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Analysis on Supporting Stability for Track Subgrade Dynamic Response In-situ Test Device based on NSGA-II

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Abstract

The dynamic response test to the subgrade plays a very important role in railway construction and a new in-situ test system is proposed. This paper presents the application of non-dominated sorting genetic algorithm-II (NSGA-II) to analyze the stability of the supporting equipment for track subgrade dynamic response in-situ test device. Its stability is related with the extension length of the hydraulic cylinders and the backward condition of the supporting equipment - the hydraulic excavator. The problem is formulated as a multi-objective optimization problem with the objective of maximizing the supporting force for the test device. An 85 tons excavator is picked as the case to study. The first optimal results show the excavator may not support the test system successfully. After redesigning the boom and adding its weight and length as new parameters, the second optimize results indicate the test device can work normally.

Keywords: *dynamic response test, hydraulic excavator, supporting stability, multi-objective optimization*

1. Introduction

The railway construction today is facing greater challenges in meeting demands of the dynamic test technology on subgrade [1]. The increase of trains' speed and the weight puts forward higher requirements on the performance of the railway embankment. Roadbed not only bears the weight of the upper structures, but also withstands the impacts from the trains repeatedly movement, therefore its stability in long-term effects is significant important. Many factors may affect the performance of the subgrade [2-3] and in order to study its property and reveal its laws more efficiently, we developed a hydraulic excitation system for in-situ track roadbed dynamic response test [4].

To provide stable support force to the test equipment and install or move it easily, the hydraulic excavator is selected as the support equipment. This paper analyses the supporting force determined by the excavator hydraulic cylinder locking force and backward condition, and then computes their maximum values with the use of non-dominated sorting genetic algorithm-II (NSGA-II) which is a newly developed algorithm for multiobjective optimization to provides a specification reference on roadbed dynamic response in-situ tests.

2. Brief Introduction of the Subgrade Dynamic Response Test Device

The core of the subgrade dynamic response test system is the excitation servo hydraulic cylinder which is shown in Figure 1 [5]. By the unique designed structure, this servo hydraulic cylinder simultaneously includes a static and a dynamic pressure chamber in one cylinder, which are able to independently output the static and the dynamic force to simulate the static load of the upper structure weight and the dynamic load of trains' movement. The coupled force is applied on the subgrade through the excitation pad. After a period of excited tests, the stability of the roadbed under long-term static & dynamic load can be analyzed by the signals through the soil pressure cell and the accelerate sensor pre-embedded in the soil.

The design requirements of the excitation hydraulic cylinder are:

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1) The maximum static force is 200 kN;
2) The maximum dynamic force is ± 100kN.

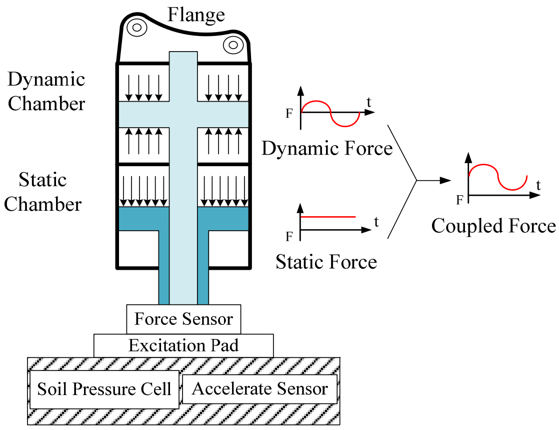

Figure 1. Excitation Servo Hydraulic Cylinder

3. Supporting Force Analysis

As shown in Figure 2, after the bucket has been demounted, the excitation hydraulic cylinder and the excavator arm can be connected by the flange. When the excavator is adjusted to a certain position, it can provide the supporting force for the excitation device after its hydraulic cylinders of the boom, arm and bucket are locked down. Point A ~ K represent the hinge points of the components, and point L is the back point contacting to the earth of the track.

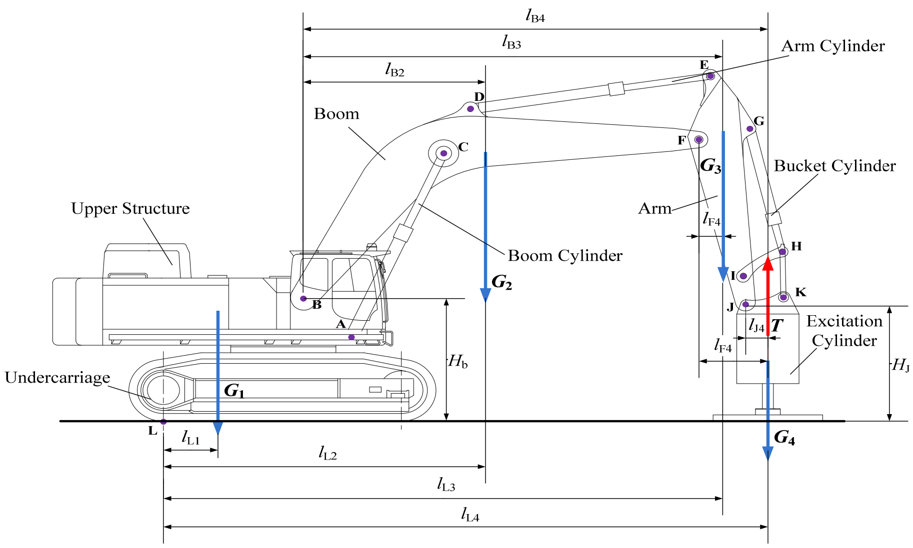

Figure 2. The Principle of Supporting Device

It's easy to know that the excavator's stability is related to the supporting force-greater the supporting force is, more stable the excavator will be; otherwise, it may lose its balance or be damaged. What's more, the supporting force is determined by the extended length of the hydraulic cylinders-the arms of the excavator components to the excitation hydraulic cylinder may be different with the changes of extended length of hydraulic cylinders of the excavators, and the supporting force might be different as well. Therefore, to ensure the excavator has the highest supporting capability and not be damaged either, and the excitation hydraulic cylinder works properly, it is necessary to analyze the relationship between extended length of the hydraulic cylinders and supporting force of the excavator.

The main factors affect the excavator stability including: the maximum locking force of the hydraulic cylinders, the backward condition (to prevent excavator overturning), wind force, etc. To simplify the calculation the author only considers the first two main factors and makes the following assumptions in the analysis process later:

(1) The device is working on the flat ground, and working equipment is in front of the excavator;
(2) The center of gravity of each component is in line with its geometric center;
(3) The force of the excitation cylinder upward and gravity is collinear;
(4) The relief valves of boom, arm and bucket hydraulic cylinder are set to the highest pressure;
(5) The mechanical structures are not damaged;
(6) The efficiency of the mechanism, the efficiency of hydraulic system and the back pressure of the hydraulic cylinders are ignored.

3.1. Mathematical Formulation
3.1.1. The Boom Mechanism Analysis [6]

As shown in Figure 3, the boom swing angle $\beta_1 = \alpha_3 - \alpha_1 - \alpha_2$, where, for a specific excavator, α_1 and α_2 can be directly measured.

In ΔBCF, $\alpha_3 = \arccos[l_{AB}{}^2 + l_{BC}{}^2 - L_1{}^2) / (2 \cdot l_{AB} \cdot l_{BC})]$,

where L_1 is the length of the boom cylinder

The boom cylinders are in tension as working, so the maximum locking force is:

$$F_1 = \frac{\pi}{4}(D_1^2 - d_1^2) \cdot P_s \cdot n_1 \qquad (1)$$

Where, F_i is the output force of the hydraulic cylinder, D_i represents the diameter of the piston, d_i represents the diameter of the piston rod, P_s is the closure pressure of the system, n_i represents the number of hydraulic cylinders, subscript i=1, 2, 3 ,4 respectively represents the terms of boom, arm, bucket and excitation cylinder, similarly hereafter.

The arm of the force to the point B of the boom cylinder locking force is:

$$e_1 = l_{BC} \cdot \sin \gamma_1 \qquad (2)$$

where $\gamma_1 = \arccos[(l_{BC}{}^2 + L_1{}^2 - l_{AB}{}^2) / (2 \cdot l_{BC} \cdot L_1)]$.

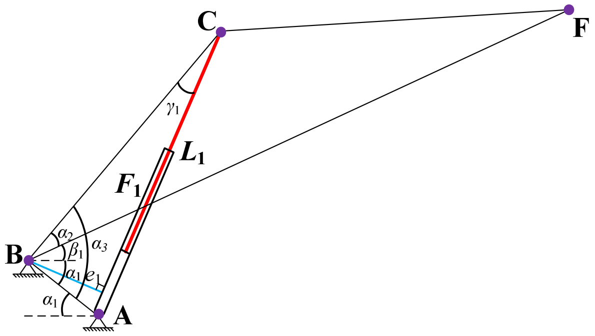

Figure 3. Diagram of Boom Mechanism

3.1.2. The Arm Mechanism Analysis

As shown in Figure 4, the maximum locking force of the arm cylinder as working will be:

When in extension:

$$F_2 = \pi \cdot (D_2^2 - d_2^2) \cdot P_s \cdot n_2 / 4 \tag{3}$$

When in pression:

$$F_2 = \pi \cdot D_2^2 \cdot P_s \cdot n_2 / 4 \tag{4}$$

The arm of the force to point F of the cylinder locking force is:

$$e_2 = l_{\mathrm{DF}} \cdot \sin \gamma_2 \tag{5}$$

Where $\gamma_2 = \arccos[(L_2{}^2 + l_{\mathrm{DF}}{}^2 - l_{\mathrm{EF}}{}^2) / (2 \cdot L_2 \cdot l_{\mathrm{DF}})]$.

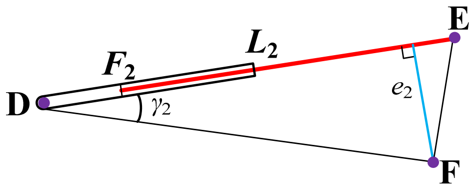

Figure 4. Diagram of Arm Mechanism

3.1.3. The Link Mechanism Analysis

As shown in Figure 5, for the designed excitation hydraulic cylinder and the flange, α_4 is known, and $H_J = H_B + l_{\mathrm{BF}} \cdot sin\beta_1 - l_{\mathrm{FJ}} \cdot sin\beta_2$

Where, H_B, H_J represents the vertical distance from point B and point J to the ground as shown in Figure 2, for a designed excavator they can be measured, β_2 is the angle between the arm and the horizontal line.

The bucket cylinder is in tension as working, and the maximum locking force is:

$$F_3 = \pi \cdot D_3^2 \cdot P_s \cdot n_3 / 4 \tag{6}$$

The arm of the force to the point J of the bucket cylinder locking force is:

$$e_3 = l_{\mathrm{GJ}} \cdot \sin \gamma_3 \tag{7}$$

Where $\gamma_3 = \arccos[(L_3{}^2 + l_{\mathrm{GJ}}{}^2 - l_{\mathrm{HI}}{}^2) / (2 \cdot L_3 \cdot l_{\mathrm{GJ}})]$.

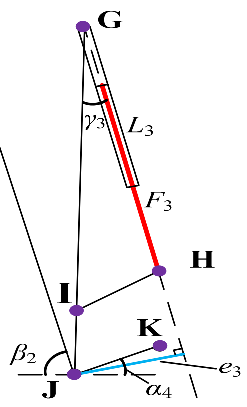

Figure 5. Diagram of link mechanism

3.2. The Supporting Capability Analysis

By theoretical mechanics knowledge, we can calculate the torque to point B, F, J respectively, and the supporting force T determined by the locking force of boom, arm and bucket cylinder can be drawn as:

$$T_1 = (G_2 \cdot l_{B2} + G_3 \cdot l_{B3} + G_4 \cdot l_{B4} + F_1 \cdot e_1) / r_1 \tag{8}$$

$$T_2 = (G_3 \cdot l_{F3} + G_4 \cdot l_{F4} + F_2 \cdot e_2) / r_2 \tag{9}$$

$$T_3 = (G_4 \cdot l_{J4} + F_3 \cdot e_3) / r_3 \tag{10}$$

And the supporting force determined by the backward condition will be:

$$T_4 = (G_1 \cdot l_{L1} + G_2 \cdot l_{L2} + G_3 \cdot l_{L3} + G_4 \cdot l_{L4}) / r_4 \tag{11}$$

Where G_1, G_2, G_3 and G_4 respectively presents the weight of the excavator body (including undercarriage, upper structure, counterweight, fuel, hydraulic oil, etc.), the boom assembly (including the boom, boom cylinders, arm cylinder, etc.), arm assembly (including arm, bucket cylinder, link assembly, etc.) and the excitation device. l_{Bj}, l_{Fj}, l_{Jj}, l_{Lj} is respectively the arm to point B, F, J, L of centre of the gravity of each component, subscript j=1, 2, 3 ,4 respectively represents the terms of excavator body, boom, arm , bucket and excitation cylinder, similarly hereafter, as shown in Figure 2. They can be got by the assumptions and analysis previously, so for space reasons, we will not repeat them.

r_1, r_2, r_3, r_4 is respectively the moment arm to point B, F, J, L of excitation hydraulic cylinder force T, it's easy to know that: $r_1 = l_{B4}, r_2 = l_{F4}, r_3 = l_{J4}, r_4 = l_{L4}$, and F_i, e_i can be get from (1)~(7).

4. Optimization Analysis of the Supporting Force

Through the analysis above, it can be seen that the supporting forces of the excavator are the functions of the length of its hydraulic cylinders. So there are 4 objective functions: T_1~T_4, and 3 variants: L_1, L_2 and L_3, obviously, it a typical multi-objective optimization problem.

4.1. Multi-objective Optimization

Mathematically a multi-objective optimization problem can be described as follows:

$$\min\{f_1(\boldsymbol{x}), f_2(\boldsymbol{x}), \cdots, f_m(\boldsymbol{x})\}$$
$$s.t. \begin{cases} lb \leqslant \boldsymbol{x} \leqslant ub \\ \mathrm{Aeq} * \boldsymbol{x} = \mathrm{beq} \\ A * \boldsymbol{x} \leqslant b \end{cases} \tag{12}$$

Where, $f_m(\boldsymbol{x})$ is the objective function to be optimized, $\boldsymbol{x}$ are the optimization variables, *lb* and *ub* respectively is the lower and upper limits of the variable x; $\mathrm{Aeq} * \boldsymbol{x} = \mathrm{beq}$ is the linear equality constraints of the variable $\boldsymbol{x}$; $A * \boldsymbol{x} \leq b$ is the linear inequality constraints of the variable $\boldsymbol{x}$.

Unlike the single-objective optimization algorithm, in the multi-objective optimization algorithms, each objective function is often contradict to the others, in most cases the objective functions can hardly reach their optimal values at the same time, so the solutions of multi-objective optimization are mutual compromised [7], and they are also known as the Pareto solutions. There are a few multi-objective optimization methods, such as weighted method, pattern search, simplex, simulated annealing, and genetic algorithms (evolutionary algorithms) [8], among of which the non-dominated sorting genetic algorithm-II (NSGA-II) is able to maintain a better spread of solutions and converge better than other multi-objective optimization algorithms, and it has been successfully applied to many problems [9]. The NSGA-II procedure used in this work is illustrated in Figure 6 [10-11]

4.2. Objective Functions and the Constraints

The purpose of the optimization is to search the maximum supporting forces as presented in the Equation (8) to (11), that is, to seek the max {T_1, T_2, T_3, T_4}. In accordance with Equation (12), the former objective functions need to be converted to the opposite to be optimized: min {-T_1,-T_2,-T_3, -T_4}. The variables are the length of the hydraulic cylinders:x = [L_1, L_2, L_3]. Their minimum and maximum values, which can be found out in the related materials, are set to be the constraints *lb* and *ub*.

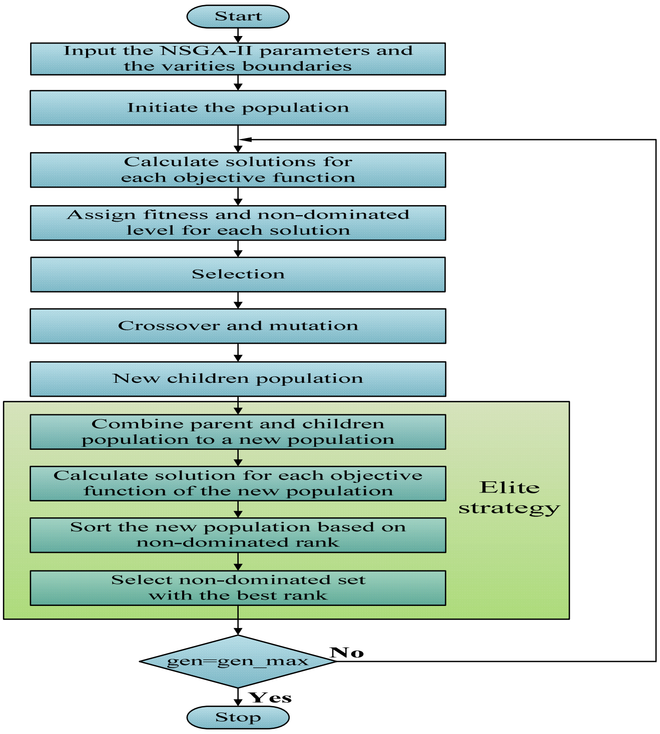

Figure 6. The Procedure of NSGA-II

4.3. Case Study

Take an 85 tons excavator as example. The parameters involved in optimization are shown in Table 1 referring to correlative drawings and manuals. The objective and constraint functions are programmed and saved as m-files respectively. Run the multi-objective optimization program based on genetic algorithms “gamultiobj” in Matlab with the default options. According to the requirements the supporting force should attain at least 300kN and part of the optimal results are picked as shown in Table 2.

Table 1. Parameters

Symbol	Value	Symbol	Value
α_2(rad)	0.44	d_1(mm)	140
α_3(rad)	0.65	D_2(mm)	185
α_4(rad)	0.26	d_2(mm)	120
l_{AB}(mm)	1200	D_3(mm)	185
l_{BC}(mm)	4040	d_3(mm)	120
l_{DF}(mm)	4370	n_1	2
l_{EF}(mm)	1355	n_2	1
l_{BF}(mm)	8200	n_3	1
l_{FJ}(mm)	3580	L_1(mm)	2930-4880
l_{GJ}(mm)	3700	L_2(mm)	2380-3990
l_{HI}(mm)	950	L_3(mm)	2590-4410
H_B(mm)	2625	G_1(N)	610000
H_Y(mm)	800	G_2(N)	100000
P_s(Mpa)	34.55	G_3(N)	44100
D_1(mm)	200	G_4(N)	8800

Table 2. Part of Optimization Results of The Supporting Force

	1	2	3	4	5
T_1(kN)	194	195	195	195	197
T_2(kN)	307	354	320	663	558
T_3(kN)	1765	4269	2364	4724	1875
T_4(kN)	183	183	183	183	182
L_1(mm)	4478	4462	4451	4441	4385
L_2(mm)	5338	5291	5352	4846	5166
L_3(mm)	4024	3452	3950	3019	4012

5. Analysis

Trough the optimized results in Table 2, it can be seen that the supporting force, determined by the locking force of the hydraulic cylinder of the boom, is about 195kN and lower than the required 300kN. This suggests that the exciting force of the excitation hydraulic cylinder probably exceeds the maximum pressure of the relief valve, and even may damage the boom cylinder. The supporting force decided by the arm cylinder locking force is from 307kN to 663kN, which basically meets the system requirements. However its safety margin is too small as being 307kN, that the excitation cylinder's operation might influence the arm hydraulic cylinder severely and its working life is possible to be shortened. The supporting force, determined by the bucket cylinder's locking force, reaches 1765kN to 4724kN, which is about 6 to 16 times to the system requirements. This indicates that the excitation device hardly has any effect on the bucket cylinder. The maximum supporting force determined by the backward condition is approximately 183kN and less than the system demand, which illustrates the exciting cylinder can possibly push up the excavator, and the excavator has the danger of being turned over.

The main reasons for the boom cylinder can not bear the excitation cylinder are the boom cylinder rod chamber is small in size and the boom is too long. Generally, the boom cylinder is under the pressure in the mining operations, and then the rodless cavity of the boom cylinder is working. While the railway subgrade dynamic response in-situ test system is on operation, boom cylinder is in tension and then its rodless cavity is working, so the locking force of the hydraulic cylinder is relatively smaller (about half of that when the rod cavity is in operation) and therefore can not provide a large supporting force for exciting cylinder. Meanwhile, the long boom also increases the arm of force of excitation force to point B. One of the solutions is to increase the net area of the rodless cavity of the boom cylinder, which can be achieved from replacing hydraulic cylinder either the inner diameter is larger or the rod diameter is relatively smaller. Secondly, it could alter a special-designed shooter boom. In the same time, we can increase the weight of the boom to "balance" part of the exciting force.

Table 3. The Improved Optimization Results as New Parameters Added

	1	2	3	4	5
T_1(kN)	336	336	335	335	335
T_2(kN)	352	371	405	502	360
T_3(kN)	4479	4478	4478	4479	4436
T_4(kN)	344	344	344	344	343
L_1(mm)	3534	4347	3024	3539	3021
L_2(mm)	4444	4564	4505	4556	4481
L_3(mm)	3019	3065	3211	3107	3528
l_B(mm)	5887	8127	6183	6459	6156
G_B(t)	11.6	18.5	17.4	10.9	16

According to Equation (11), the cause why supporting force determined by the backward condition is insufficient mainly due to the structural length of the excavator, which results in the arm of the excitation hydraulic cylinders to the point L is too long. Because the size of the excavator undercarriage can hardly be changed, it should minimize the length of the other components (such as the boom, arm) to reduce the arm of the force. And what, if possible, properly increasing the weight of the counterweight and the parts also plays a role.

The supporting force relating to the arm and bucket cylinders' locking force basically meets the requirements, so they don't need to be considered temporarily. Based on the analysis above, we adopt the means of both choosing a shooter boom and increasing its weight to enhance the supporting stability. So, two variables are added: the length l_B and the additional weight G_B of the boom. Run the optimization program again and the improved results are shown in Table 3.

6. Conclusion

In this paper the multi-objective optimal algorithm-NSGA-II has been implemented to study the supporting stability of the excavator for the track subgrade dynamic response in-situ test device. Conclusions obtained from the proposed approach are as follow:

(1) We introduced the dynamic response testing device of track subgrade, described the basic principles and conditions, and selected a hydraulic excavator as its supporting equipment. With analytical and theoretical mechanics methods the author established the mathematics models involved the length of the cylinders and backward condition of the excavator and the supporting force to the testing excitation hydraulic cylinder.
(2) Based on the non-dominated sorting genetic algorithm-II (NSGA-II), the length of the boom, arm and bucket hydraulic cylinders of an 85 tons excavator was optimized, the results show that the supporting force determined by the block force of the original boom cylinder and the backward conditions may cannot meet the requirements of the test system.
(3) The main reasons for that are the structural length is too long and the weight of the components may not enough. Then two variables were appended: the length and the additional weight of the boom and run the optimization program again. The new results show that after adjusting the parameters of the components of the excavator the maximum supporting force increases obviously. The supporting force determined by the boom cylinder locking force increases to about 336kN from 195kN previously, and that by the backward condition increases to about 343kN from 183kN previously, that respectively raises 72% and 87%. The minimum supporting force related to the arm cylinder locking force increases to 352kN, raises about 50kN. The improved results indicate the excavator can provide the stable supporting force for the track subgrade dynamic response in-situ test equipment.
(4) Considering the improved optimal results and materials saving and manufacture convenience, the specific-designed boom is recommended about 6.45ms long and 10.9 tons weight.

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