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Special aspects of calculation of lifting equipment reliability

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Abstract. Aim. *When designing lifting equipment as a whole, as well as of its elements it is desirable to perform not only deterministic strength estimations, but also a probabilistic calculation of major reliability indices. Theoretical approach to the calculation of major reliability indicators of lifting equipment is described by V.I Braude. In practice the calculation of reliability of lifting equipment is usually quite difficult, because the information about values for certain indices provided in literary sources is incomplete and discordant. It causes the necessity to use average reliability indices and to introduce different assumptions to the calculation. And the calculation results turn out to be rather approximate. At the same time an approximate calculation of reliability indices allows to decide on efficient use of one or another design layout of lifting equipment and/or its structural unit.* ***Methods.*** *To demonstrate the logical arguments that could be used at the calculation of reliability of lifting equipment, the article describes an example of calculation of probability of reliable operation for the lifting gear of an overhead crane, executed by a "detailed" scheme and consisting of nine elements: a three-phase induction electric motor with a short-circuit rotor; a parallel shaft double-stage gear box; a block brake with locking movement actuated by a coil spring and with breaking actuated by a short-stroke alternating electromagnet; flexible bolt coupling (with brake pulley); load drum; drum axle (or shaft); drum support; load cable and its mountings; hook assembly. Structurally, the elements of a lifting gear are connected in-series, i.e. in case of a failure of any element, the operable state of the gear is violated (a failure occurs).* ***Results.*** *The known experience of operation of lifting equipment shows that the most probable failures of a lifting gear's elements are the following failures: turn-to-turn short circuit of electric motor; wear out of bearings and gear teeth; turn-to-turn fault of a coil of a brake electromagnet; tearing up of a pulley of a flexible bolt coupling and break cheek wear out; fatigue breakdown of a drum and a bearing block, built into a drum; fatigue breakdown of a drum axle (or shaft); wear out of drum axle bearings, built into a drum; wear out (breakage) of wires and strands of a load cable; hook wear out and bearing freezing of a hook assembly. That is why the reference data used for calculation usually describe the probability of occurrence or a rate of these particular failures. Calculation was carried out with the following assumptions: Degradation (wear rout) failures were not taken into account, since they are anticipated during the phase of technical maintenance and repair; failures, caused by the violations of the rules of safe operation, were refer not to the crane failures, but to the failures of other systems. For descriptive reasons the elements of a lifting gear were chosen from the catalogue with a certain "margin" and without taking a loading mode into account.* ***Conclusions.*** *The calculation results showed that neglecting various load-bearing factors (for instance, a gear box underload by a rotation moment) may lead to excess reliability of a crane as a whole, its machinery and structure components.*

Keywords: *probability of reliable operation, lifting equipment, lifting gear, overhead crane, reliability indices, designing, calculation.*

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The calculation of reliability indices of lifting equipment as a whole, as well as of its components is quite difficult, because the values for certain indices provided in literary sources [1–3, etc.], are incomplete and discordant. To define missing values, average indices have to be used. And the calculation results turn out to be rather approximate.

The values of time, for which the probability of reliable operation is calculated, can be taken as $t = 1$ year. The number of hours of a gear's operation for 1 year shall be defined as follows [4]:

$$t = 8760 \cdot K_{ИГ} \cdot K_{ИС} \cdot ПВ, \text{ hour,} \quad (1)$$

where $K_{ИГ}$ and $K_{ИС}$ are the factors of use of a calendar time of the year and of a day respectively; ПВ is a duty rating.

For certain lifting, construction equipment and road machinery in general the values of these factors are listed

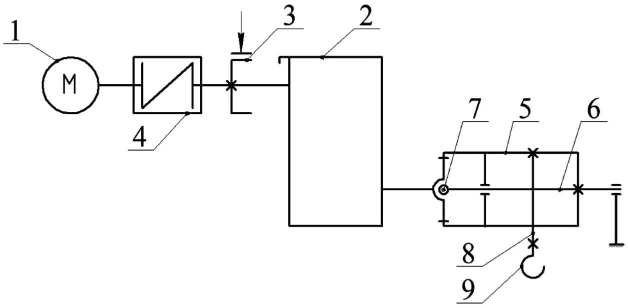

Fig. 1. Kinematic scheme of an overhead crane lifting gear: 1 – electric motor; 2 – gear box; 3 – electromagnet block brake; 4 – flexible bolt coupling (with brake pulley); 5 – drum; 6 – drum axle (or shaft); 7 drum support; 8 – load cable; 9 – hook assembly

in [4]. For crane electrical equipment the values of factors $K_{ИГ}$ and $K_{ИС}$ are given in [2] and in [5].

According to VNIIPTMASH [5] for the operation modes L; M; H; VH the estimated times of operation of crane electro motors are not more than 250; 1000; 3000; 4000 hrs/year respectively. The operation modes L; M; H; VH are given by obsolete rules of Gosgortechnadzor of 30.12.1969. Correlation between operation modes of cranes and crane gears according to the Rules of Gosgortechnadzor, GOST 25835-83, GOST 25546-82 and ISO 4301/1 is provided in [10]. Correlation between operation modes of cranes and crane gears for different foreign standards is given in Tables 1 and 2.

Theoretical approach to the calculation of major reliability indicators of lifting equipment is described by V.I Braude [6]. Let us dwell on a pragmatic angle of this issue. Standards have provisions for the probability of reliable operation as the main reliability index of an object. That is why let us consider the calculation of reliability indices of lifting equipment on the example of calculation of the probability of reliable operation of an overhead crane lifting gear (Figure 1) that consists of 9 elements [7]. Structurally, the elements of a lifting gear are connected in-series, i.e. in case of a failure of any element, the operable state of the gear is violated (a failure occurs).

1. Electric motor. From Guide [2] it is known that for 4A electric motors the probability of reliable operation is equal to 0,9 with 10000 hrs of operation time.

Based on the assumption about the exponential law of distribution:

$$P\left(10^4\right) = \exp\left(-\frac{10^4}{T_1}\right) = 0,9,$$

where T_1 is a mean time to failure of an electric motor, hrs.

$$T_1 = \frac{10^4}{\ln 0,9} = \frac{10^4}{\left(-0,105\right)} = 9,5 \cdot 10^4.$$

2. Gear box. According to VNIIPTMASH [5] a failure rate ω_2=0,2 per 1 thousand hrs. Then the gear box's mean time to failure is:

Table 1. Correlation of operation modes of a crane for foreign standards

ISO 4301/1	PN-79 M-06503 (Poland)	BS 466-84 (Great Britain)	SFS 4300-79 (Finland)	DIN 15018 (Germany)	B 4004-1 (Austria)
A1	1	A1	1	B1	T1
			2		
A2		A2	3	B2	T2
A3		A3	4	B3	T3
A4	2	A4			
A5	3	A5	5	B4	T4
A6	4	A6			
A7	5	A7	6	B5	T5
A8	6	A8		B6	T6

Table 2. Correlation of operation modes of crane gears for foreign standards

ISO 4301/1	СТ СЭВ 2077-80	CSN 27009 (Czech Republic)	BS 466-84 (Great Britain)	SFS 4020-80 (Finland)	DIN 15018 (Germany)	FEM 9.661
M1	1		M3	ImB	IEm	IDm
					IDm	
M2					ICm	ICm
M3					IBm	IBm
M4	2		M4	ImA	IAm	IAm
M5	3		M5	2m	2m	2m
M6	4		M6	3m	3m	3m
M7	5		M7	4m	4m	4m
M8	6		M8	5m	5m	5m

$$T_2 = \frac{1}{\omega_2} = \frac{1000}{0,2} = 5\cdot 10^3 \text{ hrs.}$$

If under the calculation of the gear box a loading mode was not taken into account, there is usually an underload by an equivalent rotation moment, i.e. we will have a longer mean time to failure:

$$T_2^{/} = T_2 / \mathrm{K}_Q,$$

where K_Q is an equivalent loading coefficient [8].

$$K_Q = \sqrt[m]{\sum\left(\frac{Q_i}{Q_H}\right)^m \frac{t_i}{t}}, \tag{2}$$

where Q_i is a random value of the lift load's weight (defined by a load schedule for the respective operation mode); t_i is time spent on operation with load goods Q_i; t is total time; m is a degree of durability line.

At the calculation of work surfaces of gear boxes gear teeth for back-to-back endurance m=3 [9]. If we do not know a load schedule, then we can take an assumption: common cranes lift 15% of loads with nominal weight and 85% of loads with weight 0,5 Q_{H} [10]. Then based on the formula (2) we will have:

$$K_Q = \sqrt[3]{1^3\cdot 0,15 + 0,5^3\cdot 0,85} \approx 0,63.$$

Now, with consideration of equivalent loading we get a higher value of the gear box's mean time to failure:

$$T_2^{/} = T_2 / 0,63.$$

If under the calculation of the gear box a loading mode was taken into account, then the estimations by formula (2) are not made, i.e. $T_2^{/} = T_2$.

At the arrangement of a lifting gear, unified and normalized assembly units are applied [11]. That is why a gear box is usually chosen from a catalogue with a capacity margin, i.e. there is an underload in capacity which causes a higher value of mean time to failure:

$$T_2^{//} = T_2^{/}\left(\frac{N_{\mathrm{K}}}{N_{\mathrm{H}}}\right)^3, \tag{3}$$

where N_{K} and N_{H} is the gear box's capacity according to the catalogue and its specified capacity, respectively.

Let the following values be obtained at the design phase N_{K} = 10 kW; N_{H} = 5 kW, then:

$$T_2^{//} = T_2^{/}\cdot\left(\frac{10}{5}\right)^3 = 8T_2^{/}.$$

Finally we have the gear box's mean time to failure:

$$T_2^{//} = T_2^{/}\cdot\frac{8}{0,63} = 12,7\cdot T_2 = 12,7\cdot 5000 = 6,35\cdot 10^4 \text{ hrs.}$$

At first sight such mean time to failure of the gear box seems to be incredible – about 7 years of reliable operation. But we should remember that we have almost a quadruple unload, and besides a perfect compliance with a maintenance schedule is provided (including regular change of oil and cup seal). There are no ageing components in the gear box.

3. Brake. We know from the guide [2] that MO brake magnets admit up to 600 activations per hour. However considering their limited wearing capacity, the application of this type of brake gears should be limited by the frequency of activations of not more than 300 1/hrs – for electromagnets MO 100B and not more than 150 1/hrs – for electromagnets MO 200B. Under these operation modes and voltage fluctuations within 85…105% of rated voltage, electromagnets have the probability of reliable operation about 0,95 per a year of operation. With great probability we can assume that the value of reliable operation of an electromagnet is given for the operation mode "H" (MO 100B) and for the mode "M" (MO 200B) in view of the limitation of activation frequencies, as well as for a maximum value of a braking moment M_{Tmax}, for which we have a maximum force of tightening and current in a coil.

Time of electromagnet operation per year by formula (1) is:

for MO 100B: t=2,3·10^3 hrs;

for MO 200B: t=0,6·10^3 hrs.

Based on the assumption about the exponential law of distribution:

for MO 100B:

$$P\left(2,3\cdot 10^3\right) = \exp\left(-\frac{2,3\cdot 10^3}{T_3}\right) = 0,95 \Rightarrow T_3 = 44,8\cdot 10^3 \text{ hrs;}$$

for MO 200B:

$$P\left(0,6\cdot 10^3\right) = \exp\left(-\frac{0,6\cdot 10^3}{T_3}\right) = 0,95 \Rightarrow T_3 = 11,7\cdot 10^3.$$

The most probable failures are the punctures of turn-to-turn insulation of an electromagnet's coil. They occur as the result of insulation ageing, cracks, lacquer and fabric peeling. It is encouraged by heat and coil vibration. If the brake is adjusted for a smaller braking moment M_{T} a spring force will be reduced, coil current, its heat and vibration will be reduced as well, and a mean time to failure of a coil will increase in accordance with quadratic dependence [2]:

$$T_3^{/} = T_3\left(M_{\mathrm{T_{max}}} / M_{\mathrm{T}}\right)^2. \tag{4}$$

Let (hypothetically) M_{Tmax}=2M_{T}, then:

$$T_3^{/} = T_3\left(2M_{\mathrm{T}} / M_{\mathrm{T}}\right)^2 = 4T_3.$$

It means that we have mean values of time to failure:

for MO 100B: $T_3^{/} = 4\cdot 44,8\cdot 10^3 \approx 18\cdot 10^4$ hrs;

for MO 200B: $T_3^{/} = 4\cdot 11,7\cdot 10^3 \approx 4,7\cdot 10^4$ hrs.

Let us introduce the latter value to the further calculation.

4. Flexible bolt coupling with brake pulley and brake cheeks. We suppose that if maintenance schedule is met,

loose hubs and brake cheeks are changed periodically. Tearing up of a brake pulley that causes early wear of cheeks is considered as a sudden failure, whose probability is exponentially distributed with parameter T_4 [6]. Let us nominally consider a coupling and a friction pair equally reliable in relation to an electromagnet's coil:

$$T_4 \approx T_3 = 4{,}7 \cdot 10^4 \text{ hrs.}$$

5. Drum. We consider a failure of a drum's bearing and cable mountings to be possible only when a crane is tested for a lifting with a load weight $Q=1{,}25Q_{\text{Н}}$ [11]. That is why the above listed details are considered almost failure free in operation:

$$P_5(1 \text{ year}) \approx 0{,}99.$$

6. Drum axle (or shaft). Estimation of an axle or shaft of a drum is carried out in the following order [12]:

6.1. Axle or shaft are calculated for fatigue, load factor n_t in weak section with the time of operation t=1 year is also determined. Let n_1=2.

6.2. The value of coefficients of variation of endurance limit of detail V_{-1g} is validated. This limit is determined by dispersion of a scaling factor, stress concentration factors, melting dispersion of steel chemistry. We can take V_{-1g}=0,1.

6.3. The coefficient of variation of an equivalent cycle amplitude V_a is validated. It is determined by the difference of crane operation modes form the estimated ones. We can take V_a=0,3.

6.4. The probability of axle fatigue breakdown in a weak section is defined:

$$F(t) = F_0\left(\frac{1-n_t}{\sqrt{n_t^2 \cdot V_{-1g}^2 + V_a^2}}\right), \tag{5}$$

where $F_0(x)$ is the function of normal distribution [13].

$$F_6(1 \text{ год}) = F_0\left(\frac{1-2}{\sqrt{2^2 \cdot 0{,}1^2 + 0{,}3^2}}\right) =$$
$$= F_0(-2{,}77) \approx 0{,}002.$$

As we have two weak sections (in support), let us double the obtained value. In other sections a load factor is higher, but a certain probability of a breakdown still exists, that is why let us double the value of a breakdown probability once more:

$$F_6(1 \text{ year}) = 2 \cdot 2 \cdot 0{,}002 = 0{,}008 \approx 0{,}01,$$

$$P_6(1 \text{ year}) = 1 - F_6(1 \text{ year}) = 1 - 0{,}01 = 0{,}99.$$

7. Drum axle bearing. When calculating a bearing for durability, a calendar operation time is taken into account, and based on formula (1) the number is found, and then the number of load cycles (resource), after that considering an equivalent loading the bearing is chosen [8, 14].

Under such calculation α% of bearings shall exceed the specified life, i.e. the probability of reliable operation of the chosen bearing during T years is more than α.

Let α=0,9, T_7=10 years. It is necessary to define the probability of a failure of the bearing after one year of operation, but meant for 10 years of operation.

Share of the expired operation life is 0,1 of the calculated operation life:

$$\gamma = t/T_7 = 0{,}1.$$

If the bearing is taken with a margin of lift capability $C_{\text{Д}}$, then its life increases in accordance with a cube dependence, as the exponent of the bearing's curve durability m=3 [14].

Let $C_{\text{Д}}$=1860, and according to the catalogue we have $C^{/}_{\text{Д}}$=1400, then:

$$T_7^{/} = \left(\frac{C_{\text{Д}}}{C_{\text{Д}}^{/}}\right)^3 T_7 = \left(\frac{1860}{1400}\right)^3 T_7 = 1{,}33T_7.$$

The relation $T_7/T_7^{/}$ is 1,33 smaller than t/T_7=0,1, i.e.:

$$\gamma^{/} = \frac{0{,}1}{1{,}33} = 0{,}07.$$

The probability of reliable operation of the bearings is determined by Weibull distribution [15]:

$$P_7^{/}\left(T_{7\gamma^{/}}\right) = \exp\left(-\frac{\gamma^{/}}{5{,}35}\right)^{1{,}34} = \exp\left(-\frac{0{,}07}{5{,}35}\right)^{1{,}34} = 0{,}997.$$

On the drum axle there is a joint 7 on the right end of the axle (Figure 1). The inner and outer rings of the joint do not have relative rotation, since the drum 5 and the drum axle 6 rotate with at the same speed. The bearing which does not rotate is considered less reliable. Let us take for it $P_7^{//}$=0,995. Finally we have the probability of reliable operation of the drum axle bearing and the joint:

$$P_7^{///} = P_7^{/} P_7^{//} = 0{,}997 \cdot 0{,}995 = 0{,}992 \approx 0{,}99.$$

8. Load cable with mountings. At under-control operation broken cable wires are calculated and registered in the log book [16]. As soon as a rejection number of broken wires is achieved, the cable is changed. In this case if no rules are violated [16] the cable failures are considered improbable. Let us take:

$$P_8(1 \text{ year}) \approx 0{,}99.$$

9. Hook assembly. Usually it has a multiple strength margin, it undergoes testing and is almost failure-free. Let us take:

$$P_9(1 \text{ year}) \approx 0{,}99.$$

All data obtained as the result of the above listed reasoning and calculations are brought together in table 3.

10. Calculation of probability of a lifting gear's reliable operation as a whole

Mean time to failure of the elements 1…4, connected in series:

$$\frac{1}{T}=\frac{1}{9,5\cdot 10^4}+\frac{1}{6,35\cdot 10^4}+\frac{1}{4,7\cdot 10^4}=0,47\cdot 10^4 1/\text{ hrs}$$

$$\Rightarrow T=2,1\cdot 10^4 \text{ hrs.}$$

Based on formula (1) let us define the time of operation of a lifting gear at the operation mode "H":

$$t=8760\cdot 1\cdot 0,66\cdot 0,4=2,3\cdot 10^3 \text{ hrs.}$$

The probability of reliable operation of the elements 1…4, connected in series:

$$P_{1\ldots 4}(t)=P_{1\ldots 4}(1\text{ год})=P_{1\ldots 4}\left(2,3\cdot 10^3\text{ час}\right)=$$
$$=\exp\left(-\frac{2,3\cdot 10^3}{2,1\cdot 10^4}\right)=0,853.$$

Then the probability of reliable operation of a lifting gear is:

$$P_{\text{МП}}=0,853\cdot 0,99^5\approx 0,85.$$

To calculate the reliability level of a crane as a whole let us consider that a crane usually has 3 gears (lifting gear, gear of trolley movement and gear of crane movement), steel construction and control equipment, i.e. 5 systems connected in-series. Supposing that (in order to simplify) they are equally reliable we have the following probability of a failure of a crane during 1 year:

$$P_{\text{К}}(1\text{ год})=\text{P}_{5\text{ мех}}(1\text{ год})=0,85^5=0,443.$$

Thus a mean time to failure of a crane as a whole shall be:

$$P_{\text{К}}(1\text{ год})=\exp\left(-\frac{2,3\cdot 10^3}{T_{\text{К}}}\right)=0,443\Rightarrow$$

$$\Rightarrow T_{\text{К}}=-\frac{2,3\cdot 10^3}{\ln(0,443)}\approx 2,83\cdot 10^3.$$

Failure rate of a crane as a whole will be:

$$\omega_{\text{К}}=\frac{t}{T_{\text{К}}}=\frac{2,3\cdot 10^3}{2,83\cdot 10^3}\approx 0,81\text{ 1/year.}$$

For overhead hook electric cranes of general purpose with a lifting capacity up to 50 t in the operation mode "H", the parameter of the flow of sudden failures as per [17] is $12\cdot 10^{-3}$ 1/hrs, i.e. we can tolerate 12 failures of a crane for 1000 hrs, or 28 failures per a year. The obtained value $\omega_{\text{К}}=0,81<28$ i.e. not more than 1 failure per a year, is more than acceptable.

Conclusion

1. Probability of reliable operation of a lifting gear during 1 year of operation is 0,85.

2. A crane as a whole will have not more than one failure per a year. It is much lower than the tolerable value that is why we shall consider the reliability index of the lifting gear rather high.

3. Calculated probability of reliable operation of a crane is so high due to the number of reasons:

It was determined by a lifting gear which is the most reliable of all gears, then the result was distributed to the whole crane;

The elements of a lifting gear were chosen from the catalogue with a certain "margin";

Degradation (wearout) failures were not taken into account, since they are anticipated during the phase of technical maintenance and repair;

Failures caused by the violations of [16] refer not to the crane failures, but to the failures of other systems.

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Table 3. Results of calculation of reliability indices of an overhead crane's lifting gear

	Element	Most probable failures	Failure occurrence	P_i or T_i
1	Electric motor	Turn-to-turn short circuit	Heat, smell	T_1=9,5·10^4 hrs
2	Gear box	Wear out of bearings and gear teeth	Noise, vibration	T_2=6,35·10^4 hrs
3	Brake electromagnet	Turn-to-turn fault of a coil	Pulley not released	T_3=4,7·10^4 hrs
4	Flexible bolt coupling and brake cheeks	Tearing up of a pulley Cheek wear out	Long breaking distance	T_4=4,7·10^4 hrs
5	Drum, bearing block	Fatigue breakdown	Slap	P_5=0,99
6	Drum axle	Fatigue breakdown	Slap and drum freezing	P_6=0,99
7	Drum axle bearing and a joint	Wear out	Noise, vibration	P_7=0,99
8	Load cable	Breakage	Kinking	P_8=0,99
9	Hook assembly	Wear out, bearing freezing	Noise, locking	P_9=0,99

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