HEAVY MOVABLE STRUCTURES, INC.
FOURTEENTH BIENNIAL SYMPOSIUM

October 22 – 25, 2012

PROGRESSIVE CUSHIONING FOR HYDRAULIC CYLINDERS

Michael Hanley
Electro Hydraulic Machinery Co.
www.ehmcompany.com

CARIBE ROYALE HOTEL
ORLANDO, FLORIDA

Introduction

You may be asking “What is a hydraulic cylinder cushion and why do I need one?” The answer is surprisingly simple and yet important to the safety of all heavy moving structures. Hydraulic cylinder cushions have been an inexpensive option available on pneumatic and hydraulic cylinders for over 100 years. It gets down to a simple additionally machined piece attached to the cylinder rod that has the ability to decelerate a cylinders’ velocity during the last few inches of stroke. This deceleration process is automatic, adjustable, and requires no maintenances. It’s an elegant solution that is very cost effective if properly executed.

To answer why you need one you first must have a very large and heavy moving structure like a flood gate, navigational lock gate, or movable bridge span that is powered by a hydraulic system. For mechanically operated heavy moving structures there are other hydraulic cushioning devices that accomplishing the same objective. They are called hydraulic shock absorbers and are not part of the discussions in this paper.

So if you have or are designing such a system the obvious need to decelerate and stop the structure from moving at each end of the cycle can be accomplished in many ways. Gradually reducing hydraulic fluid flow near the end of stroke is achieved with various valving schemes but they all rely on some form of electrical control system or limit switch to initiate and control the deceleration process. The problem is sometimes limit switches and control systems fail to operate. There’s little need to explain the effects of what happens when a heavy structure fails to decelerate and slams into its abutment. The resultant damage can be anywhere from mildly annoying to catastrophic.

Why not just specify a standard cylinder cushion option and be done with it? Most cylinder manufacturers’ standard cushion option does not provide progressive deceleration and can be inadequate for properly stopping a runaway heavy moving structure. This paper provides details for a multistage cushioning design that provides progressively adjustable deceleration. The structure can be intentionally driven to the end of its cycle without any need for control valve deceleration. It doesn’t eliminate the need for control valving but provides an extra measure of safety protection from ever slamming the structure into a hard stop at either end of its cycle.

Hydraulic Cylinder Progressive Cushions

Concept

Standard hydraulic cylinder cushioning options have been utilized for many years. The basic concept is a simple restriction of flow exiting the cylinder during the last inch or so of stroke at either end. A spear is added to the cylinder rod that makes contact with a closely fitted hole in the head of the cylinder. The exiting fluid is now forced through a smaller hole fitted with an adjustable needle valve that regulates the reduced rod velocity until the cylinder bottoms out of stroke. Increased pressure caused by the restriction of the exiting fluid is the mechanism for slowing the load (see Figure 1 below). A check valve is installed opposite this needle valve for return in the opposite direction so incoming fluid can be seen on the full area of the piston. The shortcoming of standard cushions for heavy civil projects is the short length of the cushion spear and the use of a “one size fits all” adjustable needle valve. Often times these needle valves are sized for a much larger flow and are not able to be properly adjusted. These subjects are investigated in greater detail below.

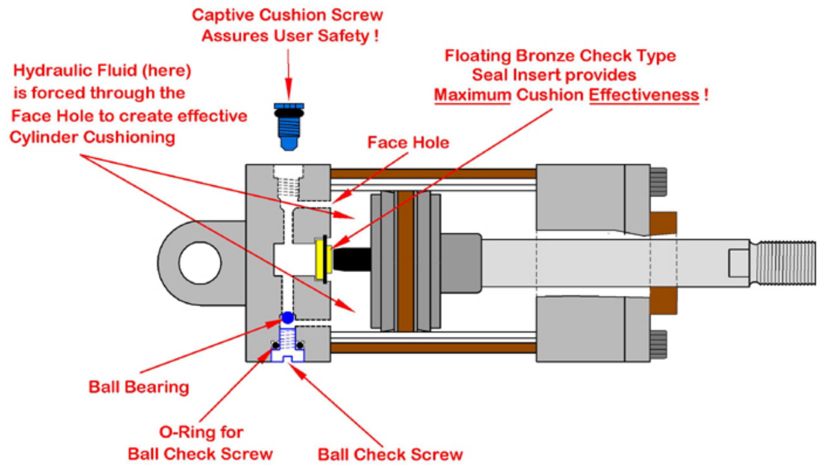


Figure 1

The progressive design expands on standard cushion options using longer cushion spears and multiple needle valves arranged in a manner that gradually reduces the number of available needle valves in the return fluid flow path (see Figure 2). Arranging the needle valves in this way allows you to program the deceleration profile to fit a particular application. Final tweaking different parts of the program profile are possible in ultimately small increments via individual needle valves.

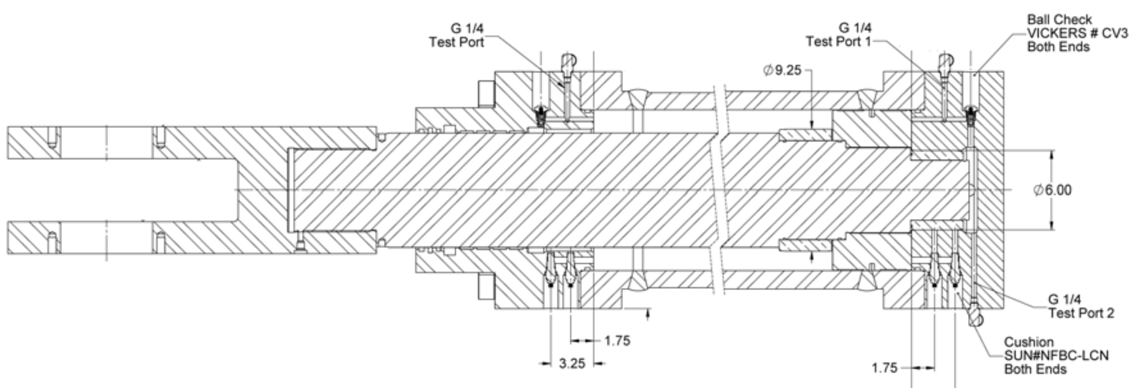


Figure 2

Designing Progressive Cushions

There are several key aspects of designing progressive hydraulic cylinder cushions to achieve maximum results.

- Length of the cushion Spear
- Calculating the load to be decelerated
- Sizing needle valves for optimum flow control

Cushion Spear Length

Cushion spear length is important because most civil engineering applications require cylinders to have reserve stroke such that the structure being powered comes to its resting position prior to the cylinder bottoming out. It's important during installation to share the reserve stroke between each end of cylinder. Setting of cylinders too shallow could result in the cylinder bottoming out before the span has closed and conversely robbing the other end from the effects of the cushions. Where more than one cylinder is involved it's important to set them all at the same level so the cushions in all cylinders actuate in unison. Figure 3 shows how reserve stroke is shared between both ends of the cylinder to maximize the amount of available cushioning.

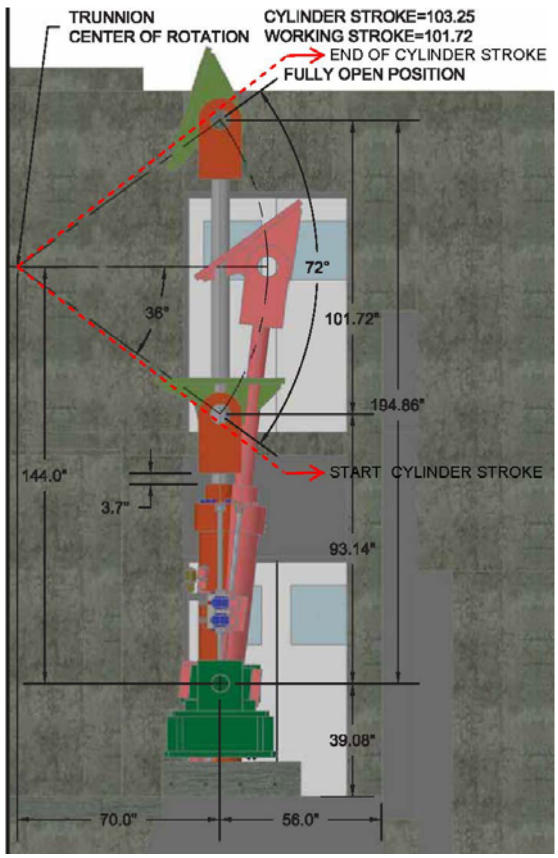


Figure 3

Reserve stroke must be taken into account when determining cushion spear length. This unused stroke is no longer available for cushioning and must be deducted from calculations. It is possible to specify reserve strokes as low as ½" - ¾". Greater lengths should only be considered if there could be need for adjustment to the point where the structure might rest.

Selecting cushion length should be approximately the same as the start point of deceleration under normal operation. A good design will have the cushion spear making contact shortly after the normal deceleration point such that no unintended back pressure from the cushioning operation is seen under normal operation. When normal cylinder deceleration is taking place, fluid flow exiting the cushion circuit is less than the needle valve setting. The cushion function is always in the background and only comes into play when required by a runaway load causing the exiting flow to be greater than normal conditions.

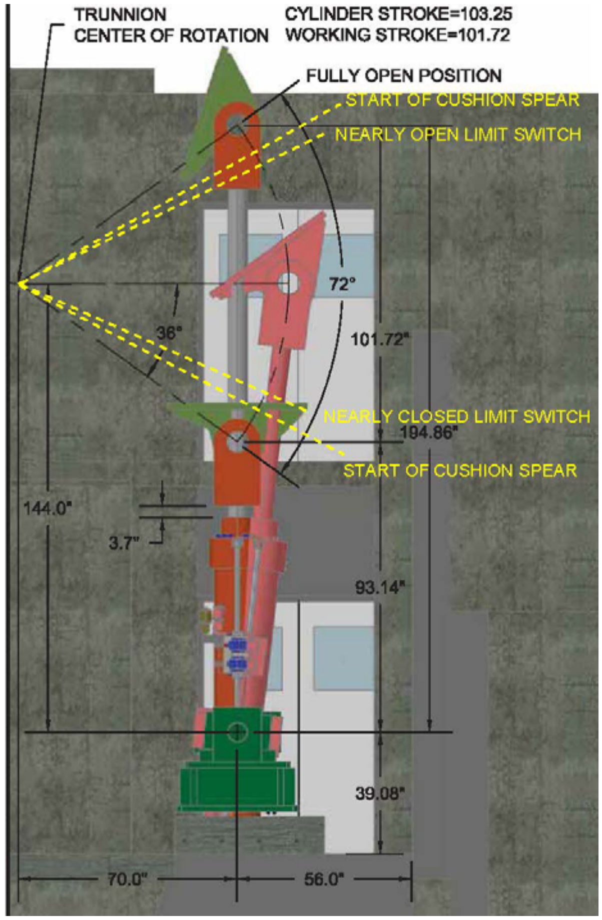


Figure 4

Generally speaking, most heavy civil engineering applications will see adequate protection with cushion spears approximately 4-6 inches in length. Remember that reserve stroke is not usable for cushioning. So a reserve stroke of ½” would require a 4.5” cushion spear.

The first ½” - ¾” of the spear is usually slightly tapered. This tapering insures the centering of the spear in the accepting return line passage but it also serves an important role in initializing deceleration. The most dramatic change in cylinder velocity happens within this first ¾” of spear engagement as available path for exiting hydraulic fluid goes from 100% to 20% within the first ¾” of spear engagement. It’s easy to see how flow exiting the cylinder is basically unrestricted until the tapered part of the spear starts closing off the fluid exit path.

The area changes rapidly until the exit passage is blocked and all fluid is directed through the needle valves. After another inch of travel the first needle valves exit path has been closed off and the final creep speed is set by the second needle valve. The deceleration velocity profile can be changed such that more or less change can be seen in the first needle valve prior to the final creep speed setting of the second needle valve.

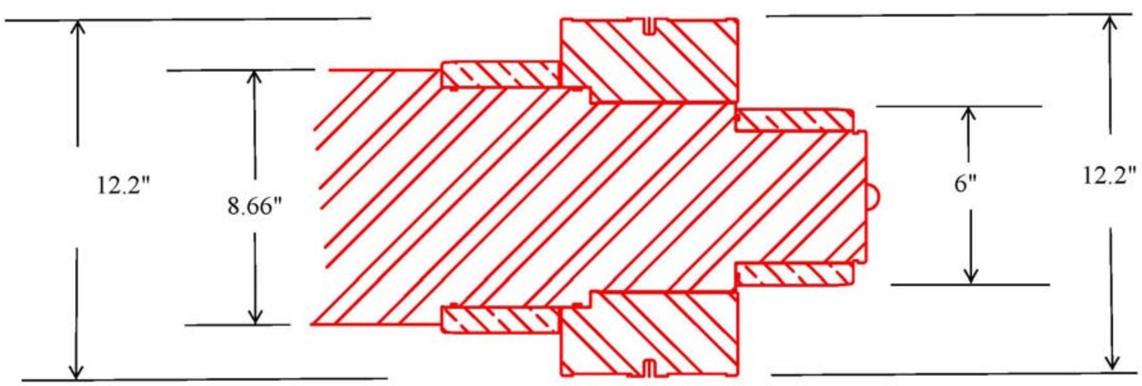
Longer cushion spears can accommodate multiple needle valves. A 6 inch spear with ¾” reserve stroke could have as many as four needle valves. A general rule to space one needle valve per inch of available cushion spear travel has been effective. With ¾” reserve stroke and ¾” spear taper there would only be 4.5” of travel available for a 6 inch spear.

Calculating Load

Determining the maximum load to be decelerated in a runaway condition may be easier than you think. In the previously mentioned condition of a heavy moving structure that's not stopping due to a limit switch failure it's not the structures' moving mass that's as important as the hydraulic system horsepower. Runaway conditions caused by moving water, wind, or ice are handled by the normal hydraulic system up to the point of failure of the counterbalance, relief valves, and basic pressure vessel integrity of the hydraulic system as a whole. Assuming these factors are taken account in initial system design, these types of failures are not part of the cushion discussion.

The inertia and mass of the structure are certainly important to overall calculations but are typically only 10-20 percent of the load seen on the cylinder cushions in the scenario where the control system fails to slow down hydraulic fluid flow exiting the cylinders. Most hydraulic systems are designed to output maximum flow until the system pressure causes the variable volume pumps to de-stroke or compensate to a lower flow. When the control system fails to regulate pump flow, the cylinder will not start to decelerate until the backpressure from the opposite side of the piston exerts a force great enough to cause the hydraulic pumps to compensate and reduce flow. Power from the hydraulic pumping system is a much greater force to deal with than load from the moving structure.

Calculating the back pressure required to compensate the pumps is a simple equation based on the available area of the piston. Keep in mind that piston area is now smaller due to the cushion spear. Figure 4 below shows a simplified version of this calculation. Basic fluid power principals of the forces exerted on both sides of the hydraulic cylinder piston must be evaluated.



12.2" Bore diameter	117 square inches
8.66" Rod diameter	- 59 square inches
Effective Annulus Area	= 58 square inches

12.2" Bore diameter	117 square inches
6" Spear diameter	- 28 square inches
Effective Cushioning Area	= 89 square inches

Figure 4

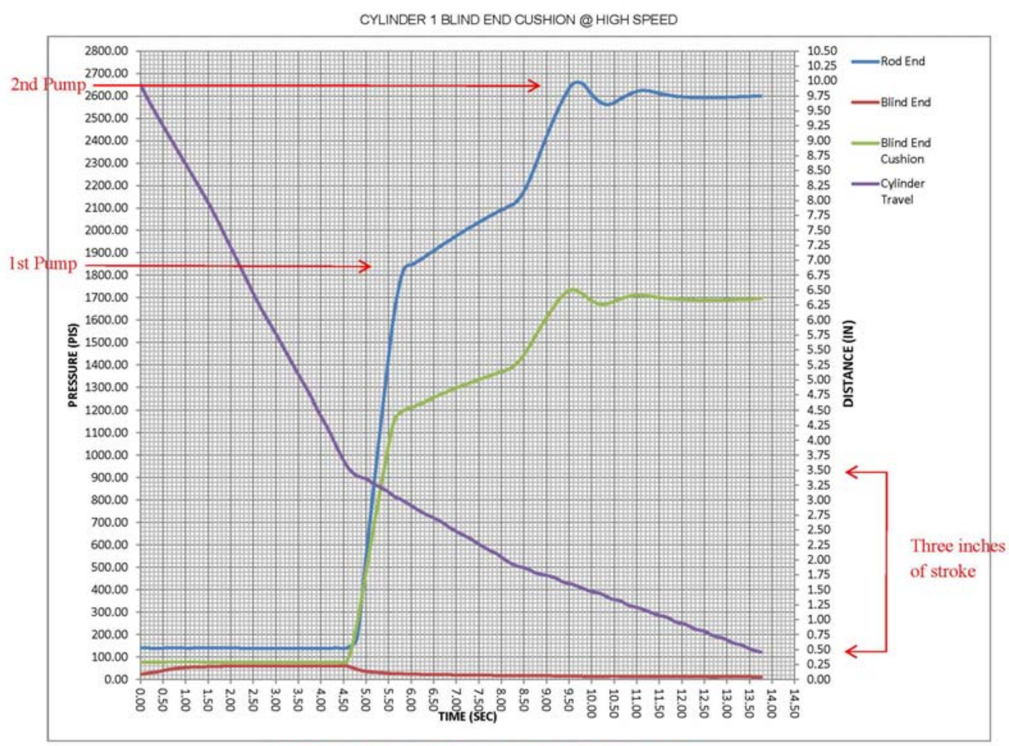
When making calculations for retracting the cylinder we will use the total available area on the rod side annulus area including the spear area. The force created by pressure on this rod side must be counteracted by the blind side piston area minus the spear area. From Figure 5 below we can see the rod side area of 58 square inches is working against an area of 89 square inches. This 89 sq in area must develop enough back pressure to raise pressure on the rod side high enough to destroke the pressure compensated pump or the pump will continue to put out full flow. When system pressure gets high enough, the pump will start to cut back output flow thus causing the cylinder velocity to slow down.

58 sq. in. x 1850 psi = 107,300 lbs / 89 sq. in. = 1205 psi required to compensate first pump lowering

58 sq. in. x 2650 psi = 153,700 lbs / 89 sq. in. = 1727 psi required to compensate sec pump lowering

Figure 5

Some systems have high and low pressure pumps. When the first pump setting is reached cylinder velocity will slow to the volume of the high pressure pump until that setting is reached. Figure 6 below shows a chart recording of a typical two pressure system using the previous area examples. You can see the velocity change in two steps and the resultant increase in ramp time to decelerate the cylinder rod. The velocity changed from 1.5 in/sec to .25 in/sec in 9 seconds over just 3 inches of cylinder travel.



Velocity change from 1.5 in/sec to .25 in/sec in 9 seconds

Figure 6

This intensification of pressure created by forcing the fluid through progressively smaller holes gradually compensates the pumps to reduce flow to the cylinders. The ratio of piston areas for retract stroke direction is such that the greater area on the blind side is less than the rod side. However, when traveling the opposite direction to extending the cylinder, it is extremely important to calculate pressure intensification for the rod side of the cylinder. The area of the blind (piston) side working on the reduced rod side area can produce extremely high pressures. These pressures can easily exceed 5000psi and should be taken into consideration when calculating the cylinder end cap bolt holding capabilities. Figure 7 shows how quickly the pressures can intensify due to the large rod and spear diameters.

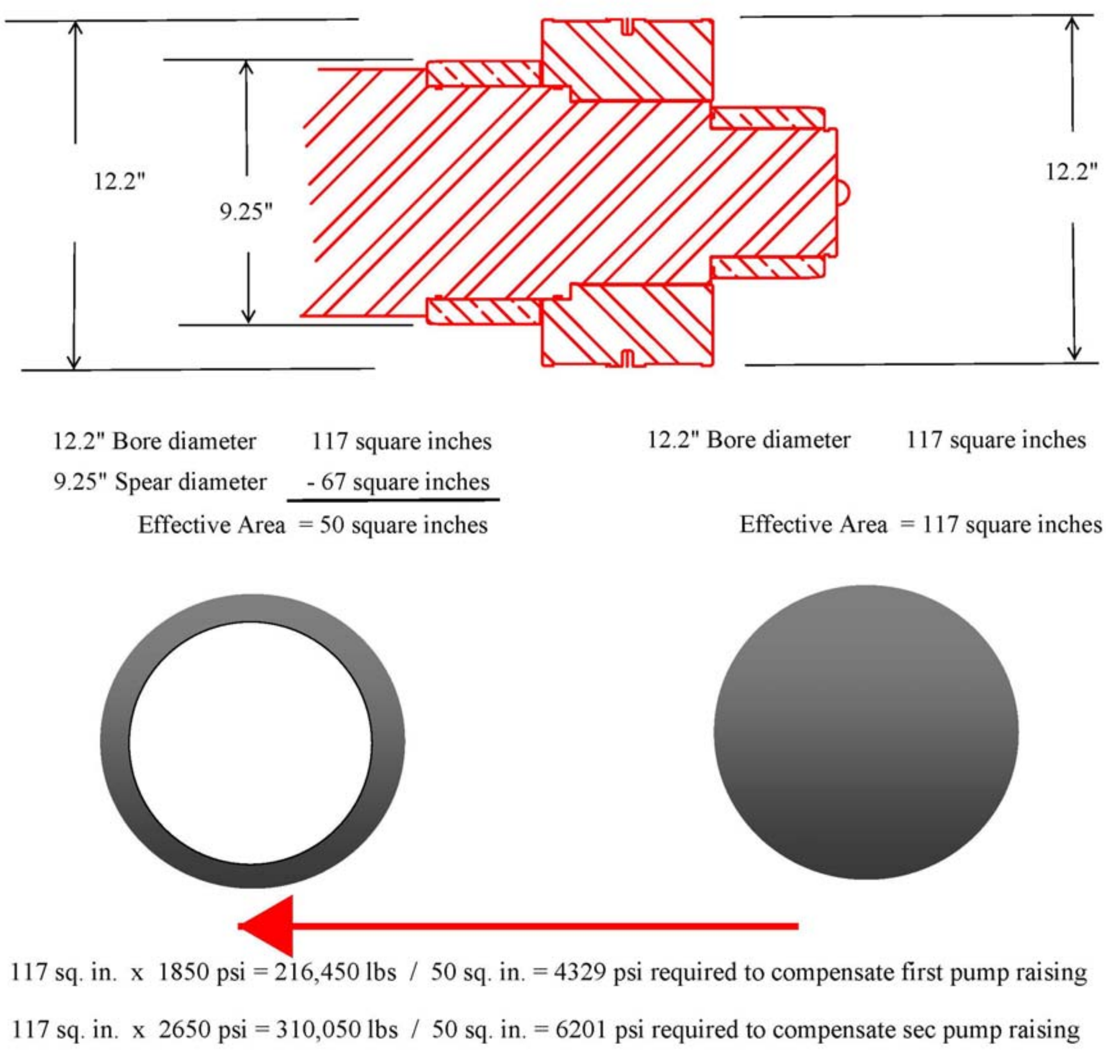


Figure 7

Sizing Needle Valves

Most standard cylinder cushion needle valves are a “one size fits all” solution and are not suited for the type of cushioning required for most civil engineering applications. They are generally sized for maximum flows and have coarse thread adjustments that give little adjustability for low flow situations. When using multiple needle valves they can be very small diameter orifices that will allow greater adjustability at low flows. Most heavy civil engineering moving structures require a creep speed of approximately 10% of the maximum travel speed. This varies of course for different applications but in general, you want to decelerate from 100% speed to 10% speed in anywhere from 3-10 seconds. Once creep speed has been achieved, it’s a matter of preference how long to creep to a stop.

Once you have established what pressures you will encounter the adjustable needle valves can be properly sized to provide the necessary change in cylinder rod velocity. A standard fluid power formula is used for calculation of flow through a sharp edge orifice (Area (sq. in.) = Q (gpm) ÷ (24.12 x √back pressure)). This formula is also used for calculating the pressure drop across the nose of the cushion spear as it first enters the cushion. Before needle valve orifice diameter can be determined you must first calculate the stage 1 pressure drop created by the initial contact of the cushion spear. This stage 1 reaction is when the high flow exiting the cylinder is very quickly restricted to the small orifices defined by the needle valves. The cushion spear is machined with a small taper over the first ½” – ¾” of cushion spear (see Figure 8). This taper provides the first stage of necessary backpressure to start de-stroking pump output.

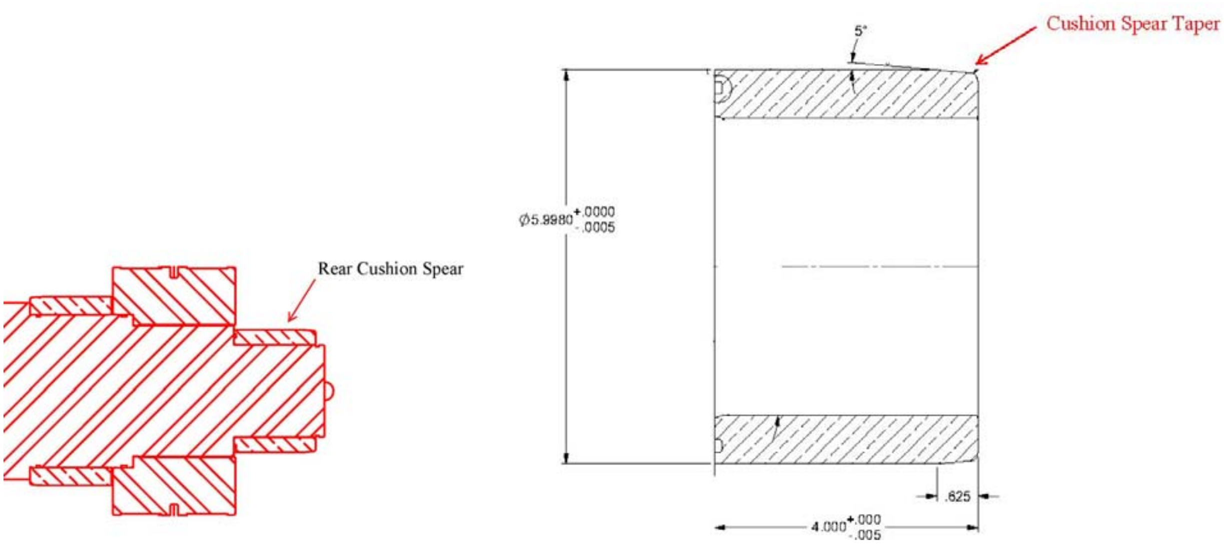
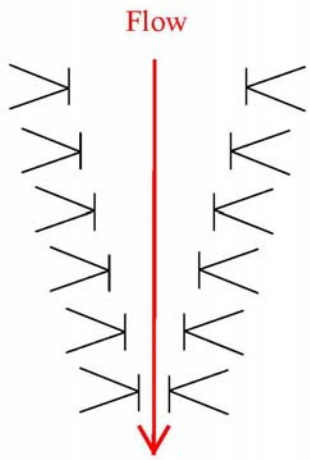


Figure 8

In order to calculate the pressure drop through the spear taper you can take a snapshot of the available flow area every 1/10” in the taper region. By adding these area snapshot pressure drops together you get the pressure drop created during the stage 1 initial rise in pressure that occurs during the first second of deceleration. Figure 9 shows a model of how this transition from full flow to restricted flow works. Once the main exit path for the oil has been closed off by the cushion spear, all oil must now flow through the needle valves. Pump output flow now starts to decrease causing the resultant flow through the needle valves to be closer to the 10% target flow required.

Model pressure drop created by spear taper into 6" diameter hole



Distance Traveled	Spear Diameter	Area Sq.In.	Pressure Drop
.1"	5.873	1.18	4
.2"	5.898	0.95	10
.3"	5.923	0.72	35
.4"	5.948	0.49	158
.5"	5.963	0.44	238
.6"	5.989	0.35	616
		Total =	**1061**

Figure 9

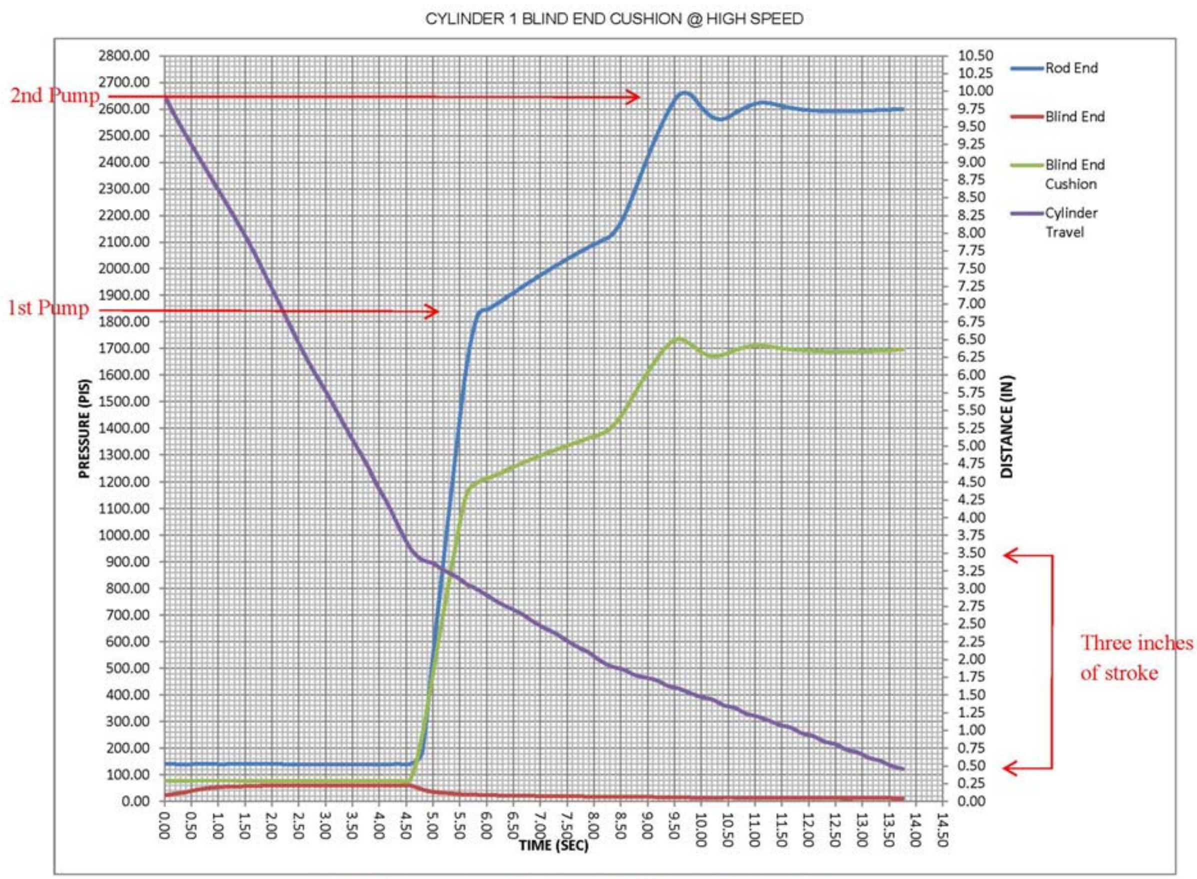
To size the needle valves take the intended creep speed flow and divide by the number of needle valves used then apply the maximum pressure drop required to keep the pumps compensated. Remember that the cushion spear has already increased the pumps to the threshold of the compensator setting. Reducing the flow further as the spear progresses through the spear cavity, very little flow is being forced through the final needle valve. Size the valves for their mid range flow capability so you will have the ability to adjust final speed up or down. Typical needle valve sizing for most civil engineering applications is approximately .16” diameter orifice sized needle valves.

Conclusion

Hydraulic cylinder cushions are a cost effective method for providing failsafe collision control of heavy civil engineering moving structures. Progressive cushioning using multiple needle valves in a series cavity arrangement can make the deceleration velocity curve infinitely adjustable to meet the needs of the application.

Example with detailed calculations

The following example was taken from calculations and resultant chart recordings of the hydraulic system used at the Matlacha Bridge in Matlacha Florida. It's a single leaf bascule bridge driven by two hydraulic cylinders 12.2"bore x 8.66" rod x 103.25" stroke. The leaf was moderately tip heavy by design and required two 60 HP hydraulic pumps which each drove a double hydraulic pump. With a total of four hydraulic pumps, two were set to compensate at 1850psi and two at 2650psi. You can see from chart recording results the shop simulation testing was very similar to the actual final installed load testing.



Velocity change from 1.5 in/sec to .25 in/sec in 9 seconds

Hydraulic Cylinder Cushion Analysis

Steps **Page**

Prepared by: Steve Hanley @ EHM

Reviewed by: Mike Hanley @ EHM *CFP #: 1145-R4*

Signature: ______________________

STEP 1: SYSTEM'S DATA COLLECTION

System Parameters (Pump Selection/Cylinder Selection)	**Extend**
Cylinder bore in inches	**12.20**
Cylinder stroke in inches (actual used stroke)	**103.25**
Rod diameter in inches	**8.66**
Cylinders per leaf	**2.00**
Pumps per leaf	**4.00**
Electric motor RPM	**1800**
Volumetric efficiency of pump	**0.900**
Horsepower of each dual pump	**60.00**
Mechanical efficiency of pump	**0.80**
Bore area in square inches	**116.8**
Annulus area in square inches	**58.0**
displacement rating required of each pump(cu in per rev)	**4.62**
Flow of each dual pump in gpm	**32.40**
Total actual pump flow available in gpm (two dual pump sets)	**129.60**

Cylinders Extend Function Calculations

Time to accelerate extension in seconds	**7.00**
Time to deccelerate extension to creep speed in seconds	**8.00**
Time in extension creep speed in seconds	**4.00**
Total time to extend cylinders	**63.00**
Pressure drop to extend	**567**
Load pressure required to extend (Assume 700 PSI)	**700**
Total pressure to extend	**1267**
Total fluid in gallons to extend both cylinders	**104.45**
Maximum pump flow required in gallons per minute to extend both cylinders	**112.92**
Flow of each dual pump in gpm	56.46
motor horsepower required for normal operations	**57.96**
Motor horsepower selected for this project	**60.00**
If required HorsePower(HP) is greater than selected HP then HP limiting will be needed	
Bridge leaf maximum velocity(feet/sec)	**0.16**

Cylinders Retract Function Calculations

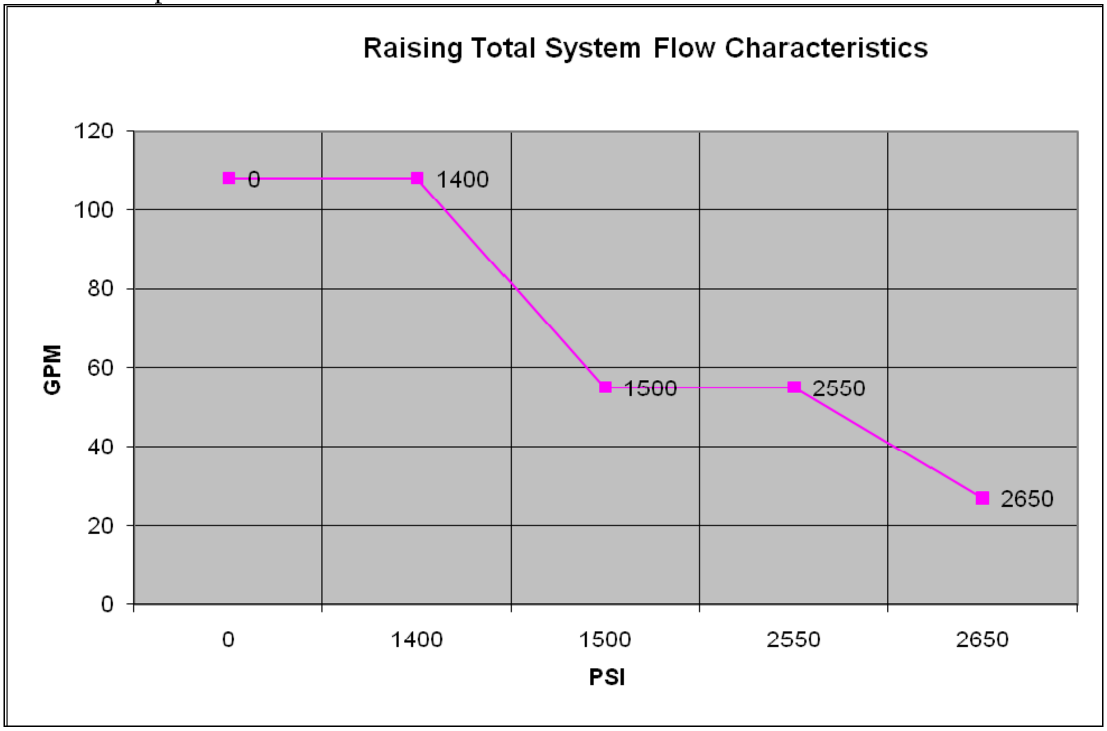
Time to accelerate retraction in seconds	**7.00**
Time to deccelerate retraction to creep speed in seconds	**8.00**
Time in retraction creep speed in seconds	**4.00**
Total time to retract cylinders	**63.00**
Pressure drop to extend	**939**
Load pressure required to retract (Assume 200 PSI)	**450**
Total pressure to retract	**1389**
Maximum pump flow required in gallons per minute to retract both cylinders	**56.02**
Total fluid in gallons to retract both cylinders	**51.82**
Flow of each dual pump in gpm	**28.01**
motor horsepower required for normal operations	**28.37**
Motor horsepower selected for this project	**60.00**
Bridge leaf maximum velocity(feet/sec)	**0.16**

Pumps Compensator Settings

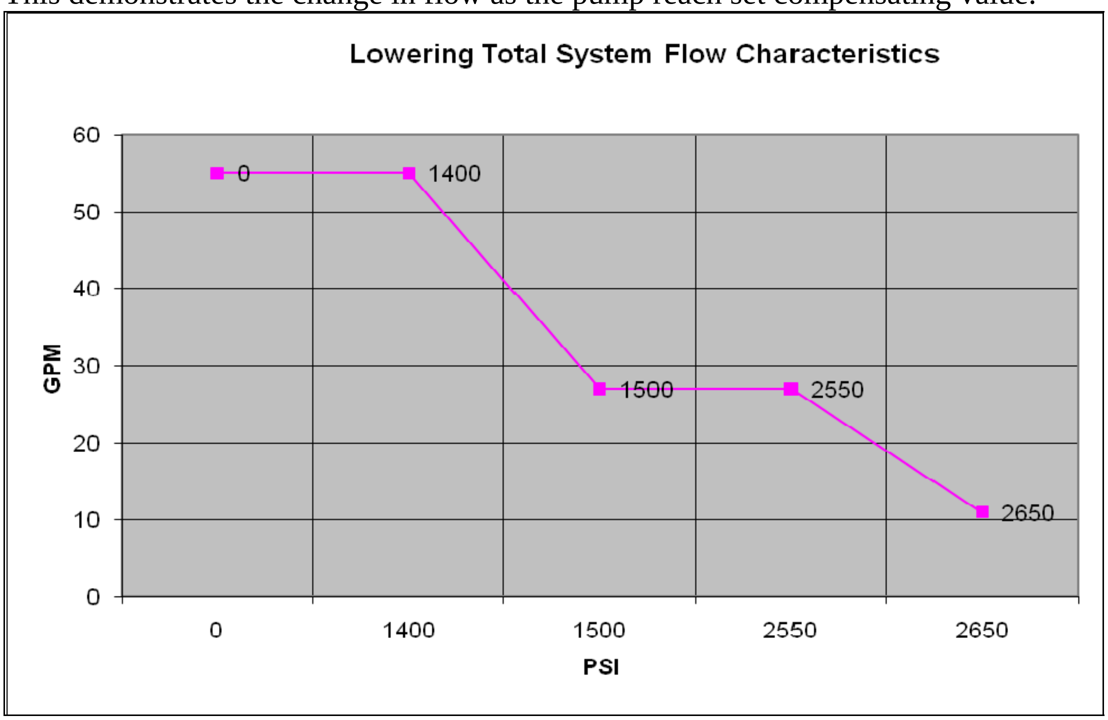
Pump 1 Compensator Setting (psi)	**1500.00**
Pump 2 Compensator Setting (psi)	**2650.00**

Step 2: Graph-Target Flows and Compensator Pressure

- This analysis is based on a worst case scenario where the limit switch has a failure not reducing the speed, in which case the cushion will build pressure and pump compensators will reduce flow to 25% of maximum seen in the charts below.



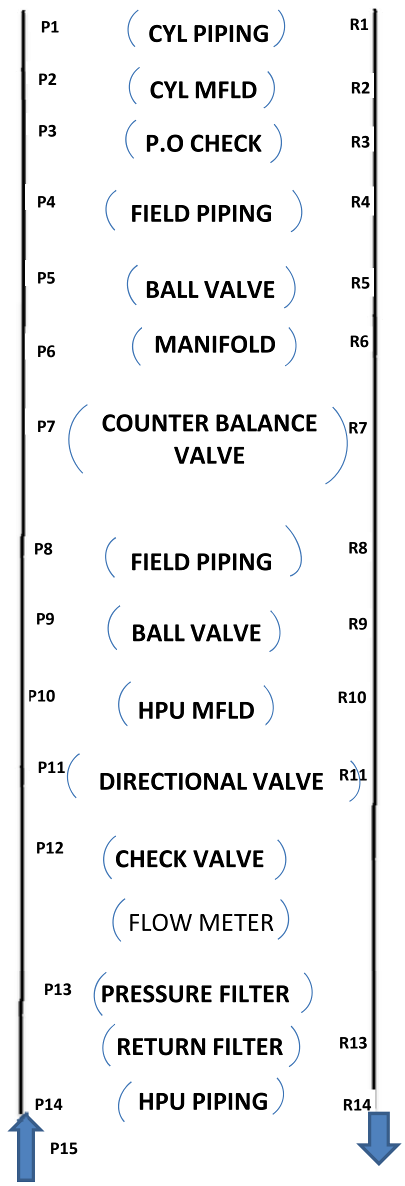
In the case of raising the bridge there are two active motors with four active pumps, two pumps having compensators set at 1500 psi and two with compensators set at 2650 psi. This demonstrates the change in flow as the pump reach set compensating value.



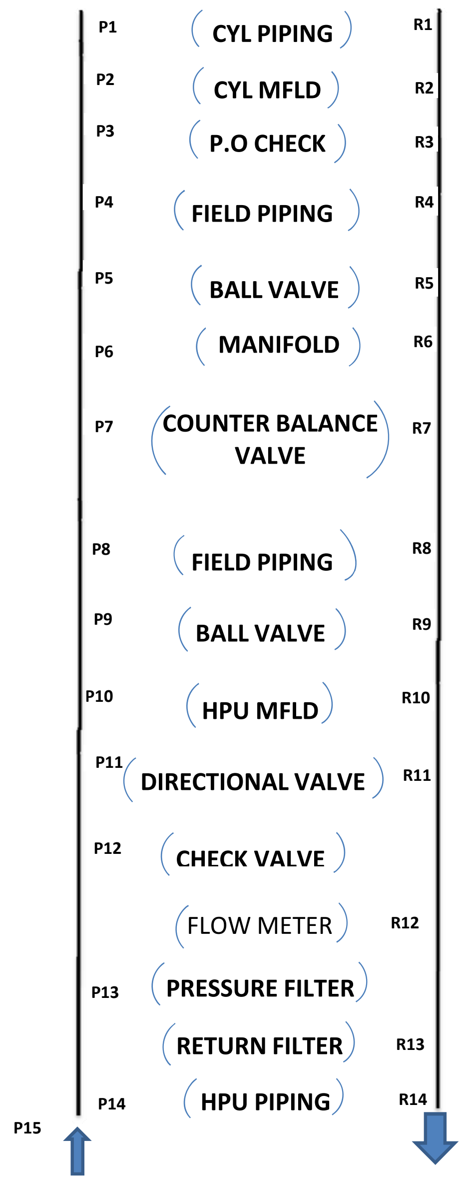
Step 2: Graph-Target Flows and Compensator Pressure

In the case of lowering the bridge there is one active motor and two active pumps, one compensator set a 1500 psi and the other set at 2650 psi. Again showing the change in flow characteristics as the pumps reach there compensating pressures.

Step 3: Pressure Drop Analysis



Raising



Lowering

Step 3: Pressure Drop Analysis

Pressure Drop for Extension

@	FLOW GPM	PRESSURE DROP
P1	54.02	4
P2	54.02	5
P3	54.02	7
P4	54.02	10
P5	54.02	10
P6	108.00	30
P7	54.02	150
P8	108.00	35
P9	108.00	17
P10	108.00	50
P11	108.00	20
P12	54.02	5
P13	54.02	8
P14	27.00	5
P15	54.02	0

Pressure Lost	356
Differential Area	189

@	FLOW GPM	PRESSURE DROP
R1	27.01	7
R2	27.01	5
R3	27.01	5
R4	27.01	7
R5	27.01	10
R6	27.01	30
R7	54.02	200
R8	54.02	31
R9	54.02	10
R10	54.02	50
R11	54.02	5
R12	54.02	15
R13	54.02	3
R14	54.02	0

Pressure Lost	378

Pressure Require to Move Fluid	567
HP Require to Move Fluid	17.88052539

Pressure Drop For Retraction

@	FLOW GPM	PRESSURE DROP
P1	27.01	2
P2	27.01	5
P3	27.01	4
P4	27.01	3
P5	27.01	5
P6	54.02	30
P7	54.02	150
P8	54.02	31
P9	54.02	8
P10	54.02	50
P11	54.02	10
P12	54.02	5
P13	54.02	8
P14	27.00	3
P15	54.02	0

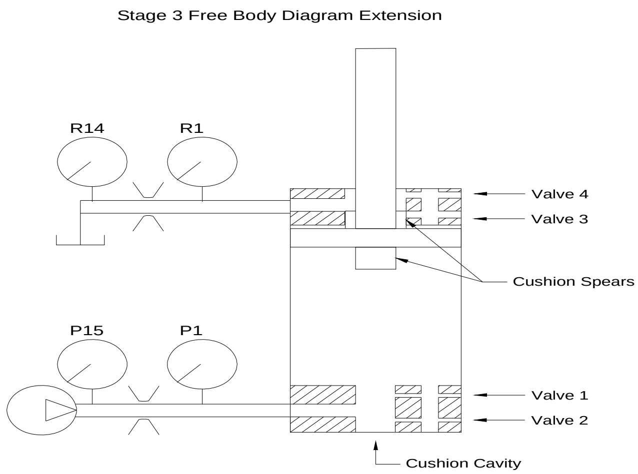
Pressure Lost	314
Differential Area	626

@	FLOW GPM	PRESSURE DROP
R1	54.02	7
R2	54.02	5
R3	54.02	5
R4	54.02	7
R5	54.02	10
R6	54.02	30
R7	54.02	100
R8	54.02	31
R9	54.02	8
R10	108	55
R11	108	20
R12	108	25
R13	108	10
R14	108	0

Pressure Lost	313

Pressure Require to Move Fluid	939
HP Require to Move Fluid	29.61166375

Step 4: Stage 3 Valve Selection



Stage 3 Parameters

@	FLOW GPM	PRESSURE
P1	13.51	2614.40
P15	******	2650.00
Compensator setting		2650.00
Pressure Lost		35.6
R1	6.75	37.8
R14	********	0
Pressure Lost		37.8

Stage 3:

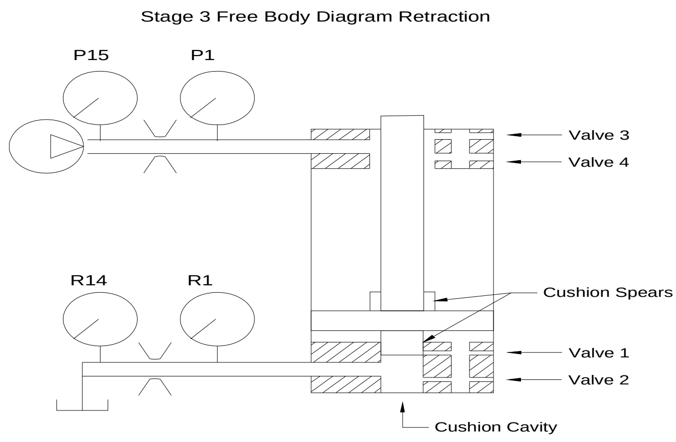
Stage 3 is one needle valve only and begins after the cushion spear is approximately 1/2 inch past Valve 1.
It is important that Valve 2 never gets covered up by the cushion spear else all flow would stop.
This cylinder will be installed such that the last 0.5 inch of stroke is not used,
so Valve 2 can be placed there and not get covered by the cushion.

Table 3:Cushion Data during extension@stage 3

Extension Flow thru valve 4 @ stage 3

Rod end Flow thru Valve 4 @ stage 3	6.75	GPM
Total rod end flow @ stage 3	6.75	GPM
Cushion pressure Raising (PRC-PR)=DP	5731	PSI
Valve 4 Diameter Raising	0.069	in
Valve 4 Area Raising	0.0037	in^2

Step 4: Stage 3 Valve Selection



Stage 3 Parameters

@	FLOW GPM	PRESSURE
P1	6.75	2618.60
P15	*******	2650.00
Compensator setting		2650.00
Pressure Lost		31.40
R1	13.505	31.30
R14	********	0
Pressure Lost		31.30

Stage 3:

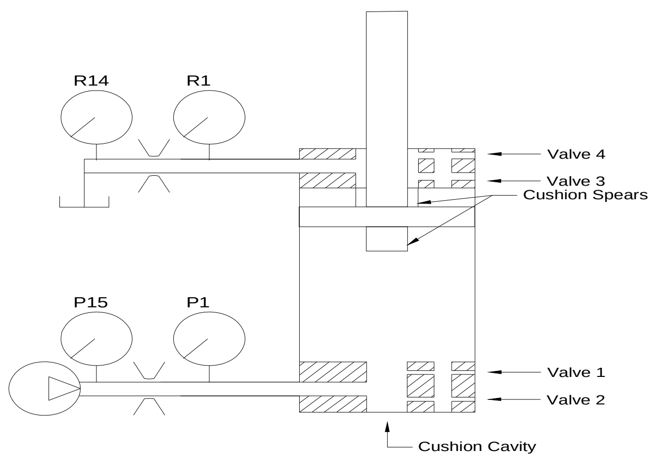
Stage 3 is one needle valve only and begins after the cushion spear is approximately 1/2 inch past Valve 1.
It is important that Valve 2 never gets covered up by the cushion spear else all flow would stop.
This cylinder will be installed such that the last 0.5 inch of stroke is not used,
so Valve 2 can be placed there and not get covered by the cushion.

Table 3:Cushion Data during retract@stage 3

Retracting Flow thru valve 2 @ stage 3

Blind end Flow thru Valve 2 @ stage 3	13.51	GPM
Total blind end flow @ stage 3	13.51	GPM
Cushion pressure lowering (PBC-PB)=DP	1642	PSI
Valve 2 Diameter lowering	0.133	in
Valve 2 Area lowering	0.0138	in^2

Step 5: Stage 2 Valve Selection



Stage 2 Parameters

@	FLOW GPM	PRESSURE
P1	27.01	1322.00
P15	********	1500.00
Compensator setting		1500.00
Pressure Lost		178
R1	13.505	189
R14	********	0
Pressure Lost		189

Table 2 Pertinent Equations

Effective Orifice Area/ Valve Diameter:

$$A(sq.in) = Q(GPM) \Big/ \left(24.12 \times \sqrt{Back \ \ pressure}\right)$$

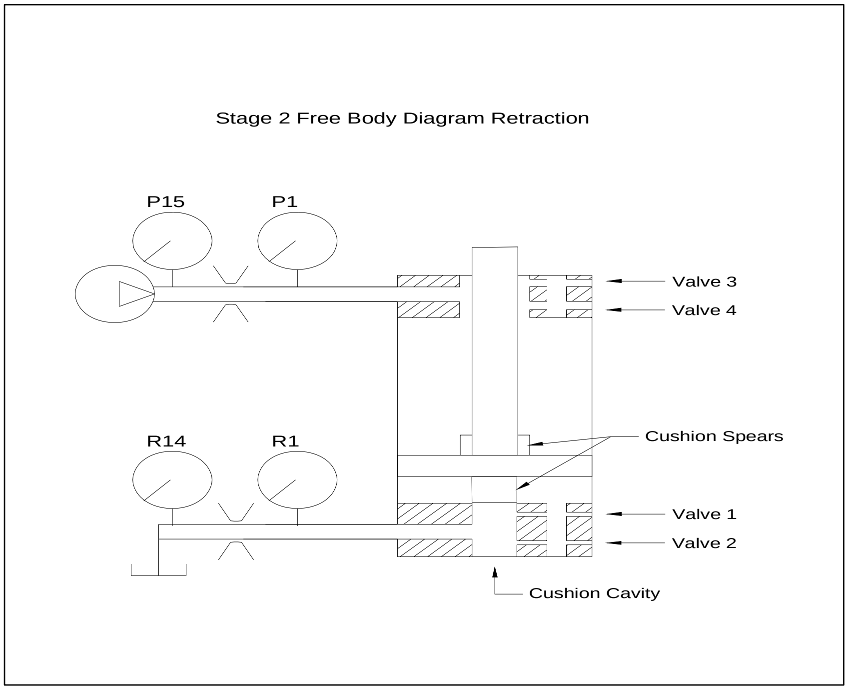
Diameter of Individual Valve:

$$D(in) = \sqrt{\left(\frac{4 \times (A(sq.in))}{\pi}\right)}$$

Table 2: Cushion Data during extension@ stage -2 at 1500 PSI

Rod end Flow thru Valve 3 @ stage 2	8.59	GPM
Rod end Flow thru Valve 4 @ stage 2	4.91	GPM
Total Rod end flow @ stage 2	13.51	GPM
Cushion pressure Raising (PRC-PR)=ΔP	3036	PSI
Valve 3 Diameter Raising	0.091	in
Valve 3 Area Raising	0.0065	in^2

Step 5: Stage 2 Valve Selection



Stage 2 Parameters

@	FLOW GPM	PRESSURE
P1	13.51	1343.00
P15	*********	1500.00
Compensator setting		1500.00
Pressure Lost		157
R1	27.01	156.5
R14	*********	0
Pressure Lost		156.5

Table 2 Pertinent Equations

Effective Orifice Area/ Valve Diameter:

$$A(sq.in) = Q(GPM) \Big/ \left(24.12 \times \sqrt{Back\ \ pressure}\right)$$

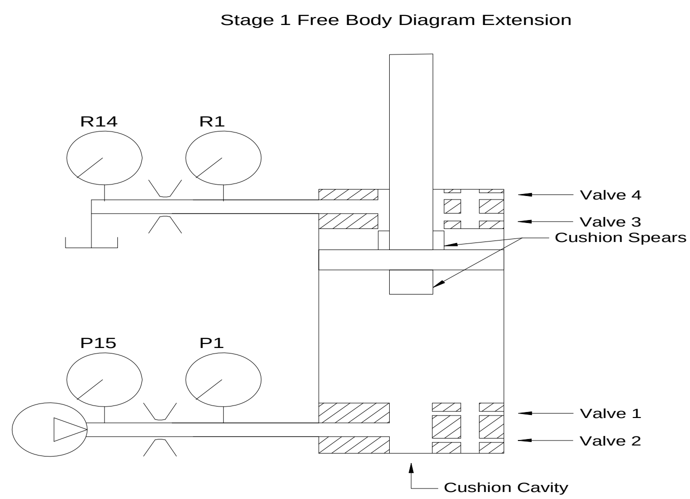
Diameter of Individual Valve:

$$D(in) = \sqrt{\left(\frac{4 \times (A(sq.in))}{\pi}\right)}$$

Table 2: Cushion Data during retract@ stage -2 at 1500 PSI

Blind end Flow thru Valve 1 @ stage 2	17.27	GPM
Blind end Flow thru Valve 2 @ stage 2	9.74	GPM
Total blind end flow @ stage 2	27.01	GPM
Cushion pressure lowering (PBC-PB)=ΔP	855	PSI
Valve 1 Diameter lowering	0.177	in
Valve 1 Area Lowering	0.0245	in^2

Step 6: Stage 1 Cushion Spear Dimension Verification



Stage 1 is just before the pump begins to compensate.
We need to know where is the cushion spear, at the moment that the pump begins to compensate
This point is 100PSI before compensator setting

Stage 1 Parameters

@	FLOW GPM	PRESSURE
P1	54.02	1044.00
P15	*******	1400.00
Compensator setting		1500.00
Pressure Lost		356
R1	27.01	378
R14	*******	0
Pressure Lost		378

When compensation begins, cushion Chamber press will be @	**2104**	**PSI**
Flow thru Valve # 4 will be	**3.7**	**GPM**
Flow thru valve # 3 will be	**6.5**	**GPM**
Flow thru the cushion spear clearance hole will be.	**16.8**	**GPM**

Cushion dimensional information			
Cavity		**spear**	**spear taper 3 degree over 0.625 inch**
9.2507		**9.2489**	

Find the distance into the cavity when the pressure drop equals the cushion chamber pressure above

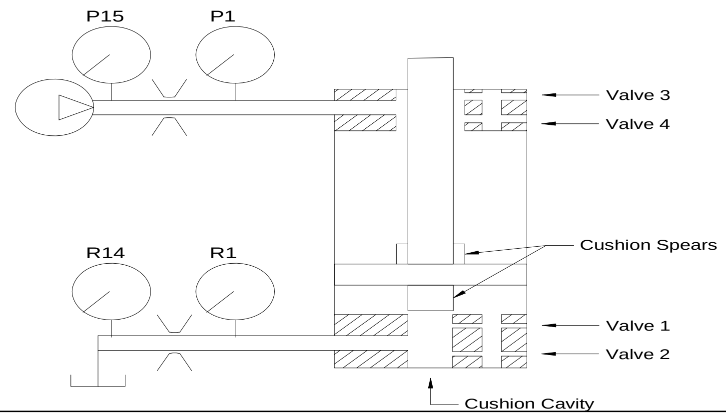
Distance	spear dia	area diff	press drop
0.1	9.1239	1.8301	0.1
0.2	9.1489	1.4713	0.2
0.3	9.1739	1.1115	0.5
0.4	9.1989	0.7507	2.5
0.5	9.2239	0.3889	34.6
0.55	9.2329	0.2584	177.6
0.575	9.2379	0.1859	663
0.6	9.2399	0.1569	1309

The pressure drop was calculated using the formula for a sharp edge orifice. The formula has limits on the travel distance. Different points were calculated to total the sum for .6" distance of travel. The table on the left represents the individual pressure drops at each distance point. The table below is the total pressure drop by adding up all the individual points.

At a distance of	**0.6**	**inch of travel**
the pressure drop is	**2187.5**	**PSI**
so it is at this point that the slowdown will begin		

Using the the free body diagram we were able to determine the flows and back pressure at each stage of cushing. This allowed us to determine the proper valve sizing for the second and thrid stage. Using the pressure difference in stage one of 378 pounds per square inch it was determine that we will have late cushioning after the critical point (.9 inch of travel) or one tenth after entering the cavity .

Step 6: Stage 1 Cushion Spear Dimension Verification



Stage 1 is just before the pump begins to compensate.
We need to know where is the cushion spear, at the moment that the pump begins to compensate
This point is 100PSI before compensator setting

Stage 1 Parameters		
@	FLOW GPM	PRESSURE
P1	27.01	1086
P15	*******	1400
Compensator setting		1500.00
Pressure Lost		314
R1	54.02	313
R14	*******	0
Pressure Lost		313

When compensation begins Pressure in cushion Chamber will be @ 538.8 PSI

Table 1: Cushion Data during retract @ Stage 1

When compensation begins, cushion Chamber press will be @	**539**	**PSI**
Flow thru Valve # 4 will be	**5.0**	**GPM**
Flow thru valve # 3 will be	**8.9**	**GPM**
Flow thru the cushion spear clearance hole will be.	**40.1**	**GPM**

Cushion dimentional information			
Cavity		**spear**	**spear taper 3 degree over 0.625 inch**
6		**5.998**	

Find the distance into the cavity when the pressure drop equals the cushion chamber pressure above

Distance	spear dia	area diff	press drop
0.1	5.8730	1.1844	2.3
0.2	5.8980	0.9533	5.5
0.3	5.9230	0.7211	16.7
0.4	5.9480	0.4880	79.5
0.45	5.9530	0.4413	118.9
0.475	5.9630	0.3477	308.5

The pressure drop was calculated using the formula for a sharp edge orifice. The formula has limits on the travel distance. Different points were calculated to total the sum for .475" distance of travel. The table on the left represents the individual pressure drops at each distance point. The table below is the total pressure drop by adding up all the individual points.

At a distance of	**0.475**	**inch of travel**
the pressure drop is	**531.2**	**PSI**

so it is at this point that the slowdown will begin

Calculation of Cushioning

Subject Matter: Raising of Bridge Leaf (extending cylinder)

The calculations below represent a compensator setting of 1500 psi. This occurs during stage 2 of cushioning.-The goal is to determine PC which is shown at the end of the next page.

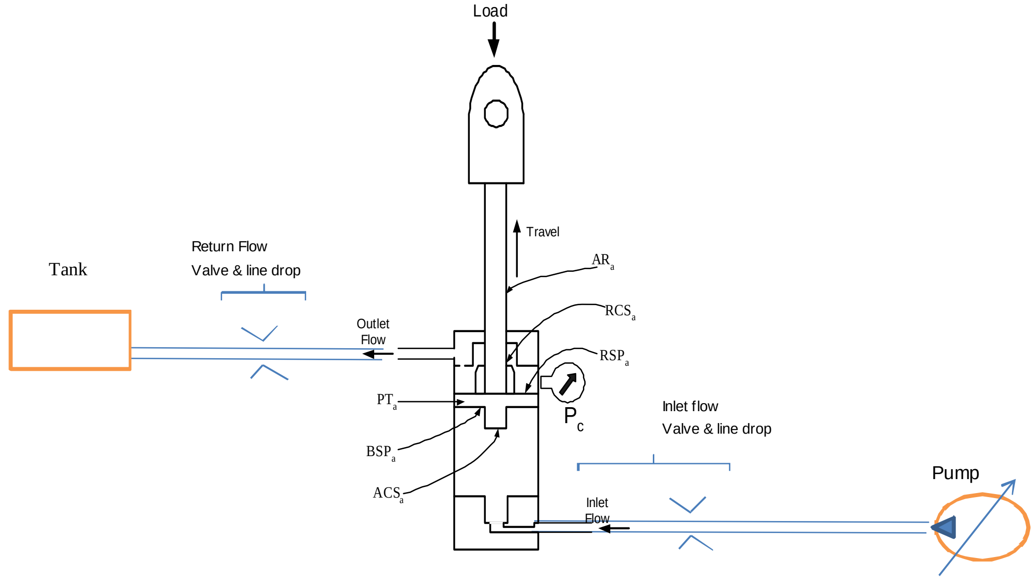


Fig. 2 Cylinder Raising

Note:

The back-pressure can be adjusted by opening or closing the needle valves thereby controlling the cushioning. This cushion device would only be used in a worst case scenario where electronic controls (limit switches) fail. First, the required back pressure to decelerate the mass of the bridge is calculated. Typically the mass energy is only about 10% to 20% of the total energy that needs to be converted to heat by the cushion.

Second the required back pressure to cause the pressure compensator control on the pump to de-stroke and therefore slow the bridge movement. Third, the required orifice area that produces the required back pressure is calculated and appropriate valves are selected (refer to table 2 and BOM).

Step 7: Maximum Cushion Chamber Pressure Calculations for Stage 2

Cushion Data:

Cushion spear diameter Hollow around the piston rod	D_s =	9.25	in
Surface area of rod end cushion spear	RCS =	8.30	in^2
Rod side piston area after cushion spear enters cavity	BSP_a =	49.67	in^2

Calculations:

Initial Velocity	v_o =	0.155	ft/secs
Final Velocity which is 50% of initial velocity	v_i =	0.074	ft/secs
Average Cushion deceleration	a_c =	0.041	$ft/secs^2$
Cushion stopping time	t_c =	2.00	secs
Estimated mass of bridge	M_b =	1463500	slugs
Total number of cylinders holding up the bridge	NC =	2	
Force of deceleration of leaf moving mass on one cylinder	$\mathbf{F_b}$ **=**	**29.65**	**kips**
Pressure @ pump outlet	PP =	1500	psi
Pressure drop across inlet flow piping and valves	PDIF =	356	psi
Pressure @ cylinder blind side	P_r =	1144	psi
Determinative pushing force Cylinder blind side:	$\mathbf{F_{rs}}$ **=**	**133.66**	**kips**
Pressure drop across outlet flow piping and valves	PDOF =	378	
Pressure @ cylinder rod side port port connection	PBP =	378	
Determinative pushing force Cylinder Rod side cushion spear:	$\mathbf{F_{cs}}$ **=**	**3.14**	**kips**
Force seen by Cylinder Rod side	$\mathbf{F_{bs}}$ **=**	**160.18**	**kips**

$$F_{bs} = F_{rs} + F_b - F_{cs}$$

Back-pressure at Gauge Pc rod side	$\mathbf{P_c}$ =	**3225**	**psi**

$$P_c = \frac{F_{bs}}{BSP_a}$$

Calculation of Cushioning

Subject Matter: Lowering of Bridge Leaf(retracting cylinder)

The calculations below represent a compensator setting of 1500 psi. This occurs during stage 2 of cushioning.-The goal is to determine PC which is shown at the end of the next page.

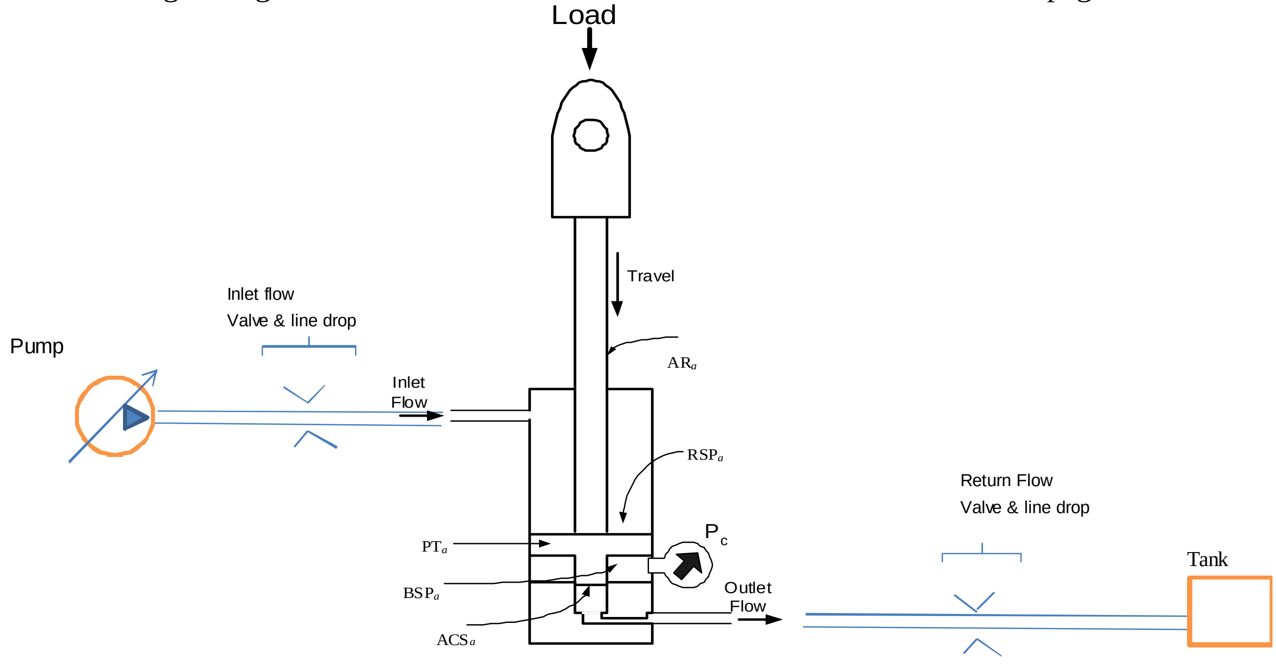


Fig. 2 Cylinder A in Retraction

Note: The back-pressure can be adjusted by opening or closing the needle valves thereby controlling the cushioning. This cushion device would only be used in a worst case scenario where electronic controls (limit switches) fail. First, the required back pressure to decelerate the mass of the bridge is calculated. Typically the mass energy is only about 10% to 20% of the total energy that needs to be converted to heat by the cushion.

Second the required back pressure to cause the pressure compensator control on the pump to de-stroke and therefore slow the bridge movement. Third, the required orifice area that produces the required back pressure is calculated and appropriate valves are selected (refer to table 2 and BOM).

Step 7: Maximum Cushion Chamber Pressure Calculations for Stage 2

Cushion Data:

Cushion spear diameter	D_s =	6	in
Surface area of cushion spear shaft	ACS_a =	28.27	in^2
Blind side piston area after cushion spear enters cavity	BSP_a =	88.57	in^2

Calculations:

Initial Velocity	v_o =	0.155	ft/secs
Final Velocity which is 50% of initial velocity	v_i =	0.074	ft/secs
Average Cushion deceleration	a_c =	0.041	$ft/secs^2$
Cushion stopping time	t_c =	2.00	secs
Estimated mass of bridge	M_b =	1463500	slugs
Total number of cylinders holding up the bridge	NC =	2	
Force of deceleration of leaf moving mass on one cylinder	$\mathbf{F_b}$ =	**29.65**	**kips**
Pressure @ pump outlet	PP =	1500	psi
Pressure drop across inlet flow piping and valves	PDIF =	314	psi
Pressure @ cylinder rod side	P_r =	1186	psi
pushing force Cylinder (rod side):	$\mathbf{F_{rs}}$ =	**68.75**	**kips**
Pressure drop across outlet flow piping and valves	PDOF =	313	
Pressure @ cylinder blind side port port connection	PBP =	313	
pushing force Cylinder blind side cushion spear:	$\mathbf{F_{cs}}$ =	**8.85**	**kips**
Force seen by Cylinder B (blind side)	$\mathbf{F_{bs}}$ =	**89.55**	**kips**

$$F_{bs} = F_{rs} + F_b - F_{cs}$$

Back-pressure at Gauge Pc (blind side)	$\mathbf{P_c}$ =	**1011**	**psi**

$$P_c = \frac{F_{bs}}{BSP_a}$$

Calculation of Cushioning

Subject Matter: Raising of Bridge Leaf (extending cylinder)

The calculations below represent a compensator setting of 2650 psi. This occurs during stage 3 of cushioning.-The goal is to determine PC which is shown at the end of the next page.

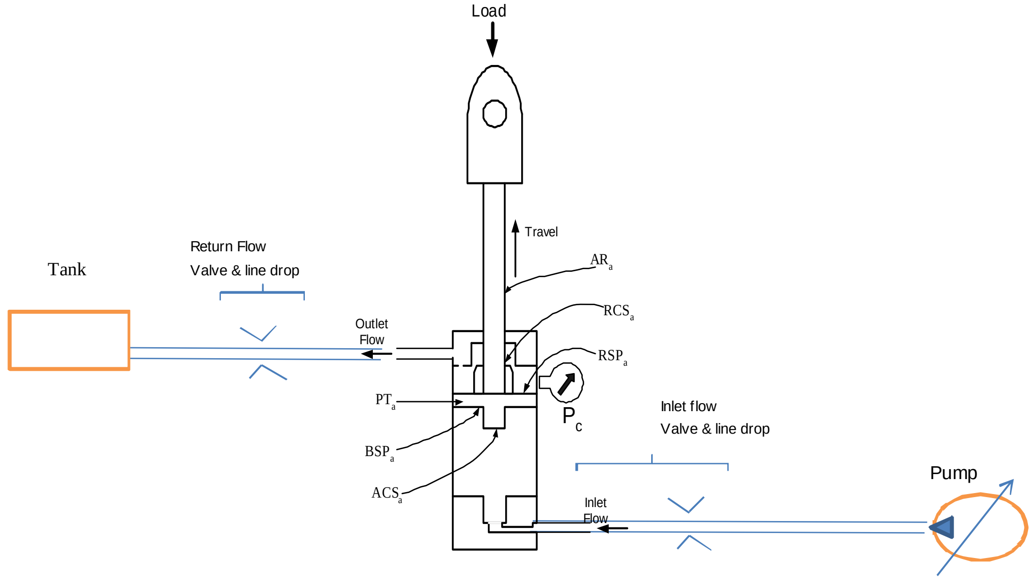


Fig. 2 Cylinder Raising

Note:

The back-pressure can be adjusted by opening or closing the needle valves thereby controlling the cushioning. This cushion device would only be used in a worst case scenario where electronic controls (limit switches) fail. First, the required back pressure to decelerate the mass of the bridge is calculated. Typically the mass energy is only about 10% to 20% of the total energy that needs to be converted to heat by the cushion.

Second the required back pressure to cause the pressure compensator control on the pump to de-stroke and therefore slow the bridge movement. Third, the required orifice area that produces the required back pressure is calculated and appropriate valves are selected (refer to table 2 and BOM).

Step 8: Maximum Cushion Chamber Pressure Calculations for Stage 3

Cushion Data:

Cushion spear diameter Hollow around the piston rod	D_s =	9.25	in
Surface area of rod end cushion spear	RCS =	8.30	in^2
Rod side piston area after cushion spear enters cavity	BSP_a =	49.67	in^2

Calculations:

Initial Velocity	v_o =	0.074	ft/secs
Final Velocity which is 20% of initial velocity	v_i =	0.015	ft/secs
Average Cushion deceleration	a_c =	0.030	$ft/secs^2$
Cushion stopping time	t_c =	2.00	secs
Estimated mass of bridge	M_b =	1463500	slugs
Total number of cylinders holding up the bridge	NC =	2	
Force of deceleration of leaf moving mass on one cylinder	$\mathbf{F_b}$ =	**21.66**	**kips**
Pressure @ pump outlet	PP =	2650	psi
Pressure drop across inlet flow piping and valves	PDIF =	356	psi
Pressure @ cylinder blind side	P_r =	2294	psi
Determinative pushing force Cylinder blind side:	$\mathbf{F_{rs}}$ =	**268.03**	**kips**
Pressure drop across outlet flow piping and valves	PDOF =	378	
Pressure @ cylinder rod side port port connection	PBP =	378	
Determinative pushing force Cylinder Rod side cushion spear:	$\mathbf{F_{cs}}$ =	**3.14**	**kips**
Force seen by Cylinder Rod side	$\mathbf{F_{bs}}$ =	**286.55**	**kips**

$$F_{bs} = F_{rs} + F_b - F_{cs}$$

Back-pressure at Gauge Pc rod side	$\mathbf{P_c}$ =	**5769**	**psi**

$$P_c = \frac{F_{bs}}{BSP_a}$$

Step 8: Maximum Cushion Chamber Pressure Calculations for Stage 3

Calculation of Cushioning

Subject Matter: Lowering of Bridge Leaf(retracting cylinder)

The calculations below represent a compensator setting of 2650 psi. This occurs during stage 3 of cushioning.-The goal is to determine PC which is shown at the end of the next page.

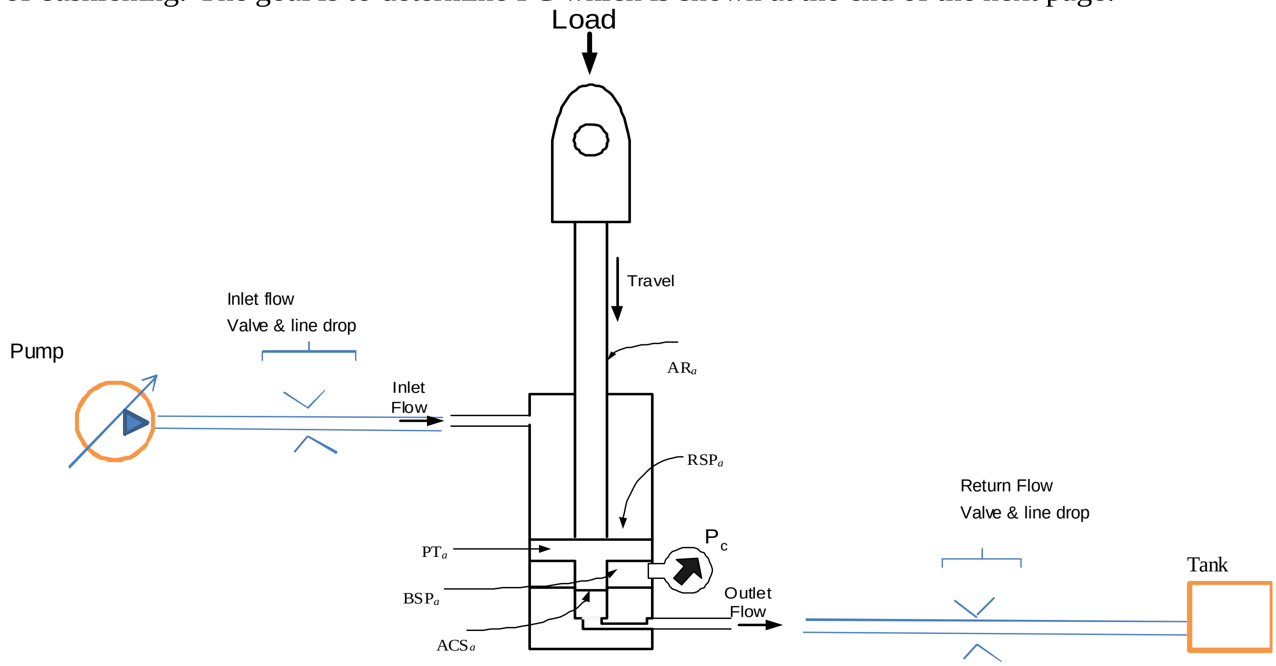


Fig. 2 Cylinder A in Retraction

Note: The back-pressure can be adjusted by opening or closing the needle valves thereby controlling the cushioning. This cushion device would only be used in a worst case scenario where electronic controls (limit switches) fail. First, the required back pressure to decelerate the mass of the bridge is calculated. Typically the mass energy is only about 10% to 20% of the total energy that needs to be converted to heat by the cushion.

Second the required back pressure to cause the pressure compensator control on the pump to de-stroke and therefore slow the bridge movement. Third, the required orifice area that produces the required back pressure is calculated and appropriate valves are selected (refer to table 2 and BOM).

Step 8: Maximum Cushion Chamber Pressure Calculations for Stage 3

Cushion Data:

Cushion spear diameter	D_s =	6	in
Surface area of cushion spear shaft	ACS_a =	28.27	in^2
Blind side piston area after cushion spear enters cavity	BSP_a =	88.57	in^2

Calculations:

Initial Velocity	v_o =	0.074	ft/secs
Final Velocity which is 20% of initial velocity	v_i =	0.015	ft/secs
Average Cushion deceleration	a_c =	0.030	$ft/secs^2$
Cushion stopping time	t_c =	2.00	secs
Estimated mass of bridge	M_b =	1463500	slugs
Total number of cylinders holding up the bridge	NC =	2	
Force of deceleration of leaf moving mass on one cylinder	$\mathbf{F_b}$ =	**21.66**	**kips**
Pressure @ pump outlet	PP =	2650	psi
Pressure drop across inlet flow piping and valves	PDIF =	314	psi
Pressure @ cylinder rod side	P_r =	2336	psi
pushing force Cylinder (rod side):	$\mathbf{F_{rs}}$ =	**135.41**	**kips**
Pressure drop across outlet flow piping and valves	PDOF =	313	
Pressure @ cylinder blind side port port connection	PBP =	313	
pushing force Cylinder blind side cushion spear:	$\mathbf{F_{cs}}$ =	**8.85**	**kips**
Force seen by Cylinder B (blind side)	$\mathbf{F_{bs}}$ =	**148.22**	**kips**

$$F_{bs} = F_{rs} + F_b - F_{cs}$$

Back-pressure at Gauge Pc (blind side)	$\mathbf{P_c}$ =	**1674**	**psi**

$$P_c = \frac{F_{bs}}{BSP_a}$$