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SOME ASPECTS OF THE LIFTING DYNAMICS OF FORK LIFT TRUCKS

Yanko SLAVCHEV
Technical University - Sofia, Bulgaria

Abstract. A research with 3D computer models on some aspects of the lifting dynamics of triple telescopic lifting gears is presented. The influence of the lifting tackle mechanisms, moving masses, etc. is taken into account. Contacts between chains and tackle rolls are also accounted for and it introduces various complexities to the model. Precise setting of integrator parameters enables the simulations to realize various model beneficial options such as: lifting for a given start time, lifting and stopping, lifting load with and without impact as well as accounting for the contact forces.

Keywords: lifting gear, chain, lifting dynamics, impact, contacts, stiff problem

1. Introduction

The paper has to be written in English. Its contents should be structured in the following way (recommendation): problem description, application field, research stages, methods used, results, further research, conclusions, and references.

Lifting dynamics of fork lift trucks is due to the structural peculiarities of the lifting gear which is primarily made up of hydraulic and mechanical systems. The hydraulic system includes one or more hydraulic drives. After the oil is accelerated, the drives act on the mechanical system by means of plungers so that the load is lifted up.

In general, the mechanical system is a multiple lifting tackle connected to telescopically arranged metal frames (masts). The frames make possible the elevated handling of loads.

Lifting dynamic response is evaluated correctly when the start time is well known. However, start time is quite difficult to be obtained since it is a complex function of numerous factors (valve actuating), time for accelerating the hydraulic system, time for accelerating the lifting tackles, etc. It is obvious that it is hard to compute theoretically this function. Therefore, the abovementioned task is handled by applying empirical relations [1, 2] such as

$$t_{start} = \lambda v_C \qquad (1)$$

where $\lambda \approx 6 - 8$ is coefficient that depends on the lifting gear parameters and design; v_c is the velocity of the piston of the hydraulic drive.

Driving force, respectively dynamic response of different lifting gear designs is investigated in 0 by the use of a single mass model where the starting time is defined from relation (1). These investigations, however, do not account for the elasticity of lifting tackles, moving masses and hydraulic system influence, etc.

It is clear that detailed research is necessary on the lifting dynamics of fork lift trucks.

Well-defined computer simulations that employ 3-D models are a key to the resolution of these problems and have been set as the objective to the present work.

2. General considerations

Lifting mechanisms with multiple lifting tackles, embedded in **n**-telescopic lifting gears, could be designed with one or two hydraulic drives. Gears with one drive (gears with low free lift) have the hydraulic drive carried by the immovable mast as shown in figure 1. Gears with two drives have one of their drives fixed and the other one lifts up along with the innermost mast, (figure 2).

The common thing to these kinematic schemes is that number of movable masts is n-1. For mechanisms with one hydraulic drive, this number is the same for the lifting tackles that are connected in series to the respective masts.

For mechanisms with two hydraulic drives, n-2 lifting tackles are connected in series and the last tackle is carried by the movable hydraulic drive 1 (figure 2).

Velocities, respectively gear ratios of movable elements in both kinematic schemes are formulated with the following relations

lifting gear with one hydraulic drive

$$v_{n-1} = 2v_{n-2} - v_{n-3} = [2(n-2)-(n-3)]v_1 = \\ = (n-1)v_1 = (n-1)v_C \quad (2)$$

Similarly, for the payload velocity v_Q

$$v_Q \equiv v_n = n \cdot v_C \quad (3)$$

The gear ratio is

$$i_n \equiv \frac{v_Q}{v_C} = n \quad (4)$$

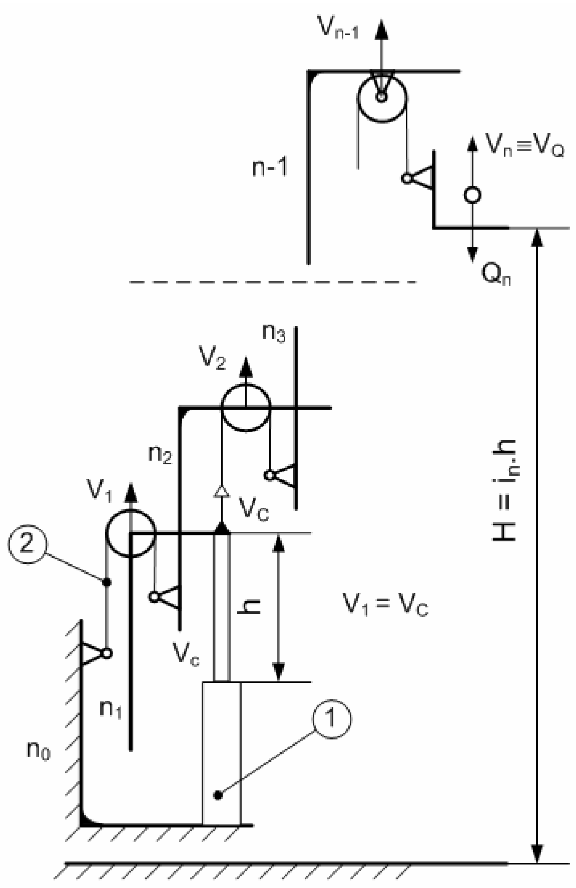

Figure 1. Lifting gear with one hydraulic drive
1 – hydraulic drive, 2 – multiple lifting tackle,
n_0 – immovable (fixed) mast, $n_{1..n-1}$ – movable masts

lifting gear with two hydraulic drives

first stroke

$$v_{Q_1} = i_1 v_C\,;\ i_1 = 2 \quad (5)$$

second stroke

$$v_{Q_2} = i_2 v_C\,;\ i_2 = n-1 \quad (6)$$

On condition that $v_{Q1} = v_{Q2}$, it follows that

$$n = 3\ (i_1 = i_2) \quad (7)$$

Clearly, lifting gears with two hydraulic drives have to be designed as triple telescopic so that the uniformity of hydraulic parameters during both strokes is conserved.

Both mechanisms have the same hydraulic scheme but the difference is that movable drive, figure 2, needs to be supplied by longer hoses. It means that, in this case, acceleration time (start time) for the hydraulic system is larger than the time for accelerating the mechanism in the presence of one drive. Hydraulic start time could be calculated with relation [2]:

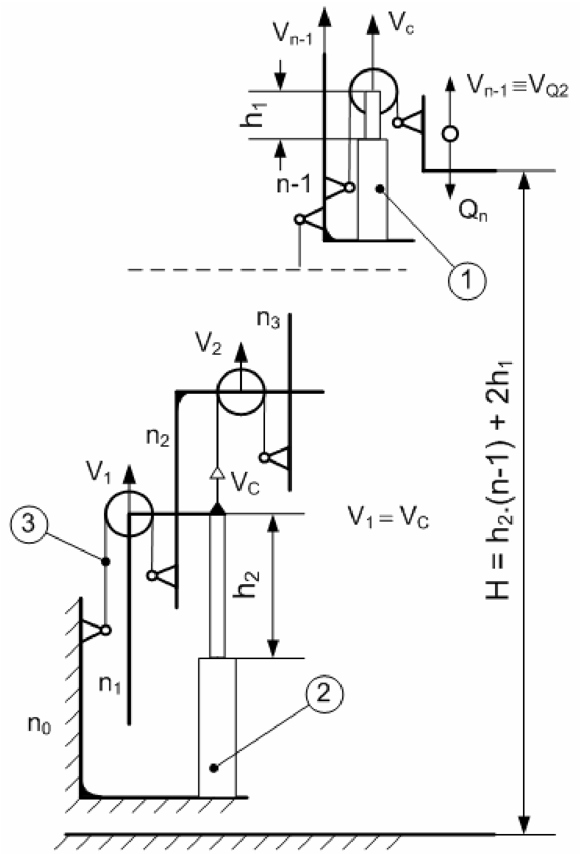

Figure 2. Lifting gear with two hydraulic drives
1 – hydraulic drive for working stroke;
2 – hydraulic drive for the multiple lifting tackle,
3 – multiple lifting tackle;
h_1, h_2 – working strokes of the corresponding hydraulic drives

$$t_{start}^{hydro} = \left(\frac{2A_{hose} l_{hose}}{E_{hose} \dfrac{\delta_{hose}}{r_{hose}}} + \frac{A_{tube} l_{tube} + A_{cyl} l_{cyl}}{E_{fluid}} \right) \cdot \\ \cdot \frac{v_C A_C \gamma}{2g(\alpha_{fluid} A_{valve})^2} \quad (8)$$

where designations are:

A, δ, r – cross-sectional area of the pass, wall thickness and inner radius; l – length, $E_{fluid} \approx 1.8 \cdot 10^3$ MPa – operating fluid modulus of elasticity, $E_{hose} = 0.2 \cdot 10^3$ MPa - hose modulus of elasticity, γ – operating fluid volumetric weight, α_{fluid} – coefficient of fluid consumption of the valve, A_{valve} – cross-sectional area of the pass of the valve

3. Generating 3D computer models

The CAD system SolidWorks is employed to design and assemble the geometry of two major 3-D models of lifting gears – model with one hydraulic drive (model 1) and model with two hydraulic drives (model 2). Real triple telescopic lifting gears are modeled and during the design stage, it is strictly monitored that model parameters such as masses, dimensions, mass moments of inertia, etc. conform precisely to the real values.

Models are in fact multibody systems of rigid bodies 0 that have 94 bodies and 90 degrees of freedom. Models dynamic response is simulated by MSC.Adams as one of the most popular packages suitable exactly for similar problems 0.

Simulations are carried out under the following conditions:

- assembly is made of steel parts;
- flexible element of the lifting tackle is block chain; contacts between chain members and tackle rolls are accounted for; chains have the same stiffness;
- gravity is considered;
- contact between payload and actuator (fork) as well as fork elasticity is accounted for;
- structural defects as well as friction between guiding rolls and masts are neglected;
- payload centre of mass is located at 500 mm from the base of the fork-arm;

Contacts between chains and tackle rolls as well as between payload and the fork arm is a key aspect of the simulations. It allows for life-like models but at the same time leads to non-linear effects and various complexities related to continuity violations that the solution is bound to account for.

One of the complexities refers to the simulation stepsize. When chains are in motion, some of the chain members fall in and others fall out of contact with the tackle rolls which causes impacts 0. It is a pre-condition for stiff differential equations and high frequency responses of the system.

Another complexity is related to the contact defined between the payload and the fork. Models enable simulating cases of payload lifting from the ground (the payload is 60 mm above the ground) and lifting from the air (the payload is located on the fork). In the cases of ground lifting, impacts inevitably occur in the instant of payload picking up by the fork. When the stepsize is too large, simulations fail to solve due to integration failures caused by excessive forces, stability issues and corrector failures.

Following an iterative procedure, it is estimated that the appropriate stepsize is 1 ms. It allows for high enough speed of solution and at the same time not sacrificing the precision. It is necessary, however, to adjust the solver settings in details. The GSTIFF integrator is used as one of the most popular types of stiff multistep integrators. The small stepsize requires that the I3 or SI2 formulation be selected [5, 6]. In order that relatively constant stepsize at high order be achieved, the following is set: max. integrator timestep = 2.10^{-4}s, local integration tolerance = 10^{-2} , max. integrator order = 12.

Model actuation is yet another complexity. Hevyside step function is applied to deal with it (figure 3). This function causes sharp changes in the kinematic parameters, but the SI2 formulation relaxes them and any further discontinuities.

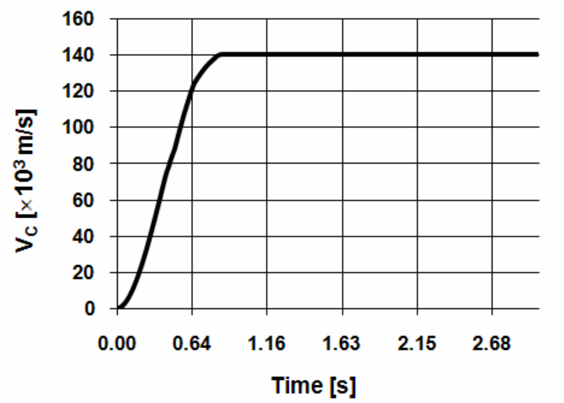

Figure 1. Hydraulic drive velocity, t_{start} = 0.84s, Vc=0.14m/s

The SI2 formulation preserves the continuity of derivatives and makes unnecessary any additional decrease in step size. Moreover, a life-like starting of the model is realized, since the steady-state velocity is reached after a certain period of starting time. The same is true for simulating a working cycle of starting, lifting and stopping the lifting gear.

Gravity is a must for the model. Providing gravity for the model, however, leads to necessary inertia relaxation prior to model actuation, so that static equilibrium is obtained and additional undesired impacts are avoided.

A real-life penetration depth of 100 μm is set for the contact between tackle rolls and chain members. It makes the system aware of their presence and relaxes impacts during simulation. Furthermore, precise geometry is used for the 3D contacts by setting values of the scaling factor for the facet tolerance. It decreases errors in forces and accelerations in contacts. After the integrator and contact settings are adjusted, simulations are solved and results are measured.

The 3D model of a triple telescopic lifting gear with one hydraulic drive (model 1) is shown in figure 4.

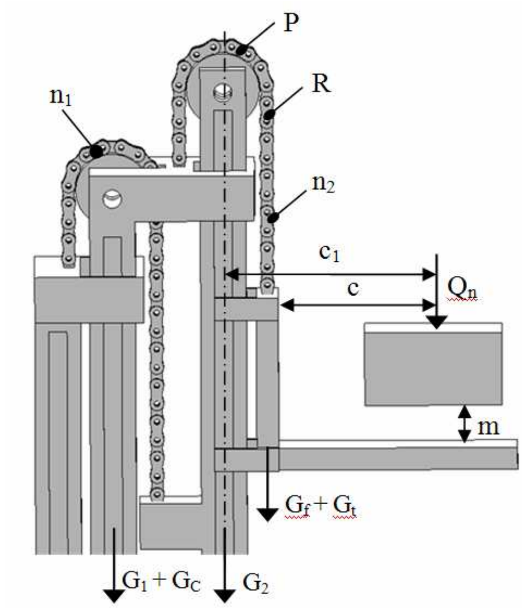

Figure 2. 3-D model 1
Q_n, G_f, G_t, G_2, G_1, G_C - weight of the payload, fork, truck, movable masts and the piston of the hydraulic drive

4. Numerical experiment and analysis

Computer simulations are performed with both models for the following input data:

<u>model 1</u> Q_n = 10kN, G_f = 0.7kN, G_t = 0.7kN, G_2 = 0.9kN, G_1 = 1.1kN, G_C = 0.3 kN, chain n1 = 30 N (chain has 93 members and step 20mm), chain n2 = 40N (95 members), chain stiffness k1≈ k2 = 11.62MN/m, c =500mm, c1 = 600mm, m = 60mm;

<u>model 2</u> all parameters are the same as for model 1 but here the weight of hydraulic drive 1 is included as G_{C1} = 0.3kN. Fork stiffness (100×40mm) – k_{fork}=4.55 MN.m/deg

Hydraulic system parameters are: $A_{hose}=A_{tube}=0.785cm^2$; $r_{hose}=r_{tube}=5mm$; $l_{hose}=1m$; $l_{tube}=2m$; $\delta_{hose}/r_{hose}\approx0.6\div0.7$; $l_{cyl}=1.4m$; $A_{cyl}=31cm^2$; $\gamma=9kN/m^3$; $\alpha_{fluid}A_{valve}\approx0.02cm^2$ (for $\alpha_{fluid}=0.6\div0.7$); for model 2, parameters are the same and l_{hose}=4m.

Simulations are performed for: lifting from the ground (case A – sharp lift, case B – smooth lift) and lifting from the air (no impact) with payload velocities v_Q=0.3, 0.36, 0.42 and 0.48m/s. For the piston velocities it is obtained: <u>for model 1</u> from formula **Error! Reference source not found.** at i=3, v_{ci} = v_Q / 3, and the start time is defined from **Error! Reference source not found.** for λ_1=6 and λ_2=8. These values are listed in Table 1.

Similarly, model 2 parameters are defined for which velocities in the first and second strokes are equal v_{c1}= v_{c2} = v_{Qi} /2. Start times are determined by:

$$t_{start\,1,2} = \frac{v_{Qi}}{2}\cdot\lambda_{1,2} + \Delta t \qquad (9)$$

Δt≈0.3s, is the time for accelerating the fluid for the longer hydraulic system (drive 1);

Start time of the second drive is determined by the same formula for Δt=0.

Table 1 lists data about the dynamic response of model 1.

Table 1 – Dynamic response of model 1 when the payload is lifted from the ground

t_{start} [s] /case	v_C [m/s]	Force [kN] / Dynamic coefficients		
		drive	chain n1	chain n2
0.6 A	0.1	66.27 1.81	42.84 1.81	20.49 1.8
0.8 B		66.07 1.80	42.76 1.8	20.46 1.79
0.72 A	0.12	72.30 1.97	47.08 1.98	22.62 1.99
0.96 B		70.32 1.92	45.16 1.91	21.63 1.91
0.84 A	0.14	74.09 2.02	49.08 2.07	23.56 2.07
1.12 B		68.35 1.86	46.01 1.94	22.11 1.94
0.96 A	0.16	76.68 2.09	50.14 2.11	24.05 2.11
1.28 B		71.01 1.94	46.29 1.95	22.20 1.95
static force [kN]		36.68	23.75	11.38

Figures 5 to 7 shows chain responses for model 1. Figure 8 and 9 show chain responses for model 2. Figure 8 illustrates contact forces in chain members P and R. Figure 10 gives chain force for model1, case A in the event of sharp stopping.

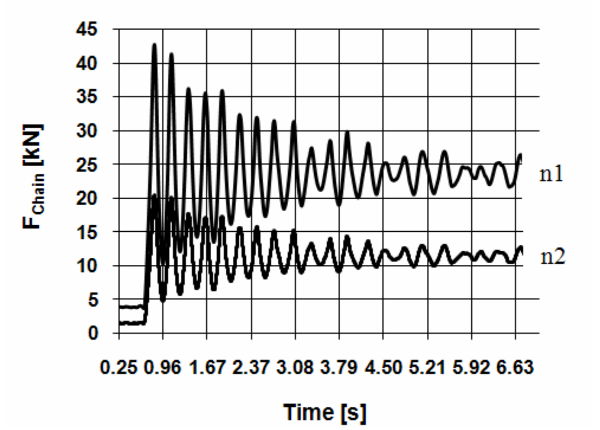

Figure 3. Forces in chains n1 and n2, model 1, case B, v_c=0.1m/s

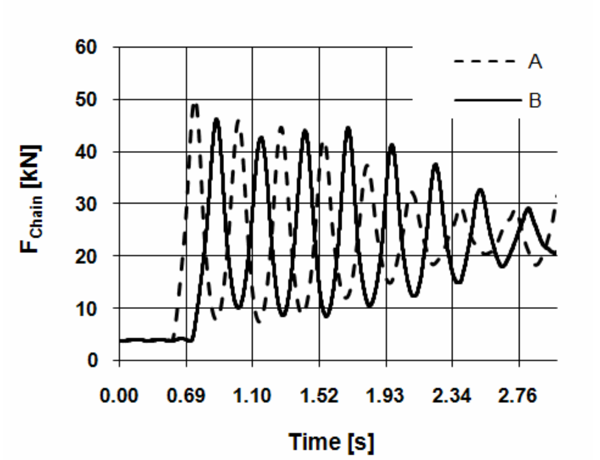

Figure 4. Forces in chain n1, model 1, cases A and B, v_c=0.16m/s

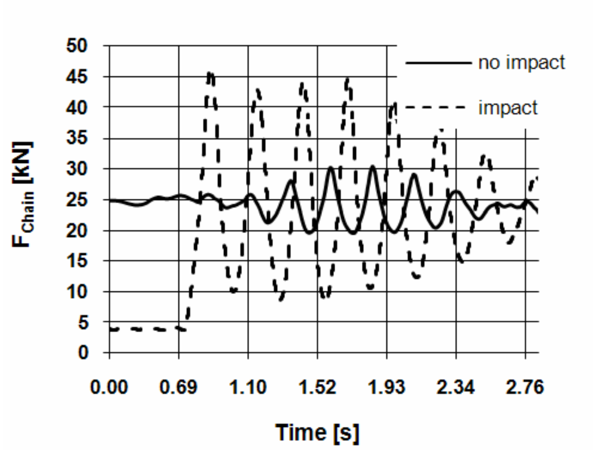

Figure 5. Forces in chain n1, model 1, lifting with and without impact, case B, v_c=0.16m/s

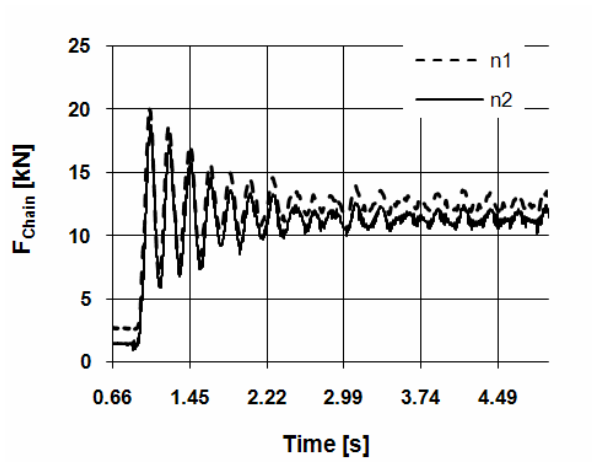

Figure 6. Forces in chains n1 and n2, model 2, lifting with impact, t_{start}=1.5s, v_c=0.15m/s

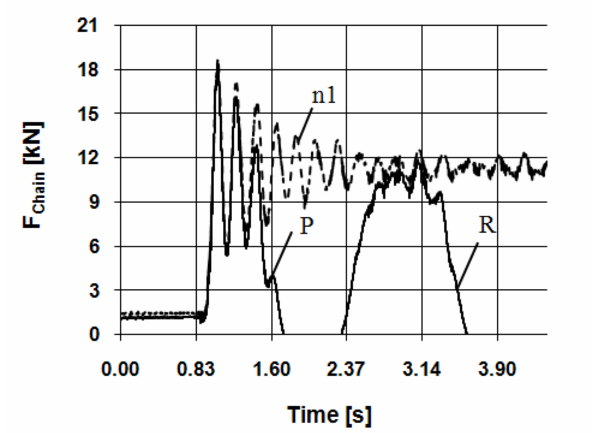

Figure 7. Forces in chain n1 and chain members P and R, model 2 lifting with impact, t_{start}=1.5s, Vc=0.15m/s

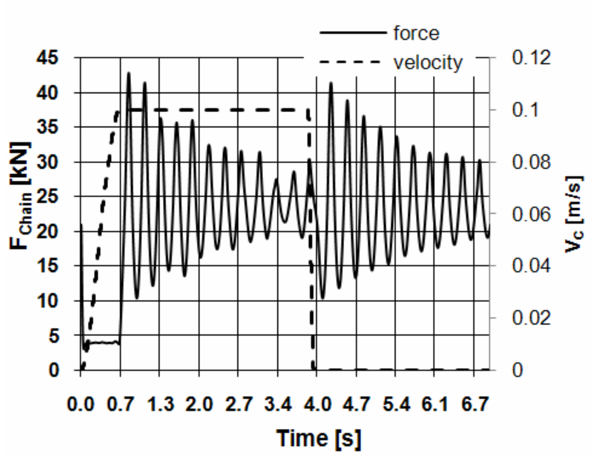

Figure 8. Force in chain n1, model 1, case A, sharp stopping t_{stop}=0.1s

The corresponding values of the fundamental natural frequencies of the models are ν1=3.2 Hz – for model 1 and ν1=4.78 Hz – for model 2.

It is established from results analyses that the dynamic response of forces in chains and drive is identical regardless of lifting velocities. When the payload is lifted from the ground, model1, the dynamic response is 50% higher compared to the case when the payload is located on the fork and then lifted up. Model 1 has shown a dynamic response that is 17% higher than in model 2.

Beating effects are observed in case B at higher lifting velocities but these are quickly dissipated. The reason for the effects is found in the external loading behaviour – Heavyside step function.

Contact forces in chain members passing over the tackle rolls are to be noticed. As shown in figure 9, members P and R are loaded differently. Member P is in contact with the roll at the instant of max. force in the chain and therefore this member turns out to be the one with the highest loading. In another instant (t=2.39s), when member Q is in contact with the roll, it is clear that this other member is not loaded as high.

Besides, when members pass over the roll, transversal vibrations occur with amplitude of 0.3÷0.5mm. These cause high frequency vibrations in the range 20 ÷ 30 Hz.

Analyzing figure 10 results has proved that lifting and sharp stopping have the same effect on the dynamic response of the system.

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