Design and Analysis Of Wing Deployment Mechanism

A Project Report

Submitted by

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in partial fulfillment for the award of the degree
Of

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"To design the Mechanism for deployment of the wing for an UAV. Wing is being considered in two pieces, port and star board wing. Wing is housed inside the fuselage. During the launch phase of UAV, the wing needs to the deployed in specified time. It involves the conceptual design and analysis during the deployment and sizing of related Mechanism."

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ABSTRACT

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To design the mechanism for deployment of the wing for an UAV. The Wing is being considered in two pieces, port and starboard wing. Wing is housed inside the fuselage. During the launch phase of UAV, the wing needs to the deployed in specified time. This report deals with Scissors type wing with two pieces one over the other with separate pivot points. Out of which two halves of wings hinged separately were selected due to its advantages over other configurations. The main advantage of having a separate hinge for each wing is early separation of two halves during deployment. The hinge points are separated by a distance of 150 mm. The hinge points are positioned symmetrically on either side of the fuselage axis. The mechanism is built to deploy the wings from aft to forward. Single pyro is used to impart required force to the wings for deployment. . It involves the conceptual design and analysis during the deployment and sizing of related mechanism.

ஆய்வுச்சுருக்கம்

இந்த ஆய்வானது யுஏவினுடைய இறக்கை விரிதலுக்கான இயங்கும் முறையை வடிவமைத்தலாகும். யுஏவி னுடைய இறக்கையானது இரண்டு பகுதிகளாக கருதப்படுகிறது. அவையாவன போர்ட் மற்றும் ஸ்டார் போர்டு ஆகும். இறக்கையானது ஃப்யூஸிலேஜினுள் பொருத்தப்பட்டிருக்கும். யுஏவிஐ செலுத்தும்பொழுது குறித்த நேரத்தில் இறக்கையானது விரிவடைய வேண்டும். இந்த ஆய்வு இரண்டு பகுதிகளை கொண்ட, ஒன்றின் மேல் ஒன்று அமைந்த, மற்றும் பிரிக்கப்பட்ட மையப்புள்ளிகள் கொண்ட கத்தரி வகை இறக்கையை பற்றியதாகும். வேறு அமைப்புகளை ஒப்பிட்டு பார்க்கும்பொழுது, இது தேர்வு செய்யப்பட்டது. தனித்தனியாக அமைக்கப்பட்டிருக்கும் இறக்கையின் முக்கிய லாபமானது அது செலுத்தும்போது முன்கூட்டியே விரிவடைவதாகும். ஹிஞ்ச் புள்ளிகளானது 150mm வித்தியாசத்தில் பிரிக்கப்பட்டிருக்கும். ஃப்யூஸிலேஜ் அச்சின் இருபக்கமும் ஹிஞ்ச் புள்ளிகள் சமமாக அமைக்கப்பட்டிருக்கும். இந்த இயங்கும் முறையானது இறக்கைகள் பின்பக்கம் இருந்து முன்பக்கமாக விரிவடையுமாறு அமைக்கப்பட்டிருக்கும். இறக்கை விரிய தேவைப்படும் விசை ஒற்றை பைரோவின் மூலம் செலுத்தப்படும். இறக்கை விரிவடையும்பொழுது ஆய்வு செய்யப்பட்ட வடிவங்கள் உட்படுத்தப்பட்டுள்ளது.

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SYMBOL	-	EXPLANATION
t	-	Time
θ	-	Angle
I	-	Inertia
α	-	Angular acceleration
T	-	Torque
F	-	Force
σ	-	Stress
Pcr	-	Buckling load
τ	-	Shear stress
E	-	Young's modulus
μ	-	Poisson's ratio
Y, δ	-	Deflection
W	-	Load applied
l	-	Length
d	-	Diameter
D	-	Mean diameter
C	-	Spring index
K	-	Wahl's stress factor
n	-	Number of turns
Pmax	-	Maximum pressure
UAV	-	Unmanned Air Vehicle
ADE	-	Aeronautical Development Establishment
PS	-	Port Side
SB	-	Star Board
FEM	-	Finite Element Analysis

CHAPTER 1

INTRODUCTION

1. INTRODUCTION

The study of mechanism involves their analysis as well as synthesis. Analysis is the study of motions and forces concerning different parts of an existing mechanism while synthesis involves the design of the different parts. In a mechanism, the various parts are so proportioned and related that he motion of one imparts requisite motions to the others, and the parts are able to withstand the forces impressed upon them. But the study of the relative motions of the parts is independent of the strength and absolute dimensions.

In a reciprocating engine, the displacement of the piston depends upon the lengths of the connecting rod and the crank. It is independent of the bearing strength of the parts or whether they are able to withstand the forces or not. Thus for the study of motions, it is immaterial if a machine part is made of mild steel, cast iron or wood. Also, it is not necessary to know the actual shape and area of cross-section of the part.

Thus to study the motions of different parts of a mechanism, the study of forces is not necessary and can be neglected.

1.1 ADE – HISTORY AND INFORMATION

ADE was established in Jan. 1959 at Bangalore as an integral part of the Defence Research and Development Organization (DRDO). Over the years ADE has developed expertise in several Aeronautical systems such as design and development of air-launched expendable target aircraft systems, aerial targets, flight simulation facility, pilot training simulators, reusable rocket pods, head up displays, electro-optic sensors for aircraft, Flight Control System (FCS) evaluation, Glass Fiber Reinforced Plastic (GFRP) Composite technology, Unmanned Air Vehicles (UAV) including Pilot-less Target Aircraft (PTA) and Remotely Piloted Vehicle (RPV).

ADE is a major contributor to the LCA development program in the areas of development of digital fly-by-wire controls, digital avionic controls and their integration.

The whole system of ADE is divided in various departments according to their research fields and establishments. The charters of duties of ADE as laid down by Ministry of Defence are as follows:

- To evolve aeronautical standards, specifications and their application and implementation.
- To evolve test procedure for evaluating new and prototype aircraft including helicopters, equipment and aircraft materials and to conduct such tests and trials as may be required. ADE would conduct there tests and trials in conjunction with the Resident Technical Office of the Directorate of Aeronautics where these offices have been established.
- To undertake research and development for improvement of safety, performance, reliability of aircraft including helicopters and their equipment.
- To design and develop such items of specialized aeronautical equipment as may be assigned to it from time to time.

Although the charter of duties covers a wide field of activities in aeronautics, the present emphasis is mainly on undertaking research and development projects either on the basis of expressed requirements of the Defence Service (Staff Projects). Therefore keeping all these views in mind, the concentration of ADE has been on the following thrust areas:

- Targets, Drones and RPVs.
- Flight Simulation for R & D and development of simulators.
- Flight Research.
- Air armament.

1.2 KINEMATICS

It deals with the relative motions of different parts of a mechanism without taking into consideration the forces producing the motions. Thus it is the study, from a geometric point of view, to know the displacement, velocity and acceleration of a part of

1.3 DYNAMICS

It involves the calculations of forces impressed upon different parts of a mechanism. The forces can be either static or dynamic.

1.4 MECHANISM AND MACHINE

If a number of bodies (usually rigid) are assembled in such a way that the motion of one caused constrained and predictable motions to the others, it is known as a mechanism. Thus, mechanism transmits and modifies a motion.

A machine is a mechanism or a combination of mechanisms which, apart from imparting definite motions to the parts, also transmits and modifies the available mechanical energy into some kind of desired work. It is neither a source of energy nor a producer of work but helps in proper utilization of the same. The motive power has to be derived from external sources.

A slider-crank mechanism covert the reciprocating motion of a slider into a rotary motion of the crank or vice-versa. However, when it is used as an automobile engine by adding valve mechanism etc. it becomes a machine which converts the available energy (force of the piston) into the desired energy (torque of the crank-shaft). The torque is used to move a vehicle.

1.5 RIGID AND RESISTANT BODIES

A body is said to be rigid if under the action of forces, it does not suffer any distortion or the distance between any two points on it remains constant.

Resistant bodies are those, which are rigid for the purposed they have to serve. Apart from rigid bodies, there are some semi-rigid bodies, which are normally flexible, but under certain loading conditions act as rigid bodies for the limited purpose and thus are resistant bodies. A belt is rigid when subjected o tensile forces. Therefore, the belt in a belt-drive acts as a resistant body. Similarly, fluids can also acts as resistant bodies when compressed as in case of hydraulic press. For some purposes, springs are also resistant bodies. These day resistant bodies are usually referred to as rigid bodies.

1.6 LINK

As mechanism is made of a number of resistant bodies out of which some may have motions relative to the others. A resistant body or a group of resistant bodies with rigid connections preventing their relative movement is known as a link. A link may also be defined as a member or a combination of members of a mechanism, connecting other members and having motion relative to them. Thus a link may consist of one or more resistant bodies. A slider-crank mechanism consists of four links: frame and guides, crank, connecting-rod and slider. However, the frame may consist of bearings for the crankshaft. The crank link may have crankshaft and flywheel also, forming one link having no relative motion of these.

A link is also known as kinematic link or element.

Links can be classified into binary, ternary, quaternary, etc. depending upon its ends on which revolute or turning pairs can be placed. The links are rigid links and there is no relative motion between the joints within the link.

a. Binary Link
b. Ternary Link
c. Quaternary Link

1.7 KINEMATIC PAIR

A kinematic pair or simply a pair is a joint of two links having relative motion between them. In a slider-crank mechanism, link 2 rotates relative to link 1 and constitutes a revolute or turning pair. Similarly, links 2, 3 and 3, 4 constitute turnings pairs. Link 4 (slider) reciprocates relative to link 1 and is a sliding pair.

1.8 DEGREES OF FREEDOM

An unconstrained rigid body moving in space can describe the following independent motions

1. Translational motions along any three mutually perpendicular axes x, y and z, and
2. Rotational motions about these axes.

Thus a rigid body possesses six degrees of freedom. The connection of a link with another imposes certain constraints on their relative motion. The number of restraints can never be zero (joint is disconnected) or six (joint becomes solid).

Degrees of freedom of a pair is defined as the number of independent relative motions, both translational and rotational, a pair can have.

Degrees of freedom = 6 - Number of restraints.

1.9 CONNECTING ROD

The connecting rod is an intermediate link between the piston and the crankshaft. It transmits force from the piston to the crankshaft. They connecting rod converts the reciprocating motion of the piston to rotary motion of the crank shaft

It has

i. An eye at the small end to accommodate piston pin bearing
ii. A long shank bearing usually of I-Section and
iii. A big end opening that is usually split to take the crank pin bearing shells the length of the connecting rod is usually kept 3 to 4.5 times the crank radius.

The materials for connecting rod ranges from mild or medium carbon steels to alloy steals in industrial engines, carbon steal with ultimate tensile strength 50 to 670 N/mm^2 is used Example: Manganese Steel In aero engines, nickel chrome steels having ultimate tensile strength of about 940 to 1350 N/mm^2 is used Connecting rods are Mostly manufactured by drop forging.

1.10 PISTON:

The piston is a disc which reciprocates with in a cylinder .it is either impulse from expanding gases and transmit force to a crankshaft via the connecting rod.A well designed piston has enormous strength and heat resistance properties withstand high gas pressures and inertia forces. Material of the piston must posses to good wearing qualities so that the piston is able to maintain the surface hardness up to the operating temperature and the weight should be minimized to reduce the inertia due to reciprocating parts.

1.11 SPRINGS

A spring is an elastic number which defects under the actions of load and regions its original shape after the load is removed springs are quit commonly used in Automobiles, railway wagons, valves, watches, etc.

Springs are usually required to perform the following functions.

i. To cushion or reduce the effect of stock or impact loading
ii. To store energy example: Clocks, toys circuit breakers and starters.
iii. To apply force and to control motions as in backers as in brakes and Clutches
iv. To control motion by maintaining contact between two elements as in Cam and Followers.

1.12 CAMS

A cam is a mechanical member used to impart desired motion to a follower by direct contact. The cam may be rotating or reciprocating whereas the follower may be rotating, reciprocating or oscillating. Complicated output motions which are otherwise difficult to achieve can easily be produced with the help of cams .cams are widely used in automatic machines, internal combustion engines , machine tools , printing control mechanisms and so on . They are manufactured usually by die-casting, milling or by punch presses.

1.13 COMPUTER- AIDED ANALYSIS OF MECHANISM

The analysis of the velocity and the acceleration depend upon the graphical approach and are suitable for finding out the velocity and the acceleration of the links of a mechanism in one or two positions of the crank. To draw velocity and acceleration diagrams again and again for different positions of the crank is not very convenient. In this project, analytical expressions for the displacement, velocity and acceleration in terms of the general parameters have been derived. A desk-calculator or a digital computer facilitates the calculation work.

1.14 VELOCITY ANAYSIS:

For a proper study of the motions of the different parts of a machine one needs to know their velocities and accelerations at different moments. To facilitate such study, a machine or a mechanism is represented by a skeleton or a line diagram, commonly known as a configuration diagram.

Velocities and accelerations in machine can be determined either analytically or graphically. With the invention of calculator and computers, it has become convenient to make use of analytical methods.

1.15 ACCELARATION ANALYSIS

Velocity of a moving body is a vector quantity having the magnitude and the direction. A change in the velocity requires any of the following conditions to be fulfilled:

A change in the magnitude only.

A change in the direction only.

A change in both direction and magnitude.

The rate of change of velocity with respect to time is known as acceleration and acts as in the direction of the change in velocity. Thus acceleration is also a vector quantity.

1.16 STATIC FORCE ANALYSIS

In the design of machine mechanisms, it is imperative to know the magnitudes as well as the directions of forces transmitted from the input to the output. The analysis helps in selecting proper sizes of the machine components to withstand the stress developed in them. If proper sizes are not selected, the components may fail during the machine operation. On the other hand ,if the members are designed to have more strength than required, the machine may not be able to compete with others due to more cost ,weight ,size , etc.,

If components of a machine accelerate, inertia forces are produced due to their masses. However, if the magnitudes of these forces are small compared to the externally applied loads, they can be neglected while analyzing the mechanism. Such an analysis is known as static-force analysis.

1.17 DYNAMIC FORCE ANALYSIS

Dynamic forces are associated with accelerating masses. As all machines have some accelerating parts, dynamic forces are always present when the machines operate. in situations where dynamic forces are dominant or comparable with magnitudes of external forces and operating speeds are high, dynamic analysis has to be carried out. For example, in case of rotors which rotate at speeds more than 80 000rpm, even the slightest eccentricity of the centre of mass from the axis of rotation produces very high dynamic forces. This may lead to vibrations, noise or even failure.

CHAPTER 2

LITERATURE SURVEY

2. LITERATURE SURVEY

Based on a recent formulation for parametric design of mechanical systems using kinematic analysis, *Karim Abdel-Malek. Jingzhou Yang(5)* reported an approach to automated design by converting mechanical part geometry into a mechanism by breaking the part into sub-parts attached by higher pair joints. In order to evaluate the performance of a mechanism where a design change has been introduced into one of its links, we extend the formulation in this paper to address the propagation of this change and its effect on the velocity of other members. This analysis will limit itself to mechanisms. Velocity propagations due to modifications in the design of a mechanical part are analytically addressed using a proposed working coordinates formulation based on the cut-joint kinematic constraint method (presented earlier). Velocity vectors in state-vector notation are derived and treated as variables such that propagations of working coordinates of a link are obtained with respect to position and orientation. The underlying theory is presented and several planar and spatial mechanisms are treated.

I. GOUDAS, I. STAVRAKIS, and S. NATSIAVAS (3) have reported that Transient and steady state dynamic response of a class of slider-crank mechanisms is investigated. Specifically, the class of mechanisms examined involves rigid members but compliant supporting bearings. Moreover, the mechanisms are subjected to non-ideal forcing. Namely, both the driving and the resisting loads are expressed as a function of the angular coordinate describing the crank rotation. First, an appropriate set of equations of motion is derived by applying Lagrange's equations. These equations are strongly nonlinear due to the large rigid body rotation of the crank and the connecting rod, as well as due to the nonlinearities associated with the bearing action and the form of the driving and the resisting loads. Consequently, the dynamics of the resulting dynamical system is examined by solving the equations of motion numerically. More specifically, transient response is captured by direct integration, while determination of complete branches of steady state response is achieved by applying appropriate numerical methodologies. Initially, mechanisms whose crankshaft is supported by bearings with rolling elements and linear stiffness characteristics are examined. Then, numerical results are presented for

focused on mechanisms supported by hydrodynamic bearings. In all cases, the attention is focused on investigating the influence of the system parameters on its dynamics. Moreover, models with constant crank angular velocity are first analyzed, since they provide valuable insight into some aspects of the system dynamics. Eventually, the emphasis is shifted to the general case of non-ideal forcing, originating from the dependence of the driving and the resisting moments
on the crankshaft motion.

JO¨ RG WAUER and PETER BU¨ HRLE (4) has told that the dynamic response of a slider crank mechanism with a flexible connecting rod driven by an electric DC motor is examined. After formulating the governing nonlinear boundary value problem, it is reduced by a one-term truncation to a system of nonlinear ordinary differential equations representing the complete dynamics of the electromechanical system. First, the steady-state behavior is analyzed. It turns out that a constant crank rotational speed is not possible: fluctuations appear which can be limited by an appropriate choice of the system and the engine data. Then, the transient startup or rundown is dealt with by using a digital simulation of the model equations. It exhibits a rich variety of dynamic effects till the stationary speed is reached. Attention is focused on the influence of the flexibility which in both categories of motion yields typical features.

ROBERT KROYER (6) in his paper discussed container launched & airborne some wings have to be folded or deployed for a minimum storage capacity. After leaving the container, the wing will be deployed to achieve the full aerodynamic performance and stability. The wing folding and deployment process is here in simulated by a finite element analysis using ADINA. The static, dynamic and strength behavior the wing mechanism is discussed for a container launched missile.

With the help of the above literature the wing deployment mechanism has been designed using the inline slider crank mechanism.

CHAPTER 3

WING DEPLOYMENT MECHANISM

3. WING DEPLOYMENT MECHANISM

3.1 INTRODUCTION

To design the mechanism for deployment of the wing for an UAV. Wing is being considered in two pieces, port and starboard wing. Wing is housed inside the fuselage. During the launch phase of UAV, the wing needs to the deployed in specified time. It involves the conceptual design and analysis during the deployment and sizing of related mechanism.

3.2 CONFIGURATION CONSIDERED FOR WINGS

Wing deployment can be achieved in two ways with mono wing and other with using two Pieces. This report deals with Scissors type wing with two pieces one over the other with separate pivot points. Out of which two halves of wings hinged separately were selected due to its advantages over other configurations. The main advantage of having a separate hinge for each wing is early separation of two halves during deployment. Other advantage being, possibility of having straight wing mounts, lower angle of rotation for deployment and reduced force requirement for deployment.

In scissor type the wing is made in two parts(Wing-PS & Wing-SB) and swings on an axis similar to common scissor in the particular case being considered, the wing are made to made to swing on different hinges due to the inherent advantages mentioned above. The hinge points are separated by a distance of 150 mm. The hinge points are positioned symmetrically on either side of the fuselage axis. The mechanism is built to deploy the wings from aft to forward. Single pyro is used to impart required force to the wings for deployment. Modular construction method is used in designing for mechanism. The deployment mechanism is built on single horizontal plate (Holder) mounted below the bulkheads, thus eliminating the multiple references. Before deployment the wings are stowed in the belly of fuselage, one above the other, just below the fuel tank. After firing the pyro, which is again mounted on the Holder on which wings are mounted, the wings rotate on hinge axes and reach flying configuration. While deployment, the Wing-PS retains its level, whereas the Wing-SB moves downwards to be wings) and Guide (at Bottom of wing) forming a 'C' frame. Upon deployment, Wing-PS, Wing-SB along with Holder in the Top and Guide in the bottom becomes single beam connected directly to the bulkheads. In the final phase of deployment, the Bottom plate (Guide) guides the Wings in to slot in the leading edge side. The spring loaded locks are mounted on the Bottom plate for locking the wings in position after deployment. Prior to deployment and during the deployment, Wing-SB is supported by a compression spring from the bottom. However, once deployed the Guide plate supports the wing-SB rigidly from the bottom. Pyro is connected to the wings by link rod with spherical bearing at the interface. Once deployed, the position of the wings is retained by engaging a slot towards a leading edge and a "spring loaded" lock at immediate aft of the trailing edge. Shock absorbers (dampeners) are used to absorb the Kinetic energy of the wings being deployed. In designing the mechanism, care had been taken to avoid clashing of the wings with each other and also with other parts of the Airframe while deploying.

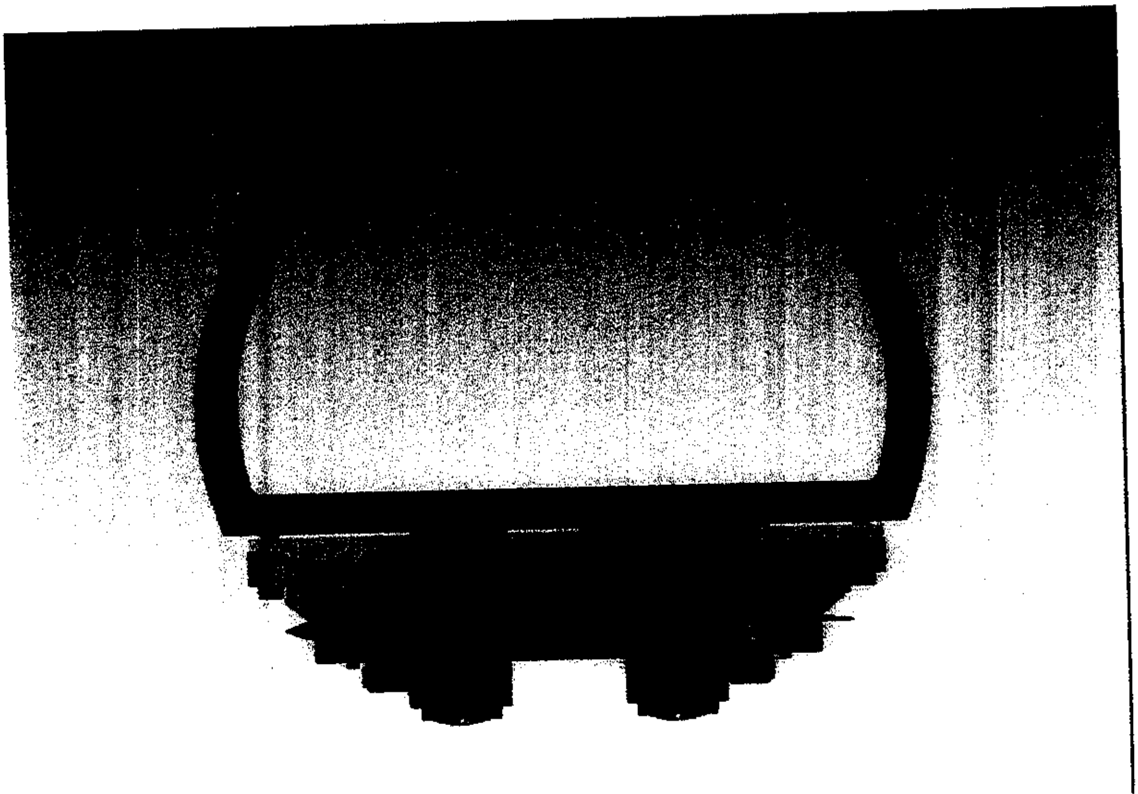


Fig 3.1 Stacking of Wing halves in the Fuselage prior to Deployment – Rear View

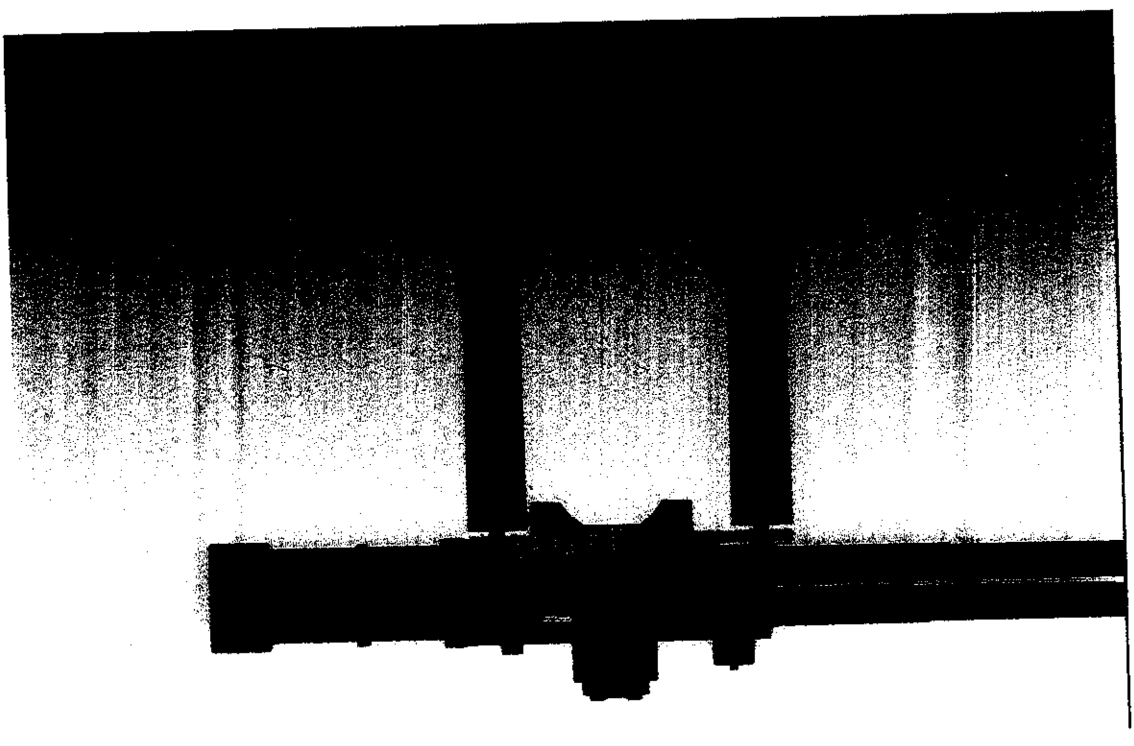


Fig 3.2 Stacking of Wing halves in the Fuselage prior to Deployment – Side View

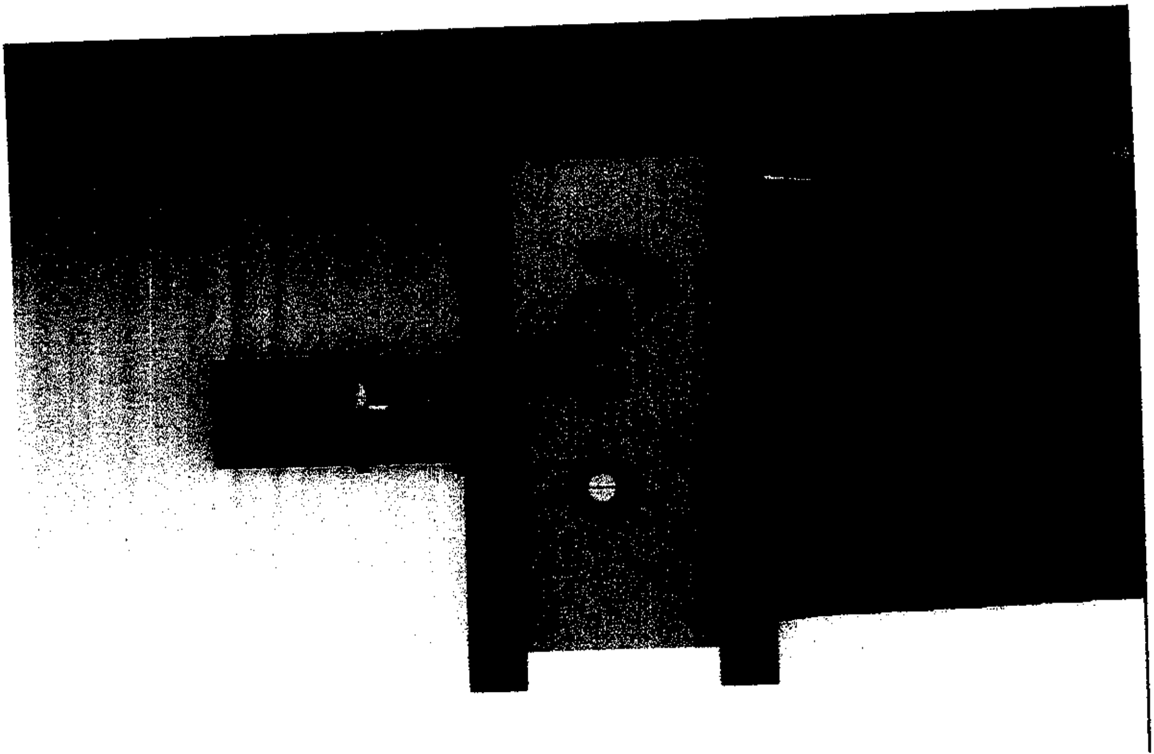


Fig 3.3 Stacking of Wing halves in the Fuselage prior to Deployment – Top view

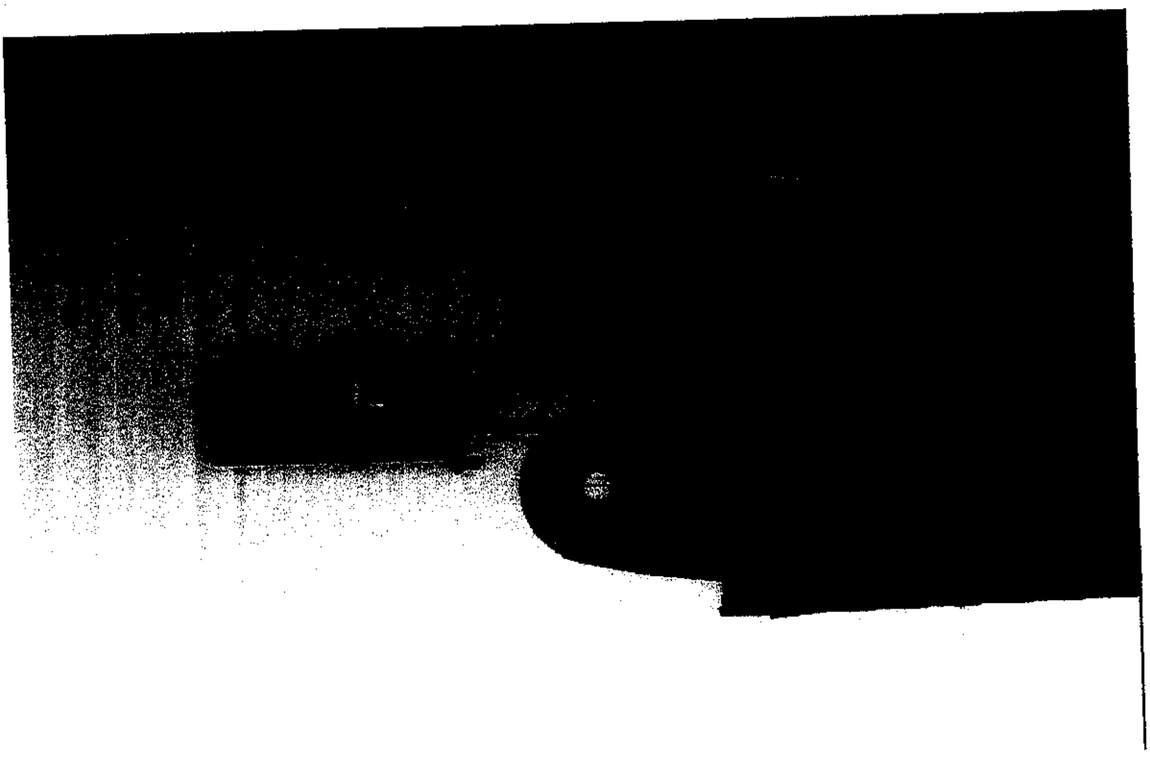


Fig 3.4Arrangement of the Pyro & Connecting Rods Prior to deployment

Fig 3.5 Wing Deployment in progress

(Please Note: Wing-SB is being pushed down with the help of Cam)

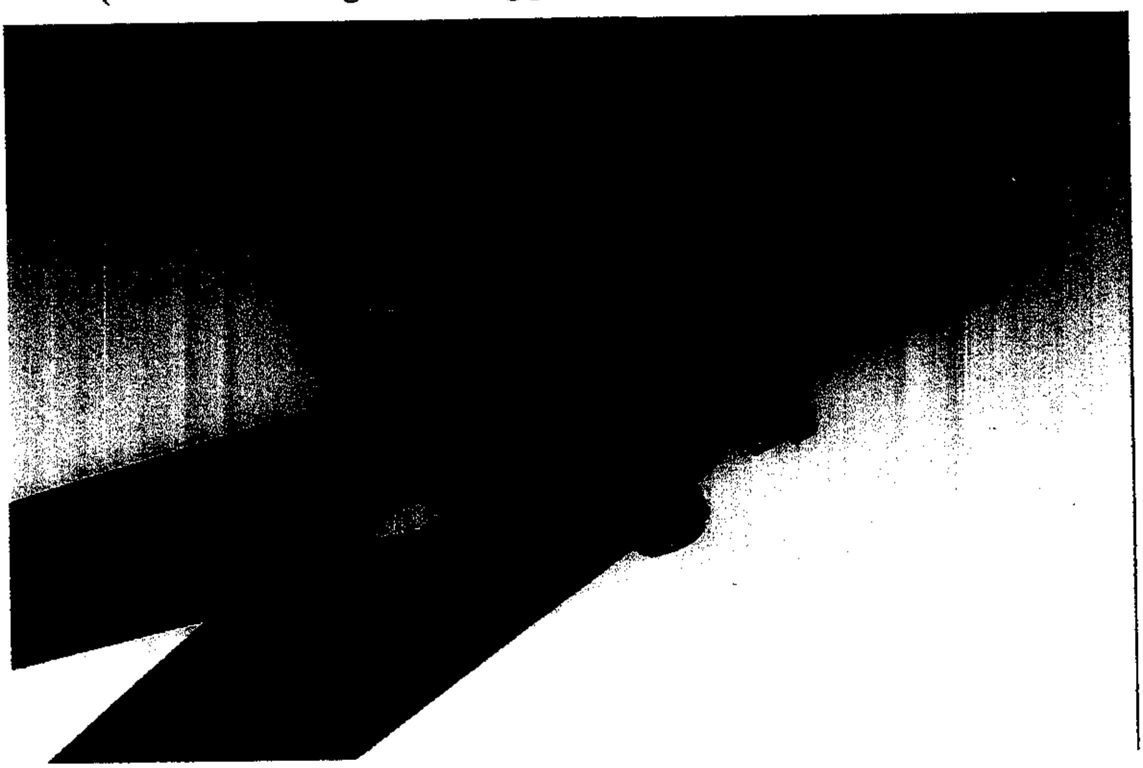


Fig 3.6 Wing-SB supported by the spring in stacked condition & during deployment.

CHAPTER 4

ESTIMATION OF THRUST REQUIRED FOR DEPLOYMENT OF WINGS

4. ESTIMATION OF THRUST REQUIRED FOR DEPLOYMENT OF WINGS

The wings are to be deployed with the help of Pyro & compression spring fitted on the same platform (Holder) where wings are mounted. From Pro/E software, it is observed that the stroke length required to deploy the wings is about 85 mm. The wings are required to be rotated by an angle of 82° for deployment.

Pyro piston Stroke length required = 85 mm

Weight of wing = 36 kg (18+18)

Wing deployment time = 0.5 seconds

Wing Inertia (I) I = 9.31 kg-m^2 (from Pro-E)

Wing Mass (one side) = 18 kg (from Pro-E)

CG dist from Pivot = 609 mm

4.1 ESTIMATION OF ANGULAR ACCELERATION (a) REQUIRED

Assuming 0.25s for wing acceleration and 0.25s for deceleration for wing rotation

Total angle of rotation required = 82^o

$t = 0.25$ sec

$\theta = 41$ degrees

$\theta = \omega 1 * t + 0.5\ \alpha\ t^2$

$41 = 0 + 0.5 * \alpha * 0.25 * 0.25$

$\alpha = 41 / (0.5 * 0.25 * 0.25)$

$= 1312$ deg / sec^2 = 22.9 rad /sec^2

Torque Required T = I a = 9.31 * 22.9 /9.81

= 217.33 N-m

Initial angle = 2.5° in vertical plane & 6.0° in horizontal plane

Initial arm = 52.12 mm= .05212 m

Pyro force required for deploying one wing (F)

F*cos 6.0 * cos 2.5 * 0.05212 = 217.33 N-m

F*0.9945 * 0.9990 * 0.05212 = 217.33 N-m

$F = 4200$ N (for deploying one wing)

Force required to deploy two wings = 4200 * 2 = 8400 N.

Angular velocity at the end of 0.25 seconds

Initial velocity = $\omega 1 = 0$

Velocity after 0.25 sec = $\omega 2$ =?

Angle of rotation $\theta = 41.0$ degrees = 0.72 radians

$\omega 2^2 - \omega 1^2 = 2. \alpha \theta$

$\omega 2^2 - 0 = 2 * 22.9 * 0.72 = 32.976$

$\omega 2 = 5.74$ rad / sec = 329.0 degrees / sec (less than 60 RPM)

From the geometry of the airframe and wing, the piston will travel a distance of 42 mm for the angular motion of the wing by 41.0 degrees (from Pro/E). That is to say the pyro should develop on an average of 8400 N for a stroke length of 42.0 mm. After the travel of 42 mm stroke length, the wings achieve about 330 degrees / sec angular velocity and gains sufficient (experiments should be conducted for verification) kinetic energy to complete the deployment of the wings. At the end of deployment, the kinetic energy of the wings is to be absorbed by providing suitable dampeners.

CHAPTER 5

STRESS ANALYSIS

5. STRESS ANALYSIS

Maximum force expected from the Pyro + spring = 10000 N. Due to uncertainties involved in pushing the Wing-SB down so that it is in plane with Wing-PS, shock load explected from the pyro, etc., the system is being designed for a force of 2000 kgf ≅ 20,000 Newtons.

5.1 PISTON ROD

Material – Steel

Piston Rod is subjected to compressive load.

Compressive Strength $\sigma = F/A = 20000/(\pi * 10^2/4)$

$= 254.6\ N/mm^2$

Buckling load (Considering one end is fixed and other is free end)

$I = \pi d^4/64 = \pi * 10^4/64$

$= 490.87\ mm^4$

Buckling load = $Pcr = \pi^2 EI/4L^2 = \pi^2 * 2.1 * 10^4 * 490.87/(4*(91^2))$

$= 30714$ N.

Compressive strength of the Piston rod is ($\pi d^2/4$ * allowable Compressive stress) is equal to $= \pi * 10^2/4 * 60 = 47123.8$ N.

Critical load carrying capacity (buckling load) 30714 N is more than expected load of 20000 N required for deployment.

5.2 PIN

Material –Steel En 24

Subjected to shear load

Shear stress = $\tau = F/2A$ (Pin is subjected to double shear)

$= 20000/(2 * \pi * d^2/4) = 20000/(2 * \pi * 6.35^2/4)$

$= 315.76\ N/mm^2$

5.3 CONNECTING ROD

Material - Steel En-24

Cross section = 12×12 mm.

$$I=\frac{bd^3}{12}=\frac{12\times12^3}{12}=1728mm^4$$

Area = 144 mm^2.

$$\text{Radius of gyration, } r=\sqrt{\frac{I}{A}}=\sqrt{\frac{1728}{144}}=3.46mm^4$$

$$\text{Slenderness ratio }=\frac{L}{r}=\frac{170}{3.46}=49.07$$

$$\text{Critical load, Pcr}=\frac{\pi^2 EI}{4L^2}=\frac{\pi^2\times21000\times1728}{4\times170^2}=30960\text{N.}$$

Actual load= 10000 N.

Factor of safety =1.5

$\therefore$ Load = 1.5×10000

=15000 N.

Bending load = 15000×sin 6^0

≈1600 N.

$$f=\frac{My}{I}=\frac{1600\times170\times6}{1728}=944N/mm^2$$

(For steel En-24 allowable direct stress = 1200 N/mm^2)

$$\text{Compressive stress }=\frac{L}{A}=\frac{15000}{12\times12}=104.2\ \text{N/mm}^2.$$

Maximum stress in the Connecting rod = 94.4+10.42

= 1048.2 N/mm^2

5.4 WING – MOUNT

Material- Al Alloy at the inter face with Pin

Crushing stress = $\frac{L}{db} = \frac{15000}{6.35 \times 7.5 \times 2} = 157.4$ N/mm^2

5.5 HOLDER

Material -Al Alloy

Holder is weak on starboard side as a cut out is made to accommodate the Cam. The shear area available around the pivot pin is

= π*D*t- (Cut surface area on Diameter D)

= π*96*10 – (2* 76.3 * 10) mm^2 (Dimension 76.3 mm is from Pro/E)

= 1490 mm^2

Shear strength of the Holder at the pivot pin = 1490 * 10

= 14,900 N.

The reaction to be transferred by the pivot pin should be limited to 14,900 N. However, this should be verified from FEM analysis owing to complex shape of the part involved.

5.6 PIVOT PIN

Material -Steel

Pivot pin along with Port spacer, Guide and Holder jointly supports the Wing-PS (Wing Mount). Air load on the wing is 700 kg-m = 700,000 kg-mm bending moment. All these members are to be analysed together. This joint is statically indeterminate structure and deflection should be considered for solving this joint.

Material of the Wing –Mount is Al Alloy

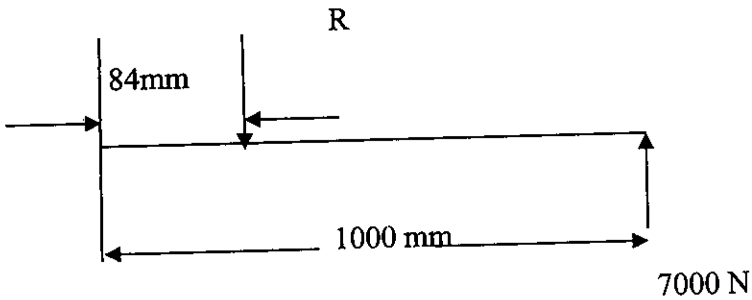
Elastic modulus E = 70000 N/mm^2

Poisson's ratio μ = 0.3 and

Shear modulus of the Material G = $\frac{E}{2(1+\mu)} = \frac{70000}{2(1+0.3)}$

= 27000 N/mm^2

Wing is assumed to be a propped cantilever fixed on pivot axis and propped at a distance of 84 mm from the pivot axis. The prop is an integral part of the Holder, a projection from the holder supporting the wing mount. The wing is subjected to a moment of 7000 N-m, is equated to a point load 7000 N applied at a distance of 1 meter from the pivot axis as shown in the following Fig.



Deflection at prop (at the support), assuming only 7000 N is acting at a distance of 1000 mm from the fixed end, in the absence of support is given by

$$\mathrm{E\,I\,y} = \mathrm{W}\left[\frac{lx^2}{2} - \frac{x^3}{6}\right]$$

Where,

E = young's modulus = 70000 N/mm^2

I = moment of inertia

y = deflection

l = length of the cantilever = 1000 mm

x = distance of the section from the fixed end = 84 mm

W = load applied = 7000 N

$\therefore$ The deflection at support due to applied load $y = \dfrac{W}{EI}\left[\dfrac{lx^2}{2} - \dfrac{x^3}{6}\right]$ ---------- (1)

When the support is present (the cantilever is propped) this deflection becomes zero. That is to say, the support should exert sufficient reaction (R) to get the same deflection as given by above equation (1).

$\therefore$ The deflection at support due to the reaction R, in the absence of applied load.

$$y = \frac{Rx^3}{3EI} \text{ ------------------------- (2)}$$

For the defection at support to be zero, the deflections obtained by equations 1 & 2 should be equal.

$$\frac{W}{EI}\left[\frac{lx^2}{2}-\frac{x^3}{6}\right]=\frac{Rx^3}{3EI}$$

By substituting W = 7000 N, l = 1000 mm, x = 84 mm

$$\frac{7000}{EI}\left[\frac{1000\times 84^2}{2}-\frac{84^3}{6}\right]=\frac{R\times 84^3}{3EI}$$

R = 12, 1500 N.↓ (The reaction from the support)

The reaction at pivot pin = 12,1500 – 7000 = 11,4500 N.↑

The moment at pivot pin M_p (assume it to be CCW) can be found by taking moments about fixed end.

M_p + (7000 * 1000) – (12,1500* 84) = 0

M_p = (12,1500* 84) - (7000 * 1000) = 320,6000 kg-mm (CCW)

Moment about free end should also be zero for the condition of equilibrium.

Moment about free end = 320,6000 – (11,4500* 1000) + (12,1500* 916) = 0

Maximum bending moment on the Pivot pin = 320,6000 N-mm

Pivot-Pin:

Material - Steel

Pin is subjected to a bending moment of 320,600 kg-mm and a tensile load of 11,4500 N due to the reaction transferred from the spacer-2.

Cross sectional Area = $\pi(d2^2 - d1^2)/4 = \pi(45^2 - 25^2)/4$ = 1099.6 mm^2

Inertia of the section = $\pi(d2^4 - d1^4)/64 = \pi(45^4 - 25^4)/64$ = 182114.2mm^4

Maximum Stress in Pivot (25/45) pin due to bending moment

= M * y / I

= 320,600 * 22.5 / 182114.2 = 396.1 N/mm^2

Direct stress due to Reaction of 11,4500 kgf. = 11,4500 /1099.6= 104.1 N/mm^2 (Tensile)

Net maximum stress = 396.1 N/mm^2 + 104.1N/mm^2

Pivot Pin is in shear at the interface with holder

Shear area = p * d * t = p * 45 * 5 = 706.86 mm^2

Shear Stress = Load / Shear area = 11,4500 / 706.86

= 162 N / mm^2

5.7 PROP

Material – Al Alloy

Prop is an integral part of the Holder. The stress in the prop because of the reaction developed in the Prop = reaction at prop / area of prop

= 12,1500 / 1710

= 71.1 N / mm^2

5.8 PORT SPACER

Material -Al Alloy

The stress in the Port Spacer because of the reaction at fixed end

= Reaction at fixed end / area of spacer = 11,4500 / 1570.8

= 72.9 N / mm^2

5.9 WING- MOUNT

Material - Al alloy

The mount is propped at a distance of 84 mm from the pivot axis. Cross sectional area of the mount at the prop is 6355.6 mm^2 and moment of inertia is 638586.52 mm^4 (from Pro/E). Maximum stress in the Mount - Wing at the prop (support) due to bending moment = M * y / I

= 7000 *(1000-84) * 17.5 / 638586.52

= 175.7 N / mm^2

5.10 GUIDE (BOTTOM PLATE)

Material -Al Alloy.

Guide is subjected to shear and compression as the reaction from the Wing – Mount is passed through the Guide to reach Bottom spacer-2 (Housing)

Shear strength:

Shear area = $\pi * 70 * 5 = 1099.6\ mm^2$.

Shear Load = 11,4500 N

(All the load coming on mount plate is to be transmitted by this member)

Shear stress = 11,4500 /1099.6

$= 104\ N/mm^2$

Area under compression = $\pi\ (70^2 - 60^2)/4$

$= 1021.0\ mm^2$.

Load = 11,450 (All the load coming on mount plate is to be transmitted by this member)

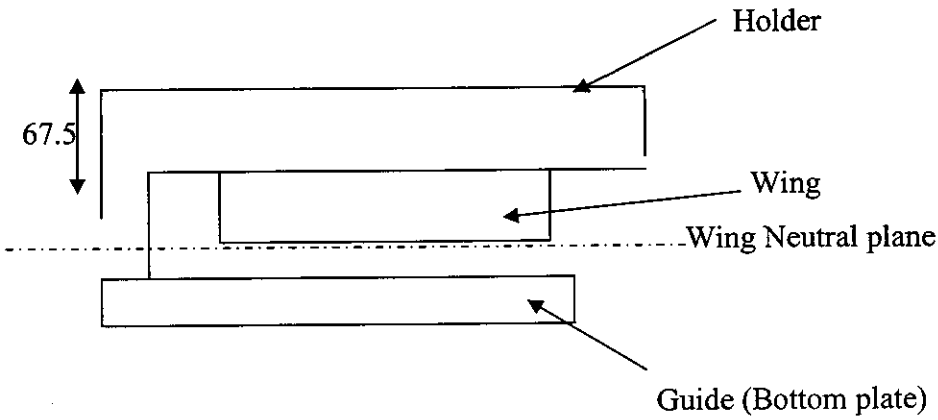
Stress (compression) = 11,4500 /1021.0

$= 112\ N/mm^2$

5.11 WINGS AS A BEAM (AFTER DEPLOYMENT)

Material- Al Alloy.

Wing as beam is subjected to a bending load of 7000 + 7000 = 14000 N-m. Maximum stresses occur at the farthest layer of the wing from neutral plane. This is the top layer of the Holder. The cross section of which is shown below.



Moment of inertia from Pro/E = 6,940,554 mm4 (Excluding wing)

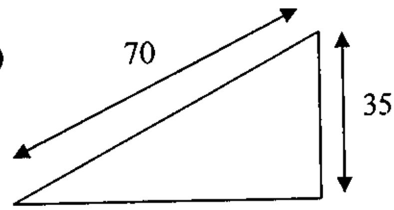
$f = \frac{My}{I} = \frac{14000000*67.5}{6940554} = 136.2 N/mm^2$

(For Al Alloy allowable stress = 270 N/mm^2)

5.12 CONTACT STRESSES AT THE INTERFACE OF CAM AND HOLDER

Curve length ≅ 70 mm (Diagonal of the triangle)

Height traveled = 35 mm (Height of the triangle)



∴ Angle of inclination $\sin^{-1}(35/70) = 30°$

Contact width is given by the formula (7)

$$b = \sqrt{\frac{2F\{[(1-v_1^2)/E_1]+[(1-v_2^2)/E_2]\}}{\pi l[(1/d_1)+(1/d_2)]}}$$

The maximum pressure is $p_{max} = \frac{2F}{\pi bl}$

Where,

b = half width of the contact (to be found)

F = load between contact surfaces

= 1000 N shared by 2 cam surfaces = (1000/2) * cos (60°) = 2500 N.

E_1, E2 = Elastic modulus 700 kg/mm^2

v_1, v_2 = Poisson's ratio = 0.3

d_1, d2 = diameter = 10, ∞

l = length of contact = 20 mm

By substituting above values, b = 0.455 mm

∴ Contact width = 2*b = 2 * 0.455 = 0.91 mm

Maximum pressure is $p_{max} = \frac{2F}{\pi bl}$

= 2*2500/ (π*0.455*20)

= 174.9 N / mm^2

Maximum shear stress = 0.3 p_{max} = 0.3 * 174.9 = 5.25 N / mm^2 and occurs at z = 0.75 * b = 0.75 * 0.455 = 0.34 mm below the surface.

5.13 PIN (LOCK)

Material- Al Alloy.

Shear area= 181.4 mm2 (from Pro/E) resulting shear strength of 181.4 * 15 = 2721 kg

Crushing area=109.1 mm^2 (from Pro/E) resulting com. strength of 109.1* 25 = 2727.5 kg

Minimum of above two values = 2721 kg

The pin is located at a distance of 50 mm away (along the span) from the hinge point.

The bending moment the pin can withstand = 2721* 50/1000= 136.05 kg-met.

(Data of Exact bending moment coming on the pin is not available)

5.14 SPRING FOR LOCK

Mass of the lock pin = 16.3 gms (from Pro/E) ≅ 20.0 gms

Distance to be ejected (s) = 18.0 mm

Time for ejection (t) = 0.02 sec

Acceleration = a =?

Initial speed (u) = 0

Form the equations of motion formula

$s = u*t + 0.5 * a * t^2$

$18 = 0 + 0.5 * a * 0.05^2$

$\therefore a = 18/(0.5 * 0.05 * 0.05)$

$= 14400 \text{ mm/sec}^2 = 14.4 \text{ met/sec}^2$

This is the acceleration required in addition to (against the) gravity of 9.81 met/sec^2

$\therefore$ Total acceleration required = 14.4 + 9.81

$= 24.21 \text{ met /sec}^2$

Force required = m * a = $\frac{20}{1000} \times 24.21 = 0.4842 \cong 0.5$ Newton

Spring is designed for 1.0 Newton

Mean dia of the spring (D) = 10.0 mm

(8) Spring index (C) $= D/d = 10/0.5 = 20$

Wahl's stress factor $K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4\times20-1}{4\times20-4} + \frac{0.615}{20} = 1.07$

Maximum shear stress $= \frac{k\times8\times W\times D}{\pi\times d^3} = \frac{1.07\times8\times0.1\times10}{\pi\times0.5^3}$

$= 218$ N/mm^2

Deflection $\delta = 30.0$ mm $= \frac{8\times W\times D^3\times n}{G\times d^4} = \frac{8\times1\times10^3\times n}{8100\times0.5^4}$

$\therefore$ Number of turns (n) $= \frac{30\times81000\times0.5^4}{8000}$

$= 18.98 \cong 19$ turns

Total number of Turns for squared & ground ends =19+2 = 21 turns

Free length of the spring = n*d + δ + 0.15δ

= (21*0.5) +27 + (0.15*27)

= 41.55≅ 42.0 mm

Pitch of the coil $= \frac{Freelength}{n-1} = \frac{42}{21-1} = \frac{41}{20} = 2.05 \cong 2.0$ mm

Mass of the spring (from Pro/E) = 1.0 gm = 0.001 kg

Stiffness of the spring = 100/30 = 3.33 gm/mm = 3.33 kg/met

= 3.33*9.81

= 32.7 Newtons /met

Natural frequency of the spring $= \frac{1}{4\pi}\sqrt{\frac{k}{m}} = \frac{1}{4\pi}\sqrt{\frac{32.7}{0.001}} =$ 14.4 cycles/sec

(Deployed length of the spring = 38 mm, Compressed length of the spring = 20 mm)

5.15 SPRING FOR DEPLOYMENT OF WINGS

Distance to be moved (s) = 50.0 mm, 7500 N

Mean dia of the spring (D) = 63.5 mm

Wire dia (d) = 12.7 mm

Spring index (C) $= D/d = 63.5/12.7 = 5$

Wahl's stress factor $K = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4\times5-1}{4\times5-4} + \frac{0.615}{5} = 1.3105$

$$\text{Maximum shear stress} = \frac{K \times 8 \times W \times D}{\pi \times d^3} = \frac{1.3105 \times 8 \times 7500 \times 63.5}{\pi \times 12.7^3}$$

$$= 776 \text{ N/mm}^2$$

$$\text{Deflection } \delta = 50.0 \text{ mm} = \frac{8 \times W \times D^3 \times n}{G \times d^4} = \frac{8 \times 7500 \times 63.5^3 \times n}{81000 \times 12.7^4} =$$

$$\therefore \text{Number of turns (n)} = \frac{50 \times 81000 \times 12.7^4}{8 \times 7500 \times 63.5^3}$$

$$= 6.858 = 7 \text{ turns}$$

Total number of Turns for squared & ground ends =7+2 = 9 turns

Free length of the spring = n*d + δ + 0.15δ

= (9*12.7) +50 + (0.15*50)

= 171.8 ≅172mm

$$\text{Pitch of the coil} = \frac{Freelength}{n-1} = \frac{172}{9-1} = \frac{172}{8} = 21.5 \text{ mm}$$

Mass of the spring (from Pro/E) = 1.576 kg

Stiffness of the spring = 750/50 = 15 kg/mm = 15 * 9.81*1000

= 147150 Newtons/ Met

$$\text{Natural frequency of the spring} = \frac{1}{4\pi}\sqrt{\frac{k}{m}} = \frac{1}{4\pi}\sqrt{\frac{1471500}{1.576}} = 77 \text{ cycles/sec}$$

(In the model the compressed length = 117.5, open length = 196.451)

5.16 SPRING FOR SUPPORTING THWE WING (S.B.):

Distance to be moved (s) = 36+14 = 50.0 mm,

Mean dia of the spring (D) = 50 mm

Wire dia (d) = 5 mm

Spring index (C) = D/d = 50/5 = 10

$$\text{Wahl's stress factor K} = \frac{4C-1}{4C-4} + \frac{0.615}{C} = \frac{4 \times 10 - 1}{4 \times 10 - 4} + \frac{0.615}{10} = 1.145$$

$$\text{Maximum shear stress} = \frac{k \times 8 \times W \times D}{\pi \times d^3} = \frac{1.145 \times 8 \times 500 \times 50}{\pi \times 5^3}$$

$$= 583.1 \text{ N/mm}^2$$

Deflection $\delta = 50.0$ mm $= \dfrac{8 \times W \times D^3 \times n}{G \times d^4} = \dfrac{8 \times 50 \times 50^3 \times n}{81000 \times 5^4} =$

$\therefore$ Number of turns (n) $= \dfrac{50 \times 81000 \times 5^4}{8 \times 500 \times 50^3}$

$= 6.858 = 5.0625$ turns

Total number of Turns for squared & ground ends =5+2 = 7 turns

Free length of the spring = n*d + δ + 0.15δ

= (7*5) +50 + (0.15*50)

= 92.5 mm

Pitch of the coil $= \dfrac{Freelength}{n-1} = \dfrac{92.5}{7-1} = \dfrac{92.5}{6} = 15.417 \cong 15.5$ mm

Mass of the spring (from Pro/E) = = 0.1434 kg

Stiffness of the spring = 500/50 = 10 kg/mm= 10 * 9.81*1000= 98100 N/ Met

Natural frequency of the spring $= \dfrac{1}{4\pi}\sqrt{\dfrac{k}{m}} = \dfrac{1}{4\pi}\sqrt{\dfrac{98100}{0.1434}} =$ 66 cycles/sec

(Compressed length of the spring should not exceed = 40 mm)

CHAPTER 6

POSITION ANALYSIS

6. POSITION ANALYSIS

(1) In the position analysis, the position of the wing is calculated in degrees for the proper motion of the wing. And the position analysis is done for the reason, so that we can know the wing has attain its flying configuration at the given time period and the force from the pyro actuator. Position analysis in machine can be determined either analytically or graphically. The following graph explains the position of the wing with respect to time.

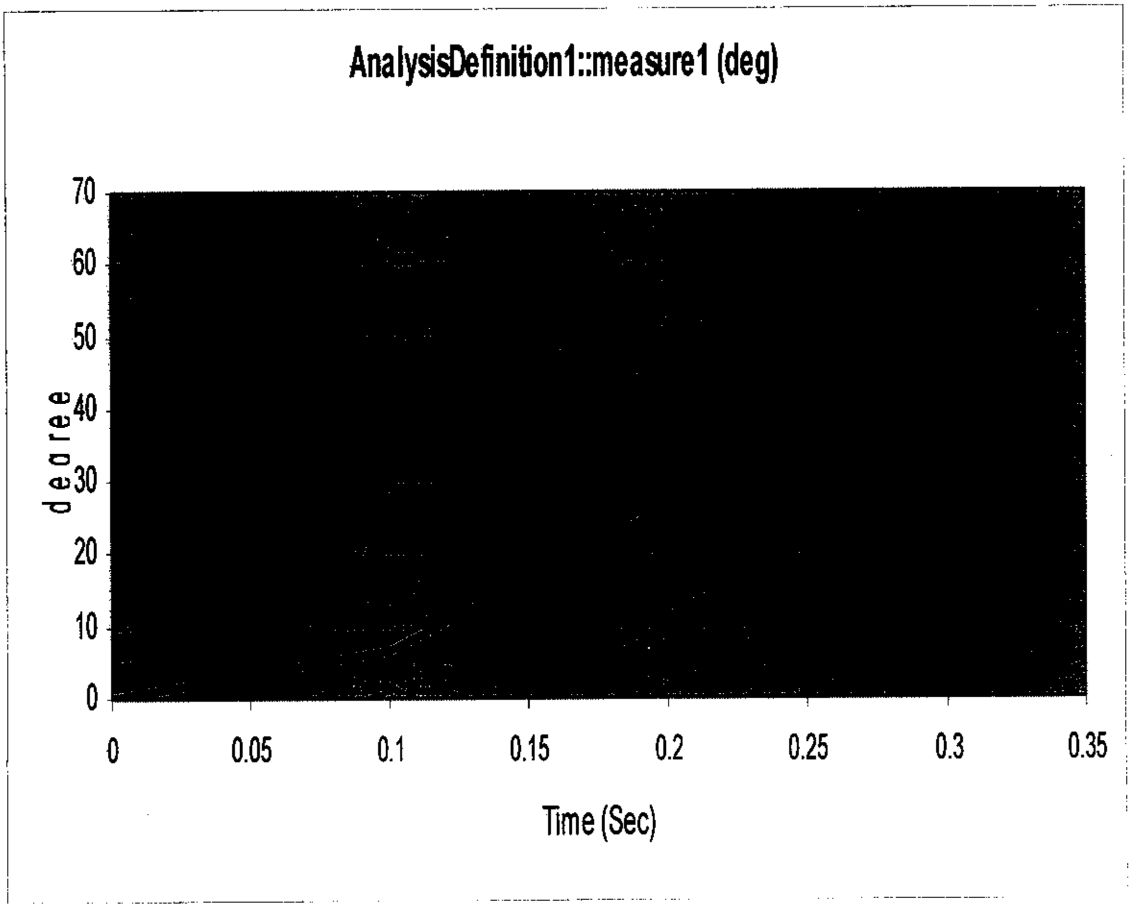


Fig 6.1 Position of wing with respect to time

Time (Sec)	Measure (deg)
0	0
0.025	1.64
0.05	3.28
0.075	4.92
0.100	6.56
0.125	11.48
0.150	16.4
0.175	21.32
0.200	26.24
0.225	34.44
0.250	42.64
0.275	50.84
0.300	59.04

Table 6.1 Position of wing in degrees with respect to time in seconds

CHAPTER 7

VELOCITY ANALYSIS

7. VELOCITY ANALYSIS

(2) For a proper study of the motions of the different parts of a machine one needs to know their velocities and accelerations at different moments. To facilitate such study, a machine or a mechanism is represented by a skeleton or a line diagram, commonly known as a configuration diagram.

Velocities and accelerations in machine can be determined either analytically or graphically. With the invention of calculator and computers, it has become convenient to make use of analytical methods.

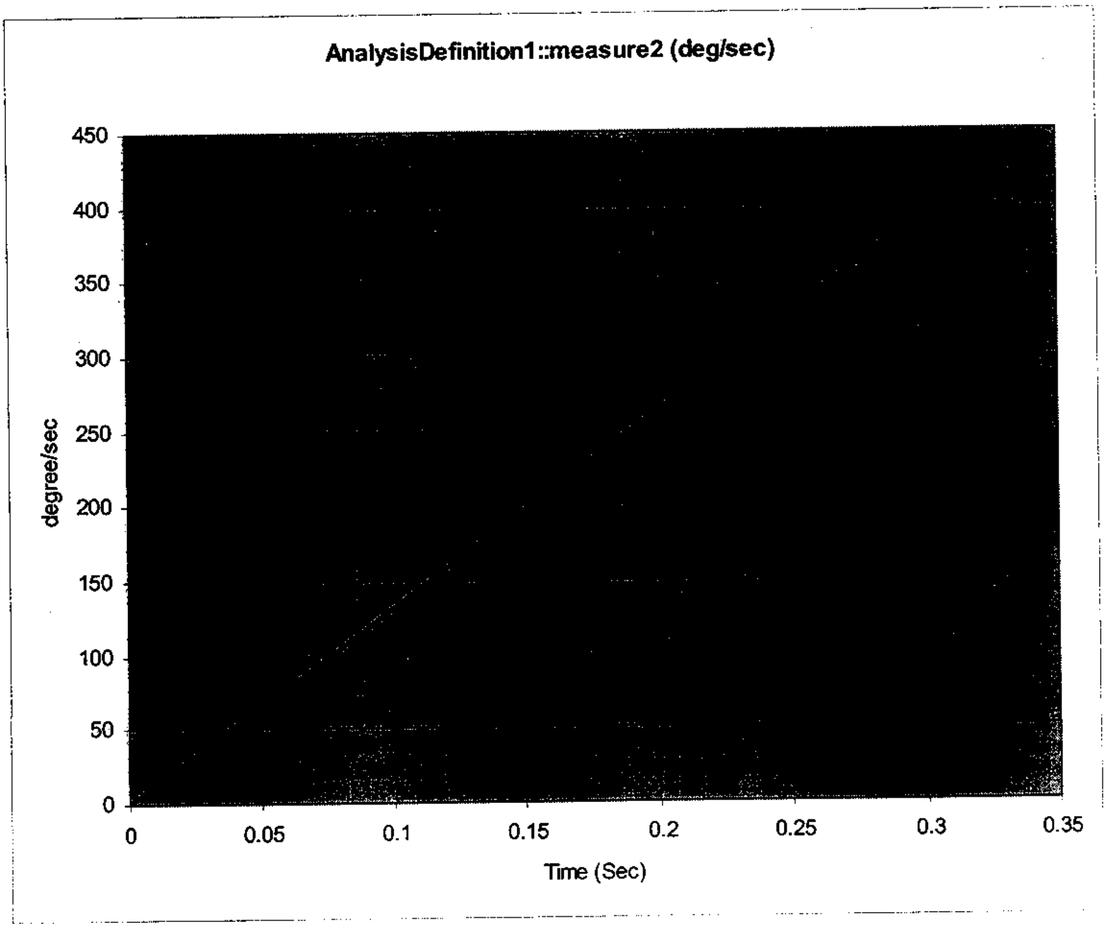


Fig 7.1 Variation of velocity with respect to time

Time (Sec)	Measure (deg/sec)
0	0
0.025	32.8
0.05	65.6
0.075	98.4
0.100	131.2
0.125	164.0
0.150	196.8
0.175	229.6
0.200	262.4
0.225	295.2
0.250	328.0
0.275	360.8
0.300	393.6

Table 7.1 Velocity variation of the wings during the velocity analysis

CHAPTER 8

RESULTS & DISCUSSION

8. RESULTS & DISCUSSION

Both the wings (PS & SB) are inter-connected during the deployment. This ensures the simultaneous deployment of the Wing-PS and Wing-SB.Cam is interfering with the fuel tank. To avoid this interference, a relief is to be made in the fuel tank. The kinetic energy of deploying wings has to be absorbed by dampeners placed suitably in the Holder to avoid shock loads on to the structure and also to avoid bouncing. Some of the dampeners being considered are Polyurethane foam embedded in flexible rubber pads, hallo lead blocks, lead blocks with a V notch, hallo aluminium blocks. The spring used for deployment exerts negative pressure in the final stages of deployment, thus reducing shock. Experiments are required to be conducted to finalise the suitable dampener.

Time available for the locks to eject is very small compared to time for wing deployment. Hence there may be a requirement of additional locks at the leading edge in addition to the Shock absorbers to prevent bouncing of the wing at the time of deployment.

The time allotted for acceleration is fifty percent of the deployment time and the remaining time is allotted for deceleration. In reality the wing will continue to move almost at the velocity gained at the end of 50% of time till it hits the Shock absorbers due to negligible decelerating force in the form of friction. So, the pyro force required may be less than the estimated 8400 N.

During deployment, the radial arm and angle of applied thrust continuously vary. The results arrived are only indicative. Further study is required. The same has to be validated by experiments.

Stress analysis is carried by Strength of Material (SOM) route. Optimisation based on FEM is likely to result in lighter deployment system.

CHAPTER 9

CONCLUSION

9. CONCLUSION

- The proposed Wing deployment mechanism, do not generate any couple (moment in horizontal plane) similar to mono wing.
- The impact on deployment is likely to result in forward thrust, which is not detrimental.
- If Scissors type wing is chosen, the axis of payload can be made to coincide with fuselage axis, which is not possible in mono wing configuration.
- Aerodynamic effect of Wing-SB moving down wards, with Wing-PS maintaining same level should be studied.
- The designed system is analysed in conventional method. The same should be analysed in FEM for optimisation. Especially the Holder & Wing mounts should be analysed in FEM for optimisation.
- Experiments should be conducted for finalising the shock absorbers and locking mechanism.

APPENDIX

WEIGHT SUMMARY OF THE COMPONENTS

Sl No	Nomenclature	Qty Nos.	Material	Weight / component Grams	Total weight Grams
1	Assy- Wing Cage	1			
2	BH-front	1	Al Alloy	2680	2680
3	Locating Pin-10x8	6	Steel	12.4	74.4
4	BH-rear	1	Al Alloy	2680	2680
5	Holder	1	Al Alloy	5673	5673
6	Pivot Pin	2	Steel	1288	2576
7	Port spacer	1	Al Alloy	194	194
8	Pyro holder	1	Al Alloy	479	479
9	Piston cap	1	Al Alloy	123	123
10	Guide	1	Al Alloy	1795	1795
11	Bush-Shear pin	2	Steel	9	18
12	Spring-Deployment	1	Steel	1576	1576
13	Socket head screw M10x50	2	Steel	43.5	87
14	Shock absorber	2	Rubber/foam/ Lead	Depends on choice of absorber	
15	Socket head screw M10x25	4	Steel	28	112
16	Socket head screw M6x25	4	Steel	8.5	34
17	Nut M6	4	Steel	3	12
18	Nut M6	1	Steel	3	3
19	Dowel Pin-5mm	2	Steel	4.5	9
20	Assy Pyro catridge	1	Assy		

Sl No	Nomenclature	Qty Nos.	Material	Weight / component Grams	Total weight Grams
21	Pyro cylinder	1		253	253
22		1		118	118
23	Bearing FMGV 4.3	4	Steel	16	64
24	Spl Bolt	1	Steel	13.6	13.6
25	Assy Wing – SB	1	Assy		
26	Wing SB	1	Assy		
27	Spl bolt	1	Steel	10	1
28	CAM	1	Al Alloy	304	304
29	Mount -Wing SB	1	Al Alloy	2080	2080
30	Assy link SB	1	Assy		
31	Link SB	1	Steel	145	145
32	Assy Wing PS	1	Assy		
33	Wing PS	1	Assy		
34	Spl. Bolt-2	1	Steel	10	10
35	Mount Wing Ps	1	Al Alloy	2080	2080
36	Assy Link PS	1	Assy		
37	Link PS	1	Steel	145	145
38	Spacer-2	2	Al Alloy	173	346
39	Spring (wing support)	1	Spring Steel	43	43
40	Nut M4	2	Steel	1	2

Sl No	Nomenclature	Qty Nos.	Material	Weight / component Grams	Total weight Grams
41	Socket head screw M4x8	4	Steel	2	8
42	Counter sunk screw M8x35	6	Steel	16	96
43	Helicoil inserts	6	Steel	2	12
44	Assy Lock	2	assy		
45	Body	2	Al Alloy	26.5	53
46	Pin	2	Al Alloy	16	32
47	Bracket	2	Al Alloy	12.5	25
48	Springs (Lock)	2	Steel	2	4
49	Spring (Support)	2	Steel	143	286
50	Arrester	2	Steel	2	4
				Total Weight =	24.2 kgs **
					(18.9 fixed, 5.3kgs moving)

** The weight includes Bulkheads (5360 gms), which are part of the fuselage. The weights are likely to reduce with optimisation of the components based on FEM analysis.

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