Electro-Mechanical Brake

ABSTRACT

Brake performance can be divided into two distinct classes:

1) Base brake performance
2) Controlled brake performance.

A base brake event can be described as a normal or typical stop in which the driver maintains the vehicle in its intended direction at a controlled deceleration level that does not closely approach wheel lock. All other braking events where additional intervention may be necessary, such as wheel brake pressure control to prevent lock-up, application of a wheel brake to transfer torque across an open differential, or application of an induced torque to one or two selected wheels to correct an under- or over steering condition, may be classified as controlled brake performance. Statistics from the field indicate the majority of braking events stem from base brake applications and as such can be classified as the single most important function. From this perspective, it can be of interest to compare modern-day Electro-Hydraulic Brake (EHB) hydraulic systems with a conventional vacuum-boosted brake apply system and note the various design options used to achieve performance and reliability objectives.

INTRODUCTION

What is EHB system?

The next brake concept. This system is a system which senses the driver's will of braking through the pedal simulator and controls the braking pressures to each wheels. The system is also a hydraulic Brake by Wire system.

Many of the vehicle sub-systems in today's modern vehicles are being converted into "by-wire" type systems. This normally implies a function, which in the past was activated directly through a purely mechanical device, is now implemented through electro-mechanical means by way of signal transfer to and from an Electronic Control Unit. Optionally, the ECU may apply additional "intelligence" based upon input from other sensors outside of the driver's influence. Electro-Hydraulic Brake is not a true "by-wire" system with the thought process that the physical wires do not extend all the way to the wheel brakes. However, in the true sense of the definition, any EHB vehicle may be braked with an electrical "joystick" completely independent of the traditional brake pedal. It just so happens that hydraulic fluid is used to transmit energy from the actuator to the wheel brakes. This configuration offers the distinct advantage that the current production wheel brakes may be maintained while an integral, manually applied, hydraulic failsafe backup system may be directly incorporated in the EHB system. The cost and complexity of this approach typically compares favorably to an Electro-Mechanical Brake (EMB) system, which requires significant investment in vehicle electrical failsafe architecture, with some needing a 42 volt power source. Therefore, EHB may be classified a "stepping stone" technology to full Electro-Mechanical Brakes.

HYDRAULIC DESIGN CONSIDERATIONS

FAILSAFE AND SYSTEM COMPLEXITIES

Analogous to a vacuum boosted system in base brake mode, EHB supplies a braking force proportional to driver input, which reduces braking effort. The boost characteristics also contribute to the pedal "feel" of the vehicle. If the boost source fails, the system resorts to manual brakes where brake input energy is supplied in full by the driver. As would be expected, the pedal forces vs. vehicle deceleration characteristics are significantly affected.

This is shown by the input **pedal force vs. Brake line pressure** output in Figure 1 of a typical vacuum boosted vehicle.

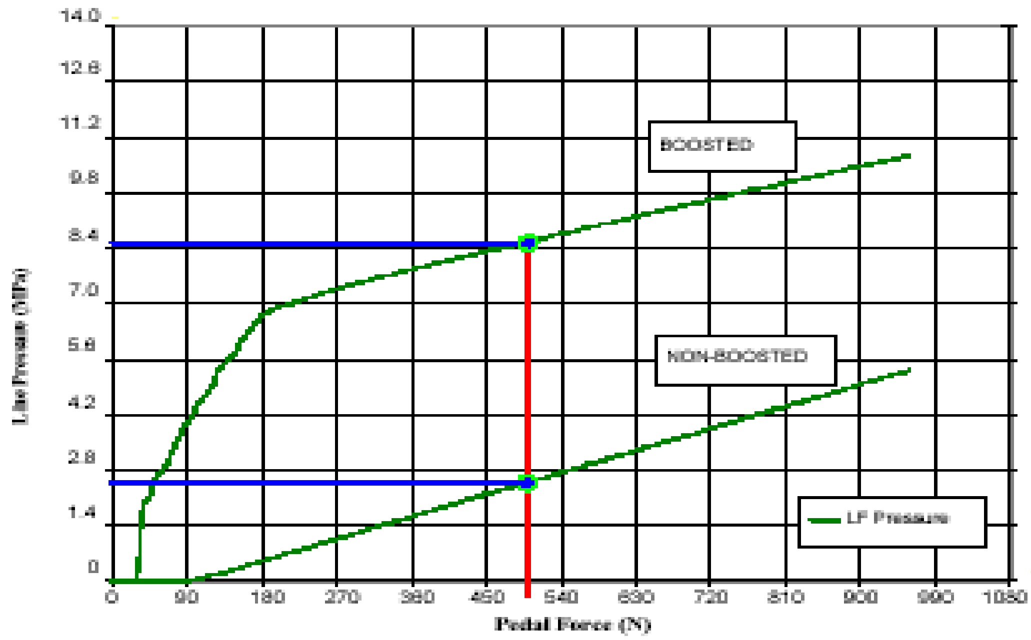
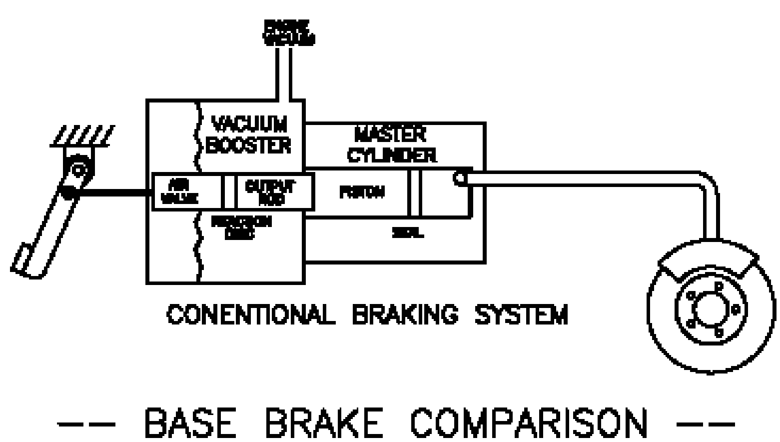


Fig. 1: Boosted vs. Non-Boosted Brake Output

Looking at a comparison using the failsafe pedal force input limit of 500 N, the difference between the resulting brake line pressure is 2.5 MPa unboosted vs. 8.5 MPa boosted. This correlates to an approximately proportional difference in vehicle deceleration. In this example there approximately correlates to 0.3 g's decel. Unboosted, and 0.9 g's boosted. With EHB systems, there is room to improve this performance, but only at the expense of pedal travel, which becomes a hydraulic lever arm of sorts. For example, to achieve a higher decel from 0.3 g to 0.5 g in failed system, the pedal travel may have to increase from 50 - 75 mm to perhaps 150 mm, which is about the practical limit for brake pedal travel. Thus, due to the consequences of boost failure, careful attention must be paid to component system design irrespective of the type of mechanism employed.
A comparison between the conventional vacuum boosted system and an EHB system is shown in Figure 2.



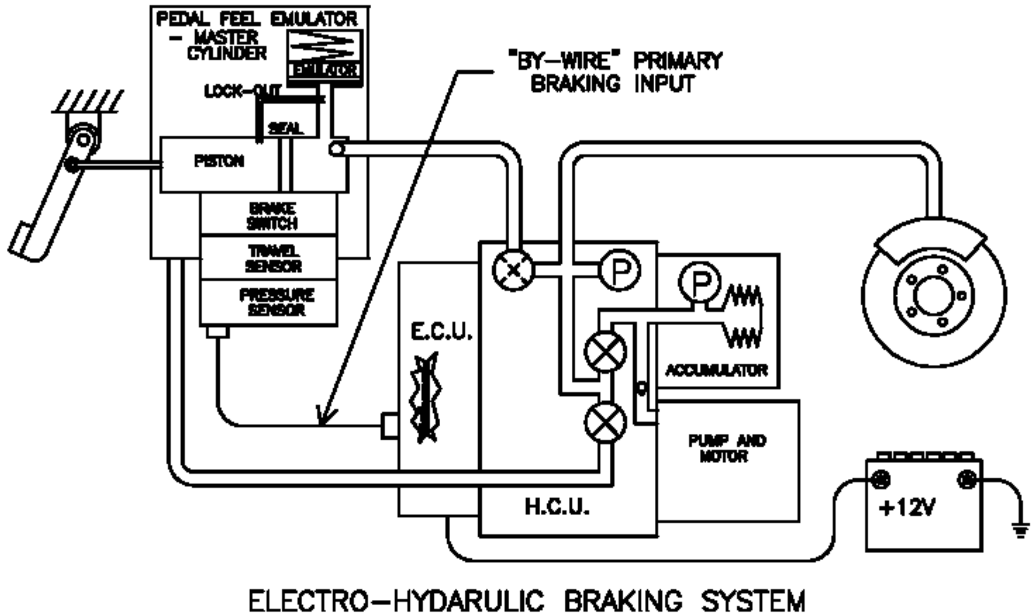


Fig. 2: Single Channel Complexity Comparison for Base Brakes

The conventional system utilizes a largely mechanical link all the way from the brake pedal through the vacuum booster and into the master cylinder piston. Proportional assist is provided by an air valve acting in conjunction with the booster diaphragm to utilize the stored vacuum energy. The piston and seal trap brake fluid and transmit the hydraulic energy to the wheel brake.

Compare this to the basic layout of the typical EHB system. First, the driver's input is normally interpreted by up to three different devices: a brake switch, a travel sensor, and a pressure sensor while an emulator provides the normal pedal "feel". To prevent unwanted brake applications, two of the three inputs must be detected to initiate base brake pressure. The backup master cylinder is subsequently locked out of the main wheel circuit using isolation solenoid valves, so all wheel brake pressure must come from a high-pressure accumulator source. This stored energy is created by pressurizing brake fluid from the reservoir with an electro-hydraulic pump into a suitable pre-charged vessel. The accumulator pressure is regulated by a separate pressure sensor or other device. The "by-wire" characteristics now come into play as the driver's braking intent signals are sent to the ECU. Here an algorithm translates the dynamically changing voltage input signals into the corresponding solenoid valve driver output current waveforms.
As the apply and release valves open and close, a pressure sensor at each wheel continuously "closes the loop" by feeding back information to the ECU so the next series of current commands can be given to the solenoid valves to assure fast and accurate pressure response.

It is obvious the EHB system is significantly more complex in nature. To address this concern, numerous steps have been taken to eliminate the possibility of boost failure due to electronic or mechanical faults. In the ECU design, component redundancy is used throughout. This includes multiple wire feeds, multiple processors and internal circuit isolation for critical valve drivers. The extra components and the resulting software to control them, does add a small level of additional complexity in itself. Thermal robustness must also carefully be designed into the unit, as duty cycles for valves and motors will be higher than in add-on type system. Thus, careful attention must be given to heat sinking, materials, circuit designs, and component selection. Special consideration must be given to the ECU cover heat transfer properties, which could include the addition of cooling fins. On the mechanical side there is redundancy in valves and wheel brake sensors in that the vehicle may still be braked with two or three boosted channels. In regards to the E-H pump and accumulator, backup components are not typically considered practical from a size, mass, and cost viewpoint. However, these few components are extremely robust in nature and thoroughly tested to exceed durability requirements.

The second area used to evaluate potential failure concerns is through the study of past warranty data of similar systems. The system chosen for comparison was an early ABS system integrated into a hydraulic booster. The data was collected from two different North American passenger vehicles built in the early 1990's at a 12-month PPM level. Both vehicles utilized a central hydraulics unit that in turn supplied power to the hydraulic brake booster and ABS block. The data in Table 1 represents an approximation of warranty comparison based upon an averaging of returns from both vehicle lines. Note any vehicles requiring a vacuum pump (such as diesel) would also have to take those failures into consideration for the baseline calculation.

Master Cylinder/Booster PPM	x	(Baseline)
Wiring	2.2x	
Sensors	0.5x	
Hydraulic Block	0.5x	
E-H Pump	0.3x	
ECU	0.7x	
TOTAL	4.2x	

Table. 1: Defective Parts per Million Vehicles (PPM) Comparison

Although the total failure frequency is higher, many of the failures may illuminate the fault light on the dashboard, but would not affect the base brakes. For example, there is sufficient redundancy in sensors and in hydraulic valve block components that the vehicle would still maintain boosted braking on the unaffected wheels. As previously noted, multiple feed wires and grounds are being employed which could therefore negate many of the concerns related to wiring harness defects. In similar fashion, many of the ECU failures would also not result in loss of the base brake boost function. Thus, when adding the E-H pump and some smaller percentage of wiring and ECU failures, the total combination that would affect base brake performance could be expected to be closely the same or even less than the conventional system. This type of comparison using ten-year-old data is only a guideline since modern technology and manufacturing methods continue to make both electronic and mechanical components more reliable.

BASE NON-ISOLATED HYDRAULIC CIRCUITDESIGN

Designing for base brake systems poses a challenge to be able to utilize the same hydraulic components to meet two extreme braking conditions. One is a panic mode situation, where an extreme amount of hydraulic energy needs to be transmitted through the brake system in a very short amount of time in order to apply the wheel brakes as quickly as possible. Current specifications typically call for reaching pressures at the wheel brakes of approximately 8 MPa in 120 milliseconds or less. For a typical midsize vehicle, this translates into average power requirements of 1,200 watts with flow rates in excess of 40 cm3/s at each wheel brake. The second challenge is to be able to modulate pressures in a very stiff system when the brakes are applied. Pressure resolution of approximately 30 kPa is required. The problem of control becomes apparent. Very small quantities of brake fluid must be sufficiently modulated to give a good base brake pedal feel. To meet the requirements the selected control valves must be designed to have very good response time characteristics (i.e. < 10 millisecond) with relatively unrestricted flow paths. The basic means to achieve wheel brake modulation comes from using two normally closed proportional control valves per wheel brake. The apply valve regulates flow from the high pressure central accumulator to the wheel brake, while the release valve regulates flow from the wheel brake back to reservoir, which is maintained at atmospheric pressure. A typical single wheel schematic is shown in Figure 3.

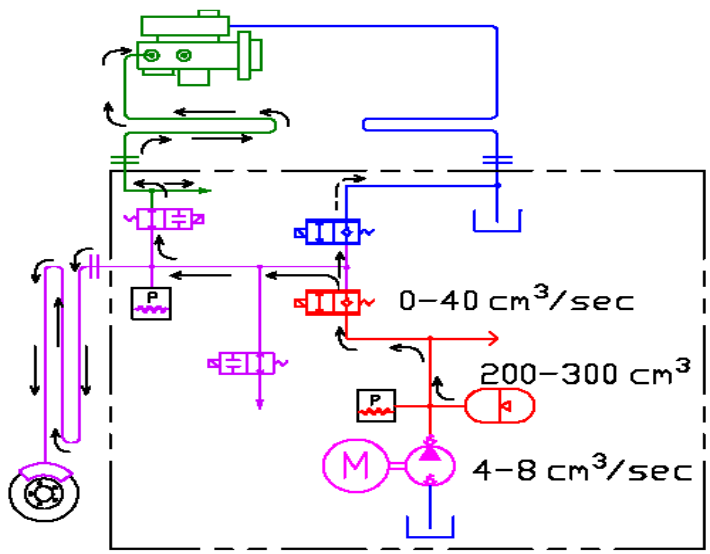


Fig. 3: Typical Non-Isolated Single Wheel Schematic

For failsafe operation, it becomes necessary to include an isolation valve between the pedal feel emulator -master cylinder (PFE-MC) assembly and wheel brake. Its functions include blocking the driver's manual output pressure during a boosted apply as well as providing a vent path back to reservoir when the brakes are not activated. Additionally, a balance valve is placed between wheel brakes on each axle to prevent momentary pressure imbalance during panic-type base brake applies. This design is especially well suited for front/rear (T-T) type of systems since the master cylinder circuits are also allocated to each axle. In this design the accumulator circuit leads directly into the master cylinder and wheel brake circuits through the apply valve as is shown by the arrows on the graph.

Most EHB's utilize brake fluid stored in a central, gas pressurized accumulator. Typical sizes for a North American midsize vehicle may range from 200 to 300 cm3. A typical accumulator pressure operating range may be 16 MPa (pump turn on) to 18 MPa (pump turn off). The gas most commonly used is nitrogen due to its relatively low cost and relative inertness. The nitrogen gas is kept separated from the brake fluid by either an elastomeric or metallic membrane or diaphragm. Most elastomeric membranes have a single, curved shape which folds back upon itself as the device fills with brake fluid. The all-metallic type of membrane is usually in the shape of a bellows with a number of folds (much like an accordion) and relies on thin plate bending with large deformations and low stress levels to accomplish the task of displacing brake fluid. Due to the small size of the nitrogen molecules, permeation is also a factor to consider, particularly with elastomeric types of diaphragms. The nitrogen gas will typically find its way through most elastomeric materials, and enter the molecular "pores" within the spaces of the pressurized brake fluid volume until all of the voids are filled. At that time, equilibrium is re-established and finally permeation diminishes.

High temperatures may also accelerate this phenomenon. With the latest multi-layer proprietary materials being developed, certain accumulator manufacturers are claiming improvements in permeation reduction of five to six times. Thus, estimated useful life is now in the range of 10 – 15 years.

Failure Mode Considerations – Non-isolated Circuit

Returning to considerations for the high pressure accumulator. This device stores significant amounts of energy, typically as much as 1,700 watt-seconds. This has the advantage of being able to supply numerous (i.e. 5 –15) reserve stops should the electro-hydraulic pump fail. It was previously noted the pressurized nitrogen gas was separated from the brake fluid by one of two types of diaphragms. Even though the latest versions of both these devices have become extremely reliable through years of development, it might not yet be possible to classify either of these types of units as a true zero defect type of device since manufacturing quality must always be considered. Therefore, the consequence of diaphragm failure must be investigated. A test was devised utilizing a non-isolated wheel brake circuit of the type shown in Figure 3. A carefully constructed accumulator with a small hole punctured in the diaphragm was installed in a vehicle. The brakes were subsequently applied and released at discreet intervals to study any change in operating characteristics. The graph in Figure 4 below shows the status of the measured brake pedal force and travel parameters after 100 powered base brake applies, where functionality was shown to be normal. (The unit was fully checked every 50 strokes.)

The pedal feel emulator-master cylinder in this test had a lockout feature. The bottom curve represents the system in normal powered mode showing the simulated pedal travel. The top curves shows brake performance in failsafe mode. At stroke number 114 of the brake pedal, the diagnostics of the ECU detected a "pressure out-of-bounds" failure indicating base brake output pressure was no longer able to follow the driver's brake pedal input commands. The system immediately reverted to the hydraulic failsafe backup mode.

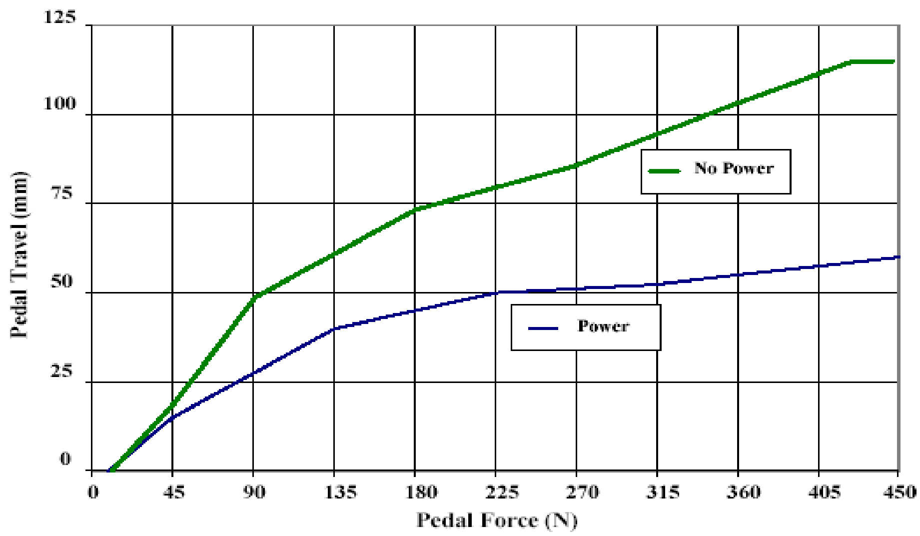


Fig. 4: Vehicle Pedal Force / Pedal Travel Checks at 100 Strokes

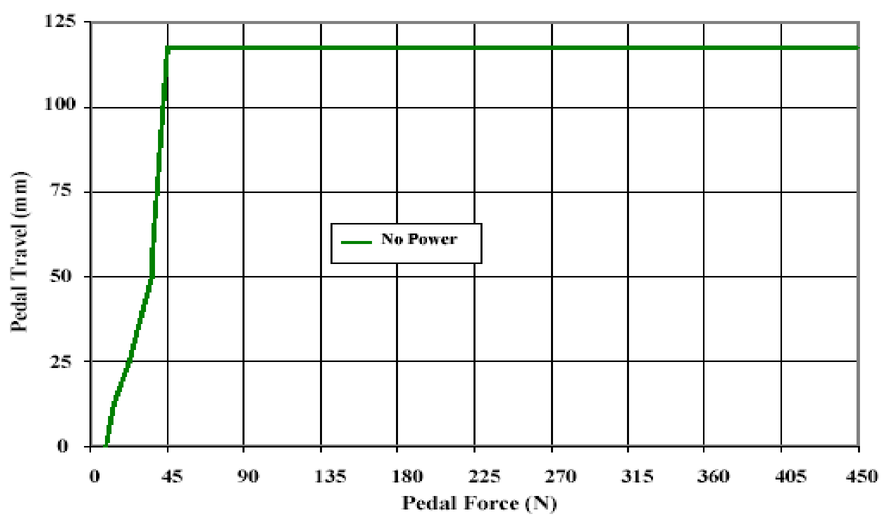


Fig. 5: Vehicle Pedal Force / Pedal Travel Checks at 114 Strokes

Figure 5 is a plot of the brake system performance just one stroke after the failure occurrence. In this case, in the failed system backup mode, the PFE-MC assembly achieved the full travel of 120 mm with an input force of only 45 N. This is the approximate force required to displace the master cylinder springs to full travel with no hydraulic load present and indicates there is minimal pressure output to the wheel brakes.

Attempts were made to find where the nitrogen gas might have migrated to in the system. The circuit was first partially re-bled just between the master cylinder and isolation valve inside the hydraulic control unit (refer again to Figure 3). The results are shown in Figure 6.

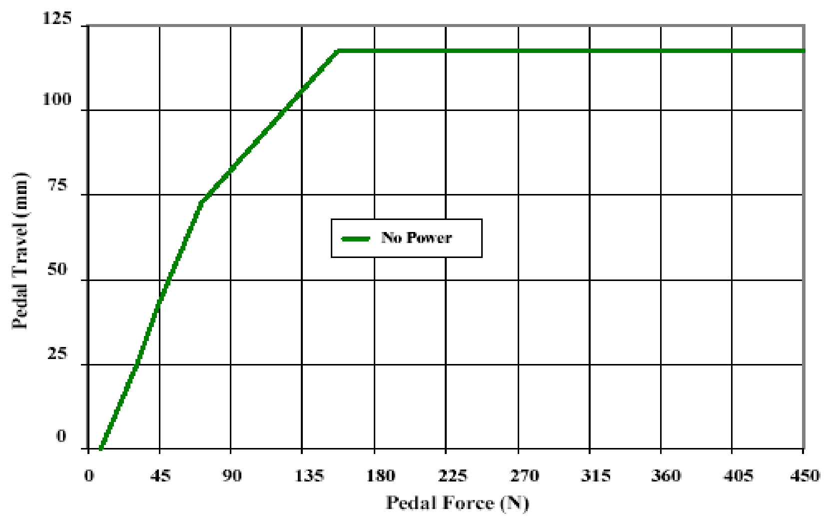


Fig. 6: After Failure – Partial Re-Bleed

The brake pedal force reached approximately 37% of its previous full-travel pedal force value prior to failure. This indicates a significant percentage of the nitrogen also made its way into the HCU-to-wheel brake circuit. As a final step the system was then bled between the HCU and wheel brakes. The results of the re-bleed are shown in Figure 7. Note this curve is nearly identical to that shown in Figure 4 indicating all of the escaped nitrogen gas had been removed from the brake circuits. The test was subsequently repeated with similar results. The gas expulsion could not be anticipated with the diagnostic methods utilized at the time.

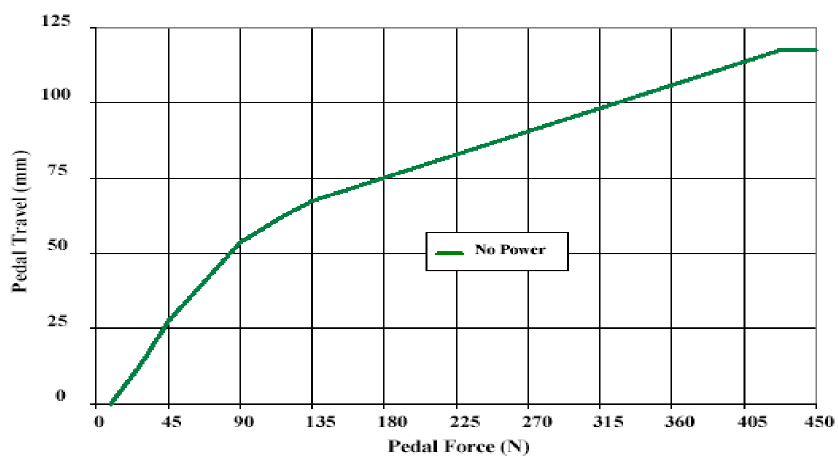


Fig. 7: After Failure – Complete Re-Bleed

These test results verify the severity level of nitrogen discharge due to a defective accumulator diaphragm. The total amount of gas which migrated into the wheel brake and master cylinder circuits was not measured but was at least equivalent to the aster cylinder volume. One technique to assure some level of braking can still be maintained with this type of failure is to allow the pump to run in a continuous mode to eventually compress the discharged gas and subsequently build wheel pressure. The effectiveness of this solution will be determined by pump flow rate and the quantity of gas discharged. For a pump with a nominal flow rate of 8 cm3/s any substantial quantity could result in relatively slow braking response times. If the gas permeates the master cylinder circuit, there could be limited or no force feedback from the pedal feel emulator, which would result in abnormal pedal feel.

One method to maintain a non-isolated circuit is to employ accumulator travel sensing. This is accomplished by incorporating a suitable sensor to track the displacement of the accumulator membrane and works especially well with the metal bellows type of unit. In addition to the travel associated with the normal filling and release of fluid, the metal bellows also has an elastic memory. Thus a defect in the bellows, which allows brake fluid to begin to fill the accumulator can be immediately detected and shut down boosted braking to prevent the possibility of gas expulsion. It is also necessary to know temperature information with this approach to be able to account for pressure variations due to temperature changes. Since the open flow path to the wheel and master cylinder circuits are still present, the sensing method must be robust.

ISOLATED HYDRAULIC CIRCUIT DESIGN

An alternative approach is to utilize an isolated base brake circuit. A typical single-wheel circuit is shown in Figure 8.

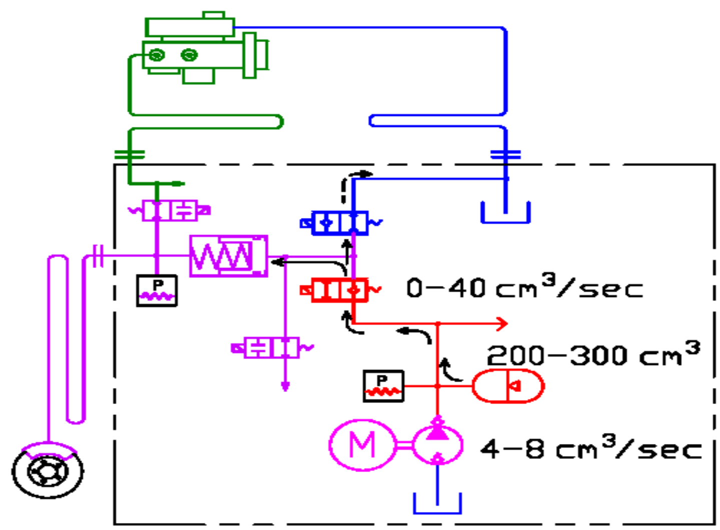


Fig. 8: Typical Isolated Single Wheel Schematic with Isolation Piston

There are some distinct differences between the isolated and non-isolated concepts. The first, and most obvious is the addition of an isolation piston assembly between the pump circuit and each wheel brake circuit which will positively stop nitrogen from entering the wheel brake circuit. The arrows in the graph highlight the restricted flow path. For design simplification, cost reduction, and improved durability, a single seal design is shown although a dual seal, vented design may also be substituted.The other change is that the proportional release valve is normally open. This provides an open flow path back to reservoir, which is independent of the wheel brake circuit. Any escaping nitrogen from the accumulator will have an unrestricted path back to reservoir in failsafe mode. Also note that the balance valve is placed in the pump circuit and may now be a normally open valve for either the T-T or X type of hydraulic circuits.

Another indirect benefit with this approach is the amount of nitrogen gas, which can be permanently trapped, is limited to the drilled holes in the HCU housing, the clearance volume behind the isolation piston, and the volumes around the proportional control valves. Therefore, running the pump to boost brake output in the event of accumulator diaphragm failure is likely to be more effective.

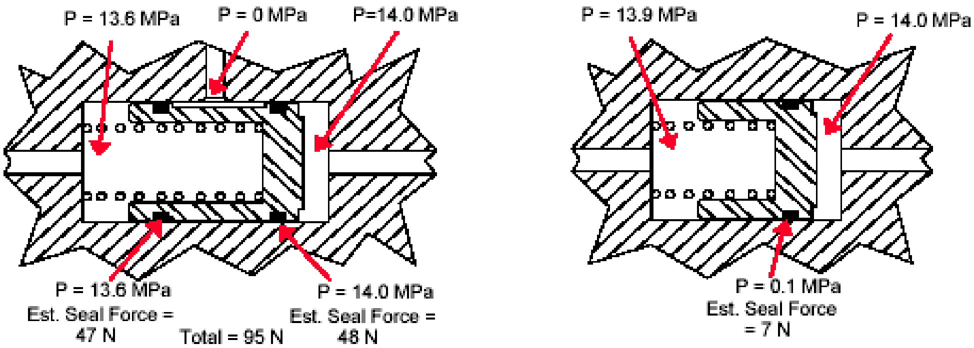
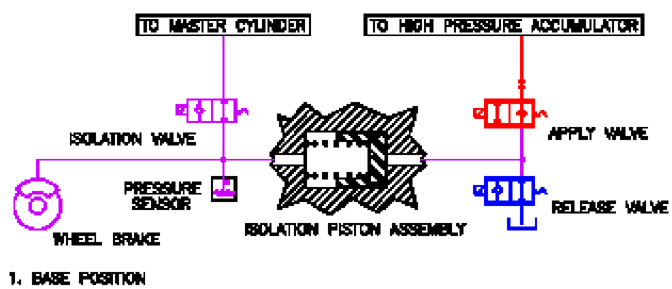
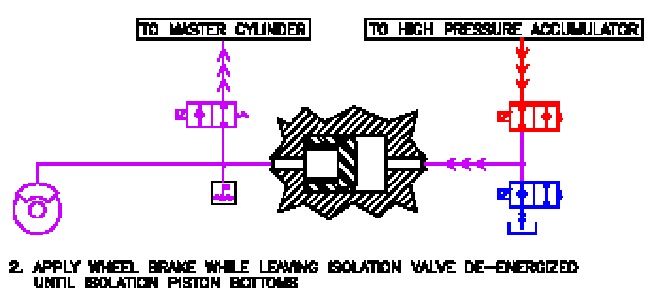


Fig. 9: Isolation Piston Comparison

As isolation piston previously mentioned included only a single seal. Although this solution raises the question of introducing a latent (i.e. undetectable) failure, there are means, both in plant and algorithmically, of detection. The many benefits of using a single seal include occupation of less space, fewer holes to drill, and fewer components, all of which translate to saving cost. In addition, seal stress loading is reduced by maintaining the seal in near pressure equilibrium. This has the added benefit of reducing wear and reducing frictional forces with the bore. The seal force values shown in Figure 9 were derived from prior generic seal testing.





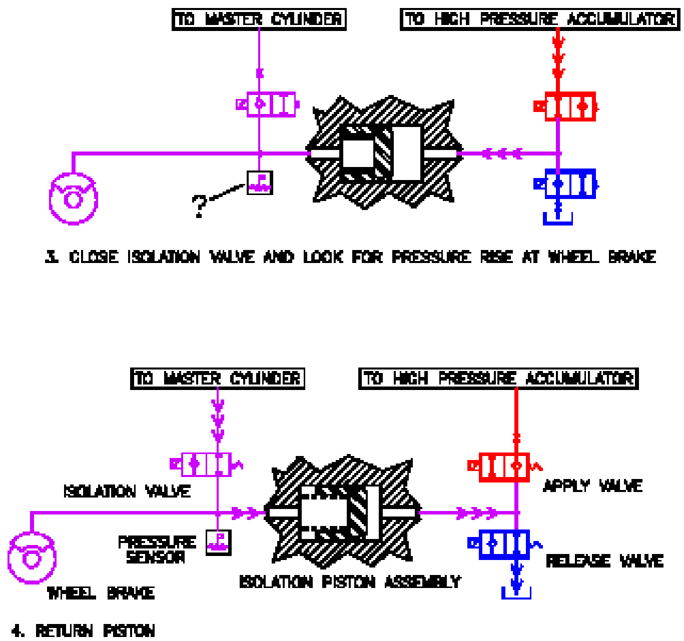


Fig. 10: Simplified Isolation Piston Seal Detection

To be able to detect a missing or defective seal, a suitable algorithm must be employed. State-of-the art assembly plant air testing can detect functionality of a seal in a bore. However, even though this is assured as a new product, it does not assure seal functionality over the ten to fifteen year design life. A leak test procedure, which can be performed on the vehicle, however, may be implemented. The procedure is outlined in detail in Figure 10.
One other step required in assuring a failsafe approach to an EHB design which utilizes isolated wheel circuits, is understanding the volume relationships between the three variable displacement devices: wheel brake, isolation piston, and master cylinder. The typical wheel cylinder and master cylinder may be simply represented as a piston inside a bore as shown in Figure 11.

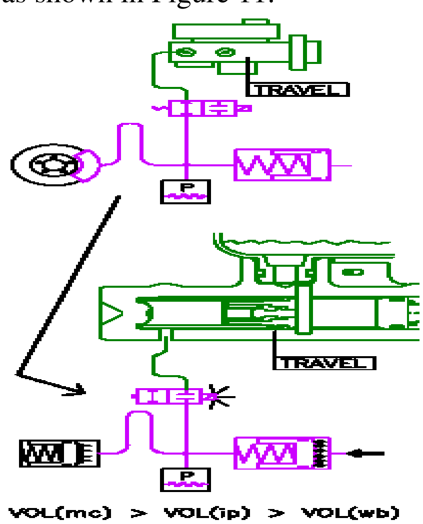


Fig. 11: Relative Volumetric Relationships

By knowing the exact volumetric relationships between wheel brake, isolation piston, and master cylinder, a system can be designed that will assure failed system performance. For example, in the lower half of the diagram above, the piston and cylinder assemblies each graphically represent the volumetric comparisons between the three aforementioned devices. First, consider the brake circuit between the HCU and master cylinder. Note the compliance this circuit may be accurately assessed from knowing the travel position of the master cylinder piston. This is true even when pedal feel emulator displacement is considered, since that is also a fixed pressure-to volume relationship and is known from the component geometry. To further test for system effectiveness, the isolation piston may be applied to a target pressure by appropriate activation of the apply valve. If the wheel brake circuit is not able to achieve the requested pressure, then system compliance is excessive, and appropriate warning can be issued. If, on the other hand, the target pressure is achieved at the wheel brake, the system is functioning properly. The isolation piston may also be utilized as a means to purge the master cylinder circuit as referred to in step no. 2 of Figure 10.

One additional factor for EHB failsafe braking must also be considered which can best be defined as “base brake system compliance allowances”. These are the factors that can contribute to a soft or spongy brake pedal such as from brake pad warpage or distortion from aging or abuse. These allowances should be recognized and included in initial component sizing. In summary, careful employment of isolation pistons, with accompanying diagnostics can be an effective solution for accumulator isolation.

2-WHEEL VS. 4-WHEEL FAILSAFE MODE

Yet another area of failsafe performance which requires consideration is 2-wheel vs. 4-wheel manual operation. Past hydraulically boosted, backup systems in the field have largely been configured with 2-wheel backup. However, utilizing a four-wheel failsafe approach offers more design flexibility, and potential stopping distance reduction on certain classes of vehicles.

The first step is to evaluate suspension and vehicle dynamics. In some vehicles, such as those with front wheel drive and a higher center of gravity, there may be insufficient normal force on the rear wheels during a moderate braking stop in the range of 0.4 to 0.6 g’s that use of rear wheel brakes is not very efficient. However, there are some classes of vehicles, particularly rear wheel- drive cars and trucks, as well as some of the larger front-wheel-drive cars, where there may be sufficient wheel-to-road braking torque available. For those vehicles use of all four wheels for emergency braking could be considered.

The next step is to measure is front vs. rear wheel brake relative efficiency. The variable in question is defined as:

WHEEL BRAKE RELATIVE EFFICIENCY

Rel.Efficiency = OUTPUT TORQUE T (p)
INPUT ENERGY = P * V (p)

In this instance, the input hydraulic energy is the brake line pressure capable of being generated by the driver times the displacement used to achieve that pressure. Thus, from an energy viewpoint, it is better to utilize the wheel brakes on a vehicle that have the highest specific torque output at the lowest displacement. Some rear wheel

brakes (disc or drum) tend to have similar specific torques to the front brakes, yet require less fluid displacement. By taking instantaneous wheel brake torque, pressure, and displacement readings, the measure of wheel brake relative efficiency can be plotted, as shown in Figure 12.
In the example given, it will be more efficient to utilize front and rear brakes together to minimize stopping distances, provided vehicle dynamic conditions are met. Delphi utilizes a in-house computer program to estimate stopping distances. This tool takes into account all of the variables mentioned and combines them with a vehicle suspension model, which captures such factors as weight transfer, to calculate decel and stopping distance.

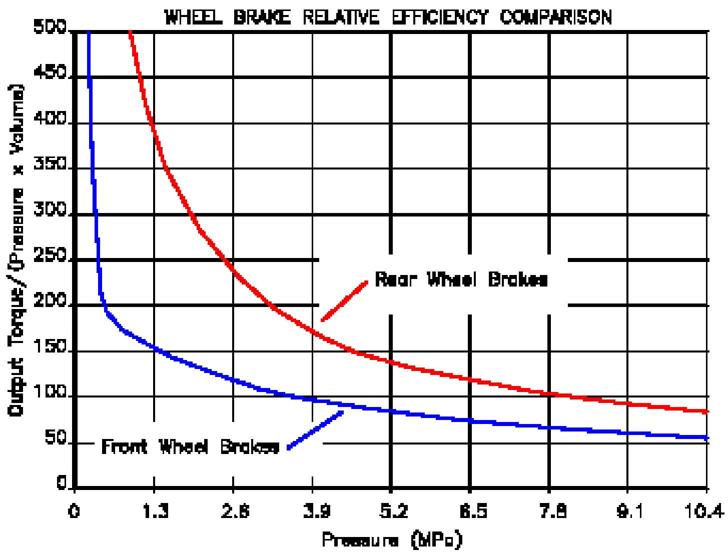
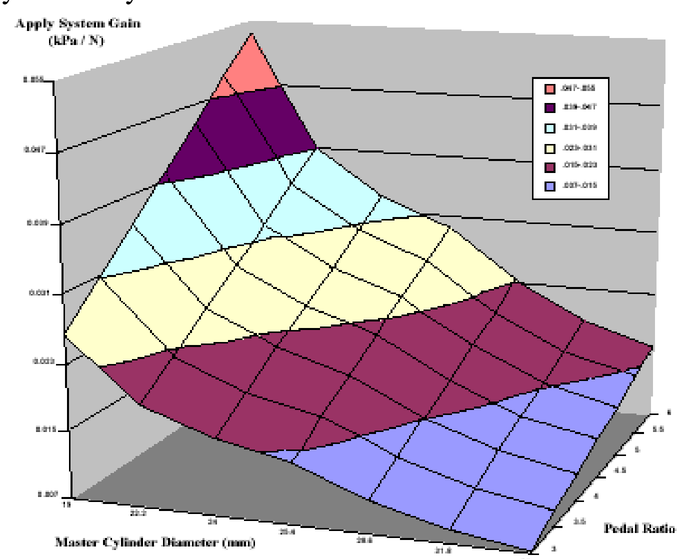


Figure 12: Light Duty Truck Front vs. Rear Brake Comparison

Actual vehicle data is taken to be able to input wheel brake specific torque. Additional information about the base brake system is then fed into the model to calculate a parameter called apply system gain. This variable is defined as the pedal ratio divided by master cylinder bore area.



A 3-dimensional plot of apply system gain is shown in Figure 13. The larger the gain, the larger the mechanical advantage in transferring energy from the driver's foot to the wheel brake, and the larger the pressure which can be applied for a given input force. The tradeoff is pedal travel. It is also increased in proportion to mechanical gain, which ultimately limits the amount of gain for the entire system. An output plot for a typical midsize vehicle is shown in Figure 14. In this single graph, the failsafe performance of the selected system may be analyzed for deceleration and pedal travel at both LLVW and GVW conditions.

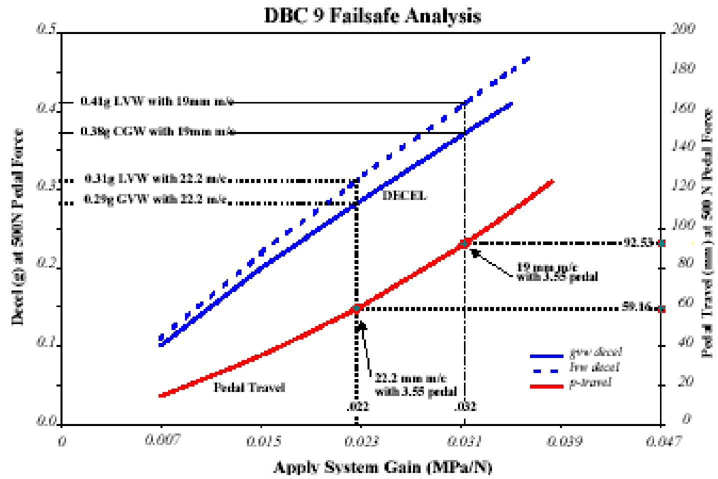


Figure 14: Typical Midsize Vehicle Performance

When evaluating the question of 2-wheel vs. 4-wheel braking, this program was used to evaluate deceleration capabilities of the same vehicle but with either one or both axles active in braking. The estimates for a specific North American light duty truck are shown in Table 2.

FULL-SIZE SUV	2-Wheel	4-Wheel
M/C SIZE	ø 22.2	ø 24.0
PEDAL RATIO	5.3 : 1	4.7 : 1
MECH. GAIN	0.014 MPa/N	0.010 MPa/N
LLVW DECEL	0.41 g	0.54 g
GVW DECEL	0.32 g	0.44 g
PEDAL TRAVEL	110 mm	110 mm

NOTE: Test run at 500 N pedal force.

Table 2: Decel Comparison of 2-wheel vs. 4-wheel failsafe

In this particular case, there was a 32% increase in estimated vehicle deceleration at LLVW and a 37% increase in estimated vehicle deceleration at GVW. It may also be observed higher apply system mechanical gains are required for 2-wheel brakes while trying to achieve equivalent vehicle braking forces. This can create some practical problems. Pedal ratios can become very large which dictate specially designed pedals to maintain a minimum master cylinder push rod arc length, or conversely, master cylinder bores become very small (i.e. less than ø 19.0) with long travel requirements.

As previously noted, not all classes of vehicles necessarily show significant improvement in moving from 2-wheel to 4-wheel backup braking mode. However, designing for only 2-wheel failsafe may exclude optimization on some types of vehicles that can benefit from 4-wheel braking. The range of improvement from the limited numbers of vehicles, which have thus far been analyzed, has run from a few percent to over 30%. In each case, cost vs. performance trade-off should be evaluated when selecting the final design since additional hydraulic components may be required for 4-wheel backup.

PEDAL FEEL EMULATOR LOCKOUT

The graph in Figure 16 shows a typical, customer requested force-displacement curve required for the emulator.

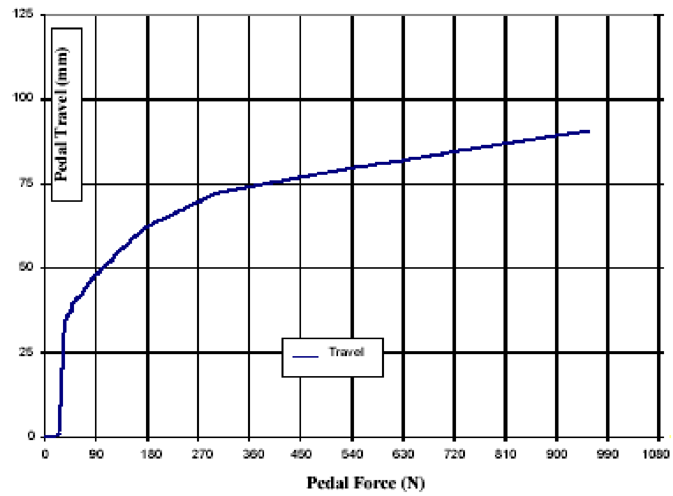


Figure 16: Pedal Force Emulator – Pedal Force vs. Pedal Travel

Figure 17 shows a typical hardware arrangement to meet the pedal feel requirements. This unit consists of a master cylinder with emulator piston and spring assembly. As the driver's foot applies the brake pedal, an input push rod displaces the primary master cylinder piston, while at the same time the isolation valves in the HCU are commanded to close. This blocks both primary and secondary master cylinder outlet ports. The secondary piston becomes locked in place due to the trapped fluid. The fluid contained by the primary piston is displaced into the drill path, which leads to the emulator assembly. As pressure continues to build, the spring begins to deform under the load from the hydraulic pressure acting on the surface of the piston. This causes additional displacement that allows the brake pedal to move in proportion to the force exerted by the driver. The force vs. travel characteristics can be "tuned" to customer directives. If the vehicle is required to stop in failed system, the isolation valves remain open so that fluid is permitted to flow to both the wheel brakes and the emulator.

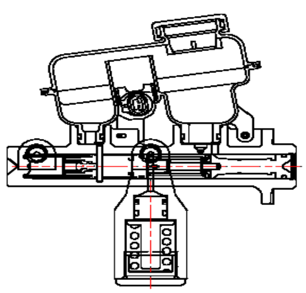


Figure 17: Pedal Force Emulator – Master Cylinder (PFE-MC)

Since emulator piston deflection occurs at relatively low pressures, the compliance of the wheel brakes and emulator are additive, which results in additional pedal travel. Thus the driver's total available input energy for failsafe braking is reduced by the additional emulator displacement which does not contribute to vehicle braking.

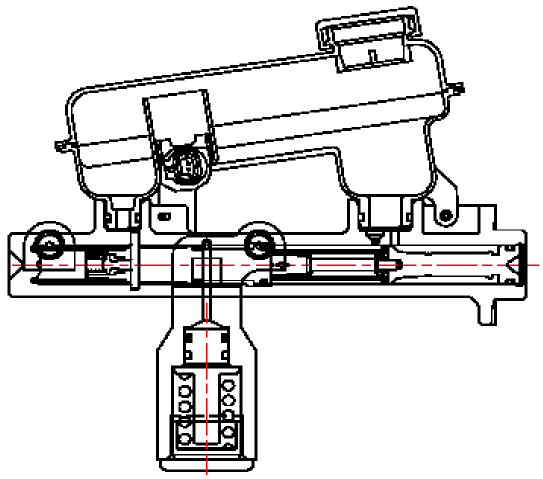


Figure 18: PFE-MC with Lockout

One solution is to incorporate a solenoid valve lockout device in-between the emulator and master cylinder. Whenever a failure is detected, the ECU would causethe valve to close to prevent the unwanted displacement. A second, more cost-effective approach to consider is to incorporate a seal bypass arrangement in the main bore as shown in Figure 18. In normal boosted operation, as previously noted, the primary and secondary outlets are both blocked by the isolation valves. Since the secondary piston is held rigidly in the bore, the fluid from the primary piston is permitted to flow around the by-pass lip seal as shown in Figure 19 and into the drilled path for the emulator.

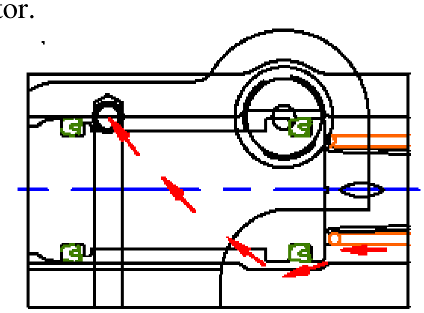


Figure 19: Pedal Feel Emulator Lockout Mechanism

As before, in failsafe braking, the isolation valves open and both primary and secondary piston circuits are now directly connected to the wheel brakes. As the secondary piston can now move forward, the lip seal re-enters the main bore and keeps the higher pressure primary circuit from losing any additional fluid to the emulator. It should be noted, however, with any types of lockout mechanism that they are ineffective should there be an "in-stop" failure. Once the PFE displacement is utilized, that amount of pedal travel is lost until the subsequent stop. Thus, worst case conditions must always be considered in the final design process assuming no lockout present.

CONCLUSION

Similar to the days of early ABS introduction, multiple EHB hydraulic design configurations have emerged. From the mid 80's through the latter part of the 1990's numerous ABS configurations ranging from hydraulically boosted open systems, to four valve flow control designs, to modulators based upon ball screws and electric motors came to market before the 8-valve, closed recirculation system became the de facto standard. As with any new technology, there are concerns and tradeoffs to be dealt with. In the case of the electro-hydraulic brake they center around increased electrical and mechanical complexity, failsafe braking performance, accumulator safety, and 2-wheel versus 4-wheel backup modes. Each of these concerns has been answered by prudent designs and incorporation of new component technologies. The configuration adopted in Delphi's EHB development has included use of four-wheel failsafe with individual isolation pistons and utilization of mechanical pedal feel lockout. This particular design allows system flexibility, inherent accumulator precharge isolation, and the ability to tune for optimum failed system stopping performance for all vehicle classes.

Ultimately, no matter which final configuration is selected for a specific vehicle platform, it will have to undergo the rigors of full brake system validation. A carefully de-signed and implemented EHB system holds the promise of enabling the new brake-by-wire features while still reliably performing the everyday task of stopping the vehicle.

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