Chapter 12. Circuit Analysis and Design: A Basic Approach to Circuit Design

12.1 The Design Steps

The basic approach to design that we take in this section was introduced by R Henke (Fluid Power Systems and Circuits, Hydraulics and Pneumatics,1983). We have, however, endeavored to streamline the preliminary steps in defining the hydraulic force (torque) profile and will introduce a set of criteria to assist in transferring information about the system to the actual placement of components to satisfy the circuit requirements. In the final analysis, there will be many possible solutions to the defined problems and each case **really is unique**. We shall attempt to provide very basic circuits which represent **worst case scenarios** and which can be optimized through compromise and redesign.

A summary of the steps for design are as follows:

1. Job to be done - understand the load
2. Choose an Actuator or Motor Size
3. Establish Flow and Pressure profiles
4. Check basic Horsepower to satisfy the hydraulic profiles
5. Design Circuit
6. Plot pressure, flow and Horsepower profiles at important parts of the circuit
7. Component selection
8. Redesign for compromise

12.1.1. Job to be done - understand the load
For many designers, this step is missed. The chances of designing a successful circuit are diminished if one does not have years of experience to base decisions on or to define what the job requirements are in detail.

(a) Establish the velocity profile
(b) Establish the Burden and Burden profile (to be defined)
(c) Establish the hydraulic force (torque) profile

This is a very difficult task. In many situations, it is very difficult to obtain the necessary information you need; thus you must be creative and use your engineering skills or intuition to come up numbers which will reflect the real situation. Consider the example of trying to estimate the forces required to cut into a rock face; except for the size of the unit you are designing, no information is known. One approach would be to find out what other possible designs there exists commercially to do the job and look at their rated horsepower. This would give you a ball park estimate of the magnitude of the forces or torques that are going to be required in your design. You should also consult the technical support groups (technicians, technologist, labourers etc) and get their input because of the experience they have in such matters.

In all our problems, we are going to be "spoon-fed" with the loading conditions because we are just starting to crawl in terms of design. As you gain experience, the definition of the job to be done will be strongly experienced based, coupled with engineering calculations and intuition.

12.1.2. Choose an Actuator or Motor Size

From the hydraulic force (torque) and velocity profile, we can size up an actuator or motor which yields a reasonable flow and pressure and still meets the mechanical constraints on the devices. This is a nice intermediate step because it gives us an idea of the size of components we are going to need.

12.1.3. Establish Flow and Pressure profiles

This is found once the hydraulic force (torque) profile and the actuator (motor) size has been established. This is a step which provides a great tool for debugging a hydraulic circuit. It also gives the designer good insight into how his/her circuit is working.

12.1.4. Check basic Horsepower to satisfy the hydraulic profiles

This is a simple intermediate step to ensure that the horsepower requirements are reasonable.

12.1.5. Design Circuit

This is what we all have been waiting to do. The fact that it is the fifth step is sometimes very frustrating to a designer and hence the tendency to jump directly to this step is quite common; don't give in. Complete the first four steps and you will find that the circuit you design will be based **on sound engineering**.

12.1.6. Plot pressure, flow and Horsepower profiles at important parts of the circuit (as a minimum, at the load and the pump)

This will give you a chance to examine the efficiency of your circuit. If you find it is unacceptable, then you will have to return to step 5 and redesign. Check circuit operation both statically and dynamically

12.1.7. Component selection

This is where you choose the best component to do the job at the least cost. (This part we will not be doing in this course. This does not mean that it is not important but usually, this is done by an experienced technician who has a better understanding of such things.)

12.1.8 Redesign for compromise

The circuits we design in this class will represent "worst case scenarios" in that the constraints placed on them will be quite severe. Most of the circuits could be "simplified" using servo controlled systems but cost and circumstances may prohibit this form being an alternative. Our philosophy is to design for the constraints producing a circuit that will meet the requirements of the client or specification and recognize that a simpler circuit is possible if compromise on the constraints can be made. If compromise is not possible, then your worst case scenario circuit becomes the best case. Regardless, if you can design a circuit that can meet the constraints, you will have a very in-depth understanding of the circuit and if problems are identified or compromise achieved, redesign is much, much easier to do.

In the following pages, we will be considering these stages of design. First, however, we must provide a few definitions to assist us in clearly defining all terms and sign conventions that will be used.

12.2 Definitions and Sign Notation

This section will introduce a sign convention and new terms relevant to the development of the hydraulic force profile (to be defined). **NOTE: IN THIS AND THE NEXT SECTIONS, THE NOTES WILL BE MORE DETAILED THAN ELSEWHERE IN THE NOTES. THIS IS BECAUSE THIS APPROACH HAS NOT BEEN DOCUMENTED ANYWHERE ELSE (YET) AS IT IS OUR UNIQUE U of S APPROACH (well as unique as one can get since it is based on good design practices and Russ Henke's approach).**

12.2.1 Sign Convention

The sign convention adopted in this study is not traditional (vector based). It is defined such that the direction of motion of the system actuator is always positive.

That is, **forces are considered to be positive if they are applied in the direction of motion of the actuator**.

All forces (torques) can therefore be categorized as to their nature (resistive, over-center, for example) with the same sign, which is independent of the direction of motion. The reasons for adopting such a nonstandard sign convention and the use of this sign convention in analyzing the motion of systems will be demonstrated with an example as follows.

Consider an example of a simple hydraulic actuator and mass system as shown in Figure 12-1(a). A container of mass m_1 is moved from point P_1 to P_2 during the forward part of the cycle for loading, and moved back to P_1 with the loaded mass m_2 during reverse. Assume that the only external force (F_{ext}) acting on the actuator is coulomb friction. The absolute magnitude of the force is

$F_1 = \mu_s \cdot m_1 \cdot g$ in the forward direction
and
$F_2 = \mu_s \cdot m_2 \cdot g$ in the reverse.
(μ_s is coulomb friction coefficient).

The sign of the friction force in this example is always negative regardless of the direction, since the friction opposes the motion of the actuator in both directions. The friction force profile for a whole duty cycle is illustrated as in Figure 12-1(b).

The negative external force represents a specific physical system condition: that is of a **"resistive" type system**. This implies that a positive force (in the direction of the motion) must be provided by the hydraulic system to overcome the friction, i.e. to "push" the actuator in order to accomplish the task. It should be noted that in this example, when the direction of motion changes, the sign of the external force does not vary as the "resistive' property of the system remains. The significance of using this sign convention is to truly reflect physical system conditions relative to the "fluid" parameters of the hydraulic system, which is not affected by the changes in the direction of the motion.

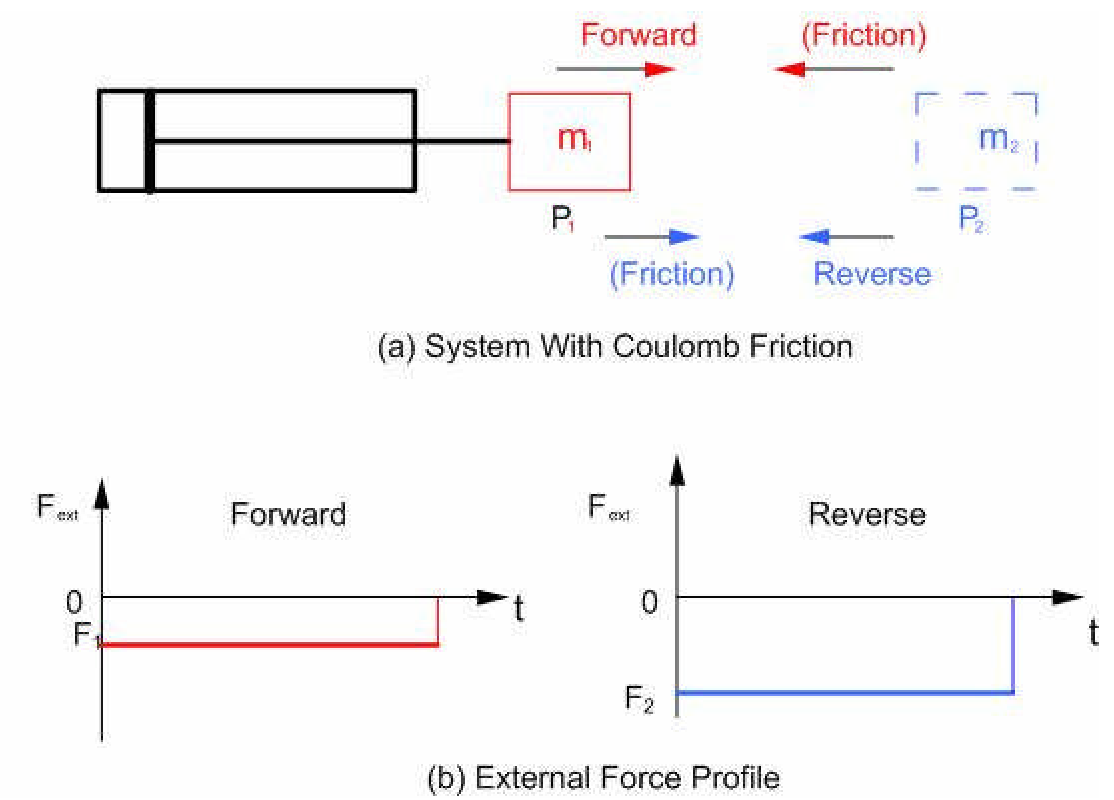


Figure 12.1 Demonstration of using the sign convention to analyze the motion of a transportation system

12.2.2 Burden and Burden profile

In the following discussions the terms "physical system" and "hydraulic system or circuit" will be frequently used. A clear distinction between the definitions of those terms is essential.

A "physical system" is referred to as the "object to be moved or acted upon".

The "hydraulic system or circuit" is that which must be configured to work against the "physical system" through its output devices.

The "hydraulic system or circuit" is that which must be configured to work against the "physical system" through its output devices (actuators). Figure 12.2 schematically illustrates this concept. Normally a hydraulic system consists of hydraulic actuators, hydraulic pumps, directional control valves, flow modulation devices etc..

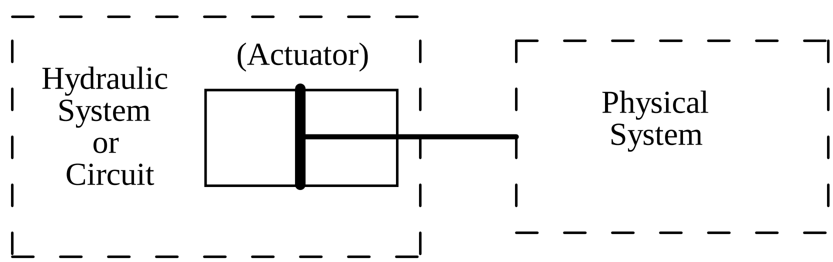


Figure 12.2 Block Diagram of "Physical System" and "Hydraulic System"

It should be pointed out that the interface between hydraulic circuits and mechanical systems may become more complex if feedback technology is involved.)

The task of hydraulic circuit design is using hydraulic components to configure a system which is able to move or act on the given physical system in the desired fashion. The design of physical systems is not considered in this work although in many situations this could be done by the same designer. It is therefore assumed that the design of the physical system is complete and all system constraints and system parameters are defined or known (calculated or approximated).

It is crucial to distinguish between the "physical systems parameters" (force/torque, velocity etc.) and the "fluid parameters" to be produced by the hydraulic circuit (pressure and flow rate) which will work against the physical system. In traditional fluid power literature, the terms "load" and "load type" are used interchangeably to describe both physical systems and hydraulic circuits (often incorrectly). To ensure that distinct terminology exists for the physical system, the hydraulic system, the physical system parameters and the hydraulic system parameters, new definitions have been adopted in this work.

Parameters of the physical system are defined as the "**burden**" of the hydraulic system. The fluid parameter of a hydraulic system, typically represented by the hydraulic force (torque) output through actuators to the physical system, is referred to as the "**hydraulic force (torque)**" (which will be discussed in subsequent section).

The burden is the sum of all external forces which act on the hydraulic actuator from the physical system.

The burden can be expressed as:

$$F_b = \Sigma F_{ext} \quad (12\text{-}1)$$

where F_b is the burden, the sum of all external forces (F_{ext}).

This definition **does** include terms such as gravity, but **does not** include inertial terms because inertial terms involve the action of hydraulic pressure.

A **burden profile** graphically represents the variations of the burden with time or displacement.

A burden and its profile can be best illustrated through an example.

Consider the system shown in Figure 12.3. In this example, the burden is comprised of friction F_f, an additional force F_e, and F_{mg} **(**numerically equal to $mg \cdot \sin\alpha$ which is the projection of gravity in the direction of motion). By using the sign convention defined earlier, the burden is given by:

$$F_b = F_e + F_f + F_{mg} \quad (12\text{–}2)$$

where F_e and F_f and F_{mg} are **negative** in our example.

The burden profile, as well as each individual force profile, would appear as shown in Figure 12.4 for the motion in the forward direction (the burden profile in reverse can be obtained in a similar fashion). It should be pointed out that the inertial term of the physical system does not appear in the burden profile. It will be shown later that inertial terms will be reflected in the "hydraulic force profile" as the requirements for the acceleration and deceleration of the physical system for a specific application are considered. The burden profile provides the designer with a complete picture of external parameters of the physical system. This does affect the eventual design of the hydraulic system as will be discussed later.

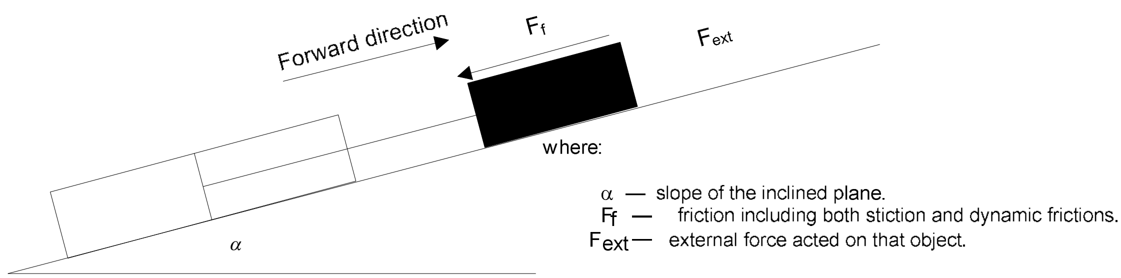


Figure 12.3 Illustration of Burden Concept

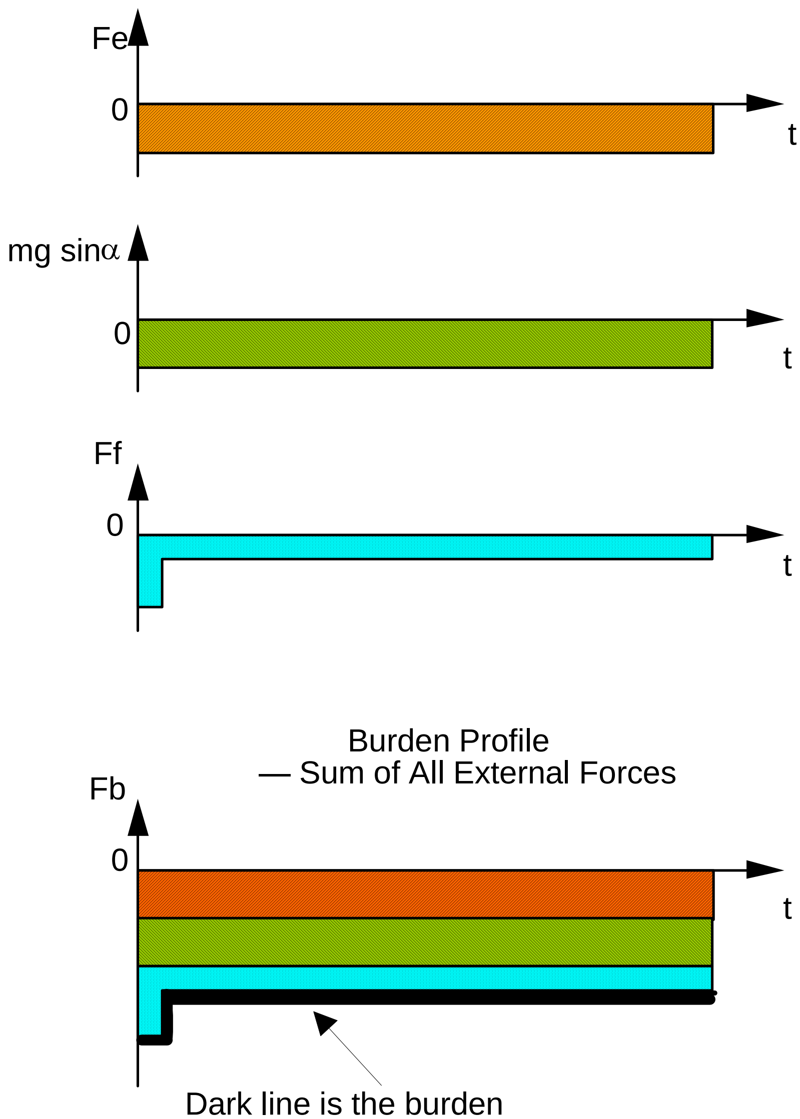


Figure 12-4 Generation of Burden Profile

12.2.3 Hydraulic Force and Hydraulic Force Profile

The **hydraulic force** (for linear motion) **or hydraulic torque** (for rotary motion) is defined as the force or torque necessary to cause the physical system to react in accordance with the specified velocity requirements. This force or torque **does include inertial terms.**

The hydraulic force in our definition is **different** from the burden in that the burden can be determined once the physical system has been identified, while the **hydraulic force (torque) depends upon both the burden and the requirements for the acceleration and deceleration of the physical system.** Indeed the hydraulic force (torque) combines the burden information with the clients' requirements and eventually represents the "fluid" parameters which the hydraulic circuit to be configured must generate in order to provide the expected motion of the physical system.

Symbolically the hydraulic force is expressed $\Delta(PA)$ or $\Delta(PD_m)$ hydraulic torque for rotary motion. The presence of the term P only indicates that the force is to be generated hydraulically. With reference to Figure 12-5 the hydraulic force (torque) can be defined mathematically as:

$\Delta(PA) = P_u \cdot A_u - P_d \cdot A_d$ For a single rod hydraulic cylinder (12-3)

$\Delta(PA) = (P_u - P_d) \cdot A$ For a double rod hydraulic cylinder (12-4)

$\Delta(PD_m) = (P_u - P_d) \cdot D_m$ For rotary motor. (12-5)

where

P_u = Pressure upstream of actuator,

P_d = Pressure downstream of actuator,

A_u = Actuator area subjected to pressure P_u ,

A_d = Actuator area subjected to pressure P_d ,

A = Actuator area for symmetric actuator,

D_m = Motor displacement.

The term **"upstream"** refers to the direction in which fluid flows **to** a component, and the term **"downstream"** refers to the direction in which fluid flows **from** a component

This is illustrated in Figure 12.5.

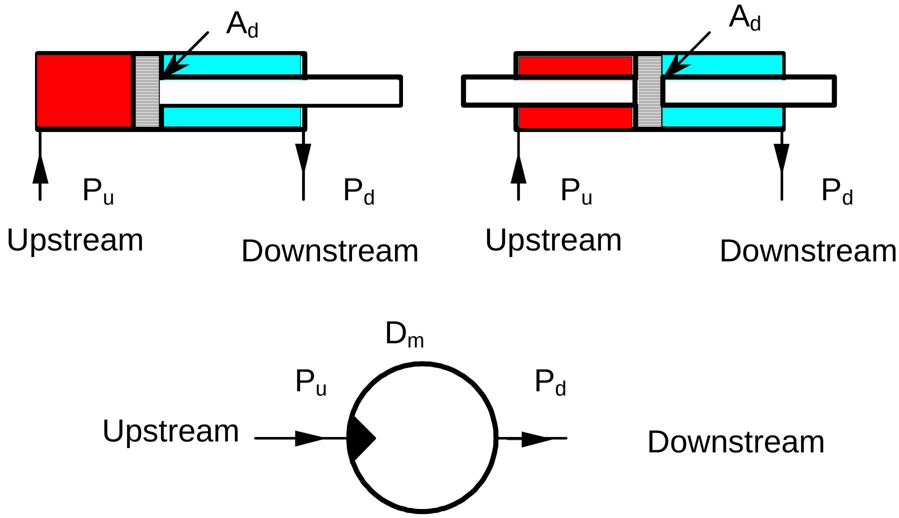


Figure 12.5 Nomenclature Used for Hydraulic Linear and Rotary Actuators

In the following discussion, linear motion is assumed; therefore only Δ(PA) (hydraulic force) is used but comments are equally valid for rotary actuators. The term Δ(PA) symbolizes the hydraulic force that a linear actuator must provide; **it is independent of circuit configuration.** Once the hydraulic force Δ(PA) has been known, the hydraulic circuit pressure can be established using Equations (12-3), (12-4) or (12-5).

A physical interpretation of Δ(PA) (in term of circuit configuration) can be summarized as follows:

(1) Δ(PA) > 0 means that the hydraulic system must provide a force in the direction of motion to push the actuator. A positive pressure difference across the actuator must exist; a back pressure in the return line is not necessary although it could be non zero.

(2) Δ(PA) = 0 indicates that no hydraulic force is required. Therefore the hydraulic force across the actuator should be maintained to zero. The system pressure could be non zero.

(3) Δ(PA) < 0 implies that a resistance to motion of the physical system must be added. This means that a back pressure P_d must be created hydraulically to provide a force opposite to the motion.

It is most important to emphasize that the sign of Δ(PA), Δ(PD_m) determines the characteristics of the hydraulic force and, consequently, will affect the decisions on the eventual circuit configuration.

The Δ(PA), Δ(PD_m) profile represents graphically the hydraulic force which must be provided over the cycle as a function of time or displacement. In order to develop the Δ(PA), Δ(PD_m) profile, sufficient information about the burden and velocity requirements of the system must be obtained through a complete understanding of the machine design of the physical system and the constraints placed on it (velocity, acceleration etc.). The burden profile reflects the constraints and parameters of the physical system to be acted upon. The velocity profile, on the other hand, reflects the requirements of the client (and hence the use of the system) and dictates how the physical system must respond.

12.2.4 Natural Acceleration or Natural Deceleration

Before proceeding, we shall use several examples which discuss the hydraulic force. To avoid having to use both hydraulic force and torque symbols every time, we shall just use hydraulic force and recognize that hydraulic torque is the same except it applies to rotary systems.

One parameter adopted in these notes to facilitate circuit design decisions is that of the "**natural acceleration**" or "**natural deceleration**" of the physical system. To illustrate this, consider a system which itself could decelerate at an acceptable rate in the absence of any hydraulic force. Hydraulically, there would be no need to provide any additional force to the system during this part of the cycle. If the natural deceleration of the system is too high (stops too fast), a hydraulic force in the direction of motion (Δ(PA)>0) must be applied to reduce this deceleration (prolong the stopping). If the natural deceleration is too small (takes too long to stop), a hydraulic force must be applied in the opposite direction of motion (Δ(PA)<0) to increase the deceleration rate (make it stop faster). This has distinct ramifications in terms of actual placement of components and hence, the natural deceleration becomes **a very useful parameter in establishing design criteria for circuit configurations.** (Natural acceleration can be interpreted in a similar fashion). As we shall see, these terms can be very valuable as a “key” in determining which set of equations we should use. This will come later so please bear with us.

Natural acceleration or deceleration is defined as **the rate of change of velocity of a physical system in the absence of any hydraulic force (Δ(PA)=0)**. The term "natural" here means the situation where the hydraulic system is absent.

Mathematically, the natural acceleration (or deceleration) can be stated as:

$$\frac{d^2x}{d^2t}\bigg|_{\Delta(PA)=0} = \frac{\Sigma F_{ext}}{M} \qquad (12\text{-}6)$$

where

ΣF_{ext} is the burden as previously defined,

$\frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0}$ is the natural acceleration or deceleration,

M is the mass of the physical system.

According to the **sign** of $\frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0}$ (remember that the direction of motion is positive) physical systems can be classified into one of the following distinct physical modes :

(1) $\frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0} \geq 0$: "Run-away"; the system accelerates and we have a **natural acceleration.**

(2) $\frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0} < 0$: "Resistive"; the system slows down by itself and we have a **natural deceleration.**

(3) $\frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0} > 0$ or $\frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0} < 0$ during a cycle: "Over-center"; we have both a natural acceleration and natural deceleration as the system passes over centre.

Consider case (1), where $\frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0} > 0$.

The physical system experiences a positive acceleration and it is deemed to be in a "run-away" condition. In the following discussions, the natural acceleration is represented by the term **A_n ($A_n = \frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0}$)** for ease of writing.

Note that $A_n = \frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0} = 0$ is the condition in which the system is moving at a constant velocity (including V = 0) and will never stop due to external forces acting on it.

Figure 12.6 illustrates a system in a "run-away" condition when the object is lowered, provided friction is less than the mg term and no hydraulic force is applied to balance this system. Its natural acceleration would be given by:

$$A_n = \frac{mg + F_f}{M} = \frac{F_b}{M} \qquad (12\text{-}7)$$

where

mg is the weight of the object and is **positive** since it acts in the direction of motion.

F_f = friction force and is **negative** in this case since it opposes motion.

m = mass of the object

NOTE: the hydraulic burden in the case would be POSITIVE since *mg > F_f* in this case. (As an aside if F_f was > than mg, the system would not move because F_f is a friction and requires motion or anticipated motion if stiction is present).

Note also that the mass of the pulley assembly is assumed very small when compared to the mass of the object, and can be neglected.)

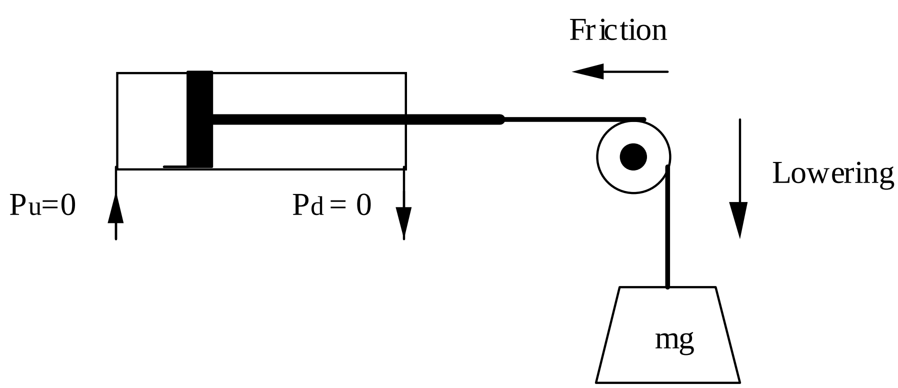


Figure 12.6 A "Run-away" System

Consider case (2) where $\frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0} < 0$.

The physical system experiences a negative natural acceleration (in fact a **natural deceleration**) and it is considered to be in a "resistive" mode. Existence of a natural deceleration indicates that the resistance to the system is sufficient to stop the system **without assistance** from the hydraulic system (whether this deceleration rate is satisfactory solely depends on whether it meets the clients' requirements). This natural deceleration D_n is written as

$\mathbf{D_n = - \frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0}}$. Note the minus sign.

We know that a positive acceleration is acceleration and a negative acceleration is deceleration. Since we normally talk about deceleration as a positive number, we use the minus sign to accommodate this. (ie a negative acceleration times a negative sign is a positive sign).

A typical example of the physical system in a "resistive' mode was illustrated in Figure 12.3, in which the object was moving up on the inclined plane by a hydraulic cylinder. With reference to Equations (12-2) and (12-6), the natural deceleration for this system is calculated from:

$$D_n = -(\frac{F_e + F_f + F_{mg}}{m}) = -F_b \tag{12.8}$$

Note that F_e , F_f , and F_{mg} (numerically equal to $mg \cdot \sin\alpha$) are actually negative according to our sign convention. Hence D_n is a positive value

As mentioned above, the minus sign outside the bracket is due to the definition of $D_n = - \frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0}$.

Consider case (3) where $\frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0} > 0$ or $\frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0} < 0$ during a cycle.

The physical system is in the "Over-center" mode. If the calculated natural acceleration can be both greater and less than zero (they both can not occur at the same time) then a situation exists in which the burden switches from being a force which opposes motion to one which assists motion. This is commonly called an "over-center" system. An application with an "over-center" property is schematically illustrated in Figure 12.7. In this example, because of the system geometry and for the direction illustrated, the external forces to the actuator initially oppose the motion ($\frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0} <0$, "resistive" mode – natural deceleration) and then changes to assist the motion ($\frac{d^2x}{d^2t}\Big|_{\Delta(PA)=0} > 0$, "run-away" mode – natural acceleration) as the system is over center (α = 90 degree). The external forces vary as the angle α changes.

In this case the system experiences both a natural deceleration ($\alpha < 90$) and a natural acceleration for $\alpha > 90$.

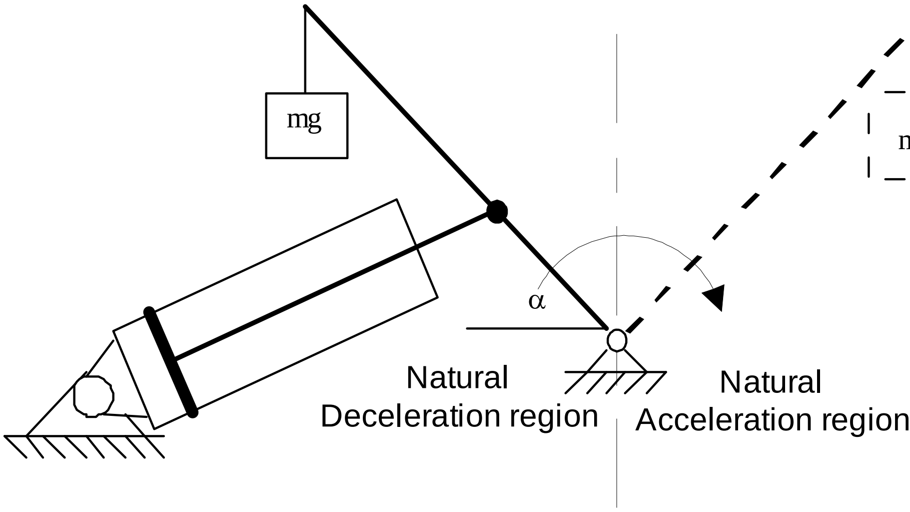


Figure 12.7 An "Over-center" Application

As will be shown, it is the natural acceleration or deceleration that dictates what is needed hydraulically to cause the desired motion in the acceleration or deceleration parts of the cycle.

12.2.5 Summary:

Natural acceleration $\boldsymbol{A_n = F_b}$ where the burden is positive
Natural deceleration $\boldsymbol{D_n = - F_b}$ where the burden is negative.

If the system has a negative force burden, then it will have a natural deceleration in the absence of a hydraulic force
If the system has a positive force burden, then it experiences a natural acceleration in the absence of a hydraulic force.

Natural angular acceleration $\boldsymbol{\alpha_n = T_b}$ where the burden is positive
Natural angular deceleration $\boldsymbol{\alpha_n = - T_b}$ where the burden is negative.

If the system has a negative torque burden, it will have a natural angular deceleration in the absence of a hydraulic torque
If the system has a positive torque burden, the it experiences a natural angular acceleration in the absence of hydraulic torque.

12.3 Establishing The Hydraulic Force Profile

The purpose of establishing the hydraulic force profile is to ascertain the "fluid" parameters which the hydraulic system must be able to produce to meet the desired system functions. This is done by analyzing the conditions or constraints of a given physical system (determining its burden profile) and the job functions (obtaining its velocity profile). The steps to develop the hydraulic force profile are as follows.

(1) Obtain the mass profile of the physical system.
(2) Obtain the velocity profile.
(3) Establish the burden profile.
(4) Calculate the range of natural acceleration or deceleration.
(5) Determine the hydraulic force required for the system to follow the desired velocity profile based on the information given in the burden profile and a comparison of natural deceleration or acceleration to the user specified acceleration or deceleration.

12.3.1 The Mass (Mass Inertia) Profile

The mass profile portrays the mass to be moved as a function of time or displacement. Any changes of the mass in mid-stroke or from cycle to cycle must be included. A typical mass profile reflects all possible mass combinations for both forward and reverse directions. There could be a maximum and a minimum mass at any particular time or displacement as illustrated in Figure 12.8. The mass profile is used to quantify external forces which are a function of mass and to evaluate the natural deceleration or acceleration as will be shown later.

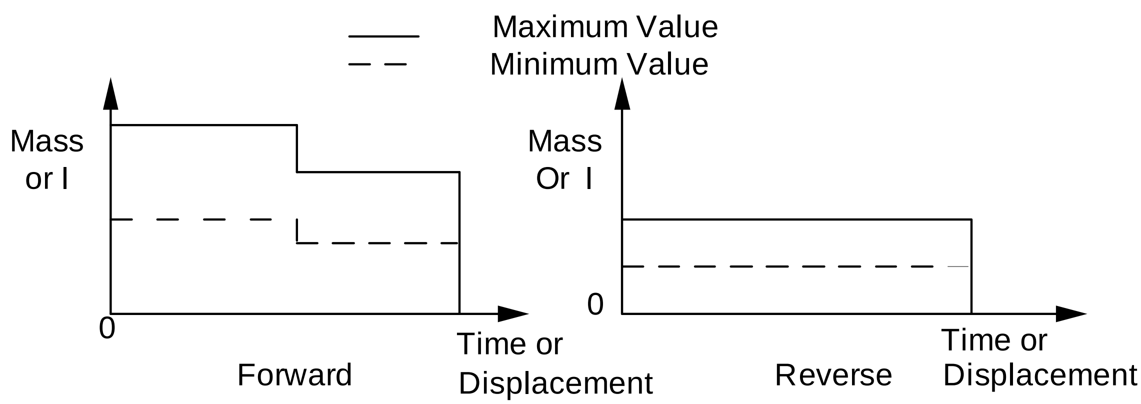
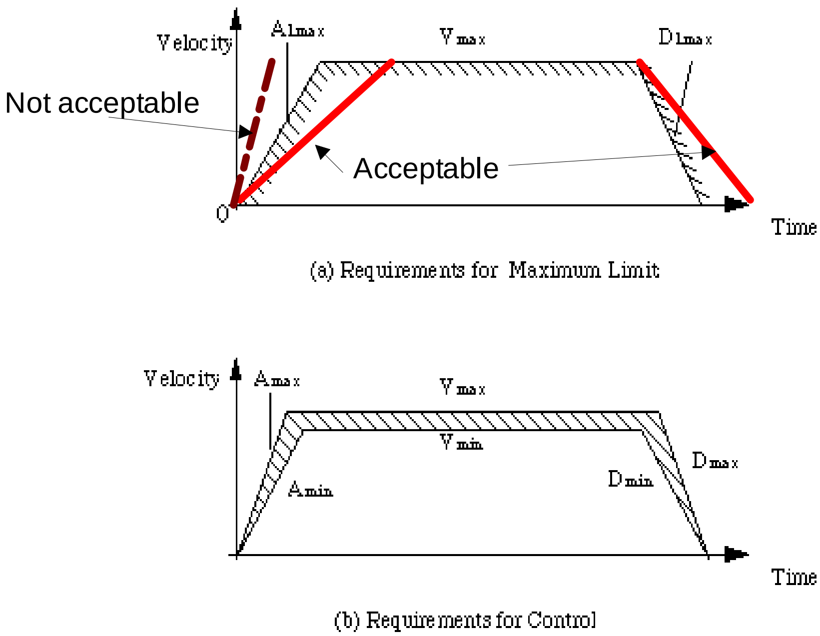


Figure 12.8 A Typical Mass Profile

12.3.2 The Velocity Profile

The velocity profile is a plot of the client's requirements for the velocity of the actuator as a function of time or displacement. The velocity profile tells the designer both the velocity and the acceleration and or deceleration requirements of the system. These requirements must be specified either as a "**limit**" (minimum or maximum) or as a "**control**" (±5% in general or ±2% for system feedback) and can be reflected on the velocity profile. Figure 12.9 (a) illustrates a typical velocity profile with requirements for limiting steady state velocity and acceleration and deceleration to be less than certain values specified. If the system velocities are equal to or fall below a specified value during the operation of the system (as shown as the dashed area), they are considered to be acceptable. In Figure 12.9 (b) the velocity profile represents the requirements for controlling steady state velocity, acceleration and deceleration in order to maintain the velocity parameters to certain values with an allowable deviation (the tolerance of error) indicated by the lined area.



The two solid lines in (a) would represent acceptable velocities of the load actuator but the dashed line in (a) would not meet the requirements and would not be acceptable.

Figure 12.9 Velocity Requirements for Limiting (a) and Controlling (b)

12.3.3 The Burden Profile

The burden, as previously defined, is the sum of all forces acting on the actuator which is a consequence of the physical system. The burden includes external forces such as that due to gravity and friction terms. All of these forces must be identified over the cycle both for the forward and reverse directions (where applicable).

To illustrate how a burden profile is established, a saw application is considered. As illustrated in Figure 12.10 (a) an external force is a result of the cutting torque produced by the rotary blade. The force on the block is in **the direction of motion** (assuming the vertical projection of this external force can be ignored). By the sign convention this external force is **positive** and is plotted as the function of displacement, shown in Figure 12.10 (b) (Note that friction is not yet included in this force profile).

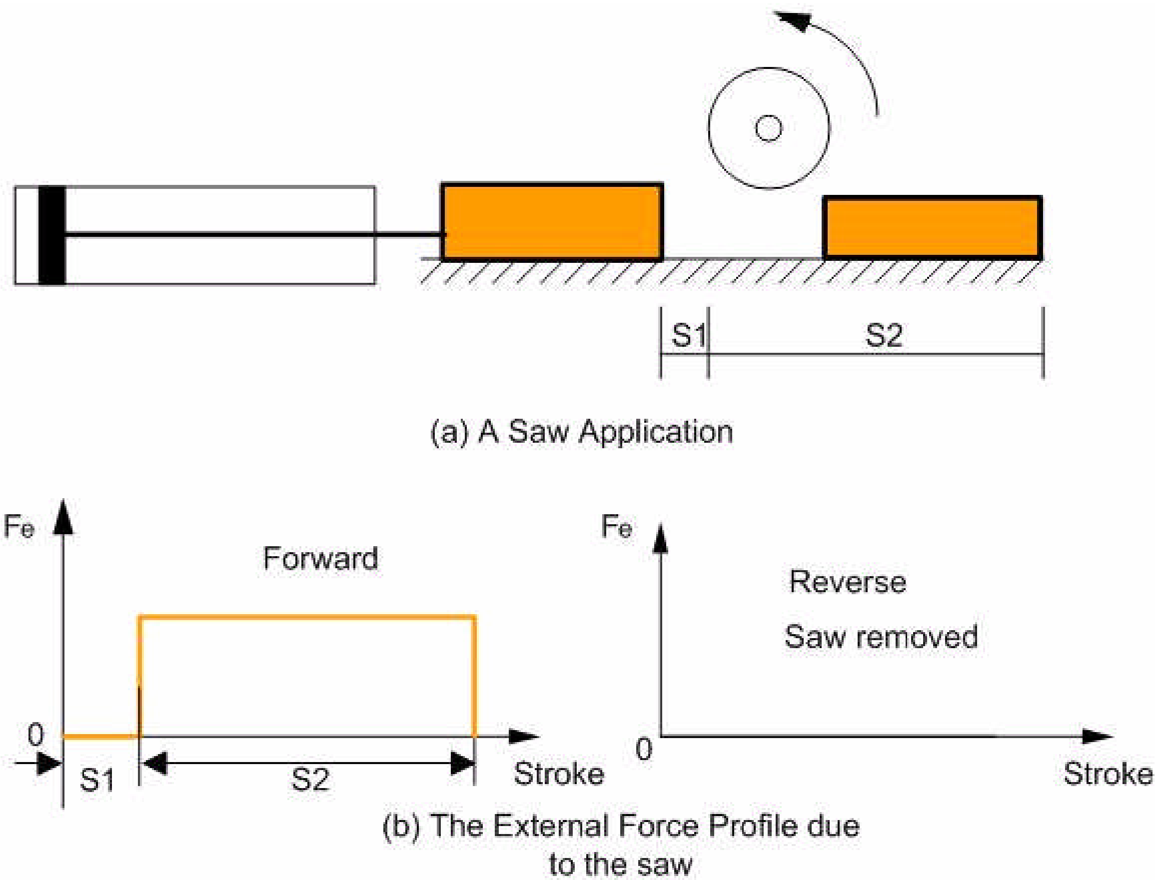


Figure 12.10 Determine the External Force for a Saw Application

The friction experienced by the block consists of stiction, dynamic friction and viscous friction. If the velocity requirement is proposed as given in Figure 12.11 (acceleration and deceleration are not specified since they are not dominant in this application), then the sum of friction forces would appear as shown in Figure 12.12. Because the block is unloaded in reverse and the friction is much less, the friction can be neglected for this part of the cycle.

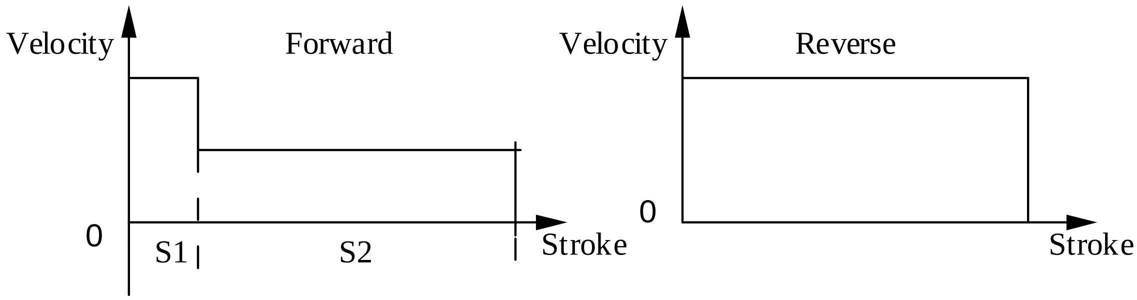


Figure 12.11 A typical Velocity Profile for the Saw Application

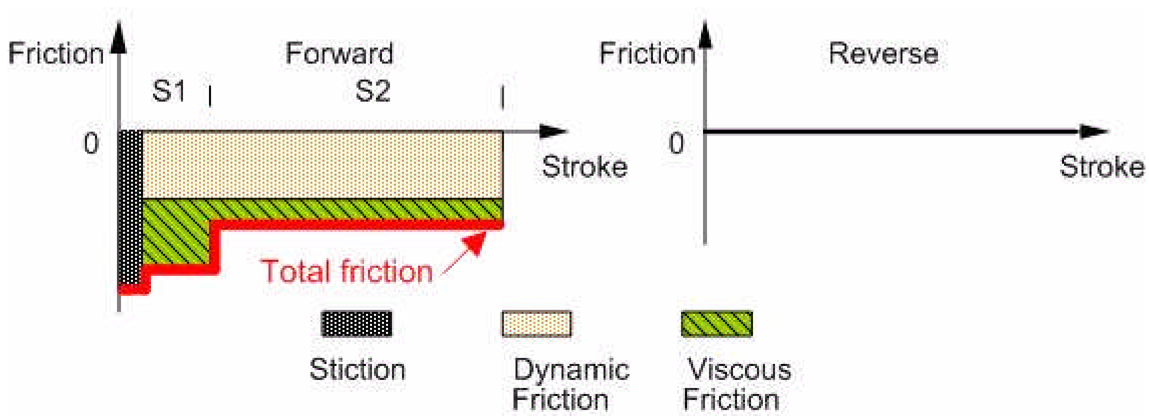


Figure 12.12 Friction Forces Profile for the Saw Application

By algebraically combining the external and friction forces together, the burden profile would appear as plotted in Figure 12.13 for this example.

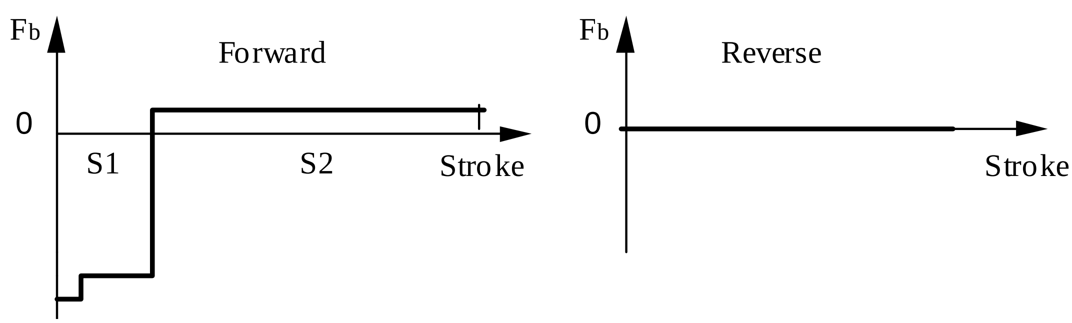


Figure 12.13 Burden Profile for the Saw Application

The burden profile in the cited example is relatively straightforward. However if during the operation of the system, external forces, friction forces or mass were known to vary, then the variations must be reflected in individual force profiles or mass profile, and subsequently the burden profile. Consequently the burden profile is usually presented using maximum and minimum values. To illustrate, consider the force profile in Figure 12.14. If the mass is expected to be changed between m_1 and m_2 as shown in the mass profile in Figure 12.14, then the friction force and gravity must be calculated using m_1 and m_2 respectively to evaluate the range in which the forces vary. The resulting burden profile would represent these variations accordingly. This is illustrated in Figure 12.14.

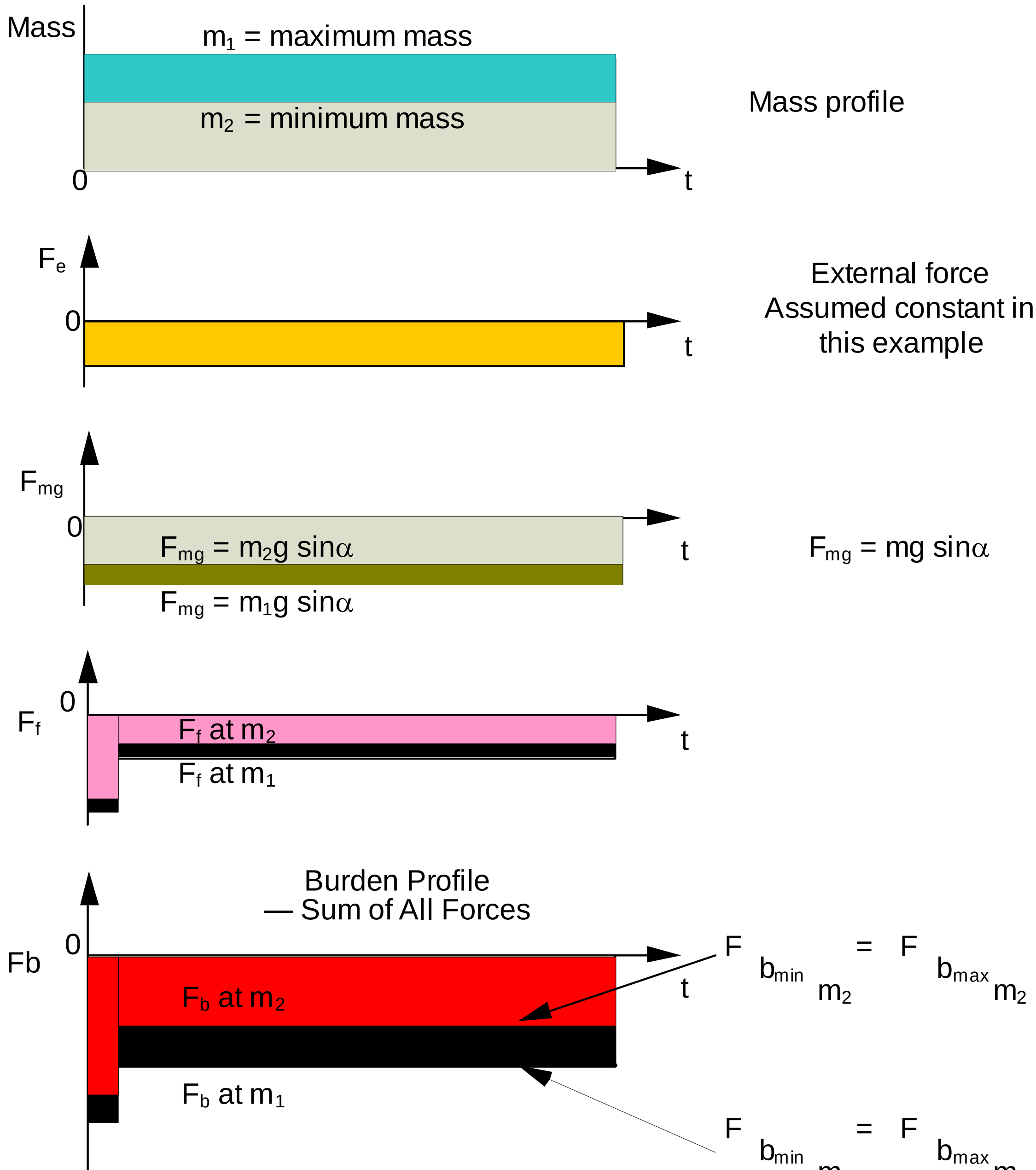


Figure 12.14 Burden Profile with Maximum and Minimum Values

It is important to note that the forces acting on the actuator are often the hardest to quantify. Sometimes it is very difficult to define the operating conditions and hence the various friction coefficients. This is a situation where "best guess" and "worst case scenarios" must be applied. Other approaches could involve observing systems with similar operating conditions and estimating the external force and friction values from them. Regardless, a best guess should be made but must always reflect the worst possible case.

In the example of Figure 12.14, it is clear that the burden will vary depending on what mass is used. This can get even more complex of say, the external force also varied from cycle to cycle. The burden profile would thus reflect an even more complex shape which makes the design of the final hydraulic circuit very difficult because of the wide variations in the operating conditions.

To facilitate the calculations of the hydraulic force profile from the burden profile and the inertial terms, we will make use of the natural acceleration and deceleration terms as a “key” in a Table of hydraulic force (torque) formulas. This will come.

12.3.4 The Natural Acceleration or Deceleration

As defined earlier, the natural acceleration or deceleration is determined by the burden and mass and mathematically is given by:

natural acceleration or deceleration	natural angular acceleration or angular deceleration
$\mathbf{A_n} = \left.\dfrac{\mathbf{d^2x}}{\mathbf{d^2t}}\right\rvert_{\Delta(PA)=0} = \dfrac{F_b}{M}, \quad F_b>0$	$\alpha_n = \dfrac{d^2\theta}{dt^2} = \dfrac{T_b}{I}, \quad T_b>0$
$\mathbf{D_n} = -\left.\dfrac{\mathbf{d^2x}}{\mathbf{d^2t}}\right\rvert_{\Delta(PA)=0} = \dfrac{-F_b}{M}, \; F_b<0$	$D\alpha_n = \dfrac{d^2\theta}{dt^2} = -\dfrac{T_b}{I}, \quad T_b<0$

(12-9)

In the following discussion, it will be assumed that the system exhibits a natural deceleration, $\left.\dfrac{d^2x}{d^2t}\right\rvert_{\Delta(PA)=0} < 0$; therefore, only natural deceleration ($D_n = -\left.\dfrac{d^2x}{d^2t}\right\rvert_{\Delta(PA)=0}$) is used. Comments, however, can be equally applied to the situations where the system has a natural acceleration (linear or angular).

In calculating natural deceleration, if the burden and mass vary during the cycle and from cycle to cycle (as the usual case), the maximum natural deceleration $D_{n.max}$ and the minimum natural deceleration $D_{n.min}$ should be determined to evaluate a range of deceleration values. To illustrate, refer to Figure 12.14 again. The mass profile indicates that there are two different masses, m_1 and m_2 to be expected over the stroke. $D_{n.min}$ and

$D_{n.max}$ should be calculated by Equation (12-9) using these two values, and the friction and gravity terms corresponding to each particular mass respectively.

$$D_{n1} = \left. \frac{F_f|_M + F_e + F_{Mg}}{M} \right|_{M=m_1} = \left. \frac{-F_b}{M} \right|_{M=m_1} \tag{12-10}$$

$$D_{n2} = \left. \frac{F_f|_M + F_e + F_{Mg}}{M} \right|_{M=m_2} = \left. \frac{-F_b}{M} \right|_{M=m_2} \tag{12-11}$$

$D_{n.max}$ or $D_{n.min}$ is found from by calculating both terms.

A typical natural deceleration with a maximum and a minimum value is graphically presented in Figure 12.15. The dotted area "bounded" by $D_{n.max}$ and $D_{n.min}$ indicates all possible natural decelerations that may be experienced when the system is in its "natural" state.

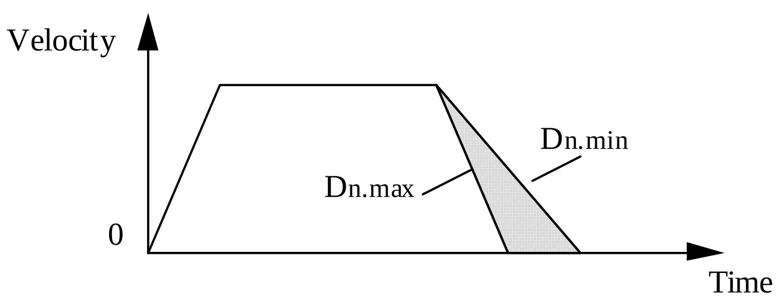


Figure 12.15 Demonstration of Natural Deceleration with Maximum and Minimum Values

It must be pointed out that the values of burden and mass used for the calculation of natural deceleration should have a one to one correspondence to avoid "over-design", and they should be chosen from the burden and mass profiles at which $D_{n.max}$ and $D_{n.min}$ occur respectively. In many applications where the system has a mass profile like the one shown in Figure 12.14, $D_{n.max}$ is calculated using the maximum burden and the minimum mass:

$$D_{n.max} = \frac{-F_{b.max}|_{M_{min}}}{M_{min}}$$

while $D_{n.min}$ is calculated with the minimum burden and the maximum mass:

$$D_{n.min} = \frac{-F_{b.min|_{M_{max}}}}{M_{max}} .$$

The easiest way to use these equations is to graph the burden first and then find D_{nmax}, D_{nmin} , A_{nmax} , A_{nmin} etc.

Note: In this example $-F_{b.max|_{M_{min}}} = -F_{b.min|_{M_{min}}}$ so problems arise. However, in some other situations $D_{n.max}$ is calculated using both maximum burden and mass or minimum burden and mass. Such a scenario is illustrated in Figure 12.16 where $D_{n.max}$ occurs at portion B of the cycle and is calculated from

$$D_{n.max} = \frac{-F_{b.min|_{M_{min}}}}{M_{min}} ,$$

and $D_{n.min}$ occurs at portion A and is calculated from

$$D_{n.min} = \frac{-F_{b.max|_{M_{max}}}}{M_{max}} .$$

Therefore Equations (12-10) and (12-11) must be applied with great care to reflect realistic scenarios.

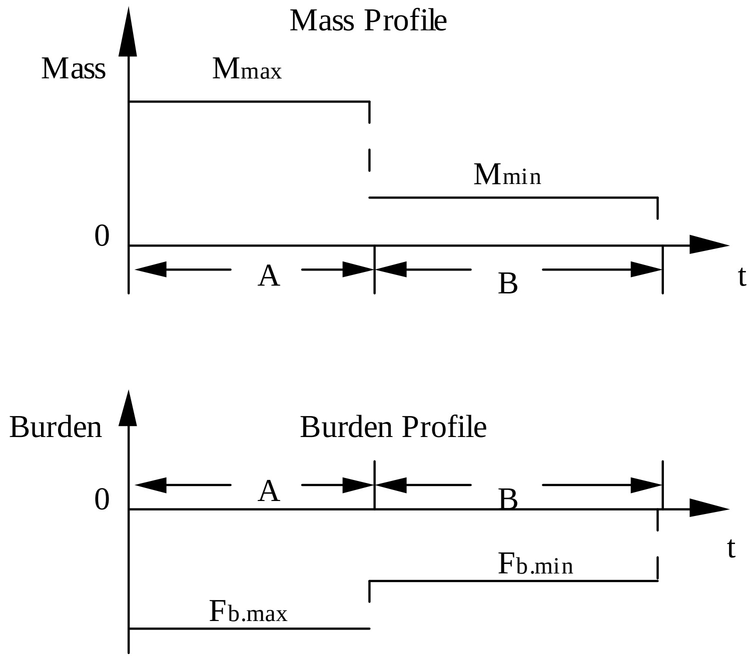


Figure 12.16 Burden and Mass Profiles

In Equation (12-9) the expression for F_b ($F_b = \sum F_{ext}$) can include constant terms (over the whole stroke) and forces which vary with the displacement (such as a spring-type force), velocity or derivatives of the velocity. In some situations, this may involve a complex procedure for evaluating the natural deceleration or acceleration. However since the only concern is the range of natural deceleration limits rather than the exact shape, it is sufficient to estimate those values ($D_{n.max}$ and $D_{n.min}$) for a "worst case" scenario. For the majority of practical applications the burden profile is dominated by gravity, friction, spring or other forces which are not a function of velocity or its derivatives. In these cases the calculation of natural deceleration is generally straightforward. For applications, which do have a complicated burden profile, usually a sophisticated servo-control technique will be necessary to achieve velocity control or limit. These applications are beyond the scope of this study at present.

It should be understood that under some circumstances, it is extremely difficulty (or indeed impossible) to give a detail description of the burden and mass. The maximum and the minimum values of burden and mass over the whole cycle must be predicted based on a "best guess" or "worst case scenario". The natural deceleration range can then be estimated using the largest burden and smallest mass (for $D_{n.max}$) and the smallest burden and largest mass (for $D_{n.min}$) as shown by the expression:

$$D_{n.min} = \frac{-F_{b.min}}{M_{max}} \qquad (12\text{-}12)$$

$$D_{n.max} = \frac{-F_{b.max}}{M_{min}} \qquad (12\text{-}13)$$

where $F_{b.min}$ and $F_{b.min}$ are magnitudes of maximum burden and minimum burden respectively.

(NOTE: if you have trouble keeping things straight, create a separate burden profile for each mass and calculate the natural deceleration/acceleration for each burden profile, then choose the maximum and minimum from these results.)

Lets us now do an example to illustrate how to calculate natural acceleration or deceleration. Consider the system shown in Figure 12.16(a)

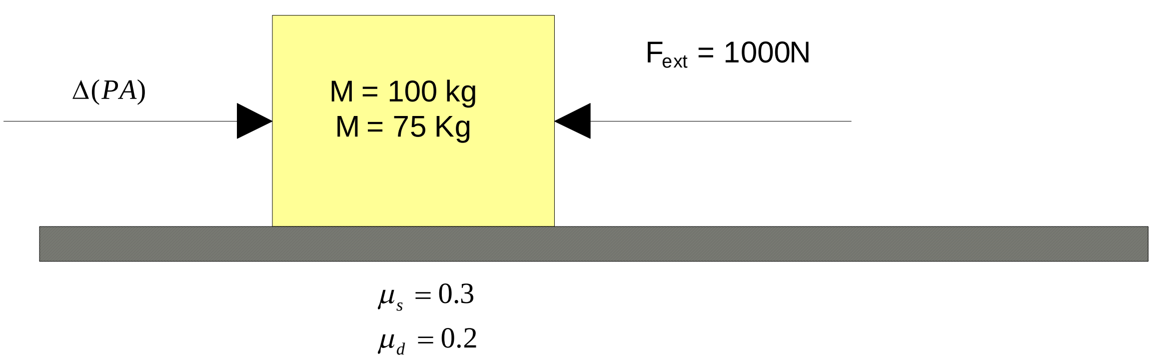


Figure 12.16(a) Example problem system

Objective: to determine the natural acceleration or deceleration for the system shown

Our first step is decide if we are taking about a natural acceleration or deceleration. The load is resistive because we have an external load and friction which act in a direction so as to slow down the system. So we are talking about **natural deceleration**.

Our first step is to establish the mass and burden profiles.

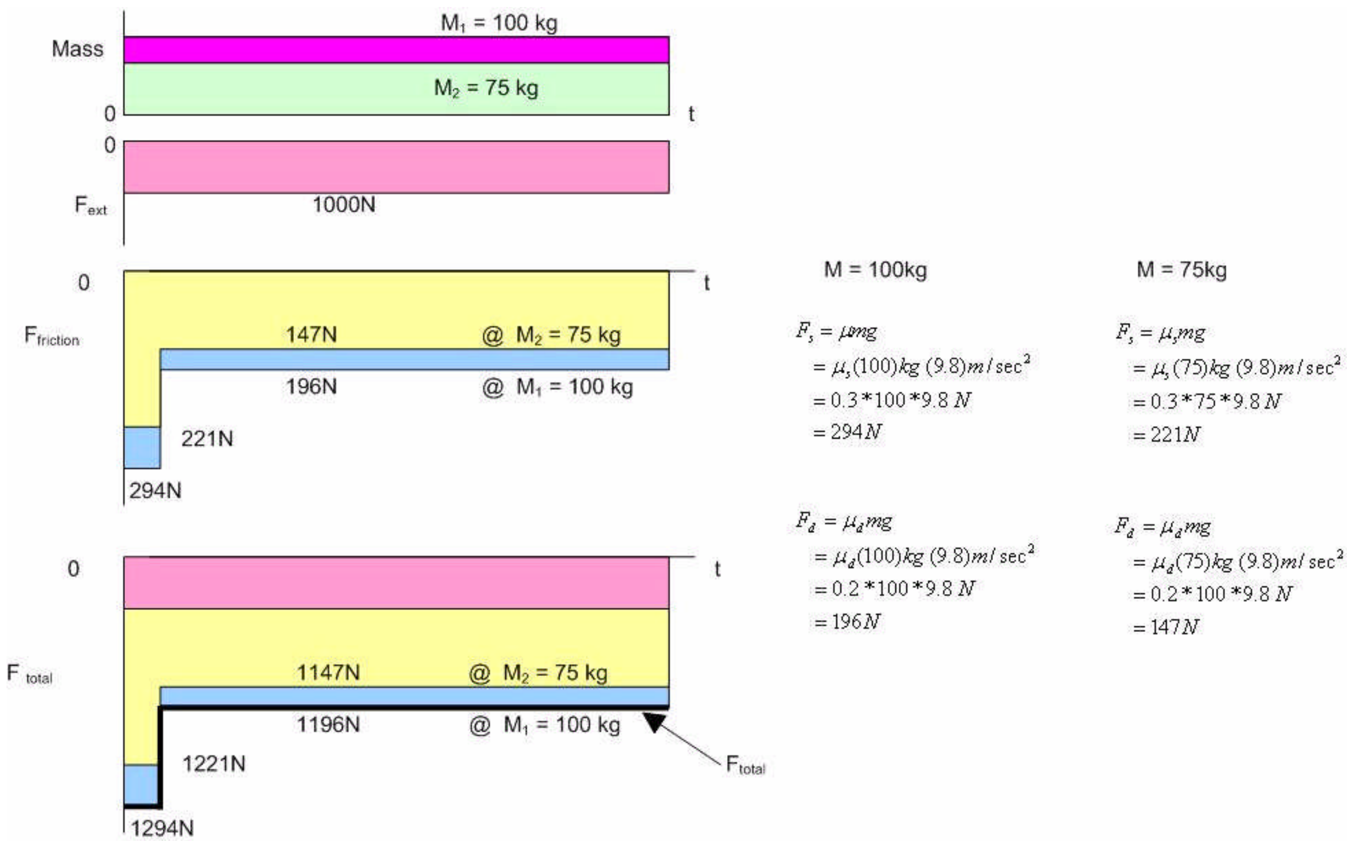


Figure 12.16(b) Mass and burden profiles

Now, we know that the burden profile is negative. Therefore

$$D_n = -\frac{F_b}{M}$$

Examine when t = 0. No motion has started so we cannot have a natural deceleration in the part, that is the mass must be moving to have a natural deceleration. So we do not need to consider this part of the cycle. Therefore we shall consider only the steady state part. We have two masses which create two regions for the burden profile. So we have two possible natural decelerations. These are:

$$D_{n1} = -\frac{-1147N}{75kg} = 15.3\frac{m}{\sec^2}$$

$$D_{n2} = -\frac{-1196N}{100kg} = 11.96\frac{m}{\sec^2}$$

Therefore our natural deceleration plot would appear as:

Figure 12.16(c) Natural deceleration limits

12.3.5 The Hydraulic Force (Torque) Profile

The step in this procedure is to combine the information given in the burden and the velocity profiles to produce the $\Delta(PA)/\Delta(PD_m)$ profile. In doing so, it has been assumed that sufficient information about the burden and velocity requirements of the system can be acquired through interrogation of the clients or at least through an "informed" best guess.

In order to develop a $\Delta(PA)/\Delta(PD_m)$ profile, the steady state, acceleration and deceleration portions of the velocity profile should be examined separately. During steady state, $\Delta(PA)/\Delta(PD_m)$ is a balance of steady state forces(torques) and is readily determined from the burden profile. However, for acceleration and deceleration portions of the cycle, specifications by the clients for acceptable acceleration and deceleration values dictate $\Delta(PA)/\Delta(PD_m)$, given the existing burden at that part of the cycle.

For example, consider a transportation system driven by a hydraulic cylinder. If tall boxes are moved over a distance, it is conceivable that the system must be decelerated at a rate less than a certain value (denoted as maximum deceleration $D_{l.max}$) to prevent the boxes from "toppling over". If the natural deceleration is larger than $D_{l.max}$ then a hydraulic force in the direction of motion **must be added** to slow down the system at a rate less than $D_{l.max,}$ The magnitude of the hydraulic force can be determined from:

$$\Delta(PA) = -F_b - M \cdot D_{l.max} \quad \text{(if the mass changes, beware: this gets more complex)} \qquad (12\text{-}14)$$

In Equation (12-14), the burden Fb is negative and $-F_b$ represents a positive value. $\Delta(PA)$ is positive because $-F_b > M \cdot D_{l.max}$. The positive value of $\Delta(PA)$ in (12-14) yields the smallest allowed hydraulic force which must be generated and applied **in the direction of** motion to meet the deceleration requirement.

In contrast, consider the situation where the system is to transport short boxes which have the same mass and burden profiles as the tall boxes; the system constraints may be that it must be stopped as quick as possible, i.e. the acceptable deceleration must exceed a certain rate $D_{l.min.}$ If the burden forces are not sufficient to decelerate the system at a rate higher than $D_{l.min}$, then a hydraulic force in the opposite direction of motion ($\Delta(PA) < 0$) is necessary to add resistance to the system in order to achieve a higher deceleration. This $\Delta(PA)$ can be calculated from:

$$\Delta(PA) = -F_b - M \cdot D_{l.min} \quad \text{(if the mass changes, again beware: this gets more complex)} \qquad (12\text{-}15)$$

$\Delta(PA)$ is negative due to $-F_b < M \cdot D_{l.min}$. The negative value of $\Delta(PA)$ means that a hydraulic force must be applied to the system in the opposite direction of motion. Although the burden profiles for both cases are similar, the hydraulic force profiles take on different shapes due to the different acceleration and/or deceleration requirements. These two scenarios are illustrated in Figure 12.17.

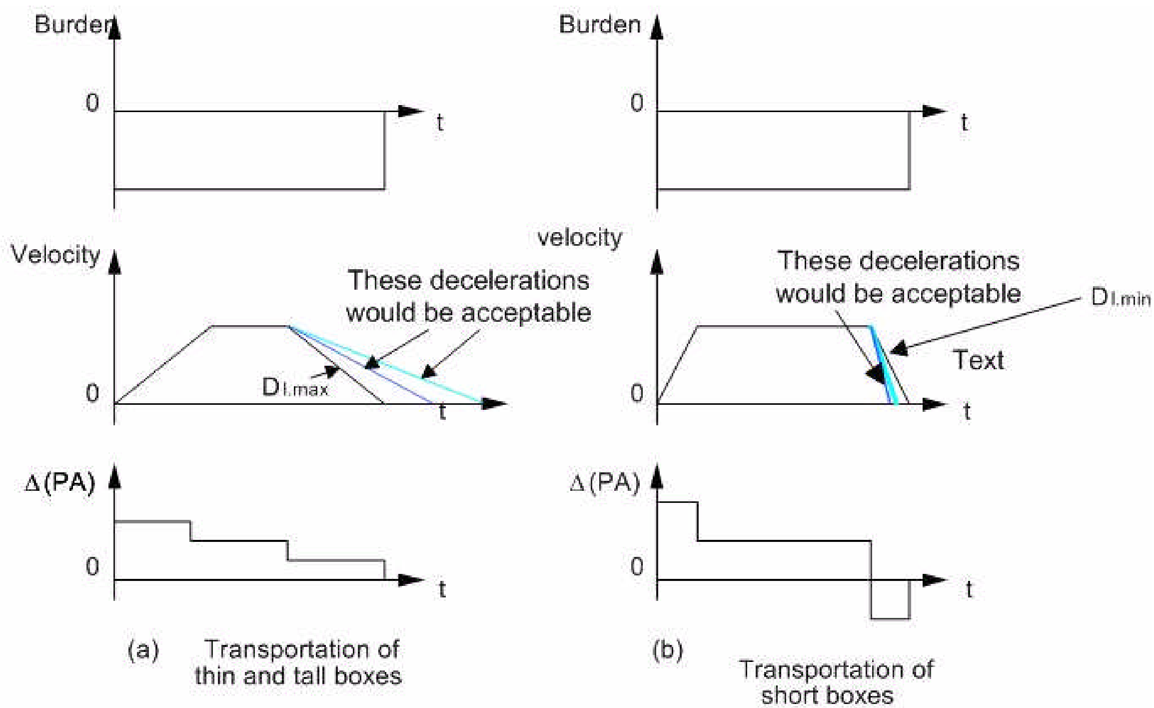


Figure 12.17 Development of Δ(PA) Profile for Transportation Systems

12.3.6 Summary

In Section 12.2 the philosophy behind the design process introduced is that the constraints or conditions of the physical system and the fluid parameters produced by the hydraulic system are separate and subsequent design decisions are based on a combination of the existing "physical parameters" and the client's requirements of the system. To implement this new approach, several new terms were adopted. The term natural deceleration or acceleration was used to identify the conditions of a physical system in its "natural" state. The burden represents all forces acting on the actuator in the "natural" state, and the hydraulic force Δ(PA) and hydraulic torque $\Delta(PD_m)$ quantitatively defines the force (torque) supplied by the hydraulic circuit to overcome the burden **and** to achieve the desired motion (meet the specifications of the acceleration and deceleration parts).

In the following sections, the discussion on the development of the hydraulic force (torque) profile will be extended. The three distinct conditions identified by the sign of the natural acceleration will be examined and the method to determine the hydraulic force (torque) under each condition will be presented. Several examples will be considered to illustrate the procedure to find the hydraulic force profile.

12.4 Development of Equations for Determining the Hydraulic Force and Torque Profile

As presented in the last sections, physical systems can be classified by their natural deceleration or acceleration as one of the following three distinct catalogues:

(1) $\left.\frac{d^2x}{d^2t}\right\vert_{\Delta(PA)=0} \geq 0$: "Run-away"; the system accelerates and we have a **natural acceleration where** $A_n = \left.\frac{d^2x}{d^2t}\right\vert_{\Delta(PA)=0} = \frac{F_b}{M}$, $F_b>0$	$\left.\frac{d^2\theta}{dt^2}\right\vert_{\Delta(PD_m)=0} \geq 0$: "Run-away"; the system accelerates and we have a **natural angular acceleration where** $\alpha_n = \frac{d^2\theta}{dt^2} = \frac{T_b}{I}$, $T_b>0$
(2) $\left.\frac{d^2x}{d^2t}\right\vert_{\Delta(PA)=0} <0$: "Resistive"; the system slows down by itself and we have a **natural deceleration where** $D_n = -\left.\frac{d^2x}{d^2t}\right\vert_{\Delta(PA)=0} = \frac{-F_b}{M}$, $F_b<0$	$\left.\frac{d^2\theta}{dt^2}\right\vert_{\Delta(PD_m)=0} < 0$: "Resistive"; the system slows down by itself and we have a **natural angular deceleration where** $D\alpha_n = \frac{d^2\theta}{dt^2} = -\frac{T_b}{I}$, $T_b<0$
(3) $\left.\frac{d^2x}{d^2t}\right\vert_{\Delta(PA)=0} > 0$ or $\left.\frac{d^2x}{d^2t}\right\vert_{\Delta(PA)=0} < 0$ during a cycle: "Over-center"; we have both a natural acceleration and natural deceleration as the system passes over centre.	$\left.\frac{d^2\theta}{dt^2}\right\vert_{\Delta(PD_m)=0} \geq 0$ or $\left.\frac{d^2\theta}{dt^2}\right\vert_{\Delta(PD_m)=0} < 0$ during a cycle: "Over-center"; we have both a natural angular acceleration and natural angular deceleration as the system passes over centre

In this section we will develop the necessary equations to calculate minimum or maximum hydraulic force (torque) values during acceleration, deceleration and steady state conditions. We must remember that if the natural acceleration/deceleration falls within the acceptable regions as defined by the user constraints, then we can let the system "coast" to a stop or accelerate on its own **without the help of any hydraulics;** that is, the ports of the actuators of the motors can be "shorted" out or connected to tank during the appropriate part of the cycle.

If the natural acceleration/deceleration falls outside the acceptable region, we must either set an upstream or a down stream pressure to force the system to satisfy the required constraints. This section will consider just one example to show how these equations are developed. The equations are very basic but defining whether one uses maximum or minimum values in the equations requires some thought. We will summarize all scenarios in the form of charts which should save a lot of time in determining which equation should be used. The one example we will discuss in detail is that in which the system exhibits a natural deceleration. (Commonly called a "resistive" load in the literature.)

Please note that we have chosen to use hydraulic force but the analysis also applies to the hydraulic torque. Our final Tables will make use of hydraulic torque as well.

12.4.1 Hydraulic Profile Equations for a $\Delta(PA) > 0$.

In order to demonstrate the procedure for developing the hydraulic force profile for a system exhibiting a natural deceleration, the hydraulic actuator and trolley system schematically illustrated in Figure 12.18 is used as an example throughout the following discussions.

In this example, the cylinder is required to move the trolley from A to C. The objects on the trolley could be added or taken off at any position (such as B), along this path. Friction forces and other forces to this system are always opposite to the motion in both forward and reverse directions (burden appears negative on the burden profile); therefore this system is considered to be in a "resistive" condition in which the system has a natural deceleration. Mathematically, the same conclusion can be ascertained from the calculation of the natural deceleration using equation:

$\frac{d^2x}{d^2t} = \frac{Fb}{M}$ (in fact, according to our definition, $D_n = - \frac{Fb}{M}$) .

The value of $\frac{d^2x}{d^2t}$ is negative because the burden remains negative. For simplicity, only the motion in the forward direction is considered in this example. However, discussions for the motion in reverse can be performed in the same manner.

The spatial and time requirements of the system can be represented as a velocity profile. Normally a velocity profile is comprised of acceleration, deceleration and steady state portions, such as that shown in Figure 12.19. At each portion of the profile, limit or control of the parameter of interest must be specified (not shown in this figure). In some situations, the control or limit of acceleration and/or deceleration may not be of prime concern; hence constraints on the acceleration and deceleration will not appear in the velocity profiles and do not have to be considered in the design process. It must be recognized that the system acceleration and/or deceleration can occur anywhere in the cycle. For example, sudden variations in the burden or emergency "shut-down" operations are not uncommon occurrences which can be experienced in the cycle.

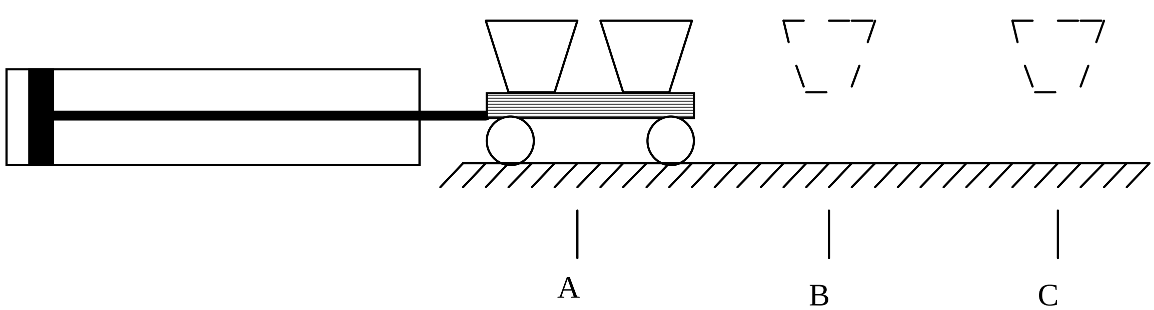


Figure 12-18 System With A Natural Deceleration

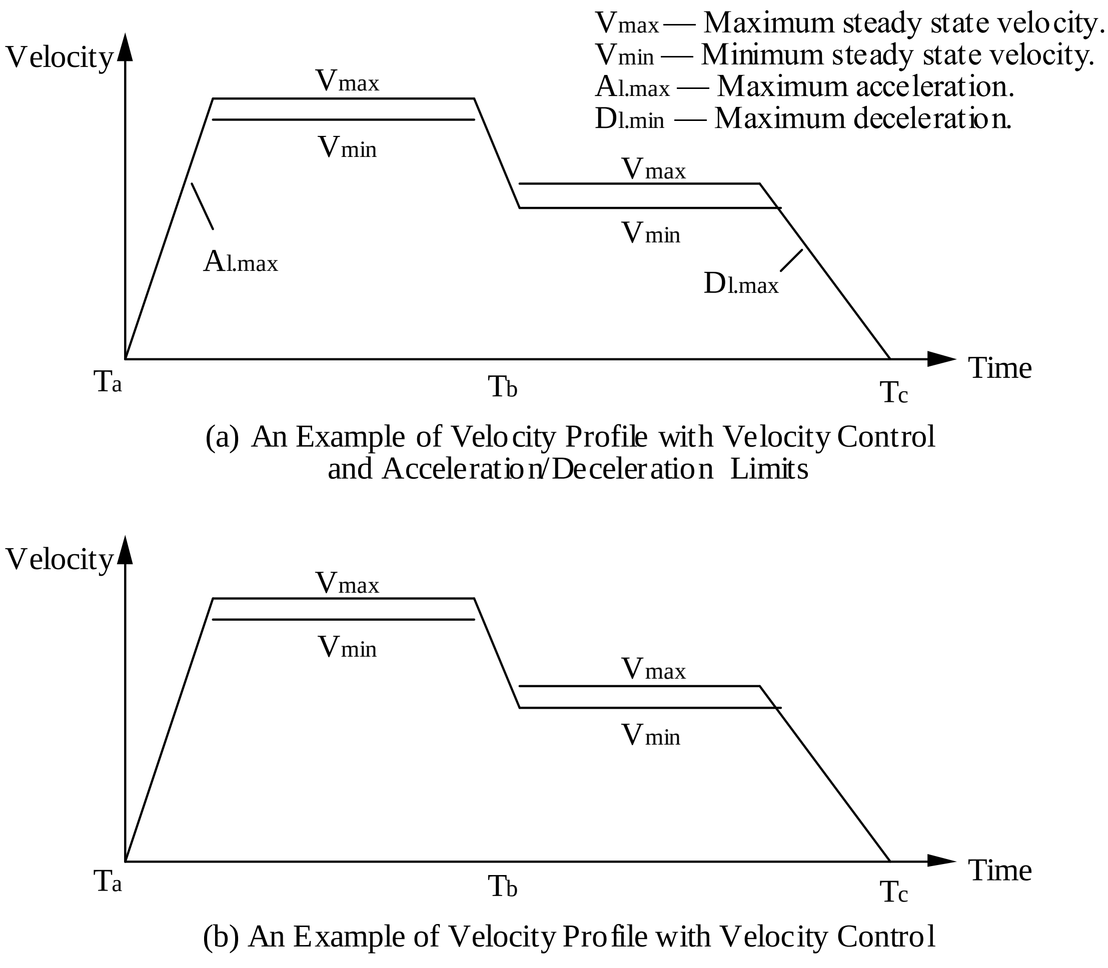


Figure 12.19 Typical Examples of Velocity Profile

In order to establish the hydraulic force profile, the mass profile and the velocity profile must be obtained first. This can be done by calculation, or by approximation based on "best guess" or "worst case scenario", or via the interrogation of the client. The burden profile can then be determined. The acceleration, deceleration and steady state must be considered separately.

For the steady state portion of the velocity profile, the hydraulic force is numerically equal to the negative of the burden: $\Delta(PA) = -F_b$

The minus sign here physically means that this hydraulic force is against the burden, i.e. applied in the direction of the trolley motion.

During the acceleration of the system, the hydraulic force must both overcome the burden and satisfy the constraints placed on the acceleration. The magnitude of the hydraulic force $\Delta(PA)$ is calculated from:

$$\Delta(PA) = - F_b + M \cdot A_{limit} \qquad (12.16)$$

where

F_b — burden force (negative in this case).

A_{limit} — specified acceleration limit.

This is nothing more than Newton's law!!!!

It is obvious that the hydraulic force for the acceleration part of the velocity profile in this case is always positive ($\Delta(PA) > 0$) because the burden is negative and acceleration positive regardless of the variations in burden, mass and acceleration constraints. Variations in burden and/or mass only affect the magnitude of the hydraulic force. Consequently, consideration of using an acceleration device in the circuit design must be given.. The selection of an appropriate acceleration device, however, does not affect the sign of $\Delta(PA)$ which is the critical parameter for the design criteria.

The following discussions will be concentrated on establishing the hydraulic force profile for the deceleration parts of the velocity profile.

12.4.1.1 Determination of The Hydraulic Force for Deceleration

In determining the hydraulic force required to maintain the desired deceleration, the natural deceleration of the system is used to define the magnitude of the hydraulic force and the direction at which the hydraulic force is applied to the physical system. In doing so, the requirement of the system deceleration must be obtained either as a control or as a limit on the velocity profile, and then the system natural deceleration must be estimated. For constant burden and mass, the natural deceleration is a single value, while for a variable burden and/or mass, a range of natural deceleration exists. Consequently, the determination of hydraulic force for deceleration involves variations in the natural system decelerations and in the specified system decelerations. It will be shown that, as the results from the combination of the natural deceleration and the client defined deceleration, it is possible for the hydraulic force necessary to produce required deceleration motion to be positive ($\Delta(PA) > 0$) or negative ($\Delta(PA) < 0$) or zero ($\Delta(PA) = 0$).

To assist in interpreting the various cases of deceleration, a pictorial legend on the velocity profile is adopted.

For visualization, the graphical legends in all following figures in this section are specified such that **the dotted areas (or shaded areas in the pdf form of the notes) represent the range of natural deceleration and the lined areas indicate the region of client required deceleration**. If deceleration constraints are imposed either as a maximum limit or as a minimum limit, a line with a note such as $D_{l.max}$ or $D_{l.min}$ on velocity profiles is also used to identify that limit.

Suppose a system has a natural deceleration range as shown in Figure 12.20. The maximum and minimum natural decelerations, $D_{n.max}$ and $D_{n.min}$, are calculated based on given burden and mass information.

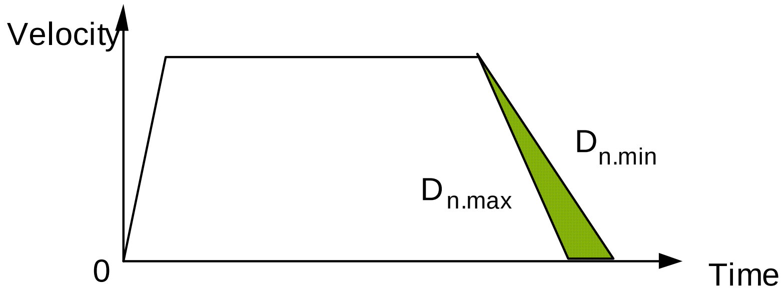


Figure 12.20 Graphical Representation of **Natural** Deceleration

The **client defined deceleration** requirement is graphically illustrated in Figure 12.21. The minimum and maximum deceleration are specified. In fact, this, by definition, is a velocity control requirement. Any deceleration values which fall (forced or naturally) into the lined region are considered to be acceptable. By superimposing the natural deceleration range (dotted area in Figure 12.20) onto the specified deceleration region (lined area in Figure 12.21, and then observing the overlap of the two, it is possible to draw conclusions pertaining to whether the natural deceleration, by itself, meets the client defined deceleration constraints. If the natural deceleration range falls completely within the acceptable deceleration region as illustrated in Figure 12.22, then no hydraulic force is required to bring the system to rest. If the natural deceleration range partially overlaps or falls outside the acceptable deceleration region, then a hydraulic force must be applied to the physical system to "force" the system deceleration completely into the acceptable region.

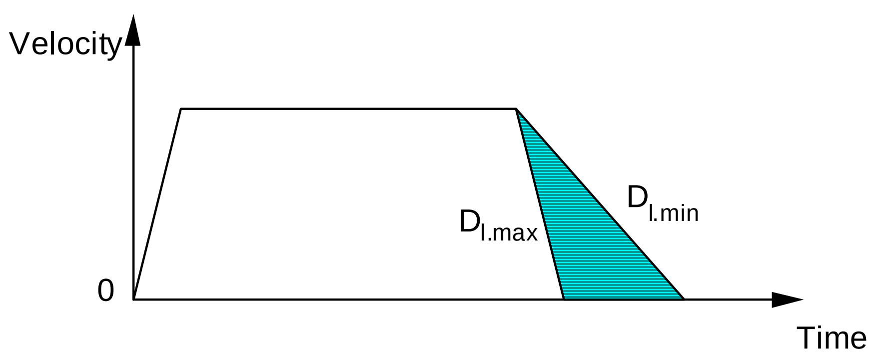
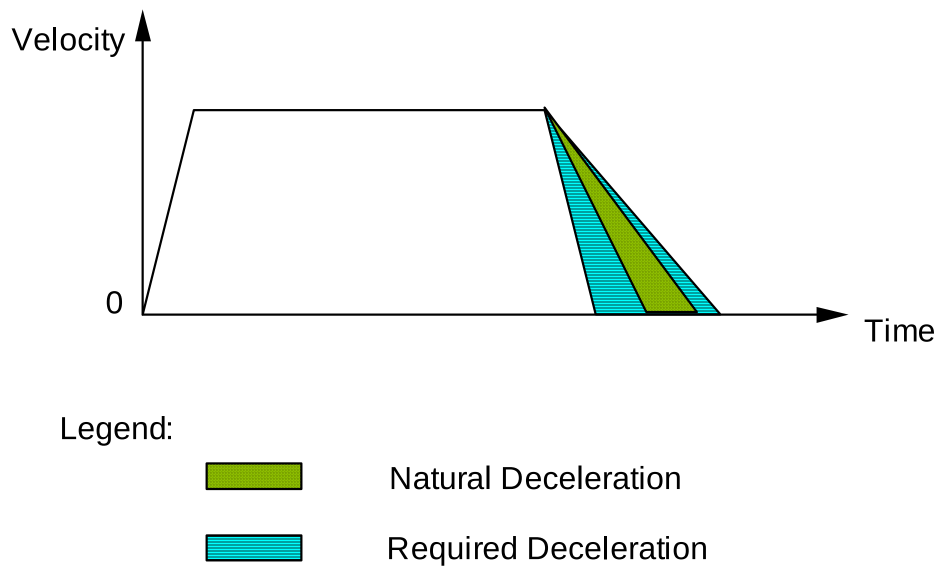
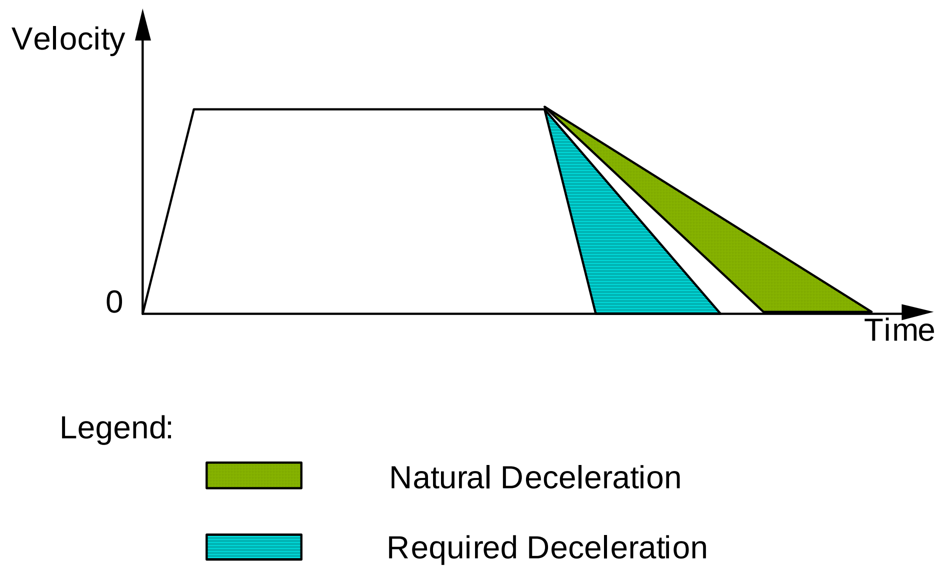


Figure 12.21 Velocity Profile with Specified Deceleration Limits



No need to force the system to stop as it can decelerate on its own and meet the system requirements.

Figure 12.22 (a) Superimposition of Natural Deceleration on the Deceleration Part of Velocity Profile.



We need to force the system to stop because it cannot decelerate on its own and meet the system requirements.

Figure 12.22 (b) Superimposition of Natural Deceleration on the Deceleration Part of Velocity Profile.

NOTE: Shading considerations: The following will be adopted for showing acceptable regions of acceleration or deceleration.

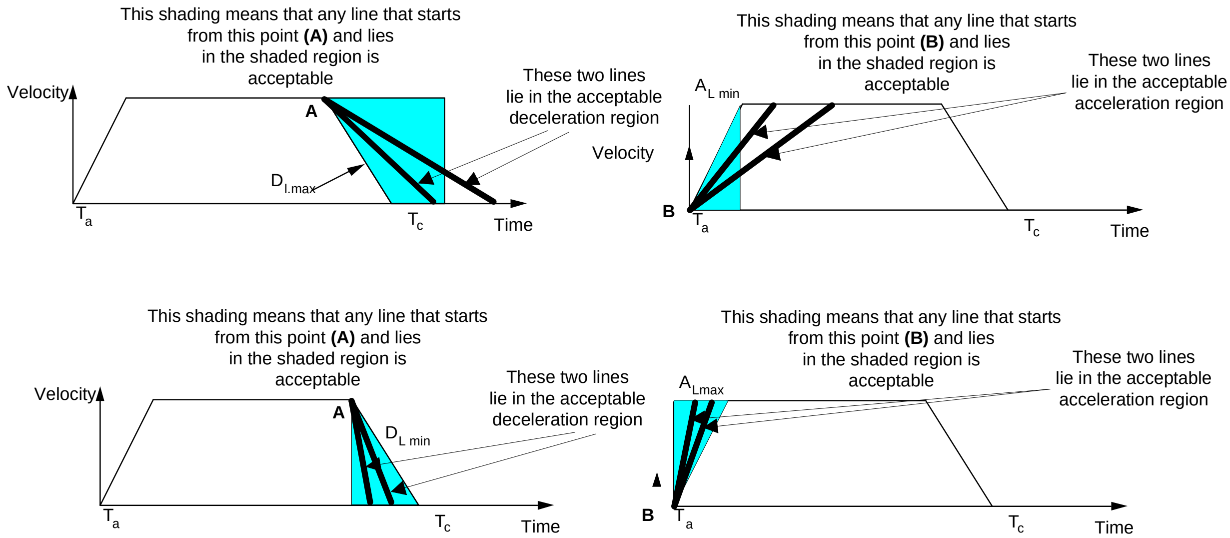


Figure 12.22(c) Shading convention

As an example, consider again the system in Figure 12.18. The trolley is carrying open containers full of liquid from position A to C. To prevent the liquid from spilling out, this trolley system must be accelerated and decelerated gently; the imposed limits on the system deceleration and acceleration are shown on its velocity profile in Figure 12.23(a). The mass of this system is expected to vary with duty cycles. The mass profile and burden profile reflecting expected variations are given in Figure 12.23(b) and Figure 12.23(c). It is assumed that the burden terms are independent of the mass in this example. The following equation reflects a mass dependency on the burden but this is so the relationship can be considered general. The range of the natural deceleration, in this example, is given by:

$$D_{n.max} = \frac{-F_{b.max}|_{M_{min}}}{M_{min}} \qquad \text{In this example, } M_{max} = M_{min} \qquad (12.17)$$

$$D_{n.min} = \frac{-F_{b.min}|_{M_{max}}}{M_{max}} \qquad (12.18)$$

After the superimposition of the natural deceleration (estimated from equations (12.17) and (12.18)) onto the deceleration part of the velocity profile in Figure 12.23(a), two possible situations arise as illustrated in Figure 12.24. (Actually, a third one when a partial overlap exists; however it is essentially the same as Figure 12.24(a); hence it is included in that case.)

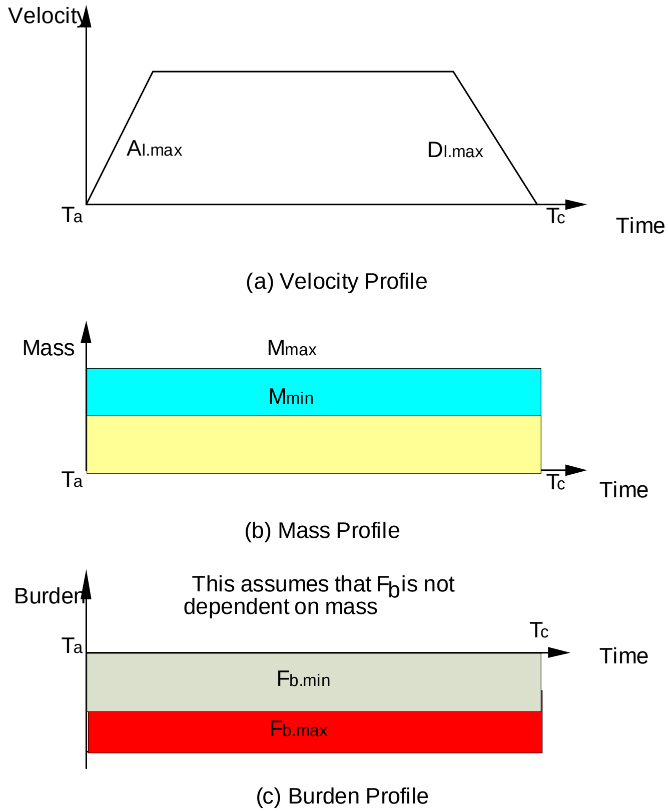
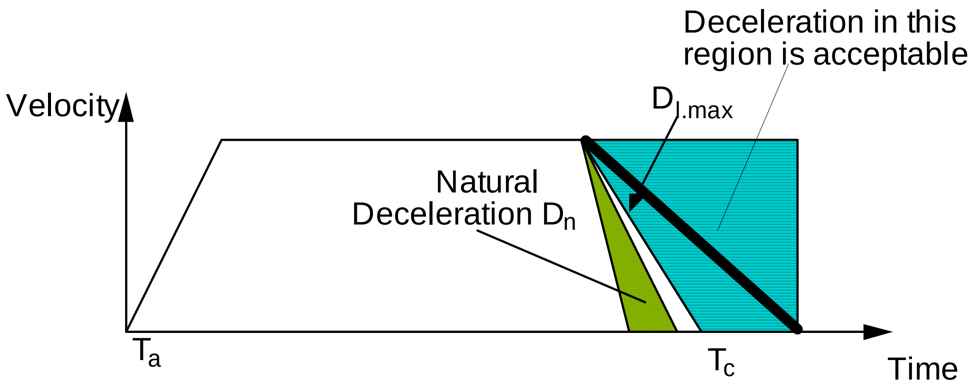
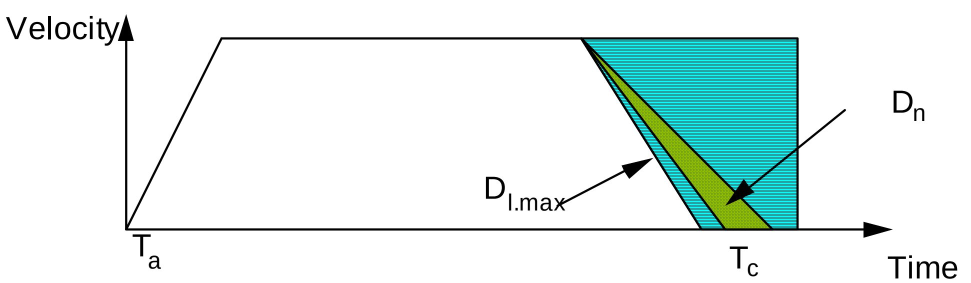


Figure 12.23 Velocity Profile and Expected Mass and Burden Profiles for Trolley System



(a) Natural Deceleration Is Too Fast



(b) Natural Deceleration Is Acceptable

Figure 12.24 Superposition of Natural Deceleration on Velocity Profile

In Figure 12.24(a), all possible values of the natural deceleration D_n are outside of the region of the specified deceleration. The system deceleration in the absence of any hydraulic forces is considered to be too fast. Therefore, a hydraulic force must be provided in the direction of the system motion ($\Delta(PA) > 0$) during deceleration in order to force the system to follow the specified deceleration. Mathematically, the magnitude of the hydraulic force necessary to force the system into the acceptable deceleration region can be calculated by:

$$\Delta(PA)_{min} = - F_{b.max}|_{M_{min}} - M_{min} \cdot D_{l.max} \qquad (12.19)$$

where

$F_{b.max}$ — Maximum burden of the system,
M_{min} — Minimum mass of the system,
$D_{l.max}$ — Maximum deceleration imposed by the client.

At this point, you may well ask how did we decide on whether each term was maximum or minimum? Lets us consider this in more detail.

From Equation (12.19) we can write

$$D_{L\max} = -\frac{\Delta(PA) + F_b\big|_M}{M} \qquad 12.19(a)$$

First we know that Δ(PA) is some positive valve. We also know that $F_b\big|_M$ is negative and that Δ(PA) < $F_b\big|_M$ for D_{Lmax} to be positive. D_{Lmax} is also maximum and hence can only get smaller.

If M increases, the D_{Lmax} gets smaller which is right on. So we can write equation 12.19(a) as:

$$D_{L\max} = -\frac{\Delta(PA) + F_b\big|_{M_{\min}}}{M_{\min}}$$

Now let us let Δ(PA) **increase**. Since $F_b\big|_M$ is negative and Δ(PA) < $F_b\big|_M$ and D_{Lmax} is positive then:

-(Δ(PA) + $F_b\big|_M$) becomes less positive (-(5 - 10) = 5 vs. -(7-10) = 3 for example) which what is we want since D_{Lmax} is a maximum and can only decrease. Therefore, Δ(PA) is a minimum and so we can write Equation 12.19(a) as

$$D_{L\max} = \frac{\Delta(PA)\big|_{\min} + F_b\big|_{M_{\min}}}{M_{\min}} \qquad 12.19(b)$$

We are not quite finished. What about $F_b\big|_{M_{\min}}$? Is this a maximum or minimum value. Let us allow $F_b\big|_{M_{\min}}$ to increase holding $\Delta(PA)\big|_{\min}$ constant . Since $F_b\big|_{M_{\min}}$ is negative, and $\Delta(PA)\big|_{\max}$ is a positive value, then -($\Delta(PA)\big|_{\max} + F_b\big|_{M_{\min}}$) gets larger (-(5-10) = 5 versus -(5-12) =7 for example). Ah hah! This is the wrong way since D_{Lmax} is a maximum value. Thus $F_b\big|_{M_{\min}}$ is a maximum value (we can only decrease it) and can be written as $F_{b\max}\big|_{M_{\min}}$ and finally we can rewrite equation 12.19(b) as

$$D_{L\max} = -\frac{\Delta(PA)\big|_{\min} + F_{b\max}\big|_{M_{\min}}}{M_{\min}} \qquad 12.19(c)$$

And finally rewriting this in its original form (Equation (12.19))

$$\Delta(PA)\big|_{\min} = - F_{b.max}\big|_{M_{min}} - M_{min} \cdot D_{l.max}$$

The hydraulic force defined in Equation (12.19 and 19(c)) is positive, and is the minimum value allowed. That is, the magnitude of hydraulic force supplied to the physical system must be equal to or higher than this value, because any hydraulic forces higher than this value will result in a deceleration rate lower than $D_{l.max}$. Once the hydraulic pressure has been set according to the value of $\Delta(PA)$ in Equation (12.19), a reduction in the burden and/or increase in the mass will lead to a lower system deceleration. **(It must be noted that a maximum limit on hydraulic force exists beyond which the system would start to accelerate.)**

In Figure 12.24(b), graphically, the “dotted” or green area is entirely encompassed by the “lined” or blue area. This means that in the natural state the external forces on the system are sufficient to bring the system to rest at the acceptable deceleration rate, i.e. the system can coast to a stop. There is no need for the hydraulic force in deceleration, and therefore, $\Delta(PA) = 0$.

It should be pointed out that the temporary absence of hydraulic force during the motion represents a special situation which often occurs when using a flow valve in a meter-in configuration to adjust flow rate, as schematically shown in Figure 12.25. A sudden reduction in flow rate (a rapid closure of the valve to reduce the orifice area, hence flow rate) may result in insufficient fluid in the chamber between the valve and the actuator. As a consequence, the hydraulic pressure in this chamber decreases drastically. The worst scenario is that this pressure drops to zero and the trolley system is moving in a natural state (assuming that the pressure in return line to tank is approximately zero). From a controllability point of view, the acceptance of the natural deceleration means that the system motion is still under control and acceptable even in the worst case. In other words when the natural deceleration is concluded to be acceptable, a meter-in configuration is able to give satisfactory controllability. If cavitation upstream to the actuator is present, a hydraulic check valve can be used to remedy the situation. This does not change the fact that a meter-in circuit can be used.

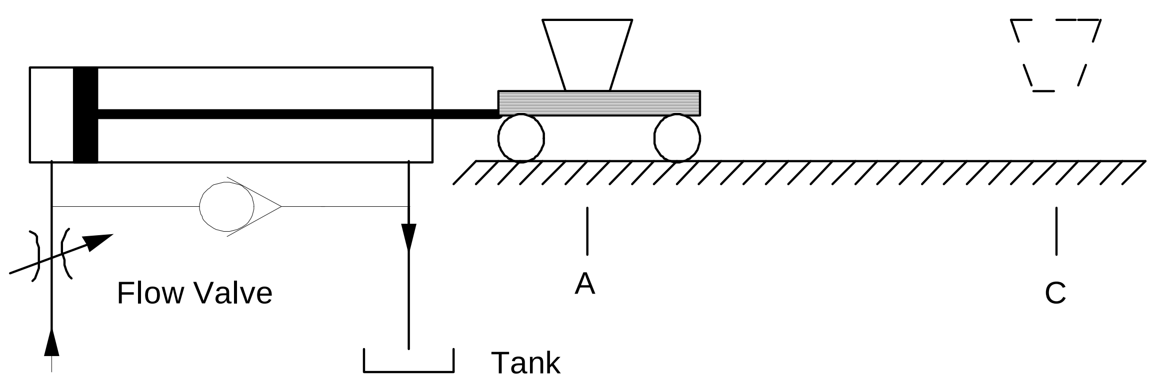


Figure 12.25 A Hydraulic Actuator and Trolley System

Consider now the situation in which the client defined deceleration is imposed as a minimum limit $D_{l.min}$. If the natural deceleration D_n is as indicated in Figure 12.26, then D_n is too small; therefore the system needs to be stopped in a shorter period of time.

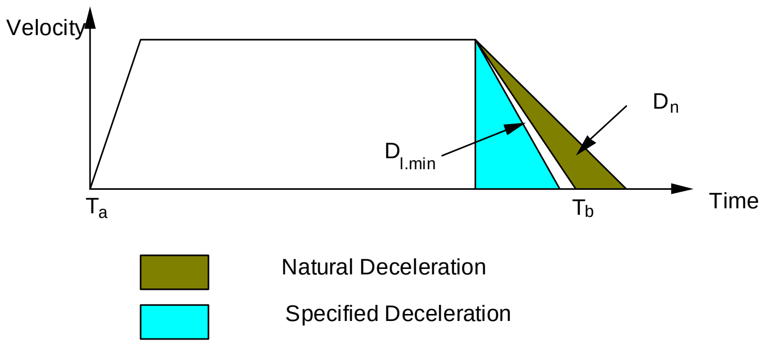


Figure 12.26 Natural Deceleration Is Too Slow

The hydraulic force which is necessary to meet the deceleration constraint is calculated by:

$$\Delta(PA)\,|_{min} = - F_{b.min}|_{M_{max}} - M_{max} \cdot D_{l.min} \qquad (12.20)$$

where

$F_{b.min}|_{M_{max}}$ — Minimum burden of the system under max mass conditions,

M_{max} — Maximum mass of the system,

$D_{l.min}$ — Minimum deceleration limit proposed by the client.

Let us do the same as we did for equation (12.19).

From Equation (12.20) we can write

$$D_{L\,\min} = -\frac{\Delta(PA) + F_b|_M}{M} \qquad 12.20(a)$$

First we know that Δ(PA) is some negative value (we have to slow the system down). We also know that $F_b|_M$ is also negative. D_{Lmin} is also minimum and hence can only get larger.

If M increases, then D_{Lmin} gets smaller which is wrong. Thus M can only decrease which means that M was a maximum. So we can write equation 12.19(a) as:

$$D_{L\,\min} = -\frac{\Delta(PA) + F_b|_{M_{\max}}}{M_{\max}}$$

Now let us let Δ(PA) **increase with** $F_b|_M$ constant. But -(Δ(PA) + $F_b|_M$) becomes more positive (-(-5 - 10)) = 15 vs. -(-7-10) = 17 for example) which is what we want since

D_{Lmin} is a minimum and can only increase. Therefore, Δ(PA) is a minimum and so we can write Equation 12.20(a) as

$$D_{L\,\min} = \frac{\Delta(PA)\big|_{\min} + F_b\big|_{M_{\max}}}{M_{\max}} \qquad 12.20(b)$$

Since $F_b\big|_M$ is also negative, the same logic applies as for Δ(PA) so we can readily deduce that $F_b\big|_M$ is a minimum. Finally we can rewrite equation 12.20(b) as

$$D_{L\,\min} = \frac{\Delta(PA)\big|_{\min} + F_{b\min}\big|_{M_{\max}}}{M_{\max}} \qquad 12.20(c)$$

And finally rewriting this in its original form (Equation (12.20))

$$\Delta(PA)\,|_{min} = -\,F_{b.min}|_{M_{max}} - M_{max} \cdot D_{l.min} \qquad (12.20)$$

No, we are not going to do all cases here. Two are enough to show our approach. We have done this for all cases and these are summarized in the following Tables for both the linear and the angular cases.

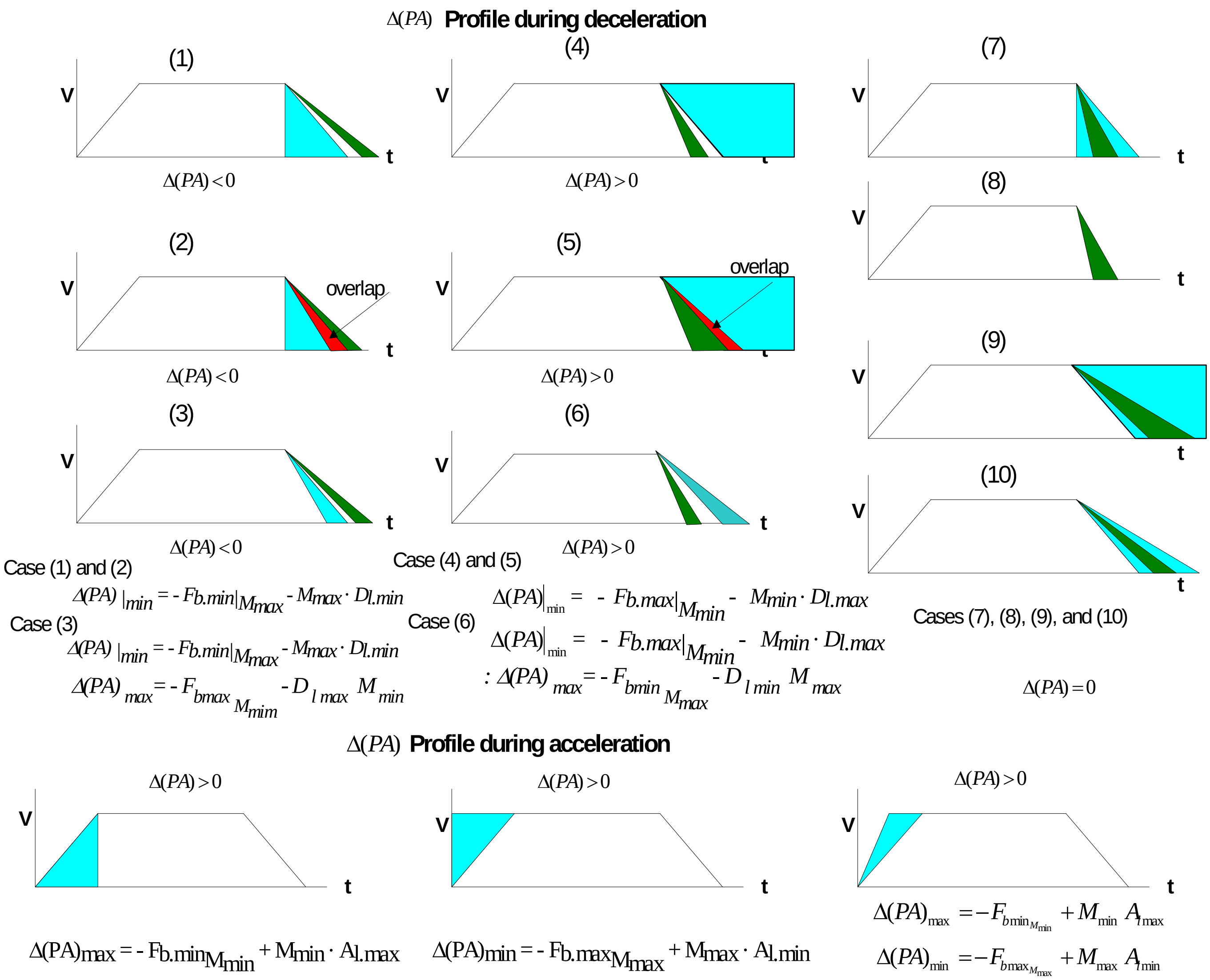
Before going too far, we should consider the whole velocity profile. If a system has a natural deceleration (negative burden), then we must force the system to accelerate. What values do we use there? Well we use exactly the same logic as above. We have therefore subsequently included these equations in our Table as well. We will use these Tables to develop our hydraulic and torque profiles.

Two situations have been excluded: (1) velocity control or limit is not specified and hence flow modulation using conventional flow valves is not necessary, and (2) precise control or limit of the velocity to ± 2% or less is required resulting in the need for a servo control system (normally with system feedback) to be used. The exclusion of these two situations is accommodated in the flow chart by employing a "Front-end" as illustrated in Figure 12.27.
In the following Tables, the color (shaded) regions are defined as follows.

Acceptable linear or angular acceleration or deceleration range

Natural linear or angular acceleration of deceleration

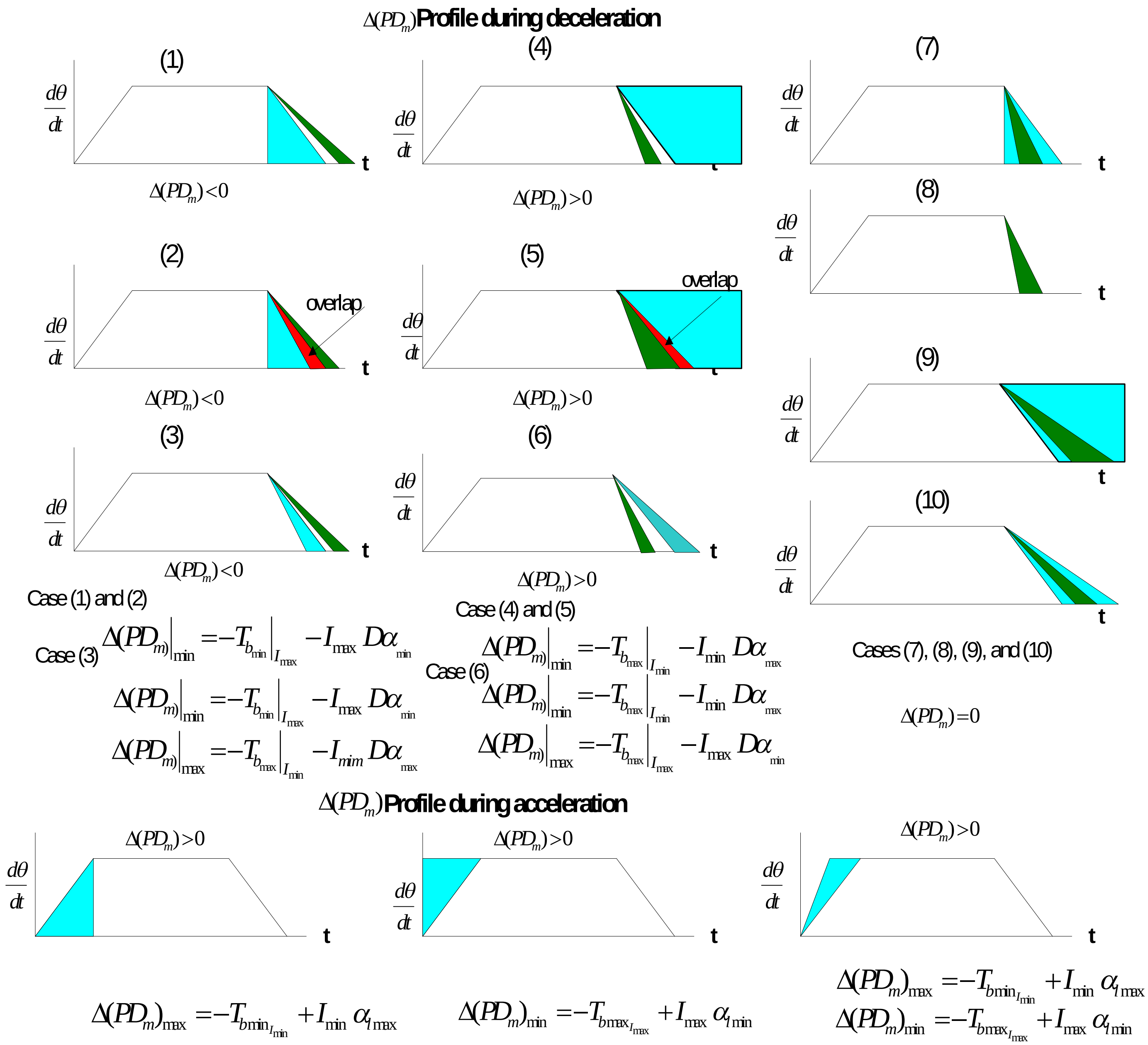
Table 12.1 Charts for hydraulic force profile with a natural deceleration



Notes: These charts are only valid if the MA term is greater than the $\Delta(PA)$ term at steady state (ie the burden)

At steady state, $\Delta(PA)$ = - (burden)

Table 12.2 Charts for hydraulic torque profile with a natural angular deceleration

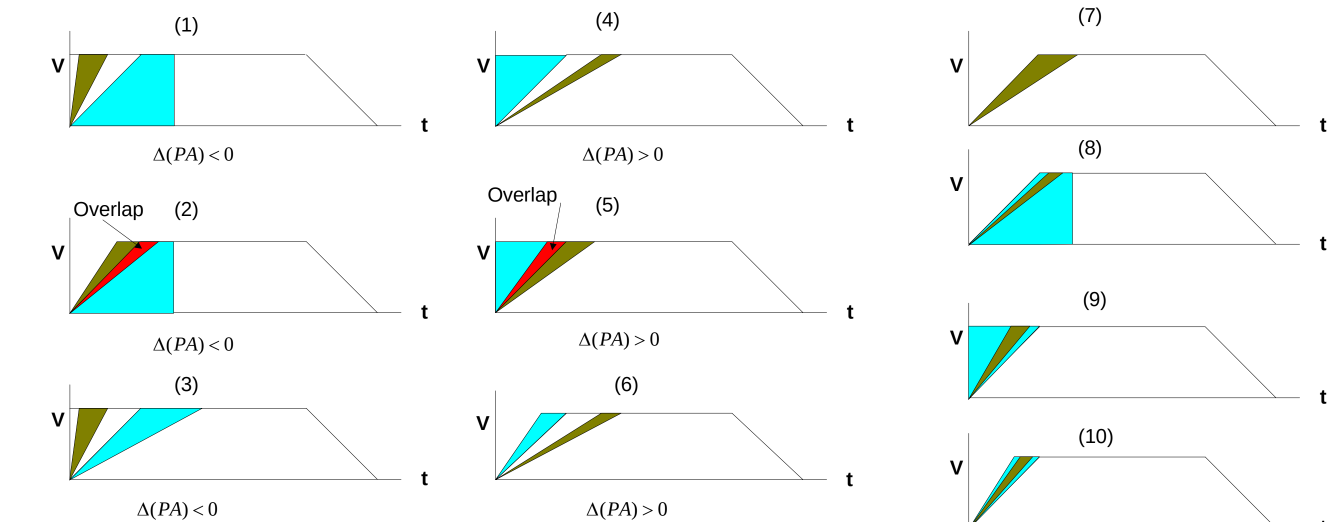


Notes: These charts are only valid if the ID$_\alpha$ term is greater than the $\Delta(PD_m)$ term at steady state (ie the burden)

At steady state, $\Delta(PD_m)$ = - (burden)

Table 12.3 Charts for hydraulic force profile with a natural acceleration

Δ(PA) Profile during acceleration



Case (1) and (2)

$$\Delta(PA)_{min} = -F_{b\,max_{M_{min}}} + M_{min}\, A_{l\,max}$$

Case (3)

$$\Delta(PA)_{min} = -F_{b\,max_{M_{min}}} + M_{min}\, A_{l\,max}$$

$$\Delta(PA)_{max} = -F_{b\,min_{M_{max}}} + M_{max}\, A_{l\,min}$$

Case (4) and (5)

$$\Delta(PA)_{min} = -F_{b\,min_{M_{max}}} + M_{max}\, A_{l\,min}$$

Case (6)

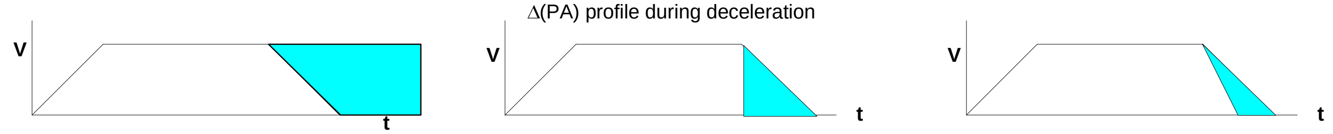
$$\Delta(PA)_{min} = -F_{b\,min_{M_{max}}} + M_{max}\, A_{l\,min}$$

$$\Delta(PA)_{max} = -F_{b\,min_{M_{min}}} + M_{min}\, A_{l\,max}$$

Case (7), (8), (9) and (10)

$$\Delta(PA) = 0$$

Δ(PA) profile during deceleration



$$\Delta(PA)_{max} = -F_{b\,min_{M_{min}}} - M_{min}\, D_{l\,max}$$

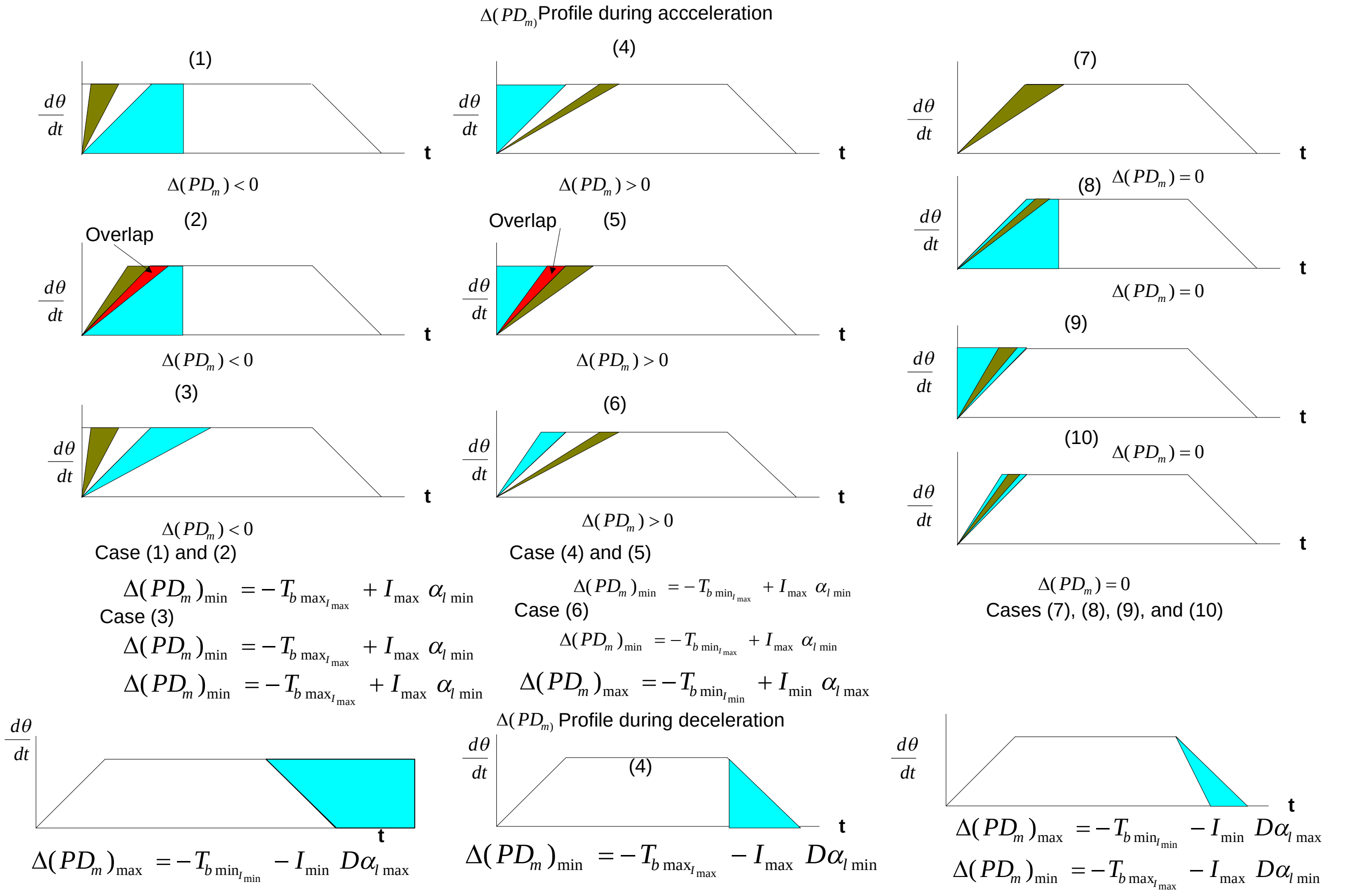
$$\Delta(PA)_{min} = -F_{b\,max_{M_{max}}} - M_{max}\, D_{l\,min}$$

$$\Delta(PA)_{max} = -F_{b\,min_{M_{min}}} - M_{min}\, D_{l\,max}$$

$$\Delta(PA)_{min} = -F_{b\,max_{M_{max}}} - M_{max}\, D_{l\,min}$$

At steady state, $\Delta(PA)$ = - (burden)

Table 12.4 Charts for hydraulic torque profile with a natural angular acceleration



At steady state, $\Delta(PD_m)$ **= - (burden)**

This flow chart should be used as a guide to design using the Tables.

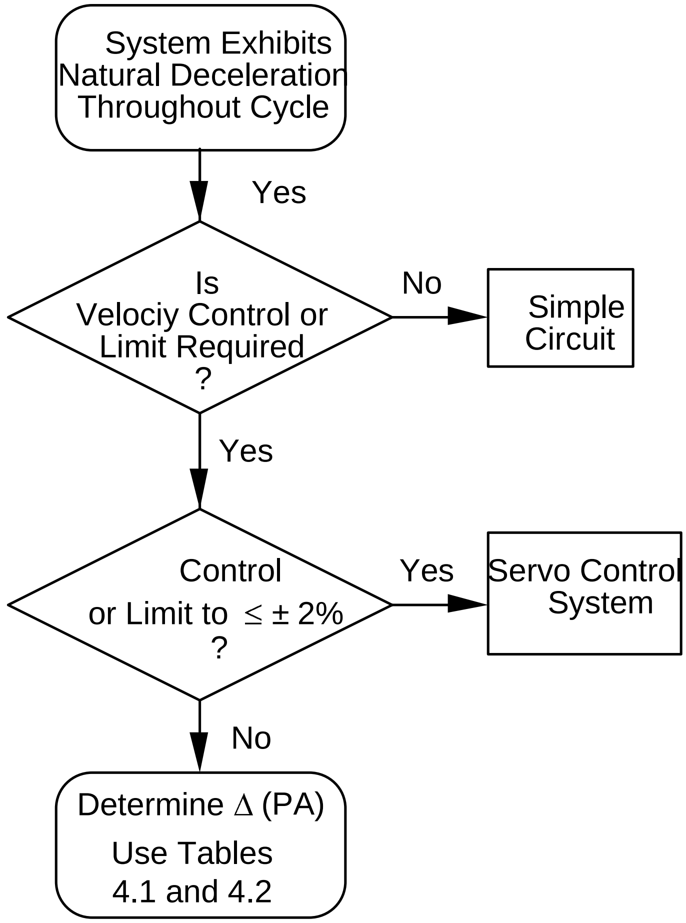


Figure 12.27 "Front-End' of the Flow Chart

12.4.1.2 Limitations on using the Charts during acceleration

As mentioned in the last section, the value of natural deceleration is used to determine the nature of the physical system (resistive, overcenter etc.) and will eventually be used to determine the placement of the flow modulation device (if applicable). However, the acceleration part of the cycle can not be neglected. Our Tables have included the acceleration portion but because there are a few problems with stiction, we will have to address these concerns separately.

It must be pointed out that in most circumstances, the maximum allowable hydraulic force defined in our Tables which satisfies the acceleration constraint is **larger than or equal to the burden at steady state**. It is conceivable, however, that the magnitudes of $F_{b\min}\big|_{M_{\min}}$ *or* $T_{b\min}\big|_{I_{\min}}$ and/or M_{min}, I_{min} could be sufficiently small such that $-F_{b\min}\big|_{M_{\min}} + M_{\min}\ A_{l\max}$ *or* $-T_{b\min}\big|_{I_{\min}} + I_{\min}\ \alpha_{l\max}$ in our Tables is smaller than -

$F_{b.max}$ or T_{bmax} at steady state. Under these conditions, choosing Δ(PA), Δ(PD_m) based on these equations would "stall" the system at steady state conditions at a large burden $-F_{b.max}$, $-T_{b.max}$. On the other hand, if Δ(PA), Δ(PD_m) is calculated on the basis of $-F_{b.max}$, $-T_{b.max}$ only, then the acceleration constraint for a small burden would be violated.

Another situation **which limits the "universal"** application of the acceleration equations is the presence of stiction in the system as illustrated in Figure 12.28. Please note that the discussion is equally valid for angular acceleration and torques so only the linear case is considered.

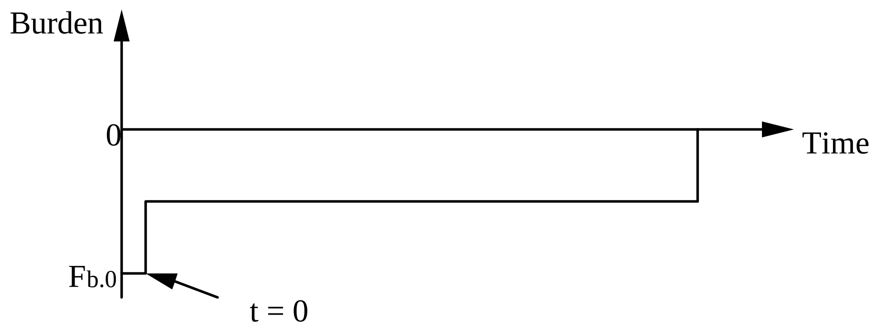


Figure 12.28 Burden Profile with Stiction Dominant

In this situation, as the system begins to move from its static state, the stiction F_{b0} is higher than the burden in steady state, and must be first overcome (indeed this is the usual case in practice). If the hydraulic force for limiting the acceleration is less than F_{b0}, then the system will not start to move. Hence *Δ(PA)* must be increased sufficiently to overcome stiction in order to "kick" the system. Because stiction only occurs at the instant t = 0, the burden value is that of the steady state value right after the system starts moving. Since the increased *Δ(PA)* to overcome stiction may be larger than *Δ(PA)* to meet acceleration constraints, the system under the higher *Δ(PA)* may accelerate in a "jerky" fashion. Thus the acceleration constraints will not be satisfied.

For these situations some means must be provided both to sustain the imposed acceleration limits and to meet the maximum burden. For the two cited cases, the system pressure of the hydraulic circuit must be set to overcome the maximum burden $F_{b.max}$ or $F_{b.0}$ (if it is stiction) and an acceleration device must be installed in the circuit to prevent the system acceleration from exceeding the imposed acceleration limit $A_{l.max}$. If the time and/or the positions where variations in burden and/or mass may occur are unpredictable and the accelerations so induced are expected to be higher than the value permitted, a servo valve with system feedback may be necessary.

In summary, if *Δ(PA), or Δ(PD_m)*, as calculated from the equations given in our Tables is sufficient to **satisfy** both dynamic (acceleration) and steady state (including stiction) constraints, the steady state and deceleration stage of the cycle can be examined without having to carry over any "configuration" constraints to the analysis. If $\Delta(PA)$, or $\Delta(PD_m)$, does not satisfy dynamic and steady state criteria, a constraint is placed on the design process and must be considered during the steady state and deceleration analysis of the cycle.

Note: if the mass or the burden changes during the cycle and the system requires that the system accelerates to a higher velocity, then we must recalculate $\Delta(PA)$, or $\Delta(PD_m)$, at **each point of the cycle** where the velocity increases.

We should note that in all tables we see a condition in which upper and lower limits have been specified.If the burden and mass remain constant during the cycle, **the control of the system acceleration may be achieved simply by the use of hydraulic system pressure which is set according to the values between $\Delta(PA)_{max}$ and $\Delta(PA)_{min}$ ($\Delta(PD_m)_{max}$ and $\Delta(PD_m)_{min}$**). With significant variations of burden and/or mass, however, this method may not be effective because maintaining the system deceleration within an acceptable region is extremely hard to achieve simply by setting a fixed hydraulic system pressure. An acceleration device with or without internal feedback may be used if the moments at which the significant variations of the burden and or mass occur can be predetermined; otherwise, a servo control system (involving internal and external feedback) must be installed.

In summary, the determination of the hydraulic force for acceleration involves the examination of variations in burden, mass and acceleration constraints. A conclusion of "$\Delta(PA) > 0$", "$\Delta(PD_m) > 0$" for acceleration can be applied to all situations where the physical systems are in a "resistive" mode. It is an objective to maintain the simplest hydraulic circuit which accomplishes the task for given constraints. (Generally, this means using hydraulic system pressure to achieve this goal.) Under some situations if the use of hydraulic system pressure cannot maintain the acceleration limits within the acceptable range due to significant variations in burden and/or mass, then an **acceleration device** may be necessary. The use of such a device, however, does not affect the conclusion of "$\Delta(PA) > 0$", "$\Delta(PD_m) > 0$".

12.4.2 Example on using the Charts.

Consider the following case. This example was done earlier so we shall just show the profiles.

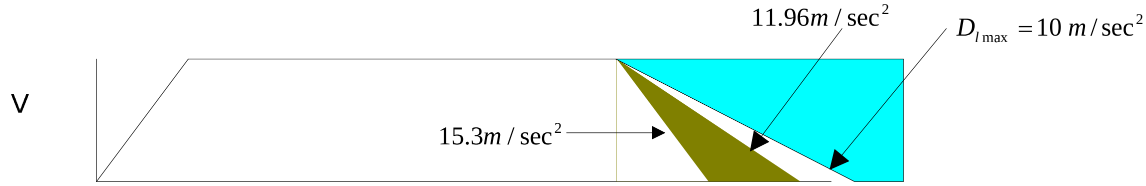


Figure 12.30 Velocity profile and deceleration constraints.

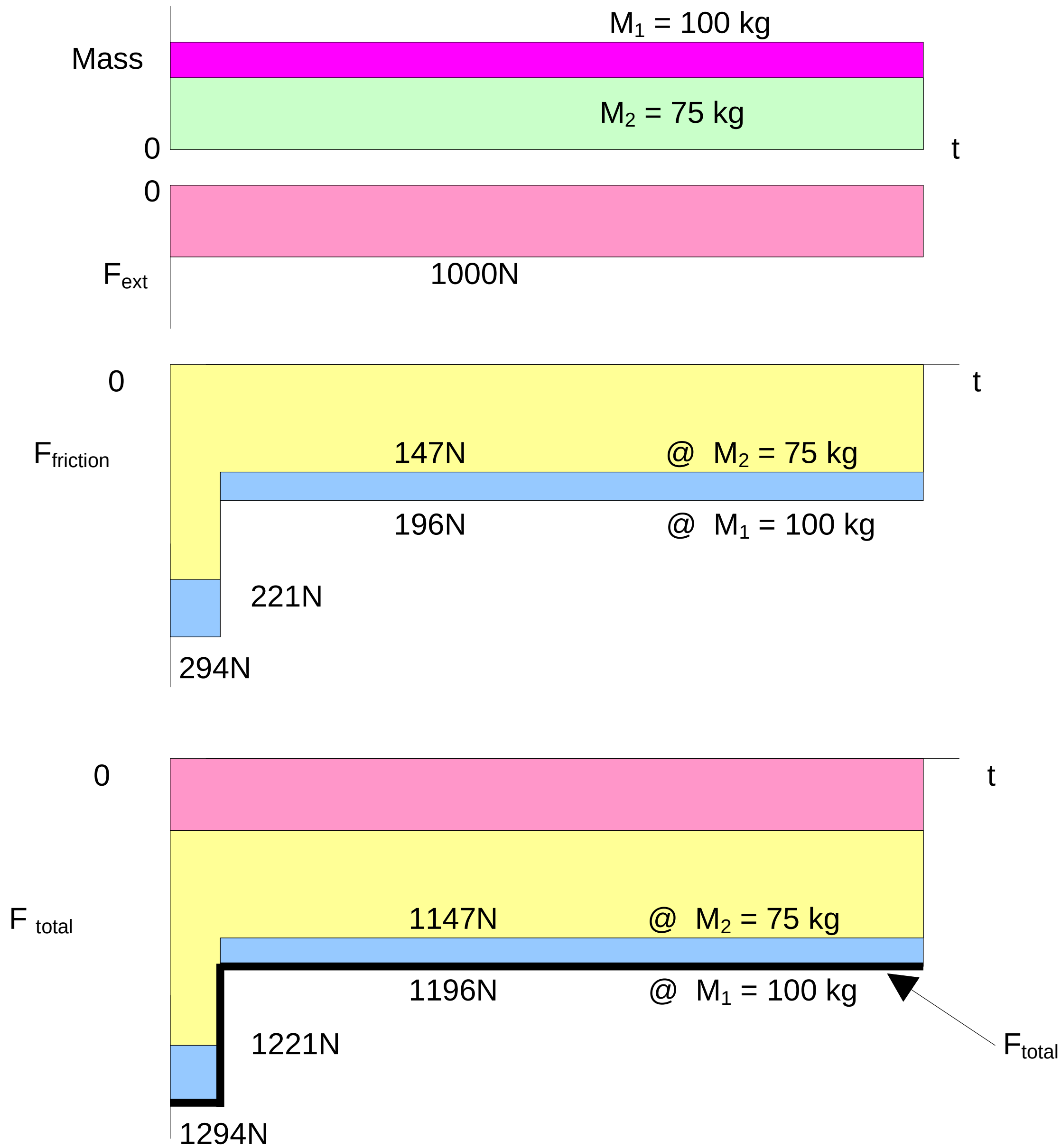


Figure 12.31 Burden profile for example.

$$D_{n1} = -\frac{-1147N}{75kg} = 15.3\frac{m}{\sec^2}$$

$$D_{n2} = -\frac{-1196N}{100kg} = 11.96\frac{m}{\sec^2}$$

We are given two limits on the natural deceleration and the upper the limit on the required deceleration as posed by the client or situation. $D_{l\max} = \cdots 10m/\sec^2$. What is the hydraulic profile for the steady state and deceleration part of the cycle. (Note: we have not specified limits on acceleration yet so we cannot do that part).

The natural deceleration is only used as a key to determine the correct chart and equation to use. That is all. It is only a key.

First look for the chart that reflects this situation. This would be Table 12.1 and case (4). From which we get that:

$$\Delta(PA)\big|_{\min} = -F_{b.max}\big|_{M_{min}} - M_{min}\cdot D_{l.max}$$

$$\Delta(PA)\big|_{\min} = (-(-1147) - 75*(10))\text{N} \quad \text{(see Figure 12.31(b) for values)}$$

$$= 397\text{N}$$

$$\Delta(PA)_{steadystate} = -F_{burden}$$

$$= 1147\text{N or } 1196\text{N, depending on the mass.}$$

The profile would appear as:

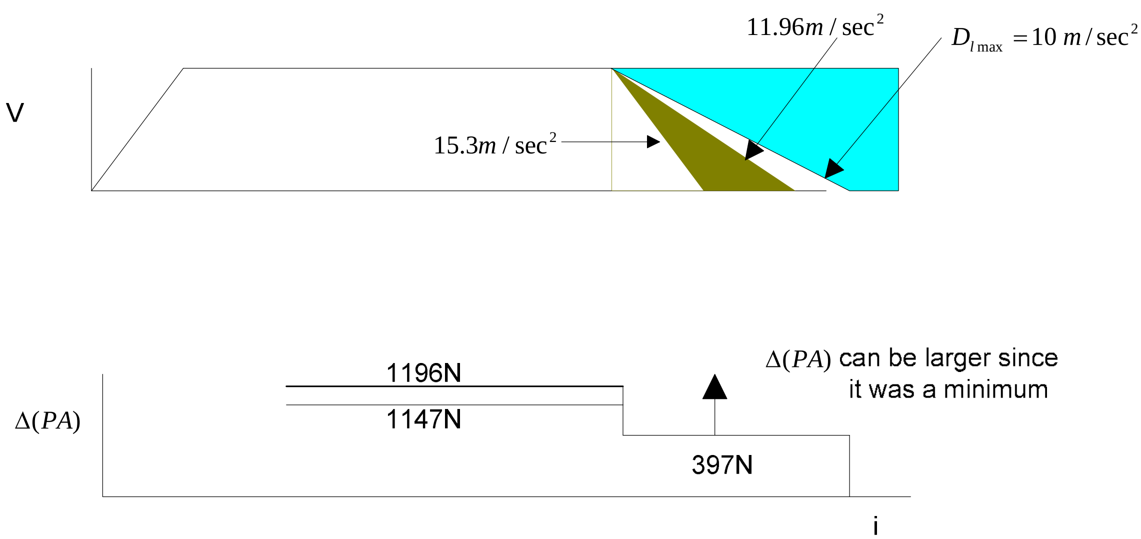


Figure 12.32 Hydraulic force profile

Now consider the same burden but a different client requirement on the deceleration.

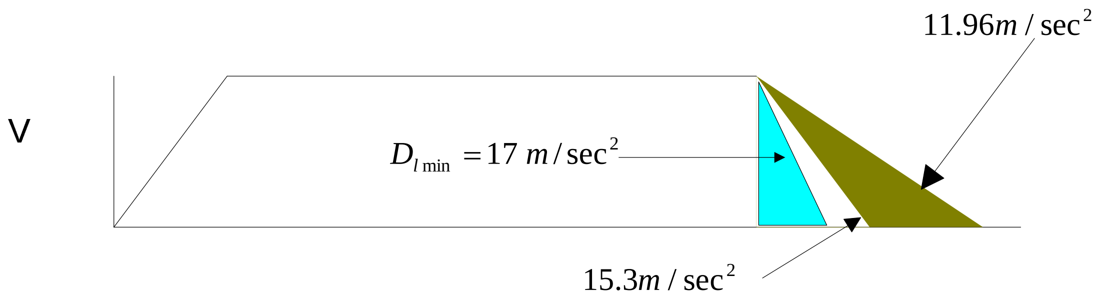


Figure 12.33 Velocity profile and clients requirements.

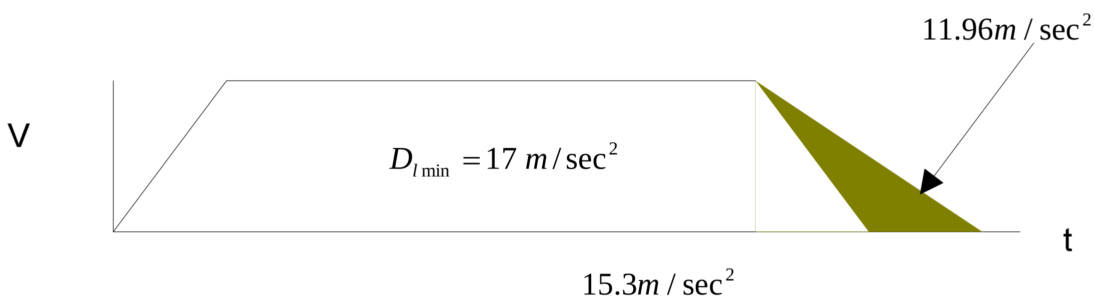
As before we use the natural deceleration as only a key to find the right chart. WE shall use chart 12.1 again and thus

$$\Delta(PA)\,|_{min} = - F_{b.min}|_{M_{max}} - M_{max} \cdot D_{l.min}$$

$$= (- (1196) - 17*100\ )\ \text{(see Figure 12.29(b) for values)}$$

$$= - 504\ N$$

The hydraulic force profile would thus be:



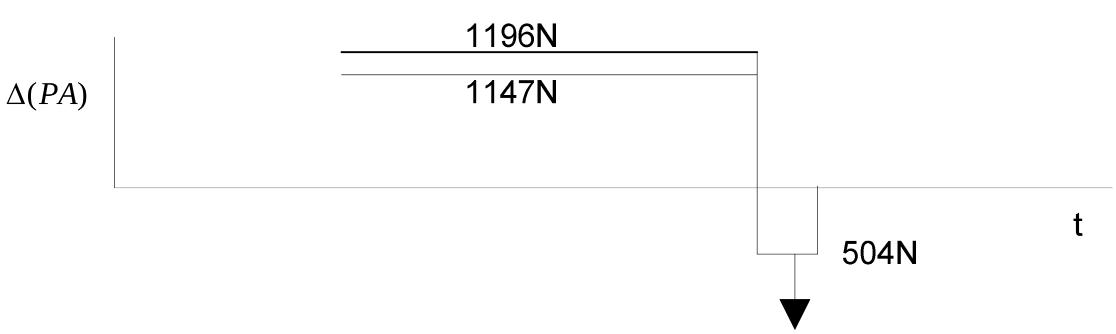


Figure 12.34 Hydraulic force profile.

Let us now look at the same problem but with acceleration limits imposed.

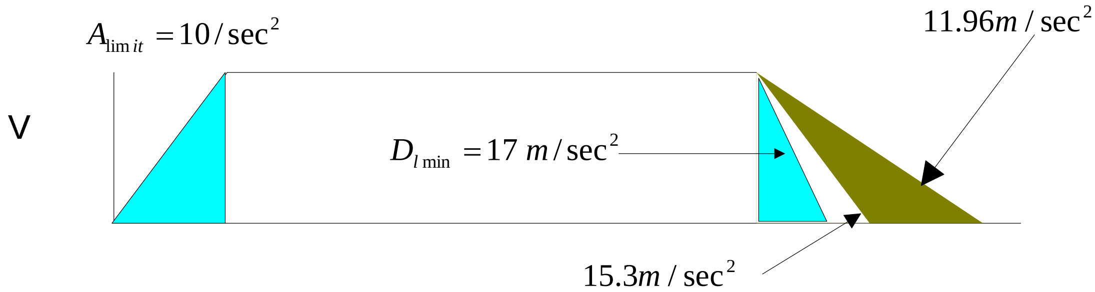


Figure 12.35 Velocity profile for example with acceleration limits.

Remember. The natural deceleration was used as a key to get the right chart and that is all. We saw that Table 12.1 was the one to use. Since we are now interested in only the acceleration part, we can look for acceleration at the lower cases and find the correct shaded one. This gives us:

$$\Delta(PA)_{max} = -F_{b.min} M_{min} + M_{min} \cdot A_{l.max}$$

$$= -(-1147) + 75 * 10$$

$$= 1897N$$

Our complete profile would now look like:

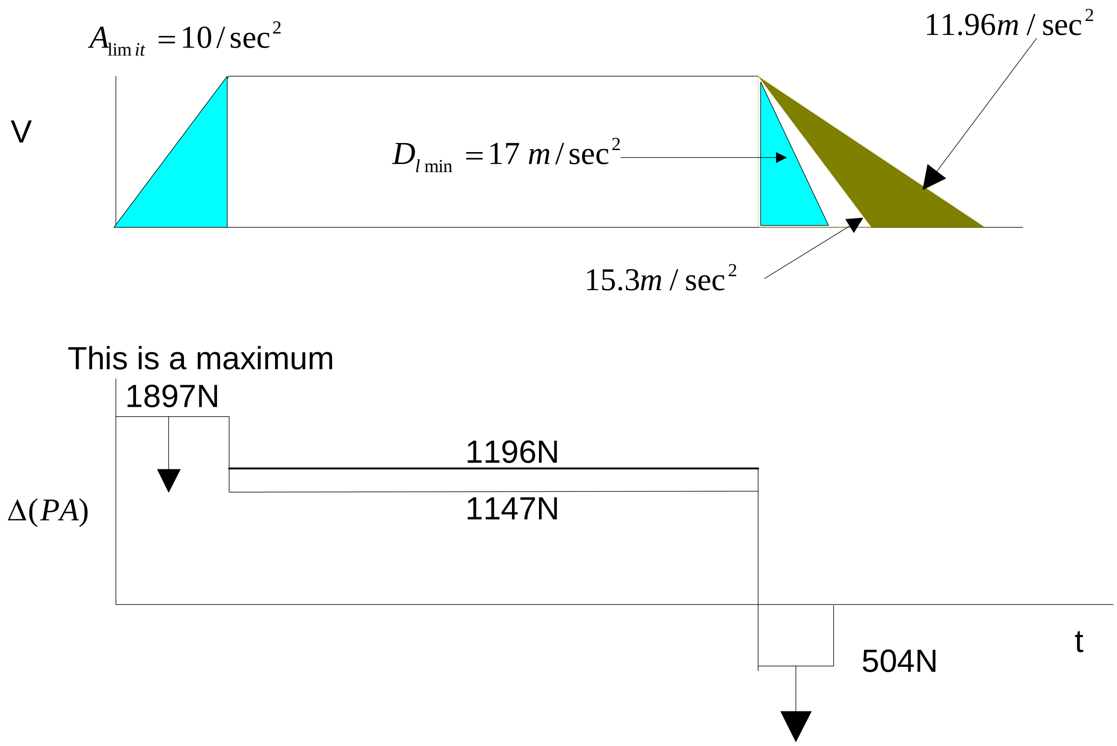
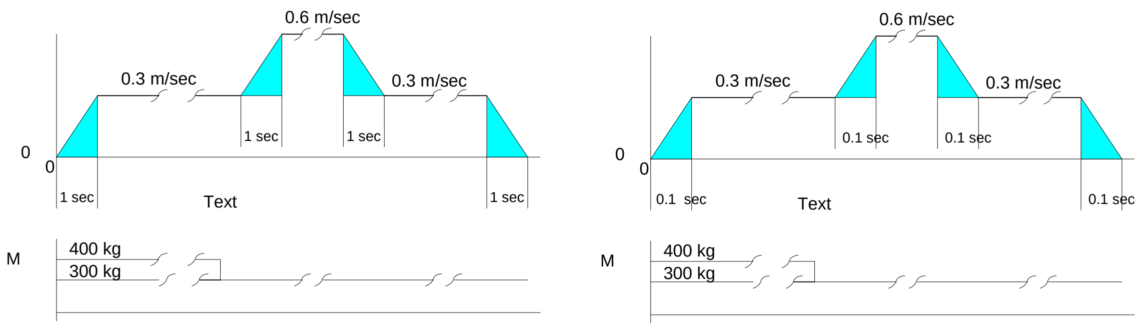


Figure 12.36 Load profile for example with acceleration limits.

In class problem:

For the two velocity and mass profiles shown determine the hydraulic force profiles.

Scenario 1 Scenario 2



Given F_{ext} = 1000N μ_d = 0.6 Resistive load	**Given F_{ext} = 1000N μ_d = 0.6 Resistive load**

Figure 12.37 and 12.38 Velocity and mass profiles for in class problems

12.5 Examples of Developing Hydraulic Force (Torque) Profiles.

In this section, we shall consider four special examples to illustrate the use of the Tables to generate the correct equations to calculate the appropriate force (torque) profiles.

12.5.1 Example #1

Consider the following rotary system. The nature of the external torque is such that it can be classified as a resistive system where $\Delta(PD_m) > 0$ throughout the cycle.

Please note that in this example, we shall use the symbol τ for torque but in the Charts, we use the letter T.

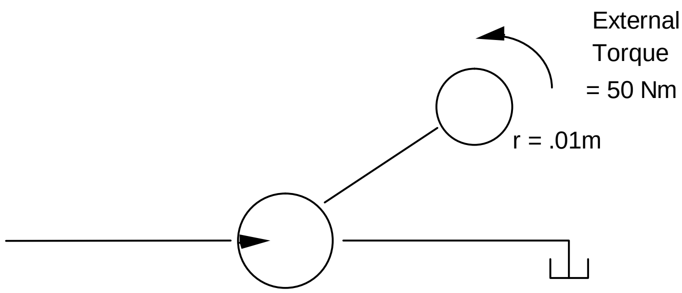


Figure 12.39 System to be examined

The system has a mass between 1000 and 2000 kg which is constant within any cycle. The radius of the inertial load is .01m. For the velocity profile specified, find the:

1. Inertia profile
2. Burden profile
3. Natural acceleration/deceleration
4. Hydraulic torque profile

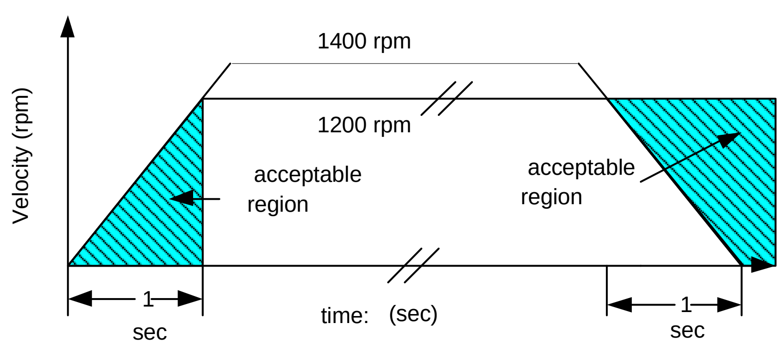


Figure 12.40 Velocity Profile

System constraints:

External torque (τ_{ext}) = 50 Nm

Viscous torque (τ_{vis}) = $\beta\dot{\theta}$ Nm

Viscous coefficient β = .01 Nm s

12.5.1.1 Calculate the Inertia Profile.

M = 1000 and 2000 kg

$I = M r^2/2$

$= 5 * 10^{-2}$ kg m^2 for 1000kg

$= 1 * 10^{-1}$ kg m^2 for 2000kg

The Inertia profile appears as:

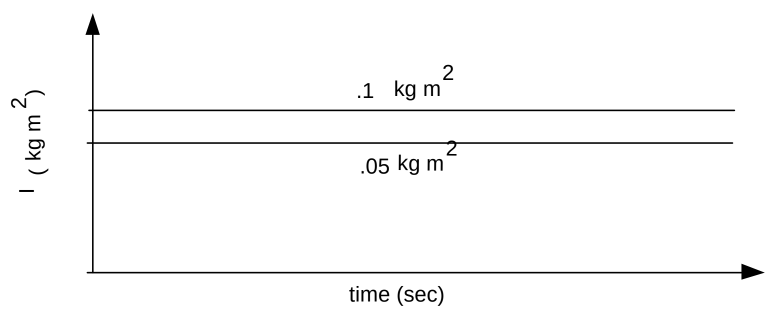


Figure 12.41 Inertia Profile

12.5.1.2 Consider the Burden Profile:

We first want to calculate any external torques on the system. We have one that is constant at 50 Nm. We have a velocity dependent term which we must calculate:

$$\tau_{vis} = \beta\dot{\theta}$$

$$= .01 \text{Nms} * 1200 \text{ rpm} * \text{min}/(60\text{s}) * 2\pi \text{rad/rev.}$$

$$\tau_{vis} = 1.26 \text{ Nm at } 1200 \text{ rpm}$$

$$= 1.47 \text{ Nm at } 1400 \text{ rpm}$$

It must be noted that in this problem, the velocity dependent terms are very small. We include them in the steady state terms for completeness; however, if they were larger, we would also have to consider them in the acceleration and deceleration parts of the cycle. Here, we neglect their contribution.

The burden profile would appear thus:

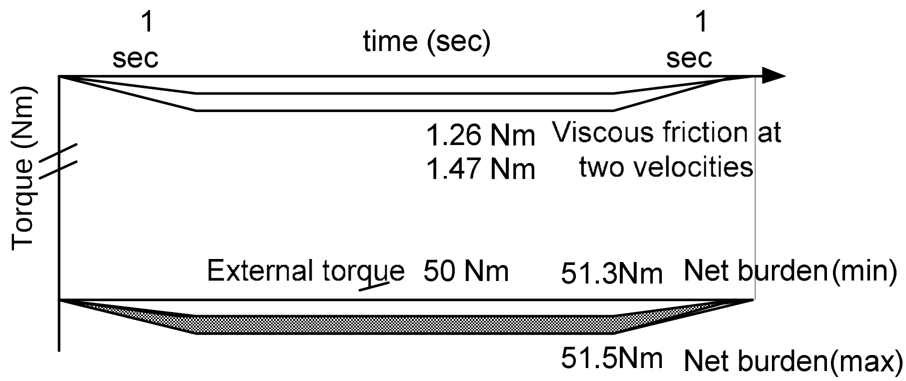


Figure 12.42 Burden Profile

12.5.1.3 Calculate the Natural Acceleration/Deceleration.

In these examples. the external torque is independent of the mass; thus all F_{bmax} terms, for example, are independent of mass.

Since there is a net negative burden, the external torque is resistive and hence the system **cannot** undergo a natural acceleration. We need only consider **natural deceleration $\mathbf{D_n}$**.

$$D_n = \Sigma\, \tau \,/\, I$$

The largest D_n occurs when I is the smallest and the burden the largest, that is:

$$D_{nmax} = 51.5\text{Nm}/.05 \text{ Kgm}^2 = 1030 \text{ rad/sec}^2$$

The smallest D_n occurs when I is the largest and the burden the smallest, that is:

D_{nmin} = 51.3Nm/.1 Kgm^2 = 512 rad/sec^2 (if one wanted to be very cautious, we could use a minimum value of 50 Nm in calculating D_{nmax} . We shall use 51.3 Nm in our calculations.

(NOTE: in using maximum and minimum values, one should make sure that the values so used occur at the same point in time or position. That is, it does not make any sense to use a mass value that does not occur at the burden value to be used. This prevents "over-design".)

From our requirements, the **acceptable deceleration** is given as;

D_{1200} = 1200 rpm/1sec = 126 rad/sec^2

This is shown in Figure 12.43.

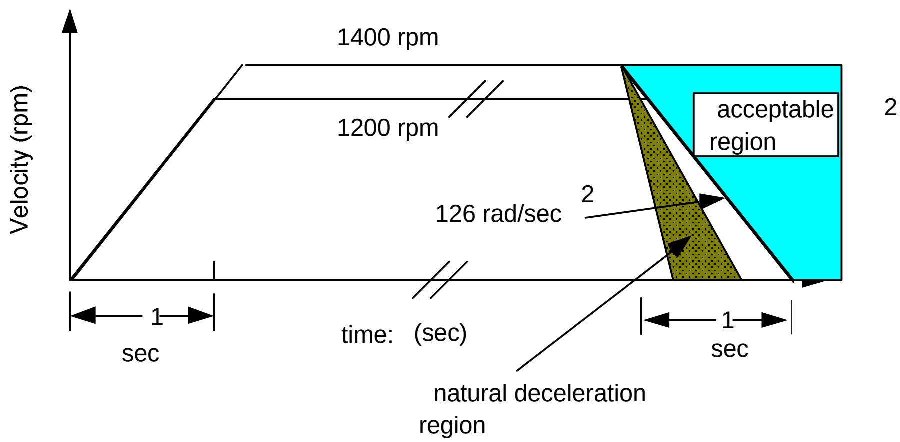


Figure 12.43 Velocity Constraints

We have a situation in that D_{nmax} and D_{nmin} do not fall in the acceptable range.

12.5.1.4. Develop the Hydraulic Force Profile.

To generate the hydraulic force profile, we go to our charts and find the appropriate case. From Table 12.2, it would appear that Case 4 is our situation. Here we see that **Δ(PD$_m$) is >0** and that

$$\Delta(PD_{m)}\Big|_{\min} = -T_{b_{\max}}\Big|_{I_{\min}} - I_{\min}\ D\alpha_{\max}$$

Here $\Delta\ (PD_m)_{min}$ is a minimum value; that is if $\Delta\ (PD_m)$ becomes smaller than this value, the actual deceleration will fall into the unacceptable range. (We have to make sure that $|\Delta(PD_m)_{min}| < |\ \tau_{bmin}\ |$ because the deceleration would turn into an acceleration. Common sense should prevail here.)

Using our maximum and minimum values ($\tau_{\beta max}$ = -51.5Nm, D_{lmax} = 126, I_{min} = .05 rad/sec^2), we find that;

$\Delta\ (PD_m)_{min}$ = (-(51.5) - .05*126)Nm
= 45.2Nm

(NOTE: From our burden profile, τ_b is negative. From our definition of deceleration, D_{limit} is positive)

Now we must consider the acceleration portion of the cycle. As before, we accept 126 rad/sec^2 as the limit for A_{limit}. From Figure Table 12.2, we find that

$$\Delta(PD_m)_{\max} = -T_{b\min_{I_{\min}}} + I_{\min}\,\alpha_{l\max}$$

Using our maximum and minimum values, we find that;

$\Delta\ (PD_m)_{max}$= (-(-51.3) + .1*126)Nm
= 63.9Nm

Finally, we consider the steady state portion of the cycle. This is straight forward because all inertial effects are gone and hence

$\Delta\ (PD_m)$ = - steady state burden. (note the minus sign). WE shall see that for negative burdens (positive hydraulic force/torque) we do not have to design for this condition because it will naturally occur. We will have to design for the acceleration and deceleration because they are forces by us.

The final hydraulic torque profile is shown below:

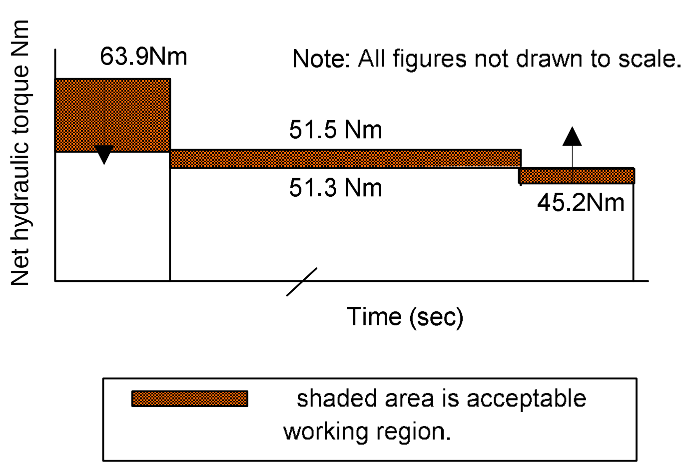


Figure 12.44 Hydraulic Torque Profile

12.5.2 Example #2

Consider the following rotary system:

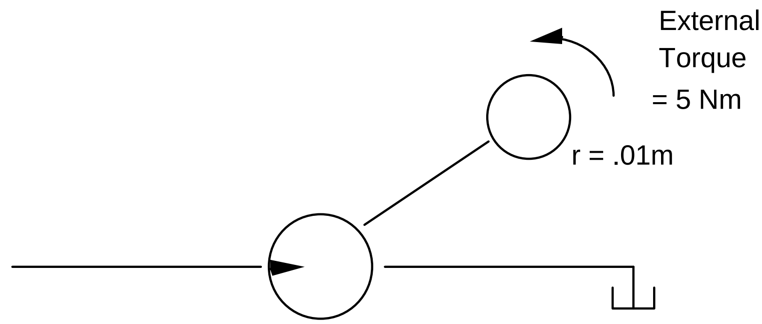


Figure 12.45. System to be examined

The system has a mass between 1000 and 2000 kg which is constant within any cycle. The radius of the inertial load is .01m. For the velocity profile specified, find the:

1. Inertia profile
2. Burden profile
3. Natural acceleration/deceleration
4. Hydraulic torque profile

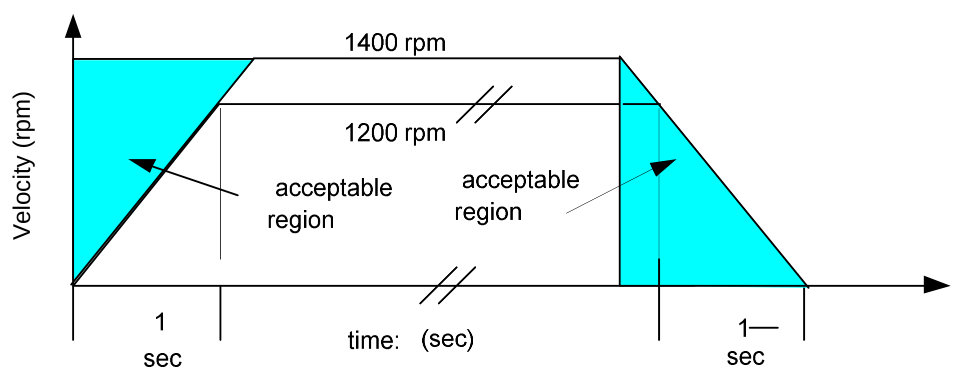


Figure 12.46 Velocity Profile

System constraints:

External torque (τ_{ext}) = 5 Nm

Viscous torque (τ_{vis}) = 0

12.5.2.1 Calculate the Inertia Profile.

$M = 1000$ and 2000 kg

$I = M r^2/2$

$= 5 * 10^{-2}$ kg m^2 for 1000kg

$= 1 * 10^{-1}$ kg m^2 for 2000kg

The Inertia profile appears as:

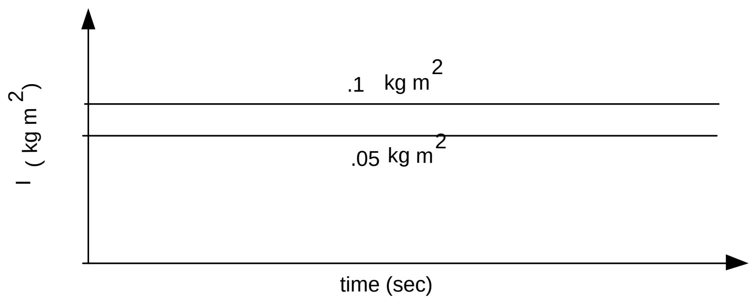


Figure 12.47 Inertia Profile

12.5.2.2 Consider the Burden Profile:

We first want to calculate any external torques on the system. We have one that is constant at 5 Nm. The burden profile would appear thus:

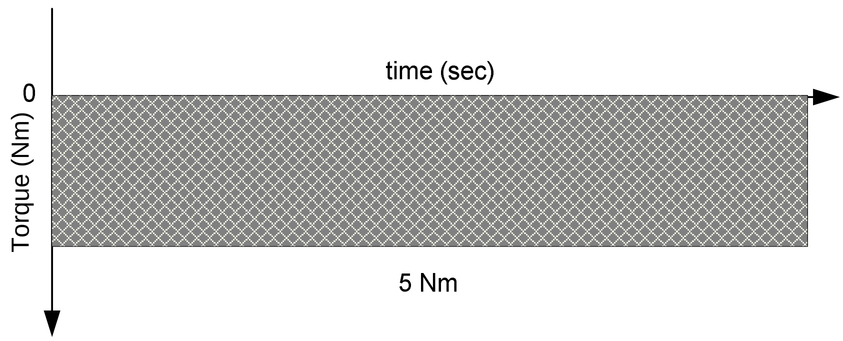


Figure 12.48 Burden Profile

12.5.2.3 Calculate the Natural Acceleration/Deceleration.

Since there is a net negative burden, the external torque is resistive and hence the system **cannot undergo a natural acceleration**. We need only consider natural deceleration D_n.

$$D_n = -\Sigma \tau / I$$

Largest D_n occurs when I is the smallest and the burden the largest, that is:

D_{nmax} = 5Nm/.05 Kgm^2 = 100 rad/sec^2

Smallest D_n occurs when I is the largest and the burden the smallest, that is:

D_{nmin} = 5Nm/.1 Kgm^2 = 50 rad/sec^2

But from our requirements, the acceptable deceleration is;

D_{1200} = 1200 rpm/1sec = 126 rad/sec^2

This is shown in Figure 12.49.

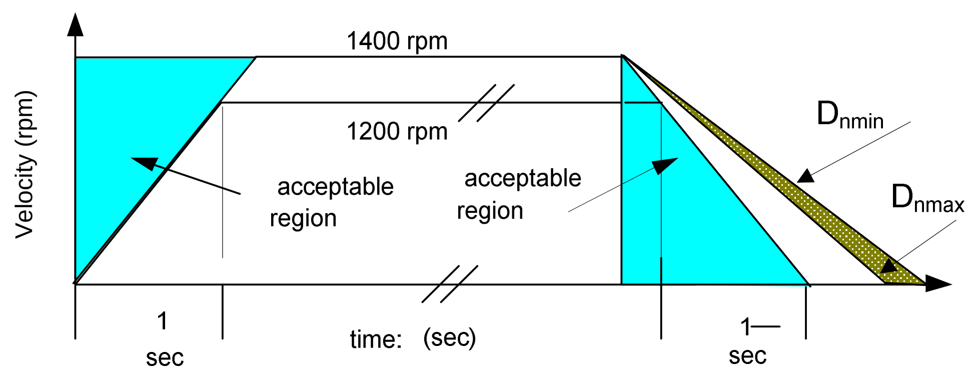


Figure 12.49 Velocity Constraints

We have a problem in that D_{nmax} and D_{nmin} do not fall in the acceptable range.

<u>12.5.2.4 Develop the Hydraulic Force Profile.</u>

To generate the hydraulic force profile, we go to our charts and find the appropriate case. From Table 12.2 it would appear that Case 1 is our situation. Here we see that **Δ (PD_m) is <0** and that

$$\Delta(PD_{m})\Big|_{\min} = -T_{b_{\min}}\Big|_{I_{\max}} - I_{\max}\ D\alpha_{\min}$$

Here Δ $(PD_m)_{min}$ is a minimum value; that is if Δ (PD_m) becomes smaller than this value, the actual deceleration will fall into the unacceptable range.

Using our maximum and minimum values, we find that;

Δ $(PD_m)_{min}$	=	(-(-5) - .1*126)Nm
	=	-7.6 Nm

(NOTE: From our burden profile, τ_b is negative. From our definition of deceleration, D_{limit} is positive)

Now we must now consider the acceleration portion of the cycle. As before, we accept 126 rad/sec^2 as the limit for α_{limit}. From Table 12.2 we find that

$$\Delta(PD_m)_{\min} = -T_{b\max_{I\max}} + I_{\max}\ \alpha_{l\min}$$

Using our maximum and minimum values, we find that;

$$\Delta\ (PD_m)_{min} = (-(-5) + .1*126)\ \text{Nm}$$
$$= 17.6\ \text{Nm}$$

Finally, we consider the steady state portion of the cycle. This is straight forward because all inertial effects are gone and hence

$$\Delta\ (PD_m) \quad = -\ \text{steady state burden.}$$

The final hydraulic force profile is shown below:

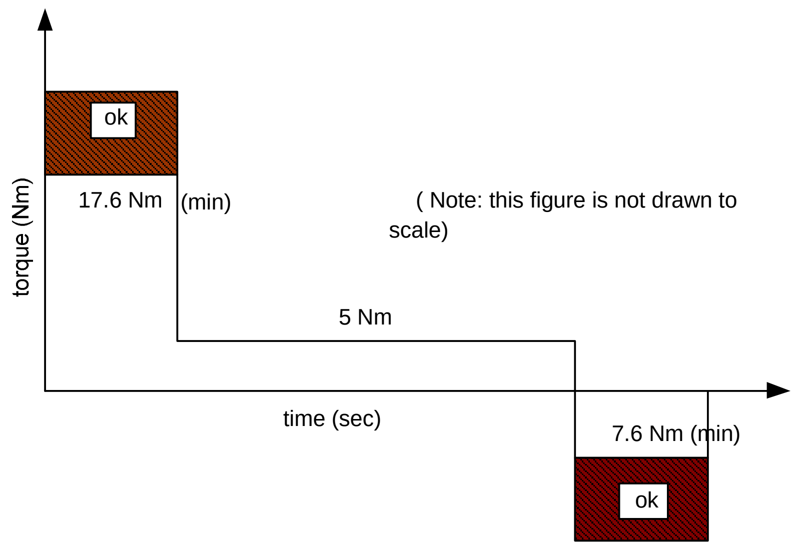


Figure 12.50 Hydraulic Force Profile

12.5.3 Example #3

Consider the following rotary system In this example, we have a resistive system with stiction dominant at t = 0.

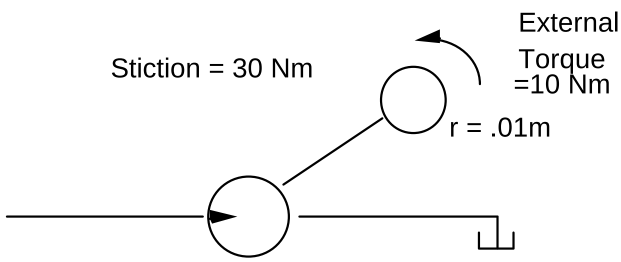


Figure 12.51 System to be examined

For the velocity profile specified, find the :

1. Inertia profile
2. Burden profile
3. Natural acceleration/deceleration
4. Hydraulic torque profile

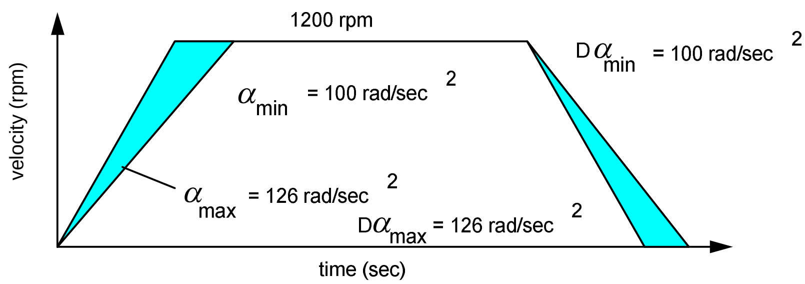


Figure 12.52 Velocity Profile

System constraints:

External torque (τ_{ext}) = 10Nm

I can vary between .1 and .11 kg m^2

Stiction is significant at 30 Nm

12.5.3.1 The Inertia Profile appears as:

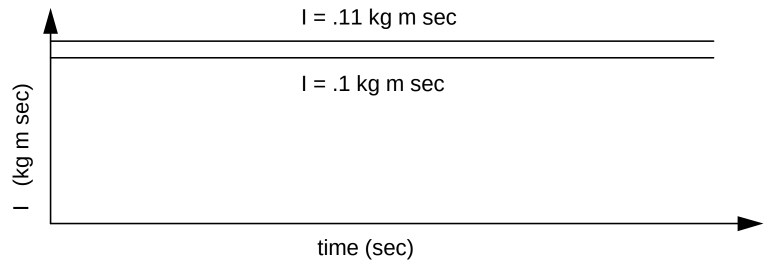


Figure 12.53 Inertia Profile

12.5.3.2 Consider the Burden Profile:

We first want to calculate any external torques on the system. For t > 0, the τ_{ext} = 10Nm; however, at t=0, we have stiction. This is reflected in the burden profile as:

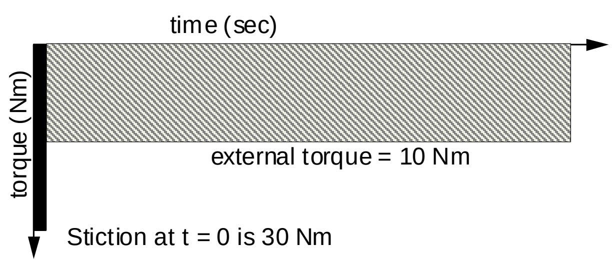


Figure 12.54 Burden Profile

12.5.3.3 Calculate the Natural Acceleration/Deceleration.

Since there is a net negative burden, the external torque is resistive and hence the system cannot undergo a natural acceleration. We need only consider natural deceleration D_n. (Note: since the system is already in motion, we can forget stiction here.)

$$D_n = -\Sigma\, \tau / I$$

The largest D_n occurs when I is the smallest and the burden the largest, that is:

$$D_{nmax} = 10\text{Nm}/.1\ \text{Kgm}^2 = 100\ \text{rad/sec}^2$$

The smallest D_n occurs when I is the largest and the burden the smallest, that is:

$$D_{nmin} = 10\text{Nm}/.11\ \text{Kgm}^2 = 90.0\ \text{rad/sec}^2$$

But from our requirements, the acceptable acceleration and deceleration is;

$$\alpha_{limits} = D\alpha_{limits} = 126\ \text{rad/sec}^2 \text{ to } 100\ \text{rad/sec}^2$$

This is shown in Figure 12.55.

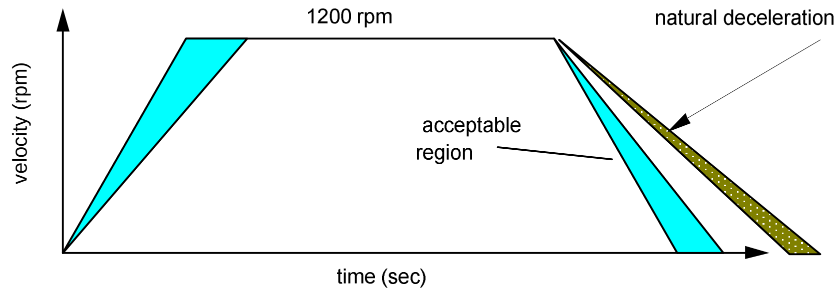


Figure 12.55 Velocity Constraints

We have a problem in that D_{nmax} and D_{nmin} do not fall in the acceptable range.

12.5.3.4. Develop the Hydraulic Torque Profile.

To generate the hydraulic torque profile, we go to our charts and find the appropriate case. From Table 12.2 it would appear that Case (3) is our situation. Here we see that **$\Delta(PD_m)$ is <0** and that

$$\Delta(PD_{m)}\Big|_{\min} = -T_{b_{\min}}\Big|_{I_{\max}} - I_{\max}\ D\alpha_{\min}$$

$$\Delta(PD_{m)}\Big|_{\max} = -T_{b_{\max}}\Big|_{I_{\min}} - I_{mim}\ D\alpha_{\max}$$

Using our maximum and minimum values, we find that;

$\Delta\ (PD_m)_{min}$ = (-(-10) - .11*100)Nm
= -1.0 Nm (the minus indicating it is a back pressure)
$\Delta\ (PD_m)_{max}$ = (-(-10) - .1*126)Nm
= -2.6 Nm

(NOTE: From our burden profile, τ_b is negative.)

During steady state, $\Delta\ (PD_m)$ = - (-10)Nm, steady state torque.

Now we must now consider the acceleration portion of the cycle. From Table 12.2 we find that

$$\Delta(PD_m)_{\max} = -T_{b\min_{I_{\min}}} + I_{\min}\ \alpha_{l\max}$$

$$\Delta(PD_m)_{\min} = -T_{b\max_{I_{\max}}} + I_{\max}\ \alpha_{l\min}$$

Using our maximum and minimum values, we find that;

$$\Delta\ (PD_m)_{max} = (-(-10) + .1*126)Nm$$
$$= 22.6\ Nm$$
$$\Delta\ (PD_m)_{min} = (-(-10) + .11*100)Nm$$
$$= 21\ Nm$$

We now must consider stiction. At t = 0, stiction is 30Nm. This value is greater than that which occurs during acceleration. This means, if we choose 30 Nm, we will exceed the acceleration specifications. This problem must be addressed hydraulically when we configure the circuit.

The final hydraulic torque profile is shown below:

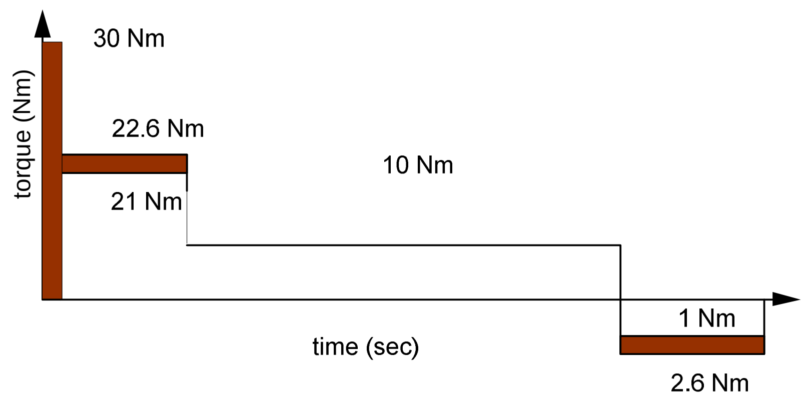


Figure 12.56 Hydraulic Torque Profile

12.6 Hydraulic Circuit Design.

In this section, we shall examine how to translate the information from the hydraulic load profile to circuit configuration. We will introduce criteria which will provide a guide to circuit design. The first relates to meter in vs meter out.

12.6.1 Meter-in vs meter-out

In general, a hydraulic force profile is comprised of three parts, corresponding to acceleration, steady state and deceleration portions of the velocity profile. With reference to Figures 12.50 and 12.56, for the conditions specified (a resistive system which exhibits a natural deceleration), the hydraulic force during acceleration and/or steady state motion of the system always remains positive; however, the hydraulic force during deceleration could be negative, positive or zero, depending upon the natural deceleration value and the imposed deceleration constraints.

The two most common situations are illustrated in Figure 12.57. In Figure 12.57(a) the hydraulic force $\Delta(PA)$ remains positive throughout the cycle and from cycle to cycle, which reflects cases (4), (5) and (6) in Tables 12.1 and 12.2. Figure 12.57(b) reflects the situations where the hydraulic force **$\Delta(PA) < 0$** occurs at sometime during the cycle (cases (1), (2) and (3) in Figure 12.1 and 12.2). The sign of $\Delta(PA)$ can be used to establish criteria for the placement of flow modulation functions in a meter-in or meter-out configuration.

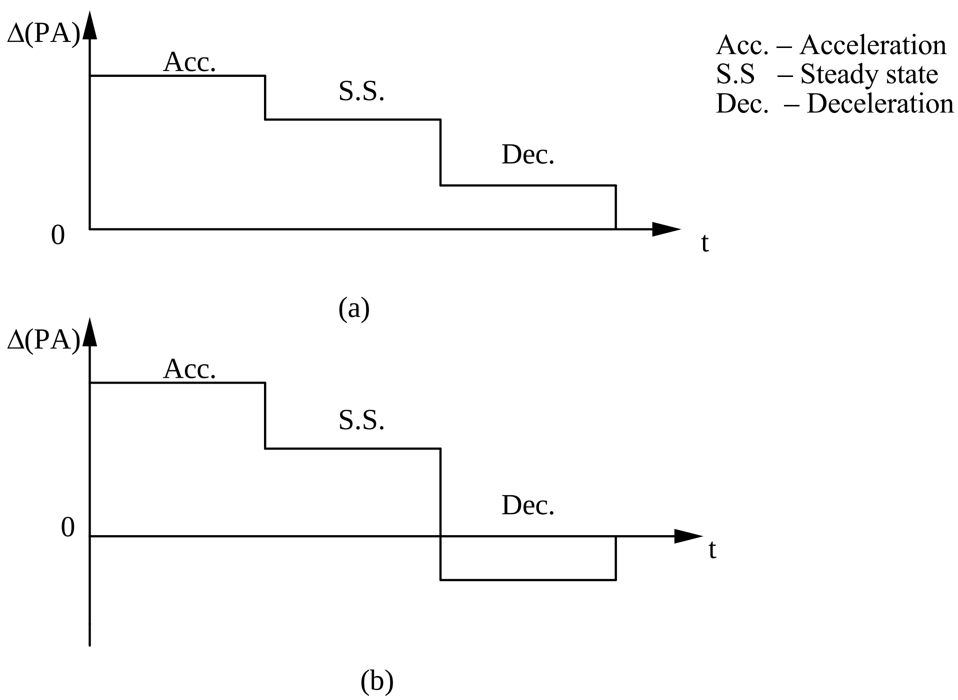


Figure 12.57 Typical Hydraulic Force Profile

Please note: in this edition of the notes, there is no Figure 12.58.
The first criterion deals with a negative hydraulic force during any part of the cycle.

Criterion One:

If, at any time during the cycle, $\Delta(PA) < 0$, ($\Delta(PD_m) < 0$), a negative hydraulic force (torque) across a hydraulic actuator (motor) is required. Since this translates into a hydraulic requirement of a back pressure on the downstream side of the actuator, then a meter-out circuit must be used*. The exception to this criterion is (1) if the negative $\Delta(PA)$, ($\Delta(PD_m)$) occurs only during deceleration and (2) the velocity is fixed during the cycle, then a meter-in circuit can be used to set the velocity and a pressure limiting device used to decelerate the system.

The implication of this first criteria is that meter-in configuration is not permitted if the hydraulic force so determined is applied in the direction opposite to the system motion (the exception duly noted).

The second criterion deals with a positive hydraulic force throughout the cycle. If $\Delta(PA) \geq 0$ is deduced, both meter-in and meter-out configurations are able to (in the steady state) satisfy the requirement of a positive hydraulic force. With significant perturbations in the burden such as a sudden reduction in friction, the potential for "jerky" motion exists. This problem will provide an additional constraint in the decision between meter-in and meter-out configurations. Experience tends to show that a meter-out circuit is more effective than a meter-in circuit in reducing "jerk" and hence minimizes the likelihood of damage to the system.

The second criterion can then be stated as:

---

* It must be noted that a counterbalance valve is often used in a hydraulic circuit to create a back pressure. The reason the counterbalance valve is not considered as an option in this chapter is that it is more preferably used in the applications where the pump modulates flow rate. If the flow modulation is to be achieved by the use of flow valves, the combination of a counterbalance valve and a flow device (often in meter-in), shortened as MI+C.B.V., unnecessarily complicates the circuit and increases the initial cost in comparison to a meter-out circuit. However this option (MI+C.B.V.) will be taken into account when the system is in a "run-away" and an "over-center" condition, as will be discussed in subsequent chapters.

Criterion two:

If $\Delta(PA) \geq 0$, ($\Delta(PD_m) \geq 0$) throughout the cycle and from cycle to cycle and sudden changes in the burden are known to occur at any time during the cycle, a force opposite to the direction of motion is required to damp out any sudden changes in system motion. This translates into a requirement of a hydraulic pressure downstream; therefore, a meter-out configuration is preferred.

This criterion reflects that in meter-out circuits, when a sudden reduction in the burden occurs, the system tends to move in a "jerky" fashion; however, the flow modulation device in the return line only allows the fluid to pass it at the rate the device permits. Consequently, the pressure in the chamber between the device and the actuator is built up instantaneously to compensate for the sudden drop of the burden. It is this dynamic compensation of a meter-out circuit that effectively dampens out the sudden acceleration — "jerk".

The third criterion deals with situations in which the burden fluctuations are considered to be insignificant.

Criterion three:

If $\Delta(PA) \geq 0$,($\Delta(PD_m) \geq 0$) throughout the cycle and from cycle to cycle and sudden burden changes do not happen at any time during the cycle, a back pressure is not necessary, then a meter-in circuit is preferable.

This criterion expresses a preference for a meter-in configuration whenever possible because such a placement could reduce the number of components in the final circuit. (Instead of using two components, one flow control device in meter-in is able to implement flow modulation for the motions in both directions.) In addition, problems associated with large pressure transients in meter-out circuits (pressure intensification for example) can be avoided.

These criteria are best illustrated with a flow chart in Figure 12.59.

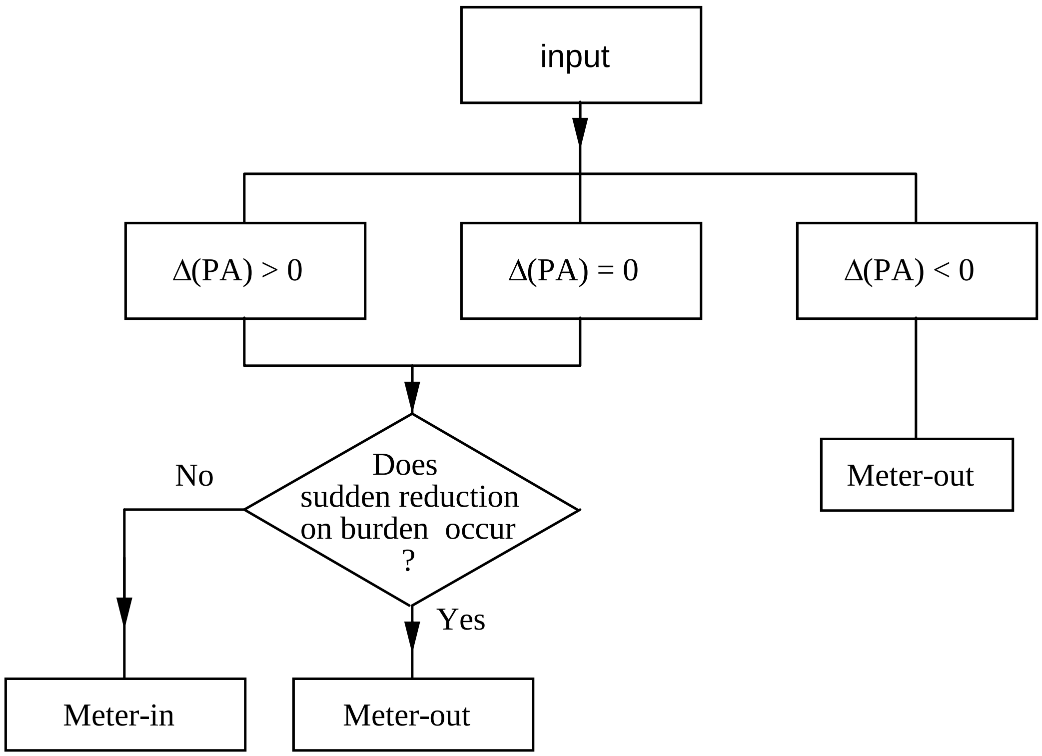


Figure 12.59 Design Criteria Based on Hydraulic Force Information

12.6.2 Limiting Acceleration

We have seen several examples in which the acceleration portion of the hydraulic force/torque profile have been limited. A question arises as to how we can design a circuit to accomplish this. We must make it clear that there are many ways to do this as you would expect in any design problem. We shall look at one example in which acceleration limits are inherent in the circuit if pressure is limited.

Consider the design problem:

Design a circuit which will satisfy the following angular velocity profile and hydraulic load profile.

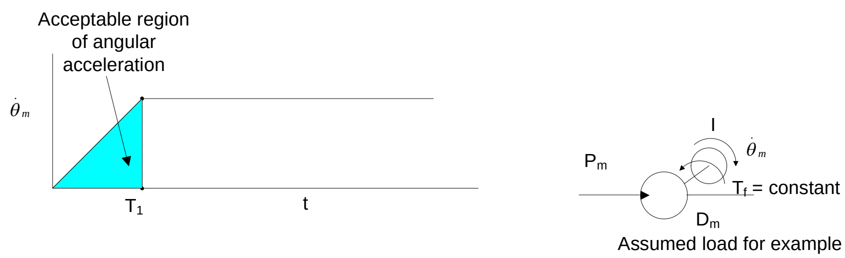


Figure 12.60(a) Angular Velocity Profile and Assumed load

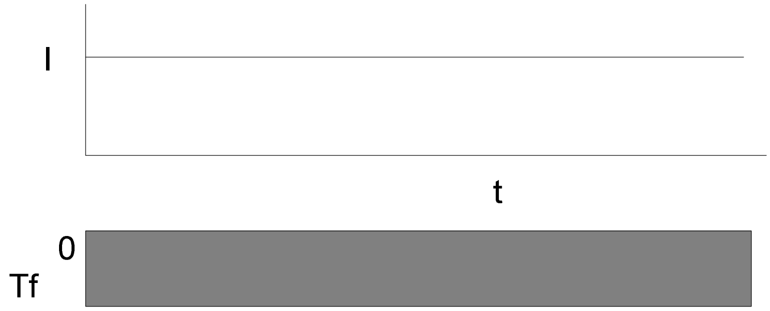


Figure 12.60(b) Mass and burden profiles

In this case, there is only one inertia, I, and the external load T_f is constant. The mass and burden profiles are very simple and constant as shown in Figure 12.60(b). From Figure 12.60(a) and our Table 12.2, we get

$$\Delta(PD_m)_{\max} = -T_{b\min_{I_{\min}}} + I_{\min}\,\alpha_{I\max}$$

Because T_f and I are single values, we can rewrite this as:

$$\Delta(PD_m)_{\max} = -T_b + I\,\alpha$$

At steady state conditions, this becomes:

$$\Delta(PD_m)_{\max} = -T_b$$

$T_b = T_f$ and T_f is negative in this case since the burden is negative.

Our hydraulic torque profile becomes:

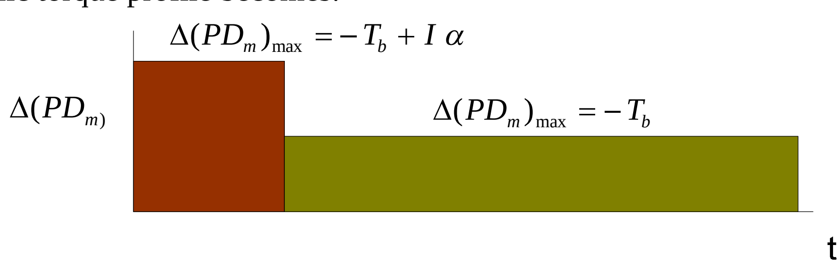


Figure 12.60(c) Hydraulic Torque Profile

But since $P_m = \dfrac{\Delta(PD_m)}{D_m}$

then $P_{m\max} = \dfrac{\Delta(PD_m)_{\max}}{D_m} = \dfrac{-T_b + I\,\alpha}{D_m}$

For steady state conditions, this becomes:

$$P_{m_{ss}} = \frac{\Delta(PD_m)_{ss}}{D_m} = \frac{-T_b}{D_m}$$

Our hydraulic pressure profile at the motor is thus:

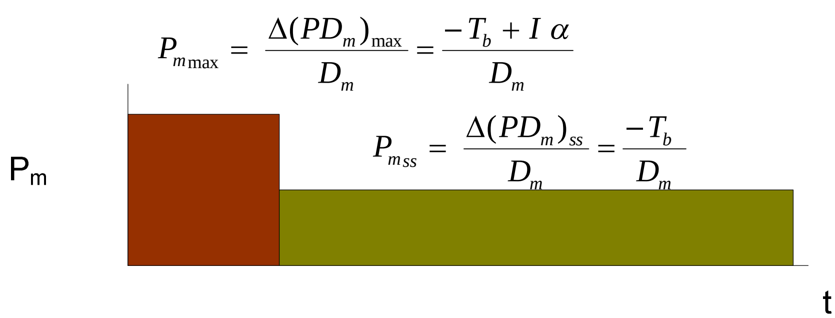


Figure 12.60(d) Hydraulic Pressure profile

So now the question arises as to how can we create the velocity profile given the hydraulic torque profiles? One circuit could be:

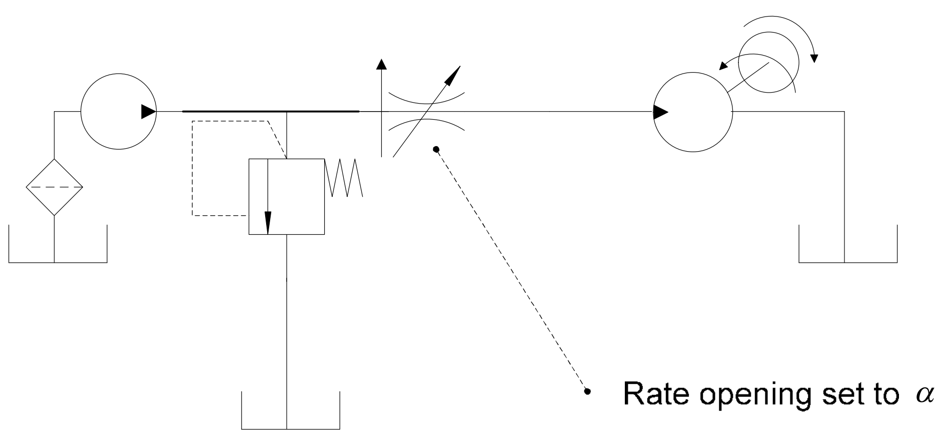


Figure 12.60(e) Flow control to change velocity

In this circuit since $\dot{\theta} = \frac{Q_m}{D_m}$ we can achieve the velocity profile by changing the flow rate through the valve at a fixed rate. If the relief valve pressure is set high enough (higher than the maximum motor pressure), then we can meet the velocity specifications. Because the flow control valve is pressure compensated, the flow rate will not change if the load conditions change which is nice (even though our burden profile says it will not).

What would the pressure profile look like at the motor and the pump?

If you look at the equation;

$$P_{m\max} = \frac{\Delta(PD_m)_{\max}}{D_m} = \frac{-T_b + I\,\alpha}{D_m}$$

Since α is controlled by the flow control valve, then the pressure profile will look exactly as we have shown it in Figure 12.60(d).

What about at the pump? Well since the pump is fixed displacement, the pump must always deliver flow but the valve accepts less due to the velocity profile it is following. So, the relief valve must be opened and it will only open at its cracking pressure. Therefore as long as the pump flow is greater than the valve requested flow, the pressure at the pump will be at the relief valve pressure. If we assume that the pump flow always exceeds the valve requested flow even at steady state, the pump pressure and motor pressure would appear as in Figure 12.60(e). In the figure the cross hatched area indicates the pressure drop which occurs across the flow control valve.

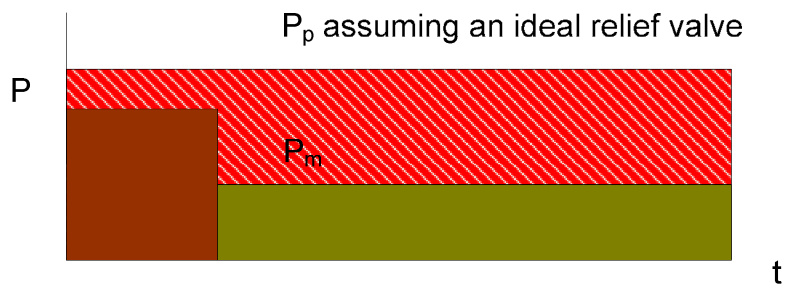


Figure 12.60(e) Pump pressure and motor pressure for $Q_v < Q_p$ at all times.

If at **steady state** the pump flow is exactly matched to the valve flow, then the pump pressure will be just slightly higher than the motor pressure due to losses across the flow control valve. Thus the pressure profiles would look like:

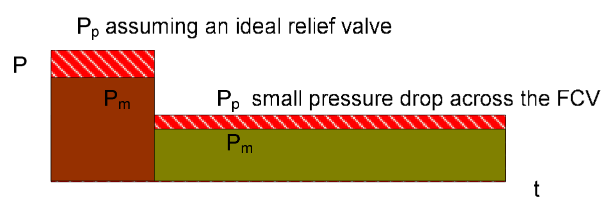


Figure 12.60(f) Pump pressure and motor pressure for $Q_v = Q_p$ at steady state

However, this is not a very cheap option. A rate controlled pressure compensated valve is expensive and the rate control in itself is very complex for this case. We do have other alternatives. Examine Figure 12.60(d). This Figure shows that if we can keep the pump pressure constant while the pump is delivering flow until the pump flow matches the desired rotary motor speed, then we can limit the angular acceleration simply using the pump pressure. At steady state, the acceleration is zero (the pump flow matches the desired motor flow) and the pressure is simply that due to the burden. This does not have to be designed for as it will occur naturally. Thus two alternates circuit to limit acceleration could be:

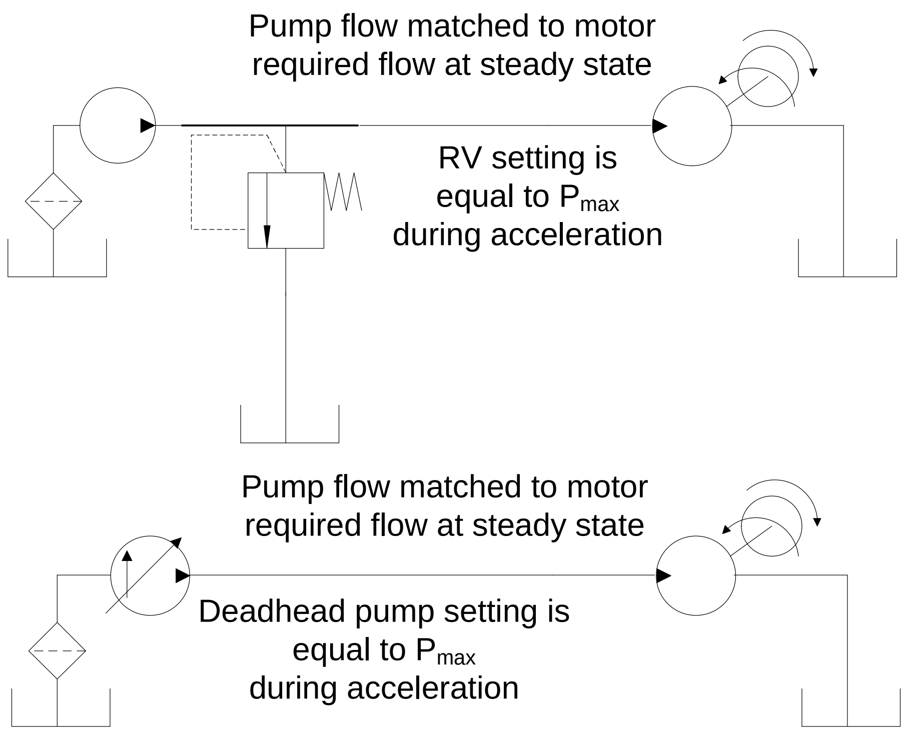


Figure 12.60(g) Using Pressure to Limit Acceleration.

There is a little problem here. Recall that:

$$P_{m\max} = \frac{\Delta(PD_m)_{\max}}{D_m} = \frac{-T_b + I\,\alpha}{D_m}$$

Since P_{mmax} is a maximum, we must ensure we set the relief valve cracking pressure (or pump cutoff pressure) lower than the desired value because the **actual pressure** when the relief valve is fully opened (or pump fully destroked) at no flow conditions is **higher** than the condition of little flow through the relief valve (or pump fully stroked). Since P_{mmax} is a maximum, we must ensure that it truly is. This is simply a problem of setting the RV cracking pressures (or pump deadhead pressures) correctly. See Figure 12.60(j).

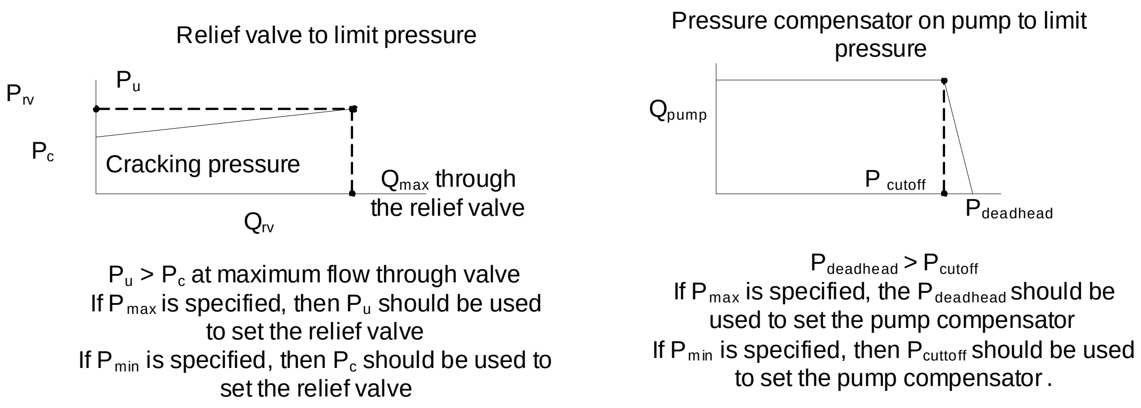


Figure 12.60(j) Practical limitation on using pressure to accelerate

Thus we can state another design suggestion:

Pump pressure via the relief valve cracking pressure or deadhead setting on the pump can be used to limit load acceleration (whether linear or rotary). This is inherent in all circuits. Care must be made to set the correct relief valve setting (or pump deadhead pressure) to meet the P_{max} or P_{min} conditions due to non-ideal pressure flow characteristics of relief valves or pressure compensators.

An interesting offshoot of this problem is what if our velocity limit was as in Figure 12.60(i). In this case our hydraulic pressure is a minimum during acceleration. (Show this). But we can still use the circuits of Figures 12.60(e) and (g) but we must ensure we set the pressure correctly.

<u>12.7.1   Examples of Simple Circuit Design</u>

To illustrate the configuration of hydraulic circuits, we shall start with very simple system constraints. Initially, we will not do any calculations but deal with "generic" situations. Later, we shall carry through several examples from start to finish including all stages of the design process.

12.7.1.1 Example 1

In our first example, we shall consider cases where acceleration/deceleration times are not important from a requirement point of view. This does not mean that one can ignore the velocity and $\Delta(PA)$, $\Delta(PD_m)$ profile. Indeed, these profiles must still be generated. For example, consider the velocity profile as illustrated in Figure 12.60 where all we are concerned with are the magnitudes of V, (i.e., not the acceleration or deceleration rates).

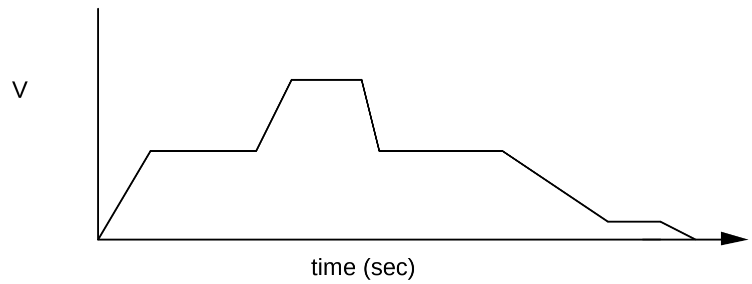


Figure 12.60 Velocity Profile

The sign of the burden profile is very important here. If it is negative, we would have a positive $\Delta(PA)$, $\Delta(PD_m)$ profile and hence a meter-in circuit would suffice. If the burden is positive, then $\Delta(PA)$, $\Delta(PD_m)$ would be negative and a meter-out circuit must be used. If we set circuit pressure to equal just the steady state value of $\Delta(PA)$, $\Delta(PD_m)$, then the system may never accelerate. Obviously, we must know what $\Delta(PA)$, $\Delta(PD_m)$ at steady state is and then set our circuit pressure (or pressure limiting device) high enough to get reasonable values. Hence we always should do a velocity and $\Delta(PA)$, $\Delta(PD_m)$ profile even though acceleration/deceleration times are not constraints.

Let us suppose that our $\Delta(PA)$, $\Delta(PD_m)$ profile is as shown in Figure 12.61.

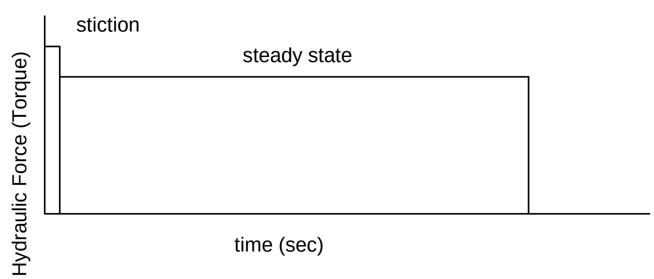


Figure 12.61 Hydraulic Force(Torque) Profile

We have not accounted for inertial effects. So when we choose a system pressure for $\Delta(PA)$, $\Delta(PD_m)$, we should choose the value high enough to overcome stiction and provide a reasonable acceleration rate there-of. Since $\Delta(PA)$, $\Delta(PD_m)$ is positive during deceleration, we can rely on "coasting" the system to a stop if we had to.

One possible circuit which could satisfy the $\Delta(PA)$, $\Delta(PD_m)$ and velocity profiles of Figures 12.60 and 12.61 could be as illustrated in Figure 12.62.

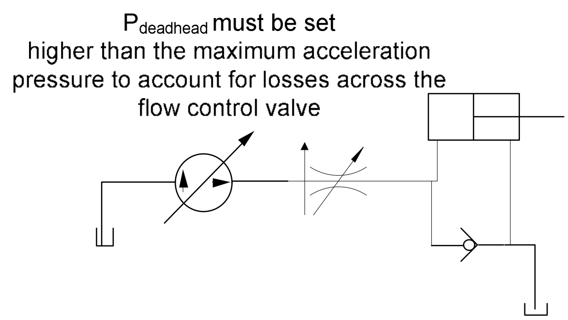


Figure 12.62 Circuit

The flow control valve is used to adjust the actual speed. If the flow control valve (FCV) is opened quickly, the acceleration is limited by the deadhead pressure ($P_{deadhead}$) of the pump. If the FCV is closed quickly, the natural deceleration of the system will "kick" in and the check valve (CV) will prevent cavitation. We could also use a meter - out circuit as shown in Figure 12.63

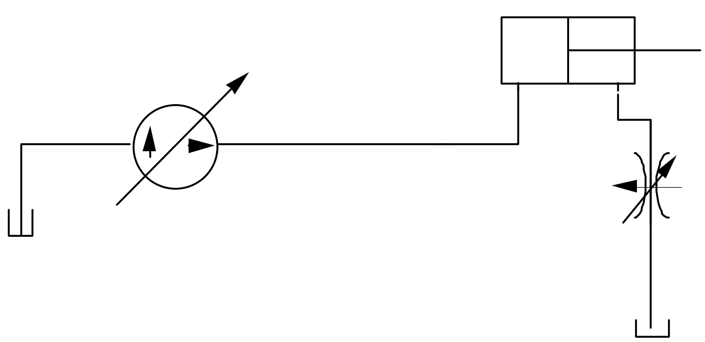


Figure 12.63 Meter –out circuit

However, if the FCV is closed too quickly, the pressure in the downstream line could increase to unacceptable levels. This is one of the reasons we chose to use a criteria which prefers a meter-in circuit.

If our example required bi-directionality, then one possible circuit might be as in Figure 12.64. Note; We have not considered the center position at this time.

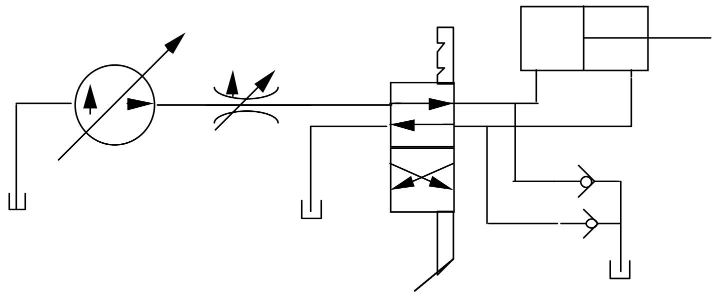


Figure 12.64 Bi-directional meter-in circuit

The check valves are used to prevent cavitation in the event the flow control valve is closed too quickly.
An alternate circuit could also be as in Figure 12.65. The variable displacement pump provides the flow control function. There are very few losses in this circuit.

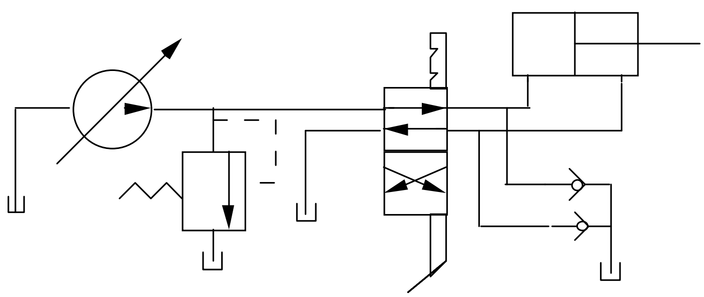


Figure 12.65 Pump control for meter-in application.

Note: If the natural deceleration is very large, then we could probably do away with the check valves from tank.

12.7.1.2 Example 2

Let us now consider the situation in which the **burden is positive** and hence $\Delta(PA)$, $\Delta(PD_m) < 0$ such as illustrated in Figure 12.66.

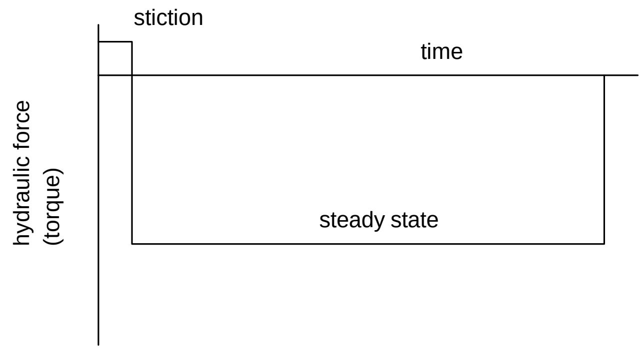


Figure 12.66 Hydraulic force (Torque) Profile

Again, we choose Δ(PA), Δ(PD_m) to overcome stiction but because Δ(PA), Δ(PD_m) <0 for most of the cycle, we must use a meter-out circuit such as illustrated in Figure 12.67.

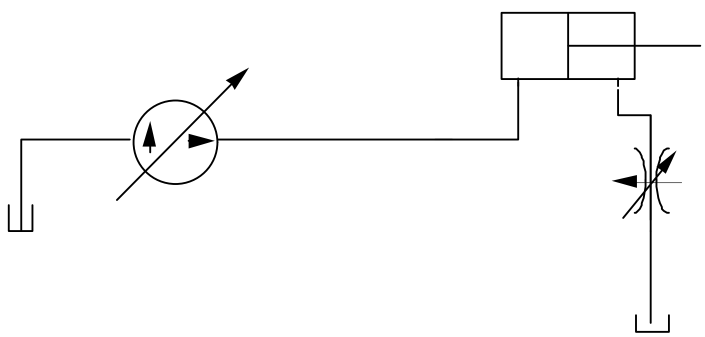


Figure 12.67 Meter-out circuit

However, this is a very poor circuit because we really need the pump only to overcome stiction. A better circuit would be as illustrated in Figure 12.68.

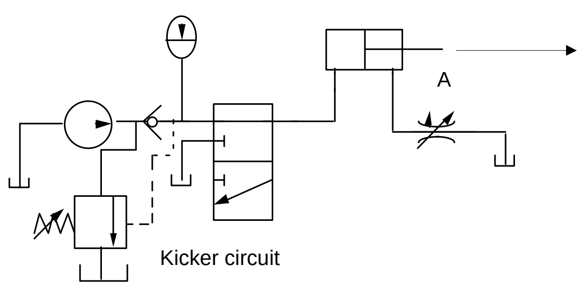


Figure 12.68 Unloading valve (Kicker circuit) to overcome friction.

If we close the FCV too quickly, we may have excessive pressure transients at A. This can be protected by using a RV across the FCV as illustrated in Figure 12.69.

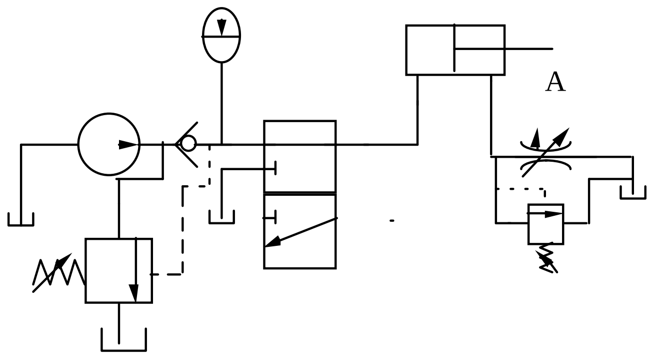


Figure 12.69 Protection of meter-out using a bypass relief valve

12.7.1.3 Example 3

Let us consider a situation in which we have a Δ(PA), Δ(PD_m) profile as shown in Figure 12.70.

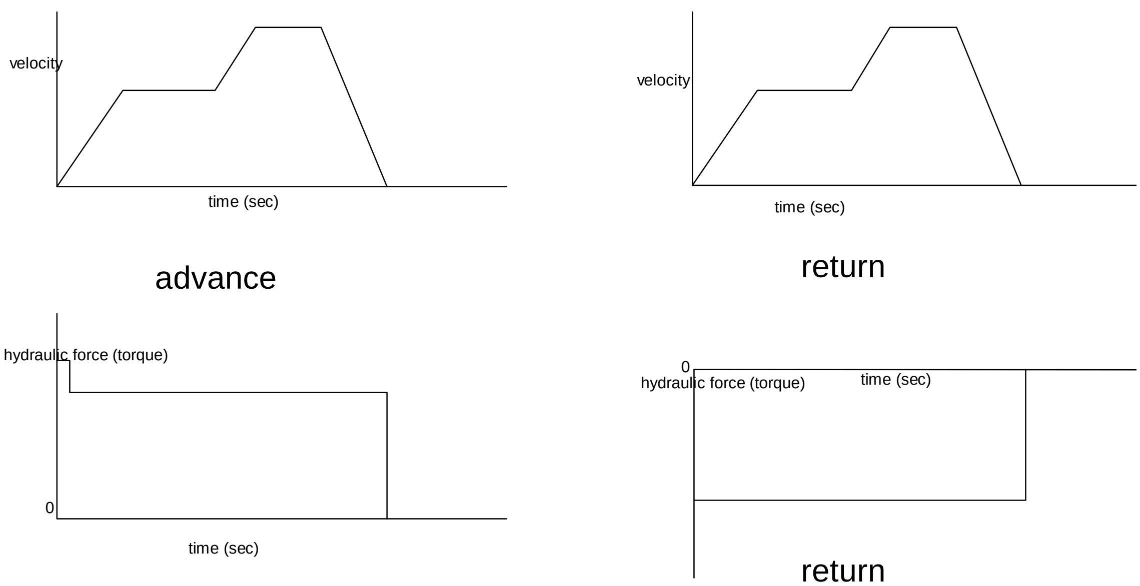


Figure 12.70. Velocity (Linear or angular) and Hydraulic Force (Torque) Profiles

Note;

- acceleration and deceleration is not a concern; it is not limited.
- we must be able to change velocity

- In forward direction, hydraulic force is positive. All valves can be upstream
- In reverse direction, hydraulic force is negative. Valves should be downsteam in the first instance.

We choose our $\Delta(PA)$, $\Delta(PD_m)$ maximum to overcome stiction and to give a reasonable acceleration term. For this case, we configure each case separately. For the forward direction, we have:

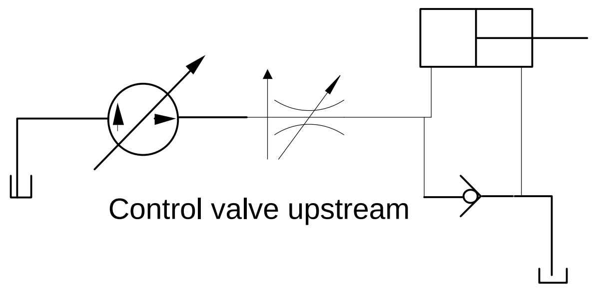


Figure 12.71 Meter-in to satisfy forward profile

In the reverse case, we have:

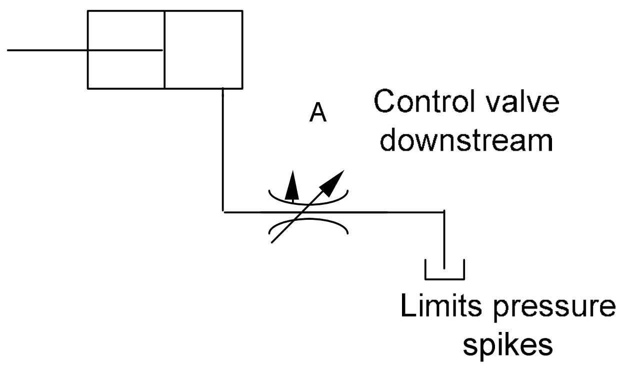


Figure 12.72 Reverse meter-out circuit

If we combine these two circuits to give one, we could end up with a circuit such as:

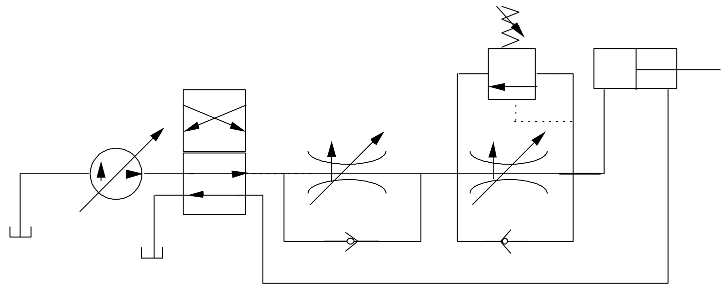


Figure 12.73 Combined forward and reverse circuits

In these circuits, if we wanted to lock the actuator, we could use the FCV's themselves. We could also use pilot operated check valves to do the same job as illustrated in Figure 12.74.

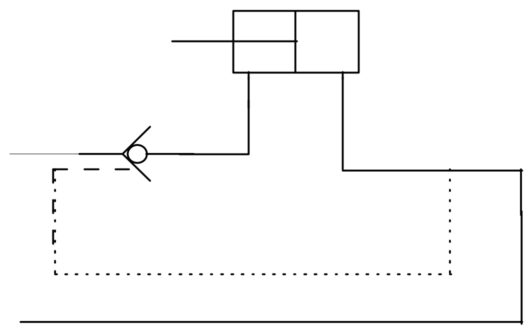


Figure 12.74 Locking using a pilot operated check valve
Note that locking is essential in reverse due to the existence of a constant negative hydraulic force

A circuit which can achieve the specified constraints is a closed system more commonly referred to as a hydrostatic unit. This is illustrated in Figure 12.75.

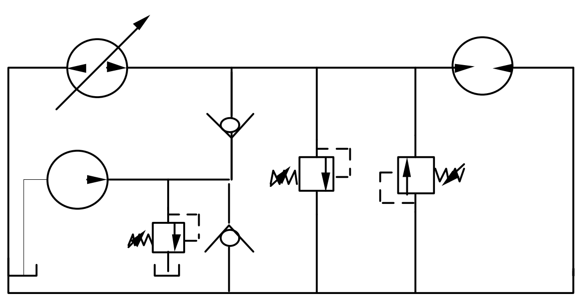


Figure 12.75 Hydrostatic circuit

This circuit is very energy efficient because no flow control valves are used. The circuit acts as a meter-in and meter-out function and although not shown, is reversible. This circuit is a closed circuit (note, **not** a closed loop circuit) in which the return line fluid from the motor is returned to the inlet of the pump directly. The secondary pump is used to maintain a positive pressure in the return line (in the range of .3 – 1 MPa, (50 – 150 psi)) to prevent cavitation due to leakage. The cross over relief valves allow a limited positive back pressure to exist on the motor in cases of an overrunning (positive burden) situation. Thus, in these cases, the relief valve is in fact a counterbalance valve.

12.7.1.4 Example 4

Let us now consider a situation in which the acceleration/deceleration range is defined. We shall place a few constraints as listed:

(a) rotary system
(b) bi-directional
(c) velocity profile same in both forward and reverse directions where applicable.
(d) External torque = constant.
(e) Mass = constant
(f) Velocity constant at steady state.

When we calculate the natural deceleration, we find that it does fall within the acceptable region as illustrated in Figure 12.76.

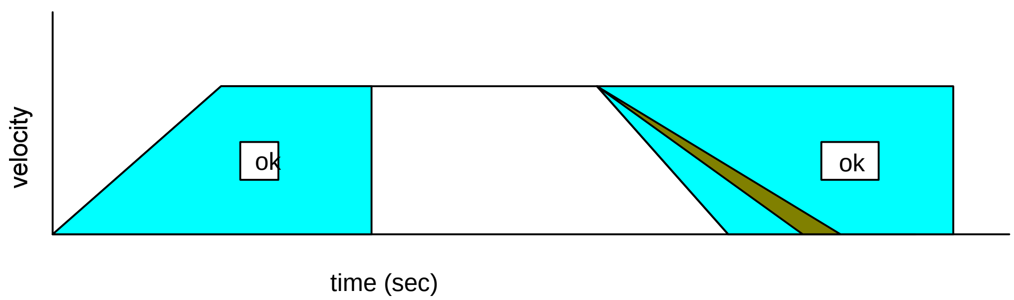


Figure 12.76 Velocity Profile with Constraints
Note: in this example, since the burden is a single value and constant, the natural deceleration would in fact be a single line.

Using our constraints, the $\Delta(PD_m)$ profile could be:

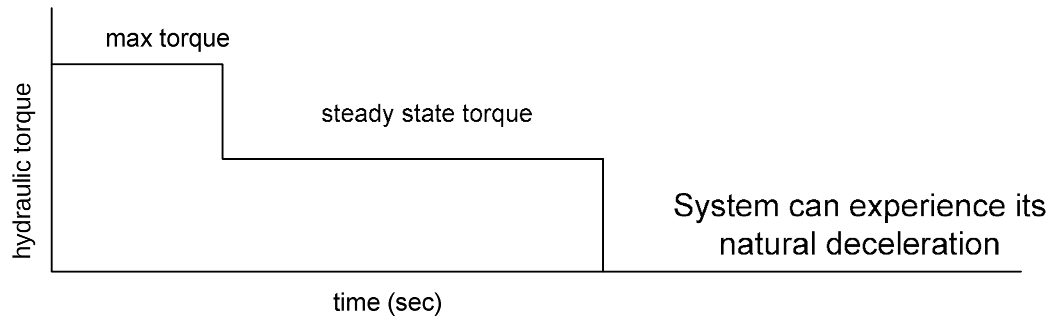


Figure 12.77 Hydraulic Profile

Some thoughts:

- Hydraulic torque is +; therefore we put "stuff" upstream,
- During deceleration, hydraulic torque is zero. We can coast here,
- There is one flow rate: we shall try to match the pump flow to meet the motor requirements (could use gears here to do this).

Note that because natural deceleration does fall within the acceptable range, $\Delta(PD_m) = 0$. we can let the system coast to a full stop. A hydraulic circuit to meet these constraints could be as in the following Figure.

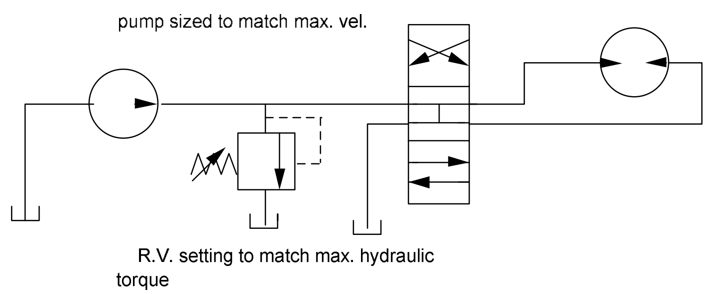


Figure 12.78 Circuit illustrating a natural deceleration in its centre position

This circuit does not lock the motor in the center position. Because we must provide some provision to allow the motor to coast to a stop, we must modify the circuit to provide locking once stopped. This is shown in Figure 12.79.

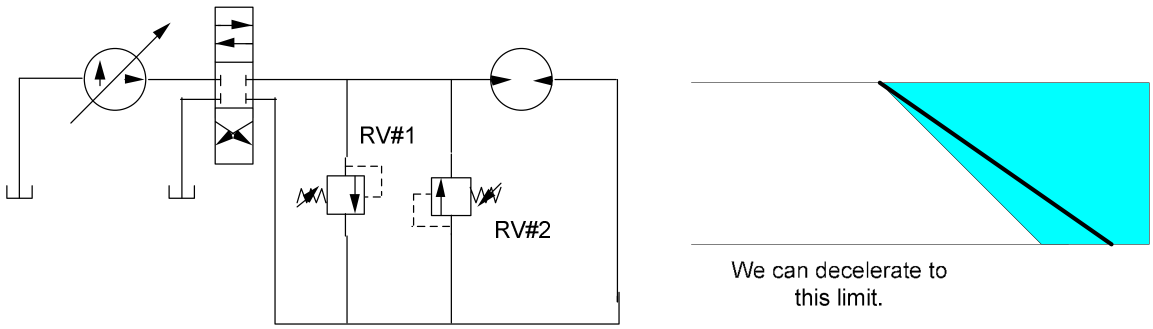


Figure 12.79 A possible circuit to lock and decelerate within acceptable limits
Note: $\Delta(PD_m) \neq 0$ in this circuit during deceleration. This is OK as long as we do not exceed the limit placed on deceleration.

This circuit would work only if the $\Delta(PD_m)$ which exactly meets the deceleration profile is greater than $\Delta(PD_m)$ during acceleration as illustrated in Figure 12.80.

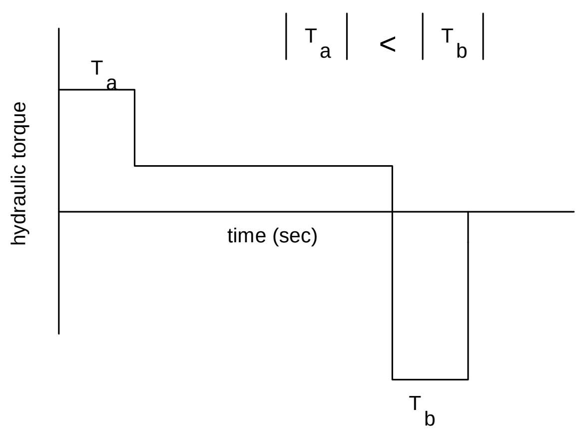


Figure 12.80 Hydraulic Torque Profile
If T_b (via relief valve 1 or 2) is less than T_a (via P.C. pump), the acceleration is limited now by RV #1 or 2

This is highly unlikely so we should look at alternatives such as that in Figure 12.81.

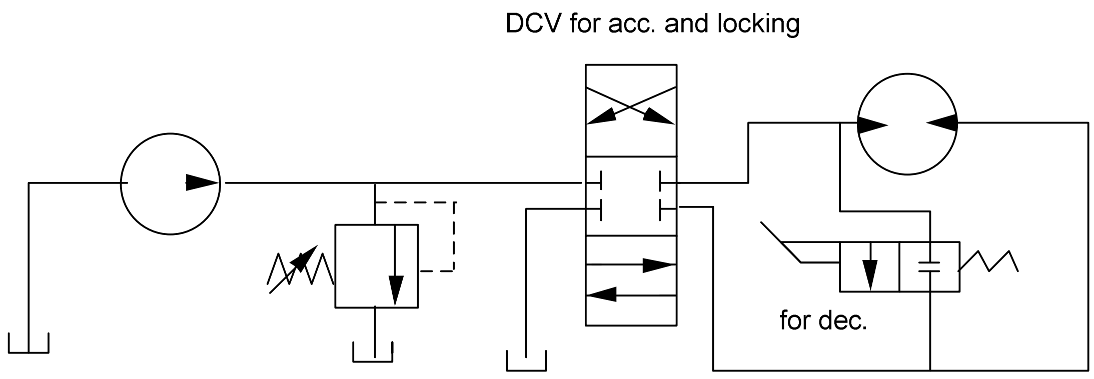


Figure 12.81 Circuit with special valve for deceleration

This circuit is awkward because the directional control valve (DCV) could be accidentally actuated into the locked position before deceleration has occurred.

Another possibility is shown as follows:

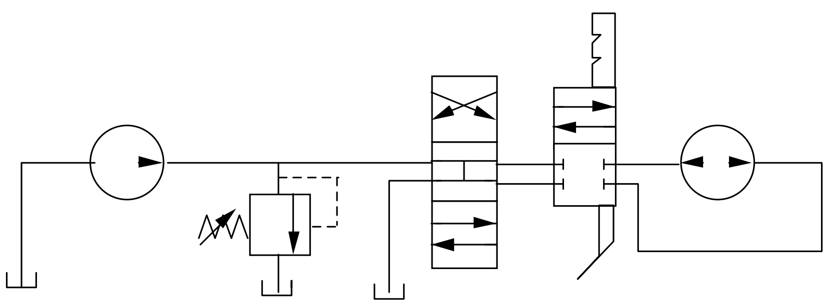


Figure 12.82 Alternate locking and deceleration circuit

This is a more practical circuit because the locking is done deliberately only after deceleration is completed.

Another possibility could be

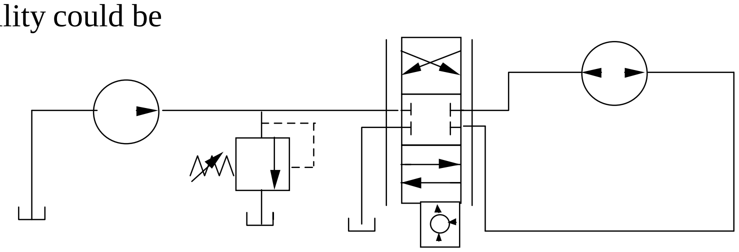
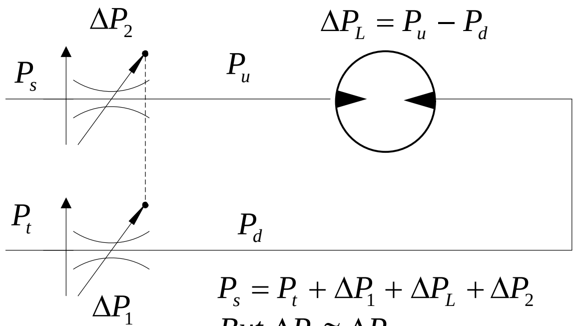


Figure 12.83(a) Proportional valve for locking and deceleration/acceleration

This same circuit can be represented as shown in Figure 12.83(b)



$$P_s = P_t + \Delta P_1 + \Delta P_L + \Delta P_2$$
$$But\ \Delta P_1 \cong \Delta P_2$$
$$P_s = \Delta P_L + 2\Delta P_1$$
$$or\ \Delta P_1 = \frac{P_s - \Delta P_L}{2}$$
$$Q_L = KA\sqrt{\frac{2}{\rho}\Delta P_1}$$

If ΔP_L is not constant, then Q_L is not constant

Figure 12.83(b) Equivalent representation of circuit in Figure 12.83(a)

In this case, a proportional valve is used to accelerate/decelerate and lock the system. This is adequate if the **rate of opening/closing** is regulated to ensure that the velocity limits are not violated. Unless FEEDBACK is used or ΔP_L is approximately constant, this circuit does not have flow control.

We must note that there is a subtle difference between various types of resistive loads. If the resistive load is present even after the system has stopped, we do require some form of locking to prevent the actuator or motor from reversing directions. If, however, the resistive load is a result of friction which in essence disappears when the system is stopped (that is as far as locking is concerned), then locking is **not** necessary when at rest. However, as a precaution, check to see if this is acceptable from a safety point of view.

12.7.1.4 Example 5

Let us consider the same example except that we don't require locking and we want to vary the velocity within the cycle as illustrated in Figure 12.84.

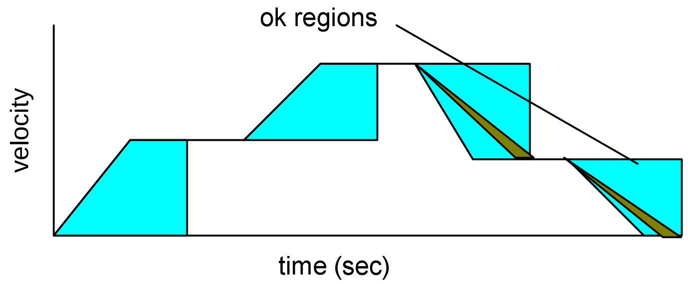


Figure 12.84 Velocity Profile with Acceptable Regions

As before, the natural deceleration is in the acceptable region. Lets us first consider a unidirectional system. A circuit to implement these requirements might be as in Figure 12.85(a).

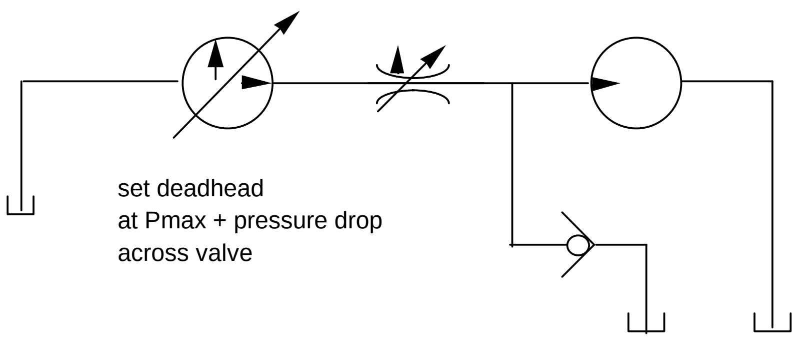


Figure 12.85(a) Circuit with variable flow requirements and acceleration limits

In this case, a CV to tank is provided in case the FCV is closed too fast. If the FCV is opened too quickly, then the pump pressure limits the acceleration and thus the profile is satisfied. We should note here that there will be some losses across the FCV when the Q to the system is less than pump flow. This is a parameter we must consider in sizing a FCV.

Suppose we had only one velocity limit such that flow through the valve was less than the maximum pump flow. Consider the operating characteristics of a pressure compensated pump in Figure 12.85(b).

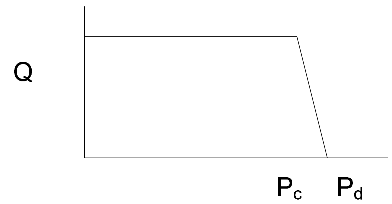
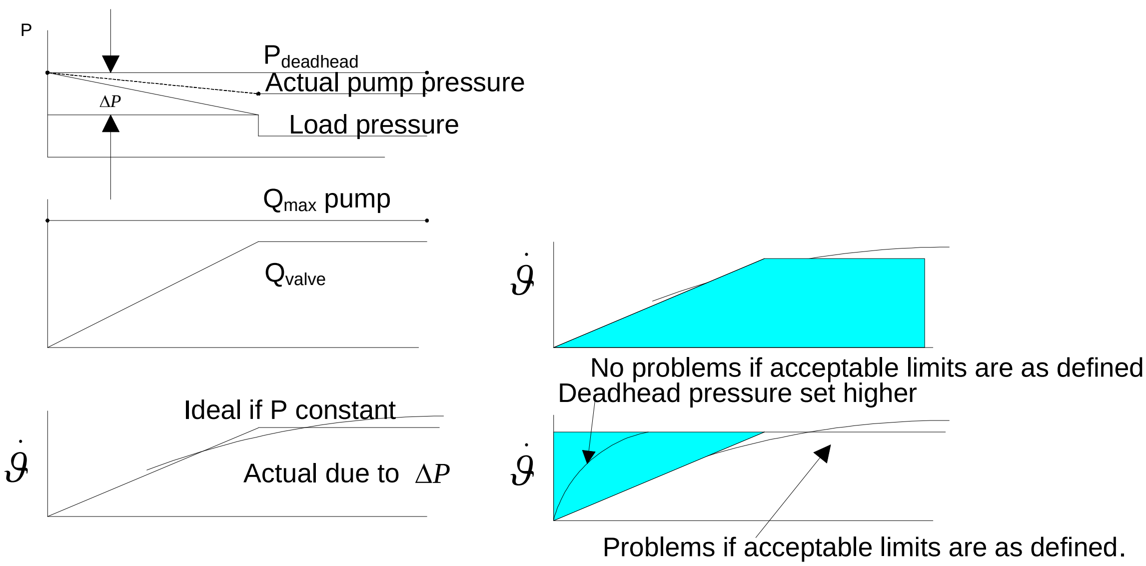


Figure 12.85(b) Pressure flow characteristics of a pressure compensated pump

It is noted that when the swash plate is fully de-stroked (no flow), the pump is at deadhead pressure. As the swash plate is stroked (flow), the pump pressure is less than the deadhead pressure. This is significant because it means that as flow is delivered to the load, the pressure at the pump is not really constant as we so often assume but drops slightly towards the cutoff pressure P_c. What this means that if we use a fixed flow control valve in the circuit of Figure 12.86(a) just to limit flow, we must be aware that the pressure at the pump will drop a bit AND a pressure drop will occur across the flow control valve. (ΔP in Figure 12.85(c)) Thus the actual pressure at the actuator or motor will NOT be at deadhead pressure but at some value which depends on the flow through the valve. This means that if we set the deadhead pressure to set the acceleration limit, then as the actuator or motor starts to accelerate (take on flow), the actual pressure will be less than deadhead and the acceleration will drop off slightly. This is acceptable if the acceleration limit was a maximum, but is not if the limit was a minimum. This effect is illustrated graphically I Figure 12.85(c).



The solution is to set the deadhead pressure higher
To force the velocity curve to lie in the acceptable region

If as in this case, the flow control valve is used to change the velocity of the actuator or motor, then as long as the deadhead pressure is set sufficiently larger than the required acceleration pressure, then the above is not a concern. However, the rate of the opening of the flow control valve must be externally controlled to yield the correct rate. This can increase the complexity and indeed the cost of the circuit as so we should always try to look at using pump pressure to limit acceleration first.

Figure 12.85(c) Actual pressures and velocities for a fixed flow control valve.
Consider the circuit in Figure 12.86

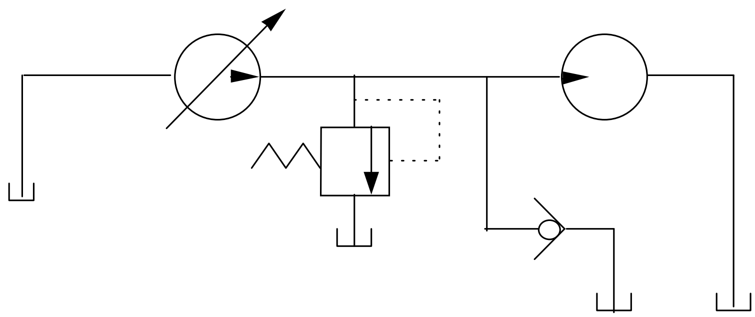


Figure 12.86 Pump controlled flow circuit

This is a very efficient circuit because there are no pressure losses across a FCV as before. The pump provides the flow function. What happens if the application requires a bi-directional motor? We are not concerned about opening the valve too quickly because the relief valve pressure limits the acceleration. However, if we close the valve too quickly, what are the effects? To answer this consider Figure 12.87.

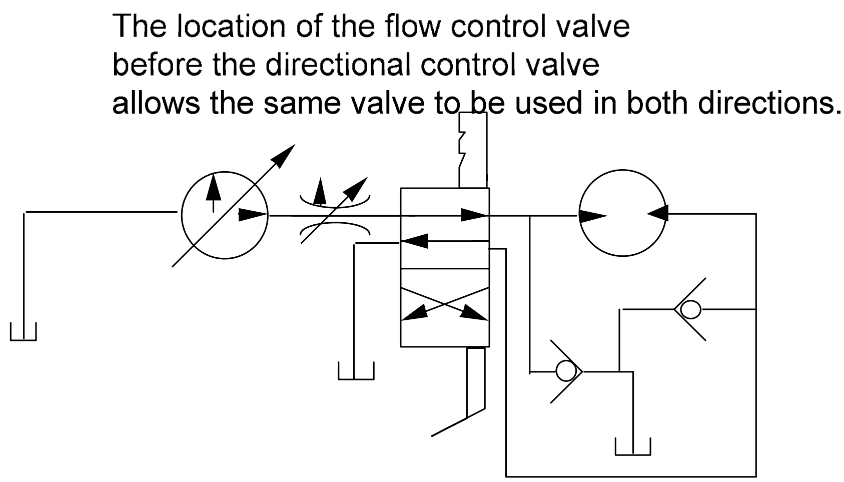
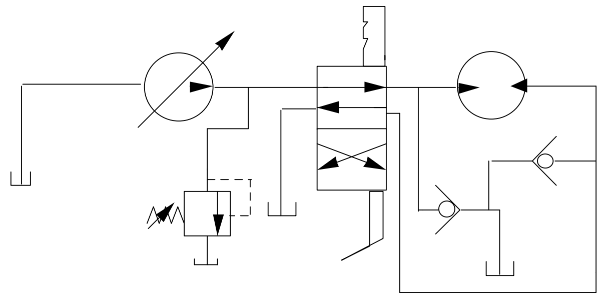


Figure 12.87 Bidirectional variable flow circuit

If we examine this circuit, we can see that rapid closure of the FCV will still satisfy the "coasting" requirements of the velocity profile. A better circuit might be:



Pressure setting limits acceleration
but as due to the characteristics of a “real” relief valve,
the pressure override is higher than the cracking pressure.

Figure 12.88 Alternate circuit

It is interesting to note that this circuit cannot be locked where as in the preceding circuit, one can "semi" lock the motor via the FCV. We say "semi" because it locks only if the external torques (burden) are negative. To completely lock the system, the following circuit could be used.

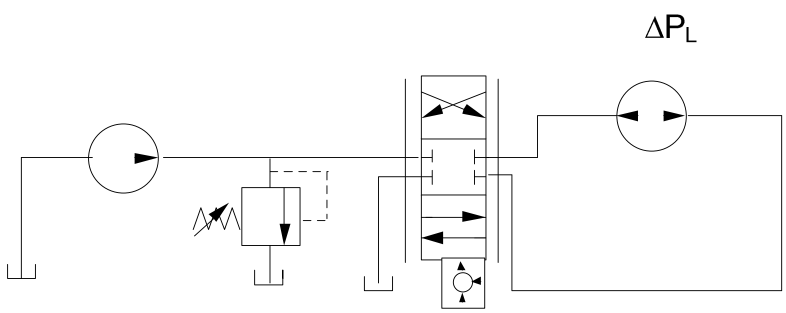


Figure 12.89 Servovalve to lock motor

Note that since ΔP_L is not necessarily constant, this is not a flow control circuit.

In this circuit, the rate of valve opening/closing must be specified to follow the velocity profile exactly.

12.8 Worked Examples

We shall now consider some of the cases we worked through in developing the hydraulic force (torque) profile. We shall follow the design process in its formal form for each case studied.

12.8.1 Example #1.

The system to be controlled is shown in Figure 12.90 (See Section 12.5.1 Example #1).

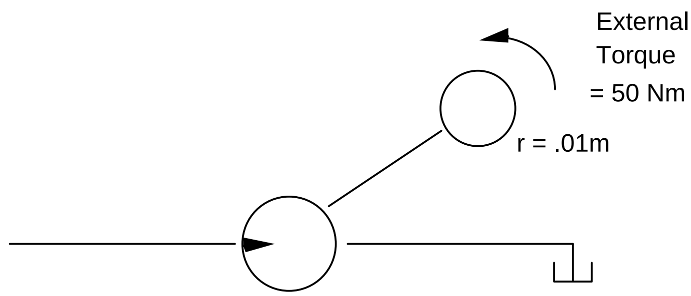


Figure 12.90 System to be Considered

1. Job to be done

This task has been done and the velocity and hydraulic torque profiles are repeated here for reference.

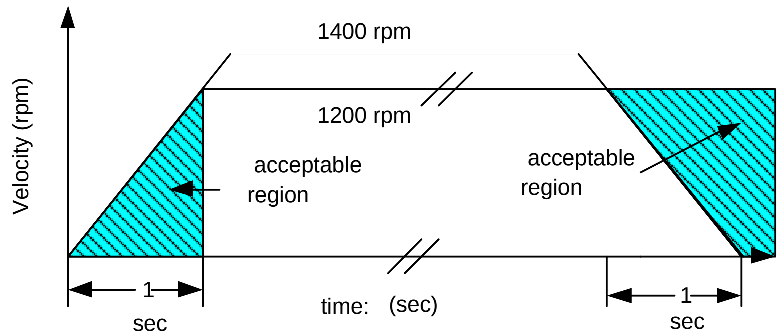


Figure 12.91 Velocity Profile

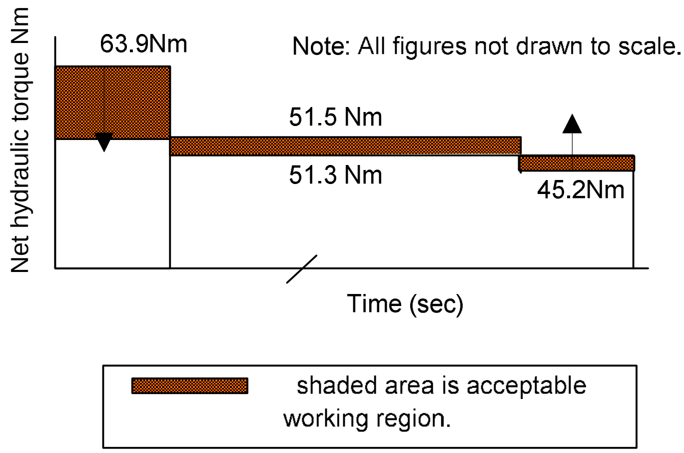


Figure 12.92 Hydraulic Torque Profile

In this problem, the hydraulic force profile is positive [$\Delta(PD_m) > 0$]. This means that we do not have to design in a back pressure on the motor.

2. Choose an Actuator or Motor Size

This is case of choosing a motor size and calculating the pressures and flows. If reasonable, use it; if not, try a new one and repeat. In this case we shall use a motor with a displacement of $6.6*10^{-6}$ m^3/rad.

3. Establish Flow and Pressure Profiles

We are assuming ideal components in this example just for simplicity. We compensate for this by oversizing our pump a small amount. Now, if we choose $D_m = 6.6 \times 10^{-6}$ m^3/rad, then for a motor,

$$\Delta(PD_m) = (P_u - P_D).\, D_m = P_u D_m \text{ (if } P_D \text{ is connected to tank.)}$$

or $$P_u = \frac{\Delta(PD_m)}{D_m}.$$

During acceleration, $$P_{umax} = \frac{63.9 \text{ Nm}}{6.6 \times 10^{-6} \text{ m}^3/\text{rad}}$$

$$= 9.68 \times 10^6 \frac{N}{m^2}$$

$$= \underline{\underline{9.68 \text{ MPa}}}$$

During steady state, P_u $= \dfrac{51.5 \text{ Nm}}{6.6 \times 10^{-6} \text{ m}^3/\text{rad}} = 7.81 \text{ MPa}$

$= \dfrac{51.3 \text{ Nm}}{6.6 \times 10^{-6} \text{ m}^3/\text{rad}} = 7.79 \text{ MPa}$

This is calculated from the burden profiles.

During deceleration, P_{umin} $= \dfrac{45.2 \text{ Nm}}{6.6 \times 10^{-6} \text{ m}^3/\text{rad}} = 6.85 \text{ MPa}$

The max. flow would become

$$Q_L = \dot{\theta} D_m$$

$$= (1400 \frac{\text{rev}}{\text{min}} \times 6.6 \times 10^{-6} \text{ m}^3/\text{rad}) \times 2\pi \frac{\text{rad}}{\text{rev}}$$

$$= 5.8 \times 10^4 \times 10^{-6} \text{ m}^3/\text{min} = .058 \text{ m}^3/\text{min}$$

$$= 58 \text{ l/min (reasonable)}$$

The pressure and flow profiles at the motor become:

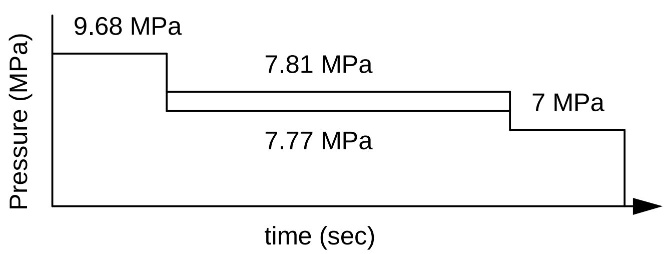


Figure 12.93 Pressure Profile

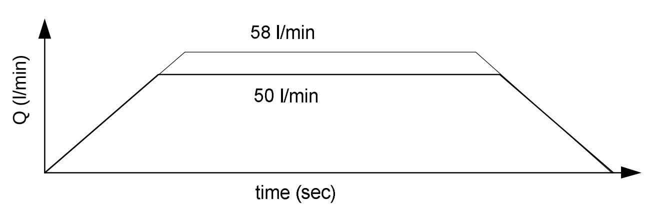


Figure 12.94 Flow Profile

It is most important to realize that if we can recreate these profiles hydraulically, then the velocity profile during acceleration and deceleration will be satisfied. In this circuit we use pressure to limit acceleration and deceleration but we must use some flow limiting device to limit the steady state velocity.

4. Check basic HP to satisfy the hydraulic profiles

Based on the pressure and flow profiles, the HP profile can be established as:

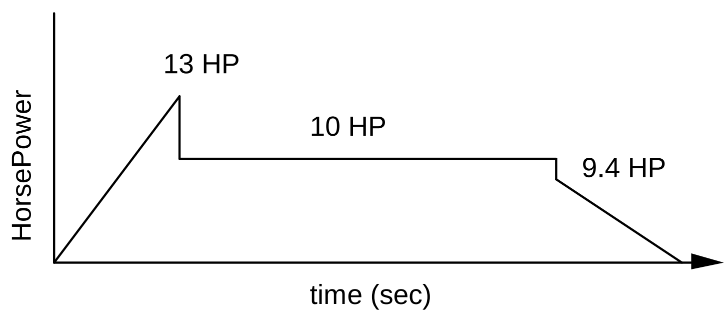


Figure 12.95 HorsePower Profile (Maximum conditions)

To illustrate a sample calculation, consider:

Power (kw)	=	P (bar)* Q (l/min)/600
	=	(9.68MPa / (.1MPa/bar)) *60 l/m /(600)
	=	9.68 kw
Power (HP)	=	9.68 kw* 1HP/ 745.7w
	=	12 HP

Note; this is the required HP at the motor which is easily found from the product of torque and angular velocity (we had to establish the torque and angular velocity right at the beginning).

5. Design Circuit

In this case our pressure profile indicated that a downstream pressure is not necessary. We can use the pump pressure to accelerate the system. To decelerate, we must drop the pressure just slightly to less than 7.75MPa but more than 7 MPa. Note: We have a situation in which it is possible for the system to actually move backwards if the deceleration pressure is not set right (due to presence of an external torque which stays even at zero velocity). **It should be noted here that if the resistive torque was due to friction, this would not be a problem since the torque cannot "reverse" motion**. A circuit to satisfy the hydraulic torque profile is as follows:

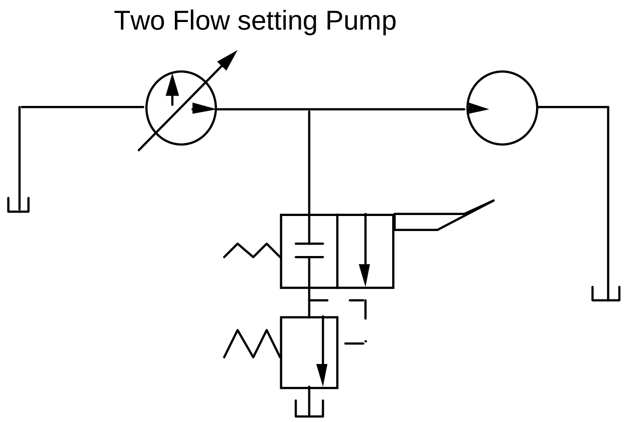


Figure 12.96(a) Circuit possibility #1
Note: the two settings of the pump to achieve the steady state angular velocity

We show here a two flow setting pressure compensated pump. If we choose to use a flow control valve with fixed settings (set between cycles), we must account for pressure losses across the valve in the pump pressure calculations and hence in the pump HP calculations. This circuit does not lock the motor. Indeed, it is would be very hard to prevent the motor from reversing directions unless either the pump is switched off (this is still dangerous because the pump could reverse act as a motor) or the external torque is exactly balanced by the upstream pressure (rather doubtful). If shutting off the pump is feasible (which means actually locking the shaft somehow), then this circuit is acceptable and is in fact a simple solution. If shutting off the pump is not convenient or possible, then an alternate solution must be found.

A very simple solution would be to use a check valve in the circuit which would prevent back flow as illustrated in Figure 12.96(b). This is a simple solution in this case.

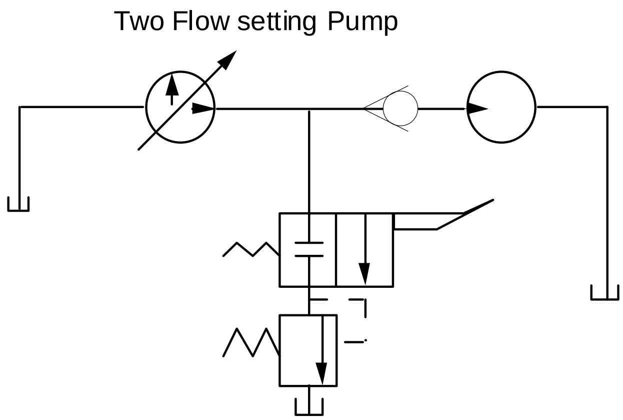


Figure 12.96(b) Circuit with check valve to prevent reverse flow in the presence of external torque

A second alternative is:

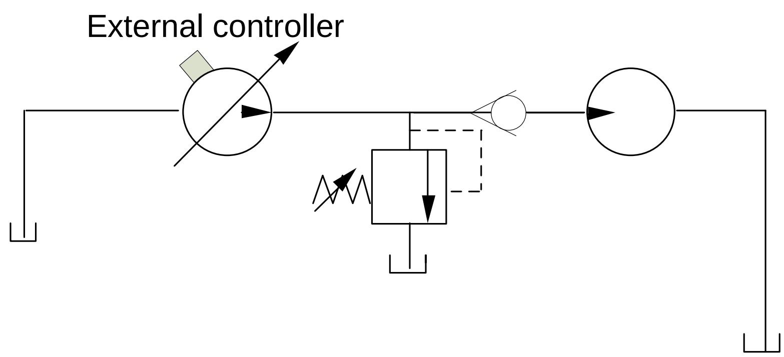


Figure 12.97 Circuit possibility #2

In this circuit, we use a variable displacement pump and vary the flow rate to follow the desired flow rate. This would require some external electronic controller and interfacing device (such as a Honeywell controller). Manual operation may be possible if the rate at which the swash plate can be moved is mechanically limited to ensure that the limits so specified on the flow profile are not violated We could use the following circuit as well:

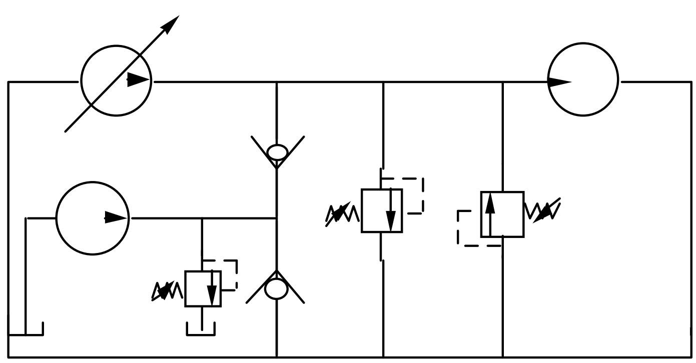


Figure 12.98 Circuit Possibility #3

This essentially is the same circuit as in Figure 12.97 except that it is a closed circuit and reversible. Both of these last circuits can be semi-locked locked by destroking the pump to its center position. (We say semi-locked because if the burden was continuous and due to an external torque and generated a pressure larger than the relief valve setting , then it could be possible to force the motor backwards.)

Another circuit which could be used is:

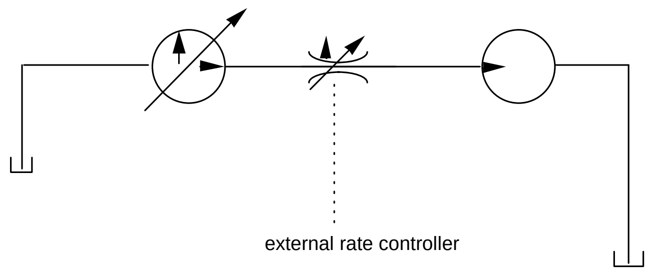


Figure 12.99 Circuit Possibility #4 (This circuit is only valid if $\Delta(PD_m) > 0$

This circuit performs the same way as the last circuits with the same restriction on the flow rate profile.

<u>6. Plot pressure, flow and HorsePower plots at important parts of the circuit.</u>

Let us consider the original circuit. The pressure profile of the motor is the same as the pump. Since a fixed displacement pump is used, the difference between the motor flow and the pump flow represents the flow across the relief valve. As such the HP profile across the RV is :

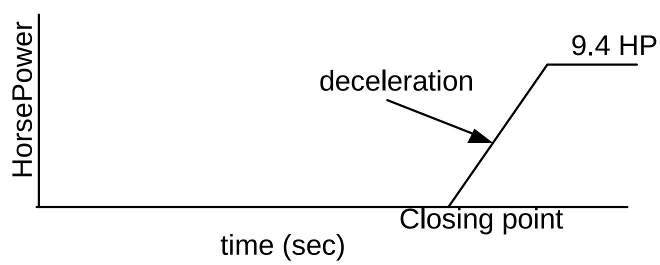


Figure 12.100 HorsePower Losses across the relief valve Plot

Our major losses would occur when the motor is being decelerated. This would cease when the pump is shut down. If the deceleration period is a small period compared to the rest of the circuit, we can neglect it .

7. Component selection

Choose a motor with a displacement of 6.6 x 10^{-6} m^3/rad.The flow should have a flow rating of 60 l/m at a maximum pressure of 14 MPa. Pressurized lines should be 3/4" and inlet lines at 2 " etc.

We could also do an efficiency calculation by looking at the output horsepower (from the hydraulic torque profile) and the input pump horsepower :

$$outputHP = T_m \dot{\theta}_m$$

$$inputHP = \frac{P_{pump} Q_{pump}}{\eta_p} \text{ where } \eta_p \text{ is the pump overall efficiency}$$

$$\text{and } \eta = \text{overall circuit efficiency} = \frac{T_m \dot{\theta}_m}{\frac{P_{pump} Q_{pump}}{\eta_p}}$$

12.8.2 Example #2

The system to be controlled is shown in Figure 12.101 (See Section 12.5.2, Example #2)

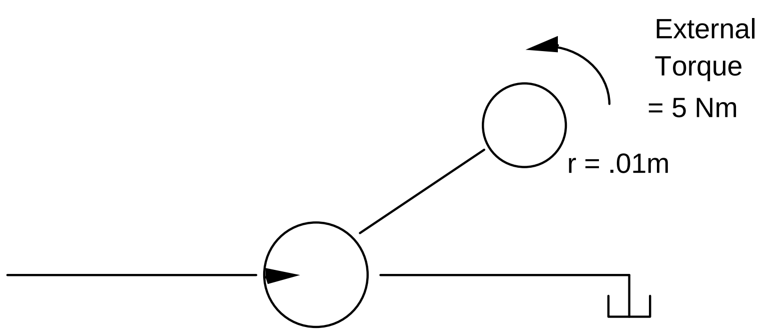


Figure 12.101 System to be considered

1. Job to be done

This task has been done and the velocity and hydraulic torque profiles are repeated here for reference.

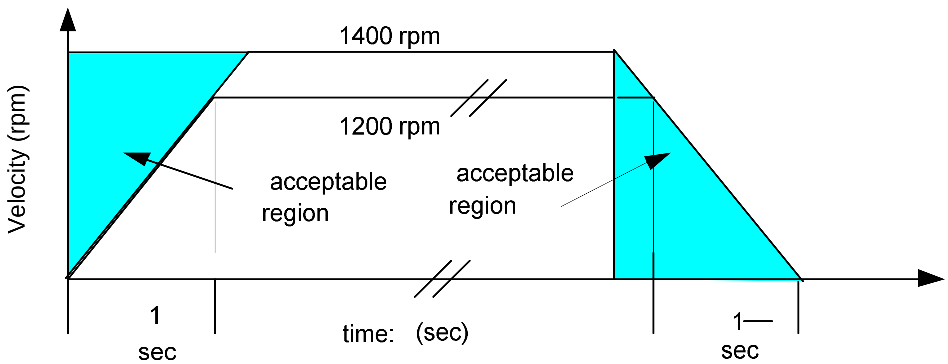
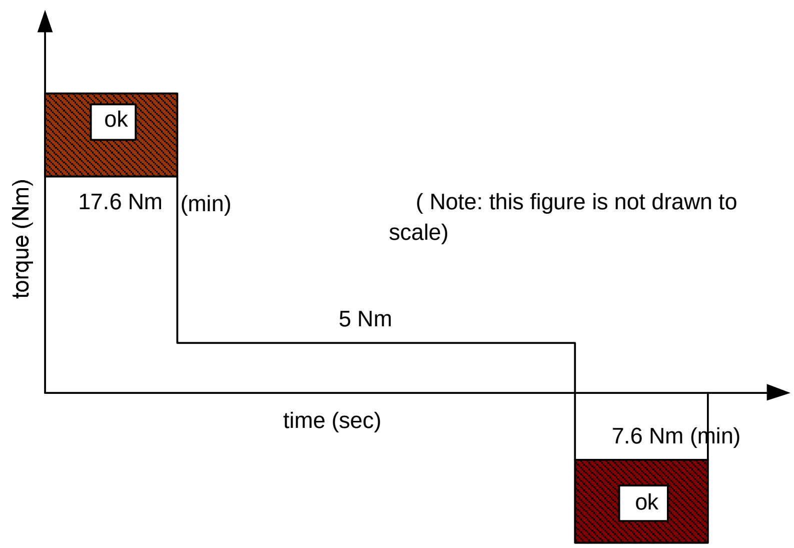


Figure 12.102 Velocity Profile



Note; that the negative hydraulic torque profile means that "stuff" has to be placed downstream; hence pressure as a means of limiting things becomes much more difficult.

Figure 12.103 Hydraulic Torque Profile

2. Choose an Actuator or Motor Size

This is the case of choosing a motor size and calculating the pressures and flows. If reasonable, use it; if not, try a new one and repeat. In this case we shall use a motor with a displacement of $6.6*10^{-6}$ m^3/rad.

3. Establish Flow and Pressure Profiles

We are assuming ideal components in this example just for simplicity. We compensate for this by oversizing our pump a small amount. In this problem, the hydraulic force profile is negative [$\Delta(PD_m) < 0$] for part of the cycle (deceleration). This means that a back pressure must be designed into the circuit. If we choose $D_m = 6.6 \times 10^{-6}$ m^3/rad, then for a motor,

$$\Delta(PD_m) = (P_u - P_D)D_m$$

We have $\Delta(PD_m)$; therefore we must first calculate the back pressure which is found from:

$$P_d = \frac{\Delta(PD_m)}{D_m} = 7.6\text{Nm}/6.6 \times 10^{-6}\ \text{m}^3/\text{rad} = 1.15\ \text{MPa*}.$$

* It must be pointed out that this assumes that $P_u = 0$ during this part of the cycle. (Note: Because during deceleration, the flow to the motor is less than that from the pump. Thus if we do not do something, the upstream pressure would be at deadhead or the relief valve setting which means we would have to make the down stream pressure very large to ensure that the pressure drop across the motor is 1.47 MPa. Alternately, we could simply design our circuit such that $P_u = 0$ and then P_d would be 1.47 MPa. Note: this might be a problem circuit wise!)

During steady state, $P_u = 5\text{Nm}/6.6 \times 10^{-6}\ \text{m}^3/\text{rad} = .76\text{MPa}$ (assuming $P_d = 0$).

During acceleration $P_u = 17.6\ \text{Nm}/6.6 \times 10^{-6}\ \text{m}^3/\text{rad} = 2.67\ \text{MPa}$ (assuming $P_d = 0$).

One possible pressure profile would be:

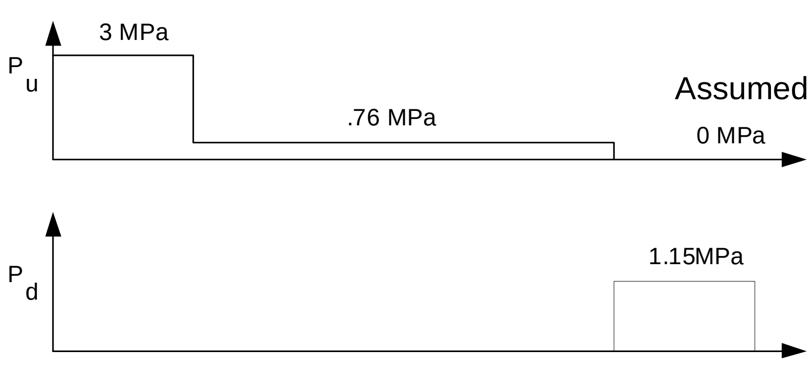


Figure 12.104 Pressure Profile #1 (This assumes that P_u can be made zero during deceleration)

If we decided to leave the back pressure in at all times, the pressure profile becomes:

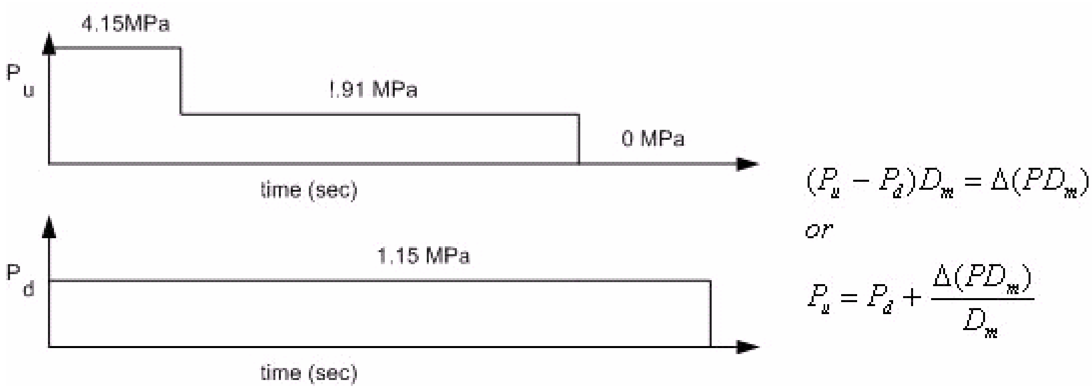


Figure 12.105   Pressure Profile #2

These two different profiles will result in different hydraulic circuits as will be examined shortly.

The max. flow would become

$$Q_L = \dot{\theta} D_m = 1400 \frac{\text{rev}}{\text{min}} \text{ x } 6.6 \text{ x } 10^{-6} \text{ m}^3\text{/rad) x } 2\pi \frac{\text{rad}}{\text{rev}}$$

$$= 5.8 \text{ x } 10^4 \text{ x } 10^{-6} \text{ m}^3\text{/min} = .058 \text{ m}^3\text{/min}$$

$$= 58 \text{ l/min (reasonable)}$$

and the flow profile becomes:

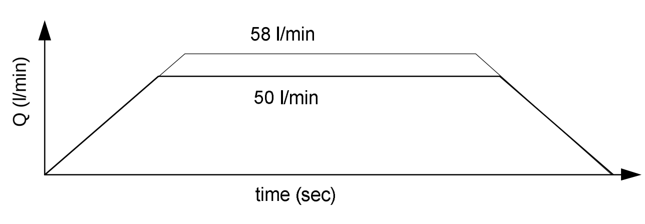


Figure 12.106   Flow Profile

4. Check basic HP to satisfy the hydraulic profiles

Regardless of whether we put a back pressure in for part of the cycle or throughout, the horsepower requirements are the same since the $\Delta(PD_m)$ profile is independent of the downstream pressure. However, the horsepower is quite different as seen by the pump. Using the pressure and flow plots for the two different cases, we get HorsePower plots of:

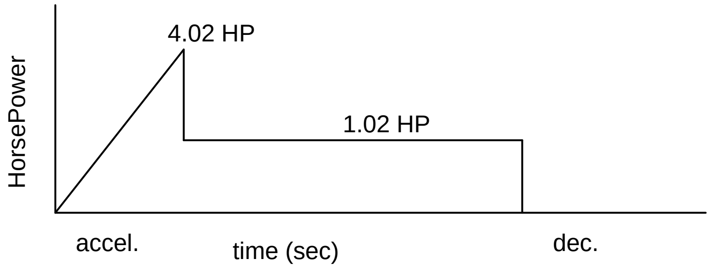


Figure 12.107 Pump HorsePower Plot for circuit #1 (maximum conditions)

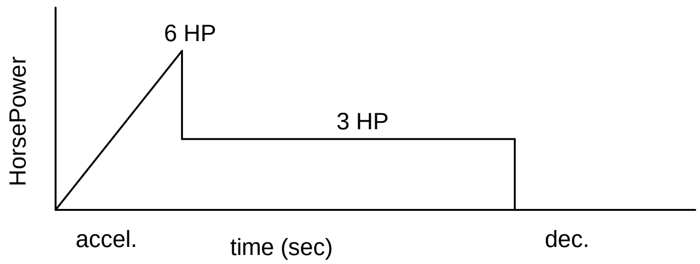


Figure 12.108 Pump HorsePower Plot for circuit #2 (maximum conditions)

The increase in the HP requirements of the second circuit is evident from a comparison of the two plots and hence must be accounted for in our design.

5. Design Circuit

Let us first consider the case illustrated in Figure 12.104. To configure a circuit based on this profile, we must port the pump to tank as we switch in the back pressure. One such circuit could be:

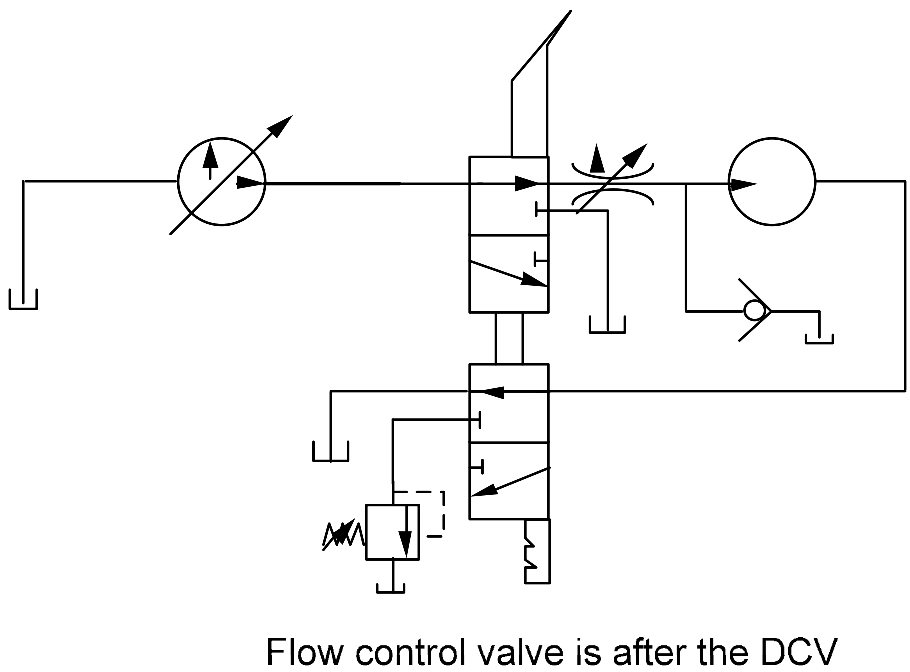


Figure 12.109 Circuit Possibility #1
Note; in this circuit, the return line of the directional control valve must be able to withstand the pressure set by the counterbalance valve.

The flow control valve is set before the cycle to the desired flow rate. There is a pressure drop across the valve which should be included when one considers the HorsePower losses. However, these are minor compared to the losses associated with the counterbalance valves.

In this circuit, two DCV's are "daisy chained" together. This system essentially locks the motor in that the upstream side of the motor is blocked via the DCV and the downstream side blocked via the CBV.

If we wish to design a circuit for the second case illustrated in Figure 12.106. We can put a CBV directly into the circuit as illustrated in Figure 12.110.

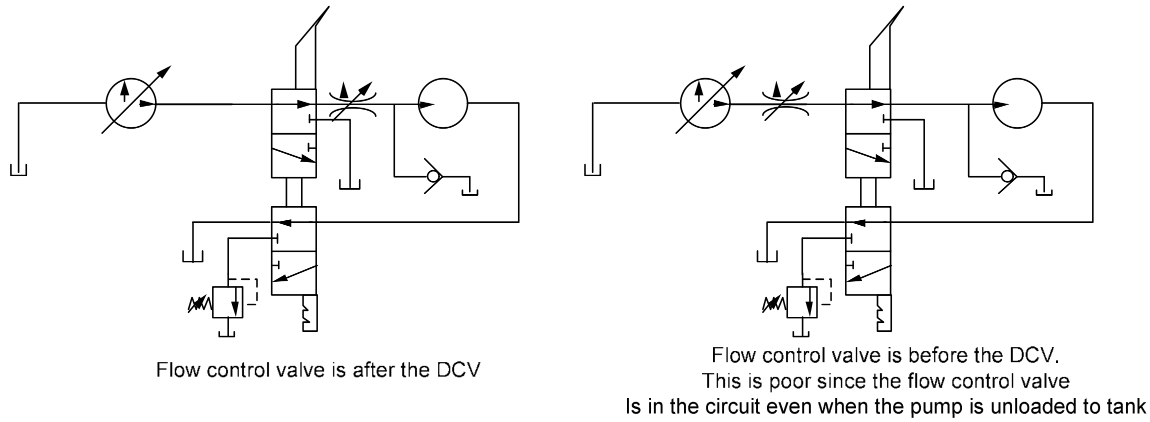


Figure 12.110(a) Circuit Possibility #2

This circuit has higher horsepower requirements but results in a much simpler circuit. This circuit also locks the motor if so required.

Another option might be as indicated in Figure 12.110(b). A pilot operated CBV is used but P_u must be set to 0 during deceleration.

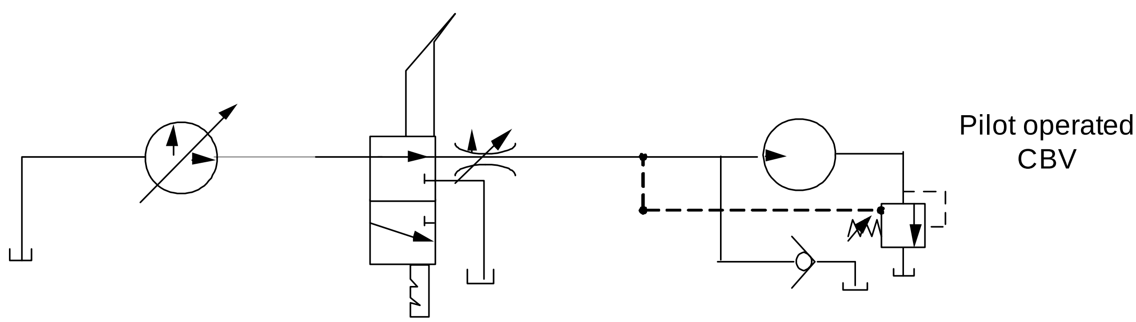


Figure 12.110(b) Circuit Possibility #3

6. Plot pressure, flow and HorsePower plots at important parts of the circuit.

Let us consider the first circuit. The only horsepower losses which occur are those due to the CBV (and some minor losses across the DCV and the FCV). These losses only occur during deceleration as illustrated:

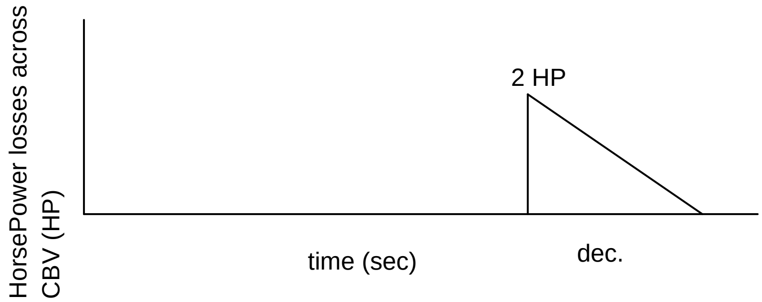


Figure 12.111 HorsePower Losses

In the second case, our horsepower losses are always present due to the CBV. The HorsePower **loss** profile would thus become:

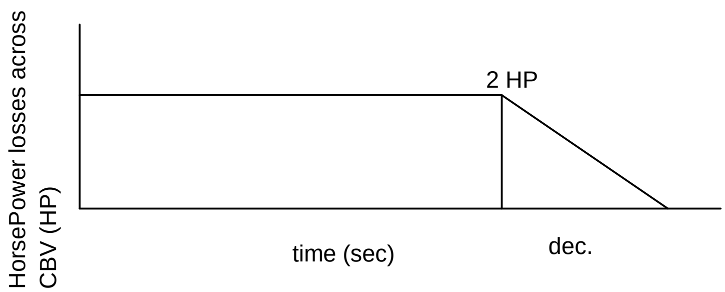


Figure 12.112 Horsepower Losses

As in the last example, we could also do an efficiency calculation by looking at the output horsepower (from the hydraulic torque profile) and the input pump horsepower :

$$outputHP = T_m \dot{\theta}_m$$

$$inputHP = \frac{P_{pump} \; Q_{pump}}{\eta_p} \text{ where } \eta_p \text{ is the pump overall efficiency}$$

$$\text{and } \eta = \text{overall circuit efficiency} = \frac{T_m \dot{\theta}_m}{\frac{P_{pump} \; Q_{pump}}{\eta_p}}$$

7. Component selection

If we go with the first circuit, we would choose a pump with a 60 lpm capacity and maximum pressure of 7 MPa. We would choose an electric motor rated at 4.5 HP (slightly larger than that specified to account for losses), etc...

12.8.3 Example #3

The system to be controlled is shown in Figure 12.113 (See section 12.5.3, Example #3)

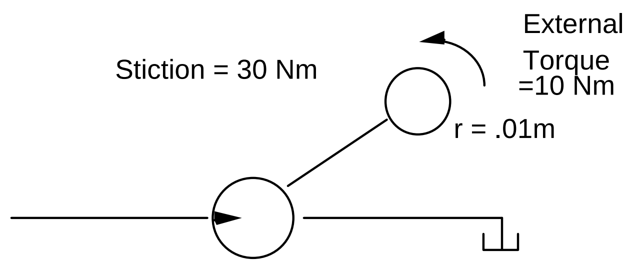


Figure 12.113 System to be Considered

1. Job to be done

This task has been done and the velocity and hydraulic torque profiles are repeated here for reference.

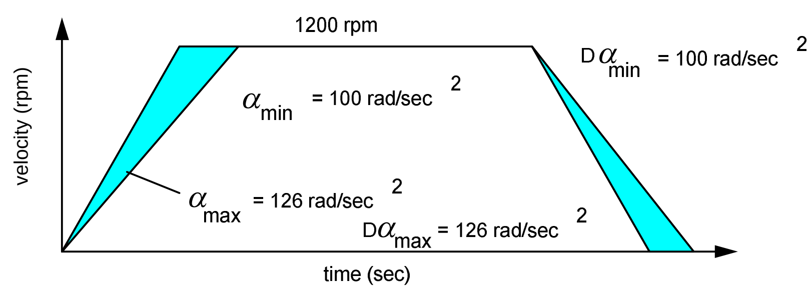


Figure 12.114 Velocity Profile

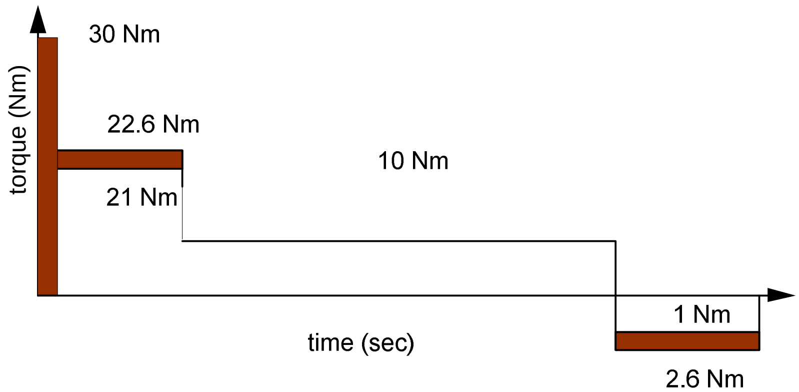


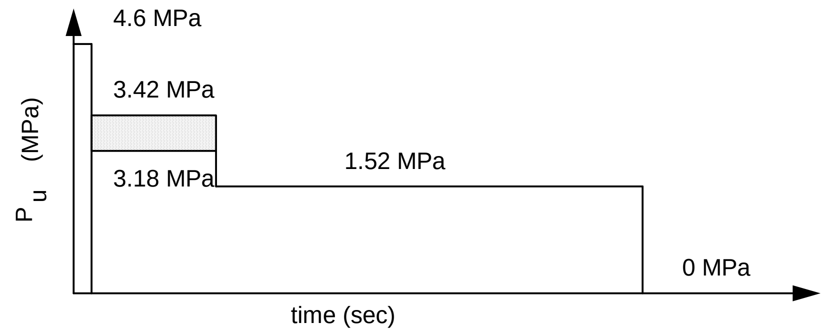
Figure 12.115 Hydraulic Torque Profile

2. Choose an Actuator or Motor Size

This is the case of choosing a motor size and calculating the pressures and flows. If reasonable, use it; if not, try a new one and repeat. In this case we shall use a motor with a displacement of $6.6*10^{-6}$ m^3/rad.

3. Establish Flow and Pressure Profiles

We are assuming ideal components in this example just for simplicity. We compensate for this by oversizing our pump a small amount. In this example, the total hydraulic profile is both positive and negative throughout the cycle. A back pressure must be applied to the motor during deceleration. or indeed, if so desired, through out the whole cycle. Thus two pressure profiles are possible as indicated in Figures 12.116 and 12.117.



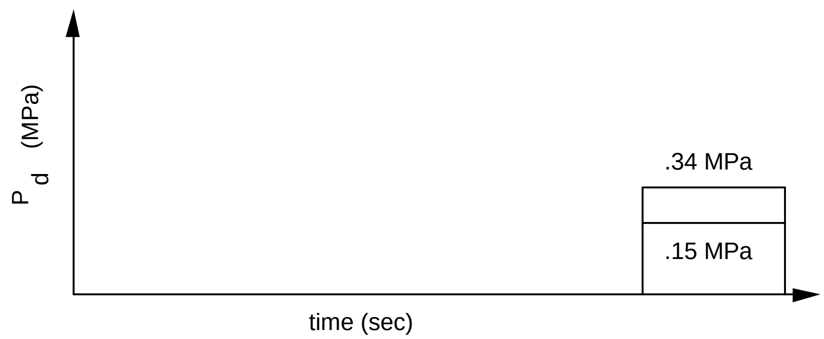
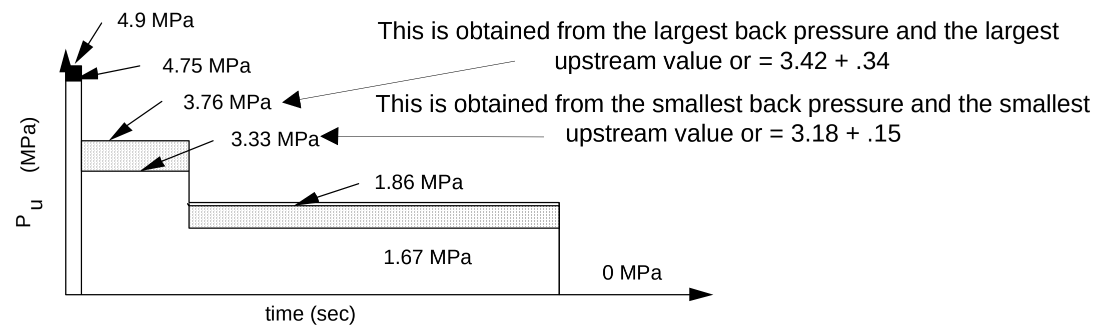


Figure 12.116 Pressure Profile #1



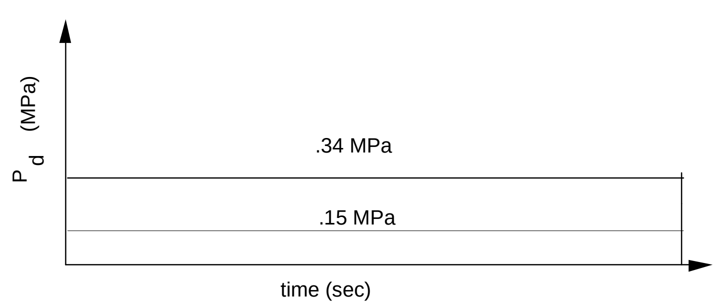


Figure 12.117 Pressure Profile #2

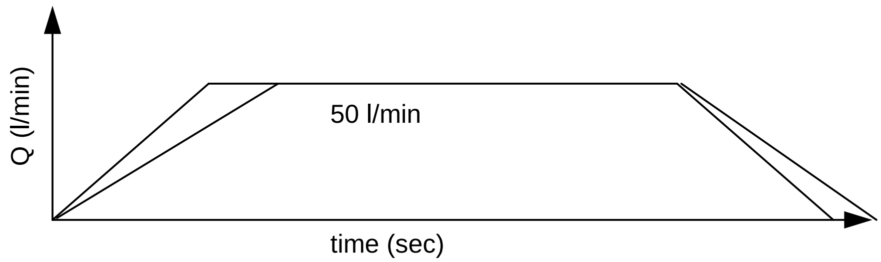


Figure 12.118 Velocity Profile

It can be observed that there are several problems which must be considered when configuring the hydraulic circuit. In the second set of profiles, there are essentially four sets of limits during acceleration which are dependent upon what back pressure is chosen (We have just shown the maximum and minimum limits in this figure) A second problem associated with stiction must also be addressed.

4. Check basic HP to satisfy the hydraulic profiles

Using the pressure and the flow profiles for the two possibilities discussed above, we have two HorsePower plots which reflect these two cases. It is noticed that this is very similar to the last case where the presence of a back pressure just increases the pump horsepower requirements but not that of the motor.

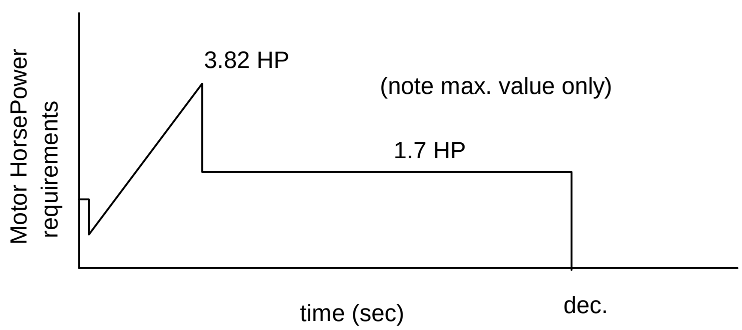


Figure 12.119 HorsePower Plot for #1

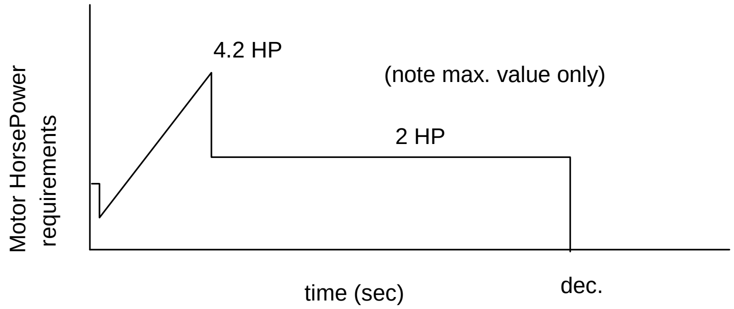


Figure 12.120 HorsePower Plot for #2

5. Design Circuit

If we don't use back pressure except to decelerate the motor, then we could use exactly the same circuit as the previous case in which two DCV are "daisy chained" **if stiction was not present.** We would have to ensure that the set pressure at the pump and the CBV fell within the limits. If we set the pump pressure high enough to overcome stiction, then we cannot satisfy the acceleration limits. Switching in a high pressure source for part of the cycle is possible but it certainly adds an extra degree of complexity to the circuit. An example of such a circuit is:

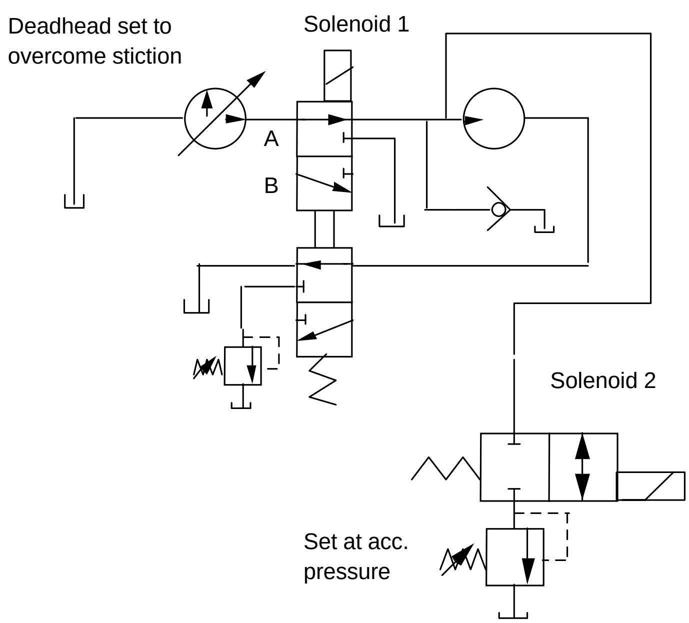


Figure 12.121 Circuit Possibility #1 This circuit could run into acceleration problems if P_d = .34 MPa

The circuit requires that after solenoid 1 is engaged into position A. Solenoid 2 engages but only after a short time delay (sufficient to overcome stiction). Once engaged it sets the pressure to accelerate the system within its limits.

This is a very poor circuit to try to make work for all conditions. It certainly would be considered a worst case scenario circuit. However, it is very useful to follow through the design process to see how constraints can make a problem very hard to design for using just pressure to achieve acceleration and deceleration limits. We should look for other alternatives which might mean we have to compromise on the constraints

.

We could also implement the circuit as follows.

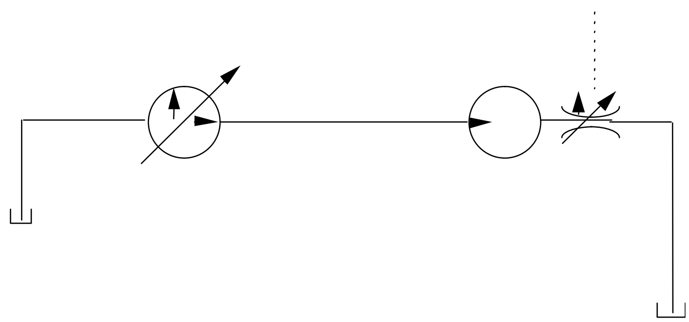


Figure 12.122 Circuit Possibility #2

In this configuration, the FCV is made to follow the flow profile manually (very doubtful) or electronically (most probable). It is interesting to note that by controlling the flow, the hydraulic torque profile will follow exactly that of Figure 12.129 (within the specified limits). We must ensure that the upstream pressure is higher than 5 MPa to guarantee that the differential pressure across the motor satisfies the hydraulic torque profile.

If we decide to use the option of having a constant pressure downstream throughout the cycle, as before, our horsepower requirements increase but in addition, we have a problem with deciding what limits can we use to set the pressures at (i.e., do we use P_{dmin} and P_{umax}, P_{dmin} and P_{umin}, P_{dmax} and P_{umax}, P_{dmax} and P_{umin} to define our limits?). This option gets difficult to implement and perhaps should not be considered.

NOTE: Why can we not use the following circuit to meet the constraints?

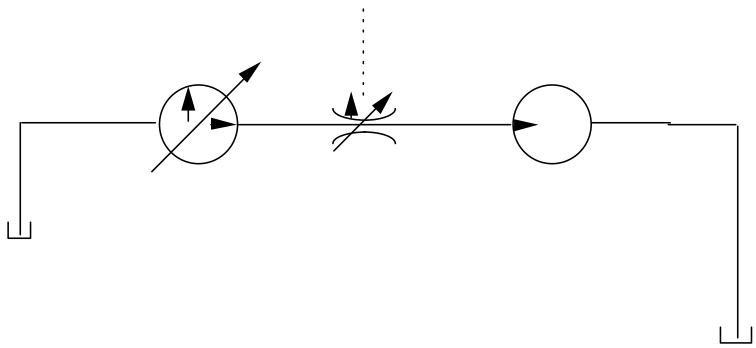


Figure 12.123(a) Circuit Possibility #3

Answer: Because we are using a FCV **to do the acc/dec.**, our design criteria specifically states that we must use a meter out circuit (because $\Delta(PD_m)$ <0.) We could however use the following circuit.

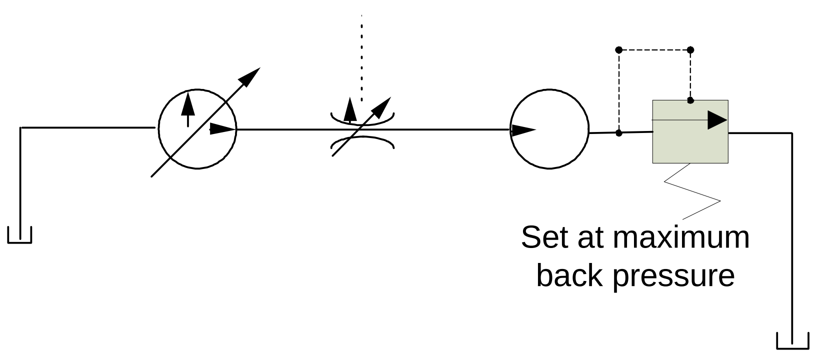


Figure 12.123(b) Alternate configuration

<u>6. Plot pressure, flow and HorsePower plots at important parts of the circuit.</u>

Considering our first circuit, we have horsepower losses at two parts of the cycle due to the RV (which limits acceleration) and the CBV (which limits deceleration). Our HorsePower plots would be:

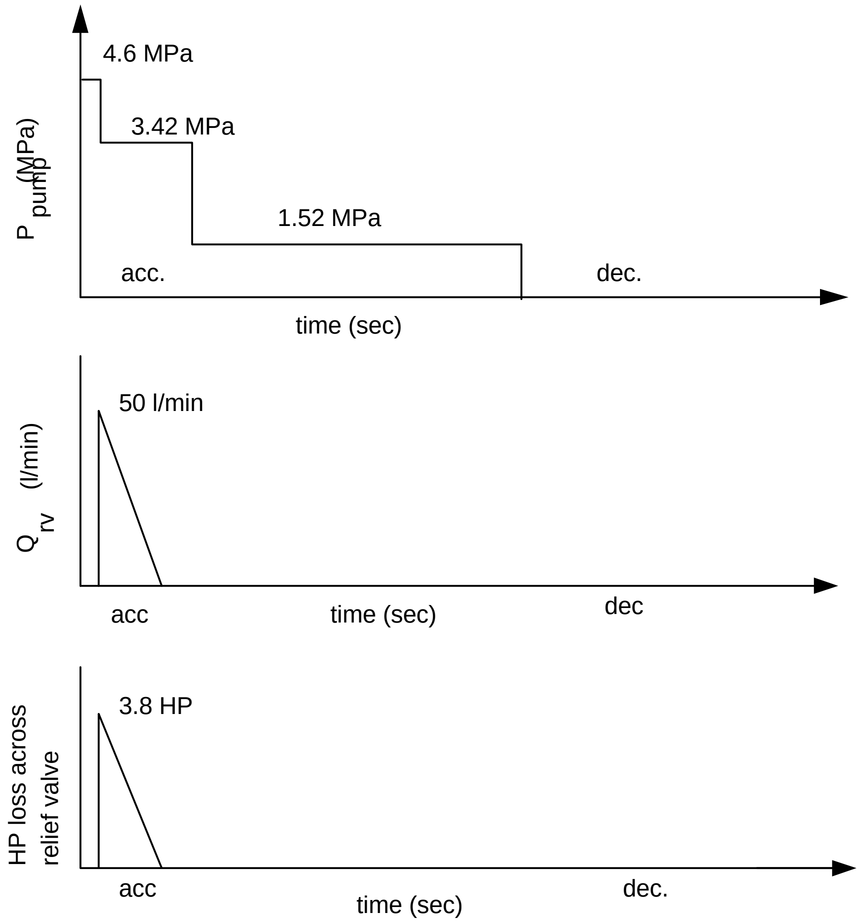


Figure 12.124 P, Q and HP Plots for #1

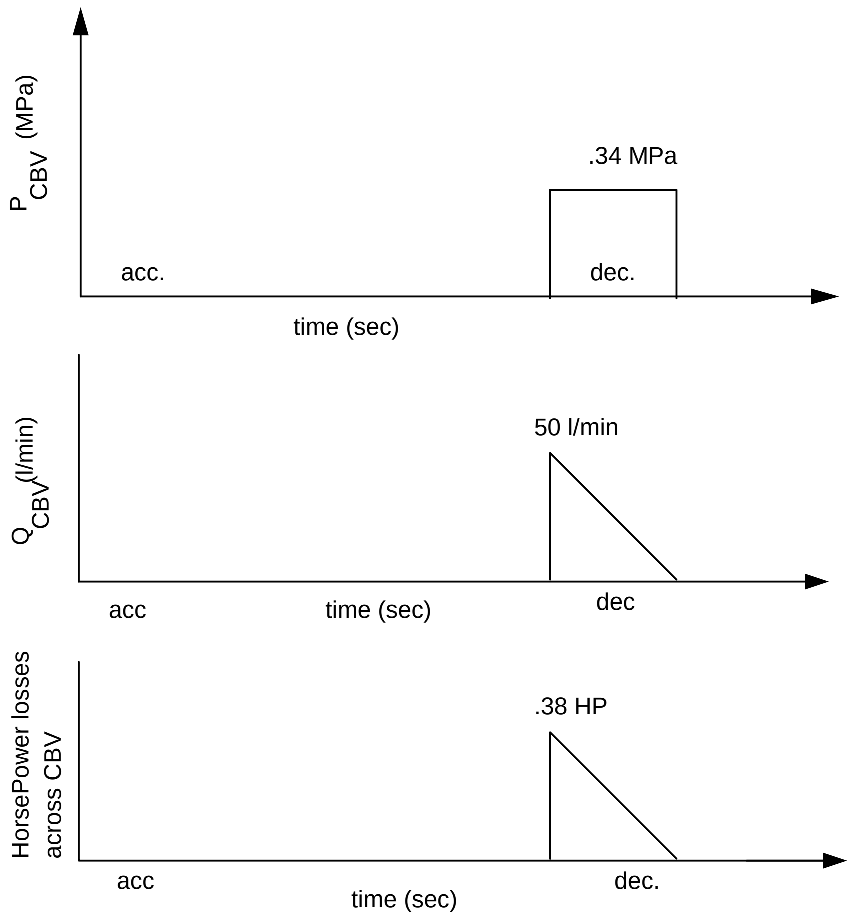


Figure 12.125 P, Q and HP Plots for #2

7. Component selection

We would choose a pump with a 50 lpm capacity and maximum pressure of 7 MPa. We would choose an electric motor rated at 4.5 HP (slightly larger than that specified to account for losses), etc...

12.8 Other worked examples

12.8.1 Position and time specified constraints

In this example, we shall consider a case in which we reflect the full design process. We shall also show how compromise is sometimes necessary to reduce complexity in the configured circuit. We must point out that there are many possible ways to configure circuits. We will present alternatives where possible but we must realize that other circuit configurations can do the same job.

In this situation consider the linear system illustrated in Figure 12.126.

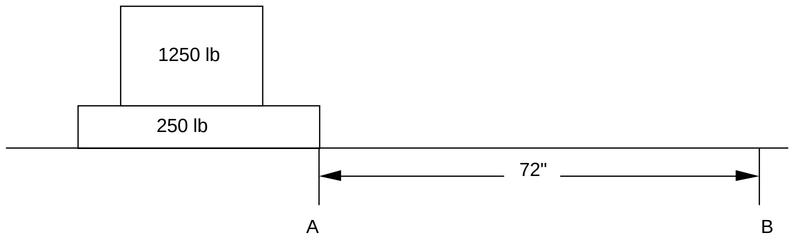


Figure 12.126 Problem to be considered

The purpose is to move the pallet and weight from A to B, unload the weight and then return the empty pallet. The information we get from the client is as follows:

(a) total time for complete cycle (advance and return) is 30 sec.
(b) advance cycle time 15 secs
(c) unload cycle time 5 secs
(d) return cycle time 10 secs
(e) total stroke is 72 inches.
(f) static friction = .25
(g) dynamic friction = .15

1. Job to be done - understand the load

In this example. we have a stroke limitation of 72 inches. Hence we must consider both position and velocity as a function of time but we have **not placed any limitations** on acceleration or deceleration. It is assumed that the mass will be constant and that the speed does not have to be changed from cycle to cycle or indeed, within a particular cycle. ***Despite the fact that acceleration/deceleration is assumed not to be critical (is it really???), we should still follow our basic procedure that we outlined earlier***.

<u>Calculate the Mass Profile.</u>

The mass is constant throughout the cycle but is different in the forward and reverse directions as illustrated in Figure 12.127

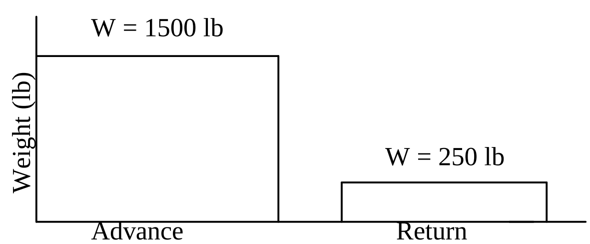


Figure 12.127 Mass profile

<u>Consider the Burden Profile</u>

Because we do not know the velocity profile as yet, we must first establish this. We do not know V but we do know the total stroke and time constraints. Hence we must use both pieces of information to establish V. Let us assume a velocity profile as illustrated in Figure 12.128.

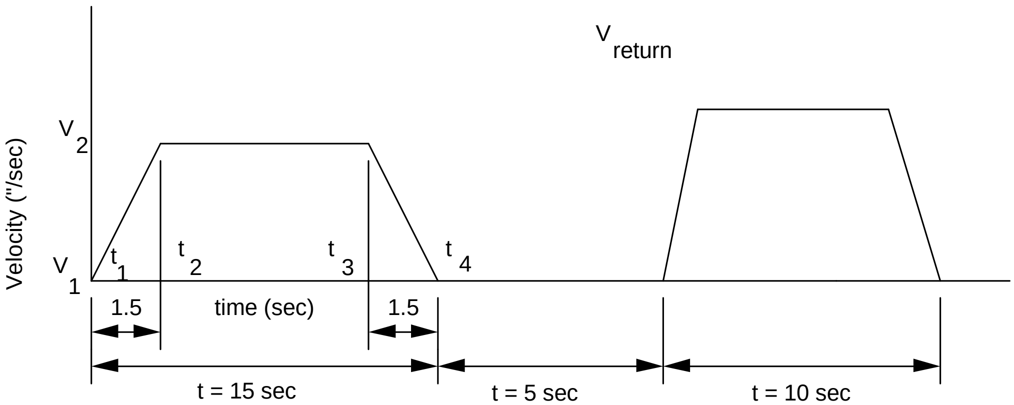


Figure 12.128 (a) Velocity Profile

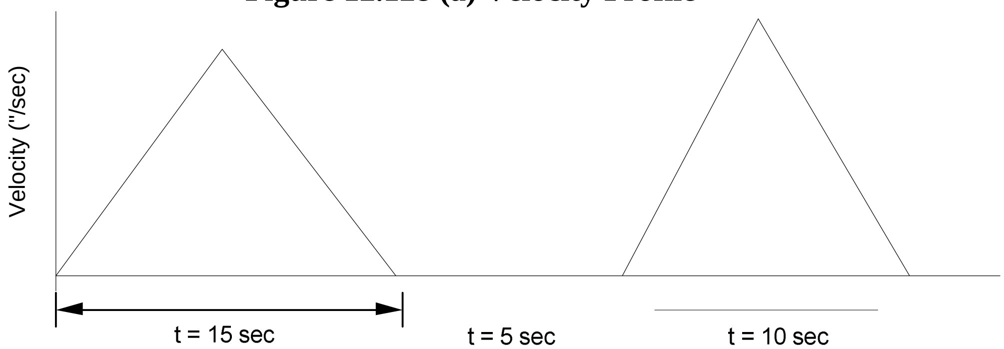


Figure 128 (b) Alternate profile choice

Note: The consequence of this choice is to increase the velocity of the actuator and hence the motor/pump size; however. The hydraulic force will decrease because the acceleration is smaller.

Note: If we choose the acceleration or deceleration too large, the boxes may fall over. So in essence, we must set our acceleration/deceleration limits based on practical and physical constraints.

We can assume any profile we want; however, our choice (if not client specified) can have a significant effect on pressure, flow and horsepower requirements. Let us first consider the advance part of the overall cycle. Using well known equations of motion, we can find the velocity required to satisfy both the velocity profile and the stroke constraints from the equation:

$$S = \frac{1}{2} a t_1^{2} + V t_2 + \frac{1}{2} d t_3^{2}$$

$$= 1/2\,((V_2-V_1)/(t_2-t_1))*(t_2-t_1)^2 + V_2(t_2-t_3) +$$

$$1/2\,((V_2-V_1)/(t_4-t_3))*(t_4-t_3)^2$$

See Figure 12.128(a) for nomenclature.

In this equation, we know t_2, t_1, t_4, t_3 ,S and V_1.

We can calculate V_2 to be .445 ft/sec. If this value is not acceptable, we can change the period during acceleration/deceleration to yield acceptable results. We defer the calculation of V for the return cycle because it is essentially unloaded. Therefore, we can apply the full pump flow to return the platen. The only constraint we have here is that the return cycle time is less than 10 sec.

Our burden profile reflects in this case, stiction and static friction. The burden profile for the advance and return parts of the cycle is illustrated in Figure 12.129.

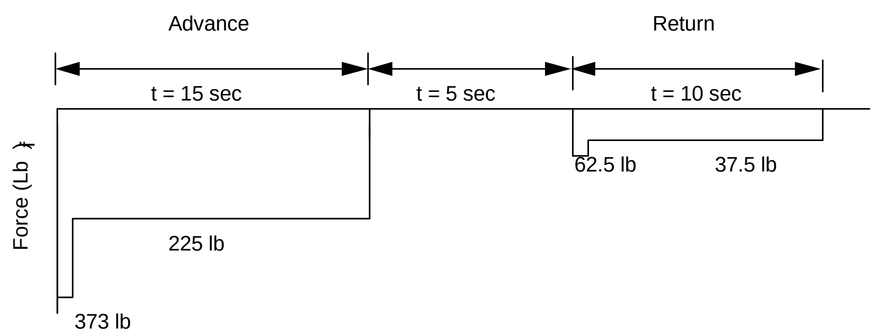


Figure 12.129(a) Burden Profile

Since friction exists in the circuit and since there are no external forces we need only concern ourselves with the natural deceleration which is found to be :

$$D_n = (225 \text{ lbf} * 32.2 \text{ ft/sec}^2 * 12\text{in/ft} / 1500 \text{ lbm}) = 58 \text{ in/sec}^2$$

The designed acceleration and deceleration are known to be

= (.445 ft/sec^2 * 12in/ft)/1.5sec = 3.6 in/sec^2. This means the system by itself slows down very fast compared to what we have specified. See Figure 12.129(b)

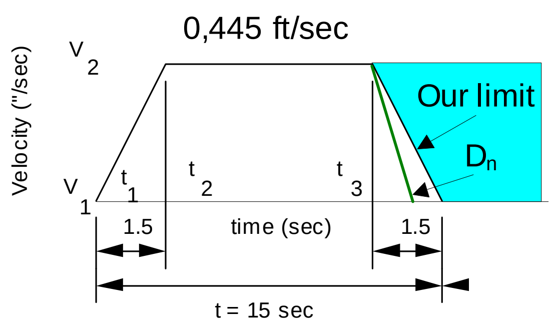
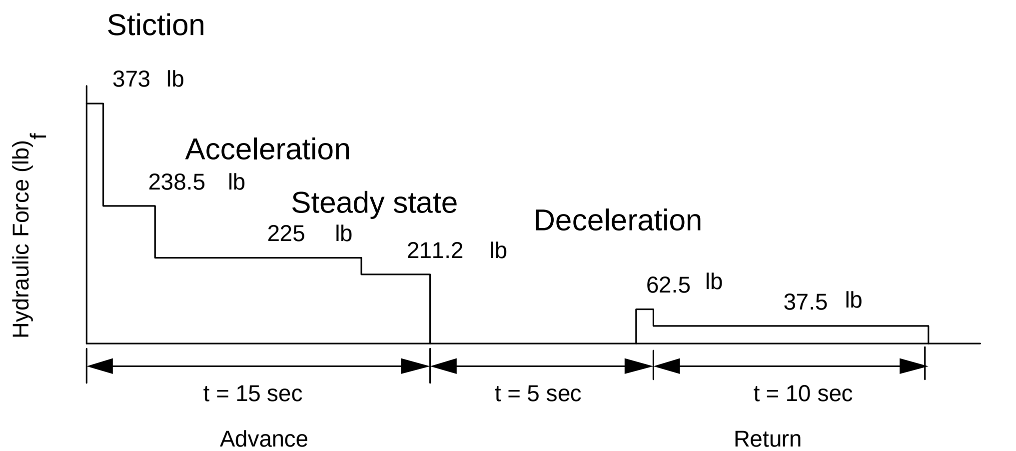


Figure 12.129(b) Limts

The force to accelerate the 1500 lb_f mass is about 240 lb_f. This is less than the static friction value. Thus if the static friction value is used to determine the pressure, then the system will accelerate at a rate which is greater than 3.6 in/sec^2 but this is acceptable. To meet both the time and stroke limitation we must assist in slowing down the platen. As it is in its natural state, it would slow down too fast. (Note: in our problem definition, we have not placed any constraints on acceleration or deceleration. In our case, the large natural deceleration could be used to our advantage as far as cycle time is concerned. Realistically, however, we must worry about the object sliding off or falling over during acceleration/deceleration. Thus our assumed velocity profile should be viewed from a physical point of view.)

<u>Develop the Hydraulic Force Profile.</u>

Since we have only one mass to consider and only one speed, the hydraulic force profile is readily established and is illustrated in Figure 12.130. We shall defer discussion of the return stroke at this point until later



Note; because we have no user constraints, we could set π(PA) to zero. But since practical considerations show that this could result in things “flying off” we shall set our own as indicated.

Figure 12.130 Hydraulic Force Profile

2. Choose an Actuator or Motor Size

The choice of the actuator size dictates the pressure and flow levels. We must try to keep pressures and flow rates reasonable. In this case, if we assume a 3/4 " bore size, we would get maximum pressures of 1000 psi. But this would mean a rod dia. of about 1/3" which is far too small for a 72" stroke actuator with forces of 400 lb. Therefore, let us choose an actuator with a 1" rod dia and a bore dia of 2". Thus the maximum pressures would be in the order of 150 psi. This is small but acceptable. The pressure and flow profiles for the advance part of the cycle is thus illustrated in Figure 12.131. It should be noted that in the pressure profile, a mechanical efficiency of 90% is assumed. We will account for volumetric efficiency when we size up the pump capacity.

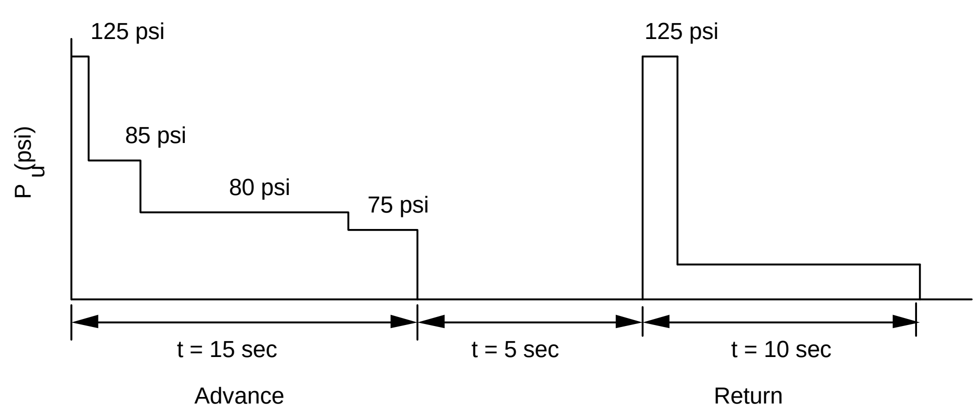


Figure 12.131 Pressure Profile assuming a mechanical efficiency of 90%

NOTE: We can use the maximum flow of 4.5 gpm (17 in^3/sec) in the reverse direction. Thus,

$$V_{maxr} = Q_f \text{ / area of rod side } (A_b - A_r)$$

$$= 17 \text{ in}^3\text{/sec/}(2.36\text{in}^2) = 7.24 \text{ in/sec}$$

But
V_{maxr} = stroke/time (neglecting acceleration/deceleration times at present),

from which

$$t_r = \text{stroke}/V_{maxr} = 72''/7.24$$

$$= 9.94\text{sec}$$

Thus, we meet the time to return constraints of 10 secs. We note that acceleration and deceleration times will affect this time but it will be a small amount if we use the same pressure levels as in the advance part of the cycle.

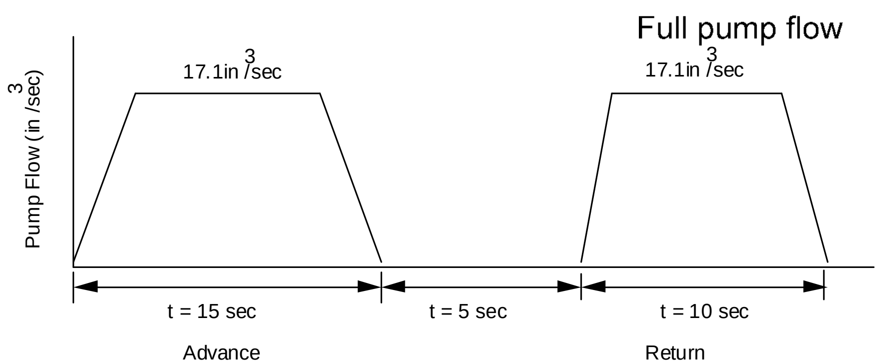


Figure 12.132 Pump flow profile

3. Establish Flow and Pressure profiles

The pressure and flow profiles based on the hydraulic force profile were shown in Figures 12.131 and 12.132. In the reverse direction, we shall assume that the actuator can satisfactorily stop the platen with no serious side effects. If this is a concern, one might choose an actuator with cushions.

4. Check basic Horsepower to satisfy the hydraulic profiles.

The horsepower profiles are shown in Figure 12.133. These values appear reasonable. It should be pointed out that the value of plotting the HP requirements lie in the fact that if we simply used the **maximum pressure and flow,** we would get a value of .33 HP compared to what we really need (.23HP). See Figure 12.133. We would thus be **oversizing** the input motor.

The easiest way to calculate horsepower is to take the appropriate graphical plots of P and Q vs. time or displacement and then take the product at various points.

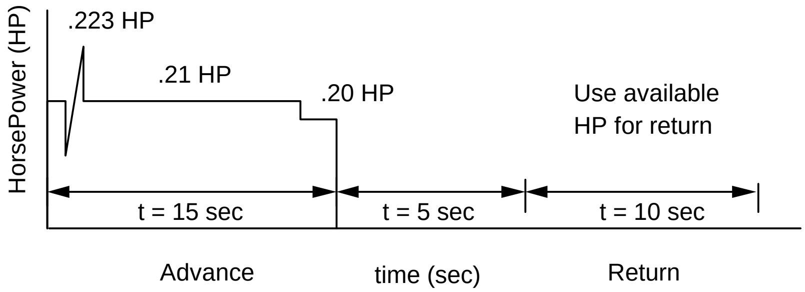


Figure 12.133 HorsePower Requirements

5. Design Circuit

A circuit to meet the constraints and hydraulic profile is shown in Figure 12.134. This circuit uses pressure to achieve acceleration and deceleration.

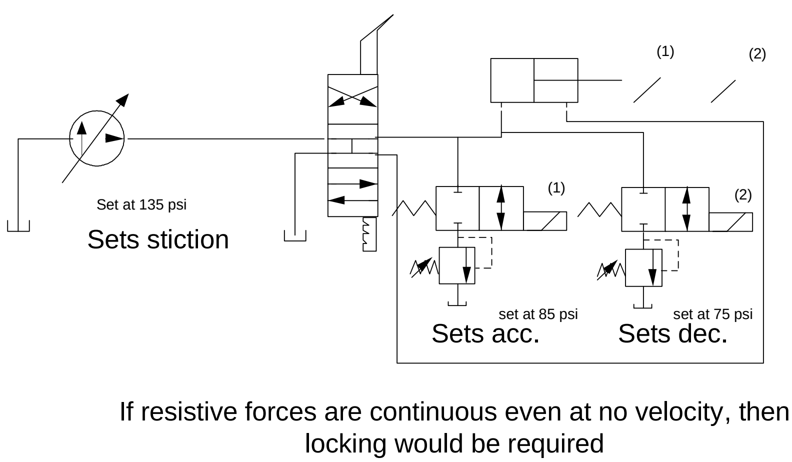


Figure 12.134 Circuit Possibility #1

This circuit does not lock the load (if so required). We could use a closed center valve; but if the valve is accidentally closed during motion, we could have pressure transient problems. In this case, since the natural deceleration due to friction is large, this probably would not create too severe of a problem.

The circuit is very complex. We have to set three pressure levels if we want to follow the velocity profile. What compromises can be made to simplify the circuit? It should be recalled that we stated initially that no constraints on acceleration or deceleration are defined by the client. Realistically, we must place some of our own constraints, because as we stated before, things could fly off if we are not careful. This is a variable that we can adjust and hence compromise in terms of circuit design. We could check analytically if the object would fall off or we could design a mechanical constraint to prevent the object from flying off during starting or stopping of the system.

Let us suppose that indeed, we can tolerate a very large acceleration/deceleration, much larger than indicated in our velocity profile. We shall use the maximum system pressure of 125 psi to accelerate the platen and load rather than just use it to overcome stiction. The system would accelerate to full speed in .1 secs as calculated by:

$$\Delta t = M*\Delta V / (P_{max}*A_b - F_{dynamic})$$

$$= 1500*(32.174*12)*5.33 / (125*\pi - 225)$$

$$= .1 sec$$

The system would accelerate in .1 sec compared to 1.5 sec as per the original design. If this is excessive, we <u>must</u> consider alternatives to overcoming stiction first. Let us assume it is not excessive. In addition, let us assume that we can use a hydraulic cushion to decelerate the mass. Our circuit would become :

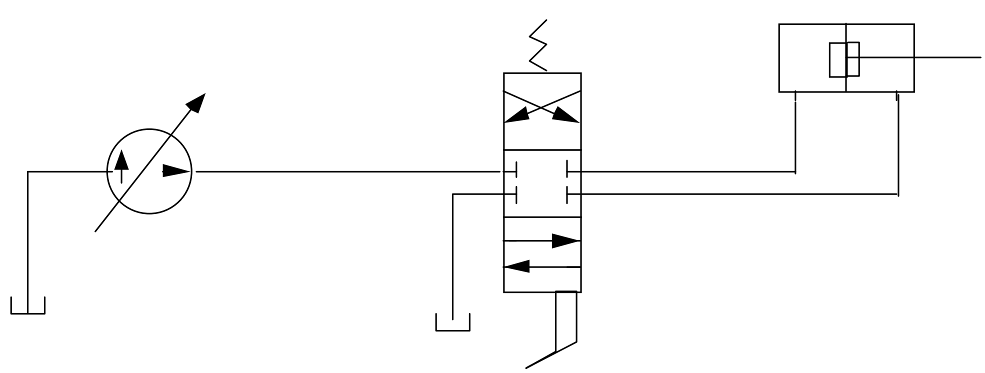


Figure 12.135 Circuit Possibility #2

If we use a deceleration valve at the end of the stroke rather than a cushion, we can decelerate the system over a larger displacement than that with a cushion. The circuit would be that as illustrated in Figure 12.136.

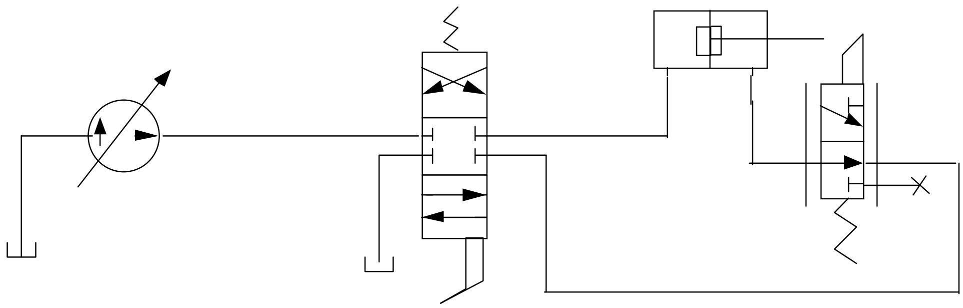


Figure 12.136 Circuit Possibility #3

Clearly, both circuits are much simpler in form. Therefore compromise to simplify a circuit is essential if possible. In many cases, the user constraints (or indeed, physical considerations) cannot be varied from; hence complexity of the circuit must be tolerated.

6. Plot pressure, flow and horsepower profiles at important parts of the circuit .

Let us examine the pressure, flow and HP profiles of the first circuit. If we assume small pressure drops across the DCV (which is reasonable in this case considering the low flow rate), the pressure, flow and HP plots of Figures 12.131, 12.132 and 12.133 respectively are essentially those at the pump.

We want to make sure that the pump input HP is greater than .25 HP to account for mechanical efficiency of the pump. What we do notice is the efficiency of the circuit is quite high because losses are minimal. If we had used a fixed displacement pump and a RV, the efficiency during acceleration and deceleration would decrease. For this particular case, the losses would be minimal and hence from a cost point of view, it may be well worth considering.

In summary, it is MOST useful to plot the pressure and flow profiles at the pump and of course at the actuator or motor (which you already have from the hydraulic force (torque) profiles. This readily accommodates overall HP and efficiency calculations.

7. Component selection

We would choose a 5 gpm pump with a pressure rating of 500 psi. We would choose a electric motor rated at .25 HP. All high pressure lines are sized at 1/2" i.d. and inlet lines at 1.5". The reservoir is sized at 15 gals(US). The actuator has a bore dia. of 2" and a rod size of 1". The stroke is 72". Limit switches are used to actuate the solenoid DCVs.

12.8.2 Example #2

Consider the system in Figure 137. Determine the hydraulic torque profile for both rising and lowering of the load. Note that T_{f1} and T_{f2} are constant resistive loading throughout the cycle in both directions.

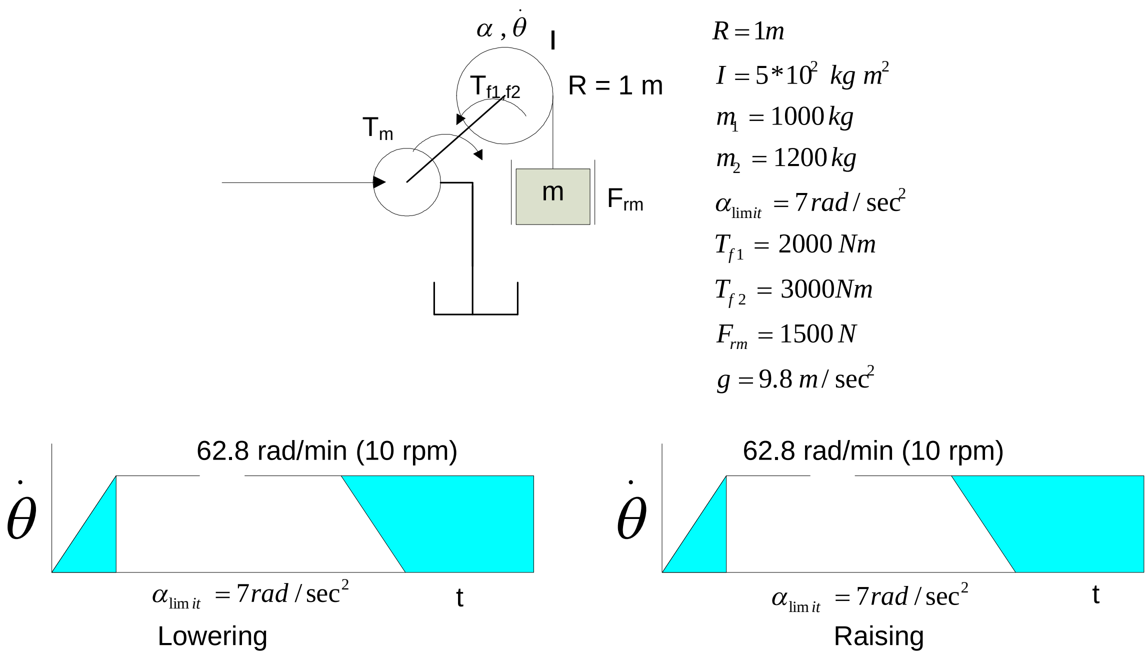


Figure 12.137 Load system and velocity profile

12.8.2.1 Step one Freebody diagram

The first step is to analyze the mechanics and dynamics of the problem. It is a fact of life that this part of the design takes the most time and in fact is very "iffy" unless we have a good handle on the actual situation. In real applications, you will have to make best guesses to come up with approximate values for parameters; even the basic describing equations can be suspect at times, but this should not deter you. Having a sense of what the expectations are on the system can result in a much better and indeed, more reliable hydraulic design.

In this problem, we are given some well defined parameter ranges and external torques on the rotary system. So we will start by drawing a free body diagram of the two masses.

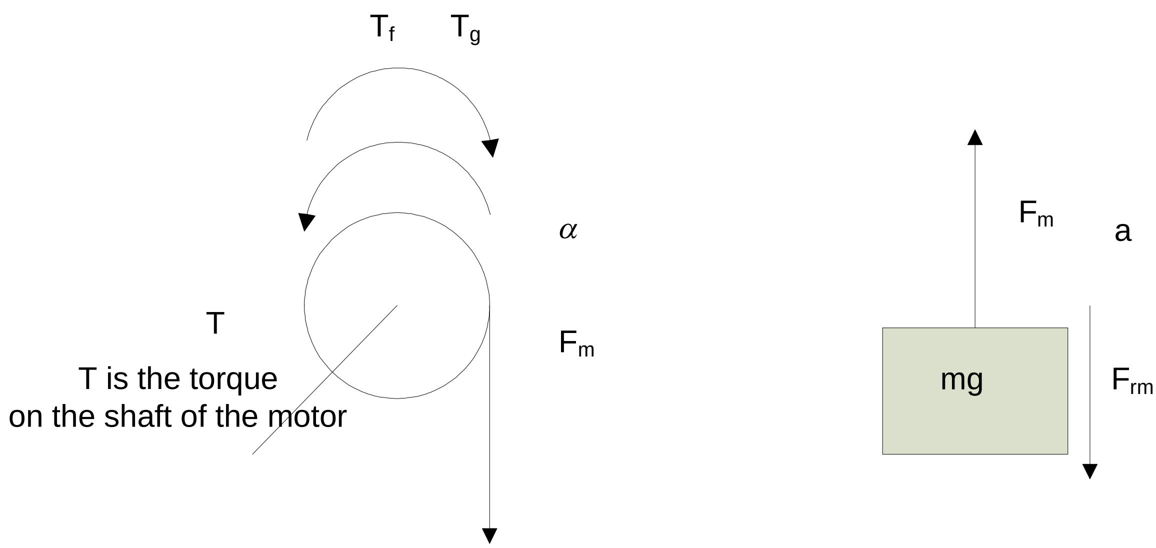


Figure12.138 Freebody Diagram

12.8.2.2 Determine the mass(inertia) profile

This is not straight forward since we have both inertia and linear mass involved. We will have to determine an effective flywheel inertia. The inertial torque of the flywheel is

$T_{Ifw} = I\alpha$

The linear acceleration torque due to the mass, m is

$T_{Ima} = ma\ r$

But $a = r\alpha$, therefore

$T_{Ima} = mr^2\alpha$

The total inertial torque is thus the sum of the two or

$$
\begin{aligned}
T_{Ieff} &= T_{Ifw} + T_{Ima} \\
&= I\alpha + mr^2\alpha \\
&= (I + Mr^2)\,\alpha
\end{aligned}
$$

So our effective inertia profile is

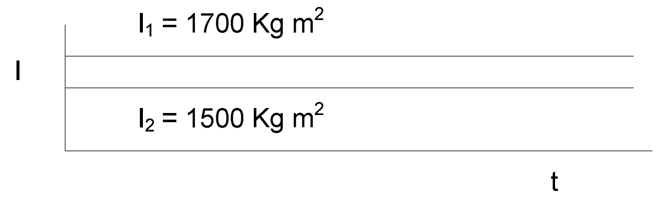


Figure 12.139 Effective Inertia profile

where $I_1 = I + m_1 r^2 = 5*10$ Kg m^2 + 1000 Kg * 1 m^2 = 1500 Kg m^2

$I_2 = I + m_2 r^2 = 5*10$ Kg m^2 + 1200 Kg * 1 m^2 = 1700 Kg m^2

12.8.2.3 Determine the burden torques

Raising the mass

Determine the burden torques (Remember, this is the steady state)

The friction torque on the flywheel is T_f, is resistive and hence is negative

$T_f = F_{rm}\, r + T_{fi}$ where i = 1,2

The gravity torque on the flywheel is T_g and is resistive and hence is negative

$T_g = mgr$ (note mgr is a positive number but T_g is negative according to our sign convention)

Substituting in for m, and T_{f1}, T_{f2} the burden profile becomes:

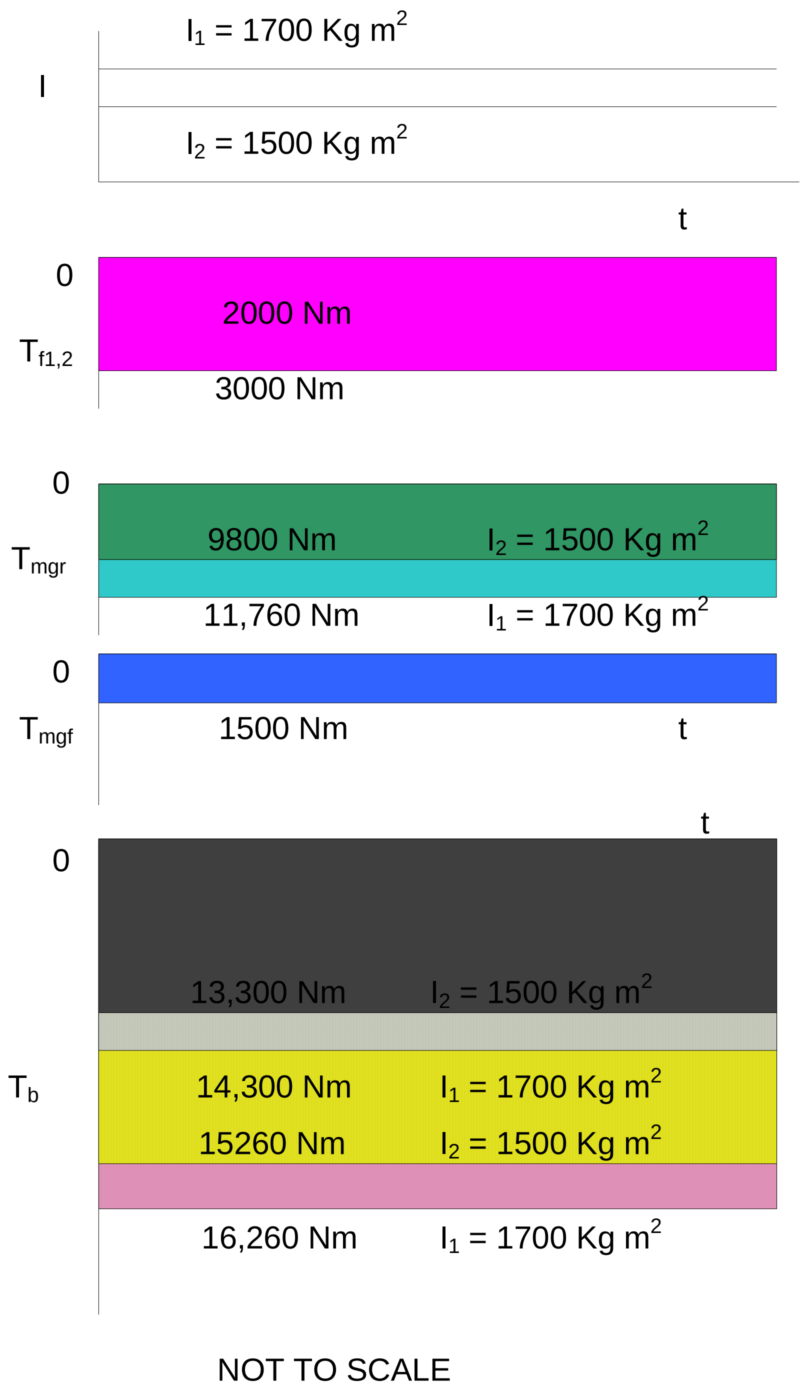


Figure 12.140 Burden Profile Lifting

Lowering the mass

Determine the burden torques (Remember, this is the steady state)

The friction torque on the flywheel is T_f, is still **resistive and hence is negative**

$T_f = F_{rm}\, r + T_{fi}$ wher i = 1,2

The gravity torque on the flywheel is T_g and is **run away and hence is positive**

$T_g = mgr$

Substituting in for m, and T_{f1}, T_{f2} the burden profile becomes:

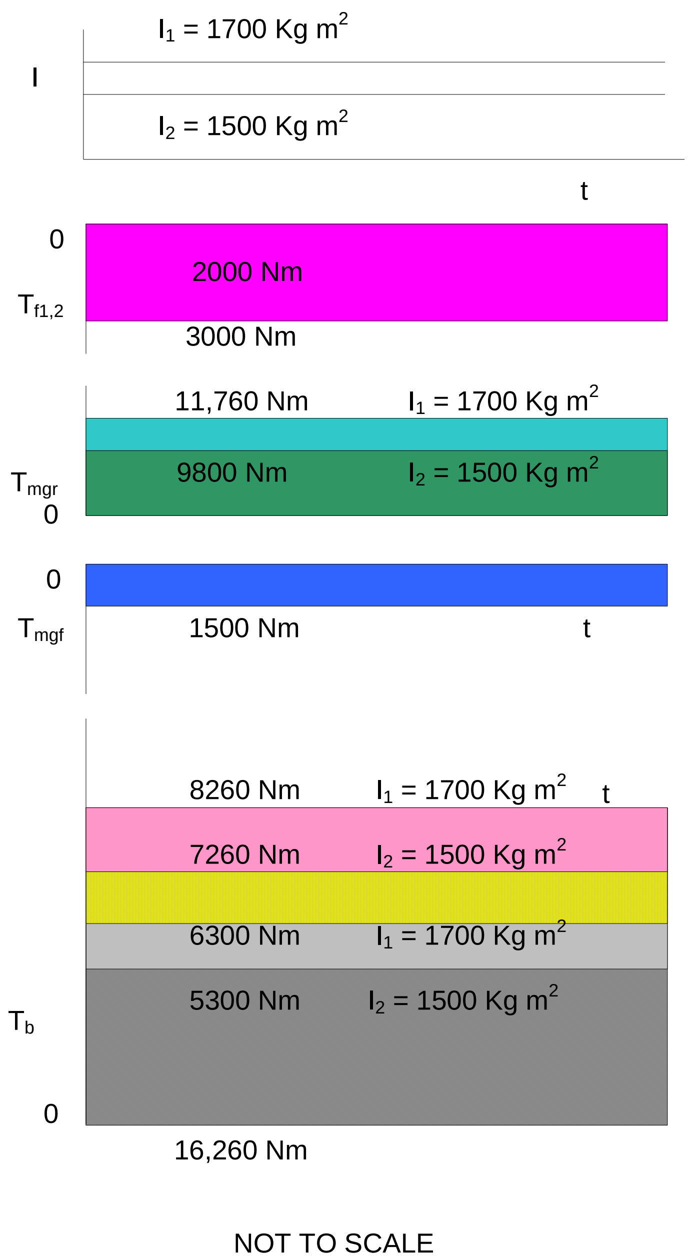


Figure 12.141 Burden profile lowering

12.8.2.4 Determine the natural acceleration or deceleration

Raising **- burden is negative, hence we have a natural deceleration, $D_{\alpha n}$**	**Lowering** **-burden is positive, hence we have a natural acceleration, α_n**
Recall $D_{\alpha n}$ = - T_b/ I_{eff} **$D_{\alpha n1}$ = 13,000/1500 = 8.87 rad/sec^2** **$D_{\alpha n2}$ = 14,300/1500 = 9.53 rad/sec^2** **$D_{\alpha n2}$ = 15,260/1700 = 8.98 rad/sec^2** **$D_{\alpha n4}$ = 16,260/1700 = 9.56 rad/sec^2**	**Recall α_n = - T_b/I_{eff}** **α_{n1} = 6300/1500 = 4.2 rad/sec^2** **α_{n2} = 5300/1500 = 3.53 rad/sec^2** **α_{n3} = 8260/1700 = 4.86 rad/sec^2** **α_{n1} = 7260/1700 = 4.27 rad/sec^2**

The coloured highlighted numbers are the two extremes

12.8.2.5 Superimpose the natural acc/dec. onto velocity profile

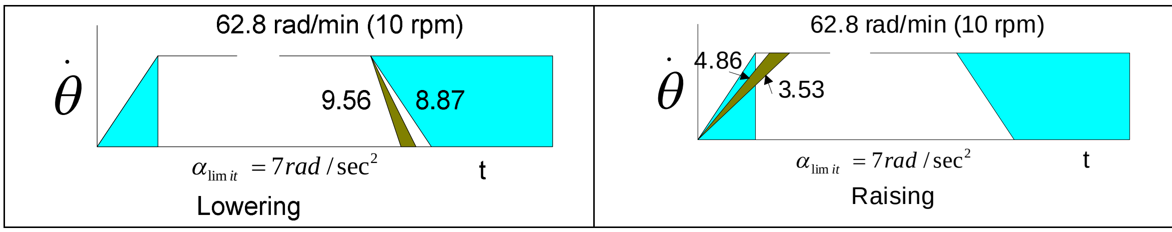


Figure 12.142 Natural Acc/Dec on velocity profiles

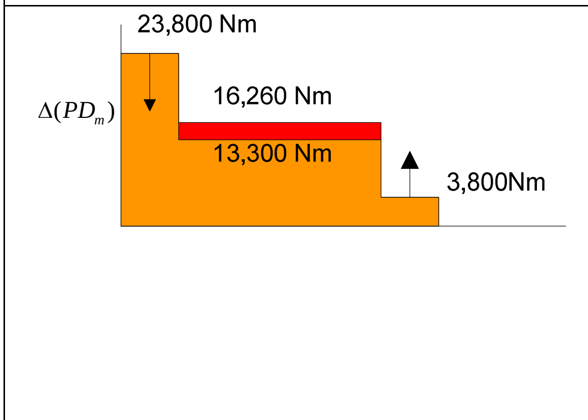
12.8.2.6 Use key to determine appropriate equations

Raise	**Lower**
Deceleration $\Delta(PD_m)\big\vert_{\min} = -T_{b_{\max}}\big\vert_{I_{\min}} - I_{\min}\ D\alpha_{\max}$ = -(-14300) - 1500*7 = 3800 Nm Cont. Acceleration	Deceleration $\Delta(PD_m) = 0$ Cont. Acceleration

$\Delta(PD_m)_{\max} = -T_{b\min_{I\min}} + I_{\min}\ \alpha_{l\max}$ = -(-13,300) + 1500(7) = 23,800 Nm	$\Delta(PD_m)_{\max} = -T_{b\min_{I\min}} - I_{\min}\ D\alpha_{l\max}$ = -(-5300) - 1500(7) = -15,800 Nm

12.8.2.7 Plot hydraulic torque profile

Raise



Lower

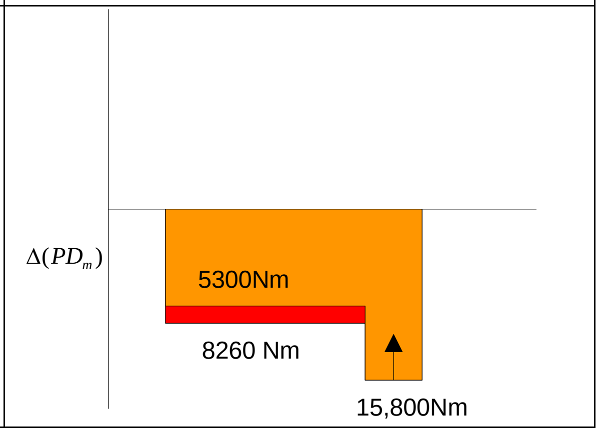


Figure 12.143 Hydraulic Torque Profile

12.8.2.8 Choose pump size to achieve reasonable pressures and flow rates.

Because of the loads involved, we should choose some gear reduction. Even with this choice, we will have to choose a motor with a large displacement.

$\Delta(PD_m)$ = 23,800Nm

$$\dot{\theta} = 10\ rpm = 62.8\ rad / \min$$

Let us choose a gear ratio N of 20. Then

$\Delta(PD_m)$ = 23,800Nm /20 = 1,190Nm

$$\dot{\theta} = 62.8 rad / \min * 20 = 1256\ rad / \min = 200\ rpm$$

We shall choose a motor size with a displacement of 6.6 $*10^{-5}$ m^3/rad. This is one mother of a motor but it is a reasonable first choice to get started.

$$P_{\max} = \frac{1190\,Nm}{6.6*10^{-5}\,m^3/rad} = 18*10^6\,N/m^2$$

$$= 18MPa \quad (quite\ reasonable)$$

$$Q_{\max} = 1256\,rad/\min * 6.6*10^{-5}\,m^3/rad$$

$$= 8.29*10^{-2}\ m^3/\min * \left(\frac{100cm}{1m}\right)^3 * \frac{1l}{10^3\,cm^3}$$

$$= 82.9\,l/\min \quad (also\ quite\ reasonable)$$

So with a choice of D_m = 6.6*10-5 m^3/rad and N = 20, our pressure and flow rates are reasonable. The final choice needs to be fine tuned so that standard commercial values for N and D_m can be used.

Our pressure and flow profiles become:

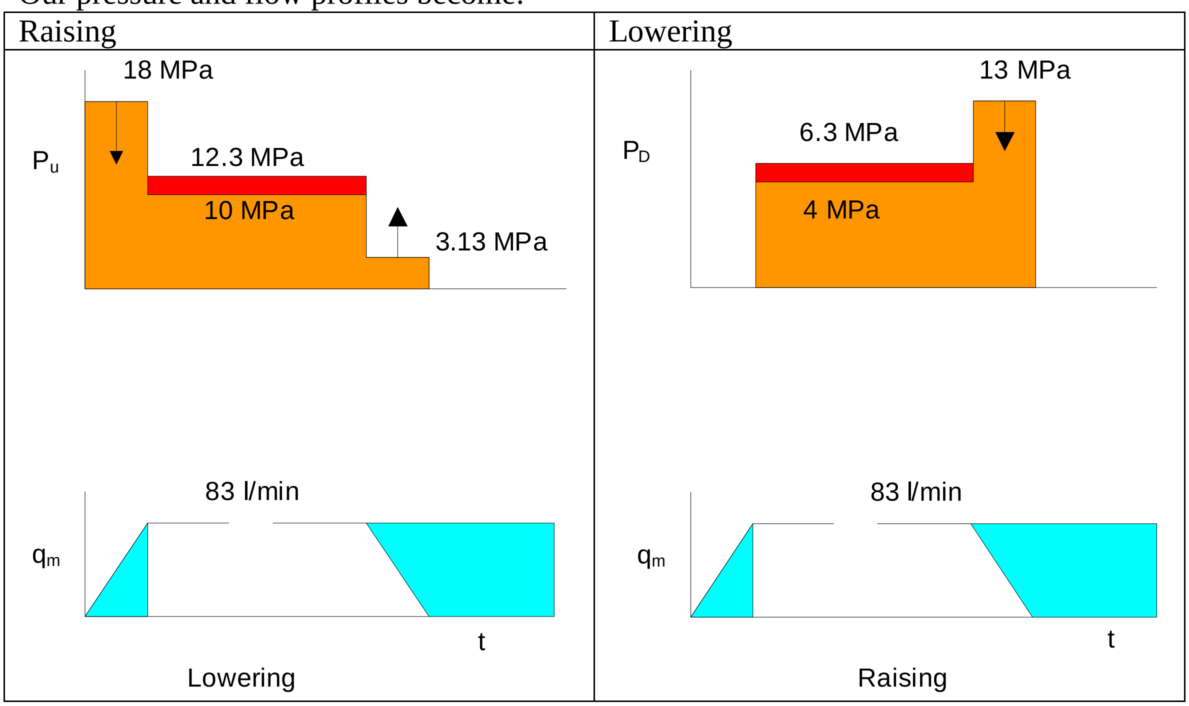


Figure 12.144 Pressure and Flow Profiles

12.8.2.9 Design Circuit (Raising):

We shall use pump pressure to limit acceleration. To decelerate, we could use a switched in pressure relief valve as shown in Figure 12.145.

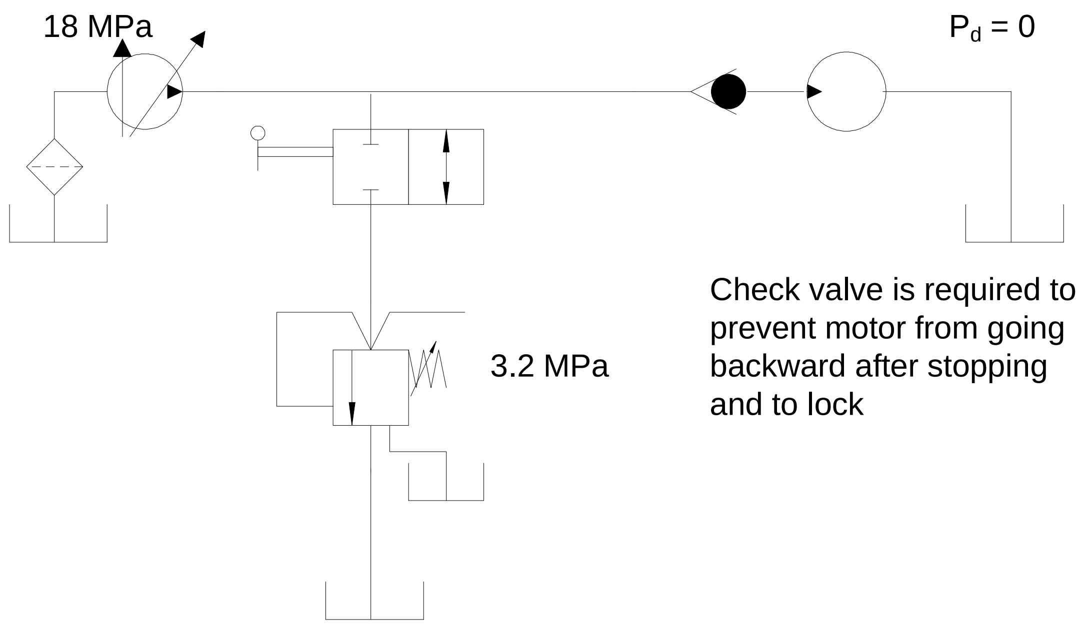


Figure 12.145 Possible raising circuit

12.8.2.10 Design Circuit (Lowering):

Since the system is runaway (burden is positive through out the cycle), motor acts as a pump. One circuit could be:

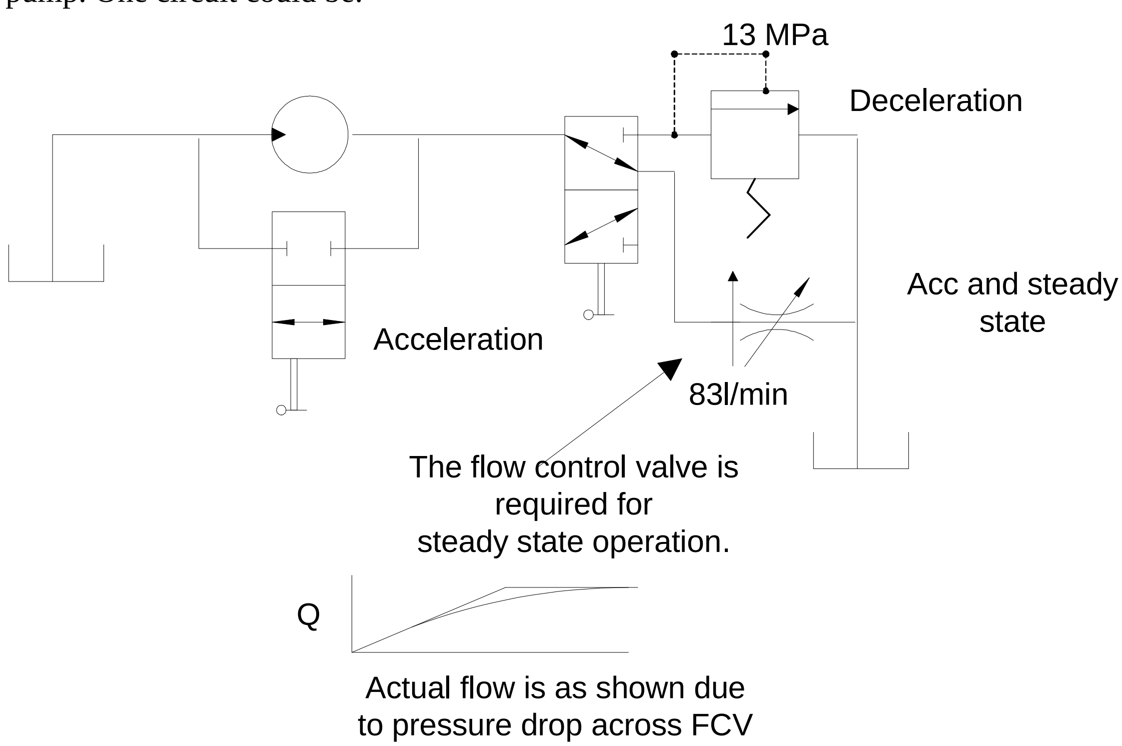


Figure 12.146 Possible lowering circuit

We must note that for lowering, **we must unload the pump during this period**. This may not be a problem because we could in fact turn it off or use an unloading circuit.

Another possible circuit is to add a counterbalance valve after the motor and convert the circuit to a resistive type one. The pressure profile upstream and downstream would now appear as:

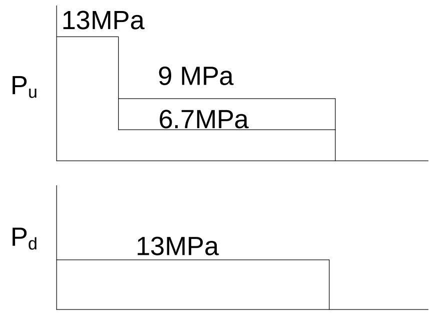


Figure 12.147 Pressure profile with CBV inserted after motor

A new circuit to meet these new pressure profiles could be:

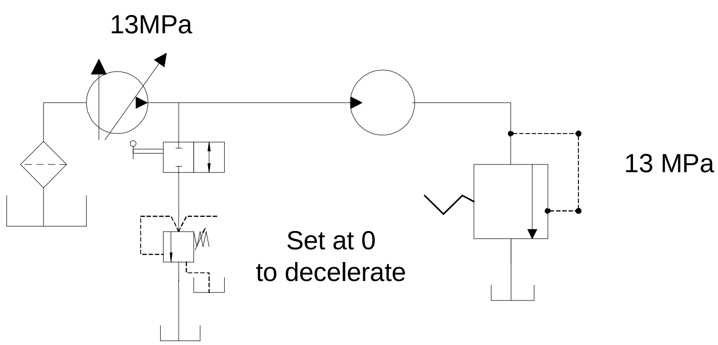


Figure 12.148 Circuit 2 with a constant back pressure

12.8.2.11 Combined Circuits (Raising and lowering):

Now if we look at the raise and lower circuits we can see some problems that could arise.

Raise **Lower**

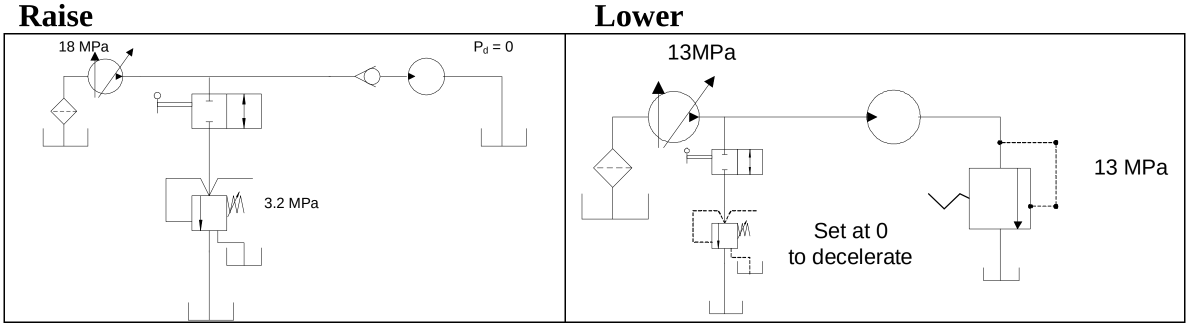


Figure 12.149 Comparison of circuits

The problem is that the pump must have two pressures to meet the requirements: 18MPa and 13 MPa. Suppose we did use 18 MPa, the acceleration in the forward direction would be satisfied. But in the lowering, the pressure drop across the motor would be 5 MPa (18 MPa - 13 MPa). This would accelerate the motor more than our limit, so this cannot work. If we made it 13 MPa, then in the raising circuit, since P_u was an upper limit, then the acceleration would be smaller which is acceptable. However, it is just 0.7 MPa higher than the maximum steady state burden pressure which could mean that under some instances, the load would take a long time to accelerate to speed. If this is acceptable, then we can combine the circuits as shown in Figure 12.150.

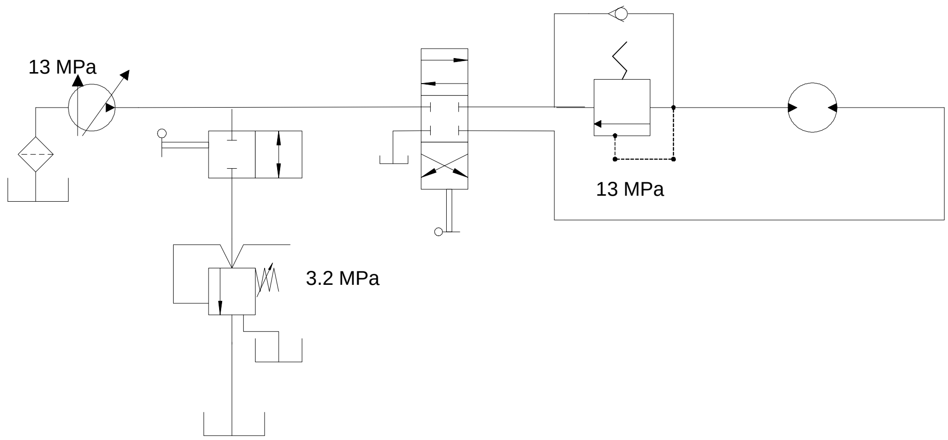


Figure 12.150 Combined circuit #1

If we feel that the reduced acceleration is really too slow, we can then increase the back pressure on the CBV to 18 MPa and hence the pressure drop across the motor during acceleration is 0 in lowering which is what we want. Raising is not affected. This combined circuit is shown in Figure 12.151

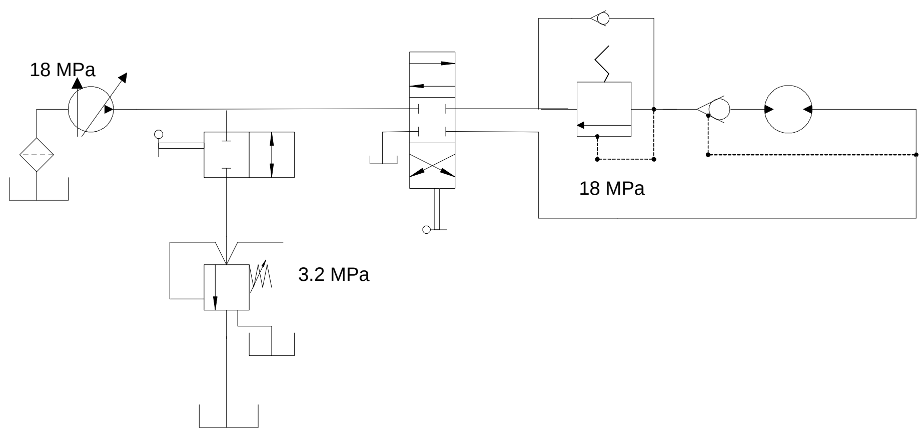


Figure 12.151 Combined circuit #2

Note that we "snuck in" a pilot operated check valve to prevent unwanted reverse motion (it is a gravity load). Raising is not a problem because the check valve is open in the direction of flow. When the load comes to a stop, the check valve prevents backward motion which is good from a safety point of view even though we are using a closed centre valve for locking. What about reverse? We know that pilot operated check valves require a positive pressure to keep them open and with a gravity load on them; this pilot pressure could be substantial (recall our formulas for pilot check valves). However, for reverse, we rely on the fact that the upstream pressure must be zero during deceleration. This would mean that the check valve would close and the system would come to a halt immediately. With a large load as we have in this case, this could most certainly blow lines. **So we cannot just simply throw in pilot check valves without some thought.** Unless a practical way of ensuring a positive and adequate pressure exists upstream during deceleration and locking (lowering), then we had best look at other ways of ensuring no reverse motion can occur when stopped. The closed centre valve, with appropriate logic employed, can serve this purpose. As soon as the load is stopped, the valve must be switched to its centre position, which does require some sensing.

We have shown all directional control valves with manual switching. These probably would have solenoids actuating them and hence could be readily integrated with a programmable logic circuit.

12.8.2.12 Horsepower and efficiency considerations.

In this example, in lowering, the system (gravity) does all the work for us. We just try to control it. So, it is very hard to talk about efficiency during lowering. So lets us concentrate on the raising part of the cycle where we must input energy to do the task. To do this simply plot the pump pressure and flow as well as the hydraulic torque and velocity profiles.

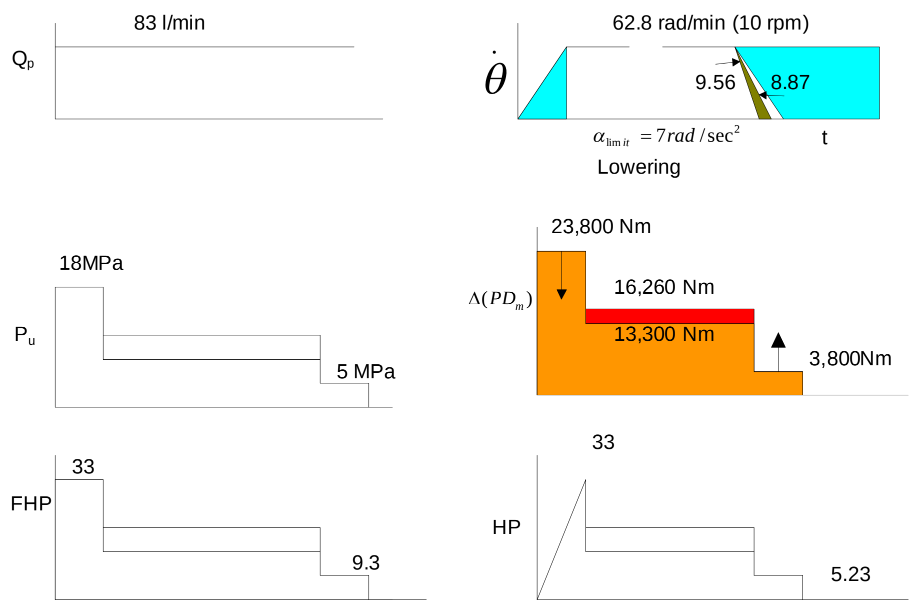


Figure 12.152 Horsepower considerations for raising only

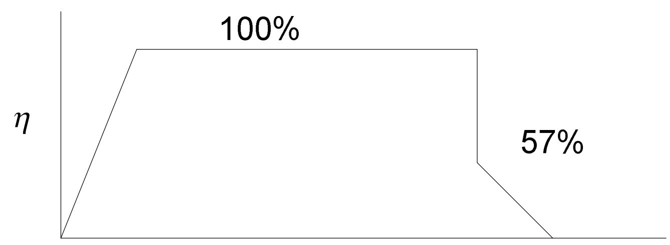


Figure 12.153 Efficiency of circuit (Raising only)

The actual efficiencies would be reduced by the inclusion of the motor and the pump efficiencies.

12.8.2.13 Other circuit possibilities

The above circuit does meet the imposed constraints. But is it a practical? There are a lot of valves to synchronize if done manually. If done with solenoid and electronic controllers, then it becomes quite feasible to do. Let us look at one alternate circuit as shown in Figure 12.154. The circuit is simple but does require the use of a rate generator to accelerate and decelerate and does require a directional control valve that can handle pressure in the down stream ports as illustrated. It is essential to note that the flow control valve does follow the required profile, THEN THE PRESSURE PROFILES WILL ALSO BE THE SAME AS IF THE MANUAL CIRCUIT WAS USED.

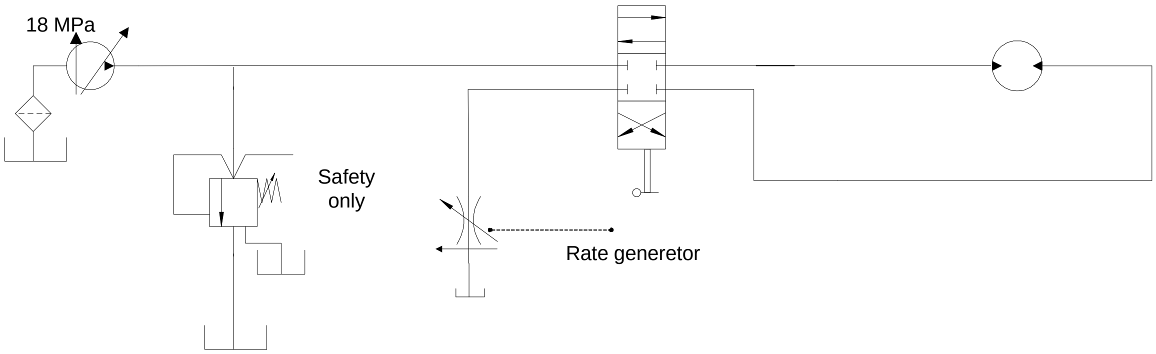


Figure 12.154 Flow control circuit in return line

There are so many other possibilities that could be used. Design does not have one answer but many. Some circuits are better than others but it is very important that we understand the constraints imposed and then design to them first (worst case scenario). Then, we can try to compromise in order to reduce complexity and cost.