Energy Efficient Hydraulic & Pneumatic Conference

Ernie Parker

Hennepin Technical College

13100 College View Drive

Eden Prairie, MN 55347

952-995-1569

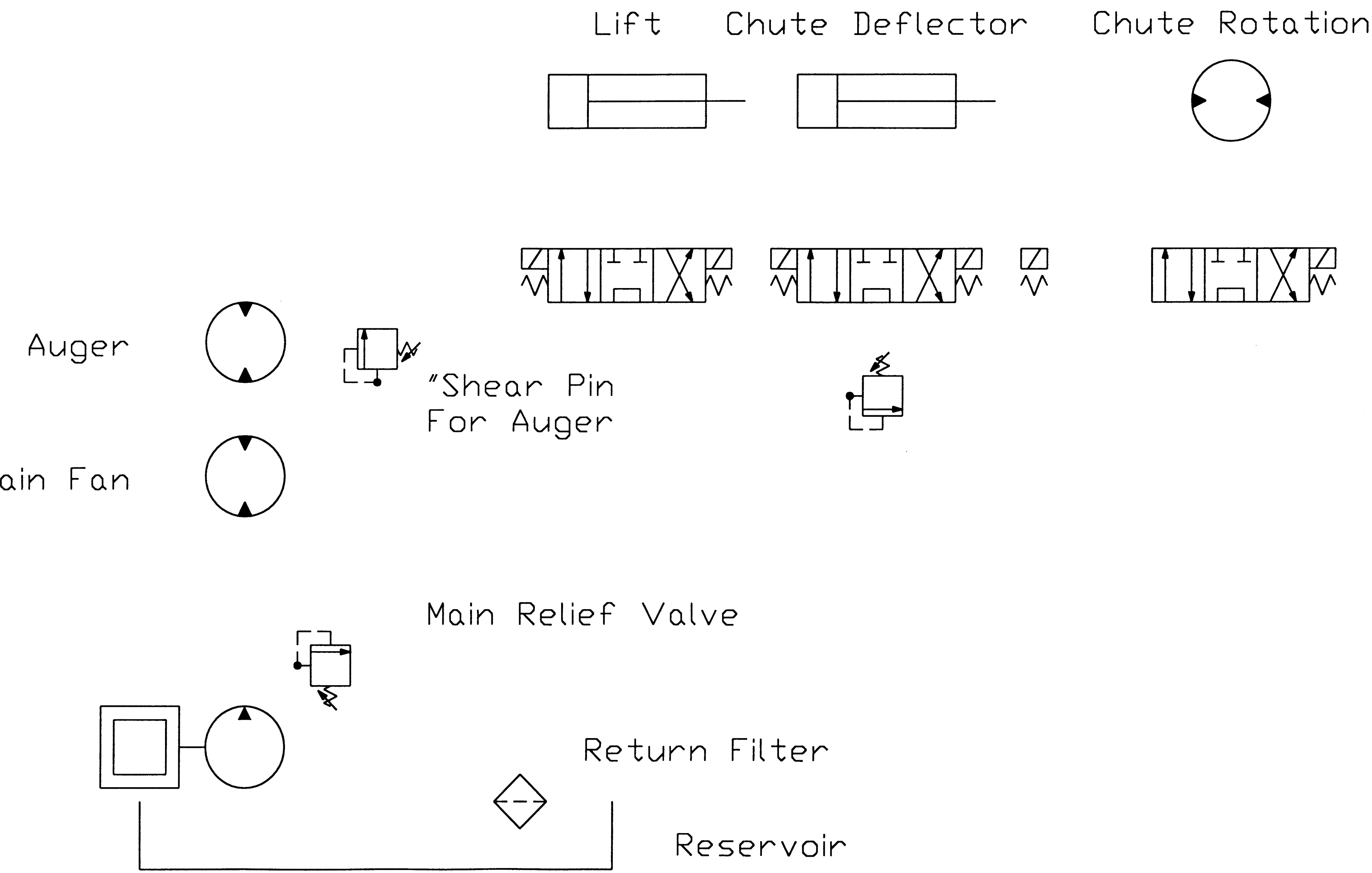
Ernie.Parker@Hennepintech.edu

Techniques for Developing Energy-Efficient Hydraulic Systems

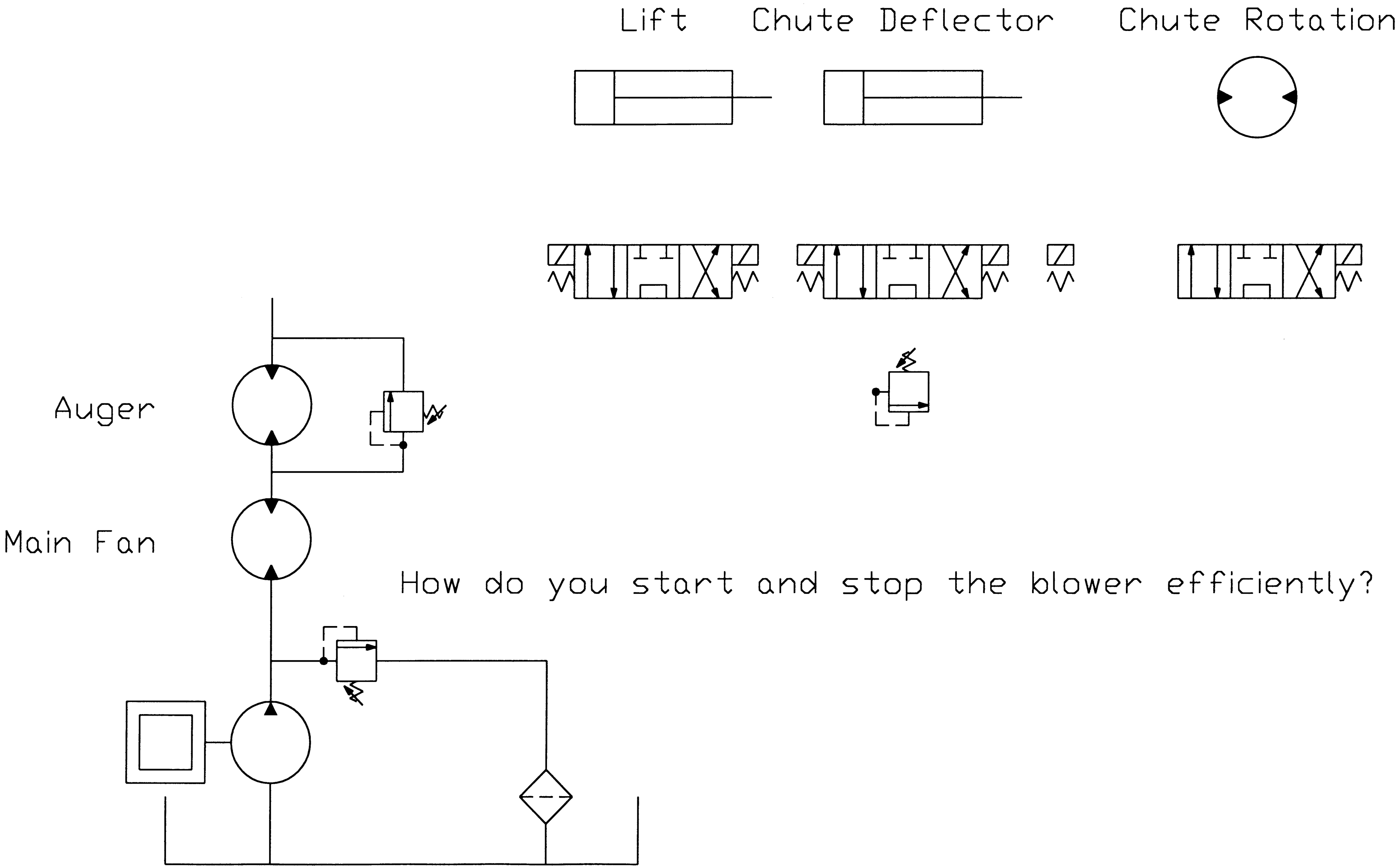
OBJECTIVE: Design a snowblower with the following specifications.

1. Top Speed: 15 MPH
2. Tire Size: 44” diameter
3. Engine RPM: 2600
4. Snowblower RPM: Fan 600 RPM Auger 300
5. Discharge Speed: 50 MPH
6. Weight of Vehicle: 1200 lbs.
7. Weight of Snowblower attachment to lift: 400 lbs.
8. Chute Rotation Speed: 180° in 2 seconds
9. Chute Deflection: Four inch cylinder in 2 seconds
10. Maximum Push Speed: 5 MPH
11. Lift Height: 12 inches
12. Zero Turn Radius: Twin hydrostatic drives
13. HP Available: 50 HP

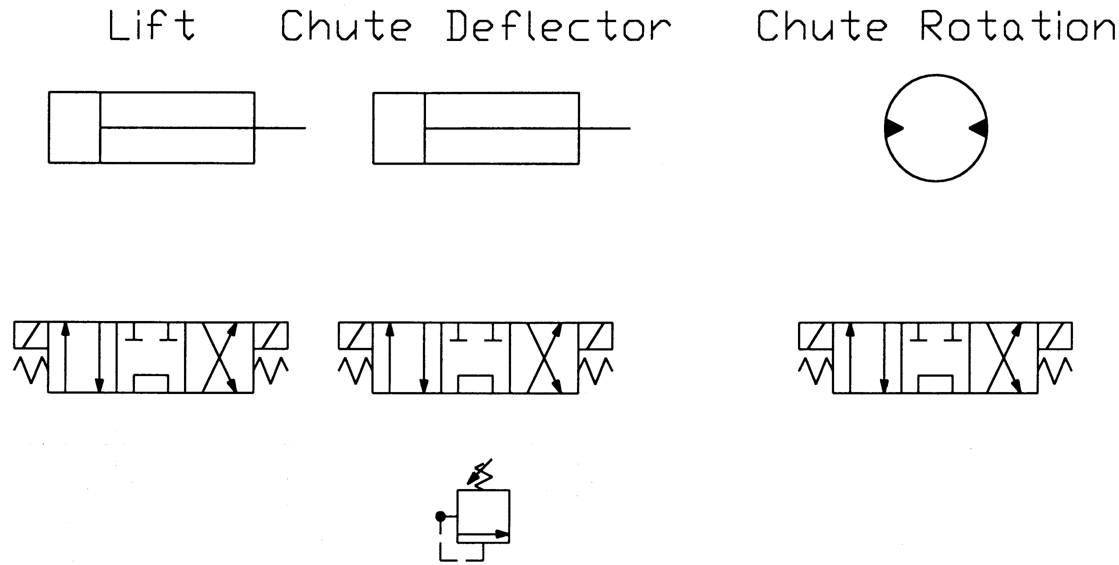
Where do we start?

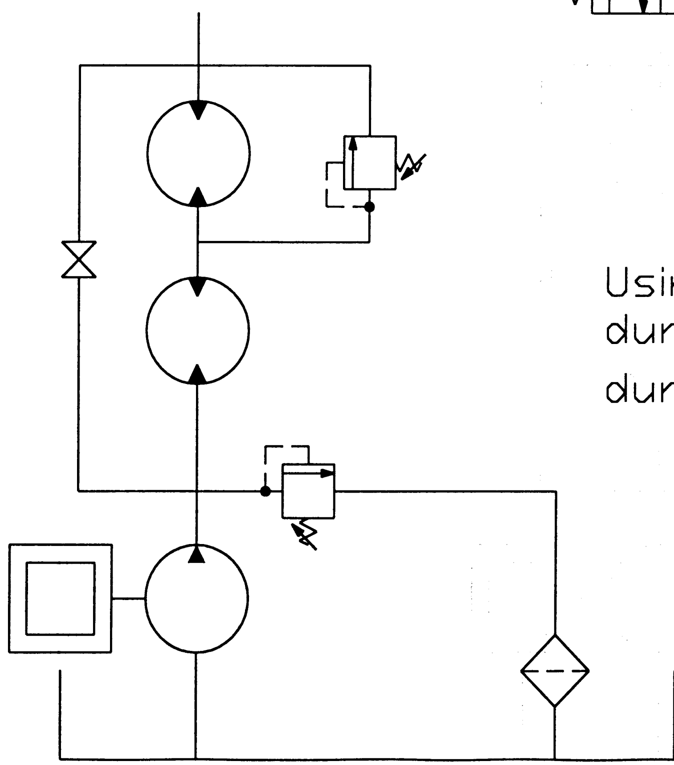

Lift
Chute Deflector
Chute Rotation
Auger
"Shear Pin
For Auger
ain Fan
Main Relief Valve
Return Filter
Reservoir

Don’t start too big. Take baby steps. No drive system yet and maybe not complete at this point.

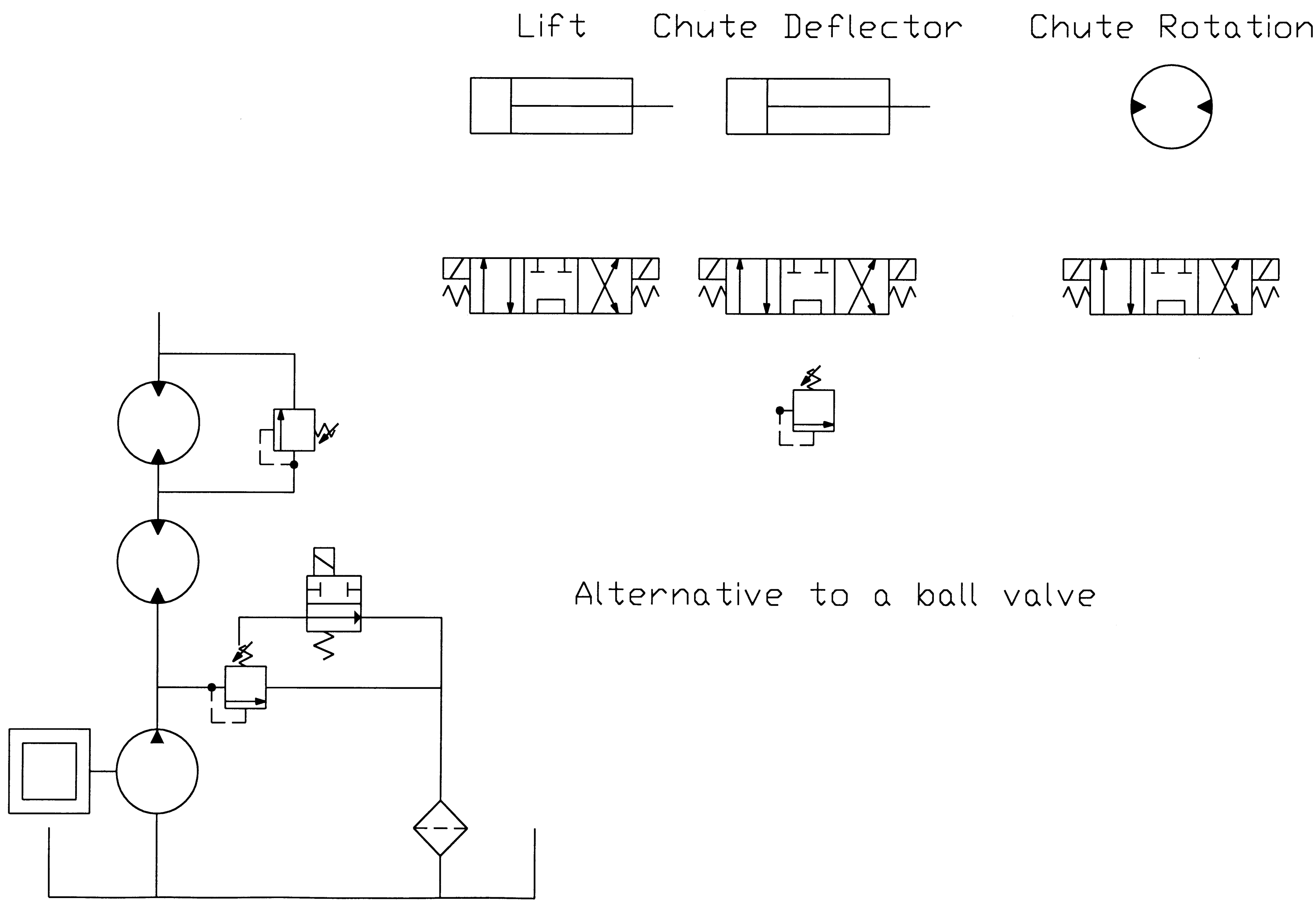


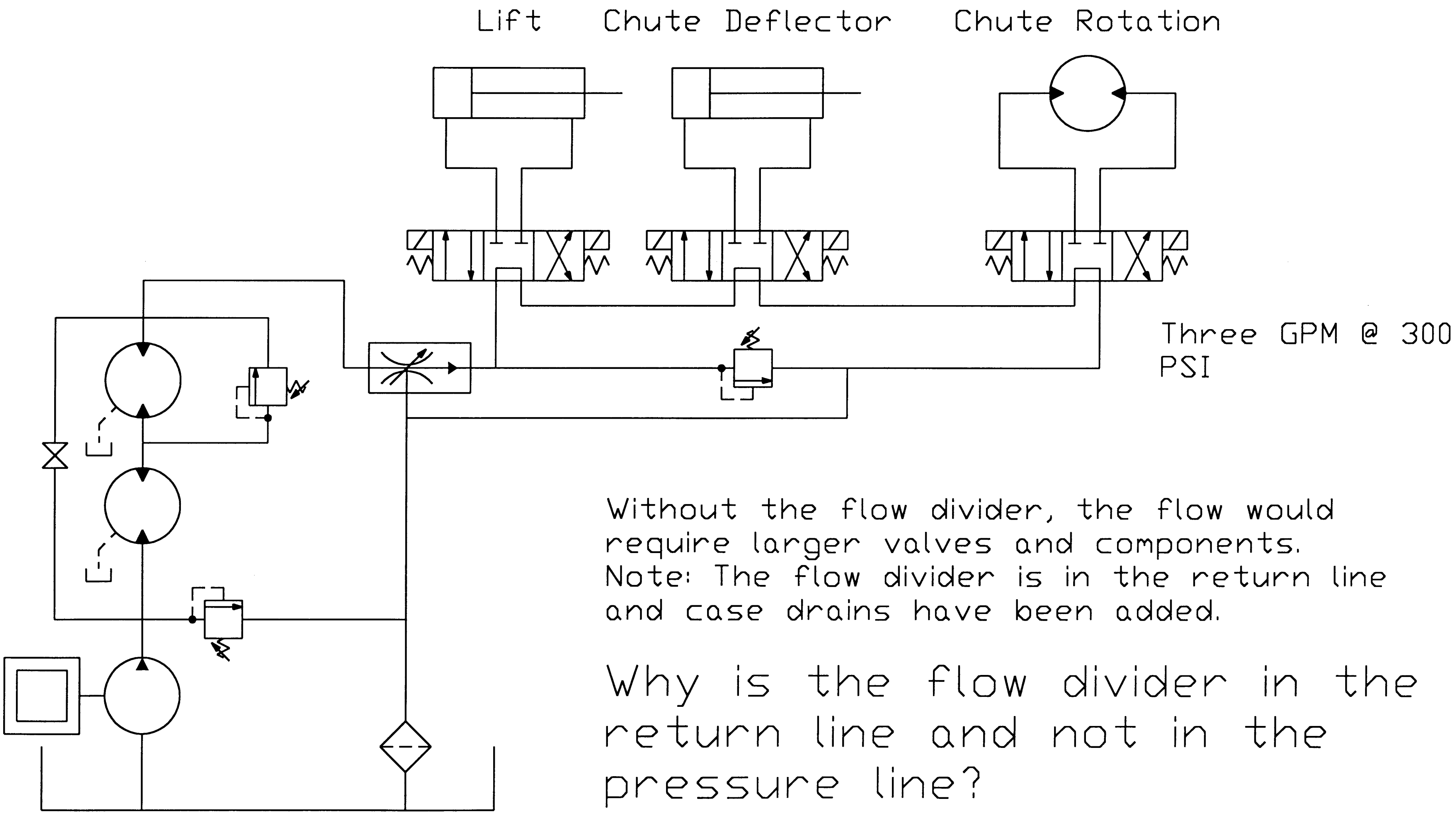
How do you start and stop the blower efficiently?





Using a ball valve has no pressure drop during operation. It prevents cavitation during shutdown. Allows for soft start.


Lift
Chute Deflector
Chute Rotation
Alternative to a ball valve



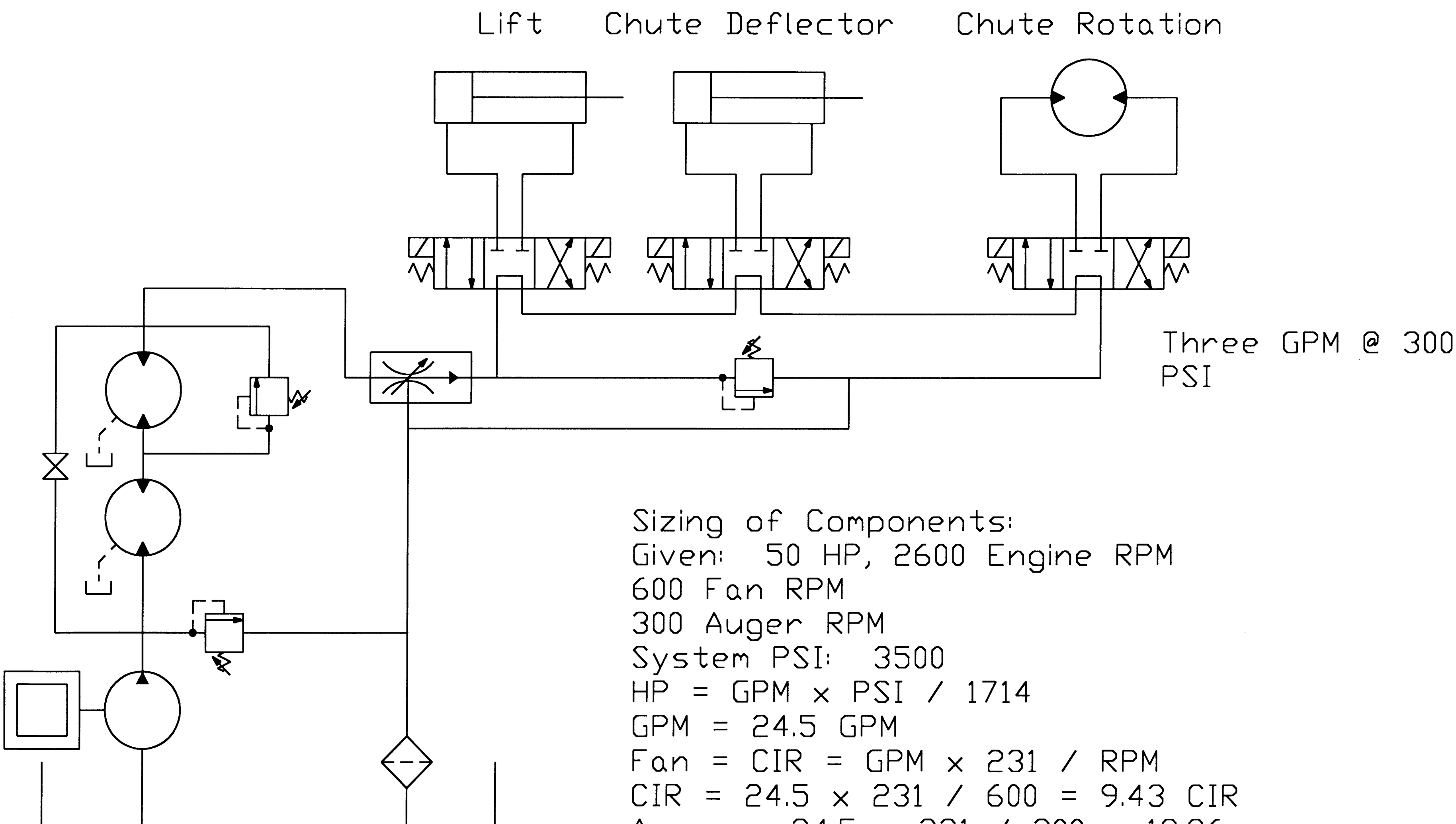
Without the flow divider, the flow would require larger valves and components.
Note: The flow divider is in the return line and case drains have been added.

Why is the flow divider in the return line and not in the pressure line?

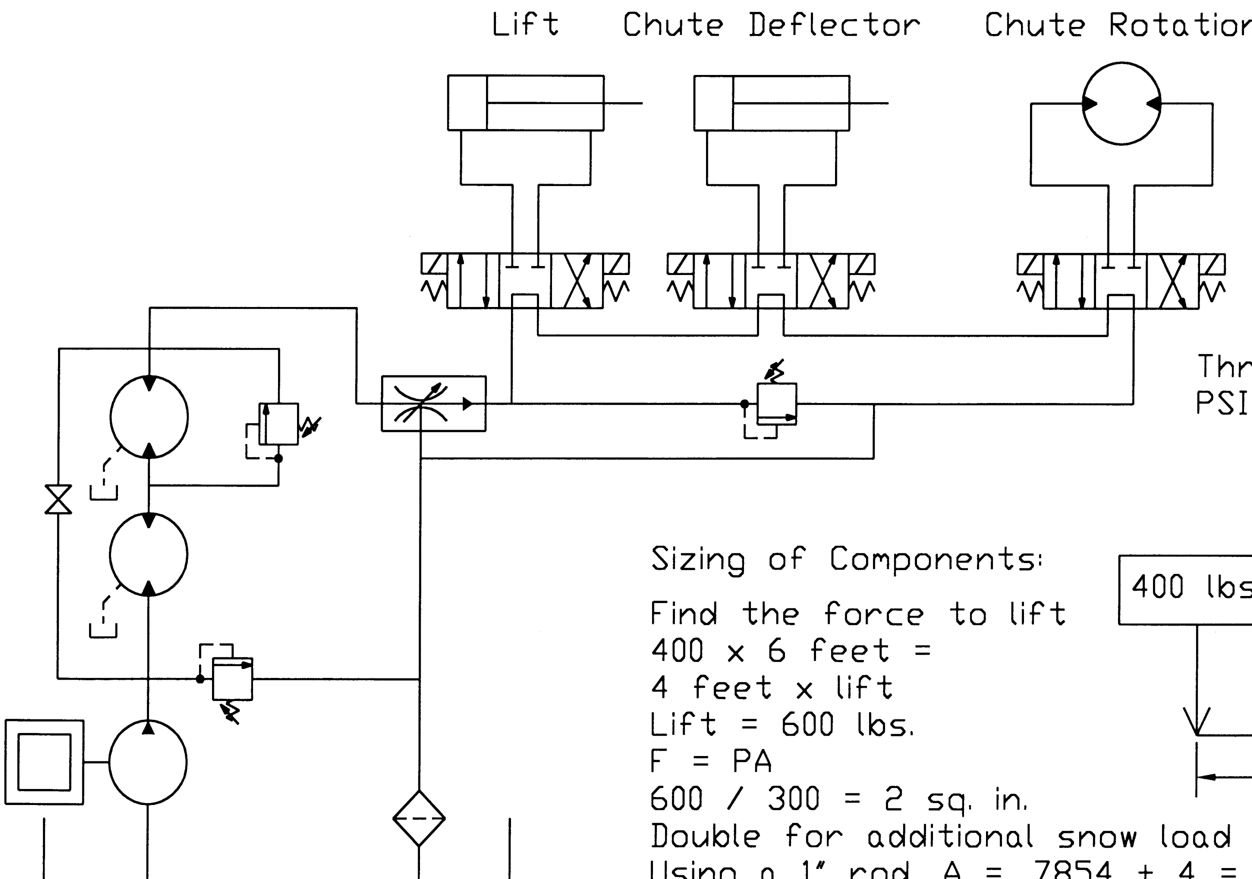
Why put the auxiliary circuit in the return line?

In the pressure line, you would be bleeding off 3 GPM at upwards of 3000 PSI when you are not using the lift or moving the chute. This is probably 95% of the time and when you do, it is only for a second or so.
Ninety-five percent of the time you would be burning energy.
3 GPM x 3000 PSI / 1714 = 5.25 HP or 13,363 BTUs / hours. If the pressure were running lower, you would waste less power.

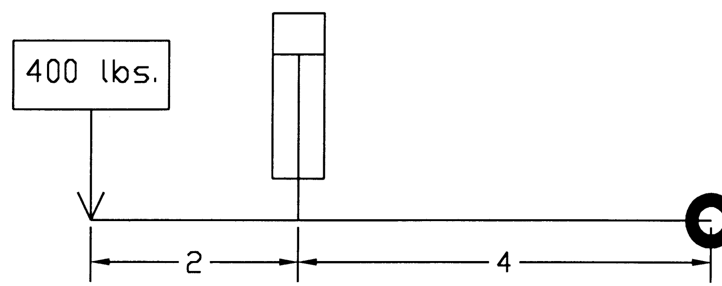
Look at the return line. The flow has the choice of going thru the priority valve and just back to tank. The back pressure may only be 50 to 100 PSI.
50 x 3 / 1714 = .09 HP or 223 watts of energy. That is less than 2% of the power or 60 times less loss then when used in the pressure line.



Sizing of Components:
Given: 50 HP, 2600 Engine RPM
600 Fan RPM
300 Auger RPM
System PSI: 3500
HP = GPM x PSI / 1714
GPM = 24.5 GPM
Fan = CIR = GPM x 231 / RPM
CIR = 24.5 x 231 / 600 = 9.43 CIR
Auger = 24.5 x 231 / 300 = 18.86
Pump = 24.5 x 231 / 2600 = 2.2 CIR

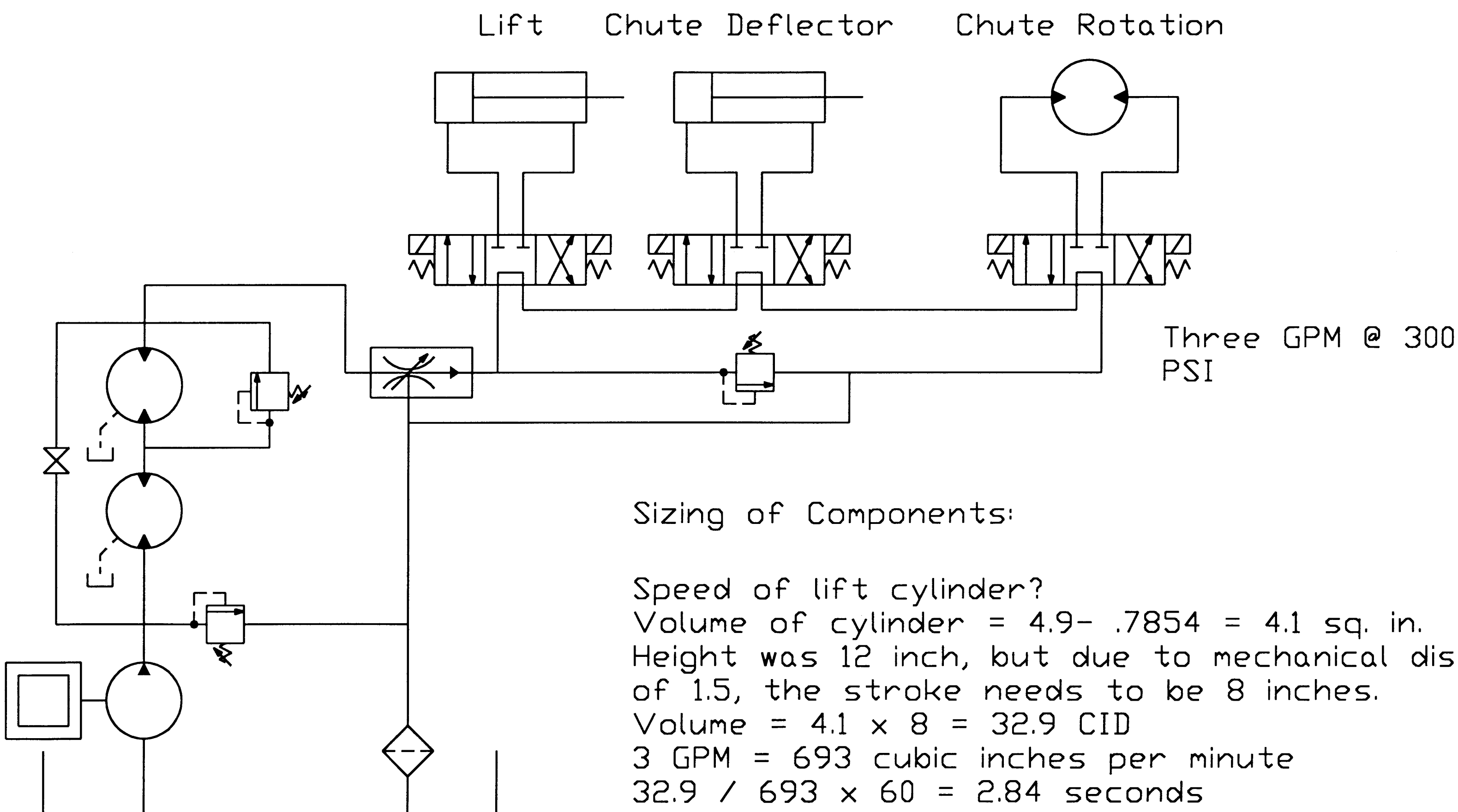


Three GPM @ 300 PSI



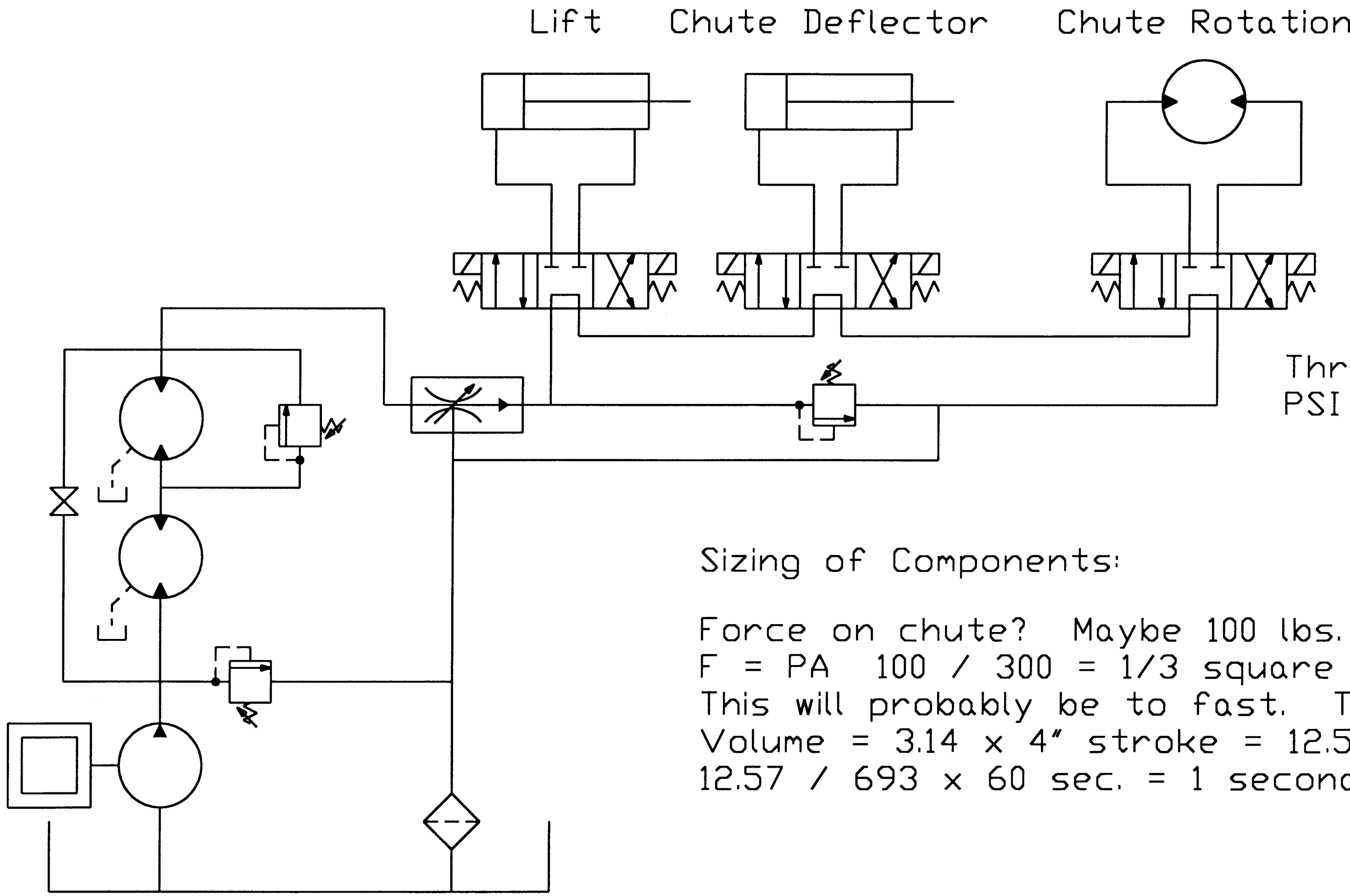
Sizing of Components:

Find the force to lift
400 x 6 feet =
4 feet x lift
Lift = 600 lbs.
F = PA
600 / 300 = 2 sq. in.
Double for additional snow load = 4 sq. in. EREA
Using a 1" rod, A = .7854 + 4 = 4.7854
A = D x D x .7854   D = 2.47 or 2.5 Inch
If 2 cylinder are used, 2.5 x .707 = 1.75 inch each



Sizing of Components:

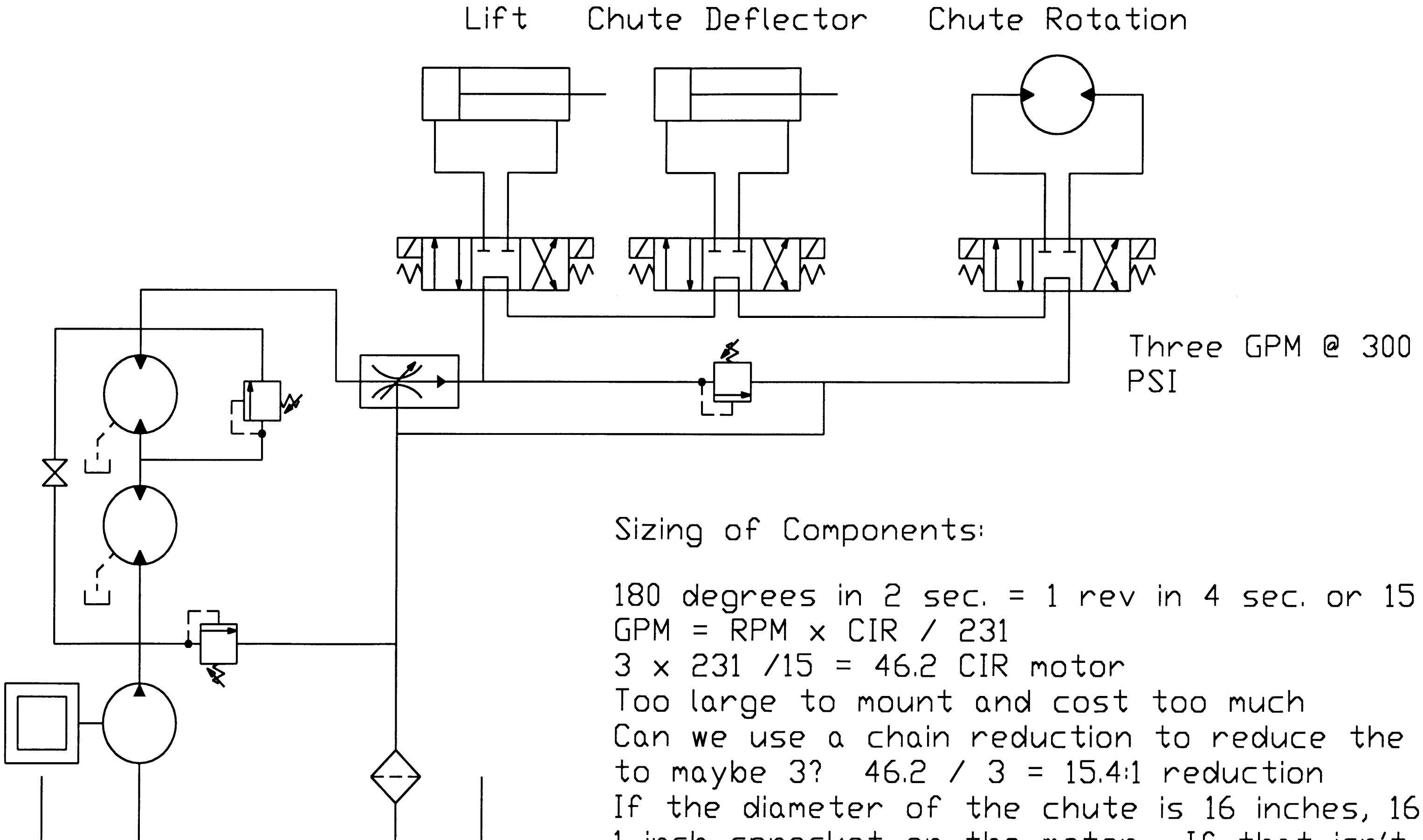
Speed of lift cylinder?
Volume of cylinder = 4.9- .7854 = 4.1 sq. in.
Height was 12 inch, but due to mechanical disadvantage of 1.5, the stroke needs to be 8 inches.
Volume = 4.1 x 8 = 32.9 CID
3 GPM = 693 cubic inches per minute
32.9 / 693 x 60 = 2.84 seconds
Is that acceptable? If not adjust at this point.



Three GPM @ 300
PSI

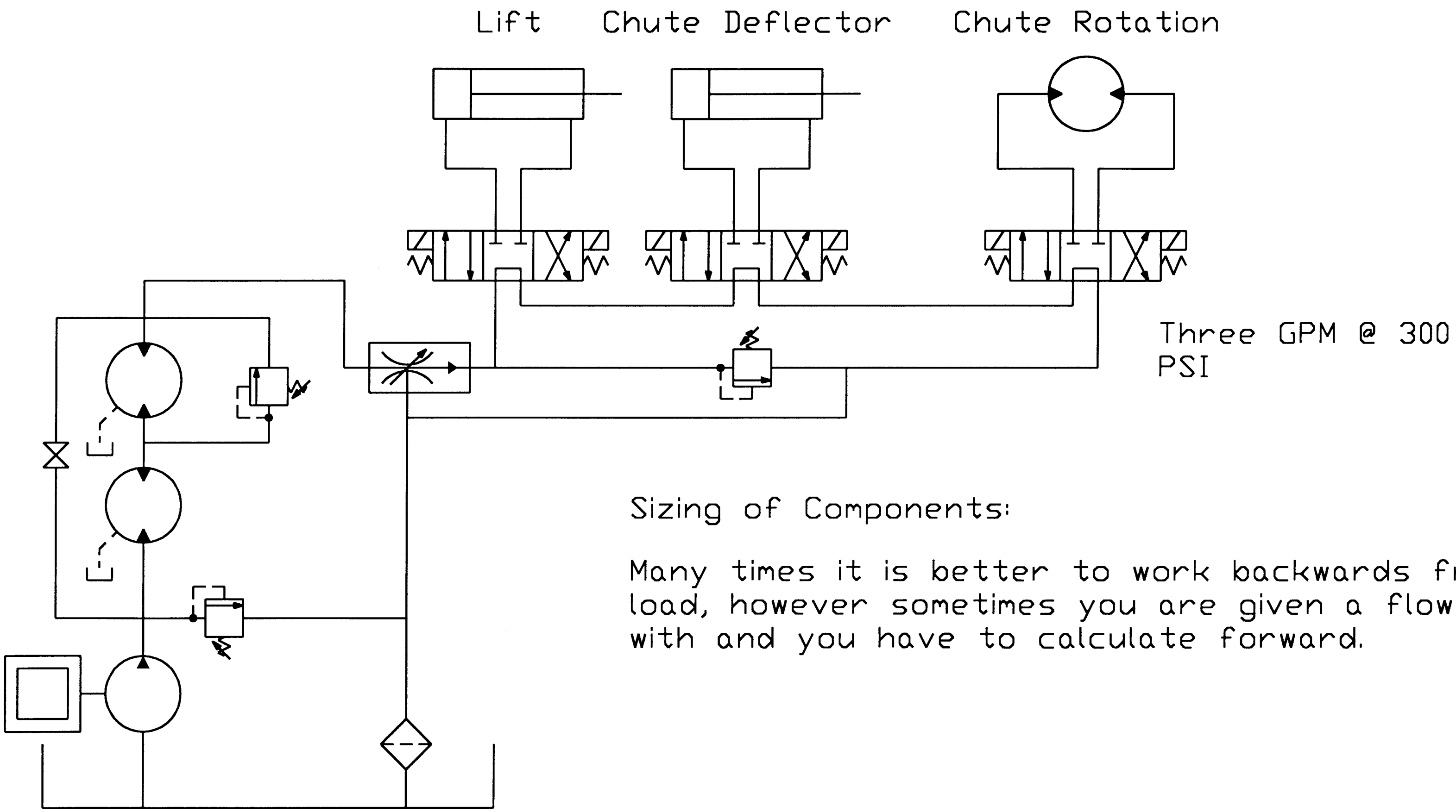
Sizing of Components:

Force on chute? Maybe 100 lbs.
F = PA 100 / 300 = 1/3 square inch = .62" diameter
This will probably be to fast. Try a 2" cylinder.
Volume = 3.14 x 4" stroke = 12.57 cu. in. needed
12.57 / 693 x 60 sec. = 1 second to deflect



Sizing of Components:

180 degrees in 2 sec. = 1 rev in 4 sec. or 15 RPM
GPM = RPM x CIR / 231
3 x 231 /15 = 46.2 CIR motor
Too large to mount and cost too much
Can we use a chain reduction to reduce the CIR
to maybe 3? 46.2 / 3 = 15.4:1 reduction
If the diameter of the chute is 16 inches, 16 / 15.4 =
1 inch sprocket on the motor. If that isn't acceptable,
change the displacement of the motor and sprocket ratio



Sizing of Components:

Many times it is better to work backwards from the load, however sometimes you are given a flow to work with and you have to calculate forward.

Sizing of lines for the Small Auxiliary Circuit

Formula: Velocity of Oil (Ft. per Sec.) = $\frac{.32 \times GPM}{\text{Net Area}}$

Let's use 15'/Second 15' x Area = .32 x 3 GPM
Area = .32 x 3 / 15
Area = .064 sq. in.

$A = D^2 \times .7854$ D = .29 inches

Note: .25 inch hose = 20 Ft./Sec. 7 PSI drop per foot with 155 SUS oil (36 CS)
.375 inch hose = 8.7 Ft./Sec. About 1.25 PSI drop / foot with 155 SUS oil

We need to make a decision as to using: ¼" or 3/8" hose?

What are the Pros and Cons?
Retraction of cylinders will produce a higher flow
Only operated for a short period of time
Flexibility
Cost
Performance
Oil temperature
Porting size
Efficiency
Heat generated

Sizing of Main System

ISO Standard: 16 Ft./Sec. for a pressure line
8 Ft./Sec. for a return line
4 Ft./Sec. for an inlet line

NFPA Standard 20 Ft./Sec for pressure and return lines
5 Ft./Sec for inlet line

SAE Standard Up to 25 Ft./Sec.

For reasons such a cost, flexibility of hoses, cost of components and conductors, payback time, life of components and oil, duty cycle, and many other reasons, I choose the 20/5 ft./second. However, it has to be a true number including retracting of cylinders and flow in tubing compared to hoses etc.

NOTE: More important than the velocity of the inlet is the plumbing. No elbows, shut off valves, strainers, etc. within the last 10 diameters going to the pump. Also flaring the pick up tubing in the reservoir is a big help. If possible, use flange fittings going into the pump for better flow characteristic. Use a hose rather than tubing to the pump. This is the most important plumbing in the entire system for performance, controllability, noise, and life of the pump as well as other components.

Sizing of Fan and Auger Motors

Given: 50 HP
600 RPM fan speed
300 RPM auger speed
50 MPH discharge speed

$$\text{Fan RPM} = \frac{336 \times \text{MPH}}{\text{Diameter of fan}} \qquad 600 = \frac{336 \times 50}{\text{Diameter}} \qquad \text{Diameter} = 28''$$

$$\text{HP} = \frac{T \times \text{RPM}}{63025} \qquad 50 = \frac{T \times 600}{63025} \qquad T = 5252 \text{ lb. inch}$$

$$\text{HP} = \frac{\text{GPM} \times \text{PSI}}{1714} \qquad 50 = \frac{\text{GPM} \times 3500}{1714} \qquad \text{GPM} = 24.5$$

$$\text{GPM} = \frac{\text{CIR} \times \text{RPM}}{231} \qquad 24.5 = \frac{\text{CIR} \times 600}{231} \qquad \text{CIR} = 9.4 \text{ in}^3$$

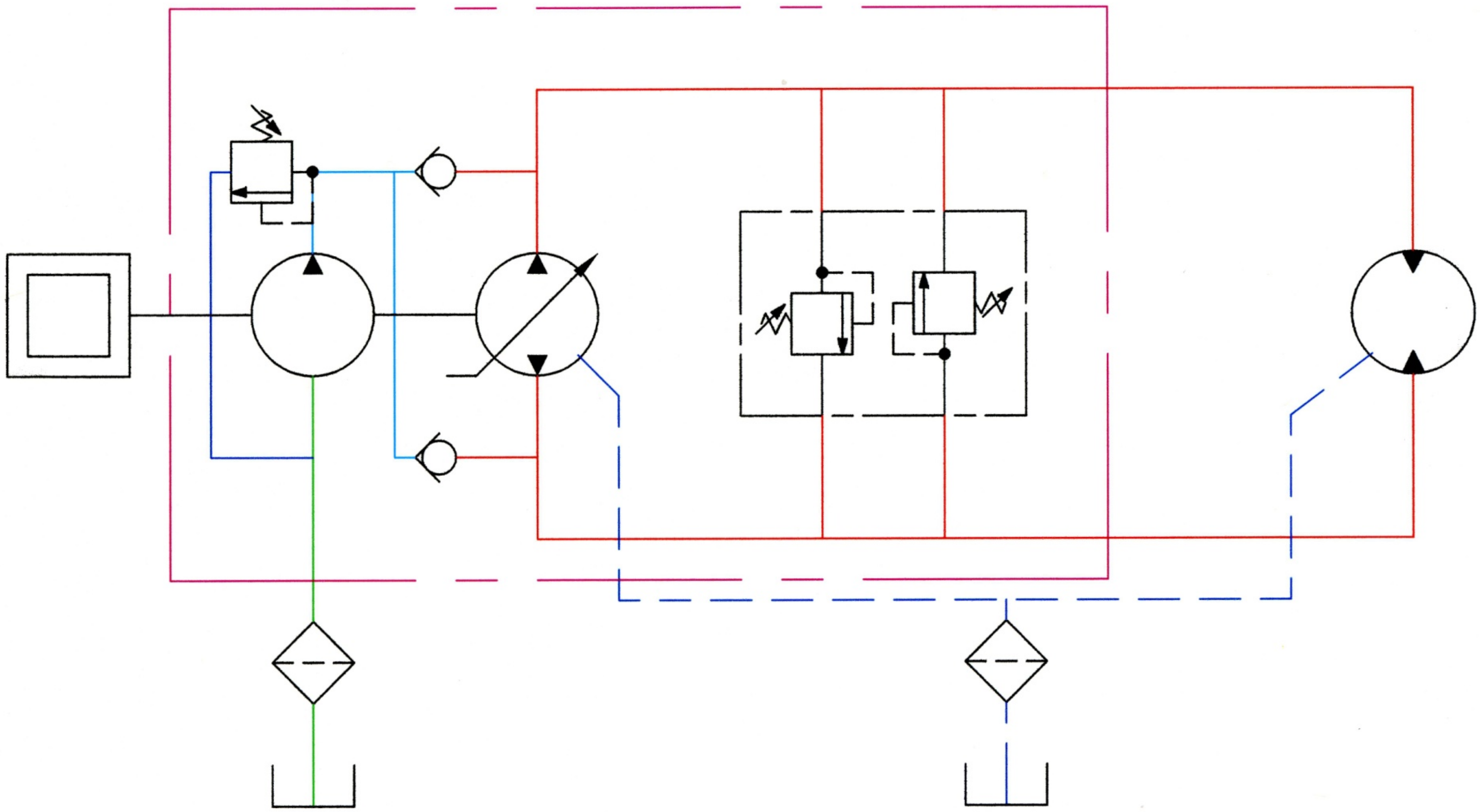
Auger CIR is ½ the speed so we will need twice the CIR because the motors are in series.

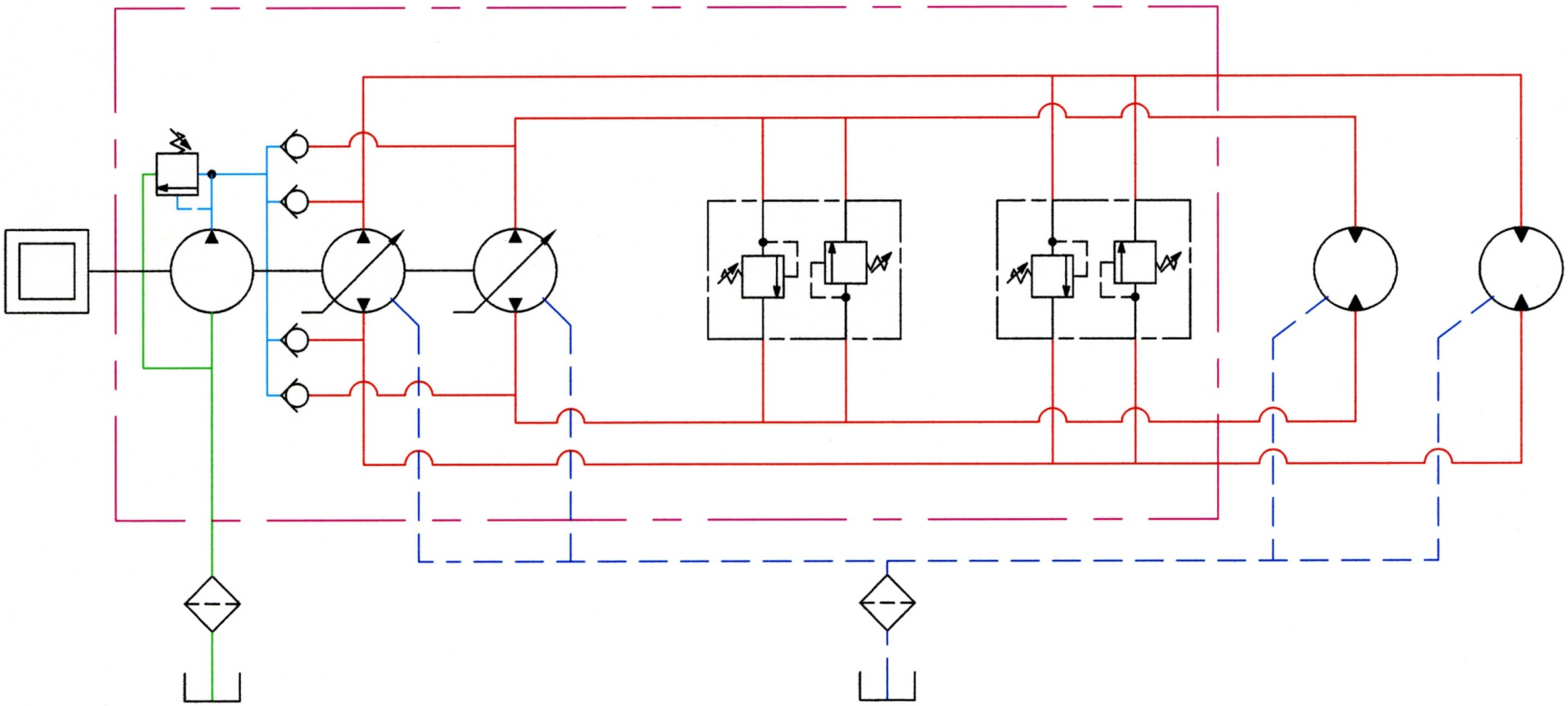
Auger CIR = 18.87 in^3

Hose size is ¾ inch

Line Sizing for System

25 GPM with ¾” hose will be at 18’ per second. Inlet to the pump will require 1-1/2” hose.





$$HP = \frac{GPM \times PSI}{1714} \qquad HP = \frac{CFM \times PSI}{229} \qquad HP = \frac{T\,(Lb.in.) \times RPM}{63025}$$

$$HP = \frac{\text{pounds of pull x Distance (in feet)}}{\text{550 x time (in sec.)}} \qquad HP = \frac{T\,(ft.lbs.) \times RPM}{5252}$$

$$HP = \frac{\text{lbs. of pull x Distance (ft..sec.)}}{550} \qquad HP = \frac{FA \times mph^3}{150{,}000}$$

$$HP = \frac{LPM \times Bars}{447.5} \qquad 1\,Ph\,HP = \frac{Volts \times Amps \times Power\ Factor}{746}$$

$$HP = \frac{NM \times RPM}{7124} \qquad KW = \frac{N-M \times RPM}{9550} \qquad 3\,Ph\,HP = \frac{Volts \times Amps \times \sqrt{3} \times PF}{746}$$

R = Radius
F = Force
A = Area
D = Diameter
C = Circumference
V = Volume
L = Length
GPM = Gallons per minute
RPM = Revolutions per minute
MPH = Miles per hour
PSI = Pounds per square inch
LPM = Liters per Minute

CID = / CIR = / Disp = } Displacement
FA = Frontal Area (Sq. Feet)
E = Volts (EMF)
I = Current (Intensity)
R = Resistance
π = 3.141592654
T = Torque
W = WATTS (Electrical Power)
NM – Newton Meters

W = EI (in phase) 1 Ph: E = IR
E = Voltage 3 Ph: Multiply x 1.732
I = Current (3) W = EI √3
R = Resistance W = Watts

Parallel Resistance: $\frac{1}{R_T} = \frac{1}{R_1} + \frac{1}{R_2} + \frac{1}{R_3}$

Series Resistance: $R_T = R_1 + R_2 + R_3$

P_2
P_3 **Absolute Values**
P_1

$P_1 \times V_1 = P_2 \times V_2 = P_3 \times V_3$
$P_1 \times V_1 = P_2 \times V_2$ = solve for V_2
$P_2 \times V_2 = P_3 \times V_3$ = solve for V_3
$V_3 - V_2$ = useable oil

7.48 gallons = 1 cubic foot
1 Micron = 39/1,000,000 of an inch
1 Micron = .000039"
1 in. Hg. = .49 PSI
231 Cubic inches / Gallon
1728 Cubic inches / Cubic Foot
62.4 pounds of water / Cubic Foot
8.3 pounds / gallon of water
490 pounds per cubic foot of steel (.283 #/in^3)
1 mile = 5280 feet

KPA = Kilopascal
0 PSIG = 14.7 PSIA
1 HP = 2545 BTU's/hr.
1 HP = 746 Watts
1 MPH = 1.47 feet/sec.
60 MPH = 88 feet/sec.
1 PSI = 27.69" H_2O
1 PSI = 2.03" HG
Absolute Zero = -460°F
Absolute Zero = -273°C

Extension only
Cyl travel rate (inches/sec.) = $\frac{4.9 \times GPM}{D^2 \text{ of cylinder}}$

Extension only
Cyl travel rate (inches/sec.) = $\frac{36.655 \times CFM}{D^2 \text{ of cylinder}}$

Cyl Ext. time (sec.) = $\frac{\text{Stroke (inches)}}{\text{Travel rate (Sec.)}}$

Cyl Ext. time (sec.) = $\frac{\text{Vol. of cyl (CID)} \times .26}{GPM}$

Retract time (sec.) = $\frac{EREV \times .26}{GPM}$

TIME FOR AN OBJECT TO FALL

H = .5g x T^2

H = Height in feet
G = 32.16
T = Time in seconds

Maximum U-Joint Operating Angle = $\frac{\text{Shaft Length}}{5}$
Angle = Degrees
Shaft Length = inches

F = PA
A = D^2 x .7854
A = πr^2
V = AL
C = πD

Lbs. In. $T = \frac{PSI \times Disp(CID)}{2\pi}$ Lbs. Ft. $T = \frac{PSI \times Disp(CID)}{24\pi}$

T = Radius x Pull

GPM = $\frac{RPM \times Disp.(CID)}{231}$

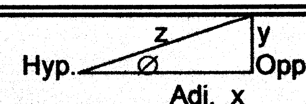
CIM = CIR X RPM

$RPM = \frac{336 \times MPH}{\text{Dia. of wheel (inches)}}$ $RPM = \frac{229 \times Ft/Se}{\text{Diameter (Inches)}}$

$\text{Velocity of oil (Ft./Sec.)} = \frac{.32 \times GPM}{\text{Net Area}}$

Use absolute values for PSI & Temp.
$P_1V_1T_2 = P_2V_2T_1$

$x^2 + y^2 = z^2$



$$Sin = \frac{\text{Opposite side}}{\text{Hypotenuse}}$$

$$Cos = \frac{\text{Adjacent side}}{\text{Hypotenuse}}$$

$$Tan = \frac{\text{Opposite side}}{\text{Adjacent side}}$$

Metric	
	1 in^3 = 16.38cc
14.5 PSI = 1 Bar	9.8N = 1 Kg
3.785 Liter = 1 Gallon	4.448N = 1 lb.
2.54 cm = 1 inch	2.2 lbs. = 1 Kg
1 MPA = 145 PSI	
16.37cc = 1ci	6.895 KPA = 1 PSI
100 KPA = 14.5 PSI	1.36 NM = 1' #
1PSI = 51,699 micron	25.4mm = 1"

Acceleration

$$Force = \frac{GVW\,(lbs.) \times MPH}{22 \times Time\,(Sec.)}$$

$$Force = \frac{Weight\,(\#) \times Feet/Sec.}{32.2 \times Time\,(Sec.)}$$

Sizing of the Hydrostatic Drive System

Given: Top Speed: 15 MPH at 1/3 of the pull
Maximum Pull: 5 MPH
One pump drives one motor

Calculations at 5 MPH

Wheel RPM = $\frac{336 \text{ x MPH}}{\text{Diameter}}$ $\frac{336 \text{ x } 15}{44}$ = 114.5 RPM

HP = $\frac{\text{Torque x RPM}}{63025}$ 25 HP per side 38.17 RPM

Torque = 25 x 63025 / 38.17 = 41,279 lb. in. per side

Torque = Radius x Pull

41,279 lb. in. / 22 in. = 1876 # of pull per side

1 MPH = 1.47 ft. / sec. 5 MPH = 7.33 Ft. / Sec. speed

Check Math

HP = $\frac{\text{Pounds of Pull x Ft. / sec.}}{550}$ HP = $\frac{1876 \text{ x } 7.33}{550}$

HP = 25 per side

Hydrostatic Sizing for High Speed

Torque is 41,279 / 3 for high speed   Torque = 13,759.67 # in.
PSI = 3500 / 3 for high speed   Pressure = 1166.67 PSI

$$T = \frac{CIR \times PSI}{2\pi} \qquad \frac{13{,}759.67 \times 2\pi}{1166.67} = 74\ CIR$$

Calculate GPM

$$GPM = \frac{RPM \times CIR}{231}$$

Use RPM for 15 MPH = 114.5 RPM

$$\frac{114.5 \times 74}{231} = 36.7\ GPM$$

Check Math

$$HP = \frac{GPM \times PSI}{1714} \quad \text{Low Speed} \quad \frac{36.7 \times 1166.67}{1714} = \textit{25 HP / Side}$$

$$HP = \frac{GPM \times PSI}{1714} \quad \text{High Speed} \quad \frac{12.32 \times 3500}{1714} = \textit{25 HP / Side}$$

Sizing of the Hydrostatic Hoses

$$\text{Ft. / Sec.} = \frac{.32 \times \text{GPM}}{\text{Net Area}}$$

$$\text{Area} = \frac{.32 \times 36.7}{20}$$

Area = .59 square inch.

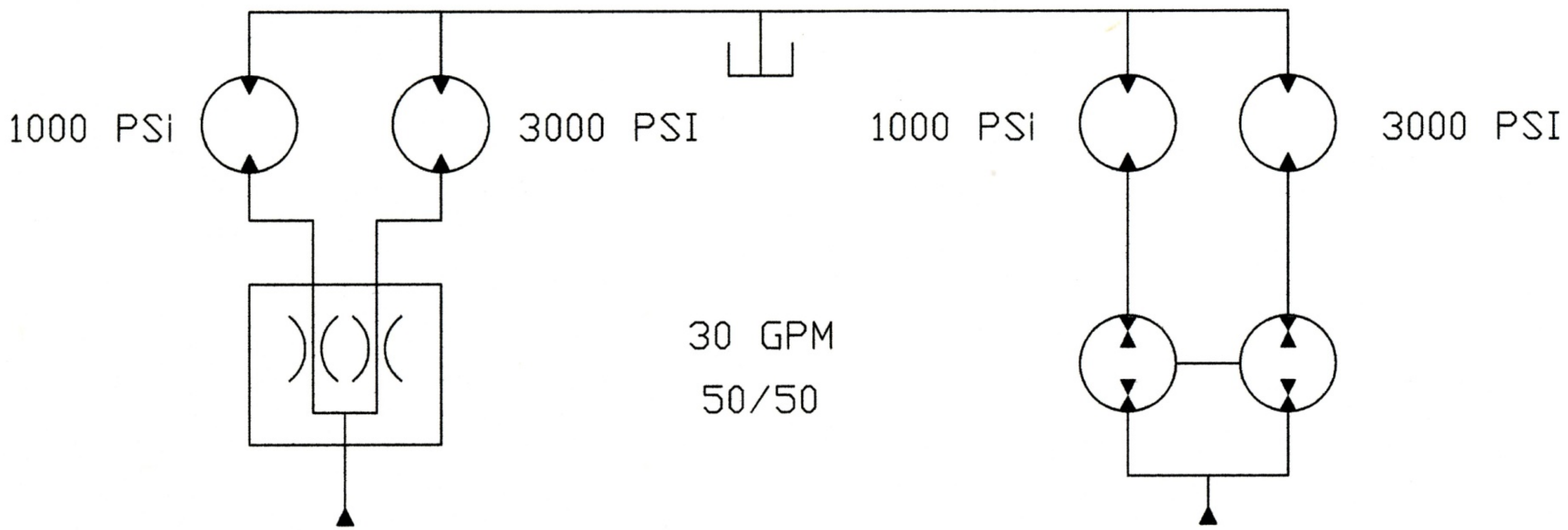
$A = D^2\ .7854$

Diameter = .86 inch *Use 1 inch hose*

1" hose will = 15 feet per second

Alternative Circuit Design

Flow Control Versa Proportionator

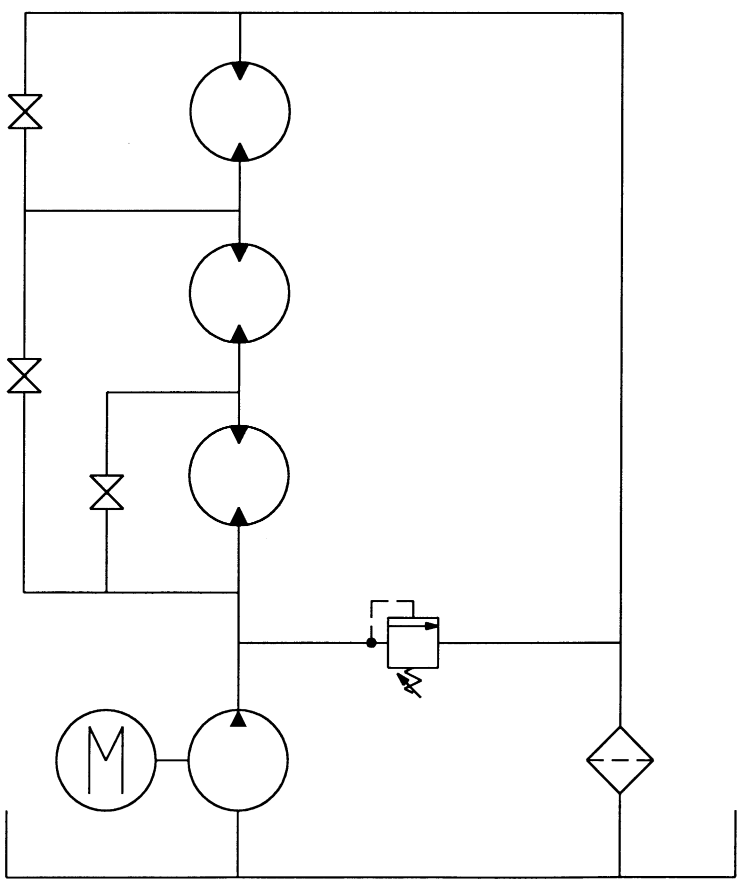


Required system pressure is 3000 PSI. Pressure drop across left side of flow control is 2000 PSI. Loss energy is 2000 x 15 / 1714 =

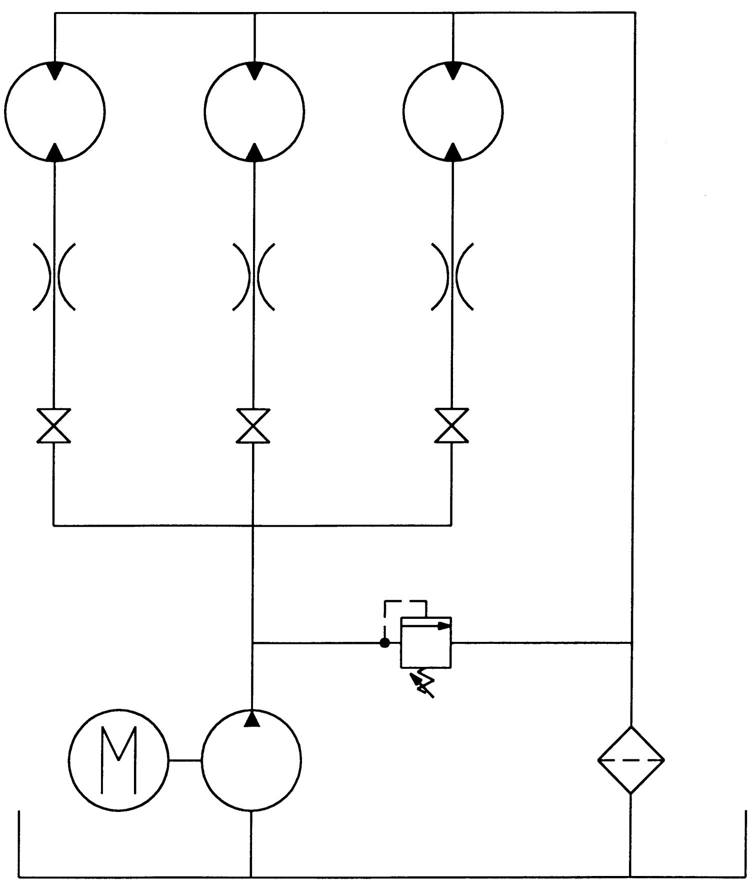
17.5 loss HP or 44,545 BTUs per hour. At $.11/hour and at 75% efficiency = $1.93/hr. Real loss will be greater.

Left side = 1000 PSI @ 15 GPM
or 8.75 HP.
Right side = 3000 PSI @ 15 GPM
or 26.25 HP
Total HP = 35 HP
HP = GPM x PSI / 1714
35 x 1714 / 30 = 2000 PSI
2000 PSI for system pressure
No loss of energy

Series



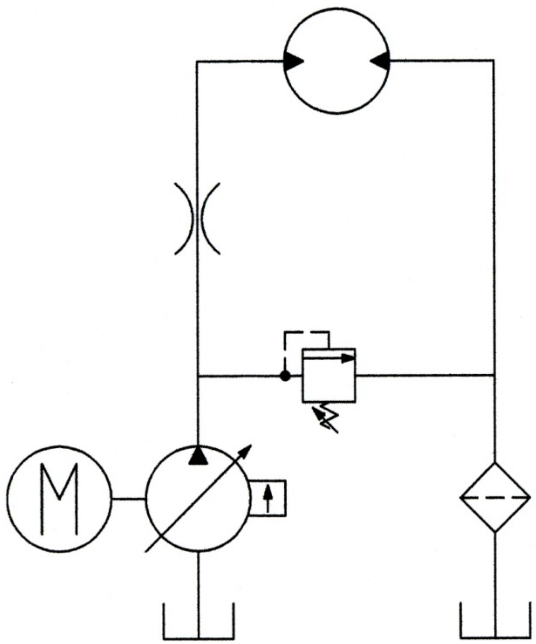
Parallel

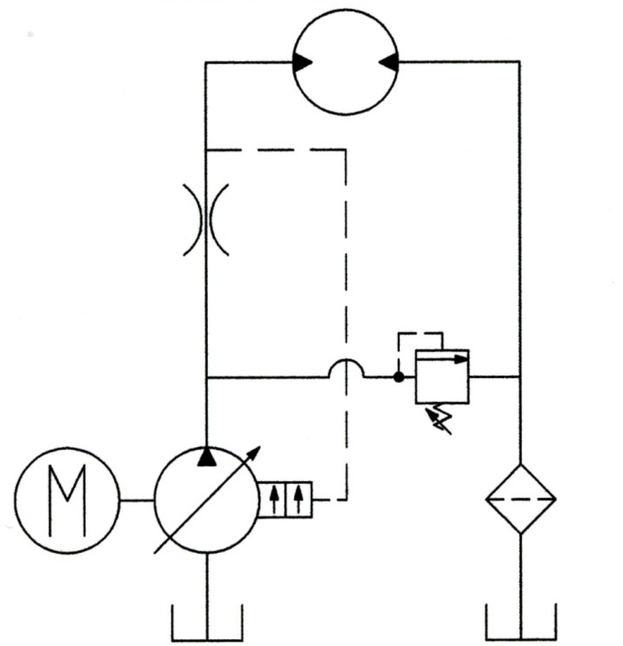


Load - 1000 PSI
Flow - 10 GPM

Lost HP
2000 PSI drop
@ 10 GPM =
11.67 HP

Lost HP
200 PSI drop
@ 10 GPM =
1.17 HP





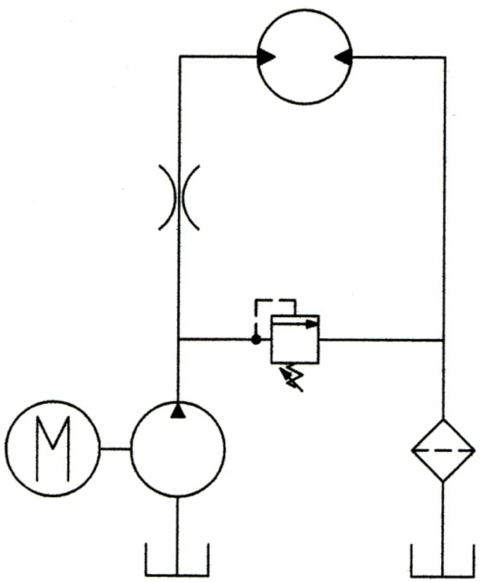
System Pressure - 3000 PSI
System Max. Flow - 30 GPM
Load Sensing Differential 200 PSI

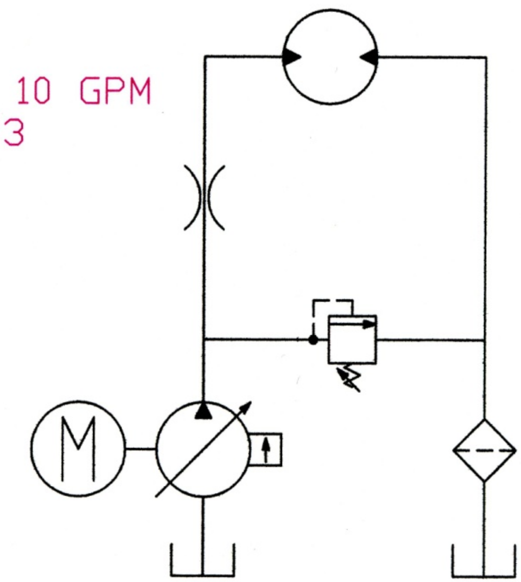
Load - 1000 PSI
Flow - 10 GPM

Flow Control - 2000 PSI @ 10 GPM = 11.67 HP
Relief Valve - 3000 PSI @ 20 GPM = 35 HP
Input HP = 30 GPM @ 3000 PSI = 52.5 HP
Efficiency 5.83 / 52.5 = 11%

Flow Control - 2000 PSI @ 10 GPM = 11.67 HP
Relief Valve = 0 GPM = 0 HP
Input HP = 10 GPM @ 3000 PSI
Input HP = 17.5 HP
Efficiency 5.83 / 17.5 = 33%

Motors - 1000 PSI @ 10 GPM
Work HP = 5.83





System Pressure - 3000 PSI
System Max. Flow - 30 GPM

Hydraulic Systems Biggest Enemies

Temperature

Running temperature should generally be between 120° F to 140° F with the right viscosity oil (100 – 300 SUS).

Too high of viscosity:

- High pressure drop
- Low mechanical efficiency
- High back pressure
  - Causes shaft seal problems
  - Bursts heat exchangers & filters
- Cavitation
- Poor lubrication

Hydraulic Systems Biggest Enemies

Temperature

Running temperature should generally be between 120° F to 140° F with the right viscosity oil (100 – 300 SUS).

Too low of viscosity (too hot of oil):

- High internal leakage
- Low volumetric efficiency
- Dangerous – One could get burned from hot components or oil leak
- Shortens seal life
- Loss of horsepower, torque and speed
- Poor lubrication
- Gerotor motors will fail with less than 70 SUS
- Draws in more moisture when cooling down
- Oil does not filter as well
- Low oil levels and build up or dirt and grease on reservoirs and components will cause heat

Hydraulic systems Biggest Enemies

Filtration

Poor Filtration has been given lots of lip service, but far too often we fail to do the basics:

1. Filter new oil and fill the case of pumps and motors
2. Use filters without indicators or by-passes
3. Use filters in inlets causing cavitation
4. Overlook the importance of a good reservoir breather
5. Missing return filters and/or pressure filters when needed
6. Not using fine enough filters. 10 micron doesn't really get the job done.
7. Use of kidney loop (off-line) filtration
8. Lack of moisture control
9. Changing filters too often or not enough based on hours, not indicators
10. Plugging hoses and ports with a grease rag

Hydraulic Systems Biggest Enemies

Filtration

11. Mounting strainers too close to the inlet of a pump
12. Flushing a new circuit without taking sensitive components out of the loop
13. Running dirty oil in production and repair test benches. Excellent place for a kidney loop
14. Leaving quick disconnects uncovered and then plugging in mating halves
15. Using a return line filter only on pressure compensated pumps that are in stand-by most of the time
16. Incorrect sizing of filters on return lines a not considering the return line flow
17. Failing to bleed filters to get full usage for the filter
18. Using cheap filters
19. Cleaning around the filter before changing
20. Mount filters vertical if possible and easy to get to

Hydraulic Systems Biggest Enemies

Cavitation

Proper inlet plumbing can solve a lot of problems in our hydraulic systems. With good plumbing, there are a number of problems that can be solved or reduced. Here are just a few:

System will run cooler
System will be much quieter
Better control of system
Faster response times
Longer pump life
Higher volumetric efficiency
Lower operating cost
Filters will last longer
Less aeration if shaft seals are weak
Higher production
Lowers back pressure capabilities of case drains
Leakage internal and external increases
Foaming in reservoir
Increases size of coolers
Higher pressure pulsations in system
Reduces lubrication

Hydraulic Systems Biggest Enemies

Proper Inlet Plumbing

Proper plumbing is far more important than appearance.

Stay away from the pump inlet port with any kind of turbulence for the last 10 diameters of the hose feeding the pump.

This means no top reservoir inlet filters drawing oil from the tank, thru the filter and a right turn right into the inlet of the pump.

No elbows, strainers, shut off valves, temperature probes, tees or anything but a clean hose of proper size going straight into a flange fitting if possible. If needed use a sweeping elbow. Illuminate any steps that will cause an interruption of laminar flow.

Hydraulic Systems Biggest Enemies

Proper Inlet Plumbing

Flooded inlets from reservoirs are the best.

Don't use pipe for inlet pickups. Use tubing that the end of the tubing is flared. This will act as a venturi and fill up much better than a pump submerged under the oil without any plumbing at all. That will act somewhat like a sharp edge orifice and vortex and reduce the intake flow. A normal bend in the inlet tubing is ok to use. Over sizing the inlet tube is ok as long as the hose going to the pump is at least 10 diameters long.

Hose should be used to reduce vibration and has a larger ID. Hose should also be used leaving the pump. It can act as a poor man's accumulator to help absorb a pressure spike until the relief valve or compensator can react.

Electric Motor & Engine Sizing

Sizing electric motors and engines is another mistake that is made often.

Under sizing is hard on a motor and will increase the amp draw and will shorten the life of the motor due to an increase of heat. Under sizing an engine will shorten it life and greatly increase fuel consumption

Over sizing a motor is a waste of money and energy. Electric motors under no load will draw about one half of the full load amps and it puts the power factor out of phase. Also there is a higher in-rush on startup and many times the electric bill for the month is based on peak usage. Why pay for increase rates when the startup time is so short? Consider soft starter controls or unloading the hydraulic system during startup.

Electric Motor & Engine Sizing

Check the true voltage that you are using for your electric motor. Running a 230-volt motor on 208 volts is within the 10% allowable limit for the motor, but if you need a true five horsepower for your circuit, you will probably only see four horsepower under those conditions. Use a motor truly rated for 208 volts and try to size the motor to use at least 80% of its rated horsepower.

As for small air-cooled gas engine, size that your load is about 70% of the rated power continuous and 85% intermittent duty. For larger liquid cool engines, try to load around 80% of rated horsepower for maximum efficiency. Overload will cause excess pollution and under burned fuel. Over sizing will also cause a waste of fuel and loading up the engine with carbon etc. One of the ways to better match engine to load is by adjusting RPM to put out the proper amount of horsepower need for the job.

Closing

The average hydraulic is about 20% efficient. This provides us with lots of opportunity to increase profits and save company many dollars with some very basic things and many of them, especially with new systems to greatly improve over the old design with minimal additional cost. There isn't any reason that hydraulic systems can't be running between 75% & 85%. The increase in efficiency will pay for the initial cost in short time.