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THE POTENTIAL FOR AUTOMOBILE WEIGHT REDUCTION OUTLOOK AS OF 1975-1976

by

Frederick J. Hooven
Francis E. Kennedy, Jr.

Thayer School of Engineering
Dartmouth College
Hanover NH 03755

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FINAL REPORT

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Frederick J. Hooven, Francis E. Kennedy

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Dartmouth College
Hanover NH 03755

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16. Abstract

A study has been carried out to evaluate the potential for weight reduction of automobiles. It is concluded that automobiles of lighter weight than those current (1976 models) in the domestic market may be built in each of three size classes, 4, 5, and 6 passenger, and that this can be done within the limits of presently prevailing designs and materials. Detailed weight breakdowns are presented for a 4-passenger vehicle of 1987 lbs. curb weight, for a 5-passenger vehicle having a curb weight of 2551 lbs., and a 6-passenger vehicle of 3271 lbs. curb weight.

Since weight reduction is without significance unless product characteristics are well defined, detailed package dimensions are presented for each of the three size classes, and an improved roominess factor is proposed for the evaluation of effective roominess in cars. Other vehicle characteristics are discussed in detail and defined. It is concluded that roominess, performance, active safety, luxury options and other product qualities can be conserved in weight reduction, but superpower options and weight concessions to appearance factors are not included in weight estimates. It is also concluded that lightweight cars may be somewhat inferior to vehicles of heavier weight and equally advanced design with respect to 1) impact with other vehicles and with certain kinds of fixed obstacles, 2) aerodynamic stability, and 3) smoothness and quietness of operation.

Detailed weight comparisons are presented between present and past designs of vehicles in each of the three size classes mentioned above. The weight history of three typical domestic vehicles is given in detail and analyzed for the period 1970-74, during which time the weight consequences of industry response to government regulations entered the picture.

The design process is outlined and illustrated by the making of design decisions for a new design of a 5-passenger vehicle. Finally, the potential for weight reduction by material substitution is examined. Three material groups are discussed in detail - aluminum, high strength low alloy steel, and plastics - and a study is made of the weight decreases that could be achieved by use of those materials in automobile components where such use is technologically feasible.

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PREFACE

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The authors also acknowledge the assistance of the Ford Motor Company, who supplied detailed weight information on Ford products, on which much of this work was based, of A.C. Sampietro, who was consulted on vehicle and engine design, of the Hollowell Engineering Company, who prepared package information and drawings, of the members of the Automotive and Truck Subcommittee of the Aluminum Association, of David Adams of Chrysler Corporation, and of John Collier and John Boyle of Dartmouth College, who obtained weight and roominess data for existing vehicles.

The work on this project was completed under the sponsorship of the U.S. Department of Transportation, National Highway Traffic Safety Administration's Automotive Fuel Economy Regulating Program at the Transportation Systems Center.

METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	square miles	2.6	square kilometers	km^2
	acres	0.4	hectares	ha
		MASS (weight)		
oz	ounces	28	grams	g
lb	pounds	0.45	kilograms	kg
	short tons (2000 lb)	0.9	tonnes	t
		VOLUME		
tsp	teaspoons	5	milliliters	ml
Tbsp	tablespoons	15	milliliters	ml
fl oz	fluid ounces	30	milliliters	ml
c	cups	0.24	liters	l
pt	pints	0.47	liters	l
qt	quarts	0.95	liters	l
gal	gallons	3.8	liters	l
ft^3	cubic feet	0.03	cubic meters	m^3
yd^3	cubic yards	0.76	cubic meters	m^3
		TEMPERATURE (exact)		
°F	Fahrenheit temperature	5/9 (after subtracting 32)	Celsius temperature	°C

Approximate Conversions from Metric Measures

Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
mm	millimeters	0.04	inches	in
cm	centimeters	0.4	inches	in
m	meters	3.3	feet	ft
m	meters	1.1	yards	yd
km	kilometers	0.6	miles	mi
		AREA		
cm^2	square centimeters	0.16	square inches	in^2
m^2	square meters	1.2	square yards	yd^2
km^2	square kilometers	0.4	square miles	mi^2
ha	hectares (10,000 m^2)	2.5	acres	
		MASS (weight)		
g	grams	0.035	ounces	oz
kg	kilograms	2.2	pounds	lb
t	tonnes (1000 kg)	1.1	short tons	
		VOLUME		
ml	milliliters	0.03	fluid ounces	fl oz
l	liters	2.1	pints	pt
l	liters	1.06	quarts	qt
l	liters	0.26	gallons	gal
m^3	cubic meters	35	cubic feet	ft^3
m^3	cubic meters	1.3	cubic yards	yd^3
		TEMPERATURE (exact)		
°C	Celsius temperature	9/5 (then add 32)	Fahrenheit temperature	°F

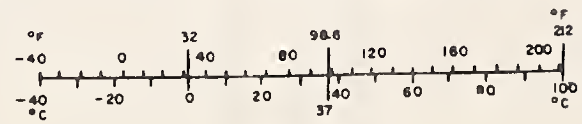


Table of Contents

LIST OF ILLUSTRATIONS

LIST OF TABLES

LIST OF TABLES (CONTINUED)

1. DESIGN ASSUMPTIONS - CONSTRAINTS AND CRITERIA

1.0 INTRODUCTION

The only basis on which automobile weights can be compared and evaluated is on the assumption that other qualities are maintained as nearly as possible without change. Consequently we define in some detail these criteria that must be evaluated in any strict comparison of automobile weights.

The primary vehicle characteristic to be conserved in weight reduction is functional size, interior space, which we have taken some pains to evaluate for purposes of comparison of different designs.

Those characteristics relating to active safety, durability, and reliability can be conserved with weight reduction, and accordingly we have adopted the position that these cannot be compromised for incremental weight advantages. Durability and reliability cannot be evaluated by the car buyer, and no savings of cost made possible by weight reduction can be justified if they involve any loss of durability and reliability. Similarly, the buyer is in no position to evaluate active safety provisions in the vehicle, and he must not be asked to accept any compromise in those qualities in addition to compromises in passive safety inherent in the lighter vehicle.

Some criteria are so fundamentally related to weight that there is no prospect of maintaining them invariable with weight reduction. These relate to impact resistance in vehicle-to-vehicle impacts, and to ride quality and noise-vibration reduction. It is true that technology is available to improve these criteria over their current state, but similar improvements are equally feasible on heavier vehicles, and the weight imbalance remains. No matter how resistant to impact a vehicle may be in collision with other vehicles, it will be better with the addition of weight in the right places, and no matter how smooth and quiet it may be, weight additions can make it smoother and quieter.

For those qualities, such as ride smoothness and noise-vibration reduction, that are not indispensible to adequate transportation, we have provided a category that we call Optional Criteria. This is in accordance with present industry practice whereby the buyer has the option of buying base vehicles for transportation purposes that lack the ultimate refinements of ride quality, smoothness and quietness, or other vehicles at extra cost with greater refinement in these characteristics. We have accordingly studied weight reduction from the standpoint of possible interchangeability between minimum weight vehicles and more expensive, heavier vehicles of greater refinement. In a later section we discuss ways and means to improve the balance between these qualities and weight.

1.1 PRODUCT PLANNING AND CUSTOMER PREFERENCES

1.1.1 Customer Preference for Conspicuous Size

The product decisions that depend on forecasting of customer preferences and judging vehicle characteristics accordingly are the most crucial of all. Customer preferences are ordinarily judged by what has sold in the past, and periods of changing conditions in which customer preferences shift from their established positions provide the most difficult conditions. An example of this is the current situation in which the fuel crisis and the economic recession combined to switch customer preferences to smaller vehicles, to which the manufacturers responded as quickly as possible with increased allocation of production resources to small cars. No sooner had these changes got well under way when a measure of economic recovery switched buyer patterns of behavior back to their traditional ways and people once more wanted big cars. A similar, but less traumatic version of the same thing accompanied the economic recession of 1958, to which the manufacturers responded with the compact cars, which sold well for their first couple of years, causing some to predict the demise of the big car, but with economic recovery the old pattern reasserted itself, and compact car sales fell off.

With the exception of these perturbations, buyer preference for large cars has been in evidence ever since the 1920's, when the last of the small, cheap cars was supplanted by the used, large car, and the pattern of growth of the so-called "low-priced three" has been a steady one from that time up until about six years ago.

It took some time for it to become apparent that the majority of buyers was more interested in a car that looked big than in one that was big. In 1949 Buick brought out the first "hardtop", which was simply a basic convertible with a metal top instead of a "rag" top. Soft tops are yielding, and can be lower than metal tops without causing difficulty with head clearance, and the general opinion was that the hardtop would not be popular because it lacked sufficient headroom. To most people's surprise, it sold like hotcakes. Shortly after that, Studebaker came out with a low-built, sporty vehicle in which rear-seat legroom and headroom were both drastically reduced for the sake of appearance, and it was very popular. It became apparent that one way to make a car look bigger was to make the passenger compartment smaller, and that people seemed not to object to these reductions in internal space.

The restriction of passenger space by lowering of roof, by inward sloping of windows ("tumblehome") and increased slope of windshield and backlight has continued to the present time. The Plymouth Valiant has been the only vehicle now in domestic production that has not been subjected to this process during the past decade.

Therefore, the most reliable criterion of buyer preference has been the preference for large-appearing cars with the almost complete indifference to interior space. The station wagon offers extra room to that minority who do require it, but even the station wagon has been subjected to the same process of shrinkage by lowering and inward-sloping glass. Today the most popular body style is the 2-door hardtop, which offers the least amount of passenger space of any commonly available.

The experience has been therefore that the public, given a choice at the same price, will invariably pick out the car that is largest, or that looks largest. Consequently the temptation has been irresistable to increase the size of each model that isn't already at the top limit of size as a means of increasing its sales. An excellent example is presented by the Torino 1971 and 1972 models. The latter weighed 304 lbs more than the former, was 1" longer and 3" wider, the latter being simply a matter of outside sheet-metal contours, on a basically carryover body structure. The 1972 car had about 5% less actual volume of passenger space by virtue of a lower roof and more extreme sloping of glass. We attach a curve that shows the sales curve of the Torino in comparison with its intermediate competitors for those years, Figure 1A shows that the Torino went to the head of the class in 1972 and remained there through 1974, despite less radical size increases on the part of its competitors the following year (Figure 1B).

The present is a time of flux in car preferences, and during such times the minority of buyers that chooses its cars on some other basis than size and glamour seems to come to the fore. This minority, sometimes called the "rational buyer" is the one who buys the imported cars and at least some of the U.S.-built compacts. It was for this buyer that the compacts were originally designed, and whose pent-up needs caused the first spurt of sales of the compacts in 1960, but when sales of these cars slacked, the manufacturers tried to sell them to the majority buyer by glamorizing them and offering high-powered

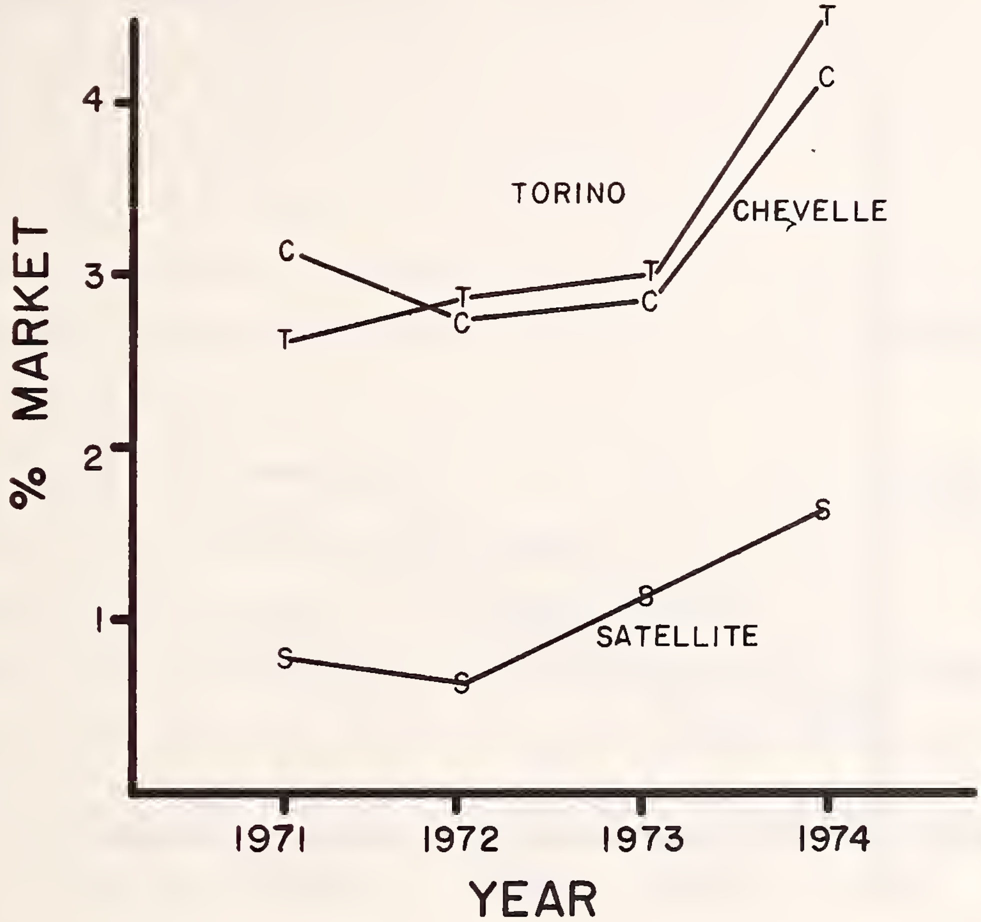


FIGURE 1A. MARKET PENETRATION INTERMEDIATES

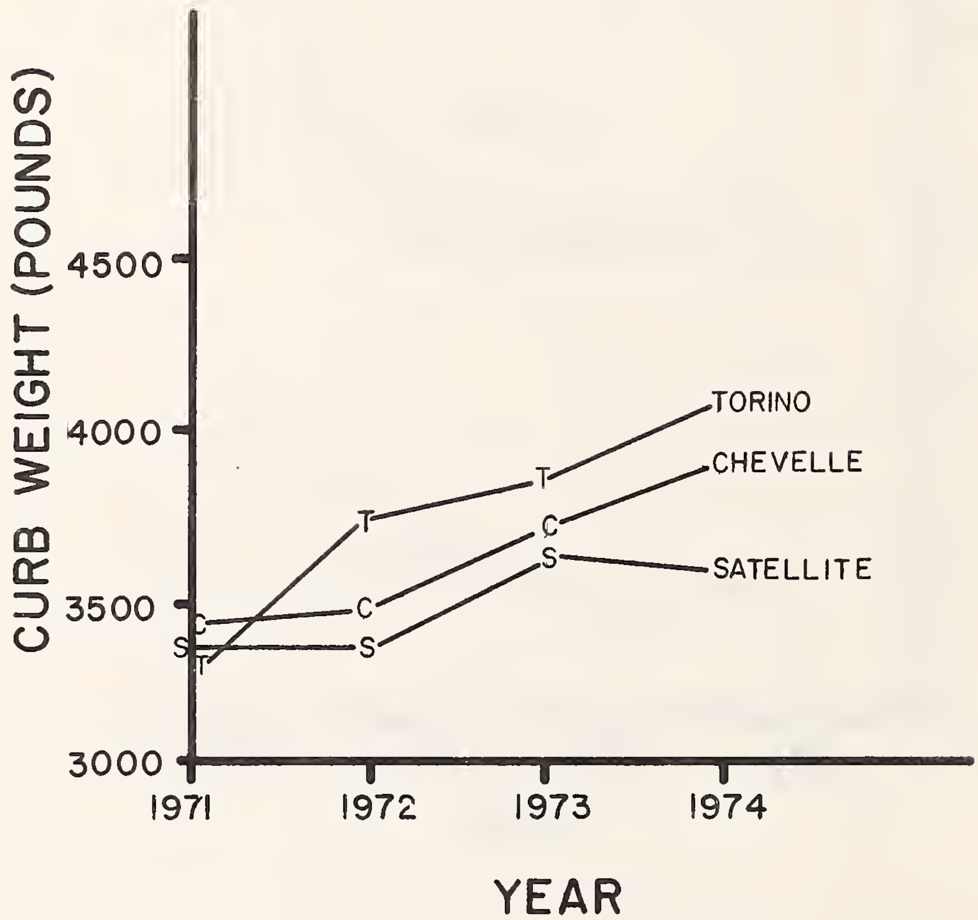


FIGURE 1B. WEIGHT TRENDS INTERMEDIATES

optional engines, with only indifferent success with the majority, and that at the expense of its other products, while the minority once more turned to the imports.

Since the minority buyer seems to emerge in times of crisis, no doubt in some degree by temporary desertions from the majority, we propose that the only way a weight study can be made is on the assumption that the buyer is a rational man who is looking for transportation value. It is obviously not a rewarding practice to base a weight-reduction program on the assumption that size and weight are constraints of primary value. It is these assumptions that have resulted in the present vehicles, which are very nearly the most cost-effective answers that could have been supplied.

So long as the majority buyer is the customer to be satisfied, no changes are called for in the current domestic vehicle. We therefore adopt the product-planning concept of the best of the imported vehicles, which is that they are designed for the rational buyer, designed and advertised in such a way as to maintain the value of cars already sold, and styled in a way that does not offend the eye but does not call for reductions in functional value or increases in cost and weight.

1.1.2 Styling Constraints

Styling is constrained by the need to make the car look as large as possible. This is accomplished, as we have remarked, by decreasing the size of the greenhouse, and by bringing the corners of the car, front and rear, out to the extremes of length and width, and generally accentuating them with brightwork.

Glamour of appearance is primarily a matter of extra decoration, brightwork, and a multiplicity of textures, surfaces and colors. Some novelty of configuration, such as the recently-popular "opera" windows helps to attract attention. Overstyling is less risky than understyling, so overstyling is the general rule.

1.1.3 Options

Table 4A lists a typical line of options, and their weight. The provision of such options is an important marketing necessity in the domestic market, and provides a competitive advantage over imports, whose flexibility to install options is much restricted. As a result of the latter, the imported vehicles have either to install a given option as standard equipment, provide something for dealer-installation, or go without.

Not all the extra weight of the option list need be a penalty on the base car, since springs, wheels, and tires, and even rear axles, can be variable, but ordinarily brakes, structure and suspension parts are constant across car models. For this reason, any penalty in those parts for the purpose of carrying the weight of optional equipment is a penalty also on those base cars which do not carry the equipment. These weights are analyzed herein in detail (Chapter 3). The seller has the option of spreading the extra cost of this weight (very small, as we shall show) across all models, or of charging to the buyer of the option-equipped car, as a matter of pricing policy. In most cases it is charged to the buyer of the option-equipped car in order to place a more attractive price on the base car, which few people buy without some extra equipment.

The heaviest single optional item on the list of many domestic builders is the optional powerplant. It is common to build compact-size vehicles to carry the largest lines of company engines, since in most cases the company "muscle-car", until recently an important market item, has been designed around interchangeability with the compact-sized car. This has involved the widening and beefing-up of front ends at considerable weight penalty, although the exact amount is not segregated anywhere for precise identification.

Considering the variable needs and desires of various buyers, the provision of at least two optional powerplants, one small, emphasizing economy, and the other enough larger to provide improved acceleration and highway performance, is a useful provision, and we will follow that approach.

Car design is constrained by the use of optional engines of widely varying dimensions to the front-engine rear-drive design, since front-drive cars do not have the flexibility of layout that permits the use of oversize engines and transmissions in a basically small vehicle.

Another constraint on car weight is the ability to pull a trailer. This is no job for a light car. We have made no attempt to maintain equal trailer-hauling capability, even as an option, which would entail a penalty in cost and weight simply to make provision for the needed increase in engine size. (See Chapter 3 for detailed listing of weight increments resulting from provision for increased power options.)

The lighter the car for its roominess index, the more it is penalized in performance, weight and fuel economy by the installation of air conditioning, since air conditioning demand is a function of roominess, while the power available to run it decreases with decreased weight and engine size. We observe here that no program to reduce the weight of the common optional items has been undertaken along with the campaign to reduce the weight of the base car, so the options in general represent a greater weight penalty on the weight-conscious car than on the standard article.

1.2 COST

1.2.1 Weight and Cost

It turns out that the incremental manufacturing cost of incremental weight in the automobile is relatively low, which is of course the answer to why weight and size increases are so commonly resorted to increase sales. It costs little or no more in investment to tool up to produce larger pieces than small ones, and no other fixed charges bear any relation to the weight of the product. Of variable costs, only materials and transportation are incremental in the heavier car, labor costs are substantially invariable with car weight.

As a result, it takes a relatively small increase in sales to pay for a proposed increase in product weight, so that increased variable costs are more than paid back by increased total profits.

So long as fuel remained very cheap just about the same thing was true of the cost of owning a heavy car as opposed to a light one. Insurance costs follow car cost rather than car weight, and fuel and tires are the principal incremental costs of the heavier car. It remains to be demonstrated that higher fuel costs will persuade the U.S. car buyer to abandon his preference for large cars. Besides the satisfaction of his needs for status symbols -- or however you define buyer desire for conspicuous size -- the buyer receives real value for the large size besides simply more passenger space. The larger and heavier car is quieter and smoother and he has a firm conviction, not without justification, that it is safer.

1.2.2. Fixed Costs and Design Changes

Fixed costs are a large portion of the total cost of building a car, since the high cost of labor enforces a need for extremely high productivity per man-hour, and this requires large investments in tooling and automation. As a consequence there is heavy financial pressure to avoid design changes because of the large investments needed for new tooling.

Constraints on new design changes relate not only to tooling investments. New designs require deployment of engineering talent that is in short supply because of the enormous amount of engineering time required to meet government-mandated requirements on a very short schedule. Likewise the toolmaking facilities of the country are limited, and only so much design change can be undertaken at any one time. These factors totally eliminate any possibility of wholesale change in the industry in a short period.

There are other reasons for resistance to design change. Each new design must be made to a higher factor of safety than the one it replaces, simply because it has not been subjected to the tests of time in the hands of the user, and hence it will ipso facto be more costly to make. It takes time for production to learn the cheapest way to make something, during which time its production costs will be higher than they will ultimately be. Finally, it will give trouble despite all the attempts to design reliability into it. No system of testing devised can discover ahead of time all the difficulties that the public can find in using hundreds of thousands of examples in everyday use.

Engineering costs are a part of fixed costs. There is no doubt that better products could be made if more time were spent engineering and testing prototypes before production. However long engineering programs require the making of decisions at an earlier time when less can be foreseen of competitive actions and other governing considerations. Good engineering planning could help to solve some of these problems, but planning is a very difficult thing to do, and it's even harder to get people to follow plans once they are made.

1.3 PACKAGE DIMENSIONS

Functional size must be accurately specified if weight estimates are to mean anything. We find that the prevailing measure of functional size is a space factor defined by Austin (2-1) and given for 52 different automobiles of current production. This rating is defined as "the sum of the six principal interior dimensions in inches plus twice the usable trunk volume in cu.ft.", and is plotted against curb weight. We reproduce these figures in Table 1A and in Figure 1C. In Figure 1C we have added lines of constant weight per unit space factor.

1.3.1 A Consistent Roominess Index

We find that the prevailing roominess index has decided shortcomings when it comes to evaluating the functional roominess of a car:

1. It is dimensionally inconsistent, consisting of the sum of a number of linear dimensions in inches plus a volume dimension in cubic feet. As a result of this the space factor for a given vehicle does not reflect the roominess per passenger between vehicles of different sizes, nor are cars of differing weights comparable in weight per unit space factor, as may be seen from Figure 1C.

2. The dimensions of headroom and legroom on which the space factor is based are highly variable, depending on the complicated conventions of measurement, which change from year to year and are different in one country from another, so that imported cars and domestic cars cannot be directly compared on the basis of published figures, nor is it possible to compare cars built in one year with those built in a different year. For example, in Table 1B we list the headroom and legroom for the 1960 Falcon as measured in the convention of 1960 and as measured in 1975, with a resulting difference of 10.3 in the computed space factor for the same vehicle.

3. It is not feasible to make direct measurements of space factor from the vehicle itself without considerable equipment and elaborate procedure (see SAE handbook specifications for seating measurements, 2-2) and unless the interior trim of the vehicle is in specification condition. This eliminates the practical possibility of measuring old cars for the purpose of making meaningful weight comparisons between cars of past design and those of current design.

TABLE 1A. INTERIOR SPACE RATING TABLE.

The interior space rating for each car consists of the sum of the six principal interior dimensions in inches plus twice the usable trunk volume in cubic feet.

No.	Automobile	Curb* Wt.	$X10^{-2}$ HP/Curb	Space Rat.	No.	Automobile	Curb Wt.	$X10^{-2}$ HP/Curb	Space Rat.
1	Chevelle	3758	2.79	298	27	VW Dasher	2108	3.37	283
2	Chevrolet	4329	3.34	316	28	VW Rabbit	1900	3.68	279
3	B.Electra	4745	4.32	328	29	Fiat 128	1900	3.47	272
4	Cad.DeVille	5292	3.59	315	30	Lada 1200	2130	2.91	289
5	Torino	4186	3.68	297	31	Fiat 132	2509*	4.26	294
6	Ford	4459	3.32	321	32	Colt 2000	2326*	4.39	276
7	Thunderbird	5105	3.80	300	33	Escort 1100	1804*	2.66	279
8	Lincoln	5242	3.93	319	34	Vaux.Viva	1987*	3.42	270
9	Monaco	4365	3.44	316	35	Ford Cortina	2456*	3.99	296
10	Chrysler	4450	4 04	319	36	Datsun 120Y	1857*	2.83	256
11	Coronet	3700	2.57	299	37	Mazda 1300	1906*	3.62	272
12	Matador	3656	2.60	298	38	Fiat 124	2144*	3.50	290
13	Vega	2495	3.13	268	39	Daf66 SL1300	1975*	2.89	269
14	Nova	3386	3.10	290	40	Alfasud	1996*	3.16	285
15	Pinto	2581	3.22	261	41	Renault 12	2149*	2.64	278
16	Mustang II	2759	3.15	260	42	Alfasud ti	1970*	3.50	285
17	Maverick	2939	2.45	278	43	Golf 1100	1909*	2.62	278
18	Granada	3243	2.22	289	44	Austin Mini	1544*	3.50	239
19	Valiant	3100	3.06	291	45	Renault 6TL	1736*	2.87	261
20	Gremlin	2739	3.65	263	46	Datsun 100A	1545*	3.82	255
21	Hornet	2827	3.54	279	47	Fiat 127	1650*	2.85	262
22	Saab 99	2500	4.40	284	48	Peugeot 104	1767*	2.83	260
23	Audi 100	2569	3.54	284	49	Renault 5TL	1627*	2.64	263
24	Volvo 144	2738	3.65	292	50	Honda Civic	1590*	3.77	254
25	Toyota MKII	2820	4.33	274	51	Austin Alleg	2004*	4.49	275
26	Peugeot 504	2860	2.87	288	52	Fiat 126	1371*	1.68	234

*100 pounds have been added to the curb weight of foreign cars which are not imported to reflect the effect of meeting U.S. Automotive Standards.

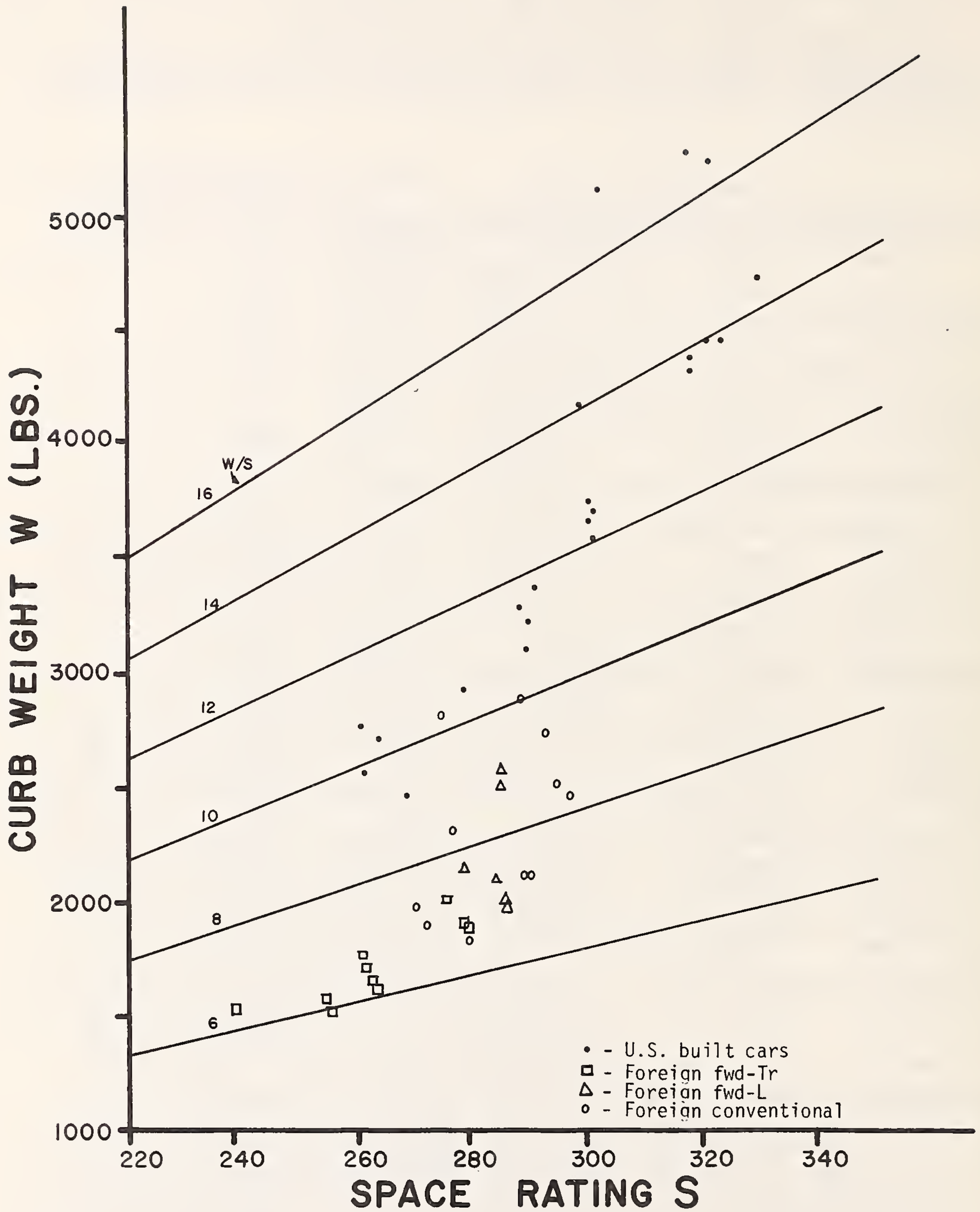


FIGURE 1C. CURB WEIGHT VS. SPACE RATING

TABLE 1B. HEADROOM AND LEGROOM FOR THE 1960 FALCON, MEASURED IN THE CONVENTIONS OF 1960 AND 1975.

	1960		1975	
	F	R	F	R
Headroom	38.6	37.3	38.2	37.0
Legroom	44.8	40.7	41.7	37.0
Shoulder room	55.6	55.4	55.6	55 4
Trunk volume	15 cu.ft.		15	
Space factor	305.2		294.9	

4. A measurement based on seating dimensions alone does not adequately evaluate the basic design of the vehicle itself, but is variable with the type and character of trim installed. It also fails to evaluate design characteristics that affect entrance and egress space. For example, excessive tumblehome restricts entrance and exit space (see Figure 1D) especially when room to open the door is restricted, as in close parallel parking, and restricts passenger and driver headroom lateral movement. Excessive slope and rearward position of A posts restricts driver entrance and exit.

We therefore propose a roominess measurement that will:

1. Permit comparisons of roominess per passenger to be made between vehicles of different size classes, and therefore to permit weight comparisons to be related to size and weight classes for the evaluation of design characteristics.

2. Produce a measurement expressed in units of volume so that passenger space and weight per unit of passenger space can have real meaning, and which can also be compared directly with existing measures of trunk volume.

3. Reflect those design characteristics that affect passenger space in ways other than headroom, shoulder room and legroom.

4. Be based on a procedure that can be conducted without special equipment

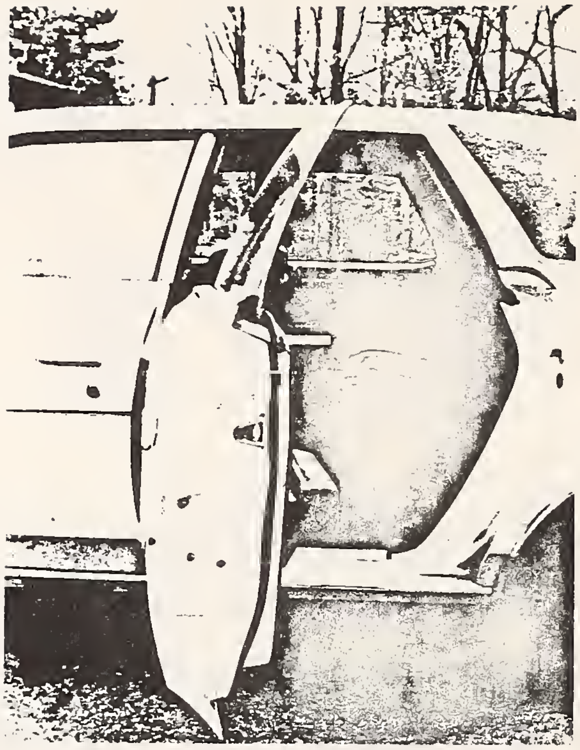


FIGURE 1D. EXCESSIVE TUMBLEHOME

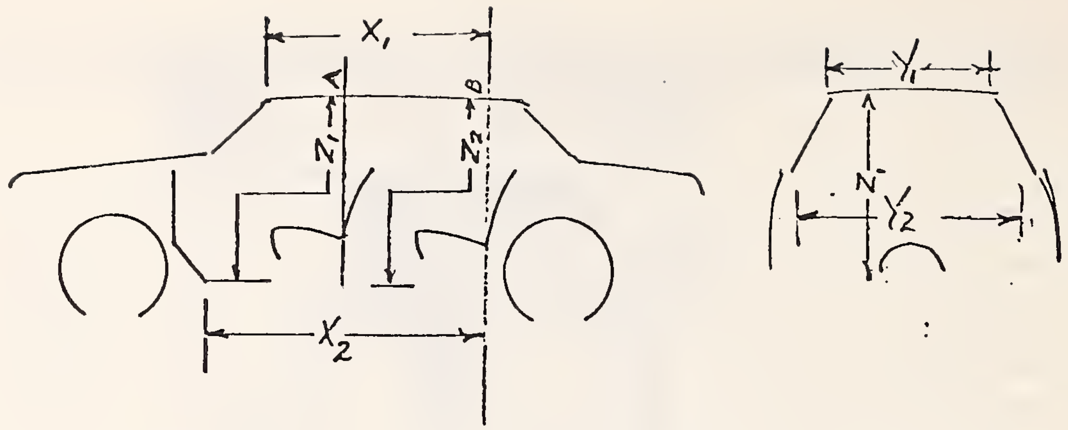
and which can be performed by unskilled personnel on an available vehicle of any age.

5. Produce a measurement that is independent of the character and condition of a vehicle's interior trim.

We therefore propose the protocol shown in Figure 1E as a means of producing a roominess index that can meet the foregoing objections and conditions. In Table 1C we list measurements taken of 40 vehicles of assorted sizes and ages on a experimental basis, using an earlier protocol shown in Figure 1F which did not include the dimension from the intersection of the roof rail and the A pillar. The values under our proposed system will be smaller than those which we display, but they will be similarly comparable. We have not, however, identified the vehicles measured other than by number and date of manufacture.

TABLE 1C. MEASUREMENTS OF 40 VEHICLES OF ASSORTED SIZES AND AGES

#	YEAR	LENGTH	WIDTH	HT.	X1	X2	Z1	Z2	Y1A	Y2A	Y1B	Y2B	R.I.	WT.	#
1	1970	170.5	66	56.5	74.75	68	46	47.25	41	55	42.5	51.5	97.0144	2360	1
2	1973	188	67	57.1	77.5	69.25	45.5	47	44.75	55.75	45	51	100.649	2767	2
3	1976	175.4	65.4	51.8	77	71	40.5	40	40	42.5	40.5	49.25	76.6494	2495	3
4	1967	219.6	80	56.4	83	73.5	45	47.5	50.5	61.25	51	61.5	122.978	4070	4
5	1973	194.3	72.4	52.5	76	69.75	44	43.75	46	56.75	47.25	55.25	99.2824	3137	5
6	1972	183	68	56.1	75.5	70.75	46.75	46.5	45.5	55.5	45.75	55.75	104.819	2440	6
7	1976	220.4	77.7	55.5	74.75	72.75	44.25	46.75	43.75	60	44	59.5	108.322	3845	7
8	1974	232.4	79.8	54.2	84	78.5	43	44.75	48	64.25	50.5	64.25	124.811	4830	8
9	1974	157.2	63.9	55.9	75.25	70.5	45	43.5	41.5	51.5	41	51.75	91.5	1980	9
10	1966	209.8	78.7	55.3	80	75	45.75	46.25	50.5	61	50.75	61.25	120.69	3840	10
11	1974	160.6	60.8	53.5	68.5	64	43	41	40	50	41.75	49.75	76.6878	1915	11
12	1974	223.6	79.4	58.5	82	77	43.5	47.5	47	61.75	48	61.5	121.609	4985	12
13	1976	171.5	77	53.6	72.5	70	44	45.5	40.5	61.5	40.5	46.25	93.2383	2995	13
14	1976	197.7	74	53.5	75.25	70	43	42.5	42.75	56.5	44	54.75	94.4397	3308	14
15	1975	169	69.4	50.5	77	69.5	42.5	40.25	40	53.25	40	51	86.1049	2613	15
16	1976	223.9	79.5	54.3	80.25	74.75	43	44.75	47.5	62	48.75	61	114.438	4502	16
17	1976	217.6	79.3	53.3	79.5	74	42.5	43	45.5	59.25	47	59	106.153	4258	17
18	1963	197.6	71.3	55.4	79.5	74.5	44	45.5	49.5	58	49.5	58	111.419	2970	18
19	1967	184.3	73.2	54.6	77	75	45	44.5	48.5	58.75	49.5	58	110.279	2677	19
20	1976	196.1	71.7	53.2	73	68	40.75	41.75	43	56.25	44.5	56.25	89.4063	3050	20
21	1976	199	71	54.1	76	73.25	44.75	42.75	47	56.25	48	56.5	102.322	3265	21
22	1976	179	66	54	71.75	70.5	41.5	41.5	42.5	51.75	44	51	84.2866	2745	22
23	1976	182.4	66.7	57	74.5	74.5	45.5	44	45.25	56.25	45.25	55.25	102.495	3000	23
24	1976	164	59.7	53.1	68.5	67.5	42	41.5	42.25	48	42.5	48.5	76.8588	2060	24
25	1960	204	71	54.7	76.5	73	46	44.75	48.25	57.5	48.5	55.5	106.913	3180	25
26	1972	165	65	52.6	67	63.5	44.5	43.5	39.25	50	42	48.5	78.2443	2125	26
27	1973	228.5	79.8	54.1	79	77	42.5	45	48	63.75	50	62.5	117.687	4943	27
28	1976	158.7	61.8	52.3	69	66.5	44.75	44	40.5	49.75	43	53	85.1967	2200	28
29	1968	214.7	79.6	55.8	79	75	43.5	42.5	52	63	54.5	62.5	115.684	3770	29
30	1974	174	64.5	56.6	76.25	69	43.5	44.25	45	51	44.75	50.75	91.0472	2083	30
31	1976	195.5	69.7	56.7	74.5	70.75	45	44.75	46.25	58	47	57.5	103.672	3205	31
32	1975	150	59.3	52.2	73	66.5	42.75	42.75	39.5	49	38.25	49	80.1858	1748	32
33	1976	172.8	63	53.5	76.25	71	42.5	45	40.5	54	40	53.5	93.902	2068	33
34	1976	155.3	63.4	55.5	71.75	67.75	45.25	45.75	40.5	52	43	50.5	89.7633	1830	34
35	1975	195	68.9	57.1	75.5	72	47.5	45.25	45	56.5	45.25	55.5	105.457	4400	35
36	1976	176	62.6	55.5	69.5	66	43.75	44	41	51.5	41	50	83.1079	3300	36
37	1976	188	69.1	56	76.5	73.5	44.75	44	45.25	55.5	45.75	55.5	102.078	2571	37
38	1967	223.9	80	56.2	82.5	76.5	42.5	42.75	53	62.5	53	61.5	117.172	4404	38
39	1964	181.6	71.6	54.5	78.5	74	44.75	45	51	56.75	51	56.5	109.342	2503	39
40	1962	183	70.8	55	81.5	77.75	42	41.5	48.25	54	48.75	55	101.962	2570	40
41	1960	180	67	51.5	74	69.5	43	40	46.5	52	48	52.5	87.8813	2495	41
42	1962	190	72.4	57.5	79.25	76	42	42	47	57	47	58.25	103.711	2993	42



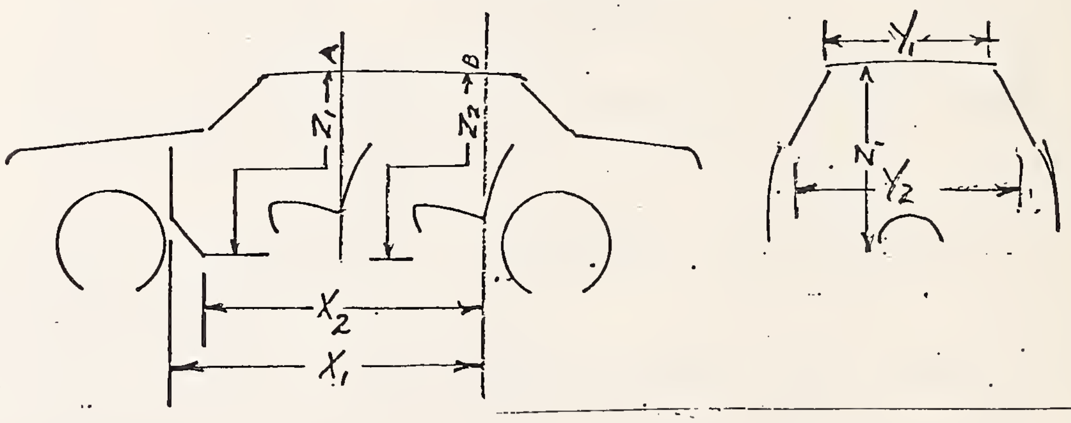
Dimensions measured in inches

Vertical transverse plane A passes through the farthest forward point of the front seatback
Vertical transverse plane B passes through the farthest forward point of the rear seatback
X1 is the longitudinal distance from the point of intersection of the roof rail and the A pillar to the B plane
X2 is the longitudinal distance from the front-seat heel point to the plane B
Y1A and Y1B are the interior widths at the roof rail in planes A and B
Y2A and Y2B are the interior widths at the seat-cushion level in planes A and B
Z1 is the vertical dimension from the front compartment floor to the intersection of the A plane and the headlining, measured at 1/3 width
Z2 is the vertical dimension from the rear compartment floor to the intersection of the B plane and the headlining, measured at 1/3 width

$$RI = V = \frac{X1+3X2}{6912} \cdot \frac{Y1A+Y1B+3Y2A+3Y2B}{13824} \cdot \frac{Z1+Z2}{3456} \text{ or } \frac{(2X1+6X2)\cdot(Y1A+Y1B+3Y2A+3Y2B)\cdot(4Z1+4Z2)}{13824}$$

RI = V = Roominess Index, cubic feet

FIGURE 1E. ROOMINESS INDEX (proposed version)



Dimensions measured in inches

Vertical transverse plane A passes through the farthest forward point of the front seatback
Vertical transverse plane B passes through the farthest forward point of the rear seatback
X1 is the longitudinal distance from the bulkhead to plane B
X2 is the longitudinal distance from the front-seat heel point to plane B
Y1A and Y1B are the interior widths at the roof rail in planes A and B
Y2A and Y2B are the interior widths at the seat-cushion level in planes A and B
Z1 is the vertical dimension from the front compartment floor to the intersection of the A plane and the headlining, measured at 1/3 width
Z2 is the vertical dimension from the rear compartment floor to the intersection of the B plane and the headlining, measured at 1/3 width

$$RI = V = \frac{X1+X2}{3456} \cdot \frac{Y1A+Y1B+3Y2A+3Y2B}{13824} \cdot \frac{Z1+Z2}{3456} \text{ or } \frac{(4X1+4X2)\cdot(Y1A+Y1B+3Y2A+3Y2B)\cdot(4Z1+4Z2)}{13824}$$

RI = V = Roominess Index, cubic feet

FIGURE 1F. ROOMINESS INDEX

In Figure 1G we have plotted RI against curb weight, along with lines of constant weight per cubic foot of roominess. It can be seen from this plot that it is possible to draw conclusions about vehicle design in its relation to weight and roominess that could not be drawn from the similar plotting of the Space Factor in Figure 1C. We believe that the greater weight per cubic foot of the larger cars is a real indication of design difference and not just a dimensional effect, since the larger vehicles have more power, give away more seating space for styling, and give away weight for improvements in noise and vibration control. We also believe that the smaller density of the front-drive cars is an indication of the superiority of the front-drive design from the package standpoint, particularly in their lower floor level as compared with rear-drive designs.

We display in Table 1D the average interior height dimensions of a number of representative imported front-drive designs as compared with those of comparable imported rear-drive and domestic rear-drive designs to illustrate this advantage.

TABLE 1D. FLOOR-TO-HEADLINING INTERIOR HEIGHT

	FLOOR-TO-HEADLINING INTERIOR HEIGHT
Front-drive imports	44.48
Rear-drive imports	44.10
Rear-drive domestic	43.41

1.3.2 Size Classes for Weight-Study Purposes

Believing, as we do, that size is the basic criterion to be conserved in weight studies, we have defined three vehicle packages of accurately specified dimensions as the basis for further weight-study comparisons, and we have evaulated the space factors and roominess indices of these packages for

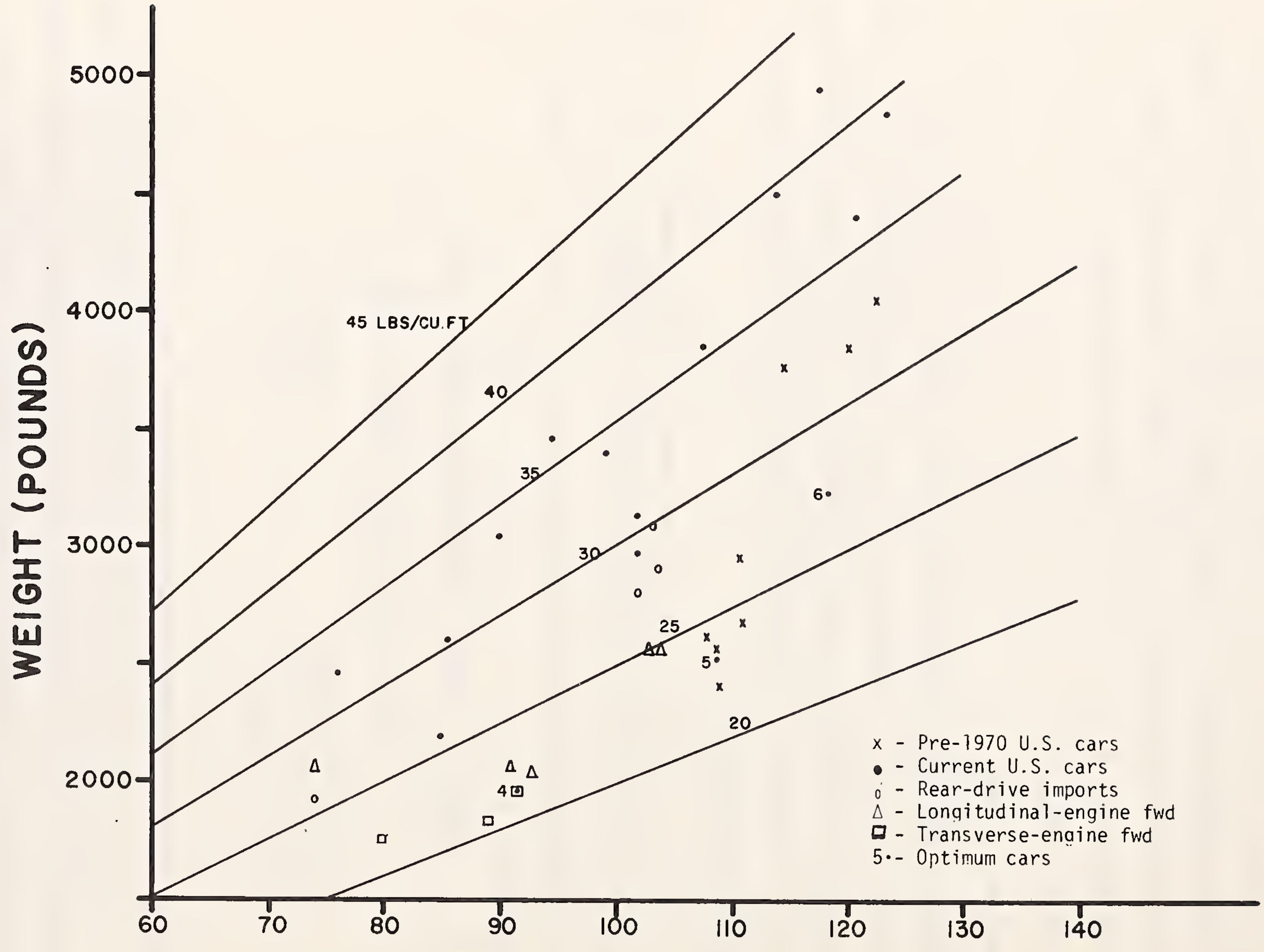


FIGURE 1G. ROOMINESS INDEX (CUBIC FEET) VS. CURB WEIGHT (POUNDS) comparisons with existing vehicles, Tables 1E, 1F, and 1G, Figures 1H, 1I and 1J.

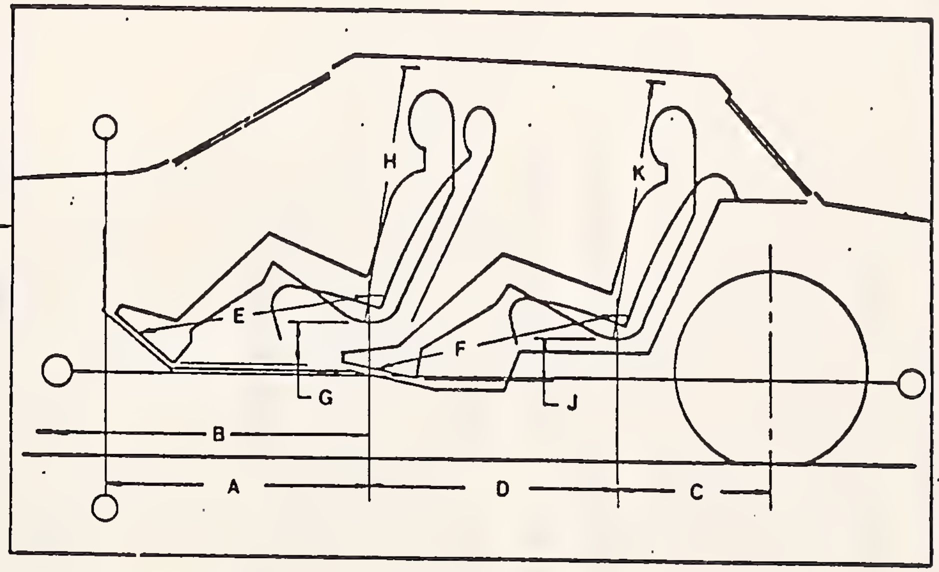
The 4-passenger class is represented by the Rabbit, Pinto, Vega, Chevette and Fiat 128. We have taken the actual dimensions of the Fiat 128 as the 4-passenger class prototype. The 5-passenger class, which may more properly be regarded as a tight, or occasional 6-passenger, is represented by the familiar compact-sized domestic cars such as the Nova, Hornet, and Valiant and their various derivatives, and amongst the imports, by the Mercedes Benz 230 and 240, the Audi 100, Saab and Peugeot 504. The 6-passenger class is represented only by the larger domestic vehicles, the Ford LTD, Chevrolet Impala and Plymouth Fury and their various derivatives. We have taken the dimensions of the 1960 Falcon as the 5-passenger prototype, and those of a 6-passenger vehicle of generally similar characteristics as the 6-passenger prototype.

1.4 ECONOMY OF OWNERSHIP

1.4.1 Fuel Economy and Weight

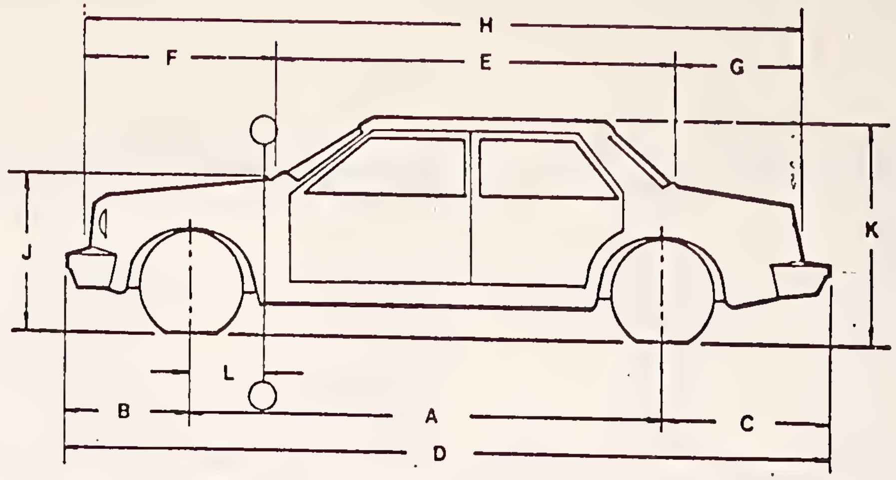
If other things remain unchanged, a change in weight produces relatively little change in overall fuel economy. Marx et al. (1-8) report that a 10% reduction in curb weight will give a 4% reduction in fuel use. However, if a comparable change is made in engine displacement, keeping cubic inches of displacement per pound constant, then a 10% change in weight will bring about a fuel use reduction of 8% (see 4.2.1 for a further discussion of engine displacement per pound per mile). Keeping displacement per pound per mile constant keeps acceleration capability constant with weight change. The change may be accomplished either by changing engine displacement or by changing engine revolutions per mile. Reference 2-5 gives an extended discussion of ways and means for improving fuel economy.

TABLE 1E. INTERIOR DIMENSIONS, OPTIMUM VEHICLE PACKAGE



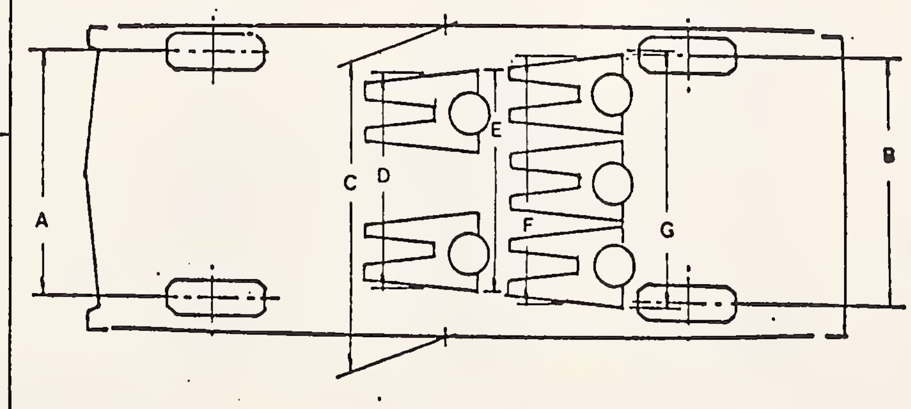
		4 PASSENGER	5 PASSENGER	6 PASSENGER
A	"O" LINE TO FRONT "H" POINT	44.40	41.00	40.66
B	C/L FRONT WHEEL TO FRONT "H" POINT	50.40	55.30	56.26
C	C/L REAR WHEEL TO REAR "H" POINT	17.40	21.20	21.56
D	COUPLE DISTANCE	30.10	33.00	37.18
E	EFFECTIVE LEG ROOM-FRONT	40.20	41.70	42.10
F	EFFECTIVE LEG ROOM-REAR	34.20	37.00	39.40
G	"D" TO HEEL-FRONT	5.30	5.70	6.23
H	EFFECTIVE HEAD ROOM-FRONT	37.60	38.20	39.00
J	"D" TO HEEL-REAR	7.50	7.80	7.50
K	EFFECTIVE HEAD ROOM-REAR	36.80	37.00	37.69
L	TRUNK VOLUME CU. FT.	14.0	15.0	19.3
M	SPACE FACTOR RATING	272	295	317

TABLE 1F. EXTERIOR DIMENSIONS, OPTIMUM VEHICLE PACKAGE



		4 PASSENGER	5 PASSENGER	6 PASSENGER
A	WHEELBASE	97.00	109.50	115.00
B	FRONT OVERHANG	28.00	29.90	34.15
C	REAR OVERHANG	34.50	43.10	47.35
D	OVERALL LENGTH	159.50	182.50	196.50
E	GREENHOUSE LENGTH	91.00	92.90	95.30
F	FRONT END LENGTH	39.00	52.30	56.70
G	DECK LENGTH	23.50	30.30	36.50
H	OVERALL LENGTH LESS BUMPERS	153.50	175.50	188.50
J	COWL HEIGHT	34.70	37.60	40.75
K	VEHICLE HEIGHT	53.00	54.60	57.65
L	"O" TO C/L OF FRONT WHEELS (REF.)	6.00	14.30	15.60
	TIRE SIZE	145 SR 13	165 SR 14	175 SR 14

TABLE 1G. WIDTH DIMENSIONS, OPTIMUM VEHICLE PACKAGE



		4 PASSENGER	5 PASSENGER	6 PASSENGER
A	TREAD-FRONT	52.10	57.00	62.00
B	TREAD-REAR	52.10	56.50	62.00
C	BODY WIDTH AT "H" POINT	60.80	67.00	71.00
D	HIP ROOM-FRONT	53.50	57.00	61.00
E	SHOULDER ROOM-FRONT	49.50	55.60	60.00
F	HIP ROOM-REAR	53.20	56.80	60.80
G	SHOULDER ROOM-REAR	49.50	55.40	59.80
H	TUMBLEHOME (SIDE GLASS ANGLE)	13.80	14.00	14.00

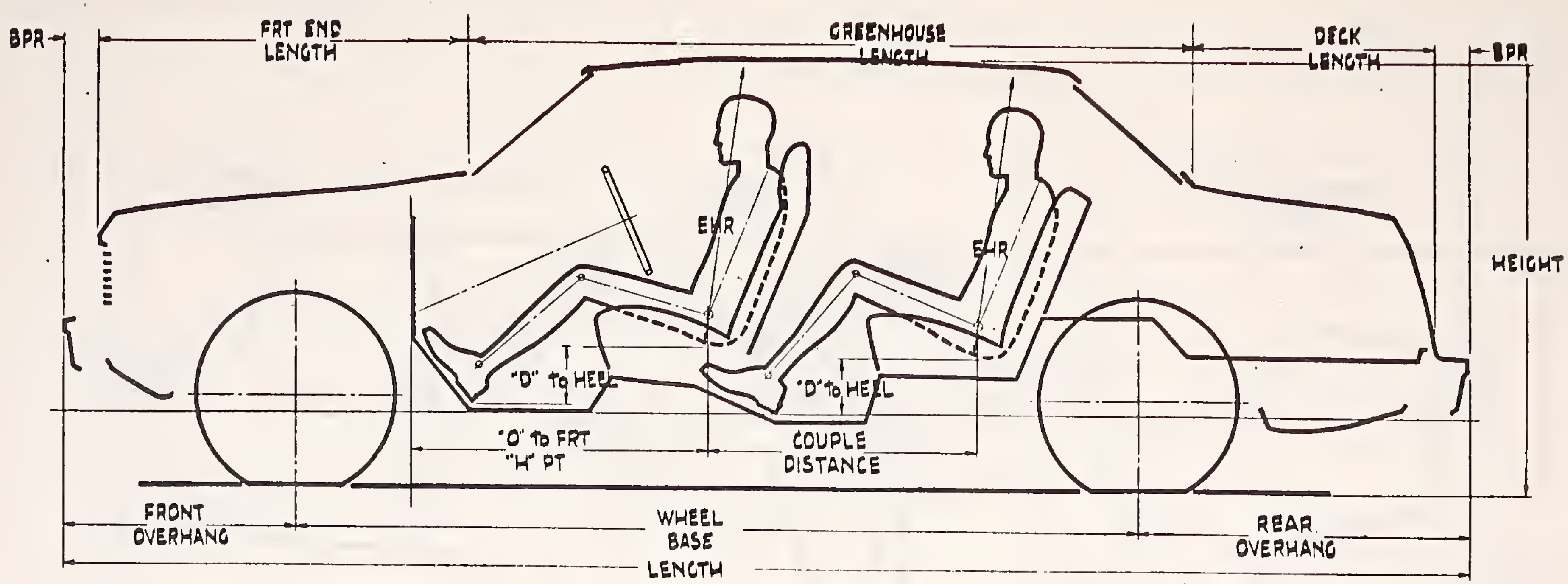


FIGURE 1H. OPTIMUM SIX PASSENGER PACKAGE

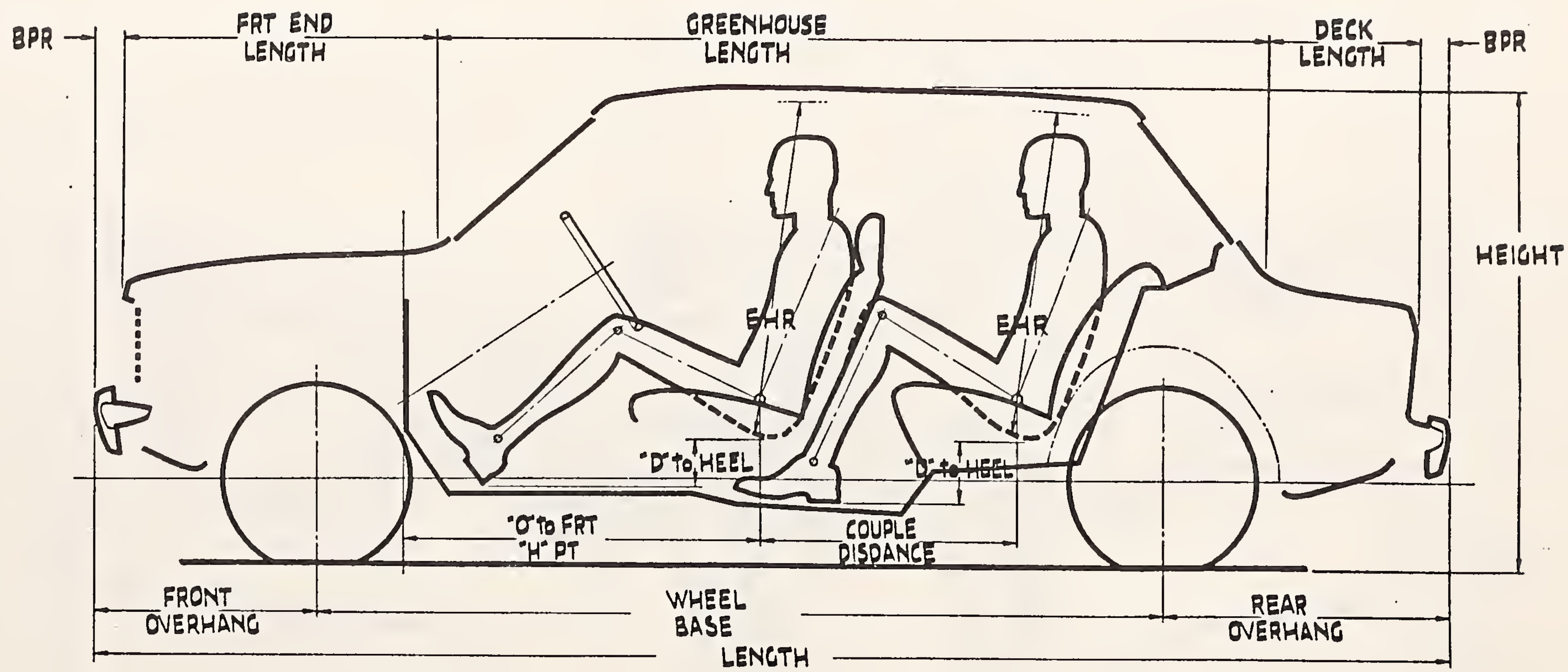


FIGURE 11. OPTIMUM FOUR PASSENGER PACKAGE

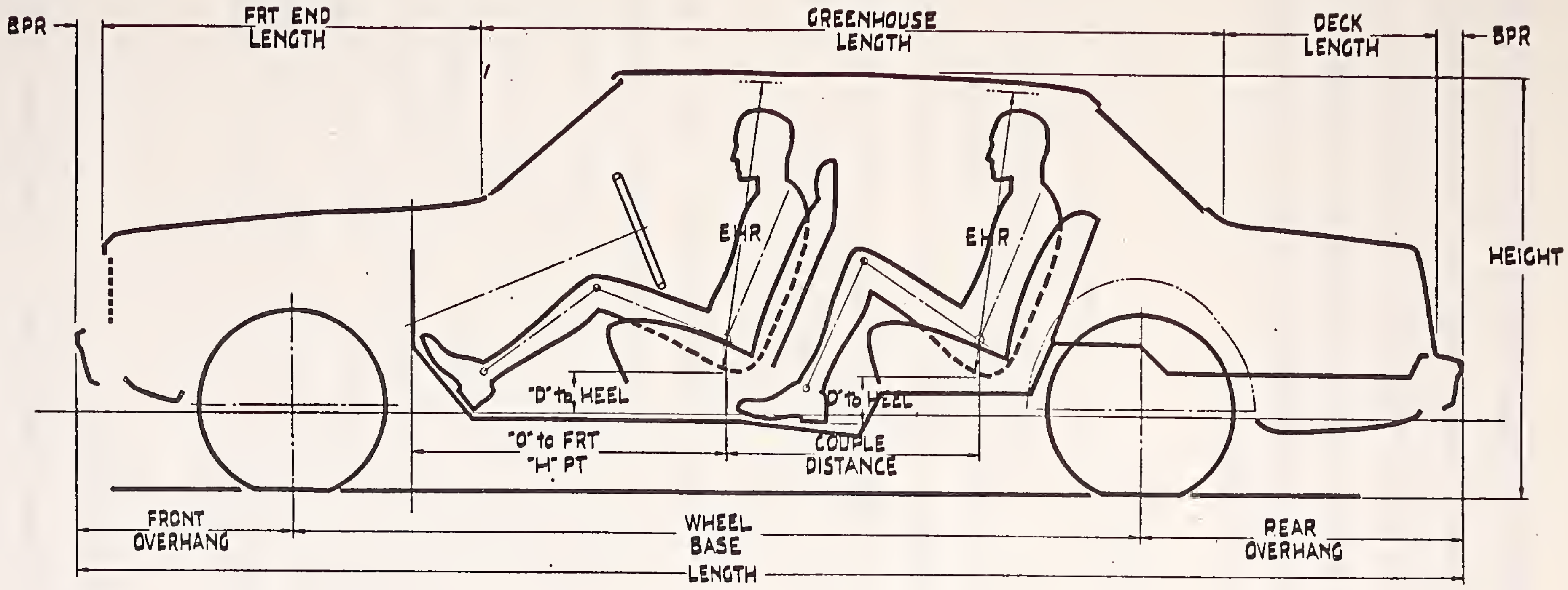


FIGURE 1J. OPTIMUM FIVE PASSENGER PACKAGE

When weight is reduced piecemeal without vehicle redesign, the weight saving is limited to the actual piece reduction. On the other hand, starting from a clean sheet of paper with a new design, it is possible to save 1 pound of weight in the vehicle body or structure and save an additional approximate .4 lbs. in weight-dependent and power-dependent weight as discussed in greater detail in Section 3 of this report.

1.4.2 Durability and Reliability

The Durability and Reliability of a new design is not known with accuracy until it has been in production and been proven in the hands of the user. As a consequence each change of design must incorporate a greater safety factor than the design it replaces, and the only way to accomplish this is to design in more weight and cost than will probably be needed eventually. As we have stated, we regard durability and reliability as characteristics that cannot be traded for weight saving, and we have entertained no proposals for weight saving that threaten to reduce either quantity. At various times in the past, and in certain present designs, durability has been sacrificed to weight saving by excessive gauge reduction of critical parts, and by reducing engine displacement and increasing engine revolutions to compensate for it. Insufficient tire dimensions, overstressed material in springs and critically-stressed chassis and driveline parts, undersized brakes, too-light flywheels (which result in shortening of clutch life in manual transmissions) and the use of dissimilar metals in contact, without proper electrolytic protection, are all techniques for weight saving at the cost of durability and reliability.

1.4.3 Cost of Maintenance and Repair

These factors follow very closely those reviewed above under Durability and Reliability. Another factor of importance in repairability is accessibility.

Certain classes of vehicles, notably those with large V-8 engines and many optional auxiliaries, as well as many designs of front-wheel drive, have the engine compartments so crowded that accessibility for adjustment, replacement and repair is difficult and expensive. We regard repairability as an unsolved problem in front-wheel drive designs that promise relatively small weight savings over comparable rear-wheel drive designs.

1.5 ACTIVE SAFETY

1.5.1 Driver Vision

Driver vision is not to be compromised in the interest of saving weight. Since metal surfaces are lighter than glass, some weight saving could result from decreasing the size of windows, windshield and backlight. Most domestic cars restrict visibility to some degree in the quarter area for the sake of styling. This is critical in some recent 2-door vehicles, which are substantially blind aft of the B-pillar except for a tiny "opera window" and most 4-door sedans are blind aft of the C-pillar to the backlight, some of them having an additional blind area due to restricted width of the backlight.

Good vision requires a side light aft of the C-pillar and A pillars to be narrow, which means less in width than the average interocular spacing. This is also antagonistic to weight reduction, which would be greater if the pillars were wider, since they could have equal strength for less weight.

Driver vision also demands a relatively high seating position relative to the beltline, a dimension in which the larger domestic cars are seriously lacking as a result of roof-lowering for styling reasons. This again is incompatible with minimum weights, as it demands larger glass areas.

Good driver vision is compatible with weight saving in that it limits the slope of the windshield. Excessive windshield slopes increase reflections and increase the probability of optical distortion as well as being heavier. Image-doubling of nighttime reflections is a function of glass angle. Windshield headers that are too far aft increase the sun and glare load on the driver. Similar considerations apply also to the backlight, and the backlight must extend high enough not to cut off the top of the scene reflected in the rear-vision mirror.

Excessively sloping windshields also increase the degree of reflection in the windshield of the shelf over the instrument panel, which in any case should be covered with some kind of non-reflective surface. This type of reflection is especially serious in driving against a dark background with the bright sky overhead. There should also be an absence of bright work in the driver's field of vision, such as at one time was common with radiator ornaments, bright windshield wiper arms, etc. (Ref. SAE Handbook, Test of Visibilty).

1.5.2 Steering and Controllability

The radial tire has done much to reduce the effects of car design on cornering ability and response to steering. It has relatively small response to side-thrust forces, and this response is much less sensitive to vertical deflection and camber than that of the bias-ply tire. As a result the dynamic steering characteristics of cars are much less affected by such factors as weight distribution, weight transfer in roll, and camber change, but they remain sensitive to steering geometry, toe-in or toe-out with roll, rear-end roll understeer or oversteer, and side-thrust steering effects.

1.5.2.1 Dynamic Stability

Steering design is a compromise between responsiveness and stability. Domestic vehicles lean in the direction of stability and are not highly responsive to steering. This condition is augmented by the emphasis on ease of control, with low-geared steering that requires many turns of the wheel to accomplish a given degree of turn, and by the softness of suspension. Dynamic stability requirements are such that if the car is caused to skid by brake application on a smooth, dry surface at 60 mph, it should, when the brakes and steering are released, resume its straight-ahead normal running from any angle of deviation up to 60° from its straight-ahead position. It is possible to attain this degree of stability without undue sacrifice of steering responsiveness. The vehicle should be capable of sustaining a turn of .6g lateral acceleration without breaking away, and on breaking away at any acceleration above .6g it should behave in an understeering manner. This is to say that its angular velocity should not increase, or that it should not "spin out".

Cars carrying more than 1400 lbs. on the front wheels, or, if front-drive, more than 1200 lbs. on the front wheels, should in our opinion be equipped with power-assisted steering.

1.5.2.2 <u>Static Stability and Effort</u>

Static steering stability should be such that when the car is driven on a flat dry surface it should pursue a straight-ahead course without wandering when the wheel is released, or when it is held in the center position. Steering effort, whether power-assisted or manual, should be in some proportion to lateral acceleration up to 10 lbs. rim-pull. Too much weight-saving in steering linkages is to blame for loss of toe-in adjustments, to which radial tires are especially susceptible, with resulting loss of static directional stability.

1.5.3 <u>Braking</u>

Braking performance cannot be specified in terms of maximum attainable rates of acceleration because these are primarily a matter of tire tread and road surface characteristics (1-2). Brakes in normal passenger-car service are called upon for two primary duties, the first is the capability for an emergency stop with the maximum available acceleration, without loss of control, the second is the capability to dissipate considerable amounts of heat in descending long grades. This does not include the capability to withstand repeated heavy application at high speeds, which is the primary requirement for sports-car and racing application, and which is the outstanding virtue of the disc brake.

Brake performance is also concerned with distribution of effect, with the relation between effort and effect, and with the maintenance of controllability during brake operation.

1.5.3.1 Braking Effort

When the car is decelerating, each occupant must press forward against something to hold himself in his seat, and many restraint systems that are governed by inertia reels do not perform this function even if the occupant happens to be using it. It is therefore of little value to provide brake pedal effort much lower than 50 lb/g. It is of much greater importance to provide pedal effort that bears some nearly linear relationship to vehicle acceleration during braking.

1.5.3.2 Braking Effect Distribution

Optimum distribution of effect between front and rear depends on the state of loading of the vehicle. Because of the disastrous effects on stability of rear-wheel skidding it is desirable that rear brakes be actuated at something less than optimum proportion. This requirement is in conflict with the desirable feature of equalizing front and rear brake wear, and some vehicles incorporate systems whereby the rear brake performs most of the work on low-effort applications. This is dangerous, however, on slippery roads, as it can cause skidding on light applications.

Another common source of unsatisfactory front-rear effect distribution is the practice of using self-energizing drum brakes in the rear in conjunction with non-self-energizing disc brakes in front. Since the self-energizing brake effort varies as the square of lining coefficient, there is undesirable variation of effect distribution with factors, notably humidity, that affect brake lining friction coefficient. This practice is a carryover from the time when many cars offered front disk brakes as an option, but since the disc brake demands a power booster in all except the smallest of cars, there is no gain in using the self-energizing brake in conjunction with it.

1.5.3.3 Non-Skid Devices

Non-skid systems are coming into common use, but they present problems. They tend to reduce the maximum available acceleration on dry pavement, or they have roughness problems (1-3). One of the difficulties with systems prevalent today is that they stem from systems developed for aircraft use and none of them include sensors of vehicle acceleration. The ultimate non-skid system will incorporate an acceleration sensor, and it will be designed to maintain a fixed relation between the angular acceleration of the wheel and the linear acceleration of the vehicle (1-4).

The most important function of a non-skid system is to prevent the rear wheels of the car from locking. This prevents the car from skidding and proceeding into an impact sidewise, with greatly increased trauma. Unskilled drivers tend to panic on the brake pedal, and the assumption is that if they do that they will not be capable of making use of the steering to avoid an impact even if it were available to them. Thus, locking of the front wheels makes the vehicle unresponsive to steering, so that it proceeds front-end first without regard to steering input so long as the rear wheels do not lock.

Four-wheel non-skid systems are much more expensive, but they maintain full control of the vehicle at all times. The cost in reduced maximum acceleration is dependent on tire and road-surface conditions (1-3).

Redundancy of actuating systems for insurance against actuation-system failures is presently universal practice.

1.5.3.4 Vehicle Stability During Braking

The problem of preserving the directional stability of the vehicle during braking, especially on rough roads, is a matter of keeping the rear wheels on the ground, which is primarily a function of suspension design. There is a compromise to be made between comfort and safety in this regard, where the choice of relatively low damping for maximum comfort on smooth roads is in conflict with the need for high damping to prevent wheel-hop on rough roads, especially during braking. This problem is particularly critical with the usual live-axle, rear-drive design because of the large unsprung weight of the rear axle, and the unfortunate presence of the heavy final drive right in the middle of the axle, where it serves as an anvil for the transmission of disturbances from one wheel to the other. This subject is discussed at greater length in connection with design.

1.5.4 Acceleration Capability

Acceleration capability is of importance to safety particularly in the passing situation on two-lane roads. Whatever the power-weight ratio of the vehicle may be it is important that it have one transmission ratio in which the maximum power of the engine is available over the range 50-70 mph to provide the best possible acceleration in this speed range, since this is the range in which the probabilities of being trapped on the wrong side of the road are the greatest, and the consequences of a collision are at a maximum. Chapter 4 herein discusses the quantitative relationships that bear on acceleration capability.

Other requirements are for a quick response to full-throttle actuation, and a quick shift into the transmission ratio that will provide the optimum acceleration for the particular speed at which the vehicle is moving. The prevalence of road-load gear ratios that optimize fuel economy by reducing engine revolutions, with resulting low acceleration capability in top gear, makes the matter of transmission ratios and their quick selection of much increased importance to safety, and it places a safety premium on the use of automatic transmissions that make the shift automatically instead of depending on the driver to select the proper gear.

The safety aspect of the automatic transmission, plus the fact that an increasing number of young drivers are being taught to drive on automatic transmission only, have persuaded us that all vehicles except sports cars should be equipped with automatic transmission as standard equipment. Reference 1-5 discusses at length the various provisions that can be taken to obtain fuel economy with the automatic transmission to equal that of the manual transmission with a skilled driver.

1.5.5 Traction on Slippery Roads

This is largely a matter of tire design and tire/road relationships. In this respect the radial tire has decided advantages over the bias-ply tire because of its lack of creep in the tire footprint. It is also a matter of proportion of total car weight on the driving wheels. The practice of installing large powerplants as optional equipment in relatively small rear-drive cars is particularly dangerous in this respect, since the heavy engine places most of the weight in front, while the high power makes the rear wheels particularly likely to slip under the application of power. Thus such a vehicle is very difficult to maintain in a stable position under acceleration, and hence it is susceptible to loss of control on slippery surfaces.

Front-drive vehicles are markedly superior to rear-drive under such conditions, since when the driving wheels slip the vehicle loses steering control but does not lose its stability, and proceeds directly forward. A front-drive car with a heavier engine will have a proportionately greater weight on the drive wheels instead of a lesser percent

1.5.6 Accessibility and Convenience of Operation of Controls

It is important that the driver have optimum relation to the steering brakes and throttle controls for safe operation. Styling considerations have led to the predominance of seating positions too low for good visibility, as has been noted, and this is particularly a critical condition for the driver of short stature. Since most seats are not adjustable vertically, this driver must sit on a cushion, which disturbs his relation to the seat, as well as moving him farther from the pedals. He must then move the seat forward to reach the pedals, which puts him too close to the steering wheel, which is ordinarily not adjustable.

The problem of adaptation to varying statures is one whose solution will involve cost and weight penalties which may not be acceptable for the base vehicle, which is adaptable to the majority of drivers. Optional adjustable seats and steering controls are available on some vehicles, but there is need for vertically adjustable seats that do not involve the extra cost and weight of a power adjustment.

There is also need for standardization of the locations and modes of operation of minor controls such as horns, heating ventilation, radio, ashtrays and lighters, windshield washer/wiper, window lifts and such. The driver who regularly drives more than one car, or who is driving a strange car, such as a rental vehicle, finds the need to look down, away from the road, to find out where the control is, and often to read labels and instructions, which introduces a considerable safety hazard. Similarly a split-second delay in locating the horn control, for example, can cost a life.

1.6 PASSIVE SAFETY

Crashworthiness and Occupant Restraint

The relation between crashworthiness and vehicle weight reduction is a subject beyond the scope of this study. In principle it may be said that in collision with a barrier or with a similar vehicle there is no reason why a light vehicle should be any less crashworthy than a heavy one. In a rollover situation it is even possible that the lighter vehicle would be less likely to structural collapse than the heavier one.

On the other hand, considering the probable mix of heavy and light vehicles that would follow a general reduction in vehicle weights, in vehicle to vehicle impact the lighter vehicles would be at a greater disadvantage relative to the heavier vehicles than would heavier vehicles of equally good design. (See also "Limitations of Weight Reduction", 1.9.) There are obstacles, such as trees and telegraph poles, which would be an effective barrier to a light vehicle, which would be demolished by a heavy one.

1.7 OPTIONAL FUNCTIONAL CHARACTERISTICS

These functional characteristics have been discussed under a separate heading because they are variable with cost, and their availability is to a considerable degree optional with the buyer, who may choose to buy an economy model in which they are available to a lesser degree, or he may choose a more expensive model, or an option at extra cost if he chooses to have them available in greater degree.

1.7.1 Riding Comfort and Weight Reduction

A vast literature exists on the subject of ride quality, most of it studied in connection with various public transportation vehicles (1.6). In the automotive industry, judgements are still mainly subjective and ride-quality development is an art, not a science. One quantitative article of faith is that low oscillation frequencies are necessary for comfort, and the best-riding domestic cars tend toward a vertical oscillation frequency of 1 Hz, corresponding to a static deflection of 9.6" (24 cm).

Automotive vehicles almost without exception employ hydraulically damped, essentially linear steel springs as suspension media. It has long been recognized that the optimum damping for comfort on a smooth road will provide insufficient damping for stability on a rough road, or at high speed. At various times adjustable dampers have been made available to compensate for this disparity, but the inevitable experience has been that users will fail to make the needed adjustment and then complain about unsuitable adjustment. In general domestic practice tends toward underdamping for low-speed, smooth-road comfort, while European practice is generally to use slightly higher oscillation frequencies and greater damping.

The only fundamental handicap of the lighter vehicle concerns the greater proportion of its disposable weight. A 2800 lb car with 1300 lbs on the rear wheels at curb will carry 2225 lbs on its rear wheels with a normal load of 6 passengers and 150 lbs of luggage, an increment of just over 71%. A 5000 lb car with 2300 lbs on the rear wheels at curb will carry 3225 lbs on the rear wheels with the same load, an increment of only 40.2%.

If the lighter vehicle had a static deflection of 9.6" at curb, it would deflect an additional 6.8" under load, a completely unacceptable change of attitude, while if it had no more than 9.6" under load it would have 5.6" at curb, and the resulting 4" of deflection under load would still be unacceptable. The heavier car, with 9.6" when loaded, would have 6.84" at curb, with an acceptable deflection of 2.76" and an oscillation rate of 1.18 without load. Clearly the only option available with the light car is to employ a higher oscillation rate or to use a self-levelling suspension. Since most light cars have been inexpensive, the choice has been to use a smaller static deflection and to accept the higher oscillation rate.

Ride quality being to a large degree a subjective matter, there is not universal agreement concerning the choice of 1 Hz as the optimum oscillation rate, and there are those who argue that it ought more nearly to approach the normal frequency of walking. Most imported vehicles employ higher oscillation rates than domestic vehicles and there is general acceptance of the ride quality of the best of these vehicles, of which the Peugeot is an example.

There are additional recourses for improving ride quality in lighter-weight vehicles that are not commonly used, in addition to the self-levelling suspension, such as acceleration-responsive dampers and mass dampers, that present opportunities that will doubtless be exploited in the future if the weight of vehicles is to come down without excessive loss of functional quality. It will be assumed here that the problem of ride quality is solvable, and the matter of its relation to design will be discussed in a later chapter.

1.7.2 Noise-Vibration Suppression and Weight Reduction

The suppression of noise and vibration from the vehicle powertrain and the road is basically a matter of adequate low-pass filtration, making optimum use of compliance and mass (1-7). Obviously the more mass available the simpler this assignment becomes. The largest domestic cars are excellent examples of the high development of this technique, and are without question the quietest and smoothest vehicles on the road. Since the most expensive vehicles have always been the heaviest, and have always had the most attention paid to noise-vibration suppression, there has always been a correlation between weight and quietness and smoothness of operation.

In lighter vehicles it is necessary to optimize noise-vibration filtration techniques by more scientific design, and the extensive use of analytical techniques by foreign makers has shown the possibilities existing for good performance without excessive weight. As weight reduction becomes a subject of interest in the U.S., more attention is being paid to this recourse and rapid progress is being made in the production of lighter-weight vehicles of good noise-vibration quality.

Nevertheless there will always be a margin whereby the heavier vehicle will enjoy superiority in this respect over the lighter one. Not only does the larger vehicle enjoy the natural advantage of greater disposable mass for filtration of noise-vibration, but with the advantages of the V-8 engine over other forms there is less noise to filter. It will be assumed here that acceptable noise-vibration quality can be obtained with lighter vehicles, but it cannot be said that these will ever be as good as heavier vehicles, equally well designed.

1.8 MANDATORY CONSTRAINTS

In Section 2 of this report we note that between 1970 and 1974 the weights of the Ford car increased 16.35%, or 619 pounds. During the same period the Maverick increased in weight by 11%, or 281 pounds. Our analysis shows that of the former, about 500 pounds was due either to direct additions of weight to meet various government regulations, added engine size to maintain acceleration capability or recoup reduced output due to emission control measures, or to added weight in structure, tire, wheels, suspension, brakes, etc., as a consequence of added weight. (See Section 3 of this report for methods for estimating total weight additions for a given arbitrary weight increment.) Engine displacement was increased 16.2% in the Ford car during this period, but acceleration capability was not maintained at the 1970 level by this change, since the engine displacement increase was no greater than the increase in car weight, leaving no margin for the recovery of reductions in specific output resulting from emission controls.

In the Ford car about 57 pounds were saved by various improvements in design, 85 pounds added in the changeover from the pillar sedan to the hard-top body, and 90 pounds for added standard equipment and quality improvements.

In the Maverick engine displacement was increased from 170 to 200 within the same basic engine structure for only 5 lbs incremental weight. Of the 308 lbs (11.5%) of total weight increases made in this car during the period 1970-74, we estimate that 250 lbs were the direct or indirect consequences of actions taken to satisfy government regulations. We estimate that acceleration capability was not maintained through the displacement increase.

The Pinto increased 26.7% in weight during the 1971-75 period. Two increases in engine displacement were made, for a total increase of 43.7%, and acceleration capability was probably maintained, thus the proportionate weight increase was greater.

33 lbs of weight reductions made the total increase 576 lbs, of which it is estimated that about 400 were the direct or indirect result of weight additions to meet government regulations.

Marx et al. (1-8) report an 18% increase in the weight of a 1670 lb 4-passenger vehicle to bring it from the point of meeting foreign requirements up to meeting U.S.-mandated requirements, with a 9% loss in acceleration capability.

We are unable to estimate the weight-increase consequences of possible future regulations, such as decreased Nox output, air-bags, and increased impact requirements. Marx discusses these subjects.

We believe that in the design of an all-new vehicle the penalties for front, rear and side impact requirements could be reduced somewhat over the reported figures for cars already designed which require to be modified to meet the requirements.

Marx reports that the base vehicle of his report was able to withstand a 30 mph frontal impact without added special provision. We believe that this statement should be true of any well-designed vehicle.

1.9 LIMITATIONS OF WEIGHT REDUCTION

1.9.1 Safety; Impact Trauma

When two vehicles collide, the lighter one will be subjected to acceleration in inverse relation to the ratio of its weight to that of the colliding vehicle, hence acceleration trauma will be even more unequal, since trauma is not linear with acceleration. Sachs (1-9) discussed the relationship of light and heavy cars in general as well as with specific regard to impact trauma. Scott (1-10) reports that large-scale statistical studies show accident severity to be in inverse proportion to vehicle weight. There are many proposals to increase crashworthiness (1-11), and more cost-effective means of improving vehicle crashworthiness are constantly being sought, but whatever the optimum degree of crashworthiness, heavier vehicles will prove to be more protective of their occupants than lighter ones, assuming the mix of old and new cars that would obtain over the coming ten years.

1.9.2 Safety; Aerodynamic Stability and Controllability

Aerodynamic disturbing forces are a function of area, while restoring and control forces are a function of weight. Hence aerodynamic forces become more serious as the car weight is decreased for a given size. Sachs (1-9) evaluates the effects of weight reduction on controllability, noting that removal of mass increases yaw amplitudes and roll deflections resulting from a given set of wind forces, hence vehicles of reduced weight are more vulnerable to disturbances and greater driver skill is required to deal with the disturbances that arise from a normal passing maneuver without loss of control. Brown et al. (1-12) and Weir et al. (1-13) show quantitative values of forces while passing or being passed by a truck in a cross wind that are shown to be rapidly varying in force and direction and to demand driver skill in inverse proportion to vehicle weight, which can reach a degree probably beyond the capabilities of the average driver. Weir et al. also show that the forces are substantially the same in two vehicles of considerably different size and configuration (a VW and a Torino). It may be inferred from this that a reduction in the average weight of vehicles will result in an increase of accidents resulting from loss of control.

1.9.3 Durability

One engineering constraint on weight reduction is to accomplish this objective without reduction in durability. The use of metal gauges too thin to resist normal impacts, of suspension parts too light to have an adequate safety factor, of small engines running at excessive revolutions (since the life of an engine is best measured in terms of revolutions), and of seats and internal appointments too light to withstand normal wear and tear, are all measures that have been worked over in the past in the interests of low weight and cost. It is not engineering design to reduce weight in this manner, it is simply a short-sighted economic and product decision.

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2. COMPARATIVE STUDY OF PAST AND PRESENT AUTOMOBILE DESIGNS

2.0 INTRODUCTION

It is an aid to understanding the product characteristics of modern cars to have some glimpse of the path whereby they evolved into their present forms, and the marketing influences that guided that evolution. As we have remarked, the influence of appearance factors on car design has been a major one ever since their recognition as an important marketing influence. It may be said that this occurred in the 1920's with the appearance of lacquer finishes that permitted quantity production of other colors than black, and with the promptly resulting demise of the Model T Ford, which one could buy in any color one wanted so long as it was black (Figure 2A).

The first lacquer finishes appeared in 1924, and Chevrolet first began to use them in 1925. With substantially the same little car that had been around since 1922, they began to advertise "Beautiful" and show pictures of multi-colored Chevrolets, along with peacocks. We plot the market penetration of the Model T Ford and the Chevrolet for the years 1924-27 to show the effect of this new approach to marketing, and 1927 was the end of the Model T (Figure 2B). 1927 was also the year that GM established its Art and Color Section, and initiated the practice of modeling cars in clay in full-scale before approving them for production.

The end of the Model T also marked the end of the small, cheap car in the U.S. market, and its replacement by the used car.

We trace the evolution of the vehicle package from that time until the present, once more showing the dominating influence of appearance factors in car design. We trace the use of alternative materials in cars and we compare vehicle weights between typical current cars and those of the 1960 period as well as those of the 1970 period.

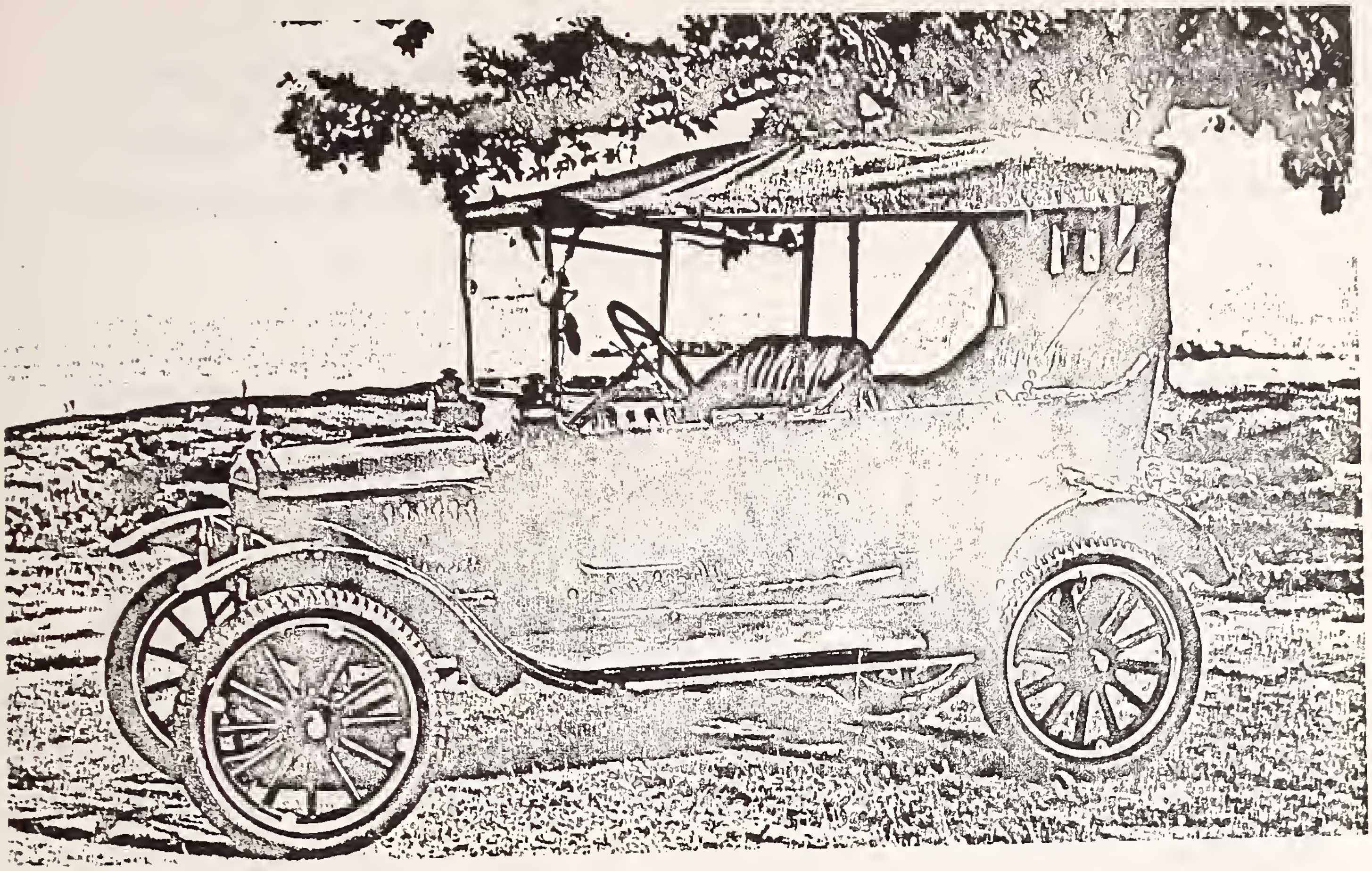


FIGURE 2A. MODEL T FORD

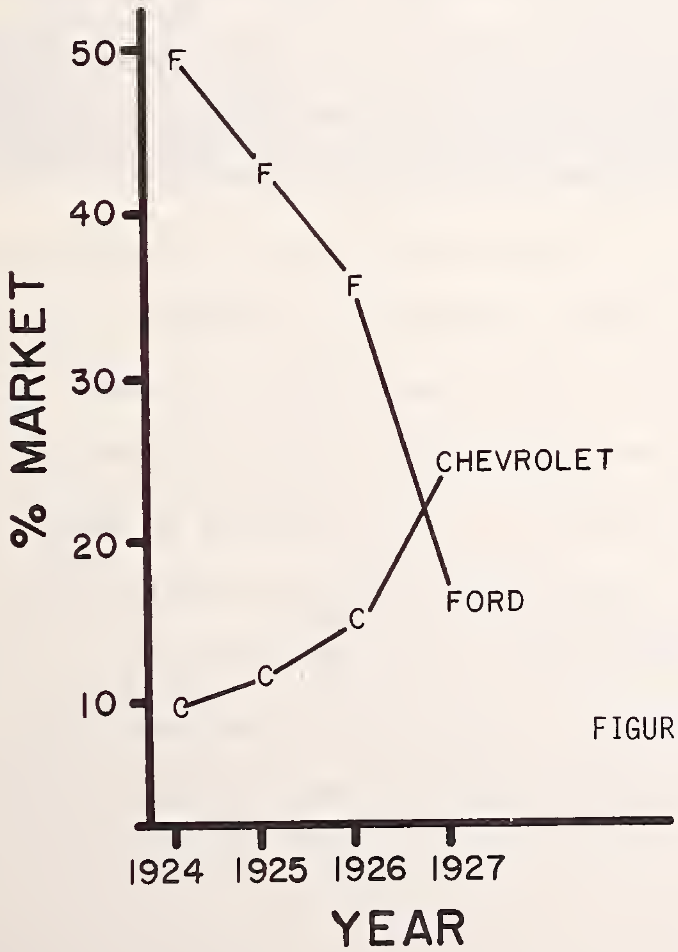


FIGURE 2B. MARKET PENETRATION 1924-27
FORD AND CHEVROLET

Qualitative comparisons are made between different alternative design characteristics, and in one instance, that of unitized vs. body-frame construction, an example is presented in which other characteristics are sufficiently alike that quantitative comparisons are possible.

2.1 THE EVOLUTION OF THE PACKAGE

The driver of a horse-drawn vehicle sits high enough to see over the horse. The driver of the early car clearly could not afford to occupy a lower position in the world than the horse-driver with whom he had to share, and often dispute, the road, so he sat equally high. The large, expensive cars of the day had engines, like the horse, out in front of the driver, while the back seat occupied a position over the rear wheels. The wheels were large, to cope with the deeply rutted roads, and spaced at the standard wagon track of 4 feet 8 inches. This meant, for a large car with 37-inch rear wheels (some of them were as much as 42") that, allowing for spring deflection and chain clearance and a well-sprung seat cushion, the seating position of the rear passenger would have to be a minimum of 50" above the road, and some were as high as 60", a good deal higher than the roofline of a present-day car (Figure 2C).

Early in the 1920's when the car had established its own position as a vehicle, the seating positions began to come down. The wheels became smaller but the back seats either had to be narrow enough to fit in between the wheelhouses, or high enough to span over them, because hoods had become even longer with the advent of the straight-8 engine, which was espoused by Packard, the prestige leader of the decade. Race cars were built with their engines far back in the chassis, and the radiator well aft of the front axle, so it was acceptable -- in fact sporty looking -- if the radiator was well back. Trucks had the engines well forward, with the radiator forward of the front axle, so

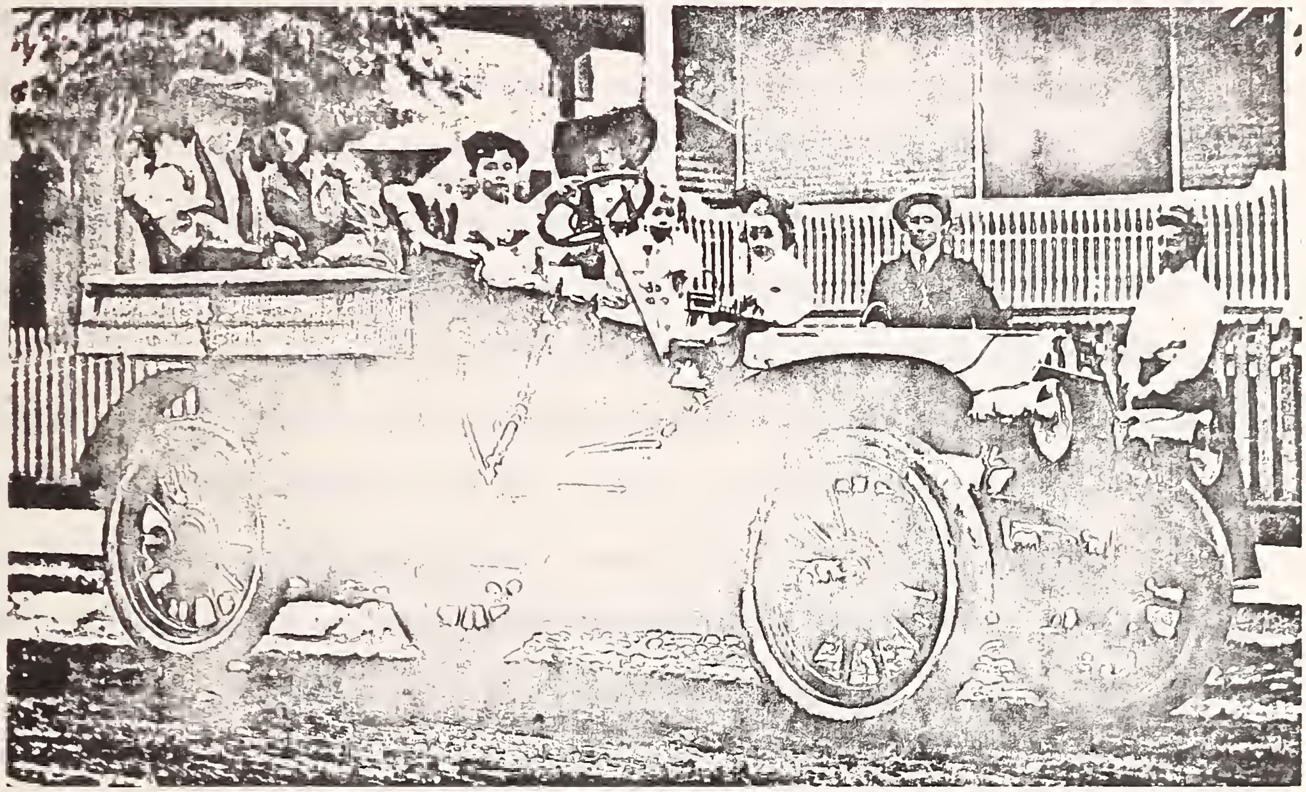


FIGURE 2C. 1904 PEERLESS

anything that had the radiator forward of the axle looked like a truck and just wasn't acceptable. Consequently a lot of good space on the chassis went to waste, while the passengers in the back seat rode over the rear axle. In those days the prestige image was for a car driven by a chauffeur, with the owner sitting in back, so the back seat had to be roomy. Often it had room enough for a couple of extra folding seats, and the car became a "7-passenger".

This image of the car carried over to the 30's. At the same time it had been recognized for some years that the sensible thing to do would be to move the engine up between the front wheels, leave the space between the front and rear wheels clear for the passengers, and house luggage, gas tank, spare wheel, etc., between the rear wheels. Meanwhile wheels had become much smaller, and it was no longer necessary to maintain their track at 4 feet 8, so the rear seat no longer had to be above the wheels, but it was still limited in width.

In 1933 Plymouth made the first move toward this more rational package. It was to move the front axle aft 5" from its earlier position under the radiator. It was also the introduction of the first Plymouth 6-cylinder engine, but the public did not warm to the new look and a mid-term increase of 5" in the car's wheelbase was made, moving the front wheels out forward once more.

In 1934 Chrysler brought out the famous Airflow. It was a car of advanced concept, structurally stiff with soft springs, with excellent handling characteristics and a tremendously improved package, with the engine forward and a rather modern-looking grill. The rear seat was moved well forward of the rear axle, but not forward enough to clear the wheel houses. It looks less out of place today than it did when it was new. People did not like its appearance, or at least the industry so interpreted its poor sales. (There was an alternative reason, which was that the car was very expensive compared with its more normal contemporaries, no small factor in 1934.) After being restyled to look more like an ordinary hood and radiator in front, it was still not popular, and was removed from the market after only four years.

In 1935 Ford boldly adopted a short hood, well over the front axle, with the rear seat moved forward of the rear axle, and they stuck with it. GM finally brought out a car in 1940 that was much along the lines of the Airflow, but to hedge its bets it continued the 1939 models having the older configuration. This time the cars were accepted, but it took 15 years from the time it became clear it should be done until the public finally bought it.

It wasn't until 1946 that Studebaker went the rest of the way to liberating the package by moving it far enough forward for the rear seat to clear the rear wheel houses entirely. This involved a further recognition of changed conditions, which was that the owner had moved from the rear seat to the front, and that all that rear-seat legroom was no longer necessary. The evolution of the modern concept of the package was complete. Hudson followed in 1948 and Ford in 1949. It now remained only to be learned how much restriction of living space the customer would tolerate in the interest of stylish appearance. The hardtop, with its lower roofline, continued to outsell the conventionally-bodied cars, the 4-door hardtop completely displaced the pillar sedan in the 6-passenger class, and at present the 2-door hardtop, with even more severely restricted rear-seat room, is the most popular body style in the U.S. Exterior size is the characteristic to which the customer is most sensitive, and interior size seems to be a matter of almost complete indifference.

The process of reducing the passenger space in the interests of appearance has continued into the '70's. The Valiant is the only major domestic product that has been carried over from the early 60's without lowering of the roof, increasing tumblehome, windshield and backlight slopes to reduce passenger space. In Table 2A we list average interior height, and width at the line of the roof-rail for a representative group of vehicles of the 1960's and their equivalents of the 1970's.

TABLE 2A. AVERAGE INTERIOR HEIGHT AND ROOF-RAIL WIDTH FOR VEHICLES OF THE 1960's AND 1970's.

	Average Interior Height	Average Roof-Rail Width
1960's 5-passenger	44.9	49.34
1970's 5-passenger	43.28	44.625
1960's 6-passenger	46.13	51.40
1970's 6-passenger	44.03	48.20
1970's imports 5 passenger	45.82	44.77

2.2 A DETAILED WEIGHT BREAKDOWN OF PAST AND PRESENT 4-, 5-, AND 6 PASSENGER AUTOMOBILES

We present detailed weight-breakdown information on a 1958 Renault Dauphine, a 1975 Fiat 128, both 4-passenger cars, a 1960 Ford Falcon and a 1974 Ford Maverick in the 5-passenger class, and an 8-cylinder Ford car in 1960 and 1974, in the 6-passenger class. These are tabulated in Table 2B by component group, with detailed body-component breakdown in Table 2C, and percentage of total curb weight by component group in Table 2D.

The Renault Dauphine of 1958 (Figure 2D) is the lightest 4-passenger, 4-door vehicle on which we have weight information. It was not a success on the U.S. market but despite its lack of market success the Dauphine is a useful study in the techniques of weight reduction, and a useful object lesson in how to carry gauge reduction too far. Later versions of the Dauphine were more satisfactory as products, and they were heavier, with more power.

The Dauphine was a rear-engine, rear-drive design with a swing-axle rear suspension that is light and cheap but unsatisfactory for reasons of poor directional stability, and an example of an effective weight-reduction measure that is unsatisfactory for functional reasons. The Dauphine uses conventional

TABLE 2B. DETAILED WEIGHT BREAKDOWN BY COMPONENT GROUP

1974 For Classification System

Group	1958 Dauphine	1975 Fiat 128	1960 Falcon	1974 Maverick	1960 Ford	1974 Ford
1. Body	693	1010	1184	1460	1668	1893
2. Frame, tools & engine mounts	18	-	17	21	340	350
3. Engine	206	284	368	430	643	693
4. Suspension	202	253	325	350	469	458
Front suspension	59	58	112	95	147	125
Rear suspension	24	65	66	78	103	91
Wheels & tires	117	130	147	179	219	241
5. Axle & driveshaft	55(2)	39(1)	90	139	155	177
Axle & lube	55	20	76	124	139	155
Driveshaft		19	14	15	16	22
6. Brakes	70	66	98	135(3)	124	208(3)
Front brake & disc	26	29	46	64	50	97
Rear brake & drum	26	23	34	43	52	63
Parking brake	6	5	9	8	11	11
Actuation & fluid	12	9	9	20	12	36
7. Transmission & fluid	36	66(4)	51	87	85	203(5)
Transmission	32		44	83	77	185
Controls	3		5	2	6	2
Fluid	1		2	2	2	16
8. Clutch	11	25	24	33	58	-
9. Exhaust	7	25	23	29	46	53
10. Fuel system & fuel	68	80	107	124	145	159
11. Steering system	24	30	42	60	53	100(7)
Gear	12	12	21	17	17	33
Linkage	4	8	11	17	19	19
Column & wheel	8	10	10	26(8)	16	27(8)
Pump lines & fluid						20
12. Heating & ventilating		30	6	21	6	30
13. Instruments & horn	8	2	8	5	22	5
14. Battery & alternator	66(6)	48	50(6)	44	69(6)	47
15. Radio						
16. Vacuum system						
17. Electrical distribution	13	17	8	11	8	19
TOTALS	1475	1975	2401	2949	3891	4397

Notes:
1. Front drive, driveshafts & universals only
2. Rear engine, no propeller shaft
3. Power front disc brakes
4. Transmission includes final drive
5. Automatic transmission
6. Generator instead of alternator
7. Power steering
8. Collapsible, energy-absorbing column

TABLE 2C. CAR BODY WEIGHT COMPARISONS

1974 Ford Classification System	1960 Ford		1974 Ford	
Group				
Subgroup				
Body structure		519		598
Roof	60		66	
Underbody	181		236	
Body cowl	96		82	
Body side & quarter	130		146	
Body rear end Structure	37		50	
Miscellaneous parts & weld	15		27	
Front End		204		218
Front fenders & aprons	87		96	
Hood & hinges	98		82	
Radiator support assy. & grille Structure	6		28	
Miscellaneous parts	13		5	
Doors, deck lid & hinges		215		246
Front doors & hinges	97		108	
Rear doors & hinges	75		83	
Deck lid & hinges	43		47	
Sound & heat control		53		83
Seats, trim & tracks		164		131
Fixed glass		78		58
Movable glass		67		42
Glass runs, tracks, mechanism		24		33
Instrument panel		19		27
Handles & locks		28		18
Exterior ornamentation		14		23
Bumpers		118		245
Body lamps		20		25
Windshield wiper/washer		9		18
Internal ornamentation		10		10
Interior trim		81		59
Paint		6		15
Seals		24		15
Safety restraint				17
Miscellandous parts		13		3
TOTAL BODY WEIGHT		1666		1893

TABLE 2D. CAR WEIGHT COMPARISONS (PERCENT CURB WEIGHT)

1974 Ford Classification System

Group	1958 Dauphine		1975 Fiat 128		1960 Falcon		1974 Maverick		1960 Ford		1974 Ford	
1. Body	47.0	48.2	51.1	51.1	49.3	50.0	49.5	50.2	41.9	50.4	42.8	51.3
2. Frame, tools & engine mounts	1.2				.7		.7		8.5		8.5	
3. Engine	14.0		14.4		15.3		14.6		16.2		15.7	
4. Front & rear suspensions	5.8		6.2		7.4		5.9		6.3		4.9	
4. Wheels & tires	7.9		6.6		6.1		6.1		5.5		5.5	
5. Axle & driveshaft	3.7		2.0		3.7		4.7		3.9		4.0	
6. Brakes	4.7		3.3		4.1		4.6		3.1		4.7	
7. Transmission & 8. Clutch	3.2		4.6		3.1		4.1		3.6		4.6	
9. Exhaust	.7		1.3		1.0		1.0		1.2		1.2	
10. Fuel system & fuel	4.6		4.0		4.5		4.2		3.6		3.6	
11. Steering	1.6		1.5		1.7		2.0		1.3		2.3	
12. Heating & ventilating	-		1.5				.7		.2		.7	
13. Instruments & horn	.5		.1		.3		.2		.6		.1	
14. Battery & alternator	4.4		2.4		2.1		1.5		1.7		1.1	
17. Electrical distribution	.9		.9		.8		.4		.2		.4	

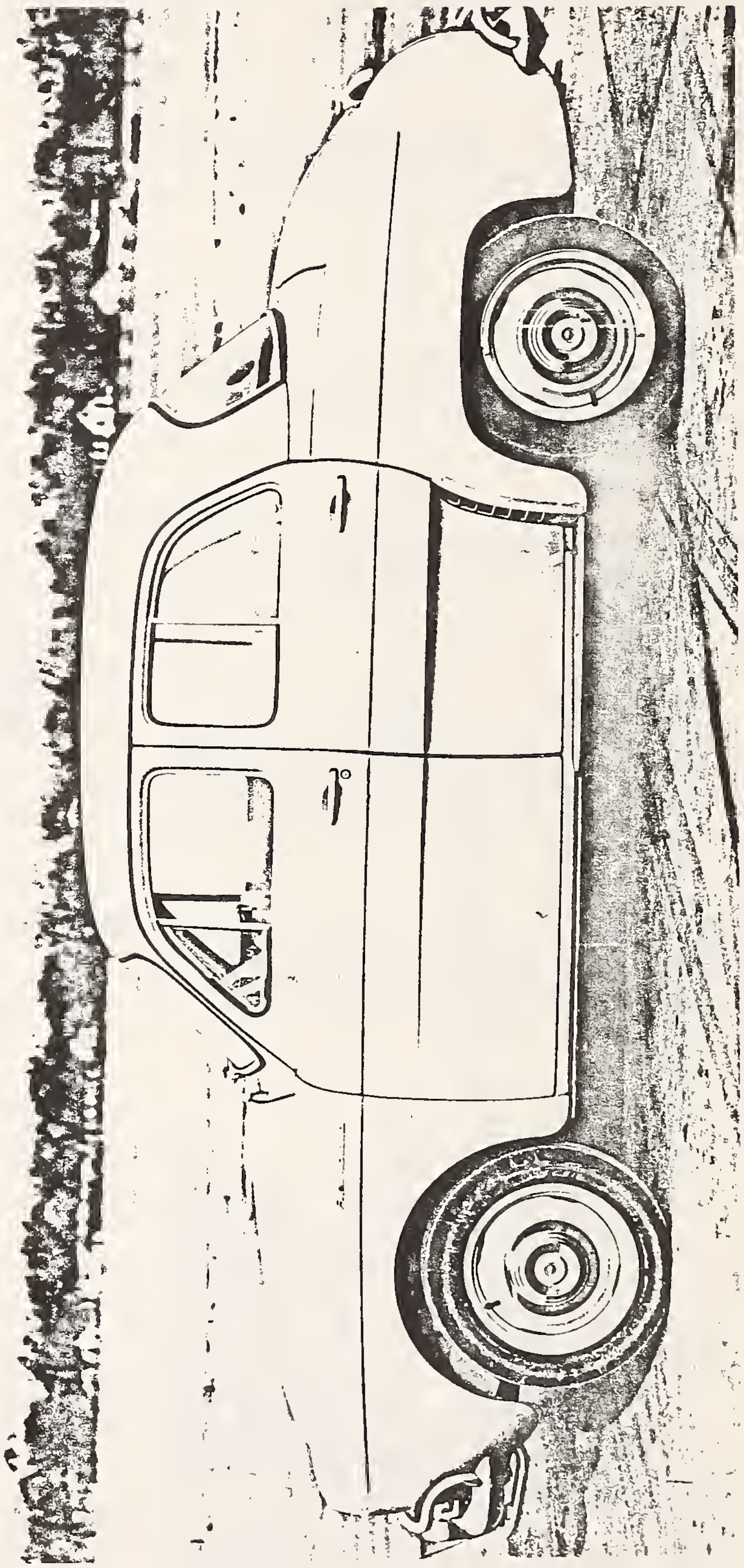


FIGURE 2D. 1958 RENAULT DAUPHINE

materials throughout, owing none of its lightness to lightweight materials.

The Fiat 128 is a front-engine, transverse-engine front-drive of modern design, well-developed, notably roomy, and adequately, if not generously, powered. While it has about the same weight per cubic inch of engine displacement as the Dauphine, its overhead-cam engine is much more capable and faster turning. The design, however, was, as late as 1972, unsatisfactory from the standpoint of rust resistance.

In comparing the rear-engine rear-drive with the front-engine front-drive, it may be seen that the former has a weight advantage in its lack of a propeller shaft and exhaust input and tail pipes, and its very light rear suspension, while it lacks the cost and complexity of the front-drive individual drive shafts and their heavy steering-pivot universal joints. However, when the rear-drive is changed to have a satisfactory rear suspension it gains most of the weight and complexity of the front-drive, with individual driveshafts and outboard, as well as inboard, universal joints. The lighter front suspension of the Fiat is to be credited to its more modern design, and not to any advantage of the front-drive. Comparing the two designs on an overall basis, it may be seen that the front-engine front-drive design enjoys functional and safety advantages at substantially no cost in increased weight, and it may be seen why it has largely supplanted the rear-engine design in modern vehicles (Figure 2E).

In the other size classes the cars are of similar design and configuration, all being conventional front-engine rear-drive cars of typical domestic configuration. In the 4-passenger class the Falcon of 1960 is compared with the smaller and heavier Maverick of 1974. The Falcon has been discussed elsewhere, and it is the lightest vehicle of its particular size class that has ever been built in this country. In durability the Falcon has been the equal of heavier

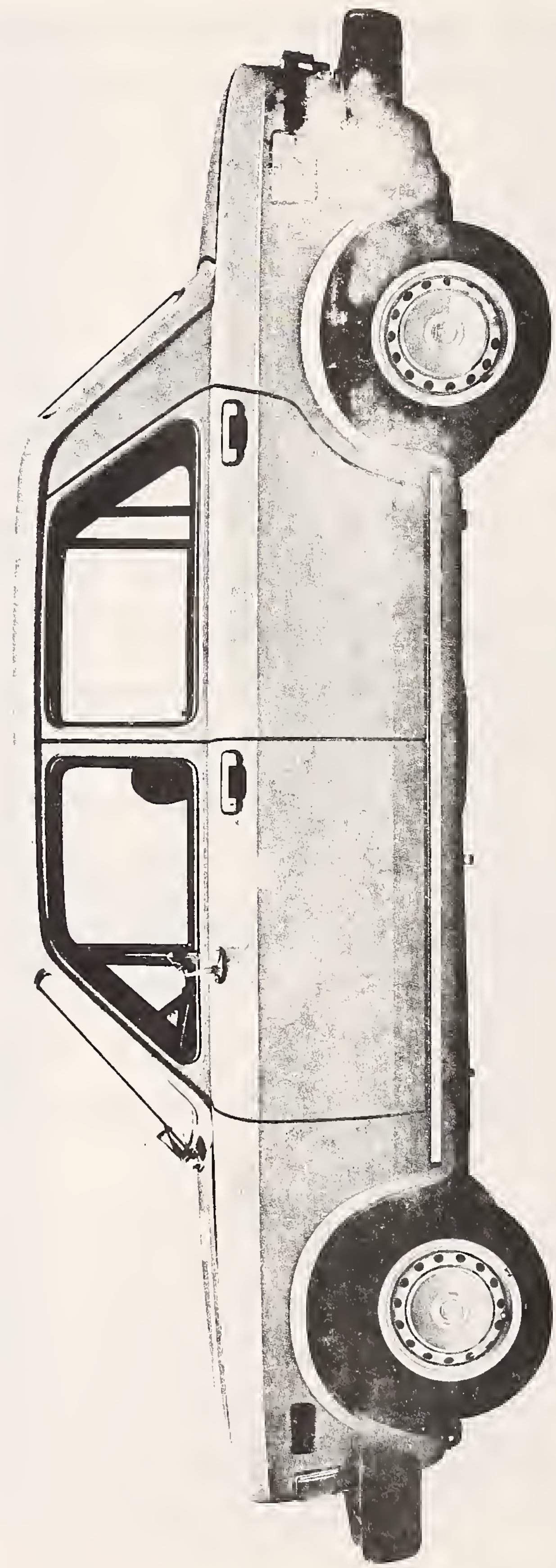


FIGURE 2E. FIAT 128

vehicles in all respects, so that it may safely be said to have involved no sacrifice of durability to weight reduction.

The reduced interior dimensions of the Maverick reflect styling influences. The lower roofline forced lower seating positions, which in turn forced the seats rearward, with the result that the Maverick rear seatback is between the wheelhouses and the rear seat is comfortable only for two people. This, as has been remarked, is the history of package development in this country, in which buyers seem to exhibit no interest at all in roominess, but to respond directly to snazzy appearance.

The two 6-passenger Fords are the only two that represent a continuity of product line and a continuous process of evolution from 1960 to 1974, and for that reason we have chosen to make a more detailed analysis of their weight differences. The 1960 Ford was newly enlarged that year in response to an enlargement of the Chevrolet in 1959. The Ford had been newly enlarged in 1957, both lengthened and lowered, with the result that Ford sales exceeded those of Chevrolet for the first time since 1935. Chevrolet responded in 1959 by leap-frogging Ford with a still bigger car. Both cars were shrunken a little for 1961.

The Ford was redesigned in 1965, with a weight reduction of 300 lbs, and it was 1971 before it had once more climbed to the 1960 high point. (See Ford car weight history, Table 2E.) The biggest single jump prior to that came in 1969 when the car grew larger again, with more extreme styling.

2.2.1 An Accounting of Weight Differences Between 1960 and 1974 Ford Cars

In order to form a basis for comparison between 1960 and 1974 designs we will study in detail the 6-passenger Ford cars, since these cars are the most nearly comparable in size, product characteristics, and design, and the weight information is most complete.

TABLE 2E. WEIGHT HISTORY OF FORD CAR, 1960-1974 (8 CLY.)

Year	Weight	Wheelbase	Overall Length	
1960	3891	119	217	
1962	3855	119	209	
1963	3800	119	209	New, lighter engine
1964	3850	119	210	
1965	3541	119	210	New chassis & body
1966	3718	119	210	
1967	3700	119	213	
1968	3726	119	213	
1969	3823	121	216	Body changes & increased length
1970	3783	121	216	
1971	3895	121	216	Side intrusion & uprated tires
1972	4093	121	216	Power steering, 351 engine & FMX transmission, safety restraint, bigger tires
1973	4290	121	219	Hardtop body, high-energy bumpers, power disc brakes
1974	4397	121	222	Bumper, frame & engine revisions

We categorize weight increases for 1974 over 1960 in this product class in Tables 2F through 2J.

TABLE 2F. EQUIPMENT AS STANDARD IN 1974 THAT WAS UNAVAILABLE OR OPTIONAL IN 1960

Power-assisted disc brakes & upgraded brakes	84 lb	1974 over 1960
Automatic transmission	15 (1)	
Heater	24	
Power steering	45	
Government-mandated equipment:		
Side-intrusion protection	35	
Bumper, 5 mph impact	150	
Safety restraint	20	
Emission controls	30 (2)	
Interchangeability penalties:(3)		
Engine (351-2V-C vs. 351-2V-W)	30	
Transmission (FMX vs. C4)	30	
TOTAL	463	

NOTES:

(1) This is the weight of the C-4 transmission over the manual transmission and clutch of the 1960 car. Note that it amounts to just about the increased weight of transmission fluid.

(2) This weight does not include increases of engine displacement designed to offset decreases in power resulting from emission controls.

(3) The 351-C engine is designed for interchangeability with the 400 engine on the Cleveland engine facility, at a weight penalty of about 30 lb. The FMX transmission uses the basic parts of the C-6 transmission, designed to accept engines up to 460 ci.

TABLE 2G. WEIGHT REDUCTIONS RESULTING FROM IMPROVED DESIGN AND MATERIALS

	lbs	
Engine design & foundry technique	78 (1)	1974 under 1960
Front suspension design	30	
Rear suspension design	20	
Glass, reduced gauge	45	
Body mouldings, trims & seals, lighter materials	30	
Alternator substituted for generator	10	
Battery, instruments & horn, improved materials & design	25	
Tires & wheels, improved materials & design	25	
Improved design of frame	50	
TOTAL	303	

NOTE:

(1) This is the weight difference between the old 292 engine and the new 289 (later 303) engine.

TABLE 2H. WEIGHT INCREASE FOR UPGRADED QUALITY, FUNCTION OR STYLING.

	lbs
Hardtop over sedan	85
Longer wheelbase & coil rear suspension	15
Ornamentation & redesigned instrument panel	20
Tires, improved load rating & safety	20
Sound & heat control	30
Structural improvements, frame & body	54
TOTAL	224

TABLE 2I. SECONDARY WEIGHT INCREASES NOT ALREADY LISTED.

	lbs
Wheels & Tires	26
351 engine & FMX transmission over 303 and C-4	76 (1)
TOTAL	102

NOTE:
(1) This does not include interchangeability penalties already listed.

TABLE 2J. WEIGHT DIFFERENCE SUMMARY BETWEEN 1960 AND 1974 FORD CARS.

	lbs
1960 Ford Car	3891
1. Equipment not included in 1960	463
2. Upgraded product quality, styling & function	224
3. Improvements in design & materials	(303)
4. Secondary weight increases not already listed	102
1974 Ford Car	4397

Only minor differences of those above listed involve improved materials. These include the use of polyester instead of rayon or nylon in tires, the use of plastic materials as body interior mouldings instead of metal, and the substitution of foam materials for solid rubber in body seals. Many minor design improvements and functional improvements have been omitted and have compensated one another. Some weight penalty doubtless accrues in accomplishing new styling objectives by the maximum use of existing tooling, as contrasted to an all-new design, but no attempt has been made to evaluate this.

Of the 303 lbs of design and material improvements listed, almost all went into the car before, or as a consequence of the 1965 redesign. The 289 (later the 302) engine replaced the old 292 in 1963, and the 1965 car was 350 lbs lighter than the 1960 car and 309 lbs lighter than the 1964 car. The 350 lbs includes most of the 303 lbs listed above plus about 80 lbs of reduction of body weight.

2.2.2 Weight Accounting of 1970-1974 Cars

We analyze the weight increases that have taken place in the Ford Car, Maverick and Pinto classes, in the interval from 1970 to 1974 as shown in Tables 2K through 2M.

TABLE 2K. WEIGHT ANALYSIS, FORD CAR, 1970-1974

		Govt. Reg.	Other
Body side intrusion	35	35	
Bumpers	159	159	
Power disc brakes	55	35	20
Hardtop over sedan	85		85
Vibration damper	8		8
Structural upgrade	27	20	7
Sound & heat control	21		21
Wheels & tires	19	19	
Safety restraint	8	8	
Engine revisions for emission control	44	44	
351-C engine for 302	104	104	
FMX transmission for C-4	39	39	
Styling & ornamentation	14		14
Suspension & axle	18	18	
Power steering	40	20	20
Miscellaneous revisions	(57)		
TOTALS	619	501	175

1970 Base Car 3784
1974 Base Car 4403

Weight increase 619

Disregarding the weight reductions, total increases were 676 lbs, of which we estimate that 500 were the direct or indirect result of government regulations.

TABLE 2L. WEIGHT ANALYSIS, FORD MAVERICK, 1970-1974 (2-DOOR SEDAN, BASE CAR)

		Govt. Reg.
Bumpers	143	143
Side intrusion	42	42
Power disc brakes	29	15
Engine revisions for emission control	13	13
Safety restraint	9	9
Tires	23	13
Axle	22	10
Front suspension	13	7
Clutch	3	
Windshield wiper/washer	5	
Miscellaneous revisions	(21)	
TOTALS	281	252

1970 Maverick base car 2562
1974 Maverick base car 2843

Total weight increase was 302 lbs, of which it is estimated that 252 were the direct or indirect result of government regulations.

TABLE 2M. WEIGHT ANALYSIS, FORD PINTO, 1971-1975 (2-DOOR SEDAN, BASE CAR)

		Govt. Reg.
Bumpers	124	124
Structure	76	40
Engine, 1.6 liter to 2.0 to 2.3 & emission control	140	140
Body side intrusion	42	42
Safety restraint	17	17
Suspension & axle	40	20
Brakes, disc brakes for drum	29	15
Transmission & clutch	8	8
Wheels & tires	18	14
Steering gear & linkage	10	10
Fuel system	10	10
Heat protection	18	
Vibration damper	6	
Sound package	29	
Trim upgrade	9	
Miscellaneous revisions	(33)	
TOTALS	543	440

1971 Pinto, base car 2030
1975 Pinto, base car 2573

Total weight increase was 576 lbs, of which it is estimated that 440 were the direct or indirect result of government regulations.

For the Maverick, total weight increases were 302 lbs, of which we estimate that 285 lbs were the direct or indirect result of provision to meet government regulations. Size of the vehicle was unchanged, and engine displacement was increased from 170 to 200 ci with negligible weight increase.

For the Pinto total weight increases were 576 lbs, of which it is estimated that 440 were the direct or indirect result of provisions to meet government regulations. Vehicle size was unchanged during the period, but engine displacement increased 43% and acceleration capability was maintained.

2.3 CONSIDERATION OF SPECIFIC DESIGN CONFIGURATIONS

2.3.1 Vehicle Configuration

We have compared the weight characteristics of the front-engine, front-drive with the rear-engine, rear-drive, finding little of a fundamental difference so far as weight characteristics were concerned, provided that the obsolete single-pivot swing-axle were ruled out. In our opinion, one of the most important advantages of the front-engine design is the fact that much of the hard mass of the car is out in front of the bulkhead, where it can do its part in decelerating the impacted object before the passengers are impacted. Any serious study of crashworthiness is beyond the scope of this study, but we do assume the foregoing as the basis for rejecting further consideration of the rear-engine car.

The front-engine, rear-drive has substantial advantages that explain its widespread predominance over all other configurations. It permits great product flexibility of design, with the maximum of interchangeability between products of different character, especially with regard to the flexibility of power-plant usage, with moderate cost and weight penalties for the flexibility to install very large engines in specialty models of small cars. For example, most of the U.S.-built compacts are adapted to receive engines of up to 460 ci for special models, which were more popular a few years ago than at present. There is still a good market for big V-8 engines in relatively small cars. Such product flexibility is very difficult to obtain with the unit powertrain design, although one could visualize a Company line of various sized unit powertrains that could be interchangeably installed on a range of vehicle sizes. We are a long way from such a situation in this country today, and even with the admitted advantages of the front-drive, it is not likely they will be considered sufficient to justify the enormous investments necessary to tool up for them.

The front-engine, rear-drive enjoys most of the impact advantage of the front drive, but it suffers from a package handicap resulting from the presence of the transmission and driveline and final drive underneath the car. Any rear-drive car, but especially the front-engine rear-drive, which has so much of its unloaded weight on the front wheels, suffers from the fact that it loses directional stability when the driving wheels slip, and the front-engine rear-drive is the worst because of the small weight on the rear wheels. This also constitutes a traction disadvantage in slippery driving.

Front-engine rear-drive vehicles in general offer better accessibility to the engine and drivetrain for repair than the integral powertrain designs. The governing reason for their predominance in the domestic industry, however, is the constraint of capital investment and the difficulty of putting radically different designs into production.

The relative advantages of the three vehicle configurations are very much tied into rear-suspension design, and will be discussed further under that heading.

2.3.2 Construction Methods

The chassis frame was an indispensable part of early cars, which were intended to be equipped with any kind of body the customer chose to have built for him. The bodies had no structural integrity of their own, being made with light wooden frames and most of them being entirely unroofed. The bodies were bolted firmly to the frame, which performed the structural functions and was the part to which all chassis components were fastened. The first steel bodies were bolted to the frame the same way, and as long as the open car persisted, the frame was a necessity. In the 1930's the first designs to appear without provision for open bodies made it possible to recognize the inherent structural integrity of the closed body, and to use a frame of reduced weight permanently welded to the body as in the Citroen "Traction Avant" and the Lincoln Zephyr. Many other unitized bodies followed on both sides of the Atlantic, but in all of them the rudimentary frame persisted, and even lightweight Renault Dauphine studied here had its frame, longitudinal rails and cross-members.

The Falcon of 1960 was the first U.S.-built car to make full use of the inherent stiffness of the body structure itself, incorporating little more than front-end structure plus sufficient added structure for attachment of chassis parts. No lighter car for its size has been built since. Its lightness derived not only from its light body and structure but from meticulous design, a weight-control procedure modeled on the usual financial control procedures, and on advanced foundry methods that permitted thinner sections of cast iron. It used substantially no light materials.

In 1960 body-frame cars and unitized cars were more alike functionally than they are today. The concept of body-frame cars, as built today only by General Motors and Ford, has been greatly refined since 1960. It depends on soft rubber mountings between frame and body to absorb road and powertrain noise before it is transmitted to the body. Proportioning and disposing of these mountings is both a science and a black art, but the result is an unprecedented degree of quietness and smoothness on reasonably smooth roads.

Since the body is so much stiffer than any frame could possibly be, because of its large moments of inertia in all dimensions, whenever a body and frame are mounted together, it is the body that resists the stress. A body must possess a certain amount of stiffness to avoid its having vibration periods within the range of wheel-rotation frequencies, so the result of these facts is that a body weighs about the same whether the car is unitized or body-frame. The frame weighs more than the needed structural parts added to make a unitized design, and this weight difference is about the weight difference between unitized and body-frame designs.

We give an example, the 1971 and 1972 Torino hardtop (Table 2N). The former was a unitized design, the latter a body-frame design, 1" longer and 3" wider -- the additional width being only a matter of sheet-metal contours, the basic body structure being the same. Of the 304 lb difference between the weights of the two cars, we attribute 221 lbs to the body-frame design, in accordance with the following, which does not include the secondary weight penalties that accompany weight increases.

This is not a comparison between equivalent cars, since no allowance has been entered in the following table's balance sheet for the superior noise-vibration characteristics of the 1972 car. We do not venture to propose any such allowance, since the functional qualities of unitized and body-frame cars are sufficiently different that it is difficult to evaluate equivalence. The unitized vehicle is more difficult to insulate from noise and vibration, while the body-frame vehicle is more difficult to control during rough-road shake. Both types, furthermore, are still under development, recent examples of both systems revealing substantial improvements in functional qualities. In neither instance is any weight-saving potential combined with functional improvement.

TABLE 2N. WEIGHTS, 4-DOOR HARDTOP, TORINO

	1971	1972	
Curb weight	3395	3699	304
More efficient rear suspension design	87	65	25
Seats & tracks	132	145	(13)
Bumpers	70	84	(14)
Brakes	140	176	(36)
Wheels & tires	209	216	(7)
Exhaust	44	48	(4)
Engine	557	561	(4)
Instruments & horn	5	7	(2)
Instrument cluster	4	7	(3)
Estimated 1" length			(10)
Estimated 3" width			(10)
Interior trim	58	63	(5)
Body-frame over unitized			221

There are intermediate varieties of structural configuration. Chrysler, in its larger cars, uses the front half of a frame bolted to the body. Chevrolet, in the Chevy II, used a bolt-on front end, and the Volkswagen Beetle has the body platform and chassis parts integral, with the upper body separable. It may be said of these designs that they add weight to the unitized design without conferring on it any of the advantages of the full body-frame configuration. They are often adopted to simplify production in a facility designed for assembling body-frame cars, or for other reasons related to production costs.

There are also vehicles, such as the Mercedes-Benz 450 and the Cadillac Seville, which are basically unitized designs having portions of a frame in the form of a front sub-frame or isolated front cross-member to assist in reducing the transmission of powerplant and road noise and vibration into the vehicle structure.

2.3.3 Engine Configurations

This subject is considered at greater length in Section 4 of this study. We will consider here more briefly the three types of engine in common use today, the 90° V-8, the in-line 6 and the in-line 4. As now used, these engines do not overlap in displacement range. There is no present production 4 larger than 146 ci (2392 cc), and there is no present production 6 smaller than 162 ci (2655 cc). There is no present production 6 (for passenger-car use) larger than 258 ci (4228 cc) and no V-8 smaller than 262 ci (4293 cc). Thus, if one is to conform to present practice, the engine configuration is chosen when the displacement has been determined. Since there are no overlapping displacements, there is no good basis on which to decide which configuration is the most weight-efficient. On a purely dimensional basis one can say that large cylinders are more weight efficient than small ones, since their volume is a function of the cube of their dimension, while their weight tends to be a function of the square of the dimension to whatever extent minimum sections are dictated by manufacturing conditions rather than stress levels.

The curve in Figure 2F bears this out, with larger displacement engines showing the smallest weight per unit displacement, and with no significant systematic difference between 4, 6, and 8 cylinder engines other than the effects of size.

Functionally, there is no contest. The V-8 is head and shoulders above other forms by virtue of its absence of torsional inertia effects, its very low primary inertial bearing loads, its overlapping power impulses, and its short, rigid structure. The in-line 6 is excellent kinematically, but its

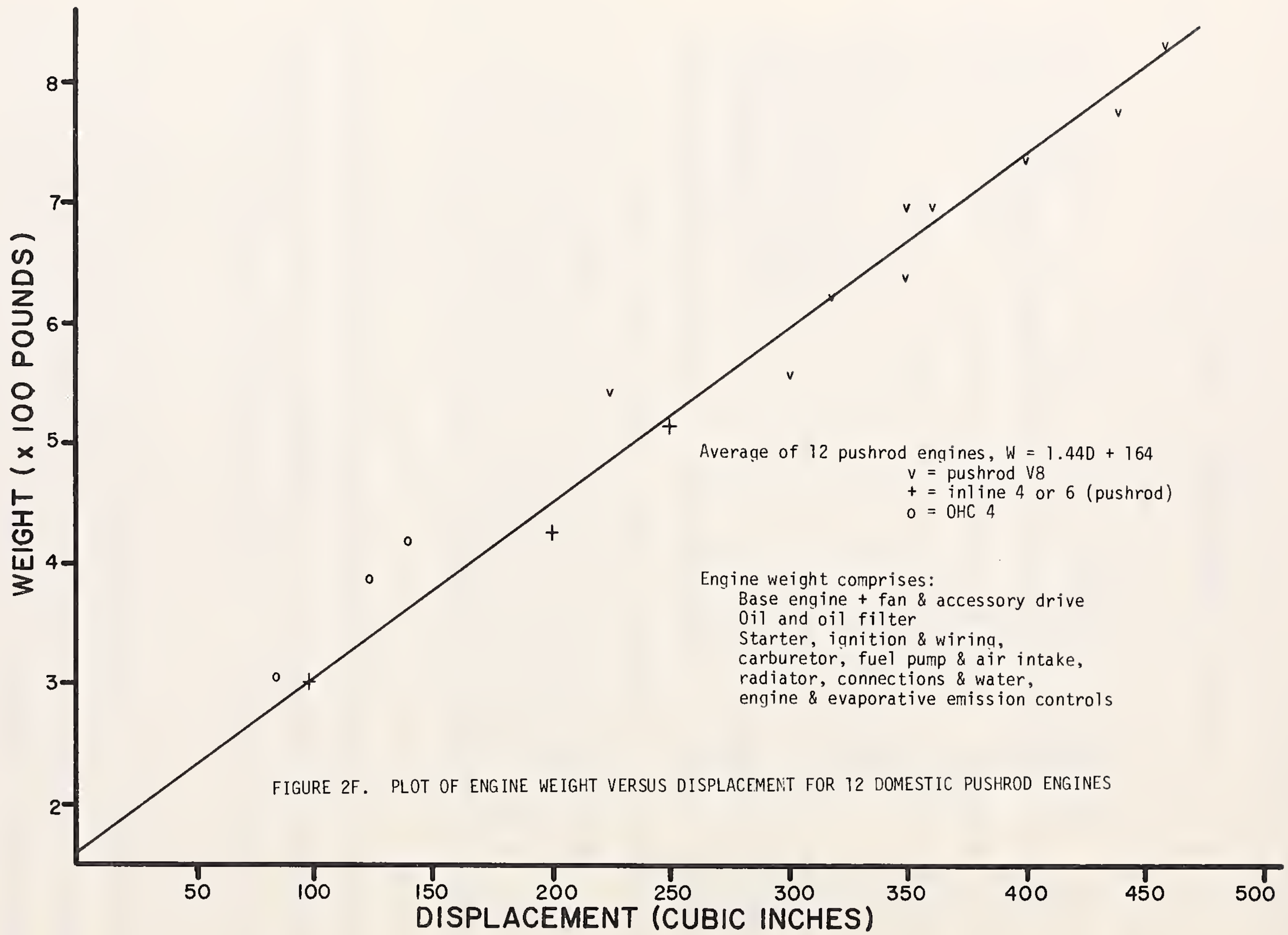


FIGURE 2F. PLOT OF ENGINE WEIGHT VERSUS DISPLACEMENT FOR 12 DOMESTIC PUSHROD ENGINES length, and the substantial bending moments due to inertia forces, bring with them resonant bending vibrations inside the range of normal operation. The in-line 4 has severe secondary reciprocating and torsional unbalance, which can be compensated at some extra cost and weight, and non-overlapping power impulses which present torsional problems that are a function of load and speed (see also 4.4).

2.3.4 Suspensions

The subject of suspension design is inextricably tied in with the important question of stability. We cannot deal with that adequately here, but some fundamental considerations of stability are essential to any understanding of the basic function of suspensions to maintain positive stability.

The essential condition of positive stability requires that whenever the car is subjected to a lateral force, as in the case of centrifugal force in rounding a turn, that the car tends to turn in that direction. In other words, whenever it is turning, the stable car tends to straighten itself out and go straight. A car can, of course, be so stable it is difficult to persuade it to turn, and this is not desirable. However, it is desirable that the car, when left to itself, will tend to run in a straight line.

In order to comply with this condition, it is required that the front end of the car be more yielding to the side force than the rear end. This establishes the difference between the functions of a front and a rear suspension: the rear suspension is called upon to resist side thrusts more than the front suspension does.

2.3.4.1 Front Suspensions

Independently sprung, as opposed to axle-mounted, front wheels are at least as old as Amédée Bollée's steam coach of 1875. Individual cars appeared with it during all stages of evolution. However, independent front-wheel suspension requires more elaborate steering linkages, and transmits more road noise than an axle suspension, and its gains were not very convincing during the day of the high-pressure tire. When low-pressure tires came into use, the compliance of the tire became a consideration in suspension dynamics, and many forms of instability began to occur that hadn't been very bothersome up until then. The very worst of these was "shimmy", a catastrophic vibration of the front wheels about the steering-pivot axis, coincidental with a violent oscillation of the axle in roll deflection. This could be so violent at its resonant period that the car would become uncontrollable, and it was a severe problem that brought forth a surprising variety of remedies, none of which was entirely effective with the surviving front axle. The problem was, of course, that the axle couples the wheels to parallel planes of rotation during movement about the roll axis, while the steering linkage couples them to parallel planes during movements about the steering axis, and these two couplings make an ideal feedback mechanism for a gyroscopic oscillator. The condition was aggravated by the rudimentary state of wheel-balancing at the time.

The remedy was to decouple the planes of rotation about the roll axis by eliminating the axle and causing the wheels to remain more nearly parallel to their rest position as they moved up and down in jounce and rebound. About the same time it was realized that very soft springs in front would eliminate the uncomfortable rear-seat jerk that was caused by excessive pitching oscillations. By 1934 most U.S. and European makes, except the English, had finally made the plunge, and eliminated the front axle. The few holdouts made similar ride improvements by greatly decreasing front spring rates and adding roll stabilizers.

The more heavily it is loaded, the less a tire is capable of resisting side thrust. Consequently, for a car to be stable directionally, the front wheels must be more heavily loaded than the rear, or carry less air pressure, or be equipped with smaller tires. Otherwise the effect is the same as putting the feathers on the front of the arrow, and the car will seek to switch ends. Furthermore, in turning, if the car is to remain stable, it is necessary to insure that the outside front wheel will be more heavily loaded by the rolling moment than the outside rear wheel. For this reason, roll stabilizers are ordinarily applied to the front suspension to give additional stiffness in roll deflection.

Thus, we have the universal application of independent front-wheel suspension to passenger cars. Not everything is gain. When the front wheel and its associated unsprung weight are accelerated in normal ride deflection, because the accelerating force does not pass through the center of gravity of the moving system, there are considerable moments to be resisted. The axle was able to resist these moments and pass them harmlessly to the opposite wheel, but with independent suspension they are directly transmitted to the structure of the vehicle, and present a noise problem that wasn't present with the front axle. A second cause of ride disturbance, with both jerks and noise, arises from the flexibility of the tire. When the tire is deflected by striking a bump, its rolling radius is momentarily decreased, and its speed of rotation must consequently increase accordingly. The inertia of the wheel and tire being considerable, this requires a sharp accelerating force to be transmitted to the tire by the ground, and the reaction from this force accelerates the wheel-hub sharply to the rear. It is therefore necessary for the wheel hub to have a degree of fore-and-aft compliance if the shock and noise are to be avoided. Since it is almost the function of the front suspension to support the system while the brakes are applied, it may be seen that these two requirements are in conflict.

Braking imposes further requirements on the front suspension. When the brakes are applied, the retarding force is at the level of the ground, while the inertia of the car resides at its center of gravity, a couple of feet above the ground. The resulting moment tending to lift the rear of the car and push down on the front is familiar to most drivers. This moment has the effect of increasing the adhesion of the front wheels and decreasing that of the rear wheels, with the result that the front wheels are ordinarily called upon to do 65-70% of the total braking effort on a maximum application. It also means that in the case of the usual softly sprung car there is an appreciable time interval between the instant of brake application and the time at which the car has assumed a sufficiently nose-down attitude to accomplish the necessary transfer of weight from rear to front. If the brake has been suddenly applied, the front wheels are likely to lock up under these conditions.

The remedy for the undesirable nose-down attitude and the resulting delay in achieving a steady-state condition of weight transfer is called anti-dive, and it is accomplished by arranging the suspension linkage so that some moment or force component from the brake reaction tends to lift the front end by an amount equal to its tendency to dive.

We have remarked how it happens that the front suspension carries the major responsibility for preventing excessive roll. In addition to the roll stabilizer, which is simply a transverse torsion rod that resists deflection of one side of the suspension relative to the other, an additional technique for resisting roll deflection is to raise the roll center by causing the point of contact between the ground and the outer wheel to move outward as the wheel moves upward in jounce deflection. Both the stabilizer and the raised roll center increase the transmission of road harshness into the car, in addition to which the raising of the roll center increases the reaction of ride deflections on the steering. So roll stiffness and ride comfort are conflicting requirements to which compromise solutions must be sought.

Anti-dive in front suspensions is likewise incompatible with ride softness since it requires that the point of contact between wheel and ground move forward if the wheel is moved vertically upward in jounce with the brakes locked, and the softness demands that the wheel move backward rather freely. In Section 4 of this report, we discuss the weight implications of various suspension designs, and conclude that the MacPherson type front end is fundamentally lighter than the prevailing SLA type. It is, however, less well adapted to providing anti-dive without sacrifice of ride comfort. If, however, anti-dive is incorporated in a MacPherson front end on a front-drive car, it also provides "anti-squat" which is the prevention of the contrary, nose-up moment during acceleration.

We therefore conclude that the MacPherson is the lightest and cheapest front suspension, but that the best functional qualities are provided by the SLA suspension, which is better adapted to an isolated cross-member, sub-frame or frame that will isolate its noise and harshness more effectively.

2.3.4.2 Rear Suspensions

The function of the rear suspension is decidedly different from that of the front suspension. The rear suspension is the foundation on which the car's stability and controllability are erected, and an axle, which keeps both rear wheels always pointed in the same direction and virtually normal to the ground under all conditions of jounce and roll, is an excellent starting point. The simplest rear axle suspension for rear-drive cars is the familiar live axle that the great majority of rear-drive cars have. The heavy weight of this axle, and the fact that much of its weight is concentrated in the heavy drive unit in the middle of it, combined with the facts that it is subjected to the torque reaction of both driving and braking, creates a host of problems that require compromises between comfort and adequate stability. Rear suspension stability is more important than front-suspension stability, just because when rear wheel adhesion is lost, the necessary condition for stability is also lost. When the axle is subjected to driving or braking torque on a rough road, there is a variety of unstable modes of oscillation that are present, amplified by the mass of the final-drive unit in the middle, which increases the translational mass of the unit without a corresponding increase in its rotational inertia. These oscillatory modes limit driving speeds on rough roads and tend to destroy stability when braking on rough roads. Their limitation within reasonable limits by damping requires reductions in ride comfort.

The live rear axle has its compensations, particularly for the vehicle that is judged, as the domestic car is likely to be, by its smoothness and softness on a relatively good surface. It has the virtue shared by all axles, of minimizing the transmission of acceleration moments to the vehicle, and its high unsprung weight renders it relatively insensitive to the higher-frequency components of these vertical accelerations, forcing the tires to absorb a greater proportion of the noise and harshness.

There are advocates, such as Mercedes-Benz, BMW and Datsun, of independently sprung rear wheels. These, it may be said, lack both the disadvantages and the advantages of the axle suspension that we have just reviewed. It is not possible to have the rear wheels remain normal to the road in both jounce and roll deflection, ordinarily a compromise is chosen, with the axis of their motion neither normal nor parallel to the car's center line. It is likewise impossible to avoid completely some steering deflection with jounce, with the result that the car's rear wheels are not properly aligned except at one specific height. Consequently the independently-sprung rear suspension must have either very stiff springs or some self-levelling attachment to avoid large changes in ride height with load change.

Independent rear suspension presents other problems that demand extra cost and weight for the proper solution. The final drive becomes a problem, since it requires to be isolated from the chassis to prevent the transmission of gear noise, and yet the means whereby it is suspended must be prepared to resist full driving torque. When the entire rear suspension is mounted on an isolated subframe the results are excellent, but it is heavy and expensive.

The ideal suspension for the rear-drive car is also heavy and expensive. It is the deDion rear suspension, which uses a rear axle, but which carries the final drive separately on the frame. It provides the substantial advantages of the axle, without its high unsprung weight and heavy central mass. However, it still faces the problem of mounting the final drive, and the expense of two universally-jointed shafts to connect the sprung final drive unit to the unsprung driving wheels.

The front-drive car bypasses these rear suspension problems, and the fact that it can possess the best possible rear suspension, which is a simple axle quite free of the complexities of the driving mechanism, is one of its outstanding virtues. It does involve the additional complexity of the universally-jointed drive shafts, mentioned above, but it does not combine these with the problems that arise from the propeller shaft and the need to mount the final drive. The final drive is mounted directly on the engine-transmission assembly, which is required to be isolated from the rest of the car no matter what its drive system may be.

Front drive demands nothing of the front suspension that it is not already required to supply because of its brakes, except the additional beef to carry the increased weight. Since the front suspension carries relatively little of the car's variable load, problems are not encountered as the result of wide variations in suspension height with load variation. Front-drive offers some potential weight saving over rear drive by virtue of its more economical housing of the final drive, but this is probably overcome by the added weight of the propeller shafts and universal joints, so we regard front and rear drives as substantially a wash. We note from Fig. 1C a significant roominess/weight advantage for front-drive cars, especially for those with transverse engines. This is inherent in the front-drive to a certain extent, in that the floor-level is lower than on rear-drive cars, but it is perhaps to a greater extent a reflection of the fact that these transverse-engine front-drives are the newest designs on the road,and they are distinguished by a lack of any compromise of roominess for styling reasons.

Further design considerations for front and rear-drive comparisons will be found in Section 4 of this report.

2.4 WEIGHT-SAVING MATERIALS

Aluminum has been used in automobiles since the beginning. Bodies were first made with wooden panels over wood frames, by carriage builders, but metal panels early replaced wood, and since aluminum is easier to form by hand methods than steel, it began to be used by body builders who used hand-forming methods, and it is so used today, with aluminum framework. However, there is no history of aluminum bodies made with mass-production methods.

Aluminum has also been used in engines since the first decade of the century. When cylinders were separate from crankcases, the latter were often made of aluminum for weight saving. During World War I aircraft engines (Hispano Suiza) were in use with aluminum crankcase and waterjackets with dry steel sleeve cylinders. Immediately post-WWI Curtiss aircraft engines used wet steel sleeves screwed into iron heads.

In the early 1920's a number of all-aluminum cars were made by the Aluminum Corporation of America. There were two designs, a four-cylinder and a six cylinder model, and these vehicles were driven a great number of miles during the 1920's. One of them survives in the Ford Museum in Dearborn. They were designed very conservatively, however, and very little work was done to realize the full potential of aluminum for weight reduction. The result was a weight saving of only about 18%, which was regarded as disappointing and not warranting the extra cost, so the project was not carried forward.

These cars used steel for highly stressed parts, but were said to be 85% aluminum with bodies that made extensive use of castings for structure. (See reference 2-1.)

General Motors built a 215 ci aluminum V-8 in the early 1960's with wet sleeves, but no weight information of sufficiently detailed nature is available to form accurate weight comparisons. It was just about this time that Ford introduced cast iron engines of thinner-wall construction due to advanced foundry methods which greatly reduced the weight advantage of the aluminum-block engines and made it possible to build larger displacements within the same basic engine structure. The aluminum engines were not continued. The 4-cylinder Vega engine was the first automobile engine to use aluminum block without iron sleeves. The engine gave trouble for reasons that were not all connected with the aluminum engine or its cylinder walls, but at this juncture we believe it unlikely to be repeated.

Aluminum is indispensible as a piston material, for which it is ideally suited by its low density and high heat conductivity, and it is universally used for that purpose. It is also growing in its application to transmission housings. Present-day use of aluminum is covered in greater detail in Section 5 of this report.

Fiberglass-reinforced thermoplastic resins were introduced about 1950 as a material for car bodies, and the Chevrolet Corvette, in 1953, was the first production version of such a body. It is especially valuable as a material for low-volume products for which tooling costs for a metal body are not justified. The Corvette has continued in small-scale production and the Studebaker Avanti was produced briefly by similar materials and methods briefly in the 1960's.

Numerous truck bodies utilize fiberglass-reinforced polyester materials and also a foam-backed butadiene-styrene material. Current interest in plastic materials, especially for outside panels subject to impact, is discussed in Section 5 of this report.

Some experimental work was done in the 1960's with butadiene-styrene materials for hoods and deck lids, but difficulties were encountered with the widely differing expansion coefficients in extreme temperature conditions.

3. DEVELOPMENT OF INFLUENCE COEFFICIENTS AND WEIGHT-CONSCIOUS DESIGNS

3.0 INTRODUCTION

It has already been stated that significant vehicle weight reductions can be achieved by making design decisions that decrease the vehicle's non-functional size. In order to evaluate the magnitude of these potential weight reductions, it is first necessary to analyze existing designs to determine the weights of different vehicle components or systems.

It is well known that the different vehicle system weights are not independent of one another. For example, if a heavier engine or body is included, a heavier suspension system will be required. In this section a vehicle weight analysis will be used as the basis for development of influence coefficients which relate a weight change in a vehicle subsystem to the total vehicle weight change that could result.

The influence coefficients developed here will be used, along with the results of the vehicle weight analysis, in the synthesis of weight-conscious designs based on conventional configurations and materials. Weight-conscious designs will be developed for the 4-, 5-, and 6-passenger size classes discussed in Section 1.3.

3.1 WEIGHT CLASSIFICATION

3.1.1 Weight Classification Method

The large number of components that make up the automobile of today may be collected into several vehicle subsystems or component groups. Although each automobile manufacturer uses some classification scheme to group vehicle components into subsystems, the classification schemes differ among manufacturers and sometimes vary from year to year. This makes any industry-wide comparison of vehicle system weights somewhat difficult.

For the purposes of this study the subsystem classifications shown in Table 3A will be used. In the analysis of the weights of a large number of vehicles it has been noted that some subsystem weights are relatively independent of the weights of other subsystems, but are dependent instead on the basic character of the car. Other subsystem weights are primarily determined by the maximum gross weight of the vehicle, while still others are dependent primarily on the power or displacement of the vehicle powerplant. Because of this, the vehicle subsystem weights have been further combined into three categories: product-dependent weight, weight-dependent weight and displacement- (or power-) dependent weight.

3.1.2 Product-Dependent Weight

The subsystems that contribute to the product-dependent weight are those whose weight is controlled by the functional size, functional characteristics such as comfort and noise/vibration control, safety requirements, and styling. The product-dependent weight is effectively determined when the basic character of the vehicle is originally established, and is independent of either vehicle weight or power.

The composition of the product-dependent weight, designated W1, is given in Table 3B. It should be noted that the subsystem weights are those for the standard body style, without options. The options are included, along with the weight difference between the standard model and the heaviest body style, in category W4, the disposable weight.

3.1.3 Weight-Dependent Weight

The weight-dependent weight is assumed to be a direct function of the maximum gross vehicle weight. The subsystems that contribute to the weight-dependent weight, designated W2, are given in Table 3B. Note that, in contrast to the product-dependent category, the weight-dependent weight is

TABLE 3A. VEHICLE SUBSYSTEMS (COMPONENT GROUPS)

Subsystem	Components Included in Subsystem
Body	Body structure, doors, front end sheet metal, body electrical, interior and exterior trim, paint, glass, all other body components not listed in other systems
Bumpers	Front and rear bumpers, energy absorbing units
Frame and tools	Frame, engine mounts, tools, jacks
Engine	Basic engine, lubricants and filter, carburetor, air cleaner, emission control, starter, ignition system, cooling system, coolant
Suspension	Front and rear suspensions, shock absorbers and mounting brackets
Wheels and tires	Wheels, tires
Final drive	Drive shaft, axle, differential lubricants
Brakes	Front and rear brakes and backplates, drums or disks, brake lines, brake fluid, parking brake, power brake booster, master cylinder, skid control
Transmission	Transmission (manual or automatic), transmission controls, torque converter, clutch and controls
Exhaust	Exhaust pipes, catalytic converter, muffler
Fuel	Fuel tank, fuel lines, throttle control, gasoline
Steering	Steering linkage and gear with lubricant, steering wheel and column, power steering unit and pump with fluid
Heating and ventilating	Heater, body ventilation, air conditioner
Instruments	Instrument cluster, panel lighting, fuel gauge and sender, speedometer and cable, clock, engine operation gauges, horn
Battery and alternator	Battery, alternator and regulator
Electrical distribution	Wiring and miscellaneous electrical devices

TABLE 3B. WEIGHT CLASSIFICATION DEFINITIONS

1. Independent variables, specified weights

   Product-Dependent Weight, W1
   - Body
   - Frame and Tools
   - Instruments and Electrical
   - 80% of Battery and Alternator
   - 50% of Bumpers
   - 50% of Exhaust System
   - 50% of Steering System
   - 20% of Fuel System
   - Constant, n, of powerplant weight Wp
   - Heating and Ventilating

   Disposable Weight, W4
   - Full load of passengers (@150 lbs/passenger)
   - Luggage (@25 lbs/passenger)
   - All available options
   - Incremental weight of heaviest available body style over that of standard model

   Performance Load, Wx
   - Vehicle load specified for performance evaluation = 300 lbs.

2. Dependent variables, derived weight categories

   Weight-Dependent Weight, W2
   - Suspension
   - Wheels and Tires
   - Brakes
   - 80% of Final Drive
   - 50% of Bumpers
   - 50% of Steering System
   - 50% of Transmission

   Powerplant Weight, Wp
   - Engine
   - 80% of Fuel System
   - 50% of Exhaust System
   - 20% of Final Drive
   - 20% of Battery and Alternator
   - 50% of Transmission

   Displacement-Dependent Weight, W3
   - The portion of the powerplant weight that is dependent on engine power or displacement. W3 = Wp - n

   Curb Weight, Wc
   - The weight of the standard vehicle with liquids but without disposable weight. Wc = W1 + W2 + W3

   Maximum Gross Vehicle Weight, Wg
   - Curb weight plus disposable weight. Wg = W1 + W2 + W3 + W4

a dependent variable, with the size and weight of components in this category being determined by the weights of all other vehicle components.

3.1.4 Displacement-Dependent Weight

The displacement- or power-dependent weight, designated W3, is the portion of the weight of the powerplant and related components, which is a direct function of the engine displacement. It is assumed that engine power is proportional to engine displacement.

As will be shown in Section 3.3.3, analysis of the weights of many vehicles has shown that a portion, n, of the powerplant weight (as defined in Table 3B) is independent of displacement, with the remainder being approximately directly porportional to engine displacement. Thus, if Wp is the powerplant weight,

$$Wp = W3 + n$$

and

$$W3 = m\ D$$

where m and n are constants

and D is the engine displacement

The weight n is included in W1, the product-dependent weight since it is independent of both vehicle weight and engine displacement.

3.2 DERIVATION OF INFLUENCE COEFFICIENTS

Whenever the weight of one of the components contributing to W1 is decreased, as would occur in an automobile weight reduction program, secondary weight reductions can occur in the supporting components (W2). Similarly, if a less powerful engine is used, the powerplant weight would decrease and this would enable further reductions in the weight-dependent components. Thus, any weight change brought about by a decrease in either W1, W4, or D will allow an even greater reduction in curb weight Wc. This assumes, of course, that the weight-dependent components are redesigned to meet the new.

less-stringent, load carrying requirements. In order to predict the magnitude of the curb weight reduction resulting from a change in either W1, W4, or D it is necessary to develop influence coefficients, which are defined as the resultant change of vehicle weight as a consequence of a unit change in one of the independent variables.

Before deriving the influence coefficients it is necessary to define several quantities:

Weight per unit displacement, p

$$p = \frac{Wc+Wx}{D} = \frac{W1+W2+W3+Wx}{D}$$

The value of p normally varies from vehicle to vehicle and is determined by choice of engine displacement, D. When either p or D is independently specified, the other becomes a dependent variable.

Weight-dependent factor, c

$$c = \frac{W2}{W1+W3+W4}$$

This constant determines the variation in W2 when either W1, W3, or W4 is changed. It is presumed that the value of c is the same for all vehicles and an analysis of existing designs proves this to be approximately true.

Using the above definitions and the definitions of Table 3B, it is easy to derive the following equations.

$$D = \frac{Wc+Wx}{p} = \frac{W1+W2+Wx}{p-m}$$

$$W3 = \frac{m(W1+W2+Wx)}{p-m}$$

$$W2 = \frac{cp(W1+W4)-cm(W4-Wx)}{p-m-cm}$$

$$Wc = \frac{(1+c)(pW1+mWx)+cpW4}{p-m-mc}$$

The influence coefficients relating a change in Wc to a unit change in W1, W4, or D are now found to be

$$Q1 \equiv \frac{\partial Wc}{\partial W1} = \frac{p(1+c)}{p-m-cm}$$

$$Q4 \equiv \frac{\partial Wc}{\partial W4} = \frac{pc}{p-m-cm}$$

$$QD \equiv \frac{\partial Wc}{\partial D} = m(1+c)$$

It is presumed that Q1, Q4, and QD are all approximately constant for all vehicles and a comparison of existing designs (Section 3.3.4) bears this out.

3.3 WEIGHT ANALYSIS OF EXISTING DESIGNS

3.3.1 Availability of Weight Information

One of the most serious constraints imposed on this analysis of automobile weights is the difficulty in obtaining sufficiently detailed weight breakdowns of recently produced cars. Weight data are available in which weights of major subsystems, such as body, frame, engine, etc., are given. Although the total weights of these subsystems may add up to the published curb weight for the given car, they are not broken down into sufficient detail to allow adequate comparisons between different vehicles. As mentioned earlier, each automaker has his own system of classification and these classification schemes frequently change from year to year.

For example, it might seem that one maker's engines are much heavier than those of another maker. Yet there is no way to know what is included under the term, "engine". Is the radiator included, or is it a part of the body? Does the engine figure include the starter and the alternator, or are these part of the electrical system? Is the carburetor included with the engine, or is it part of the fuel system? Are the front fenders part of the body, or are they listed under "chassis"? Are the brake backing plates included with the axle, or with the brakes? What is meant by a listing for "front axle" on a car that has independent front suspension, with no axle?

Because of the above-mentioned difficulties the weight data presented in the next section may not be completely fault-free. Every effort was made, however, to adjust all weight breakdowns in accordance with the component groups given in Table 3A.

The sparsity of data in the next section for foreign-made cars and for cars from some American producers stems mainly from the unavailability of that data from authoritative sources. In addition, the available weight figures for European-built cars are generally given for the European versions, which differ substantially, especially in the weights of the bumper and engine subsystems, from the versions built for sale in the U.S.

3.3.2 Weight Comparisons

Detailed weight information was obtained for about 25 passenger cars produced for the U.S. market within the past six years. The data for 17 of these vehicles were analyzed in depth and are listed in Table 3C. The data for the remaining vehicles either were deficient in some respect or were too similar to data for one of the other cars to provide any additional information. It is felt that the 17 vehicles included in the study constitute a representative sampling. Four of the vehicles fit in the 4-passenger class, seven are 5-passenger cars, and six are in the 6-passenger class.

The weight information was in some cases (cars 1-6, 9-12) obtained from the vehicle manufacturer, and in other cases (cars 7, 8, 13-15, 17) was provided by the DOT Transportation Systems Center. The data for car 16 (Fiat 128) were obtained at the Thayer School by disassembly and weighing of the subject vehicle components. In all cases the weights are for a 4-door version with standard engine and no optional equipment.

TABLE 3C. WEIGHT BREAKDOWNS FOR SEVENTEEN ACTUAL VEHICLES (ALL WEIGHTS GIVEN IN POUNDS)

	VEHICLE																
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
BODY	1518	1576	1652	1220	1273	1266	1668	2168	1418	1469	2112	1809	1190	1310	1101	837	760
FRAME & TOOLS	305	358	375	25	22	21	10	12	10	11	12	12	30	120	9	8	8
BUMPERS	112	108	246	37	36	171	147	230	173	151	208	216	120	150	42	55	63
SUSPENSION	210	184	192	151	169	171	220	272	164	225	225	225	130	260	186	110	112
WHEELS & TIRES	219	245	241	156	174	179	200	265	169	174	264	252	160	240	165	130	147
DRIVE AXLE	173	172	177	117	139	139	168	193	126	164	176	169	120	170	50	39	51
TRANSMISSION	159	204	203	117	118	120	165	173	135	140	175	160	100	90	80	85	64
BRAKES	152	154	208	108	114	135	158	207	149	163	218	203	100	150	110	57	41
EXHAUST	56	54	53	28	29	29	44	68	33	48	73	70	40	40	29	25	21
FUEL	168	161	159	126	125	125	144	192	120	135	191	192	110	140	112	80	118
STEERING	64	100	100	58	58	60	48	77	45	48	73	70	40	60	35	30	24
HEATING & VENT	30	30	30	27	22	21	22	22	21	21	23	22	10	20	22	30	30
BATTERY & ALT	49	56	47	48	50	44	50	58	45	45	50	49	40	40	33	47	50
ELECTRICAL	29	28	27	14	14	17	30	32	23	25	31	30	20	20	16	20	22
ENGINE	540	662	693	427	430	430	698	740	527	588	648	590	390	590	310	270	319
W1-N	2056	2183	2338	1396	1444	1503	1905	2488	1644	1699	2418	2114	1392	1643	1240	989	923
W2	887	927	1057	614	675	772	892	1139	759	863	1076	1038	616	936	579	413	416
WP	841	983	1007	647	657	654	977	1082	755	845	985	917	592	821	481	420	491
W4	1700	1700	1791	1380	1380	1414	1750	1900	1580	1580	1800	1625	1043	1482	1000	950	950
WX	300	300	300	300	300	300	300	300	300	300	300	300	300	300	300	300	300
WC	3784	4093	4403	2657	2776	2929	3774	4709	3158	3407	4479	4069	2600	3400	2300	1823	1830
WG	5484	5793	6194	4037	4156	4343	5524	6609	4738	4987	6279	5694	3643	4882	3300	2773	2780

VEHICLES STUDIED

1. 1970 FORD CUSTOM 500
2. 1972 FORD CUSTOM 500
3. 1974 FORD GALAXIE 500
4. 1970 MAVERICK
5. 1972 MAVERICK
6. 1974 MAVERICK
7. 1974 PLYMOUTH SATELLITE
8. 1974 CHRYSLER NEW YORKER
9. 1976 VALIANT
10. 1976 PLYMOUTH VOLARE
11. 1976 PLYMOUTH GRAN FURY
12. 1976 CHARGER SE
13. 1975 VEGA
14. 1975 NOVA
15. C6
16. 1972 FIAT 128
17. 1975 VW RABBIT

The weight breakdowns given in Table 3C are in accordance with the classification scheme discussed in Section 3.1.1. In some instances the original weight information had been classified using a different scheme, and this necessitated a modification of the weight classification before analysis for those vehicles. Although every attempt was made to accurately fit the available data to the chosen classification scheme, lack of detailed weight information may have led to some small inaccuracies in the data presented in Table 3C.

Also given in Table 3C are the values determined in the weight analysis for W1-n, W2, Wp, W4, Wc and Wg, as defined in Section 3.1.

3.3.3 Derived Constants

It has been assumed that the three constants, m, n, and c, are the same for all vehicles. One purpose of this weight analysis is the determination of those constants and a check on the validity of the assumption of their constancy.

In Figure 3A the weight Wg-W2 (=W1+W3+W4) is plotted vs. W2 (weight dependent weight) for each of the 17 vehicles. It can be seen from the plot that the data fall very nearly on a straight line, proving the validity of the assumed equation

$$W2 = c(W1+W3+W4)$$

The mean value of the constant c (the slope of the straight line) was found by statistical analysis to be

$$c = 0.201$$

with a standard deviation of .016.

In order to determine the values of m and n it is necessary to plot the powerplant weight Wp vs. engine displacement D. This is done in Figure 3B. The engine displacement values used in the plot were those for the

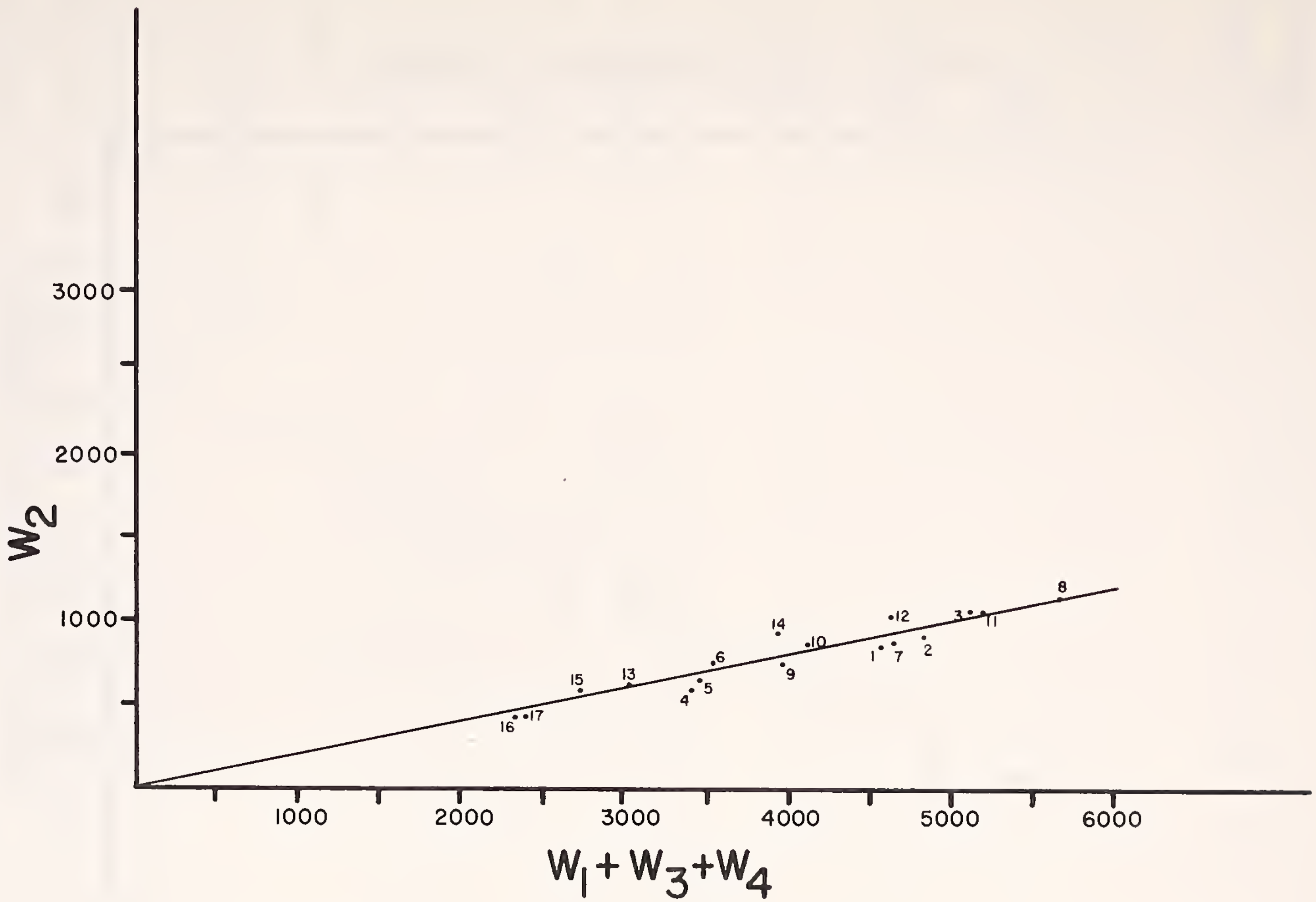


FIGURE 3A. WEIGHT-DEPENDENT WEIGHT (W_2) VS. WEIGHT ($W_1 + W_3 + W_4$)

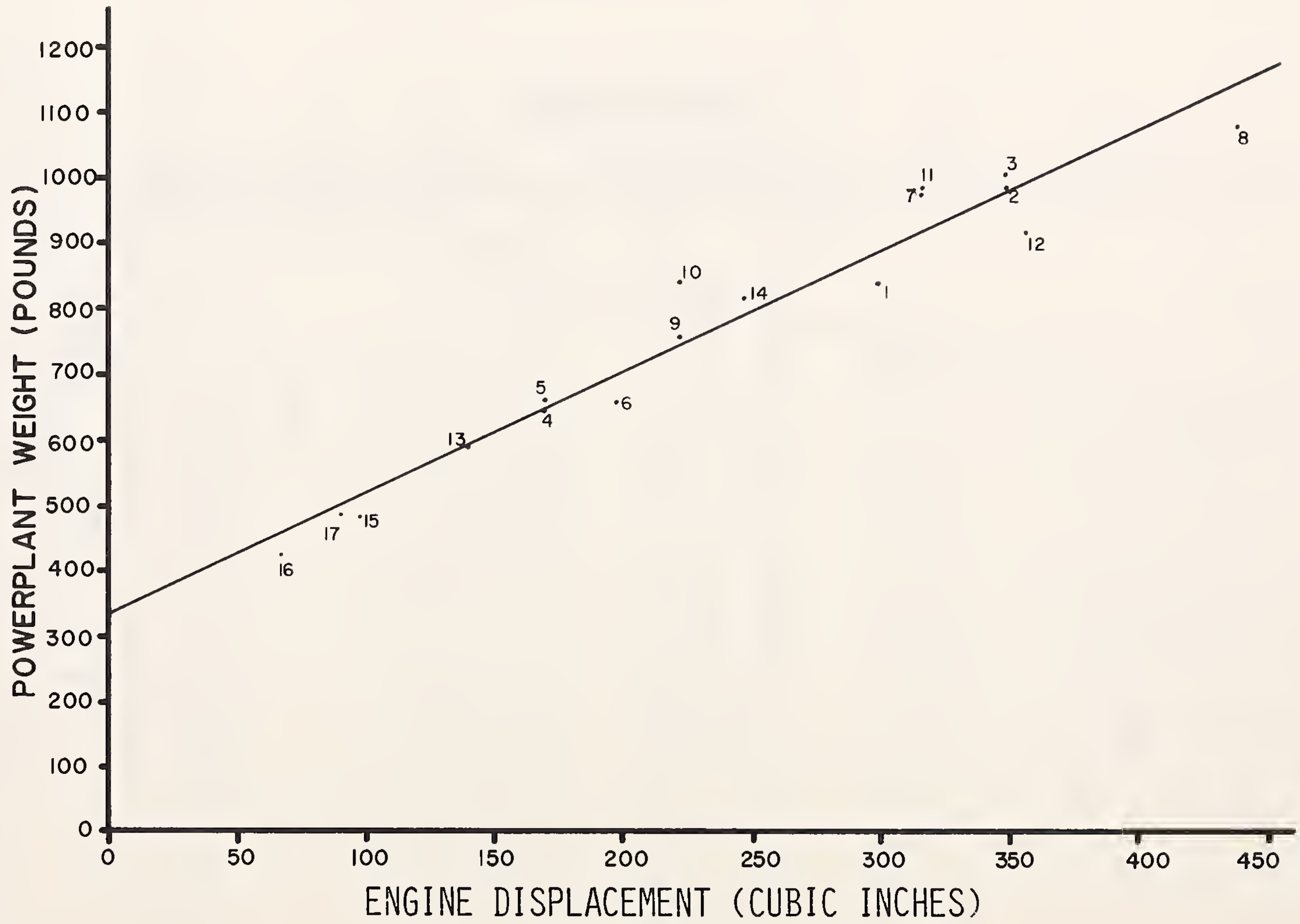


FIGURE 3B. POWERPLANT WEIGHT (POUNDS) VS. ENGINE DISPLACEMENT (CUBIC INCHES) standard engine of the given vehicle. These values are given in Table 3D. A least-squares curve fit procedure was used to determine the best mathematical fit to the data. Despite some scatter of the data, a straight line approximation proved to be optimal, with the line being described by

$$Wp = 330+1.85D$$

Thus m $\approx$ 330, n $\approx$ 1.85. The scatter of data, which can be noted in Figure 3B, is probably caused mainly by the fact that the power-dependent components of a vehicle are not necessarily designed for the so-called standard engine, especially if there are optional engines available. Other reasons for the scatter in Figures 3A and 3B might be inaccuracies in the weight breakdowns. It is felt that the scatter is not too severe and does not affect the validity of the analysis.

As can be seen from Table 3D, the values for p vary considerably from vehicle to vehicle. As was indicated earlier, p and D are co-dependent variables. When D is specified, as it was in this analysis, p is a dependent variable and indicates the vehicle weight per unit displacement. It might be mentioned that a high value of p (vehicle weight per unit displacement) usually necessitates a high engine speed (revs/mile). This will be discussed in more detail in Chapter 4.

3.3.4 Influence Coefficients

After having determined the values of the constants c, m, and n, along with a value for p for each vehicle, it was possible to calculate the influence coefficients Q1, Q4, and QD. The expressions developed in Section 3.2 for the influence coefficients in terms of p, c, and m were used in the calculations. The results of the calculations are given in Table 3D.

As can be seen from Table 3D, the values for Q1, Q4, and QD are nearly the same for all vehicles. Therefore, as was anticipated, the influence

TABLE 3D. DERIVED DATA FOR SEVENTEEN ACTUAL VEHICLES

	VEHICLE 1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
D (CU. IN.)	302	351	351	170	170	200	318	440	225	225	318	360	140	250	100	68	90
P (LB/CU.IN)	13.5	12.5	13.4	17.4	18.1	16.1	12.8	11.4	15.4	16.5	15.0	12.1	20.7	14.8	26.0	31.2	23.7
WEIGHTS IN POUNDS																	
W1	2386	2513	2668	1726	1774	1833	2235	2818	1974	2029	2748	2444	1722	1973	1570	1319	1253
W2	887	927	1057	614	675	772	892	1139	759	863	1076	1038	616	936	579	413	416
W3	511	653	677	317	327	324	647	752	425	515	655	587	262	491	151	90	161
CURB WEIGHT	3784	4093	4403	2657	2776	2929	3774	4709	3158	3407	4479	4069	2600	3400	2300	1823	1830
INFLUENCE COEFFICIENTS																	
Q1	1.43	1.44	1.45	1.35	1.36	1.41	1.44	1.50	1.39	1.40	1.42	1.50	1.35	1.46	1.33	1.26	1.29
Q4	0.23	0.23	0.25	0.21	0.22	0.25	0.23	0.26	0.22	0.24	0.24	0.27	0.23	0.28	0.23	0.19	0.19
QD	2.21	2.20	2.23	2.18	2.21	2.25	2.21	2.24	2.20	2.24	2.23	2.26	2.23	2.29	2.24	2.17	2.18

VEHICLES STUDIED

1. 1970 FORD CUSTOM 500
2. 1972 FORD CUSTOM 500
3. 1974 FORD GALAXIE 500
4. 1970 MAVERICK
5. 1972 MAVERICK
6. 1974 MAVERICK
7. 1974 PLYMOUTH SATELLITE
8. 1974 CHRYSLER NEW YORKER
9. 1976 VALIANT
10. 1976 PLYMOUTH VOLARE
11. 1976 PLYMOUTH GRAN FURY
12. 1976 CHARGER SE
13. 1975 VEGA
14. 1975 NOVA
15. C6
16. 1972 FIAT 128
17. 1975 VW RABBIT

coefficients are constants and do not depend on the vehicle being studied.

A statistical analysis was performed on the influence coefficient values given in Table 3D. The mean values are as follows:

$$Q1 = 1.400$$

$$QD = 2.22$$

$$Q4 = 0.235$$

The standard deviations for the coefficients are, respectively, $\sigma_{Q1} = 0.067$, $\sigma_{QD} = 0.030$, $\sigma_{Q4} = 0.024$.

Thus, a ten-pound decrease in the weight of a body component, for example, could lead to a total vehicle weight reduction of 14 pounds, assuming that the weight-dependent components are redesigned to account for the smaller load they need to support. Similarly, if a vehicle's engine size is decreased by 10 cubic inches and the weight-dependent (W2) components are reduced in size accordingly, a total vehicle weight reduction of 22.2 lbs could ensue. From the value for Q4 it can be seen that if the total weight of available options is reduced by 10 lbs, and if this reduction is taken into account in the design of the W2 components, the base vehicle (without optional equipment) could be 2.35 lbs lighter.

Each of these influence coefficients should be of much interest to the vehicle designer and to anyone else interested in the potential effect of weight reduction measures.

It might be noted that other investigators have determined values for an influence coefficient similar to Q1, because of its importance in weight reduction studies. Artinian and Terry (Ref. 3-1) determined a value of 1.539, Cochran (Ref. 3-2) used 1.5, and Adams et al. (Ref. 3-3) found a variable coefficient that averaged slightly greater than 2. It is felt that the value Q1 = 1.4 obtained in this study is the most valid because it is based on a study of a greater number and variety of vehicles.

3.4 SYNTHESIS OF WEIGHT-CONSCIOUS DESIGNS

3.4.1 Basic Assumptions

As was shown in Section 1.3, a comparison of vehicles of a given functional size (roominess) shows that there are vast differences in curb weights of functionally-equivalent cars. In the design of some vehicles, weight has been an important consideration, but for most vehicles weight considerations have taken a back seat to other factors, such as non-functional appearance. This has been discussed in depth in earlier sections of this report.

In this section we will develop vehicle designs for which reduced weight is of primary importance. Weight-conscious designs will be synthesized for each of the size classes discussed earlier: 4-passenger, 5-passenger, and 6-passenger. Each of the designs will be functionally equivalent to the best of current U.S. practice for its particular size classification. It is felt that these designs are optimum designs for their roominess class, at least from a weight standpoint. For this reason the vehicles are designated Optimum 4, Optimum 5, and Optimum 6.

No engineering marvels are assumed in the design of these optimum vehicles. They are of conventional construction and use conventional materials and production techniques. The vehicles could, therefore, be built without large investments in new facilities and without requiring newly-engineered vehicle components.

The major difference between these weight-conscious designs and the majority of vehicles that are being produced today are smaller, non-functional exterior size, fewer provisions for high-powered options, and fewer non-functional appearance items. The vehicles are, therefore, designed for the "rational buyer" mentioned in Section 1.1.

Although the optimum vehicles will lack many of the size and appearance features of some of today's automobiles, they are designed to meet all of the customer's functional expectations. The designs are fully equivalent to existing cars in their size class in each of the following respects: steering and handling, braking, acceleration, traction, durability, reliability, repairability, driver vision, accessibility of instruments, emissions control, heating and ventilation, and seating comfort. In addition, the cars would meet all applicable safety and impact standards.

Most of the optional convenience and appearance options that are available on present-day automobiles of a given roominess would also be available on the equivalent weight-conscious design. A list of such options will be found in Section 4.1.1. Both 5- and 6-passenger designs would have one optional engine available, but the optional equipment would not include the exceptionally high-powered engines available on many cars today.

Each of the optimum designs is assumed to be of unitized construction, rather than the body-frame construction found on many American cars. The noise-vibration and riding characteristics of these unitized designs would be slightly different than those of the body-frame cars, as was discussed in Section 2.3.2. It is assumed that the optimum vehicles would have the best ride and noise-vibration characteristics that could be obtained without significant weight penalty. As was discussed earlier, however, the lower weight of the optimum designs could possibly result in slightly inferior performance in these respects when compared with current heavier vehicles.

The body style chosen for the weight-conscious designs is the four-door pillar sedan, as represented by the Plymouth Valiant or Chevrolet Nova. The pillar sedan possesses full doors which include window frames attached to the doors, with a full center pillar between the doors.

3.4.2 Optimum Four-Passenger Vehicle

The Fiat 128 four door sedan was used as a starting point in the development of the weight conscious 4-passenger design. The choice of the Fiat 128 was prompted by the availability of reliable weight information for that vehicle (obtained by disassembly and weighing of the car's components), and by its reasonably large ratio of roominess/curb weight. Other four passenger vehicles with similarly large roominess/curb weight values, such as the VW Rabbit, could just as easily have been used as the basis for the Optimum 4 design. Our choice of the Fiat 128 does not indicate any feeling that the vehicle is superior to other vehicles of its size class.

The Fiat has a transversely-mounted engine with front wheel drive, a combination that was chosen for the Optimum 4 design because it has been used very effectively in a number of roomy, light, four passenger vehicles in recent years. The advantages of a front engine, front drive configuration are discussed in more detail in Section 4.3. It should be noted, however, that a front engine, rear drive design with the same roominess, weight, and functional characteristics of the Optimum 4 could be designed, with the 1976 Chevette being an example.

A breakdown of the weight of the Optimum 4 design is shown in Table 3E. Although the OP4 design was based on that of the Fiat 128, the OP4 weights differ from those shown in Table 3C for the 1972 Fiat. It was felt that modification of the 1972 design was desirable in order to meet the design specifications set out in Section 3.4.1 for the

TABLE 3E. WEIGHT BREAKDOWNS FOR WEIGHT-CONSCIOUS DESIGNS

(ALL WEIGHTS GIVEN IN POUNDS)

	VEHICLE		
	OPTIMUM 4	OPTIMUM 5	OPTIMUM 6
BODY	909	1138	1394
FRAME & TOOLS	8	17	110
BUMPERS	80	100	125
SUSPENSION	123	144	177
WHEELS & TIRES	134	156	190
DRIVE AXLE	42	107	123
TRANSMISSION	90	97	128
BRAKES	66	120	155
EXHAUST	30	33	46
FUEL	90	115	140
STEERING	30	47	80
HEATING & VENT	23	21	30
BATTERY & ALT	43	40	43
ELECTRICAL	19	20	22
ENGINE	300	396	508
W1	1398	1659	2061
W2	457	628	787
W3	132	264	423
W4	975	1427	1750
WX	300	300	300
CURB WEIGHT	1987	2551	3271
GROSS WEIGHT	2962	3978	5021

weight-conscious designs. The weight differences between the 1972 Fiat 128 component groups and those of the OP4 are given in Table 3F, along with reasons for the differences. The major differences are due to more stringent impact requirements and to a larger engine displacement. It is felt that OP4 engine should have a 100 cubic inch displacement, in order to give improved acceleration performance, a lower value of p (vehicle weight/unit displacement), and lower engine speed.

3.4.3 Optimum Five-Passenger Vehicle

The five-passenger car, which is representative of the "compact" U.S. vehicle and some of the "intermediate" U.S. and foreign cars, has seating room for 2½ passengers in both front and rear. In actual fact it is a rather tight six-passenger vehicle. As can be seen from Figures 1-C & 1-G few five-passenger cars, especially those produced by U.S. makers, have been light weight. In fact, most weigh more than 3000 lbs., with much unnecessary weight being contributed by non-functional size and provisions for heavy optional equipment.

In the early 1960's, however, several five-passenger vehicles were produced in the U.S. which were weight-conscious designs. One of these vehicles, the Ford Falcon of 1960-1965 serves as the prototype for the optimum five-passenger design. The Chevy II, which was very similar in most respects to the Falcon, could just as easily have been chosen.

A number of modifications were required to bring the Falcon design up to current standards and to take advantage of current materials and production techniques. The weight changes resulting from these modifications are listed in Table 3G, along with an explanation of the changes. The major weight increases were due to upgraded safety and impact standards, and an increase in engine size (to about 175 C.I.D.) for improved

TABLE 3F. WEIGHT COMPARISON BETWEEN OP-4 AND 1972 FIAT 128

Subsystem	Weights in Pounds			Explanation
	1972 Fiat 128	OP-4	Difference	
Body	837	909	+72	Improved impact resistance, better rust protection
Frame & Tools	8	8	-	
Bumpers	55	80	+25	5 mph impact requirement
Wheels & Tires	130	134	+ 4	Heavier weight-dependent components due to increased vehicle weight
Suspension	110	123	+13	
Drive Axle	39	42	+ 3	
Transmission	85	90	+ 5	
Brakes	57	66	+ 9	
Exhaust	25	30	+ 5	Added catalytic converter
Fuel	80	90	+10	Increased size of fuel tank
Steering	30	30	-	
Heating	30	23	- 7	Improved design
Battery & Alt.	47	43	- 4	Lighter battery case
Electrical	20	19	- 1	Improved wiring
Engine	270	300	+30	Increased engine displacement

TABLE 3G. WEIGHT COMPARISON BETWEEN OP-5 AND 1960 FALCON

Subsystem	Weights in Pounds			Explanation
	1960 Falcon	OP-5	Difference	
Body	1129	1138	+ 9	Improved impact resistance (+51), lighter materials for soft trim and weatherstrips (-24), thinner gauge glass (-37), added safety restraint (+12), addition and/or redesign of miscellaneous parts (+7)
Frame & Tools	17	17	-	
Bumpers	55	100	+45	5 mph impact requirements
Suspension	178	144	-34	Improved design
Wheels & Tires	147	156	+ 9	Upgraded standards (+25), improved materials & design (-16)
Drive Axle	90	107	+17	Axle durability improvement
Transmission	75	97	+22	Improved design (more durable 3-speed synchromesh)
Brakes	98	120	+22	Upgraded standards, front disc brake
Exhaust	23	33	+10	Improved durability, added catalytic converter
Fuel	107	115	+ 8	Safer fuel tank
Steering	42	47	+ 5	Collapsible column (+16), rack and pinion design (-11)
Heating & Vent	6	21	+15	Heater standard on OP-5
Battery & Alt.	50	40	-10	Lighter battery case, lighter alternator instead of generator
Electrical	16	20	+ 4	Improved wiring
Engine	368	396	+28	Greater displacement (+50), improved design (-54), emission controls (+32)

performance led to a further weight rise. These increases were partially offset by weight reductions made possible by modern production techniques, recent design improvements, and modern materials, such as tempered glass and plastics.

The weight breakdown for the OP-5 is given in Table 3E. A list of the optional equipment that would be available with the OP-5 will be found in Section 4.1.1 and the design decisions involved in the OP-5 design will be discussed in detail in Chapter 4.

3.4.4 Optimum Six-Passenger Vehicle

The six-passenger size class includes the American "full-size" cars, as well as some of the larger "intermediate" sedans. A study of the curb weights of vehicles in this size class shows that there are no U.S. six-passenger cars that could be considered as weight-conscious designs. Non-functional exterior size, appearance items, and provisions for heavy body styles account for a large amount of unnecessary weight in these cars.

Instead of attempting to redesign one of the heavy production vehicles, the optimum six-passenger study was derived from a vehicle that was designed along with the Falcon about 1960, but which was never put into production. That design was subjected to modifications similar to those discussed in the last section in order to make the design consistent with current safety standards and production techniques.

The overall character of the car would be somewhat similar to that of the OP-5, except for size differences. There would, however, be more noise-vibration control and upgraded interior trim and seating on the six-passenger vehicle. The OP-6 also has a sub-frame, not found on the OP-5.

The weight breakdown for the weight-conscious six-passenger design is given in Table 3E.

References for Section 3

3-1. Artinian, L. and S.L. Terry, "The Total Cost of Weight", SAE Paper 325D, 1961.

3-2. Cochran, C.N., "Aluminum-Villain or Hero in Energy Crisis", Automotive Engineering, Vol. 81, No. 6, p. 59, 1973.

3-3. Adams, D.G., et al., "High Strength Materials and Vehicle Weight Reduction Analysis", SAE Paper 750221, 1975.

4. SUMMARY OF DESIGN DECISION TREE, EVALUATION OF DESIGN CYCLE AND MARKET PRESSURES

4.0 INTRODUCTION

The design process will be illustrated here by the actual evaluation of major design decisions for a hypothetical all-new vehicle. The design will be optimized for weight and economy within the limits of presently accepted design characteristics and conventional materials, on the assumption that it is desired to get the vehicle on the market in as short a time as possible, which is the usual situation in the domestic industry.

In order to best outline the nature of the design process, certain unrealistic assumptions will have to be made. It will be assumed, for example, that carryover of components and systems already in production will not be necessary, which is ordinarily unrealistic financially, and it will be assumed that styling design will be restrained within the limit that functional and weight and size compromises will not be made in the interests of appearance factors, which is generally assumed to be unrealistic marketing-wise, since styling and non-functional size have repeatedly been proven to be of great importance in appeal to the majority of U.S. buyers.

Weight reduction is related to design in a number of ways. 1) By substitution of materials, which almost always requires changes of design, and is treated separately in the present study. 2) Weight reduction by improved manufacturing techniques, as for example, the reduction of weight of cast-iron parts by improved foundry techniques that permit casting of thinner sections (4-1) or improved fastening methods that permit the use of thinner sheet metal (4-2). 3) Weight reduction by more efficient design, as illustrated in 4.7, and, 4) Weight reduction by more efficient use of materials in existing designs. This latter results from a meticulous study of each part, with elimination of redundant material; this is a matter of good design, which is a commodity often in short supply, and which often demands additional time, which is usually in even shorter supply.

4.1 PRODUCT-PLANNING ASSUMPTIONS

The design objective of this study will be to design a five-passenger (compact size) vehicle that can be built in two versions, an economy version and a luxury version, with the maximum interchangeability of major components, and a minimum weight and cost penalty for interchangeability. It is assumed to be a clean-sheet-of-paper approach, since only in this way can the process of design compromise be illustrated cleanly. The imposition of requirements for carryover components introduces arbitrary design considerations that affect other components and arrangements as well, and carryover designs have already been presented for weight optimization.

The reason for the choice of the 5-passenger vehicle is that there is more potential for development of luxury characteristics in that size range than in the 6-passenger range, where these are already highly developed. The 5-passenger stands more to gain package-wise from the application of front-drive than does the 4-passenger, because of the need to accommodate center passengers, and the luxury-version requirement to package more powerful engines provides a better example of the kinds of problems and compromises involved in packaging drivetrain components in different configurations.

Weight and fuel economy are to be optimized, super-power options will not be provided for. Styling requirements provide for conventional configuration without cost and weight penalties for appearance provisions, and provide optimum comfort and active safety provisions (see 1.5 and 1.7).

4.1.1 Equipment and Option Assumptions and Weights

It will be assumed that these two vehicles will be equipped with automatic transmissions and power steering throughout. The manual transmission imposes limitations on choice of engines because of uneveness of torque impulses with engines having less than 6 cylinders, or (like the 90° V-6) having unevenly distributed power impulses. Furthermore, new drivers are generally not being taught to drive on manual-transmission-equipped cars because of the high cost of maintenance in instruction service. Cars of this weight and size are too heavy for manual steering unless unsuitably low ratios are chosen, and this condition is much augmented by front-drive and by radial-ply, steel-belted tires.

It is also contemplated that both models will be made available with all the presently available luxury and convenience options, provided that the resulting gross weight does not impose on the economy model too great a weight penalty for interchangeability with the luxury model. A list of these, with their weights, follows:

Optional engine	54	(W3)*
Light group	2	
Rear window defogger	3	
Heavy-duty suspension	8	(W2)
Vinyl roof	4	
Luxury decor group	116	
Interior decor group	40	
Bumper guards	11	
Air conditioning	95	
Radio, A/M, F/M, stereo	11	
Heavy-duty battery	7	
Larger brakes	20	(W2)
Appearance protection group	7	
Lock group	2	
Oversize tires	30	(W2)
Levelling rear suspension	10	
Speed control	4	
Deluxe wheel covers	10	(W2)
Power windows	24	
Power door locks	7	
Heated backlight	7	
Power seat adjustment	15	
Larger gas tank	65	
TOTAL	552	

W4 = 429

W3 = 54

W2 = 69

*The items marked are included within the marked weight groups. All unmarked items are assumed to be included in W4 for purposes of weight allocations.

Since the two vehicles will be interchangeable, each will be basically capable of carrying the entire list of options in addition to its normal maximum disposable load. The penalty of interchangeability will be this incremental load's accompanying increment in the interchangeable W2 parts of the lighter vehicle. For purposes of preliminary calculation the weight of the basic vehicle will be taken as that of the OP5 with power steering and automatic transmission added. The disposable load will be taken to be that of 5 passengers, each with 20 lbs of luggage, a total of 850 lbs, plus options.

4.1.2 Preliminary Weight Estimates

Economy Model:

Base car, OP5	2566
Power steering	30
Interchangeability penalty	30
Design weight reduction task	(50)
Preliminary curb weight, economy model	2576
Performance weight (Wc+300)	2876
Base loaded weight (Wc+875)	3451
Total options	536
GVW economy model	3987

Luxury Model:

Base car (economy model	2576
Noise & vibration control	100
Isolated front suspension	60
Larger engine	54
Larger tires & wheels	30
Heavier brakes	20
Load leveller	20
Air conditioning	95
Luxury & appearance items	100
Preliminary curb weight, luxury model	3055
Performance weight	3355
Base loaded weight	3930
Options	217
GVW luxury model	4147

4.2 PERFORMANCE AND ECONOMY

4.2.1 Engine Size and Speed Range

Acceleration is commonly measured in terms of horsepower/weight ratios. However, contemporary published horsepower figures are measured on widely differing assumptions and are presently in a considerable state of flux with changing emission-control requirements and no established means of meeting them. It will therefore be assumed instead that engine mean-effective pressures are comparable between different alternative engine types, and that car performance is a function of displacement per unit weight per mile, or cubic inches per ton per mile (cppm).

Cppm is a function of engine displacement and engine revolutions per mile (erpm) and is an inverse function of weight. In domestic cars it is obtained with relatively large displacements at relatively small erpm, while in imported cars displacements are smaller and erpm are greater.

Engine durability is a subject beyond the scope of this study, but the assumption will be made that other things being equal, the smaller the erpm, the higher the durability; in other words, engine life is measured in revolutions rather than in miles. On the other hand, the smaller the engine the less the weight, so we have an interest in using the smallest engine that will meet the criteria for durability, performance and quietness and smoothness of operation. Since engine mechanical efficiency decreases with increasing piston speed (4-3), and the length of stroke of similar engines is a function of the cube root of displacement, it follows that the higher-speed engine will have a lower mechanical efficiency than the larger, lower speed engine, a factor that will tend to negate the economy gain resulting from its lower weight. The higher-speed engine will also make more noise and vibration.

In Table 4A we list a number of 1976 domestic and foreign 5-passenger vehicles, their inertia weight, displacement, erpm and cppm. In 4B we list some of the same makes in 1973 form, showing the considerable reductions in cppm and erpm that have been made to improve fuel economy in recent years in U.S. cars. This has in all instances been accomplished simply by reducing the final drive ratio, as can be seen. This reduces top-gear acceleration proportionately, and also reduces the maximum possible low-gear torque by the same proportion.

TABLES 4A. AND 4B. DISPLACEMENT PER POUND MILE FOR 5-PASSENGER CARS

Make & Model	Inertia Weight (W+300)	Displacement	Gear Ratio	Tire Size	Wheel Revs/Mile	Engine Revs/Mile (erpm)	Cubic Inches/lb/mi (cppm)
1973 Models:							
AMC Hornet 6	3184	232	2.73	6.95×14	807	2203	80.26
Buick Apollo 6	3642	250	3.08	E78 ×14	776	2390	82.03
Chevrolet Nova 6	3469	250	3.08	E78 ×14	776	2390	86.12
Dodge Dart 6	3310	198	3.23	6.95×14	807	2607	77.97
Ford Maverick 6	3036	200	3.00	6.45×14	830	2489	82.01
Plymouth Valiant	3265	198	3.23	6.95×14	807	2607	79.04
Average of U.S. 1973 Models						2448	81.24
1976 Models:							
AMC Hornet 6	3266	232	2.73	6.95×14	807	2203	78.24
Buick Apollo V6	3795	231	2.56	FR78×14	766	1961	59.68
Chevrolet Nova 6	3716	250	3.08	FR78×14	766	2359	79.36
Dodge Dart 6	3484	225	2.45*	6.95×14	807	1977	63.84
Ford Maverick 6	3290	200	2.79	C78 ×14	806	2250	68.41
Plymouth Valiant	3430	225	2.45*	6.95×14	807	1977	64.84
Average of U.S. 1976 Models						2121	69.06
1976 Imports:							
Audi 100	2871	115	4.11*	165SR×14	827	3397	67.44
Mercedes Benz 230	3485	140	3.92	175R×14	808	3167	63.62
Peugeot 504	3205	120	3.89*	175R×14	808	3143	58.84
Saab 99GL	2870	121	3.89*	165R×15	804	3128	65.94
Volvo 244	3256	130	4.10*	C78×14	806	3305	65.97
Average of 1976 Imports						3218	64.36

*4-speed transmission

We take advantage of the assumption that the 1973 U.S. cars listed in Table 4A have adequate engine durability, and that the 1976 cars in Table 4B have adequate top-gear acceleration, so we will base our assumed displacement on the erpm of the 1973 U.S. cars and the cppm of the 1976 cars, and choose 2500 top-gear erpm and 63 cppm. In other words, we will accomplish the switch to lower cppm by reducing displacement instead of erpm from their 1973 values.

On this basis assuming 2500 erpm neglecting torque converter slip, our 2876 lb inertia-weight vehicle will require 145 ci and the 3355 lb vehicle will require 169 ci.

These displacements are relatively small for the weights of the cars, there being 18 lbs curb weight per ci, so we conclude that the engines should have overhead camshafts in order to provide a high specific output, and we assume that they will peak at 4800 rpm, with a maximum permissible speed of 6000 rpm.

We will defer a choice of engine configuration until a decision has been made on vehicle configuration and drive.

Our weight range calls for tires of 165R14 on the lighter vehicle and 175R14 on the heavier. These tires roll 827 and 808 revolutions per mile, respectively (4-4), and to provide 2500 erpm will then call for overall ratios of 3.02 and 3.09 in top gear. When overall ratios are discussed in connection with transmission ratios they will be referred to the 3.02 final ratio of the lighter vehicle. The transmission ratios will be the same in both vehicles.

We estimate that the weight saving achieved by the adoption of these smaller engines will be 40 and 48 lbs, respectively, as compared with lower-speed pushrod engines of the same maximum output, which we estimate would have displacements of 180 and 210 ci, respectively. The estimated weights of these engines are as follows:

180 pushrod engine	400 lbs
210 pushrod engine	445 lbs
145 OHC engine	360 lbs
169 OHC engine	397 lbs

4.2.2 Transmission Characteristics and Ratios

Transmission requirements are three in number: 1) to provide a quick downshift into a ratio suitable for city driving and moderate acceleration of the order of that provided by the normal top-gear ratios of the 1973 cars in Table 4B, 2) to provide a passing gear that will make the maximum engine power available in the important passing range from 50-70 mph, 3) to provide a low gear that will give .5 lbs of tractive effort per pound of car weight. Unless the first and second requirements can be satisfied with a single intermediate speed, a four-speed transmission will be required.

The low gear, to provide .5 lbs of tractive effort per pound of weight, assuming that the engine develops 125 lbs/sq.in. bmep at converter stall speed, will be the equivalent of 253 cppm. Assuming a converter ratio of 2.0 and a transmission efficiency of .8, this is the equivalent of 158 cppm and an overall reduction of 8.24, or a transmission ratio of 2.72. Assuming overhead-cam engines that peak at 4800 and have a maximum speed of 6000, and 5% converter slip at peak speed, to provide maximum power at 55 mph will require a reduction ratio of 2.0, which, at 6000 rpm will have a maximum speed of just about 70 mph, and will satisfy the requirement for the passing gear. However, this ratio is far too high for a satisfactory downshift and city-driving speed, from which it is concluded that 4 speeds are necessary. To give even intervals we choose a ratio of 10/7 for the third speed, which gives an overall ratio in that speed of 4.32.

These ratios are obtainable by various dispositions of gearsets, and we will defer the decision concerning the transmission specifics until we have determined the vehicle configuration. Top speeds at 6000 rpm in the three indirect gears, assuming 3% torque-converter slip at 6000 rpm, will be 49, 70 and 98 mph.

4.3 VEHICLE CONFIGURATION (see also 2.3.1)

Alternatives:

1) Front-engine front-drive, (Audi, Saab)

2) Front-engine rear-drive, (Nova, Maverick, Valiant, Mercedes-Benz, Hornet)

3) Rear-engine rear-drive (Corvair)

We believe the rear-engine configuration represents an impact hazard by virtue of possessing too small a proportion of its mass forward to the bulkhead. The rear-engine car provides less favorable luggage accommodation than the front-engine. Its advantages are good traction, by virtue of the high proportion of weight on the driving wheels, and the advantages of the unit powertrain. It also presents the advantage that the seating package is not infringed by the presence of the transmission and propeller shaft under the floor and seats. Because of its assumed safety problems it will not be given further consideration.

The front-engine rear-drive is the most widely used and best-proven configuration. It permits the greatest freedom of choice in engine configuration, and permits the use of large engines in small cars. The underhood area is less congested than that of the front-drive with similar-sized engine, and is more easily accessible for repair for that reason. It is generally regarded as cheaper to build than the front-drive, which is partly a reflection of the fact that nearly all the cost comparisons that have been made have been between a presently-used and well-developed rear-drive design and a new, untried front-drive, to be assembled in facilities designed for assembling rear-drive cars.

We reemphasize at this point the excellent financial reasons for staying with the front-engine rear-drive design, simply because these are presently tooled and designed, and the additional investment required to tool and design and develop a completely new type vehicle is not justified by probable increased sales and profits of the new vehicle over the old, even if the needed large amounts of capital were available for the purpose. However, this is a design study, admittedly based on unrealistic financial assumptions.

The principal disadvantages of the front-engine rear-drive design relate to the package, to the rear suspension, and to traction under slippery conditions. In particular, for those vehicles that require the accommodation of a center passenger, the rear-drive requirement for a transmission and driveshaft under the floor of the passenger compartment imposes a severe penalty on leg-space and seating comfort for the center passenger. This is even more critical in the case of a low-built vehicle with low seating position, in which it becomes necessary to restrict the depth of the seat cushion over the transmission-propeller-shaft tunnel.

Most front-engine rear-drive cars use a driven rear axle having the final drive mounted directly in the middle of it. This large mass, being unsprung and in a position to form an anvil whereby disturbances are communicated from one wheel to the other, causes rear-suspension stability problems of which the solutions are compromises between stability and comfort, usually involving compromises in both. Rear suspensions in which the final drive is suspended are costly and complicated, and in the case of the independent rear suspension they place limitations on allowable rear-spring travel.

Rear-drive cars, especially those with relatively heavy engines, air conditioning, etc., carry more of their curb weight on the front wheels, and when they are lightly loaded they have poor traction on slippery roads. Vehicle stability is lost when the rear wheels lose their adhesion, and this constitutes a safety hazard for other than very skillful drivers.

The advantages of the front-drive relate to: package, traction on slippery roads, quality and cost of assembly operations, safety.

Most front-drive cars have a small tunnel in the middle of the passenger-compartment floor, both for structural reasons and to accommodate the exhaust system, but this is low and narrow and offers little impediment to comfortable center-passenger seating. In addition to improved seating, the lack of need to accommodate the final drive makes available a considerable amount of increased room behind the rear seat back for disposal of luggage and the fuel tank.

Traction in slippery going, and especially on ice and snow, is considerably better in the front-drive than the rear-drive, even for equal proportions of weight on the driven wheels, in spite of the fact that forward acceleration increases the load on the reqr wheels while decreasing that on the front wheels. This is because when the front wheels are turned they travel over a longer path than the rear wheels do, so that front-wheel traction is improved to the degree that the car is travelling on a curved path. It is possible to drive out of snow, sand or mud that would mire a rear-drive, simply by travelling in a wavy path (see also 4.6).

The unit powertrain, in which engine, transmission and final drive are combined in a single assembly, offers potential advantages in weight, simplicity and assembly cost and quality, the last being true only where facilities are designed especially for the purpose. The entire powertrain can be manufactured and assembled in a centralized facility in which automation and quality-control can be optimized to a much higher degree than is possible in a number of decentralized assembly operations.

We make the assumption that safety in a forward impact is improved to the degree that the mass of the vehicle is located forward to the bulkhead. The difference between front and rear drive in this respect is the mass of the transmission and final drive, which is forward to the bulkhead in the front-drive and behind it in the rear-drive.

The disadvantages of the front-drive relate to the need to accommodate under the hood components that are behind the bulkhead in the rear-drive. This imposes severe limitations on the size and configuration of the engine and presents problems in accessibility for repair. It is for this reason that the most successful front-drive designs are almost all amongst the smallest class of vehicles (exceptions in the 5-passenger class, the Audi 100 and the Saab) in which the package advantages are least because of the lack of center-passenger accommodation.

It should be noted that the above disadvantage becomes a potential gain in the event that it is proposed to improve the crashworthiness of the vehicle. Lengthening the car forward of the bulkhead can be an effective way to improve crashworthiness. It will at the same time alleviate the crowded conditions under the hood.

Front-Drive Powertrain Configurations

Alternatives: 1) Longitudinal engine (Audi, Saab)

2) Transverse engine (Citroen CX2000 also in the 4-passenger class, Rabbit, Fiat 128, Peugeot 304, Honda.

The advantage of the longitudinal-engine configuration is that it uses basically the same type of final drive as the rear-drive vehicle, the hypoid bevel. This is also one of the disadvantages, since the hypoid bevel is heavier, more costly and less efficient than the spur-gear or chain final drive that is used with the transverse engine.

The transverse-engine configuration permits the use of the more efficient final drive, and also allows a more complete integration of engine and transmission housings for improved simplicity and lighter weight. The chain or spur gear can be lubricated from the transmission lubrication system, while the hypoid bevel requires its own specialized lubrication system.

The transverse arrangement, however, severely restricts the length of the engine, or alternatively it requires a second transfer drive from engine to transmission to permit the transmission to operate on an axis parallel to that of the engine instead of coaxially with it. For this reason it is most applicable to cars with relatively short engines.

4.4 ENGINE CONFIGURATION AND SIZE (see also 2.3.3)

Alternatives:
1) 90° V-8 (Mercedes 450, optional on Nova, Maverick, Valiant).
2) 90° V-6 (Buick, Volvo, Peugeot).
3) In-line 6 (Nova, Maverick, Valiant, Hornet, Mercedes 280).
4) 60° V-6 (Optional on Capri, Mustang).
5) In-line 4 (Mercedes 230, Peugeot, Audi, Saab).
6) Opposed 4 (Lancia Gamma, Subaru and Beetle 4-passenger).

We make note of the displacements of production engines today as compared with their numbers of cylinders, and we observe there is no overlap between production engines of various numbers of cylinders. The 8-cylinder engines range from 260 to 500 ci, the 6-cylinders from 162 to 258, and the 4-cylinders are all below 146 ci. Thus, in a sense, we have chosen our engine type in the process of choosing its displacement.

Larger cylinders, having a more favorable surface/volume ratio, are more efficient, both thermally and mechanically, and produce fewer unburned hydrocarbon pollutants (4-5). On the other hand, the more cylinders an engine has the more smooth-running and quiet it will be. The 90° V-8 and V-6 share the unique advantages that primary main-bearing loads can be completely balanced out by counterweighting, and that there is a complete absence of secondary inertia torque. The V-8, in addition, has the smoothest torque curve, so it is quieter and smoother than other engines (4-6).

The four-cylinder engine is simple and cheap and efficient, but it has undesirable sources of noise and vibration. Secondary reciprocating and torsional forces are large, but can be balanced out by double-speed balance weights (4-7). However, there is presently in use no satisfactory method of compensating for the large torque variations that result from a relatively few non-overlapping power impulses. These variations are less objectionable when used with the torque converter.

The 6-cylinder in-line engine is widely used, cheap, simple and efficient. Kinematically it is well balanced, but balancing involves internal stresses that cause bending vibrations at some points in the operating range (4-8). Its length would cause difficulties in accommodating it to a front-drive powertrain design.

The 90° V-6 is kinematically good, but it has uneven power impulses, with the power impulses coming at alternate intervals of 90° and 150°. It is more complex, heavier and more expensive than the in-line 6, but its short length permits compensating savings in vehicle length, or, in a front-drive, in powertrain complexity. The 60° V-6 has the complexity without the uneven power impulses, but it has a less favorable proportion between its 6-throw crankshaft and its 3-cylinder blocks, enforcing too short a shaft for good bearings or too long a cylinder block for economical proportions.

The decision between front-drive and rear-drive presents difficulties, especially when we consider the requirement that the economy vehicle and the luxury version be essentially interchangeable. The rear-drive enjoys great flexibility in choice of engines, with few penalties in accommodating the 6-cylinder in-line or V-8 engines, whereas the 6 in-line is ruled out of any front-drive package because of its length, or alternatively by virtue of a complex drive configuration to accommodate it. There is no adequate substitute for the 6-cylinder in-line engine for good dynamic properties at low cost, with moderate weight.

We nevertheless persist in choosing the front-drive configuration for its safety and functional advantages, with the ultimate gain in quality and economy resulting from the use of the integral powertrain. The economy engine falls within the size range of normal 4-cylinder engines, although this engine lacks the inherent smoothness of operation of the 6-cylinder with which it must compete in its field. We therefore allow for the possible need to equip this engine with secondary balancing means similar to that described in connection with the Japanese-built small Chrysler car (4-7).

In the case of the luxury version of the vehicle, its displacement falls at the low end of the 6-cylinder displacement range, but we conclude that the V-6 is not an adequate substitute for the V-8 in a luxury vehicle because of its uneven firing order and its residual secondary unbalance. We therefore conclude that a V-8 is necessary even in this displacement range; that in fact the V-8 is indispensible in a luxury vehicle expected to have the highest functional quality.

4.5 VEHICLE CONSTRUCTION; UNITIZED OR BODY FRAME (see also 2.3.2)

We conclude that unitized construction is lighter than body-frame, at least for the economy model. Body-frame construction is admittedly an excellent method of noise and vibration control, but for a vehicle such as our luxury model, which is expected to share major interchangeability with lower-cost unitized models there is an alternative method of noise-vibration control, which is the isolated front subframe, or crossmember, such as used in the Seville, which shares major parts with the lower-cost unitized Nova, or the Mercedes-Benz 450, which shares major parts with lower-cost unitized cars.

In Section 2.3.2 of this report we compare weights for the 1971 Torino, a unitized car, and the 1972 Torino, a body-frame car having the same major body dimensions as its 1971 predecessor, with the same powertrain components. These figures indicate a weight-penalty of 200-230 lbs chargeable to the body-frame version of the car. Admittedly, the noise-vibration characteristics of the 1972 car are better than those of the 1971, and a considerable weight penalty would have to be added to the 1971 car to make it the equivalent in that respect to the 1972 model.

4.6 WEIGHT DISTRIBUTION AND WHEELBASE

We have tentatively arrived at a design synthesis of a vehicle having front-drive, transverse-engine, with chain final drive, in which practically the entire W3 weight is forward of the bulkhead, whereas the W1 and W2 components are pretty much as they are in the rear-drive configuration. The result is a large concentration of curb weight on the front wheels.

We are faced at this point with a decision concerning weight distribution and its implications for wheel and tire dimensions, cost and weight. Front-drive vehicles suffer a penalty in wheel weight owing to the need to locate the plane of the mounting-bolt circle farther outboard of the center-plane of the tire (Figure 4A) than the rear drive, involving a rotating moment that demands heavier construction to resist the resulting fatigue stresses. In Table 4C there are plotted wheel and tire weights for a number of representative front and rear-drive cars. From this plot it will be noted that the lighter vehicles have the heaviest proportion of the weight in wheels and tires, but there is also a significant difference between the front-drive and the rear-drive cars of about .5% of loaded vehicle weight in wheels and tires. Except for the Toronado, the average front-drive car is lighter and smaller than the average rear-drive in the list.

The size of wheels and tires is determined by the normal loaded weight, so it is seen as desirable to optimize wheel and tire weight by equalizing the load on front and rear wheels at this load level, which, in our economy model is 3438 lbs. We will not optimize wheel and tire weights for the luxury model, since it affords larger wheels and tires than the economy model. In the front-drive car this is done by positioning the rear axle, which can be moved without substantial penalty over a range of positions aft of the package position.

Figure 4B is a moment diagram representative of our tentative front-drive design, in which are shown the individual weights of various parts, with their distances, positive behind the front-wheel centerline and negative in front of it. The sum of these moments, less that of the weight that is movable with the axle, (Wa), will be the weight at the rear axle less Wa, multiplied by the wheelbase, B. We see that W-Wa = 3211, Wa = 215, and that the center of gravity of W-Wa is at position +52.17. The desired weight at the rear axle is W/2 = 1713. This, less Wa, is 1498, so we can write:

$$B = \frac{(3211 \times 52.17)}{1498} = 111.8''$$

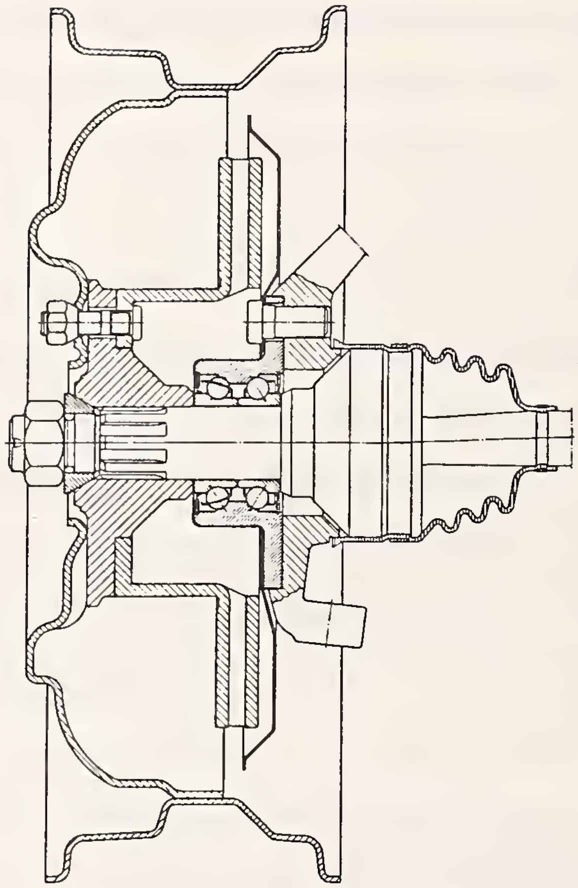
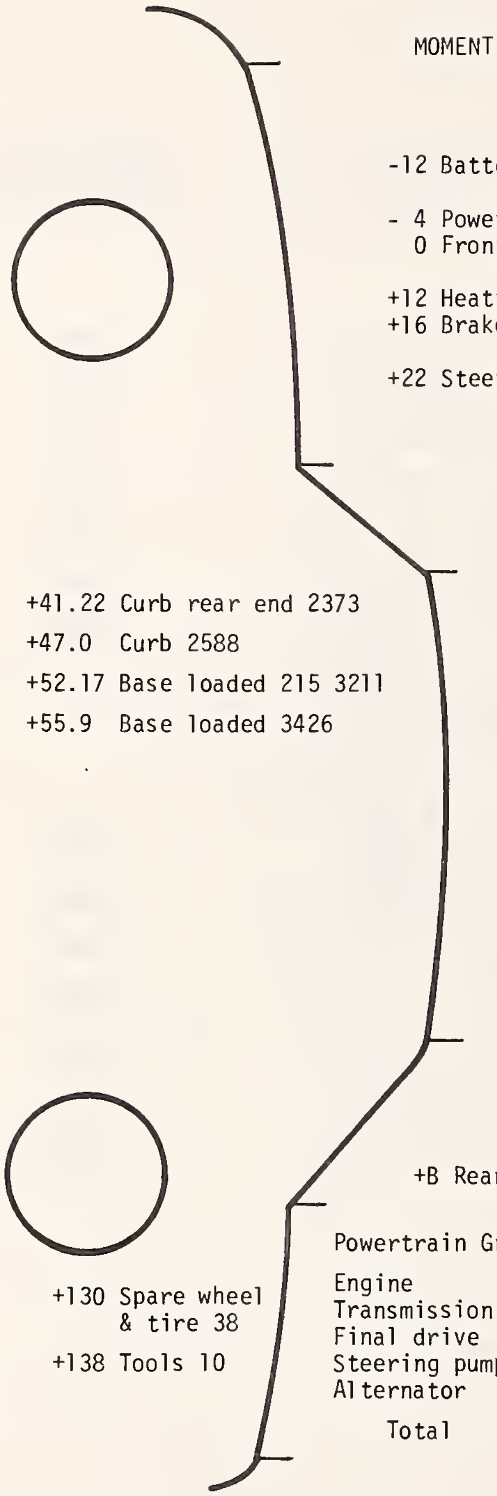


FIGURE 4A. FRONT-DRIVE

TABLE 4C. WEIGHTS OF WHEELS & TIRES FOR FRONT AND REAR-DRIVE CARS

	Loaded Wt.	Wheels & Tires Wt.	Percentage Wheels & Tires
Front-Drive Cars			
Rabbit	2430	147	.060
Fox	2588	153	.059
Saab	3300	185	.056
Alfa Sud	2623	166	.063
C-6	3000	165	.055
Toronado	5879	309	.053
Average Front-Drive	3303		.0577
Rear-Drive Cars			
Kadett	2372	123	.052
Vega	3150	172	.054
Pinto	3264	165	.051
Maverick	3850	179	.046
Ford Car	5521	241	.045
Olds 98	5950	279	.047
Average Rear-Drive	4017		.0495

MOMENT WORK SHEET - BASE CAR



Powertrain Group		Front End Group		Rear End Group	
Engine	363	Front Susp.	75	Rear Susp.	65
Transmission	123	Wheels & tires	75	Rear axle	36
Final drive	30	Brakes	57	Brakes	38
Steering pump	10	Driveshafts	30	Wheels & tires	76
Alternator	11	Steering	40		
Total	536	Total	278	Total	215

FIGURE 4B. TENTATIVE FRONT-DRIVE DESIGN

This places the center of gravity of the car at curb at +47, with the cg. at fully loaded condition at 55.8.

The cg of the variable load is at +82, which means that 73% of the variable weight is on the rear wheels. The effect of this is that at all loads less than its normal maximum, the front-drive car has a larger proportion of its weight on the driving wheels, while the rear-drive car has a smaller proportion. Hence the front-drive car has the advantage in traction at all normal loads under the maximum, and the vehicle with the heavier optional engine has improved traction.

On grades, and under acceleration, a certain amount of vehicle weight is transferred to the rear wheels, so that the rear-drive gains and the front-drive loses a certain amount. Part of this gain is sacrificed in the rear-driven car with the live axle because of the tendency of the propeller-shaft torque to lift the right rear wheel. The curve, Figure 4C, shows the coefficient of friction at wheel-slip for front and rear-drive cars under different loading conditions. This shows that the front-drive car enjoys the superiority under low-friction conditions, except with a full load. See Addendum for the complete analysis of this condition.

4.7 DESIGN DETAILS AND WEIGHT REDUCTION

4.7.1 Suspension

Figure 4D illustrates the comparison between the SLA suspension commonly used, and the MacPherson suspension shown in Figure 4E whereby the latter eliminates the upper arm and its pivots, utilizing instead the guiding effect of the shock absorber, and supporting the load on the coil spring directly on the front-end structure. This construction is especially adaptable to unitized construction, and it is both lighter and simpler than the SLA design.

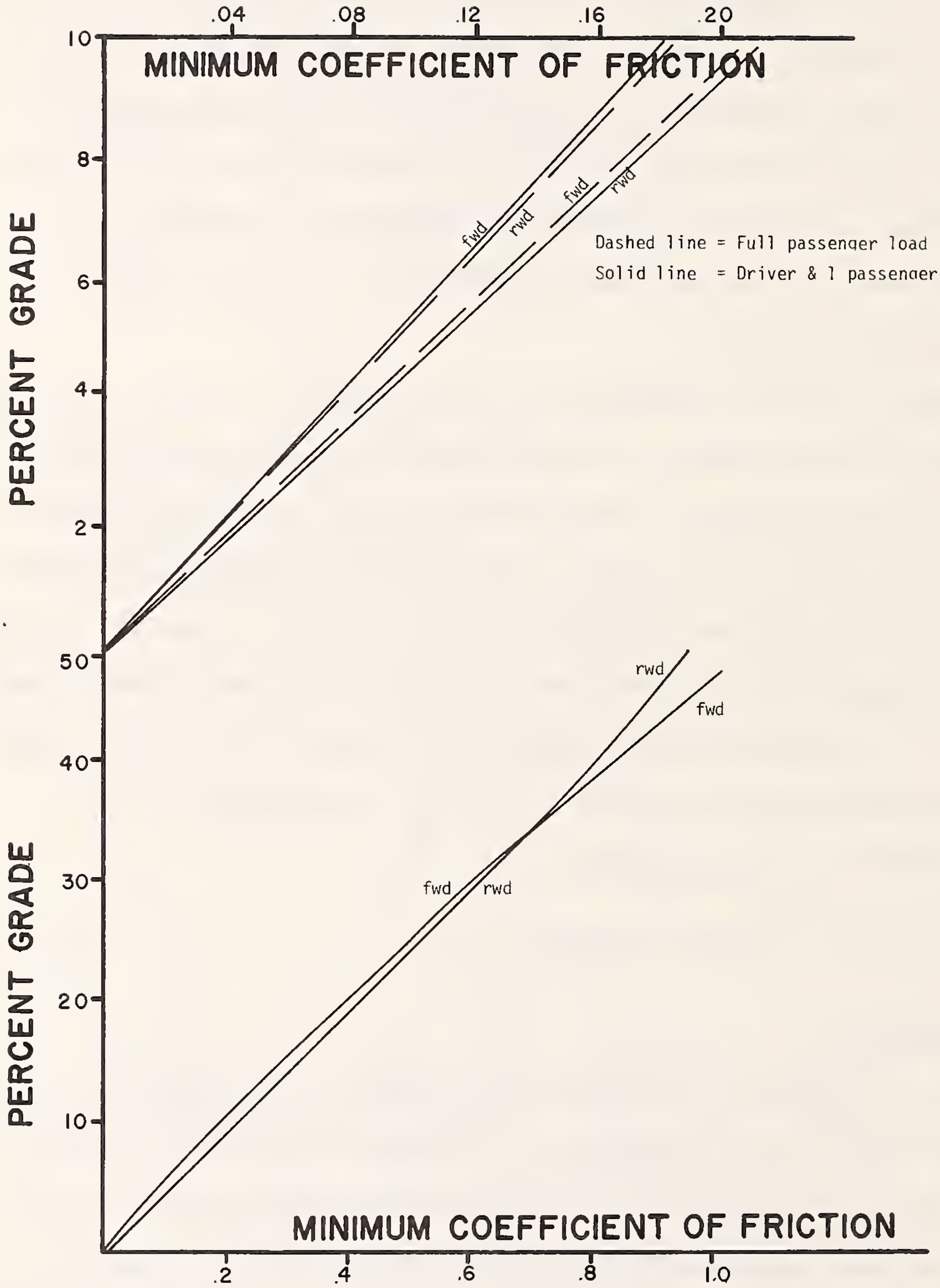


FIGURE 4C. MINIMUM FRICTION COEFFICIENT TO AVOID SLIPPING VS. PERCENT GRADE FRONT AND REAR-DRIVE CARS.

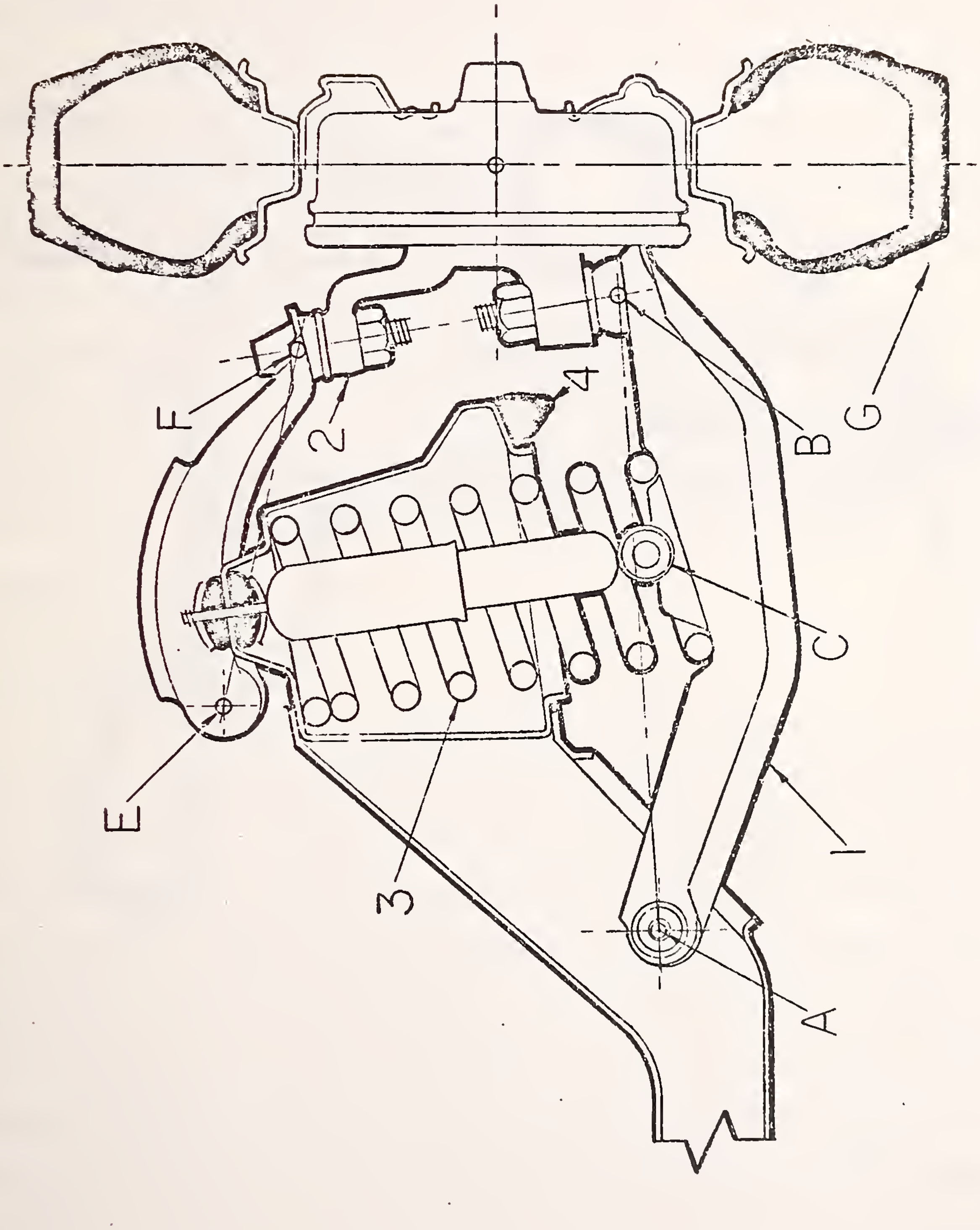


FIGURE 4D. SLA SUSPENSION

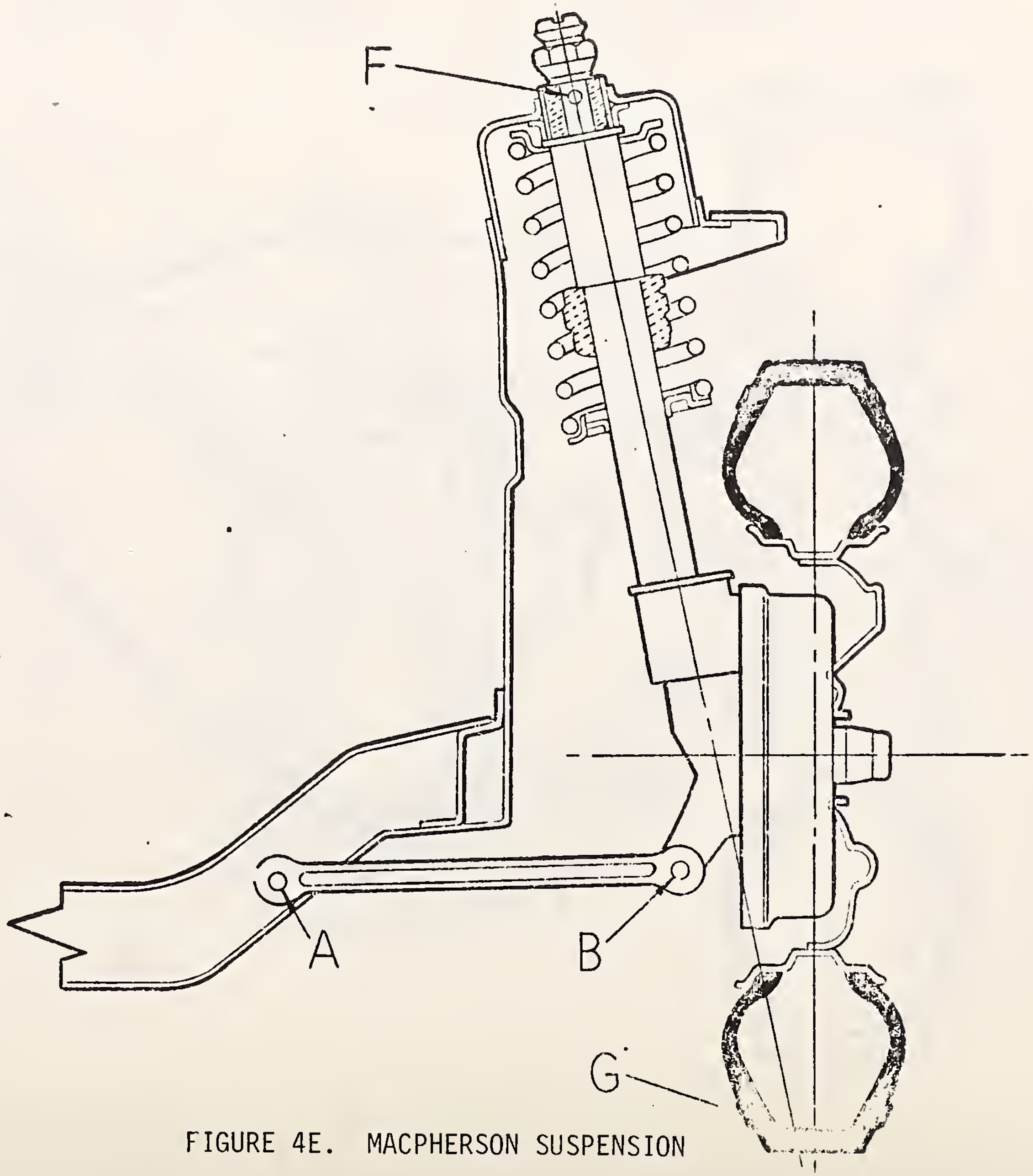


FIGURE 4E. MACPHERSON SUSPENSION

4.7.2 Steering Gear and Linkage

Figures 4F and 4G compare the rack-and-pinion steering gear with the ball-nut gear used on all but a few small U.S.-built cars. The rack-and-pinion gear itself replaces the cross-link and auxiliary arm, having the tie-rods connected directly to it. There is no unfavorable mechanical advantage such as that which multiplies the internal stresses of the ball-nut gear as well as increasing its compliance, so that it is feasible to use an aluminum casting for the rack-and-pinion housing. Lost-motion is less of a problem with the rack-and-pinion gear because of this lack of multiplication, and there is no need for the critical tolerances required by the ball-nut and its associated balls and screws.

A rack-and-pinion manual gear for a 3000 lb car has a weight of 14 lbs, plus 6 lbs for the linkage, a total of 20 lbs. A ball-nut gear for a similar car weighs 21 lbs, linkage 17 lbs, total 38 lbs. The ball-nut gear and linkage has a total of 61 parts and 52 balls, the rack-and-pinion gear and linkage has a total of 38 parts.

4.7.3 Final Drive

Figure 4H depicts a hypoid bevel gear, such as used in final drives of live axles and longitudinal-engine front-drive vehicles. Being a bevel gear, it is critical with respect to alignment of gear teeth, requiring that both shafts be accurately positioned both axially and radially, and mounted in structures of the lowest possible compliance, the pinion overhangs the bearing, adding to the stress to which the bearing and its housing are subjected. The gear teeth are subject to sliding friction, which requires a special lubricant and introduces losses that limit the gear's efficiency, especially at part loads. The result of these factors is that the gears, their bearings

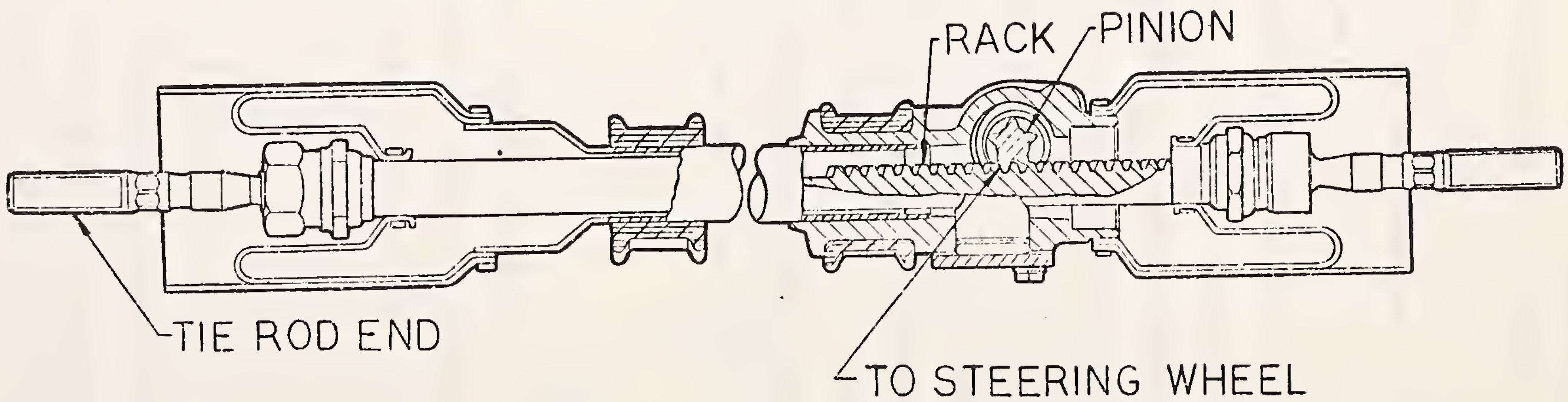


FIGURE 4F. RACK AND PINION STEERING GEAR

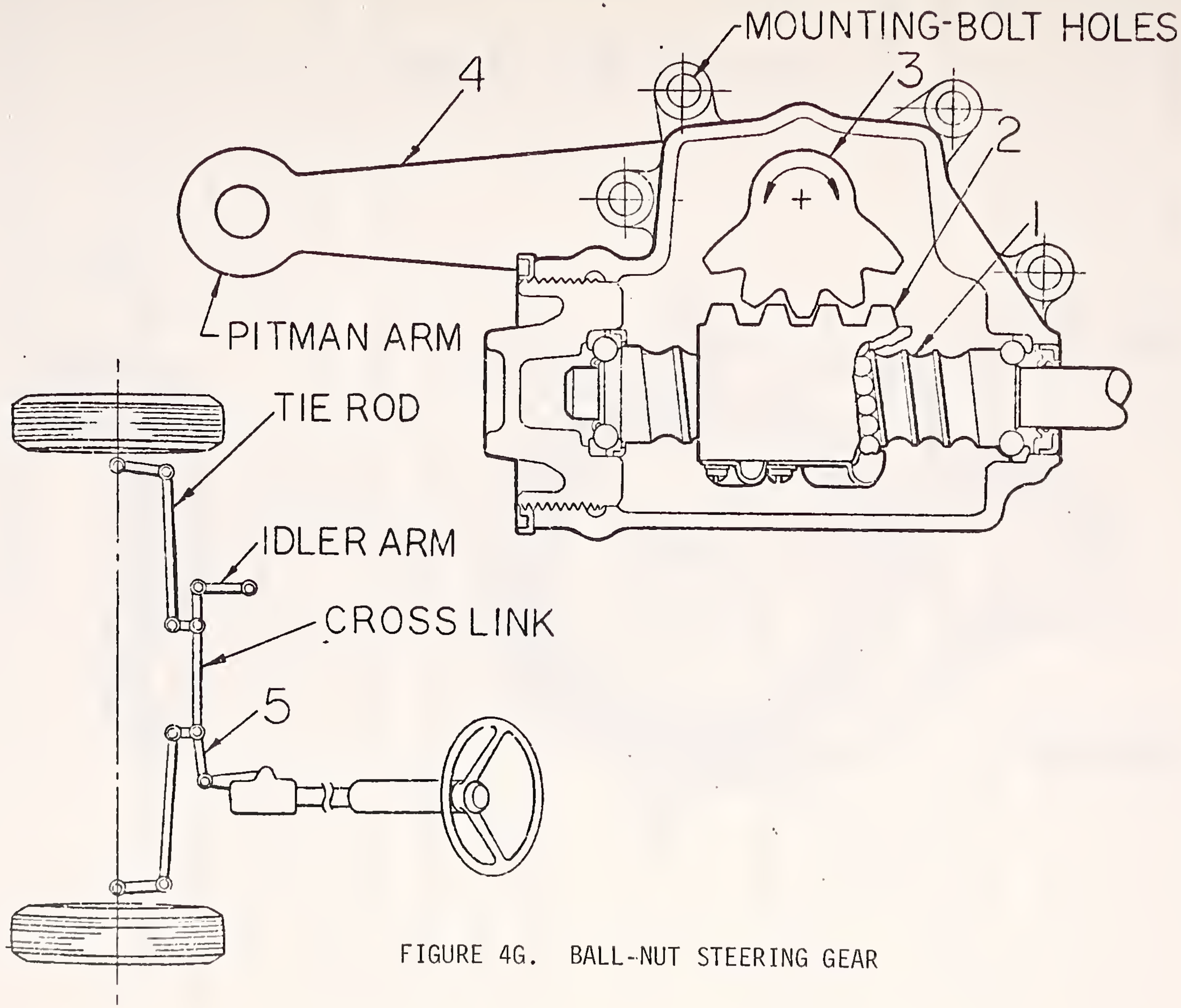


FIGURE 4G. BALL-NUT STEERING GEAR

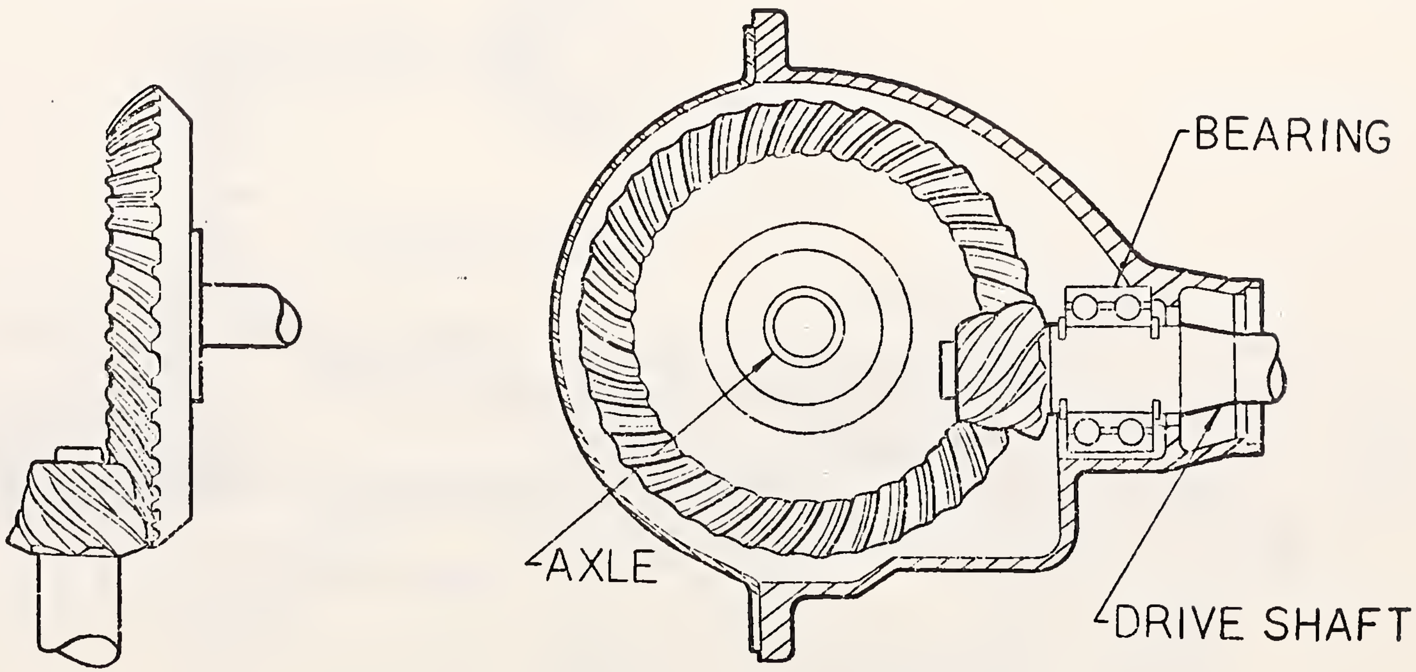


FIGURE 4H. HYPOID-BEVEL GEAR

and their housings are heavy, and the friction losses are relatively high.

Figure 4I shows a chain drive, in which the driving and driven shafts are parallel instead of orthogonal to one another. At any given instant a large number of teeth on each sprocket are engaged in driving, and the location of the two shafts with respect to each other is uncritical, with reasonable tolerances required only in radial spacing. Lubrication is similar to that of any gear system, and the chain drive can be lubricated from the same lubrication system as a transmission. Losses are moderate and tend mostly to be proportionate to load, hence the part-load efficiency is good.

Because of their geometrical dissimilarity the two systems are difficult to compare directly in weight, but the fundamental weight advantages of the chain are apparent from the above considerations.

Figure 4J illustrates the general layout of a transverse-engine front-drive powertrain unit.

These examples illustrate a couple of important design fundamentals for weight-saving: avoid overhanging, cantilevered loads, especially in heavily-loaded rotating systems such as the hypoid gear, avoid unfavorable mechanical advantages which increase stresses, and multiply compliance, requiring heavy, rigidly-aligned housings and close tolerances, such as the ball-nut steering gear; avoid redundant components and pivots in kinematic systems, such as the SLA front suspension, and the conventional steering gear, linkage system. Simplicity is not only weight and cost saving, it is also a way to improve reliability and durability.

4.8 REVIEW OF DESIGN DECISIONS AGAINST CRITERIA

4.8.1 Package Dimensions and Roominess

We have noted those ways in which the front-drive design chosen is superior to the rear-drive in roominess. These relate to the absence of the

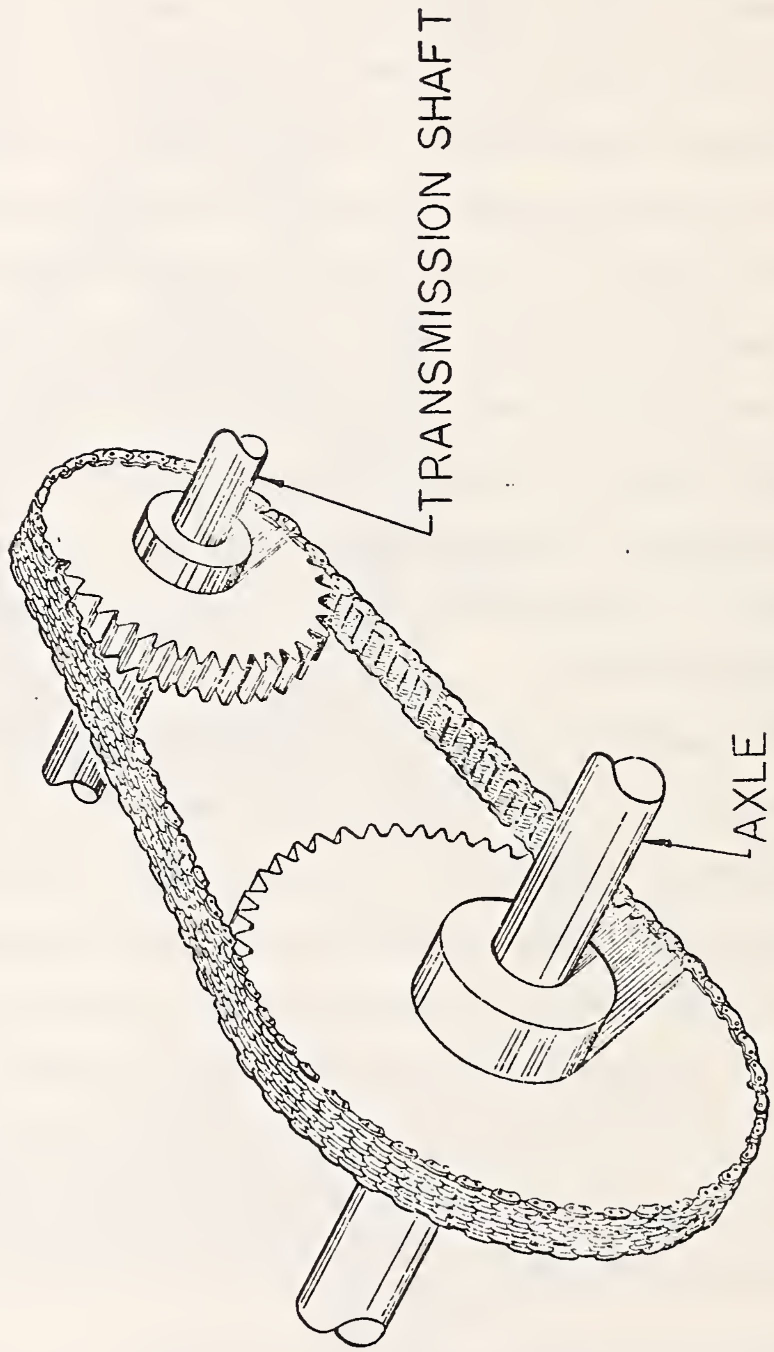


FIGURE 41. CHAIN DRIVE

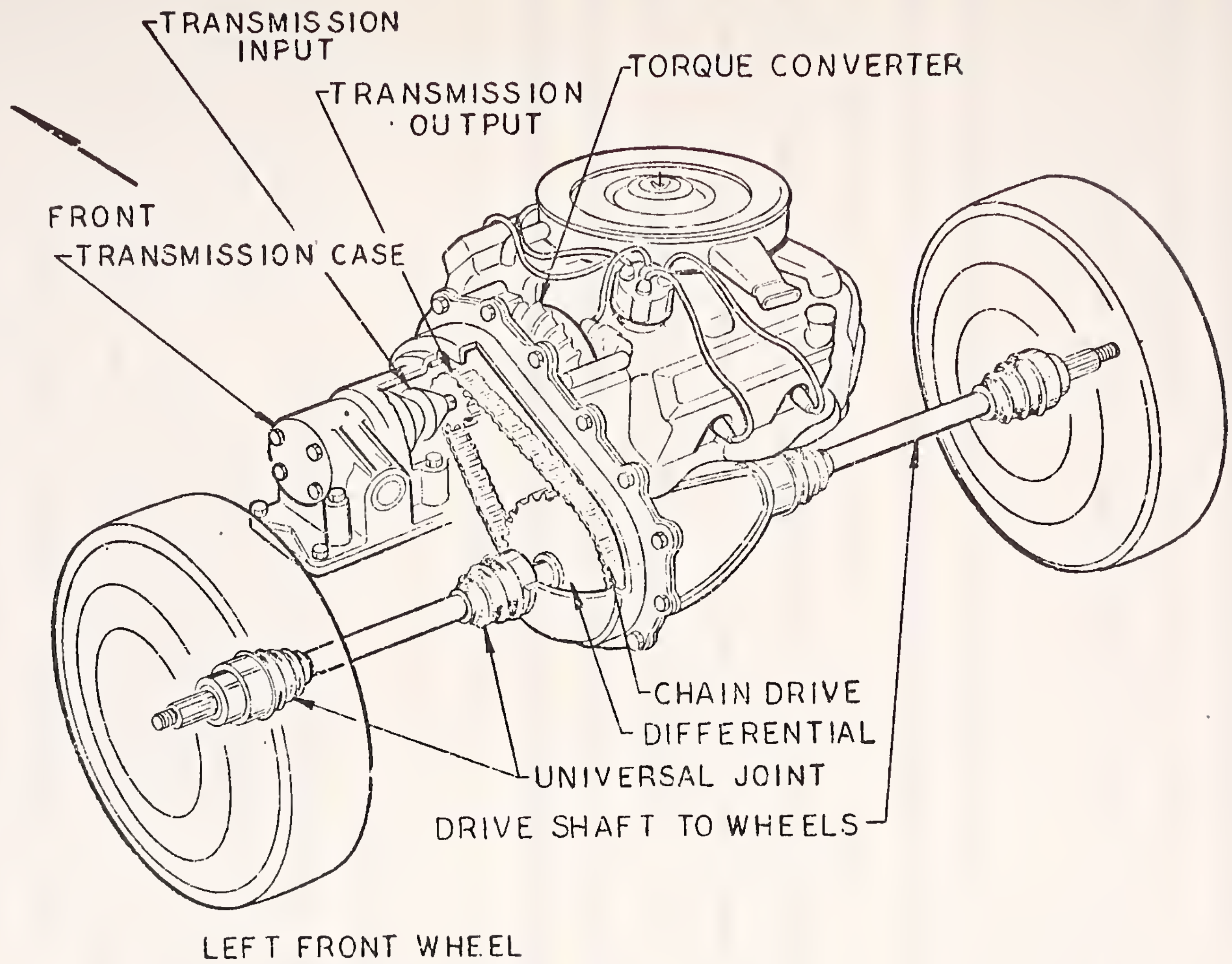


FIGURE 4J. GENERAL LAYOUT OF A TRANSVERSE-ENGINE FRONT-DRIVE POWERTRAIN UNIT drivetrain and its components under the passenger-compartment floor. There is no transmission tunnel in front, and it is not necessary to thin out the seat cushions over the tunnel area. In the rear there are no restrictions needed on rear-axle movements such as those required in the rear drive to keep the axle driveshaft extension from striking the floor of the passenger compartment, nor is there a tunnel under the rear seat cushion to accommodate this movement.

In front-drive designs there is greater freedom of location of the rear wheels without the penalty of extended propeller-shafts, so that the rear seats may be well forward of the rear wheelhouses and thus unrestricted as to hip and shoulder room. Since in the front-drive the rear axle may be dropped below the wheel centers and small in diameter, little space is subtracted from the interior package to allow for its vertical movement. This permits a safe and convenient location for the fuel tank without restriction of trunk space, hence the front-drive enjoys better trunk volume than an equal-size rear-drive.

4.8.2 Active Safety

Front-drive steering characteristics differ from those of the rear-drive in several respects that favor safety. In dynamic stability the front-drive shows greater stability than the rear-drive by virtue of its front-heavy weight bias, but this also detracts to some extent from quick steering response so that it protects the unskilled driver more than the skilled one.

When there is a loss of adhesion of the driving wheels, the front-drive car goes in a straight line almost without regard for the direction of steering, while in the rear-drive the vehicle loses directional stability and goes into a skid.

The static stability of the front-drive is notably superior to that of the rear-drive, which is an important safety factor in maintaining the directional stability of the car when the driver's attention is momentarily diverted from the road. The front-drive actively seeks a straight-line course by virtue of friction in the differential.

For the above reason and because of its greater forward weight-bias the front-drive requires greater steering effort than the rear-drive and therefore stands more in need of power steering than the rear-drive. It is our opinion, however, that cars in the 5-passenger class require power steering in either case.

The front-drive is at some disadvantage with respect to braking effect distribution due to its forward weight bias and the consequent greater weight-variation in the rear wheels, so that a smaller proportion of total effort is on the rear wheels than is required when fully loaded. It is possible to deal with this condition at moderate cost penalty and negligible weight penalty by a proportioning device, but the best way to deal with it is with a non-skid system on the rear wheels, which we believe to be one of the most important safety additions that can be made to any car.

A positive advantage of the front-drive car as compared with the rear drive with conventional live rear axle resides in its superior rear suspension. This lacks the excessive and badly-placed unsprung weight of the live axle, and so the wheels hold the road better on a rough road, thus reducing the probability of a loss of adhesion in the rear, with consequent loss of vehicle stability. This advantage more than compensates for the above-cited weight-distribution effect on all but the smoothest of roads.

Acceleration capability on normal roads is dependent on power/weight ratio and choice of transmission capabilities, as well as on carburetion and powerplant characteristics, and is not affected by vehicle configuration. We have discussed the matter of transmission characteristics earlier.

We have also discussed the matter of traction on slippery roads. Here the front-drive configuration has the advantage over the rear drive in all but the maximum loaded condition. On rough roads or at speed the superior wheel adhesion of the front-drive driven wheels over that of the live axle constitutes an additional advantage in favor of the front-drive. As noted in Appendix 4-1 the live axle imposes an additional degradation of adhesion due to the effect of the propeller-shaft torque tending to lift the right rear wheel.

4.8.3 Passive Safety

As we have stated, we are not qualified to discuss crashworthiness in detail, and it is discussed in references cited in 1.6 and 1.9. However, we believe that the concentration of powertrain weight ahead of the bulkhead in the front-drive design is an advantage in most frontal impact situations, and that crashworthiness can be improved at moderate weight penalty by lengthening the vehicle ahead of the bulkhead by increasing the distance from the bulkhead to the powerplant.

4.8.4 Economy of Ownership

We perceive no important weight difference between front and rear drive designs. However, reference 1-5 discusses means of improving fuel economy by reductions in cppm, and we have shown that this recourse has been resorted to generally by domestic makers. We have shown that the use of smaller engines at normal erpm rather than reductions in erpm cannot only reduce cppm but can also reduce car weight. To make use of smaller engines without reducing available horsepower, however, requires overhead-cam engines, which are heavier for their displacement, while lighter, but more expensive, for their power.

Durability and reliability are subjects for separate study, and cannot be dealt with adequately here. However, they are of overruling importance in car design, and weight-saving measures which adversely affect either of them are not permissible. As we have remarked, durability and reliability can be definitively established only by statistical studies of service in the hands of owners, and this is a lengthy and expensive process for new designs. We have not knowingly advocated any design measures for weight reduction that present questions of this nature.

Cost of maintenance and repair is to a considerable extent a matter of accessibility as well as of what the repair facilities of this country are accustomed to working on. Any new design, in addition to facing the hazards we have already discussed, faces the hazard of the independent service man who is unfamiliar with it.

Front-drive cars face this kind of handicap as well as a handicap of accessibility. The advantage of having all the powertrain components under the hood has already been recited, but the penalty for this is additional crowding, at the expense of accessibility. It is lighter to have a transmission and final drive integral with the engine, but it is likely to be more difficult to remove the transmission, torque-converter, or final drive for repair. Making the entire power package removable, as in the VW Beetle, is advantageous, but it becomes more difficult to manage in more complicated installations with air-conditioning and engine-driven auxiliaries, heating, vacuum connections, etc. Thus we believe that cost of maintenance and repair in larger front-drive vehicles is at present a handicap and a problem to be solved. It must be added, in fairness, that large, V-8 rear-drive luxury vehicles do not represent ideal solutions to this problem either.

4.8.5 Optional Functional Characteristics

Front-drive cars have a riding-comfort advantage because of the superiority of their rear suspension, for reasons already discussed. Far less compromise of comfort for stability is required for the light-unsprung rear suspension. Also, the front-heavy bias of the front-drive car makes it less susceptible to pitching, and offers a better anvil for road-noise reduction. The absence of the propeller shaft, with the vibration problems resulting from unbalance and universal-joint moments is a further advantage in the suppression of noise and vibration. The hypoid gear in the usual live rear axle presents a potential noise problem greater than the same gear would present as part of a unit powertrain of greater mass than the differential housing.

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ADDENDUM
COMPUTATION OF REQUIRED FRICTION COEFFICIENT, TIRE TO ROAD, FOR ACCELERATION AND HILL-CLIMB, FRONT AND REAR DRIVE.

$$z_r = \frac{x}{G} - y$$

$$\mu_r = \frac{W}{z_r} = \frac{WG}{x-Gy} = \tan\beta$$

$$z_f = \frac{W-x}{G} + y$$

$$\mu_f = \frac{W}{z_f} = \frac{WG}{W-x+Gy}$$

$$z_r' = z_r\left(1 - \frac{R}{nt}\right)$$

$$\mu_r' = \mu_r\left(\frac{nt}{nt-R}\right)$$

W = wheelbase

G = grade % = $\tan\alpha$ = acceleration/g

x = horizontal distance, front wheel to C.G.

y = vertical distance, road to C.G.

R = effective radius, drive wheel

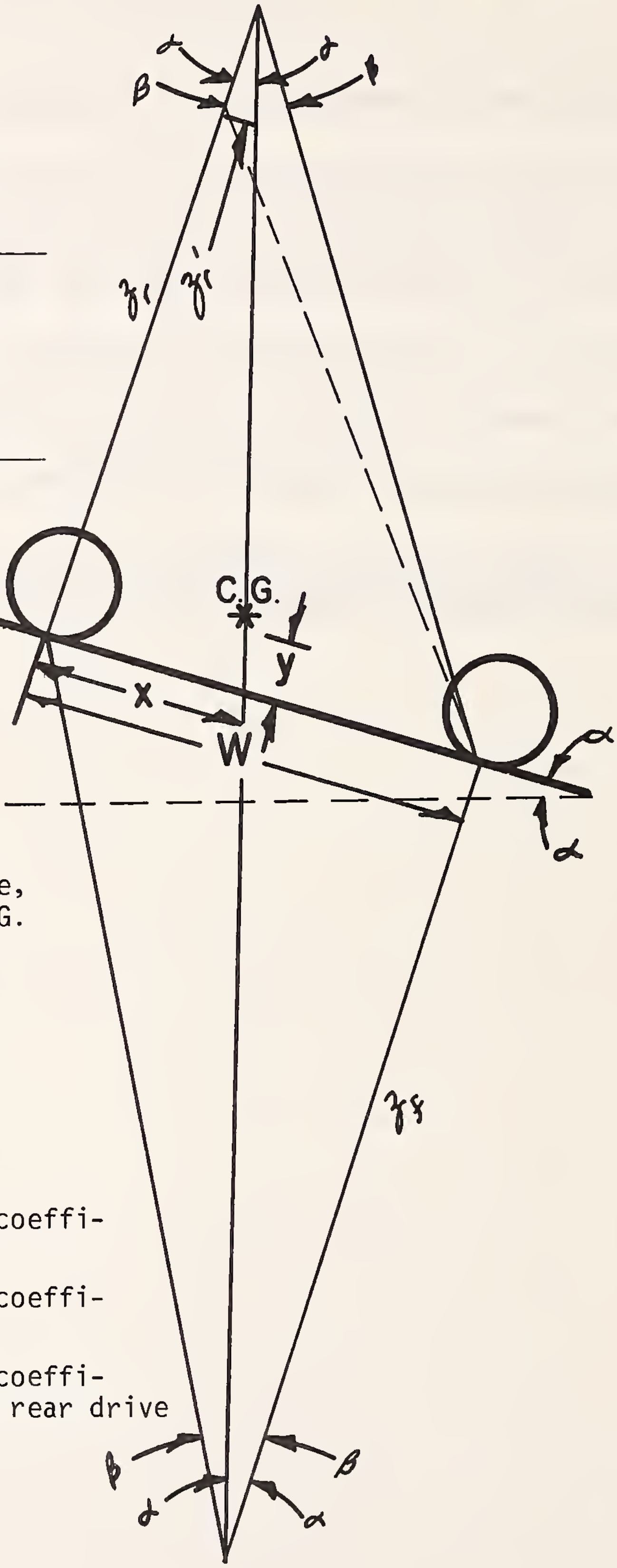
n = rear axle ratio

t = rear axle track

μ_f = required friction coefficient front drive

μ_r = required friction coefficient rear drive

μ_r' = required friction coefficient solid axle, rear drive



5. WEIGHT REDUCTION BY MATERIAL SUBSTITUTION

5.0 INTRODUCTION

Three main approaches to weight reduction of functionally-equivalent vehicles are being considered:

(a) reduction of non-functional size

(b) improvement of design

(c) substitution of materials.

The first two approaches have already been discussed in some detail. In this section the substitution of different materials for currently-used metals will be considered. The weight savings that could result from material substitution are assumed to be above and beyond those achieved by the weight-conscious design procedures discussed in earlier sections of this report. It should be noted that the total weight reduction that could result from a combination of design improvements and material substitution may be limited by the functional and safety constraints discussed in Section 1.

There are two major ways to achieve weight reduction by means of material substitution: use of lighter weight materials with reasonable strength, and use of higher strength materials of reasonably low density. Of the lighter weight materials, the most promising from a cost, availability, and strength viewpoint appear to be aluminum alloys and plastics (including fiber-reinforced composites). The best possibilities from among the higher strength materials appear to be the high strength, low alloy (HSLA) steels that have been developed in recent years. These three material groups, aluminum, plastics, and HSLA, will be considered in more detail below.

Substitution of either a lighter weight material, such as aluminum or plastic, or a higher strength material, such as HSLA, in a component currently made of carbon steel, cast iron, or even zinc, can lead to a lighter component. Lighter components require less structural support from other vehicle subsystems, such as chassis, suspension system, tires, etc., so these components may be reduced in size. Utilization of lighter components may also allow a smaller, less powerful engine to be used. Thus a secondary weight reduction may result. Use of the influence coefficients developed earlier in this report shows the magnitude of this reduction. Coefficient Q1 gives an indication of the total reduction in curb weight resulting from a reduction in the weight of a vehicle part. The functional expression for Q1 developed in Section 3.2 is

$$Q1 = \frac{P(1+C)}{P-m-Cm}$$

where C and m are the weight-dependent factor and the power dependent factor, respectively, (Section 3.1.4). The statistical analysis of existing designs carried out in Section 3.3 showed that $C \approx 0.2$, $m \approx 1.85$, and P ranges from 12 to 30, depending on the vehicle. Use of these values in the above expression for Q1 gives $Q1 \approx 1.4 \pm 0.1$. Thus, a one pound reduction in the weight of a vehicle component will lead to approximately a 1.4 pound reduction in vehicle curb weight.

5.1 ALUMINUM ALLOYS

Aluminum and its alloys have been used in automobiles for many years, but the use of large amounts of aluminum in mass-produced vehicles has never become widespread. In recent years the amount of aluminum used in automobiles has been increasing, from 12 lb/car (average) in 1950 to about 87 lbs per car in 1976. Still, however, aluminum makes up only 2 to 2½% of an automobile's total mass, compared with over 50% for plain carbon steels and 10-15% for cast iron.

The density of aluminum, about .1 lb/in^3, is much lower than that of carbon steel (.284 lb/in^3) and cast iron (.266 lb/in^3), indicating a potential for possible weight reduction by substitution of aluminum for iron or steel in some components. The material substitution decision is not a simple one, however, with many factors being involved, including: amount of material necessary to meet strength and/or rigidity requirements, cost and availability of material, ease of production, and durability. Other considerations such as energy requirements for production and recyclability may also enter into the decision-making process.

In order to evaluate completely the potential for aluminum usage in automobiles, it is necessary to look at each component separately, analyze its requirements, and determine if an aluminum alloy is able to meet those requirements. Such a complete evaluation is not possible in this report, so a more general evaluation based on component groups will be described.

5.1.1 Body Parts

Since a large portion of an automobile's weight (up to 45%) is concentrated in the body components, it is a natural target for weight reduction programs. Several aluminum alloys have been proposed by the aluminum industry especially for use in vehicle bodies (5-1): 2036 (containing 2-3% Cu and less than .6% of other elements), and two alloys containing over 2% Mg, 5020 and 5182. It has been proposed that the latter alloy be used in its annealed condition with the former two being used in T4 temper. Typical yield strengths for these materials after a forming and paint bake cycle range from 28 ksi (for 5182) to 42 ksi (for 2036-T4) (Ref. 5-1), and therefore are not too different from the yield strength of standard low carbon steel.

Body parts, in general, are not designed on a basis of strength alone, but for a combination of strength, stiffness, and dynamic response characteristics. Because of this, aluminum cannot usually replace steel on a gage-for-gage basis but must often be thicker than the steel it would replace. In a report published in 1974 (5-2), engineers from an aluminum company described the design of an all-aluminum automobile body to replace an all-steel body for one particular configuration. It was shown that for some body components, such as doors and outside body skin parts including hood and trunk lids, significant weight reductions can be achieved, with aluminum components weighing up to 60% less than the equivalent steel component. Other components, however, especially structural parts such as sills, saw very little weight decrease upon substitution because the aluminum components were much thicker than the equivalent steel part. The thickness increase was due mainly to the necessity to maintain good stiffness with the low modulus aluminum ($E_{Al}/E_{steel} \approx 1/3$). Additional increases stemmed from the requirement that spot welded joints maintain sufficient strength. Since aluminum welded joints are frequently weaker than comparable (same gage) steel joints, the aluminum flanges must often be thicker than the steel to get equivalent joint strength. The use of other joining methods, to be discussed later, could eliminate much of this thickness increase at joints.

The above-mentioned study concluded that an <u>all-aluminum body</u> would weight 39% less than the equivalent steel body, for the vehicle configuration investigated. There would be a secondary weight decrease in the suspension, etc., so that the total weight reduction could be about 55% of the steel body weight ($1.4 \times .39 = .55$). For other vehicle designs the total weight reduction achieved by using an all-aluminum body would probably differ from this 55% figure, with even greater weight reductions being anticipated in many cases.

The total cost of the aluminum-bodied car would be greater than that of the steel car, however, since the aluminum body alloys cost about 4 times as much as the carbon steels they would replace (5-3). The total cost of materials for the aluminum-bodied car could be up to 1.3 times that of the original steel-bodied car (Appendix 5A), assuming that 40% of a steel-bodied car weight rests in the body. If manufacturing costs remained the same, this material cost increase would cause a higher selling price for the aluminum-bodied car. The increase in selling price, plus the additional problems of availability of the large amounts of aluminum that would be required and the unresolved production difficulties with some aluminum parts (to be discussed later) lead to the conclusion that an all-aluminum vehicle body is not feasible at this time.

An alternative to an all-aluminum body is a composite steel-aluminum body. Because little weight reduction results from a material change in the structural parts of the body, those parts will probably remain as steel. Since there is a strong possibility of galvanic corrosion at steel-aluminum joints in the presence of corrosive environments, such as road salt solutions, it is probable that body parts joined intimately (welded) to the structural steel parts will also remain as steel. Various bonding techniques, such as adhesive bonding, which would insulate steel from aluminum at joints, have been proposed to eliminate the corrosion problems, and proper painting procedures can inhibit corrosion, but at the present time it seems that it is better to avoid steel-aluminum joints in potentially troublesome locations.

The most promising body parts for conversion to aluminum are the "hang-on" components - hoods, doors, load decks, fenders and deck lids. Auto manufacturers have recently marketed vehicles with aluminum inner hood panels (Toronado and Feather Duster) or aluminum load decks (Vega), and all manufacturers have development programs underway on aluminum "hang-on" body sheet applications.

Since most of the "hang-on" parts do not have as severe strength or rigidity requirements as do the structural parts of the body, a switch to aluminum would not mean large increases in thickness, so that significant weight reductions could potentially be attained. A 33 lb savings resulted from use of an aluminum inner hood panel in the 1975 Toronado (5-3) while use of an aluminum alloy load deck in the 1973 Vega resulted in a 65% weight reduction of that component (5-4),

Because the aluminum alloys used in these substitutions cost about 4 times as much as the original carbon steel, even large weight reductions are not usually enough to avoid a material cost penalty arising from the substitution. Assuming that the substructure, tires, etc., are reduced in weight when a lighter body component is produced from aluminum, it can be shown that the weight decrease must be at least 62.5% if a material cost penalty is to be avoided (see Appendix A). A decrease this large will seldom be achieved in practice but, in a well designed component, significant weight reductions could be achieved with only a small material cost penalty.

In determining the total economic effect of the change to aluminum, the fuel savings potential should also be investigated. It has been shown that a one pound reduction in vehicle weight can lead to a 1.4 gallon savings in fuel over the vehicle lifetime (100,000 miles) (Ref. 5-5). Considering that the price/gallon of gasoline may average about 5 times the price/lb of carbon steel, it can be shown (Appendix A) that, for a 20% or greater decrease in component weight by a change to aluminum, the fuel savings over the lifetime of the car will pay for the increased material cost. Since weight reductions of much larger magnitude seem to be attainable, and since inflation in fuel prices over the lifetime of the car could decrease the 20% figure significantly, it is expected that most of the material cost penalty associated with a substitution of aluminum for steel would be negated within the first two or three years of operation.

Although the "hang-on" components have, in general, fewer strength and stiffness requirements than other body parts, there are some specifications that may not be met very well by a reduced-weight, all-aluminum part design. For example, the specifications for side impact resistance may require a heavy intrusion beam in an aluminum door, in order to improve the strength and stiffness of the door. While an aluminum guard beam is feasible, some Chrysler engineers have shown (5-6) that more weight reduction can be achieved by employing a higher-modulus, high-strength, steel, intrusion-resistant door-guard beam inside aluminum door panels.

A number of production problems must be overcome before the weight reduction potential of aluminum in doors, hoods, trunk lids, etc., can become reality. The aluminum alloy most often promoted for external body sheet applications (2036) has, in the eyes of many auto industry materials engineers, inadequate formability for use in some typical hood and deck lid outer panels. Low ductility and variable springback have been frequent causes of poor quality outer panels and high scrap rates in recent development programs.

The 5182-0 alloy has better formability than 2036, but its susceptibility to the formation of Lüders bands during forming makes it ill-suited for use in most external body panels. The Lüders bands cause an uneven surface appearance that persists after painting. This condition matters little for inner body panels, for which these alloys are prime contenders, but their usefulness in exterior applications is limited. A new version of the 5182 alloy, for which the Lüders band problem is much less severe (5-7), is currently being evaluated in exterior body panel prototypes. The alloy, designated 5182-SSF, has mechanical properties which are similar to those of conventional 5182-0. Both 5182 alloys are more formable than 2036-T4, but have lower yield strength and are less dent-resistant. Although each of the alloys proposed for body sheet alloys is the subject of large-scale development programs, both in the automobile and aluminum industries, no aluminum external body parts have yet gone into production. It is felt that until significant success in production runs (scheduled to begin with one 1977 model) is achieved, the application of aluminum in exterior hood, trunk lid, and door panels may be limited.

Another difficulty that must be overcome with the aluminum body alloys is in the area of high production welding and joining. Due to the presence of oxide layers on aluminum surfaces and to the high thermal conductivity of aluminum, high current levels are required in spot welding the material. This contributes to short electrode lifetimes and low productivity. Although much improvement has been achieved in recent years in electrode longevity and in alloy weldability, special welding practices must be followed with most of the proposed body materials in order to get satisfactory welds and production rates. As mentioned earlier, aluminum sheet metal spot-welded joints must also frequently be thicker than steel joints to achieve equal weld strength. More improvement in high production welding methods for aluminum seems to be desirable.

An alternative joining method for automotive body panels is adhesive bonding. An advantage of structural adhesive bonding is that stress is more evenly distributed along the joint and that the body panels are better able to carry load and to contribute to the stiffness of the body. Tests on adhesive bonding of aluminum indicate (5-8) that the bond strength is somewhat affected by cyclic temperature and humid environments. Some of the problems in getting good bond performance on aluminum parts are due to surface oxide films on the metal. Alloy 2036 seems to be especially affected by this (5-9). With the proper choice of adhesive, however, relatively strong and durable adhesive bonds can be obtained with each of the aluminum body sheet alloys, especially if the surfaces are pretreated (5-9).

Despite a limited amount of production experience with adhesives and some unresolved quality control questions, it is expected that adhesive bonding will be employed more frequently in the future, especially as a weight reduction measure. It is very possible that bonding will be combined with spot welding in order to avoid the need to keep the bond fixtured while curing and at the same time to achieve stronger, stiffer, more uniform, "weldbonded" joints with fewer spot welds. Improved production methods will have to be perfected before this becomes widespread, but it has already been proven feasible on a small scale for both aluminum and steel joints (e.g., Vega's weldbonded aluminum load floor).

A number of the proposed aluminum body parts will probably be joined to other components by mechanical means. One method that has proven successful for joining aluminum body panels to steel, or aluminum to aluminum, is by mechanical spotclinching (5-14). Other mechanical fasteners, such as steel clench nuts, possibly phosphate-coated to cut down the susceptibility to galvanic corrosion, have also been proposed for use with aluminum body panels (5-2,5-14) and have been tested in prototypes. Self-piercing rivets are another alternative being investigated.

Mechanical hem-flanging was initially thought to be a poor method for joining aluminum body panels because of the cracks that were frequently encountered at the aluminum hems. It seems that modified hemming techniques have largely overcome the cracking problems, since a representative of the Oldsmobile Division recently stated that they believe hemming is the best way to join their proposed aluminum inner and outer hood panels (5-10).

Other factors that may enter into a decision to adopt aluminum body parts are dent resistance and repairability. Although dent resistance varies from alloy to alloy, it can be stated that, in general, the dent resistance of aluminum is somewhat less than that of steel at both low and high strain rates. This may force most aluminum exterior body panels to be thicker than equivalent steel panels. A recent study (5-11) concluded that panels made from 2036-T4 aluminum must be 10-13% thicker than equivalent 1010 steel panels in order to have equal dent resistance. Equivalent 5182 alloy panels would have to be even thicker. Repairability is a problem which should diminish in importance when manufacturing and service personnel have gathered more experience working with aluminum. One repair item which requires more development work is a good filler for use in repairing small dents in exterior aluminum body panels.

Since many of the production problems mentioned above seem amenable to solution in the near future, and since both the aluminum and automobile industries seem to assign a relatively high priority to the use of aluminum in hang-on body parts, it is felt that total usage of aluminum in body parts will increase significantly in the future. A breakdown of weight savings that could be attained by such substitution will be given in Section 5.1.5. It should perhaps be noted that lighter-weight hang-on components, such as hoods and trunk lids, will probably be welcomed by people who have occasion to lift or open these components and could also lead to further weight reduction in the hinges, counterweights, etc., used with the hang-on components.

5.1.2 Bumpers

In recent years, due in large part to industry's response to government regulations, bumpers on American cars have become much heavier. For example, the Pinto bumpers increased from 60 lbs in 1973 to 147 lbs in 1974. Since the regulations are probably not going to be rescinded, any weight reduction in bumpers will require redesigns and material substitutions. The auto industry seems to be seriously considering many new, lighter weight, bumper system designs. Most of the designs make use of energy absorbing plastic materials and/or HSLA steels and/or aluminum, with each design having its advocates. As might be expected, there is a wide variance of opinions on the future of bumper designs and materials (5-12).

In the past few years several production cars have been equipped with bright anodized, one-piece extruded aluminum bumpers (Vega, Camaro, and Astre), with some of them having a rubber facing strip. Other vehicles (nearly 10 American models in 1976) use aluminum reinforcing bars in their bumper systems. Significant weight reductions have resulted from these applications. For example, each of the two aluminum reinforcing bars in the Feather Duster weighs about 13 lbs, which is 23 lbs less than the steel re-bar it replaces (5-10). In addition to these production bumpers, development programs are underway in which aluminum face bars, as well as reinforcements and one-piece extrusions, are being evaluated on other vehicle prototypes.

Bumpers are designed mainly on a strength basis, so the aluminum alloys being used and considered for bumper applications are high strength alloys, including 6061, with yield strength of 40 ksi in T6 temper, 7016 (46 ksi, T5 temper), and 7046 (55 ksi-62 ksi, T63 temper) (Ref. 5-1). Although other new high strength aluminum alloys are also being evaluated for potential bumper applications, most of the current aluminum effort seems to be centered on the two zinc-containing alloys 7016 and 7046. The 7016 alloy is being used in each one-piece extruded bumper on 1976 cars but, since the alloy is available only in extruded form, its applicability is more limited than that of 7046, which can be supplied as either plate or extrusion. Most of the current and proposed reinforcing bars as well as the proposed face bars use 7046.

Several factors could limit the amount of aluminum used in bumpers in the near future. In the opinion of some bumper design engineers, the formability of 7046 sheet is inadequate for some bumper applications (5-3). It is understood that development programs are underway in an attempt to improve the tool design and forming practices for the alloy. New alloys with more formability but less strength than 7046 are also being evaluated for bumper applications.

Another factor which could deter the adoption of aluminum bumpers is surface appearance. Although a lustrous, bright-anodized finish can be produced on some of the bumper alloys, in particular 7016, a number of designers would prefer a chrome-plated bumper, which they feel has more decorative appeal (5-3). No commercial chrome-plating process had yet been adopted for these high strength aluminum alloys, although development of such a process has evidently reached an advanced stage.

Despite the formability and appearance drawbacks, the use of aluminum alloys in bumpers will probably continue to increase rapidly during the next few years, particularly in reinforcing bars and in extruded one-piece bumpers of simple shape. The total usage of these alloys will probably depend on the success of the chrome-plating and forming practice development programs, on the incentive for bumper designers to modify bumper designs to enable use of lighter materials, and on the competition from light-weight bumper designs utilizing other materials, such as HSLA and energy-absorbing plastics.

5.1.3 Castings

The majority of the aluminum presently used on automobiles is in the form of castings. Automatic transmission housings, carburetors, fuel pumps, distributor housings, and alternator housings are frequently aluminum castings, with many of them being die cast. The largest single aluminum component on an American car is the sand cast Vega engine block. In recent years, other aluminum castings have begun to appear in some production vehicles. The 1976 Chevette, for example, has a die cast camshaft housing and a sand cast intake manifold on its overhead cam engine, along with a die cast cover and extension on its manual transmission and a die cast steering-gear housing. Weight reductions resulting from these aluminum applications, e.g., a 35 lb saving in the manual transmission (5-10), help keep the Chevette's curb weight at about 2000 lbs, lowest of any U.S.-built car.

It is expected that the number of aluminum castings on new automobiles will continue to increase in the coming years. Some components, such as steering gear and alternator housings, oil pumps and intake manifolds, will probably more and more frequently be cast from aluminum. Other components, such as cylinder heads and manual transmission covers may also be introduced as aluminum castings, and some items now appearing as zinc could possibly be changed to aluminum. It seems rather doubtful that many more cast aluminum engine blocks will be introduced by American automakers in the near future, however.

Development programs leading to material substitutions in cast components will probably be initiated by the automobile industry itself, where much of the casting expertise for these materials, including aluminum, can be found. This is in contrast to the wrought components mentioned earlier, where the primary aluminum industry will provide much of the initiative.

Substitution of aluminum in cast components could be affected by several factors. A major deterrent could be cost, since a material cost penalty would be incurred in most switches from cast iron to aluminum. This cost increase could be overcome if the aluminum part could be made as a die casting, with fewer costly machining operations than an equivalent sand casting. A factor that could work in aluminum's favor is a continuation of the shortage of iron casting capacity.

The problem of porosity that has been encountered in some aluminum castings will probably be overcome by impregnating critical castings with epoxy or similar substances after machining. Except in severe applications, porosity problems should not prevent substitution of aluminum for cast iron. One factor that will limit such substitution is the relatively low temperature range in which aluminum can be used. This will prevent its use in such components as the exhaust manifold.

5.1.4 Other Components

A number of other vehicle components are presently made from aluminum, such as air conditioner components, electrical wire and cable, heat shields for catalytic converters, brake splash shields, seat parts, and trim items. Use of aluminum in such applications will probably continue to grow. Other components could also appear as aluminum, including air cleaner covers, forged aluminum wheels, radiator support brackets and engine brackets. The determining factor in a material substitution decision for most of the above parts will probably be cost.

Several other components could be made from aluminum if development programs improve the feasibility. For example, radiators made from aluminum instead of copper have recently been introduced (5-13) and have begun to appear on production models of several European cars. Before aluminum radiators can be accepted for use on American cars, however, a better method will have to be found for repairing them and stopping leaks, both in the production shop and in the field.

5.1.5 Potential for Aluminum Usage

As was indicated earlier, the feasibility of the substitution of aluminum for a currently-used material in a given vehicle component, and the weight reduction that could result from such substitution, is governed by a number of factors. Important factors include cost, availability, production difficulties, technical problems during service, and competition from other materials. Although cost, availability, and competition could certainly limit aluminum usage in automobiles, these factors will not be considered here and the potential usage will be examined only from a technical point of view.

Four categories of vehicle components will be considered.

I. Components currently made from aluminum on a large percentage of passenger vehicles.

II. Components for which an immediate switch to aluminum would present no major technical problems.

III. Components for which technical problems exist but will probably be overcome by development efforts during the next few years.

IV. Components for which major technical problems exist that will likely impede any widespread switch to aluminum.

The estimated weight of the components that fall in each of the first three categories will be given for the Op 4, Op 5, and Op 6 designs developed earlier in this report. This will enable an estimate to be made of further weight reduction through material substitution that could be achieved in these weight-conscious designs.

Category I Components

A number of automobile parts are now made predominantly from aluminum. In the Optimum 4, 5 and 6 weight-conscious designs discussed earlier, it is assumed that aluminum is the incumbent material for some components. A list of these aluminum components and their approximate total contribution to the weights of the weight-conscious designs of Section 3 is given in Table 5A.

TABLE 5A. CATEGORY I COMPONENTS AND WEIGHTS

Components: Pistons, transmission cover and extensions, distributor and pump housings, electrical wire and cable, moldings and other trim items, miscellaneous small parts.

Approximate Weights:	Op 4	Op 5	Op 6
	57 lb	67 lb	83 lb

It may be noted that these weights fall below the 84 lb per car average quoted earlier for aluminum. There are two reasons for this. The weight-conscious designs for which the figures were given do not include optional equipment such as air conditioners, which are big users of aluminum. In addition, the weight-conscious designs do not have any of the large aluminum items, such as engine block or bumpers, which are found on some vehicles and which are partly responsible for the increase in aluminum usage during the past few years. Many of these items are included among the Category II components in this discussion.

Category II Components

Development programs and small-scale production runs in recent years have shown that some components can be made from aluminum without encountering major difficulties in either production or service. In particular, inner panels for hoods and trunk lids, one-piece extruded bumpers, bumper reinforcing bars, castings, and other small components have already been used successfully on selected production automobiles, as was indicated earlier. These components will be considered to make up Category II, and they are listed, along with the associated primary weight savings and changes in curb weight, in Table 5B. In determining the curb weight change, ΔW_c, the factor Q1 $\approx$ 1.4 was used.

Although the feasibility of the aluminum Category II components has been demonstrated in production vehicles, it is unlikely that all of these components will be made from aluminum on every American car. In particular, it is doubtful whether aluminum will capture the wheel market completely, especially in view of the relatively small weight reductions that have been achieved by use of aluminum wheels in recent years. These less probable items are identified by * in the table.

Category III Components

Extensive efforts are now underway to develop improved aluminum alloys and production methods that could enable regular use of aluminum in components for which it has not yet been proven completely feasible. It is expected that these development programs will meet with enough success to permit some of the aluminum components now being tested on prototypes to go into production on a limited scale within the next few years, after the remaining minor technical problems have been worked out. Successful production trials would then enable more widespread use of these aluminum components, which have been grouped together in Category III.

TABLE 5B. CATEGORY II ALUMINUM COMPONENTS AND WEIGHTS

COMPONENT		OPTIMUM 4			OPTIMUM 5			OPTIMUM 6		
		C	A	Δ	C	A	Δ	C	A	Δ
Inner panels of hood & deck lid		20	9	11	37	17	20	42	19	23
Bumpers without formability or appearance problems		47	21	26	50	22	28	57	25	32
Castings (intake manifold, steering gear & alternator housings, pump housings)		51	20	31	60	23	37	76	32	44
Others (air cleaner cover, hub caps, brackets, seat supports, heat shield, splash shield, trim items, etc.)		60	25	35	70	27	43	85	33	52
*Wheels		60	52	8	75	65	10	100	85	15
TOTALS	Excluding * items	178	75	103	217	89	128	260	109	151
	Including * items	238	127	111	292	154	138	360	194	166
ΔW_c	Excluding * items			-143 lb			-180 lb			-211 lb
	Including * items			-156 lb			-194 lb			-232 lb

* Indicates components in Category II least likely to be made from aluminum on industry-wide scale.
Column C - Weight of components (in pounds) made from current material.
Column A - Weight of components (in pounds) made from aluminum.
Column Δ - Decrease in weight (in pounds) resulting from substitution of aluminum for current material.

The Category III components are listed in Table 5C. As in the earlier table, items for which an industry-wide switch to aluminum is least probable are identified by *.

Category IV Components

It may be observed that the only body parts considered to fall in Categories II and III are the hang-on parts. It is felt that significant problems, both technical and non-technical, will keep aluminum usage in the remainder of the body relatively low. It is quite possible that the experience that will be gained in producing the hang-on parts, especially those in Category III, will lead to the use of aluminum in some other body parts, such as roof skin and rear fenders, in some cars. It is unlikely that aluminum body parts other than the hang-on items will become standard, however.

Another major vehicle component that falls in Category IV is the engine block. Although aluminum engine blocks have already been used successfully, it is felt that U.S. automakers will be hesitant to introduce any more aluminum engine blocks in the near future. Two reasons for this hesitancy are the improved capability to cast reasonably light weight cast iron engine blocks and a remembrance of the problems that have plagued some aluminum engine blocks in recent years. Many other engine parts will be made from aluminum, however, as can be seen from the list of castings included under Categories II and III.

Other vehicle parts which will continue to be made from materials other than aluminum include components of the exhaust and suspension systems.

It should be emphasized that considerations of cost, availability, and competition from other materials did not enter into the above evaluation. All of these factors could prove important in determining whether the potential

TABLE 5C. CATEGORY III COMPONENTS AND WEIGHTS

COMPONENT		OPTIMUM 4			OPTIMUM 5			OPTIMUM 6		
		C	A	Δ	C	A	Δ	C	A	Δ
Outer panels of hood & deck lid		27	12	15	51	23	28	57	26	31
Bumpers not included in Category II		25	10	15	40	18	22	53	23	30
Castings (mainly cylinder head)		30	12	18	40	15	25	55	20	35
Others (radiator & radiator supports)		28	14	14	37	18	19	45	22	23
Doors(4) - inner & outer panels		95	41	54	114	50	64	120	52	68
*Doors(4) - intrusion beams		15	10	5	20	14	6	28	19	9
*Fenders - front (2)		18	9	9	30	15	15	40	20	20
TOTALS	Excluding * items	205	89	116	282	124	158	330	143	187
	Including * items	238	108	130	332	153	179	398	182	216
ΔW_c	Excluding * items			162 lb			221 lb			262 lb
	Including * items			182 lb			250 lb			302 lb

* Indicates components in Category III that are least likely to be made from aluminum on industry-wide scale.
Column C - Weight of components (in pounds) made from current materials.
Column A - Weight of components (in pounds) made from aluminum
Column Δ - Decrease in weight (in pounds) resulting from substitution of aluminum for current material.

usage of aluminum in automobiles is realized.

5.1.6 Energy Requirements for Aluminum Production

Although reduced weight will improve automobile fuel economy, there will be no net saving in energy consumption if more energy is required to produce the aluminum than is saved by the gain in fuel economy over the vehicle's life. Aluminum requires a very energy-intensive production process. The total amount of energy required to produce a part from aluminum depends on the fabrication process, ranging from 43,300 Btu/lb for castings to 102,600 Btu/lb for sheet products and 152,400 for forgings (5-5). Greater use of recycled aluminum could reduce these figures (see Section 5.1.7). In contrast, the energy required to produce steel sheet is about 25,800 Btu/lb (5-5).

Despite the high energy requirements of aluminum, use of aluminum to replace a heavier part made from steel will usually result in an overall saving in energy. For example, in Appendix B it is shown that if a steel body sheet component is replaced by an aluminum part weighing half as much, the total energy saved over the lifetime of the vehicle by the lighter component would be 3.5 times the energy required to produce the aluminum component.

5.1.7 Recycling of Aluminum

Although large amounts of energy are required to produce an aluminum part from virgin aluminum, the energy requirement for recycling aluminum is relatively small - only about 5% of the energy required for virgin aluminum. The process of recycling aluminum from junked cars has become much simpler and better in recent years with the advent of car shredders and dense media separators. It is now feasible to reclaim 85-90% of the aluminum used in an automobile.

At the present time most of the aluminum found in junked cars is in the form of castings, so the chemical composition of the recovered aluminum is similar to that of the casting alloys, which contain a high percentage of silicon and, possibly, several percent of copper. The composition of the casting alloys differs considerably from that of the 2XXX, 5XXX, and 7XXX alloys proposed for use in body components and bumpers. Therefore, when more structural alloys are found in the junked cars being recycled, the chemical composition of the reclaimed scrap may be incompatible with any of the commonly used casting or wrought alloys. Although this problem should not develop before the mid-1980's, it could prove to be an obstacle to the growth of aluminum usage.

One way to avoid the problem is to have most of the wrought aluminum alloys concentrated in components such as hang-on body parts and bumpers that could be removed and separated before the junked vehicle is crushed. This would involve more effort in the recycling process but would enable separation of most wrought aluminum parts from the cast ones. It is unknown whether this procedure would enable the chemical composition of the recycled scrap to be kept within the tolerances required for its reuse in automobile components.

This recycling problem certainly requires more R & D effort before aluminum-laden cars start appearing in greater numbers at car recycling plants.

5.2 HIGH STRENGTH, LOW ALLOY STEEL

Although the use of aluminum in automobiles has been increasing in the past few years, mainly as a weight reduction measure, even greater increases have been attained in the use of steel alloys known as high strength, low alloy (HSLA) steels. These latter materials are low carbon (0.1 to 0.3%) steel alloys containing manganese (0.6 to 1.7%) and alloyed with minute amounts of chromium, molybdenum, vanadium, titanium, columbium, or zirconium. The presence of these micro-alloying elements, along with controlled processing leads to improved properties over conventional low carbon steel.

The most important property of the HSLA steels is their relatively high strength. Fine grain size and carbide and nitride precipitation enable significant improvements in yield strength for this group of steels, in comparison with normal low carbon (mild) steel. HSLA grades are available with yield strengths ranging from 40 ksi to 90 ksi, compared with about 33 ksi for 1010 steel. Since the density of the HSLA grades is approximately the same as that of mild steel, the higher strength values for HSLA lead to higher strength/weight ratios. This means that a part made from HSLA can be thinner, and lighter, than the equivalent mild steel part, and still withstand the same applied load.

The cost per lb. of HSLA is somewhat greater than that of mild steel, with price, in general, increasing with yield strength. In 1975, typical 50 ksi (yield strength) grades were about 24% more expensive than 1010 steel, with 60, 70, and 80 ksi alloys being, respectively, 29%, 36%, and 46% more expensive than 1010 (Ref. 5-17). The price of HSLA is, however, well below that of the ultra high strength alloy steels. The HSLA steels, therefore, fall between mild steel and alloy steel in both strength and price.

It should be noted at this time that the discussion in this section will be limited to applications of the family of high strength, low alloy steels. There are other applications for which the higher strength alloy steels, such as martensitic steels and full-hard cold rolled steel, are suitable. Some of these materials have been used for a number of years in automobiles, but limited formability and relatively high cost restrict their usage to relatively few components. It is expected that the newer, more formable, less expensive, HSLA steels will have a much wider range of applications in automobile weight reduction programs.

Despite the higher cost/lb. of the HSLA steels, use of the materials in place of mild steel frequently enables weight reductions large enough to result in a lower total material cost. A cost/weight analysis (Appendix C) shows that a 12% or greater decrease in the weight of a part, using a 50 ksi HSLA grade, will pay for itself (material costs) by weight reductions both in the part itself and in secondary reductions in other vehicle components. Weight reductions of this magnitude or greater should be feasible through redesign of heavily loaded components to take advantage of the higher strength/weight ratio of HSLA steels (50% higher for 50 ksi grade in tensile loading).

Use of HSLA in components for which strength is the primary design parameter has increased significantly in the past few years, and has enabled a reduction in thickness and weight of those parts. Many parts, however, have other criteria to meet besides strength, such as limited deflection or avoidance of buckling. Since deflection, buckling and other stiffness-related effects occur more easily in thin sections, these criteria would act to limit the thickness reduction (and weight reduction) potential of HSLA in many components.

The stiffness constraints apply especially to thin panels, where "oil-canning" must be avoided, and in side rails and frame parts which contribute significantly to the vehicle's deflection characteristics. A recent study (Ref. 5-24) showed that strict imposition of stiffness constraints could eliminate potential thickness and weight reduction in those components through substitution of HSLA. Other computer studies, however, show that some thickness reduction is allowable in those components (Ref. 5-6), but the reductions may not be cost effective.

In addition to the stiffness limitation, there are several other factors which could limit the weight reduction potential of high strength steel. Although the HSLA steels are much more formable than the ultra high strength steel alloys, they have inherently lower formability, because of their higher yield strength and slightly lower ductility, than mild steel. This decreased formability can be seen in forming limit diagrams of the various HSLA materials (Ref. 5-18). Their greater strength also requires forming pressures and greater springback allowances than for mild steel. In spite of these drawbacks, which are more pronounced for the higher strength HSLA grades, reasonable success has been achieved in stamping the steels at room temperatures to relatively severe automotive shapes.

Intensive forming investigations, especially by the auto makers, have led to improved forming practices for HSLA and to the development of suggested minimum bend radii for different HSLA grades and thicknesses (Refs. 5-19, 5-20). With the use of these special practices, forming of the lower strength HSLA grades is not much more difficult than forming mild steel, except for severe radial draws (Ref. 5-21).

The more severe forming problems with the higher strength HSLA grades have, to some extent, restricted their use in automobiles. In the past few years a great deal of materials development work has been carried out in an attempt to find more formable high strength grades of HSLA. Some progress has been made with the development of more formable alloys with inclusion shape control (Ref. 5-22) and an annealing heat treatment prior to forming has been developed which seems to offer improved formability (Ref. 5-23). It is expected that even more progress will be made in the forming area in the not-too-distant future, and this would open up more potential weight reduction applications for HSLA steels.

Another problem, related to formability, that has restricted the usage of HSLA steels is the limited range of available thicknesses, especially in cold-rolled gages. At present, the materials are available in quantity in gage thicknesses ranging from .071" to .23". Improved availability of thinner gages, particularly cold rolled gages for external body sheet applications, would open up more potential high volume applications.

If more use is made of thin gages of HSLA steels, corrosion could become a limiting factor, in addition to stiffness. Most HSLA steels, particularly those containing copper or chromium, demonstrate more resistance to atmospheric corrosion than mild steel (Ref. 5-19), and their corrosion life in other environments, such as road salt, seems to be at least equivalent to that of mild steel of the same thickness. However, when thinner gages of HSLA are used to achieve weight reduction, corrosion problems may be encountered where they weren't previously experienced, mainly because of reduced thickness (Ref. 5-17). This may require more use of galvanized or zinc-coated sheets for extra corrision protection at susceptible locations.

A final factor that could limit thickness reductions is dent resistance. Since dent resistance is a function of thickness, the thickness reduction of external body parts may be limited by the requirement that dent resistance be maintained. In one recent experimental program, although a 20% thickness decrease in body panels would have been allowed from strength and stiffness considerations, the thickness decrease was cut to 10-15% in order to maintain acceptable dent resistance (Ref. 5-18).

Considering the above limitations, it is apparent that the weight reduction potential of the HSLA steels is greatest with strength-designed parts for which stiffness is not too critical, forming is not too difficult, and corrosion is controllable. A number of such parts are found in the body and bumper groups, as well as elsewhere in the automobile.

5.2.1 Body Parts

A number of body parts are designed on a basis of strength. Prime examples are the intrusion-resistant door guard beams mentioned in Section 5.1.1. HSLA steels are used for door intrusion beams in many cars because of their high strength and reasonable formability, and there is much potential for more such applications. In a recent study at Chrysler, a specially-processed, roll-formed HSLA steel of yield strength 150 ksi was used to achieve a weight savings of 50% in the intrusion beam (Refs. 5-18, 5-25, 5-26). Weight reductions of a smaller magnitude would be expected with an 80 ksi grade of HSLA.

In the same Chrysler study, a number of other body parts were fabricated from HSLA steels and installed on a developmental vehicle. Among the body parts that were converted from mild steel to HSLA were: door inner and outer panels, front fenders, side rails, body sills, center pillar, wheelhouse structure, deck panels, door hinges, crossmembers, and windshield cowl and dash structure, in addition to the intrusion beam. It was found that a thickness reduction of 10% (1 gage reduction) was possible in cold-rolled exterior body panels, such as outer door panels (Ref. 5-18), while 15% reductions were realized in cold rolled interior door panels. 45 and 50 ksi grades of HSLA were used in these applications. It is expected that reductions of a similar magnitude could be achieved in hood and deck lid inner and outer panels (Ref. 5-26). A few problems were encountered in the forming of these HSLA parts in tools designed for mild steel parts. These problems would have to be resolved, by using improved die design and forming practices, before similar parts could be made in production quantities.

Most of the remaining HSLA body parts on the developmental vehicle were made from hot rolled sheet of 50 ksi yield strength. Thickness reductions of about 15% were obtained on deflection-critical components, such as rails, with decreases in the 20-25% range being reported for other components (Refs. 5-18, 5-25, 5-26). There were some minor forming difficulties encountered with these parts, also, which would require further development work before any production runs.

The results of the referenced study show clearly the potential for substantial weight reductions in body parts through use of HSLA steels. Before this potential can be realized in production, however, much more development work will have to be carried out in the areas of part and tool design, forming practices, and material property refinement. It should also be pointed out that some of the potential weight reductions might not be large enough to avoid imposition of a cost penalty.

5.2.2 Bumpers

Automobiles have already started to appear which use HSLA steels in bumper face bars and reinforcement members, as well as in other bumper parts, such as brackets, plates, restricters, and supports. In fact, the majority of current automotive usage of HSLA is in bumper components.

A prime example of the application of HSLA steel is in the single piece bumper of the 1976 Chevette. The bumper uses a formable grade of 55 ksi HSLA, .095" thick, in a chrome plated face bar without reinforcement. The HSLA bumper is 1/2 lb. lighter and 32% cheaper than an aluminum bumper of similar design (Ref. 5-12). Success with the Chevette bumper seems to have prompted a redesign of the Vega bumper to utilize HSLA instead of aluminum. Utilization of a .105" thick face bar made from the same material as is used in the Chevette bumper will enable elimination of the reinforcement used in the current bumper design, with a resultant total weight saving of 23 lb./car over the present aluminum design (Ref. 5-27).

A number of other bumper designs make use of the weight-saving potential of HSLA steels. A 15 lb. weight reduction in each bumper resulted from use of a 50 ksi HSLA reinforcement in the Pinto bumper (Ref. 5-17). Similar reductions have resulted from use of HSLA reinforcements or face bars in other 1975 and 1976 cars.

A recent evaluation of several bumper designs (Ref. 5-28) shows that a 54 lb. weight savings can be achieved in a currently-produced 148 lb. bumper/grille system for a 4000 lb. vehicle by substituting 60 ksi HSLA steel face bar for the present mild steel face bar and reinforcement. Even greater decreases can be obtained by combining the HSLA face bar with an elastomeric energy absorption unit and using plastic instead of zinc in the grille. A design with an HSLA reinforcement with elastomeric face bar cover is slightly heavier, while an aluminum face bar design was found to be lighter, but slightly more expensive than the HSLA design (Ref. 5-28). As was indicated in Section 5.1.2, each of the many bumper designs has proponents and each has the potential to reduce the weight of bumper systems.

5.2.3 Other Components

A number of other vehicle components have been produced from HSLA steel in the past few years. Among them are: engine mounts, wheels, ball joint sockets, crankshaft flywheels, pedal mounting plates, and brackets. These applications have nearly all resulted in vehicle weight reductions. For example, use of a 50 ksi HSLA steel to replace mild steel in the wheel spiders of some 1974 Chevrolets resulted in a weight saving of over 10% (Ref. 5-29). Similarly, reductions of 2 lb./wheel were achieved by use of HSLA steel in reduced gauges instead of mild steel in the wheels of the Charger (Ref. 5-25), and a 20% decrease in weights of both torque converter plate and engine mounts on the same car resulted from a switch to 50 ksi HSLA. Other components, such as seat supports, are also good candidates for substitution of HSLA steel in vehicle weight reduction programs.

5.2.4 Potential for HSLA Usage

The discussion in the above three sections points out that there is much potential for automobile weight reduction through substitution of high strength, low alloy steel for mild steel in selected vehicle components. As is the case with aluminum, whether the potential is realized depends on technical factors, such as production difficulties and problems during service, as well as on such non-technical factors as cost and competition from other materials. As was done in Section 5.1.5, we will now consider how the technical factors could affect the potential usage of HSLA steels in weight reduction programs. It should be kept in mind that, although the non-technical factors are not being considered here, those factors could determine whether or not HSLA steels will be used in a given application.

The same categories of vehicle components that were considered in Section 5.1.5 will be used again in this discussion. The components falling in those categories, and their weights, will be discussed for the particular case of the Optimum 5 weight-conscious design. The weight reduction potential of HSLA for the OP-4 and OP-6 design can be extrapolated from the OP-5 information.

Category I Components

The weight derivation for the Optimum 5, as given in Section 3.4.3 assumes no HSLA steel components. There are, however, a few components, especially in the power train, that are assumed to be made from alloy steel.

Category II Components

A number of vehicle components are now being produced successfully from HSLA steels on some vehicles without encountering major problems in either production or service. These components, and the approximate weight reductions that could be expected from their use on the Optimum 5 are shown in Table 5D. In determining the curb weight change, ΔWc, the influence coefficient Q1 = 1.4 was used.

Since the Category II Components listed in Table 5D have already been made from HSLA steel on production automobiles, there are no major technical problems that would prevent their immediate adoption.

Category III Components

The development efforts now underway in both the steel and automobile industries to resolve the problems that now limit more extensive use of

TABLE 5D. HSLA COMPONENTS AND WEIGHTS - OPTIMUM 5

CATEGORY II

Component	C_S	C_H	Δ
Door intrusion beam	20	10	10
Bumpers (excluding energy absorber)	90	60	30
Wheels	75	65	10
Other components (engine mounts, brackets, pedal mounting plates, ball joint sockets, crankshaft flywheel, seat supports)	70	55	15
TOTALS	255	190	65
ΔWc			-91 lb.

CATEGORY III

Component	C_S	C_H	Δ
Doors (excluding intrusion beam)	114	100	14
Hood, deck lid, hinges	88	77	11
Front fenders	30	27	3
Body structure	427	395	32
Front end (excluding fenders, hood, grille)	75	68	7
TOTALS	734	667	67
ΔWc			-93 lb.

Column C_S - Weights of components (in pounds) made from current material.

Column C_H - Weights of components (in pounds) made from HSLA steel.

Column Δ - Decrease in weight (in pounds) resulting from material substitution.

HSLA steel are meeting with some success, as has been demonstrated by papers such as References 5-18 and 5-25. There is still a great deal of work to be done, but it is expected that some of the technical limitations now facing HSLA will be overcome within the next few years. This would allow more widespread use of these high strength steels in the components listed under Category III in Table 5D.

It should be noted that, since the expected weight reductions in some of the components are relatively small, the cost effectiveness of those material substitutions might be questionable. In fact, there are few vehicle components now being made from mild steel that couldn't be made from HSLA steel once the limitation of formability is resolved. For many of those components, however, stiffness requirements would keep thickness and weight reduction to a minimum, while forcing a cost increase. It is very doubtful that any such substitutions will occur.

5.3 PLASTICS

The present day automobile contains a large number of components made from plastics. Although the total weight of these components varies from vehicle to vehicle, it is estimated that the average 1976 American automobile contains about 150 lbs of plastic parts. These plastic parts have been introduced over the years for a number of reasons, but economic factors have generally been the primary motivation for increased use of plastics. With the emphasis now being placed on weight reduction, it is felt that the low density and light weight of plastics will now be one of the main considerations leading to their use. For many material substitution applications plastics are the most cost-competitive light weight material available, and this should lead to increases in plastics usage during the next few years.

Besides their light weight and relatively low cost, plastics have a number of other advantages which favor their use in automobiles. Among them are corrosion-resistance, ease of molding to desirable shapes, and relatively high strength/weight ratio for some plastics (especially when reinforced). Each of the above factors has played a part in the increased utilization of plastics in future materials decisions. A number of other considerations could, however, affect the adoption of more plastic vehicle components. Among them are: (Ref. 5-30)

-Special handling requirements. Plastic parts require different processing and handling during assembly, painting, and repair than do the more common metallic components. This requires some modification of assembly and repair practices.

-Investment costs. Because of the special processing requirements of plastics, large scale switches to plastics would require vast capital investment in new equipment by the automotive industry. In order to reduce investment requirements, there will probably be more emphasis placed on low investment, high productivity processes, such as thermoplastic stamping and liquid reaction molding, instead of on high investment processes such as injection or compression molding.

-Cost and availability of plastics. Although the price increases and potential shortages of petroleum have affected and will continue to affect the cost and availability of polymers, there is hope that most plastics will be readily available in the foreseeable future. Alternative sources for plastic raw materials, such as coal, oil shale, wood, and agricultural products could be used to produce almost all automotive plastics (Ref. 5-31). Fairly substantial price increases must be anticipated for many of the plastics, however (Ref. 5-32). One factor that could limit the rate at which new plastic components are introduced is the raw material and resin capacity of the plastics industry. The facilities of the plastics producers would possibly not be able to expand fast enough to handle an increase in consumption much greater than 10 lb/car/year (Ref. 5-34).

-Energy requirements. Despite the fact that most polymers come from petrochemical feedstocks, it has been shown that the total energy requirements of non-reinforced plastics are much less than those required for the production of most metals, while energy requirements of reinforced plastics are less than half the requirements of aluminum on a Btu/lb basis (Ref. 5-33).

-Recyclability. Although there are methods for recovering and recycling thermoplastic resin from automobile scrap, it is frequently not cost effective. Thermosets cannot be recycled but can be burned as fuel. It is felt that in the near future most plastic scrap automotive parts will be burned as a fuel, with only large thermoplastic parts being recycled (Ref. 5-30). The problem of recycling requires further attention, however.

-<u>Other factors</u>. Other factors such as painting, stiffness, and durability will be discussed later in this section.

At the present time the majority of the plastics used in cars are in interior trim components, such as foam seats, seat covers, instrument panels, floor mats, etc. While new trim applications are being developed, plastics are also candidates for exterior body parts, bumper components, and other applications.

5.3.1 Trim

Large amounts of flexible and rigid plastics and elastomers are found in the interior of every American automobile. Among the common polymeric components are: injection molded ABS door and instrument panels, sometimes glass-reinforced; urethane foam seating; vinyl seat covers and floor mats; and vinyl/paper headliners. Advances will continue to be made in these components to decrease weight and lower production cost while frequently also improving comfort, safety, or appearance. For example, polyethylene foam is being laminated to non-woven carpet or PVC film to produce a lower cost floor covering which is lighter than present coverings (Ref. 5-35). New foam plastic seating materials, such as polyester, are being developed that can sometimes be used without a metallic spring base (Ref. 5-32). Decorative plastics, either chrome plated or heat sealed/embossed hot stampings, are being used more frequently, not only inside the vehicle in the dashboard area, but also on the exterior, for side bump strips or as trim on the grille.

Despite the new plastic trim applications, it is not expected that significant increases in the amount of plastics in automobile interiors will occur. Some weight reductions will be brought about by advances in technology and materials, however, as pointed out above. In addition there will be changes in the composition of the plastic trim items. For example, some polypropylene door panels, filled with calcium carbonate for impact resistance, are replacing ABS panels (Ref. 5-35) and Noryl phenylene oxide is now being used instead of ABS for some instrument panel covers (Ref. 5-32). These material changes often result in slight weight reductions.

5.3.2 Body Parts

Although most automotive plastics are now found in interior trim, the greatest potential for increased utilization of plastics seems to be in body parts. It is estimated that the average 1976 automobile has about 20 pounds of plastic body parts (Ref. 5-30). It is predicted that this figure could increase substantially within the next five years. Extensive development programs are now underway in which plastics are being tested in a wide variety of body applications, ranging from decorative trim to rigid load-bearing components. There is similar variation in the plastics being studied for body applications, ranging from soft, flexible elastomers to stiff and strong fiber reinforced thermosets.

Plastics applications in automobile bodies, both current and contemplated, can be classified in the following groups: flexible exterior parts, including fascia; rigid grilles and fascia; hang-on parts; and other body components.

Flexible body parts. Since first appearing on a 1968 Pontiac, elastomeric exterior body parts have become fairly common on American automobiles, with some of the impetus being provided by federal impact regulations. Among the major parts being made from elastomeric materials on some vehicles are front and rear fascia (as on several compact and intermediate GM cars), fender and quarter extensions, and filler panels or end caps. These parts, in general, support little load but are located at front and/or rear end where their elastic properties enable them to maintain their shape in minor collisions.

Most of today's flexible exterior parts are made from either of three materials: thermosetting urethane, thermoplastic urethane, or rubber-like EPDM (ethylene-propylene diene monomer). The thermoset urethane parts are made by the reaction injection molding (RIM) process (sometimes called liquid reaction molding), while parts made from the latter two materials are usually injection molded. An RIM process for molding thermoplastic urethane has recently been developed (Ref. 5-37) but is not yet being used for production. Other thermoplastic elastomers have been developed, such as polypropylene and thermo-plastic olefins, but they have not yet captured much of the automotive market (Refs. 5-38,5-39). While the EPDM parts are frequently glass-reinforced, most of the urethanes have so far not been reinforced. Recent tests (Ref. 5-40) have shown that glass reinforcing of urethane can be compatible with the RIM process. It is felt that, since the strength and stiffness properties of each of the elastomeric materials are improved by the addition of reinforcing fibers, more use will be made of reinforced elastomers in the future.

Because of the inherent low stiffness of the elastomeric materials, it is expected that their use will be limited to components for which stiffness is not a major factor. The flexibility of the parts requires that their exterior paint also be flexible. Elastomeric paints have been developed for use on such parts (Ref. 5-41), but they are still much more expensive than the enamels used on metallic components. There is occasional difficulty in getting a good color match between the two types of paint.

It is expected that flexible body parts will be found in greater numbers on vehicles produced in the coming years, but the total usage of the elastomeric parts will depend on a number of factors, one of which is the styling of the front and rear ends of the vehicles.

Rigid grilles and fascia. For those components on the front and rear ends of vehicles where flexibility is not desirable, such as headlamp housings and chrome-plated grilles, more rigid plastic components are being introduced. The properties of the components range from the rigid RIM structural urethane foams with modulus $E \approx 10^5$ psi, elongation $\approx$ 70%, tensile strength $\approx$ 4000 psi to fiberglass reinforced plastics with $E \leq 5 \times 10^6$ psi, elongation $\approx$ 5% and tensile strength up to 80,000 psi. The majority of present applications use plastics with properties somewhere in between these extremes. For example, injection molded grilles made from ABS or polypropylene have replaced metallic grilles (zinc) on some cars, leading to significant weight reductions (Ref. 5-38). Headlamp housings and fender extensions made from glass-reinforced or mineral-filled nylon are also in production. These components are produced mainly by injection molding. Some parts for which rigidity is important, such as lamp housings and front end panels, are being compression molded from fiberglass-reinforced sheet molding compound (SMC). Parts made from rigid RIM, such as the front and rear fascia on the 1976 Corvette, generally have lower tooling costs than those made by injection or compression molding, but do not have the stiffness required in some applications.

Although most of these so-called rigid materials do not require the elastomeric paints, they still must be painted separately from the metallic body assembly. If a chrome coating is desired, as on plastic grilles, it is likely that sputtering will be employed.

Hang-on body parts. At the present time there are no large-scale applications of plastics in the hang-on components, such as doors, hoods, and deck lids. In the bolt-on components (fenders) the only high volume use of plastics has been in fender liners. Substitution of a glass reinforced polypropylene liner for one made of mild steel resulted in a 30% weight reduction, along with improved corrosion resistance and sound deadening (Ref. 5-38). The only other production uses of plastics in the hang-on or bolt-on body parts have been in the low-volume Corvette. In spite of this, it is felt that there is much potential for high-volume applications of plastics in hoods, decklids, doors and fenders in the not-too-distant future. In fact, it is felt by many that the largest long-term gains in use of automotive plastics will occur in these components.

Development efforts are now underway in which a variety of plastics are being tested in hood, decklid and door panels, and fenders. The success of some of these programs, coupled with the current emphasis on weight reduction, should lead to production applications within the next few years. The first high volume component may be a deck lid inner panel, with hood inner panels being, perhaps, not too far behind (although high under-hood temperatures may be a limiting factor). Inner door panels should also appear, along with some fenders, at least on low volume cars. Finally, there may be production vehicles with exterior panels of decklids, doors, and hoods (Ref. 5-30).

Because of the stiffness and strength requirements of the hang-on and bolt-on components, it is felt that the most likely plastics candidates for these applications are glass-reinforced structural plastics. In particular, the polyester-based thermoset sheet molding compound (SMC), and the new, high modulus compounds XMC and HMC (Ref. 5-43), seem to be good possibilities for the inner panels of the hang-on components. Some surface quality problems may have to be overcome before SMC is suitable for large exterior panels, however, and the other two compounds are still relatively untested. These glass-reinforced molding compounds would all be processed by compression molding. Superior mechanical properties could be obtained by using graphite or Kevlar fibers instead of glass in the reinforced thermosets, but the high cost of those fibers should preclude their use in automotive applications for quite a while.

Although the glass-reinforced thermoplastics being considered for hang-on body parts do not have the strength or stiffness of the thermoset molding compounds, they are still prime candidates for those applications without high stiffness requirements. Stampable glass-reinforced thermoplastics, such as polypropylene, are particularly interesting because of their low density, good strength, and relatively low investment requirements, although their surface appearance is not yet good enough for external panels. The use of injection molded reinforced thermoplastics will probably continue to grow, but the lower modulus of those materials, due to fiber length and volume limitations, may restrict their adoption for hang-on components. Some success has been achieved in the use of rigid RIM urethane in hang-on components, such as the fenders of the experimental Pontiac Phoenix (Ref. 5-44). The surface quality of the RIM parts seems to be good enough for exterior use, but the material's stiffness is too low to allow it to be used without a higher modulus reinforcing panel.

Two important factors that may limit the use of plastics in hang-on components are cost and surface appearance. Because of relatively slow production rates, substitution of plastic for steel in these parts would often result in a cost penalty even though weight reductions of up to 50% could be achieved. In addition to the surface quality problems with SMC and stamped thermoplastics, mentioned above, other appearance difficulties that need to be worked out are color match of both freshly-painted and weathered surfaces and repairability. A good deal of research and development work seems to be underway in an attempt to overcome these problems.

It should be noted that, although there seems to be a great potential for use of plastics in hang-on or bolt-on body parts, the realization of this potential will not occur until the development efforts and production tryouts prove that the materials can meet required automotive standards in a cost-effective manner.

Other body parts. Plastics are now being used in a number of other body applications and are proposed for even more. Examples of current usage include: spoilers, window frames, window louvre panels, exterior trim, fender liners, and mounting brackets. Parts for which plastics are being proposed and tested include: catwalks, radiator supports, door guard beams, and splash shields.

Because many of these components have strength and stiffness requirements, it is expected that glass-reinforced plastics will be prime candidates for these applications. For example, it has been shown that weight reductions of 40-50% are possible by using high modulus structural molding compounds in place of steel in door guard beams (Ref. 5-43). Since the door guard beams and some of the other parts mentioned above do not have an exterior surface, they could easily be made from plastic materials on production vehicles before the surface appearance problems of the materials are resolved.

Some of the exterior parts, such as spoilers, could be made from rigid RIM urethane foams and, in fact, some successes have already been achieved in that direction. Injection molded thermoplastics will also be in the running for many of these components.

Another plastic item which could begin appearance on high-volume automobile bodies in the near future is an epoxy replacement for lead-tin body solder. Development of this plastic "body solder" is evidently in an advanced stage. Another potential item, but one that is further away from production, is the plastic window. GM has developed a specially-coated polycarbonate that was used for the fixed rear windows of the Phoenix (Ref. 5-44), but production runs will probably be delayed until its scratch resistance is proven. Even if plastic replaces glass only on the fixed side windows, a significant weight reduction could be realized.

5.3.3 Bumpers

A number of the new bumper designs that are being proposed, tested, and/or produced by the automakers include plastics in one form or another. Soft elastomeric materials are appearing on a number of bumpers, mainly because of their elastic energy absorbing characteristics. Other energy absorbing systems containing plastics have also been developed, and the first fiber reinforced plastic bumper has recently appeared on the U.S. market. Thus it appears that the plastics industry will be in the thick of the battle for supremacy in the bumper market, along with aluminum and HSLA.

Three different plastics-containing bumper systems are being studied: the "hybrid" bumper consisting of a metallic reinforcing bar with elastomeric face, the "soft" bumper system composed of elastomeric bumper with either thermoplastic honeycomb or urethane foam energy absorber, or the SMC system consisting of fiber reinforced plastic bumper with an elastomeric face.

Hybrid bumper systems with elastomeric face bars or bumper guards have been produced since 1968 and are now found on a number of U.S. automobiles, e.g., Mustang II. The soft, flexible plastics are resistant to permanent deformation during low energy impact and have been used in bumper systems to enable damage-free 5 mph collisions to be sustained. Both RIM urethane and EPDM have been used effectively in such parts. Although some weight reductions have probably resulted from use of these elastomeric bumper guards, the magnitude of the reductions is hard to determine since most occurred during complete bumper redesigns. It is expected that more hybrid bumper applications of this type will occur in the future, especially in combination with elastomeric front and rear fascia.

The all-soft bumper system is a relatively new development, with the only applications so far being the 1976 Corvette and a fleet of New York City taxicabs. A higher volume production run, the Pontiac Firebird, is evidently in the works (Ref. 5-12). Two different soft bumper systems are presently available, one using lightweight urethane foam energy absorbing blocks and the other an injection molded polyolefin thermoplastic honeycomb energy absorber. In both systems the energy absorbing unit is attached to a rigid (metallic) reinforcement bar and are covered with an elastomeric face, which could be part of a unitized soft fascia. Although the soft bumper systems do not handle energy as effectively as a hard faced bumper equipped with shock absorbers, they inflict less damage on other cars and pedestrians (Ref. 5-12). The weight reductions that have been achieved with these systems vary from 38% to 53% over all-steel production bumpers (Refs. 5-12,5-28).

The newest plastic bumper system on the U.S. market is the SMC bumper on the imported Renault 5. The system uses a bumper made of a new high strength sheet molding compound (oriented roving SMC) with an elastomeric (urethane) covering and a steel reinforcing bar. The SMC face bar has been shown to absorb more kinetic energy than either steel or aluminum and is lighter in weight (Ref. 5-45). SMC is not the only possible glass-reinforced plastic under consideration for bumper face bars or reinforcements. XMC and HMC are also possibilities (Ref. 5-43). It appears that significant weight reductions are possible by using glass-reinforced plastic instead of steel in bumpers, since a 12 lb. weight savings was realized in the Renault 5 (Ref. 5-32). Potential drawbacks to the use of such bumpers are cost and ease of repair.

In each of the above bumper systems, the face bars, whether elastomeric or fiber-reinforced, are made of plastic which can be colored with pigments to most desired colors. Although this has proven quite acceptable on a number of cars, many stylists, and perhaps consumers, desire bright chrome-plated bumper face bars, especially on larger cars. Plastic face bars may lose out to chrome-plated steel or aluminum for those applications. Another factor that might favor the use of metallic face bars over plastics is their capability for towing (with bumper-mounted hitches), pushing, or jacking (Ref. 5-28). Even bumpers with hard metallic face bars, however, offer an opportunity for plastics usage since elastomeric buckling columns are being proposed as replacements for hydraulic energy absorbers in such systems (Ref. 5-46).

5.3.4 Other Components

Quite a few other vehicle components that don't fit within the trim, body, or bumper groups could be made from plastics. Some of these components have already been produced in volume, others could start appearing in the near future, while others are further down the line.

Among present applications are polyethylene insulation for electrical wiring, phenolic distributor caps and rotors, nylon gears, glass-reinforced phenolic brake pistons and valve bodies, polypropylene air conditioner and heater housings, nylon radiator fans, and polypropylene fan shrouds. These components are not all made from light plastics on every automobile, but their use is spreading and their adoption can bring measurable weight reduction. For example, an air conditioner heater housing made from reinforced polypropylene can weigh about 40% less than one made from gunk molding compound (Ref. 5-38). Polypropylene battery cases have replaced rubber on several lines of automobiles, with weight reduction being a major motivating factor. A plastic case weighs about 1.4 lbs., over 75% less than the hard rubber case it replaced (Ref. 5-38).

Another plastic item which seems to be nearly ready for adoption on production vehicles is the gas tank. Although no American car has yet appeared with a plastic primary gas tank, Ford has been using auxiliary gas tanks made of high density polyethylene (HDPE) for several years and several European cars now have HDPE fuel tanks. A weight reduction of about 25% can be achieved by replacing sheet steel gas tanks by blow molded HDPE (Ref. 5-32).

The molded phenolics that have been adopted for disk brake pistons in the past few years are now being tested for other applications. Among the promising uses of phenolics are brake master cylinders, automatic transmission pistons, and carburetor components (Ref. 5-47). Some phenolic ignition components, however, could possibly be shifted to other, less dense, plastics. A great many potential applications of glass-reinforced plastics are being studied, some of which could be adopted in the near future. The first new applications might be in torque converter and timing belt covers, for which reinforced polypropylene looks promising, while a reinforced polyester computer module housing will probably be appearing in more Chrysler cars. Further away, but undergoing tests, are glass-reinforced plastic wheels. The wheels for GM's experimental Phoenix were molded in converted die casting machines from glass-reinforced polyester (Ref. 5-44). Although significant weight reductions (60% or more) are possible with the plastic wheels, they probably won't be ready for production for several years. The new high-modulus structural molding compounds are being tested in several components, the most promising of which are transmission supports (Ref. 5-43). Other potential applications include radiator supports and leaf springs, although the latter are probably further down the line.

One other possible use of automotive plastics is in tires. A urethane tire, made by the RIM process, is currently being tested in Europe. It is reported to be lighter and more durable than conventional rubber tires (Ref. 5-36).

5.3.5 Potential for Plastics Usage

From the above discussion it is clear that a wide variety of plastic parts could be produced for U.S. automobiles in the future and their use could result in considerable weight reductions. Because of the large variety of plastics that could be used for the parts, and their different densities and design requirements, it is rather difficult to predict their exact weight reduction potential. The task is further complicated by the fact that the different plastics compete with each other for the same components, and sometimes replace one another.

In spite of these difficulties, an attempt will be made, with the aid of Refs. 5-48, 5-28, 5-38, and 5-43, to predict the potential effect of plastics usage on the weight of the Optimum 5 weight-conscious design. As in Sections 5.1.5 and 5.2.4, this analysis will consider only the technical feasibility of the plastic parts, neglecting the important considerations of cost and competition from other materials. The same categories of vehicle components that were used in Sections 5.1.5 and 5.2.4 will again form the basis for the discussion of weight reduction potential.

Category I Components

The weight derivation for the Optimum 5 vehicle given in Section 3.4.3 includes approximately 100 pounds of plastics. Among the parts included in this group are urethane foam seating, polypropylene battery case, polyethylene electrical insulation, polyvinyl chloride seat and instrument panel covers, nylon fan, urethane armrests, and ABS instrument panel components.

Category II Components

The components that fall in this category are those that have already been developed to a point where they could immediately be used on production vehicles without encountering major technical problems. These components, and the approximate weight reductions that could result from their adoption, are listed in Table 5E. The influence coefficient Q1 = 1.4 was used in the determination of the curb weight change ΔWc.

It should be noted that the new component weights C_N listed in Table 5E include materials other than plastics. The weight of plastic in the components is listed under W_{pl}. The new bumper weight listed in Category II is that for an all-soft system such as is used on the Corvette. It could be used with an elastomeric front fascia.

Category III Components

The components in this category are those which could start appearing on production vehicles in a few years, after some of the current technical problems with the plastic parts are resolved. These parts are listed in Table 5E, along with the potential weight reductions that their use could bring about.

The new bumper system listed in this category consists of a reinforced plastic bumper covered by an elastomeric fascia and attached through elastomeric energy absorbers to a steel supporting member. The weight change shown in Table 5E for the Category III bumper is the difference between it and the Category II bumper system.

Actually the list of components in Category III could have been expanded to include other components such as springs, tires, drive shafts, and structural members, all of which could eventually be made from plastic. In fact, it has been suggested that sometime in the future we may see a car which is composed mainly of plastics. Such a car will not likely appear for a good many years, however, and the plastic components that were left out of Category III will probably not be ready for production automobiles for quite a while.

TABLE 5E. PLASTIC COMPONENTS AND WEIGHTS-OPTIMUM 5

CATEGORY II

Component	C_C	C_N	W_{pl}	Δ
Bumpers (all-soft)	100	70	63	30
Fascia (replaces grille, headlight housings, etc.)	20	9	9	11
Trim	10	4	4	6
Door intrusion beam	20	10	10	10
Other components (heater housing, fender liners, brackets, brake parts, body solder)	30	17	16	13
TOTALS	180	110		70
ΔWc				-98 lb.

CATEGORY III

Component	C_C	C_N	W_{pl}	Δ
Doors	115	78	50	37
Hood, deck lid	88	45	40	43
Front Fenders	30	18	18	12
Wheels	75	30	30	45
Bumpers (see p. 192)	70	50	43	20
Other components (gas tank, structural supports, fixed windows)	42	24	24	18
TOTALS	420	245		175
ΔWc				-245 lb.

Column C_C - Weight of component (in pounds) made from current material.
Column C_N - Weight of component (in pounds) made from new material (plastics).
Column W_{pl} - Amount (in pounds) of plastics in new component.
Column Δ - Weight decrease (in pounds) $\Delta = C_C - C_N$

It should be pointed out that some of the components listed as potential applications of plastics were also listed for aluminum and/or HSLA steel. A large number of factors, including cost and styling considerations, will affect the material choice decision for those components. No attempt will be made here to determine which of the materials is superior for a given application.

5.4 CLOSURE

It is apparent that significant automobile weight reductions are possible by taking advantage of some of the light weight or high strength materials available today. Attention must be paid, however, to the limitations of the materials, and more development work will be required on the part of both material suppliers and the auto industry to eliminate some of these limitations.

In designing vehicle components using the newer materials, engineers must endeavor to use the materials as effectively as possible, and not just use different materials in the same old designs. In fact, if really effective design techniques were used it would be possible to achieve significant weight reductions without any substitution of materials. For example, modern die casting technology has enabled use of lighter, thin-walled zinc castings in such applications as grilles, headlamp bezels, door handles, wheel covers, and mirror mounts. Another example is the significant reduction in wall thickness and weight of cast iron engine blocks that has occurred in the past decade or so due to the use of more effective foundry techniques.

It is felt that there is much potential for further weight reduction in automobiles both by making more efficient use of conventional materials and by substituting light weight or high strength materials in applications where they can be proven effective. In order to realize these potential weight savings, however, automotive engineers will have to make maximum use of new advances in materials technology and of modern design and production techniques. As the late C.F. Kettering of General Motors once said, "Engineering is a combination of brains and materials. The more brains, the fewer materials."

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APPENDIX A

COST/WEIGHT CALCULATIONS FOR ALUMINUM

Assumptions:

(1) body weight (steel body) = (.4) (curb weight).

(2) decrease in component weight by one lb results in curb weight by 1.4 lb.

(3) secondary weight reduction (.4 lb) occurs in components that cost twice as much (per pound) as steel body sheet (on average).

(4) 1 lb weight reduction results in 1.4 gallon lifetime fuel savings.

Definitions:

B_s - weight of all-steel body

C_s - weight of steel component

C_{AL} - weight of aluminum component

$\$s$ - cost per lb of steel

$\$AL$ - cost per lb of aluminum ($\approx 4\$s$)

$\$g$ - cost per gallon of gasoline ($\approx 5\$s$)

Original Vehicle (no material substitution)

Component weight = C_s

Body weight = B_s

Vehicle weight = $W_c = 2.5\ B_s$

Material cost $\approx \$s(B_s) + 2\$s(1.5\ B_s) = 4\$s\ B_s$

Vehicle with Aluminum Replacement Components

Component weight = C_{AL}

Body weight = $B_s - C_s + C_{AL}$

Vehicle weight = $2.5\ B_s - 1.4(C_s - C_{AL})$

Material cost $= \$s[4B_s - C_s - .8(C_s - C_{AL})] + \$_{AL}C_{AL}$

$= \$s[4B_s - 1.8C_s + 4.8C_{AL}]$

Decrease in weight $= 1.4(C_{AL}-C_s)$

Increase in Material Cost $= \$s[4B_s-1.8C_s+4.8C_{AL}-4B_s]$

$= \$s(4.8C_{AL}-1.8C_s)$

Fuel Savings (lifetime of car) $= (1.4\ \frac{gal}{lb})$(decrease in weight)

$= 1.4(1.4)(C_s-C_{AL})$ gallons

Dollar Savings in Fuel $= \$g(1.96)(C_s-C_{AL})$

(lifetime of car) $= 5\$s(1.96)(C_s-C_{AL})$

(Inflation in fuel prices neglected) $= \$s(9.8)(C_s-C_{AL})$

<u>For No Increase in Material Cost</u>

$$\$s(4.8C_{AL}-1.8C_s) \leq 0$$

$$\underline{C_{AL} \leq 0.375C_s}$$

(at least 62.5% reduction in component weight)

<u>For No Increase in Total Cost</u>

Increase in total cost ≡ increase in material cost - lifetime fuel saving

$$\$s(4.8C_{AL}-1.8C_s) = \$s(9.8C_s-9.8C_{AL})$$

$$\underline{C_{AL} \leq .80C_s}$$

(at least 20% reduction in component weight)

<u>All-Aluminum Body</u>

If $C_s=B_s$ and $C_{AL}=.61C_s=.61B_s$

(entire body changed to aluminum, with 39% weight reduction)

Increase in material cost $= \$s[4.8(.61B_s)-1.8B_s] = \$s(1.13B_s)$

Original material cost $= \$s(4B_s)$

New material cost $\approx (1.3)$(original material cost)

APPENDIX B

ENERGY SAVINGS WITH ALUMINUM COMPONENTS

ASSUMPTIONS:
(1) aluminum component replaces steel component weighing twice as much
(2) energy expenditure for aluminum component is 102,600 Btu/lb (for sheet products)
(3) energy expenditure for steel sheet component is 25,800 Btu/lb
(4) one lb weight reduction results in 1.4 gallon lifetime fuel savings
(5) energy content of gasoline = 150,000 Btu/gal.

Lifetime fuel savings (from Appendix A)

$$F = 1.96\ (C_s - C_{AL})\ \text{gal.}$$

$$C_s = 2\ C_{AL} \longrightarrow F = 1.96\ C_{AL}\ \text{gal.}$$

Energy savings from fuel savings

$$E_1 = 150{,}000\ F = 294{,}000\ C_{AL}\ \text{Btu}$$

Reduction in steel weight (from Appendix A)

$$W_s = 1.4\ C_s - .4\ C_{AL} = 2.4\ C_{AL}\ \text{lb}$$

Energy saving from steel

$$E_s = 25{,}800\ (2.4\ C_{AL}) = 62{,}000\ C_{AL}\ \text{Btu}$$

Addition in aluminum weight

$$W_{AL} = C_{AL}$$

Energy required to produce aluminum part

$$E_{AL} = 102{,}600\ C_{AL}\ \text{Btu}$$

$$R = \frac{\text{Total energy saved}}{\text{Total energy required}} = \frac{294{,}000\ C_{AL} + 62{,}000\ C_{AL}}{102{,}600\ C_{AL}}$$

get $\underline{R \approx 3.5}$

∴ 3.5 times as much energy saves as is used in production of aluminum component

APPENDIX C

COST/WEIGHT CALCULATIONS FOR 50 KSI HSLA STEEL

Assumptions: Same as in Appendix A

Definitions: C_H - weight of HSLA component

$\$_H$ - cost per lb. of 50 KSI HSLA

($\$_H = 1.24\ \$_s$)

plus definitions from Appendix A

Using results of Appendix A

-Decrease in vehicle weight using HSLA component instead of mild steel component = $1.4(C_H - C_s)$

-Increase in material cost

$$= \$_s[4B_s - C_s - .8(C_s - C_H)] + \$_H C_H - 4\$_s B_s$$

$$= \$_s[2.04\ C_H - 1.8\ C_s]$$

-Fuel savings (lifetime of car)

$$= 1.96\ (C_s - C_H)$$

-Dollar savings in fuel (lifetime of car)

$$= \$_s\ (9.8)(C_s - C_H)$$

<u>For No Increase in Material Cost</u>

$$\$_s(2.04\ C_H - 1.8\ C_s) \leq 0$$

$$\underline{C_H \leq .88\ C_s}$$

(at least 12% reduction in component weight)

<u>For No Increase in Total Cost</u>

$$\$_s(2.04\ C_H - 1.8\ C_s) \leq \$_s(9.8\ C_s - 9.8\ C_H)$$

$$\underline{C_H \leq .98\ C_s}$$

(at least 2% reduction in component weight)

APPENDIX D

REPORT OF NEW TECHNOLOGY

The contract, DOT-TSC-996, required the contractor, Thayer School of Engineering, Dartmouth College, to collect and evaluate information regarding the potential for reducing the weight of automobiles within the limits of prevailing designs and materials.

A diligent review of work performed under this contract has revealed no innovations, discoveries, or improvements of inventions.

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