DETROIT DIESEL



Engineering Bulletin No. 35

SHORT SHAFT POWER TAKE-OFF V-BELT DRIVES FOR DETROIT DIESEL ENGINE APPLICATIONS

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INTRODUCTION

In addition to the familiar vehicular drive arrangement wherein engine power is transmitted to the vehicle through a direct mounted automotive transmission, there are various applications of Detroit Diesel engines that utilize power take-off assemblies (PTO's) mounted directly to the flywheel housing. These usually incorporate a flywheel mounted disconnect clutch and a pulley (sheave) on the output end of the PTO shaft for belt driven equipment. For this type of application, a PTO assembly should be selected that is capable of transmitting full rated engine power and maximum engine torque.

Some applications (chiefly marine installations) may require the use of a front mounted power take-off assembly. The power capacities of front mounted PTO assemblies offered by DDC for use on Detroit Diesel engines are limited by the torque capacity of the PTO clutch. They are not capable of transmitting full engine power.

Front or rear PTO drive arrangements should be carefully analyzed to insure that they will not subject the engine or the driven equipment to detrimental loads which would result in unsatisfactory operation or inadequate life of any component.

SHORT SHAFT POWER TAKE-OFF - V-BELT DRIVE

Transmission of engine power through a power take-off belt system requires careful sheave and belt selection. Proper sheave and belt design is important to insure the life of both the power take-off (PTO) bearings and PTO shaft. Good design includes the proper selection of sheave diameter, sheave location relative to the PTO main bearing, number and type of belts and belt tension. These sheaves and belt factors must be worked out so that recommended limits on shaft side loads are not exceeded. Maximum allowable side loading is different for each PTO assembly and varies with the location of the centerline of the sheave load relative to the PTO main bearing. For a short shaft PTO (no outboard bearing), this load centerline should be as close to the PTO main bearing as possible.

Too small a sheave diameter limits belt life and power transmitting capacity due to the small amount of pulley-to-belt contact. Too large a sheave increases the belt velocity for a given shaft rpm thereby increasing the centrifugal tension in the belt. The mass of an excessively large sheave also magnifies the unbalancing effect of any shaft deflection (and consequently the bearing loads) under dynamic conditions. Belt cross section selected also affects belt centrifugal tension. See Fig. 1.

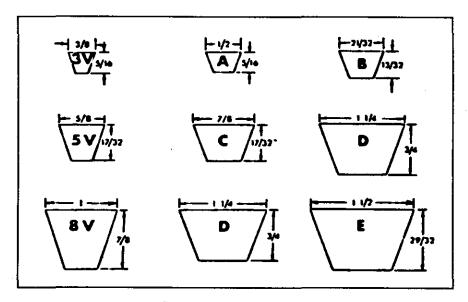


Fig. 1 V BELT SECTION

With the engine running and the clutch disengaged, life of the pilot bearing may become critical. The disconnect clutch of each PTO is rated as to torque capacity. A clutch service factor corresponding to the type of duty expected in the application should be used when considering input torque to insure that shock loading will not cause early clutch failure.

Before an analysis or component selection can be made concerning proposed PTO belt drive, certain information must be available. (See Table 1)

TABLE 1

Information Required for Drive Analysis

ITEM	FUNCTION
Type of application and duty rating.	Determines required horsepower and belt & clutch service factors (SF). Consult belt manufacturer and PTO manufacturer for service factors.
Drive and driven shaft speeds (drive ratio) and center distance between drive and driven shafts.	Determines belt wrap and driven pulley diameter (D2). Determines tension ratio.
Maximum engine horsepower at rated engine speed.	Determines effective belt pull (EP), equals $(T_1 - T_2)$ and total belt pull load $(T_1 + T_2)$ at rated speed.
Engine horsepower and speed at maximum torque condition.	Determines effective belt pull (EP), equals (T_1+T_2) and total belt pull load (T_1+T_2) at maximum torque condition.
V-Belt characteristics from belt manufacturer's design manual.	Supplies belt service factor, belt section, HP per belt and belt mass constant.
Assembly number of desired PTO and desired location of sheave load centerline (X-dimensions).	Determines allowable shaft load limits. See Table 3, p. 17, or consult PTO manufacturer.
Maximum allowable shaft load. See Table 3, p. 17, or consult PTO manufacturer.	Determines limiting values for maximum dynamic sheave load (Ld) and maximum static sheave load (Ls).

A theoretical example will illustrate the type of analysis which can be performed to evaluate a proposed PTO V-Belt drive, provided all the information listed in Table 1 is available.

DDC distributors have access to a computer program through the Sales Engineering Department for performing analyses and can provide assistance to customers who are not confident of their own ability to perform the necessary analysis.

PROBLEM: Determine the appropriate PTO assembly number, drive and driven sheave diameters and the proper number and size of V-Belts for the following application (See Table 2 on page 7):

TABLE 2

Problem Data

Application: Sawmill

Duty Rating: Intermittent

Power Required: Approximately 140 HP

Driven Equipment RPM: 1000-1100 RPM

Engine Selected: 4-71T with 7E75 injectors

Engine Rated Power and Speed: *200 BHP @ 2100 RPM

Engine Power & Speed at Maximum *120 BHP @ 1200 RPM

Torque Condition: (Peak Torque = 533 LB-FT)

Drive Ratio:

Drive shaft (engine)

speed to be twice driven

shaft speed.

Shaft Center Distance: Approximately 80 inches.

Desired Sheave Load Location: 5 inches (X-Dimension on Table 3, p. 17)

*BHP values are SAE basic engine power for a conservative design approach. For a less conservative design, accurate net BHP values could be used.

PROBLEM ANALYSIS

In addition to the information given in Table 2, it is necessary to consult the belt manufacturer's design manual or Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers' Association, Inc. to obtain the required V-Belt characteristics (required belt section, service factor, horsepower rating per belt and belt mass factor).

For an intermittent duty sawmill application, the standards specify a belt service factor (SF) of 1.3. From this, design horsepower is calculated to be engine horsepower x 1.3 = 200 bhp x 1.3 = 260 bhp @ 2100 RPM.

The analysis procedure and solution appear on the following pages.

PROCEDURE

Step 1. Determine tentative belt section.

Selection can be made from Figure 2, page 18, or belt manufacturers manual for the design horsepower and rated engine speed of the application.

Engine speed is the speed of the smaller sheave of the two-sheave system.

Step 2. Determine tentative drive sheave diameter (D1).

For this method the drive sheave diameter (D1) is always the smaller diameter.

From belt manufacturer's catalog select, for the belt section determined in Step 1, as large a diameter as belt speed and application permits. The larger the sheave diameter, the smaller the number of belts and the smaller the static pulley load, see Steps 10 and 11. D1 is in inches. Record horsepower rating per belt for use in Step 10a.

Step 3. Determine tentative driven sheave diameter (D2).

D2 (in.) =
$$\frac{D1 \times engine \text{ speed}}{Driven \text{ shaft speed}}$$
 or D1 x Drive Ratio

Step 4. Determine wrap angle.

Wrap angle = 180° - $2 \propto$ where, $\propto = Sin^{-1} \frac{(D2 - D1)}{2C}$. See Figure 3, page 19 for \propto and C locations. When C is not known, problem can be evaluated with a tentative, C = 1/2 (D1 + 3D2).

Step 5. Determine tension ratio (R).

Read R for wrap angle calculated in previous step, from Tension Ratio curve, Figure 4, page 19. Tension ratio (R) is used in Step 8a.

Step 6. Determine belt speed (S).

S(fpm) =
$$\frac{\pi(D1)N}{12}$$
, where $\pi = 3.1416$. D1 = Drive sheave diameter determined in Step 2. N = Rated engine speed, rpm — (See Table 2, Page 7).

PROCEDURE (CONT'D)

Step 7. Determine effective belt pull (EP).

 $EP (lb) = \frac{33,000 \times HP \times SF}{S}, \text{ where HP} = \text{Maximum engine power at rated engine speed. (See Table 2, page 7) } SF = Service factor. This is a function of application and is available from the belt manufacturers manual. S = Belt speed as determined in Step 6.$

- Step 8. Determine total beit pull $(T_1 + T_2)$.
 - 8a. Determine T₂.

 T_2 (lb) = $\frac{EP}{R-1}$, where, EP = Effective pull (lb) determined in Step 7. R = Tension ratio determined in Step 5.

8b. Determine T..

 T_1 (lb) = R x T_2 , where, R = Tension ratio determined in Step 5. T_2 = Belt pull determined in Step 8a.

8c. Determine total belt pull $(T_1 + T_2)$.

Total belt pull (lb) = $T_1 + T_2$ where, T_1 = Belt pull determined in 8b. T_2 = Belt pull determined in 8a.

Step 9. Determine belt tension load.

Belt tension load (lb) = .3 $(T_1 + T_2)$, where, $T_1 + T_2 =$ Belt pull determined in 8c.

NOTE: Experience indicates that 30% of the belt pull load is an adequate tension load.

Step 10. Determine belt centrifugal force load (CNF).

CNF (lb) = $2 \times B \times Fc$, where, B = Number of belts (See Step 10a), Fc = Lb per belt strand (See Step 10b).

10a. Determine number of belts (B).

 $B = \frac{HP \times SF}{HP \text{ Rating per belt}}, \text{ where HP} = \text{Engine horsepower (Table 2, page 7)}. SF = Service factor (see step 7). HP Rating per belt (See Step 2).}$

PROCEDURE (CONT'D)

10b. Determine Fc (Force in lb per belt strand).

Fc = M ($\frac{S}{1000}$)², where M = Belt mass factor. Use value from Table 4, page 19 for tentative belt size selected in Step 1. S = Belt speed (See Step 6).

Step 11. Determine static sheave load (L_s).

- 11a. Minimum L_s (lb) = Total belt pull load + Belt centrifugal force load = [(T₁ + T₂) + CNF] CF, where, (T₁ + T₂) = Belt pull determined in Step 8c. CNF = Belt centrifugal force load determined in Step 10. CF = Arc-of-contact correction factor. (See Engineering Standards of Mechanical Power Transmission Association and Rubber Manufacturers Association, Inc. or belt manufacturers manuals.)
- 11b. Maximum L_s (lb) = Total belt pull load + Belt centrifugal force load + Belt tension load = $[(T_1 + T_2) + CNF + .3 (T_1 + T_2)]$ CF, where, $(T_1 + T_2) = Belt$ pull determined in Step 8c. CNF = Belt centrifugal force load determined in Step 10. .3 $(T_1 + T_2) = Belt$ tension load determined in Step 9. CF = Arc-of-contact correction factor, see Step 11a.

NOTE: When this value (11b) exceeds critical load in Table 3, page 17, tentative belt section (Step 1) and tentative drive sheave diameter (Step 2) CANNOT BE USED. Another selection must be made and evaluated in Steps 3 through 11b.

Step 12. Determine belt tension (BT).

- 12a. Minimum BT (lb) = $\frac{\text{Min. L}_s}{2B}$, where, Min. L_s = Minimum static sheave load determined in Step 11a. B = Number of belts determined in Step 10a.
- 12b. Maximum BT (lb) = $\frac{\text{Max. L}_s}{2B}$, where, Max L_s = Maximum static sheave load determined in Step 11b. B = Number of belts, see Step 12a.

NOTE: Belt tension values are always a function of static sheave loads (Min. L_s and Max. L_s) determined in Steps 11a and 11b.

PROCEDURE (CONT'D)

- Step 13. Determine dynamic sheave load (L_d).
 - 13a. Minimum L_d (lb) = Total belt pull load = $(T_1 + T_2)$ CF, where, $(T_1 + T_2)$ = Belt pull determined in Step 8c. CF = Arc-of-contact correction factor, see Step 11a.
 - 13b. Maximum L_d (lb) = Total belt pull load + belt tension load = [(T₁ + T₂) + .3 (T₁ + T₂)]
 CF, where, (T₁ + T₂) = Belt pull determined in Step 8c.
 .3 (T₁ + T₂) = Belt tension load determined in Step 9. CF = Arc-of-contact correction factor, see Step 11a.

NOTE: When this value (Step 13b) exceeds critical load in Table 3, page 17, tentative belt section (Step 1) and tentative drive sheave diameter (Step 2) CANNOT BE USED. Another selection must be made and evaluated in Steps 3 through 13.

SOLUTION

Trial A

Trial B

Condition a

Condition b

Condition c

(Maximum Horsepower)

(Maximum Torque)

(Maximum Horsepower)

Step 1. Determine Belt Section.

Rated speed = 2100 rpm
For these two values,
from Section I, Figure 2,
page 18, SELECT 5V BELT
SECTION.

Note: Drive configuration, belt section and D1, determined by "Condition a" will be evaluated for hp and speed for this condition.

For design horsepower of 182 (See Trial A, Condition a) and 2100 rpm from Section II, Figure 2, page 18, SELECT C BELT SECTION.

Step 2. Determine Drive Sheave Diameter.

$$^{*}D1 = 11.5 in.$$

$$^*HP/belt = 17.78$$

Step 3. Determine Driven Sheave Diameter.

$$D2 = 11.5 in. x 2$$

$$= 23 in.$$

Step 4. Determine Wrap Angle.

$$\approx = \sin^{-1} \frac{23 - 11.5}{2 \times 80}$$

^{*}See Engineering Standards of Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc. However, belt manufacturers catalogs & manuals may also be used.

SOLUTION (CONT'D)

Step 5. Determine Tension Ratio (Fig. 4, p. 19).

$$R = 4.60$$

$$R = 4.60$$

$$R = 4.64$$

Step 6. Determine Belt Speed.

$$S = \frac{\pi (12) 2100}{12}$$

$$S = \frac{\pi (12) 1200}{12}$$

$$S = \frac{\pi (11.5) 210}{12}$$

$$= 6600 \text{ fpm}$$

$$= 6350 \text{ fpm}$$

Step 7. Determine Effective Belt Pull.

$$EP = \frac{33,000 (140) 1.3}{6600}$$
$$= 910 lb.$$

$$EP = \frac{33,000 (89) 1.3}{3771}$$
$$= 1014 lb.$$

$$EP = \frac{33,000 (140) 1.3}{6350}$$
$$= 94.5 \text{ lb.}$$

Step 8. Determine Total Belt Pull $(T_1 + T_2)$.

Step 8a.

$$T_2 = \frac{910}{4.60 - 1.00}$$
$$= 253 \text{ lb.}$$

$$T_2 = \frac{1014}{4.60 - 1.00}$$
$$= 282 \text{ lb.}$$

$$T_2 = \frac{945}{4.64 - 1.00}$$
$$= 260 \text{ lb.}$$

Step 8b.

$$T_1 = 4.60 (253)$$

= 1164 lb.

$$T_1 = 4.60 (282)$$

= 1300 lb.

$$T_1 = 4.64 (260)$$

= 1205 lb.

Step 8c.

$$T_1 + T_2 = 1164 + 253$$

= 1417 lb.

$$T_1 + T_2 = 1300 + 282$$

= 1582 lb.

$$T_1 + T_2 = 1205 + 260$$

= 1465 lb.

Step 9. Determine Belt Tension Load.

Belt tension load

Belt tension load = .3 (1582) = 475 lb. Belt tension load = .3 (1465) = 440 lb.

Step 10. Determine Centrifugal Force Load (CNF).

$$CNF = 2 (7) 42.3$$

= 592 lb.

$$CNF = 2 (11) 60.1$$

= 1322 lb.

SOLUTION (CONT'D)

Step 10a. Number of Belts.

$$B = \frac{140 (1.3)}{27.07} \qquad B = \frac{89 (1.3)}{20} \qquad B = \frac{140 (1.3)}{17.78}$$
= 6.7 = 5.8 = 10.25 = 11 belts because Condition a requires it.

Step 10b. Force per Strand

Fc = .97
$$(\frac{6600}{1000})^2$$
 Fc = .97 $(\frac{3771}{1000})$ Fc = 1.49 $(\frac{6350}{1000})$ = 42.3 lb. Fc = 0.1 lb.

Step 11. Static Sheave Load (L_s).

Step 11a.

Min.
$$L_s = (1417 + 592)$$
 .993 Min. $L_s = (1582 + 193.5)$.993 Min. $L_s = (1465 + 1322)$.993 = 1762 lb. = 2768 lb.

Step 11b.

Per Table 3, page 17, PTO No. 5132201 and PTO No. 5136547 both would be adequate for Trial A, but 5136547 provides more reserve capacity. Critical load is static loading of 2640 lb. For PTO No. 5136547, 2100 rpm, and "X" location given in the problem, **Trial A drive parameters are satisfactory** because static loading for conditions a and b do not exceed the critical value (2417 lb. and 2235 lb. are less than 2640 lb.). **Trial B drive parameters are not satisfactory** because static loading for condition c exceeds critical value (3205 lb. is greater than 2640 lb.).

Step 12. Belt Tension (BT).

Step 12a. $\frac{1995}{2(7)} = 143 \text{ lb.}$		
Min. BT = $\frac{1000}{2(7)}$ = 143 lb.	No calculations required —	No calculations — Drive
2(7)	Max. L _s is less than for	parameters are not
	this condition than it is	satisfactory, see note at
Step 12b. 2417	for "Condition a".	conclusion of Step 11b
Step 12b. Max. BT = $\frac{2417}{2(7)}$ = 175 lb.		calculations.

SOLUTION (CONT'D)

Step 13. Dynamic Sheave Load.

No calculations — critical load for this problem was the static load, see discussion after calculations for Step 11b.

Trial A

Trial B

Conclusions

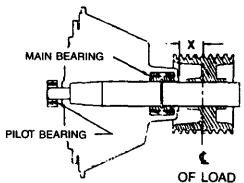
Drive sheave diameter of 12 in. and seven (7) 5V belts are satisfactory.

Drive sheave diameter of 11.5 in. and eleven (11) C belts are not satisfactory.

TABLE 3

MAXIMUM SHAFT LOADS FOR POWER TAKE-OFFS

	Maximum Shaft Load — Lb.								
PTO Asm. No.*	Nominal Clutch	Engine		At "X"	Location	n — In			
Engine Mod. Usage	Capacity Lb-Ft	RPM	0	1	2	3	4	5	6
				0400					4.400
5115884	3235	1600-1800	9580	9100	8560	6980	5890	5095	4490
8V71N		2000-2100	9145	8690	8170	6660	5620	4860	4285
5147643	460	2000-2200	3385	3125	2350	1850	1530	1300	
3-53N, 4-53N	400	2400-2500	3260	3010	2270	1790	1480	1260	
0 0011, 4 0011		2 100 2000	0200] 00.0		1700	1 100	1200	
5128701	1225	2000-2200	5220	3440	2530	2000	1650	1410	
3L, 4L & 6V-53N		2400-2500	5030	3320	2450	1940	1600	1370	
		!	L	ł					
5132201	700	1600-1800	3640	3370	3140	2940	2770	2610	2470
3L & 4L-71N		2000-2100	3470	3215	3000	2810	2640	2490	2360
5136547	1050	1600-1800	7450	6220	4740	3830	3210	2760	2430
3-4-6-71 N, 6V-71N		2000-2100	7100	5950	4540	3660	3070	2640	2320
5136550		1600-1800	7535	6835	5230	4000	3555	2005	0005
						4230		3065	2695
6-71N, 6V-71N		2000-2100	7190	6525	4990	4045	3395	2925	2570
5141604	4140	1600-1800	9905	9515	9155	8820	7880	6820	6005
8,12&16V-71N,8&12V-		2000-2100	9455	9080	8740	8420	7520	6505	5720
	· · · , · — · · · · · · · · · · · · · · · · ·				J	J J			. <u></u> I
5171519	NO	1600-1800	3640	3370	3140	2940	2770	2610	
3L, 4L & 6L-71N	CLUTCH	2000-2100	3470	3215	3000	2810	2640	2490	
д									



Never locate "X" dimension beyond the end of the shaft.

Loads to the left of the heavy line, in the tabulation above, are dynamic loads. This loading occurs when the shaft is running at engine speed; life of the main bearing is critical.

Loads to the right of the heavy line are static loads. This loading occurs when the shaft is stopped and the engine is running at the speeds shown; life of the pilot bearing is critical.

^{*}Contact DDC Sales Engineering for other available PTO's that are not covered by this table.

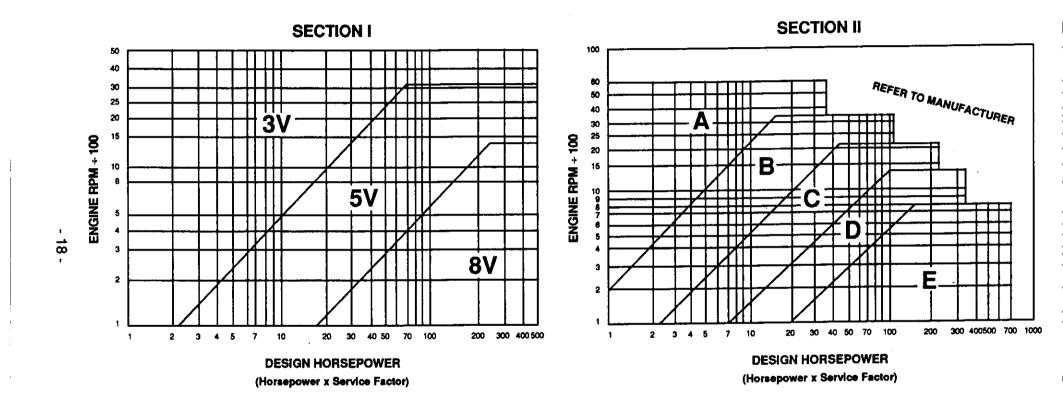


Fig. 2 V BELT SECTIONS FOR DESIGN HORSEPOWER AND ENGINE SPEED

 T_1 = Tight-Side Tension (lb) T_2 = Loose-side Tension (lb)

DRIVE SHEAVE

DRIVEN SHEAVE

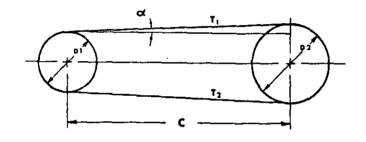
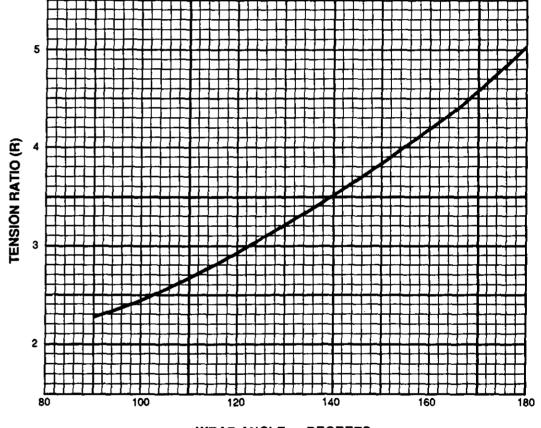


Fig. 3 SCHEMATIC DRIVE DIAGRAM

Table 4

MASS FACTORS (M) FOR V-BELT SECTION			
Belt Section	М		
3V	0.34		
5V	0.97		
8V	2.30		
В	0.83		
С	1.49		



WRAP ANGLE - DEGREES

Fig. 4 TENSION RATIO vs. WRAP ANGLE

BELT TENSION

Proper drive alignment and belt tension are necessary for good drive efficiency. For proper alignment, drive and driven shafts should be parallel and V-Belts should run at right angles to the shafts. For proper belt tension, tension values should be set within the limits calculated in the drive analysis. Tension may be checked with a tension gauge or with a weight scale.

With a gauge, tension values are usually read directly. Check with gauge manufacturer if calibration is required for change in belt configuration. A gauge, in position on a belt is shown below:

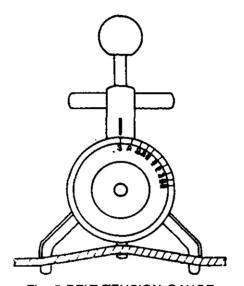


Fig. 5 BELT TENSION GAUGE

With the weight scale, both force and deflection are measured. Weight-scale method is pictured in Figure 6, page 21. To illustrate, the solution for the example (see page 16) lists an acceptable tension range of 143 to 175 lbs. for the 5V belt. For this range, to be correctly tensioned, force values must measure between 10.0 and 12.0 lbs. for a drive having an 80 inch span; belt deflection is 1-1/4 inch.

Tension values apply only for the sheave diameter, belt configuration, "X" location, load and duty cycle shown.

When belts are over-tensioned the shaft is loaded beyond the permissible limits in Table 3, page 17. Power take-off bearings and shaft can fail; engine damage is also possible.

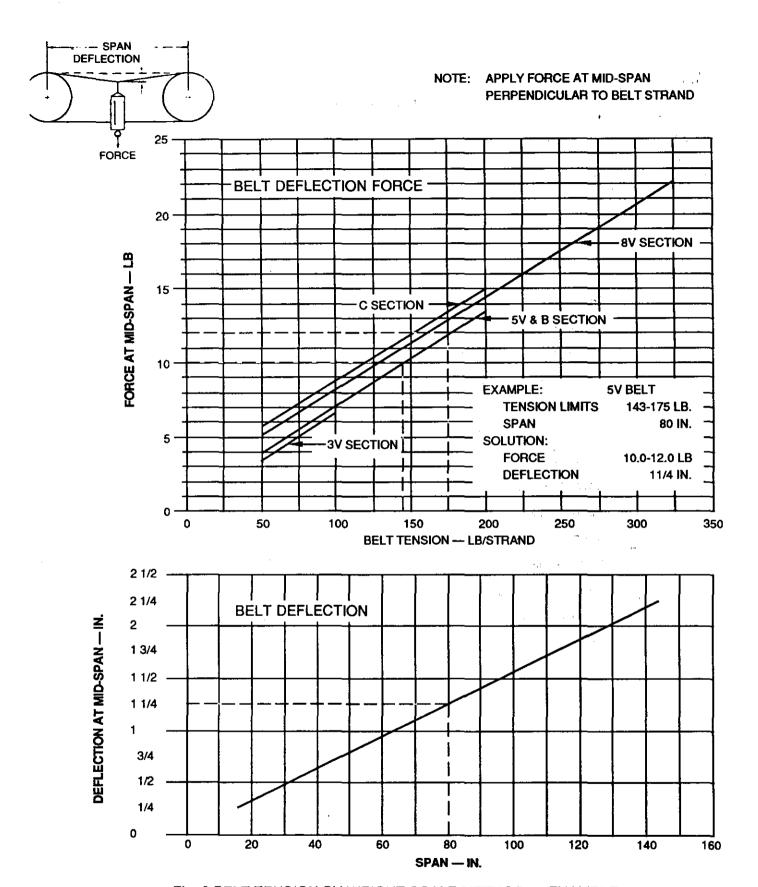


Fig. 6 BELT TENSION BY WEIGHT SCALE METHOD — EXAMPLE

CLUTCH CAPACITY

In addition to analyzing the belt drive to be connected to the PTO, it is necessary to verify that the disconnect clutch in the PTO assembly selected has adequate capacity for the intended application. Clutch service factors are available from the PTO manufacturer. See Table 5.

TABLE 5
PTO CLUTCH SERVICE FACTORS FOR TYPICAL APPLICATIONS

Application	Service Factor (SF)
Blower	4.0
Snowblower	2.0
Compressor — Reciprocating	4.0
Compressor — Turbine Type	2.0
Conveyor	1.5
Drill — Core or Boring	2.0
Farm Tractor PTO	2.0
Power Generator	2.0
Pump — Centrifugal	1.5
Pump Dredge	2.0
Pump — Mud	3.0
Pump — Reciprocating	3.0
Pump — Turbine	1.5
Rock Crusher	3.0
Saw Mill	3.0
Wood Chipper	3.0

^{*}Feramic clutch material recommended. Service Factors courtesy Rockford Clutch.

To determine design torque (Td), multiply peak engine rated torque (Tp) by the clutch service factor (SF): Td = Tp (SF). Compare this value to the nominal clutch capacity of the PTO selected from Table 3, page 17.

For maximum clutch life, design torque (Td) should not exceed the nominal clutch capacity of the PTO. For the proposed sawmill application, Tp = 389 lb-ft (See Table 2, page 7), and SF (clutch) = 3 (See Table 5, page 22).

```
Td = Tp (SF)
= 389 (3)
= 1167 lb-ft.
```

This value far exceeds the nominal clutch capacity of PTO 5132201 (700 lb. ft — Table 3, page 17). Therefore, PTO 5132201 would be completely unsuitable for this application.

The Td of 1167 lb-ft also exceeds the nominal clutch capacity of PTO 5136547, but only by approximately 10%. This is somewhat marginal, but may be acceptable if the user is willing to accept something less than maximum clutch life. If maximum clutch life is essential, the proposed installation should be reviewed with the PTO manufacturer to determine if a change to the clutch facing material or use of a double disc clutch would extend the life of the clutch.

It should be noted that the values given in Table 5 are average service factors. Special conditions such as unusually high starting loads, frequent sever shock loads, considerable clutch slippage during engagement, frequent engagement and disengagement of the clutch and frequent reversals of load on the output shaft may call for a higher service factor. Conversely, the absence of any of these severe conditions may permit a reduction of the service factor.

Whenever doubt exists as to the proper service factor to use, the PTO manufacturer should be consulted.

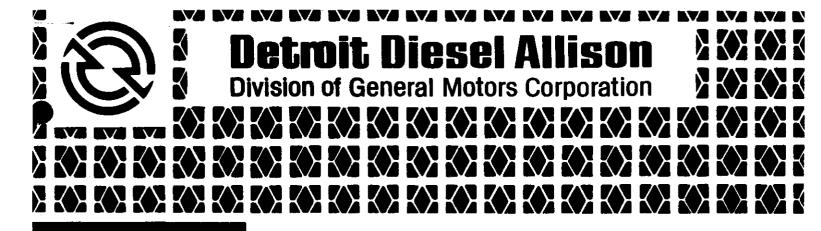
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POWER TAKE-OFF V-BELT DRIVES

FOR DETROIT DIESEL ENGINE APPLICATIONS

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INTRODUCTION

Transmission of engine power through a power take-off belt system requires careful sheave and belt selection. Proper sheave and belt design is important to insure the life of both the power take-off bearings and the shaft. Good design includes the proper selection of sheave diameter, sheave location from the end of the power take-off housing, number and type of belts and belt tension. These sheave and belt factors must be worked out so the shaft side loads tabulated in Table 5, page 15, are not exceeded. Maximum allowable side loading is different for each power take-off and varies with engine speed and with the location, "X", of the centerline of the sheave load.

This bulletin contains a discussion of suggested sheave and belt recommendations which begin on page 1, a detailed drive analysis procedure which begins on page 3, belt tension information which begins on page 12, and drive sheave and belt charts in the Appendix beginning on page 16.

In this initial release, standard V-belts for two-sheave systems used with short-shaft power take-offs are covered. Later bulletin additions will discuss Poly V-belts, long shaft power take-offs, torsional drives, and clutch ratings as defined by horse-power, torque and usage.

The purpose of this bulletin is to assist in the selection of adequate drive sheave diameters and belt sections. Specific dimensioning of sheave and belts can be obtained from manufacturers' manuals and catalogs.

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SHEAVE AND BELT RECOMMENDATIONS

Suggested sheave and belt recommendations are summarized in chart form in the Appendix. To locate the correct Sheave and Belt Chart, consult the listing D1-0000-00-3 shown as Table 6, page 16. For the desired power take-off and engine combination, select the Sheave and Belt Chart which covers the proper power rating. On the Sheave and Belt Chart, the sheave diameter, maximum and minimum belt tension and the number of belts are tabulated versus load centerline location. When possible, the chart includes information for more than one belt section.

Sections for conventional belt designations are illustrated below:

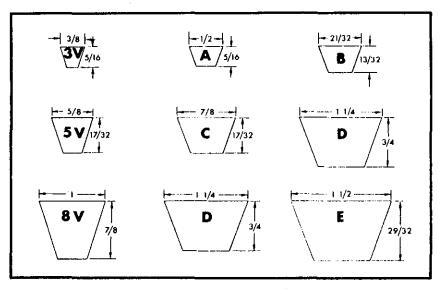


Fig. 1 V BELT SECTION

For a short shaft power take-off, the centerline of the load on the shaft should be as close to the bearings as possible. Since the type of drive is decided by the attached machinery, it is necessary for the owner to determine the centerline position of the belts. Once the centerline is determined, he may lay a scale on the power take-off housing shaft and measure the distance from the power take-off housing to the centerline. This distance is designated as "X" and is illustrated on any Sheave and Belt Chart, page 17, for an example. The tabulations below the illustration show the recommended minimum sheave diameter and the number of belts corresponding to this centerline load position on the shaft.

Maximum recommended sheave size is also listed. When sheave size exceeds the maximum dimension shown, the centrifugal force of the system is increased to values which definitely reduce bearing life.

The following problem illustrates usage of the listing and the charts.

PROBLEM: Determine minimum sheave diameter and proper belt size for the following application:

Engine	4-71, HV7
Application	Saw Mill
Power Rating	Intermittent
Power Take-off Part Number	5136547
Desired Location of Centerline, "X"	5 in.

SOLUTION: The chart listing, page 16, for power take-off 5136547, refers to D1-1043-00-1 which appears on page 24. From the information on this chart, two of the possible solutions are:

Belt Section	3 V	5 V	5 V
Minimum Sheave Diameter	10 in.	11 in.	12 in.
Number of Belts	16	8	7
Minimum Belt Tension/Strand	68 lb.	140 lb.	146 lb.
Maximum Belt Tension/Strand	85 lb.	171 lb.	177 lb.

Belt tensioning is covered on page 12 through 14.

DRIVE ANALYSIS PROCEDURE

Following procedure determines, for given loading conditions, sheave diameters and belts which do not exceed the capacity of the power take-off shaft and power take-off bearings. Parameters which describe belt configuration for the drive are belt type, belt size, belt capacity and the number of belts. Belt data is found in catalogs and manuals available from belt manufacturers. Table 1 lists particulars required for an analysis and usage of these particulars.

Table 1

INFORMATION REQUIRED FOR DRIVE ANALYSIS			
<u>Item</u>	Usage		
Maximum engine horsepower at engine rated speed.	Determines effective pull (EP) and belt pull load ($T_1 + T_2$).		
Engine horsepower at "maximum torque" engine speed.	Determines effective pull (EP) and belt pull load ($T_1 + T_2$).		
Power Take-off Part Number.	Allowable load limits, see Table 3 page 9.		
Type of application and power rating.	Determines service factor (SF).		
Drive and driven shaft speeds (drive ratio).	Determines wrap angle. Determines driven pulley diameter (D2).		
Distance between drive and driven shaft centers.	Determines wrap angle.		
V belt characteristics from belt manufacturers design manual.	Supplies service factor (SF), tension ratio (R), HP rating for belt, and belt mass constant (M).		
Maximum allowable shaft load.	Limiting values for maximum dynamic pulley load (Ld) and maximum static pulley load (Ls), see Table 3, page 9.		

For an adequate drive, components must be selected so the critical condition, whether it is the dynamic or the static load, does not exceed the values listed in Table 3, page 9. <u>Maximum dynamic load</u> is the sum of the belt pull loads and the belt tension load. <u>Maximum static load</u> is the sum of the belt pull loads, the belt tension load, and the belt centrifugal force load. Evaluate drive components operation at maximum horsepower and maximum torque conditions.

PROBLEM:

Determine minimum drive sheave diameter and proper belt size for the following application:

Engine 4-71, HV7 Injectors

Application Saw Mill
Power Rating Intermittent
Power Take-off Part Number 5136547

Desired Location of Centerline, "X" 5 in.

Drive Ratio Drive shaft speed twice driven shaft speed.
Span 80 in.

As shown on Chart D1-1043-00-1, page 24, there are a number of drive possibilities. Two of these are examined on pages 5 through 8; for convenience, the procedures and the solutions are paralleled on these pages. Maximum-torque-condition for second drive (Trial B) is not shown because maximum-horsepower-condition represents a loading in excess of the critical value, see page 8.

Table 2

INFORMATION FOR DRIVE ANALYSIS - EXAMPLE

<u>Item</u> <u>Value</u>

Maximum engine horsepower at engine rated speed. *143 hp, 2100 rpm.

Engine horsepower at "maximum-torque" engine *114 hp, 1600 rpm.

speed.

Power take-off part number. 5136547

Type of application and power rating. Application: Saw mill. Rating: Intermittent

Drive and driven shaft speeds (drive ratio). Given drive ratio is 2:1.

Distance between drive and driven shaft centers. Given span (C) is 80 in.

V-belt characteristics. **Service factor (SF), tension ratio (R), horsepower rating

for belt.

See Table 4, page 11, for belt mass constant (M).

Maximum allowable shaft load. See Table 3, page 9, for limiting values for maximum

dynamic pulley load (Ld) and maximum static pulley load

(Ls).

*Both values are SAE basic engine power for a conservative approach, accurate net values could be used.

**Specifications for example are from Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc. However, belt manufacturers catalogs and design manuals may be used.

4

PROCEDURE

When C is not known, problem can be evaluated

with a tentative:

 $C = \frac{1}{2} (D1 + 3D2)$

SOLUTION

	<u>Trial A</u>		Trial B
	Condition a Maximum horsepower	Condition b Maximum torque	Condition c Maximum horsepower
Step 1. Determine tentative belt section. Selection can be made from Figure 2, page 10, or belt manufacturers manual for the design horsepower and rated engine speed of the application. Engine speed is the speed of the smaller sheave of the two-sheave system.	From Table 2, page 4 Design hp = 143 x 1.3 = 186 Rated speed = 2100 rpm For these two values, from Section I, Figure 2, page 10, SELECT 5V BELT SECTION	Note: Drive configuration, belt section and D1, deter- mined by 'Condition a'' will be evaluated for hp and speed for this condition.	For design horsepower of 186 (See Trial A, Condition a) and 2100 rpm from Section II, Figure 2, page 10, SELECT C BELT SECTION.
Step 2. Determine tentative drive sheave diameter (D1).			
For this method the drive sheave diameter (D1), is always the smaller diameter.	*D1 = 12 in.	D1 = 12 in. (See Condition a)	*D1 = 11.5 in.
From belt manufacturer's catalog select, for the beit section determined in Step 1, as large a diameter as belt speed and application permits. The larger the sheave diameter, the smaller the number of belts and the smaller the static pulley load, see Steps 10 and 11. Dis in inches. Record horsepower rating per belt for use in Step 10a.	*HP/belt = 27.07	*HP/belt = 24.15	*HP/belt = 17.78
Step 3. Determine tentative driven sheave diameter (D2).	D2 = 12 in. x 2 (See Table 2, page 4)		D2 = 11.5 in. x 2
D2 (In.) = D1 x engine speed or D1 x Drive Ratio Driven shaft speed	= 24 in.		= 23 in.
Step 4. Determine wrap angle. Wrap angle = 180° - 2 Sin 1 (D2-D1) 2C See Figure 3, page 11 for and C locations.	$ \approx = \sin^{-1} \left(\frac{24 - 12}{2 \times 80} \right) $ (See Table 2, page 4)		

≪ = 4.1°

(5)

^{*}See Engineering Standards of Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc. However, belt manufacturers catalogs and manuals may also be used.

 $= 180^{\circ} - 8.6^{\circ}$

R = 4.60

 $S = \frac{77(12) \ 2100}{12}$

= 6600 fpm

 $= 171.4^{\circ} \text{ or } 171^{\circ}$

Maximum torque #Wrap angle = 171°

 $s = \frac{77(12) \ 1600}{12}$

= 5025 fpm

Condition b

Condition c

Maximum horsepower Wrap Angle $\ll = 180^{\circ} - 2 \ll$ $= 180^{\circ} - 8.7$ $= 171.8^{\circ} \text{ or } 172^{\circ}$

#R = 4.60

R = 4.64

 $S = \frac{77 (11.5) 2100}{12}$

= 6350 fpm

Step 5.

Step 4 - Continued

Determine tension ratio (R).

Read R | for wrap angle calculated in previous step, from Tension Ratio curve, Figure 4, page 11. Tension ratio (R) is used in Step 8a.

Step 6.

Determine belt speed (S).

π ⋅ 3, 1416

D1 Drive shaft diameter determined in Step 2.

Rated engine speed, rom -(See Table 2, page 4).

Step 7.

9

Determine effective pull (EP)

EP (1b) $\frac{33,000 \times HP \times SF}{S}$, where:

Maximum engine power at rated

engine speed. (see Table 2, page 4). SF Service factor (see Table 2, page 4). This is a function of application and is available from the belt manufacturers manual.

S - Belt speed as determined in Step 6.

 $EP = \frac{33,000(143)\ 1.3}{6600}$ = 930 lb

 $EP = \frac{33,000 (114) 1.3}{5025}$ = 974 lb

 $EP = \frac{33,000 (143) 1.3}{6350}$

968 lb

Determine total best pull (T1 + T2).

sa. Determine T2.

 T_2 (1b) $=\frac{EP}{R-1}$, where:

EP Effective pull (lb) determined in Step 7.

 $T_2 = \frac{930}{4.60 - 1.00}$

= 258 lb

 $T_2 = \frac{968}{4.64 - 1.00}$ $T_2 = \frac{974}{4.60-1.00}$ = 270 lb= 266 lb

R Tension ratio determined in Step 5,

sh. Determine T1.

T1 (ib) Rx T2, where:

R Tension ratio determined in Step 5. T. Bolt pull determined in Step Sa.

 $T_1 = 4.60 (258)$ = 1187 lb

 $T_1 = 4.60 (270)$ = 1242 lb

 $T_1 = 4.64 (266)$ = 1234 lb

Se, Determine total belt pull (T1 T2).

Total best pull (lb) - $T_1 \rightarrow T_2$ where:

T1 Belt pull determined in 8b.

To Bely out determined in Sa.

 $T_1 + T_2 = 1187 + 258$

= 1445 lb

 $T_1 + T_2 = 1242 + 270$

= 1512 lb

 $T_1 + T_2 = 1234 + 266$

= 1500 lb

#Use ''Condition a'' calculation.

PROCEDURE

SOLUTION

Step 9.	Condition a
Determine belt tension load.	Maximum horsenower

Belt tension load (ib) =
$$.3 (T_1 + T_2)$$
, where:

 $T_1 + T_2 = \text{Belt pull determined in 8c.}$

Belt to some series and series that 30% of the belt pulley load is an adequate tension load.

Determine belt centrifugal force load (CNF)

CNF (1b) = 2 x B x Fc, where:

B=Number of belts (See Step 10a) Fc=Lb per belt strand (See Step 10b)

 $B^{\perp} \frac{HP \times SF}{HP \ Rating \ per \ belt, \ where;}$

IIP = See Table 2, page 4)

SF = Service factor (See Step 7)

HP Rating per belt (See Step 1)

10b. Determine Fc (Force in 1b per belt strand)

Fc =
$$M\left(\frac{S}{s}\right)^2$$
, where:

M = Belt mass factor. Use value from Table 4, page 11 for tentative belt size selected in Step 1.

S = Belt speed (See Step 6)

Step 11.

Determine static sheave load, (L_n) .

11a. Minimum L₀ (ib) = Total belt pull load +
Belt centrifugal (orce load

(T₁ + T₂) = Belt pull determined in Step 8c

CNF = Belt centrifugal force load determined in Step 10.

CF = Arc-of-contact correction factor.

Trial A

Condition a Maximum horsepower Belt tension load = .3(1445) = 434 lb

$$CNF = 2(7) 42.3$$

= 592 lb

$$B = \frac{143(1.3)}{27.07}$$

$$Fc = .97 \left(\frac{6600}{1000} \right)^2$$

$$= 42.3 lb$$

Min.
$$L_S = (1445 + 592) .993$$

$$= 2023 lb$$

Condition b

Maximum torque
Belt tension load
= .3(1512)
= 454 lb

$$CNF = 2(7) 24.5$$

= 343 lb

$$B = \frac{114(1.3)}{24.15}$$

$$\mathbf{Fc} = .97 \left(\frac{5025}{1000} \right)^2$$

$$= 24.5 lb$$

Min.
$$L_8 = (1512 + 343) .993$$

$$= 1842 lb$$

Trial B

Condition c

Maximum horsepower
Belt tension load
= .3(1500)

$$CNF = 2(11) 60.1$$

= 1322 lb

$$B = \frac{143(1.3)}{17.78}$$

$$Fc = 1.49 \left(\frac{6350}{1000} \right)^2$$

$$= 60.1 lb$$

Min.
$$L_{s} = (1550 + 1322) .993$$

$$= 2822 lb$$

^{*}See Engineering Standards of Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc. However, belt manufacturers catalogs and manuals may also be used.

Condition a

11b. Maximum La (lb) = Total belt pull load + Belt centrifugal force load + Belt tension

> CNF = Belt centrifugal force load determined in Step 10.

> . 3 (T₁ + T₂) = Belt tension load determined in Step 9.

CF = Arc-of-contact correction factor. see Step Lia.

NOTE: When this value (11b) exceeds critical load in Table 3, page 9, tentative helt section (Step 1) and tentative drive sheave diameter Sten 2) CANNOT BE USED Another selection must be made and evaluated in Steps 3 through 11b.

Step 12,

Determine belt tension (BT).

12a. Minimum BT (1b) =
$$\frac{\text{Min. } L_S}{2B}$$
 , where:

Min. Lg = Minimum static sheave load determined in Step 11a. B = Number of belts determined in Step 10s

12b. Maximum BT (1b) =
$$\frac{\text{Min. L}_8}{2B}$$
 , where:

Max. Lg = Maximum static sheave load determined in Step 11b. B = Number of belts, see Step 12s.

NOTE: Belt tension values are always a function of static sheave loads (Min. La and Mrx. La) determined in Store 11 and 11b.

Step 13.

Determine dynamic sheave load (Ld).

13a. Minimum Ld (1b) = Total helt pull load = (T1 + T2) CF, where (T1 + T2) = Belt pull determined in Step 8c. CF = Arc-of-contact correction factor see Step 11a.

13b. Maximum Let (1b) . Total belt pull load + helt tension load.

= $(T_1 + T_2) + .3 (T_1 + T_2)$ CF where: (T1+T2) = Belt pull determined in Step Sc. .3 (T + T) = Belt tension long determined

in Sten 9. CF = Arc-of-contact correction funtor. see Step 11a.

NOTE: When this value (Step 13b) exceeds critical load in Table 3, page 9, tentative belt section (Step 1) and tentative drive sheave diameter (Step 2) CANNOT BE USED Another selection must be made and evaluated in Steps 3 through 13.

Trial A

Condition b Maximum horsepower

Max. L.

$$= (1445 + 592 + 434) .993$$

 $= (2471) .993$

Maximum torque Max. L.

$$= (1512 + 343 + 454) .993$$

 $= (2309) .993$

$$= 2295 \text{ lb}$$

Trial B

Condition c Maximum horsepower

Max. L.

= (1500 + 1322 + 450)

= (3272).993

= 3249 lb

Critical load is static loading of 2640 lb. This value appears on Table 3, page 9, for the power take-off, speed and 'X" location given in the problem. Trial A drive parameters are satisfactory because static loading for conditions a and b do not exceed the critical value (2454 lb and 2295 lb are less than 2640 lb). Trial B drive parameters are not satisfactory because static loading for condition c exceeds critical value (3249 lb is greater than 2640 lb).

Min. BT =
$$\frac{2023}{2(7)}$$
 = 145 lb

Max. BT
$$=\frac{2454}{2(7)}$$
 = 175 lb

No calculations required -Max. Ls is less for this condition than it is for "Condition a.

No calculations - Drive parameters are not satisfactory, see note at conclusion of Step 11b calculations.

No calculations - critical load for this problem was the static load, see discussion after calculations for Step 11b.

Trial A

Trial B

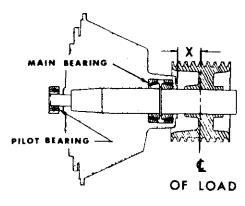
CONCLUSIONS

Drive sheave diameter of 12 in. and seven (7) 5V belts are satisfactory.

Drive sheave diameter of 11.5 in. and eleven (11) C belts are not satisfactory.

20

MAXIMUM SHAFT LOADS FOR POWER TAKE-OFFS



(8

Never locate "X" dimension beyond the end of the shaft.

Loads to left of the heavy line, in the tabulation below, are dynamic loads.

This loading occurs when the shaft is running at engine speed; life of the main bearing is critical.

Loads to the right of the heavy line are static loads. This loading occurs when the shaft is stopped and the engine is running at the speeds shown; life of the pilot bearing is critical.

Table 3

DOWNER TAKE OFF	ENGINE	-	MA	XIMUM AT ''X''				
POWER TAKE-OFF PART NUMBER	RPM	0	T i	2	3	4	5	6
5115882	1600 - 1800	9580	9100	8560	6980	5890	5095	4490
	2000 - 2100	9145	8690	8170	6660	5620	4860	4285
5115884	1600 - 1800	9580	9100	8560	6980	5890	5095	4490
	2000 - 2100	9145	8690	8170	6660	5620	4860	4285
5116493	2000 - 2200	3385	3125	2350	1850	1530	1300	
	2400 - 2500	3260	3010	2270	1790	1480	1260	
5116764	1600 - 1800	9580	9085	8640	8240	7875	7540	7230
	2000 - 2100	9140	8670	8250	7865	7515	7195	6900
5128701	2000 - 2200	5220	3440	2530	2000	1650	1410	
	2400 - 2500	5030	3320	2450	1940	1600	1370	
5132201	1600 - 1800	3640	3370	3140	2940	2770	2610	2470
	2000 - 2100	3470	3215	3000	2810	2640	2490	2360
5136547	1600 - 1800	7450	6220	4740	3830	3210	2760	2430
	2000 - 2100	7100	5950	4540	3660	3070	2640	2320
5136549	1600 - 1800	7535	6835	5230	4230	3555	3065	2695
	2000 - 2100	7190	6525	4990	4045	3395	2925	2570
5141604	1600 - 1800	9905	9515	9155	8820	7880	6820	6005
	2000 - 2100	9455	9080	8740	8420	7520	6505	5720
5171519	1600 - 1800	3640	3370	3140	2940	2770	2610	
	2000 - 2100	3470	3215	3000	2810	2640	2490	
5189512	1600 - 1800	3640	3370	3140	2940	2770	2440	2130
	2000 - 2100	3470	3215	3000	2810	2640	2335	2035

(Horsepower x Service Factor)

SECTION I

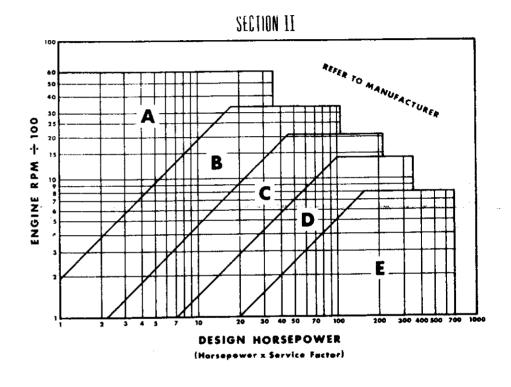


Fig. 2 V BELT SECTIONS FOR DESIGN HORSEPOWER AND ENGINE SPEED

T₁ = Tight-Side Tension (lb)
T₂ = Loose-Side Tension (lb)

DRIVE SHEAVE

DRIVEN SHEAVE

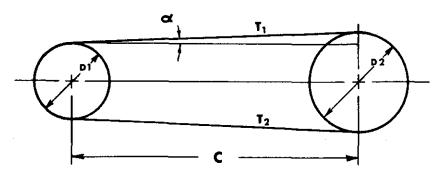


Fig. 3 SCHEMATIC DRIVE DIAGRAM

Table 4

MASS FACTORS (M) FOR V-BELT SECTION							
Belt Section	<u>M</u>						
3V	0.34						
5 V	0.97						
8V	2.30						
В	0.83						
С	1.49						

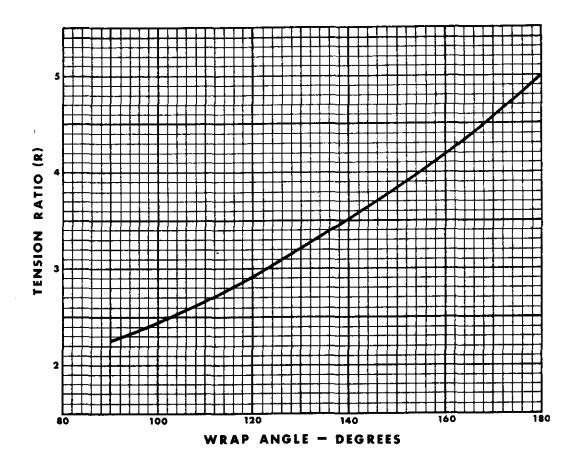


Fig. 4 TENSION RATIO vs. WRAP ANGLE

BELT TENSION

Proper drive alignment and belt tension are necessary for good drive efficiency. For proper alignment, driver and driven shafts should be parallel and V-belts should run at right angles to the shafts. For proper belt tension, tension values should check out within the limits given on the Sheave and Belt Charts or limits calculated.

Tension may be checked with a tension gauge or with a weight scale.

With a gauge, tension values are usually read directly. Check with gauge manufacturer if calibration is required for change in belt configuration. A gauge, in position on a belt, is shown below.

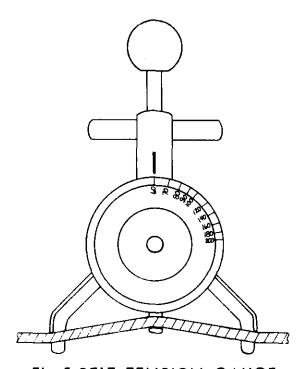
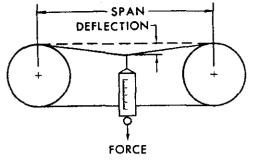


Fig. 5 BELT TENSION GAUGE

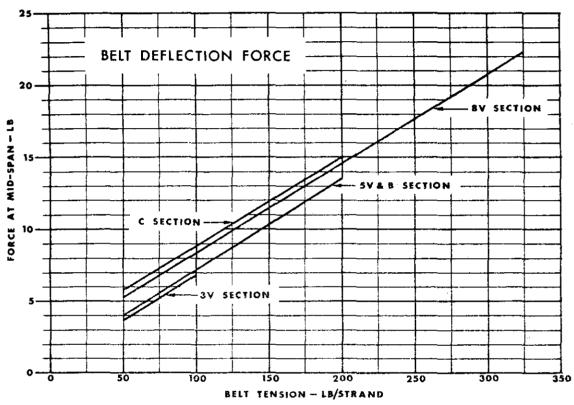
With the weight scale, both force and deflection are measured. Weight-scale method is pictured in Figure 6, page 13. To illustrate, the solution for the example (see page 2) lists an acceptable tension range of 146 to 177 lbs. for the 5V belt. For this range, to be correctly tensioned, force values must measure between 10.0 and 12.0 lbs. for a drive having an 80-inch span; belt deflection is 1 1/4 inch. The solution is shown in Figure 7, page 14.

Tension values, apply only for the sheave diameter, belt configuration, "X" location, load and duty cycle shown.

When belts are over-tensioned the shaft is side loaded beyond the permissible limits in Table 3, page 9. Power take-off bearings and shaft can fail; engine damage is also possible.



NOTE: APPLY FORCE AT MID-SPAN PERPENDICULAR TO BELT STRAND



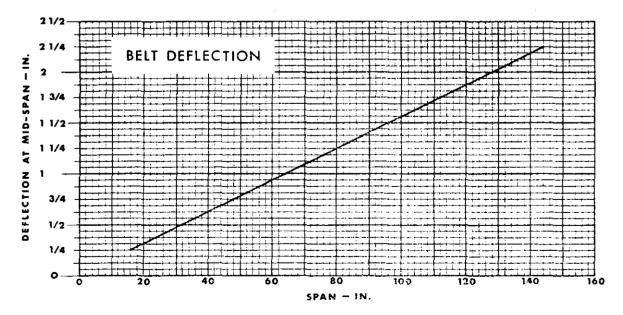
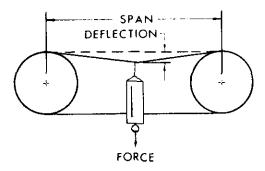
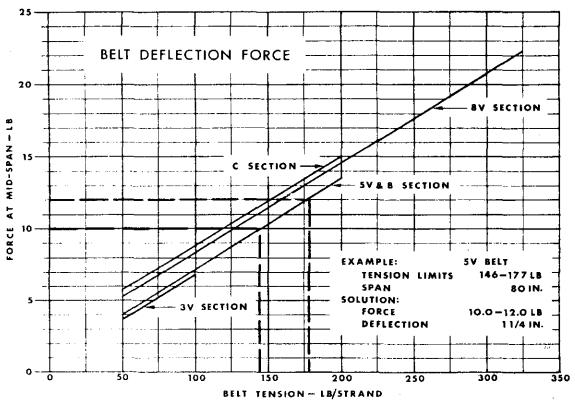


Fig. 6 BELT TENSION BY WEIGHT SCALE METHOD



NOTE: APPLY FORCE AT MID-SPAN PERPENDICULAR TO BELT STRAND



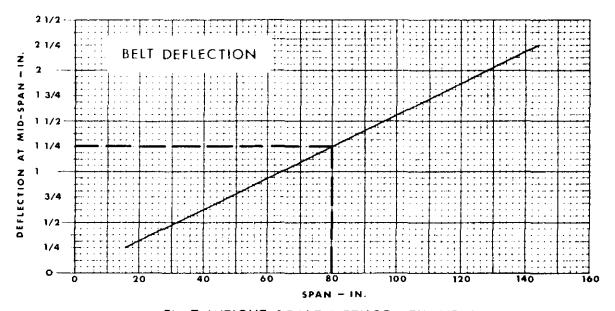
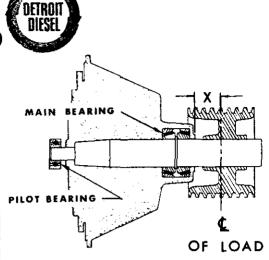


Fig. 7 WEIGHT SCALE METHOD-EXAMPLE



MAXIMUM SHAFT LOADS FOR POWER TAKE-OFFS

Never locate "X" dimension beyond the end of the shaft.

Loads to left of the heavy line, in the tabulation below, are dynamic loads. This loading occurs when the shaft is running at engine speed; life of the main bearing is critical.

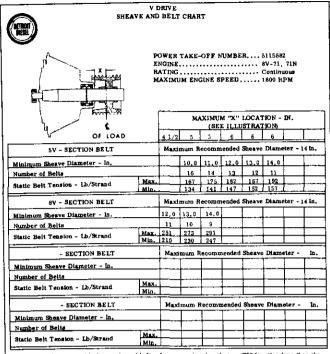
Loads to the right of the heavy line are static loads. This loading occurs when the shaft is stopped and the engine is running at the speeds shown; life of the pilot bearing is critical.

	I LOAD	MAXIMUM SHAFT LOAD - LB.									
POWER TAKE-OFF	ENGINE		,	AT 'X'	LOCAT	ON - IN					
PART NUMBER	RPM	0	1	_ 2	3	4	5	6			
5115882	1600 - 1800	9580	9100	8560	6980	5890	5095	4490			
	2000 - 2100	9145	8690	8170	6660	5620	4860	4285			
5115884	1600 - 1800	9580	9100	8560	6980	5890	5095	4490			
	2000 - 2100	9145	8690	8170	6660	5620	4860	4285			
5116493	2000 - 2200	3385	3125	2350	1850	1530	1300				
	2400 - 2500	3260	3010	2270	1790	1480	1260				
5116764	1600 - 1800	9580	9085	8640	8240	7875	7540	7230			
	2000 - 2100	9140	8670	8250	7865	7515	7195	6900			
5128701	2000 - 2200	5220	3440	2530	2000	1650	1410				
	2400 - 2500	5030	3320	2450	1940	1600	1370				
5132201	1600 - 1800	3640	3370	3140	2940	2770	2610	2470			
	2000 - 2100	3470	3215	3000	2810	2640	2490	2360			
5136547	1600 - 1800	7450	6220	4740	3830	3210	2760	2430			
	2000 - 2100	7100	5950	4540	3660	3070	2640	2320			
5136549	1600 - 1800	7535	6835	5230	4230	3555	3065	2695			
	2000 - 2100	7190	6525	4990	4045	3395	2925	2570			
5141604	1600 - 1800	9905	9515	9155	8820	7880	6820	6005			
	2000 - 2100	9455	9080	8740	8420	7520	6505	5720			
5171519	1600 - 1800	3640	3370	3140	2940	2770	2610				
	2000 - 2100	3470	3215	3000	2810	2640	2490				
5189512	1600 ~ 1800	3640	3370	3140	2940	2770	2440	2130			
	2000 - 2100	3470	3215	3000	2810	2640	2335	2035			

Table 6

DRIVE SHEAVE CHARTS FOR POWER TAKE-OFFS

POWER TAKE-OFF PART NUMBER	ENGINE	RATING	CHART NUMBER	PAGE NUMBER
5115882	8V-71, 71N 8V-71, 71N	Intermittent Continuous	D1-7083-02-1 D1-7083-02-3	17 17
5115884	8V-71, 71N 8V-71, 71N	Continuous Intermittent	D1-7083-02-4 D1-7083-02-2	17 17
5116493	2-53 2-53 3-53N 3-53 3-53, 53N(4V)	Intermittent Continuous Continuous Continuous Intermittent	D1-5023-11-1 D1-5023-11-2 D1-5033-52-1 D1-5033-10-1 D1-5033-02-1	18 18 18 18 19
	3-53 (2V) 4-53N 4-53 4-53, 53N(4V) 4-53 (2V)	Intermittent Continuous Continuous Intermittent Intermittent	D1-5033-11-1 D1-5043-52-1 D1-5043-10-1 D1-5043-02-1 D1-5043-11-1	19 20 20 20 20
5116764	8V-71, 71N 8V-71, 71N	Continuous Intermittent	D1-7083-02-5 D1-7083-02-6	21 21
5128701	6V-53 6V-53N	Continuous Continuous	D1-5063-02-1 D1-5063-52-2	22 22
5132201	3-71 3-71 4-71, 71E, 71N 4-71, 71E, 71N	Intermittent Continuous Intermittent Continuous	D1-1033-11-6 D1-1033-11-5 D1-1043-00-8 D1-1043-00-7	23 23 23 23
5136547	3-71 3-71 4-71, 71E, 71N 4-71, 71E, 71N 6-71, 71E, 71N 6-71, 71E, 71N 6V-71, 71N 6V-71, 71N	Intermittent Continuous Intermittent Continuous Intermittent Continuous Intermittent Continuous Intermittent Continuous	D1-1033-11-2 D1-1033-11-4 D1-1043-00-1 D1-1043-00-6 D1-1063-00-1 D1-1063-00-3 D1-7063-00-1 D1-7063-00-3	24 24 24 23 25 25 25 25 25
5136549	6-71, 71E, 71N 6-71, 71E, 71N 6V-71, 71N 6V-71, 71N	Intermittent Continuous Intermittent Continuous	D1-1063-00-2 D1-1063-00-4 D1-7063-00-2 D1-7063-00-4	26 26 26 26 26
5141604	16V-71, 71N 12V-71, 71N	Continuous Continuous	D1-7163-02-1 D1-7123-02-2	27 27
5171519	3-71 3-71 4-71, 71E, 71N 4-71, 71E, 71N	Intermittent Continuous Intermittent Continuous	D1-1033-11-1 D1-1033-11-3 D1-1043-00-2 D1-1043-00-5	28 28 28 28
5189512	2-71 2-71	Intermittent Continuous	D1-1023-11-1 D1-1023-11-3	29 29

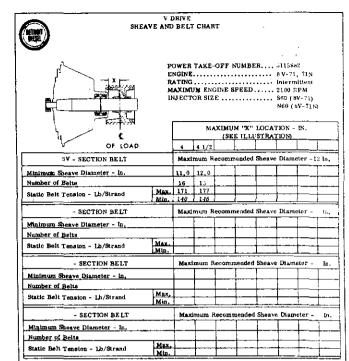


Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°.

Engine speed is the speed of the smaller sheave of the two-sheave system.

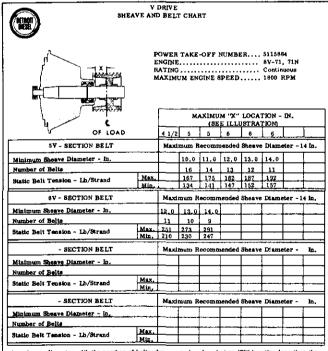
Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc.
Other standards exist for certain drives for automotive, agricultural and oil field machinery.
D1-7083-02-3
11-1-68



Any sheave diameter with the number of belts shown may be pisced at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°. Engine speed is the speed of the smaller sheave of the two-sheave system.

tandard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc. Other standards exist for certain drives for automotive, agricultural and oil field machinery. D1-7683-02-1 11-1-68

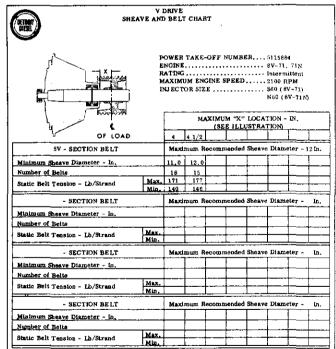


Any sheave diameter with the number of beits shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°.
Engine speed is the speed of the smaller sheave of the two-sheave system.

Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc. Other standards exist for certain drives for automotive, agricultural and oil field machinery.

DI-7083-02-1



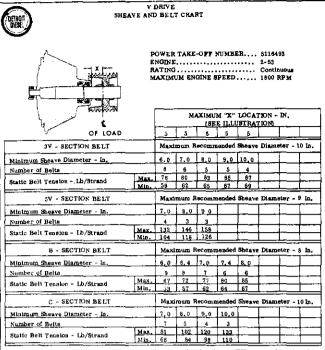
Any sheave diameter with the number of belts shown may be placed at an "%" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°. Engine speed is the speed of the smaller sheave of the two-sheave system.

Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc. Other standards exist for certain drives for automotive, agricultural and oil field machinery.

D1-7083-02-2

11-1-68



Any sheave diameter with the number of helts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°.
Engine speed is the speed of the smaller sheave of the two-sheave system.

Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, inc. Other standards exist for certain drives for automotive, agricultural and oil field mechanisty. Di-5023-11-2

SHE,	V AVE AN	DRIVE DBELT	CHAI	RT								
	OWER NGINE ATING IAXIMI NJECT	TAKE	-off	PEED.		2-53 Interr 2000 1	nittent					
Fred Visit			м	XIMU (SE	M 'X"			IN.				
V OF LOAD		4 1/2	3	5_	5	. 5	l		1.			
3V - SECTION BELT		Maximum Recommended Sheave Diameter - 10 In.										
Minimum Sheave Diameter - In.		6,0	7.0	8.0	9.0	10,0						
Number of Belts		9	7	6	6	5						
Static Belt Tension - Lb/Strand	Mex.	75	79	62	82	86						
State Bert Tension - 107 att and	Min.	-59	62	64	64	58						
5V - SECTION BELT		Maxi	mum R	ecomn	repded	Sheave	Diam	eter - I	O In,			
Minimum Sheave Diameter - In.		7.0	B. 0	9,0	10.0		•					
Number of Belts		5	4	3	3			Τ-	1			
Static Belt Tension - Lb/Strand	Max.	129	144	156	165				L			
DEED DOWN TENSION - DOUGHT MIN	Mlo.	103	115	125	133				l_			
B - SECTION BELT		Maxi	mum R	есопъп	ended	Sheave	Diam	eter -	8 fn.			
Minimum Sheave Diameter - In.		6.0	7.0	6.0								
Number of Belts	11	8	7		·		T	{				
Static Belt Tension - Lb/Strand	Tax. 1											
- I Dy Straig	Min.	53	61	67			ΙΞ.	1				
C - SECTION BELT		Maximum Recommended Sheave Diameter - 11 in.										
Minimum Sheave Diameter - In.		7.0	8.0	9,0	10.0	11.0	i	1	í			
Number of Belts		9	7	5	4	4		1				

Any sheave diameter with the number of beits shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, beits must be tensioned to values shown.

Static Belt Tension - Lb/Strand

Max, 78 99 117 131 143 Min, 65 83 97 110 118

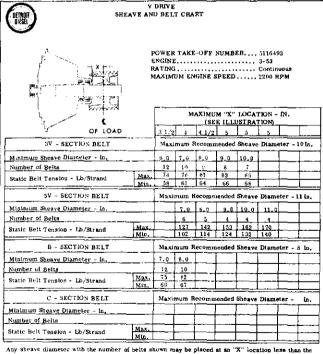
Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°. Engine speed is the speed of the smaller sheave of the two-sheave system.

andard belt specification, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, inc.

Other standards exist for certain drives for automotive, agricultural and oil field machinery.

DI-5023-1-11

11-1-08

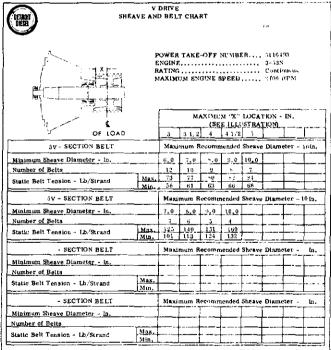


Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, built must be tensioned to values shown.

Belt and shouse information is a function of load factors to 1.4 and wrap angles from 140° to 180°. Engine speed is the speed of the smaller sheave of the two-sheave system.

Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, inc. Other standards exist for certain drives for automotive, agricultural and oil field machine.

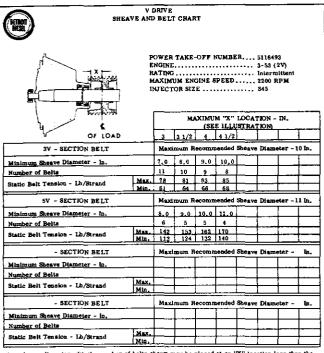
D1-5033-10-1
11-1-68



Any sheave diameter with the number of belts shown may be placed at an "X" lucation less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

selt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°, Engine speed is the speed of the smaller sheave of the two-sheave system.

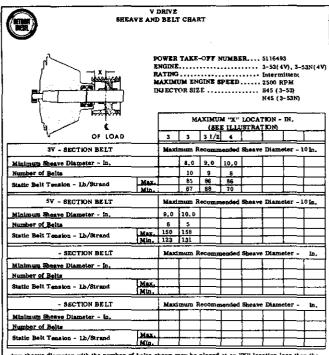
Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc. Other standards exist for certain drives for automotive, agricultural and oil field machinery, D1-5033-52-1 1-11-86



Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and aheave information is a function of load factors to 1.4 and wrap angles from 140° to 190° . Engine speed is the speed of the smaller aheave of the two-sheave system.

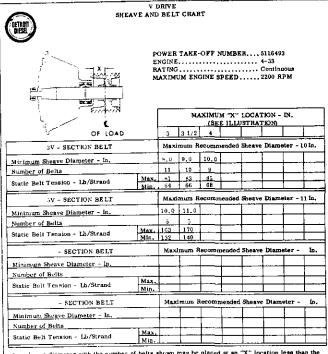
Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, inc. Other standards exist for certain drives for automotive, agricultural and oil field machine. D1-5033-11-1



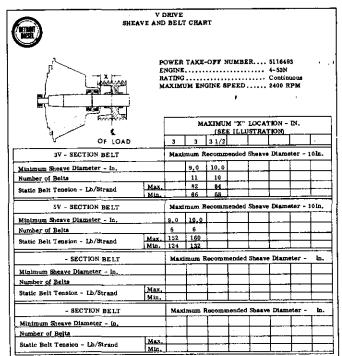
Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°, Engine speed is the speed of the smaller sheave of the two-sheave system.

Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, inc. Other standards exist for cartain drives for automotive, agricultural and oil field machinery. D1-5033-02-1



- Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, helts must be tensioned to values shown.
- Be't and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°. Engine speed is the speed of the smaller sheave of the two-sheave system.
 - ...hard best specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, inc. Other standards exist for certain drives for automotive, agricultural and oil field machine. Di-5043-10-1

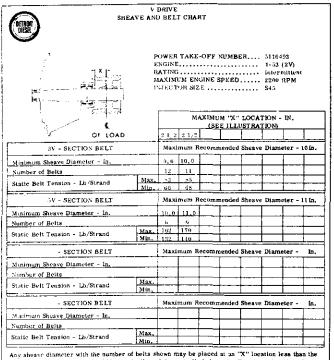


- Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensimed to values shown.
- Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°.
 Engine speed is the speed of the smaller sheave of the two-sheave system.
- Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, inc.

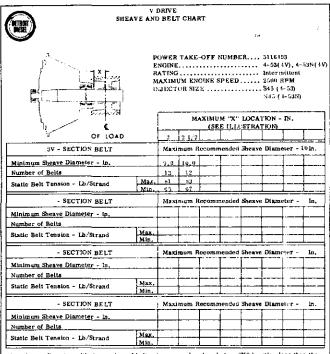
 Other standards exist for certain drives for automotive, agricultural and oil field machinery.

 D1:5043-52-1

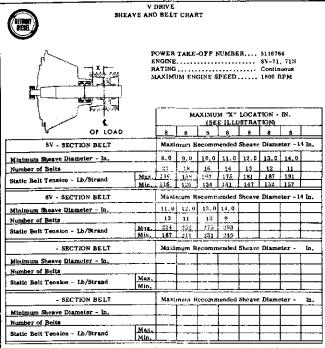
 11:1-1-68



- Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.
- Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°, Engine speed is the speed of the smaller sheave of the two-sheave system.
- Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc.
 Other standards exist for certain drives for automotive, agricultural and oil field machinery.
 D1-5043-11-1



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 D1-3043-02-1

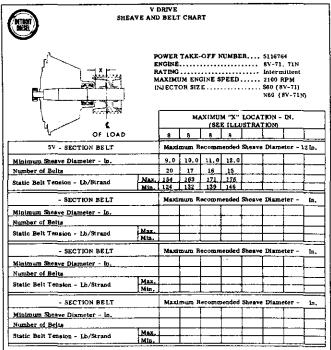


Any sheave diameter with the number of bette shown may be placed at an 'X' location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°.

Engine speed is the speed of the smaller sheave of the two-sheave system.

Standard belt specifications, where used are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc. Other standards exist for certain drives for automotive, agricultural and oil field machinery. D1-7063-02-5 11-1-68

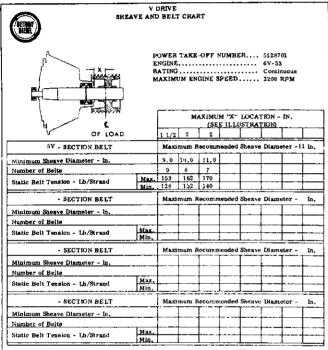


Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°. Engine speed is the speed of the smaller sheave of the two-sheave system.

Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc. Other standards exist for certain drives for automotive, agricultural and oil field machinery. D1-7083-02-16

11-1-68



Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect hearings from overload, belts must be tensioned to values shown.

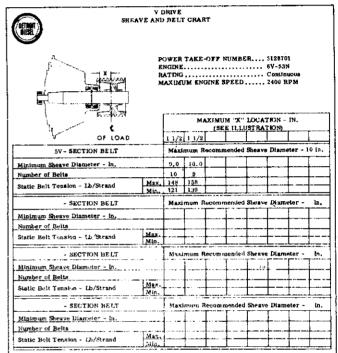
Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°. Engine speed is the speed of the smaller sheave of the two-sheave system.

Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc.

Other standards exist for certain drives for automotive, agricultural and oil field machinery.

D1-5033-02-1

11-1-68



Any sheave diameter with the number of bolts shown may be placed at an "X" location less than the limit limited. Sheave chould be statishly said dynamically halanced. To protect bearings from overload, bolts must be tensioned to whos shown.

Belt and sheave information is a fraction of load factors to 1.4 and wrap angles from 140° to 190°.

Engine speed in the speed of the amatic in sheare of the two shears system.

Standard belt specifications, where used, are from the recommended Engliseering Standards of the Mechanical Power Transmission descending and the Rubber Manufacturers Association, Inc. Other standards exist for certain drives for submodified, agricultural and oil field machinery.

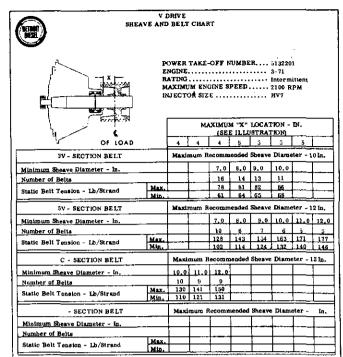
11-1-083-53-2
11-1-88

		RIVE										
SHEAVE AND BELT CHART												
	NGINE ATING		OFF N			3-71 Cantin	ious					
with the state of		MAXIMUM "X" LOCATION - IN. (SEE !LLUSTRATION)										
OF LOAD		5	5	5	5	5	5	5	5			
3V - SECTION BELT		Maximum Recommended Sheave Diameter - 10 In.										
Minimum Sheave Diameter - In.		6.0	7,0	8,0	9,0	10.0						
Number of Belts		16	13	12	10	9						
Static Belt Tension - Lh/Strand	Max.	7 <u>6</u> 39	80 62	83 65	85 67	87 69						
5V - SECTION BELT		Maximum Recommended Sheave Diameter - 14In,										
Minimum Sheave Diameter - In.		7.0	8,0	9,0	10.0	11.0	12,0	13.0	14,0			
Number of Belts		5	7	6	ā	5	4	4	4_			
Static Belt Tension - Lb/Strand	Max. Min.	132	146	158 126	167	175	182	157	191			
B - SECTION BELT		Maxi	mum R	ecomn	ended	Sheave	Diame	ter - 8	In.			
Minimum Sheave Diameter - In.		8.0										
Number of Belts		12										
Static Belt Tension - 1b/Strand	85 68											
C - SECTION BELT		Maxi	mum F	lecomn	iended	Sheave	Diam	ster - 1	4 ln.			
Minimum Sheave Diameter - in.	8,0	9.0	10.0	11.0	12.0	13.0	14.0					
Number of Belts		11	9	7	7	6	6	5				
Static Belt Tension - Lb/Strand	May						162 138	170 144				

Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°.
Engine speed is the speed of the smaller sheave of the two-sheave system.

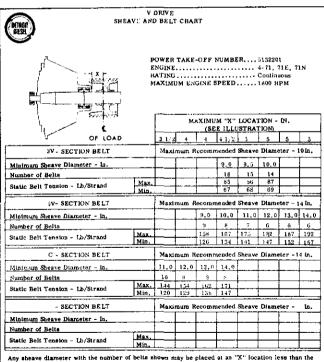
Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, inc. Other standards exist for certain drives for automotive, agricultural and oil field machines. D1-1033-11-5



Any sheave diameter with the number of belts shown may be pisced at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

eit and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°. Engine speed is the speed of the smaller sheave of the two-sheave system.

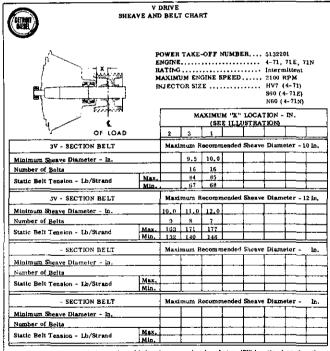
Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, inc. Other standards exist for certain drives for automotive, agricultural and oil field machinery, DI-103-11-6
11-1-68



Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°. Engine speed is the speed of the smaller sheave of the two-sheave system.

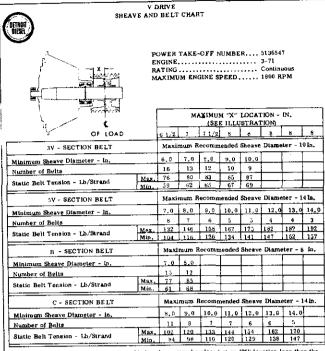
Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, inc. Other standards exist for certain drives for automotive, agricultural and oil field machinery. D1-1043-00-7



Any sheave diameter with the number of belts shown may be placed at an 'X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°. Engine speed is the speed of the smaller sheave of the two-sheave system.

Standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc. Other standards exist for certain drives for automotive, agricultural and off field machinery, D1-1043-90-8 [1-1-68]



Any sheave diameter with the number of helts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, helts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°.

Engine speed is the speed of the smaller sheave of the two-sheave system.

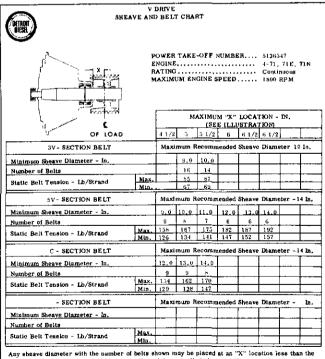
standard belt specifications, where used, are from the recommended Engineering Standards of the Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc. Other standards exist for certain drives for automotive, agricultural and oil field maching. D1-1033-11-4

SHEA	V I	RIVE	CHAR	т				•									
	OWER NGINE, ATING AXIMU NJECTO	M ENC	TNE S	PEED.		3-71 Intern 2100 F	ittent	•									
Strafter W			MA			LOCAT	CION -	IN.									
OF LOAD		5	ţ.	5 1/2	6	6 1/2	7	7 1/2									
3V - SECTION BELT	Maximum Recommended Sheave Diameter - 10 In.																
Minimum Sheave Diameter - In.				7.0	6.0	9.0	10.0										
Number of Belts				16	14	13	11										
Static Belt Tension - Lb/Strand	Max.			76 61	81 64	52 65	85 68										
3V - SECTION BELT		Maximum Recommended Sheave Diameter - 12 In.															
Minimum Sheave Diameter - In.			7.0	6. 9	9.0	10,0	11.0	12.0									
Number of Belts			10	8	7	6	6	5									
Static Belt Tension - Lb/Strand	Max.		128	143	154		171	177									
Static Dett 1 days 257 517 213	Min.		102	114	124	132	140	146	<u> </u>								
C- SECTION BELT		Maxi	mum R	ecomu	ended	Sheave	Diam	ster - 1	2 ln.								
Minimum Sheave Diameter - In.		10.0	11.0	12.0		ļ											
Number of Belts		10	9	8		<u> </u>	ļ	<u> </u>	L								
Static Belt Tension - Lh/Strand	130		151		1	 		-									
- SECTION BELT	- SECTION BELT							Maximum Recommended Sheave Diameter - In.									
Minimum Sheave Diameter - In.						Ţ	\Box	\Box									
Number of Belts	1			Γ		1	\Box	T									
Static Belt Tension - Lb/Strand				.		-											

Any sheave diameter with the number of belts shown may be placed at an 'X' location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°. Engine speed is the speed of the smaller sheave of the two-sheave system.

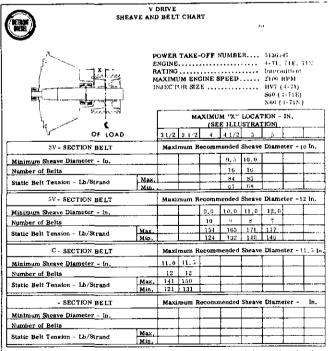
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Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

Belt and sheave information is a function of load factors to 1.4 and wrap angles from 140° to 180°. Engine speed is the speed of the smaller sheave of the two-sheave system.

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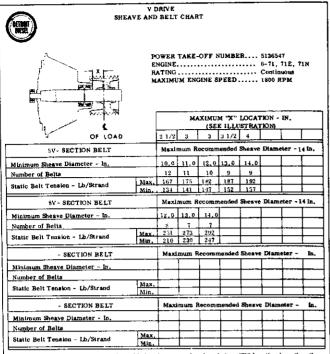
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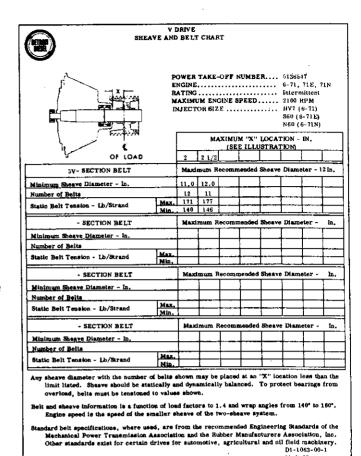
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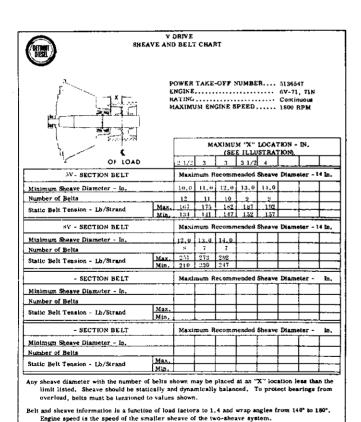


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Mechanical Power Transmission Association and the Rubber Manufacturers Association, Inc.
Other standards exist for certain drives for automotive, agricultural and oil field machinery.

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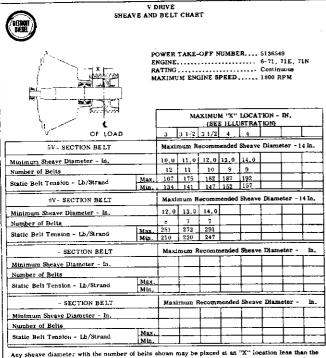
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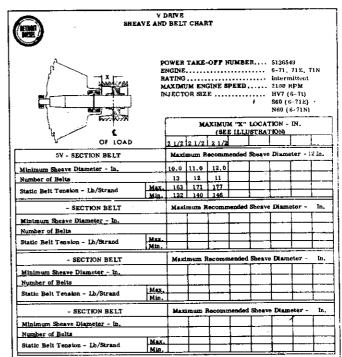


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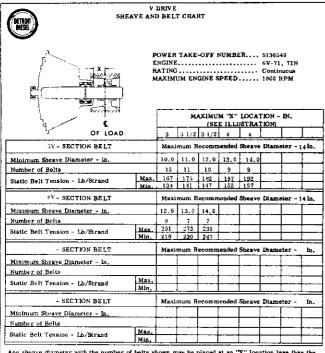
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Other standards exist for certain drives for automotive, agricultural and oil field machinery.

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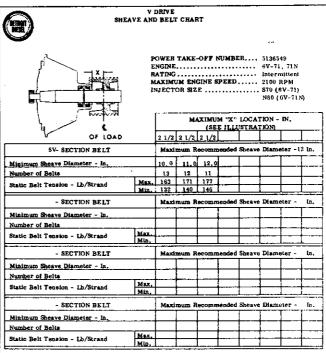
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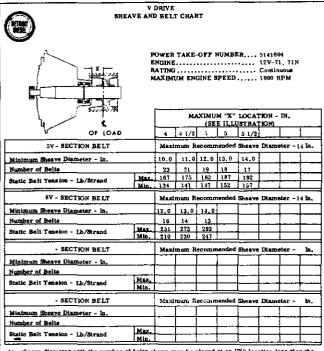
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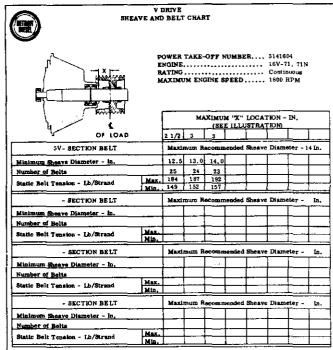


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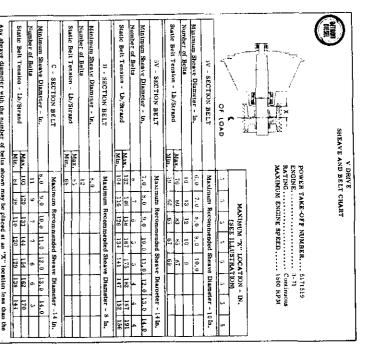
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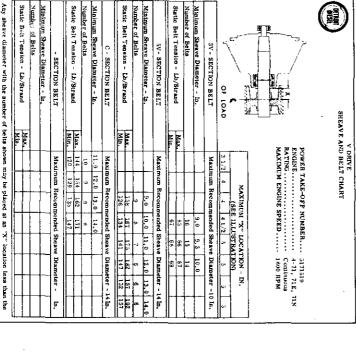
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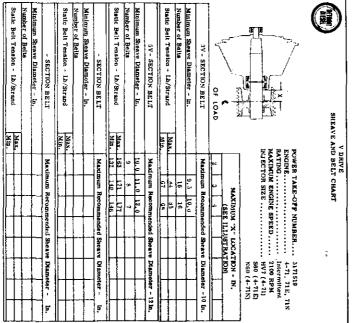
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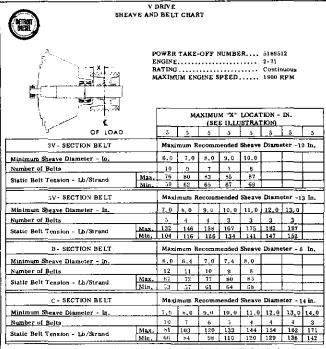
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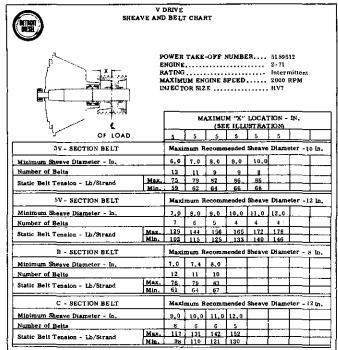
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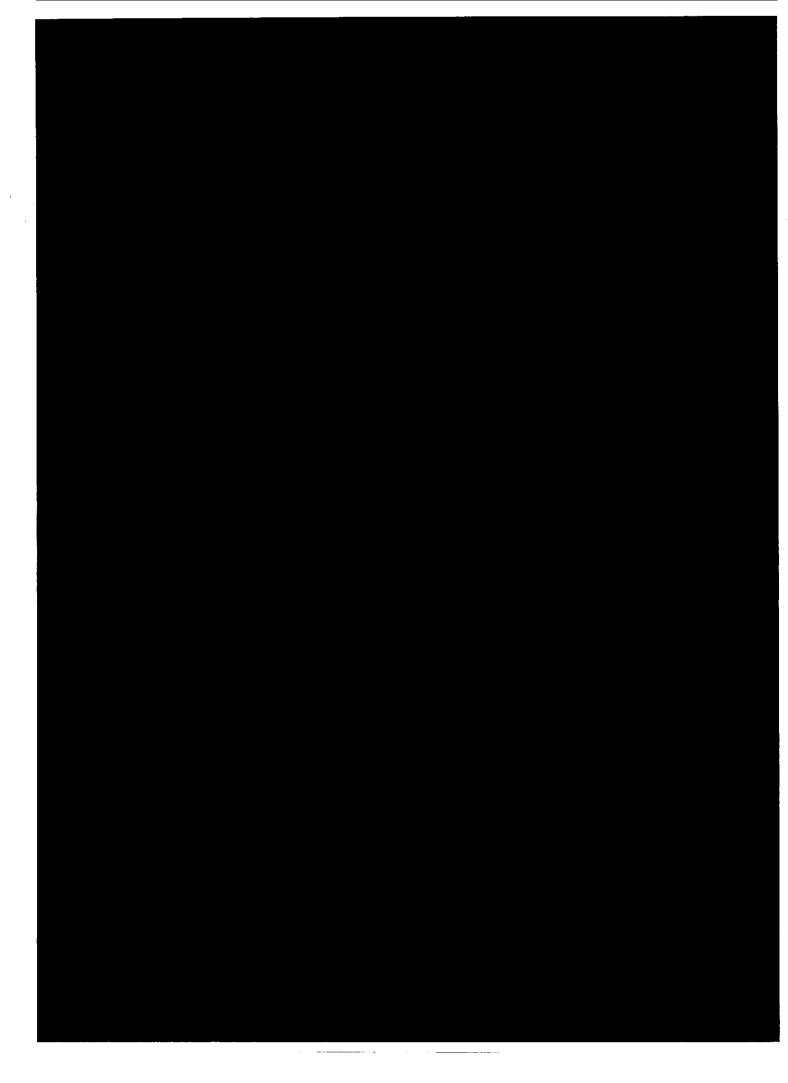
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Any sheave diameter with the number of belts shown may be placed at an "X" location less than the limit listed. Sheave should be statically and dynamically balanced. To protect bearings from overload, belts must be tensioned to values shown.

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Other standards exist for certain drives for automotive, agricultural and oil field machinery,
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BOOK II: ENGINE APPLICATION AND VEHICLE POWER REQUIREMENTS

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SHIFT PATTERN

The shift pattern is a graph showing vehicle speed at any engine RPM in each gear of the transmission and range of the axle. It does not indicate whether the vehicle will be capable of satisfactory performance at these speeds. Each line on the shift pattern represents the only MPH versus RPM relationship for the specific gear; the vehicle will always be operating along one of these lines — never at some intermediate point. The exception would be a vehicle equipped with a torque converter transmission which has no fixed ratio of input to output shaft speed when not in "lock-up."

To determine the geared road speed for a given vehicle, the following information is required:

- 1. Governed engine speed (full load).
- 2. Transmission ratios.
- 3. Auxiliary transmission ratio (if used).
- 4. Rear axle ratio(s).
- 5. Tire revolutions per mile.

Geared road speed formula:

 $MPH = \frac{RPM \times 60}{Total Reduction X tire Revs/Mile}$

Where: MPH = Road speed in miles per hour

RPM = Governed engine speed

Total reduction = transmission ratio(s) X axle ratios.

An examination of the above formula shows all factors to be constant except for the total reduction which is the product of the transmission ratio(s) and the axle ratio(s). The geared road speed in direct gear of the transmission (1:1) in conjunction with a given axle ratio may be determined by using the above formula. The geared road speed in any other gear of the transmission may be determined by dividing the road speed achieved in direct gear of the transmission by the remaining ratios of the transmission as shown in the following example:

EXAMPLE NO. 1: A. 8V-71 at 2100 RPM

- B. 4,11:1 axle ratio
- C. 10:00 x 20 tires (504 revs/mile)
- D. 10 speed transmission

Transmission ratios: 1st 10.05:1, 2nd 7.68:1, 3rd 6.15:1, 4th 4.75:1, 5th 3.63:1, 6th 2.75:1, 7th 2.13:1, 8th 1.58:1, 9th 1.21:1, 10th 1:1.

$$\begin{array}{ll} \text{MPH} \\ \text{(Direct)} &=& \frac{2100 \times 60}{4.11 \times 504} = 60.8 \\ \\ \text{MPH} \\ \text{(9th)} &=& \frac{60.8}{1.21} = 50.3 \\ \\ \text{MPH} \\ \text{(8th)} &=& \frac{60.8}{1.58} = 38.5 \\ \\ \text{ETC.} \end{array}$$

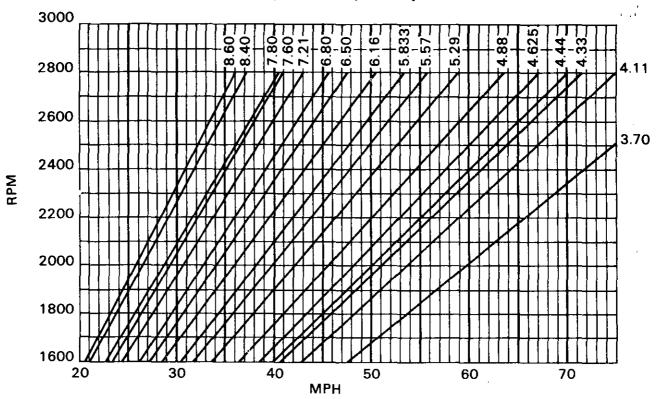
Arranging this data in columns will simplify the procedure and reduce the chance of error.

TRANS	SMISSION	MPH
Gear	Ratio	4.11:1 Axle
1	10.05	6.50
2	7.68	7.92
3	6.15	9.90
4	4.75	12.80
5	3.63	16.70
6	2.75	22.10
7	2.13	28.60
8	1.58	38.50
9	1.21	50.30
10	1.00	60.80

If a multi-speed axle is used, the procedure remains the same as above, i.e., determine the geared road speed in direct with each axle ratio and divide by the remaining transmission ratios. An auxiliary transmission can be treated the same as a multi-speed axle. Determine the geared road speed in direct of the main transmission and whatever combination of axle ratios or auxiliary transmission ratios used and divide by the remaining ratios of the main transmission. Shown in Figs. 1, 2, 3 are graphs of direct gear MPH at various engine speeds for common axle ratios in conjunction with various tire sizes. These graphs eliminate the need to calculate the direct gear road speed.

The road speeds calculated are plotted on the shift pattern paper at the governed engine speed Fig. 4. A straight line is drawn from the zero MPH and RPM point to each of the speed points (MPH). A vertical line drawn from the speed points will intersect the next higher geared speed line at the transmission or axle shift point.

GEARED ROAD SPEED CURVE 8.25 X 20 OR 9.00 X 22.5 TIRES (542 REV/MILE)



GEARED ROAD SPEED CURVE 9.00 X 20 OR 10.00 X 22.5 TIRES (520 REV/MILE)

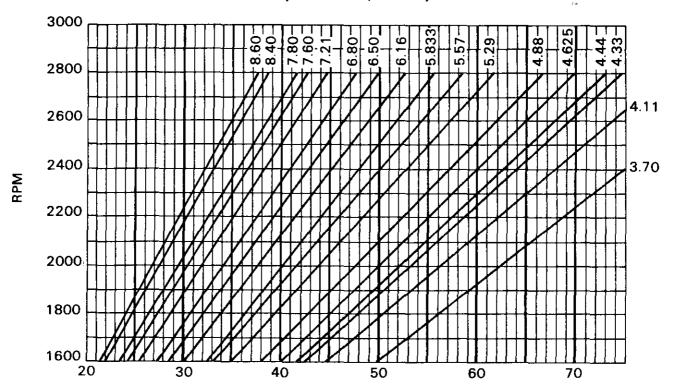
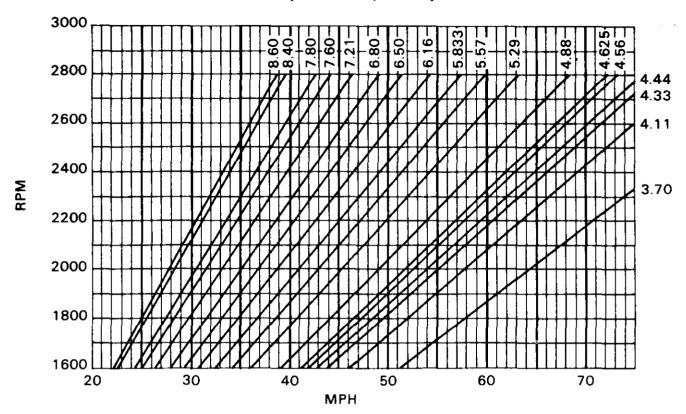


FIGURE 1—
GEARED ROAD SPEED CURVE

GEARED ROAD SPEED CURVE 10.00 X 20 OR 11.00 X 22.5 TIRES (504 REV/MILE)



GEARED ROAD SPEED CURVE 10.00 X 22 OR 11.00 X 24.5 TIRES (480 REV/MILE)

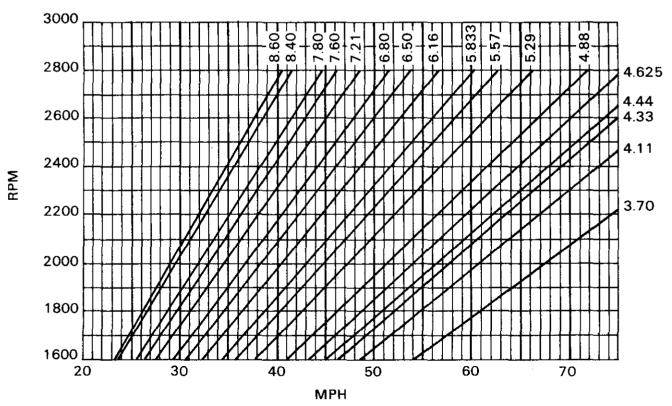
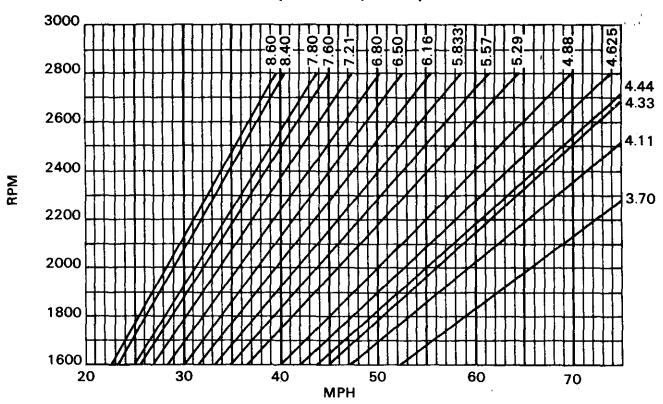


FIGURE 2
GEARED ROAD SPEED CURVE

GEARED ROAD SPEED CURVE 11.00 X 20 OR 12.00 X 22.5 TIRES (492 REV/MILE)



GEARED ROAD SPEED CURVE 11.00 X 22 OR 12.00 X 24.5 TIRES (470 REV/MILE)

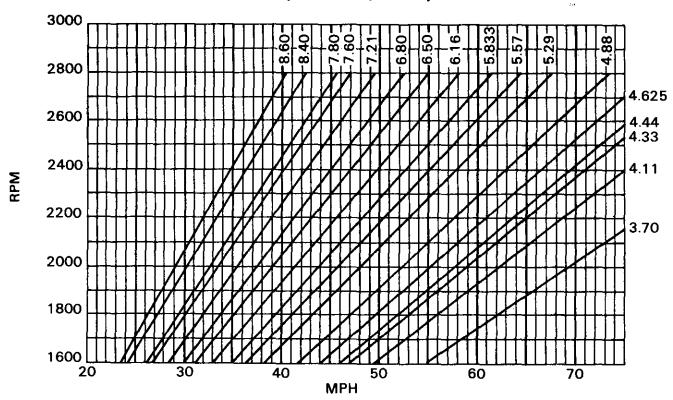


FIGURE 3
GEARED ROAD SPEED CURVE

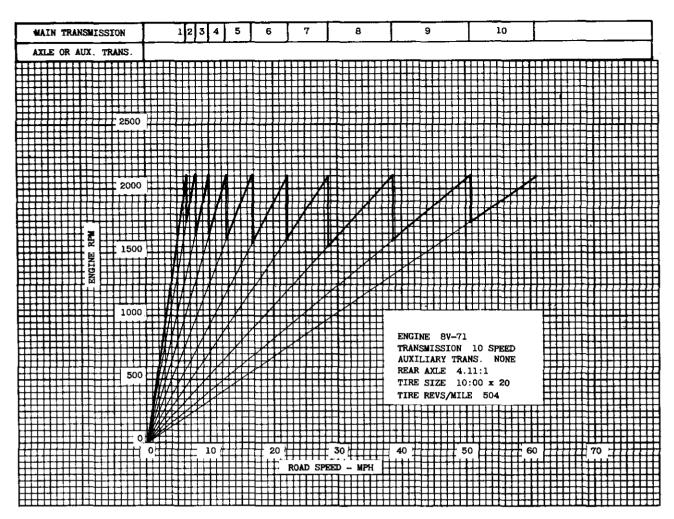


FIGURE 4
SHIFT PATTERN CHART

INTERPRETING THE SHIFT PATTERN

After drawing a shift pattern what can be learned from it? The example chosen was an 8V-71 (N55) at 2100, 10 speed transmission 4.11:1 axle and $10:00 \times 20$ tires. The shift pattern shows a maximum geared speed of 60.8 MPH at 2100 RPM in 10th gear. At 50 MPH the engine is turning at 1725 RPM in 10th or 2100 RPM in 9th.

With a given load, frontal area, drive line efficiency, etc. will the vehicle actually be able to operate as the shift pattern indicates? At the 50 MPH cruising speed will the truck always remain in 9th gear or is it possible to operate some of the time in 10th gear at a reduced engine speed? To answer these and other questions of performance, the shift pattern must be used in conjunction with the engine power curves and the vehicle power requirement curves.

7

VEHICLE POWER REQUIREMENTS

A basic requirement in selecting a truck or a truck-tractor for a given job is performance. Broadly speaking, performance provides a measure of the truck's ability to move a load economically under varying conditions of operation. The ability of a truck to move a load depends on the engine, transmission, and axle ratio matched to the expected gross weight, road conditions and road speed. The engine must provide adequate power to maintain the desired road speed with the expected load, the transmission must provide an adequate selection of speeds to permit the engine to operate within its optimum range and the axle ratio must be compatible with the required road speed. Deviation from these requirements can result in less than optimum fuel mileage, shortened engine life and/or frequent driver complaints of poor performance.

The power requirements for a given vehicle must be known before an engine and driveline can be selected. The power required is the sum of the air resistance, the rolling resistance and the grade resistance. Assuming the road surface to remain constant, the grade resistance may be considered as increased rolling resistance. The separate effect of these variables is discussed under their appropriate heading. The total resistance (the sum of the above), or power to overcome this resistance to movement, can be determined if the following items are known:

- 1. Gross weight (GVW or GCW).
- 2. Maximum cross sectional frontal area.
- 3. Vehicle speed (MPH).
- 4. Road surface.

If the above factors are known, equations for determining the vehicle power requirements can be solved.

While the solution of these equations is not complicated, it is time consuming. To aid in the solution of these problems, vehicle power requirement curves (hereafter referred to as V.P.R. curves) have been prepared which cover several vehicle configurations from city delivery vans to large highway vehicles for turnpike double bottom operations.

A V.P.R. curve is a plot of the power required at the drive wheels (Demand Wheel Horsepower) versus road speed. Each curve is drawn covering specific conditions of grade, load, and frontal area (Sample Fig. 5). The curves are identified by gross load (GCW or GVW) and frontal dimensions (height and width) of the vehicle. The vehicle height shown on the various curves is measured from the road surface to the top of the vehicle. Computed conditions are for still air, 85° F ambient temperature, 500 ft. altitude, and class "A" road surface. Air resistance is based on flat nose trailer or van body with 6 to 8 inch radiused corners and smooth or horizontally ribbed sides. Deviations from these conditions will be discussed later.

The power developed at the rear wheels of a given vehicle may be plotted directly on the V.P.R. curve at the road speeds corresponding to the shift pattern for the vehicle. This results in a graphic picture of the vehicle's performance capabilities throughout its operational range.

The gross engine output will not be available at the drive wheels of the vehicle due to engine parasitic losses and driveline efficiency. Wheel horsepower curves have been prepared for all Detroit Diesel automotive engines which were developed from basic engine curves and reflect the average parasitic power losses from engine accessories such as the cooling fan, the air compressor and the generator. The various Wheel Horsepower curves (Fig. 6 thru 24) also include the driveline efficiency that corresponds to the anticipated driveline of the vehicle.

A truck performance work sheet has been provided to aid in transposing the available wheel horse-power to the V.P.R. curve. The use of this work sheet and the associated material will be discussed in detail beginning on page 24.

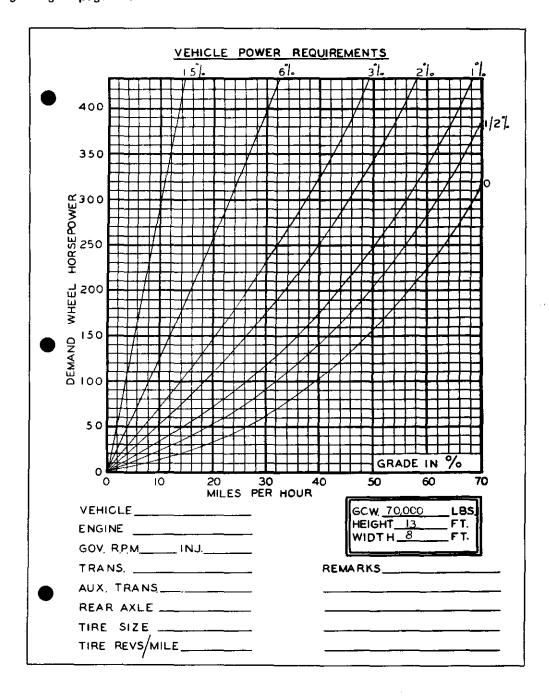
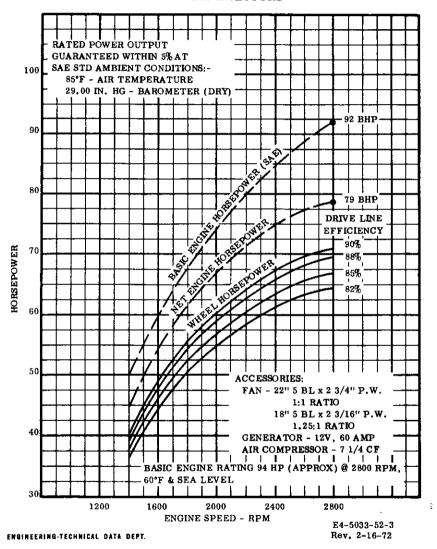


FIGURE 5
SAMPLE VEHICLE POWER REQUIREMENTS CHART

9

3-53 ENGINE — AUTOMOTIVE ADVANCED CAMSHAFT TIMING C45 INJECTORS

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3-53 ENGINE — AUTOMOTIVE ADVANCED CAMSHAFT TIMING C50 INJECTORS

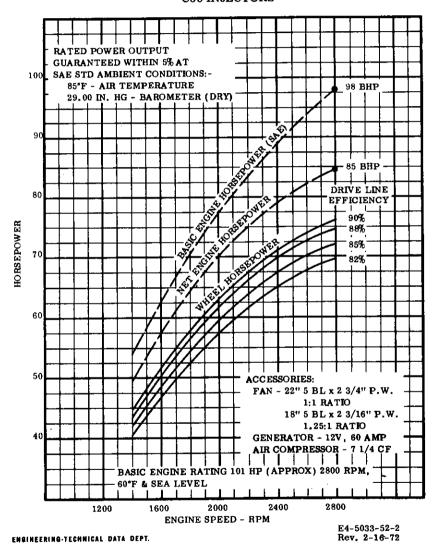
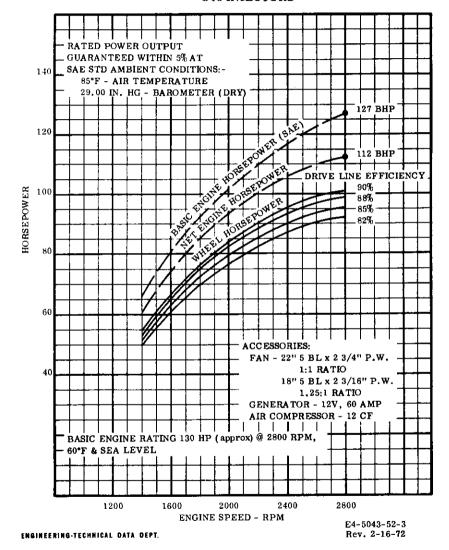


FIGURE 6

FIGURE 7

4-53 ENGINE — AUTOMOTIVE ADVANCED CAMSHAFT TIMING C45 INJECTORS



4-53 ENGINE — AUTOMOTIVE ADVANCED CAMSHAFT TIMING C50 INJECTORS

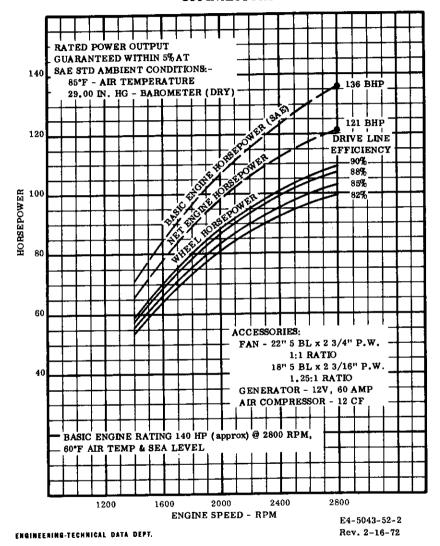
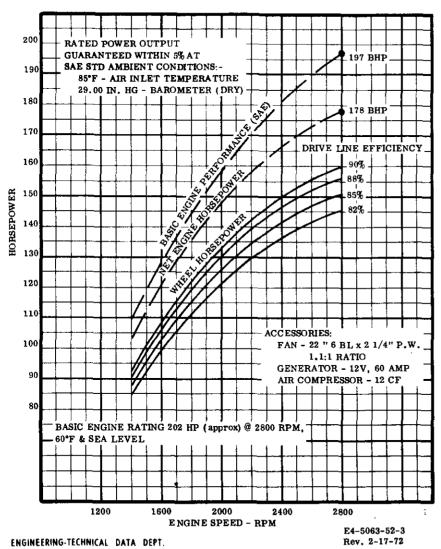


FIGURE 8

6V-53 ENGINE — AUTOMOTIVE ADVANCED CAMSHAFT TIMING C45 INJECTORS



6V-53 ENGINE — AUTOMOTIVE ADVANCED CAMSHAFT TIMING C50 INJECTORS

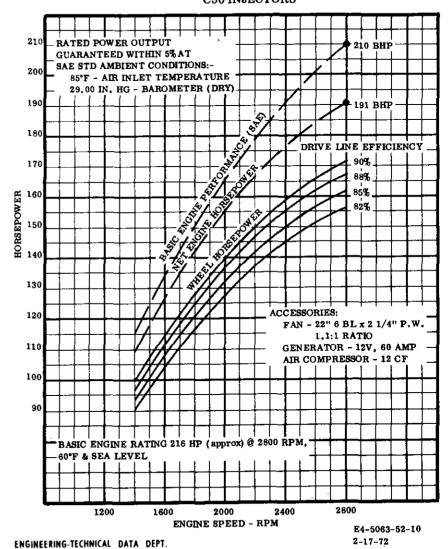


FIGURE 10

4-71 ENGINE C60 INJECTORS RATED POWER OUTPUT . **GUARANTEED WITHIN 5% AT** SAE STD AMBIENT CONDITIONS:-85°F - AIR TEMPERATURE 29.00 IN. HG - BAROMETER (DRY) 140 140 BHP. 130 BHP DRIVE LINE EFFICIENCY 120 HORSEPOWER ACCESSORIES: FAN - 24" 6 BL x 2 3/8" P.W. .95:1 RATIO GENERATOR - 12V, 50 AMP AIR COMPRESSOR - 12 CF BASIC ENGINE RATING 145 HP (APPROX) 2100 RPM, 60°F & SEA LEVEL 1800 2000 2200 1200 1400 1600 **ENGINE SPEED - RPM** E4-1043-52-12

4-71 ENGINE ADVANCED CAMSHAFT TIMING C65 INJECTORS

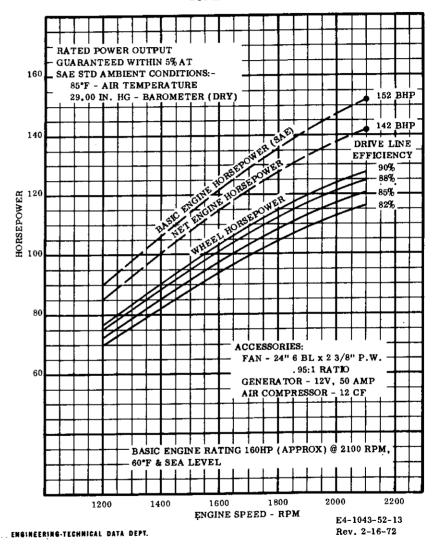


FIGURE 12

Rev. 2-16-72

FIGURE 13

ENGINEERING-TECHNICAL DATA DEPT.

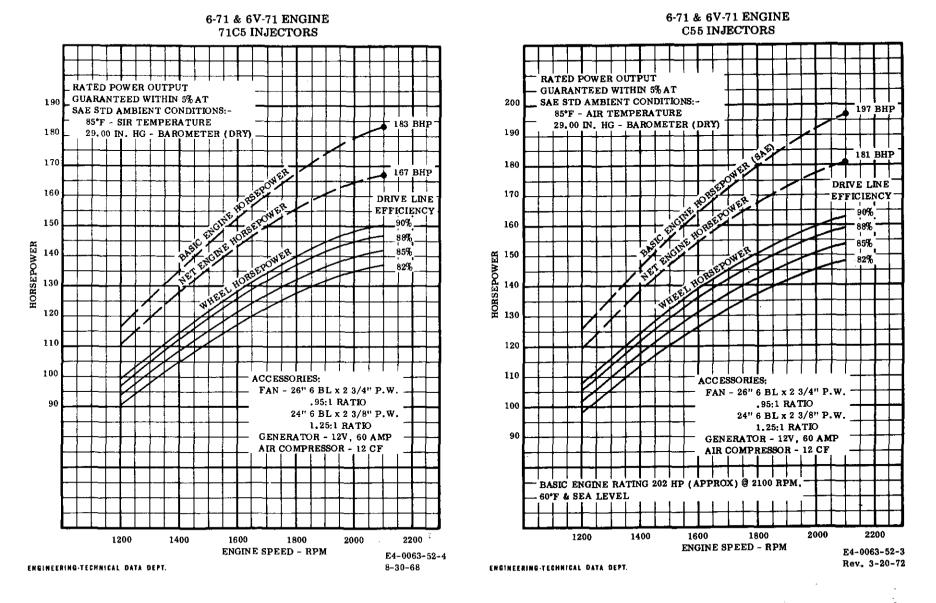


FIGURE 14 FIGURE 15

6-71 & 6V-71 ENGINE C60 INJECTORS RATED POWER OUTPUT 210 GUARANTEED WITHIN 5% AT SAE STD AMBIENT CONDITIONS:-85°F - AIR TEMPERATURE 200 29,00 IN. HG - BAROMETER (DRY) 190 DRIVE LINE 180 90% 170 HORSEPOWER 120 170 130 120 ACCESSORIES: 110 FAN - 26" 6 BL x 2 3/4" P.W. .95:1 RATIO 24" 6 BL x 2 3/8" P.W. 100 1.25:1 RATIO GENERATOR - 12V, 60 AMP AIR COMPRESSOR - 12 CF

BASIC ENGINE RATING 218 HP (APPROX) @ 2100 RPM.

1600

1400

60°F & SEA LEVEL

1200

ENGINEERING-TECHNICAL DATA DEPT.

6-71 & 6V-71 ENGINE ADVANCED CAMSHAFT TIMING C65 INJECTORS

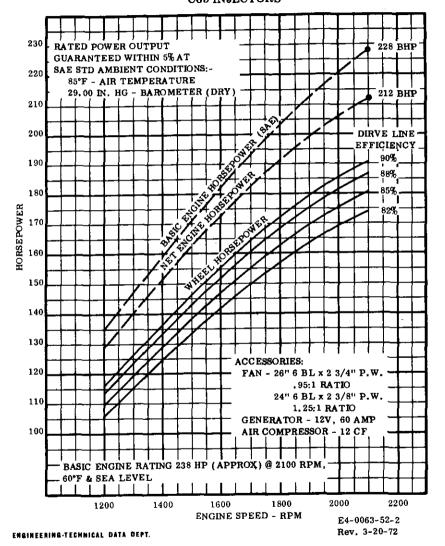


FIGURE 16

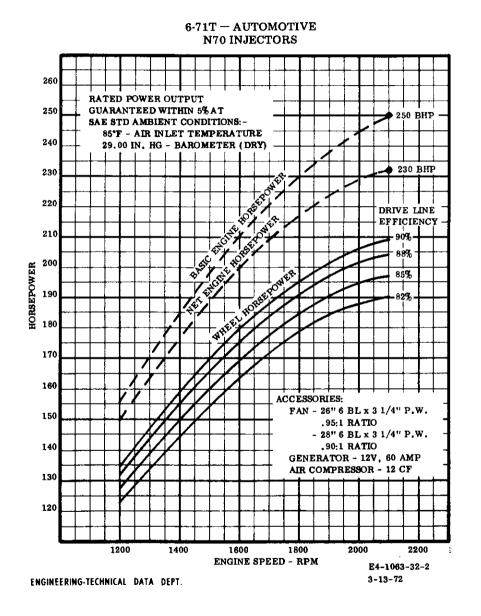
ENGINE SPEED - RPM

1800

2200

E4-0063-52-1

Rev. 3-20-72



6-71T — AUTOMOTIVE N75 INJECTORS

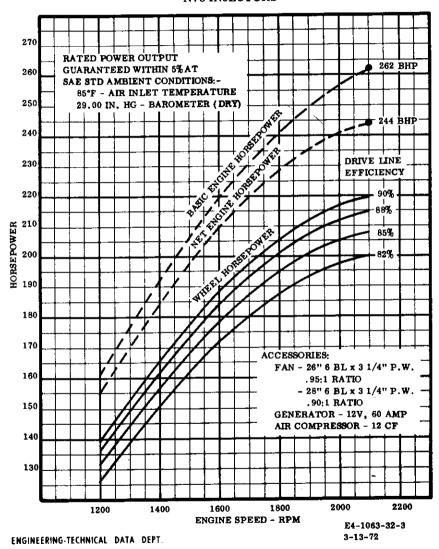
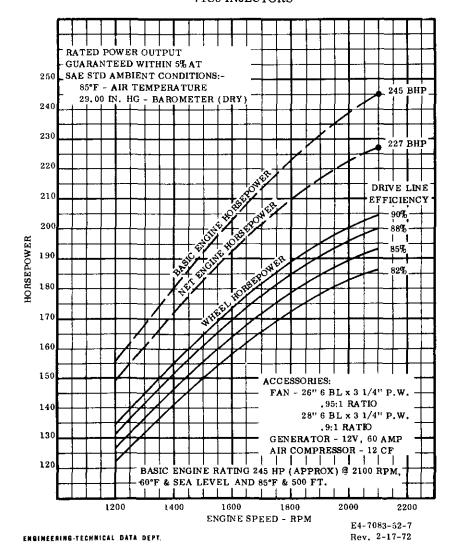


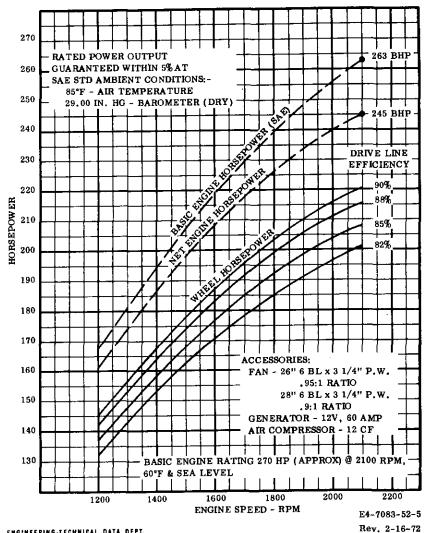
FIGURE 18

FIGURE 19

8V-71 ENGINE — AUTOMOTIVE 71C5 INJECTORS



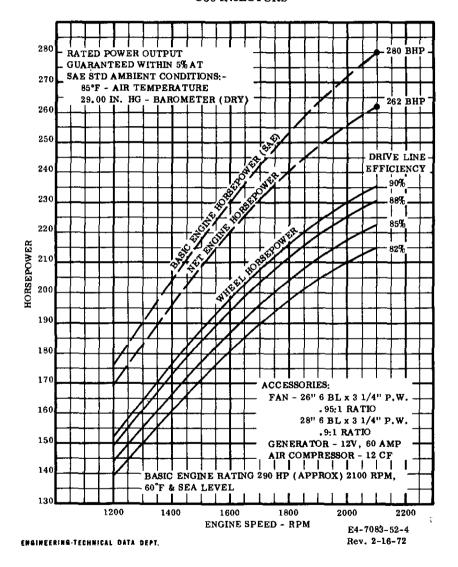
8V-71 ENGINE - AUTOMOTIVE C55 INJECTORS



ENGINEERING-TECHNICAL DATA DEPT.

FIGURE 20

8V-71 ENGINE — AUTOMOTIVE C60 INJECTORS



8V-71 ENGINE ADVANCED CAMSHAFT TIMING C65 INJECTORS

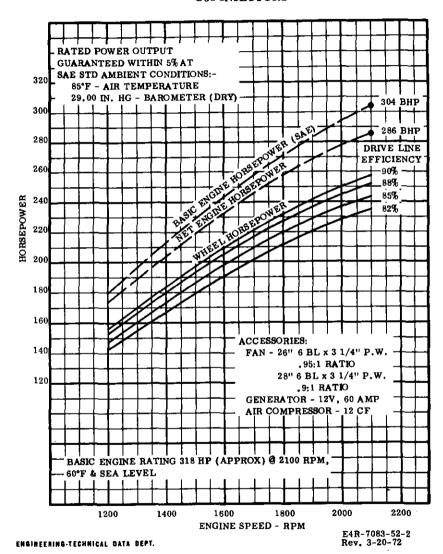
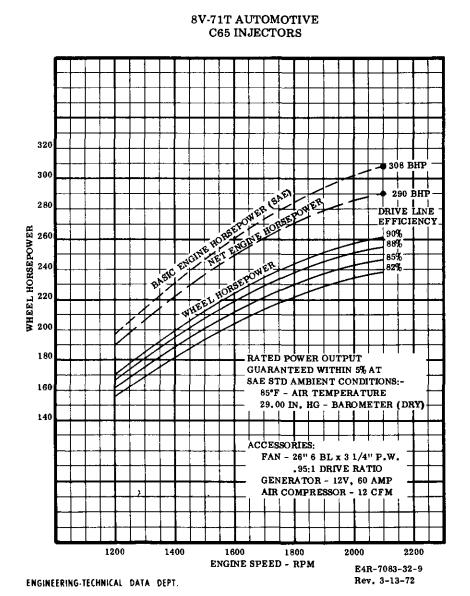
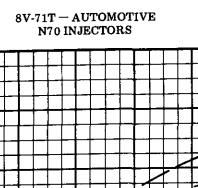


FIGURE 22





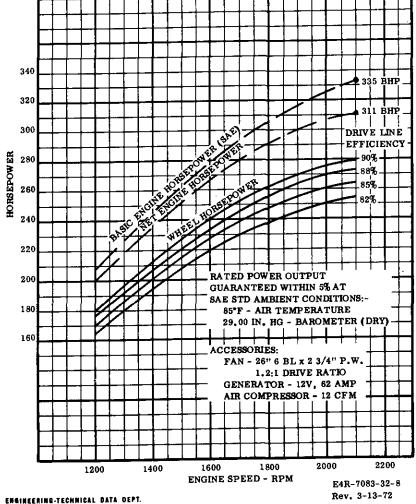


FIGURE 24

FIGURE 25