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MILITARY TECHNICAL COLLEGE

CAIRO - EGYPT

# ON MATERIALS HANDLING MACHINES
## Minimizing Adhesive Tractive Forces, Using GPOT

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ABSTRACT

The ability of application of GPOT(Geometric Programming Optimization Technique) to design some members of materials handling machines with preventing the tipping over phenomenon is discussed. The application is done on FEL(Front-End-Loaders) and finds out the values of the design variables which are the nominal value of minimum machine weight and of maximum lifting load. These values are used to compute the minimum value of the adhesive tractive forces with avoiding the tipping over of the machine. The signomial GPOT is applied to construct design nomograms by which the designer of FEL can easily estimate values of the design variables and make correlation between these values and other parameter value. Then the preliminary information concern the method of determining stochastically the optimum real values of such design variables is given.

## 1. INTRODUCTION

To the author's knowledge no work on the optimum design of materials handling machines have been appeared so far while few papers, some of which are listed in [1,2], have been reported on dynamic force analysis in such machines taking into account the dynamic stability of the machine. Moreover, the researchs should be directed to the field of optimum design of materials handling machines considering the imposed constraints to fill the gap in such field. Therefore, this work aims to extend the author's work [1] and points out the application of signomial GPOT on FEL. The objective function and the constraint are minimizing the adhesive tractive force and avoiding the turning over of the machine respectively. The signomial GPOT which is reported in [3,4] is used, since the objective function has negative term.

## 2. FORMULATIONS

The most of the following formulations have been already given in [1]. These formulations are of FEL which is shown in Fig.1, and can be modi-

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fied for other material handling machines.

A- Power required by the machine P is estimated by

$$P = C\, F_t\, V$$

where C : Constant depends upon the transmission efficiency,
$F_t$ : Tractive Forces and
V : Machine Speed.

B- The Tractive Forces $F_t$ is given by

$$F_t = F_1 W + F_2 Q + F_3 W V + F_4 Q V + F_5 V^2 \qquad (1)$$

where $F_1, \ldots F_5$ : Coefficients can be determined as follows,

$F_1 = K_1 \cos\beta + \sin\beta - V_s/t_a$

$F_2 = K_2 \cos\alpha + \sin\alpha - V_s/t_a$

$F_3 = K_2 \cos\beta + 1/t_a$

$F_4 = K_2 \cos\alpha + 1/t_a$

$F_5 = K_w A$

$K_1$ & $K_2$ : Coefficients [2]

$K_w$ : Coefficient of resistance of air(wind)

A : Front end area of the machine

$t_a$ : Time for acceleration

$V_s$ : Starting velocity of the machine

$\alpha = \beta + \theta$ ( see Fig.1)

W and Q are machine weight and lifting load respectively.

Then
$$P = P_1 W V + P_2 Q V + P_3 W V^2 + P_4 Q V^2 + P_5 V^3 \qquad (2)$$

where $P_1, \ldots P_5$ : Coefficients can be determined by

$P_i = F_i\, C$ , $i = 1, 2, \ldots 5$.

C- The Adhesive Tractive Forces $F_d$ is computed by

$$F_d = (K_a - K_r)\, R_r$$

where $K_a$ : Adhesion coefficient

$K_r$ : Rolling coefficient $= (K_1 + K_2)\, V$ (see[2] )

$R_r$ : Reaction Force at point of contact between the rear wheel and the road

Note that $F_d$ is given for rear wheel drive condition, and that $R_r$ is calculated by

$$R_r = R_1 W A_f + R_2 W h_c + R_3 W h_c V + R_4 V^2 h_w + R_5 h_q Q V + R_6 h_q Q + R_7 \ell Q V - R_8 \ell Q$$

where $R_1, \ldots R_8$ : Coefficicnts and estimated by

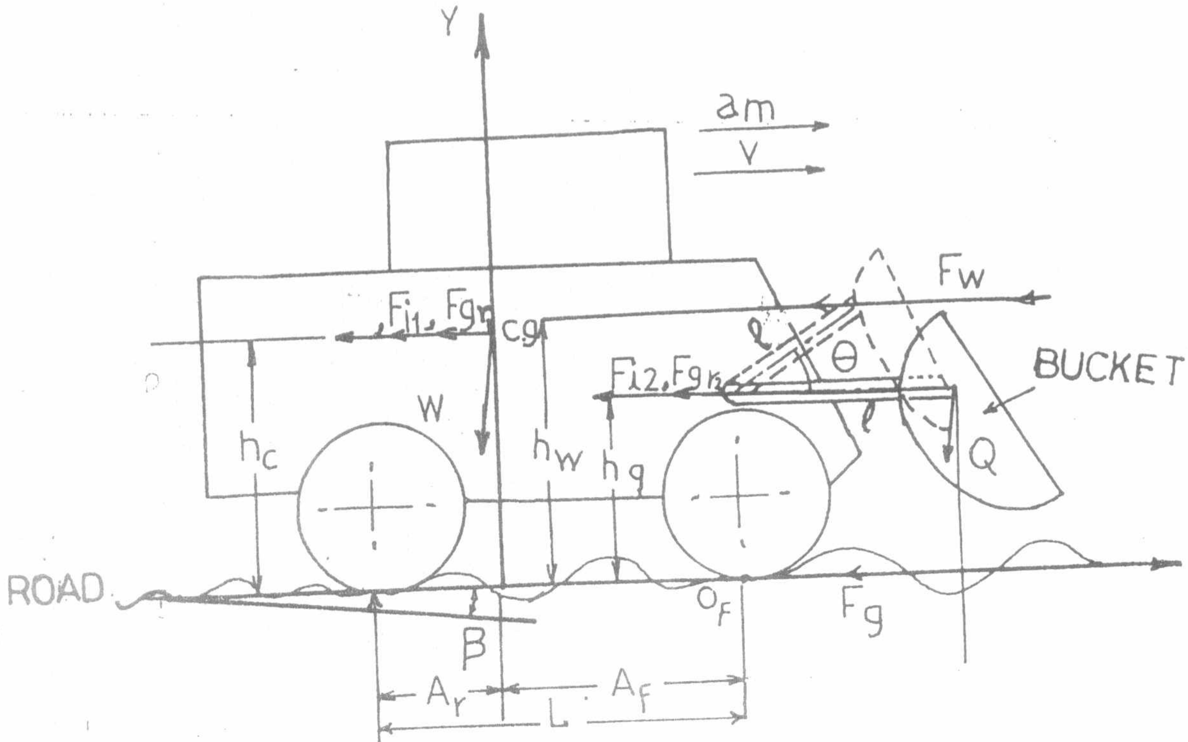
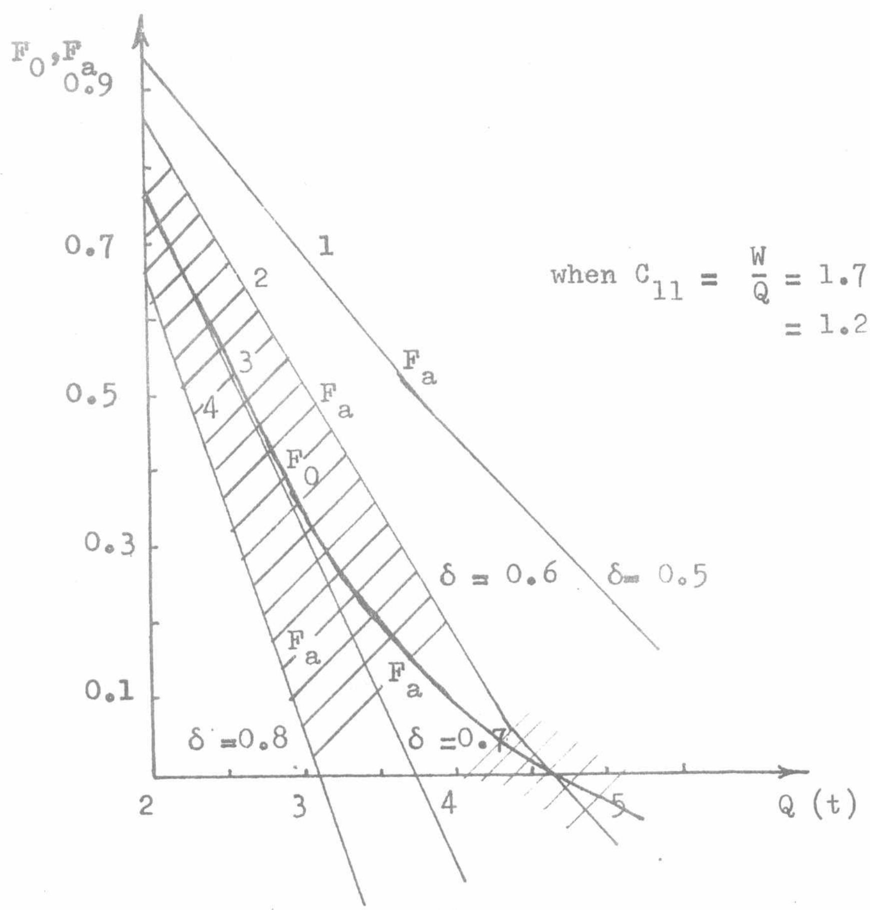



Fig .1




when $C_{11} = \frac{W}{Q} = 1.7 \sim 2, \ \delta \cong 0.7$

$= 1.2 \sim 1.5, \ \delta \cong 0.6$

Fig. 2

$$R_1 = \cos \beta / L$$
$$R_2 = (\sin \beta - V_s/t_a)/L$$
$$R_3 = R_5 = 1/L\ t_a$$
$$R_4 = K_w\ A/L$$
$$R_6 = (\sin \alpha - V_s/t_a)/L$$
$$R_7 = \sin \theta / L\ t_a$$
$$R_8 = [\cos \alpha \cos \theta - \sin \theta (\sin \alpha - V_s/t_a)]/L$$

Substituting these values into equation of $F_d$, we get

$$F_d = d_1 W A_f + d_2 W h_c + d_3 W h_c V + d_4 h_w V^2 + d_5 h_q Q V + d_6 h_q Q + d_7 \ell Q V - d_8 \ell Q - d_9 W A_f V - d_{10} W h_c V^2 - d_{11} h_w V^3 - d_{12} h_q Q V^2 - d_{13} \ell Q V^2 \quad (3)$$

where $d_1 = C_0 R_1$, $d_2 = C_0 R_2$, $d_3 = C_0 R_3 - K_2 R_2$,
$d_4 = C_0 R_4$, $d_5 = C_0 R_5 - K_2 R_6$, $d_6 = C_0 R_6$,
$d_7 = C_0 R_7 + K_2 R_8$, $d_8 = C_0 R_8$, $d_9 = K_2 R_1$, $C_0 = K_a + K_1$
$d_{10} = K_2 R_3$, $d_{11} = K_2 R_4$, $d_{12} = K_2 R_5$ and $d_{13} = K_2 R_7$

### 2.1 Limit Design Equations

As stated in [1]

1. To prevent tipping over of the machine, the load stability factor $K_t$ must be greater than unity and is governed by

$$K_t = [T_1 W A_f + T_2 W h_c + T_3 W h_c V + T_4 h_w V^2 + T_5 h_q Q V + T_6 h_q Q + T_7 \ell Q V + T_8 \ell Q] / T_9 \ell Q \quad (4)$$

2. To eliminate the machine sliding, the sliding factor $K_s$ must be greater than one and is given by

$$K_s = \mu (W \cos \beta + Q \cos \alpha) / F_t \quad (5)$$

where $T_1, .. T_9$ : Coefficients can be estimated by
$T_i = R_i L$, $i = 1,2,..7$, $T_8 = \sin \theta (\sin \alpha - V_s/t_a)$,
$T_9 = \cos \alpha \cos \theta$, $\mu$ : Coefficient of friction and
for other notations, the figure 1 can be referred.

### 2.2 Standard GPOT

In order to use the GPOT which has been stated in [3,4], the following concepts must be taken into account, 6

1. The objective function and the constraints must be in either posynomial or signomial form,
2. The costraint must be equal to or less than unity,
3. The degree of difficulty should be equal to or greater than zero and restricted to be as small as possible and
4. The costraints of the dual objective function must be satisfied.

In addition, the signomial GPOT which is applied here exactly can be consulted in [3] . To achieve these concepts, the previous equations are posynomial except eqn (3) of $F_d$ which is signomial and the $K_t$, eqn(4), and $K_s$, eqn(5), can be rearranged to be $G_1$ and $G_2$ respectively as,

$$G_1 = C_{11} W^{-1} Q A_f^{-1} 1 - C_{12} h_c V A_f^{-1} - C_{13} h_c A_f^{-1} - C_{14} W^{-1} A_f^{-1} h_w V^2 - C_{15} Q W^{-1} V A_f^{-1} h_q - C_{16} Q W^{-1} A_f^{-1} h_q - C_{17} Q V W^{-1} A_f^{-1} \ell \leq 1 \qquad (6)$$

$$G_2 = - C_{21} W Q^{-1} + C_{22} W Q^{-1} V + C_{23} V + C_{24} Q^{-1} V^2 \leq 1 \qquad (7)$$

These become signomial forms with the following coefficients

$C_{11} = ( T_9 - T_8 )/T1$ , $C_{12} = T_2/T_1$ , $C_{13} = T_3/T1$ , ....., $C_{17} = T_7/T_1$ , $C_{21} = ( S_1 - F_1)/C_{02}$ , $C_{22} = F_3/C_{02}$ , $C_{23} = F_4/C_{02}$ , $C_{24} = F_5/C_{02}$ , $C_{02} = S_2 - F_2$ , $S_1 = \mu \cos \beta$ , $S_2 = \mu \cos \alpha$

Since the first case of this study is to investigate the optimum values of the machine weight W and the lifting load Q, the number of terms must be decreased to be three. This gives the optimization problem zero degree of difficulty, accordingly, the following is done

$$F_d = K_{01} W - K_{02} Q + C_d \qquad (8)$$

$$G_1 = K_{11} W Q^{-1} + K_{12} Q^{-1} \geq 1 \qquad (9)$$

where $K_{01} = ( d_1 - d_9 V ) A_f + ( d_2 + d_3 V - d_{10} V^2) h_c$ ,

$K_{02} = ( d_8 - d_7 V + d_{13} V^2) \ell - ( d_6 + d_5 V - d_{12} V^2) h_q$

$C_d = ( d_4 - d_{11} V) V^2 h_w$

$K_{11} = T_1 A_f + ( T_2 + T_3 V ) h_c /C_t$

$K_{12} = T_4 V^2 h_w/ C_t$ , $C_t = ( T_9 - T_8 - T_7 V) \ell - (T_6 + T_5 V) h_q$

The coefficient of air resistance and loader speed are small, therefore, the wind force can be neglected. Hence the terms $C_d$ of eqn (8) and $C_{12}$ of eqn (9) become zero. Consequently, the objective function $F_d$ and the constraint $G_1$ can be restated as

$$F_d = K_{01}\ W - K_{02}\ Q \qquad (10)$$

and $$G_1 = K_{11}\ W\ Q^{-1} \geqslant 1 \qquad (11)$$

To enable the application of GPOT, the objective function is taken as

$$F_{ob} = F_d^2 = K_{01}^2\ W^2 - 2\ K_{01}\ K_{02}\ W\ Q + K_{02}^2\ Q^2 \qquad (12)$$

Let $$F_{ob} = C_{01}\ W^2 - C_{02}\ W\ Q^{\delta} \qquad (13)$$

and $$G_1 = C_{11}\ Q\ W^{-1} \leqslant 1 \qquad (14)$$

where $C_{01} = K_{01}^2$ , $C_{02} = 2\ K_{01}\ K_{02}$ and $C_{11} = 1/K_{11}$

Not that the power of Q, $\delta$ , in eqn (13) is assumed to be as ,
if $Q > 1$, $\delta < 1$ ( from 0.5 to 0.8) and if $Q < 1$, $\delta > 1$ ( from 1.2 to 1.6)
(see disoussion), and that $K_t$ is much significant than $K_s$, thus it is considered and $K_s$ is neglected as constraint.
Then, the optimization problem is that finding W and Q which minimize $F_{ob}$,eqn(13) and satisfy $G_1$, eqn(14). To find these optimum values $W^*$, $Q^*$, the constraints of the dual objective function are

1. The normality condition is

$$x_{01} - x_{02} = \sigma_0\ (= \pm 1) \qquad (15)$$

2. The orthogonality conditions are

$$x_{01} - x_{02} - x_{11} = 0 \qquad (16)$$

$$-\delta\ x_{02} + x_{11} = 0 \qquad (17)$$

3. Then, the nonnegativity condition is

$$x_{10} = x_{11} > 0 \qquad (18)$$

Solving Eqns (15),(16) and (17), we obtain the dual design variables $x_{01}$ $x_{02}$ and $x_{11}$ as

$$x_{01} = \sigma_0\ (1 + \delta)/(\delta - 1)$$

$$x_{02} = 2\ \sigma_0\ /(\ \delta - 1\ ) \qquad (19)$$

$$x_{11} = x_{10} = 2\ \sigma_0\ \delta/(\ \delta - 1)$$

Once these values are known, the minimum objective function can be computed by

$$f_d^* = \sigma_0 \left[ (C_{01}/x_{01})^{x_{01}}\ (C_{02}/x_{02})^{-x_{02}} (C_{11})^{x_{11}} \right]^{\sigma_0} \qquad (20)$$

The final step is to determine the optimum values of machine weight $W^*$

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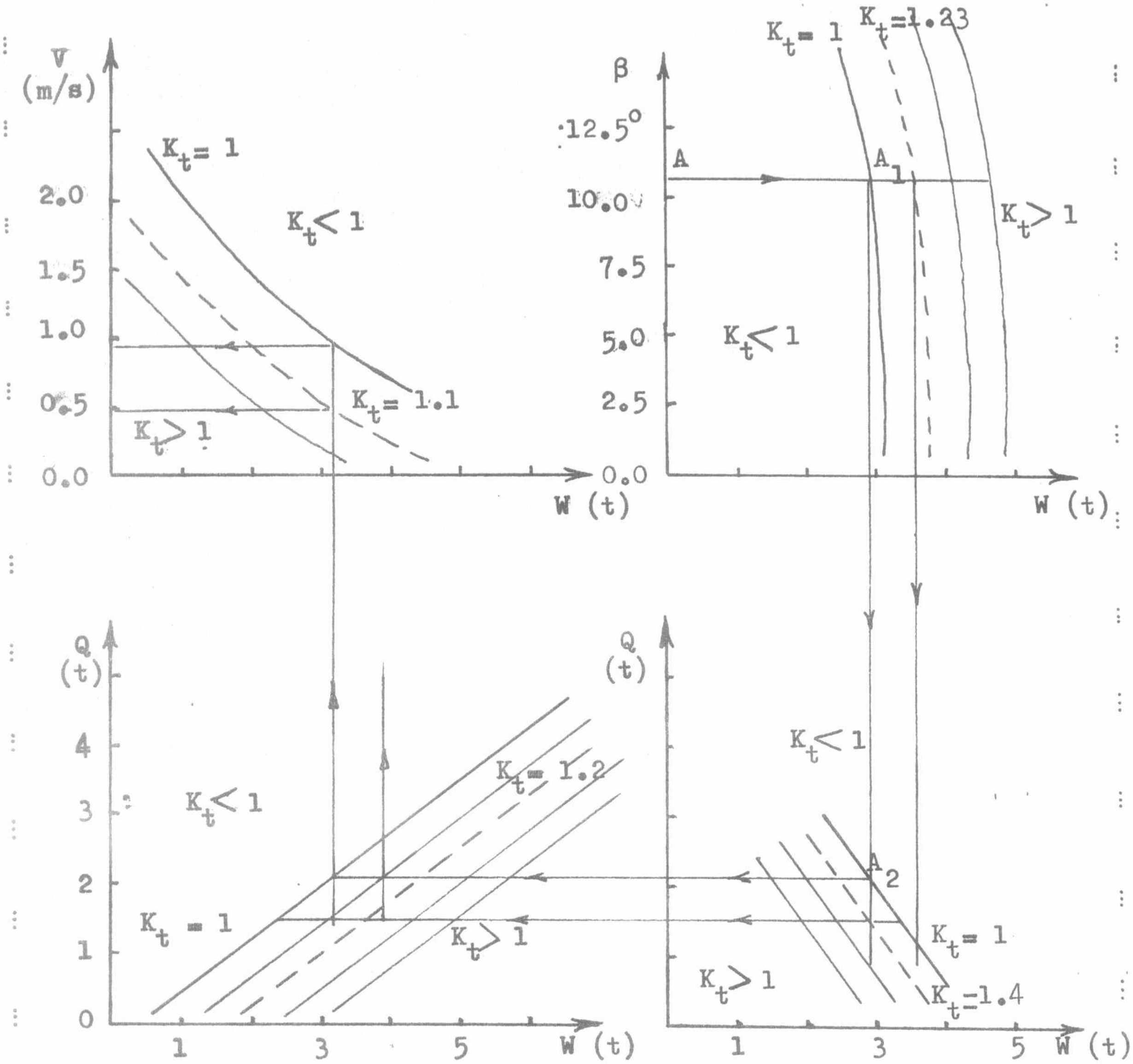


Fig. 3 Design Nomogram For β & V

N.B. $K_t$ > 1 Feasible Design Region

$K_t$ < 1 Avoiding Design Region

$K_t$ =1 Critical Condition (Tipping Load)

$K_t$ = 1~1.4 Optimum Design Domain

$F_d$ = 0.1~0.22 ton

and lifting load $Q^*$ by

$$W^* = \sqrt{x_{01} \, f_d^* / C_{01}} \quad \text{and}$$

$$Q^* = W^*/C_{11} \tag{21}$$

It is clear that these valus are function of other parameters such as $A_f$, $h_c$, $h_q$, $\ell$, $\beta$, $\theta$, V etc.. To illustrate the influence of any of such parameters on either $W^*$ or on $Q^*$, let the following numerical example be investigated. Hereby the following design nomograms are constructed.

### 3. IMPLEMENTATION

#### 3.1 Initial Values

$\beta = \theta = 0$, $\ell = 1$ m, $h_q = 0.25$ m, $h_c = 0.6$ m, $A_f = 0.75$ m, $L = (A_r + A_f)$ m, $L = C_1 A_f$ m, $C_1 = 2$, $\delta \cong 0.7$(see discussion), $t_a = 5$ s, $V = 1$ m/s, $V_s^f = 0$, $K_1 = 0.01$, $K_2 = 0.0001$, W & Q in ton(t), $K_a = 0.51$ , $K_w = 0.0$.

#### 3.2 Calculation Steps

1. The computer feeds by the initial values, the $W^*$ and $Q^*$ are calculated for various values of $\delta$,
2. The influence of any parameter on $W^*$ and $Q^*$ is computed while other parameters are kept at their initial values and then
3. Because of the space limitations due to the conf. com.requirements, only two design nomograms are given in Figs 3 and 4.

### 4. DISCUSSION OF THE RESULTS

Firstly, the $F_Q$ and $F_a$ are calculated by eqns (12) and (13) respectively when the $K_{01}$, $K_{02}$ and W are fixed at 0.25, 0.3 and 6 respectively. These values are selected to be closed to their numerical values, the results are plotted in Fig 2. In this figure the lines labbled 1, 2, 3 and 4 are for $F_a$ at $\delta = 0.5$, 0.6, 0.7 and 0.8 respectively. It can be seen that the accurate value of $\delta$ is the mean value which is located in dashed area. Then the value of $\delta$ is selected to be approximately 0.7 and the signum function $\sigma_Q$ is taken to be -1, hence all of the numerical values are known and fixed.

Secondly, Figs 3 and 4 illustrate the design nomograms by which the correlation between two investigated parameters or more than two can be made. As example, the line between $A$-$A_1$-$A_2$-$A_3$ and $A_4$ in Fig 3 makes correlation between the investigated parameters $\beta$ and V or vice versa, such line in Fig 4 makes also correlation between $A_f$ and $h_c$ or vice versa.

Thirdly, Fig. 3 indicates the influence of $\beta$ and of V on $W(W^*)$ and shows as $\beta$ increases the W decreases by small rate, as well as as V increases the W decreases rapidly. Also it can be seen that the increasing in $\beta$ and in V give increasing in Q $(Q^*)$ and decreasing in Q respectively.

Fourthly, Fig. 4 shows the influence of $A_f$ and $h_c$,by which the C.G. of the machine can be known, on the W and on Q. This influence is similar to the preceeding influence with different rate.

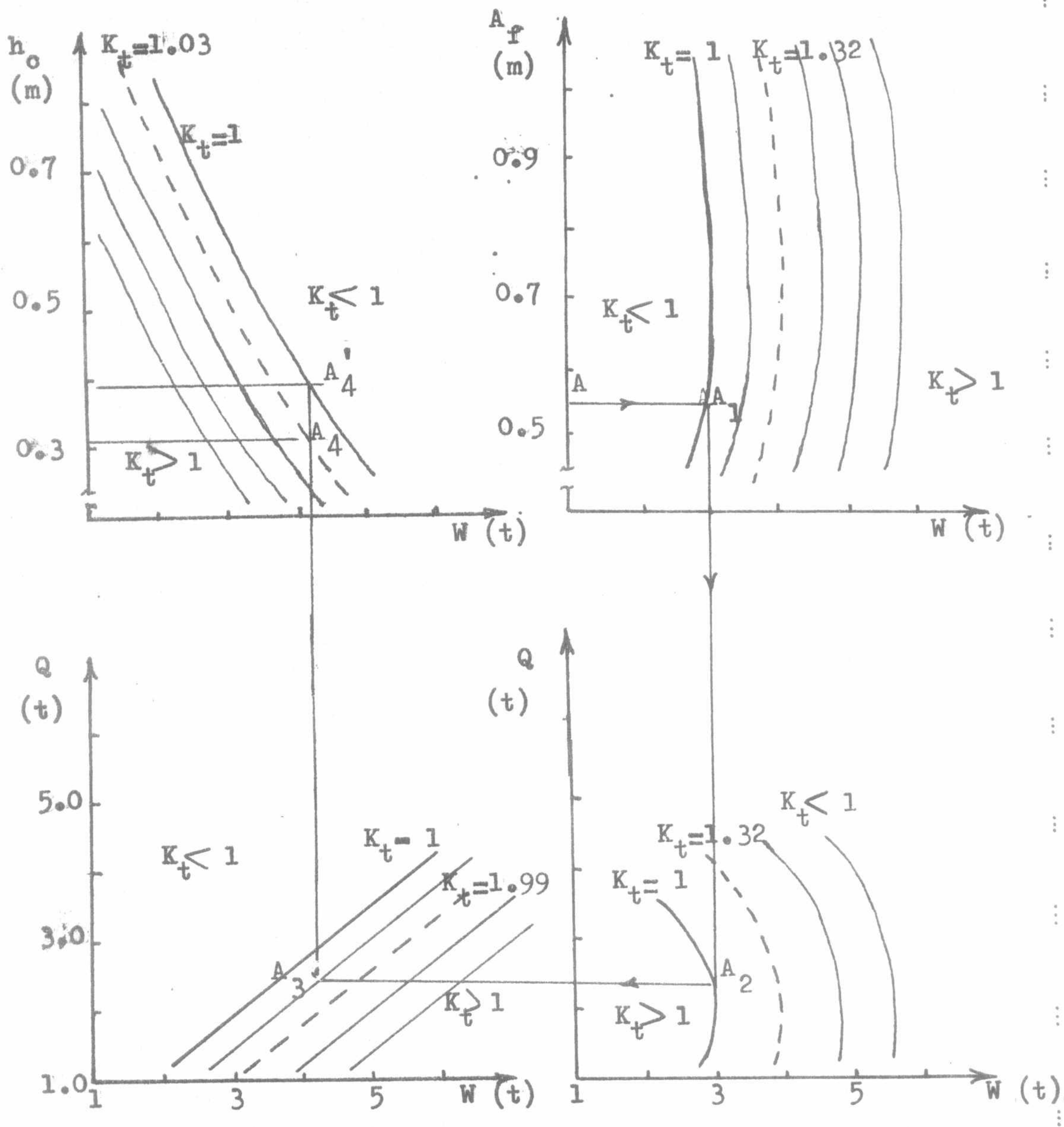


**Fig. 4 Design Nomogram For $A_f$ & $h_c$**

(see N.B.of Fig. 3)

$F_d = 0.1 \sim 0.3$ ton

Finally, to get good longitudinal dynamic stability which is considered in this study, W and Q should be increased and decreased respectively. And to lift maximum load Q, the $h_c$ and V should be decreased and $\beta$ may be increased as far as possible. The $Q^*$ equals to the critical or tipping load which has been defined in [1]. The Q must be less than $Q^*$ as shown in the nomograms. Moreover, the $W^*$ and $Q^*$, eqns (21), are the nominal values of the optimum design variables, W and Q.
As stated in 1 that such these variables should be treated as random variables, therefore, to calculate the real optimum values of such variables the chance constrained technique with GPOT is expected to be more realistic technique for such optimization.

## 5. CONCLUSIONS

The following conclusions can be drawn from this study,
1. The GPOT is applicable in the field of optimum design of materials handling machines,
2. No difficulty can be existed if the design variables are changed,
3. If the actual lifting load is less than one ton, the $\delta$ may be taken 1.2 or so to start. For accurate values of the optimum design variables, the $\delta$ should be computed in high accuracy within the computation steps,
4. The maximum required lifting load must be less than $Q^*$ which is the tipping load, this can be estimated fast by such as given nomograms,
5. To increase the lifting load, the value of $C_{11}$, eqn (14), should be decreased and then
6. To obtain the optimum real values of such design variables, the chance constrained programming with GPOT is suggested, this will be in further work with determining the high accuracy of $\delta$.

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## 7. REFERENCES

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