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SAFETY OF WINDER DISC BRAKES - A STRUCTURAL APPROACH

Summary. The paper deals with safety assessment of ASEA disc brakes assembled in the Polish - made winders. Having defined a safety of winder brake the author has presented reliability diagrams referring to brake tasks essential from point of wiev of winding installation safety. When deriving expressions determining brake system safety, probabilities of brake element failures have been taken into account. Two versions of brakes (furnished with one and two discs) have been discussed with emphasis on working assembly. An example of brake safety assessment (under static conditions) has been also given.

1. INTRODUCTION

The reliability problems of technical object in the view of safety requirements were considered among other papers in [1, 2, 3], while articles [4, 5, 6, 7] delt with the question at issue, namely with some aspects of safe exploitation of winder brakes.

The term "safety" is understood here as an abbreviation of "reliability of safety".

As [5] has it, a winder brake safety is brake fastness to the so-called dangerous failures resulting either in:

- complete destruction of the brake itself (this event being remote) or,
- producing a catastrophic effect in winding installation, e.g. multiple injuries and falatities in the case of man winding or substantial financial loss due to a halt in mineral winding.

This can be an overtravel resulting in striking the crash beams.

If the most important criterion of hoist safety is to stop its motion harmlessly, than the safety of brake operation depends on correct fulfilment of:

- emergency braking.
- brake holding.

These two tasks of the brake have been discussed in the paper. However safety assessment has been confined to the brake holding only, due to the relatively simple adoption of statutory requirements under static conditions of braking.

2. RELIABILITY DIAGRAMS OF DISC BRAKE SYSTEM

Fig. 1 depicts a certain version of ASEA hydraulic disc brake [9], while indications of components are described in the text. This functional diagram has been a basis to form reliability structures at three (I - III) levels of brake system decomposition. As it has been proved in [4,5], reliability diagrams (structures) illustrating the accomplishment of brake tasks (influencing safety) can be reduced, unlike those of "operational" reliability. Let us analyse brake reliability diagram at decomposition level I, fig. 2 This is a series system comprising two components: A_1 (A_2) - pump part and B - the so-called common part of brake. At level II (fig. 2), reliability diagram consists of the following components:

Z1 - Working assembly (brake units, frames, discs),

Z2 - Feeding assenbly (electric motor, hydraulic pump, valves),

Z2′- Feeding assembly of component B (valves, pipes, hoses),

Z3 - Control assembly (electrohydraulic valces, pressure accumulator, etc.),

(Z4 - Switch gear)

The above - mentioned assemblies compose either series (single disc brake - version "a") or series / parallel system (two disc brake - version "b").

Allowance must only be made for swith gear (Z1) in a certain version of the brake. Hence, this component has not been taken furter into consideration. Connections of inner reliability structures of working and feeding (part B) assemblies vary, so to simplify them a series link has been assumed.

Level III of the brake decomposition corresponds with partition of assemblies into elements.

A reliability diagram of working assembly (fig. 3) is formed with: frames (stands, S), brake disc (T) and brake units (SH) in series/parallel system, while safety of a set of brake units can be modelled after a pattern of system of n identical components. Fig. 4.1 presents reliability diagram of feeding assembly (Z2), comprising pipes (R) and valves

(V1) connected in series, then fig. 4.2 - diagram of feeding assembly of component B (Z2′) with pipes, hoses (G) and valves (V) also in series. When analysing fig. 4.2 one can notice that hydraulic pump (P) has not been included in the emergency braking reliability diagram.
For it is this brake task when pump is being switched off, thus becoming control assembly component (fig. 5).

The control assembly (under emergency braking) is characterized by a complicated reliability structure first of all consisting of electrohydraulic multiway valves (e.g. V2), pressure valves (V3, V4), pressure accumulator (H), etc. The most important elements participating in emergency braking have been marked on fig. 1. Fig. 5 shows a simplified reliability diagram of this assembly. Allowance has been only made for the valves which dangerous failures consisted in oil flow blocking (valve stoppage). As on fig. 5, three unequally loaded branches form series/ parallel reliability diagram of the control assembly. Needless to say, the diagram should be supplementd, for instance, with pressure accumulator and cooperating valves.

3. THEORETIC ASSESSMENT OF BRAKE SAFETY

Having constructed reliability diagrams some indices characterizing brake safety can be considered.

To do so, a probability of brake to fail when fulfiling its task essential from safety wievpoint must be determined.

In general from, this probability is as follows:

$$Q = Q\left\{A \cup B\right\} \tag{1}$$

Where:

$Q\left\{A\right\}$ - probability (stationary) of failure of pump part A_1 or A_2 (either operating),

$A\left\{B\right\}$ - probability of failure of common part of brake

$$Q\left\{A\right\} = Q\left\{Z2 \cup Z3\right\} \tag{2}$$

$$Q'\left\{B\right\} = Q\left\{Z1 \cup Z2' \cup Z4\right\} \text{ - one disc version} \tag{3}$$

$$Q''\left\{B\right\} = Q\left\{\left[(Z1 \cup Z2') \cap (Z1 \cup Z2')\right] \cup Z4\right\} \text{ - two disc version} \tag{4}$$

Assuming failures be independent and ignoring product probabilities, expressions (1)-(4) can be given in form:

$$Q' = Q\left\{Z2\right\} + Q\left\{Z3\right\} + Q\left\{Z1\right\} + Q\left\{Z2'\right\} + Q\left\{Z4\right\} \tag{5}$$

or

$$Q'' = Q\left\{Z2\right\} + Q\left\{Z3\right\} + \left[Q\left\{Z1\right\} + Q\left\{Z2'\right\}\right]^2 + Q\left\{Z4\right\} \tag{6}$$

where

$$Q\left\{Z1\right\} = Q\left\{\left[(S_1U\binom{n_1}{k_1}\ SH) \cap (S_2U\binom{n_2}{k_2}\ SH)\right]\ UT\right\} \tag{7}$$

and $k_1 + k_2 = k, \quad n_1 + n_2 = n$

$$Q\left\{Z2\right\} = Q\left\{RUGUV\right\} \tag{8}$$

$$Q\left\{Z2'\right\} = Q\left\{RUV1\right\} \tag{9}$$

$$Q\left\{Z3\right\} = Q\left\{\left[V3U(V2 \cap P)\right]\ U\left[V4 \cap (V2UP)\right]\right\} \tag{10}$$

remark: according to functional analysis of the control assembly version.

$$Q\left\{Z4\right\} = 0$$

remark: for the majority of disc brake versions.

In the case of brake holding expression (1) reduces itself (chapter 4) to:

$$Q = Q\left\{Z1\right\}$$

Generally, brake safety level can be determined from expression: $R_s = 1 - Q(1 - Q'$ or $1 - Q'')$ where R_s is a safety index (reliability of safety index).

Application of probabilities of failures as a basis for safety index calculations results from the fact that safety objectives have been established similiarly [8]. As stated in the cited report probability of catastrophic affect shold not exceed 10^{-8} (or 10^{-9} if considering man winding). Hence, the safety index obligatory for (disc) brakes must be of higher order which means lower acceptable probability of brake system failure Q.

4. EXAMPLE OF DISC BRAKE SAFETY ASSESSMENT

For the time being, safety level can be exclusively determined in the case of brake holding. Working assembly is the one that fulfills the brake task under static operation conditions (fig. 3). Examinations of exploitation processes have revealed no brake frames failures. Hence, there has been assumed that $Q_s \gtrsim 0$. Working assembly reliability diagram has turned thereby its form into that presented on fig. 6.

If "n" hydraulic brake units (SH) stand for a sub-system of working assembly, then such sub-system is in down-time state given at most "k" brake units operate.

Let us calculate the number of "k" brake units in relation to factor of brake holding safety (abbr. FBHS) understood as a ratio of brake holding torque to maximum static torque (due to an overweight of winding installation). The values of "k" and corresponding number "n" of brake units used in a given type of winder are presented in table 1 for man winding (FBHS $\geqslant 3$) and mineral winding (FBHS $\geqslant 2{,}5$).

Table 1

Number of brake units assembled n			Number of brake untis operating k	
			Mineral winding	Man winding
Winders with	one disc	$n_{min} = 6$ [10]	2	2
		10	4	3
		12	4	4
		14	5	4
		16	6	5
	two discs	20	8	6
		24	9	8
		28	11	9
		32	12	10

Probability of failure of brake units sub-system is given by:

$$Q_{k/n} = \binom{n}{k} Q_{SH}^{n}(1 - Q_{SH})^{n-k} \qquad (11)$$

where:

Q_{SH} - probability of brake unit failure causing substantial (or complete) drop in its braking force, e.g. piston stoppage.

Probability $Q_{k/n}$ values calculated for $Q_{SH} = 4 \times 10^{-4}$ according to [4] are presented in table 2.

Table 2

		Probability of sub-system failure	
		Mineral winding	Man winding
Winders with	one disc	$Q_{2/6} = 6{,}1 \times 10^{-20}$	$Q_{2/6} = 6{,}1 \times 10^{-20}$
		$Q_{4/10} = 2{,}2 \times 10^{-32}$	$Q_{3/10} = 1{,}2 \times 10^{-32}$
		$Q_{4/12} = 8{,}3 \times 10^{-39}$	$Q_{4/12} = 8{,}3 \times 10^{-39}$
		$Q_{5/14} = 5{,}3 \times 10^{-45}$	$Q_{4/14} = 2{,}7 \times 10^{-45}$
		$Q_{6/16} = 3{,}4 \times 10^{-51}$	$Q_{5/16} = 1{,}9 \times 10^{-51}$
	two discs	$Q_{8/20} = 1{,}4 \times 10^{-63}$	$Q_{6/20} = 4{,}2 \times 10^{-64}$
		$Q_{9/24} = 3{,}6 \times 10^{-76}$	$Q_{8/24} = 2{,}0 \times 10^{-76}$
		$Q_{11/28} = 1{,}5 \times 10^{-88}$	$Q_{9/29} = 4{,}9 \times 10^{-89}$
		$Q_{12/32} = 4{,}0 \times 10^{-101}$	$Q_{10/32} = 1{,}1 \times 10^{-101}$

When analysing table 2 one can find that probabilities of sub-system failures are extremely remote even for the least advisable version equipped with three pairs of brake units ($Q_{2/6}$).

As reads the paper [8], probability of brake unit failure is of 10^{-6} order.

The maximum acceptable value of this probability could equal $Q_{SH} = 10^{-2}$, since for the six brake unit version:

$$Q_{2/6} = 1{,}4 \times 10^{-11}$$

while for the remaining ones the values of $Q_{k/n}$ are respectively lover.

There are some problems how to determine safety of brake disc. Practically, dangerous failures affecting discs resolve themselves mainly to oil contaminations. These could result either from effluents in main shaft bearings or leakages in hydraulic elements of the brake. Taking no account of reasons the probability of brake disc contamination can roughly be estimated at $Q_T \approx 5 \times 10^{-7}$ with reference to [4, 5, 6].

As proved in [11], oil contaminations make static and kinetic coefficients of friction decrease almost three times. So does braking torgue. However, these assumptions seem to be very strict since all brake linings are not likely to be contaminated to the same degree.

The triple drop in braking holding torque:

a) does not meet the reqiurement of FBHS > 1 in the case of one disc brake (FBHS = 0,83 for mineral winding, FBHS = 1 for man winding),

b) meets the requirement of FBKS > 1 in the case of two disc brake and single disc be contamined (FBHS = 1,66 for mineral winding, FBHS = 2 for man winding),

c) does not meet the reguirement in the case of two disc brake and both discs be contamined (FBHS as in a).

The foregoing statement justifies the assumed method for brake discs interconnection, as seen on the reliability diagram, fig. 6. Probability of brake failure (when holding task is being carried out), is,

- for one disc brake:

$$Q_H = Q_{k/n} + Q_T \tag{12}$$

- for two disc brake:

$$Q''_H = Q_{k/n} + Q_T^2 \tag{13}$$

provided that as well $Q_{k/n} \ll 1$, $Q_T \ll 1$ as simultameous occurence of brake units stoppage and oil contamination of discs be improbable.

Analysis of expressions (12) and (13) displayes that for all $Q_T < 1$ the safety level of two disc brake is higher then that of one disc version, since:

$$R''_{SH} = 1 - Q''_H > R'_{SH} = 1 - Q'_H \tag{14}$$

When considering a certain winder, say of $\frac{4L - 3400}{2400}$ type, the probabilities are as follows:

$$Q_{k/n} = Q_{4/12}$$

$$Q_{4/12} = 4{,}6 \times 10^{-22} \quad \text{for} \quad Q_{SH} = 10^{-2}$$

Hence:

$$Q'_H = 4{,}6 \times 10^{-22} + 5 \times 10^{-7} \tag{15}$$

$$Q''_H = 4{,}6 \times 10^{-22} + (5 \times 10^{-7})^2 \text{, should this version be produced} \tag{16}$$

Having ignored the first constituent in expressions (14) and (15) it can be concluded that the probability of two disc brake to fail dangerousl is about 2×10^6 less then the probability of failure for the brake furnished with single disc (under the static condition).

In the case of two disc brake the level of safety can be assessed positively. Then, brake discs are the most important elements influencing the safety of brakes.

If thermal deformations of brake discs are taken into account the probability Q_T will increase, thus decreasing the safety of brakes. However, this lower level of brakes safety is supposed to be acceptable.

5. CONCLUSIONS

The presented procedure enables safety assessment of disc brake if reliability diagrams both with reliability indices of elements are given. So far, the safety level has been exclusively determined for brake holding (static condition) since in the case of emergency braking such a method is more complicated, i.e. analysis of reliability diagrams, braking requirements under dynamic conditions with regard to winder types (drum or friction winders), etc. Of course, the majority of results obtained can be adopted to the other condition of braking (dynamic).

A simulation method is intended to be used to safety assessment of the brakes. It would be also instrumental to obtain more precise reliability indices of brake elements (e.g. valves) through laboratory testing, exploatation examinations or simply as manufacturer's cataloque data. Allowance should be made for friction between brake lining and brake path (disc).

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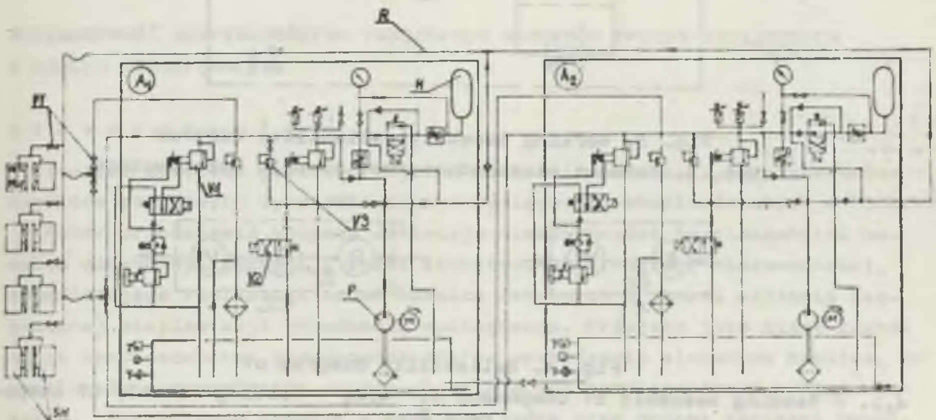


Fig. 1. ASEA hydraulic disc brake

Rys. 1. Hydrauliczny hamulec tarczowy ASEA

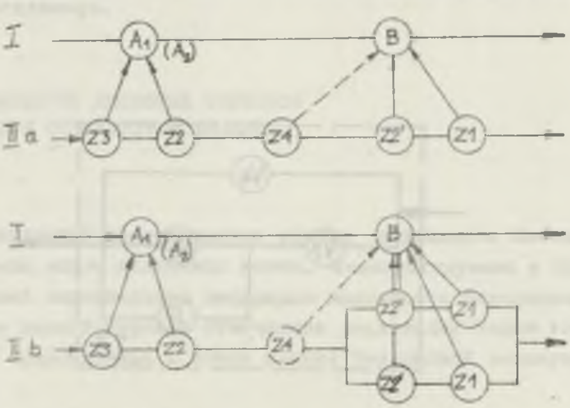


Fig. 2. Disc brake reliability diagram

Rys. 2. Schemat niezawodnościowy hamulca tarczowego

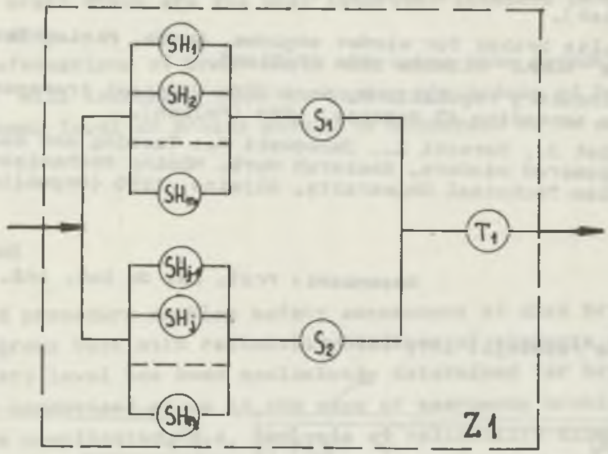


Fig. 3. Working assembly reliability diagram

Rys. 3. Schemat niezawodnościowy zespołu wykonawczego

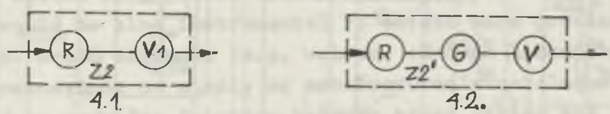


Fig. 4. Reliability diagram of

4.1. - feeding assembly of component A, 4.2. - feeding asembly of component B

Rys. 4. Schemat niezawodnościowy

4.1. - zespoły zasilającego części A, 4.2. - zespołu zasilającego części B

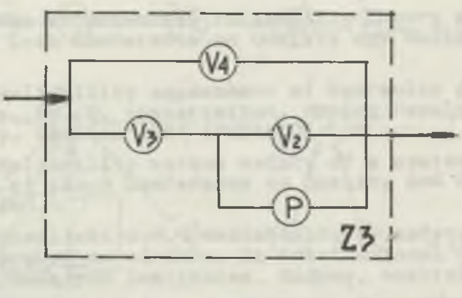
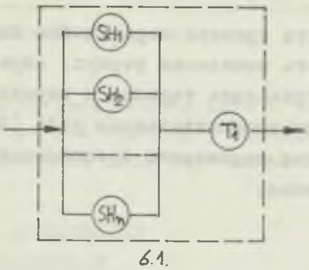


Fig. 5. Control assembly reliability diagram

Rys. 5. Schemat niezawodnościowy zespołu sterowania



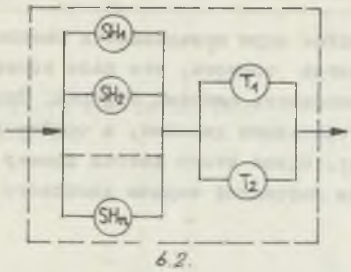


Fig. 6. Modyfied reliability diagram of working assembly with 6.1. - one brake disc, 6.2. - two brake discs

Rys. 6. Zmodyfikowany schemat niezawodnościowy zespołu wykonawczego 6.1. - z jedną tarczą, 6.2. - z dwoma tarczami

NIEZAWODNOŚĆ BEZPIECZEŃSTWA TARCZOWYCH HAMULCÓW MASZYN WYCIĄGOWYCH W UJĘCIU STRUKTURALNYM

S t r e s z c z e n i e

Tematem artykułu jest teoretyczna ocena niezawodności bezpieczeństwa hamulców tarczowych typu ASEA maszyn wyciągowych eksploatowanych w Polsce.

Autor przedstawił opisową definicję niezawodności bezpieczeństwa hamulca maszyn wyciągowej. Z kolei skonstruował struktury niezawodności, odpowiadające realizacji zadań hamulca istotnych z punktu widzenia bezpiecznej eksploatacji urządzenia wyciągowego. Przyjęto jako miary zawodności bezpieczeństwa prawdopodobieństwa uszkodzenia elementów hamulca, co umożliwiło wyprowadzenie wzorów określających bezpieczeństwo systemu hamulca. Rozpatrzono hamulce w wersji z jedną oraz dwiema tarczami hamulcowymi, ze szczególnym uwzględnieniem zespołu wykonawczego. Ponadto podano przykład oceny niezawodności bezpieczeństwa dla realizacji wybranego zadania hamulca tarczowego.

НАДЕЖНОСТЬ БЕЗОПАСНОСТИ ДИСКОВЫХ ТОРМОЗОВ ПОДЪЁМНЫХ МАШИН ПРИ СТРУКТУРНОМ ПОДХОДЕ

Р е з ю м е

Темой статьи является теоретическая оценка надёжности безопасности дисковых тормозов типа ASEA подъёмных машин, эксплуатируемых в Польше.

Астор представил описательную дефиницию надёжности безопасности тормоза подъёмника, Затем сконструировал отвечающие реализации задач тормоза структуры надёжности, существенные с точки зрения безопасной эксплуатации подъемного устройства.

В качестве меры ненадёжности безопасности принята вероятность повреждения элементов тормоза, что дало возможность выведения формул, определяющих безопасность системы тормоза. Рассматривались тормоза в версии о одним и двумя тормозными дисками, с особым учётом исполнительного узла (блока, комплекса). Кроме этого даётся пример оценки надёжности безопасности для реализации избранной задачи дискового тормоза.