SAE Aerospace An SAE International Group	**AEROSPACE INFORMATION REPORT**	**SAE AIR1739**	**REV. B**
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(R) Information on Antiskid Systems

RATIONALE

Significant changes in brake control have taken place since the last revision of this document.

INTRODUCTION

The summary of experience in antiskid system usage will be used by design personnel to avoid the pitfalls experienced by others in application of antiskid to retrofitted and new systems. The information will be presented as General System Description and Operation, Hardware Details and Functions, System Performance Evaluation, System Development Process, and Service Problems.

Need and Requirements:

While antiskid system, component, and feature problems and limitations are well known to the designer and specific user, there is no documentation available which describes the broad field of applications. This document is a compendium of field experience and can form the basis for establishing system requirements. The requirements should reflect the intended operating surfaces, the desired performance, required system efficiency and the method of determining this efficiency. Required system characteristics or features should be initially established along with performance requirements prior to contracting for a system.

Antiskid control systems originated in response to the inability of the pilot to maintain control of the brake system and to avoid inadvertent wheel lockup and possible tire failure. With most power brake systems, the "feel" of the overall aircraft system to the pilot is inadequate to maintain knowledge of the state of rotation of all the aircraft wheels. This is especially true for large multi-wheel aircraft. Therefore, some degree of assistance was needed to detect and react to deep skids and wheel lockups. This initial basic need resulted in the inclusion of an antiskid system. As antiskid systems evolved, their primary purpose shifted from lockup prevention and tire saving to optimal stopping performance. All modern brake control systems include highly efficient stopping performance as one of their primary goals.

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1. SCOPE

This SAE Aerospace Information Report (AIR) has been prepared by a panel of the SAE A-5A Committee and is presented to document the design approaches and service experience from various applications of antiskid systems. This experience includes commercial and military applications.

1.1 Purpose

The purpose of this document is to describe antiskid system configurations, features, modes of operation and to define various methods used to calculate antiskid system performance.

2. 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.

2.1 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

AS483B Skid Control Equipment

AIR764C Skid Control System Vibration Survey

ARP862B Skid Control Performance

AIR1064D Braking System Dynamics

ARP1070C Design and Testing of Antiskid Brake Control Systems for Total Aircraft Compatibility

AIR5372 Information on Brake-by-Wire (BBW) Brake Control Systems

2.2 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/

14 CFR 25.735e Antiskid Systems

14 CFR 25.109 Accelerate-Stop Distance

14 CFR 25.1301 Function and Installation

14 CFR 25.1309 Equipment, Systems and Installations

AC 25-7B Flight Test Guide for Certification of Transport Category Airplanes

AC 25.735-1 Brakes and Braking Systems Certification Tests and Analysis

MIL-W-5013L Military Specification Wheel and Brake Assemblies, Aircraft General Specification for

MIL-PRF-5041K Performance Specification, Tires, Ribbed Tread, Pneumatic, Aircraft, General Specification for

MIL-B-8075D Brake Control Systems, Antiskid, Aircraft Wheels, General Specification for

2.3 Other Documents

AFFDL-TR 74-118 “Test and Performance Criteria for Airplane Antiskid Systems," October 1974

Boeing Report D6-41115, "Research Study on Antiskid Braking Systems for the Space Shuttle," NASA Contract 8-27864

ASD-TR-74-41, FAA-RD-74-211, Vols. I and II, "Combat Traction II, Phase II," October 1974

ASD-TR-77-6, Vols. I and II, "An Extended Prediction Model for Airplane Braking Distance and a Specification for a Total Braking Prediction System," March 1977

NASA TP-1051, “Behavior of Aircraft Antiskid Braking Systems on Dry and Wet Runway Surfaces - A Slip-Velocity-Controlled, Pressure-Bias-Modulated System”, December 1979

NLR-TP-2001-242, “Hydroplaning of Modern Aircraft Tires,” Nationaal Lucht- en Ruimtervarrtlaboratorium NLR, November 1999

3. GENERAL SYSTEM DESCRIPTION AND OPERATION

3.1 General System Configuration

Antiskid Control System is defined as a group of interconnected components which interact to prevent inadvertent tire skidding and contribute to shorter aircraft stopping distances by controlling excessive clamping force on the brakes. The basic system normally consists of a wheel speed transducer, a control circuit and a brake pressure control valve or electric motor actuation controller as shown in Figure 1. The antiskid system is a sub-system of the braking system which also includes the pilot control, parking brake functions, and the wheel/tire/brake assemblies. The braking system may also include other features such as automatic and gear retract braking. In some cases, the pilot’s control function uses the antiskid system hardware, as is the case in many brake-by-wire systems. All are subsystems of the gear, and they are all interrelated.

3.2 System Operation

The basic purpose of the antiskid system is to modulate pilot commanded clamping force on the brakes, modulating it to levels compatible with optimum aircraft deceleration, while preventing excessive wheel skidding. In hydraulic systems this clamping force is produced by hydraulically driven pistons. In electric brake systems this force is produced by electrically driven actuators. Typical operation is as follows. The pilot applies brakes by applying force to the brake pedals resulting in an increase in clamping force in the brakes. As the clamping force increases, the wheel begins to slow down and the force between the tire and the runway increases. When the available friction force between the tire and the runway is exceeded, the wheel begins to decelerate rapidly. This is sensed by the antiskid system which in response outputs a signal to reduce clamping force. With release of clamping force, the wheel accelerates to a speed somewhat less than synchronous speed while maintaining as much braking force as possible, the release signal is removed and clamping force is reapplied. This sequence then repeats as needed.

3.3 Antiskid Control Classification

Many different antiskid systems are in use today. They represent a broad spectrum of an evolving technology from the late 1940's and different approaches to the solution of a complex problem. The very early systems utilize on-off control concepts, while later systems provide different degrees of brake pressure modulation in response to wheel speed changes.

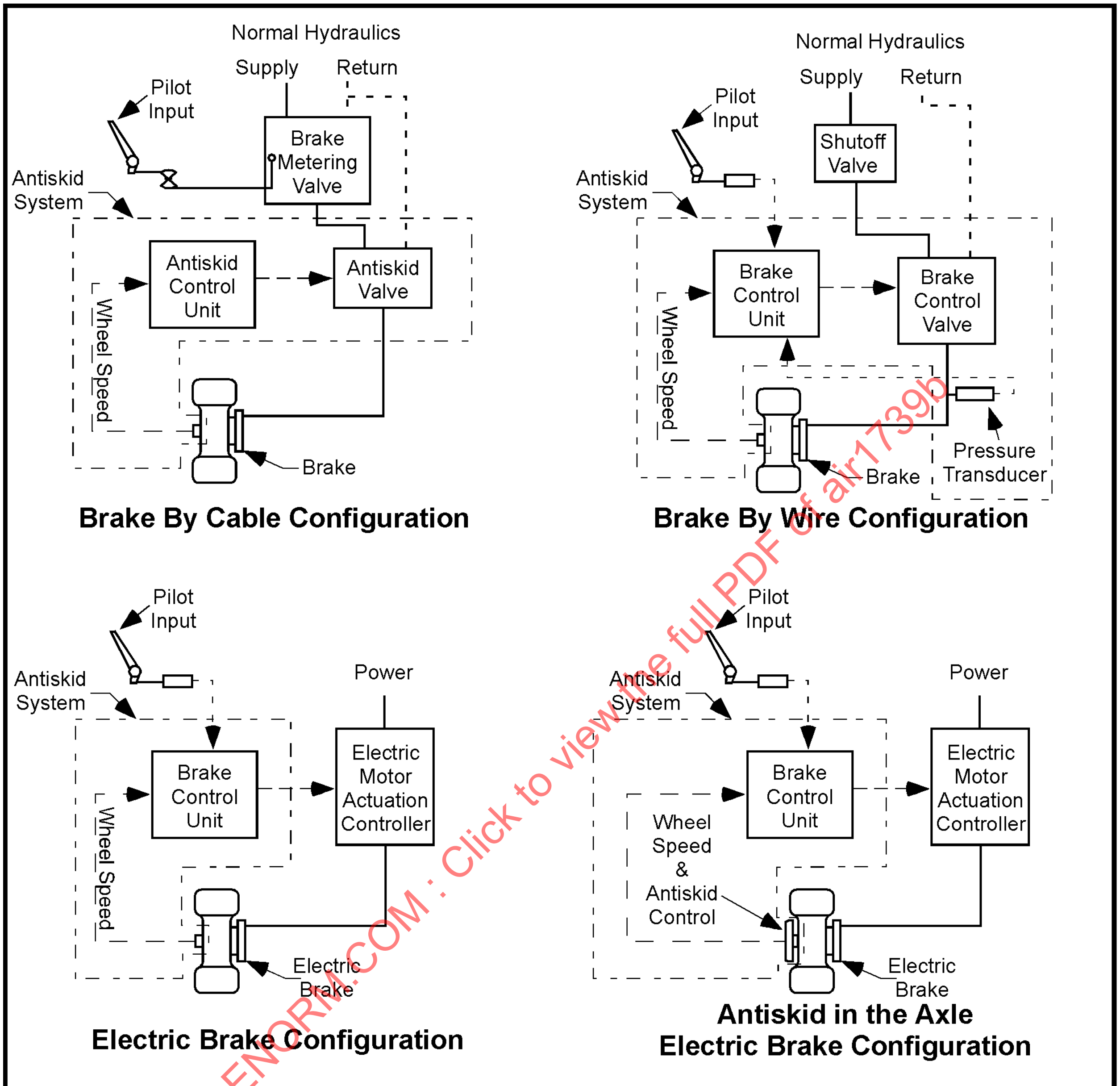


FIGURE 1 - ANTISKID SYSTEM SCHEMATIC

3.3.1 ON-OFF Systems

A typical on-off system is in use on the B-52 and consists of three components: a skid and locked wheel detector, a control shield, and a solenoid valve. The detector is an electro-mechanical device, providing logic signals to the control shield. The control shield is a power conditioner containing a series of relays. The relays interpret logic signals from the detector and apply an electrical signal to the hydraulic solenoid valve, to dump or reapply metered brake pressure.

The detector is the heart of this antiskid system. Its function in the system is to detect wheel decelerations above a preset value. As implemented in the B-52, the detector is mounted in the axle and is driven by wheel hub cap rotation. The skid sensing portion of the detector consists of an inertia flywheel and an overload-release clutch. The flywheel's inertia causes the spring loaded clutch to release when wheel deceleration exceeds a predetermined rate. Because the wheel is slowing down faster than the released inertia, a relative displacement occurs between the inertia and the drive, attached to the wheel, causing a set of electrical points to contact, thereby completing the skid circuit to the control shield. The shield in turn provides an electrical signal to the antiskid solenoid valve, resulting in a release of brake pressure. Feedback to the pilot to indicate that the wheel is skidding is provided either through pedal vibration or a cockpit light.

As the wheel accelerates back to synchronous speed, the contacts in the detector open, removing the signal from the antiskid solenoid valve. Thus, the pilot's metered pressure is again applied to the brake, allowing the skid cycle to repeat. In this type of system, the pilot must adjust the applied pressure level to minimize cycling in order to achieve good performance. Figure 2 shows an example of typical control using an ON-OFF system on a dry and a wet runway.

3.3.2 Modulating (Quasi-Modulating) Systems

Modulating antiskid systems generally rely on a wheel speed transducer which generates a signal proportional to wheel speed. Based on the input wheel speed information, the electronic control circuit determines if the wheel is going into a skid. A signal is supplied by the control circuit to an electro-hydraulic valve (the antiskid valve) which regulates brake pressure in a manner inversely proportional to control signal, releasing pressure when a skid is detected and reapplying it when the wheel recovers. But, unlike the ON-OFF system, the pressure is reapplied to a lower level and then ramped up until the wheel starts to go into a skid again. This eliminates the need for interaction by the pilot.

The first generation modulating systems generally act to release brake pressure when the computed wheel deceleration exceeds a preset value indicating an incipient skid. Brake pressure is held off for a time duration depending on the depth of the skid. As the wheel recovers from the skid, brake pressure is first stepped up to a low value and then gradually increased until a new incipient skid is sensed. During the wheel skid essentially all corrective action is based on a pre-programmed sequence, rather than the wheel speed/time history. These modulating systems can be tuned to provide very good dry runway performance; however, on slippery runways they are only able to extract a fraction of the maximum available friction because too much time is consumed in opening and closing the brake stack. The control dynamics of the modulating systems are not optimized for maximum performance on slippery runways. Figure 3 shows an example of typical control for a Modulating System on a dry and a wet surface.

Most modulating systems rely on an electronic circuit for intelligence. However, one system is in use which consists of only two hydro-mechanical components. A hydraulic modulator, upstream of the brake, controls initial pressurization volume and pressure application rate. The sensor, downstream of the brake, is located in the wheel axle and controls pressure release rate when an incipient skid is sensed.

3.3.3 Adaptive Systems (Fully Modulating)

Adaptive antiskid systems represent advanced control concepts which give optimized performance for both dry and wet runways. Usually high frequency wheel speed transducers are used which result in improved signal fidelity and response. Multiple data control functions, feedback of valve current, and nonlinear computing elements such as multipliers and dividers are combined to result in the adaptive control system. During the skid, corrective action on brake pressure is based on the sensed wheel speed signal. The major difference between the modulating and adaptive systems is found in the implementation of control in the electronic circuitry.

Adaptive systems, typically, compare braked wheel speed to a reference wheel speed. In almost all systems the reference wheel speed is derived electronically from the braked wheel during spin-up and brake release. However, one system is in use which measures the reference wheel speed with a separate and additional sensor in the nose wheel. In some systems, the control circuit cycles the braked wheel about a fixed slip ratio. Slip (or slip ratio) is defined as one minus the wheel rotational speed divided by the wheel rotational speed of a free rolling wheel. At least one system observes wheelspeed characteristics to seek out and modulate brake clamping force about the level that will just result in a skid. This is the level at which optimal braking occurs. Figure 4 shows an example of typical control for an Adaptive system on a dry and a wet runway.

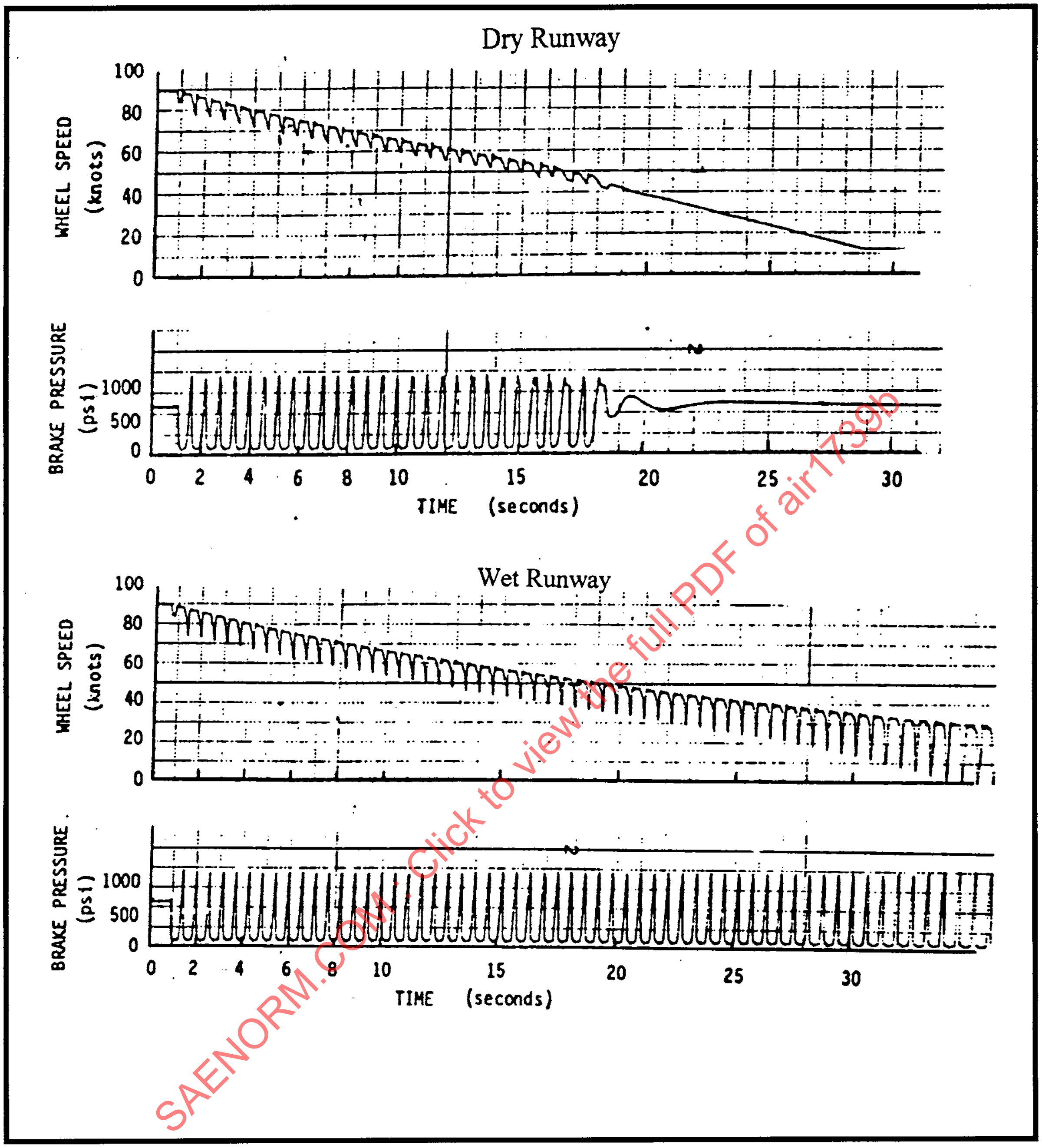


FIGURE 2 - ON-OFF SYSTEM OPERATION

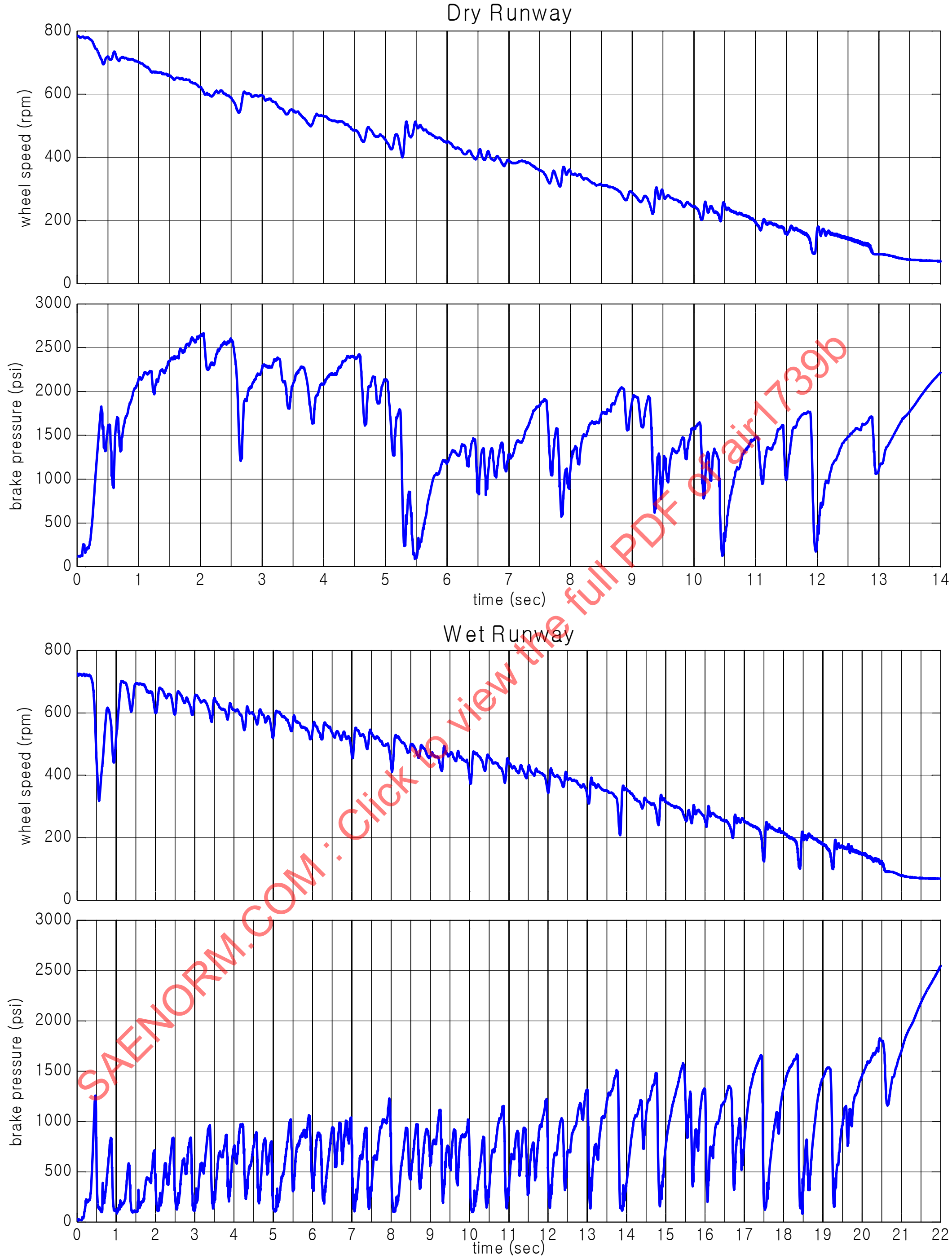


FIGURE 3 - MODULATING SYSTEM OPERATION

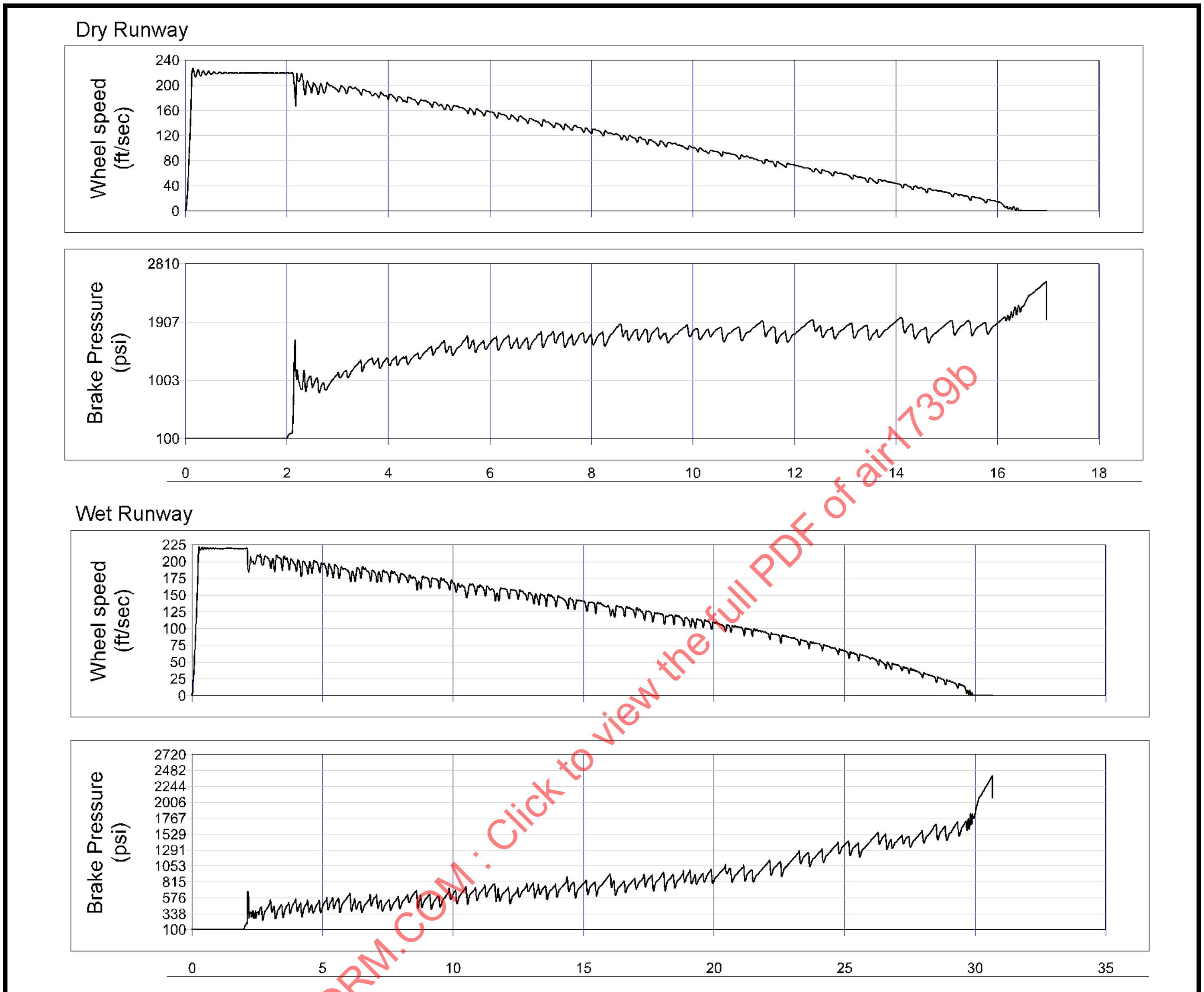


FIGURE 4 - ADAPTIVE SYSTEM OPERATION

3.4 Control Features

Experience has shown that several ancillary functions contribute to making the system work under all circumstances. These include wheel control grouping, touchdown protection, locked wheel protection, hydroplaning protection, and fault detection.

3.4.1 Individual Wheel Control

Individual wheel control is a system configuration in which each braked wheel is controlled by a separate valve and control circuit. Individual wheel control is the predominant control configuration in use because of its high efficiency.

3.4.2 Paired Wheel Control

In a paired wheel control configuration, two or more brakes are controlled together utilizing one control signal. This may be accomplished in two or more ways: the brake controller can produce one signal controlling one valve supplying pressure to two brakes or the brake controller can produce one signal controlling two valves with each valve supplying pressure to a separate brake, This configuration, while not as efficient as individual wheel control, has been used to assist in maintaining directional control where individual control can produce yaw moments which are difficult to control by the pilot. Paired wheel control has been used for the backup system on large aircraft with multiple braked wheels on each gear to reduce the amount of hardware required with only a small loss in performance. Differential control, by reducing pilot commanded pressure to the side opposite the direction of the intended turn, is possible in paired wheel control as long as there are separate valves for the left and right wheels of the airplane.

3.4.3 Locked Wheel Protection

Generally all antiskid systems for modern jet aircraft incorporate a locked-wheel protection feature. The wheel speed signals, on two (or more) wheels, are used to produce an “airplane moving” reference. In the event one of the wheels should lock up, a comparison of that wheel speed with the reference signal provides a basis for the release of the brake pressure of the slow wheel. A reference memory may be provided to maintain the reference should the two (or more) wheels lock up at the same time. The combination/location of wheels grouped (or paired) for locked wheel protection should be selected on the basis of airplane configuration, antiskid system configuration and failure modes. Typical pairing used include: inboard and outboard wheels across the airplane, adjacent wheels on a landing gear, and fore/aft wheel pairs on a truck (4-wheel bogie). Locked wheel protection can protect against hyroplaning (or aquaplaning) where loss of tire footprint contact with the runway may result in friction characteristics that fool the brake control into slowly controlling the wheel into a deep, prolonged skid.

3.4.4 Touchdown Protection

The purpose of touchdown protection is to make sure that braked wheels are free to rotate at touchdown even though the pilot had inadvertently applied brakes. The protection is normally implemented using the wheel speed signal and a signal from the airplane’s AIR/GROUND logic system. If the AIR/GROUND signal indicates that the airplane is in the air and the wheel speed is lower than a preset level, a full dump signal is applied to the antiskid valve. A transition from air to ground indication will result in removal of the dump signal, normally after a delay to allow wheel spin-up. Spin-up of the wheel to a level higher than the preset level will also remove the dump signal even with an airplane in the air indication.

3.4.5 Hydroplaning Protection

Hydroplaning Protection provides an extended release of clamping force to a braked wheel which fails to spin up due to hydroplaning at high speed on a flooded runway. Hydroplaning protection may be implemented by the use of an airplane ground speed reference which is external to the antiskid system, such as an Inertial Reference/Navigation System, or via an internal antiskid system reference signal. Brake release is based on the wheel speed being less than a set percentage of the ground speed reference. The hydroplaning protection would also provide the touchdown protection feature. Note that the locked wheel protection feature provides some amount of hydroplaning protection without the need for an airplane ground speed reference, as long as at least one wheel in the locked wheel groupings does spin up.

3.4.6 Fault Detection

Antiskid systems normally contain fault detection capability to provide indication to the pilot that the system is not fully operational, so that the pilot can use appropriate stopping techniques and allow for timely maintenance actions. At the simplest, the fault detection may do nothing more than check continuity to valves and wheel speed transducers and the availability of power. More sophisticated systems run end to end checks to ensure the health of the system and to isolate problems to the line replaceable unit (LRU) level or lower. A significant issue with fault detection systems is that they can result in numerous no fault found removals of brake system components.

A significant improvement to fault detection systems in microprocessor based systems is the ability to save application and landing parameters to memory to be used in the determination of system degradation and precise troubleshooting decisions. Stored parameters may include: wheel speed, brake pressure, pilot pedal inputs, and valve currents or clamping force commands.

4. HARDWARE DETAILS AND FUNCTIONS

A discussion and description of the hardware components more frequently encountered in antiskid systems is included as an aid in understanding overall system operation. For the sake of brevity, only the more commonly used components will be discussed.

4.1 Wheel Speed Transducer

The purpose of the wheel speed transducer is to provide an electric signal equivalent to the angular speed of the braked wheel. The transducer is mounted in the wheel axle and must be rugged in design to withstand high linear and torsional accelerations as well as high temperatures and high vibration levels. Two basic transducers are in use, the AC Generator and the DC Generator. The use of an AC sensor yields better signal fidelity and improved noise immunity than the DC generator as long as a sufficient number of teeth are used on the rotor. Fiber optic based wheel speed transducers have been developed and lab tested but have not entered service. A non-contact Hall effects based wheel speed transducer has recently completed certification.

4.1.1 DC Generator

The DC Generator produces a voltage proportional to wheel speed. The device is completely self-contained and is mounted in the axle. The armature of the generator is attached to the wheel and rotates with it. The basic parts of the transducer as shown in Figure 5 are a permanent magnet, armature, commutator, and brush assembly. Rotation of the commutator in the magnetic field induces an electrical signal proportional to wheel speed.

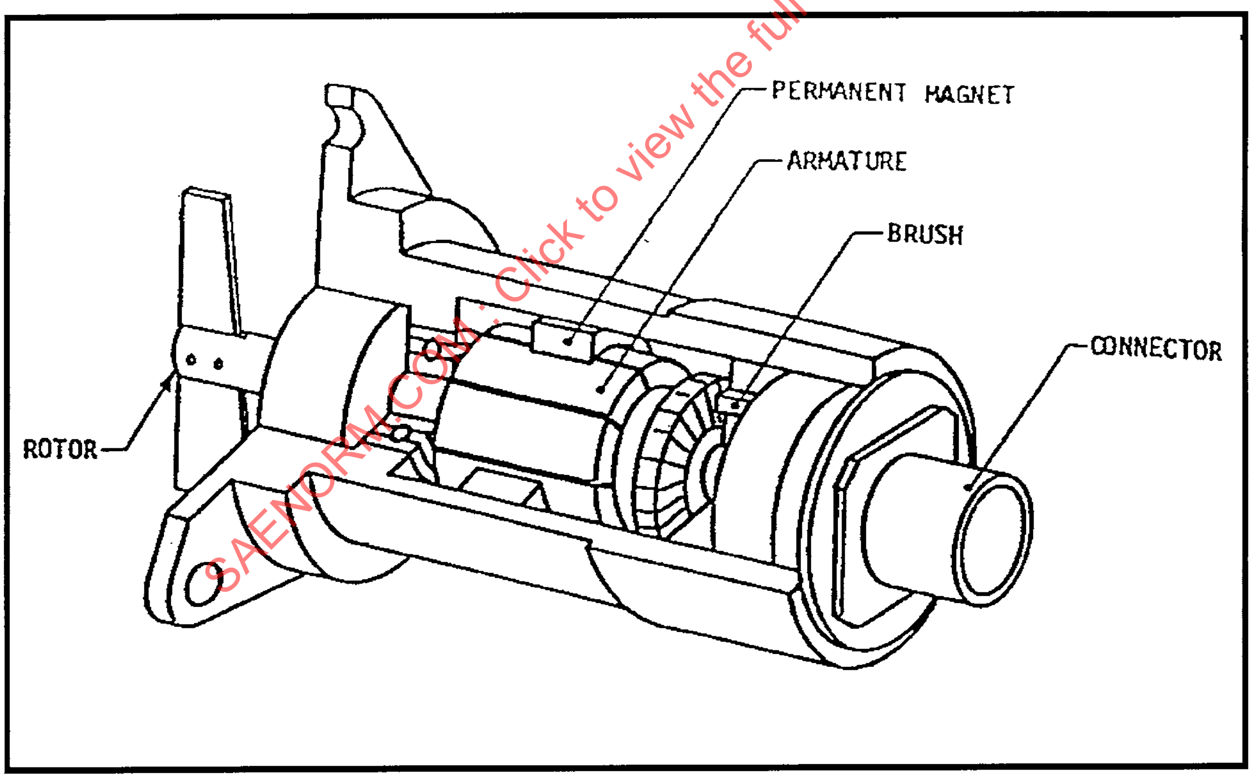


FIGURE 5 - DC WHEEL SPEED TRANSDUCER

4.1.2 AC Generator

The AC Generator is similar in appearance to the DC Generator but contains no brushes. The AC Generator produces an output voltage signal with frequency proportional to rotational speed and number of rotor teeth. A magnetic field is generated either with a DC current or a permanent magnet. As the rotor turns, the alternating alignment and misalignment of the teeth in the rotor and the stator vary the reluctance in the magnetic current. This results in an alternating current with frequency proportional to wheel speed. A typical AC wheel speed transducer is shown in Figure 6.

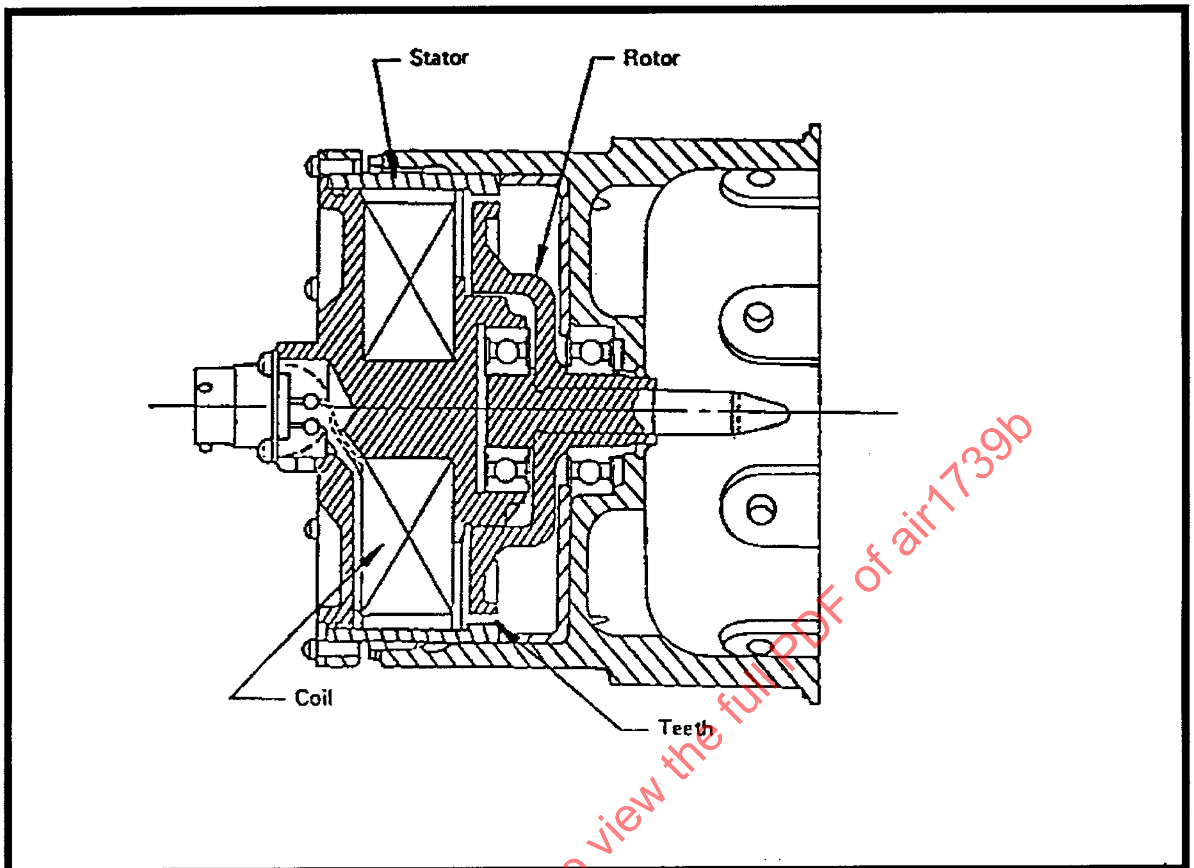


FIGURE 6 - AC WHEEL SPEED TRANSDUCER

4.1.3 Fiber-Optic Wheel Speed Transducers

Fiber-optic wheel speed transducers use light signals rather than electrical signals. A light source is normally located in the control unit and the light is transmitted through the transducer and back to a detector in the control unit via fiber optic cable. Within the transducer, the light is alternately transmitted and interrupted as the wheel rotates producing a detected signal with a frequency proportional to wheel speed.

4.1.4 Wheel Speed Transducer Couplings

The function of the transducer coupling is to connect the rotor of the wheel speed transducer to the rotating wheel, normally the wheel hubcap. In doing so, it must compensate for any misalignment between the axle centerline and the rotational axis of the wheel. There are a variety of methods for doing so. The normal method used is to have a drive arm mounted on the transducer shaft that fits in a radial slot in the part mounted in the hub cap. Figure 7 shows a spider/dog-bone coupling using this method. A shortcoming of this method is that if there is misalignment between the two axis, a one cycle per revolution sine wave is imposed on the wheel speed due to the varying arm length.

Another coupling currently in use is a bellows coupling, Figure 8. This coupling will accommodate large offsets, both angular and radial, with minimal effect on the signal quality.

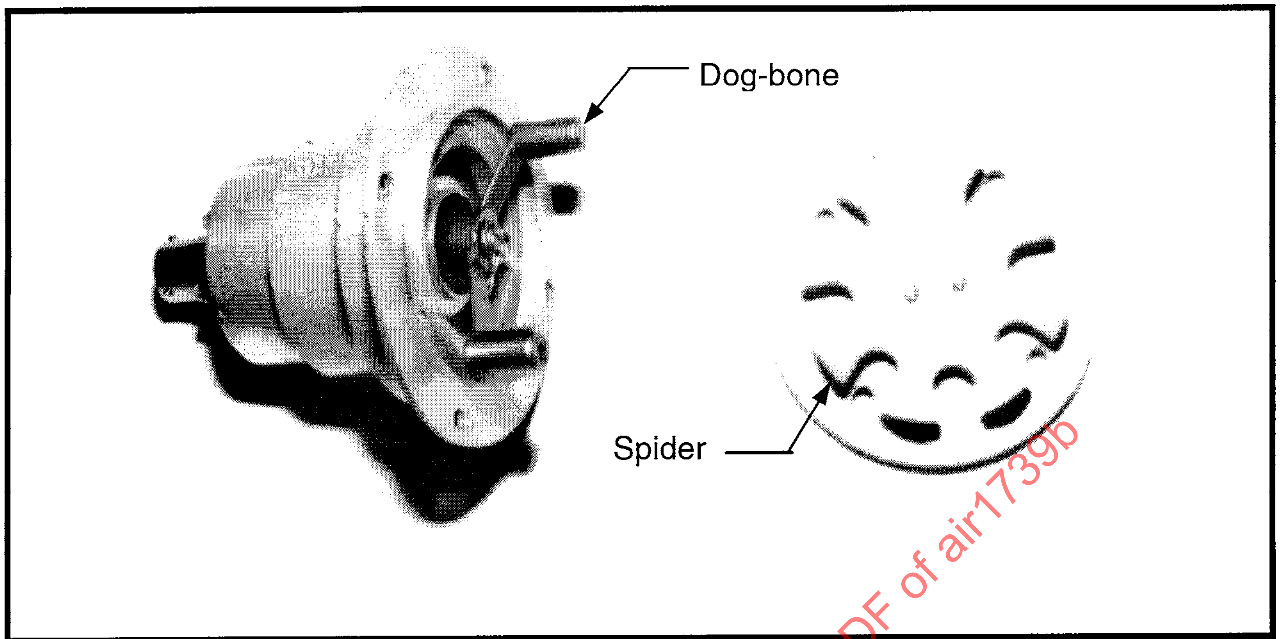


FIGURE 7 - SPIDER/DOG-BONE COUPLING

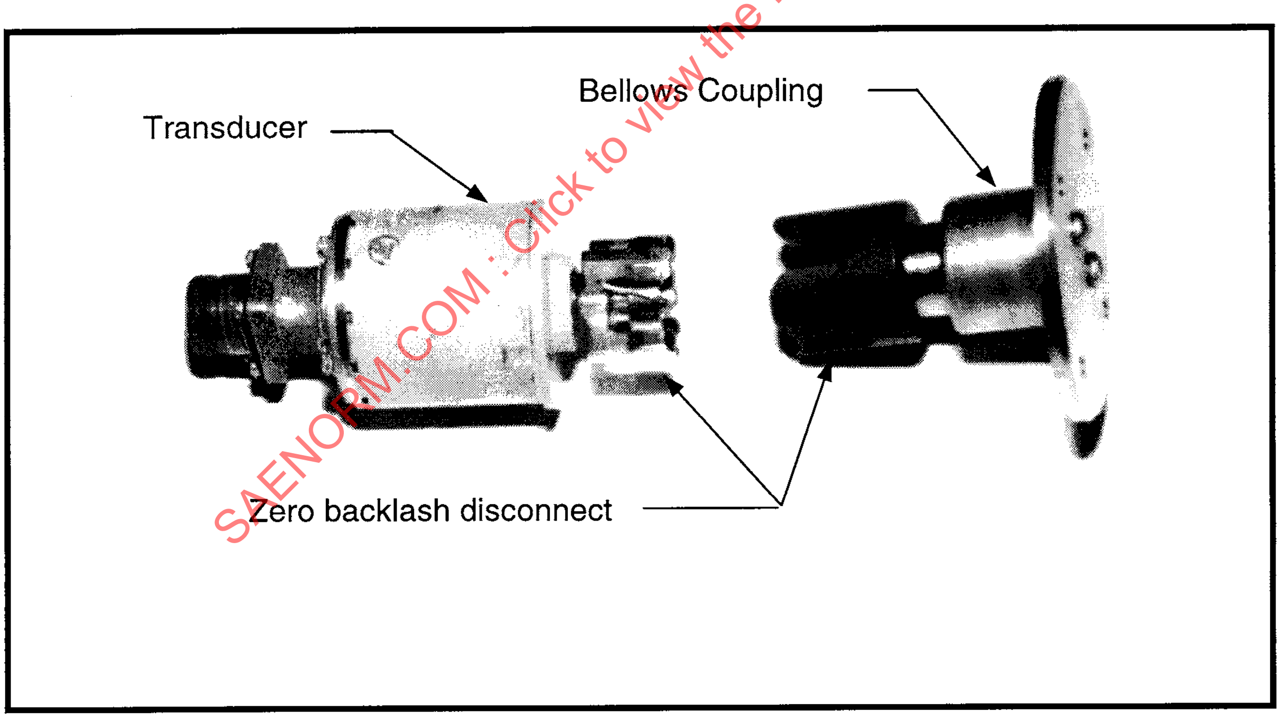


FIGURE 8 - BELLOWS COUPLING

4.1.5 Sensor/Exciter Ring, Wheel Speed Transducers

An alternate approach is the use of a sensor/exciter ring. The exiter ring is mounted to the rotating portion of the wheel/brake assembly while the sensor is mounted to the stationary portion. A sensor and exciter ring are shown in Figure 9.

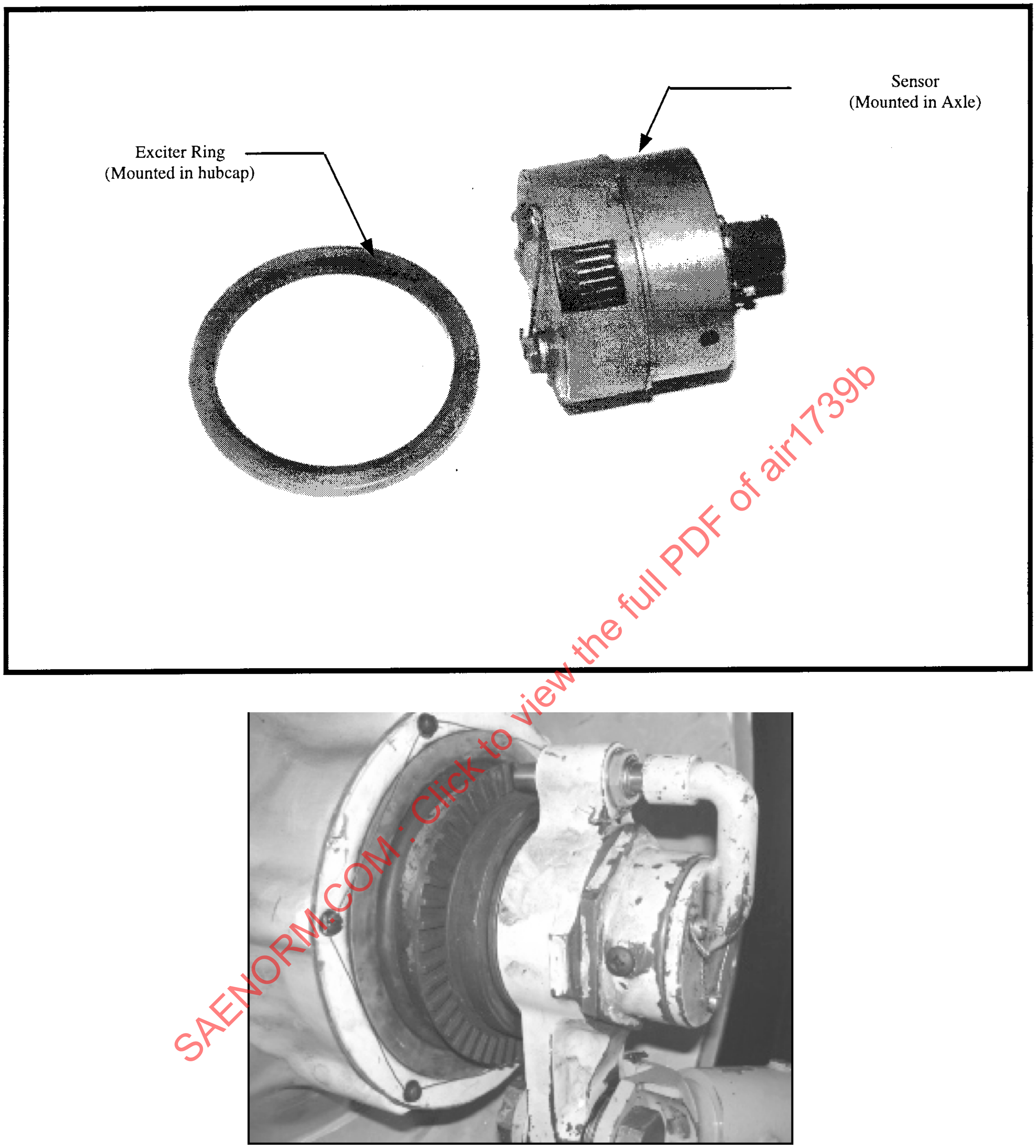


FIGURE 9 - SENSOR - EXCITER RING

4.1.6 Hall Effects Wheel Speed Transducers

Another “contactless” wheel speed transducer that has recently entered into service uses a ring of hall effects sensors mounted to the axle that measures the precise position of a ring of magnets mounted to the wheel. It can detect wheel speed down to zero and can differentiate between forward and backwards rotation. In one implementation, the antiskid control computer is located on the same circuit board as the hall effects sensors. The hall effects wheel speed transducer may be seen in Figures 10 and 11.

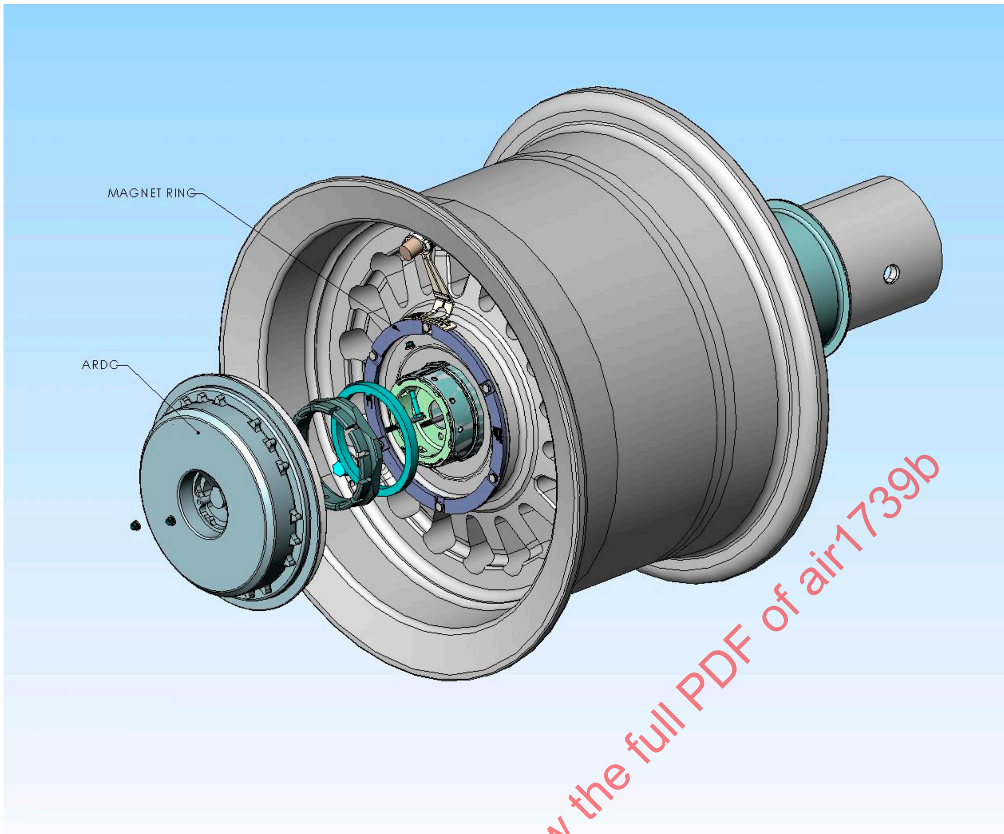


FIGURE 10 - AXLE RDC, BLOW UP

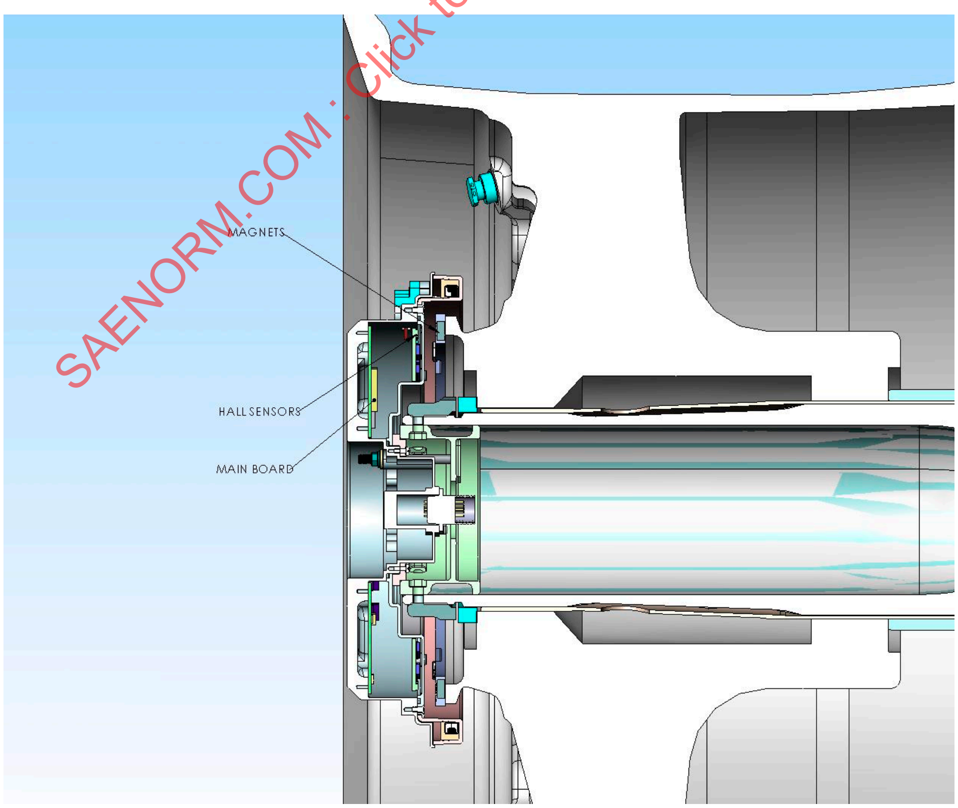


FIGURE 11 - AXLE RDC, SIDE VIEW

4.2 Controller

The control circuit is the brain of the antiskid system. Normally several functions are provided, regardless of the detail of the control concept. These functions include conversion of the wheel speed signal, computation, valve current generation on hydraulic braking systems, and digital control signal generation on electric braking systems. In addition, locked wheel protection and built in test features are provided.

When an AC wheel speed sensor is used, the frequency content of the sensor output must be converted to a value representative of wheel speed, either a DC voltage in analog circuitry or a digital word in microprocessor based controllers. The computational section of the circuit utilizes a number of control functions which are designed to command the maximum tire-runway friction coefficient regardless of airplane weight, speed, or runway condition. Manufacturers utilize different computational concepts to meet this basic requirement. For hydraulic braking systems, a valve driver is required to amplify low level voltages from the computational circuits to current levels required by the antiskid valve. Proper valve driver design can reduce the induction lag of the valve. For electric braking systems a digital signal is transmitted over a digital bus to the electric motor controller that drives the electric actuators.

In the past, the control functions were often implemented as electronic analog circuitry. Virtually all of the newer systems being developed today use microprocessor based (digital) controllers, in which the functions within the controller are implemented in a program that executes on the microprocessor. This approach has allowed the implementation of more complex control functions and has improved system efficiency over the range of operation. However, digital control does introduce a small lag in response, both sampling and computational, that may require particular attention to avoid degradation in system stability.

Use of microprocessor based systems allowed improved fault detection within the system. The additional complexity has added the potential for more failure modes and increased removals of brake system components. Fault detection circuitry and logic must be carefully designed and implemented to avoid unwarranted removal of brake control system components.

4.3 Antiskid Valve

In a hydraulic brake system the antiskid valve is the interface between the low power electronic control circuit and high power hydraulic brake system.

A schematic of a typical antiskid valve is shown in Figure 12. The figure on the left is a view of a first + second stage antiskid valve cartridge that may be plugged into a manifold. The figure on the right is a schematic of the first and second stages. The typical antiskid valve is a two-stage valve with a flapper/nozzle first stage and spool and sleeve second stage. A permanent magnet torque motor in the first stage operates the flapper. Both 3-way and 4-way first stage valves are used. In a 3-way valve, the application of an electrical signal to the torque motor from the skid control box causes the flapper to move from its zero command position against the return nozzle (maximum pressure). Movement of the flapper unbalances the bridge and or control chamber pressure, with a resultant command pressure to the second stage spool. The forces on the spool work to position it until an equilibrium position is reached and the feedback brake port pressure balances the command pressure. The output of the antiskid valve provides the control pressure to the brakes..

In a classical antiskid system, movement of the flapper from the zero command position serves to reduce pilot metered pressure to the brake. Because of fail-safe requirements the valve is normally designed to port the pressure commanded by the pilot to the brake when no current is present. As current is increased, the valve reduces the brake pressure.

In a brake by wire system, the functions of pressure application and reduction are provided by a single valve, with pilot pedal position fed to the brake control computer to be combined with antiskid and the other control functions. In this case movement of the flapper from the relaxed position typically serves to increase pressure to the brake. Increasing current to the valve increases brake pressure. In a brake by wire system, some method for insuring that pressure is not applied when it is not commanded is required. The inclusion of feedback pressure transducer at some point in the brake lines may be used to detect pressure greater than commanded (or less than commanded). A solenoid may be used to remove pressure from the system if pressure greater than commanded is detected. The implementation of the feedback pressure for pressure greater than commanded detection may include a high speed pressure control loop, which can significantly affect antiskid control. The interaction of the normal, wheel speed driven, antiskid control loop and the pressure control loop must be carefully considered in the design of such an antiskid system.

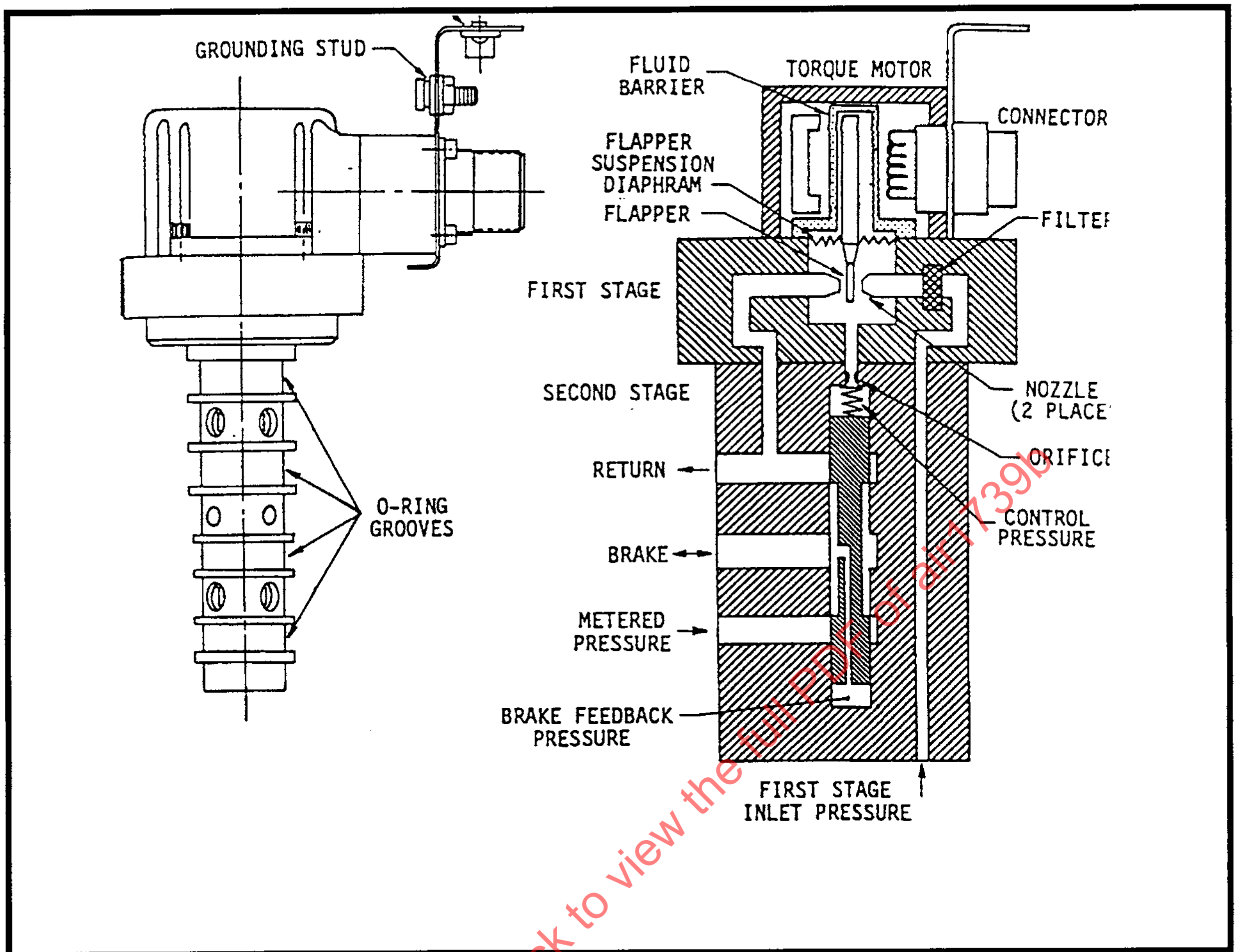


FIGURE 12 - ANTISKID SERVO VALVE (3-WAY)

Direct drive antiskid drives also exist, in which the first stage is replaced by an electric motor which drives the second stage spool directly.

4.4 Electric Brake Actuation

Figure 13 shows two views of an electric brake.

FIGURE 13 – ELECTRIC BRAKES

A cut away of a first generation electric brake actuator may be seen in Figure 14.

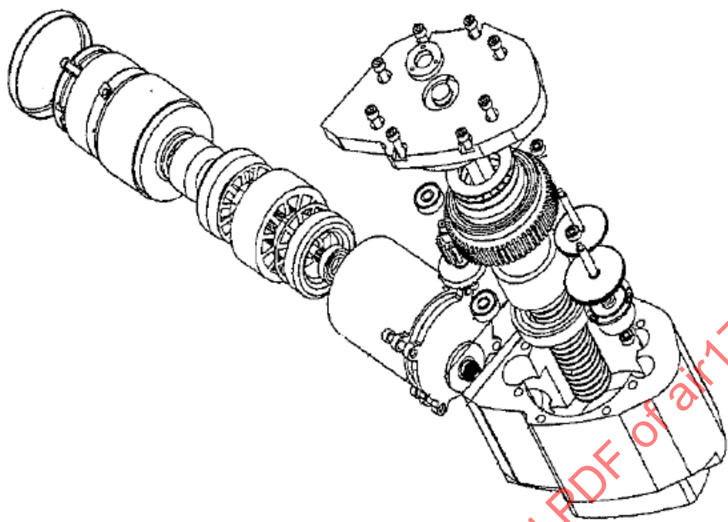
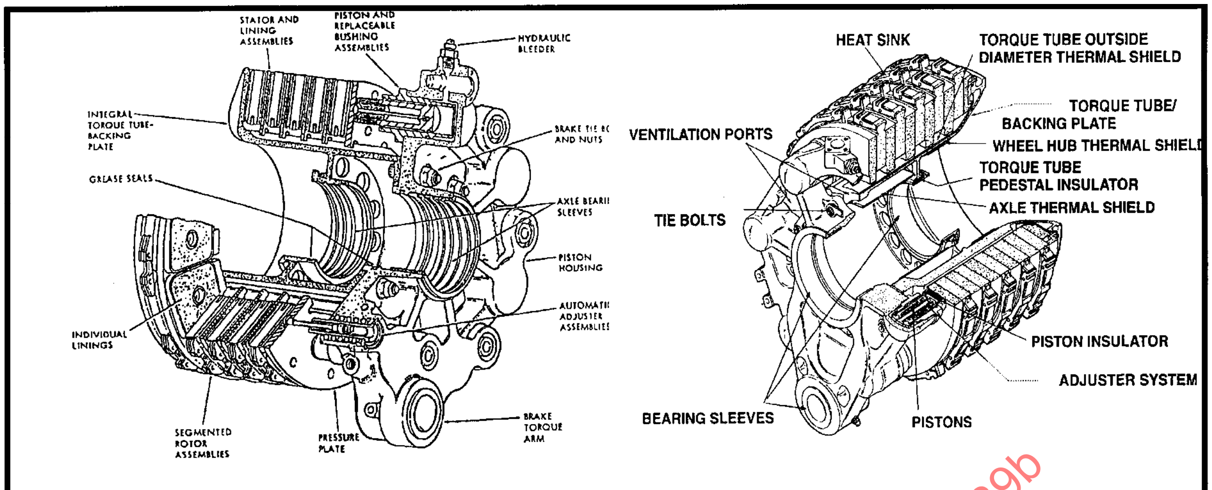


FIGURE 14 – ELECTRIC BRAKE ACTUATOR (FIRST GENERATION)

An electric brake actuator produces clamping force on a brake stack (in a fashion similar to hydraulically driven pistons). At present, these actuators typically consist of an electric motor driving a ball screw and a gearing arrangement which drives a piston which produces clamping force on the brake stack. An electric motor actuation controller (EMAC) provides the power and controls the motor. This control may be based on position, electric current, and force/torque feedback. Electric braking is a rapidly evolving technology so future, and even current, electric brake control system actuators may operate differently and be controlled differently than described here.

4.5 Brake and Tire

The hydraulic brake is normally a multi-stator and rotor disk type designed to absorb the kinetic energy of the airplane. A sectional view of typical steel and carbon heat sink brakes for use on a jet transport plane are shown in Figure 15. In a hydraulic brake system, hydraulic pressure is converted into a mechanical clamping force with the pistons located peripherally in the brake housing, around the axle. In an electric brake, electrically driven motor and gear assemblies apply the clamping force. As clamping force is applied, the rotors and stators are forced together and against the backing plate. The resulting brake torque is a function of the applied clamping force. High brake temperatures can cause fading of brake torque, while at low speed torque can increase rapidly. As brake linings wear, automatic adjusters provide for a stroke limit of the pressure plate to minimize piston free travel. In a hydraulic brake, a small amount of oil volume is required to pressurize the brake to full system pressure through the full range of brake wear.



a. Steel Heatsink Brake

b. Carbon Heatsink Brake

FIGURE 15 - TYPICAL AIRCRAFT BRAKE ASSEMBLIES

The tire, in conjunction with the runway, generates the retarding friction forces as the brake clamping force is applied and provides a critical function in stopping the aircraft. Friction is generated at the interface between the ground and the tire footprint by forcing the wheel to rotate slightly slower than the ground speed demands. The difference between the ground speed (or synchronous speed) and the tire rotational speed times the rolling radius is called slip velocity. The slip ratio is defined as one minus the ratio of linear wheel speed (angular wheel speed times the rolling radius) to the ground speed. As the slip ratio increases, friction forces increase up to some slip velocity (typically 6% to 20% of the ground speed, depending on the size and stiffness of the tire). Above this slip velocity friction either decreases rapidly or levels off. The function of tire-runway coefficient of friction versus the slip velocity, or slip ratio, is called the mu-slip curve. Figure 16 gives a rough idea of the shape of the mu-slip curve.

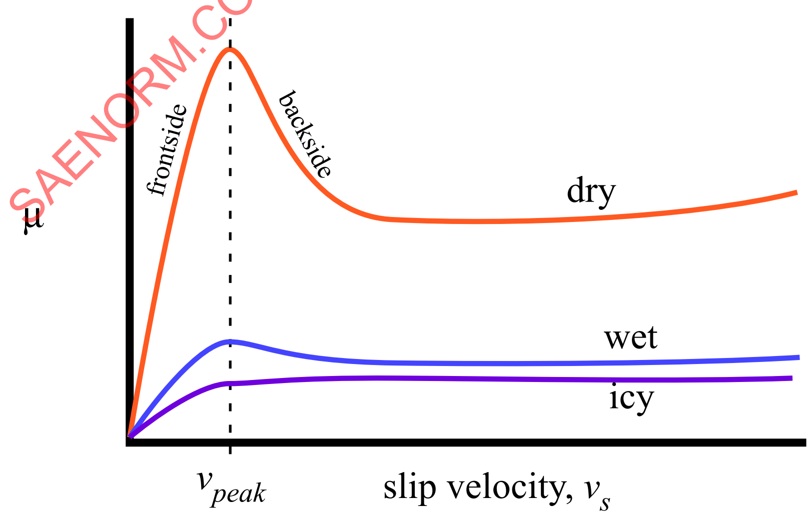


FIGURE 16 – MU VS SLIP CURVE

There is differing data, theory, and opinion about the mu-slip curve. One theory concerning dry pavements posits that tire heating plays a dominant role in limiting friction. As the speed difference increases beyond a certain amount of slip, friction decreases continuously all the way to lock up. On wet pavements tire heating plays a less dominant role and friction stays nearly constant beyond the peak of the mu-slip curve. There is data to support this explanation but other data and opinion suggests that on slick surfaces the curve is not flat on the backside but that the coefficient falls off much as it does on dry surfaces.

It is clear that the exact shape of the mu-slip curve is not completely understood. Some types of tires (radial for one) may have a broad, flat peak, even on a dry runway. The peak is normally considered to be located at a certain percentage of the aircraft velocity (percent slip), so the slip velocity at the peak becomes lower as the aircraft speed reduces. On wet and icy runways, some experience and testing suggests that the percent slip at the peak may be lower than it is on dry. Other experience suggests exactly the opposite, that the percent slip is higher on slippery surfaces. Below some aircraft velocity, the slip velocity at the peak may be constant, instead of being a percentage of the aircraft velocity. In addition, on the frontside of the mu-slip curve, the tire footprint is not actually slipping relative to the ground. The stretching of the tire rubber when brakes are applied causes the rotational speed at the axle, where the wheel speed transducer is located, to be slower than the rotational speed of the tire at the tire/ground interface. The best way to visualize this is perhaps the tire-pulley analogy which may be seen in Figure 17.

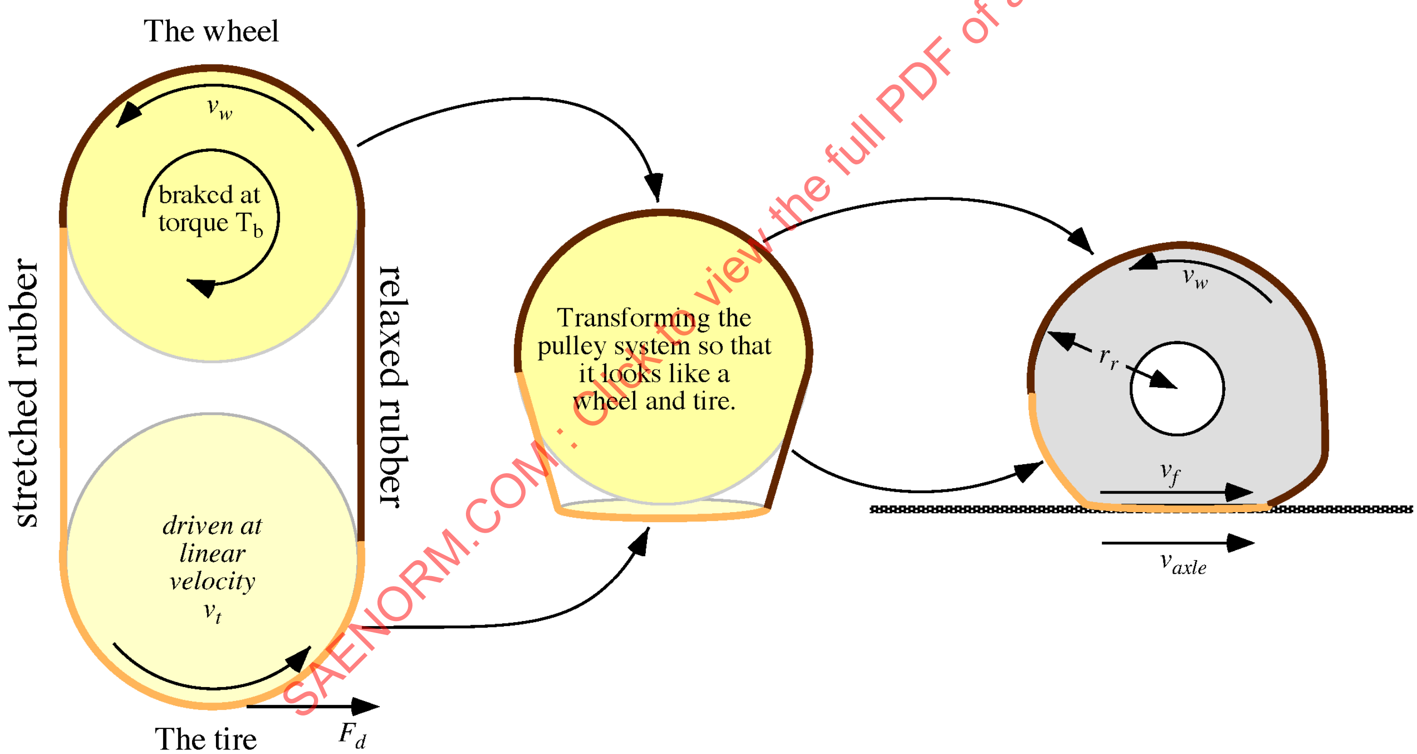


FIGURE 17 – TIRE PULLEY ANALOGY

At any rate, the shape of this curve and the location of the peak is very important to antiskid control.

Aircraft tires also provide the distribution of gear load into the ground and the generation of side forces for directional control. The cross section of typical bias and radial aircraft tires may be seen in Figure 18 and 19. Aircraft tires must operate over a wide temperature range. Compared to automotive tires, aircraft tires run at much greater deflection, usually 30 to 35% deflection of the section height. The side wall does not provide much stiffness by itself. (In contrast, the side walls of automobile tires do provide a significant amount of the total stiffness. Truck tires are typically more like aircraft tires in this respect.) The wheel serves to complete the toroidal pressure vessel and the inflation pressure, which can range from 45 psi to over 450 psi depending on application, provides most of the tire load bearing characteristics.

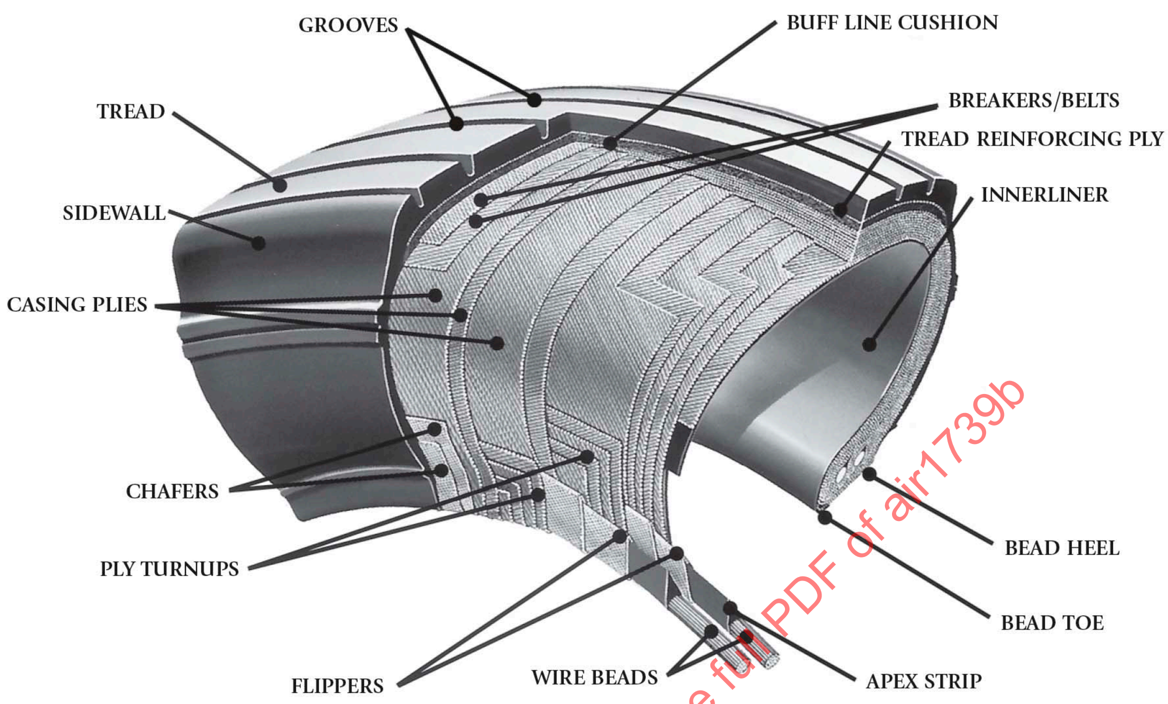


FIGURE 18 - TYPICAL AIRCRAFT TIRE CONSTRUCTION (BIAS)

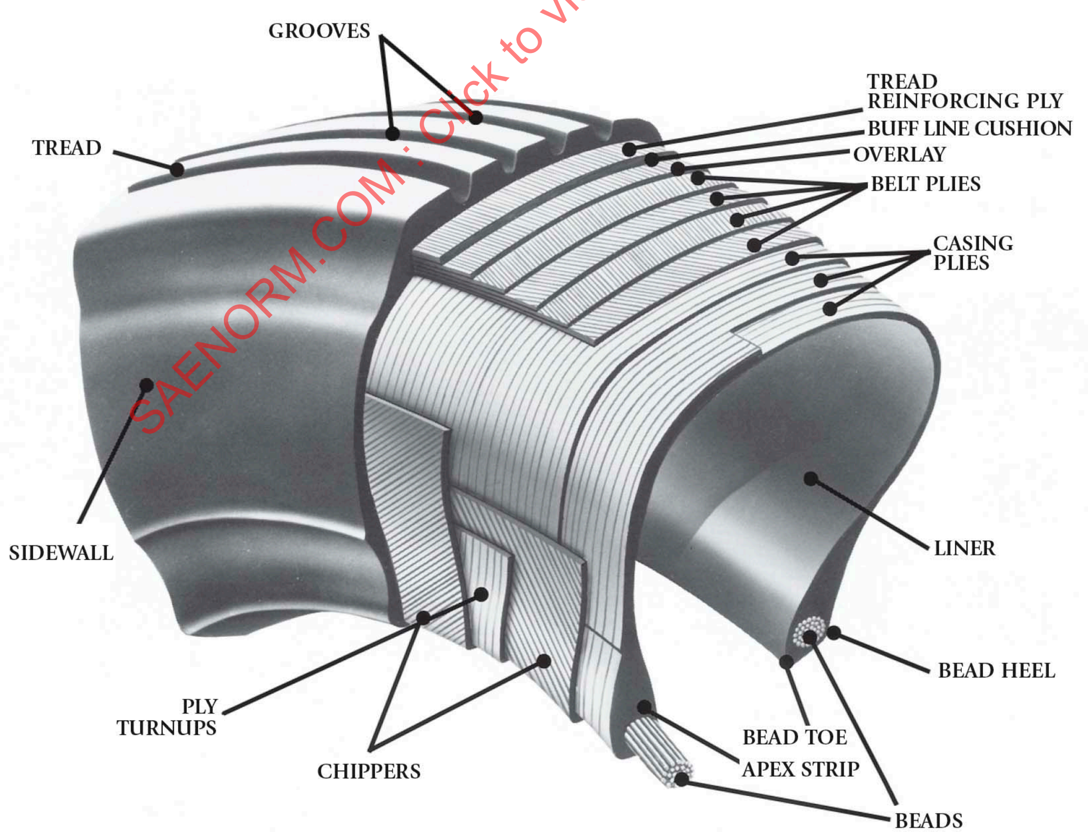


FIGURE 19 - TYPICAL AIRCRAFT TIRE CONSTRUCTION (RADIAL)

5. SYSTEM PERFORMANCE EVALUATION

System performance evaluation (stopping distance and various efficiency calculations) may be performed in the following environments:

- Aircraft Tests: According to AC25-7B, 3 Determination of a Specific Wet Runway Anti-Skid System Efficiency, (ii), "Two acceptable methods, referred to as the torque method and the wheel slip method, for determining the wet runway anti-skid efficiency value from wet stopping tests are described below. Other methods may be acceptable if they can be shown to give equivalent results." The descriptions of these methods below in sections 5.1.2 and 5.1.3 may differ in wording and detail from the descriptions in AC25-7B. Both of these methods, at this time, require human interpretation of flight test data.

  Note that flight tests are extremely expensive, so the number of antiskid performance stops is limited. In addition, there are many unknowns, including the tire-runway coefficient of friction at various points on the runway and various aircraft speeds.

- Simulation Tests: Simulation tests fall into two basic categories: Pure Simulation and Hardware-in-the-Loop (HITL or RTHITL Simulation.

  Pure Simulation:

  In pure simulation, everything is simulated, including the control and the brake actuation. Pure simulation may be either real-time or non-real-time, depending on the type of computer used for simulation. Note that, at the present time, high fidelity, fast running, predictive modeling of hydraulic systems and electric brake actuators may be difficult.

  HITL Simulation:

  In HITL simulation (sometimes called RTHITL simulation, Real-Time-HITL simulation) a real time simulation computer running models of the aircraft, landing gear, tire, and wheel dynamics is coupled with a real brake control computer and with real brake actuation hardware. The brake actuation hardware typically consists of either a hydraulic rig or electric brake rig duplicating as closely as possible the actual aircraft brake actuation system. On occasion, either the brake control computer or the brake actuation hardware may be simulated as well.

  In the early years of antiskid control development, simulations were performed on analog simulation computers. They were relatively simple and the fidelity was low. Modern simulations are performed on digital computers. The fidelity of the simulations is constantly increasing as processor capabilites increase and as experience in matching simulations to aircraft flight test results is accumulated.

  There is a multiplicity of methods for evaluating performance in simulation environments. Since all of the state variables (the variables on the left hand side of the differential equations that define the state of the system) and variables associated with them are known, efficiency methods listed in Section 5.2 are exact and do not call for human interpretation in their implementation. They are entirely algorithmic and may be programmed into the simulation. The limitation is the fidelity of the match between the aircraft and simulation. Note that the aircraft efficiency methods outlined in Section 5.1 may be performed on simulation generated data in the same manner they are performed on flight test data.

  At present, antiskid tuning (adjusting the parameters of antiskid control for acceptable performance throughout the envelope of operation) is best performed in HITL simulation. In HITL simulations, hundreds of stops may be performed per day. Test matrices that cover every conceivable point and corner of the operational envelope, along with every failure condition and combination of failure conditions, can be devised and implemented. The number and types of tests that are conducted is limited more by the ability of human beings to evaluate the results than limitations in the simulation itself.

- Dynamometer Tests: Dynamometer testing with antiskid in the loop is not required in the design of antiskid systems, as current and past systems have demonstrated. Dynamometer tests including antiskid in the loop are performed in some circumstances, typically in cases where investigation of the interaction between antiskid control and rotating hardware (brakes, wheels, tires, wheel speed transducers, wheel speed transducer couplings) is required and there is some question in the fidelity of the simulation (or suspicion of unknowns) in modeling the behavior. The aircraft efficiency methods described in Section 5.1 are appropriate to apply to dynamometer tests. Caution in the use of efficiency calculated on a dynamometer should be observed, as antiskid may perform differently on a dynamometer than it does on an aircraft. Note that, even when antiskid in the loop testing is performed on a dynamometer, the number of test cases is limited both by cost and the number of dynamometer stops that can be performed each day.

5.1 Aircraft Tests

Stopping distance performance for the airplane should be determined from measured stopping distance, and may be presented as an average friction coefficient or as an antiskid system efficiency. The average friction coefficient or antiskid system efficiency can be calculated by any of the following flight test methods.

5.1.1 Average Friction Coefficient Calculation - Basic Method

The average friction coefficient is calculated by dividing the average braking force by the average vertical load on the braked wheels. The first step is calculating an average braking force. This can be done in several ways as indicated in Alternate Methods 1-3 (Sections 5.1.1.1, 5.1.1.2, and 5.1.1.3). In the basic method average braking force is calculated from brake energy. The average braking force is obtained by recording the instantaneous values of the brake torque and wheel rotational speed for each braked wheel during the stop. Brake energy is calculated by integrating the product of brake torque and wheel speed from brake application to stop. (Note that brake torque is difficult to measure on many airplanes so this is not always an accurate or appropriate method.)

$$BE = \int_{t_A}^{t_R} (TB \cdot \omega)\,dt \qquad \text{(Eq. 1)}$$

where:

BE = Net brake energy from a single brake

TB = Measured brake torque

ω = Measured wheel angular speed

t_A = Time of brake application

t_R = Time of brake release

The net brake energies from all the wheels are summed (Equation 2) and the average braking force is obtained by dividing the total energy by the measured stopping distance (Equation 3). This method does not account for the energy absorbed by the tire and the wheel bearing rolling resistance is assumed to be negligible. Measurement of brake torque may include errors due to frictional losses. Correction for these losses should be considered.

$$BET = \sum_{i=1}^{NB} BE_i \qquad \text{(Eq. 2)}$$

$$FBA = \frac{BET}{SB} \qquad \text{(Eq. 3)}$$

where:

FBA = Average braking force

BET = Total net brake energy

BE_i = Brake energy for brake i

NB = Number of brakes

SB = Measured stopping distance from brake application

Furthermore, Equation 3 is derived such that the energy absorbed by the brakes and tires is isolated from all the other braking forces associated with an aircraft stop that are not related to the brakes. Examples of these ancillary forces include aerodynamic drag and retardation forces from such sources as deceleration parachutes and reverse thrust.

The average friction coefficient is the average braking force divided by the aircraft weight minus the average aerodynamic lift minus the average load on the unbraked nose wheels.

$$\mu_A = \frac{FBA}{(WT - LIFT_A - F_N)} \quad \text{(Eq. 4)}$$

where:

μ_A = Average friction coefficient

WT = Aircraft weight

$LIFT_A$ = Average aerodynamic lift

F_N = Average load on unbraked nose wheels

The expression in the denominator of Equation 4 represents the load on the main-gear wheels (F_M). The relationship between main-gear wheel loads, aircraft geometry, and braking drag is illustrated in Figure 20 and the formula for the nose gear vertical load is given in Equation 5. (Note that the aerodynamic lift is difficult to measure, reducing the utility of this method.)

$$F_N = \frac{(WT - LIFT_A) * X + A_H * FBA}{L} \quad \text{(Eq. 5)}$$

where:

L = Spacing between nose and main gears

L-X = Distance from center of gravity to nose gear

X = Distance from center of gravity to main gear

A_H = Height of the center of gravity above ground contact

FBA = Braking force

F_M = Main-gear load

F_N = Nose-gear load

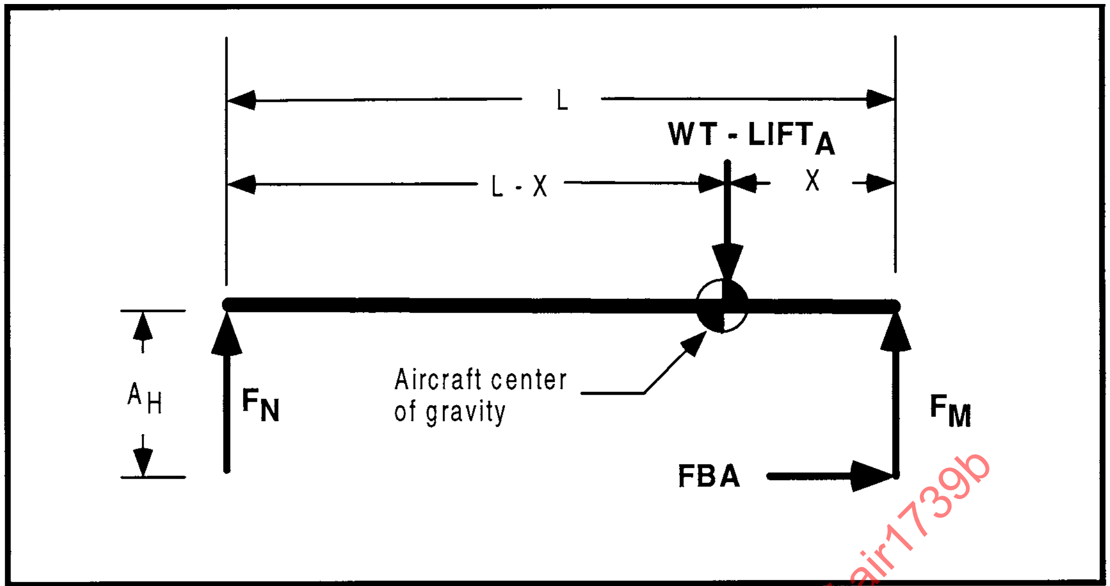


FIGURE 20 - AIRCRAFT LANDING GEAR GEOMETRY

5.1.1.1 Average Friction Coefficient Calculation - Alternate Method 1

This method is identical to the previous calculation method except that the percent of total energy absorbed by the brakes variable, PCEB, is introduced into Equation 3 to account for the energy absorbed by the tire, so that:

$$FBA = \frac{BET}{SB} \cdot \frac{100}{PCEB} \qquad \text{(Eq. 6)}$$

where:

PCEB = Percent of total energy absorbed by brakes

and FBA then becomes the braking force due to the brakes and tires. The value of PCEB must be determined from other testing or estimated. Some manufacturers consider the previous method to be a better indication of aircraft braking performance than this method.

5.1.1.2 Average Friction Coefficient Calculation - Alternate Method 2

In this method the average friction coefficient is calculated from measured brake torque and the moment of inertia of the wheels and tires according to Equations 7 and 8:

$$\mu \equiv \frac{1}{NB}\sum_{i=1}^{NB}\mu_i = \frac{1}{NB}\sum_{i=1}^{NB}\frac{I_i\dot{\omega}_i + TB_i}{F_{M,i}R_{H,i}} = \sum_{i=1}^{NB}\frac{I_i\dot{\omega}_i + TB_i}{NB \cdot F_{M,i}R_{H,i}} \approx \sum_{i=1}^{NB}\frac{I_i\dot{\omega}_i + TB_i}{F_{M,tot}R_{H,avg}} \qquad \text{(Eq.7)}$$

$$\mu_A = \frac{1}{t_A - t_R}\int_{t_A}^{t_R}\mu\, dt \qquad \text{(Eq.8)}$$

where:

μ = Instantaneous friction coefficient

ω = Wheel angular speed

I = Wheel and tire moment of inertia

R_H = Deflected tire radius (axle height above ground contact)

NB = Number of brakes

TB = Brake torque

F_M = Main landing gear load

t_A = Time of brake application

t_R = Time of brake release

Implementation of this method involves instrumentation to measure wheel speeds, brake torque, and main-gear wheel loads.

5.1.1.3 Average Friction Coefficient Calculation - Alternate Method 3

In this method the average friction coefficient is calculated indirectly from the measured stopping distance. A value for the average friction coefficient is assumed and the stopping distance is calculated based on braking force, engine thrust and aerodynamic lift and drag. The calculated stopping distance is compared to the measured stopping distance and the friction coefficient is adjusted and a new stopping distance is calculated. This iterative procedure is repeated until the difference between measured and calculated stopping distance is less than a specified tolerance. One significant drawback to this method is the difficulty of accurately measuring or calculating aerodynamic lift and drag, particularly after noting that ground effects must be taken into account.

5.1.2 Antiskid System Efficiency Calculation - Drag Force/Torque/Pressure Efficiency Method

As an alternative to stopping distance performance, efficiency calculations based on measured parameters can be made for those aircraft tests which include direct measurements of drag force, brake torque, or brake pressure. For this method the drag force, brake torque, or brake pressure curves should be integrated from brake application to brake release to get the actual area under the curve. The peaks (not transients) of the curves may be connected and integrated to get the optimum area under the curves. Care should be taken in selection of the peaks to insure that they can be clearly identified as being coincident with a friction peak. The efficiency of braking is defined as the ratio of the actual area divided by the optimum area as shown in Figure 21.

A better approximation can be made by using a velocity-weighted summation. This summation is done by multiplying both measured and peak values of drag force, brake torque, or brake pressure by velocity before the integration is performed. This approach addresses the fact that braking at high speed has a larger impact than braking at low speed.

For this method of estimating antiskid braking performance, the drag force curve would be the first choice for analysis. If this parameter is not available, then the brake torque curve or the brake pressure curve may be substituted. Antiskid braking performance estimates based on the brake pressure curve will probably be the least accurate in this group.

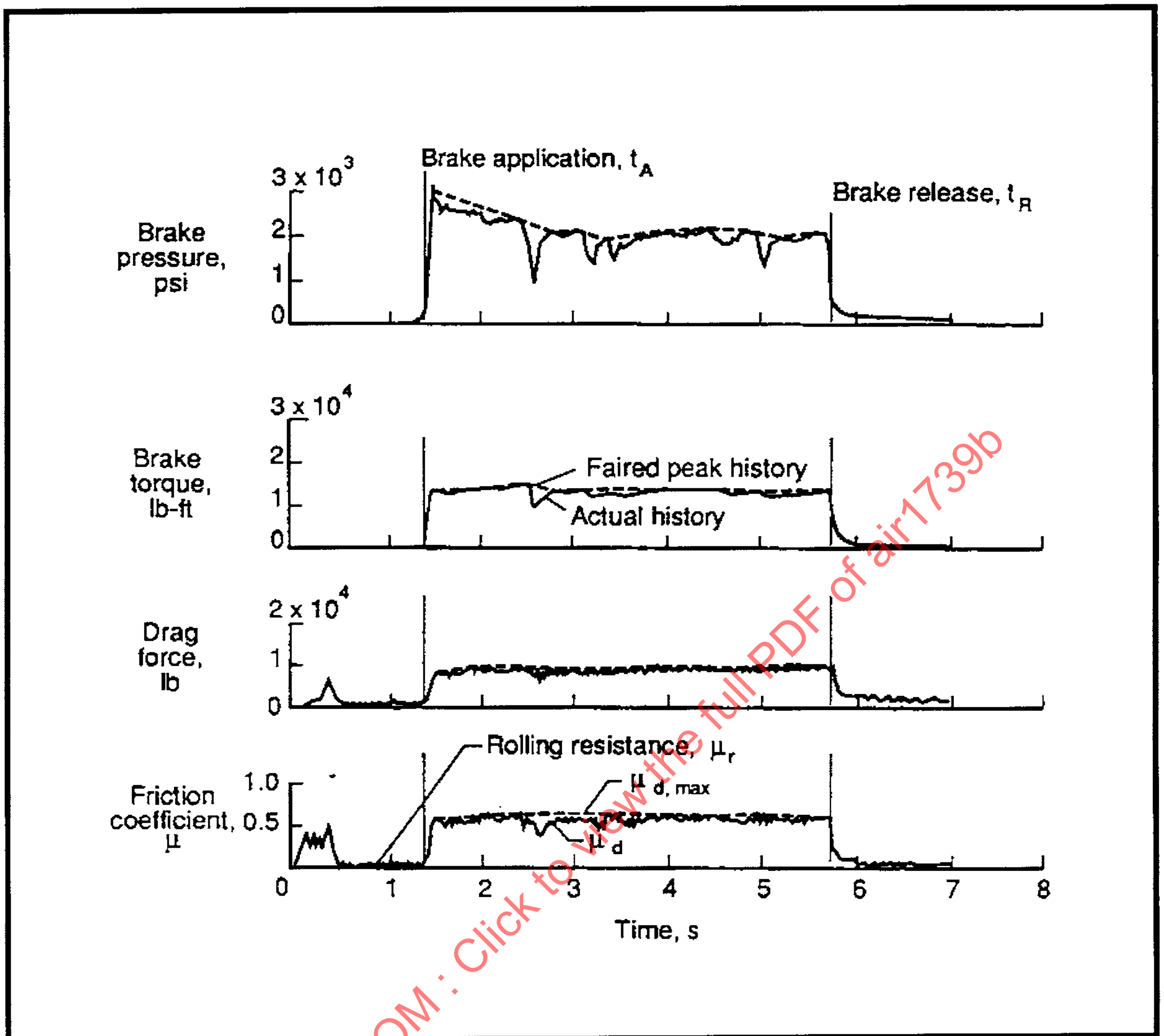


FIGURE 21 - BRAKE TORQUE/PRESSURE/DRAG FORCE EFFICIENCY

5.1.3 Antiskid System Efficiency Calculation - Wheel Slip Method

At brake application, the tire begins to slip with respect to the runway surface, i.e., the wheel speed slows down with respect to the airplane's ground speed. As the amount of tire slip increases, the brake force also increases until an optimal slip is reached. If the amount of slip continues to increase past the optimal slip, the tire will begin to skid, which reduces the braking force.

Using the wheel slip method, the antiskid efficiency is determined by comparing the actual wheel slip measured during a stop to the optimal slip. Since the wheel slip varies significantly during the stop, sufficient wheel and ground speed data must be obtained to determine the variation of both the actual wheel slip and the optimal wheel slip over the length of the stop. A sampling rate of at least 16 samples per second for both wheel speed and ground speed has been found to yield acceptable fidelity. It may be necessary to perform an analysis of tire radius, brake+wheel+tire rotating moment of inertia, and wheel loading to determine the data rate necessary to capture the relevant dynamics.

For each wheel and ground speed data point, the instantaneous antiskid efficiency value should be determined from the relationship shown in Figure 22.

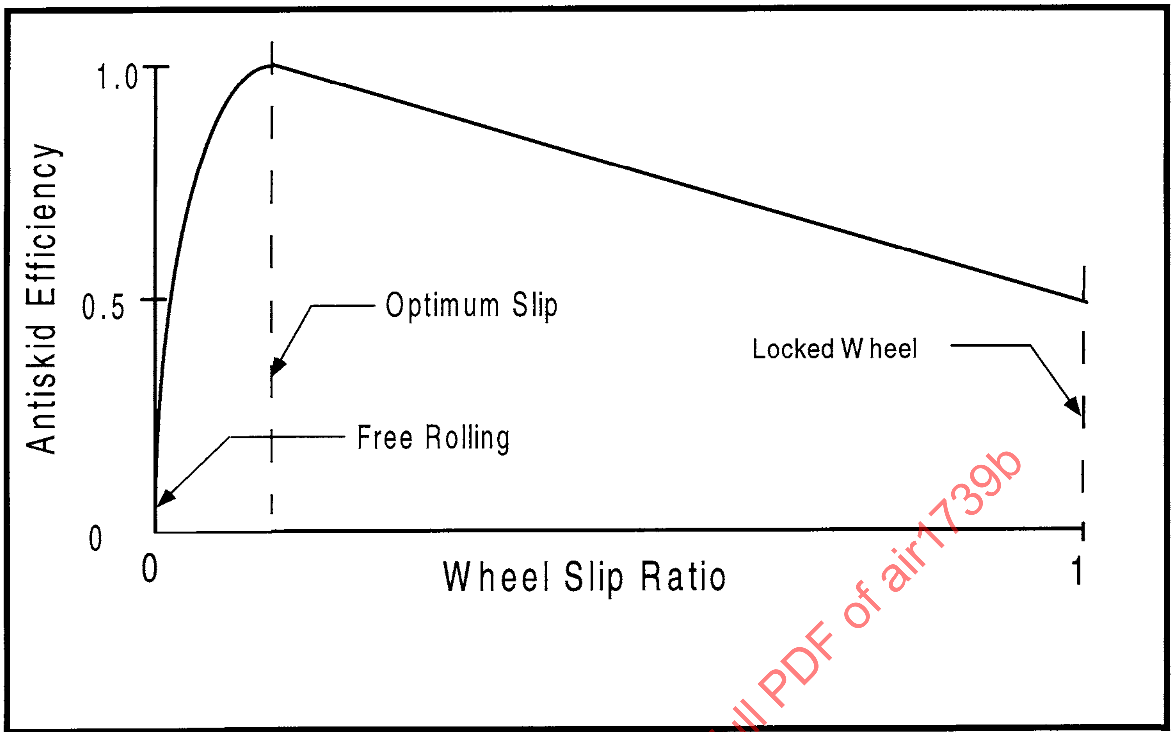


FIGURE 22 - ANTISKID EFFICIENCY VERSUS WHEEL SLIP RATIO

For

$$\text{WSR} < \text{OPS Efficiency} = 1.5\left(\frac{\text{WSR}}{\text{OPS}}\right) - 0.5\left(\frac{\text{WSR}}{\text{OPS}}\right)^3$$

$$\text{WSR} = \text{OPS Efficiency} = 1.0$$

$$\text{WSR} > \text{OPS Efficiency} = 0.5\left(1 + \frac{(1 - \text{WSR})}{(1 - \text{OPS})}\right)$$

where:

$$\text{WSR} = \text{The wheel slip ratio} = 1 - \left(\frac{\text{Linear wheel speed}}{\text{Ground speed}}\right) \tag{Eq.9}$$

and OPS is the optimal slip ratio

The optimal wheel slip ratio value(s) (OPS) must be determined to use this method. A method for determining the optimal slip value(s) is to compare time history plots of the brake force and wheel slip data obtained during the stopping tests. If, during a skid, the wheel slip continues to increase after a reduction in the brake force, the optimal slip is the slip value corresponding to the brake force peak. Figure 23 shows an example of this. Note that both the actual wheel slip and the optimal wheel slip can vary during the stop. For brake installations where measuring brake force directly is impractical, brake force may be determined from other parameters (e.g., brake pressure) if a suitable correlation is available.

To determine the overall antiskid efficiency value the instantaneous antiskid efficiencies should be integrated with respect to distance and divided by the total stopping distance:

$$\text{antiskid efficiency} = \frac{1}{S}\int \text{instantaneous efficiency ds} \tag{Eq.10}$$

where:

s = Stopping distance

The stopping distance is defined as the distance traveled during the specific runway stopping demonstration, beginning when the full braking configuration is obtained and ending at the lowest speed at which antiskid cycling occurs (i.e., the brakes are not torque-limited), except that this speed need not be less than 10 knots.

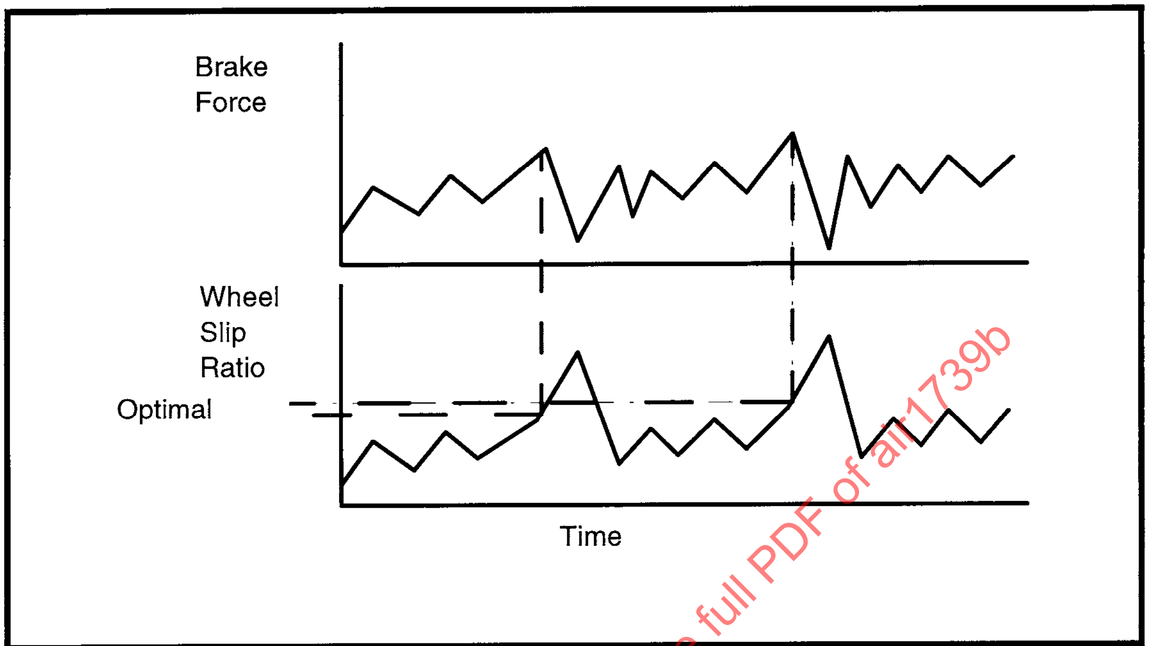


FIGURE 23 - OPTIMUM SLIP RATIO DETERMINATION

Although this is a flight test method, it can be used in simulation. If this method is used in simulation there are two choices for how to implement it:

- Collect data from the simulation exactly as it would be collected in flight test and reduce it using exactly the same process that would be used with flight test data.
- Since WSR and OPS are exactly known in simulation, use them. This implementation is very similar to the developed mu efficiency method but not as exact, since the shape of the mu-slip curve used in simulation may differ from the curve defined for this method.

5.2 Simulator Tests

Methods described in 5.1 can be used for performance evaluation to match the simulator. The simulator can then be used to assess and tune the antiskid system. The availability and controllability of variables not measurable on aircraft tests allows direct calculation of antiskid system efficiency on a computer simulator. Several efficiency calculation methods are available on a simulator as described in the following paragraphs.

Some notes concerning all simulation stopping efficiency methods

- The criterion for starting the efficiency measurement should be clearly defined. Possible triggers for initializing the efficiency calculation might be (all potentially with a defined time delay after the trigger):
    - Brake application
    - Brake pressure has reached the skid pressure or the pedal has reached 100% application (for individual wheel efficiency methods)
    - Brake pressure at the first wheel has reached the skid level or the pedal has reached 100% application (for full aircraft efficiency methods)
    - Brake pressure at all wheels have reached the skid level or the pedal has reached 100% application (for full aircraft efficiency methods)

- The stopping conditions in which efficiency is measured for credit should be clearly defined.
    - The more realistic the simulation, the harder it will be to achieve high efficiency numbers. Runway roughness, runway crowning, high frequency runway surface variations in friction, crosswind gusts, gear stiction, aircraft roll, etc. will all reduce efficiency.
    - An efficiency of 95% or higher is almost certainly obtainable only under ideal conditions (perfectly smooth runway, etc.) which will never occur in the real world.
    - Consideration should be given to how the formulation of the efficiency requirement affects tuning. Will a better performing real world system be produced by tuning for 95% on an unrealistic, perfect runway, or by tuning to a lower efficiency number using a simulation that is as realistic as possible?
    - Note, that for tuning and robustness testing purposes, the envelope of simulation tests may be designed intentionally to be more challenging to antiskid than most normal real world operating conditions. While this may be good practice for insuring the robustness of the antiskid system, it will put the system at a disadvantage when it is expected to produce high efficiency numbers.
    - Note also that tuning to a perfectly smooth runway may actually be dangerous in some cases. For example: on an airplane with a step in the shock strut, or a sticking gear (as described in 7.5.1), a small amount of roughness in the runway can excite significant weight on wheels variations. If tuning is focused on producing 95% efficiency on a smooth runway, the result in the real world can be severe directional control problems.

- In paired wheel control with uneven loading of the wheels (due to a runway crown or axle vertical offsets) or a tire runway coefficient that differs from one wheel in the pair to the other, the wheel with the lower product of weight on the wheel times the coefficient of friction will control the brake pressure to both wheels. The more heavily loaded wheel will not reach its full stopping potential. For a fair efficiency measurement, this effect must be taken into account.

5.2.1 Stopping Distance Efficiency

Stopping distance efficiency is the ratio of the minimum distance required by the aircraft to stop for a given condition and the actual distance required to stop. Stopping distance efficiency is calculated by first conducting a simulated stop with the antiskid system off and the friction force held equal to the maximum available friction coefficient times the instantaneous vertical tire load. This defines the minimum (or perfect) stopping distance. A stop is then made with the antiskid system active to define the actual stopping distance. The efficiency is then obtained by dividing the perfect stopping distance by the actual stopping distance. This calculation is defined by Equation 9:

$$\eta = \frac{X_{perfect}}{X_{actual}} * 100 \qquad \text{(Eq. 11)}$$

where:

η = Stopping distance efficiency

$X_{perfect}$ = Minimum aircraft stopping distance

X_{actual} = Actual aircraft stopping distance

NOTE: When the simulation assumes a constant deceleration stop, then the perfect stopping distance can be defined by Equation 10:

$$X_{perfect} = \frac{\dot{x}_0^2}{2\mu_{max} g} \qquad \text{(Eq. 12)}$$

where:

μ_{max} = Maximum available friction coefficient

g = Gravitational constant

$\dot{x}_0$ = Initial aircraft velocity

5.2.1.1 Potential Issues with Stopping Distance Efficiency Calculation

Although this may seem to be the simplest of the system performance evaluation methods in simulation, there are potential issues with it.

The first has to do with pressure application. In a perfect stop the pedal position input from the pilot might be used to calculate a metered (or pilot commanded pressure). To calculate the maximum deceleration, the lower of this commanded pressure and the skid pressure would be used as brake pressure. What if the control contains some sort of fill algorithm? Then the brakes might actually reach the skid pressure in a normal stop more quickly than they would in an optimal stop. Since brake application occurs at high speed this could increase the optimal stopping distance significantly compared to normal stopping distance and result in much higher efficiency numbers than should be the case. Thus, attention must be paid to how brakes are applied in an optimal stop, as compared to a normal stop, to make sure that the efficiency number is a fair representation of stopping performance.

The second issue has to do with what brake pressure should be applied to the brake during the optimal stop, after the pedal commanded pressure has exceeded the skid pressure. The natural, and fair, pressure to use is the skid pressure. The problem is that in a simulation that includes realistic amounts of runway roughness and tire-runway mu variation this may result in high frequency variations in brake pressure as the skid pressure rises and falls to chase the product of the weight on the wheel and the tire-runway mu. Depending on the amplitude and frequency, fore-aft motion of the gear may be excited. In a normal stop real brake control will filter out these variations. In fact one of the major requirements placed on antiskid is to avoid excitation of the gear at its natural frequency (gear walk) and also to avoid exciting other modes of aircraft oscillation such as pitch and roll. Taking this into account, the “perfect” stop may not be so perfect.

5.2.2 Developed μ Efficiency

Developed μ efficiency is the ratio of the average friction coefficient actually developed during the stop and the average available friction coefficient. The developed μ efficiency is calculated as follows:

a. Calculate the instantaneous actual and available friction coefficients by dividing the instantaneous braking forces by the instantaneous vertical main gear loads.

b. Calculate the distance-weighted-average actual and available friction coefficients for the entire stop by using the instantaneous friction coefficients in the equation below:

$$\text{Distance} - \text{weighted} - \text{average}\ \mu = \frac{1}{s}\int \mu_i\, ds \qquad \text{(Eq. 13)}$$

Substituting $ds = v_i\, dt$ gives

$$\mu = \frac{1}{s}\int \mu_i v_i\, dt \qquad \text{(Eq.14)}$$

where:

μ_i = Instantaneous friction coefficient

v_i = Instantaneous velocity

S = Total stopping distance

NOTE: The integration is carried out from brake application to the end of the stop.

c. Calculate the developed efficiency by dividing the distance-weighted average actual friction coefficient by the distance-weighted average available friction coefficient. If the available friction coefficient varies during the stop, then the average must be determined by distance average integration in a similar manner.

5.2.3 Developed μ Efficiency - Alternate Method 1

A variation of Developed μ Efficiency is suggested in NASA TP 1051 (see 2.3) wherein the friction coefficient ratios are modified to account for the rolling resistance coefficient, μ_r. This method is not recommended. The drawbacks to it are outlined in 5.2.3.1. Equations 13, 14, and 15 describe this method:

$$\bar{\mu}_{avail} = \frac{1}{t_R - t_A}\int_{t_A}^{t_R} \mu_{avail}\,dt \quad \text{(Eq.15)}$$

$$\bar{\mu}_d = \frac{1}{t_R - t_A}\int_{t_A}^{t_R} \mu_d\,dt \quad \text{(Eq.16)}$$

$$\eta = \frac{\bar{\mu}_d - \mu_r}{\bar{\mu}_{avail} - \mu_r} \quad \text{(Eq.17)}$$

where:

$\bar{\mu}_{avail}$ = Average maximum friction coefficient available to the antiskid system

μ_{avail} = Maximum friction coefficient

$\bar{\mu}_d$ = Average friction coefficient developed by the antiskid system

μ_d = Instantaneous friction developed by the antiskid system

μ_r = Rolling resistance friction coefficient

η = Efficiency of antiskid system

t_A = Time of brake application

t_R = Time of brake release

5.2.3.1 Developed μ Efficiency - Alternate Method 1, Drawbacks

A free body diagram of a tire may be seen in Figure 24.

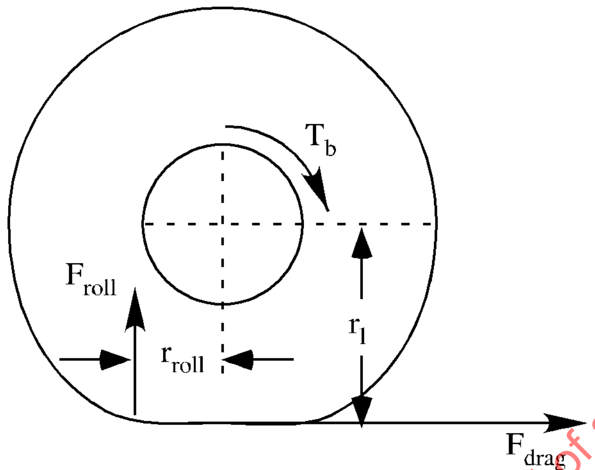


FIGURE 24 - TIRE FREE BODY DIAGRAM

F_{drag} = Tire/runway friction force

T_b = Brake torque

F_{roll} = Rolling resistance

r_l = Tire loaded radius

r_{roll} = Moment arm of the rolling resistance

The tire/runway friction force tends to keep the wheel rolling. Brake torque slows the wheel down. An increase in the tire/runway coefficient of friction will result in an increase in the tire/runway friction force. Rolling resistance does the exact opposite, as it tends to slow the wheel down. Thus, rolling resistance cannot properly be thought of as an increase in the tire/runway coefficient of friction. Rolling resistance is more properly thought of as a retarding torque on the wheel, analagous to brake torque. Rolling resistance may be thought of as equivalent to tire/runway mu only in the same same sense that brake torque may be thought of as being equivalent to a level of tire/runway mu (see Figure 25 outlining the mu versus slip curve below).

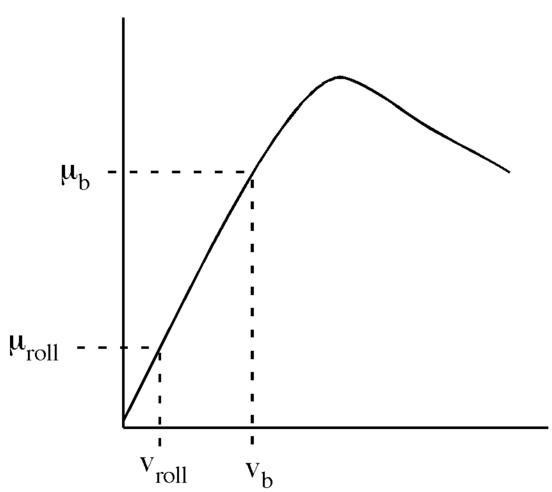


FIGURE 25 - MU-SLIP AND MU ROLL

μ_b = Mu level required to balance the brake torque

μ_{roll} = Mu level required to balance the rolling resistance

v_b = Tire/runway slip required to generate the mu required to balance the brake torque

v_{roll} = Tire/runway slip required to generate the mu required to balance the rolling resistance

$T_b = \mu_b W_w r_l = F_{dragb} r_l$

$F_{roll} r_{roll} = \mu_{roll} W_w r_l = F_{dragr} r_l$

Developed μ Efficiency - Alternate Method 1 penalizes antiskid efficiency by positing that rolling resistance increases the tire/runway coefficient of friction and that that increase should be subtracted from the friction coefficient developed by antiskid. This is not a fair penalty. It is equivalent to penalizing antiskid for a dragging brake.

Rolling resistance is equivalent to some amount of brake torque always being present. If that amount of retarding torque is enough to skid the wheel then there is nothing that antiskid can do about it.

Developed μ Efficiency - Alternate Method 1 should not be used for antiskid efficiency calculation.

5.2.4 Developed Acceleration Efficiency

Developed acceleration efficiency is useful because it can be calculated as the simulation runs and the resulting number is generally comparable to stopping distance efficiency (which requires two simulation stops for every condition).

Note the velocity weighting included in the equations. This weighting has the effect of making the efficiency contribution at high speeds have a larger contribution to total efficiency than efficiency at low speeds. Since poor efficiency at high speeds will result in greater increases in stopping distance, this weighting makes sense.

$$\eta = \frac{\int_{t_{brake}} v_{AC} \frac{a_{AC}}{a_{\max}} dt}{\int_{t_{brake}} v_{AC} dt} = \frac{1}{x_b} \int_{t_{brake}} v_{AC} \frac{a_{AC}}{a_{\max}} dt \qquad \text{(Eq.18)}$$

t = time

t_{brake} = Time at the end of braking

v_{AC} = Aircraft velocity

a_{AC} = Aircraft acceleration (negative for deceleration)

a_{max} = Aircraft acceleration that would be obtained if the tire were operating at the peak of the mu-slip curve (negative for deceleration). Note that a_{max} must be less than or equal to a_{AC},, or $|a_{max}| >= |a_{AC}|$.

x_b = Distance travelled since the start of braking

η = Efficiency

Note that the formula above does not work correctly if engine thrust causes the aircraft to accelerate during braking. The following modifications to the method are suggested for cases where engine thrust is high enough to accelerate the aircraft. Create a secondary velocity, u, as shown in FIGURE 26.

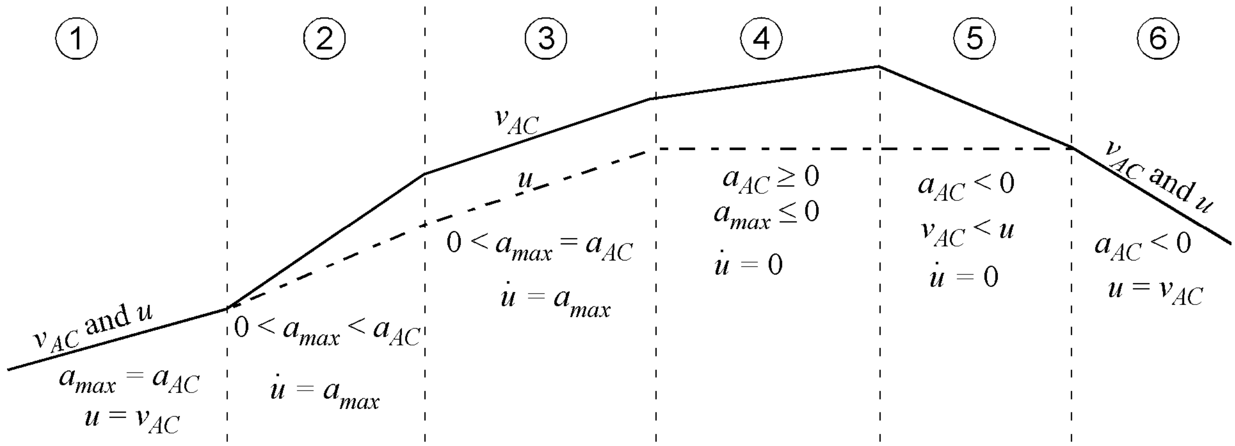


FIGURE 26 - EFFICIENCY FOR AN ACCELERATING AIRCRAFT

Regions 1, 2, and 3

Since a_{AC} becomes greater than a_{max} v_{AC} will increase more rapidly than u and the pseudo distance, x_e, should be calculated accordingly.

$$\dot{x}_e = u \qquad \text{(Eq.19)}$$

Regions 4 and 5

In these regions x_e is being held constant while waiting for the v_{AC} to drop to meet u.

$$\dot{x}_e = 0 \qquad \text{(Eq.20)}$$

Region 6

This is the normal case once aircraft deceleration has been established.

$$\dot{x}_e = v_{AC}\frac{a_{AC}}{a_{max}} \qquad \text{(Eq.21)}$$

$$\eta = \frac{x_e}{x_b} \qquad \text{(Eq.22)}$$

In no case should x_e ever be allowed to be greater than x_b.

5.3 Dynamometer Tests

For testing of antiskid systems conducted on a dynamometer, the efficiency methods defined in 5.1.1 through 5.1.3 may be used.

6. SYSTEM DEVELOPMENT PROCESS

Antiskid Systems are developed in several identifiable steps. These include control concept selection, computer simulation, hardware manufacture, system tuning on airplane, and final performance evaluation. Some of the details of these steps are discussed below.

The control concept selection is usually based on the experience gained on prior programs, on the progress of product development since that time, and, to some extent, the level of performance required. The antiskid system is an adaptive, highly non-linear control system which defies analysis in the classical sense. Computer simulations have contributed extensively to the cost effective laboratory evaluation of different systems. Such simulations normally include representative hydraulic hardware for the brake metering valve, the antiskid valve, the brakes, and the hydraulic lines, or, for electric brakes, the associated power conditioning equipment, the electro-mechanical actuator controller (EMAC), electrical cabling (power and bus) and the brakes. Airplane and landing gear parameters must be defined for the simulation (Table 1 shows a selection of typical parameters). The required degrees of freedom and complexities of the simulation depend heavily upon the difficulties presented to brake control by the aircraft under development as well as the robustness of the antiskid control. Actual antiskid control circuits are normally used to close the loop from computed wheel speed to the antiskid valve command signal. Representation of the antiskid circuit or digital control algorithms on a simulation computer is also possible, especially during early concept evaluation.

Once the hardware has been manufactured, it must be exposed to the usual qualification and acceptance test procedures. These include various environmental and fatigue tests suitable for each component to satisfy the relevant certification authorities. In addition, the hardware must be tested on a system basis. For this purpose the computer/hardware simulation is the most acceptable approach. For a full end-to-end laboratory test, the dynamometer with the real tire, wheel and brake assembly can be used. Performance validation, though, is difficult due to the tire-road wheel interface of rubber on steel, the curved surface, limited control of tire to road wheel friction coefficient, run to run variation due to brake torque variability, and the difficulty in accounting for the effects of aerodynamic drag and engine thrust on mechanical inertia dynamometers. Electrical Inertia dynamometers overcome some of these difficulties due to their capability to control deceleration rate to account for engine thrust and aerodynamic drag effects. In addition, the 168i dynamometer at Wright-Patterson Air Force Base has the capability to put representative materials on the flywheel surface.

Once laboratory testing has been completed, full airplane testing can commence. This is initiated with a number of landings to check out the antiskid system. These early flight tests serve to validate the simulator testing and provide a performance baseline for further tuning changes. The computer/ hardware simulator is also updated to reflect actual gear and airplane dynamic behavior and at the same time is used to validate tuning changes to the antiskid control. Several iterations of check-out landings and tuning changes may be required before the antiskid hardware configuration is frozen for airplane stopping performance tests. For performance landings on wet or dry runways the following conditions generally apply:

1. The landing is preceded by a steady glide approach down to the 50-ft height and at a calibrated airspeed close to 1.4 times the stall speed of the airplane.

2. The time elapsed between the 50-ft height and touchdown is not less than 7 seconds.

3. The landing is made on a level, smooth, hard-surfaced runway.

4. If demonstrated performance credit is desired for wet runways, the wheels are fitted with tires that have been worn to a point where no more than 20% of the original tread remains.

The antiskid system is required to cope with touchdown dynamics, varying loads on the main wheels and gear dynamics such as gear walk, shimmy, and truck pitching. The preceding tests on simulator and dynamometer should minimize as much as possible the risk of testing on the airplane. During the airplane tests the antiskid system should demonstrate normal operation during the braking run. Operation should indicate minimal depth of skids in the observation of the wheel-speed data. Brake pressure modulation should always try to search for the maximum available tire-runway friction coefficient. The antiskid system should not allow a locked-wheel condition and bring the airplane to a full stop when the pilot applies maximum braking effort.

Table 1 is indicative of the types of parameters which are required for antiskid simulation but not inclusive. It is not a comprehensive list of required parameters and should not be used as such. Depending upon the fidelity of the simulation, considerably more parameters may be required than are shown in the table. The exact parameter requirements will depend upon, and be tightly coupled with, the targetted simulation.

TABLE 1 - AIRPLANE AND LANDING GEAR PARAMETERS

	Definition	Unit
Airplane Parameters	Effective wing area	ft^2
	Drag coefficient	-
	Drag coefficient without spoilers	-
	Lift coefficient	-
	Lift coefficient without spoilers	-
	Engine idle thrust at zero velocity	lbf
	Height of CG above ground	ft
	Mass moment of inertia, pitch	ft-lbf-s^2
	Change of idle thrust with velocity	lbf-s/ft
	Nose gear to CG distance	ft
	Main gear to CG distance	ft
	Number of brakes per main strut	-
	Number of main gear brakes per airplane	-
	Number of nose gear wheels	-
	Number of main gear struts per airplane	-
	Mass density of air	lbf-s^2/ft^4
	Initial airplane velocity	ft/s
	Final airplane velocity	ft/s
	Weight of airplane	lbf
Brake Parameters	Torque peaking gain	(ft-lbf/psi)/(rad/s)
	Mass of brake heat sink	lbf-s^2/ft
	Wheel velocity at start of torque peaking	rad/s
	Retractor spring pressure	psi
	Torque gain	ft-lbf/psi
	Temperature at initiation of fade	°F
	Natural frequency of torque response	Hz
	Damping ratio of torque response	
	Brake stiffness	
	Dynamometer test results for various conditions	
Hydraulic Parameters	Line lengths, diameters, and material	
	Flex hose length and diameters	
	Fittings, swivels, etc.	
	Valves	
	Schematic of hydraulic system	
Tire Parameters	Tire diameter	in
	Tire deflection	in
	Mass moment of inertia of tire, wheel, and brake	ft-lbf-s^2
	Tire operating inflation pressure	psi
	Tire rated inflation pressure	psi
	Tire rolling radius	ft
	Tire torque radius	ft
Strut Parameters	Main gear vertical damping coefficient	lbf-s/ft
	Nose gear vertical damping coefficient	lbf-s/ft
	Main gear fore-aft damping coefficient	lbf-s/ft
	Torsional damping between strut and brake	lbf-ft-s
	Mass moment of inertia of main gear strut	ft-lbf-s^2
	Main gear vertical stiffness	lbf/ft
	Nose gear vertical stiffness	lbf/ft
	Main gear fore-aft stiffness	lbf/ft
	Effective strut length	ft
	Effective mass of strut	lbf-s^2/ft

7. SYSTEM ISSUES

During design of an aircraft, design choices are often made that significantly affect the brake control system. This section is included to make sure that the effects of these choices on brake control are included when considering the trade offs with other aircraft needs.

7.1 Actuator/System Response

Generally the targeted frequency response of modern brake control systems is 10 Hz or higher, measured from the antiskid or brake control command output to pressure at the brake bleed port (or brake piston clamping force) The frequency called out here is the point where the phase on a Bode plot crosses 90 degrees. In any control system actuator response is critical to the performance of the system. This is very much the case for antiskid control systems. It is not at all uncommon for aircraft space availability considerations to drive antiskid system components into areas that result in suboptimal response for antiskid control. Cost and reliability considerations may also lead to design decisions resulting in slow response

Slow frequency response will almost certainly lower the achievable antiskid efficiency of the system, thus increasing stopping distances. The ability to obtain high antiskid efficiency while maintaining gear stability (avoiding gear walk) may also be compromised. It will also limit the ability of the brake control supplier to adjust the control to deal with unforeseen system issues.

System design is a process of trade-offs and compromises which may drive the design toward slower antiskid system response. However, these design decisions should be made in full awareness of their possible affect on antiskid control.

7.1.1 Long Hydraulic Lines

Long hydraulic lines present a number of issues which all slow the step and frequency response of the system. These include:

- Water hammer
- Increased flow resistance
- Increased compliance
- Transport delay

7.1.1.1 Water Hammer When the Pistons Contact the Brake Stack

In a system with very long lines water hammer may be an issue. Figure 27 shows a typical brake displacement curve.

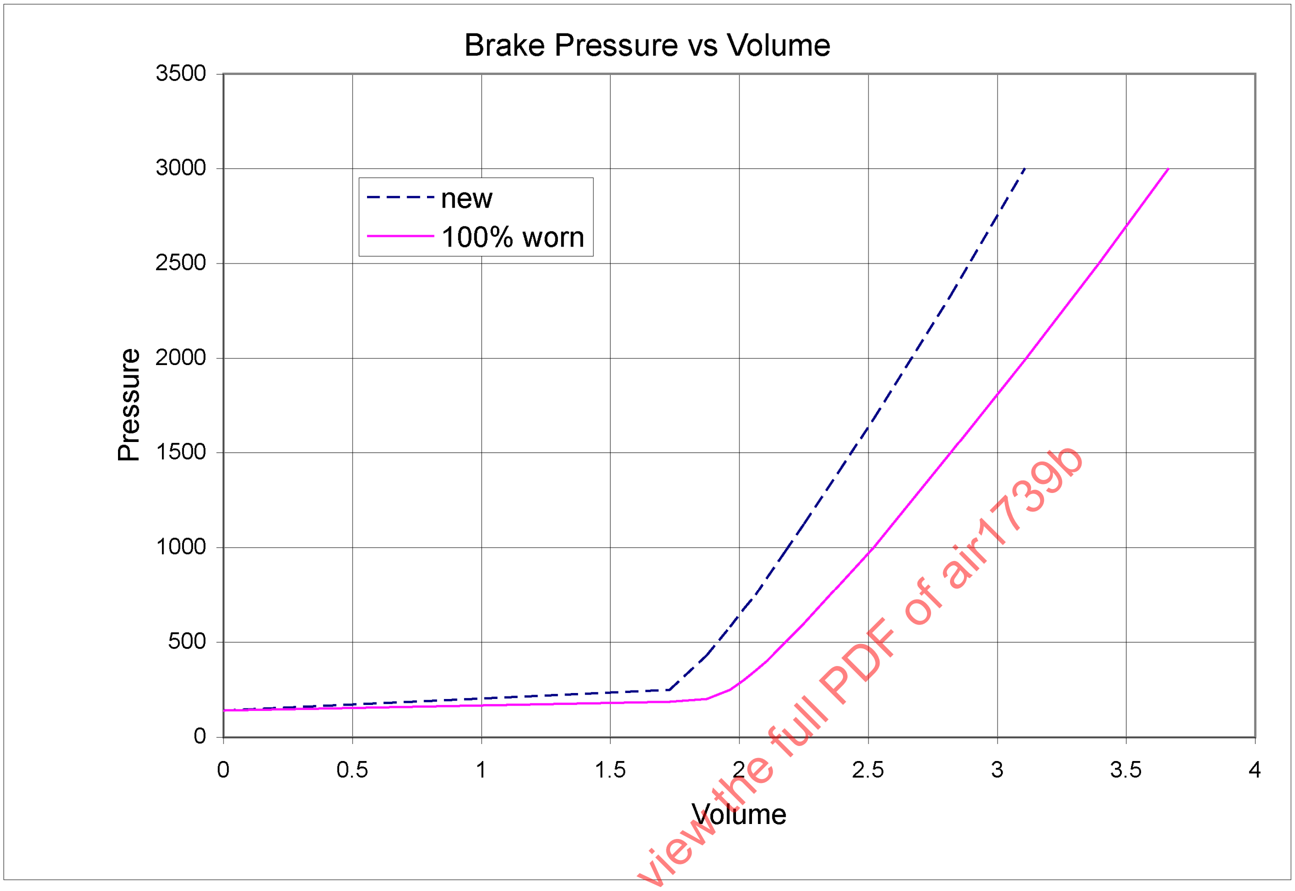


FIGURE 27 - BRAKE DISPLACEMENT CURVE

Prior to contacting the stack of rotors and stators, only a light spring opposes the pressure of the fluid. This may be seen in Figure 28.

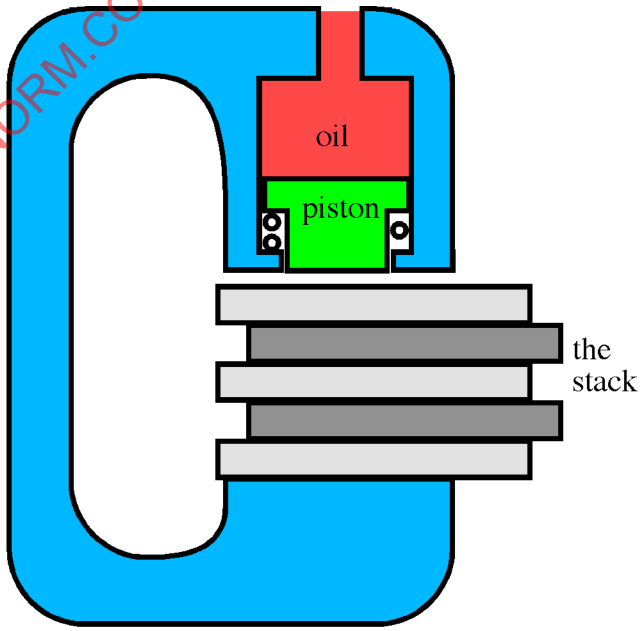


FIGURE 28 - BRAKE PISTON AND STACK

To fill the brake within a reasonable amount of time, flow must be high. If the lines are long the mass of fluid in motion is substantial. When the piston contacts the stack the stiffness goes up by an order of magnitude or more. The result can be water hammer. The momentum of the fluid continues the rush of fluid into the brake, compressing the fluid and causing pressure to spike. As a result a wave of pressure will be reflected back up the line toward the valve. Figure 29 shows water hammer using lumped masses and multiple springs as analogs for the fluid mass and compressibility.

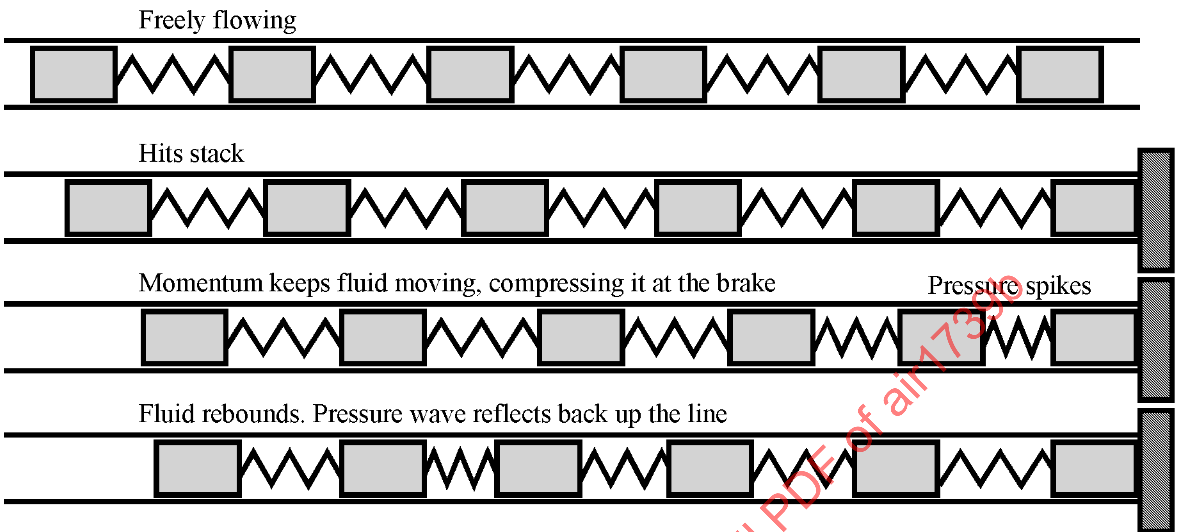


FIGURE 29 - WATER HAMMER

This pressure surge is problematic. It can cause a deep skid. It may be felt by the passengers and crew. If it is asymmetric, particularly on an aircraft with only two wheels and a large track (lateral distance between the wheels) it can cause controllability issues. Fill algorithms are often used to try to limit water hammer but they have limitations. The physics of the situation and the need to fill the brake quickly make water hammer very difficult to avoid. The rate of flow for a given delta pressure varies considerably at different temperatures. Thus an algorithm based solely on timing will not achieve its desired result under all conditions. Airplanes often have brake line lengths that vary considerably from brake to brake. The timing that works on for the longest line may not work for the shortest.

7.1.1.2 Excessive Resistance to Flow

Long lines increase the resistance to flow. It takes a larger delta in pressure between the valve and the brake to drive the same amount of flow as a system with shorter lines. This slows both the step and frequency response of the system making the job of brake control more difficult.

7.1.1.3 Excessive Hydraulic Compliance

Long lines also increase the compliance of the system due both to increased fluid volume and added line material with its own compliance. This means that it takes more fluid to fill the line and brake to the same pressure than it would with shorter lines. This slows both the step and frequency response of the system making the job of brake control more difficult. Because of this compliance effect, increasing the diameter of the line to reduce resistance to flow is only effective up to the point where the reduced response due to additional compliance overcomes the increased response due to reduced resistance to flow.

7.1.1.4 Transport Delay

A pressure wave moving through a hydraulic line moves at the speed of sound in that line, which is:

$$c = \sqrt{\frac{B}{\rho}} \qquad \text{(Eq.23)}$$

$$\frac{1}{B} = \frac{1}{B_t} + \frac{1}{B_f} \quad \text{(Eq.24)}$$

c = Speed of sound

ρ = Fluid density

B = Bulk modulus of the line

B_t = Bulk modulus of the tube (depends on the diameter, thickness, and material of the tube)

B_f = Bulk modulus of the fluid (including the effect of entrained air)

The time for a pressure wave to travel down a twenty foot long, quarter inch diameter brake line (including two feet of flex line) is about 0.04 seconds. This delay is large enough to have a significant effect on the total frequency response of the system.

7.1.2 Line Diameter

Line diameter should be optimized for the required flow rates. Line diameters that are too small will restrict the flow, lowering response. Line diameters that are too large will increase the total compliance of the system, lowering response. Thus there must be a balance. Smaller is not necessarilly better. Larger is not necessarily better.

7.1.3 Restrictions in Brake Lines

If the diameters of the passages through fittings, swivels, fuses, and other components in the brake line are smaller than the inner diameter of the line, the resistance to flow can be significantly increased, slowing both the step and frequency response of the system.

7.1.4 Flex Lines

Flex lines typically are much more compliant than hard lines. Long flex lines can significantly increase the compliance of the system.

7.1.5 Brakes with Excessive Compliance

The brake is almost always the predominant contributor to the total compliance of the system. Too compliant a brake may make adequate step and frequency response impossible to achieve.

7.1.6 Air in the System/Low Return Pressure

At low pressures excessive amounts of air in the system can significantly lower the bulk modulus of the hydraulic fluid, lowering the response of the system. Since the bulk modulus of air is approximately equal to the pressure level, the effect on fluid bulk modulus of entrained air is much greater at low pressures. At high pressures air goes into solution and has less effect on the total bulk modulus. As a rough rule of thumb, the transition between entrained air and air in solution occurs at about 1000 psi, although the exact pressure will depend upon the fluid and the temperature.

Systems that operate at low return pressures and systems with brakes where piston contact with the brake stack occurs at low pressures will be more likely to have to operate at pressures low enough for entrained air to significantly affect response, particularly for stops on wet and icy runways.

7.1.7 Hydraulic Line Bends and Routing

Excessive numbers of bends can increase the resistance to flow. In addition, attention should be paid to the routing of lines, particularly deadheaded rising lines, due to the possibility of trapping air in the system.

7.1.8 Valves With Inadequate Flow Capacity

The valve must be sized correctly to drive levels of flow through the system commensurate with the total compliance of the system, both lines and brake.

7.1.9 Inadequate Hydraulic Supply

Limited supply flow capacity can limit brake system response, particularly during brake fill.

7.1.10 Inadequate Hydraulic Return

If the return lines are not adequate to keep return pressure below the piston stack contact pressure, the antiskid system will be unable to fully release the brake and there may be enough pressure to lock the wheel even though the antiskid system is commanding zero pressure. This problem is exacerbated when devices outside of the brake system are active during brake control and share the same return line as brake control.

7.1.11 Poor Brake Metering Valve Response

Metering valves often have inadequate flow capacity and response. Metering valve design must take into account the flow modulation that will take place during antiskid activity. They must also take into account flow during intial brake application.

7.1.12 Contaminated Hydraulic Fluid

Contaminated hydraulic fluid can cause numerous system failures. The most vulnerable point in the brake control system is the first stage of the brake control valve. The clearance between the flapper and the nozzles is on the order of only several thousandths to ten thousandths of an inch, roughly the diameter of a human hair or thickness of a piece of paper. A particle trapped between the flapper and the nozzle is shown in Figure 30.

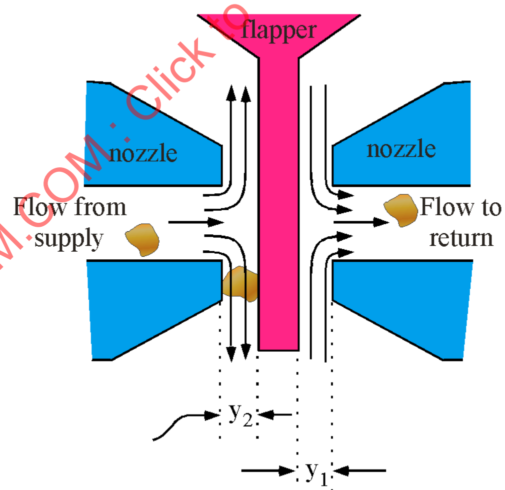


FIGURE 30 - FLAPPER-NOZZLE CONTAMINATION

A piece of contaminant lodged between the flapper and a nozzle can cause the valve to be unable to achieve full pressure or to be unable to dump to return. It can also cause inconsistent behavior as successive pieces of contaminant with varying amounts of compliance become lodged between the flapper and the nozzle and then are freed by commands to the flapper.

7.1.13 Electric Actuator Response

The trade offs resulting from slow antiskid system response described in Section 7.1 apply to electric brakes as well as hydraulic brakes. The response of an electric brake should be measured from the antiskid command to clamping force at the brake. This will include the transport delay due to data transmission through the various buses in the system as well as the response of the actuator itself.