CORVETTE NEWS SUPPLEMENT TO VOLUME 8, NUMBER 3

HIGH CALIPER BRAKING

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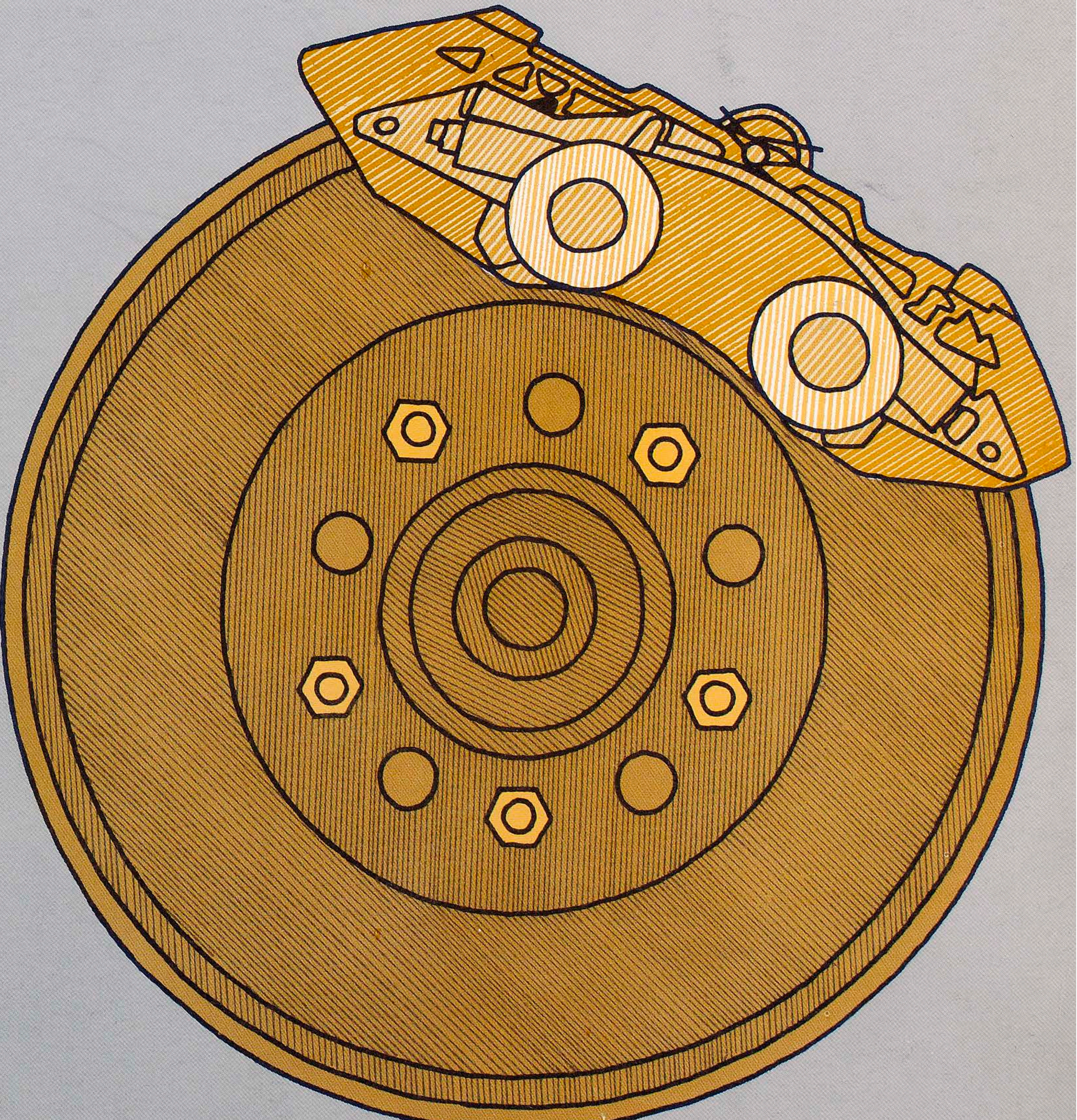

As Presented to the 1965 International Automotive Engineering Congress, Detroit, Michigan, January 11-15, 1965

INTRODUCTION

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The announcement that the 1965 Corvette will have disk brakes was greeted by the motoring press in unison with "Finally."

This implied that we have done something we should have done a long time ago and were blind to the facts obvious even to a casual student of the subject.

In actual fact, we were not blind, backward, or obstinate. We had a considerable background with disc brakes which indicated that just slapping units of existing accepted design on the Corvette will not produce desired results.

To put things in the proper perspective, it must be realized that for a number of years Corvette had available three drum brake choices:

1. Base system with organic lining.
2. Optional metallic lining with essentially the same brake drums.
3. Heavy duty brake with metallic linings, special drums and forced draft ventilation.

Even the base brake had the lowest ratio of vehicle weight to lining area of all American passenger cars and with standard drums and optional metallic linings the brake capacity, per se, was more than adequate for the most severe usage the vehicle could be subjected to on the road.

With these, Corvette did not lack stopping ability or suffer from fade or rapid wear.

However, Corvette, being the instrument of driving pleasure, could stand an improvement in brake modulation.

For a vehicle of this nature, it is not sufficient to be steered, accelerated, and braked. It is important how well the car responds to the driver in the performance of these tasks.

Duo-servo brakes do not lend themselves to good modulation. With metallic linings having a greater variation of coefficient of friction the modulation is further impaired.

For attainment of linear relationship between driver effort and vehicle retardation, system without self-energization offered the best prospects.

Two trailing shoe drum brakes were not seriously considered, but disc brakes were, at all times, under study and consideration.

However, one of the restraining factors in a commitment to a disc brake was our doubt that we would be able to match our then current heavy duty metallic brake performance.

This drum brake, although brutal, had the highest energy dissipating ability and durability of all brakes we could visualize on our Corvette.

It was somewhat paradoxical because a modern disc brake was designed by Dunlop, and introduced by Jaguar in the 1952 LeMans race.

It is via racing that disc brakes proceeded to conquer sports car and, to some extent, passenger car fields. Yet in the case of Corvette, it is in this type of application that the then current disc brakes held little promise.

We were aware of "sex appeal" of the disc brake, and the opinion of some sector which held unless you have disc brake "you don't have a brake."

However, with fortitude, we worked toward the brake which, at least, would hold the promise to equal or outdo our heavy duty drum brakes.

What sets Corvette apart from other sports cars, subjected to similar usage, is its unparalleled combination of weight and performance packaged in comparatively low drag body. The effect of such a combination was apparently not fully clear to most experienced brake firms, who embarked on the Corvette project without a shadow of a doubt as to the successful outcome.

In estimating brake requirements, the expression for kinetic energy to be dissipated,

$$\frac{MV^2}{2},$$

tends to attract attention to V^2 more than it does to M.

The energy equation, on the road level deceleration, will read as follows:

$$\frac{MV^2}{2} = \text{Braking} + \text{roadload}$$

or

$$\frac{MV^2}{2} = \text{Braking} + \text{engine drag} + \text{rolling resistance and chassis friction} + \text{aerodynamic drag}$$

or

$$\frac{MV^2}{2} = \text{Braking} + aV^n + bV^m + cV^2$$

V is the speed

M is the equivalent mass including inertia of rotating parts.

a, b, and c are constants and $n > 1$, $m > 1$.

From the above, we can see that the right hand side of the equation also contains a member which varies with the square of speed and two members which vary with exponents larger than 1.0.

The significance of this can be seen on the following example:

Figure 1 represents time-speed diagrams for two almost identical vehicles except for their weight.

If we assign to the light car a weight value of 100%, the standard car will weigh 135.5% or 35.5% more.

The average decelerations between 140 and 80 mph, taken from these graphs, are:

For the Light Car	In neutral	4.9 ft./sec.^{2}
	In high gear	7.46 ft./sec.^{2}
For the Std. Car:	In neutral	2.60 ft./sec.^{2}
	In high gear	5.16 ft./sec.^{2}

The make-up deceleration to be supplied by brakes for, let's say, total deceleration of 1.0 g is:

24.74 ft./sec.^{2} for light car

and

27.04 ft./sec.^{2} for standard car.

If we do apply these figures to the respective vehicles, we will see that 35.5% heavier vehicle will require 48.25% higher braking force.

The light car in this example, with comparatively modest power source at the time, had fairly satisfactory brakes.

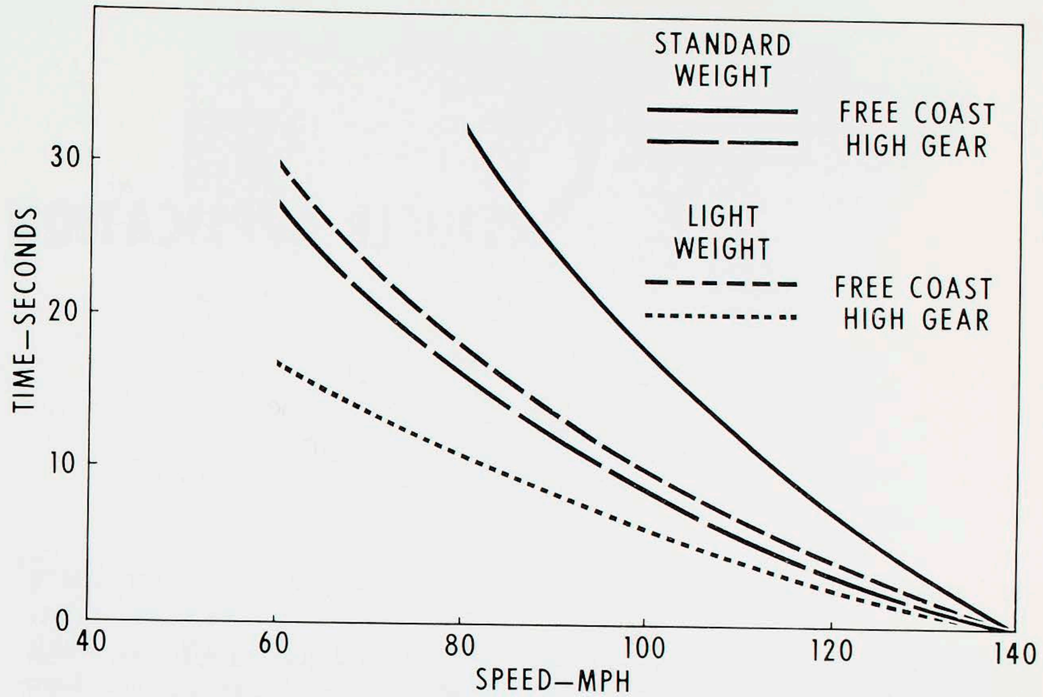
They were solid rotors conforming to best European practice but it was evident that for higher energy inputs, our standard Corvette would require a configuration with higher heat rejection ability than the solid rotors could provide.

When we received management directive to proceed with design and development toward 1965 model year release, we had enough background to establish basic premises.

1. The brake to fit in existing environment.
2. Vented discs with air passages big enough to be effective.
3. Largest lining area compatible with two pistons per pad.
4. Heat insulation of working pistons to protect heat sensitive parts and prevent fluid from boiling.

FREE COAST AND HIGH GEAR DECELERATION CURVES

TWO SIMILAR VEHICLES EXCEPT FOR WEIGHT



In other words, we were designing a potentially maximum brake within the given surroundings.

We did not believe that we would be able to design this brake to operate without power assist. However, our management felt that we would not accept the cost penalty of power assist on the base brake.

This decision contributed to orientation of design in a way, which in retrospect, has proven most beneficial.

Apart from stiffness of mechanical and hydraulic components thus required, the brake pad became zero clearance rather than retracted when not in use.

The benefits we reaped were:

1. Instantaneous response due to zero travel.
2. Freedom from "knock back" of pads on heavy competition type cornering.
3. Freedom from "knock back" due to the axle wheel bearing clearances.
4. Low sensitivity to weather and icing.

The price we are paying for this feature was negligible power consumption.

The selection of identical discs, front and rear, although a departure from optimized brake was indicated by economics.

The brake distribution, of which Arnold Brown will go into in detail, is such that under heaviest deceleration, 1.0 g or slightly over, lock up will occur on the front wheels first.

If conditions are such that wheel lock up can occur at deceleration of less than one g, the front wheel will lock up first, and the ability to regain direction control is enhanced.

It is of interest to note that the maximum decelerations, that is to the limit of frictional ability of the tire, is not easily attainable.

It follows that deceleration rates due to the combined effect of frictional and aerodynamic forces can exceed 1.2 g especially in higher speed brackets.

We are particularly appreciative of the help and suggestions we got along the way. Successful developments of this type are usually the result of much effort by many people. We learned from developments already on the market and had the assistance of Moraine Products, Engineering Staff and GM Research as well as the Chevrolet Engineering Groups.

VEHICLE APPLICATION

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The design program for the Corvette disc brake system was guided by certain parameters and objectives established by previous development investigations. In general, these guideposts were:

Vehicle -

- Braking tailored for sports car performance without power assist.
- Four wheel disc system.
- Minimum change to existing vehicle design.

Component -

- Fixed vented discs.
- Heat sink capacity of disc limited only by space available.
- Precision tolerances required for flatness, parallelism, balance, and surface finish.
- Splash shields mandatory.
- Fixed calipers with floating pistons.
- Separate parking brake system.
- Large reservoir master cylinder required with tailored piston diameter and travel.

To obtain true sports car brake performance, a design for one g deceleration was considered mandatory. A sports car lends itself to this objective since its load carrying capacity is physically limited. No compromise is necessary because of extremes in load capabilitites and the attendant distribution changes. Early development work showed convincingly that front discs only produced a better brake system, but discs both front and rear produced the best system. This is especially true if a one g deceleration capability is to be achieved.

A problem basic to all brake systems is heat. Solid discs, shown in figure 1, were initially designed into the experimental Grand Sport Corvette program, but brake performance was not satisfactory. Changing the front discs from 1/2 inch solid to the 1 inch vented units in figure 2 produced a substantial reduction in lining temperature for a weight increase of only 0.25 pounds. Temperatures recorded during stops from 100 MPH at 20 and 25 ft/sec^2 decelerations are plotted in figure 3. Brake performance became highly satisfactory with vented discs.

Since the heat reservoir capability of any drum or disc is a function of its effective mass, the physical size of disc was to be limited only by the space package available.

Within practical limits, disc flatness, parallelism, and balance have little effect on brake performance. These factors, however, influence the feel of the system; hence, extremely tight limits were imposed.

Although splash shields are a deterrent to heat rejection, their absence decreases lining life and, of course, increases water sensitivity.

The Delco-Moraine caliper design was selected. As will be shown later, the constant contact feature of this design plays an important part in making the brake system effective without power assist.

Since both the disc and the caliper were to be fixed rather than floating, a separate parking brake system became necessary. A mechanically actuated drum type duo-servo parking brake had been successfully designed in the Grand Sport Corvette program. The same basic design was adapted into the production 1965 Corvette.

Totally new master cylinders were necessary for this program. With the caliper design used, lining wear is automatically made up by brake fluid requiring an increased fluid reserve. Since a larger integral reservoir required a new housing casting, a completely new master cylinder was designed for the system.

HYDRAULICS

Within the limits established by the vehicle and component parameters, physical design of the system began with the selection of caliper piston size. Major factors in this determination were the need for 1 g deceleration and the desire to have a resultant pedal effort that would not require power assist. The front-to-rear proportioning was determined by front-to-rear weight distribution at one g deceleration:

Vehicle design weight (W) 3477 lbs.

Front 1665 lbs.
Rear 1812 lbs.

$$\text{Weight Transfer} = \frac{uWh}{L} = 628 \text{ lbs.}$$

where: h is the center of gravity (17.7 inches)
L is the wheelbase (98 inches)
u is coefficient of traction (1 at one g)

Percent weight distribution -

Front 66% (2293 lbs.)
Rear 34% (1184 lbs.)

Therefore, the front-to-rear ratio of the effective hydraulic areas had to be approximately 2-to-1.

The 1964 brake pedal geometry is used to retain the sports car feature of a relatively low pedal height. This established a 1.1 inch master cylinder stroke for 4.98 inches of pedal travel. The resulting 4.54-to-1 mechanical ratio imposed a high hydraulic ratio requirement for successful manual operation.

Development indicated that an overall ratio of approximately 200-to-1 would be required to exclude the need for power assist. With the established 4.54-to-1 mechanical ratio, a hydraulic ratio of 44-to-1 would be necessary. As will be shown later, a one inch diameter master cylinder was necessary. Using this, required total

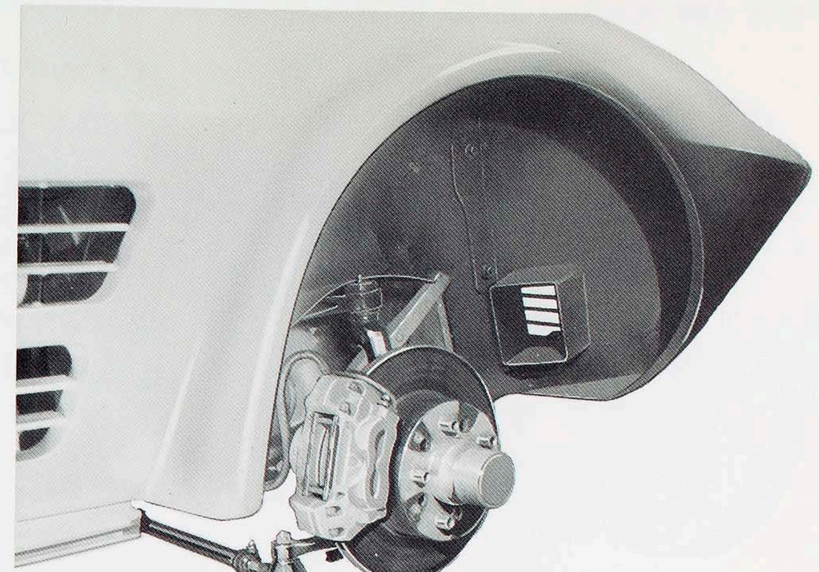

Fig. 1

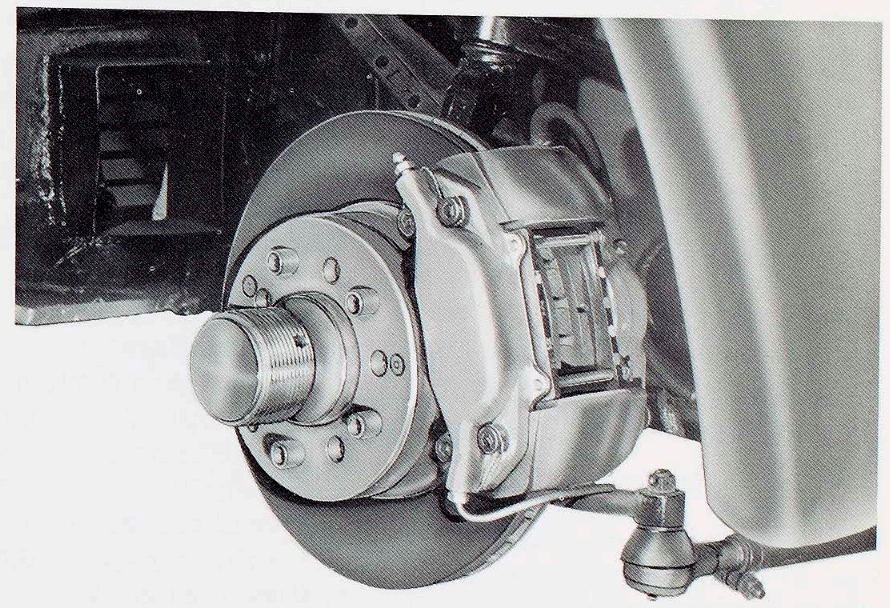

Fig. 2

LINING TEMPERATURE COMPARISON
SOLID VERSUS VENTED DISCS

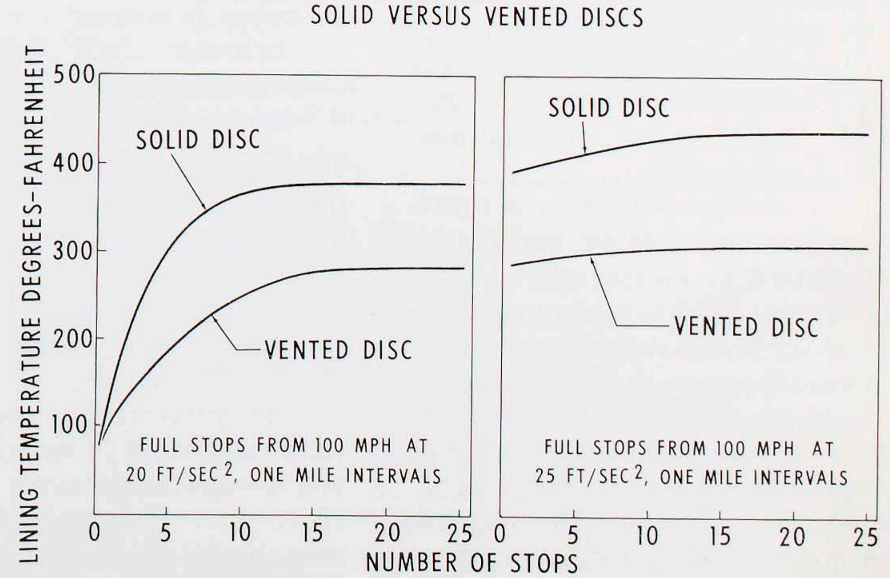

Fig. 3

REAR DISC AND PARKING BRAKE DETAILS

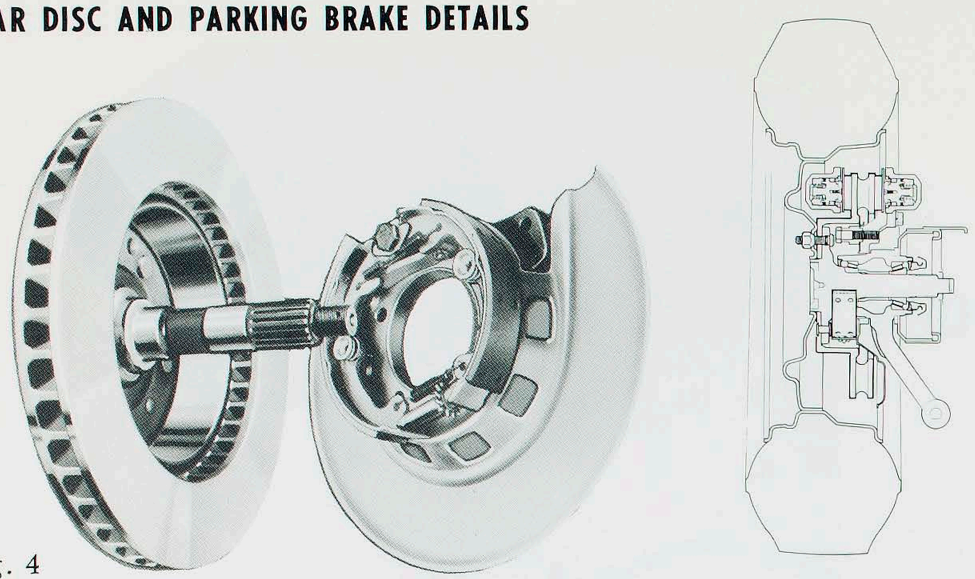

Fig. 4

piston area becomes 34.6 square inches.

The area of 34.6 square inches had to be proportioned 2-to-1 front-to-rear with physical size limited by available space. The final piston sizes selected were:

	Front	Rear
Diameter	1.875 in.	1.375 in.
Area	2.76 sq.in.	1.48 sq.in.
Actual Distribution	65%	35%

Actual total piston area became 33.9 square inches producing a 43.2-to-1 hydraulic ratio and a 196-to-1 overall ratio. This high overall ratio requires one inch of travel for every 0.005 inches of lining movement. To avoid excessive pedal travel and remain within the original parameters, the caliper was designed to give constant lining contact with the disc rather than retract.

For design purposes, a maximum caliper piston travel was estimated as 0.010 inches. This produces a fluid volume requirement of 0.339 cubic inches. A 100% allowance, a suggested rule of thumb figure that later tests proved relatively accurate, was added to provide for system expansion and deflection. The required master cylinder displacement became 0.678 cubic inches. To exceed this requirement, with the 1.1 inch stroke previously established, a one inch diameter master cylinder was selected.

In a later laboratory check of displacement on a prototype installation, total fluid displaced was 0.619 cubic inches at 1200 psi line pressure. Component expansion and displacement was found to be:

Expansion in steel brake lines	nil
Expansion in flexible brake lines	0.0910 cu.in.
Spread of front caliper	0.1105 cu.in.
Spread of rear caliper	0.0297 cu.in.
Compression of front lining	0.1328 cu.in.
Compression of rear lining	0.0595 cu.in.
Losses in master cylinder	0.0525 cu.in.
Total accounted for	0.476 cu.in.

This total of 0.476 cu.in. is 77% of the displaced volume of 0.619 cubic inch and the remaining 23% was used in fluid compression, caliper seal deflection, and piston insulator compression which were not directly measurable.

It is interesting to note that the fluid displaced in caliper spread and lining compression is 0.3325 cubic inch. This compares favorably with the 0.339 cubic inch displacement required for the estimated maximum piston travel of 0.010.

In similar manner, the total displacement of 0.619 cubic inch at 1200 psi compares favorably with the 0.678 cubic inch displacement projected. The rule of thumb allowance of 100% was generous, but in the safety factor direction.

DISCS

The cast iron disc for the 1965 Corvette is 11.75 inches in diameter and 1.25 inches thick at the braking surfaces. The braking surfaces swept by the linings have an outer radius of 5.82 inches and an inner radius of 3.94 inches. This produces a swept area of approximately 460 square inches. The brake surfaces are separated by internal radial fins that move air over many exposed surfaces to dissipate heat. The fins themselves act as heat sinks and conduct heat away from the braking surfaces (figure 4). Except for an added finishing operation for the parking brake, front and rear discs are identical. All finishing operations are held to tight tolerances to provide optimum feel and performance. The two braking surfaces are flat and parallel within 0.001 inch on any radial line and within 0.0005 inch on the circumference of any radius. These surfaces must be square and run true with their respective bearing diameters within 0.002 inch total indicator reading.

To achieve these tolerances, all finish operations are performed using the vehicle's actual bearing races as centers. Semi-finished discs are riveted to the front hub or the rear stub axle, and, in the finish operation, both brake surfaces simultaneously honed to a 30-to-50 microinch nondirectional finish. Balance of the assemblies is held to 3 inch ounces.

With the mechanical and hydraulic specifications of the system established, other factors were determined. Of primary importance was the brake pedal force required for a one g stop. For simplification, it was assumed that rotary inertia would be offset by rolling resistance, wind resistance, and engine braking. Therefore, all these factors were neglected at this stage. Again, this omission would shade results toward the plus side of the ledger, and vehicle brake performance could only be better in the final product. On this basis, pedal effort was approximated as shown in figure 5.

A pedal force of 169 pounds for a one g stop was considered to be excessive, and the only factor used to determine this figure that was not irrevocable was the assumed lining coefficient of 0.3. Assuming a value of 0.4 for the lining material reduced the maximum line pressure to 665 psi and the maximum pedal force to 127 lbs. Therefore, another specification was established in that the coefficient of friction of the lining would have to approach 0.4.

PARKING BRAKE

Two major obstacles were encountered in adapting the Grand Sport parking brake to the production design. The first of these was effectiveness and the second was economics. Since the front and rear discs were to be basically the same, the diameter of the rear drum brake could be no greater than the inner diameter of the disc hat section. This became 6.5 inches. The width of the drum was also limited by space available, and 1.25 inches became the maximum width possible. Total lining area became 33.8 square inches.

In conventional drum brake designs, the actual parking brake is basically a low cost item since service brake drums and shoes are utilized. A separate brake system can become costly, and every effort was made to keep costs at a minimum and still provide an adequate brake.

In the Corvette design, the brake drum was obtained by adding a finish turn operation to the inside diameter of the disc hat section.

PROJECTED PEDAL FORCE CALCULATION

PEDAL FORCE (F)

$$F = \frac{A_m \times P}{r \times E}$$

where: A_m is master cylinder area (.785 sq.in.)
P is required line pressure
r is pedal ratio (4.54-to-1)
E is pedal efficiency (.9)

$$F_p = 0.192P$$

LINE PRESSURE (P)

$$P = \frac{T}{A_p R_e u}$$

where: T is brake torque
A_p is total caliper piston area (33.9 sq.in.)
R_e is effective radius
u is lining coefficient of friction (0.3 assumed)

$$P = \frac{0.0983T}{R_e}$$

BRAKE TORQUE (T)

$$T = W \times a/g \times R_r$$

where: W is vehicle design weight (3477 lb.)
a is acceleration rate (32.2 ft/sec^2)
R_r is tire rolling radius (12.8 in.)

$$T = 44{,}500 \text{ lb.in.}$$

EFFECTIVE RADIUS (R_e, as defined by M. Olley)

$$R_e = \frac{R_1}{6} \times \frac{(1-M)^2}{1\ M} + R_m$$

where: R_1 is lining outer radius (5.82 in.)
R_2 is lining inner radius (3.94 in.)
R_m is lining mean radius (4.88 in.)
m is ratio $\frac{R_2}{R_1}$ (.677)

$$R_e = 4.94 \text{ in.}$$

therefore:

$$P = 885 \text{ psi}$$

and

pedal force (F) = 169 lb. required for a one g deceleration with an assumed lining coefficient of 0.3.

The same calculations, assuming a lining coefficient of 0.4, gives a pedal force of 127 pounds.

Fig. 5

A backing plate was necessary. Since a splash shield was also required, the two were combined into a single component. The anchor pin reaction point was designed into the rear caliper attaching bracket. The only added expense parts were the brake shoes and their related hardware.

The parking brake is not self-adjusting. There should be no wear of the friction surfaces in parking brake service. To facilitate assembly and provide for production variations, a conventional star wheel adjuster is provided. Access is through the disc and axle flange after the wheel has been removed. Other than lengthening the rear cable, the parking brake actuating mechanism is unchanged from 1964.

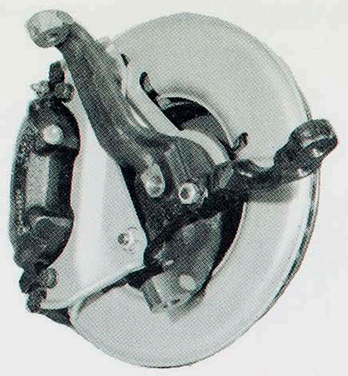

Fig. 6

ADAPTATION

The radial location for the front caliper was limited by the rear mounted steering linkage and jounce and wheel turning clearances. The sum of these factors dictated the selected production location, forward of the wheel centerline and above the horizontal.

Wheel bearing loads induced by braking were calculated and found to be less for this location than those of the drum brake. No bearing revisions were required.

This factor, coupled with the hat section design of the disc, meant that the front bearing hub would require only minor machining changes. The steering knuckle and spindle support forging required machining changes in the anchor boss area. Here, material was removed for disc clearance, and the boss itself became the caliper adapter bracket upper attaching hole. The second attaching hole for the adapter is the forward attaching point for the steering arm (figure 6). Attaching hardware was revised accordingly.

Considerably more problems were encountered in the adaptation at the rear. Again, the caliper location was restricted by the existing design, and the selected location 18 degrees aft of the vertical centerline required the least change. Here, the rear caliper adapter replaced the drum brake backing plate and was located on the same piloting diameter. This allowed the carryover of the spindle support. The drum type parking brake was designed into the area between the disc hat section and the adapter. The adapter was designed to serve as the parking brake anchor pin reaction point. The dust shield was located to the adapter and serves as the parking brake backing plate. Additional material was added to the rear spindle forging in the flange and wheel pilot area.

A local depression was added to the frame to insure caliper clearance in jounce. This, coupled with the compression bumper relocation, were the only frame revisions required. The rear suspension torque arm is unchanged except for the parking brake cable guide welded to it.

Wheel revisions were made to provide additional caliper clearance. The 15x5-1/2K steel wheel became a 15x5-1/2JK wheel with wheel disc revisions to accommodate the drop section change. The optional aluminum wheel was qualified on the under side of the rim to again assure adequate caliper clearance.

Since the normal mounting faces of both the front hub and the rear spindle were unchanged, the vehicle wheel tread was increased by approximately 1/2 inch. This is due to the increased thickness of the disc hat section web over the drum web it replaced. Sufficient clearance had initially been designed into the wheel house openings of the body to accommodate this tread increase without body changes.

No major changes were required in the parking brake actuating system. Since the rear cable was externally attached to the parking brake actuating lever, the cable was revised to a one-piece construction and is installed at final assembly.

Other than compatibility changes of the brake pipes at the master cylinder and the master cylinders themselves, no other chassis changes were required.

BRAKE TESTS

Early in the disc brake development program it became obvious that standard brake tests and schedules were not sufficiently severe to insure a final design that would meet the goals established. For example, the first road trip for Corvettes with prototype installations was completely inconclusive.

The test course selected was mountainous roads in western West Virginia, specifically Peters and Potts Mountains. The schedule established through the years in this area has long been considered as severe a test of a brake system as is necessary for customer acceptance. With the Corvette's relatively light weight and good handling, however, the schedule quickly expanded itself into a sports car rally with speeds and decelerations well beyond the idiot limit for conventional cars. Other than a relatively obvious conclusion that splash shields would be necessary to control water sensitivity, no significant engineering data was obtained.

The results were accepted enthusiastically since the braking potential of the system appeared to be limitless in view of established baselines.

Proving Ground brake schedules were adequate for effectiveness checks, but the fade and recovery section of the S.A.E. Procedure produced no discernible fade characteristics between the various linings under consideration. For this reason, a special brake abuse test was incorporated into the schedule. This consists of twenty stops made at one mile intervals from 100 MPH at 20 ft/sec^2 deceleration. Such a deceleration rate is close to wheel slide on some conventional brake systems and therefore approaches a panic stop.

EFFECTIVENESS

Comparisons between Corvette drum and disc brake S.A.E. Procedure decelerations curves reveal several expected results:

Duo-servo action produces an apparent increase in effectiveness and decrease in pedal effort.

Change in deceleration rate is linear with disc brakes and exponential with drum brakes.

Effectiveness relative to time is better for drum brakes at low speeds, equal for both systems at intermediate speeds, and better for disc brakes at higher speeds.

No wheel lock-up is encountered with the disc brake system.

Disc brakes require less pedal travel than drums.

Response is more immediate with discs due to constant contact linings.

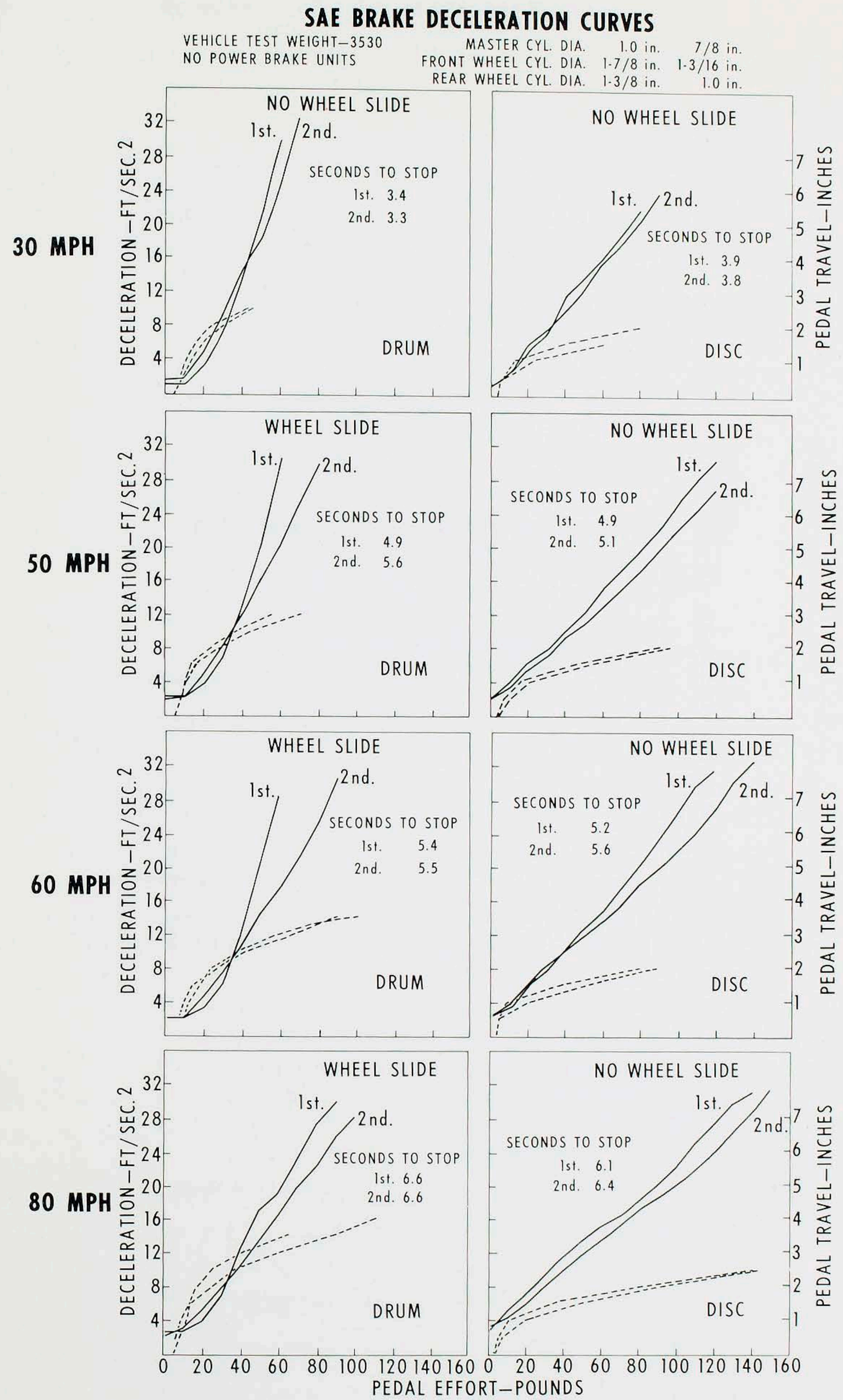

Fig. 7

Comparing the 30 MPH deceleration curves for each system (figure 7) shows that neither produces wheel slide and that the duo-servo system reaches a higher deceleration rate with a complete stop being obtained 1/2 second faster. Pedal effort for the drum system is lighter. No advantages can be claimed for the disc brake at this speed except for the linear increase in deceleration rate over the variable rate of the drum system. This is experienced and appreciated as good "pedal feel."

At 50 MPH, 60 MPH, and 80 MPH, the effectiveness of drum system appears at first glance to be better than the disc system. If effectiveness is to be measured strictly in terms of pedal effort, this would be true. Comparing the times required for each stop, however, shows a slight edge for disc brakes for the 80 MPH deceleration. It should be noted that no wheel slide was produced by the disc brakes, whereas four wheel slide was present on the drum brake vehicle at all three speeds. Translating this into terms of vehicle control under deceleration offers a paramount advantage for the disc system. Because a self-energized brake requires a major reduction in pedal force to allow wheel rotation after lock-up, the linear deceleration rate of the disc system is manifestly better than the non-linear drum system, especially under less than ideal road conditions.

If the comparison of the two systems were ended here, little fault could be found with the drum brake system. In fact, for the average driver of an average American made automobile, the disc brake system offers slight advantages. The drum brake system is more than adequate. The Corvette, however, is a sports car, and the average Corvette driver drives accordingly. Therefore, the comparison of the fade characteristics should reveal a major advantage of disc brakes.

FADE AND RECOVERY

Fade and recovery tests were run under the standard S.A.E. passenger car procedure. First fade and recovery curves for the two systems are shown in figure 8. From these, it can be seen that drum brakes experienced a moderate fade with excellent recovery whereas the disc brakes produced virtually no fade. On the second fade and recovery (figure 9), the drum brakes faded excessively and had slow recovery and the disc brakes were again fade free.

The special test of 20 stops from 100 MPH at 20 ft/sec^2 was added to the brake schedule. These results are shown in figure 10. Although the drum brake system completed the 20 stops, one front lining was found to be shredded on post-test inspection, and all linings showed considerable cracking. The disc brake system faded slightly between the sixth and eighth stops, but recovered within the test and became more effective. In a subsequent test, an accumulative total of 200 of these stops were made without lining failure.

Metallic linings were offered as optional equipment for the 1964 Corvette for those who felt the need for better fade resistance. It is of interest to compare the drum brake system with metallic linings (figure 11) to the organic disc brake system. It can be seen that the effectiveness, time to stop, pedal travel, pedal effort, and occurrence of wheel slide closely parallels the organic drum system. Fade characteristics, however, more closely follow the disc brake system (figure 12). The metallic lined drum brake has the effectiveness of the drum system and most of the fade resistance of the disc system. Its major drawback is erratic operation which has long been the nemesis of drum brakes used under severe conditions, and metallic linings generally add to problem. Normally, this is due to variations in effectiveness because of temperature

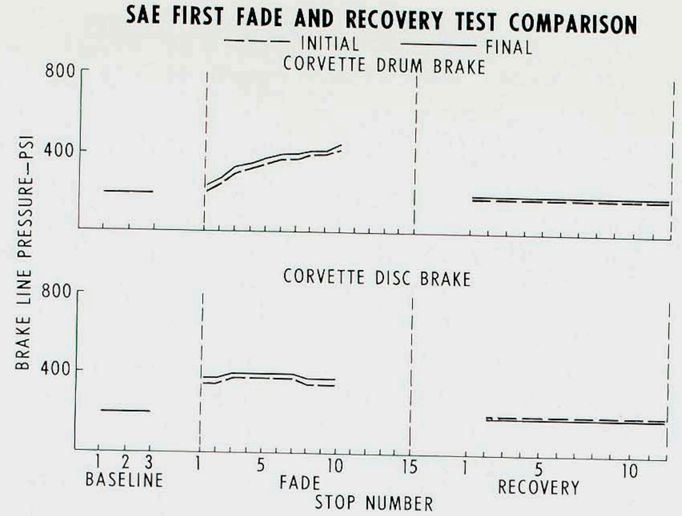

Fig. 8

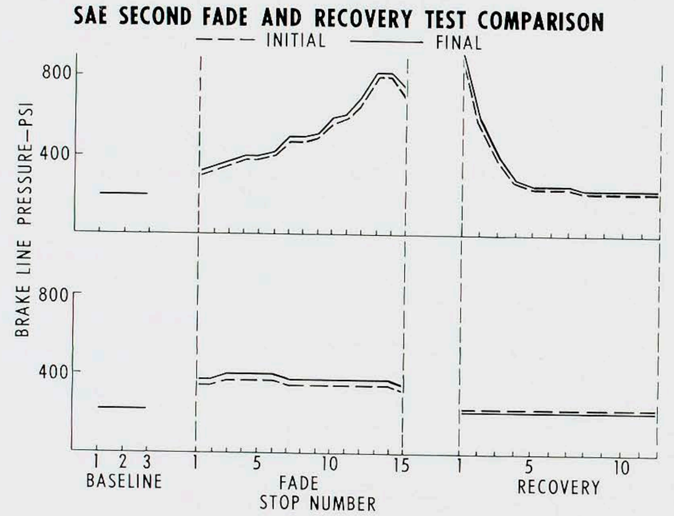

Fig. 9

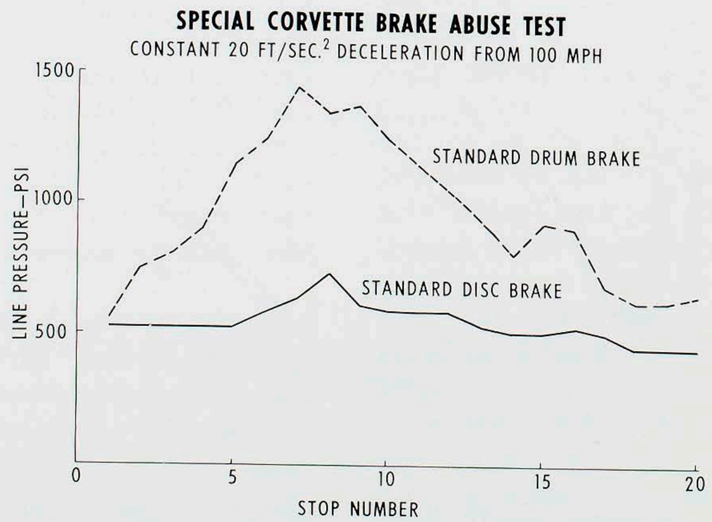

Fig. 10

change during a stop. Coupled with self-energization, a sensitive system can be produced. Since the disc brake is virtually free of erratic operation and fade, it has superseded the metallic optional lining in Corvette usage.

Results of an additional test graphically summarize the brake performance differences between the two systems. During vehicle durability tests, a 70 MPH constant pedal pressure fade out test is run periodically. From this speed, a 16 ft/sec^2 stop is made and required pedal force recorded.

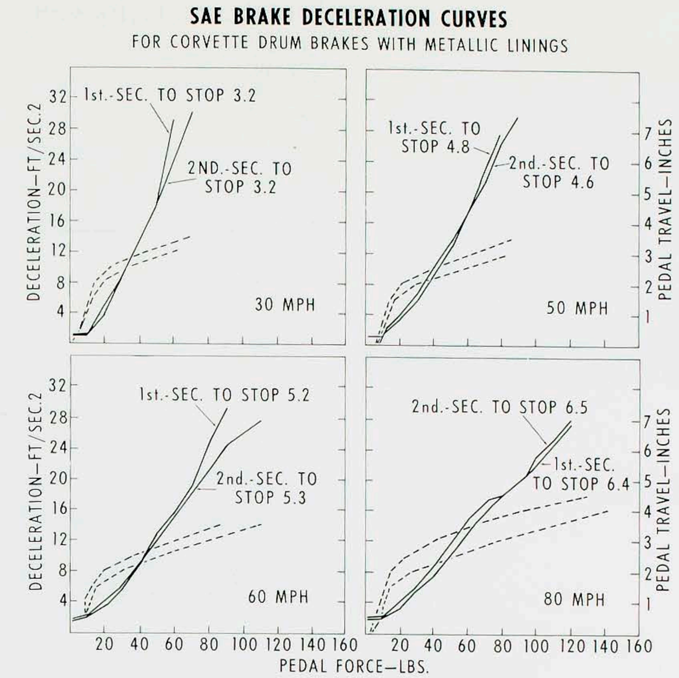

Fig. 11

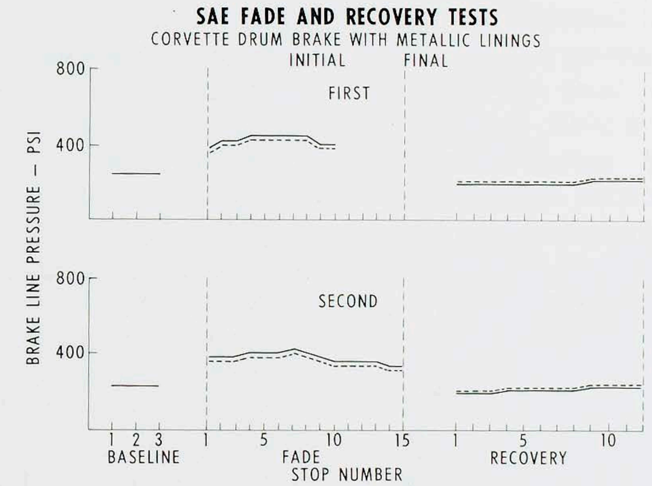

Fig. 12

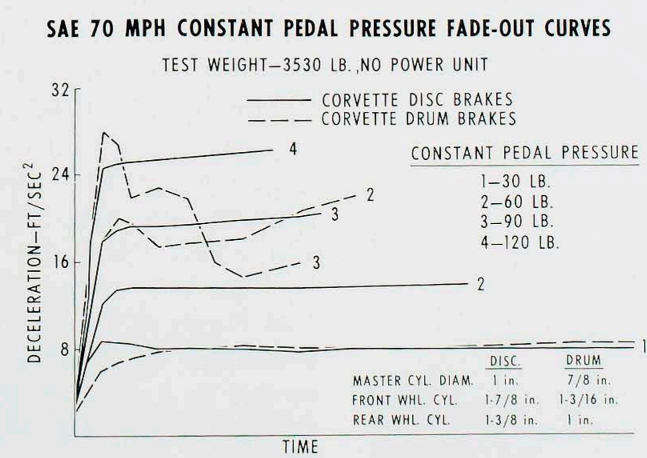

Fig. 13

This fade-out test was modified by holding the pedal force constant and recording the resulting deceleration rate. Pedal forces of 30, 60, 90, and 120 pounds were selected and the results are plotted in figure 13. At 30 pounds constant pedal the two systems respond identically. The 8 ft/sec^2 deceleration is an average braking rate encountered quite frequently in normal driving. With a 60 pound pedal, the drum brake system exhibits better effectiveness, some fade and some self-energization. At 90 pounds, the safest statement to make would be that the vehicle stopped. No attempts were made with a 120 pound pedal with the drum brake. Under the same conditions, the disc brake shows almost constant deceleration with rates proportional to the applied pedal force. For those who may feel that a 120 pound pedal is excessive for a stop approaching 30 ft/sec^2, optional power assist is available.

ADDITIONAL ROAD TESTS

All conventional Proving Ground durability tests and highway evaluations produced results favoring the disc brakes.

Lining wear rates were established on the Chevrolet 36,000 mile durability schedule. On this schedule the bogey for lining life had been established as 25,000 miles for front drum linings, full schedule for rear. Results for the disc brakes far exceeded this baseline. For those who believe that constant contact must cause lining wear, these figures must be disconcerting. Maximum disc wear on this test was negligible.

Dust tests were run at the Phoenix Proving Ground with excellent results. A total of over 1700 dust miles were run at 50 MPH with one stop per mile. Lining wear was very slight, the maximum being 0.005 inch at one point. Disc wear was 0.001 inch for the total test.

City traffic tests were run in Phoenix and Los Angeles again with excellent results. With an ambient of 85 degrees, the maximum lining temperature recorded was 300 degrees. These tests were included in a cross-country evaluation that encompassed every type of customer driving short of all out competition.

The final acceptance of the disc brakes came from a separate road test at Pike's Peak. Here, runs were made from the summit to the Gate House with the transmission in neutral whenever braking was required. Full brake effectiveness was available at the end of each run.

CALIPER DEVELOPMENT

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DELCO MORAINE DIVISION

HISTORICAL BACKGROUND

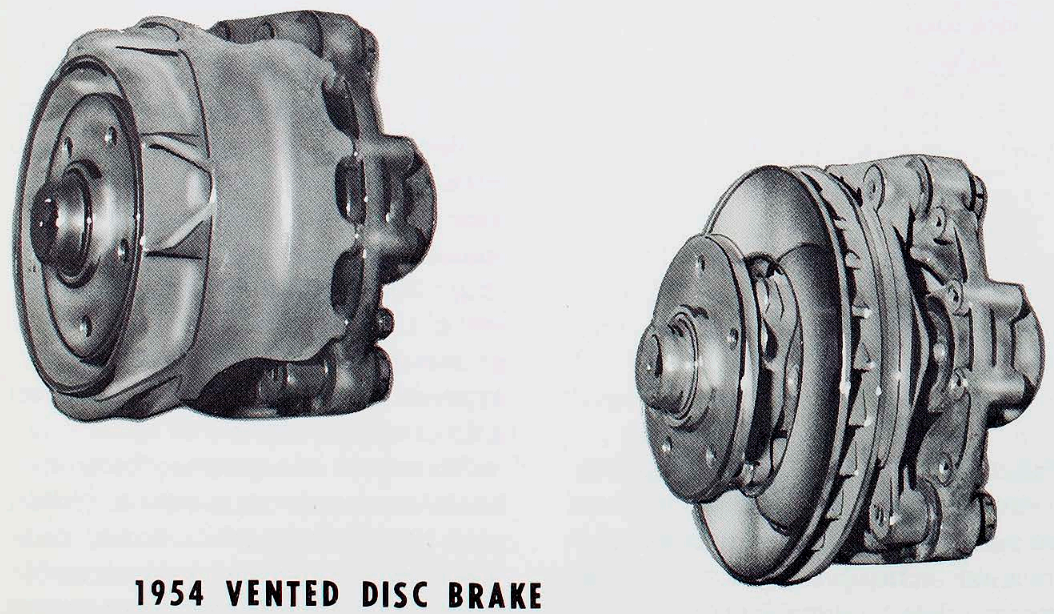
Much has occurred in the field of disc brakes since the first test of a disc brake was made at the Delco Brake Division twenty-seven years ago, in September, 1937. In fact, the name of the General Motors brake division itself has gone through some evolutionary changes before it became known as the Delco Moraine Division.

That brake was a dismal failure. Many exploratory designs were built and tested including an internal expanding clutch type similar to one later used on a passenger car in 1949. Cone and self-energizing types, floating disc, floating piston, and floating caliper types were designed and tested. All seemed inadequate for the job.

The European disc brakes in general were of the spot type, clamping a disc about 3/8 inch thick. These were completely inadequate for American-size cars. Lining life was poor because of high lining pressure and high temperature resulting in part from low disc mass. The pedal travel was too great, even when a booster was used, because of high caliper deflections, shoe clearance, and hydraulic deflections resulting from the high operating line pressure required.

Several designs with increased lining area were built at Delco Moraine, but it was not until a vented disc was applied to wheel brakes in 1954 that possibilities could be seen. Several four wheel installations were made on Buick, Oldsmobile, Cadillac, and Corvette cars and tested with varying degrees of success. One of these designs with a floating vented disc and metal lining, as illustrated in Figure 2, was adapted to the GM Firebird II in 1955 and was described in an SAE paper presented by J. B. Bidwell and R. E. Owens at the June meeting in 1956.

Several more car installations of this configuration were made and tested extensively at the GM Proving Ground, Pikes' Peak, and on the West Virginia mountain schedule, with very encouraging results. Early in 1963, a Corvette with four wheel discs of this type, as illustrated in Figure 3, was tested at Sebring.



1954 VENTED DISC BRAKE

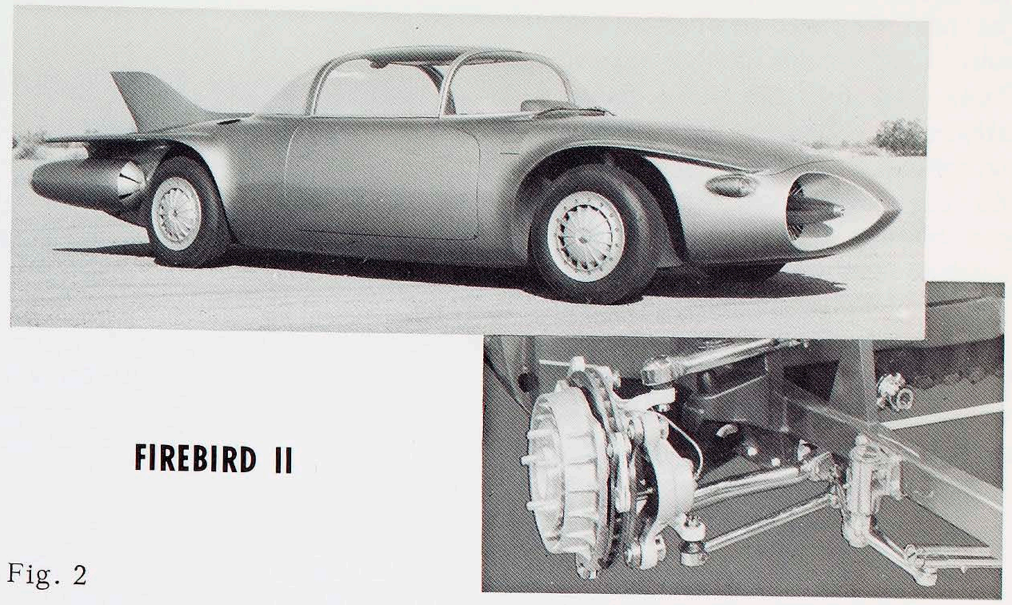
Fig. 1

At this time the brake performance of large European cars with disc brakes, weighing 3700 to 4100 lbs. and American cars with European disc brake adaptation generally were not considered acceptable by American standards.

Typical shortcomings of such systems are:

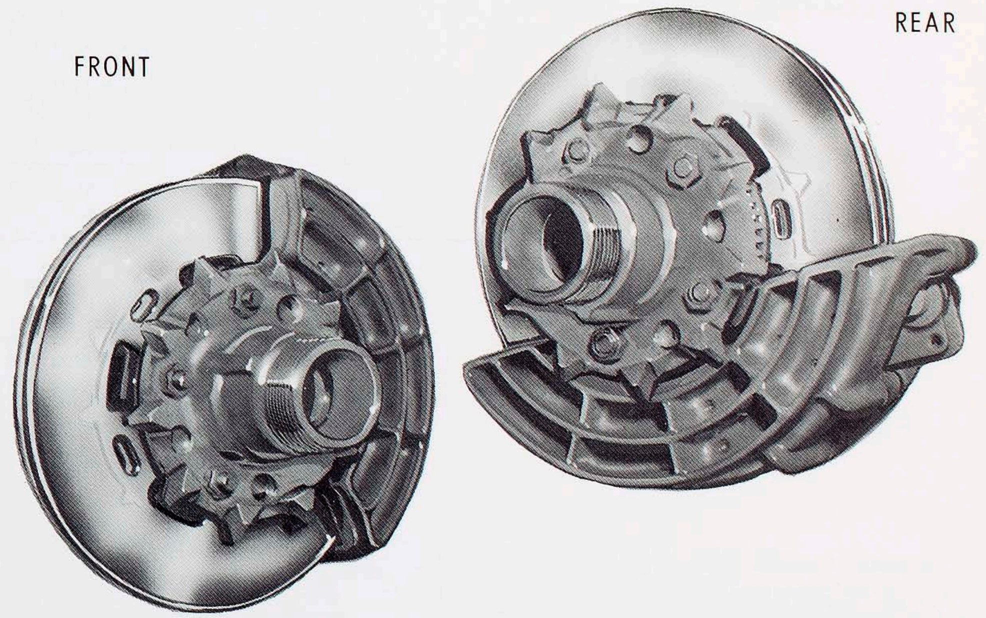
- Long pedal travel
- Spongy pedal
- Necessity of a booster
- Poor no power operation
- Low lining life
- Cornering pedal loss
- Poor fade
- Unsatisfactory parking brake with rear discs

However, the many attractive and desirable virtues of disc brakes kept development programs active. Inherent stability, linearity of output, and the freedom from wheel lockup as the limit of road adhesion is approached were outstanding characteristics. The excellent fade characteristics achieved on lighter cars and that attained with experimental designs on heavy cars added stimulus.



FIREBIRD II

Fig. 2



1963 TEST DISC BRAKE

Fig. 3

DESIGN PARAMETERS

To develop a disc brake design which would meet the operating requirements in this country, the following design objectives were set down.

1. Improved stability
2. Better fade resistance
3. Better durability
4. Apply system compatibility with present rear brakes for possible hybrid systems
5. No power pedal force equal to present booster system
6. Adaptability to present booster system
7. Pedal force equal to present brake system
8. Pedal travel equal to or less than present brake system

Many of these parameters dictate specific design features. Improved stability necessitates non-energization. Better fade resistance required a ventilated disc. Extended life indicated a ventilated disc for lowered lining temperature and adequate lining area for lowered lining pressure. Compatibility with present drum rear brakes, capability of operation with present boosters and acceptable no power operation dictated present drum brake operating line pressures. This in turn, defines the apply piston areas and results in the same pedal forces. Comparable pedal travel requires that deflections and lost travel be reduced to a minimum. Keeping deflections to a minimum requires a rigid and structurally efficient design, and again indicates low line pressures for minimizing hydraulic deflection. Reducing lost travel to a minimum necessitates minimum lining to disc clearances.

DESIGN ANALYSIS

The next step was to analyze the basic disc brake design types. These fall into the three basic categories - floating disc, floating caliper, and floating piston. The advantages and disadvantages of each of these types were analyzed and weighed as shown.

After giving consideration to these factors, it was decided to pursue the floating piston configuration, largely because of the experience background.

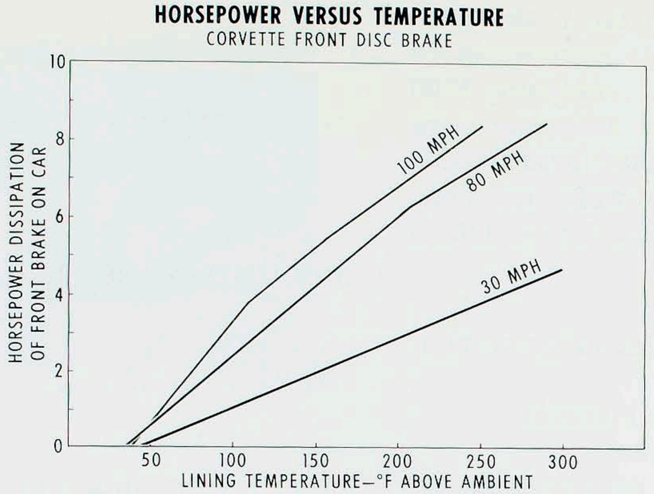


Fig. 4

FLOATING PISTON	Advantages: Fixed disc for firm torque reaction. Fixed caliper for firm torque reaction. Protected sliding contact via pistons. Freedom from noise and inertial forces of heavy sliding parts. Greater usage and experience.	Disadvantages: Cost of pistons required on both sides of discs. Cost of transfer plumbing. Cost of separate parking brake. Piston knock-back.
FLOATING CALIPER	Advantages: Fixed discs for firmly mounted torque reaction. Fewer pistons - required on one side of disc only. No transfer plumbing required. Use of service brake for parking.	Disadvantages: Anti-rattle device required for caliper. Unprotected sliding contacts. Cost and design problem of mounting caliper. Lack of successful experience.
FLOATING DISC	Advantages: Fixed caliper for firmly mounted torque reaction. Fewer pistons - required on one side of disc only. No transfer plumbing required. Use of service brake for parking.	Disadvantages: Fretting and wear of unprotected disc attachment. Noise of disc attachment from inertial and gyroscopic forces. Knock-back from inertial and gyroscopic forces of disc. Cost of adapting disc attachment in normal environment. Thermal stress cracking without hat section. Lack of successful experience.

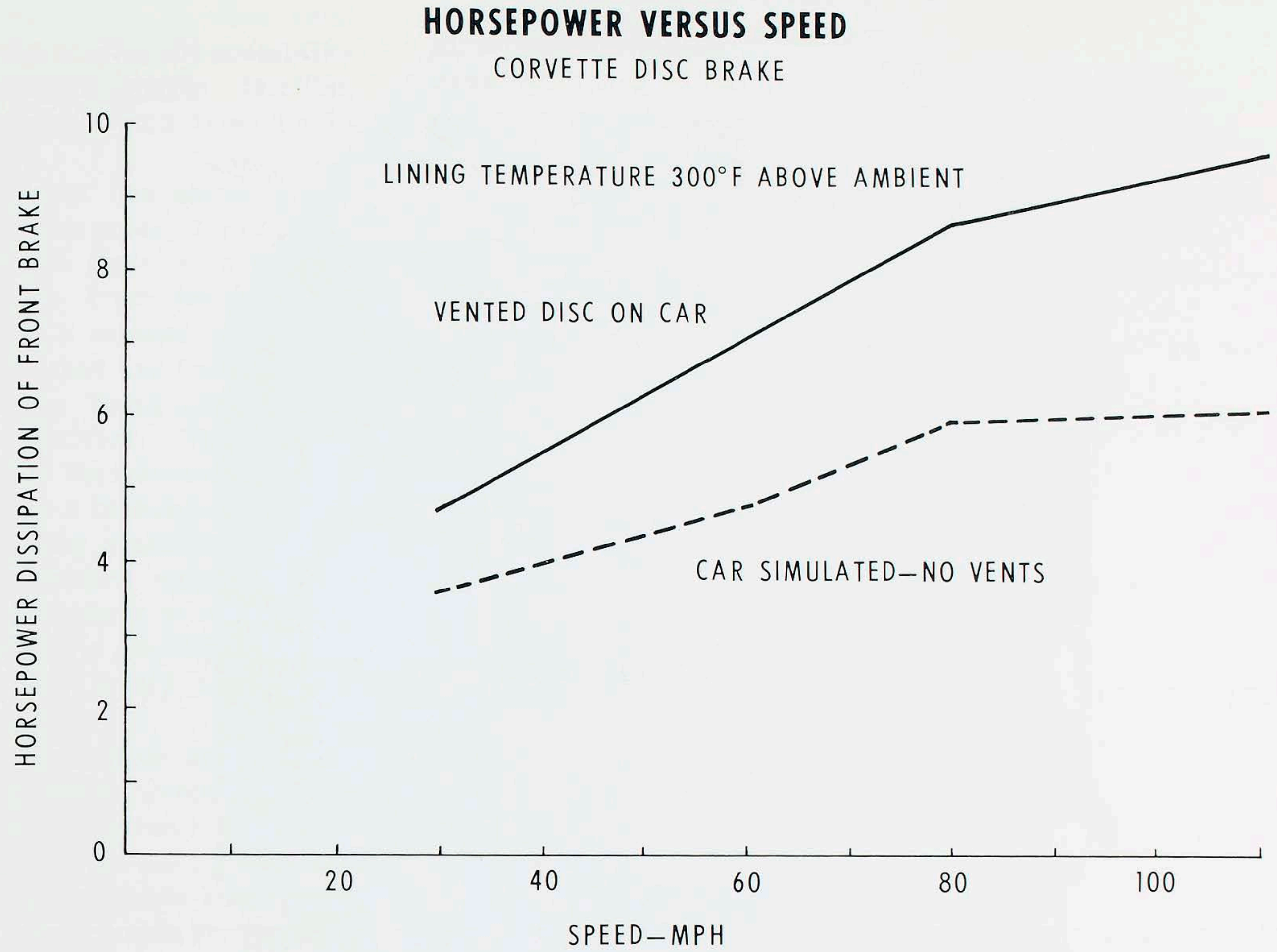


Fig. 5

DEVELOPMENT

A development program followed in which many design features were tested and evaluated.

VENTILATED DISC

The first convincing evidence of the merit of this feature was found on a dynamometer schedule devised four years ago when a disc brake with metal linings was under development for use on a Corvette for Sebring. It comprises slow downs from 120 MPH to 30 MPH at 18 ft/sec^2 deceleration and at a cycle frequency which would maintain an 800 degrees F shoe temperature.

A ventilated disc permitted a cycle frequency of 59 sec. A non-vented disc of the same weight stabilized at a cycle frequency of 105 sec., showing the vented design had a 78 per cent greater heat dissipating capacity. These represented brake horsepower ratings of 20.6 and 11.6 respectively.

More recently, a dynamometer horsepower test schedule has been used. In this schedule, the dynamometer is run at a steady speed between slow downs, which are made at regular intervals, until the lining temperature has stabilized. Several such cycle intervals are run to establish points on a horsepower-temperature chart, as shown in Figure 4. The curve thus generated is extrapolated to an arbitrarily established lining temperature. In this test, the thermocouple is placed 1/8 inch from the rubbing surface, 300 degrees F above ambient was selected as a temperature that would represent a condition which would not induce fade or cause damage to the lining or impair its inherent performance characteristic.

This procedure can be repeated at various cool-out speeds to show the effect of speed on horsepower capacity.

Figure 5 shows the relative horsepower capacity of a ventilated disc and one which has had the vents closed off.

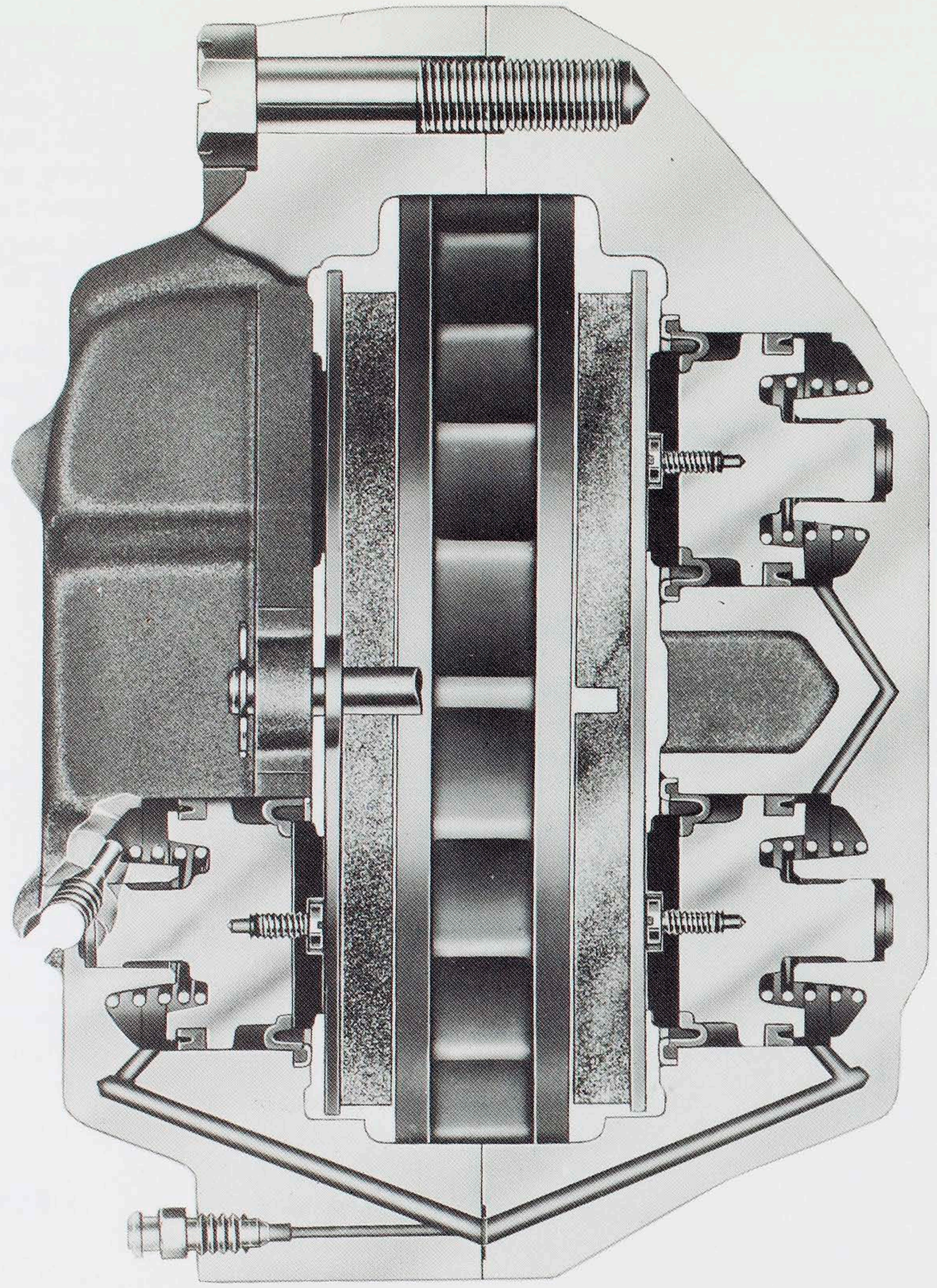


Fig. 6

BOOT DESIGN

Some of the earlier designs followed the then popular design having a single radial convolution. These frequently deteriorated rapidly from heat or were even "branded" by the shoe back. Heat shields were experimented with, but were found to be of questionable merit, used valuable space, and added complexity to the design.

A rolling diaphragm type was selected and, as shown in Figure 6, is almost completely shielded from radiant heat from the shoe.

SEAL DESIGN

In the floating piston type brake, pedal loss resulting from piston knock-back presents a problem. Knuckle and spindle deflections, primarily from cornering forces, are greatest at the vertical centerline and cause piston knock-back. This is minimized if the calipers can be mounted on the horizontal centerline. There are many chassis interferences at this location and if the experiences of other brake manufacturers have been like ours, we have never been in the Utopian position of having a customer design a car around our brakes. It has been necessary, usually, to position the caliper in a location somewhat less than ideal in this respect.

To cope with these deflections, it has been found to be necessary that the piston be free to follow the disc. This is accomplished with a spring urging the piston toward the disc. The spring force must be light enough to avoid a dragging situation. In turn, the seal friction must be less than the spring force. A lip type seal as shown in Figure 6 satisfies this requirement and provides complete freedom from the piston knock-back problem. As a side benefit, it is much less difficult to properly control the production of a seal groove in the piston rather than in the cylinder face.

PISTON GUIDING

In many of the earlier designs, self-aligning pistons with short lands were used to save axial space, and prevent the possibility of increased piston friction, which might result from cocking forces.

Experience has shown piston guiding is necessary for two purposes. The first is to prevent the pistons from being tipped over, like a damper in a stove pipe, when they are being pushed back during lining replacement or other service. This tipping would allow the fluid to be dumped and require a bleeding job. Secondly, guiding stabilizes the lining wear pattern which may become inconsistent at times, thus controlling the development of a taper which would reduce the effective life.

Space limitation again influenced the design evolution. The axial space required for a rolling diaphragm boot, the lip seal and adequate piston guide length added up to unacceptable proportions when placed end to end. The stem guide was the best solution for telescoping these features, as shown in Figure 6.

PISTON

Aluminum was selected as the piston material for two reasons. It is the most economical material from which a piston of this design can be produced and, when anodized, offers exceptional corrosion resistance.

INSULATORS

Piston insulators, also shown in Figure 6, are used to protect the brake fluid from heat transfer through the aluminum pistons. Not one instance of vapor lock has been encountered in test experience. The maximum fluid temperature recorded during an SAE fade schedule is 60 F above ambient. A series of 70 stops from 100 MPH at 1 mile intervals produces a shoe temperature of only 330 degrees F, 1/8 inch from the surface. Metal linings on the same test raise the shoe temperature to 620 degrees F.

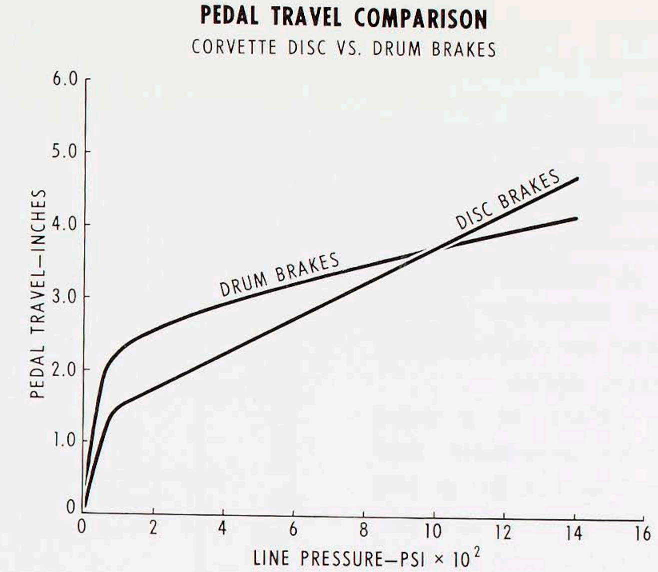


Fig. 7

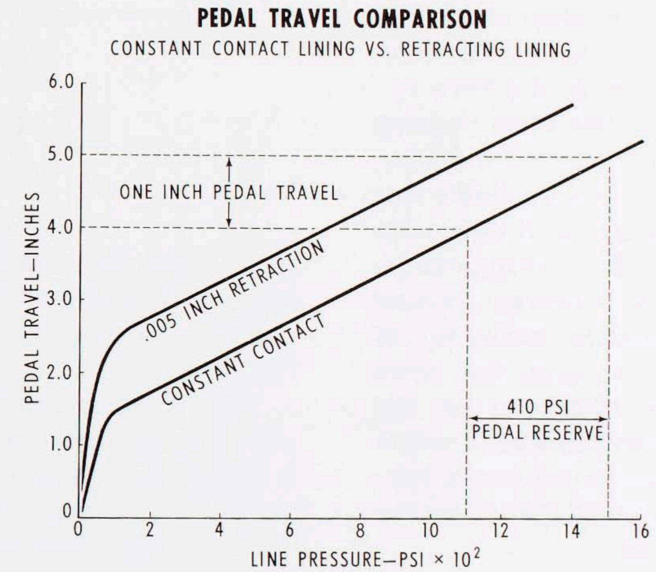


Fig. 8

CONSTANT CONTACT SHOE

A constant contact shoe feature made it possible to design a disc brake that will produce the same brake torque at the same line pressure as the duo-servo brake and at equal, or less, pedal travel up to wheel slide conditions. This is shown graphically in Figure 7. It is obvious then that booster and master cylinder requirements of the brake system remain unaffected.

Another benefit of the constant contact feature, one that has become important, is that of improved brake response by reducing pedal travel before line pressure is generated, as shown in Figure 8. In the operation of any control device, there is an intangibly pleasant reaction in having an immediate response to the movement of that control. In the operation of a brake there is also a reassuring feeling of security.

In Figure 8 it can be seen that .005 lining clearance on Corvette causes a 1 inch pedal travel loss. It is also apparent that with zero lining clearance this 1 inch pedal travel is available for pedal reserve. This is an important additional advantage which in this case amounts to 400 psi line pressure.

Better water immunity is another advantage of zero clearance. Figure 9 shows the comparative recovery of brakes with zero and .005 shoe clearance. This test is performed by making a series of recovery stops from 40 MPH at 10 ft/sec^2 and 1/2 mile intervals after the brakes had been soaked for 1 minute with a 1/2 GPM water spray.

The drag effect of a zero clearance brake assembly has been measured at 0.8 HP at 100 MPH. This has proved to be so insignificant that we have not been successful in measuring the effect on gas mileage, because it is within the test method accuracy. Experience has shown that some lining materials do not have acceptable lining life when running in constant contact with the disc. However, many are available that provide lining life well above the necessary level of performance.

On the GM Proving Ground durability schedule, lining life of the selected material has been 50,000 miles minimum on the Corvette. Forty thousand miles has been usual on the intermediate size cars and 35,000 miles for a 5800 lb. car.

LINING SELECTION

In selecting a lining material the following properties are considered:

1. Fade resistance
2. Wear resistance
3. Effectiveness
4. Noise supression
5. Modulus

A material which rates high in each of these properties would be ideal for any type of use. Unfortunately, this material is still in the development stage, and less than perfection must be accepted. It is necessary that these properties be blended with varying emphasis depending upon the specific application.

For a highly energized duo-servo brake, it is generally accepted that a degree of fade is desirable for stability. In a disc

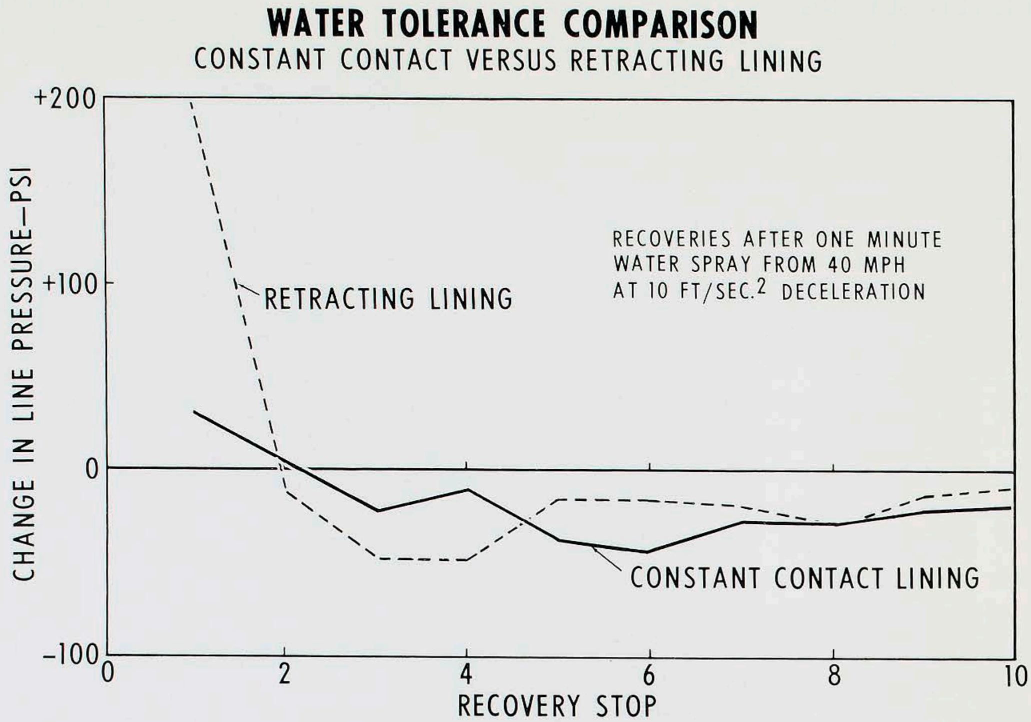


Fig. 9

brake with no energization, this need is greatly reduced, and the use of a more constant lining is feasible.

To evaluate lining materials, new procedures were developed. A dynamometer fade schedule has proved to be an accurate evaluation of fade, effectiveness, and noise, and of wear under severe service. This is similar to the SAE flat road schedule and comprises a burnish schedule, progressively increasing pressure stops from 30, 60 and 80 MPH, constant pressure stops from 30, 60, and 80 MPH and a stop from 100 MPH at a near wheel slide pressure. This is followed by 20 decelerations from 100 MPH to 30 MPH at 20 ft/sec^2 deceleration and at 75 sec. intervals. This is followed by a repeat of the initial stop schedule.

Delco Moraine uses a Road test in the Dayton area which has been found to closely correlate with the General Motor Proving Ground schedule.

As an additional laboratory test, a dynamometer schedule has been developed which is comparable to both of these car test schedules for lining life evaluation. A constant output Link Test machine schedule has proven to be a quick appraisal device for lining life.

In the course of our search for the best lining material for the Corvette, 173 dynamometer tests were run on 127 compounds from 12 suppliers.

CONCLUSION

Delco Moraine has been designing and building drum brakes for nearly thirty years. During this time there has been a continuing improvement of their performance and manufacturing methods through design refinement.

Disc brakes are the most recent product addition to the family of brake components. Knowledge and experience is being applied to the development and refinement of design to improve performance and value of disc brakes.

As a brake system supplier, this provides Delco Moraine customers with a broader opportunity to select the brake most suited to their requirements.