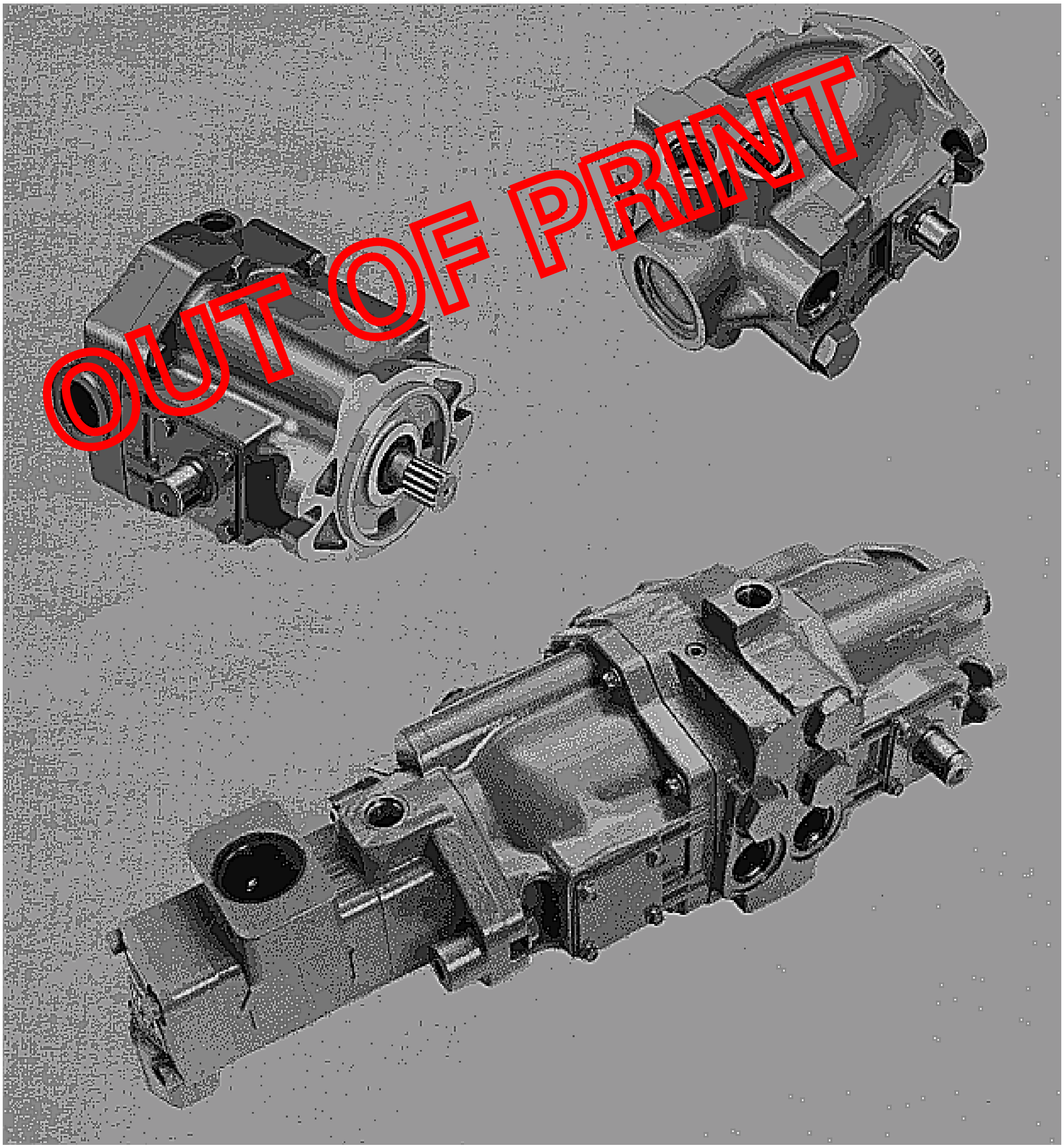
Vickers®

Transmissions

E Series 19 Size Transmission Guide



Forward

This manual was prepared to give the application engineer in the field sufficient engineering information and procedures to make the judgements and calculations necessary for reliable and successful applications of the E Series, 19 size hydrostatic transmissions.

In addition, it attempts to answer the most frequently asked questions that are referred to Vickers pertaining to the application of the 19 size transmission.

There are instances cited in this manual which require Vickers Engineering review and approval. Requests for engineering review and approval should be made through a Vickers representative.

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Section I – General Information

Introduction

The E Series, 19 size transmission was developed to expand the Vickers product line to include a highly reliable, low cost, medium duty transmission for mobile applications. It was designed specifically for the agricultural, construction equipment and material handling industries, to provide greater machine capability and reliability while still maintaining a competitive price.

Application of hydraulic units as hydrostatic transmissions is not new, and Vickers has always been a pioneer in the field. Our successful applications are due largely to the application engineer having a thorough knowledge of the product, its capabilities and limitations, as well as an understanding of the machine on which it will be used. The information contained in this manual is intended to provide the application engineer with reference data and procedures which will enable him to fulfill his responsibility in properly applying the transmission.

General Description

The transmission consists of a pump and motor, both of conventional inline piston design, together with the elements essential to the function of a mobile hydrostatic transmission. In addition, Vickers can provide in the transmission package some additional capacity for auxiliary functions on a machine, depending on the application.

In order to provide for a higher reliability which is increasingly demanded by the mobile market, transmission components were designed for maximum rigidity. This yields low stress levels for maximum life and low deflections for optimum performance.

Figures 1 and 2 show the arrangement of parts and the circuit of a transmission.

Rotating Group

The rotating group is similar to the proven B Series units. State-of-the-art improvements in materials, processes and design have resulted in higher speed and pressure capabilities. The group consists of the cylinder block assembly, nine piston/shoe subassemblies, a spherical washer and shoe return plate. A steel-backed, bronze wafer plate completes the assembly.

Pump

The pump is of over-center construction. The yoke is designed to rotate in either direction from the neutral position; this provides variable flow in either direction in the closed loop transmission circuit. The yoke is positioned manually through the pintle which is an integral part of the yoke casting. Internal, nonadjustable stops that limit the maximum yoke angle to 18°F are machined into the yoke and housing.

The valving essential to closed loop operation is contained within the pump valve block. A supercharge relief valve regulates supercharge pressure and directs the excess supercharge flow through the pump housing for cooling. High pressure crossline relief and replenishing check valves distribute supercharge fluid to the low pressure side of the loop and limit maximum pressure in the high pressure side.

Motors

Motors are similar in construction to pumps and can be either fixed or variable displacement depending upon the requirements of the application. Motors are bidirectional in rotation and are completely reversible depending upon direction of flow.

The variable motor has fixed, nonadjustable, internal yoke stops. Maximum yoke angle is 18° and minimum angle stops may be pre-selected at assembly for various yoke angles, depending upon the application.

Supercharge Pump (See Figure 1)
The closed loop design of the transmission requires that the system be supercharged under all operating conditions. All standard pump configurations provide for supercharge supply from either an integral gear type or a vane type pump mounted externally and driven by the main pump shaft.

Auxiliary Pump (See Figure 2)
On TA19V and TA1919V transmission models, the supercharge vane pump can be supplied with covers that will provide some flow for auxiliary machine functions in addition to that required for supercharge. TA19V2010 double vane pump models provide an independent V20 vane pump for auxiliary functions.

The use of a vane pump for auxiliary functions will always require special application analysis. This is necessary to ensure adequate supercharge pressure and flow.

Customer-Supplied Elements

The following items are supplied by the customer:

1. Reservoir
2. Plumbing including supercharge lines, main loop lines and case drains.
3. Filter
4. Air to oil cooler
5. Control linkages

Specifications and Ratings

Rated input horsepower/rotating group	22.5 hp/1000 rpm
Maximum displacement/rotating group	2.5 CIR
Maximum pump input speed*	3600 rpm
Maximum motor output speed	
@ maximum stroke	3600 rpm
@ reduced stroke	4000 rpm
Minimum pump operating speed	800 rpm

*TA19V models maximum speeds are limited by the maximum allowable speeds of the vane pumps.

Specifications and Ratings (Cont'd.)

Minimum pump operating speed	800 rpm
Minimum input speed	
@ rated pressure	1200 rpm
@ maximum pressure	1800 rpm
Rated pressure – continuous	3000 psig
Maximum pressure – intermittent	5000 psig
Minimum supercharge pressure	75 psig
Maximum case pressure	20 psig
Minimum supercharge pump inlet pressure (TA19)	10 in hg Vacuum
Minimum vane pump inlet pressure	5 in hg. Vacuum
Maximum operating loop temperature	
continuous	180°F
intermittent	220°F
TA19 supercharge pump Displacement (SAE)	.91 CIR

Circuit Schematics

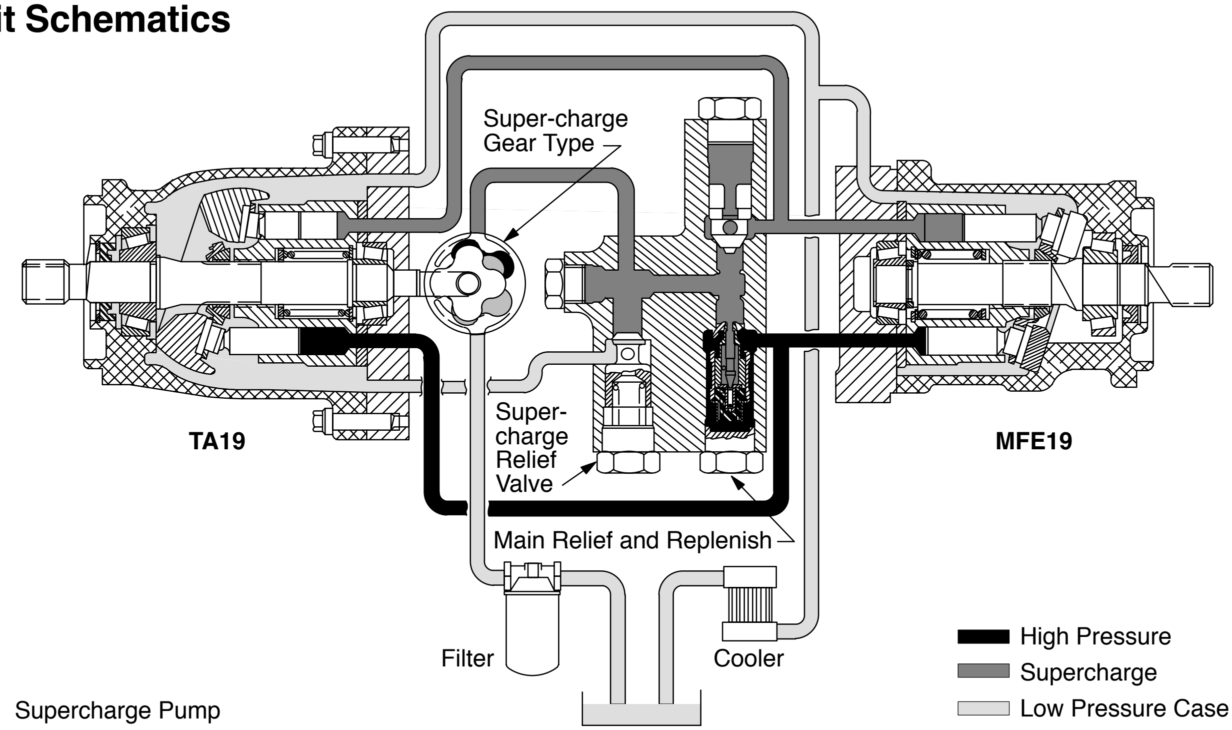


Figure 1. Supercharge Pump

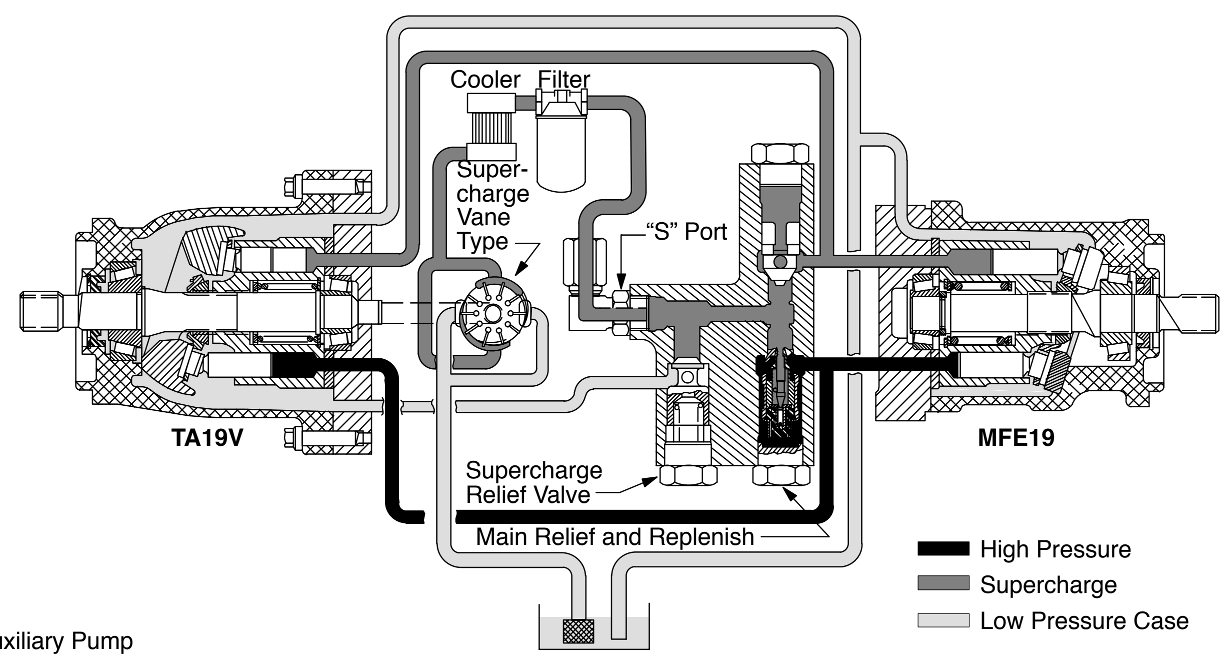


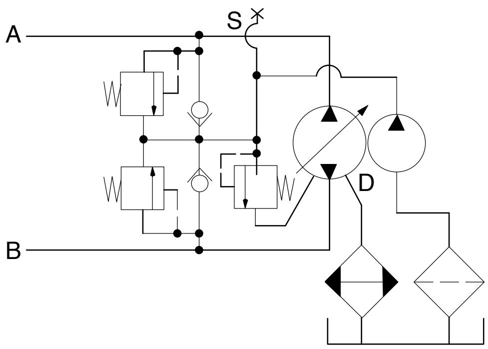
Figure 2. Auxiliary Pump

Transmission Family

The E Series, 19 size transmission family consists of various basic models with standard options of both pumps and motors. Judicious application of the models available enables Vickers to provide economic transmissions for a wide range of applications. (See Table 1.)

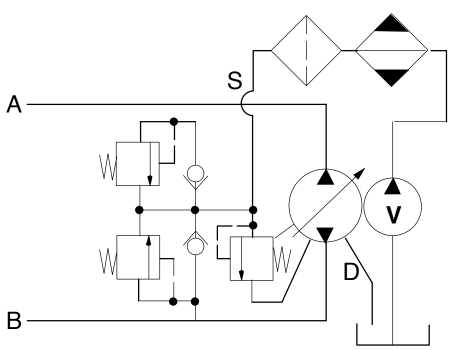
Three basic 19 size transmission components are available, as described below: a variable displacement over-center pump, a fixed displacement bidirectional motor, and a variable displacement bidirectional motor.

TA19 Pump



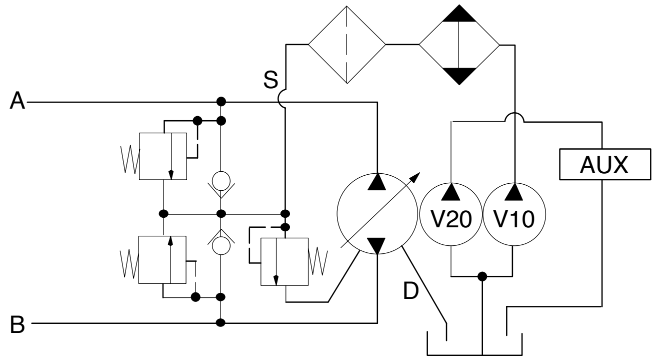
The TA19 is a single piston pump with integral gear type supercharge pump. This is the basic transmission pump and provides the simplest and most economical package when combined with an MFE19 or MVE19 motor. It is intended for single path transmissions which require no auxiliary functions.

TA19V Pump



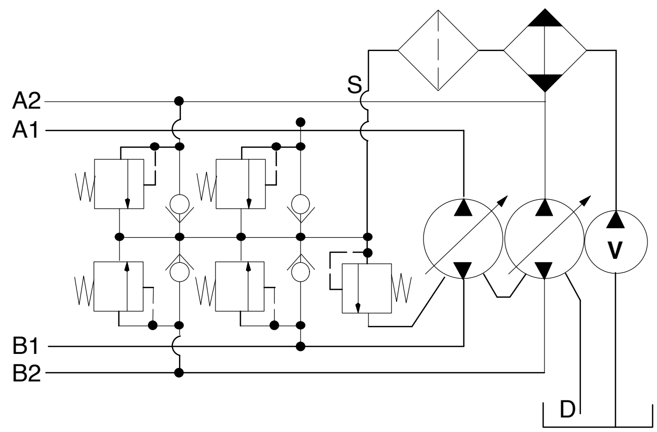
The TA19V is a single piston pump with a vane pump driven from the main shaft for supercharge. It is intended for use in multiple motor applications or applications using motors of another manufacturer, in which supercharge demand exceeds that of a TA19 supercharge pump. The vane pump can be provided with covers that can use vane pump flow in excess of supercharge requirements for auxiliary function. Models can be provided with either a V10 or V20.

TA19V2010 Pump



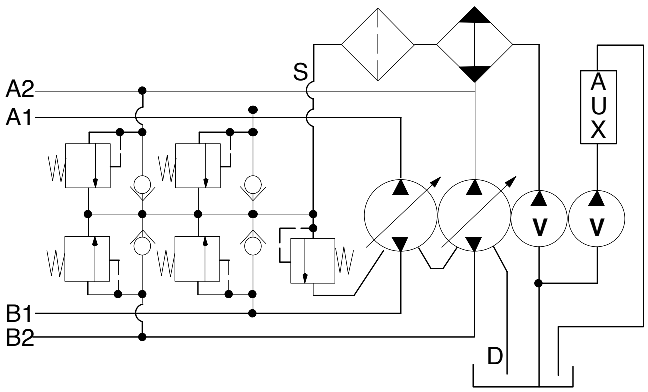
The TA19V2010 is a single piston pump with a V2010 driven from the main pump shaft. The V10 is primarily for transmission supercharge, although surplus flow over that required for supercharge can be used for auxiliary function if properly applied. The V20 is independent of the transmission circuit and is for auxiliary function only.

TA1919V Pump



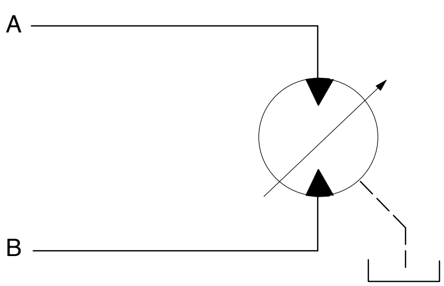
The TA1919V is a double piston pump with vane pump. It consists of two 19 size transmission pumps and a vane pump mounted inline and driven from a single shaft. This unit is intended primarily for dual path applications such as windrowers and skid steer loaders which require both propel and steering functions. It is attractive because it offers multiple pump capability from a single drive pad. A vane pump, either V10 or V20, provides supercharge and can also be used for some auxiliary functions.

TA1919V1010 Pump



The TA1919V1010 is a double piston pump with double vane pump. It consists of two 19 size transmission pumps and a V1010 vane pump mounted inline and driven from a single shaft. It is intended primarily for dual path transmissions which require an independent source of flow for auxiliary functions. The cover end pump is for these auxiliary functions. The shaft end vane pump is used only for transmission supercharge and has a 22 l/min (6 USgpm) ring with a maximum pressure rating of 35 bar (500 psi).

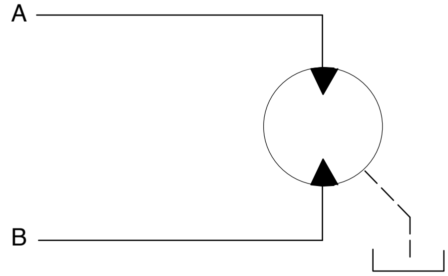
MVE19 Motor



The MVE19 is a variable displacement piston motor for application with a 19 size pump. It is intended for use in those applications requiring an extended speed range. Minimum stroke is established by internal, nonadjustable mechanical stops. The motor is also available with the shaft extended through the valve block for installation of a vehicle brake. This model is designated MVE19X.

The MVE19 is available with either manual or pilot pressure control of displacement. The pilot control models require system pressure to fully stroke the yoke, and units are available with the yoke normally in the full stroke or minimum stroke condition.

MFE19 Motor



The MFE19 (or MFE15) is a fixed displacement piston motor for application with a 19 size pump. It can be used in all transmission circuits within the E Series rating. The motor is available with a through shaft extension intended for installation of a vehicle brake.

Model	Standard Options	
TA19	None	
TA19V	TA19V10 TA19V10D TA19V10F TA19V10P TA19V20	TA19V20V TA19V20P TA19V2010 TA19V2010F TA19V2010P
TA1919V	TA1919V10 TA1919V10D TA1919V10F TA1919V10P	TA1919V1010 TA1919V20 TA1919V20F TA1919V20P
MVE19	MVE19	MVE19X
MFE19	MFE19	MFE19X
MFE15	MFE15	MFE15X

Table 1

The above components can be connected to form three fundamental transmission arrangements, of which there are many variations, to accommodate a variety of applications.

1. **Single Path with Single Motor** (Figure 3)
   Consists of a single over-center pump and a single bidirectional motor, either fixed or variable displacement, connected in a closed loop circuit.

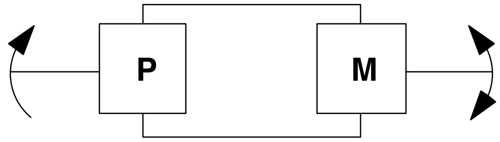


Figure 3. Single Path, Single Motor

2. **Single Path with Multiple Motors** (Figure 4)
Consists of a single over-center pump and two or more motors connected in parallel. Motors may be fixed or variable displacement or a combination of both.

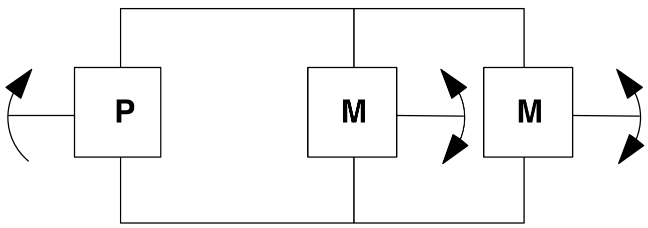


Figure 4. Single Path, Multiple Motors

3. **Dual Path** (Figure 5)
Consists of two single path transmissions with both pumps driven by the same power source. Each transmission is connected to a drive wheel or track and is independent of the other. Provides both propel and steering functions within the transmission. Independent control of each drive wheel permits spin turns about the vertical axis.

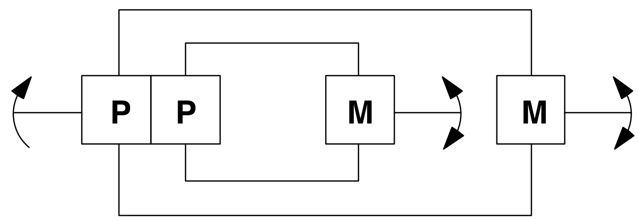


Figure 5. Dual Path

Section II – Application Review

Guidelines

The purpose of an application review is to make certain that the transmission is capable of performing all of the necessary functions of the machine within the rating of the unit, with maximum safety and reliability and at minimum cost to the customer. An orderly review following the procedures shown and using the information contained in this manual will minimize the risk of misapplication.

It must be understood that a review of this nature is only as good as the input data upon which it is based. As a vehicle design evolves, the review must be updated based upon new input. Very few machines reach production with the same parameters as the original development concept. The following items must be considered:

1. Transmission application value vs. vehicle application value
2. Available horsepower vs. rated horsepower
3. Maximum tractive effort
4. Maximum vehicle speed (FLG)
5. Supercharge requirements
6. Auxiliary requirements
7. Circuit analysis:
   a. supercharge
   b. auxiliary
   c. main loop
   d. case drain
   e. filtration
   f. cooling
   g. reservoir
8. Bearing life and shaft stress
9. Installation requirements

It is essential that the transmission arrangement is thoroughly understood. It is generally useful to make a simple block diagram of the arrangement which will enable the application engineer to visualize it, so that all aspects are considered. An example of a block diagram for a dual path transmission drive line is shown in Figure 6.

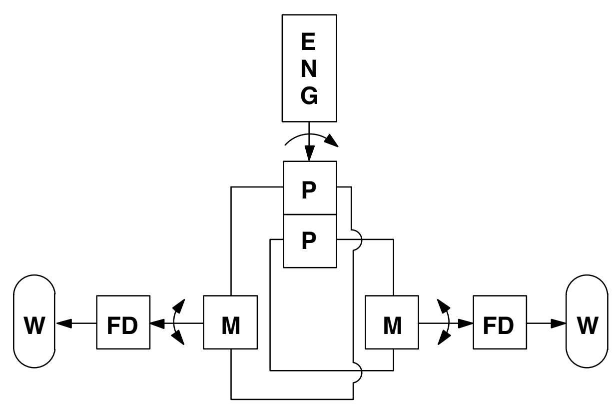


Figure 6. Block Diagram, Dual Path Transmission

The diagram shows that engine output is divided between two pumps, each driving a wheel. It is a reminder of the effects of vehicle weight distribution, engine power distribution, and torque split between the two drive wheels.

Horsepower Distribution

In dual path applications, the power going to the two pumps is rarely equally divided because of vehicle dynamics, weight distribution, terrain and steering requirements. For application analysis purposes, when calculating maximum tractive effort or maximum pressure that can be obtained, assume that the maximum horsepower to one pump is 80% of the total available to the two pumps.

Weight Distribution

Mobile machines are required to function under terrain conditions that will cause unequal weight distribution on the drive wheels. We normally assume that 60% of the total weight on the drive wheels can be on one side. There are some machines, because of their nature that will exceed this value.

Some machines can operate under conditions in which the effective weight on the drive wheels can exceed the static loaded weight. Examples are the effect of down draft of a plow, and the reaction force of an endloader bucket breaking out a load while being driven into the pile. This effect is sometimes expressed as coefficient of slip and can be greater than 1, as much as 1.15.

Cross Port Relief Valves

Cross port relief valves in a closed loop transmission circuit are intended only to protect the pump, motor and final drive from overpressure. When relief valves are actuated in a closed loop circuit, a relatively small volume of fluid at high flow rate is recirculated across the valve, resulting in a rapid fluid temperature rise. Theoretically, fluid temperatures can reach astronomical values in just a few seconds of operation. The cross port relief valves are intended for intermittent transient service only and are not to be used to regulate system pressure.

For example, the relief valve should not be used to regulate input power to prevent engine stall in normal modes of operation. This should not be construed as meaning that, in an application using a variable displacement motor at reduced displacement for road range, the motor torque required to accelerate the vehicle should not result in a pressure exceeding the relief valve pressure setting. In this case, the relief valve is doing precisely what it is intended for: protecting the pump, motor, hoses and drive train against overpressure during an intermittent condition of limited duration.

Good application practice is to size the transmission so the maximum torque requirement (i.e. wheel slip) results in a pressure below the relief valve setting. Applications using variable displacement motors nearly always require relief valves set at 5000 psi because wheels may not slip at reduced displacement.

For most applications, relief valves should be set at the maximum pressure capability of pump or motor, whichever is lowest. For example, a customer may select a motor of another manufacturer which has a maximum pressure rating of 4000 psi. The relief valves should be set at 4000 psi to protect this motor.

Some applications using fixed displacement motors, if properly analyzed and thoroughly tested in the prototype stage, may not require relief valves. The analysis would consider the heaviest machine, fully equipped with maximum payload, as well as the worst condition of weight distribution on the drive wheels. If wheels will slip below 5000 psi at the highest slip coefficient that can be encountered, application without relief valves can be considered.

During prototype testing, these same conditions are to be duplicated to verify calculations. Such applications are subject to Vickers engineering analysis and approval.

Relief valve settings are from 3000 to 5000 psi, in 500 psi increments. Settings below 3000 psi are not available. Those applications requiring relief valve settings below 3000 psi should have external cross port relief valves added to the circuit. For these applications, Vickers can provide transmission pumps without relief valves.

Series Operation of Motors

Connection of motors in series, while not absolutely prohibited, requires Vickers engineering application review and approval. Application of motors in series is often a very attractive approach to the solution of a problem. However, upon thorough analysis, the disadvantages nearly always outweigh the advantages and usually a better way can be devised. The following points must be considered in applying motors in series:

1. **Overpressurization**

   The pressure differential (ΔP) across motors connected in series has a cascading effect, and the first motor in the series will experience the highest total pressure. For application of motors in series, the sum of the inlet and outlet pressures for any motor in the series must not exceed the pressure rating of the motor (e.g. 3000 psig continuous).

   The ΔP across one motor for any number of motors in a series can be calculated as follows:

   $$\Delta P = \frac{\text{Rated Pressure}}{N + (N - 1)}$$

   where N = number of motors in series.

   For example, if two motors are connected in series, the inlet pressure of the first motor of 2000 psi plus the outlet pressure of 1000 psi equals the 3000 psi rating of each motor.

   Therefore, per the formula $3000 \div [2 + (2 - 1)]$, the maximum ΔP across each motor will be 1000 psi for continuous operation.

2. **Cavitation**

   Cavitation will result in any application where the motor shafts are not completely independent of each other. Leakage of the upstream motor subtracts from the inlet flow to the downstream motor, which then tends to turn slower than the upstream motor. In traction drives, the condition is aggravated in steering situations because differential action between the two wheels does not exist. Unequal tire wear, tire inflation and load distribution can cause variation in the loaded radius of the wheel, again affecting motor speed and flow demand. The tendency to cavitate can be alleviated by the addition of a replenishing check valve between motors.

3. **Increased Volumetric and Mechanical Losses**
Overall efficiency will suffer because of the double pressure loading of the upstream motor(s). Volumetric losses for a motor connected in series will be equivalent to the sum of the leakage of one motor at inlet pressure plus one motor at outlet pressure. Mechanical losses are even more pronounced. Laboratory test results show that they are equivalent to the torque loss of one motor at inlet pressure plus the torque loss of a pump operating at the same pressure as motor outlet pressure.

Transmission Sizing

Definition of Terms

Transmission	As used in this manual, refers to one or more variable displacement, over-center hydraulic pumps driven by a prime mover and connected to one or more bidirectional hydraulic motors in a closed loop circuit. The transmission transmits power to and provides a means of control of speed and direction of the live axle of the vehicle.
Gross Vehicle Weight	The total weight on the machine including operator, accessories, attachments and payload.
Weight on Drive Wheels	That portion of the gross vehicle weight supported by the driving wheels. It may also include the effect of downdraft due to external loads.
Tractive Effort (or Rim Pull)	The total force available to propel the vehicle.
Rolling Resistance	That portion of tractive effort required to overcome friction and move the vehicle.
Drawbar Pull	The force available to pull a load; equals tractive effort minus rolling resistance.
Coefficient of Rolling Resistance	A constant which expresses the relative resistance to rolling over a given surface. In this manual it is expressed as lbs. per 1000 lbs. gross vehicle weight.
Gradeability	The slope a vehicle is capable of ascending without slipping the wheels or exceeding the rating of the transmission, expressed in percent.
Percent Grade	Tangent of the grade angle $\times$ 100.

Symbols Used In Calculations

Symbols	Definition	Expressed in
a	Vehicle acceleration	FT/SEC^2
C_r	Coefficient of rolling resistance	–
C_s	Coefficient of slip	–
DBP	Draw bar pull	LBS
Dm	Motor displacement	CIR
D_p	Pump displacement	CIR
D_s	Supercharge pump displacement	CIR
E_{fd}	Efficiency, final drive	Percent
E_o	Efficiency, overall	Percent
E_t	Efficiency, torque	Percent
E_v	Efficiency, volumetric	Percent
FD	Final drive ratio	–
G	Gradeability	Percent
g	32.2	FT/SEC^2
GVW	Gross vehicle weight	LBS
HP	Horsepower	–
Ne	Engine speed	RPM
N_m	Motor speed	RPM
N_p	Pump speed	RPM
N_w	Wheel Speed	RPM
P_c	Supercharge pressure	PSI
P_d	Drain (case) pressure	PSI
ΔP	$P_s – P_c$	PSI
P_s	System pressure	PSI
Q_c	Supercharge flow	GPM
Q_d	Drain (case) flow	GPM
Q_m	Motor inlet flow	GPM
Q_p	Pump outlet flow	GPM
R_L	Loaded radius	IN
RR	Rolling resistance	LBS
t	Time	SEC
TE	Tractive effort	LBS
T_e	Engine torque	LB FT
T_m	Motor output torque	LB IN
T_p	Pump input torque	LB IN
T_w	Wheel torque	LB IN
V	Velocity	FT/SEC
W_w	Weight on drive wheel(s)	LBS

Formulae

Horsepower

$$HP = \frac{\text{Torque} \times \text{RPM}}{63025}$$

$$HP = \frac{\text{GPM} \times \text{PSI}}{1714}$$

$$^{*}HP = \frac{\text{Disp.} \times \text{PSI} \times \text{RPM}}{396000}$$

Torque

$$^{*}T = \frac{\text{Displ.} \times \text{PSI}}{2\pi}$$

*These formulae are useful for theoretical calculations. If used to obtain actual values, care must be taken in selection and efficiencies.

Flow

$$* \text{GPM} = \frac{\text{Displ.} \times \text{RPM}}{231}$$

Flow Through Orifice

$$Q = 100A\sqrt{\Delta P}$$

where: Q = Inches^3/sec, A = Inches^2

This formula provides a close approximation when applied to standard hydraulic fluids at normal operating temperatures.

Flow Velocity

$$V = .321 \times \frac{\text{GPM}}{A}$$

where: V = FT/sec, A = Inches^2

Required Cooling Flow

$$\text{GPM} = \frac{\text{HP} \times 3.6}{\Delta\text{Temp. } ^\circ\text{F}}$$

This formula assumes a 30% HP loss and 10-W oil at 200°F.

Displacement, 19 Size

$$D = \frac{\tan \text{yoke angle} \times 2.5}{\tan 18^\circ}$$

*These formulae are useful for theoretical calculations. If used to obtain actual values, care must be taken in selection and efficiencies.

Motor Speed (RPM)

$$N_m = \frac{N_p \times D_p \times E_{vp} \times E_{vm}}{D_m}$$

Vehicle Speed

$$\text{MPH} = \frac{N_m \times R_L}{\text{FD} \times 168}$$

Wheel Torque To Accelerate Vehicle

$$T_w = \frac{\text{GVW} \times R_L \times a}{g}$$

where: a = FT/sec^2, g = 32.2 FT/sec^2

Pitch Diameter of Roller Chain Sprocket

$$\text{PD} = \frac{\text{Chain pitch in inches}}{\sin(180^\circ/\text{No. of teeth})}$$

Grade Angle

$$\text{Grade Angle} = \sin^{-1}\frac{\text{TE-RR}}{\text{GVW}}$$

Gradeability

$$\%\ \text{Grade} = 100 \times \tan \text{grade angle}$$

Surface	Rubber Tire	Crawler
Asphalt or Concrete	0.8 – 1.0	.05
Clay (dry)	0.5 – 0.7	.09
Sand and Gravel	0.3 – 0.6	.04
Soil (firm)	0.5 – 0.6	.09
Soil (loose)	0.4 – 0.5	.06
Snow (packed)	0.2	–

Figure 7. Coefficient of Slip (C_s)

Surface	Rubber Tire	Crawler
Concrete	10 – 20	40
Asphalt	12 – 22	40
Gravel	15 – 37	60
Soil	25 – 37	110
Mud	37 – 150	–
Sand	60 – 150	–
Snow	25 – 37	–

Figure 8. Coefficient of Rolling Resistance (C_r)

Application Data Sheet

The application data sheet is a convenient form for recording data required for analysis of the application. It also establishes a record of the application parameters, which becomes a permanent part of the application file. In general, all of the applicable information is important, and can be categorized as;

1. Essential data for basic calculations:
   a. Engine rated (gross) horsepower @ RPM
   b. Gross vehicle weight
   c. Weight on drive wheels
   d. Rolling radius
   e. Maximum tractive effort required
   f. Maximum vehicle speed required
   g. Engine to pump drive ratio
   h. Final drive ratio
   i. Circuit schematic
   j. Duty cycle
2. Required for calculations but, if not given, assumed values will be used:
   a. Net horsepower to transmission pump at Full Load Governed (FLG)
   b. No Load Governed (NLG) engine speed
   c. Weight distribution on drive wheels
   d. Coefficient of slip
   e. Rolling resistance
   f. Grade requirements

3. Background information
    a. Customer
    b. Machine description
    c. Auxiliary pump(s) function
    d. Pump drive arrangement
    e. Final drive arrangement

Application Value

When applying the 19 size transmission to any vehicle, the machine requirements must first be compared with the transmission capability to ensure that the transmission is applied within its rating. A convenient and simple way to establish this relationship is to use the Application Value method.

Machine Application Value. The machine requirements can be expressed as a numerical value by multiplying the tractive effort at wheel slip by the maximum speed in miles per hour. The result is machine application value in LBS MPH as shown in Figure 9.

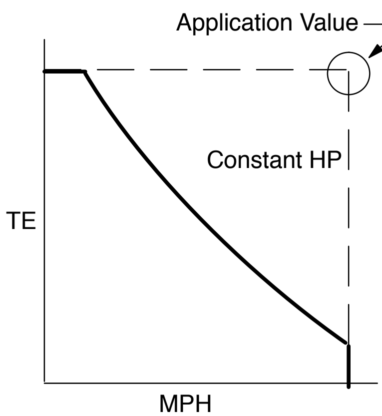


Figure 9. Machine application Value

Transmission Application Value. Constant horsepower performance of a motor can be shown with the same type of curve using motor torque vs. RPM, and a transmission application value can be calculated in terms of IN LBS RPM. In order to be useable for a direct comparison with machine application value, the motor capability must be expressed in the same terms – LBS MPH. The transmission application value is a function of pump speed, pressure and displacement with a constant applied to convert units to LBS MPH.

The transmission application value (AV) for either a PVMF or a PVMV transmission is shown in Figure 10. Enter the curve at the pump speed and read up to the type of transmission being used, then across to the application value.

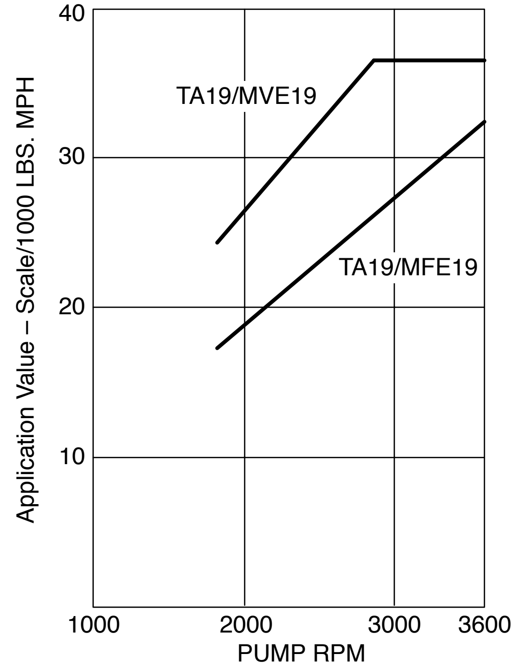


Figure 10. Transmission Application Value

If the transmission AV is *greater* than the machine AV, the application is viable. If the transmission AV is *less* than the machine AV and a fixed motor is being used, consider increasing pump speed and/or using a variable motor. A second alternative would be to reduce the machine AV by lowering the required maximum vehicle speed.

Guidelines For Using Application Value

1. The transmission application value must be greater than the machine application value.
2. In dual path systems, calculate for only one side of the machine (one transmission), with tractive effort to slip calculated with 60% of the total weight on the drive wheels on that side.
3. Application value method does not guarantee that the transmission will be within rating at the customer's design parameters. Pump speed and/or final drive ratio may have to be adjusted for application within rating.

Transmission Sizing Procedure (Using information given on the Application Data Sheet) See list of symbols for those used in the following calculations:

1. Check application value
    a. Calculate tractive effort to slip/wheels for single path applications (TE = weight on wheels $\times$ C_s). For dual path applications, weight on wheels is 60% total weight on both wheels (TE = weight on wheels $\times$ .6 $\times$ C_s).
    b. Machine AV = TE $\times$ max. vehicle speed (FLG).
    c. Determine transmission application value (see Figure 10). Transmission AV must be greater than machine AV.

VICKERS

TRANSMISSION APPLICATION DATA

FORM 83 (REV 5/85)

CUSTOMER		
ADDRESS		
CITY	STATE	ZIP
CONTACT	PHONE	

MACHINE DESCRIPTION

NAME	MODEL
FUNCTION	

TRANSMISSION FUNCTION

☐ PROPEL ☐ OTHER (DESCRIBE)

TRANSMISSION AUXILIARY PUMPS FUNCTION

☐ CYLINDER CIRCUIT ☐ AUXILIARY MOTOR ☐ CONTROL CIRCUIT ☐ OTHER (DESCRIBE)

ENGINE DATA

NAME	MODEL	
RATED (GROSS) HP @ RPM	NET HP @ FLG @ RPM	
NO LOAD GOVERNED SPEED (HI IDLE) RPM	LOW IDLE SPEED RPM	MAX. ENGINE TORQUE FT LB @ RPM

MACHINE DATA

GROSS VEHICLE WEIGHT MAX. LBS.	MIN. LBS.	WEIGHT ON DRIVEN WHEELS TOTAL LBS.	WEIGHT ON ONE (SET) OF DRIVE WHEELS MAX. LBS.	MIN. LBS.
TIRE SIZE AND TYPE	ROLLING RADIUS IN.	COEFF. SLIP	ROLLING RESISTANCE LBS/1000 # GVW	

REQUIRED MACHINE PERFORMANCE – TRACTION APPLICATIONS

MAX. TRACTIVE EFFORT LBS.	GRADEABILITY – MAX. %	MAX. WORK SPEED MPH	MAX. TRAVEL SPEED MPH
TRANSMISSION LIFE HRS./YEAR FOR YEARS			

TORQUE APPLICATIONS

STARTING TORQUE IN LB	MAX. SPEED RPM	MAX. RUNNING TORQUE IN LB @ RPM
DESCRIBE LOAD CONDITIONS:		

DRIVE TRAIN DATA PUMP(S)

NET HP AVAILABLE TO TRANSMISSION DRIVE @ FLG HP	PUMP DRIVE TYPE ☐ DIRECT ☐ SPUR GEAR ☐ BELT	SHAFT COUPLING ☐ RIGID ☐ FLOATING SPLINE ☐ U-JOINT ☐ FLEXIBLE
ENGINE TO PUMP DRIVE RATIO	EFFICIENCY %	PUMP SPEED @ FLG RPM

DRIVE TRAIN DATA (FINAL DRIVE)

FINAL DRIVE TYPE
☐ DIRECT ☐ GEAR BOX ☐ ROLLER CHAIN ☐ RIGID ☐ FLOATING SPLINE ☐ U-JOINT ☐ FLEXIBLE

MAKE AND MODEL OF FINAL DRIVE

FINAL DRIVE RATIO(S) – TOTAL INCLUDING DIFFERENTIAL
1ST 2ND 3RD FD EFFICIENCY %

ESTIMATED DUTY CYCLE

COND	% TIME	TE (TORQUE)	MPH (RPM)
1			
2			
3			
4			

AUXILIARY CIRCUITS

IN APPLYING AUXILIARY CIRCUITS, PRIMARY CONSIDERATION MUST BE GIVEN TO TRANSMISSION SUPERCHARGE REQUIREMENTS FOR ALL CONDITIONS OF OPERATION.

A COMPLETE CIRCUIT SCHEMATIC SHOWING ALL AUXILIARY FUNCTIONS MUST BE PROVIDED FOR ANALYSIS TO ENSURE SUPERCHARGE REQUIREMENTS ARE MET.

TRANSMISSION MOTORS

MOTORS FROM OTHER SUPPLIERS CAN USUALLY BE SUCCESSFULLY APPLIED WITH 19 SIZE PUMPS PROVIDED ALL SUPERCHARGE REQUIREMENTS ARE FULFILLED. THE FOLLOWING INFORMATION IS REQUIRED FOR ANALYSIS:

MOTOR MAKE		MODEL	
DISPL. C.I.R.	MAX. SPEED RPM	PRESSURE RATING PSI	CONTINUOUS PSI MAX

COMPLETE DESCRIPTION AND PERFORMANCE OF INTERNAL MOTOR CIRCUITRY INCLUDING SHUTTLE VALVES, BLEED VALVES, ORIFICES, DRAINS. ETC.

PROVIDE MOTOR PERFORMANCE CURVES INCLUDING LEAKAGE DATA AND EFFECT OF PUMP SUPERCHARGE PRESSURE ON LOW PRESSURE (MOTOR OUTLET) SIDE OF CLOSED LOOP.

ADDITIONAL INFORMATION

IS THIS APPLICATION A REPLACEMENT FOR ANOTHER MAKE OR MODEL TRANSMISSION? ☐ YES ☐ NO IF YES, THE FOLLOWING INFORMATION MUST BE PROVIDED:

TRANSMISSION SUPPLIER

MODEL NO. AND COMPLETE DESCRIPTION. INCLUDE EXISTING CIRCUIT INCLUDING ALL AUXILIARY FUNCTIONS

REASON FOR CHANGING SUPPLIERS

ARE VEHICLE PARAMETERS CHANGED ☐ YES ☐ NO

IF YES, DESCRIBE CHANGES:

2. Horsepower rating (HP)

a. Single Path $= \dfrac{\text{Input HP} \times 1000}{\text{RPM}} < 22.5$

b. Dual path (input HP is split between two pumps; use 80% of total input HP) =

$$\frac{\text{Input HP} \times .8 \times 1000}{\text{RPM}} < 22.5$$

Note: At this point, it has only been determined that the transmission has the capability of meeting vehicle requirements within rating. It is not known if the unit will meet the requirements using the final drive ratio, wheel radius and pump speed given on the data sheet.

3. Maximum tractive effort (TE)

Enter Motor Torque vs PSI curve (Figure 11) at maximum pressure &usually relief valve setting) and read motor torque at 18° yoke angle.

$$TE = \frac{T_m \times FD \times E_{fd} \times \text{Number of Motors}}{R_L}$$

For single path systems, compare TE maximum directly to TE to slip. TE maximum should be at least 4% higher than TE to slip for good application because of tolerance on relief valve setting.

For dual path systems, divide TE maximum by number of motors and compare to TE to slip with 60% weight on one wheel. TE maximum should be 4% above TE to slip.

If TE maximum is less than TE to slip, an adjustment in final drive ratio or wheel radius will be required.

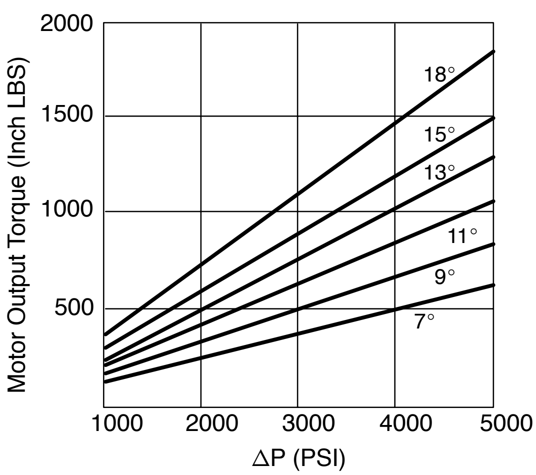


Figure 11. Motor Torque vs PSI

4. Maximum vehicle speed

Enter Flow vs Speed curve (Figure 12) at pump input speed (FLG). Read flow and motor RPM. Divide motor RPM by number of motors connected in parallel. Calculate vehicle speed:

$$MPH = \frac{N_m \times R_L}{FD \times 168 \times \text{Number of Motors}}$$

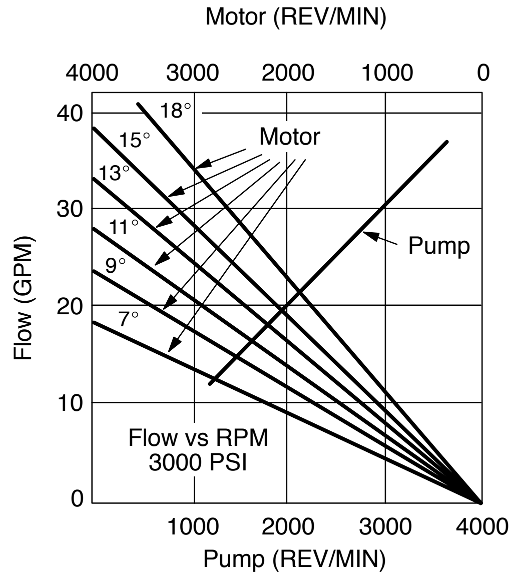


Figure 12. Flow vs Speed

5. Tractive effort to slip

Compare calculated values to data sheet requirements for maximum tractive effort and speed. Tractive effort to slip one wheel at worst condition should occur at least 4% below maximum tractive effort calculated.

Worst condition for a single path system is with weight equally divided between drive wheels. Worst condition for a dual path system will be a condition with assumed 60% weight on one wheel. Maximum TE is calculated for total machine, so it must be modified for comparison to TE-to-slip by calculating: TE = TE max. $\times$ .6.

If the maximum speed is acceptable and maximum tractive effort is at least 4% above the tractive effort at wheel slip, the application is acceptable and the application review can proceed. If either tractive effort and/or speed is not acceptable, proceed as follows:

a. Speed too low; tractive effort acceptable

OPTION 1. Maximum speed was calculated for a constant horsepower condition. Actual vehicle speed will be slightly higher, since volumetric efficiencies increase with decreased pressure and pump speed increases with decreased engine load. Calculate maximum theoretical speed using NLG pump speed and 100% volumetric efficiency.

$$MPH = \frac{\text{Pump RPM} \times R_L}{\text{Number of Motors} \times FD \times 168}$$

OPTION 2. Consider increasing pump speed to achieve desired vehicle speed.

OPTION 3. Consider use of variable motor.

OPTION 4. Consider decreasing final drive ratio, but only if tractive effort can be maintained at an acceptable level.

OPTION 5. Consider 2-speed final drive.

b. Tractive effort low; speed acceptable

OPTION 1. Increase final drive ratio and increase pump speed. Use variable motor, or 2-speed final drive to maintain speed.

OPTION 2. Increase motor displacement by using motors in parallel. One or both may be variable displacement to maintain speed.

6. Calculation of final drive ratio

If a single speed final drive is to be used, the ratio is always determined by the maximum tractive effort required to slip the wheels, plus 4% to allow for relief valve tolerance. Using Motor Torque vs. PSI curve (Figure 11), enter at maximum pressure (relief valve setting) and read motor torque at 18° yoke angle.

$$FD = \frac{TE_{SLIP} \times 1.04 \times R_L}{T_m \times E_{fd}}$$

7. Calculation of variable motor minimum yoke angle

Using maximum required vehicle speed from data sheet, calculate maximum motor speed:

$$N_m = \frac{MPH \times FD \times 168}{R_L}$$

N_m must be less than 4000 RPM at No Load Governed (NLG) pump speed and 100% volumetric efficiencies:

$$D_m = \frac{N_p \times D_p}{N_m} = \text{minimum displacement}$$

$$\text{Yoke Angle} = \tan^{-1}\left[\frac{D_m \times \tan 18^\circ}{2.5}\right]$$

Use next lower yoke angle offered (7°, 9°, 11°, 13°. 15°) to be sure vehicle speed can be reached. Recheck maximum motor speed at NLG engine speed to be sure motor speed does not exceed 4000 RPM.

$$N_m = \frac{N_p \times D_p}{D_m}$$

If motor speed exceeds 4000 RPM, use next higher increment of yoke angle and recheck.

8. Rolling resistance

Calculate vehicle rolling resistance using coefficient of rolling resistance given on data sheet, or from Figure 8, for the type of vehicle and soil conditions to be encountered.

$$RR = GVW \times C_r$$

The rolling resistance must be less than the tractive effort available under any condition of operation.

9. Gradeability

Gradeability is usually expressed in percent and is the equivalent to the tangent of the angle of the grade from the horizontal times 100. A 100% grade is 45°. Calculate the percent grade at maximum tractive effort and tractive effort at maximum speed. Where $\measuredangle$ = Grade angle,

$$\sin \measuredangle = \frac{TE-RR}{GVW}$$

$$\%\ Grade = \tan \measuredangle \times 100$$

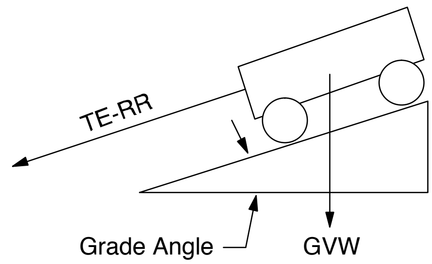


Figure 13.

Example – Single Path Transmission, Two Motors Connected In Parallel

The application is a tandem road roller for soil compaction and asphalt finishing. The conditions were selected to give a wide range of application possibilities. Given:

Engine HP @ FLG	60 @ 2800 RPM
Gross Vehicle Weight	8000 LBS
Weight on drive rolls	8000 LBS
Final drive ratio	30:1
Final drive efficiency	90%
Radius of drive rolls	20 IN
Coefficient of slip	.4
Coefficient of rolling resistance	55 LBS/1000 LBS GVW
Pump input speed @ FLG	2800 RPM
Pump input speed @ NLG	3000 RPM

Required:

Maximum Tractive Effort	3200 LBS
Maximum speed	8 MPH
Maximum grade	15%

Pump is direct drive from engine through a flex drive coupling. Pump is coupled to two motors in parallel, one motor on each of the front and rear rolls. Motors drive through a fixed ratio gearbox. The pump is a TA19V2010 with the V20 supplying an independent auxiliary circuit. The V10 is to supply supercharge.

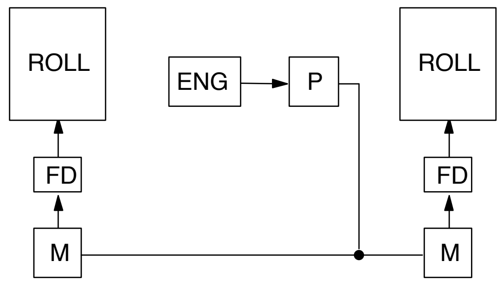


Figure 14.

STEP 1. APPLICATION VALUE

$$TE_{slip} = W_w \times C_s = 8000 \text{ LBS} \times .4 = 3200 \text{ LBS}$$

$$\text{Machine AV} = TE \times MPH = 320 \text{ LBS} \times 8 \text{ MPH} = 25{,}600 \text{ LBS MPH}$$

Transmission AV from Figure 10 for a TA29/MVE at 2800 RPM FLG is 35,800, so application is well within rating.

STEP 2. INPUT HORSEPOWER vs RATED HORSEPOWER

Pump input horsepower equals engine horsepower less horsepower required for supercharge, plus auxiliary no-load loss. Determine horsepower consumed by the V2010 for both supercharge and auxiliary functions from vane pump curves. For this example, we will assume they total 3.5 horsepower. Pump input horsepower/1000 RPM will be,

$$HP = \frac{(60\text{–}3.5) \times 1000}{2800} = 20.17 \text{ HP}$$

which is less than 22.5, so we are applied within the horsepower rating.

STEP 3. MAXIMUM TRACTIVE EFFORT

Data sheet indicates that 3200 pounds of tractive effort is required. Determine maximum tractive effort that can be realized with the 19 size transmission. Using Motor Torque vs PSI curve (Figure 11, enter at maximum pressure.

$$TE = \frac{1850 \text{ IN LBS} \times 30 \times .9\, E_{fd} \times 2}{20\, R_L} = 4995 \text{ LBS}$$

Value calculated is well above required 3200 pounds.

STEP 4. MAXIMUM VEHICLE SPEED

A maximum speed of 8 miles per hour is desired. From the Flow vs Speed curve (Figure 12), enter at 2800 pump RPM and project to pump curve; then across to 18° motor curve and read motor RPM. Divide result by two, because two motors are connected to the pump.

$$N_m = \frac{2450}{2} = 1225 \text{ RPM}$$

$$MPH = \frac{1225 N_m \times 20 \text{in.}}{30 FD \times 168} = 4.86 \text{ MPH}$$

The required 8 miles per hour is not met with fixed displacement motors, so alternatives must be explored. There are two courses of action open: 1) change Final Drive Ratio or 2) use some arrangement employing variable motors.

OPTION 1. NEW FINAL DRIVE RATIO

A change in final drive ratio is the best choice from a cost standpoint, unless the customer is "locked in" on the established ratio. Since sizing for wheel slip is required, use tractive effort to determine the new final drive ratio. A slip of 4% below the relief valve setting is desired; therefore, the maximum tractive effort for calculating the new final drive ratio will be:

$$TE_{max} = TE_{slip} \times 1.04 = 3200 \times 1.04 = 3328 \text{ LBS}$$

$$\text{New FD} = \frac{3328 \times 30}{4995} = 20 : 1$$

Recalculate the vehicle speed using new final drive ratio:

$$MPH = \frac{4.86 \text{ MPH} \times 30}{20} = 7.29 \text{ MPH}$$

This speed is based on Full Load Governed speed and horsepower. Reduced pressure required to overcome rolling resistance will result in higher volumetric efficiency and increased pump speed because of lower torque required. Therefore, check the maximum speed at No Load Governed (NLG) pump speed and 100% volumetric efficiency. From the data sheet, NLG pump speed is 3000 RPM.

$$N_M = \frac{3000}{2} = 1500 \text{ RPM}$$

$$MPH = \frac{1500 \times 20}{20 \times 168} = \mathit{8.9} \text{ MPH}$$

The result exceeds the required 8 miles per hour, so there is a good chance of customer acceptance. If the ratio change or speed is not acceptable, variable motors must be considered. Again, there are two options: either one fixed and one variable, or both variable motors.

OPTION 2. CHANGE TO VARIABLE MOTORS

One fixed and one variable motor is the most economical, both from cost and simplicity of controls. Calculate the motor speed for 8 miles per hour.

$$N_m = \frac{MPH \times FD \times 168}{R_L} = \frac{8 \times 30 \times 168}{20} = 2016 \text{ RPM}$$

Calculate the combined motor displacement of the fixed motor at full stroke and the variable motor at minimum stroke, to obtain a motor speed of 2016 RPM. Pump flow at 2800 RPM, from the Flow vs Speed curve (Figure 12), is 29 GPM. Therefore, the combined theoretical motor displacement,

$$D_m \text{ total} = \frac{29 \times 231}{2016} = 3.32 \text{ CIR}$$

Variable motor minimum displacement will be 3.32 minus 2.5, or .82 CIR. Minimum yoke angle is

$$\mathrm{Tan}^{-1}\left[\frac{.82 \times \tan 18^\circ}{2.5}\right] = 6.08^\circ$$

Motor displacement at a yoke angle of 7° has been established as the minimum displacement that will provide acceptable torque efficiency. Therefore, yoke stops for yoke angles of less than 7° are not offered. A 7° yoke angle is available, so calculate speed using one fixed motor at 2.5 CIR and one variable motor at 7° (or .94 CIR). Total motor displacement = 2.5 + .94, or 3.44 CIR.

To calculate motor speed and MPH,

$$N_m = \frac{29 \text{ GPM } Q_p \times 231}{3.44\ D_m} = 1947 \text{ RPM}$$

$$\text{MPH} = \frac{1947 \times 20}{30 \times 168} = 7.7 \text{ MPH}$$

If speed is acceptable, also calculate the maximum motor speed in the event the roll with the variable motor slips and total pump flow of 29 GPM is consumed by that motor.

$$N_m = \frac{29 \text{ GPM} \times 231}{.94 \text{ CIR}} = 7126 \text{ RPM}$$

Maximum motor speed must not exceed 4000 RPM, so some device is required which will limit maximum speed.

An alternate approach to using one fixed and one variable motor is to set the variable motor minimum stop at zero degrees. This approach has been used successfully on a number of applications requiring an extended speed range. If such a motor is used, the total pump flow of 29 GPM will be consumed by the fixed displacement motor. From the Flow vs Speed curve (Figure 12), the intersection of 29 GPM and the 18° motor curve is found at 2600 RPM motor speed.

$$\text{MPH} = \frac{2600\ N_m = 20 \text{ In. } R_L}{30 \text{ FD} \times 168} = 10.3 \text{ MPH}$$

If this approach is to be considered, the next step will be to determine if tractive effort to overcome rolling resistance at maximum speed can be achieved at a reasonable pressure.

$$\text{RR} = \frac{8000 \times 55}{1000} = 440 \text{ LBS}$$

Then, motor torque required,

$$T_m = \frac{\text{RR} \times R_L}{\text{FD} \times E_{fd}}$$

$$= \frac{440 \times 20}{30 \times .9}$$

$$= 326 \text{ IN LBS}$$

Using the Motor Torque vs. PSI curve (Figure 11), enter at 1000 PSI to 325 inch pounds. The necessary torque to overcome rolling resistance is achieved at slightly less than 1000 PSI.

If 10.3 MPH is too fast, it will be necessary to use two variable motors. Total displacement at 8 miles per hour was calculated to be 3.32 CIR. Therefore, minimum displacement per motor should be 1.66 CIR.

$$\text{Yoke Angle} = \tan^{-1}\left[\frac{1.66 \times \tan 18^\circ}{2.5}\right] = 12.17^\circ$$

Use either 11° or 13° yoke stops in the motor, determined by the 8 MPH requirement. Half the total pump flow of 29 GPM (for two motors in parallel) is 14.5. From Figure 13 and the formula for Vehicle Speed, the motor speed at both stop sizes can be determined.

11° = 2050 RPM, or 8.13 MPH
13° = 1650 RPM, or 6.55 MPH

Therefore, the 11° stops will be used.

Check the maximum motor speed. Because variable motors are used in parallel, total pump flow can go to one motor in the event it slips out. Be sure the maximum motor speed does not exceed 4000 RPM. Assume 100% volumetric efficiency and NLG pump speed of 3000 RPM.

$$N_m = \frac{3000 \times 2.5}{1.5} = 5000 \text{ RPM}$$

Maximum speed exceeds maximum rated speed, so some device is required which will limit maximum speed. With the exception of a 2-speed final drive, most of the feasible approaches to the application have been explored. Each one has a compromise in either economics or performance, but there are alternatives and data upon which to base a decision. Figure 15 summarizes these findings.

Option	Max. TE	Max. Work MPH	Max. Travel MPH	Remarks
2 MFE19's FD, 30:1	4995	4.86	4.86	Most economical.
2 MFE19's New FD, 20:1	3328	7.29	7.29	Economical. Requires FD ratio change.
1 MFE19 & 1 MVE19 @ 17° FD, 30:1	4995	4.86	7.70	Requires control to 1 motor & flow limiter to prevent overspeed.
1 MFE19 & 1 MVE19 @ 0° FD, 30:1	4995	4.86	10.30	Requires control to 1 motor.
2 MFE19's @ 11° FD, 30:1	4995	4.86	8.13	Requires control to both motors & 2 flow limiters. Most expensive.

Figure 15.

Note that the maximum travel speed calculated is at full load governed pump RPM at elevated pressure. actual maximum travel speed will be at a reduced horsepower and pressure level, that level required only to overcome rolling resistance. It will fall somewhere between calculated speed at FLG and maximum theoretical speed at NLG pump RPM (at 100% volumetric efficiency). Calculate the maximum theoretical speed by first determining motor RPM at NLG pump RPM and total motor displacement. Then calculate vehicle speed.

$$N_m = \frac{N_p\,NLG \times 2.5\ CIR\ D_p}{\text{Total } D_m}$$

$$MPH = \frac{N_m \times R_L}{FD \times 168}$$

Supercharge and Auxiliary Circuits

Supercharge Requirements

Maintenance of adequate supercharge flow along with system cleanliness are the two most important factors in the application of a closed loop transmission. Supercharge flow is required to charge the low pressure side of the loop and replenish fluid lost through leakage. Supercharge flow also provides cooling oil for the transmission.

Pressure

Supercharge pressure is required to prevent cavitation, erosion and high noise levels. The supercharge relief valve cracking pressure is 75 PSI on all standard models. Relief valve override will normally results in “S” port pressures in the 100-120 PSI range. For those applications which require higher pressure (e.g. actuation of a fail-safe brake), a setting of 150 PSI is available. However, the TA 19 model with integral gear type supercharge pump is available only with the standard 75 PSI setting.

Replenishing Flow

Flow into the closed loop must be sufficient to replenish fluid lost due to leakage at any operating condition during the useful life of the transmission. Inadequate replenishing flow results in a void being developed in the closed loop. This void causes loss of supercharge pressure, cavitation, erosion, high noise level, and the localized overheating of parts.

Operating Speeds

Minimum pump speed at idle conditions is 800 RPM. The transmission should be operated only at very light loads, such as moving the machine short distances on level ground. The pump should never be operated at less than 1200 RPM for any working load condition up to 3000 PSI. Supercharge pressure must be monitored and if it drops below 65 PSI plus case pressure, the pump speed must be increased accordingly. Minimum pump speed for operation at full load is 1800 RPM.

Supercharge Pump Sizing

The supercharge pump must be sized for each application and must be sufficient to satisfy the requirements noted above. Use of supercharge pump flow for auxiliary functions can be considered only after flow requirements to make up leakage are satisfied (with one exception: if the total flow into the auxiliary circuit is continually returned to the supercharge circuit without interruption).

When considering the TA19 with integral gear type supercharge pump (model TA19*-***-21), the following limitations apply:

1. This model may be used only with a single MFE19 motor unless otherwise approved by Vickers Engineering.
2. Supercharge relief valve is set at 75 PSIG and the case pressure must not exceed 20 PSI. Higher relief valve settings are not permitted with this model.
3. Supercharge pump flow may not be used for auxiliary functions in this model.

Procedure For Sizing

1. Establish the total number of pump and motor rotating groups contributing to the total leakage which must be made up by the supercharge pump.
2. Determine the total maximum leakage from Figure 16 for each rotating group in the circuits, at minimum and full load governed pump speeds.
3. Add the leakage of all rotating groups in the system (Q_L) at minimum and full load governed speeds.
4. Determine the maximum auxiliary circuit flow (Q_A).
5. Determine the minimum auxiliary flow (if any) returned to supercharge (Q_R).
6. Determine the vane pump flow required: $Q_L + Q_A - Q_R$.
7. Determine the vane pump ring size from Figure 17. Select the next highest size which will satisfy requirements of vane pump flow at minimum and full governed speeds.

Unit	RPM	Maximum Leakage – GPM 5000 PSI	Maximum Leakage – GPM 3000 PSI
Motor	Stall[1]	3.50	2.50
Pump	1200[2]	–	.75
	1800	1.40	.85
	2400	1.75	1.00
	3000	2.10	1.20
	3600	2.50	1.50

Figure 16. Maximum Leakage for Supercharge Pump Sizing

[1]Nearly all transmission applications require operation at or near motor stall, especially when starting under load. Therefore, stall leakage of motors should be used unless it can be definitely established that stall or starting under load will not occur.

[2]Minimum recommended working speed. Minimum idle speed is 800 RPM

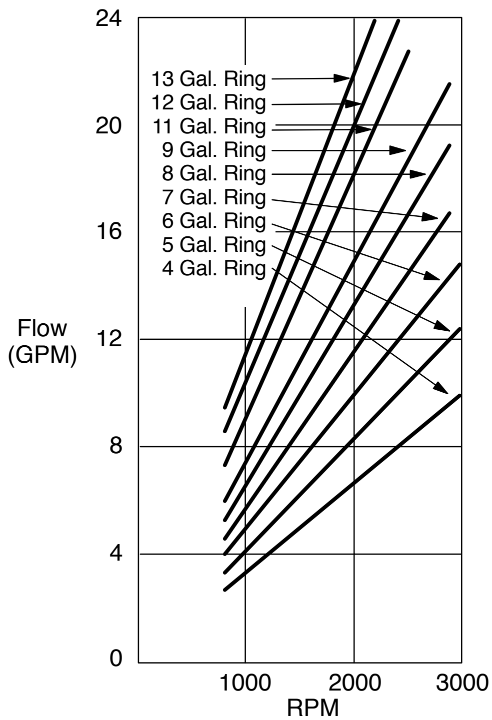


Figure 17. Vane Pump Ring Sizing

Cooling Flow

Supercharge flow in excess of that required for replenishing is used to remove heat from the transmission circuit. This excess flow is discharged over the supercharge relief valve into the pump case(s) where it absorbs heat from the rotating group(s) and exits from the case drain.

Auxiliary Functions

One of the attractive functions of the 19 size transmission is its ability to provide power for some auxiliary functions, in addition to its main function. However, since this feature is prohibited on the Ta19 model with internal gear type supercharge, the following discussion applies only to those models which use vane pumps or remote supercharge pumps.

The supercharge pump must have adequate capacity to supply the requirements of both supercharge and auxiliary functions. Remember that transmission supercharge is the primary function of the auxiliary pump. Applications which result in even a momentary drop of supercharge pressure (below 75 PSI above case pressure) are to be avoided. It must also be remembered that a low speed, low flow condition is nearly always the critical point as far as supercharge supply is concerned, and must be considered when applying auxiliary functions. It is sometimes necessary to install a lockout device which will prevent operation of the auxiliary functions when operating at critical conditions of supercharge flow.

Auxiliary circuit directional control valves should not cause interruption of supercharge when actuated. The use of a priority, flow control or flow divider valve will extend the application possibilities for auxiliary functions.

Relief valves in a combined supercharge and auxiliary circuit should always discharge into the supercharge line before the cooler and filter; they should never be connected directly to the reservoir. Applications using a V2010, in which the V20 supplies an independent circuit, will have an independent return to reservoir from the V20 and not into the supercharge.

Use a double rod cylinder or two double acting single rod cylinders connected to provide equal fill and return flow whenever possible. This will provide a constant flow from the auxiliary circuit to the replenishing circuit. If the cylinders are connected so that fill and return flows are unequal, the extra fluid required could exceed the total output of the supercharge pump. If the reduction of flow from the auxiliary circuit is sufficient to drop supercharge pressure below its required value, the transmission pump may cavitate and reduced life will result. If return flow is insufficient to maintain supercharge under every condition, a priority, flow control or flow divider cover will be required to ensure adequate supercharge.

Use of single acting cylinders will nearly always require use of one of these controls to ensure supercharge. In any case, auxiliary circuits used in conjunction with the supercharge circuit must be very carefully analyzed. If any doubt exists, a double pump such as the V2010 or a remotely driven pump should be recommended. The analysis must consider that the transmission pump input speed is determined by the rated speed of the vane pump used. Figures 18 through 21 are representative samples of acceptable auxiliary circuits.

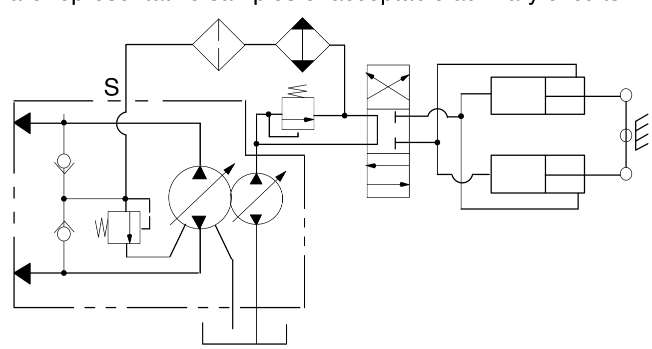


Figure 18. Auxiliary steering for an articulated loader using two double acting cylinders. Note that they cylinders are connected sos that return flow is equal to inlet flow, resulting in constant supercharge flow under all conditions.

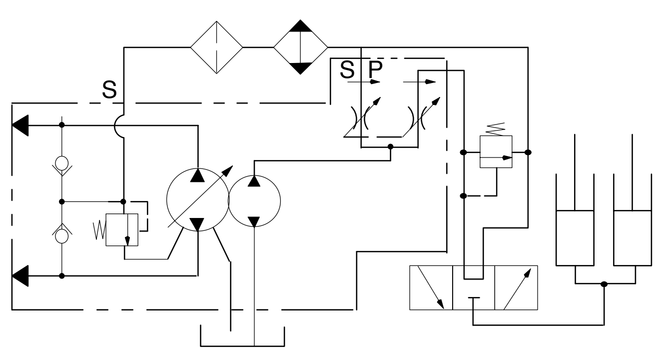


Figure 19. Header lift circuit for a windrower using two single acting cylinders. Note that a flow divider cover is used in this circuit to ensure supercharge flow as the header is lifted. A flow control or priority valve also could be used.

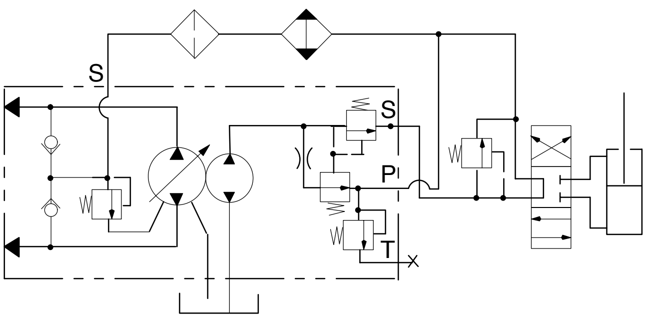


Figure 20. Steering circuit using a double acting cylinder. A priority valve is shown but, again, a flow control or flow divider could be used depending upon the application. The circuit as shown requires an external relief valve to protect the auxiliary circuit.

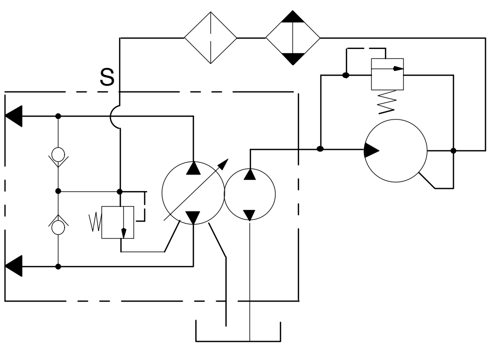


Figure 21. A hydraulic motor is series with the supercharge circuit. The motor case must be ported to the outlet to prevent loss of fluid from the supercharge circuit. The motor case and seals must be capable of withstanding replenishing pressure. A quick acting cross port relief valve is required to prevent interruption of supercharge if the motor is stalled.

The above are only a few examples of the many approaches possible. The charge pump can nearly always accommodate some auxiliary functions within the limits of torque required and flow available. The only exception is the TA19 model, as noted earlier.

Vane Pump Covers

The use of flow control covers, priority covers and flow divider covers requires special considerations. Their application is discussed below.

Flow Control Cover

In some cases, a flow control cover can be used on the vane pump to provide power to auxiliary functions and still ensure adequate supercharge flow. The function of the flow control cover is to provide a relatively constant flow regardless of total pump flow. When used in a transmission circuit in which vane pump flow is used primarily for supercharge, selection of ring size, orifice size and minimum speed at which the auxiliary function is energized will be dependent on supercharge requirements and the auxiliary circuit used. Some examples:

1. A single acting cylinder circuit is frequently employed for lift functions in which gravity is used to retract the cylinder. When this type of circuit is actuated, total controlled flow is directed to the cylinder. The flow available for supercharge will be the difference between controller flow and total vane pump flow.

   Figure 22 shows flow versus speed for a 7-gallon ring and controlled flow from a 4- and a 6-gallon flow control cover. At 1000 RPM, there is about 6 GPM total vane pump flow available. In a circuit with a 6-gallon flow control, the cylinder demand (when actuated) is over 6 GPM, leaving less than 1 GPM for supercharge. With a 4-gallon flow control, the cylinder demand is 4 GPM, leaving only 2 GPM for supercharge.

   In either case, in most transmission applications, the resulting supercharge flow will be too low to prevent damage to the transmission. Increasing pump speed to 1500 RPM shows that there will be slightly over 5 GPM for supercharge with the 4-gallon orifice, which probably would be sufficient to maintain supercharge under most conditions. With a 6-gallon orifice, there is less than 3 GPM available for supercharge.

   A flow control cover should be avoided when the circuit involves single acting cylinders. A flow divider cover or priority cover could be used, or supplemental supercharge flow should be provided.

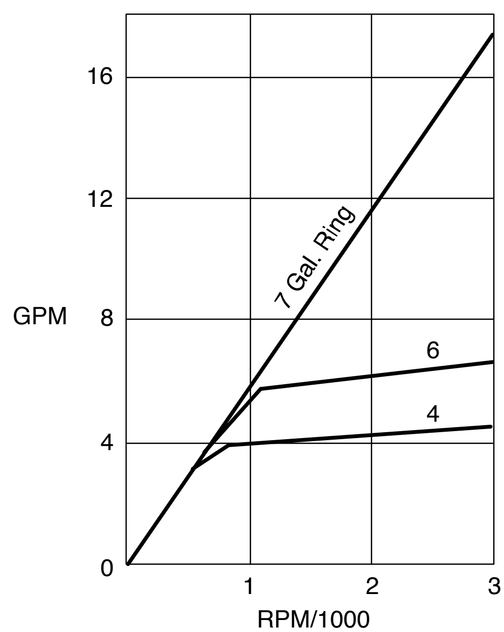


Figure 22.

2. When double acting cylinders are used, the condition is improved because flow from the rod end is returned to supercharge. The rod end flow will be proportional to the areas of the rod end and head end of the cylinder, and low from the rod end must be calculated on this basis. Flow control covers used with double acting cylinders should be applied with caution when all of the controlled flow is directed to the head end of the cylinders.

If the circuit contains two or multiples of two mechanically connected, double acting cylinders, and controlled flow is directed to the head end and rod end of respective cylinders, the return flow to supercharge will equal controlled flow; thus, total vane pump flow is available for supercharge. The only reason for using a flow control cover in this type of circuit is to provide a constant stroking rate of the cylinders regardless of vane pump speed.

Priority Cover

In those applications where the auxiliary circuit must be energized at low engine speeds, a priority cover can be used. Either the primary or secondary port can be used to supply supercharge, with selection being dependent upon the application requirements. (See Figure 23)

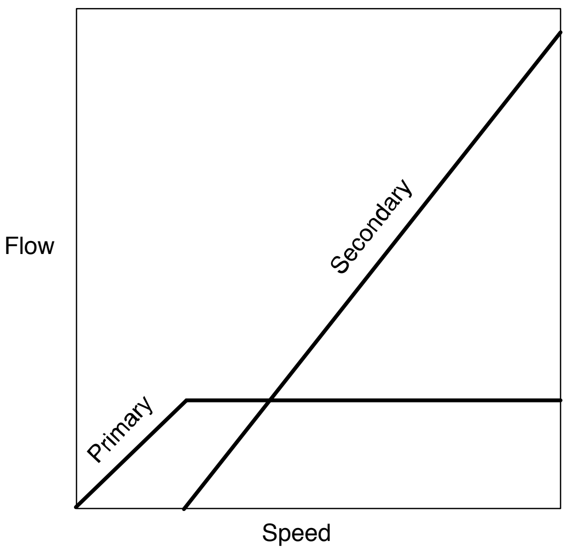


Figure 23. Priority Cover Characteristics

Secondary port to supercharge, primary to auxiliary:

This arrangement is used when a relatively constant flow at low flow levels is required for the auxiliary circuit. It is ideal for steering applications where steering response should be independent of pump speed. Care should be used when applying the priority cover in this manner, because secondary port (supercharge) flow is zero until sufficient pump speed to supply controlled primary (auxiliary) flow is available.

In sizing, the flow required for the auxiliary circuit should be established in order to select orifice size for primary port flow. Ring size is then selected to provide minimum supercharge flow plus auxiliary flow at the minimum pump speed required. A typical circuit is shown in Figure 24.

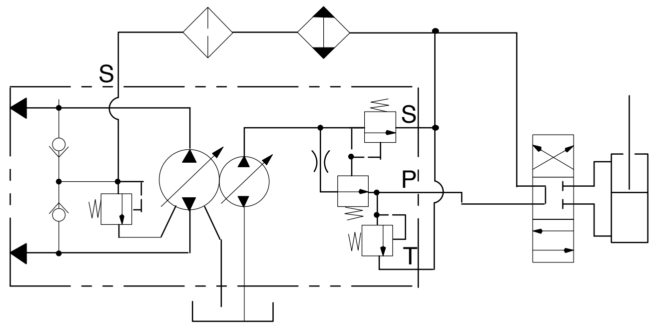


Figure 24.

Note that in this circuit the relief valve in the cover protects the auxiliary circuit, and the priority cover tank and secondary ports are connected to the supercharge supply to prevent flow loss to the supercharge circuit. The transmission supercharge relief valve protects the secondary circuit.

Primary port to supercharge, secondary to auxiliary:

This circuit should be used where higher flows are required for the auxiliary circuit and can vary with pump speed. In sizing, the cover orifice should first be selected to provide minimum supercharge flow at minimum pump speed. Ring size should then be selected to provide sufficient supercharge flow plus flow required for the auxiliary circuit at normal speed. A typical circuit is shown in Figure 25.

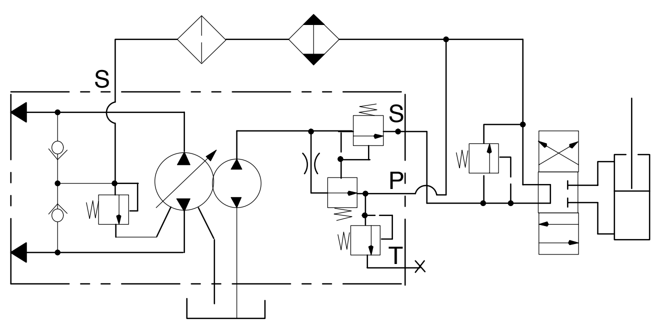


Figure 25.

Note that in this circuit the relief valve in the cover is non-functional and must be plugged. An external relief valve must be provided to protect the auxiliary circuit.

Flow Divider Cover

There are some applications in which neither the flow control or the priority control will satisfy supercharge and auxiliary requirements at low pump speeds. An example is a dual path transmission in which supercharge must make up leakage of four rotating groups at low pump speeds. At the same time, some auxiliary flow is required to actuate single acting cylinders.

The flow divider cover splits the flow so that approximately 60% goes to supercharge while the remaining flow is available for the auxiliary circuit. It is characteristic of the valve that the initial 2.5 GPM output of the vane pump on start-up is directed to supercharge before the valve starts to divide flow. (See Figure 26.) Ring size only needs to be sized and is determined by the supercharge required at minimum speed, with 60% of the total flow to supercharge.

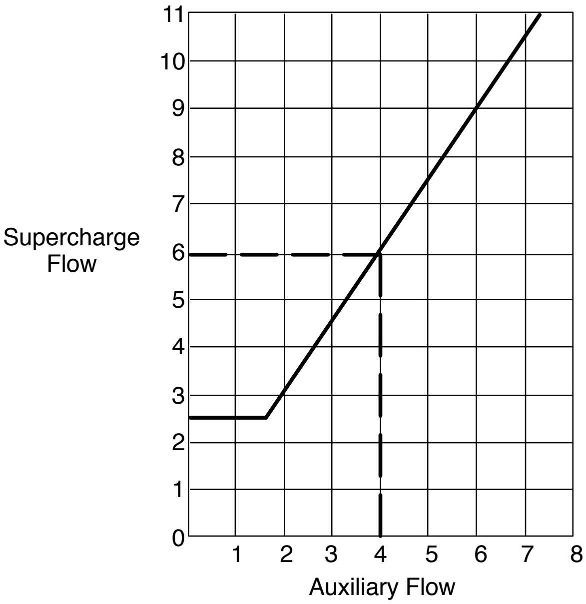


Figure 26.

A typical circuit is shown in Figure 27. Note that the flow divider cover does not provide a relief valve for protection of the auxiliary circuit, and one must be provided.

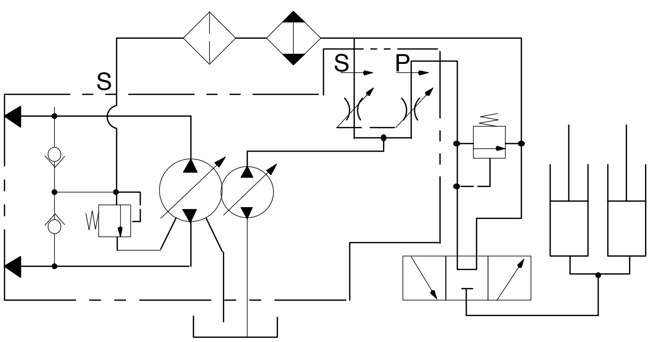


Figure 27.

Example, application using flow control cover:

The application is a traction transmission using a TA19V*-F driving two MFE19 motors connected in parallel. The vane pump supplies both the transmission supercharge and a steering circuit which requires 4 GPM flow from the “F” (flow control) cover. The steering circuit has a cylinder with a ratio of rod end to head end area of .6. Return flow from auxiliary circuit goes to the supercharge circuit. Full load governed speed is 2400 RPM, and a capability of operating as low as 1200 RPM at 3000 PSI is desired. The transmission circuit is as shown in Figure 28. Proceed as follows:

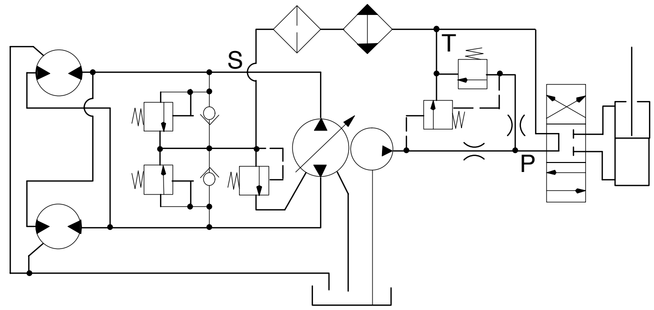


Figure 28.

1. Three rotating groups: one pump and two motors.
2. Total leakage to be made up (Q_L) @:
   a. 2400 RPM pump speed and 5000 PSI:

One pump	1.75 GPM
Two motors at stall	7.00 GPM
Total leakage @ 2400 RPM	8.75 GPM

   b. 1200 RPM pump speed and 3000 PSI:

One pump	.75 GPM
Two motors at stall	5.00 GPM
Total leakage @ 1200 RPM	5.75 GPM

3. Steering circuit requires auxiliary flow (Q_A) of 4 GPM.
4. Calculate minimum fluid returned to the supercharge circuit from the auxiliary circuit (Q_R): 4 GPM $\times$ .6 = 2.4 GPM.
5. Total flow required from the vane pump at both:
   a. 2400 RPM and 5000 PSI:

   $8.75\ Q_L + 4\ Q_A - 2.4\ Q_R = 10.35$ GPM

   b. 1200 RPM pump speed and 3000 PSI:

   $5.75\ Q_L + 4\ Q_A - 2.4\ Q_R = 10.35$ GPM
6. Vane pump ring size required (refer to Figure 17):

   @ 2400 RPM, 10.35 GPM = 6-gallon ring
   @ 1200 RPM, 7.35 GPM = 8-gallon ring
7. Analysis: operation at the minimum speed requirement dictates the use of an 8-gallon ring, which is available only in a V20 vane pump. In this case, it would be more economical to consider increasing the minimum speed requirement to 1400 RPM, which would permit the use of a 7-gallon ring in a V10 vane pump.

Section III – Drive Shafts and Bearings

Available Shaft Ends

The SAE BB splined shaft was selected as the standard 19 size transmission shaft, because its load carrying ability most nearly matches the torque requirements of the unit in transmission applications. Other shafts can be supplied for special application requirements. These shafts are either currently being manufactured or can be made if design and order quantity justify release, provided they comply with application limitations.

Shaft Limitations and Restrictions

Any shaft, including motor through-shafts, which will be subjected to side (bending) loading will impose additional bearing loads and bending stress on the shaft. It is very important that such applications be reviewed and approved by Vickers Engineering.

Transmission motors with through-shaft extensions are provided for application of parking or emergency brakes. They are not intended for use as vehicle service brakes.

Shafts with SAE B splines (13 teeth) are limited to applications with a maximum system pressure of 3500 PSI, established by relief valve or wheel slip. Straight keyed shafts are discouraged for transmission applications because of the limited torque capacity of the keyway. If used, system pressure must not exceed 3000 PSI unless approved by Vickers Engineering.

Spline Wear

The most common cause of spline wear is fretting corrosion, which is the result of tooth compressive stress (torque), the amount and rate of relative motion between mating teeth (misalignment), oxidation of the tooth surface (lack of lubrication), and material and hardness of the mating teeth.

Assuming all other factors being equal, the spline wear will become a function of the tooth compressive stress. The SAE B spline compressive stress is 1.75 times greater than that of the SAE BB spline, when operated at the same load. Thus, while the spline strength of an SAE B spline may be adequate for an application, the spline wear may be unacceptable.

An adequate supply of lubricant is essential for good spline life. One-time greasing at assembly usually results in short spline life. Life can be prolonged somewhat by using a molybdenum disulphide compound. Periodic greasing at intervals determined by torque load and environment will usually give adequate spline life. The installations for best spline life are so-called wet flange mountings, in which gear box lubricant is always present at the spline.

Shaft Indirect Loading

Applications which require the shaft extension to be subjected to side or thrust loads require Vickers Engineering review and approval. Indirect loads on the shaft affect both shaft stress and bearing life, and the calculations required to determine the values of each are quite complex and beyond the scope of this manual. Indirect loads on the shaft can be imposed by misalignment, shaft deflection, and loads due to belt, chain and gear drive arrangements.

Misalignment

Alignment of the unit with the mating mounting pad and drive must meet the requirements noted on the applicable installation drawing. Excessive misalignment causes high stress in the spline teeth and greatly affects bearing loads.

Shaft deflection is inherent in inline piston units because of the load imposed between bearings by the 90° force component, due to pressure and yoke angle. In order to realize maximum reliability, the coupling between the transmission shaft and the rest of the drive train should be sufficiently flexible to accommodate all of the misalignment in both the driving and driven members. Rigid couplings are not recommended and should be avoided.

Chain, Belt and Gear Drives

With the exception of a planetary gear box, these drive arrangements always result in an indirect load on the shaft and, as such, require Vickers review and approval. The review consists of shaft stress and bearing life calculations based on the duty cycle provided by the customer.

Additional information on the installation is required in order to establish magnitude, direction and location of shaft load. This includes orientation of units on both right and left side of a machine, if applicable. Referring to Figure 29, provide the following:

1. *Yoke Position 1 or 2* (motors will always have position 1, with drain port D1 up).
2. *L:* distance of external load to the mounting flange.
3. *Angle* β*:* angle of the ground plane or horizontal centerline of the machine to the vertical centerline of the unit, as shown on the installation drawing. (Units are often rotated to some mounting angle to obtain optimum shaft stress and bearing life.)
4. *Angle* θ*:* angle of the vertical centerline of the unit to a line connecting the center of the driving and driven sprockets.
5. *Center Distance:* the distance between the centers of the two sprockets.
6. *PD 1:* pitch diameter of the sprocket mounted on the unit.

7. *PD 2:* pitch diameter of the other sprocket.

If pitch diameters of the sprockets are not readily available, they can be calculated from the chain pitch and number of teeth in the sprocket, using the formula:

$$PD = \frac{\text{Chain Pitch in Inches}}{\text{Sin } (180^\circ/\text{No. of Teeth})}$$

8. *R:* direction of rotation.

For belt drives, in addition to the above, belt pretension must be given. For straight spur gears, pressure angle must be given but center distance is not required. For helical spur, bevel, spiral bevel and other types of gears, a full description of the bear is required.

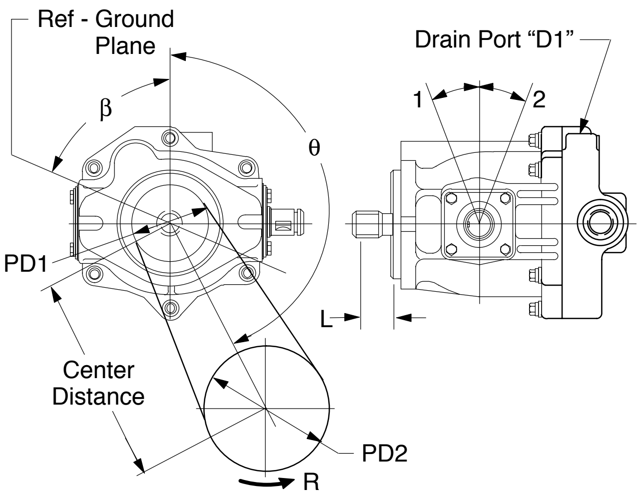


Figure 29.

Bearing Life

Life Expectancy

Life expectancy is a very important consideration in the application of a transmission, and one of the first questions a customer will ask is, “How long will it live?”. This is affected by any number of factors, including temperature, speeds, loads, lubrication, alignment of parts, materials and contamination level. Because bearings are an essential element in all rotating machines, methods of calculating bearing life have been established over many years and have become a dependable yardstick in predicting the useful life of such machines.

B10 Life

Bearing life is expressed in hours, and B10 life in hours is the accepted method of presentation.

Any bearing will eventually fail through fatigue, and in any sample some will survive longer than others. Therefore, it has been established that the criteria for expression of bearing life is defined as the number of hours that 90% of the bearings in a given sample will survive when run under the same conditions. It follows that the *failure* rate is then 10%; hence, the designation of B10 hours for expressing bearing life.

Effect of Speed and Load

All other factors being equal, speed and load have the greatest effect on bearing life, with load being the more influential. For example, decreasing the speed by one-half will nearly double the B10 life, while decreasing the load by one-half will increase the B10 life about ten times. The converse is also true: double the load and B10 life will be reduced to about one-tenth of the previous life.

Calculation of Bearing Life

Theoretical B10 life is easily calculated using procedures outlined in all bearing handbooks. The B10 life so calculated is based on the bearing running under specific and controlled conditions. Actual B10 life must consider the variables of the installation and application conditions. Therefore, calculations for transmissions require modifications of the theoretical bearing life by applying factors for misalignment, end play and lubrication. In addition, a factor must be applied which is determined by the bearings and bearing arrangement peculiar to the 19 size E Series units.

Because of the varying duty conditions to which a mobile transmission is subjected, bearing life for a single condition would be misleading and unrealistic. A vehicle transmission must operate at varying speeds and loads during its normal duty cycle. Calculation for actual B10 life for a single condition of the duty cycle is meaningless. Therefore, a method of calculating the expected or weighted average B10 life is used.

A machine duty cycle must be established either by actual test or experience and should be given as shown in the application data sheet. A minimum of three conditions should be considered to give credibility to the calculation: maximum torque, normal working condition and maximum speed. Anticipated percent of total life at each condition is the weighing factor and should be as realistic as possible. Tractive effort (torque) values listed must be equatable to system pressure, and miles per hour (RPM) must be equatable to pump or motor RPM and yoke angle.

Actual B10 life is calculated for each condition. Percent of time at each condition is applied to arrive at the weighted average B10 life for the duty cycle given.

Life for applications which have no overhung or thrust loads on the shaft may be calculated using the following procedure (tractive effort and MPH given):

1. Calculate motor speed:

$$N_m = \frac{MPH \times FD \times 168}{R_L}$$

2. Determine pressure:

   a. Calculate motor torque:

$$T_m = \frac{TE \times R_L}{FD \times E_{fd} \times \text{Number of Motors}}$$

b. Enter appropriate motor torque vs. P curve (Figure 11); read pressure at motor torque.

$$T_m = \frac{TE \times R_L}{FD \times E_{fd} \times \text{Number of Motors}}$$

Record motor inlet flow from flow vs. RPM curve (Figure 12).

3. If parameters other than tractive effort and MPH are given, calculate pressure, motor yoke angle and motor speed as required.

4. B10 life for the motor can be read from the bearing life curve (Figure 30). Enter the curve at the pressure for the duty cycle condition. Read up to the yoke angle, then across to the RPM. Read down for the B10 life. Repeat for each condition of the duty cycle and record.

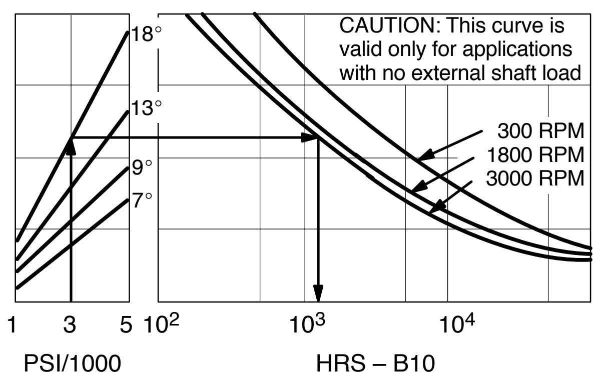


Figure 30. Bearing Life, E Series 19 Size Pumps and Motors

5. Calculate pump yoke angle using motor inlet flow, previously recorded, and pump RPM:

$$D_p = \frac{GPM \times 231}{N_p}$$

6. Calculate the yoke angle.

7. Pump B10 life can now be determined for each condition of the duty cycle using the same procedure outlined above (4).

8. Calculate the weighted average B10 life for each pump and motor using the following equation:

$$B10_{AV} = \frac{1}{\frac{\%\text{ Time Cond. 1}}{\text{B10 Cond. 1}} + \frac{\%\text{ Time Cond. 2}}{\text{B10 Cond. 2}} + \frac{\%\text{ Time Cond. N}}{\text{B10 Cond. N}}}$$

(% Time is a fraction, i.e. 1% = .01)

Remember that the above procedure is valid only for those applications with no external shaft loads. If the application has external shaft loads, the duty cycle and other required information must be given (see “Shaft Indirect Loading”). The procedures are quite complex and the information required to calculate by hand is quite extensive. There, it is more expedient for Vickers Engineering to perform the calculations. Shaft stress also becomes critical with external loads and must be calculated.

Section IV – Controls

Control Linkage

Variable displacement transmission units are furnished with a pintle shaft extension for the attachment of necessary displacement controls. Because of the reversing nature of the control forces, it is essential that, in addition to the locating key, the attached level arm be securely clamped to the pintle shaft. Inadequate clamping can lead to fatigue failures of control elements. A groove is provided on the pintle shaft so a snap ring may be installed as a safety measure.

The section on Yoke Moments contains all of the information required on control forces generated in the pump or motor for the design of customer-supplied control elements. Structureborne noise transmitted to the operator’s station through control linkage can be reduced by inserting a resilient member in the linkage as close to the pump or motor as possible.

Yoke Stops

Internal, nonadjustable yoke stops are used in all variable displacement transmission units. Maximum angle stops are machined integral with the housing and yoke, and no machining allowance for other than standard 18° yoke angle is provided.

Minimum angle stops for variable displacement motors are available for nominal 15°, 13°, 11°, 9° and 7° yoke angles. Incremental yoke angles between these angles are not offered because manufacturing tolerances establish the limits of practicality. In those applications which require precise tuning, the customer is expected to provide an external, adjustable stop in the control linkage.

Minimum motor yoke angles less than 7° are not provided and not recommended, because torque (mechanical) efficiency decreases rapidly in an unpredictable manner at yoke angles less than 7°. Exception: Vickers offers a 0°

minimum yoke angle stop for those applications using multiple motors connected in parallel, where it would be advantageous to stroke one motor to zero stroke in order to extend the speed range of the vehicle. This approach can eliminate expensive drive line components and/or controls.

The application engineer is again reminded that the maximum motor speed at reduced yoke angles (13° or less) must not exceed the rated 4000 RPM under any condition of operation for reasons of safety and liability. Maximum motor speed at 15° and 18° is rated 3600 RPM.

Yoke Moments

The yoke of any variable displacement inline piston unit has a tendency to rotate about the pintle centerline under any condition of operation. The torque required to restrain this tendency and hold the yoke in a given position or angle is known as the control *yoke moment* or total moment.

The control yoke moment is the algebraic sum of the contributing effects of pressure, speed and piston friction at a given angle of the yoke. These are generally referred to as pressure moment, inertia moment and friction moment.

Pressure Moment

Pressure moment is the torque-producing effect of the difference in area under pressure on opposite sides of the pintle centerline. It is affected by speed, wafer plate timing, yoke angle and pressure. Pressure moments in the pumping mode are always stroke-reducing; that is, the moment tends to rotate the yoke towards neutral or the zero stroke position. Pressure moments in the motoring mode are stroke-increasing, towards maximum displacement.

Low Speed Pressure Moments

At very low speeds, the pressure moment is oscillatory in nature and is often referred to as the AC moment or reversing moment. The cause is exposure of alternating four and then five pistons to system pressure above and below the pintle centerline. This results in a maximum stroke-reducing moment, 18 times (9 pistons $\times$ 2) with each revolution of the cylinder block.

The effects of these moments are always present in both pumps and motors. They are most noticeable at shaft speeds below 50 RPM and seem to disappear at speeds above 300 RPM, only because the amplitude decreases as frequency increases. Variable motors, because they must operate at these low speeds, must have the control linkage designed to withstand these forces in a near static condition. Control linkage on both pump and motor must be designed to withstand the fatigue effects of these moments. Figure 31 shows the magnitude of these moments as related to system pressure.

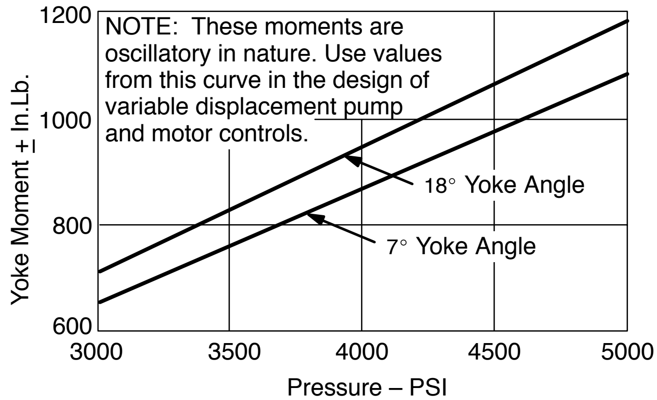


Figure 31.

Inertia Moment

Inertia moment is the torque tending to rotate the yoke about the pintle centerline, due to the inertia of moving parts. This results from the reaction to the force required to accelerate the pistons in the cylinder block, speed, weight of the piston/ shoe assembly, cylinder block pitch diameter and yoke angle. Inertia moment is always stroke-increasing and varies with the square of the speed.

Friction Moment

Friction moment is the torque tending to rotate the yoke about the pintle centerline due to piston friction and is affected by pressure and yoke angle and, to some extent, by speed. The friction moment in the pumping mode is stroke-increasing, and in the motoring mode is stroke-reducing. The friction moment is most pronounced in new units. As the unit breaks in and the coefficient of friction decreases, the effect of the friction moment on total moment decreases.

Effect of Variables on Yoke Moments

As stated, pressure moments in a pump are always stroke-reducing. Inertia moments are always stroke-increasing, in either the pumping or motoring mode. Friction moments in a pump are always stroke-increasing, and stroke-reducing in a motor. The *total* moment can be either stroke-reducing or stroke-increasing, depending upon the combination of variables producing the total moment.

Optional wafer plates that reduce sound levels of the 19 size transmissions from 5 to 8 dB(A) are available. However, use of these plates results in a much stronger stroke-increasing tendency than with standard plates (see Figures 32 and 33) and a neutral centering device must be provided to prevent movement of the vehicle when controls are unattended at any time the pump is running.

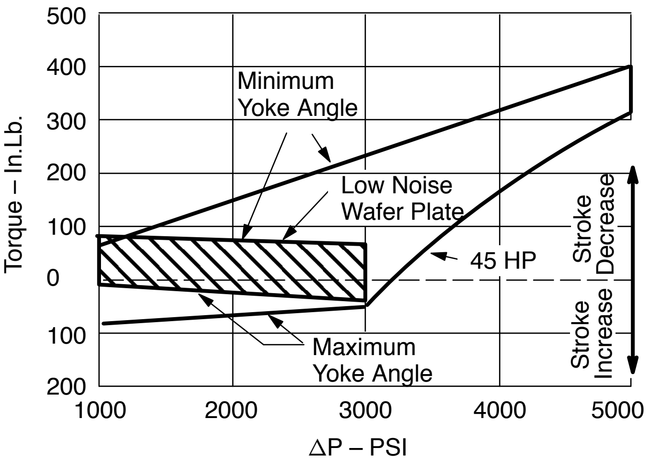


Figure 32. Pump yoke moments @ 2000 RPM

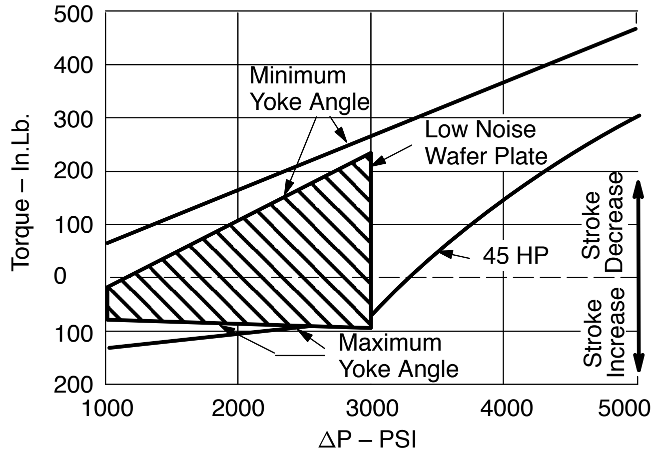


Figure 33. Pump Yoke Moments @ 3000 RPM

Section V – Selection of Circuit Elements

Fluid Requirements and Recommendations

Selection of fluid for transmission use should be based on Vickers "Oil Recommendations for Mobile Hydraulic Systems" (Form M-2950-S, rev. 11/87). Vickers maintains a modern and well equipped Fluids Laboratory. The knowledge and experience accumulated over the years is available to our customers as an aid in the selection of fluids for their application.

When requesting Vickers opinion on the suitability of a fluid, complete information is required, including: manufacturing, brand name, any identifying numbers or letters, and formulation if possible. Do not send fluid samples unless requested by the Fluids Laboratory.

The fluid should have at least the minimum recommended antiwear additive for satisfactory service life. Multiple viscosity oils may be used, but these oils tend to shear in service and result in a permanent decrease in viscosity which can affect service life.

Oils other than mineral based fluids are not recommended for general use. In those applications which require use of fire resistant, water based fluids or other types of fluids, consult Vickers before using.

If a fluid other than SAE 10W is used, the viscosity of the fluid should not be less than 50 SUS for continuous operation. The maximum viscosity for cold start conditions must not exceed 5000 SUS, which equates to approximately 15°F with SAE 10W fluid.

Contamination Level

Much research has been done in recent years in an attempt to develop a definition of an allowable contamination level. To date, no completely satisfactory definition has been established, especially as applied to closed loop transmission applications.

Most of the definitions are based on a controlled contaminant which is rarely encountered in actual practice.

A typical oil sample removed from an existing application will show particles consisting of silica (sand), aluminum oxide (grinding wheel or road dust), iron oxides (welding slag), iron (wear particles or machine chips), aluminum (wear particles or chips), copper (wear particles), plastic, fiber, rubber, paint, etc., in various sizes, shapes and concentrations. Some of these particles are more abrasive or heavier than others.

The following are some expressions used in defining contamination level, with a brief comment on each.

Beta Ratio: an expression of filter efficiency. A Beta 10 ratio is the ratio of the number of particles greater than 10 micron size upstream of the filter, divided by the number of particles greater than 10 micron size downstream of the filter.

Gravimetric: a measurement of the weight of contaminating particles in a given sample volume of fluid.

Filter Micronic Rating: defines the size of particle a filter will trap. Absolute rating is defined as the largest hard, spherical particle which will pass through the filter element. Nominal rating is an arbitrary value assigned by the filter manufacturer. It indicates a particle size of which the filter removes a given percentage. A 25-micron absolute rated filter will usually have about a 10-micron nominal rating, which will remove about 90% of the particles over 10-micron size.

Particle Count: an actual count of the number of particles of various sizes in a given sample volume of fluid.

Discussion

Beta Ratio serves a useful purpose in defining the effectiveness of filters but does not define a fluid cleanliness level.

Gravimetric measurement gives a measurement of cleanliness level but does not take into account shape or hardness. An exaggerated example would be to compare gravimetric measurements of lead and aluminum oxide. The lead sample would show a high reading but, because of the softness, would do little damage in a transmission. Aluminum oxide, being relatively light, would show a low reading but, because of its abrasiveness, would result in rapid wear.

Micronic rating of a filter, because it is based on a specified hard spherical particle, does not satisfy the requirements for establishing a definable contamination level. Contaminate particles are in all shapes; a long, slender particle will either be trapped or passed through the filter element depending upon its orientation.

Particle count, while it does define a cleanliness level in theoretical terms, does not give a reliable actual indication because it does not discern particle shape or hardness. It only determines size and numbers without regard to relative abrasiveness which affects wear.

NFPA Standard Test

The National Fluid Power Association standard test is generally accepted as the best available method for determining the effect of contamination on hydraulic units. However, the test is based on a single pump rotating group. Acceptance is based upon flow loss degrading to 80% of rated flow.

In a closed loop, supercharged transmission, at least two rotating groups are required to function. In addition, the useful life of such a unit is determined by the supercharge flow available to make up leakage due to wear. If flow degradation to 80% of rated flow per rotating group is used as a limit of acceptance, an intolerably large supercharge pump would be required for transmission applications. Acceptable flow degradation for a transmission (one pump and one motor) is in the order of 15-20%, and permits reasonable supercharge pump size.

In practice, the best defense against contamination failures is to get the closed loop clean at assembly (see "Installation and Operation"), provide adequate filtration of oil, and minimize contaminant ingestion through cylinder rod seals, reservoir breathers, etc.

Filtration

Filter Rating

Filtration of all fluid entering the transmission is required. The filter should have a Beta 10 rating of 10 or higher.

Filter Location

All transmissions, other than the TA19 with integral gerotor type charge pump, should have the filter located between the outlet of the supercharge pump and the supercharge inlet to the transmission. There must be a filter bypass valve to protect the filter element from rupture during cold starts. The bypass valve should be set for 10 PSI minimum and should be a nonbackwash type to prevent flushing dirt, which has collected on the element, into the transmission.

The TA19 model requires an inlet filter because the charge pump outlet is internally ported to the transmission pump circuit. The inlet filter used on the TA19 should have a bypass valve set at 5 PSI and should be a nonbackwash type. An inlet filter must not be used on a transmission equipped with a vane pump.

Pressure Rating of Filters

The filter housing must be capable of withstanding the highest surge pressure that will be encountered. Supercharge pressures of over 500 PSI have been recorded during cold starts on transmissions. Pressure surges can be minimized by proper sizing and design of case drain lines, cooler and other external circuitry.

Cooling

Cooler Types

The cooler, if required, should be the air-to-oil type, water-to-oil coolers can be used on stationary applications if they are designed so there is no possibility of cooling water contaminating the oil. Air-to-oil coolers should be plumbed with the inlet at the bottom so that any air trapped can be naturally purged from the cooler core.

It should be pointed out that some engines, especially diesels, may be supplied with an engine coolant-to-oil heat exchanger as a standard accessory. These heat exchangers are sometimes labeled as "transmission coolers". This type of exchanger is intended for use with torque converter type transmissions. It must not be used to cool hydrostatic transmissions because the coolant temperature is seldom below the allowable transmission operating temperature.

Cooler Sizing

Preliminary sizing of the cooler should be based on the horsepower loss of the transmission at the rated pressure condition. Cooling flow will be supercharge pump output at full load governed engine speed.

The normal deterioration of transmission performance over its useful life should also be considered. "Ballpark" estimates of cooling requirements can also be made by assuming that 30% of the pump input horsepower will be expended as heat. Actual cooler size required will have to be established through test of the complete machine in the environment and duty cycle in which it will be used.

Cooler Location

There is no hard and fast rule that determines location of the cooler in the circuit. The only requirement is that transmission loop temperature and case pressures be held within the prescribed limits.

Cooling of the transmission may be augmented by locating the elements where there is a free flow of air. Louvers may be put in sheet metal parts that will allow air circulation. Pumps and motors should be located so that dirt and debris will not collect and affect natural radiation and convection cooling.

Horsepower Loss

Because of mechanical and volumetric losses, there is, for any condition of transmission operation, horsepower expended simply to turn the rotating parts and move the fluid before any useful work is done. The sum total of these losses is not constant but varies with speed, displacement, pressure and fluid viscosity.

Effect of Speed: Horsepower loss due to speed is caused by friction of sliding parts and energy required to move both the fluid within the circuit itself and the fluid surrounding the rotating parts.

Effect of Displacement: Increasing pump displacement increases piston velocity and, thus, affects horsepower loss. Increased pump displacement also increases flow, resulting in increased horsepower loss. Decreasing motor displacement increases speed which increases horsepower loss.

Effect of Pressure: Increasing pressures result in increased loading of sliding parts. It also causes an increase in leakage from the main transmission loop to the case.

Effect of Viscosity: This effect is most evident at low temperature starting conditions. The variation at normal operating temperatures is usually not significant.

Other Considerations: Horsepower losses in the connecting circuits have not been included in the above. The customer can realize some economy in such losses by careful sizing and design of circuit elements.

Any time there is a loss in pressure due to unnecessary line restriction, an avoidable horsepower loss exists in the form of heat. Line sizes should be equal to or larger than the port size on the transmission pump or motor. Sharp bends and fittings should be kept to a minimum.

Line lengths should be as short as possible. If lines are long, increasing line size to reduce pressure drop may offset the cost of additional cooling. If two lines are "Tee'd" together, the area of the outlet should be at least equal to the combined area of the two inlet lines.

Fluid velocity for pressure lines should not exceed 20 ft./sec. and velocity in inlet (suction) lines should not exceed 5 ft./sec.

Case pressure should be limited to a maximum of 20 PSI as measured at the "D" port. Case pressures moderately higher than 20 PSI (up to 40 PSI) can be tolerated, but they result in decreased seal life and are an unnecessary expenditure of horsepower. That, in turn, results in more heat being generated; consequently, a larger capacity cooler is required. Case pressure can be maintained within the recommended limits by proper circuit design and, generally, will result in a more economical installation.

Case pressure must always be below the pressure on the low pressure side of the main loop, even in transient conditions. Case pressure above low loop pressure will result in shoe roll and other mechanical damage to the transmission. Extremely high spikes in case pressure (80-100 PSI) will cause permanent damage to seals.

Reservoir Design

Proper reservoir design is a most important consideration in the application of a hydrostatic transmission to a vehicle. Often, the reservoir is considered only as a receptacle for fluid with little thought as to the functions it will perform. In addition to maintaining a reserve supply of fluid, the reservoir also allows deaeration of the fluid, provides a settling basin for contamination, and can present a large surface area for dissipation of heat.

Design Considerations

Size: The fluid volume should be one-half and the total volume should be five-eighths of the maximum supercharge and auxiliary flow. If auxiliary functions are involved, the maximum and minimum cylinder volumes should be included.

Location: The reservoir should always be located above the pump inlet to maintain a positive head of oil. It should also be placed where there is a free flow of air for cooling. The ideal location is directly above the pump inlet. If located remote from the pump, it should be high enough to permit the inlet line to bleed itself of trapped air.

Outlet: The outlet port should be at least equivalent in size to the pump inlet port. If located in the bottom of the tank, it should have a standpipe that will prevent settled contamination from entering the pump. It should be located where it will always be covered with fluid regardless of vehicle attitude, effects of sloshing, or vortex effects of fluid flow in the tank.

Inlet: The inlet port is to be located so that oil enters below the minimum fluid level, and positioned so that there is no direct shunting of flow to the outlet port. It should be sized at least as large as the outlet to reduce the velocity of the fluid entering the tank.

Baffles: Baffling is essential to good reservoir design. Baffling minimizes face surface movement and provides long path circulation, which aids in deaeration and improves heat rejection capability. It also reduces the tendency to vortex at the outlet and prevents shunting of inlet oil to outlet.

Clean-Out: A clean-out panel is included in every good reservoir design. Clean-out is required to remove welding slag, rust and other contaminants when new and also for periodic maintenance.

Filler: With the exception of initial assembly or opening the system for maintenance, the reservoir filler is the primary place where dirt can enter a closed loop transmission. It should be designed so that dirt will not collect and fall into the tank when opened. The cap should be easily cleaned before removal.

Breather: A breather with filter should be provided. The filter should be of 10-micron size and have sufficient capacity for the machine environment.

Fluid Level Indicator: A sight glass is the preferred method. If a dipstick is used, it should be part of the filler cap to reduce points of dirt entry.

Drain: A drain should be provided at the bottom of the reservoir, not only to change the oil but (more importantly) to allow removal of condensed moisture at periodic intervals. This is especially critical if the machine is to be used in areas of temperature extremes and high humidity.

Sealed Reservoirs: Sealed reservoirs have no particular advantage over an open or free breathing system, but have the disadvantage of inducing air through shaft seals as the system cools. Minute particles of dirt can be carried into the seal and create sealing problems. Reservoir pressure must be controlled and limited to a few PSI above atmospheric pressure. Reservoir pressure is an additive to all system pressures and must be considered. Increased reservoir pressures also increase the amount of air in solution with the oil.

Frame Members as Reservoir: Structure members of the vehicle frame are undesirable for use as reservoirs, because it is difficult to incorporate the features necessary for good reservoir design. Cleaning is especially difficult. Most vehicle frames are made of weldments of hot rolled steel. Welding slag and scale are difficult to remove and continue to crack off as the frame deflects in service. Proper baffling is difficult to install and, if installed, only makes the cleaning process more difficult. Location is generally such that little or no head is available at the pump inlet, leading to the possibility of poor inlet conditions and air ingestion.

Screens: Use of screens on filler and outlet are recommended to prevent large particles from entering the system. Use 50-mesh screen on filler and 100-mesh on the reservoir outlet.

Mechanical and Hydraulic Resonance

Mechanical and hydraulic resonance may occur in some circuits at specific speed and pressure conditions, and are unpredictable because they are dependent upon circuit design. Operation at resonant conditions should be avoided because it can affect fatigue life of the transmission or circuit elements. If this condition does occur, speed and/or pressure conditions should be changed or the circuit modified.

Section VI – Installation and Operation

The importance of proper installation, start-up and break-in procedures cannot be overemphasized. Nearly all infant mortality failures are directly traceable to inadequate care in these areas.

Preinstallation

Inspect shaft splines and mounting flanges for evidence of damage in shipping. Lightly stone or file to remove minor imperfections. Any part that is damaged so as to create a stress riser should be rejected.

Leave shipping plugs in place until necessary to remove for assembly or installation.

Hoses and tubing should be flushed with oil at high velocity (at least 30 ft./sec.) in both directions of flow. Hoses should be inspected and slivers of rubber from skiving should be removed. Tubing should have all scale, rust and weld spatter removed, either mechanically or by pickling. Parts should be thoroughly cleaned after any welding or brazing operation. Threaded fittings and flared tube ends should be deburred to remove any lose burrs. A brush operation is frequently required to loosen dirt in hoses or tubing during the flushing and pickling procedure.

After cleaning, all openings must be sealed to prevent moisture, airborne dirt, etc., from contaminating the parts. Use plastic plugs, plastic bags or other noncontaminating closures. Do not use rags or waste. Closures should remain in place until final assembly.

Mounting of Units

Transmission units should be mounted in a horizontal position whenever possible, with the case drain in the uppermost position to insure that the case is kept full of oil. Other mounting positions are possible provided each rotating group is submerged at least halfway in oil and the case drain line is designed to prevent syphoning of fluid from the case.

Unit should install on the mounting pad smoothly and easily. Support unit as required. If spline or pilot does not fully engage, it usually indicates either improper alignment or “out of print” mating parts.

The shaft-to-pilot concentricity on the 19 size transmission units is .006 inches TIR. Squareness of the shaft axis to the mounting pad face is .0015 inches/inch. These tolerances, plus the normal shaft deflection due to load between bearings, require that any coupling between the transmission pump or motor shaft and the mating drive member be sufficiently flexible to prevent side loading of either the driving or driven member. Failure to provide this flexibility can seriously affect both shaft and bearing life.

Drive splines must be adequately lubricated.

Install all plumbing, reservoir, cooler, filters, etc. Reservoir must be located so there is a positive heat of oil at the supercharge pump inlet.

Priming is accomplished by first manually filling the inlet line and inlet port of the supercharge pump with oil. Cases of all units must be filled with prefiltered oil prior to first start-up and must be maintained in this condition at all times. Neglecting to fill cases with oil can result in instant failure and will always result in component damage. Using a clean container, remove case drain shipping plug and fill cases with oil.

Bleed air from the system by cracking the highest fitting in each circuit. Prefill main loop fluid tubes or hoses manually if possible.

Start-Up

Start engine and run at low idle. (NOTE: vane pump models may require a pump speed of 800-1000 RPM to extend the vanes; do not exceed 1000 RPM.) Monitor supercharge pressure. If supercharge pressure is not observed within 30 seconds, shut down and clear the problem.

Supercharge Pump Priming

In order for any hydraulic pump to prime, it must develop sufficient pressure to overcome the discharge load. In some cases, a pump will fail to prime on the initial start because air on the discharge side will not be compressed sufficiently to overcome the relief valve. Venting the discharge to remove air can be accomplished by loosening the “S” port fitting until the air is removed and a prime is obtained.

Some applications, though not recommended, may have the reservoir located below the inlet of the supercharge pump. In these applications, the difficulty is compounded and the possibility of loss of prime is constant throughout the life of the transmission. These applications require the installation of a foot valve (check valve) on the inlet line and located so it is always below the surface of the reservoir fluid. This valve is required to minimize draining of fluid from the inlet line, creating an air lock. It must be sized so that losses due to head and flow at pump inlet do not exceed the specified supercharge pump inlet vacuum.

Priming is accomplished by first manually filling the inlet line and inlet port of the supercharge pump with oil. If the unit does not prime on start, it will be necessary to vent the discharge as described above.

Filtration of Closed Loop

Install a 3-micron filter in the closed loop circuit and, with the machine on blocks so wheels are free, run pump at 1800 RPM one-half stroke. Circulate oil through the main loop for ten minutes. Remove filter and reconnect lines.

Break-In

The first few hours constitute the most critical period in the life of the transmission, so the severity of operation during this time will greatly influence the success of the application.

1. For the first ten minutes of operation, exercise the machine in both directions of travel at 1800-2000 RPM pump speed and at pressures up to 2000 PSI. Avoid rapid starts and stops. Correct any leaks and conditions in controls or installation that could affect operation or operator safety.

2. Continue to exercise machine for an additional twenty minutes at full pump speed and pressures up to 3000 PSI, again avoiding rapid starts and stops. Recheck for leaks.

3. Machine may now be operated at rated conditions (3000 PSI continuous), but limit operation at pressure levels above this point to not more than five applications at relief valve pressure in each forward and reverse for the next two hours of operation. Allow at least ten minutes between relief valve applications and limit applications to not more than one second.

4. Frequent checks during the break-in period to discern leaks, aeration of oil. localized high temperatures, unusual noise, fluid level, etc., should be made and the discrepancy corrected throughout the break-in period.

5. Change filter elements and examine for unusual types and amounts of contaminate. If found, determine source and correct.

The transmission is now ready for full service.

Vehicle Towing

The features of a closed loop hydrostatic transmission that make it most attractive as a device for power transmission in mobile applications, are the same features that make it difficult to move the machine if a breakdown occurs.

The motor output shaft is usually connected to the drive wheels through a fixed final drive reduction. The wheel torque imposed by towing is transmitted to the motor shaft, driving it as a pump. Because the engine is not turning, the system cannot accept the flow and the motor is essentially dead-headed. The only escape for the fluid being pumped by the motor is either over the cross port relief valve or leakage. In a very short period of time, the loop will be void of oil through leakage and resistance to rotation will cease.

From the beginning, supercharge pressure is not available, so shoes start to roll. As resistance to rotation decreases, the speed will increase, causing irreparable damage in a very short time.

When it is necessary to tow a vehicle, the following courses of action are available:

1. Tow vehicle with drive wheels off the ground. Towing dollies or heavy duty wrecker may be used.
2. Disconnect the final drive. On chain drive machines, this can be accomplished by disconnecting the chains. Most gear type wheel drives are available with means to disengage and should be used if possible.
3. Remove the wheel drive motor(s) enough to disengage the spline shaft.

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