and location of centre of gravity of different members. The origin of the coordinate system has been selected at the main hinge with X- and Y-axis along the directions, shown in the figure. The kinematic relations can be easily derived (Fig. 3a). From the external force and inertia forces of different members (Fig. 3b), the force relations are derived. The derivations are given in the Annexure.

The governing equation of the system is obtained as:

\[ K_A \frac{d^2 \theta}{dt^2} + K_B \left(\frac{d\theta}{dt}\right)^2 + K_C = 0 \]  ... (1)

where, \( m \), mass of arm assembly; \( I_a \), moment of inertia of arm assembly about C G; \( m_t \), mass of turnbuckle; \( I_t \), moment of inertia of turnbuckle about C G; \( m_p \), mass of piston unit; and \( F_p \), piston force.

\[ K_A = I_a - k_{f5} l_2 \cos \theta - k_{f6} l_2 \sin \theta \]

\[ K_B = - k_{f6} l_2 \cos \theta - k_{f5} l_2 \sin \theta \]

\[ K_C = 0.5 F_p l_2 \tan \alpha \cos \theta + 0.5 F_p l_2 \sin \theta \]

\[ k_{f1} = m_i K_5 + 0.5 m_i K_9; \ k_{f2} = m_i K_6 + 0.5 m_i K_{10} \]

\[ k_{f3} = 0.5 k_{f1} \tan \alpha + 0.25 m_i K_9 \tan \alpha - \left( l_5 K_1 / l_3 \sin \alpha \right) + 0.5 m_i K_7 \]
Fig. 3 (a) — Geometrical parameters of first stage support

Fig. 3 (b) — Dynamic forces on first stage support system
In deriving the force relation, pressure in the cylinder is assumed constant throughout the piston travel. Eq. (1) has been solved for $\theta$ and $d\theta/dt$ by Runge–Kutta method from the initial values of $\theta$ and $d\theta/dt$ to the final value of $\theta$ corresponding to the initial and final positions of the arms using the known geometrical parameters.

**Experimental Methodology**

A test rig, developed to simulate opening function of the arms under different load conditions (Fig. 4), consists of two units fixed on the base. One side of FSS is installed and other consists of the loading arrangement, in which two sliders can be moved up and down along four guide rods. Six hydraulic cylinders are fixed on the sliders along two perpendicular directions for applying forces on the FSS through a loading frame. Two hydraulic cylinders have been provided at the base to lift the slider, and detach the loading frame from FSS. Sensors are attached at suitable locations to measure opening time. A hydraulic pack, consisting of pressure control, flow control, solenoid operated direction control valve etc. have been provided to

\[
k_{j4} = 0.5 \, k_{j2} \tan\alpha + 0.25 \, m_p K_{j0} \tan\alpha + (l_1 K_2)/(l_3 \cos\alpha) + 0.5 \, m_t K_8
\]

\[
k_{j5} = k_{j3} - m_t K_7; \quad k_{j6} = k_{j4} - m_t K_8
\]

\[
K_1 = l_2 \cos\theta / (l_3 \cos\alpha);
\]

\[
K_2 = l_2 \sin\theta / (l_3 \cos\alpha) - l_2^2 \tan\alpha \sec^2\alpha \cos^2\theta / l_3^2
\]

\[
K_3 = -l_2 \sin (\phi - \delta); \quad K_4 = -l_2 \cos (\phi - \delta)
\]

\[
K_5 = -l_2 \sin\theta - 0.5 \, l_3 \, K_1 \sin\alpha
\]

\[
K_6 = -l_2 \cos\theta - 0.5 \, l_3 \, K_1^2 \cos\alpha + 0.5 \, K_2 \, l_3 \sin\alpha
\]

\[
K_7 = 0.5 \, l_3 \, K_1 \cos\alpha; \quad K_8 = -0.5 \, l_3 \, K_1^2 \sin\alpha - 0.5 \, l_3 \, K_2 \cos\alpha
\]

\[
K_9 = -l_2 \sin\theta - l_3 \, K_1 \sin\alpha
\]

\[
K_{10} = -l_2 \cos\theta - l_3 \, K_1^2 \cos\alpha + l_3 \, K_2 \sin\alpha
\]
apply loads of desired magnitude on the unit. The pneumatic cylinder of FSS is connected to an air compressor to apply pneumatic pressure on the system. An accumulator between the compressor and FSS reduces the variation of air pressure inside the cylinder during the movement of the piston rod.

The loading frame is connected to the FSS. Desired loads, mainly forces and moments are applied through hydraulic cylinders in two perpendicular directions separately or simultaneously. The opening of the arm is initiated by sliding the slider vertically, consisting of load bar, by hydraulic cylinders.

Results and Discussion

In order to estimate the opening time theoretically, the following values have been used: \( l_1 = 0.705 \text{ m}, \ l_2 = 0.2 \text{ m}, \ l_3 = 0.611 \text{ m}, \ h_1 = 0.095 \text{ m}, \ h_2 = 0.203 \text{ m}, \ m_t = 4.8 \text{ kg}, \ m = 37 \text{ kg}, \ m_p = 7.5 \text{ kg}, \ I_a = 6.03 \text{ kg-m}^2, \ I_t = 0.196 \text{ kg-m}^2 \). These values have been obtained from the nominal dimensions of the designed system and subsequently checked with those of the fabricated unit. The components of FSS have been modelled in I-DEAS, a CAD based 3-d modelling software. The values of mass and moment of inertia of the relevant parts have been obtained from the solid models of these components. The solution has been made from the initial values of \( \theta = 112^0 \) and \( \frac{d\theta}{dt} = 0 \) at \( t = 0 \) to final value of \( \theta = 11^0 \) corresponding to desired opening angle under different pneumatic pressures.

The opening times under different pneumatic pressures have been experimentally measured for each pneumatic pressure (Table 1). Four different trials for a certain pressure have been carried out on a simulated test rig. The opening time that is measured after detachment of the FSS from the vehicle is independent of any load on the unit. The variations (+10 %) among theoretical and experimental values are due to the following reasons: a) Difference in theoretical and actual weights and moments of inertia of different components; b) Inaccuracy in sensor settings at proper locations during experimental trials; c) Variation of pneumatic pressure in the cylinder during travel of the piston; d) Friction in the hinges of the actual system; and e) Deviations in dimensions and symmetry of the fabricated unit from the ideal system.

The fabricated unit consists of different components but some small components have not been accounted in the solid model. So, theoretically estimated moments of inertia of different parts differ from the corresponding fabricated parts. Every attempt has been made to ensure the same setting of the sensors during the trials. However, because of high speed of the unit, there were slight variations in the settings leading to differences in the identification of initial and final positions of the arms. In the theoretical formulation, pneumatic pressure has been considered constant throughout the travel. However, in reality, pneumatic pressure in the cylinder varies, but could not be recorded. Although rolling element bearings have been used in the fabricated unit, friction may exist in the hinges. In theoretical estimation, the same has not been considered. The exact symmetry of the unit during movement could not be ensured or monitored. Since the variations of opening time from the theoretical values are less than ±10 percent, the theoretical formulation has been considered satisfactory for further analysis of the system during design.

During the initial period of swing, arm moves slowly; the speed increases subsequently (Fig. 5) mainly because the transmission ratio of the system is low during the initial period of movement in comparison with that during the later part of travel. So, effective turning moment on the arms increases gradually during the motion. However, to avoid shock

<table>
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<th>Pneumatic Pressure bar</th>
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<th>Theoretical</th>
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<tr>
<td></td>
<td>(i) 247.0</td>
<td>239.0</td>
</tr>
<tr>
<td></td>
<td>(ii) 240.0</td>
<td>230.0</td>
</tr>
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<td></td>
<td>(iii) 266.3</td>
<td>222.0</td>
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<td></td>
<td>(iv) 234.0</td>
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<td>Mean 246.8</td>
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<td>7.0</td>
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<tr>
<td>7.5</td>
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<tr>
<td>8.0</td>
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<td>8.5</td>
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<td>9.0</td>
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load developed owing to stoppage of the arm at the end of the travel, the terminal velocity needs to be as low as possible. So, for information needed for design, the final angular speed, acceleration and the maximum force at the hinge have been evaluated (Table 2). The values are required for the selection of the shock absorber and estimation of the corresponding loads on the structure.

**Conclusions**

The comparison among theoretical and experimental values of opening time ensures validity of the theoretical formulation, which has been satisfactorily used for detailed investigation of the mechanism and estimation of different parameters required for design of different subsystems. The opening time for FSS, under investigation varies from 203 to 239 m sec corresponding to 9.0 bars and 6.5 bar pneumatic cylinder pressure respectively.

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### References


### Annexure

**Governing equation**

(a) The geometric of the unit leads to the following relations [Fig. 3 (a)]:

\[ \phi = \gamma + \theta \quad \quad h = h_2 - h_1 \]

and \( \sin \alpha = (h + l_2 \sin \theta)l_3 \)

\[
\frac{d^2 \alpha}{dt^2} = K_1 \frac{d^2 \theta}{dt^2} - K_2 \left( \frac{d\theta}{dt} \right)^2 \quad \text{...}(A1)
\]

\[ x_c = l_c \cos(\phi - \alpha), \quad y_c = l_c \sin(\phi - \alpha) \]

\[
\frac{d^2 x_c}{dt^2} = K_3 \frac{d^2 \theta}{dt^2} + K_4 \left( \frac{d\theta}{dt} \right)^2 \quad \text{...}(A2)
\]

\[
\frac{d^2 y_c}{dt^2} = K_3 \frac{d^2 \theta}{dt^2} - K_4 \left( \frac{d\theta}{dt} \right)^2 \quad \text{...}(A3)
\]

\[ x_c = l_2 \cos \theta + 0.5 l_3 \cos \alpha \]
\[
y_{c} = 0.5 \ l_{3} \sin \ \alpha - h
\]

\[
\frac{d^{2}x_{c}}{dt^{2}} = K_{5} \ \frac{d^{2}\theta}{dt^{2}} + K_{6} \left( \frac{d\theta}{dt} \right)^{2}
\]  
\ldots (A4)

\[
\frac{d^{2}y_{c}}{dt^{2}} = K_{7} \ \frac{d^{2}\theta}{dt^{2}} + K_{8} \left( \frac{d\theta}{dt} \right)^{2}
\]  
\ldots (A5)

\[
x_{b} = l_{2}\cos \ \theta + l_{3}\cos \ \alpha, \ y_{b} = -h
\]

\[
\frac{d^{2}x_{b}}{dt^{2}} = K_{9} \ \frac{d^{2}\theta}{dt^{2}} + K_{10} \left( \frac{d\theta}{dt} \right)^{2}
\]  
\ldots (A6)

where \( K_{1}, K_{2}, K_{3} \ldots \ldots K_{10} \) are defined in the text.

(b) Following relations are obtained by balancing the forces and moments on the members (Fig. 3b):

\[
F_{01x} + F_{21x} = m \ \frac{d^{2}x_{c}}{dt^{2}}
\]  
\ldots (A7)

\[
F_{01y} + F_{21y} = m \ \frac{d^{2}y_{c}}{dt^{2}}
\]  
\ldots (A8)

\[
F_{21x}l_{2}\cos \theta - F_{21x}l_{2}\sin \ theta = I_{h} \ \frac{d^{2}\theta}{dt^{2}}
\]  
\ldots (A9)

where, \( m \) is the mass of the arm assembly consisting of the link, crank and staybrace and \( I_{h} \) is its moment of inertia about its center of gravity.

Again,

\[
F_{12x} + F_{32x} = m_{l} \ \frac{d^{2}x_{c}}{dt^{2}}
\]  
\ldots (A10)

\[
F_{12y} + F_{32y} = m_{l} \ \frac{d^{2}y_{c}}{dt^{2}}
\]  
\ldots (A11)

\[
F_{23x} + 0.5F_{p} = 0.5m_{p} \ \frac{d^{2}x_{b}}{dt^{2}}
\]  
\ldots (A12)

\[
0.5F_{32x}l_{3}\sin \alpha + 0.5F_{32y}l_{3}\cos \alpha - 0.5F_{12x}l_{3}\sin \alpha
\]

\[-0.5F_{12y}l_{3}\cos \alpha = I_{l} \ \frac{d^{2}\alpha}{dt^{2}}
\]  
\ldots (A13)

where \( m_{p} \) is the mass of the moving piston and crosshead, \( m_{l} \) and \( I_{l} \) are the mass and moment of inertia about center of gravity of the turnbuckle respectively.

Using the relations of \( \frac{d^{2}\alpha}{dt^{2}}, \ \frac{d^{2}x_{c}}{dt^{2}}, \ \frac{d^{2}y_{c}}{dt^{2}}, \ \frac{d^{2}x_{b}}{dt^{2}} \), and \( \frac{d^{2}y_{b}}{dt^{2}} \) in terms of \( \frac{d\theta}{dt} \) and \( \frac{d^{2}\theta}{dt^{2}} \) from Eqs (A1) to (A6) and rearranging the terms given in the Eqs (A7) to (A13), through mathematical manipulation, the governing equation of the system is given by,

\[
K_{A} \ \frac{d^{2}\theta}{dt^{2}} + K_{B} \ \frac{d\theta}{dt} + K_{C} = 0.
\]

where \( K_{A}, K_{B} \) and \( K_{C} \) are defined in the text.