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Modeling of Natural Frequency and Amplitude on a Non-Driven Vibrating Potato Harvester Shank

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SUMMARY

A non driven vibrating potato harvester has been attempted to design to produce exactly same type of vibration function with the forced vibratory soil digging shank. The analysis examines the rotation angle and position characteristics of the shank as the soil force that acts on the blade. Because the soil digging shank was assumed to be clamped with an elastic material on to a solid potato harvester frame, the shank was assumed in a certain vibration in the soil during to soil digging. A system of expression was generated to describe the general motion of the shank, and equations of the motion were solved by analytically in concerning with necessary assumptions.

Key words: Natural Frequency, Amplitude, Vibration, Potato Harvester, Modeling.

ÖZET

Tahriksiz Patates Hasat Makinalarının Titreşiminde Doğal Frekansın ve Genliğin Modellenmesi

Titreşimli toprak işleme aletlerinin toprakta yaptığı titreşim hareketine benzer bir titreşim yapabilecek bir patates hasat makinasına ait patates sökme organı tasarlanmaya çalışılmıştır. Analiz, sökücü bıçağa etki eden toprak kuvvetlerinden hareketle sökücü organın dönme açısı ile durum karakteristiklerini

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saptar. Aletin toprak kazan kısmına ait taşıyıcı organın, elastik bir materyal ile birlikte ana şaseye sabitleştirilmiş olması yüzünden toprak kazıcının toprak işleme süresi boyunca belirli bir titreşim gösterdiği kabul edilmiştir. Sistem kazıcı ayağın genel hareket eşitlikleri ile tanımlanmış ve eşitlikler denklemlerin sınır koşulları dikkate alınarak analitik yolla çözümlenmiştir.

Anahtar sözcükler: Doğal Frekans, Genlik, Titreşim, Patates Hasat Makinası, Modelleme.

INTRODUCTION

A non-driven vibrating soil digger for proposed use on a potato harvester generates natural vibration frequency when it moves through the soil with a certain velocity. The main discussion point is to determine the natural frequency and amplitude of a non-driven vibrating blade oscillation in a specific soil conditions. Generally, potato harvesters are designed by using a mechanical driver which forces the vibration of the digger blades and tines (Hammerle, 1970). Therefore, their construction are complex and are resulted higher design cost compared with the non-driven vibrating machines.

Most non-driven vibrating potato harvesters are usually designed by using S-type shanks which serve as their own spring (Johnson, 1974). Elastomer clamps are also widely used as non-driven vibrating soil digging devices in order to generate similar dynamic situations.

In this work, all analysis and calculations were performed on a elastomer clamp type soil digging blade in the sandy loam. The dynamic soil resistance force was derived as a function of the soil and tool parameters such as soil-tool friction, working depth and velocity. This force causes the rotation of the machine axis from x-y to x'-y' (Figure 1c). When the system rotates at certain rotation angle, the elastic material is squeezed by the clamps and it acts as a helical spring with a spring constant k. Therefore, deformed elastomer stores energy that creates vibration on the system. As a result, the solution of the problem takes place for defining dynamic soil forces on the blade and deriving equations of motion about the system under these forces.

MATERIAL AND METHOD

The soil-tool system is under influence of several factors which are related either soil or tool and soil-tool interaction (Sial, 1977). In the model, the soil depth, soil cohesion, the soil internal friction angle, the soil bulk density and the soil surface forward failure angle are used as soil parameters. Similarly, the tool sharpness angle, blade dimension (thickness, width, length) and soil-blade friction angle are also used as blade parameters. The main approach is to make some assumptions in order to drive equations of the dynamic soil forces acting on the blade in specific soil condition. These equations are solved analytically by concerning soil and tool parameters.

The soil forces on inclined tool was given in Figure 1a. By the force equilibrium concept, the total force acting on the center of the blade in horizontal direction is,

$$R = N_0 Sin\delta + \mu' N_o Cos\delta + k' b \qquad (1)$$

where R is draft force, μ' is soil metal friction coefficient, N_0 the normal load on the inclined tool, k' is pure cutting resistance of soil per unit width, b is tool width and δ is lift angle of the tool (Gill and Vanderberg, 1968). The normal load on a inclined soil tillage tool was described as a function of the weight of the soil segment (G) and other soil and tool parameters such as cohesion of the soil, lift angle of the blade (δ), angle of forward failure surface angle (β) etc.

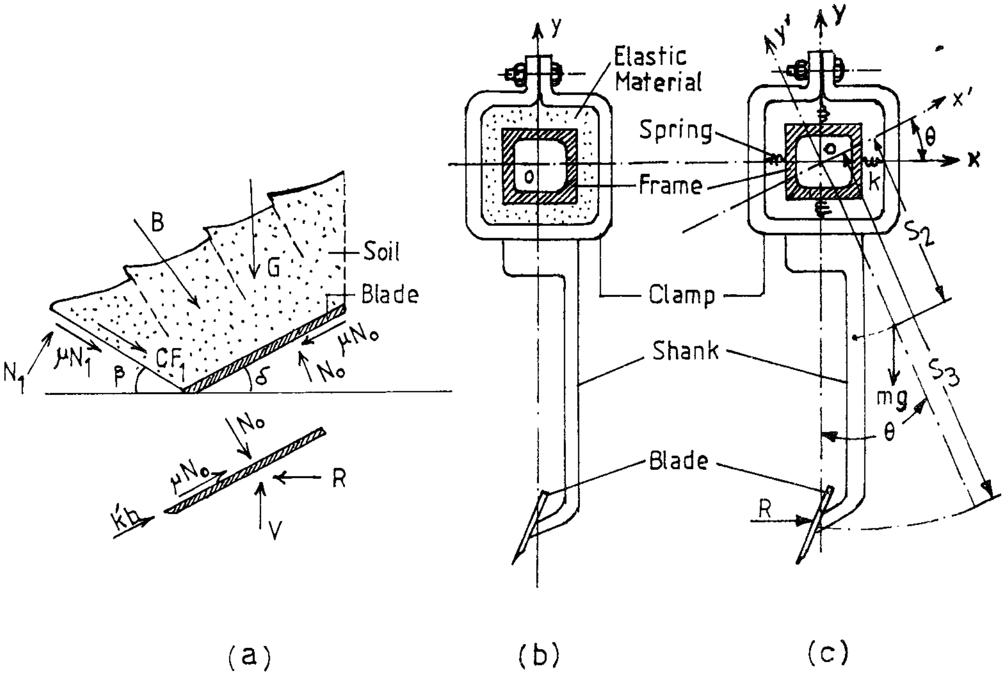

Figure 1
(a) Soil Forces Acting to inclined Tine, (b) A Non-Driven Vibrating Potato Harvester Shank, (c) Simulation of the Shank

The second approach for finding natural frequency and the amplitude of the system is to simulate elastomer clamp as a set of helical spring which is assumed to be fixed in the clamp. From the Figure 1b and Figure 1c, the total moment at the point O is,

$$J_0 \frac{d\theta(t)^2}{dt^2} = RS_3 Cos(\theta) - 2[kS_1 Sin(\theta)]\,[S_1 Cos(\theta)] - 9.8\,mg\,S_2\,Sin(\theta)] \tag{2}$$

where J_0 is the mass moment of inertia of the system, k is spring constant, m is system mass, θ is rotation angle on the x-y axis. For small oscillation that means the small rotation angle, we can assume $Sin\theta = 0$ and $Cos\theta = 1$ then the equation becomes,

$$\frac{d\theta(t)^2}{dt^2} + \left[\frac{2kS_1^2 + 9.8\,mg\,S_2}{J_0}\right]\theta = \frac{RS_3}{J_0} \tag{3}$$

If the system has a damping, the analysis should be done by considering the damping coefficient of the system. Therefore, dynamic equations of the system con be rewritten by including the damping coefficient which is a function of several factors such as soil blade friction, soil moisture content, plasticity and adhesion. In general, if D and λ represent the damping coefficient and the damping ratio, the equation can be reduced in small fraction. Therefore,

$$J_0 \frac{d\theta(t)^2}{dt^2} - \left(\frac{d}{J_0}\right)\frac{d\theta(t)}{dt} = \frac{RS_3}{J_0} Cos(\theta) - \frac{2kS_1^2}{J_0} Sin(\theta)\,Cos(\theta) \tag{4}$$

$$- \frac{9.8mg\,S_2}{J_0} Sin(\theta)$$

where $2\lambda = D/J_0$. Assuming $Sin\theta = 0$ and $Cos\theta = 1$ for the small oscillation (Zill, 1982), the final equation of the motion is turn out to second order linear differential equation with force function. That is also,

$$\frac{d\theta(t)^2}{dt^2} + 2\lambda\frac{d\theta(t)}{dt} + \omega^2\theta = \frac{RS_3}{J_0} \tag{5}$$

where the ω^2 is given by the following formula;

$$\omega^2 = \frac{2kS_1^2 + 9.8mg\,S_2}{J_0} \tag{6}$$

The differential equation of the system solution was performed with computer program that is written in FORTRAN. The flowchart of the program is given in Figure 2. In the Appendix 1, the program is also given.

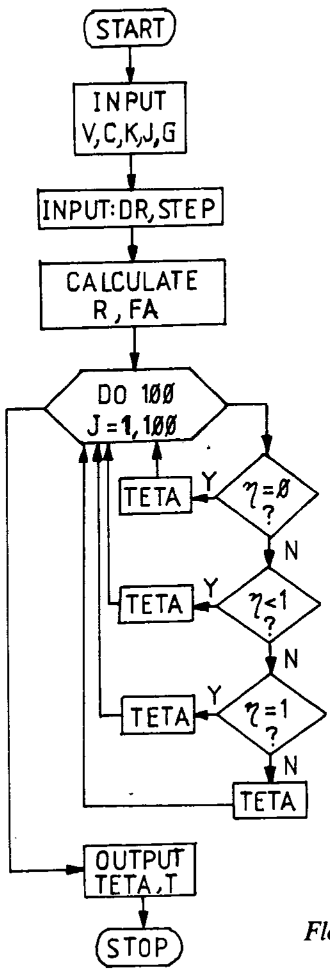

Figure 2
Flowchart of the Computer Program

DISCUSSION AND RESULT

The solution of the linear second order differential equations with force function consist on the particular and the homogenous solution with necessary initial and boundary condition. For calculations, dimension and the total weight of the shank and clamp have been arbitrarily chosen. The elastic material in the clamp was assumed as a set of helical spring which has a spring constant as 10^5 N/m. The spring mass and the soil adhesion force were neglected for the calculations. The initial conditions were assumed as, $\theta(0) = 0$ and $d\theta(0)/dt = 0$. For the calculation, the blade cutting angle, soil cohesion coefficient, the soil internal friction angle and soil-metal friction angle were chosen as 16°, 700 N/m, 25° and 20°, respectively. The soil force calculation were performed for the sandy loam, which has 1660 N/m density, and the blade velocity was 5 km per hour in that soil.

The biggest difficulty was to determine the soil damping coefficient, because it is a function of the soil plasticity, the soil moisture content. However, the equations of motion about the system were solved using several damping ratio which are shown in Figure 3. The frequency model has maximum amplitude around 0.07 radian if the damping coefficient is 0.25. The system can reach steady state in 2 second. The lower value in damping ratio produced high amplitude and frequency as it was expected. The critical value for the damping ration is 1, and there is no oscillation in that case. The system is also able to produce a sinus cycles which has no steady state equilibrium if the damping of the system is not taking account.

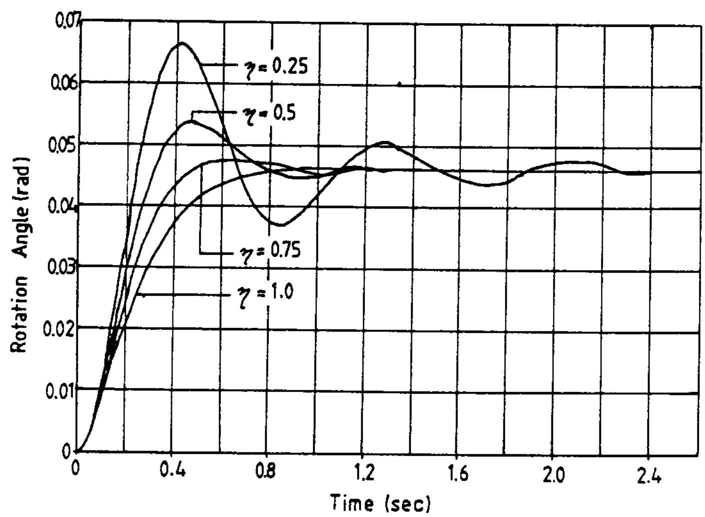

Figure 3

System Vibration for Critical and Underdamping Situation

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Appendix 1. FORTRAN Program

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C************************************************************
C     CALCULATION OF NATURAL FREQUENCY AND AMPLITUDE
C     A NON-VIBRATING POTATO HARVESTER SHANK EITHER
C     DAMPED OR UNDAMPED CONDITION
C
C                Dr. Rasim OKURSOY
C                ULUDAG UNIVERSITESI ZIRAAT FAKULTESI
C                TARIM MAKINALARI BOLUMU
C
C     NOTE: ALL UNITS ARE IN METRIC SYSTEM
C************************************************************
      REAL J0,L,LAM,M,KS,K,MU1,MU2,N1
      DATA VC,PI,WD,ZERO,BW,BL/5.,3.1416,0.25,0.,0.55,0.11/
      DATA GRAV,GAMA,C,STEP/9.81,1660.,700.,0.025/
      DATA M,S1,S2,S3,J0/1.1151,0.03,0.035,0.03,3.68/
      OPEN(2,FILE='VIBRA,OUT',STATUS='NEW')
      WRITE(*,*)'ENTER SPRING CONSTANT AND DAMPING RATIO'
      READ(*,*)KS,DR
      WRITE(*,*)'----------------------------------------'
      V=(100./360)*VC
      K=PI/180.
      RHO=16.*K
      RHO1=20.*K
      PHI=25.*K
      BETA=(PI/4.)-(PHI/2.)
      MU1=TAN(RHO1)
      MU2=TAN(PHI)
C*********** PRINT INPUT DATA *******************************
```

```
      WRITE(2,*)'BETA,V',BETA,V
      WRITE(2,*)'-------------------------------------------'
      WRITE(2,*)'DR,STEP',DR,STEP
      B1=(WD*SIN(RHO+BETA))/(SIN(BETA))
      B2=COS(RHO+BETA)+(TAN(RHO)*SIN(RHO+BETA)
      G=GRAV*GAMA*BW*B1*)BL+(WD/(2.*SIN(BETA)))*B2)
      C1=(GAMA/GRAV)*BW*B1*(BL+(WD/(2.*SIN(BETA)))*B2)
      C2=SIN(RHO)/SIN(RHO+BETA)
      FA=GRAW*(C1*C2)
      AF=(WD*BW)/SIN(BETA)
      A1=(COS(RHO)-MU1*SIN(RHO))*(SIN(BETA)+MU2*COS(BETA))
      A2=(-(SIN(RHO)+MU1*COS(RHO)))*(COS(BETA)-MU2*SIN(BETA))
      DELTA=A1-A2
      A3=(SIN(BETA)+MU2*COS(BETA))*(G+(C*AF+FA)*SIN(BETA))
      A4=(COS(BETA)-MU2*SIN(BETA))*(-(C*AF+FA)*COS(BETA))
      DELTA1=A3-A4
      N1=DELTA1/DELTA
      R=(N1*(SIN(RHO)+MU1*COS(RHO)))/2
C********** SOIL FORCE CALCULATION ************************
      ETA=ABS(DR)
      OM=SQRT((2.*KS*S1**2+M*GRAV*S2*GRAV)/J0)
      LAM=ETA*OM
      VAL=(R*S3/(J0*OM**2.))
      DO 100 J=1,100
      T=J*STEP
      EF=EXP(-LAM*T)
      IF(ETA.EQ.ZERO) GO TO 75
      IF(ETA.LT.1) GO TO 76
      IF(ETA.EQ.1) GO TO 77
      IF(ETA.GT.1) GO TO 78
 76   RT2=SQRT(OM**2.-LAM**2.)
      TETA=(-VAL)*EF*(COS(RT2*T)+(LAM/RT2)*SIN(RT2*T))+VAL
      GO TO 80
 77   TETA=(-VAL)*EF*(1.+LAM*T)+VAL
      GO TO 80
 78   RT1=SQRT(LAM**2.-OM**2.)
      VAL1=LAM+RT1
      VAL2=LAM-RT1
      VAL3=(-VAL1*EXP(RT1*T)+VAL2*EXP(-RT1*T))
      TETA=(VAL/(2.*RT1))*EF*(VAL3)+VAL
      GO TO 80
 75   TETA=VAL*(1.-COS(OM*T))
C********** PRINT OUTPUT DATA ******************************
 80   WRITE(2,35) J,T,TETA
 35   FORMAT(1X,I3,2(1X,F15,3))
100   CONTINUE
      CLOSE(2,STATUS='KEEP')
      STOP
      END
```