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ГЕОЛОГИЯ ЖӘНЕ ТЕХНИКАЛЫҚ ҒЫЛЫМДАР СЕРИЯСЫ

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СЕРИЯ ГЕОЛОГИИ И ТЕХНИЧЕСКИХ НАУК

♦

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CENTRIFUGAL PUMP ROTOR DYNAMICS STUDY

Abstract. Rotary machines just like other complex technical devices, are the subject to vibrations that can lead to harmful effects during operation, and sometimes to destruction of individual elements, for example, reference nodes . The main source of vibration in such machines is a rotating element - the rotor, to which the centrifugal forces act in presence of shell elements or their residual imbalance. This is the main and inevitable type of vibration of any rotary machine. The unbalanced rotor always oscillates with the reference frequency, that is, the rotation velocity. In this case, the resulting centrifugal forces can cause not only vertical and horizontal vibrations, but also, under certain conditions, axial vibrations.

The analysis of the dynamic behavior of the rotor under the influence of these forces should be carried out to any rotary machine both as at the design and operational stages, and so in case of operational accidents.

The purpose of this work is to determine the main dynamic parameters and characteristics of the rotor of a centrifugal pump, taking into account the shell elements, determination of critical rotation velocities and the derivation of results in the form of the Campbell diagram with visualization of the rotor motion paths to determine the danger of resonance modes. To achieve this goal, the NASTRAN engineering analysis system of the standardized DMAP procedure "Rotodynamics" was used. The rotor was modeled as a shaft of piecewise constant cross section with shell elements modeled in the form of concentrated masses with inertia of rotation. In the article the results of calculations of two design schemes of a rotor of the pump with fastening the end of a shaft at a guide support, in the form of a caprolon sleeve and at a bearing support are given.

Key words: rotor, centrifugal pump, dynamic analyses, critical velocities, Campbell diagram, calculating scheme, NASTRAN.

Introduction. Reliability and life of a centrifugal pump is largely determined by its vibrational state. The technology of calculating the critical rotational velocity of the pump rotor is complicated, and to date it is impossible to accurately determine it because of the impossibility of reliable prediction of the coefficients taking into account the effect of all possible factors that have an impact on the vibrational state of the pump [1, 2].

At the present stage of development of computer technologies, the problem of determining the eigenfrequencies of rotor systems based on linear mathematical models is well defined in automatic mode [3,4,5]. Dynamic processes occurring in rotor systems can be numerically modeled using ANSIS, as it was implemented in [6]. In [7, 8], methods are considered that allow one to take into account the gyroscopic moments of inertia of the shell parts.

The Patran module of the NASTRAN software package allows calculating the critical velocities of the rotor. The inclusion of gyroscopic terms in the computational model of the rotor occurs automatically with the help of the standardized DMAP procedure "Rotodynamics" included in the solution sequence of the MSC Nastran dynamic problem (in all versions, since version 2004) [9,10].

Methods

1. Theoretical provisions of numerical analysis of rotor dynamics. Free oscillations fully determine the dynamic properties of the mechanical system and are of primary importance in the analysis of forced oscillations [11], so using the finite element model we primarily determine the spectrum of the eigenoscillation frequencies of the rotor of the Centrifugal pump.

To describe the motion only under the action of the restoring (elastic) force without taking into account the energy dissipation, use the equation [12,13,14]

$$[M]\{\ddot{q}\} + [C]\{q\} = 0, \quad (1)$$

where [M], [C] – mass matrix (inertia) and rigidity of the system; $\{\ddot{q}\}$, [q] – generalized node displacements and their derivatives.

The solution of equation (1) is sought in the form [13]

$$\{q\} = \{q_0\} \sin \omega_0 t \quad (2)$$

where ω_0 – values of eigen-frequencies, $\{q\}$ – complete vector of nodal displacements of the system, $\{q_Q\}$ – Column-matrixof amplitudes.

The total vector q is a function of independent components of displacement and angles of rotation with respect to the corresponding axes. The complete displacement vector is represented in the form

$$\{q\} = \left\{\{q^{(1)}\}\{q^{(2)}\} \dots \{q^{(n)}\}\right\}^T. \quad (3)$$

In this case, the problem reduces to calculating the eigenvalues of the frequencies ω0 and the eigenvalues of the vectors of the generalized displacement *q*,this implies that *q* determines the shape of the eigen-oscillations at the corresponding value of the frequency ω0. When implementing the automated finite-element method for determining the eigen-oscillations, the numerical solution of the system of algebraic equations (3) with the algorithms of the PATRAN program is carried out using the Lanczos method [15,16].

Forced oscillations of the rotor occur under the action of the harmonic centrifugal inertia force of the unbalanced rotor masses, which is represented in the form $F_u = m\omega^2 e\cos(\omega t)$, then the equation of forced oscillations will be written as following

$$[M]\{\ddot{q}\} + [B]\{\dot{q}\} + [C]\{q\} = [Me]\omega^2 \cos(\omega t), \quad (4)$$

where [M], [B], [C] – matrices of mass (inertia), damping and rigidity of the system; $\{q\}$, $\{\dot{q}\}$, $\{\ddot{q}\}$ – generalized node displacements and their derivatives, ω – angular velocity of rotation, e – specific imbalance.

The solution of equation (6) is sought in the form

$$\{q\} = \{q_0\} \sin \omega_0 t + [Me]\omega^2 \cos(\omega t) \quad (5)$$

2. Features of simulation of rotors for various purposes. As an object of the research, the rotor of a multistage vertical submergible pump for aggressive environment was chosen. Figure 1 shows the design of the 3D rotor model with shell elements. The upper end of the shaft is attached to the motor, the lower end is fitted with a guide support in the form of a caprolon sleeve. The rotor is modeled by rod elements, to such a representation a shaft of almost any rotary machine can be adduced (turbine, compressor, expander, generator, etc.). Impellers shell elements are of complex configuration, for them the location of the center of mass on the shaft, mass, equatorial and polar moments of inertia must be known [17]. Currently used bearings (rolling, sliding, magnetic) can be modeled as rigidly clamped, hinged or resilient damper piers, depending on the degree of proximity to these options for the type of bearings used. In addition, for carrying out computational dynamics studies, such parameters of the rotor as the properties of the shaft material, residual imbalances of the shell elements, the acceleration characteristic and the range of operating rotor velocities should be known [18].

3. Automated calculation of eigen-frequencies and mode shapes of a rotor of a centrifugal pump. The initial data for the calculation are the physical properties of the material of the shaft (density ρ = 7850 kg / m3 and modulus of elasticity of the first kind E = 2,1 · 1011 N/m2), length *L*, outer *D* and inner *d* diameters of sections, mass *m*, as well as the rigidity of bearing piers. Figure 1 shows a 3D model of the design of the seven sectional centrifugal pump.

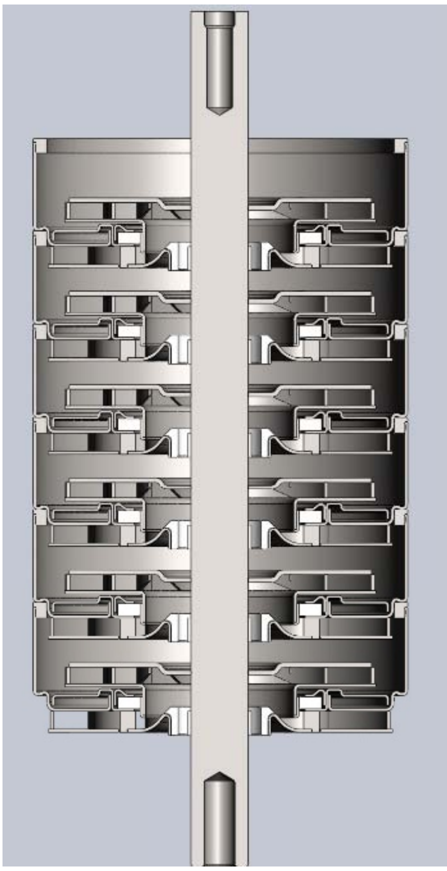
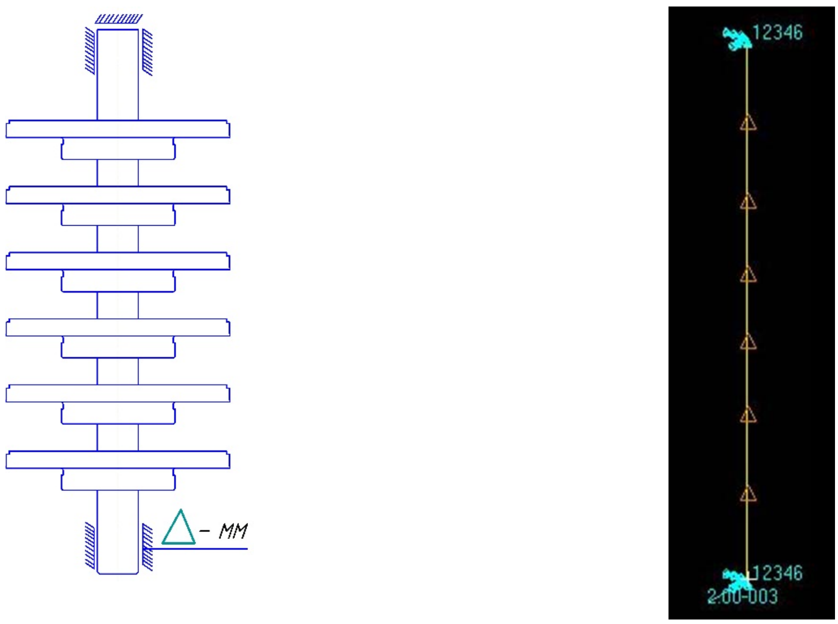


Figure 1 – 3D Model of Rotor System

According to the design scheme (figure 2), the complete model was built in Patran, it includes 7 elements of CBEAM (rotor shaft), 6 elements of CONM2 (a concentrated mass element simulating the rotor wheel).



a) b)

Figure 2 – The rotor of CP a) the design model b) the rod model of a rotor constructed in Patran

Simulation of fixedpiersis carried out by fixing the nodes of the rotor model to the corresponding degrees of freedom, in the design of the pump under study the pier was modeled taking into account the gap Δ-mm (figure 2).

The values of the rotor eigen-frequencies were found using the NORMAL MODELS solver (modal analysis).

Calculated values of eigen-frequencies

Parameter	Value
f_1- first eigen-frequency, Hz	8
f_2- second eigen-frequency, Hz	434
f_3- third eigen-frequency, Hz	447
f_4- fourth eigen-frequency, Hz	696
f_5- fifth eigen-frequency, Hz	732
f_6- sixth eigen-frequency, Hz	892

The coincidence of often disturbed oscillations with frequencies of natural oscillations presented in the table can lead to resonance phenomena.

4. Determination of disturbed oscillations frequencies. To determine the disturbed oscillations frequencies we use the COMPLEX EIGENVALUE solver (complex frequencies).

It is available to use the option of asynchronous precession (ASYNC) in the program to determine the response of the system to an external action that is independent of the rotation velocity. When using the synchronous precession option (SYNC), the system respond to an imbalance or other excitation, which depends on the rotor speed, is determined. With the help of complex shape analysis, it is possible to determine the oscillation frequencies corresponding to direct and retrograde precession, as well as the critical rotational velocities [19,20].

In the Spin Profile menu, the user sets individual rotation velocities of the rotor, for our centrifugal pump rotor problem, the angular velocity value is ω = 3000 rpm. Also the required moments of inertia of the hook-up wiring elements were defined in the CAD system of Solid Works.

Results and discussion. When choosing the calculation type, the calculation of complex eigenvalues SOL 107 is a direct method, the frequency diagram is obtained by calculating the complex eigenvalues by a direct method using the option ASYNC, at rotation velocities of 0, 100, 200, 300, 700 rpm.

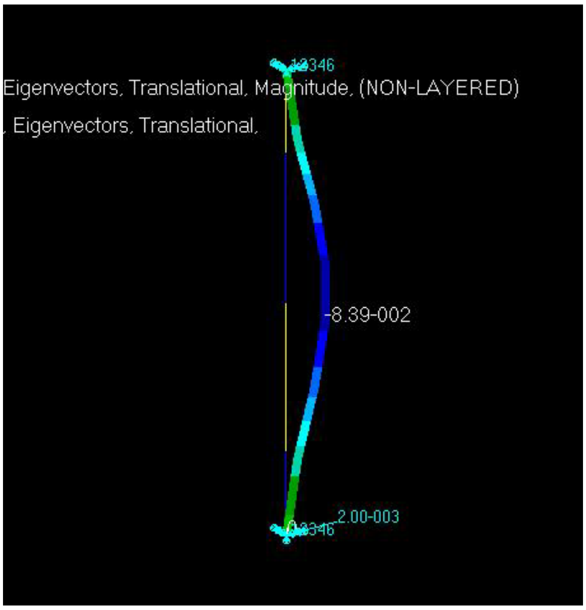


Figure 3 – The 5th form of disturbed oscillations at a frequency of 100 rpm

Obtained values of disturbed oscillations frequencies

Parameter	Value
f_1- first frequency, Hz	0
f_2- second frequency, Hz	463
f_3- third frequency, Hz	488
f_4- fourth frequency, Hz	732
f_5- fifth frequency, Hz	736

The critical velocities are determined based on which eigenvalues are identical to the rotation velocity of the rotor. To do this, a straight line corresponding to w = W is constructed on the diagram, i.e. (oscillation frequency = angular rotation velocity of the rotor). The points of intersection of the straight line with the eigen-frequency curves correspond to the critical rotational velocities of the rotor.

The automatically calculated eigenvalues (figure 4), corresponding to the identical oscillation forms, form a series of curves, which are the functions of changing the oscillation frequency from the angular rotational velocity of the rotor. At the shown Campbell diagram, all the multiple critical rotational velocities for the first waveforms 366, 488.732 Hz are in the 47 Hz (2200 rpm) range for retrograde and direct precession, respectively.

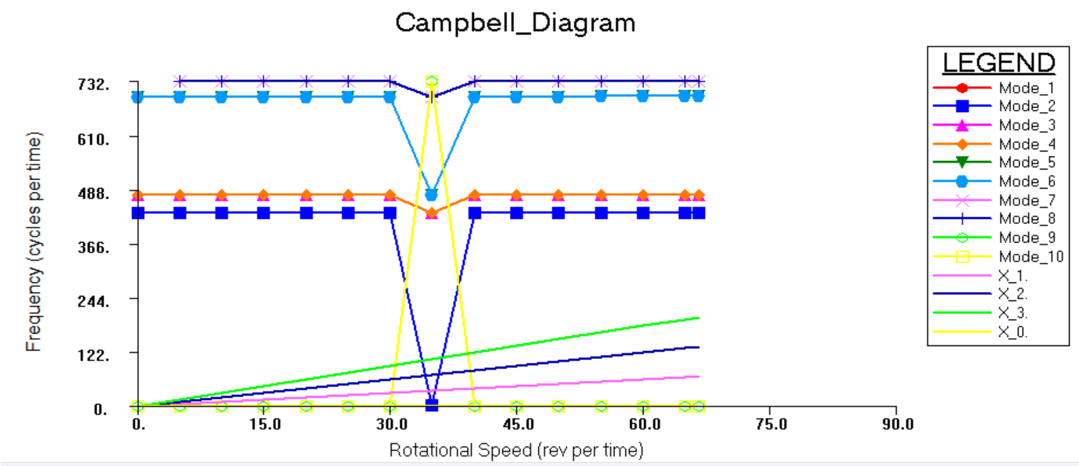


Figure 4 – The Campbell diagram

The polygonal character of the frequency lines indicates the unstable character of the vibrations associated with the structural features of the guide supports in the form of a gap of size Δ.

In order to optimize the vibration and predict more stable launch of the machine into operation, a calculation scheme was simulated with a pier, that imposes restrictions on the movement in the plane perpendicular to the plane of the axis of the rotor shaft. The task was calculated with the same input data as for the scheme presented above.

The frequencies of the disturbed oscillations in the second case of holdfastening are summarized in table.

Parameter	Value
f_1- firstfrequency, Hz	0
f_2- second frequency, Hz	315
f_3- third frequency, Hz	388
f_4- fourthfrequency, Hz	631
f_5- fifthfrequency, Hz	636

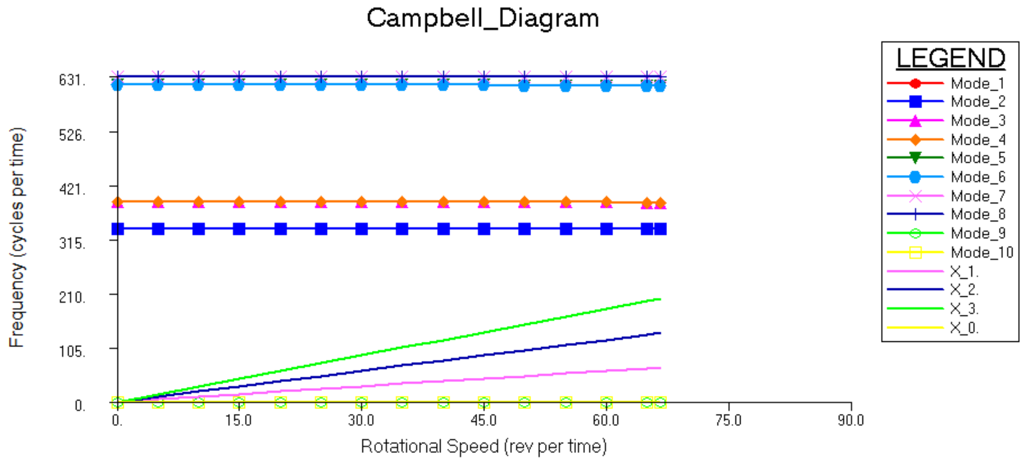


Figure 5 – Campbell diagram for a model with a support without a gap

Despite the fact that the vibration tones of the second model are lower by 20% than the models in the guide bearings, it can be said that the points of critical velocities (the intersection points of the frequency multiplicity lines with the frequency lines) are absent and the machine enters the operating mode relatively stably.

Conclusions. The determination of critical velocities is important for the system constancy assessment. When balancing real rotors, the residual imbalance valueiskept in the system. Any imbalance will cause oscillating forces in the rotor or support element. If the velocity of the rotor rotation is equal to the critical velocity, then the system undergoes strong enough vibrations, which can lead to damage or even functional failure. Knowledge of critical velocities allows the user to determine safe operating ranges. For the studied pump the critical velocities are 30,37,42 Hz.

To date, there is a sufficient number of analytical methods for analyzing the dynamics of rotary systems, however only modern computer technologies such as NASTRAN solver allow to quickly and adequately study the vibration parameters of machines and mechanisms, which also excludes the costs for vibration testing.

The researches carried out and the automated calculations have allowed to define tones of natural and disturbed oscillations, and also values of critical rotor velocities of the model of an operating vertical submersible centrifugal pump.

Two design schemes of the rotor system were distinguished, differing in the ways of securing the shaft, the first existing version is the guide support, the second variant is hypothetical, the support with two degrees of freedom without a gap. Comparison of the results of the calculation showed a difference between the tones of the disturbed oscillations in 20%, and also the greater stability of the mode of entry into the operating speeds of the rotor on the support without a gap. The second diagram of Campbell clearly shows the absence of points of intersection of the lines of frequencies of perturbed oscillations with direct corresponding $w = W$, i.e. (oscillation frequency = angular velocity of rotation of the rotor).

As a recommendation to give, it is possible to propose to the manufacturer of the centrifugal pump to change the design of the pump shaft holdfast, as a factor affecting the performance of the machinery.

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ОРТАЛЫҚТАН ТЕПКІШ СОРҒЫ РОТОРДЫҢ ДИНАМИКАСІН ЗЕРТТЕУ

Аннотация. Роторлы машиналары басқа да күрделі техникалық құралдары сияқты діріл әсеріне ұшырауға бейім келеді, бұл пайдалану процесінде зиянды зардаптарға әкелуі мүмкін, оларға жататындар пайдалы әсер ету коэффициентінің төмендеуі, мысалы, тірек түзілімдердің жекеленген элементтерінің қирауы. Мұндай машиналарда негізгі діріл көзі ретінде айналмалы элементі болып табылады – ротор, ол кезде аспалы элементтердің эксцентрикті қондырылуы немесе олардың қалдық теңгерімсіздігінен тепкіш күштер әсер етеді.Бұл кез келген роторлы машиналардың басты және шарасыздықты діріл түрі болады. Байсалды емес ротор әрқашан негізгі жиілігімен тербелістер жасайды, яғни ротордың айналу жиілігімен. Бұл ретте пайда болған ортадан тепкіш күштер тек тік және көлденең дірілдер ғана емес, сол сияқты белгілі жағдайда, осьтік дірілдерді тудыруы мүмкін.

Аталған күштердің әсерінен ротордың динамикалық тәртібінің талдауына - кез келген роторлы машина, өйткені жобалау және жетілдіру кезеңдерінде, сондай-ақ пайдалану авариялар туындаған кезде ұшырауы тиіс.

Бұл жұмыстың мақсаты болып орталықтан тепкіш сорғы ротордың сипаттамалары мен негізгі динамикалық параметрлерін анықтау болып табылады, аспалы элементтерін ескере отырып, критикалық айналу жылдамдығын анықтау және роторының қозғалыс траекториясын Кэмпбелла диаграммалар нәтижелерін шығару түрінде резонансты режимдер с қауіптілік анықтау үшін қолданылады.Алға қойылған міндеттерді іске асыру үшін инженерлік талдау NASTRAN жүйесі пайдаланылады стандартталған DMAP рәсімдері "Rotodynamics", ротор тілім тұрақты қималы аспалы элементтерімен біліктің түрінде модельденген болады, бұл бұрылу инерциясына ие шоғырланған масса түрінде модельденген. Мақалада екі құрылымдық ротордың сорғы схемаларын есептеу нәтижелері келтірілген, ол білік ұшының бағыттаушы тіректе, капоролонды төлке түрінде және мойынтіректі тіректер түрінде бекітуімен келтірілген.

Нәтижелері сорғы конструкциясын оңтайландыру үшін ұсынылуы мүмкін, сондай-ақ, көп сатылы ортадан тепкіш сорғы критикалық жылдамдығын компьютерлік есептеу әдісі ретінде ұсынылуы мүмкін.

Түйін сөздер: ротор, сорғы, динамикалық зерттеу, критикалық жылдамдық, Кэмпбелл диаграмасы, жиілік, есептеу сулбасы, NASTRAN.

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ИССЛЕДОВАНИЕ ДИНАМИКИ РОТОРА ЦЕНТРОБЕЖНОГО НАСОСА

Аннотация. Роторные машины, как другие сложные технические устройства, испытывают воздействие вибраций, которые могут приводить в процессе эксплуатации к пагубным последствиям, таким как снижение коэффициента полезного действия, разрушению отдельных элементов, например, узлов опоры. Главным источником такой вибрации в машинах является вращающийся элемент – ротор, на который при наличии посадки с эксцентриситетом навесных элементов, либо их остаточной несбалансированности действуют центробежные силы. Это основной и неизбежный вид вибраций любой роторной машины. Неуравновешенный ротор всегда совершает колебания с основной частотой, то есть с частотой вращения ротора. Центробежные силывозникающие при этом могут вызывать не только вертикальные и горизонтальные вибрации, но и, при определенных условиях, осевые.

Анализу динамического поведения ротора под воздействием указанных сил должна подвергаться любая роторная машина как на этапах проектирования и доводки, так и при возникновении эксплуатационных аварий.

Целью данной работы является определение основных динамических параметров и характеристик ротора центробежного насоса, с учетом навесных элементов, определение критических скоростей вращения и выведение результатов в виде диаграммы Кэмпбелла с визуализацией траекторий движения ротора для определения опасности резонансных режимов. Для реализации поставленной задачи использовалась система инженерного анализа NASTRANстандартизированной DMAP процедуры “Rotodynamics”, ротор был смоделирован в виде вала кусочно-постоянного сечения с навесными элементами, смоделированными в виде сосредоточенных масс обладающими инерцией поворота. В статье приводиться результаты расчётов двух конструктивных схем ротора насоса с закреплением конца вала в направляющей опоре, в виде капролоновой втулки и в подшипниковой опоре.

Результаты могут быть рекомендованы для оптимизации конструкции насоса, также как методика компьютерного расчета критических скоростей многоступенчатого центробежного насоса.

Ключевые слова: ротор, насос, динамический анализ, критическая скорость, диаграмма Кэмпбела, частота, расчетная схема, NASTRAN.

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