



FALCON V-8



INTERNATIONAL ENGINEERING CONFERENCE
APRIL 22-MAY 3, 1963 Dearborn, Michigan

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**BY
J. J. PRENDERGAST
FORD DIVISION PRODUCT ENGINEERING OFFICE

FORD MOTOR COMPANY**

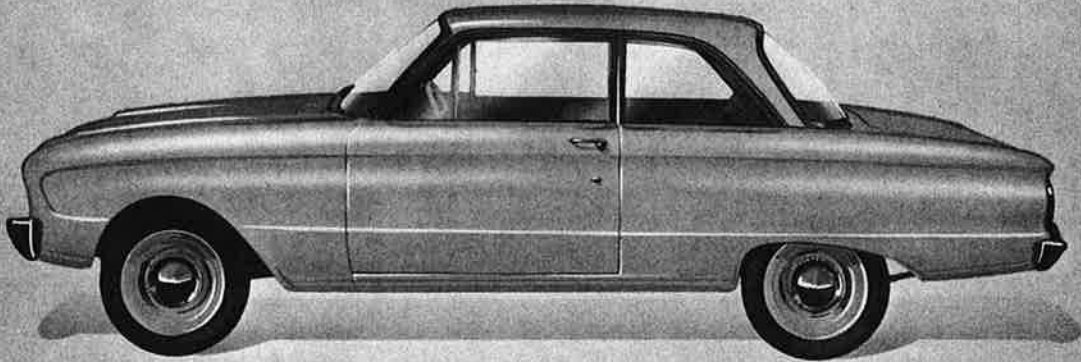
FALCON V-8

As you know, we have just recently completed the introduction of the Falcon V-8 and the Falcon V-8 Sprint models. The history of the Falcon, if you can call four model years history, is a most interesting subject to discuss. It is interesting because of the radical changes which have taken place in regard to the concept of the vehicle. This change in vehicle image or concept naturally generates a great deal of engineering activity and, as we shall see, results in many changes to the basic vehicle, some of which are not readily apparent to the casual observer.

The prime purpose of this paper is to outline for you the changes which were required for the basic Falcon vehicle to accommodate the V-8 engine. A by-product of this study is a hypothetical question; namely, would it have been economically possible back in 1959 and 1960 to make provisions in the basic design of the Falcon to minimize the changes required in 1963 for the V-8?

To refresh your memories, we would like, first, to review briefly the original Falcon objectives as you knew them in 1959, and then proceed thru the design of the basic components, outlining the design changes required for the V-8.

1960 FALCON PROGRAM OBJECTIVES



**MODEL-2 DOOR SEDAN
BODY-UNITIZED CONSTRUCTION
PASSENGERS - SIX**

**WEIGHT 2350
FUEL ECONOMY 23 MPG**

**ENGINE - 6 CYL. IN LINE - 144.3 CID 86.5 HP
TRANSMISSION - 3 SPEED SYNCHRO MESH
PERFORMANCE - TOP SPEED - 82 MPH
CRUISING SPEED - 70 MPH
ACCELERATION 0-60 - 18 SEC.**

Figure 1

FIGURE 1

As you will recall, the original Falcon program was established to provide a 2-door, six-passenger sedan. The powerplant was a 6-cylinder, 144.3 C.I.D. engine coupled to a 3-speed Synchromesh transmission. The total weight objective was 2350 pounds. This package was to provide a top speed of 82 M.P.H. with a cruising speed of 70 M.P.H. The average fuel economy objective was established at 23 M.P.G. As most of you know, the 1960 car met, within reasonable limits, all of its objectives, namely, piece cost, weight, fuel economy and performance. It met these tight objectives simply because it was designed to do so. Now, let's see what happens to this design when we change the objectives and put in a V-8 powerplant.

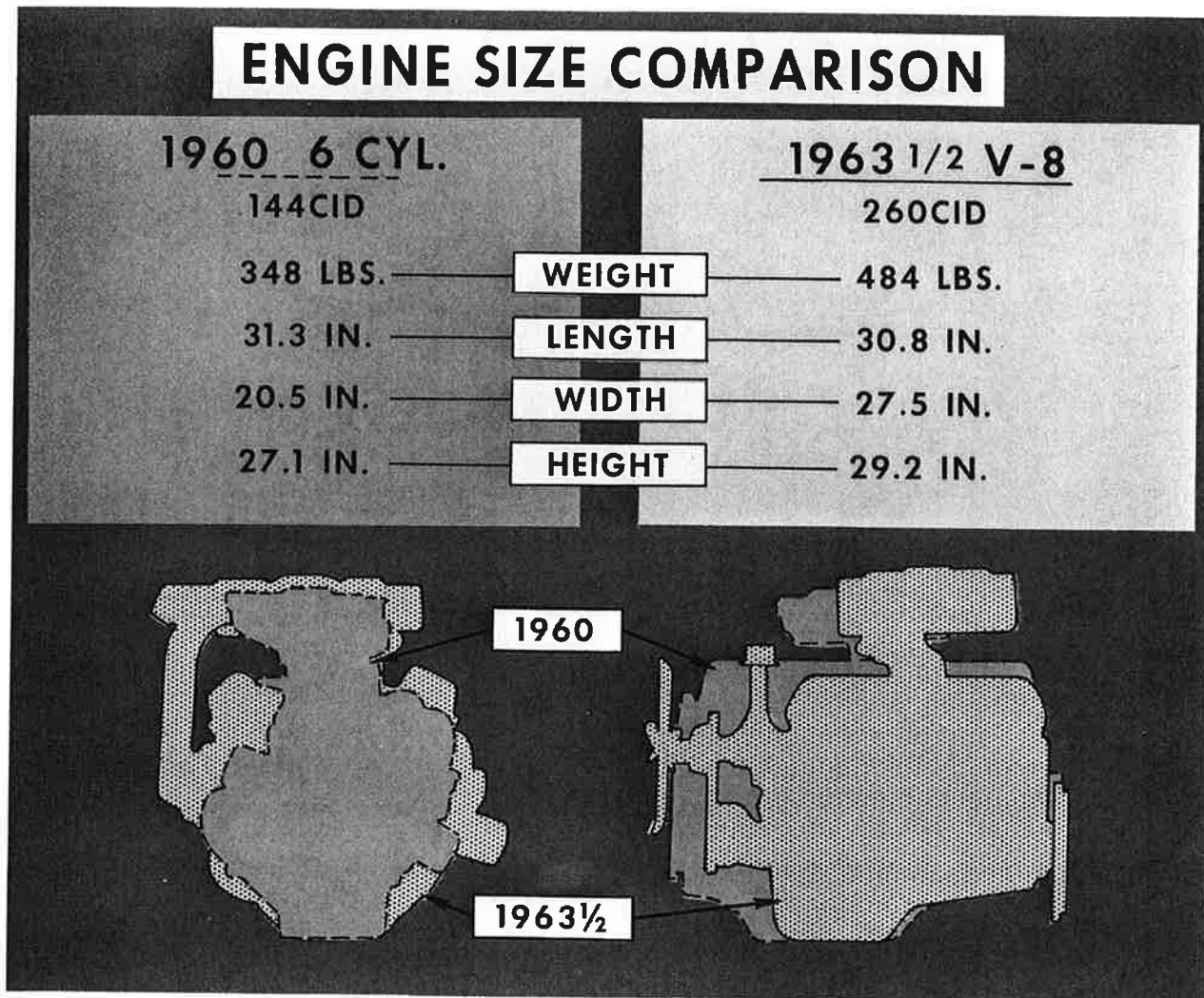


Figure 2

FIGURE 2

The basic engine planned for the Falcon was the 260 C.I.D. V-8 used in the Fairlane, Meteor and Ford. The first problem which we faced was one of simply not having enough engine compartment space. Although the 260 C.I.D. V-8 is .5 of an inch shorter than the 144 C.I.D. I-6, it is 7 inches wider and 2.1 inches higher. These increases in overall engine size necessitated some changes in the engine compartment sheet metal, which we will discuss later, and some changes to the basic 260 C.I.D. engine. The engine changes included a new left-hand exhaust manifold, one spark plug connector elbow, and a flatter hot/cold carburetor heat duct. The increased engine height also required redesigned steering linkage. When you compare the weight of 484 pounds for the V-8 to the 348 pounds for the I-6, it is obvious that structural and suspension changes are inevitable.

144 CID-260 CID ENGINE COMPARISON

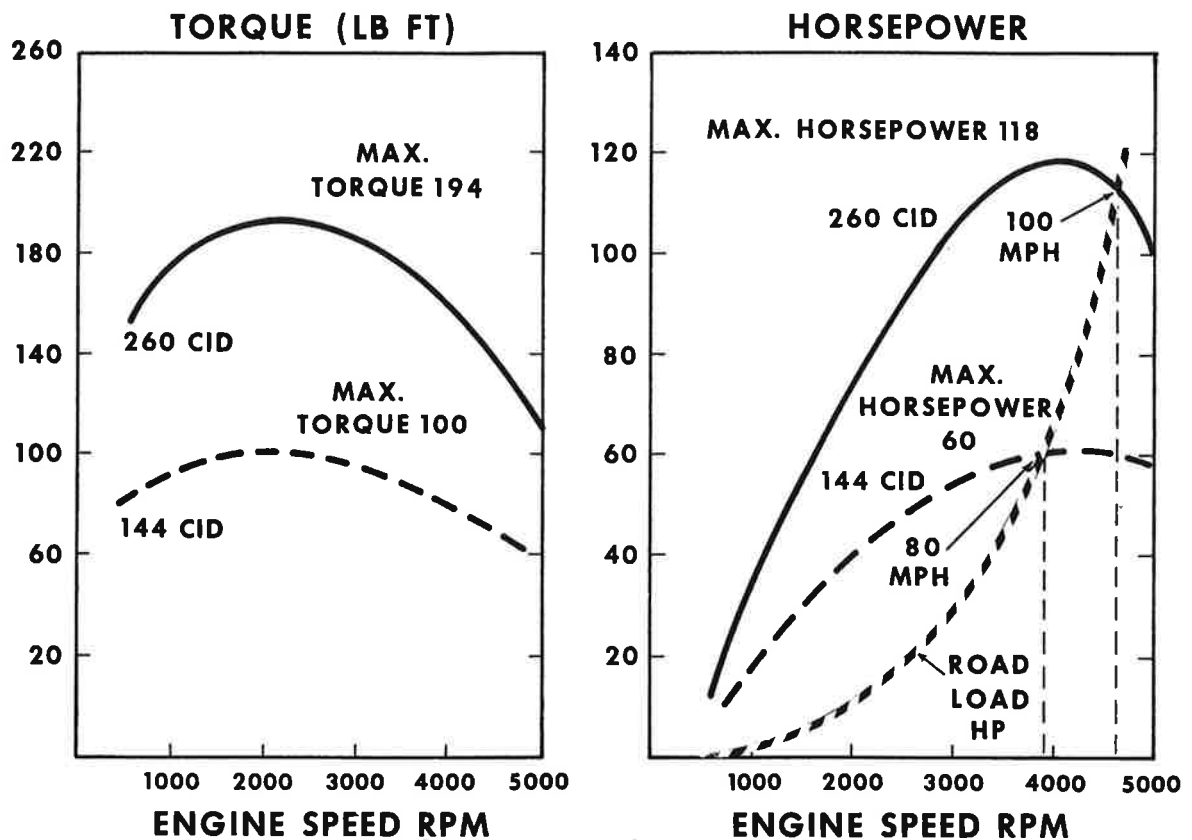


Figure 3

FIGURE 3

From a design standpoint, the increased power of the V-8, as compared to the I-6, is of greater significance than is the increase in size and weight. On an "as installed" basis, the V-8 engine provides a 94% increase in torque and a 97% increase in horsepower. Even when compared on a torque output to design weight ratio, you see that the V-8 presents a greater problem. The I-6 has a ratio of .29 lb.-ft. torque per lb. of design weight, while the V-8 has .40 lb.-ft. of torque per lb. of design weight. The result of this is reflected at the flywheel where the I-6 has available a maximum torque of 100 lb.-ft. at 2000 R.P.M. compared to the maximum torque of 194 lb.-ft. at 2000 R.P.M. for the V-8. The available horsepower is 60 for the I-6 and 118 for the V-8. The V-8 provides more than 150% increase in reserve horsepower at a cruising speed of 50 M.P.H. This additional power increases the possible maximum car speed from approximately 80 M.P.H. to 100 M.P.H. Naturally, this large increase in power and speed dictates the redesign or re-evaluation of all supporting components in the vehicle.

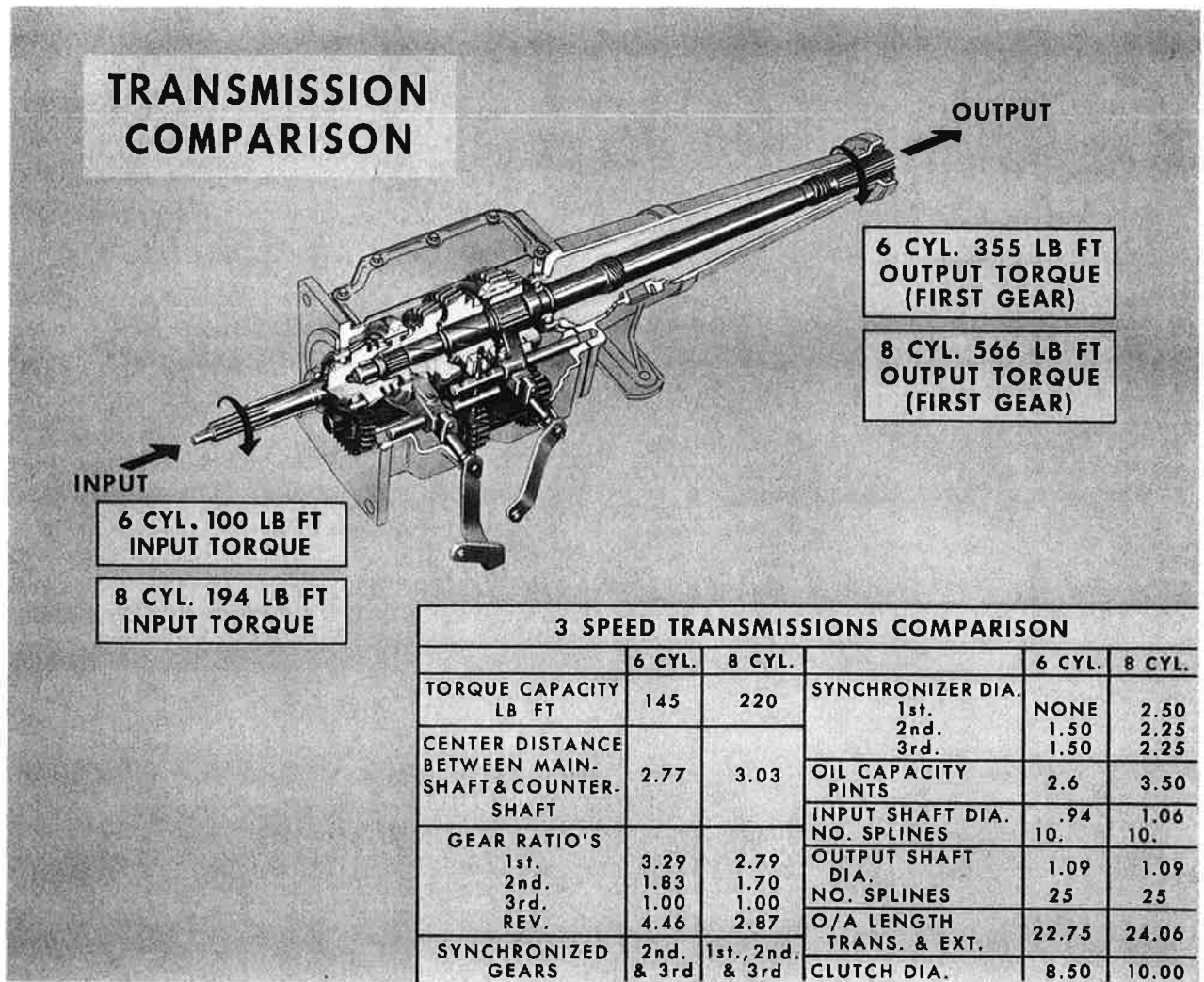


Figure 4

FIGURE 4

The 8.5-inch diameter Falcon clutch has a torque capacity of 150 lb.-ft., obviously too low to accommodate the 194 lb.-ft. produced by the V-8. The 10-inch Fairlane clutch, however, has a torque capacity of 250 lb.-ft. and, because it was already available, it was decided to adapt this clutch to the Falcon. The increased size of the clutch housing interfered with the underbody and required that the underbody be redesigned.

Naturally, the same analysis of available torque and horsepower can be applied to the transmission. The 3-speed manual transmission used in the 6-cylinder Falcon has a torque capacity of 145 lb.-ft. When you recall that the V-8 provides a torque of 194 lb.-ft., you can understand why it was necessary to provide a new transmission for the V-8. The new transmission is an adaptation of the Fairlane transmission and has a torque capacity of 220 lb.-ft. A review of the basic design dimensions of the two transmissions indicates the degree of design differences between them.

FIGURE 4 (Continued)

**COMPARISON BETWEEN 6- & 8-CYLINDER
3-SPEED TRANSMISSIONS**

		<u>6 - CYL.</u>	<u>8 - CYL.</u>
Torque capacity — lb. - ft.		145	220
Center distance between mainshaft and countershaft		2.77	3.03
Gear ratios	1st	3.29	2.79
	2nd	1.83	1.70
	3rd	1.00	1.00
	rev.	4.46	2.87
Synchronized gears		2nd & 3rd	1st, 2nd, & 3rd
Synchronizer dia.	1st	None	2.50
	2nd	1.50	2.25
	3rd	1.50	2.25
Oil capacity — pints		2.6	3.50
Input shaft dia.		.94	1.06
No. of splines		10	10
Output shaft dia.		1.09	1.09
No. splines		25	25
O/A length trans. and ext.		22.75	24.06
Clutch dia.		8.50	10.00

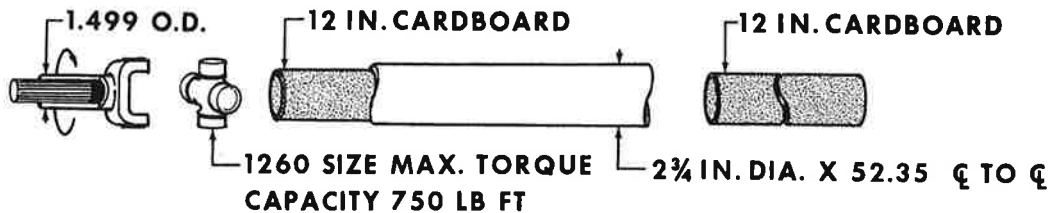
Next, it was necessary to review the remaining drivetrain components to ascertain their capabilities as related to the increased power and weight data.

DRIVE SHAFT AND JOINTS

1963 FALCON 260 CID V-8

OUTPUT SHAFT TORQUE
(LOW GEAR) 566 LB FT

MAX. OPERATING SPEED 4850 RPM
% OF CRITICAL SPEED 74.5%



1960 FALCON 144 CID 6-CYL.

OUTPUT SHAFT TORQUE
(LOW GEAR) 355 LB FT

MAX. OPERATING SPEED 3910 RPM
% OF CRITICAL SPEED 74.0%

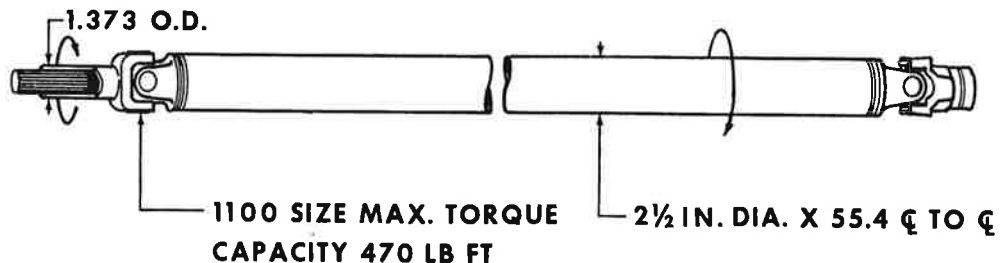


Figure 5

FIGURE 5

With better than a 60% increase in transmission shaft torque output, it was a foregone conclusion that the universal joints would have to be increased in size, which naturally increased the size and weight of the universal joint yoke and flange. The I-6 has an output shaft torque in low gear of 355 lb.-ft. and uses an 1100 size universal joint with a rated maximum capacity of 470 lb.-ft. while the V-8 installation has an output shaft torque in low gear of 566 lb.-ft. This torque requires universal joints of the 1260 size which have a maximum torque capacity of 750 lb.-ft.

The higher vehicle speeds possible with the V-8 naturally led us to review the driveshaft capabilities. A brief analysis of the pertinent data is sufficient to illustrate why, although the V-8 shaft is shorter than the I-6, it was necessary to increase its size.

LENGTH ϕ TO ϕ	TUBE DIA.	WALL THICKNESS	MAX. SPEED	N/V	MAX. SHAFT RPM	THEO. CRITICAL	MAX. SPEED AS % OF THEO. CRITICAL SPEED
144-55.4"	2.5	.065	85	46.0	3910	5280	74.0%
260-52.35"	2.75	.065	105	46.2	4850	6522	74.5%

FIGURE 5 (Continued)

This results in a maximum speed which is 74.5% of the critical speed on the V-8 while we have been producing Falcons with the 6-cylinder engine whose maximum speed is 74.0% of the critical speed. Tooling and production costs were minimized for the 1260 size universal joint V-8 shaft by utilizing production Fairlane slip yokes and shortened Ford tube assemblies. If a 2-1/2 diameter shaft had been incorporated, new weld yoke tooling would have been required. Driveline angles are comparable to those on the 144 C.I.D. installation. You will also notice that it was necessary to incorporate two twelve-inch cardboard liners, one at each end of the shaft, to eliminate driveshaft ring.

REAR AXLE CARRIERS

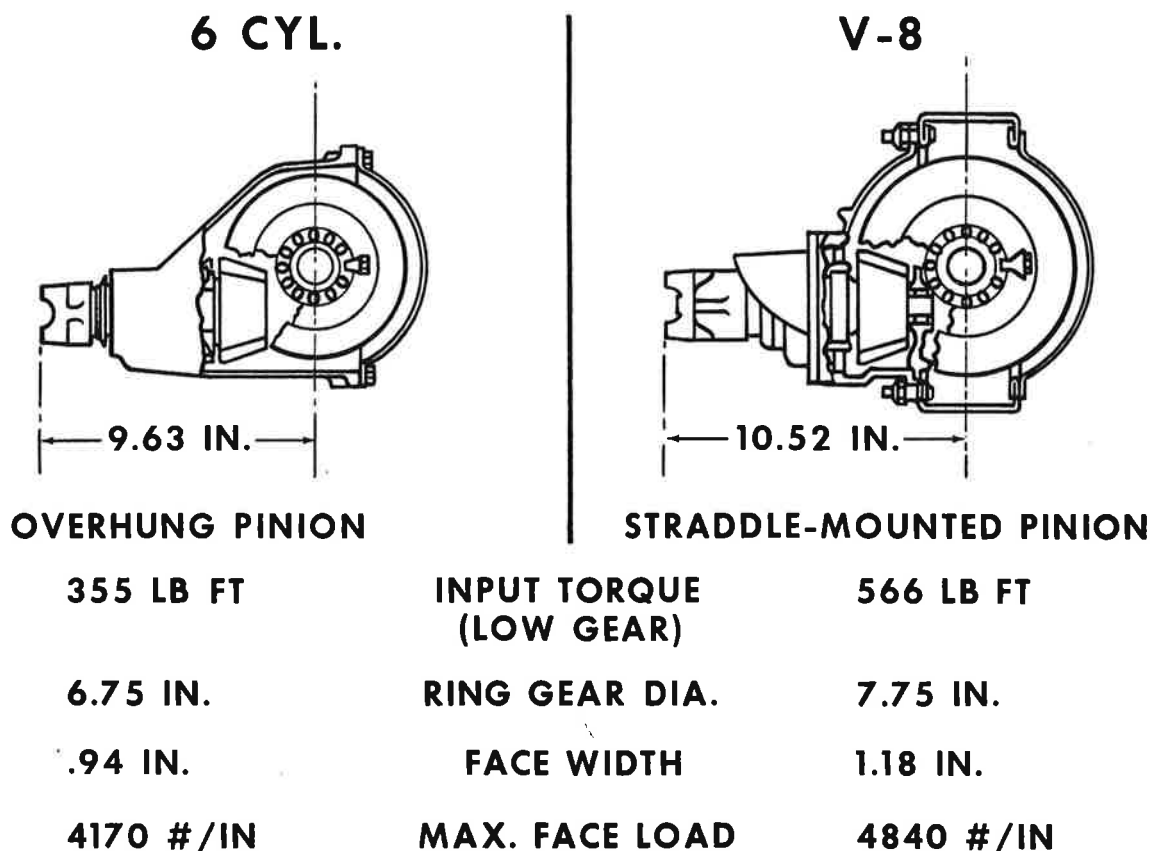


Figure 6

FIGURE 6

Past experience and tests indicate that the face loading on the axle ring gear should be kept below 5600 lbs/inch of force. In order to accomplish this, a modified Fairlane straddle-mounted axle was developed. This axle has a ring gear diameter of 7.75 inches which results in a maximum face loading of 4840 lbs/inch. This compares with the 6-cylinder overhung axle which has a 6.75 inch diameter ring gear with a maximum face loading of 4170 lbs/inch.

FINAL DRIVE

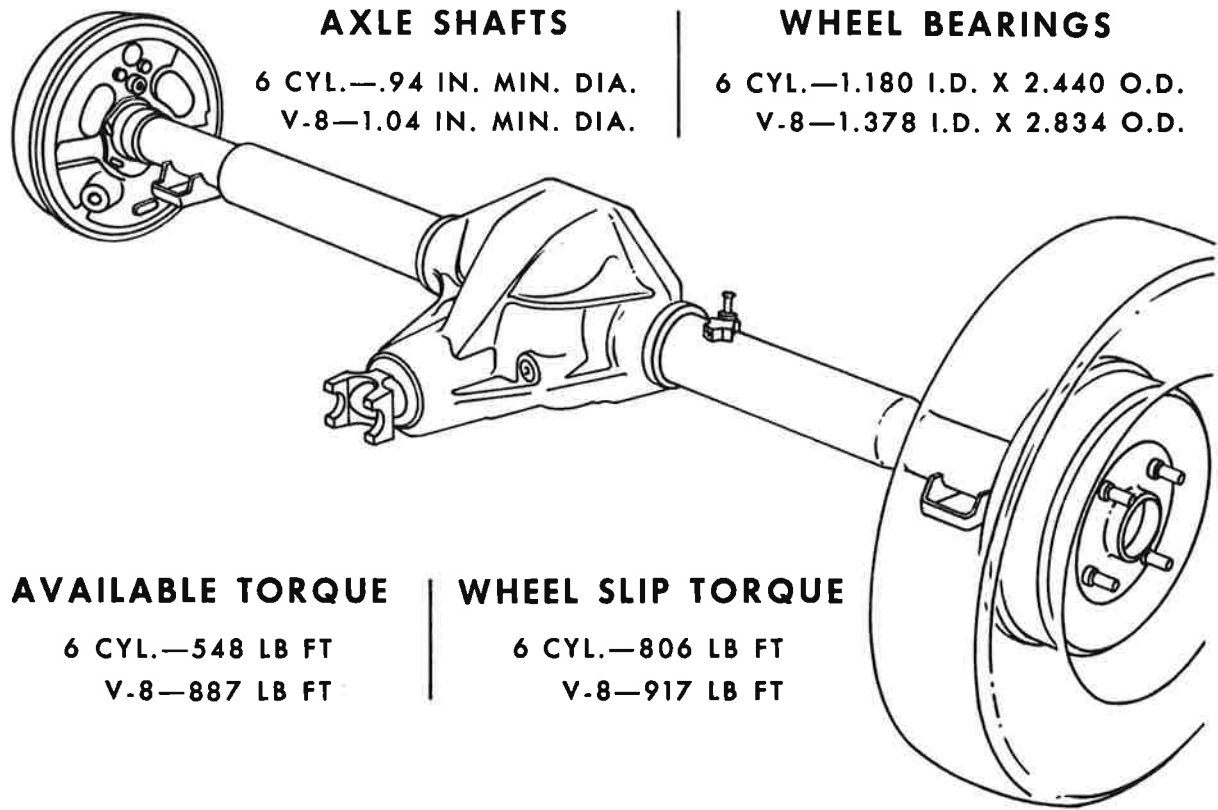


Figure 7

FIGURE 7

With the additional torque and vehicle weight, it was necessary to increase the axle shaft diameter from the .94 inch used on the 6-cylinder to 1.04 inch for the 8-cylinder vehicle. Wheel bearings also had to be increased in size; the 6-cylinder bearings are 1.18" I.D. x 2.44" O.D., with a design factor of 1.23. The bearings released for the 8-cylinder Falcon are 1.378" I.D. x 2.834" O.D. with a design factor of 1.41.

All of the above actions are, of course, caused by the increase in available torque and, to some extent, by the additional weight which is accumulated as we move thru the design. In the final analysis, there is available at the wheels a torque of 548 lb.-ft. with the 6-cylinder and 887 lb.-ft. with the 8-cylinder engine. The wheel slip torque for the 6-cylinder is 806 lb.-ft. and for the 8-cylinder 917 lb.-ft.

REAR SUSPENSION LOADS

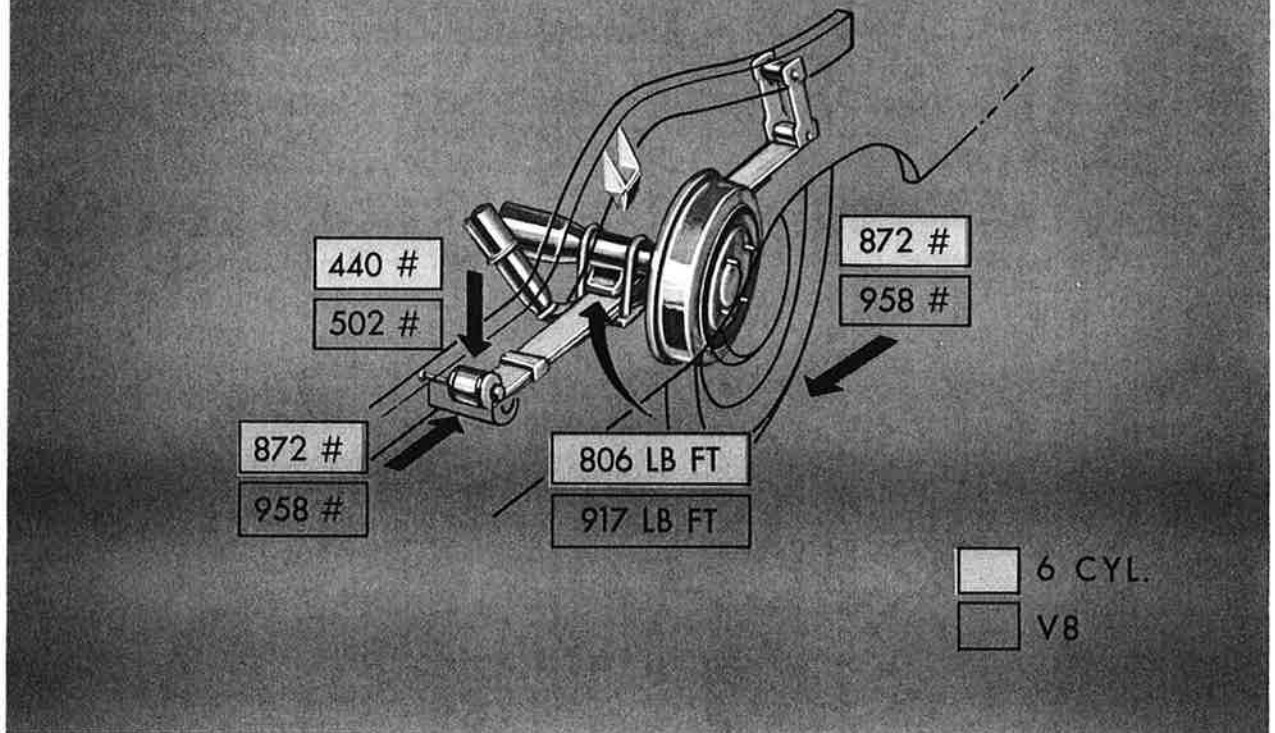


Figure 8

FIGURE 8

As the torque at the rear wheels is increased, the components reacting to this force, namely the rear springs and attachments, had to be analyzed. The 6-cylinder rear springs had a loaded stress of 170,000 P.S.I. With the new loads, this level would have been increased to an unacceptable 204,000 P.S.I. Therefore, a new 6-leaf rear spring was designed for all of the V-8 models which gives a stress load of 163,000 P.S.I. The rear spring front eye bushing and attachments were also redesigned. The new design incorporates a Harris-type bushing with an increased diameter attaching bolt and brackets. This resulted in an increase in projected bearing area of 22%.

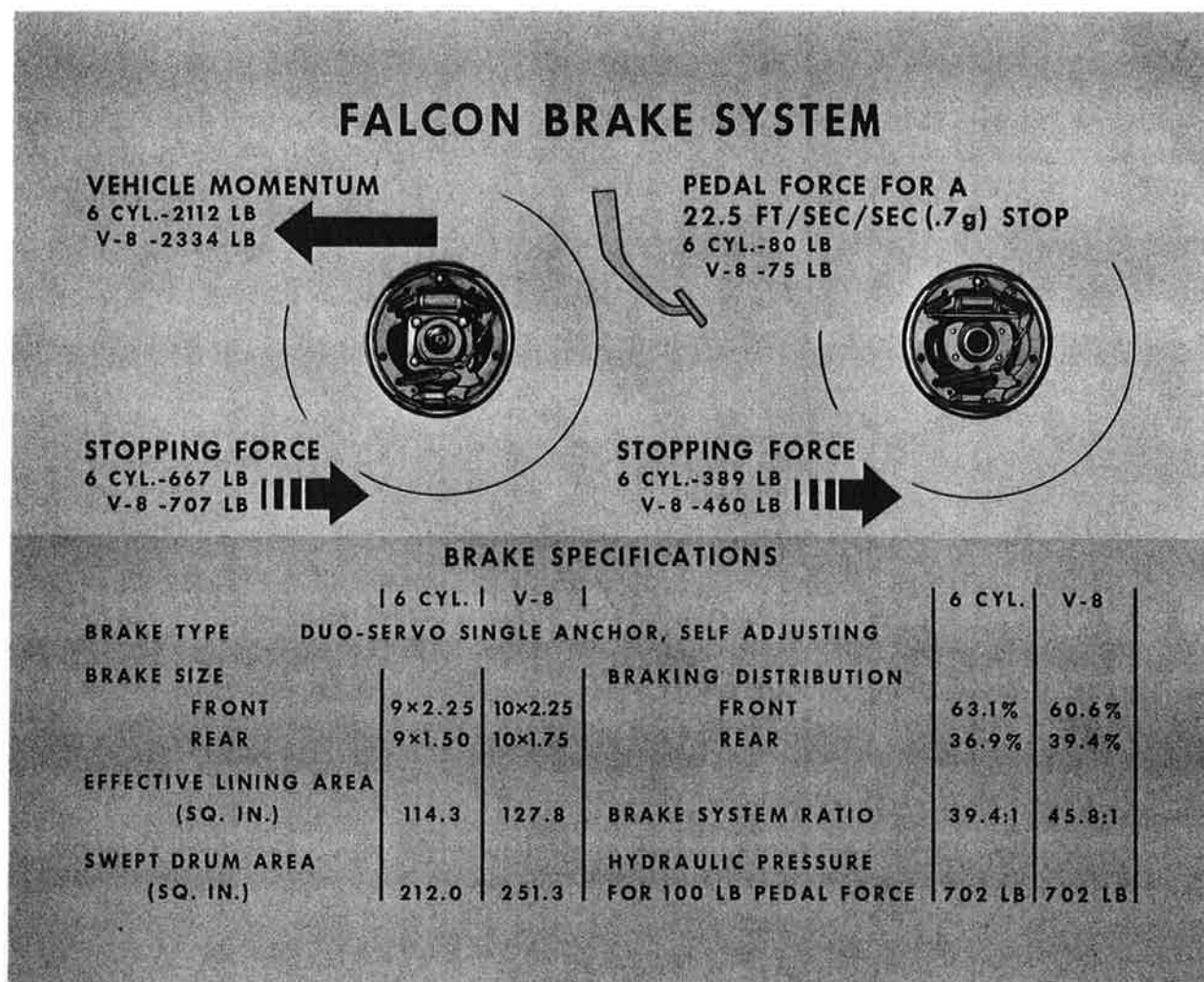


Figure 9

FIGURE 9

The 6-cylinder Falcon brake system has proven itself to be adequate for the vehicle during the past 3-1/2 years. However, the added vehicle weight resulting from incorporating the V-8 engine and the increased engine performance necessitated larger brakes.

In order to determine the new brake size requirements, brake loading which may be defined as the rate of energy absorbed by each front and rear brake under specific braking conditions was used as the criteria for making the selection. The loading is based on the energy absorbed during a 22.5 ft./sec./sec. rate of deceleration stop from a vehicle speed of 70 M.P.H. and expressed as kinetic energy per square inch of effective lining area per second of stopping time.

Using the base Falcon sedan as an example, the 6-cylinder model has the following loads:

Front brake — 997 ft.-lbs./sq. in./sec.
Rear brake — 873 ft.-lbs./sq. in./sec.

and the 8-cylinder model:

Front brake — 988 ft.-lbs./sq. in./sec.
Rear brake — 868 ft.-lbs./sq. in./sec.

FIGURE 9 (Continued)

If no changes in brake size were made, the added vehicle weight would have resulted in the 6-cylinder figures increasing to:

Front brake — 1104 ft.-lbs./sq. in./sec.

Rear brake — 970 ft.-lbs./sq. in./sec.

Changing the braking distribution to the 8-cylinder system ratio while retaining the 6-cylinder system brake sizes unloads the front brake only slightly while overloading the rear brakes as shown:

Front brake — 1061 ft.-lbs./sq. in./sec.

Rear brake — 1038 ft.-lbs./sq. in./sec.

Past experience with Duo-Servo brakes has established that if the rate of energy absorption is allowed to exceed 1100 ft.-lbs./sq. in./sec. for the front brake and 1000 ft.-lbs./sq. in./sec. for the rear brake on the heaviest model using the brake system, inadequate fade resistance, poor vehicle stability and low lining life can be anticipated. A spread of 100 ft.-lbs./sq. in./sec. is common for both the front and rear brake between the base vehicle and the heaviest vehicle.

Rear brake loading which approaches the front brake value has been found to be unacceptable from the lining life standpoint because of the cooling problems resulting from the restricted air flow under the rear half of the vehicle.

Once the lining area requirements for the 8-cylinder vehicle were established, it was found that the current Fairlane brake system met the requirements and would be more economical than tooling new, wider 9-inch brakes.

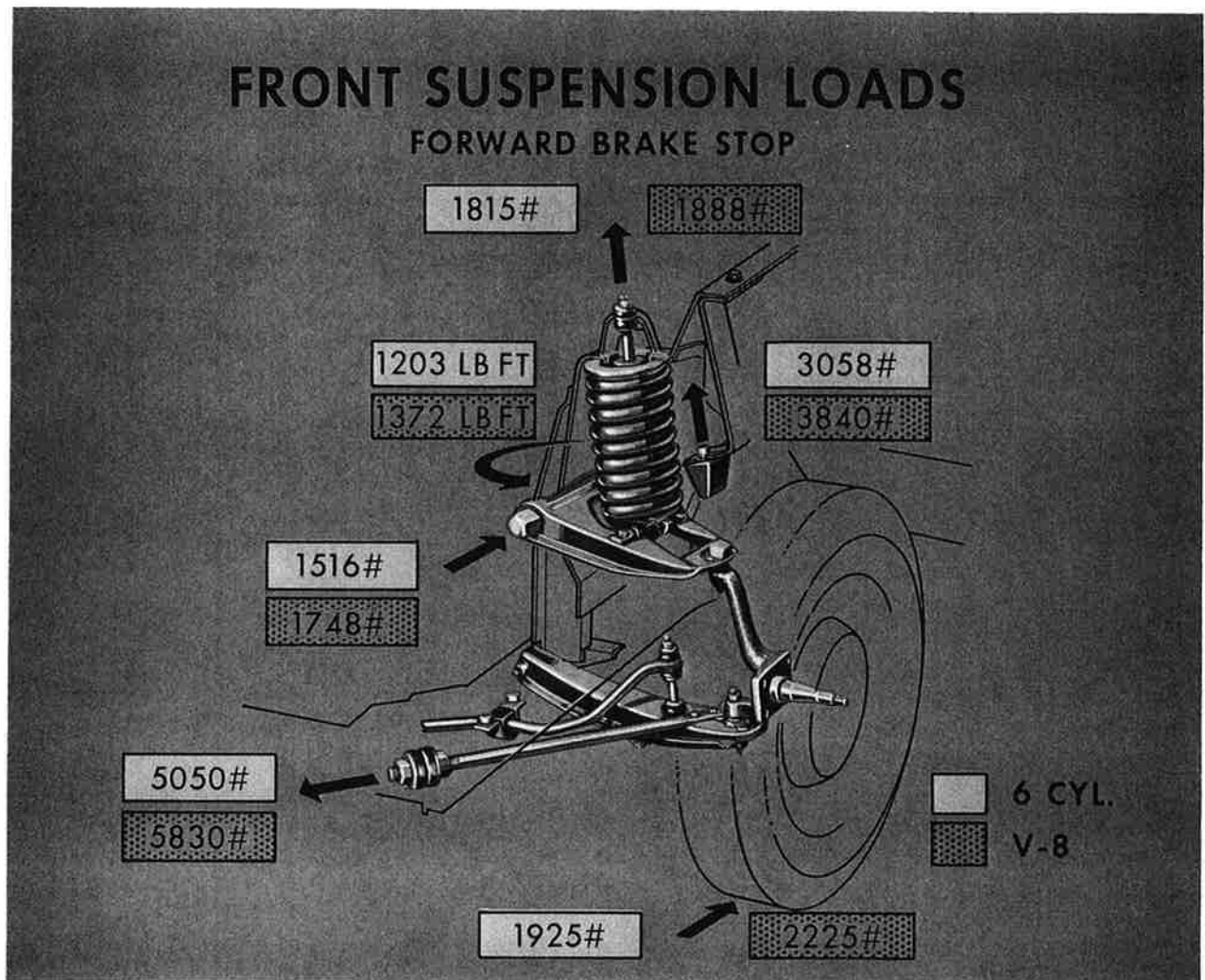


Figure 10

FIGURE 10

The additional weight of the V-8 package materially changed the jounce and spike stop loading of the front suspension components. Spring loads were increased approximately 190 lbs. per model to correct ride heights. Other components that changed are upper arm and ball joint assembly, spindle, strut, jounce bumper, and stabilizer bar.

A heavier spindle was required to accommodate the increased brake capacity. With the heavier spindle, a revised steering stop was needed. A new strut incorporating an integral steering stop was released for V-8 usage. The upper arm and joint assembly had to be strengthened to withstand the loads of spike brake stops. As shown in the slide, the couple applied above the upper arm inner pivot shaft in the horizontal plane has increased from 1203 lb.-ft. to 1372 lb.-ft. The longitudinal load increased 232 pounds or 15%. Similarly, the tensile load in the strut increased from 5050 lbs. to 5830 lbs.

Clearance problems were encountered in several areas. A tire size increase, coupled with a tread increase of .28 inch per side, resulted in tire to sheet metal interference at fender wheel cutout. Also, the heavier front springs introduced coil clash in some models. These conditions were relieved by restricting front jounce travel to 3.5 inches instead of the standard 4.0 inches. This was done by lengthening the jounce bumper and incorporating a metal stop at the base of the bumper. This solution was not without a penalty. The .5 inch restriction increased jounce bumper loads as much as 38% for a given load input at the road wheel.

The stabilizer bar interfered with the fuel filter used on the V-8 engine. A new configuration was developed for use on both 6-cylinder and 8-cylinder models.

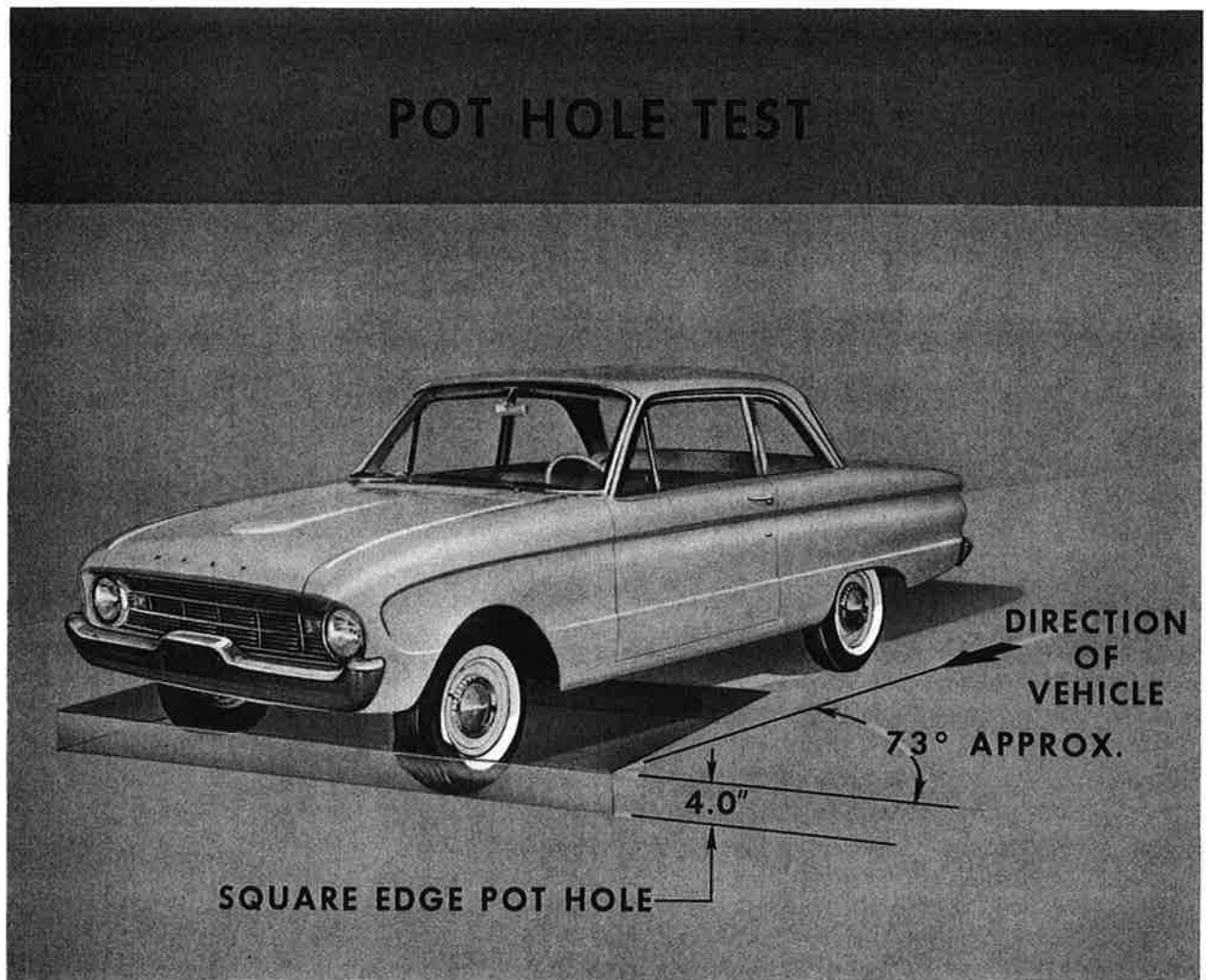


Figure 11

FIGURE 11

In the case of the initial design of a steering system, the forces applied to various system components (gear, linkage, gear mounting, etc.) are estimated based on static forces imposed by means of a maximum steering wheel rim pull load of 135 pounds. This force related thru the steering gear ratio and considering various suspension and road wheel turn positions is used to determine the size of steering components required.

The next stage used in determining the adequacy of the system to withstand the required durability is the program and special test load determination. These tests are used to establish the maximum stresses that various steering components are subjected to and are accomplished as follows:

The steering system is stress coated and subjected to forces so that the high stress areas are determined.

Strain gages are then placed in the high stress and other strategic areas to determine compressive, tensile, torsional and bending loads on all components (tie rods, center link, gear, idler arm, etc.). These gages are used in conjunction with an electrical recorder that indicates load, amplitude and frequency of occurrence. Then the instrumented

FIGURE 11 (Continued)

parts are installed in a test vehicle and subjected to an established load determination test course and procedure which includes road conditions and special tests, such as:

- Belgian blocks
- Cobblestones
- Chuck holes
- Gravel
- Sine Waves
- Running into curbs at various angles
- Jacking the vehicle away from the curb by the road wheel (static)
- Spike brake stops
- Panic brake stops

At the completion of these tests, the data is analyzed and curves are plotted to show load and occurrence relationship. This data is then used to establish laboratory fatigue, impact, and wear tests.

STEERING LINKAGE LOADS (IN POUNDS)

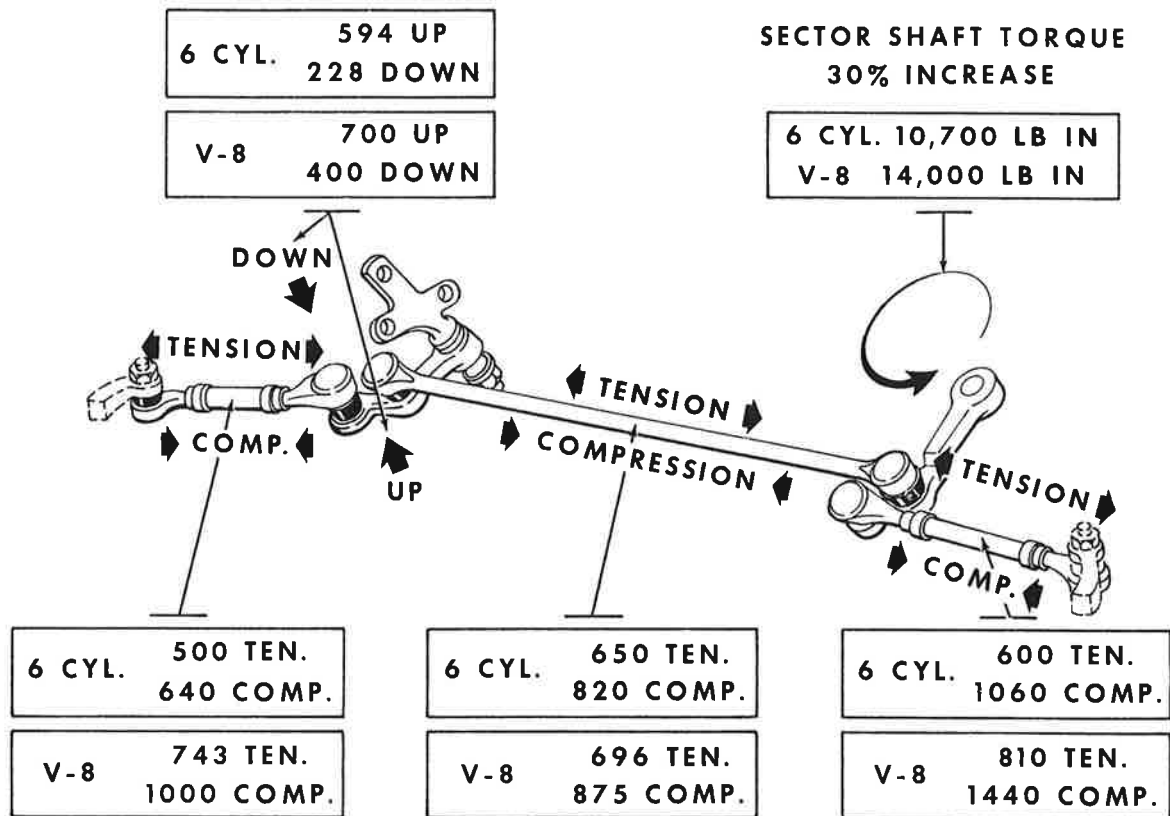


Figure 12

FIGURE 12

The entire steering linkage, including the steering gear, was redesigned to adequately withstand the increased loads encountered with the V-8 installation. This illustration of the steering linkage shows a comparison of loads obtained in 6-cylinder and 8-cylinder vehicles using the procedure for program load determination. Note that tension and compression loads in the tie rods increased as much as 36%. Because of the short tie rods used on the Falcon and the resulting linkage angles, a considerable portion of this increase in longitudinal load is taken by the idler and pitman arms as a bending load. In the V-8 installation, the idler arm load has increased 18% upward and 75% downward. Such loads made it necessary to increase the shank diameter of the idler arm mounting bracket. This change, together with the conical load increase, required a greatly enlarged idler arm rubber bushing. The mounting bolt pattern of the idler arm bracket was changed from 2 bolts to 3 bolts to provide better load distribution to the body rail.

STEERING IDLER ARM FORCES

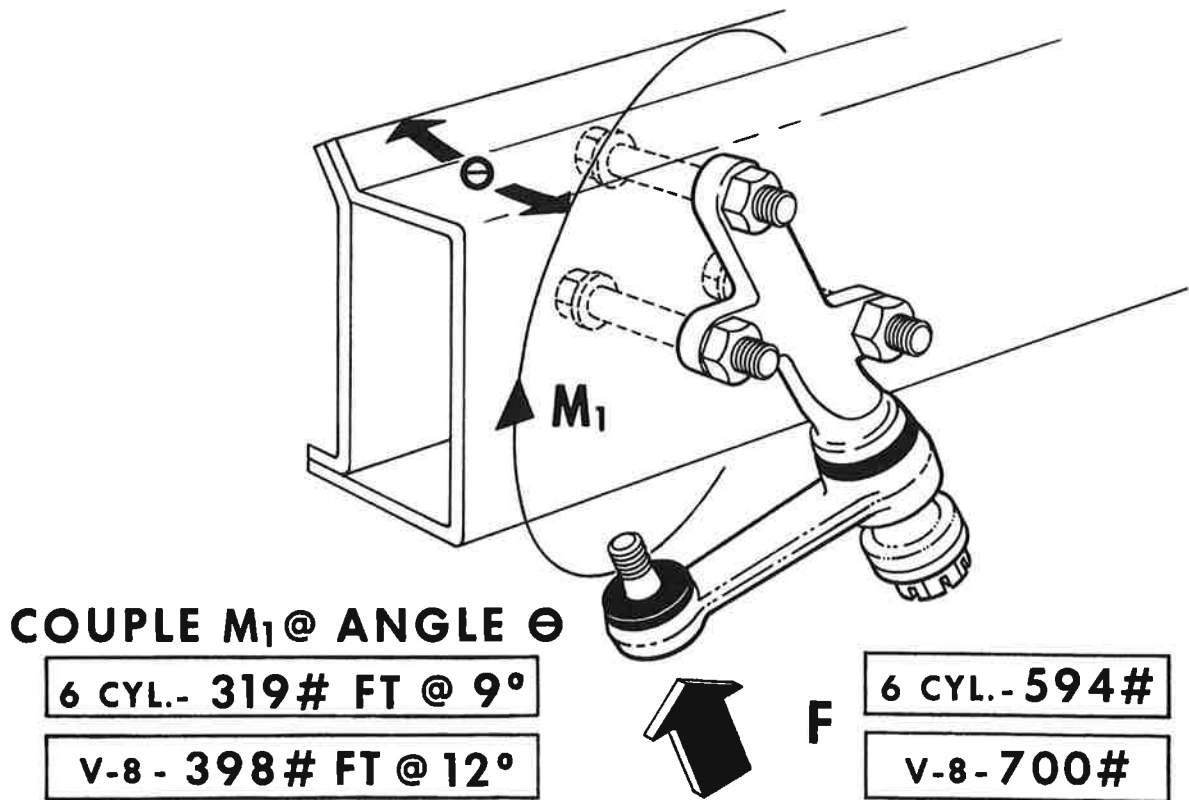


Figure 13

FIGURE 13

As shown here in the straight-ahead position, the axial load along the idler arm mounting bracket and the effective lever arm increased. The result was a 25% increase in the couple acting at a small angle across the rail structure. Although it only takes a minute to review this data with you, the effect of it on the car was significant. As a matter of fact, you will see, when we get to the body structure, the effect that these loads had on the idler arm and steering gear spacers in the rail.

STEERING GEAR TORQUE

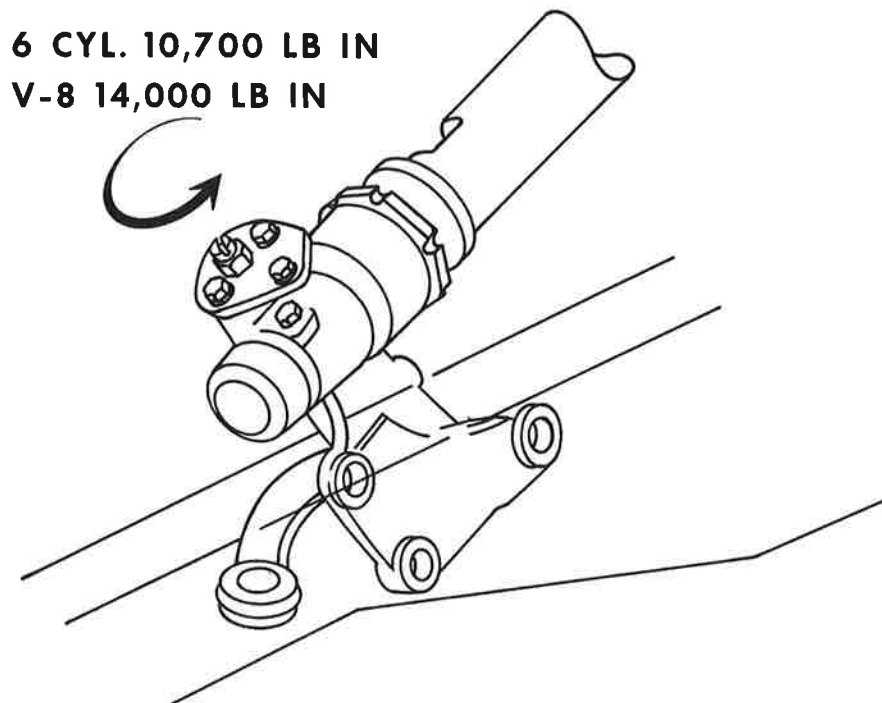


Figure 14

FIGURE 14

Similar loads will occur at the steering gear attachments, but the attaching bolt pattern provides a more advantageous load distribution.

Though the steering gear attachment was satisfactory, the gear sector shaft required additional strength. You will note that sector shaft torque has jumped 30% from 10,700 lb.-in. to 14,000 lb.-in. To provide added capacity in the steering gear, the sector shaft diameter was increased from 1 inch to 1.12 inches.

The enlarged idler arm, bushing, the increased weight of the car, and the enlarged steering linkage joints all tended to increase steering efforts, particularly under static conditions. To partially counteract this, needle bearings at the sector shaft are used in place of the usual bronze bushings. This results in approximately a 5% reduction in efforts, needle bearings vs. bronze bushings.

FRONT SIDE MEMBER REVISIONS

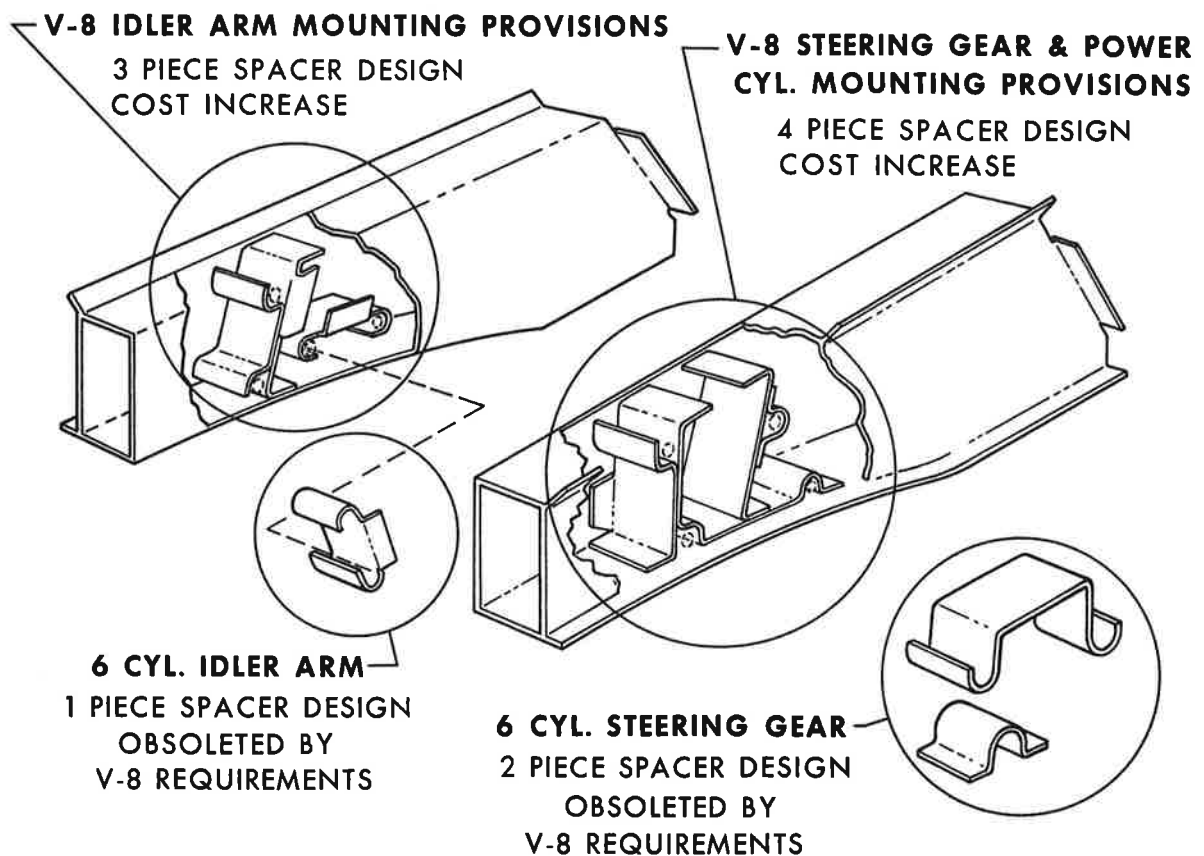


Figure 15

FIGURE 15

As you would expect, the loads requiring redesign of the idler arm and steering gear, also, affected the associated body components. The 6-cylinder Falcon had a relatively uncomplicated single-piece spacer at the idler arm bracket and a light two-piece spacer at the steering gear attachment. With the increased loads and larger attaching bolts required for the V-8 installation, it became evident that these spacers were not adequate to resist the forces being put into them. Consequently, it was necessary to design a relatively heavy and complicated three-piece spacer for the idler arm attachment and a four-piece spacer for the steering gear attachment. This latter spacer also has provisions for power steering mounting. Power steering now being a desirable feature for the heavier steering V-8's.

FRONT END STRUCTURE MODIFICATIONS

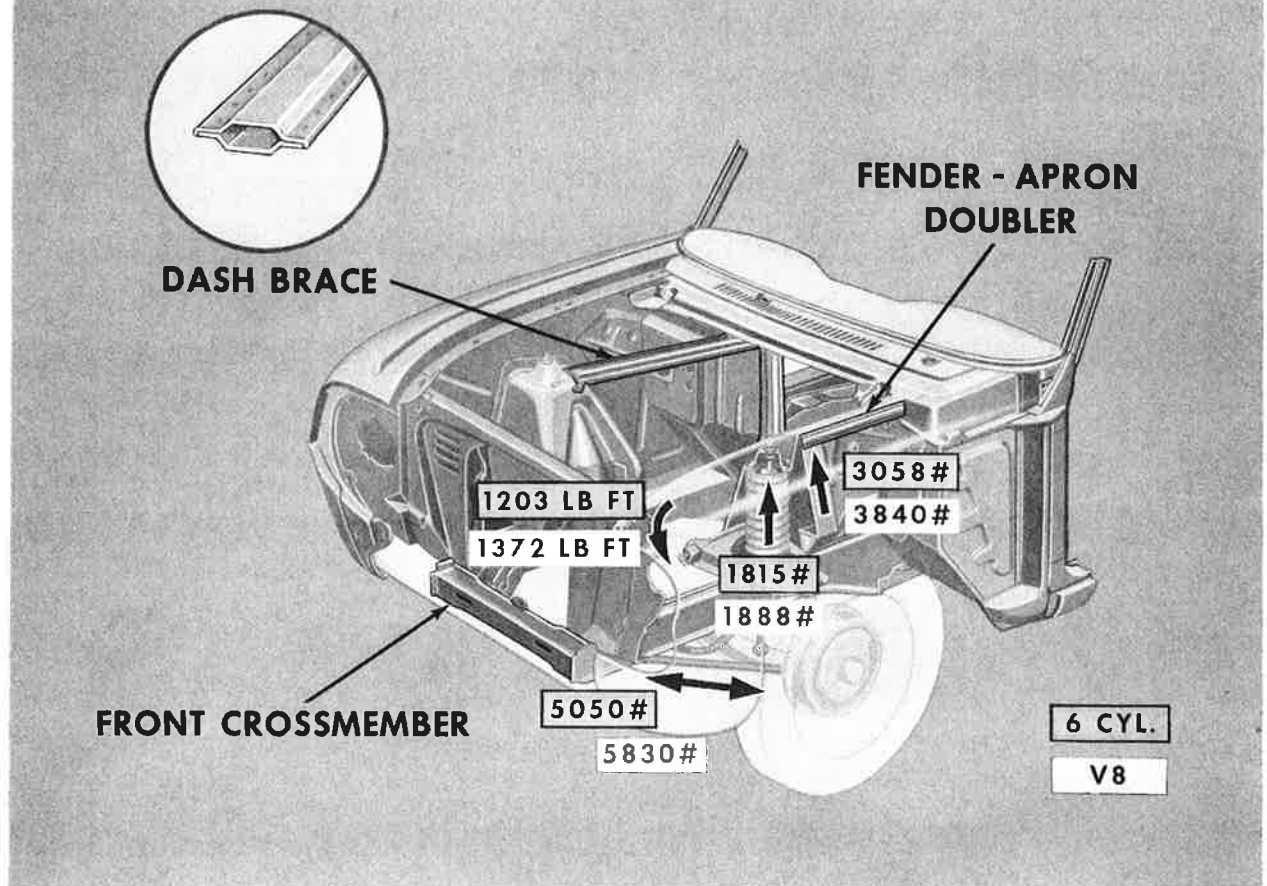


Figure 16

FIGURE 16

From the "programmed load" study, previously discussed, it was evident that structural improvements in the body were required. The combined torsional and bending loads imposed on the front crossmember were increased by 15% and necessitated an increase in gage of this member. Restricting the jounce to 3.5 inches coupled with the added weight of the V-8 engine resulted in loads into the structure considerably higher for the 8-cylinder than those measured on the 6-cylinder installation. To counteract these increased forces, a hat section was added to the dash brace, and a doubler was added to the fender apron.

BODY STRUCTURE REVISIONS

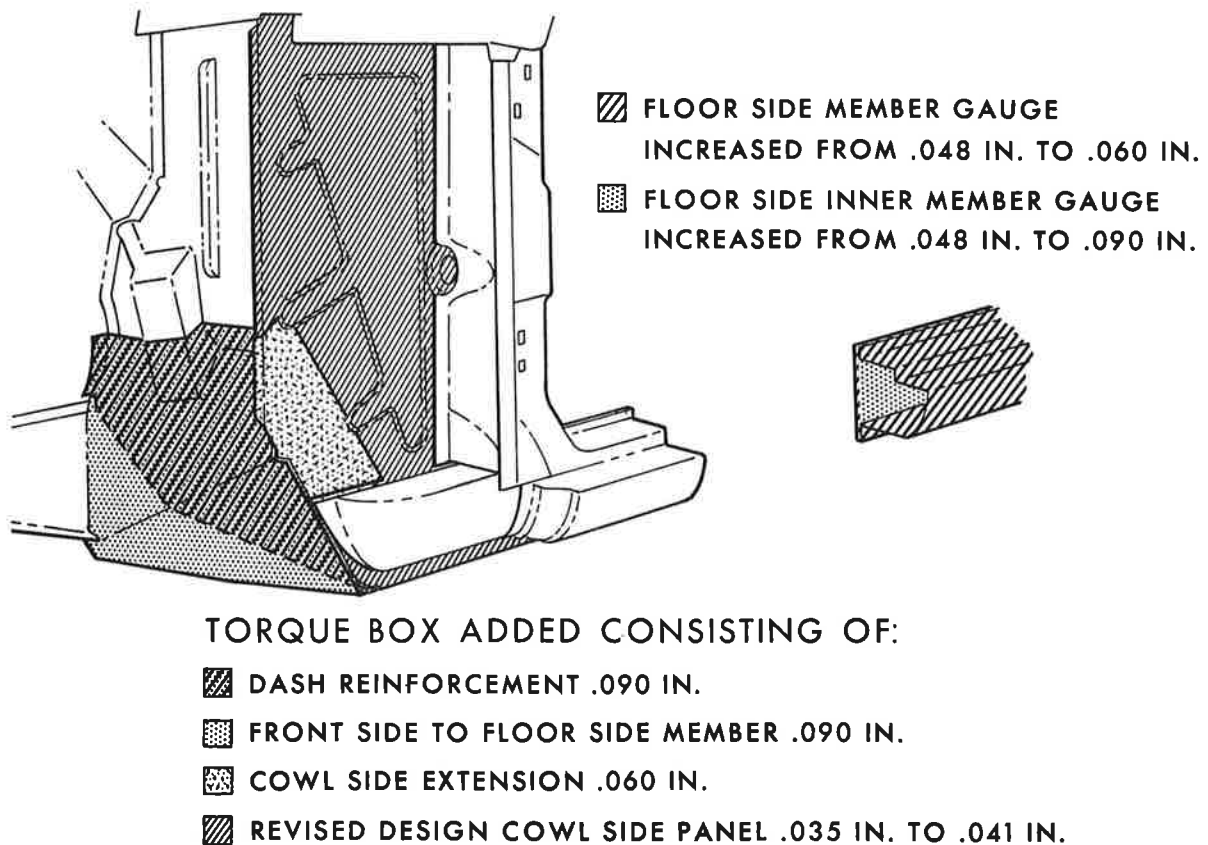


Figure 17

FIGURE 17

Along with the modifications that were designed to accept the additional loads imposed by the 8-cylinder engine forward of the dash, it was also necessary to accomplish a redesign of those body areas which transmit these loads into the basic body structure. The addition of torque boxes between the front side rail and the body sill along with gage increases in the cowl side panel from .039 to .041 accomplished this function. Additional bending rigidity was obtained by increasing the floor side inner member from .048 to .090 and the floor side member from .048 to .060. The windshield pillar and the roof structure were examined to determine their capacity to accommodate the 8-cylinder engine load input and were found to be adequate.

PACKAGE MODIFICATIONS

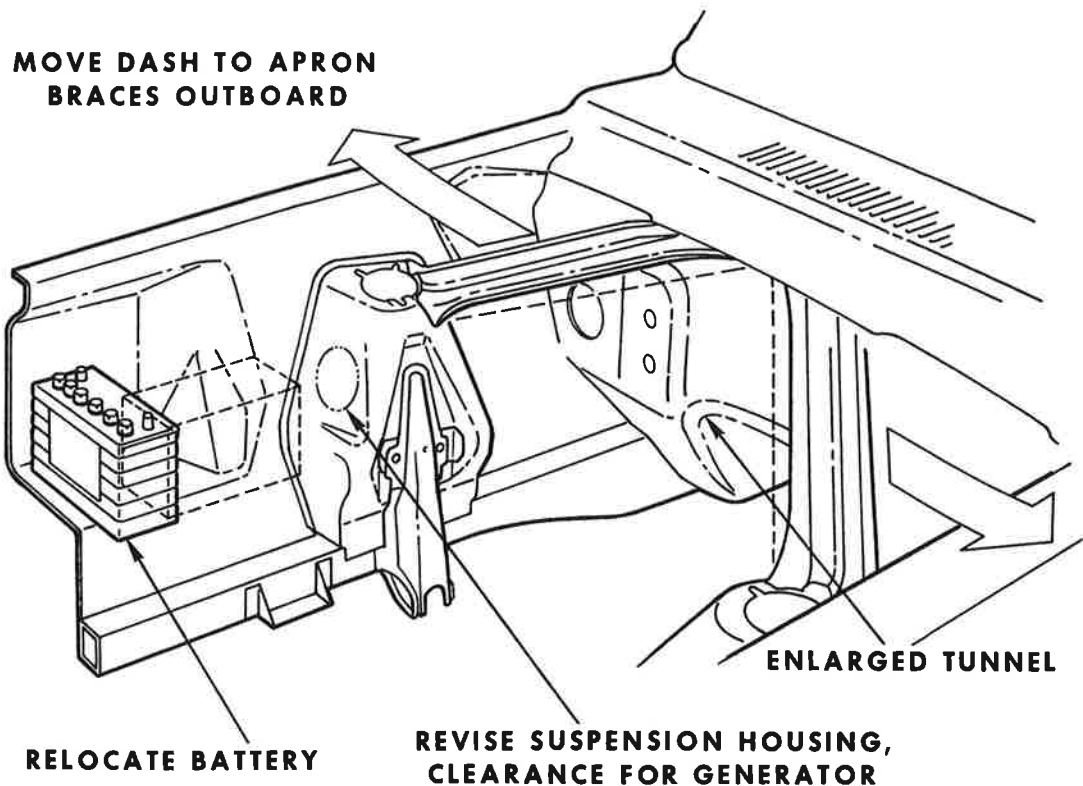
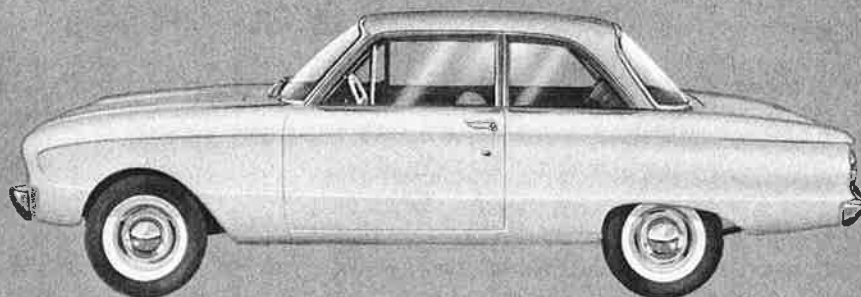


Figure 18

FIGURE 18

You will recall that the V-8 engine was 7" wider and 2.1" higher than the 6-cylinder engine. You will also remember that it was necessary to provide a unique flywheel housing and transmission for the V-8 installation. These requirements necessitated a rearrangement of the engine compartment package and, in fact, also required changes to the tunnel. In order to deck the V-8 from the top into the body, it was necessary that the dash to apron braces be moved outboard. In order to provide clearance for the V-8 generator and the exhaust manifolds, the battery had to be moved from its 6-cylinder location. This necessitated a redesign of the front fender apron and the radiator support panel. During this redesign, provisions were made to accommodate power steering. The tunnel in the passenger compartment was increased in height and width to accommodate the new flywheel housing and transmission. All of these redesigns were made in such a way that they would be compatible with 6-cylinder installation at Job 1, 1963-1/2. Thus, the basic body shell can be used for both V-8 and 6-cylinder engine installations.

TIRE DATA



1963 6 CYL. SEDAN

749.5 #/WHEEL

2904#

702.5 #/WHEEL

1963 V-8 SEDAN

W/ACCESSORIES

948 #/WHEEL

3381#

742.5 #/WHEEL

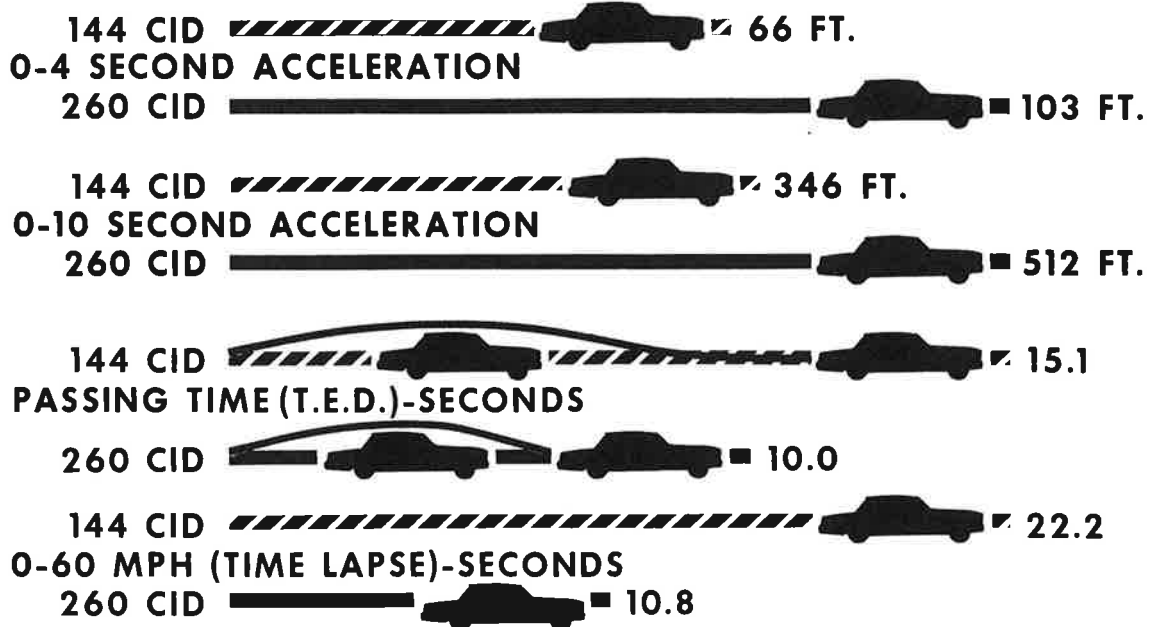
RECOMMENDED TIRE SIZE	TIRE & RIM LOAD RATING	MAXIMUM TIRE OVERLOAD (UNDERLOAD)	
		6 CYL.*	V-8 •
6.00x13 *	725	3.5%	31.0%
6.50x13 •	835	(10.2)%	13.6%
7.00x13	920	—	3.5%

Figure 19

FIGURE 19

One of the major concerns in regard to the installation of the V-8 was the considerable weight penalty which resulted from the various design actions necessary to accommodate it. You will notice that with tire and rim association loading, the 1963 6-cylinder sedan weighs 2904 pounds while the 1963-1/2 V-8 sedan with accessories under the same conditions weighs 3381 pounds. This is a weight spread of 477 pounds or a 17% increase for the V-8. The major portion of this additional weight is carried by the front wheels with an increase of 26% for the V-8 or a total of 948 pounds per wheel on the front. If we had attempted to maintain the standard 6.00 x 13 size tire released for the 6-cylinder on the 8-cylinder, we would have been operating with the tire 31% overloaded. Therefore, it was necessary to release the 6.50 x 13 tire for some V-8 models and the 7.00 x 13 tire for the heaviest models. This provides the most economical usage consistent with good safety practice. Because it was necessary to use Fairlane-type brakes on the V-8, the Falcon V-8 has unique wheels which are compatible with the brake system. The basic difference between the 6-cylinder and 8-cylinder wheels is that the 6-cylinders are a 4 lug arrangement while the V-8 has the standard 5 lug attachments. In effect, all of the actions which were taken from a design standpoint in order to accommodate the V-8 result in practically a completely redesigned vehicle. From the wheels through the powertrain and into the body, each component was analyzed from a load and stress standpoint and design action taken where necessary to provide a vehicle which would have the same good durability and reliability characteristics as has been enjoyed by the 6-cylinder version.

PERFORMANCE



FUEL ECONOMY



Figure 20

FIGURE 20

Aside from the improvement in sound level and engine smoothness of the V-8, most customers are interested in the performance advantages associated with the greater displacement engine. Of course, some of the advantage is lost due to the added weight of the V-8 package. But even so, the performance data for the Falcon V-8 is pretty respectable. 103 ft. in 0 to 4 seconds as compared to 66 ft. for the I-6. In 0 to 10 seconds, the V-8 covers 512 ft. while the 6 goes 346 ft. In measuring 0 to 60 MPH performance, the V-8 requires only 10.8 seconds while the I-6 needs 22.2 seconds.

In time exposed to danger, the V-8 requires only 10 seconds while the I-6 requires 15.1 seconds. This time is that required by a vehicle running at 50 MPH to accelerate from a position one car length behind to one car length in front of a second vehicle traveling in the same direction at 50 MPH. With this performance, the V-8 has actual customer fuel economy of 17.6 M.P.G. This compares with 22.1 M.P.G. measured for the I-6 under the same conditions.

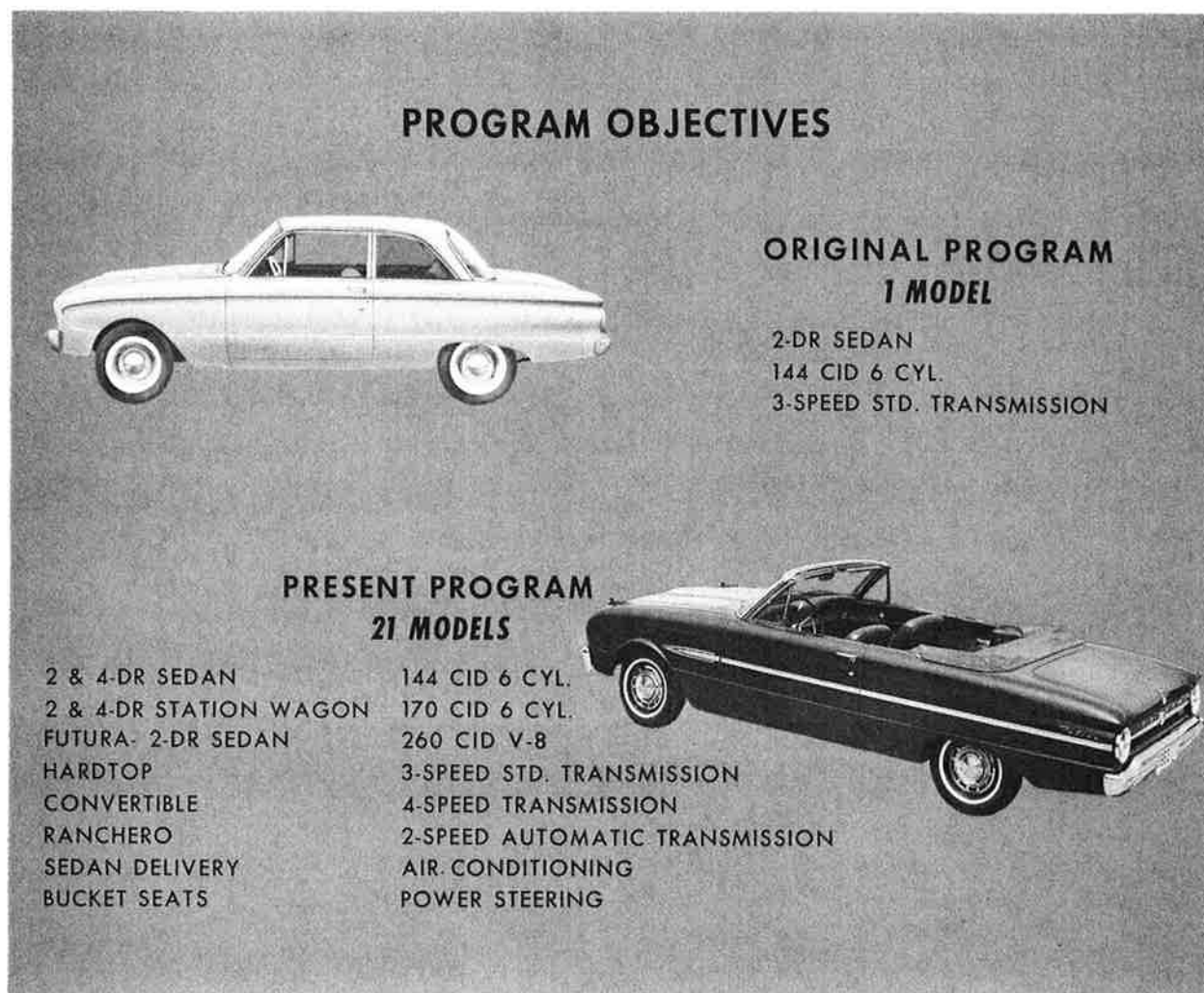


Figure 21

FIGURE 21

In retrospect, it is interesting to speculate, at least from an engineering standpoint, just how many of these design changes should have been incorporated into the original design of the Falcon. Certainly some of them could have been included at very little, if any, penalty to the basic car; others, of course, are too expensive to be carried in a program which was directed at minimum weight and cost objectives.

We thought that, in conclusion, you might like to see what has happened to the minimum weight, minimum cost Falcon in its short four-year life.

ORIGINAL PROGRAM
1 MODEL

2-door sedan
144 C.I.D. 6-cylinder
3-speed standard transmission

PRESENT PROGRAM
21 MODELS

2- and 4-door sedan
2- and 4-door station wagon
Hardtop

Futura 2-door sedan
Convertible
Ranchero
Sedan delivery
Bucket seats
144 C.I.D. 6-cylinder
170 C.I.D. 6-cylinder
260 C.I.D. V-8
3-speed std. transmission
4-speed transmission
2-speed automatic transmission
Air conditioning
Power steering

