L.S.M.R.E.

LARGE SCALE MODEL RAILWAY ENGINEERING

MILWAUKEE
LIGHT ENGINEERING
SOCIETY
Ltd.

DIESEL

Volume I

MLES

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December 15, 1994

Fellow members & modelers:

In the following volumes of L.S.M.R.E. you will find reprints of the articles that Tom Artzberger has written for the GAZETTE starting back in 1991.

These reprints are compiled pretty much in the order in which Tom has submitted them for print, with some changes.

On the following page you will find a reprint from an old *M.L.E.S.* BULLETIN on Tom's background. I think you'll find it interesting as well.

Tom continues to write articles for the GAZETTE, and additional volumes will follow, or updates will be made available as necessary.

I'm sure you'll find these volumes extremely helpful and easy to follow to get you started on some easy and nice looking equipment.

Happy Steaming

Ken Rodig

Ken Rodig, Sec.

PHONE 414-246-8215

From the Secretary's Desk

In the future I promise to get a little more thorough on the minutes of the last meeting's events in the BULLETIN for those who didn't make the meeting. I'm not used to taking notes yet. But your best bet would be to make the meetings. Tom A's talk on hydraulics was very enlightening, and with more talks in the future like his, you wouldn't want to miss any more free knowledge. I did video the last meeting talk by Tom and we're thinking of starting a video library of the special talks coming up too.

Now that the holidays are behind us I can get going on the new "New Membership Information" packet I started a month ago. It will contain the revised club bylaws and other pertinent information on the club. This packet will be available to all present members also in the event they never received one when they joined. Just call or write me to order one.

This month in the "Getting to Know Your Fellow Member" column I'm covering a gentleman that's well known to many of us in the club for a long time. However with all the new members joining recently, and for the organizations that receive the news letters outside the club, it might be nice to have a little background on him with the series he's writing for the GAZETTE and the talks he gave and will be giving.

Getting to Know Your Fellow Member: THOMAS ARTZBERGER

Tom has been a member of the M.L.E.S. almost from the beginning. He missed the very first or charter meeting, but from the second meeting on, he's been instrumental in the organization and formation of the society to date. Tom received his Bachelors degree in Engineering at the Milwaukee School of Engineering in 1971. That was two years after the club's formation in 1969. Today he's the Chief Design Engineer of the M.B.W. Ground Pounders Corp. in Slinger, WI which produces industrial concrete finishing equipment.

In 1970 when the R.R. Div. of the M.L.E.S. was formed this was right down Tom's alley. His interest in trains and Steam Engines goes back to his childhood. Tom's grandfather was even the shop superintendent for the Porter Locomotive Works and after his retirement was commissioned to construct a 2" scale 4-8-4 Northern loco for a private party, which is still running today.

In 1971 Tom started writing the technical articles for the then R.R. Div. STACK TALK paper. At the time Tom was the Division Chief and has held that and a number of other offices in the club's history. Then when the land was acquired for the R.R. Park, Tom did most of the surveying and grading of the mainline with his own small crawler loader and through the years has been a major contributor in park maintenance and construction with his personally owned equipment. He's always been willing to bring it out any time to help with projects and grass cutting.

Tom's first live steam engine was a light two truck shay he built some 20 years ago. Since then he's been setting his machine shop up and has built a number of engines, including a EMD SW-1200 diesel switcher, a Southern Pacific 4-6-0 Ten-wheeler, and a 40 Ton Climax geared logging loco. At this time he's in the process of building two EMD GP-7 diesels, one gas powered, one electric powered--all 1-1/2" scale, along with a 15" gauge diesel locomotive for the Riverside and Great Northern Society, intended for the Milwaukee County Zoo Line R.R. nearing completion for summer 1992 operation. He's also built about a dozen cars to include flat cars, gondolas, hoppers, a caboose, and even a drop bottom gondola, along with helping with the major rebuilding of a 0-4-2, 2-1/2" scale plantation loco into a 2-4-4, 1-1/2" scale logging engine.

Those who know Tom, know that he's always willing to share his knowledge or lend a helping hand, and the club wouldn't be the same without him.

Rule One - SAFETY FIRST !
Rule Two - HAVE FUN !

Ken Rodig

Introduction

Designing a good performing model locomotive is a relatively simple job. During the next several months I will describe the procedure that I use when I build a diesel or steam locomotive. This month we will begin by calculating the train weight and from this determine the weight and horsepower required. In future months design parameters for a diesel and steam locomotive along with construction suggestions will be covered.

So lets get started.

Section 1 The load

The first thing one needs to know is the load that one wants to pull. We will use 1 1/2 scale as an example, but this procedure also applies to the small and larger scales. One may wish to pull 6 cars with 2 people in each car. A 1 1/2 scale car weighs about 100 lbs and we will use a average weight of 125 lbs. for each person. We can then calculate the total train weight as follows:

1 car 100 lbs.
2 people 250 lbs.
(1) total for each car = 350 lbs.

350 lbs / car X 6 cars = 2100 lbs.

We must also add the weight of the locomotive which we will say is 400 lbs. The total train weight is therefore:

locomotive 400 lbs.
train 2100 lbs.
(2) total train weight = 2500 lbs.

The rolling resistance of a train on straight level track is approximately 10 lbs. per ton of train weight. Using our results from equation 2 above we can calculate the rolling resistance as follows:

(3) 2500 lbs / 2000 = 1.25 tons
(4) 1.25 tons x 10 = 12.5 lbs rolling resistance

To the rolling resistance we must add the resistance due to grade and curvature. At high speed (over 30 mph) we must also add air resistance, but since our speed are much slower than this we can ignore this factor. The grade resistance is approximately 20 lbs. per ton for each 1% of grade (1 ft in 100 ft.). In our example we will assume a 2% grade.

(5) 20 lbs per ton X 2% grade = 40 lbs. per ton

Since our train weighs 1.25 tons (equation 3).

(6) 1.25 tons x 40 lbs. per ton = 50 lbs. grade resistance

The curve resistance has been found to be the following:

(7)

35 ft. radius	=	16 lbs. per ton
45 ft. radius	=	12 lbs. per ton
60 ft. radius	=	10 lbs. per ton

We will use a 45 ft radius in our example. To find the total train resistance we will add the values for the rolling resistance (equation 4), the grade resistance (equation 6) and the curve resistance (equation 7).

	rolling resistance	=	12.5 lbs.
	grade resistance	=	50 lbs.
	curve resistance	=	12 lbs.
(8)	total train resistance	=	74.5 lbs.

This means that the locomotive must have at least 74.5 lbs. of tractive effort to pull our train. In order to generate this tractive effort we need two things. First we need enough power on the wheels and second enough weight to prevent the wheels from spinning. This last factor is a function of the friction between the wheel and the rail. This is known as the coefficient of adhesion, which for steel or cast iron is approximately 0.25 under ideal conditions. This value drops off rapidly under wet conditions. The maximum tractive effort that can be generated is equal to the weight on the driving wheels times the coefficient of adhesion. In this discussion we will assume that all the weight are on the drivers.

(9) 400 lbs x .25 = 100 lbs. tractive effort

In equation 8 we calculated that we need 74.5 lbs of tractive effort to pull our train and equation 9 tells us that we can generate 100 lbs.

The next thing we must consider is the speed that the train is to run. This is typically 4 - 6 miles per hour with a maximum speed of 8 mph when dealing with 1 1/2 scale. The speed is a function of the wheel diameter and revolutions per minutes(rpm). In this example we will use a 40 inch diameter wheel which is 5" in diameter in 1 1/2 scale. To calculate the rpm required for a top speed of 8 miles per hour we proceed as follows:

(10) 1 mile per hour = 88 ft per minute

(11) 88 ft per minute = 1056 inches per minute

The circumference of a wheel is found by multiplying the diameter by the value of "pi" (3.1416).

(12) 5 in diameter x 3.1416 = 15.70 inches

(13) 8 miles per hour = 8,448 inches per minute

To find our rpm divide 8448 inches per minute by the circumference of the wheel (equation 12).

(14) 8,448 / 15.70 = 538 rpm

We now have the speed (rpm) and the required tractive effort. The last step is to calculate the axle torque required to generate this tractive effort. To do this we will use the maximum tractive effort calculated in equation 9 which was 100 lbs., and the radius of the wheel (1/2 the diameter). The total axle torque is tractive effort times the wheel radius.

(15) 100 lbs x 2.5 inches = 250 in lbs of torque

Our last step is to calculate the approximate horsepower required to generate this torque at this speed. This will be a rough approximation which we will refine latter. The horsepower is found by multiplying the torque (in in-lbs) by the speed (rpm) and dividing the result by 63025.

(16) hp = 250 in-lbs x 538 rpm / 63025 = 2.13 hp.

To sum up we have determine that we will need 250 in lbs of torque at 538 rpm to meet our performance expectations.

That's it for this month. Next month we will examine the power train for a miniature diesel locomotive.

Section 2 The Diesel Locomotive

Last month we began our discussion with the calculations required to determine the power required to move a train over a railroad. This month we will look at how we can generate this power and get it to the rails, and next month do the calculations to select and size the components.

There is four common ways to build a drive train for a diesel model; mechanical, electrical, hydraulic or some combination of the above. The prime mover is usually a gas engine, diesel engine or batteries in combination with electric motors.

Straight mechanical drives, although once popular, has largely been supplanted by hydraulic or electrical systems because of their flexibility and smoothness. The mechanical configuration usually consists of an engine coupled to a forward and reverse gear box by way of a clutch. The gear box output would drive the trucks via universal joints and drive shafts.(see fig.1) One of the problems with this drive setup is obtaining a wide speed range, although many units have been made over the years with variable speed gear boxes or modulating clutches.

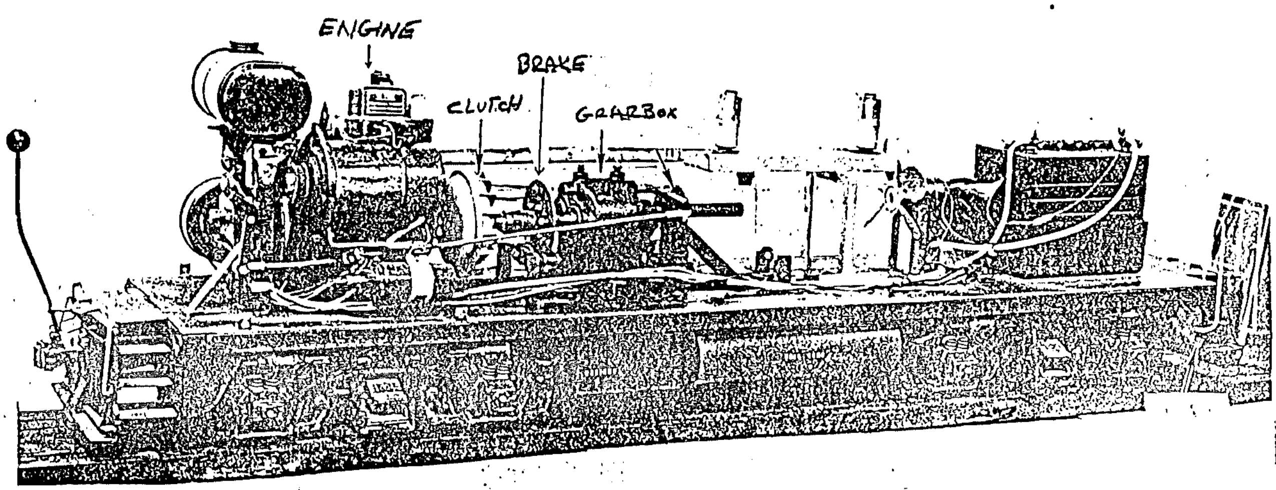


figure 1

The electrical drive system has also been popular, especially in the smaller scales. This configuration generally consists of one or more motors on each truck driving the axles through mechanical means (ie gears, belts or chain). Power is obtained from batteries or in some cases an engine powered generator, batteries being the most common approach.

From an engineering standpoint there are two shortcomings with this drive system. First is the batteries take up a lot of space and give a limited run time before needing recharged. Second is the power density available in small electric motors. Just like in a full size locomotive the tractive effort that the locomotive can generate is limited by the size of the motor that will fit in the available space. On a 1 1/2 scale locomotive this has historically been about 1/2 to 3/4 hp per axle, and on a 1" scale closer to 1/2 hp per truck. The availability of small dc motors and power controllers also present a problem to the model builder. The electrical drive system does have many good points such as smoothness and quietness along with realistic control.

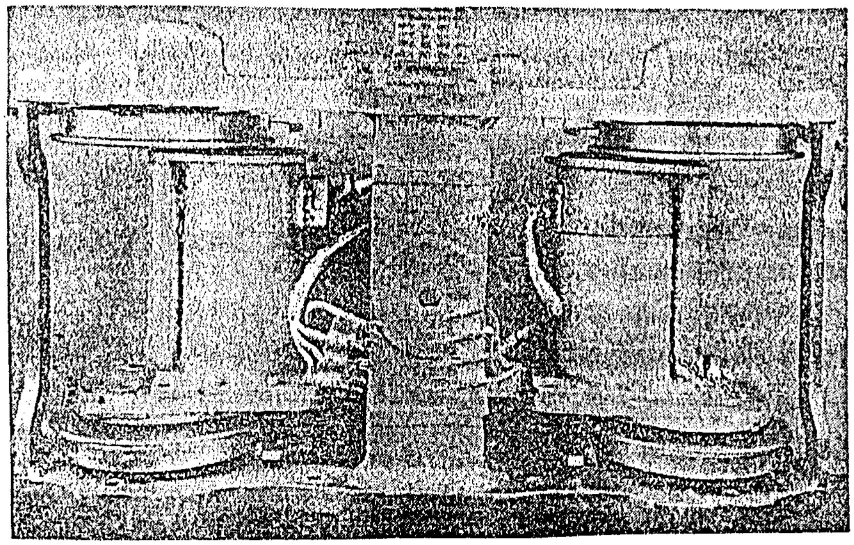


figure 2

Today the most popular way to propel a model diesel is through the use of hydraulics. There are two types of systems in use; the closed loop hydraulic system and the open loop. In either case hydraulics offer a number of advantages to the builder. The power density of hydraulic system is much greater than in electrical systems letting one transmit much more power in a smaller package. also hydraulic motors can run over a wide speed range and can even be stalled under load for long periods of time without damage. Speed control and circuit protection are also easy to accomplished.

The first decision one needs to make in applying a hydraulic system is weather to use a closed loop or open loop system. Figure 3 shows the open loop arrangement and figure 4 the closed loop. In a open loop system a pump draws oil from the tank and feeds the motors by way of some valving. After driving the motor the oil is returned to the tank to be reused. The pump can be of fixed or variable displacement, in the case of a fixed displacement pump speed control and direction must be obtained by way of the valving, while in the case of a variable pump speed control and direction may be may be obtained internally to the pump requiring less external valving.

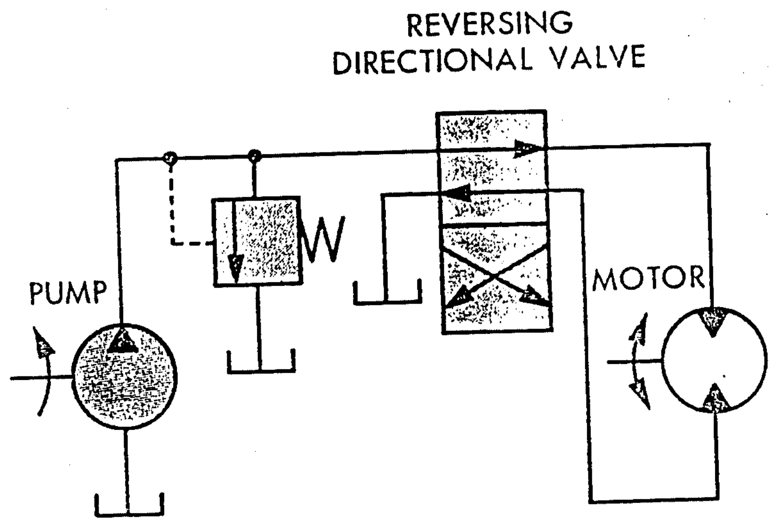


figure 3

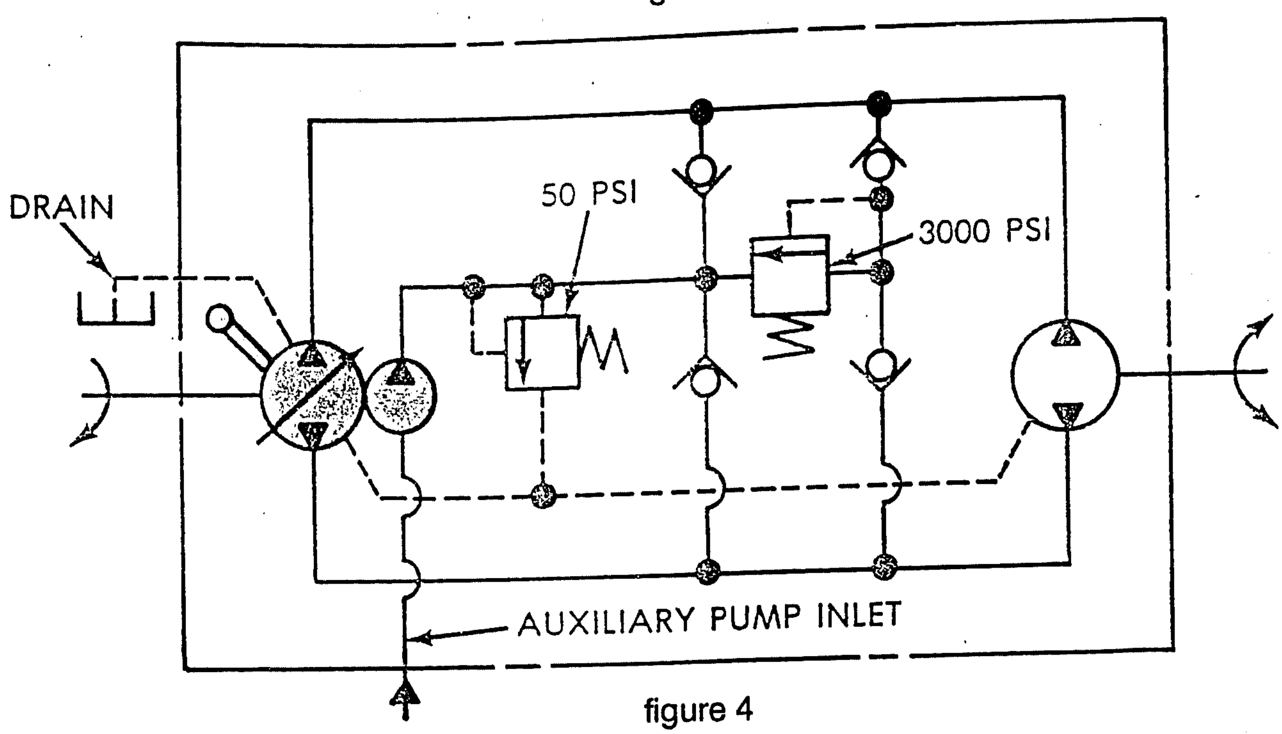


figure 4

In a closed loop system oil flows from the pump to the motors and back to the pump forming a closed loop. To keep the loop filled with oil an auxiliary pump is used, called a charge pump, that draws oil from the tank and keeps the loop filled with oil. Closed loop systems have many advantages from a control and efficiency standpoint and today most hydrostatic transmissions are closed loop.

There are two ways in which hydrostatic transmissions can be applied when building a locomotive. The first is to use a unit that has the pump and motor in a single unit. The Eaton model 6 or 11 is a good example of this.(fig 5) With this type of unit one can use the mechanical drive setup from figure 1, replacing the clutch and gear box with the hydrostatic transmission. This is often seen in locomotives from several suppliers.

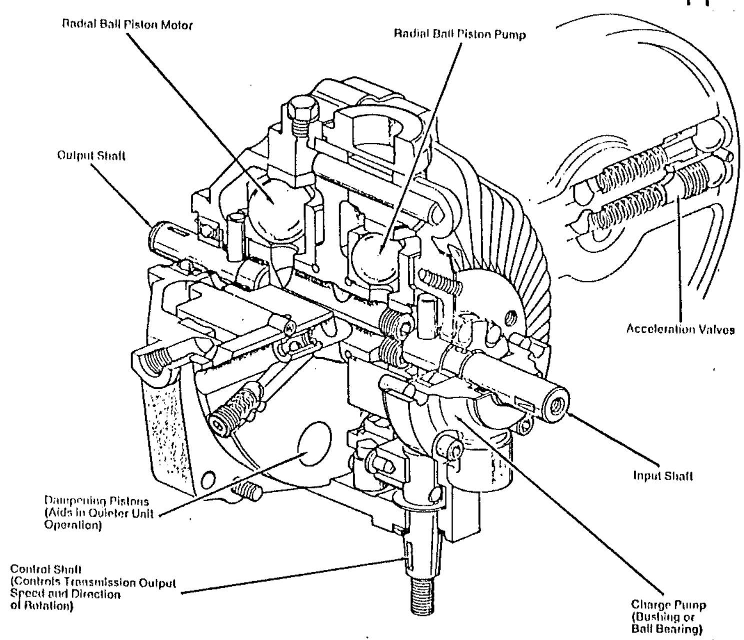
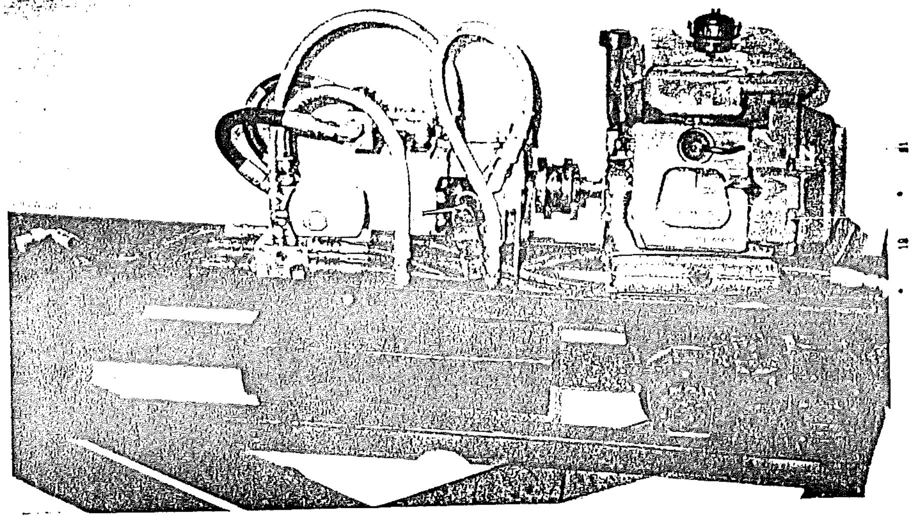
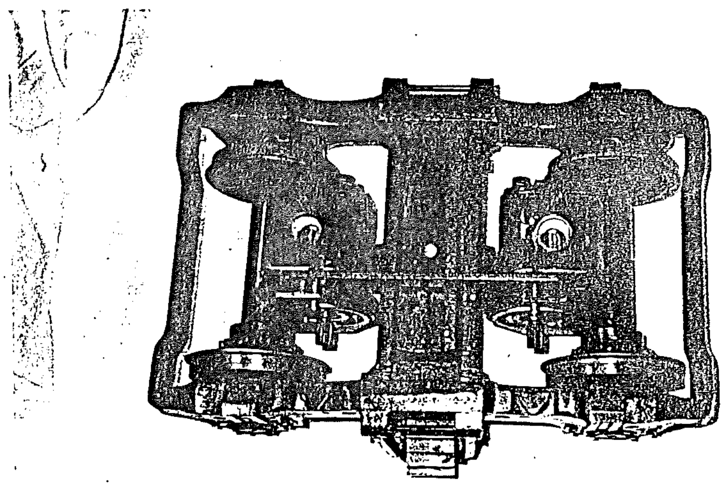


figure 5

The second method is to use an independent pump and motors as shown in figure 6. I have found this arrangement to be the most satisfactory and cost effective in most applications.

Next month we will do the calculations required to select and size the major components for the drive train.





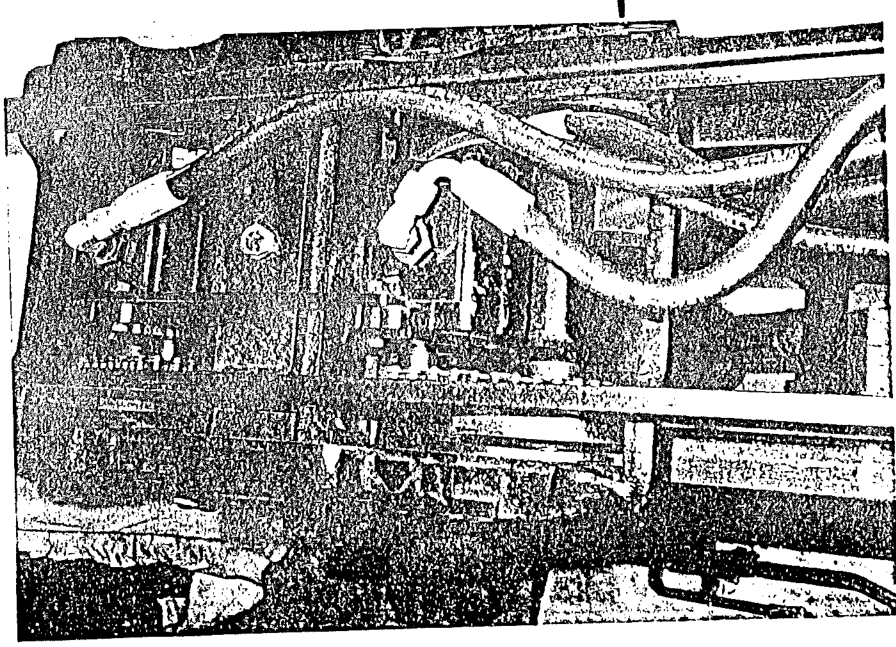


figure 6

Section 3 The Diesel Drive Train

This month we are going to use the information presented in the first two section to design a drive train for a diesel locomotive. Because hydraulic drive systems are probable the most common we will begin our discussion with this method and cover electrical and mechanical methods next month.

Sizing the pump and motors is a trail and error procedure, but we can start with what we know and go from there. We know from section 1 that we would need 250 in lbs of torque at 538 rpm at the wheels to meet our performance expectations. Also we can decide on the speed of the prime mover, in most cases a gas engine. In order to keep the noise level down we should keep this speed as low as possible and still have a smooth running engine. I have often used an engine speed approximately half way between idle and full speed. On an engine with an idle speed of 1200 rpm and a full speed of 3600 rpm our design speed would be 2400 rpm.

Dividing the engine speed by the wheel speed we can determine the reduction that is required between the engine and the wheels.

(1) 2400 rpm / 538 rpm = 4.46 to 1 reduction

This reduction can be obtained by mechanical means, such as chains or gears, or by selecting the displacement of the hydraulic components or some combination of both. The next step that i usually take is to select the pump size because there is usually a much better selection of motors to choose from then there is pumps. In this discussion we will concentrate on variable volume pumps, fixed volume pumps can be used but there is a number of technical problems that has to be overcome to have a successful application. The other design decision that has to be made at this point is weather to use a unitized unit, that is the pump and motor in one housing or a split system with the pump and motor in separate housings interconnected by hoses. In either case the calculations are the same.

For our design we will first look at using a unitized design and then a split design. Two units that come to mind when thinking of single case hydrostatic transmissions is the Eaton model 11 or the Sundstrand model 15. Both are used in many types of lawn and garden equipment. Looking at the eaton unit first (fig 1) the displacement of the pump(the quantity of oil displaced in one revolution) is 1.10 cu in /rev and the motor has a displacement of 2.09 cu in /rev. This means that one revolution of the pump input shaft would yield just over 1/2 revolution of the output shaft

(2 to 1 reduction). These displacements are typically given for the unit at a maximum output condition. The output is made variable by changing the displacement of the pump and or motor therefore changing the ratio of the output to input. In equation 1 we determined that we need a reduction of 4.46 to 1 (4.46 revolutions on the input to 1 turn of the output).

(2) 4.46 to 1 reduction required
2.00 to 1 reduction in hydrostatic unit

2.46 to 1 reduction required in mechanical drive.

The above setup is very similar to what RR Supply uses in their diesel units.

Using the Sundstrand unit (fig 2) in place of the Eaton unit requires a larger mechanical reduction since the ratio of displacements of pump to motor is 1 to 1 (pump and motor same displacement).

(3) 4.46 to 1.00 reduction required
1.00 to 1.00 reduction in hydrostatic unit

4.46 to 1.00 reduction required in mechanical drive.

Now that we know the reductions that are required, or next job is to check our torque requirements to be sure that our hydraulics will handle the load. Checking our Eaton unit again we see that our unit has a continuous torque rating of 360 in lbs. and from our calculation above we have an additional reduction of 2.46 to 1.00 This gives us an effective torque of

(4) 360in lbs. X 2.46 to 1.00 = 885.6 in lbs

From section 1 we figured that we need 250 in lbs and our drive train can deliver 885 so we are in good shape. Like wise or Sundstrand unit has plenty of capacity.

Designing a split system is a bit more complicated. We have much more flexibility in selecting the displacements and we will usually have more than one motor involved. Our first decision is how many motors we should employ and how they should be interconnected. One motor on each truck is usually sufficient however a motor on each axle is sometimes used. On one diesel that i have built i used a motor on each axle (section 2 page 5) not because of power reasons but because it would have been difficult to drive the second axle because of the swing bolster was in the way of a chain drive. We will take the more common approach of one motor on each truck. Once this is decided our next choice is whether to connect the motors in a serial or parallel fashion. If the motors are used in series the oil from the first motor flows to the second and then back to the pump. This makes the effective displacement equal to the displacement of a single motor. If the motors are used in parallel the oil splits equally (one hopes) between the two motors. This makes the effective displacement twice the displacement of a single motor. Parallel connection is usually the prefered method although serial is sometimes used in certain applications. When motors are used in parallel a device called a flow divider is usually used to make sure that the oil is always split between the two motors. If not the oil tends to take the path of least resistance causing the motors to run a different speeds. In smaller scale diesels i have found flow dividers not necessary but they are required on larger higher horsepower applications.

For our example we will use a 15 series Sundstrand pump con figured as a split system (fig 3). From the specifications we see that at 2400 rpm or pump puts out approximately 8 gal per minute and if split between two motors, each motor would receive 4 gpm. Knowing that we need 538 rpm on the wheels we can calculate the displacement of a motor that will give us this speed at 4 gpm input.

(5) rpm = (231 X gpm)/displacement
538 rpm = (231 X 4 gpm.)/displacement
displacement = 1.71 cu in.

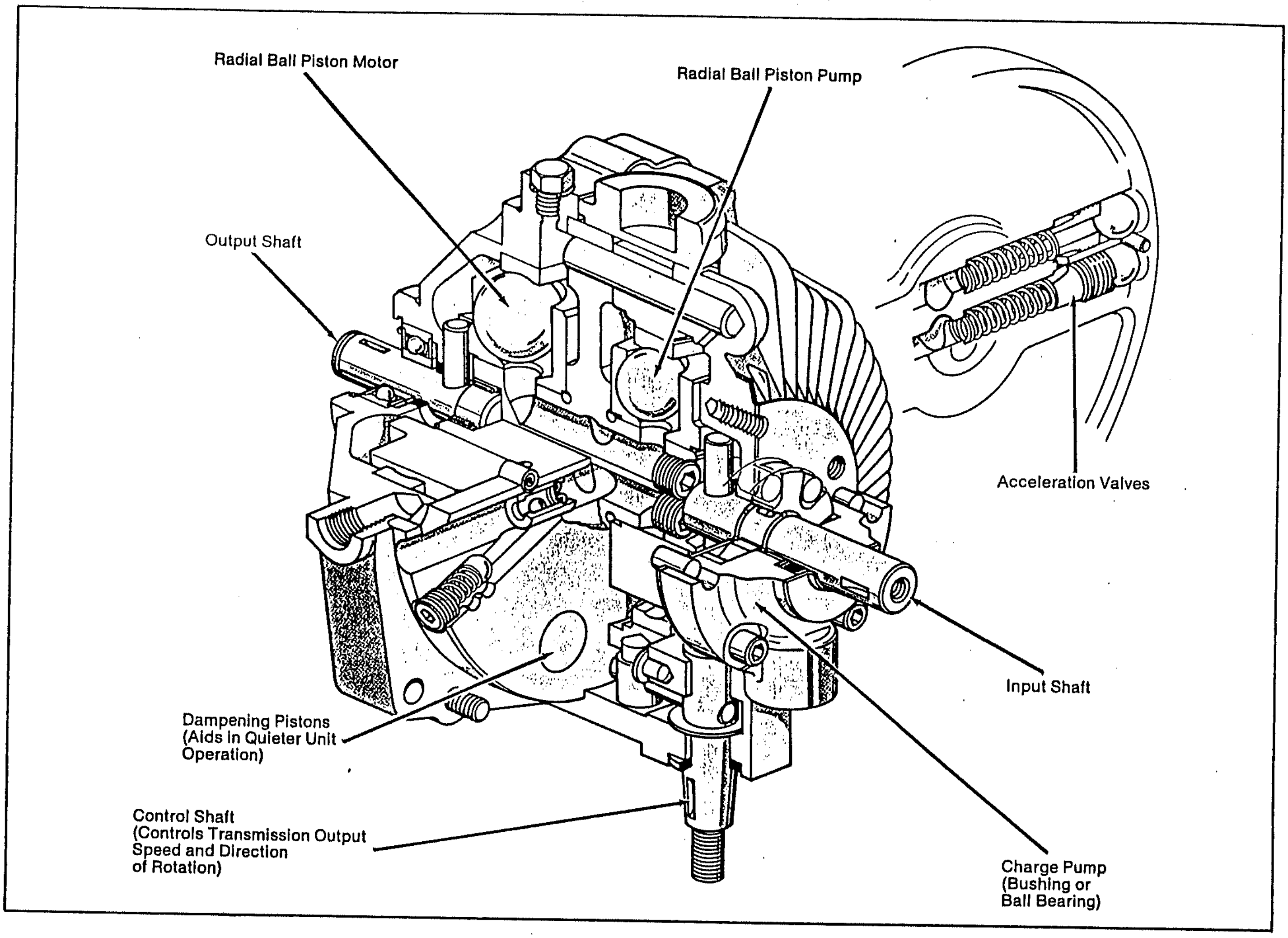
This tells us that we need a 1.71 cu in. per rev motor on each truck with a one to one reduction to the axle or a smaller displacement motor with some mechanical reduction. A small displacement motor runs faster so we have to reduce the speed with some mechanical means. The reduction required for a smaller displacement motor can be found by finding the speed of the motor using equation 5 and then dividing by the desired axle speed. Lets say we wanted to use a .89 cu inch motor.

(6) rpm = (231 X 4 gpm.) / .89 cu in
rpm = 1038

(7) 1038 rpm. / 538 rpm = 1.92 to 1.00 reduction

We could use a 1.71 cu inch per revolution motor with a 1 to 1 reduction or a .89 cu in per evolution motor with a 1.92 to 1 reduction. Both would give the same speed.

Next month we will finish our design and look at mechanical and electrical methods.



Performance Data

Displacement (Theoretical)
- Pump, Variable .. 1.10 cu. in./rev.
- Motor, Fixed ..... 2.09 cu. in./rev.

Speed (Maximum)
- Input .... 3600 RPM
- Output .... 0-1850 RPM

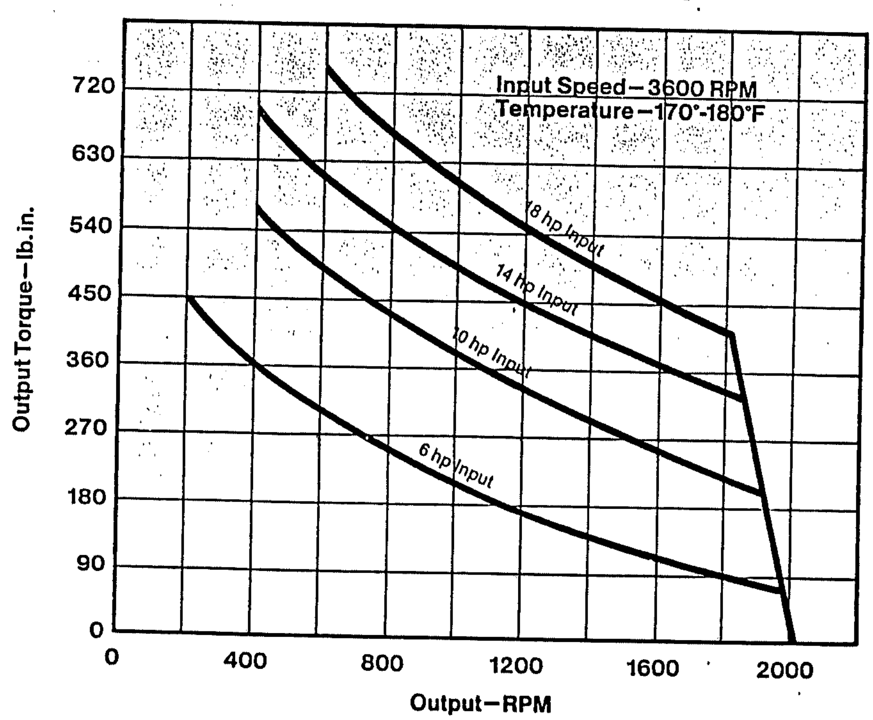
Horsepower, Input
- @ 3600 RPM (Max.) .... 20 HP

Torque, Output (Max.)
- Continuous .... 360 lb. in.
- Intermittent .... 540 lb. in.

Operating Temperature
- (Max. Continuous) .... 180°F

Intermittent Use Only

Output Torque vs. Output Speed



Intermittent Use Only

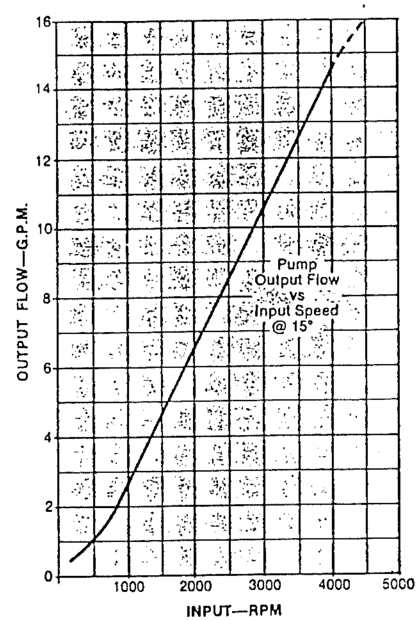
figure 1

15 Series

SPECIFICATIONS

FEATURES

The 15 Series transmission consists of a variable displacement, over center swashplate, axial piston pump and a fixed displacement, reversible axial piston motor. The basic transmission includes the charge pump and check valves; the only elements required to complete the hydraulic power train are a filter, reservoir and, possibily, supplemental cooling. A manual control lever directly connected to the pump swashplate controls the amount and direction of flow to the motor determining the speed and direction of the motor output shaft.



PUMP DISPLACEMENT: 0.913 In. ^{3}/Rev.
PUMP INPUT SPEED: Up to 4000 rpm
MOTOR DISPLACEMENT: .913 In. ^{3}/Rev.
MOTOR OUTPUT SPEED: 0-4000 rpm
MAX. SYSTEM OPERATING PRESS: (with written approval: 4500 psi)
CONT. WORKING PRESS: 1500 psi
NORMAL CHARGE PRESS: 120 psi
IMPLEMENT FLOW & PRESS: 1-3 gpm with relief pressures from 700 to 1000 psi (optional)
PORT SIZES:
Suction: 3/4-16 SAE "O" Ring
Implement Circuit: 9/16-18 SAE "O" Ring
Split System Main Ports: 3/4-16 SAE "O" Ring
FILTRATION: 25 Micron (Nominal)
HEAT EXCHANGER: Depends on application (fan usually adequate)

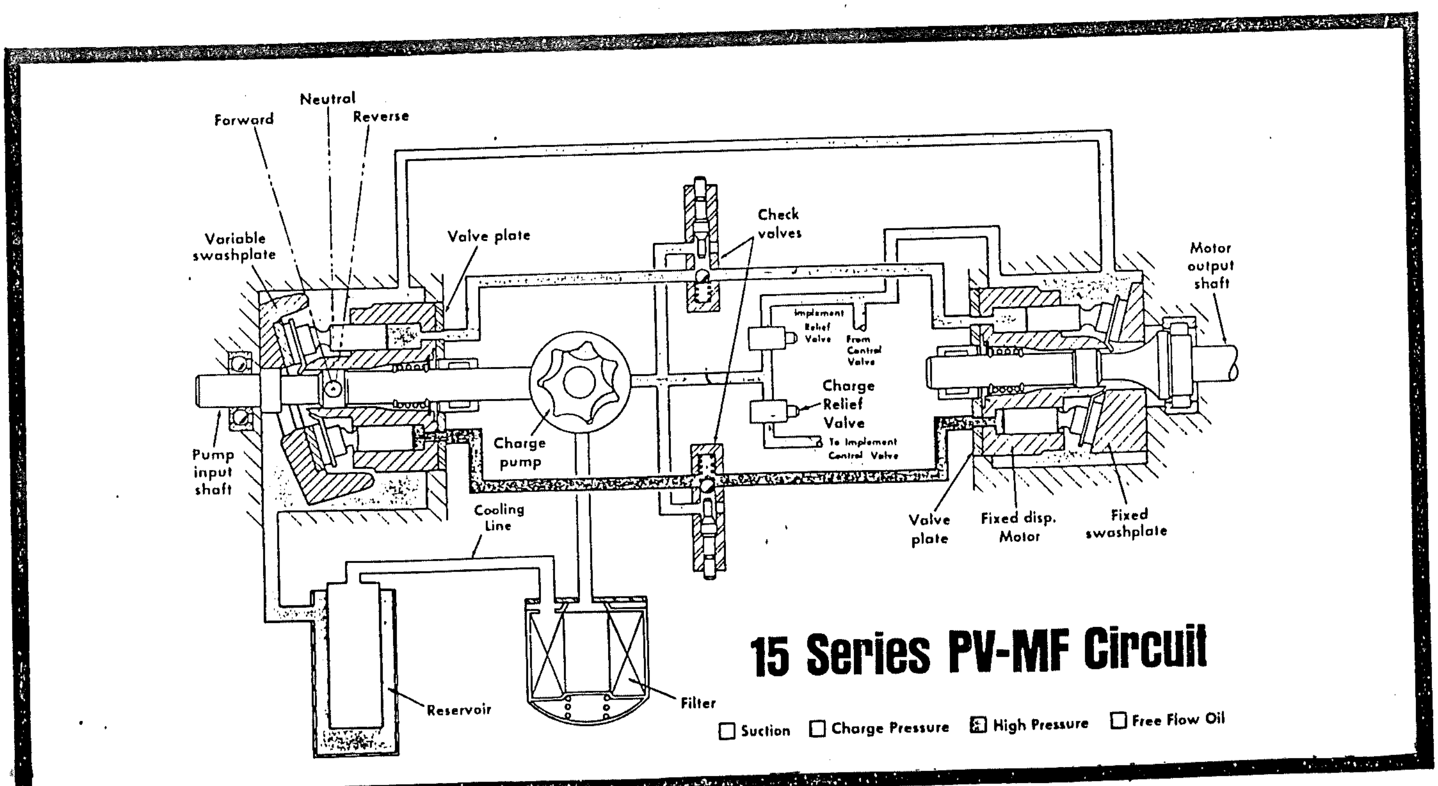
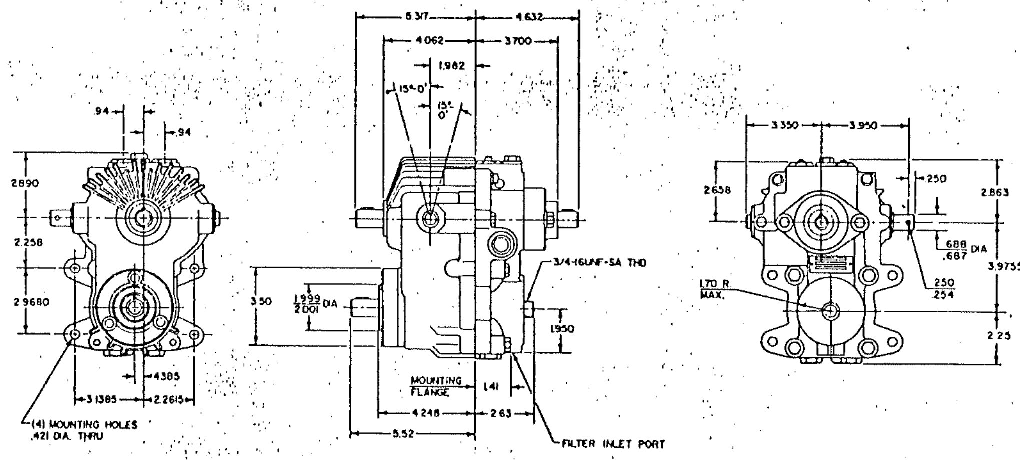


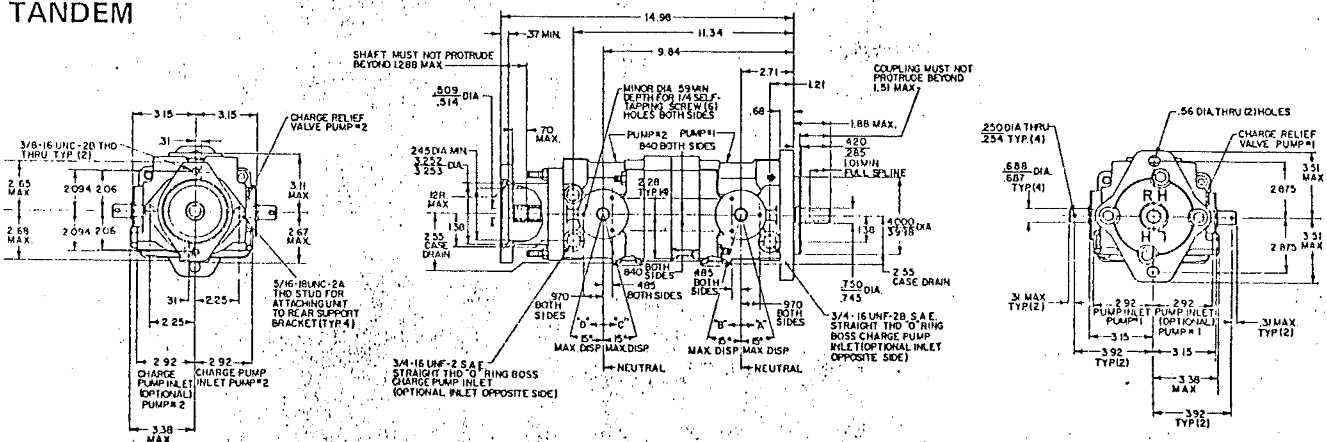
figure 2

INSTALLATION DIMENSIONS

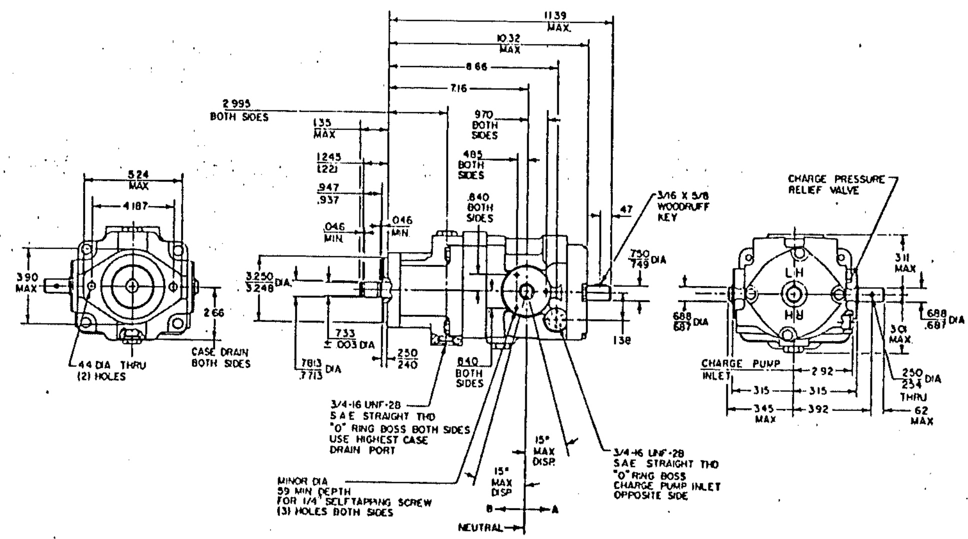
"U"



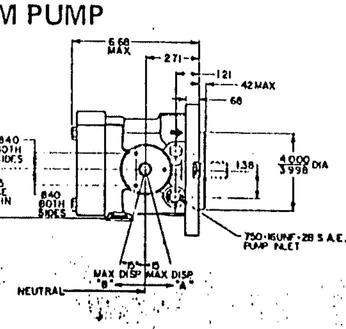
TANDEM



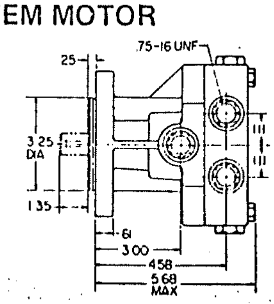
INLINE



SPLIT SYSTEM PUMP



SPLIT SYSTEM MOTOR



–OPTIONAL SHAFTS AVAILABLE–

Section 3 The Diesel Drive Train (continued)

Last month we began our discussion of a drive train for our diesel by selecting a pump and determining the displacement and mechanical reduction required to obtained the speed and torque needed to meet our performance requirements. This month we will look at the various types of motors that are available, select one and the work backwards to find the actual horsepower required from the engine.

There are three types of motors that are useful for our application; gear,gerotor and piston. Gear motors (figure 1) work well for low horsepower application where high speeds are needed. These motors are very similar to gear pumps and in fact a motor can usually be used as a pump and a pump can be used as a pump. The chief difference between a pump and a motor besides some minor internal geometry changes is that pumps are usually internally drained, that is the oil that leaks past the gears and bearings is routed to the intake port of the pump instead of being returned to the oil supply by way of an external line. The Internally drained pump can be used as a motor if only rotation in one direction is required and the pump is connected so that the pressure at the intake port is only a few psi. This is done by using the intake port as the outlet port and thus the outlet port becomes the inlet port. If we pressurize the wrong port we will in most likelihood blow out the shaft seal and could damage the pump internally. Motors are usually externally drained thus allowing them to run in either direction and even to act as a pump if the load starts to overrun the motor.

Gerotor motors or Char-Lynn type motors as they are often called are made by many manufactures and are usually used in applications calling for high torque at low speed (figure 2). They have been found to be a good choice for 15 and 24 in gauge equipment and just recently a new "M" series motor has been introduced that will be ideal for 1 1/2 inch scale equipment. Gerotor motors are bi-directional and in many applications a drain line are not required. Piston motors are the most universal as they can handle low and high speeds over a wide torque range and are very efficient. They can be built to handle very high pressures and put out large amount of power in a very small package. For high horsepower application they are the motor of choice.

Looking at our application, we determined last month that we will need 2 motors with an effective displacement of 1.71 cubic inches per revolution per motor to give us the correct top speed with our 8gpm output from the pump. (equation 5 last month)

Since this is a 1 1/2 gauge locomotive and we are talking about several horsepower we will chose gear motors for our example, although the same procedure holds true for gerotor or piston motors. Early we selected a displacement of .89 cubic inches for our motors, so our first steep is to look at our motor specification an see what is available (figure 1). Checking, we find that a motor is available with a displacement of .884 cubic inches with a maximum pressure of 3000 psi. and a maximum output speed of 4000 rpm. Figure 6 of section 2 shows these motors mounted on one of my diesel trucks. I used a motor on each axle but for this example we will only use one motor per truck with the motor mounted in the center of the truck and a chain drive from the motor to each axle. A reduction of 1.92 to 1 is required between the motor and each axle.

To select the components for the chain drive we can go to a chain manufacturer catalog and check the horsepower capacity of various size chain and sprockets. Going way back to section 1 equation 15 and 16 we calculated that 250 in lbs of torque at 538 rpm would be required. Since we are going to use 2 motors we will assume that each motor will have to supply 1/2 of the total or 125 in lbs at the 538 rpm which is 1.07 horsepower. Since we are using a chain to each axle, each chain will see half of the horsepower or .54 hp. From this information we can look at the horsepower rating tables and select a chain and sprocket size. A number 40 chain is a readily available size so we will look at this table first. Since we know that we are going to need a 1.92 to 1 reduction(small sprocket to large sprocket) we want to pick a relatively small one for the motor so that our large one will not inter fear with the track. I used a 9 tooth sprocket on my units so we will pick the same size. Using a 9 tooth sprocket on the motor we would need a 18 tooth sprocket on the axle

(1) $$9 \text{ teeth} \times 1.92 = 17.28 \text{ teeth} \quad \text{(use 18 teeth)}$$

Checking the catalog, a 18 tooth sprocket is 3.14 inches in diameter which is about right for a 5 inch diameter wheel. Going to our table we find that a 9 tooth sprocket running at 500 rpm is rated at 1.48 horsepower which is indeed more than the .54 horsepower we calculated above, therefore our chains should provide good service.

Now that we have all of our components selected we can do our final horsepower calculations and finish our design. The first step is to determine our maximum system pressure and from this we can find pump pressure which in turn will give us the required engine horsepower. We have calculated that we need 250 in lbs of torque output and since we are going to use two motors in parallel, each will have to supply 1/2 of the total or 125 in lbs. Since we are using a 1.92 to 1 reduction this reduces our actual motor torque to requirement to:

(2) 125 in lbs / 1.92:1 = 65.10 in lbs at the motor shaft

Using the motor curves for a .884 cu inch motor(figure 1) we see that at 65 in lbs will be generated at 475 psi. system pressure. There will be some line losses between the pump and motors. In our case we will approximate this at 10% or approximately 47 psi. This means that our pump must put out approximately 522 psi to get 475 psi at the motors. The next step is to use the pump curves or equations to find the horsepower required to produce 522 psi at the required flow. We will use the following equation to find the input horsepower.

(3) input hp = (flow x pressure) /1714

= (8 gpm X 522 psi) / 1714

= 2.43 hp

To this value we have to add the mechanical efficiency and in the case of a closed loop system we have the charge pump horsepower to add. The mechanical efficiency is about 10% and the charge pump horsepower is 5 to 10% of the main pump horsepower, in this case we will be conservative and use the 10% figure. This makes our total input horsepower about 2.91 horsepower. We can now select an engine that puts out 3 horsepower at 2600 rpm and we will be all set.

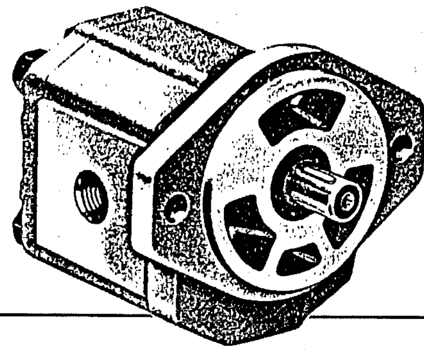
So that it for another month. Next time we will finish our discussion of our hydraulic system and look at the mechanical and electrical alternatives.

GEAR MOTORS

SERIES TKM 200

TKM 200

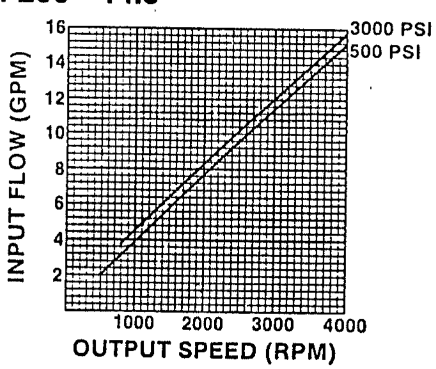
- Six sizes from .488 to 1.56 cu. in./rev. (8.5 to 25.5 cc/rev.)
- SAE 2-Bolt "A" Mounting Flange
- SAE Splined or Straight Keyed Shaft
- Speeds to 4000 RPM
- Bi-Directional Rotation



SPECIFICATIONS

MODEL	DISPLACEMENT		MAX. PRESSURE		MAX. SPEED	MAX. TORQUE OUTPUT	
	Cu. In./Rev.	cc/Rev.	PSI	BAR	RPM	Ft. Lbs.	Kg. M.
8.5	.488	8.5	3000	210	4000	18.1	2.5
11	.671	11	3000	210	4000	23.9	3.3
14.5	.884	14.5	3000	210	4000	31.1	4.3
17	1.03	17	3000	210	4000	36.9	5.1
19.5	1.18	19.5	2600	180	3500	36.1	5.0
26.5	1.56	25.5	2000	140	3500	38.3	5.3

TKM 200 - 14.5



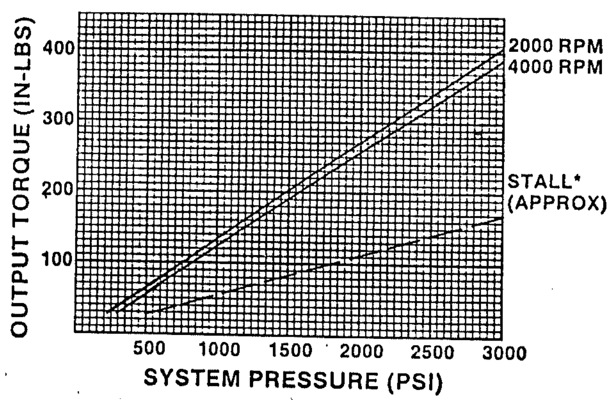


figure 1

Char-Lynn® "M" Series
Low Speed, High Torque Hydraulic Motor

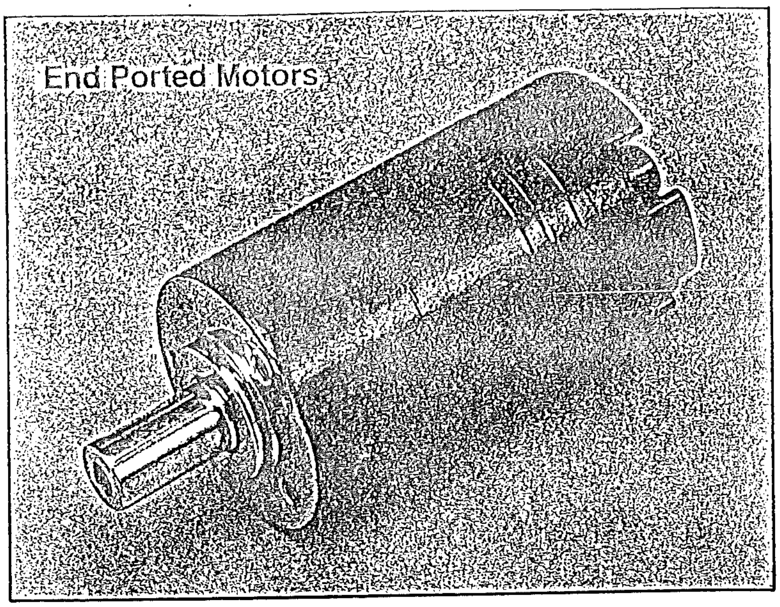
The Char-Lynn "M" Series motor packs a lot of power in a very small package. Less than 2.375 in. [60mm] in diameter and 4 in. [102mm] long, this motor will produce up to 4 HP [3 kw] up to 450 lb. in. [50 Nm] of torque. The "M" Series motor is the smallest in the Eaton line of Char-Lynn low speed, high torque hydraulic motors. It uses the same basic gerotor principle as the other Char-Lynn motors. It has high starting and running torque and operates with equal torque in either direction at speeds up to 2000 RPM. It can accept high radial shaft loads because of the hydrodynamic bearing which absorbs radial forces from sprockets or pulleys.

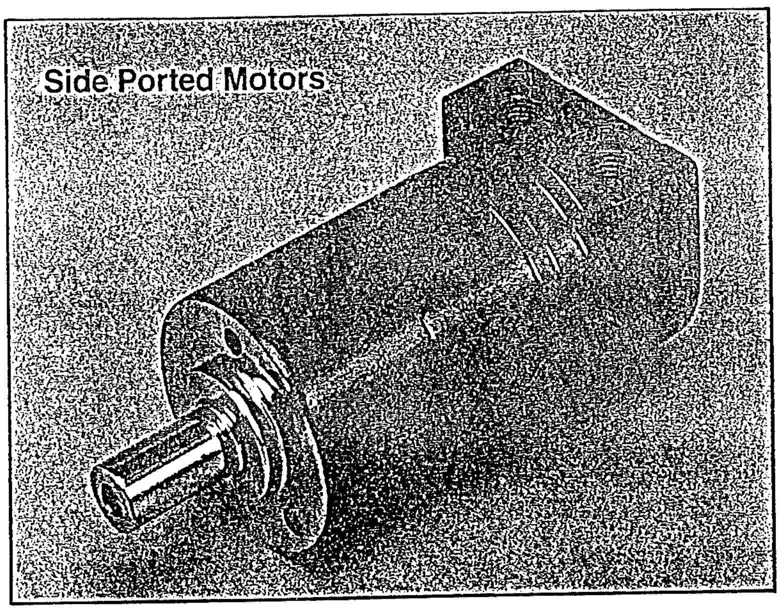
Shaft seals allow a high return pressure, 360 PSI [24,8 Bar], without an external case drain. Check valves in the motor assure that pressure on the shaft seals does not exceed the return pressure when case line is not used.

The "M" Series motor can be mounted directly on the driven device. Speed is controlled easily and smoothly over a wide range. It is ideal for applications such as hand tools where a hexagon socket or a drill chuck can be fitted directly on the output shaft.

Other uses for the Char-Lynn "M" Series motors include:

- Machine tools
- Drilling and tapping machines
- Seed drills
- Augers
- Conveyors
- Textile machinery
- Sweepers
- Fans





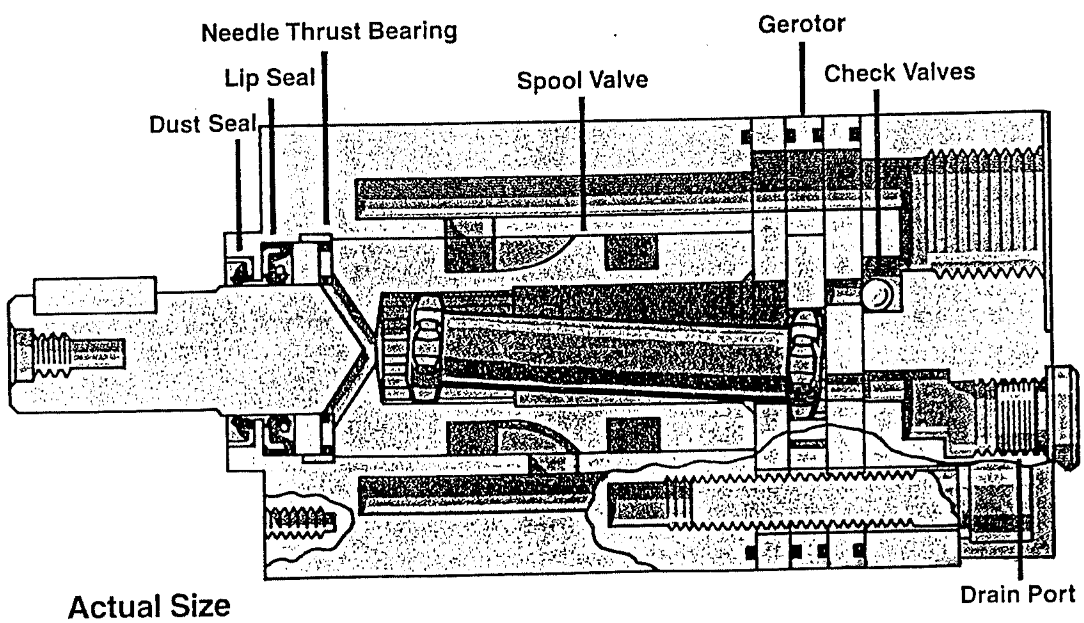


figure 2

Section 3 The Diesel Drive Train (continued)

Last month we finished the selection of components for our hydraulic system and calculated the horsepower required. To finish up I would like to briefly touch on the hydraulic tank and filtering.

The general rule of thumb for sizing a hydraulic tank is that the tank size should be at least big enough so that the oil remains in the tank about a minute before going through the system again. This give time for the oil to cool down and for any air trapped in the oil to dissipate. In the case of the Sundstrand 15 series pump, that we have been talking about, it is the change pump that is circulating the oil and this pump is rated at about 2 gals. per minute at 2600 rpm; Therefore we should use at least a 2 gal. tank. This is sufficient for light duty drives but high horsepower drives usually require a heat exchanges of some kind in the main loop to avoid excessive heat buildup. The hydraulic component vendors usually like to see the oil at about 160 - 180 degree F range for best component life.

Two things are required for proper filtering. First, that the oil must be clean before getting to the pump and second the oil in the system must be filtered to remove any contamination that is generated internally. Again in a low horsepower application filters are usually not used in the main loop but often are employed in higher horsepower applications. Filtering is always needed on the inlet side of the pump however. This is often a fine mesh strainer to catch the large particles (#100 mesh), but yet not restrict the oil flow to the pump which is not good for pump life. A fine filter in the range of 25 micron is then used in the return line back to the tank to filter the oil as it returns.

I am not going to spend a lot of time on mechanical drive trains since they are really the same as we have been talking about. we are just replacing the pumps and motors with gear boxes and clutches etc. One point to remember however is when sizing the components, the maximum horsepower that we have to transmit is the horsepower required to spin the wheels, or the horsepower of the prime mover which ever is less. This will minimize the size of the components needed.

Besides the mechanical power transmission problems discussed previously, the electrical motor driven locomotive has several other parameters to be concerned with. The two most important are the current required by the motors and the temperature that the motors can run at without damage. Figures 1, 2, and 3 show the specifications for a typical 2 1/2 hp DC motor. This particular motor is setup to drive a hydraulic pump but could be adapted to our purposes.

From section 1 we determined that 250 in-lbs of torque @ 538 rpm was required to move our train. Assuming we were going to use 2 motors, one on each truck, this means that each motor would have to supply 1/2 of the 250 in-lbs or 125 in-lbs The easiest way to proceed from here is to pick a convenient reduction for our gearing between the motor and the axles. We can then find the motor speed required to give us 538 rpm on the wheels, then using the motor curves (fig 2) determine the torque that the motor can produce at this speed. For example, lets assume that we are going to have a 4 to 1 reduction.

538 rpm X 4 to 1 Reduction= 2152 rpm motor speed

Using the speed versus torque curve of Fig. 2 we see that at 2150 motor rpm we get approximately 4.2 ft-lbs of torque. Multiplying 4.2 ft-lbs by 12 to convert to in-lbs we get 50.4 in-lbs and multiply by our 4 to 1 reduction we arrive at 201 in-lbs of torque, which is in deed more than the 125 in-lbs needed.

Now that we know our reduction and that the motor will be able to generate enough power we can find the maximum motor current and thermal loads. We have calculated that the maximum torque we can transmit before the wheels slip is 125 in-lbs per truck, and with our 4 to 1 reduction this will be 32.25 in-lbs at the motor. Again, converting in-lbs to ft-lbs by dividing by 12 gives us 2.68 ft-lbs of torque as our maximum motor output. Using the current vs. torque curve of fig. 2 we see that at 2.68 ft-lbs the motor will draw about 66 amps at 24 volts.

Just like in the full size locomotive, the locomotive can only run for a certain period of time at a given current draw before the traction motors will overheat and burn up. Using the thermal performance curve (fig. 3) we can determine the length of time our motor will run at a given output before overheating. Using a torque value of 2.68 ft-lbs we see that we can run for approximately 9 minutes at full power condition with damage. This should be long enough.

The speed of a DC motor (permanent magnet type) is controlled by the voltage applied to the motor and thus this same analysis should be done under starting and intermediate conditions. I am not go in to talk about methods for controlling the motor voltage, at least not now.

The last step is to estimate the size of the batteries required (if this is going to be battery powered), which can be done by estimating the number of minutes that the motor will run at various current draws. For example, the following might apply to a given track for one trip around:

2 minutes	60 amps =	120 amp-minutes
5 minutes	20 amps =	100 amp-minutes
3 minutes	30 amps =	90 amp-minutes
	total	310 amp-minutes

Batteries are rated by the amount of current they can deliver for a period of time (reserve capacity minutes). The Society of Automotive Engineers define this as the amount of time the battery can supply 25 amps at a temperature of 80 degree F. For our example we will use a 12 volt battery with a 100 minute rating. Therefore;

100 minutes X 25 amps = 2500 amp-minutes of power

Placing two such batteries in series will give us our 24 volts for the motors while maintaining our 2500 amp-minute output. Placing the two batteries in parallel would give us twice the amp-minute output but at only 12 volts, which is not enough for our motors. We can then estimate the amount of time our locomotive will run on a battery charge.

2500 amp-minutes/ 310 amp-minutes per trip = 8 trips before the batteries need recharging.

This is it for Diesel Engineering. Nest month Ken has something different and after that I will go into steam locomotive design.

K+E 10 X 10 TO ½ INCH 7 X 10 INCHES KEUFFEL & ESSER CO MADE IN U.S.A. 46 1323

DC835

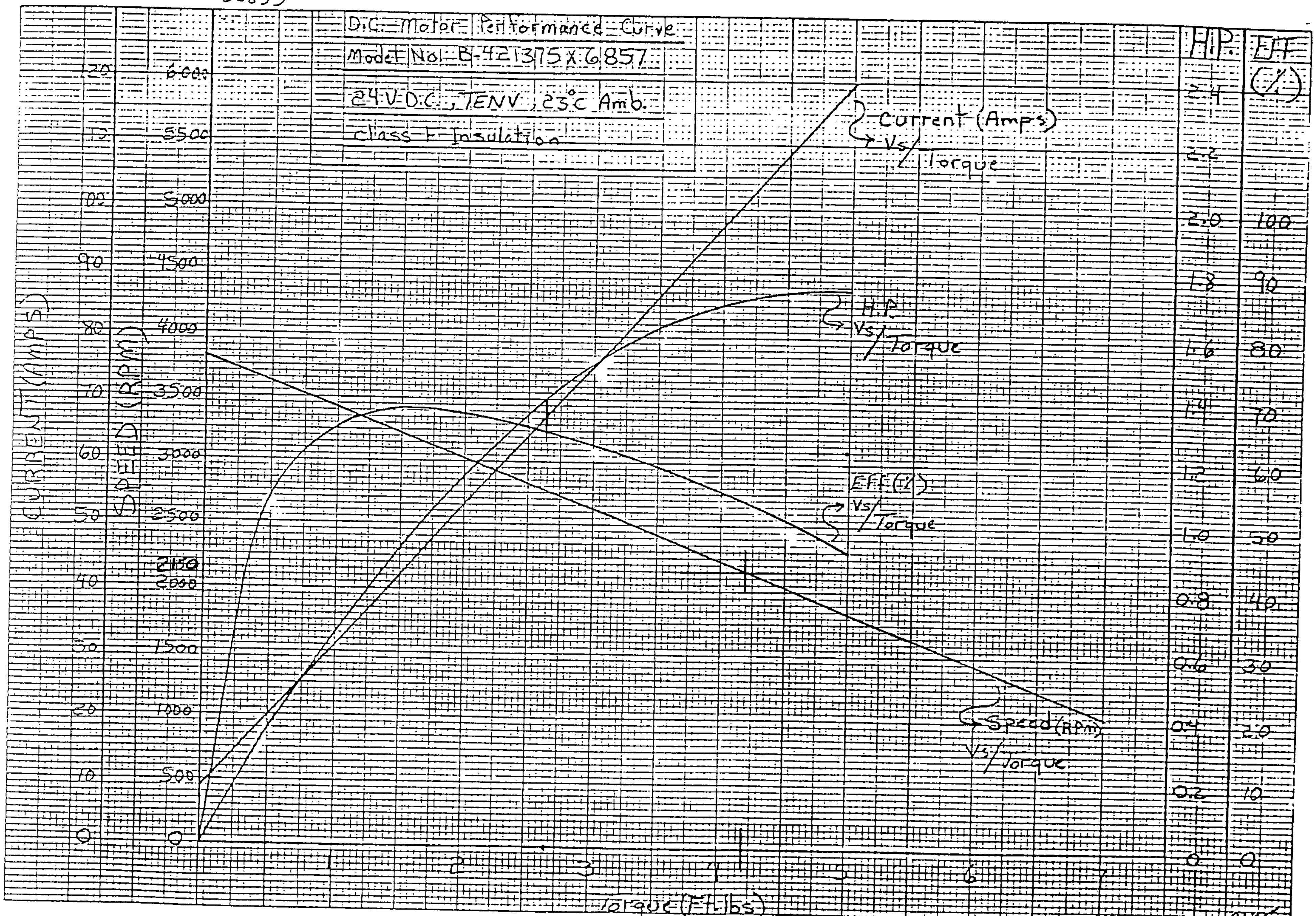


FIG 2

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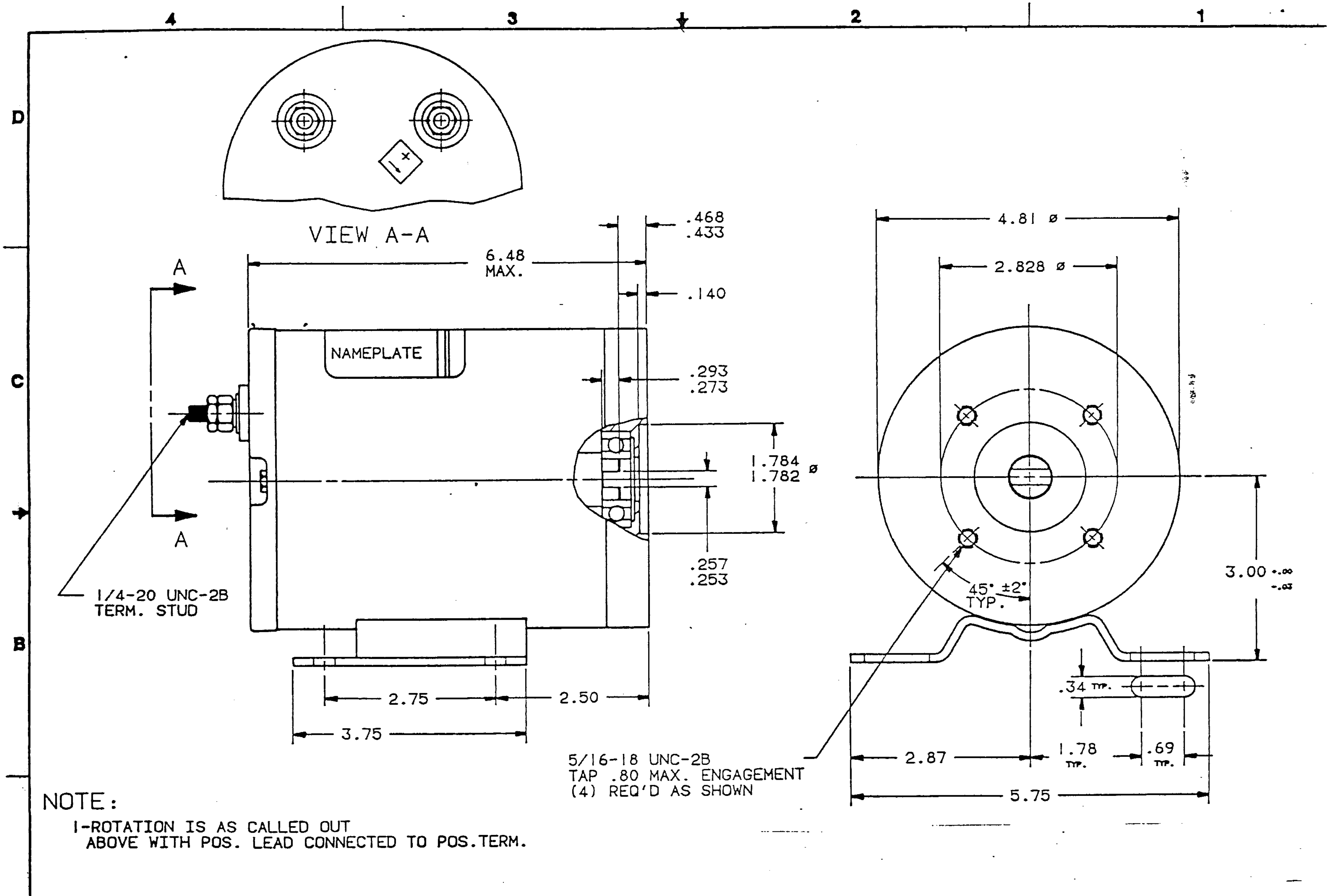
DC835



VIEW A-A
A
A
6.48
MAX.
.468
.433
.140
.293
.273
NAMEPLATE
1.784
1.782 ø
.257
.253
1/4-20 UNC-2B
TERM. STUD
2.75
2.50
3.75
5/16-18 UNC-2B
TAP .80 MAX. ENGAGEMENT
(4) REQ'D AS SHOWN
4.81 ø
2.828 ø
45° ±2°
TYP.
3.00
.34 TYP.
2.87
1.78 TYP.
.69 TYP.
5.75
NOTE:
1-ROTATION IS AS CALLED OUT
ABOVE WITH POS. LEAD CONNECTED TO POS.TERM.

FIG 1

K+E 10 X 10 TO ½ INCH 7 X 10 INCHES KEUFFEL & ESSER CO. MADE IN U.S.A. 46 1323

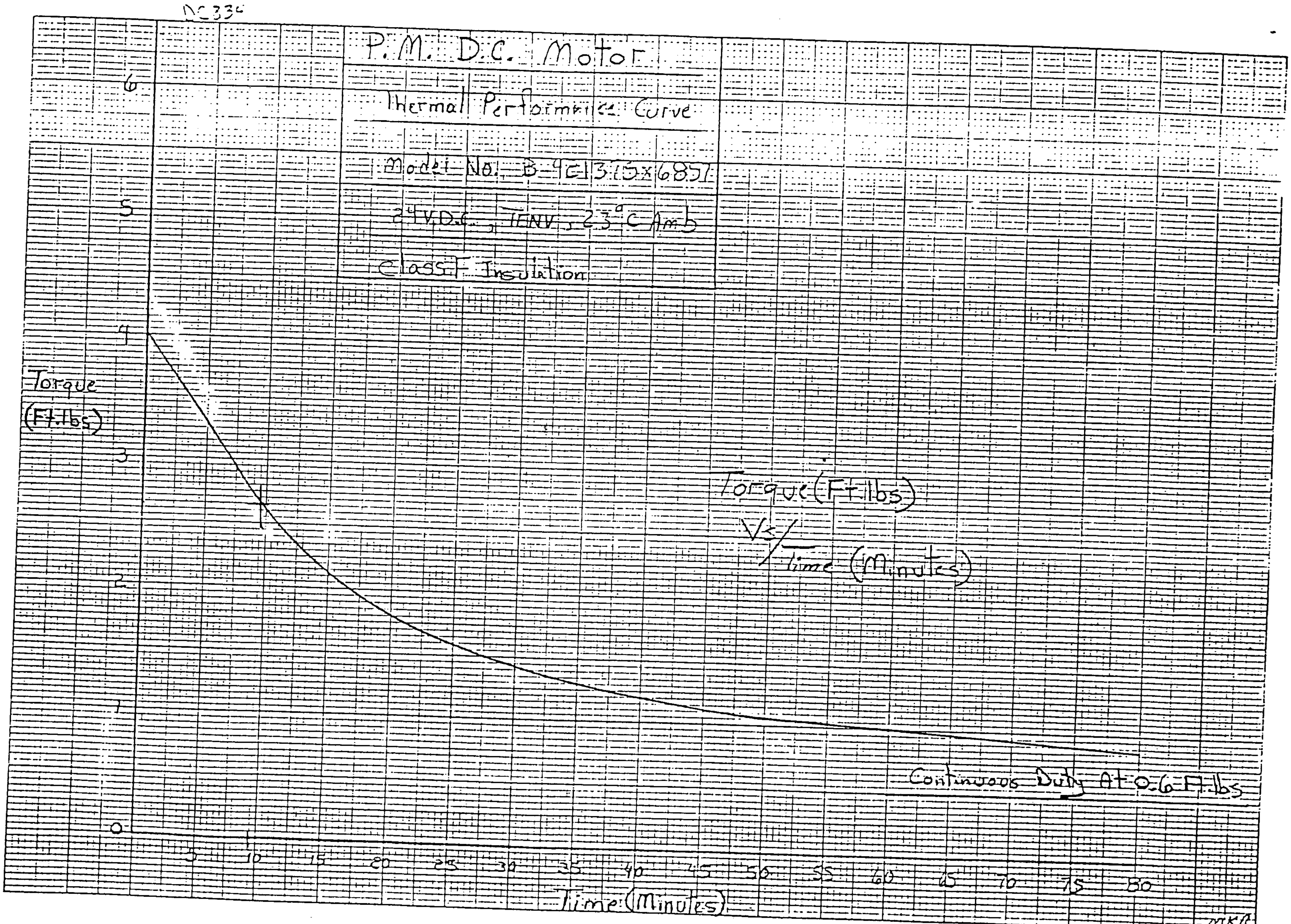


FIG 3

12-10-86

Large Scale Model Railway Engineering

Last month Ken published the specification sheet that covers steam locomotives that we are asking all owners to fill out for our upcoming 25th anniversity publication. This month Ken made one up for electric and diesel electric prototypes. My job is to explain how to fill it out and how to calculate the required information.

The top third of the questionnaire is the same as last month and should present no problem to fill out. The section labeled PRIME MOVER is new and is for the purpose of specifying the type of engine that powers the locomotive. In the case of an electrically driven locomotive just enter batteries for the type and leave the other entries blank.

The TRACTION MOTORS section mainly concerns electrically powered locomotives were motors are applied to each truck or axle. Type refers to the style of motor ie. series wound, permanent magnet etc, and horsepower and speed if they are known.

Under the LOCOMOTIVE section the drive wheel diameter and wheel base are readily found along with the operating weight. Tractive effort and drawbar horsepower is another matter however and does require a little calculation. From an early section when we talked about designing a model diesel locomotive we determined that the maximum tractive effort is a function of the torque available on the axles and the weight on the driving axles. From a weight standpoint we have been using a maximum coefficient of adhesion of .25 so our maximum tractive effort would be 1/4 of the weight on the drivers. This assume that we have enough axle torque to produce this much pull. The maximum axle torque may be difficult to calculate so 1/4 of the weight on the drivers can be used as the maximum tractive effort can be used if the axle torque is not known. The drawbar horsepower can easily be found if you know the tractive effort and the revolutions of the wheels at maximum speed. The best way to find the speed is to measure the distance the locomotive travel in 30 seconds with a load just short of where the wheels start to slip (on dry rail). As we covered several times before the distance the wheel travels in one revolution is equal to the tread diameter multiplied by 3.1416. This gives the distance for one revolution of the wheel in inches. Dividing this value by 12 will give the distance in ft. As an example if we have a 5" diameter wheel one revolution would give us:

5inch X 3.1416 = 15.70 inches per revolution

15.70 / 12 = 1.309 feet per revolution

If the train traveled 150 ft in 30 seconds we can double this and get 300 ft in one minute. If we divide the 300ft by the feet per revolution of the wheels, in this case 1.309 ft per rev we get 229 revolutions per minute. Lastly we need the axle torque in order to find the horsepower and this can be approximated by multiplying the tractive effort by half the wheel diameter(radius). As above we approximated the maximum tractive effort as 1/4 the weight on the drivers so as an example if we have a locomotive with 300 pounds on the drivers our maximum tractive effort would be 300 /4 or 75 pounds. The torque is then 75 pounds times 2.50 inches or 187.50 inch pounds.

Horsepower is found by multiplying the torque by the speed and dividing by 63025.

$$HP = \frac{187.50 \text{ inch pounds} \times 229 \text{ rpm}}{63025}$$

$$HP = .681$$

The last section to be filled out is the MISCELLANEOUS section which has space for control type, drive line type and mu capability. The control type is ment to indicate the type of operator interface ie. manual, electrical, radio etc. The drive line references the type of power train such as electrical, mechinical, hydraulic and the mu capable indicate weather the unit is made to run with other units.

That's it for the diesel (and electric) locomotive specification questionnaire. Next month back to another technical article.

MILWAUKEE LIGHT ENGINEERING SOCIETY

R A I L R O A D   D I V I S I O N

Membership Diesel Locomotive Specification Questionnaire

Name of Railroad: ____________________

Name of Builder: ____________________

Name of Owner: ____________________

Serial No.: __________ Years Built ___/___/___ to ___/___/___

Type Name: __________ Truck type: __________

Road No.: ______ R.R. Class: __________ Builder Class: __________

Prototype Builder & Railroad: ____________________

Scale Built: ______ Gauge of Track: ______ Overall Length: __________

PRIME MOVER

Manufacturer Brand	Type Design	Rated Horse power	Maximum RPM	Fuel Kind
__________	__________	__________	______	____

TRACTION MOTORS

Manufacturer Brand	Type Design	Rated Horse power	Maximum RPM
__________	__________	__________	______

LOCOMOTIVE

DRIVING WHEEL Diameter	WHEEL BASE Truck	WHEEL BASE Engine
______	______	______

AVERAGE WEIGHT IN WORKING ORDER, POUNDS: __________

Tractive Effort __________ Draw Bar Horsepower __________

Miscellaneous

Control Type ____________________

Drive line Type ____________________

MU capable ______, Type ____________________