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Superseding AIR1064C

(R) Braking System Dynamics

RATIONALE

AIR1064D has been reaffirmed to comply with the SAE five-year review policy.

FOREWORD

Revisions from Revision C to Revision D are major. The title has been changed from "brake" to "braking system" dynamics to reflect primarily the broader treatment of antiskid issues involved. More material has been added in this Revision regarding identification and diagnosis of vibration of landing gear system components. Appendices have also been added for relevant documents and patents. It should be noted by readers unfamiliar with the different kinds of SAE documents that this document is an Aerospace Information Report (AIR), and not an Aerospace Recommended Practice (ARP). As such it represents the combined information and experience of many practitioners, whose opinions and practices are different. Differences of opinion are noted, and the review process is expected to help assure they are fairly represented without prejudice. There is much information that can only be alluded to generally because many of the participants in this process are business competitors in one or more fields, and issues of proprietary intellectual property must be respected.

Many "rules of thumb" that are discussed herein have a built in caveat, namely that ours is an ever-changing field, where such rules are often broken. What may seem logical and intuitively true today may be discovered to be at least debatable when newer methods of testing and analysis are employed by engineers who will continue our endeavors to solve the problems and refine the solutions of the future.

1. SCOPE

The aircraft landing gear is a complex multi-degree of freedom dynamic system, and may encounter vibration or dynamic response problems induced by braking action. The vibratory modes can be induced by brake and tire-ground frictional characteristics, antiskid operation, brake design features, landing gear design features, and tire characteristics. The impact of this vibration can range from catastrophic failure of critical system components or entire landing gears, to fatigue of small components, to passenger annoyance. It is therefore important that the vibration is assessed during the design concept phase, and verified during the development and testing phases of the system hardware.

This SAE Aerospace Information Report (AIR) has been prepared by a panel of the A-5A Subcommittee to present an overview of the landing gear problems associated with aircraft braking system dynamics, and the approaches to the identification, diagnosis, and solution of these problems. All pertinent system modes of vibration are described. In addition, facilities and techniques available for test and evaluation are presented and discussed, and useful references are cited. The terminology used is intended to be consistent with AIR1489, “Aerospace Landing Gear Systems Terminology”, but some terminology herein is not yet included in AIR1489. The panel members include representatives from major brake, landing gear, aircraft, and brake control system manufacturers. In addition, drafts of the document were circulated for input beyond the SAE to other experts in the field.

2. APPLICABLE DOCUMENTS

The references included in the appendices of this document are not necessarily cited in the text. They are included as extensions of the text for the sake of providing a more comprehensive source of information on braking system dynamics than this document can provide. In all cases, 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. Patents cited may be discussed in the text of this document inasmuch as they may apply to one subject area or another. Preliminary searches have shown that the list of papers is quite lengthy, and many papers contain exhaustive bibliographies. It is not the intent of this AIR to reproduce all such bibliographies, nor to cite the documents listed. The papers and patents listed herein are not intended to be exhaustive, but should rather be regarded as representative of the documents available on the subject of this AIR.

2.1 SAE Publications are listed in Appendix A of this AIR.

2.2 Other Useful Publications are listed in Appendix B of this AIR.

2.3 Applicable Patents are listed in Appendix C of this AIR. Patents pending are not listed.

3. GEAR/BRAKE SYSTEM MODES INFLUENCED BY BRAKING

There are several modes of vibration of the aircraft, landing gear, and brake which can be affected by braking and anti-skid action. It is not uncommon for most of these modes to be described as “chatter”, “growl”, “roar”, “screech”, and other terms descriptive of the sounds they may make. It is preferable, however, to refer to them by terminology standards established in AIR1489. Many of these modes cannot be identified by sound, because the frequency ranges of some are often the same. Consequently proper diagnosis requires a combination of test and analysis discussed in Sections 8 and 9. Some of the modes described below are NOT yet listed in AIR1489. They are not discussed in order of their typical vibration frequencies.

3.1 Gear Walk: a fore-aft pitch-plane motion of the landing gear relative to the aircraft

In large aircraft and gears, the fundamental walk mode occurs in the range of 5 to 10 Hz. In smaller regional and business jets, it is in the range of 20 to 30 Hz. There are symmetric modes, in which both left and right gears move in phase, and anti-symmetric ones, in which they move out of phase. Both are governed by the gear and its support structure compliance in pitch-torsion. The support structure can include the fuselage. Some investigators have found that the portions of the fuselage forward and aft of the wing are more involved in symmetric modes, which lower their walk frequencies, with the logic that the fuselage is a node in antisymmetric modes, so affords no compliance to the gear modes. This is not always true however. The fundamental walk mode on the L1011 was found to be an antisymmetric one. In any case, the strut's bending elasticity is a secondary factor in most such modes. It should be noted that all aircraft have many walk modes, since fore-aft motion of the gear can occur during many other dynamic modes of the aircraft. It is important to discern which of all of those modes is the fundamental gear walk mode most likely to be excited by the braking system. That discernment demands insight and skill from the analysts and engineers.

One of the ways to identify the dominant gear mode is a normal mode analysis of the undamped system, and the rationale for selection is based on what percentage of modal energy resides in the gear mass for each mode. A mode with 5% energy is much more difficult to excite by the brakes than one with 90% energy in the gear. For example, the effective gear mass in a 90% mode is the actual gear mass divided by 0.9, hence representing all of the energy of the system in the gear that is simulated on the dynamometer. This was the method used by Lockheed for identifying the walk mode of the L-1011 in the 1970s.

Historically, it was believed that the minimum damping associated with such modes was on the order of 5 to 7% critical, but in the 1990s, measurements on T-geared and trailing-link geared aircraft clarified that the damping is in fact lower and possibly nonlinear. On one T-gear aircraft the apparent damping was very low at low levels of displacement, and increased to 3 to 4% as the amplitude of the gear walk increased. Such behavior can be expected in pin-jointed structures with small clearances in their joints, where rigid body motion transitions to motion with a greater proportion of strain energy as deflection amplitudes increase. It is important to consider sources of amplitude-dependent damping when designing landing gear, and when conducting dynamometer tests for gear walk stability margins.

The damping is believed to be higher for bogie gears. It was measured to be 5 to 7% on the L1011 in the 1970s. Multi-axle gears have greater walk-mode damping because of the contribution of friction at the joints of the linkages required to react brake torque between the brakes and aircraft through their bogey beams. The measured higher damping of bogie gear walk alone is enough to regard link-less gears to have lower walk damping than 5%. Because of the high damping of bogie gear walk, the walk mode in multi-axle gears is regarded by some dynamicists as only a minor concern for brake-induced vibration. It remains a concern for anti-skid system engineers however, because they are generally concerned with transient response, and not dynamic stability.

Note that gear walk is called "judder" in some circles (Britain), but the preferred terminology is "walk", according to AIR1489.

3.2 Aircraft Pitch: a "porpoising" motion of the entire aircraft under sudden braking or spoiler action

The effect of this motion is to pitch the aircraft forward and offload the braked landing gears, which reduces the traction available to antiskid systems. It is a concern to antiskid engineering because this mode occurs at less than 5 Hz (often less than 1 Hz), which is within the range of their operating bandwidth, and the transient action is initiated at high speed when brakes are first applied and stopping distance is most affected. Part of the concern is maintaining passenger comfort with smooth braking. Some practitioners do not regard this as a transient or "sudden" effect at all, because the pitch moment is proportional to braking drag force, which is continually modulated by antiskid cycling. This can be a low-damped mode and continue for as long as brakes are applied.

3.3 Aircraft Bounce: a vertical motion of the entire aircraft during the landing transient before full and stable weight on wheels is achieved

This is primarily a concern to anti-skid engineers because it affects available traction at high speeds. The frequency range is also below 5 Hz, and can be less than 1 Hz. Except in the rare case of perfect 3-point landings, initial transients are a rich mixture of pitch and bounce. The brake control system must cope with this mixture and the tire load variations associated with it.

3.4 Bogie Pitch: a “porpoising” motion of the bogie of a multi-axle gear relative to the aircraft

Such gears are used only on large heavy aircraft, and the mode typically occurs between 5 and 15 Hz. Again, this is within the range of concern for antiskid engineers. It is a unique scenario because it can be affected by runway profiles. Tire loads fore and aft on the same gear vary out of phase during bogie pitching, along with tire radii and therefore transients in wheel speed. Antiskid engineers must be cognizant of this. It is also a concern regarding pivot joint frictional overheating on rough runways.

3.5 Wheel Chatter: a torsional oscillation of the wheel against the elasticity of the tire in the pitch-plane of motion

This motion has a natural frequency between 25 and 50 Hz in large aircraft, and can lead to violent shaking of the aircraft. Experience has shown it can occur under extreme brake temperatures and at very low ground speeds (typically less than 5 mph). It can pose structural fatigue concerns if encountered often, and it is prudent to assess the possibility of its occurrence during dynamometer tests.

3.6 Brake Whirl: a rotating bending motion of the axle out-of-phase with the brake torque tube

This mode of vibration became a major concern with the advent of carbon brakes, because of their light disk weight, low gyroscopic resistance to out-of-plane motion, and low damping relative to the segmented linkage design of metallic brake disks. One of the characteristics of classical brake whirl in hydraulic brakes is that the rotating bending causes a brake fluid pressure wave to circulate in the brake housing. Consequently, one of the remedies for whirl has been to incorporate orifices in the housings to absorb energy by means of turbulent losses as the fluid traveled from piston to piston around the housing. Different designers have used different approaches to address the fluid pressure waves evident during whirl. Some have used blocks to prevent fluid circulation. Others have used a variety of orifice sizes to add damping to the whirl mode while still allowing rapid response for antiskid pressure modulation. Still others use a single orifice size.

It is commonly accepted that the initiation of brake whirl normally requires a cyclically asymmetric, and hence unbalanced, application of clamping force to the brake disk stack. One source of such asymmetric force occurs in hydraulic brakes naturally because the pistons are fed fluid sequentially from the inlet. Another source is axle deflection, which is discussed below. Be aware that cyclic asymmetry is not a necessary condition for whirl to occur, because any disturbance large enough to change the state of the system to an unstable region can initiate the motion. Regardless of origin, the result of unbalanced clamping forces is that the resulting unbalanced frictional forces act to deflect the system radially 90 degrees away from the unbalance, which sets up yet another unbalance which chases itself around as a “follower force” that drives the whirl motion.

The recent advent of electrically actuated brakes has posed questions for brake designers about their options for preventing whirl in the absence of fluid circulation orifices. Some engineers are concerned that orifices are unavailable as a whirl weapon. Others are of the opinion that orifices are required only because fluid circulation exists, and they are the only way to restrict passing the whirl disturbance from piston to piston. It remains to be seen whether whirl will be a concern with electric brakes. Electric actuators do not act as fluid pistons, and cannot relay force variations around the “housing” at the rate necessary to sustain or reinforce whirl vibrations in the range of 200 Hz. They may have other idiosyncrasies, however, which will come to light during their evolution as an alternate to hydraulic actuation.

Regardless of actuation mechanics, it is important to realize that clamping forces are always unbalanced in multi-disk aircraft brakes, because the wheel is located further outboard on the cantilevered axle than the brake is. Consequently both normal and drag loads acting to deflect the axle have the effect of misaligning the rotating and non-rotating parts of the brake. Axial friction forces between the wheel and brake rotors are continuously acting to force the rotors inboard near 12 o’clock and outboard near 6 o’clock.

3.7 Brake Squeal: historically defined as a pitch-plane torsional oscillation of the brake against the elasticity of its torque-reaction interface with the landing gear

In a flange-mounted brake, such as the 727 and 737 and A320, the torque is transferred to the gear and aircraft through the axle, and the elasticity that governs the squeal mode is the torsional windup of the axle. In such systems there is only one fundamental squeal mode. A 2nd order squeal mode exists, defined as a situation in which the backplate of the brake oscillates out-of-phase with the piston housing, but it is rarely a concern.

In multi-axle brake systems, there are multiple squeal modes, because the brakes interact with each other in such gears through the linkages that join them, and they can do so in more than one way. This is illustrated schematically in Figure 1, representing 5 squeal modes of a 2-axle, 4-brake bogie gear, in which the brake torque is reacted through their brake rods at the inner cylinder in such a way that the fore and aft brake rods share a common pin-joint. The 5 modes are differentiated by which brakes oscillate in-phase with one another and which oscillate out-of phase. Different combinations are also differentiated by the degree to which components of the landing gear participate in the elastic restraint to the brake motion. The manner in which energy transfer and energy-sharing influences which landing gear structural elements are involved in a given mode is not perfectly understood. Since most test sites accommodate only one brake, it is important to identify which of these modes is likely to have the lowest stability margin so that test efforts can be dedicated to that one as the worst-case. The rationale may vary from one supplier to another, but it is wise to test the worst case if it can be justified, because it reduces all costs, and shortens schedules. It is incumbent upon the brake supplier to convince the customer of the validity of his methods.

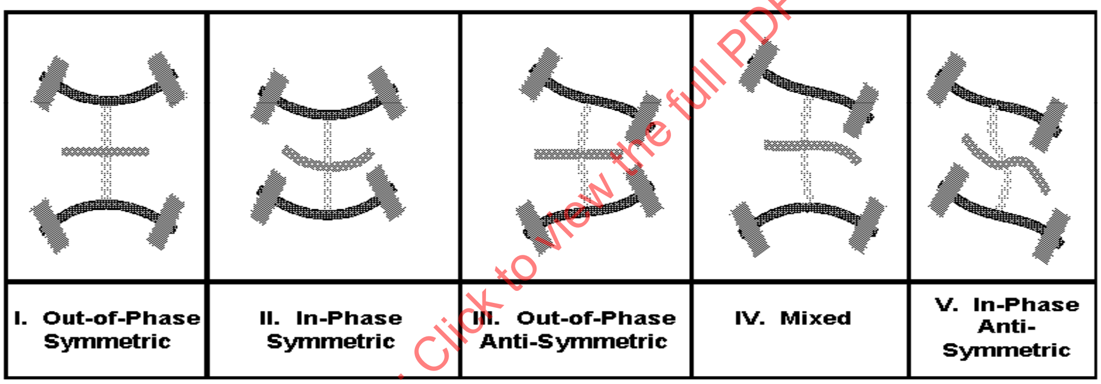

FIGURE 1 - FIVE BRAKE SQUEAL MODES IN A BOGIE GEAR

The elastic compliance anchoring the inertia of such brakes in a squeal mode must be the series combination of the pitch-plane housing deflection, the rod extension, and the fixture to which the rod is attached to ground. It is NOT the rod alone. In a series combination, the stiffness is dominated by the most compliant member. Hence one should be aware that in some cases, perhaps mostly in dynamometer fixtures, the squeal vibration frequency may not be dominated by the rod.

As a caveat to all of the above discussion, it should be realized that any such assumptions that fundamental modes are least stable, for instance, are never hard and fast. Some dynamicists remain convinced that stability cannot be inferred from mode shapes and related sources of available damping. Nevertheless, it remains an important issue to identify the worst case situations in multi-brake systems, because it will always be a priority for the brake designer to minimize the test and analysis effort required to provide a brake with optimum dynamic stability, and justify his approach to the customer.

3.8 Rod-Bending Modes in Multi-Axle Gears

Rod bending has been observed to accompany brake squeal in multi-axle landing gears. The alternating compression or tension forces imposed on the rod by brake torsion oscillations excite the rod to vibrate in bending. Since the rods can have many bending modes, it is not surprising that one or another can participate in the fundamental brake squeal mode. Some analytical modeling has shown that the bending stiffness of the rod does not affect the mode shape of the brake, but that the associated bending participation can affect the mode’s stability: when rod bending is prevented, the stability of the associated brake squeal modes increases 10-fold. Other analytical studies have indicated that such vibration can be chaotic in nature. The unique role of torque reaction by means of a brake rod is that the rod dynamics enter the system in a significant manner, and bending vibration is often the highest vibration recorded during brake squeal events. Proper assessment of the system modes should include data from the rod and the fixture it is attached to. Rod bending under oscillating axial loading is described best by Mathieu equations, which show many frequency regions of alternating stability and instability, which help to understand how instability is not necessarily associated with the most fundamental mode.

Care should be taken in evaluation of the interaction between brake and rod, as this situation has the potential for other (non-squeal) brake modes at higher frequencies in which the dominant motion can become localized in either the forward or the aft positions.

3.9 Pitch and Lift of Trailing-Link Gears

Brake torque acts to raise the axle of trailing-link gears against the elastic restraint of the shock strut. This mode is often in the range of the gear walk natural frequency, but is more heavily damped due to high shock strut damping. Nevertheless, the axle lift causes the tire to temporarily offload while the aircraft settles, and available traction for anti-skid can be reduced, which can trigger a skid unjustifiably at brake pressures well below that which can otherwise be sustained. This effect is very brief, but not so brief that the anti-skid ignores it. The magnitude of the tire force offloading is related to the rate of brake application. There is in addition a more sustained effect, which shifts the operating point of the oleo up its load/stroke curve to a higher spring rate, at which it is regarded by some researchers to degrade its dynamic stability. This mode and its effect are unique to these gear designs.

3.10 Shimmy

This gear mode, involving lateral and yaw oscillation of the gear is normally a concern for nose gears, but it is possible in main gears as well. Ordinarily, this is not a major dynamic stability problem during braking, but it is of some concern to brake control system designers. According to some investigators, such a strut torsional mode of vibration can be excited during a pivoting turn, and may be induced or aggravated by a combination of brake torque and tire slippage.

3.11 Rotor Cycloidal Vibration

Rotor cycloidal vibration is classified as the vibration of internal rotors having radial motion which can be characterized as epicycloidal, which is, the disks roll circumferentially, bounded by their dead-band, and follow a circular path within the wheel. Those dead-bands come mainly from the wear ledge on the friction surfaces and design tolerances. It occurs only at the low pressure which allows stack breathing motion coupled with whirl, and is present only on worn brakes. See 3.20. This mode can be avoided by designing the brake disks with “wear grooves” to prevent radial interference as the disks wear.

Typical rotor cycloidal vibration can be detected from vibration measurements, and usually consists of two stages: The first stage is only a few fraction of a second consisting of a wide band noise from a series of repeated hammer impacts. The second stage is a self-sustained vibration, longer in duration, at frequencies ranging from 100 Hz for large brakes to 150 Hz for smaller ones.

3.12 Disk-Flexural Modes

In integral brake disks, such as carbon or ceramic constructions, traveling and standing waves can be excited by frictional forces, and are inherently unstable, in the sense that negative damping is not required for their natural excitation. The stability of such modes depends on the internal damping of their materials and construction, and on the boundary conditions of the brake design, such as numbers of splines on their peripheries and axial clamping locations. The lowest order mode of such disks is a 2-diameter flexural mode and is often in the same frequency range as brake assembly modes such as squeal and whirl. There are an infinite number of these modes, and the higher frequency ones can be amplified by nearby reverberant structures such as the drum-like structure of the wheel.

3.13 Other Gear Modes

There are some large aircraft that have body-mounted gears as well as wing-mounted gears. The walk modes are different, and the engineers tasked with determining critical modes must account for both situations. There are also some landing gears which have unique designs, and can have other combinations or variations of the modes described above because of the unconventional ways used to transfer brake torque and drag forces to the gear. The KC-135 and C-17 are examples of such gears. Others are described in AIR1489.

3.14 Other Brake Modes in Electric Brakes

The actuation of electric brakes is accomplished by relatively large and heavy motor/gear assemblies mounted on a plate that replaces the more traditional piston housing. If design economy dictates that no more than four of these assemblies can be used, it leads to the expectation that such designs may exhibit modes similar to 2-diameter disk modes, due to the lumped masses of the motor/gear assemblies. They may also exhibit modes due to the actuators themselves, which may resonate with more traditional brake modes.

3.15 Coupling Between Modes

It is not unusual for some of the modes described above to couple with others. For instance, because they share the same frequency ranges and excitation sources, squeal and whirl modes are often seen to coexist in what some dynamists call a “squirl” mode. At the same time, disk modes can also be inferred to participate with both, but cannot be as easily diagnosed because mode shapes cannot be directly measured during operation. Suffice it to say that coupling can exist, and proper diagnosis is an artful application of analytical and experimental techniques. It is rare to expect that any vibration mode in any braking system is clearly described by only one of the many described above or below without influence or participation by any other.

3.16 Energy Sharing Between Pitch-Plane Modes

One example not well understood is the sharing of energy that occurs in single-axle landing gear with steel brakes, where squeal and gear walk modes blossom and die out of phase to each other sharing the available energy in cycles. Another example, presented at an International ADAMS User Conference, is that the relative stability of walk, squeal, and chatter modes depends on their relative damping, and when the damping of one mode is increased, it can reduce the stability of the others due to the redistribution of available energy.

3.17 High Frequency Brake Vibration

Vibrations in the range of 2 to 20 KHz have small displacements, but can occur at acceleration amplitudes greater than 500 g at the piston housing. They are not a threat to the landing gear, but can be loud and can lead to fatigue failure of small parts such as fasteners, fittings, and connectors.

3.18 Aircraft Yaw

The aircraft may experience yaw from unintended differential braking, especially at brake onset. Yaw oscillations are not an uncommon result. One cause may be the statistical variation in friction produced by the brakes. Another may be that brake response can be affected by differences in left-right hydraulic symmetry.

3.19 Wheel Order Related Vibration

This type of vibration will be manifested as a linear function of wheel speed. As wheel speed decreases, its vibration frequencies decrease. Although steel brakes are more prone to this type of vibration because of the obvious discontinuities in the disk construction, carbon brakes are not immune. The source in carbon brakes is related to the waviness of the friction surfaces, wheel ovalization, non-uniform engagement of rotors, and unevenness of the tire and its wear-dependent dimensions. It is also related to cyclic symmetry of the rotor properties or boundary conditions, such as number of wheel splines engaged to transfer torque.

3.20 Axial Brake Modes of Vibration

This vibration is frequently associated with other modes because it serves mainly to modulate the clamping force on the friction surfaces, which serves to excite other modes. The brake has many axial modes of vibration, ranging from pure axial motion of the entire heat stack (all rotors and stators) against the elastic restraint of the Belleville-spring action of the housing and backplate to others which involve axial wave motion that can result in a sequence of contact pressure changes from one friction pair to another. It is difficult to ascertain which mode is active when it occurs, because empirical verification requires instrumentation at both ends of the brake (which is problematic because of the limitations of temperature and space at the backplate). Some researchers call the axial traveling mode a “breathing” mode because of its similarity to the out-of-phase action of an accordion, and associate it with rotor-cycloidal vibration.

3.21 Rotor-Wheel Whirl and Squeal

These modes can occur in brakes whose rotors transmit torque to the wheel through “torque bars” which are fixed to the wheel only at their ends. As such, they can flex relative to the wheel both radially and tangentially. The result is a degree of freedom wherein the rotor inertias can move relative to the wheel. Unlike brake whirl, there is no natural out-of-plane motion involved. The frequencies are between 200 and 1000 Hz. These modes do not exist in wheels with integral splines.

4. EXCITATION SOURCES

4.1 Self-Excitation by the Frictional Characteristics of the Brake

These problems generally are more pronounced with steel heat sink brakes with lining material containing mixtures of ceramic and metallic particles or plain metal particle lining materials. They exhibit a variation of the coefficient of friction with instantaneous slip velocity which can result in a dynamic condition of "negative damping", which can reduce dynamic stability or even render the system unstable. The condition of negative damping is real and is proportional to the negative slope of the brake’s torque-speed function. Negative damping in metallic brakes is caused by constituents in the brake lining that form compounds on the friction surface, and it is often temperature dependent. Carbon on carbon friction surfaces can also exhibit such a speed dependency, but it is usually much less pronounced. Little has been documented about such tendencies in other friction materials, such as ceramics or cermets. It is important to recognize that the sensitivity of friction to speed is not a straightforward characteristic to measure, because friction is also sensitive to pressure and especially temperature, which changes at the same time as speed. On a micro scale, the friction as affected by factors such as the adhesion force, roughness, interference, friction film, and other environmental factors. The effective rate of change of friction with speed is further affected by the rate at which the speed is changing, and during an oscillation at a high rate the sensitivity will be different than at a lower rate. Hence, it is possible that higher slopes may exist during chatter-mode oscillations than during, say, walk-mode oscillations. Care must be taken when defining these characteristics in order to assess the potential for destabilization from this cause, because speed dependence can be masked by the other factors cited. Lastly, it should be noted that measurement of negative damping is often problematic because compounds on the brake lining surface that influence speed sensitivity can change during a braking event.

4.2 Speed-Dependent Order-Forced Oscillation

The most basic of these excitations is a once per wheel revolution change in force or wheel speed caused by tire flat-spotting or unequal brake rotor stack thickness. But higher order forcing possibilities exist. These sources of vibration arise because cyclically discontinuous features of the brake can cause torque changes independent of the friction characteristics and can cause resonant responses in many system modes.

This is more of a concern with metallic heat-sink construction, which requires thermal expansion slot discontinuities and segmented disk constructions to prevent thermal distortion problems. In the extreme, it is easy to understand that if a stator had say 6 thermal expansion slots, and the rotor it mated with had 6 segments such a brake would have 6 friction/torque discontinuities for every revolution of the wheel, and hence would exhibit a 6/rev excitation function, which could serve to excite every mode of the system with a natural frequency within 6 times that brake's operating (angular) speed range.

These potential problems may be controlled by proper detail design of the brake, guided by suitable analysis and test. One of the rules of thumb has historically been to avoid using cyclic symmetry of the same order or multiples in the brake design. Odd numbers and prime numbers have been preferred. However, that rule of thumb has been broken often, and may be challenged further with the advent of electric braking systems, which are optimally configured to use fewer numbers of actuators. A brake with the same number of discrete actuators, rotor-wheel splines, and stator-torque tube splines is more likely to exhibit vibration attributable to this cause.

4.3 Whirl Excitation

Whirl has been associated with an initial excitation and eccentricity between the rotating and nonrotating parts of the brake. The eccentricity arises naturally in the brake because the wheel (and the rotors keyed to it) rotate in a non-vertical plane prior to brake application because the wheel is mounted further outboard than the brake on the cantilever beam of the axle. When braking stack clamping forces are applied, the rotors are forced into alignment with the brake. During the transition in this angular alignment, there is a greater force at 12 o'clock than at 6. This unbalanced clamping force creates an eccentricity that sets up the follower forces noted in the description of the whirl mode (see 3.6). It is important to note, however, that some researchers do not consider this eccentricity to be necessary for whirl to occur, and consider a perturbation to be sufficient.

In hydraulic brakes, the unbalance is exacerbated by the fact that actuation itself typically proceeds from 12 to 6 o'clock because the fluid pistons are actuated in sequence according to their distance from the brake's fluid inlet port. In electric brakes, the actuation can be simultaneous, which theoretically will reduce the unbalanced excitation. For this reason, some engineers believe electric brake actuation will reduce whirl excitation and probability of occurrence. This belief has yet to be confirmed or refuted by service operation experience, but some laboratory evaluation implies that it is valid.

4.4 Excitation from Anti-Skid System Feedback

This type of feedback can cause instability and limit cycle motion of the low frequency walk mode of the complete landing gear. It is not dependent on adverse brake frictional properties but is a function of the anti-skid characteristics and the fore and aft natural frequency of the landing gear. It is caused by the fact that wheel speed variations due to transient fore-aft motion of the gear can be misinterpreted by the antiskid logic as impending skids, and brake pressure modulation intended to minimize skidding can instead act to reinforce the transient motion and drive it unstable. This is more of a concern with large landing gears, which have natural walk frequencies below 20 Hz. In those gears, the problem can be avoided by designing the antiskid to be insensitive to the gear frequency. However, smaller aircraft, which typically have walk modes near 20 Hz, have still been shown to be responsive to antiskid excitation. Gears with natural walk frequencies in the 30 Hz range and above are not as susceptible because the band-width of skid control systems is typically lower. These gears are rare.

There is a caveat to the aforementioned statement. If an antiskid system modulates pressure and torque by near-step release and reapplication, the frequency content of such disturbance may be broad enough to serve as excitation to modes of higher frequencies.

In addition to gear walk excitation, it should be noted that there are other modes of the gear and aircraft which can be excited by the antiskid, because their natural frequencies are within the operating range of the antiskid system. Aircraft pitch and bounce mode transients occur below 5 Hz, and the bogie pitch mode of larger aircraft occurs below 20 Hz. These are significant because antiskid systems are designed to accommodate scenarios where braking is initiated before these typically transient motions die out. Information regarding these other modes is important to the designers of brake control systems, so that inadvertent excitation of these modes can be avoided. The issue of stability in these circumstances is not as significant as the fact that any such motion can change the tire-ground traction and transient tire radii.

It is a matter of conjecture whether a sufficiently sophisticated anti-skid system can be designed to discern low frequency modes (such as gear walk or pitch) and provide active damping to increase their stability and perhaps even compensate for destabilizing effects of the brake friction characteristics. It remains to be seen whether such a system would sacrifice anti-skid efficiency to do so. One of the obstacles to such a capability would be the time necessary for such discernment. If it takes too long, it would seem that stopping distance would have to suffer. But the tradeoff may be positive.

4.5 Tire Lockup

Friction-induced instability of landing gear vibration first became an issue on the Electra and Constellation because of tire-ground friction. These aircraft predate the development of antiskid systems that prevent tire lockup. When the brakes locked and deep skids occurred, the friction between the tire and runway was found to be speed-dependent and could cause gear walk instability severe enough to destroy landing gears. Such concerns have been largely eliminated by the use of anti-skid braking control. Yet they can still occur at very low speeds, because such control systems become ineffective at some low speed during braking with most wheel–speed sensors. The instability is not dependent on adverse brake frictional properties. It is the result of the negative slope of the tire-runway friction characteristics beyond that level of slip which corresponds to maximum drag. The instability is increased if the landing gear has a forward cant angle.

4.6 Torque Gain and Humidity: Absorption and Adsorption

One source of brake vibration in carbon-carbon composite brake linings has been shown to be aggressive and sudden increase in brake lining surface friction due to water desorption. Sudden very large changes in friction coefficient can occur when heat generated by braking energy raises surface temperatures and alters the residence time of water molecules adsorbed on the surface. The highly transient adsorption and desorption phenomena should not be confused with the slow absorption and off-gassing of water molecules that occurs when brakes are left in humid environments for extended periods of time. For the latter case, the absorbed water molecules typically produce low brake friction (often called morning sickness), and subsequent use of the brake off-gases the water and returns normal performance.

Desorption is different. When desorption occurs, the sudden inability of the carbon to maintain a presence of adsorbed molecules removes their lubricative influence on the carbon surface, and the result is an abnormally large step increase in friction coefficient. If it occurs simultaneously across many of the brake interfaces, the result is an abnormally high spike in brake torque. Beyond initiating squeal vibration, the step disturbance can excite many brake system modes, and cause excessive torque gain, commonly referred to as grabbiness, or ‘brake snatch’ in the UK. Sudden decreases have been observed as well. These changes also serve as step functions in friction coefficient, which contain many frequency components, and hence can serve to excite many modes of vibration. It is logical to expect that this problem should be less likely to occur in brakes containing more rotors because their larger number of surfaces reduces the statistical likelihood of simultaneous desorption from each rotor. It should be noted however that a low likelihood of occurrence is not necessarily desirable, since the problem may not be detected during screening testing, and thereby become a much larger problem later. The primary factor governing the severity of desorption has been shown to be inherent to the heat sink material, independent of the number of rotors, with some materials exhibiting marked sensitivity compared to others.

While it is true that carbon is reactive with many substances besides water, there is no published evidence that any other than water is a factor in brake grabbiness. The chemicals in de-icer fluids and oxidation inhibitors can indeed have a significant effect on reducing brake friction, but it is not known to be regarded as a factor in excitation of vibration.

4.7 Nonlinear Compressibility of Brake Disks and Fluid

Carbon disks have a markedly non-linear compressibility. Some analytical studies have indicated that this results in a region of instability dependent on coefficient and pressure. Other studies at Southern Illinois University have suggested that the phenomenon is not limited to carbon, because the same nonlinearity can be caused in metallic brakes due to surface asperities. Axial compliance nonlinearity also extends to the fluid stiffness in hydraulic brakes, which is markedly affected by air content. This area deserves more study, and the situation can be expected to be different in electric brakes, which have no fluid compliance issues.

4.8 Distortion

Many brakes are designed to react torque through linkages or single-point lug interfaces with the landing gear. In such designs, the brake torque generates bending moments reacted by the brake's interface with the axle. These moments cause the housing and backplate of the brake to distort in a non-axisymmetric manner, which results in load distributions on the disks that act to excite whirl, squeal, axial modes, and disk-mode flexure. Some designers regard this as a major source of coupling among brake modes. It must be noted that distortion can be minimized by designing the brake to eliminate such bending moments. Unfortunately, when the bending moments are eliminated, it can result in lightly loaded outboard bushing reactions, which reduce damping for whirl mode vibration stability. As is often the case, the decision is not clear cut and depends on the vibration mode that is the greatest threat to the system.

4.9 Duty Cycle and Pilot Technique

Empirical evidence suggests that aircraft duty cycle and pilot technique can have a significant impact on brake vibration. For example, some vibration modes only occur at particular airlines on particular routes. This is not yet completely understood, but may be linked to differing cooling cycles, longer exposure to lack of humidity at altitude, different intervals of brake applications, and thus oxidation times, and so on. At one airline, a vibration problem associated with brake wear through lengthy revenue-flight investigation was eventually cured by an alteration of pilot technique.

4.10 Friction Surface Wear Grooves

Due to contamination, use of fans which remove wear debris, and other environmental factors, carbon friction surfaces can deteriorate and develop a wear pattern called "record grooving". This was once believed to be related to the excitation of some brake vibration modes. There is no published evidence that this is the case. Grooved and tapered wear patterns do in fact occur on some brakes in revenue service, but there is no published evidence that it is related to vibration, either as a cause or as an effect. Until there is published evidence to the contrary, friction wear grooves are regarded as benign.

4.11 Wear Lips

As carbon disks wear, they can develop lips at their inner and outer diameters. They are a cause for rotor cycloidal vibration. See 3.11.

4.12 Hydraulic Pressure Oscillations

Some investigators have found that symmetric gear walk turned out to be not a stability situation, but a forced response caused by hydraulic pressure oscillations. The experience of some practitioners has been that many gear stability related problems that were initially assumed to be caused by brakes during braking actuation were ultimately related to improper anti-skid control and tuning for gear walk and/or shimmy. In the experience of others, valve characteristic can also play a role in destabilizing a system regardless of the antiskid logic.

5. DIAGNOSIS

Vibration is most commonly measured by accelerometers, but other instrumentation, such as strain gages, pressure sensors, thermocouples, load cells, potentiometers, and cameras can be necessary to identify the conditions under which the vibration may occur. Diagnosis requires a combination of instrumented hardware and valid mathematical models. It is very different in laboratory situations than on the aircraft.

5.1 Hardware Instrumentation in the Laboratory

There is virtually no limit to the amount of instrumentation that can be used in a laboratory situation. Some practitioners instrument heavily with triaxial accelerometers and pressure transducers at or in every port and piston. Others use the minimum necessary to define a limited number of modes. In linkage-mounted brakes it is important to also instrument the brake rods, usually at two locations in long rods to identify 1st and 2nd bending modes. It is rare that the backplates of such brakes are instrumented, because the temperatures are too high in general, and space is limited. To assist in diagnosis of the role of rods and fixtures, especially with regard to squeal vibration, it is prudent to instrument both.

The records from accelerometer data are commonly used to identify frequencies of vibration and the animated mode shapes associated with each frequency of interest. In addition, waterfall plots are useful to identify speed-related vibration.

5.2 Aircraft Instrumentation

It is more difficult to obtain extensive data from aircraft tests, particularly regarding the dynamics of each brake. This is because aircraft manufacturers do not typically instrument the landing gear and brake for comprehensive brake vibration surveys unless a problem is suspected. Due to the expense of dedicated flight testing and instrumenting a large number of brakes, usually only a minimum instrumentation set is used to monitor the brake during the many landing and braking cycles of the flight test program. This remains an issue for the brake designers, because it is more difficult to establish correlations with their dynamometer tests and analyses. It has been an historical problem, because brake laboratories are concerned that their fixturing accurately represents the aircraft systems, and confidence in the fidelity of laboratory systems is more difficult to establish without correlation with aircraft data. It is an unfortunate fact that the high cost to instrument airplanes usually results in minimum instrumentation. Aircraft are instrumented for vibration only if a problem is suspected to be occurring, and the problems suspected cannot be identified without instrumentation.

5.3 Data Processing

Merely measuring vibration at available locations on the system structure is not sufficient to diagnose its cause and/or cure. Various post processing techniques are necessary to interpret the data and understand the phenomena. These include the following.

a. Fast Fourier Transforms (FFTs) can be used to identify the frequencies of vibration that occur. These are often the first step in comparison to mathematical models to identify the modes of vibration that may be occurring.

b. Histograms can be used to help put vibration occurrences into statistical perspective. If a vibration exceeds what may be regarded as an acceptable level only a very small percentage of the time or circumstances of the brake operation, it may be regarded as statistically insignificant.

c. Orbit Plots, where for instance biaxial accelerometer data is plotted according to its direction, can be useful in ascertaining whether a vibration mode is predominantly whirl, bending, or squeal. If the orbit is completely circular, a whirl solution may be appropriate. If it is predominantly bending, stiffness alterations such as hydraulic blocks may be better.

d. Waterfall Plots, which portray vibration as a function of frequency, amplitude, and time or speed, can be useful in ascertaining the degree to which a vibration is speed-dependent and further interpreted to discern what order of rotation it may be dependent upon. These are especially useful in correlating vibration to design orders of cyclic symmetry, such as 6 drives per wheel reinforcing the effects of 6 pistons at one speed.

e. Animated Mode Shapes are useful to understand the motion of any particular vibration frequency. Some practitioners use many accelerometers to define the shape, while others prefer to use a minimum number and map the motion of uninstrumented locations by making various assumptions regarding flexibility between the instrumented locations.

5.4 Limitations

Unfortunately, brakes cannot be instrumented to completely identify all vibration modes that may occur, because instrumentation cannot be put at locations which become too hot for the sensors to function, or which afford insufficient space for mounting them. Because of this it is necessary to augment the test data with valid mathematical models to infer the motion of parts that cannot be measured. An example would be that the vibration of the heat sink itself, or the parts they interface with, such as torque bars in wheels, cannot be practically instrumented because of their high temperatures and lack of space.

5.5 The Role of Models

Mathematical models can be used to extrapolate vibration at measured locations to locations internal to the brake which cannot be measured. They are also necessary to establish confidence that the laboratory system is truly representative of the aircraft system. See Section 9 for elaboration.

5.6 Ambiguity

Despite the judicious use of models and test data, it remains difficult to discern some subtle differences in modes of vibration, or to infer them correctly. And the fact remains that the output of models is only as good as the input. If an aircraft system is not defined accurately, the laboratory or computer systems meant to simulate it cannot be expected to do so either. Even if the aircraft system is defined as precisely as possible, it still remains an artful endeavor to produce models that correlate completely with measured performance. The target should really be models that perform adequately, or exhibit “acceptable” correlation. The ambiguity unfortunately extends to a judgment of what is adequate or acceptable.

6. SYSTEM DEFINITION

6.1 Some gear configurations are more responsive to brake vibration than others. A friction pair suitable for one gear is not necessarily satisfactory for a different design. Probability of compatibility is increased by testing and analysis during the development program. The objective of such efforts should be to match gear and brake characteristics to minimize adverse vibration throughout the anticipated operating range and to uncover any adverse wheel and brake features that promote or aggravate vibrational motion. It is of greatest importance that the gear is accurately portrayed for the brake designer to provide a brake designed to be dynamically compatible with it.

6.2 Additional or supplemental review of compatibility can be accomplished analytically if proper input data are generated during the development program. The airframe manufacturer must provide basic aircraft parametric data for such analysis, laboratory simulation testing, or both, to enable the brake designer to provide a high-fidelity dynamometer system representing the aircraft system. These would include:

6.2.1 Fore and aft spring rate and gear walk mode shape and natural frequency of the landing gear, or flexibility matrix for 6 degree of freedom motions of the truck or axle center point (multiple axle gears) or wheel center (single axle gears). This can be accomplished in several ways, depending on whether the aircraft and its gear(s) are in the design stage or actually available as hardware.

6.2.1.1 Hardware Available Options

If the hardware is available, the gear mode frequency and shape can be defined by shake tests of the gear on a suspended aircraft. Alternatively, the same information can be derived from “stab” braking which simulates step application or release of brake torque on a rolling aircraft. Both techniques have been used. In either case, defining the mode shape requires sufficient instrumentation to define both strut and axle motion. The angular and linear displacements in the pitch-plane are both important. It is best if the gear tested is under a true drag load or one simulated by such things as bungee cords to remove clearances that may affect the frequency and damping.

6.2.1.2 Hardware Unavailable Options

If hardware is not available, the gear mode shapes and frequencies must be provided by an appropriate analysis of the structure. A static unit load analysis can define the shape of the gear mode and its deflected position under different decel conditions, as it is generally accepted that the static shape is the same as the fundamental dynamic shape for all practical purposes. But the frequencies must be defined from a normal mode analysis of the gear and aircraft as a system. Obviously this alternative can also be employed if the actual gear exists as well. It is important to recognize that the gear walk mode may in some instances be governed more by the gear support structure flexibilities than by the flexibility of the strut itself. It is not effective to construct a strut model with thousands of elements and assign it poor boundary conditions or mate it with an aircraft model containing an insufficient number of structural degrees of freedom.

6.2.1.3 Damping coefficient associated with the fore and aft walk motion of the landing gear system or loss factor for the spring matrix mentioned. Unfortunately, this will almost always be an estimate, and dependent upon the experience of both the airframer and brake designer. Ultimately, appropriate levels of damping for different gear designs rests with experimental evidence provided by aircraft test of one sort or another. This type of information is rare, but should add to the experience bank of the brake designers. As an example, the walk-mode damping of single-axle gears was long assumed to be on the order of 5 to 7%, when tests from more than one airframer eventually suggested the appropriate value was perhaps 3 to 4% and amplitude dependent. This is a very significant parameter in the brake designer's attempts to simulate the system correctly on the dynamometer or in mathematical models. Overestimating the damping associated with a particular gear mode has the effect of overestimating the stability margin of the brake during qualification tests on the dynamometer, and consequently impugning the fidelity of the dynamometer simulation unfairly if damping levels simulated are too high.

6.2.2 Angular Spring Rate Associated with Squeal Motion of the Landing Gear System

For a single-axle T-gear or trailing-link gear, the brake torque is transferred to the gear either by a flange with a combination of studs and bolts, or by a single-point lug. In either case the pertinent spring rate can be easily derived by the brake designer from axle (and trailing link gear) dimensions, if the air-framer provides them.

In the case of multi-axle gears the brake designer needs more information from the air-framer. As discussed earlier, there are several possible squeal modes of such systems, dependent on both the inner cylinder interface design and the rod design itself. The geometry and material properties of the gear and brake rod design are still essential, but it is also necessary for the air-framer to provide the landed aftward gear deflection under a given drag load or aircraft deceleration. For high-fidelity analysis of the system and dynamometer simulation, it is necessary for the air-framer to provide at least a static deflection of the gear as part of a complete aircraft system model. This is most commonly understood as a unit-load analysis, but the drag load associated with a nominal aircraft deceleration, such as 8 or 10 fpss, must also be provided.

Also in multi-axle gears, the fore and aft rods may be designed differently, because the forward rods have to react a compression load, and so must be designed to resist buckling. The detail design of both rods must be provided to the brake designers, including the dimensions of auxiliary hardware such as pins, bushings, and fasteners.

6.2.3 Angular Damping Coefficient Associated with the Squeal Motion of the Landing Gear System

Again, this is a parameter that depends heavily on experience, because it is difficult to measure. There are some useful rules of thumb however. For instance, a flange-mounted brake can be expected to have very low squeal mode damping because few interfaces are involved in the motion. But a link or lug interface provides a source of coulomb damping, which is very high at low amplitudes and decreases drastically with vibration amplitude.

In some multi-axle gears, the fore and aft brakes are raked towards each other. One result of this design is that under drag loads, when the inner cylinder deflects and rotates aftward, the fore and aft brakes have different rake angles, and the brakes have different radial distances from the brake rods (see Figure 2). Equal brake torques fore and aft are reacted by unequal brake rod loads. The forward brake rod load is higher than the aft rod load. Thus the friction at the pin joints of the forward rod is higher, which implies higher damping.

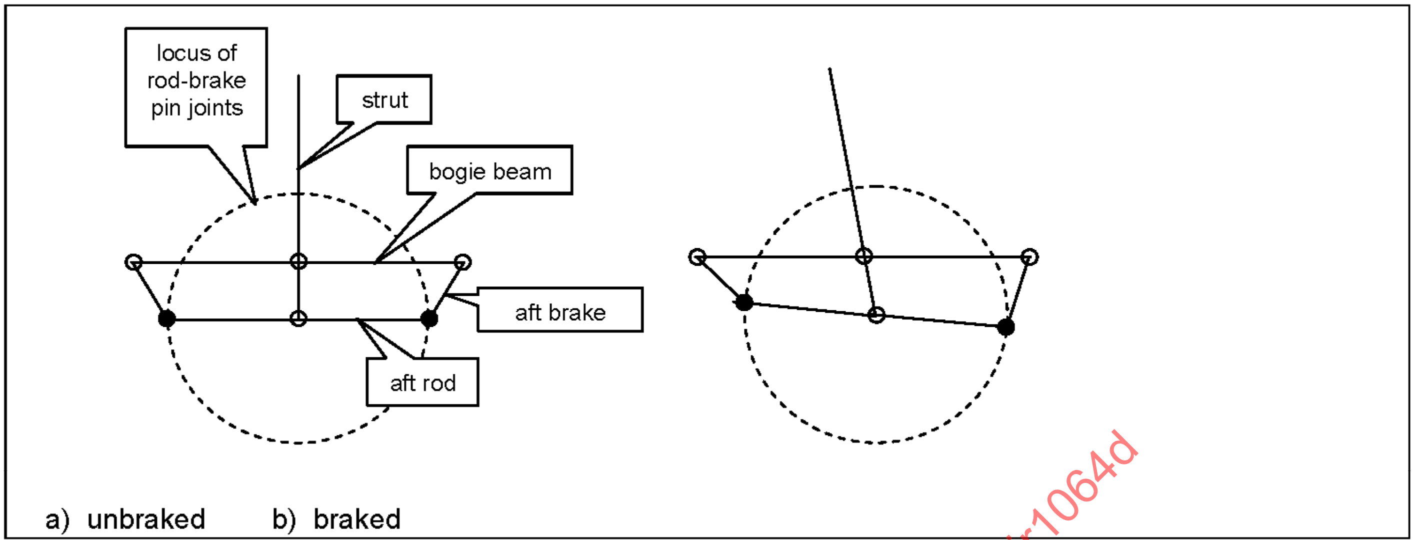

FIGURE 2 - CHANGE IN FORE-AFT RAKE ANGLES OF BRAKES UNDER TORQUE

6.2.4 Axle Geometry, Stiffness and Damping Characteristics, Configuration, and Mounting Flange Details

Alternatively, it is sufficient for the airframe designer to provide accurate dimensioned descriptions of the axle and mounting details, or solid models of the components. It is well within the capabilities of the brake designers to analyze the pertinent dynamic characteristics of the axle and its junction with the landing gear if it is described to a sufficiently clear and complete degree in a timely manner.

6.2.5 Various Tire Characteristics

The tire is a very significant component for effective modeling, perhaps more so to brake-control engineers than dynamicists concerned with vibrational stability. Most research on tire characteristics has been focused on the issue of shimmy. These are of negligible concern to braked landing gears. Yet there are several parameters that are critical, and remain difficult to define. TR-64 has been a classic reference, but it is probably fair to say that most researchers are aware that it is out-dated, mostly because it predates the advent of radial tires, and consequently different engineering organizations have developed their own more or less proprietary modifications and interpretations. The following parameters are of greatest concern:

6.2.5.1 Rolling Radius

The rolling radius of tires has been long assumed to be equal to the free radius less 1/3 of the radial deflection. This has been shown to be a reliable assumption in general for bias tires only. The mathematical basis for it can be found in the Hadekal paper referenced in the bibliography of the TR-64 reference in Appendix B. NASA tests at Langley have documented that for radial tires the rolling radius is closer to the free radius less 1/5 of the deflection. In either case, the rolling radius is NOT a constant, but is dependent on speed, due to centrifugal effects, and the variation in tire load that occurs as the aircraft actually lands and rolls out.

6.2.5.2 Braking Radius

Braking radius is defined as the ratio of brake torque to drag force. When torque and drag forces are applied to the tire, the footprint translates such that the centroid of the vertical force on the tire is not through the axle centerline. When this happens, the normal force either adds to the torque or subtracts from it, depending on whether the centroid moves forward or aft of the axle centerline. The equations are clear. Refer to Figure 3B, which is taken from the oft-quoted TR-64. In the scenario shown, the centroid of normal force is assumed to move aft of the axle by distance x in the direction of the drag force. Thus the vector loads on the tire from drag and normal ground loading combine to affect the torque. The ground load contributes to torque by virtue of the footprint translation. Thus the torque is NOT equal to the product of drag and deflected tire radius, more conventionally referred to as axle height. It is more correctly represented by the expression

$$T = Dh + Nx = DR_{Braking} \qquad \text{(Eq. 1)}$$

from which $R_{Braking} = \frac{T}{D} = h + \left(\frac{N}{D}\right)x > h$

where:

D = drag force

h = axle height above the ground

N = normal tire load

x = aftward displacement of the centroid of the N distribution

Clearly, if the centroid of N deflects forward, opposite of the direction of D, then x would be negative in these equations and the braking radius would be less than the axle height, which is implied by Figure 3A. The apparent problem is that some engineers claim to have measured T, D, and h, calculated $R_{Braking}$, and found it to be greater than axle height h and nearly equal to the tire's free radius.

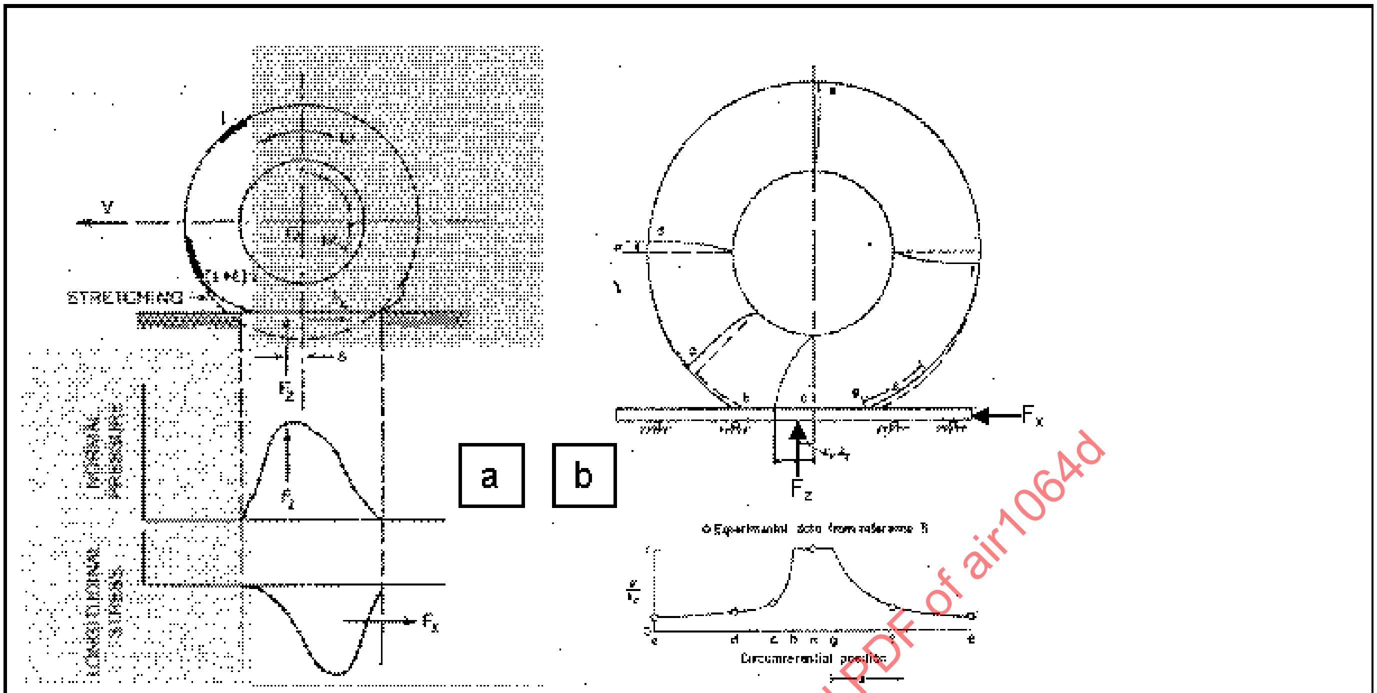

FIGURE 3 - CONTRADICTORY VIEWS OF THE TIRE LOAD CENTROID MOVEMENT UNDER BRAKING ACTION (FIGURE 3B TAKEN FROM TR-64)

It is significant to note that the footprint translation under drag is exaggerated during roadwheel dynamometer testing by roadwheel curvature effects. Some testing purports to show that the braking radius is at least 95% of the free tire radius, but it is unclear whether that is an average or instantaneous conclusion. It is of concern to those seeking to understand the effect more in the antiskid than braking dynamics fields. Other than that, there are no clear rules of thumb correlating braking radius to axle height.

There is a body of literature supporting the view that the N-centroid moves forward under braking. Most of this comes from the automotive industry, and some of it seems to be self contradictory. One reference for Figure 3A shows it to be from "Theory of Ground Vehicles" by J.Y. Wong., and is based on a Masters Thesis, but the assertion appears in others as well. Moreover, this viewpoint is shared by some engineers in the aircraft industry, and on the panel responsible for this document.

Beyond this discussion, this AIR cannot resolve the apparent contradiction cited, but only call your attention to the fact that such a difference of opinion and understanding does exist. We recommend that efforts be taken by the A-5 Committee to resolve it.

6.2.5.3 Tire Radial Stiffness

The radial stiffness helps to define the amount of tire off-loading associated with axle motion during any of several gear modes. It is significant to antiskid designers who must maintain maximum traction in the face of these load variations. It is especially significant to skid-control engineers that this stiffness is nonlinear, because braking is often initiated before the gear and tire are fully loaded and deflected.

6.2.5.4 Tire Torsional Stiffness in the Pitch-Plane

It is interesting to note that tire torsional stiffness in almost all documents regarding tire properties refers to ground-plane yaw stiffness, because that is a parameter significant to shimmy analysis. This AIR takes issue with that definition, and suggests that it be specifically described as yaw stiffness for shimmy discussions. The pitch-torsional stiffness is significant to pitch-plane dynamics associated with brake-induced vibration. It can be estimated from the footprint translation under drag, but there is no apparent consensus regarding how to do so. Yet it is an important parameter in the definition of the tire-wheel chatter mode of vibration. Some researchers derive this parameter from measured tire chatter frequencies and assumptions regarding the amount of tire and wheel inertia participating. Needless to say, there is little obvious consensus on the assumptions, either.

6.2.5.5 Tire Damping

It is commonly believed that radial tires have lower damping than bias-ply tires. This can be a concern if an airframer is contemplating a switch from one to the other, or to tire construction variants as the NZG tire. The literature suggests that radial tires have less damping, so that the stability margins of vibration modes that involve the tire would be decreased.

It is possible to define all of these characteristics from dynamometer tests that are properly instrumented to measure tire normal and drag loads and axle positions during a brake test, but the techniques for doing so have to be regarded as unsubstantiated in the literature to date. Attempts by some researchers have led to the realization that roadwheel curvature is a significant influence, and must be accounted for. Even if axle position is held constant, translation of the footprint on a dynamometer has a larger tire-offloading effect than on the aircraft, because the load vectors rotate. It is relatively simple to derive average values for these parameters from the summation of data over a stop, but instantaneous values are of greater value to the optimization of antiskid control systems.

6.2.6 Landing gear configuration (2/4/6 wheels), including the design of the inner cylinder attach lug. In the case of a single-axle gear, the most important information remains whether it is a post gear or a trailing link gear. Multi-axle gears always feature a linkage torque-transfer design to prevent bogie pitch-down during braking. These can have several different methods of joining the brake rods of the linkages to the inner cylinder of the gear. And each has its unique effect on brake interaction with the gear,

6.2.6.1 Common Fore-Aft Gear Pin Joints

In this design, the fore and aft brake rods overlap at the inner cylinder and transmit their braking loads through a common pin. If fore and aft brakes vibrate in squeal in phase, their dynamic loads are reacted by bending of the inner cylinder lug. If they vibrate out-of-phase, the lug pin becomes a stationary node. The two possibilities can lead to the requirement for different fixturing on dynamometer tests. Needless to say, both modes involve significant brake rod bending, which can lead to significant frictional heating of the common fore-aft pin joint on the aircraft.

6.2.6.2 Common Left-Right Gear Pin Joints

In this design, the fore-aft rod interaction is less direct, and the left-right reactions dominate. The consequence is that the common left-right inner cylinder pin is subjected to bending loads, which complicates the design of dynamometer fixtures representing the reaction by requiring them to bend in response to the rod loads. This situation naturally results in a greater amount of brake rod bending, but reduces the threat of frictional heating of the pins on the aircraft from rod vibration.

6.2.6.3 Four-Pin Gear Joints

In this design, all four brake rods are reacted by separate pins on a more elaborate inner cylinder lug. For dynamometer simulation, this situation is very much like the common fore-aft pin joint, but for the gear designer, the possibility of frictional heating of a common pin by rod vibration is eliminated.

Lastly, it should be recognized that in trailing-link landing gears, there are two gear modes responsive to excitation from torque or drag variations. In addition to gear walk, the torque loading also has the effect of lifting the trailing link of such gears against the restraint of the shock strut. This mode is more highly damped than the walk mode, but is often in the same frequency range. Little is documented in the literature about the coupling effects between these modes.

6.3 Suitable flight testing should be planned for the development program to verify system compatibility. This compatibility must include demonstration with the full range of skid control operation on wet and dry runway surfaces. Data collection should include information on speeds where vibration occurs, past history of the brake, frequency, and amplitude of vibrations. Ideally, instrumentation would include the minimum number of accelerometers in the minimum number of locations and directions that would allow mode shapes associated with each vibration to be defined. This will assist in pinpointing problem areas and will also provide insight to problem solution. Moreover, it would go a long way in establishing confidence that the brake designers are providing both dynamometer and computer models of the system with high fidelity.

6.4 Suitable dynamometer testing should also be planned to provide the brake control system designers with sufficient data to optimize their control models. Currently, it is common only to provide frequency response test results, because that is typically a specification control document requirement. Some control system designers feel that it would be helpful to provide data from more extensive dynamometer tests specifically designed to define other parameters. It should be realized that such additional tests constitute a burden on the brake providers, who are under considerable schedule restraints to satisfy other test requirements. The ultimate solution may be to coordinate and integrate the requirements of all suppliers for any gear system and conduct integrated dynamometer system tests. All component suppliers should consider the benefits of greater integration of their resources. Integrated dynamometer system tests, including PIL, have been conducted for some systems. With the advent of electric brakes, there is increased interest in the characterization of brake clamping force by the brake supplier for the control system supplier. It would seem the future is in integrated efforts by each.

7. DATA CLASSIFICATION

7.1 In spite of the scatter in results of identical stops that prevents pinpointing or predicting a specific event for any one stop, continued effort should be devoted to identifying the dynamic characteristic pattern. Meaningful information can result through use of statistical analysis when a large amount of seemingly inconsistent data is processed. Sustained analysis and testing by brake manufacturers and cooperation with the users on vibration problems is essential. Regard this as an area that is sadly underdeveloped. The industry has the potential to perform much better in this regard, and it should endeavor to do so.

7.2 A uniform method of classifying brake performance on a statistical basis is desirable. The following would classify a brake assembly as a complete unit and as applied to a specific gear. The suggested classification of brake characteristics is:

7.2.1 Coefficient of Friction

a. Static - cold, hot, and dynamic breakaway (termination of skid)

b. Dynamic - initial, average, and maximum

c. Transition - dynamic to static (entering skid)

d. Carbon brakes - wet, dry, and transition wet to dry

7.2.2 Dynamic Coefficient of Friction Variation With

a. Kinetic energy absorption rate

b. Temperature

c. Amount of absorbed kinetic energy

d. Velocity

e. Unit Pressure

f. Number of prior landings or applications, or both

g. Amount of water applied (carbon brakes)

7.2.3 Wear-Variation With

a. Level of kinetic energy absorption/mass loading/temperature

b. Rate of kinetic energy absorption/power loading/velocity

c. History of past usage, including potential vibration induced wear

d. Unit pressure

e. Wet or dry condition on carbon brakes

7.2.4 Torque Versus Pressure Characteristics

a. Response to ramp increases and decreases in pressure at various mean pressures

b. Frequency response characteristics (gain and phase) for pressure variations from 0 to 50 Hz

c. Both of the above as a function of wear and brake operating temperature

d. Brake pressure versus displacement (required fluid volume) as a function of brake wear

7.2.5 Landing Gear Dynamic Characteristics

a. Gear walk natural frequency and mode shape

b. Squeal natural frequencies and shapes

c. Positive damping associated with walk, squeal, and whirl

8. LABORATORY RESOURCES

The following is a discussion of techniques, equipment and facilities that can be used for brake and skid control evaluation in the laboratory:

8.1 Shaft Dynamometers

Direct drive shaft dynamometers have been used to evaluate the brake as an assembly and for evaluation of brake components. Attempts have been made to simulate the flexibility of gear mounting and to operate with a system, which responds to vibration in both the "walk" and the "squeal" mode similar to the aircraft, but the success of such attempts has not been documented publicly. Yet according to a previous revision of this document, some investigators feel that single wheel simulation has been accomplished on a shaft dynamometer with fixtures simulating the flexibility of the mounting on a conventional brake test variable inertia dynamometer.

8.1.1 The major advantage of this type of machine is that comparative evaluation of geometry, material, and performance can be achieved at relatively low cost in a timely manner. It is an excellent screening device for:

a. Wear rate

b. Temperature of brake elements (but not relative cooling rates)

c. Static and dynamic friction coefficients

d. Variation of friction coefficient under controlled test environments

8.1.2 Noted disadvantages of this type of equipment are primarily that it is not generally regarded as appropriate for the evaluation of dynamics or skid control:

a. The flywheel kinetic energy is transmitted directly to the brake, thus removing the tire effects on brake system energy absorption.

b. The critical influence of tire or strut elasticity on brake dynamic characteristics is absent.

c. There is no axle deflection, hence no excitation for whirl or non-axisymmetric effects. Hence dynamic stability cannot be assessed well.

d. It is always limited to evaluation of one brake assembly.

8.2 Conventional Roadwheel Dynamometers

The most common method of brake testing consists of landing one or two complete wheel, brake, axle, and tire assemblies against a rotating inertial wheel. There are two kinds of these dynamometers that have been in use.

a. Sliding carriages. In this type of dynamometer the test assembly is mounted in a fixture that is powered to translate into the roadwheel. Such fixturing usually includes multiple force cells to monitor and control landing and drag loads. Yet their inertia limits their ability to respond quickly to simulate landing load profiles or reactions at typical landing gear sink rates.

b. Pivoting load arms. In this type of dynamometer, the load arm pivots about a point near floor level and landing motion and force is applied from above the axle or below the floor. These designs predated the use of antiskid systems, and part of their utility was that they self-energized in the sense that brake torque and tire drag acted to increase the tire load to inhibit tire skidding. The downside of this technique is that it is difficult to control tire load. Such systems are commonly actuated either pneumatically or hydraulically, but electric actuation is used in some laboratories as well, in order to improve control of axle load and/or position.

   Pivoting load arms have the opposite effect of de-energizing if the roadwheel is run backwards, thus decreasing tire load in proportion to brake and drag load generation. Some researchers have used that characteristic to simulate the main gear off-loading effect that occurs when weight is transferred to the nose gear upon braking.

Both of the above kinds of roadwheel dynamometers typically are operated with one wheel-tire-brake-axle assembly, and the inertia of the roadwheel is expressed in an “Inertia Equivalent” representing the aircraft weight per brake, and consisting of a variable number of steel disks joined together as the “roadwheel”. The largest such dynamometer has a 192 in diameter and is at WPAFB laboratories.

Both of these methods tend to share seven disadvantages:

a. The motors used to accelerate the roadwheel to proper speeds are much less powerful than the engines on an aircraft. The consequence is that the timing between taxi and landing stop events is often compromised.

b. Inertia and landing speed is commonly a compromise, because none naturally account for the aerodynamic drag of the aircraft's spoilers and reverse thrust that helps to decelerate the aircraft. Common practice has been to either match landing speed and reduce inertia, or match inertia and reduce landing speed. This is a compromise especially undesirable to antiskid engineers, because they must begin to operate at highest speeds, but prefer not to cope with the thermal variation with friction that results with the excessive energy involved with doing so with an unrealistically low inertia. There are methods which can eliminate this compromise, but none have been disclosed in the literature.

c. Natural vibration modes of the axle fixture restraints of conventional dynamometer sites can be in the same frequency range as critical modes of the mounted brake. Consequently, unrealistic coupling can occur and true dynamic interactions can be masked, unless measures are taken to simulate the significant gear modes, or isolate the natural fixture modes.

d. Tire-to-flywheel friction becomes erratic during antiskid testing because of the patchy rubber buildup on the flywheel.

e. It is difficult to simulate runway friction for antiskid testing.

f. It is also difficult to simulate the weight transfer from main to nose gears that occurs during landing and brake application.

g. Tire footprint translation on the curved surface of a roadwheel dynamometer is not the same as on an aircraft, because the directions of the load vectors rotate.

Both of these methods also tend to share six advantages:

a. At the very least the assembly to be tested is mounted on a fixture that can be made to embody or simulate the aircraft axle. The axle is always included and its cantilevered bending contribution to whirl-mode excitation is a natural advantage to assessment of whirl stability to various degrees.

b. The tire is always included in the system, and its energy absorption is properly represented. Its chatter mode is represented as well.

c. The thermal environment is more accurately represented during the cooldown event following landing, because the tire is included and forced airflow can be simulated with fans. Thus the thermal effects on friction are well represented.

d. Preliminary assessment of anti-skid compatibility is possible, but limited to such things as the transition of dynamic to static characteristics, torque response, and ability to move actuation fluid.

e. There are various possibilities for enhancing the test fixturing to represent gear dynamics.

f. For antiskid developers, it is very significant that on the dynamometer the brakes truly heat up and friction changes as a result, and that the tires truly slip under the drag loads generated. These cannot be simulated well on computers, regardless of the data generated to feed the programmers.

8.3 Unconventional Roadwheel Dynamometers

There are two types of roadwheel dynamometers that are unconventional for different reasons. They have some unique advantages but share most of the characteristics described above for conventional roadwheels.

8.3.1 The “electric inertia” dynamometer can function as either a conventional dynamometer, or in an electric motor torque mode which simulates mechanical inertia using a digital computer in a closed-loop control system

The machines can be capable of simulating inertia and surface speeds for testing brakes for large transport aircraft all the way down to light military fighters.

One of the advantages is that a wide road wheel can be used to accommodate large braking systems, such as two-wheel axles, without the landing speed/inertia compromise necessary on conventional roadwheels.

The machine has controls capable of running stops in deceleration control, with the option of varying inertia, torque profiles, and rolling radius throughout the braking application.

On such electric-inertia dynos, it would seem that with proper programming, drag proportional to speed could be implemented to simulate aerodynamic drag as well.

8.3.2 A special inside-out dynamometer at WPAFB test labs has the unique capability of landing a braked assembly inside of a large drum, as opposed to all other roadwheel dynamometers. Clearly roadwheel curvature effects are different on this site. The big advantage of such a set-up is that various runway surfaces can be simulated, including grit, water, and slush.

8.4 Two-Wheel Gear Simulation on Roadwheel Dynamometers

Recognition of the elastic nature and complex interaction of each element of the landing gear system has resulted in the desire to test as much of the actual system as possible. Various degrees of simulation have been achieved with the use of each type of test facility. Under certain conditions, it has been possible to duplicate or simulate very closely the full-scale landing gear dynamics from the trunnion attachment interface to the tire flywheel interface. Several aircraft systems have been investigated in this manner; and in some cases, the simulation has been extended to include the actual aircraft attachment structure. There is a difference of opinion in the industry regarding the value of such simulation, particularly in regards to the compromises represented in simulating the effects of the aircraft structure to which the gear is attached, without unrepresentative fixturing modes.

Some (perhaps most) investigators maintain that most successful simulation has been achieved by overhead mounting of a gear or simulated gear on a conventional brake test variable inertia dynamometer. This has always been limited to a two-wheel gear or a two-wheel simulator producing dynamic response of a four-wheel gear on the aircraft.

Other investigators with significant experience with “overhead rig” testing of two-wheel landing gear systems feel it is a flawed approach, because the contribution of the wing and fuselage structure is very difficult to replicate, and is so critical to the simulation that the use of the actual gear in the structure is of negligible advantage. In the experience of these investigators, fixture modes are inevitably introduced that interfere with and obfuscate the simulation, and roadwheel curvature effects cast doubt on the fidelity of such simulations.

Nevertheless, “overhead rig” simulation remains the simulation of choice at some, if not most facilities. It is ultimately up to the customers of the brake suppliers to judge the fidelity and suitability of such simulation techniques for qualification testing.

8.4.1 The major advantages in the use of this type of equipment are as follows:

a. Dynamic compatibility of brake and structure is verified before aircraft installation, provided that the contribution of the wing and fuselage compliance is faithfully replicated by the fixturing supporting the gear, and the effects of roadwheel curvature are compensated.

b. Numerous design discrepancies and compatibilities can be evaluated.

c. It is a relatively simple extension of such simulation to include brake rods used in multi-axle gears.

8.4.2 Some of the disadvantages or limitations in the use of this type of test equipment are as follows:

a. Physical size of the system. Most present test dynamometers do not have the necessary width to accommodate very wide twin gears. It is necessary to install special spacers. When this is done, the inertia requirements may conflict with the width requirements of the system simulation. A good example of this is the simulation of the L-1011 gear by a half-gear equivalent. In this simulation, the roadwheel width associated with the required axle length resulted in an inertia that was 70% higher than the aircraft equivalent. Consequently, the stopping time resulted in prolonging unstable walk events to unrealistic levels, and subsequent post processing by computer models required months to reconcile. This is apparently not a disadvantage with electric inertia dynamometers (see 8.3.1).

b. The aircraft must be in an advanced state of construction to utilize actual gear components. Simulation requires detail knowledge and availability of actual strut parameters and hardware. Cost and size can be a limiting consideration of this testing technique. Regardless of the availability of actual landing gear hardware, the fact remains that gear dynamics are very much dependent on aircraft wing and fuselage compliance, which must still be represented by auxiliary structure in such installations. These can be the major concern in the fidelity of such rigs.

c. The planning of tests and the analysis of test results should take into account road wheel curvature, which can affect tire parameters and effective strut fore and aft stiffness.

d. Truck-type gears cannot be tested with this technique. Or any other dynamometer techniques for that matter.

e. Though it has been used for shimmy testing, some dynamicists feel it is inappropriate, due to roadwheel curvature effects which tend to add stability.

8.5 Pitch-Plane Gear Simulation on Roadwheel Dynamometers

The interaction of the braking system with vertical, yaw, roll, and shimmy modes of the landing gear are often of secondary interest compared to the landing gear's walk, squeal, and whirl stability. Accurate simulation of the pitch-plane dynamics of the landing gear can be accomplished by replacing the relatively rigid axle restraint on conventional dynamometers with a flexible one representing the gear support flexibility. When the stiffness is properly sized, the resulting low frequency angular degree of freedom is dynamically equivalent to the fore-aft gear walk mode, based on the fact that the equations of pitch-plane motion of both systems are the same when second order effects are eliminated. This technique and the design criteria for dynamic equivalence are discussed in SAE Paper 851937, Laboratory Simulation of Landing Gear Pitch-Plane Dynamics.

8.5.1 The major advantages of this technique derive from its relative simplicity and the fact that the simulation is accomplished without fore-aft gear motion:

a. Roadwheel curvature effects on walk vibration are eliminated. The curvature effect on tire load vectors (see 8.2g) of course remains.

b. Actual landing gear hardware is unnecessary.

c. The brake assembly is isolated from the masking effects of conventional dynamometer fixture modes.

d. Damping can be reduced to a small fraction of full-gear damping, thus deliberately destabilized configurations can be tested to verify on-aircraft stability margins and help develop accurate computer models. If gear walk damping is reduced, however, be aware that the stability of squeal and chatter modes may be artificially increased (see 3.16).

e. The equipment and its installation are relatively inexpensive.

f. The equipment is a major advantage for antiskid tuning in the laboratory, because the actual tire and brake friction are represented. No computer simulations can replicate the variability of brake and tire friction affected by speed and temperature changes

8.5.2 Some of the limitations in the use of this simulation technique are:

a. It is limited to a single brake. Thus, interactions between brakes on a gear cannot be evaluated. Note that this is a limitation shared with other techniques.

b. Only one gear walk mode can be simulated with each fixture. Thus, the effect of interactions with other gear modes cannot be evaluated. Again this is not a unique limitation.

c. Shimmy modes cannot be simulated.

d. Some investigators believe that this technique does not properly account for coupling effects on brake squeal due to tire-flex damping. That is to say that some investigators believe the second-order effects ignored are truly significant, especially with regard to the use of this simulation technique for antiskid tests. Because of this, those investigators regard the simulation to be conservative in some respects. The reasoning is based on the fact that the tire and axle do not translate during a simulated walk event, because the translation is deliberately incorporated in the angular motion of the axle. Thus, wheel speed variations are really gear velocity variation. Antiskid engineers are very concerned with the phasing of the wheel speed variation during gear walk events, because they really have only wheel speed variation to base their control strategies on. The key point of their opinion that this technique is flawed is based on the belief that the lag between gear walk motion and wheel speed motion is not represented accurately due to the absence of tire inertial effects on that lag. This is a viewpoint that is not unanimous, however, and future modeling work may clarify the issue.

e. For purposes of antiskid performance verification, this technique does not adequately represent the aircraft runway roughness, traction, or irregularities. Then again, it must be said that no dynamometer techniques can do so, with the possible exception of the internal-drum dynamometer in the WPAFB laboratory, or track facilities. Even the internal drum dynamometer cannot replicate all surface characteristics of a runway, though. A common drawback of all dynamometer testing is the fact that the tire continuously rolls over the same surface, which changes unnaturally because of tire rubber buildup that does not occur on runways. Many experimenters acknowledge such imperfect representation of the runway and still value the dynamometer as a tool to flex the capabilities of their products.

f. The technique is not appropriate for simulation of the walk mode of multi-axle gears, because the brake rod's essentially horizontal orientation cannot be easily maintained independent of the simulated walk motion. There is a patented method of extending the capability of pitch-plane simulators, however, to accomplish simulation of both walk and brake modes of bogie gears. But it has not been found justifiable to date, because the generally accepted high walk mode stability of bogie gears has made simulation of that mode unnecessary.

g. The pitch mode of trailing-link gears is difficult to simulate.

8.6 Subscale Friction Testers

Subscale friction test apparatus have been used for a number of years and different purposes. Subscale testing can be of extreme value if conducted early in a program and if focused on the right deliverables. For example, a great deal can be learned about the variations in friction that are possible with a given carbon material in subscale tests because the averaging effects of the built up brake are not present. This can help the brake manufacturer understand a proposed material's sensitivity to desorption, a thermochemical reaction that has a very large influence on wear and friction variation. Because they are economical to conduct, subscale tests can be a powerful and financially attractive screening tool for evaluating tailoring and downselecting manufacturing processes. Another use is to evaluate proposed modifications to existing carbon manufacturing processes. While not wholly comprehensive, subscale test data can help reveal if a proposed process change is an obvious concern that warrants further verification.

For some time airframe manufacturers have been reluctant to accept the use of subscale tests for lot to lot quality assurance verification, reasoning that structural and thermal effects in the built-up brake cannot be simulated with subscale coupons. Similarly, there is reluctance to accept subscale data predictions for wear. Nonetheless subscale testing has a definite place and usefulness. Two SAE A-5 papers, 2005-01-3437 and 2005-01-3436 discuss subscale test procedures, test data, and data interpretation methods in detail.

8.7 Dyno Simulation of Brake Systems with Brake Rod Torque Reaction

There are patented techniques for simulating gear walk of multi-axle systems with pitch-plane techniques, but they have not been found necessary to implement, because studies have shown that the gear walk mode is relatively stable in such systems due to the high damping provided by the frictional interfaces between the linkage component and the pins that join them. Consequently some investigators ignore the walk mode of multi-axle gears and instead concentrate on identification of the least stable brake modes and construction of fixtures to represent them. In these situations, computer models of the system are critical to identify the least stable brake modes, and fixtures are designed to replicate them. Some of the literature does document the use of overhead rig simulation to test walk modes of systems that include brake rods.

8.8 Pilot in the Loop (PIL) Simulations on Dyno Tests

Anyone who has ever visited an amusement park such as Universal Cities or Epcot knows that simulation of deceleration and directional motion is not a rocket science. Flight simulators are a big business. Braking simulators can be devised to be so as well. It is not a common feature of most brake dynamic labs, but it is definitely an area where simulation fidelity can be enhanced. Pilot feel is a very significant element of customer satisfaction, especially for the regional-business market, and is very difficult to quantify. It is an area ripe for development. This is especially true for those aircraft manufacturers who rely on the opinion of the pilot for their assessment of performance quality. It is a simple matter to arrange for a pilot to operate the brake on a dynamometer with foot-pedals, but it is more difficult to arrange feedback that represents “feel”. This is a gap in dynamometer simulation fidelity. It should be at least a goal for aircraft manufacturers to strive to quantify pilot feel, and for brake laboratories to develop methods to simulate it accurately.

8.9 Track Facilities

There are facilities available which permit mounting of an actual gear or major component, which is then propelled down a track to test velocity and loaded in a manner similar to that experienced on the aircraft. The two known facilities that have this or similar capabilities are NASA at the Langley Loads Track at Langley AFB, Virginia, and the U.S. Navy at the Naval Air Engineering Center, Lakehurst, New Jersey. They differ in propulsion method and method of applying vertical load. However, the NASA Langley facility is more experienced with many more tests being conducted in this technical area. Therefore, the facility is more advanced in sophistication of test techniques.

8.9.1 The major advantages in the use of this type of equipment are as follows:

a. The actual gear can be tested along with specific component or subsystem, or both.

b. This method of testing will permit running over different types of runway surfaces and introducing water and slush effects.

c. Dynamic interplay between the brakes and gear can be evaluated

d. NAEC and NASA facilities have 250 knot speed range

e. NASA facility employs very experienced technical support to assist in interpretive analysis.

f. Provides realistic excitation from runway surface

g. This is the only facility that can test multi-axle gear stability.

8.9.2 Some of the disadvantages or limitations in the use of this type of test equipment are as follows:

a. Availability of NASA track is restricted to NASA schedule

b. Vertical load capacity of 70,000 to 160,000 lb (31,000 to 73,000 kg) maximum on NASA facility

c. Inexperience in testing at Navy facility

d. Track length limited to 2800 ft (850 m) at NASA facility.

e. Expense of redesigning test carriage for each application at NAEC facility.

f. Production design probably released before test is complete. Data generation occurs much too late to permit design modifications at a meaningful time.

g. Number of runs limited by recycle time of the facility.

h. The structural dynamics of the aircraft wing and fuselage restraint to the gear still have to be accounted for accurate simulation of gear mode dynamics, because they dominate the elastic restraint to the landing gear.

8.10 Drop Test Facilities

There are drop towers at many landing gear facilities, airframers, and at WPAFB laboratories. In some of them (or maybe all?) the tires can be spun up to landing speeds. In these tower tests, the gear is mounted to a mass representing the aircraft load per gear and it is dropped to the ground to test reaction to landing loads. Some of the data collected from such tests includes the transient fore-aft response of the gear to the tire friction resulting from stopping the rotating tires. This data is sometimes used to infer landing gear walk mode frequencies. However, such frequencies are usually quite dependent on the wing and fuselage restraint to the gear, and if they are not properly represented in the installation, the gear walk frequencies measured can be regarded as suspect.

9. ANALYTICAL RESOURCES: COMPUTER

Analytical resources are computer programs and techniques that aid in the process of identifying, diagnosing, or solving vibration problems, or optimizing the performance of dynamic systems such as the braking and control of aircraft landing gear. There has been an explosion of commercial software for dealing with dynamic systems over the years, and much can be written about the evolution of that software, but that is beyond the scope of this document. The purpose of this section is to describe the types and purposes of analyses that are helpful for engineers to assist in providing dynamically stable braked landing gear systems, and optimizing their dynamic performance. It will be acknowledged here, however, that the difficulties involved in actually using many commercial software packages in this climate of their perpetual evolution are not minor. No single piece of software can really do it all, and many companies write their own proprietary code for the analysis of the idiosyncrasies of particular systems.

9.1 As a supplement to testing, computer analysis can be valuable for assessing the effects of assignable variables on brake dynamic characteristics. Utility of the information from this tool is dependent upon an accurate assessment of both strut and brake design parameters. Many investigators and their customers believe that neither tests nor computer simulation alone are adequate for establishing confidence in performance on an aircraft.

9.2 Computer simulation regarding braking system dynamic stability is primarily effective when used as an extension of the tests performed for the purpose of assessing the effects of compromises made on any test site. These include:

a. Roadwheel curvature effects on dynamometer-simulated gear walk stability

b. Tire load effects, such as self-energizing on some dynamometers, but are absent on the aircraft

c. Runway surface characteristics, which are absent on most dynamometers.

d. Understanding modes of the system which may not be measured conveniently for one reason or another.

9.3 Computer simulation regarding antiskid compatibility and efficiency is more complex in the sense that the system modeled includes time and speed-dependent gear position, which gear walk modes are sensitive to, and variable runway profiles, contamination, and roughness. It is less complex in the sense that the simulation need not concern itself with brake dynamic modes such as squeal and whirl at all. The structural dynamics models for dynamic stability and brake control tend to overlap.

Some practitioners believe it is critical that an antiskid system is capable of releasing brake pressure (and drag force) at a higher rate than a skid can occur, reasoning that the system is unable to control tire skids otherwise. Others believe that it only need be fast enough to turn the wheel around before the skid becomes too deep, and in fact must actually force a skid to ascertain where the peak of the mu-slip curve is for any particular runway condition. The system must also be able to rapidly reapply pressure to maintain high efficiency. To do this requires knowledge of specific parts of the braking system.

First, it's critical to understand and model the feedback system. This includes the accuracy of the sensor, including its dynamic accuracy. Typically, this is related to the resolution, in radians, of the angular displacement of the aircraft's wheel. Velocity and acceleration are derived from this fundamental measurement as well as the change in time (slope). (Time is usually at least an order of magnitude more accurate in both analog and digital systems than other errors. Therefore, errors related to time measurement will be ignored.) In an analog computer, although very fast, filter and charge delays have to be considered in the dynamic response of the feedback signal. In a digital system, models should also include computational delays and quantization errors.

A typical feedback system therefore must be simulated by modeling the accuracy of the sensor and any inherent systemic and random errors. Transport (or latency) associated with signal conditioning, quantization or computation of the fundamental signal must be considered too. The result is a model that can be used to simulate the dynamic and error characteristics of the feedback mechanism.

Next the computational errors and delays are modeled. Computational errors can be due to, noise (from many sources), quantization of analog signals, round off, etc. depending on the computational engine employed. Additional latencies, such as computational time to derive data (like velocity) or antiskid algorithm execution time must be included in the overall model.

Finally, the actuator's dynamic characteristics are modeled. Typically the actuator is a hydraulic servo valve or in the case of an electric brake, an electric motor and gear train that drives an aircraft brake, wheel and tire. For example, if the controller determines that the drag force must be lowered significantly to avoid a skid, then the command must be sent to the servo, the valve and hydraulics must respond to release pressure, the brake's single ended (spring return) pistons push back, brake friction surfaces relax, drag force falls and the energy stored in the spring of the rubber tire, gear strut and airframe are released. There are many different gear configurations, trailing links, twin gears (post) and bogie gears. In each of these a specific gear model is created. The way the normal force is applied is an important modeling effort for antiskid and brake control systems. Fore-aft structural dynamics are significant for twin gears, but less so for bogie gears. Lateral forces are typically modeled by some practitioners, but are usually less significant than the other axes. In all cases the geometry of the gear, joint characteristics, material properties, structural dynamic characteristics, viscous damping of the shock strut, aircraft center of gravity, and weight play a significant factor in development of an antiskid control system

Each of the components that make up the control loop must be modeled to some level of fidelity. The latency (transport delays), step response characteristics, frequency response, harmonic sensitivity, frictional characteristics, structural dynamics, damping, mass, displacement, velocity, acceleration, fluid properties, and all compliances need to be characterized. That said, there is a school of thought that favors the simplest possible system representation because such a formulation is much easier to understand and optimize. In this approach, greater detail and fidelity is only needed if the performance optimization requires it.

Obviously some characterizations are more important for some components than others. It requires real-life systems to verify these models accurately represent the most significant aspects of the system. This is done by a successive scale up of real aircraft hardware and removal of the math models. If at every stage of testing, workstation, hydraulic mockup, dynamometer, and finally aircraft the model and real hardware can be successfully interchanged, then an accurate system has been simulated. However, to realistically achieve this requires a good understanding of the limitations and correlations to the testing environment.