

SOLUTIONS MANUAL TO ACCOMPANY

SYSTEM

Third Edition

DYNAMICS

Modeling and
Simulation of
Mechatronic Systems

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DONALD L. MARCOLIS

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**Solutions Manual to Accompany System Dynamics:
Modeling and Simulation of Mechatronic Systems
Third Edition**

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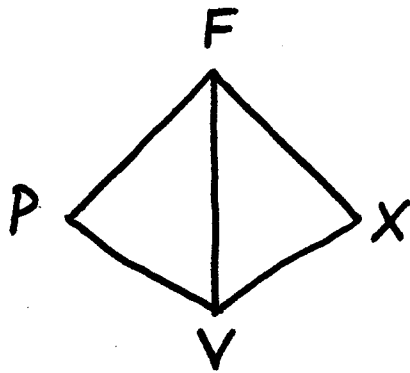
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1-1 to 1-5 These are mainly discussion questions.

1

2-1

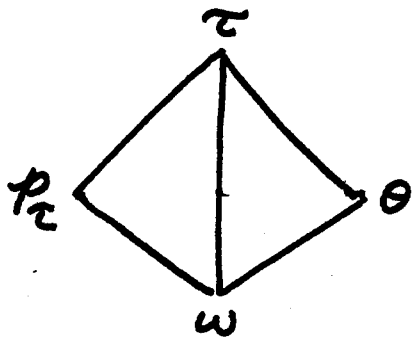


$$[F] - N$$

$$[V] - m/s$$

$$[P] - N \cdot s$$

$$[X] - m$$

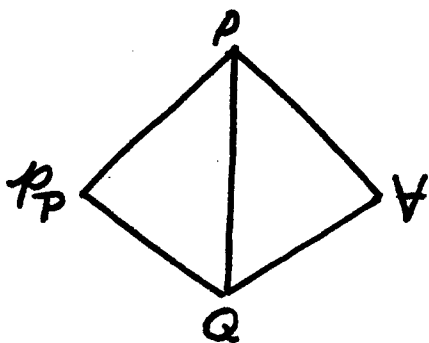


$$[\tau] - N \cdot m$$

$$[\omega] - rad/s$$

$$[P_\tau] - N \cdot m \cdot s$$

$$[\theta] - rad$$

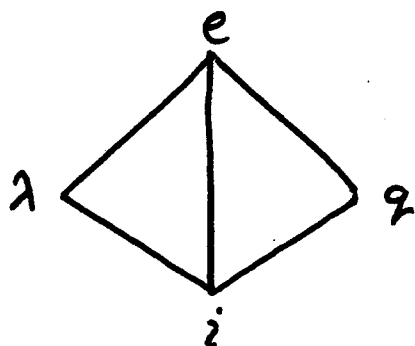


$$[P] - N/m^2$$

$$[Q] - m^3/s$$

$$[P_P] - N \cdot s/m^2$$

$$[V] - m^3$$



$$[e] - v$$

$$[i] - a$$

$$[\lambda] - v \cdot s$$

$$[q] - a \cdot s = c$$

2-2

$$\frac{\tau}{\omega} \text{ Electric Motor } \frac{e}{i}$$

(a)

2

$$\frac{\tau}{\omega} \text{ Hydraulic Pump } \frac{P}{Q}$$

(b)

$$\frac{\tau}{\omega_1} \text{ Shaft } \leftarrow \frac{\tau}{\omega_2}$$

(c)

$$\frac{F}{V_1} \text{ Shock Absorber } \frac{F}{V_2}$$

(d)

$$\frac{e_1}{i_1} \text{ Transistor } \frac{e_2}{i_2}$$

(e)

$$\frac{e}{i} \text{ Speaker}$$

(f)

$$\frac{\tau}{\omega} \text{ Crank } \frac{F}{V}$$

(g)

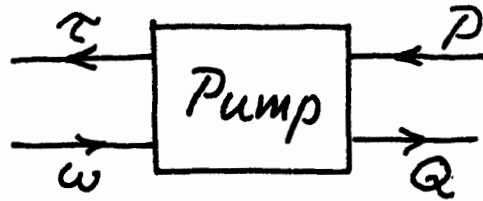
$$\frac{F}{V} \text{ Wheel } \leftarrow \frac{\tau}{\omega}$$

(h)

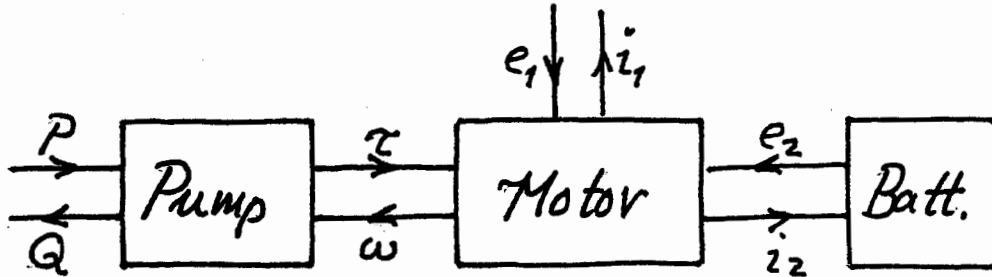
$$\frac{\tau}{\omega} \text{ Motor } \frac{e_a}{i_a} \quad e_f / i_f$$

(i)

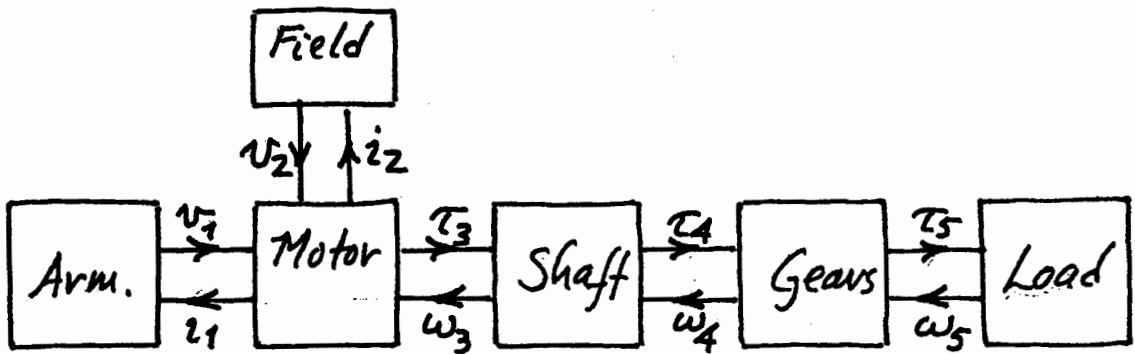
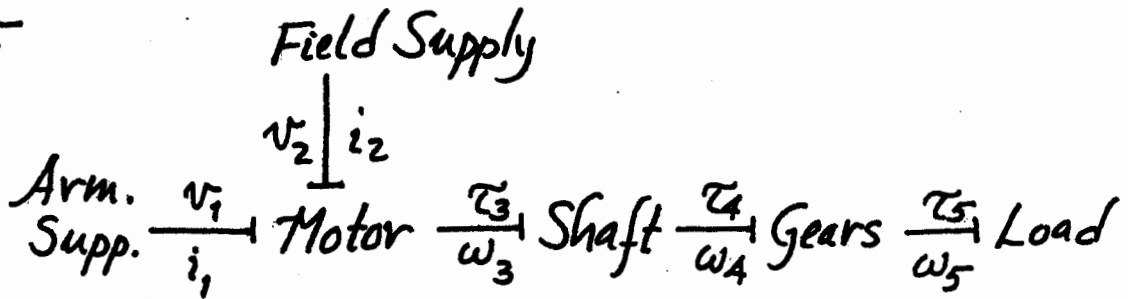
2-3



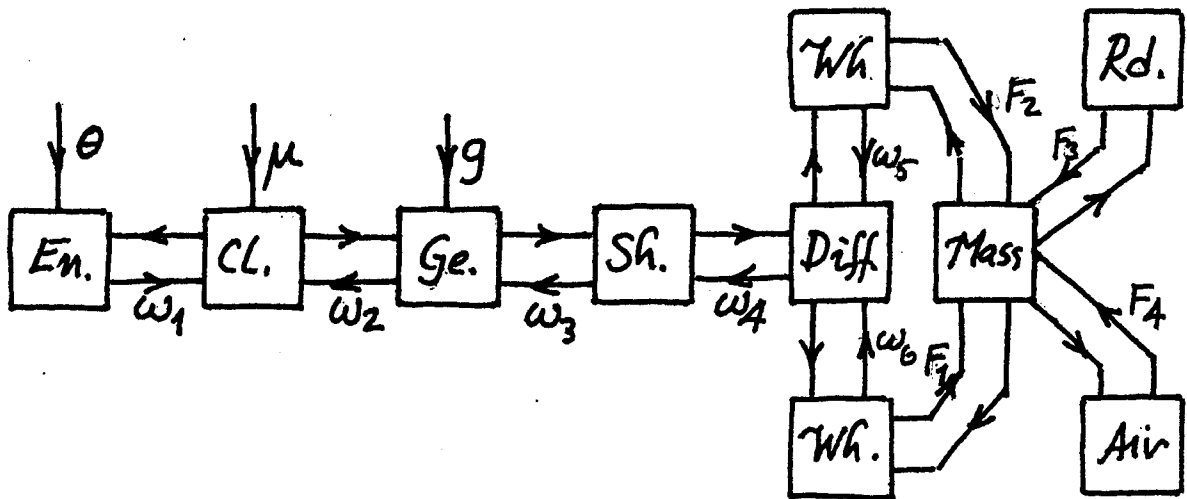
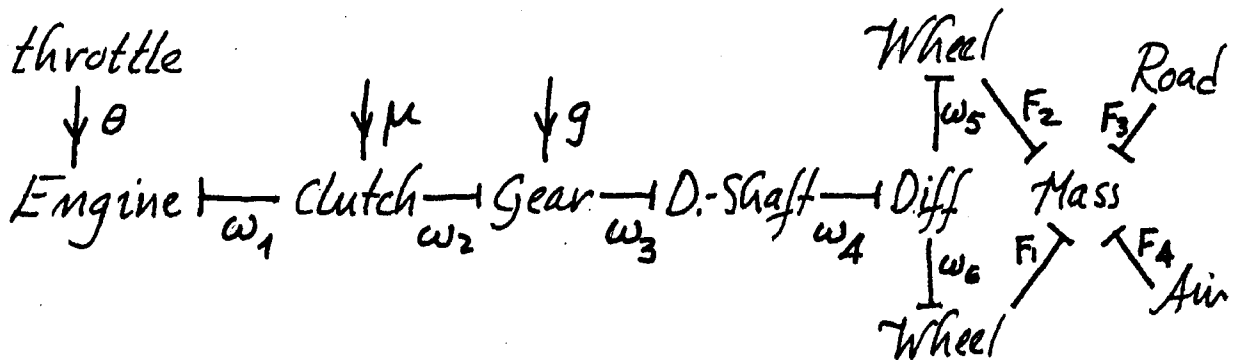
2-4



2-5



2-6

2-7 Inputs: P, e_1, e_2 Outputs: Q, i_1, i_2

2-8

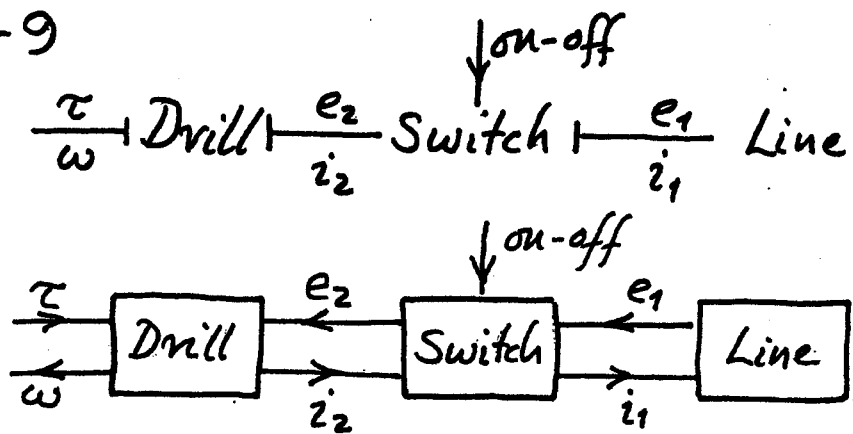
$$\text{Power} \times \text{Time} = \text{Energy}$$

$$P \cdot t = mgh$$

$$100 \cdot t = 10 \cdot (9.81) \cdot 30$$

$$t = 29.43 \text{ s}$$

2-9



2-10

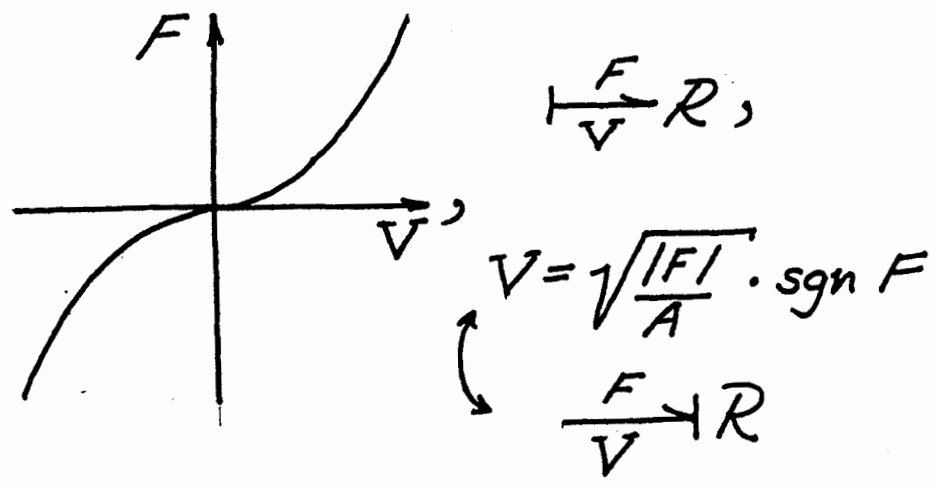
$$\tau \omega = P Q$$

$$\omega = \frac{P}{\tau} \cdot Q = \frac{7.0 \times 10^6}{5} Q$$

$$\omega = 1.4 \times 10^6 Q$$

$$\left[\frac{\text{rad}}{\text{s}} \right] = \left[\frac{1}{\text{m}^3} \right] \cdot \left[\frac{\text{m}^3}{\text{s}} \right]$$

3-1



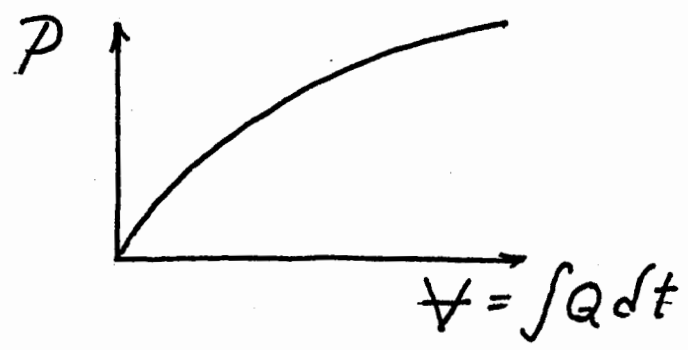
3-2

$$hA = V = \int Q dt,$$

$$P = \rho g h = \frac{\rho g}{A} \int Q dt$$

$$\therefore C = A / \rho g$$

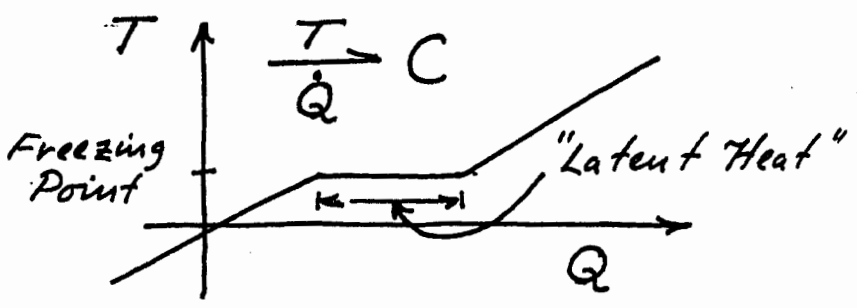
3-3



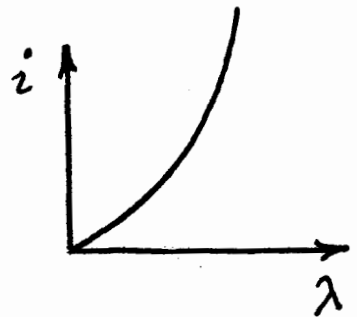
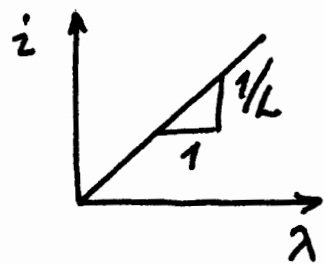
3-4

$$\frac{F}{\dot{x}} \rightarrow C, \quad F = \left(\frac{3EI}{L^3} \right) x.$$

3-5



3-6 $Li = \lambda = \int e dt$



$i = i(\lambda)$

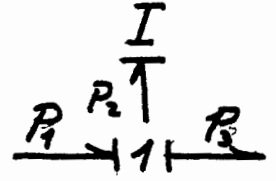
or $\lambda = \lambda(i)$

$$\frac{di}{dt} = \frac{di(\lambda)}{d\lambda} \cdot \frac{d\lambda}{dt}$$

$$= \frac{di(\lambda)}{d\lambda} \cdot e$$

$$\frac{d\lambda}{dt} = e = \frac{d\lambda(i)}{di} \cdot \frac{di}{dt}$$

3-7 $\bar{F} = m\bar{a} = m\dot{\bar{v}}$

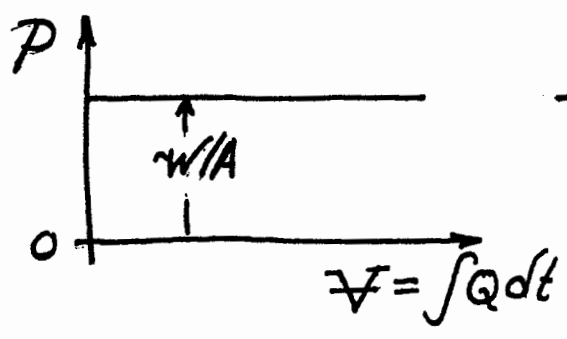


$$P_1 A - P_3 A = \rho A L \cdot \frac{dQ_2}{dt} A$$

or $P_1 - P_3 = \left(\frac{\rho L}{A}\right) \cdot \frac{dQ_2}{dt} = P_2$

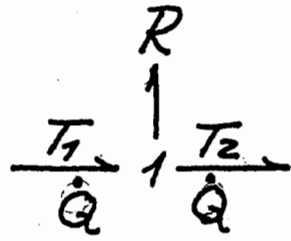
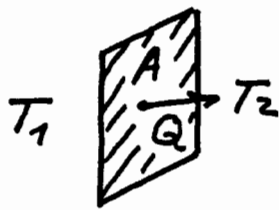
$$P_2 = \int (P_1 - P_3) dt = \frac{\rho L}{A} \cdot Q_2$$

3-8



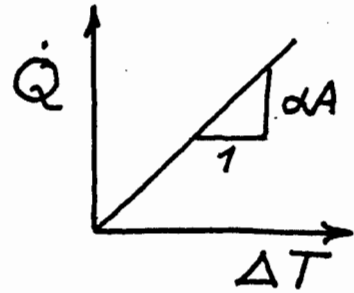
$$\frac{P}{Q} \rightarrow C$$

3-9



8

$$\dot{Q} = \alpha A (T_1 - T_2)$$



3-10

$$\tau = I \alpha, \quad p_{\tau} = \int \tau dt = I \omega$$

$$I = \frac{1}{2} m R^2 = \frac{1}{2} \rho \cdot \pi R^2 t \cdot R^2$$

$$= \frac{0.28 \pi \cdot 1 \cdot (5)^4}{2 \cdot 386} = 0.712 \text{ (lb s}^2 \text{ in)}$$

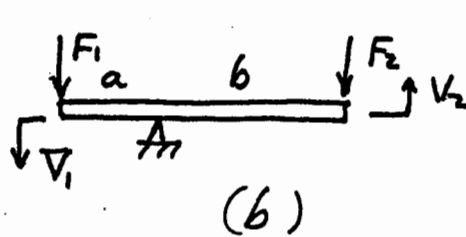
3-11

$$\frac{F}{V} = TF = \frac{P}{Q}, \quad \text{Area} = A$$

$$F = A P$$

$$AV = Q$$

3-12



$$\frac{F_1}{V_1} = TF = \frac{F_2}{V_2}$$

$$a F_1 = b F_2$$

$$\frac{V_1}{a} = \frac{V_2}{b}$$

(c)

$$\frac{\tau_1}{\omega_1} = TF = \frac{\tau_2}{\omega_2}$$

$$\omega_1 r_1 = \omega_2 r_2,$$

$$\tau_1 / r_1 = \tau_2 / r_2,$$

} r_1, r_2 radii of gears

3-13 Ang. Momentum = \bar{H} 9

$|\dot{\bar{H}}| \cong J\Omega$, torques associated

with F_1 and F_2 cause change in direction of \bar{H} not magnitude. Consider F_1 first; let shaft length be L . Torque is then $F_1 L$, $\dot{\bar{c}} = \dot{\bar{H}}$ means that tip of \bar{H} vector must move up with angular rate V_2/L . $|\dot{\bar{H}}| = J\Omega \cdot V_2/L = F_1 L$

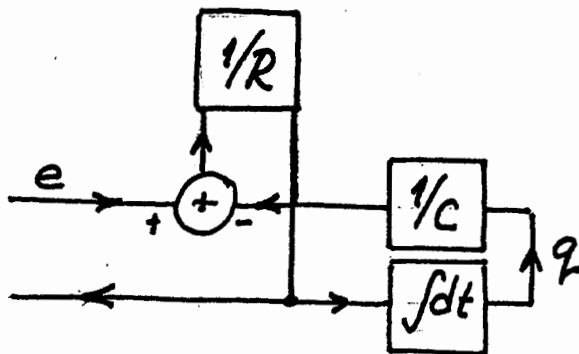
so

$$F_1 = \left(\frac{J\Omega}{L^2} \right) V_2$$

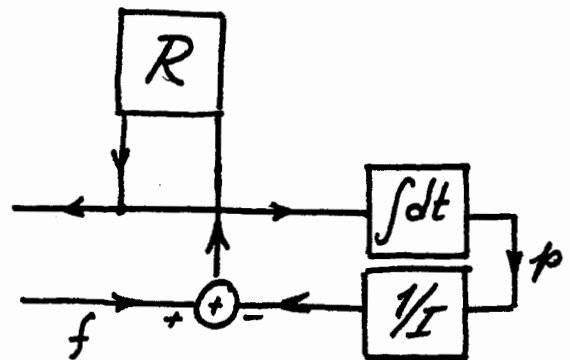
Similarly $F_2 = \left(\frac{J\Omega}{L^2} \right) V_1$

$$\left. \begin{array}{l} F_1 = \left(\frac{J\Omega}{L^2} \right) V_2 \\ F_2 = \left(\frac{J\Omega}{L^2} \right) V_1 \end{array} \right\} \frac{F_1}{V_1} GY \frac{F_2}{V_2}$$

3-14

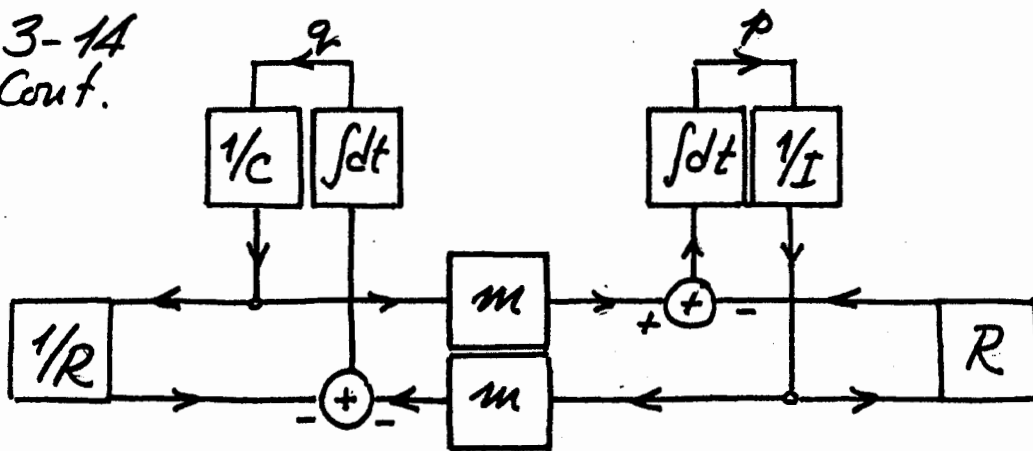


(a)

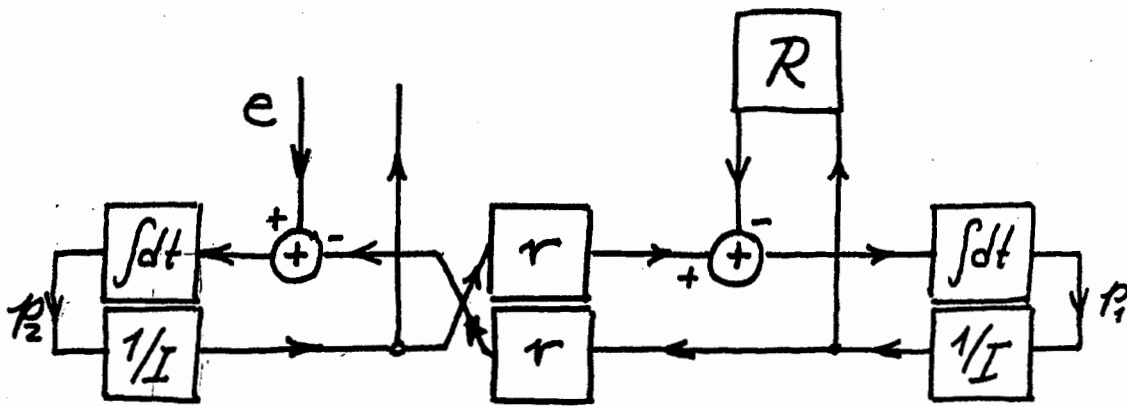


(b)

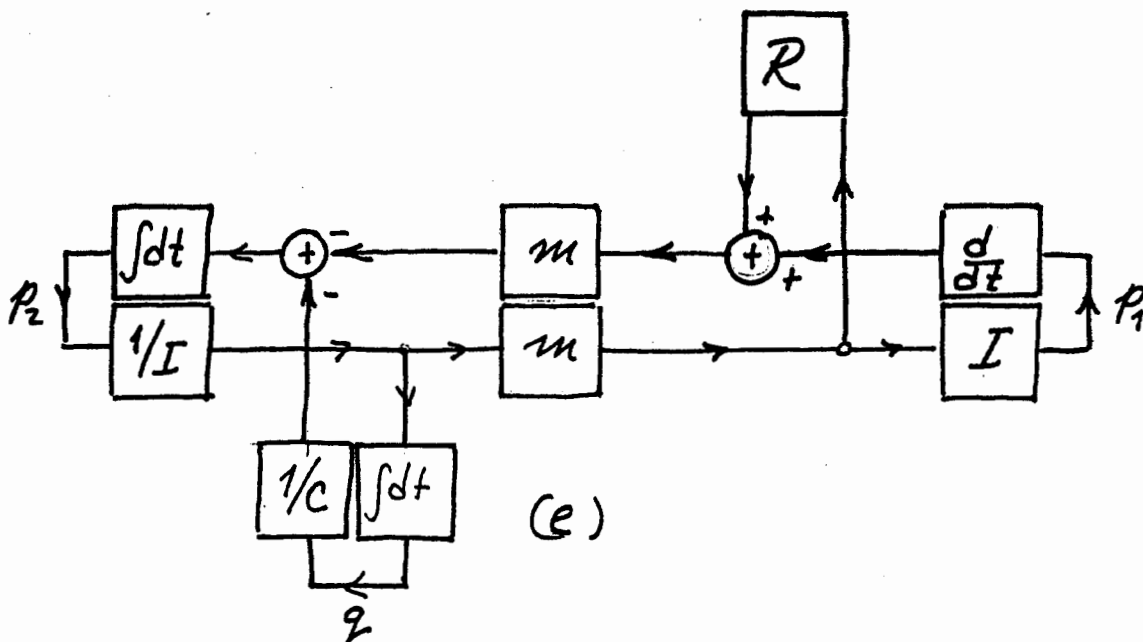
3-14
Cont.



(c)



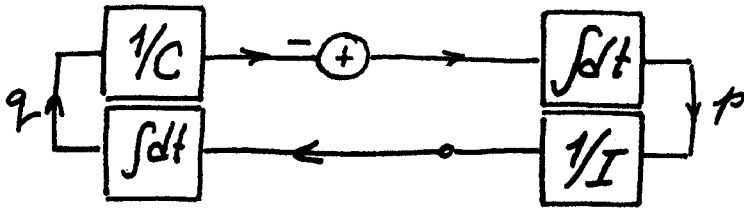
(d)



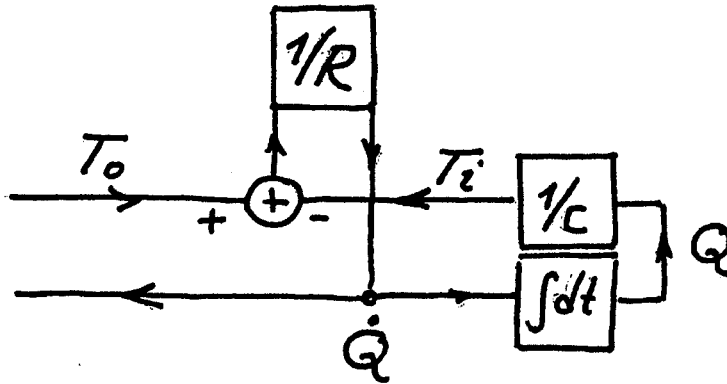
(e)

3-15

11



3-16



3-17

$$\frac{\tau}{\omega} \rightarrow TF \frac{F}{V} \quad \tau = r F$$

$$r \omega = V$$

3-18 $\frac{e}{i} \rightarrow G \ddot{y} \frac{F}{V} \rightarrow I$

$$e = TV = T \frac{p}{I} = \frac{T \int F dt}{I}$$

$$= \frac{T}{I} \int T i dt = \frac{T^2}{I} \int i dt = \frac{T^2}{I} \cdot q$$

so $\frac{1}{C} \leftrightarrow \frac{T^2}{I}$

3.19

$$P = P_0 \frac{V_0^n}{V^n} = \frac{P_0 V_0^n}{(V_0 - A_p X)^n} = \frac{P_0}{\left[1 - \frac{A_p X}{V_0}\right]^n}$$

P is absolute pressure:

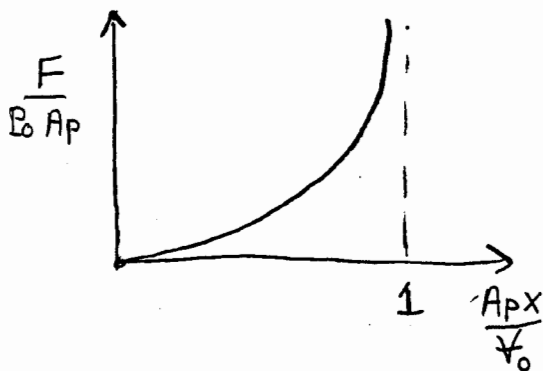
$P - P_0$ is gage pressure in cylinder,

$$(P - P_0) A_p = F$$

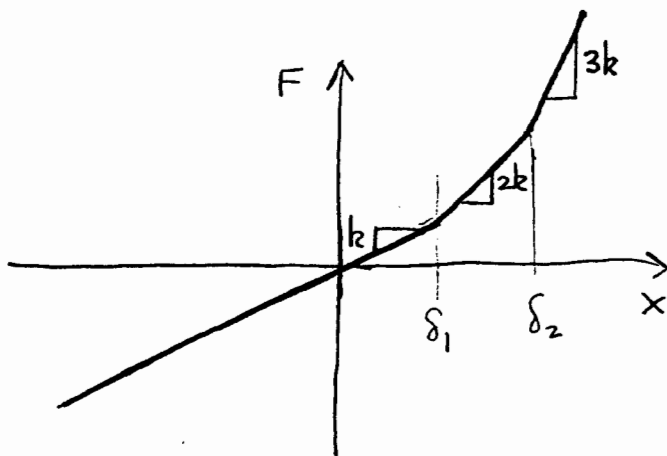
or

$$F = \left\{ \frac{P_0}{\left[1 - \frac{A_p X}{V_0}\right]^n} - P_0 \right\} A_p$$

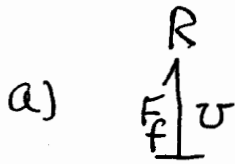
$$F = P_0 A_p \left[\left(1 - \frac{A_p X}{V_0}\right)^{-n} - 1 \right]$$



3.20



3.21

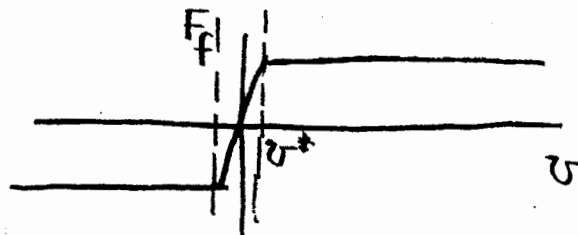


The only possible causality for the friction model shown in the problem is "effort" out, "flow" in.

For any specified velocity, F_f can be computed, but if F_f is specified, v is indeterminate.

- (b) If used simply as $F_f = \mu N$, then F_f will be applied when $v=0$, which is not correct. When $v=0$, the mass "sticks", and the friction force exactly balances all other forces on the mass. When the other forces exceed the "stick" force, then F_f returns to $F_f = \mu N$.

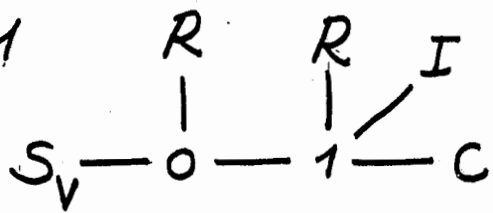
A possible change in the constitutive law might be,



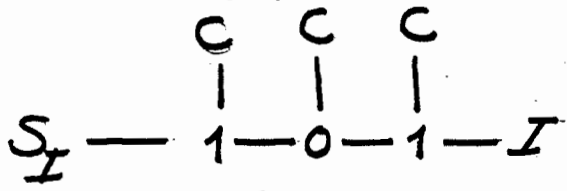
For v^* very small, the fundamental character of friction is maintained.

4-1

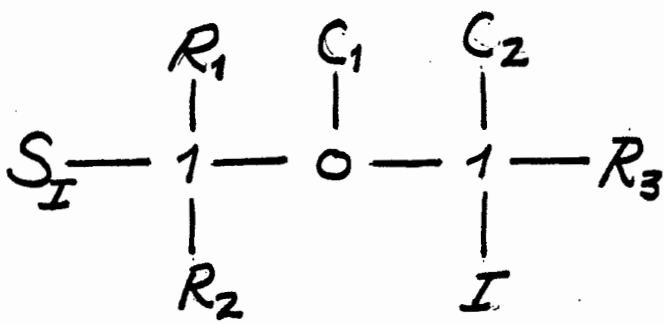
14



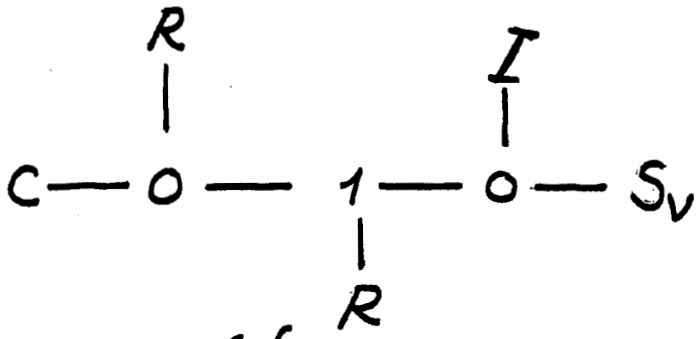
(a)



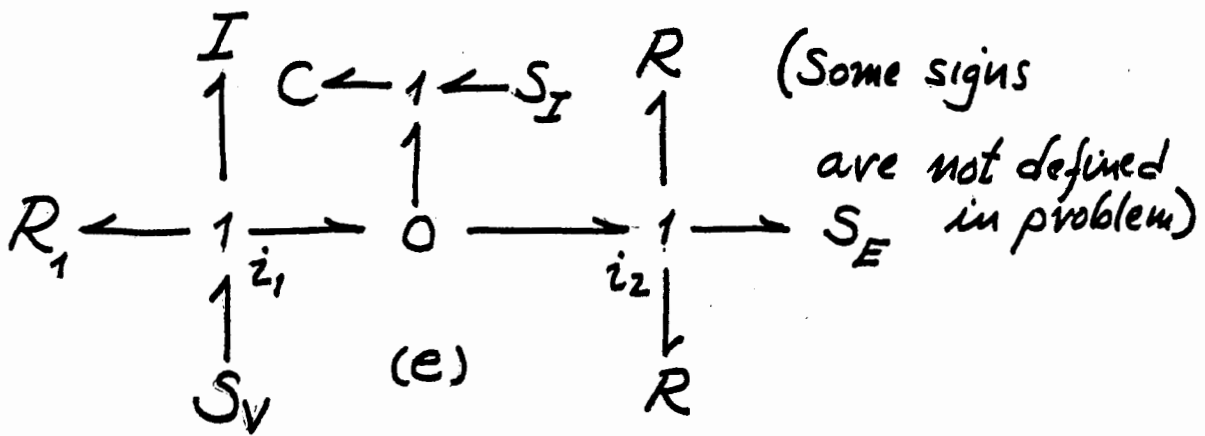
(b)



(c)



(d)



(e)

(Some signs are not defined in problem)