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COMPARATIVE LOGGING TRUCK SPECIFICATIONS AND PERFORMANCE

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INTRODUCTION

Truck logging as a particular method of log transportation does not change the basic principles of a logging enterprise. One of the most important of these principles is to move logs from the stump to the mill at a reasonable cost.

Location of the timber and the limitations of the equipment have caused more variation in these costs than any other factors. Ever since the first logging in this country was started more than 300 years ago, new methods and equipment have been adopted when timber became inaccessible to the existing methods.

Just as railroad equipment became more widely used as the timber close to drivable water diminished, so the use of trucks is becoming more numerous every year. Since the first logging railroad was built in New York in 1851, railroad logging expanded until at one time there were more than 30,000 miles of logging railroads in the United States. The history of logging truck transportation is so recent that the man who made the first attempts in 1914 in the Douglas Fir region is still in the business. Yet today motor truck logging is the biggest thing in the business and is still growing. Last year the amazing figure of 20,000,000 tons or an estimated volume of 5 billion board feet of logs were moved by trucks in the eleven western states.

The original development of the use of trucks was stimulated by the inability of logging railroads to reach all of the merchantable timber. Railroad logging has been confined to those areas where grades of any distance have not exceeded 2 per cent adverse nor 3½ per cent favorable and where curves have not exceeded 40 degrees. Truck operation has expanded these limitations so that 10 per cent adverse and 15 per cent favorable grades and curves not exceeding a 50 foot radius are standard limitations.

Long circuitous routes have been used in some railroad development in spite of the narrow grade limitations where sufficient bodies of timber were found over most of the location. However, the modern truck has been made so efficient that it is now a legitimate and preferable way of operating even some of these areas.

HISTORY

The first attempt to overcome some of the difficulties of railroad transportation was made with moderate success by the use of the road or traction engine. The "Holt Three-Wheeled Traction Engine" was a horizontal boiler mounted on three steel wheels which were propeled by a chain drive from a 11x12 inch steam valve. Loads were pulled in trailer fashion behind this "donkey on wheels" on stone or earth roads to the railroads. Manufacturers claimed that a 60 horse-power engine could haul a load of from 40 to 60 tons at a speed of from 2 to 3 miles per hour, ascending grades as high as 10 per cent. One of its more serious limitations was its slow maximum speed of 3 to 4 miles per hour even when empty.

J. T. McDonald, who undertook the first truck logging operations in the Douglas Fir region in 1914, used Knox iron tired trucks with a fair degree of success. About 1916, L. A. Christensen, who is still in the truck logging business now at Seaside, Oregon, started using trucks in a serious way.

Since that start, truck development has gone through a number of stages. As listed by Truman W. Collins of the Lakeview Logging Company, Lakeview, Oregon, in the August, 1939, "The Timberman," these are:

"1. Relatively large trucks; solid rubber tires, no trailers.
2. Same as first stage, but with high pressure tires; some six wheel equipment.
3. Lighter trucks with single-axle trailers.
4. Same as third stage with low pressure tires.

5. Trucks of varying conventional sizes, with dual axle trailers, auxiliary transmission.

6. The appearance of the very large, custom built trucks, diesel-powered for use entirely on private logging roads.

7. The successful development of diesel power for trucks of conventional highway size."

About 1920 the average 5 ton truck with an 8½ ton trailer operating over private roads was hauling an average of 4000 board feet. Now practical grades do not exceed 10 per cent against the loads and 15 per cent against the empties although with proper gear reductions and power these may be doubled if necessary for short distances.

RESULTS OF EXPERIENCE

It is worthwhile to note in detail what those pioneers who have had the longest and also the most successful experience in logging truck operation state about the proper selection of a motor truck.

Lollyd Christensen, mentioned above, is the contract hauler for the Crown-Willamette Paper Company, Seaside, Oregon. He claims, "The main consideration in selecting a motor truck is whether or not the truck will perform satisfactorily the job in hand. Few people realize the value of getting a thoroughly engineered truck for the job. You must have a unit of ample power to negotiate grades or to give sustained speeds that will enable you to operate on a scheduled trip basis, and along with the motor you must have a frame of the right construction and strength to withstand the loads you wish to haul. The same is true of axles, wheel bearings, tires, gear ratios, etc."

"In buying trucks, the average operator does not study and understand the technical and mechanical details of trucks sufficiently. All steam operators have had some technical knowledge of locomotives and steam logging equipment. Trucks are more complicated, and technical knowledge is even more important." *

Truman Collins, who is an associate of J. T. MacDonald, the first pioneer in truck logging, states in regard to the selection of the right motor truck:

" "As this branch of logging has come of age, we find evidence of a more scientific approach than formerly prevailed. Our truck logging operators are recognizing certain basic principles, such as the following:

1. A logging truck must be carefully engineered for the conditions under which it is to operate. The transmission and rear end must

*"West Coast Lumberman," April, 1939

have a reasonable factor of safety over the torque of the motor, and the frame and axles over the weight to be carried.

2. Trucks must be adequately tired, reasonable consideration being given to the manufacturer's recommendations.
3. Good service facilities must be made available so that proper care may be taken of the eqipment.
4. It is desirable that there by uniformity in performance characteristics of the units in a fleet.
5. It is necessary to reduce the vehicle and trailer weight consistent with the proper factor of safety by the use of newer alloy metals.
6. The superiority of the dual-axle trailer is generally recognized.
7. The desirability of planning a truck logging show for years ahead.

PLANNING AHEAD

In the large scale operations there has been an increasing tendency to plan many years agead on the various phases of logging. For the West Fork Logging Company this type of engineering research is but one of the many phases of a broad sustained yield program. Fundamentally, the broad purposes of this company are to provide secure livelihood for its employees and to grow and harvest successive crops of raw material for domestic and industrial consumption.

This necessitates a practical plan of forest management and harvest. It is vital to the success of this long-range and continuous plan that current operations be successful. A profit must be made in spite of changing and more difficult logging conditions. On this basis I have interested myself as forester for the West Fork Logging Company in this coming problem of truck selection and operation with the hope that it will aid in current operations and will properly fit into the general program of forest management.

In keeping with this planning program for this area, photographs have been made from which an accurate contour and type map has been prepared. The land in this particular area is owned in checker board fashion, one-half by the United States Forest Service and one-half by the West Fork Logging Company.

With this sound basis of topography and timber location, long range plans will be made in advance cooperatively to develop not only actual permanent logging roads and landings but also to fit the entire system into the broad plans of forest growth, fire protection, and watershed control.

It is possible that in connection with the capital depletion allowances now available to selective logging following a forestry program, that a study of timber values, their location, and their volume will be made. This more accurate knowledge will be of great value in planning the location and volume of annual cuts.

LOGGING CONDITIONS

It probably would be wise to describe the logging conditions which must be met before proceeding to discuss how they are to be met.

Location:

In the eastern sixth of the working circle now under forest management is the largest body of the present-day merchantable timber. Into this area, development has started. Since the greatest effect upon the second crop determined by the manner in which the old-growth is harvested, and since this area is to be operated for the next twenty or thirty years, it becomes highly desirable to concentrate on a broad plan of operation which will cover this entire period.

Area:

It covers 35 to 40,000 acres of which I estimate 50 per cent is covered with present-day operable timber which contains as a guess 500 million to 1 billion feet of merchantable volume. In any case, it is evident that a substantial volume and life is possible.

Topography:

The topography of this part of the working circle is rugged amd steep. Elevations range from 1000 to 5000 feet, but most of the timber will be found between 20000 and 4000 feet elevation. It covers all of one and parts of two watersheds. All timber will be trucked eventually up through a pass at 3000 feet elevation to a railroad which has already been developed.

It is impractical to haul the timber all down hill through a steep rocky gorge and thus entail excessive road construction costs and in addition a longer mainline haul. Already as evidence of the difficulties of logging and construction, the 6 miles of 5 per cent adverse 18 foot road which has been constructed from the pass to the main creek has cost an average of $5000 per mile. Additional roads will probably average $4000 to $5000 per mile.

Timber:

The timber to be cut during the first cutting cycle is largely Douglas Fir which will probably average about 1500 feet to the log.

Length of Haul:

Hauls for the first three or four years at least, probably will not exceed 20 to 24 miles on a round trip, 6 miles of which will be hauling the loads up a 5 per cent grade.

Production Expected:

I assume that about 200,000 feet per day will be the desired average production extending over about four months of ideal weather, four months of intermittently wet, and one to two months of continually wet weather.

Costs:

There is no better equipment to be found anywhere on the coast for moving the logs from the stump to the landing. A substantial part of this consists of ten RD8 Caterpillar Diesel tractors. It is natural to expect that the same quality of truck will supplement this equipment. However, it is prudent to admit that in spite of this equipment and careful management, costs will probably increase because of the longer haul and more diffucult logging conditions. At least this is certain--even though stump to landing costs remain the same, one more operation between the landing and the railroad is necessary and no matter how cheaply it is done it will add to the total costs. This factor should be considered in determining stumpage appraisal and in preparing a budget of income and expense for this period.

COMPARATIVE SPECIFICATIONS

Class of Motor Truck Necessary:

The West Fork Logging Company has a winding private road 6 miles long with a 5 per cent adverse grade which will be used for many years to come. This fact has more effect upon the choice of a motor truck than any other. This fact eliminates from consideration more than $\frac{3}{4}$ of all the models of trucks now being used to haul logs. Most of those makes and models which do remain for study are so much alike in their important specifications that if the cab were removed it would be difficult to tell one from the other. More differences occur between different models of the same manufacture than between comparative models of different manufacture. This is due to the fact that most of the heavy-duty trucks are assembled jobs. Timken axles are the basis for most of these assemblies, and the same motors with few exceptions are available in all of these makes.

Choice of a motor truck for logging at the West Fork Logging Company is dependent more upon the relative performance of various motors on the 5 per cent adverse grade. For this reason the major portion of this report is devoted to the method of determining performance and the relative performance of various motors with different gear reductions, tire sizes, and gross weights.

In the process of this work I have contacted at least forty people all over the San Francisco Bay Region who are earning their livings in same phase of truck, tire, or motor work. I also made a four day trip over the Yosemite Sugar Pine Company operations with the Logging Superintendent and a truck logging contractor who was going to bid on a 70 million feet job including road construction. The proposed output was about 300,000 feet per day to a railroad head.

To these people I am deeply indebted. I am also indebted to the many articles in "The Timberman" and the "West Coast Lumberman" magazines which supplied worthwhile information. Most of all I am grateful for the opportunity I have been given to make a practical investigation under the auspices of the West Fork Logging Company in the pleasant atmosphere of the University of California.

Weight of Truck and Load:

Basically the purpose of a logging truck is to move logs from one place to another. The cost with which this movement is made is dependent upon many factors, but only insofar as the weight of the truck and its load affects these costs are we concerned in this section.

The greatest effect of weight on the transportation of logs is whether or not the load is to be both pulled and carried or primarily carried. Since the West Fork Logging Company has a 6 mile 5 per cent adverse grade which covers about ½ the distance the loads must be moved, its problem is just as much a pulling problem as a carrying one.

This means that although a number of light trucks such as the Ford, Chevrolet, and Dodge with dual axle trailers are capable of carrying huge weights as compared to larger trucks, yet their ability to pull comparable loads up adverse grades is hardly worth considering. Even with the International Model 426F equipped with a451 cubic inch displacement motor and recommended by the salesman for this job, its capability as indicated in Table XXIV on page ____ of hauling up the 5 per cent grade is only 44,000 pounds (including weight of truck) at 7 miles per hour as compared to the ability of the Cummins Diesel HBS-6 in, let us say, the Kenworth Model 548 of hauling 93,000 pounds at 7 miles per hour.

As shown in Table I, the weight of the International truck is 13,900 pounds and of the Kenworth 20,160. With appropriate trailer combinations the combined weights of trailer and truck of each would be about 20,000 and 29,000 pounds respectively. This would mean a pay load of 24,000 pounds or about 4,000 board feet and 54,000 pounds or about 9000 board feet respectively. This additional 5000 board feet at this speed can be hauled with little if any additional labor cost.

The list price on the Kenworth and the Cummins HBS-6 is about 5 per cent higher so that no accompanying saving in investment per thousand of hauling ability is affected. If anything, probably depreciation per M on the larger truck and motor will be greater.

These figures are substantiated by the experience of the Grande Ronde Pine Company in 1937. An analysis of the costs figures presented by Truman Collins reveals that the heavy diesel trucks reduced the operating labor cost by 32¢ per M or 48 per cent over the light trucks. Depreciation costs, although they increased 7¢ per M on 27 per cent, still showed a net saving of about 25¢ per M on these two items alone without considering repairs and supplies.

The Kenworth, Autocar, Sterling, and Peterbilt trucks are built of almost the same equipment. From any one of these companies can be purchased upon order almost the identical equipment which may be purchased from any other. There are, however, published specifications of certain standard models which I have used to illustrate features important in choosing a truck. Table I shows the respective weights of various models of trucks.

TABLE I
Weight of Truck Empty in Pounds

MAKE	CHASSIS AND CAB	WEIGHT ON AXLE					
		FRONT	%	FRONT REAR	REAR REAR	TOTAL REAR	%
Kenworth 548	20,160	8560	43	5800	5800	11600	57
Sterling HCS 255H	17,025	---	--	---	---	---	--
Autocar DC 10,064	17,545	827	47	5360	3915	9275	53
Mack FK	16,545	8485	57	one axle only		8110	49
Peterbilt-Logger	no data available						
International DR-426-F	13,900	5100	37	4400	4400	8800	63

The distribution of weight between the front and rear axles has some bearing on the way the payload should be distributed in order to utilize the maximum carrying capacity of each axle without overloading any one axle.

Ordinarily, for negotiating the winding and rougher logging roads, it is best to place the log bunk on the truck in a way so that only about 5 per cent of the payload is carried on the front axle. This is accomplished by making the distance of this bunk to the center of the rear axle 5 per cent of the length of the wheel base.

If, for example, the 216 inch wheel base in the Kenworth Model 548 shown in Table II were used, the bunk on the truck should be placed 11 inches in front of the center of the rear axles. This would leave 107 inches or 9 feet between the cab and the bunk in which to shift the load so that a proper balance with the trailer axles can be affected. It probably would be better to shorten this wheelbase to about 190 inches for easier maneuverability and still have about 8 feet for adjusting the loads. A shorter frame also will help prevent a tendency to buckle the frame with the load.

On smooth highways it is common to find 15 per cent of the payload weight carried on the front axles, but where roads are winding and comparatively rough, steering ease is very important.

TABLE II
Important Dimensions
of some Standard Truck Models*

MAKE	WHEELBASE	CAB TO CENTER REAR AXLE	CAB TO END OF FRAME	OVERALL WIDTH
Kenworth 548	216	129	181	--
Sterling HCS 255H	211	123	192	--
Autocar CD 10064	145	97	159	98¼
Mack FK	160	91	126	106
International DR-426-F	215	138	222	90¼

* The wheelbase dimensions in each model can be varied considerably. These are intended to be illustrative of the important dimensions.

Along the line of truck dimensions, the test engineer for the Peterbilt factory told me that the maximum width of a truck with Timken axles should not exceed 115 inches. Widths over this amount puts too much of the load on the inside wheel bearings when the load is placed on the truck. On the other hand the width dimensions should be large enough so that more frame structure is available and also to prevent excessive side sway. Transfer of the load from one side of the truck to the other as it moves over the road causes overloading on the individual wheel bearings and tires. It is apparent in Table II that the overall width of the International Model

DR-426-F is comparatively narrow to the Mack or the Autocar. In view of the fact that it will be necessary to provide at least 9 foot bunks, the minimum width should not be less than 100 inches.

Weight of Logs:

The standard measurement of logs is volume in terms of board feet, but as far as truck operation is concerned the weight hauling ability in terms of pounds is a better measure of relative performance. From one region to another, within different species, within various sizes, and even within the species itself, the weight of logs per cubic foot will vary.

It becomes necessary then to determine the weight of the average Douglas Fir log per board foot at the West Fork Logging Company. R. W. Pratt, Logging Engineer, has made a study of the weights of logs and has published a table of weights according to diameter and length in the March, 1939, issue of "The Timberman."

I have found from a study of the records of the West Fork Logging Company that the average log size of Douglas Fir in 1939 was 1704 board feet net, Scribner scale. From a previous study of 1600 logs with regard to their sound scale percentages I found that a log of this size was appoximately 86 per cent sound scale. Hence the gross volume of the average log is about 1980 board feet. This would be a log 35 inches in diameter and 36 feet long. According to Pratt's table, a Douglas Fir log of this size would weigh 1150 pounds or 5.8 pounds per board feet. For a safety factor and for convenience, 6 pounds per board foot has been used.

Weight of Average Load:

The average railroad carload of Douglas Fir during 1939 contained 8800 board feet net scale or 10,240 board feet gross scale. At 6 pounds per board foot, the average load would weigh 61,440 pounds. An approximate weight of a heavy-duty truck and trailer is 29,000 pounds and if hauling the average railroad carload the total weight would be about 90,000 pounds.

According to my calculations in Table XXIV a HBS-6 Cummins Diesel motor could pull this weight up a 5 per cent grade at about 7 miles per hour.

Weight of Maximum Load:

I doubt if many, if any, railroad cars contained over 12,000 board feet net or about 13,000 feet gross scale. This maximum load would weigh about 84,000 pounds. This amount plus the 29,000 pounds of truck and trailer would be 113,000 pounds gross; this is a tremendous weight to be moved up a 6 mile 5 per cent grade. The HBS-6 motor as shown in Table XXV could move this amount at 6 miles per hour.

Carrying and Moving Capacities:

Before determing what size loads should be moved, it is wise to learn what size loads can be carried without overloading and what size loads can be moved at efficient speeds. This is primarily a problem, first, of axle, frame, and spring capacities; second, of driving units and gear reductions, and third, motor power abilities.

Front Axle:

As stated above only 5 per cent of the payload is carried on the front axle. Nevertheless, the capacity of these axles should be sufficient to carry even this load with safety.

As shown in Table III the axles are practically identical in design and manufacture. If, as in the case of the Autocar, there are 8270 pounds of truck weight (Table I) and 80,000 pounds of carried weight, one-half of which is on the truck and one-half on the trailer, the front axle would have to carry 5 per cent of 40,000 pounds or 2000 pounds additional weight or a gross weight of 10,270 pounds. This is within the rated capacity of 12,000 pounds of this axle. This and other capacity ratings are probably conservative, but too much weight should not be placed on the front axle in order to permit steering ease.

Steering ease is also facilitated by the use of smaller dimension tires such as 9.75x34 instead of the ones used on the rear axles such as 11.25x24. Since the carrying capacity of these tires is about 18 per cent less, it becomes increasingly important to keep the weight on the front axles at a minumum.

cent are worn out. This, he claims, is due to the fact that the power is applied wrong through improper use of the clutch and to poor selection of gear ratios and is not due to too much power.

Capacities:

It has been extremely diffucilt to secure the capacities and torque resistance ratings of each of the axles shown in Table IV . The correspondence on page ____ indicates the unwillingness of the Timken Detroit Axle Co. to make any commitments without very detailed specifications from the manufacturer. I was finally able to uncover these ratings at the Peterbilt truck plant in Oakland. There I learned that all axles in the so-called "400" and "66,000" Timken series are equipped with #742-749 outer and #752-759 inner wheel bearings which have a combined rated capacity of 16,710 pounds. For each axle as shown in Table IV, the capacity of each axle is then 33,420 pounds, and for each rear end with a dual axle there is a total capacity of 66,840 pounds.

The heaviest dual-axle trailer I have seen is the Isaacson Karry Quad Kolossal with a total carrying capacity of 50,000 pounds. However, I have not in any comparable manner made a study of logging trailers so that I do not feel capable of making any specific recommendations. Nevertheless, it is apparent that the combined rated carrying capacity of the axles of the truck and trailer of this type is 120,182 pounds (5 per cent of [illegible]

TABLE III
Front Axle

MAKE	MANUFAC-URER	TYPE	MODEL#	RATED CAPACITY
Kenworth 548	Timken	Reverse Elliot I Beam	36,100 TW	10,000*
Sterling HCS 255H	do	do	27,452	12,000
Autocar DC 10064	do	do	do	do
Mack FK	---	do	---	---
Peterbilt-Logger	do	do	27,052 N	12,000
International DR-426-F	---	---	---	---

* approximate

REAR AXLE AND DRIVING UNIT

Aside from the motor, there is no more important unit in a truck than the rear axle. More serious trouble is found here because it is constantly receiving the power of the motor and also feeling the resistance of the load. When anything gives under these stresses, it is likely to be in the rear axle. When the torque of high powered motors is increased as much as 143 times (Table IX) through gear reductions it must be built to withstand these terrific pressures.

The general manager of the Cummins Diesel Sales Corporation, R. P. Meehan, has told me that he estimates 90 per cent of the rear-ends are broken and only 10 per load carried on front axle.) Deducting 29,000 pounds of truck and trailer weight leaves 91,000 pounds of payload or about 15,000 pounds of board feet gross scale or about 13,000 feet net or water scale. This rating contains a safety factor of some unknown amount to me and which naturally varies according to the operating conditions to which the trucks are subject. However, this gives a fair idea of the weight these axles will withstand. Combined with this information, it is necessary to determine what size tires are necessary to carry this weight and the miles per hour with which this maximum load can be brought up the 5 per cent adverse grade before it is decided that this maximum capacity should be utilized and not limited by some other factor. This is done in progressive steps in this report.

Returning more specifically to the rear axle, I learned that there is one larger axle made by Timken (model #SW-520) which has in total on the two rear axles a bearing capacity of 39,560 pounds. This is 4 per cent more capacity, but its increase in weight is far out of proportion to its increase in capacity.

TABLE IV
Rear Axle and Driving Unit

MAKE	MANUFACTURER	MODEL#	STANDARD RATIOS*	# OF AXLES	RATED CAPACITY		TYPE		
					Bearings, 1 Axle	Torque	Worm	Double Reduction	Chain
Kenworth									
548	Timken	5D-463	8,7:1	2	33,420	5320		✓	
550	do	59,000	8.94:1	1	----	--			✓
Sterling									
HWS 235H	do	SW 452	----	2	33,420	5320	✓		
HCS 255H	do	59,000	----	1	----	--			✓
Autocar DC 10,064	do	SD 452W	10.01	2	33,420	5320		✓	✓
Mack FK	----	------	3.45:1	1	----	--			✓
Peterbilt-Logger	Timken	66,796	8.5:1	2	33,420	--	✓		
International DR-426-F	Inter.	RF-1701	8.05:1	2	17,500	--		✓	
GMC ADCW-974	Timken	SDD-452	7.97:1	2	33,420	5320		✓	

* Ratios that I know are available in Timken Models range from 6.0:1 up to 11.67:1.

If this maximum load were to be moved, we assume that the weight placed on each axle is proportional to its capacity. However, this is not possible unless the load is carefully placed on the bunks with relation to the distance between the bunks, the amount of overhang on the front bunk, and the effective length of the load in feet. Proper placement will save, even an average loads, uneven and undue tire wear due to overloading.

R. W. Pratt has worked out a method of distributind these weights correctly in the August, 1939, issue of "The Timberman," but I shall not go into its details here.

Dual versus Single Rear Axle:

Both driving units of two rear axles on 8 tires and of one rear axle on 4 tires are used extensively in hauling logs. Both have their particular advantages.

The two axle unit with 11.25x24 heavy duty sixteen ply tires has a load capacity of 56,480 pounds as against 39,320 pounds with four 13.50x24 sixteen ply tires, or a difference of 17,160 pounds or an additional 3000 board feet capacity on the front end of the load. This two axle unit, in addition, is easier on the road because it spreads the weight over more surface.

However, on empty trips where insufficient traction is available because the rear end weight is spread over almost twice as much surface, a single axle unit is useful. For example, the Mack truck shown in Table I has 8110 pounds on its one axle while the Kenworth has only 5800 pounds on each rear axle. Hence, the Mack could climb a 31 per cent grade empty at 4 miles per hour as compared to 15 per cent at 8 miles per hour for the Kenworth. However, when grades are so steep against the empties, the trailer could be loaded on the rear-end of the Kenworth and enable it to climb a 34 per cent grade at 4.8 miles per hour with an HBS-6 motor, 11.25x24 tires, and a 45.7 total gear reduction on a dry gravel road.

Some units are manufactured with two rear axles, but with only one axle driving. This type, although it adds to the load carrying capacity, acts as a dead weight on adverse grades. Two rear axles make the truck less easy to manuever than does one axle, but the additional carrying capacity, as long as both axles are driving, outweigh this disadvantage.

Four Wheel Drive:

A number of manufacturers such as the Walter Truck Company or the Four Wheel Drive Auto Company feature a four wheel drive, two axle truck. These have a number of good features such as the ability to roll up out of ruts and holes with some of the pulling power on the front wheels. Some of these are now in use in the Douglas Fir region.

However, when a rim pull of 15,000 pounds is developed as shown in Table XXI there must be at least 15,000 pounds on each axle in order to utilize this power without slipping. The chassis weight on the front axles is no more than 10,000 pounds, and even though 13.50x24 tires capable of hauling 20,000 pounds were used, this amount of weight on the front axle would further complicate the steering already made more difficult because of the fact that the front wheels are being driven.

Torque Resistance:

The related torque resistance factor of the SW 452 and the SDD 452 is given in the Peterbilt Factory table as 5320 pounds. To determine the ability to resist torque of a motor for any gear ratio, this factor is divided by the total gear reduction. For example, as shown in Table IX the total gear reduction in under drive low is 68.4; then the amount of motor torque which supposedly could be resisted is $\frac{5320}{68.4}$ or 78 pounds. However, the operating torque of the HBS-6 is 607 foot pounds or about 8 times the capacity rating. At the present time the HBS-6 motor is not available in even the largest Mack trucks because of this large torque in comparison to the capacity of the drive.

In actual practice, though, the use of an auxiliary transmission in order to increase the pulling power through greater gear reduction was originally opposed by axle manufacturers. Experience has proved that it is not the total pressure which was applied which broke rear ends but the manner in which this power was applied. A sudden jerk due to large differences in gear ratios from one shift to another has caused more trouble than when auxiliary gears are used to provide intermediate steps between these large jumps and to reduce the lowest gear to a still lower gear known as under drive low.

Rear Axle Ratios:

As shown in Table IV, a number of rear axle ratios are available and may be varied to meet the particular hauling conditions. This ratio constitutes the last of the three gear reductions from the motor to the rear wheels. Tables VII, VIII, and IX indicate the effect on total gear reduction and available road speeds with the use of different rear axle ratios. These tables are also basic to determining power performance shown in Tables XXI, XXII, and XXIII. The use of a 6.1:1 ratio instead of the 10.25:1 ratio will produce with Model #703 auxiliary transmission 20 more miles per hour or 48.9 miles in over drive 4th but 70 per cent less power in compound low. A fair compromise between power and speed lies in a ratio similar to 8.7:1 in Table XXI. Herein is found all the power which can be used with regard to carrying capacities of the tires and axles and yet produce usable top road speeds.

Driving Unit:

As shown in Table IV the type of driving unit is the distinguishing characteristic of each of the models indicated. I do not claim to know the value of a worm drive over a double reduction drive. Both are enclosed the worm is a little simpler but I have received no indication that it is less likely to give trouble.

In the case of the chain drive, there are a number of engineering points in its favor over the enclosed drives. It has greater mechanical efficiency and therefore can deliver more tractive effort to the wheels; it bives a drive at each end of the axle instead of at the center only. It tends to pull the wheel down whereas the gear or enclosed drives have a tendency to lift the wheels. Also it is less likely to give final drive trouble. Nevertheless, some loggers conplain that the chains bind in mud or dust, and that if sticks or brush get caught the chain might break. At the present time, I would not like to recommend one over the other until I have had an opportunity to look into the matter further.

MAIN TRANSMISSION

The transmission is the first step in the conversion of power at the motor speed to greater power at less speed. Until recently, only four or five different reductions which were in the main transmission were possible. Consequently, as shown in Table IX, this limited number causes a wide gap between shifts. Between 1st and 2nd, the ratio for Model #7341 is 2.84. This means as indicated in Table IX that a truck must almost double its road speed (4 miles per hour to 7.4) before a shift can be made. This is very difficult on adverse grades with capacity loads.

Various ratios are available as shown in Table V, and selection should be based upon the particular hauling conditions. However, I strongly recommend the Brown-Lipe make over the International or Mack make, because this can be supplemented with an auxiliary also made by the Brown-Lipe Company in which 12 forward gear shifts are possible but in the case of the Mack TR15 only 8 and in the case of the International only 5. The spread between the low and the high in the 5 speeds is the same as in the 12 speeds in the Brown-Lipe models and may account to some degree for the rear-end trouble the Diamond Match Company has experienced with large diesel motors in International Trucks. As previously explained, jamming from one gear shigt to another has been the chief cause of rear-end trouble. This is certain to be minimized when it is possible to obtain increasing or decreasing road speeds through gradual shifts.

AUXILIARY TRANSMISSION

Table IX indicates that if mothing but a main transmission were used and the HBS-6 motor kept operating at its most efficient speed of 1650 RPM, the only road speeds available would be 4.0 miles per hour, 7.4 miles, 14.6 miles, and 25.3 miles. An auxiliary transmission by means of an over drive and an under drive gear in a separate transmission installed behind the standard transmission provides two additional shifts for each shift in the transmission; thereby with the Model #703, 8 additional speeds ranging from 1.5 to 33.8 miles per hour are provided.

TABLE V
Transmission

MAKE	TYPE	GEAR RATIOS 5th	4th	3rd	2nd	1st	Reverse
Kenworth 548	Brown-Lipe	optional	1.00:1	1.73:100	3.43:1	6.27:1	8.15:1
Sterling HCS 255H	do	do	do	do	do	do	do
Autocar DC 10064	do	do	do	do	do	do	do
Mack FK	Mack TR15	do	1.00:1	1.75:1	3111:1	5.48:1	5.53:1
Peterbilt-Logger	Brown-Lipe	do	1.00:1	1.73:1	3.43:1	6.27:1	8.15:1
International DR-426-F	International F-54B	0.776:1	1.00:1	1.72:1	3.50:1	7.07:1	7.11:1
GMC ADCW-974	-----	0.69:1	1.00:1	1.76:1	3.02:1	5.07:1	5.07:1

The Model #703 auxiliary was designed for lighter trucks so the heavy loads could be pulled with the motors available and to produce high speeds when out on the open highway. Hence, a large under drive ratio of 2.62 and a small over drive ratio of 0.75 was set to accomplish this. Little or no regard was paid to the ease of shifting or to the effect of the ratio on fuel consumption and motor speed until more powerful motors were developed.

During the summer of 1939, the Long Transportation Company of Los Angeles, cooperating with the Cummins Engine Company of Columbus, Indiana, conducted a series of road tests to determine the causes and find the remedy for smoky engine exhaust. These tests showed that the grade of fuel used, combined with inefficient air cleaners and mufflers, together with irregularities in transmission ratios were the principal contributing factors.

Of the three, the transmission ratios were determined upon as the greatest offender. It was noted that the heaviest smoke occurred during and immediately after a shift of gears.

In Table IX, it can be seen by following the arrow that to shift from a lower speed to a higher speed requires quite a bit of irregular shifting. For example, starting out in under drive low at 1.5 miles per hour, then to under drive second at 2.8 miles, then back to direct first at 4.0 miles, then to over drive low, then all the way up to under drive third, etc.; this requires an undue amount of double

AUXILIARY TRANSMISSIONS AVAILABLE

MAKE	TYPE	# SPEEDS	RATIO OVER DRIVE	RATIO UNDER DRIVE
Kenworth	Brown-Lipe			
	703	3	.747 to 1.00	2.62 to 1.00
	703A	3	.84 to 1.00	1.25 to 1.00
Sterling HCS 225H	do	do	do	do
Autocar DC 10064	do	do	do	do
Peterbilt-Logger	do	do	do	do
Mack FK	Mack TRA 12	2	none	2.33
International DR-426-F		2	none	1.207 to 1 (power divided)
GMC ADCW-974		3	0.77 to 1	1.99 to 1

shifting and therefore a consequent loss of fuel and power.

From this study there has been developed and just put on the market auxiliary transmission #703A which corrects these difficulties. In the new set of gears no two transmission ratios overlap. There is direct progression as shown in Table IX from under drive low to over drive high. In addition the spacings between these shifts are uniform. In the #703, until the road speed is 9.8 miles per hour, the differences in speed from one shift to the other vary between 1 and 2 miles per hour for the first 6 shifts, but between underdrive 4th and over drive 2nd it is increased by only 0.1 of a mile per hour, then suddenly jumps 4.8 miles per hour into direct third. In the 703A, the differences start slowly with 0.8 of a mile difference and gradually increase until over drive fourth is reached with an increase of 5.1 miles per hour.

It probably will not be possible to use this new auxiliary transmission in any of the lighter trucks which must haul heavy loads. However, in the larger trucks with powerful motors, a slight change in the rear axle ratios will give sufficient total gear reduction for pulling ability without unduly sacrificing speed. At the present time I recommend for the West Fork Logging Company 5 per cent adverse road a 8.7:1 rear axle ratio, Model 7341 main transmission and Model 703A auxiliary transmission.

The G. M. C. heavy duty Model ADCW-974 recommended for this job has an auxiliary transmission ratio as indicated in Table VI of 0.77:1 over drive and 1.99:1 under-

TABLE VII

TOTAL GEAR REDUCTION AND ROAD SPEEDS

TRUCK Kenworth 548

UNIT TRANSMISSION 7351   AXLE 10.25:1   TIRE SIZE 11.25x24

ENGINE Cummins Diesel HBS-6   OPERATING SPEED 1650

GEAR USED		Using 703 Auxiliary 2.62 Under & .75 Over			Using 703A Auxiliary 1.25 Under & .84 Over		
IN AUXILIARY	IN MAIN	Trans Ratio	Total Ratio	MPH at 1650 RPM	Trans Ratio	Total Ratio	MPH at 1650 RPM
UNDERDRIVE LOW			167.9	1.3		80.4	2.6
	DIRECT LOW	6.27	64.5	3.4	6.27	64.3	3.4
OVERDRIVE LOW			48.1	4.6		53.9	4.1
UNDERDRIVE 2nd			92.8	2.4		44.0	5.0
	DIRECT 2nd	3.43	35.2	6.2	3.43	35.7	6.2
OVERDRIVE 2nd			26.3	8.3		29.5	7.5
UNDERDRIVE 3rd			46.5	4.6		22.2	10.1
	DIRECT 3rd	1.73	17.7	12.4	1.73	17.7	12.4
OVERDRIVE 3rd			13.3	16.5		14.9	14.7
UNDERDRIVE 4th			26.8	18.2		12.8	17.2
	DIRECT 4th	1.0	10.25	21.4		10.25	21.4
OVERDRIVE 4th			7.7	28.5		8.6	25.5

TABLE VIII

TOTAL GEAR REDUCTION AND ROAD SPEEDS

TRUCK Kenworth 548

UNIT TRANSMISSION 7351 AXLE 6.1:1 TIRE SIZE 11.25x24

ENGINE Cummins Diesel HBS-6 OPERATING SPEED 1650

GEAR USED		Using 703 Auxiliary 2.62 Under & .75 Over			Using 703A Auxiliary 1.25 Under & .84 Over		
IN AUXILIARY	IN MAIN	Trans Ratio #7351	Total* Ratio	MPH at 1650 RPM	Trans Ratio #7351	Total* Ratio	MPH at 1650 RPM
UNDERDRIVE LOW			98.5	2.2		47.0	4.7
	DIRECT LOW	6.27	37.6	5.9	6.27	37.6	5.8
OVERDRIVE LOW			28.1	7.8		31.6	7.0
UNDERDRIVE 2nd			55.5	4.0		25.7	8.5
	DIRECT 2nd	3.43	20.6	10.7	3.43	20.6	10.7
OVERDRIVE 2nd			15.9	13.8		17.3	12.7
UNDERDRIVE 3rd			27.2	8.1		13.0	17.0
	DIRECT 3rd	1.73	10.4	21.2	1.73	10.4	21.2
OVERDRIVE 3rd			7.7	28.6		8.7	25.3
UNDERDRIVE 4th			15.7	14.0		7.5	29.4
	DIRECT 4th	1.0	6.0	36.6	1.0	6.0	36.6
OVERDRIVE 4th			4.5	48.9		5.05	43.5

REMARKS:

*Total Ratio found by multiplying transmission ratio by auxiliary ratio by axle ratio.

TABLE IX

TOTAL GEAR REDUCTION AND ROAD SPEEDS

TRUCK Kenworth 548

UNIT TRANSMISSION 7351 AXLE 8.7:1 TIRE SIZE 11.25x24

ENGINE Cummins Diesel HBS-6 OPERATING SPEED 1650

GEAR USED		Using 703 Auxiliary 2.62 Under 7 .75 Over			Using 703A Auxiliary 1.25 Under & .84 Over		
IN AUXILIARY	IN MAIN	Trans Ratio #7351	Total Ratio	MPH at 1650 RPM	Trans Ratio	Total Ratio	MPH at 1650 RPM
UNDERDRIVE LOW			143.0	1.5		68.4	3.2
	DIRECT LOW	6.27	54.6	4.0	6.27	54.6	4.0
OVERDRIVE LOW			41.0	5.4		54.7	4.8
UNDERDRIVE 2nd			78.2	2.8		37.4	5.9
	DIRECT 2nd	3.43	29.8	7.4	3.43	29.8	7.4
OVERDRIVE 2nd			22.4	9.8		25.0	8.8
UNDERDRIVE 3rd			39.4	5.6		18.8	11.7
	DIRECT 3rd	1.73	15.1	14.6	1.73	15.1	14.6
OVERDRIVE 3rd			11.3	19.5		12.6	17.5
UNDERDRIVE 4th			22.8	9.7		10.9	20.2
	DIRECT 4th	1.0	8.7	25.3	1.0	8.7	25.3
OVERDRIVE 4th			6.5	33.8		7.3	30.2

drive. This combination does not result in a matched ratio as in the case of Model 703A. Table X indicates in the G. M. C. models that a shift must be made from under drive low to over drive low to under drive second to over drive second, then back to direct first, then to under drive third, etc. This not only wastes fuel and power as indicated by the study made in Los Angeles but requires the driver to memorize jumbled shifts in their various orders in order to progress from one shift to another.

It might also be noted that between under drive low and over drive low there is only 0.1 of a mile per hour. The next shift increases by 2.6 miles and the next by 0.2 miles. In other words the steps are not equally divided between the different gears. This unequal division is one of the most serious handicaps of the G. M. C. Model and is over come completely in the Brown-Lipe Model 705A.

The International Model, as explained above, has only five shifts between each of which the road speed as shown in Table X must be doubled for the first three and most difficult shifts in which to pick up speed.

TOTAL GEAR REDUCTION AND ROAD SPEED
Various Gear Ratios

GEAR SHIFT	BROWN-LIPE 7341 and 703A 8.7 AXLE RATIO		TRANSMISSION MODELS GMC 7.97 AXLE RATIO		INTERNATIONAL 1.207 Pr. Reduction 8.05 AXLE RATIO	
	TGR	MPH	TGR	MPH	TGR	MPH
U 1	68.4	3.2	80.4	3.0		
D 1	54.6	4.0	40.4	5.9	68.7	3.9
O 1	45.7	4.8	77.5	3.1		
U 2	37.4	5.9	47.9	5.0	34.01	7.9
D 2	29.8	7.4	24.0	10.0		
O 2	25.0	8.8	46.2	5.2		
U 3	18.8	11.7	27.9	8.6		
D 3	15.1	14.6	14.0	17.7	16.72	16.0
O 3	12.6	17.5	26.9	8.0		
U 4	10.9	20.2	15.9	15.1		
D 4	8.7	25.3	8.0	30.0	9.72	27.5
O 4	7.3	30.2	15.3	15.7		
U 5			10.9	22.0		
D 5			5.5	43.5	7.45	35.4
O 5			10.6	22.6		

CLUTCH

Although not as important an item as the above mentioned specifications, proper choice of a clutch may prevent a number of delays and breakdowns.

The best model now installed in heavy duty trucks is the 14 inch, two plate Brown-Lipe model with an estimated engaging area of 430 square inches. This involves no greater cost than the 13 inch, two plate now installed in many heavy duty trucks with an engaging area of only 360 square inches. The 13 inch, two plate, as shown in Table XI is capable of resisting a torque when standing still of 900 foot pounds without slipping. Most operators have found this sufficient resistance, but as long as the cost is no greater, the additional capacity might as well be obtained.

The Autocar, Mack, and International models have less engaging area and therefore are less desirable on this item.

TABLE
Clutch

MAKE	TYPE	DIAMETER	# PLATES	ENGAGING AREA	PULL OFF TORQUE RATING
Kenworth 548	Brown-Lipe	14 inches	2	430 sq.in. (est.)	1000 ft.lbs.
Sterling HCS 255H	Brown-Lipe	13 inches	2	360 sq. in.	900 ft. lbs.
Autocar DC 10064	W. C. Lipe	15 inches	1	256 sq, in.	-----
Mack FK	W. C. Lipe Z42SX	15 inches	1	253.8 sq. in.	-----
Peterbilt Logger	Brown-Lipe	13 inches	2	360 sq. in.	900 ft. lbs.
International DR-426-F	International R-14-15	14 inches	1	174.4 sq. in.	-----

FRAME

Weakness of frame construction has been one of the most serious causes of breakdowns, yet it is one of the easiest troubles to avoid of construction is planned right in manufacture. An attempt has been made to combine strength in an alloy known as corten which is a pressed chrome manganese heat treated steel.

Except in the Sterling and the International models shown in Table XII, this alloy is available. The Sterling frame is the weakest of the various frames shown, not only because of its composition, but also because a wood insert is used inside the channel section. This wood shrinks and swells with a change in atmospheric conditions and loosens the bolts in the frame.

The Kenworth uses an I beam type of construction, but all other makes use a channel section with an additional steel channel inserted within the outside channel for additional strength. Which type affords greatest strength I do not know, but I believe both will perform satisfactorily if deep and wide enough. As indicated in Table XII, these frame dimensions are practically the same.

One very important item which is too often neglected in manufacture is the number of cross members supporting the frame. This number can easily be varied upon order and should be specified. In my judgment, there should be at least one cross member for every three feet of frame length.

TABLE XII

FRAME

MAKE	MATERIAL	TYPE	DIMENSIONS--INCHES DEPTH	WIDTH	THICKNESS	# CROSS MEMBERS
Kenworth 548	Corten Pressed chrome-manganese heat treated steel	I beam	10	--	--	--
Sterling HWS 235H HCS 255H	Pressed chrome-nickle heat treated steel	Channel section without wood insert	9¼	3 5/8	3/8	--
Autocar DC 10064	Alloy steel heat treated	Channel section with steel liner	10½	3	5/16	--
Mack FK	Chrome-manganese heat treated steel	Channel section with inside channel	10 5/8	3½	5/16	6
Peterbilt Logger	Pressed chrome-manganese heat treated steel	Channel section with steel and torque rods, deep center section	10 3/8 (center section)	3½	¼	7
International DR-426-F	Pressed steel	Channel section plate reinforcements and torque rods	11 1/8 (center section) 8 end of section	3 1/16	3/8	6

In order to take off some of the weight on the rear axles, torque rods running from the rear axles to the frame about three feet in front of the rear axles place more stress upon the frame at this point.

Most of the stress cracks in frames have occurred between the front and rear axles just behind the rear of the cab. In order to compensate for these stresses and for the additional stress placed on it from the torque rods, some frames such as the Peterbilt and the International are made deeper in the center section than at the ends.

COOLING SYSTEM

Table XIII indicates the comparative frontal areas of various truck models and the capacity of each. The foreman of the Cummins Diesel agency in San Francisco recommends a total of 48 quarts as a minimum for the Cummins Diesel HBS-6. The Peterbilt is the only model listed which has this capacity, but I understand the latest models of the Kenworth now have a capacity of 54 quarts.

TABLE XIII

COOLING SYSTEM

MAKE	FRONTAL AREA	CAPACITY COOLING SYSTEM
Kenworth 548	640 sq. in.	40 qts.
Sterling HCS 255H	818 sq. in.	---
Autocar DC 10064	723 sq. in.	46 qts.
Mack FK	---	---
Peterbilt	894.25 sq, in.	56 qts.
Inter[illegible]onal	---	31 qts.

BRAKES

It is just as important to keep the trucks under control with motor compression and brakes coming downhill as it is when using power to control the loads coming uphill.

Some models of trucks shown in Table XIV have brakes on only four of the six wheels. The important difference between one and the other is the total lining area. For example, Kenworth, Autocar, and International each have brakes on all six wheels, but the total lining area is 1194, 998, and 800 square inches respectively. An addition of two more brakes to the Sterling model, for example, would involve only 225 pounds of additional weight and would insure sufficient braking power for emergencies.

Aside from this variation, there is little difference in make, type, or in the capacity of the compressor. The number of brakes and their sizes can be specified when ordering a truck. I personally favor the largest sized brakes on all wheels in order to insure safe travel from the pass down to the railroad head.

TABLE XIV

SERVICE BRAKES

TRUCK	MAKE	TYPE	LOCATION	LINING AREA SQ. IN.	COMPRESSOR CU. FT.	HAND BRAKE AREA SQ. IN.
Kenworth 548	Westinghouse	air	all 6 wheels	1194	7.25	120
Sterling HCS 255H	Westinghouse	air	4 rear wheels	868	7.25	147
Autocar DC 10064	----	air	all 6 wheels	998	7.25	120
Mack FK	Westinghouse	air	all 4 wheels	849	12	86
Peterbilt Logger	Westinghouse	air	4 rear wheels	700	7.25	120
International DR-426-F	Westinghouse	air	all 6 wheels	800	7.25	120

SPRINGS

Broken springs have caused breakdowns and delays because of overloading or inadequate capacity in relation to the other capacities of the truck.

Table XV shows a rather wide variation between the models shown. The Kenworth model provides the largest springs and therefore probably the greatest capacity.

TABLE XV

SPRINGS

MAKE	FRONT #LEAVES	FRONT LENGTH	FRONT WIDTH	REAR #LEAVES	REAR LENGTH	REAR WIDTH
Kenworth 548	--	48 in.	4 in.	--	56 in.	5 in.
Sterling HCS 255H	10	48 in.	4 in.	16	52 in.	3½ in.
Autocar DC 10064	15	41 in.	3 in.	12	54½ in.	5 in.
Mack FK	--	50 in.	3½ in.	--	50 in.	5 in.
Peterbilt Logger			data not available			
International DR-426-F	14	44½ in.	3 in.	12	34. in	4 in.

MOTOR

INTRODUCTION

The motor is the most important part of a logging truck. No other piece of the unit affects the amount and speeds of loads, costs of fuel, repairs, and maintenance in as large a measure. Tires and wheel bearings have some limits, but where a 6 mile 5 per cent winding adverse grade must be climbed (which can not be emphasized too strongly as the controlling physical factor at the West Fork Logging Company), then motor capacities become the controlling mechanical factor.

Motor Prices and Total Truck Cost:

The list price on the highest priced unit I have found is $13,281 on the Kenworth model 548. The Karry-Quad Kolossal logging trailer model KQ-2400 with a rated capacity of 48,000 pounds has a list price of $3,605 on 11.25x24 tires. A total price for truck and trailer is $16,886 of which the highest priced and most powerful diesel motor is $3,200 or only 19 per dent of the total cost.

The Kenworth agent in San Francisco has told me that 70 per cent of the repair costs on trucks of this type is because of the motor and only 30 per cent because of the chassis. As evidence of this fact, since the Cummins Diesel Corporation maintains its own service on its motors, this agent does not maintain a shop of his own but has all his chassis work done outside because there is so little.

Other motors and trucks covered in this report reveal that only 15 to 20 per cent of the total cost is in the motor. Since the motor cost is a relatively small part of the cost of a truck and trailer and yet contributes more than any other feature to the repair cost and to the truck's performance, it is evident that a full investigation should be made of this part of a truck without too much emphasis on its initial costs.

Motors Available:

It was stated above that most of the heavy duty trucks are assemblied units. For this reason, a number of different motors are available at the option of the purchaser. The range of choice is in most cases wider than the selection of axles, frame, or tires.

A few of the motors available for each manufacturer are shown in Table XVI. In the case of the Mack truck, the manufacturers are waiting for a longer experience on the HBS-6 Cummins Diesel before making it available in their trucks. It is feared that the motor may be too powerful for the torque resistance of the rear axles.

The GMC Diesel motor 6-71 is not available in any new trucks except GMC models but can be installed in used equipment to replace used motors. The Mack-Lanova motor, I believe, is handled in the same way.

TABLE XVI

MOTORS AVAILABLE

MAKE			DIESEL				GAS
	HBS-6	HB6	GMC 6-71	MACK-LANOVA ED	CATER-PILLAR D468	HALL-SCOTT 177	INTERNA-TIONAL FBB-450
Kenworth 548	/	/			/	/	
Sterling HCS 255H	/	/			/	/	
Autocar DC 10064	/	/			/	/	
Mack FK		/		/	x		
Peterbilt Logger	/	/			/	/	
Interna-tional DR-426-F		/			x		/
GMC ADCW-974			/				

The Caterpillar Tractor Company diesel truck motor D468 is relatively new on the motor market. I cannot state specifically that all the truck manufacturers indicated will install this motor, but I see no reason why this would not be possible.

In any case, it can be seen that for each manufacturer a number of motors are available. The proper choice is as much a responsibility of the purchaser as of the truck assembly companies, because the purchaser knows best what operating conditions the motor must overcome.

Reason for Extensive Study of Motors:

It might pay to pause here a moment to realize why I have made such an extensive study of truck specifications and especially of motors. The other factors which affect the cost of truck logging, such as road location and surface, methods of loading and unloading, and labor, all can be varied to a considerable extent in the field and from day to day. But when the truck is finally purchased, all other factors are made subservient to its abilities. The road surface can certainly be made to sustain the heaviest loads that can be moved, loads can easily be made large or small, and hours of labor can be changed to meet operating conditions. Once a truck with a certain motor is selected, the amount of the loads and speed with which these loads can be moved is practically fixed. Hence, I hope to have gathered information which will help make this selection a wise one. I do not think the final answer will be found in this report, but it will serve to eliminate useless consideration of innumerable types of trucks and best of all to educate us to a better understanding of truck capacities and performance which will aid as much in operation as in selection. Nevertheless, after putting on paper for future reference the facts and understanding I have gathered, I shall recommend a definite piece of equipment for the West Fork Logging Company. Naturally, since selection will not be made for probably a year, new developments and machinery may become more desirable.

Motor Ratings:

For a complete understanding of the comparative performance of various motors all of which are good high-powered motors, it is necessary to learn what the various ratings mean and how they are obtained. Every manufacturer expresses the abilities of his motor in a little different way. To first learn the meanings and use of these ratings and then to reduce each to comparable operating conditions has been the most difficult and yet the most interesting part of this report. It should be borne in mind in the pages to follow that many of the terms and ratings such as torque, horsepower, and break mean effective pressure, will be used in performance formulas. Also, I shall not attempt to go into detail on all of the specifications but only those which have a bearing on comparative performance.

In Table XVIII will be found assembled the most complete set of comparative specifications on the most powerful and most efficient heavy duty truck motors in the United States. I have also included for comparison specifications on the Caterpillar Tractor Company motor D13000 which is used in the Caterpillar Tractor RD8 with which we are so familiar. It is expected that the reader will constantly have this table in front of him while reading this section so that irritating constant references to this table can be avoided.

Number of Cylinders:

Oddly enough, all of the motors shown here have

TABLE XVII

COMPARATIVE SPECIFICATIONS

VARIOUS TRUCK MOTORS AND CATERPILLAR TRACTOR (RD8)

SPECIFICATIONS	HALL SCOTT GAS 177	CUMMINS DIESEL HBS-6	GMC DIESEL 6-71	CUMMINS DIESEL HB-6	MACK ED DIESEL	INTER. FBB-450	CATERPILLAR DIESEL TRUCK D468	CATERPILLAR DIESEL TRACTOR RD 8
NUMBER OF CYLINDERS	6	6	6	6	6	6	6	6
# STROKES TO CYCLE	4	4	2	4	4	4	4	4
BORE (DIAMETER IN INCHES)	$5\frac{1}{2}$	$4\frac{7}{8}$	$4\frac{1}{2}$	$4\frac{7}{8}$	$4\frac{3}{8}$	$4\frac{3}{8}$	$4\frac{1}{4}$	$5\frac{3}{4}$
STROKE (INCHES)	6	6	5	6	$5\frac{1}{4}$	5	$5\frac{1}{2}$	8
PISTON DISPLACEMENT (CUBIC INCHES)	855.3	672	425	672	519	451	468	1246
MOTOR SPEEDS: RPM								
GOVERNED	1800	1800	2000	1800	2200	2400	1800	1000
OPERATING	1600	1650	1800	1650	2000	2000	1800	850
SPEED AT MAXIMUM TORQUE	1100	900	900	700	1350	800	900	650
HORSEPOWER AT:								
GOVERNED SPEED	203	200	165	150	133	108	90	120
OPERATING SPEED	195	194	160	144	131	100	90	108
MAXIMUM TORQUE	146	112	94	66	98	48	40	68
H.P. OF ACCESSORIES NOT DEDUCTED (OPER. SP.)	9	5	27	24	11	0	0	0
OPERATING SPEED WITH ALL ACCESSORIES	184	189	133	120	120	100	90	108
BREAK MEAN EFFECTIVE PRESSURE AT OPERATING SPEED	115	136	106	99	101	765	85	81
TORQUE AT:								
GOVERNED SPEED	570	580	450	435	350	228	262	656
OPERATING SPEED	640	607	472	442	348	262	262	667
MAXIMUM TORQUE	677	670	570	500	382	320	305	766

COMPARATIVE SPECIFICATIONS
(con't.)

SPECIFICATIONS	HALL SCOTT GAS 177	CUMMINS DIESEL HBS-6	GMC DIESEL 6-71	CUMMINS DIESEL HB-6	MACK ED DIESEL	INTER. FBB-450	CATERPILLAR DIESEL TRUCK D468	CATERPILLAR DIESEL TRACTOR RD 8
NUMBER OF MAIN BEARINGS	7	7	7	7	7	7	7	7
AREA OF MAIN BEARINGS (SQ. IN.)	35.9	70.3	27.6	70.3	40.3	32.4	118	214
COMPRESSION RATIO	5:1	17:1	16:1	17:1	14.57:1	5.2:1	17:1	15:1
CAPACITY OF LUBRICATING SYSTEM (QTS.)	---	5	15.5	5	----	----	4¼	6¾
OVERALL DIMENSIONS	60x33 x29	69x49 x33	73x52 x32	63x49 x33	----	----	60x47 x25	100x 60x43
ENGINE SHIPPING WEIGHT (lbs.)	1830	2500	2000	2181	----	1030	2120	5610
POUNDS OF WEIGHT PER H.P. AT OPERATING SPEED	9.4	13.0	12.5	15.2	----	10.3	23.5	51.9
FUEL CONSUMPTION:								
AT OPERATING SPEED lb./BHP/HR	.560	.495	.475	.445	.467	.605	.470	.438
GALLONS PER HOUR UNDER FULL LOAD	17.5	13.1	10.2	8.6	8.2	9.7	5.7	6.4
LIST PRICE $	2400	3200	3200	2400	----	----	1650	3255
FLEET PRICE $	2000	2800	----	2000	----	----	1650	3255

six cylinders, but the power difference between them varies as much as 60 per cent. This is primarily either because of the piston displacement or because of the number of strokes to the cycle.

Piston Displacement:

Piston displacement for motors of the same number of strokes to the cycle offers the quickest method of judging potential power. It does not necessarily follow that the motor with the largest piston displacement will give the best performance, but it does mean that it will produce the most power under certain operating conditions. Larger piston displacement is obtained by either adding to the length of the stroke or the diameter of the bore. Total piston displacement in inches is found by determining the cubic inch volume of each cylinder and multiplying by the number of cylinders. For example, the diameter of the bore of the HBS-6 is $4^{7}/8$ inches and the stroke 6 inches or a volume of 112 cubic inches per cylinder or for all 6 cylinders a total of 672 cubic inches.

This is also the piston displacement for the Cummins HB-6, but in the case of the HBS-6 motor, it has been supercharged to make the 40 per cent more powerful HBS-6. In all other mechanical respects, the two motors are alike.

The other motors shown in the table of the same stroke cycle at the same RPM's vary in power almost directly with the piston displacement.

Number of Strokes per Cycle:

In the case of the GMC Diesel 6-71, it is a two-stroke cycle instead of a four cycle engine so that its piston displacement of 425 cubic inches might be compared to an 850 cubic inch four-cycle motor. However, this does not completely follow, for it produces 20 per cent less horsepower at 1800 RPM than the Hall-Scott 855.3 cubic inch motor. All other motors investigated follow the four-stroke cycle principle.

The General Motors Corporation has made the following statement: "General Motors has gone a step further in perfecting the Diesel engine, by its development of the two-stroke cycle principle. General Motors elected to develop the two cycle principle because it offered the greatest possibilities for reducing the Diesel's size and weight and increasing smoothness." In spite of this statement, actually as far as size is concerned, its overall dimensions are 8 per cent greater than the HBS-6 and although 500 pounds lighter, it weighs 12.5 pounds per brake horsepower at operating speed as compared to 13.0 pounds for the HBS-6.

Motor Speed:

Some motors are designed to turn over faster than others. Generally speaking, the higher the operating RPM, the less the power and vice-versa. The most efficient operating speed of the motor determines to a large extent the road speeds possible. Hence, the selection of the proper motor for the West Fork Logging Company becomes a compromise between power and speed. As will be pointed out in the discussion on performance in terms of hauling power and road speed, power is the most important factor, because speed could be varied only on the loaded trip on the 5 per cent adverse grade which is only one-third of the total round trip distance.

Expressions of horsepower and torque given will vary according to the speed of the motor. Hence, it is very important when comparing motors to specify the RPM (revolutions per minute.) All motors are governed not to exceed a certain speed. This is necessary in order to avoid excessive heat and strain. This, however, is not the operating speed. The difference of 200 RPM, for example, in the GMC motor, between governed and operating speed is set not only for more efficient fuel consumption at operating speeds, but also to permit the motor to be speeded up when shifting from one gear to another, so that by the time the shift is completed, the motor has not slowed down too far if any below operating RPM's.

Operating RPM is the speed recommended by the manufacturer as the most efficient speed of the motor. It is more likely to include not only a definite point, but also a range. Modern truck operation efficiency demands the use of a tachometer, which is a gauge placed in the panel. This gauge indicated the RPM of the motor and is watched carefully by the driver to see that the motor is running at a speed within the recommended range.

All motors possess the ability to pull more loads at less than their operating speeds. At a certain point below these operating speeds, there is developed what is known as maximum torque. The RPM range between this point and the operating point is one of the best ways to judge a motor's performance from the standpoint of RPM. The longer this range, the better the motor for hauling purposes. For example, the distances between these points on the Hall-Scott are 700 RPM; on the RBS-6, 750; on the HB-6, 950; and on the International Harvester FBB-450, 1200. The operation at or near this point is known as lugging the motor. This vibrates bearings, crankshafts, and pistons, and should be avoided if possible.

With the use of auxiliary transmissions and the tachometer, modern operation calls for speeds constantly in the operating range. If additional power is needed, even for short distances, a lower shift is made without changing the RPM of the motor. Theoretically, the only place where this lugging ability was needed would be in compound low where no lower shifts are available.

Trucks such as the International DR-426-F which have only five gear shifts need a longer range between these points so that slight changes in the necessary pulling power can be accomplished by slowing down the motor instead of resorting to a shift in gears. Both methods result in a lower road speed.

Brake Horsepower:

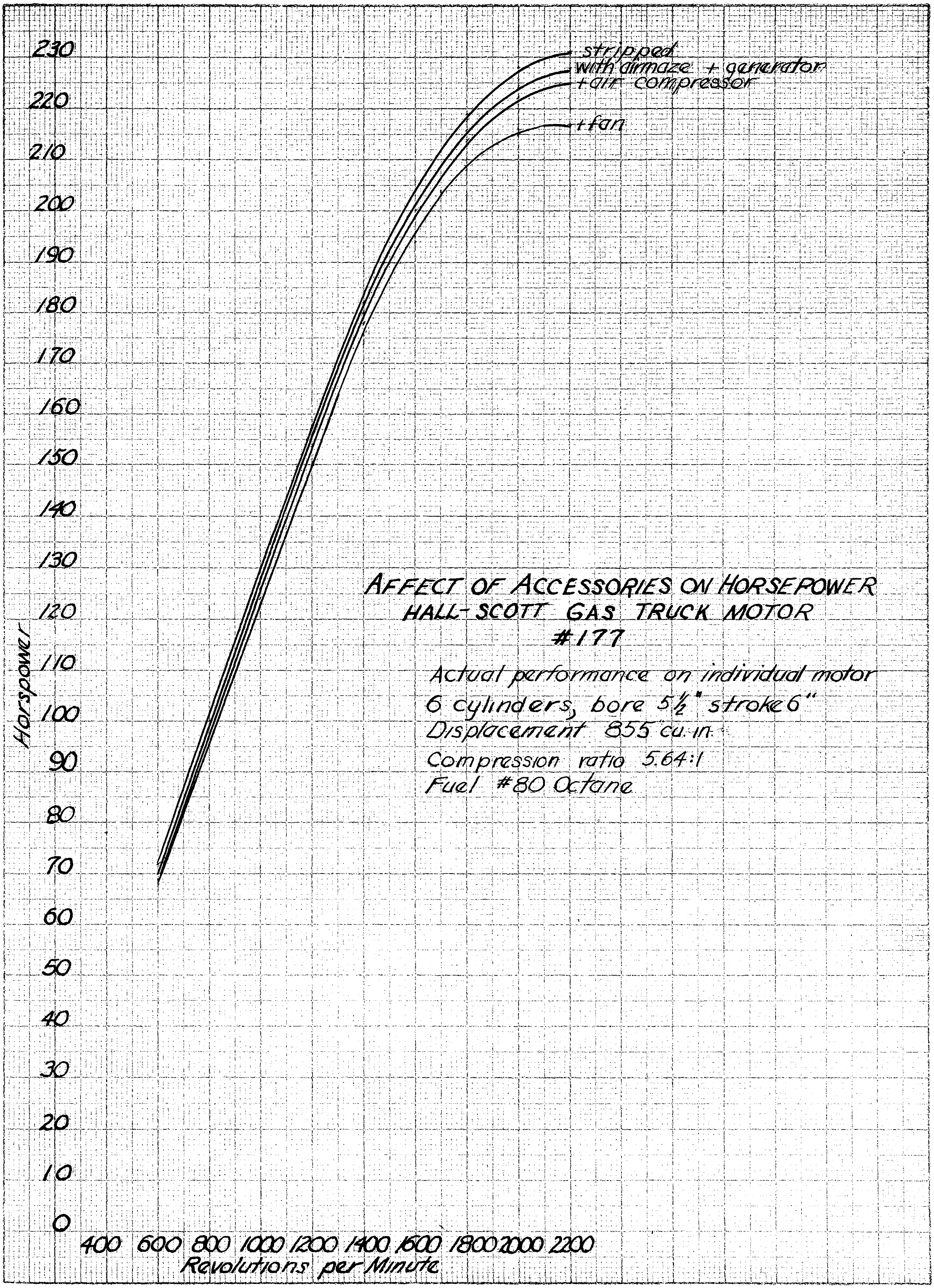
Horsepower is one of the most misused terms in the motor industry. It can be expressed in so many different ways that people who are not familiar with it are often misled or confused. It actually has very little use in computing power and speed performance.

Brake horsepower is merely a combined expression of the power (called torque) and the motor speed. Hence, at different motor speeds and torque there is derived different amounts of horsepower. It is found by the formula:

$$BHP = \frac{\text{Torque x RPM}}{5252.1}$$

One would therefore expect that the higher the RPM, the higher the horsepower. This is true, but the amount of this increase is not constant. It can be seen in Chart I that all of the horsepower curves tend to flatten out in the higher RPM's. This can be explained by the fact that the torque as seen in Chart II increases up to a certain point along with an increase in RPM and then declines rapidly as the higher RPM's are reached. The RPM figure in the formula is such a higher mathematical figure than the torque figure just because of the way motor speed is expressed, that the decline in torque is not large enough to cause a decline in horsepower. Torque in foot pounds at all RPM's is the best and only reliable expression of motor power. The use of horsepower for this purpose may be misleading.

To further exemplify this point, it can be seen in Chart I that the horsepower curve for the Caterpillar D13000 used in the Caterpillar Tractor RD8 has a maximum of 140



stripped
with airmaze + generator
+air compressor
+fan
AFFECT OF ACCESSORIES ON HORSEPOWER
HALL-SCOTT GAS TRUCK MOTOR
#177
Actual performance on individual motor
6 cylinders, bore 5½" stroke 6"
Displacement 855 cu. in.
Compression ratio 5.64:1
Fuel #80 Octane
Horspower
230
220
210
200
190
180
170
160
150
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
400 600 800 1000 1200 1400 1600 1800 2000 2200
Revolutions per Minute

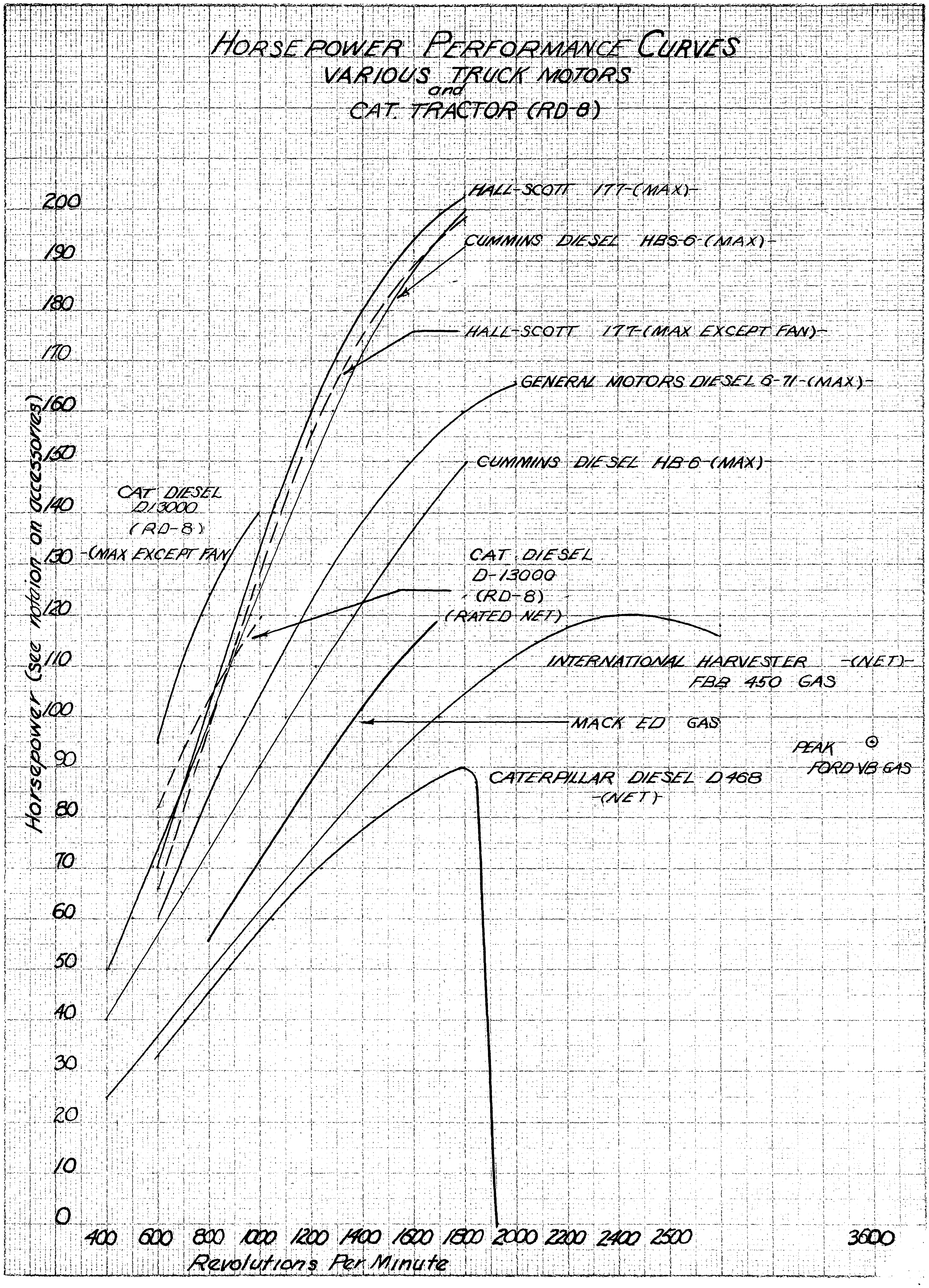
horsepower at its governed speed of 1000 RPM. The Hall-Scott 177 ahs a maximum of 203 horsepower at its governed speed of 1800 RPM. One motor ordinarily in public is given a maximum rating of 140 horsepower and the other, of 203 horsepower. Yet the torque ratings at these points are 730 foot pounds and 570 foot pounds respectively.

Accessories and Their Effect on Ratings:

Beside the fact that the horsepower rating itself is misleading, the way manufacturers have expressed motor ratings varies. In order to display the highest horsepower or torque possible, the performance charts are sometimes based on a motor without accessories or with all accessories except the fan.

Chart II represents a test on an individual motor with various accessories; I copied this chart from the files of the Hall-Scott Company. It illustrates that the horsepower rating varies according to the number of accessories used. The air maze, generator, and air compressor deduct 2 to 3 per cent from the available horsepower, and the fan itself deducts an additional 2 to 3 per cent. This particular motor has a higher rating than the average model 177 motors; this is because of the higher quality of fuel used in the test plus the fact that the published curve represents the average results of a number of motors each of which will vary from this average.

Accessories reduce torque by the same percentage they reduce horsepower. Since I used only published curves for


HORSEPOWER PERFORMANCE CURVES
VARIOUS TRUCK MOTORS
and
CAT. TRACTOR (RD-8)
HALL-SCOTT 177-(MAX)-
CUMMINS DIESEL HBS-6-(MAX)-
HALL-SCOTT 177-(MAX EXCEPT FAN)-
GENERAL MOTORS DIESEL 6-71-(MAX)-
CUMMINS DIESEL HB-6-(MAX)-
CAT DIESEL
D13000
(RD-8)
-(MAX EXCEPT FAN
CAT. DIESEL
D-13000
(RD-8)
(RATED NET)
INTERNATIONAL HARVESTER -(NET)-
FBB 450 GAS
MACK ED GAS
PEAK
FORD V8 GAS
CATERPILLAR DIESEL D468
-(NET)-
Horsepower (See notation on accessories)
200
190
180
170
160
150
140
130
120
110
100
90
80
70
60
50
40
30
20
10
0
400
600
800
1000
1200
1400
1600
1800
2000
2200
2400
2500
3600
Revolutions Per Minute

my comparative performance tables shown under TRUCK PERFORMANCE rather than reduce the operating torque by an arbitrary percentage, the actual performance abilities of some motors may vary 5 per cent. However, this is too small to affect a good comparison between the motors.

Chart I shows a dotted line for the Hall-Scott model 177 motor as well as a heavy line. This dotted line represents the maximum horsepower production at various RPM's with all accessories except the fan. The HBS-6 motor is also on the latter basis so that a direct comparison can be made.

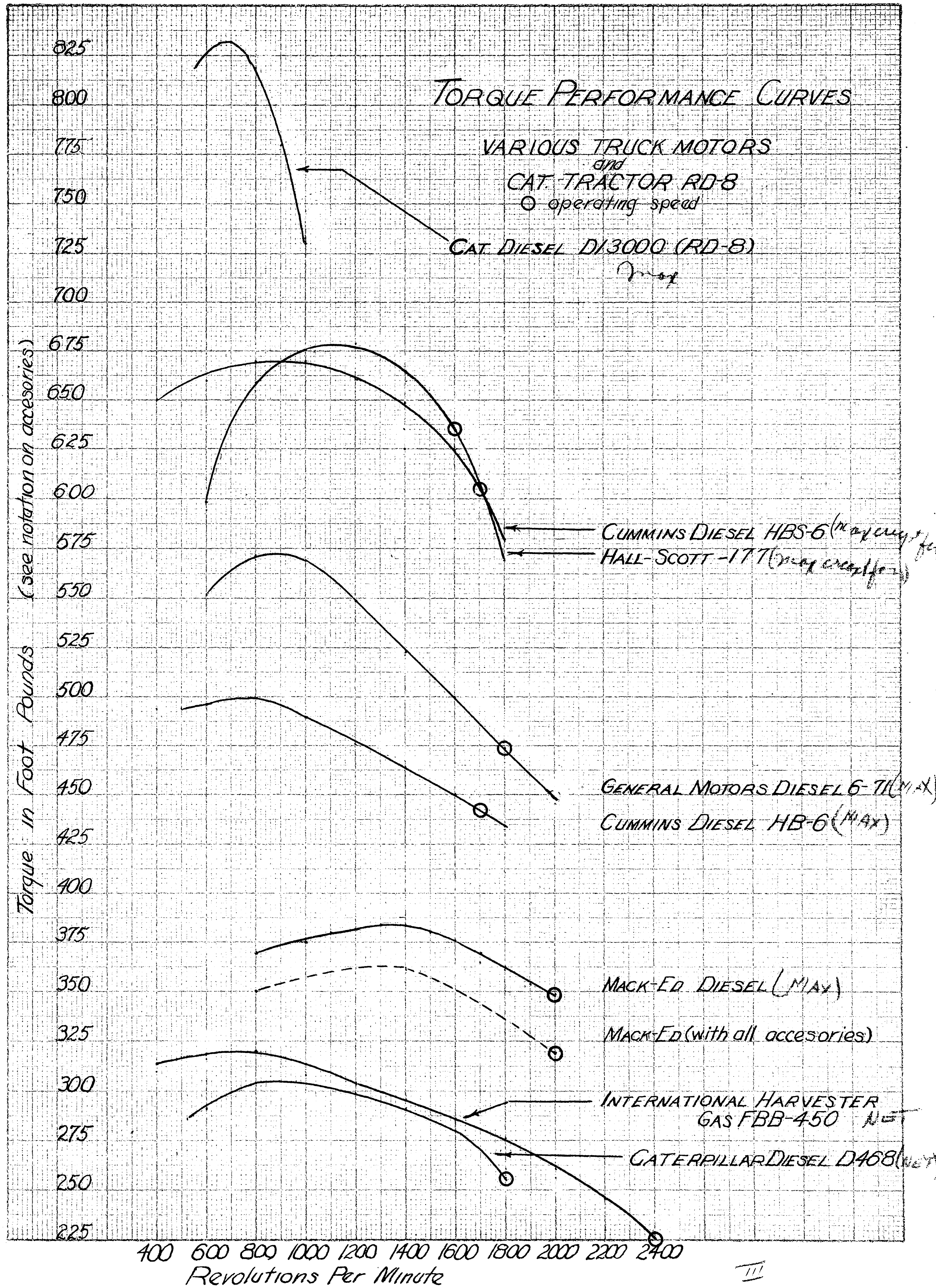
Use of Motor and Effect on Ratings:

Manufacturers publish curves of motor performance which vary according to the service in which they are to be used. All of the curves except the Caterpillar Diesel RD8 heavy line shown in these charts are based on rated output which generally means the load the engine is capable of carrying for a period of twelve hours from each cold start. The difference between rated output shown by the dotted line and maximum output shown by the heavy line is seen in Chart I on the curves for the Caterpillar Diesel D13000 (RD8). The dotted line durve is the one with which we, in using the RD8, have been accustomed. This does not represent the horsepower available on the draw bar, because it must be reduced by the friction loss or mechanical efficiency percentage which is caused by the transfer of power through the clutch, transmission, and differential to the trucks. This reduction is also done for trucks motors as is shown later in order to arrive at available tractive effort at the tires.

Torque:

Torque at operating speeds is the best way of comparing the power of one motor to another. It also serves as the basis for the computation of truck performance. Specifically, torque is the rotating force expressed in foot pounds by the crankshaft as it revolves at a given rate of speed.

Table ____ and Chart III reveal for the first time to my knowledge the relative performance of eight powerful truck motors on the market, now being used extensively in hauling logs. The torque at recommended operating speeds is the final judge of performande, but there are other characteristics which deserve examination. For example, the period or number of revolutioms this operating torque or a higher one is maintained will determine the ability of a motor as explained under motor speeds to continue pulling over a small increase in grade or road resistance. For example, the GMC Model as shown in Chart III is able to pull at an increasingly greater power from its operating speed at 1800 RPM down to 900 RPM while the Hall-Scott can do this over only 500 RPM's. However, as explained before, a better gear shifting system enables the driver to stay within the narrow range of economy RPM's. Nevertheless, constant shifting of gears is wasteful of fuel and time.


TORQUE PERFORMANCE CURVES
VARIOUS TRUCK MOTORS
and
CAT. TRACTOR RD-8
○ operating speed
CAT. DIESEL D13000 (RD-8)
CUMMINS DIESEL HBS-6
HALL-SCOTT -177
GENERAL MOTORS DIESEL 6-71 (MAX)
CUMMINS DIESEL HB-6 (MAX)
MACK-ED DIESEL (MAX)
MACK-ED (with all accesories)
INTERNATIONAL HARVESTER
GAS FBB-450
CATERPILLAR DIESEL D468
Torque in Foot Pounds (see notation on accesories)
825
800
775
750
725
700
675
650
625
600
575
550
525
500
475
450
425
400
375
350
325
300
275
250
225
400 600 800 1000 1200 1400 1600 1800 2000 2200 2400
Revolutions Per Minute

Hence, the amount with which the torque is increased within this operating range will determine the necessity for making shifts for small obstructions.

It is not practical to attempt to describe torque more in detail; Chart III and Table ____ do this more clearly.

Main Bearing Area:

Although all of the motors investigated have the same number of bearings, the bearing surface provided varies a great deal. The GMC Diesel has only 27.6 square inches of bearing surface as compared to 118 square inches for the Caterpillar D468, or to 70.3 square inches for the HBS-6. This may not be a fair comparison between the two-cycle and four-cycle motors. Between the four-cycle motors, the HBS-6 produces 2 1/3 times as much operating torque as the D468 but has only 3/5 the bearing surface. It is natural to expect more bearing trouble with the HBS-6. However, it is not claimed there will be any undue amount of trouble, for probably the Caterpillar has an over-supply of bearing surface. This and other features of the Caterpillar D468 make it a very dependable motor, but it does not produce enough power to assure the least per M cost at the West Fork Logging Company.

Compression Ratio:

The ratio of fuel to air in a Diesel engine is not a function of forming an explosive mixture as it is in a gasoline engine. Hence, in a Diesel, sufficient air can be provided to burn the fuel more completely. A greater com-

compression ratio can be used; consequently, more power is derived from a given amount of fuel, and exceptional economy is obtained at partial loads and speeds.

This is shown in the fact that the gas motors listed in Table XVIIhave only a 5:1 compression ratio while the diesel engines have as much as 17:1 ratio.

Lubricating Oil System:

The large capacity of the GMC oil system is explained by the fact that this is used to help cool the cylinder wall of the motor1 Competitors claim that some of this is burned up as fuel and therefore fuel consumption is greater than that coming from the fuel pump alone.

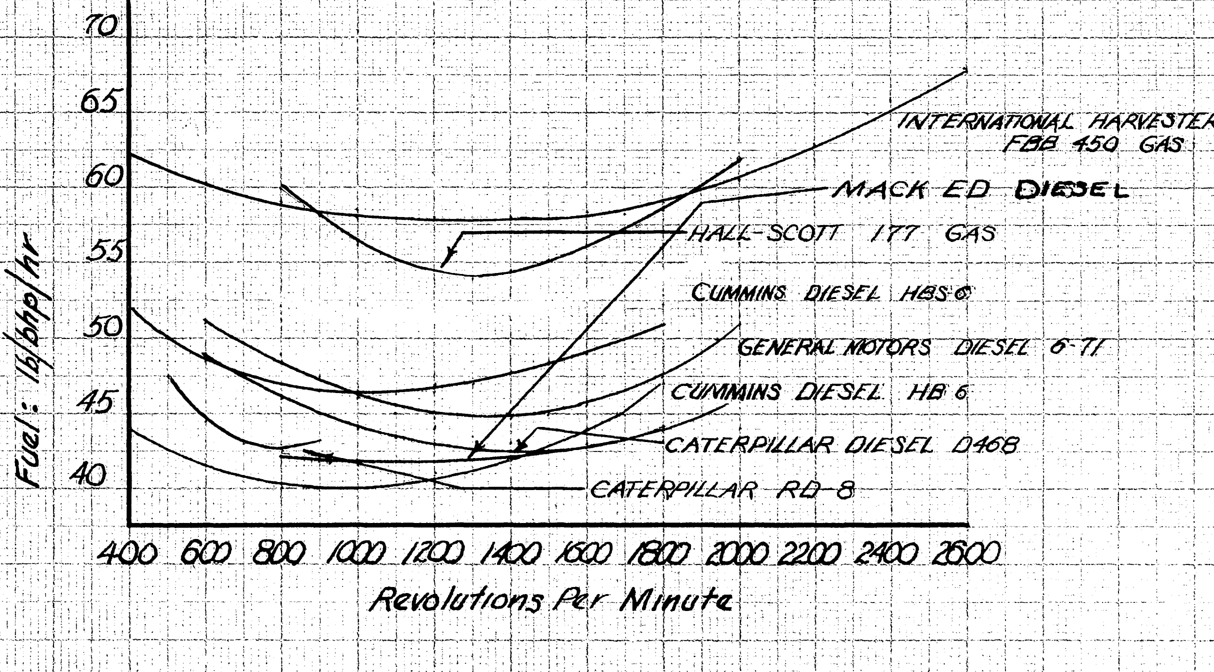
Engine Weights:

Since a few hundred pounds does not make much difference on a private road, this element does not bear much influence. Nevertheless, a fair idea of strength and weight of the metal used for the power produced is shown by dividing the total weight by the horsepower at operating speeds. The D468 pounds of weight per horsepower are nearly twide that of the GMC.

Fuel Consumption:

Relative fuel consumption is shown is Chart IV. This is expressed in terms of pounds of fuel per brake horsepower per hour under full load. To convert to gallons of fuel consumption per hour at certain BHP, the following formula us used for gasoine motors:

FUEL CONSUMPTION CURVES
VARIOUS TRUCK MOTORS
and
CAT. TRACTOR (RD-8)



$$\#\text{gal./hr. at operating speeds} = \frac{\text{lbs. of fuel} \times \text{BHP(at operating RPM)}}{6.25}$$

In the Hall-Scott motor, for example, this would be:

$\frac{.560 \times 195}{6.25} = 17.5$ gallons of gasoline per hour at 1600 RPM.

For diesel motors the formula is:

$$\#\text{ gallons/hour} = \frac{\text{Pounds of fuel} \times \text{BPH}}{7.425}$$

In the Caterpillar D13000 (RD8) this would be, under full load: $\frac{.438 \times 108}{7.425} = 6.4$ gallons per hour at 850 RPM. This would amount to 49.2 gallons in 8 hours under full load. Since we are using about 30 gallons per day, this indicates that our RD8's are operating at about 60 per cent of full load every day.

Gas versus Diesel:

Diesel motors have won so much favor recently that gas motors have been almost completely forgotten. Yet today there exists on the market a gasoline truck motor more powerful yet lighter in weight than any diesel motor of comparative performance. This is the Hall-Scott Model 177 manufactured in Berkeley, California. I have visited this plant and have found it is well equipped, busy, and turning out a good product. However, I hesitate to recommend this motor for logging work at the West Fork Logging Company for several reasons. First, the difference in weight of 700 pounds between it and the Cummins Diesel HBS-6 (its nearest power competitor) is not important enough on private truck roads to make this a deciding point; second, the fuel consumption is (Table XVII and Chart III) 17.5 gallons per hour under full load as compared to 13.1 gallons of diesel fuel for the GBS-6 under full load. Naturally, neither motor is forced to operate under these strenuous conditions continually, but these figures act as the best and only means of comparison. Hence, at a cost of gasoline at 18¢ per gallon and diesel fuel at 6¢ per gallon there is an operating fuel cost item in favor of the Diesel HBS-6 of 4 to 1. There is in addition to this item the problem of handling greater quantities of fuel plus the serious question of greater fire risk with the gasoline as compared with the diesel. It is true that that the HBS-6 costs $800 more than the Hall-Scott and repairs will possibly be more expensive, but all things considered I regard a diesel motor a better investment than a gasoline motor.

There is also another gasoline motor which probably has wider use in logging trucks than the one mentioned above. This is the International Harvester FBB-450. It has an operating torque exactly equal to the new Caterpillar Diesel motor made for truck use. Here the gas motor fuel cost has a 5 to 1 disadvantage as compared to the diesel.

Break Mean Effective Pressure:

The average amount of pressure per square inch which is applied against the top of the piston is called the break mean effective pressure. This acts as an index on the amount of heat which must be dissipated. This is obtained by the following formula: $\text{BMEP} = \frac{150.8 \times \text{Torque}}{\text{Displacement}}$

As indicated in Table XVII the BMEP of the HBS-6 is considerably higher than its counterpart HB-6. The additianal amount of heat which must be dissipated because of the application of a supercharger to the HB-6 is an indication that repair costs are likely to be higher on the HBS-6. It is true that the blowing action of the supercharger carries away some heat but not enough to compensate completely for the increase in pressures. The cooling system on the Kenworth has been increased 14 quarts to aid in dissipating the additional heat.

TIRES

Tires are a constant source of expense, for they are continually wearing out or blowing out. In five operations using heavy trucks in the Douglas Fir region, the expense amounted to between 9 and 13 per cent of the total operating costs.

These operating costs are affected not only by the conditions of the road, but also by the amount of load carried as compared to the rated capacity of the tires.and the amount of air pressure maintaindd, as compared to the recommended pressures. Maintenance of dorrect inflation pressure in tires of sufficient carrying capacity is necessary to obtain 100 per cent service. Chart IV shows at a glance hew neglect of air pressure and overloading affecgs tire servide. When under-inflation and overload conditions are found together, the resulting percentage of service can be determined by multiplying together the percentages of service found for inflation and load.

Over-inflation is also an erroneous practice of many operators who believe they are increasing the load capacity of tires, lengthening the life of the tires, or eliminating the necessity for frequent air pressure checks. The detrimental effects of over-inflation on a tire are increased stress on the cords of the body which lowers the safety factor, accelerates the natural depreciation and resiliency of the cord body, and renders the tire more susceptible to cuts, stone bruises, impact, or penetration breaks.

There are many types of tires on the market, each of which is adapted to certain work. The best tire I have found for the use of the West Fork Logging Company is the Firestone "Ground Grip Excavator Heavy-Duty Tire." It is a tire made especially for slower speeds with a maximum speed of 25 miles per hour with rated load. This slow speed tire increases capacities by about 6 per cent.

Most tires manufactured with the specifications shown in Table XVIII, have the same rated capacities. Nevertheless, the Firestone tire in this class has a tread design whidh has the largest number of square inches of contact on the road at the same rated load and yet has a sufficient amount of tracitve effort. This type of tread will give a longer life. I notice from pictures in trade journals that the heavy load of 22,140 board feet of pine of the Lakeview Logging Company was carried on Firestone Tires of this type.

TABLE XVIII

SIZES, LOADS, AND DIMENSIONS
GROUND GRIP EXCAVATOR H.D. TIRE

SIZE OF TIRE	NO. OF PLIES	SIZE OF RIM	MAX. LOAD LBS., 25 MPH	LBS. PRESSURE	DIMENSIONS, NEW TIRES, INCHES		
					SECTION DIA.	OVERALL DIA.	LOADED RAD.
9.00-24	10	8"	4600	55	9.70	44.2	20.6
9.75-20	12	*9-10"	4670	60	10.63	41.2	19.4
9.75-20	14	*9-10"	5120	70	10.63	41.2	19.4
9.75-24	12	*9-10"	5270	60	10.63	45.2	21.4
9.75-24	14	*9-10"	5770	70	10.63	45.2	21.4
10.50-24	12	9-10"	5830	60	11.10	46.4	21.
10.50-24	14	9-10"	6390	70	11.10	46.4	21.4
11.25-24	14	9-10"	6450	60	11.60	47.8	22.4
11.25-24	16	9-10"	7060	70	11.60	47.8	22.4
12.00-24	14	11"	7700	60	12.95	48.8	22.7
12.00-24	16	11"	8450	70	12.95	48.8	22.7
13.50-24	16	11"	9830	65	14.30	52.2	23.9
13.50-24	18	11"	10690	75	14.30	52.2	23.9

* 8" rim permissible.

Aside from the tread design, tractive effort is also influenced by the diameter of the tire. The smaller the tire, the more the power available. However, this is not so serious a problem as the carrying capacities and costs of the various size tires. Table XIX shows the comparative performance and prices of various tire sizes. An increase in the tire size from 9.75-20 to 12.00-24 decreases the rim pull or tractive effort by 14 per cent but increases carrying capacities by 65 per cent and prices by 111 per cent. With powerful motors and various gear reductions, larger tire sizes can be used in spite of their effect on power in order to obtain larger carrying capacities. However, as stated, initial tire cost increases about twice as rapidly as capacities so that cost becomes a major consideration.

TABLE XIX

COMPARATIVE PERFORMANCE AND PRICE

OF VARIOUS TIRE SIZES

Firestone Heavy Duty Ground Grip Excavator Tires

TIRE SIZE	LOADED RADIUS INCHES	NO.[1] OF PLIES	LBS. PRES-SURE	MAXIMUM RATED LOAD LBS., 25 MPH		RIM PULL[2]	DEALER PRICES[3]		
				1 TIRE	16 TIRES	HBS-6	TIRE	TUBE	TOTAL
9.75-20	19.4	12	60	4670	74600	17300	75.00	8.00	83.00
9.75-24	21.4	12	60	5270	84300	15700	81.40	8.54	89.94
10.50-24	21.4	12	60	5830	93200	15700	96.50	11.30	107.80
11.25-24	22.4	14	60	6450	103000	15000	123.50	14.00	137.50
12.00-24	22.7	14	60	7700	123000	14800	157.40	16.50	173.90

1. 14 ply tires no longer available under 11.25-24; 16 ply tires not available under 13.50-24.

2. Based on 8.7:1 axle ratio, 6.27:1 transmission ratio, and 1.25:1 auxiliary transmission ratio with motor at operating speed. This is the power available at the tires at operating speed with these ratios and the HBS-6 Cummins Diesel Motor.

3. Prices as of April, 1940, are 15 per cent off list prices to represent franchise dealer prices. If a sufficient number of tires are purchased, the West Fork Logging Company can be given a discount.

Table XX reveals the tire cost of fully equipped units with various sizes of tires. A four axle truck and a four axle trailer are capable of carrying 88,000 pounds as compared to 113,000 pounds for a 6 axle truck and a four axle trailer equipped with the same size tires. These additional tires cost $550 more but increased the total capacity by about 28 per cent and the payload capacity by about 4,000 board feet.

TABLE XX

COMPARATIVE CARRYING CAPACITIES AND TIRE PRICES

Various Size Tires and Trucks

TIRE SIZE	TIRE LOCATION	TOTAL CAPACITY	TOTAL DEALER PRICE
	4 Axle Truck and 2 Axle Trailer		
9.75-24	2 front tires	10,540 pounds	$180.00
11.25-24	12 rear tires	77,400 pounds	1650.00
Total	14 tires	87,950 pounds	$1830.00
	6 Axle Truck and $ Axle Trailer		
9.75-24	2 front tires	10,540 pounds	$180.00
10.50-24	16 rear tires	93,200 pounds	1720.00
Total	18 tires	103,740 pounds	$1900.00
	6 Axle Truck and 4 Axle Trailer		
9.75-24	2 front tires	10,540 pounds	$180.00
11.25-24	16 rear tires	103,000 pounds	2200.00
Total	18 tires	113,540 pounds	$2380.00

PERFORMANCE

Introduction:

Performance of a motor truck is the final test of its suitability. The best way of judging this performance would be to take each model of truck and motor, place on each equal amounts of weight, run them over the road as it will be used, and keep records of performance and costs.

However, this method would be not only inconvenient, but the final results would not show enough difference to alter a decision which could be based on computed performance. Some studies have been made of performance in the field, but they have not taken into account so many things which have nothing to do with the abilities of the truck but do affect its performance, such as the skill of the driver, the age of the truck, the condition of the tires, and the dimensions of the load. Hence, comparisons between abilities from field studies of certain makes and models are not entirely without errors.

The purpose of the discussion to follow is to convert the expressions of power discussed under motors and gear reductions explained under transmissions and rear axles to actual road performance. All the important factors which affect the abilities of the truck will be included. By the method shown, all trucks are given equal consideration; the variations in performance are caused only by the inherent abilities of the truck and motor.

Mechanically speaking, these abilities are displayed in the amount of rim pull or tractive effort available at the tires. In order to convert this tractive effort to actual road performance, the following factors are taken into account; first, gross weight moved; second, the road surface; and third, the grade negotiated. In the computations to follow which allow for these factors, the only variation caused in final results between one motor and another is the torque at operating speeds.

Tractive Effort Available:

Tractive effort is becoming the yard stick of vehicle performance. The old basis of rating on pay load ton capacity was one man's recommendation and another's guess; this ridiculous basis of rating is passing out.

Tractive effort is the pounds pull available at the tires. It corresponds to line pull on donkey engines. It is computed by the following formula:

$$TE = \frac{T \times TGR \times ME \times A \times 12}{LR}$$

TE = Tractive Effort
T = Torque
TGR = Total Gear Reduction
ME = Mechanical Efficiency
A = Altitude Factor
LR = Loaded Radius of Tires (in inches)

Torque used is ordinarily based on torque at operating speeds, but as explained previously, all motors have abilities to pull more at lower or lugging speeds. However, this larger torque can not be used for determining operating performande even though some manufacturers publish this maximum torque of the motor without explaining what it means and without also including the operating torque. This is done, for example, in the specifications of the Mack-Lanova Diesel engine Model ED where 381.5 foot pounds at 1200 RPM is given when its operating torque is actually 348 at 2000 RPM.

The total gear reduction is obtained by multiplying the main transmission ratio by the auxiliary transmission ratio (if used) by the rear axle ratio. These total gear reductions for various rear axle ratios and transmissions are shown in Tables VII, VIII, and IX.

The mechanical efficiency percentage is based on the amount of power lost when the torque of the motor is transferred through the clutch, transmission, and axle differential to the rear wheels. When shifts in the main transmission are used, the percentage of efficiency is set at 85 per cent, and when a shift involving both the main and auxiliary transmission is used, this is set at 75 per cent.

The altitude factor is based on the fact that at higher altitudes, the percentage of oxygen in the air is less than at sea level. Hence, motors are unable to burn the fuel as completely, and efficiency is reduced. Temperature of the air and density of the air also affect the efficiency. All of these factors are extremely variable, but in order to reduce the torque from its rated sea level performance to somewhat nearer actual road performance, I have used R. W. Pratt's "Altitude and Volumetric Efficiency Table"[1] in which 90 per cent of motor efficiency is shown at 3,000 feet elevation, which is the approximate elevation of the pass.

The loaded radius in inches of various size tires is

1. Motor Logging Supplement--"The Timberman," March, 1938.

shown in Table XVIII. This loaded radius is used in order to arrive at the actual radius of the tire while under load.

In order to illustrate this formula, let us take the following example: if in a Kenworth truck a Model 7341 main transmission, a Model 703A auxiliary transmission, and an 8.7:1 rear axle were installed, and the truck were equipped with 11.25x24 heavy-duty slow speed tires and the HBS-6 Cummins Diesel motor, the number of pounds of tractive effort available at operating speed in over drive low would be computed as follows:

$$TE = \frac{T \times TGR \times ME \times A \times 12}{LR}$$

$$= \frac{607 \times 45.7 \times .75 \times .90 \times 12}{22.4}$$

$$= 10,000 \text{ pounds}$$

Other amounts of tractive effort for the various motors investigated are shown in Tables XXI, XXII, and XXIII. The gear reductions used are not available in all makes of trucks in which these motors are installed, but the prime purpose of computations is to demonstrate the tractive effort available with different motors under exactly the same conditions.

Actually all tractive effort amounts to in the above formula and tables is this; the torque is increased by gear combinations but reduced by mechanical efficiency, altitude, and the size of the tires. The tables I have computed on comparative tractive effort available describe the respective abilities of the motors much more clearly than I can in words.

TABLE XXI

TRACTIVE EFFORT AVAILABLE *

AT TIRES WITH VARIOUS TRUCK MOTORS

Model 7341 Transmission
Model 703 A Auxiliary Transmission
Rear Axle Ratio 8.7:1
Tire Size 11.25x24

GEAR USED	HALL-SCOTT GAS 171	CUMMINS DIESEL HBS-6	GMC DIESEL 6-71	CUMMINS DIESEL HB6	MACK-ED DIESEL	INTER-NATIONAL FBB-450 GAS	CATER-PILLAR DIESEL D-468
			POUNDS TRACTIVE EFFORT				
U 1	15500	15000	11700	10900	8600	6500	6500
D 1	14000	13600	10600	9900	7800	5900	5900
O 1	10300	10000	7800	7300	5800	4300	4300
U 2	8500	8200	6400	6000	4600	3500	3500
D 2	7600	7400	5800	5400	4200	3200	3200
O 2	5700	5500	4300	4000	3200	2400	2400
U 3	4300	4000	3200	3000	2400	1800	1800
D 3	3900	3800	2900	2700	2100	1600	1600
O 3	2800	2800	2200	2000	1600	1200	1200
U 4	2500	2400	1900	1700	1400	1000	1000
D 4	2200	2200	1700	1600	1200	900	900
O 4	1700	1600	1300	1200	900	700	700
* operating torque foot pounds	640	607	472	442	348	262	262

mechanical efficiency--85% in direct gear, 75% in overdrive and underdrive

altitude factor-90% efficiency
loaded radius--22.4 inches

TABLE XXII

TRACTIVE EFFORT AVAILABLE

AT TIRES WITH VARIOUS TRUCK MOTORS

Model 7341 Transmission
Model 703 A Auxiliary Transmission
Rear Axle Ratio 6.1:1
Tire Size 11.25x24

GEAR USED	HALL-SCOTT GAS 171	CUMMINS DIESEL HBS-6	GMC DIESEL 6-71	CUMMINS DIESEL HB6	MACK-ED DIESEL	INTER-NATIONAL FBB-450 GAS	CATER-PILLAR DIESEL D-468
			POUNDS TRACTIVE EFFORT				
U 1	10600	10300	8000	7600	5900	4500	4500
D 1	9600	9300	7300	6800	5400	4000	4000
O 1	7200	7000	5400	5100	4000	3000	3000
U 2	5800	5700	4400	4100	3200	2400	2400
D 2	4700	5100	4000	3700	2900	2200	2200
O 2	3900	3800	3000	2800	2200	1600	1600
U 3	2900	2900	2200	2100	1600	1200	1200
D 3	2700	2600	2000	1900	1500	1100	1100
O 3	2000	1900	1500	1500	1100	800	800
U 4	1700	1600	1300	1200	900	700	700
D 4	1500	1500	1200	1100	800	600	600
O 4	1100	1100	900	800	700	500	500

TABLE XXIII

TRACTIVE EFFORT AVAILABLE

AT TIRES WITH VARIOUS TRUCK MOTORS

Model 7341 Transmission
Model 703 A Auxiliary Transmission
Rear Axle Ratio 10.25:1
Tire Size 11.25x24

GEAR USED	HALL-SCOTT GAS 171	CUMMINS DIESEL HBS-6	GMC DIESEL 6-71	CUMMINS DIESEL HB6	MACK-ED DIESEL	INTER-NATIONAL FBB-450 GAS	CATER-PILLAR DIESEL D468
	POUNDS TRACTIVE EFFORT						
U 1	18200	17200	13700	12900	10100	7600	7600
D 1	16800	16000	12400	11700	9200	6900	6900
O 1	12200	11900	9200	8600	6800	5100	5100
U 2	10000	9700	7500	7000	5500	4200	4200
D 2	9200	8900	6900	6500	5000	3800	3800
O 2	6700	6500	5000	4700	3700	2800	2800
U 3	5000	4900	3600	3500	2800	2100	2100
D 3	4500	4400	3400	3800	2500	1900	1900
O 3	3500	3300	2500	2400	1900	1400	1400
U 4	4800	2800	2200	2000	1600	1200	1200
D 4	2600	2500	2000	1900	1500	1100	1100
O 4	1900	1900	1500	1400	1100	800	800

Resistance to Tractive Effort:

Tractive effort is resisted by the weight of the truck and load, the rolling resistance of the road, and the grade resistance or gravity. All of these resistance factors are expressed in terms of per 1000 pounds of gross weight which can be moved on a certain grade over a certain road surface. The formula is as follows:

$$\text{Gross Weight Movable} = \frac{\text{Tractive Effort}}{\text{Grade Factor plus Rolling Resistance Factor}}$$

$$\text{GWM} = \frac{\text{TE}}{\text{GF plus RRF}} = \frac{\text{T x TGR x ME x A x 12}}{\text{GF plus RRF}}$$

The grade factor is determined from a simple calculation. For all grades up to a 50 per cent grade it has been found that each 1 per cent of adverse grade offers a resistance to tractive effort of 10 pounds for each 1000 pounds of gross weight moved. Hence for a 5 per cent grade, the grade factor is 5 x 10 or 50.

Rolling resistance is that resistance expressed in pounds per 1000 pounds of gross weight offered by the road surface. This varies from 7.5 pounds for asphalt roads to 50 pounds for sand or clay roads. The best logging roads have a resistance of about 20 pounds, and the poorest, about 40. This resistance changes from time to time on the same surface, depending upon weather conditions. It increases when it is wet or frozen. For the purposes of comparative performance, I have used 30 pounds per 1000 pounds of gross weight. The effect of road surface on hauling ability will be shown later.

TABLE XXIV

WEIGHT HAULING ABILITY ON 5 PER CENT ADVERSE GRADE

VARIOUS TRUCK MOTORS

Rear Axle Ratio 8.7:1

Model #7341 Main Transmission
Model #703A Auxiliary transmission
Tire Size 11.25x24

GEAR USED	HALL-SCOTT GAS-171		CUMMINS DIESEL HBS-6		GMC DIESEL 6-71		CUMMINS DIESEL HB-6		MACK ED DIESEL		INTERNATIONAL FBB-450 GAS		CAT. DIESEL	
	M.lbs.	MPH	M. lbs.	MPH	M.lbs.	MPH	M.lbs.	MPH	M.lbs.	MPH	M.lbs.	MPH	M.lbs.	MP
U 1	194	3.1	188	3.2	148	3.5	136	3.2	107	3.9	81	3.9	81	3.5
D 1	175	3.9	170	4.0	132	4.4	124	4.0	98	4.9	74	4.9	74	4.4
00 1	129	4.7	125	4.8	98	5.3	91	4.8	73	5.8	54	5.8	54	5.3
U 2	1066	5.7	102	519	90	6.4	75	5.9	58	7.1	44	7.1	44	6.4
DD22	95	7.2	93	7.4	73	8.1	68	7.4	53	9.0	40	9.0	40	8.1
0 2	71	8.5	69	8.8	54	9.6	50	8.8	40	10.7	30	10.7	30	9.6
UU 3	54	11.3	50	11.7	40	12.8	38	11.7	30	14.2	22	14.2	22	12.8
D 3	49	14.1	48	14.6	36	15.9	34	14.6	26	17.7	20	17.7	20	15.9
0 3	35	16.9	35	17.5	28	19.1	25.	17.5	20	21.2	15	21.2	18	19.1
U 4	31	19.6	30	20.2	24	22.0	21	20.2	18	24.4	13	24.4	13	22.0
D 4	28	24.5	28	25.3	21	27.6	20	25.3	15	30.6	11	30.6	11	27.6
m0 4	21	29.2	20	30.2	16	33.0	15	30.2	11	36.6	9	36.6	9	33.0
Motor operating speed (RPM)	1600		1650		1800		1650		2000		2000		1800	
Torque at operating speed (RPM)	626 626[1]-1		607 -1		472-2		442 -2		348 -2		262-3		262 -3	

1. Include all accessories except fan
2. Include all accessories
3. Include no accessories

TABLE XXV

WEIGHT HAULING ABILITY ON 5 PER CENT ADVERSE GRADE

VARIOUS TRUCK MOTORS

Rear Axle Ratio 10.25

GEAR USED	HALL-SCOTT GAS-171		CUMMINS DIESEL 6-71		GMC DIESEL 6-71		CUMMINS DIESEL HB-6		MACK ED DIESEL		CAT. DIESEL D-468	
	M.lbs.	MPH	M. lbs.	MPH	M.lbs.	MPH	M. lbs.	MPH	M.lbs.	MPH	M. lbs.	MPH
U 1	228	2.6	215	2.6	172	3.0	161	2.6	126	3.3	95	3.0
D 1	210	3.3	200	3.4	155	3.7	148	3.4	115	4.2	86	3.7
O 1	152	4.0	149	4.1	115	4.5	108	4.1	85	5.0	64	4.5
U 2	125	4.9	121	5.0	94	5.5	88	5.0	69	6.1	53	5.5
D 2	115	6.0	111	6.2	86	6.7	81	6.2	63	715	41	6.7
O 2	84	7.2	69	7.5	63	8.1	59	7.5	46	9.1	35	8.1
U 3	63	9.6	61	10.1	45	1.8	44	10.1	35	12.0	28	10.8
D 3	56	2.0	55	12.4	43	13.6	40	12.4	31	15.1	24	13.6
O 3	44	14.3	41	14.7	31	16.1	30	14.7	24	17.9	18	16.1
U 4	35	16.7	35	17.2	38	18.7	25	17.2	20	20.9	15	18.7
D 4	33	20.8	31	21.4	25	23.4	24	21.4	19	26.0	14	23.4
O 4	24	24.8	24	25.5	19	27.9	18	25.5	14	31.1	13	27.9
Motor operating speed (RPM)	1600		1650		1800		1650		2000		1800	

If we substitute these values in the formula given above and use the example of 10,000 pounds on tractive effort available, we find that on a five percent adverse grade with a 30 pound rolling resistance factor, we could move the following computed gross weight:

$$GWM = \frac{TE}{GF \text{ plus } RRF} = \frac{10,000}{(5 \times 10 \text{ plus } 30} = 125 \text{ M pounds}$$

With the tractive effort available for various rear axle ratiosshown in Tables XXI, XXII, and XXIII, the gross weights movable up a 5 per cent adverse grade were computed with this formula and are shown in Tables XXIV, ---, and XXV. The performance on the 5 per cent grade is the final judge of the motor's power as far as the West Fork Logging Company is concerned. Any loads which can be moved up this five per cent grade can easily be moved on the level or down hill. Please examine these tables, carefully, especially the one with the 8.7:1 rear axle ratio, for they are the final expression of the comparative abilities on this 5 per cent adverse grade of the motors investigated.

It must be emphasized here that figures in Tables XXIV, --, and XXV are based on the fact that the tire size, total gear reductions, and efficiency of each truck in which these motors were installed were exactly the same; therefore <u>the only variation in weight moving ability is because of the differences in power.</u>

Road Speed:

Although power is the prime consideration, road speed

TOTAL GEAR REDUCTION AND ROAD SPEEDS

TRUCK Kenworth 548

UNIT TRANSMISSION 7351 AXLE 8.7:1 TIRE SIZE 11.25x24

ENGINE Cummins Diesel HBS-6 OPERATING SPEED 1650

GEAR USED		Using 703 Auxiliary 2.62 Under & .75 Over			Using 703A Auxiliary 1.25 Under & .84 Over		
IN AUXILIARY	IN MAIN	Trans Ratio #7351	Total Ratio	MPH at 1650 RPM	Trans Ratio	Total Ratio	MPH at 1650 RPM
UNDERDRIVE LOW			143.0	1.5		68.4	3.2
	DIRECT LOW	6.27	54.6	4.0	6.27	54.6	4.0
OVERDRIVE LOW			41.0	5.4		45.7	4.8
UNDERDRIVE 2nd			78.2	2.8		37.4	5.9
	DIRECT 2nd	3.43	29.8	7.4	3.43	29.8	7.4
OVERDRIVE 2nd			22.4	9.8		25.0	8.8
UNDERDRIVE 3rd			39.4	5.6		18.8	11.7
	DIRECT 3rd	1.73	15.1	14.6	1.73	15.1	14.6
OVERDRIVE 3rd			11.3	19.5		12.6	17.5
UNDERDRIVE 4th			22.8	9.7		10.9	20.2
	DIRECT 4th	1.0	8.7	25.3	1.0	8.7	25.3
OVERDRIVE 4th			6.5	33.8		7.3	30.2

TOTAL GEAR REDUCTION AND ROAD SPEEDS

TRUCK Kenworth 548

UNIT TRANSMISSION 7351 AXLE 6.1:1 TIRE SIZE 11.25x24

ENGINE Cummins Diesel HBS-6 OPERATING SPEED 1650

GEAR USED		Using 703 Auxiliary 2.62 Under & .75 Over			Using 703A Auxiliary 1.25 Under & .84 Over		
IN AUXILIARY	IN MAIN	Trans Ratio #7351	Total* Ratio	MPH at 1650 RPM	Trans Ratio #7351	Total* Ratio	MPH at 1650 RPM
UNDERDRIVE LOW			98.5	2.2		47.0	4.7
	DIRECT LOW	6.27	37.6	5.9	6.27	37.6	5.8
OVERDRIVE LOW			28.1	7.8		31.6	7.0
UNDERDRIVE 2nd			55.5	4.0		25.7	8.5
	DIRECT 2nd	3.43	20.6	10.7	3.43	20.6	10.7
OVERDRIVE 2nd			15.9	13.8		17.3	12.7
UNDERDRIVE 3rd			27.2	8.1		13.0	17.0
	DIRECT 3rd	1.73	10.4	21.2	1.73	10.4	21.2
OVERDRIVE 3rd			7.7	28.6		8.7	25.3
UNDERDRIVE 4th			15.7	14.0		7.5	29.4
	DIRECT 4th	1.0	6.0	36.6	1.0	6.0	36.6
OVERDRIVE 4th			4.5	48.9		5.05	43.5

REMARKS:

*Total Ratio found by multiplying transmission ratio by auxiliary ratio by axle ratio.

TOTAL GEAR REDUCTION AND ROAD SPEEDS

TRUCK Kenworth 548

UNIT TRANSMISSION 7351 AXLE 10.25:1 TIRE SIZE 11.25x24

ENGINE Cummins Diesel HBS-6 OPERATING SPEED 1650

GEAR USED		Using 703 Auxiliary 2.62 Under & .75 Over			Using 703A Auxiliary 1.25 Under & .84 Over		
IN AUXILIARY	IN MAIN	Trans Ratio	Total Ratio	MPH at 1650 RPM	Trans Ratio	Total Ratio	MPH at 1650 RPM
UNDERDRIVE LOW			167.9	1.3		80.4	2.6
	DIRECT LOW	6.27	64.5	3.4	6.27	64.3	3.4
OVERDRIVE LOW			48.1	4.6		53.9	4.1
UNDERDRIVE 2nd			92.8	2.4		44.0	5.0
	DIRECT 2nd	3.43	35.2	6.2	3.43	35.7	6.2
OVERDRIVE 2nd			26.3	8.3		29.5	7.5
UNDERDRIVE 3rd			46.5	4.6		22.2	10.1
	DIRECT 3rd	1.73	17.7	12.4	1.73	17.7	12.4
OVERDRIVE 3rd			13.3	16.5		14.9	14.7
UNDERDRIVE 4th			26.8	8.2		12.8	17.2
	DIRECT 4th	1.0	10.25	21.4		10.25	21.4
OVERDRIVE 4th			7.7	28.5		8.6	25.5

with this power is also important. Road speed can be quite accurately determined in the following formula:

$$\text{Miles per Hour} = \frac{\text{Engine RPM} \times D}{TGR \times 336.134}$$

Engine RPM in these tables was based on the recommended operating speed of each motor. D = 2 times the loaded radius in inches; TGR = total gear reduction; 336.134 = factor.

For example, to find the miles per hour of, let us say, the Autocar installed with the Model #7341 main transmission, #703A auxiliary transmission, and an 8.7:1 rear axle ratio, a Caterpillar Diesel motor #D 468 operating at 1800 RPM and on 11.25x24 tires, the spped in overdrive second shift would be:

$$\text{Miles per Hour} = \frac{1800 \times 44.8}{25.0 \times 336.134}$$

$$= 9.6$$

Actual Truck Performance:

All makes of trucks do not have the same gear reductionssystems available. The actual gear reductions for different makes is given in the discussion under Transmissions and Auxiliary Transmissions. This variation causes a differance in the performance of the truck in addition to the power of the mmtor. Also, the total body weight of the trucks and trailers vary. Hence, to compute actual payload performance of individual trucks, the weight of the equipment must be subtracted from the gross weight movable according to the gear reductions available. To convert this derived weight to gross board feet, I divided by 6 (no. pounds to the Board foot, see page___) and multiplied by .86 to arrive at net or water scale. These results are shown in Table XXVI. The footnotes to this table describe the specifications used.

Now that we have finally arrived at some figures in terms of board feet hauling power, let us reexamine the capacities of the various trucks.

It was stated under capacities of the REAR AXLE AND DRIVING UNIT on page ____, that dual-axle trucks equipped with the "400" or "66,000" Timken series and the Karry-Quad Kolossal trailer have a total axle capacity of 120,182 pounds. This same combination equipped with sixteen 11.25x24 rear tires and two 9.75x24 fronttires, has a tital tire capacity of 113,540 pounds.

If it is assumed that we should remain within the rated axle capacities until we learn from experience what additional weight the axles could withstand under our particular conditions,tthen it would be likely that the average total weight probably would be about 106,000 pounds. The number of times the tire capacity would be exceeded would not justify larger tires because it would not be exceeded by more than six per cent if not loaded beyond the axle capacity. Since the weight of a heavy-duty truck and trailer is about 29,000 pounds this would mean that with an average total load of 106,000 pounds, the average payload would be 87,000 pounds. Translated into board feet net scale, the average load would be 12,500 feet on the 5 per cent adverse grade.

The Kenworth, Sterling, Autocar, and Peterbilt will provide models which will carry this load and perform,

TABLE XXVI

PAYLOAD IN BOARD FEET (NET SCALE)[1] AND ROAD SPEED

VARIOUS TRUCKS AND MOTORS
ON 5 PER CENT ADVERSE GRADE

GEAR USED	HALL-SCOTT GAS 171		KENWORTH 548 [2] CUMMINS DIESEL HBS-6		KENWORTH 548 [2] CUMMINS DIESEL HB-6		CAT. DIESEL D-468		GMC [3] GMC DIESEL 6-71		MACK FK [4] MACK ED DIESEL		INT. DR-426-F [5] INTERNATIONAL GAS FB-450	
	BD.FT	MPH	BD.FT.	MPH	BD.FT.	MPH	BD.FT.	MPH	BD.FT.	MPH.	BD.FT.	MPH	BD.FT	MPH
U 1	23600	3.1	22800	3.2	115300	3.2	7500	3.5	20400	3.0	20400	2.4		
D 1	20900	3.9	20200	4.0	113600	4.0	6500	4.4	9700	5.9	7000	5.4	9500	3.9
O 1	14300	4.7	13800	4.8	8900	4.8	3600	5.3	19400	3.1				
U 2	11000	5.7	10500	5.9	6600	5.9	2200	6.4	10400	5.0	10000	4.1		
D 2	9500	7.2	9200	7.4	5600	7.4	1600	8.1	4200	10.0	2600	9.5	3300	7.8
O 2	6000	8.5	5700	8.8	13000	8.8	** ----	----	9900	5.3				
U 3	3600	11.3	15000	11.7	13000	11.7			4400	8.6	4500	7.2	----	----
D 3	2900	14.1	2700	14.6	----	----			----		----			
O 3	860		860	----					4000	8.9				
U 4	----		----	----							1100	13.0		
D 4											----	----		

1. Net scale = 86 per cent of gross scale; gross scale wt. = 6 pounds per board foot.
2. Weight truck and trailer 29,000 pounds. Axle ratio 8.7:1; transmission #7341; auxiliary transmission #702A; sixteen 11.25x24 tires; two 9.75x24bfront tires.
3. Weight of truck and trailer 29,000 pounds. Axle ratio 7.97:1; for main and auxiliary transmission ratios for GMC (see Table ___); sixteen 11.25 x24 tires; 9.75 x24 front tires.
4. Weight of truck and trailer 22,000 pounds. 3.45:1 jackshaft ratio; 2.53:1 sprocket ratio; Mack TR15 main and TRA12 auxiliary transmissions. Tires: front two 11.25x24; rear four 13.50x24; trailer eight 11.25x24.
5. Weight of trudk and trailer 20,000 pounds. Transmission ratios shown in Table___. Tires: sixteen 11.25x24 rear; two 9.75x24 front.

except for differences pointed out under specifications, in the same way as indicated for the Kenworth in Table XXVI. The GMC Models are equipped with a differant type of gear reduction and will perform as shown in this same table.

This average load of 12,500 board feet can be moved up the 5 per cent grade with the HBS-6 in over drive first at 4.8 miles per hour. The maximum load of about 15,500 (based on axle capacities) can be moved at 4.0 miles per hour. Correspondingly, the GMC, because of the large step in its gear reductions between over drive first and under drive second, would have to haul the average load and the maximum load at 3.1 miles per hour. In addition, as we know, some loads with small logs probably will not exceed 9000 board feet in spite of an average of 12,500. This minimum load on the Kenworth can be moved at 7.4 miles, and on the GMC at 5.2 miles.

It is quite evident here that the additional power of the HBS-6 plus the transmissions with which it is usable, make it the more desirable unit over the GMC 6-71 installed in a GMC truck for work at the West Fork Logging Company.

The Cummins HB-6 motor could haul the average load at 4.0 miles, the maximum load at 3.2 miles, and the minimum load at 4.8 miles.

When hauling this maximum load it would be in under drive low and would have no reserve shigts for extra power if an emergency should arrise. In the case of the HBS-6, the additional power gives it not only faster road speeds, but also a reserve shift which could produce 25 per cent more power if necessary. The Mack Model

The Mack Model FK recommended for this job can be seen from the table as wholly incapable of performing anywhere near as satisfactorily as the HBS-6, not only because of the lack of power, but also because of the lack of suitable shifts to handle the average load.

As can be seen in the Kenworth with the HBS-6 motor, there is a range of 4 shifts between the maximum and the minimum loads which can be used to match the power to the load, and the road conditions. In the GMC there are only 3 poorly spaced shifts for this purpose; in the Mack, only 2 and in the International, only 1.

CONCLUSIONS ON RECOMMENDED SELECTION

Hence, I come to the conclusion of the recommended selection of a logging truck for the West Fork Logging Com Company.

On the basis of performance of the HBS66 motor and transmissions available with its use, I strongly recommend it as the motor to be used for our logging work. The fact that the Kenworth Truck Company is located in Seattle gives it a $500 freight advantage over eastern assemblies, plus the minor differences in clutches, frame, and brakes, plus the added advantage of service facilities when needed, give it an edge over its similar competitors as long as the selling price is equal to assembly units of similar specification.

ESTIMATED PRODUCTION

Now that we have arrived at a definite conclusion as to the most suitable truck and motor, let us examine its probable performance over the entire trip with the average load of 12,500 board feet.

The road grade itself, from one of the landings, if I am not assuming too much, will be as follows: two miles of 5 per cent favorable, 6 miles of 5 per cent adverse, and then 2 miles of 5 per cent favorable. Speeds with the loads on these grades will be 14.6 miles per hour, 4.8, and 14.6 respecitvely. The total running time with the loads would then be 92 minutes. The empty trip would be as follows:

20.2 miles per hour, 25.3 miles per hour, and 20.2 miles per hour respectively, or a total running time of 26 minutes. Making allowance for slowing up for turns and small delays will add possibly ten minutes to the trip time or a round trip running time of 1 hour and 30 minutes. Loading and unloading will probably take about 25 minutes.

In the course of 8 hours it is therefore likely that about 4 round trips could be made, hauling a total of about 50,000 board feet per day per truck. A production of 200,000 board feet per day will therefore require 4 trucks.

Investment on the basis of the list price of $13,281 per truck and $3605 per trailer, will require a total investment of $67,544. Because the motor truck field is so competitive, because other trucks assembled from these same specifications will perform satisfactorily, and because a discount off list price for psychological pruposes is always given, I know that a substantial reduction could be made from the quoted investment price.

A set of published specifications for the Kenworth Model 548 and the Karry-Quad Kolossal trailer is found on the next pages, undoubtedly minor changes will be made.

"PETERBILT FLEX"

The Peterbilt Motor Company of Oakland, California, has been experimenting with a new type of logging truck called the "Peterbilt Flex" which is not ready for the market, but which will be available in a short time. It has a chain drive on each of the 8 rear truck wheels, each of which has an independent axle and vertical action. The frame is placed on a walking arm between the two outside and two inside wheels replacing the dual-wheel type in which both wheels are on the outside of the frame. I have seen this experimental model; after it had been tested in the sugar cane fields of Hawaii, it appears to have a number of highly desirable features plus the fact that the famous Peterman logging trailer will also be sold along with it at a total price of about $10,500. Those who have made tests with it in logging work claim that it is far superior to the standard type of truck. I would advise that if this truck is ready for manufacture and sale before we need ours, that a full investigation be made of its adaptability to our operation.

THE TIMKEN-DETROIT AXLE CO.

General Offices
100-400 Clark Avenue
Detroit, Michigan

March 15, 1940

Mr. L. W. Schatz
International House
Berkeley, California

Dear Sir:

We have for acknowledgment your letter of March 13th requesting information concerning the capacity of Timken axles as used in various makes of trucks.

For your information, we have no published axle ratings and the bulletin which we once published on the subject has been withdrawn. Since that time it has been our practice to make recommendations for adequate axle equipment after we have received the complete specifications for the vehicle from the vehicle builder.

We are attaching hereto a copy of an article which appeared some years ago in the leading automotive trade papers calling attention to the fact that this axle rating bulletin was declared obsolete.

Under the circumstances it is necessary for us to refer you to the vehicle manufacturers for the information you have requested. We regret that we are unable to serve you in this instance.

Very truly yours,
(Signed) C, F. Lundgren
SALES ENGINEER

June 24, 1935

On September 1, 1933, The Timken-Detroit Axle Company issued a bulletin giving complete engineering data for the application of Timken and Wisconsin axles to motor vehicles. This bulletin has been used extensively in the trade as a reference and, although now out of date, has been found still in use occasionally for capacity of Timken and Wisconsin axles.

The Timken-Detroit Axle Company advises the automotive industry that the above bulletin is very definitely obsolete and has no bearing on current axles. In every case the vehicle specifications are submitted by the vehicle manufacturer and axle recommendations are made in writing by Timken and Wisconsin to the vehicle manufacturer.

Purchasers of motor vehicles equipped with Timken or Wisconsin axles whould disregard any reference to the above mentioned bulletin.