Mechanical Control Design Guide

RATIONALE

Mechanical systems for transmitting pilot commands to flight control surfaces, secondary controls, utility controls or servo control actuators are used less as electrical command transmission is used more frequently. But, mechanical transmission of primary, secondary, utility or backup commands is likely to be the appropriate choice for some aircraft of the future. There is a need of having a document which provides recommended practice for design of Mechanical Control Systems; this ARP satisfies that need.

FOREWORD

Mechanical controls have been used since the advent of aviation to transform pilot commands into control surface deflections and secondary and utility control actuation. Mechanical controls were initially used to position control surfaces directly. (Smaller aircraft, currently being developed, use mechanical controls almost exclusively because of reduced complexity, weight and cost, and proven reliability.) But as aircraft size, speed and maneuverability were increased, the power and/or accuracy required to position a control surface increased to a level where the exclusive use of mechanical controls was inadequate. On some aircraft, the pilot forces became too high for pure mechanical control. This resulted in the introduction of servoactuation. As control requirements became more complex, modifiers were installed upstream or integral with the servoactuators to enhance controllability, maneuverability and/or ride comfort. Some of these latter mechanical control systems became relatively complex. Fly-By-Wire (FBW) and Control-By-Wire (CBW) were recognized as alternate solutions to complex mechanical controls. As electronic reliability improved, airframe company confidence in electronic control reached a level where FBW and CBW were introduced and increased in popularity. But, even on FBW and CBW controlled aircraft, mechanical controls are still utilized in a limited way as backup systems or to convert pilot commands into FBW and CBW command signals.

This ARP addresses mechanical controls that are found in aircraft primary, secondary and utility control systems. These control systems typically use cables and pulleys, pushrods and cranks, and combinations of both to transmit manual pilot commands into control surface, secondary and utility control displacements. The mechanical control guidelines provided in this ARP were accumulated from inputs from a number of prime aircraft manufacturers and personal experiences of a number of control system engineers.

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1. SCOPE

This SAE Aerospace Recommended Practice (ARP) provides guidelines for the configuration and design of mechanical control signal transmission systems and subsystems. It is focused on the recommended practices for designing cable and pulley, pushrod and bellcrank and push-pull flexible cable control systems. These systems are typically used in some combination to transmit pilot commands into primary, secondary and utility control system commands (mechanical or electrical) or aircraft surface commands. On mechanically controlled aircraft, most pilot control commands are initiated through cockpit mounted wheels, sticks, levers, pedals or cranks that are coupled by pushrods or links to cable systems. The cable systems are routed throughout the aircraft and terminated in close proximity to the commanded surface or function where cranks and pushrods are again used to control the commanded function.

2. REFERENCES

2.1 Applicable Documents

The following publications form a part of this document to the extent specified herein. The latest issue of SAE publications shall apply. The applicable issue of other publications shall be the issue in effect on the date of the purchase order. In the event of conflict between the text of this document and references cited herein, the text of this document takes precedence. Nothing in this document, however, supersedes applicable laws and regulations unless a specific exemption has been obtained.

Requirements from many standards and regulations are listed throughout the body of this ARP for the reader's convenience. Before these requirements are used for design, the current requirements in the governing revision of the referenced standard should be checked.

2.1.1 SAE Publications

Available from SAE International, 400 Commonwealth Drive, Warrendale, PA 15096-0001, Tel: 877-606-7323 (inside USA and Canada) or 724-776-4970 (outside USA), www.sae.org.

AS21150 Bearing, Ball, Rod End, Double Row, Precision, Solid Shank, Self-Aligning, Airframe, Type I, -65 to 300 °F

AS21151 Bearing, Ball, Rod End, Double Row, Precision, External Thread, Self-Aligning, Airframe, Type II, -65 to 300 °F

AS21152 Bearing, Ball, Rod End, Double Row, Precision, Hollow Shank, Self-Aligning, Airframe, Type III, -65 to 300 °F

AS94900 Aerospace - Flight Control Systems - Design, Installation and Test of Piloted Military Aircraft, General Specification For

2.1.2 EASA Publications

Available from European Aviation Safety Agency, Postfach 10 12 53, D-50452 Koeln, Germany, Tel: +49-221-8999-000, www.easa.eu.int.

CS-23 Certification Specifications for Normal, Utility, Aerobatic and Commuter Aeroplanes

CS-25 Certification Specifications for Large Aeroplanes

CS-27 Certification Specifications for Small Rotorcraft

CS-29 Certification Specifications for Large Rotorcraft

2.1.3 FAA Publications

Available from Federal Aviation Administration, 800 Independence Avenue, SW, Washington, DC 20591, Tel: 866-835-5322, www.faa.gov.

AC 23.1309 Advisory Circular - System Safety Analysis and Assessment for Part 23 Airplanes

AC 25.1309 Advisory Circular - System Design and Analysis

AC27-!B Advisory Circular- Certification of Normal Category Rotorcraft

AC29-2C Advisory Circular- Certification of Transport Category Rotorcraft

14 CFR 23 Airworthiness Standards: Normal, Utility, Acrobatic and Commuter Category Airplanes

14 CFR 25 Airworthiness Standards: Transport Category Airplanes

14 CFR 27 Airworthiness Standards: Normal Category Rotorcraft

14 CFR 29 Airworthiness Standards: Transport Category Rotorcraft

2.1.4 Joint Aviation Authorities Committee Documents

Available from Global Engineering Documents, 15 Inverness Way, Englewood, CO 80112, Tel: 800-854-7179, www.global@ihs.com.

JAR-23 Joint Airworthiness Requirements, Normal, Utility, Aerobatic and Commuter Category Aeroplanes

JAR-25 Joint Airworthiness Requirements, Large Aeroplanes

JAR-27 Joint Airworthiness Requirements, Small Rotorcraft

JAR-29 Joint Airworthiness Requirements, Large Rotorcraft

2.1.5 National Aerospace Standards

Available from Aerospace Industries Association, 1000 Wilson Boulevard, Suite 1700, Arlington, VA 22209-3928, Tel: 703-358-1000, www.aia-aerospace.org.

NAS428 Bolt, Machine, 125,000 PSI - Crowned Hexagon Head, Adjusting

NAS509 Nut, Drilled Jam

NAS513 Rod End Locking Washer

NAS537 Bushing, Sleeve - Press Fit Undersize Inside Diameter

NAS538 Bushing-Flanged, Press Fit Undersize Inside Diameter

NAS559 Lock - Rod End, Key Type

NAS620 Washer, Flat, Reduced Outside Diameter

NAS623 Screw, Machine, Aircraft, Pan Head, Cross Recess

NAS1149 Washer, Flat, Airframe

NAS1193 Locking Device, Positive Index, Rod End

NAS1368 Grommet - Plastic Flip Type

NAS6203 Bolt, Hex Head, Close Tolerance,

NAS6204 Alloy Steel, Short Thread with

NAS6205 .1875, .2500 and .3125 Diameter Shanks

NASM21042 Nut, Self-Locking, 450°F, Reduced Hexagon, Reduced Height

NASM21209 Insert, Screw Thread, Coarse and Fine, Screw Locking, Helical Coil

2.1.6 Military Guide Specifications

Available from ASC/ENSI, Building 560, 2530 Loop Rd. West, Wright-Patterson AFB, OH 45433-7101, Tel: 937-255-6296.

AFSC-DH1-5 Air Force Systems Command Design Handbook - Electromagnetic Interference

AFSC-DH2-2 Air Force Systems Command Design Handbook - Crew Station Accommodations

JSSG-2006 Joint Service Specification Guide - Aircraft Structures

2.1.7 U.S. Government Publications

Available from the Document Automation and Production Service (DAPS), Building 4/D, 700 Robbins Avenue, Philadelphia, PA 19111-5094, Tel: 215-697-6257, http://assist.daps.dla.mil/quicksearch/.

AN665 Threaded, Clevis, Tie Rod, Terminal

MIL-B-8584 Brake Systems, Wheel, Aircraft Design of

MIL-DTL-5541 Detail Specification, Chemical Conversion Coatings on Aluminum and Aluminum Alloys

MIL-DTL-5688 Detail Specification, Wire Rope Assemblies; Aircraft, Proof Testing and Prestretching of

MIL-DTL-6117 Detail Specificatin, Terminal, Wire Rope Assemblies, Swaged Type

MIL-DTL-83420 Detail Specification, Wire Rope, Flexible, for Aircraft Control

MIL-DTL-87218 Detail Specification, Cable, Lockclad, for Aircraft Control

MIL-F-8785 Flying Qualities of Piloted Airplanes

MIL-PRF-7958 Military Performance Specification, Controls Push-Pull, Flexible and Rigid

MIL-PRF-16173 Military Performance Specification, Corrosion Preventive Compound, Solvent Cutback, Cold Application

MIL-PRF-23827 Military Performance Specification, Grease, Aircraft

MIL-STD-203	Aircrew Station Controls and Displays: Location, Arrangement and Actuation of, for Fixed Wing Aircraft
MIL-STD-461	Interface Standard - Requirements for the Control of Electromagnetic Interference Characteristics of Subsystems and Equipment
MIL-STD-464	Interface Standard - Electromagnetic Environmental Effects Requirements for Systems
MIL-STD-1472	Design Criteria Standard - Human Engineering
MS21251	Turnbuckle Body, Clip Locking
MS21256	Clip, Locking, Turnbuckle
MS21259	Terminal, Wire Rope, Swaging, Stud, Long Thread

2.2 Abbreviations, Acronyms, Symbols

AFCS	Automatic Flight Control System
AFSC	Air Force Systems Command Design Handbook
ARP	Aerospace Recommended Practice
CBW	Control-By-Wire
CFR	Code of Federal Regulations
CG	Center of Gravity
CM	Centimeters
CRES	Corrosion Resistant Steel
CTR	Cable Tension Regulator
D, d, dia	Diameter
EA	Elongation
EASA	European Aviation Safety Agency
F	Friction
FAA	Federal Aviation Administration
FBW	Fly-By-Wire
FCS	Flight Control System
FMECA	Failure Mode, Effect and Criticality Analysis
FTA	Fault Tree Analysis
F/T	Friction, Tension ratio
g	Gravitational Acceleration
in-oz	Inch-Ounce (Torque)
in-lb	Inch-Pound (Torque)
JSSG	Joint Service Specification Guide
JAR	Joint Aviation Authorities, Joint Airworthiness Requirements
LBS	Pounds
LRU	Line Replaceable Unit
M	Meters
MBS	Minimum Breaking Strength
MFCS	Manual Flight Control System
MIL	Military
N	Newtons
NAS	National Aerospace Standard
OD	Outside Diameter
OS	Operational State
PCU	Power Control Unit
PFCS	Primary Flight Control System
PTFE	Polytetrafluoroethylene
RA	Roughness Attribute

RSS	Root Sum of Squares
SCS	Secondary Control System
T	Tension
TED	Trailing Edge Down
TEU	Trailing Edge Up
TSO	Technical Standard Order
UCS	Utility Control System
14 CFR	Title 14 of the Codes of Federal Regulations

3. SYSTEM CONSIDERATIONS

3.1 Control System Classifications

Aircraft control systems are classified into three categories:

- Primary Flight Control Systems (PFCS) - See AS94900 for definition
- Secondary Control Systems (SCS) - See AS94900 for definition
- Utility Control Systems (UCS)

3.1.1 Primary Flight Control Systems (PFCS)

These systems are divided into:

a. Manual Flight Control Systems (MFCS): See ARP4386 for definition.

b. Automatic Flight Control Systems (AFCS): See ARP4386 for definition.

3.1.2 Secondary Control Systems (SCS)

Secondary controls are flight controls that are typically not assigned a primary control function, such as flaps, slats, speed brakes, and spoilers.

3.1.3 Utility Control Systems (UCS)

Utility controls are typically such functions as landing gear and arresting hooks. Brakes and steering may be considered either a secondary or utility control.

3.1.4 Flight Control System Operational State Classifications

These are Military classifications. Transport, Normal, Utility and Commuter Category Airplanes, Normal and Transport Category Rotorcraft usage is recommended. For mechanical flight control systems on EASA/FAA/JAR category airplanes, refer to CS-23/ -25/ -27/ -29 or 14CRF Part 23/ Part 25/ Part 27/ Part 29 or JAR-23/ -25/ -27/ -29 requirements.

a. Operational State I (Normal Operation): See AS94900 for definition.

b. Operational State II (Restricted Operation): See AS94900 for definition.

c. Operational State III (Minimum Safe Operation): See AS94900 for definition.

d. Operational State IV (Controllable to an Immediate Emergency Landing): See AS94900 for definition.

3.1.5 Flight Control System Criticality Classifications

These are Military classifications. Transport, Normal, Utility and Commuter Category Airplanes and Transport and Normal Rotorcraft usage is recommended. (Refer to CS/14CFR or JAR requirements.)

a. Essential: See AS94900 for definition.

b. Flight Phase Essential: See AS94900 for definition.

c. Non-critical: See AS94900 for definition.

3.2 Control System Operational Requirements

3.2.1 Control System Redundancy

In general, redundancy is designed into the system as necessary to provide safe flight and landing following single and/or multiple failures. Typically, mechanical control of pitch and roll for CS-25, 14CFR 25, JAR-25 Category Aircraft is redundant. Mechanical control of yaw may be redundant or simplex. See CS/14CFR/JAR or AS94900, for minimum redundancy requirements.

Pitch and roll mechanical controls are typically configured redundant to provide fail operative control capability after a mechanical jam or open. There are usually two parallel control systems consisting of cable, pushrod, and bellcrank runs. The two parallel control systems are coupled in the cockpit and at the fuselage station where the control systems divert laterally to control either the ailerons or the elevators. Both couplings are usually manually overridable. The fore and aft couplings allow either pilot to command all of the control surfaces if one of the parallel cable or pushrod systems fails open. Likewise, either pilot can override one or both of the couplings if the parallel system jams, allowing continued fail operative control, usually with 50% control authority. Some couplings reset and recouple automatically. Others have to be accessed and mechanically recoupled by maintenance. The latter is usually preferred because a jam occurs very infrequently. It is prudent to perform inspection and maintenance of the control system after a jam to minimize the potential of a jam reoccurrence.

3.2.2 Vulnerability

Vulnerability of essential flight control systems shall be minimized. Design must limit the degradation of these systems from any cause including those listed in Table 1. See CS/14 CFR/JAR and AS94900 for the requirements.

TABLE 1 - DESIGN REQUIREMENTS FOR ESSENTIAL FLIGHT CONTROL SYSTEMS

Factor	Cause and Effect	Design Requirements
Natural Environment	Pressure cycling Icing Condensation and runoff Structural Deflections Temperature variations Gusts and turbulence	No permanent degradation of performance except at adverse extremes of the operational envelope (not normally encountered) where temporary degradation to OS II is allowed. 14CFR 25.341 or EASA CS25.341
Bird Strike	Cockpit penetration Control system damage	Prevent damage or protect control system elements for continued safe flight. 14CFR 25.631 or EASA CS25.631
Induced Environment	Control systems and other systems generation of: Temperatures, Thermal shocks Vibrations and buffeting, Noise, Shocks Induced pressures Electrical or magnetic emissions or potentials	Prevent or protect Flight Control System from performance degradation. Military requirements - Commercial/business reference: MIL-STD-464A JSSG-2006 Guidance 14CFR 25.251 or EASA CS25.251 MIL-STD-461 14CFR 25.365 or EASA CS25.365 AFSC-DH 1-5, DN IBI and DNICI Guidance
Lightning and Static Electricity	Electric potential buildup and discharge	Prevent performance degradation below OS II, except temporary recoverable loss to OS III is allowed for a direct lightning strike. Military requirements - Commercial/business reference. MIL-STD-464 14CFR 25.581 or AFSC DH 1-5 Guidance EASA C25.581
Flight Crew Inaction or Error	Mispositioned or maladjusted controls	Prevent improper position or sequencing of controls by positive action gates, warnings, making pilot reactions to failure instinctively correct.
Maintenance Error	Improper assembly or installation	Prevent installation errors by requiring two or more overt actions or modifications. Design LRUs (Line Replaceable Units) with bench adjustment only and replaceable without rerigging of the system. Minimize possibilities of jamming from mislaid tools or equipment.
Control System Component Failures	Jam or severance of system Loss of hydraulic system Loss of electrical power	Aircraft performance shall not be degraded below OS III. Failure analyses usually determines redundancy requirements.
Other System or Equipment Failures	Damage to FCS components from any failure	Prevent FCS degradation below OS III.
Loss of Main Landing Gear	Excessive aft or up load	Complete lateral control after failure. For example: On some aircraft - Route lateral controls over gear trunnion near rear spar as forward trunnion bearing is fused to fail with an excessive gear aft load. Route lateral controls under the outboard end of the landing gear beam which is fused to fail with excessive up load.
Engine Burst	Fragments of turbine or compressor discs are sprayed across the width of the airplane. Spray angles and maximum fragment sizes are defined by Propulsion Staff	FCS must be routed and designed to minimize damage from single fragment in engine burst zone: 1. Limit FCS components in the zone to those actually required and separate them as much as possible. 2. Use aircraft structure to shield FCS components as much as possible. 3. Fragment size determines minimum separation of dualized systems. 14CFR25.903 or EASA CS25.903
Structural Failures	Fatigue Explosions Cabin depressurization Etc.	FCS must remain operational if aircraft is structurally capable of flight through such methods as duplication, separation, routing for maximum structural protection, etc.

TABLE 1 - DESIGN REQUIREMENTS FOR ESSENTIAL FLIGHT CONTROL SYSTEMS (CONTINUED)

Factor	Cause and Effect	Design Requirements
Tire Burst in Main Wheel Well	Pressurization of wheel well by small, destructive jet (from tire burst) whose impact pressure levels off about 9 in from the tire.	FCS components in wheel well shall be minimized, separated, screen shielded if within approximately 9 in of any tire, and designed along with their supports, for tire burst impact loads. Prime airframe companies typically have tire burst test data that supports the resolution of local burst issues: 14 CFR 25.729 or EASA CS25.729

3.2.3 Reliability

The reliability and flight safety requirements for the mechanical control system are based on probability of all of those single and combined failures that would cause loss of control of the aircraft. These probabilities of failure are allocated from requirements for the maximum aircraft loss rate from flight control system failures per flight hour. The reliability requirements may require some portions of the mechanical controls to be dual.

Quantitative flight safety probability of loss of control and the equivalent maximum acceptable aircraft loss rate due to relevant flight control system failures is provided in AS94900 for military aircraft and 14CFR 23, 25, 27 or 29 for FAA certified aircraft and CS-23, -25, -27 and -29 for EASA certified aircraft.

3.2.4 Control System Load Requirements

The control system manual load requirements for numerous civil aircraft categories are outlined in the following FAA Federal Aviation Regulations and European Aviation Safety Agency Certification Specifications:

14CFR Part 23 Airworthiness Standards: Normal, Utility, Acrobatic, and Commuter Category Airplanes

CS-23 Certification Specifications for Normal, Utility, Aerobatic and Commuter Category Aeroplanes

14CFR Part 25 Airworthiness Standards: Transport Category Airplanes

CS-25 Certification Specifications for Large Aeroplanes

14CFR Part 27 Airworthiness Standards: Normal Category Rotorcraft

CS-27 Certification Specifications for Small Rotorcraft

14CFR Part 29 Airworthiness Standards: Transport Category Rotorcraft

CS-29 Certification Specifications for Large Rotorcraft

a. Control System Limit/Ultimate Design Loads

Limit Load

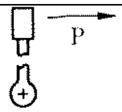
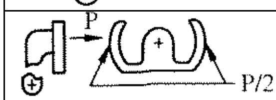
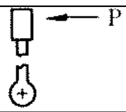
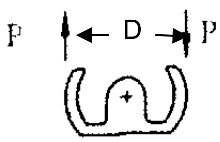
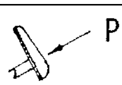
Lateral, Longitudinal, Directional (Primary) = Maximum expected applied load

All Other Controls (Secondary) = Maximum expected applied load

Limit Design Loads of control systems for 14CFR Part 23 / EASA CS-23 Normal, Utility, Acrobatic and Commuter Category Aircraft/Aeroplanes and Part 25 / EASA CS-25 Transport Category Airplanes/Aeroplanes are shown in Table 2.

Ultimate Design Load = 1.5 x limit load

TABLE 2 - 14CFR PART 23 AND 25 /EASA CS-23 AND -25 CONTROL SYSTEM LIMIT DESIGN LOADS

Aircraft Control	Cockpit Control		Limit Design Load		Load Location and Direction
			14CFR Part 23.397 EASA CS23.397 Note 1	14CFR Part 25.397 EASA CS25.397	
Longitudinal		Stick (Floor Mounted)	P = 167 lb or P = 445 N (min) P = 743 N (max)	P = 250 lb or P = 1112 N (max) P = 445 N (min)	A longitudinal force at top of stick and perpendicular to a radial from stick's pivot.
		Wheel	P = 200 lb or P = 890 N (max) P = 178 N (min)	P = 300 lb or P = 1335 N (max) P = 445 N (min)	Two forces on wheel rim, diametrically opposite, acting perpendicular to wheel plane in same direction.
Lateral		Stick (Floor Mounted)	P = 67 lb or P = 298 N (max) P = 178 N (min)	P = 100 lb or P = 445 N (max) P = 178 N (min)	A lateral force at top of stick and perpendicular to radial from stick pivot.
		Wheel	Torque=50XD Inch-Pounds or Torque = 222xD Newton - Meters (maximum) Torque = 178xD Newton - Meters (minimum)	Torque= 80XD Inch-Pounds or Torque = 356xD Newton - Meters (maximum) Torque =178xD Newton- Meters (minimum)	Two tangential forces on wheel rim, diametrically opposite, and acting in opposite direction. The critical parts of the lateral control system must be designed for a single tangential force with a limit value equal to 1.25 times the couple force determined by the specified requirement.
Directional and Brake		Pedals	P = 200 lb or P = 890 N (max) P = 667 N (min)	P = 300 lb or P = 1335 N (max) P = 578 N (min)	Force acting on line connecting pedal foot contact point and hip joint with pilot's seat in mean landing position.
Flap, Tab, Stabilizer, Spoiler, Landing Gear Controls	Crank, wheel, or lever operated by push or pull force		14CFR Part 23.405 or CS23.405 Maximum force that a pilot is likely to apply.	14CFR Part 25.405 $\left[\frac{1+R}{3}\right]$ x 50 but not less than 50 nor more than 150 lb (R = radius in inches) or EASA CS25.405 $\left[\frac{25.4+R}{76.2}\right]$ x222 N but not less than 222 N nor more than 667 N (R = radius in mm)	A force on the circumference of the wheel, or grip of wheel, or lever that acts at any angle within 20 degrees of the plane of the control.
	Small wheel or knob			133 in-lb or 15 N m if operated by twisting	
	Dual Control System		14CFR Part 23.399 or CS23.399	14CFR Part 25.399 or CS25.399	Note 2

NOTE 1: For aircraft design weight more than 5000 lb (2268 kg), the specified maximum values must be increased linearly with weight to 1.18 times the specified values at a design weight of 12 500 lb (5670 kg) and for commuter category aircraft, the specified values must be increased linearly with weight to 1.35 times the specified values at a design weight of 19 000 lb (8618 kg).

NOTE 2: Each dual control system must be designed to withstand the force of the pilots operating in opposition, using individual pilot forces not less than the greater of:

- 0.75 times those obtained under 14CFR Part 23.395 or Part 25.395 or EASA CS23.395 or CS25.395
- the minimum forces specified in 14CFR Part 23.397 or Part 25.397 or EASA CS23.397 or CS25.397.

Each dual control system must be designed for pilot forces applied in the same direction, using individual pilot forces not less than 0.75 times those obtained under 14CFR Part 23.395 or Part 25.395 or EASA CS23.395 or CS 25.395.

b. Control System Operational Forces

1. Maximum Operational Forces: Lateral, longitudinal and directional flight controls have their maximum operational forces specified for each flight condition by MIL-F-8785. FAR 14CFR Part 23.143 and Part 25.143 or EASA CS23.143 and CS25.143 specify the maximum forces permitted during aircraft certification testing as shown in Table 3 For 14CFR Part 23 or EASA CS-23 aircraft, with a stick cockpit control, the maximum operational forces are 60 pounds or 267 N in pitch and 30 lb or 133 N in roll.

TABLE 3 - MAXIMUM OPERATIONAL FORCES

Force, in pounds, applied to the control wheel or rudder pedals	Pitch	Roll	Yaw
For short term application for pitch and roll control - two hands available for control ……............	75 lb or 334 N	50 lb or 222 N	
For short term application for pitch and roll control - one hand available for control ……...............	50 lb or 222 N	25 lb or 111 N	
For short term application for yaw control ……............................			150 lb or 667 N
For long term application …….....	10 lb or 44.5 N	5 lb or 22 N	20 lb or 89 N

The engine thrust levers should have an operational force of about 3 lb or 13.4 N per lever. Pilots consider 5 lb or 22 N objectionable, even though AFSC DH2-2 allows up to 7.5 lb per lever and 30 lb for all engine thrust levers at the same time.

Brake pedal maximum operational force of 85 lb is specified by MIL-B-8584 for a powered brake control system, and 125 lb for a manual brake system.

Maximum operating forces are not specified for other control systems; however, they should be kept low to prevent pilot complaints, but high enough to prevent actuation by vibration or acceleration forces.

2. Breakout Forces and Control Centering - Breakout Forces: Breakout force is that force on the control when it's control surface begins to move, and includes system friction, centering, balance, and feel forces.

The recommended breakout force ranges for the pitch, roll, and yaw flight controls from the trim position are shown in Table 4 for military aircraft. These breakout force levels are also appropriate for FAA/EASA category aircraft.

TABLE 4 - BREAKOUT FORCES

Control		Breakout Force in Pounds			
		MIL-F-8785 Aircraft Classes			
		Class I - Small, Light Class II-C - Carrier, medium weight, low to medium maneuverability Class IV - High maneuverability		Class II-L - Land, medium weight, low to medium maneuverability Class III - Large, heavy, low to medium maneuverability	
		Minimum	Maximum	Minimum	Maximum
Longitudinal	Stick	1/2	3	1/2	5
	Wheel	1/2	4	1/2	7
Lateral	Stick	1/2	2	1/2	4
	Wheel	1/2	3	1/2	6
Directional		1	7	1	14

Ground measurement of breakout forces is usually considered as compliance with the requirements. Breakout forces on the ground shall be measured when the trailing edge of the control surface has visibly moved.

Actual breakout forces in flight are usually lower than those measured on the ground due to aircraft vibration tending to make all friction moving, rather than static.

Centering: With absolute centering, a cockpit control will always return exactly to its trim position when released. Positive centering is a tendency to return: upon release, the control will move toward the trim position, but friction may prevent absolute centering.

Pitch, roll, and yaw controls should show positive centering in flight at any normal trim setting. Although absolute centering is not required, the combined effects of centering, breakout force, stability, and force gradient shall not produce objectionable flight characteristics.

3. Manual Override Forces: It shall always be possible for the pilot to manually overpower or countermand the automatic flight control action using the normal pilot controls. Required pilot forces shall not exceed pilot capability as defined by MIL-STD-1472. Table 5 lists recommended maximum override forces.

TABLE 5 - MANUAL OVERRIDE FORCES

Control		AFCS Operation Maximum Manual Override Force	
		Single Channel Single Servo Engaged	Redundant Channel Servos Engaged in Redundant Configuration
		Pounds	Pounds
Longitudinal	Stick	20	30
	Wheel	30	50
Lateral	Stick	10	20
	Wheel	20	30
Directional	Pedals	80	120

4. Servoactuator Control Valve Chip Shear Forces

A servoactuator slide valve must be capable of shearing a metallic chip which may become lodged across the sleeve port and slide metering edge. The valve and its actuating linkage must sustain a maximum of 200 to 1000 lb or 890 to 4450 N (depending on application/customer requirements) at the valve slide without any damage or deformation, in order to shear the chip. All of the servoactuator linkage and the control system must have adequate stiffness and strength to transmit the valve chip shear force. Damage to the valve and linkage is acceptable, but the valve and linkage must still be capable of full displacement with minimal additional friction. (There are some servoactuator designs that have lower level detents that breakout when a slide valve becomes jammed. The breakout allows continued operation as a function of the PCU override features or by control of an adjacent servoactuator coupled to the same control surface.)

3.3 System Friction and Breakout Force Calculations

System friction is critical for pitch, roll, and yaw flight control systems which have centering and low breakout force requirements. For a system with a centering device, every pound or 4.45 N of system friction, measured under static conditions on the ground, adds approximately 1.7 lb or 7.56 N to the breakout force, the additional 0.7 lb or 3.11 N being required to overcome the centering (detent) force.

Design for a specified system breakout force requires that the friction of system components be minimized.

A careful friction analysis is required to have the actual breakout force within 10% of the calculated breakout force.

System friction is calculated, usually at 70 °F, along with the control centering and any unbalance forces, to determine the breakout force for the system.

Table 6 lists the methods for calculating system friction for system components. Cable routing may be developed by using CATIA (Computer-Graphics Aided Three-dimensional Interactive Application).

TABLE 6 - COMPONENT FRICTION CALCULATION

Component	Calculation Method
Ball Bearing	Use the average no-load breakout torque of bearing or similar bearing in Table 7.
Sheave and Cable (Carbon Steel, Corrosion Resistant Steel or Nylon Jacketed)	Compute friction from ratio *F/T* values from Figure 20 or Figure 21 depending upon groove conformance with cable diameter. Interpolate *F/T* values for wrap angle not shown between wrap angles that are shown.
Fairlead Pulley with Zero Wrap	Compute friction on the assumption that only half of the pulleys are contacted by the cable under two conditions: 1. The fairlead pulley bearing breakout friction. 2. The sliding friction between the cable and a non-rotating fairlead pulley. Assume friction coefficient of 0.04 and the weight of the cable between adjacent sheaves. Take the lesser of the two values and divide by 2 to account for only half of pulleys being contacted.
Rub Strips and Fixed Fairleads	Friction based on maximum anticipated deflection of supporting structure (typically wing bending) that induces cable deflection. Compute cable friction using cable tension induced side load and 0.04 friction coefficient.
Seal and Cable Eyeball Pressure Seal Douglas Grommet Firewall Seal	Compute friction using friction coefficient of 0.04 and the weight of the cable between adjacent sheaves.
Hydraulic Control Valve	Use the breakout friction called out in the specification or functional test for the valve or similar valve.

Table 7 gives average bearing friction breakout torques of commonly used bearings at 70 °F. These no-load breakout values should be used for large systems with many bearings. If the system is friction critical with a small number of bearings, these values should be increased by 25 to 50%.

TABLE 7 - AFBMA (ANTI-FRICTION BEARING MANUFACTURER ASSOCIATION)
BEARING AVERAGE NO-LOAD BREAKOUT TORQUE AT 70 °F 1/

AFBMA No.	SAE/MIL-Spec Designation	Average No-Load Breakout Torque		AFBMA No.	SAE/MIL-Spec Designation	Average No-Load Breakout Torque	
		in-oz	Gram - CM			in-oz	Gram - CM
AS/MS27640				AS/MS27641			
KP3	-3	0.09	6.5	KP3A	-3	0.08	5.8
KP4	-4	0.45	32.4	KP4A	-4	0.16	11.5
KP5	-5	0.40	28.8	KP5A	-5	0.23	16.6
KP6	-6	0.58	41.8	KP6A	-6	0.74	53.3
KP8	-8	1.06	76.4	KP8A	-8	1.30	93.6
KP10	-10	1.28	92.2	KP10A	-10	1.45	104.4
AS/MS27642				KP12A	-12	1.54	110.9
KP21B	-21	5.38	387.4	KP16A	-16	3.01	216.7
KP25B	-25	6.01	432.8	KP20A	-20	3.88	279.4
KP29B	-29	7.02	505.5	MKP4A	-4R	0.16	11.5
KP33B	-33	8.04	579.0	MKP5A	-5R	0.21	15.1
KP37B	-37	9.43	679.0	MKP12A	-12R	1.33	95.8
KP47B	-47	12.15	874.9	AS/MS21151			
KP49B	-49	13.42	966.4	REP4M6	-8	0.47	33.9
AS/MS27647				AS/MS21443			
DW4K	-4	0.80	57.6	P4K	-4B	0.20	14.4
DW4		0.85	61.2	P5K	-5A	0.25	18.0
DW5	-5	0.90	64.8	PD5K	-5B	0.50	36.0
DW6	-6	1.10	79.2	P8	-8	1.30	93.6
				P10K	-10	0.80	57.6

1/ The average radial no-load breakout test results of multiple bearings of each size by bearing and airframe manufacturers. If the control installation is subjected to misalignment by design or installation deflections, the misalignment bearing torque needs to be accounted for. The misalignment torque can be several times greater than the radial no-load frictional torque.

Tables 8 and 9 show a friction analysis for a hypothetical elevator control system which consists of:

1. An isometric diagram of the elevator control system showing all of the friction producing components of the system. - Figure 1.

2. A data sheet tabulating the friction of each component in the system. - Table 8.

3. Three Table 9 calculation sheets showing the typical computations for the friction of each component of the system, the resultant centering detent, and the total breakout force.

The maximum total breakout force should not exceed 7 lb or 31.1 N based on MIL-F-8785 criteria. Most aircraft manufacturers prefer a maximum total breakout force of 5 lb or 22.2 N. Based on this latter criteria, the centering detent should be greater than total friction plus column unbalance (i.e., 2.5 lb or 11.12 N max), but not less than 70% of the total friction with no column unbalance. If friction cannot be minimized to achieve this condition, consideration should be given to the use of cable tension regulators which reduce friction by permitting a lower rigging load. This hypothetical elevator control system utilizes tension regulators with a 70 lb or 311 N cable preload.

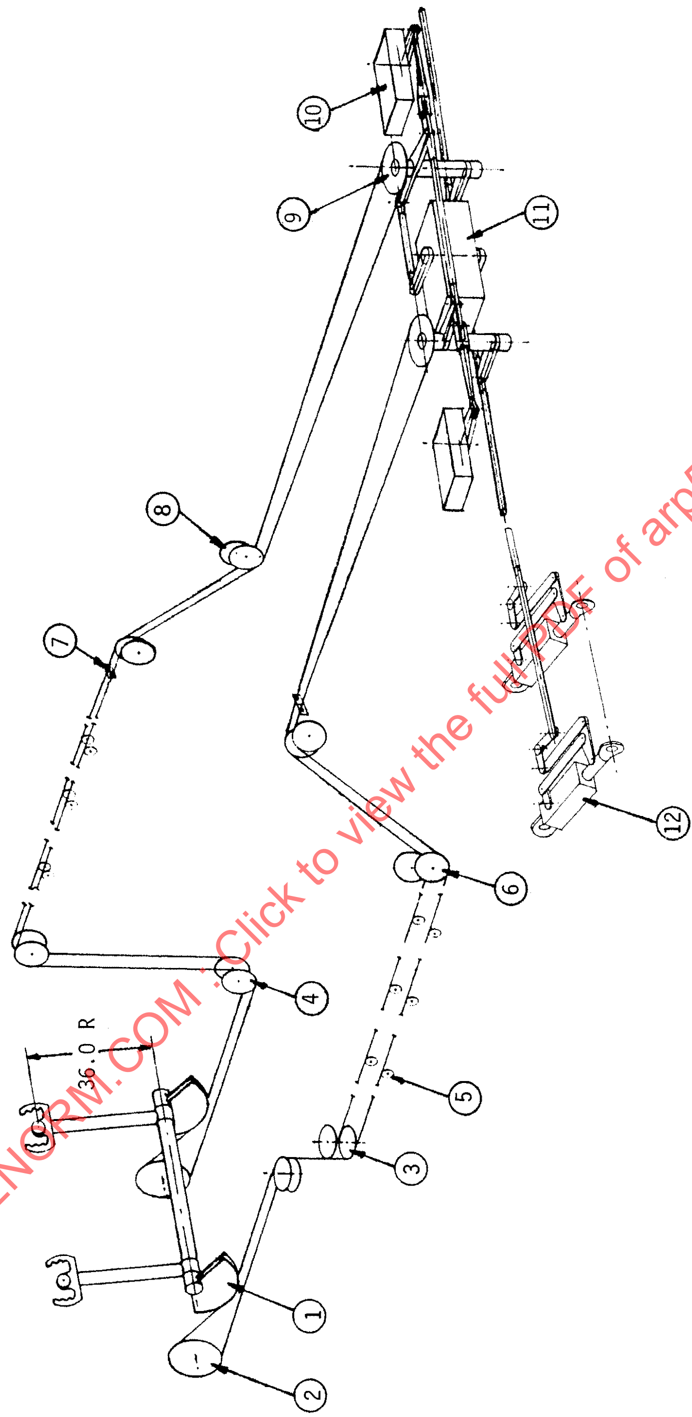


FIGURE 1 - CHARACTERISTIC TRANSPORT AIRCRAFT
ELEVATOR CONTROL SYSTEM ISOMETRIC

TABLE 8 - CHARACTERISTIC ELEVATOR CONTROL SYSTEM COMPONENT FRICTION ANALYSIS

Item	Item	No. of Items	Friction at Column Pounds	Friction at Column Newtons
1	Control Columns	2	0.155	0.689
2	Forward Pulleys	2	0.051	0.227
3	Intermediate Pulleys - Pilot Side	4	0.109	0.485
4	Intermediate Pulleys - Copilot Side	4	0.117	0.520
5	Fairlead Pulleys	24	0.036	0.160
6	Intermediate Pulleys - Pilot Side	4	0.109	0.485
7	Eyeball Pressure Seals	4	0.027	0.120
8	Intermediate Pulleys - Copilot Side	4	0.097	0.431
9	Rear Quadrants	4	0.077	0.342
10	Pitch Autopilot Actuators	2	0.196	0.872
11	Force Feel Unit	1	0.185	0.823
12	Servoactuator Linkage and Valves	4	1.584	7.046
	Subtotal:		2.743 lb	12.200 N
	70% Detent:		1.920 lb	8.540 N
	Total Pitch Axis Breakout:		4.66 lb	20.74 N

Examination of the summary of calculated component friction levels highlights the magnitude of the friction associated with control assemblies such as the force feel unit, pitch autopilot actuators, and servoactuators.

TABLE 9 - CHARACTERISTIC ELEVATOR CONTROL SYSTEM
COMPONENT FRICTION ANALYSIS

Control Column Cable Quadrants	-	10.00R (20.00 pitch diameter)
Forward Pulleys and Rear Quadrants	-	12.00 pitch diameter
Intermediate Pulleys	-	8.00 pitch diameter
Idler Pulleys	-	3.00 pitch diameter
Cable Diameter	-	0.125 dia
Cable Tension	-	70.0 lb

1. Control Columns

2 - Column Boots	-	0.45 lb at 4.00R
2 - Column Bearings	-	KP8A - 1.30 in-oz
2 - Column Quadrant Assemblies	-	20.00 dia x 30 degree wrap (2 wraps per quadrant assembly)
4 - Remove PD5K Pulley Bearing Friction	-	0.50 in-oz

Quadrant $D/d = (20 - 0.125)/0.125 = 159,\ F/T = 0.0007$

$$\text{Friction} = \frac{2 \times 0.45 \times 4}{36} + \frac{2 \times 1.30 - 4 \times 0.5}{36 \times 16} + \frac{4 \times 0.0007 \times 70 \times 10}{36}$$

$$\text{Friction} = 0.1000 + 0.0010 + 0.0544 = 0.155 \text{ lb} \ (0.689 \text{ N})$$

2. Forward Pulleys

2 - Pulleys - 12.0 pitch dia x 210 degree wrap

$D/d = (12 - 0.125)/0.125 = 95,\ F/T = 0.0012 \times 1.08 = 0.0013$ (Extrapolated from 180 to 210 degrees)

$$\text{Friction} = \frac{2 \times 0.0013 \times 70 \times 10}{36} = 0.051 \text{ lb} \ (0.227 \text{ N})$$

3. Intermediate Pulleys - Pilot Side

4 - Pulleys - 8.0 pitch dia x 60 degree wrap

$D/d = (8 - 0.125)/0.125 = 63,\ F/T = 0.0014$

$$\text{Friction} = \frac{4 \times 0.0014 \times 70 \times 10}{36} = 0.109 \text{ lb} \ (0.485 \text{ N})$$

4. Intermediate Pulleys - Copilot Side

4 - Pulleys - 8.0 pitch dia x 90 degree wrap

$D/d = 63,\ F/T = 0.0015$

$$\text{Friction} = \frac{4 \times 0.0015 \times 70 \times 10}{36} = 0.117 \text{ lb} \ (0.520 \text{ N})$$

Table 9 - Sheet 1

5. Fairlead Pulleys - Select lesser value of sliding or pulley bearing friction assuming one-half of pulleys contacted.
 24 - Pulleys - 3.0 pitch dia x 0 degree wrap
 24 - Pulley Bearings - P5K - 0.25 in-oz

$$\text{Bearing Friction} = \frac{0.25}{16} \times \frac{2}{3.00 - 0.125} \times \frac{24}{2} \times \frac{10}{36} = 0.036 \text{ lb } (0.160 \text{ N})$$

Sliding Friction

$\mu = 0.04$

Weight of jacketed cable = 3.62 lb/100 ft

Average weight/every other pulley - $\frac{3.62}{100} \times 16 = 0.579$ lb

$$\text{Friction}_S = 12 \times 0.579 \times 0.04 \times \frac{10}{36} = 0.077 \text{ lb } (0.342 \text{ N})$$

Use the bearing friction = 0.036 lb (0.160 N)

6. Intermediate Pulleys - Pilot Side
 4 - Pulleys - 8.0 pitch dia x 60 degree wrap
 $D/d = (8 - 0.125)/0.125 = 63, \; F/T = 0.0014$

$$\text{Friction} = \frac{4 \times 0.0014 \times 70 \times 10}{36} = 0.109 \text{ lb } (0.485 \text{ N})$$

7. Eyeball Pressure Seals
 4 - Seals - Friction Coefficient - 0.04
 Weight of 200 in of cable at 3.62 lb/100 ft - 0.6033 lb

$$\text{Seal Friction - } 4 \times 0.04 \times 0.6033 \times \frac{10}{36} = 0.027 \text{ lb } (0.120 \text{ N})$$

8. Intermediate Pulleys - Copilot Side
 4 - Pulleys - 8.0 pitch dia x 30 degree wrap
 $D/d = 63, \; F/T = 0.00125$

$$\text{Friction} = \frac{4 \times 0.00125 \times 70 \times 10}{36} = 0.097 \text{ lb } (0.431 \text{ N})$$

9. Rear Quadrants
 2 - Quadrant Assemblies - 12.0 pitch dia x 28 degree wrap (2 wraps per quadrant assembly)
 4 - Pivot Bearings - KP8A - 1.30 in-oz
 4 - Remove PD5K Pulley Bearing Friction - 0.50 in-oz
 10 - Rod End Bearings on 8 in cranks - REP4M6 - 0.47 in-oz
 Quadrant $D/d = 95, \; F/T = 0.0007$

$$\text{Friction} = \frac{4 \times 0.0007 \times 70 \times 10}{36} + \frac{4 \times 1.30 - 4 \times 0.50 + 10 \times 0.47}{16 \times 6} \times \frac{10}{36} = 0.077 \text{ lb } (0.342 \text{ N})$$

Table 9 - Sheet 2

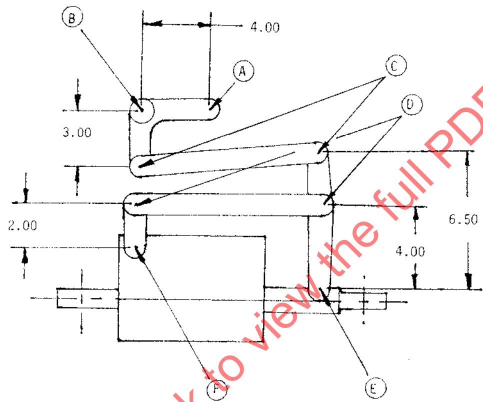
10. Pitch Autopilot Actuators
    2 - Actuators - 34 in-oz breakout friction with 8.0 crank

$$\text{Friction} = \frac{2 \times 34 \times 8}{16 \times 8} \times \frac{10}{6} \times \frac{1}{36} = 0.196 \text{ lb } (0.872 \text{ N})$$

11. Force Feel Unit
    1 - Feel Unit
    2 - Inputs - Internal friction at breakout - 2 in-lb

$$\text{Friction} = \frac{2 \times 2}{8} \times 8 \times \frac{10}{6} \times \frac{1}{36} = 0.185 \text{ lb } (0.823 \text{ N})$$

12. Servoactuator Linkage and Valve
    4 - Servoactuators/ Aircraft



Servoactuator

A - REP4M6 Bearing	0.47 in-oz
B - 2 MKP5A Bearings	0.42 in-oz
C - 2 MKP4A Bearings	0.32 in-oz
D - 2 MKP4A Bearings	0.32 in-oz
E - 2 MKP4A Bearings	0.32 in-oz
F - Valve Breakout Friction	72 in-oz

$$\text{Friction}_{\text{ACTUATOR}} = \left[\frac{0.47}{16 \times 4} + \frac{0.42}{16 \times 4} + \frac{0.32}{16 \times 4} + \frac{0.32 \times 3}{16 \times 6.5 \times 4} + \frac{0.32 \times 3}{16 \times 6.5 \times 4} + \frac{0.32 \times 3}{16 \times 6.5 \times 2} + \frac{0.32 \times 3}{16 \times 6.5 \times 4} + \frac{72 \times 4 \times 3}{16 \times 2 \times 6.5 \times 4}\right]$$

$$\text{Friction}_{\text{ACTUATOR}} = 0.0073 + 0.0066 + 0.0050 + 0.0023 + 0.0023 + 0.0046 + 0.0023 + 1.0385$$

$$\text{Friction}_{\text{ACTUATOR}} = 1.069 \text{ lb}$$

$$\text{Friction}_{\text{COLUMN}} = 4 \times 1.069 \times \frac{8}{6} \times \frac{10}{36} = 1.584 \text{ lb } (7.046 \text{ N})$$

Table 9 - Sheet 3

3.4 Control System Positionability/Threshold

The ultimate quality of a control system is determined by the ability of the pilot/copilot to accurately position the commanded surfaces with minimum overshoot/undershoot and maintain that position as commanded. The pilot should be able to command a surface response with relative ease and achieve that response consistently. Positionability/Threshold, in itself, is not a mechanical control system design criteria. Surface positionability/threshold is a by-product of the control inceptor travel range, control system stiffness, backlash, friction, and control forces. There are numerous additional control system factors that impact positionability such as the characteristics of the control surface and/or the servoactuator that positions that surface, servoactuator backup structural stiffness, and aircraft moment of inertia and dynamic response. These latter topics are not the subject of this ARP and are not addressed.

Typically, control system positionability is most critical when the control surface effectivity is the highest and the pilot commands a specific flight attitude, maneuver or response level. These are all general terms and subject to a lot of interpretation. Positionability improves as the control inceptor travel range and system stiffness increase and the backlash and friction decrease.

The pitch axis is typically the most control critical axis. Pitch control is most critical when the aircraft is at high airspeed with an aft center of gravity (CG) and low gross weight. This is the condition where the elevator/canard effectiveness is the highest and where control positionability/threshold is most critical. This ARP will focus on the pitch axis. A general guideline (for some airframe companies) that addresses positionability/threshold in the pitch axis is the 1/20 rule. The pilot should be able to position the elevators or canards accurately and have the aircraft maintain the commanded flight path within ±1/20 of a “g” acceleration. The ability to maintain a commanded pitch attitude within a ±1/20 g band is influenced by the factors noted above. Each will be addressed in subsequent paragraphs.

a. Control Inceptor Travel

The cockpit command inceptor travel range should be large enough so it is not too sensitive, yet not so large that it results in uncomfortable pilot motion or requires unusual manual skill.

Typical inceptor travels for mechanical control systems are:

Pitch Axis

Control Column (Wheel or Stick)

Forward no-load travel from neutral (TED)	-	4.0 to 4.5 in	10.2 to 11.4 cm
Forward travel to column stop	-	5.0 to 5.5 in	12.7 to 14.0 cm
Aft no-load travel from neutral (TEU)	-	5.5 to 6.0 in	14.0 to 15.2 cm
Aft travel to column stop	-	6.5 to 7.0 in	16.5 to 17.8 cm

Roll Axis

Wheel

Clockwise and counterclockwise rotation	-	±60 to ±70 degrees	
CW and CCW travel to wheel stops		±80 to ±85 degrees	

Stick

Right and left displacement	-	±5 to ±6 in	12.7 to 15.2 cm
Right and left travel to stick stops		±6 to ±7 in	15.2 to 17.8 cm

Yaw Axis

Pedals

Pedals displacement from neutral	-	±2 to ±3.5 in	5.1 to 8.9 cm
Pedal travel to pedal stops		±3 to ±4.5 in	7.6 to 11.4 cm

b. Control Forces and Friction

Control system operational forces and friction have been addressed in detail in 3.2.4.

c. Control System Backlash

Control system backlash has a detrimental effect on the ability of the pilot/autopilot to control the aircraft flight path. In a mechanical control system with a single load path, backlash is generated at every pushrod and lever joint. Examples of mechanical features that generate backlash are:

- Pivot bearing radial play

- Rod end radial play

- Pivot joint radial clearances

- Rod end joint radial clearances

All of these contributors to backlash can be minimized by the use of precision bearings, bolts, and fitting/bellcrank bushings. Bearing radial play can be specified and controlled to a low level, but not eliminated. Clamped pivot joints reduce the backlash under normal radial loads but can shift and ultimately loosen under repeated high loads. Typically, backlash increases as the aircraft ages. Control loads and vibration tend to increase joint clearances over time, resulting in increased overall backlash.

On some aircraft, the pushrods and levers are dualized with identical parallel paths and preloaded against one another to eliminate the control freeplay associated with backlash. Dualized control systems require careful design because the duality raises the overall friction level. The friction level is amplified by the pushrod preload force level. If the duality is installed downstream of the feel force generator, the preload can be controlled at a lower level and the friction can be minimized. Preload springs have been installed on some aircraft to eliminate the effects of backlash. Preload springs are usually not a wise choice because a preload spring failure can induce uncommanded surface movement if the springs are installed at the extreme ends of a pushrod run in opposition to one another. On some aircraft, a cable system is installed in parallel with the mechanical linkage to preload out the linkage backlash. This is usually an effective approach because a cable break or a pushrod/lever open failure results in minimal uncommanded surface movement.

The backlash associated with pushrods and cranks highlights one of the advantages of a preloaded cable system. The ideal mechanical control system utilizes as much cable control transmission as possible, limiting the pushrods and cranks to the immediate areas of the control input inceptor, autopilot, damper, feel unit, and surface/servoactuator interfaces.

d. Control System Stiffness

Control cable installation stiffness mathematics is addressed in some detail in 5.1.h. But the cable installation stiffness required for control system positionability/threshold has not been addressed. Likewise, pushrod and crank stiffness have not been addressed.

e. Control System Positionability/Threshold Analysis

The hypothetical elevator control system presented isometrically in Figure 1 and analyzed for friction in Tables 8 and 9 provides a basis for positionability/threshold evaluation. For this analysis it is assumed that the maximum surface effectivity with an aircraft aft center of gravity, low gross weight and high airspeed is 1.2 degrees of elevator per delta 'g'. It is also assumed that the control column no-load displacement is 5.75 in aft and 4.25 in forward from neutral, with resultant elevator displacements of 27 degrees TEU and 20 degrees TED. The elevator control system linkage rise and fall is split with ±5.00 in at the control column, the surface travel is split ±23.5 degrees and the assumed elevator surface horn radius of 5.541 in generates a split output travel of ±2.2095 in.

If 1.2 degrees of elevator equates to one delta 'g', then 0.06 degrees equates to 1/20 of a 'g' or 0.0058 in of servoactuator output. The rear quadrant coupled feel unit serves as a mechanical detent for the elevator control surfaces. The stiffness between the rear quadrant mounted feel and centering unit and the servoactuator output with the reflected local friction augmented by the local backlash define the ranges of a hysteresis loop when subjected to normally encountered aerodynamic variations. The local servoactuator friction per the analysis of Table 9 is 1.069 lb per servoactuator. For two elevator servoactuators, the input friction is 2.138 lb. The linkage bearing friction between the feel unit and the two servoactuators is:

Linkage Bearing Friction = $5 \times 0.47/16/8 = 0.018$ lb

Total servoactuator Input Reflected Friction = $2.138 + 0.018 = 2.156$ lb

The pushrods between the feel unit and the mid-position between the two servoactuators are assumed to be 50 in long with an installed stiffness of 2310 lb/in. The ability of the control system to maintain a fixed elevator position is a function of the local hysteresis loop defined partially by the ratio of local friction to local stiffness referenced to a detent. For this example that ratio per elevator is:

Ratio = $\pm 2.156/2310 = \pm 0.00093$ in at the servoactuator input

There are five precision, reduced backlash anti-friction bearings (with 0.0004 in radial play each) in the feel unit to servoactuator input control linkage with a combined absolute maximum backlash of 0.0020 in. The accepted practice is for the aircraft manufacturer to RSS all of the individual airframe contributors of backlash, resulting in 0.00159 in mean effective input pushrod backlash.

The servoactuator manufacturer is responsible for the servoactuator backlash. The servoactuator linkage has an absolute maximum backlash of 0.002800 in reflected at the servoactuator input. Again, the accepted practice is to RSS all of the individual servoactuator contributors to backlash, resulting in 0.00228 in mean effective servoactuator input backlash. The total backlash reflected at the servoactuator input is 0.00387 in.

The resultant input effective deadband is 0.00387 in +0.00093 in, or 0.00480 in. Reflected at the servoactuator output, this deadband is 0.00576 in. This equates to a 1/20.1 'g' or 0.0497 'g' steady state flight path positionability/threshold variation.

Another positionability/threshold consideration is the ability to position and maintain an elevator position as commanded at the control column. A conservative approach is to consider all of the control system friction reflected at the rear cable quadrants. A review of Table 8 shows that 74% of the total control system friction is at or aft of the rear cable quadrants. So this conservative distribution of friction is not unreasonable. The total friction reflected at the control column is 2.743 lb. At the cable level, the total friction is 9.875 lb.

The cable run from the control columns to the rear quadrants are assumed to be 60 ft long of 0.125 in diameter carbon steel with 7 x 19 construction. There are four cables, preloaded to 70 lb, operating in parallel. With a 70 lb preload, the cable EA value is assumed to be 84 200 lb/in/in each. With a system rigidity factor of 1.09 (explained later in 5.1.h.1), the resultant cable system stiffness is $\frac{84\,200 \times 4}{60 \times 12 \times 1.09} = 429.2\ \text{lb/in at the cable level}$.

The resultant positionability hysteresis/threshold at the rear quadrant is $\pm 9.875 / 429.2 = \pm 0.0230\ \text{in}$ at a 6 in quadrant pitch radius, or ±0.220 degrees.

This control cable rear quadrant deadband equates to ±0.0368 in servoactuator output. This control cable reflected deadband is additive to the elevator control pushrod deadband calculated above. The net control column positionability deadband is $\pm 0.0368 \pm 0.5 \times 0.00480 = \pm 0.0392\ \text{in}$ at the servoactuator output. This equates to ±0.0405 degrees of elevator surface or ±0.034 ‘g’ positionability at the most effective elevator condition of 1.2 degrees/g. This hysteresis approach to control column positionability is conservative. It assumes the pilot is commanding an elevator position from opposing pre-maneuver conditions at maximum friction. Positionability repeatability is a more realistic consideration. With repeatability, the pilot is working with much smaller variations in friction and backlash and he senses the command response, adjusting the command input accordingly. But the hysteresis approach provides a quantitative assessment of the quality of a control system during the design phase.

4. CONTROL SYSTEM INSTALLATION GUIDELINES

4.1 Control System Routing/Orientation/Installation

Control system routing/orientation of cables, pushrods, push-pull cables, hydraulic lines, and electrical wiring should be done in the most direct manner possible taking into consideration the separation of duplicate systems, structure, other systems, hazard areas of 3.2.2, and easy access for routing maintenance.

Mechanical control systems should be run as close to the structure’s neutral axis as possible to minimize the effect of structural deflections of aircraft due to maneuvers in flight, and on the ground. Cable systems should keep the cables of the same system on the same side of, and at the same distance from the neutral axis of the structure to prevent control actuation due to structural deflections. Control rod systems should use reversing cranks frequently to equalize the deflections due to structural flexure and to minimize the control forces associated with accelerations.

If several different installations of components such as brackets, rods, bellcranks, splines, torque tubes, electrical connectors, trim actuators, spring struts, cable turnbuckles are feasible, the designer should ensure that the various installations are acceptable. Installations that are not acceptable must be made physically impossible. Where such detail features as inadequate spline engagement cannot be made physically impossible, the male spline should incorporate a groove that indicates the correct engagement when not visible.

4.2 Cockpit Controls

Cockpit controls have a high degree of commonality between cockpits of aircraft of the same type to minimize the possibility of human error as pilots transfer from one aircraft model to another. Basically, cockpit controls are designed and located on commercial aircraft, as well as military aircraft, in accordance with MIL-STD-203, AFSC DH 2-2, DN 2A1, paragraph 3 (Aircrew Controls), and AFSC DH 2-2, DN 2A5 (Flight Controls) for conventional and short takeoff and landing aircraft.

4.3 Control System Stops

Control system stops should be provided to limit system travel at the output end of the control system. Other auxiliary stops frequently used in control systems are output load limiting stops, input travel stops for cockpit controls, AFCS authority stops, and mechanism overtravel stops. All adjustable stops should be positively locked or safety wired in the adjusted position. Plain jam nuts are not considered adequate as a locking device.

Output stops should be located near the item that is controlled. Each control surface should have its own travel stops and these stops may be incorporated in the servo control actuator, if one is used, to position the surface. A control surface must have its own stops if the actuator is remotely located from the surface, unless the connecting linkage can be shown to have only an extremely remote probability (1.0×10^{-9} or less) of failure. Servo control actuators should be designed with input travel stops that are set inside the actuator output piston travel limits, to minimize actuator fatigue damage. (The servoactuator output piston travel limits shall be designed to withstand bottoming during ground checkout and/or wind gusts.) Output load limiting stops are also used in a servoactuator's input linkage to react the cockpit control's design load, so that the load downstream of the stop can be reduced to that needed to shear a chip in the actuator's control valve. Coat stop bolt threads with corrosion preventative compound.

Input stops are located near the cockpit controls to limit their movement. Input stops must allow the output stops to be contacted first. The additional force to move the cockpit control to the input stop, after the output stop is contacted, is called the stop cushion force in cable systems.

With the output and input stops set for maximum control system travel, there shall be clearances elsewhere in the control system between links, cranks, structure, etc.

AFCS authority stops are used to limit AFCS inputs into primary MFCS systems to safe levels which the pilot can override.

Mechanism overtravel stops are used to prevent a mechanism from going into an irreversible position at an extreme travel condition.

4.4 Control System Separation and Clearances

Control system separation for redundant cable, push rod, push-pull cable, electrical wiring, and hydraulic systems should be the maximum possible to minimize the probability of simultaneous failure of all redundant control paths from the same cause. An example of good design is where one system is in the ceiling and the other is underneath the cabin floor.

Table 10 lists desirable and minimum clearances for cable, flexible cable, and control rod systems.

TABLE 10- CONTROL SYSTEM CLEARANCES

Type of Control System	Condition	Inches (Centimeters) in Clearance	
		Desirable	Minimum
Cable	Between cables	2.00 (5.1)	0.50 (1.3) by supports
	Structure, electrical wiring, tubing and fixed equipment	1.50 (3.8) Down 1.00 (2.5) Other Directions	0.50 (1.3) fairlead (Note 1) 0.20 (0.5) rub strip (Note 1)
	Gear, doors, moving components	4.00 (10.2)	2.00 (5.1)
Push-Pull Rods Push-Pull Flexible Cable	Between push-pull rods and push-pull flexible cables.	2.00 (5.1)	0.50 (1.3) other systems 0.25 (0.6) same system
	Structure, electrical wiring, tubing and fixed equipment	1.00 (2.5)	0.50 (1.3)
	Gear, doors, moving components	4.00 (10.2)	2.00 (5.1)

NOTE 1: Under normal pre-flight and flight structural deflections (1 g ± 0.20 g)

Clearance for moving parts should be shown to be adequate for a minimum of 5 degrees overtravel or 0.50 in (1.3 cm), whichever is greater. In each design the amount of overtravel should be evaluated for excess of this amount. Overtravel and clearance should be shown on design layouts. This is particularly important on bellcrank clevises with respect to rod ends and similar joints.

Valve overtravel on actuators should be determined based on system travel and all tolerances to avoid the necessity for extremely precise rigging.

Quadrant overtravel should be 5 degrees beyond the maximum travel range before the cable attains a force that would tend to pull the cable retention terminal out of its quadrant seat. Effect of cable stretch should be included.

Cable systems should be analyzed under all loading conditions including conditions where the cable might slack to ensure adequate clearances.

Access panels or doors, which have to be removed for inspection or maintenance shall not interfere with the full freedom and functioning of the controls.

4.5 Control System Rigging Provisions

In order for a control system to operate properly, cable runs, levers, sectors and bellcranks should be rigged to a definite neutral or stowed position. Rigging pins provide a quick and accurate means of establishing this position.

In a long run of pushrods, it is only required to have rigging provisions on every third bellcrank. The positions of the bellcranks on each side of one that is rigged are established by fixed length pushrods. The third pushrod should be adjustable. Locate adjustable rods at fuselage production breaks since a dimension crossing a break has a larger than normal production tolerance. See Figure 2.

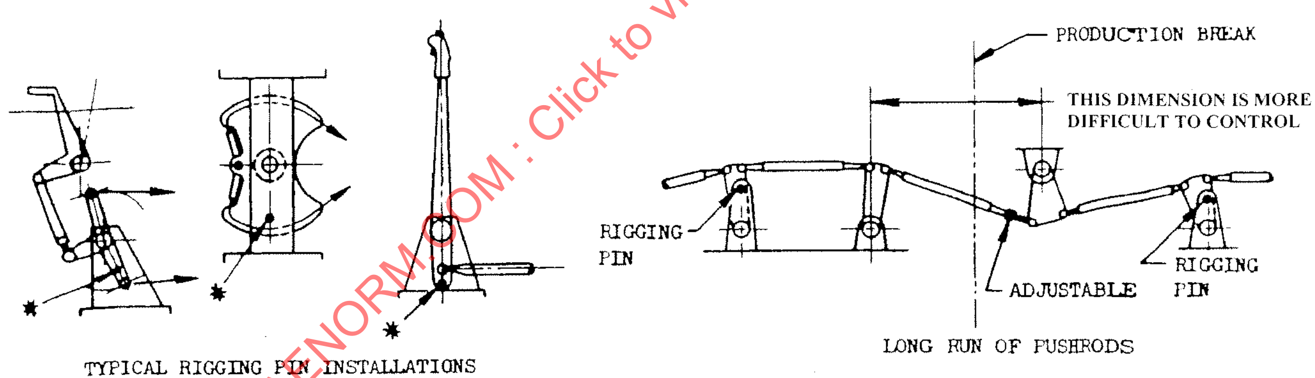


FIGURE 2 - RIGGING PIN PROVISIONS

Double shear rigging pin installations are preferred. If single shear pins must be used, one hole should be sufficiently long to prevent cocking of the pin. Avoid single shear pins if possible. See Figures 3 and 4.

To eliminate the possibility of interference in the control system, rigging pin supports should be strong enough to withstand normal ground handling without significant bending.

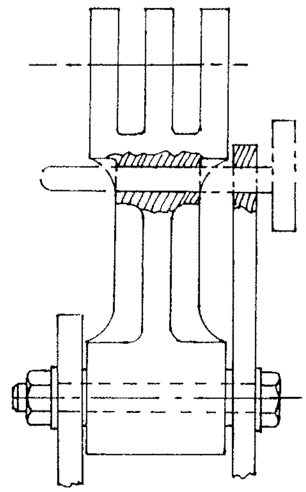


FIGURE 3 - SINGLE SHEAR RIGGING PIN INSTALLATION - POOR DESIGN

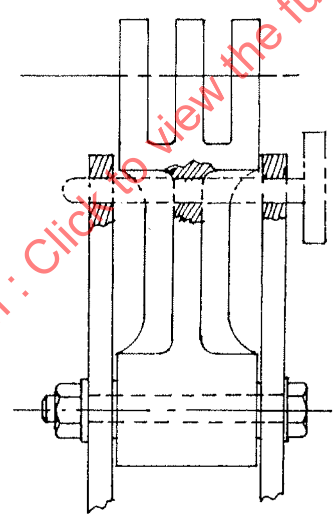


FIGURE 4 - DOUBLE SHEAR RIGGING PIN INSTALLATION - GOOD DESIGN

Rigging of control systems should be minimized by designing components which can be removed and replaced without affecting the system rigging. The replacement of a component in one system should not affect the rigging of another control system.

No rigging pin shall be used on the power side of the system.

Complex components requiring rigging should be readily accessible and have adequate working space.

Equipment that requires sensitive or very precise adjustments which can only be properly made when removed from the airplane, should be designed with these adjustments inaccessible with the equipment installed.

For rigging pins, use bar stock of comparable diameter or commercially available jig pins. Close tolerance pins and holes may be required if positional accuracy of the rigged components is required. Rigging pins shall have a large red flap attached to highlight the presence of the installed rig pin and the need to remove after rigging is complete.

Where possible, design equipment and system to eliminate the need for adjustment and special tools.

Angular measurements should be shown in linear dimensions whenever possible.

Measurements that must be made when adjusting and rigging should be specified between easily measured features. Dimensions should be between flats, index marks, etc., not between centers of holes or centerlines of parts. Do not depend on the use of clay or putty to measure clearances, etc. Loose shims shall not be used, if avoidable, in particular where their presence can result in a jam, malfunction or linkage strain. Likewise, loose shims shall not be used where their emission can cause a control system strain or maladjustment.

For rigging load and rigging instructions, see 5.1.h.7 and 5.1.h.8.

Considering the cable's non-linear load versus deflection characteristics, systems should be designed for tension at 2.5 times the specified rig load.

4.6 Cable System Installations

a. Cable Support Spacing: Cable wear and fatigue are accelerated if a section of cable rides on more than one sheave; therefore, the minimum spacing between sheaves is that distance which prevents a point on the cable from traveling over the groove of two adjacent sheaves plus an overtravel factor.

   The maximum longitudinal cable support spacing by pulleys, fairleads or cranks is approximately 200 in (510 cm) in the body. The maximum cable support spacing by pulleys in the wing is approximately 75 in (190 cm). The closer cable support spacing in the wing is to minimize cable wear from slapping fairleads and sheave grooves due to the wing bending and vibratory environment.

   Cable runs in the wing (or other vibratory areas) must keep the cable restrained by positive wrap (2 degrees minimum) on pulleys or by grommets which have zero wrap unless the wing deflects. Both sheaves and grommets shall bend the cable with radii greater than a 30 *D/d* ratio.

b. Cable Travels: No-load cable travel is defined as cable travel without stretch, due to load, to move a controlled item from one travel extreme to the other. Stop cushion force is defined as the force required at 70 °F to move the control to its input stop after the system output stop is contacted. The force is created by cable stretch, and is used to assure the output stops will be contacted with normal system operating forces. These conditions normally apply to control systems with servoactuators as well as purely manual control systems.

   Table 11 lists the suggested no-load cable travels and stop cushion forces for PFCS. Cable travel is selected as a compromise between system friction, system stiffness, and backlash. When cable travel is reduced, system stiffness becomes more critical. Small travel also makes backlash and other lost motion more critical.

TABLE 11 - SUGGESTED CABLE SYSTEM TRAVEL AND STOP CUSHION FORCES

Control System	No-Load Cable Travel		Stop Cushion Force	
	Inches	Centimeters	Pounds	Newtons
Lateral	8	20.3	50	222
Longitudinal	8	20.3	75	334
Directional	8	20.3	150	667
Engine Military Requirement of AFSC-DH-2-2 DN2B3	4 minimum	10.2 minimum	200 to 250% of normal operating force	
Other Controls	4 minimum	10.2 minimum	10 maximum	44 maximum

c. Cable Sheaves: Cable alignment with a sheave is defined as the angle between the cable centerline and the plane of the sheave groove. Design should strive for zero misalignment, although a maximum of 1/4 degree is allowed. This amount of design misalignment, along with manufacturing tolerances, still allows the shop to meet the maximum 2 degrees misalignment permitted on the aircraft, provided the cable does not rub against the side of the pulley groove. Typically, 2 degrees misalignment is permitted on PFCS and 3 degrees misalignment is permitted on SCS and UCS. Figure 5 provides a graphic indication of the impact of misalignment on cable and pulley friction.

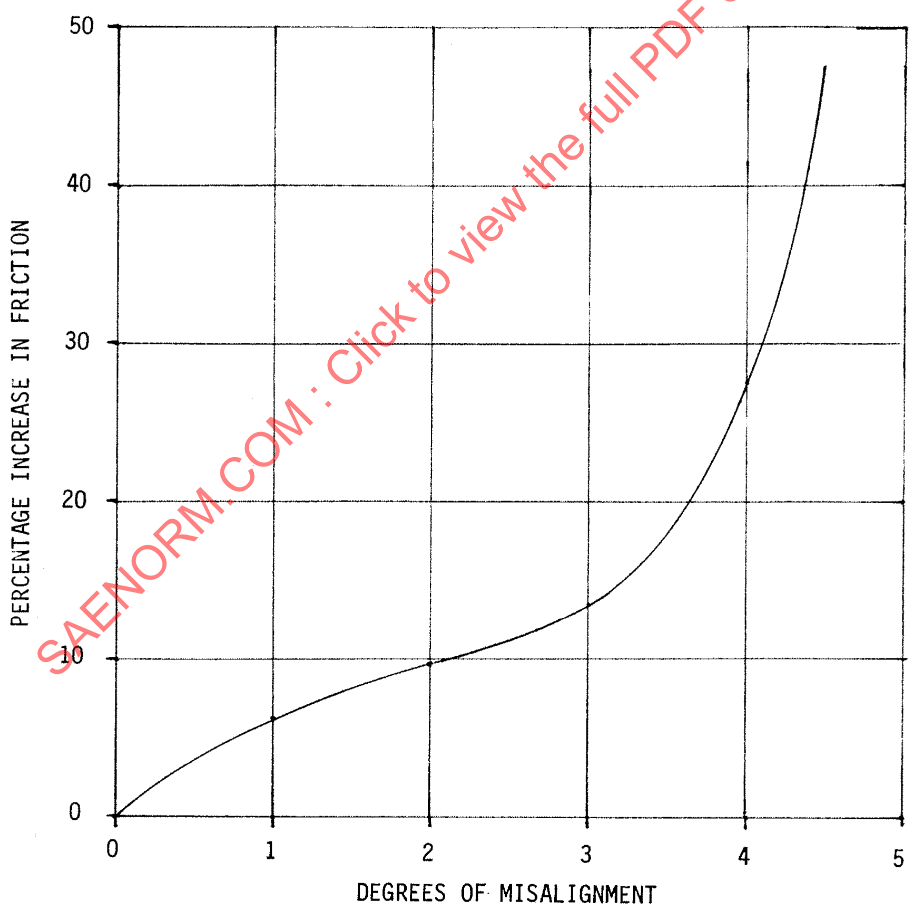


FIGURE 5 - MISALIGNMENT IMPACT ON CABLE FRICTION

On sheave brackets which direct the cable upward, a pulley guard, similar to Figure 6, should be installed to deflect debris which might fall from above or ride down the cable.

In unpressurized areas, sheave support structure should be designed to prevent jamming of the control system by the freezing of water or condensation flowing downward over the sheave or bracket, and the accumulation of water slush or ice. Cables routed below galleys and toilets must be protected from liquid spillage by covers. If cables pass close to waste water pipes, the cables should be routed above those pipes.

Lightening holes in the sides of pulley brackets should be placed so that the cables on sheaves are easily inspectable, and small ingested particles can drop out rather than wedge between the pulley and the bracket.

Pulleys of different systems shall not be installed on the same bracket, or use the same pivot bolt unless a dual load path is provided.

Cable runs should be designed and configured so a replacement cable can be installed by pulling it into position by the cable to be replaced. Obviously, cable guards need to be removed.

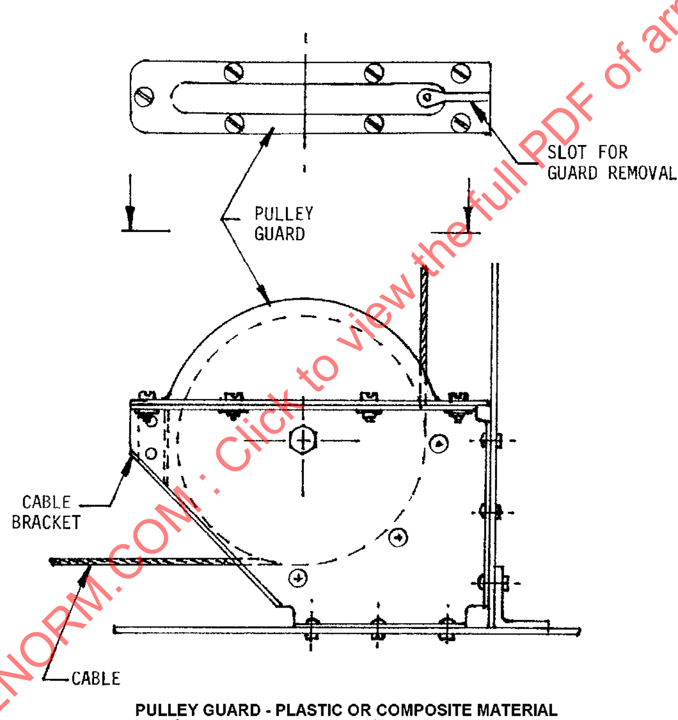


FIGURE 6 - PULLEY GUARD

d. Cable and Crank Combinations: Cables can be used in combination with cranks to accomplish the same function as cables in combination with quadrants. Examples of cable and crank combinations are shown in Figures 7 and 8. Cable and crank installations generally have less friction because the cables are not being wrapped and unwrapped at the bellcrank interfaces like with cable quadrants. The anti-friction ball bearing terminals provide a very low friction interface and allow the cables to be preloaded to a higher level resulting in increased control installation stiffness. The cable diameter can be increased on applications like Figure 8 where there are no pulleys involved. This increases the installed stiffness with no increase in friction. The cable/crank mechanical gain can be non-linear, especially if a crank is used in combination with a conventional quadrant. This characteristic can be an advantage or a disadvantage, depending on the application. The crank and cable installation motions need to be analyzed and indexed accurately so the incremental motions of both cables are as equal as possible. Splitting the cable rise and fall, similar to the approach presented in 4.7 for pushrods and cranks, equalizes the cable incremental displacements. This incremental equality of cable strokes is necessary to maintain the cable preloads as constant as practical.

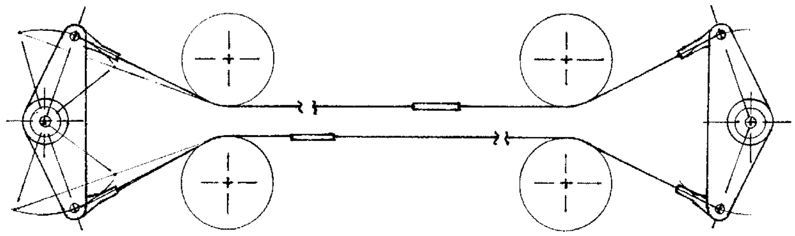


FIGURE 7 - CABLE AND CRANK INSTALLATION

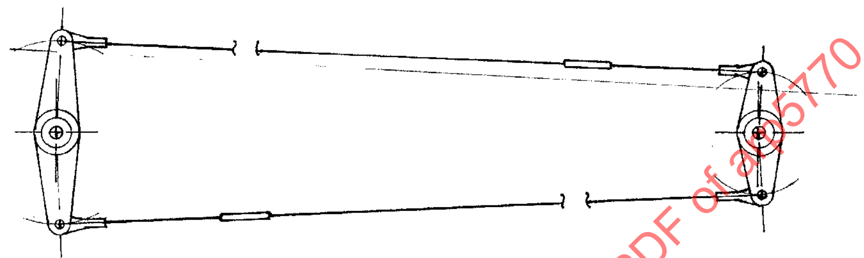


FIGURE 8 - CABLE AND CRANK INSTALLATION

4.7 Control Pushrod Installations

a. Control Pushrod Length: The maximum length of a control pushrod is usually based upon the length which will allow removal and installation without the removal of other items. Most of the time this will restrict rod length to approximately 75 in.

It is preferred that control pushrods have a maximum slenderness ratio of *L/p* = 100, where L is the length of the rod and p is the mean radius of gyration. This value may be exceeded, provided that the rod is within acceptable stress limitations. The rod compressive stress, *Sc*, without local buckling, is calculated by use of the Euler column formula,

$$Sc = \frac{P}{A} = \frac{\pi^2 E}{(L/p)^2} \quad \text{(Eq. 1)}$$

where:

P = total axial load
A = area of section
E = modulus of elasticity of the material
L/p = slenderness ratio

$$p = \frac{1}{4}(d_o^2 + d_i^2)^{0.5} \quad \text{(Eq. 2)}$$

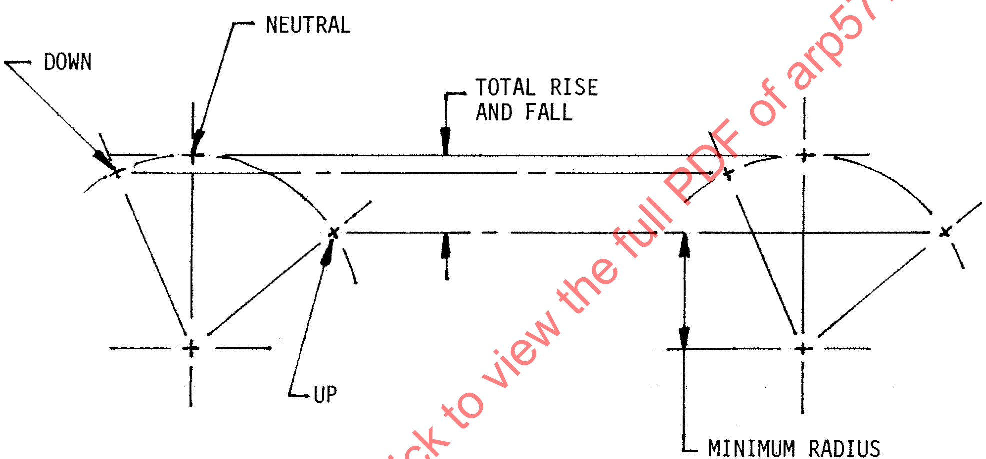
where:

d_o = pushrod outside diameter
d_i = pushrod inside diameter

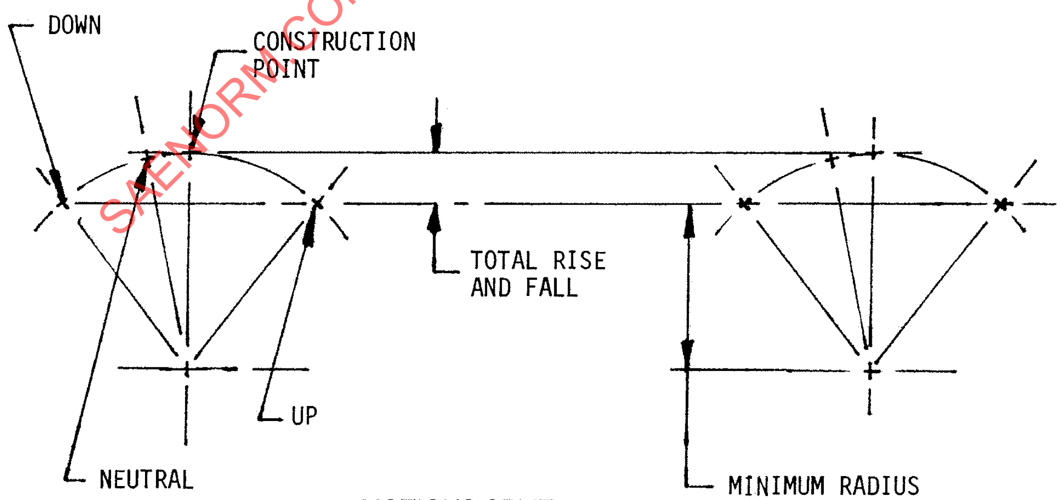
The most efficient rod design is a thin walled tube whose ends are swaged down for rod ends and whose slenderness ratio is close to 100. Minimum wall thickness in inches is 0.035 for aluminum and 0.020 for steel.

Preliminary clearance studies can be quickly made by assuming a solid rod which has a maximum outside diameter of d_o = 0.04L and a slenderness ratio of 100. For example, the maximum possible OD of a 75 in long pushrod with a slenderness ratio of 100 will be 3.00 in (0.04 x 75). Use this maximum diameter for evaluating the clearance between the pushrod and other systems or structure during preliminary design.

b. Control Pushrod Travel and Motion Linearity: Control pushrod travel should be set as large as practical for high mechanical advantage and low rod loads. The lower the loads, the lighter the system. Control pushrod axial travel is not linear with the angular travel of the cranks. This non-linear travel characteristic is not significant if the crank angular travel is kept within ±30 degrees of the position where the rod centerline is normal to the crank centerline. Total crank rotation of more than 60 degrees causes rapid increase in rise and fall of the control pushrod with smaller incremental gains in linear travel. The more the angular travel, the more the rise and fall of the pushrod, and as a result, more space is required for routing the control system. For typical control pushrod systems, the clockwise and counterclockwise crank motions are split (made equal) for optimum mechanical advantage and linearity. Examples of installations with the motions not split and split are shown in Figure 9.



MOTIONS NOT SPLIT



MOTIONS SPLIT

FIGURE 9 - MOTION LAYOUTS

Split motion arrangements can be used to optimize the overall mechanical advantage with cranks of equal or unequal radii, with the cranks on the same or opposite sides of the control pushrod and with non-coplanar crank pivots. Geometric layouts are used to optimize the pushrod motions before the cranks, pushrods and pivot fitting designs are initiated. Elevator and aileron control systems are usually designed with more trailing edge up travel than trailing edge down travel. The elevator and aileron pushrod system can best be configured with the extreme end points of up and down surface travel split and the neutral surface position offset with respect to the control pushrod neutral position.

c. Control Linkage Non-linearity: On many control systems, there is a preference to keep the command motions versus surface motions as linear as possible. This is best achieved by splitting the pushrod and cranks rise and fall. But in some applications, there is a need to provide non-linear gains. This can be achieved with a pushrod and crank system, but not with a conventional cable system. A typical example of this is a stabilator or canard command control for a high performance jet powered aircraft. The preferred mechanical command versus surface motion is low gain at neutral and high gain at larger surface travels. An example of the preferred motions are shown in Figure 10. This motion plot provides a very low gain at neutral for accurate small output positional control at high aircraft speeds and large incremental outputs versus input at low aircraft speeds. The crank and pushrod arrangement that provides this non-linearity is shown in Figure 11. This very non-linear motion characteristic is usually used in combination with series trim so the command neutral position is held constant with the non-linear motion neutral position, regardless of trim position. (Series trim actuator output motion is inserted between the non-linear linkage output and control surface input. The cockpit control and non-linear function remain motionless while the series trim output resets the control surface position. A control system schematic with the non-linear mechanization and series trim included is shown in Figure 12). Conversely, there have been some unique control applications where the opposite motion characteristics were needed. An example of this motion plot is shown in Figure 13. (This unusual non-linear gain feature was used for pitch control of a V/Stol aircraft where pitch was induced by a propeller mounted horizontally at the aircraft rear. The propeller provided low pitch gain around neutral. The unique linkage high gain at neutral command motion compensated for the low propeller pitch gain.) This motion arrangement is provided by the pushrod and crank arrangement shown in Figure 14. This is reversal of the linkage arrangement shown in Figure 11.

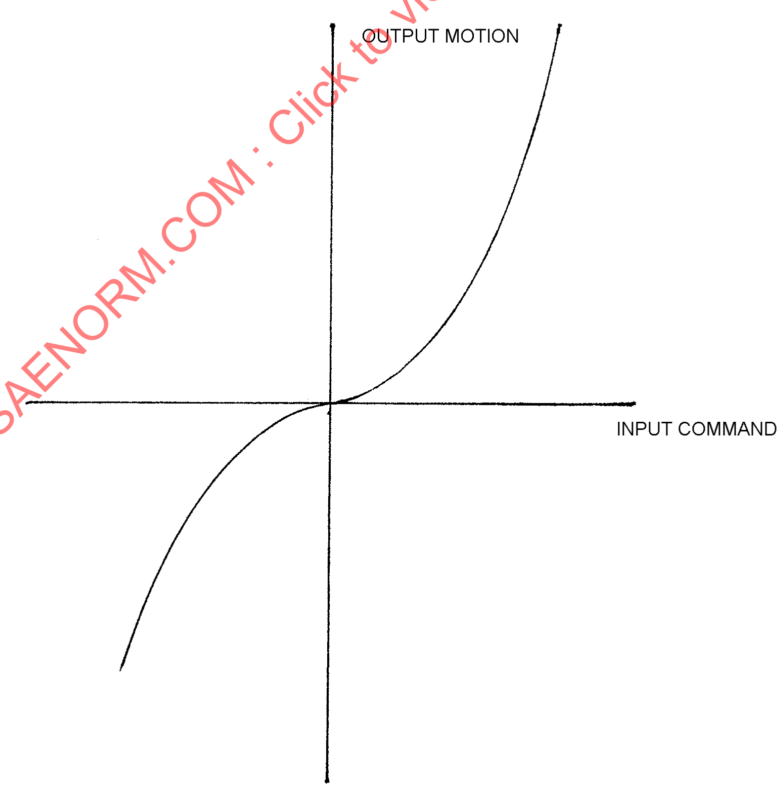


FIGURE 10 - PREFERRED MOTIONS - HIGH PERFORMANCE JET

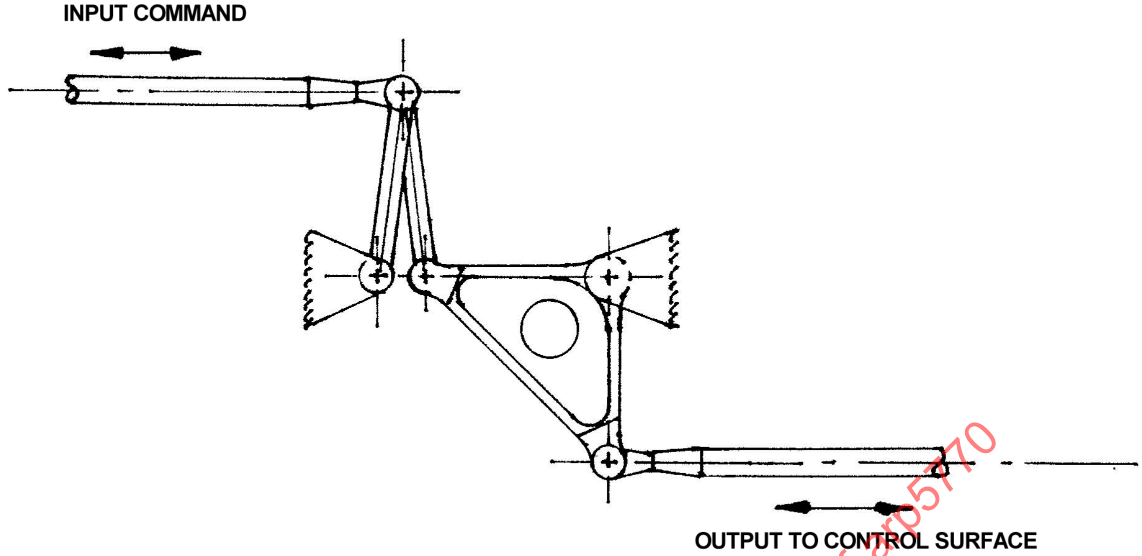


FIGURE 11 - CRANK AND PUSHROD ARRANGEMENT

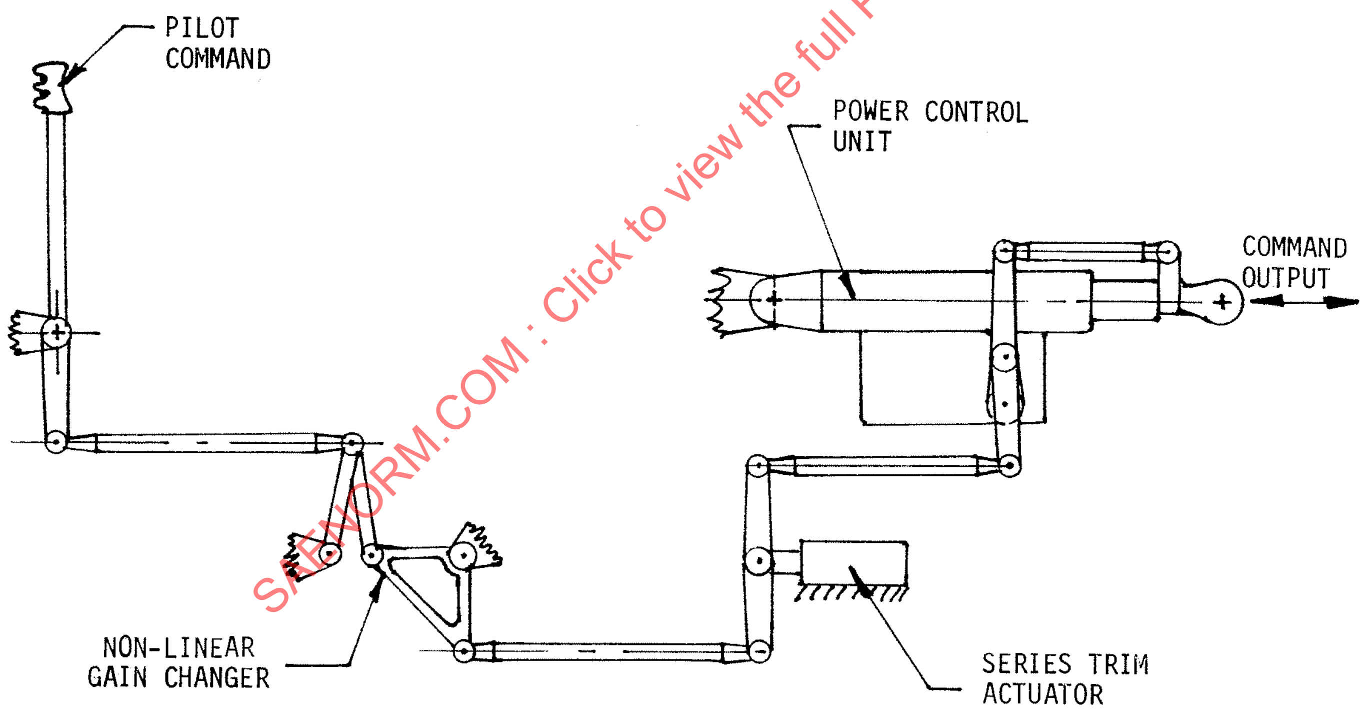


FIGURE 12 - CONTROL SYSTEM SCHEMATIC WITH NON-LINEAR GAIN CHANGER AND SERIES TRIM ACTUATOR

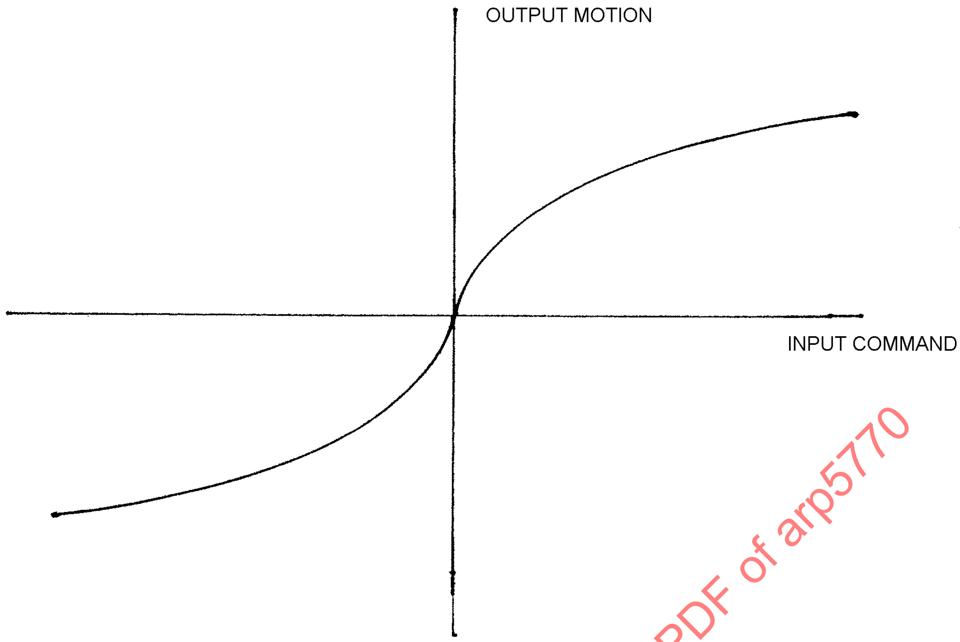


FIGURE 13 - UNIQUE CONTROL MOTIONS

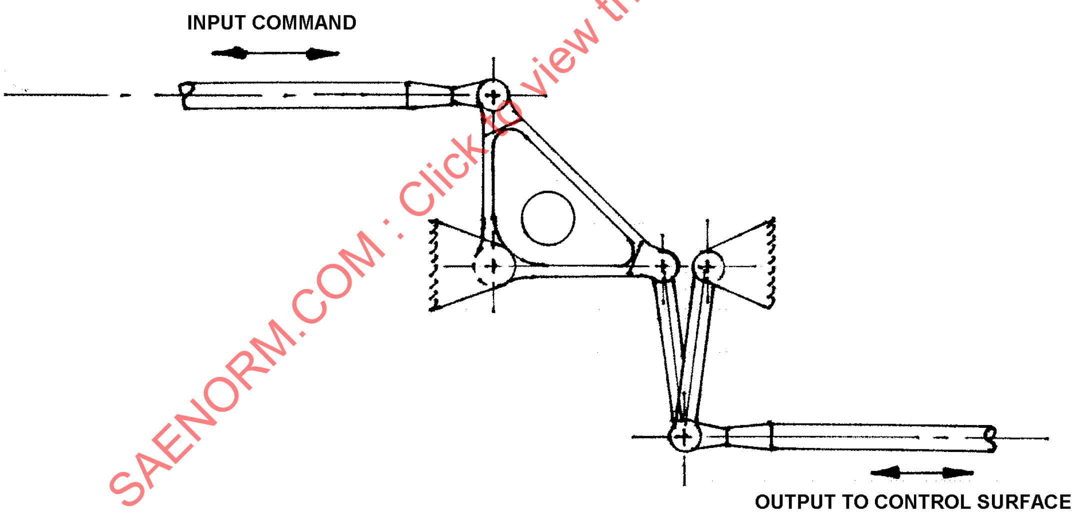
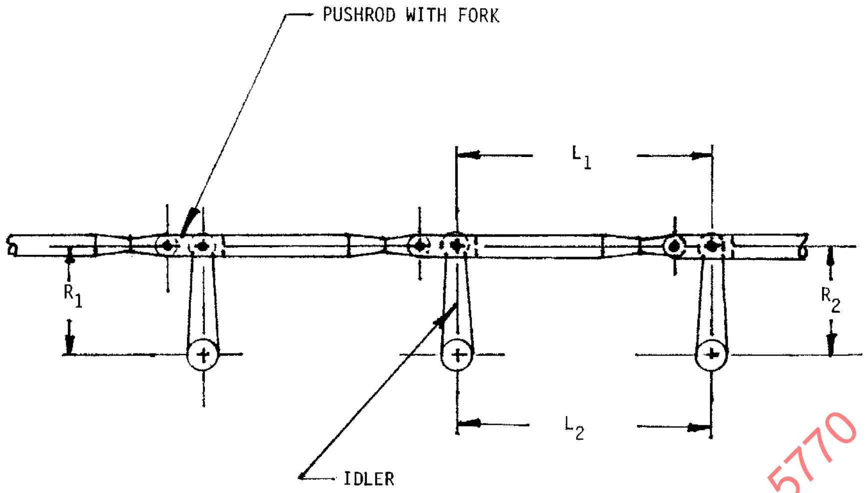


FIGURE 14 - UNIQUE CRANK AND PUSHROD ARRANGEMENT

There are computer based four-bar linkage programs that provide a large variety of motion characteristics.

d. Long Pushrod Runs: Long pushrod runs are typically not used on aircraft because cable systems are usually more weight and cost effective and have less backlash. But there may be some applications where two or more pushrods are used in series. On multiple parallel pushrod runs, a series of parallelogram cranks and pushrods can be arranged as shown in Figure 15. This arrangement minimizes bearing loads, friction and backlash because the idler bearings carry very low loads. This combination of pushrods and cranks should not be used where the rod system must deviate from a straight line because it will impose a bending load on the forks and pushrods. In such an application, a bellcrank should be used to turn the corner, then the parallelograms may be continued.



Parallelogram Linkage Arrangement
($L_1 = L_2$ and $R_1 = R_2$)

Idlers function as linkage stabilizers only and carry low loads.

FIGURE 15 - LONG PUSHROD RUNS

Airframe flexure, if ignored in control linkage design, can introduce unwanted signals into the control system and, in some cases, compromise aircraft safety. When airframe bending occurs, the distance between the mounting points at the ends of the linkage system will change. The linkage however, is not subjected to the airframe loads, remains the same length, and introduces a signal into the control system. If a walking beam is properly located in the system, the pushrod position will then vary as required to accommodate the relative motion of the mounting points. This same walking beam may also be useful in canceling out inertial effects. See Figure 16.

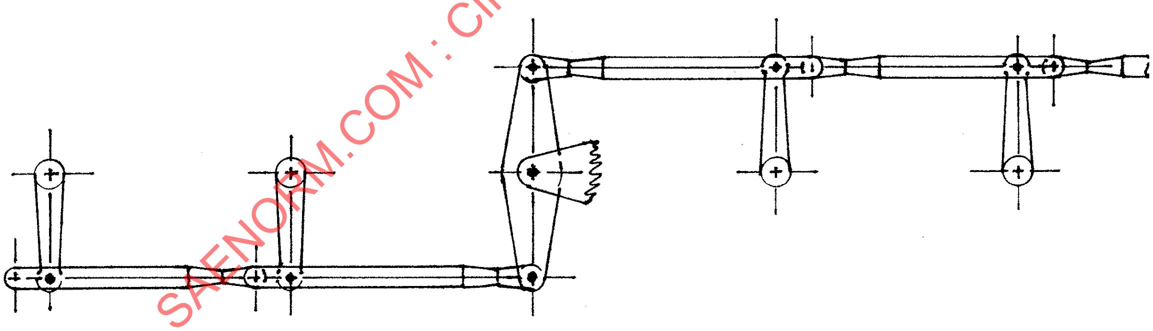


FIGURE 16 - WALKING BEAM INSTALLATION

e. Control Pushrod Failure Restraints: Control pushrod adjustments should be minimized and each pushrod should be carefully evaluated to determine if a fixed length pushrod would be satisfactory in a particular application.

Quarter (1/4) inch diameter bolts should be the smallest fastener used for the installation of control pushrods. If a 3/16 in diameter or smaller fastener is used, use a recessed head fastener to reduce potential for over torqueing.

The effect of a jam condition must be carefully reviewed in a control pushrod system to determine whether this type of failure can be tolerated, or if an override device is required to assure adequate control at all times.

Restraints for a control pushrod must be provided if it could jam other systems should it become disconnected or severed. Any rod or link that can possibly be used as a handhold or step during construction, maintenance or inspection shall be designed for the following ultimate lateral loads applied at any point along its length.

Handhold - 150 lb (667 N)
Step - 300 lb (1334 N)

4.8 Push-Pull Flexible Control Installations

a. Push-pull flexible controls are sometimes used where the space available for the mechanical control path is restricted. They have been used occasionally for primary flight control and on numerous aircraft for secondary flight control applications. Their use for primary flight control is not recommended because of the hysteresis introduced by their combination of friction and compliance. This hysteresis can increase as a result of handling damage and installation or maintenance errors and cause flying qualities unacceptable for precision control tasks. The use of cable or pushrod approaches should be pursued energetically for primary flight control because of their promise of dependably acceptable friction and threshold.

b. Push-pull flexible controls shall comply with MIL-PRF-7958. There are two basic types of push-pull flexible controls, the sliding friction type and the ball bearing type. The ball bearing type has a higher efficiency. Extreme care must be taken in design to prevent installation errors and provide shielding so controls cannot be used as handholds or stepped on by maintenance personnel. Installation design of the sliding friction type should minimize the number of bends, use large curves, and avoid reverse bends to keep friction and backlash within acceptable limits. Installation design of the ball bearing type should keep all bends in a single plane, or the control helixed at a controlled rate into another plane to prevent binding or damage. Both types must be supported at frequent intervals and both ends, but not restrained so that motion of the internal parts may be restricted axially. Consideration of environmental conditions is very important.

4.9 Electrical Bonding

Bonding and grounding of brackets and cables is per MIL-STD-461 and MIL-STD-464.

4.10 Corrosion Protection

Cavities likely to retain fluids (condensation water, cleaning fluids, etc.) shall be avoided as far as possible. If it is not possible, the cavities shall be filled with compound or have a suitable corrosion protection or be drained via holes. Floating bearings, splines or bushings should be avoided. If not, precautions against fretting corrosion should be incorporated.

4.11 Bearings and Lubrication

If a sealed bearing requires periodic lubrication, its installation should be such that the maximum lubrication pressure cannot expel the seals. The maximum lubrication gun pressure is typically 6000 psi. Use of needle bearings where needles are able to skew should be avoided. Caged needle bearings are acceptable for some applications.

4.12 Control System Maintenance

All parts of the mechanical control system should be readily accessible for inspection, repair, lubrication, and rigging adjustment of cables, linkage, and components. Inspection doors should be provided at pulleys, quadrants, connections and components, not otherwise accessible. It should be possible to inspect the entire lengths of cables, and push-pull rods for corrosion and signs of wear without disconnecting the system. Components should be designed so their removal and replacement can be accomplished without disturbing the rigging to the maximum extent that is practical.

- Cable rigging loads should be verified and reset on a scheduled basis.

- Consideration should be given to sources of possible injury or damage to personnel or equipment during maintenance.

- Access to mechanical control components shall be sufficient to allow maintenance operations.

- All tools should be standard tools if practical.

- There should be sufficient space to enable the various maintenance tools to be used.

- Quick release fasteners should be used according to the maintenance time scale, where possible.

- If shall not be necessary to remove or partially detach structural parts for any installation/removal operations.

- Labels provided for maintenance (identification, lubrication, adjustment, graduated quadrants on regulator ...) should be accessible for inspection by removing the inspection doors only.

- If an item of equipment is installed in different areas, several labels should be provided for maintenance as necessary, so that at least one of them (or one of each) is visible in each installation case.

- Dual load path controls: special care should be taken for their installation; it should be possible to inspect or check each path without removing these controls or equipment from the aircraft.

- Replacement of any position transducer should be possible without adjustment within the aircraft electronics.

- LRU removal and installation time: 30 min (design goal).

- It should be possible to use lifting/hoisting equipment for the removal/installation of units whose weight exceeds 50 lb.

Maintenance Checklist

Total Control System:

- Pivot bolts are properly installed, preloaded, and retained.

- Pivot brackets are properly aligned (not bent) so pulleys and quadrants have alignment with the cable. Cranks have adequate clearance with pivot brackets over the full stroke.

- All elements of the control system have adequate clearance with respect to adjacent components over the full stroke.

- All rig pins can be installed. (If the system was designed accordingly.)

Cable Systems:

- Cable guards have clearance with pulley, quadrant, and drum.
  Cable guards have no more than 1/2 cable diameter clearance with pulley, quadrant, and drum.

- Cables have no broken or kinked wires.
  Clad cables have no kinks.
  Carbon cables are lubricated.
  Corrosion resistant cables are not lubricated.

- Turnbuckles are properly installed and locked.
  Cable attachments to quadrants, cranks, and drums are secure.

- Cables are not deflected by fairleads, rub strips or seals.
- Cables have clearance with respect to cable grommets.
- Cable tension is within specified limits.

Pushrod and Crank Systems:

- Rod and nuts, stop nuts, and adjustments are preloaded and secured.
- Rod ends are aligned with respect to crank clevii, have adequate clearance, and do not bind over full travel.

5. COMPONENT DESIGN GUIDELINES

5.1 Control Cable

a. Control Cable Assemblies: Use control cable assemblies with swaged terminal fittings that are fabricated per MIL-DTL-6117. Control cable assemblies should be prestretched and proof tested per MIL-DTL-5688 after terminal swaging. In calling out cable assemblies where turnbuckles are involved, it is conventional to have the right-hand thread of the turnbuckle forward or inboard. One possible exception to this rule is where there are two quick-disconnects or turnbuckles in one system at a fuselage breakpoint. Having one right-hand and one left-hand thread forward eliminates the possibility of the cables being cross-coupled.

   Use 3/32 or 1/8 in diameter wire rope per MIL-DTL-83420 Type I, Composition A for normal applications. Corrosion resistant steel cables per MIL-DTL-83420 Type I, Composition B may be used for applications in areas with a highly corrosive environment. If strength requirements dictate larger sizes, use wire rope per MIL-DTL-83420. Composition A carbon steel wire is available with zinc or tin over zinc coating. Tin over zinc coating is preferred. Composition A or B Type I wire rope is rated for operation within a -65 to +250 °F temperature range. 14CFR Part 23 and Part 25 allow no cable smaller than 1/8 in diameter in aileron, elevator, and rudder control systems.

   Cable systems with adequate turnbuckle adjustment shall have cable lengths designed at zero rig load to have a nominal engagement of the turnbuckle at minimum cable length. Standard cable length tolerances are ±0.0005 in/in with a minimum tolerance of ±0.12 in, and a maximum tolerance of ±0.50 in. Also see 6.1.h.8 for turnbuckle calculations.

b. Cable Lubrication and Protection: Lubrication of control cables is necessary for fatigue strength and corrosion protection.

   Internal lubrication is provided by the cable manufacturer so the wire rope will meet the tensile strength requirement after a specification endurance test. Each wire of the wire rope is coated with a friction-preventative, non-corrosive lubricant. Cable life is dependent upon the retention of this lubricant in service. The loss of the internal lubrication results in rapid deterioration of the cable by wear of the individual wires as they move relative to each other when passing over sheaves. The cable should be re-lubricated upon installation and periodically throughout its service life.

   Cable exposure to contaminants should be assessed and the control designer has the responsibility to ensure that cables are adequately protected for the entire run.

   A thin coating of external lubrication over the length of all bare carbon steel (composition A) cable is required. This external lubrication is to provide cable lubrication over sheaves, and additional corrosion protection for the rest of the cable.

This does not apply to corrosion resistant steel cables (composition B) and jacketed cables. Lubrication attracts abrasive particles to corrosion resistant steel cables resulting in wear at fairleads, pulleys, and pressure seals. Jacketed cables do not require external lubrication.

Composition A cable relubrication in service is a necessary maintenance item in some areas where the lubrication is depleted by high temperatures, or by the washing action of cleaning solutions used by aircraft operators.

c. Control Cable Characteristics and Usage: Wire rope construction characteristics and usage are shown in Table 12.

Nylon jacketed wire rope should be considered for new applications because of the superior wear and corrosion resistance provided by the nylon jacket which prevents the loss of, or contamination of, the cable lubricant. Composition A or B Type II nylon jacketed wire rope is rated for operation within a -65 to +200 °F temperature range.

Lock-clad control cable has aluminum tubing swaged over MIL-DTL-83420, Type I wire rope to improve stiffness for all airframe applications and thermal expansion characteristics for aluminum airframes only. The clad is not flexible and therefore is only applied between sheaves.

TABLE 12 - AIRCRAFT CONTROL CABLE USAGE

Specification	Construction	Construction Characteristic	Where to Use
MIL-DTL-83420 Type I Composition A	3 x 7 1/32 7 x 7 3/64, 1/16 & 3/32 7 x 19 1/16 to 3/8	Unjacketed cable Carbon steel wires, tin over zinc coat preferred. Operating temperature -65 to 250 °F. Moderate corrosion protection. 7 x 7-3/32 diameter is preferred over 7 x 19 because of superior wear resistance.	All areas except those with a corrosive environment where lubrication can be lost by elevated temperatures or the washing action of aircraft fluids, and relubrication is impractical. Military Applications - Require a deviation to AS94900 for usage.
MIL-DTL-83420 Type I Composition B	3 x 7 1/32 7 x 7 3/64, 1/16 & 3/32 7 x 19 1/16 to 3/8	Same as Composition A except: 1. Corrosion resistant steel wires without tin or zinc coating. 2. Superior corrosion resistance.	Commercial - Areas impractical to replace lubricant lost by elevated temperature. Military - All areas where it is impractical to replace lubricant lost due to washing action of aircraft fluids.
MIL-DTL-83420 1/ Type II Composition A or B	3 x 7 1/32 7 x 7 3/64, 1/16 & 3/32 7 x 19 1/16 to 3/8	Nylon jacketed Type I cable with: 1. Superior wear resistance. 2. Superior corrosion resistance. 3. Less maintenance to relubricate. Wire rope failures hidden by jacket. Jacket ends must be sealed to prevent corrosion under jacket. Consider heat shrinkable sleeve. Possible higher friction unless jacket is stripped where it passes over sheaves.	Composition A preferred, but requires a deviation for military usage, in areas where lubrication can be lost by the washing action of aircraft fluids, and relubrication is impractical.
Lock-Clad 1/ MIL-DTL-87218	7 x 7 1/16 & 3/32 7 x 19 3/32 to 1/4	Aluminum tube swaged over MIL-DTL-83420 Type I cable of either composition. 1/ Clad areas have very high EA values. Clad sections can't be bent over sheaves but straight runs can be supported by sheaves in the body. Easily damaged in handling, installation, and maintenance. Low rig load may demand tension regulator. Purchased as a complete cable assembly.	Composition A preferred, but requires a deviation for Military usage. Use only for special system requirements, such as stiffness. Avoid usage where there is activity by personnel for maintenance or other reasons.

1/ Tin over zinc coating for Composition A preferred.

Detailed construction and physical properties of MIL-DTL-83420 Type I carbon and corrosion resistant wire rope is listed on Table 13. Construction and dimensional properties of MIL-DTL-83420 Type II jacketed wire rope is listed on Table 14. The breaking strengths of MIL-DTL-83420 Type I wire rope and Type II jacketed wire rope after endurance testing are listed on Tables 15 and 16. The construction, physical properties and performance properties of MIL-DTL-87218 lock-clad wire rope are shown in Tables 17 and 18.

d. Cable Strength: New cable minimum strength, ultimate design strength, and end of endurance strength as listed in Tables 13, 15, and 16 for different cable diameters and constructions, are defined as follows:

1. New Cable Minimum Breaking Strength (MBS) as required by the wire rope specification is the minimum breaking strength for the cable as received from the manufacturer.

2. Ultimate Design Strength is the maximum allowable design strength for a new cable assembly with imperfect swaged or spliced terminals and equals 85% of the new cable minimum strength for 14CFR certified aircraft. Some prime military aircraft manufacturers limit the maximum cable assembly ultimate allowable load to 75% of the new cable minimum breaking strength.

   The reduction of cable strength due to sheave/bending induced stresses can be calculated with the Macwhyte empirical formula:

$$\text{Design Strength} = 100 - 76/(D/d)^{.73}\,\%\ \text{when } D/d \text{ is greater than } 6 \qquad \text{(Eq. 3)}$$

   NOTE: *D/d* ratio of less than 40 is not recommended and 25 is the absolute minimum.

$$D/d = \frac{\text{Pulley root diameter}}{\text{cable diameter}} \qquad \text{(Eq. 4)}$$

   The swaging factor and the sheave bending stress factors are not cumulative as they do not occur at the same place in the cable. The 15 to 25% swaging factor is usually the largest factor and therefore determines the cable assembly design strength. There is no factor in the design strength for cable wear in service.

3. Endurance Strength is the minimum breaking strength of cable after a specification endurance test. There is no correlation of endurance strength to cable strength after the cable has been in service. The cable breaking strength after endurance testing is 50 to 60% of the new cable strength.

4. Limit Load is the maximum load that the cable will experience in service. It should not exceed 66.67% or 2/3 of the new cable assembly minimum strength for all cable diameters except 3/32 in or smaller. For 3/32 in or smaller cable, a design limit load of less than 50% (of the new cable assembly minimum strength) is recommended. Designers should avoid design limit loads in excess of 50% for small cables because the required proof load may cause a reduction in the cable life.

   The thermal coefficient of expansion of MIL-DTL-83420 Composition A carbon steel wire rope is 6.5×10^{-6} in/in/°F. The thermal coefficient of expansion of MIL-DTL-83420 Composition B Corrosion Resistant Steel wire rope is 9.4×10^{-6} in/in/°F.

TABLE 13 - CONSTRUCTION, PHYSICAL PROPERTIES OF MIL-DTL-83420 TYPE I, CARBON STEEL AND CORROSION RESISTANT STEEL WIRE

Nominal Rope Diameter of Wire (Reference) inch	Minimum Rope Diameter of Wire inch	Tolerance on Diameter (plus only) inch	Construction	Minimum Breaking Strength Composition A pounds	Minimum Breaking Strength Composition B pounds	Approximate Weight per 100 ft pounds
1/32	0.031	0.006	3 x 7	110	110	0.16
3/64	0.046	0.008	7 x 7	270	270	0.42
1/16	0.062	0.010	7 x 7	480	480	0.75
1/16	0.062	0.010	7 x 19	480	480	0.75
3/32	0.093	0.012	7 x 7	920	920	1.60
3/32	0.093	0.012	7 x 19	1000	920	1.74
1/8	0.125	0.014	7 x 19	2000	1760	2.90
5/32	0.156	0.016	7 x 19	2800	2400	4.50
3/16	0.187	0.018	7 x 19	4200	3700	6.50
7/32	0.218	0.018	7 x 19	5600	5000	8.60
1/4	0.250	0.018	7 x 19	7000	6400	11.00
9/32	0.281	0.020	7 x 19	8000	7800	13.90
5/16	0.312	0.022	7 x 19	9800	9000	17.30
3/8	0.375	0.026	7 x 19	14 400	12 000	24.30

TABLE 14 - CONSTRUCTION AND DIMENSIONAL PROPERTIES OF MIL-DTL-83420 TYPE II WIRE ROPE JACKET TOLERANCE

Nominal Diameter of Wire Rope inch	Construction	Outside Diameter (OD) of Jacket inch	Tolerance on Jacket OD (plus only) inch	Jacket Wall Thickness (Reference) inch	Approximate Weight per 100 ft pounds
1/32	3 x 7	0.046 (3/64)	0.008	0.008	0.22
3/64	7 x 7	0.062 (1/16)	0.010	0.008	0.49
3/64	7 x 7	0.078 (5/64)	0.012	0.016	0.76
1/16	7 x 7	0.093 (3/32)	0.012	0.016	0.93
1/16	7 x 7	0.125 (1/8)	0.014	0.031	1.18
1/16	7 x 19	0.093 (3/32)	0.012	0.016	0.93
1/16	7 x 19	0.125 (1/8)	0.014	0.031	1.18
3/32	7 x 7	0.125 (1/8)	0.014	0.016	1.85
3/32	7 x 7	0.156 (5/32)	0.016	0.031	2.18
3/32	7 x 19	0.125 (1/8)	0.014	0.016	1.99
3/32	7 x 19	0.156 (5/32)	0.016	0.031	2.32
1/8	7 x 19	0.187 (3/16)	0.018	0.031	3.62
5/32	7 x 19	0.218 (7/32)	0.018	0.031	6.10
5/32	7 x 19	0.281 (9/32)	0.022	0.063	7.51
3/16	7 x 19	0.250 (1/4)	0.018	0.031	7.75
3/16	7 x 19	0.312 (5/16)	0.022	0.063	9.20
7/32	7 x 19	0.281 (9/32)	0.020	0.031	9.76
7/32	7 x 19	0.343 (11/32)	0.024	0.063	11.55
1/4	7 x 19	0.312 (5/16)	0.020	0.031	12.30
1/4	7 x 19	0.375 (3/8)	0.024	0.063	14.42
9/32	7 x 19	0.406 (13/32)	0.024	0.063	16.18
5/16	7 x 19	0.437 (7/16)	0.024	0.063	19.80
3/8	7 x 19	0.500 (1/2)	0.027	0.063	27.20

TABLE 15 - BREAKING STRENGTH OF MIL-DTL-83420 TYPE I AFTER ENDURANCE TESTING

COMPOSITION		A		B	
Nominal Diameter of Bare Wire Rope inch	Construction	Number of Reversals 1/	MBS 2/ pounds	Number of Reversals 1/	MBS 3/ pounds
1/32	3 x 7	70 000	55	70 000	66
3/64	7 x 7	70 000	135	70 000	162
1/16	7 x 7	70 000	240	70 000	288
1/16	7 x 19	70 000	240	70 000	288
3/32	7 x 7	70 000	460	70 000	552
3/32	7 x 19	70 000	500	70 000	552
1/8	7 x 19	70 000	1000	70 000	1056
5/32	7 x 19	130000	1400	130000	1440
3/16	7 x 19	130 000	2100	130 000	2220
7/32	7 x 19	130 000	2800	130 000	3000
1/4	7 x 19	130 000	3500	130 000	3840
9/32	7 x 19	130 000	4000	130 000	4680
5/16	7 x 19	130 000	4900	130 000	5400
3/8	7 x 19	130 000	7200	130 000	7200

1/ 1 cycle = 2 reversals
2/ Equal to 50% of the original MBS.
3/ Equal to 60% of the original MBS.

TABLE 16 - BREAKING STRENGTH OF MIL-DTL-83420 TYPE II AFTER ENDURANCE TESTING

COMPOSITION		A		B	
Nominal Diameter of Bare Wire Rope inch	Construction	Number of Reversals 1/	MBS 2/ pounds	Number of Reversals 1/	MBS 2/ pounds
1/32	3 x 7	200 000	66	200 000	66
3/64	7 x 7	200 000	162	200 000	162
1/16	7 x 7	200 000	288	200 000	288
1/16	7 x 19	200 000	288	200 000	288
3/32	7 x 7	200 000	552	200 000	552
3/32	7 x 19	200 000	600	200 000	552
1/8	7 x 19	200 000	1200	200 000	1056
5/32	7 x 19	500 000	1680	500 000	1440
3/16	7 x 19	500 000	2520	500 000	2220
7/32	7 x 19	500 000	3360	500 000	3000
1/4	7 x 19	500 000	4200	500 000	3840
9/32	7 x 19	500 000	4800	500 000	4680
5/16	7 x 19	500 000	5880	500 000	5400
3/8	7 x 19	500 000	8640	500 000	7200

1/ 1 cycle = 2 reversals
2/ Equal to 60% of the original MBS.

TABLE 17 - MIL-DTL-87218 LOCK-CLAD CLASSES AND PHYSICAL PROPERTIES

Class Identity	Base Wire Rope 1/ Diameter Nominal (Reference)	Base Wire Rope 1/ Diameter Minimum	Base Wire Rope 1/ Construction (Reference)	Finished Lock-Clad Cladding Diameter inch ± 0.003	Lock-Clad Weight Maximum pounds/100 ft
2A	1/16	0.062	7 x 7	0.1495	2.60
2B	1/16	0.062	7 x 7	0.1695	3.10
3A	3/32	0.093	7 x 7	0.1695	3.80
3B	3/32	0.093	7 x 7	0.2010	4.80
3C	3/32	0.093	7 x 19	0.1695	3.80
3D	3/32	0.093	7 x 19	0.2010	4.80
4A	1/8	0.125	7 x 19	0.2010	5.56
4B	1/8	0.125	7 x 19	0.2500	7.45
5A	5/32	0.156	7 x 19	0.2500	8.61
5B	5/32	0.156	7 x 19	0.2950	10.71
6A	3/16	0.187	7 x 19	0.2500	9.71
6B	3/16	0.187	7 x 19	0.2700	11.02
6C	3/16	0.187	7 x 19	0.2950	12.02
6D	3/16	0.187	7 x 19	0.3020	12.49
8A	1/4	0.250	7 x 19	0.3320	17.43
8B	1/4	0.250	7 x 19	0.3580	19.32
8C	1/4	0.250	7 x 19	0.3750	20.47

1/ Wire rope per MIL-DTL-83420 Type 1 Composition A or B.

TABLE 18 - MIL-DTL-87218 LOCK-CLAD PERFORMANCE PROPERTIES

Class (see Table 17)	Maximum Percent Elongation at Designated Load 5% MBS	Maximum Percent Elongation at Designated Load 60% MBS	Minimum EA Value at Designated Load x 10^3 (pound/inch/inch) Type I Lock-Clad 1/ 5% MBS	Type I Lock-Clad 1/ 60% MBS	Type II Lock-Clad 2/ 5% MBS	Type II Lock-Clad 2/ 60% MBS	Coefficient of Thermal Expansion (Minimum) 32 - 80 °F (inch/inch X 10^6 per °F) Type I 1/	Type II 2/
2A	0.06	0.4	111	166	111	166	11.9	12.2
2B	0.06	0.4	144	216	144	216	12.4	12.4
3A	0.06	0.4	165	247	157	235	11.1	11.3
3B	0.06	0.4	213	320	209	314	11.8	12.0
3C	0.06	0.4	165	247	157	235	11.1	11.3
3D	0.06	0.4	213	320	209	314	11.8	12.0
4A	0.06	0.4	237	355	228	343	10.4	10.5
4B	0.06	0.4	330	495	322	483	11.5	11.5
5A	0.06	0.4	372	558	362	543	10.6	10.7
5B	0.06	0.4	459	689	449	674	11.4	11.4
6A	0.06	0.4	402	603	382	573	9.3	9.3
6B	0.06	0.4	452	678	432	655	9.8	10.1
6C	0.06	0.4	501	751	482	723	10.3	10.4
6D	0.06	0.4	534	801	515	773	10.6	10.7
8A	0.06	0.4	720	1080	680	1019	9.5	9.7
8B	0.06	0.4	780	1170	741	1112	10.0	10.2
8C	0.06	0.4	870	1305	830	1246	10.3	10.4

1/ Type I lock-clad is manufactured with MIL-DTL-83420, Type I, Composition A carbon steel, zinc or tin over zinc coated wire rope. Tin over zinc coating is preferred.
2/ Type II lock-clad is manufactured with MIL-DTL-83420, Type I, Composition B, corrosion resistant steel wire rope.

e. Elongation (EA) values of lock-clad cable are controlled by specification MIL-DTL-87218 to +10/-5% of the values listed in Table 18. Most lock-clad cable assemblies have sections of bare, or Type I, cable and the *EA* value of the assembly is:

$$E_{assy} = \frac{L_{assy}}{\frac{L_{lock-clad}}{EA_{lock-clad}} + \frac{L_{bare}}{EA_{bare}}} \quad \text{(Eq. 5)}$$

As manufactured elongation is not controlled for Type I and II cables per MIL-W-83420, and have a wide range of values. The amount of unit deflection at the 60% proof load is allowed to be a maximum of 1.5%.

Cables are pre-stretched and proof loaded after swaging per MIL-DTL-5688. This process stretches the cables, improving stiffness over the manufactured cable stiffness. Cable is typically more flexible at low loads and improves as the load is increased. As the load increases, the gaps between the individual wires are eliminated. Table 19 provides stiffness data for various pre-stretched MIL-DTL-83420 Type 1 cable sizes as a function of applied load. Most control cable systems are preloaded within the 2% MBS to 10% MBS range. Table 19 shows the results of two independent sets of cable tests and highlights the variability associated with cables and cable testing. The cable system designer needs to be cautious where installed cable stiffness becomes a significant design consideration. The EA (load/deflection) units are in pound/inch/inch of cable length.

TABLE 19 - BARE CABLE DEFLECTION PROPERTIES

Composition A (Carbon Steel)

Cable Size	EA Values under 2% MBS		EA Values under 10% MBS	
	TEST A	TEST B	TEST A	TEST B
1/16 in 7x7	20 400	19 100	40 200	38 800
3/32 in 7x7	39 800	32 900	81 000	69 400
1/8 in 7x19	62 200	53 900	129 300	111 800
5/32 in 7x19	89 100	Not Tested	187 100	Not Tested
3/16 in 7x19	129 900	128 700	275 900	283 600
1/4 in 7x19	222 900	Not Tested	478 900	Not Tested

Composition B (CRES)

Cable Size	EA Values under 2% MBS		EA Values under 10% MBS	
	TEST A	TEST B	TEST A	TEST B
1/16 in 7x7	18 400	20 700	36 100	33 200
3/32 in 7x7	35 800	39 300	72 700	65 300
1/8 in 7x19	55 800	60 200	116 200	104 100
5/32 in 7x19	80 000	85 200	168 100	152 600
3/16 in 7x19	116 700	113 600	247 900	221 800
1/4 in 7x19	200 200	Not Tested	430 300	Not Tested

f. Cable-Pulley Friction: The friction developed by a cable-pulley combination is related to the energy necessary to wrap and form the cable to the pulley and groove diameters and then unwrap it as the cable is stroked. Energy is used to bend the individual strands which make up the wire cable. Energy is used to stretch the outer strands and to compress the inner strands during bending. Because of the twisted strand construction, the outer and inner strands reverse positions every half twist along the cable. Most energy is used to overcome the sliding friction between strands as they shift to minimize the resultant tension and compression stress cycles along their length. Cable tension tends to press the twisted strands together. This action results in an increase of cable-pulley friction that is proportional to an increase of cable tension. Internal cable friction is also increased on the smaller pulleys where the pulley groove diameter is larger than the cable diameter, which allows the cable to flatten in the groove. On the larger pulleys, groove bearing and cable bending stresses become too low to flatten the cable. Pulley bearing no-load torque may be an appreciable part of cable-pulley friction. Depending on the bearing and seal construction and the relative cable tension and radius on the pulley, bearing torque may be 4 to 40% of the friction at a pulley.

Some useful relations for cable-pulley friction determinations are shown in Figures 17 and 18 which summarize the data recorded from room temperature tests of carbon steel cable applied to all pulley sizes. The cable friction force (F) increases with cable tension (T) as an inverse function of pulley diameter and non-linear function of wrap angle. Pulley groove diameter also impacts cable friction.

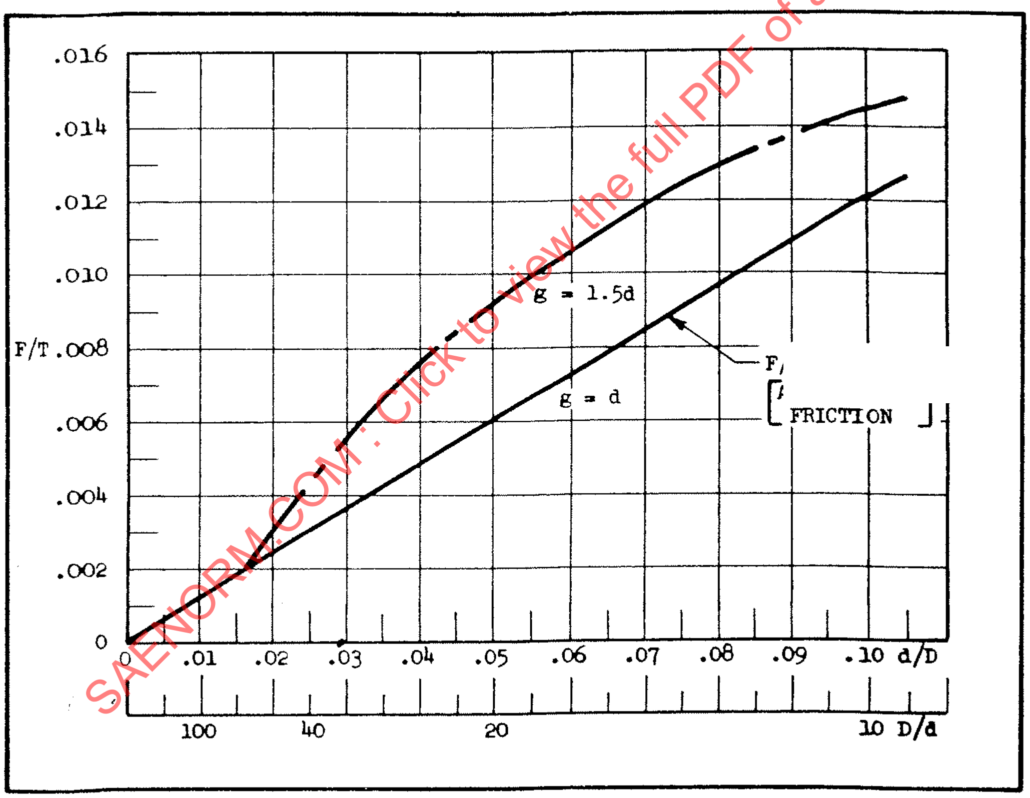


FIGURE 17 - CARBON STEEL CABLE FRICTION AT 180 DEGREE WRAP VERSUS PULLEY AND GROOVE DIAMETER

NOTE: Pivot bearing friction not included.

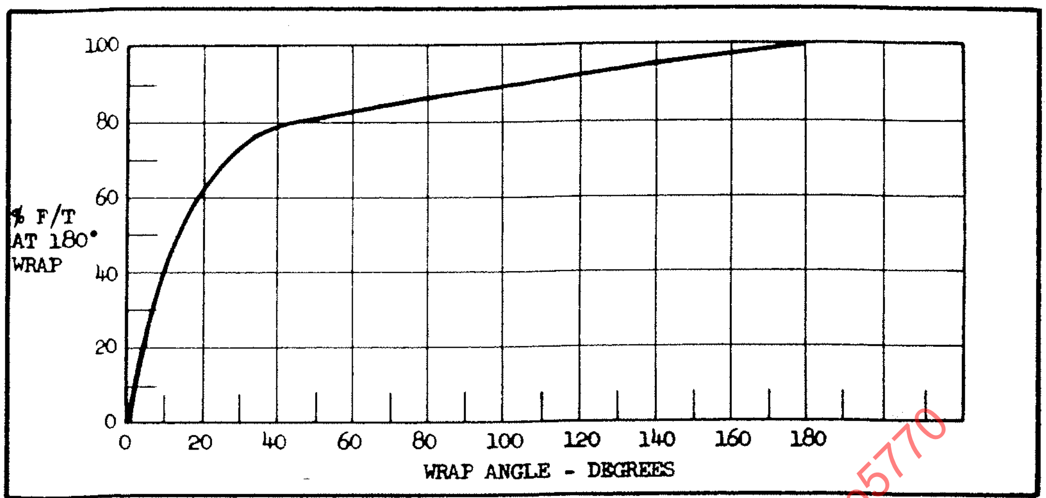


FIGURE 18 - CARBON STEEL CABLE FRICTION VERSUS PULLEY WRAP

Examination of Figures 17 and 18 will reveal the desirability of a system containing a few large pulleys with large wrap angles rather than one containing numerous small pulleys with small wrap angles.

g. Cable Friction and Cable Life: Cable friction and wear should be minimized because friction increases operating forces and wear decreases life. Table 20 shows design criteria for low cable friction and wear by limiting cable contact and minimizing the effects of the relative sliding of individual wires when the cable is bent over sheaves.

Figures 20 and 21 show the effects of pulley size and wrap angle on the friction between cable and pulley. This data provides more detail than Figures 17 and 18. This data is used to calculate total system friction.

To improve cable service life, pulleys should be spaced so the cable wrapped on one pulley does not wrap onto an adjacent pulley.

TABLE 20 - FRICTION AND CABLE LIFE DESIGN CRITERIA FOR MECHANICAL CONTROLS

Factor	Design Criteria	Reason
Number of Pulleys and Seals or Grommet Supports	Use minimum number of pulleys or supports with maximum spacing of: 1. 200 in (510 cm) in body 2. 75 in (190 cm) in wing or other high vibration areas	Each pulley and cable support adds to friction, but cable life is affected only by spacing of supports which prevents cable contact and wear with structure or other components.
D/d Ratio (Sheave groove dia. to cable dia. ratio)	Use largest *D/d* ratio possible consistent with weight requirements: 60 *D/d* desirable for low friction 40 *D/d* minimum for good cable life 25 *D/d* absolute minimum with penalty for friction and life	Cable friction and wear are reduced by an increase in the *D/d* ratio. See Figure 19 for life effect and Figure 17 for friction effect.
Wrap Angle	Use wrap angles as required except in wing, or other vibrational areas, where a positive wrap of 2 degrees is required.	Wrap angle is a friction parameter (see Figure 18) but affects wear only in a vibratory environment if the cable is allowed to slap groove flanges.
Sheave Groove Radius	Use groove radii that conforms to cable plus tolerance.	Groove radius of d/2 + 0.010 minimizes friction and cable wear by minimizing cable ovalization and cable "wedging" where d = cable dia.
Cable Diameter	Use 3/32 or 1/8 in diameter cables if possible. 14CFR Part 23 and Part 25 require 1/8 diameter or larger cables for aileron, elevator, and rudder control systems.	Smallest cable that meets strength and performance requirements minimizes system weight.
Rig Load	Use smallest rig load possible with a no-slack cable condition at low temperature and the maximum normal operating load.	Rig load is a parameter of friction (see Figures 17 and 18) and wear.
Cable Lubrication	Retain cable lubrication by protective environment or by re-lubrication. **Exception:** Do not lubricate CRES cables.	Lubrication is necessary for cable fatigue strength and corrosion protection. See 5.1.b. Grease collects grit which wears CRES cable prematurely.
Cable Construction	Use 3/32 dia 7 x 7 or 1/8 dia 7 x 19 construction for normal installations.	3/32 dia 7 x 7 construction has larger wire diameter than 7 x 19 and can withstand more wear without wire breakage.
Composition of Coating Over Cable Wires	Use tin over zinc coating.	Zinc coating alone has high friction and high rig load decay.

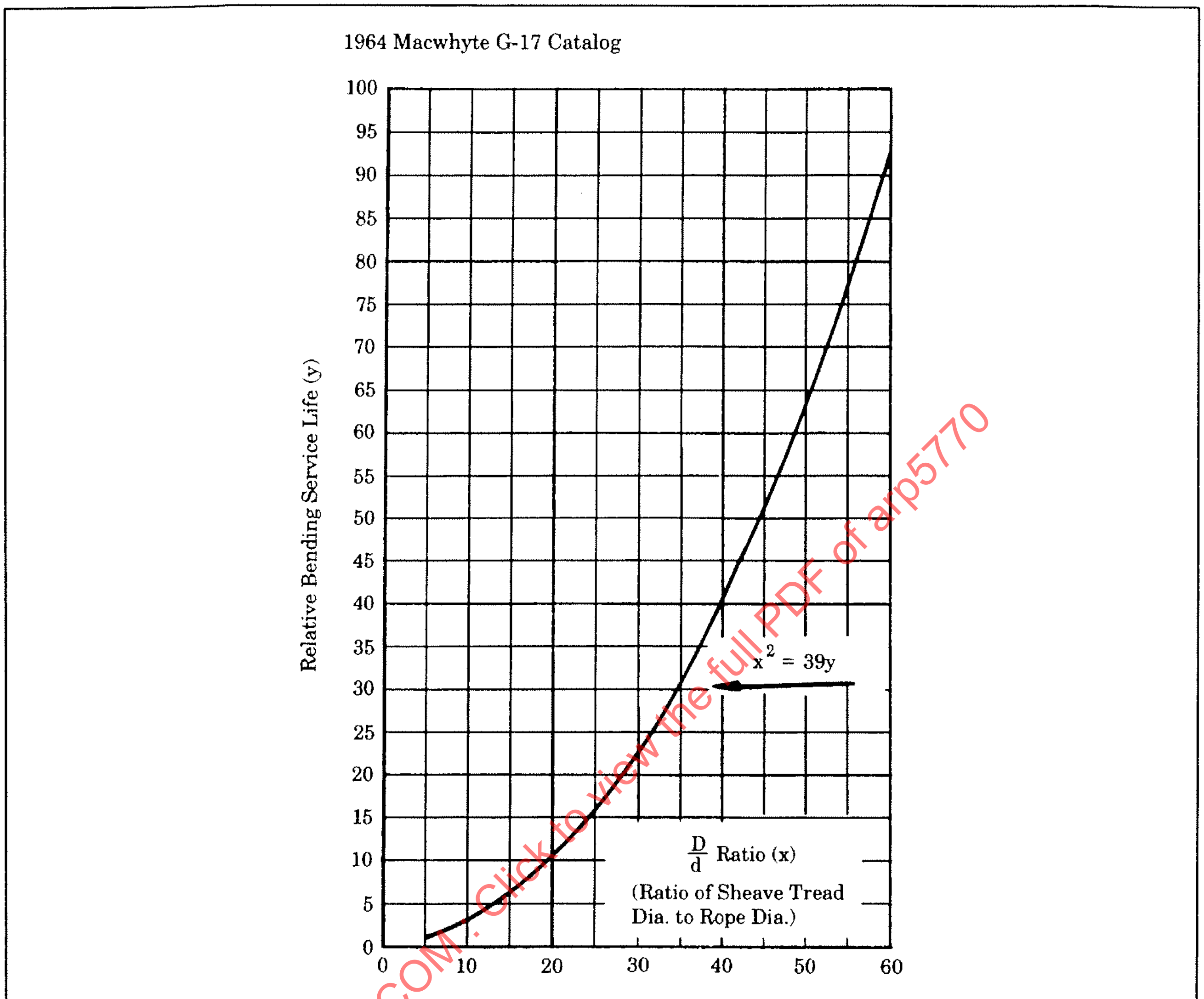


FIGURE 19 - SHEAVE-CABLE DIAMETER RATIO EFFECT ON SERVICE LIFE

NOTE: Sheave tread diameter is pulley minor diameter that cable contacts.

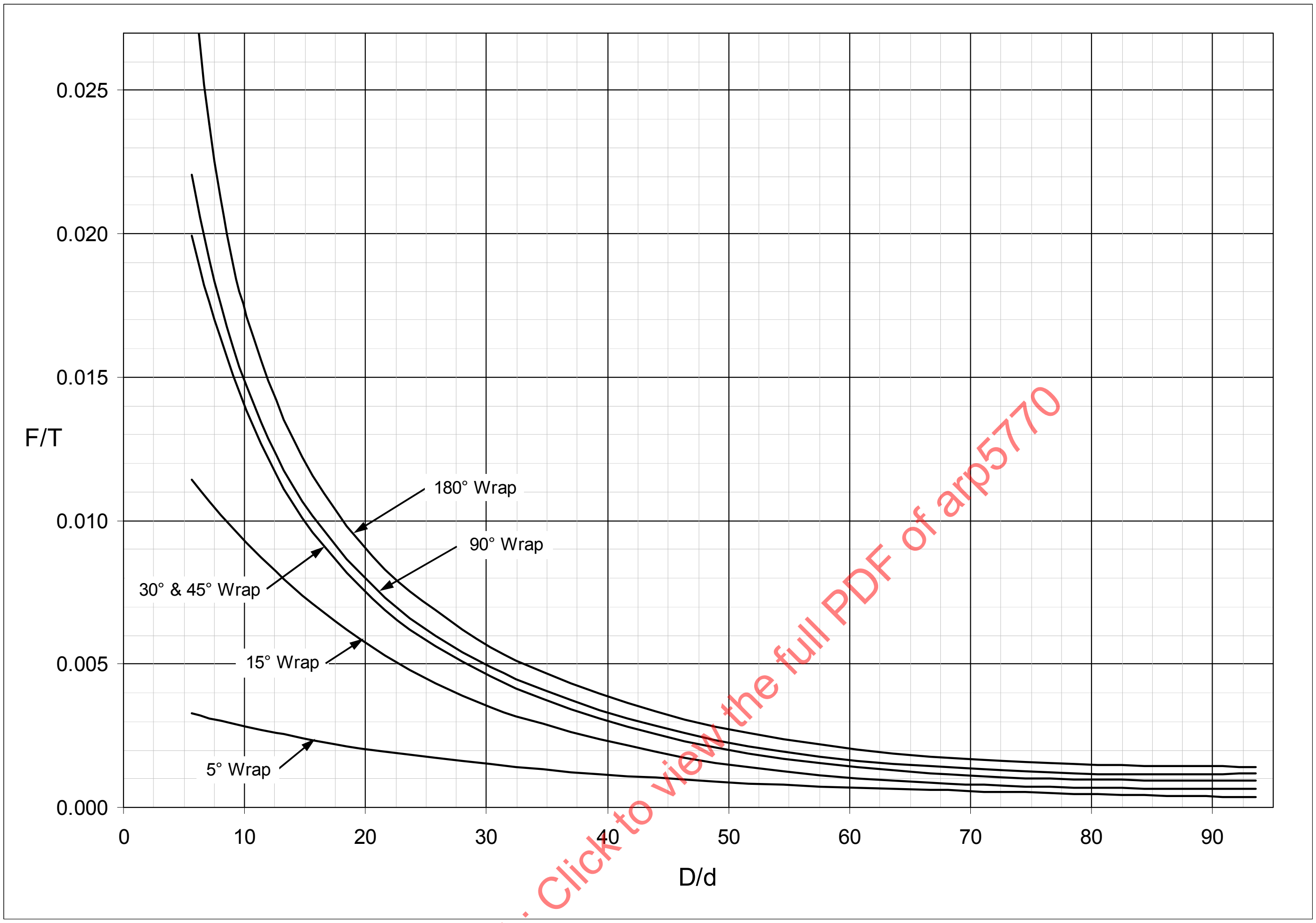


FIGURE 20 - CABLE-SHEAVE FRICTION WITH GROOVE RADIUS = (*D*/2) + 0.020 in (0.050 cm)

F = Friction - pounds or Newtons at 70 °F
T = Cable tension - pounds or Newtons at 70 °F
D = Pulley root diameter - inches or centimeters
d = Cable diameter - inches or centimeters

Cables are zinc coated

NOTE: These friction curves include 0.5 in-oz (36 g-cm) of friction for one PD5K bearing. If more or larger bearings are used, add extra bearing friction per Table 7.

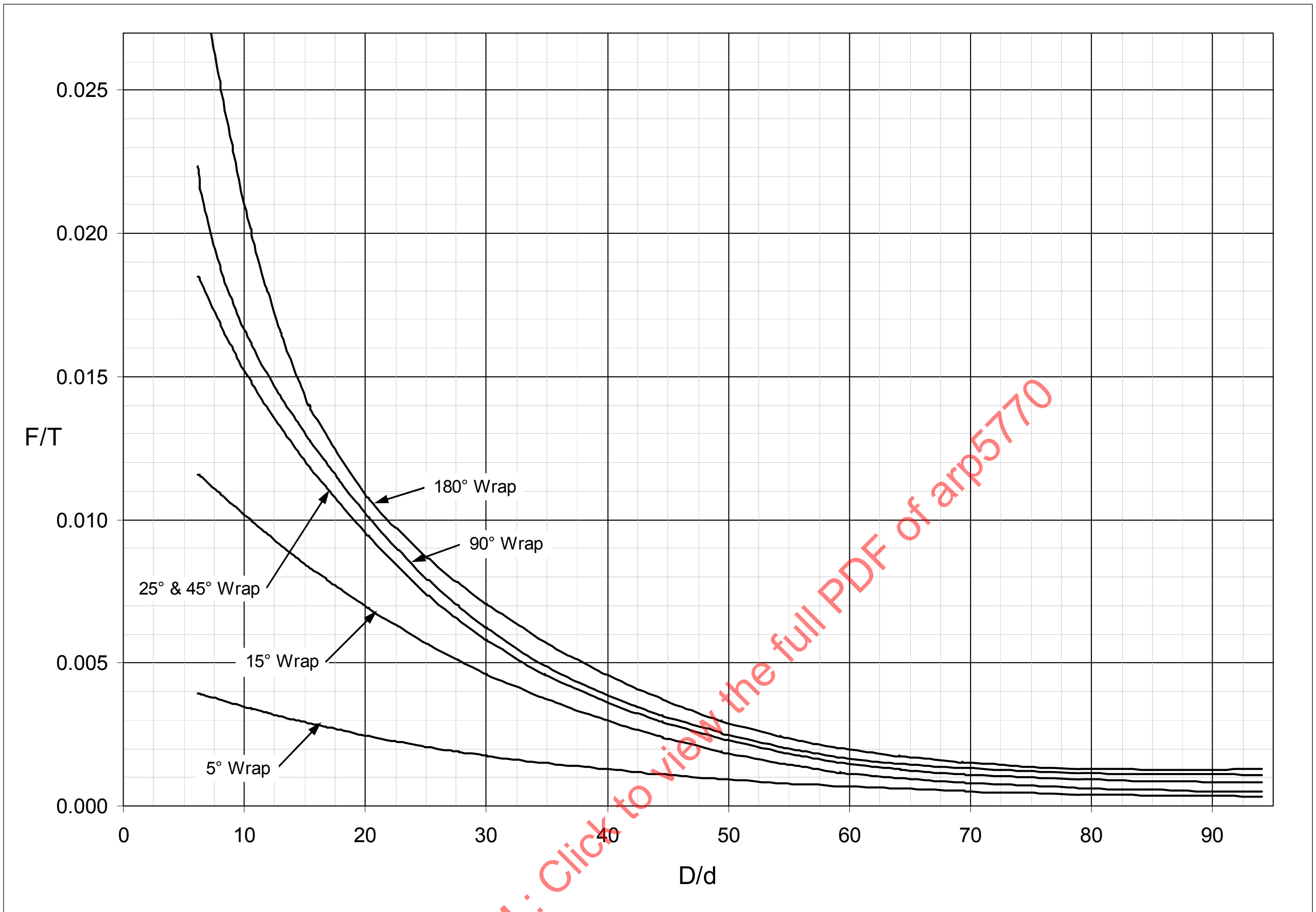


FIGURE 21 - CABLE-SHEAVE FRICTION WITH GROOVE RADIUS = (*D*/2) + 0.036 in (0.091 cm)

F = Friction - pounds or Newtons at 70 °F
T = Cable tension - pounds or Newtons at 70 °F
D = Pulley root diameter - inches or centimeters
d = Cable diameter - inches or centimeters

Cables are zinc coated

NOTE: These friction curves include 0.5 in-oz (36 g-cm) of friction for one PD5K bearing. If more or larger bearings are used, add extra bearing friction per Table 7.

h. Cable Rigging Load: A closed cable system is rigged to a predetermined load that keeps the cables taut and minimizes control system backlash. The rig load also eliminates approximately 50% of the cable deflection, due to operating loads in the cable system, because the operating load is divided almost equally between both input quadrant cables. This reduction in handle movement, for cable deflection due to load, can be seen by the study of the diagrams of 5.1.h.2 and 5.1.h.3 for cable systems without and with a rig load.

In the diagram Without a Rig Load, the operating load, *P*, is taken by only a single cable, while in the diagram with a Rig Load, the load, *P*, is shared equally, *P/2* by both cables, if the *EA* value for both cables is the same. This results in 50% less movement in control input due to cable deflection under load. Note that the decrease in control input due to cable deflection occurs only as long as the cable rigging load is half of the operating load or larger, shown below:

$$T \geq P/2 \qquad \text{(Eq. 6)}$$

The formula of 5.1.h.3 assumes that the *EA* value remains constant. However, the formula will give close results when the *EA* value changes with load, as each cable's share of the load, *P*, also changes.

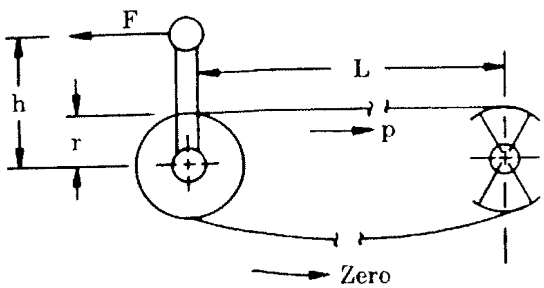
For definition of applicable symbols and terminology, see Table 21.

TABLE 21 - DEFINITION OF SYMBOLS AND TERMINOLOGY FOR CABLE APPLICATIONS

Common Terminology	**Temperature Change Terminology**
δ = cable stretch due to handle load, F	A_a = average temperature change in body skin temperature from 70 °F
L = cable path length at 70 degrees	A_n = change in cable temperature from 70 °F in zone n
P = Fh/r, cable load from handle load, F	C_a = thermal coefficient of aircraft structural expansion: - aluminum structure - 12.5 x 10^{-6} in/in/°F - composite structure - 6.7 x 10^{-6} in/in/°F
Pf = cable final load	
Pi = cable initial load	C_c = thermal coefficient of expansion of the cable: - MIL-DTL-83420 Composition A Carbon steel 6.5 x 10^{-6} in/in/ °F - MIL-DTL-83420 Composition B corrosion Resistant Steel 9.4 x 10^{-6} in/in/ °F - MIL-DTL-87218 Lockclad - See Table 17
T = cable rig load	
ΔT = change in cable rig load	
R = system rigidity factor	
EA = cable's primary modulus at an average cable load (includes any rig load), $1/2\,(P_f + P_i)$ (Eq. 7)	L_n = length of cable at 70 °F in zone n
	δ_t = difference in the length of the cable and aircraft due to a temperature change from 70 °F
Structure Deflection Terminology	**Cabin Pressurization Terminology**
δ_s = change of cable length due to structural deflection	δ_p = cable length change due to cabin pressurization
δ_{sd} = structural deflection from the wing center section	E_o = modulus of elasticity of aircraft skin - aluminum - 10.5 x 10^6 psi - composite - 10 x 10^6 psi
D_{na} = cable path distance from the structure's neutral axis	
L_{sd} = cable path length which is deflected	L_p = longitudinal length of cable path in pressurized cabin $(P/A)_p$=change in longitudinal body skin stresses due to cabin pressurization

1. System Rigidity Factor: The system rigidity factor, *R*, (Table 21) accounts for deflection of brackets, their support structure, and an allowance for the actual *EA* value of the average cable. The rigidity factor cannot be calculated, but is determined by system load-deflection tests. The factor varies from 1.05 to 1.8, depending upon the system. In design, use a factor of 1.3, or a factor of 1.0 plus 0.015 for each major directional change by a pulley, each quadrant, and each drum in the cable path.

2. Cable Deflection without a Rig Load

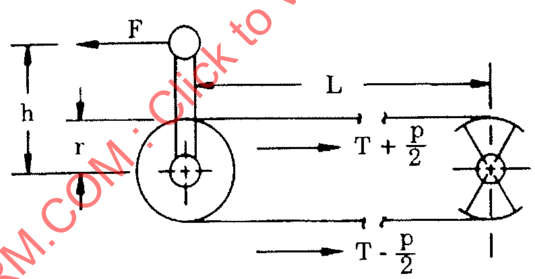


$$P_i = O$$

$$P_f = P \qquad \text{(Eq. 8)}$$

$$\delta = \frac{(P_f - P_i)LR}{EA} = \frac{PLR}{EA}$$

3. Cable Deflection with a Rig Load



$$P_i = T$$

$$P_f = T \pm P/2 \qquad \text{(Eq. 9)}$$

$$\delta = \frac{(P_f - P_i)LR}{EA} = \frac{PLR}{2EA}, \text{ if } P \leq 2T$$

4. Temperature Change Effect on Cable Rig Load: The change in cable rig load, *T*, due to a change in ambient temperature requires a detailed survey to determine the average temperature change of the aircraft skin, and the cable in significant temperature zones in the aircraft (see Figure 22). Calculate ΔT by:

$$\Delta T = \delta_t EA / LR$$

$$\delta t = A_a C_a L - C_c(A_1L_1 + A_2L_2 + \cdots + A_nL_n) \qquad \text{(Eq. 10)}$$

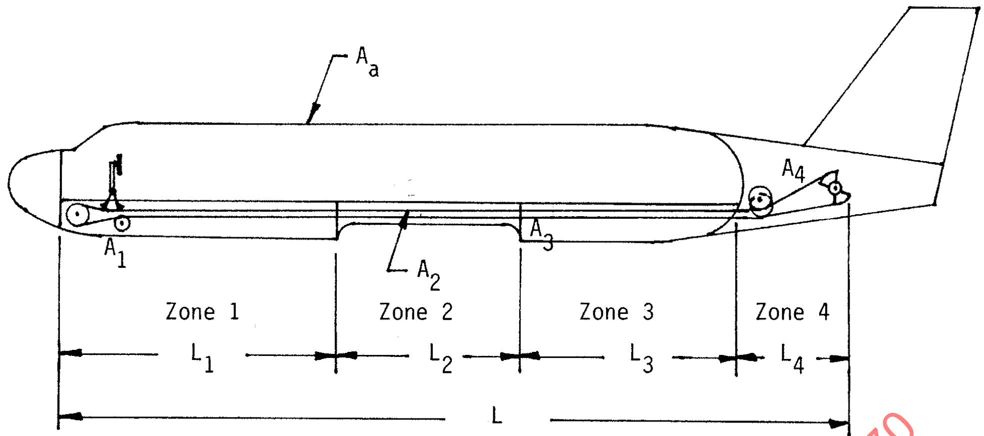


FIGURE 22 - SIGNIFICANT CABLE TEMPERATURE ZONES

5. Structure Deflection Effect on Cable Rig Load: The cable rig load will change due to airplane structural deflections if the cable path is not located exactly on the neutral bending and torsional axes of the aircraft.

   Structural deflections which have an effect are the movement on the aircraft's nose, tail, and wing tips in respect to the relatively rigid wing center section. Calculate ΔT by:

$$\Delta T = (\delta_s EA)/RL \qquad \text{(Eq. 11)}$$

$$\delta_s = \delta_{sd}(D_{na}/L_{sd})$$

6. Cabin Pressurization Effect on Rig Load: Cable rig load will increase when the cabin is pressurized due to the increase in the length of the cabin and the cable. Calculate ΔT by:

$$\Delta T = (\delta_p EA)/RL \qquad \text{(Eq. 12)}$$

$$\delta_p = (P/A)_p (L_p/E_O)$$

7. Minimum Rigging Load: The minimum rig load shall be taken as 1/2 of the cable load generated by the maximum normal operating force (not limit or design load) at the worst flight operating condition in the normal service envelope that includes structural deflections as well as low temperatures.

8. Rig Load Creep and Rigging Instructions: Rig load creep is the loss of rig load due to the system not being initially rigged to stabilize the rig load. Rigging instructions should require cables to be rigged by the following method:

   - Tension cables to twice the specified rig load.
   - Cycle system 25 or more times through complete travel.
   - Reduce cable tension to specified rig load.

   Cable lengths should be designed to allow sufficient turnbuckle adjustment to obtain a cable tension equal to twice the system rig load with cables having the maximum 1.5% elongation. The number of standard turnbuckles necessary to meet these requirements is:

$$n = \frac{L\left(0.001 + \frac{2T}{(EA)_{min}}\right)}{L_T - 0.4} \quad \text{(Eq. 13)}$$

where:

n = number of turnbuckles = no.

L = length of cable path = inches

L_T = length of turnbuckle adjustment = inches

T = cable rig load = pounds

$(EA)_{min}$ = primary modulus at rig load T for cable having 1.5% elongation at 60% proofload (for standard cable without lock-clad) = pounds

0.001 = manufacturing tolerance, inch/inch of assembly length

0.4 = locking clip clearance in center at turnbuckle and structure tolerance.

i. Cable Adjustment: Cable turnbuckles are used to set the rigging load in closed cable systems. Turnbuckle barrels have a right-hand thread on one end and a left-hand thread on the opposite end. Turnbuckles also serve as a production break between major aircraft structural assemblies. Turnbuckles should be installed in an accessible area close to a production break and staggered to prevent cross-coupling. A typical cable turnbuckle is shown on Figure 23.

1. For minimum safe thread engagement, no more than three male threads should be exposed when rigging is complete.

2. For the desired available adjustment for future cable stretch and structural deflection, the cable terminal threads should not be buried more than three threads when the original rigging is accomplished.

3. Turnbuckles are locked with two MS21256 clips as shown on Figures 23 and 24. The clips should be installed one time and not reused when removed.

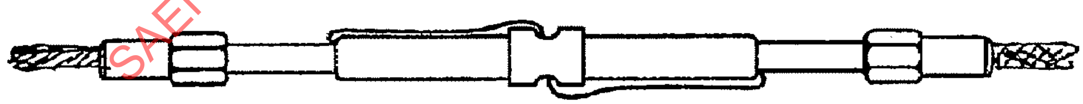
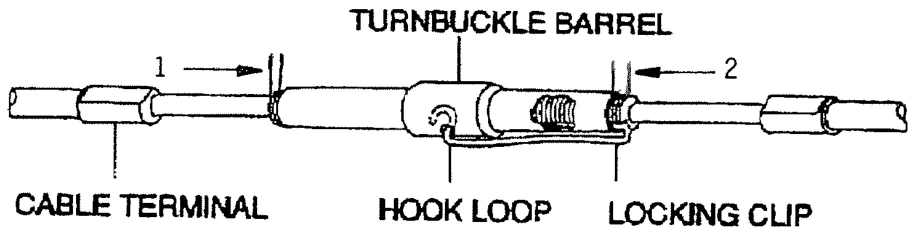


FIGURE 23 - CABLE TURNBUCKLE WITH LOCKING CLIPS



NOTE 1 and 2: Adjustment range when installed with proper cable tension load - three threads exposed to three threads buried.

FIGURE 24 - CABLE TURNBUCKLE LOCKING CLIP INSTALLATION

A very long small diameter cable would require more take-up to achieve rigging tension than that available in one turnbuckle. The small amount of turnbuckle adjustment (three threads exposed to three threads buried) makes it necessary to have some other adjustment in the cable system to compensate for tolerances in cable length, bracket locations, fuselage length, etc. In a cable system that terminates at a bellcrank, the use of an AN665 fork and MS21259 male threaded terminal will usually provide sufficient adjustment for the turnbuckle to be set properly. Refer to Figure 25. If more take-up is required, a special fork and terminal with less or greater length of female thread as shown in Figure 26 may be required.

NOTE: Because the AN665 or special fork must be disconnected from the bellcrank in order to turn it on the threads, the control rigging load should not be adjusted at this point.

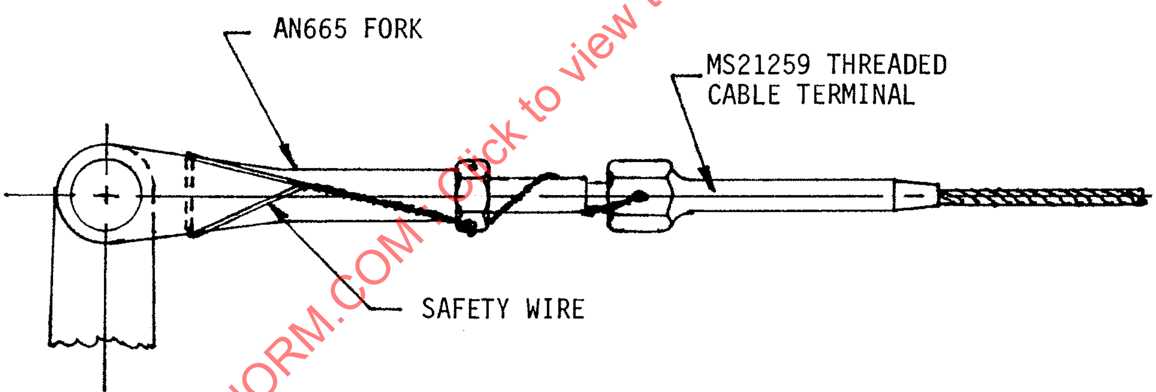


FIGURE 25 - FORK CABLE TERMINAL

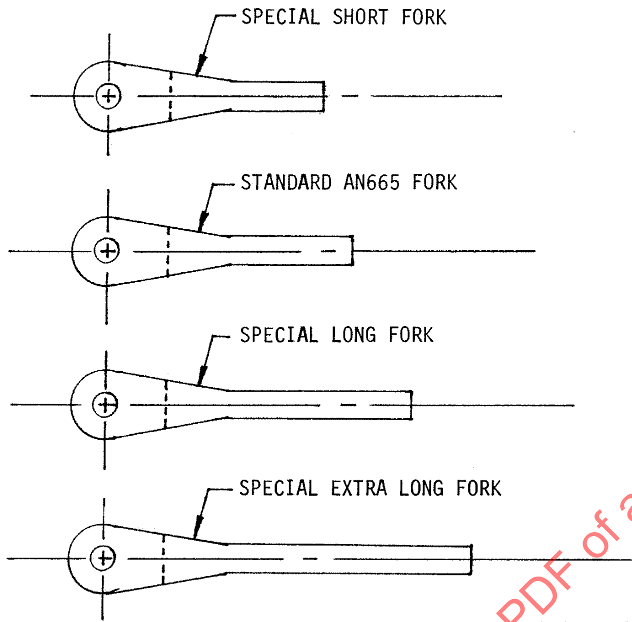


FIGURE 26 - SPECIAL FORKS

Rather than use a multiplicity of turnbuckles on a quadrant/quadrant control system, the cable should be taken up by use of a hydraulic cable tensioning device which draws the two ends of the cable together before the turnbuckle is connected. The ground equipment tensioning device is removed after the turnbuckle is connected. Even with this approach, multiple turnbuckles may be required.

Turnbuckles per MS21251B (Brass) shall be used with locking clips per MS21256.

Turnbuckle assemblies must comply with the requirements of TSO-C21b. This is a Technical Standard Order issued by the FAA. It prescribes the minimum performance standards that turnbuckle assemblies and/or safetying devices must meet. Marking of the assembly to show compliance is required.

Cross-connection of control cables should be made impossible by proper spacing of turnbuckles and opposite handed threads.

Cable rigging should be possible with the airplane on or off jacks and with or without fuel.

5.2 Cable Tension Regulators

Cable tension regulators (CTRs) are used to maintain a more constant cable tension under all operational and environmental conditions. Because the cable tension is more constant with CTRs, the cable tension can be set at a lower nominal level, resulting in lower system friction. The intervals between cable re-rigging can be increased with the use of CTRs.

Cable tension regulators add weight, cost, and complexity to the cable system. These disadvantages are typically outweighed by the advantages for primary control and engine control cable systems. The following are some of the advantages of CTRs:

a. Thermal Expansion

Aircraft structure and control cables are typically fabricated from different materials and expand at different rates due to thermal changes. The control cable system temperature typically lags the aircraft structural temperature changes. CTRs compensate for all control system thermal differences.

b. Structural Deflections

It is not always possible or practical to install the cable system on the aircraft fuselage or wing structural neutral axes. As a result, the cable system installed length will vary with structural deflections, varying cable tension loads. CTRs compensate for structural deflection variations in cable length and maintain a near constant cable tension preload.

c. Airframe Pressurization

Pressurization of the airframe cabin area can result in significant changes in control cable installed length versus cable length. CTRs compensate for pressurization variations in control cable length.

d. Cable Rig Tension

Without CTRs, high initial tension rig loads must be utilized to prevent the cables from becoming slack under some thermal or structural deflection conditions. Under other thermal or deflection conditions, the cable tension preload will be higher than the initial rig load. Cable/pulley friction varies proportional to the rig load, resulting in less than ideal friction levels. CTRs maintain a more nearly constant cable rig load, allowing the use of lower cable rig loads and avoiding the higher cable friction loads.

e. Wear and Resonances

Lower cable tension loads reduce control cable installation wear. The narrower band of cable tension variations allows the resonant frequency of the cables to be controlled away from critical airframe resonances.

A cable system with a CTR installed is shown in Figure 27. The CTR consists of two cable sectors that are preloaded by springs and links to preload the control cables. The crosshead slides on the lock shaft and preloads the control cables. When rotated, the crosshead locks on the lock shaft and transmits torque in the control system. Some CTRs are designed to lock in position if one of the cables becomes slack or breaks.

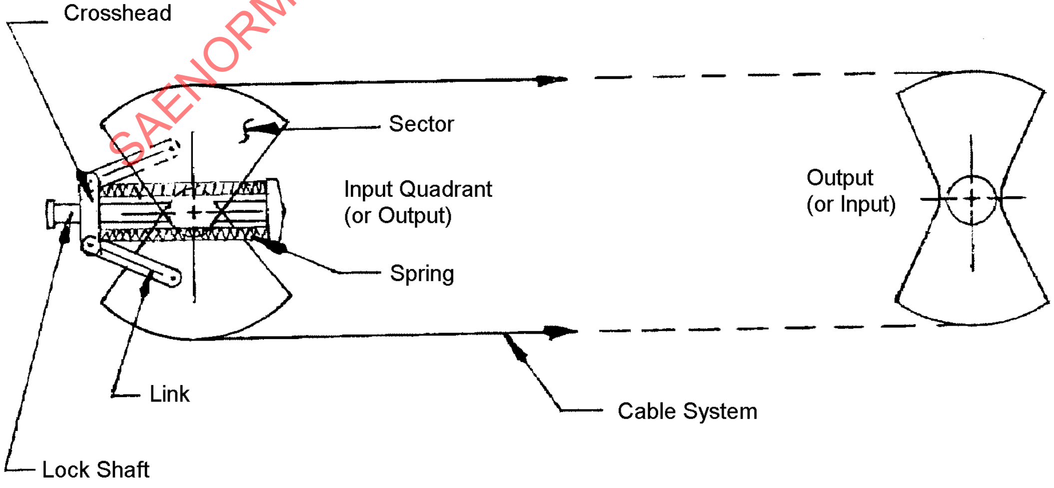


FIGURE 27 - CABLE SYSTEM WITH CTR INSTALLED

5.3 Sheaves

Sheaves are grooved wheels or sectors of wheels, such as pulleys, quadrants or drums, over which cables travel.

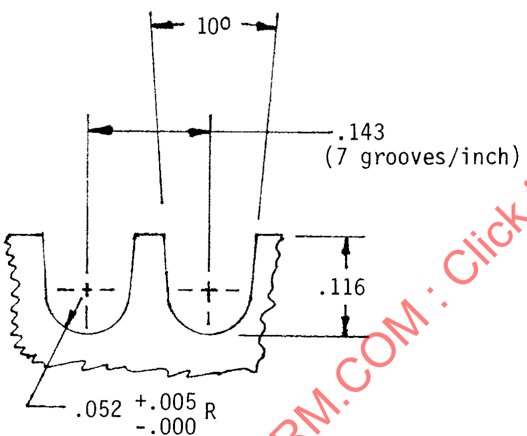
a. Pulleys

1. Only conformance groove pulleys with a *D/d* ratio greater than 40 should be used in new designs, but a 25 to 40 ratio is allowable in low fatigue applications. The pulley grooves should have a root radius of *d/2* + 0.010 where *d* is the cable diameter) to minimize friction and cable wear, since control cable diameter tends toward the maximum tolerance.

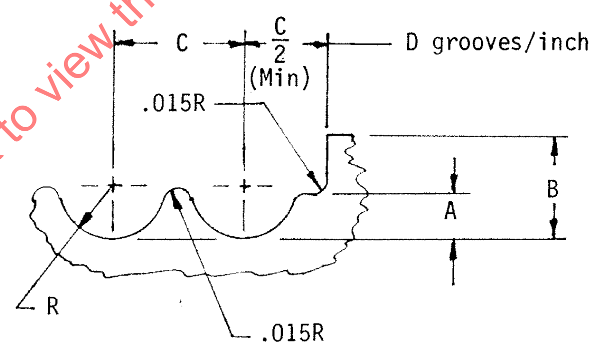
b. Drums: Drums are used to terminate cable systems when the sheave travel is more than 270 degrees. Drums shall have at least 10 degree wrap of the driving cable after limits of travel (including overtravel) in both directions are reached.

1. Drum Cable Grooves: Recommended callouts for cable grooves on a drum are shown in Figures 28 and 29 and Table 22.

   These groove dimensions provide for a 1/16 and 3/32 cable leaving the drum up to 4 degrees from normal drum axis without rubbing on the side of the groove. Note: There is a change in alignment of the cable as it is wound onto a drum. This may result in change of initial cable tension.



1/16 and 3/32 Cable



1/8, 5/32, 3/16 Cable

FIGURE 28 - DRUM GROOVE DIMENSIONS

TABLE 22 - DRUM GROOVE DIMENSIONS

Cable Dia.	A	B ± 0.005	C	D (Ref.) Grooves/inch	R +0.005/-0.000
1/8	0.062	0.144	0.182	5.50	0.076
5/32	0.078	0.177	0.216	4.63	0.093
3/16	0.094	0.211	0.250	4	0.109

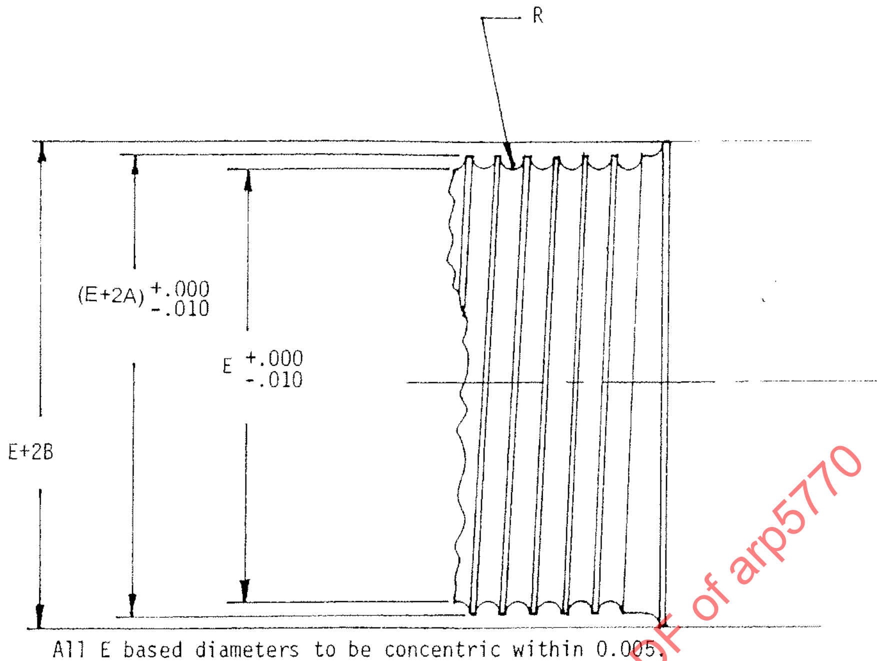


FIGURE 29 - DRUM GROOVES FOR 1/8, 5/32, AND 3/16 CABLE
(SEE FIGURE 28 FOR 1/16 AND 3/32 CABLE)

2. Drum Cable Terminations: For drum cable terminations, use swaged cylindrical terminals. On large drums where space permits, swaged eye terminals are optional. Figures 30 and 31 show typical drum cable terminations.

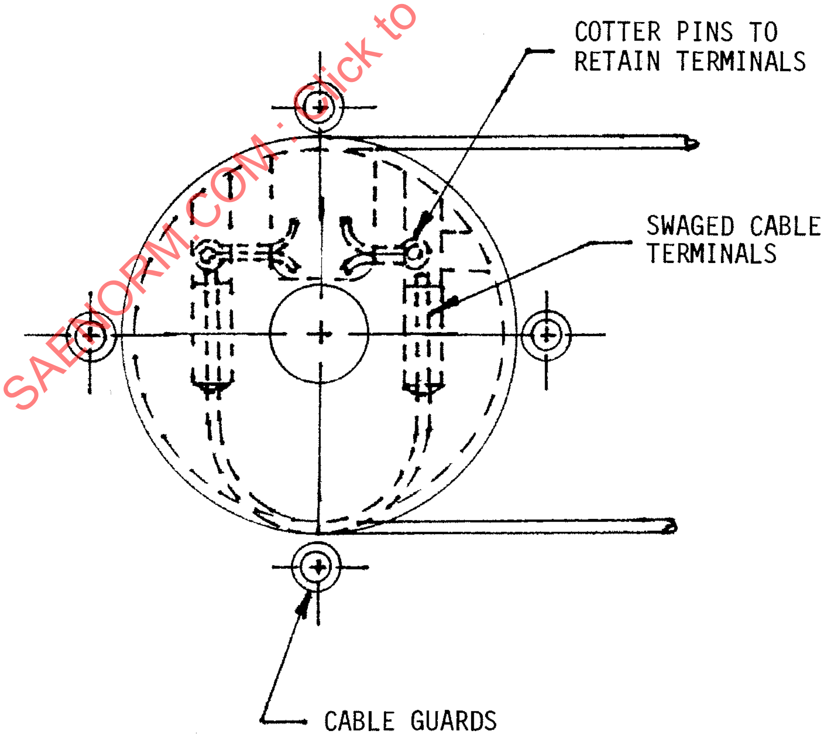


FIGURE 30 - TYPICAL DRUM CABLE TERMINATION AND GUARDS

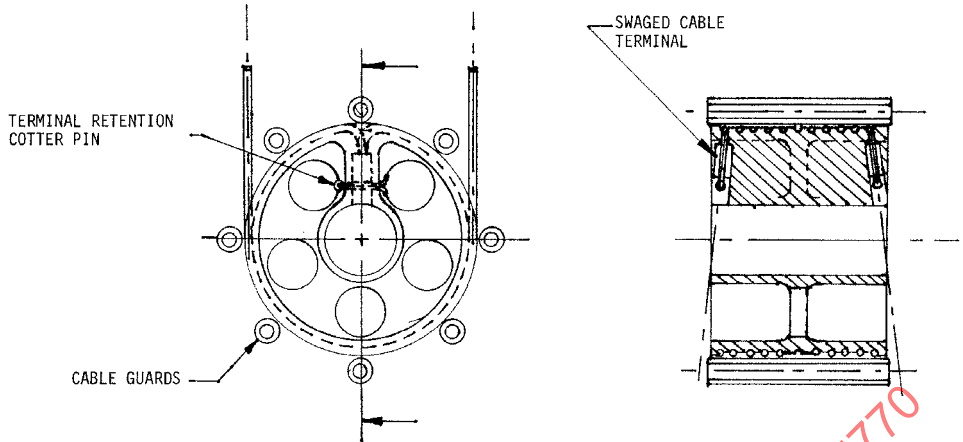


FIGURE 31 - LARGE DRUM CABLE TERMINATION AND GUARDS

c. Quadrants: Quadrants are used to terminate cable systems when the sheave travel is less than 270 degrees.

1. Quadrant Cable Groove Length: The length of the cable groove should be the no-load cable travel plus an overtravel allowance for aircraft tolerances and, on input quadrants, cable stretch at the system's maximum operational load. Overtravel is usually set at 10 degrees minimum wrap or 1.00 in of groove length, whichever is less.

2. Quadrant Cable Terminals: For quadrant cable terminals, use swaged cylinder terminals, swaged lugs, swaged threaded terminals, or swaged ball terminals. Usage of the swaged ball terminals is usually restricted to secondary control system applications because their swaging strength is only 60 to 80% of the cable breaking strength.

   Terminal retention methods and making provision for adequate bearing area are often products of a particular design requirement. Some of the standard terminations used are shown in Figures 32, 33, and 34. The quadrant terminals shown in Figures 35 and 36 are more robust and capable of high loads. The threaded cable terminals shown in Figure 36 allow some cable length adjustment that is not available with other types of cable terminals. These types of terminals are sometimes used on large cable drums.

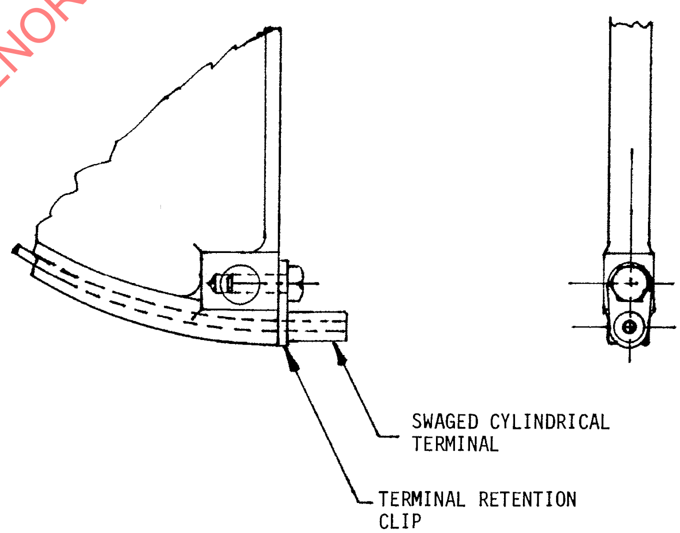


FIGURE 32 - PRIMARY FLIGHT CONTROL QUADRANT CABLE TERMINATION

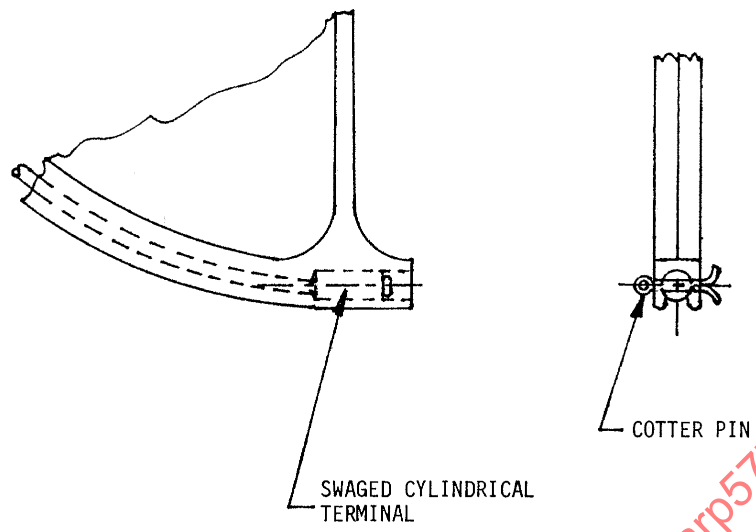


FIGURE 33 - PRIMARY FLIGHT CONTROL QUADRANT CABLE TERMINATION

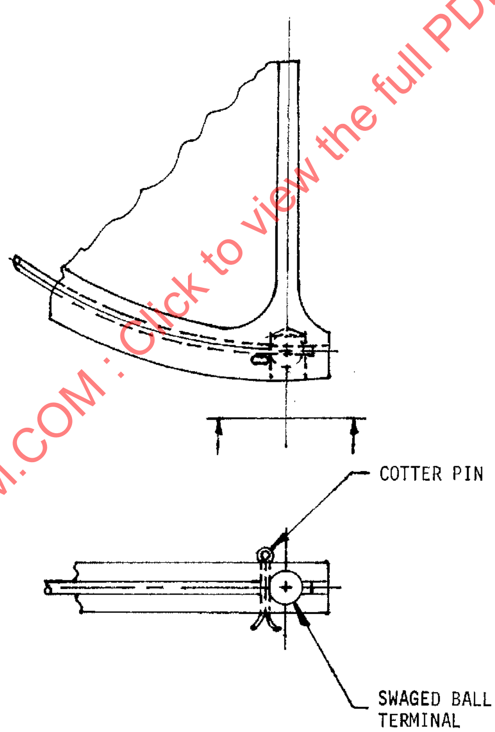


FIGURE 34 - SECONDARY CONTROL QUADRANT CABLE TERMINATION

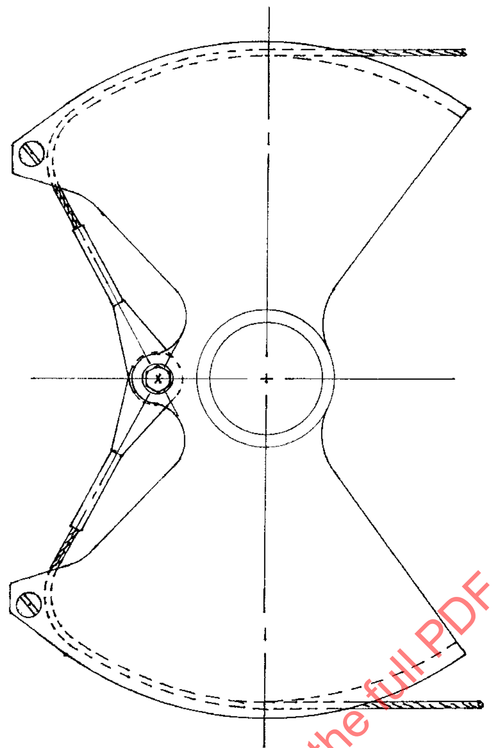


FIGURE 35 - PRIMARY FLIGHT CONTROL QUADRANT CABLE TERMINATION

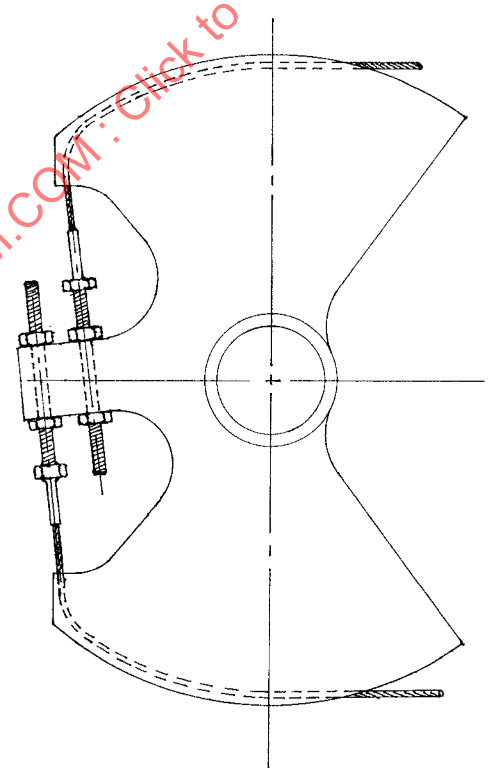


FIGURE 36 - PRIMARY FLIGHT CONTROL QUADRANT CABLE TERMINATION

5.4 Sheave Mounting Brackets

Sheave mounting brackets may be made of sheet metal construction, or machined from castings, forgings, or bar stock.

Sheet metal parts are usually the lightest and most economical for simple, small brackets, but machined parts may become cost effective for larger more complex brackets.

a. Sheave Bracket Load: All sheave brackets should be designed to accept a side load equal to 5% of the maximum load on the sheave.

   Dual load paths must be considered for any bracket which has pulleys for more than one control system. When dual load paths are required, they must extend from the pulley mounting bolt to the fasteners which attach the bracket to basic structure.

b. Sheet Metal Sheave Brackets: Sheet metal should be used for simple brackets for pulleys, and small quadrants or drums.

   Sheet metal sheave brackets:

   1. Should be designed for tension, rather than compression, loading whenever possible to reduce complexity, parts, and weight. A design with the load path passing through the base of the bracket is the preferred configuration.

   2. Should be channel-shaped rather than formed of two angles which require additional parts and fasteners to connect.

   3. Must space the sheave from the bracket sides with dimples, rather than washers, because of fewer parts and increased stiffness.

   4. Should have side braces added to react compression loads and side loads. Pulley brackets should be designed to withstand 3 degree cable misalignment.

   5. Brackets that might be stepped upon should be designed for 300 lb ultimate load.

   6. Brackets that might be used as a handhold should be designed for 150 lb ultimate load.

   7. When supplementary information (such as rigging instructions, cable identification, etc.) is required at a pulley bracket, it should be on a marker plate. Locate where it is conveniently visible.

   8. Lightening holes should be flanged and should be placed so that foreign objects cannot fall in through the hole(s) and become trapped in a cable-pulley intersection or between a crank and structure to jam the system or to restrict system motion.

   9. A hat section bracket should not be loaded sideways as shown in Figure 37. This loading imposes differential bending on the bracket. The preferred configuration imposes shear on the bracket.

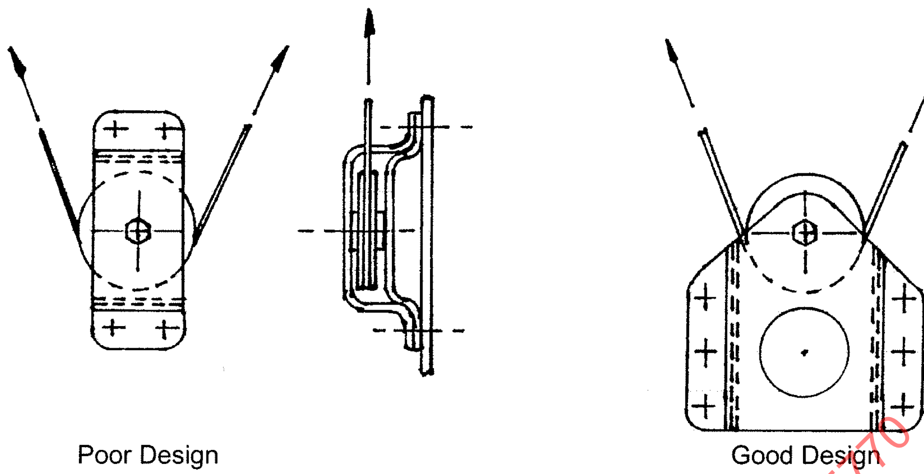


FIGURE 37 - HAT SECTION PULLEY BRACKET

10. Figure 38 provides an example of a well-designed bracket where the cable load passes through the bracket base, and a poor design where the cable load does not go through the base. If tension loads are high, attach bracket with bolts or screws. Also, a bracket can be mounted on a slope, as shown.

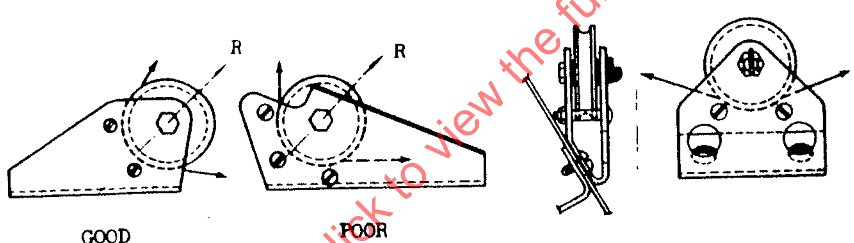


FIGURE 38 - BRACKET MOUNTED ON SLOPE

11. Pulleys can be set in the web of a bulkhead of beam, or between two webs. Pulleys of different sizes may be used on the same bolt when it is desirable to separate the cables. See Figure 39.

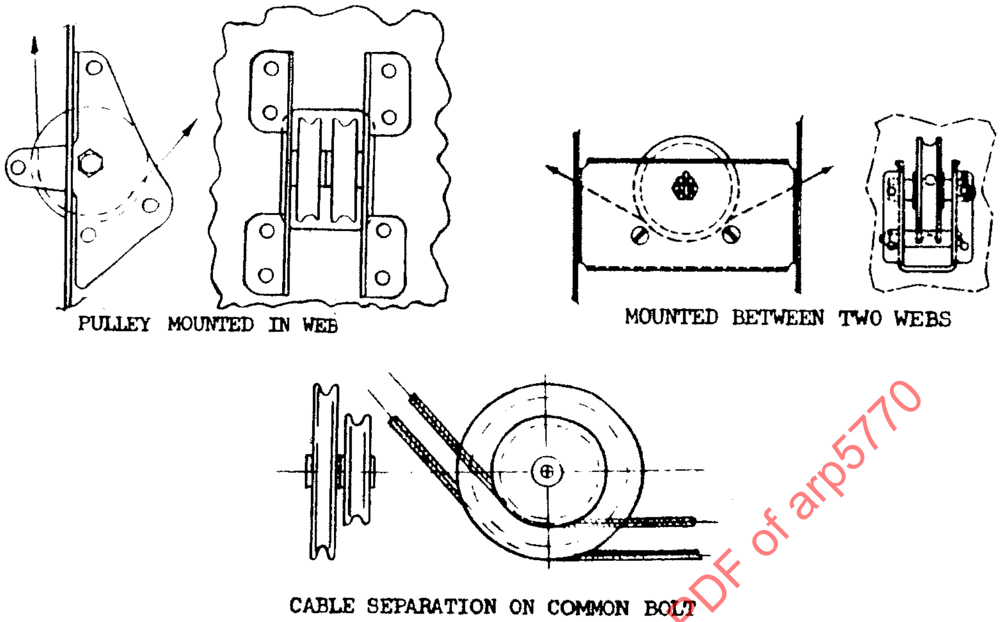


FIGURE 39 - PULLEY MOUNTING

5.5 Cable Guards

Cable guards are required for all sheaves (pulleys, quadrants, and drums) to prevent a slack cable from escaping the sheave cable groove.

All cable guards should be supported by the supporting bracket of the sheave which they guard to avert binding of the sheave on the guard due to relative deflections between the sheave and aircraft structure.

Guards should be installed within 15 degrees of the cable tangency points. When the cable wrap exceeds 25 degrees, one or more intermediate guards should be installed. Figure 40 can be used as a general guide (as to the number of guards required). Figures 39, 41, and 42 show the location of 1, 2, and 3 guards, respectively.

Angle of Cable Wrap (1)	No. of Guards
Up to 25°	1
26 to 90°	2
91 to 180°	3

(1) Values are approximate ranges.

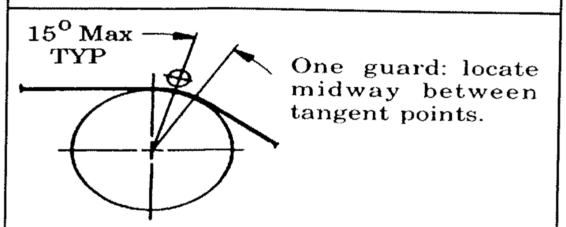


FIGURE 40 - SHEAVE GUARDS

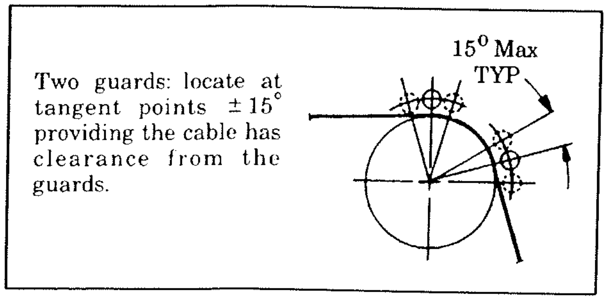


FIGURE 41 - TWO SHEAVE GUARDS

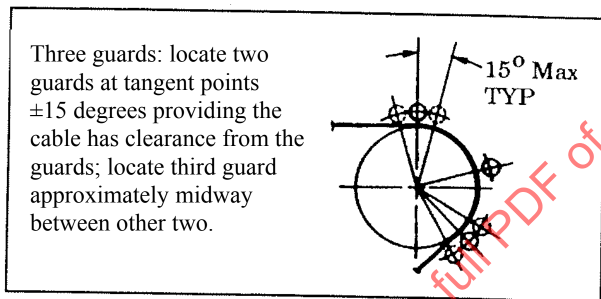


FIGURE 42 - THREE SHEAVE GUARDS

The nominal clearance between the guard and the sheave should be 0.032 in for 3/32 in cables and larger. Cumulative tolerances of all parts and holes should not decrease this clearance to less than 0.015 in, to prevent binding of the sheave by the guard, and not more than 1/2 of the cable diameter to prevent wedging of the cable between the guard and the sheave rim. Cables 1/16 in and smaller in diameter require smaller clearances and closer tolerances to prevent cable wedging or guard sheave binding.

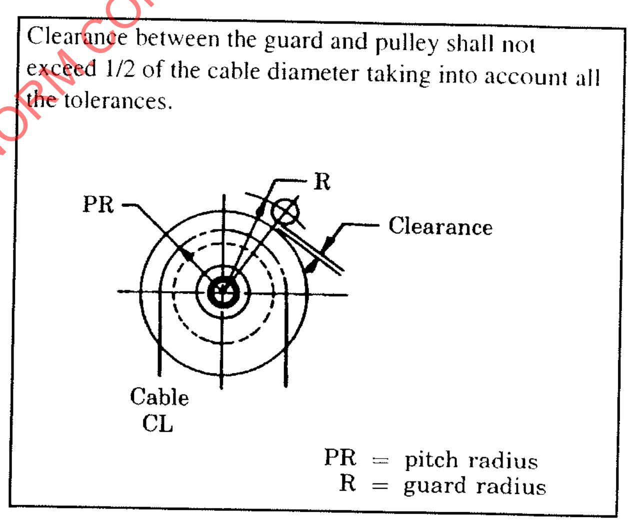


FIGURE 43 - PULLEY TO GUARD CLEARANCE

The type of guard selected will usually be determined by the ease with which the cable can be removed or replaced. In simple brackets less than 1 in wide, and where the sheave can be removed, fixed guards, such as rivets and rivet spacers, can be used. In complex bracket assemblies, where the sheave cannot be removed easily, removable guards, such as pins or bolts with rivet spacers, can be used. Avoid the use of cantilevered cable guards if possible.

a. Drum Cable Guards: Drum cable guards should be designed so that a cable cannot ride out of its groove. The diameter of the drum flanges shall be the nominal groove diameter plus two cable diameters. The design clearance of 0.032 +0.000/-010 should be provided between each guard and the drum flanges for 3/32 in cables and larger. There should be a minimum of one guard every 90 degrees of circumference around the drum. Typical cable guard installations are shown in Figures 30 and 31. Note that the guards are removable around 180 degrees of the drum to allow drum removal when replacing cables.

b. Quadrant Cable Guards: Quadrant cable guards can be either fastened to the quadrant mounting bracket, or a part of the quadrant itself. Quadrant guards should be removable to allow cable replacement.

   Quadrant mounted cable guards are usually the most effective type, and result in less weight for large diameter quadrants. Quadrant mounted cable guards require deep grooves with a pin or bolted rivet spacer installed through the groove flanges to prevent cable escape from the groove, as shown in Figure 44. The quadrant mounted guard must be located so the guard does not deflect the cable at either travel extreme.

   Stationary cable guards are used for small diameter quadrants or for quadrants where inadequate space exists for the deep grooves necessary for quadrant mounted guards. These guards use the criteria for pulley cable guards.

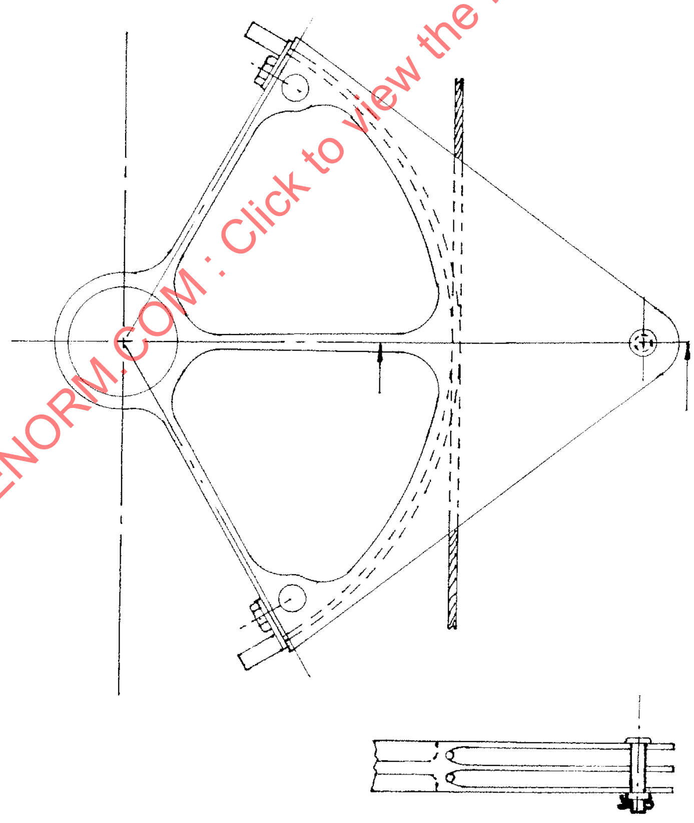


FIGURE 44 - QUADRANT MOUNTED GUARD

5.6 Fairleads and Rub Strips

Fairleads and rub strips are used to prevent control cables from contacting each other, structure and other equipment. Cables must not be bent or deflected by fairleads or rub strips, unless due to structural deflection. Allowable deflection should be a function of the cable rig tension.

The following guidelines are recommended:

Deflection ±3 degrees - 50 lb or 222 N max cable tension
Deflection ±2 degrees - 100 lb or 445 N max cable tension
Deflection ±1 degree - 150 lb or 667 N max cable tension
No deflection - Cable tension over 150 lb or 667 N

Fairleads and rub strips are made of non-abrasive material, such as nylon, nylatron (nylon impregnated with molybdenum disulphide), delrin, or laminated phenolic, to minimize abrasion of the cable when it contacts the fairlead and rub strips.

Grommets, which are standard fairleads, should be used whenever possible. When fairleads or rub strips are required for a specific location, they shall conform to the requirements of Figures 45 and 46.

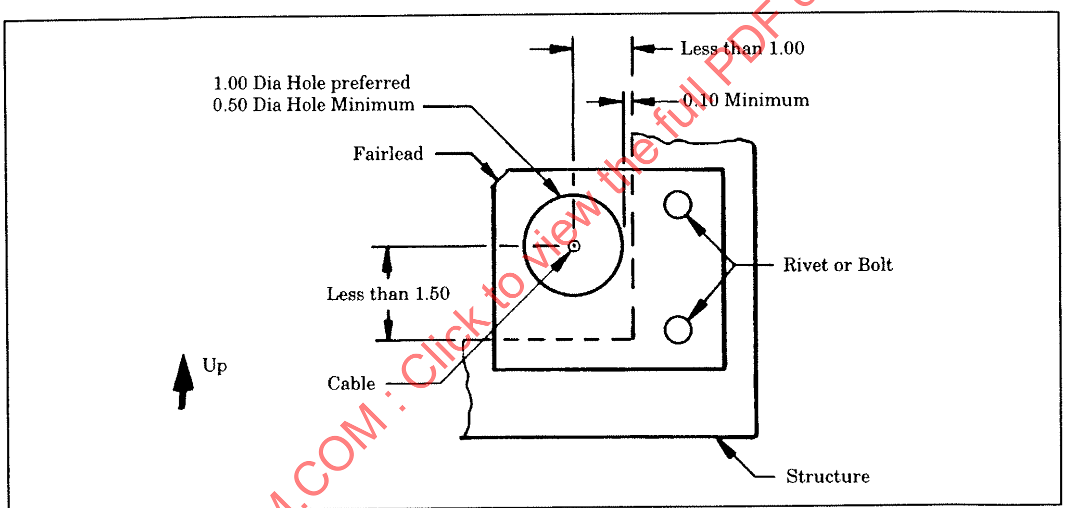


FIGURE 45 - TYPICAL FAIRLEAD INSTALLATION

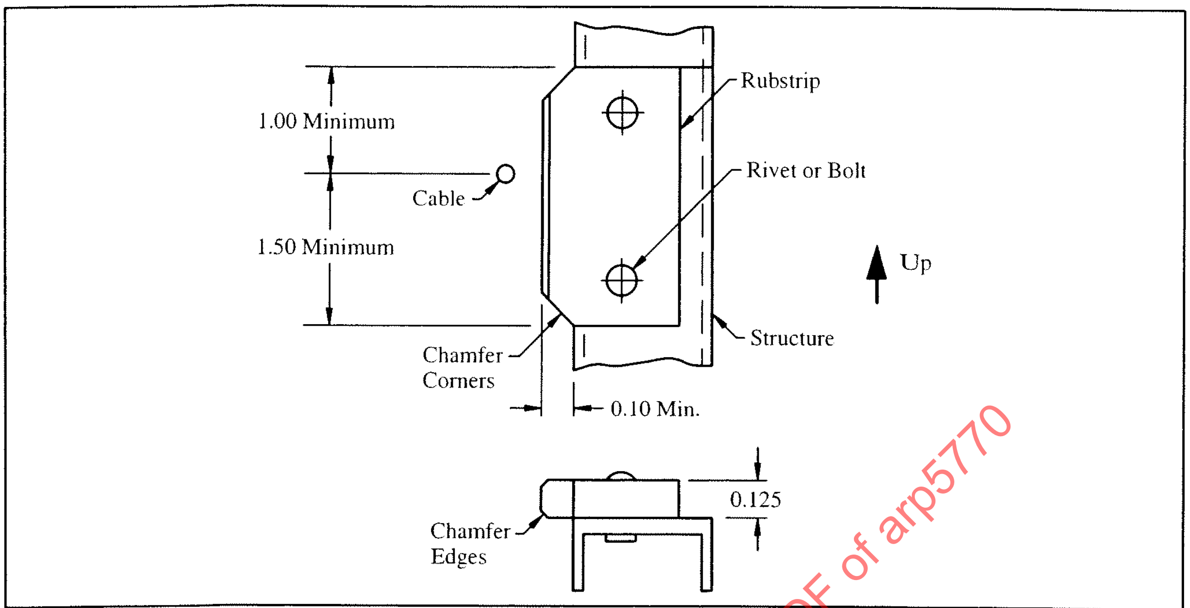


FIGURE 46 - TYPICAL RUB STRIP INSTALLATION

Fairleads and grommets should be split and removable if their hole diameter is smaller than 0.50 in, which is the minimum hole size through which a cable, with its terminal, can be removed or installed.

Cable runs should be examined in mockups and/or under slack cable conditions during proof load tests to determine if and where additional fairleads and rubstrips are required to prevent cable hangup or rubbing with structure or equipment.

a. NAS1368 Grommets: NAS1368 grommets should be used whenever possible to route cables through holes in structure.

   The NAS1368 grommet is a solid ring grommet that is usually installed with a tool before the cable is installed. Bonding is required only if the ring is split to allow its installation, and then the grommet shall be bonded.

b. Tapered Grommet: This grommet is used on straight cable runs in the wing to dampen cable radial movement due to vibration, and to deflect cables up to ±3 degrees due to structural deflections. The grommet bends the cable over a radius equivalent to 30 *D/d* ratio in an hour-glass like hole. Usage of these grommets results in good cable life, but with an increase in system friction.

   Figure 47 shows a typical installation in which grommets are installed in a phenolic mounting plate whose grommet and bolt mounting holes are located within 0.020 in of their true position.

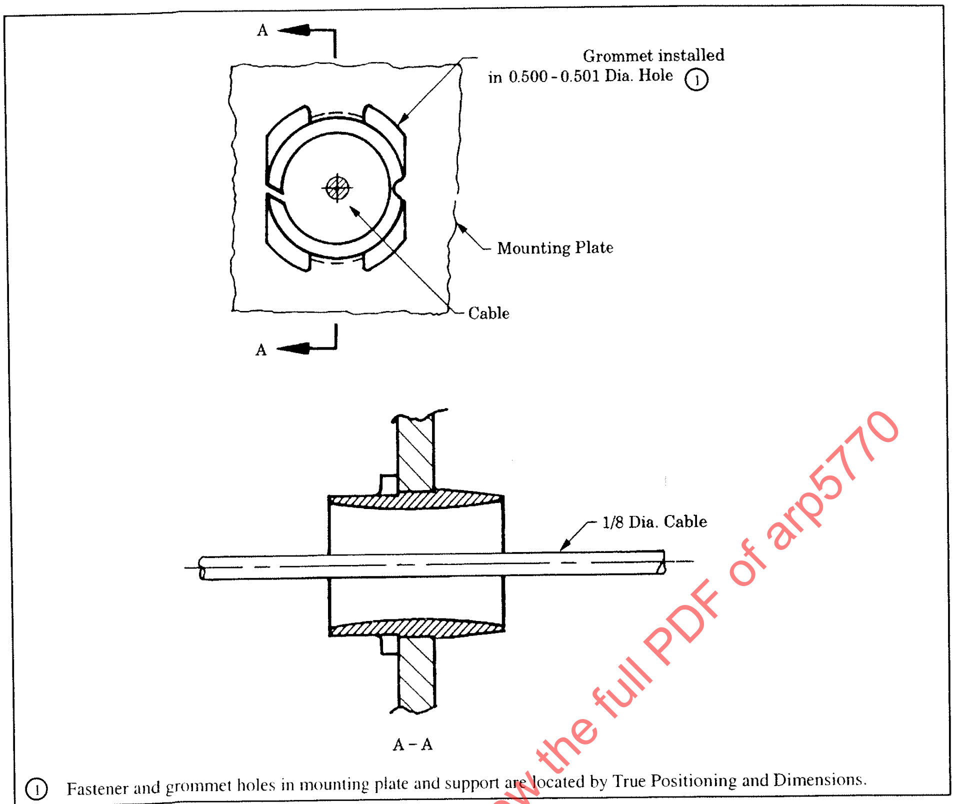


FIGURE 47 - TAPERED GROMMET INSTALLATION

Straight cable runs in the wing are supported by grommets approximately every 40 in.

c. Fairlead Pulleys: Fairlead pulleys, sometimes called idler pulleys, are not recommended on wing cable runs due to short cable life caused by the vibratory environment and large structural deflections.

Fairlead pulleys can be used in the body to support straight cable runs with zero cable wrap, where vibration and structural deflections are not significant. *D/d* ratios and conformance groove radii are not applicable to fairlead pulleys as they do not bend the cable.

5.7 Cable Seals

a. Pressure Seal: The delrin eyeball pressure seal of Figure 48 should be used for all pressure seal applications. This type of seal minimizes difficulties experienced with elastomer fin type seals, such as leakage, wear, friction, and susceptibility to mislocation and misalignment. The double plate retention provides alignment of the seal with the pierce point.

The seal should be left free until cable rigging is complete. When the seal is bolted into place, the cable should not be deflected by the seal.

Cable seals in areas subject to moisture build up from condensation, should not be mounted so that seal is in standing water if the drain holes are blocked. Use seal stand-off if necessary.

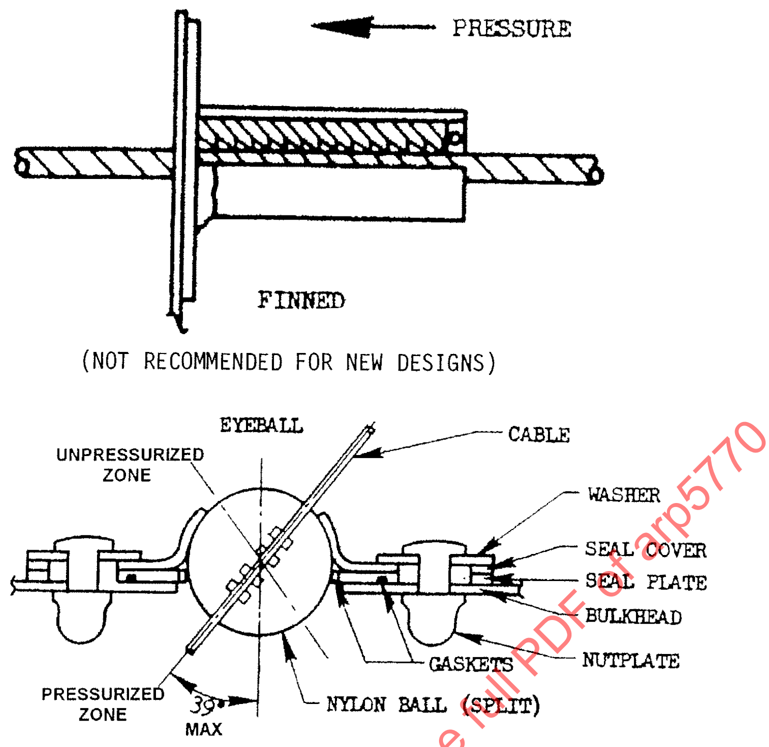


FIGURE 48 - EYEBALL AND FINNED CABLE SEALS

b. Cable Seals: Cable seals are used at firewall bulkheads at the engine, in the nacelle and at other locations to minimize the spread of fuel vapors or to transition from a pressurized to an unpressurized zone. Cable seals are usually made of laminated or molded plastic. The eyeball and finned seals are shown in Figure 48. The cable shall be lubricated in the seal zone.

Figure 49 shows a simple two piece seal which provides a hole whose diameter is 0.045 greater than the cable. Each piece is slotted for the cable so that the seal can be installed after the cable has been rigged. Mounting holes in the seal plates are oversize to allow the seal to be shifted to obtain best cable alignment.

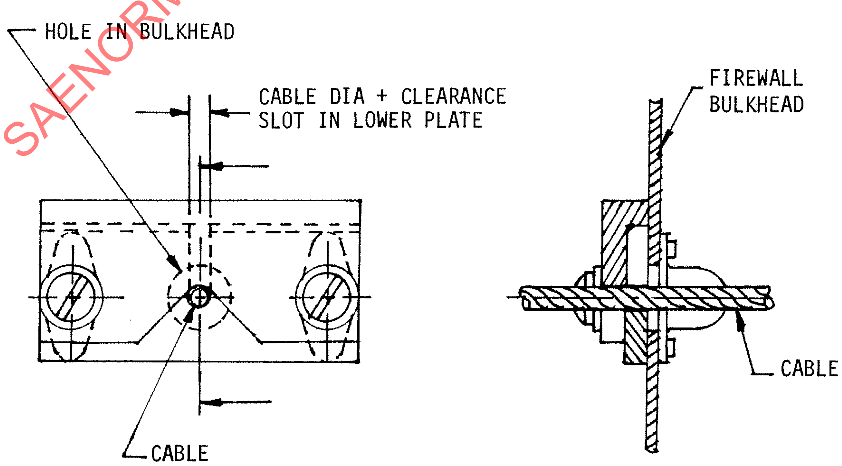


FIGURE 49 - SIMPLE CABLE FIREWALL SEAL

5.8 Control Pushrods and Bellcranks

Control cable systems are most efffective for long or straight control runs with control functions at the extreme ends. Pushrods and bellcranks are typically used for shorter control runs and/or where there are frequent changes in direction or intermediate attachment of control features.

Pushrods and bellcranks are typically fabricated from aluminum. Pushrods are typically fabricated from aluminum tubing that is swaged on both ends to interface with smaller diameter clevi and/or rod ends. The end fittings can be riveted in-place as depicted on Figures 50 and 51. Or the end fittings may be secured to the pushrod by electro-magnetic-formed (EMF) indentation of the pushrod into end fitting pockets. The EMF attachment process has been proven structurally sound on numerous military aircraft applications. The rod ends may be fixed or threaded depending on the control system rigging requirements. Bellcranks are typically machined from aluminum plate for coplanar designs. More complex bellcranks are fabricated from cast or riveted aluminum assemblies. There have been some high temperature applications where the pushrods and bellcranks were fabricated from steel or titanium. More recently, composite pushrods have been introduced into service. Composites offer weight and corrosion resistance advantages over metallic designs. The thermal expansion characteristics of the pushrod material should be compatible with the airframe structure.

In the past, dual load path pushrods, bellcranks and attachments were used extensively to provide primary control structural redundancy and continuity. These dual load path designs were expensive and difficult to inspect. In most cases the strength of the design was somewhat compromised by the duality features. In more recent applications, completely dualized parallel control systems have been implemented to provide structural redundancy and continuity. For new applications, completely separate redundant control systems are favored over dual load path designs because of inspection and verification issues with dual load paths. Dualized parallel control pushrods and bellcranks must be carefully configured and analyzed to minimize any thermal differentials or structural deflections that would result in one load path preloading the adjacent load path to unacceptable levels. Dualized parallel control designs can be preloaded against one another at low levels to eliminate backlash in the control system.

5.8.1 Control Pushrods

Control pushrods should be designed for a fatigue service life of 20 years or the aircraft life, whichever is greater. Typical service loads and lateral vibration should be considered. Additionally, control pushrods should meet the following criteria:

1. Single-Path, Non-Adjustable (or single half turn adjustment at one end) shall have fixed ends and adjustable ends integrally threaded as shown in Figures 50, 51, 54, and 55.

2. Single-Path, Vernier Adjustable Rods are of two types as follows:

    - Where only one end of the rod is easily accessible (such as a tab-rod with one end visible and the other end buried deep in the surface), rods shall be of the type shown and shall use a sleeve for adjustment similar to the one shown in Figure 52.

    - Where the rod is fairly short and both ends are readily visible and accessible, use the differential thread double-ended adjustment type shown in Figure 53. Turnbuckle type adjustment (i.e., combination of right and left hand threads) shall not be used in control rods because of the possibility of a disconnect or gross length change if the jam nuts loosen. Turnbuckle type adjustment can be used if each rod end utilizes a NAS513, NAS559, or NAS1193 type locking feature with safety wired jam nuts.

3. Figure 54 shows a typical composite pushrod. Composite pushrods with threaded rod ends or Vernier adjustment are accepted practice.

4. Any adjustable control pushrod that attaches directly to an airplane control surface should use steel threads in the adjustment feature. Use integral threads in a steel rod or provide a threaded steel fitting.

5. Access must be provided to allow easy adjustment and check of all adjustable control pushrods.

6. Corrosion protection of aluminum alloy control pushrods should include all the following steps:

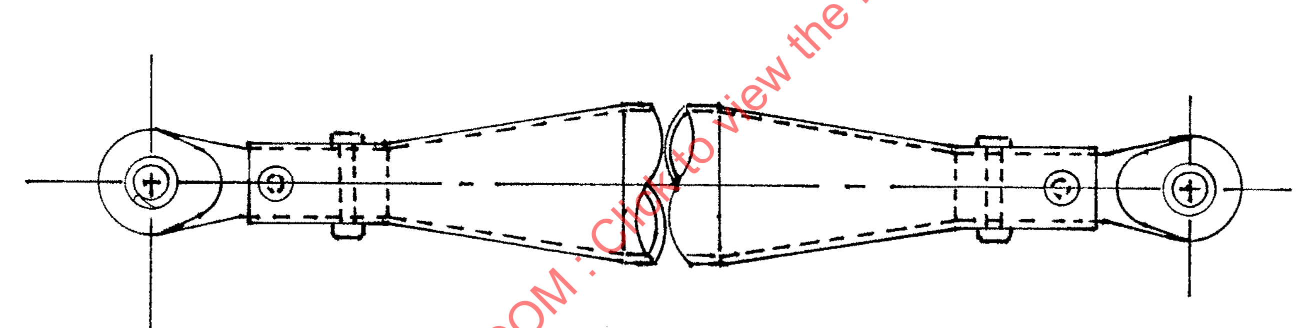
   - Alodine interior and exterior per MIL-DTL-5541.
   - Apply one coat of zinc chromate primer on interior and exterior.
   - Apply corrosion inhibiting compound on interior only.
   - Assemble control rod with wet zinc chromate primer applied to non-threaded rod ends and corrosion preventative compound MIL-PRF-16173 applied to internal and external surfaces of mating threads.

7. Excess misalignment of self-aligning rod ends on control pushrods should be prevented. Ensure that clevis width or some other restriction limits bearing misalignment to within its capability to avoid damage to bearing seals.

8. All control pushrods should be designed as stepped beam columns with initial mid-span deflection of $L/400$, where L is the rod length (inches) measured between end bearing centers.

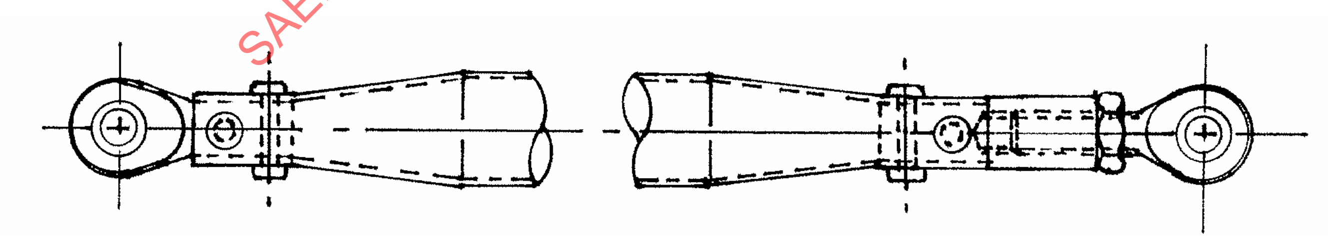
9. Where lubricated or PTFE spherical bearings or bushings are used in lieu of anti-friction bearings, the initial column eccentricity should be increased to include a structural deflection due to end moments of μx (rotating radius) x (compression load) where μ is the bearing coefficient of friction.

a. Control Pushrod Types: Six basic types of control pushrods used are shown in Figures 50 through 55.



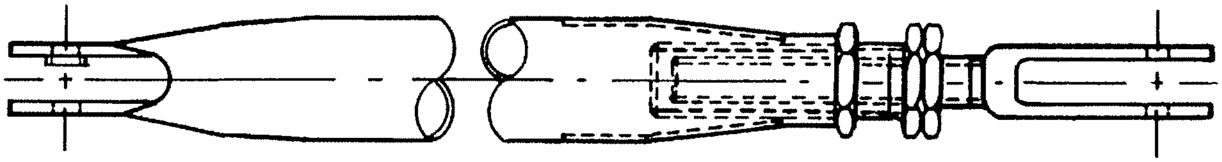
Fixed Non-Adjustable: Fabricated from swaged aluminum tubing. The end fittings are riveted to the tube and cannot be adjusted.

FIGURE 50 - SINGLE LOAD PATH NON-ADJUSTABLE PUSHROD



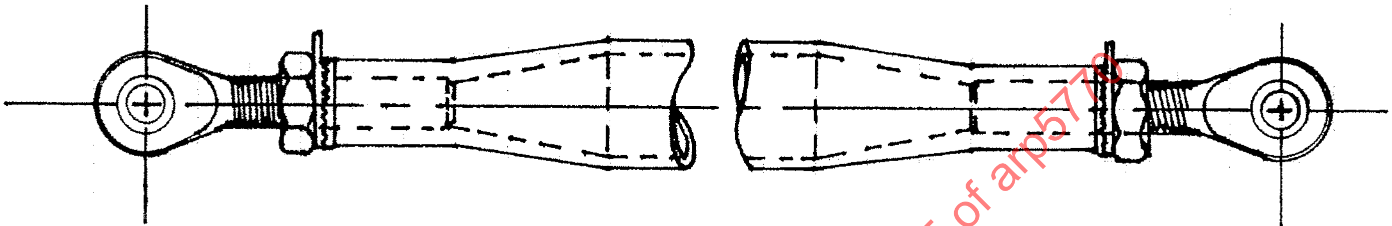
Half-Turn Adjustable: One end is screwed into a sleeve and secured with a locknut. When the nut is loosened, the rod end may be rotated in half-turn increments for length adjustment.

FIGURE 51 - SINGLE LOAD PATH PUSHROD WITH HALF-TURN ADJUSTABLE ROD



One End Vernier Adjustable: One rod end is screwed into a sleeve which is threaded into another sleeve. Both are secured with locknuts. Infinite adjustment is possible.

FIGURE 52 - SINGLE LOAD PATH VERNIER ADJUSTABLE PUSHROD



Both Ends Adjustable: Each rod end is threaded into the pushrod with a different RH thread pitch and secured with a locknut and NAS1193 locking device. This permits infinite adjustment.

FIGURE 53 - SINGLE LOAD PATH WITH BOTH ENDS ADJUSTABLE PUSHROD

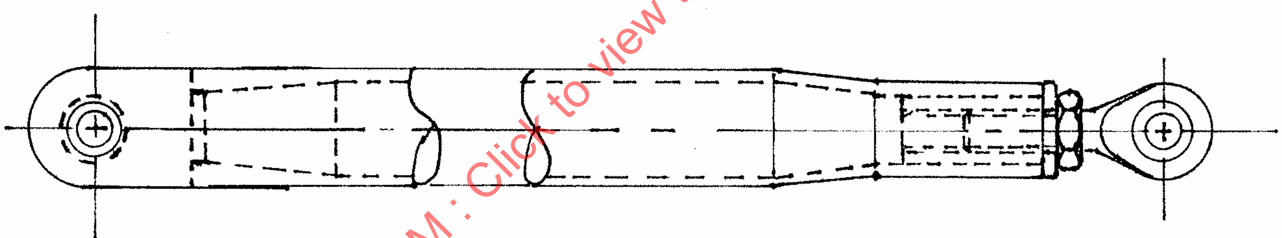
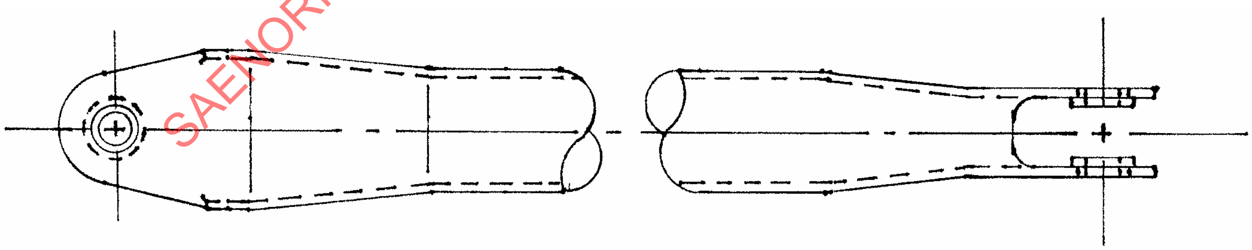


FIGURE 54 - COMPOSITE ADJUSTABLE PUSHROD



Fixed Non-Adjustable: Pushrod fabricated from tubing. Flanged bushings are press fitted and reamed at both ends.

FIGURE 55 - SINGLE LOAD PATH NON-ADJUSTABLE PUSHROD

b. Rod Ends: Threaded rod ends should use check nuts to eliminate end play and working of the threads. Check nuts should be torqued, and lockwired.

Rod end locking devices are required on Vernier adjustable rods. Typical rod end locking devices are shown in Figures 56 through 58.

1. Positive Index Washers, NAS1193, shall be used on applications where positive control in linear adjust is critical, such as Vernier adjustments or struts.

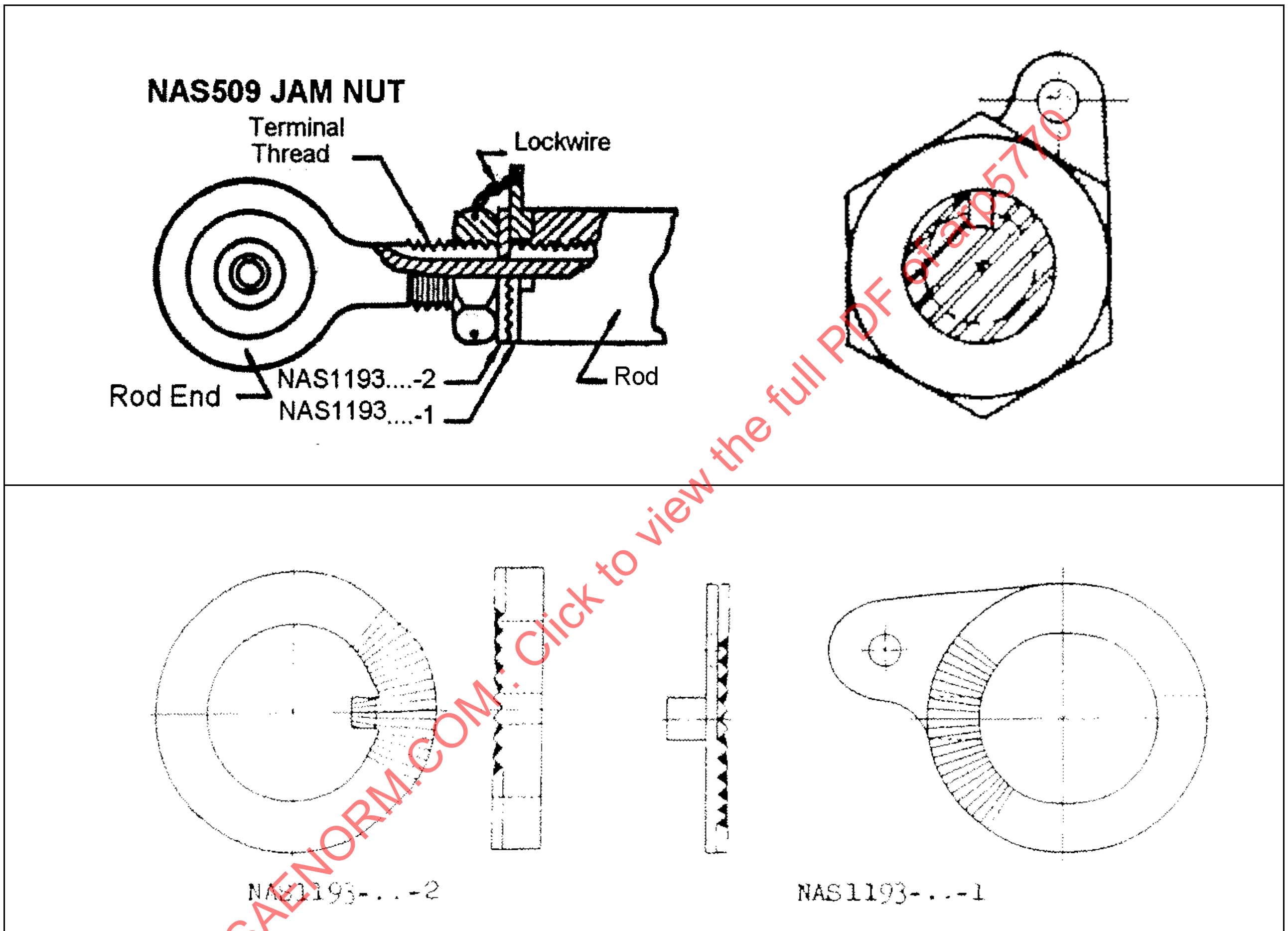


FIGURE 56 - TYPICAL INSTALLATION OF NAS1193 POSITIVE INDEX WASHER