Design, Fabrication and Testing of Mooring Masts for Remotely Controlled Indoor and Outdoor Airships

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Received: 26 November 2013 / Accepted: 12 February 2016 / Published online: 4 March 2016
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Abstract This paper presents the design and structural details of two mooring masts, one for remotely controlled outdoor airships and another one for remotely controlled indoor airships. In a previous study, a mast for outdoor remotely controlled airship was designed to meet several user-specified operating requirements, and a simplified version of the same was fabricated. A spring loaded device was incorporated that sounds an alarm when the wind-loads exceed a threshold value, so that the airship can be taken indoors. The present study started with a critical analysis of that mast, and a new mast was designed and fabricated to remove several of its shortcomings. This mast consists of power screw operated telescopic module made of aluminium, mounted on a five legged base with castor wheels, for ease in mobility. Components of the existing mast were used to the possible extent, and the design was simplified to meet the assembly and transportation requirements. The spring mechanism used in alarming device was also modified to ensure higher sensitivity in the range of maximum expected wind-loads acting on the airship. A lightweight mooring mast for indoor remotely controlled airships was also designed and fabricated, which can accommodate non-

rigid indoor airships of length up to 5 m. The mast consists of an elevating bolt operated telescopic module mounted on a tripod adapter base, with lockable castor wheels, and has a specially designed mooring-clamp at the top. The various modules and components of the mast were designed to enable quick assembly and transportation.

Keywords Mooring mast · Indoor airships · Outdoor airships · Ground-handling · Yaw · Pitch · Roll

List of Symbols

L	Length of the airship envelope (m)
D	Maximum diameter of the airship envelope (m)
C_{DV}	Drag-coefficient based on the volume of the envelope
v	Velocity (m/s)
ρ	Density of air at the place of flight-test (kg/m^3)
μ	Coefficient of viscosity of air at the place of flight-test (Pa s)
V	Total volume of the envelope of airship (m^3)
F_{DE}	Drag force on the bare envelope (N)
F_D	Drag force on the airship (N)
S_{yt}	Yield-tensile strength in tension of the material under consideration (MPa)
S_{yc}	Yield-tensile strength in compression of the material under consideration (MPa)
S_{sy}	Yield-strength in shear of the material under consideration (MPa)
M	Bending moment (Nm)
T	Torque (or twisting moment) (Nm)
σ	Stress under consideration (MPa)
σ_b	Bending stress (MPa)
τ	Shear stress (MPa)

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σ_n	Normal stress (MPa)
σ_c	Axial compressive stress (MPa)
Z	Section modulus (m^3)
F.O.S.	Factor of safety
d	Diameter under consideration (mm)
σ_t	Tensile stress (MPa)
I	Area moment of inertia (m^4)
y	Perpendicular distance of the section under consideration from the neutral axis (mm)
HSS-1	Hollow Square Section-1 (in the mooring-mast for indoor airships)
HSS-2	Hollow Square Section-2 (in the mooring-mast for indoor airships)

Introduction

Airships are used for surveillance, demonstrations, product promotion and several other commercial and scientific applications. But, being buoyant and large sized, their ground-handling is somewhat problematic. An airship tends to have vertical oscillations while in the vicinity of ground. So, a ground-crew is needed to ground-handle it and this manual-handling has several drawbacks. An airship is very sensitive towards atmospheric disturbances due to its large surface-area and volume. Hence, an airship should not be completely constrained (using multi-point mooring) to move at ground in the presence of air-gusts, so as to avoid damages to its delicate envelope and large loads at the restraining attachments. Hence, for safe and secure ground-handling of an airship, a structure is proposed termed as “Mooring Mast”. In single point mooring, the airship is constrained only at its nose and is allowed to yaw, pitch and roll about this point. Such a mast enables limited three axes rotational/oscillatory motion to the airship in moored condition, substantially reducing the structural load due to atmospheric gusts and random turbulence. It is also desirable to have a wind-load measuring device on the mast (for outdoor airships), with an alarming system to provide an aural warning to the ground crew to move the airship and the mast inside the storage area when the atmospheric conditions are not suitable for outdoor storage. Mooring mast when employed for indoor airship facilitates us in safely carrying the airship indoors, in mounting and demounting of fins, any other components (e.g. a camera) on the airship safely, by giving the freedoms of yawing, pitching, rolling and height-adjustability to the moored airship. Howe [1] has described the ground-handling problems of airships, and has also outlined design features of mooring masts to tackle them.

Critical Review of Existing Mooring Masts

A mooring mast for remotely controlled outdoor airships was designed by Kale and Pant [2]. A critical evaluation of that mast was carried out by Goyal et al. [3], and they observed several shortcomings. They noticed that its structure used different aluminium sections of varying cross sections in three different modules, which required a lot of time and money to fabricate. Further, they noticed that the modules had to be made using a matrix i.e., the four aluminium sections had to be fastened together in a nested fashion, demanding a lot of time for assembly. It was felt that drilling too many holes in the aluminium section would decrease its strength drastically. Goyal et al. [3] then re-designed a mast with drag force measuring device for remotely controlled outdoor airships consisting of two modules made of aluminium sections, clamped together using bolts. A wooden base was used and four legs were connected to it. However, the mast was found to be unstable since it had four legs. It was difficult to assemble and disassemble the mast, or to adjust its height, as it needed some bolts to be unfastened. Also, it was designed for a maximum wind load of only 100 N, which is quite less.

Keeping in mind the shortcomings of the existing masts, two mooring masts, one for outdoor airships and another one for indoor airships were designed, fabricated and tested. This paper describes their design and the structural analysis carried out for the same.

Design Requirements for Mooring Mast and Drag Device for Outdoor Airships

The design requirements for the mooring mast for outdoor airship were arrived on the basis of the experience gained during field trials of several remotely controlled airships, and are summarized as follows.

The gross weight of the mooring mast should be less than 60 kg. It should be possible to assemble or disassemble the mast, by two persons within 20 min. It should fit in dismantled condition in a container placed below berths of a long-distance train in India. Also, height from ground should be adjustable from 1.85 to 3.00 m and it should provide pitch freedom of $\pm 30^\circ$ and yaw freedom of 360° to the airship in moored condition.

Additionally, it was required to fit the mast with a wind-load measuring device and alarming system, which could display wind-load within accuracy of 100 g, and sound an alarm when ambient wind conditions became hazardous. Its gross weight should be less than 1 kg and it should be easy to install it on the mast. It should not restrict the motion of the moored airship. It should also be sensitive in the region of expected wind-loads on the airship.

Design of Improved Mast

Keeping in mind the above-mentioned design-requirements of the mooring-mast and the drag force measuring device (with alarming system), a mooring mast was designed and fabricated as shown in the Fig. 1. The improved mast consists of power screw operated telescopic module made of aluminium, mounted on a five legged base for better stability with lockable castor wheels, for mobility. The design was simplified to meet the assembly and transportation requirements.

Conceptual Design of Modified Mast

The modified mast consists of four sub-assemblies, namely:

Adapter Assembly

This assembly comprises of adapter base, mast legs and castor wheels, and it was designed for maximum load condition. The mast has bending tendency due to horizontal drag force on itself and the airship. It has been estimated that the maximum wind load of 214 N can act on a remotely controlled outdoor airship under a wind velocity of 25 m/s, using the formulae of drag force [4]. The maximum bending moments at mast base were calculated for the maximum extended height of 3 m at this wind load,

and various components of the mast were checked for their ability to resist these bending loads. Various components of the adapter assembly are described as follows:

Adapter Base

This is one of the most important components in the assembly as it connects all the five legs to the power screw operated telescopic module. Legs are fixed to the U-shaped brackets and can be hinged or fixed depending upon the usage as shown in Fig. 1.

Mast Legs

These are the main load bearing members, since the horizontal load from wind-loads on the airship and self-weight of the mast acts on the mast legs, from where it gets transferred to the ground through castor wheels. The legs were fixed to the U-shaped brackets at an angle of 10 degree for having proper ground clearance. The legs can be folded inside the mast, making the whole mast system compact for transportation and to reduce assembly time. The hollow rectangular sections of the legs were placed in a manner that they provide maximum moment of inertia ($2.85 \times 10^{-7} \text{ m}^4$), hence increasing the strength of the section in bending [5]. The length of the each leg is 1.8 m, providing adequate ground stability.

Fig. 1 CAD model (left) and photograph (right) of the modified mast for outdoor airships

Caster Wheel Assembly

A standard 76.2 mm diameter castor wheel with load bearing capacity of 150 kg was selected for this application, which results in sufficient factor of safety.

Telescopic Module

This module was designed to provide ease in adjustment of mast height from 1.85 to 3.00 m, by using a power screw. This module consists of a square member with elevating bolt and nut, as described.

Square Member

A square section of cast iron (as it damps vibration caused by random turbulence of wind) was selected for satisfying the geometrical constraints in design and ease in attachment with the adapter.

Elevating Bolt and Nut

The elevating bolt was designed based on ISO standards with square thread of pitch 6.35 mm and major diameter of 50.8 mm [5]. Both bolt and nut were made of Aluminum.

Gimbal Mechanism

This mechanism was designed to provide the required three axes motion to the moored airship. This mechanism consists of three components:

The Yaw Bearing Setup

This setup is made of mild steel so that it can resist the impact loads caused due to sudden gusts or random turbulence of wind. It consists of a bearing having I.D. (inner diameter) of 40 mm and O.D. (outer diameter) of 54 mm. It is designed in such a way that, it provides 360 degree freedom in yaw to the airship.

Pitching Clamp

The pitching clamp provides desired freedom to the airship in pitch axis, made of stainless steel so as to resist the impact loads caused due to sudden gusts or random turbulence of wind. The pitching clamp is attached to the yaw bearing setup using pitching pin made of stainless steel.

Rolling Clamp

The rolling clamp provides a 360 degree freedom in roll to the airship. This clamp carries the nose hook that is attached to the airship nose. The crucial component of the clamp is its mild-steel pin.

Drag Measurement and Alerting Device

A spring-operated force-measuring device was designed to measure the drag force on the airship, as shown in Fig. 2. A pointer attached to telescopic square section containing spring, slides over the scale to indicate the wind-load. To enable use of the same mast and wind-load measuring device for larger airships or during higher ambient winds, it

Fig. 2 Drag device and gimbal mechanism

Fig. 3 Photograph of an outdoor airship moored on the fabricated mast

was decided to design the device to be capable of measuring wind-loads up to 500 N. An electrical alarm system is attached to the pointer, which gets activated if the pointer moves beyond a particular pre-set value of wind-load.

Figure 3 shows an outdoor airship moored on this mooring mast. Slight modifications were incorporated in the design of nose-hook and associated sub-assembly later-on to facilitate the attachment and rolling-freedom of the airship, while fabrication work was being carried-out. The airship had a length of 10.6 m and max. diameter of 2.65 m.

Table 1 provides details and illustration of the various sub-assemblies of the mast, and lists the stresses that were checked for relevant types of failure modes that were expected.

Stress Analysis for Mooring Mast for Outdoor Airships

The volumetric drag coefficient C_{DV} for an airship envelope with $L/D = 4$ is estimated to be 0.0242, using the empirical formula suggested by Hoerner [4].

The mooring mast should be strong enough to bear safely the wind loads on the moored airship.

Also, $\rho = 1.225 \text{ kg/m}^3$ and $V = 39.3 \text{ m}^3$, hence,

$$F_{DE} = \frac{1}{2} \rho V^{2/3} v^2 C_{DV} = 107 \text{ N}$$

As is clear from the airships flight-tested in the past, the drag force on the bare hull is approximately 50 % of the

drag force on the airship as a whole [6], hence, the drag force on the airship is given by

$$F_D = 2(107) = 214 \text{ N}$$

Stress Analysis for Elevating Bolt

Material: Al-1060 aluminium alloy

$S_{yt} = 103 \text{ Mpa}$

Major diameter (d) = 50.8 mm

Thread pitch (p) = 6.35 mm

Pitch diameter (d_p) = 47.625 mm

Minor diameter (d_m) = 44.45 mm

Length (l_b) = 1220 mm

Screw dimensions: O.D = 1.5 d and $h = 0.8 d$

Mass of elevating bolt = 6 kg

Lead angle [5] is given by

$$\tan \lambda = \frac{p}{\pi d_p}$$

$$\tan \lambda = \frac{6.35}{(3.14)(47.625)} = 0.04246$$

$$\lambda = 2.43^\circ$$

Lifting force required [5] is given by

$$P_1 = \frac{F(\sin \lambda + \mu \cos \lambda)}{\cos \lambda - \mu \sin \lambda}$$

Coefficient of friction (μ) = 1.05

Force (F) = 60 N

(including the weight of the elevating bolt itself and the

Table 1 Design considerations for various sub-assemblies

Sub-assembly	CAD model	Stresses checked	Type of failure	Material
Elevating Bolt and Nut		Bending Shear Bearing Axial	Bolt rupture Ductile failure Abrasive wear Impact fatigue	Al alloy-1060
Adapter assembly		Bending Axial	Ductile failure	Mild steel
Flange and Flange bolts		Bending Shear (including fillet weld)	Bolt rupture Ductile failure	Mild steel
Yaw setup U-clamp and Pitch clamp		Shear Impact Tensile	Bolt rupture Impact fatigue	Stainless steel
Square section		Bending stress Axial Compressive Stress Shear stress	Brittle fracture	Cast iron

mooring clamp plus drag-device installed at the top of elevating bolt)

$$P_1 = \frac{60[\sin(2.43^\circ) + 1.05\cos(2.43^\circ)]}{\cos(2.43^\circ) - 1.05\sin(2.43^\circ)}$$

$$P_1 = 68.57 \text{ N}$$

Lowering force required [5] is given by

$$P_2 = \frac{F(\mu \cos \lambda - \sin \lambda)}{\cos \lambda + \mu \sin \lambda}$$

$$P_2 = \frac{60[1.05\cos(2.43^\circ) - \sin(2.43^\circ)]}{\cos(2.43^\circ) + 1.05\sin(2.43^\circ)}$$

$$P_2 = 57.86 \text{ N}$$

Torque required for lifting

$$T_1 = \frac{(P_1)(d_p)}{2}$$

$$T_1 = \frac{(68.57)(47.625 \times 10^{-3})}{2}$$

$$T_1 = 1.6 \text{ Nm}$$

Torque required for lowering

$$T_2 = \frac{(P_2)(d_p)}{2} = \frac{(57.86)(47.625 \times 10^{-3})}{2}$$

$$T_2 = 1.38 \text{ Nm}$$

Shank diameter of the elevating bolt may be used for the design purpose without taking into account the effect of stress-concentration [7].

Bending Stresses

$$\text{Section modulus (Z)} = \frac{\pi d_m^3}{32}$$

Also,

$$\sigma_b = \frac{M}{Z}$$

Now,

$$M = (214)(1.22) = 261.08 \text{ Nm}$$

$$\sigma_b = \frac{261.08}{8.617 \times 10^{-6}} = 30.3 \text{ MPa}$$

Axial Compressive Stress

$$\sigma = \frac{F}{A} = \frac{60}{\pi(0.04445)^2/4}$$

$$\sigma = 0.04 \text{ MPa}$$

Thus, the axial compressive stress due to the weight is negligible as compared to the axial stress due to bending, hence can be neglected in comparison with the latter.

Thus we take normal stress (σ_n) as

$$(\sigma_{\max.})_{\text{tensile}} = 30.3 \text{ MPa}$$

$$(\sigma_{\max.})_{\text{compressive}} = 30.3 \text{ MPa}$$

Maximum Shear Stress

Though the torque transmission across the elevating bolt is zero, but a lifting torque (T_1) as calculated above is required to be applied to overcome the friction. So, this torque induces torsional shear stress in the elevating bolt.

$$\tau = \frac{16T_1}{\pi d_m^3} = \frac{16(1.6)}{\pi(44.45 \times 10^{-3})^3}$$

$$\tau = 0.09 \text{ MPa}$$

Net Stress and F.O.S

Thus, the elevating bolt is subjected to a combination of bending and torsional moments. The principal stress and principal shear stress can be calculated by constructing Mohr's circle [7]. Since the elevating bolt has been made of Al-1060 aluminium-alloy, which is a ductile material, hence "maximum shear stress" theory is applicable for its design [7, 8].

Principal shear stress [7] is calculated as

$$\tau_{\max.} = \sqrt{\left[\left(\frac{\sigma_n}{2}\right)^2 + \tau^2\right]} = \sqrt{\left[\left(\frac{30.3}{2}\right)^2 + (0.09)^2\right]}$$

$$\tau_{\max.} = 15.15 \text{ MPa}$$

Since the material used is ductile, yielding can be considered as the likely mode of failure and hence can be used as basis for designing.

According to maximum shear stress theory, $S_{sy} = 0.5 S_{yt}$ [7].

Therefore, $S_{sy} = 51.5 \text{ MPa}$

Thus,

$$\text{F.O.S.} = \frac{S_{sy}}{\tau_{\max.}} = \frac{51.5}{15.15} = 3.4$$

Bearing Stress [5]

$$\sigma_{\text{bearing}} = \frac{2F_1}{\pi d_m n p}$$

$$F_1 = 0.38F = 0.38(60) = 22.8 \text{ N}$$

$$n = 1$$

$$\sigma_b = \frac{2 \times 22.8}{\pi(44.45)(1)(6.35)(10^{-6})}$$

$$\sigma_b = 0.05 \text{ MPa}$$

Stress Analysis for Flange

The flange-assembly along with its two major dimensions has been shown in Fig. 4.

Material: Mild steel

$$S_{yt} = 207 \text{ MPa}$$

$$b = 200 \text{ mm} = 0.2 \text{ m}$$

$$t = 16 \text{ mm} = 0.016 \text{ m}$$

Bending Stress

$$\text{Section Modulus } (Z) = \frac{bt^2}{6} = 8.5 \times 10^{-6}$$

$$M = (214)(1.22 + 0.016) = 264.5 \text{ Nm}$$

$$\sigma_b = \frac{M}{Z} = 31.1 \text{ MPa}$$

Flange experiences bending stress in a plane perpendicular to the mast-axis, i.e., in a plane parallel to the ground, being maximum tensile at the bottom surface facing to the ground and maximum compressive at the upper surface.

Axial Compressive Stress

Total weight of the elevating bolt and the mooring clamp plus drag-device installed at the top of elevating bolt = 60 N.

Mass of the whole flange-assembly = 5 kg, which is equivalent to a weight of 49 N.

Therefore, total axial compressive load at the base of the flange = 109 N.

Axial compressive stress in the flange along the mast-axis (i.e., perpendicular to the ground) is given by

$$\sigma_c \approx \frac{109}{(0.2)(0.2)} = 0.0027 \text{ MPa}$$

Net Stress and F.O.S

Thus the flange is subjected to biaxial stress [8, 9]. Maximum shear stress is given by

Fig. 4 Some dimensions of the flange-assembly

$$\tau_{\max.} = \frac{\sigma_b - \sigma_c}{2} = 15.55$$

(at an oblique plane, which is at an angle of 45° with the plane of flanges)

Using Maximum shear stress theory, since the material is ductile [7, 8], we have, $S_{sy} = 0.5 S_{yt}$ [7]

$$\text{Therefore, } S_{sy} = 103.5 \text{ MPa}$$

Thus,

$$\text{F.O.S.} = \frac{S_{sy}}{\tau_{\max.}} = \frac{103.5}{15.55} = 6.65$$

Stress Analysis for Flange Bolts

Material: Mild steel

$$S_{yt} = 207 \text{ MPa}$$

$$d_{\text{bolt}} = \text{bolt diameter} = 6 \text{ mm}$$

Tensile Stress

Balancing the moments for equilibrium of the flange-assembly [5], we get

$$M - \frac{Pd_m}{2} = F_b \left(d_b + \frac{d_m}{2} \right)$$

where, F_b = Tensile force induced in each of the flange bolts; d_b = Distance between centre of flange and the centre of bolt hole $\approx 130 \text{ mm}$

$$M = 264.5 \text{ Nm}$$

$$P = \text{Dead load} = 109 \text{ N}$$

$$d_m = 44.45 \text{ mm}$$

This gives

$$F_b = 1721 \text{ N}$$

Also,

$$(\sigma_t)_{\text{bolt}} = \frac{F_b}{\pi d_{\text{bolt}}^2 / 4} = 60.9 \text{ MPa}$$

Hence,

$$\text{F.O.S.} = \frac{207}{60.9} = 3.4$$

This F.O.S. has been evaluated assuming only one of the bolts taking up the whole load. Hence, actual F.O.S. per bolt will be higher than 3.4, since the load is actually shared by the four bolts.

Stress Analysis for Square Section

Figure 5 shows the square section with some of its main dimensions.

Fig. 5 Dimensions of the square section

Grey Cast Iron was used, since it damps out the vibrations caused by random turbulence of wind.

Tensile strength = 276 MPa

Side (B) = 76.2 mm

Length (l) = 1829 mm

Thickness (t) = 3 mm

Bending Stress

M = max. at the base of square – section
 $= (214)(1.22 + 0.016 + 1.829) = 656 \text{ Nm}$

$$I = \frac{B^4}{12} - \frac{(B - 2t)^4}{12} = 7.86 \times 10^{-7}$$

Maximum bending stress will be at the pair of opposite outer vertices (max. tensile at one and max. compressive at the other) of the square section at its base, when the drag force on the airship is perpendicular to the diagonal of the square section passing through the pair of other two opposite vertices. Since the moored airship has yaw-freedom of 360° , this situation can be easily encountered with during field testing.

$$\sigma_b = \text{max. at } y = \frac{\sqrt{2}(76.2)}{2} \text{ mm} = 53.88 \text{ mm}$$

$$\frac{M}{I} = \frac{\sigma_b}{y}$$

This gives

$$\sigma_b = 45 \text{ MPa}$$

Axial Compressive Stress due to weight

Designing the base of the square-section to support whole of the mass of the mast, axial compressive load = 549 N.

Cross-sectional area of the section = 0.0008784 m^2 .

This gives axial compressive stress as

$$\sigma_c = 0.625 \text{ MPa}$$

Net Stress and F.O.S

Thus we take normal stress (σ_n) as

$$(\sigma_{\text{max.}})_{\text{tensile}} = 44.375 \text{ MPa}$$

$$(\sigma_{\text{max.}})_{\text{compressive}} = 45.625 \text{ MPa}$$

Since grey cast iron is a brittle material and its compressive strength is much greater than the tensile strength [10], the likely mode of failure is brittle fracture due to tensile stress.

Therefore,

$$\text{F.O.S.} = \frac{276}{44.375} = 6.2$$

Stress Analysis for Adapter Base

The isometric view of the adapter assembly has been shown in Fig. 6 with some of its dimensions.

Material: Mild Steel

$$S_{yt} = 250 \text{ MPa} = S_{yc} [7]$$

D = Outer diameter = 200 mm

d = Inner diameter = 120 mm

Bending Stress

$$M \approx (214)(1.22 + 0.016 + 1.829) = 656 \text{ Nm}$$

Now,

$$\text{Section modulus (Z)} = \frac{\pi(D^4 - d^4)}{32D}$$

This gives,

$$Z = 6.8326 \times 10^{-4} \text{ m}^4$$

Fig. 6 Dimensions of the adapter assembly

Thus,

$$\sigma_b = \frac{M}{Z} = 0.96 \text{ MPa}$$

Axial Compressive Stress Due to Weight

Designing the adapter-base to support whole of the mass of the mast, axial compressive load = 549 N.

Cross-sectional area of the section = 0.02 m².

This gives axial compressive stress as

$$\sigma_c = 0.03 \text{ MPa}$$

Net Stress

Thus we take normal stress (σ_n) as

$$(\sigma_{\max.})_{\text{tensile}} = 0.93 \text{ MPa}$$

$$(\sigma_{\max.})_{\text{compressive}} = 0.99 \text{ MPa}$$

Since the material used is ductile, yielding can be considered as the likely mode of failure and hence can be used as basis for designing. Since the compressive stress produced here is far less than the S_{yc} value [7], there are no chances of yielding or crushing.

Stress Analysis for Adapter-Assembly Bolts

Material: Mild steel

$S_{yt} = 207 \text{ MPa}$

$d_{\text{bolt}} = \text{bolt diameter} = 10 \text{ mm}$

Tensile Stress

Balancing the moments for equilibrium of the adapter-assembly [5], we get

$$M - \frac{PB}{2} = F_b \left(d_b + \frac{B}{2} \right)$$

where, F_b = Tensile force induced in each of the bolts;
 d_b = Distance between the adapter centre and the centre of bolt hole = 180 mm; $M = (214)(1.22 + 0.016 + 1.829)$
 $= 656 \text{ Nm}$; P = Dead load = 549 N; $B = 76.2 \text{ mm}$

This gives

$$F_b = 2912 \text{ N}$$

Also,

$$(\sigma_t)_{\text{bolt}} = \frac{F_b}{\pi d_{\text{bolt}}^2 / 4} = 37 \text{ MPa}$$

Hence,

$$\text{F.O.S.} = \frac{207}{37} = 5.6$$

This F.O.S. has been evaluated assuming only one of the bolts taking up the whole load. Hence, actual F.O.S. per bolt will be higher than 5.6, since the load is actually shared by the 4 bolts.

Stress Analysis for Mast Legs

The cross-sectional dimensions of a mast-leg have been shown in Fig. 7.

Material: Al-6351 aluminium alloy

$S_{yt} = 150 \text{ MPa} = S_{yc}$ and Fatigue strength = 90 MPa

Length of each leg = 1.8 m, and there are a total of five legs.

Bending Stress

Total mass of the mast = 56 kg, which gives a weight of 549 N. Each of the five mast-legs support (1/5)th of the total mast-weight. Hence, each leg should be strong enough to safely bear that load. So, we take the mass supported by each of the legs as “56/5 kg” (which is approx. equivalent to a weight of 110 N) for design purpose.

The leg touches the ground only through the castor-wheel attached to its end. The reaction force, exerted by the ground on the castor-wheel tends to bend the leg.

Bending moment due to the weight of the mast is max. at the hinged-end of the leg and is given by

Fig. 7 Dimensions of the cross-section of each of the mast-legs

$$M_1 = (110)(1.8) = 198 \text{ Nm}$$

Bending moment per leg due to air-drag on the moored airship is given by

$$M_2 = \frac{1}{5}(214)(1.22 + 0.016 + 1.829) = 131.2 \text{ Nm}$$

“ M_2 ” will add up into “ M_1 ”, when the drag force on the airship is in a direction parallel and opposite to the mast-leg, which is under consideration, hence gives the maximum possible bending moment for a leg, for a particular orientation of the moored airship (which is having 360° yaw-freedom).

Therefore, maximum possible bending moment is given by

$$M = M_1 + M_2 = 329.2 \text{ Nm}$$

Each leg is having hollow-rectangular cross-section with the dimensions as shown above.

$$I = 2.85 \times 10^{-7} \text{ m}^4$$

$$\sigma_b = \text{max. at } y = \frac{62}{2} \text{ mm} = 31 \text{ mm}$$

$$\frac{M}{I} = \frac{\sigma_b}{y}$$

This gives:

$$\sigma_b = 35.8 \text{ MPa}$$

Bending stresses act along the longitudinal axis of the leg itself.

Compressive Stress Along the Mast-Axis

The weight of the mast (acting at the pivot-joint of the leg with the adapter-base) and the normal reaction (offered by the ground through the castor-wheel) together induces the compressive stress in the mast-leg in a direction parallel to the vertical mast-axis, or we can say perpendicular to the longitudinal axis of the leg itself.

Therefore, total compressive load per leg = $(56)(9.8)/5 \approx 110 \text{ N}$.

Maximum Compressive stress is given by

$$\begin{aligned} \sigma_c &\approx \frac{110}{\text{Min.cross-sectional area}} \\ &= \frac{110}{2(0.0025)(1.8)} = 0.012 \text{ MPa} \end{aligned}$$

Net Stress and F.O.S

Thus the leg is subjected to biaxial stress [8, 9]. Maximum shear stress is given by

$$\tau_{\text{max.}} = \frac{\sigma_b - \sigma_c}{2} = 17.9 \text{ MPa}$$

(at an oblique plane, which is at an angle of 45° with the axis of leg)

Since the material used is ductile, yielding can be considered as the likely mode of failure and hence can be used as basis for designing.

Using Maximum shear stress theory, since the material is ductile [7, 8], we have, $S_{sy} = 0.5S_{yt}$ [7]

Therefore, $S_{sy} = 75 \text{ MPa}$

Thus,

$$\text{F.O.S.} = \frac{S_{sy}}{\tau_{\text{max.}}} = \frac{75}{17.9} = 4.2$$

Stress Analysis for Pitch Clamp (U-Clamp)

The isometric view of the set-up being used for providing yaw and pitch freedoms to the moored airship has been shown in Fig. 8 with its principal dimensions.

Material: Stainless Steel

$S_{yt} = 240 \text{ MPa}$

$S_{sy} = 0.5 S_{yt} = 120 \text{ MPa}$ [7]

Tensile Stress

Under the estimated wind-load of 214 N, each arm of “U” takes half of this load, viz., 107 N. Maximum tensile stress is in each of the two arms of U-clamp at the location of bolt passing through it (where the resisting cross-sectional area is minimum), and is given by

$$\sigma_t = \frac{107}{(0.031)(0.005) - (0.012)(0.005)} = 1.13 \text{ MPa}$$

Therefore, considering yielding as the mode of failure,

$$\text{F.O.S.} = \frac{240}{1.13} = 212.4$$

Fig. 8 Dimensions of the yaw-setup and pitch clamp

Shear Stress

Wind-load on the airship tends to induce double-shear in each of the two arms of U-clamp beyond the location of bolt passing through it.

$$\tau = \frac{107}{2(0.0185)(0.005)} = 0.6 \text{ MPa}$$

Therefore, considering yielding as the mode of failure,

$$\text{F.O.S.} = \frac{120}{0.6} = 200$$

Stress Analysis for Bolts of Yaw and Pitch Setup

Material: Mild steel

$S_{yt} = 207 \text{ MPa}$

$S_{sy} = 103.5 \text{ MPa}$ [7]

Shear Stress

Wind-load on the airship tends to induce double-shear in each of the two bolts passing across an arm of U-clamp [not taking into account the shearing action over the bolt due to the yaw-setup cup at the top of elevating bolt, we are on more safe side, since triple or quarter shear (due to additionally double or single shearing action by the wall of the cup, depending upon whether the end of the bolt is passed across the inner surface of the cup-wall or not) offers even more resistance to the shearing action].

Shear stress in the bolt is given by

$$\tau = \frac{107}{2 \left[\frac{\pi(0.012)^2}{4} \right]} = 0.47 \text{ MPa}$$

Therefore, considering yielding as the mode of failure,

$$\text{F.O.S.} = \frac{103.5}{0.47} = 220$$

Design Requirements for Mooring Mast for Indoor Airships

The design requirements for the mooring mast for remotely controlled indoor airships were arrived based on the user-specified requirements, which have been summarised as

- The gross weight of the mast should be less than 10 kg.
- It should allow easy assembly and de-assembly within 10 min by two persons.
- It should be easily movable on the ground, and should have height adjustable from 0.8 to 1.4 m. Also, its legs should be foldable.
- The various components of the mast should also fit in a standard suitcase (0.68 m × 0.49 m × 0.28 m).

- It should also allow the airship to move freely; 360° in yaw and roll, and −90° to +30° in pitch in the moored position. Further, it should be very easy to operate, adjust, fold and unfold it.

Fabricated Design of the Mooring Mast for Indoor Airships

Keeping in mind the key design-requirements given above, a mooring mast for indoor airships was designed and fabricated as shown in the Fig. 9.

Conceptual Design for the Mast for Indoor Airships

The mooring mast for remotely controlled indoor airships consists of an elevating bolt operated telescopic module made of standard aluminium 6351, mounted on a tripod adapter base which consists of standard aluminium (6351) legs (which are locked in the right position by inclined Al-strips) and lockable castor wheels, for ease in mobility. The mast can accommodate non-rigid indoor airships of length up to 5 meter. The various modules and components of the mast were designed to enable quick assembly and transportation. The mast has a specially designed mooring-clamp at the top. The mast-structure is composed of mainly three sub-assemblies, as described in the sections that follow.

Adapter Assembly

This assembly consists of the robust mast-base with three U-clamps. The three legs of the mast are hinged into these U-clamps (one in each) and are locked by the inclined Aluminium (Al)-strips (using bolts). The Al-strips are hinged into separate U-clamps attached on a block mounted on the lower telescopic module using allen-bolts.

Telescopic Modules' Assembly

There are two modules providing the desired height-adjustability to the mast. The upper one, hereby referred to as Hollow Square Section (HSS)-2, is moved in-or-out of the lower one, which is referred to as Hollow Square Section (HSS)-1, to adjust the mast's height just by rotating a pulley. This sub-system employs a bevel-gear assembly which transfers the rotary-motion from pulley to an elevating bolt supported in a bearing. This causes a nut (fitted into the base of upper telescopic module) to move up or down depending upon the sense of rotation.

Fig. 9 CAD model (*left*) and photograph (*right*) of the fully assembled mooring mast for indoor airships

Mooring Clamp Assembly

It consists of a hollow bent Al-rod (bent by 90°), with its lower end fitted into a bearing at the top of upper telescopic module for providing yaw-freedom (360°) and other end carrying a mooring arrangement. The mooring arrangement consists of a U-clamp, where a solid rod passing across the two arms of “U” (each arm containing a ball-bearing) provides pitch-freedom to the airship. A properly designed stopper attached to the end of each of the two arms of “U” clamp limits the pitch-freedom up to $+30^\circ$. A protruding solid rod is supported in two bearings contained in a hollow-rod attached to the cross-rod passing across the two arms of “U” clamp, and this rod provides the required rolling-freedom (360°) to the moored airship. This rod is having two thorough holes in it, exactly matching with the holes in a frontal protruding hollow rod at airship-nose. Then, the airship can be moored with the mast just by putting the protruding solid-rod of the mast into the hollow rod protruding from the nose of the airship and simply putting the bolts-with-nuts into the matching holes.

Figure 10 shows an indoor airship moored to this mooring mast. The airship envelope is of Zhiyuan shape [11], with length of 4.2 m and max. diameter of 1.27 m.

The design considerations for various subassemblies of the mooring mast are listed in Table 2.

Stress Analysis for Mooring Mast for Indoor Airships

The volumetric drag coefficient C_{DV} for an airship envelope with $L/D = 4$ is estimated to be 0.0255, using the empirical formula suggested by Hoerner [4]. Though the forward velocity of indoor airship is quite small, but it could face higher winds while being relocated. Hence, the mooring mast was designed to bear a wind load of 25 m/s.

$$\rho = 1.225 \text{ kg/m}^3$$

$$V = 3.6 \text{ m/s}$$

As is clear from the airships flight-tested in the past, the drag force on the bare hull is approximately 50 % of the drag force on the airship as a whole [6], hence, the drag force on the airship is given by

$$F_D = 2(F_{DE}) = (2) \frac{1}{2} \rho V^{2/3} v^2 C_{DV} \approx 47 \text{ N}$$

Mass of the overhanging end-portion of the mooring clamp is 0.8 kg, hence Bending Moment (M_W) at any transverse

Fig. 10 Photograph of an indoor airship moored on the fabricated Mast

cross-section due to its weight (about the centroidal axis of any transverse section, say mast base, or hollow square section) = $(0.8)(9.8)(0.13)$ i.e., $M_w = 1$ Nm. It is assumed that the weight acts at approximately 0.13 m horizontal distance from the vertical axis passing through the centroids of the respective transverse sections of the mast.

Stress Analysis for Elevating Bolt

Material: Mild Steel [MS-2062 (MS-Bright)]

$S_{yt} = 250$ MPa

Major diameter (d) = 20 mm

Thread pitch (p) = 3.8 mm

Pitch diameter (d_p) = 17.5 mm

Minor diameter (d_m) = 15 mm

Length of threaded portion (l_b) = 609.6 mm

Length and diameter of the shank of the elevating bolt below the threaded portion (through which the elevating bolt is supported in a bearing placed just above the mast-base) = 46 mm and 15 mm respectively.

Lead angle [5] is given by

$$\tan \lambda = \frac{p}{\pi d_p}$$

$$\tan \lambda = \frac{3.8}{(3.14)(17.5)}$$

$$\lambda = 3.95^\circ$$

Lifting force required [5] is given by

$$P_1 = \frac{F(\sin \lambda + \mu \cos \lambda)}{\cos \lambda - \mu \sin \lambda}$$

Coefficient of friction (μ) = 0.2

Force (F) = $(3.6)(9.8) = 35.3$ N

$$P_1 = \frac{35.3[\sin(3.95^\circ) + 0.2 \cos(3.95^\circ)]}{\cos(3.95^\circ) - 0.2 \sin(3.95^\circ)}$$

$$P_1 = 9.6$$
 N

Lowering force required [5] is given by

$$P_2 = \frac{F(\mu \cos \lambda - \sin \lambda)}{\cos \lambda + \mu \sin \lambda}$$

$$P_2 = \frac{35.3[0.2 \cos(3.95^\circ) - \sin(3.95^\circ)]}{\cos(3.95^\circ) + 0.2 \sin(3.95^\circ)}$$

$$P_2 = 4.5$$
 N

Torque required for lifting

$$T_1 = \frac{(P_1)(d_p)}{2}$$

$$T_1 = \frac{(9.6)(17.5 \times 10^{-3})}{2}$$

$$T_1 = 0.084$$
 Nm

Torque required for lowering

Table 2 Design considerations for various sub-assemblies

Sub-assembly	CAD model	Stresses checked	Type of failure	Material
Elevating Bolt and Nut		Bending Shear Bearing Axial	Bolt rupture Ductile failure Abrasive wear Impact fatigue	Mild steel
Adapter assembly		Bending Axial	Ductile failure	Al-6351 alloy
Telescopic Modules' assembly		Bending Compressive	Rupture Ductile failure	Al-6351 alloy
Mooring clamp		Impact Tensile Bending	Rupture Impact fatigue	Al-6351 alloy

$$T_2 = \frac{(P_2)(d_p)}{2} = \frac{(4.5)(17.5 \times 10^{-3})}{2}$$

$$T_2 = 0.04 \text{ Nm}$$

Shank diameter of the elevating bolt may be used for the design purpose without taking into account the effect of stress-concentration [7].

Bending Stress

$$M = \text{max. at the bottom most section of elevating bolt}$$

$$= \text{Bending Moment due to drag}$$

$$\text{force on the airship} + M_w$$

$$= 47(0.12 + 0.5 + 0.7) + 1$$

$$= 63.04 \text{ Nm}$$

$$I = \frac{\pi}{64} [(0.015)^4] = 2.48 \times 10^{-9} \text{ m}^4$$

$$\sigma_b = \text{max. at } y = \frac{15}{2} \text{ mm} = 7.5 \text{ mm}$$

$$\frac{M}{I} = \frac{\sigma_b}{y}$$

This gives:

$$\sigma_b = 190 \text{ MPa}$$

Axial Compressive Stress

Total mass exerting the axial load (due to weight) at the bottom most section of elevating bolt (taking into account the masses of mooring clamp, HSS-2, nut, bevel-gears, elevating bolt itself, etc.) = 3.6 kg. Hence, total axial load = (3.6)(9.8) = 35.3 N.

Area of cross-section of shank of elevating bolt = 0.00017 m^2 .

Therefore, axial compressive stress at the bottom of the elevating bolt due to the weight is calculated as

$$\sigma_c = 0.2 \text{ MPa}$$

Thus, the axial compressive stress due to the weight is negligible as compared to the axial stress due to bending, hence can be neglected in comparison with the latter.

Thus we take normal stress (σ_n) as

$$(\sigma_{\text{max.}})_{\text{tensile}} = 190 \text{ MPa}$$

$$(\sigma_{\text{max.}})_{\text{compressive}} = 190 \text{ MPa}$$

Torsional Shear Stress

Maximum Shear Stress in the screw body (at the shank surface) is given by

$$\tau = \frac{16T_1}{\pi d_m^3} = \frac{16(0.084)}{\pi(15 \times 10^{-3})^3}$$

$$\tau = 0.13 \text{ MPa}$$

Net Stress and F.O.S

Thus, the elevating bolt is subjected to a combination of bending and torsional moments. The principal stress and principal shear stress can be calculated by constructing Mohr's circle [7]. Since the elevating bolt has been made of Mild steel, which is a ductile material, hence "maximum shear stress" theory is applicable for its design [7, 8].

Principal shear stress [7] is calculated as

$$\tau_{\text{max.}} = \sqrt{\left[\left(\frac{\sigma_n}{2}\right)^2 + \tau^2\right]} = \sqrt{\left[\left(\frac{190}{2}\right)^2 + (0.13)^2\right]}$$

$$\tau_{\text{max.}} = 95 \text{ MPa}$$

Since the material used is ductile, yielding can be considered as the likely mode of failure and hence can be used as basis for designing.

According to maximum shear stress theory, $S_{sy} = 0.5 S_{yt}$ [7]. Therefore, $S_{sy} = 125 \text{ MPa}$

Thus,

$$\text{F.O.S.} = \frac{S_{sy}}{\tau_{\text{max.}}} = \frac{125}{95} = 1.3$$

Stress Analysis for Mast-Base

Material: Al-6351 aluminium alloy

$S_{yt} = 150 \text{ MPa} = S_{yc}$ [7] and Fatigue strength = 90 MPa

The cross-sectional and longitudinal dimensions of the mast-base have been shown in Fig. 11.

Fig. 11 Dimensions of the Mast-base

Bending Stress

$$\begin{aligned}
 M &= \text{max. at the bottom of Mast – base} \\
 &= \text{Bending Moment due to drag} \\
 &\quad \text{force on the airship} + M_w \\
 &= 47(0.12 + 0.5 + 0.7 + 0.03) + 1 \\
 &= 63.45 \text{ Nm}
 \end{aligned}$$

$$I = \frac{\pi}{64} \left[(0.09)^4 - (0.03)^4 \right] = 3.18 \times 10^{-6} \text{ m}^4$$

$$\sigma_b = \text{max. at } y = \frac{90}{2} \text{ mm} = 45 \text{ mm}$$

$$\frac{M}{I} = \frac{\sigma_b}{y}$$

This gives:

$$\sigma_b = 0.912 \text{ MPa}$$

Axial Compressive Stress

For the structural arrangement of the mast, whole of the mast-weight can be supposed to be balanced by the mast-base. Further, assuming this load at the mast base and designing HSS-1 accordingly, we are on more safe side.

Thus, taking axial compressive load = $(9.055)(9.8) = 88.7 \text{ N}$ and area of cross-section of the mast-base = 0.005652 m^2

Axial compressive stress at the bottom of the HSS-2 due to the weight is calculated as

$$\sigma_c = 0.016 \text{ MPa}$$

Net Stress and F.O.S

Therefore,

$$\begin{aligned}
 (\sigma_{\text{max.}})_{\text{tensile}} &= 0.896 \text{ MPa} \\
 (\sigma_{\text{max.}})_{\text{compressive}} &= 0.928 \text{ MPa}
 \end{aligned}$$

Since the material used is ductile, yielding can be considered as the likely mode of failure and hence can be used as basis for designing.

Thus,

$$\text{F.O.S.} = \frac{150}{0.928} = 161.6$$

Stress Analysis for Mast-Legs

Material: Al-6351 aluminium alloy

$S_{yt} = 150 \text{ MPa} = S_{yc}$ [7] and Fatigue strength = 90 MPa

Length of each leg = 0.434 m, and there are a total of three legs.

Bending Stress

Total mass of the mast = 9.055 kg, which gives a weight of “88.7 N”. Each of the three mast-legs support $(1/3)^{\text{rd}}$ of the total mast-weight. Hence, each leg should be strong enough to safely bear that load. So, we take the mass supported by each of the legs as “9.055/3 kg” (which is approx. equivalent to a weight of 29.6 N) for design purpose.

The leg touches the ground only through the castor-wheel attached to its end. The reaction force, exerted by the ground on the castor-wheel tends to bend the leg.

Bending moment due to the weight of the mast is max. at the hinged-end of the leg and is given by

$$M_1 = (29.6)(0.434) = 12.8 \text{ Nm}$$

Bending moment per leg due to air-drag on the moored airship is given by

$$M_2 = \frac{1}{3}(47)(0.12 + 0.5 + 0.7 + 0.03) = 21.15 \text{ Nm}$$

“ M_2 ” will add up into “ M_1 ”, when the drag force on the airship is in a direction parallel and opposite to the mast-leg, which is under consideration, hence gives the maximum possible bending moment for a leg, for a particular orientation of the moored airship (which is having 360° yaw-freedom).

Therefore, maximum possible bending moment is given by

$$M = M_1 + M_2 = 33.95 \text{ Nm}$$

Each leg is a hollow-cylinder with internal diameter of “0.028 m” and outer diameter of “0.032 m”.

Hence, for bending consideration,

$$I = \frac{\pi}{64} \left[(0.032)^4 - (0.028)^4 \right] = 2.13 \times 10^{-8} \text{ m}^4$$

$$\sigma_b = \text{max. at } y = \frac{32}{2} \text{ mm} = 16 \text{ mm}$$

$$\frac{M}{I} = \frac{\sigma_b}{y}$$

This gives

$$\sigma_b = 25.5 \text{ MPa}$$

Bending stresses act along the longitudinal axis of the leg itself.

Compressive Stress Along the Mast-Axis

The weight of the mast (acting at the pivot-joint of the leg with the adapter-base) and the normal reaction (offered by the ground through the castor-wheel) together induces the compressive stress in the mast-leg in a direction parallel to

the vertical mast-axis, or we can say perpendicular to the longitudinal axis of the leg itself.

Therefore, total compressive load per leg = $(9.055)(9.8)/3 \approx 29.6$ N

Maximum compressive stress is given by

$$\sigma_c \approx \frac{29.6}{\text{Min. cross-sectional area}} = \frac{29.6}{2(0.002)(0.434)} = 0.017 \text{ MPa}$$

Net Stress and F.O.S

Thus the leg is subjected to biaxial stress [8, 9]. Maximum shear stress is given by

$$\tau_{\max} = \frac{\sigma_b - \sigma_c}{2} = 12.7 \text{ MPa}$$

(at an oblique plane, which is at an angle of 45° with the axis of leg).

Since the material used is ductile, yielding can be considered as the likely mode of failure and hence can be used as basis for designing.

Using Maximum shear stress theory, since the material is ductile [7, 8], we have, $S_{sy} = 0.5 S_{yt}$ [7].

Therefore, $S_{sy} = 75$ MPa

Thus,

$$\text{F.O.S.} = \frac{S_{sy}}{\tau_{\max}} = \frac{75}{12.7} = 5.9$$

Stress Analysis for the Bolts at hinges

Material: Mild Steel

$S_{yt} = 207$ MPa

Shear Stress

Each of the bolts used for hinging of a mast-leg can be designed to be strong enough to safely bear the weight of the mast as a whole. So, we take the mass supported by each of the bolts as “9.055 kg” for design purpose, which is equivalent to a load of 88.7 N.

Each of those bolts used is of 6 mm dia. (and 2 inch length).

Each bolt is subjected to double-shear under the combined action of weight of the mast and the reaction from the ground.

Shear stress in the bolt is given by

$$\tau = \frac{88.7}{2 \left[\frac{\pi(0.006)^2}{4} \right]} = 1.6 \text{ MPa}$$

Since the material used is ductile, yielding can be considered as the likely mode of failure and hence can be used as basis for designing.

According to maximum shear stress theory, $S_{sy} = 0.5 S_{yt}$ [7]

Therefore, $S_{sy} = 103.5$ MPa

Thus,

$$\text{F.O.S.} = \frac{S_{sy}}{\tau} = \frac{103.5}{1.6} = 64.7$$

Stress Analysis for HSS-1 and HSS-2 (portions of Telescopic Modules' Assembly)

Now the Airship moored to the mast has 360° Yaw-freedom. Thus the airship can rotate freely about the (vertical) axis of the mast. Due to the air-gusts, the airship will be at different positions around the mast at different instants of time (due to yaw-freedom). But at whatever position it is at an instant of time, the air-flow through around the airship will exert the air-drag. This force will act on the airship along the (horizontal) axis of the airship. (Note that in the moored position, our airship was exactly horizontal due to perfect balancing of the forces along vertical and the mast was not contributing at all towards its (airship's) vertical equilibrium).

Therefore, the drag force is acting parallel to the ground. The airship exerts this force on the mooring mast directed parallel to the ground. As airship will rotate around the mast under the effects of air-gusts, the drag force will also keep on rotating around the mast as shown in Fig. 12. Now as the mast stands straight on the ground, this force will try

Fig. 12 Rotation of the drag force on the airship and hence the bending-axis for hollow square sections due to yaw-freedom of the moored airship

Fig. 13 Dimensions of HSS-1 and HSS-2

to bend the HSS-1 and HSS-2 about the axis parallel to the ground and perpendicular to the instantaneous direction of the drag force. Thus the axis of bending of the HSS-1 and HSS-2 will keep on changing continuously with the changing orientation of the drag force.

The bending stresses will be maximum at the outer vertex of the hollow square cross-section of HSS-1 or HSS-2 when the drag force is perpendicular to the diagonal of the square cross-section, as shown in Fig. 12a or e, since, in this case, “y” is maximum.

Bending Stress = Maximum tensile at the vertex opposite to the vertex at which the force acts

Bending Stress = Maximum compressive at the vertex at which the force acts

The cross-sectional dimensions of the hollow square sections (HSS-1 and HSS-2) have been indicated in Fig. 13.

For Hollow Square Section-2 (HSS-2),

Material: Al-6351 aluminium alloy

$S_{yt} = 150 \text{ MPa} = S_{yc}$ [7] and Fatigue strength = 90 MPa

Bending Stress

$M = \text{max. at the base of HSS - 2}$

= Bending Moment due to drag

force on the airship + M_w

= $47(0.12 + 0.5) + 1$

= 30.14 Nm

$$I = \frac{1}{12} [(0.0508)^4 - (0.0458)^4] = 1.88 \times 10^{-7} \text{ m}^4$$

$$\sigma_b = \text{max. at } y = \frac{\sqrt{2}(50.8)}{2} \text{ mm} = 35.9 \text{ mm}$$

$$\frac{M}{I} = \frac{\sigma_b}{y}$$

This gives:

$$\sigma_b = 5.75 \text{ MPa}$$

Axial Compressive Stress

Mass of HSS-2 = 0.665 kg

Mass of mooring clamp = 0.955 kg

Mass of nut = 0.150 kg

Hence, total axial load = $(1.77)(9.8) = 17.3 \text{ N}$

Area of cross-section of HSS-2 = 0.000483 m^2

Therefore, axial compressive stress at the bottom of the HSS-2 due to the weight is calculated as $\sigma_c = 0.036 \text{ MPa}$.

Net Stress and F.O.S

Therefore

$$(\sigma_{\text{max.}})_{\text{tensile}} = 5.7 \text{ MPa}$$

$$(\sigma_{\text{max.}})_{\text{compressive}} = 5.8 \text{ MPa}$$

Since the material used is ductile, yielding can be considered as the likely mode of failure and hence can be used as basis for designing.

Thus,

$$\text{F.O.S.} = \frac{150}{5.8} = 25.8$$

For Hollow Square Section-1 (HSS-1),

Material: Al-6351 aluminium alloy

$S_{yt} = 150 \text{ MPa} = S_{yc}$ [7] and Fatigue strength = 90 MPa

Bending Stress

$M = \text{max. at the base of HSS-1}$

= Bending Moment due to drag

force on the airship + M_w

= $47(0.12 + 0.5 + 0.7) + 1$

= 63.04 Nm

$$I = \frac{1}{12} [(0.0635)^4 - (0.0585)^4] = 3.789 \times 10^{-7} \text{ m}^4$$

$$\sigma_b = \text{max. at } y = \frac{\sqrt{2}(63.5)}{2} \text{ mm} = 44.9 \text{ mm}$$

$$\frac{M}{I} = \frac{\sigma_b}{y}$$

This gives:

$$\sigma_b = 7.47 \text{ MPa}$$

Axial Compressive Stress

Due to inclined aluminium-strips joining HSS-1 with the legs and support to whole mast through the mast-base

joined with the legs, whole of the mast-weight can be supposed to be balanced by the base of HSS-1. Further, assuming this load at the mast base and designing HSS-1 accordingly, we are on more safe side.

Thus, taking axial compressive load = $(9.055)(9.8) = 88.7$ N and area of cross-section of HSS-1 = 0.00061 m^2

Axial compressive stress at the bottom of the HSS-2 due to the weight is calculated as

$$\sigma_C = 0.145 \text{ MPa}$$

Net Stress and F.O.S

Therefore

$$(\sigma_{\max.})_{\text{tensile}} = 7.325 \text{ MPa}$$

$$(\sigma_{\max.})_{\text{compressive}} = 7.615 \text{ MPa}$$

Since the material used is ductile, yielding can be considered as the likely mode of failure and hence can be used as basis for designing.

Thus,

$$\text{F.O.S.} = \frac{150}{7.615} = 19.7$$

Stress Analysis for Bent Rod (a part of Mooring Clamp)

Material: Al-6351 aluminium alloy

$S_{yt} = 150 \text{ MPa} = S_{yc}[7]$ and Fatigue strength = 90 MPa

Bending Stresses

M = maximum at the base of the bent rod (90°)

$$\begin{aligned} &= \text{Bending Moment due to drag} \\ &\quad \text{force on the airship} + M_w \\ &= 47(0.12) + 1 \\ &= 6.64 \text{ Nm} \end{aligned}$$

$$I = \frac{\pi(0.02)^4}{64} = 7.85 \times 10^{-9} \text{ m}^4$$

$$\frac{M}{I} = \frac{\sigma_b}{y}$$

$$\sigma_b = \frac{6.64}{7.85 \times 10^{-9}} \times (10 \times 10^{-3}) = 8.46 \text{ MPa}$$

Axial Compressive Stress

Mass of the mooring clamp as a whole = 0.955 kg.

There, axial compressive stress at the bottom of the bent-rod due to the weight of the mooring-clamp is given by

$$\sigma_C = \frac{0.955 \times 9.81}{\pi(0.02)^2/4} = 0.03 \text{ MPa}$$

Net Stress and F.O.S

Therefore,

$$(\sigma_{\max.})_{\text{tensile}} = 8.43 \text{ MPa}$$

$$(\sigma_{\max.})_{\text{compressive}} = 8.49 \text{ MPa}$$

Since the material used is ductile, yielding can be considered as the likely mode of failure and hence can be used as basis for designing.

Thus,

$$\text{F.O.S.} = \frac{150}{8.49} = 17.6$$

Stress Analysis for Protruding Solid Mooring Rod

Material: Al-6351 aluminium alloy

$S_{yt} = 150 \text{ MPa}$ and Fatigue strength = 90 MPa

Tensile Stress

Diameter of this rod = 8 mm

$$\sigma_t = \frac{47}{\pi(0.008)^2/4} = 0.94 \text{ MPa}$$

Since the material used is ductile, yielding can be considered as the likely mode of failure and hence can be used as basis for designing.

Thus,

$$\text{F.O.S.} = \frac{150}{0.94} = 159.5$$

Conclusions and Suggestions for Improvement

Regarding the mooring mast for outdoor airships, the total weight of the whole system was around 56 kg, which is within the upper limit of 60 kg with a 15 % margin for growth. During several trials, it was seen that the mast could easily be assembled in less than 20 min by a two-member team, as desired. The use of power screw operated telescopic module ensures that the alteration of the height requirement for the mast is met easily, without the need to remove and tighten any bolts. Desired pitch and yaw freedom is available through the gimbal mechanism. The five legged base with adapter assembly make the mast stable. Thus, it can be concluded that this modified mast designed for ground handling of remotely controlled outdoor airships met all the user specified requirements.

However, few improvements to this basic design can be incorporated. First of all, the rotation of the elevating nut can be made more convenient by replacing the nut handle with a bevel gear assembly along with pulleys. In the

adapter assembly, the mast legs can be provided with spring activated quick response mechanism and all terrain lockable wheel system, to enable quick folding and hence reducing the effort and time required for de-assembly.

Regarding the mooring mast for indoor airships, the total mass of the mast was found to be around 9.055 kg, thus well within the required limits. It was very easy to assemble and dis-assemble by 2 persons in 10 min. The mooring-clamp provided the airship the required yawing-freedom (360°), pitching-freedom (-90° to $+30^\circ$) and rolling-freedom (360°). The telescopic-modules' assembly provided the mast the height adjustable from 0.8 to 1.5 m. The tripod-base made the mast stable on ground and the castor-wheels made it easily movable on the ground. The inclined Al-strips allowed folding the mast-legs by removing just three bolts. It was possible to pack all the components of this mast in a standard suitcase, making it highly portable. Thus, it can be concluded that this mast successfully met all the user specified requirements.

But, still some improvements are possible in its structural design. The legs can be made very-easily foldable and lockable by using a spring-activated bolt besides a fixed bolt used for hinging of each leg into a U-clamp of the adapter assembly.

It is to be noted that some of the factors of safety as calculated above are quite high. At first look it may appear that the dimensions of various components could be reduced to bring down these numbers. But that will reduce the efficacy of the mast, for e.g., if the size of the mast-base is reduced, it might become unstable. Also, it is very difficult to predict the magnitude and direction of the ambient wind that the mast will have to face while deployed in the field. Hence, the mast may have to face much larger forces than what are expected. Further, the mast may also experience impact-loads due to sudden gusts of the air. Thus, the mast components, in reality, may be subjected to far greater impact and/or dynamic loads. Further, the effects of stress-concentration have also not been taken into account. Keeping these uncertainties in mind, incorporation of high factors of safety can be justified.

Acknowledgments The authors are highly thankful to the students, researchers and interns of the Lighter-Than-Air Systems Laboratory

of Aerospace Engineering Department of IIT Bombay where this work was carried out. We are also thankful to the fellow students at Department of Mechanical Engineering, DCRUST Murthal, who helped indispensably in the brain-storming technical discussions, and also in preparation and uploading of the manuscript. Further, we are highly grateful to IRCC, IIT Bombay and KVPY (which is being administered by IISc Bangalore and funded by DST, Govt. of India) for financially assisting two of the co-authors for the project-work carried out at LTA Systems Lab, IIT Bombay, and the last, but not the least, we are also highly thankful to Mr. Nilesh Dhanve, LTA Systems Lab, Department of Aerospace Engineering, IIT Bombay for helping indispensably in completing all the official formalities and arranging the funds for the project.

References

1. D. Howe, Mooring, in *Airship Technology*, 2nd edn., Cambridge Aerospace Series: 10, ed. by G.A. Khoury (Cambridge University Press, London, 2012), pp. 258–277
2. S.M. Kale, R.S. Pant, Structural design of mooring mast for a remotely controlled airship, in *Proceedings of the 5th International Convention of The Airship Association*, 2004, Oxford, England
3. A. Goyal, D. Pandey, R.S. Pant, Design of a portable mooring mast with drag force measuring device for remotely controlled airships, AIAA-2011-6833, in *Proceedings of AIAA's 19th Lighter-Than-Air (LTA) Technology Conference*, 20–22 Sep 2011, Virginia Beach, Norfolk, Virginia
4. S.F. Hoerner, *Fluid Dynamic Drag* (Dr.-Ing. S.F. Hoerner, Midland Park, N.J., 1957), pp. 11.1–11.4
5. R.L. Norton, *Design of Machinery*, 3rd edn. (Tata McGraw Hill, New Delhi, 2005)
6. G.A. Khoury, *Airship Technology*, 2nd edn. (Cambridge University Press, Cambridge, 2012), pp. 25–34. ISBN 978-1-107-01970-6
7. V.B. Bhandari, *Design of Machine Elements*, 8th reprint of 3rd edn. (Tata McGraw Hill Education Private Limited, New Delhi, 2012) ISBN-13: 978-0-07-068179-8
8. G.H. Ryder, *Strength of Materials*, reprint of 3rd edn with SI units (1969) (Macmillan Publishers India Limited, Chennai, 2009) ISBN-13: 978 0333-93536-1
9. D.S. Kumar, *Elements of Mechanical Engineering*, 2nd edn. (S.K. Kataria & Sons, New Delhi, 2008), pp. 467–469
10. R. Singh, *Introduction to Basic Manufacturing Processes and Workshop Technology*, 2nd edn. (New Age International Publishers, New Delhi, 2010), pp. 69–70. ISBN 978-81-224-3070-7
11. P. Liu, G. Fu, L. Zhu, X. Wang, Aerodynamic Characteristics of Airship Zhiyuan-1. J. Shanghai Jiaotong Univ. (Sci.) **18**(6), 679–687 (2013)