RESEARCH PAPER | Engineering | Volume : 3 | Issue : 5 | May 2013 | ISSN - 2249-555X

Simulation of the Dynamic Behaviour of LEC Based Controller to Ensure Operational Continuity of a Low Cost Automation Device Energised by Human Powered Flywheel Motor (HPFM)

KEYWORDS: HPFM, Finger Type Clutch, LEC based Controller

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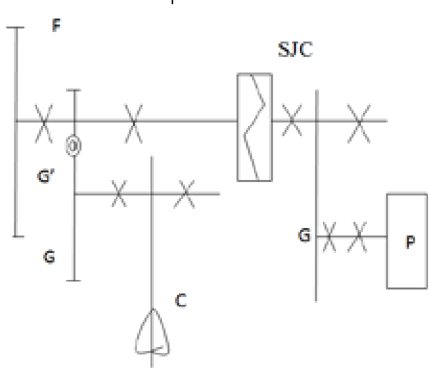
ABSTRACT *In our previous attempt stress was given on modeling and control of a phenomenon to enhance the operational reliability of a process unit. In that we have suggested that for enhancing the operational reliability we can use a LEC based controller. But the point of discussion was whether this controller will be capable of changing the control from one clutch to another in a shortest possible time may be in terms of μsec. In the present paper the solution of this problem is approached by replacing the motor connected to the clutches by their non-linear models. These models could be actuated by designing a linear electronic circuit simulating their behavior.*

A. HPFM Energized Process Machine:
Modak and his associates [1 &2] have developed a HPFM energized process machines for various applications [3&4] which are rural based and necessary to improve the life of the people of the third world. The machine consists of three subsystems namely, HPFM (Human Powered Flywheel Motor), comprising of a flywheel F, speeded up through speed amplification gears G' and a conventional bicycle operated by a human operator), Spiral jaw clutch SJC and torque amplification gear G &, Process unit. The schematic arrangement of the machine in its plan view is as described in the Fig 1.



Fig(1):Schematic Arrangement of the machine

B. Machine parameters, Operation & Operational Characteristics:
A flywheel is arranged (1 m rim Φ & 10 cm rim width, 2cm rim thickness) in which a man pumps the energy at a rate convenient to him (human power approximately 0.13 hp continuous duty) for about a minute's time by operating a pedaling mechanism through a speed rise gear pair having speed ratio equal to 4.5:1.At the end of a minute's duration, flywheel is accelerated to about 700-800 rpm speed. Then the pedaling is stopped, spiral jaw clutch is engaged and the kinetic energy stored in the flywheel is communicated to a process unit through a pair of torque amplification gear [G=4.1].

Process unit exhausts the energy stored in the flywheel during a very small time 3-10 seconds (Process HP up to 3) depending on the resistance of the process unit. Thus the processes which could be of an intermittent nature and needing power far in excess of human capacity can be energized manually by such machine concept. The concept is tried for various applications such as water lifting, wood turning, wood strip cutting, potter's wheel, brick making, algae formation etc.

C. The need for development of Torsionally Flexible Clutches:
During the period of clutch engagement the mechanical system is subjected to severe shock due to instantaneous momentum exchange. On account of this, spiral jaw clutch is subjected to unpredictable malfunction. This is one of the serious drawbacks of this system.

The basic reason for the spiral jaw clutch failure is, it does not have torsional flexibility very much needed in this situation. A clutch with torsional flexibility will permit momentum exchange at a slow rate. The exhaustive literature survey [5&6] shows that clutches with torsional flexibility are not developed excepting the attempts of Modak and his research scholars [7 to 10]. These types of clutches permit momentum exchange at a fairly slow rate. Though literature indicates development of plate clutches with axial flexibility [5&6] these are not useful for the present purpose.

The present research addresses to the generation of design data through the development of generalized experimental model for a finger tip load and subsequent vibrations of fingers in a finger type torsionally flexible clutch.It also details inclusion of LEC based [14] controller for getting operational velocity of process unit.

II. FINGER TYPE TORSIONALLY FLEXIBLE CLUTCHES:
A. Construction:
Clutch comprises of two members. Member I is connected to the flywheel shaft through splines. Multiple numbers of fingers 3, 4, 6 are provided integral with the hub of the member I. Fingers have rectangular section as shown. Member II is carrying jaws J (3, 4, 6, depending on the number of fingers provided, but no. of jaws equals no. of fingers). Member II is integral with the jaw which provides drive to the process unit.

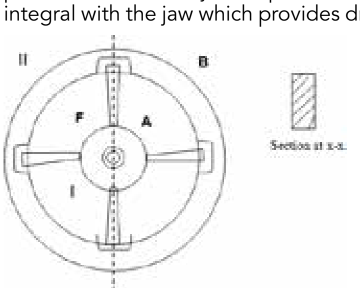


Fig (2): Schematic of Finger Type Clutch

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B. Working:

When the flywheel F of the main machine is being accelerated, member I is so located on the flywheel shaft that fingers F do not have contact with jaws J and the clutch is in disengaged position. After getting the desired flywheel speed, peddling is stopped, member I is axially slided over flywheel shaft. The moment when fingers dash on jaws J, the process of momentum transfer from flywheel shaft to load shaft commences. Fingers F, structurally behave as short cantilever having spring like action because of its elasticity. Thus the fingers provide relative angular displacement between load shaft and the flywheel shaft during the period of clutch engagement. This is how torsional flexibility is provided by this type of clutch.

III. DYNAMICS OF THE CLUTCH:

Clutch engagement duration is defined to be the time interval in which the flywheel shaft and the load shaft attain the identical speeds after first contact of fingers with the jaws. During this period, flywheel shaft speed& load shaft speed are different. Applying De-Alembert's formulation, the general equations of motion for the flywheel shaft and the load shaft respectively can be derived as –

$$-I_F\,\ddot{\theta}_F - b_F\,\dot{\theta}_F - K\alpha t = 0 \qquad \text{(I)}$$

$$K\alpha t - I_L\,\ddot{\theta}_L - b_L\,\dot{\theta}_L - T_L = 0 \qquad \text{(II)}$$

In equations (I) & (II) I_F and I_L are moment of inertia's of flywheel and load shaft. b_F and b_L are bearing friction torque constants of flywheel and load shaft respectively. T_L= Load Torque on the load shaft as imposed by the process resistance. K= Stiffness of the fingers, α_t = instantaneous slope of the finger at its fixed end. Careful examination of Equas (I) & (II) shows that these equations are not solvable unless experimental feedback of the behavior of the system is known.

Hence, it amounts to establishing the generalized experimental data based models for & or eventually of finger tip load and finger vibrations which can be considered to be a function of&

IV. DESIGN OF EXPERIMENTATION

Generalized experimental models for W & S (maximum stress induced in the fingers due to finger load and vibrations) are established adopting methodology of experimentation [13].

As per this methodology all independent parameters and/or physical quantities are varied over widest possible range. Huge response data is collected. Based on this entire data Generalized models are formed. The detailed steps are (1) Establishing the dimensional equations of the mechanics of a clutch (2) Test envelopes, Test points and Test sequence (3) Design and building up of an experimental set up ,(4) Performing experimentation (5) Publication of Experimental Data(6)Establishing the exact mathematical function of the dimensional equation based on Experimental Data.

A. Dimensional Equations:

Applying Buckingham-II theorem and a Raleigh's method [13], the dimensional equations for (WD/T_L) & (SD^3/T_L) are formulated as under:

$$\frac{WD}{T_L} = f\left[\left(\frac{I_F}{T_L*t^2}\right)\cdot\left(\frac{I_L}{T_L*t^2}\right) \square \left(\frac{b_F}{T_L*t}\right) \square \left(\frac{b_L}{T_L*t}\right)\cdot \left(\frac{R}{D}\right)\cdot\left(\frac{W}{D}\right)\cdot\left(\frac{d}{D}\right)\cdot\left(\frac{ED^3}{T_L}\right).(U).(N).\left(\frac{gt^2}{D}\right)\right] \qquad \text{(III)}$$

$$\frac{SD^3}{T_L} = f\left[\left(\frac{I_F}{T_L*t^2}\right)\cdot\left(\frac{I_L}{T_L*t^2}\right) \square \left(\frac{b_F}{T_L*t}\right) \square \left(\frac{b_L}{T_L*t}\right)\cdot \left(\frac{R}{D}\right)\cdot\left(\frac{W}{D}\right)\cdot\left(\frac{d}{D}\right)\cdot\left(\frac{ED^3}{T_L}\right).(U).(N).\left(\frac{gt^2}{D}\right)\right] \qquad \text{(IV)}$$

Where

I_F=Moment of Inertia of Flywheel Shaft

I_L=Moment of Inertia of Load Shaft

T_L=Load Torque on Load Shaft

R= Radius of the finger tip measured from centre O

D= Diameter of the finger

W= Width of the finger

d= Depth of the finger

E=Modulus of Elasticity of Finger Material

t=Time

g= Acceleration due to gravity

D. Generalised Models:

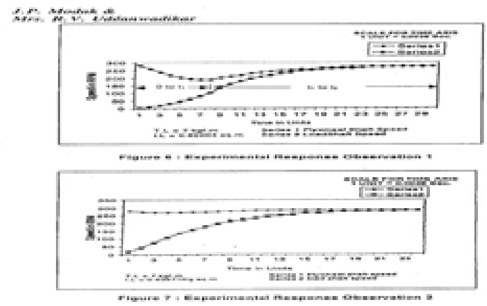

Fig 4 Dynamic Response

Using the principle of (I) force balance & (II) energy balance variation of W verses time is established as depicted in Fig 5 by the curves (a) & (b) respectively corresponding to the experimental data of Fig 4.

Fig 5 shows erratic variation of W. This appears to be because of unpredictable bearing friction torque. This situation may be perhaps due to much more severe loading on the flywheel shaft & load shaft. Hence hereafter the anticipated finger tip load based on force balance concept is no more considered. The vibration response is evaluated approximating the finger as a single degree of freedom spring-mass-damper system.' ζ ' is assumed to be 0.9 in view of the fact that the system damping is due to the friction between the finger and the jaws due to axial siding of fingers under severe tip load during the period of engagement is not reflected in Equations I &II . To account for this ζ is assumed to be very high and of the order of 0.9.

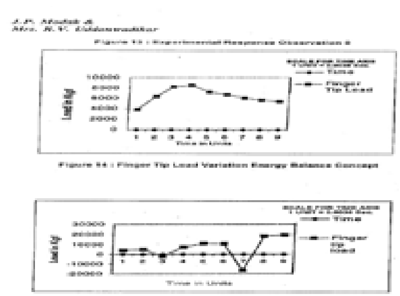

Fig 5 (a) Finger Tip load (W) variation with time (t)

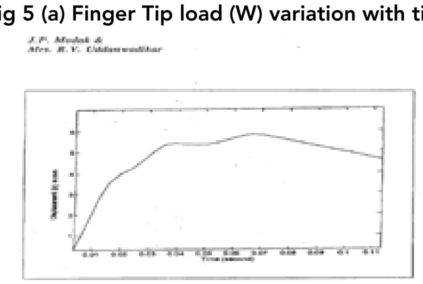

Fig 5(b): Vibration response of fingers

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Fig 5 (a) shows the variation of finger tip load W vs time and fig5 (b) shows vibration response of fingers . The above variations are obtained for the entire experimental information gathered in this work. Based on this research, exact mathematical form of dimensional Equations (I&II) are obtained after performing necessary mathematical operations & are reproduced below.

$$=-0.063[(IF/T_L t^2)^{1.569}(I_L/T_L t^2)^{-0.07}(RwdN/D^3)^{-0.031}$$

$$(ED^3/T_L)^{-0.237}\ (gt^2/D)^{1.512}] \qquad (V)$$

$$(SD^3/T_L) = -0.1438\ [(IF/T_L t^2)^{0.0027}\ (IL/T_L t^2)^{0.0289}\ (RwdN/D^3)^{0.2766}\ (ED^3/T_L)^{0.0638}\ (gt^2/D)^{-0.128}] \qquad (VI)$$

V. DISCUSSION OF RESULTS

In this section it is proposed to discuss the mechanics of energy transfer from flywheel shaft to the load shaft through this finger type clutch.

Fig 4 shows that in the early part of clutch engagement there is a drop in flywheel speed but afterwards it keeps on increasing right from t=t3 . however keeps on increasing right right from t=t1 till t=t3 with some spells of time in which is reducing. Similar behavior is observed for entire variation of independent quantities.

This indicates that immediately upon colliding of fingers on the jaws considerable energy is stored in the finger in the form of elastic strain energy. Maximum energy should be stored during $\delta t=t_2-t_1$. In fact this gets confirmed from subsequent calculations of estimation of finger tip load W. Curve of Fig5(b) shows maximum W at $t=t_2$ when is minimum .As W is maximum at $t=t_2$ tip load determination is maximum at t=t2 , hence is the maximum storage of strain energy. Interestingly, it is observed that $\delta t=t_2 - t_1$ varies with system independent variables. Infact, generalized experimental model should be formulated for δt which will reveal the influence of independent quantities on δt & hence on impact phenomenon.

Further interesting observation is at times is increasing and at times reducing during $\delta t=t_3 - t_2$.This solidly confirms that load shaft at times demands energy from fingers and at times pumps the energy in the fingers. This should cause severe superimposed oscillation over and above that caused by variation of W vs time. Finger vibrations during $\delta t=t_3 - t_2$ and also during $\delta t=t_2 - t_1$ as W is rising with steep gradient, fingers are subjected to transient vibrations . On the whole therefore fingers are subjected to vibrations which need estimation of stress under vibrations.

Curve a of Fig 5 shows higher value of W as compared to those of curve b. This is obvious because frictional energy loss is not assumed for information presented by curve a. Bearing friction phenomenon appears to be pretty erratic because curve b shows W=0 for some instants. This is because at those instants frictional resistance itself is enough to impose necessary retardation even if not reaching to limiting value of friction.

The experimental set up may need additional instrumentation to solidify ascertain the influence of friction.

In the estimation of vibration response over simplifying assumptions are made which are as under (1) Entire finger mass is assumed to be at the tip. (2) Finger elasticity is assumed to be linear. In fact it may be non-linear leading to the sever finger oscillations.(3) ζ is assumed to be 0.9 , a very high value without which stress under vibrations could not have been a practically acceptable figure. ζ =0.9 may be justified because considerable axial frictional rub or the fingers during $\delta t = t_3 - t_1$ is not modeled. Of course this will be only possible by sophisticated instrumentation like telemetry.

VI. FORMULATION OF THE PROPOSED MODEL:

Equation (VI) can be rearranged to deduce necessary number of fingers for the specified T_L and specified material of the fingers. However since the reliability of estimation of clutch performance based on equation (VI) being not adequate it is necessary to change the mathematical form of the model. What follows is this change in the form of the model.

$$S=f(I_F,I_L,T_L,t,g,E,R,w,d,N,D) \qquad (VII)$$

In Equation (VII)

I_F= Moment of Inertia of Flywheel Shaft

I_L=Moment of Inertia of Load Shaft

The most compact form of the dimensional equation for equation (VII), could be as under as per Raleigh's Method.

$$SD^3/T_L= f\ ((I_F.R.w.d.N.E.g.t^2) / (I_L.T_L.D)) \qquad (VIII)$$

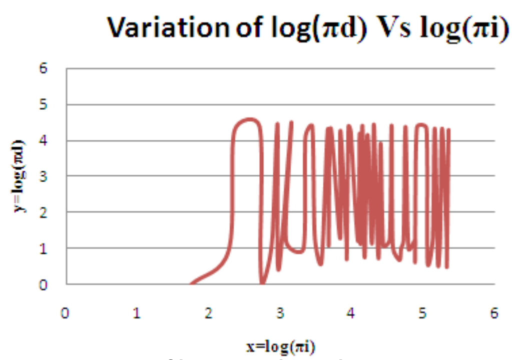
Let Equation(VIII) be written in the form

$$\pi_d=f(\pi_i) \qquad (VIII\text{-}A)$$

Based on the experimental data, the variation $\log(\pi_d)$ Vs $\log(\pi_i)$ is as shown in the Fig 6.

Figure (6) indicates that the variation of log (π_d) Vs log (π_i) is having several peaks and crusts. Hence, the most appropriate form of model could be either the higher order polynomial form or the sinusoidal form. The polynomial form of model deduced is as under:

$$Y=1.0e+006 *(0.0002x^9 -0.0479x^8+0.1244x^7-2.2663x^6+3.6081x^5-1.7813x^4+1.6889x^3-1.6133x^2+0.3033x-0.0162) \qquad (IX)$$



Fig(6)Variation of log(πi) Vs log(πd)

This form is having better reliability as compared to the reliability of the model depicted in Equation (VI). Further, this form of the model is more convenient from the point of view of designing controller based on linear electronic circuits[14].

The matter presented through articles I to VI given above is already presented in the earlier paper [16] of the present authors ,however it is again reproduced here as a matter of ready reference because this background has considerable relevance with the matter of this paper which follows here onwards.

VII.LEC BASED CONTROLLER

Equations (IX) simulate operation of the system under consideration.

Now as and when, the operating conditions change i.e. inputs change, it becomes inevitable that responses also change. But, it is necessary to maintain responses within cer-

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tain limits.

For example in this case the load torque T_L changes say increases, it will increases the finger tip load (W) and in turn it will increase induced stress (S). But S should not increase beyond a certain limit allowable stress in bending. Only way, to do this in this case will be to change (i) either cross section of fingers (ii) increase D (iii) change the material of the fingers (iv) change I_F or I_L (v) change no. of fingers, N.

Amongst these alternatives easiest is to change the number of fingers N. This amounts to provision of more than one clutch. So one may provide three clutches with N=3, 4 and 6.

Further, there has to be a provision of (i) measuring T_L (ii) comparing measured T_L with designed T_L (iii) if T_L actual is more than designed then, there has to be a provision of measuring how much in excess is T_L. Then subsequently there has to be a provision to decide a clutch with number of N to be engaged. Finally, a necessary physical system is to be provided to shift energy flow from flywheel F to process unit through a proper clutch.

Entire system including a controller schematically is shown in Figure 7. This schematic represents (i) main system to be controlled and (ii) the Linear Electronic Circuit Based controller. This controller comprises of T_M the torque meter, A/D converter, Linear Electronic Circuit calculating necessary number of fingers in a clutch corresponding to load torque T_L , selection of motors M1, M2, M3, corresponding linkages converting rotary motion into rectilinear motion of slider of dog clutches DC1, DC2, DC3. It is obvious that A/D converter,LEC based minicomputer estimating necessary number of fingers ,N and selector of M1, M2, M3, could be on one chip denoted here as LEC BASED CONTROLLER.

VIII . MODEL
According to equation (VIII-A) ,i.e.

$SD^3/T_L = f((I_F.R.w.d.N.E.g.t^2) / (I_L.T_L.D))$ (VIII-A)

Now put $y=SD^3/T_L$ and $x= (I_F.R.w.d.N.E.g.t^2) / (I_L.T_L.D)$ in equation(VIII-A), then Equation(VIII-A) changes to

$y= f(x)$ (XI)

Plotting y and x on ordinate and abscissa respectively, one gets graphic plot of variation of y as x varies. This graphic plot is converted into polynomial form of model as under

$Y=A_0+ A_1x+ A_2x^2+ A_3x^3+$………………. (XII)

CASE (i):
Now only considering second term of the model i.e .Equation (XII), one gets

$y=A_1x$ (XII-A)

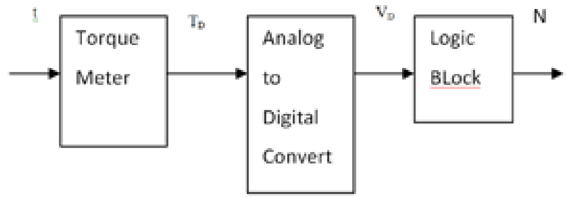
Substituting in Equation (XII-A) and substituting for original variables for y and x, equation (XII-A) takes the following form

$SD^3/T_L= A_1((I_F.R.w.d.N.E.g.t^2) / (I_L.T_L.D))$

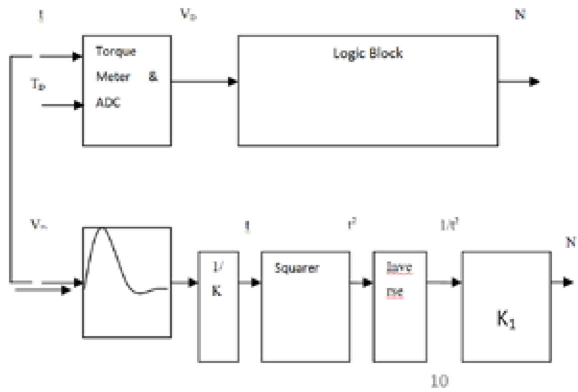
$N=(1/A_1). (SD^3/T_L). ((I_L.T_L.D)/ (I_F.R.w.d.E.g.t^2))$

$N=(1/A_1). (SD^3). ((I_L.D)/ (I_F.R.w.d.E.g.t^2))$ (XII.B)

Here't' is a specific time instant during the period of clutch engagement. However, from the point of view of the physics of the system, it is T_L which in fact decides N for specified material of the finger i.e. the parameters S and E .In fact instead of't', V_D (digital voltage)as detailed below should be substituted for 't'.



Fig(8).
Perhaps it could be as under



Fig(9):
In the above block

$K_1=(SD^3.I_L.D)/(A_1.I_F.R.w.d.g.E)$ (XII.C)

CASE (ii):
Now only considering third term of the model i.Equation (XII), and let us denote this contribution to total y as y'' , then

$y''= A_2x^2$ (XIII)

Upon substituting for $y''=SD^3/T_L$ and

$x== (I_F.R.w.d.N.E.g.t^2) / (I_L.T_L.D)$

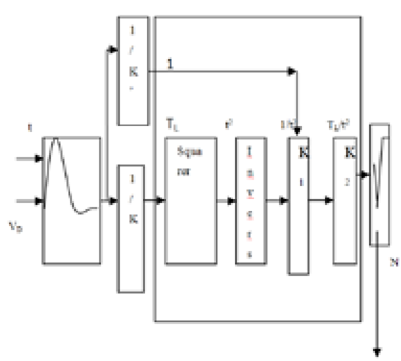
Equation (XIII) would take the form

$SD^3/T_L= A_2 ((I_F.R.w.d.N.E.g.t^2) / (I_L.T_L.D))^2$

$N^2= (1/A_2). (SD^3/T_L). ((I_L.T_L.D)/ (I_F.R.w.d.E.g.t^2))^2$

$N=[(S.D^5T_LI_L^2)/A_2(I_FR.w.d.g.t^2.E)^2]^{1/2}$ (XIII.A)

Here also 't' is a specific time instant during the period of clutch engagement. However from the point of view of the physics of the system .it is T_L which in fact decides N for specified material of the finger i.e. the parameters S and E .In fact instead of't', V_D (digital voltage) as detailed below should be substituted for 't'.



Fig(10):

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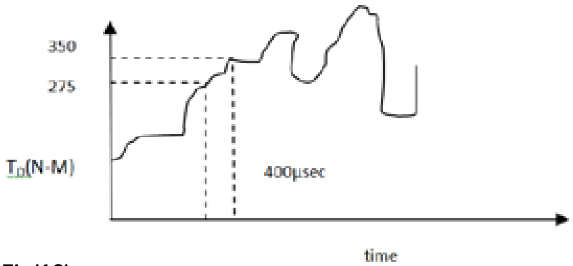
Let $K_2=(SD^5 I_L)/(A_2 I_F (Rwd) .g.E\ )$

Similarly for 3rd ,4th ,5th -------9th component of polynomial form one should develop complete block diagram. [Refer Figure (11)]



Fig(12):

Supposing fig(12) describes demand torque (T_D)variation of the process unit over a substantial time of operation. Let t = t' be the time instant at which the demand torque is say 275 N-m and the necessary number of fingers are say 3.1 as calculated by model portion of LEC controller. As seen in fig(12), this torque gets increased from 275 N-m to 350 N-m in the next 400 µsec and is subsequently remaining constant for say next 1000 µsec and then subjected to decline. In other words from time t= t' onwards we need to have a finger type clutch with 4 number of fingers. Hence at t= t' the process of power flow has to change from clutch with 3 number of fingers to clutch with 4 number of fingers. For this purpose LEC is subjected to a ramp change of torque from 275 N-m to 350 N-m ie. The ramp change of 75 N-m in about 100 µsec is required.

It is becoming essential to confirm whether the designed LEC would be capable of meeting this requirement. The issue could be looked upon in two ways 1) either assume that at t= t' there is a step change of demand torque of 75 N-m or there is imposition of ramp change of torque of 75 N-m in 100 µsec. The more preferred option would be to consider as if there is a step change in demand torque of 75 N-m. What then needs to be ascertained is

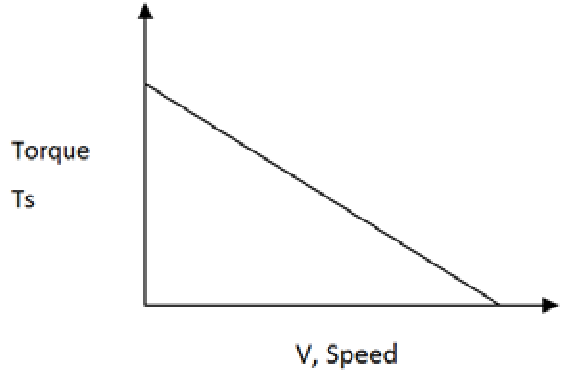
1] What will be the settling time?
2] What is the maximum overshoot?
3] What is the steady state error of the portion of the system comprising of battery L3 providing the energy to motor M3, pinion P3, rack R3, friction between 1]bearings, 2] rack and pinion , 3] slight jerk at the time of engagement of rack with DC ,4] sliding friction between rack 4 and the slot in the dog clutch. The governing equation of this Electromechanical system can be considered to be as under:

$$Ts - I' (d^2\theta/dt^2) - b\ (d\theta/dt) - T_L = 0 \qquad (A)$$

In the above equation Ts is the electromagnetic torque, generated by the fractional horse power M3 at instant t = t', I' is the total moment of inertia of the mechanical system (comprising of inertia of rotor of the motor, motor shaft, pinion P3, rack R3), b represents viscous coefficient of friction. Load torque in this specific application could be that due to rubbing and/ or dashing action of the bottom of the rack with circular surface of the stationary dog clutch.

Though equation (A) is an ordinary linear differential equation of the second order with constant coefficient it may be that the coefficient b' may be an intermediate case of viscous friction and coulomb friction and the external resisting torque to the movement of the rack may be represented by a exponentially decaying function, some constant (kt^{-n}) where 'n' is a positive integer. In that case equation (A) would turn out to be ordinary non-linear differential equation of the form as under

$$Ts - I' (d^2\theta/dt^2) - b\ (d\theta/dt)^{n1} - k\ t^{-n2} = 0 \qquad (B)$$



Fig(13):

Fig (13) describes a probable motor which could be used for these applications. One such motor is a dc servo motor with its torque speed characteristics as shown in fig (13). The analytical representation of this torque speed characteristics could be given by the equation

Ts=C-m N

Which could be put in an alternative form as

$$Ts = C - m\ (w/2\pi) \qquad (C)$$

Substituting for Ts from equation (C) into equation (B), the equation for the transient electromechanical dynamics of the drive part of the LEC can be simulated by the equation

$$\{C - m\ (w/2\pi)\} - I'(dw/dt) - b\ (w)^{n1} - k\ t^{-n2} = 0 \qquad (D)$$

In equation (D) C, m, I, b, k, n1, n2 are constants. The equation (D) is obviously non – linear with fairly high degree of non – linearity because in equation (D) there are two non – linear terms

1) $b\ (w)^{n1}$ and
2) $K\ t^{-n2}$.

In order to reduce the complexity of solution to this non-linear equation alternatively the form of the equation (D) can be made in stages slightly less non-linear by altering it as under:

$$\{C - m\ (w/2\pi)\} - I'\ (dw/dt) - b\ (w) - k\ t^{-n2} = 0 \qquad (D1)$$

In equation (D1) the frictional torque between motor shaft and its bearings, rack – pinion and their bearings is assumed to be proportional to angular speed. This is of course true in the event of the use of light engine oil as a lubricant rather than Grease and/or dry lubricant (carbon based lubricant) as a lubricant. However the resisting torque during the period of clutch engagement being un – avoidable that needs to be considered and presented by the term (Kt^{-n2}). The interpretation of the term (Kt^{-n2}) is at time t = 0^+, a resisting torque would be very high which is an expected reality.

Hence equation (D1) can be considered as a mathematical model representing somewhat lesser non-linearity than the equation (D1) which comprises of both the non-linear terms viz (1) $b\ w^{n1}$ and (2) $k\ t^{-n2}$.

An approach to solution of these non-linear differential equations will be two ways

1) Refer the treatment of solution to ordinary non-linear differential equations or
2) Alternatively solution to these non-linear differential

equations by (DC analog computers ie. What is known as a modern electronics using OP-AMP or LECs) or else

3) by adding LEC which is simulating above detailed non-linear differential equation which in turn is simulating the phenomenon of engagement of a new dog clutch.

VIII. CONCLUSION

This paper is an extension of our previous paper[16] in which we are suggesting an approach to develop a controller which will be capable of providing instantaneous transfer of control from one clutch to another (ie.in μsec). For this we have proposed to design one more Linear Electronic Circuit depicting/simulating the actual non-linear behavior of the motor used in the schematic of our LEC Based Controller.

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